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DESIGN OF FINNED-TUBE HEATERS AND COOLERS

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DESIGN OF FINNED TUBE HEATERS AND COOLERS

Finned tube shell and tube heat exchangers are used in a wide variety of applications in the petroleum industry. Typical uses are lube oil coolers, compressor interstage coolers, pre-heaters, bottoms coolers, stabilized gasoline coolers, and other applications. External finned tubes can be used to advantage if the shell side heat transfer resistance is appreciably greater than the tube side heat transfer resistance. The purpose of this article is to present a method of designing finned tube shell and tube heat exchangers based on the work of D. L. Katz and associates at the University of Michigan[1-3] and to compare the economics involved with bare tube exchangers.

The low finned tubes under consideration have plain ends that can be rolled into standard tube sheets in the usual manner. The diameter over the fins is slightly smaller than the diameter of the plain ends. Heat exchangers currently in use can often be retubed to advantage by retubing with a low finned tubing.

Figure 1 illustrates a finned tube shell and tube unit currently in use. Table 1 of Part I[4] presents typical dimensions of selected 19-fins-per-inch tubes used in commercial exchangers.

Six tube bundles were studies in a finned tube heat transfer investigation by Katz and associates[1,2]. Three of the bundles were tubed with 1/2, 5/8, and 3/4 inch 19-fins-per-inch tubes whereas the three remaining bundles were tubed with 1/2, 5/8, and 3/4 inch bare tubes. The 1/2 inch and 3/4 inch plain and finned tube units were studied in an eight inch shell and the two 5/8 inch units were housed in six inch shells. All bundles were 4 feet long. In all cases the shell circle design described by Tinker was used[6]. The tube sheet layouts are indicated in Figs. 3, 4, and 5 of Reference 1. Heat transfer measurements were made for water, lubricating oil and glycerine on the shell side. Figure 2 presents a comparison of the heat transferred by 3/4 inch finned and 3/4 inch plain tube bundles with hot oil on the shell side. The water flow rate on the tube side in both cases was 50,000 pounds per hour. An examination of Fig. 2 indicates that the finned tube unit transferred 50% more heat with an oil rate of 20,000 pounds per hour and 75% more heat with an oil rate of 45,000 pounds per hour.

The shell side pressure drops were also reported. Figure 3 presents the shell side pressure drop data for the 3/4 inch finned tube units reported in Fig. 2. An examination of Fig. 3 indicates that the shell side pressure drop of the plain tube unit with oil flowing at 20,000 pounds per hour was 64% greater than for the corresponding finned tube unit. With an oil flow rate of 45,000 pounds per hour the plain tube unit shell side pressure drop was also 64% greater than the corresponding plain tube bundle.

Design Relationships

The shell side heat transfer coefficients can be determined by use of the following relationships:[1]

For unbored shells:

$$\frac{h_o' De}{k} = 0.155 \left(\frac{DeG}{\mu}\right)^{0.6} \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (1)$$

For bored shells:

$$\frac{h_o' De}{k} = 0.175 \left(\frac{DeG}{\mu}\right)^{0.6} \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (2)$$

where:

h_o' = actual fin-side coefficient

De = the diameter of a bare tube having the same I.D. and volume of metal per foot length as the low finned tube, ft.

$$De = D_r + (D_o - D_r)\, Ny \qquad (3)$$

where:

D_r = root diameter, ft.

D_o = diameter over fins, ft.

N = number of fins per inch

y = fin thickness, inches

$$G = \sqrt{G_{bw} \cdot G_{cf}} \qquad (4)$$

where:

G_{bw} = mass flow rate through the baffle window, lb./sq. ft. - hr.

G_{cf} = mass flow rate on shell side pass the row of tubes nearest the centerline of the exchanger, lb./sq. ft. - hr.

Equations 1 or 2 can be used with Eq. 15 and Eq. 2 of reference 4 to obtain the overall coefficient U_o. The two alternate methods are illustrated in Part I of this series(4).

The shell side pressure drop consists of the pressure drop through the baffle windows plus the pressure drop in cross flow in the exchanger or:

$$\Delta P_t = \Delta P_{bw} + \Delta P_{cf} \qquad (5)$$

where:

ΔP_t = total pressure drop

ΔP_{bw} = pressure drop in baffle windows

ΔP_{cf} = pressure drop in cross flow

The baffle window pressure drop can be predicted by Donohue's equation[12]

$$\Delta P_{bw} = 0.01392\, V_w^2\, (\text{Sp. Gr.})\, n \qquad (6)$$

where:

ΔP_{bw} = lbs./sq. in.

V_w^2 = velocity in baffle window, ft./sec.

n = number of baffle windows

The shell side pressure drop for 19-fin-per-inch tubes can be predicted from Fig. 4. The equation is:

$$\Delta P_{cf} = \frac{N\, G_{cf}^2\, f}{(9.35 \times 10^8)\, \rho\, g_c \left(\frac{\mu}{\mu_w}\right)^{0.14}} \qquad (7)$$

where:

ΔP_{cf} = lbs. per sq. inch pressure drop

N = total number of rows of tubes crossed by shell side fluid (from baffle window centroid to baffle window centroid)

G_{cf} = see Eq. 4

f = friction factor from Fig. 4

ρ = fluid density, lbs./cu. ft.

g_c = 32.2

Liquid-Liquid Cooler Design Problem

A shell-and-tube heat exchanger is required to cool 20,200 gal./hr. of S.A.E.-40 lube oil from 200°F to 140°F. Treated cooling tower water is available at 90°F, and the maximum outlet temperature is limited to 110°F. The sizes of a low finned tube unit and a plain tube unit using 3/4 inch Admiralty tubing are required for comparison purposes.

