BOILING HEAT TRANSFER
FLOW BOILING

MCADAMS

$$q/A = 2.253\,(\Delta T_x)^{3.96} \qquad 0.2 < p < 0.7 \text{ MPa}$$

LEVY

$$q/A = 283.2\,p^{4/3}(\Delta T_x)^3 \qquad 0.7 < p < 14 \text{ MPa}$$

JACOB & HAWKINS

$$h = h_{ATMOSPHERIC}\,(p\,/\,p_{ATMOSPHERIC})^{0.4}$$

$h_{ATMOSPHERIC}$ SEE TABLE

JACOB

$$h = 2.54\,(\Delta T_x)^3\,e^{p/1.551}$$

$$\Delta T_x = T_{WALL} - T_{SATURATION}$$

p IN MEGAPASCALS

Table 9-3 Simplified Relations for Boiling Heat-Transfer Coefficients to Water at Atmospheric Pressure, Adapted from Ref. 15. $\Delta T_x = T_w - T_{sat}$, °C.

Surface	$\frac{q}{A}$, kW/m²	h, W/m² · °C
Horizontal	$\frac{q}{A} < 16$	$1042\,(\Delta T_x)^{1/3}$
	$16 < \frac{q}{A} < 240$	$5.56\,(\Delta T_x)^3$
Vertical	$\frac{q}{A} < 3$	$537\,(\Delta T_{ax})^{1/7}$
	$3 < \frac{q}{A} < 63$	$7.96\,(\Delta T_x)^3$

5.4 Boiling Heat Transfer

Heat transfer coefficients for boiling are naturally very variable as they are influenced by flow and pressure as well as by the degree of agitation caused by the boiling itself. A number of empirical equations are available. The following are from *J.P. Holman.*

- At atmospheric pressure:

Horizontal tubes	$q / A < 16$	$h = 1042\,(\Delta T_x)^{1/3}$
	$16 < q / A < 240$	$h = 5.56\,(\Delta T_x)^3$
Vertical tubes	$q / A < 3$	$h = 537\,(\Delta T_x)^{1/7}$
	$3 < q / A < 63$	$h = 7.96\,(\Delta T_x)^3$

$$\Delta T_x = T_{wall} - T_{saturation}$$

- At higher pressures:

The values given above are corrected as follows:

$$h_{high\ pressure} = h\,(p_{high} / p_{atmospheric})^{0.4}$$

The units applicable to the above are q/A in kW m^{-2}, h in W $m^{-2}\,°C^{-1}$ and T in °C.

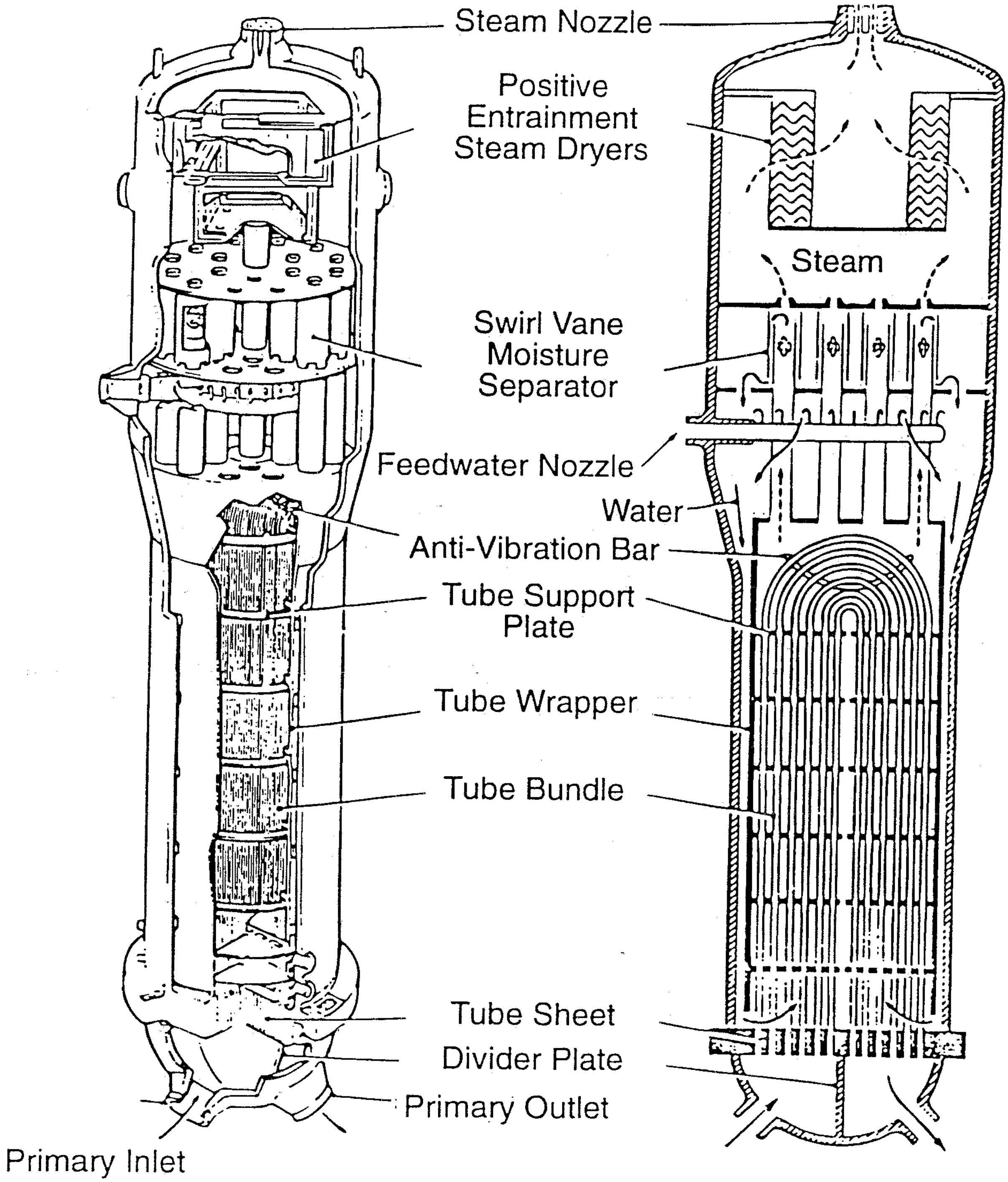
- At all pressures inside vertical tubes:

