King Abdulaziz University

Mechanical Engineering Department

MEP 460

Heat Exchanger Design

Design and rating of Shell and tube heat Exchangers

Bell-Delaware method

March 2022

Bell Delaware method for heat exchangers

1-Introduction
2-Main flow streams
3-Ideal shell side heat transfer coefficient and pressure drop
4-Shell side heat transfer coefficient and correction factors
5-Shell side pressure drop and correction factors
6- Example

1-Intoduction

Shell and tube heat exchangers are the most commonly used heat exchangers

Use extensively in power plants, refineries and industrial and commercial sectors

TEMA Standards of shell and tube layouts are available

Very well-known analysis methods:

- Simple Kern method
- Bell-Delaware method

2- Main flow streams for shell and tube heat exchanger

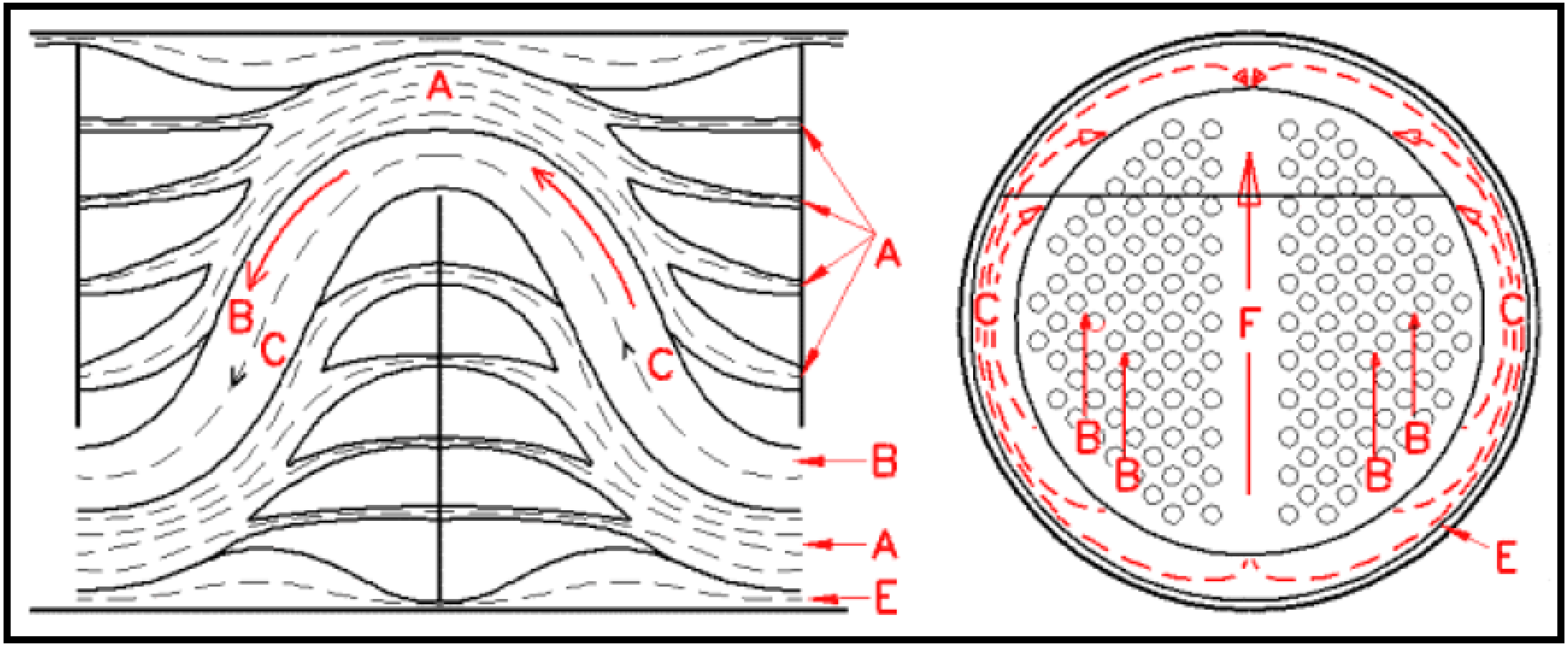


Figure 3.1. Shell-side flow paths in a baffled heat exchanger according to Tinker (1951).

Main flow streams for shell and tube heat exchanger

Stream A is the leakage between the baffle and tubes

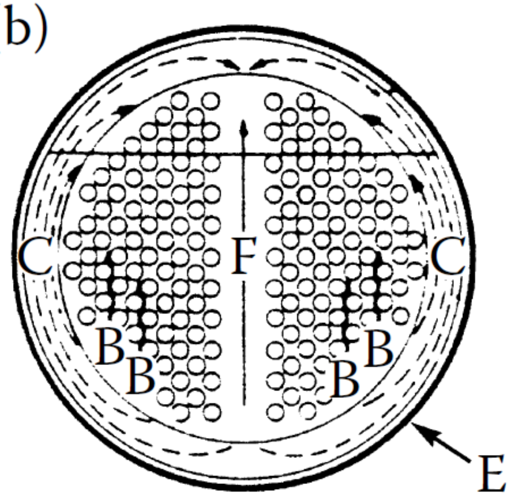
Stream B is the main effective cross flow stream over tube bundle

Stream C is the bundle bypass between the tube bundle and the shell wall

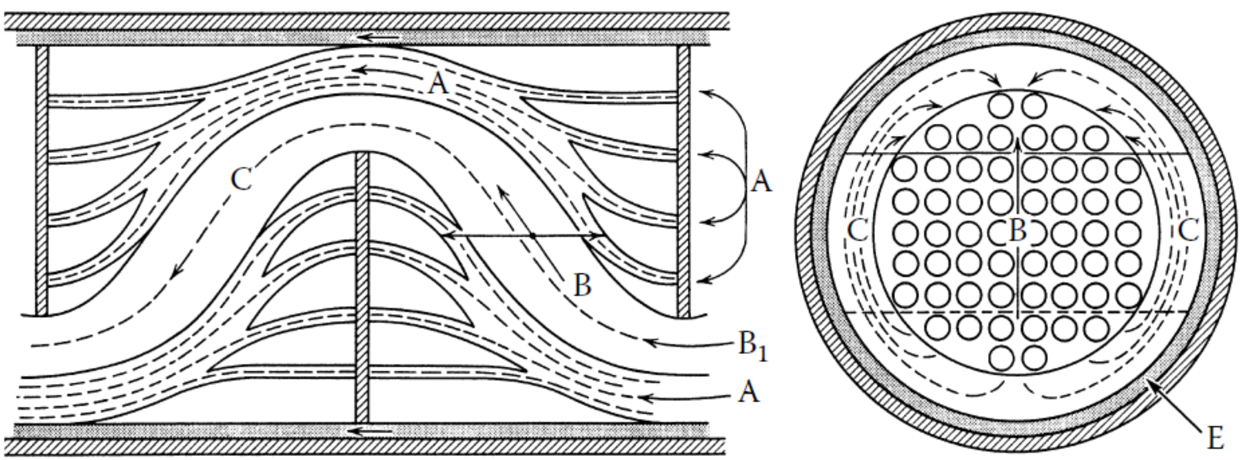
Stream E is the leakage between the baffle edge and the shell wall

Stream F is the bypass stream in flow channel partition due to omissions of tubes in tube pass partition

(b)



(a)



Main flow streams for shell and tube heat exchanger

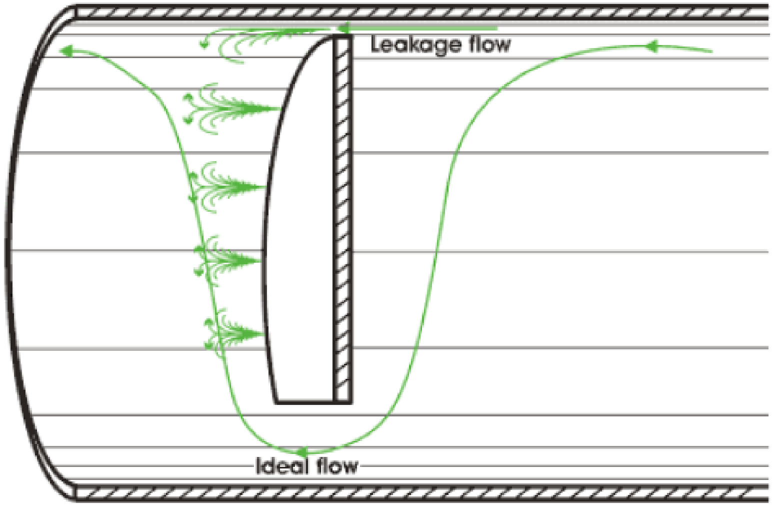


Figure 3.4. Schematic of shell-to-baffle bypass stream E.