A. Finned Tube Characteristics (see Table 1 of Part I[4])

No. fins/in.	= 19
d_o	= 0.737 in.
d_r	= 0.640 in.
d_i	= 0.510 in.
Root wall thickness	= 0.065 in.
Mean fin thickness, y	= 0.016 in.
Fin height	= 0.0485 in.
A_o	= 0.438 sq. ft./ft.
$\left(\frac{A_o}{A_i}\right)$	3.39
A_{cs}	= 0.001418 ft.^{2}/tube
D_e (for fluid flow)	= 0.0559 ft. (by Eq. 3)

B. Plain Tube Characteristics

3/4 in. O.D. - 16 BWG Admiralty

O.D.	= 0.750 in.
I.D.	= 0.620 in.
Wall thickness	= 0.065 in.
A_o	= 0.1963 sq. ft./ft.
$\left(\frac{A_o}{A_i}\right)$	= 1.210
A_{cs}	= 0.0021 sq. ft./tube

Finned Tube Cooler

Preliminary Design

Assume a 23-inch-I.D. shell containing one pass on the shell side and two passes on the tube side. The unit is to contain 420[7] tubes on a 15/16-in. triangular pitch. The length of the unit is to be determined.

A. Physical Properties of SAE 40 Oil[1]

	Temperature, °F			
	100	140	180	220
Density, Gm/cc	0.89	0.88	0.86	0.85
Viscosity, Centipoise	160	48	21	12
Thermal Conductivity, Btu/hr °F ft	0.082	0.081	0.081	0.080
Specific Heat, Btu/lb °F	0.46	0.48	0.50	0.52

B. Calculation of Average Shell Fluid Temperature[8]

Density of oil at 60°F = .905
A.P.I. of oil = 24.85[9]
Temperature range of oil in shell = 200-140 = 60°F
∴ C factor = 0.36

Δt_c = 140-90 = 50
Δt_h = 200-110 = 90

∴ $\Delta t_h/\Delta t_h$ = 50/90 = .555

∴ F factor = 0.42

∴ Average shell side temperature = 140 + .42 (200-140)
= 165.3°F
or 165°F will be used for design.

C. Heat Duty

$$C_p = 0.49, \text{ Sp. Gr.} = 0.86$$

$$Q = WC_p\Delta t = (20{,}200)(8.33)(0.86)(0.49)(60)$$

$$= 4{,}260{,}000 \text{ Btu/hr.}$$

D. Temperature Difference

$$\Delta T_{IM} = \frac{(200-110)-(140-90)}{\ln \frac{90}{50}} = 68°F$$

Correction to ΔT_{IM} (8)

$$P = \frac{110-90}{200-90} = 0.182$$

$$R = \frac{200-140}{20} = 3.0$$

$$F = 0.95$$

$$(\Delta T_{IM})_{corrected} = (68)(0.95) = 64.6°F.$$

E. Flow Areas

1. Baffle Flow Area

Assume a baffle spacing of 12 inches and a baffle cut of 30% on the diameter.

Total window area (10) $= (0.19817)(23)^2 = 104.8$ sq. in.

Shell cross-sectional area $= \dfrac{\pi(23)^2}{4} = 415$ sq. in.

Number of tubes in baffle window $= \dfrac{104.8}{415}(420) = 106$ tubes.

Projected area of tubes $= \dfrac{(0.737)^2\pi}{4} \cdot 106 = 45.1 \text{ in.}^2$

Free flow area in baffle window $= 104.8 - 45.1 = 59.7$ sq.in.
$= 0.415$ sq. ft.

2. Cross Flow Area

Shell area on centerline = (23)(12) = 276 sq. in.
No. of tubes on centerline = 23
Projected area of tubes = (23)(0.67)(12) = 185 sq. in.
Free flow area on centerline = 276-185 = 91 sq. in. = 0.632 sq. ft.

3. Geometric Mean Area

$$A_{GM} = \sqrt{(0.632)(0.415)} = 0.512 \text{ sq. ft.}$$

F. Water Flow Rate

$$W_W = \frac{4{,}260{,}000}{20} = 213{,}000 \text{ lb/hr.}$$

Cross-sectional area for water flow/pass $= \frac{420}{2}(0.001418) = 0.298$ ft.

$$\rho_w = 62 \text{ lbs. per cu. ft.}$$

$$V_t = \frac{213{,}000}{(62)(3600)(0.298)} = 3.22 \text{ ft./sec.}$$

G. Resistances to Heat Transfer

From Eq. 20 of Part I[4]:

$$\frac{1}{U_o} = \frac{1}{h_o} + r_o' + r_f + r_w \frac{A_o}{A_m} + \frac{A_o}{A_i} r_i + \frac{A_o}{A_i h_i}$$

1. Inside Coefficient[11]

$$h_i = 150(1 + 0.011\, t_w) \frac{V_t^{0.8}}{(d_i)^{0.2}}$$

at the average water temperature,

$$h_i = \frac{150(2.1)(2.54)}{(0.874)} = 915 \text{ Btu/hr-°F-ft}^2 \text{ (inside area).}$$

$$\frac{A_o}{A_i h_i} = \frac{3.39}{915} = 0.0037 \frac{\text{hr-°F-ft}^2 \text{ (outside area)}}{\text{Btu}}$$

2. Inside Fouling[8]

$$r_i = 0.001 \frac{\text{hr-°F-ft}^2 \text{ (inside area)}}{\text{Btu}}.$$

$$r_i \frac{A_o}{A_i} = 0.0039 \frac{\text{hr-°F-ft}^2 \text{ (outside area)}}{\text{Btu}}$$

3. Outside Fouling[8]

$$r_o' = 0.001 \frac{\text{hr-°F-ft}^2 \text{ (outside area)}}{\text{Btu}}$$

4. Fin Resistance

r_f is obtained from Table 2 of Part I of this series[4].

$$r_f = 0.00011$$

This value is valid over a wide range of coefficients as indicated in Part I.

5. Metal Resistance

$$r_w \frac{A_o}{A_m} = \frac{XA_o}{KA_m}$$

$$X = \frac{D_r - D_i}{2} = \frac{0.64 - 0.51}{(12)(2)} = 0.0054 \text{ ft.}$$

$$A_o = 0.438 \text{ sq. ft./ft.}$$

$$K = 65 \text{ Btu/hr-°F-ft}^{(11)}$$

$$A_m = \pi \frac{(D_r - D_i)}{\ln \frac{D_r}{D_i}} = 0.15 \text{ sq. ft./ft.}$$

$$r_w \frac{A_o}{A_m} = 0.00024 \frac{\text{hr-°F-ft}^2 \text{ (outside area)}}{\text{Btu}}$$

6. Liquid Film Resistance

The outside liquid film resistance will be calculated by Equation 1 with a constant of 0.155 for unbored shells.

$$\left(\frac{h_o' D_e}{k}\right) = 0.155 \left(\frac{D_e G}{\mu}\right)^{0.6} \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Rewriting Eq. 20 of Part I, (Reference 4):

$$\frac{1}{U_o} - \frac{1}{h_o'} = r_o' + r_f + r_w \frac{A_o}{A_m} + r_i \frac{A_o}{A_i} + \frac{1}{h_i} \frac{A_o}{A_i}$$

substituting:

$$\frac{1}{U_o} - \frac{1}{h_o} = 0.001 + 0.00011 + 0.00024 + 0.00339 + 0.00037 = 0.00844$$

From the physical properties of the oil at 165°F:

C_p = 0.49 Btu/lb.

