KATHMANDU UNIVERSITY
End Semester Examination
August, 2018

Marks scored:

Level : B.E.	Course : CHEG 314
Year : III	Semester: II
Exam Roll No.: Time: 30 mins.	F.M. : 10
Registration No.:	Date AUG 2 1 2018

SECTION "A"
[20 Q.×0.5=10 marks]

Encircle the most appropriate answer.

1. Which one of the following statements about baffles in a shell and tube heat exchanger is false? Baffles
   a. act as a support to the tube bundle
   b. reduce the pressure drop on the shell-side
   c. alter the shell-side flow pattern
   d. help in increasing the shell-side heat transfer coefficient

2. The following list of options P, Q, R and S are some of the important considerations in the design of a shell and tube heat exchanger.
   P: Square pitch permits the use of more tubes in a given shell diameter.
   Q: The tube side clearance should not be less than one-fourth of the tube diameter.
   R: Baffle spacing is not greater than the diameter of the shell or less than one fifth of the shell diameter.
   S: The pressure drop on the tube side is less than 10 psi.
   Pick out the correct combination of true statements from the following
   a. P, Q and R b. Q, R and S c. R, S and P d. P, Q, R and S

3. Baffles are used in heat exchangers in order to
   a. increase the tube side fluid's heat transfer coefficient
   b. promote vibration in the heat exchanger
   c. promote cross flow and turbulence in the shell side fluid
   d. prevent shell expansion due to thermal effects

4. If the baffle spacing in a shell and tube heat exchanger increases, then the Reynolds number of the shell side fluid
   a. remains unchanged
   b. increases
   c. increases or decreases depending on number of shells passes
   d. decreases

5. The flooding velocity in a plate column, operating at 1 atm pressure is 3 m/s. If the column is operated at 2 atm pressure under otherwise identical conditions, the flooding velocity will be
   a. $3/\sqrt{2}$ b. 3/2 c. 1 d. 3/4

6. The feed to a binary distillation column has 40 mol% vapour and 60 mol% liquid. Then, the slope of the q -line in the McCabe-Thiele plot is
   a. –1.5 b. –0.6 c. 0.6 d. 1.5

7. In a tray column, separating a binary mixture, with non-ideal stages, which one of the following statements is true?
a. Point efficiency can exceed 100%
b. Murphree efficiency cannot exceed 100%
c. Murphree efficiency can exceed 100%
d. Both Murphree and point efficiencies can exceed 100%

8. Minimum reflux ratio in a distillation column results in
a. optimum number of trays
b. minimum reboiler size
c. maximum condenser size
d. minimum number of trays

9. Multiple effect evaporators are used to
a. increase the steam economy & decrease the capacity
b. increase the steam economy & the capacity
c. decrease the steam economy & the capacity
d. decrease the steam economy & increase the capacity

10. In distillation column sizing calculations by short cut methods, match the following.

Column I	**Column II**
P. Underwood's equation	1. Number of real trays
Q. Fenske's equation	2. Colum diameter
R. Gilliland's equation	3. Minimum number of ideal trays
S. Vapour velocity at flooding	4. Actual number of ideal trays
	5. Minimum reflux ratio
	6. Tray efficiency

a. P-1, Q-3, R-4, S-6
b. P-2, Q-5, R-1, S-3
c. P-5, Q-3, R-6, S-2
d. P-5, Q-3, R-4, S-2

11. If a single tube passes heat exchanger is converted to two passes; for the same flow rate, the pressure drop per unit length in tube side will __________ times.
a. increase by $1^{0.8}$
b. decrease by 2^2
c. increase by $2^{1.6}$
d. decrease by 2^3

12. Tube side heat transfer co-efficient for turbulent flow of liquid through tubes is proportional to
a. $G^{0.2}$
b. $G^{0.5}$
c. $G^{0.8}$
d. $G^{1.5}$

13. In the agitators, the power required will be changed with the increase of diameter of agitator (D) as
a. D^2
b. D^5
c. D
d. D^9

14. If moisture content of solid on dry basis is X, then the same on wet basis is
a. $\frac{X}{X+1}$
b. $\frac{X}{X-1}$
c. $\frac{X+1}{X}$
d. $\frac{1-X}{X}$

15. The inside heat transfer co-efficient in case of turbulent flow of liquid in the tube side in a 1-2 shell and tube heat exchanger is increased by __________ times, when the number of tube passes is increased to 8.

a. $2^{0.8}$ b. $4^{0.8}$ c. $4^{0.4}$ d. $2^{0.4}$

16. For condensation of pure vapors, if the heat transfers co-efficient in film wise and drop-wise condensation are respectively h_f and h_d, then

a. $h_f = h_d$ b. $h_f > h_d$
c. $h_f < h_d$ d. h_f could be greater or smaller than h_d

17. Peclet number (Pe) is given by

a. Pe = Re.Pr b. Pe = Re/P_r c. $P_e = P_r/R_e$ d. Pe = Nu.Re

18. The unit of heat transfer co-efficient in SI unit is

a. J/M^2,^{0}K b. W/m^2.^{0}K c. W/m.^{0}K d. J/m.^{0}K

19. Steam economy in case of a triple effect evaporator will be

a. 1 b. <1 c. >1 d. between 0 and 1

20. Which is the best tube arrangement (in a shell and tube heat exchanger) if the fluids are clean and non-fouling?

a. Square pitch b. Triangular pitch
c. Diagonal square pitch d. Rectangular pitch

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SECTION "B"

Attempt *ALL* questions.

1. a. What is meant by weeping in a distillation column? How can it be eliminated? [5×1= 5]
   b. What is meant by steam economy in the design of evaporators?
   c. What are the differences between condenser and a reboiler?
   d. Why low pressure steam is used in evaporators?
   e. What is Entrainment and Weep Point?

2. A binary feed mixture containing equimolar quantities of components S and T is to be distilled in a fractionating tower at atmospheric pressure. The distillate contains 96 mol% S. The q –line (feed line) intersects the equilibrium line at $x' = 0.46$ and $y' = 0.66$, where x' and y' are mole fractions. Assume that the McCabe-Thiele method is applicable and the relative volatility is constant. Find the minimum reflux ratio. [5]

3. Draw a neat sketch of a triple effect evaporation with all accessories and write the mass and energy balance equations. [5]

   OR

   Draw a neat sketch of a distillation column with all the necessary accessories shown.

4. Oil at 120 °C is used to heat water at 30°C in a 1-1 cocurrent shell and tube heat exchanger. The available heat exchange area is $S1$. The exit temperatures of the oil and the water streams are 90°C and 60°C respectively. The concurrent heat exchanger is replaced by a 1-1 countercurrent heat exchanger having heat exchange area $S2$. If the exit temperatures and the overall heat transfer coefficients are same, find the ratio of $S1$ to $S2$. [5]

5. A sieve-plate column operating at atmospheric pressure is to produce nearly pure methanol from an aqueous feed containing 50 mol percent methanol. The distillate product rate is 5000 kg/h. The molecular weight and boiling point of methanol are 32 and 65°C respectively, density and surface tension of methanol at 65°C are 750 kg/m^3 and 19 dyn/cm respectively. [3×2=6]
   (a) For a reflux ratio of 3 and a plate spacing of 18 in., calculate the allowable vapor velocity and the coulmn diameter.
   (b) Calculate the pressure drop per plate if each sieve tray is 1/8 in, thick with ¼-in. holes on a 3/4-in. triangular spacing and a weir height of 2 in.
   (c) What is the fourth height in the downcomer?

6. Gas oil at 200°C is to be cooled to 40°C. The oil flow rate is 22,500 kg/h. Cooling water is available at 30°C and the temperature rise is to be limited to 20°C. The pressure drop allowance for each stream is 100 kN/m^2. Design a suitable shell and tube heat exchanger and write the summary of proposed design. [8]

Physical properties:
Water

Temperature (°C)	30	40	50
Specific heat (kJ/kg.°C)	4.18	4.18	4.18
Thermal conductivity (W/m.°C)	0.618	0.631	0.643
Density (kg/m^3)	995.2	992.8	990.1
Viscosity (mNm^{-2}.s)	0.797	0.671	0.544

Gas oil

Temperature (°C)	200	120	40
Specific heat (kJ/kg.°C)	2.59	2.28	1.97
Thermal conductivity (W/m.°C)	0.13	0.125	0.12
Density (kg/m^3)	830	850	870
Viscosity (mNm^{-2}.s)	0.06	0.17	0.28

OR

Design a shell and tube exchanger for the following duty and write the summary of proposed design.

Kerosene, 20,000 kg/h (42° API), leaves the base of a kerosene side-stripping column at 200°C and is to be cooled to 90 °C by exchange with 70,000 kg/h light crude oil (34° API) coming from storage at 40 °C. The kerosene enters the exchanger at a pressure of 5 bar and the crude oil at 6.5 bar. A pressure drop of 0.8 bar is permissible on both streams. Allowance should be made for fouling by including a fouling factor of 0.0003 (W/m^2°C)$^{-1}$ on the crude stream and 0.0002 (W/m^2°C)$^{-1}$ on the kerosene stream.

