Fundamentals of Chemical Engineering Process Equipment Design

by

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Chapter 3

Heat transfer equipment

The transfer of heat to and from process fluids is an essential part of most chemical processes. The most commonly equipment used type of heat transfer equipment is the ubiquitous shell and tube heat exchanger.

3.1 Basic design procedure and theory

The general equation of heat transfer across a surface is:

$$Q = UA\Delta T_m \tag{3.1}$$

where Q is the heat transfer per unit time (W), U is the overall heat transfer coefficient ($W \cdot m^{-2} \cdot {}^oC^{-1}$), A is the heat transfer area (m^2), and ΔT_m is the mean temperature difference, the temperature driving force (^{o}C.).

The main point required for the heat exchanger design is the required surface area for the specified duty.

Different heat transfer resistance must be considered when specific heat transfer is required. The over all coefficient is the reciprocal of the over all resistance to heat transfer, which is the sum of the over all resistance to heat transfer.

The relation between the overall coefficient and the individual resistance are as follows:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{d_o ln(\frac{d_o}{d_i})}{2k_w} + \frac{d_o}{d_i} \times \frac{1}{h_{id}} + \frac{d_o}{d_i} \times \frac{1}{h_i} \tag{3.2}$$

where U_o is the overall coefficient based on the outside area of the tube (W·m^{-2}·^{o}C^{-1}), h$_o$ is the outside film coefficient (W·m^{-2}·^{o}C^{-1}), h$_i$ inside fluid film coefficient (W·m^{-2}·^{o}C^{-1}), h$_{od}$ outside dirt coefficient (W·m^{-2}·^{o}C^{-1}), h$_{id}$ is the inside dirt coefficient (W·m^{-2}·^{o}C^{-1}), k_w is the thermal conductivity of the tube wall material (W·m^{-1}·^{o}C^{-1}), d_i is the tube inside diameter (m), and d_o is the outside diameter (m).

The steps in a typical design procedure are:

1 Define the duty:heat transfer rate, fluid flow rates, temperatures

2 Collect together the fluid physical properties required: density, viscosity, thermal conductivity

3 Decide on the type of heat exchanger to be used

4 Select a trial value for the overall coefficient (U)

5 Calculate the mean temperature difference (ΔT_m)

6 Calculate the area required from Equation 3.1

7 Calculate the individual coefficient

8 Calculate the overall coefficient and compare with trial value. If the calculated value differs significantly from the estimated value, substitute the calculated for the estimated value and return to step 6

9 Calculate the exchanger pressure drop; if unsatisfactory return to steps 7 or 4 or 3 in that order of preference.

11 Optimise the design: repeat 4 to 10 as necessary to determine the cheapest exchanger that will satisfy the duty. (Note: smallest area is the cheapest)

3.2 Over all heat transfer coefficient

Typical values of the overall heat transfer coefficient are given in Figure 3.1. The values estimated from Figure 3.1 can be used for the preliminary sizing of equipment for detailed thermal design.

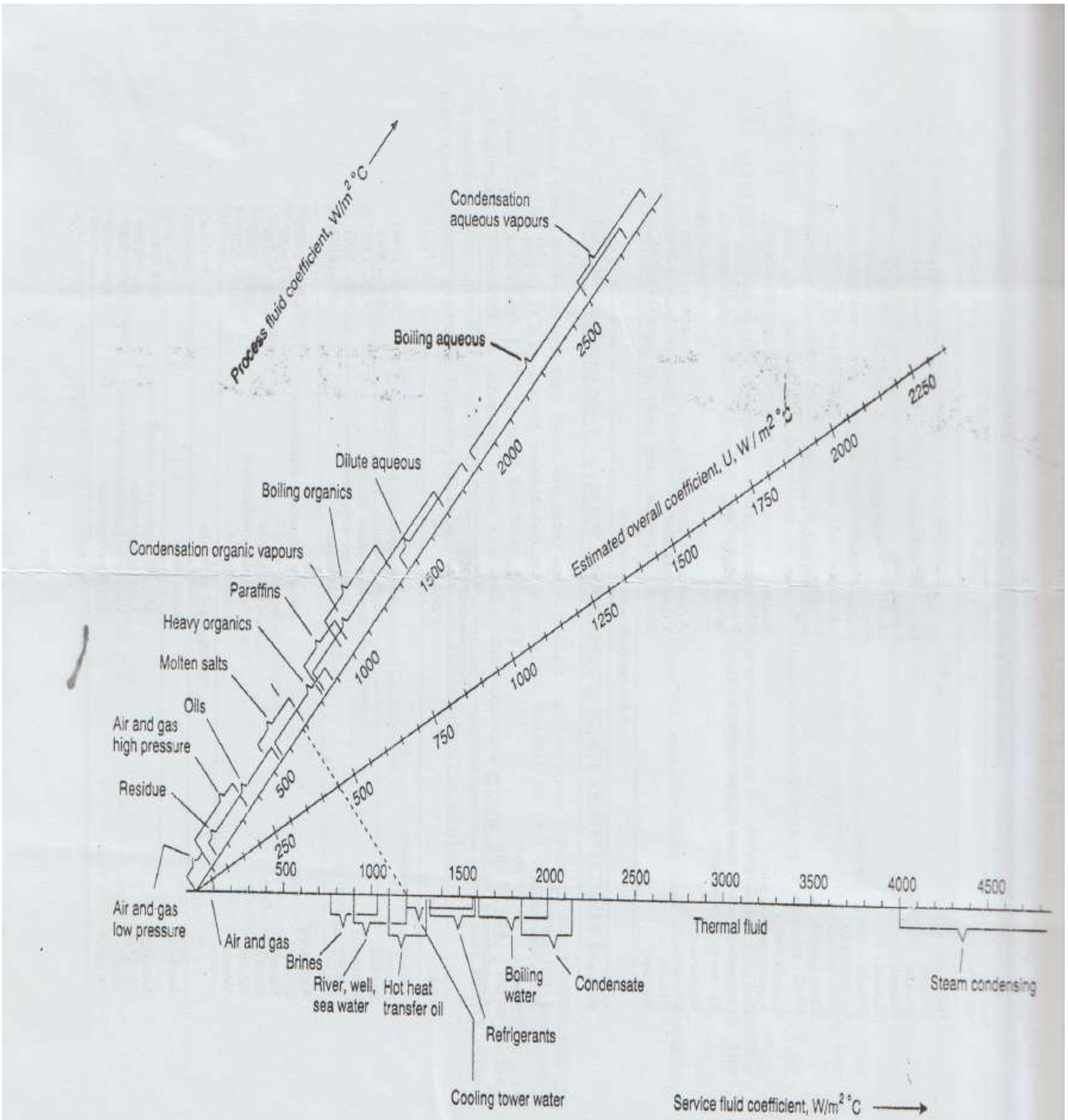

Figure 3.1: Overall coefficient (Join process side duty to service side and read U from centre scale)

3.3 Tubes

3.3.1 Dimensions

3.3.1.1 Diameter

Selection of tube diameters can be selected according to the duties required. The smaller diameter is 16-25 mm are preferred. Larger tubes are selected for heavily fouling liquid.

Note: Tube diameter, as guide, 19mm is a good trial diameter with which to start design calculation.

3.3.1.2 Thikness

Tube thickness is another consideration in the heat exchanger design. The selection of the thickness is chosen according to internal pressure and adequate corrosion allowance.

3.3.1.3 Materials

The standard of steel tubes of the heat exchangers are covered by the **BS 3606** and the standard applicable for other materials is **BS 3274**.

3.3.1.4 Length

The preferred tubes' length of heat exchanger are 1.83 m, 2.44m, 3.66m, 4.88m, 6.1m, 7.32m. The use of longer tube will reduce the shell diameter, which will result in a lower cost exchanger especially for high shell pressure.

3.3.2 Tube arrangments

Different arrangement of tubes are used in the heat exchanger design which are as follows: equilateral triangular, square, or rotated square pattern as shown in Figure 3.2.

The triangular and rotated square patterns gives higher heat transfer but in the expense of high pressure drop than the square pattern.The square or rotated square pattern are used for heavily fouling fluids(**Why?** where it is necessary to mechanically clean the outside of tubes.).