μ = 29 centipoise

k = 0.081 Btu/hr °F ft

ρ = 0.865 grams/cc.

$$\left(\frac{C_p \mu}{k}\right)^{1/3} = \left[\frac{(0.49)(29)(2.42)}{0.081}\right]^{1/3} = 7.5$$

$$G_m = \frac{(20,200)(8.33)(.865)}{0.512} = 284,000$$

$$Re = \left(\frac{D_e\ G_m}{\mu}\right)$$

$$= \frac{(0.67)(284,000)}{(12)(29)(2.42)}$$

$$= 226$$

$$Re^{0.6} = 25.7$$

Substituting into Eq. 1:

$$\left(\frac{h_o' D_e}{k}\right) = (0.155)(25.7)(7.5)\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$= 29.8\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\therefore h_o' = \frac{(29.8)(0.081)(12)}{0.67}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$= 43.3\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

The value of μ_w is a function of the temperature drop across the outside film. The temperature drop in turn is a function of the outside film resistance. Therefore a trial and error procedure is required to satisfy the above equation.

$$t_w = t_{oil} - (\Delta t)_{film}$$

$$(\Delta t)_{film} = \frac{U_o}{h_o'} (\Delta T_{LM})_{corrected}$$

First Trial: assume $h_o' = 38$

$$\frac{1}{U_o} = \frac{1}{h_o'} + 0.00844$$

$$\therefore \frac{1}{U_o} = \frac{1}{38} + 0.00844 = 0.03474$$

$$U_o = 28.8$$

$$\Delta t_{film} = \left(\frac{U_o}{h_o'}\right)(\Delta T_{LM})_{corr.} = \left(\frac{28.8}{38}\right)(64.4) = 49.0°F$$

$$\therefore t_w = 165° - 49° = 116°F$$

$$\therefore \mu = 91 \text{ centipoises}$$

$$\left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{29}{91}\right)^{0.14} = 0.854$$

$$h_o' = (43.3)(0.854) = 37 \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ (outside area}}$$

which checks closely the assumed value of 38.

H. Determination of Outside Area

$$Q = U_o A_o \Delta T_{LM}$$

$$A_o = \frac{(4,260,000)}{(28.8)(64.6)} = 2290 \text{ sq. ft.,}$$

Total length of tubing required

$$= \frac{2290 \text{ sq.ft.}}{0.438 \text{ sq.ft./ft.}} = 5220 \text{ ft.}$$

$$\text{length/tube} = \frac{522}{420} = 12.4 \text{ ft.}$$

Using a 14-ft. unit, the excess area will be:

excess area = 12.9%.

I. Pressure Drop on Tube Side

Reference is made to the friction factor plot on page 836 of Kern.[7

1. Friction Pressure Drop

$$Re = \frac{DG}{\mu} = \frac{(0.51)(3.22)(62)}{(12)(0.69)(6.72\cdot 10^{-4})} = 18,300$$

$$f = 0.00023$$

$$\Delta P_f = \frac{f\ G^2\ LN}{5.22\cdot 10^{10} D\ Sp\ Gr}$$

$$= \frac{(0.00023)(7.15\cdot 10^5)^2(14)(2)(12)}{(0.51)(5.22\cdot 10^{10})}$$

$$= 1.48 \text{ psi.}$$

2. Header Losses[7]

$$\Delta P_h = \frac{4N\ V^2 \rho}{2g_c} = \frac{4V^2\ (62.4)}{144\ g_c} = 0.56$$

3. Total Pressure Drop-Tube-Side

$$\Delta P_T = (-\Delta P_f) + (-\Delta P_h)$$

$$= 1.48 + 0.56 = 2.04 \text{ psi.}$$

J. Pressure Drop on Shell Side

The shell-side pressure drop for the baffle window losses for the finned-tube unit will be computed from Donohue's correlation, Equation 12A[12].

1. Baffle Window Losses

$$\Delta P_{bw} = 0.01392\ V_w^2\ (\text{Sp Gr})N$$

$$\text{Sp Gr} = 0.86$$

$$n = 13 \text{ baffle windows}$$

$$A_w = 0.415 \text{ sq. ft.}$$

$$V_w = \frac{145{,}500}{(0.86)(62)(3600)(0.415)} = 1.81 \text{ ft./sec.}$$

$$V_w^2 = 3.28$$

$$\Delta P_{bw} = 0.01392\ (3.28)(0.865)(13)$$

$$= 0.512 \text{ psi.}$$

2. Cross Flow

Reference is made to the friction factor plot of Fig. 4. From the definition of the friction factor,

$$(\Delta P_{cf}) = \frac{N\ G_c^2 f}{9.35\cdot 10^8\ \rho\ g_c \left(\frac{\mu}{\mu_w}\right)^{0.14}}$$

Determination of number of rows of tubes cross from the centroid of one baffle window to the centroid of the adjacent baffle.