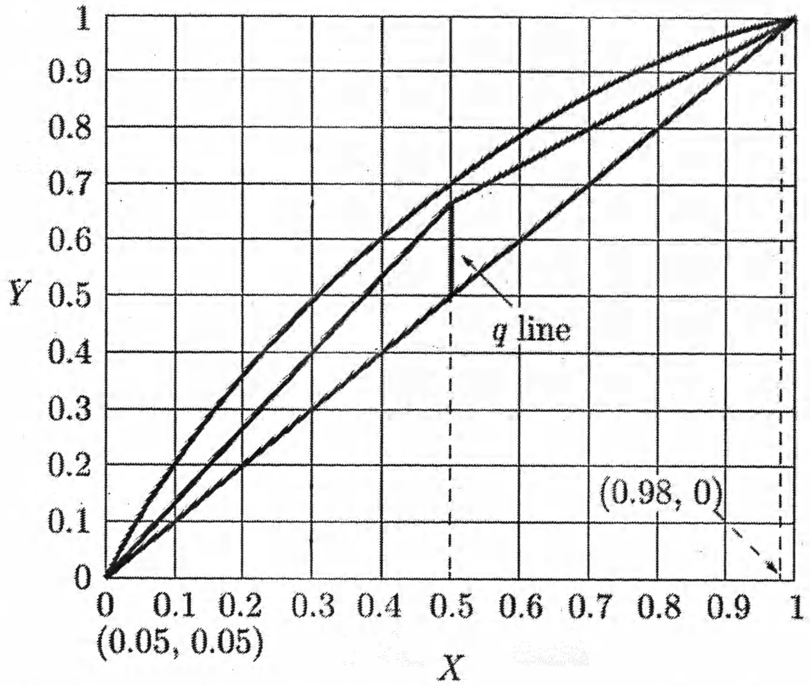
Physical properties:
Kerosene

Temperature (°C)	200	145	90
Specific heat (kJ/kg.°C)	2.72	2.47	2.26
Thermal conductivity (W/m.°C)	0.130	0.132	0.135
Density (kg/m^3)	690	730	770
Viscosity (mNm^{-2}.s)	0.22	0.43	0.80

Crude oil

Temperature (°C)	78	59	40
Specific heat (kJ/kg.°C)	2.09	2.05	2.01
Thermal conductivity (W/m.°C)	0.133	0.134	0.135
Density (kg/m^3)	800	820	840
Viscosity (mNm^{-2}.s)	2.4	3.2	4.3

7. A binary distillation column separates 100 mol/h of a feed mixture into distillate D and residue W. The McCabe-Thiele diagram for this process is given below. The relative volatility for the binary system is constant at 2.4. [3×2=6]



a. Find the distillate and residue flow rates (in mol/h).
b. What is the ratio of liquid to vapour molar flow rates in the rectifying section?
c. What is the number of theoretical stages (inclusive of reboiler) for this process?

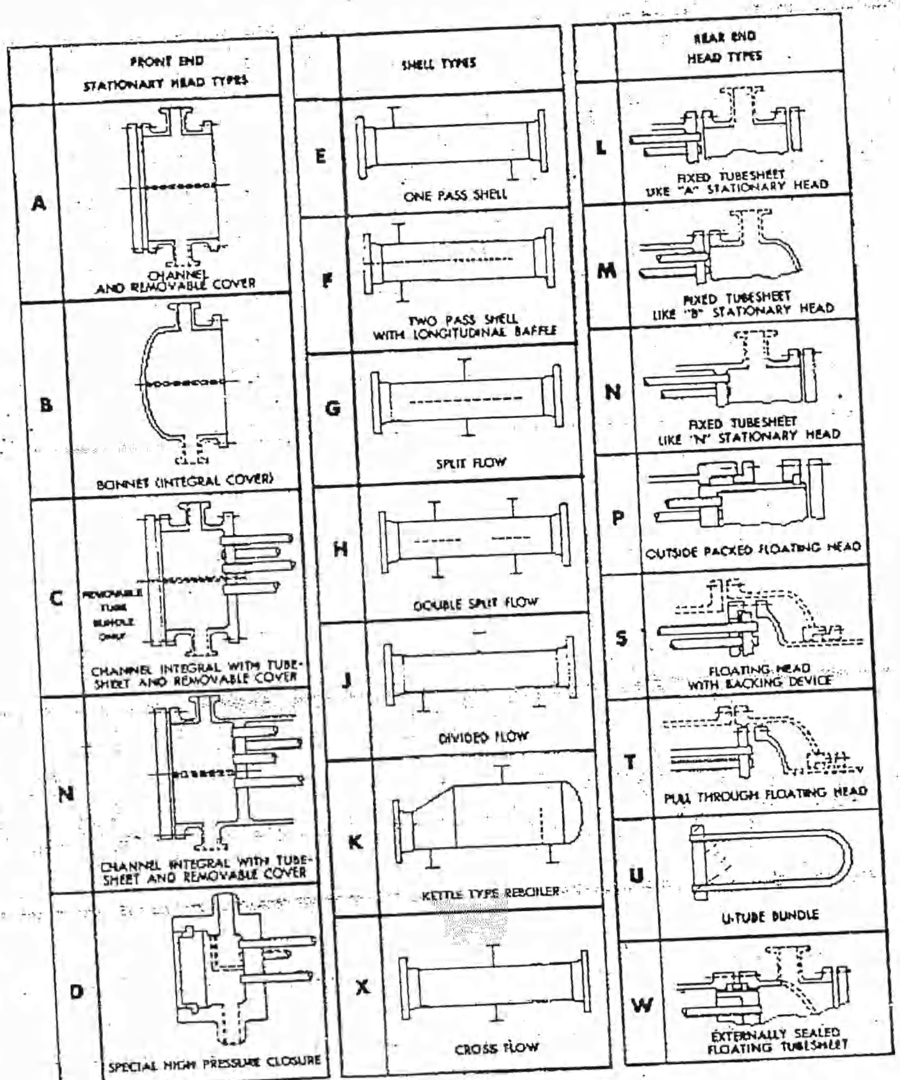


Figure 12.9. TEMA designations for shell and tube heat exchangers (reproduced with permission from the Tubular Exchanger Manufacturers Association).

thicknesses correspond to even-numbered B.W.G. (Birmingham Wire Gauge) units. Standard diameters and wall thicknesses for steel tubes are given in Table 12.3.

The preferred lengths of tubes for heat exchangers are: 6 ft (1.83 m), 8 ft (2.44 m), 12 ft (3.66 m), 16 ft (4.88 m), 20 ft (6.10 m) and 24 ft (7.32 m). For a given surface area, the use of longer tubes will reduce the shell diameter. This will generally result in a lower cost exchanger, particularly for high shell pressures, but will lead to an

TABLE 12.3. Standard Dimensions for Steel Tubes

Outside Diameter (mm)	Wall Thickness (mm)				
16	1.2	1.7	2.1	—	—
19	—	1.7	2.1	2.8	—
25	—	1.7	2.1	2.8	3.4
32	—	1.7	2.1	2.8	3.4
38	—	—	2.1	2.8	3.4
50	—	—	2.1	2.8	3.4

increase in pressure drop and pump work. The optimum tube length to shell diamet ratio will usually fall within the range of 5 to 10.

If U-tubes are used, the tubes on the outside of the bundle will be longer than tho on the inside. The average length needs to be estimated for use in the thermal desi U-tubes will be bent from standard tube lengths and cut to size.

The tube size is often determined by the plant maintenance department standa as clearly it is an advantage to reduce the number of sizes that have to be held in sto for tube replacement.

As a guide, ¾ in (19 mm) is a good trial diameter with which to start desi calculations.

Tube Arrangements

The tubes in an exchanger are usually arranged in an equilateral triangular, square, rotated square pattern; see Figure 12.10.

The triangular and rotated square patterns give higher heat-transfer rates, but the expense of a higher pressure drop than the square pattern. A square, or rota square arrangement, is used for heavily fouling fluids, where it is necessary mechanically clean the outside of the tubes. The recommended tube pitch (dista between tube centres) is 1.25 times the tube outside diameter, and this will norm be used unless process requirements dictate otherwise. Where a square pattern is for ease of cleaning, the recommended minimum clearance between the tub 0.25 in (6.4 mm).

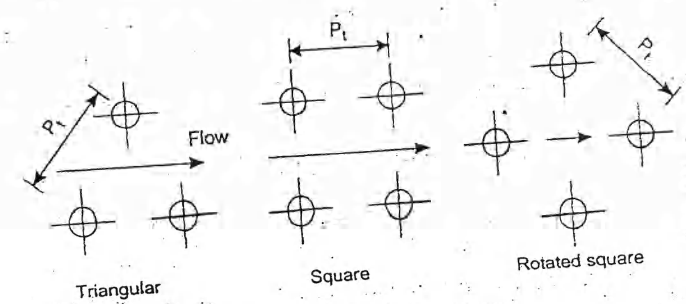


Figure 12.10. Tube patterns.

Table 12.1. Typical Overall Coefficients—*Cont'd*

Immersed Coils		
Coil	**Pool**	***U* (W/m² °C)**
Natural circulation		
Steam	Dilute aqueous solutions	500–1000
Steam	Light oils	200–300
Steam	Heavy oils	70–150
Water	Aqueous solutions	200–500
Water	Light oils	100–150
Agitated		
Steam	Dilute aqueous solutions	800–1500
Steam	Light oils	300–500
Steam	Heavy oils	200–400
Water	Aqueous solutions	400–700
Water	Light oils	200–300

Jacketed Vessels		
Jacket	**Vessel**	
Steam	Dilute aqueous solutions	500–700
Steam	Light organics	250–500
Water	Dilute aqueous solutions	200–500
Water	Light organics	200–300

Gasketed-Plate Exchangers		
Hot Fluid	**Cold Fluid**	
Light organic	Light organic	2500–5000
Light organic	Viscous organic	250–500
Viscous organic	Viscous organic	100–200
Light organic	Process water	2500–3500
Viscous organic	Process water	250–500
Light organic	Cooling water	2000–4500
Viscous organic	Cooling water	250–450
Condensing steam	Light organic	2500–3500
Condensing steam	Viscous organic	250–500
Process water	Process water	5000–7500
Process water	Cooling water	5000–7000
Dilute aqueous solutions	Cooling water	5000–7000
Condensing steam	Process water	3500–4500

Figure 12.1, which is adapted from a similar nomograph given by Frank (1974), can be used to estimate the overall coefficient for tubular exchangers (shell and tube). The film coefficients given in Figure 12.1 include an allowance for fouling.

The values given in Table 12.1 and Figure 12.1 can be used for the preliminary sizing of equipment for process evaluation, and as trial values for starting a detailed thermal design.

