Overall fouling factor: 000625 $(Wm^{-2}\,°C^{-1})^{-1}$.

Pressure drops, including drop over nozzles:

Tube-side, estimated 0·4 bar.

Shell-side, estimated 0·6 bar.

12.9.4. Bell's method

In Bell's method the heat-transfer coefficient and pressure drop are estimated from correlations for flow over ideal tube-banks, and the effects of leakage, bypassing and flow in the window zone are allowed for by applying correction factors.

This approach will give more satisfactory predictions of the heat-transfer coefficient and pressure drop than Kern's method; and, as it takes into account the effects of leakage and bypassing, can be used to investigate the effects of constructional tolerances and the use of sealing strips. The procedure in a simplified and modified form to that given by Bell (1963), is outlined below.

The method is not recommended when the by-pass flow area is greater than 30% of the cross-flow area, unless sealing strips are used.

Heat-transfer coefficient

The shell-side heat transfer coefficient is given by:

$$h_s = h_{oc} F_n F_w F_b F_L \tag{12.27}$$

where h_{oc} = heat transfer coefficient calculated for cross-flow over an ideal tube bank, no leakage or bypassing,

F_n = correction factor to allow for the effect of the number of vertical tube rows,

F_w = window effect correction factor,

F_b = bypass stream correction factor,

F_L = leakage correction factor.

The total correction will vary from 0·6 for a poorly designed exchanger with large clearances to 0·9 for a well-designed exchanger.

h_{oc}, ideal cross-flow coefficient

The heat-transfer coefficient for an ideal cross-flow tube bank can be calculated using the heat transfer factors j_h given in Fig. 12.31. Figure 12.31 has been adapted from a similar figure given by Mueller (1973). Mueller includes values for more tube arrangements than are shown in Fig. 12.31. As an alternative to Fig. 12.31, the comprehensive data given in the Engineering Sciences Data Unit report on heat transfer during cross-flow of fluids over tube banks, ESDU No. 73031 (1973), can be used; see Butterworth (1977).

The Reynolds number for cross-flow through a tube bank is given by:

$$Re = \frac{G_s d_o}{\mu} = \frac{u_s \rho d_o}{\mu}$$

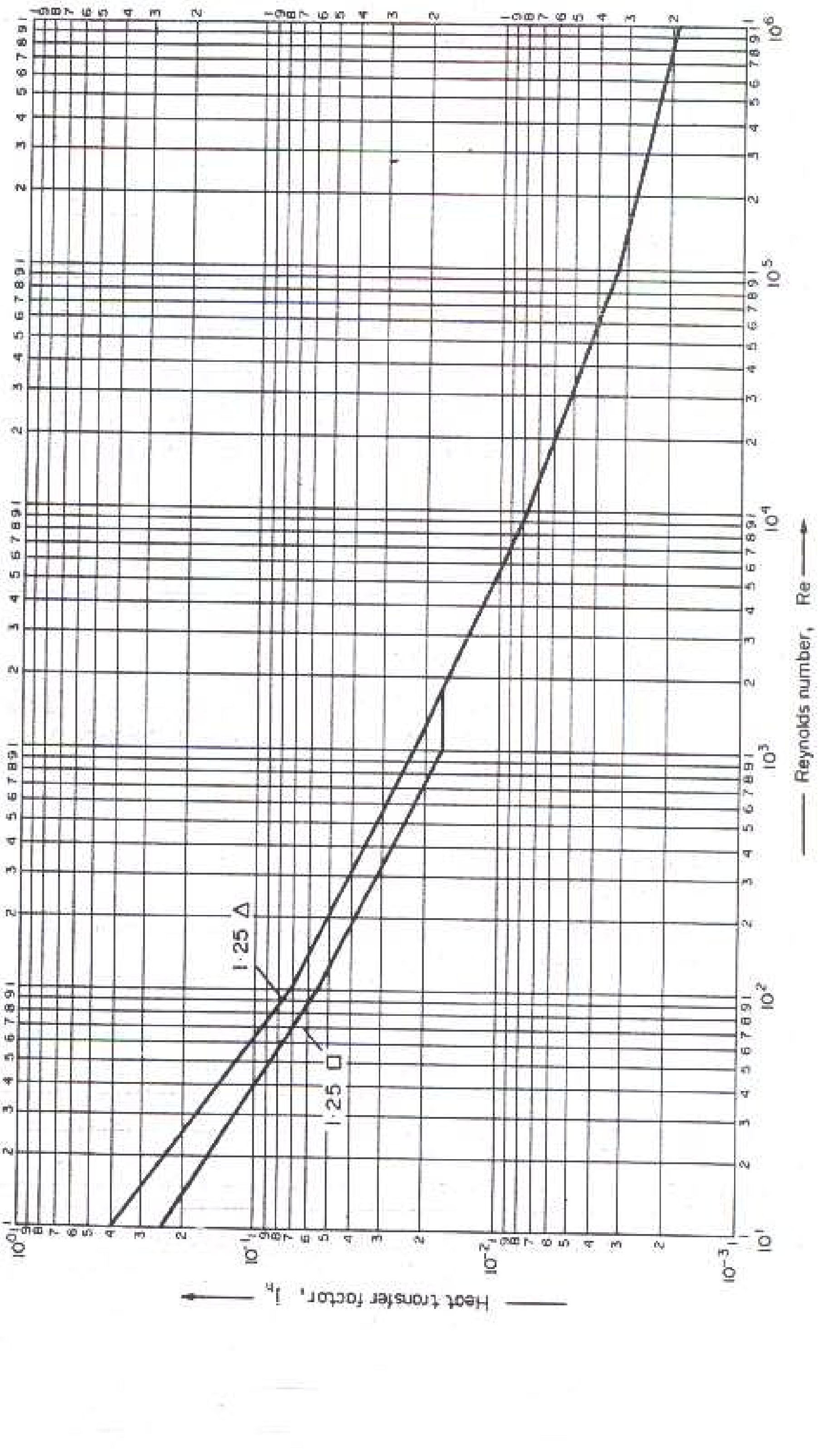


FIG. 12.31. **Heat-transfer factor for cross-flow tube banks**

where G_s = mass flow rate per unit area, based on the total flow and free area at the bundle equator. This is the same as G_s calculated for Kern's method,

d_o = tube outside diameter.

