Table 2 .... Gas Constants for Common Gases[a]

Gas	R	$R_g = mR$
Carbon dioxide	1,130	1544
Oxygen	1,555	1546
Air	1,716	1546
Nitrogen	1,777	1546
Hydrogen	24,697	1534
Methane	3,041	1542

[a] Adapted from page 291 of Reference 1.

Table 3 .... Specific Heats of Common Gases[a]

Gas	Specific heat, Btu/(lb)(F deg)		$k = c_p/c_v$
	c_p	c_v	
Air	0.241	0.1725	1.40
Carbon dioxide	0.205	0.1599	1.28
Hydrogen	3.42	2.435	1.40
Methane	0.593	0.4692	1.26
Nitrogen	0.247	0.1761	1.40
Oxygen	0.217	0.1549	1.40

[a] Adapted from page 291 of Reference 1.

Specific heats of refrigerants are given in Tables 7 to 10 of Chapter 19.

Types of Fluids

An ideal fluid is one which is incompressible and has zero viscosity. With zero viscosity, the fluid offers no resistance to shearing forces; and hence, during flow and deformation of the fluid, all shear forces are zero. Many flow problems can be simplified by assuming that the fluid is ideal. All real fluids have finite viscosity; and in most cases of flow in ducts and over immersed bodies, it is necessary to consider the viscosity and the related shearing stresses associated with deformation of the fluids. Real fluids are also called viscous fluids. Non-viscous fluids are those having zero viscosity, but they may or may not be incompressible. Flow of an ideal fluid is called non-viscous flow.

Real fluids are further subdivided into two main classes. Newtonian fluids are those for which the viscosity coefficient is independent of the rate of shear (velocity gradient). Non-Newtonian fluids are those in which the viscosity at a given pressure and temperature is a function of the velocity gradient. Gases and thin liquids can be considered Newtonian fluids while thick liquids and gases near their critical points are non-Newtonian. Such fluids as colloidal suspensions, emulsions, and gels are included in the classification of non-Newtonian fluids.

THEORY OF FLUID FLOW

Most of the fluid flow problems encountered in the design or analysis of refrigeration and air-conditioning systems are such that they may be treated as one-dimensional steady flows in ducts. The governing equations for such flows are the dynamic equation:

$$\frac{dp}{dz_L} + \rho\frac{d}{dz_L}\left(\frac{V^2}{2}\right) + \gamma\frac{dz}{dz_L} + \frac{f_D}{D}\rho\frac{V^2}{2} = 0 \qquad (1)$$

and the continuity equation:

$$\frac{d}{dz_L}(\rho A V) = 0 \qquad (2)$$

where

f_D = D'Arcy-Weisbach friction factor (dimensionless).
A = duct cross sectional area, square feet.
D = equivalent diameter of duct, feet.
z = elevation, feet.
z_L = position along duct, feet.
p = static pressure, pounds per square foot.
V = fluid velocity, feet per second.

The equivalent diameter D is defined by:

$$D = \frac{4A}{P_w} \qquad (3)$$

where P_w is the wetted perimeter of the duct (feet). The friction factor f_D is usually obtained from a chart such as that shown in Fig. 5. It is a function of Reynolds number N_{Re} and relative roughness ϵ/D, where:

$$N_{Re} = \frac{VD}{\nu} \qquad (4)$$

and

ϵ = absolute roughness, feet.

Flow of Incompressible, Frictionless Fluids

Many fluid flow problems are such that the frictional contribution to the pressure gradient is negligible compared to contributions due to momentum change and change of elevation. This will be true, for example, if the duct diameter is large compared to the rate at which the duct area changes with respect to position along the duct. If, in addition, the fluid can be considered incompressible, Equations 1 and 2 can be written in the form:

$$\frac{dp}{dz_L} + \frac{d}{dz_L}\left(\frac{1}{2}\rho V^2\right) + \frac{d}{dz_L}(\gamma z) = 0 \qquad (5)$$

$$\frac{d}{dz_L}(AV) = 0 \qquad (6)$$

Since all the differentials in Equations 5 and 6 are exact, these equations can be integrated between any two points z_1 and z_2 of the duct to obtain:

$$\tfrac{1}{2}\rho V_2^2 + p_2 + \gamma z_2 = \tfrac{1}{2}\rho V_1^2 + p_1 + \gamma z_1 \qquad (7)$$

and

$$A_2 V_2 = A_1 V_1 \qquad (8)$$

Equation 8 shows that the volumetric flow rate $Q = AV$ is constant in the duct. Equation 7 is known as Bernoulli's equation. These equations can be used to determine the pressure drop associated with area changes or change of elevation for liquids or for gases in which the pressure ratio p_2/p_1 is greater than 0.95.

The sum of the kinetic energy and static pressure terms is called the total pressure P.

$$P = p + \tfrac{1}{2}\rho V^2 \qquad (9)$$

Equation 9 shows that the total pressure remains constant in a horizontal duct when friction is neglected.