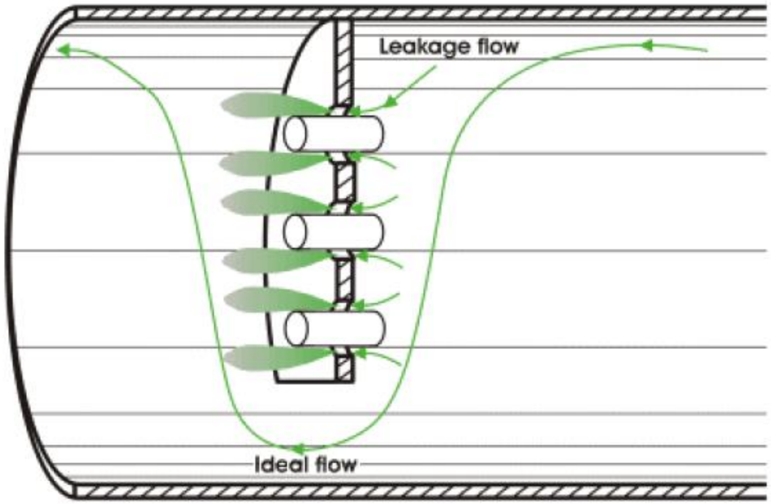


Figure 3.2. Diagram of tube hole leakage stream A.

Stream E

Leakage between the baffle and the shell

Stream A

leakage between tubes and baffle

Using sealing strip to reduce bundle-shell by pass

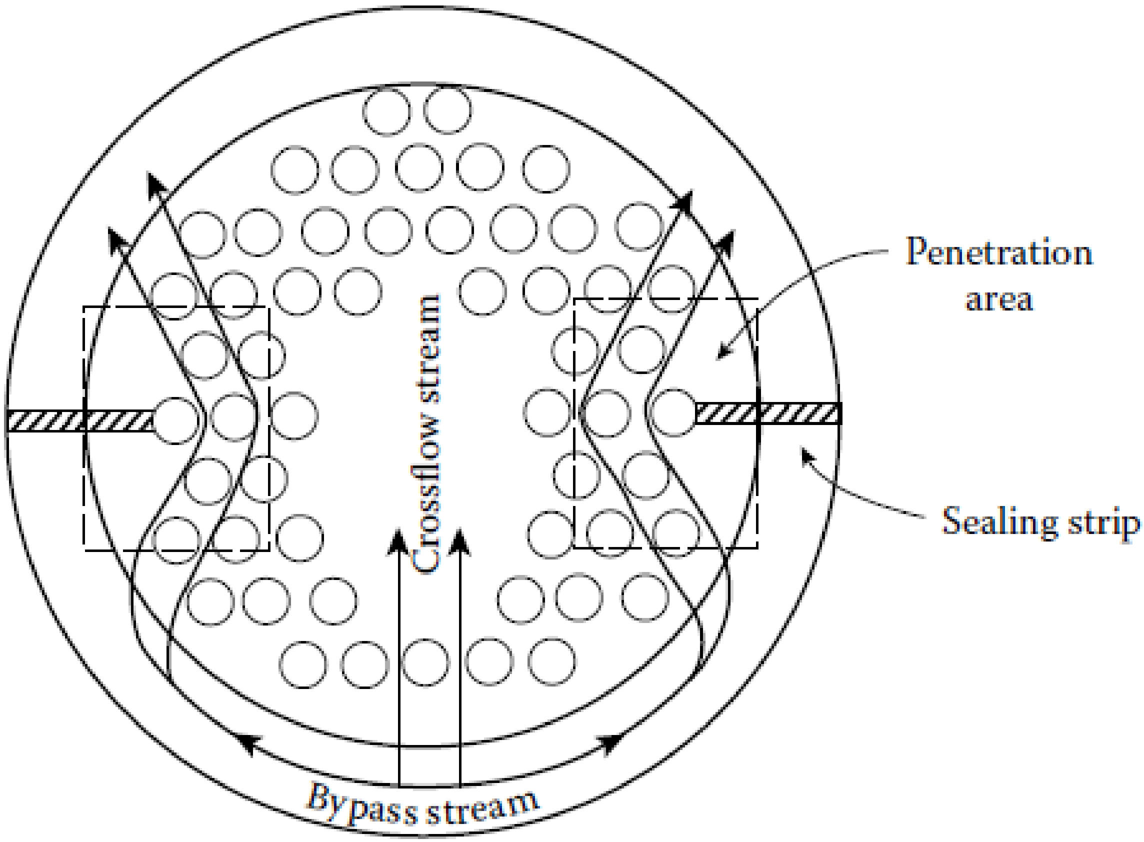
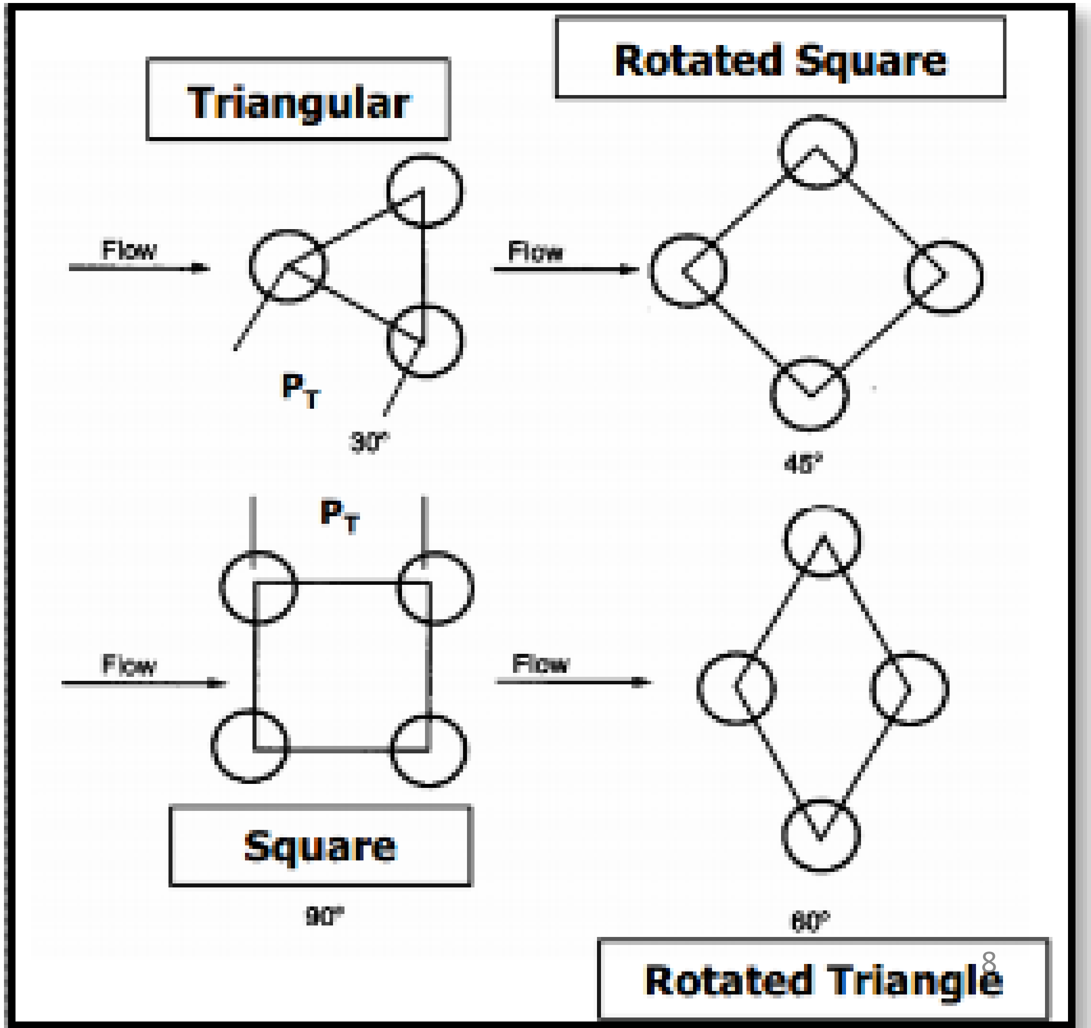


FIGURE 9.14
Radial baffles designed to reduce the amount of bypass flow through the gap between the side of the tube matrix and the shell.[2,8] (Adapted from Spalding, D. B. and Taborek, J., *Heat Exchanger Design Handbook*, Section 3.3, Hemisphere, Washington, D.C., 1983. With permission.)

