27

Heat-Exchange Equipment

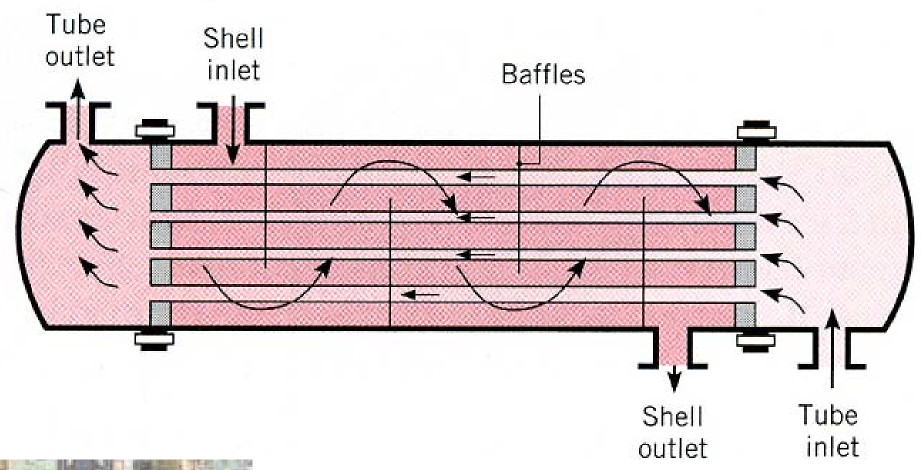
Heat Exchangers

In this chapter we shall consider the application of heat transfer theory to the design and operation of certain types of heat exchange equipment.

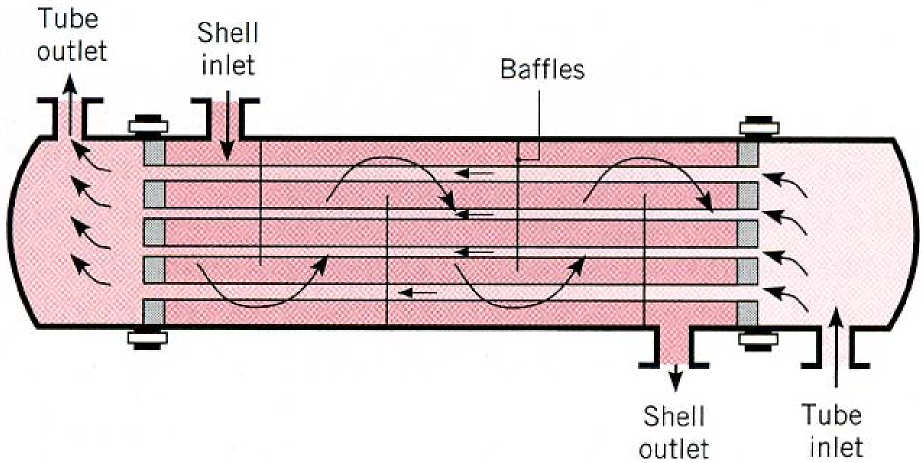
We have covered a number of such applications in previous chapters, but the following topics are best discussed after all the principal mechanisms of heat transfer have been studied.

Shell-and-Tube Heat Exchangers

The heat exchangers found most commonly in industry contain a number of parallel tubes enclosed in a single shell and for that reason are called shell-and tube exchangers.



Baffles

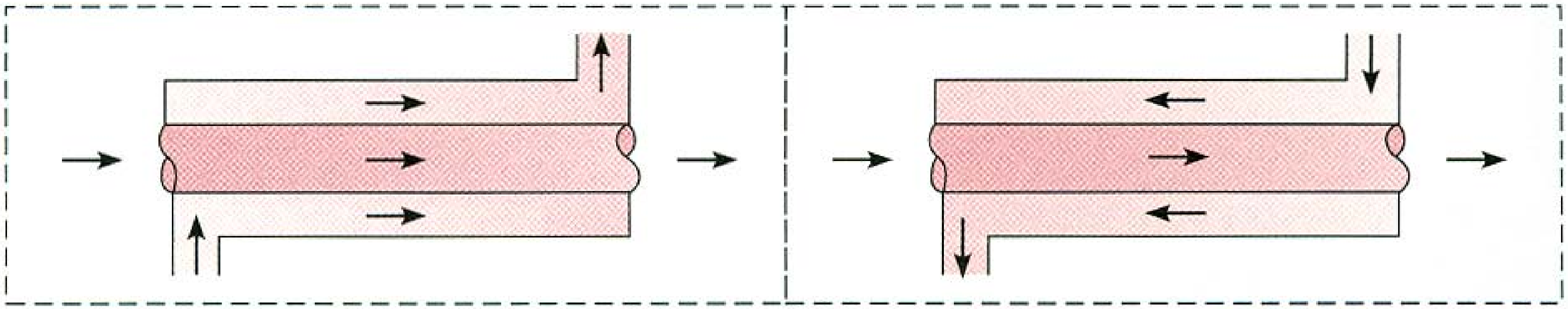


The main purpose of the baffles is to minimize channeling by which some of the fluid flows preferentially in certain paths that have little contact with the heat transfer surface. Each baffle may extend to half of the shell cross section; they are spaced at distances as close as one-fifth of the shell diameter.

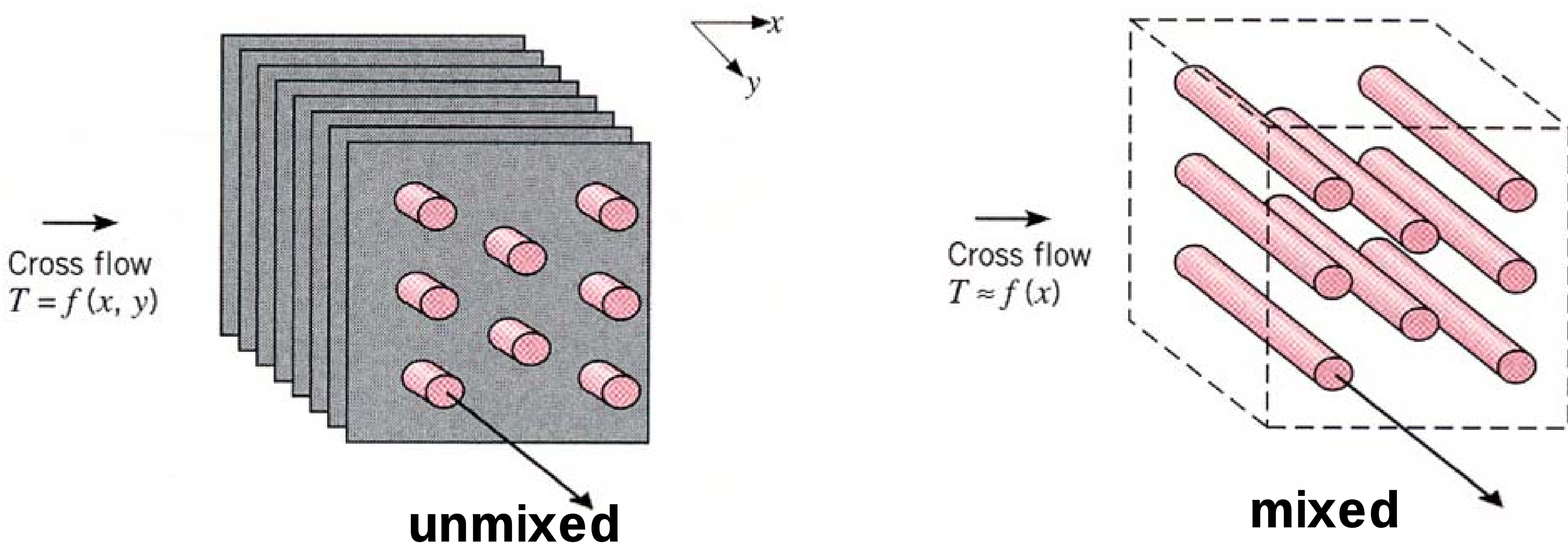
In addition to providing more uniform flow and heat transfer characteristics, the baffles also help to support the tubes. In heat exchangers in which condensation or boiling is taking place on the shell side, baffles are not necessary; hence tube supports are installed.