The heat-transfer coefficient is given by:

$$\frac{h_{oc} d_o}{k_f} = j_h Re\, Pr^{1/3} (\mu/\mu_w)^{0.14} \tag{12.28}$$

F_n, tube row correction factor

The mean heat-transfer coefficient will depend on the number of tubes crossed. Figure 12.31 is based on data for ten rows of tubes. For turbulent flow the correction factor F_n is close to 1·0. In laminar flow the heat-transfer coefficient may decrease with increasing rows of tubes crossed, due to the build up of the temperature boundary layer. The factors given below can be used for the various flow regimes; the factors for turbulent flow are based on those given by Bell (1963).

N_{cv} is number of constrictions crossed = number of tube rows between the baffle tips; see Fig. 12.39, and Section 12.9.5.

1. $Re > 2000$, turbulent;
   take F_n from Fig. 12.32.
2. $Re > 100$ to 2000, transition region,
   take $F_n = 1{\cdot}0$;
3. $Re < 100$, laminar region,
   $F_n \propto (N_c')^{-0.18}$, (12.29)

where N_c' is the number of rows crossed in series from end to end of the shell, and depends on the number of baffles. The correction factor in the laminar region is not well established, and Bell's paper, or the summary given by Mueller (1973), should be consulted if the design falls in this region.

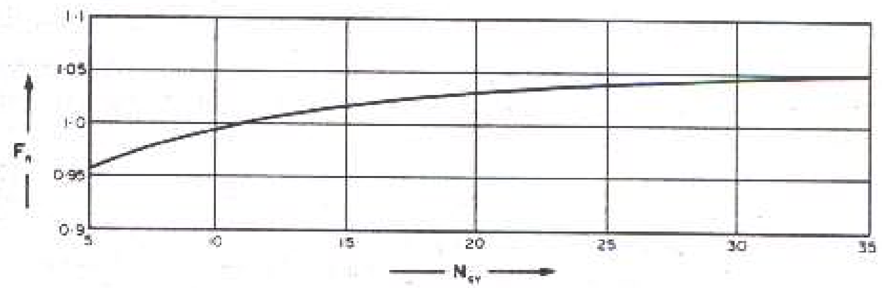


FIG. 12.32. Tube row correction factor, F_n

F_w, window correction factor

This factor corrects for the effect of flow through the baffle window, and is a function of the heat-transfer area in the window zones and the total heat-transfer area. The correction factor is shown in Fig. 12.33 plotted versus R_w, the ratio of the number of tubes in the window zones to the total number in the bundle, determined from the tube layout diagram.

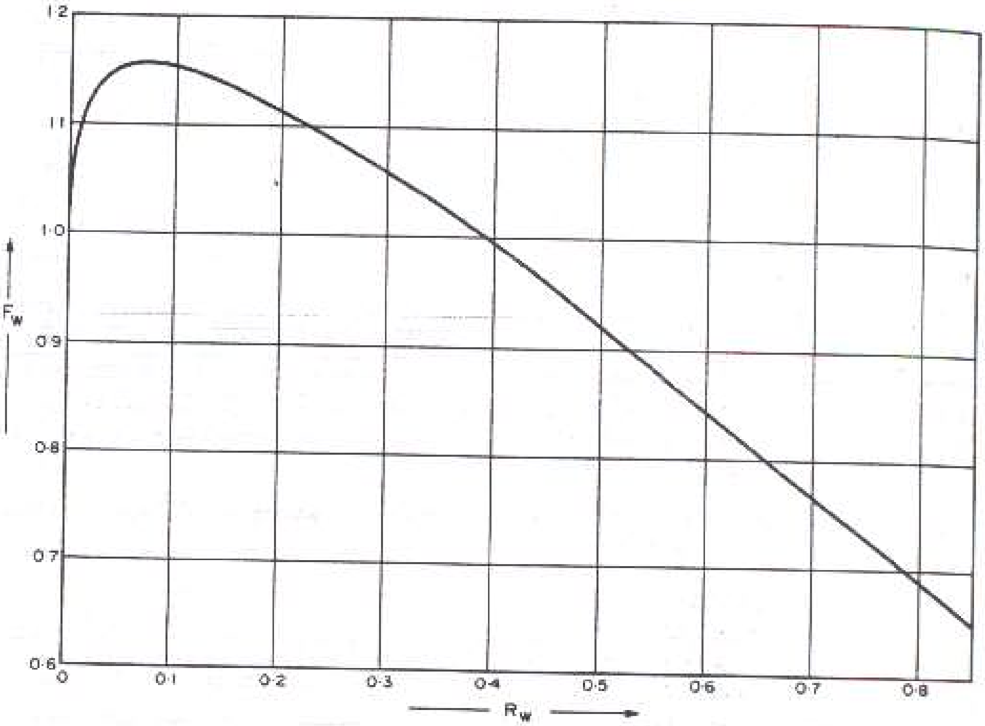


FIG. 12.33. Window correction factor

For preliminary calculations R_w can be estimated from the bundle and window cross-sectional areas, see Section 12.9.5.

