Process Heat Transfer

Lec 11: Heat Exchanger Design

classification of heat exchangers, overall heat transfer coefficient in HEX, HEX design: Kern method, NTU analysis, compact HEX

Dr. Zayed Al-Hamamre

Introduction

- Heat exchangers are devices that facilitate the *exchange of heat* between *two fluids* that are at different temperatures while keeping them from mixing with each other.
- Heat exchangers are commonly used in practice in a wide range of applications, from heating and air-conditioning systems in a household, to chemical processing and power production in large plants.
- 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.

- Heat transfer in a heat exchanger usually involves *convection* in each fluid and *conduction* through the wall separating the two fluids

The general equation for heat transfer across a surface is:

$$Q = UA\Delta T_m$$

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, °C.

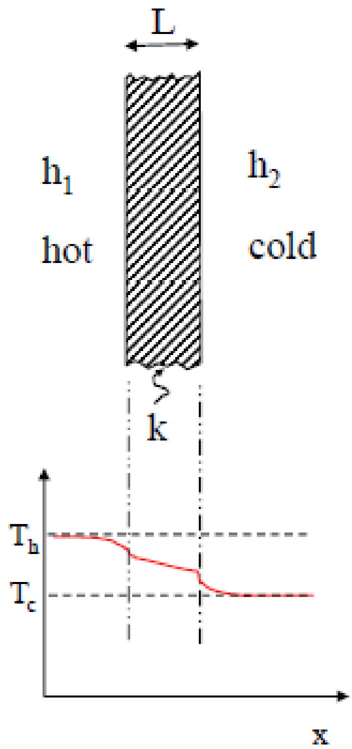
- For a flat plate, overall resistance is the sum of the individual resistances

$$\frac{Q}{A}\left(\frac{1}{h_1}+\frac{L}{k}+\frac{1}{h_2}\right) = T_h - T_c$$

- Hence overall heat transfer coefficient, U is given by

$$Q = UA(T_h - T_c)$$

$$\frac{1}{U} = \frac{1}{h_1}+\frac{L}{k}+\frac{1}{h_2}$$



➢ For heat exchange across a typical heat exchanger tube the relationship between the overall coefficient and the individual coefficients, which are the reciprocals of the individual resistances, is given by:

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

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

➢ The magnitude of the individual coefficients will depend on the
- nature of the heat transfer process (conduction, convection, condensation, boiling or radiation),
- physical properties of the fluids, o
- fluid flow-rates,
- physical arrangement of the heat-transfer surface.

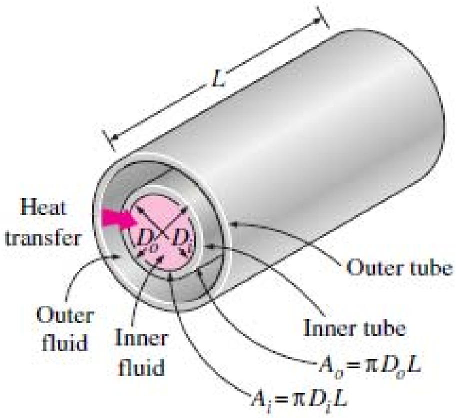
➢ As the physical layout of the exchanger cannot be determined until the area is known the design of an exchanger is of necessity a trial and error procedure.

Tubular Heat Exchangers...........................
- Double-Pipe Heat Exchangers......
- Shell-and-Tube Heat Exchangers.
- Spiral-Tube Heat Exchangers........

Plate Heat Exchangers...............................
- Gasketed Plate Heat Exchangers.
- Spiral Plate Heat Exchangers.......
- Lamella Heat Exchangers.............

Extended Surface Heat Exchangers...........
- Plate-Fin Heat Exchanger...............
- Tubular-Fin Heat Exchangers.......

Cylindrical Geometry (Tubes)



$$\dot{Q} = \frac{\Delta T}{R} = UA\,\Delta T = U_i A_i\,\Delta T = U_o A_o\,\Delta T$$

$$R = R_{\text{total}} = R_i + R_{\text{wall}} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{1}{h_o A_o}$$

Add terms for fouling:

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o}$$

Chemical Engineering Department | University of Jordan | Amman 11942, Jordan
Tel. +962 6 535 5000 | 22888

Overall heat transfer coefficient

TABLE 13–1

Representative values of the overall heat transfer coefficients in heat exchangers

Type of heat exchanger	U, W/m^2 · °C*
Water-to-water	850–1700
Water-to-oil	100–350
Water-to-gasoline or kerosene	300–1000
Feedwater heaters	1000–8500
Steam-to-light fuel oil	200–400
Steam-to-heavy fuel oil	50–200
Steam condenser	1000–6000
Freon condenser (water cooled)	300–1000
Ammonia condenser (water cooled)	800–1400
Alcohol condensers (water cooled)	250–700
Gas-to-gas	10–40
Water-to-air in finned tubes (water in tubes)	30–60†
	400–850†
Steam-to-air in finned tubes (steam in tubes)	30–300†
	400–4000‡

*Multiply the listed values by 0.176 to convert them to Btu/h · ft^2 · °F.

†Based on air-side surface area.

‡Based on water- or steam-side surface area.

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
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

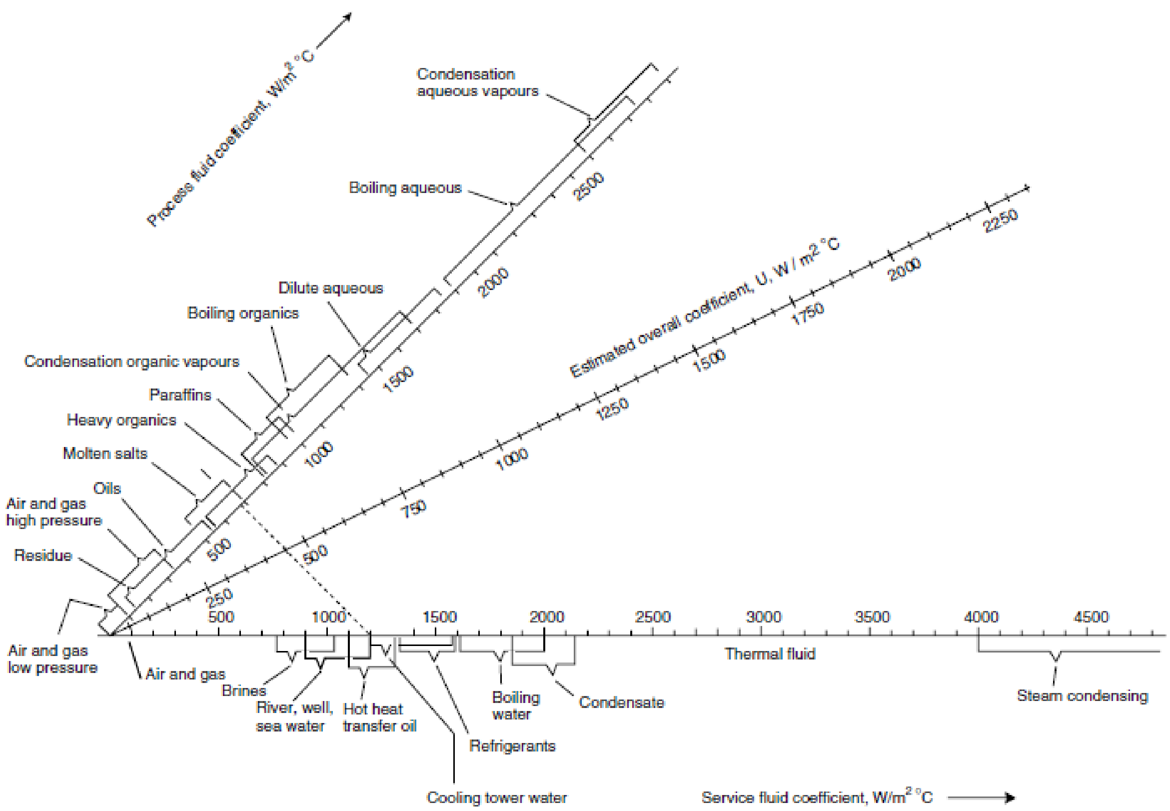


Figure 12.1. Overall coefficients (join process side duty to service side and read U from centre scale)

Overall heat transfer coefficient

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

TABLE 13–2

Representative fouling factors (thermal resistance due to fouling for a unit surface area)

(*Source:* Tubular Exchange Manufacturers Association.)

