2.7. Examples of Design Problems for Low- and Medium-Finned Trufin in Shell and Tube Heat Exchangers

In this section two examples of heat exchanger design using Trufin are worked out to illustrate the use of the methods just described. The first problem is for a water cooled air compressor aftercooler, cooling air from 350°F to 125°F. This is the same problem that was previously used to illustrate the use of the preliminary design procedure, and the results of that problem will be used as the starting point for the solution. The second problem involves heat recovery, using a high temperature gas oil stream to preheat an incoming medium crude; this problem will be worked from scratch using the preliminary procedures already developed to get started.

2.7.1. Design Of A Compressor Aftercooler

The problem is to design a heat exchanger to cool 13,000 SCFM (58,500 lb/hr) of air at 65 psig from 350 to 125°F, using cooling water available at 80°F. The unit was specified to be a U-tube configuration using type S/T Trufin 3/4 in. O.D., 26 fins per in. (catalog No. 65-265058), of phosphorus - deoxidized copper. The tubes are to be laid out on a I in. equilateral triangular pitch.

The important tube dimensions are

d_o =	0.750 in.	d_i =	0.509 in.
d_r =	0.625 in.	A_o =	0.640 ft^2/ft
H =	0.0625 in.	A_i =	0.133 ft^2/ft
Y =	0.012 in.	S_i =	0.206 in^2. (inside cross-sectional flow area)
s =	0.026 in.		
Δx_w =	0.058 in.	k_w =	170 Btu/hr ft°F

The air properties are evaluated at 65 psia and at a mean air side temperature of 240°F, except for the wall viscosity which is evaluated at 130°F (a rough approximation ahead of time, but the solution is very insensitive to this value.)

The values used are:

Density	0.300 lb/ft^3
Specific heat	0.241 Btu/lb_m °F
Viscosity (bulk)	5.40 X 10^{-2} lb_m/ft hr
Viscosity (wall)	4.68 x 10^{-2} lb_m/ft hr
Thermal conductivity	0.0188 Btu/hr ft °F

The water outlet temperature will be assumed to be 110°F and the properties evaluated at a mean bulk temperature of 95°F, with an estimated mean wall temperature of 130°F. The values used are:

Density	62.0 lb_m/ft^3
Specific heat	1.00 Btu/lb_m °F
Viscosity (bulk)	1.84 lb_m/ft hr
Viscosity (wall)	1.31 lb_m/ft hr
Thermal conductivity	0.36 Btu/hr ft°F

The fouling factor of the water will be taken as 0.001 hr ft^2 °F/Btu; no fouling factor will be assigned to the air.

The preliminary design estimate suggested the following units:

Inside shell diameter, in.	Effective tube length, ft.
15 1/4	12
17 1/4	10
19 1/4	8

For low pressure compressor intercoolers or aftercoolers, pressure drop of the gas is a major, consideration. This suggests looking at the larger diameter, shorter shells (contrary to the usual case for high pressure gases or liquids). Therefore, the following design is directed towards the 19 1/4 in. inside diameter shell.

In the same philosophy, we select a baffle spacing near the maximum allowable under TEMA standards, which is the inside diameter of the shell; the value tried will be 18 in. Also, we select the maximum baffle cut, which provides only enough overlap to ensure that the central tube rows pass through all baffles. Since this is a U-tube bundle and some of the tubes near the centerline must be omitted because of bend radius restrictions, the maximum allowable cut will be about 8 in., which is the value chosen. We simply step through the various calculations in the order they come:

Shell-side geometry data:

d_r	=	0.625 in.
d_o	=	0.750 in.
s	=	0.026 in.
Y	=	0.012 in.
p	=	1 in., equilateral triangular layout
D_i	=	19 1/4 in.
$D_{ot\ell}$	=	18 3/4 in.

Effective tube length: This will be about 8 feet, but the exact determination will be left until after the heat transfer coefficients have been calculated. Then the required heat transfer area and the corresponding length will be calculated.

$$\ell_c = 8\ in.$$

$$\ell_s = 18\ 3/4\ in.$$

N_{ss} = 0; This is indicated in this case for several reasons: The small clearance between D_i and $D_{ot\ell}$ the small crossflow distance due to the large baffle cut, and the fact that at least two of the center line rows are missing anyway due to the minimum bend radius.

Calculation of shell-side geometrical parameters

1. $N_t = \dfrac{282}{1.08}$ = 260; the fixed tube sheet, two-tube pass value of N_t is 282 by Table 2.6. This must be corrected for the U-tube construction using F_3 from Table 2.4.

2. p_p = 0.866 in: p_n = 0.500 in.

3. $N_c = \dfrac{19.25\left[1-2\left(\dfrac{8}{19.25}\right)\right]}{0.866} = \quad 4 \rightarrow 2$ (2.38)

While N_c = 4 by the calculation, at least two tube rows will be lost by the minimum bend radius requirement for the U-tube construction. So N_c = 2 will be used for the remainder of the calculation.

4. $\ell_c / D_i = 8/19.25 = 0.416$
F_c = 0.25 from Fig.2.28.

However, since at least two crossflow rows are lost near the centerline, F_c will be reduced to 0. 15 for the remainder of the calculations.

5. $N_{cw} = \dfrac{0.8(8)}{0.866} = \quad 7.4 \rightarrow 8$ (2.40)

6. N_b will be calculated after the tube length required for heat transfer area is known.

7. $S_m = 18\left\{19.25 - 18.75 + \left(\dfrac{18.75-0.75}{1}\right)\left[(1-0.75) + 2(0.0625)\left(\dfrac{0.026}{0.038}\right)\right]\right\} = 119 \;\; in.^2$ (2.43)

8. $F_{sbp} = \dfrac{(19.25-18.75)(18)}{119} = 0.076$ (2.44)

9. S_{tb} = 0.0148(260)(1.15) = 5.5 in^2

10. S_{sb} = 2.5 $in.^2$ from Fig. 2.29

11. From Fig. 2.30, S_{wg} = 114.6 $in.^2$
From Eq. (2.51), S_{wt} = 48.8 $in.^2$
Therefore, S_w = 65.8 $in.^2$ from Eq. (2.49)

12. Not needed, since the shell-side flow is turbulent.

Calculation of shell-side heat transfer characteristics:

1. $\mathrm{Re}_s = \left(\dfrac{0.625}{12}\right)\dfrac{(58{,}500)(144)}{(119)(0.054)} = 68{,}300$ (2.54)

2. j_s = 0.0055 from Fig 2.15

3. $$h_{o,i} = 0.0055(0.241)\left[\frac{(58{,}500)(144)}{119}\right]\left[\frac{0.0188}{(0.241)(0.054)}\right]^{2/3} \times \left(\frac{0.0540}{0.0468}\right)^{0.14} = 122 \;\; Btu/hr \;\; ft^2 \, ^\circ F \qquad (2.55)$$