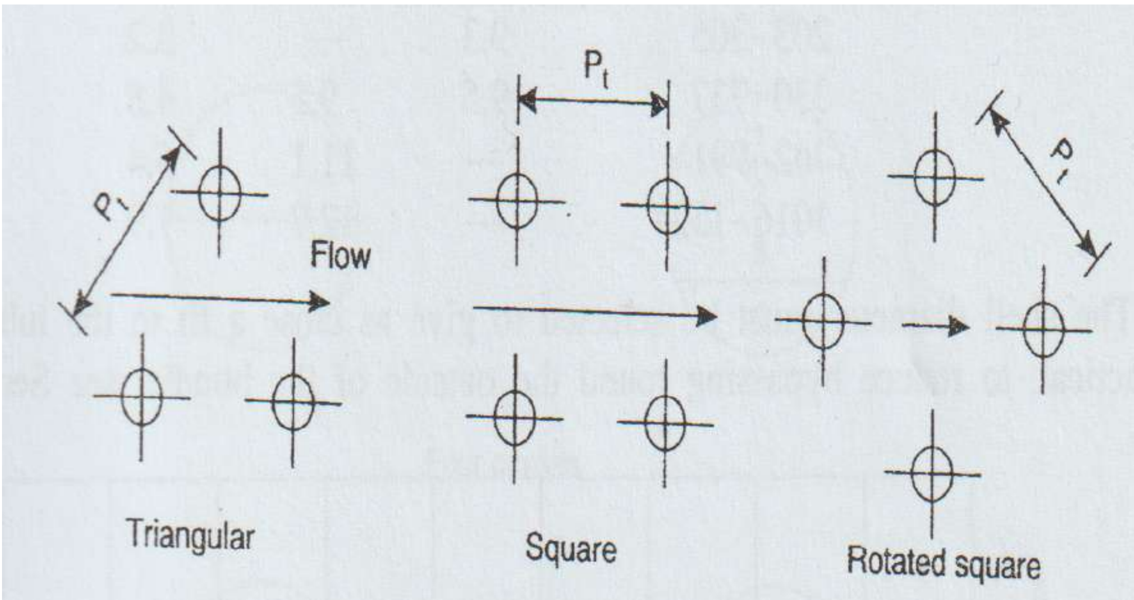

Figure 3.2: Tube patterns

Tube pitch is recommended to be 1.25 times the tube outside diameter. The square pattern recommended minimum clearance between the tubes to be 6.4 mm. The square pattern is used for ease of cleaning, the recommended clearance between the tubes is 6.4mm

3.3.3 Tube-side passes

In order to increase the tube length, number of passes are used. The fluid is usually directed back and forth for a number of passes. The arrangement of pass partitions for 2,4,and 6 tubes passes are shown in Figure 3.3

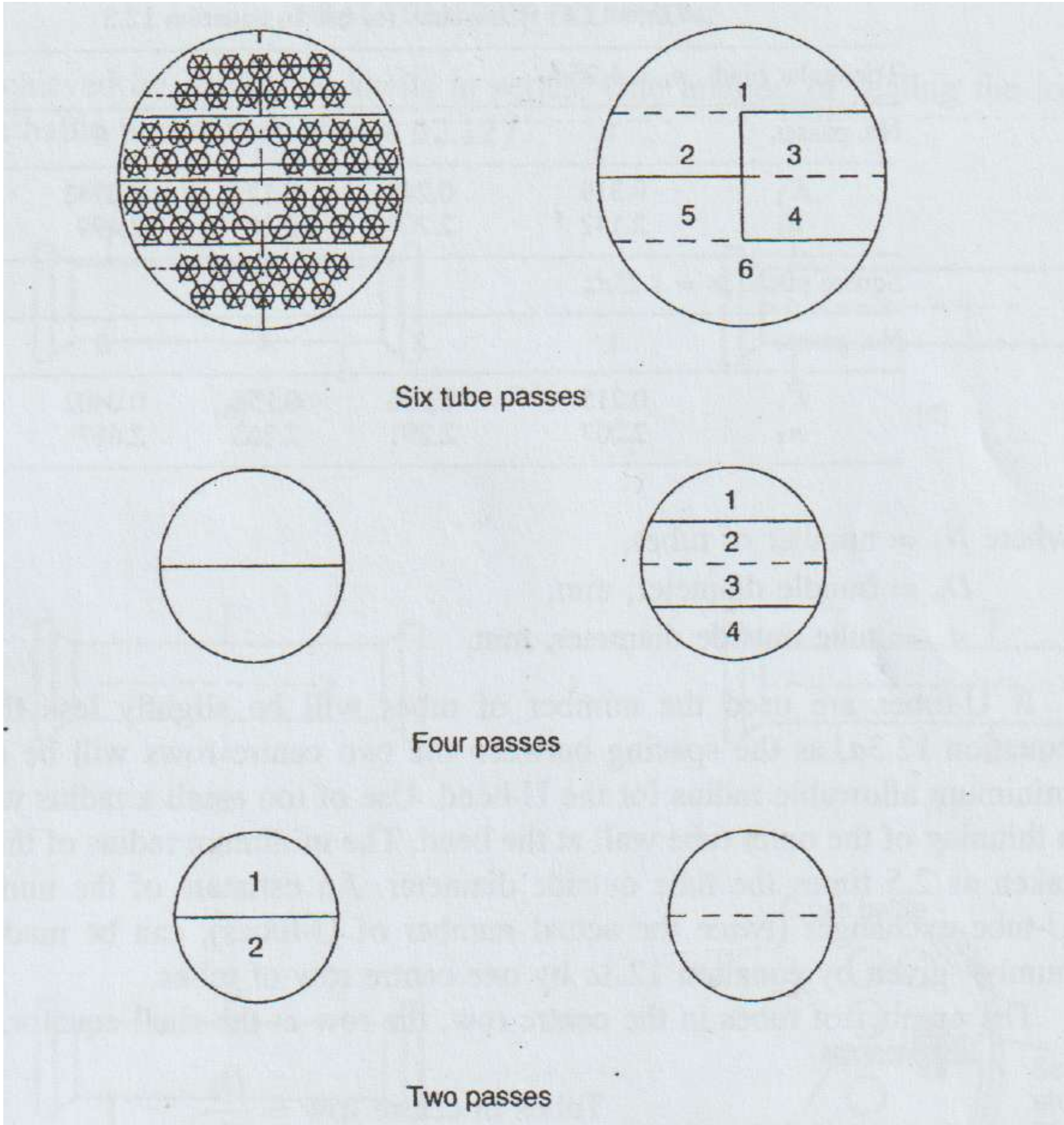

Figure 3.3: Tube arrangements, showing pass-partitions in headers

3.4 Shells

British standard (BS 3274) covers exchangers from 150 mm to 1067 mm diameter. The shell diameter must be selected to give as close fit to the tube bundle.

The clearance required between the outer most tubes in the bundle and the shell inside diameter will depend on the type of exchanger and the manufacturing tolerance. Typical values for the clearance are given in Figure 3.4.

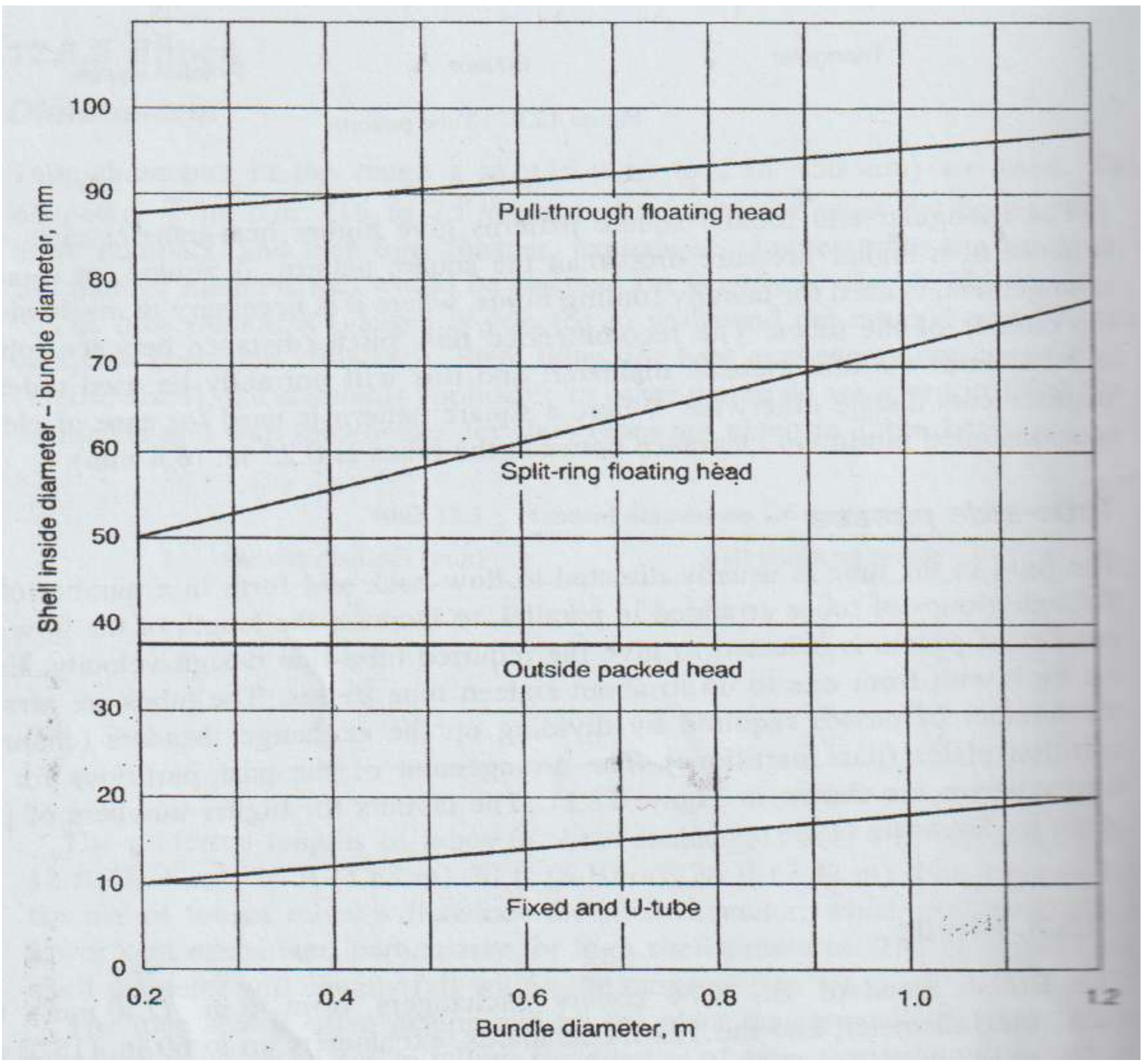

Figure 3.4: Shell-bundle clearance

3.5 Tube count

The bundle diameter will depend on the number of tubes and the number of passes since there is space must be left in the pattern of tubes to accommodate the pass partition plates.