F_b, bypass correction factor

This factor corrects for the main bypass stream, the flow between the tube bundle and the shell wall, and is a function of the shell to bundle clearance, and whether sealing strips are used:

$$F_b = \exp\left[-\alpha \frac{A_b}{A_s}(1-(2N_s/N_{cv})^{1/3})\right] \tag{12.30}$$

where α = 1·5 for laminar flow, $Re < 100$,

α = 1·35 for transitional and turbulent flow $Re > 100$,

A_b = clearance area between the bundle and the shell, see Fig. 12.39 and Section 12.9.5,

A_s = maximum area for cross-flow, equation 12.21,

N_s = number of sealing strips encountered by the bypass stream in the cross-flow zone,

N_{cv} = the number of constrictions, tube rows, encountered in the cross-flow section.

Equation 12.30 applies for $N_s \leqslant N_{cv}/2$.

Where no sealing strips are used, F_b can be obtained from Fig. 12.34.

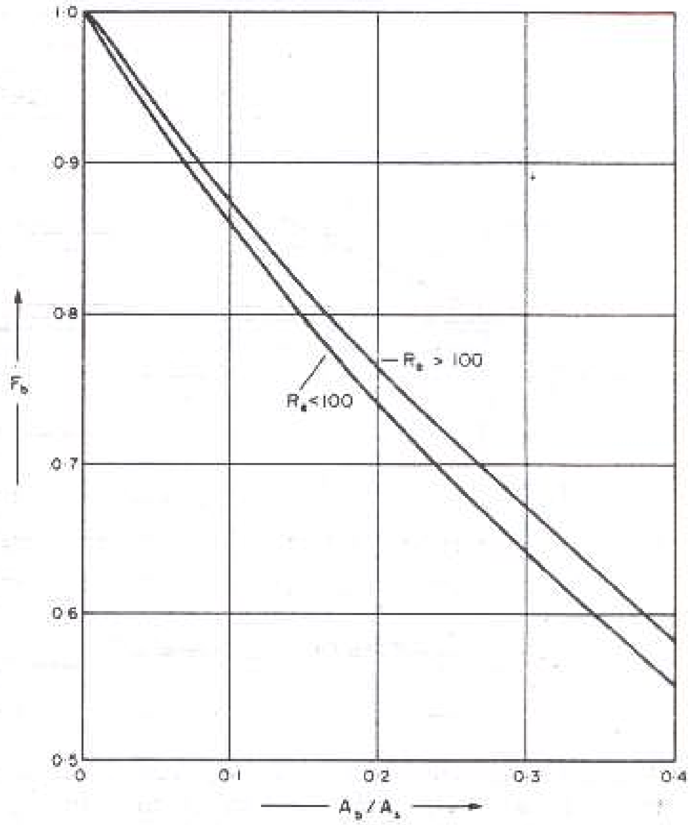


FIG. 12.34. Bypass correction factor

F_L, Leakage correction factor

This factor corrects for the leakage through the tube-to-baffle clearance and the baffle-to-shell clearance.

$$F_L = 1 - \beta_L[(A_{tb} + 2A_{sb})/A_L] \qquad (12.31)$$

where β_L = a factor obtained from Fig. 12.35,
A_{tb} = the tube to baffle clearance area, per baffle, see Fig. 12.39 and Section 12.9.5,
A_{sb} = shell-to-baffle clearance area, per baffle, see Fig. 12.39 and Section 12.9.5,
A_L = total leakage area = $(A_{tb} + A_{sb})$.

Typical values for the clearances are given in the standards, and are discussed in Section 12.5.6. The clearances and tolerances required in practical exchangers are discussed by Rubin (1968).

Pressure drop

The pressure drops in the cross-flow and window zones are determined separately, and summed to give the total shell-side pressure drop.

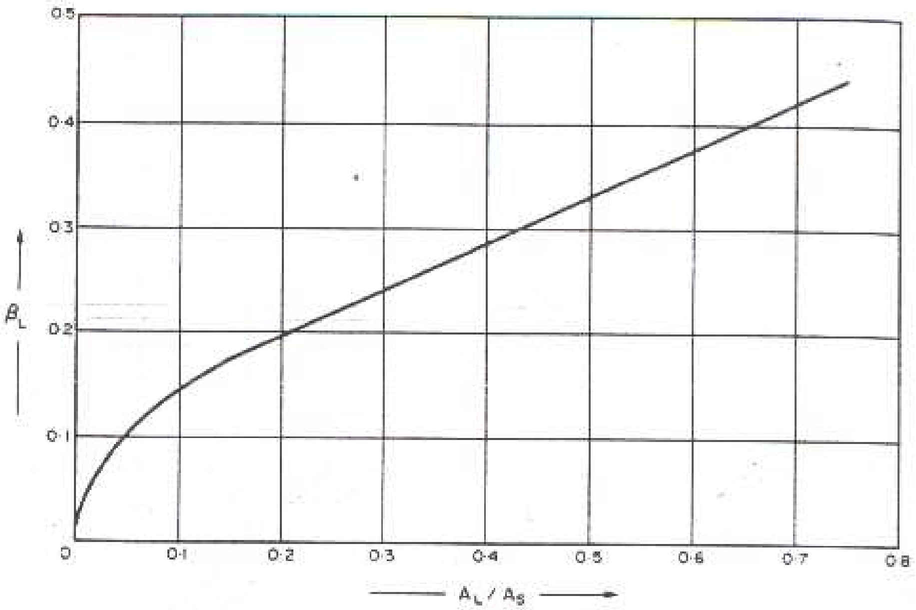


FIG. 12.35. Coefficient for F_L, heat transfer

Cross-flow zones

The pressure drop in the cross-flow zones between the baffle tips is calculated from correlations for ideal tube banks, and corrected for leakage and bypassing.

$$\Delta P_c = \Delta P_i F'_b F'_L \tag{12.32}$$

where ΔP_c = the pressure drop in a cross-flow zone between the baffle tips, corrected for by-passing and leakage,

ΔP_i = the pressure drop calculated for an equivalent ideal tube bank,

F'_b = by-pass correction factor,

F'_L = leakage correction factor.