4. J_c=0.65 (at F_c = 0.15 from Fig. 2.33)

5. $$\frac{S_{tb} + S_{sb}}{S_m} = \frac{5.50 + 2.50}{119} = 0.067$$

$$\frac{S_{sb}}{S_{tb} + S_{sb}} = \frac{2.5}{8} = 0.313$$

$$J_\ell = 0.86 \;\; from \;\; Fig. \; 2.34$$

6. J_b = 0.91 from Fig. 2.35

7. J_r = 1.00 since Re_s > 100.

8. h_o = 122 (0.65)(0.86)(0.91) (2.56)
h_o = 62.1 Btu/hr ft^2°F

Calculation of required heat transfer area:

1. Calculate heat load

Q_T = (58,500)(0.241)(350 – 125) = 3.17 x 10^6 Btu/hr (2.32)

2. Water flow rate, assuming an outlet temperature of 110°F

$$W_i = \frac{3.17 \times 10^6}{(110 - 80)(1.00)} = 105{,}700 \frac{lb}{hr} \qquad (2.33)$$

3. Water velocity, assuming two tube-side passes

$$V_i = \frac{105{,}700(144)}{(260/2)(0.206)(62.0)(3600)} = 2.55 \quad ft/\sec$$

This is possible value, but good design would generally call for a water velocity above 3 ft/sec in the tubes. If we go to four tube-side passes, N_t is about 240 tubes, and the tube-side velocity becomes

$\dfrac{(260/2)}{(240/4)}(2.55) = 5.53 \quad ft/\sec,$ which is better practice.

4. Using Fig. 2.19,

$$h_i = 1280(1.015) = 1300 \quad Btu/hr \;\; ft°F$$

5. Using the equation developed in Chapter 1 for fin resistance, or from Fig. 1.52

   R_{fin} = 8 x 10^{-5} hr ft^2 °F/Btu

6. Using Eq. 2.2

$$U_o = \cfrac{1}{\frac{1}{62.1} + (8\times10^{-5}) + \frac{0.058}{12(170)}\left(\frac{0.640}{0.148}\right) + \left(0.001 + \frac{1}{1300}\right)\left(\frac{0.640}{0.133}\right)}$$

$$= \frac{1}{1.61\times10^{-2} + 8\times10^{-5} + 1.23\times10^{-4} + 8.51\times10^{-3}}$$

$$= 40.3 \quad Btu/hr \;\; ft^2 °F$$

7. $$LMTD = \frac{(350-110)-(125-80)}{\ell n\left(\frac{350-110}{125-80}\right)} = 116.5°F \qquad (2.11)$$

8. $$P = \frac{110-80}{350-80} = 0.111 \qquad (2.14)$$

$$R = \frac{350-125}{350-80} = 7.5 \qquad (2.13)$$

   F = 0.9 From Fig. 2.5

9. MTD = 0.9(116.5) = 104.8°F (2.10)

10. $$A_o = \frac{3.17\times10^6}{40.3(104.8)} = 751 \quad ft^2 \qquad (2.36)$$

which gives a required effective tube length of

$$\frac{751}{240(0.640)} = 4.9 \quad ft.$$

If we choose an effective tube length of 6 feet, then we require 3 baffles (= 4 baffle spaces, each 18 in. = 1 ½ feet baffle spacing.) This puts both nozzles on the same side of the shell, which we shall assume is satisfactory in this case.

Calculation of shell-side pressure drops:

1. f_s = 0.20 from Fig. 2.17.

2. $$\Delta P_{b,i} = \frac{4(0.20)(58{,}500)^2(2)(144)^2}{2(0.30)(4.17\times10^8)(119)^2}\left(\frac{4.68\times10^{-2}}{5.40\times10^{-2}}\right)^{0.14} = 31.4\ \ lb_f\ /\ ft^2 \tag{2.57}$$

3. $$\Delta P_{w,i} = \frac{(58{,}500)^2\left[2+0.6(8)\right](144)^2}{2(4.17\times10^8)(119)(65.8)(0.3)} = 246\ \ lb_f\ /\ ft^2 \tag{2.58}$$

4. $R_\ell = 0.67 \quad from\ \ Fig.\ 2.38.$

5. R_b = 0.73 from Fig. 2.39. This value is actually high since we have had to go to four tube passes. There will be an internal bypass channel, which in usual practice will be partially blocked by tie rods. The effect on heat transfer will be small, but it is possible that the effect on pressure drop might be to drop R_b to as low as 0.55. We will use the higher value here.

6. Then, using Eq.2.60 and N_b = 3:

$$\Delta P_s = \left\{\left[2(31.4)(0.73)+3(246)\right]0.67 + 2(31.4)(0.73)\left(1+\frac{8}{2}\right)\right\}\left(\frac{1}{144}\right)$$

$$\Delta P_s = 5.24\ \ lb_f\ / in.^2$$

This is a feasible value though perhaps higher than we would like; but as discussed previously, this calculation is likely to be conservative. If we had to reduce this value, we could do any of the following:

a. Increase the shell diameter. But the chosen shell is already quite large in diameter compared to its length.

b. Use a TEMA J shell ("split flow"), which would reduce the pressure drop by about a factor of four at the cost of substantially more heat transfer area.

c. Use double-segmental baffles, which would have roughly the same effect as using a J shell.

d. Use a "no-tubes-in-the-window-design", reducing the baffle cut somewhat and substantially increasing the number of crossflow tubes compared to the present design, and probably increasing the length substantially.

The Delaware method has not at this point been developed to apply to the geometry modifications suggested in b, c, and d above.

Calculation of tube-side pressure drop:

1. $$\mathrm{Re}_i = \left(\frac{0.509}{12}\right)\left(\frac{(62.0)(5.53)(3600)}{(1.84)}\right) = 28{,}500 \tag{2.19}$$

2. f_i = 0.006 from Fig. 2.20.

3. $$L = 4(6) + 2\left(\frac{1}{2}\right)\pi\left(\frac{18.75}{12}\right)$$

L = 28.9 ft.

where the last term is a conservative estimate of the added length of flow in the U-bends.

4. From Eq. (2.26)

$$\Delta P_i = \frac{2(0.006)(62.0)(5.53)^2(28.9)}{\left(\frac{0.509}{12}\right)(32.2)}\left(\frac{1.31}{1.84}\right)^{0.14} = 459 \ \ lb_f \ / \ ft^2 = 3.19 \ \ lb_f \ / in.^2$$

5. The loss for two tube entrances is, from Eq. (2.25),

$$\Delta P_{ent} = 2\left[\frac{3(62.0)(5.53)^2}{2(32.2)}\right] = 177\frac{lb_f}{ft^2} = 1.23 \ \ lb_f \ / in.^2$$

These values are well within standard practice.

If the shell-side pressure drop of possibly as much as 5.3 lb_f / $in.^2$ is acceptable, the unit designed above will do the job.

Summary of Major Design Parameters

Shell dimensions: 19 1/4 in. ID x 6 ft. effective tube length (tube sheet face to tangent line.)