To estimate the bundle diameter according to the Equation 3.3

$$D_b = d_o(\frac{N_t}{K_1})^{1/n_1} \tag{3.3}$$

where N_t is number of tubes, D_b is the bundle diameter (mm), and d_o is the tube outside diameter (mm).

where N_t can be estimated as follows:

$$N_t = K_1(\frac{D_b}{d_o})^{n_1} \tag{3.4}$$

Keep in mind:

- If (U-tubes) are used, the number of tubes will less than that estimated by Equation 3.3.

- Use of too small radius will cause too great thinning of the tube wall at the bend. In this case, the minimum radius of the bend is normally taken as 2.5 times the outside diameter.

- The number of tubes in the U-Tube exchangers can be reduced by one centre raw of tubes according to following Equation 3.5.

$$\text{Tubes in centre raw} = \frac{D_b}{P_t} \tag{3.5}$$

where P_t is the tube pitch.

Note: K_1 and n_1 can be estimated according to the Table 3.1 for triangular and square patterns

3.6 Shell types (passes)

The letters (E,F,G,H, and J) are letters used to in the TEMA standards to differentiate the various types of shells. The shell E is the most commonly used arrangements,

Table 3.1: Constants for use in tube counts

Triangular pitch, p_t=1.25d_o					
No. Passes	1	2	4	6	8
K_1	0.319	0.249	0.175	0.0743	0.0365
n_1	2.142	2.207	2.285	2.499	2.675
Square pitch, p_t=1.25d_o					
No. of passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

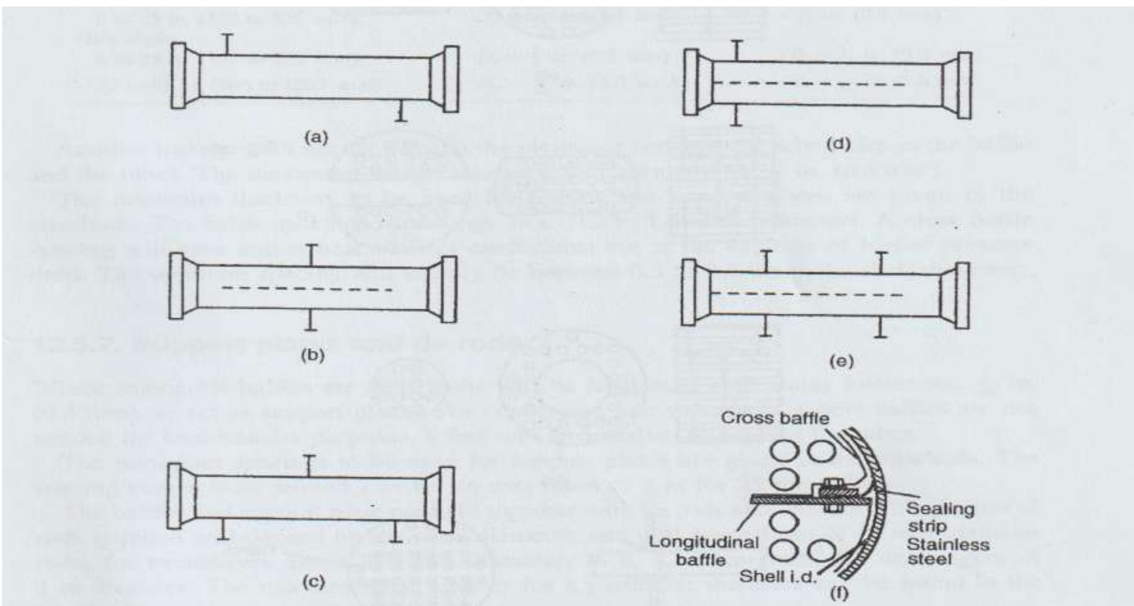

Figure 3.5: Shell type passes arrangements. (a) One-pass shell (E shell), (b) Split flow (G shell), (c)Divided flow (J shell), (d) Two-pass shell with longitudinal baffle(F Shell), (e) Double split flow (H shell)

3.7 Baffles

The purpose of baffles is to direct the fluid stream across the tubes to increase the velocity and the transfer rate.Different types of baffles are used (Figure 3.6). The most common one used is Figure 3.6a. The baffle cut is taken between (15-45)%, and the optimum baffle cut is between (20-25)%.

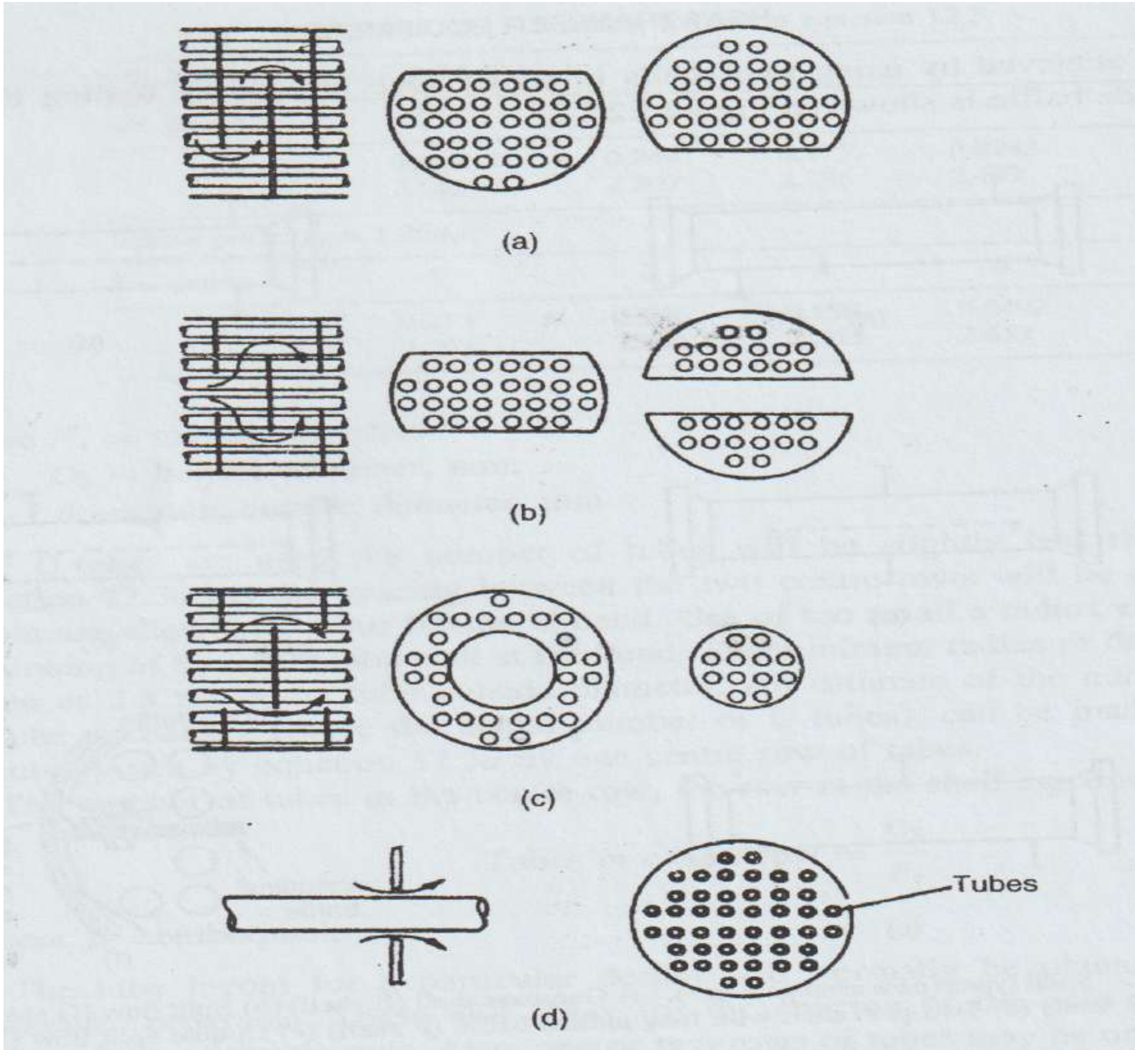

Figure 3.6: Types of baffles used in shell and tube heat exchangers. a. segmental, b. segmental and strip, c. disc and doughnuts, d. orifice

3.8 Mean temperature difference(Temperature driving force)

For counter-current flow (Figure 3.7), the logarithmic mean temperature difference is given by Equation B.1:

$$\Delta T_m = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln(\frac{(T_1 - t_2)}{(T_2 - t_1)}} \tag{3.6}$$

where:

ΔT_m= Log mean temperature difference

T_1= inlet shell-side fluid temperature

T_2= outlet shell-side fluid temperature

t_1= inlet tube-side temperature

t_2= outlet tube-side temperature

This equation is the same for co-current flow according to the following condition: 1- The terminal temperature difference is $(T_1 - t_1)$ and $(T_2 - t_2)$ 2- There is no change in the specific heats. 3- The overall heat transfer coefficient is constant. 4- There are no heat losses.