Flow of Compressible, Frictionless Fluids

Flow processes involving gases and vapors which involve pressure changes greater than approximately 5 percent require the use of the compressible fluid form of the flow equations. In this case, the assumption of incompressibility can be replaced by the isentropic relationship between pressure and density.

$$p\left(\frac{1}{\rho}\right)^k = \text{constant} \qquad (10)$$

Substituting Equation 10 into Equation 1, neglecting the frictional and elevation terms, and integrating between two points z_1 and z_2 of the duct, obtain:

$$\tfrac{1}{2}(V_2^2 - V_1^2) + \left(\frac{k}{k-1}\right)\frac{p_1}{\rho_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \qquad (11)$$

The Mach number, M, at a point is defined to be the ratio of the velocity to the speed of sound at the point.

$$M = \frac{V}{a} \qquad (12)$$

where

$$a^2 = k\frac{p}{\rho} \qquad (13)$$

and

a = sonic velocity, feet per second.

Upon substituting Equations 12 and 13 into Equation 11, it is possible to show that:

$$\frac{p_2}{p_1} = \left[\frac{1 + \left(\frac{k-1}{2}\right)M_1^2}{1 + \left(\frac{k-1}{2}\right)M_2^2}\right]^{\frac{k}{k-1}} \qquad (14)$$

The total pressure for compressible flow can be found by setting $M_2 = 0$ in Equation 14. It is found that:

$$P = p\left(1 + \frac{k-1}{2}M^2\right)^{\frac{k}{k-1}} \qquad (15)$$

If the Mach number in Equation 15 is set equal to unity, a critical pressure ratio is obtained, namely:

$$\frac{p}{P} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad (16)$$

Equation 16 shows that if the ratio of the static pressure to the total pressure at a point is equal to the critical pressure ratio, the velocity at that point will be equal to the speed of sound at that point. This principle is frequently applied in the case of critical flow orifices. If the pressure ratio across an orifice is equal to or less than the critical pressure ratio, the velocity at the orifice will be sonic. The mass flow rate is then:

$$w_2 = \rho_2 A_2 V_2 \qquad (17)$$

where

w_2 = mass flow rate, slugs per second

and subscript 2 denotes conditions at the orifice.

Equations 11 and 17 can be combined to show that:

$$w_2 = A_2\sqrt{k\rho_T P\left(\frac{2}{k+1}\right)^{\frac{(k+1)}{(k-1)}}} \qquad (18)$$

where

$$\rho_T = \frac{P}{RT} \qquad (19)$$

ρ_T = density corresponding to total pressure.

The flow through an orifice having a pressure ratio less than the critical ratio as defined by Equation 16 is therefore a function only of the upstream pressure and density. If the upstream pressure and temperature are held constant, the flow will be constant regardless of the downstream conditions as long as the pressure ratio remains less than critical. The critical flow is also the maximum flow which will pass through the orifice for a given upstream total pressure and temperature.

Flow of Real Fluids

In a large number of fluid-flow problems, the effect of viscosity is insignificant compared with the effect of other quantities such as pressure and inertia force. Consequently, it can be neglected. This is particularly true of fluids having low viscosity, such as water and the common gases. Typical problems in which such an assumption is valid relate to flow through orifices in large tanks, in channel and duct entrances, and in converging and diverging nozzles. In such problems, the movement of the main mass of fluid is of particular interest, and conditions immediately adjacent to the solid boundary, where viscosity has a significant effect, are not of interest. However, in problems dealing with flow in closed conduits or past a stationary immersed object, viscosity plays an important role in determining the flow patterns and energy requirements to move the fluid.

As a nonviscous fluid flows past a solid boundary, the fluid immediately adjacent to the boundary has a finite velocity determined by the conditions of flow and the geometry of the boundaries. On the other hand, for the flow of viscous fluid, there is no slip between the fluid and the boundary. In other words, as the fluid moves past a solid boundary, the fluid immediately adjacent to the boundary has zero velocity relative to the boundary. The vicinity of the boundary is therefore a region where the fluid velocity increases from zero to that of the main stream, and in this region velocity gradients exist in the fluid. This region of changing velocity is known as the *boundary layer*.

With real fluids, two flow regimes exist—laminar and turbulent. In laminar or streamline flow, the particles of fluid move in a direction parallel to solid boundaries with no component of velocity normal to the axis of the conduit. In turbulent flow there is considerable mixing of fluid during flow. A particle of fluid has an average velocity parallel to the axis of the conduit, but its motion is chaotic, with instantaneous velocities both parallel and perpendicular to the axis.

The dimensionless quantity VD/ν, known as the Reynolds number, N_{Re} represents a ratio of the inertia to the viscous forces in a fluid. At high Reynolds numbers, the inertia forces predominate over the viscous forces and turbulent flow re-

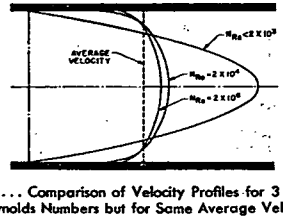


Fig. 1 .... Comparison of Velocity Profiles for 3 Different Reynolds Numbers but for Same Average Velocity