Shell type: U-tube

Baffles: Segmental, 41.6 percent cut; 3 baffles, spaced 18 in. apart.

Tubes: Wolverine Type S/T Trufin, 65-265058-01 (3/4in. OD, 26 fins per in., 0.058 in. wall, phosphorous deoxidized copper). Four tube-side passes.

Tube layout: 3/4 in. OD tubes on I in. triangular pitch.

Sealing strips: None.

2.7.2. Design of A Gas Oil to Crude Heat Recovery Exchanger

The problem is to design a split ring floating head exchanger to heat 49,800 bpd (597,000 lb/hr) of 34° API MidContinent Crude from 125°F to 180°F, using 13,200 bpd (152,000 lb/hr) of 28° API Gas Oil at 410°F, cooling it to 220°F. The unit will use type S/T Trufin I in. OD, 19 fins/in. (Catalog No. 60-197083), of low carbon steel. The tubes are to be laid out on a 1 1/4 in. rotated square. Pressure drop is limited to 15 psi on each side.

The important tube dimensions are:

d_o	=	1.00 in.	d_i	=	0.709 in.
d_r	=	0.875 in.	A_o	=	0.688 ft^2/ft
H	=	0.0625 in.	A_i	=	0.186 ft^2/ft
Y	=	0.017 in.	S_i	=	0.395 $in.^2$
s	=	0.036 in.	K_w	=	26 Btu/hr ft°F
Δx_w	=	0.083 in.			

The properties of the shell-side fluid (34° API crude) at a mean fluid temperature of 150°F are:

Density	51.2 lb_m/ft^2
Specific Heat	0.51 Btu/lb_m°F
Viscosity (bulk)	7.0 lbm/ft hr
Viscosity (wall, at 2000F)	4.4 lb_m/ft hr
Thermal conductivity	0.071 Btu/hr ft°F

The properties of the tube-side fluid (28° API Gas Oil) at a mean fluid temperature of 315°F are:

Density	49.3 lb_m/ft,
Specific heat	0.58 Btu/lb_m °F
Viscosity (bulk)	2.90 lb_m/ft hr
Viscosity (wall, at 2000F)	7.50 lb_m/ft hr
Thermal conductivity	0.061 Btu/hr ft°F

The fouling factor for both the crude and the gas oil will be taken as 0.002 hr ft^2°F/Btu for each stream.

The next step is to estimate the dimensions of the heat exchanger required, using the procedure for approximate size estimation given previously in this section.

$$Q = 597{,}000\,(0.51)(180-125)$$
$$= 1.67 \times 10^7 \text{ Btu/hr for the crude} \qquad (2.32)$$
$$Q = 152{,}000\,(0.58)(410-220)$$
$$= 1.68 \times 10^7 \text{Btu/hr for the gas oil} \qquad (2.33)$$

$$LMTD = \frac{(410-180)-(220-125)}{\ell n\left(\frac{410-180}{220-125}\right)} = 152.7°F \qquad (2.11)$$

$$P = \frac{220-410}{125-410} = 0.667 \qquad (2.14)$$

$$R = \frac{125-180}{220-410} = 0.289 \qquad (2.13)$$

From Fig. 2.5, F = 0.92

U_o may be estimated from Table 2.1, using the value for a medium organic fluid in the tubes and a heavy organic fluid on the shell. A median value of 30 Btu/hr-ft^2°F will be sufficient for present purposes.

Then the actual area required may be estimated from

$$A_o = \frac{1.67x10^7}{(152.7)(0.92)(30)} = 3960 \quad ft^2 \tag{2.36}$$

To obtain a value to enter Fig. 2.26, the following correction factors are needed:

F_1 = 1.54 for 1 in. tubes on a 1 1/4 in. rotated square pitch

F_2 = 1.03 for two tube passes and a shell inside diameter between 25 and 33 in. (to be checked later, if necessary.)

F_3 = 1.09 for a split ring floating head and a 23 1/4 - 35 in. shell inside diameter.

F_4 = 0.97 for I in. OD S/T Trufin, 19 fins/in.

A_o = 3960(1.54)(1.03)(1.09)(0.97) = 6640 ft2 for entry into Fig. 2.26. (2.37)

From Fig. 2.26, we see the following combinations answer to this requirement:

Shell inside diameter, in.	Effective tube length, ft.	L/D
37	10	3.2
35	11.5	3.9
33	13	4.7
31	14.5	5.6
29	17	7.0
27	19.5	8.7
25	23	11.0
23 1/4	27	13.9

Undoubtedly several of these could be chosen and designed to meet the thermal-hydraulic performance. Because of the high shell-side flow rate, let us choose the 31 in. ID shell for at least preliminary evaluation. Before proceeding through the complete Delaware method, we can check the tube-side velocity to ensure that it is within reasonable limits:

For two passes:

$$N_t = \frac{417}{F_3} = \frac{417}{1.09} = 382$$

(Here we have taken the fixed tube sheet tube count of 417 from Table 2.6 for the given tube layout and divided it by F_3 (= 1.09) for the split ring floating head configuration to obtain the estimate of 382 tubes in the bundle, or 191 per pass.) Then the tube-side velocity is

$$V_i = \frac{(152{,}000)(144)}{(191)(0.395)(49.3)(3600)} = 1.63 \quad ft/\sec$$

This value is too low, and we can readily estimate that going to six tube-side passes would give a velocity of about 5 ft/sec, which would be a better design for fouling control and probably still be acceptable for tube-side pressure drop. So six tube-side passes will be used for the tube-side.

Entering now upon the Delaware method, we list the following basic shell-side geometry values:

d_r	=	0.875 in.
d_o	=	1.00 in.
Y	=	0.017 in.
p	=	1 1/4 in., rotated square (45°) layout
D_i	=	31 in.
$D_{ot\ell}$	=	29 3/8 in.

L (Effective tube length): As in the previous examples, this will be determined once the heat transfer coefficients have been calculated.

ℓ_c = 10.8 in. This is a "35 percent cut", based on the diameter. This must be adjusted up or down somewhat in the final design to correspond to the actual centerline of a row of tubes.

ℓ_s = 16 in. This is about half the shell diameter, a common first choice for liquid flow on the shell side. This value can be adjusted in either direction to give evenly spaced baffles in the shell, or to adjust the heat transfer coefficient or pressure drop up or down as needed.

N_{ss} will be chosen later to give one pair of sealing strips for approximately every six rows of tubes in crossflow (N_c).

Calculation of shell-side geometrical parameters:

1. $N_t = \frac{387}{1.09}$ = 355, from Tables 2.6 and 2.4.