$$h = 2.54\,(\Delta T_x)^3\, e^{p/1.551}$$

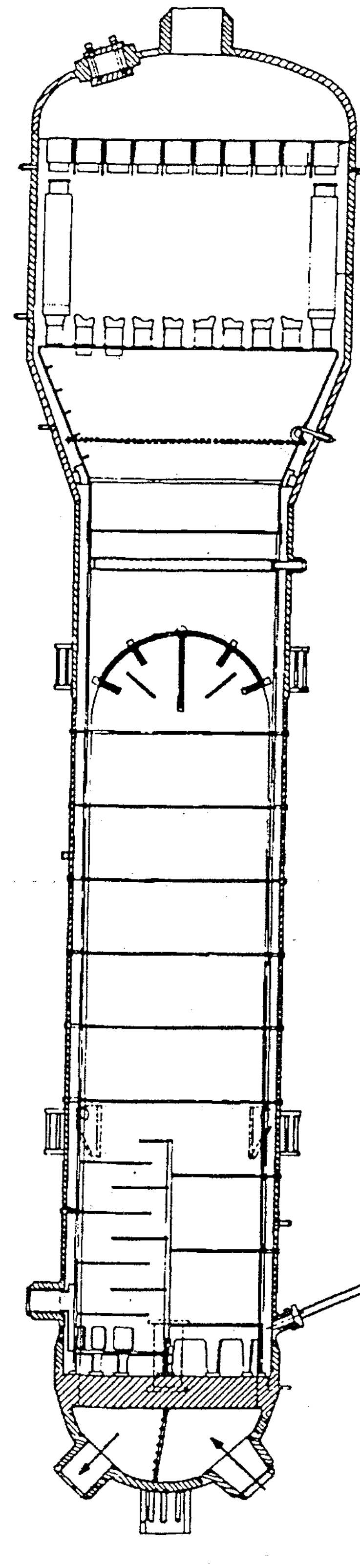
$$\Delta T_x = T_{wall} - T_{saturation}$$

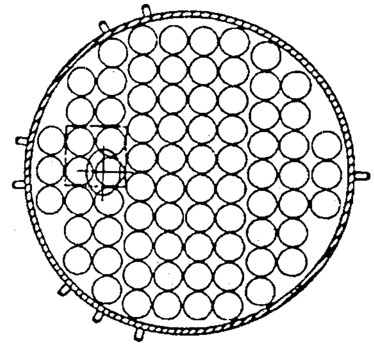
Here h is in W $m^{-2}\,°C^{-1}$, p in MPa and T in °C.

Actual Boiler
(Westinghouse)



Actual Boiler (Point Lepreau)