ΔP_i ideal tube bank pressure drop

The number of tube rows has little effect on the friction factor and is ignored.

Any suitable correlation for the cross-flow friction factor can be used; for that given in Fig. 12.36, the pressure drop across the ideal tube bank is given by:

$$\Delta P_i = 8 j_f N_{cv} \frac{\rho u_s^2}{2} (\mu/\mu_w)^{-0.14} \tag{12.33}$$

where N_{cv} = number of tube rows crossed (in the cross-flow region),

u_s = shell side velocity, based on the clearance area at the bundle equator, equation 12.21,

j_f = friction factor obtained from Fig. 12.36, at the appropriate Reynolds number, $Re = \frac{\rho u_s d_o}{\mu}$.

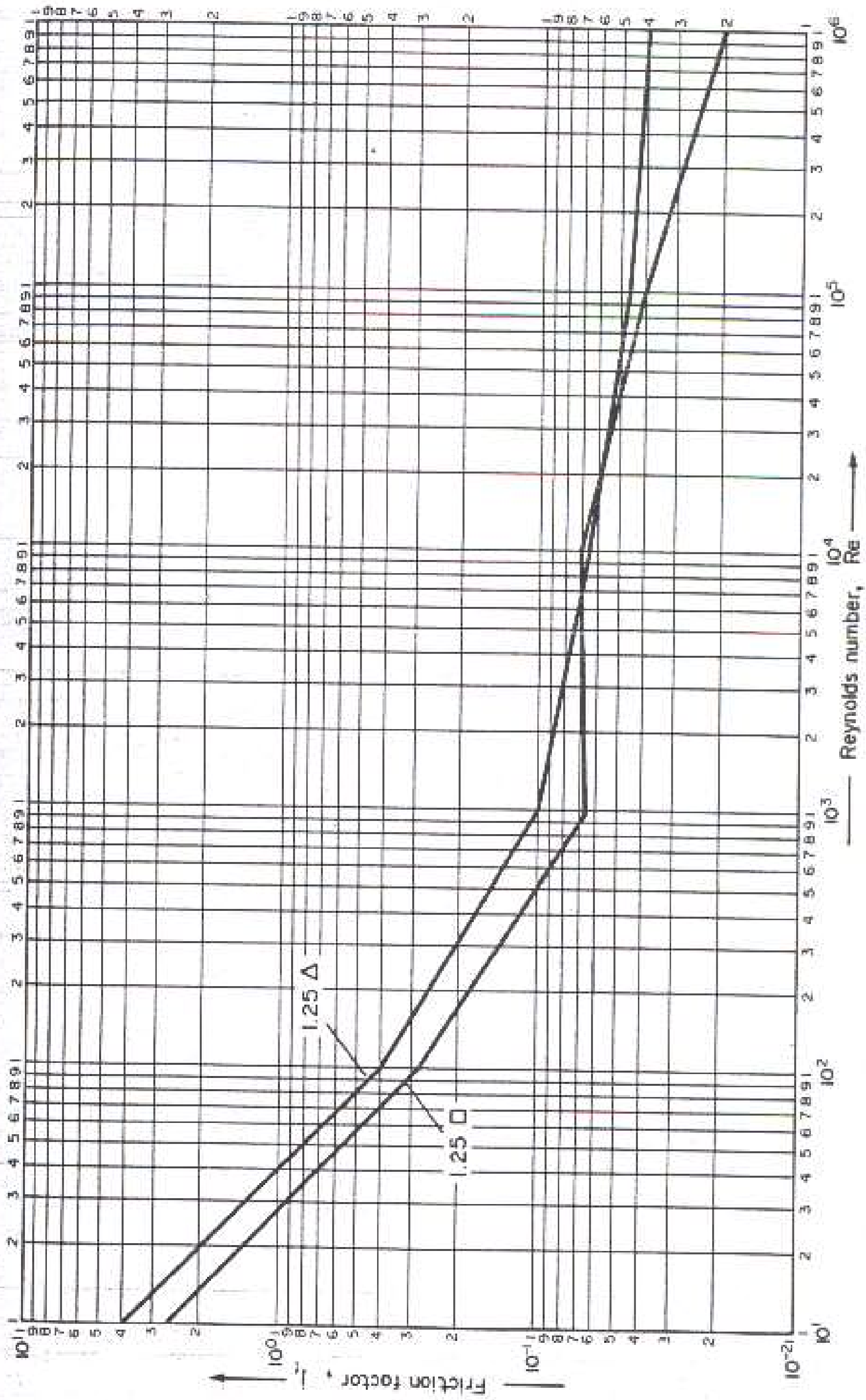


FIG. 12.36. Friction factor for cross-flow tube banks

F'_b, bypass correction factor for pressure drop

Bypassing will affect the pressure drop only in the cross-flow zones. The correction factor is calculated from the equation used to calculate the bypass correction factor for heat transfer, equation 12.30, but with the following values for the constant α.

Laminar region, $Re < 100$, $\alpha = 5.0$
Transition and turbulent region, $Re > 100$, $\alpha = 4.0$

The correction factor for exchangers without sealing strips is shown in Fig. 12.37.