Fluid	R_f, $m^2 \cdot °C/W$
Distilled water, sea water, river water, boiler feedwater:	
Below 50°C	0.0001
Above 50°C	0.0002
Fuel oil	0.0009
Steam (oil-free)	0.0001
Refrigerants (liquid)	0.0002
Refrigerants (vapor)	0.0004
Alcohol vapors	0.0001
Air	0.0004

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

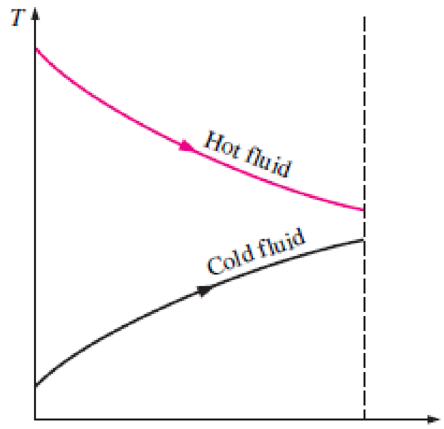
Heat Exchanger Analysis

assumptions.

1. The heat exchanger is insulated from its surroundings, in which case the only heat exchange is between the hot and cold fluids.
2. Axial conduction along the tubes is negligible.
3. Potential and kinetic energy changes are negligible.
4. The fluid specific heats are constant.
5. The overall heat transfer coefficient is constant.

Parallel flow Heat Transfer

- both the hot and cold fluids enter the heat exchanger at the same end and move in the *same* direction

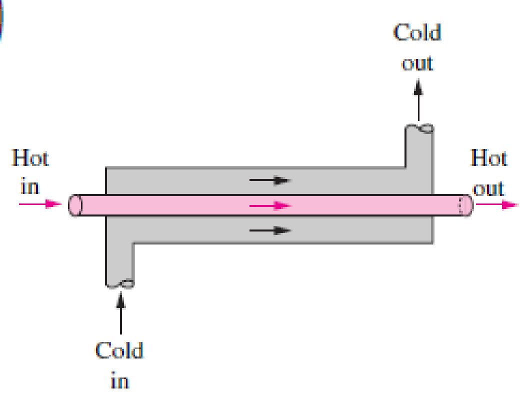


$$\delta\dot{Q} = -\dot{m}_h C_{ph}\, dT_h \qquad \delta\dot{Q} = \dot{m}_c C_{pc}\, dT_c$$

$$dT_h - dT_c = d(T_h - T_c) = -\delta\dot{Q}\left(\frac{1}{\dot{m}_h C_{ph}} + \frac{1}{\dot{m}_c C_{pc}}\right)$$

$$\delta\dot{Q} = U(T_h - T_c)\, dA_s$$

$$\Rightarrow \quad \frac{d(T_h - T_c)}{T_h - T_c} = -U\, dA_s\left(\frac{1}{\dot{m}_h C_{ph}} + \frac{1}{\dot{m}_c C_{pc}}\right)$$



Integrating from the inlet of the heat exchanger to its outlet,

$$\Rightarrow \quad \ln\frac{T_{h,\text{out}} - T_{c,\text{out}}}{T_{h,\text{in}} - T_{c,\text{in}}} = -UA_s\left(\frac{1}{\dot{m}_h C_{ph}} + \frac{1}{\dot{m}_c C_{pc}}\right)$$

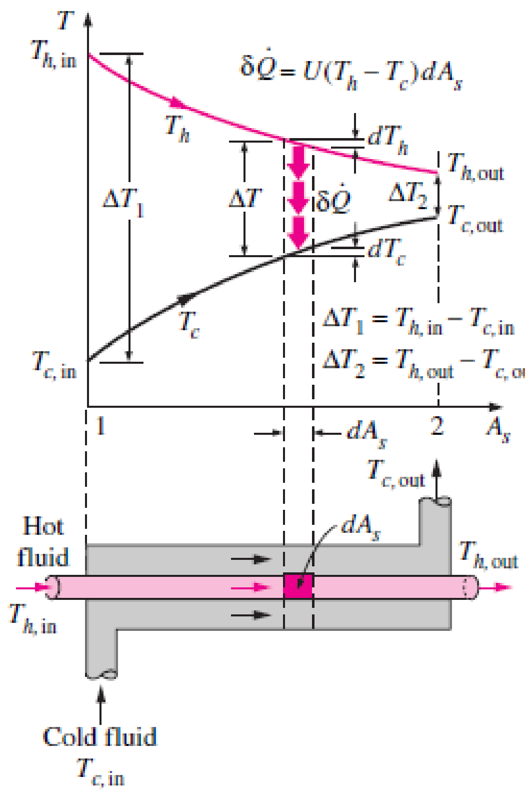
But

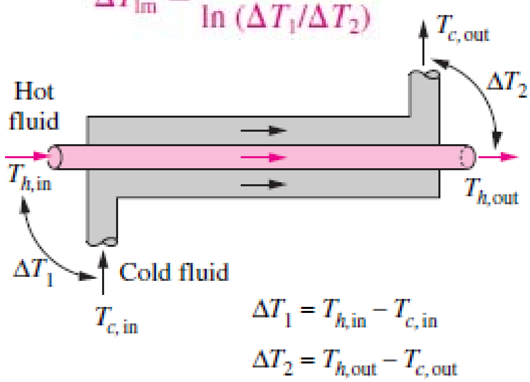
$$\dot{Q} = \dot{m}_c C_{pc}(T_{c,\text{out}} - T_{c,\text{in}}) \qquad \dot{Q} = \dot{m}_h C_{ph}(T_{h,\text{in}} - T_{h,\text{out}})$$

$$\Rightarrow \quad \ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -UA\left(\frac{T_{h,i} - T_{h,o}}{q} + \frac{T_{c,o} - T_{c,i}}{q}\right)$$

$$\Rightarrow \quad \dot{Q} = UA_s\, \Delta T_{\text{lm}}$$

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

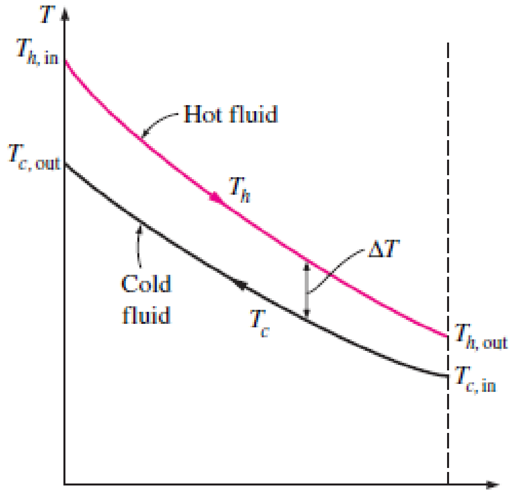


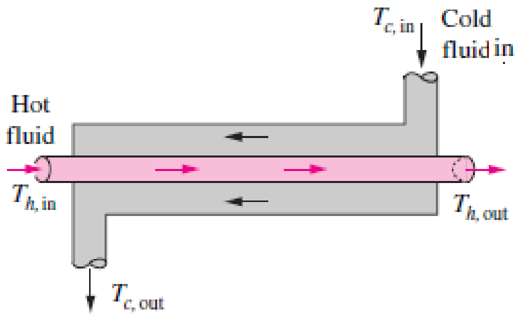


(*a*) Parallel-flow heat exchangers

Counter Current Heat Transfer

- the hot and cold fluids enter the heat exchanger at opposite ends and flow in *opposite* directions