2. P_p = 0.884 in.; p_n = 0.884 in., from Table 2.7.

3. $$N_c = \frac{31[1 - 20(0.35)]}{0.884} = 10.5 \approx 10 \qquad (2.38)$$

   Use $N_{ss} \approx \frac{10}{6}$; i.e., 2 pairs of sealing strips

4. F_c = 0.40 from Fig. 2.28.

5. $$N_{cw} = \frac{0.8(10.8)}{0.884} = 9.8 \approx 10 \qquad (2.40)$$

6. Calculate N_b later

7. $S_m = 16\left\{31 - 29\tfrac{3}{8} + \frac{29\tfrac{3}{8} - 0.75}{0.884}\left[1\tfrac{1}{4} - 1 + 2(0.0625)\left(\frac{0.036}{0.036 + 0.017}\right)\right]\right\} = 200\ in.^2$ (2.42)

8. $F_{sbp} = \frac{(31 - 29\tfrac{3}{8})16}{200} = 0.130$ (2.44)

9. S_{tb} = 0.0245(355)(1.40) = 12.2 in.^{2} (2.47)

10. S_{sb} = 9.2 in.^{2} from Fig. 2.29

11. S_{wg} = 235 in.^{2} from Fig. 2.30
S_{wt} = 84 in.^{2} from Eq. 2.51
S_w = 235-84 = 151 in.^{2}

12. Not needed for this case.

Calculation of shell-side heat transfer coefficient:

1. $\mathrm{Re}_s = \frac{0.875(597{,}000)(144)}{12(7.0)(200)} = 4480$ (2.54)

2. j_s = 1.1 x10^{-2} from Fig. 2.15.

3. $h_{o,i} = 1.1 \times 10^{-2}(0.51)\left[\frac{597{,}000(144)}{200}\right]\left[\frac{0.071}{0.51(7.0)}\right]^{2/3}\left(\frac{7.0}{4.4}\right)^{0.14} = 189\ \ Btu/hr\ \ ft^2\,{}^\circ F$ (2.55)

4. J_c = 0.845 from Fig. 2.33.

5. $\frac{S_{sb} + S_{tb}}{S_m} = \frac{9.2 + 12.2}{200} = 0.017$

$\frac{S_{sb}}{S_{sb} + S_{tb}} = \frac{9.2}{9.2 + 12.2} = 0.430$

$J_\ell = 0.80\ \ from\ Fig.\ 2.34$

6. $\frac{N_{ss}}{N_c} = \frac{2}{10} = 0.20$

J_b = 0.95 from Fig. 2.35

7. h_o = 189(0.845)(0.80)(0.95) (2.56)
=121 Btu/hr-ft^2°F

Calculation of tub-side and overall heat transfer coefficients:

1. Calculate tube-side velocity:

No. of tubes/pass = $\frac{355}{6}$ = 59

$$V_i = \frac{152{,}000(144)}{3600(49.3)(59)(0.395)} = 5.29\frac{ft}{\sec}$$

2. Calculate tube-side Reynolds number:

$$\mathrm{Re}_i = \frac{0.709(49.3)(5.29)(3600)}{12(2.90)} = 19{,}100 \qquad (2.19)$$

3. Calculate the tube-side heat transfer coefficient:

$$h_i = 0.023\left[\frac{0.061(12)}{0.709}\right](19{,}100)^{0.8}\left[\frac{0.58(2.90)}{0.061}\right]^{1/3}\left(\frac{2.90}{7.50}\right)^{0.14} = 167 \ \ Btu/hr \ \ ft^2\,{}^\circ F \qquad (2.23)$$

4. Calculate the overall heat transfer coefficient, using Eq. (2.2). R_{fin} may be obtained from Chapter 1 as 4.9 x 10^{-4} hr $ft^2{}^\circ F$/Btu.

$$U_o = \frac{1}{\frac{1}{121} + 0.002 + 4.9\times10^{-4} + \frac{0.083}{26(12)}\frac{0.688}{0.196} + 0.002\frac{0.688}{0.186} + \frac{1}{167}\frac{0.688}{0.186}}$$

$$= \frac{1}{8.26\times10^{-3} + 2.0\times10^{-3} + 4.9\times10^{-4} + 9.34\times10^{-4} + 7.40\times10^{-3} + 2.21\times10^{-2}}$$

= 24.3 Btu/hr $ft^2{}^\circ F$

5. Calculate required area and length of exchanger:

$$A_o = \frac{Q}{U_o F(LMTD)} = \frac{1.68\times10^7}{24.3(0.92)(152.7)} \qquad (2.36)$$

A_o = 4920 ft^2

$$L = \frac{4920}{355(0.688)} = 20 \ \ ft.$$

For a 16 in baffle spacing, this corresponds to 15 baffle spaces or 14 baffles, putting the nozzles on opposite sides of the shell. If this were not satisfactory, a slightly shorter or longer baffle spacing could be investigated.

Calculation of shell-side pressure drop.

1. From Fig. 2.17, at Re_s = 4480, f_s = 0.38

2. Pressure drop across one ideal crossflow section:

$$\Delta P_{b,i} = \frac{4(0.38)(597{,}000)^2(10)(144)^2}{2(51.2)(4.17\times 10^8)(200)^2}\left(\frac{4.4}{7.0}\right)^{0.14} \tag{2.57}$$

$$\Delta P_{b,i} = 61.6 \ \ lb_f \ / \ ft^2$$

3. Pressure drop through one ideal window section:

$$\Delta P_{w,i} = \frac{(597{,}000)^2[2+0.6(10)](144)^2}{2(4.17\times 10^8)(200)(151)(51.2)} = 45.8 \ \ lb_f \ / \ ft^2. \tag{2.58}$$

4. $R_\ell = 0.58 \ \ from\ Fig.\ 2.38$

5. R_b = 0.87 from Fig. 2.39

6. $\Delta P_s = [13(61.6)(0.87)+14(45.8)]0.58 + 2(61.6)(0.87)\left(1+\frac{10}{10}\right) = 990 \ \ lb_f \ / \ ft^2 = 6.88 \ \ lb_f \ / in.^2$

Even if nozzle losses are added this is well within allowable design limits.

Calculation of tube-side pressure drop.

1. From Fig. 2.20, at Re_i = 19,100.

   f_i = 0.007

2. From Eq. (2.26), with L = 6 x 20 = 120 ft.

$$\Delta P_i = \frac{2(0.007)(49.3)(5.29)^2(120)(12)}{0.709(32.2)}\left(\frac{7.0}{4.4}\right)^{0.14}$$

$$\Delta P_i = 1300 \ \ lb_f \ / \ ft^2 = 9.02 \ \ lb_f \ / in.^2$$

Additionally, entrance/exit losses must be assessed at each nozzle and tube entrance (one per pass) by Eq. (2.24 and 2.25).

These are

$$\Sigma\Delta P_{ent} = 8(3)\left[\frac{49.3(5.29)^2}{2(32.2)}\right] = 514 \ \ lb_f \ / \ ft^2 = 3.57 \ \ lb_f \ / in.^2 \tag{2.24}$$

The total tube-side loss of 12.6 psi is within limits.