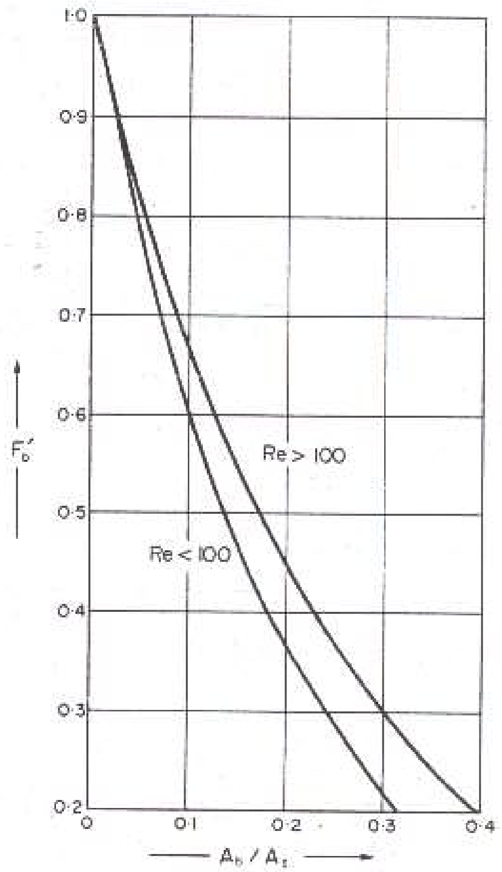


FIG. 12.37. Bypass factor for pressure drop F_b'

F'_L, leakage factor for pressure drop

Leakages will affect the pressure drop in both the cross-flow and window zones. The factor is calculated using the equation for the heat-transfer leakage-correction factor, equation 12.31, with the values for the coefficient β'_L taken from Fig. 12.38.

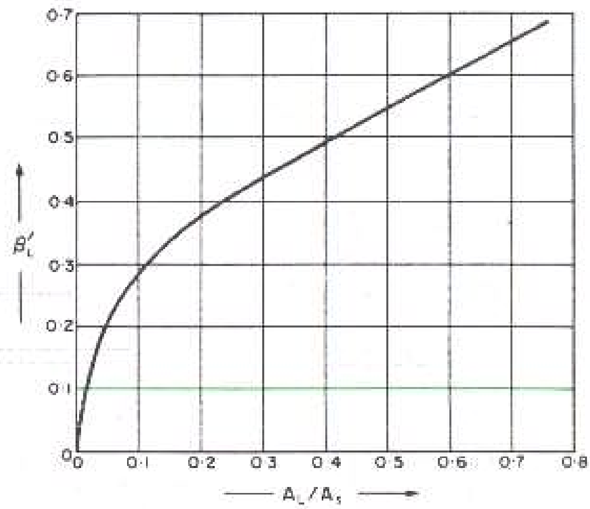


FIG. 12.38. Coefficient for F'_L, pressure drop

Window-zone pressure drop

Any suitable method can be used to determine the pressure drop in the window area; see Butterworth (1977). Bell used a method proposed by Colburn. Corrected for leakage, the window drop for turbulent flow is given by:

$$\Delta P_w = F'_L(2 + 0{\cdot}6N_{wv})\frac{\rho u_z^2}{2} \tag{12.34}$$

where u_z = the geometric mean velocity,

$u_z = \sqrt{u_w u_s}$,

u_w = the velocity in the window zone, based on the window area less the area occupied by the tubes A_w, see Section 12.9.5,

$$u_w = \frac{W_s}{A_w \rho} \tag{12.35}$$

W_s = shell-side fluid mass flow, kg/s,

N_{wv} = number of restrictions for cross-flow in window zone, approximately equal to the number of tube rows.

End zone pressure drop

There will be no leakage paths in an end zone (the zone between tube sheet and baffle). Also, there will only be one baffle window in these zones; so the total number of restrictions in the cross-flow zone will be $N_{cv} + N_{wv}$. The end zone pressure drop ΔP_e will therefore be given by:

$$\Delta P_e = \Delta P_i[(N_{wv} + N_{cv})/N_{cv}]F'_b \tag{12.36}$$

Total shell-side pressure drop

Summing the pressure drops over all the zones in series from inlet to outlet gives:

$$\Delta P_s = 2 \text{ end zones} + (N_b - 1) \text{ cross-flow zones} + N_b \text{ window zones}$$

$$\Delta P_s = 2\Delta P_e + \Delta P_c(N_b - 1) + N_b \Delta P_w \quad (12.37)$$

where N_b is the number of baffles $= \left[\frac{L}{l_B} - 1\right]$.

An estimate of the pressure loss incurred in the shell inlet and outlet nozzles must be added to that calculated by equation 12.37; see Section 12.9.3.

End zone lengths

The spacing in the end zones will often be increased to provide more flow area at the inlet and outlet nozzles. The velocity in these zones will then be lower and the heat transfer and pressure drop will be reduced slightly. The effect on pressure drop will be more marked than on heat transfer, and can be estimated by using the actual spacing in the end zone when calculating the cross-flow velocity in those zones.

12.9.5. Shell and bundle geometry

The bypass and leakage areas, window area, and the number of tubes and tube rows in the window and cross-flow zones can be determined precisely from the tube layout diagram. For preliminary calculations they can be estimated with sufficient accuracy by considering the tube bundle and shell geometry.

With reference to Figs. 12.39 and 12.40:

H_c = baffle cut height = $D_s \times B_c$, where B_c is the baffle cut as a fraction,

H_b = height from the baffle chord to the top of the tube bundle,

B_b = "bundle cut" = H_b/D_b,

θ_b = angle subtended by the baffle chord, rads,

D_b = bundle diameter.

Then:

$$H_b = D_b/2 - D_s(0.5 - B_c) \quad (12.38)$$

$$N_{cv} = (D_b - 2H_b)/p_t' \quad (12.39)$$

$$N_{wv} = H_b/p_t' \quad (12.40)$$

where p_t' is the vertical tube pitch

$p_t' = p_t$ for square pitch,

$p_t' = 0.87 p_t$ for equilateral triangular pitch.

The number of tubes in a window zone N_w is given by:

$$N_w = N_t \times R_a' \quad (12.41)$$

where R_a' is the ratio of the bundle cross-sectional area in the window zone to the total bundle cross-sectional area, R_a' can be obtained from Fig. 12.41, for the appropriate "bundle cut", B_b.