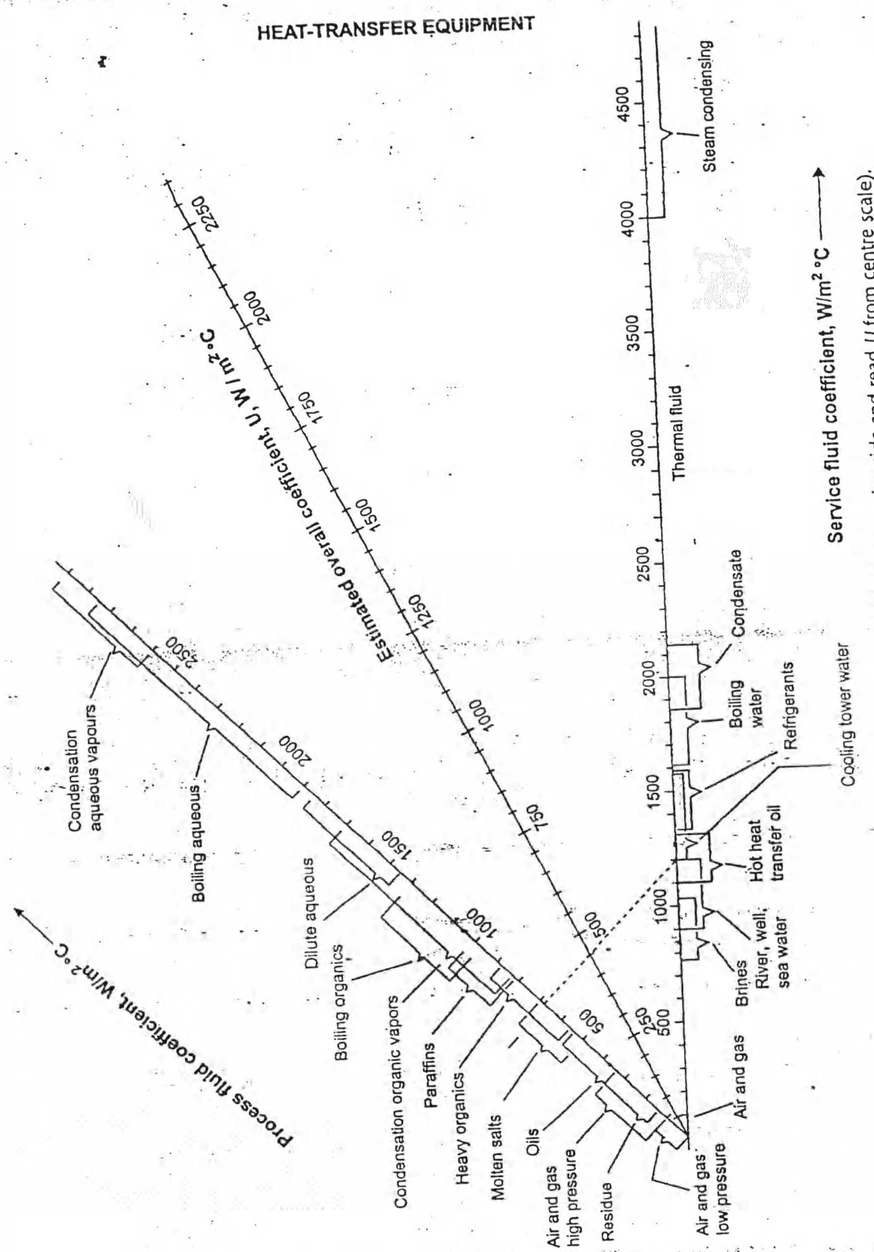


Figure 12.1. Overall coefficients (join process side duty to service side and read *U* from centre scale).

Tube-Side Passes

The fluid in the tube is usually directed to flow back and forth in a number of 'passes' through groups of tubes arranged in parallel, to increase the length of the flow path. The number of passes is selected to give the required tube-side design velocity. Exchangers are built with from one to up to about 16 tube passes. The tubes are arranged into the number of passes required by dividing up the exchanger headers (channels) with partition plates (pass partitions). The arrangement of the pass partitions for two, four and six tube passes are shown in Figure 12.11. The layouts for higher numbers of passes are given by Saunders (1988).

Shells

The British standard BS 3274 covers exchangers from 6 in (150 mm) to 42 in (1067 mm) diameter; and the TEMA standards, exchangers up to 60 in (1520 mm).

Up to about 24 in (610 mm), shells are normally constructed from standard, close tolerance, pipe; above 24 in (610 mm) they are rolled from plate.

For pressure applications the shell thickness would be sized according to the pressure vessel design standards, see Chapter 13. The minimum allowable shell thickness is given in BS 3274 and the TEMA standards. The values, converted to SI units and rounded, are given below:

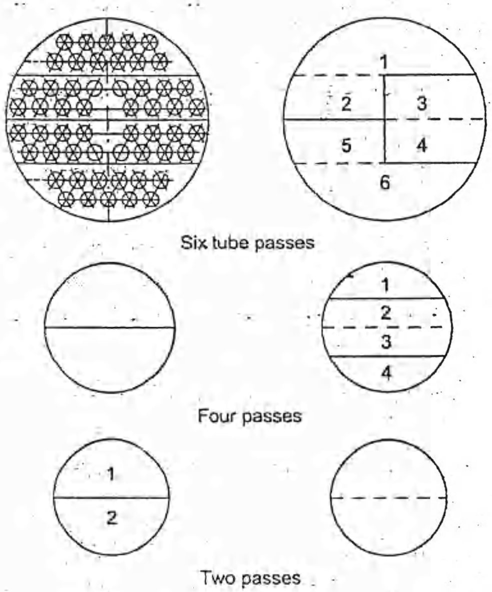


Figure 12.11. Tube arrangements, showing pass-partitions in headers.

Minimum Shell Thickness (mm)

Nominal Shell dia. (mm)	Carbon Steel Pipe	Carbon Steel Plate	Alloy Steel
150	7.1	—	3.2
200–300	9.3	—	3.2
330–580	9.5	7.9	3.2
610–740	—	7.9	4.8
760–990	—	9.5	6.4
1010–1520	—	11.1	6.4
1550–2030	—	12.7	7.9
2050–2540	—	12.7	9.5

The shell diameter must be selected to give as close a fit to the tube bundle as is practical; to reduce bypassing round the outside of the bundle; see Section 12.9. The clearance required between the outermost tubes in the bundle and the shell inside diameter will depend on the type of exchanger and the manufacturing tolerances; typical values are given in Figure 12.12.

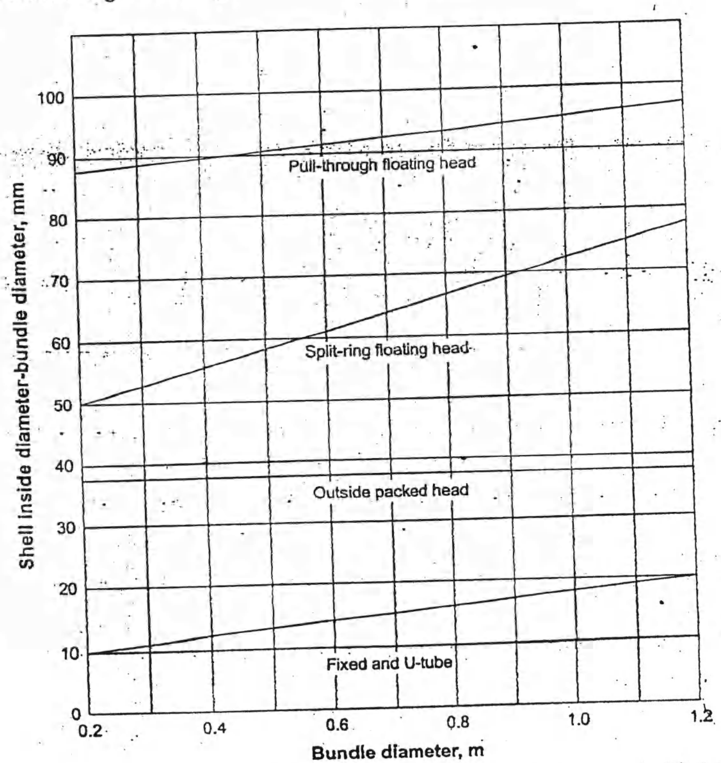


Figure 12.12. Shell-bundle clearance.

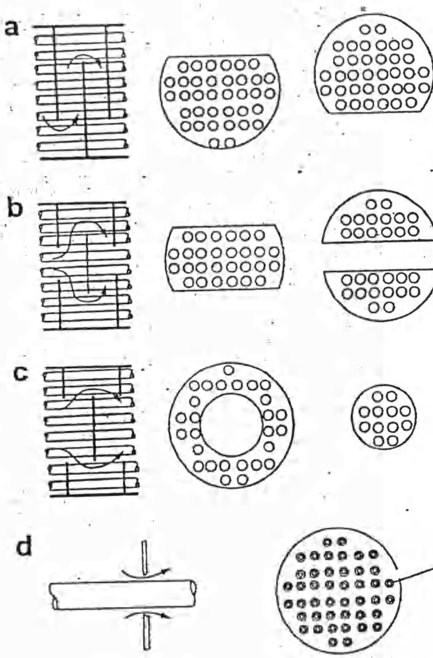


Figure 12.13. Types of baffle used in shell and tube heat exchangers. (a) Segmental. (b) Segmental and strip. (c) Disc and doughnut. (d) Orifice.

baffle as a clearance must be allowed for assembly. The clearance needed will depend on the shell diameter; typical values, and tolerances, are given in Table 12.5.

Another leakage path occurs through the clearance between the tube holes in the baffle and the tubes. The maximum design clearance will normally be 1/32 in. (0.8 mm).

The minimum thickness to be used for baffles and support plates is given in the standards. The baffle spacings used range from 0.2 to 1.0 shell diameters. A close baffle spacing will give higher heat-transfer coefficients, but at the expense of higher pressure drop. The optimum spacing will usually be between 0.3 to 0.5 times the shell diameter.

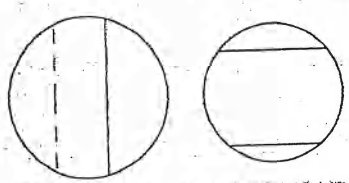


Figure 12.14. Baffles for condensers.

TABLE 12.5. Typical Baffle Clearances and Tolerances

Shell Diameter, D_s	Baffle Diameter	Tolerance
Pipe shells		
6 to 25 in (152 to 635 mm)	$D_s - \frac{1}{16}$ in (1.6 mm)	$+\frac{1}{32}$ in (0.8 mm)
Plate shells		
6 to 25 in (152 to 635 mm)	$D_s - \frac{1}{8}$ in (3.2 mm)	$+0, -\frac{1}{32}$ in (0.8 mm)
27 to 42 in (686 to 1067 mm)	$D_s - \frac{3}{16}$ in (4.8 mm)	$+0, -\frac{1}{16}$ in (1.6 mm)

12.5.8. Support Plates and Tie Rods

Where segmental baffles are used, some will be fabricated with closer tolerances, $\frac{1}{64}$ in (0.4 mm), to act as support plates. For condensers and vaporizers, where baffles are not needed for heat-transfer purposes, a few will be installed to support the tubes.