Heat Exchangers

parallel-flow **counterflow**



cross flow



Compact heat exchanger

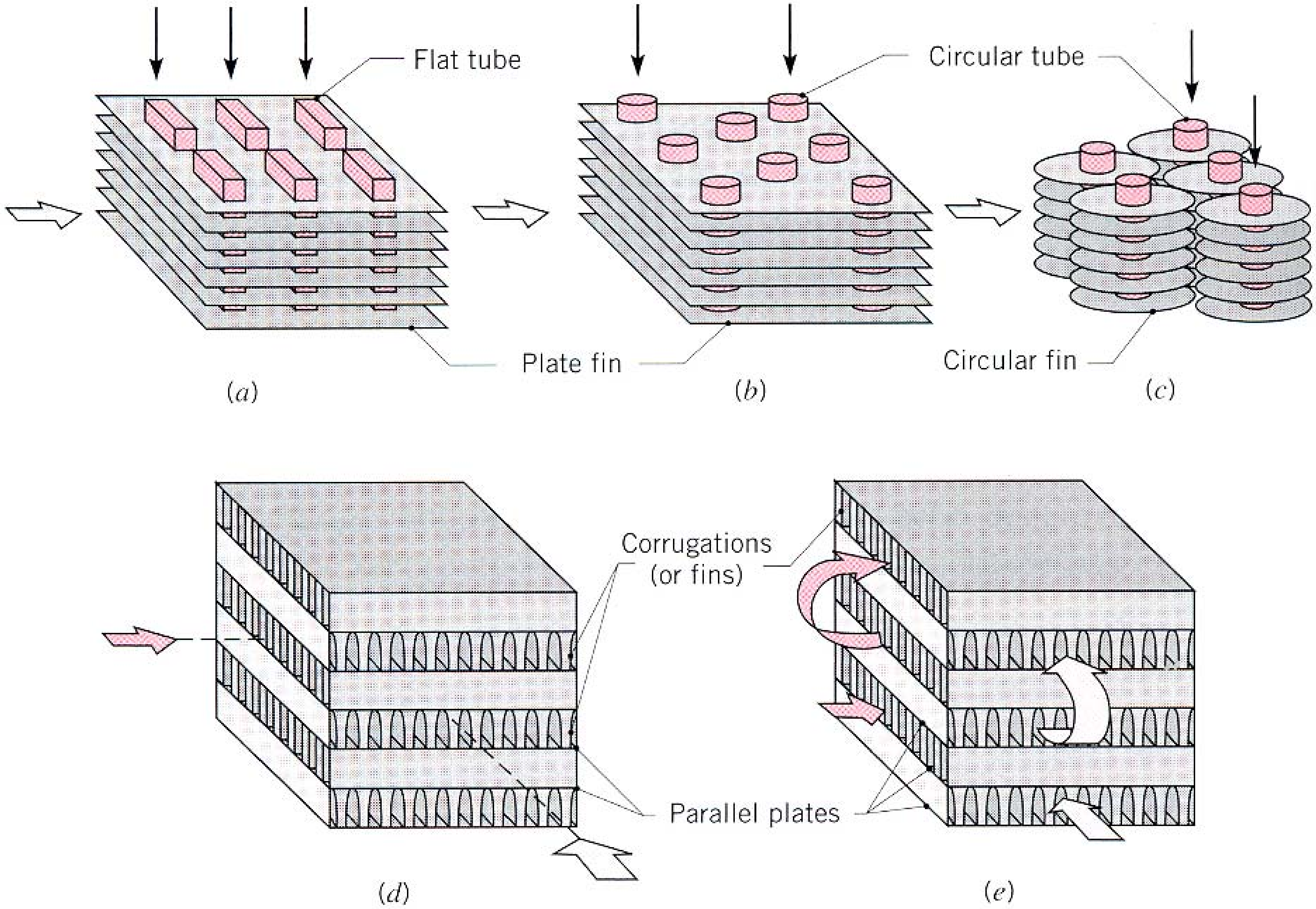


FIGURE 11.5 Compact heat exchanger cores. (*a*) Fin–tube (flat tubes, continuous plate fins). (*b*) Fin–tube (circular tubes, continuous plate fins). (*c*) Fin–tube (circular tubes, circular fins). (*d*) Plate–fin (single pass). (*e*) Plate–fin (multipass).

Heat transfer area

$$dq = w_c C_{pc} dt_c = w_h C_{ph} dt_h = U_0 (t_h - t_c) dA_0 \qquad (21-25)$$

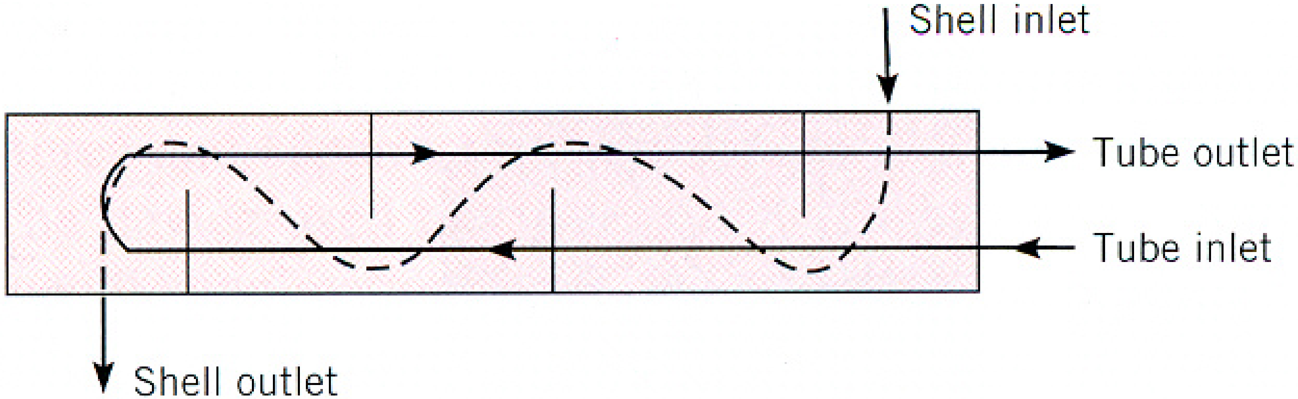
$$U_0 \ is \ const.: q = U_0 A_0 \frac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2 / \Delta t_1)} \qquad (21-29)$$

$$U_0 \neq f(\Delta t): w_h C_{ph} \int_{t_{h2}}^{t_{h1}} \frac{dt_h}{U_0 (t_h - t_c)} = \int_0^{A_0} dA_0 \qquad (21-35)$$

$$U_0 = a + b\Delta t: \quad q = \frac{U_{01}\Delta t_2 - u_{02}\Delta t_1}{\ln(U_{01}\Delta t_2 / U_{02}\Delta t_1)} A_0 \qquad (21-34)$$