Summary of Major Design Parameters:

Shell dimensions: 31 in. ID x 20 ft effective tube length

Shell type: Split ring floating head

Baffles: Segmental, 35 per cent cut; 14 baffles, spaced 16 in. apart

Tubes: Wolverine Type S/T Trufin 60-197083-63 (1 in. OD, 19 fins per inch, 0.083 in. wall, carbon steel.) Six tube passes.

Tube layout: 1 in. OD tubes on 1 ¼ in. rotated square pitch

Sealing strips: Two pairs

TABLE 2.1 TYPICAL OVERALL DESIGN COEFFICIENTS FOR TRUFIN TUBED HEAT EXCHANGERS

TUBE-SIDE FLUID	SHELL-SIDE FLUID	TOTAL FOULING RESISTANCE IN hr ft^2°F/Btu	U_o Btu/hr ft^2°F
Water	Gas, about 10 psig	0.002	15-20
Water	Gas, about 100 psig	0.002	25-35
Water	Gas, about 1000 psig	0.002	50-75
Water	Light organic liquids	0.0025	70-120
Water	Medium organic liquids	0.003	50-80
Water	Heavy organic liquids	0.0035	30-65
Water	Very heavy organic liquids (cooling)	0.005	5-30
Condensing Steam	Gas, about 10 psig	0.0005	15-20
Condensing Steam	Gas, about 100 psig	0.0005	25-40
Condensing Steam	Gas, about 1000 psig	0.0005	60-85
Condensing Steam	Light organic liquids	0.001	100-150
Condensing Steam	Medium organic liquids	0.0015	75-130
Condensing Steam	Heavy organic liquids	0.002	50-85
Condensing Steam	Very heavy organic liquids	0.0035	10-40
Light organic liquids	Light organic liquids	0.0017	60-90
Light organic liquids	Medium organic liquids	0.0022	40-70
Light organic liquids	Heavy organic liquids	0.0027	25-55
Light organic liquids	Very heavy organic liquids	0.0042	5-25
Medium organic liquids	Heavy organic liquids	0.0037	20-40
Medium organic liquids	Very heavy organic liquids	0.0055	5-25

General Notes on Table 2. 1.

1. The total fouling resistance and the overall heat transfer coefficient are based on the total outside tube area, including fins.

2. Allowable pressure drops on each side are assumed to be about 10 psi except for (a) low pressure gas, when the pressure drop is assumed to be about 5 per cent of the absolute pressure, and (b) heavy organics where the allowable pressure drop is assumed to be about 20 to 30 psi.

3. Aqueous solutions give approximately the same coefficients as water.

4. Liquid ammonia gives about the same results as water.

5. "Light organic liquids" include liquids with viscosities less than 0.5 cp, such as hydrocarbons through C8, gasoline, light alcohols and ketones, etc.

6. "Medium organic liquids" include liquids with viscosities between about 0.5 cp and 1.5 cp, such as kerosene, straw oil, hot gas oil, absorber oil, and light crudes.

7. "Heavy organic liquids" include liquids with viscosities greater than 1.5 cp, but not over 50 cp, such as cold gas oil, lube oils, fuel oils, and heavy and reduced crudes.

8. "Very heavy organic liquids" include tars, asphalts, polymer melts, greases, etc., having viscosities greater than about 50 cp. Estimation of coefficients for these materials is very uncertain, and frequently their fouling characteristics are such as to render the use of finned tubes unwise. Values of U_o for heating these liquids are usually significantly higher than for cooling them.

TABLE 2.2 F_1 FOR VARIOUS UNIT CELLS

Tube Outside Diameter, In.	Tube Pitch, In.	Layout	F_1
5/8	13/16	→ ◁	0.90
5/8	13/16	→ ◇, □	1.04
3/4	15/16	→ ◁	1.00
3/4	15/16	→ ◇, □	1.16
3/4	1	→ ◁	1.14
3/4	1	→ ◇, □	1.31
1	1 1/4	→ ◁	1.34
1	1 1/4	→ ◇, □	1.54

TABLE 2.3 F_2, FOR VARIOUS NUMBERS OF TUBE-SIDE PASSES*

INSIDE SHELL DIAMETER, IN.	F_2 NUMBER OF TUBE-SIDE PASSES			
	2	4	6	8
Up to 12**	1.20	1.40	1.80	--
13 ¼ to 17 ¼**	1.06	1.18	1.25	1.50
19 ¼ to 23 ¼	1.04	1.14	1.19	1.35
25 to 33	1.03	1.12	1.16	1.20
35 to 45	1.02	1.08	1.12	1.16
48 to 60	1.02	1.05	1.08	1.12
Above 60	1.01	1.03	1.04	1.06

* Since U-tube bundles must always have at least two passes, use of this table is essential for U-tube bundle estimation.

** Use of this table for the small shell diameters gives very approximate answers.

TABLE 2.4 F_3 FOR VARIOUS TUBE BUNDLE CONSTRUCTIONS

TYPE OF TUBE BUNDLE CONSTRUCTION	F_3 INSIDE SHELL DIAMETER IN.				
	UP TO 12	13 ¼ - 21 ¼	23 ¼ - 35	37 - 48	ABOVE 48
Split Backing Ring (TEMA S)	1.30	1.15	1.09	1.06	1.04
Outside Packed Floating Head (TEMA P)	1.30	1.15	1.09	1.06	1.04
U – Tube* (TEMS U)	1.12	1.08	1.03	1.01	1.01
Pull-Through Floating Head (TEMA T)	—	1.40	1.25	1.18	1.15

* Since U-tube bundles must always have at least two tube-side passes, it is essential to use Table 2.3 also.

TABLE 2.5 F_4 FOR VARIOUS TUBE AREA ENHANCEMENTS

TUBE DESCRIPTION	F_4	TUBE DESCRIPTION	F_4
Plain (unfinned) tube, any outside diameter*	2.56	S/T Trufin, 26 fins/in.: 5/8 in. O.D. 3/4 in. O.D. 1 in. O.D.	 0.76 0.79 0.75
S/T Trufin, 11 fins/in.: 3/4 in. O.D. 7/8 in. O.D. 1 in. O.D.	 0.87 0.83 0.81	S/T Trufin, 28 fins/in.: 3/4 in. O.D. 7/8 in. O.D. 1 in. O.D.	 0.97 0.96 0.96
S/T Trufin, 16 fins/in.: 1/2 in. O.D. 5/8 in. O.D. 3/4 in. O.D. 7/8 in. O.D. 1 in. O.D.	 1.29 1.23 1.20 1.17 1.16	S/T Trufin, 32 fins/in.: 5/8 in. O.D. 3/4 in. O.D. 7/8 in. O.D. 1 in. O.D.	 1.01 1.00 0.99 0.99
S/T Trufin, 19 fins/in.: 3/8 in. O.D. 1/2 in. O.D. 5/8 in. O.D. 3/4 in. O.D. 7/8 in. O.D. 1 in. O.D.	 1.12 1.05 1.02 1.00 0.99 0.97	S/T Trufin, 40 fins/in.: 3/4 in. O.D. 3/4 in. O.D.	 0.78 0.54

*Outside diameter effects are taken into account for F_1.