The minimum spacings to be used for support plates are given in the standards. The spacing ranges from around 1 m for 16-mm tubes to 2 m for 25-mm tubes.

The baffles and support plate are held together with tie rods and spacers. The number of rods required depends on the shell diameter, and ranges from four 16-mm diameter rods, for exchangers under 380 mm diameter; to eight 12.5-mm rods, for exchangers of 1 m diameter. The recommended number for a particular diameter can be found in the standards.

12.5.9. Tube Sheets (Plates)

In operation, the tube sheets are subjected to the differential pressure between shell and tube sides. The design of tube sheets as pressure-vessel components is covered by the ASME BPV Code and BS 5500 and is discussed in Section 13.11. Design formulae for calculating tube-sheet thicknesses are also given in the TEMA standards.

The joint between the tubes and tube sheet is normally made by expanding the tube by rolling with special tools (Figure 12.15). Tube rolling is a skilled task; the tube must be expanded sufficiently to ensure a sound leak-proof joint, but not over-thinned, weakening the tube. The tube holes are normally grooved (Figure 12.16a), to

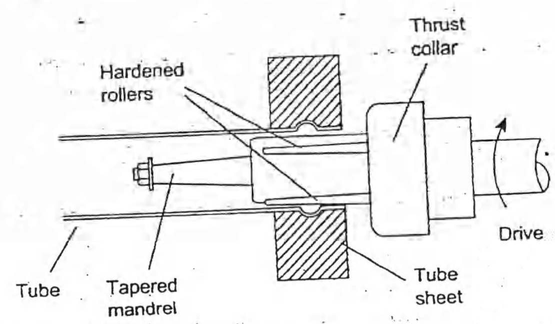


Figure 12.15. Tube rolling.

5.4. Tube-sheet Layout (Tube Count)

The bundle diameter depends not only on the number of tubes but also on the number of tube passes, as spaces must be left in the pattern of tubes on the tube sheet to accommodate the pass partition plates.

An estimate of the bundle diameter D_b can be obtained from equation 12.3b, which is an empirical equation based on standard tube layouts. The constants for use in this equation, for triangular and square patterns, are given in Table 12.4.

$$N_t = K_1 \left(\frac{D_b}{d_o}\right)^{n_1} \tag{12.3a}$$

$$D_b = d_o \left(\frac{N_t}{K_1}\right)^{1/n_1} \tag{12.3b}$$

where

N_t = number of tubes
D_b = bundle diameter, mm
d_o = tube outside diameter, mm.

If U-tubes are used, the number of tubes will be slightly less than that given by equation 12.3a, as the spacing between the two centre rows will be determined by the minimum allowable radius for the U-bend. The minimum bend radius will depend on the tube diameter and wall thickness. It will range from 1.5 to 3.0 times the tube outside diameter. The tighter bend radius will lead to some thinning of the tube wall.

An estimate of the number of tubes in a U-tube exchanger (twice the actual number of U-tubes), can be made by reducing the number given by equation 12.3a by one centre row of tubes.

The number of tubes in the centre row, the row at the shell equator, is given by:

$$\text{Tubes in centre row} = \frac{D_b}{p_t}$$

where p_t = tube pitch, mm.

The tube layout for a particular design will normally be planned with the aid of computer programs. These will allow for the spacing of the pass partition plates and

TABLE 12.4. Constants for Use in Equation 12.3

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. 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

the position of the tie rods. Also, one or two rows of tubes may be omitted at the top and bottom of the bundle to increase the clearance and flow area opposite the inlet and outlet nozzles.

Tube count tables that give an estimate of the number of tubes that can be accommodated in standard shell sizes, for commonly used tube sizes, pitches and number of passes, can be found in several books: Kern (1950), Ludwig (2001), Green and Perry (2007), and Saunders (1988).

Some typical tube arrangements are shown in Appendix H.

12.5.5. Shell Types (Passes)

The principal shell arrangements are shown in Figure 12.9. The letters E, F, G, H, J are those used in the TEMA standards to designate the various types. The E shell is the most commonly used arrangement.

Two shell passes (F shell) are occasionally used where the shell and tube side temperature differences are unsuitable for a single pass (see Section 12.6); however, it is difficult to obtain a satisfactory seal with a shell-side baffle and the same flow arrangement can be achieved by using two shells in series.

The divided-flow and split-flow arrangements (G and J shells) are used to reduce the shell-side pressure drop, where pressure drop, rather than heat transfer, is the controlling factor in the design.

12.5.6. Shell and Tube Designation

A common method of describing an exchanger is to designate the number of shell and tube passes: m/n or m:n; where m is the number of shell passes and n the number of tube passes. So 1/2 or 1:2 describes an exchanger with one shell pass and two tube passes, and 2/4 an exchanger with two shell passes and four tube passes.

12.5.7. Baffles

Baffles are used in the shell to direct the fluid stream across the tubes, to increase the fluid velocity and so improve the rate of transfer. The most commonly used type of baffle is the single segmental baffle shown in Figure 12.13a, other types are shown in Figure 12.13b, c and d.

Only the design of exchangers using single segmental baffles will be considered in this chapter.

If the arrangement shown in Figure 12.13a were used with a horizontal condenser the baffles would restrict the condensate flow. This problem can be overcome either by rotating the baffle arrangement through 90°, or by trimming the base of the baffle (Figure 12.14).

The term 'baffle cut' is used to specify the dimensions of a segmental baffle. The baffle cut is the height of the segment removed to form the baffle, expressed as a percentage of the baffle disc diameter. Baffle cuts from 15% to 45% are used. Generally, a baffle cut of 20% to 25% will be the optimum, giving good heat-transfer rates, without excessive pressure drop. There will be some leakage of fluid round the

2.1. INTRODUCTION

The transfer of heat to and from process fluids is an essential part of most chemical processes. The most commonly used type of heat-transfer equipment is the ubiquitous shell and tube heat exchanger; the design of which is the main subject of this chapter.

The fundamentals of heat-transfer theory are covered in Coulson *et al.* (1999), and in many other textbooks: Holman (2002), Ozisik (1985), Rohsenow *et al.* (1998), Kreith and Bohn (2000) and Incropera and Dewitt (2001).

Several useful books have been published on the design of heat-exchange equipment. These should be consulted for more details of the construction of equipment and design methods than can be given in this book. A selection of the more useful texts is listed in the bibliography at the end of this chapter. The compilation edited by Schlünder (1983), see also the edition by Hewitt (2002), is probably the most comprehensive work on heat-exchanger design methods available in the open literature. The book by Saunders (1988) is recommended as a good source of information on heat-exchanger design, especially for shell and tube exchangers.

As with distillation, work on the development of reliable design methods for heat exchangers has been dominated in recent years by commercial research organizations: Heat Transfer Research Inc. (HTRI) in the USA and Heat Transfer and Fluid Flow Service (HTFS) in the UK. The HTFS program was developed by the UK Atomic Energy Authority and the National Physical Laboratory, but is now available from Aspen Technology Inc. and as part of the Honeywell UniSim Design Suite; see Chapter 4, Table 4.1. Their proprietary methods are not available in the open literature. They will, however, be available to design engineers in the major operating and contracting companies, whose companies subscribe to these organizations.

The principal types of heat exchanger used in the chemical process and allied industries, which will be discussed in this chapter, are listed below:

1. Double-pipe exchanger: the simplest type, used for cooling and heating
2. Shell and tube exchangers: used for all applications
3. Plate and frame exchangers (plate heat exchangers): used for heating and cooling
4. Plate-fin exchangers
5. Spiral heat exchangers
6. Air cooled: used for coolers and condensers
7. Direct contact: used for cooling and quenching
8. Agitated vessels
9. Fired heaters.

The word 'exchanger' really applies to all types of equipment in which heat is exchanged but is often used specifically to denote equipment in which heat is exchanged between two process streams. Exchangers in which a process fluid is heated or cooled by a plant service stream are referred to as heaters and coolers. If the process stream is vaporized, the exchanger is called a vaporizer if the stream is essentially completely vaporized, a reboiler if associated with a distillation column, and an evaporator if used to concentrate a solution (see Chapter 10). The terms fired exchanger or fired heater are used for exchangers hea...ed by co...tion such boilers; other exchangers are referred to as 'unfired exchangers'.

12.2. BASIC DESIGN PROCEDURE AND THEORY

The general equation for heat transfer across a surface is:

$$Q = UA\Delta T_m \tag{12.1}$$

where

Q = heat transferred per unit time, W
U = the overall heat-transfer coefficient, W/m^2°C
A = heat-transfer area, m^2
ΔT_m = the mean temperature difference, the temperature driving force, °C.

The prime objective in the design of an exchanger is to determine the surface area required for the specified duty (rate of heat transfer) using the temperature differences available.

The overall coefficient is the reciprocal of the overall resistance to heat transfer, which is the sum of several individual resistances. For heat exchange across a typical heat-exchanger tube the relationship between the overall coefficient and the individual coefficients, which are the reciprocals of the individual resistances, is given by:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k_w} + \frac{d_o}{d_i} \times \frac{1}{h_{id}} + \frac{d_o}{d_i} \times \frac{1}{h_i} \tag{12.2}$$

where

U_o = the overall coefficient based on the outside area of the tube, W/m^2°C
h_o = outside fluid film coefficient, W/m^2°C
h_i = inside fluid film coefficient, W/m^2°C
h_{od} = outside dirt coefficient (fouling factor), W/m^2°C
h_{id} = inside dirt coefficient, W/m^2°C
k_w = thermal conductivity of the tube wall material, W/m°C
d_i = tube inside diameter, m
d_o = tube outside diameter, m.

