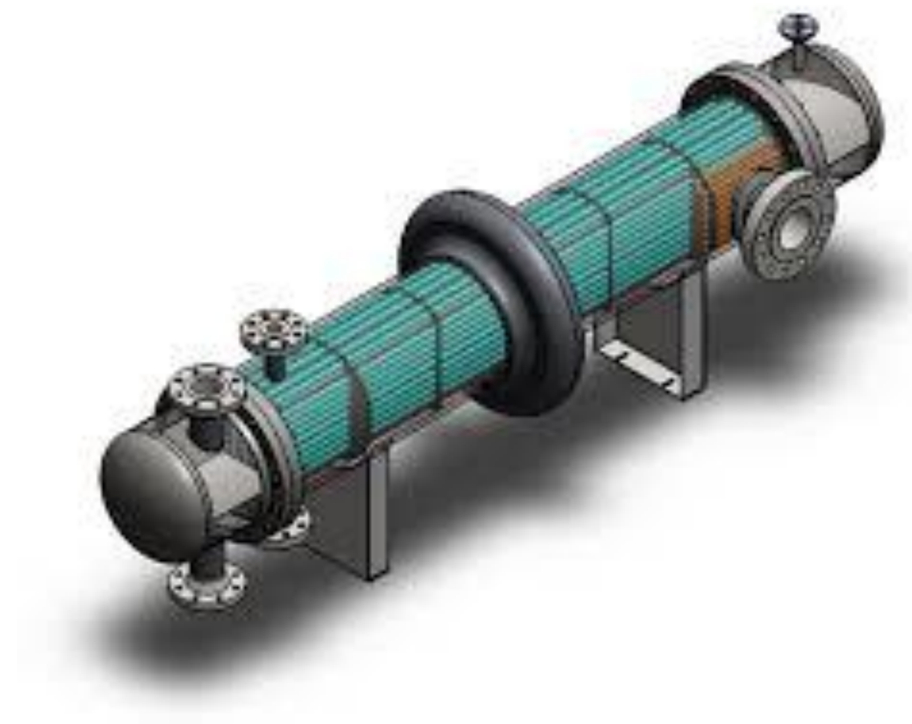
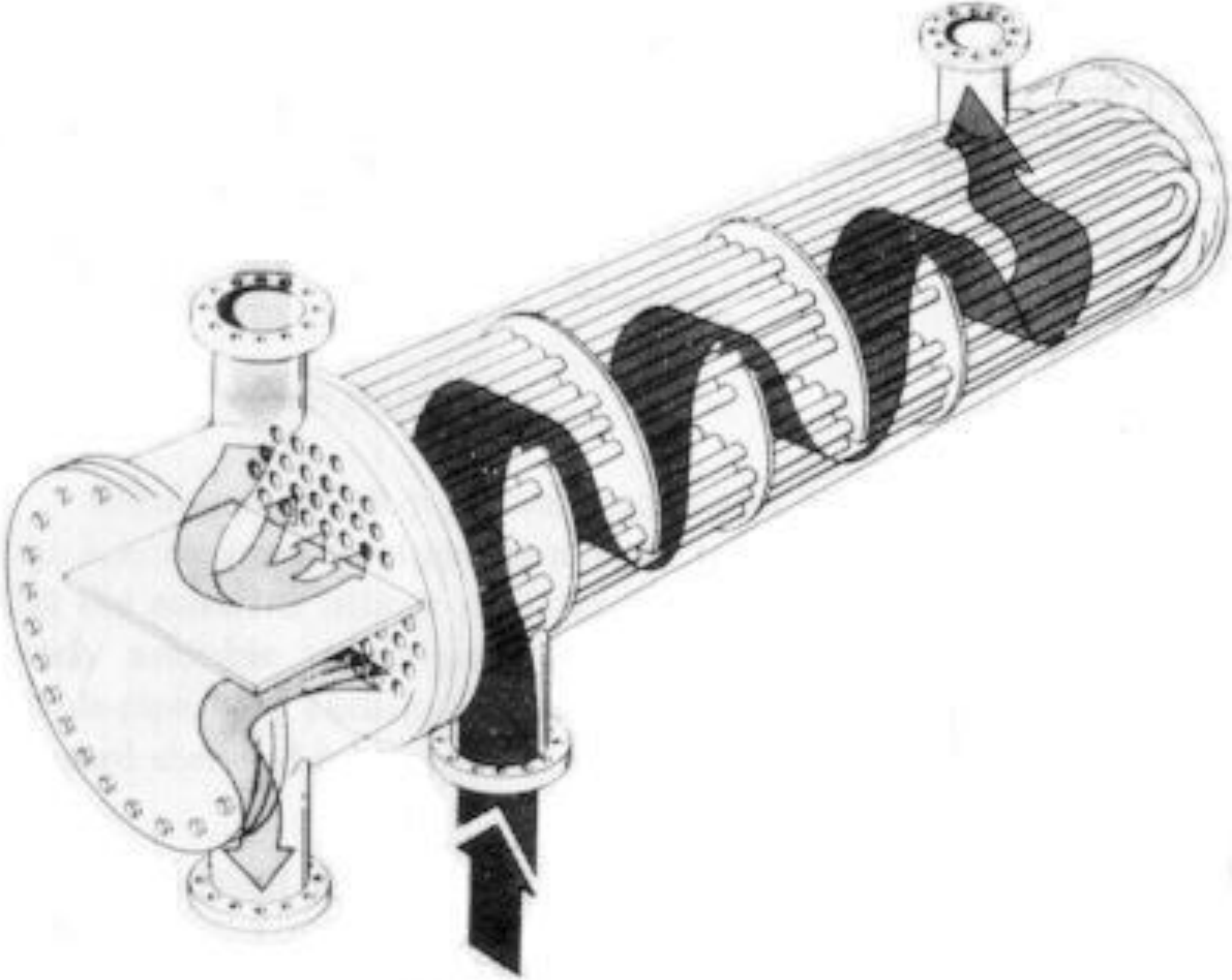
Shell-and-Tube Heat Exchangers



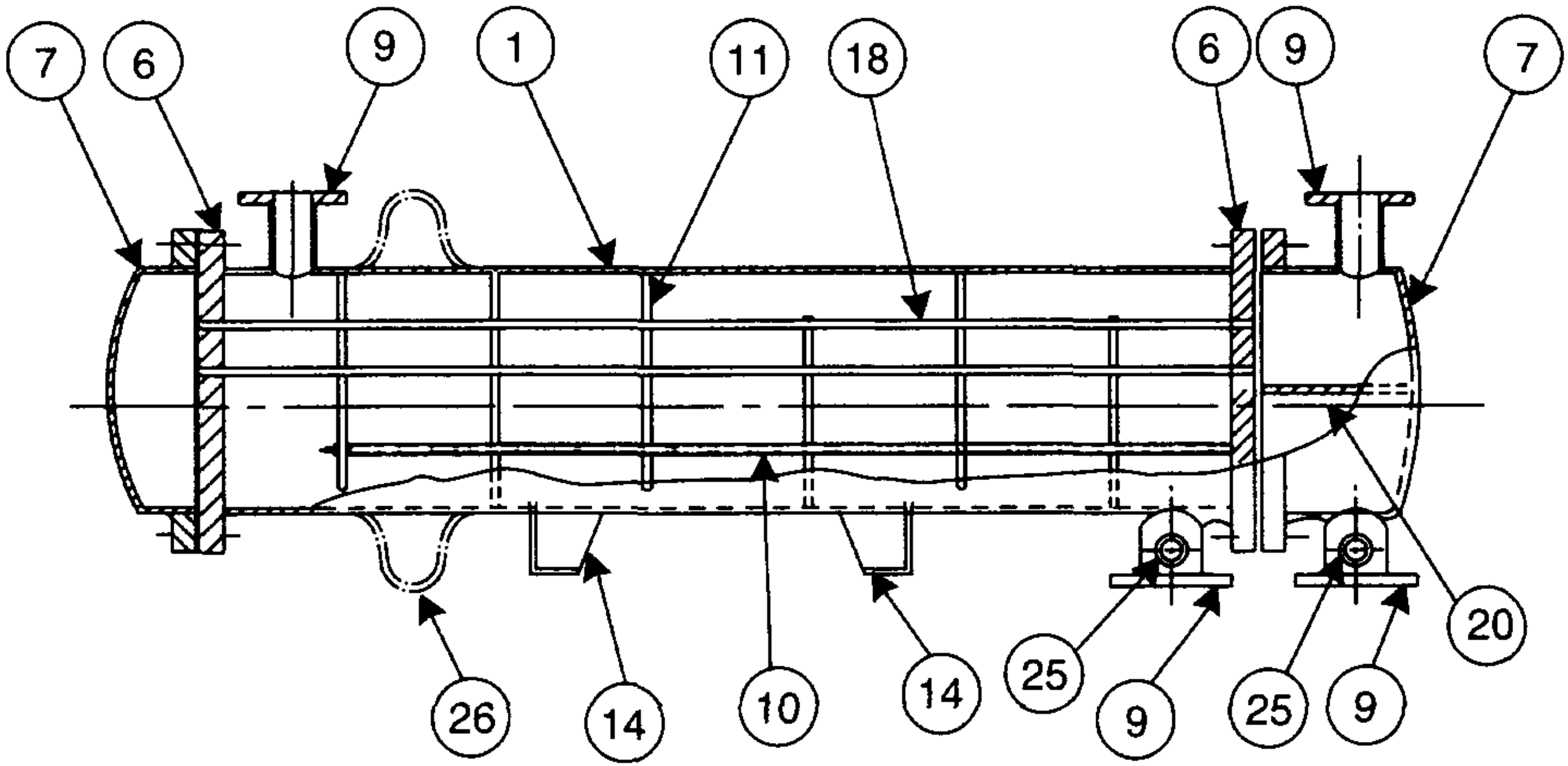
The shell and tube exchanger is by far the most commonly used type of heat-transfer equipment used in the chemical and allied industries. The advantages of this type are:

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.

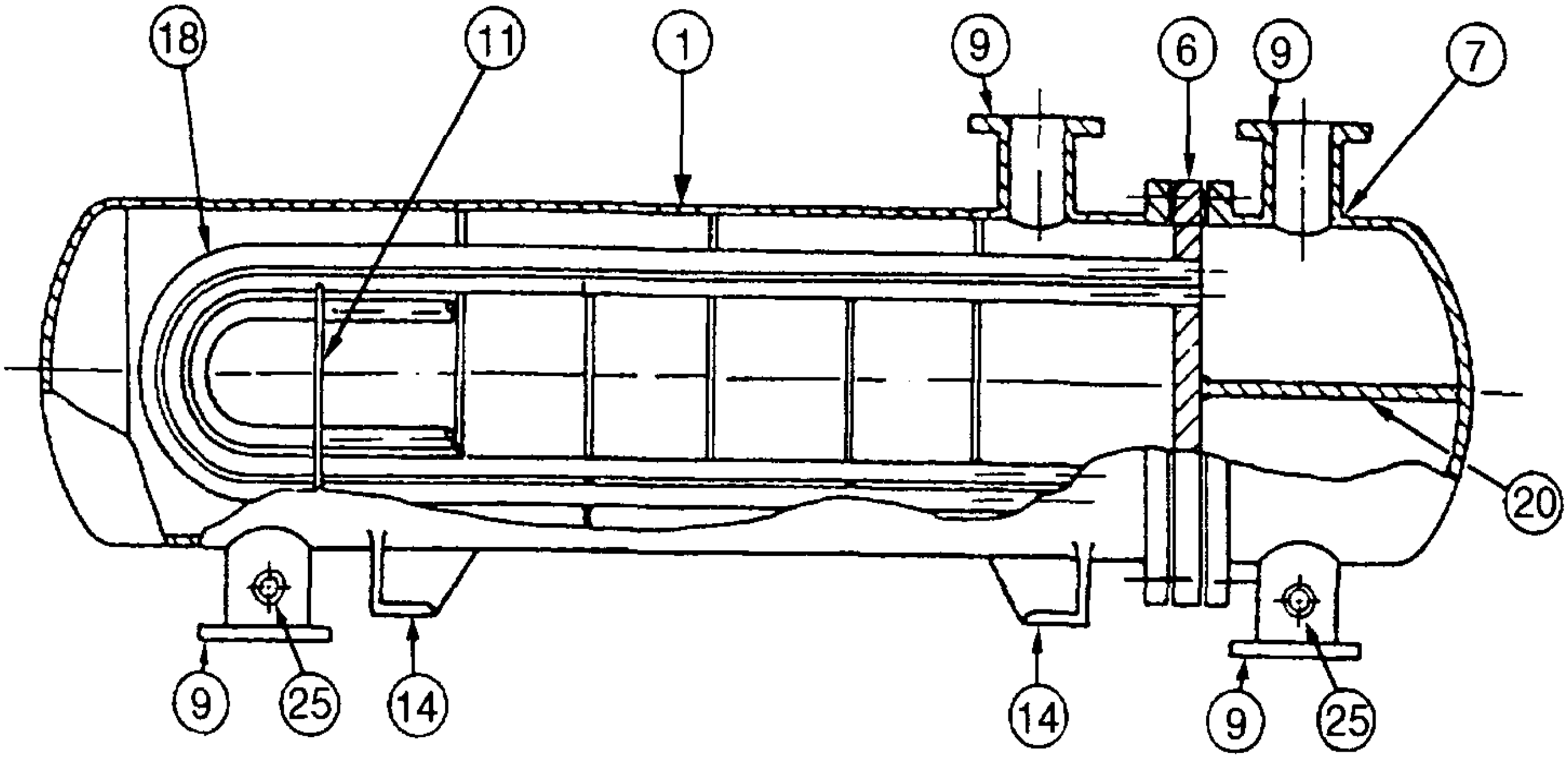
Essentially, a shell and tube exchanger consists of a bundle of tubes enclosed in a cylindrical shell. The ends of the tubes are fitted into tube sheets, which separate the shell-side and tube-side fluids. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly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