The distance from the center of the shell to the centroid of the baffle window segment[13]:

$$X_o = \frac{2/3 \ r \sin^3 c}{\text{rad. } c - \cos c \sin c}$$

where c is the one half the included angle of the segment.

$$\cos c = \frac{11.5 - (0.30)(23.0)}{11.5} = \frac{4.6}{11.5} = 0.40$$

$$\sin c = 0.916; \ \sin^3 c = 0.769 \text{ and rad } c = 1.16.$$

Substituting:

$$X_o = \frac{2/3 \ (11.5)(0.769)}{1.16 - 0.366} = 7.43 \text{ inches}$$

Therefore centroid to centroid distance $= 2X_o = 14.86$ inches

$$\text{rows per cross flow} = \frac{14.86}{(15/16)(.866)} = 18.3$$

$$N = 18.3\ (14) = 256 \text{ rows of tubes.}$$

$$G_{cf} = \frac{145{,}000}{0.632} = 2.30 \cdot 10^5$$

$$Re = \frac{(2.30 \cdot 10^5)(0.67)}{(12)(29)(2.42)} = 183.0$$

$$\frac{P}{D} = \frac{15/16}{3/4} = 1.25$$

$$f = 0.290$$

$$\rho = (0.865)(62.4) = 54 \text{ lbs. per cu. ft.}$$

$$g_c = 32.2$$

$$\left(\frac{\mu}{\mu_w}\right)^{0.14} = 0.854$$

$$\therefore \Delta P_{cf} = \frac{(256)(0.290)(2.30\cdot 10^5)^2}{(9.35\cdot 10^8)(0.865)(62.4)(32.2)(0.854)}$$

$$= 2.84 \text{ psi}$$

3. Total Pressure Drop

$$\Delta P_T = (\Delta P_{cf}) + (\Delta P_{bw})$$

$$= 2.84 + 0.512 = 3.35 \text{ psi.}$$

Plain Tube Design

Preliminary Layout

Assume a 31-inch-ID shell containing one pass on the shell side and six passes on the tube side. The unit is to contain 722 tubes[7] on a 15/16-in. triangular pitch. The length of the unit is to be determined.

A. Average Shell Side Temperature = 165°F.

B. Heat Duty

$$Q = WC_p\Delta t = (20{,}200)(8.33)(0.86)(60)(0.49)$$

$$= 4{,}260{,}000 \text{ Btu/hr}$$

C. Temperature Difference

The log mean temperature difference is the same as the finned tube unit.

$$\Delta T_{lM} = 64.6°F$$

D. Flow Areas

1. Baffle Flow Area

Assume a baffle spacing of 12 in. and a baffle cut of 30% on the shell diameter.

Total window area[10] = $(0.19817)(31)^2 \doteq 191$ sq. in.

$$\text{Shell cross-sectional area} = \frac{\pi(31)^2}{4} = 755 \text{ sq. in.}$$

$$\text{No. of tubes in baffle window} = \frac{191}{755}(722) = 183$$

$$\text{Projected area of tubes in baffle window} = \frac{\pi}{4}(0.75)^2(183)$$

$$= 80.8 \text{ sq. in.}$$

$$\text{Free flow area in baffle window} = 191-80.8 = 110.2 \text{ sq. in.}$$

$$= 0.765 \text{ sq. ft.}$$

2. Cross Flow Area

Tubes on centerline = 32
Projected area of tubes = (0.75)(32)(12) = 288 sq. in.
Shell free flow area on centerline = 372 sq. in.
Free flow area = 372-288 = 84 sq. in. = 0.584 sq. ft.

3. Geometric Mean Area

$$A_{GM} = \sqrt{(0.584)(0.765)} = 0.67 \text{ sq. ft.}$$

E.. Water Flow Rate

$$\text{Mass rate} = \frac{4{,}260{,}000}{20} = 213{,}000 \text{ lb./hr.}$$

$$\text{Cross-sectional area for water flow/pass} = (0.0021)\frac{722}{6}$$

$$= 0.253 \text{ sq. ft.}$$

$$V_t = \frac{213{,}000}{(3600)(62)(0.253)} = 3.79 \text{ ft./sec.}$$

F. Resistances to Heat Transfer

$$\frac{1}{U_o} = \frac{1}{h_o} + r_o + r_w + \frac{A_o}{A_i} r_i + \frac{A_o}{A_i h_i}$$

1. Inside Coefficient[11]

$$h_i = \frac{150(1 + 0.011t_w)\ V_t^{0.8}}{(d_i)^{0.2}}$$

At the average water temperature

$$\frac{A_o}{A_i h_i} = \frac{1.21}{1008} = 0.00120\ \frac{\text{hr-°F-ft (outside)}}{\text{Btu}}$$

2. Inside Fouling[8]

$$r_i = 0.001$$

$$\frac{A_o}{A_i} r_i = 0.00121$$

3. Outside Fouling[8]

$$r_o = 0.001$$

4. Metal Resistance

$$r_w = \frac{XA_o}{KA_m}$$

$$X = 0.065/12 \text{ ft.}$$

$$A_o = 0.1963 \text{ ft}^2/\text{ft.}$$

$K = 65$ Btu/hr-°F-ft.[11]

$$A_m = 0.1792 \text{ ft.}^2/\text{ft.}$$

$$r_w = \frac{(0.065)(0.1963)}{(12)(65)(0.1792)} = 0.000083$$

5. Liquid Film Resistance

$$\frac{1}{U_o} - \frac{1}{h_o} = r_o + r_w + \frac{A_o}{A_i} r_i + \frac{A_o}{A_i h_i}$$

$$= 0.001 + 0.000083 + 0.00121 + 0.0012 = 0.00349$$

The outside coefficient will be determined by Equation (4b) of the Katz-Williams reference[1] using a constant of 0.22 for unbored shells.

$$\left(\frac{h_o D_o}{k}\right) = 0.22 \left(\frac{D_o G_m}{\mu}\right)^{0.6} \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

As calculated earlier:

$$\left(\frac{C_p \mu}{k}\right)^{1/3} = 7.5$$

$$G_m = \frac{(20{,}200)(8.33)(0.865)}{0.67} = 217{,}000$$

$$Re = \frac{(217{,}000)(0.75)}{(12)(29)(2.42)} = 193$$

$$Re^{0.6} = 23.5$$

$$\left(\frac{h_o D_o}{k}\right) = (0.22)(23.5)(7.5)\left(\frac{\mu}{\mu_w}\right)^{0.14} = 38.8\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$\therefore h_o = \frac{38.8(0.081)(12)}{(0.75)}\left(\frac{\mu}{\mu_w}\right)^{0.14} = 50.3\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Trial and error on h_o

(1) Assume $h_o = 43$

(2) $\dfrac{1}{U_o} = \dfrac{1}{43} + .0035 = 0.0267$

(3) $U_o = 37.4$

(4) $\Delta t_{film} = \left(\dfrac{37.4}{43}\right)(64.4) = 56.2°F$

(5) $t_w = (165°F - 56.2) = 109°F$

(6) $\mu_w = 105$ centipoises

(7) $\therefore \left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{29}{100}\right)^{0.14} = 0.84$

(8) $\therefore h_o = (50.3)(0.84) = 42.3$ (which checks the assumed value of 43).