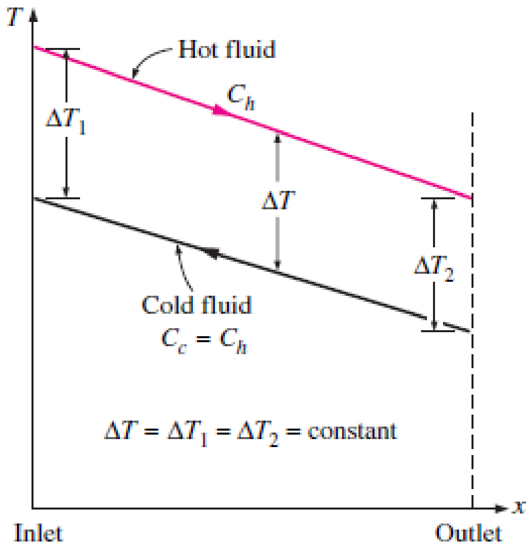
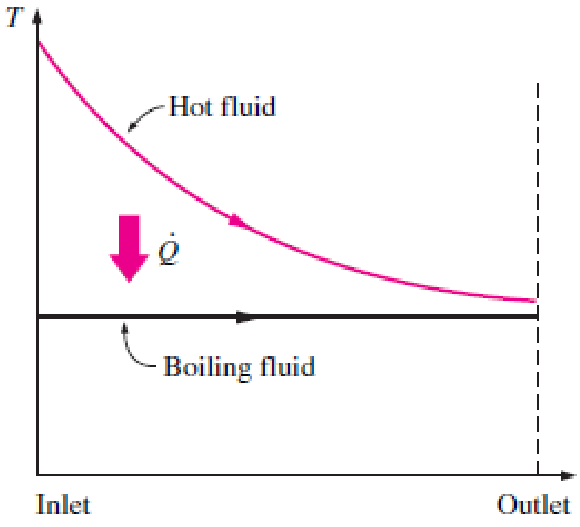
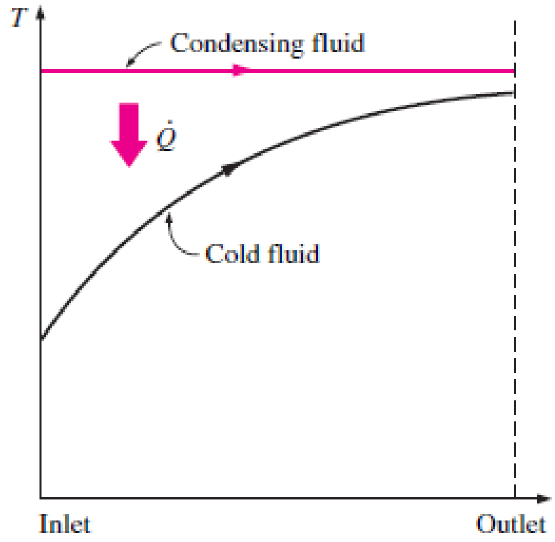


FIGURE 13–12
Two fluids that have the same mass flow rate and the same specific heat experience the same temperature change in a well-insulated heat exchanger.



(b) Boiler ($C_c \to \infty$)



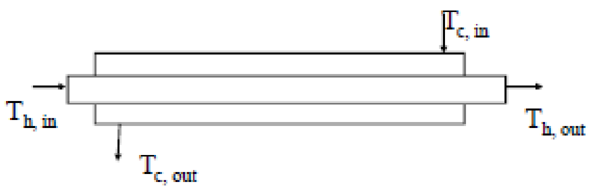
(a) Condenser ($C_h \to \infty$)

$$C_h = \dot{m}_h C_{ph} \quad \text{and} \quad C_c = \dot{m}_c C_{pc}$$

Counter Current Heat Transfer

Hot End

$\Delta T_h = T_{h,in} - T_{c,out}$



Cold End

$\Delta T_c = T_{h,out} - T_{c,in}$

- $Q = U\ A\ \Delta T_m$
- For perfect counter current flow, ΔT_m is the log mean temperature difference:

$$\Delta T_{lm} = \frac{\left(\left(T_{h,in} - T_{c,out}\right) - \left(T_{h,out} - T_{c,in}\right)\right)}{\ln\left(\dfrac{\left(T_{h,in} - T_{c,out}\right)}{\left(T_{h,out} - T_{c,in}\right)}\right)}$$

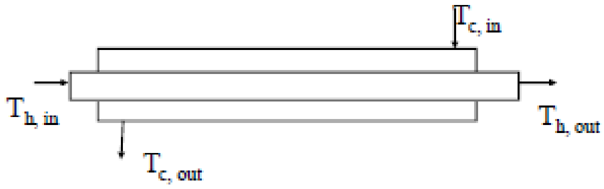
- Easiest to remember as:

$$\Delta T_{lm} = \frac{\left(\Delta T_h - \Delta T_c\right)}{\ln\left(\dfrac{\Delta T_h}{\Delta T_c}\right)}$$

Counter Current Heat Transfer

Hot End

$\Delta T_h = T_{h,in} - T_{c,out}$



Cold End

$\Delta T_c = T_{h,out} - T_{c,in}$

$$\Delta T_{lm} = \frac{(\Delta T_h - \Delta T_c)}{\ln\left(\frac{\Delta T_h}{\Delta T_c}\right)}$$

A useful shortcut to know:

$\Delta T_h/\Delta T_c$	1.0	1.5	2.0	3.0	4.0
$\Delta T_{lm} / \Delta T_c$	1.0	1.2	1.4	1.8	2.2
$\Delta T_{geom\ mean} = \sqrt{(\Delta T_c \times \Delta T_h)}$					
$\Delta T_{geom\ mean} / \Delta T_c$	1.0	1.22	1.41	1.73	2.0

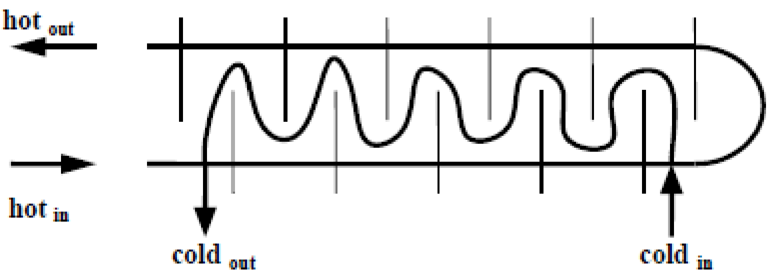
So $\Delta T_{lm} \approx \Delta T_{geom\ mean}$ if ratio is 3 or less!

Multi-pass and Cross-Flow Heat Exchangers

- Most real heat exchangers do not have pure counter-current flow
- We apply a correction factor for this in design

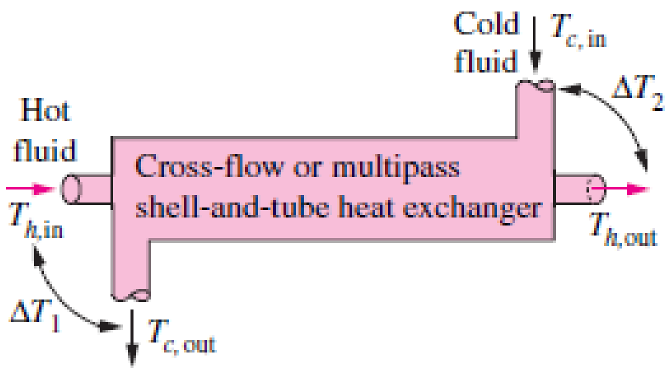
$$Q = U\,A\,F\,\Delta T_{lm} \qquad \Delta T_{lm} = F\,\Delta T_{lm,\,CF}$$

- F is usually > 0.8 unless the design is poor (see later)
- F is sometimes called F_t $\qquad F = \phi(P, R, \text{flow arrangement})$



where F is the **correction factor,** which depends on the *geometry* of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. The $\Delta T_{lm,\,CF}$ is the log mean temperature difference for the case of a *counter-flow* heat exchanger with the same inlet and outlet temperatures

The correction factor is less than unity for a cross-flow and multipass shell-and-tube heat exchanger. That is, $F \leq 1$. The limiting value of $F = 1$ corresponds to the counter-flow heat exchanger. Thus, the correction factor F for a heat exchanger is *a measure of deviation of the* ΔT_{lm} *from the corresponding values for the counter-flow case.*



