CHAPTER 12

Heat-transfer Equipment

12.1. 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 Volume 1, Chapter 9; 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 fuller 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 (1983ff), see also the edition by Hewitt (1990), 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 organisations: Heat Transfer Research Inc. (HTRI) in the United States and Heat Transfer and Fluid Flow Service (HTFS) in the United Kingdom. HTFS was developed by the United Kingdom Atomic Energy Authority and the National Physical Laboratory, but is now available from Aspentech, see Chapter 4, Table 4.1. Their methods are of a proprietary nature and are not therefore 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 organisations.

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: coolers and condensers.
7. Direct contact: 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 vaporised the exchanger is called a vaporiser if the stream is essentially completely vaporised; a reboiler if associated with a distillation column; and an evaporator if used to concentrate a solution (see Chapter 10). The term fired exchanger is used for exchangers heated by combustion gases, such as 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, $W/m^2{}^\circ C$,
A = heat-transfer area, m^2,
ΔT_m = the mean temperature difference, the temperature driving force, $^\circ C$.

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:

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

where U_o = the overall coefficient based on the outside area of the tube, $W/m^2{}^\circ C$,
h_o = outside fluid film coefficient, $W/m^2{}^\circ C$,
h_i = inside fluid film coefficient, $W/m^2{}^\circ C$,
h_{od} = outside dirt coefficient (fouling factor), $W/m^2{}^\circ C$,
h_{id} = inside dirt coefficient, $W/m^2{}^\circ C$,
k_w = thermal conductivity of the tube wall material, $W/m^\circ C$,
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.
3. Decide on the type of 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 12.1.
7. Decide the exchanger layout.
8. Calculate the individual coefficients.
9. Calculate the overall coefficient and compare with the trial value. If the calculated value differs significantly from the estimated value, substitute the calculated for the estimated value and return to step 6.
10. 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 steps 4 to 10, as necessary, to determine the cheapest exchanger that will satisfy the duty. Usually this will be the one with the smallest area.

Procedures for estimating the individual heat-transfer coefficients and the exchanger pressure drops are given in this chapter.

12.2.1. Heat exchanger analysis: the effectiveness—NTU method

The *effectiveness—NTU* method is a procedure for evaluating the performance of heat exchangers, which has the advantage that it does not require the evaluation of the mean temperature differences. *NTU* stands for the Number of Transfer Units, and is analogous with the use of transfer units in mass transfer; see Chapter 11.

The principal use of this method is in the rating of an existing exchanger. It can be used to determine the performance of the exchanger when the heat transfer area and construction details are known. The method has an advantage over the use of the design procedure outlined above, as an unknown stream outlet temperature can be determined directly, without the need for iterative calculations. It makes use of plots of the exchanger *effectiveness* versus *NTU*. The effectiveness is the ratio of the actual rate of heat transfer, to the maximum possible rate.

The *effectiveness–NTU* method will not be covered in this book, as it is more useful for rating than design. The method is covered in books by Incropera and Dewitt (2001), Ozisik (1985) and Hewitt *et al.* (1994). The method is also covered by the Engineering Sciences Data Unit in their Design Guides 98003 to 98007 (1998). These guides give large clear plots of *effectiveness* versus *NTU* and are recommended for accurate work.

12.3. OVERALL HEAT-TRANSFER COEFFICIENT

Typical values of the overall heat-transfer coefficient for various types of heat exchanger are given in Table 12.1. More extensive data can be found in the books by Perry *et al.* (1997), TEMA (1999), and Ludwig (2001).

Table 12.1. Typical overall coefficients

Shell and tube exchangers		
Hot fluid	Cold fluid	U (W/m^2°C)
Heat exchangers		
Water	Water	800–1500
Organic solvents	Organic solvents	100–300
Light oils	Light oils	100–400
Heavy oils	Heavy oils	50–300
Gases	Gases	10–50
Coolers		
Organic solvents	Water	250–750
Light oils	Water	350–900
Heavy oils	Water	60–300
Gases	Water	20–300
Organic solvents	Brine	150–500
Water	Brine	600–1200
Gases	Brine	15–250
Heaters		
Steam	Water	1500–4000
Steam	Organic solvents	500–1000
Steam	Light oils	300–900
Steam	Heavy oils	60–450
Steam	Gases	30–300
Dowtherm	Heavy oils	50–300
Dowtherm	Gases	20–200
Flue gases	Steam	30–100
Flue	Hydrocarbon vapours	30–100
Condensers		
Aqueous vapours	Water	1000–1500
Organic vapours	Water	700–1000
Organics (some non-condensables)	Water	500–700
Vacuum condensers	Water	200–500
Vaporisers		
Steam	Aqueous solutions	1000–1500
Steam	Light organics	900–1200
Steam	Heavy organics	600–900
Air-cooled exchangers		
Process fluid		
Water		300–450
Light organics		300–700
Heavy organics		50–150
Gases, 5–10 bar		50–100
10–30 bar		100–300
Condensing hydrocarbons		300–600
Immersed coils		
Coil	Pool	
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

(*continued overleaf*)

Table 12.1. (*continued*)

Coil	Pool	U (W/m^2°C)
Immersed coils		
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.

12.4. FOULING FACTORS (DIRT FACTORS)

Most process and service fluids will foul the heat-transfer surfaces in an exchanger to a greater or lesser extent. The deposited material will normally have a relatively low thermal conductivity and will reduce the overall coefficient. It is therefore necessary to oversize an exchanger to allow for the reduction in performance during operation. The effect of fouling is allowed for in design by including the inside and outside fouling coefficients in equation 12.2. Fouling factors are usually quoted as heat-transfer resistances, rather than coefficients. They are difficult to predict and are usually based on past experience.

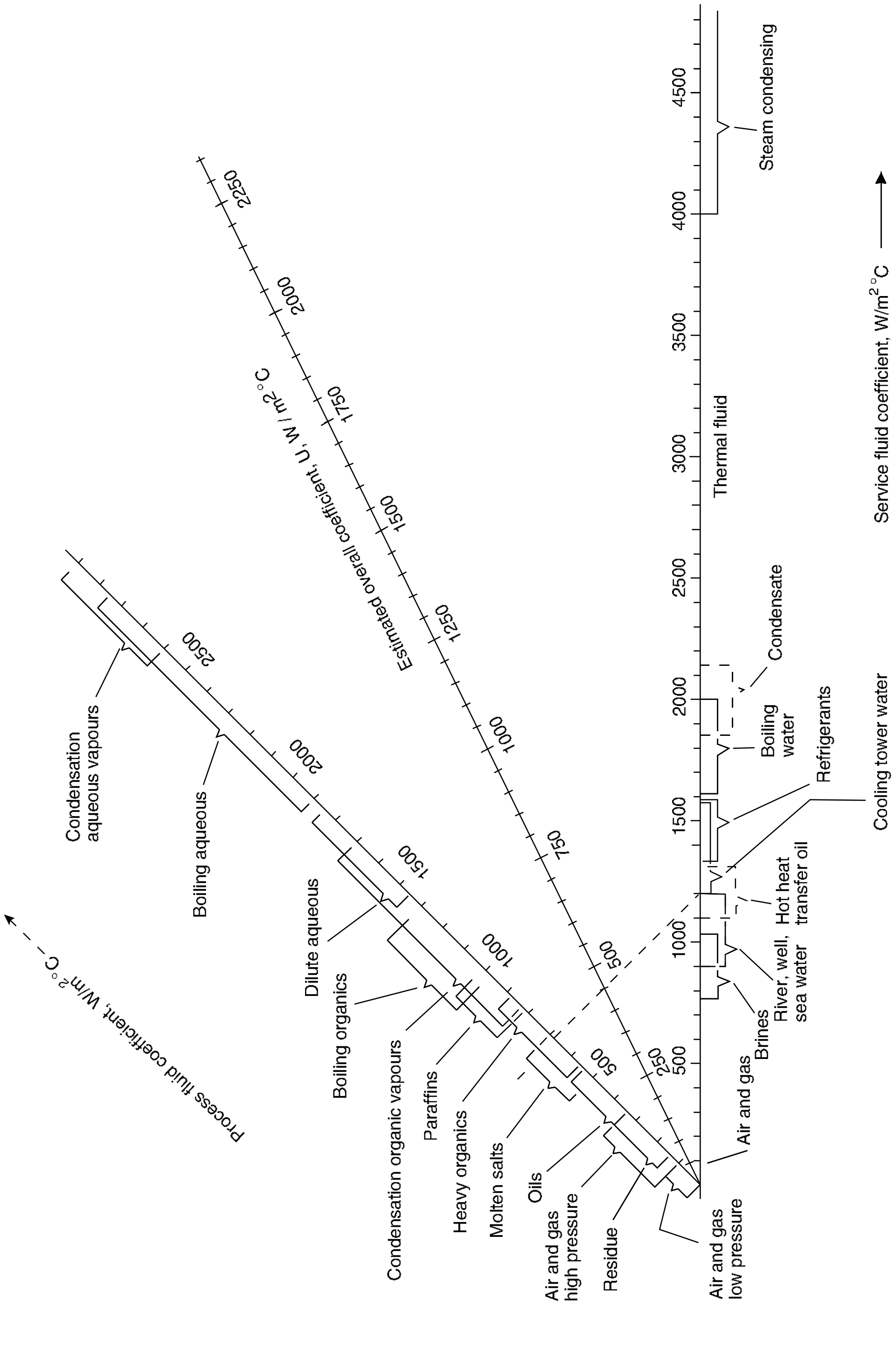


Figure 12.1. Overall coefficients (join process side duty to service side and read U from centre scale)

Estimating fouling factors introduces a considerable uncertainty into exchanger design; the value assumed for the fouling factor can overwhelm the accuracy of the predicted values of the other coefficients. Fouling factors are often wrongly used as factors of safety in exchanger design. Some work on the prediction of fouling factors has been done by HTRI; see Taborek *et al.* (1972). Fouling is the subject of books by Bott (1990) an Garrett-Price (1985).

Typical values for the fouling coefficients and factors for common process and service fluids are given in Table 12.2. These values are for shell and tube exchangers with plain (not finned) tubes. More extensive data on fouling factors are given in the TEMA standards (1999), and by Ludwig (2001).

Table 12.2. Fouling factors (coefficients), typical values

Fluid	Coefficient (W/m^2°C)	Factor (resistance) (m^2°C/W)
River water	3000–12,000	0.0003–0.0001
Sea water	1000–3000	0.001–0.0003
Cooling water (towers)	3000–6000	0.0003–0.00017
Towns water (soft)	3000–5000	0.0003–0.0002
Towns water (hard)	1000–2000	0.001–0.0005
Steam condensate	1500–5000	0.00067–0.0002
Steam (oil free)	4000–10,000	0.0025–0.0001
Steam (oil traces)	2000–5000	0.0005–0.0002
Refrigerated brine	3000–5000	0.0003–0.0002
Air and industrial gases	5000–10,000	0.0002–0.0001
Flue gases	2000–5000	0.0005–0.0002
Organic vapours	5000	0.0002
Organic liquids	5000	0.0002
Light hydrocarbons	5000	0.0002
Heavy hydrocarbons	2000	0.0005
Boiling organics	2500	0.0004
Condensing organics	5000	0.0002
Heat transfer fluids	5000	0.0002
Aqueous salt solutions	3000–5000	0.0003–0.0002

The selection of the design fouling coefficient will often be an economic decision. The optimum design will be obtained by balancing the extra capital cost of a larger exchanger against the savings in operating cost obtained from the longer operating time between cleaning that the larger area will give. Duplicate exchangers should be considered for severely fouling systems.

12.5. SHELL AND TUBE EXCHANGERS: CONSTRUCTION DETAILS

The shell and tube exchanger is by far the most commonly used type of heat-transfer equipment used in the chemical and allied industries. The advantages of this type are:

1. The configuration gives a large surface area in a small volume.
2. Good mechanical layout: a good shape for pressure operation.
3. Uses well-established fabrication techniques.
4. Can be constructed from a wide range of materials.

5. Easily cleaned.
6. Well-established design procedures.

Essentially, a shell and tube exchanger consists of a bundle of tubes enclosed in a cylindrical shell. The ends of the tubes are fitted into tube sheets, which separate the shell-side and tube-side fluids. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly of baffles and tubes is held together by support rods and spacers, Figure 12.2.

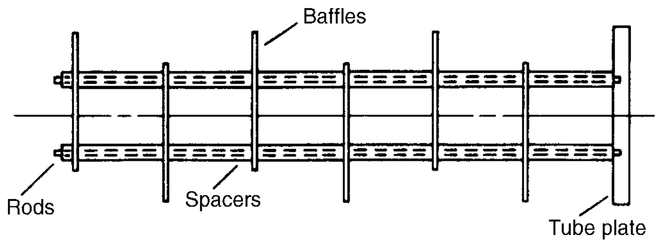


Figure 12.2. Baffle spacers and tie rods

Exchanger types

The principal types of shell and tube exchanger are shown in Figures 12.3 to 12.8. Diagrams of other types and full details of their construction can be found in the heat-exchanger standards (see Section 12.5.1.). The standard nomenclature used for shell and tube exchangers is given below; the numbers refer to the features shown in Figures 12.3 to 12.8.

Nomenclature

Part number

1. Shell
2. Shell cover
3. Floating-head cover
4. Floating-tube plate
5. Clamp ring
6. Fixed-tube sheet (tube plate)
7. Channel (end-box or header)
8. Channel cover
9. Branch (nozzle)
10. Tie rod and spacer
11. Cross baffle or tube-support plate
12. Impingement baffle
13. Longitudinal baffle
14. Support bracket
15. Floating-head support
16. Weir
17. Split ring
18. Tube
19. Tube bundle
20. Pass partition
21. Floating-head gland (packed gland)
22. Floating-head gland ring
23. Vent connection
24. Drain connection
25. Test connection
26. Expansion bellows
27. Lifting ring

The simplest and cheapest type of shell and tube exchanger is the fixed tube sheet design shown in Figure 12.3. The main disadvantages of this type are that the tube bundle cannot be removed for cleaning and there is no provision for differential expansion of the shell and tubes. As the shell and tubes will be at different temperatures, and may be of different materials, the differential expansion can be considerable and the use of this type is limited to temperature differences up to about 80°C. Some provision for expansion can be made by including an expansion loop in the shell (shown dotted on Figure 12.3) but their use is limited to low shell pressure; up to about 8 bar. In the other types, only one end of the tubes is fixed and the bundle can expand freely.

The U-tube (U-bundle) type shown in Figure 12.4 requires only one tube sheet and is cheaper than the floating-head types; but is limited in use to relatively clean fluids as the tubes and bundle are difficult to clean. It is also more difficult to replace a tube in this type.

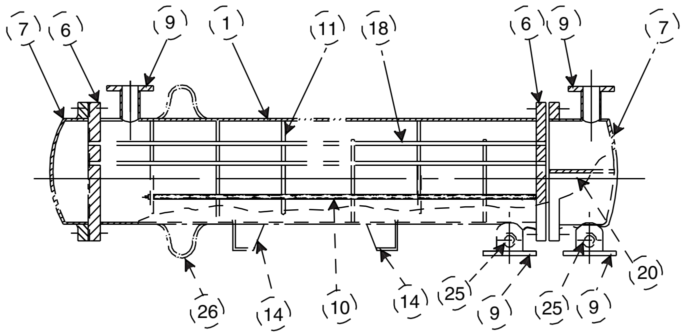


Figure 12.3. Fixed-tube plate (based on figures from BS 3274: 1960)

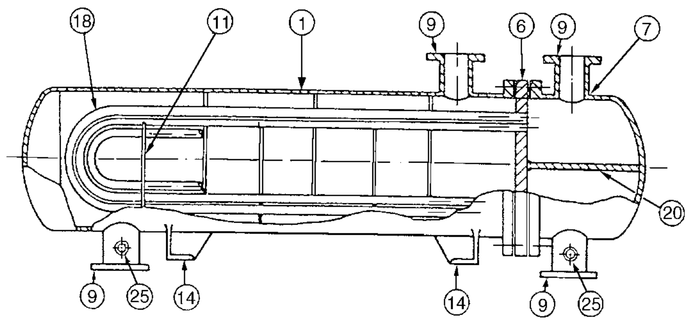


Figure 12.4. U-tube (based on figures from BS 3274: 1960)

Exchangers with an internal floating head, Figures 12.5 and 12.6, are more versatile than fixed head and U-tube exchangers. They are suitable for high-temperature differentials and, as the tubes can be rodded from end to end and the bundle removed, are easier to clean and can be used for fouling liquids. A disadvantage of the pull-through design, Figure 12.5, is that the clearance between the outermost tubes in the bundle and the shell must be made greater than in the fixed and U-tube designs to accommodate the floating-head flange, allowing fluid to bypass the tubes. The clamp ring (split flange design), Figure 12.6, is used to reduce the clearance needed. There will always be a danger of leakage occurring from the internal flanges in these floating head designs.