The magnitude of the individual coefficients will depend on the nature of the heat transfer process (conduction, convection, condensation, boiling or radiation), on the physical properties of the fluids, on the fluid flow rates, and on the physical arrangement of the heat-transfer surface. As the physical layout of the exchanger cannot be determined until the area is known, the design of an exchanger is of necessity a trial-and-error procedure. The steps in a typical design procedure are given below:

1. Define the duty: heat-transfer rate, fluid flow rates, temperatures.
2. Collect together the fluid physical properties required: density, viscosity, thermal conductivity.

the fluid temperatures at the inlet and outlet of the exchanger. The well known 'logarithmic mean' temperature difference is only applicable to sensible heat transfer in true co-current or counter-current flow, with linear temperature–enthalpy curves. This situation occurs when the heat capacities of both streams are constant and there is no phase change, or if there is a phase change at constant pressure for a stream that contains a single component. These conditions are only approximated in reality. For counter-current flow (Figure 12.18a), the logarithmic mean temperature difference is given by:

$$\Delta T_{lm} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \dfrac{(T_1 - t_2)}{(T_2 - t_1)}} \tag{12.4}$$

where

ΔT_{lm} = log mean temperature difference
T_1 = hot fluid temperature, inlet
T_2 = hot fluid temperature, outlet

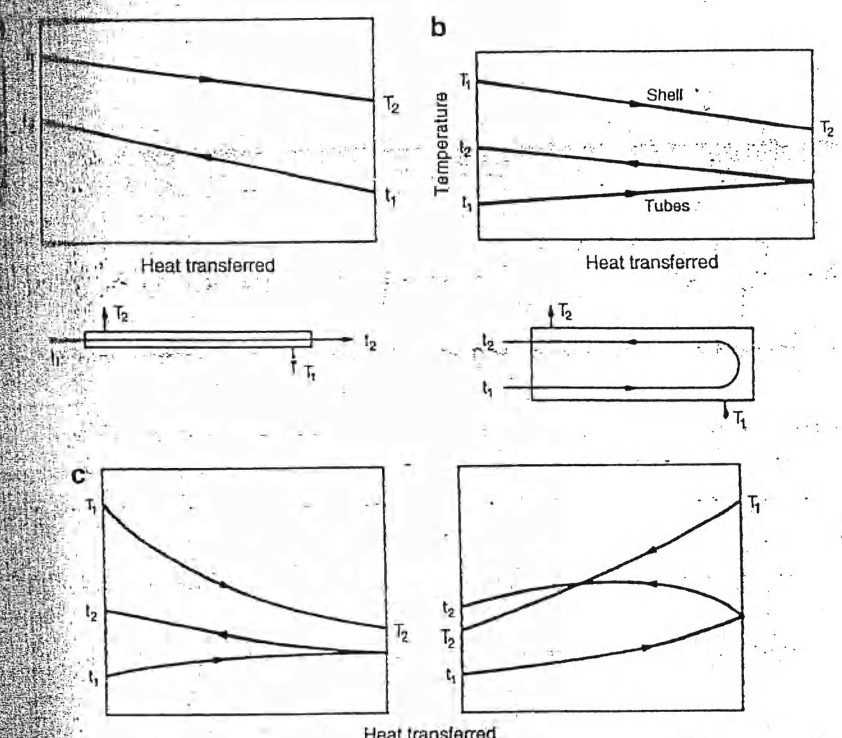


Figure 12.18. Temperature profiles. (a) Counter-current flow. (b) 1:2 exchanger. (c) Temperature cross.

t_1 = cold fluid temperature, inlet
t_2 = cold fluid temperature, outlet.

The equation is the same for co-current flow, but the terminal temperature differences will be $(T_1 - t_1)$ and $(T_2 - t_2)$. Strictly, equation 12.4 will only apply when there is no change in the specific heats, the overall heat-transfer coefficient is constant, and there are no heat losses. In design, these conditions can be assumed to be satisfied providing the temperature change in each fluid stream is not large.

In most shell and tube exchangers, the flow will be a mixture of co-current, counter-current and cross flow. Figures 12.18b and c show typical temperature profiles for an exchanger with one shell pass and two tube passes (a 1:2 exchanger). Figure 12.18c shows two different cases of temperature cross, where the outlet temperature of the cold stream is above that of the hot stream.

The usual practice in the design of shell and tube exchangers is to estimate the 'true temperature difference' from the logarithmic mean temperature by applying a correction factor to allow for the departure from true counter-current flow:

$$\Delta T_m = F_t \Delta T_{lm} \tag{12.5}$$

Where ΔT_m = true temperature difference, the mean temperature difference for use in the design equation 12.1

F_t = the temperature correction factor

The correction factor is a function of the shell and tube fluid temperatures, and the number of tube and shell passes. It is normally correlated as a function of two dimensionless temperature ratios:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \tag{12.6}$$

and

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)} \tag{12.7}$$

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 is a measure of the temperature efficiency of the exchanger.

For a 1:2 exchanger, the correction factor is given by:

$$F_t = \frac{\sqrt{(R^2+1)}\ln\left[\dfrac{(1-S)}{(1-RS)}\right]}{(R-1)\ln\left[\dfrac{2 - S[R + 1 - \sqrt{(R^2+1)}]}{2 - S[R + 1 + \sqrt{(R^2+1)}]}\right]} \tag{12.8}$$

The derivation of equation 12.8 is given by Kern (1950). The equation for a 1:2 exchanger can be used for any exchanger with an even number of tube passes, and is plotted in Figure 12.19. The correction factor for two shell passes and four, or multiples of four, tube passes is shown in Figure 12.20, and that for divided-flow and split-flow shells in Figures 12.21 and 12.22.

Temperature correction factor plots for other arrangements can be found in the TEMA standards and the books by Kern (1950) and Ludwig (2001). Mueller (1973) gives a comprehensive set of figures for calculating the log mean temperature correction factor, which includes figures for cross-flow exchangers.

The following assumptions are made in the derivation of the temperature correction factor F_t, in addition to those made for the calculation of the log mean temperature difference:

1. Equal heat transfer areas in each pass
2. A constant overall heat-transfer coefficient in each pass
3. The temperature of the shell-side fluid in any pass is constant across any cross-section
4. There is no leakage of fluid between shell passes.

Though these conditions will not be strictly satisfied in practical heat exchangers, the F_t values obtained from the curves will give an estimate of the 'true mean temperature difference' that is sufficiently accurate for most designs. Mueller (1973) discusses these assumptions, and gives F_t curves for conditions when all the assumptions are not met; see also Butterworth (1973) and Emerson (1973). Values of F_t are calculated for heat exchangers in most process simulation programs, as described in Chapter 4.

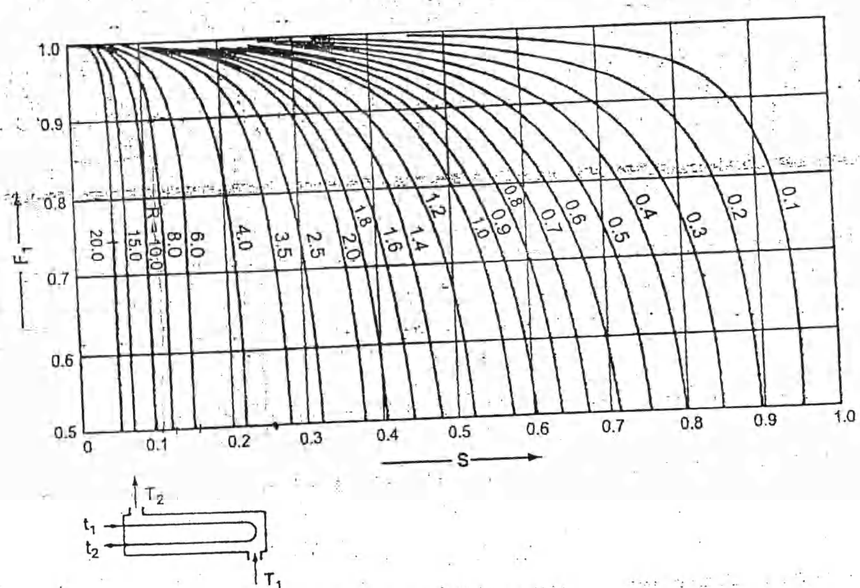


Figure 12.19. Temperature correction factor: one shell pass; two or more even tube passes.

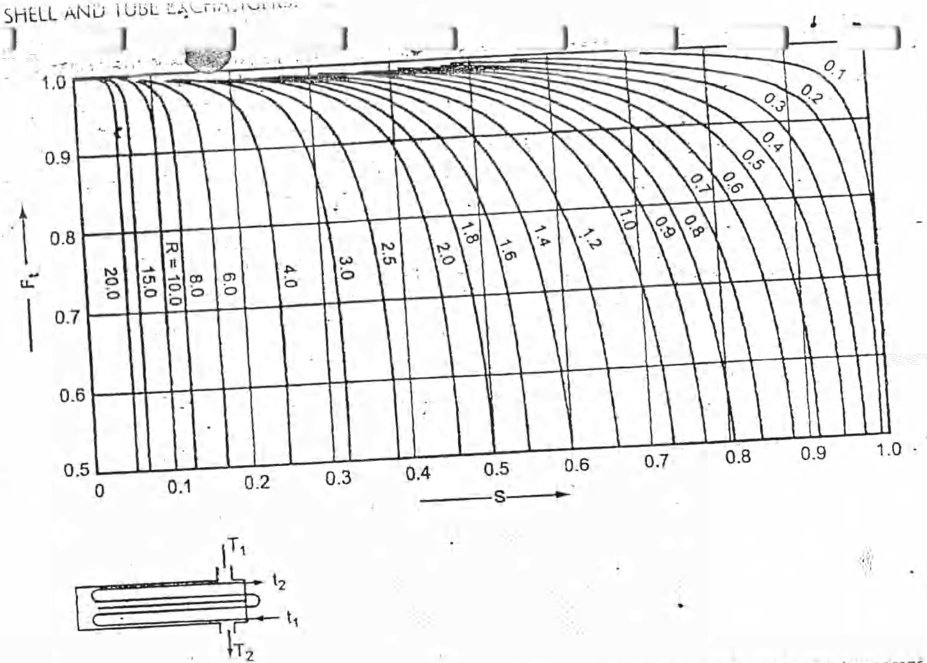


Figure 12.20. Temperature correction factor: two shell passes; four or multiples of four tube passes.

The shell-side leakage and bypass streams (see Section 12.9) will affect the mean temperature difference, but are not normally taken into account when estimating the correction factor F_t. Fisher and Parker (1969) give curves that show the effect of leakage on the correction factor for a 1:2 exchanger.