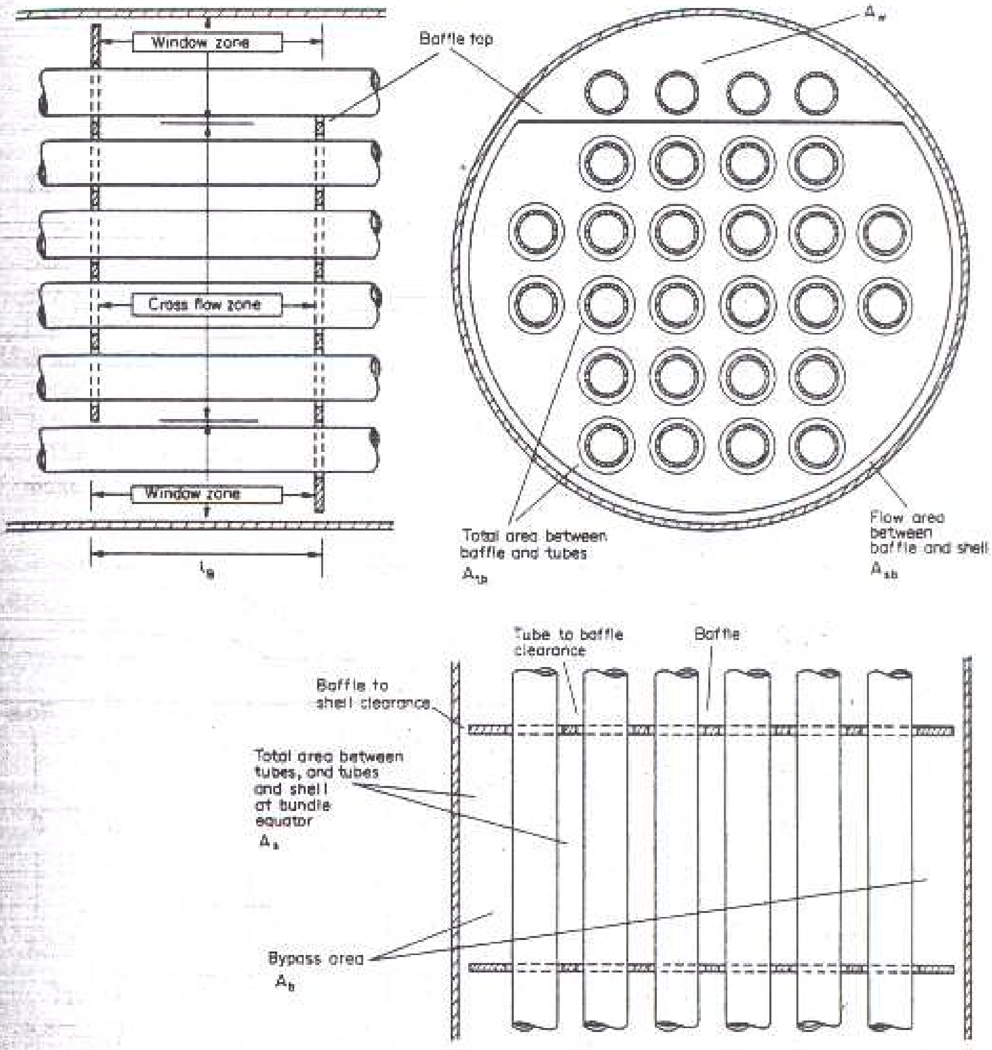


FIG. 12.39. Clearance and flow areas in the shell-side of a shell and tube exchanger

The number of tubes in a cross-flow zone N_c is given by

$$N_c = N_t - 2N_w \tag{12.42}$$

and

$$R_w = 2N_w / N_t \tag{12.43}$$

$$A_w = \left(\frac{\pi D_s^2}{4} \times R_a\right) - \left(N_w \frac{\pi d_o^2}{4}\right) \tag{12.44}$$

