2.4. Heat Transfer and Pressure Drop Inside Tubes

2.4.1 Heat Transfer And Pressure Drop In Single Phase Flow Inside Round Tubes

1. Flow Regimes for Heat Transfer. The fundamental mechanisms of heat transfer inside tubes were discussed in Chapter 1, and a few of the salient points will be repeated here in connection with presenting detailed correlations for calculating the heat transfer coefficient.

The particular correlation to be used for calculating in tube heat transfer coefficients depends upon the flow regime existing inside the tube - laminar, turbulent, or transition. For the majority of applications for which Trufin is suitable, the flow will be turbulent; however, it is necessary to present a complete set of correlations to cover the entire range.

The flow regime existing in a tube may be determined by calculating the Reynolds number, Re_i :

$$\mathrm{Re}_i = \frac{d_i \rho_i V_i}{\mu_i} \tag{2.19}$$

where d_i is the inside diameter of the finned portion of the tube, ρ_i and μ_i are respectively the density and viscosity of the fluid flowing and V_i is the average fluid velocity inside the finned portion. Any set of dimensions may be used in the equation as long as the final result is dimensionless.

If Re_i for a given flow is less than about 2000, the flow is laminar, though some disturbances from the entrance of the tube or from an upstream pump may persist to significant distances down the tube.

If Re_i is greater than about 10,000, the flow is fully developed turbulent and good heat transfer correlations exist.

For Re_i between 2000 and 10,000, the heat transfer coefficient is between the values for laminar and turbulent and cannot be predicted with precision. This is the so called transition regime and it is generally recommended that the designer attempt to keep the flow conditions out of this range. This can usually be done by increasing the velocity sufficiently (e.g., going to multiple tube passes) to get into the fully developed turbulent flow regime.

2. *Heat Transfer in Laminar Flow*. A number of correlations exist for the laminar flow regime, but the one most widely recommended is the Hausen correlation:

$$\overline{h_i} = \frac{k_i}{d_i}\left\{3.65 + \frac{0.0668\,\mathrm{Re}_i\,\mathrm{Pr}_i(d_i/L)}{1+0.04\left[\mathrm{Re}_i\,\mathrm{Pr}_i(d_i/L)\right]^{2/3}}\right\}\left(\frac{\mu_i}{\mu_{w,i}}\right)^{0.14} \tag{2.20}$$

where $\overline{h_i}$ is the mean coefficient for the entire length L of a single tube. Pr_i is the Prandtl number of the fluid flowing defined as

$$\mathrm{Pr}_i = \frac{c_{pi}\mu_i}{k_i} \tag{2.21}$$

Examination of Eq. (2.20) shows that the mean coefficient decreases with increasing length of tube, L. This is a consequence of the buildup of an adverse temperature gradient in laminar flow. The value of L to be used is the length of a single pass, or in a U-tube bundle, the length of the straight tube from the tube sheet to the tangent point of the bend. In other words, the adverse temperature gradient is assumed to be completely destroyed by the turnaround or by the strong secondary flow induced in the U-bend. The dependence of $\overline{h_i}$ on L means that the usual design calculation is in principle reiterative; that is, L must be estimated, $\overline{h_i}$ calculated, and a revised value of L calculated from the area requirement based upon that $\overline{h_i}$. This reiterative process ordinarily converges very rapidly.

For hand calculations, sufficient accuracy is achieved by evaluating all physical properties (except μ_w) at the mean bulk temperature of the stream $\bar{t}$ or $\overline{T}$ as defined in Fig. 2.18.

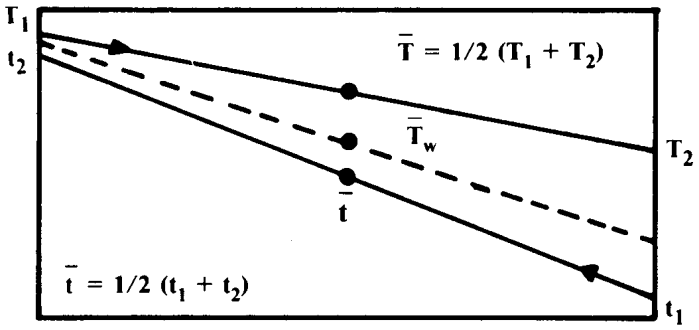


Fig. 2.18 Temperature Definitions

The term $(\mu_i / \mu_{w,i})^{0.14}$ included at the end of Eq. (2.20) is the so-called Sieder - Tate term, which corrects the coefficient for the effect of a viscosity difference between the bulk fluid and that at the wall. For example, if a liquid is being heated in the tube, the wall temperature and therefore the temperature of the liquid at the wall is higher than the bulk temperature. Thus the viscosity of the liquid at the wall is less than the bulk viscosity and the boundary layer of the liquid on the wall is thinner, resulting in a small increase in the film heat transfer coefficient over that calculated for the constant viscosity case. It is sufficiently accurate to calculate the mean wall temperature $\overline{T_w}$, at which μ_w is evaluated as shown in Fig. 2.18.