In the external floating head designs, Figure 12.7, the floating-head joint is located outside the shell, and the shell sealed with a sliding gland joint employing a stuffing box. Because of the danger of leaks through the gland, the shell-side pressure in this type is usually limited to about 20 bar, and flammable or toxic materials should not be used on the shell side.

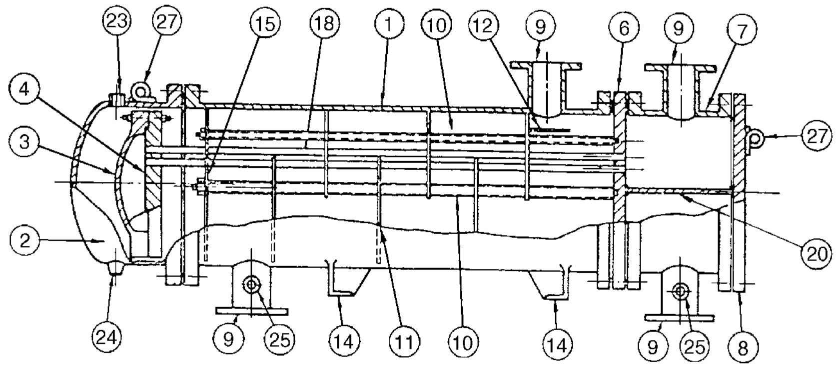


Figure 12.5. Internal floating head without clamp ring (based on figures from BS 3274: 1960)

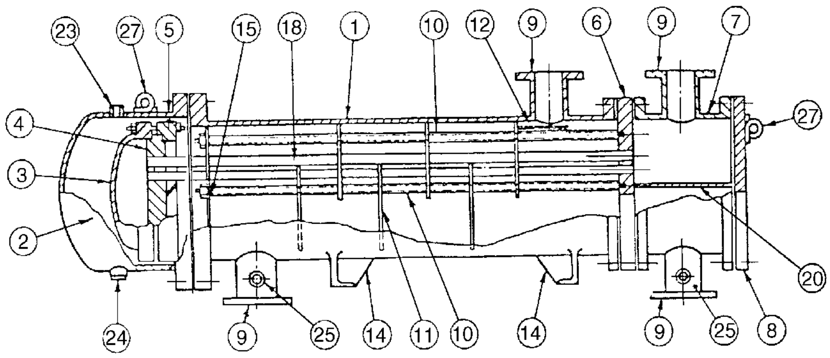


Figure 12.6. Internal floating head with clamp ring (based on figures from BS 3274: 1960)

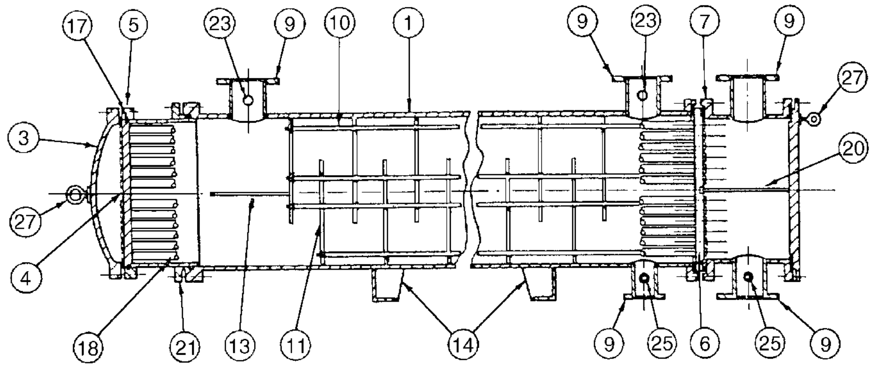


Figure 12.7. External floating head, packed gland (based on figures from BS 3274: 1960)

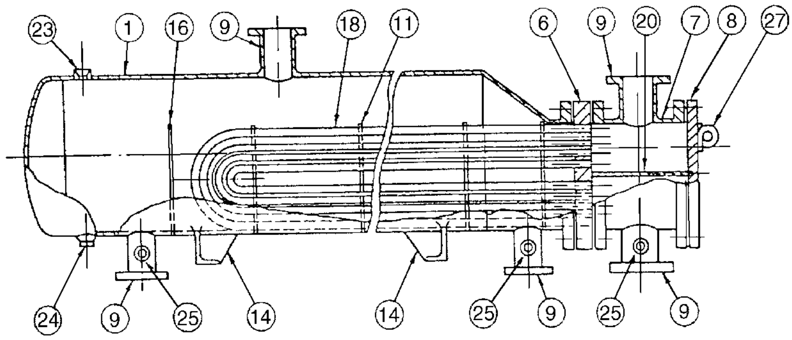


Figure 12.8. Kettle reboiler with U-tube bundle (based on figures from BS 3274: 1960)

12.5.1. Heat-exchanger standards and codes

The mechanical design features, fabrication, materials of construction, and testing of shell and tube exchangers is covered by British Standard, BS 3274. The standards of the American Tubular Heat Exchanger Manufacturers Association, the TEMA standards, are also universally used. The TEMA standards cover three classes of exchanger: class R covers exchangers for the generally severe duties of the petroleum and related industries; class C covers exchangers for moderate duties in commercial and general process applications; and class B covers exchangers for use in the chemical process industries. The British and American standards should be consulted for full details of the mechanical design features of shell and tube exchangers; only brief details will be given in this chapter.

The standards give the preferred shell and tube dimensions; the design and manufacturing tolerances; corrosion allowances; and the recommended design stresses for materials of construction. The shell of an exchanger is a pressure vessel and will be designed in accordance with the appropriate national pressure vessel code or standard; see Chapter 13, Section 13.2. The dimensions of standard flanges for use with heat exchangers are given in BS 3274, and in the TEMA standards.

In both the American and British standards dimensions are given in feet and inches, so these units have been used in this chapter with the equivalent values in SI units given in brackets.

12.5.2. Tubes

Dimensions

Tube diameters in the range $\frac{5}{8}$ in. (16 mm) to 2 in. (50 mm) are used. The smaller diameters $\frac{5}{8}$ to 1 in. (16 to 25 mm) are preferred for most duties, as they will give more compact, and therefore cheaper, exchangers. Larger tubes are easier to clean by mechanical methods and would be selected for heavily fouling fluids.

The tube thickness (gauge) is selected to withstand the internal pressure and give an adequate corrosion allowance. Steel tubes for heat exchangers are covered by BS 3606 (metric sizes); the standards applicable to other materials are given in BS 3274. Standard diameters and wall thicknesses for steel tubes are given in Table 12.3.

Table 12.3. Standard dimensions for steel tubes

Outside diameter (mm)	Wall thickness (mm)				
16	1.2	1.6	2.0	—	—
20	—	1.6	2.0	2.6	—
25	—	1.6	2.0	2.6	3.2
30	—	1.6	2.0	2.6	3.2
38	—	—	2.0	2.6	3.2
50	—	—	2.0	2.6	3.2

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), 24 ft (7.32 m). For a given surface area, the use of longer tubes will reduce the shell diameter; which will generally result in a lower cost exchanger, particularly for high shell pressures. The optimum tube length to shell diameter 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 those on the inside. The average length needs to be estimated for use in the thermal design. U-tubes will be bent from standard tube lengths and cut to size.

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

As a guide, $\frac{3}{4}$ in. (19 mm) is a good trial diameter with which to start design calculations.

Tube arrangements

The tubes in an exchanger are usually arranged in an equilateral triangular, square, or rotated square pattern; see Figure 12.9.

The triangular and rotated square patterns give higher heat-transfer rates, but at the expense of a higher pressure drop than the square pattern. A square, or rotated square arrangement, is used for heavily fouling fluids, where it is necessary to mechanically clean

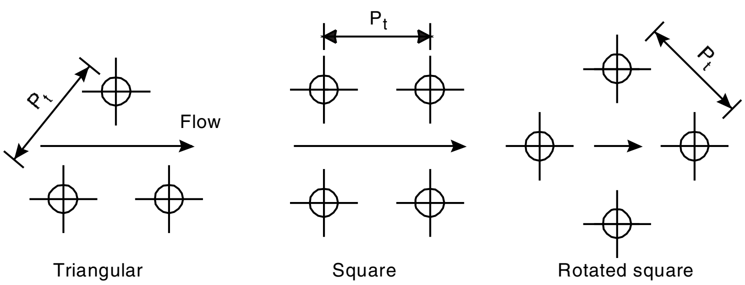


Figure 12.9. Tube patterns

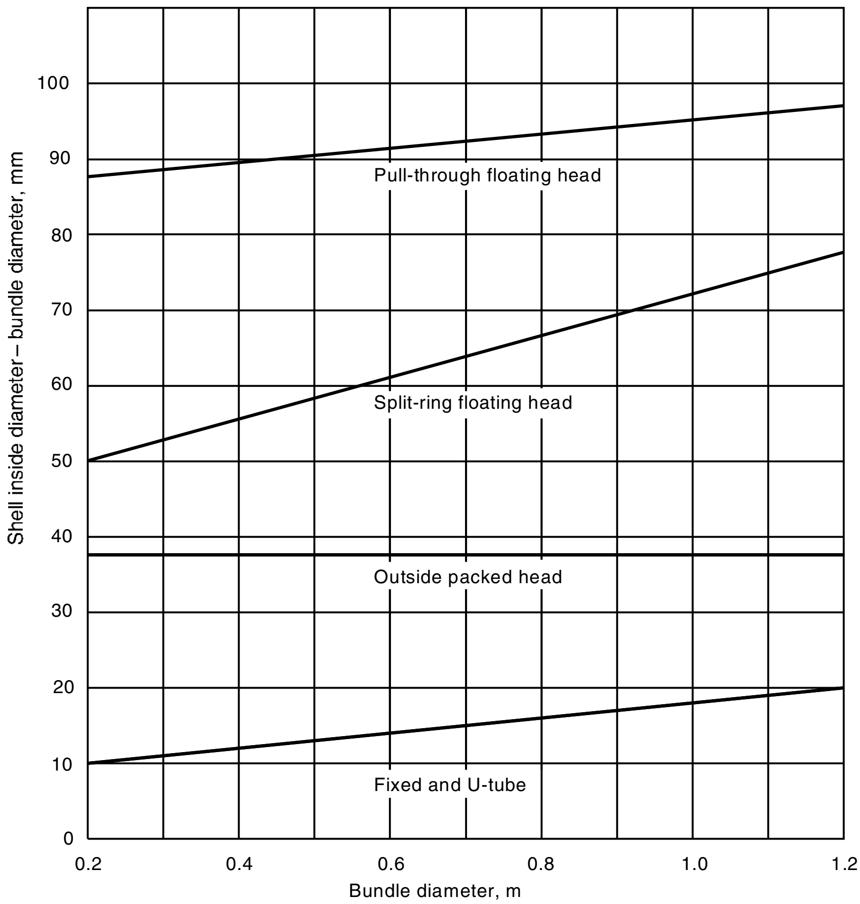


Figure 12.10. Shell-bundle clearance

the outside of the tubes. The recommended tube pitch (distance between tube centres) is 1.25 times the tube outside diameter; and this will normally be used unless process requirements dictate otherwise. Where a square pattern is used for ease of cleaning, the recommended minimum clearance between the tubes is 0.25 in. (6.4 mm).

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 sixteen 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 2, 4 and 6 tube passes are shown in Figure 12.11. The layouts for higher numbers of passes are given by Saunders (1988).

12.5.3. 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:

Minimum shell thickness

Nominal shell	Carbon steel		Alloy
dia., mm	pipe	plate	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 12.10 (as given on p. 646).

12.5.4. Tube-sheet layout (tube count)

The bundle diameter will depend 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.

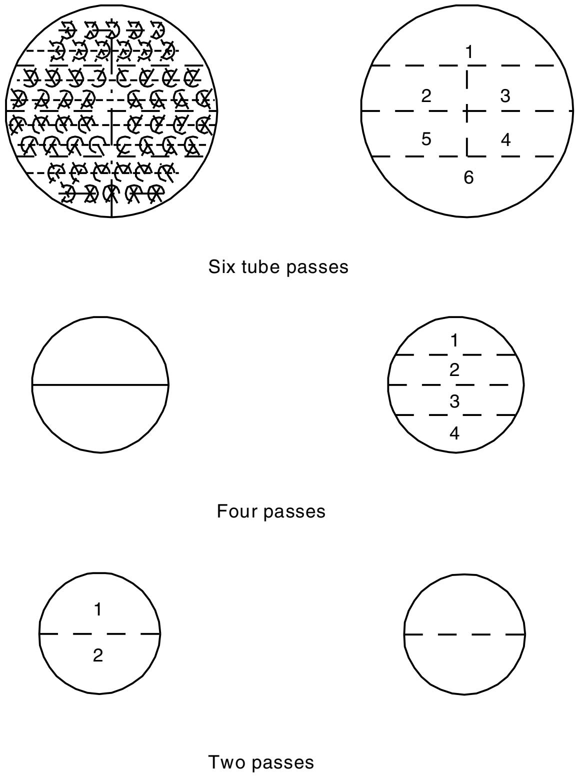


Figure 12.11. Tube arrangements, showing pass-partitions in headers

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 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 which 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), Perry *et al.* (1997), and Saunders (1988).

Some typical tube arrangements are shown in Appendix I.

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

12.5.5. Shell types (passes)

The principal shell arrangements are shown in Figure 12.12a–e. 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 will be 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. One method of sealing the longitudinal shell-side baffle is shown in Figure 12.12f.

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; where m is the number of shell passes and n the number of tube passes.

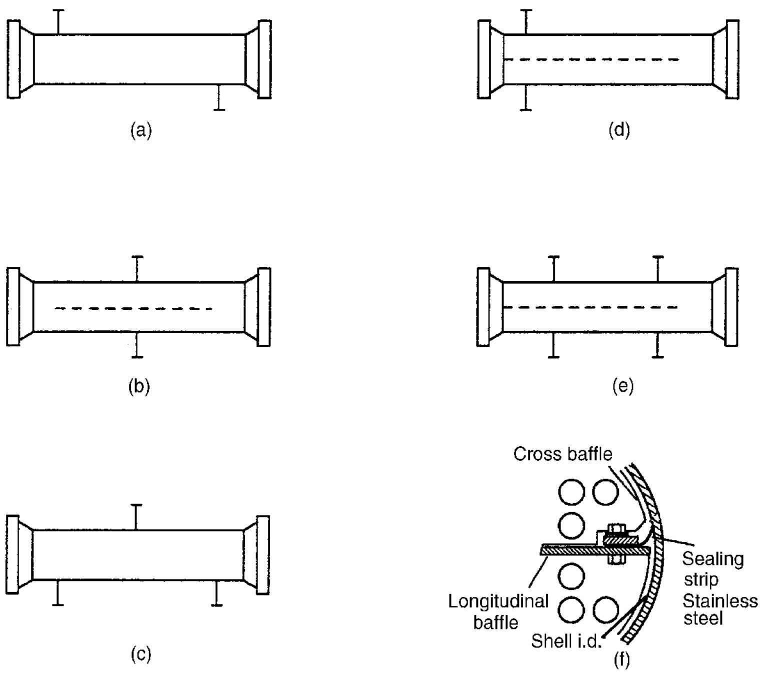


Figure 12.12. Shell types (pass arrangements). (*a*) One-pass shell (E shell) (*b*) Split flow (G shell) (*c*) Divided flow (J shell) (*d*) Two-pass shell with longitudinal baffle (F shell) (*e*) Double split flow (H shell)

So 1/2 describes an exchanger with 1 shell pass and 2 tube passes, and 2/4 an exchanger with 2 shell passes and 4 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 12.13*a*, other types are shown in Figures 12.13*b*, *c* and *d*.

