CHAPTER 3

Heat Exchanger Design

This chapter will provide the framework on how to design a Double Pipe and Shell and Tube type Heat Exchanger. Detailed theoretical discussion on the principles of heat transfer was consciously left to readers to read standard textbook in heat transfer. A brief review of every topic that would be needed in the understanding of heat exchanger design has been provided to assure continuity and comprehension.

A double-pipe heat exchanger is consists of two concentric pipes with one fluid flowing through the inner pipe while the other fluid flowing through the annular space .While a shell-and-tube heat exchanger consists of tube bundles enclosed in a cylindrical shell with one fluid flowing through the tubes and the other fluid flowing outside of the tube bundles enclosed in the shell.

The three fundamental mode of heat transfer are (1) conduction, (2) convection and (3) radiation. Conduction occurs when the transfer of heat is through molecular action. There is a need of physical contact but without significant displacement or movement of molecules. It could take place within a solid or non-moving fluid medium. On the other hand, convection requires the movement or mixing of fluids. It takes place between a solid surface and a contacting moving fluid that are at different temperatures. Radiation is the transfer of heat in the form of electromagnetic waves. This mode of transfer does not require the presence of an intervening material or physical contact .

HEAT EXCHANGER DESIGN EQUATIONS

Rate of heat Transfer

Similar to other transport phenomena, the rate of heat transfer is also expressed in terms of driving force and resistance. The general rate expression for heat transfer is conveniently expressed in terms of overall heat-transfer coefficient, *U*, which is defined in an analogous manner to Newton's law for convective heat transfer. For Double Pipe heat exchanger, the rate of heat transfer may be computed using Equation 3-1.

$$q = UA\Delta T \ln \qquad \textit{eq 3-1}$$

However, to correct mixed flow in a shell and tube type of heat exchangers, a geometric correction factor, Y, has to be added to account for the flow deviation from double Pipe heat exchangers**.** Thus,

$$q = UAY\Delta T \ln \qquad \textit{eq 3-2}$$

where:

A = total heat-transfer area in the heat exchanger

$\Delta T_1, \Delta T_2$ = temperature difference between hot and cold streams at heat exchanger terminals

$\Delta \overline{T} \ln$ = logarithmic mean temperature difference (LMTD)

$$\Delta \bar{T}_{\ln} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \qquad \textit{eq 3-3}$$

Y is correlated to two dimensionless temperature ratios namely the heat capacity Z, and the effectiveness of the heat exchanger X. Values of the geometric correction factor for different number of tube to shell passes may be derived from **Figure 3-1** and **3-2**.

$$\begin{cases} Z = \dfrac{T_{A1} - T_{A2}}{T_{B2} - T_{B}1} \\ \text{and} \\ X = \dfrac{T_{B2} - T_{B1}}{T_{A1} - T_{B1}} \end{cases} \qquad \textit{eq 3-4}$$

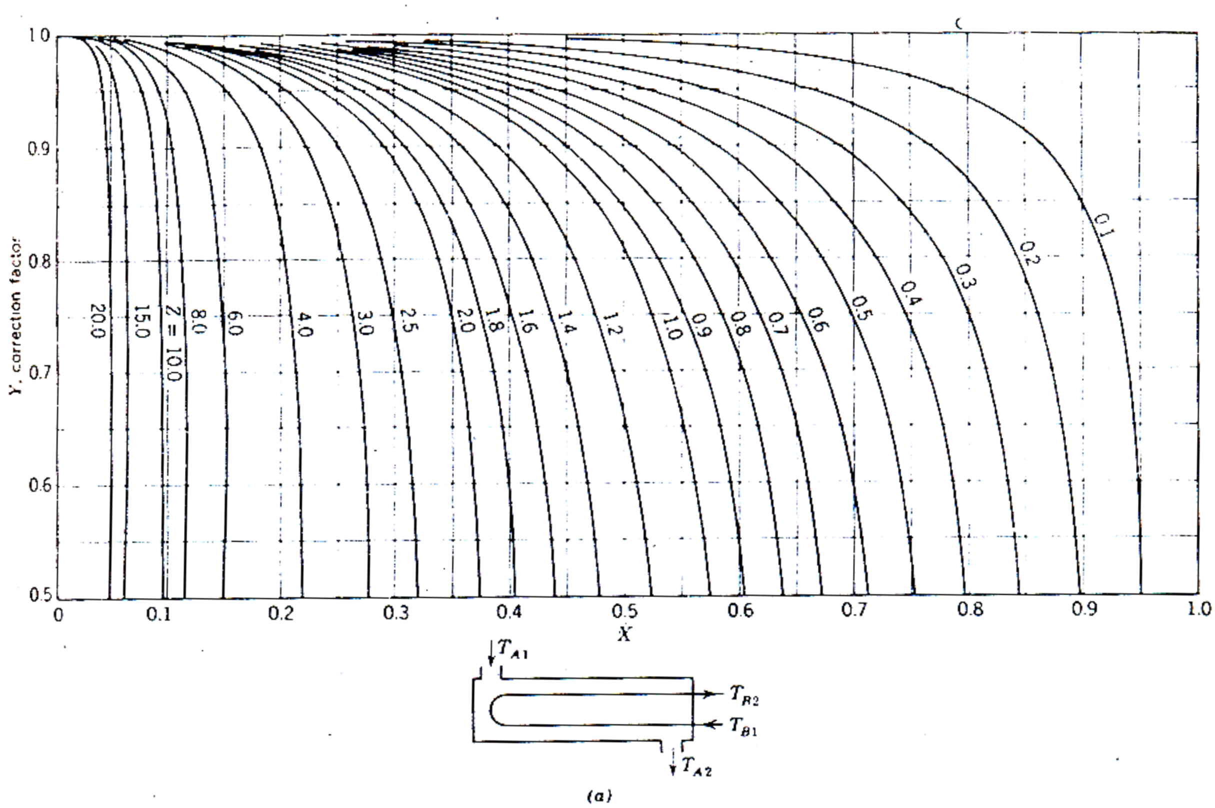

Figure 3-1. Correction factor for mixed-flow Type Heat Exchangers one shell pass; two or more tube passes. Adapted from ASME as cited by Foust, 1980.