G. Determination of Required Area

$$Q = U_o A_o \Delta T_{LM}$$

$$A_o = \frac{4,260,000}{(37.4)(64.6)} = 1765 \text{ ft.}^2$$

Length of tubing required

$$L = \frac{1765}{0.1963} = 8990 \text{ ft.}$$

$$\text{Length/tube} = \frac{8990}{722} = 12.4$$

Using a 14-ft.-long unit, the excess area will be the same as for the finned tube unit.

excess area = 12.9%

H. Pressure Drop on Tube-Side

Reference is made to the friction factor plot on page 836 of Kern.(7)

1. Friction Pressure Drop

$$Re = \frac{DG}{\mu} = \frac{(0.62)(3.79)(62)}{(12)(0.69)(6.72\cdot 10^{-4})} = 26,200$$

$$f = 0.00021$$

$$-\Delta P_f = \frac{f\ G^2\ LN}{5.22\cdot 10^{10}\ D\cdot Sp\ Gr}$$

$$-\Delta P_f = \frac{(0.00021)(8.45\cdot 10^5)^2(14)(6)(12)}{(5.22\cdot 10^{10})(0.62)}$$

$$= 4.67 \text{ psi.}$$

2. Header Losses[7]

$$-\Delta P_h = \frac{4NV^2\rho}{2g_c} = \frac{12V^2\ 62.4}{g_c\ 144} = 2.32\text{ psi.}$$

3. Total Tube-side Pressure Drop

$$-\Delta P_T = 2.32 + 4.67 = 6.99\text{ psi.}$$

I. Shell-Side Pressure Drop

The shell-side pressure drop is considered to be the sum of the pressure drop of the fluid through the baffle window and the pressure drop of the fluid in cross flow through the tube bundle.

1. Baffle Pressure Drop

From Donohue's correlation, Equation 12A,[12]

$$-\Delta P_{bw} = 0.01392\ V_w^2\ (\text{Sp Gr})N$$

$$\text{Sp Gr} = 0.86$$

$$n = 13\text{ baffles}$$

$$A_w = 0.765\text{ ft.}^2$$

$$V_w = \frac{(20{,}200)(8.33)(0.86)}{(62.4)(0.86)(0.765)(3600)} = 0.982\text{ ft./sec.}$$

$$V_w^2 = 0.965$$

$$-\Delta P_{bw} = (0.86)(0.01392)(0.965)13 = 0.151\text{ psi.}$$

2. Cross-flow Pressure Drop

Reference is made to the friction factor plot of Donohue[12] (Fig. 26).

$$-\Delta P_{cf} = \frac{N\ G_c^2\ f}{(9.35\ 10^8)\ \rho\ g_c\left(\frac{\mu}{\mu_w}\right)^{0.14}}$$

$$\text{Re} = \frac{DG_{cf}}{\mu}$$

$$A_{cf} = 0.584 \text{ sq. ft.}$$

$$G_{cf} = \frac{145{,}500}{0.584} = 250{,}000 \text{ lb./hr.-ft.}^2$$

$$Re = \frac{(0.75)(250{,}000)}{(12)(29)(2.42)} = 222$$

$$P/D == \frac{15/16}{3/4} = 1.25$$

From Donohue's Fig. 26:

$$f = 0.37$$

For N, the same method used for finned tubes applies,

$$\therefore X_o = 7.43\left(\frac{31}{23}\right) = 10$$

$$\therefore N = \frac{(10)(2)(16)(14)}{(.866)(15)} = 299$$

$$-\Delta P_c = \frac{(299)(250{,}000)^2(0.37)}{(9.35)10^8)(62.4)(0.86)(32.2)(0.84)}$$

$$= 5.1 \text{ psi.}$$

3. Total Pressure Drop

$$-\Delta P_T = (-\Delta P_{bw}) + (-\Delta P_{cf})$$

$$-\Delta P_T = 0.151 + 5.10 = \underline{\underline{5.25 \text{ psi.}}}$$

Summary

Shell Side	Finned	Plain
ID, in.	23	31
No. of shell passes	1	1
Length, ft.	14	14
No. of baffles	13	13
Baffle spacing, in.	12	12
Baffle cut, % on diameter	30	30
Geometric mean area, sq. ft.	0.512	0.670
Inlet oil temperature, °F	200	200
Outlet oil temperature, °F	140	140
Lb/hr, oil	145,000	145,000
Pressure drop, psi	3.35	5.25
Tube Side		
No. of tubes	420	722
No. of tube passes	2	6
Water flow rate, lb/hr	213,000	213,000
Water velocity, ft/sec	3.22	3.79
Water temperature in, °F	90	90
Water temperature out, °F	110	110
Tube pitch, in.	15/16	15/16
Tube arrangement	60° triangular	60° triangular
Pressure drop, psi	2.04	6.99
Heat Transfer		
Heat duty, Btu/hr	4,260,000	4,260,000
Overall coefficient, U_o	28.8	37.4
Log mean temperature difference	64.6	64.6
Shell side coefficient	38	42.3
Tube side coefficient	915	1008
Shell side fouling factor	0.001	0.001
Tube side fouling factor	0.001	0.001
External heat transfer area	2,580	1,990
% excess area	12.9	12.9

$$\frac{(U_L)_{finned}}{(U_L)_{plain}} = \frac{(28.8)(0.438)}{(37.4)(0.1963)} = 1.72 = 72\% \text{ greater for finned tube.}$$

Cost: Cost data for 16 foot long finned tube and 16 foot long bare tube heat exchangers has appeared in the literature(14):

Cost of 16 ft. bare tube unit of 1,990 sq. ft. = $11,500

Cost of 16 ft. finned tube unit of 2,580 sq. ft. = $9,400

∴ Savings = $2,100.

