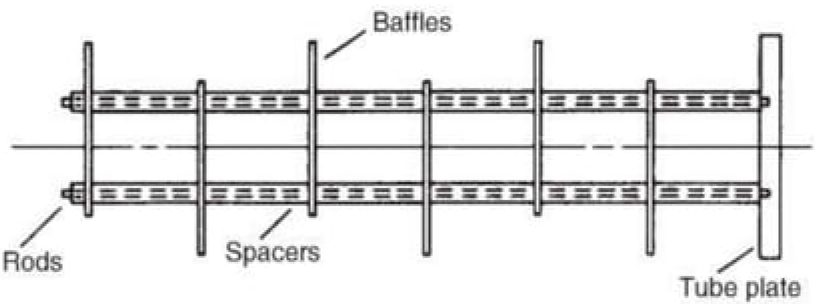
Shell and Tube Heat Exchanger (STHE)

Introduction

- Shell and tube heat exchangers can be constructed with a very large heat transfer surface in a relatively small volume, fabricated from alloy steels to resist corrosion and
- used for heating, cooling and for condensing a very wide range of fluids, they are The most widely used form of heat transfer equipment.
- The tubes are connected so that the internal fluid makes several passes up and down the exchanger thus enabling a high velocity of flow to be obtained for a given heat Transfer are a and through put of fluid.

It 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



- The fluid flowing in the shell is made to flow first in one sense and then in the opposite sense across the tube bundle by fitting a series of Baffles along the length.
- These baffles are frequently of the segmental form with about 25 percent cut away to provide the free space to increase the velocity of flow across the tubes, thus giving higher rates of heat transfer.

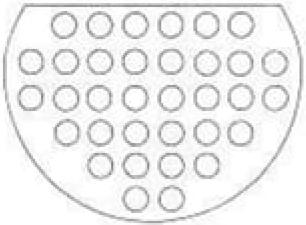


Figure 9.29 Baffle for heat exchanger

- 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.

Fixed Tube Plate STHE

- It is simplest and cheapest type but has 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 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.

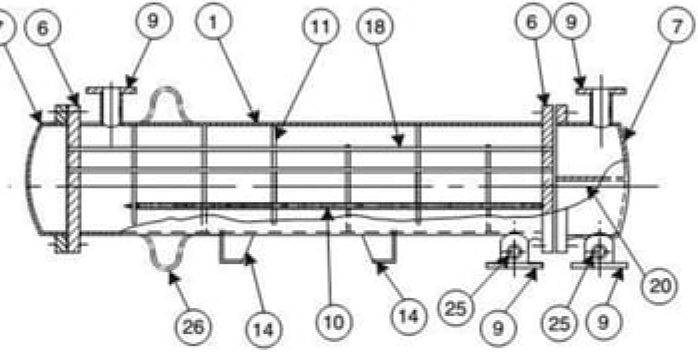


Figure 12.3. Fixed-tube plate (based on figures from BS 3274: 1960)

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

U Tube HX

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.

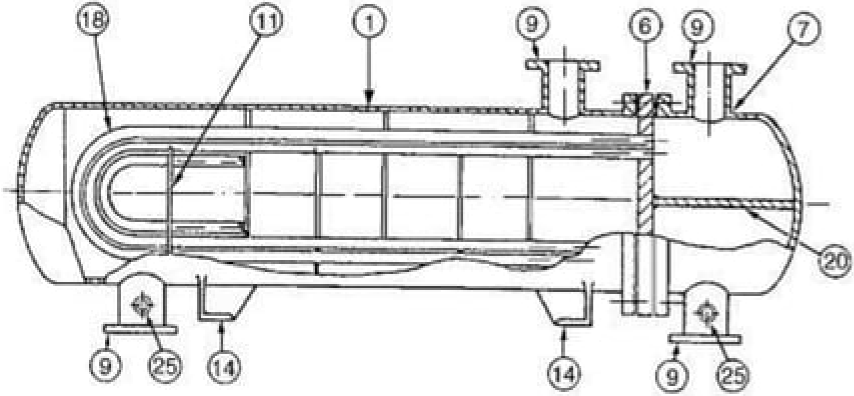


Figure 12.4. U-tube (based on figures from BS 3274: 1960)

Pull Through STHE

- Pull through types 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 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.