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

If the arrangement shown in Figure 12.13*a* 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 per cent are used. Generally, a baffle cut of 20 to 25 per cent will be the optimum, giving good heat-transfer rates, without excessive drop. There will be some leakage of fluid round the 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.

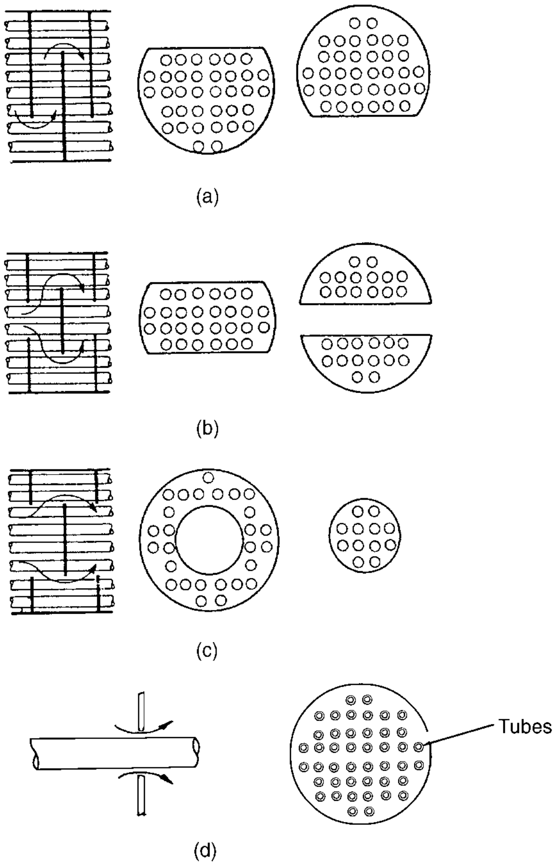


Figure 12.13. Types of baffle used in shell and tube heat exchangers. (*a*) Segmental (*b*) Segmental and strip (*c*) Disc and doughnut (*d*) Orifice

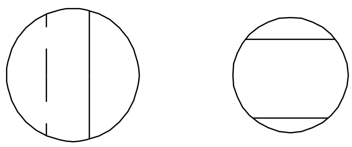


Figure 12.14. Baffles for condensers

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)

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

The minimum thickness to be used for baffles and support plates are 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.

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 vaporisers, 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 will depend on the shell diameter, and will range from 4, 16 mm diameter rods, for exchangers under 380 mm diameter; to 8, 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 BS 5500 and is discussed in Chapter 13. Design formulae for calculating tube sheet thicknesses are also given in the TEMA standards.

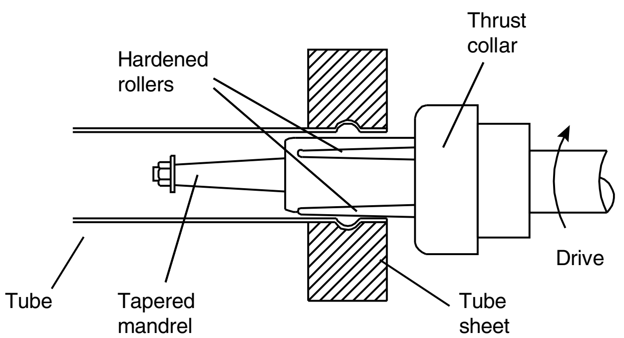


Figure 12.15. Tube rolling

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 leaf-proof joint, but not overthinned, weakening the tube. The tube holes are normally grooved, Figure 12.16*a*, to lock the tubes more firmly in position and to prevent the joint from being loosened by the differential expansion

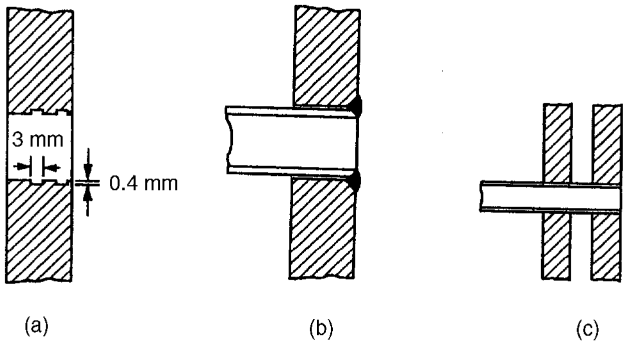


Figure 12.16. Tube/tube sheet joints

of the shell and tubes. When it is essential to guarantee a leak-proof joint the tubes can be welded to the sheet, Figure 12.16*b*. This will add to the cost of the exchanger; not only due to the cost of welding, but also because a wider tube spacing will be needed.

The tube sheet forms the barrier between the shell and tube fluids, and where it is essential for safety or process reasons to prevent any possibility of intermixing due to leakage at the tube sheet joint, double tube-sheets can be used, with the space between the sheets vented; Figure 12.16*c*.

To allow sufficient thickness to seal the tubes the tube sheet thickness should not be less than the tube outside diameter, up to about 25 mm diameter. Recommended minimum plate thicknesses are given in the standards.

The thickness of the tube sheet will reduce the effective length of the tube slightly, and this should be allowed for when calculating the area available for heat transfer. As a first approximation the length of the tubes can be reduced by 25 mm for each tube sheet.

12.5.10. Shell and header nozzles (branches)

Standard pipe sizes will be used for the inlet and outlet nozzles. It is important to avoid flow restrictions at the inlet and outlet nozzles to prevent excessive pressure drop and flow-induced vibration of the tubes. As well as omitting some tube rows (see Section 12.5.4), the baffle spacing is usually increased in the nozzle zone, to increase the flow area. For vapours and gases, where the inlet velocities will be high, the nozzle may be flared, or special designs used, to reduce the inlet velocities; Figure 12.17*a* and *b* (see p. 654). The extended shell design shown in Figure 12.17*b* also serves as an impingement plate. Impingement plates are used where the shell-side fluid contains liquid drops, or for high-velocity fluids containing abrasive particles.

12.5.11. Flow-induced tube vibrations

Premature failure of exchanger tubes can occur through vibrations induced by the shell-side fluid flow. Care must be taken in the mechanical design of large exchangers where

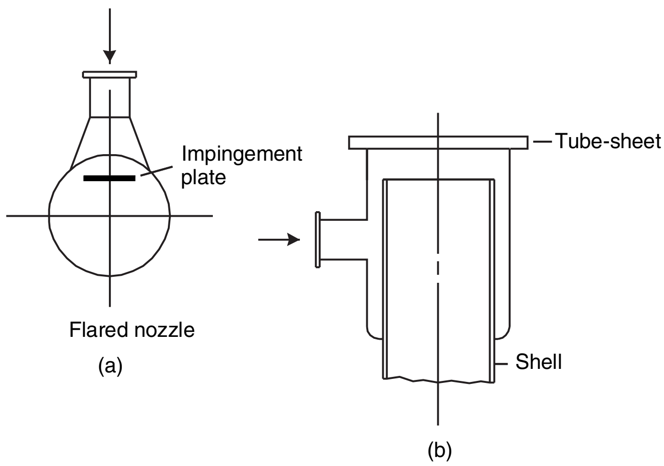


Figure 12.17. Inlet nozzle designs

the shell-side velocity is high, say greater than 3 m/s, to ensure that tubes are adequately supported.

The vibration induced by the fluid flowing over the tube bundle is caused principally by vortex shedding and turbulent buffeting. As fluid flows over a tube vortices are shed from the down-stream side which cause disturbances in the flow pattern and pressure distribution round the tube. Turbulent buffeting of tubes occurs at high flow-rates due to the intense turbulence at high Reynolds numbers.

The buffeting caused by vortex shedding or by turbulent eddies in the flow stream will cause vibration, but large amplitude vibrations will normally only occur above a certain critical flow velocity. Above this velocity the interaction with the adjacent tubes can provide a feed back path which reinforces the vibrations. Resonance will also occur if the vibrations approach the natural vibration frequency of the unsupported tube length. Under these conditions the magnitude of the vibrations can increase dramatically leading to tube failure. Failure can occur either through the impact of one tube on another or through wear on the tube where it passes through the baffles.

For most exchanger designs, following the recommendations on support sheet spacing given in the standards will be sufficient to protect against premature tube failure from vibration. For large exchangers with high velocities on the shell-side the design should be analysed to check for possible vibration problems. The computer aided design programs for shell-and-tube exchanger design available from commercial organisations, such as HTFS and HTRI (see Section 12.1), include programs for vibration analysis.

Much work has been done on tube vibration over the past 20 years, due to an increase in the failure of exchangers as larger sizes and higher flow-rates have been used. Discussion of this work is beyond the scope of this book; for review of the methods used see Saunders (1988) and Singh and Soler (1992).

See also, the Engineering Science Data Unit Design Guide ESDU 87019, which gives a clear explanation of mechanisms causing tube vibration in shell and tube heat exchangers, and their prediction and prevention.

12.6. MEAN TEMPERATURE DIFFERENCE (TEMPERATURE DRIVING FORCE)

Before equation 12.1 can be used to determine the heat transfer area required for a given duty, an estimate of the mean temperature difference ΔT_m must be made. This will normally be calculated from the terminal temperature differences: the difference in the fluid temperatures at the inlet and outlet of the exchanger. The well-known "logarithmic mean" temperature difference (see Volume 1, Chapter 9) is only applicable to sensible heat transfer in true co-current or counter-current flow (linear temperature-enthalpy curves). For counter-current flow, Figure 12.18*a*, the logarithmic mean temperature is given by:

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

where ΔT_{lm} = log mean temperature difference,
T_1 = hot fluid temperature, inlet,
T_2 = hot fluid temperature, outlet,
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 12.18*b* and *c* show typical temperature profiles for an exchanger with one shell pass and two tube passes (a 1 : 2 exchanger). Figure 12.18*c* shows a 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:

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

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:

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

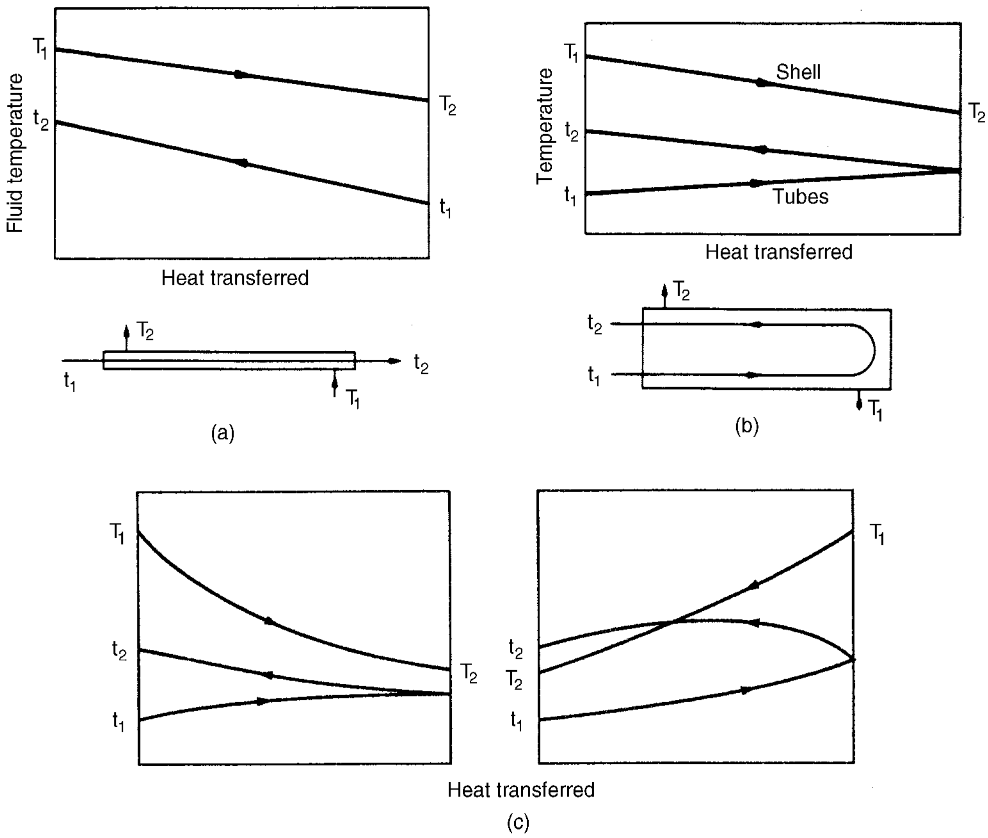


Figure 12.18. Temperature profiles (*a*) Counter-current flow (*b*) 1 : 2 exchanger (*c*) Temperature cross

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 shell : 2 tube pass 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 shell : 2 tube pass 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 2 shell passes and 4, or multiples of 4, tube passes is shown in Figure 12.20, and that for divided and split flow shells in Figures 12.21 and 12.22.

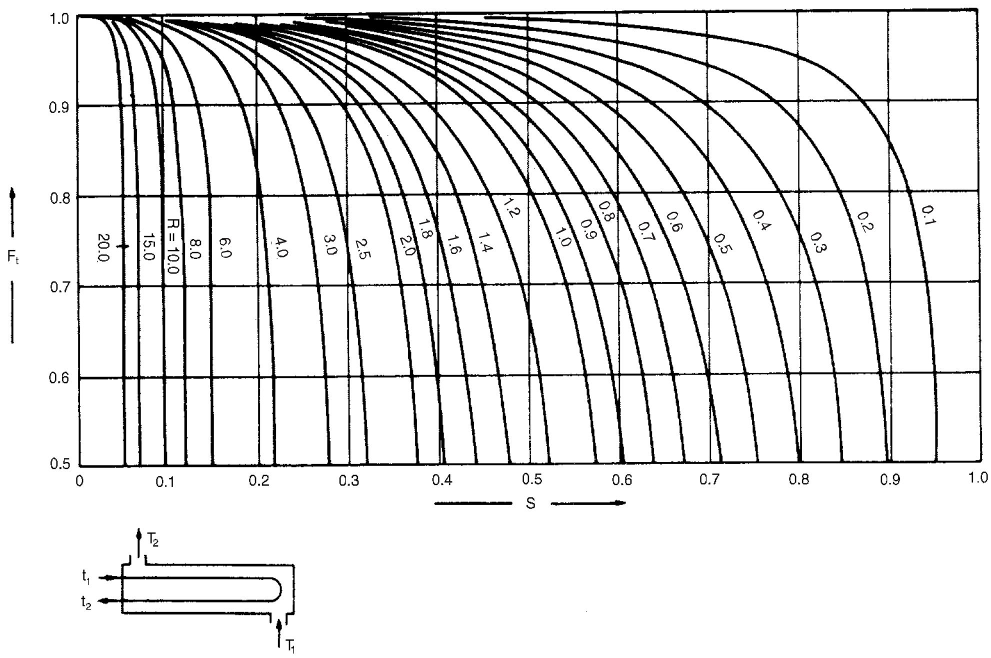


Figure 12.19. Temperature correction factor: one shell pass; two or more even tube passes