TABLE 1

TYPICAL DIMENSIONS OF SELECTED 16 GAUGE WALL,
19-FIN-PER-INCH, FINNED TUBING (Courtesy of Wolverine Tube)*

		Plain Ends		Finned Section				
Nominal O.D.(inches)	Alloy	O.D. inches	Wall Thickness inches	O.D. of Fins inches	Root Dia. inches	Wall Thickness inches	A_o sq.ft./ft.	A_o/A_i
5/8	Copper	0.625	0.071	0.614	0.499	0.065	0.405	4.19
5/8	Cu-Ni	0.625	0.072	0.614	0.499	0.065	0.405	4.19
5/8	Admiralty	0.625	0.077	0.612	0.515	0.065	0.386	3.80
5/8	Nickel	0.625	0.077	0.612	0.515	0.065	0.386	3.80
3/4	Copper	0.750	0.077	0.739	0.624	0.065	0.496	3.84
3/4	Cu-Ni	0.750	0.077	0.739	0.624	0.065	0.496	3.84
3/4	Admiralty	0.750	0.082	0.737	0.640	0.065	0.438	3.39
3/4	Nickel	0.750	0.082	0.737	0.640	0.065	0.438	3.39
7/8	Copper	0.875	0.083	0.864	0.749	0.065	0.588	3.63
7/8	Cu-Ni	0.875	0.082	0.864	0.749	0.065	0.588	3.63
7/8	Admiralty	0.875	0.087	0.862	0.765	0.065	0.520	3.12
7/8	Nickel	0.875	0.087	0.862	0.765	0.065	0.520	3.12
1	Copper	1.000	0.083	0.989	0.874	0.065	0.678	3.48
1	Cu-Ni	1.000	0.082	0.989	0.874	0.065	0.678	3.48
1	Admiralty	1.000	0.082	0.987	0.890	0.065	0.598	3.01
1	Nickel	1.000	0.082	0.987	0.890	0.065	0.598	3.01

*Mean fin thickness is about 0.015 inches.