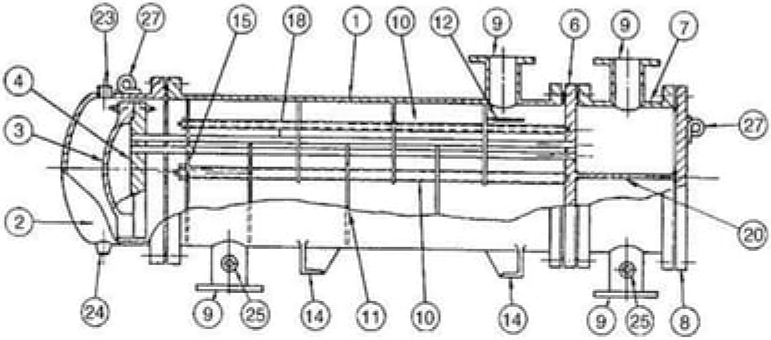


Figure 12.5. Internal floating head without clamp ring (based on figures from BS 3274: 1960)

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.

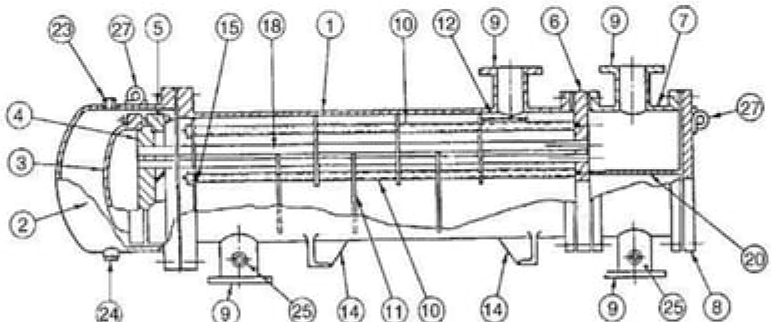


Figure 12.6. Internal floating head with clamp ring (based on figures from BS 3274: 1960)

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.

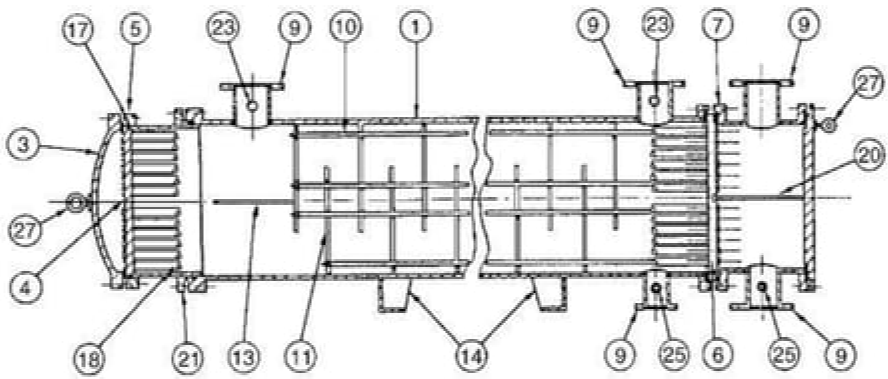


Figure 12.7. External floating head, packed gland (based on figures from BS 3274: 1960)

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 standards give the preferred shell and tube dimensions; the design and manufacturing tolerances; corrosion allowances; and the recommended design stresses for materialsof 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

Essential requirements in the design

- The essential requirements in the design of a heat exchanger are firstly, The provision of a unit which is reliable and has the desired capacity.
- secondly, the need to provide an exchanger at minimum overall cost.
- In general, this involves using standard components and fittings and making the design as simple as possible.
- In most cases, it is necessary to balance the capital cost in terms of the depreciation against the operating cost.