TABLE 2.6
TUBE COUNTS FOR FIXED TUBE SHEET EXCHANGERS, ADAPTED FROM REF. (9)

SHELL ID D_i, IN.	TUBE OD, d_o, IN.	TUBE PITCH, p. IN. AND LAYOUT	NUMBER OF TUBE PASSES				
			1 PASS	2 PASS	4 PASS	6 PASS	8 PASS
8	¾	15/16 ◁	64	48	34	24	
	¾	1 □ ◇	40	36	24	20	
	¾	1 ◁	42	40	26	24	
	1	1 ¼ □ ◇	24	20	16	12	
	1	1 ¼ ◁	27	26	18	14	
10	¾	15/16 ◁	85	72	52	50	
	¾	1 □ ◇	64	55	39	38	
	¾	1 ◁	73	66	52	44	
	1	1 ¼ □ ◇	36	35	29	20	
	1	1 ¼ ◁	42	40	34	24	
12	¾	15/16 ◁	122	114	94	90	88
	¾	1 □ ◇	92	86	71	68	67
	¾	1 ◁	109	102	88	80	72
	1	1 ¼ □ ◇	55	52	44	38	34
	1	1 ¼ ◁	64	60	52	44	40
13 ¼	¾	15/16 ◁	151	142	124	112	106
	¾	1 □ ◇	115	108	94	85	80
	¾	1 ◁	136	128	112	102	96
	1	1 ¼ □ ◇	70	64	54	48	45
	1	1 ¼ ◁	81	74	62	56	52
15 ¼	¾	15/16 ◁	204	192	166	162	152
	¾	1 □ ◇	154	145	125	122	115
	¾	1 ◁	183	172	146	140	132
	1	1 ¼ □ ◇	92	90	83	79	76
	1	1 ¼ ◁	106	106	96	92	88
17 ¼	¾	15/16 ◁	264	254	228	220	216
	¾	1 □ ◇	200	192	172	166	160
	¾	1 ◁	237	228	208	192	180
	1	1 ¼ □ ◇	127	116	107	98	90
	1	1 ¼ ◁	147	134	124	114	104
19 ¼	¾	15/16 ◁	332	326	290	280	268
	¾	1 □ ◇	251	246	220	212	203
	¾	1 ◁	295	282	258	248	232
	1	1 ¼ □ ◇	158	152	138	131	124
	1	1 ¼ ◁	183	176	160	152	144
21 ¼	¾	15/16 ◁	417	396	364	349	336
	¾	1 □ ◇	315	300	275	263	254
	¾	1 ◁	361	346	318	312	302
	1	1 ¼ □ ◇	196	190	176	161	152
	1	1 ¼ ◁	226	220	204	186	176

TABLE 2.6 (cont.)

SHELL ID D_i, IN.	TUBE OD, d_o, IN.	TUBE PITCH, p. IN. AND LAYOUT	NUMBER OF TUBE PASSES				
			1 PASS	2 PASS	4 PASS	6 PASS	8 PASS
23 ¼	¾	15/16 ◁	495	478	430	420	408
	¾	1 □ ◇	375	362	325	318	309
	¾	1 ◁	438	416	382	372	360
	1	1 ¼ □ ◇	232	226	204	197	190
	1	1 ¼ ◁	268	262	236	228	220
25	¾	15/16 ◁	579	554	512	488	472
	¾	1 □ ◇	439	419	387	369	357
	¾	1 ◁	507	486	448	440	430
	1	1 ¼ □ ◇	273	261	237	226	214
	1	1 ¼ ◁	316	302	274	272	248
27	¾	15/16 ◁	676	648	602	584	560
	¾	1 □ ◇	512	490	456	442	424
	¾	1 ◁	592	574	536	516	496
	1	1 ¼ □ ◇	324	311	290	280	266
	1	1 ¼ ◁	375	360	336	324	308
29	¾	15/16 ◁	785	762	704	688	664
	¾	1 □ ◇	594	577	533	521	503
	¾	1 ◁	692	668	632	604	580
	1	1 ¼ □ ◇	372	360	337	329	318
	1	1 ¼ ◁	430	416	390	380	368
31	¾	15/16 ◁	909	878	814	792	776
	¾	1 □ ◇	688	665	616	600	587
	¾	1 ◁	796	774	732	708	680
	1	1 ¼ □ ◇	428	417	391	387	374
	1	1 ¼ ◁	495	482	452	448	432
33	¾	15/16 ◁	1035	1002	944	920	896
	¾	1 □ ◇	784	759	715	696	678
	¾	1 ◁	909	886	836	812	780
	1	1 ¼ □ ◇	501	479	450	463	422
	1	1 ¼ ◁	579	554	520	504	488
35	¾	15/16 ◁	1164	1132	1062	1036	1012
	¾	1 □ ◇	881	857	804	784	766
	¾	1 ◁	1023	1002	942	920	896
	1	1 ¼ □ ◇	558	538	507	498	484
	1	1 ¼ ◁	645	622	586	576	560
37	¾	15/16 ◁	1304	1270	1200	1168	1136
	¾	1 □ ◇	987	962	909	884	860
	¾	1 ◁	1155	1124	1058	1032	1004
	1	1 ¼ □ ◇	631	616	573	561	547
	1	1 ¼ ◁	729	712	662	648	632

TABLE 2.6 (cont.)