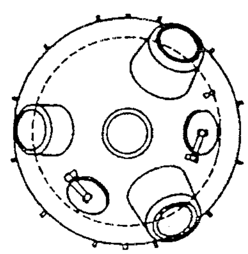


Figure 2.3

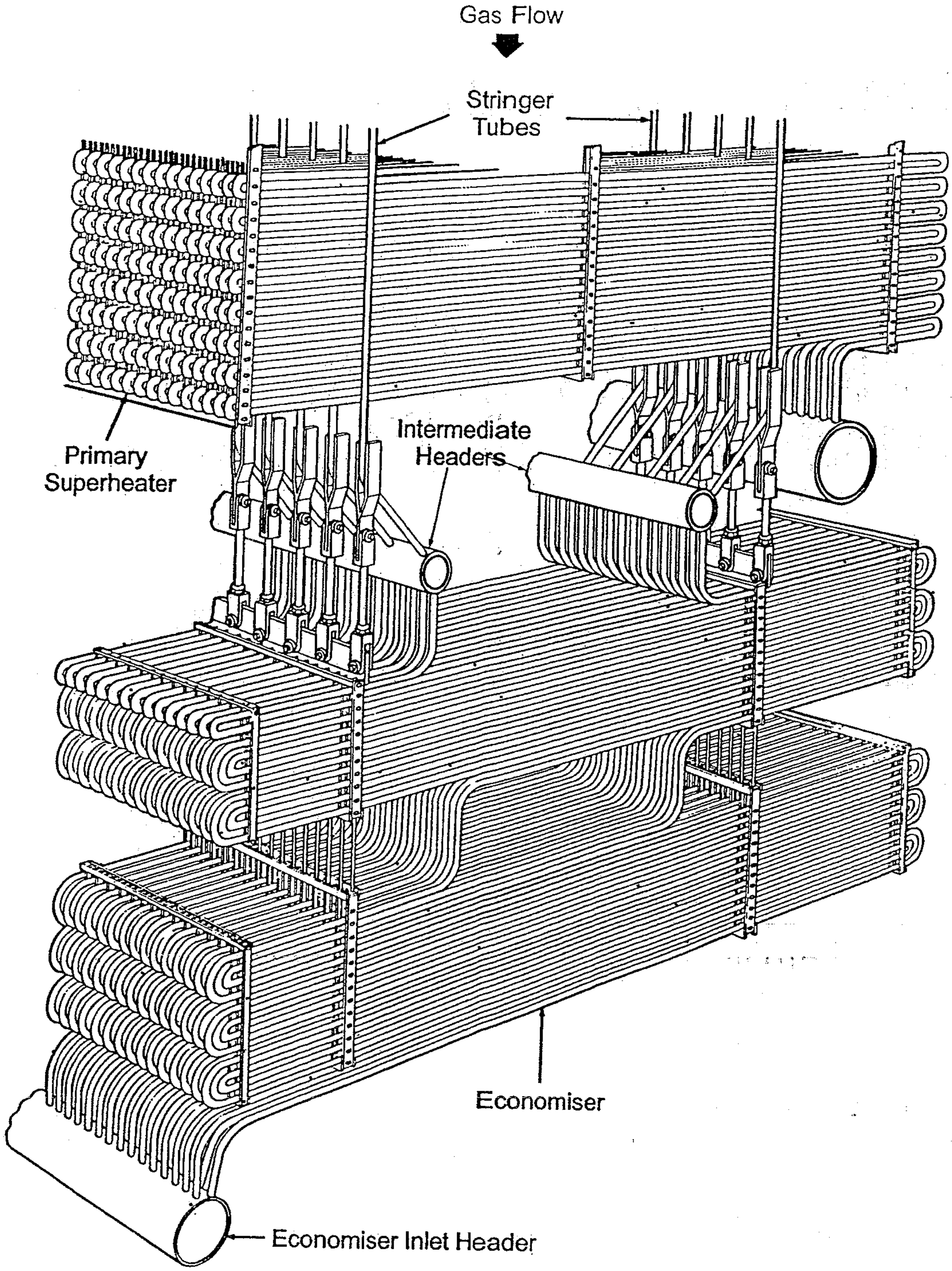


Figure 12 Economiser tube bank (courtesy of Babcock & Wilcox)

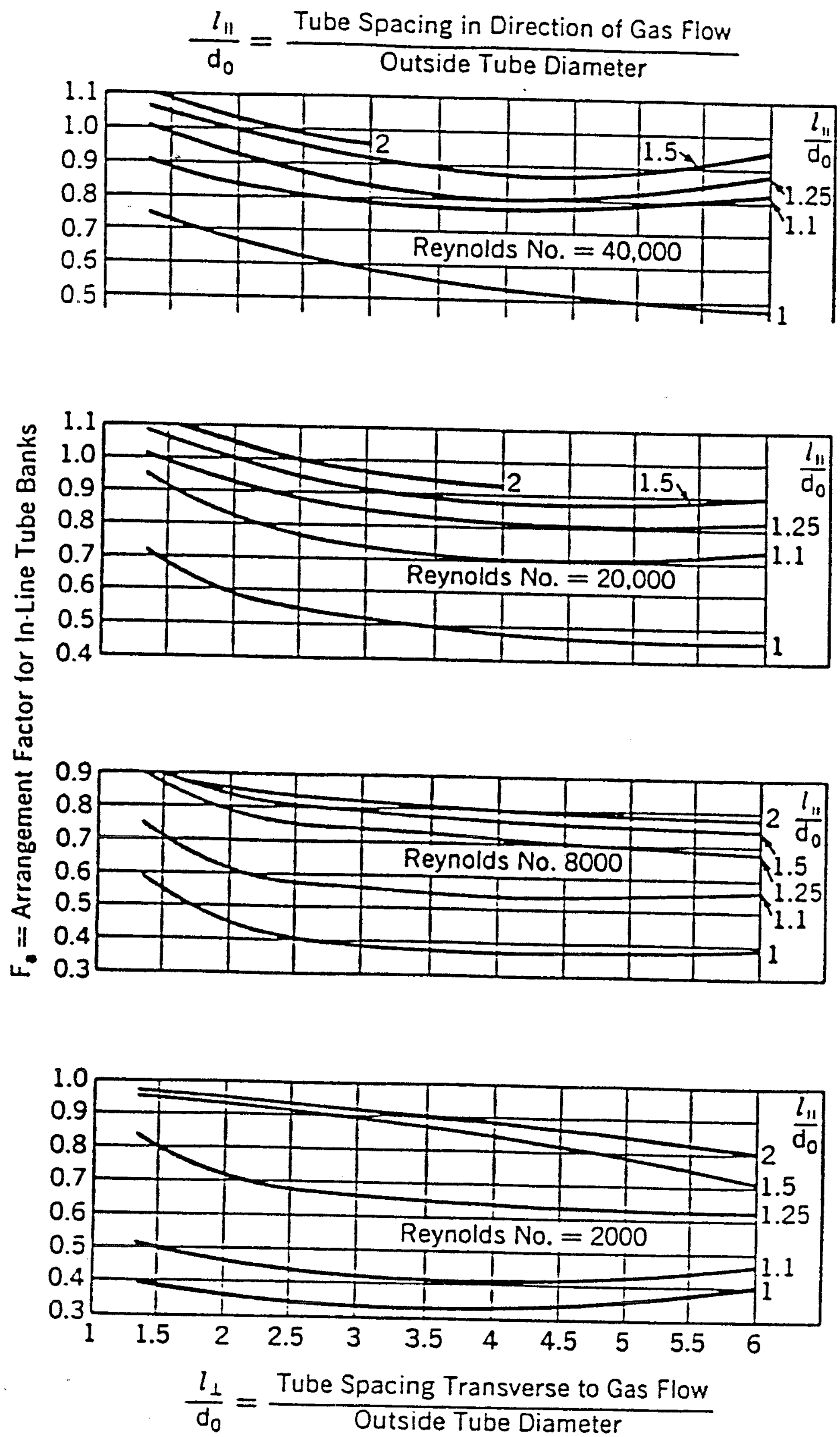


Fig. 14 Arrangement factor, F_a, as affected by Reynolds number for various in-line tube patterns, crossflow gas or air.