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 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).

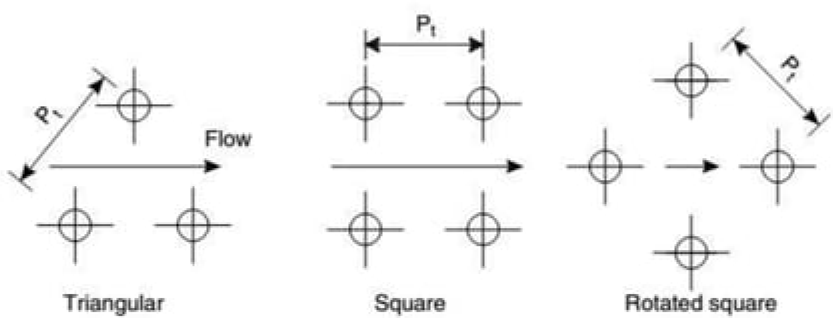
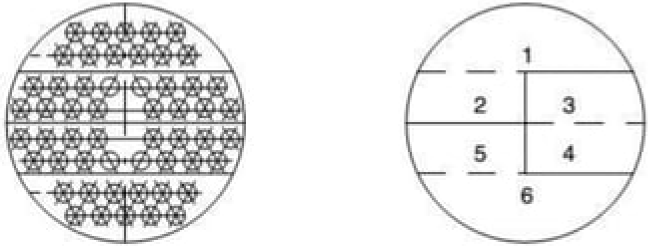


Figure 12.9. Tube patterns

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).



Six tube passes

Four passes

Two passes

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 dia., mm	Carbon steel pipe	Carbon steel plate	Alloy steel
150	7.1	—	3.2
200–300	9.3	—	3.2
330–580	9.5	7.9	3.2
610–740	—	7.9	4.8
760–990	—	9.5	6.4
1010–1520	—	11.1	6.4
1550–2030	—	12.7	7.9
2050–2540	—	12.7	9.5

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.

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.

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

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.

Shell Types

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

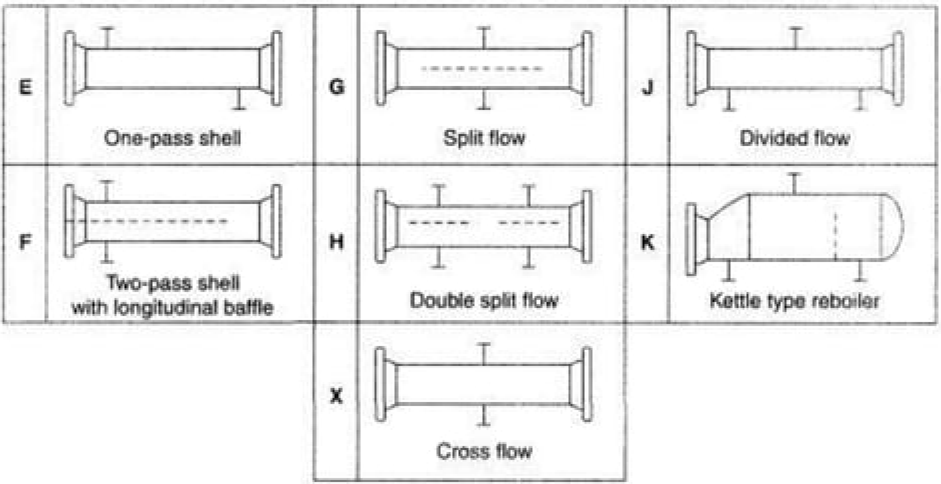


Figure 9.67. TEMA shell types

Shell Types

- **TEMA E-type** on which most design methods are based, although these may be adapted for other shell types by allowing for the resulting velocity changes.
- **TEMA F-type** has a longitudinal baffle giving two shell passes and this provides an alternative arrangement to the use of two shells required in order to cope with a close temperature approach or low shell-side flow rates.
- The pressure drop in two shells is some eight times greater than that encountered in the E-type design although any potential leakage between the longitudinal baffle and the shell in the F-type design may restrict the range of application.