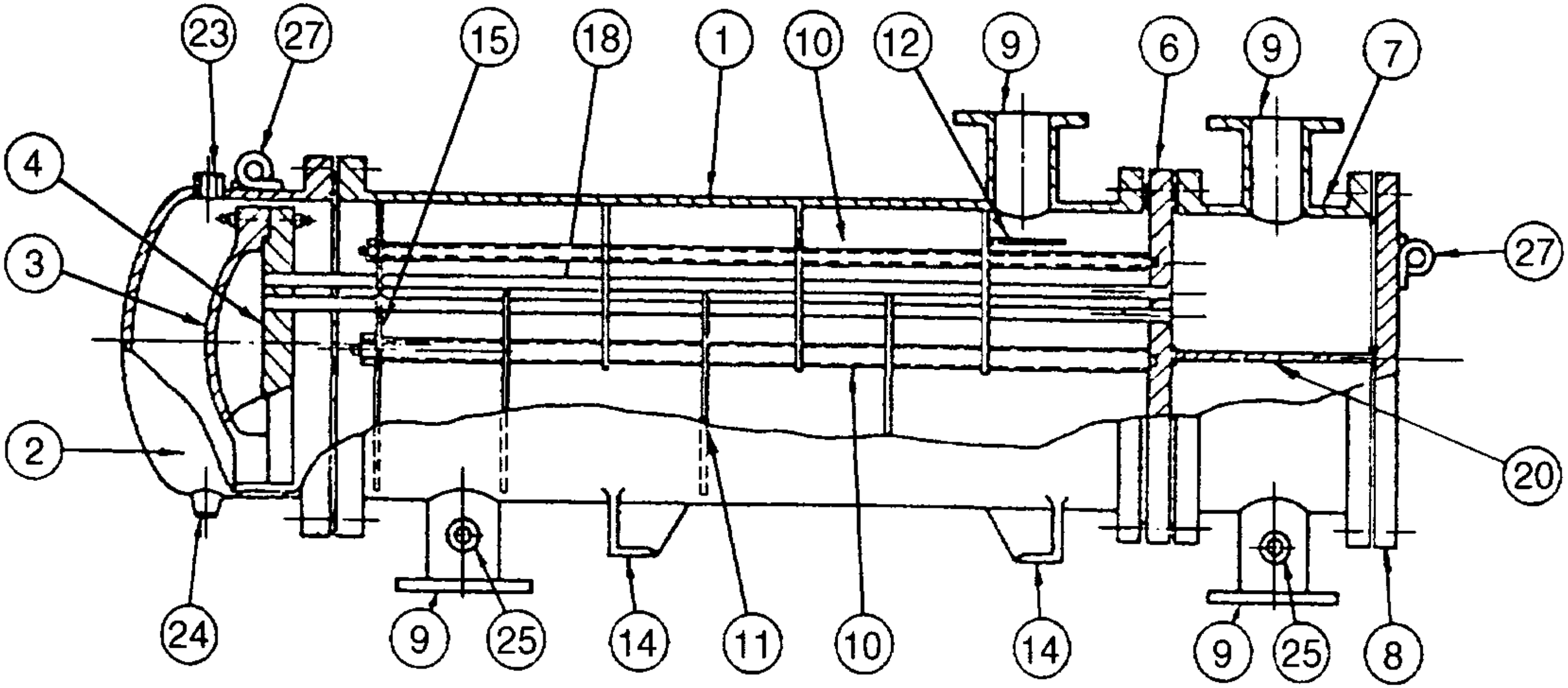
- ***Exchanger types***
- The principal types of shell and tube exchanger are shown in Figures below,
- Part number
- 1. Shell
- 2. Shell cover
- 3. Floating-head cover
- 4. Floating-tube plate
- 5. Clamp ring
- 6. Fixed-tube sheet (tube plate)
- 7. Channel (end-box or header)
- 8. Channel cover
- 9. Branch (nozzle)
- 10. Tie rod and spacer
- 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



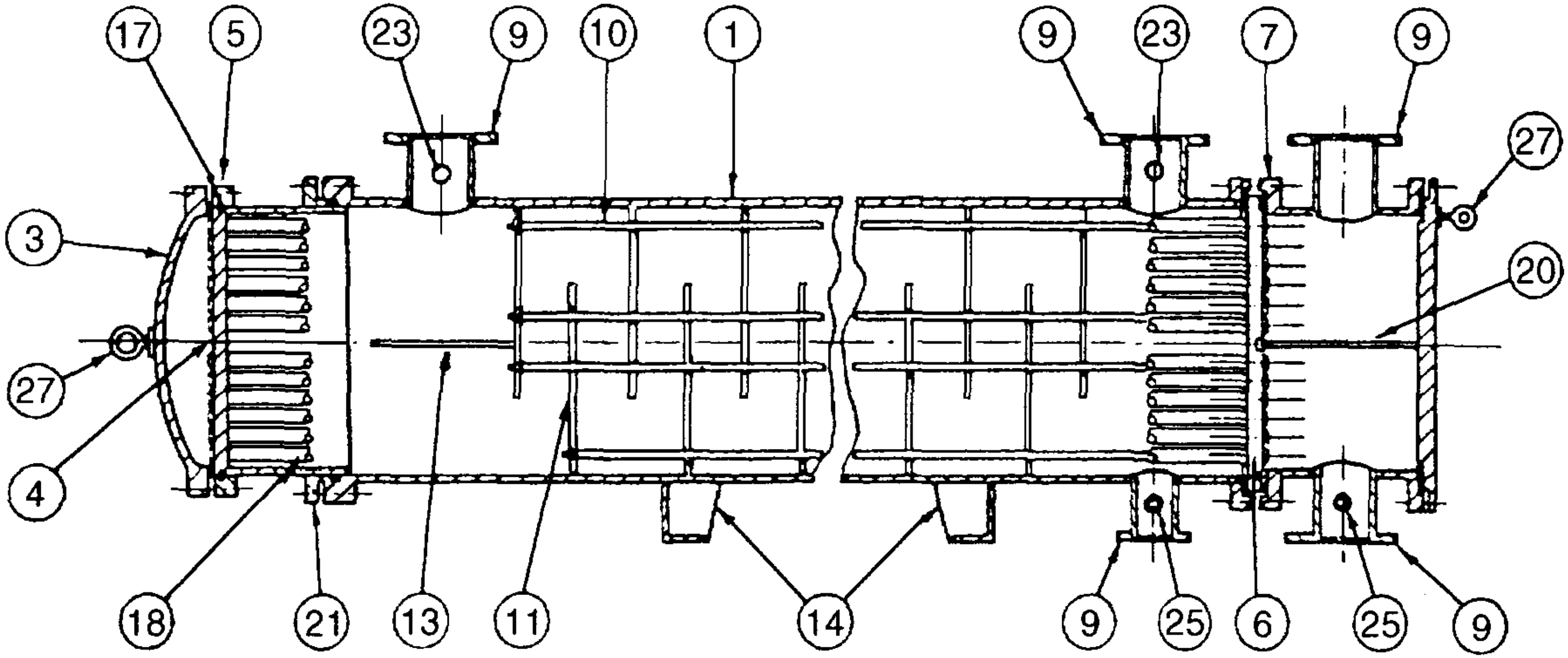
Fixed-tube plate (based on figures from BS 3274: 1960)



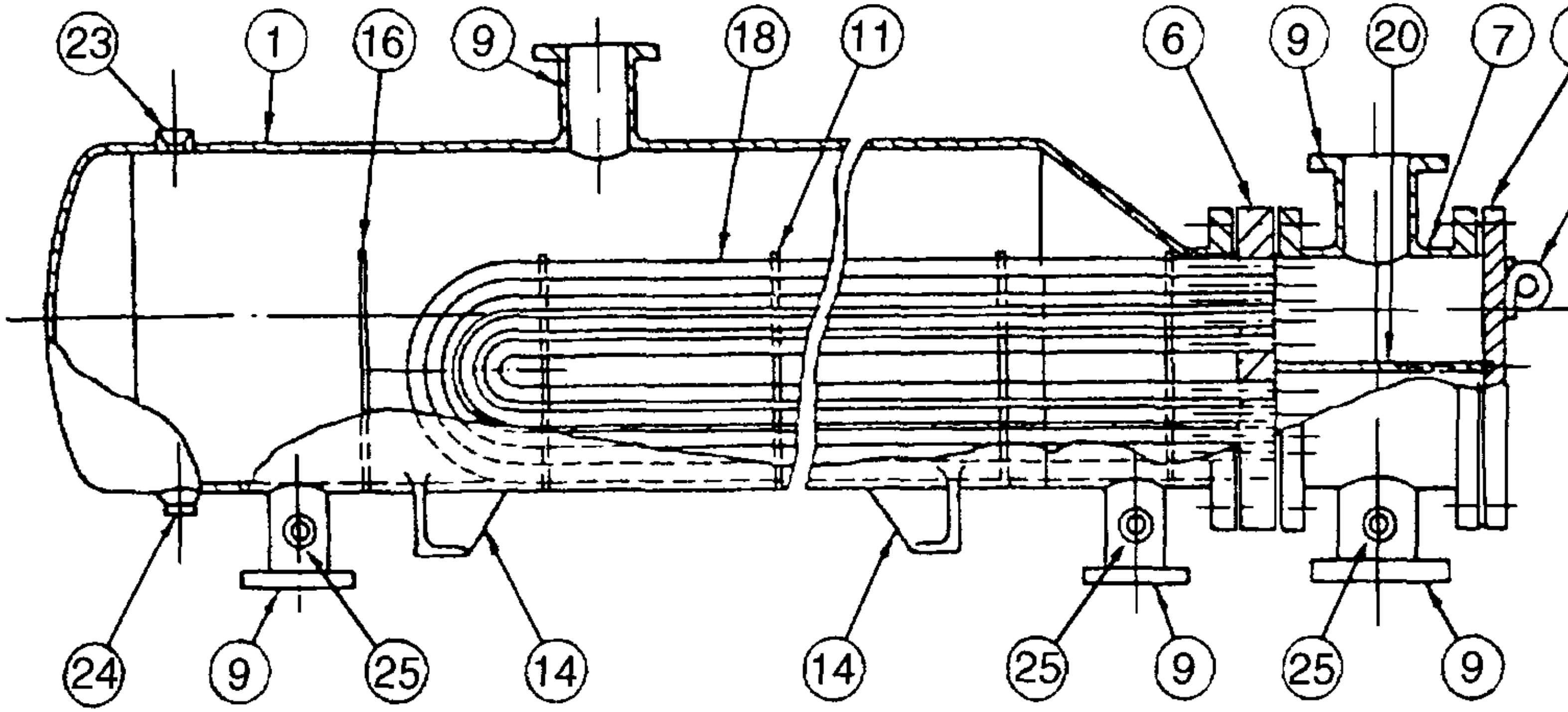
U-tube (based on figures from BS 3274: 1960)



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



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



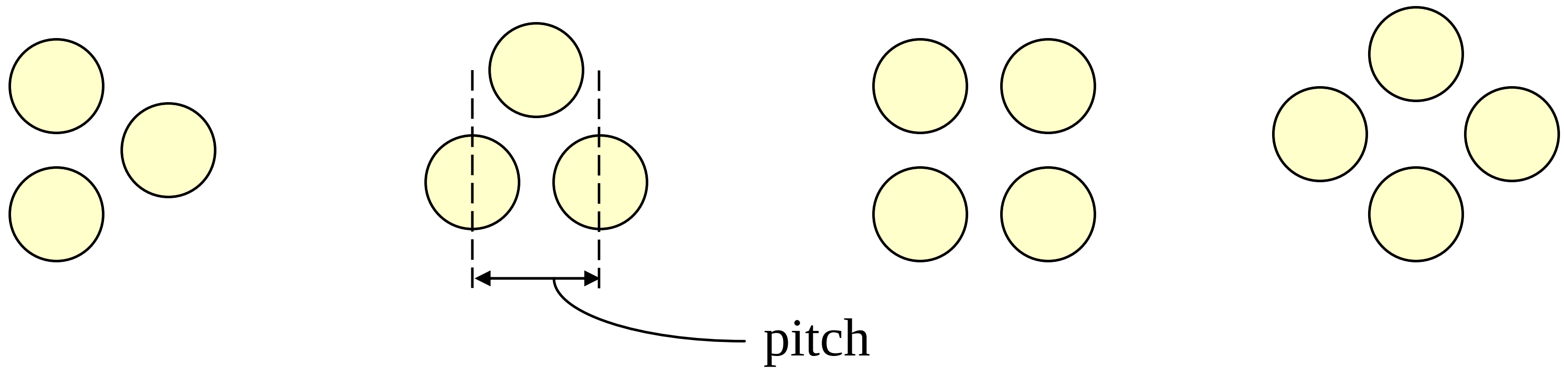
Kettle reboiler with U-tube bundle (based on figures from BS 3274: 1960)

- ***Tube arrangements***

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

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 centers) 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).

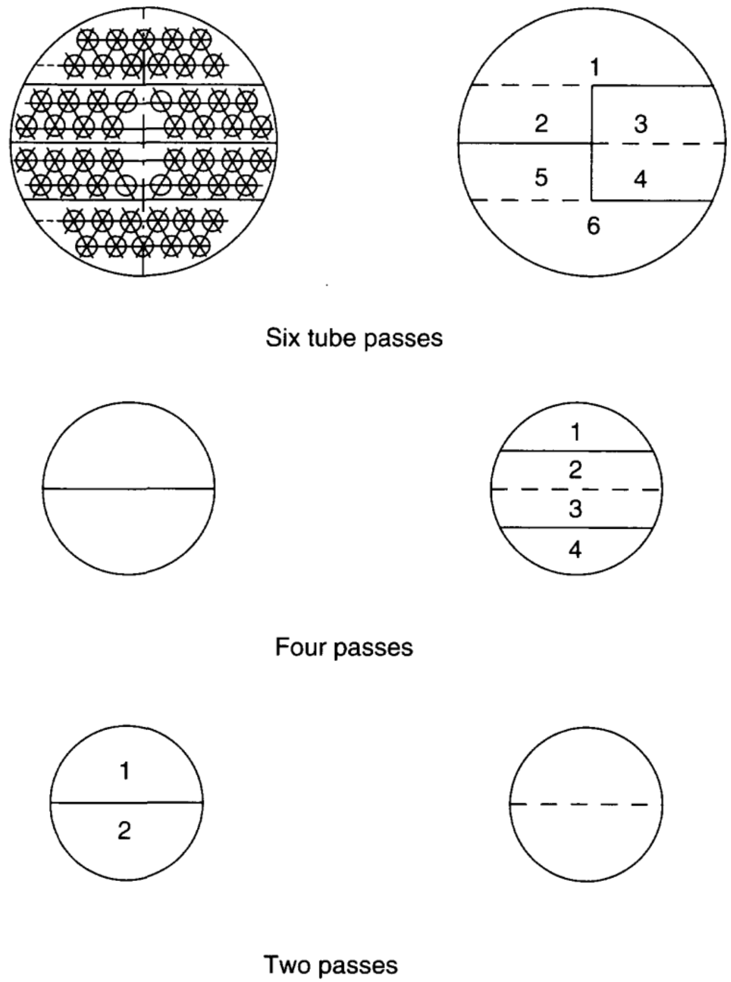
Tube layouts



Triangular 30^{o} Rotated triangular 60^{o} Square 90^{o} Rotated square 45^{o}

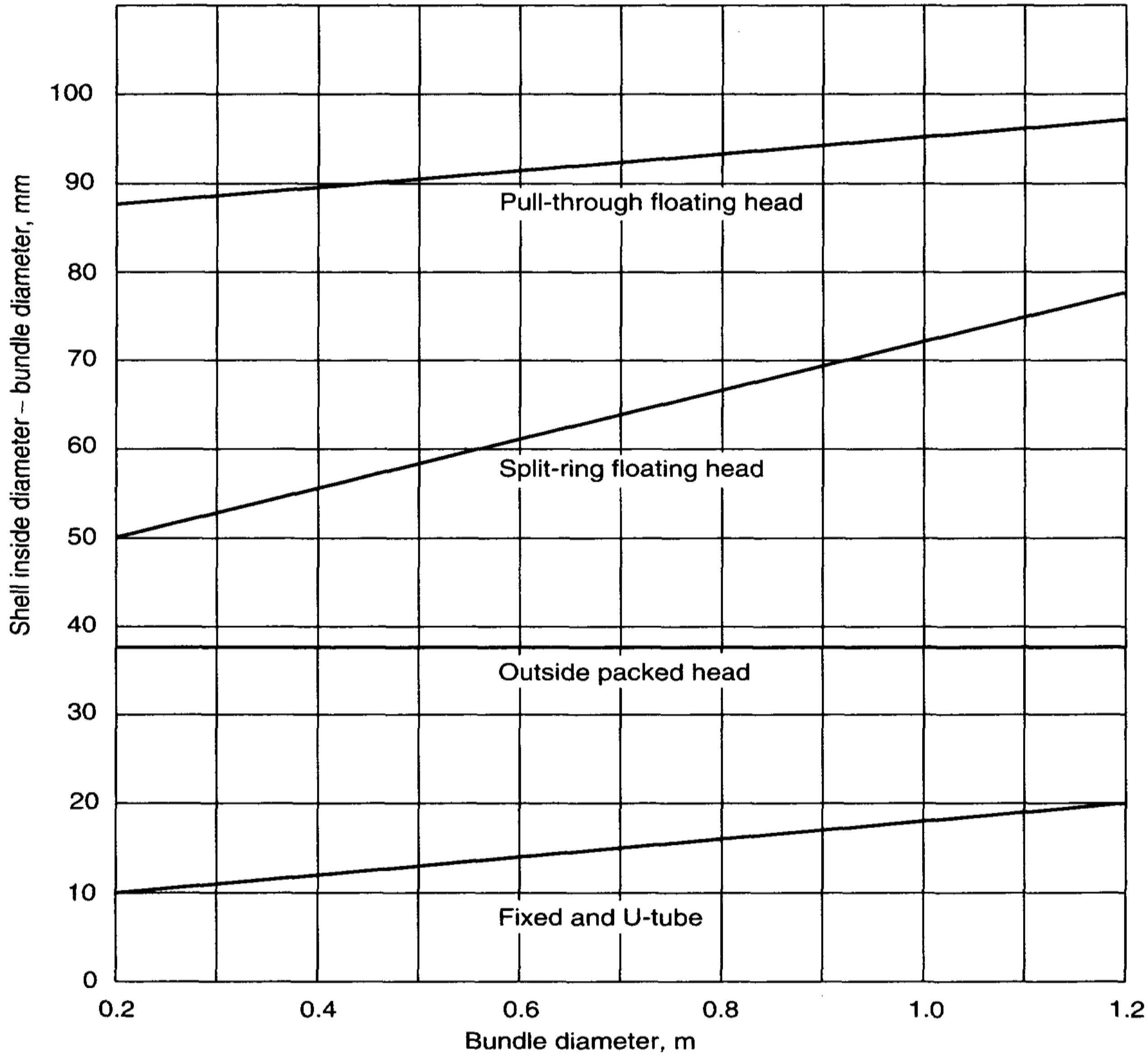
- ***Tube-side passes***

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. The arrangement of the pass partitions for 2, 4 and 6 tube passes are shown in Figure below;



Tube arrangements, showing pass-partitions in headers

Shell-to-bundle clearance (on diameter)



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

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

where

N_t = number of tubes,

D_b = bundle diameter, mm,

d_o = tube outside diameter, mm.