SHELL ID D_i, IN.	TUBE OD, d_o, IN.	TUBE PITCH, p. IN. AND LAYOUT	NUMBER OF TUBE PASSES				
			1 PASS	2 PASS	4 PASS	6 PASS	8 PASS
39	¾	15/16 ◁	1460	1422	1338	1320	1296
	¾	1 □ ◇	1106	1077	1013	1000	981
	¾	1 ◁	1277	1254	1194	1164	1120
	1	1 ¼ □ ◇	699	685	644	633	623
	1	1 ¼ ◁	808	792	744	732	720
42	¾	15/16 ◁	1703	1664	1578	1552	1528
	¾	1 □ ◇	1290	1260	1195	1175	1157
	¾	1 ◁	1503	1466	1404	1372	1344
	1	1 ¼ □ ◇	820	795	756	751	741
	1	1 ¼ ◁	947	918	874	868	856
45	¾	15/16 ◁	1960	1918	1830	1800	1776
	¾	1 □ ◇	1484	1453	1386	1363	1345
	¾	1 ◁	1726	1690	1622	1588	1552
	1	1 ¼ □ ◇	948	924	885	866	845
	1	1 ¼ ◁	1095	1068	1022	1000	976
48	¾	15/16 ◁	2242	2196	2106	2060	2032
	¾	1 □ ◇	1698	1663	1595	1560	1539
	¾	1 ◁	1964	1936	1870	1828	1792
	1	1 ¼ □ ◇	1074	1056	1018	994	963
	1	1 ¼ ◁	1241	1220	1176	1148	1112
54	¾	15/16 ◁	2861	2804	2682	2660	2632
	¾	1 □ ◇	2167	2124	2031	2015	1993
	¾	1 ◁	2519	2466	2380	2352	2320
	1	1 ¼ □ ◇	1365	1361	1307	1281	1264
	1	1 ¼ ◁	1577	1572	1510	1480	1460
60	¾	15/16 ◁	3527	3476	3360	3300	3264
	¾	1 □ ◇	2671	2633	2545	2500	2472
	¾	1 ◁	3095	3058	2954	2928	2896
	1	1 ¼ □ ◇	1700	1680	1629	1586	1548
	1	1 ¼ ◁	1964	1940	1882	1832	1788
66	¾	15/16 ◁	4292	4228	4088	4044	3967
	¾	1 □ ◇	3251	3203	3096	3063	3005
	¾	1 ◁	3769	3722	3618	3576	3508
	1	1 ¼ □ ◇	2069	2045	1976	1957	1920
	1	1 ¼ ◁	2390	2362	2282	2260	2217
72	¾	15/16 ◁	5116	5044	4902	4868	4776
	¾	1 □ ◇	3875	3821	3713	3687	3618
	¾	1 ◁	4502	4448	4324	4280	4199
	1	1 ¼ □ ◇	2477	2449	2378	2345	2300
	1	1 ¼ ◁	2861	2828	2746	2708	2656

TABLE 2.6 (cont.)

SHELL ID D_i, IN.	TUBE OD, d_o, IN.	TUBE PITCH, p. IN. AND LAYOUT	NUMBER OF TUBE PASSES				
			1 PASS	2 PASS	4 PASS	6 PASS	8 PASS
78	¾	15/16 ◁	6034	5964	5786	5740	5631
	¾	1 □ ◇	4571	4518	4383	4348	4265
	¾	1 ◁	5309	5252	5126	5068	4972
	1	1 ¼ □ ◇	2916	2878	2802	2785	2732
	1	1 ¼ ◁	3368	3324	3236	3216	3155
84	¾	15/16 ◁	7005	6934	6766	6680	65553
	¾	1 □ ◇	5306	5253	5125	5060	4965
	¾	1 ◁	6162	6108	5964	5900	5788
	1	1 ¼ □ ◇	3394	3361	3277	3235	3173
	1	1 ¼ ◁	3920	3882	3784	3736	3665
90	¾	15/16 ◁	8093	7998	7832	7708	7562
	¾	1 □ ◇	6131	6059	5933	5839	5729
	¾	1 ◁	7103	7040	6898	6800	6671
	1	1 ¼ □ ◇	3896	3859	3784	3748	3677
	1	1 ¼ ◁	4499	4456	4370	4328	4246
96	¾	15/16 ◁	9203	9114	8896	8844	8677
	¾	1 □ ◇	6971	6904	6739	6700	6573
	¾	1 ◁	8093	8026	7848	7796	7648
	1	1 ¼ □ ◇	4454	4420	4318	4274	4194
	1	1 ¼ ◁	5144	5104	4986	4936	4842
108	¾	15/16 ◁	11,696	11,618	11,336	11,268	11,055
	¾	1 □ ◇	8,860	8,801	8,587	8,536	8,375
	¾	1 ◁	10,260	10,206	9,992	9,940	9,752
	1	1 ¼ □ ◇	5,669	5,623	5,507	5,455	5,353
	1	1 ¼ ◁	6,546	6,494	6,360	6,300	6,181
120	¾	15/16 ◁	14,459	14,378	14,080	13,984	13,720
	¾	1 □ ◇	10,953	10,892	10,666	10,593	10,394
	¾	1 ◁	12,731	12,648	12,450	12,336	12,103
	1	1 ¼ □ ◇	7,029	6,961	6,815	6,765	6,637
	1	1 ¼ ◁	8,117	8,038	7,870	7,812	7,664

TABLE 2.7
TUBE PITCHES PARALLEL AND NORMAL TO FLOW

TUBE O.D. 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
¾ = 0.750	15/16 = 0.938	→ ◁	0.814	0.469
¾ = 0.750	1.000	→ □	1.000	1.000
¾ = 0.750	1.000	→ ◇	0.707	0.707
¾ = 0.750	1.000	→ ◁	0.866	0.500
1	1 ¼ = 1.250	→ □	1.250	1.250
1	1 ¼ = 1.250	→ ◇	0.884	0.884
1	1 ¼ = 1.250	→ ◁	1.082	0.625

NOMENCLATURE

A	Heat transfer area. A_I, inside tube heat transfer area; A_o, outside tube heat transfer area, including fins; A'_o, effective heat transfer area for use with Fig. 2.26; A^*, arbitrary convenient reference area for heat transfer; A_m, mean wall area for heat transfer, defined by eq. 2.3; A_{root}, heat transfer area of the bare tube remaining between the fins; A_{fin} is the total heat area of all of the fins on a tube.	$in.^2$ or $ft.^2$
c_p	Specific heat of fluid flowing. $c_{p,i}$ refers to the tube-side fluid; $C_{p,s}$ refers to the shell-side fluid.	$Btu/lb_m°F$
D_i	Shell inside diameter.	in. or ft.
$D_{0t\ell}$	Diameter of the outer tube limit.	in. or ft.
D_w	Equivalent diameter of the window.	in. or ft.
d	Tube diameter. d_o, outside diameter of tube, or diameter over the fins; d_i, inside tube diameter, d_r, root diameter of finned tube.	in. or ft.
F	Configuration correction factor for the logarithmic mean temperature difference.	dimensionless
F_c	Fraction of the total tubes that are in crossflow.	dimensionless
F_{sbp}	Fraction of total crossflow area that is available for bypass flow around tube bundle.	dimensionless
f	Friction factor. f_i is the friction factor inside tubes; f_s is the friction factor for crossflow or shell-side flow.	dimensionless
G	Mass velocity. G_m is the mass velocity of the shell-side fluid through the minimum free-flow area on the shell side, S_m.	$lb_m/hr\ ft^2$
g_c	Gravitational conversion constant.	$32.2\ lb_m ft/lb_f\ sec^2$ or $4.17 \times 10^8\ lb_m\ ft/lb_f$
H	Fin height. This is the distance from the root to the outer tip of the fin.	in. or ft.
h	Film heat transfer coefficient. h_i is the coefficient based on inside area for turbulent flow; $\bar{h}_i$ is the average coefficient for laminar flow inside a tube of length L; $(h_i)_T$ is the coefficient for flow in the transitional flow; h_o is the coefficient based on outside heat transfer area; $h_{o,i}$ is the coefficient for ideal crossflow on the shell-side of an exchanger.	$Btu/hr\ ft^2°F$
J_b	Correction factor on the shell-side heat transfer coefficient to account for	dimensionless

