$$\underline{X}_{l-l} = \left[\frac{\rho_g}{\rho_f}\left(\frac{\mu_f}{\mu_g}\right)^{0.2}\left(\frac{1}{x}-1\right)^{1.8}\right]^{0.5} \quad (36)$$

$$\underline{X}_{v-l} = \left[\frac{\rho_g}{\rho_f}\left(\frac{\mu_f}{\mu_g}\right)\left(\frac{1}{x}-1\right)\left(\frac{348}{(N_{Re})_g^{0.8}}\right)\right]^{0.5} \quad (37)$$

$$\underline{X}_{l-v} = \left[\frac{\rho_g}{\rho_f}\left(\frac{\mu_f}{\mu_g}\right)\left(\frac{1}{x}-1\right)(0.00286)(N_{Re})_f^{0.8}\right]^{0.5} \quad (38)$$

$$\underline{X}_{v-v} = \left[\frac{\rho_g}{\rho_f}\left(\frac{\mu_f}{\mu_g}\right)\left(\frac{1}{x}-1\right)\right]^{0.5} \quad (39)$$

It is seen that the determination of Δp_{fr} involves a numerical evaluation of the integral in Equation 35 in which the integrand factor ϕ_l^2 is determined from Fig. 6. Furthermore, if the pressure drop is appreciable (say as much as 10 percent of the absolute pressure) an iterative procedure is also required to establish the pressure at each step of the numerical integration.

Although a rigorous solution to the problem involves the complicated procedure discussed above, it should be remembered that extreme precision is unwarranted due to the many assumptions that have been made. In most cases, it is only an approximation of the pressure drop which is required. This can be obtained very simply by approximating the integral of Equation 35 by

$$\phi_{l\,avg}^2(1-x_{avg})^{1.8} = \frac{1}{x_e - x_i}\int_{x_i}^{x_e}\phi_l^2(1-x)^{1.8}dx \quad (40)$$

where $\phi_{l\,avg}$ is evaluated from Fig. 6 at the value $\underline{X}_{avg}$. For turbulent-turbulent flow

$$\underline{X}_{avg} = \left[\left(\frac{\rho_g}{\rho_f}\right)_{avg}\left(\frac{\mu_f}{\mu_g}\right)^{0.2}\left(\frac{1}{x_{avg}}-1\right)^{1.8}\right]^{0.5} \quad (41)$$

where

$$x_{avg} = \frac{x_e + x_i}{2} \quad (42)$$

A comparison of the more rigorous solution and the approximate solution will be given for a specific case in *Example 1*.

Example 1. Calculate the two-phase pressure drop across a coil which is 11 tubes wide, 66 in. long, 4 rows deep, ⅝ in. OD tubing, and has a 22-ton capacity using Refrigerant 12. Suction temperature is assumed as 40 F, with condensing temperature 105 F.

The loading is 2 tons per circuit, and, since the standard vapor compression cycle is being assumed, the flow rate is 3.9 lb /(min) (ton). Thus, the total flow rate, w, in the circuit is 7.8 lb/min, or 4.04×10^{-3} slug/sec.

Solution: The quality entering the coil is found to be $x_i = 0.235$, so that the liquid fraction of the flow rate is $w_f = 5.95$ lb/min, and the vapor fraction is $w_g = 1.85$ lb/min. It will be assumed that the fluid leaving the coil is saturated vapor, i.e., $x_e = 1.00$. At 40 F, the viscosity of the liquid and vapor are:

$\mu_f = 0.286$ centipoise $= 5.95 \times 10^{-6}$ slug/(ft)(sec).
$\mu_g = 0.0119$ centipoise $= 2.48 \times 10^{-7}$ slug/(ft) (sec).

The densities are:

$\rho_f = 86.4$ lb/cu ft $= 2.68$ slug/cu ft.
$\rho_g = 1.295$ lb/cu ft $= 0.0402$ slug/cu ft.

The inside diameter of ⅝ in. OD tubing is 0.0455 ft, and the inside area is 0.00162 sq ft.