The value of F_t will be close to one when the terminal temperature differences are large, but will appreciably reduce the logarithmic mean temperature difference when the temperatures of shell and tube fluids approach each other; it will fall drastically when there is a temperature cross. A temperature cross will occur if the outlet temperature of the cold stream is greater than the outlet temperature of the hot stream, Figure 12.18c.

Where the F_t curve is near vertical, values cannot be read accurately, which will introduce a considerable uncertainty into the design.

An economic exchanger design cannot normally be achieved if the correction factor F_t falls below about 0.75. In these circumstances, an alternative type of exchanger should be considered that gives a closer approach to true counter-current flow. The use of two or more shells in series, or multiple shell-side passes, will give a closer approach to true counter-current flow, and should be considered where a temperature cross is likely to occur.

When both sensible and latent heat is transferred, it will be necessary to divide the temperature profile into sections and calculate the mean temperature difference for each section. The overall heat-transfer coefficient should also be different in each section.

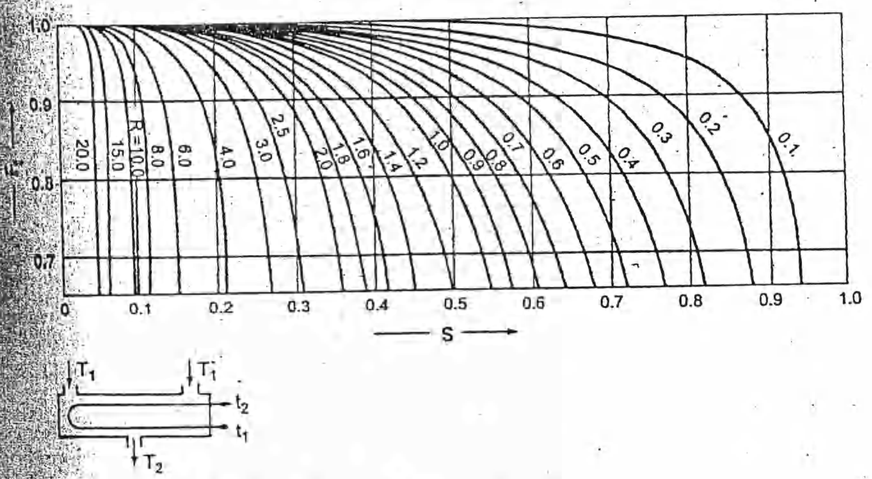


Figure 12.21. Temperature correction factor: divided-flow shell; two or more even tube passes.

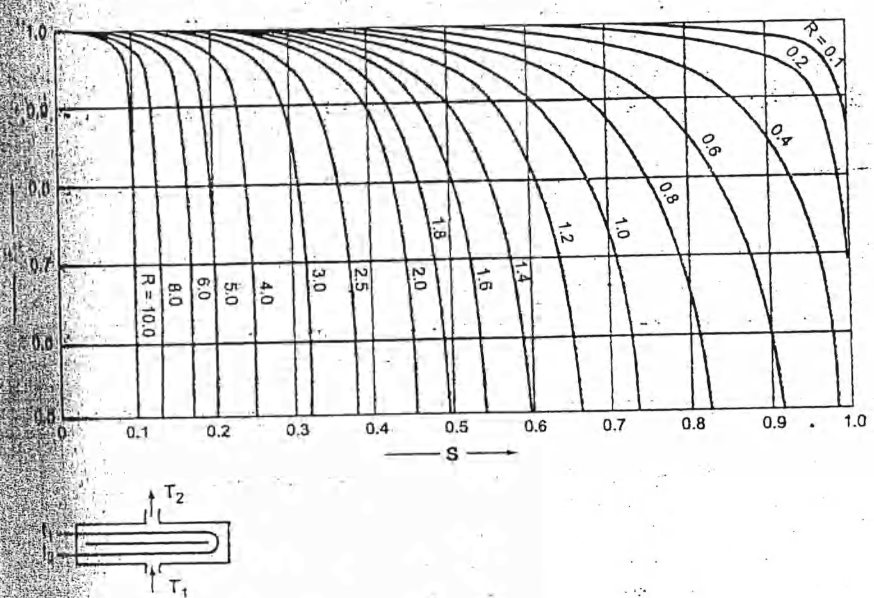


Figure 12.22. Temperature correction factor: split-flow shell, two tube passes.

12.7. SHELL AND TUBE EXCHANGERS: GENERAL DESIGN CONSIDERATIONS

12.7.1. Fluid Allocation: Shell or Tubes

Where no phase change occurs, the following factors determine the allocation of the fluid streams to the shell or tubes.

Corrosion. The more corrosive fluid should be allocated to the tube side. This will reduce the cost of expensive alloy or clad components.

Fouling. The fluid that has the greatest tendency to foul the heat-transfer surfaces should be placed in the tubes. This gives better control over the design fluid velocity, and the higher allowable velocity in the tubes will reduce fouling. Also, the tubes will be easier to clean.

Fluid temperatures. If the temperatures are high enough to require the use of special alloys, placing the higher temperature fluid in the tubes will reduce the overall cost. At moderate temperatures, placing the hotter fluid in the tubes will reduce the shell surface temperatures, and hence the need for lagging to reduce heat loss, or for safety reasons.

Operating pressures. The higher pressure stream should be allocated to the tube side. High-pressure tubes will be cheaper than a high-pressure shell. The required tube thickness is less for high internal pressure than high external pressure and an expensive high-pressure shell may be avoided.

Pressure drop. For the same pressure drop, higher heat-transfer coefficients will be obtained on the tube side than the shell side, and fluid with the lowest allowable pressure drop should be allocated to the tube side.

Viscosity. Generally, a higher heat-transfer coefficient will be obtained by allocating the more viscous material to the shell side, providing the flow is turbulent. The critical Reynolds number for turbulent flow in the shell is in the region of 200. If turbulent flow cannot be achieved in the shell, it is better to place the fluid in the tubes, as the tube-side heat-transfer coefficient can be predicted with more certainty.

Stream flow rates. Allocating the fluids with the lowest flow rate to the shell side will normally give the most economical design.

12.7.2. Shell and Tube Fluid Velocities

High velocities will give high heat-transfer coefficients but also a high pressure drop. The velocity must be high enough to prevent any suspended solids settling, but not so high as to cause erosion. High velocities will reduce fouling. Plastic inserts are sometimes used to reduce erosion at the tube inlet. Typical design velocities are given below.

Liquids

Tube side, process fluids: 1 to 2 m/s, maximum 4 m/s if required to reduce fouling; water: 1.5 to 2.5 m/s.
Shell side: 0.3 to 1 m/s.

the coefficient should be evaluated using both equations 12.11 and 12.13 and the lower value taken.

Heat-transfer Factor, j_h

It is often convenient to correlate heat-transfer data in terms of a heat-transfer 'j' factor, which is similar to the friction factor used for pressure drop. The heat-transfer factor is defined by:

$$j_h = \mathrm{St}\mathrm{Pr}^{0.67}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \quad (12.14)$$

The use of the j_h factor allows data for laminar and turbulent flow to be represented on the same graph (Figure 12.23). The j_h values obtained from Figure 12.23 can be used with equation 12.14 to estimate the heat-transfer coefficients for heat-exchanger tubes and commercial pipes. The coefficient estimated for pipes will normally be conservative (on the low side) as pipes are rougher than the tubes used for heat exchangers, which are finished to closer tolerances. Equation 12.14 can be rearranged to a more convenient form:

$$\frac{h_i d_i}{k_f} = j_h \mathrm{Re}\mathrm{Pr}^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (12.15)$$

Note: Kern (1950), and others define the heat-transfer factor as

$$j_H = \mathrm{Nu}\mathrm{Pr}^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

The relationship between j_h and j_H is given by

$$j_H = j_h \mathrm{Re}$$

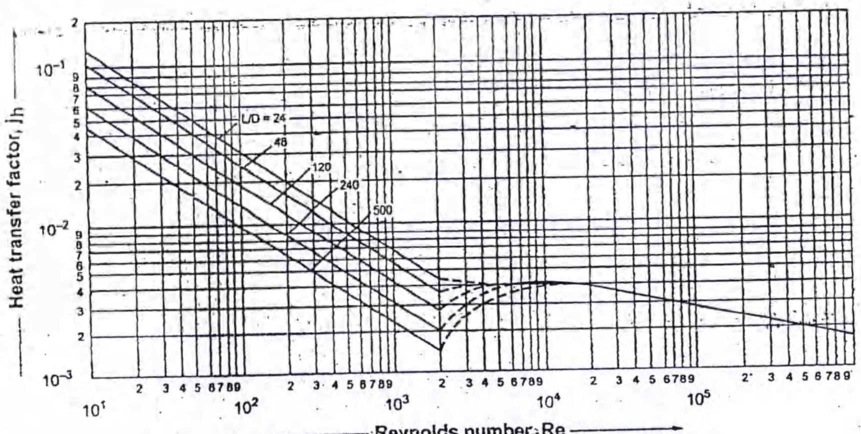


Figure 12.23. Tube-side heat-transfer factor.

Viscosity Correction Factor

The viscosity correction factor will normally only be significant for viscous liquids.

To apply the correction, an estimate of the wall temperature is needed. This can be made by first calculating the coefficient without the correction and using the following relationship to estimate the wall temperature:

$$h_i(t_w - t) = U(T - t) \quad (12.16)$$

where

t = tube-side bulk temperature (mean)
t_w = estimated wall temperature
T = shell-side bulk temperature (mean).

Usually an approximate estimate of the wall temperature is sufficient, but trial-and-error calculations can be made to obtain a better estimate if the correction is large.

Coefficients for Water

Though equations 12.11 and 12.13 and Figure 12.23 may be used for water, a more accurate estimate can be made by using equations developed specifically for water. The physical properties are conveniently incorporated into the correlation. The equation below has been adapted from data given by Eagle and Ferguson (1930):

$$h_i = \frac{4200(1.35 + 0.02t)u_t^{0.8}}{d_i^{0.2}} \quad (12.17)$$

where

h_i = inside coefficient, for water, W/m^2°C
t = water temperature, °C
u_t = water velocity, m/s
d_i = tube inside diameter, mm.