In most shell heat exchanger, the flow will be a mixture of co-current, counter current, and cross flow. Figure 3.7 and Figure 3.8 show typical temperature profile exchanger with one shell and 2 tube passes. Figure 3.9 shows the cross flow of exchangers.

The temperature difference is required to be corrected in order to get the "**True temperature difference** by correction factor to transfer the value from true counter-current flow:

$$\Delta T_m = F_t \Delta T_{Im} \tag{3.7}$$

where:

ΔT_m= true temperature difference which is used in Equation 3.1

F_t = the temperature correction factor which is estimated according to procedure below.

3.9 Correction factor for mean temperature difference

The correction factor for the mean temperature difference is a function of two dimensionless temperature ratios:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \tag{3.8}$$

R is equal to the shell-side fluid flow rate times the fluid mean specific heat, divided by the tube-side fluid flow rate times the tube-side fluid specific heat.

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)} \tag{3.9}$$

S is the measure of the temperature efficiency of the exchanger.

For 1 shell: 2tube passe exchanger, the correction factor is given by Equation 3.10 and is plotted in Figure 3.7:

$$F_t = \frac{\sqrt{(R^2+1)}\, ln\, [\frac{(1-S)}{(1-RS)}]}{(R-1)\, ln\, [\frac{2 - S[R+1-\sqrt{R^2+1}]}{2 - S[R+1+\sqrt{R^2+1}]}]} \tag{3.10}$$

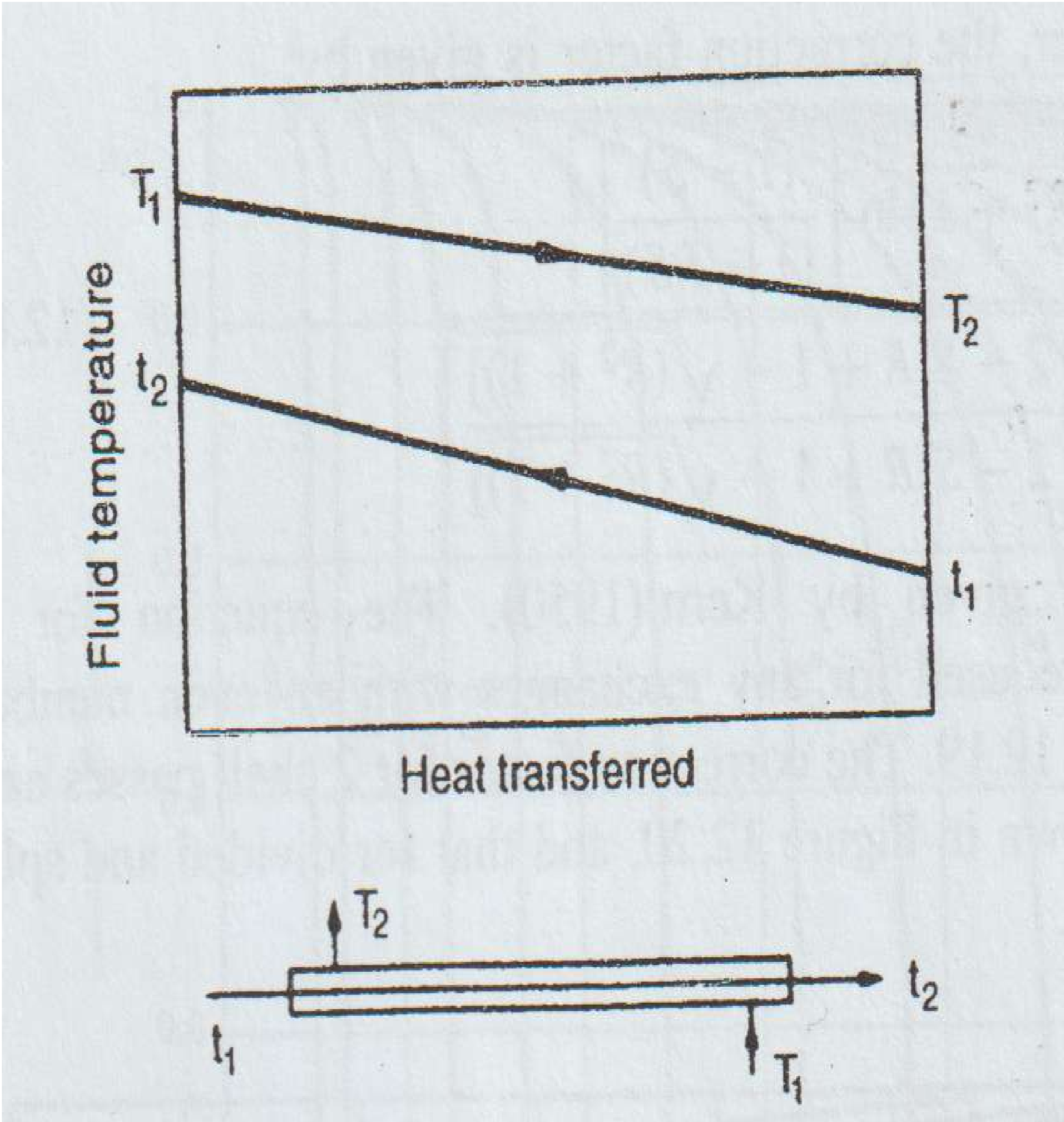

Figure 3.7: Temperature profile for counter current flow

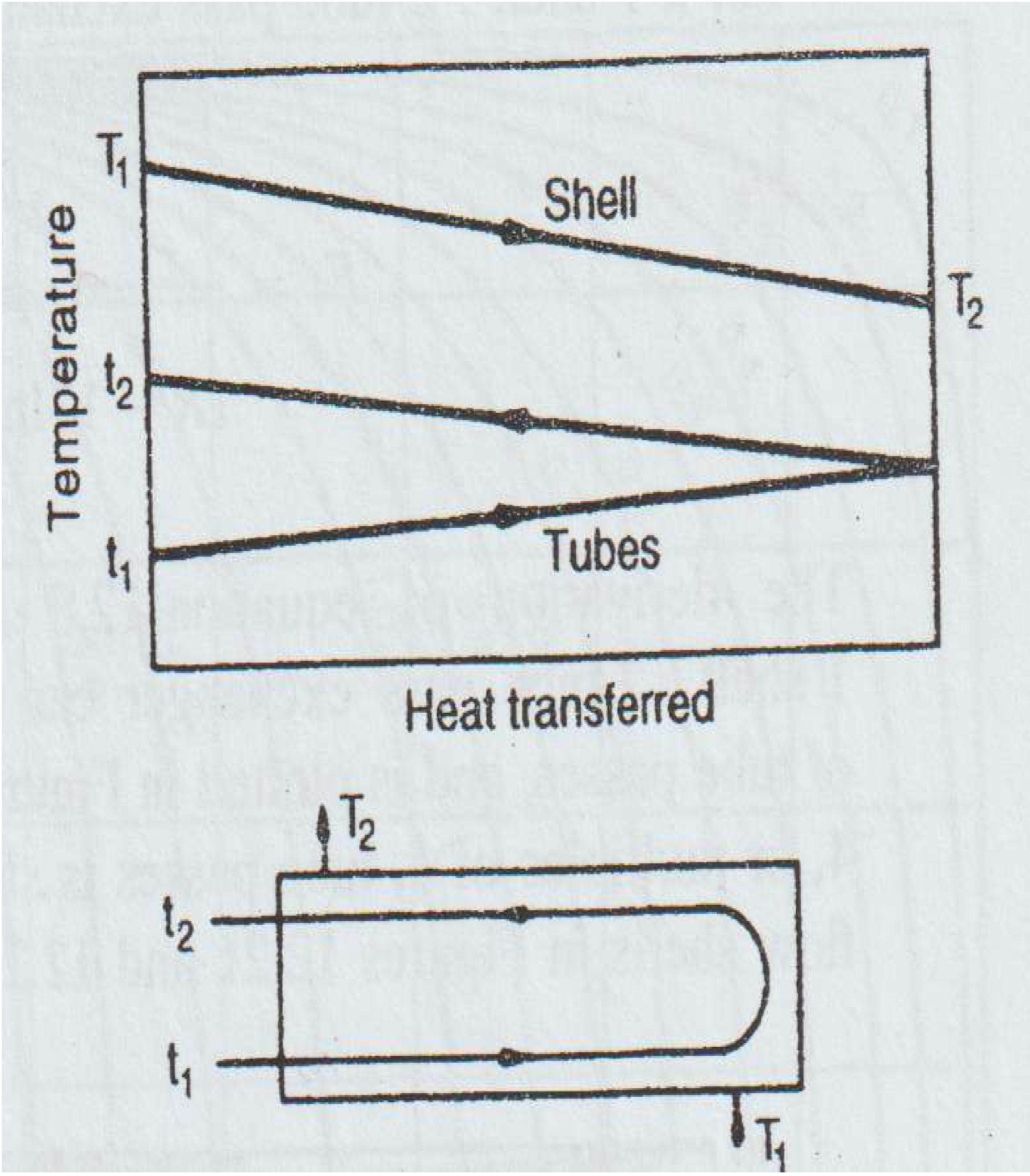

Figure 3.8: Temperature profile for 1:2 exchanger

3.10 Tube-side heat transfer coefficient and pressure drop

3.10.1 Heat transfer

3.10.1.1 Turbulent flow

Heat transfer inside conduits of uniform cross sectional area are correlated by Equation 3.11:

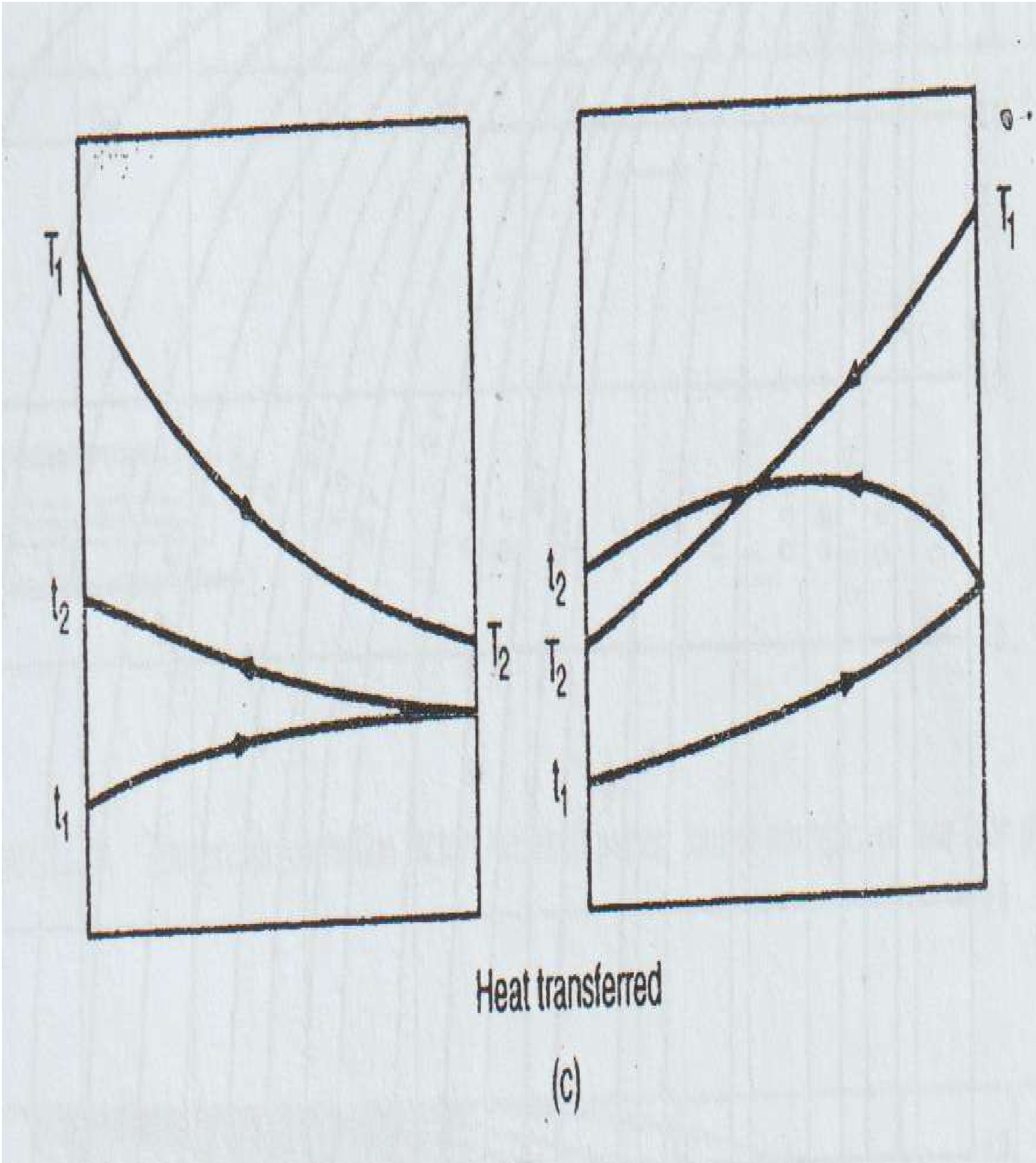

Figure 3.9: Temperature profile for cross flow

$$Nu = C\,Re^a Pr^b\,(\frac{\mu}{\mu_w})^c \tag{3.11}$$

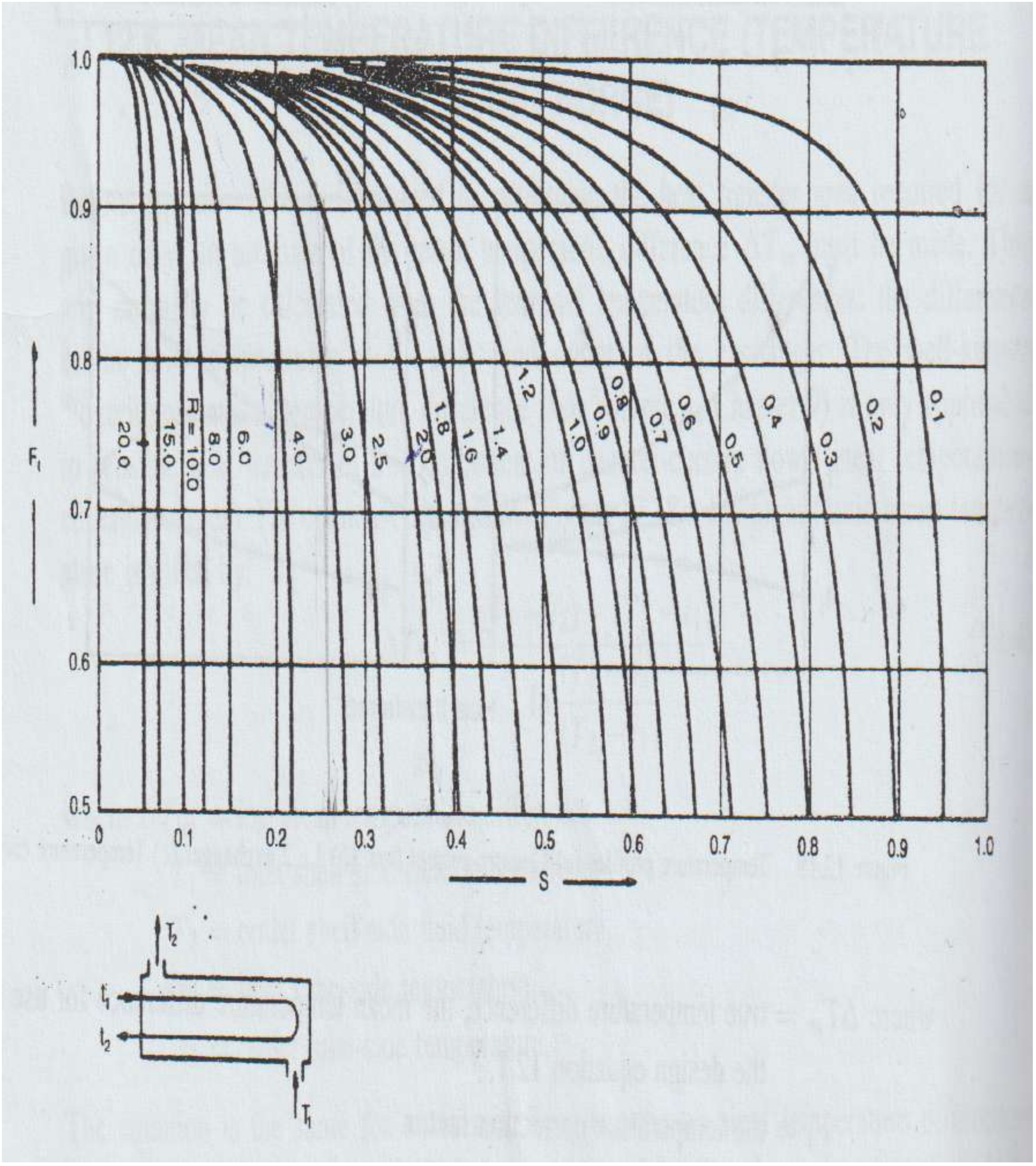

Figure 3.10: Temperature correction factor: one shell pass; two or more even tube passes