3-Ideal shell side heat transfer coefficient and correction factors

The three common tube layout used in shell and tube heat exchangers are:

1-Triangular pitch
2-Square pitch
3-Rotated square pitch



3-Ideal shell side heat transfer coefficient and correction factors

Pitch angle $\theta_p = 30°$

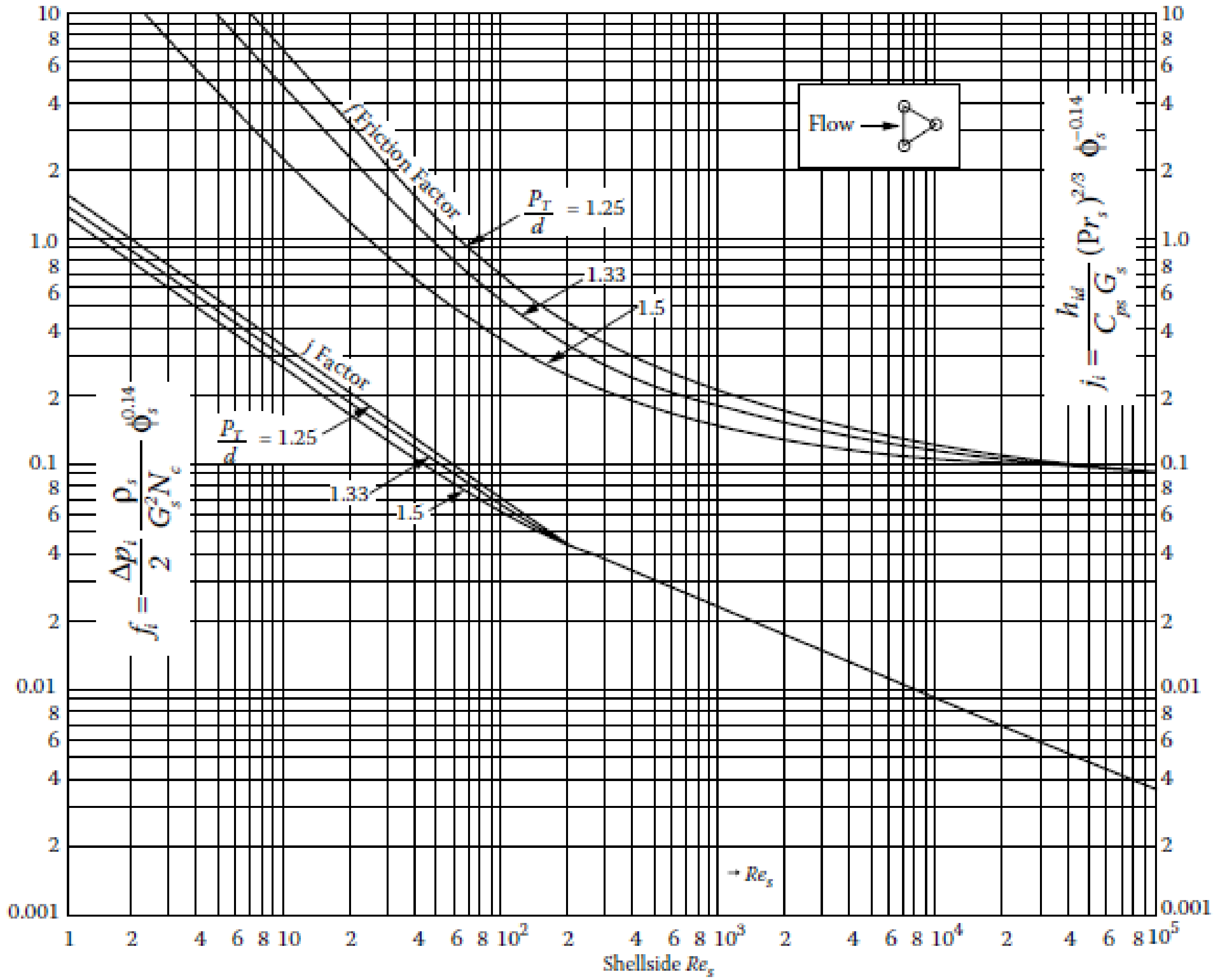


FIGURE 9.15
Ideal tube bank j_i and f_i factors for 30°C staggered layout. (From Spalding, D. B. and Taborek, J., *Heat Exchanger Design Handbook*, Section 3.3, Hemisphere, Washington, D.C., 1983. With permission.)

3-Ideal shell side heat transfer coefficient and correction factors

Pitch angle $\theta_p = 45°$

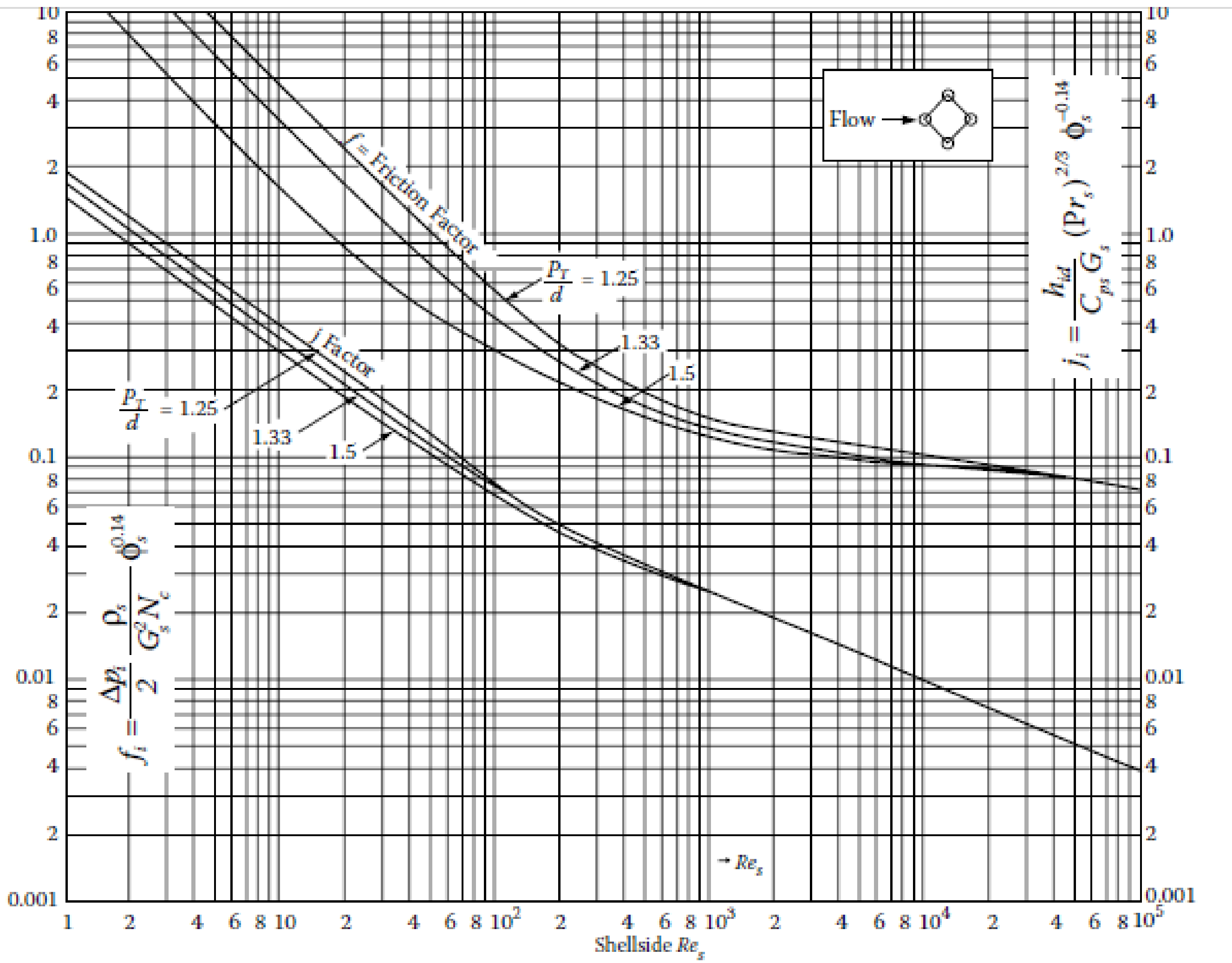


FIGURE 9.16
Ideal tube bank j_i and f_i factors for 45°C staggered layout. (From Spalding, D. B. and Taborek, J., *Heat Exchanger Design Handbook*, Section 3.3, Hemisphere, Washington, D.C., 1983. With permission.)