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

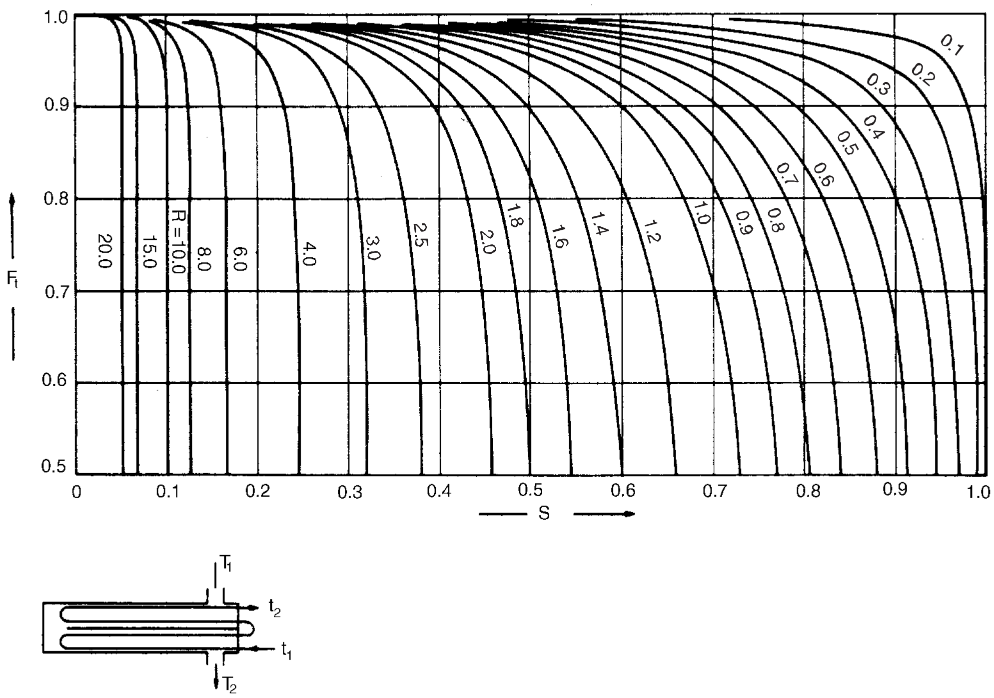


Figure 12.20. Temperature correction factor: two shell passes; four or multiples of four tube passes

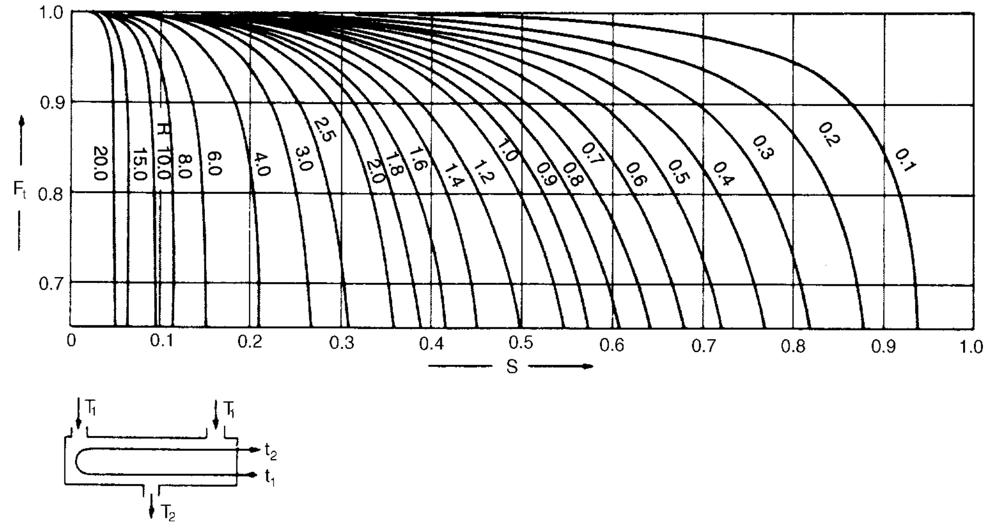


Figure 12.21. Temperature correction factor: divided-flow shell; two or more even-tube passes

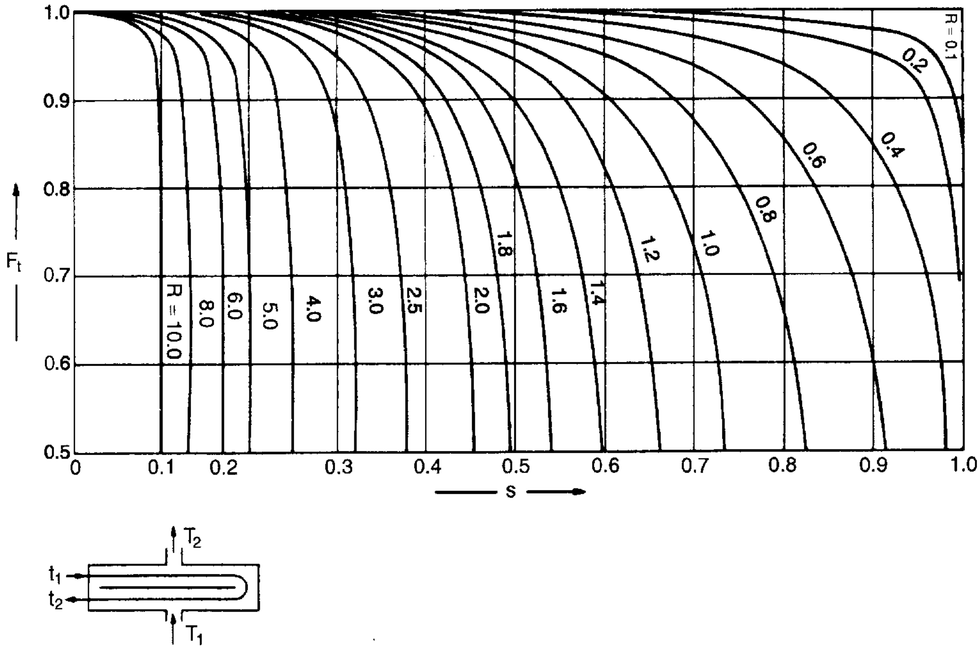


Figure 12.22. Temperature correction factor, split flow shell, 2 tube pass

assumptions, and gives F_t curves for conditions when all the assumptions are not met; see also Butterworth (1973) and Emerson (1973).

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 which show the effect of leakage on the correction factor for a 1 shell pass : 2 tube pass 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 inlet temperature of the hot stream, Figure 12.18*c*.

Where the F_t curve is near vertical values cannot be read accurately, and this 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 which 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.

Where 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.

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 will 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 will give 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.

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.

Vapours

For vapours, the velocity used will depend on the operating pressure and fluid density; the lower values in the ranges given below will apply to high molecular weight materials.

Vacuum	50 to 70 m/s
Atmospheric pressure	10 to 30 m/s
High pressure	5 to 10 m/s

12.7.3. Stream temperatures

The closer the temperature approach used (the difference between the outlet temperature of one stream and the inlet temperature of the other stream) the larger will be the heat-transfer area required for a given duty. The optimum value will depend on the application, and can only be determined by making an economic analysis of alternative designs. As a general guide the greater temperature difference should be at least 20°C, and the least temperature difference 5 to 7°C for coolers using cooling water, and 3 to 5°C using refrigerated brines. The maximum temperature rise in recirculated cooling water is limited to around 30°C. Care should be taken to ensure that cooling media temperatures are kept well above the freezing point of the process materials. When the heat exchange is between process fluids for heat recovery the optimum approach temperatures will normally not be lower than 20°C.

12.7.4. Pressure drop

In many applications the pressure drop available to drive the fluids through the exchanger will be set by the process conditions, and the available pressure drop will vary from a few millibars in vacuum service to several bars in pressure systems.

When the designer is free to select the pressure drop an economic analysis can be made to determine the exchanger design which gives the lowest operating costs, taking into consideration both capital and pumping costs. However, a full economic analysis will only be justified for very large, expensive, exchangers. The values suggested below can be used as a general guide, and will normally give designs that are near the optimum.

Liquids

Viscosity	<1 mN s/m^2	35 kN/m^2
	1 to 10 mN s/m^2	50–70 kN/m^2

Gas and vapours

High vacuum	0.4–0.8 kN/m^2
Medium vacuum	$0.1 \times$ absolute pressure
1 to 2 bar	$0.5 \times$ system gauge pressure
Above 10 bar	$0.1 \times$ system gauge pressure

When a high-pressure drop is utilised, care must be taken to ensure that the resulting high fluid velocity does not cause erosion or flow-induced tube vibration.

12.7.5. Fluid physical properties

The fluid physical properties required for heat-exchanger design are: density, viscosity, thermal conductivity and temperature-enthalpy correlations (specific and latent heats). Sources of physical property data are given in Chapter 8. The thermal conductivities of commonly used tube materials are given in Table 12.6.

Table 12.6. Conductivity of metals

Metal	Temperature (°C)	k_w(W/m°C)
Aluminium	0	202
	100	206
Brass	0	97
(70 Cu, 30 Zn)	100	104
	400	116
Copper	0	388
	100	378
Nickel	0	62
	212	59
Cupro-nickel (10 per cent Ni)	0–100	45
Monel	0–100	30
Stainless steel (18/8)	0–100	16
Steel	0	45
	100	45
	600	36
Titanium	0–100	16

In the correlations used to predict heat-transfer coefficients, the physical properties are usually evaluated at the mean stream temperature. This is satisfactory when the temperature change is small, but can cause a significant error when the change in temperature is large. In these circumstances, a simple, and safe, procedure is to evaluate the heat-transfer coefficients at the stream inlet and outlet temperatures and use the lowest of the two values. Alternatively, the method suggested by Frank (1978) can be used; in which equations 12.1 and 12.3 are combined:

$$Q = \frac{A[U_2(T_1 - t_2) - U_1(T_2 - t_1)]}{\ln\left[\dfrac{U_2(T_1 - t_2)}{U_1(T_2 - t_1)}\right]} \tag{12.9}$$

where U_1 and U_2 are evaluated at the ends of the exchanger. Equation 12.9 is derived by assuming that the heat-transfer coefficient varies linearly with temperature.

If the variation in the physical properties is too large for these simple methods to be used it will be necessary to divide the temperature-enthalpy profile into sections and evaluate the heat-transfer coefficients and area required for each section.

12.8. TUBE-SIDE HEAT-TRANSFER COEFFICIENT AND PRESSURE DROP (SINGLE PHASE)

12.8.1. Heat transfer

Turbulent flow

Heat-transfer data for turbulent flow inside conduits of uniform cross-section are usually correlated by an equation of the form:

$$Nu = CRe^a Pr^b \left(\frac{\mu}{\mu_w}\right)^c \tag{12.10}$$

where Nu = Nusselt number = $(h_i d_e / k_f)$,
Re = Reynolds number = $(\rho u_t d_e/\mu) = (G_t d_e/\mu)$,
Pr = Prandtl number = $(C_p \mu / k_f)$
and: h_i = inside coefficient, W/m^2°C,
d_e = equivalent (or hydraulic mean) diameter, m

$$d_e = \frac{4 \times \text{cross-sectional area for flow}}{\text{wetted perimeter}} = d_i \text{ for tubes,}$$

u_t = fluid velocity, m/s,
k_f = fluid thermal conductivity, W/m°C,
G_t = mass velocity, mass flow per unit area, kg/m^{2}s,
μ = fluid viscosity at the bulk fluid temperature, Ns/m^2,
μ_w = fluid viscosity at the wall,
C_p = fluid specific heat, heat capacity, J/kg°C.

The index for the Reynolds number is generally taken as 0.8. That for the Prandtl number can range from 0.3 for cooling to 0.4 for heating. The index for the viscosity factor is normally taken as 0.14 for flow in tubes, from the work of Sieder and Tate (1936), but some workers report higher values. A general equation that can be used for exchanger design is:

$$Nu = C Re^{0.8} Pr^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{12.11}$$

where C = 0.021 for gases,
= 0.023 for non-viscous liquids,
= 0.027 for viscous liquids.

It is not really possible to find values for the constant and indexes to cover the complete range of process fluids, from gases to viscous liquids, but the values predicted using equation 12.11 should be sufficiently accurate for design purposes. The uncertainty in the prediction of the shell-side coefficient and fouling factors will usually far outweigh any error in the tube-side value. Where a more accurate prediction than that given by equation 12.11 is required, and justified, the data and correlations given in the Engineering Science Data Unit reports are recommended: ESDU 92003 and 93018 (1998).

Butterworth (1977) gives the following equation, which is based on the ESDU work:

$$St = E Re^{-0.205} Pr^{-0.505} \tag{12.12}$$

where St = Stanton number = $(Nu/RePr) = (h_i/\rho u_t C_p)$
and $E = 0.0225 \exp(-0.0225 (\ln Pr)^2)$.

Equation 12.12 is applicable at Reynolds numbers greater than 10,000.

Hydraulic mean diameter

In some texts the equivalent (hydraulic mean) diameter is defined differently for use in calculating the heat transfer coefficient in a conduit or channel, than for calculating the pressure drop. The perimeter through which the heat is being transferred is used in place of the total wetted perimeter. In practice, the use of d_e calculated either way will make little difference to the value of the estimated overall coefficient; as the film coefficient is only, roughly, proportional to $d_e^{-0.2}$.

It is the full wetted perimeter that determines the flow regime and the velocity gradients in a channel. So, in this book, d_e determined using the full wetted perimeter will be used for both pressure drop and heat transfer calculations. The actual area through which the heat is transferred should, of course, be used to determine the rate of heat transfer; equation 12.1.

Laminar flow

Below a Reynolds number of about 2000 the flow in pipes will be laminar. Providing the natural convection effects are small, which will normally be so in forced convection, the following equation can be used to estimate the film heat-transfer coefficient:

$$Nu = 1.86(RePr)^{0.33}\left(\frac{d_e}{L}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{12.13}$$

Where L is the length of the tube in metres.

If the Nusselt number given by equation 12.13 is less than 3.5, it should be taken as 3.5.

In laminar flow the length of the tube can have a marked effect on the heat-transfer rate for length to diameter ratios less than 500.