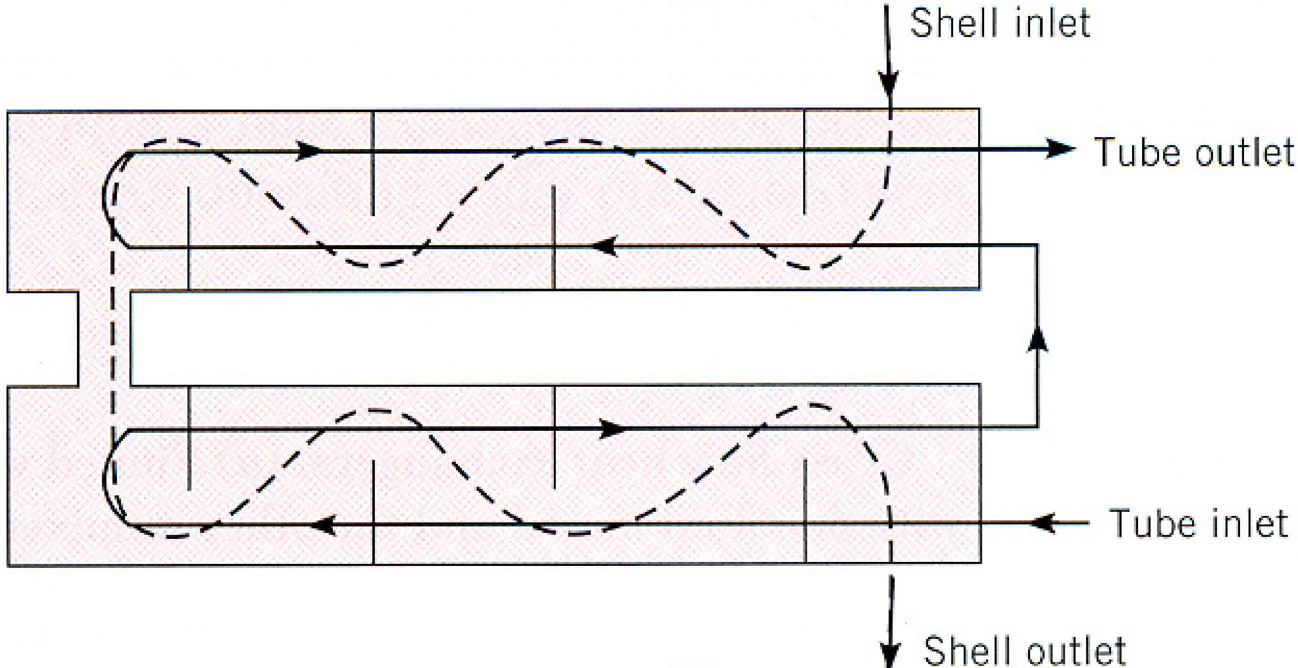
Shell-and-Tube Heat Exchangers

One shell pass & two tube passes



(a)

Two shell passes & four tube passes



(b)

Multipass and cross-flow heat exchangers

Bowman, Mueller, and Nagle

Introducing the correction factor Y
for cross flow passes of the shell-side fluid

$$q = U_0 A_0 Y \frac{\Delta t_2 - \Delta t_1}{\ln\left(\Delta t_2 / \Delta t_1\right)} \qquad (27-1)$$

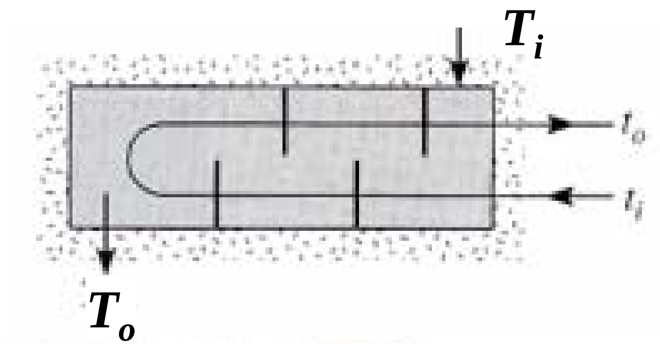
$$Y(F): correction\ factor$$

Δt_1 and Δt_2 are the terminal temperature differences taken as if the fluids in the cross-flow heat exchanger were in countercurrent flow.

Multipass and cross-flow heat exchangers

$$q = U_0 A_0 Y \frac{\Delta t_2 - \Delta t_1}{\ln\left(\Delta t_2 / \Delta t_1\right)} \qquad (27-1)$$