Rearranging equation---- and expressing in terms of resistance,

$$q = \frac{\Delta T}{R} = UAY\Delta T \ln \qquad \textit{eq 3-5}$$

Thus

$$\frac{1}{R} = UA \qquad eq\ 3\text{-}6$$

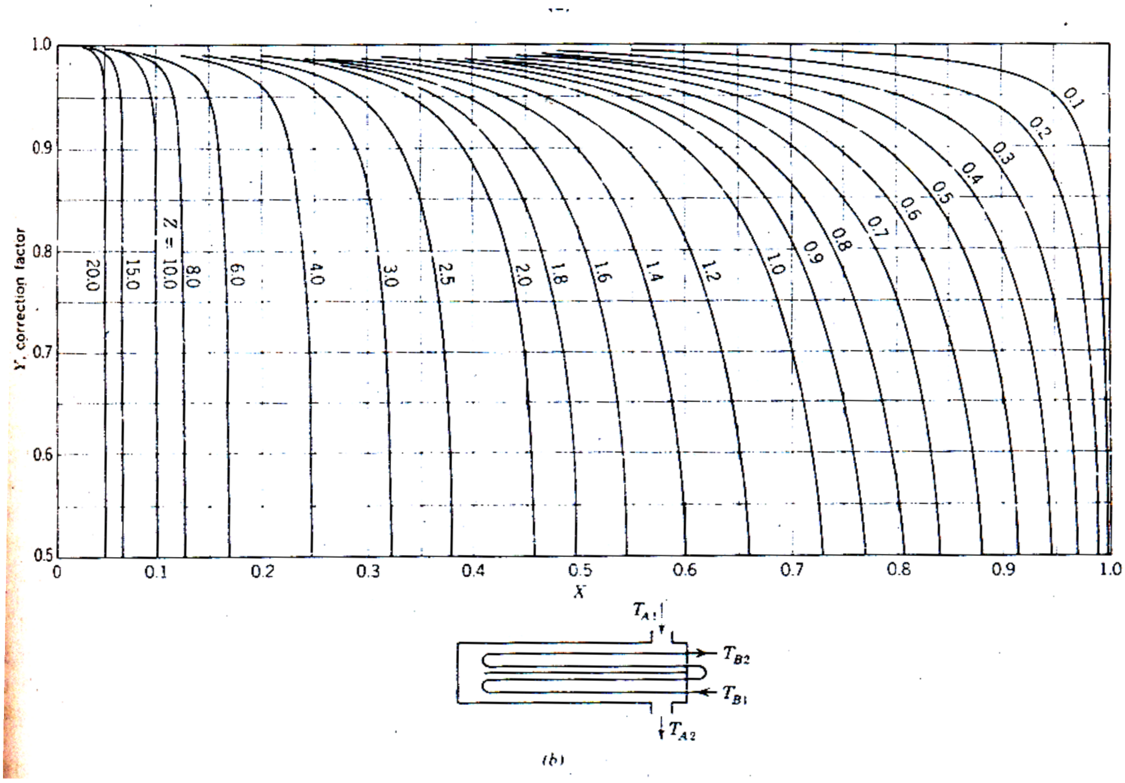

Figure 3-2. Two Shell passes; four tube passes. Adapted from ASME as cited by Foust, 1980.

$$X = \frac{T_{B2} - T_{B1}}{T_{A1} - T_{B1}} \qquad Z = \frac{T_{A1} - T_{A2}}{T_{B2} - T_{B1}}$$

The Overall heat transfer coefficient maybe based on the inside, U_i or based on the outside, U_o . The choice of overall heat transfer coefficient would depend on which of the resistances would be controlling or bigger. While overall heat transfer coefficient is directly proportional to convective heat transfer coefficient, convective resistance is inversely proportional to the convective heat transfer coefficient. Thus, the lower the convective heat transfer coefficient the more controlling its resistance.

If

ho <<< hi,

then

Ro >>> Ri

thus

$$q = U_o A_o Y \Delta T \ln \qquad eq\ 3\text{-}7$$

On the other hand, if

hi <<< ho

Then

Ri >>> Ro

Thus

$$q = U_i A_i \Delta T \ln \qquad eq\ 3\text{-}8$$

In a steady-state transfer of heat from a hot fluid stream outside the tube to a cold fluid stream inside the tube, the following steps are involved: convection from the hot fluid outside the tube to a tube wall surface, conduction through the tube wall, and convection from the surface to the cold fluid flowing inside the tube. Thus, a 3- resistance overall heat transfer coefficient is,

$$R = \frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{\Delta x_w}{k_w \bar{A}m} + \frac{1}{h_o A_o} \qquad eq\ 3\text{-}9$$

If outside film resistance is controlling, *Equation 3-9* is reduced to,

$$\frac{1}{U_o} = \frac{Do}{h_i Di} + \frac{\Delta x_w Do}{k_w Dm} + \frac{1}{h_o} \qquad eq\ 3\text{-}10$$

Whereas if the inside film resistance is controlling , the working equation would be:

$$\frac{1}{Ui} = \frac{1}{h_i} + \frac{\Delta x_w Do}{k_w Dm} + \frac{Di}{h_o Do} \qquad eq\ 3\text{-}11$$

However if fouling exits in both sides of the wall, additional scale resistances will be incorporated,

$$R_{f,i} = \frac{1}{h_{f,i} Ai} \quad and \quad R_{f,o} = \frac{1}{h_{f,o} Ao} \qquad eq\ 3\text{-}12$$

where

$h_{f,i}$ and $h_{f,o}$ are the fouling coefficients of the inside and outside surfaces of the tube, respectively.

Thus , the overall resistance , U , is:

$$R = \frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{1}{h_{f,i} A_i} + \frac{\Delta x_w}{k_w \bar{A}m} + \frac{1}{h_o A_o} + \frac{1}{h_{f,o} A_o} \qquad eq\ 3\text{-}13$$

A similar simplified equation may be derived for known controlling film resistance.

Table 3-1 shows typical fouling factor for different liquids at different fluid velocity.

Table 3-1. Typical Fouling factor (Foust, 1980).

	Fouling Factor, $R_d = 1/h_d$ hr °F ft^2/Btu Water Velocity	
	3 ft/s or less	3 ft/s or more
Seawater (up to 125°F)	0.005	0.0005
Well water	0.001	0.001
Delaware and Lehigh river waters	0.003	0.002
Brine	0.001	0.001
Fuel oil	0.005	0.005

Conductive Heat Transfer

Fourier's Law of Heat Conduction states that the heat flux **,** *q/A,* (the rate of heat transfer per unit time per unit area) is proportional to the temperature gradient (-dT) and inversely wall thickness:

$$\frac{q}{A} = -k\frac{dT}{dx} \qquad eq\ 3\text{-}14$$

$$= -k\frac{(T_2 - T_1)}{(x_2 - x_1)} \qquad eq\ 3\text{-}15$$

where:

$\Delta x = (x_2 - x_1)$, thickness of the wall

$\Delta T = (T_1 - T_2)$, temperature drop across wall

$R = \Delta x/(kA)$, conductive thermal resistance of wall

Where the proportionality constant *k is* the thermal conductivity that indicates how good the material conducts thermal energy. In most engineering applications *k* may be considered constant except in high temperature drops. In cases where *k* varies with temperature a linear relationship such as in *Equation 3-16* may be used (McCabe, 2001):

$$k = a + bT \qquad eq\ 3\text{-}16$$

Where *a* and *b* are empirical constants and *T* is the temperature of the medium.