R_a is obtained from Fig. 12.41, for the appropriate baffle cut B_c

$$A_{tb} = \frac{c_t \pi d_o}{2}(N_t - N_w) \tag{12.45}$$

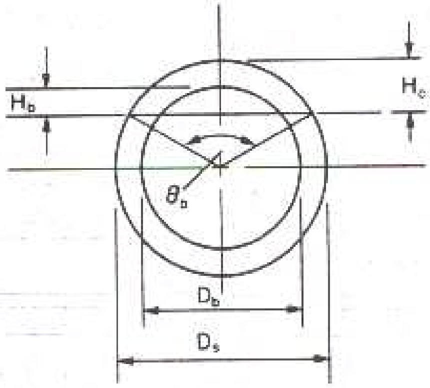


FIG. 12.40. Baffle and tube geometry

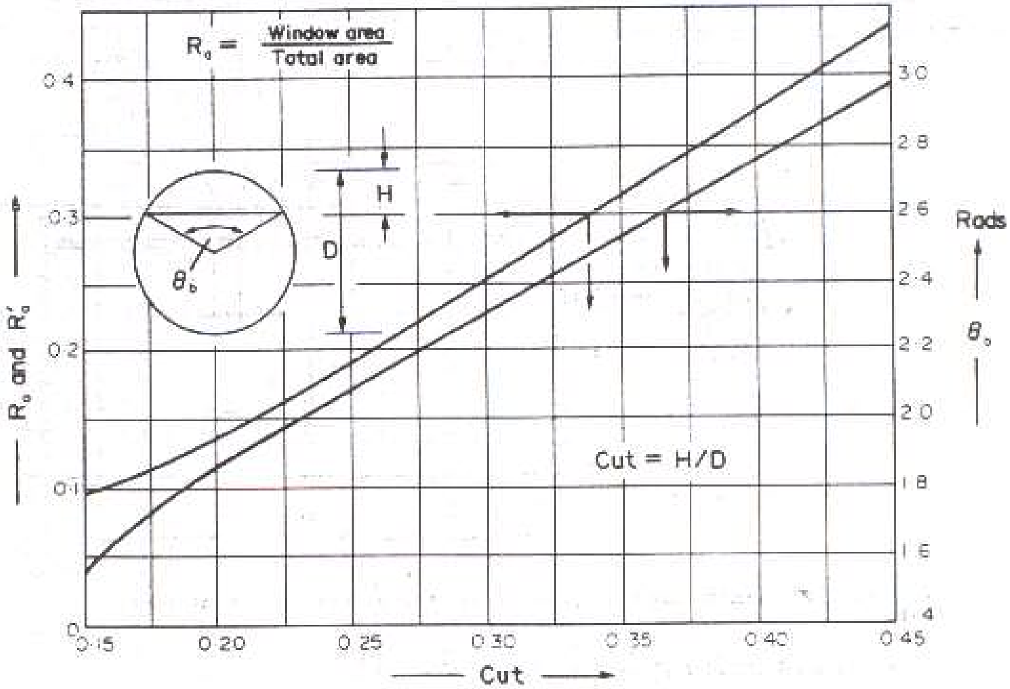


FIG. 12.41. Baffle geometrical factors

where c_t is the diametrical tube-to-baffle clearance; the difference between the hole and tube diameter, typically 0·8 mm.

$$A_{sb} = \frac{c_s D_s}{2}(2\pi - \theta_b) \qquad (12.46)$$

where c_s is the baffle-to-shell clearance, see Table 12.5.

θ_b can be obtained from Fig. 12.41, for the appropriate baffle cut, B_c

$$A_b = l_B(D_s - D_b) \tag{12.47}$$

where l_B is the baffle spacing.

12.9.6. Effect of fouling on pressure drop

Bell's method gives an estimate of the shell-side pressure drop for the exchanger in the clean condition. In service, the clearances will tend to plug up, particularly the small clearance between the tubes and baffle, and this will increase the pressure drop. Devore (1961) has estimated the affect of fouling on pressure drop by calculating the pressure drop in an exchange in the clean condition and with the clearance reduced by fouling, using Tinker's method. He presented his results as ratios of the fouled to clean pressure drop for various fouling factors and baffle spacings.

The ratios given in Table 12.7, which are adapted from Devore's figures, can be used to make a rough estimate of the effect of fouling on pressure drop.

TABLE 12.7. *Ratio of fouled to clean pressure drop*

Fouling coefficient	Shell diameter/baffle spacing		
(W/m² °C)	1·0	2·0	5·0
Laminar flow			
6000	1·06	1·20	1·28
2000	1·19	1·44	1·55
< 1000	1·32	1·99	2·38
Turbulent flow			
6000	1·12	1·38	1·55
2000	1·37	2·31	2·96
< 1000	1·64	3·44	4·77

12.9.7. Pressure-drop limitations

Though Bell's method will give a better estimate of the shell-side pressure drop than Kern's, it is not sufficiently accurate for the design of exchangers where the allowable pressure drop is the overriding consideration. For such designs, a divided-flow model based on Tinker's work should be used. If a proprietary computer program is not available, the EDSU method (ESDU 83038) is recommended. Devore's method can also be considered, providing the exchanger layout conforms with those covered in his work.

Example 12.3

Using Bell's method, calculate the shell-side heat transfer coefficient and pressure drop for the exchanger designed in Example 12.1.

Summary of proposed design

Number of tubes	= 918
Shell i.d.	894 mm
Bundle diameter	826 mm
Tube o.d.	20 mm

Pitch 1.25 Δ	25 mm
Tube length	4830 mm
Baffle pitch	356 mm

Physical properties from Example 12.1

Solution

Heat-transfer coefficient

Ideal bank coefficient, h_{oc}

$$A_s = \frac{25-20}{25} \times 894 \times 356 \times 10^{-6} = 0{\cdot}062 \text{ m}^2$$

$$G_s = \frac{100{,}000}{3600} \times \frac{1}{0{\cdot}062} = 448 \text{ kg/s m}^2$$

$$Re = \frac{G_s d_o}{\mu} = \frac{448 \times 20 \times 10^{-3}}{0{\cdot}34 \times 10^{-3}} = 26{,}353$$

From Fig. 12.31 $j_h = 5{\cdot}3 \times 10^{-3}$.
Prandtl number, from Example 12.1 = 5·1.
Neglect viscosity correction factor (μ/μ_w).

(12.28) $$h_{oc} = \frac{0{\cdot}19}{20 \times 10^{-3}} \times 5{\cdot}3 \times 10^{-3} \times 26{,}353 \times 5{\cdot}1^{1/3} = 2272 \text{ W/m}^2\,°\text{C}$$

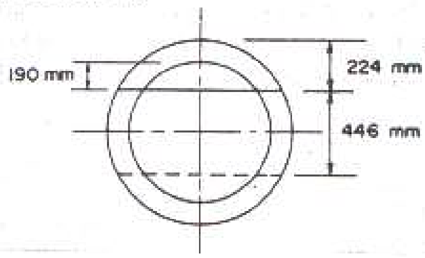
Tube row correction factor, F_n

Tube vertical pitch $p_t' = 0{\cdot}87 \times 25 = 21{\cdot}8$ mm
Baffle cut height $H_c = 0{\cdot}25 \times 894 = 224$ mm
Height between baffle tips = 894 − 2 × 224 = 446 mm

$$N_{cv} = \frac{446}{21{\cdot}8} = 20$$

From Fig. 12.32 $F_n = 1{\cdot}03$.