	by-pass flow effects.	
J_c	Correction factor on the shell-side heat transfer coefficient to account for baffle configuration effects.	dimensionless
J_ℓ	Correction factor on the shell-side heat transfer coefficient to account for baffle leakage effects.	dimensionless
J_r	Correction factor on the shell-side heat transfer coefficient to account for buildup of adverse temperature gradient.	dimensionless
J^*_r	Base correction factor on the shell-side heat transfer coefficient to account for buildup of adverse temperature gradient.	dimensionless
j	Colburn j factor for heat transfer. j_s is the value for crossflow or shell-side flow, defined by Eq. (2.15).	dimensionless
k	Thermal conductivity. k_i is the thermal conductivity of the fluid flowing inside the tube; k_ℓ is the thermal conductivity of a liquid; k_s is the thermal conductivity of shell side fluid; k_w is the thermal conductivity of tube wall material.	Btu/hr ft_2°F
L	Tube length effective for heat transfer.	in. or ft.
LMTD	Logarithmic mean temperature difference, defined for countercurrent flow by Eq. (2.11).	°F
ℓ_c	Baffle cut, from baffle tip to inside of shell.	in. or ft.
ℓ_s	Baffle spacing, face to face.	in. or ft.
m	Parameter in fin efficiency and fin resistance calculations. Defined in Eq. (2.7).	dimensionless
N_b	Number of baffles in exchanger.	dimensionless
N_c	Number of major restrictions crossed in a tube bank on one cross-flow section of a baffled shell and tube exchanger.	dimensionless
N_{cw}	Number of effective crossflow rows in each window or turnaround section of an exchanger.	dimensionless
N_f	Number of fins per unit length of tube.	$(in.)^{-1}$ or $(ft.)^{-1}$
N_{ss}	Number of pairs of sealing strips or equivalent obstructions to bypass flow encountered by the stream in one crossflow section.	dimensionless
N_t	Total numbers of tubes in the exchanger. For a U-tube bundle, N_t is equal to the number of holes in the tubesheet.	dimensionless

n	Number of tube-side passes in series.	dimensionless
P	Parameter in MTD calculations defined by Eq. (2.14).	dimensionless
ΔP	Pressure drop. $\Delta P_{b,i}$ is the pressure drop across one ideal (no leakage or bypass) cross-flow section in a baffled shell and tube exchanger. $\Delta P_{w,i}$ is the pressure drop through one ideal (no leakage) window section of a baffled shell and tube exchanger. ΔP_s is the pressure drop across an ideal tube bank. ΔP_i is the frictional pressure loss inside a tube. ΔP_{ent} is the pressure loss due to acceleration and friction at the entrance of a tube, and ΔP_{noz} is the pressure loss during flow through one nozzle.	lb_f/in^2 or lb_f/ft^2
Pr	Prandtl number defined by Eq. (2.21). Pr_i is the Prandtl number for the fluid flowing inside a tube; Pr_s is the Prandtl number of the fluid on the shell-side; Pr_ℓ refers to the liquid properties.	dimensionless
p	Tube pitch: distance between centers of nearest tubes in tube layout; p_n, tube pitch normal to flow; distance between centers of tubes in the same tube row normal to the flow; p_p tube pitch parallel to flow; distance between centers of adjacent tube rows in the direction of flow.	in. or ft.
Q	Heat duty. Q_T is the total heat load to be transferred in an exchanger.	Btu/hr
R	Parameter in MTD calculations defined by Eq. (2.13).	dimensionless
R_b	Correction factor for effect of bundle bypass on pressure drop.	dimensionless
R_f	Fouling resistance. R_{fi}, fouling resistance on inside tube surface; R_{fo}, fouling resistance on outside tube surface.	hr ft^2°F/Btu
R_{fin}	Fin resistance defined by Eq. (2.5).	hr ft^2°F/Btu
R_ℓ	Correction factor for effect of baffle leakage on pressure drop.	dimensionless
Re	Reynolds number; Re_i is the Reynolds number for flow inside round tubes, Eq. (2.19); $Re_{i,\ell}$ refers to the liquid Reynolds number inside a tube; Re_s is the Reynolds number for shell-side flow, defined by Eq. (2.17).	dimensionless
S	Cross-sectional area for flow. S_m is the minimum free flow area through one crossflow section; for a circular tube field, S_m is evaluated at or near the centerline. S_{sb} is the shell to baffle leakage area for a single baffle; S_{tb} is the tube to baffle leakage area for a single baffle. S_w is the area available for flow through a single window; S_{wg} is the flow area through a single window with no tubes; S_{wt} is the window area that is occupied by tubes.	ft^2
s	Fin spacing.	in. or ft.

T,t	Temperatures of the two streams in the beat exchanger. Usually, T refers to the hot stream and t to the cold stream, though T may also be used to refer to the shell-side stream and t to the tube-side stream, irrespective of their relative values. Subscripts 1 and 2 usually refer to the inlet and outlet values, respectively. $\overline{T}$ and $\bar{t}$ are the mean temperatures at which properties are evaluated, as defined by Fig. 2.18. $\overline{T}_w$ is the mean wall temperature, evaluated by means of Eq. (2.22).	°F
U	Overall heat transfer coefficient. U^*, overall coefficient referenced to an arbitrary heat transfer area A^*. U_i, overall coefficient based upon inside heat transfer are; U_o, overall coefficient based upon total outside heat transfer area.	Btu/hr ft^2°F
V	Velocity of flow. V_i is the average velocity of a fluid flowing inside a tube; $V_{i,\ell}$ is the velocity of a two-phase flow computed as if the entire flow were or liquid. V_{max} is the velocity of a fluid flowing across a tube bank at the minimum crossflow are, S_{min}.	ft/sec or ft/hr
W	Weight (or mass) flow rate. W_s denotes the shell-side flow rate; w_i denotes the tube-side flow rate.	lb/hr
x	Local quality of a two phase flow: ratio of mass of vapor to total mass present; x_i is inlet quality; x_o is the outlet quality.	dimensionless
Δx_w	Wall thickness of tube. Usually, in this Manual, the thickness of interest is the finned section.	in. or ft.
Y	Mean thickness of a single fin.	in.

GREEK

δ_{sb}	Diametral clearance between baffle and shell.	in. or ft.
ε	Roughness of tube/pipe inside surface.	ft.
θ	Baffle cut angle.	radians
μ	Fluid viscosity. μ_i refers to the tube-side fluid; μ_s refers to the shell-side fluid; μ_ℓ refers to the liquid viscosity and μ_v to the vapor viscosity in a twophase flow.	lb$_m$/ft hr
ρ	Fluid density. ρ_i refers to the tube-side fluid; ρ_s refers to the shell-side fluid; ρ_ℓ refers to the liquid density and ρ_v to the vapor density in a two phase flow.	lb$_m$/ft^3
Φ	Fin efficiency.	Dimensionless

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