In case of radial conduction of heat in a hollow cylindrical vessels or pipes, the area perpendicular to the direction of heat flow is not constant but is proportional to the radius ($A = 2\pi rL$). Thus, for hollow cylindrical configurations with outside radius r_o to inside radius r_i, ratio is greater than 1.4 (Mc Cabe):

$$q = \frac{k(2\pi L)(T_i - T_o)}{\ln(r_o / r_i)} = k\overline{A}_L \frac{(T_i - T_o)}{(r_o - r_i)} = \frac{\Delta T}{R} \qquad eq\ 3\text{-}17$$

where:

$$R = \frac{(r_o - r_i)}{k\bar{A}_L} \qquad \textit{eq 3-18}$$

$$\bar{A}_L = (2\pi\bar{r}_L)L \qquad \textit{(Logarithmic mean surface area)} \qquad \textit{eq 3-19}$$

$$\bar{r}_L = \frac{(r_o - r_i)}{ln(r_o / r_i)} \qquad \textit{(Logarithmic mean radius)} \qquad \textit{eq 3-20}$$

In cases where the outside to inside radius ratio is less 1.4, logarithmic mean radius will just be equal to average radius.

Series of Resistances

At steady-state, the rate of heat transfer through a wall consisting of a series of layers of different conducting media *A*, *B*, *C …* that are in excellent thermal contact will be equal to the rate of heat transfer in each layer,

$$q = q_A = q_B = q_C \qquad \textit{eq 3-21}$$

The total temperature drop across the multilayer wall is equal to the sum of the temperature drops across each layer,

$$\Delta T = \Delta T_A + \Delta T_B + \Delta T_C \qquad \textit{eq 3-22}$$

Whereas the total resistance of the multilayer wall is equal to the sum of the individual resistances,

$$\Delta T / q = R = R_A + R_B + R_C \qquad \textit{eq 3-23}$$

Where resistance for each layer is:

$$R_A = \Delta T_A / q_A = \Delta x_A / (k_A A) \qquad \textit{eq 3-24}$$

$$R_B = \Delta T_B / q_B = \Delta x_B / (k_B A) \qquad \textit{eq 3-25}$$

$$R_C = \Delta T_C / q_C = \Delta x_C / (k_C A) \qquad \textit{eq 3-26}$$

Convective Heat Transfer

Newton's Law for Convective Heat Transfer states that the convective heat flux is proportional to the difference between the surface temperature, T_w, and the temperature of the fluid T_f along the heat flow path:

$$\frac{q}{A} = h(T_w - T_f) \qquad \textit{eq 3-27}$$

$$R = \frac{1}{hA} \qquad \textit{eq 3-28}$$

Where:

R = Convective resistance
h = Convective heat transfer coefficient (local film coefficient)
A = heat transfer area in-contact with the fluid

Several empirical equations were derived to estimate convective heat transfer applicable coefficient for different types of heat exchangers that operates as heaters, condensers, reboiler and evaporators.

Heat Transfer without Phase Change

This section covers the estimation of convective heat transfer coefficient of fluids involve in heating and cooling in Double Pipe and Shell and tube type of heat Exchangers.

Double Pipe Heat Exchanger

The Sieder-Tate equation is applicable in the estimation of the convective heat transfer coefficient of fluids flowing inside the tube of a double pipe heat exchanger. It could also be use to estimate the convective coefficient of the annulus fluid provided the annular diameter is reflected as equivalent diameter, *Deq. .* Sieder-Tate equation is applicable only for non-metallic fluid with Prandtl number between 0.5 to 100 and Reynold's number of at least 10,000.

For long tubes (L/D) >50 (Mc Cabe 2001).

The convective heat transfer coefficient for the tube fluid is:

$$\frac{h_i D_i}{k} = 0.023\left(N_{Re}^{0.8}\right)\left(N_{Pr}^{1/3}\right)\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \textit{eq 3-29}$$

where

$$N_{\mathrm{Re}} = \frac{D\overline{V}\rho}{\mu} \quad , \quad N_{\mathrm{Pr}} = \frac{c_p\mu}{k} \qquad \textit{eq 3-30}$$

The convective heat transfer coefficient for the annular fluid is:

$$\frac{h_o D_{eq}}{k} = 0.023\left(N_{\mathrm{Re},O}^{0.8}\right)\left(N_{\mathrm{Pr}}^{1/3}\right)\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \textit{eq 3-31}$$

where

$$D_e q = \frac{4S}{P} \qquad \textit{eq 3-32}$$

D_{eq} = equivalent or hydraulic diameter
S = cross sectional area of the fluid stream
P = wetted perimeter

and

$$S = \frac{\pi}{4}\left(D_{oi}^2 - d_{io}^2\right) and\ P = \pi\left(D_{oi} + d_{io}\right) \qquad \textit{eq 3-33}$$

thus:

$$D_{eq} = \frac{4\left(\frac{\pi}{4}\right)\left(D_{oi} - d_{io}\right)\left(D_{oi} + d_{io}\right)}{\pi\left(D_{oi} + d_{io}\right)} \qquad \textit{eq 3-34}$$

$$D_{eq} = D_{oi} - d_{io} \qquad \textit{eq 3-35}$$

The Reynold’s Number of the annulus fluid is then equal to

$$N_{\mathrm{Re,O}} = \frac{D_{eq}\overline{\upsilon}\rho}{\mu} \qquad \textit{eq 3-36}$$

For a short tube with L/D < 50, (McCabe, 2001)*:*

$$\frac{his}{hi} = 1 + \left(\frac{D}{L}\right)^{0.7} \qquad \textit{eq 3-37}$$

Where:

h_{is} = average heat transfer coefficient over the short tube length
h_i= heat transfer coefficient for fully developed turbulent flow (long tube)

Laminar Flow Forced Convection **(Geankoplis, 1995*)***

$$N_{Nu} = 2N_{Gz}^{1/3}\phi_v^{0.14} \qquad \textit{eq 3-38}$$

where: μ = viscosity at the mean bulk temperature of the fluid

μ_w =viscosity of the fluid evaluated at the tube wall temperature T_w

$$N_{Nu} = \frac{h_i D}{k} \quad , \quad N_{Gz}^{1/3} = \frac{\dot{m}c_p}{kL} \quad , \quad \phi_v{}^{0.14} = \frac{\mu}{\mu_w}$$

Shell and Tube Type Heat Exchanger

To compute for the outside heat transfer coefficient, ho, in a shell and tube heat exchanger, *Figure 3-3* has been commonly used . The choice of pitch to diameter ratio and tube pitch arrangement are used simultaneously with the Reynold's number to determine the j'g max value which will in turn use to estimate the average coefficient.