$$\overline{T_w} = \bar{t} + \frac{U_i}{h_i}\left(\overline{T} - \bar{t}\right) \tag{2.22}$$

where U_i is calculated based upon the inside surface area.

One element of conservatism that has been introduced into the calculations of this section is the neglect of natural convection effects in calculating $\overline{h_i}$. These effects arise from the density differences caused by the temperature gradients and almost always act in heat exchangers to increase the heat transfer coefficients, the effect increasing with increasing temperature differences. However, the computation of

these effects adds complications to the calculation and it is usually much more convenient and somewhat conservative to neglect them.

3. *Heat Transfer in Turbulent Flow.* In the fully developed turbulent flow regime at Re_i above 10,000, the most widely applicable correlation for hi is the Sieder Tate equation:

$$h_i = 0.023 \frac{k_i}{d_i} \mathrm{Re}_i^{0.8} \mathrm{Pr}_i^{1/3} \left(\frac{\mu_i}{\mu_{i,w}} \right)^{0.14} \tag{2.23}$$

Actually, in various sources the prefactor constant is quoted variously at 0.019 to 0.027, and the 0.023 is a plausible average. This variation should be considered by the designer when deciding what confidence to place in an answer and the precision to which it should be calculated. For L/d_i values less than 60 the mean heat transfer coefficient is somewhat higher than that given by Eq. (2.23) because of entrance effects; however, most exchangers greatly exceed that ratio and in any case it is usually conservative to ignore the improvement.

For water, which will very commonly be the tube-side fluid in Trufin applications, the Eagle-Ferguson chart (7) is regarded as both very convenient for hand calculations and more accurate than most others. This chart is given in
Fig. 2.19. Recall that the values of the velocity and inside diameter referred to are those of the finned portion of the tube.

4. *Heat Transfer in Transition Flow.* For Re_i between 2000 and 10,000, the heat transfer coefficient is very unpredictable, being largely dependent upon the structuring of the flow caused by upstream flow conditions. There is some possibility of flow oscillations because of hydrodynamic instability between laminar and turbulent flow, and the designer is best advised to avoid this region if possible.

However, the coefficient will be bounded by Eqs. (2.20) and (2.23) and a plausible if not very exact procedure in this range is as follows:

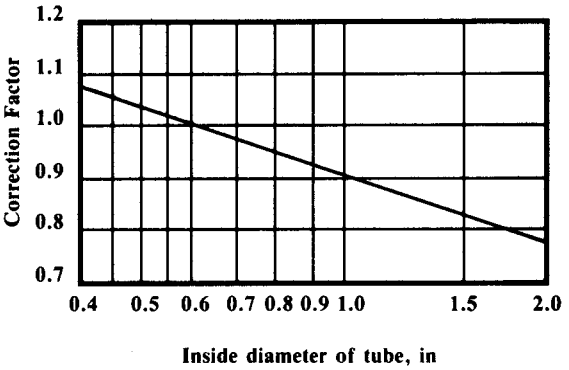
A. Calculate $\overline{h_i}$ by Eq. (2.20) as if the flow were laminar.

B. Calculate h_i by Eq. (2.23) as if the flow were turbulent.

C. Estimate $(h_i)_T$

$$(h_i)_T = \overline{h_i} + \left[h_i - \overline{h_i} \right] \left(\frac{\mathrm{Re}_i - 2000}{8000} \right) \tag{2.23A}$$

5. *Pressure Drop Inside Round Tubes*. The pressure drop on the tube-side of the heat exchanger is composed of several different terms: the pressure losses in the inlet and outlet nozzles, the pressure losses in the headers or channels, the pressure losses associated with accelerating the fluid and establishing the flow profile in the tube and finally the frictional losses of the flow in the tube.