- The so-called "**split-flow" type** of unit with a longitudinal baffle is classified as the TEMA G-type whose performance is superior although the pressure drop is similar to the E-type.
- This design is used mainly for reboilers and only occasionally for systems where there is no change of phase.
- The so-called "**divided-flow" type**, the TEMAJ-type, has one inlet and two outlet nozzles and, with a pressure drop some one-eighth of the E type, finds application in gas coolers and condensers operating at low pressures.
- **The TEMA X-type** shell has no cross baffles and hence the shell-side fluid is in pure counter flow giving extremely low pressure drops and again, this type of design is used for gas cooling and condensation at low pressures.

Shell Material and Geometry

- The shell of a heat exchanger is commonly made of carbon steel and standard pipes are used for the smaller sizes and rolled welded plate for the larger sizes (say0.4-1.0m).
- The thickness of the shell may be treated as similar to thin-walled cylinders and a minimum thickness of 9.5 mm is used for shells over 0.33m o.d. and 11.1 mm for shells over 0.9m o.d.
- A corrosion allowance of 3.2mm is commonly added to all carbon steel parts and thickness is determined more by rigidity requirements than simply internal pressure.
- A shell diameter should be such as to give as close a fit to the tube bundle as practical in order to reduce by passing round the outside of the bundle.

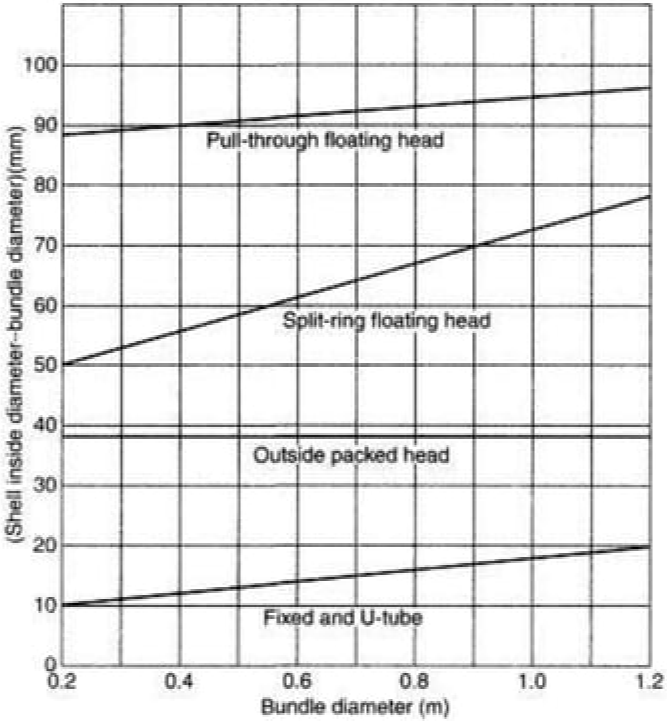
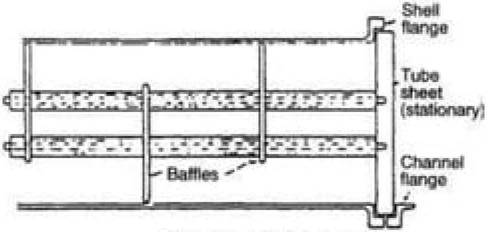


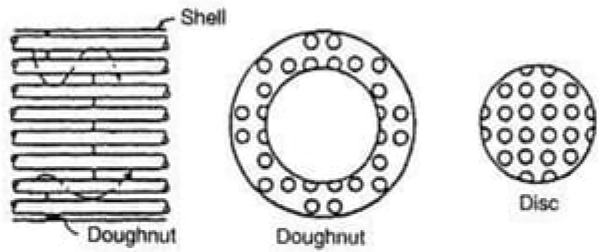
Figure 9.68. Shell-bundle clearance

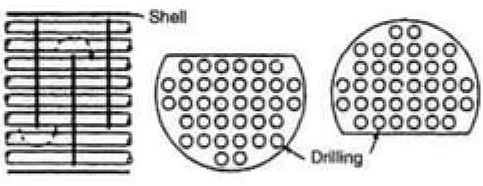
12.5.7. Baffles

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*.