Constants for use in equation above

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

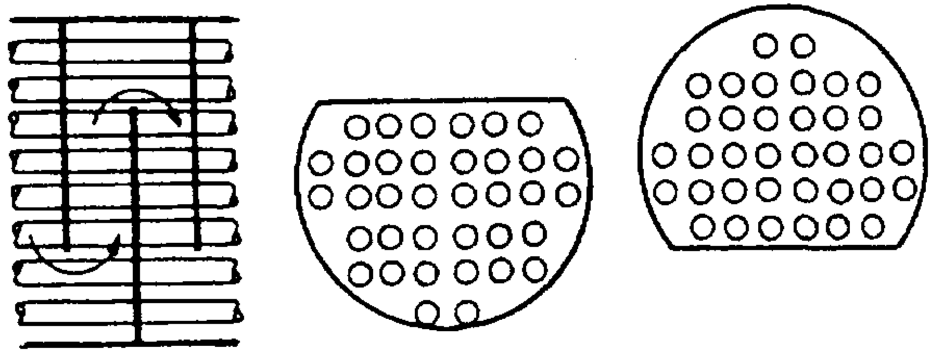
Square pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

Baffles

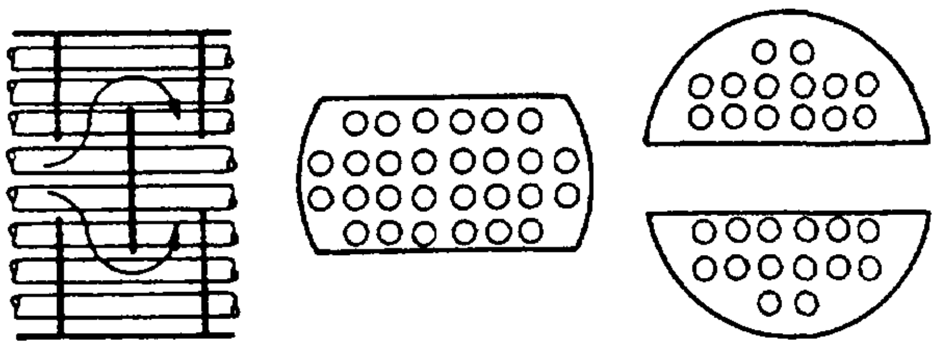
Baffles are used in the shell to direct the fluid stream across the tubes, to increase the fluid velocity and so improve the rate of heat transfer. The most commonly used type of baffle is the a single segmental baffle shown in Figures below. The minimum thickness to be used for baffles and support plates are given in the standards. The baffle spacing's 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. The clearance needed will depend on the shell diameter; typical values, and tolerances, are given in Table below;

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)

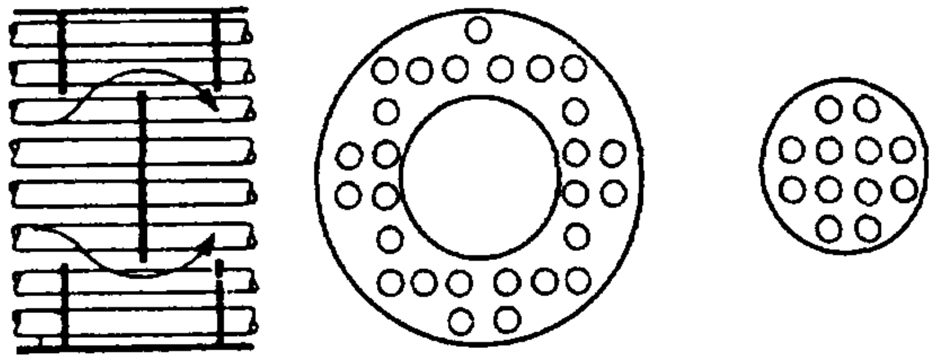


(a)

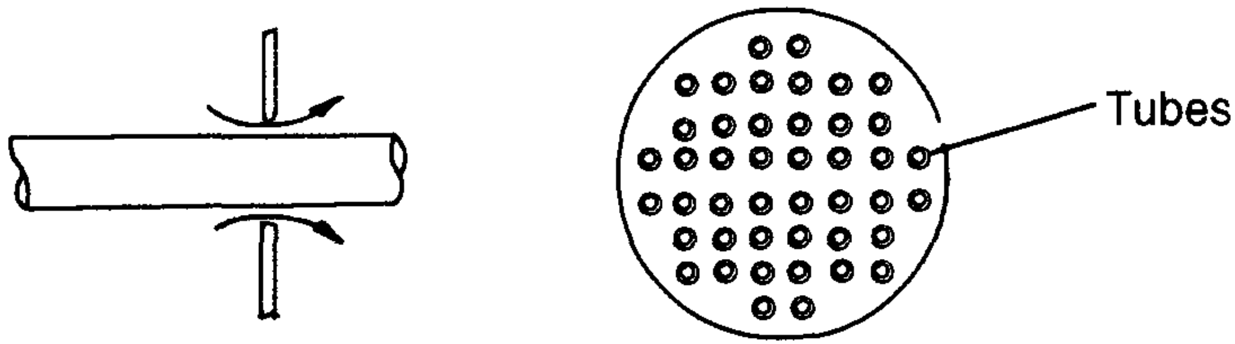


(b)

Types of baffle used in shell and tube heat exchangers,
(a) Segmental (b) Segraental and strip



(c)



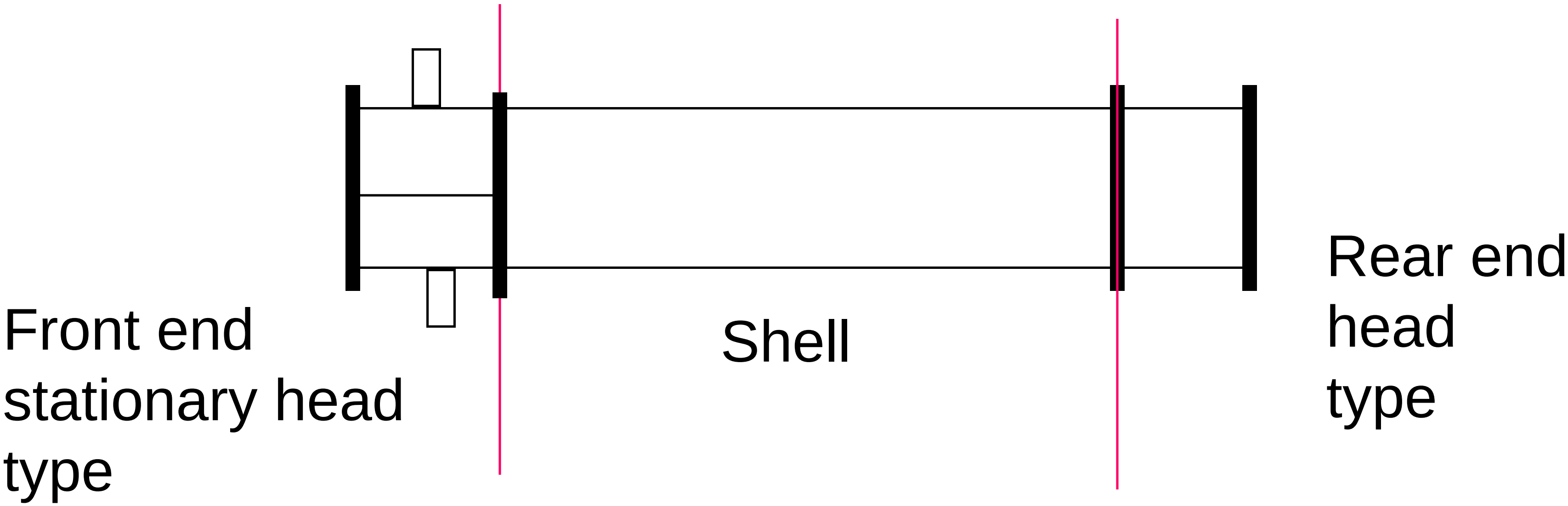
(d)

Types of baffle used in shell and tube heat exchangers,
(c) Disc and doughnut *(d)* Orifice

TEMA standards

- The design and construction is usually based on TEMA 8th Edition 1998
- Supplements pressure vessel codes like ASME and BS 5500
- Sets out constructional details, recommended tube sizes, allowable clearances, terminology etc.
- Provides basis for contracts
- Tends to be followed rigidly even when not strictly necessary
- Many users have their own additions to the standard which suppliers must follow

TEMA terminology

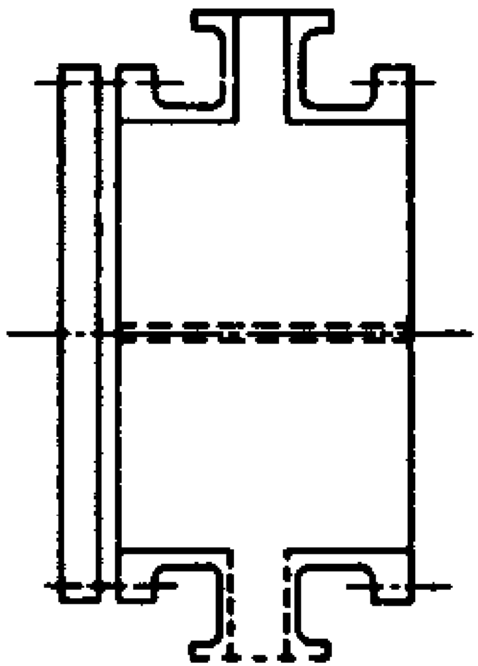


- Letters given for the front end, shell and rear end types
- Exchanger given three letter designation Above is AEL

Front head type

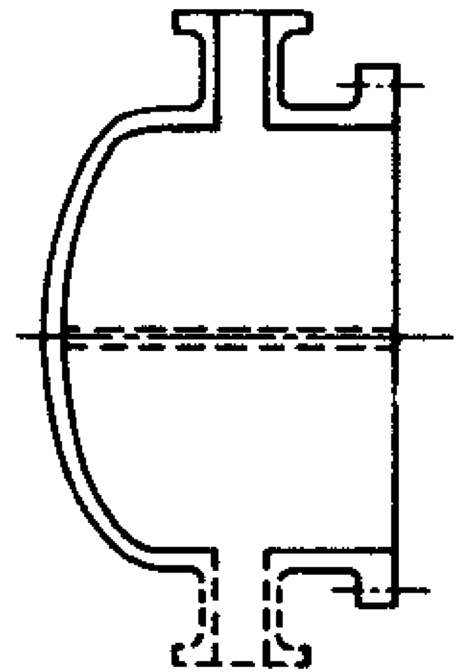
- A-type is standard for dirty tube side
- B-type for clean tube side duties. Use if possible since cheap and simple.

A



Channel and removable cover

B

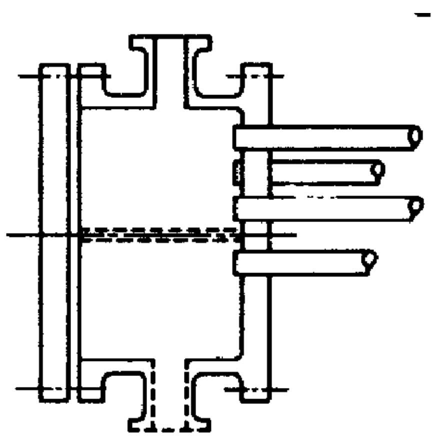


Bonnet (integral cover)

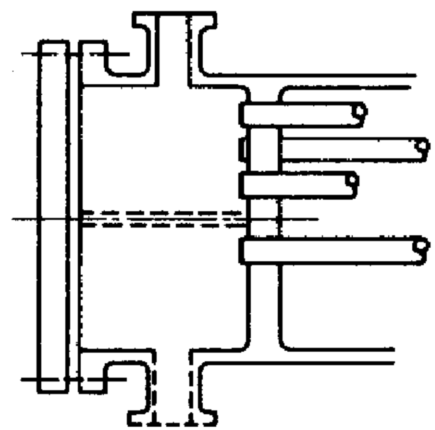
More front-end head types

- C-type with removable shell for hazardous tube-side fluids, heavy bundles or services that need frequent shell-side cleaning
- N-type for fixed for hazardous fluids on shell side
- D-type or welded to tube sheet bonnet for high pressure (over 150 bar)

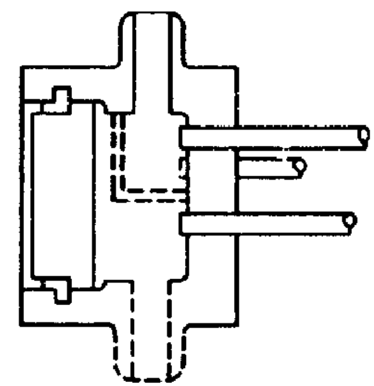
C



N

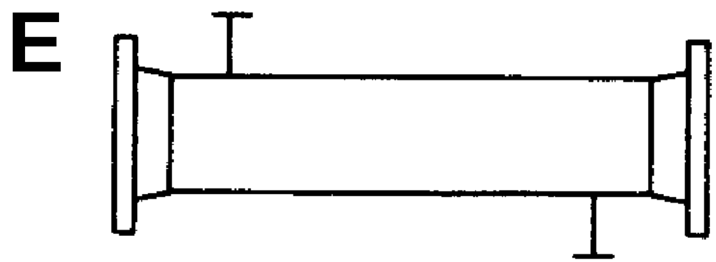


D

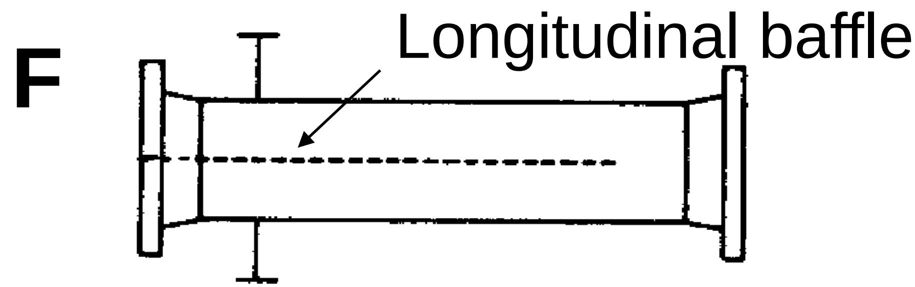


Shell type

- E-type shell should be used if possible but
- F shell gives pure counter-current flow with two tube passes (avoids very long exchangers)