$Y(F): correction\ factor$



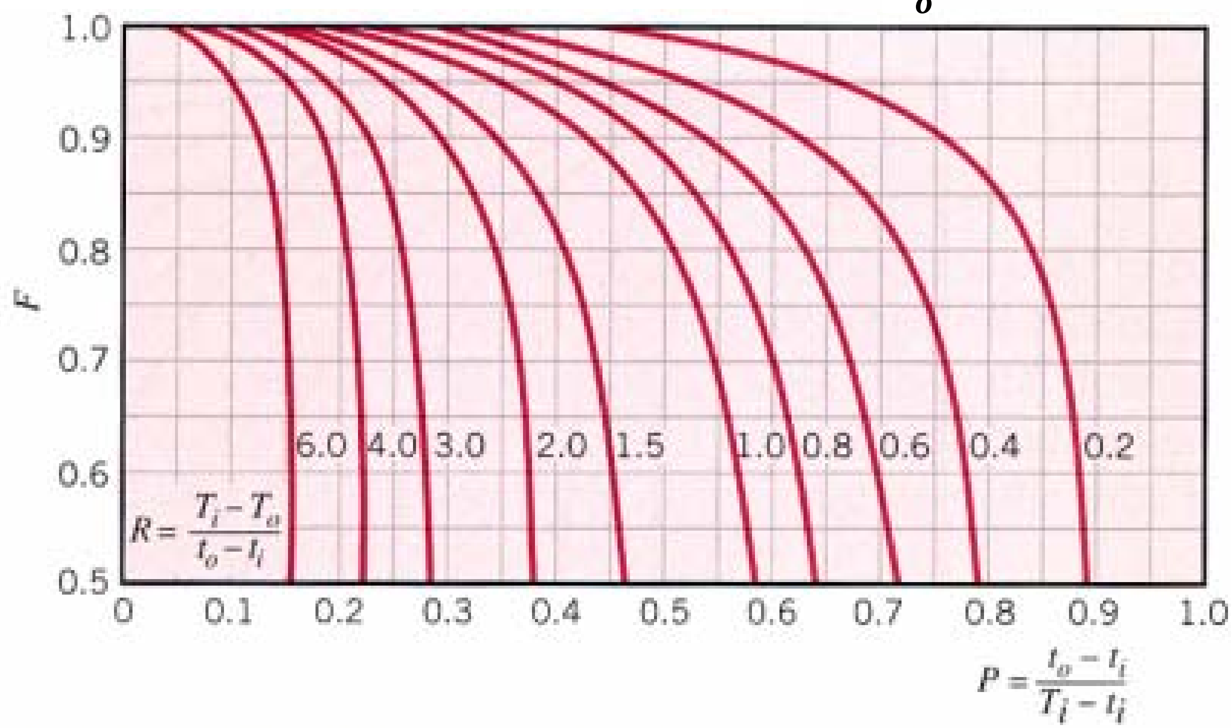
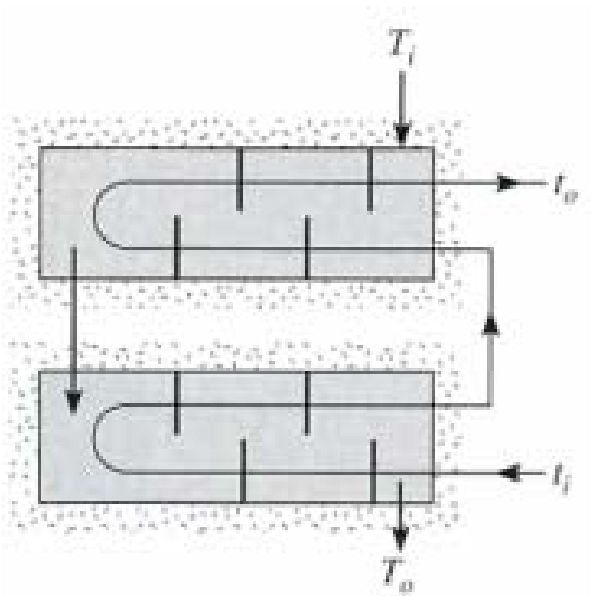


FIGURE 11.10 Correction factor for a shell-and-tube heat exchanger with one shell and any multiple of two tube passes (two, four, etc. tube passes).

Multipass and cross-flow heat exchangers

$$q = U_0 A_0 Y \frac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2 / \Delta t_1)} \qquad (27-1)$$

$$Y = F : correction\ factor$$



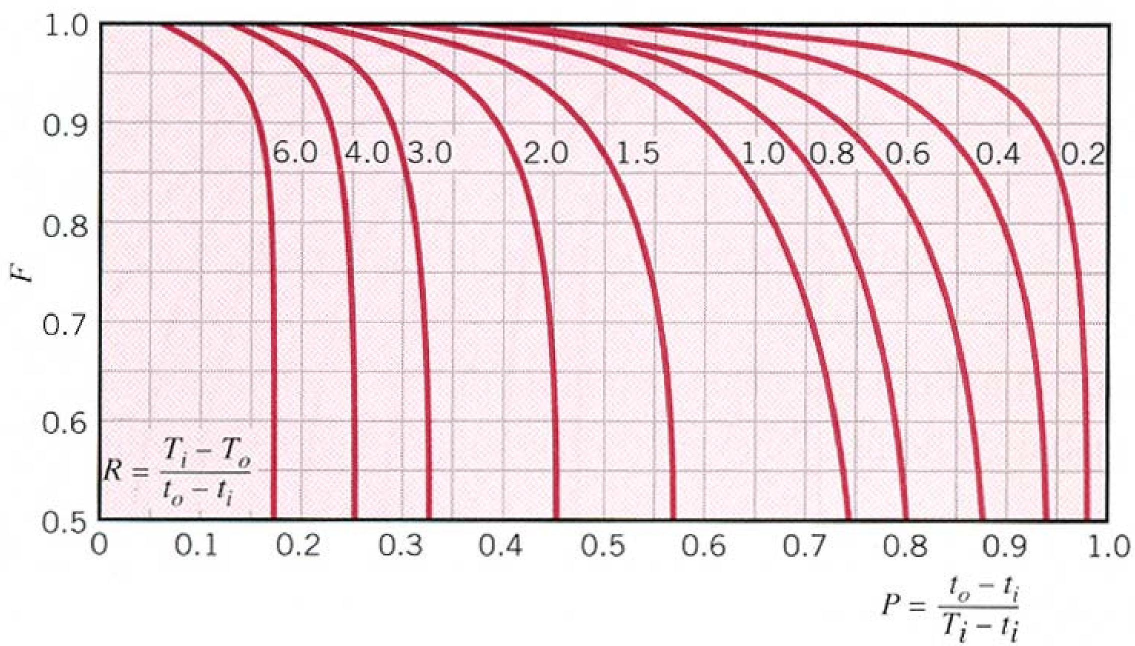
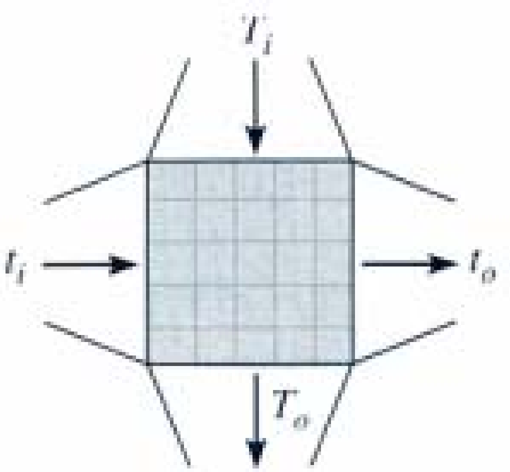


FIGURE 11.11 Correction factor for a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes (four, eight, etc. tube passes).

Multipass and cross-flow heat exchangers

$$q = U_0 A_0 Y \frac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2 / \Delta t_1)} \qquad (27-1)$$