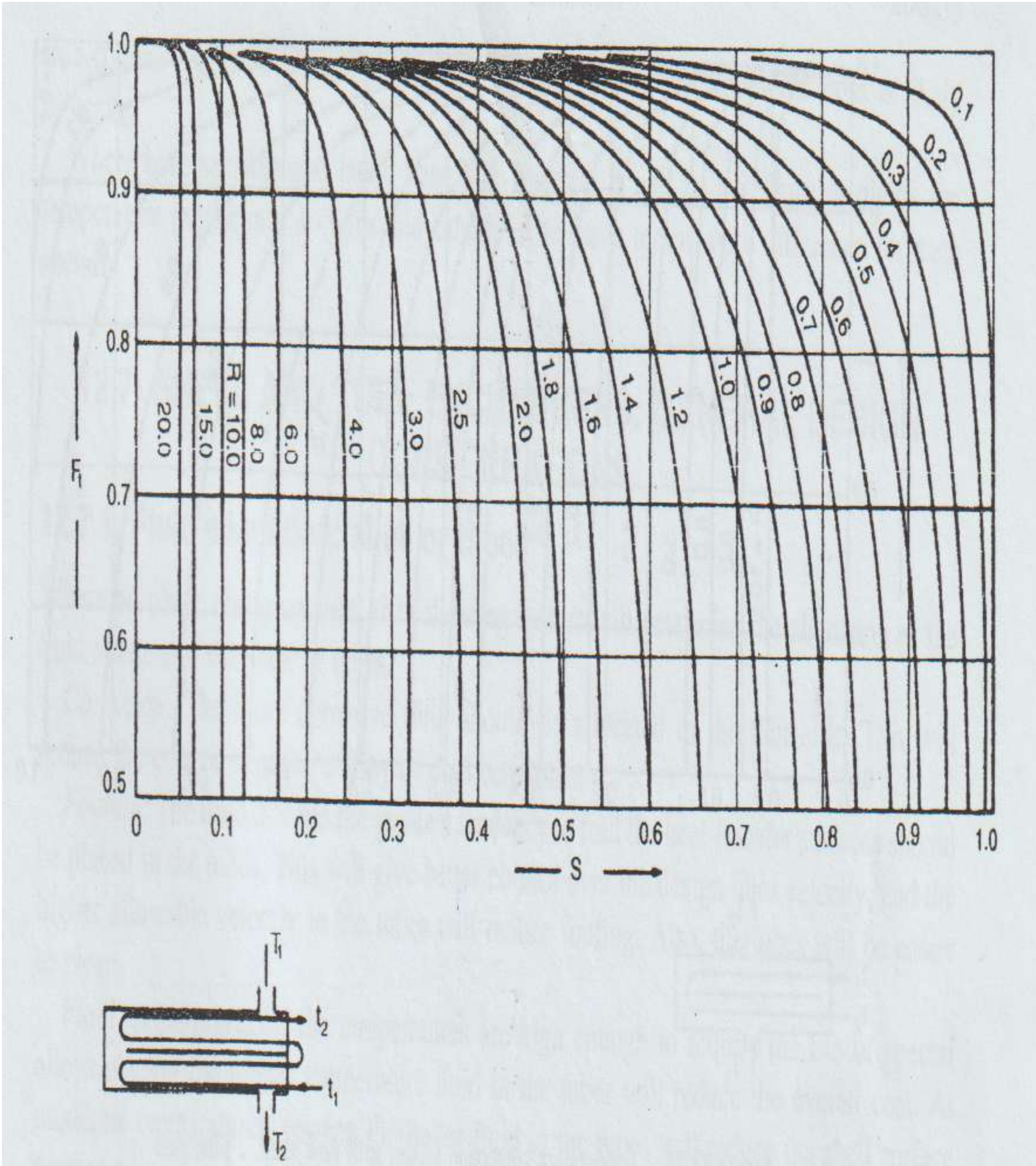

Figure 3.11: Temperature correction factor: two shell pass; four or multiples of four tube passes

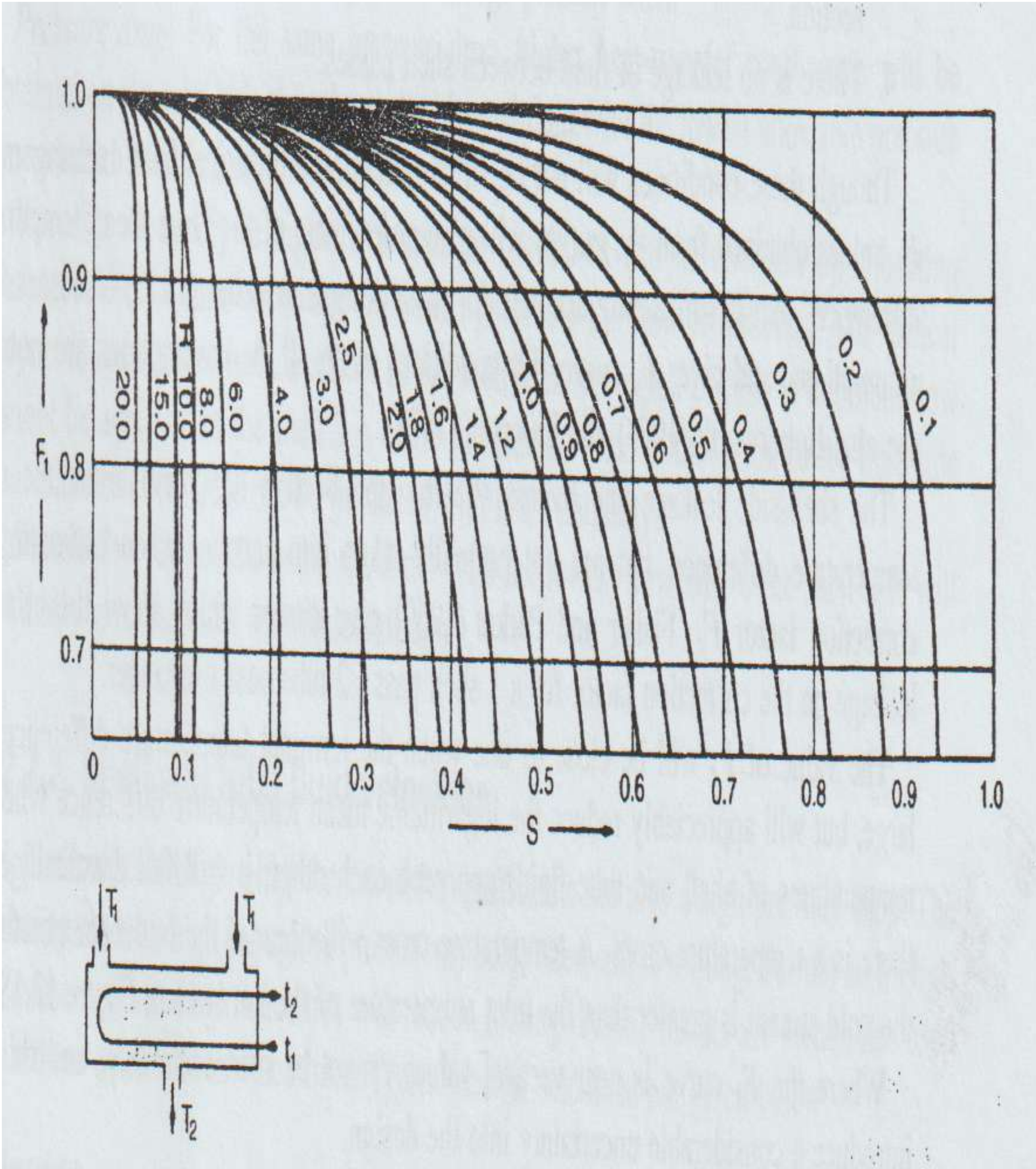

Figure 3.12: Temperature correction factor: divided-flow shell; two or more even-tube passes