Window correction factor, F_w



(12.38) $$H_b = \frac{826}{2} - 894(0{\cdot}5 - 0{\cdot}25) = 190 \text{ mm}$$

"Bundle cut" $= \frac{190}{826} = 0{\cdot}23$ (23 per cent)

From Fig. 12.41 at cut of 0·23

$$R'_a = 0{\cdot}18$$

(12.41) Tubes in one window area, $N_w = 918 \times 0{\cdot}18 = 165$

(12.42) Tubes in cross-flow area $N_c = 918 - 2 \times 165 = 588$

(12.43) $$R_w = \frac{2 \times 165}{918} = 0{\cdot}36$$

From Fig. 12.33 $F_w = 1{\cdot}02$.

Bypass correction, F_b

(12.47) $$A_b = (894 - 826)356 \times 10^{-6} = 0{\cdot}024 \text{ m}^2$$

$$A_b/A_s = \frac{0{\cdot}024}{0{\cdot}062} = 0{\cdot}39$$

(12.30) $$F_b = \exp[-1{\cdot}35 \times 0{\cdot}39] = 0{\cdot}59$$

Very low, sealing strips needed; try one strip for each five vertical rows.

$$N_s/N_{cv} = 1/5$$

(12.30) $$F_b = \exp[-1{\cdot}35 \times 0{\cdot}39(1 - (2/5)^{1/3}] = 0{\cdot}87$$

Leakage correction, F_L

Using clearances as specified in the Standards,

tube-to-baffle	$\frac{1}{32}$ in.	= 0·8 mm
baffle-to-shell	$\frac{3}{16}$ in.	= 4·8 mm

(12.45) $$A_{tb} = \frac{0{\cdot}8}{2} \times 20\pi(918 - 165) = 18{\cdot}9 \times 10^3 \text{ mm}^2 = 0{\cdot}019 \text{ m}^2$$

From Fig. 12.41, 25 per cent cut (0·25), $\theta_b = 2{\cdot}1$ rads.

(12.46) $$A_{sb} = \frac{4{\cdot}8}{2} \times 894\ (2\pi - 2{\cdot}1) = 8{\cdot}98 \times 10^3 \text{ mm}^2 = 0{\cdot}009 \text{ m}^2$$

$$A_L = (0{\cdot}019 + 0{\cdot}009) = 0{\cdot}028 \text{ m}^2$$

$$A_L/A_s = \frac{0{\cdot}028}{0{\cdot}062} = 0{\cdot}45$$

From Fig. 12.35 $\beta_L = 0{\cdot}3$.

(12.31) $$F_L = 1 - 0{\cdot}3[(0{\cdot}019 + 2 \times 0{\cdot}009)/0{\cdot}028] = 0{\cdot}60$$

Shell-Side Coefficient

(12.27) $$h_s = 2272 \times 1{\cdot}03 \times 1{\cdot}02 \times 0{\cdot}87 \times 0{\cdot}60 = \underline{\underline{1246 \text{ W/m}^2\,°\text{C}}}$$

Appreciably lower than that predicted by Kern's method.

Pressure drop

Cross-flow zone

From Fig. 12.36 at $Re = 26,353$, for 1·25 Δ pitch, $j_f = 5{\cdot}6 \times 10^{-2}$

$$u_s = \frac{G_s}{\rho} = \frac{448}{750} = 0{\cdot}60 \text{ m/s}$$

Neglecting viscosity term (μ/μ_w).

(12.33) $$\Delta P_i = 8 \times 5{\cdot}6 \times 10^{-2} \times 20 \times \frac{750 \times 0{\cdot}6^2}{2} = 1209{\cdot}6 \text{ N/m}^2$$

(12.30) $(\alpha = 4{\cdot}0)$

$$F'_b = \exp[-4{\cdot}0 \times 0{\cdot}39(1-(2/5)^{1/3})] = 0{\cdot}66$$

From Fig. 12.38 $\beta'_L = 0{\cdot}52$.

(12.31) $$F'_L = 1 - 0{\cdot}52[(0{\cdot}019 + 2 \times 0{\cdot}009)/0{\cdot}028] = 0{\cdot}31$$

$$\Delta P_c = 1209{\cdot}6 \times 0{\cdot}66 \times 0{\cdot}31 = 248 \text{ N/m}^2$$

Window zone

From Fig. 12.41, for baffle cut 25 per cent (0·25) $R_a = 0{\cdot}19$.

(12.44) $$A_w = \left(\frac{\pi}{4} \times 894^2 \times 0{\cdot}19\right) - \left(165 \times \frac{\pi}{4} \times 20^2\right) = 67{\cdot}4 \times 10^3 \text{ mm}^2 = 0{\cdot}067 \text{ m}^2$$

$$u_w = \frac{100{,}000}{3600} \times \frac{1}{750} \times \frac{1}{0{\cdot}067} = 0{\cdot}55 \text{ m/s}$$

$$u_z = \sqrt{u_w u_s} = \sqrt{0{\cdot}55 \times 0{\cdot}60} = 0{\cdot}57 \text{ m/s}$$

(12.40) $$N_{wv} = \frac{190}{21{\cdot}8} = 8$$

(12.34) $$\Delta P_w = 0{\cdot}31(2 + 0{\cdot}6 \times 8)\frac{750 \times 0{\cdot}57^2}{2} = 257 \text{ N/m}^2$$

End zone

(12.36) $$\Delta P_e = 1209{\cdot}6[(8+20)/20]0{\cdot}66 = 1118 \text{ N/m}^2$$

Total pressure drop

$$\text{Number of baffles } N_b = \frac{4830}{356} - 1 = 12$$

(12.37) $$\Delta P_s = 2 \times 1118 + 248(12-1) + 12 \times 257 = 8048 \text{ N/m}^2 = \underline{\underline{8{\cdot}05 \text{ kPa}}} \text{ (1·2 psi)}$$

This for the exchanger in the clean condition. Using the factors given in Table 12.7 to estimate the pressure drop in the fouled condition

$$\Delta P_s = 1{\cdot}4 \times 8{\cdot}05 = \underline{\underline{11{\cdot}3 \text{ kPa}}}$$