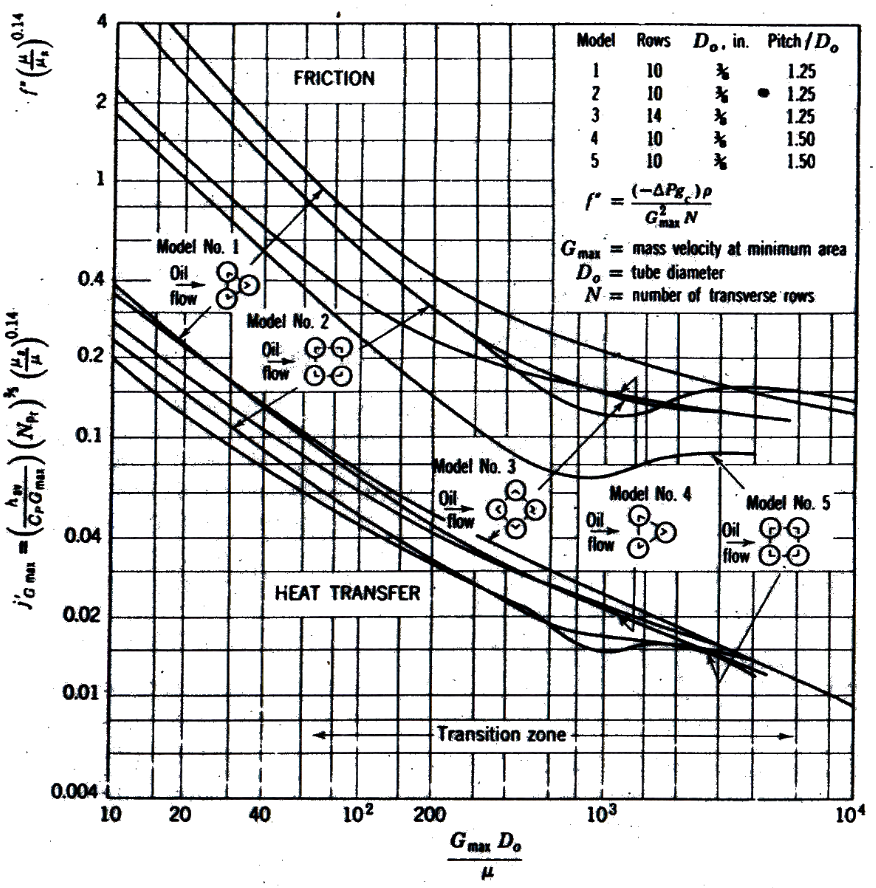

Figure 15.16. Heat-transfer coefficients and friction factors for flow normal to tube banks (3). (With permission of ASME, copyright © 1952.)

Figure 3-3. Heat Transfer Coefficient for Flow Normal to tube banks. Adapted from ASME as cited by Foust et.al., 1980.

To compute for the inside heat transfer coefficient, hi, in a shell and tube heat exchanger requires tube length to diameter ratio, L/D. *Figure 3-4* which is applicable for both laminar and turbulent flows could be used to estimate the convective coefficient. L/D is graph against the Reynold's number to determine the j'g value which will in turn be use to estimate the average convective coefficient.

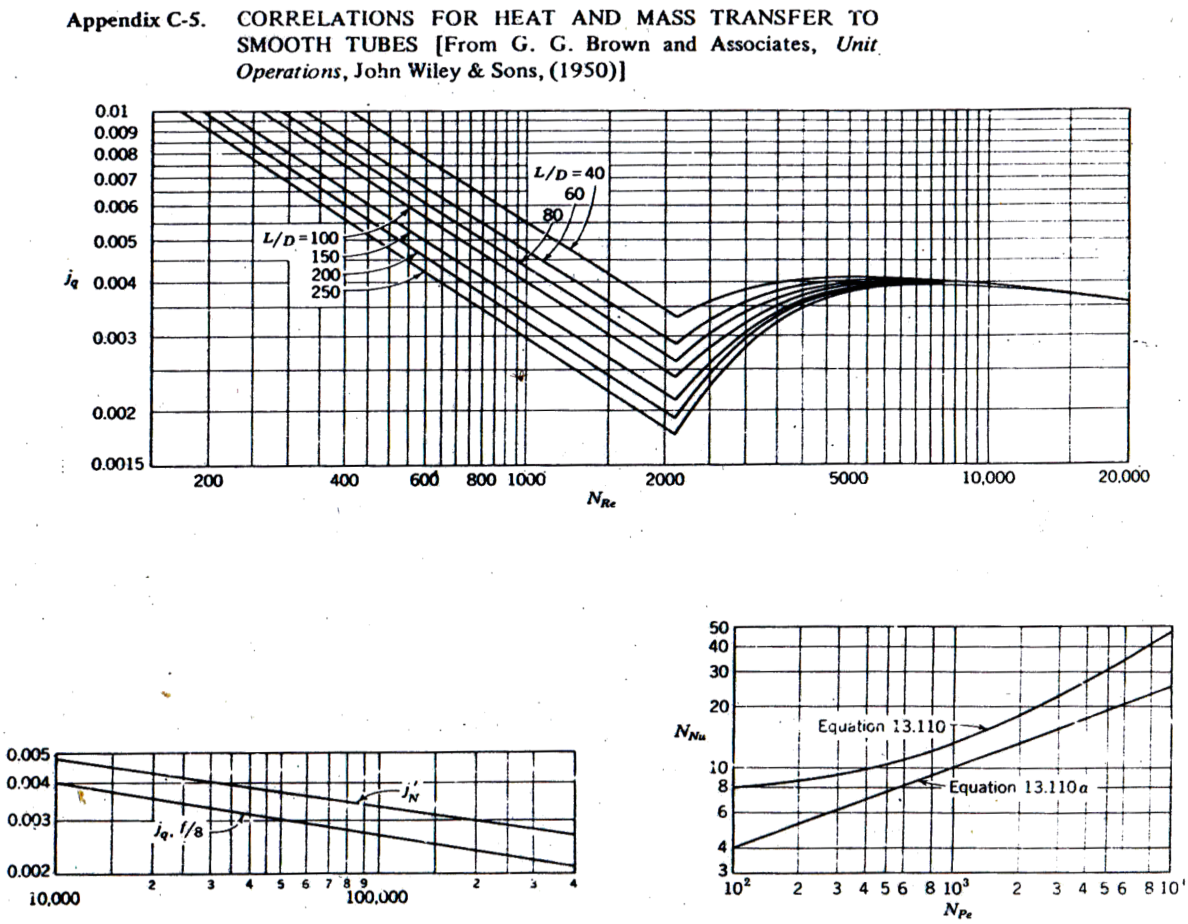

Figure 3-4. Correlations for Heat and Mass Transfer to Smooth Tubes.
Adapted from Brown et.al., 1950.