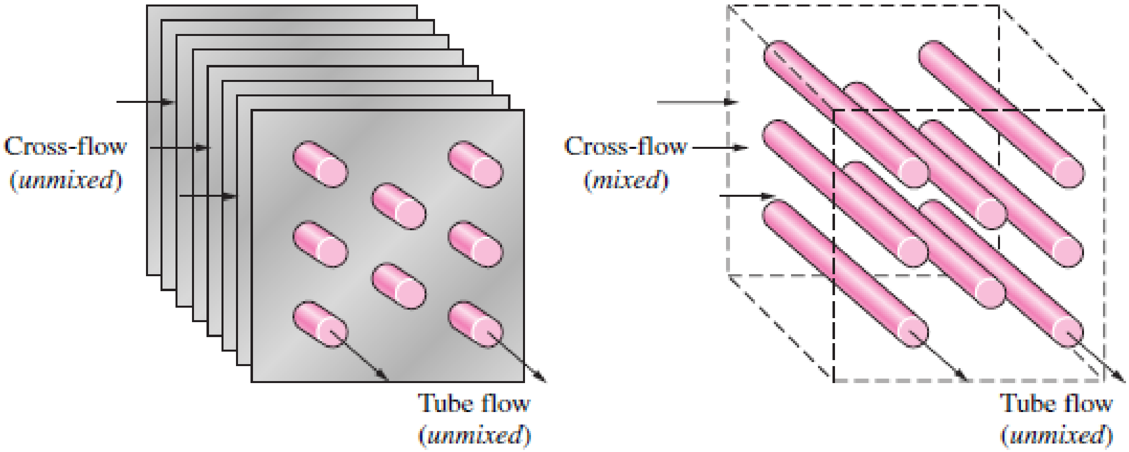
Heat transfer rate:

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$

where
$$\Delta T_{lm,\,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$

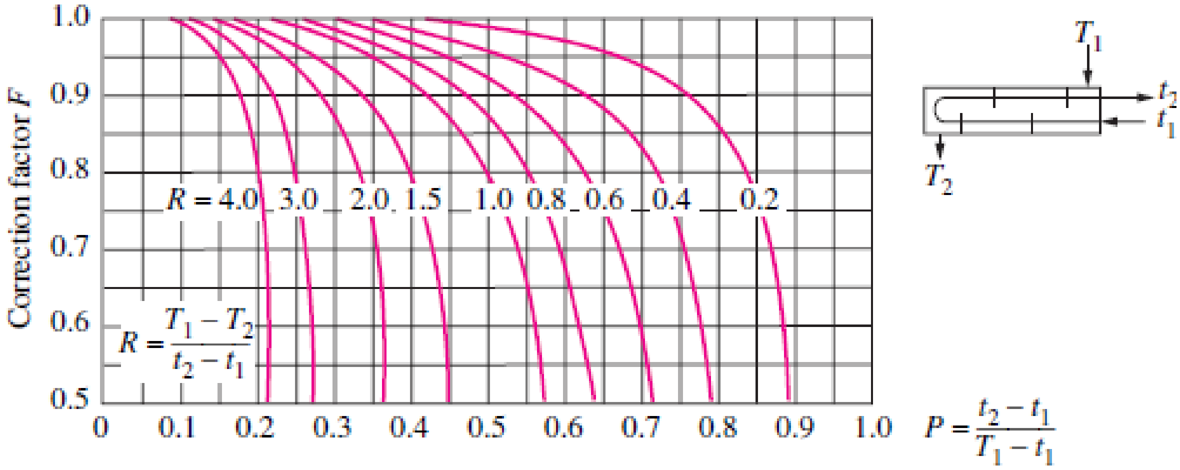
Cross-flow heat transfer



(a) Both fluids unmixed

(b) One fluid mixed, one fluid unmixed

Correction Factor

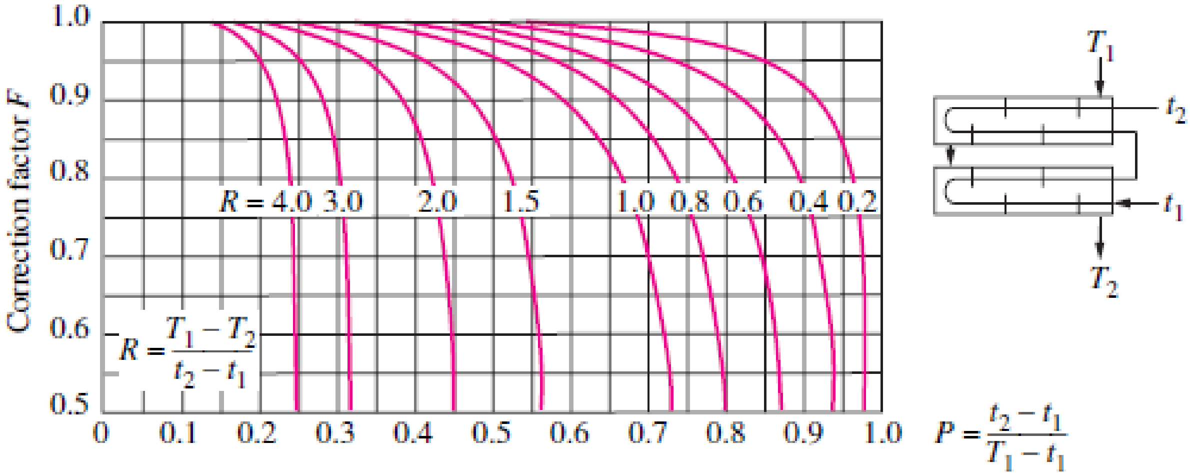


(a) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}C_p)_{\text{tube side}}}{(\dot{m}C_p)_{\text{shell side}}}$$

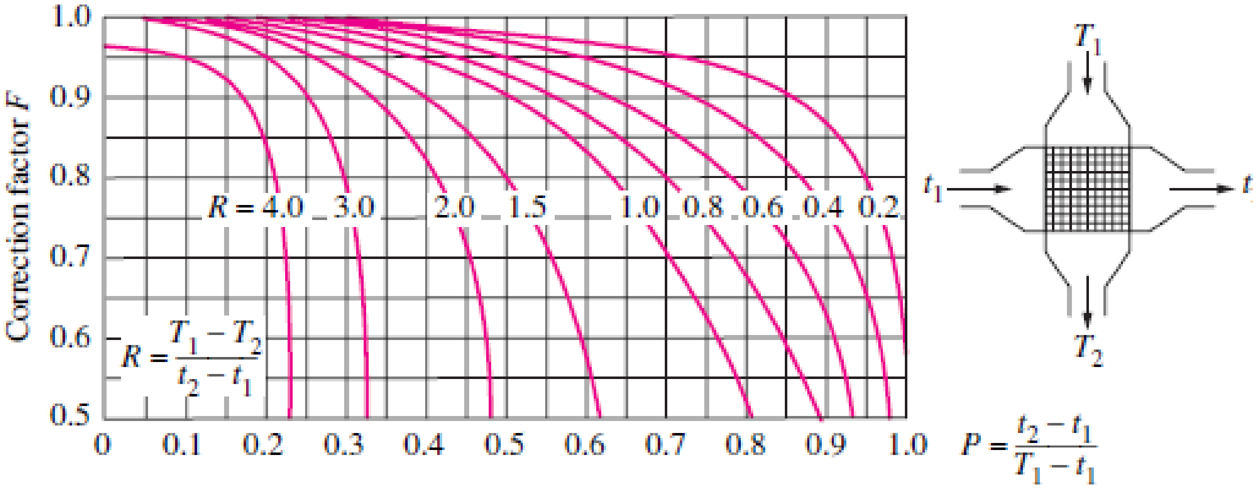
where the subscripts 1 and 2 represent the *inlet* and *outlet*, respectively.