Transition region

In the flow region between laminar and fully developed turbulent flow heat-transfer coefficients cannot be predicted with certainty, as the flow in this region is unstable, and the transition region should be avoided in exchanger design. If this is not practicable the coefficient should be evaluated using both equations 12.11 and 12.13 and the least 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 (see Volume 1, Chapters 3 and 9). The heat-transfer factor is defined by:

$$j_h = StPr^{0.67}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{12.14}$$

The use of the j_h factor enables 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 high 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:

$$\frac{h_i d_i}{k_f} = j_h RePr^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{12.15}$$

Note. Kern (1950), and other workers, define the heat transfer factor as:

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

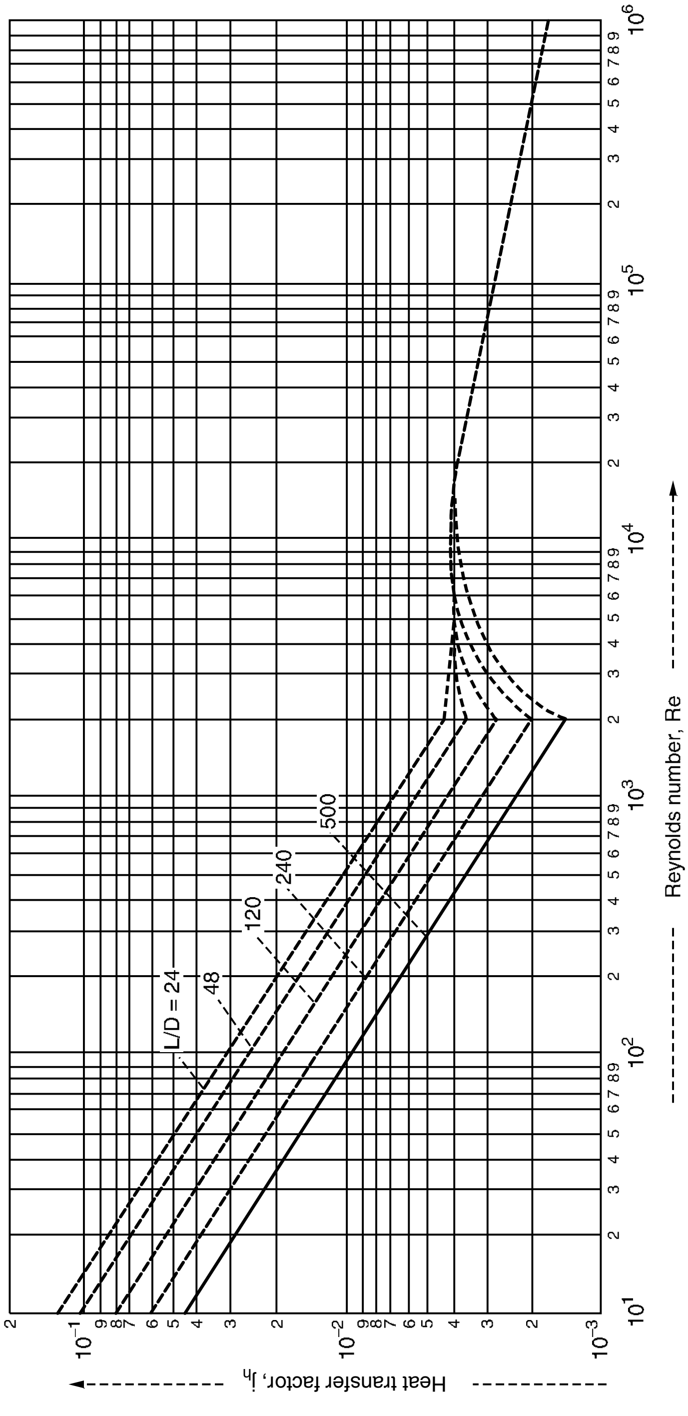


Figure 12.23. Tube-side heat-transfer factor

The relationship between j_h and j_H is given by:

$$j_H = j_h Re$$

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:

$$h_i(t_w - t) = U(T - t) \tag{12.16}$$

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):

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

where h_i = inside coefficient, for water, $W/m^2{}^\circ C$,
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 loss in pipes (see Volume 1, Chapter 3). The basic equation for isothermal flow in pipes (constant temperature) is:

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

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

The flow in a heat exchanger will clearly not be 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:

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

$m = 0.25$ for laminar flow, $Re < 2100$,
$\quad = 0.14$ 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 (see Volume 1, Chapter 3) 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/m^2 (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 will be 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.

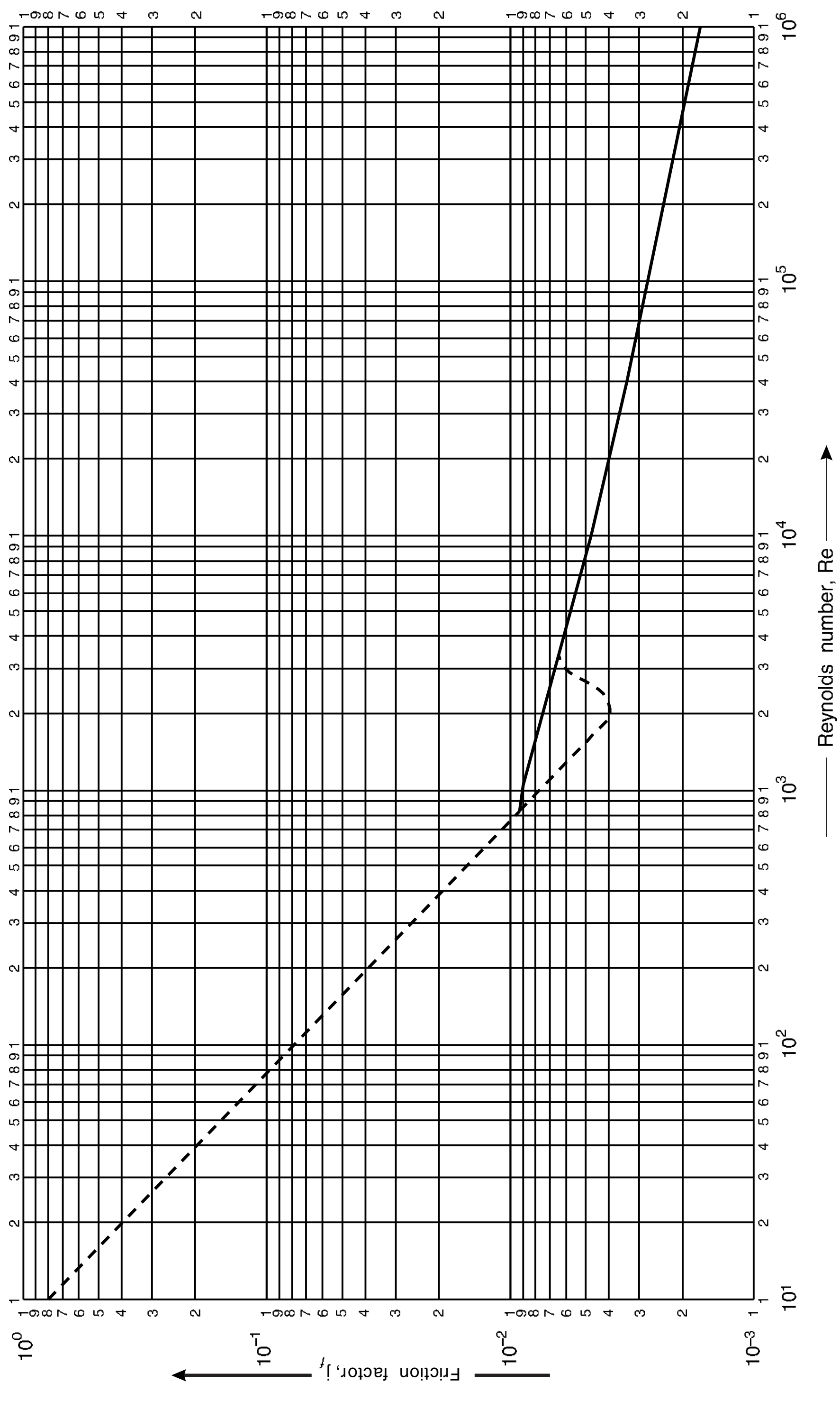


Figure 12.24. Tube-side friction factors

Note: The friction factor j_f is the same as the friction factor for pipes $\phi(= (R/\rho u^2))$, defined in Volume 1 Chapter 3.

12.9. SHELL-SIDE HEAT-TRANSFER AND PRESSURE DROP (SINGLE PHASE)

12.9.1. Flow pattern

The flow pattern in the shell of a segmentally baffled heat exchanger is complex, and this makes the prediction of the shell-side heat-transfer coefficient and pressure drop very much more difficult than for the tube-side. Though the baffles are installed to direct the flow across the tubes, the actual flow of the main stream of fluid will be a mixture of cross flow between the baffles, coupled with axial (parallel) flow in the baffle windows; as shown in Figure 12.25. Not all the fluid flow follows the path shown in Figure 12.25; some will leak through gaps formed by the clearances that have to be allowed for fabrication and assembly of the exchanger. These leakage and bypass streams are shown in Figure 12.26, which is based on the flow model proposed by Tinker (1951, 1958). In Figure 12.26, Tinker's nomenclature is used to identify the various streams, as follows:

Stream A is the tube-to-baffle leakage stream. The fluid flowing through the clearance between the tube outside diameter and the tube hole in the baffle.

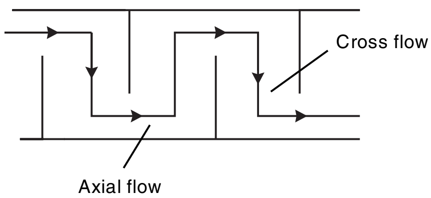


Figure 12.25. Idealised main stream flow

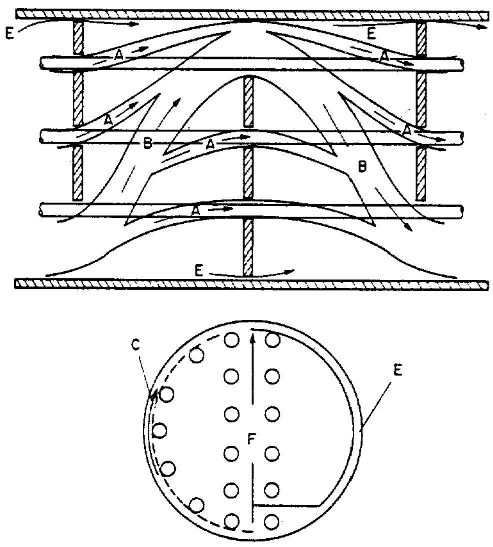


Figure 12.26. Shell-side leakage and by-pass paths

Stream B is the actual cross-flow stream.

Stream C is the bundle-to-shell bypass stream. The fluid flowing in the clearance area between the outer tubes in the bundle (bundle diameter) and the shell.

Stream E is the baffle-to-shell leakage stream. The fluid flowing through the clearance between the edge of a baffle and the shell wall.

Stream F is the pass-partition stream. The fluid flowing through the gap in the tube arrangement due to the pass partition plates. Where the gap is vertical it will provide a low-pressure drop path for fluid flow.

Note. There is no stream D.

The fluid in streams C, E and F bypasses the tubes, which reduces the effective heat-transfer area.

Stream C is the main bypass stream and will be particularly significant in pull-through bundle exchangers, where the clearance between the shell and bundle is of necessity large. Stream C can be considerably reduced by using sealing strips; horizontal strips that block the gap between the bundle and the shell, Figure 12.27. Dummy tubes are also sometimes used to block the pass-partition leakage stream F.

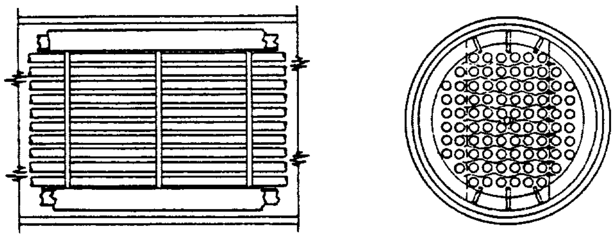


Figure 12.27. Sealing strips

The tube-to-baffle leakage stream A does not bypass the tubes, and its main effect is on pressure drop rather than heat transfer.

The clearances will tend to plug as the exchanger becomes fouled and this will increase the pressure drop; see Section 12.9.6.

12.9.2. Design methods

The complex flow pattern on the shell-side, and the great number of variables involved, make it difficult to predict the shell-side coefficient and pressure drop with complete assurance. In methods used for the design of exchangers prior to about 1960 no attempt was made to account for the leakage and bypass streams. Correlations were based on the total stream flow, and empirical methods were used to account for the performance of real exchangers compared with that for cross flow over ideal tube banks. Typical of these "bulk-flow" methods are those of Kern (1950) and Donohue (1955). Reliable predictions can only be achieved by comprehensive analysis of the contribution to heat transfer and pressure drop made by the individual streams shown in Figure 12.26. Tinker (1951, 1958) published the first detailed stream-analysis method for predicting shell-side heat-transfer coefficients and pressure drop, and the methods subsequently developed have been based on his model. Tinker's presentation is difficult to follow, and his method difficult and tedious to apply in manual calculations. It has been simplified by Devore (1961, 1962); using standard tolerance for commercial exchangers and only a limited number of baffle cuts. Devore gives nomographs that facilitate the application of the method in manual calculations. Mueller (1973) has further simplified Devore's method and gives an illustrative example.

The Engineering Sciences Data Unit has also published a method for estimating shell-side the pressure drop and heat transfer coefficient, EDSU Design Guide 83038 (1984). The method is based on a simplification of Tinker's work. It can be used for hand calculations, but as iterative procedures are involved it is best programmed for use with personal computers.

Tinker's model has been used as the basis for the proprietary computer methods developed by Heat Transfer Research Incorporated; see Palen and Taborek (1969), and by Heat Transfer and Fluid Flow Services; see Grant (1973).

Bell (1960, 1963) developed a semi-analytical method based on work done in the cooperative research programme on shell and tube exchangers at the University of Delaware. His method accounts for the major bypass and leakage streams and is suitable for a manual calculation. Bell's method is outlined in Section 12.9.4 and illustrated in Example 12.3.

Though Kern's method does not take account of the bypass and leakage streams, it is simple to apply and is accurate enough for preliminary design calculations, and for designs where uncertainty in other design parameters is such that the use of more elaborate methods is not justified. Kern's method is given in Section 12.9.3 and is illustrated in Examples 12.1 and 12.3.

12.9.3. Kern's method

This method was based on experimental work on commercial exchangers with standard tolerances and will give a reasonably satisfactory prediction of the heat-transfer coefficient for standard designs. 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 and friction factors are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and shell diameter. As the cross-sectional area for flow will vary across the shell diameter, the linear and mass velocities are based on the maximum area for cross-flow: that at the shell equator. The shell equivalent diameter is calculated using the flow area between the tubes taken in the axial direction (parallel to the tubes) and the wetted perimeter of the tubes; see Figure 12.28.

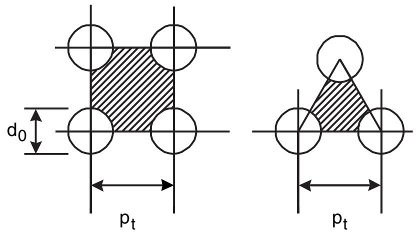


Figure 12.28. Equivalent diameter, cross-sectional areas and wetted perimeters

Shell-side j_h and j_f factors for use in this method are given in Figures 12.29 and 12.30, for various baffle cuts and tube arrangements. These figures are based on data given by Kern (1950) and by Ludwig (2001).

The procedure for calculating the shell-side heat-transfer coefficient and pressure drop for a single shell pass exchanger is given below:

Procedure

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

$$A_s = \frac{(p_t - d_o)D_s l_B}{p_t} \tag{12.21}$$

where p_t = tube pitch,
d_o = tube outside diameter,
D_s = shell inside diameter, m,
l_B = baffle spacing, m.

The term $(p_t - d_o)/p_t$ is the ratio of the clearance between tubes and the total distance between tube centres.

2. Calculate the shell-side mass velocity G_s and the linear velocity u_s:

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

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

where W_s = fluid flow-rate on the shell-side, kg/s,
ρ = shell-side fluid density, kg/m^3.

3. Calculate the shell-side equivalent diameter (hydraulic diameter), Figure 12.28. For a square pitch arrangement:

$$d_e = \frac{4\left(\dfrac{p_t^2 - \pi d_o^2}{4}\right)}{\pi d_o} = \frac{1.27}{d_o}(p_t^2 - 0.785 d_o^2) \tag{12.22}$$

For an equilateral triangular pitch arrangement:

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

where d_e = equivalent diameter, m.