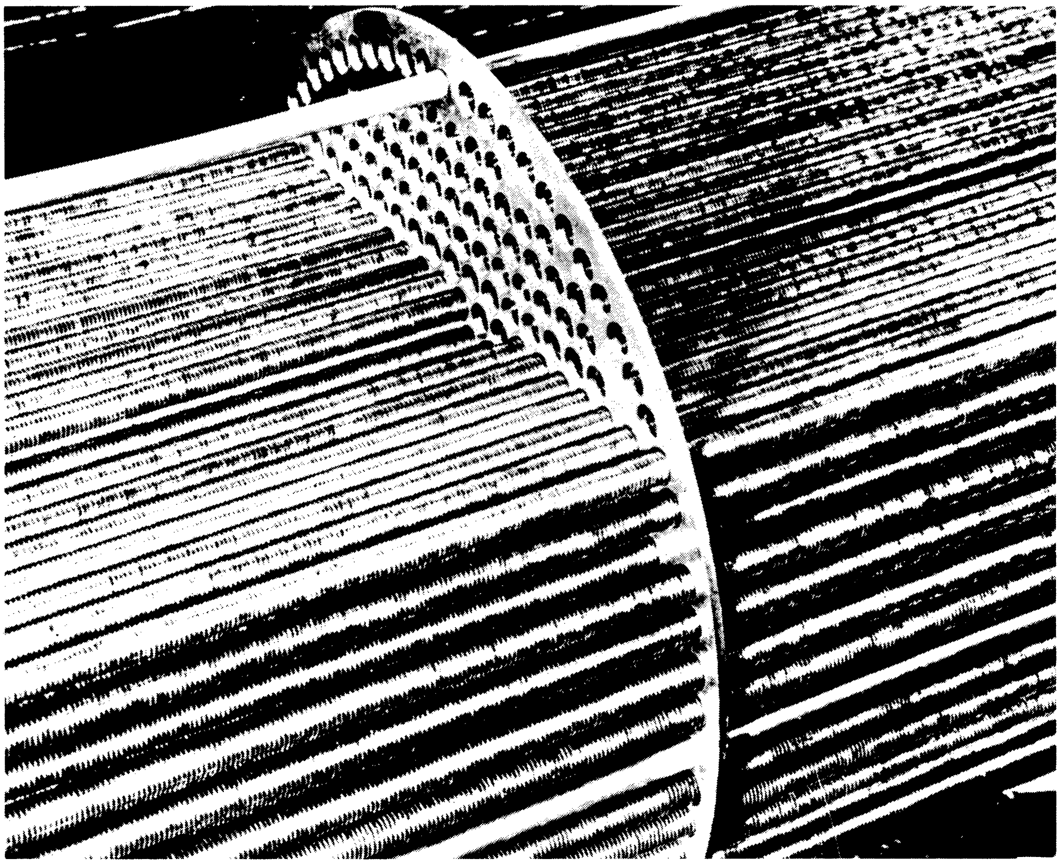

Figure 1. Finned Tube Bundle in Process of Fabrication

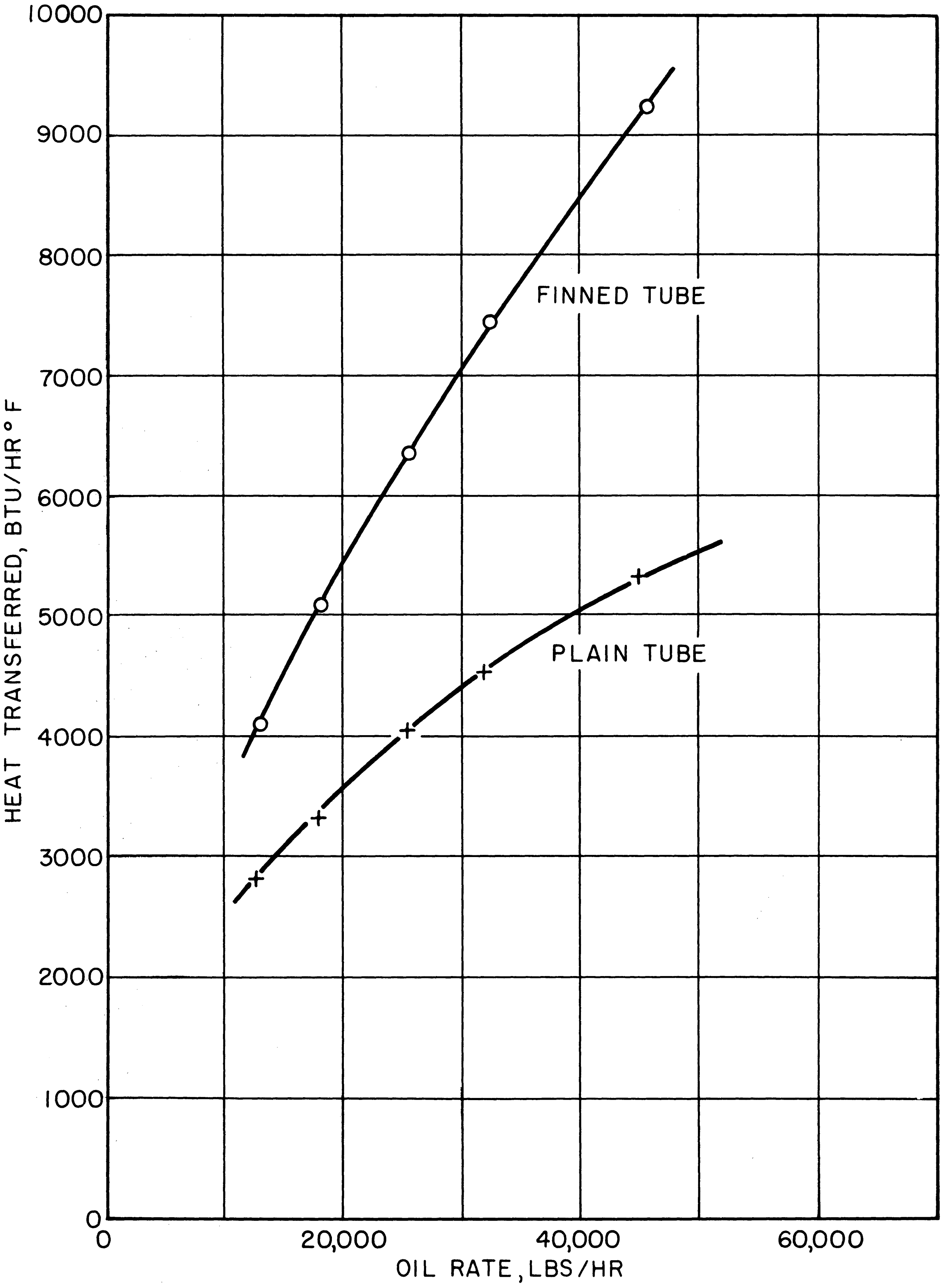

Figure 2. Comparison of Heat Transfer Rates for 3/4 inch Finned and Plain Tubes in 8 inch Shells.

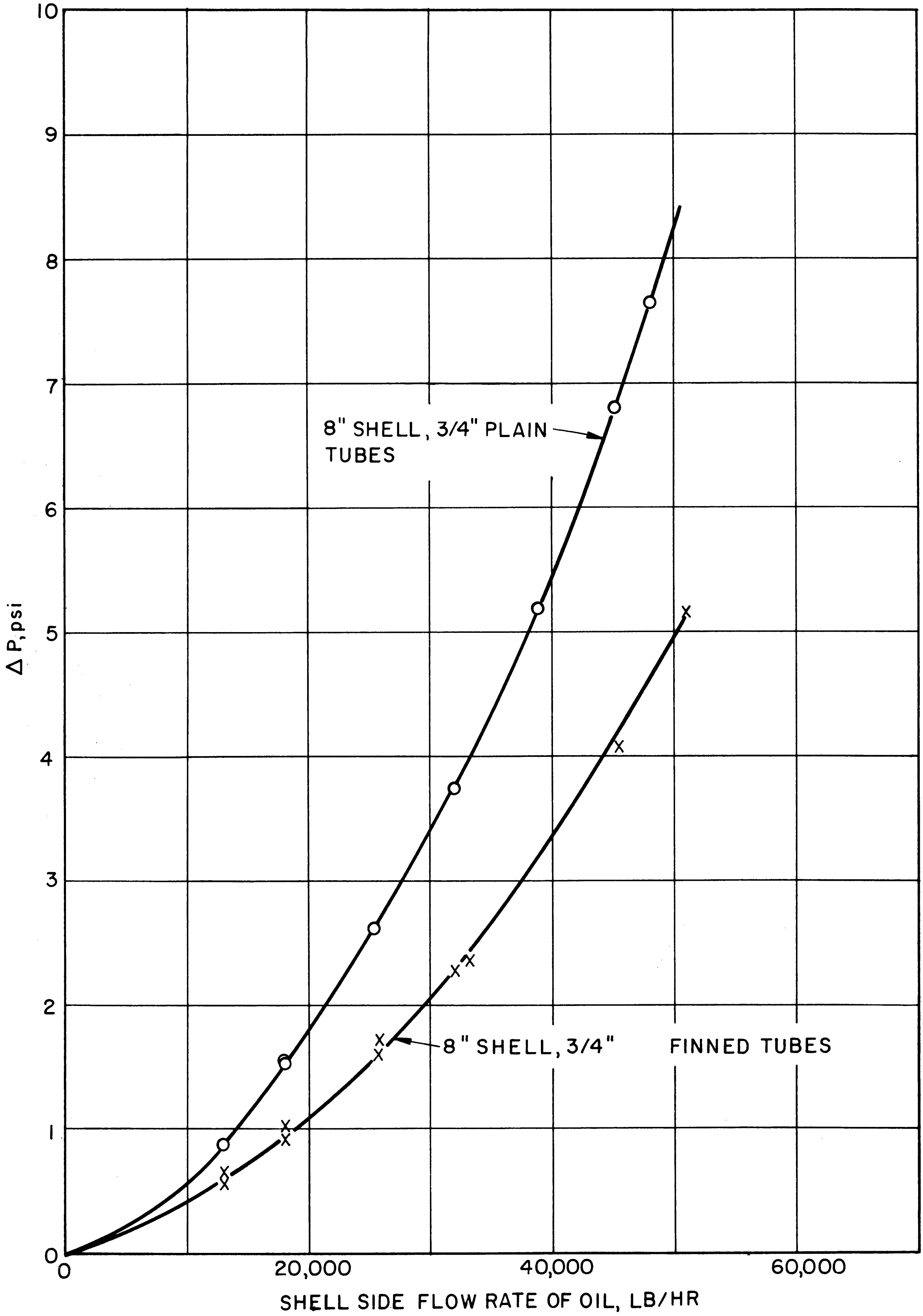

Figure 3. Comparison of Shell Side Pressure Drop for 3/4 inch Plain and Finned Tubes.