3-Ideal shell side heat transfer coefficient and correction factors

Pitch angle $\theta_p = 90\,°$

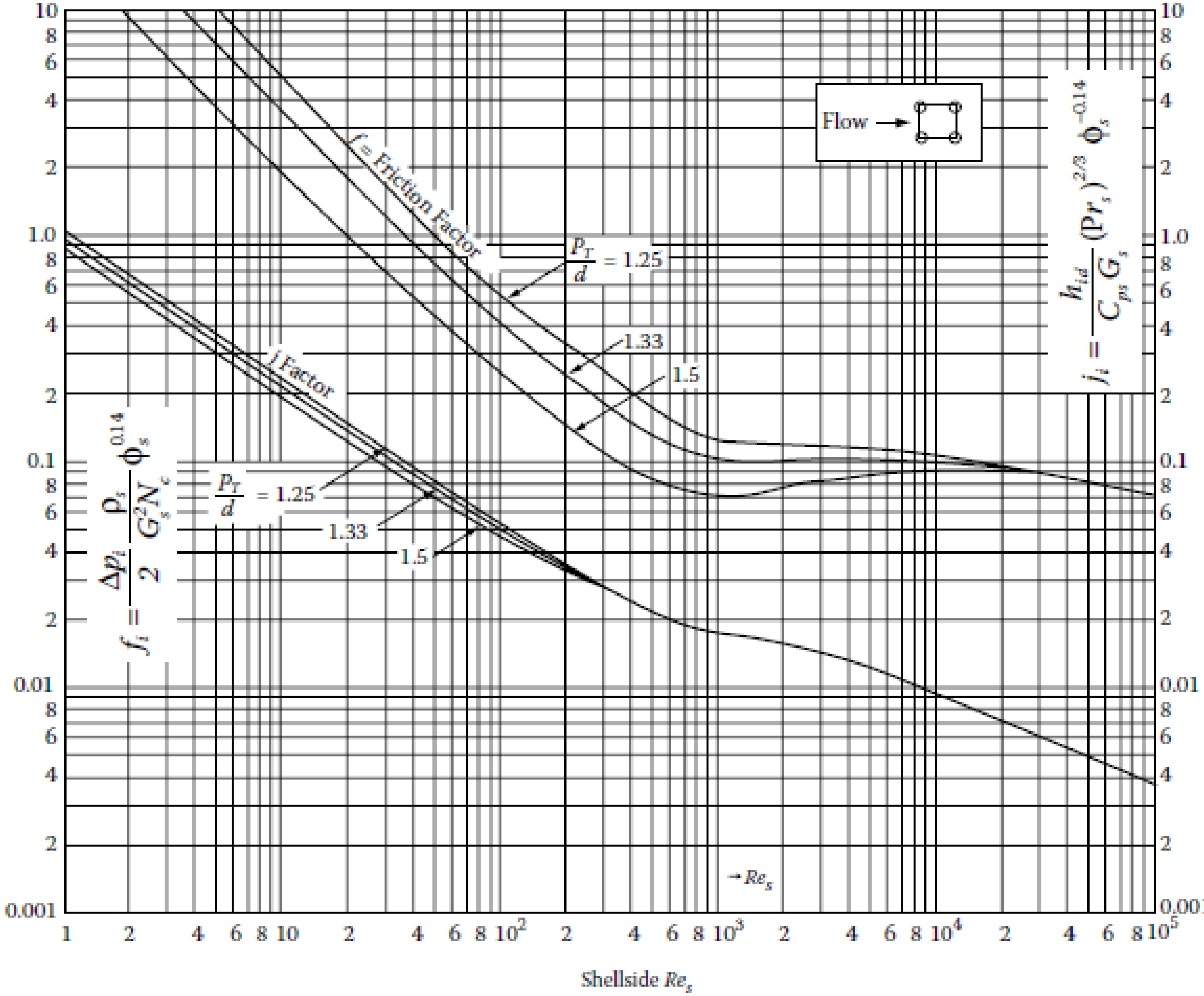


FIGURE 9.17
Ideal tube bank j_i and f_i factors for 90°C inline layout. (From Spalding, D. B. and Taborek, J., *Heat Exchanger Design Handbook*, Section 3.3, Hemisphere, Washington, D.C., 1983. With permission.)

3-Ideal shell side heat transfer coefficient and correction factors

Heat transfer coefficient

$$j_i = a_1 \left(\frac{1.33}{P_T/d_o}\right)^{a} (Re_s)^{a_2}$$

$$a = \frac{a_3}{1 + 0.14\,(Re_s)^{a_4}}$$

Friction coefficient

$$f_i = b_1 \left(\frac{1.33}{P_T/d_o}\right)^{b} (Re_s)^{b_2}$$

$$b = \frac{b_3}{1 + 0.14\,(Re_s)^{b_4}}$$

$$h_{id} = j_i C_{ps} \left(\frac{\dot{m}}{A_s}\right) (Pr)^{-2/3} \left(\frac{\mu_s}{\mu_{s,w}}\right)^{0.14}$$

Ideal heat transfer and pressure drop factors

TABLE 9.6

Correlation Coefficients for j_i and f_i Equations 9.25 and 9.26

Layout Angle	Reynolds Number	a_1	a_2	a_3	a_4	b_1	b_2	b_3	b_4
30°	10^5–10^4	0.321	−0.388	1.450	0.519	0.372	−0.123	7.00	0.500
	10^4–10^3	0.321	−0.388			0.486	−0.152		
	10^3–10^2	0.593	−0.477			4.570	−0.476		
	10^2–10	1.360	−0.657			45.100	−0.973		
	<10	1.400	−0.667			48.000	−1.000		
45°	10^5–10^4	0.370	−0.396	1.930	0.500	0.303	−0.126	6.59	0.520
	10^4–10^3	0.370	−0.396			0.333	−0.136		
	10^3–10^2	0.730	−0.500			3.500	−0.476		
	10^2–10	0.498	−0.656			26.200	−0.913		
	<10	1.550	−0.667			32.00	−1.000		
90°	10^5–10^4	0.370	−0.395	1.187	0.370	0.391	−0.148	6.30	0.378
	10^4–10^3	0.107	−0.266			0.0815	+0.022		
	10^3–10^2	0.408	−0.460			6.0900	−0.602		
	10^2–10	0.900	−0.631			32.1000	−0.963		
	<10	0.970	−0.667			35.0000	−1.000		

Shell side heat transfer coefficient

$$h_o = h_{id}\, J_c J_l J_b J_s J_r$$

J_c, J_l, J_b, J_s, J_r

Correction factors

The ideal heat transfer coefficient for pure cross flow is given by

$$h_{id} = j_i C_{ps} \left(\frac{\dot{m}_s}{A_s}\right)(Pr)^{-2/3}\left(\frac{\mu_s}{\mu_{s,w}}\right)^{0.14}$$

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{\dot{m}_s}{S_m}$$

Heat transfer correction factors

$$h_o = h_{id}\, J_c J_l J_b J_s J_r$$

factor	Due to	Typical values
J_c	baffle cut and spacing (heat transfer in the window)	0.53-1.15
J_l	leakage effects **(streams A & E)**	0.7-0.8
J_b	bundle bypass flow **(streams C & F streams)**	0.7-0.9
J_s	variable baffle spacing in the inlet and outlet sections(	0.85-1.0
J_r	adverse temperature gradient build-up **(Re_s<100)**	1.0 for Re_s>100
$J_c J_l J_b J_d J_s$	Product of all factors (for well design HX	0.6