$$Y = F : correction\ factor$$



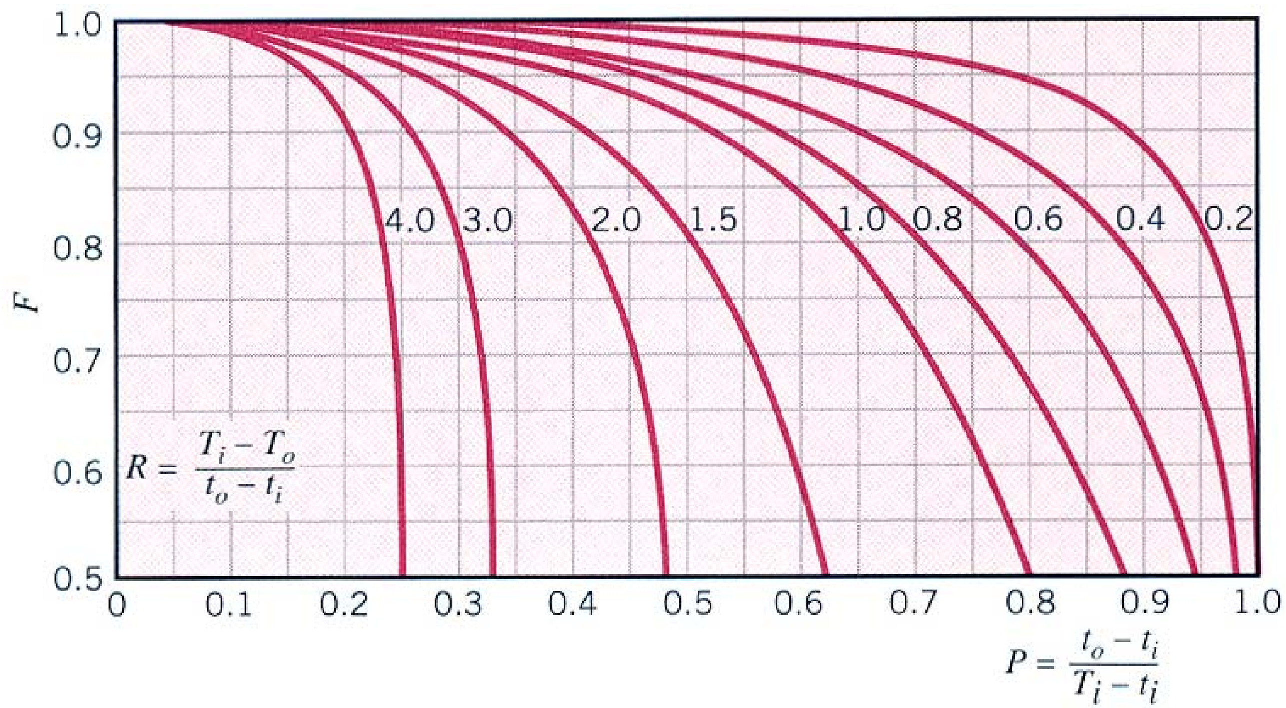
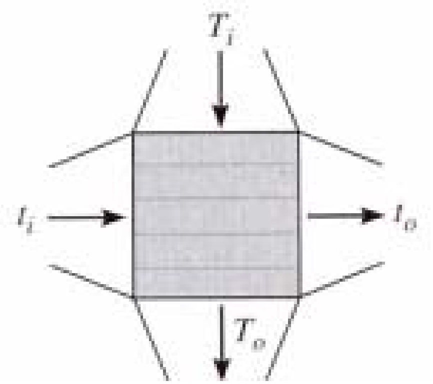


FIGURE 11.12 Correction factor for a single-pass, cross-flow heat exchanger with both fluids unmixed.

Multipass and cross-flow heat exchangers

$$q = U_0 A_0 Y \frac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2 / \Delta t_1)} \qquad (27-1)$$

$$Y = F : correction\ factor$$



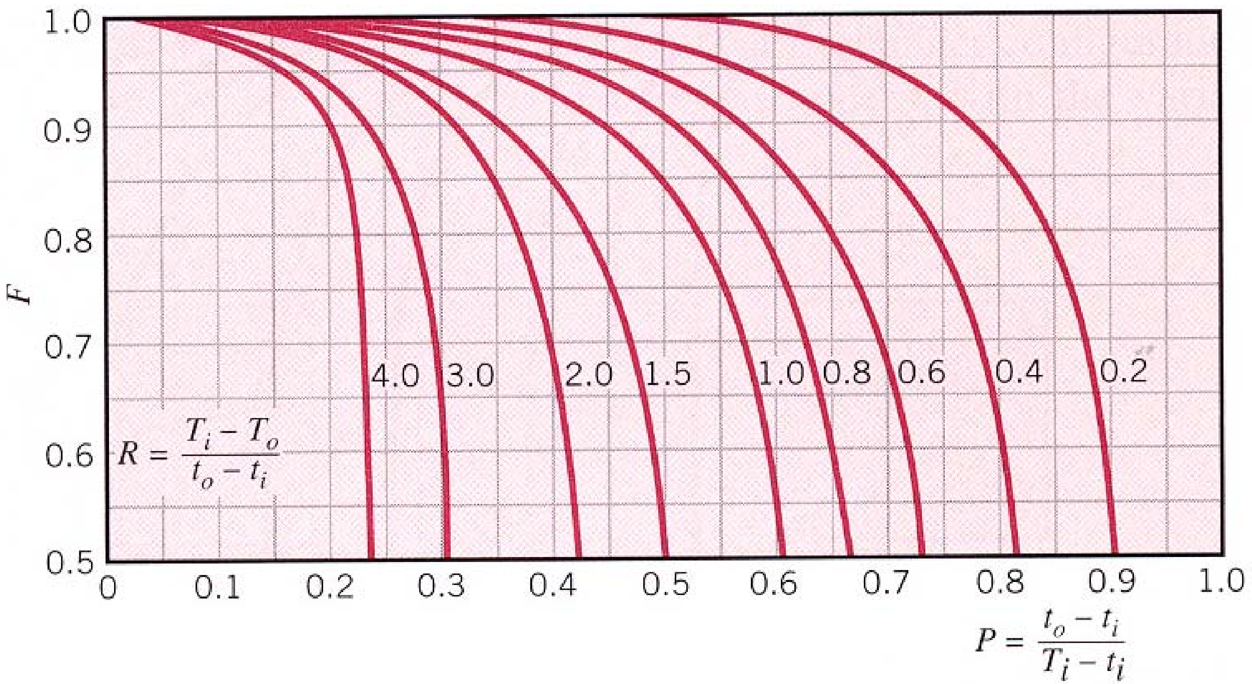


FIGURE 11.13 Correction factor for a single-pass, cross-flow heat exchanger with one fluid mixed and the other unmixed.

Example 27-1

1 shell : hot gas $t_{si}=160^\circ F,\ t_{so}=102^\circ F$

8 tubes : a liquid $t_{ti}=52^\circ F,\ t_{to}=87^\circ F$

$q=100{,}000\,Btu/h$ $U_0=5.5\,Btu/(h)(ft^2)(^\circ F)$

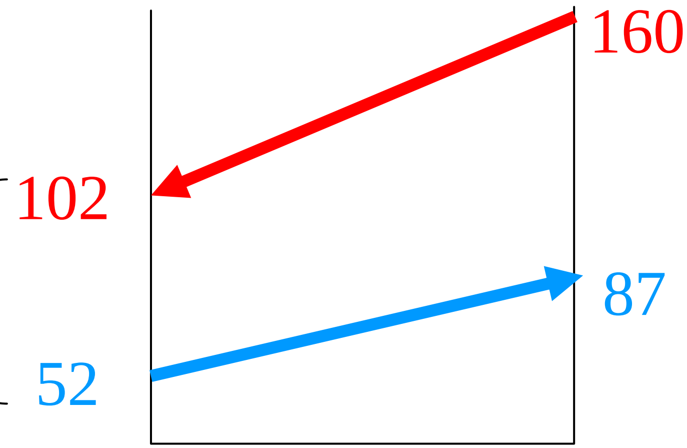
$$x=\frac{t_{t0}-t_{ti}}{t_{si}-t_{ti}}=\frac{87-52}{160-52}=0.32$$

$$z=\frac{t_{si}-t_{s0}}{t_{t0}-t_{ti}}=\frac{160-102}{87-52}=1.66$$

Example 27-1

from Fig 27-4, **Y=0.89**

Δt_1 and Δt_2 are the terminal temperature differences taken as if the fluids in the cross-flow heat exchanger were in countercurrent flow.