9.3. Convection in Flow across Tube Bundles

It is not possible here to say much about this topic, except that a great deal of experimental work has been done to find suitable empirical relationships for heat transfer coefficients. The reader is referred to volumes such as Kays and London[7] for extended coverage of the present situation.

In general, results correlate in the usual form, i.e., $\overline{Nu} = f(RePr)$, provided the characteristic dimension is expressed as some function of tube diameter and tube spacing. Various definitions have been proposed by different investigators. These results for average convection coefficients must be used with caution since local coefficients do not as a rule become constant until at least the third row of tubes in the bundle.

In addition to the heat transfer measurements, a great deal of work has been done to measure friction coefficients, expressible as functions of tube spacing and Reynolds number.

6-4 FLOW ACROSS TUBE BANKS

Because many heat-exchanger arrangements involve multiple rows of tubes, the heat-transfer characteristics for tube banks are of important practical interest. The heat-transfer characteristics of staggered and in-line tube banks were studied by Grimson [12], and on the basis of a correlation of the results of various investigators, he was able to represent data in the form of Eq. (6-17). The values of the constant C and the exponent n are given in Table 6-4 in terms of the geometric parameters used to describe the tube-bundle arrangement. The Reynolds number is based on the maximum velocity occurring in the tube bank, i.e., the velocity through the minimum-flow area. This area will depend on the geometric tube arrangement. The nomenclature for use with Table 6-4 is shown in Fig. 6-14. The data of Table 6-4 pertain to tube banks having 10 or more rows of tubes in the direction of flow. For fewer rows the ratio of h for N rows deep to that for 10 rows is given in Table 6-5.

Table 6-4 Correlation of Grimson for Heat Transfer in Tube Banks of 10 Rows or More, From Ref. 12, for Use with Eq. (6-17).

$\frac{S_p}{d}$	$\frac{S_n}{d}$ = 1.25		1.5		2.0		3.0	
	C	*n*	*C*	*n*	*C*	*n*	*C*	*n*
				In line				
1.25	0.386	0.592	0.305	0.608	0.111	0.704	0.0703	0.752
1.5	0.407	0.586	0.278	0.620	0.112	0.702	0.0753	0.744
2.0	0.464	0.570	0.332	0.602	0.254	0.632	0.220	0.648
3.0	0.322	0.601	0.396	0.584	0.415	0.581	0.317	0.608
				Staggered				
0.6	—	—	—	—	—	—	0.236	0.636
0.9	—	—	—	—	0.495	0.571	0.445	0.581
1.0	—	—	0.552	0.558	—	—	—	—
1.125	—	—	—	—	0.531	0.565	0.575	0.560
1.25	0.575	0.556	0.561	0.554	0.576	0.556	0.579	0.562
1.5	0.501	0.568	0.511	0.562	0.502	0.568	0.542	0.568
2.0	0.448	0.572	0.462	0.568	0.535	0.556	0.498	0.570
3.0	0.344	0.592	0.395	0.580	0.488	0.562	0.467	0.574

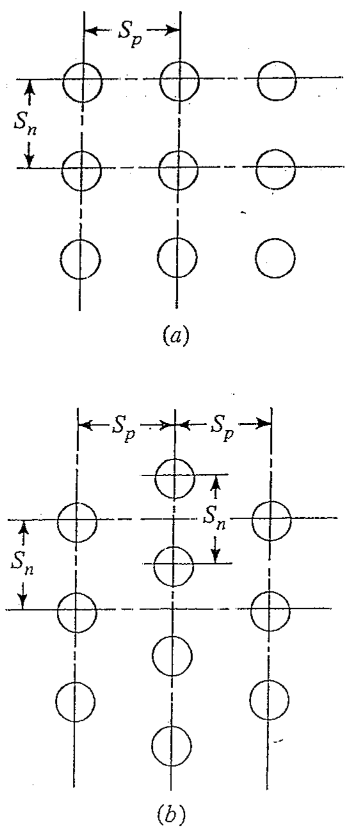


Fig. 6-14 Nomenclature for use with Table 6-4: (*a*) in-line tube rows; (*b*) staggered tube rows

Table 6-5 Ratio of h for N Rows Deep to that for 10 Rows Deep. From Ref. 17.

N	1	2	3	4	5	6	7	8	9	10
Ratio for staggered tubes	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99	1.0
Ratio for in-line tubes	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99	1.0

Pressure drop for flow of gases over a bank of tubes may be calculated with Eq. (6-31), expressed in pascals:

$$\Delta p = \frac{2f' G_{\max}^2 N}{\rho} \left(\frac{\mu_w}{\mu_b}\right)^{0.14} \tag{6-31}$$

where $G_{\max}$ = mass velocity at minimum flow area, kg/m$^2 \cdot$ s
ρ = density evaluated at free-stream conditions, kg/m^3
N = number of transverse rows
μ_b = average free-stream viscosity

The empirical friction factor f' is given by Jakob [18] as

$$f' = \left\{0.25 + \frac{0.118}{[(S_n - d)/d]^{1.08}}\right\} \mathrm{Re}_{\max}^{-0.16} \tag{6-32}$$

for staggered tube arrangements, and

$$f' = \left\{0.044 + \frac{0.08 S_p/d}{[(S_n - d)/d]^{0.43 + 1.13d/S_p}}\right\} \mathrm{Re}_{\max}^{-0.15} \tag{6-33}$$

for in-line arrangements.