Shell side Reynold's number

$$Re_s = \frac{d_o \dot{m}_s}{\mu_s A_s}$$

Notice that Reynold's number is based on tube outside diameter d_o

A_s is the minimum flow area at the shell centerline

$$A_s = (D_s - N_{TC} d_o)B$$

or

$$A_s = \frac{D_s C B}{P_T}$$

$$N_{TC} = \frac{D_s}{P_T}$$

B is the baffle spacing

N_{TC} number of tubes at the centerline of the shell

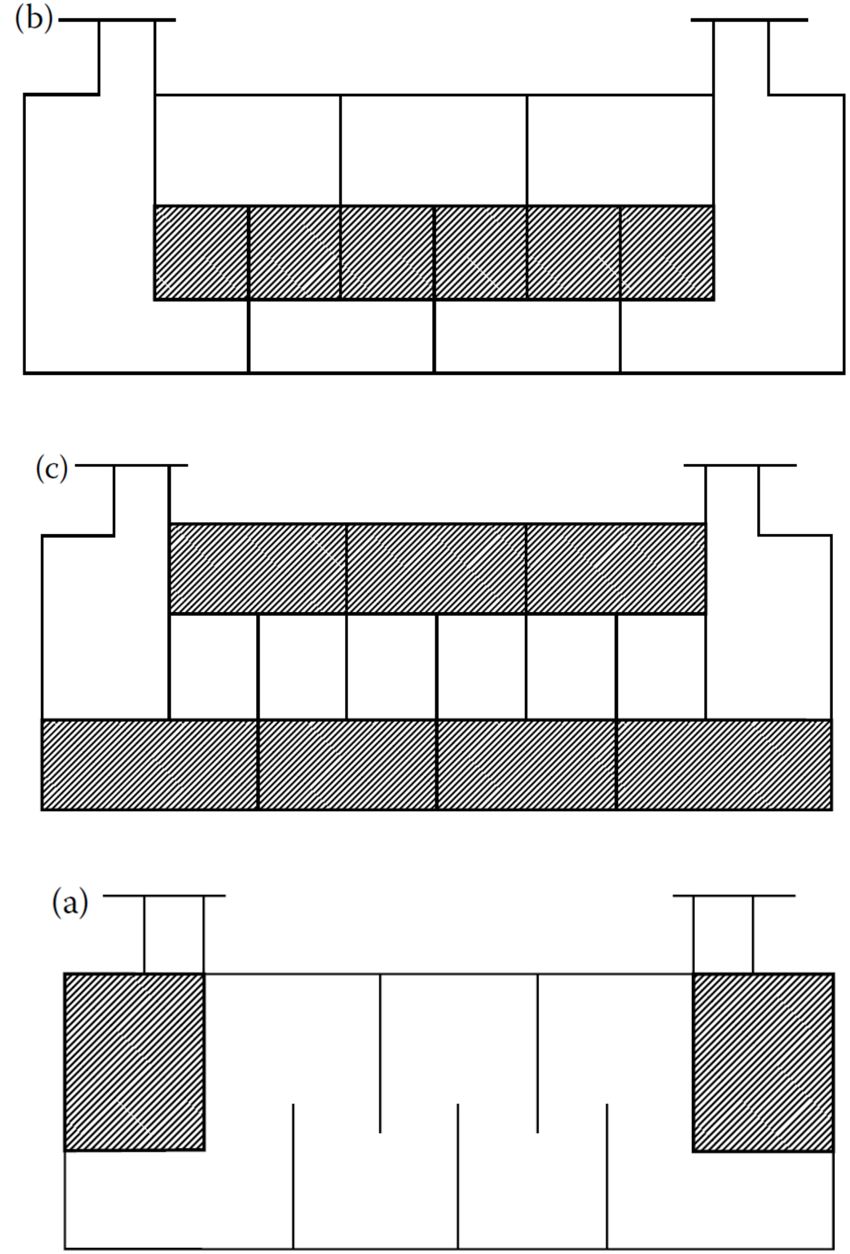
$$Re_s = \frac{d_o \dot{m}_s}{\mu_s A_s}$$

Shell side pressure drop

Internal Crossed Δp_c

Windows Δp_w

Entrance Δp_e



$$\Delta p_s = \Delta p_c + \Delta p_w + \Delta p_e$$

Pressure drop in cross flow between baffle edges

$$\Delta p_c = \Delta p_{bi}(N_b - 1)R_l R_b$$

$$\Delta p_{bi} = 4\, f_i \frac{G_s^2}{2\rho_s} \left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14} N_c$$

Eq. 9.32

R_l leakage correction factor due streams A and C (typical value between 0.4 and 0.5)

R_b by pass correction factor due to stream C and F (typical value between 0.5 and 0.8)

N_b is the number of baffles

N_c is the number of tubes crossed between baffle tips

Pressure drop in the window

Pressure drop in windows $\Delta p_w = \Delta p_{wi} N_b R_l$

$$\Delta p_{wi} = \frac{\dot{m}_s^2 (2 + 0.6\, N_{cw})}{2 \rho_s A_s A_w} \qquad Re_s \geq 100$$

$$\Delta p_{wi} = 26 \frac{\mu_s \dot{m}_s}{\sqrt{A_s A_w \rho}} \left(\frac{N_{cw}}{P_T - d_o} + \frac{B}{D_w^2} \right) + \frac{\dot{m}_s}{A_s A_w \rho_s}$$

Eq. 9.34 $Re_s < 100$

No of tube rows crossed from tip to tip of baffle

$$N_c = \frac{D_s \left(1 - 2 \frac{L_c}{D_s}\right)}{P_p}$$

Eq. 9.35

No. of tube rows in window

$$N_{cw} = \frac{0.8 L_c}{P_p}$$

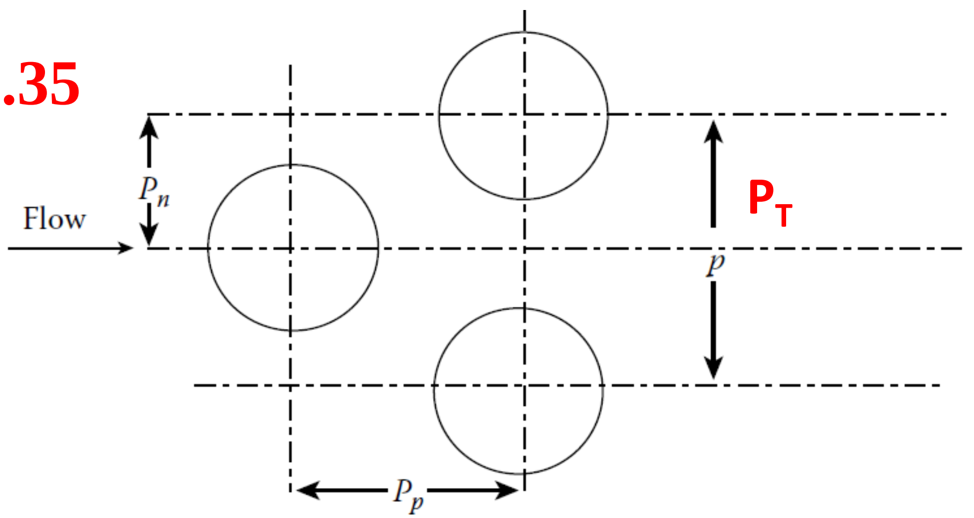


FIGURE 9.19
Tube pitches parallel and normal to flow (equilateral triangular arrangement shown).