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

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

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

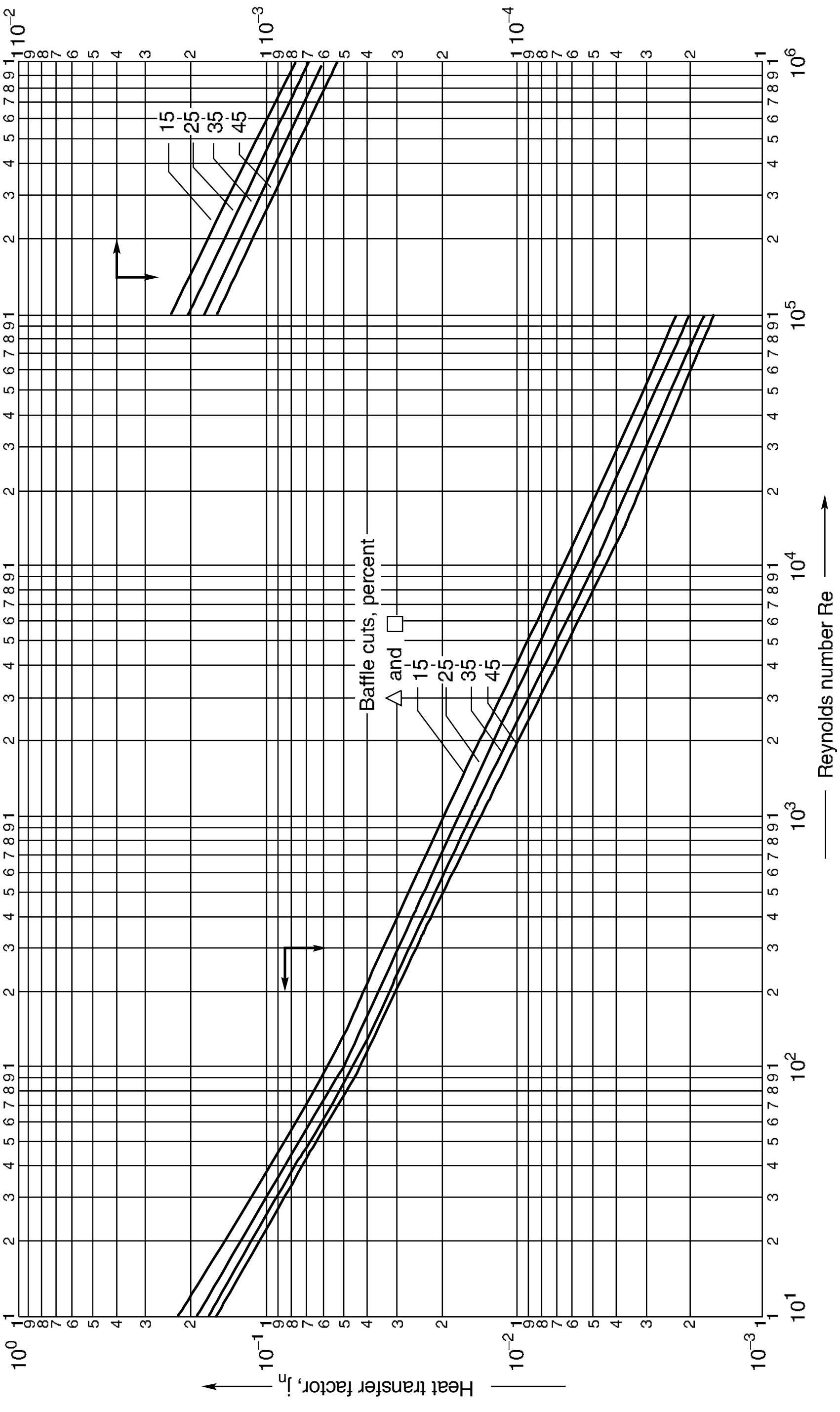


Figure 12.29. Shell-side heat-transfer factors, segmental baffles

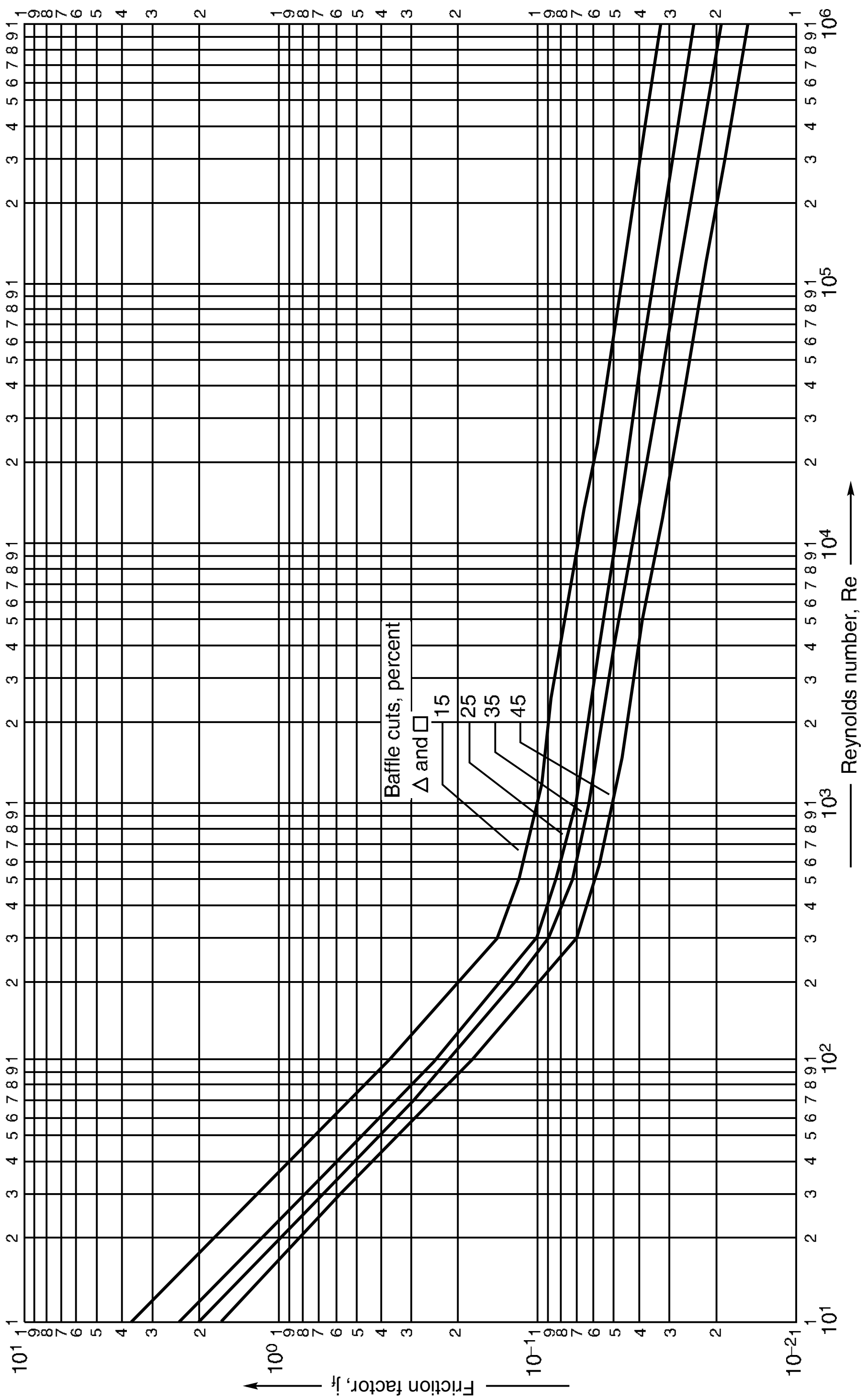


Figure 12.30. Shell-side friction factors, segmental baffles

coefficient h_s from:

$$Nu = \frac{h_s d_e}{k_f} = j_h Re Pr^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (12.25)$$

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:

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

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

The pressure loss in the shell nozzles will normally only be significant with gases. The nozzle pressure drop can be taken as equivalent to $1\frac{1}{2}$ velocity heads for the inlet and $\frac{1}{2}$ for the outlet, based on the nozzle area or the free area between the tubes in the row immediately adjacent to the nozzle, whichever is the least.

Example 12.1

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

Solution

Only the thermal design will be considered.

This example illustrates Kern's method.

Coolant is corrosive, so assign to tube-side.

Heat capacity methanol = 2.84 kJ/kg°C

$$\text{Heat load} = \frac{100{,}000}{3600} \times 2.84(95 - 40) = 4340 \text{ kW}$$

Heat capacity water = 4.2 kJ/kg°C

$$\text{Cooling water flow} = \frac{4340}{4.2(40 - 25)} = 68.9 \text{ kg/s}$$

$$\Delta T_{\text{lm}} = \frac{(95 - 40) - (40 - 25)}{\ln \dfrac{(95 - 40)}{(40 - 25)}} = 31°\text{C} \quad (12.4)$$

Use one shell pass and two tube passes

$$R = \frac{95 - 40}{40 - 25} = 3.67 \qquad (12.6)$$

$$S = \frac{40 - 25}{95 - 25} = 0.21 \qquad (12.7)$$

From Figure 12.19

$$F_t = 0.85$$

$$\Delta T_m = 0.85 \times 31 = 26^\circ\text{C}$$

From Figure 12.1

$$U = 600\ \text{W/m}^2\,{}^\circ\text{C}$$

Provisional area

$$A = \frac{4340 \times 10^3}{26 \times 600} = 278\ \text{m}^2 \qquad (12.1)$$

Choose 20 mm o.d., 16 mm i.d., 4.88-m-long tubes $\left(\frac{3}{4}\text{in.} \times 16\ \text{ft}\right)$, cupro-nickel.
Allowing for tube-sheet thickness, take

$$L = 4.83\ \text{m}$$

$$\text{Area of one tube} = 4.83 \times 20 \times 10^{-3}\pi = 0.303\ \text{m}^2$$

$$\text{Number of tubes} = \frac{278}{0.303} = \underline{\underline{918}}$$

As the shell-side fluid is relatively clean use 1.25 triangular pitch.

$$\text{Bundle diameter } D_b = 20\left(\frac{918}{0.249}\right)^{1/2.207} = 826\ \text{mm} \qquad (12.3b)$$

Use a split-ring floating head type.
From Figure 12.10, bundle diametrical clearance = 68 mm,

$$\text{shell diameter, } D_s = 826 + 68 = 894\ \text{mm.}$$

(*Note*. nearest standard pipe sizes are 863.6 or 914.4 mm).
Shell size could be read from standard tube count tables.

Tube-side coefficient

$$\text{Mean water temperature} = \frac{40 + 25}{2} = 33^\circ\text{C}$$

$$\text{Tube cross-sectional area} = \frac{\pi}{4} \times 16^2 = 201\ \text{mm}^2$$

$$\text{Tubes per pass} = \frac{918}{2} = 459$$

$$\text{Total flow area} = 459 \times 201 \times 10^{-6} = 0.092\ \text{m}^2$$

$$\text{Water mass velocity} = \frac{68.9}{0.092} = 749 \text{ kg/s m}^2$$

$$\text{Density water} = 995 \text{ kg/m}^3$$

$$\text{Water linear velocity} = \frac{749}{995} = 0.75 \text{ m/s}$$

$$h_i = \frac{4200(1.35 + 0.02 \times 33)0.75^{0.8}}{16^{0.2}} = 3852 \text{ W/m}^2\,^\circ\text{C} \qquad (12.17)$$

The coefficient can also be calculated using equation 12.15; this is done to illustrate use of this method.

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

$$\text{Viscosity of water} = 0.8 \text{ mNs/m}^2$$

$$\text{Thermal conductivity} = 0.59 \text{ W/m}^\circ\text{C}$$

$$Re = \frac{\rho u d_i}{\mu} = \frac{995 \times 0.75 \times 16 \times 10^{-3}}{0.8 \times 10^{-3}} = 14{,}925$$

$$Pr = \frac{C_p \mu}{k_f} = \frac{4.2 \times 10^3 \times 0.8 \times 10^{-3}}{0.59} = 5.7$$

$$\text{Neglect} \left(\frac{\mu}{\mu_w}\right)$$

$$\frac{L}{d_i} = \frac{4.83 \times 10^3}{16} = 302$$

From Figure 12.23, $j_h = 3.9 \times 10^{-3}$

$$h_i = \frac{0.59}{16 \times 10^{-3}} \times 3.9 \times 10^{-3} \times 14{,}925 \times 5.7^{0.33} = 3812 \text{ W/m}^2\,^\circ\text{C}$$

Checks reasonably well with value calculated from equation 12.17; use lower figure.

Shell-side coefficient

Choose baffle spacing $= \dfrac{D_s}{5} = \dfrac{894}{5} = 178$ mm.

$$\text{Tube pitch} = 1.25 \times 20 = 25 \text{ mm}$$

$$\text{Cross-flow area } A_s = \frac{(25 - 20)}{25} 894 \times 178 \times 10^{-6} = 0.032 \text{ m}^2 \qquad (12.21)$$

$$\text{Mass velocity, } G_S = \frac{100{,}000}{3600} \times \frac{1}{0.032} = 868 \text{ kg/s m}^2$$

$$\text{Equivalent diameter } d_e = \frac{1.1}{20}(25^2 - 0.917 \times 20^2) = 14.4 \text{ mm} \qquad (12.23)$$

$$\text{Mean shell side temperature} = \frac{95+40}{2} = 68^\circ\text{C}$$

Methanol density = 750 kg/m^3

Viscosity = 0.34 mNs/m^2

Heat capacity = 2.84 kJ/kg°C

Thermal conductivity = 0.19 W/m°C

$$Re = \frac{G_s d_e}{\mu} = \frac{868 \times 14.4 \times 10^{-3}}{0.34 \times 10^{-3}} = 36{,}762 \quad (12.24)$$

$$Pr = \frac{C_p \mu}{k_f} = \frac{2.84 \times 10^3 \times 0.34 \times 10^{-3}}{0.19} = 5.1$$

Choose 25 per cent baffle cut, from Figure 12.29

$$j_h = 3.3 \times 10^{-3}$$

Without the viscosity correction term

$$h_s = \frac{0.19}{14.4 \times 10^{-3}} \times 3.3 \times 10^{-3} \times 36{,}762 \times 5.1^{1/3} = 2740 \text{ W/m}^2\,^\circ\text{C}$$

Estimate wall temperature

Mean temperature difference = 68 − 33 = 35°C
across all resistances

$$\text{across methanol film} = \frac{U}{h_o} \times \Delta T = \frac{600}{2740} \times 35 = 8^\circ\text{C}$$

Mean wall temperature = 68 − 8 = 60°C

$$\mu_w = 0.37 \text{ mNs/m}^2$$

$$\left(\frac{\mu}{\mu_w}\right)^{0.14} = 0.99$$

which shows that the correction for a low-viscosity fluid is not significant.

Overall coefficient

Thermal conductivity of cupro-nickel alloys = 50 W/m°C.

Take the fouling coefficients from Table 12.2; methanol (light organic) 5000 Wm^{-2}°C^{-1}, brackish water (sea water), take as highest value, 3000 Wm^{-2}°C^{-1}

$$\frac{1}{U_o} = \frac{1}{2740} + \frac{1}{5000} + \frac{20 \times 10^{-3} \ln\left(\frac{20}{16}\right)}{2 \times 50} + \frac{20}{16} \times \frac{1}{3000} + \frac{20}{16} \times \frac{1}{3812} \quad (12.2)$$

$$U_o = \underline{\underline{738 \text{ W/m}^2\,^\circ\text{C}}}$$

well above assumed value of 600 W/m^2°C.

Pressure drop

Tube-side

From Figure 12.24, for $Re = 14{,}925$

$$j_f = 4.3 \times 10^{-3}$$

Neglecting the viscosity correction term

$$\Delta P_t = 2\left(8 \times 4.3 \times 10^{-3}\left(\frac{4.83 \times 10^3}{16}\right) + 2.5\right)\frac{995 \times 0.75^2}{2} \qquad (12.20)$$

$$= 7211 \text{ N/m}^2 = 7.2 \text{ kPa (1.1 psi)}$$

low, could consider increasing the number of tube passes.

Shell side

$$\text{Linear velocity} = \frac{G_s}{\rho} = \frac{868}{750} = 1.16 \text{ m/s}$$

From Figure 12.30, at Re = 36,762

$$j_f = 4 \times 10^{-2}$$

Neglect viscosity correction

$$\Delta P_s = 8 \times 4 \times 10^{-2}\left(\frac{894}{14.4}\right)\left(\frac{4.83 \times 10^3}{178}\right)\frac{750 \times 1.16^2}{2} \qquad (12.26)$$

$$= 272{,}019 \text{ N/m}^2$$

$$= 272 \text{ kPa (39 psi) too high,}$$

could be reduced by increasing the baffle pitch. Doubling the pitch halves the shell-side velocity, which reduces the pressure drop by a factor of approximately $(1/2)^2$

$$\Delta P_s = \frac{272}{4} = 68 \text{ kPa (10 psi), acceptable}$$

This will reduce the shell-side heat-transfer coefficient by a factor of $(1/2)^{0.8}$ ($h_o \propto Re^{0.8} \propto u_s^{0.8}$)

$$h_o = 2740 \times (\tfrac{1}{2})^{0.8} = 1573 \text{ W/m}^2\,{}^\circ\text{C}$$

This gives an overall coefficient of 615 W/m^2°C – still above assumed value of 600 W/m^2°C.