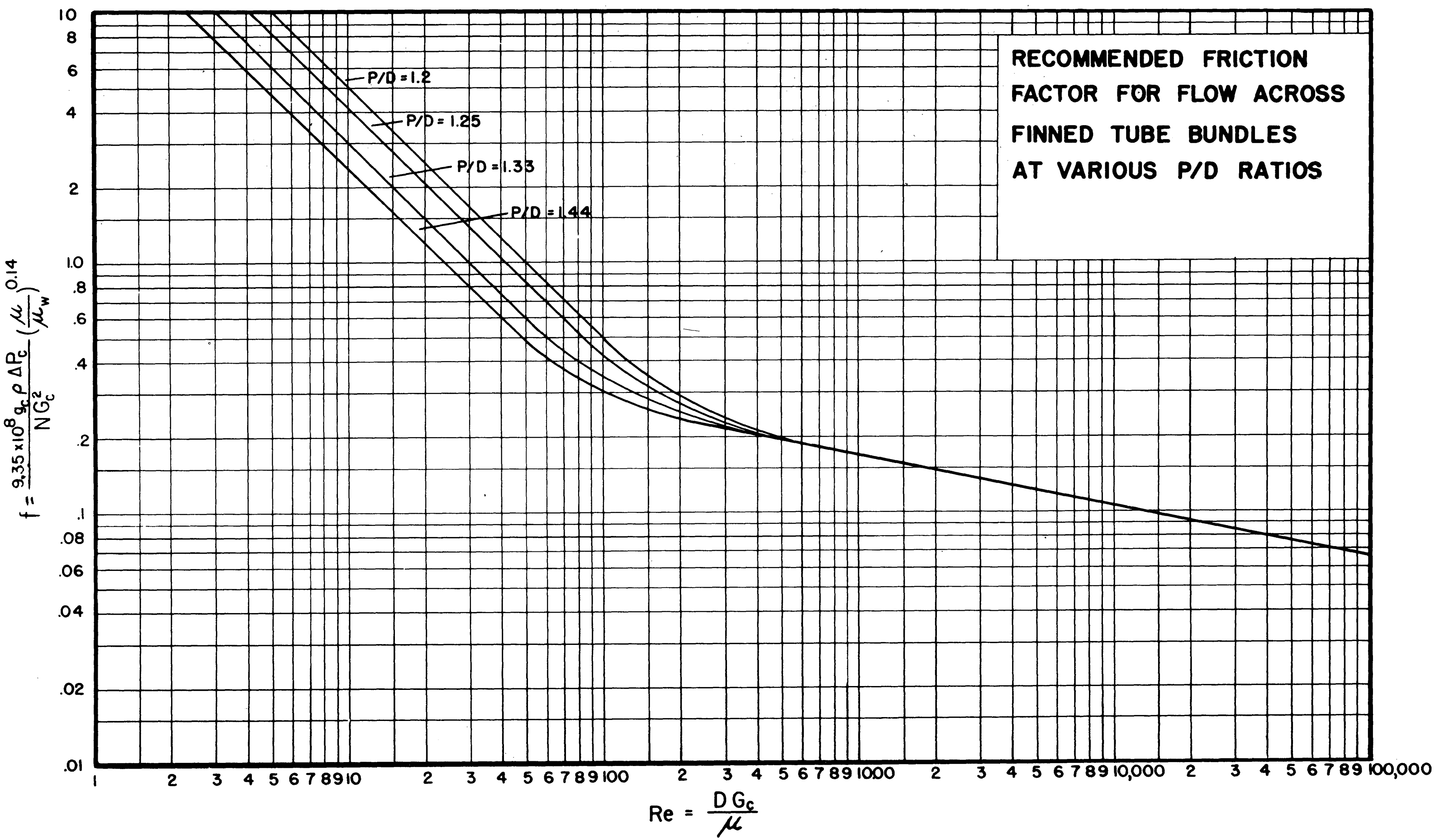

Figure 4. Recommended Friction Factors for Flow Across Finned Tube Bundles.

REFERENCES

1. Williams, R. B. and Katz, D. L., Trans. ASME, 74, pp. 1307-1320, 1952.
2. Williams, R. B. and Katz, D. L., Petroleum Refiner, Vol. 33, No. 3, pp. 145-149, 1954.
3. Katz, D. L., Knudsen, J. G., Balekjian, G., and Grover, S. S., Petroleum Refiner, Vol. 33, No. 8, pp. 123-125, 1954.
4. Young, E. H. and Ward, D. J., Fundamentals of Finned Tube Heat Transfer, IP-172, University of Michigan, August 1956.
5. Young, E. H. and Ward, D. J., Design of Finned Tube Condensers, IP-184, University of Michigan, October 1956.
6. Tinker, T., "General Discussion on Heat Transfer," Institution of Mechanical Engineering, London, England, 1951, pp. 89-116.
7. Kern, D. Q., Process Heat Transfer, McGraw-Hill, 1950.
8. "Standards of Tubular Exchanger Manufacturers Association," 3rd Edition, TEMA, New York, 1952.
9. ASTM - IP, Petroleum Measurement Tables, 1953, p. 171.
10. Perry, J. H., Chemical Engineers' Handbook, McGraw-Hill, 3rd Edition, 1950, p. 32.
11. McAdams, W. H., Heat Transmission, McGraw-Hill, 3rd Edition, 1954.
12. Donahue, D. A., "Heat Transfer and Pressure Drop in Heat Exchangers," IEC, 41, 1949.
13. Marks, L. S., Mechanical Engineers' Handbook, 4th Edition, McGraw-Hill, 1941, p. 208.
14. Kern, D. Q. and Associates, "Compare Exchanger Costs Quickly," Petroleum Refiner, Vol. 34, No. 8, 1956.