Correction Factor



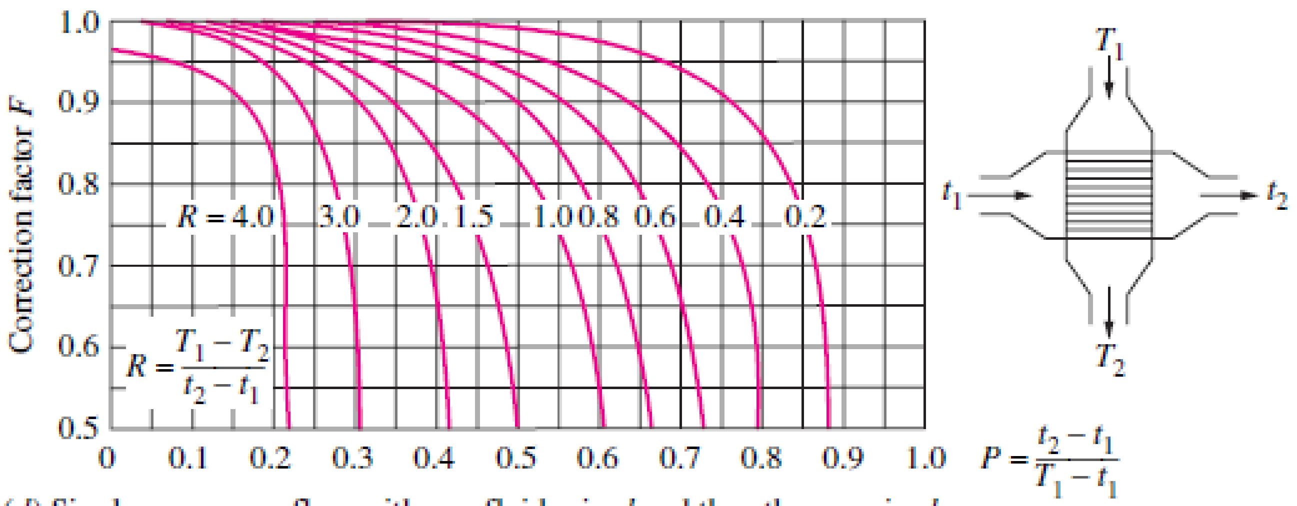
(b) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes

Correction Factor



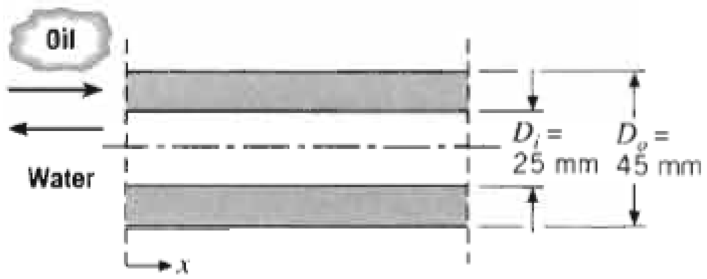
(c) Single-pass cross-flow with both fluids *unmixed*

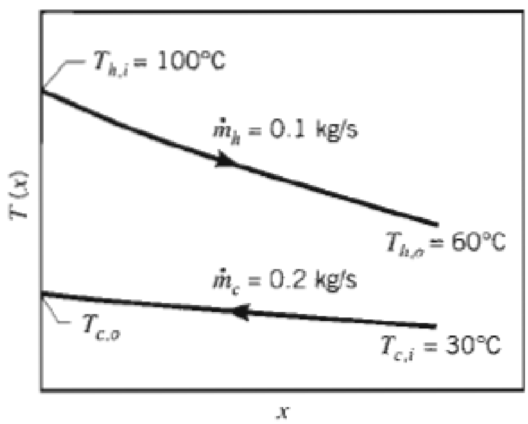
Correction Factor



(*d*) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

A counterflow, concentric tube heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube (D_i = 25 mm) is 0.2 kg/s, while the flow rate of oil through the outer annulus (D_o = 45 mm) is 0.1 kg/s. The oil and water enter at temperatures of 100 and 30°C, respectively. How long must the tube be made if the outlet temperature of the oil is to be 60°C?





Known: Fluid flow rates and inlet temperatures for a counterflow, concentric tube heat exchanger of prescribed inner and outer diameter.

Find: Tube length to achieve a desired hot fluid outlet temperature.

A 2-shell passes and 4-tube passes heat exchanger is used to heat glycerin from 20°C to 50°C by hot water, which enters the thin-walled 2-cm-diameter tubes at 80°C and leaves at 40°C (Fig. 13–21). The total length of the tubes in the heat exchanger is 60 m. The convection heat transfer coefficient is 25 W/m² · °C on the glycerin (shell) side and 160 W/m² · °C on the water (tube) side. Determine the rate of heat transfer in the heat exchanger (*a*) before any fouling occurs and (*b*) after fouling with a fouling factor of 0.0006 m² · °C/W occurs on the outer surfaces of the tubes.

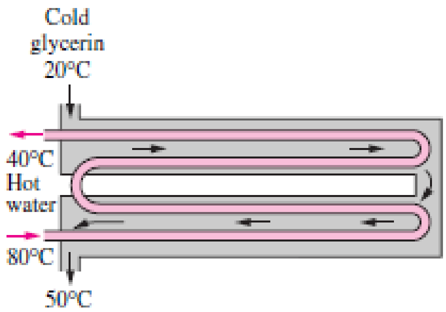
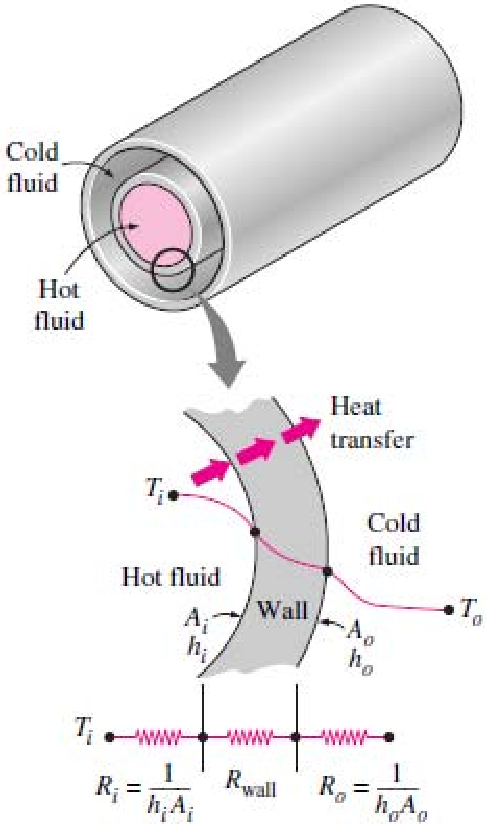


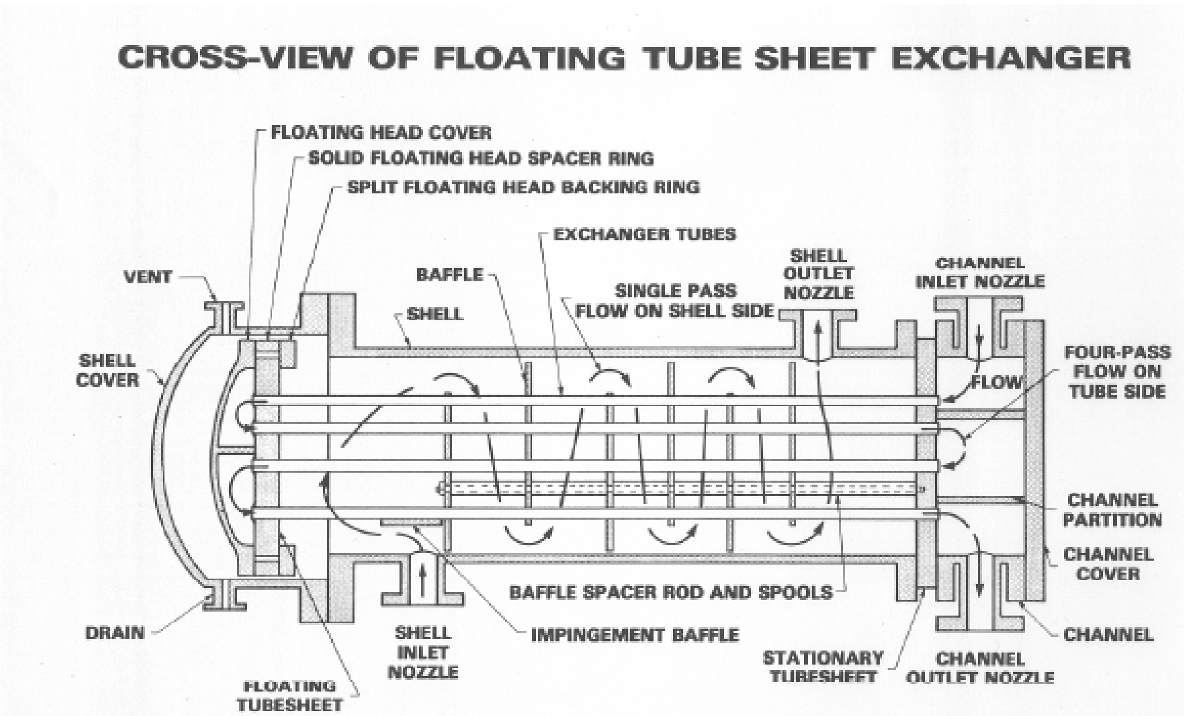
FIGURE 13–21
Schematic for Example 13–5.