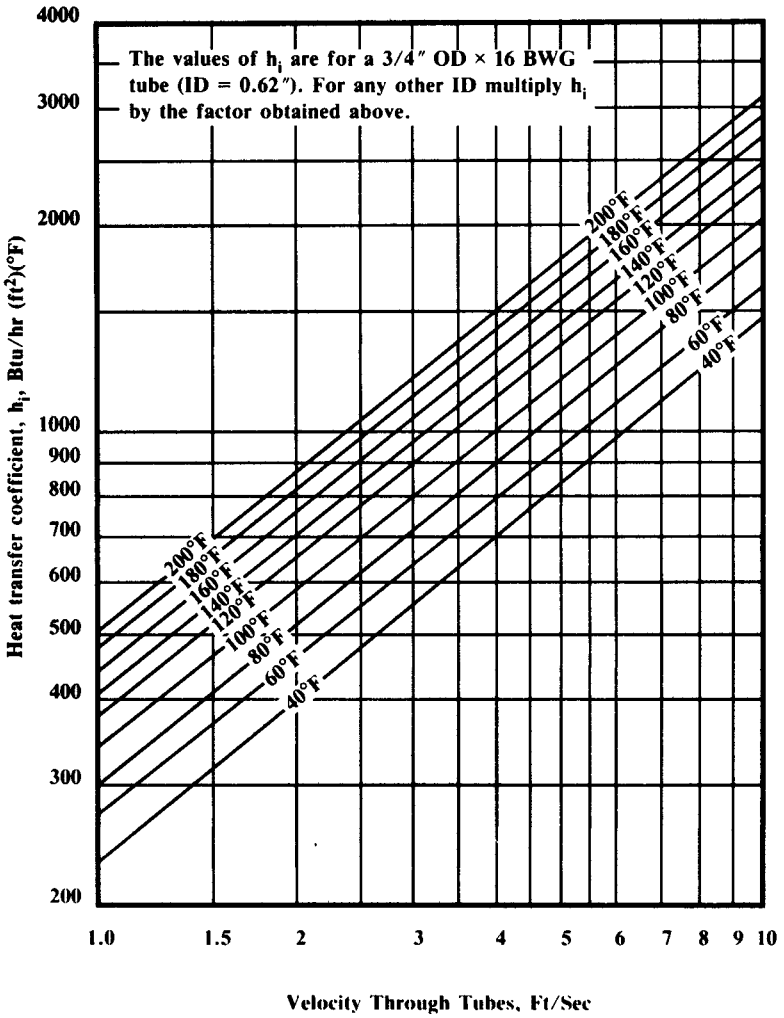


Fig. 2.19 Charts of h_i for Water Flowing Inside Tube

The latter term is usually the major contributor, and is the only one that can be calculated with a fair degree of accuracy.

For the nozzle losses, it is usually sufficient to calculate the loss for each nozzle at about three times the velocity head in the nozzle, Eq. (2.24):

$$\Delta P_{noz} = 3\frac{\rho\ V^2{}_{noz}}{2g_c} \tag{2.24}$$

where V_{noz} is calculated at the smallest cross-section area for flow (i.e., highest velocity) in the nozzle.

The combined header and tube entrance losses are estimated in a similar way, but using the velocity inside the tube, V_i:

$$\Delta P_{ent} = 3\left(\frac{\rho\ V_i^2}{2g_c}\right) \tag{2.25}$$

In a multipass exchanger, this loss is incurred for each pass, so Eq. (2.25) should be multiplied by the number of tube-side passes.

The frictional losses inside the tube are calculated using a friction factor chart similar to Fig. 2.20.

The ordinate of Fig. 2.20 is the Fanning friction factor, which is related to the frictional pressure loss by

$$\Delta P_i = \frac{2f_i\rho_i\ V_i^2 L}{d_i g_c}\left(\frac{\mu_w}{\mu}\right)_i^{0.14} \tag{2.26}$$

where L is the tube length; if there are n tube passes, the total in tube friction loss is n times the value given by (2.26).

The differing behavior of laminar and turbulent flow is clearly indicated in Fig. (2.20) by the curves above and below
Re_i = 2100. However, because of the disturbance of the flow caused by the entrance, the flow may display higher pressure drops in the Re_i range above 600 than indicated by the laminar flow curve. Therefore, the turbulent flow curves have been extrapolated to lower Re_i (dashed curve) until they intersect the laminar flow curve.

In the turbulent flow regime, the frictional pressure loss increases with the increasing roughness of the surface. For new tube, the friction factor is given by the "smooth" curve in Fig. 2.20.