Zukauskas [39] has presented additional information for tube bundles which takes into account wide ranges of Reynolds numbers and property variations. The correlating equation takes the form

$$\mathrm{Nu} = \frac{\bar{h}d}{k} = C\,\mathrm{Re}_{d,\max}^n\,\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{1/4} \tag{6-34}$$

where all properties except Pr_w are evaluated at T_∞ and the values of the constants are given in Table 6-6 for greater than 20 rows of tubes. This equation is applicable for $0.7 < \mathrm{Pr} < 500$ and $10 < \mathrm{Re}_{d,\max} < 10^6$. For gases the Prandtl number ratio has little influence and is dropped. Once again, note that the Reynolds number is based on the maximum velocity in the tube bundle. For less than 20 rows in the direction of flow the correction factor in Table 6-7 should be applied. It is essentially the same as for the Grimson correlation. Additional information is given by Morgan [44]. Further information on pressure drop is given in Ref. 39.

Table 6-6 Constant for Zukauskas Correlation [Eq. (6-34)] for Heat Transfer in Tube Banks of 20 Rows or More. From Ref. 39.

Geometry	$Re_{d,max}$	C	n
In-line	10–100	0.8	0.4
	$100-10^3$	Treat as individual tubes	
	$10^3 - 2 \times 10^5$	0.27	0.63
	$>2 \times 10^5$	0.21	0.84
Staggered	10–100	0.9	0.4
	$100-10^3$	Treat as individual tubes	
	$10^3 - 2 \times 10^5$	$0.35\left(\frac{S_n}{S_L}\right)^{0.2}$ for $\frac{S_n}{S_L} < 2$	0.60
	$10^3 - 2 \times 10^5$	0.40 for $\frac{S_n}{S_L} > 2$	0.60
	$>2 \times 10^5$	0.022	0.84

The reader should keep in mind that these relations correlate experimental data with an accuracy of about ± 25 percent.

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	$>2 \times 10^5$	0.022	0.84

The reader should keep in mind that these relations correlate experimental data with an accuracy of about ± 25 percent.