12.8.2. Tube-side Pressure Drop

There are two major sources of pressure loss on the tube side of a shell and tube exchanger: the friction loss in the tubes and the losses due to the sudden contraction and expansion and flow reversals that the fluid experiences in flow through the tube arrangement.

The tube friction loss can be calculated using the familiar equations for pressure drop in pipes (see Chapter 5). The basic equation for isothermal flow in pipes (constant temperature) is

$$\Delta P = 8 j_f \left(\frac{L'}{d_i}\right)\frac{\rho u_t^2}{2} \quad (12.18)$$

where j_f is the dimensionless friction factor and L' is the effective pipe length.

The flow in a heat exchanger is clearly not isothermal, and this is allowed for by including an empirical correction factor to account for the change in physical properties with temperature. Normally only the change in viscosity is considered:

$$\Delta P = 8j_f(L'/d_i)\rho\frac{u_t^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-m} \tag{12.19}$$

$m = 0.25$ for laminar flow, $Re < 2100$
$= 0.14$ for turbulent flow, $Re > 2100$.

Values of j_f for heat-exchanger tubes can be obtained from Figure 12.24; commercial pipes are given in Chapter 5.

The pressure losses due to contraction at the tube inlets, expansion at the exits, and flow reversal in the headers, can be a significant part of the total tube-side pressure drop. There is no entirely satisfactory method for estimating these losses. Kern (1950) suggests adding four velocity heads per pass. Frank (1978) considers this to be too high, and recommends 2.5 velocity heads. Butterworth (1978) suggests 1.8. Lord *et al.* (1970) take the loss per pass as equivalent to a length of tube equal to 300 tube diameters for straight tubes and 200 for U-tubes, whereas Evans (1980) appears to add only 67 tube diameters per pass.

The loss in terms of velocity heads can be estimated by counting the number of flow contractions, expansions and reversals, and using the factors for pipe fittings to estimate the number of velocity heads lost. For two tube passes, there will be two contractions, two expansions and one flow reversal. The head loss for each of these effects is: contraction 0.5, expansion 1.0, 180° bend 1.5; so for two passes the maximum loss will be

$$2 \times 0.5 + 2 \times 1.0 + 1.5 = 4.5 \text{ velocity heads}$$
$$= \underline{\underline{2.25 \text{ per pass}}}$$

From this, it appears that Frank's recommended value of 2.5 velocity heads per pass is the most realistic value to use.

Combining this factor with equation 12.19 gives

$$\Delta P_t = N_p\left[8\,j_f\left(\frac{L}{d_i}\right)\left(\frac{\mu}{\mu_w}\right)^{-m} + 2.5\right]\frac{\rho u_t^2}{2} \tag{12.20}$$

where

ΔP_t = tube-side pressure drop, N/m² (Pa)
N_p = number of tube-side passes
u_t = tube-side velocity, m/s
L = length of one tube.

Another source of pressure drop is the flow expansion and contraction at the exchanger inlet and outlet nozzles. This can be estimated by adding one velocity head for the inlet and 0.5 for the outlet, based on the nozzle velocities.

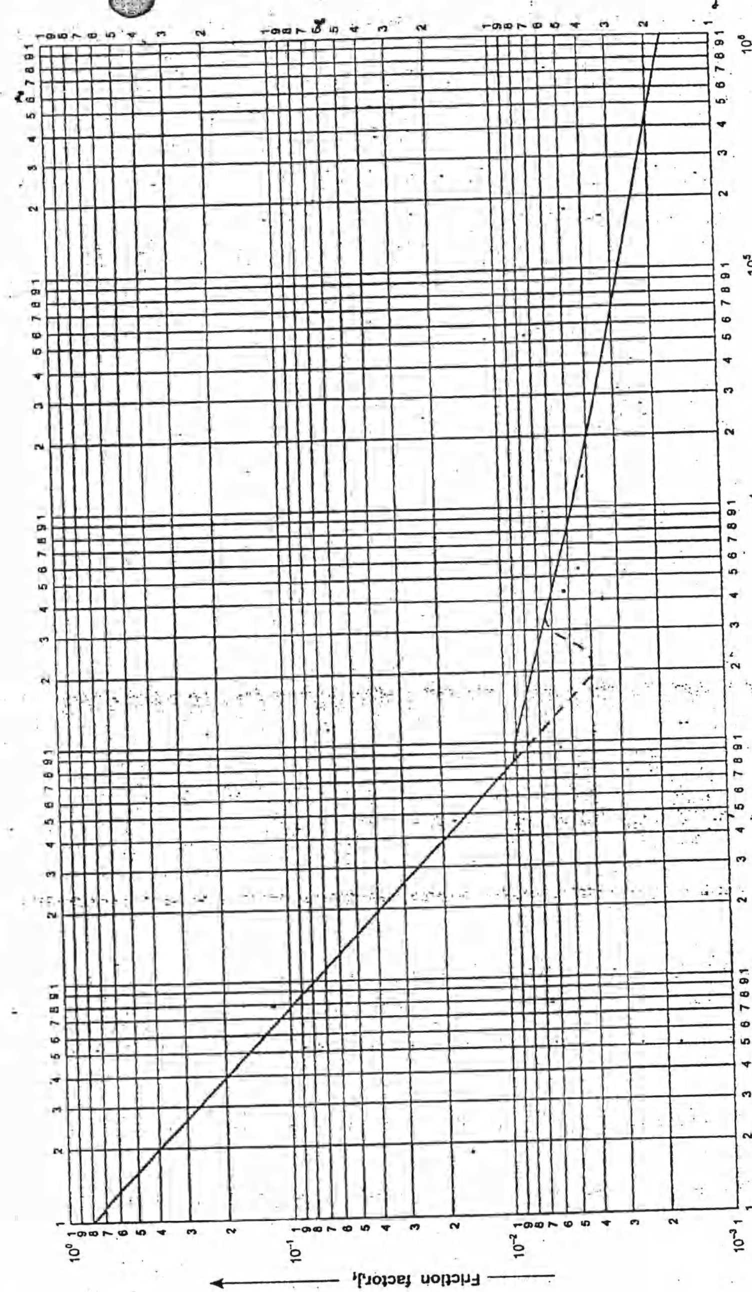


Figure 12.24. Tube-side friction factors. *Note:* The friction factor j_f is the same as the friction factor for pipes ϕ (= $(R/\rho u^2)$).

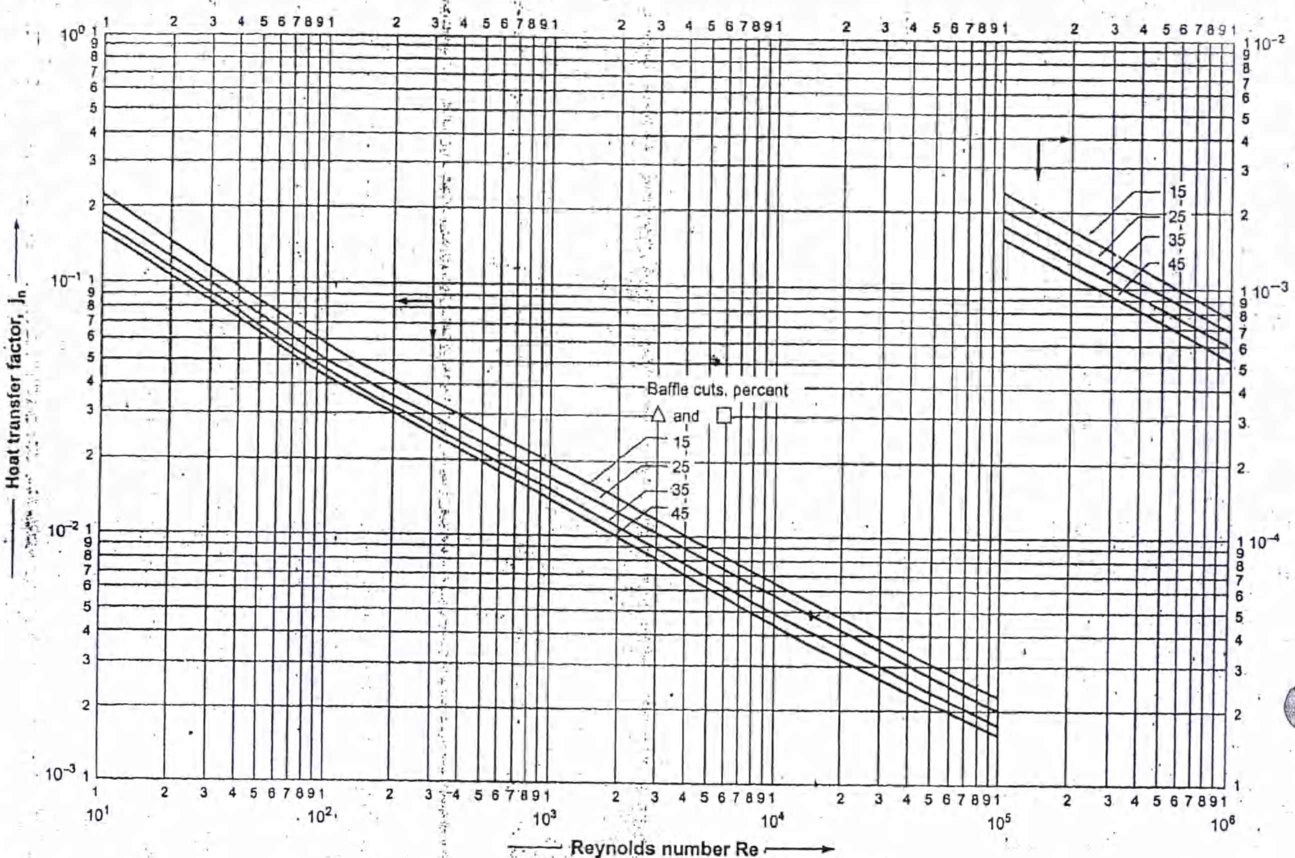


Figure 12.29. Shell-side heat-transfer factors, segmental baffles.