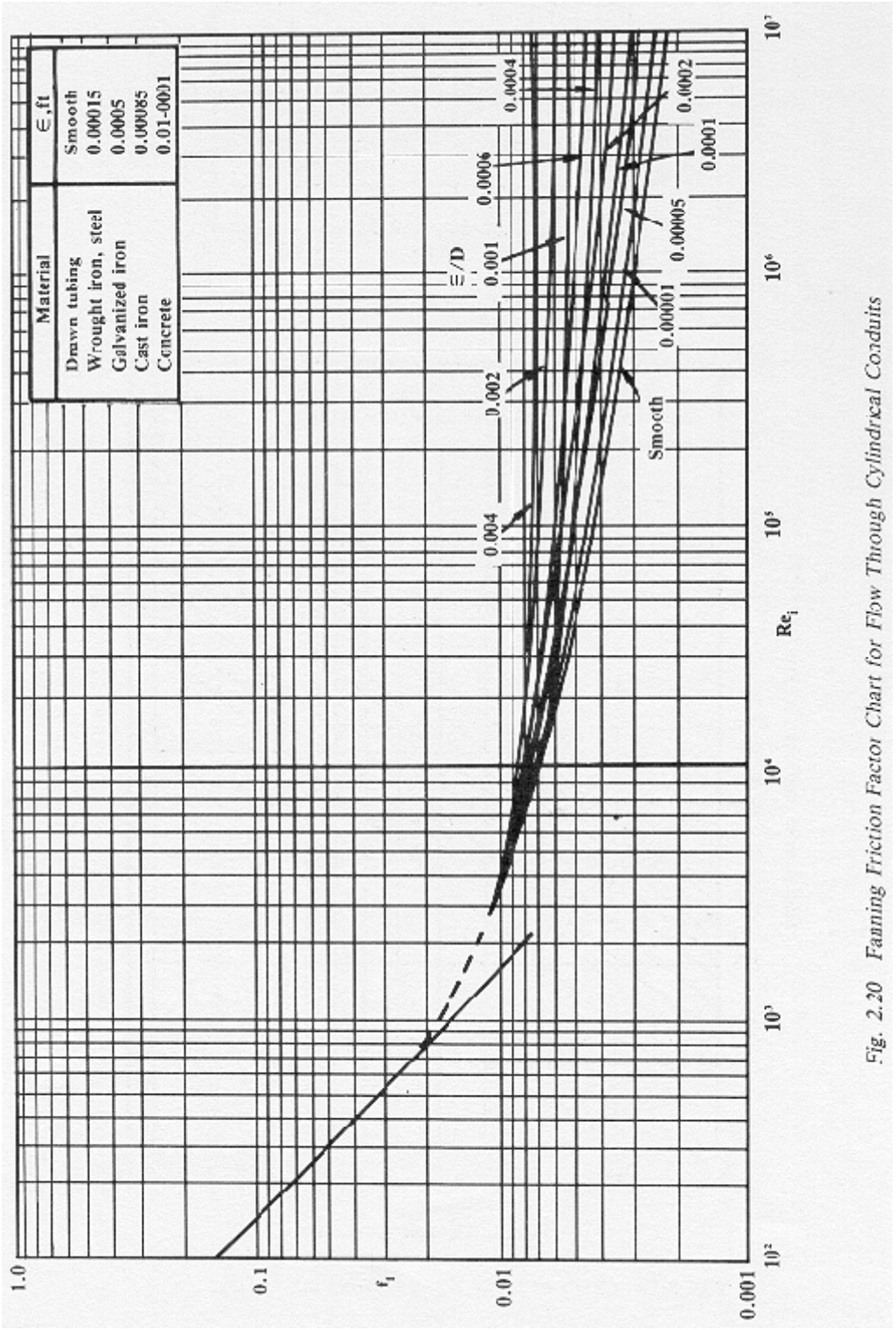


Fig. 2.20 Fanning Friction Factor Chart for Flow Through Cylindrical Conduits

2.4.2. Heat Transfer in Two-Phase Flow Inside Tubes

1. *Heat Transfer During In-Tube Condensation*. Frequently, type S/T Trufin will be used where the fluid in the tubes is condensing. Usually the condensing vapor will be steam, although other pure or nearly pure vapors will be involved. The condensation of a pure component under low pressure drop conditions (the usual case and the only one that can be practically handled without an elaborate computer program) is nearly isothermal, but there is a substantial variation in the local condensing heat transfer coefficient from the entrance (nearly all vapor and a high coefficient) to the exit (usually nearly all liquid and a low coefficient).