One-pass shell

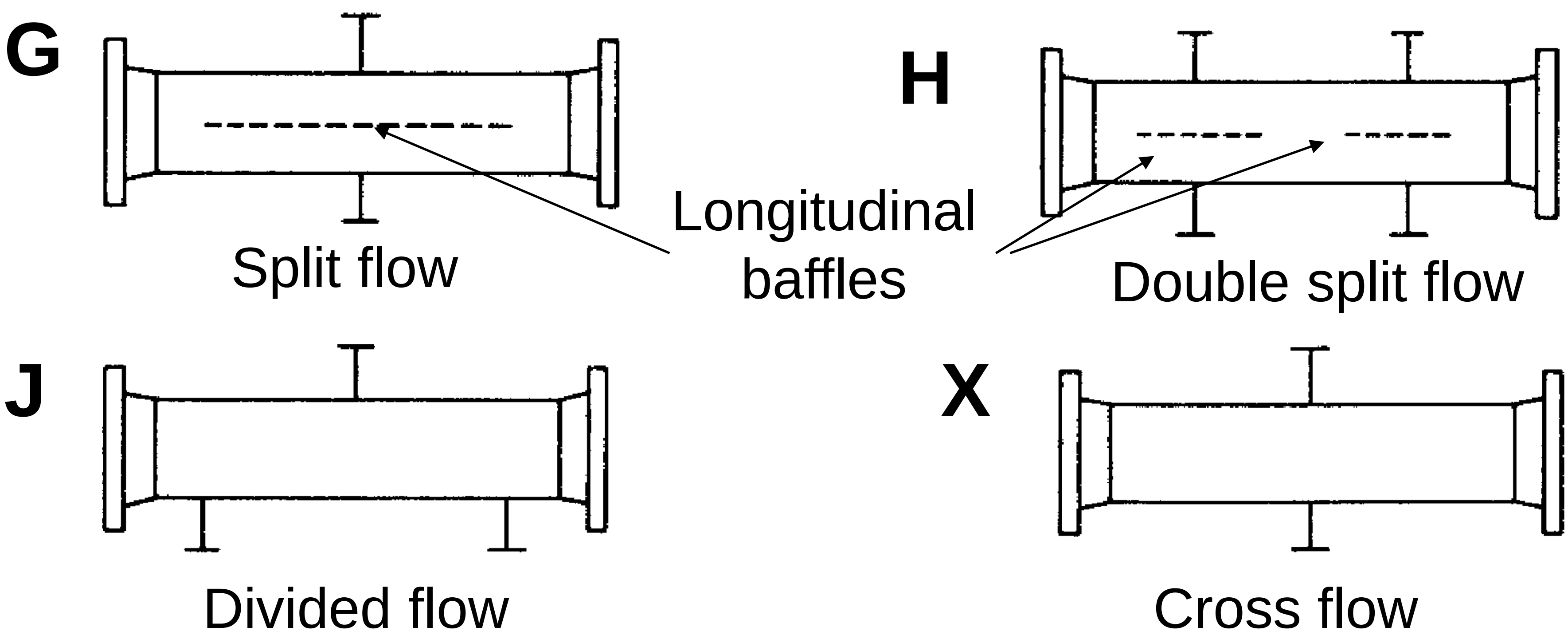


Two-pass shell

Note, longitudinal baffles are difficult to seal with the shell especially when reinserting the shell after maintenance

More shell types

- G and H shells normally only used for horizontal thermosyphon reboilers
- J and X shells if allowable pressure drop can not be achieved in an E shell



Rear head type

These fall into three general types

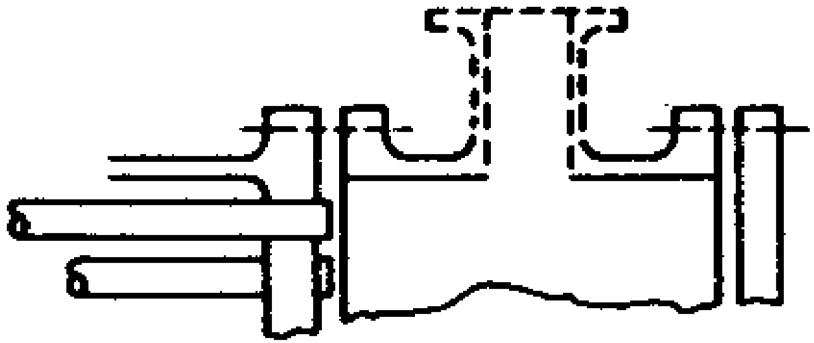
- fixed tube sheet (L, M, N)
- U-tube
- floating head (P, S, T, W)

Use fixed tube sheet if ΔT below $50^{o}C$, otherwise use other types to allow for differential thermal expansion

You can use bellows in shell to allow for expansion but these are special items which have pressure limitations (max. 35 bar)

Fixed rear head types

L



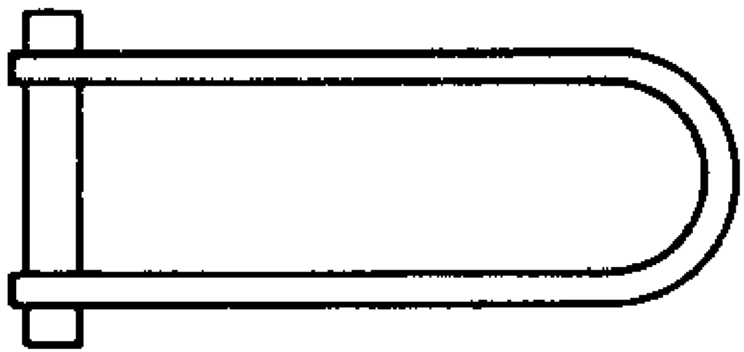
Fixed tube sheet

- L is a mirror of the A front end head
- M is a mirror of the bonnet (B) front end
- N is the mirror of the N front end

Floating heads and U tube

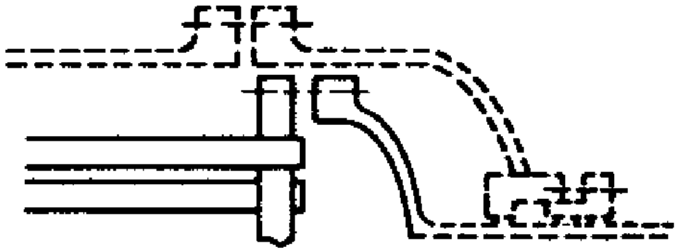
Allow bundle removal and mechanical cleaning on the shell side

- U tube is simple design but it is difficult to clean the tube side round the bend



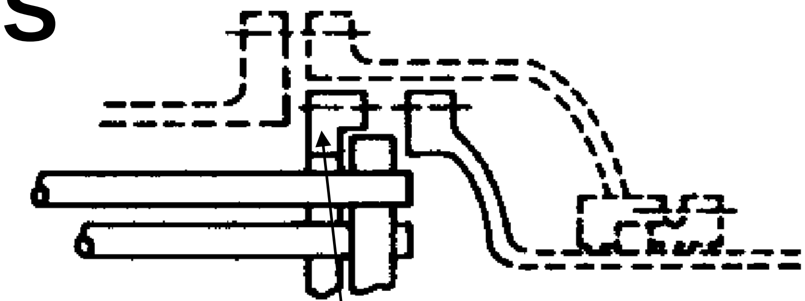
Floating heads

T

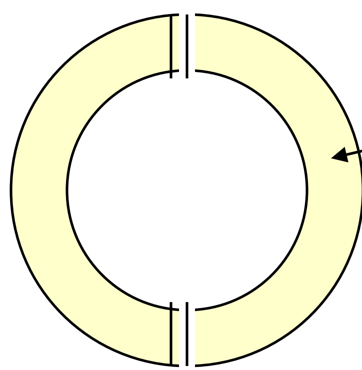


Pull through floating head
Note large shell/bundle gap

S



Similar to T but with smaller shell/ bundle gap

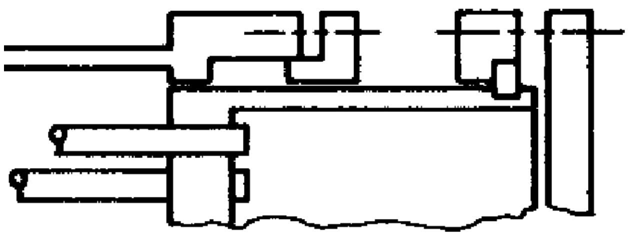


Split backing ring

Other floating heads

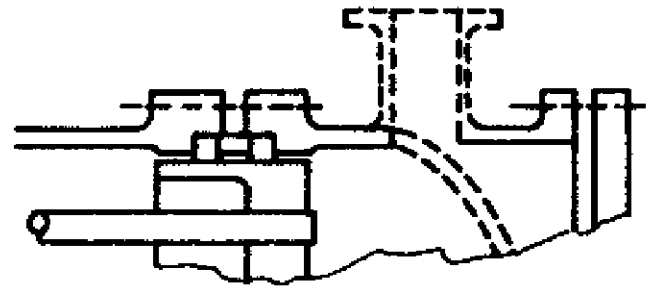
- Not used often and then with small exchangers