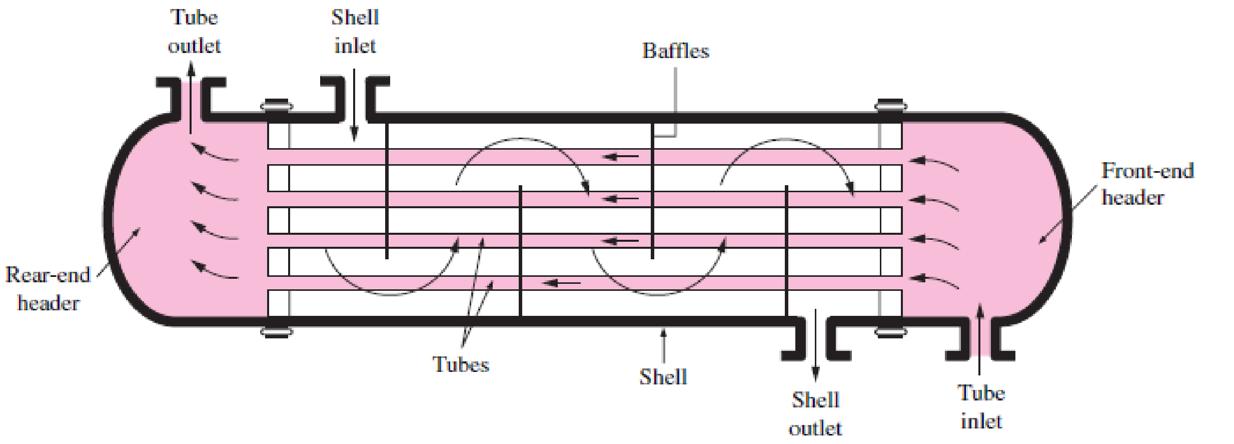
Tubular Exchangers

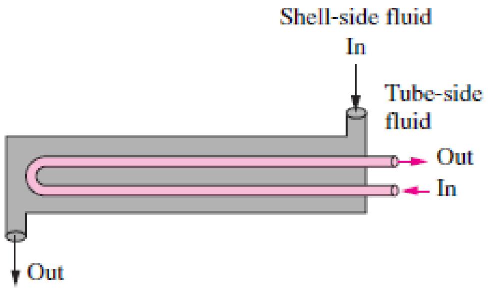
Double-pipe heat exchanger



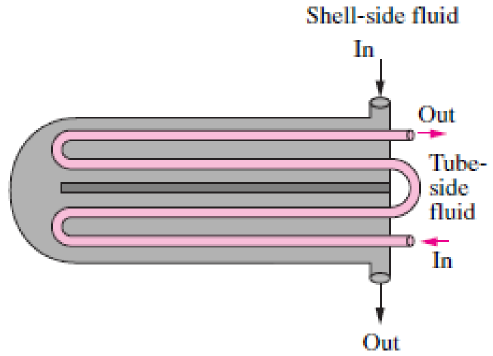
Shell and Tube Heat Exchangers



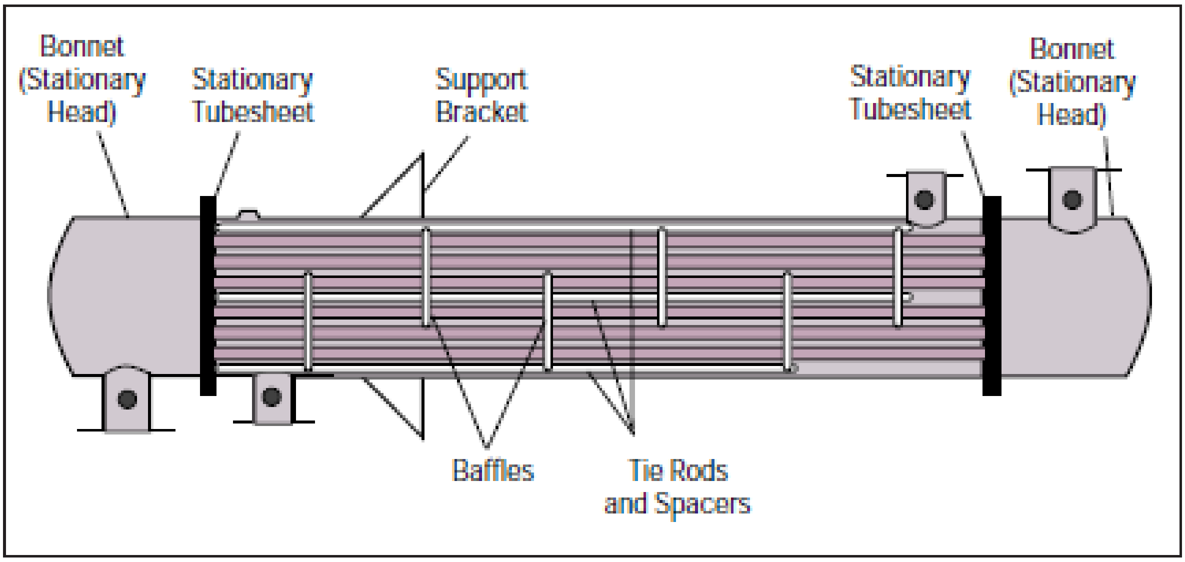




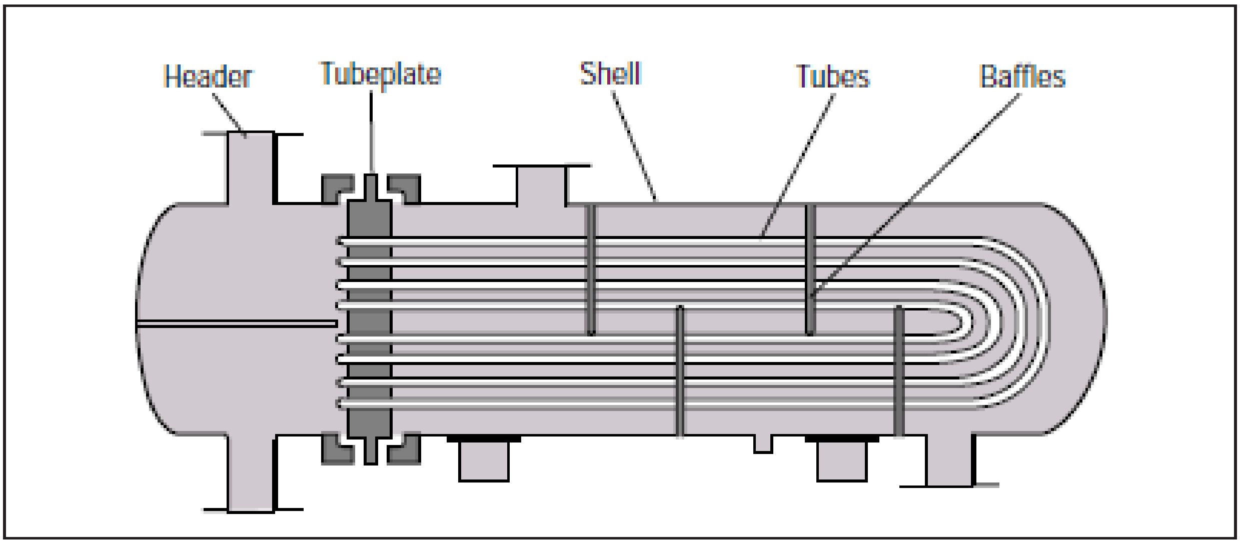
(*a*) One-shell pass and two-tube passes



(*b*) Two-shell passes and four-tube passes



Fixed-tubesheet heat exchanger.



U-tube heat exchanger.

Advantages

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.