Areas in the window of a baffle

Window gross area A_{wg} or S_{wg}

Tube area in the window A_{wt} or S_{wt}

Flow area in window=A_w=A_{wg}-A_{wt}

L_c=L_{bhc}

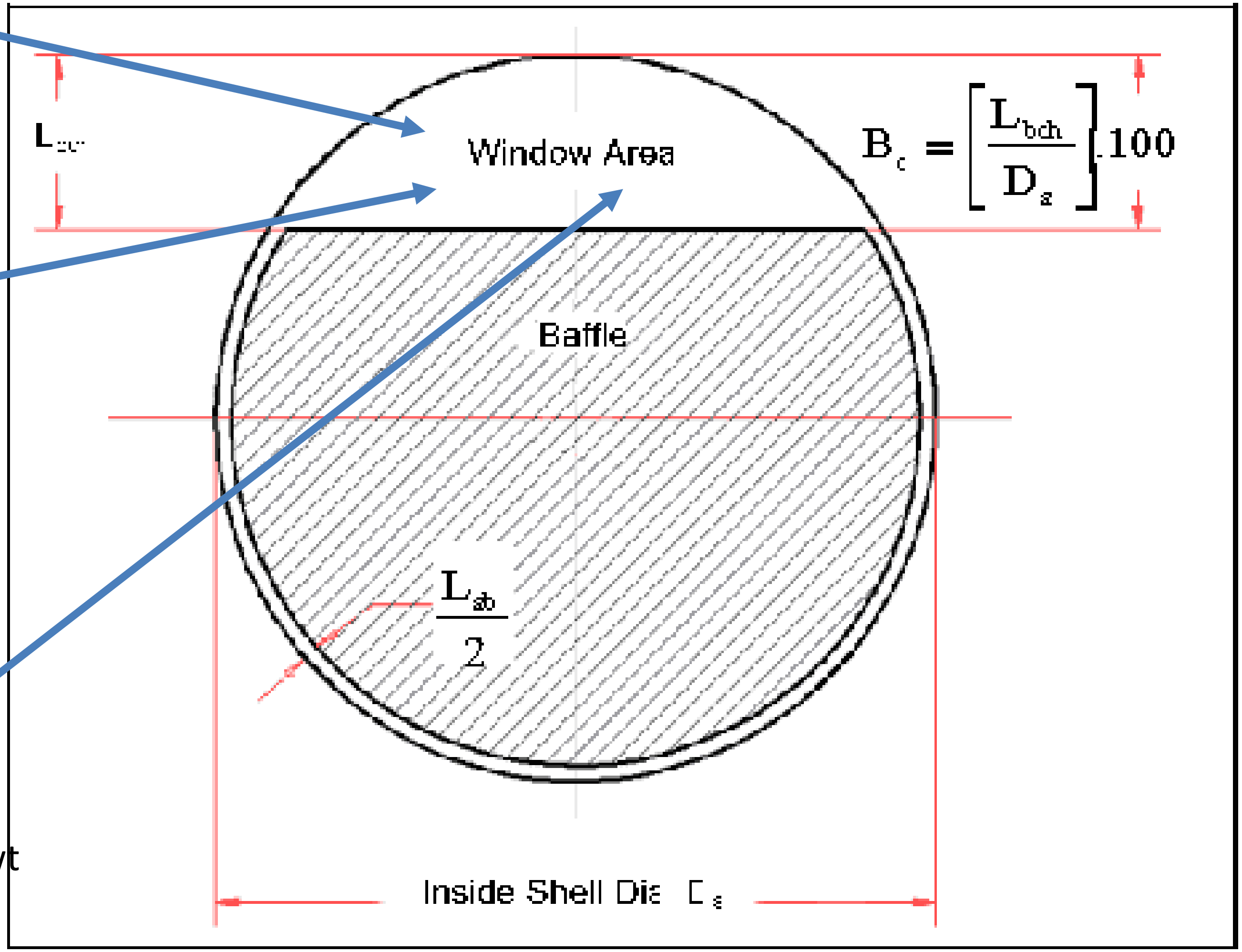


Fig. 2 Baffle Cut and Clearance [6]

Parallel and normal tube pitch definition

TABLE 9.7

Tube Pitches Parallel and Normal to Flow

Tube OD (d_o, in.)	**Tube Pitch (p, in.)**	**Layout**	**p_p (in.)**	**p_n (in.)**
5/8 = 0.625	13/16 = 0.812	→ ◁	0.704	0.406
3/4 = 0.750	15/16 = 0.938	→ ◁	0.814	0.469
3/4 = 0.750	1.000	→ □	1.000	1.000
3/4 = 0.750	1.000	→ ◊	0.707	0.707
3/4 = 0.750	1.000	→ ◁	0.866	0.500
1	1 1/4 = 1.250	→ □	1.250	1.250
1	1 1/4 = 1.250	→ ◊	0.884	0.884
1	1 1/4 = 1.250	→ ◁	1.082	0.625

Source: From Bell, K. J., *Heat Exchangers — Thermal–Hydraulic Fundamentals and Design*, 1981. With permission.

Pressure drop at the entrance and exit

Pressure drop in entrance and exit

$$\Delta p_e = 2\Delta p_{bi} \frac{N_c + N_{cw}}{N_c} R_b R_s$$

N_c is the number of tube rows crossed in the heat exchanger (baffle tip-to-tip)
N_{cw} is the number of tube rows crosses in the window
R_s is the correction factor entrance and exit section having different baffle spacing than internal section due existence of inlet and outlet nozzles

Total Shell side pressure drop

$$\Delta p_s = \Delta p_c + \Delta p_w + \Delta p_e$$

or

$$\Delta p_s = [(N_b - 1)\Delta p_{bi} R_b + N_b \Delta p_{wi}] R_l + 2\Delta p_{bi}\left(1 + \frac{N_{cw}}{N_c}\right) R_b R_s$$

Number of baffles

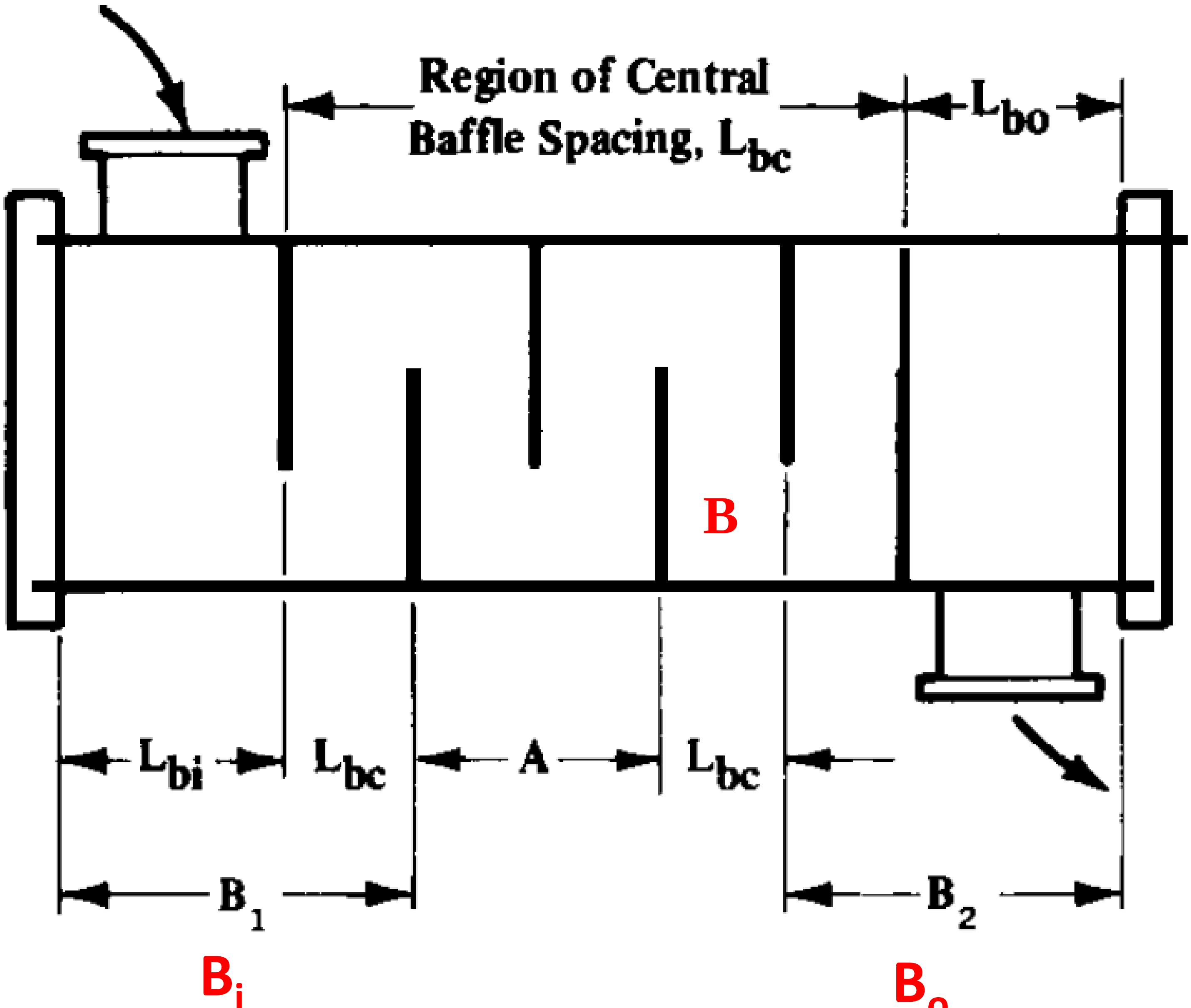
Number of baffles can be calculated using

$$N_b = \frac{L - B_i - B_o}{B} + 1$$

B_i and B_o are the baffle spacing at inlet and exit

If B_i=B_o=B then

$$N_b = \frac{L}{B} - 1$$


Region of Central
Baffle Spacing, Lbc
Lbo
B
Lbi
Lbc
A
Lbc
B1
B2
Bi
Bo

Example 9.4

Assume that a preliminary analysis of a heat exchanger is performed using the Kern method as in Example 9.3 (or assume that such a heat exchanger is available), and the design parameters are as given below. D_s = 0.58 m, number of tubes N_t = 374, the length of the heat exchanger L = 5.0, the tube diameter is 3/4 in (19 mm OD with 16 mm ID), and tubes are laid out on a 1 in. square pitch. The baffle spacing B = 0.5 m, and the baffle cut is 25% of the shell inside diameter D_s. Inlet and outlet baffle spacing and central baffle spacing are equal. Flow specifications on the shell side and the tubes are specified in Example 9.3. The shell-side mass flow rate is 50 kg/s. Allowable pressure drop on the shell side is 15 kPa. The flow area through the window is calculated to be A_w = 0.076 m2. Calculate the shell-side pressure drop using the Bell–Delaware method and state if this heat exchanger is suitable.