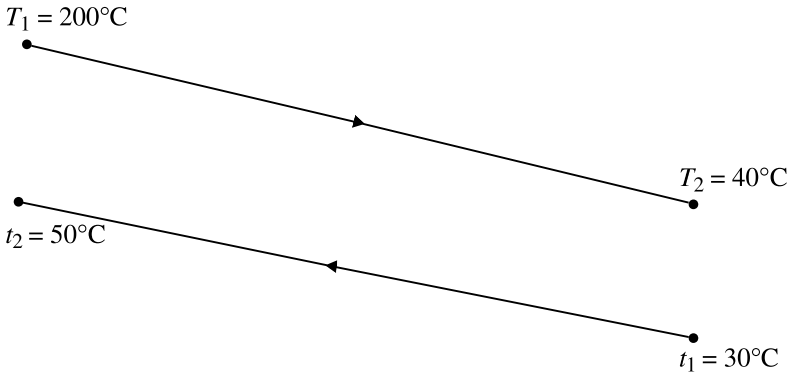
Example 12.2

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 exchanger for this duty.

Solution

Only the thermal design will be carried out, to illustrate the calculation procedure for an exchanger with a divided shell.



$$\Delta T_{lm} = \frac{(200 - 40) - (40 - 30)}{\text{Ln}\,\dfrac{(200 - 50)}{(40 - 30)}} = 51.7^{\circ}\text{C} \qquad (12.4)$$

$$R = (200 - 50)/(50 - 30) = 8.0 \qquad (12.6)$$

$$S = (50 - 30)/(200 - 30) = 0.12 \qquad (12.7)$$

These values do not intercept on the figure for a single shell-pass exchanger, Figure 12.19, so use the figure for a two-pass shell, Figure 12.20, which gives

$$F_t = 0.94, \text{ so}$$

$$\Delta T_m = 0.94 \times 51.7 = 48.6^{\circ}\text{C}$$

Physical properties

Water, from steam tables:

Temperature, °C	30	40	50
C_p, kJ kg^{-1}°C^{-1}	4.18	4.18	4.18
k, kWm^{-1}°C^{-1}	618×10^{-6}	631×10^{-6}	643×10^{-6}
μ, mNm^{-2}s	797×10^{-3}	671×10^{-3}	544×10^{-3}
ρ, kg m^{-3}	995.2	992.8	990.1

Gas oil, from Kern, Process Heat Transfer, McGraw-Hill:

Temperature, °C	200	120	40
C_p, kJ kg^{-1}°C^{-1}	2.59	2.28	1.97
k, Wm^{-1}°C^{-1}	0.13	0.125	0.12
μ, mNm^{-2}s	0.06	0.17	0.28
ρ, kg m^{-3}	830	850	870

Duty:

$$\text{Oil flow-rate} = 22{,}500/3600 = 6.25 \text{ kg/s}$$

$$Q = 6.25 \times 2.28 \times (200 - 40) = 2280 \text{ kW}$$

$$\text{Water flow-rate} = \frac{2280}{4.18(50 - 30)} = 27.27 \text{ kg/h}$$

From Figure 12.1, for cooling tower water and heavy organic liquid, take

$$U = 500 \text{ Wm}^{-2}\text{C}^{-1}$$

$$\text{Area required} = \frac{2280 \times 10^3}{500 \times 48.6} = 94 \text{ m}^2$$

Tube-side coefficient

Select 20 mm o.d., 16 mm i.d. tubes, 4 m long, triangular pitch $1.25d_o$, carbon steel.
Surface area of one tube $= \Pi \times 20 \times 10^{-3} \times 4 = 0.251$ m^2
Number of tubes required $= 94/0.251 = 375$, say 376, even number
Cross-sectional area, one tube $= \dfrac{\Pi}{4}(16 \times 10^{-3})^2 = 2.011 \times 10^{-4}$ m^2
Total tube area $= 376 \times 2.011 \times 10^{-4} = 0.0756$ m^2
Put water through tube for ease of cleaning.
Tube velocity, one pass $= 27.27/(992.8 \times 0.0756) = 0.363$ m/s
Too low to make effective use of the allowable pressure drop, try 4 passes.

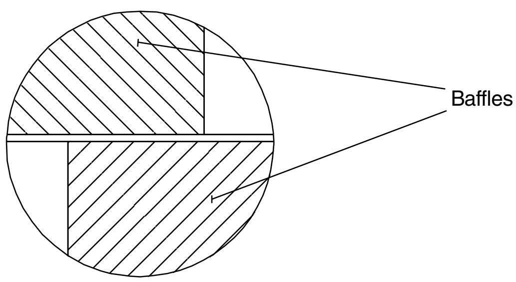
$$u_t = 4 \times 0.363 = 1.45 \text{ m/s}$$

A floating head will be needed due to the temperature difference. Use a pull through type.
Tube-side heat transfer coefficient

$$h_i = \frac{4200(1.35 + 0.02 \times 40)1.45^{0.8}}{16^{0.2}} = 6982 \text{ Wm}^{-2\circ}\text{C}^{-1} \qquad (12.17)$$

Shell-side coefficient

From Table 12.4 and equation 12.3*b*, for 4 passes, $1.25d_o$ triangular pitch
Bundle diameter, $D_b = 20(376/0.175)^{1/2.285} = 575$ mm
From Figure 12.10, for pull through head, clearance = 92 mm
Shell diameter, $D_s = 575 + 92 = 667$ mm (26 in pipe)
Use 25 per cent cut baffles, baffle arrangement for divided shell as shown below:



Take baffle spacing as 1/5 shell diameter = 667/5 = 133 mm
Tube pitch, $p_t = 1.25 \times 20 = 25$ mm
Area for flow, A_s, will be half that given by equation 12.21

$$A_s = 0.5 \times \left(\frac{25 - 20}{25} \times 0.667 \times 0.133\right) = 0.00887 \text{ m}^2$$

$$G_s = 6.25/0.00887 = 704.6 \text{ kg/s}$$

$$u_s = 704.6/850 = 0.83 \text{ m/s, looks reasonable}$$

$$d_e = \frac{1.10}{20}(25^2 - 0.917 \times 20^2) = 14.2 \text{ mm} \qquad (12.23)$$

$$\text{Re} = \frac{0.83 \times 14.2 \times 10^{-3} \times 850}{0.17 \times 10^{-3}} = 58{,}930$$

From Figure 12.29, $j_h = 2.6 \times 10^{-3}$

$$\text{Pr} = (2.28 \times 10^3 \times 0.17 \times 10^{-3})/0.125 = 3.1$$

$$\text{Nu} = 2.6 \times 10^{-3} \times 58{,}930 \times 3.1^{1/3} = 223.4 \qquad (12.25)$$

$$h_s = (223.4 \times 0.125)/(14.2 \times 10^{-3}) = 1967 \text{ Wm}^{-2}{}^\circ\text{C}^{-1}$$

Overall coefficient

Take fouling factors as 0.00025 for cooling tower water and 0.0002 for gas oil (light organic). Thermal conductivity for carbon steel tubes 45 $\text{Wm}^{-1}{}^\circ\text{C}^{-1}$.

$$1/U_o = 1/1967 + 0.0002 + \frac{20 \times 10^{-3} \ln(20/16)}{2 \times 45}$$
$$+ 20/16(1/6982 + 0.00025) = 0.00125$$

$$U_o = 1/0.00125 = 800 \text{ Wm}^{-2}{}^\circ\text{C}^{-1} \qquad (12.2)$$

Well above the initial estimate of 500 $\text{Wm}^{-2}{}^\circ\text{C}^{-1}$, so design has adequate area for the duty required.

Pressure drops

Tube-side

$$\text{Re} = \frac{1.45 \times 16 \times 10^{-3} \times 992.8}{670 \times 10^{-6}} = 34{,}378 \qquad (3.4 \times 10^{-4})$$

From Figure 12.24, $j_f = 3.5 \times 10^{-3}$. Neglecting the viscosity correction

$$\Delta P_t = 4\left[8 \times 3.5 \times 10^{-3} \times \left(\frac{4}{16 \times 10^{-3}}\right) + 2.5\right] 992.8 \times \frac{1.45^2}{2} = 39{,}660$$
$$= 40 \text{ kN/m}^2 \qquad (12.20)$$

Well within the specification, so no need to check the nozzle pressure drop.

Shell-side

From Figure 12.30, for Re = 58,930, $j_s = 3.8 \times 10^{-2}$
With a divided shell, the path length = $2 \times (L/l_b)$
Neglecting the viscosity correction factor,

$$\Delta P_s = 8 \times 3.8 \times 10^{-2}\left(\frac{662 \times 10^{-3}}{14.2 \times 10^{-3}}\right) \times \left(\frac{2 \times 4}{132 \times 10^{-3}}\right) \times 850 \times \frac{0.83^2}{2} = 251{,}481$$

$$= 252 \text{ kN/m}^2 \qquad (12.26)$$

Well within the specification, no need to check nozzle pressure drops.

So the proposed thermal design is satisfactory. As the calculated pressure drops are below that allowed, there is some scope for improving the design.

Example 12.3

Design a shell-and-tube exchanger for the following duty.

20,000 kg/h of kerosene (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 $(\text{W/m}^2\,{}^\circ\text{C})^{-1}$ on the crude stream and 0.0002 $(\text{W/m}^2\,{}^\circ\text{C})^{-1}$ on the kerosene stream.

Solution

The solution to this example illustrates the iterative nature of heat exchanger design calculations. An algorithm for the design of shell-and-tube exchangers is shown in Figure A (see p. 684). The procedure set out in this figure will be followed in the solution.

Step 1: Specification

The specification is given in the problem statement.

20,000 kg/h of kerosene (42° API) at 200°C cooled to 90°C, by exchange with 70,000 kg/h light crude oil (34° API) at 40°C.

The kerosene pressure 5 bar, the crude oil pressure 6.5 bar.

Permissible pressure drop of 0.8 bar on both streams.

Fouling factors: crude stream 0.00035 $(\text{W/m}^2\,{}^\circ\text{C})^{-1}$, kerosene stream 0.0002 $(\text{W/m}^2\,{}^\circ\text{C})^{-1}$.

To complete the specification, the duty (heat transfer rate) and the outlet temperature of the crude oil needed to be calculated.

The mean temperature of the kerosene = (200 + 90)/2 = 145°C.

At this temperature the specific heat capacity of 42° API kerosene is 2.47 kJ/kg°C (physical properties from D. Q. Kern, Process Heat Transfer, McGraw-Hill).

$$\text{Duty} = \frac{20{,}000}{3600} \times 2.47(200 - 90) = 1509.4 \text{ kW}$$

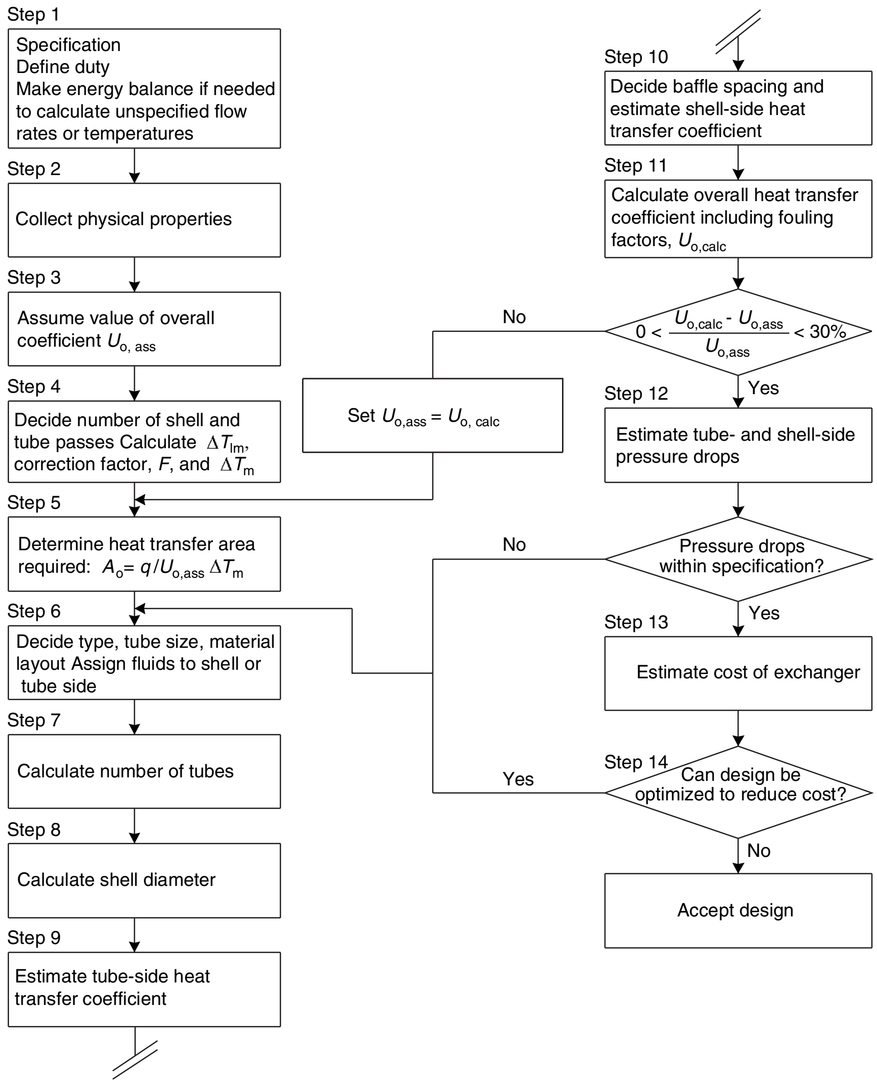


Figure A. Design procedure for shell-and-tube heat exchangers

Example 12.2 and Figure A were developed by the author for the Open University Course T333 *Principles and Applications of Heat Transfer*. They are reproduced here by permission of the Open University.

As a first trial take the mean temperature of the crude oil as equal to the inlet temperature, 40°C; specific heat capacity at this temperature = 2.01 kJ/kg°C.

An energy balance gives:

$$\frac{7000}{3600} \times 2.01(t_2 - 40) = 1509.4$$

$t_2 = 78.6$°C and the stream mean temperature $= (40 + 78.6)/2 = 59.3$°C.

The specific heat at this temperature is 2.05 kJ/kg°C. A second trial calculation using this value gives $t_2 = 77.9°C$ and a new mean temperature of 58.9°C. There is no significant change in the specific heat at this mean temperature from the value used, so take the crude stream outlet temperature to be 77.9°C, say 78°C.

Step 2: Physical Properties

Kerosene	inlet	mean	outlet	
temperature	200	145	90	°C
specific heat	2.72	2.47	2.26	kJ/kg°C
thermal conductivity	0.130	0.132	0.135	W/m°C
density	690	730	770	kg/m^3
viscosity	0.22	0.43	0.80	$mN\ sm^{-2}$
Crude oil	outlet	mean	inlet	
temperature	78	59	40	°C
specific heat	2.09	2.05	2.01	kJ/kg°C
thermal conductivity	0.133	0.134	0.135	W/m°C
density	800	820	840	kg/m^3
viscosity	2.4	3.2	4.3	$mN\ sm^{-2}$

Step 3: Overall coefficient

For an exchanger of this type the overall coefficient will be in the range 300 to 500 $W/m^2{}°C$, see Figure 12.1 and Table 12.1; so start with 300 $W/m^2{}°C$.

Step 4: Exchanger type and dimensions

An even number of tube passes is usually the preferred arrangement, as this positions the inlet and outlet nozzles at the same end of the exchanger, which simplifies the pipework.