P



Outside packing to give
smaller shell/bundle gap

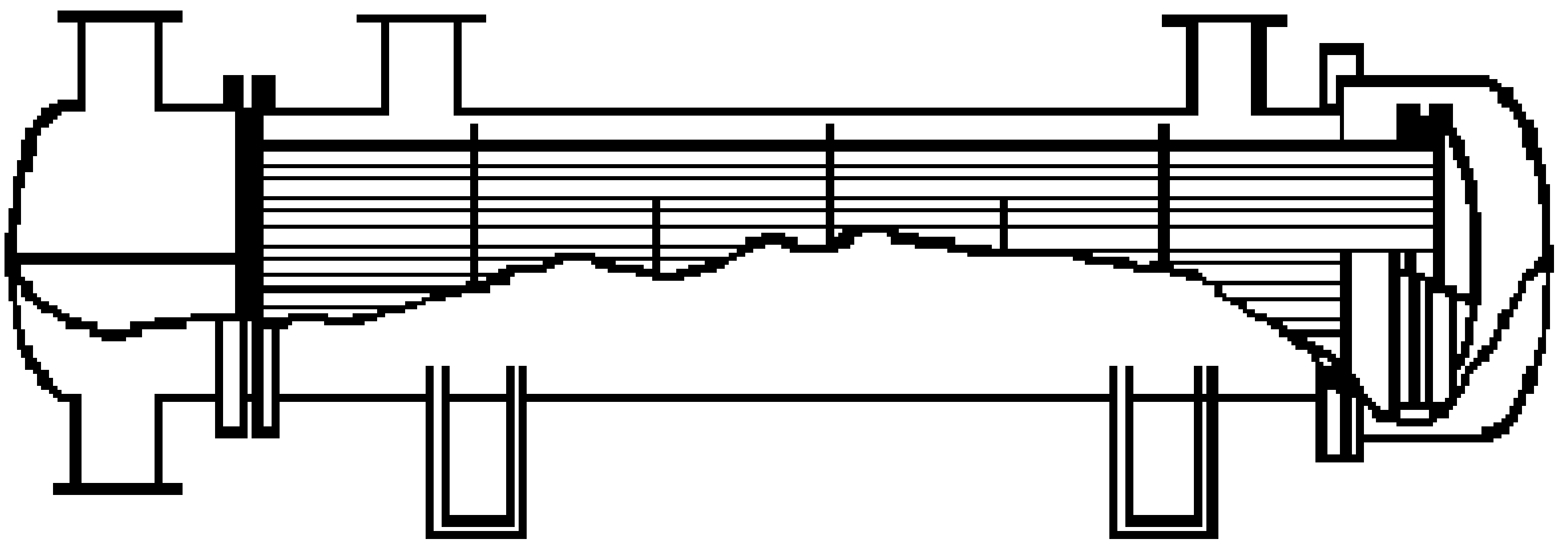
W

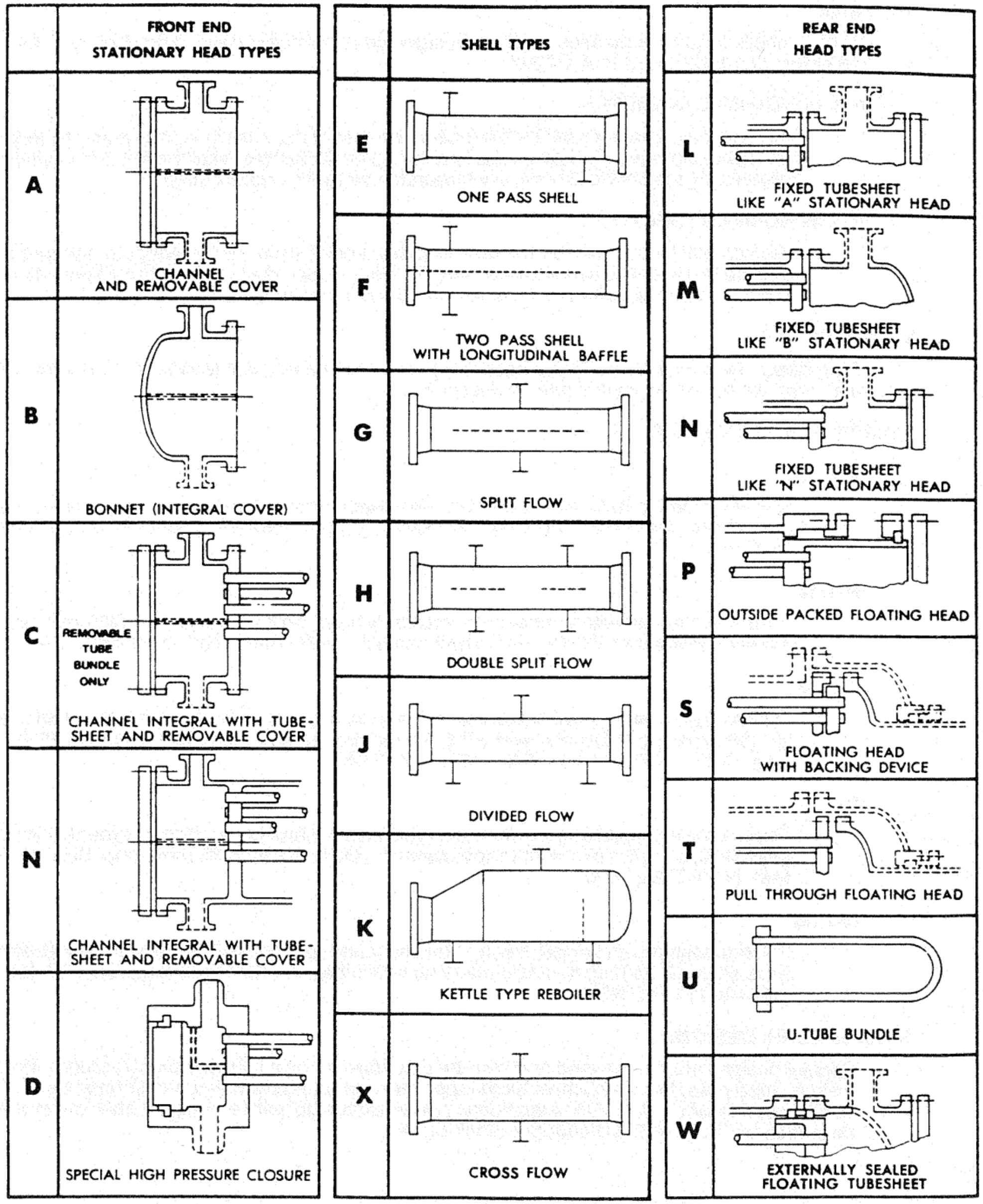


Externally sealed floating tube sheet
maximum of 2 tube passes

Example

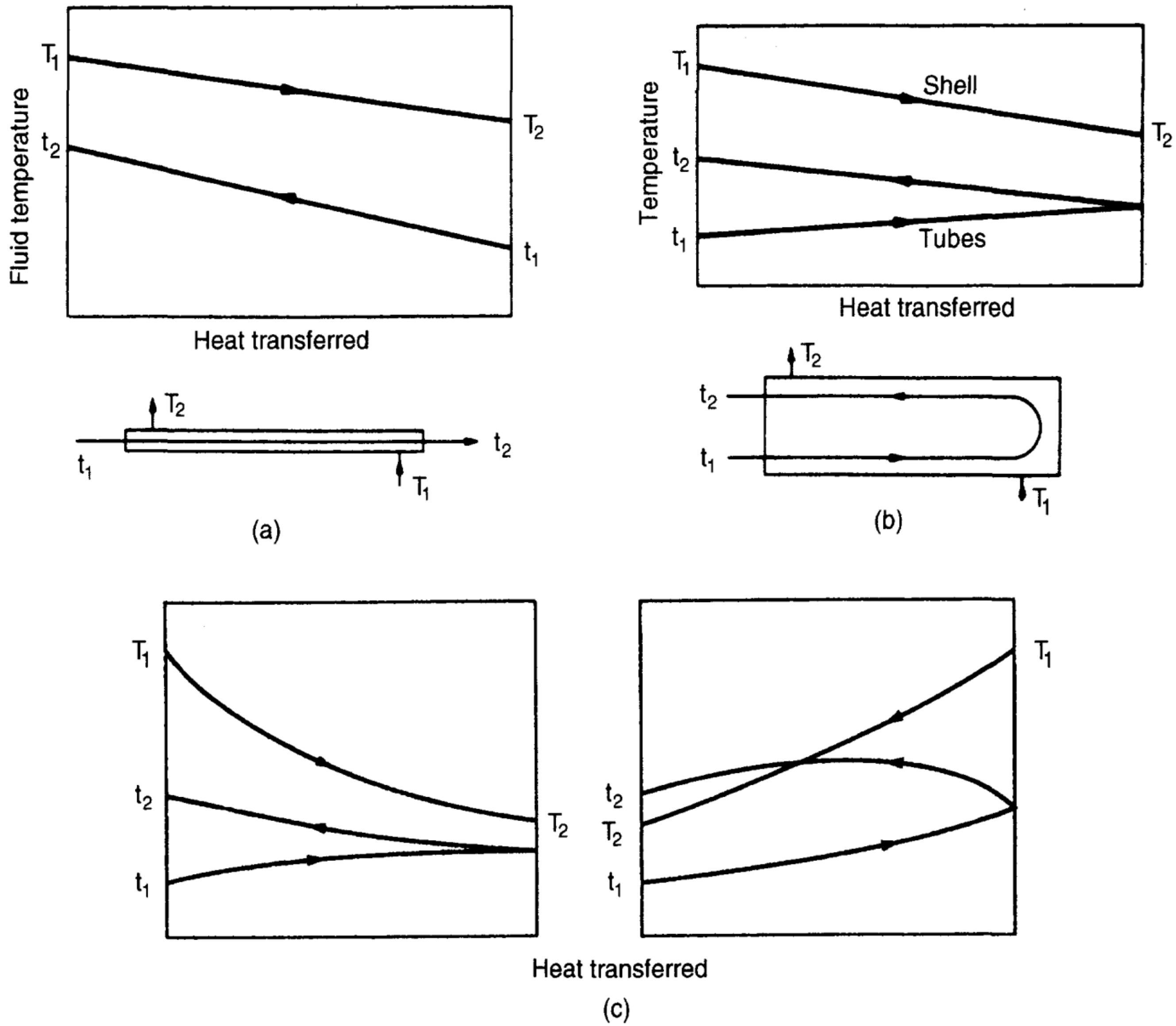
- BES
- Bonnet front end, single shell pass and split backing ring floating head




FRONT END
STATIONARY HEAD TYPES
A
CHANNEL
AND REMOVABLE COVER
B
BONNET (INTEGRAL COVER)
C
REMOVABLE
TUBE
BUNDLE
ONLY
CHANNEL INTEGRAL WITH TUBE-
SHEET AND REMOVABLE COVER
N
CHANNEL INTEGRAL WITH TUBE-
SHEET AND REMOVABLE COVER
D
SPECIAL HIGH PRESSURE CLOSURE
SHELL TYPES
E
ONE PASS SHELL
F
TWO PASS SHELL
WITH LONGITUDINAL BAFFLE
G
SPLIT FLOW
H
DOUBLE SPLIT FLOW
J
DIVIDED FLOW
K
KETTLE TYPE REBOILER
X
CROSS FLOW
REAR END
HEAD TYPES
L
FIXED TUBESHEET
LIKE "A" STATIONARY HEAD
M
FIXED TUBESHEET
LIKE "B" STATIONARY HEAD
N
FIXED TUBESHEET
LIKE "N" STATIONARY HEAD
P
OUTSIDE PACKED FLOATING HEAD
S
FLOATING HEAD
WITH BACKING DEVICE
T
PULL THROUGH FLOATING HEAD
U
U-TUBE BUNDLE
W
EXTERNALLY SEALED
FLOATING TUBESHEET

Flow Patterns

- Parallel Flow
- Counter Current Flow
- Shell and Tube with baffles
- Cross Flow



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

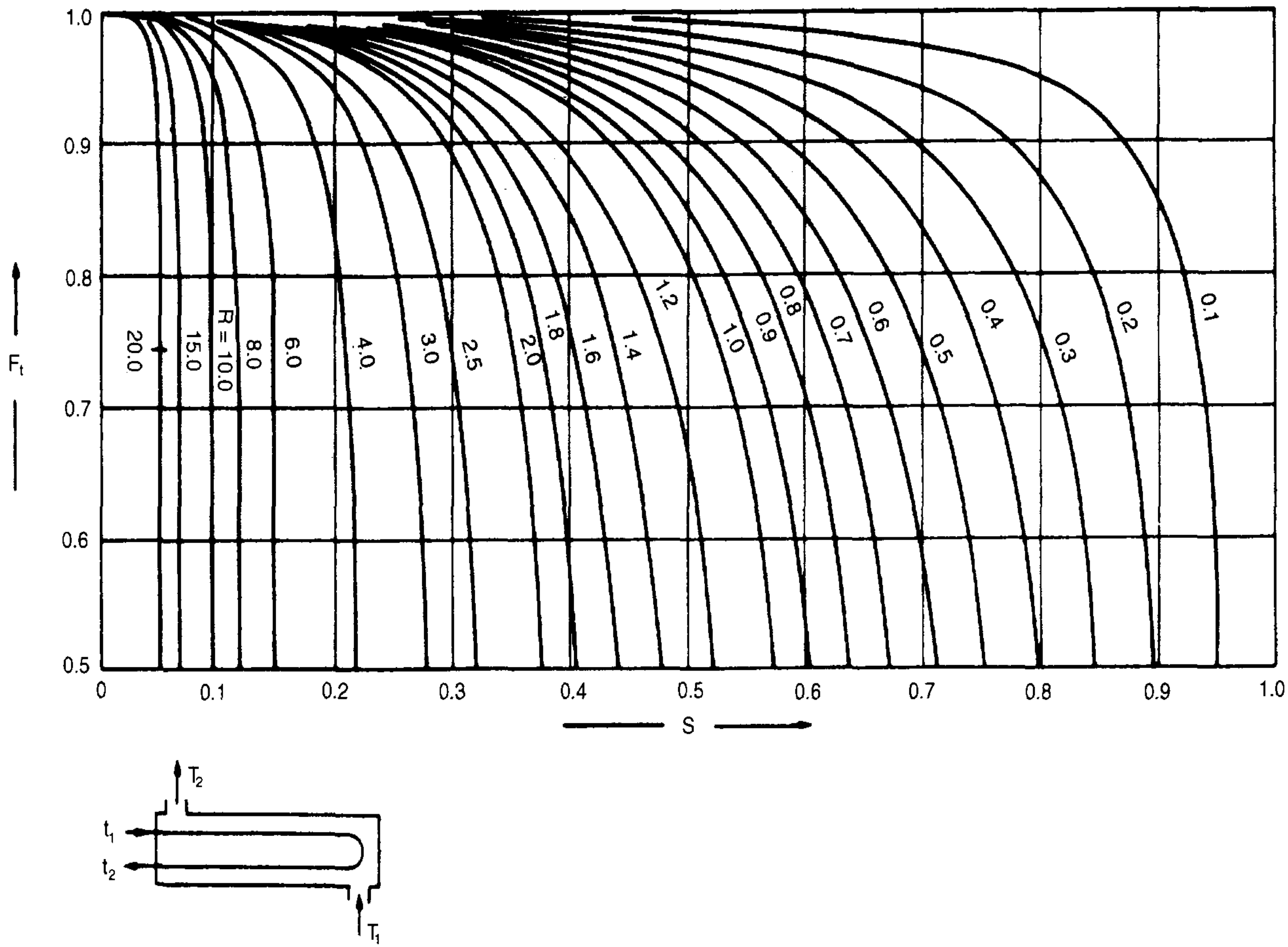
Flow Structure

$Q = U\,A\,F\,\Delta T_{lm\text{-}counter}$

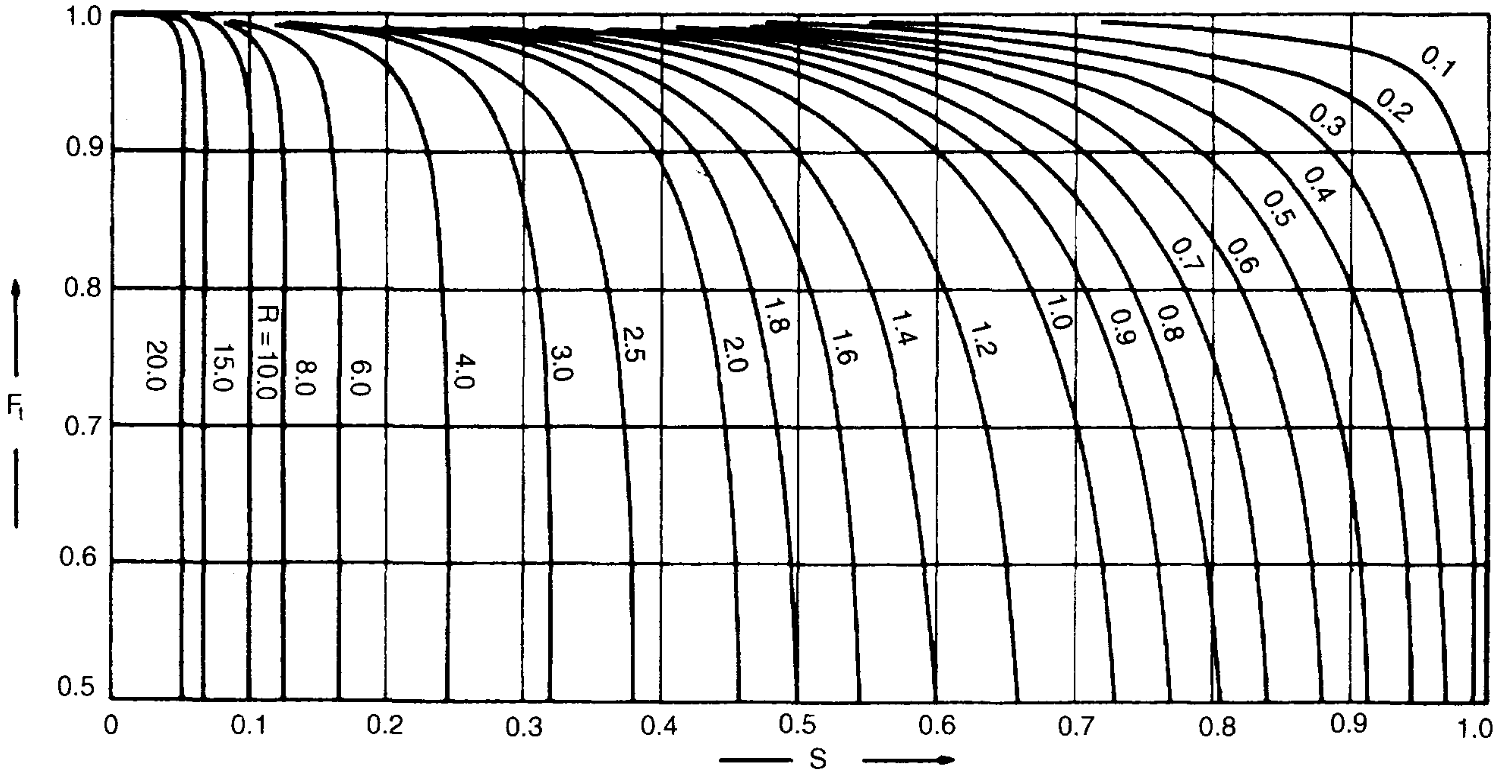
The correction factor is a function of the shell and tube fluid temperatures, and the number of tube and shell passes. It is normally correlated as a function of two dimensionless temperature ratios (*R* and *S*);

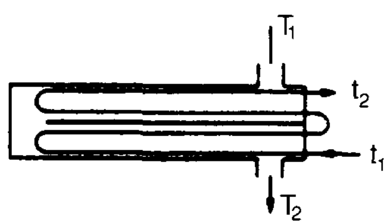
$$F = \frac{\sqrt{R^2+1}\,\ln\left[\frac{(1-S)}{1-RS}\right]}{(R-1)\ln\left[\frac{2-S\left(R+1-\sqrt{R^2+1}\right)}{2-S\left(R+1+\sqrt{R^2+1}\right)}\right]}$$

$$R = \frac{T_{hot\ in} - T_{hot\ out}}{T_{cold\ out} - T_{cold\ in}} \qquad S = \frac{T_{cold\ out} - T_{cold\ in}}{T_{hot\ in} - T_{cold\ in}}$$