$$\Delta t_2 = 73^\circ F$$

$$\Delta t_1 = 50^\circ F$$

$$\therefore \quad A_0 = \frac{q}{Y \cdot U_0} \frac{\ln\left(\Delta t_2 / \Delta t_1\right)}{\Delta t_2 - \Delta t_1} = \frac{(100{,}000)}{(\mathbf{0.89})(5.5)} \frac{\ln 73/50}{73-50} = 336(ft^2)$$

Extended Surfaces

$$\frac{1}{U_0 A_0} = \Sigma R = \frac{1}{h_0 A_0} + \boxed{\left(\frac{\Delta r}{k A_{lm}}\right)_{metal}} + \frac{1}{h_i A_i}$$

$$assume: normally\ negligible$$

Extended surface ↑, $\boldsymbol{A_0}$ ↑

$$i)\ if\ \ h_0 \cong small,\ A_0 \uparrow,\ \Sigma R \downarrow$$

$$ii)\ if\ \ h_0 \cong big,\ A_0 \uparrow,\ negligible\ reduction\ in\ \Sigma R$$

Example 27-2

Air is to be heated by the condensation of steam.

$$h_{air} = 60 W/m^2K$$

$$h_{steam} = 6000 W/m^2K \;\; in \; \frac{3''}{4}, 18 - guage \; Cu \; tube$$

$$q = ?$$

basis : 1m length

$A_i = 0.052 m^2$, $A_0 = 0.060 m^2$

Fin = 600 circular fins/m of 0.00132 m height

Example 27-2

(a) steam inside; air outside

tube = copper with no fins

$$\Sigma R = \frac{1}{(6000)(0.052)} + \frac{1}{(60)(0.060)} = \frac{1}{U_0 A_0} = 0.281 \left(K/W\right)$$

$$\therefore\ q = \frac{\Delta t}{\Sigma R} = \frac{55}{0.281} = 196 \left(W\right)$$

Example 27-2

(b) steam inside; air outside

tube = copper with fins

$$\Sigma R = \frac{1}{(6000)(0.052)} + \frac{1}{(60)(0.161)} = 0.104 (K/W)$$

$$\therefore q = \frac{\Delta t}{\Sigma R} = \frac{55}{0.104} = 514 (W)$$

Example 27-2

(b) steam outside; air inside

tube = copper with fins

$$\Sigma R = \frac{1}{(6000)(0.161)} + \frac{1}{(60)(0.052)} = 0.322\left(K/W\right)$$

$$\therefore\ q = \frac{\Delta t}{\Sigma R} = \frac{55}{0.322} = 171\left(W\right)$$

Example 27-2

(a) steam inside; air outside; tube without fins, q=196W

(b) steam inside; air outside; tube with fins, q=514W

(c) steam outside; air inside; tube with fins, q=171W

In system (b), the effect of the fins in reducing the major resistance is cause a significant increase in the heat-transfer rate over the rate in system (a).

In system (c), however, the effect of the fins is merely to reduce further the resistance, which was negligible even on an unfinned surface.

The major resistance, which is that of the air, is at inside surface of the tube, which is smaller in area than the outside of the smooth tube. Thus the total resistance in system (c) with the finned tubes is greater than in system (a), when smooth tubes were used.

Fin efficiencies

Although the resistance to heat transfer of the metal wall was neglected in Example 27-2, this procedure is not always justifiable. Fins as high as 1 in and only a few hundreds of an inch in thickness are not uncommon, and even when made of copper, such fins may offer a significant resistance to heat transfer. This resistance is, of course, more likely to be significant when the fluid resistances are small.

A quantitative method of describing the effect of the fin resistance is by the use of **fin efficiency**. <u>This quantity is defined as the actual heat-transfer rate from a fin divided by the rate that would be obtained if the entire fin were at the temperature of the base of the fin, i.e., the outer cylindrical surface of the tube.</u>

Fin efficiencies

$$\text{Fin efficient } (\eta_f) = \frac{\text{actual heat - transferred}}{\begin{array}{c}\text{heat which would be transferred} \\ \text{if entire fin area were at base temperature}\end{array}}$$

The heat flow from a tube surface to a fluid can be written as

$$q = hA_f \eta_f \Delta t + hA_t \Delta t \qquad (27\text{-}3)$$

where A_f = fin area

A_t = area of tube surface between fins

η_f = fin efficiency

h = heat-transfer coefficient,

assumed constant at all points on fin and tube surface

Δt = temperature difference between base of fin and bulk-fluid phase

Fin efficiencies

$$q = hA_f \eta_f \Delta t + hA_t \Delta t \qquad (27\text{-}3)$$

When written in Ohm's law form, Eq. (27-3) becomes

$$q = \frac{\Delta t}{\dfrac{1}{h(A_f \eta_f + A_t)}} = \frac{\Delta t}{\sum R} \qquad (27\text{-}4)$$

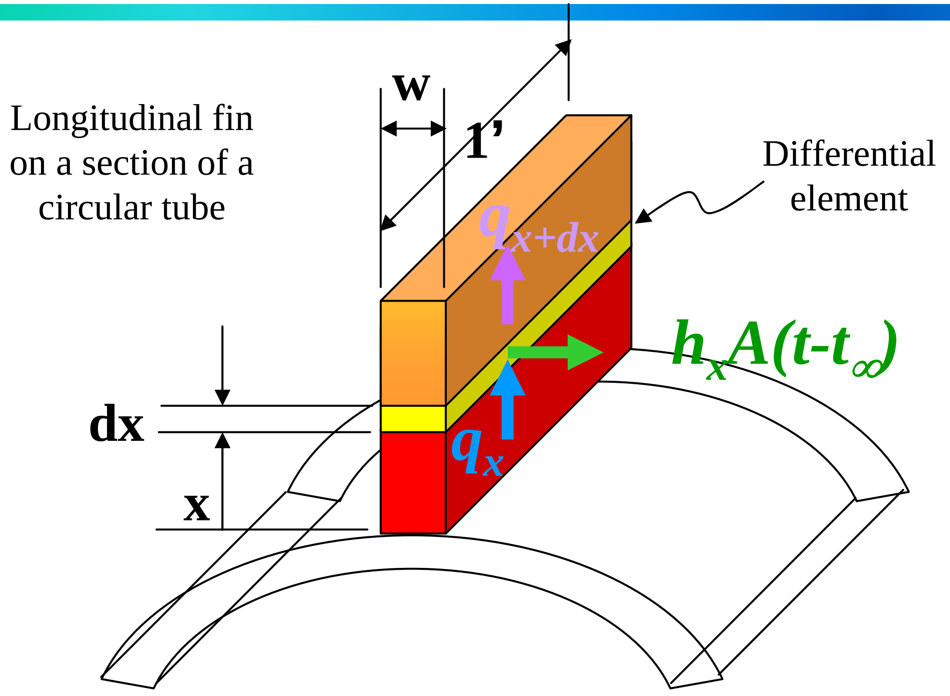
Fin efficiencies have been calculated for a number of configurations. A simple system is examined in Example 27-3, and a chart for determining the efficiency of a circular fin is given in Fig. 27-5.