HEAT TRANSFER

CONDUCTION IN FLAT PLATES

$$Q_x = -kA\frac{dT}{dx}$$

$$Q_x/A = -k\,(T_2 - T_1)/(x_2 - x_1)$$

CONDUCTION IN CYLINDRICAL SHELLS

$$Q_r = -k\,A_r\frac{dT}{dx}$$

$$Q_r/A_r = -k\,(T_2 - T_1)/\left(r_r \ln(r_2/r_1)\right)$$

CONVECTION AT SURFACE

$$Q = hA\Delta t$$

$$Q/A = h\Delta t$$

OVERALL HEAT TRANSFER COEFFICIENT U

$$Q = UA\Delta T$$

$$\Delta T = (Q/A)(1/U)$$

$$= (Q/A)\left(\frac{1}{h_1} + \frac{x}{k} + \frac{1}{h_2}\right)$$

$$\frac{1}{U} = \frac{1}{h_1} + \frac{x}{k} + \frac{1}{h_2}$$

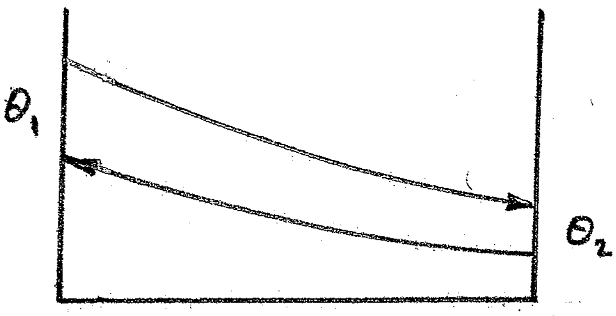
TEMPERATURE DIFFERENCE

AVERAGE TEMPERATURE DIFFERENCE

$Q = UA\,\theta_{AVE}$

$\theta_{AVE} = \frac{1}{2}(\theta_1 + \theta_2)$

DIFFERENCE < 40%



LOG MEAN TEMPERATURE DIFFENCE

$Q = UA\,\theta_m$

$\theta_M = (\theta_1 - \theta_2) / Ln(\theta_1 / \theta_2)$

COUNTER-FLOW ARRANGEMENT

SAME FOR PARALLEL-FLOW ARRANGEMENT

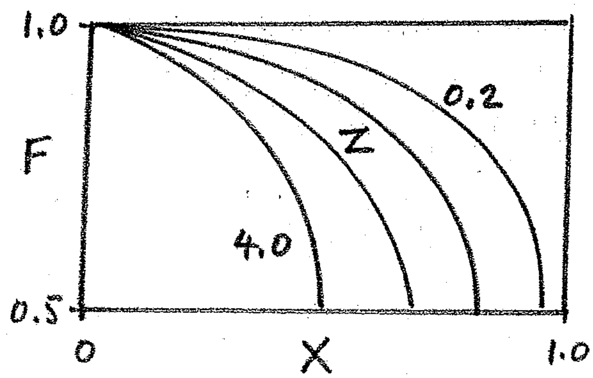
CROSS FLOW CORRECTION FACTOR

$Q = UAF\,\theta_m$

F FROM CHART

$Z = \frac{(t_1 - t_2)_{SHELL}}{(t_2 - t_1)_{TUBE}}$

$X = \frac{(t_2 - t_1)_{TUBE}}{(\Delta T)_{INLETS}}$



MIXED ON SHELL SIDE

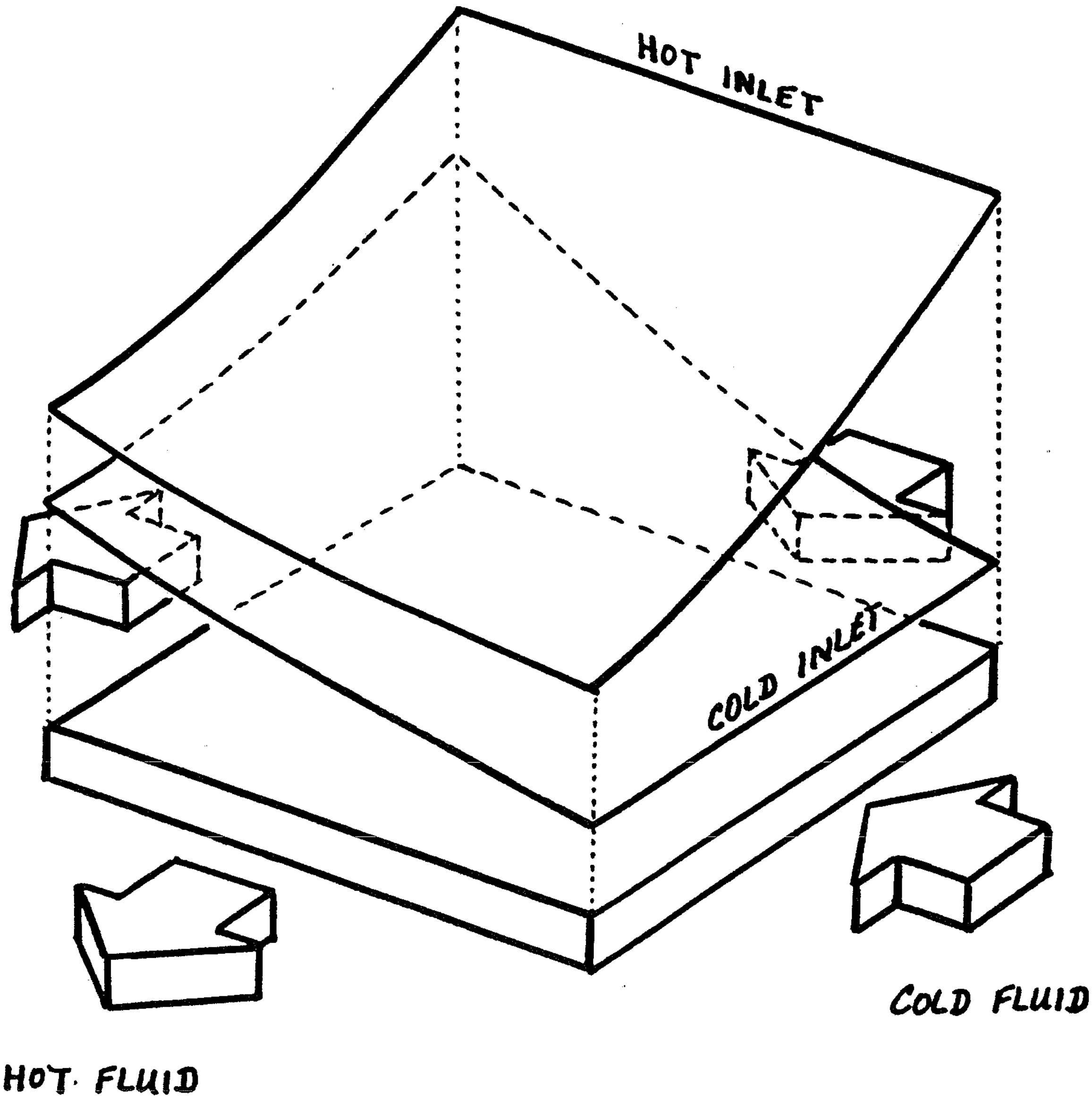
CROSS FLOW

Cross flow or transverse flow heat exchangers are common in many practical applications. One advantage is that the fluid path inlets and outlets are well separated leading to a convenient design. Another advantage is that improved mixing of one or sometimes both fluids can be achieved thus improving the heat flow between that fluid and the heat exchange surface. This is particularly true of a fluid flowing across the outside of a tube bank. Mathematical analysis of such flows is very difficult and much design is based on empirical relationships. Thus correction factors are available and can be applied as follows:

$$Q = UAF\Theta_m$$

where F is a correction factor for a particular configuration and conditions. Under certain conditions it is possible for F to be slightly greater than unity if the configuration actually creates an improvement in heat transfer over that of a single tube. Normally F is less than unity since the heat transfer of a tube in a tube bundle is less than that of single tube.

CROSS FLOW



9.3. Convection in Flow across Tube Bundles

It is not possible here to say much about this topic, except that a great deal of experimental work has been done to find suitable empirical relationships for heat transfer coefficients. The reader is referred to volumes such as Kays and London[7] for extended coverage of the present situation.

In general, results correlate in the usual form, i.e., $\overline{Nu} = f(RePr)$, provided the characteristic dimension is expressed as some function of tube diameter and tube spacing. Various definitions have been proposed by different investigators. These results for average convection coefficients must be used with caution since local coefficients do not as a rule become constant until at least the third row of tubes in the bundle.

In addition to the heat transfer measurements, a great deal of work has been done to measure friction coefficients, expressible as functions of tube spacing and Reynolds number.