The velocity of the vapor at the coil inlet (assuming no liquid present) would be

$$V_g = \frac{w_g}{\rho_g A} = \frac{1.85}{1.295(.00162)(60)}$$

$$V_g = 14.7 \text{ ft/sec}$$

The kinematic viscosity of the vapor is

$$\nu_g = \frac{\mu_g}{\rho_g} = \frac{2.48 \times 10^{-7}}{0.0402}$$

$$\nu_g = 6.16 \times 10^{-6} \text{ sq ft/sec.}$$

Then, the associated Reynolds number is

$$(N_{Re})_g = \frac{V_g D}{\nu_g} = \frac{(14.7)(0.0455)}{6.16 \times 10^{-6}}$$

$$(N_{Re})_g = 1.085 \times 10^5$$

Thus, the vapor is definitely flowing turbulently since the liquid actually present assures that the vapor Reynolds number is higher than the value calculated above.

In a similar way, the Reynolds number for the liquid fraction at coil inlet, assuming no vapor present, is found to be $(N_{Re})_f + 1.45 \times 10^4$ which assures that the liquid is flowing turbulently. The dissipative flow model is, therefore, turbulent liquid-turbulent vapor (t-t).

The Lockhart-Martinelli parameter to be used is therefore

$$(\underline{X}_{t-t})_{avg} = \left[\left(\frac{\rho_g}{\rho_f}\right)_{avg}\left(\frac{\mu_f}{\mu_g}\right)_{avg}^{0.2}\left(\frac{1}{x_{avg}}-1\right)^{1.8}\right]^{0.5}$$

where

$$x_{avg} = \frac{1.00 + 0.235}{2}$$

$$x_{avg} = 0.618$$

The average value to be used for the fluid properties is not known at this stage because the pressure drop is not yet known. For a first approximation, the fluid properties will be evaluated at inlet temperature (assumed to be 40 F).

$$(\underline{X}_{t-t})_{avg} = \left[\left(\frac{1.295}{86.4}\right)\left(\frac{0.286}{0.0119}\right)^{0.2}\left(\frac{1}{0.618}-1\right)^{1.8}\right]^{0.5}$$

or

$$(\underline{X}_{t-t})_{avg} = 0.109$$

From Fig. 6, it is found that

$$(\phi_l)_{avg} = 18$$

$$(\phi_l^2)_{avg}(1-x_{avg})^{1.8} = (18)^2(1-0.618)^{1.8} = 57.4$$

The liquid pressure drop Δp_o is calculated next. This is given by Equation 31. The velocity V_o is given by:

$$v_o = \frac{w}{\rho_f A} = \frac{7.8}{(86.4)(0.00162)(60)}$$

$$v_o = 0.93 \text{ ft/sec}$$

The Reynolds number $(N_{Re})_o$ is:

$$(N_{Re})_o = \frac{V_o D}{\nu_f} = \frac{(0.93)(0.0455)}{2.22 \times 10^{-6}}$$

$$(N_{Re})_o = 1.9 \times 10^4$$

From Fig. 5 (assuming a smooth tube) the friction factor f_o is found to be:

$$f_o = 0.026$$

Neglecting the bends, the length of one circuit of the coil is:

$$L = \frac{4(66)}{12} \doteq 22 \text{ ft}$$

Substituting these various values into Equation 31:

$$\Delta p_o = 0.026\left(\frac{22}{.0455}\right)\left(\frac{1}{2}\right)(2.68)(0.93)^2$$

$$\Delta p_o = 14.6 \text{ lb/sq ft}$$

$$\Delta p_o = 0.101 \text{ psi}$$

The frictional component of the two-phase pressure drop is thus found to be:

$$\Delta p_{fr} = \Delta p_o(\phi_l^2)_{avg}(1 - x_{avg})^{1.8}$$
$$= (0.101)(57.4)$$

or

$$\Delta p_{fr} = 5.8 \text{ psi}$$

The acceleration component is given by Equation 28. Assuming as a first approximation that $\rho_{fe} = \rho_{fi}$ and $\rho_{ge} = \rho_{gi}$, it is found that:

$$\Delta p_a = \frac{16(4.04 \times 10^{-3})^2}{\pi^2(.0455)^4}\left[\frac{1.00 - 0.235}{0.0402} - \frac{1.00 - 0.235}{2.68}\right]$$

$$\Delta p_a = 100 \text{ lb/sq ft}$$

$$\Delta p_a = 0.7 \text{ psi}$$

The total two-phase pressure drop is thus found to be:

$$\Delta p_{TP} = \Delta p_f + \Delta p_a$$
$$= 5.8 + 0.7$$
$$\Delta p_{TP} = 6.5 \text{ psi}$$

This value could now be improved by reworking the problem using properties evaluated for the average evaporator temperature, which is about 36 F. If this is done, it is found that $\Delta p_{TP} = 6.7$ psi.