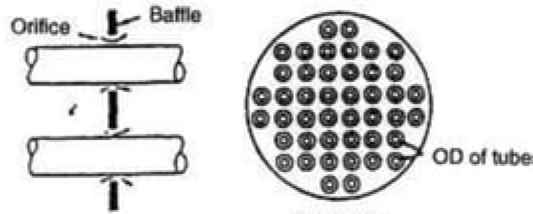


Baffle space detail (enlarged).



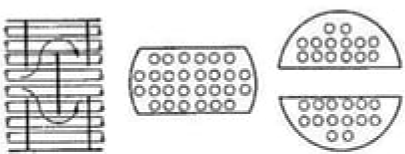


Segmental baffle detail



(a) Detail

(b) Baffle
Orifice baffle



Baffle designs

- The cross-baffle is designed to direct the flow of the shell side fluid across the tube bundle and to support the tubes against sagging and
- possible vibration, and the most common type is the segmental baffle which provides a baffle window.
- The ratio, baffle spacing/ baffle cut, is very important in maximizing the ratio of heat transfer rate to pressure drop.
- Where very low pressure drops are required, double segmental or "disc and doughnut "baffles are used to reduce the pressure drop by some 60 percent.
- Triple segmental baffles and designs in which all the tubes are supported by all the baffles provide for low pressure drops and minimum tube vibration

- TEMA recommends that segmental baffles should not be spaced closer than 20 percent of the shell inside diameter and maximum spacing should be such that the unsupported tube lengths are not exceeded.
- It may be noted that the majority of failures occur due to vibration when the unsupported tube length is in excess of the TEMA maximum limit; the best solution is to avoid having tubes in the baffle window.

Table 9.14. Maximum unsupported spans for tubes

Aproximate tube OD. (mm)	Maximum unsupported span (mm)	
	Materials group A	Materials group B
19	1520	1321
25	1880	1626
32	2240	1930
38	2540	2210
50	3175	2794

Materials
Group A: Carbon and high alloy steel, low alloy steel, nickel-copper, nickel, nickel-chromium-iron.
Group B: Aluminium and aluminium alloys, copper and copper alloys, titanium and zirconium.

SHELL AND TUBE EXCHANGERS: GENERAL DESIGN CONSIDERATIONS

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. 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.

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

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.
- 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.

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 milli-bars 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.

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 × absolute pressure
1 to 2 bar	0.5 × system gauge pressure
Above 10 bar	0.1 × 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.

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).***
- 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.

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

Turbulent flow

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

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

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.

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.

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

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

- The use of the **jh** factor enables data for laminar and turbulent flow to be represented on the same graph