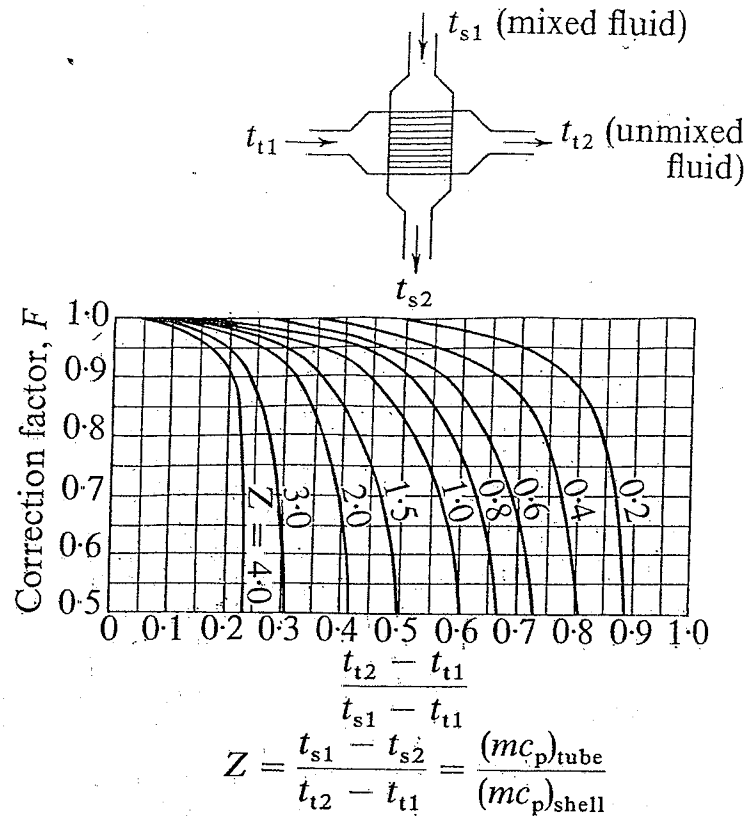


Fig. 13.5. Logarithmic temperature difference correction factor for cross flow, one fluid mixed, one fluid unmixed. From R. A. Bowman, A. E. Mueller, and W. M. Nagle. Trans. ASME, Vol. 62, p. 283 (1940). By permission of the American Society of Mechanical Engineers.

13.2.2. Cross Flow. Analysis of the cross-flow heat exchanger is more complicated owing to temperature variation across the flow. This variation will depend on whether the fluid is *mixed* or *unmixed*. A mixed fluid is free to move across the flow direction; an unmixed fluid is constrained in parallel flow passages. Thus, if an exchanger consisted of a bank of tubes placed across a duct, the fluid in the duct would be mixed while the fluid in the tubes would be unmixed.

Results of analyses of this type of exchanger are available as correction factors.[1,2] Equation (13.5) would become

$$Q = U_A A F \theta_m$$

where F is a factor to be obtained from the appropriate graph, and θ_m is the mean temperature difference, (13.17), calculated for counter flow with the same inlet and outlet temperatures as for cross flow. Figure 13.5 shows F for a cross-flow exchanger with one fluid mixed and one fluid unmixed. In applying the factor F it does not matter whether the hotter fluid is mixed or unmixed.

Multipass and Cross-Flow Heat Exchangers: Use of a Correction Factor

The log mean temperature difference ΔT_{lm} relation developed earlier is limited to parallel-flow and counter-flow heat exchangers only. Similar relations are also developed for *cross-flow* and *multipass shell-and-tube* heat exchangers, but the resulting expressions are too complicated because of the complex flow conditions.

In such cases, it is convenient to relate the equivalent temperature difference to the log mean temperature difference relation for the counter-flow case as

$$\Delta T_{lm} = F\,\Delta T_{lm,\,CF} \qquad (23\text{–}26)$$

where F is the **correction factor,** which depends on the *geometry* of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. The $\Delta T_{lm,\,CF}$ is the log mean temperature difference for the case of a *counter-flow* heat exchanger with the same inlet and outlet temperatures and is determined from Eq. 23–25 by taking $\Delta T_1 = T_{h,\,in} - T_{c,\,out}$ and $\Delta T_2 = T_{h,\,out} - T_{c,\,in}$ (Fig. 23–17).

The correction factor is less than unity for a cross-flow and multipass shell-and-tube heat exchanger. That is, $F \leq 1$. The limiting value of $F = 1$ corresponds to the counter-flow heat exchanger. Thus, the correction factor for a heat exchanger is *a measure of deviation of the ΔT_{lm} from the corresponding values for the counter-flow case.*

The correction factor F for common cross-flow and shell-and-tube heat exchanger configurations is given in Fig. 23–18 versus two temperature ratios P and R defined as

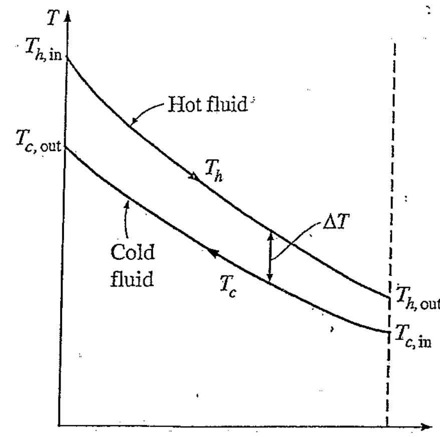
$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad (23\text{–}27)$$

and

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}C_p)_{\text{tube side}}}{(\dot{m}C_p)_{\text{shell side}}} \qquad (23\text{–}28)$$

where the subscripts 1 and 2 represent the *inlet* and *outlet,* respectively. Note that for a shell-and-tube heat exchanger, T and t represent the *shell-* and *tube-side* temperatures, respectively, as shown in the correction factor charts. It makes no difference whether the hot or the cold fluid flows through the shell or the tube. The determination of the correction factor F requires the availability of the *inlet* and the *outlet* temperatures for both the cold and hot fluids.

Note that the value of P ranges from 0 to 1. The value of R, on the other hand, ranges from 0 to infinity, with $R = 0$ corresponding to the phase-change (condensation or boiling) on the shell-side and $R \to \infty$ to phase-change on the tube side. The correction factor is $F = 1$ for both of these limiting cases. Therefore, the correction factor for a *condenser* or *boiler* is $F = 1$, regardless of the configuration of the heat exchanger.