where

$$\left(\frac{h_{am}}{C_p \bar{\upsilon} \rho}\right)(N_{Pr})^{2/3}\left(\frac{\mu_1}{\mu}\right)^{0.14} = j_g = 1.86(N_{Re})^{\frac{2}{3}}\left(\frac{D}{L}\right)^{\frac{1}{3}} \qquad \textit{eq 3-39}$$

Heat Transfer with Phase Change

Condesation

Condensation of vapor may be of Filmwise or Dropwise condensation. Filmwise Condensation occurs when vapor of organic substances condense on cold metallic surfaces forming a continuous film. While Dropwise condensation occurs when vapor condenses on cold metallic surfaces and does not form continuous film but rather droplets condensates only.

Film – Type Condensation on Vertical Surface

As cited by Foust et al (1980), Nusselt derived the basic working equation for the evaluation of film type condensation on the vertical surface. The derived equation requires several assumptions as presented by Foust:

1.) Pure vapor is at its saturation temperature.
2.) The condensate film flows in laminar regime and heat is transferred through the film by condensation.
3.) The temperature gradient through the film is linear.
4.) Temperature of the condensing surface is constant.
5.) The physical properties of the condensate are constant and evaluated at a mean film temperature.
6.) Negligible vapor shear exists at the interface.

$$\bar{h} = 0.943\left[\frac{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)\Delta H_v g}{\mu_\ell L(T_v - T_l)}\right]^{1/4} \qquad \textit{eq 3-40}$$

where

ρ_l, ρ_v = densities of liquid and vapor
g = gravitational acceleration
k_e = liquid thermal conductivity
μ_e = liquid viscosity
ΔH_v = latent heat of vaporization
T_v = vapor saturation temperature
T_1 = surface temperature
W = condensate, #/hr

The above equation has been found to be fairly in agreement with experimental values. However, assumption 2 has to deal with because the condensate does not flow strictly on the laminar regime. Most experimental values are higher than the predicted values from the above equation. An increase of 20 percent of the predicted value has been recommended , thus,

$$\bar{h} = 1.13\left[\frac{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)\Delta H_v g}{\mu_\ell L(T_v - T_l)}\right]^{1/4} \qquad \textit{eq 3-41}$$

Film Type Condensation on Horizontal Surface

As cited by Foust et al, on the same assumptions, Nusselt derived the working equation for the condensation of vapor on horizontal surface as:

$$\bar{h} = 0.725\left[\frac{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)\Delta H_v g}{\mu_\ell D_2(T_v - T_1)}\right]^{1/4} \qquad eq\ 3\text{-}42$$

For N_{Re} > 40, *h* is multiplied by 1.2 to account for the effect of rippling (McCabe, 2001).

However, when the amount of the condensate is known, Mc Adams (1954) presented an alternative equation,

$$\bar{h} = 0.95\left[\frac{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)gL}{\mu_\ell W}\right]^{1/3} \qquad eq\ 3\text{-}43$$

Where W is the amount of condensate in lb/hr

Banks of Horizontal Tubes

The result on the single horizontal tube may be extended to estimate the average convective heat transfer coefficient, hN of bank of tubes on N vertical rows (Brown ,1950),

$$\bar{h}_N = \bar{h}N^{-\frac{1}{4}} \qquad eq\ 3\text{-}44$$

where:

$$N^{1/4} = \frac{N_1 + N_2 + N_3 \ldots + N_n}{N_1^{3/4} + N_2^{3/4} + N_3^{3/4} + \ldots + N_n^{3/4}} \qquad eq\ 3\text{-}45$$

N is number of tubes per row while subscripts 1, 2, 3, 4…etc. are row numbers.

Combining the above equations,

$$\bar{h} = 0.725\left[\frac{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)\Delta H_v g}{\mu_\ell N D_2(T_v - T_1)}\right]^{1/4} \qquad eq\ 3\text{-}46$$

However if splashing occurs as the condensate flows from tube to tube Kern (1950) as cited by Foust (1980), proposed a modification of the working equation as follows;

$$\bar{h} = 0.725\left[\frac{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)\Delta H_v g}{\mu_\ell D_2 N^{2/3}(T_v - T_l)}\right]^{1/4}$$ *eq 3-47*

For turbulent condensation on vertical surfaces , as cited by Foust (1980), Kirkbride (1934) and Colburn (1933) proposed a reasonable estimation of the convective heat transfer coefficient,

$$\bar{h}_c\left[\frac{\mu_\ell^2}{K_\ell^{3}\rho_\ell(\rho_\ell - \rho_\upsilon)g}\right]^{1/3} = 0.011(N_{ReC})^{1/3}\cdot\sqrt[4]{N_{Pr\ell}}$$ *eq 3-48*

Condensate properties are evaluated at a mean film temperature, T_f,

where:

$$T_f = T_{sv} - \frac{3}{4}(T_{sv} - T_s)$$ *eq 3-49*

Ts_V = saturation temperature of condensing vapor

T_s = temperature of solid surface

Film Boiling on Submerged Horizontal Cylinder or Sphere

Boiling heat transfer coefficient may estimated using McNelly equation (Perry and Green, 1997)

$$h = 0.225\left[\frac{qC_P}{A\lambda}\right]^{0.69}\left[\frac{Pk}{\sigma}\right]^{0.31}\left[\frac{\rho_\ell}{\rho_\upsilon} - 1\right]^{0.33}$$ *eq 3-50*

Where

$Cpe ; \frac{BTU}{\#° F}$ = heat capacity

$X = 970\ \frac{BTU}{\#}$ = latent heat evaporation

$K_e : \frac{BTU\, ft}{ft^2 hr° F}$ = thermal conductivity

$\sigma\ 4.97 x 10^{-3} \frac{\#f}{ft}$ = surface tension

P Psia, Pressure of system = pressure of system

$\frac{Q}{A}$ = rate of heat flux

ρ_e, ρ_υ = density of liquid and vapor

These properties are evaluated at 212°F or vapor side temperature

Cengel, (2003) proposed an alternative equation for the estimation of film boiling coefficient applicable for both horizontal cylinder or a sphere,

$$\frac{q}{A} = C\left[\frac{gk_v^3\rho_v(\rho_l - \rho_v)[\lambda + 0.4c_{pv}(T_s - T_{sat})]}{\mu_v D(T_s - T_{sat})}\right]^{1/4}(T_s - T_{sat}) \qquad eq\ 3\text{-}51$$

where:

A = area of surface of cylinder or sphere in contact with fluid
C = 0.62 for horizontal cylinder; C = 0.67 for sphere
λ = heat of vaporization
g = gravitational acceleration
ρ_V, ρ_L = density of liquid and vapor, respectively
μ_V , k_V = viscosity and thermal conductivity of vapor, respectively
T_s = surface temperature of hot cylinder or sphere
T_{sat} = boiling temperature at the specified pressure
c_{pv} = specific heat capacity of vapor

HEAT EXCHANGER DESIGN SPECIFICATION

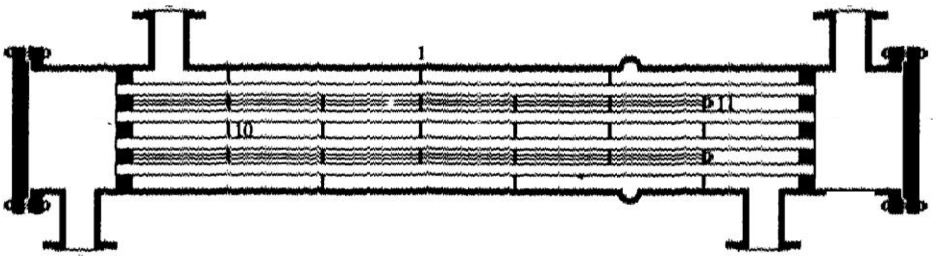

Figure 3-5. *Shell-and-Tube Heat Exchanger*.

Layout and Pitch Arrangement

Tubes are usually arranged in a triangular or square pitch arrangement. Pitch is the center-to-center distance between tubes. Rotated square pitch, a variation of square pitch is the third commonly used tube arrangement as presented in ***Figure 3-6***. While a triangular pitch arrangement offers more heat transfer area per unit volume of a heat exchanger, the square pitch arrangement offers ease in cleaning and maintenance operations. A minimum of 1.25 pitch to diameter ratio and/ or a minimum webb thickness between tubes of approximately 3.2 mm could ensure sufficient strength for tube rolling. Whereas a 6.4 mm clearance is suggested for mechanical cleaning requirement (Hewitt, et. al., 1994). In most design, the pitch to diameter ratio range from 1.25 to 1.5 (Peters et. al, 2004).

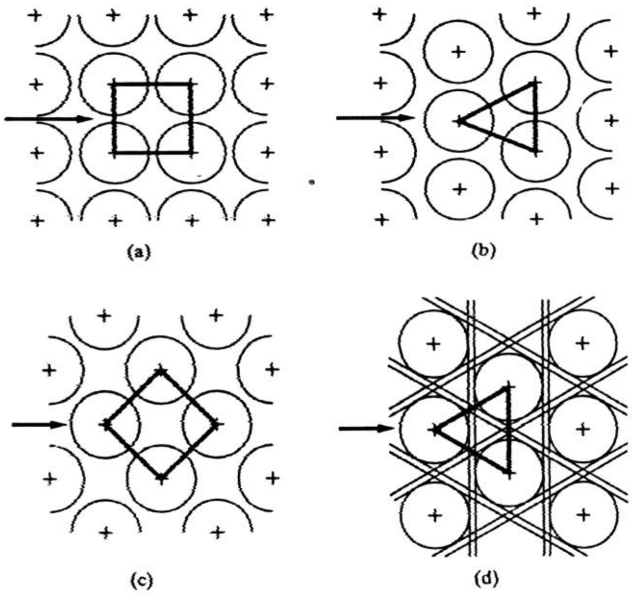

Figure 3-6. *Tube Layout Patterns: (a) Square Pitch; (b) Triangular Pitch; (c) Square Pitch Rotated; (d) Triangular Pitch with Cleaning Lanes.*

Tube layout normally follows symmetrical arrangement having the largest number of tubes at the center. With an appropriate pitch to diameter ratio and optimum pipe diameter chosen, known total heat transfer area, would lead to the shell diameter specification. Minimum shell diameter is calculated by:

$$\textbf{Shell Diameter}_{Min} = N_c D_o + (N_c + 1) C \qquad eq\ 3\text{-}52$$

Where

N_c = Number of tubes at the Center

C = Clearance

$$Clearance = Pitch - Diameter \qquad eq\ 3\text{-}53$$

where $$Pitch = \left(\frac{Pitch}{Dia}\right) Diameter$$

Total heat transfer Area (A_T) is equal to the product of total number of tubes and heat transfer area per tube.

$$A_T = N_T \pi DL \qquad eq\ 3\text{-}54$$

where N_T = Total number of tubes

d = Either inside or outside diameter

L = Length of tube

At a set pipe diameter d, a compromise is chosen between N_T and L using $\left(\frac{L}{D_{shell}}\right)$ ratio of 5 to 10 (Hewitt, et. al., 1994).

A consequence of choosing higher ratio would result to longer tube length and eventually smaller shell diameter. With smaller shell diameter, tube sheets, shell diameter etc. would be thinner, hence, cheaper. At a known volumetric flow rate, longer tubes (less number of tubes) would provide for higher tube velocity, however such design might encounter difficulty on baffle support on the shell side. Standard tube lengths in a shell and tube type heat exchanger come in 8, 12 or 16 ft and are available in a variety of different diameters and wall thickness (Peters and Timmerhaus, 1991). However 20 ft tube lengths are now the most commonly used (Perry and Green, 1997).

Table 3-2 shows the number of tubes in a conventional tube sheet layout at different shell diameter, pitch arrangement and number of passes for ¾ and 1 inch tube diameters.

Table 3–2. Number of Tubes in Conventional Tubesheet Layouts.

Shell ID, in.	One-pass		Two-pass		Four-pass	
	Square Pitch	Triangular Pitch	Square Pitch	Triangular Pitch	Square Pitch	Triangular Pitch
¾-in.-OD tubes on 1-in. pitch						
8	32	37	26	30	20	24
12	81	92	76	82	68	76
15¼	137	151	124	138	116	122
21¼	277	316	270	302	246	278
25	413	470	394	452	370	422
31	657	745	640	728	600	678
37	934	1074	914	1044	886	1012
1-in.-OD tubes on 1¼-in. pitch						
8	21	21	16	16	14	16
12	48	55	45	52	40	48
15¼	81	91	76	86	68	80
21¼	177	199	166	188	158	170
25	260	294	252	282	238	256
31	406	472	398	454	380	430
37	596	674	574	664	562	632

Adapted from Table 7, p 608. Plant Design and Economics for Chemical Engineers 4th ed.
O.5.2. Baffle and Tube Sheet.