Types of shell and tube exchanger

1. Fixed-tube plate
2. U-tube
3. Internal floating head without clamp ring
4. Internal floating head with clamp ring
5. External floating head, packed gland
6. Kettle reboiler with U-tube bundle

Shell and Tube Exchangers




Source: Riggins Company:
www.rigginscompany.com

Welding the shell

Baffle assembly

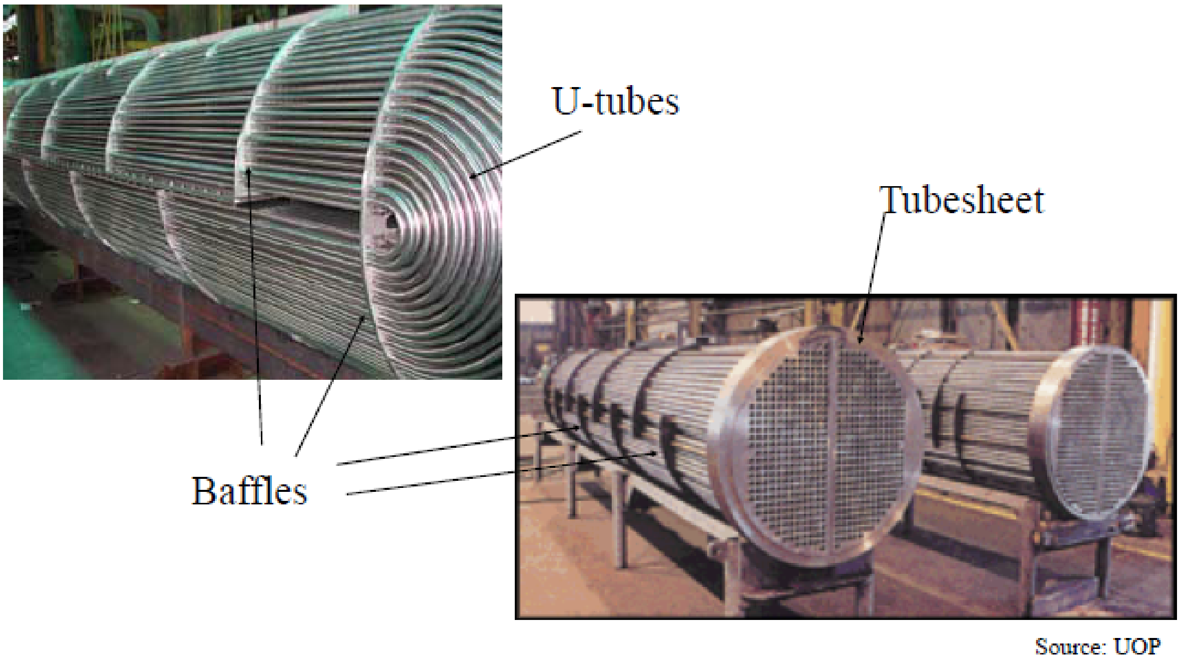
Inserting tubes

Tubesheet

Final product

Source: Bos-Hatten Inc.: www.Bos-Hatten.com

Tube Bundles



Fixed-tube plate

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
- the use of this type is limited to temperature differences up to about 80^{o}C

The U-tube (U-bundle) type

- requires only one tube sheet and
- It 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.

Exchangers with an internal floating head,

- They are more versatile than fixed head and U-tube exchangers.
- They are suitable for high-temperature differentials
- 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.
- There will always be a danger of leakage occurring from the internal flanges in these floating head designs

External floating head designs,

- 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.

Components of STHEs

It is essential for the designer to have a good working knowledge of the mechanical features of STHEs and how they influence thermal design. The principal components of an STHE are:

- shell;
- shell cover;
- tubes;
- channel;
- channel cover;
- tubesheet;
- baffles; and
- nozzles.

Other components include tie-rods and spacers, pass partition plates, impingement plate, longitudinal baffle, sealing strips, supports, and foundation.

The Standards of the Tubular Exchanger Manufacturers Association (TEMA) *(1)* describe these various components in detail.

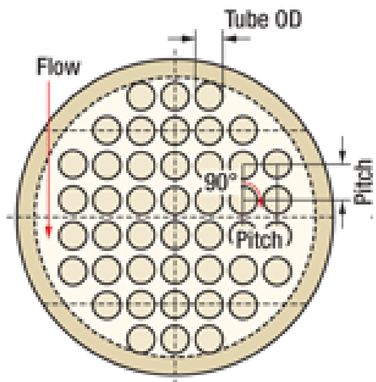
Number of Tubes

- The flow rate inside the tube is a function of the density of the fluid, the velocity of the fluid, cross-sectional flow area of the tube, and the number of tubes.

$$\dot{m}_{tube} = \rho_t u_t A_c N_t$$

By using above Eq. and replacing A_c by $\pi d_i^2/4$, number of tubes can be calculated as