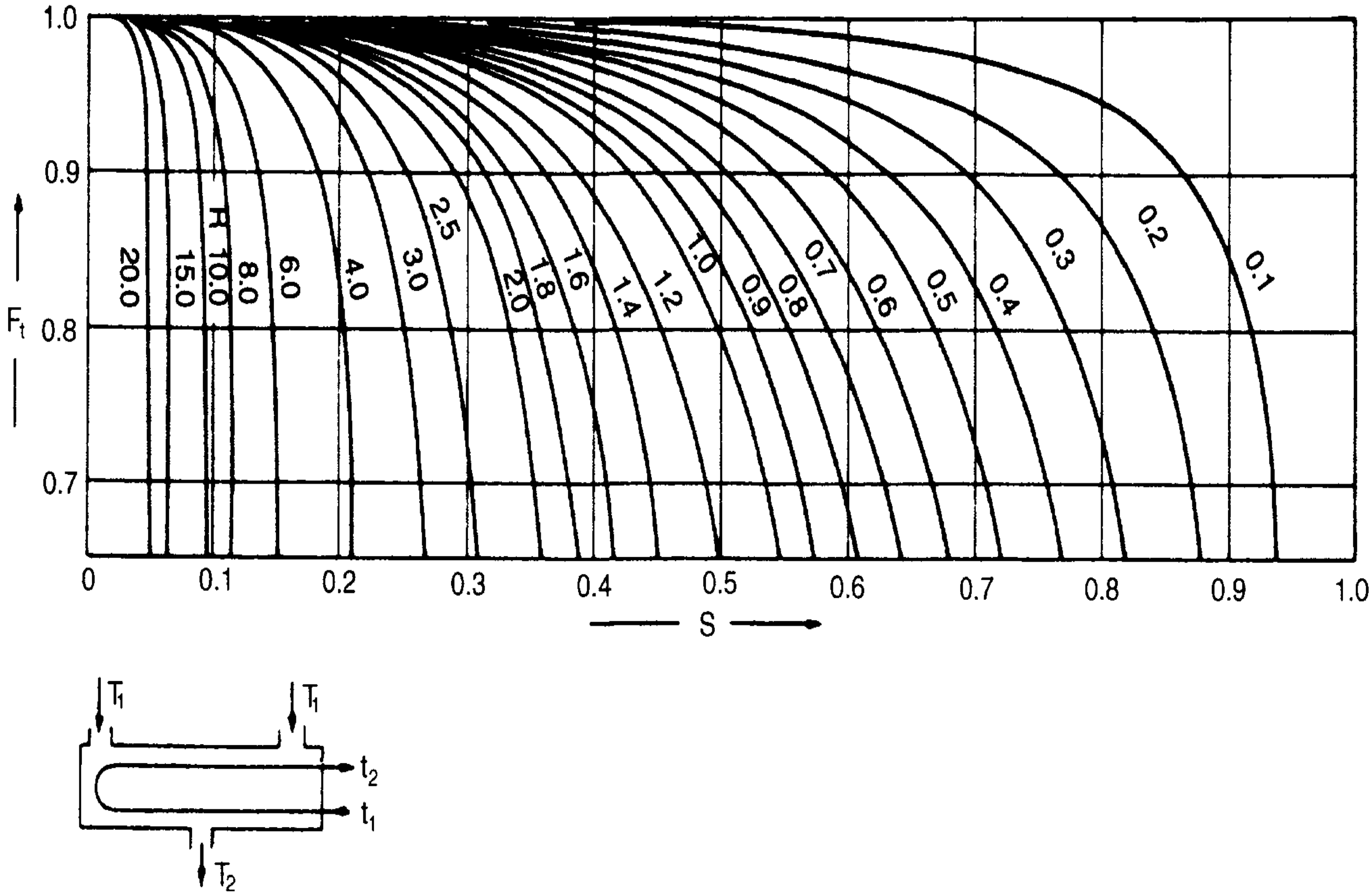


Temperature correction factor: one shell pass; two or more even tube 'passes

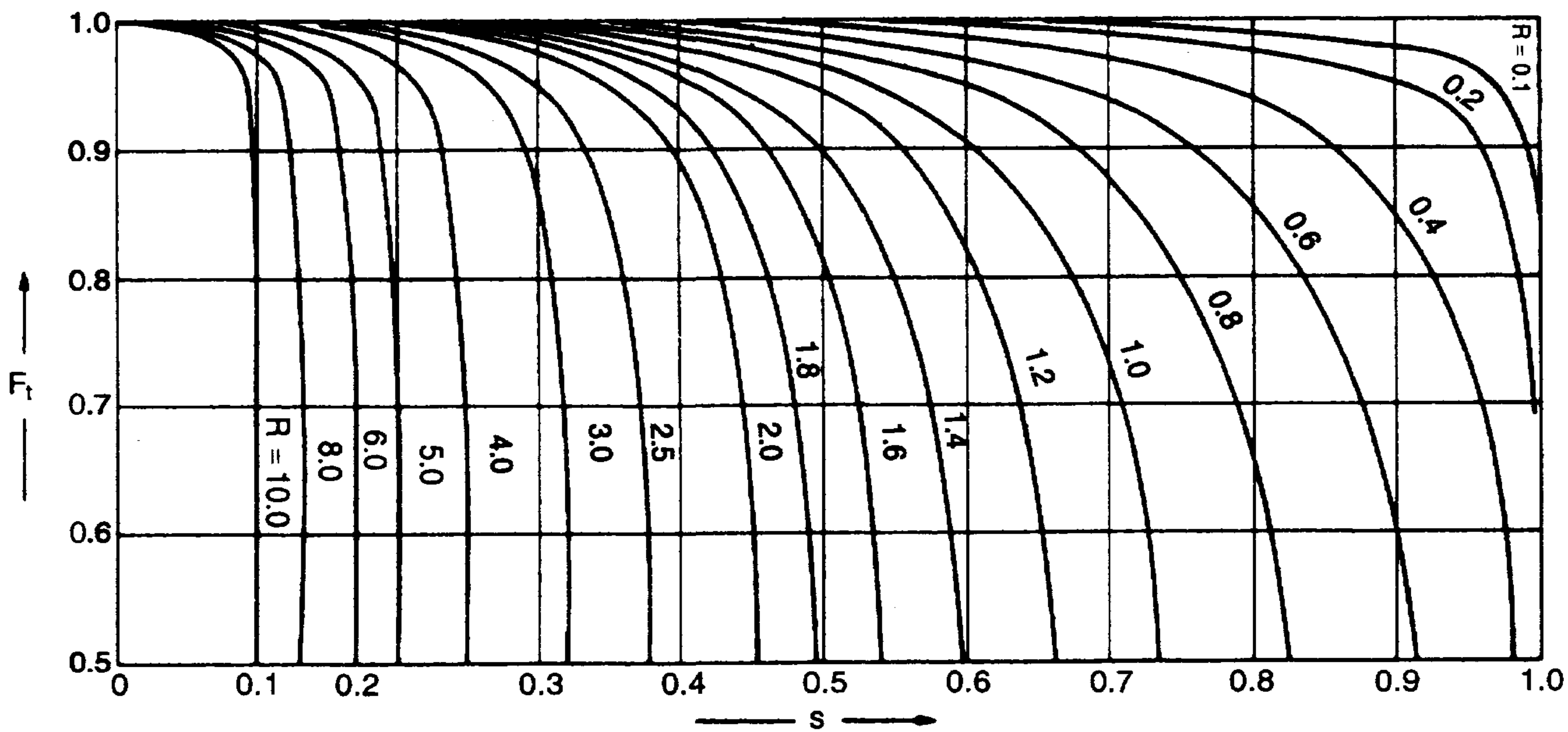


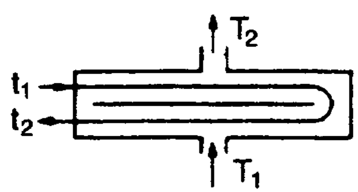


Temperature correction factor: two shell passes; four or multiples of four tube passes



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





Temperature correction factor, split flow shell, 2 tube pass

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. 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.

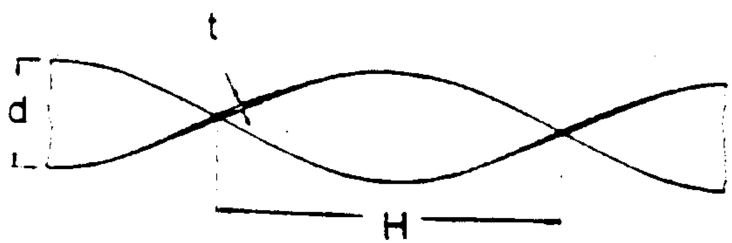
Note

If no obvious benefit, try streams both ways and see which gives best design

Tube-side enhancement using inserts

Spiral wound wire and twisted tape

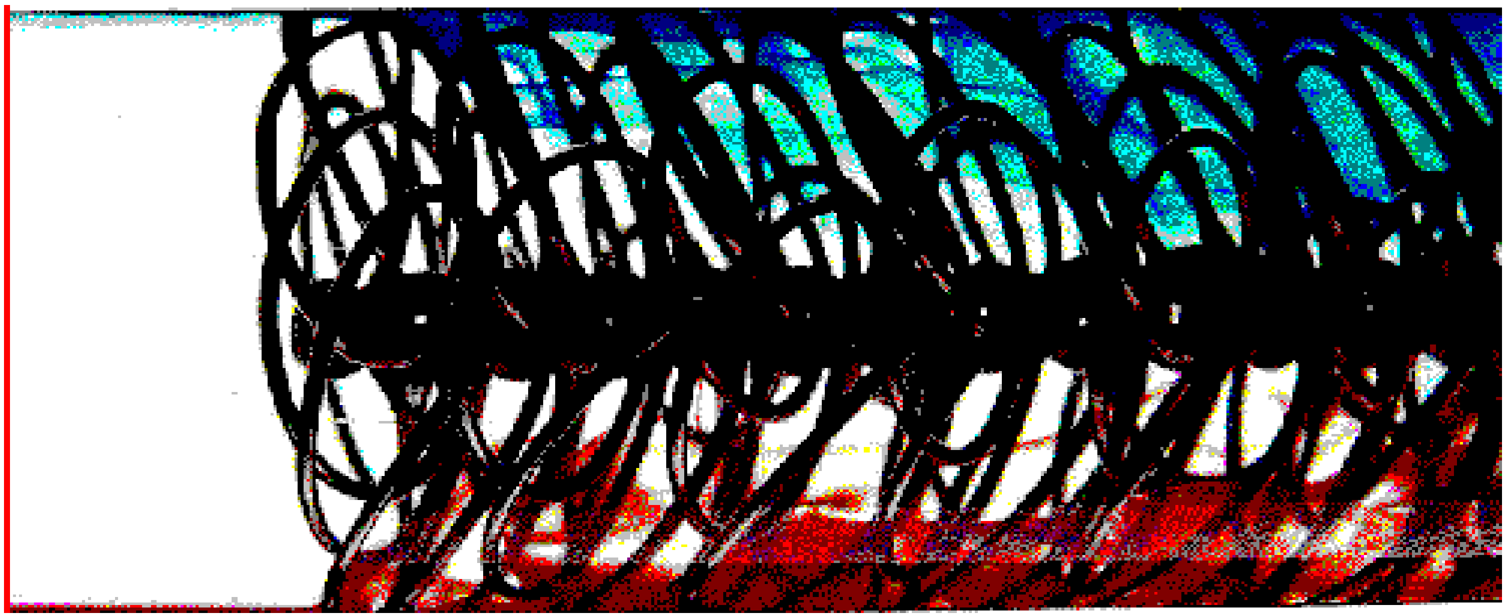
- Increase tube side heat transfer coefficient but at the cost of larger pressure drop (although exchanger can be reconfigured to allow for higher pressure drop)
- In some circumstances, they can significantly reduce fouling. In others they may make things worse
- Can be retrofitted



Twisted tape

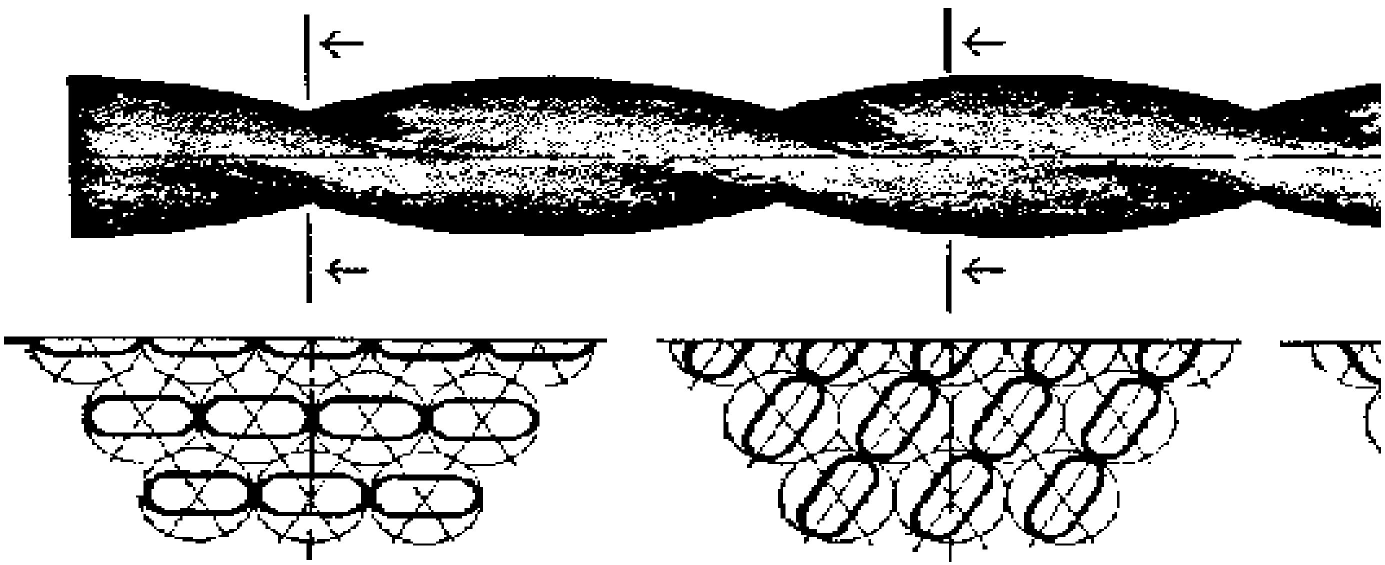
Wire-wound inserts

- Both mixes the core (radial mixing) and breaks up the boundary layer
- Available in range of wire densities for different duties