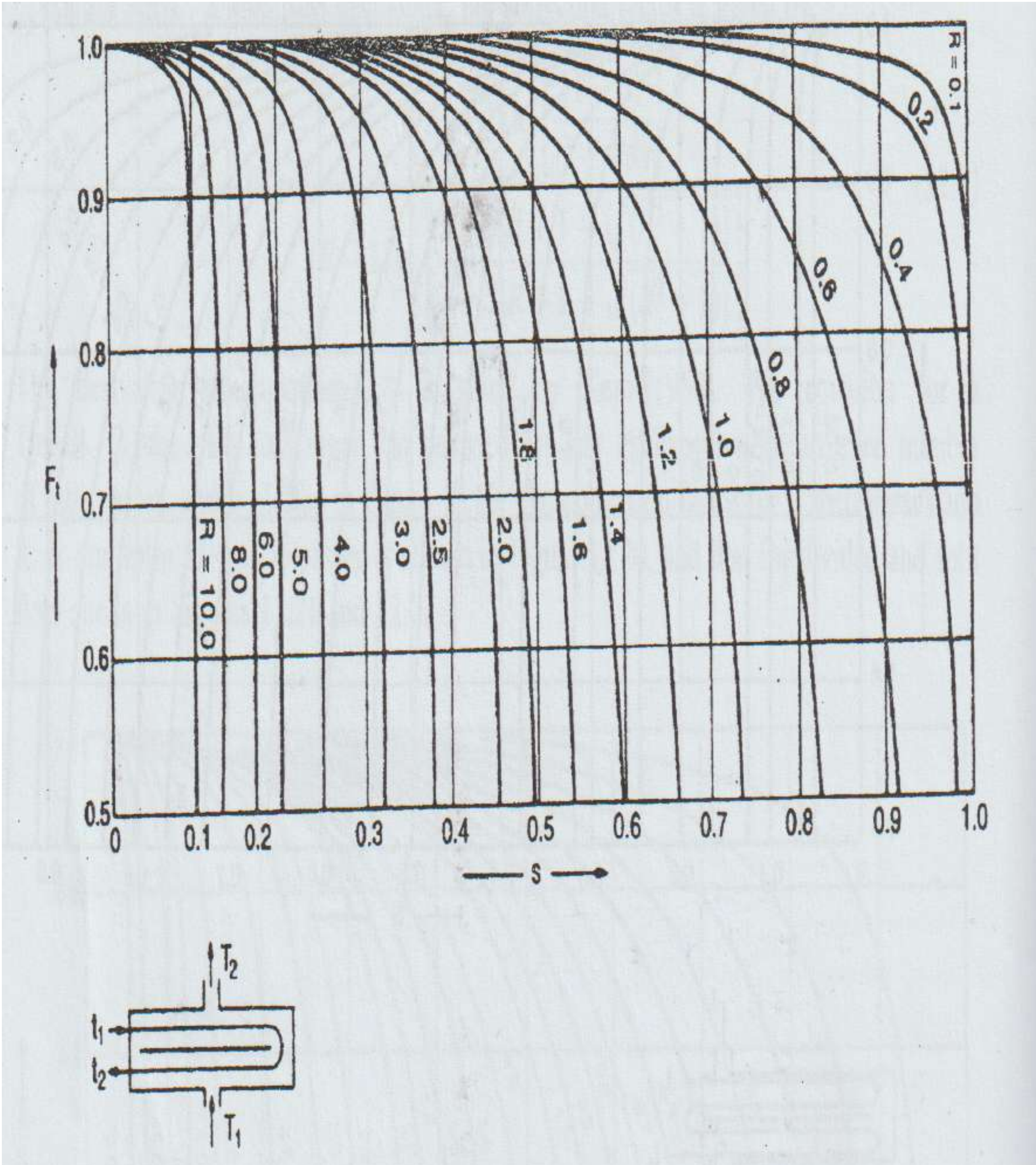

Figure 3.13: Temperature correction factor: split flow shell; 2 tube pass

where:

Nu= Nusselt number=$(h_i d_e/k_f)$

Re= Reynolds number=$(\rho u_t d_e/\mu)$=$(G_f d_e/\mu)$

Pr= Prandtl number=$(C_p\mu/k_f)$

h$_i$= inside coefficient, W·m^{-2}·^{o}C^{-1}

d$_e$= equivalent or hydraulic mean diameter (m)=$\dfrac{4 \times \text{cross-sectional area for flow}}{\text{Wetted perimeter}}$=d$_i$

u_t= fluid velocity (m·s^{-1})

k$_f$= fluid thermal conductivity (W·m^{-1}·^{o}C^{-1})

G$_t$= mass velocity, mass flow per unit area (kg·m^{-2}·s^{-1})

μ= fluid viscosity at the bulk fluid temperature (N·s·m^{-2})

μ_w= fluid viscosity at the wall

C_p= Heat capacity or fluid specific heat (J·kg^{-1}·^{o}C^{-1})

The index (a) for the Reynolds number is taken as 0.8. The index for Prandtl (b) can range from 0.3 for cooling and 0.4 for heating. For the viscosity ratio (c) is taken as 0.14.

$$Nu = C\,Re^{0.8}Pr^{0.33}\,(\frac{\mu}{\mu_w})^{0.14} \tag{3.12}$$

where C can be taken as 0.021 for gases, 0.023 for non-viscous liquids, and 0.027 for viscous liquids.

3.10.1.2 Laminar flow

For laminar flow (Reynolds Number<2000), Nusselt number can be estimated according to the following:

$$Nu = 1.86(RePr)^{0.33}(\frac{d_e}{L})^{0.33}(\frac{\mu}{\mu_w})^{0.14} \tag{3.13}$$

where L is the length of pipe in meters. (**Note**:) Nusselt number given by Equation 3.13 $<$ 3.5, Nusselt number must be taken as 3.5

3.10.1.3 Transition region

Transition region should be avoided in the design of exchangers.

3.10.2 Heat-transfer factor(j_h)

The heat transfer factor is given by Equation 3.14:

$$j_h = StPr^{0.67}(\frac{\mu}{\mu_w})^{-0.14} \tag{3.14}$$

where:

St= Stanton number = (Nu/RePr)=($h_i/\rho u_t C_p$)

Equation 3.14 can be rearranged and written as follows:

$$\frac{h_i d_i}{k_f} = j_h RePr^{0.33}(\frac{\mu}{\mu_w})^{0.14} \tag{3.15}$$

3.10.3 Viscosity correction factor

This factor is significant for viscous liquids. To estimate this factor, an estimate of the wall temperature temperature is needed. The wall temperature can be estimated according to the following relation:

$$h_i(t_w - t) = U(T - t) \tag{3.16}$$

where:

t= tube-side bulk temperature (mean)

t_w**=** estimated wall temperature

T= shell side bulk temperature (mean)

3.11 Tube-side pressure drop

Two major sources of pressure loss inside the tube: 1- Friction in the tubes 2- Friction due to sudden contraction and expansion

The basic Equation for estimating the friction in the **isothermal** flow in pipes according as follows:

$$\Delta P = 8\, j_f (\frac{(L')}{d_i})(\frac{\rho u_t^2}{2}) \tag{3.17}$$

The flow in the heat exchanger will be **not isothermal**, and the change in the physical properties must be considered. The ΔP can be estimated according to the following equation:

$$\Delta P = 8\, j_f (\frac{(L')}{d_i})\rho(\frac{u_t^2}{2})(\frac{\mu}{\mu_w})^{-m} \tag{3.18}$$

where:

m= 0.25 for laminar flow, Re<2100

m= 0.14 for turbulent flow, Re>2100

The loss in terms of velocity heads can be estimated by counting the number of flow contractions, contraction and reversal, and using the factors for pipe fittings to estimate the number of velocity head lost.

Combining this factor with Equation 3.18

$$\Delta P = N_p[8\, j_f (\frac{(L)}{d_i})(\frac{\mu}{\mu_w})^{-m} + 2.5](\frac{\rho u_t^2}{2}) \tag{3.19}$$

where:

ΔP= tube-side pressure drop (Pa)

N_p= number of tube side passes

u_t= tube-side velocity (m/s)

L= length of one tube

Note: j_h and j_f can be taken from Figure **??** and Figure **??**, respectively.