Example 9.4 continue

Solution

Now we have to rate the heat exchanger using the Bell–Delaware method and assuming $D_s = 0.58$ m, $N_t = 374$, $L = 5$ m, $d_o = 19$ mm, $d_i = 16$ m, $B = 0.50$ m, a 25% baffle cut, $P_T = 0.0254$ m, $L_c = 0.25\ D_s$, the tubes are laid out in square pitch, and the area of the baffle windows, $A_w = 0.076\ \text{m}^2$.

The estimated crossflow area at the shell diameter from Example 9.3 is

$$A_s = 0.073\ \text{m}^2$$

$$A_s = (D_s - N_{TC} d_o)B$$

$$N_{TC} = \frac{D_s}{P_T}$$

The number of the rows crossed in one crossflow section, N_c, can be calculated from Equation 9.35:

$$\frac{L_c}{D_s} = \frac{0.25 D_s}{D_s} = 0.25$$

From Table 9.7,

$$P_p = 1\ \text{in.} = 0.0254\ \text{m}$$

$$N_c = \frac{D_s\left[1 - 2\dfrac{L_c}{D_s}\right]}{P_p}$$

$$N_c = 0.58 \times [1 - 2 \times (0.25)]/0.0254 \cong 12$$

TABLE 9.7

Tube Pitches Parallel and Normal to Flow

Tube OD (d_o, in.)	Tube Pitch (p, in.)	Layout	p_p (in.)	p_n (in.)
5/8 = 0.625	13/16 = 0.812	→ ◁	0.704	0.406
3/4 = 0.750	15/16 = 0.938	→ ◁	0.814	0.469
3/4 = 0.750	1.000	→ □	1.000	1.000
3/4 = 0.750	1.000	→ ◊	0.707	0.707
3/4 = 0.750	1.000	→ ◁	0.866	0.500
1	1 1/4 = 1.250	→ □	1.250	1.250
1	1 1/4 = 1.250	→ ◊	0.884	0.884
1	1 1/4 = 1.250	→ ◁	1.082	0.625

Source: From Bell, K. J., *Heat Exchangers — Thermal–Hydraulic Fundamentals and Design*, 1981. With permission.

Example 9.4 continue

The Reynolds number is based on the tube outside diameter and the velocity on the crossflow area at the diameter of the shell. Note that distilled water circulates on the shell side:

$$Re_s = \frac{\rho u_s d_o}{\mu} = \frac{\dot{m}}{A_s}\frac{d_o}{\mu}$$

which was calculated in Example 9.3 as

$$Re_s = 15,968$$

Now, calculate the Fanning friction coefficient, which is given by Equation 9.26:

$$f_i = b_1\left(\frac{1.33}{P_T/d_o}\right)^b (Re_s)^{b_2}$$

Since $P_T/d_0 \approx 1.33$, this equation can be simplified to

$$f_i = b_1(Re_s)^{b_2}$$

Example 9.4 continue

From Table 9.6, $b_1 = 0.391$, $b_2 = -0.148$; therefore, the friction coefficient

$$f_i = 0.391(15{,}968)^{-0.148} = 0.093$$

If there were no leakage or bypass, the pressure drop in one crossflow section can be calculated from Equation 9.32:

$$\Delta p_{bi} = 4 f_i \frac{G_s^2}{2\rho_s}\left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14} N_c$$

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{50}{0.073} = 684.9 \text{ kg/m}^2\text{s}$$

$$\Delta p_{bi} = 4 \times 0.0933 \times \frac{684.9^2}{2 \times 995.9} \times 12 = 1055 \text{ Pa}$$

Assuming $R_b = 0.60$ and $R_l = 0.4$, the combined pressure drop of the entire interior crossflow section can be calculated from Equation 9.27:

$$\Delta p_c = \Delta p_{bi}(N_b - 1)R_l R_b$$

$$\Delta p_c = 1137(9 - 1)0.60 \times 0.4 = 2.18 \text{ kPa}$$

Example 9.4 continue

where N_b is the number of baffles:

$$N_b = \frac{L}{B} - 1$$

For an ideal baffle window section, Δp_{wi} is calculated from Equation 9.33. The number of effective crossflow rows in each window, N_{cw}, can be estimated from Equation 9.36:

$$N_{cw} = \frac{0.8\,L_c}{P_p}$$

where P_p is given in Table 9.7 as 0.0254 m. L_c is the baffle cut distance from the baffle tip to the shell inside diameter:

$$L_c = 0.25\,D_s = 0.25 \times 0.580 = 0.145 \text{ m}$$

$$N_{cw} = \frac{0.8 \times 0.145}{0.0254} = 4.6 \cong 5$$

The area of flow through the baffle windows is

$$A_w = A_{wg} - A_{wt}$$

Example 9.4 continue

where A_{wg} is the gross window area and A_{wt} is the window area occupied by tubes. The expressions to calculate A_{wg} and A_{wt} are given by Taborek[6] and Bell.[11,12] Here it is given as $A_w = 0.076\ m^2$, then, from Equation 9.33,

$$\Delta p_{wi} = \frac{\dot{m}_s^2 (2 + 0.6 N_{cw})}{2\rho A_s A_w}$$

$$\Delta p_{wi} = \frac{50^2 (2 + 0.6 \times 5)}{2 \times 995.9 \times 0.073 \times 0.076} = 1131\ \text{Pa}$$

The total pressure drop in all the windows is

$$\Delta p_w = \Delta p_{wi} N_b R_l$$

$$\Delta p_w = 1131 \times 9 \times 0.4 = 4072\ \text{Pa}$$

The total pressure drop over the heat exchanger on the shell side can be calculated from Equation 9.31:

$$\Delta p_s = [(N_b - 1)\Delta p_{bi} R_b + N_b \Delta p_{wi}]\, R_l + 2\Delta p_{bi}\left(1 + \frac{N_{cw}}{N_c}\right) R_b R_s$$

Baffle spacing in the inlet, exit, and central regions are equal, so $R_s = 1$:

$$\Delta p_s = [(9 - 1) \times 1105 \times 0.60 + 9 \times 1131]\, 0.4 + 2 \times 1105 \times \left(1 + \frac{5}{12}\right)$$

Example 9.4 continue

The total pressure drop over the heat exchanger on the shell side can be calculated from Equation 9.31:

$$\Delta p_s = \left[(N_b - 1)\Delta p_{bi} R_b + N_b \Delta p_{wi}\right] R_l + 2\Delta p_{bi}\left(1 + \frac{N_{cw}}{N_c}\right) R_b R_s$$

Baffle spacing in the inlet, exit, and central regions are equal, so $R_s = 1$:

$$\Delta p_s = \left[(9-1)\times 1105 \times 0.60 + 9 \times 1131\right] 0.4 + 2 \times 1105 \times \left(1 + \frac{5}{12}\right)$$

$$\times 0.60 = 8.07 \text{ kPa}$$

which is less than the allowable pressure drop, so the heat exchanger is suitable. The shell-side pressure drop could be overestimated if it were calculated without baffle leakage and without tube bundle bypass effects.

Δp_c [kPa]	Δp_w [kPa]	Δp_e [kPa]	Δp_s [kPa]
2.03	4.07	1.79	7.9

Example 9.4 continue

Shell side pressure drop using Kern procedure

Now calculate the shell-side pressure drop by the use of the Kern method, which does not take into account the baffle leakage and bypass effects. The shell-side pressure drop can be calculated from Equation 9.17:

$$\Delta p_s = \frac{f G_s^2 (N_b + 1) D_s}{2 \rho D_e \phi_s}$$

where D_e is the equivalent diameter, which is calculated from Equation 9.13 and is given in Example 9.3 as

$$D_e = 0.024 \text{ m}$$

The friction coefficient is calculated from Equation 9.18, where

$$Re_s = \left(\frac{\dot{m}_s}{A_s} \right) \frac{D_e}{A_s} = G_s \frac{D_e}{\mu}$$

Example 9.4 continued

which is given in Example 9.3 as

$$Re_s \approx 20170$$

Then,

$$f = \exp(0.576 - 0.19 \ln Re_s)$$

$$f = 0.271$$

Assuming that $\phi_s = 1$ and inserting the calculated values into Equation 9.17, Δp_s becomes

$$\Delta p_s = f G_s^2 \frac{(N_b + 1) D_s}{2 \rho D_e \phi_s} = 0.271 \left(\frac{50}{0.073} \right)^2 \frac{(9+1) \times 0.58}{2 \times 995.9 \times 0.024 \times 1}$$

$$= 15400 \text{ Pa} = 15.4 \text{ kPa}$$

The shell-side pressure drop obtained by Bell-Delaware method is about 48% lower than that obtained by the Kern method.