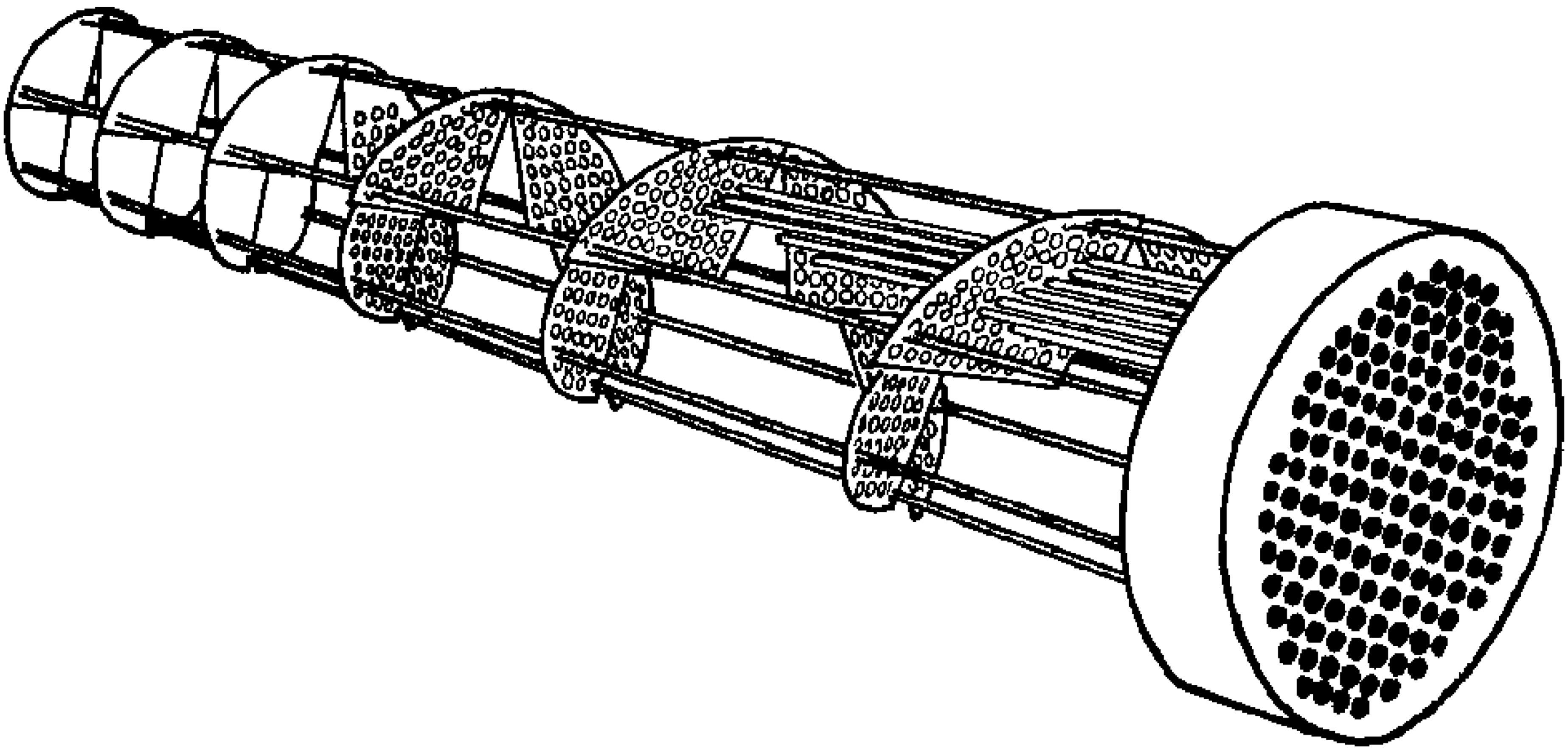
Twisted tube (Brown Fintube)

- Tubes support each other
- Used for single phase and condensing duties in the power, chemical and pulp and paper industries



Shell-side helical flow (ABB Lummus)

Independently developed by two groups in Norway and Czech Republic



Design Procedure for Shell and Tube Heat Exchanger

1. Define heat transfer, mass flowrate and temperature deference for heat exchanger

$$Q = mC_p\Delta T$$

2. Collect physical properties at bulk temperature (ρ, μ, *Cp*, *k*…etc.)
3. Select a trial value for overall heat transfer coefficient table below;
4. Decide the heat exchanger layout (Number of passes for tube and shell side
5. Calculate the mean deference temperature (LTMD)
6. Calculate R and S value in order to find the correction factor (F_T)
7. Calculate the heat transfer area
8. Decide the heat exchanger layout and tube size in order to calculate number of tubes, Bundle and shell diameter

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

9. Calculate the heat transfer coefficients for each stream (h_i, h_o)

$$Nu = \frac{h_i D_{ht}}{k_f} = J_{ht} \operatorname{Re} \operatorname{Pr}^{0.33} (\frac{\mu}{\mu_w})^{0.14}$$

$$Nu = \frac{h_s D_{hs}}{k_f} = J_{hs} \operatorname{Re} \operatorname{Pr}^{0.33} (\frac{\mu}{\mu_w})^{0.14}$$

10. Predict the fouling factor for two sides of heat exchanger (1/hiD, 1/hoD).
11. Calculate the overall heat transfer coefficient exchanger using equation below;

$$\frac{1}{U} = \frac{D_0}{D_i}\frac{1}{h_i} + \frac{1}{h_o} + \frac{D_o \ln \frac{D_o}{D_i}}{2K_w} + \frac{D_o}{D_i}\frac{1}{h_{iD}} + \frac{1}{h_{oD}}$$

12. Recalculate the heat transfer coefficient using equation below and compare it with assumed value.

$$Q = UAF_T \Delta T_{lm}$$

13. If the calculated value within (0-10)%, the calculated value is satisfied and if not return to step ***No. 4***
14. Calculate the pressure drop for each side of heat exchanger if the results unsatisfactory return to steps 4 or 8.

$$\Delta P_t = n\left[8 j_f \left(\frac{L}{D_i}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14} + 2.5\right]\frac{\rho u_t^2}{2}$$

$$\Delta P_s = 8J_f \left(\frac{D_s}{D_e}\right)\left(\frac{L}{L_B}\right)\left(\frac{\rho u_t^2}{2}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

15. Optimize the design calculation by repeating steps 4-10 as necessary to design cheapest heat exchanger which satisfy (smallest area)

$$D_e = \frac{1.27}{d_o}\left[p_t^2 - 0.785 d_o^2 \right]$$ *Square pitch*

$$D_e = \frac{1.1}{d_o}\left[p_t^2 - 0.917 d_o^2 \right]$$ *Triangle pitch*

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

$$D_s = D_b + Clearance$$

Clearance can be found from TEMA depend on type of tube sheet and Bundle diameter