For most heat exchanger design purposes at the hand calculation level, it is both necessary and sufficient to use an average heat transfer coefficient on the condensing side. One correlation that is simple to use, generally conservative, and sufficiently accurate for most purposes is due to Boyko and Kruzhilin (8):

$$h_i = 0.024 \frac{k_\ell}{d_i} \mathrm{Re}_{i,\ell}^{\ 0.8} \mathrm{Pr}_{i,\ell}^{\ 0.43} \left[\frac{\sqrt{(\rho/\rho_m)_i} + \sqrt{(\rho/\rho_m)_o}}{2} \right] \tag{2.27}$$

where

$$(\rho/\rho_m)_i = 1 + \left(\frac{\rho_\ell - \rho_v}{\rho_v} \right) x_i \tag{2.28}$$

and

$$(\rho/\rho_m)_o = 1 + \left(\frac{\rho_\ell - \rho_v}{\rho_v} \right) x_o \tag{2.29}$$

In the above equations x_i is the quality of the inlet vapor and x_o is the quality of the outlet vapor. For the usual case, x_i = 1.00 and x_o = 0; then the bracketed term in Eq. (2.27) reduces to

$$\left[\frac{1 + \sqrt{\rho_\ell / \rho_v}}{2} \right]$$

The term $\mathrm{Re}_{i,\ell}$ in Eq. (2.27) requires some comment; it is the Reynolds number computed as if the entire condensing flow were *liquid*:

$$\mathrm{Re}_{i,\ell} = \frac{d_i w_i}{\mu_\ell \left(\frac{\pi}{4} \right) d_i^{\ 2}} = \frac{d_i \rho_\ell \ V_i}{\mu_\ell} \tag{2.30}$$

where W_i is the total weight flow rate of fluid through one tube. $\mathrm{Pr}_{i,\ell}$ is the Prandtl number for the liquid.

The frictional pressure drop during condensation can usually be estimated with sufficient accuracy to be about 1/4 to 1/2 of the pressure drop calculated as if the entire vapor flow traveled the total length of the tube without condensing.

A more detailed analysis of flow and heat transfer during condensation is given in Chapter 3.

2. *Heat Transfer During In-tube Boiling.* Occasionally a boiling coolant is to be used in the tubes of a finned shell and tube exchanger. The general question of boiling heat transfer is an extremely complicated one. For non-critical situations the following remarks may suffice.

The first matter is to ensure that the conditions for nucleating a vapor phase exist. Generally this condition will be satisfied if the wall temperature exceeds the local saturation temperature of the liquid by 5 to $10°F$. The exact value depends upon the substance to be boiled, being greater for water than for non-aqueous liquids, and upon the pressure, becoming less as the pressure increases. Even if boiling doesn't occur immediately at the entrance, to the tube, the single phase heat transfer heats up the liquid until conditions for nucleation are satisfied. The Sieder - Tate equation (2.23) can be used to estimate the sensible heat transfer coefficient.

Once a vapor phase is formed, the Boyko-Kruzhilin equation (2.27) can be used to estimate the average coefficient for the boiling region up to an exit quality of about 0.5. Above this value, there is a significant probability that the boiling flow may go into the "dry-wall" regime, in which the liquid phase is mostly in the form of mist in the vapor flow and the wall is not completely wetted. This leads to very low heat transfer rates and possibly deposition of solid matter on the tube wall.

Pressure drop in a boiling flow is very difficult to predict using methods suitable for hand calculation. The basic concepts and some applicable correlations are discussed in Chapter 5.

3. *Heat Transfer During Gas-Liquid Flow*. The in-tube flow may occasionally be a gas-liquid mixture in which little or no phase change (boiling or condensation) takes place. Any calculations on such a system must be very uncertain because in practical heat transfer equipment it is impossible to guarantee that the flow of each phase will be more-or-less uniformly distributed among each of the tubes.

A rough estimate of the coefficient may be obtained from the constant-quality version of the Boyko-Kruzhilin equation:

$$h_i = 0.024 \frac{k_\ell}{d_i} \mathrm{Re}_{i,\ell}^{\ 0.8}\, \mathrm{Pr}_\ell^{\ 0.43} \sqrt{1 + \left(\frac{\rho_\ell - \rho_v}{\rho_v} \right) x} \qquad (2.31)$$