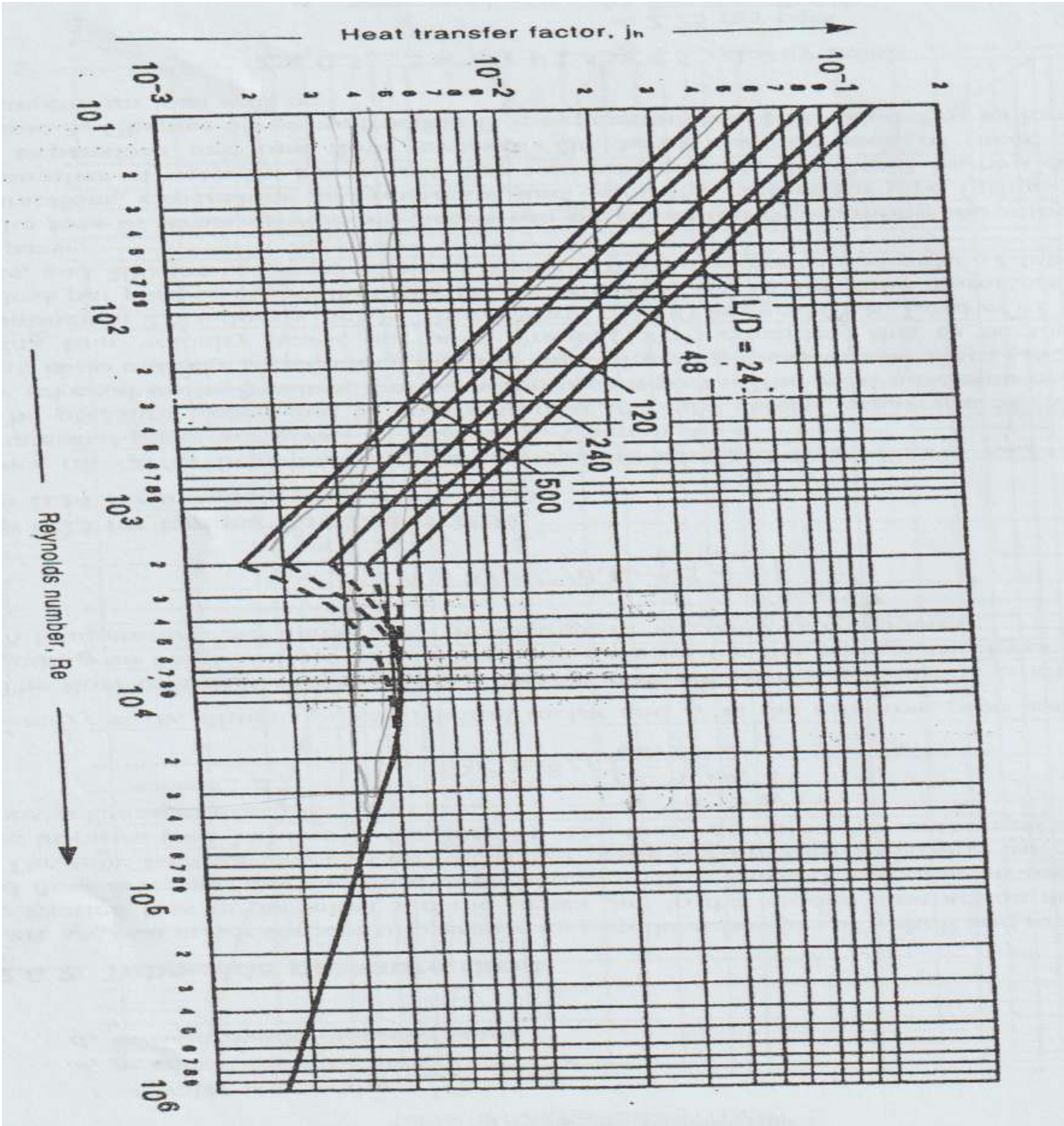

Figure 3.14: Tube side heat transfer factor

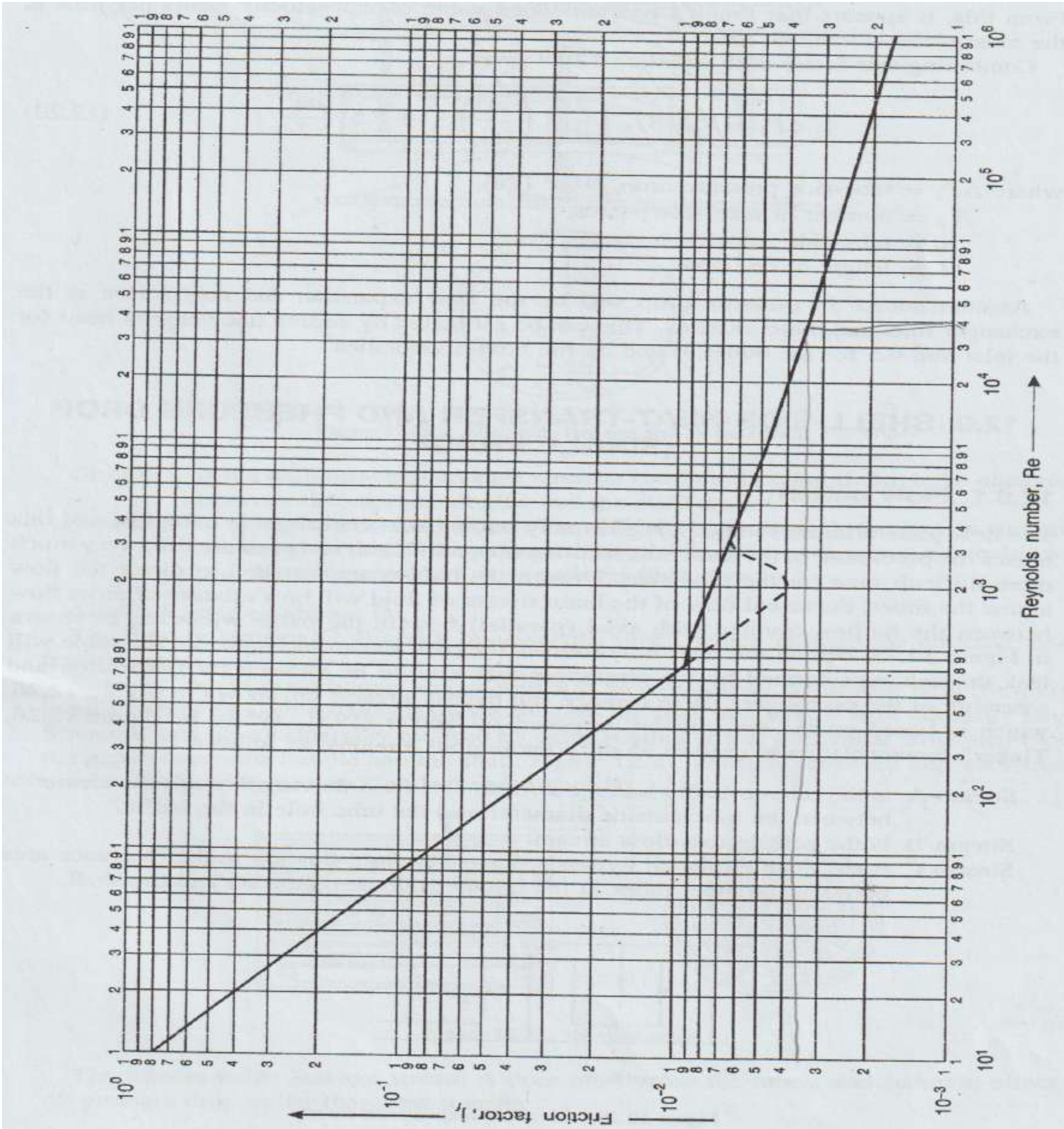

Figure 3.15: Tube side friction factor

3.12 Kern's method

This method was based on experimental work on commercial exchangers with standard tolerance and will give a reasonably satisfactory prediction of the heat transfer coefficient for standard design.

The prediction of pressure drop is less satisfactory, as pressure drop is more affected by leakage and bypassing than heat transfer.

The shell-side heat transfer coefficient and friction factors are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and shell diameter. The shell equivalent diameter is calculated using the flow area between the tubes taken in the axial direction(parallel to the tubes)and wetted perimeter of the tubes(see Figure **??**)

Procedure :

1- Calculate the area for cross-flow (A_s) for the hypothetical row of tubes at the shell equator, given by:

$$A_s = \frac{(P_t - d_o)D_s l_b}{P_t} \tag{3.20}$$

where:

$P_t =$ tube pitch

$d_o =$ tube outside diameter

$D_s =$ shell inside diameter

$l_B =$ baffle spacing

2- Calculate the shell-side mass velocity (G_S) and the linear velocity (u_s):

$$G_s = \frac{W_s}{A_s} \tag{3.21}$$

$$u_s = \frac{G_s}{\rho} \tag{3.22}$$

where:

$W_s =$ fluid-flow rate on the shell side (kg/s)

$\rho =$ shell-side fluid density (kg/m^3)

3- Calculate the shell-side equivalent diameter according to the tube patterns arrangements (Figure **??**)

a- Square pitch arrangements:

$$d_e = \frac{1.27}{d_o}(P_t^2 - 0.785d_o^2) \tag{3.23}$$

b- Equilateral triangular pitch arrangements:

$$d_e = \frac{1.10}{d_o}(P_t^2 - 0.917d_o^2) \tag{3.24}$$

where d_e equivalent diameter (m).

4- Calculate the shell side Reynolds number which is given by:

$$Re = \frac{G_s d_e}{\mu} = \frac{u_s d_e \rho}{\mu} \tag{3.25}$$

5- Read the value of j_h from Figure **??** for the selected baffle cut and tube arrangements, and calculate the shell-side heat transfer coefficient h_s from:

$$Nu = \frac{h_s d_e}{k_f} = j_h Re Pr^{\frac{1}{3}} (\frac{\mu}{\mu_w})^{-0.14} \tag{3.26}$$

The heat transfer coefficient for **water** can be estimated according to the following Equation 3.27. The physical properties are incorporated into the correlation. This equation has been adopted and given by Eagle and Ferguson:

$$h_i = \frac{4200(1.35 + 0.02 \times t)u_t^{0.8}}{d_i^{0.2}} \tag{3.27}$$

where h_i inside coefficient for water (W·m^{-2}·^{o}C^{-1})

6- Read the friction factor from Figure **??**, and calculate the pressure drop according to the following:

$$\Delta P = 8\, j_f(\frac{(D_s)}{d_e})(\frac{L}{l_B})(\frac{\rho u_s^2}{2})(\frac{\mu}{\mu_w})^{-0.14} \tag{3.28}$$

where:

$L =$ Tube length

l_B Baffle spacing

Example:

Design an exchanger to sub-cool condense from a methanol condenser from 95^{o}C to 40^{o}C. Flow rate of methanol 100,000 kg/h. Brackish water will be used as the coolant, with temperature rise from 25^{o}C to 40^{o}C.

Solution:

Coolant is corrosive, so assign to tube-side.

Note:

- Decide which fluid in the shell or tubes. In our case, methanol will pass through the shell and water through the tube since the effect of water will appear on the inside diameter. - Physical properties of fluids can be found in the appendices of Volume 6.

Heat capacity methanol=2.84 kJ·kg^{-1}·^{o}C.

1- Heat load can be estimated from mass flow rate, heat capacity of methanol, and temperature difference of methanol according to the following:

2- Estimate the mean temperature difference after choosing the type of heat exchanger: In our case we will choose heat exchanger with one shell and two tube passes, The mean temperature difference will be estimated according to the following:

3- Correct the mean temperature difference according to the following procedure: a- estimate R and S according to the following:

b- Find the correction factor according to the type of heat exchanger, R, and S