$$A_s = \frac{(p_t - d_o) D_s L_B}{p_t}$$

$$L_B = \left(0.2 - 1\right) D_s$$

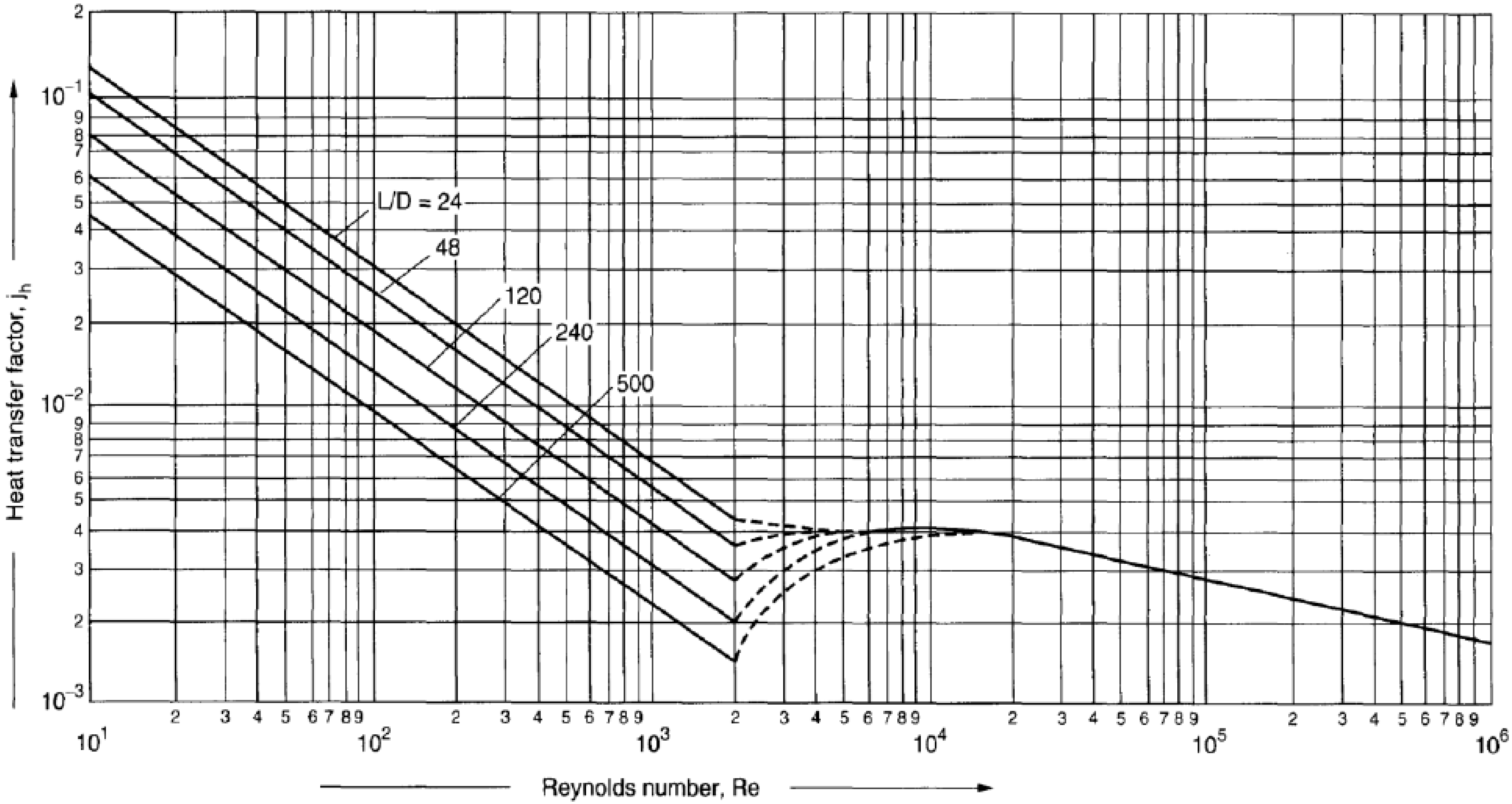


Figure 1. Tube-Side heat-transfer factor

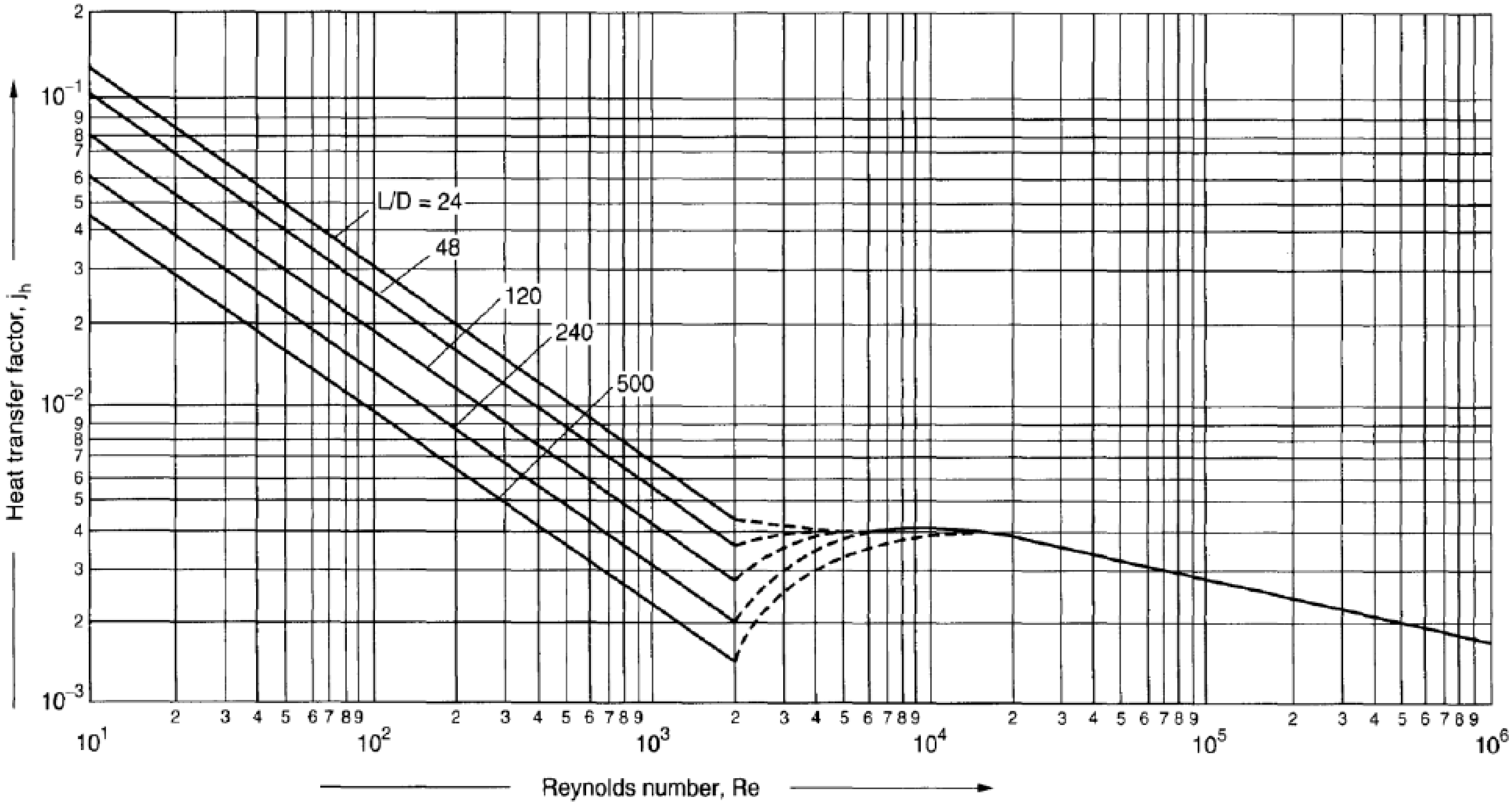


Figure 1. Tube-Side heat-transfer factor

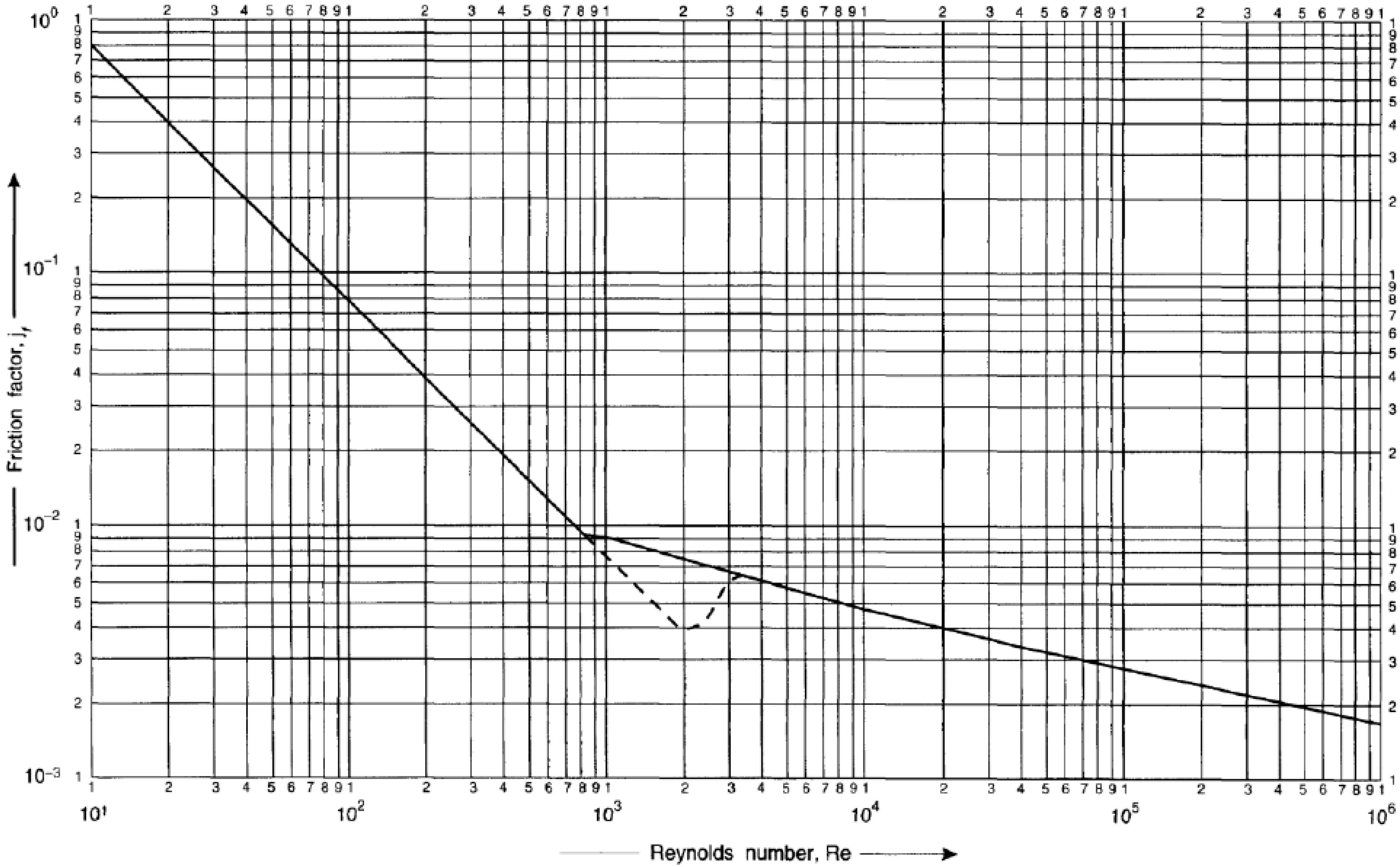


Figure 2. Tube-Side Friction factors

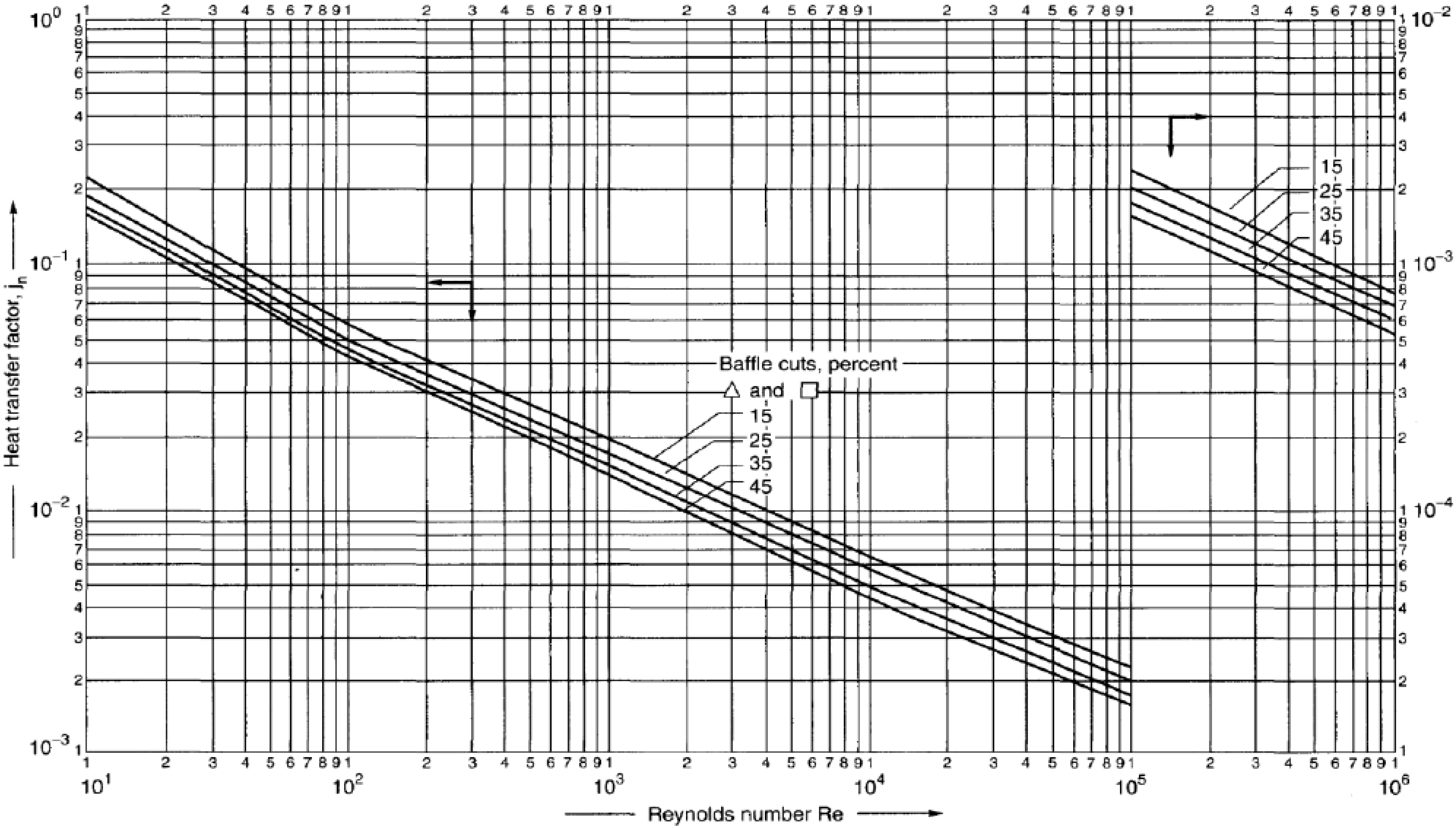


Figure3. Shell-Side heat-transfer, segmental baffles

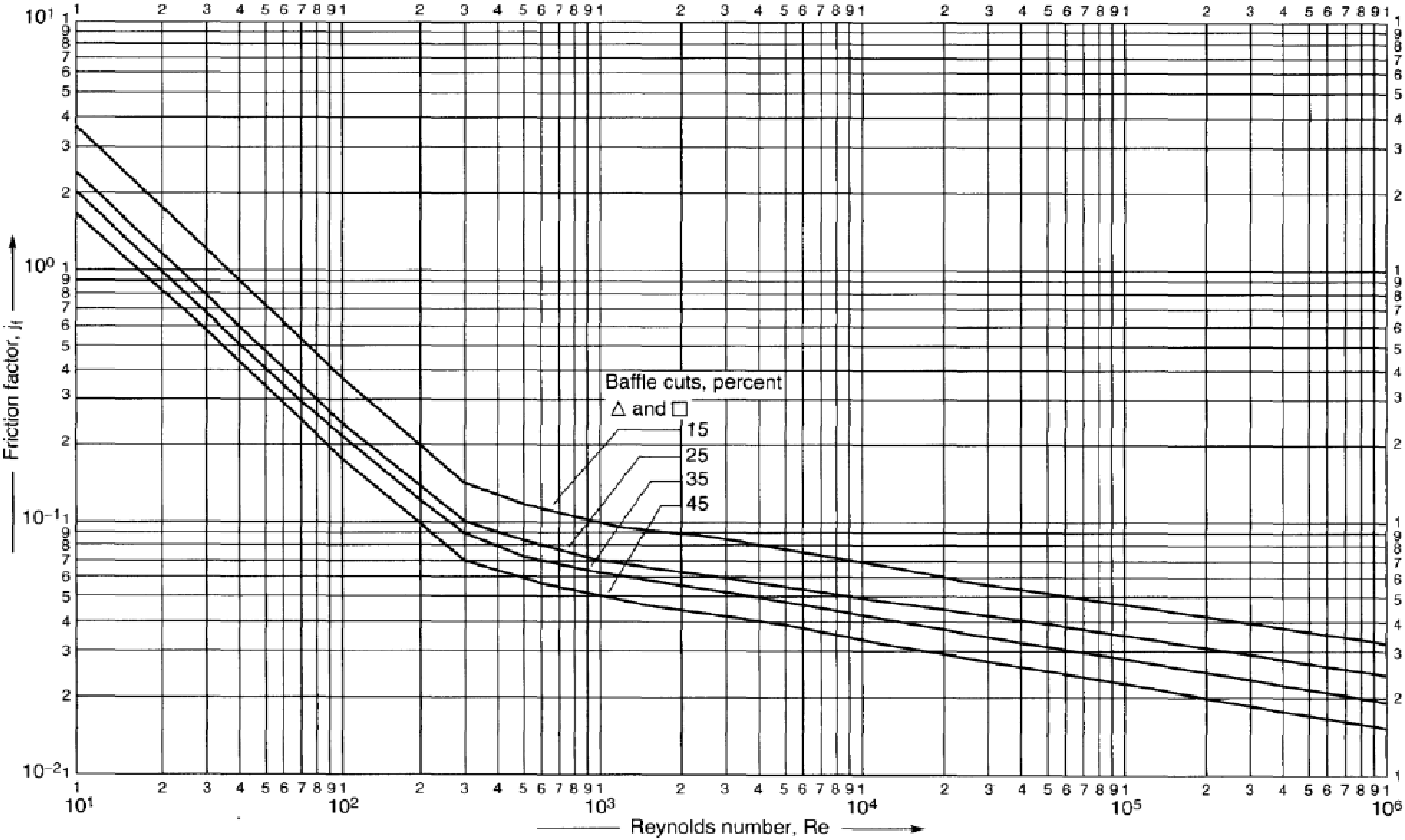


Figure4. Shell-Side friction factor, segmental baffles

Example: It is desired to methanol from 95°C to 40°C with flow rate 10000 Kg/hr. Sea water used as a coolant with temperature rise from 25°C to 40 °C. Design shell and tube heat exchanger using data below ;

- ***Methanol (Hot stream)***
- **Entering temp. = 95°C**
- **Leaving temp. = 40°C**
- **Specific heat = 2.84 kJ/kg °C**
- **Viscosity = 0.34 m Pa.s**
- **Density = 750 kg./m^3**
- **Thermal conductivity = 0.19 W/m K.**
- **Fouling coefficient = 2000 w/m^2 °C**

- ***Sea water (Cold stream)***
- **Entering temp. = 25°C**
- **Leaving temp. = 40°C**
- **Specific heat = 4.2 kJ/kg °C**
- **Viscosity = 0.8 m Pa.s**
- **Density = 900 kg./m^3**
- **Thermal conductivity = 0.58 W/m K.**
- **Fouling coefficient = 3000 w/m^2 °C**

Solution

Methanol 95 °C → 40 °C

Sea water 40 °C ← 25 °C

$\Delta T_1 = 95 - 40 = 55°C$ $\Delta T_2 = 40 - 25 = 15°C$

$$LMTD = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2)$$

$$LMTD = (55 - 15) / \ln(55/15)$$

$$= 31°C$$

Heat Duty $\quad Q = m\, Cp\, (T_{h1} - T_{h2})$

$= (100000 \text{ kg,/h})(2.84 \text{ kJ/kg °C})(95 - 40)\text{°C}$

$= \mathbf{15624000\ KJ/h} = \underline{\mathbf{4340\ \ KW}}$

Heat given by the hot stream = Heat taken by the cold stream

Water is heated from 25°C to 40°C Therefore,

Water flow rate = $Q\, /\, Cp \times (T_{c2} - T_{c1})$

$= 4340/(4.2 \times 15)$

$=68.9$ kg/s

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{95-40}{40-25} = 3.67 \qquad S = \frac{t_2 - t_1}{T_1 - t_1} = \frac{40-25}{95-25} = 0.21$$

From Figure for one shell pass; two tube 'passes $F_T = 0.85$

$$\Delta T_m = 31 * 0.85 = 26^o C$$

Select U= 600 w/m^2 °C from table

$$A = \frac{Q}{U \Delta T_m} = \frac{4340000}{600 \times 26} = 278 \text{ m}^2$$

Select material of construction as Cupro-Nickal
Thermal conductivity = 50 W/m K.
Pipe nominal size ¾ in x 20 ft
O.D = 20 mm
I.D = 16 mm

Area of one tube = 6x20xπ/1000 = 0.372 m^2

Using allowing thickness for tube sheet = 5 cm

$$L = \frac{20}{3.28} - 0.05 = 6m$$

Number of tubes = 278/0.372=738

Use 1.25 triangular pitch (Shell-Side fluid is clean)

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

K_1=0.249 n_1=2.207

$$D_b = 20\left(\frac{738}{0.249}\right)^{1/2.207} = 715mm$$

From Figure of Shell-to-bundle clearance, we find that the clearance = 64 mm

$$D_s = 715 + 64 = 779mm$$

Tube X-section area = $A = \frac{\pi}{4}d^2 = \frac{\pi}{4}x16^2$ = 201 mm^2

Number of tubes per pass = (738/2) = 369

Total flow area = 369 x 201 x 10^{-6} = 0.07416 m^2

Water mass flux = 68.9/0.07416 = 929 Kg/s. m^2

Water linear velocity =929/900 = 1.0323 m/s

Prandtl number, Pr = Cpμ/k

= (4.2)(10^3 1000)(0.8 x 10^{-3})/0.59

= 5.7

Reynolds number, Re = ρudi/μ

= 900 x 1.0323 x 16 x 10^{-3}

= 18576

$$\frac{L}{d_i} = \left(\frac{6\times10^3}{16}\right) = 375$$

From Figure 1 $j_h = 3.9\times 10^{-3}$

Neglect the effect of viscosity

$$h_i = \frac{0.59}{16\times10^{-3}}\times3.9\times10^{-3}\times18576\times5.7^{0.33} = 4744.46 w/m^2c$$

Select baffle spacing = 0.4

$$L_B = 0.4\times D_s = 0.4\times779 = 311.5\ mm = 312\ mm$$

Tube pitch = 1.25x20 = 25

$$A_s = \frac{(p_t - d_o)D_sL_B}{p_t} = \frac{(25-20)\times312\times779\times10^{-6}}{25} = 0.0485m^2$$

Mass flux = 100000/(36000 x 0.0485) = 512.18 Kg/s. m^2

Select baffle cut 45%

$$D_e = \frac{1.1}{d_o}\left[p_t^2 - 0.917 d_o^2 \right] = \frac{1.1}{20}\left[25^2 - 0.917\times 25^2 \right] = 14.4\times 10^{-3} m^2$$

Re = Gde/μ=512.12 x 14.4x10^{-3} x/(0.34x 10^{-3})

From Figure 3 j_h = 2.8 x 10^{-3}

$$h_s = \frac{0.19}{14.4\times 10^{-3}} \times 2.8\times 10^{-3} \times 124333.5 \times 5.1^{0.33} = 1539 w/m^2 c$$

$$\frac{1}{U} = \frac{d_o}{d_i}\frac{1}{h_i} + \frac{1}{h_o} + \frac{d_o \ln \frac{d_o}{d_i}}{2K_w} + \frac{d_o}{d_i}\frac{1}{h_{iD}} + \frac{1}{h_{oD}}$$

U = 635.1 w/m^2 °C Error% = (635.1-600)/635.1X100% = 5.52%

Pressure Drop

Tube side

$$\Delta P_t = n\left[8 j_f \left(\frac{L}{d_i}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14} + 2.5\right]\frac{\rho u_t^2}{2}$$

From Figure 2 $j_f = 4.1 \times 10^{-3}$

$$\Delta P_t = 2\left[8 \times 4.1 \times 10^{-3}\left(\frac{6 \times 10^3}{16}\right) + 2.5\right]\frac{900 \times 1.032^2}{2} = 14186.119\, N/m^2 = 2.05\, psi$$

Shell side

Linear velocity = G_s/ρ = 572/750 = 0.7629 m/s

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

From Figure 4 $J_f = 2.8 \times 10^{-2}$

$$\Delta P_s = 8\times 2.8\times 10^{-2}\left(\frac{779}{14.4}\right)\left(\frac{6\times 10^3}{312}\right)\left(\frac{750\times 0.7629^2}{2}\right) = 50862.034\,N/m^2 = 7.317\,psi$$