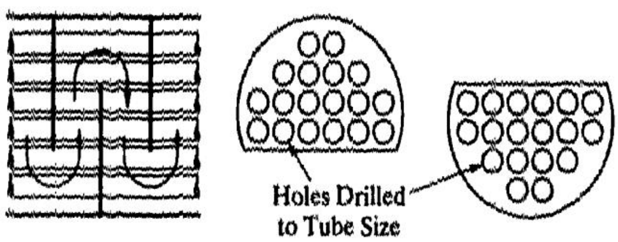

Figure 3–7. *Segmental Baffles.*

Baffles and Tubesheet

Baffles are used to support tubes mechanically against sagging and possible vibration. They also direct fluid flow and control velocities. Among the different types of baffle, the segmental types are the most commonly used. Segmental baffle cut is normally 25% of the disk diameter (maximum of 45%) (Peters et. al, 2004). The ratio of baffle spacing to baffle cut is a major design parameter for the effective conversion of pressure drop to heat transfer (Hewitt, et. al., 1994). However to accommodate low pressure drop requirement, the disk use of and doughnut baffle type reduces pressure drop by 50% - 60% (Hewitt, et. al., 1994 and Peters et. al, 2004). Common baffle spacing for Shell and Tube Type heat exchanger ranges from 20% to 100% of the shell diameter (Peters and Timmerhaus, 1991). TEMA recommends a minimum spacing of 50 mm or 20% of shell diameter which ever is greater (Perry and Green, 1997).

When maximum spacing is sought, it should not exceed the value indicated in ***Table 3–3*** (Hewitt, et. al., 1994). This has been set to anticipate effects of temperature on materials of construction. It should be noted that at smaller baffle spacing, shell side pressure drop is high.

For unsupported tube span, the maximum unsupported tube span should equal to $74d^{0.75}$ (d is outside tube diameter). The unsupported span is reduced by 12% for aluminum, copper and their alloys (Perry and Green, 1997).

Table 3–3. Maximum Unsupported Straight Tube Length (all dimensions in millimeters).

	Maximum Unsupported Span Tube Materials and Temperature Limits (°C)	
Tube OD (approx.)	Carbon and High Alloy Steel (400) Low Alloy Steel (454) Nickel-Copper (316) Nickel (454) Nickel-Chromium-Iron (538)	Aluminum & Aluminum Alloys Copper & Copper Alloys Titanium & Zirconium at Code Max. Allowable Temperature
19	1520	1321
25	1880	1626
32	2240	1930
38	2540	2210
50	3175	2794

Adapted from Table 6.1 p 271. Process Heat Transfer. (Hewitt 1994).

TEMA sets minimum thickness for baffle and support plate thickness as shown in Table O – 15.

For tubesheet thickness calculations, bending and shearing of tubesheet have been considered by TEMA (Jawad and Farr, 1988).

The TEMA equation for the required thickness in bending is

$$T = \frac{GF}{2}\sqrt{\frac{P}{S}}$$ *eq 3-55*

where

T = Required thickness of tubesheet
G = Diameter
P = Applied Pressure
S = ASME allowable tensile stress
F = 1.25 For Simply supported plate
= 1.0 For Fixed plate

The required thickness for the shearing stress at the outer tube perimeter is:

$$T = \frac{0.31\, D_L}{1 - \frac{d_o}{p}}\left(\frac{P}{S}\right)$$ *eq 3-56*

where

$D = 4\,{}^{A}\!/_{C}$
A = Area of tubesheet within outer tube perimeter
C = Perimeter of outer tube
d_o = Outside diameter of tube
p = Distance between tubes

The resulting thickness resulting from equations 3-55 and 3-56 are compared to the thickness as specified by TEMA (Table 3-4). Whichever is thickner will be specified for used.

Table 3–4. Baffle and Support Plate Minimum Thickness (TEMA).

DISTANCE BETWEEN ADJACENT SEGMENTAL PLATES, (Inches)	NOMINAL SHELL INSIDE DIAMETER (Inches)			
	8 - 14	15 - 28	28 - 38	>39
<12	⅛	3/16	¼	¼
12 – 18	3/16	¼	5/16	⅜
18 – 24	¼	⅜	⅜	½
24 – 30	⅜	⅜	½	⅝
>30	⅜	½	⅝	⅝

Pressure Drop Across the Heat Exchanger

Convective heat transfer coefficient increases with increasing fluid velocity. However, this large film coefficient can be counter-balanced by the detrimental effects of increased pressure drop due to increase in fluid velocity (Perry and Green, 1997). Judgment on the compromised velocity for net beneficial effect could only be done by economic analysis considering smaller equipment cost (smaller heat transfer area) versus high pumping cost (at higher pressure drop).

Kern (1950) provides for the estimation of pressure drop in both shell side and tube side. These equations presented by Backhurst and Harker (1973) are reproduced in the following paragraph:

Tube Side

On the tube side, the pressure drop consists of two components, the friction losses in the tubes and the loss due to change in direction:

$$\Delta \mathrm{P}_t = \frac{fG_t^2 Ln}{5.22\times 10^{10} D_t S\phi_t} + \frac{4n\mu^2 \times 62.4s}{2\times 144g}\left[\frac{lb}{in^2}\right] \qquad \textit{eq 3-57}$$

$$\Delta P_t = \frac{fG_t^2 L_n}{2\times 10^6 D_t S\phi_t} + 2.26n\mu^2 s\left[\frac{KN}{m^2}\right] \qquad \textit{eq 3-58}$$

where

G_t = Mass velocity in the tubes $\left(\frac{m_t}{a_t}\right)\left[\frac{lb}{ft^2hr}\right]\left[\frac{kg}{m^2s}\right]$ *eq 3-59*

$$a_t = \frac{Na^1}{n}\left[ft^2\right]\left[m^2\right] \qquad \textit{eq 3-60}$$

N = Total number of tubes
n = Number of tube side passes
a^1 = Cross-sectional area of flow per tube $\left[ft^2\right]\left[m^2\right]$
D_t = Tube diameter $ft \quad m$
μ = Fluid velocity $\left[\frac{ft}{s}\right]\left[\frac{m}{s}\right]$
f = Friction Factor $\left[\frac{ft}{in^2}\right]\left[\frac{m^2}{m^2}\right]$

$$f = 0.05N_{\mathrm{Re}}^{-0.33}\left[\frac{ft^2}{in^2}\right] \quad or \qquad \textit{eq 3-61}$$

$$f = 0.72N_{\mathrm{Re}}^{-0.33}\left[\frac{m^2}{m^2}\right] \qquad \textit{eq 3-62}$$