Example 27-3: Fin efficiencies

A longitudinal steel fin 1 in high and 1/8 in thick is exposed to an air stream at 70^{o}F. The air moving past the fin has a uniform convection coefficient ***h***=15 Btu/(h)(ft^2)(^{o}F). The fin has a thermal conductivity of 25 Btu/(h)(ft)(^{o}F) and a base temperature of 250^{o}F.

Calculate the fin efficiency and the heat flow per linear foot assuming that all temperature gradients in planes parallel to the base of the fin are negligible. A section of the fin is shown in Fig. 27-6.

Example 27-3: Fin efficiencies



At steady state

$$q_x - q_{x+dx} = h_x A(t\text{-}t_\infty)$$

$$-kw\frac{dt}{dx}\bigg|_x - \left(-kw\frac{dt}{dx}\right)\bigg|_{x+dx} = h_x\left(2\cdot\Delta x\right)\left(t-t_\infty\right)$$

Rate of heat conduction in | **Rate of heat conduction out** | **Rate of heat convection out**

Fin efficiencies

$$-kw\frac{dt}{dx}\bigg|_{x} - \left(-kw\frac{dt}{dx}\right)\bigg|_{x+dx} = h_x(2\cdot\Delta x)(t-t_\infty)$$

$$\div\,\Delta x,\ \lim\ \Delta x \to 0$$

$$kw\frac{d^2t}{dx^2} = 2h_x(t-t_\infty)$$

$$\boxed{\begin{array}{l} \dfrac{d^2t}{dx^2} - \dfrac{2h_x}{kw}(t-t_\infty) = 0 \\ \\ B.C.'s \quad at\ \ x = 0, \quad t = 250 \\ \qquad\qquad at\ \ x = H,\ -kw\dfrac{dt}{dx} = hw(t-t_\infty) \end{array}}$$

Fin efficiencies

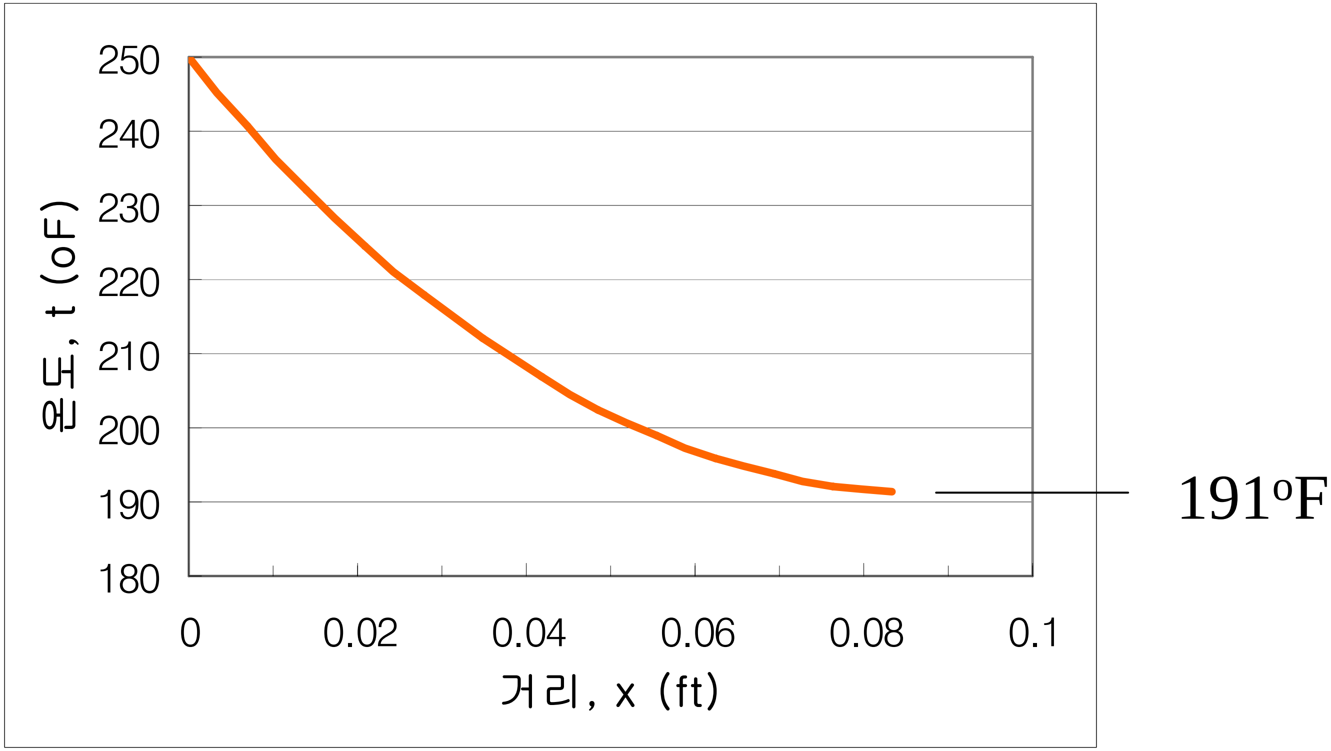
$$t-70 = C_1 e^{\alpha x} + C_2 e^{-\alpha x}$$

$$\alpha = \sqrt{\frac{2h}{kw}} = \sqrt{\frac{(2)(15)}{(25)(1/96)}} = 10.7\ ft^{-1}$$

$$C_1 = 23.4, \quad C_2 = 156.6$$

Fin efficiencies

$$t = 70 + 23.4e^{10.7x} + 156.6e^{-10.7x}$$



Fin efficiencies

total heat loss:
= The rate of heat conduction into the fin at the base

$$q = -kA\left(\frac{dt}{dx}\right)\Bigg|_{x=0}$$

$$= -(25)\left(\frac{1}{12}\right)\left(\frac{1}{8}(23.4\times 10.7 - 156.6\times 10.7)\right)$$

$$= \underline{370\,Btu/(h)(ft)}$$

total heat loss:
If the fin were at $250^\circ F$ through

$$q = hA(250-70)$$

$$= (15)\left(\frac{1}{12}+\frac{1}{8}\times\frac{1}{10}+\frac{1}{12}\right)(250-70)$$

$$= \underline{478\,Btu/(h)(ft)}$$

$$\therefore \boxed{\eta_f = \frac{370}{478} = 0.78}$$

Wilson's Methods of Analysis

A useful method of determining the individual convective heat-transfer coefficients in a heat exchanger is that of Wilson.