Example 9.5

Distilled water with a mass flow rate of 80,000 kg/hr enters the shell-side of an exchanger at 35°C and leaves at 25°C. The heat will be transferred to 140,000 kg/hr of raw water coming from a supply at 20°C. The baffles will be spaced 12 in. apart. A single shell and single tube pass is preferable. The tubes are 18 BWG tubes with a 1 in. outside diameter (OD = 0.0254 m, ID = 0.0229 m) and they are laid out in square pitch. Shell diameter is 15.25 in. A pitch size of 1.25 in. and a clearance of 0.25 in. are selected. Calculate the length of the heat exchanger and the pressure drop for each stream. If the shell-side allowable maximum pressure drop is 200 kPa, will this heat exchanger be suitable?

Solution

The tube-side specifications are

Outer diameter	$d_o = 1 \text{ in.} = 0.0254 \text{ m}$
Inner diameter	$d_i = 0.902 \text{ in.} = 0.0229108 \text{ m}$
Flow area	$A_c = 0.639 \text{ in.}^2 = 0.00041226 \text{ m}^2$
Wall thickness	$t_w = 0.049 \text{ in.} = 0.0012446 \text{ m}$

Calculate the mass flow rate (from problem statement) by

$$\dot{m}_t = \frac{14{,}000 \text{ kg/hr}}{3600 \text{ s/hr}} = 38.89 \text{ kg/s}$$

Example 9.5 continue

The shell-side specifications are

Pitch size	$P_T = 1.25$ in. $= 0.03175$ m
Clearance	$C = 0.25$ in. $= 0.00635$ m
Baffle spacing	$B = 12$ in. $= 0.3048$ m
Shell diameter	$D_s = 15.25$ in. $= 0.38735$ m

Calculate the mass flow rate (from problem statement) by

$$\dot{m}_s = \frac{80{,}000 \text{ kg/hr}}{3600 \text{ s/hr}} = 22.22 \text{ kg/s}$$

For a single pass shell-and-tube heat exchanger with a diameter of 15.25 in., from Table 9.3, the number of tubes, N_t, in a 1.25 in. square pitch with an outer tube diameter of 1 in. is 81 tubes.

Example 9.5 continue

The correction factor is assumed to be $F \approx 1$.

Calculate the shell-side heat transfer coefficient by

$$A_s = \frac{(D_s C B)}{P_t} = \frac{(0.38735 \times 0.00635 \times 0.3048)}{0.03175} = 0.02361 \text{ m}^2$$

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{22.22 \text{ kg/s}}{0.02361 \text{ m}^2} = 941.107 \text{ kg/m}^2 \cdot \text{s}$$

$$D_e = \frac{4(P_t^2 - \pi d_o^2/4)}{\pi d_o} = \frac{4\left[(0.03175)^2 - \pi \cdot (0.0254)^2/4\right]}{\pi \cdot 0.0254} = 0.02513 \text{ m}$$

$$Re_s = \frac{D_e G_s}{\mu} = \frac{0.02513 \cdot 941.107}{0.000797} = 29673.8$$

Therefore, the flow of the fluid on the shell side is turbulent. Using McAdam's correlation, Equation 9.11, we get the Nusselt number:

$$Nu = 0.36\left(\frac{D_e G_s}{\mu_b}\right)^{0.55}\left(\frac{c_p \mu_b}{k}\right)^{0.33}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$= 0.36\left(\frac{0.02513 \times 941.107}{0.000797}\right)^{0.55}\left(\frac{4178.5 \times 0.000797}{0.614}\right)^{0.33}\left(\frac{0.000797}{0.00086}\right)^{0.14}$$

$$= 179.39$$

It is assumed that the tube wall temperature is 26°C and $\mu_w = 0.00086$ kg/m· s.

The shell-sidve heat transfer coefficient, h_o, is then calculated as

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{179.30 \times 0.614}{0.02513} = 4{,}383.09 \text{ W/m}^2 \cdot \text{K}$$

Example 9.5 continue

Calculate the tube side heat transfer coefficient by

$$A_t = \frac{\pi d_i^2}{4} = \frac{\pi \times (0.02291)^2}{4} = 0.0004122\ \text{m}^2$$

$$A_{tp} = \frac{N_t A_t}{\text{Number of passes}} = \frac{81 \times 0.0004122}{1} = 0.03339\ \text{m}^2$$

$$G_t = \frac{\dot{m}_t}{A_{tp}} = \frac{38.889}{0.03339} = 1164.6\ \text{kg/m}^2 \cdot \text{s}$$

$$u_t = \frac{G_t}{\rho} = \frac{1164.6}{997} = 1.1682\ \text{m/s}$$

$$Re_t = \frac{u_t \rho d_i}{\mu} = \frac{1.1682 \times 997 \times 0.02291}{0.00095} = 28{,}087.5$$

Example 9.5 continue

Therefore, the flow of the fluid on the tube side is turbulent. Using the Petukhov–Kirillov correlation,

$$Nu = \frac{(f/2)Re\,Pr}{1.07 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)}$$

where $f = (1.58 \ln Re - 3.28)^{-2} = [1.58 \times \ln(28{,}087.5) - 3.28]^{-2} = 0.0060$

$$Nu = \frac{(0.006/2) \times 28087.5 \times 6.55}{1.07 + 12.7 \times (0.006/2)^{1/2} \times (6.55^{2/3} - 1)} = 196.45$$

The tube-side heat transfer coefficient, h_i, is then found as

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{196.45 \times 0.6065}{0.02291} = 5200.5\ \mathrm{W/m^2 \cdot K}$$

The overall heat transfer coefficient, U_o, is determined by the following equation:

$$U_o = \frac{1}{\dfrac{d_o}{d_i h_i} + \dfrac{d_o \ln(d_o/d_i)}{2k} + \dfrac{1}{h_o}}$$

$$= \frac{1}{\dfrac{0.0254}{0.02291 \times 5200.5} + \dfrac{0.0254 \times \ln(0.0254/0.02291)}{2 \times 54} + \dfrac{1}{4383.09}}$$

$$= 2147.48\ \mathrm{W/m^2 \cdot K}$$

Example 9.5 continue

To find the area and, consequently, the length of heat exchanger, the required heat transfer rate must first be determined. This heat transfer rate is determined by

$$Q = \dot{m}_s c_p \left(T_{h_1} - T_{h_2}\right) = 22.22 \times 4178.5 \times (35 - 25) = 928.5 \text{ kW}$$

The heat transfer rate is also defined as

$$Q = U_o A F \Delta T_{lm,cf}$$

Therefore, the area can be determined by

$$A = \frac{Q}{U_o F \Delta T_{lm,cf}} = \frac{928.5 \times 1000}{2147.48 \times 1 \times 7.2135} = 59.93 \text{ m}^2$$

and the length is

$$L = \frac{A}{N_t \pi D_o} = \frac{59.77}{81 \times \pi \times 0.0254} = 9.28 \text{ m}$$

Example 9.5 continued

The shell-side pressure drop can be calculated from Equation 9.17:

$$\Delta p_s = \frac{f G_s^2 (N_b + 1) D_s}{2 \rho D_e \phi_s}$$

$$N_L = \frac{L}{B} - 1 = \frac{9.28}{0.3048} - 1 \approx 29$$

$$f = \exp(0.576 - 0.19 \ln R_s)$$

$$f = \exp(0.576 - 0.19 \ln 29{,}673.8) = 0.2514$$

$$\Delta p_s = \frac{0.2514 \times (941.107)^2 \times (29+1) \times 0.38735}{2 \times 995.7 \times 0.02513 \times \left(\frac{7.97}{8.6}\right)^{0.14}} = 52{,}257 \text{ Pa} = 52.3 \text{ kPa}$$

$$\Delta p_s = 52.3 \text{ kPa} < 200 \text{ kPa}$$

Therefore, this heat exchanger is suitable.

The tube-side pressure drop can be calculated from Equation 9.22:

$$\Delta p_t = \left(4 f \frac{L N_p}{d_i} + 4 N_p\right) \frac{\rho u_m^2}{2}$$

$$\Delta p_t = \left(\frac{4 \times 0.0060 \times 9.28 \times 1}{0.02291} + 4 + 1\right) \times 997 \times \frac{1.1682^2}{2}$$

$$= 9300 \text{ Pa} = 9.3 \text{ kPa}$$