If a more accurate numerical integration of Equation 35 is carried out rather than using the average value of Equation 40, it is found that $\Delta p_{TP} = 5.2$ psi.

This gives an idea of the order of magnitude of differences associated with different computational methods. It also may be of interest to note that the manufacturer of this particular coil states that the pressure drop is 5.6 psi.

DYNAMIC PRESSURE LOSSES

As was discussed previously, total pressure, P, is equal to the sum of the static pressure and kinetic energy head (velocity pressure). If there are no losses in a system, the total pressure remains constant, but if losses exist, the total pressure decreases in the direction of flow.

The loss of total pressure due to wall friction in uniform pipes has been discussed previously. In addition to the frictional losses, losses can be imposed on the flow, due to separation of the boundary layer and the subsequent dissipation into turbulence.

Table 5 . . . . Location of Pressure Loss Data in the 1964 and 1965 GUIDE AND DATA BOOK

Component	Volume	Location	
		Chapter	Pages
Ducts and duct fittings	1965	31	562–569
Duct fittings and intakes	1964	7	78–79
Registers	1964	7	85–86
Pipe fittings	1964	8	97
	1964	10	139–140
	1964	76	849
Pipe friction losses, copper and iron	1964	10	136–137
Pipe friction losses, commercial steel, schedule 40	1964	10	138
Pipe and tubing friction losses	1964	82	921
Valves and fittings friction losses	1964	82	922

Dynamic Losses

Dynamic flow losses represent a conversion of kinetic energy of a vector sense parallel to the direction of flow to kinetic energy of a random vector sense or *turbulence*. Whenever any factor in a flow system changes the magnitude or direction of the fluid kinetic energy, turbulence occurs with an accompanying loss in total pressure. Flow losses, in general, are proportional to the kinetic energy of the fluid for any given system configuration.

Because flow losses are proportional to the kinetic energy of the fluid it is convenient to express them non-dimensionally in terms of the velocity pressure or dynamic head existing at some point in the system, usually immediately upstream of the section under consideration. This non-dimensional expression, commonly referred to as a *loss coefficient*, is defined as:

$$C_L = \frac{\Delta P}{p_v} \quad (43)$$

where

C_L = loss coefficient.
ΔP = pressure loss.
$p_v = \rho\dfrac{V^2}{2g_c}$ = velocity pressure.

The units used in Equation 43 should be consistent for all terms.

If the losses are expressed as loss coefficients based on local velocities, they are nearly independent of scale, velocity, and fluid properties so long as the Reynolds number is well into the turbulent region, no change in the mode of flow occurs, and, for compressible fluids, the Mach number is less than 0.3.

Loss coefficients for duct elbows and duct area changes are given in Tables 3 and 4 of Chapter 31. Additional loss coefficient data for elbows, bends, area changes, diffusers, turning vanes, branch connections, manifolds, valves, and screens are given in Section 1, of Reference 12.

Pressure loss data for ducts and duct fittings and various types of pipe and pipe fittings, expressed as equivalent lengths rather than as loss coefficients, are included in other chapters of the 1964 and 1965 volumes of the Guide And Data Book as shown in Table 5.

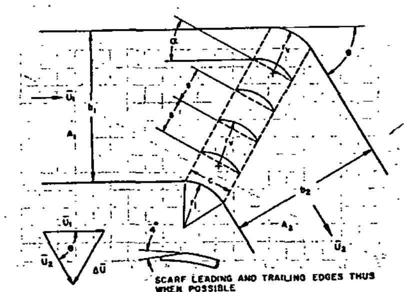

Fig. 7 . . . . Turning Vanes Geometry[12]