If a single-phase fluid is flowing in developed turbulent flow inside the tubes of an exchanger, the convective coefficient h_i can be estimated by the Dittus-Boelter equation.

$$\frac{h_i D}{k} = 0.023\left(\frac{D u_b \rho}{\mu}\right)^{0.8}\left(\frac{C_p \mu}{k}\right)^{0.3} \qquad (24\text{-}4)$$

This equation shows that $\boldsymbol{h_i}$ is proportional to $\boldsymbol{u_b^{0.8}}$ if everything else is held constant. However, we shall find it desirable to change the average temperature of the fluid as $\boldsymbol{u_b}$ changes; the effect of the temperature if the fluid is water is represented by the term (1 + 0.011$\boldsymbol{t}$) in the equation.

$$h_i = a u_b^{0.8}(1 + 0.011t) \qquad (27\text{-}5)$$

Wilson's Methods of Analysis

The constant a could be calculated by equating (24-4) and (27-5), but it is usually determining from the results of the Wilson-line experiment described below. The temperature ***t*** is the average water temperature in degrees Fahrenheit.

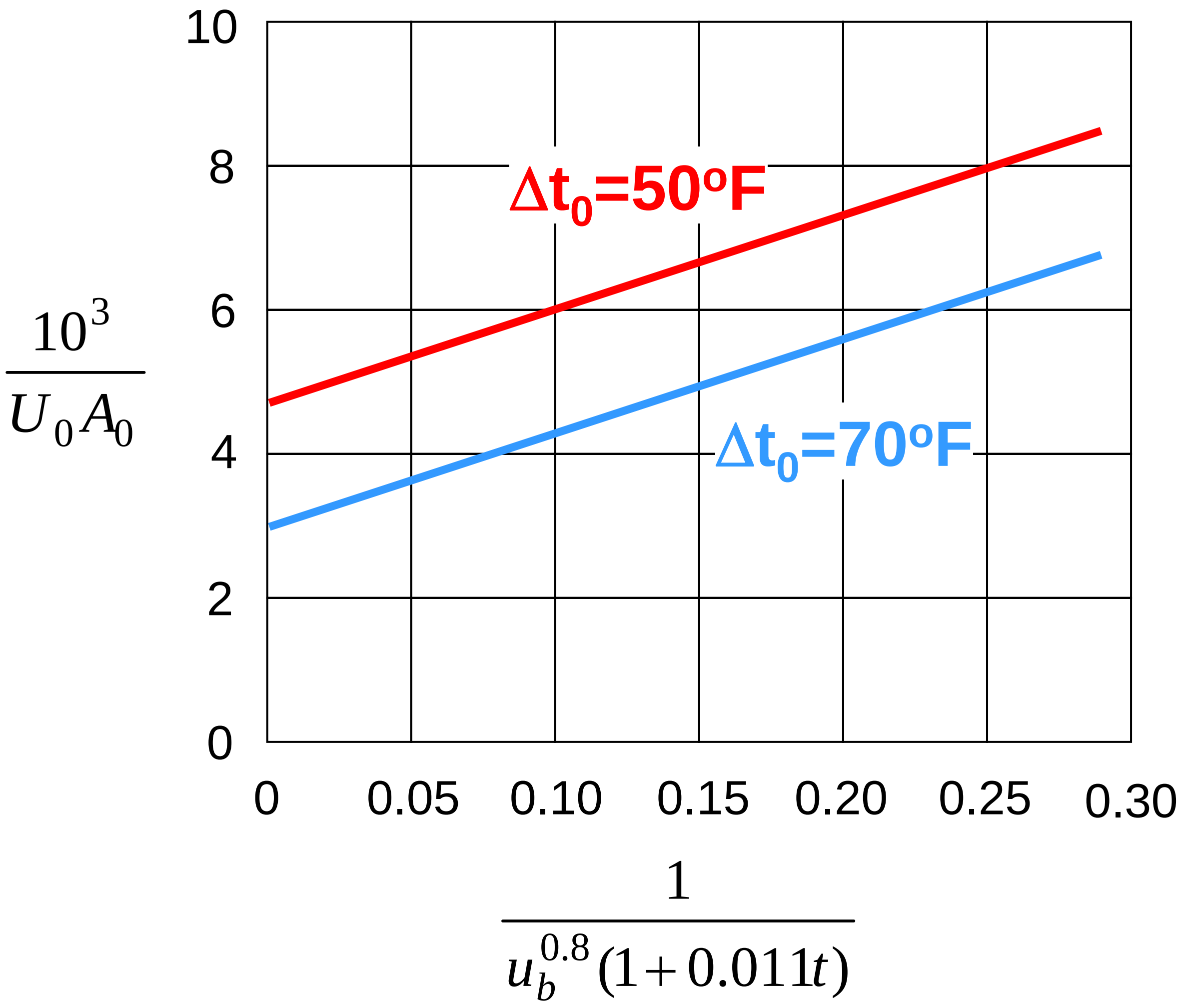
If a series of runs is conducted in which the velocity of the water in tubes is varied, the overall coefficient can be represented by the equation.

$$\frac{1}{U_0 A_0} = \frac{1}{h_0 A_0} + \frac{\Delta r}{k A_{lm}} + \frac{1}{a u_b^{0.8}(1+0.011t)A_i} \qquad (27\text{-}6)$$

The first two terms on the right-hand side are the resistances of the outside fluid and the tube wall, respectively. If they are held constant for all runs, a linear relation exist between $1/U_0A_0$ and $1/u_b^{0.8}(1+0.011t)$.

Fig. 27-7: Wilson plot for boiling methanol on 1½” tube

R.P. Lance, AIChE J., **4**, 75(1958)



Wilson's Methods of Analysis

It is essential that all resistances other than the inside resistance be held as nearly constant as possible during runs at different velocities. This means that the outside coefficient $\boldsymbol{h_0}$ must be held constant. Because $\boldsymbol{h_0}$ is a function of temperature drop for such system as boiling, condensation, and natural convection, care must be taken that the temperature drop over the fluid outside the tube is held constant for successive runs.

This is achieved by altering the temperature level of the fluid inside the tube each time its velocity is changed, so that the inclusion of the factor (1+0.011t) is essential for accurate results. Conditions are usually chosen so that the temperature change of the water is small from inlet to outlet.

Wilson’s Methods of Analysis

Fouling coefficients can be estimated by the Wilson method if the outside fluid coefficient $\boldsymbol{h_0}$ can be predicted or is negligible. The fouling resistance $\boldsymbol{1/h_{d0}A_0}$ is a part of the intercept value.

In some pieces of equipment the value of the exponent on $\boldsymbol{u_b}$ in Eq. (27-5) may not be known. Upstream disturbances may cause it to differ from the value of 0.8 and for developed turbulent flow in circular pipes. If an incorrect exponent is used in the plot (for example, 0.8 when it should still extrapolate to the proper value of the ordinate representing all the other resistances. However, the line will no longer be quite straight, and extrapolation is more difficult. Nevertheless, the curvature is often not great enough to cause serious errors, and the method remains, in spite of this deficiency, a useful technique.

Homework

27-1

27-2