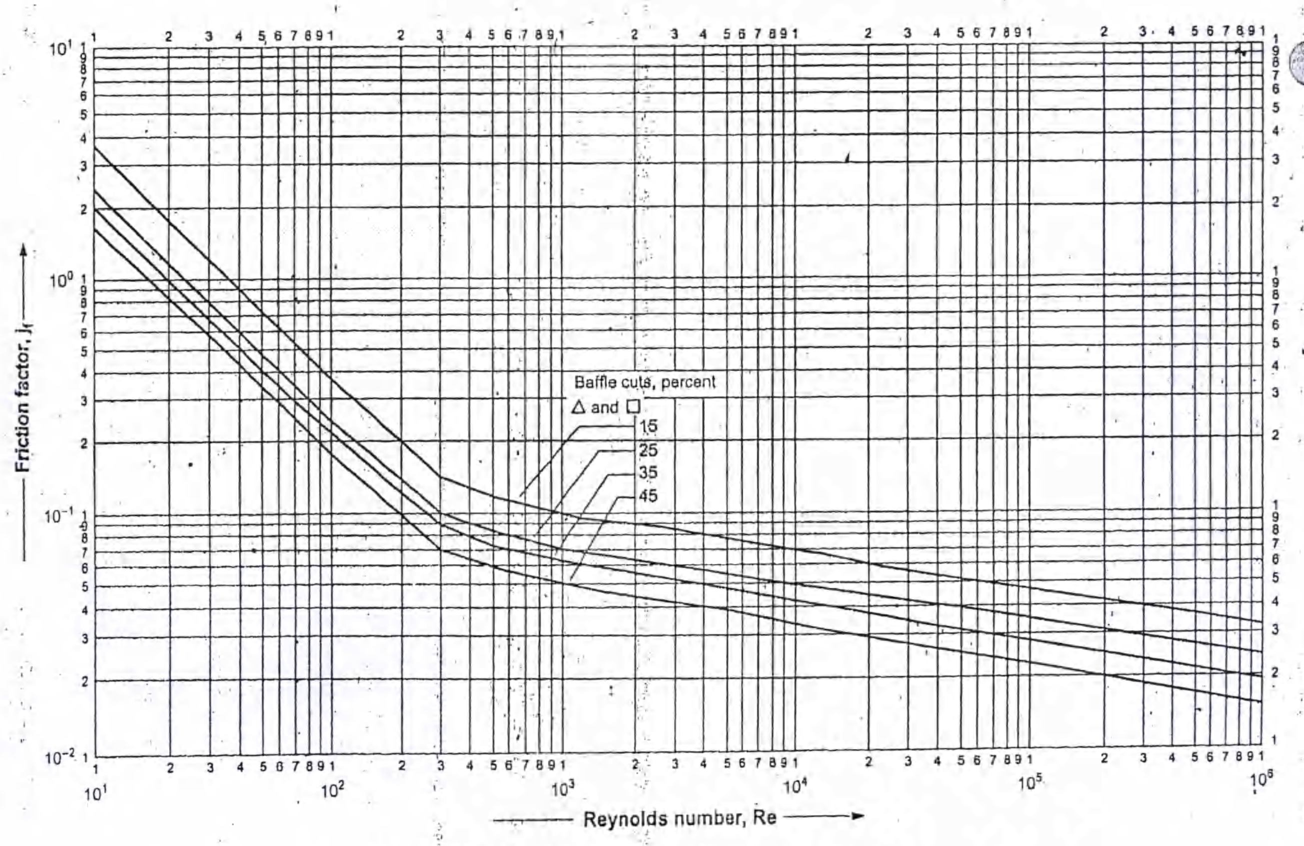


Figure 12.30. Shell-side friction factors, segmental baffles.

For an equilateral triangular pitch arrangement:

$$d_e = \frac{4\left(\frac{p_t}{2} \times 0.87\, p_t - \frac{1}{2}\pi\frac{d_o^2}{4}\right)}{\frac{\pi d_o}{2}} = \frac{1.10}{d_o}(p_t^2 - 0.917 d_o^2) \tag{12.23}$$

where d_e = equivalent diameter, m.

4. Calculate the shell-side Reynolds number, given by:

$$\text{Re} = \frac{G_s d_e}{\mu} = \frac{u_s d_e \rho}{\mu} \tag{12.24}$$

5. For the calculated Reynolds number, read the value of j_h from Figure 12.29 for the selected baffle cut and tube arrangement, and calculate the shell-side heat-transfer coefficient h_s from:

$$\text{Nu} = \frac{h_s d_e}{k_f} = j_h \,\text{Re}\text{Pr}^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{12.25}$$

The tube wall temperature can be estimated using the method given for the tube side; Section 12.8.1.

6. For the calculated shell-side Reynolds number, read the friction factor from Figure 12.30 and calculate the shell-side pressure drop from:

$$\Delta P_s = 8\, j_f\left(\frac{D_s}{d_e}\right)\left(\frac{L}{l_B}\right)\frac{\rho u_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{12.26}$$

where

L = tube length

l_B = baffle spacing.

The term (L/l_B) is the number of times the flow crosses the tube bundle $= (N_b + 1)$, where N_b is the number of baffles.

Shell-Nozzle Pressure Drop

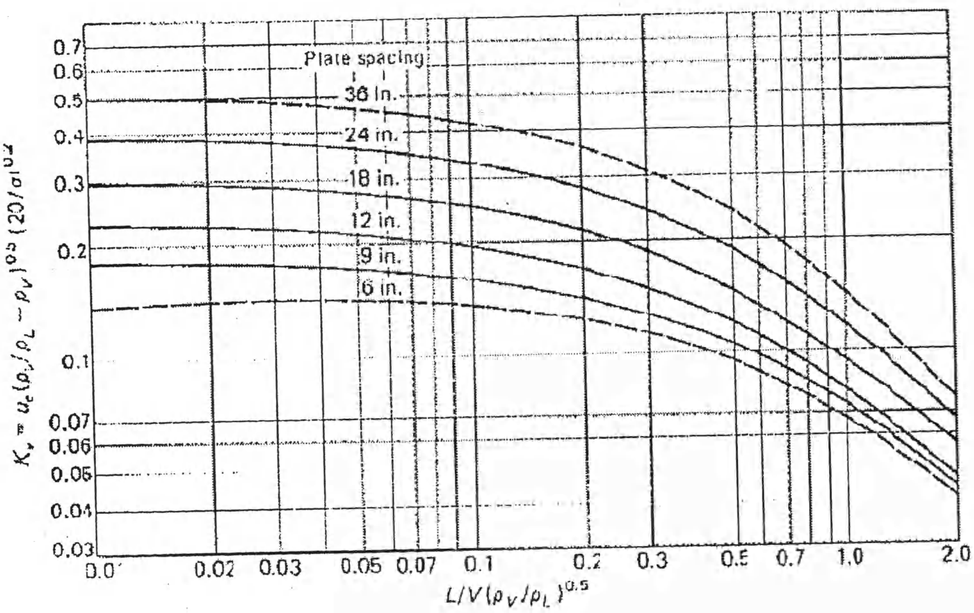


FIGURE 18.28
Values of K_v at flooding conditions for sieve plates; L/V = ratio of mass flow rate of liquid to vapor, u is in feet per second, and σ is in dynes per centimeter. [*J. R. Fair, Petrol. Chem. Eng.*, 33(10):45, 1961. *Courtesy Petroleum Engineer.*]

$$h_d = \left(\frac{u_0^2}{C_0^2}\right)\left(\frac{\rho_V}{2g\rho_L}\right) = 51.0\left(\frac{u_0^2}{C_0^2}\right)\left(\frac{\rho_V}{\rho_L}\right)$$

where u_0 = vapor velocity through holes, m/s
ρ_V = vapor density
ρ_L = liquid density
C_0 = orifice coefficient

$$Z_c = 2\beta(h_w + h_{ow}) + h_d + h_{f,L} \qquad h_l = \beta(h_w + h_{ow})$$

$$h_{ow} = 43.4\left(\frac{q_L}{L_w}\right)^{2/3}$$

where h_{ow} = height, mm
q_L = flow rate of clear liquid, m^3/min
L_w = length of weir, m

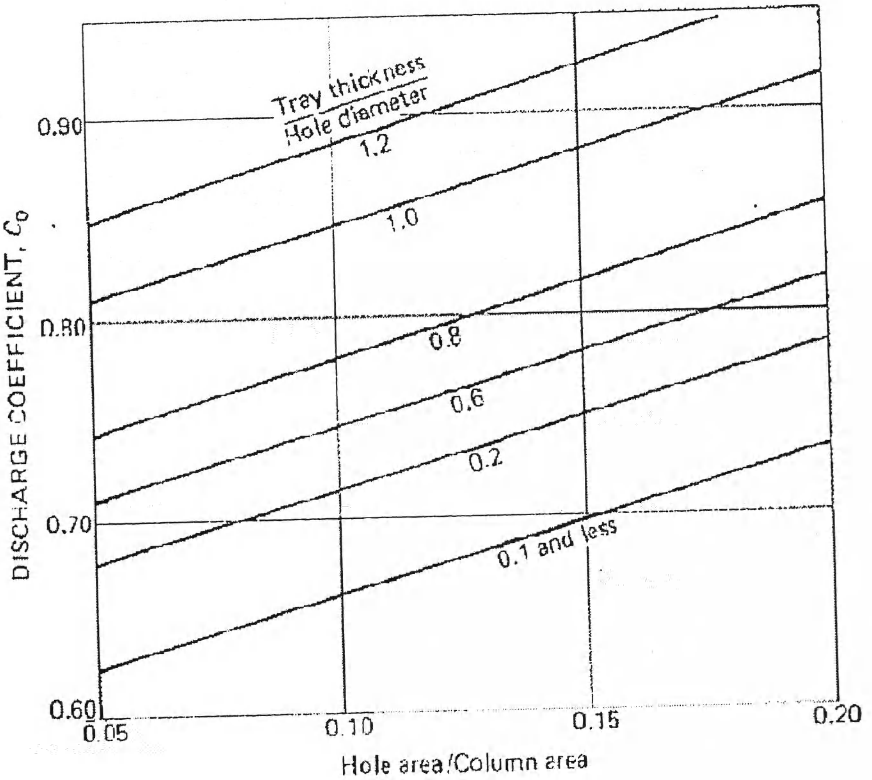


FIGURE 18.27
Discharge coefficients for vapor flow, sieve trays. [*I. Liebson, R. E. Kelley, and L. A. Bullington, Petrol. Refin.*, 36(2):127, 1957; 36(3):288, 1957.]