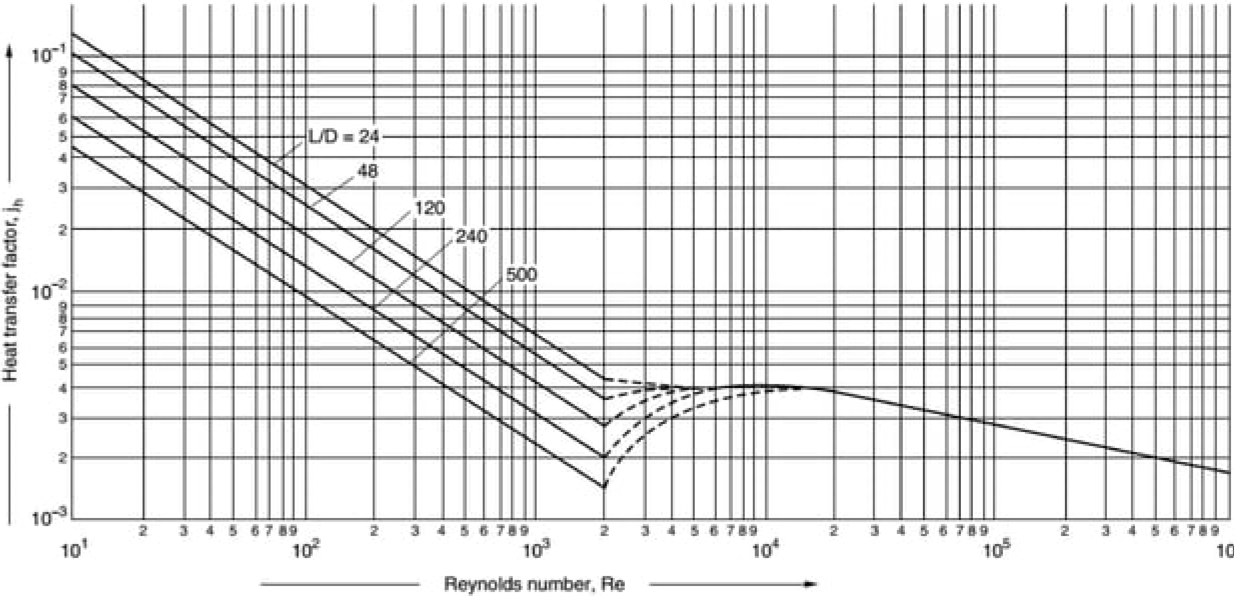


Figure 12.23. Tube-side heat-transfer factor

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

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

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

where h_i = inside coefficient, for water, W/m^2°C,

t = water temperature, °C,

u_t = water velocity, m/s,

d_i = tube inside diameter, mm.

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.

For Isothermal Flow

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

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, $Re < 2100$,

$= 0.14$ for turbulent flow, $Re > 2100$.

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.

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

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

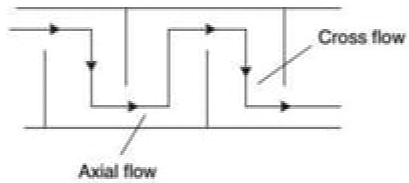


Figure 12.25. Idealised main stream flow

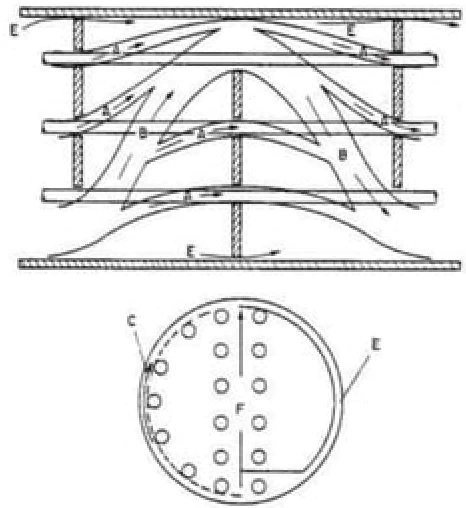


Figure 12.26. Shell-side leakage and by-pass paths

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.

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.

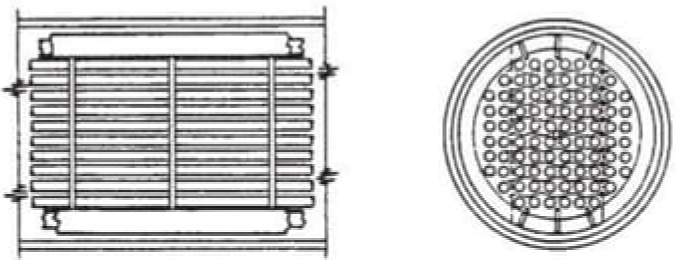


Figure 12.27. Sealing strips

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