Start with one shell pass and 2 tube passes.

$$\Delta T_{lm} = \frac{(200 - 78) - (90 - 40)}{\ln \dfrac{(200 - 78)}{(90 - 40)}} = 80.7°C \tag{12.4}$$

$$R = \frac{(200 - 90)}{(78 - 40)} = 2.9 \tag{12.6}$$

$$S = \frac{(78 - 40)}{(200 - 40)} = 0.24 \tag{12.7}$$

From Figure 12.19, $F_t = 0.88$, which is acceptable.

So, $$\Delta T_m = 0.88 \times 80.7 = 71.0°C$$

Step 5: Heat transfer area

$$A_o = \frac{1509.4 \times 10^3}{300 \times 71.0} = 70.86 \text{ m}^2 \tag{12.1}$$

Step 6: Layout and tube size

Using a split-ring floating head exchanger for efficiency and ease of cleaning.

Neither fluid is corrosive, and the operating pressure is not high, so a plain carbon steel can be used for the shell and tubes.

The crude is dirtier than the kerosene, so put the crude through the tubes and the kerosene in the shell.

Use 19.05 mm (3/4 inch) outside diameter, 14.83 mm inside diameter, 5 m Long tubes (a popular size) on a triangular 23.81 mm pitch (pitch/dia. = 1.25).

Step 7: Number of tubes

Area of one tube (neglecting thickness of tube sheets)

$$= \pi \times 19.05 \times 10^{-3} \times 5 = 0.2992 \text{ m}^2$$

Number of tubes = 70.89/0.2992 = 237, say 240
So, for 2 passes, tubes per pass = 120

Check the tube-side velocity at this stage to see if it looks reasonable.

$$\text{Tube cross-sectional area} = \frac{\pi}{4}(14.83 \times 10^{-3})^2 = 0.0001727 \text{ m}^2$$

$$\text{So area per pass} = 120 \times 0.0001727 = 0.02073 \text{ m}^2$$

$$\text{Volumetric flow} = \frac{70{,}000}{3600} \times \frac{1}{820} = 0.0237 \text{ m}^3/\text{s}$$

$$\text{Tube-side velocity, } u_t = \frac{0.0237}{0.02073} = 1.14 \text{ m/s}$$

The velocity is satisfactory, between 1 to 2 m/s, but may be a little low. This will show up when the pressure drop is calculated.

Step 8: Bundle and shell diameter

From Table 12.4, for 2 tube passes, $K_1 = 0.249$, $n_1 = 2.207$,

so,
$$D_b = 19.05\left(\frac{240}{0.249}\right)^{1/2.207} = 428 \text{ mm } (0.43 \text{ m}) \tag{12.3b}$$

For a split-ring floating head exchanger the typical shell clearance from Figure 12.10 is 56 mm, so the shell inside diameter,

$$D_s = 428 + 56 = 484 \text{ mm}$$

Step 9: Tube-side heat transfer coefficient

$$Re = \frac{820 \times 1.14 \times 14.83 \times 10^{-3}}{3.2 \times 10^{-3}} = 4332,\ (4.3 \times 10^3)$$

$$Pr = \frac{2.05 \times 10^3 \times 3.2 \times 10^{-3}}{0.134} = 48.96$$

$$\frac{L}{d_i} = \frac{5000}{14.83} = 337$$

From Figure 12.23, $j_h = 3.2 \times 10^{-3}$

$$Nu = 3.2 \times 10^{-3}(4332)(48.96)^{0.33} = 50.06 \qquad (12.15)$$

$$h_i = 50.06 \times \left(\frac{0.134}{14.83 \times 10^{-3}}\right) = 452\ \text{W/m}^2\,^\circ\text{C}$$

This is clearly too low if U_o is to be 300 W/m^2 °C. The tube-side velocity did look low, so increase the number of tube passes to 4. This will halve the cross-sectional area in each pass and double the velocity.

New $\quad u_t = 2 \times 1.14 = 2.3$ m/s

and $\quad Re = 2 \times 4332 = 8664(8.7 \times 10^3)$

$$j_h = 3.8 \times 10^{-3}$$

$$h_i = \left(\frac{0.134}{14.83 \times 10^{-3}}\right) \times 3.8 \times 10^{-3}(8664)(48.96)^{0.33}$$

$$= 1074\ \text{W/m}^2\,^\circ\text{C}$$

Step 10: Shell-side heat transfer coefficient

Kern's method will be used.

With 4 tube passes the shell diameter will be larger than that calculated for 2 passes. For 4 passes $K_1 = 0.175$ and $n_1 = 2.285$.

$$D_b = 19.05\left(\frac{240}{0.175}\right)^{1/2.285} = 450\ \text{mm},\ (0.45\ \text{m}) \qquad (12.3b)$$

The bundle to shell clearance is still around 56 mm, giving:

$$D_s = 506\ \text{mm (about 20 inches)}$$

As a first trial take the baffle spacing = $D_s/5$, say 100 mm. This spacing should give good heat transfer without too high a pressure drop.

$$A_s = \frac{(23.81 - 19.05)}{23.81} 506 \times 100 = 10{,}116\ \text{mm}^2 = 0.01012\ \text{m}^2 \qquad (12.21)$$

$$d_e = \frac{1.10}{19.05}(23.81^2 - 0.917 \times 19.05^2) = 13.52\ \text{mm} \qquad (12.23)$$

Volumetric flow-rate on shell-side $= \dfrac{20{,}000}{3600} \times \dfrac{1}{730} = 0.0076\ \text{m}^3/\text{s}$

$$\text{Shell-side velocity} = \frac{0.076}{0.01012} = 0.75 \text{ m/s}$$

$$Re = \frac{730 \times 0.75 \times 13.52 \times 10^{-3}}{0.43 \times 10^{-3}} = 17{,}214,\ (1.72 \times 10^4)$$

$$Pr = \frac{2.47 \times 10^3 \times 0.43 \times 10^{-3}}{0.132} = 8.05$$

Use segmental baffles with a 25% cut. This should give a reasonable heat transfer coefficient without too large a pressure drop.
From Figure 12.29, $j_h = 4.52 \times 10^{-3}$.
Neglecting the viscosity correction:

$$h_s = \left(\frac{0.132}{13.52} \times 10^3\right) \times 4.52 \times 10^{-3} \times 17{,}214 \times 8.05^{0.33} = 1505 \text{ W/m}^2\,^\circ\text{C} \qquad (12.25)$$

Step 11: Overall coefficient

$$\frac{1}{U_o} = \left(\frac{1}{1074} + 0.00035\right)\frac{19.05}{14.83} + \frac{19.05 \times 10^{-3}\text{Ln}\left(\frac{19.05}{14.83}\right)}{2 \times 55} + \frac{1}{1505} + 0.0002$$

$$U_o = 386 \text{ W/m}^2\,^\circ\text{C} \qquad (12.2)$$

This is above the initial estimate of 300 W/m^2 °C. The number of tubes could possibly be reduced, but first check the pressure drops.

Step 12: Pressure drop

Tube-side

240 tubes, 4 passes, tube i.d. 14.83 mm, u_t 2.3 m/s,
$Re = 8.7 \times 10^3$. From Figure 12.24, $j_f = 5 \times 10^{-3}$.

$$\Delta P_t = 4\left(8 \times 5 \times 10^{-3}\left(\frac{5000}{14.83}\right) + 2.5\right)\frac{(820 \times 2.3^2)}{2} \qquad (12.20)$$

$$= 4(13.5 + 2.5)\frac{(820 \times 2.3^2)}{2}$$

$$= 138{,}810 \text{ N/m}^2,\ 1.4 \text{ bar}$$

This exceeds the specification. Return to step 6 and modify the design.

Modified design

The tube velocity needs to be reduced. This will reduce the heat transfer coefficient, so the number of tubes must be increased to compensate. There will be a pressure drop across the inlet and outlet nozzles. Allow 0.1 bar for this, a typical figure (about 15% of the total); which leaves 0.7 bar across the tubes. Pressure drop is roughly proportional

to the square of the velocity and u_t is proportional to the number of tubes per pass. So the pressure drop calculated for 240 tubes can be used to estimate the number of tubes required.
Tubes needed $= 240/(0.6/1.4)^{0.5} = 365$
Say, 360 with 4 passes.
Retain 4 passes as the heat transfer coefficient will be too low with 2 passes.
Second trial design: 360 tubes 19.05 mm o.d., 14.83 mm i.d., 5 m long, triangular pitch 23.81 mm.

$$D_b = 19.05\left(\frac{360}{0.175}\right)^{1/2.285} = 537 \text{ mm, } (0.54 \text{ m}) \qquad (12.3b)$$

From Figure 12.10 clearance with this bundle diameter = 59 mm

$$D_s = 537 + 59 = 596 \text{ mm}$$

$$\text{Cross-sectional area per pass} = \frac{360}{4}(14.83 \times 10^{-3})^2\frac{\pi}{4} = 0.01555 \text{ m}^2$$

$$\text{Tube velocity } u_t = \frac{0.0237}{0.01555} = 1.524 \text{ m/s}$$

$$Re = \frac{820 \times 1.524 \times 14.83 \times 10^{-3}}{3.2 \times 10^{-3}} = 5792$$

L/d is the same as the first trial, 337

$$j_h = 3.6 \times 10^{-3}$$

$$h_i = \left(\frac{0.134}{14.83} \times 10^{-3}\right) 3.6 \times 10^{-3} \times 5792 \times 48.96^{0.33} = 680 \text{ W/m}^2\,{}^\circ\text{C} \qquad (12.15)$$

This looks satisfactory, but check the pressure drop before doing the shell-side calculation.

$$j_f = 5.5 \times 10^{-3}$$

$$\Delta P_t = 4\left(8 \times 5.5 \times 10^{-3}\left(\frac{5000}{14.83}\right) + 2.5\right)\frac{(820 \times 1.524^2)}{2} = 66{,}029 \text{ N/m}^2\text{, } 0.66 \text{ bar} \qquad (12.20)$$

Well within specification.
Keep the same baffle cut and spacing.

$$A_s = \frac{(23.81 - 19.05)}{23.81}596 \times 100 = 11{,}915 \text{ mm}^2\text{, } 0.01192 \text{ m}^2 \qquad (12.21)$$

$$u_s = \frac{0.0076}{0.01193} = 0.638 \text{ m/s}$$

$$d_e = 13.52 \text{ mm, as before}$$

$$Re = \frac{730 \times 0.638 \times 13.52 \times 10^{-3}}{0.43 \times 10^{-3}} = 14{,}644\text{, } (1.5 \times 10^4)$$

$Pr = 8.05$

$j_h = 4.8 \times 10^{-3}, \quad j_f = 4.6 \times 10^{-2}$

$$h_s = \left(\frac{0.132}{13.52 \times 10^{-3}}\right) 4.8 \times 10^{-3} \times 14{,}644 \times (8.05)^{0.33} = 1366 \text{ W/m}^2\,^\circ\text{C}, \text{ looks OK} \quad (12.25)$$

$$\Delta P_s = 8 \times 4.6 \times 10^{-2} \left(\frac{596}{13.52}\right)\left(\frac{5000}{100}\right)\frac{(730 \times 0.638^2)}{2} = 120{,}510 \text{ N/m}^2, \text{ 1.2 bar} \quad (12.26)$$

Too high; the specification only allowed 0.8 overall, including the loss over the nozzles. Check the overall coefficient to see if there is room to modify the shell-side design.

$$\frac{1}{U_o} = \left(\frac{1}{683} + 0.00035\right)\frac{19.05}{14.83} + \frac{19.05 \times 10^{-3} \ln\left(\frac{19.05}{14.88}\right)}{2 \times 55} + \frac{1}{1366} + 0.0002 \quad (12.2)$$

$$U_o = 302 \text{ W/m}^2\,^\circ\text{C}$$

$$U_o \text{ required} = \frac{Q}{(A_o \Delta T_{\text{lm}})}, \quad A_o = 360 \times 0.2992 = 107.7 \text{ m}^2,$$

$$\text{so } U_o \text{ required} = \frac{1509.4 \times 10^3}{(107.7 \times 71)} = 197 \text{ W/m}^2\,^\circ\text{C}$$

The estimated overall coefficient is well above that required for design, 302 compared to 192 $\text{W/m}^2\,^\circ\text{C}$, which gives scope for reducing the shell-side pressure drop.

Allow a drop of 0.1 bar for the shell inlet and outlet nozzles, leaving 0.7 bar for the shell-side flow. So, to keep within the specification, the shell-side velocity will have to be reduced by around $\sqrt{(1/2)} = 0.707$. To achieve this the baffle spacing will need to be increased to $100/0.707 = 141$, say 140 mm.

$$A_s = \frac{(23.81 - 19.05)}{23.81} 596 \times 140 = 6681 \text{ mm}^2, 0.167 \text{ m}^2 \quad (12.21)$$

$$u_s = \frac{0.0076}{0.0167} = 0.455 \text{ m/s},$$

Giving: $Re = 10{,}443$, $h_s = 1177$ $\text{W/m}^2\,^\circ\text{C}$, $\Delta P_s = 0.47$ bar, and $U_o = 288$ $\text{Wm}^{-2}\,^\circ\text{C}^{-1}$.

The pressure drop is now well within the specification.

Step 13: Estimate cost

The cost of this design can be estimated using the methods given in Chapter 6.

Step 14: Optimisation

There is scope for optimising the design by reducing the number of tubes, as the pressure drops are well within specification and the overall coefficient is well above that needed. However, the method used for estimating the coefficient and pressure drop on the shell-side (Kern's method) is not accurate, so keeping to this design will give some margin of safety.

Viscosity correction factor

The viscosity correction factor $(\mu/\mu_w)^{0.14}$ was neglected when calculating the heat transfer coefficients and pressure drops. This is reasonable for the kerosene as it has a relatively low viscosity, but it is not so obviously so for the crude oil. So, before firming up the design, the effect of this factor on the tube-side coefficient and pressure drop will be checked.

First, an estimate of the temperature at the tube wall, t_w is needed.

$$\text{The inside area of the tubes} = \pi \times 14.83 \times 10^{-3} \times 5 \times 360 = 83.86 \text{ m}^2$$

$$\text{Heat flux} = Q/A = 1509.4 \times 10^3/83.86 = 17{,}999 \text{ W/m}^2$$

As a rough approximation

$$(t_w - t)h_i = 17{,}999$$

where t is the mean bulk fluid temperature = 59°C.

So,
$$t_w = \frac{17{,}999}{680} + 59 = 86^\circ\text{C}.$$

The crude oil viscosity at this temperature = 2.1×10^{-3} Ns/m^2.

$$\text{Giving} \left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{3.2 \times 10^{-3}}{2.1 \times 10^{-3}}\right)^{0.14} = 1.06$$

Only a small factor, so the decision to neglect it was justified. Applying the correction would increase the estimated heat transfer coefficient, which is in the right direction. It would give a slight decrease in the estimated pressure drop.

Summary: the proposed design

Split ring, floating head, 1 shell pass, 4 tube passes.

360 carbon steel tubes, 5 m long, 19.05 mm o.d., 14.83 mm i.d., triangular pitch, pitch 23.18 mm.

Heat transfer area 107.7 m^2 (based on outside diameter).

Shell i.d. 597 mm (600 mm), baffle spacing 140 mm, 25% cut.

Tube-side coefficient 680 W/m^2°C, clean.

Shell-side coefficient 1366 W/m^2°C, clean.

Overall coefficient, estimated 288 W/m^2°C, dirty.

Overall coefficient required 197 W/m^2°C, dirty.

Dirt/Fouling factors:

Tube-side (crude oil) 0.00035 (W/m^2°C)$^{-1}$.

Shell-side (kerosene) 0.0002 (W/m^2°C)$^{-1}$.

Pressure drops:

Tube-side, estimated 0.40 bar, +0.1 for nozzles; specified 0.8 bar overall.

Shell-side, estimated 0.45 bar, +0.1 for nozzles; specified 0.8 bar overall.