Applicable for N_{Re} >1000

Shell side

$$\Delta \mathrm{P}_s = \frac{fG_s^2 D_s L}{4.35\times 10^9 D_e S \phi_s B}\left[\frac{lb}{m^2}\right] \quad \textit{eq 3-63}$$

$$\Delta P_S = \frac{fG_s^2 D_s L}{2\times 10^6 D_e S \phi_s B}\left[\frac{KN}{m^2}\right] \quad \textit{eq 3-64}$$

where

G_s = Mass velocity in the shell $\left(\frac{lb}{ft^2 hr}\right)$ $\left(\frac{kg}{m^2 s}\right)$

D_s = Shell inner diameter [ft] [m]

S = Specific gravity of the fluid

$$\phi_s = \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad \textit{eq 3-65}$$

L = Tube length [ft] [m]

B = Baffle spacing [ft] [m]

f = Friction factor $\left[\frac{ft^2}{in^2}\right]\left[\frac{m^2}{m^2}\right]$

$$f = 0.013 N_{\mathrm{Re}}^{-0.2}\left[\frac{ft^2}{in^2}\right] \quad \textit{eq 3-66}$$

$$f = 1.87 N_{\mathrm{Re}}^{-0.2}\left[\frac{m^2}{m^2}\right] \quad \textit{eq 3-67}$$

Applicable for N_{Re} >500

Peters and Timmerhaus, (1991) provide an alternative equation for pressure drop across the tube and shell as reproduced in the following paragraph:

For tube – side

$$-\Delta P_i = \frac{B_i 2 f_i G^2 L n_p}{g_c \rho_i D_i \phi} \quad \textit{eq 3-68}$$

where subscript *i* refers to inside of tube at bulk temperature

f_i = Fanning friction factor for isothermal flow based on conditions at the arithmetic-average temperature of the fluid

n_p = Number of tube passes

g_c = Conversion factor in Newton's law of motion,

$$g_c = 32.17 \times \ 3600^{\ 2} \ \frac{ft \cdot lbm}{hr^2 \cdot lbf} \qquad \textit{eq 3-69}$$

Φ

Φ_I = Correction factor for non-isothermal flow

$$\phi_i = 1.1\left(\frac{\mu_i}{\mu w}\right)^{0.25} \qquad \textit{eq 3-70}$$

when D_i/μ is less than 2100 and

$$\phi_i = 1.02\left(\frac{\mu_i}{\mu_w}\right)^{0.14} \qquad \textit{eq 3-71}$$

when $\dfrac{D_i G}{\mu_i}$ is greater than 2100;

μ_i = Viscosity at arithmetic – average (bulk) temperature of fluid

μ_w = Viscosity of fluid at average temperature of the inside tube wall surface

B_i = Correction factor to account for friction due to sudden contraction, expansion and reversal of flow direction

$$B_i = \frac{1 + \ F_e + F_e + F_r}{2 f_i G^2 L \Big/ g_c \rho_i^2 D_i \phi_i} \qquad \textit{eq 3-72}$$

For flow across tubes, the following equation can be used to approximate pressure drop due to friction:

$$-\Delta P_o = \frac{B_o 2 f^1 N_r G_s^2}{g_c \rho_o} \qquad \textit{eq 3-73}$$

where subscript *o* refers to outside of tube at bulk temperature

f^1 = Special friction for shell-side flow

$$f^1 = b_o\left(\frac{D_o G_s}{\mu_f}\right)^{-0.15} \qquad \textit{eq 3-74}$$

$$b_o = 0.23 + \frac{0.11}{X_T - I^{\ 1.08}} \qquad \textit{eq 3-75}$$

$$b_o = 0.044 + \frac{0.08 X_L}{X_T - 1^{\left(0.43 + \frac{1.13}{X_L}\right)}} \qquad \textit{eq 3-76}$$

$$where \quad X_T = \frac{ratio \;\; of \;\; pitch \;\; transverse \;\; to \;\; flow}{tube \;\; diameter} \qquad eq\ 3\text{-}77$$

$$X_L = \frac{ratio \;\; of \;\; pitch \;\; pallalel \;\; to \;\; flow}{tube \;\; diameter} \qquad eq\ 3\text{-}78$$

N_r = Number of rows of tubes across which shell fluid flows
B_o = Correction factor to account for friction due to reversal in directional flow re-crossing of tubes, and variation in cross section

$B_o =$ 1 when the flow is across unbaffled tubes or

$B_o =$ Number of tubes crosses as a rough approximation

Variation of Kern method and other estimations by Bell-Delaware method and Wills and Johnson method are discussed in Process Heat Transfer (Hewitt, et. al., 1994).

Heat Exchanger Temperature Limits

The most common heat exchanger medium used for cooling is water. Aside from its abundance and cost, water exhibits relatively high heat capacity. In the design of heat exchanger, it is obvious that either large quantity of cooling will be used or greater water temperature change should be anticipated to come up with smaller heat exchanger. Large quantity of cooling water would result to higher water velocity. This high velocity will reduce fouling but increases water and pumping costs. On the other hand, large water temperature increase will require less water and pumping costs. However, at high temperatures, water exerts considerable corrosive action on steel, particularly if water contains dissolved oxygen (Peters et. al, 2004). Furthermore at high water temperature scaling tends to increase (Backhurst and Harker, 1973). To minimize scale formation, water temperature should not be more than 120ºF (Backhurst and Harker, 1973; Peters et. al, 2004). To protect against fouling and corrosion, water temperature (outlet) should not be heated above 158°F (Baasel, 1974). Again a good compromise has to be set between large quantity of cooling water and greater water temperature change.

For the cooling water, on an open circulation systems such as cooling towers and spray ponds, the temperature of the cooled water is 8-13ºF above the wet bulb temperature (Baasel, 1974). However since oxygen is picked-up in every pass, treatment of water is necessary if corrosion and growth of microorganism is to be controlled (Peters et. al, 2004).

When using cooling water to cool or condense a process stream, assume a water inlet temperature of 90^{o}F (from a cooling tower) and a maximum water outlet temperature of 120^{o}F (Seider et al, 2004).

As to the temperature difference, the rule of thumb is that the greatest temperature difference in an exchanger should be at least 36°F and the minimum temperature difference should be at least 10^{o}F hot (Lord et. al., 1970).