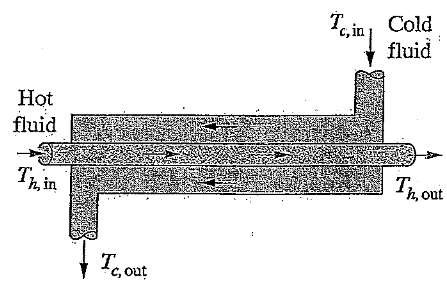
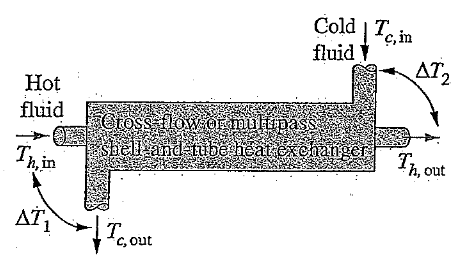


FIGURE 23–16
The variation of the fluid temperatures in a counter-flow double-pipe heat exchanger.



Heat transfer rate:

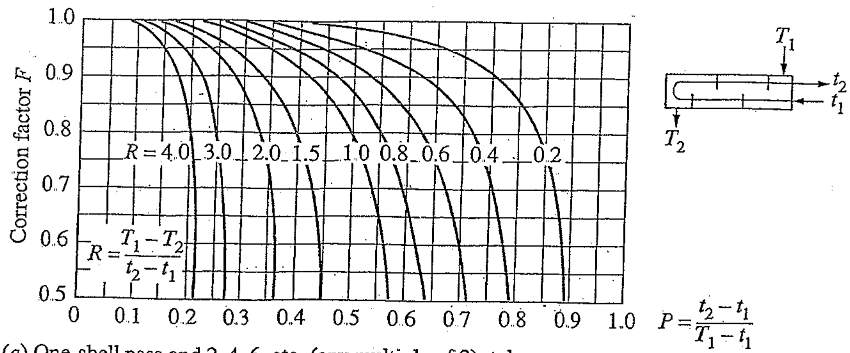
$$\dot{Q} = UA_sF\Delta T_{lm,CF}$$

where $\Delta T_{lm,CF} = \dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$

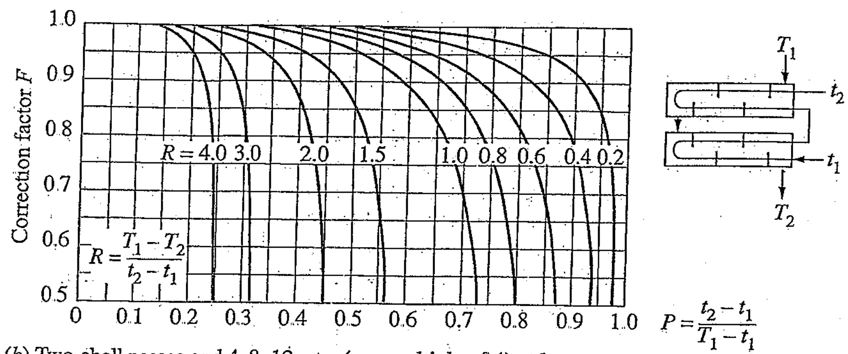
$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$

and $F = \ldots$ (Fig. 23–18)

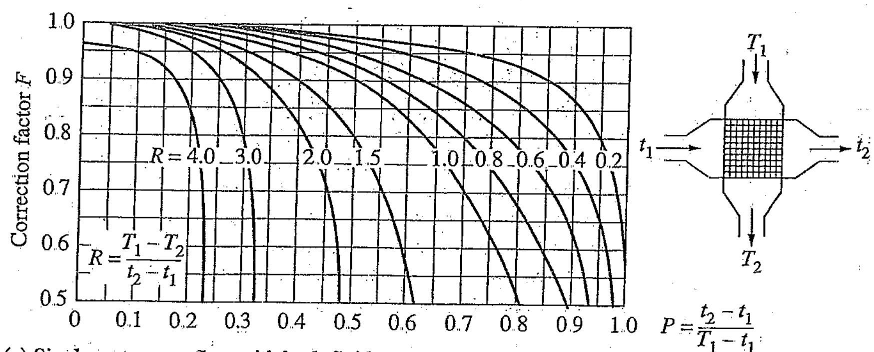
FIGURE 23–17
The determination of the heat transfer rate for cross-flow and multipass shell-and-tube heat exchangers using the correction factor.



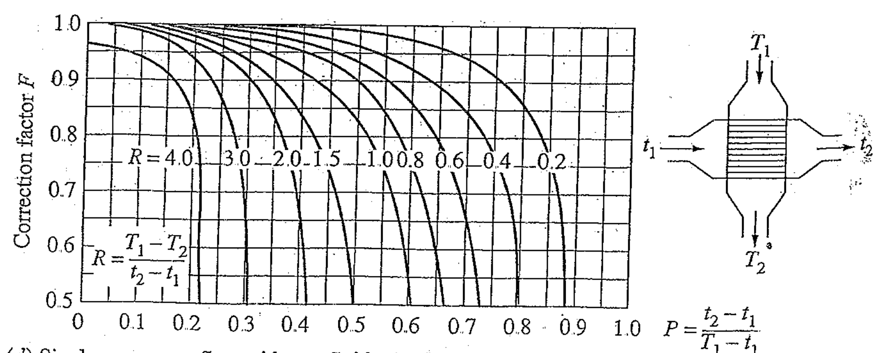
(*a*) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



(*b*) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



(*c*) Single-pass cross-flow with both fluids *unmixed*



(*d*) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

FIGURE 23–18
Correction factor F charts for common shell-and-tube and cross-flow heat exchangers (from Bowman, Mueller, and Nagle).