$$N_t = \frac{4\times \dot{m}_{tube}}{\rho_t u_t \pi d_i^2}$$

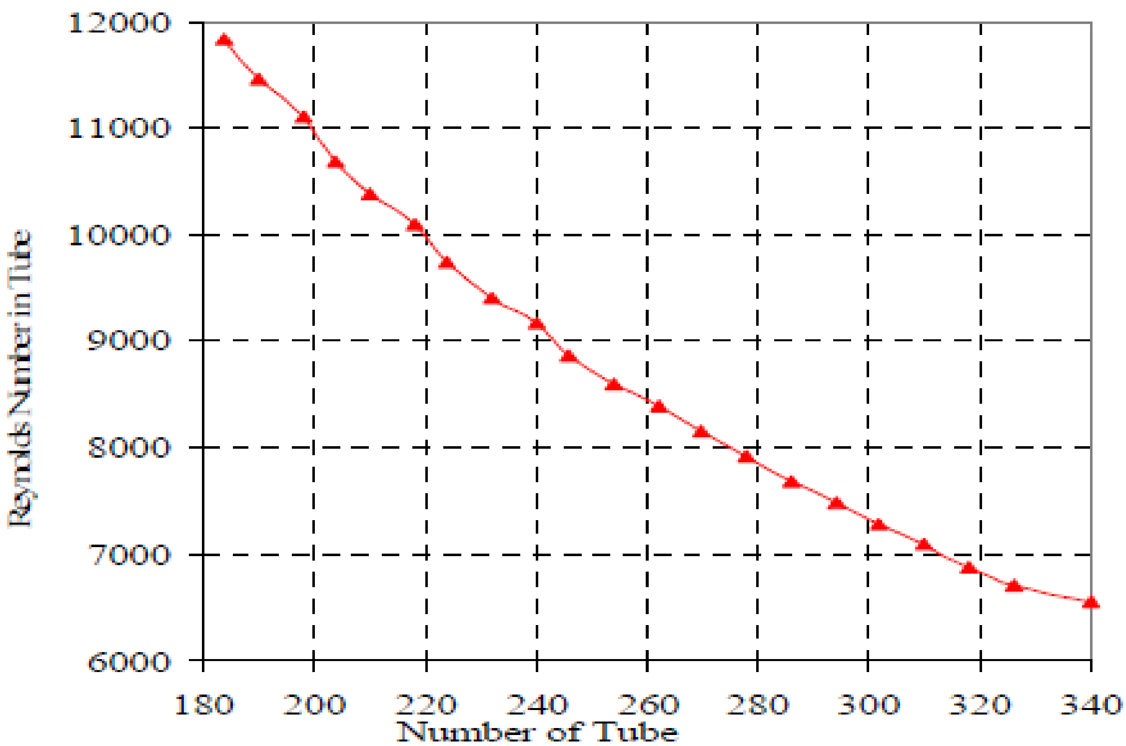


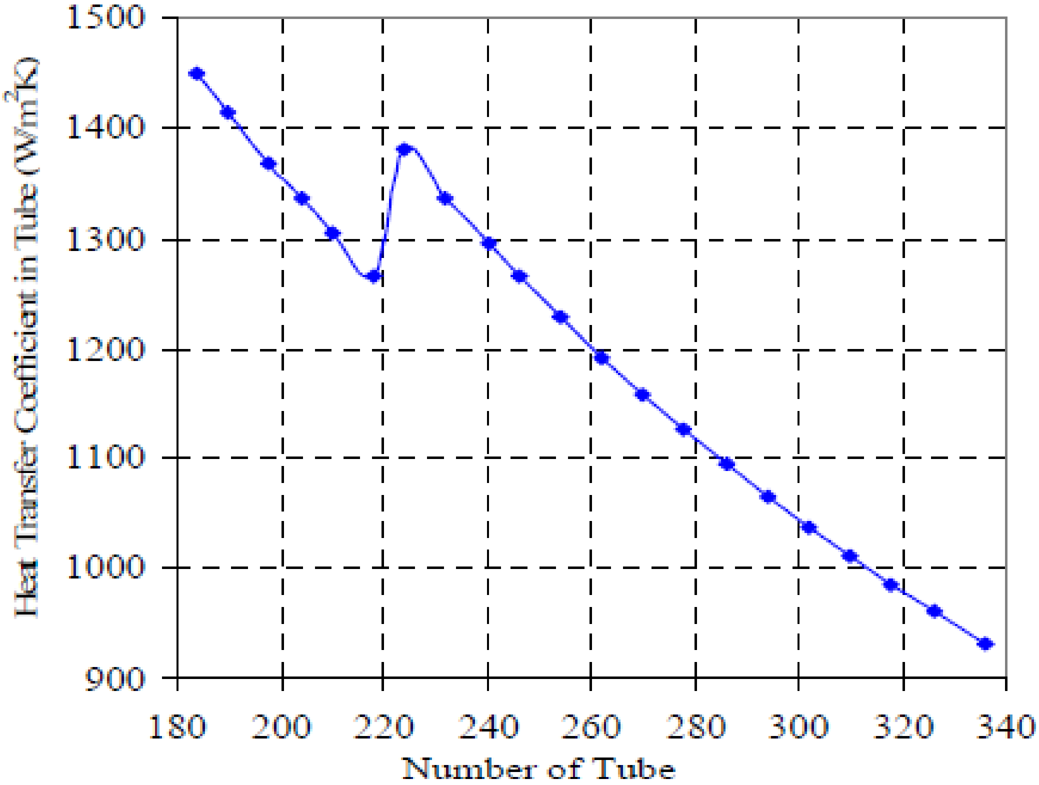
where d_i is the tube inside diameter.

Tubes in Shell and Tube Hx

- The number and size of tubes in an exchanger depends on the
  - Fluid flow rates
  - Available pressure drop.
- The number and size of tubes is selected such that the
  - **Tube side velocity** for water and similar liquids ranges from
  - 0.9 to 2.4 m/s.
  - **Shell-side velocity** from 0.6 to 1.5 m/s.
  - The lower velocity limit corresponds to limiting the **fouling**, and the
  - upper velocity limit corresponds to limiting the rate of **erosion**.
- When sand and silt are present, the velocity is kept high enough to prevent settling.

Number of Tubes Vs Reynolds Number





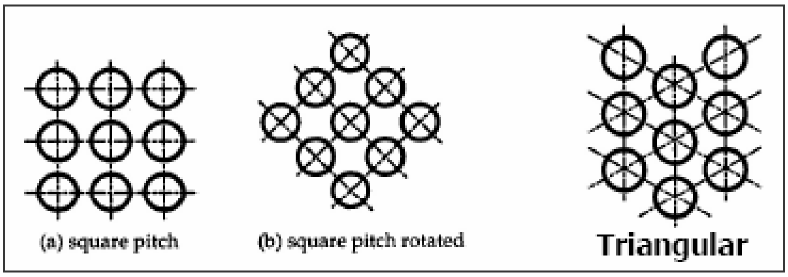
Tube Layout

- Triangular pitch (30^o layout) is better for heat transfer and surface area per unit length (greatest tube density.)
- Square pitch (45 & 90 layouts) is needed for mechanical cleaning.
- Note that the 30°,45° and 60° are staggered, and 90° is in line.
- For the identical tube pitch and flow rates, the tube layouts in decreasing order of shell-side heat transfer coefficient and pressure drop are: 30°,45°,60°, 90°.
- **The 90° layout** will have the lowest heat transfer coefficient and the lowest pressure drop.
- **The square pitch** (90° or 45°) is used when jet or mechanical cleaning is necessary on the shell side.

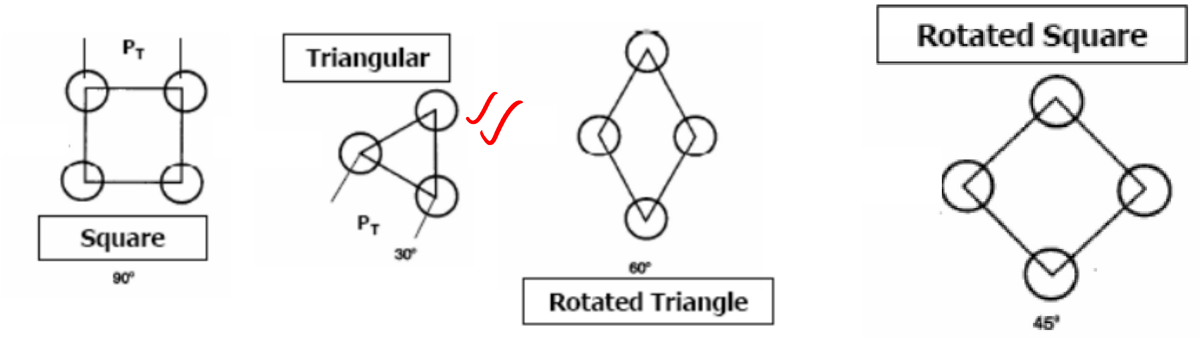
- The square pitch is generally not used in the fixed header sheet design because cleaning is not feasible.

Tube Layout & Flow Scales

There are four tube layout patterns, as shown in Figure 6: triangular (30°), rotated triangular (60°), square (90°), and rotated square (45°).



A Real Use of Wetted Perimeter !



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.

Steel tubes for heat exchangers are covered by BS 3606

- The wall thickness of heat exchanger tubes is standardized in terms of Birmingham Wire Gage BWG of the tube.
- Small tube diameters (8 to 15mm) are preferred for greater area to volume density but are limited for the purposes of cleaning.
- Large tube diameters are often required for condensers and boilers.

Chemical Engineering Department | University of Jordan | Amman 11942, Jordan
Tel. +962 6 535 5000 | 22888

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

Tube Outside Diameter

- The most common plain tube sizes have 15.88,19.05, and 25.40 mm (5/8, ¾, 1 inche) tube outside diameters.
- From the heat transfer viewpoint, smaller-diameter tubes yield higher heat transfer coefficients and result in a more compact exchanger.

- However, larger-diameter tubes are easier to clean and more rugged.
- The foregoing common sizes represent a compromise.
  - For mechanical cleaning, the smallest practical size is 19.05 mm.
  - For chemical cleaning, smaller sizes can be used provided that the tubes never plug completely.

Tube Length

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.

Tube Length

- Tube length affects the cost and operation of heat exchangers.
  - Longer the tube length (for any given surface area),
    - Fewer tubes are needed, requiring less complicated header plate with fewer holes drilled
    - Shell diameter decreases resulting in lower cost
- Typically tubes are employed in 8, 12, 15, and 20 foot lengths. Mechanical cleaning is limited to tubes 20 ft and shorter, although standard exchangers can be built with tubes up to 40 ft.
- There are, like with anything limits of how long the tubes can be.
  - **Shell-diameter-to-tube-length ratio** should be within limits of 1/5 to 1/15
- Maximum tube length is dictated by
  - Architectural layouts
  - Transportation (to about 30m.)
    - The diameter of the two booster rockets is dictated by the smallest highway tunnel size between the location of manufacturer and Florida. Scientific hah!

Tube Passes

- **A pass** is when liquid flows all the way across from one end to the other of the exchanger. We will count *shell passes* and *tube passes*.
  - An exchanger with one shell pass and two tube passes is a 1-2 exchanger. Almost always, the tube passes will be in multiples of two (1-2, 1-4, 2-4, etc.)
  - Odd numbers of tube passes have more complicated mechanical stresses, etc. An exception: 1-1 exchangers are sometimes used for vaporizers and condensers.
- A large number of tube passes are used to increase the tube side fluid velocity and heat transfer coefficient and minimize fouling.
  - This can only be done when there is enough pumping power since the increased velocity and additional turns increases the pressure drop significantly.

- The number of tube passes depends on the available pressure drop.
  - Higher velocities in the tube result in higher heat transfer coefficients, at the expense of increased pressure drop.
- Therefore, if a higher pressure drop is acceptable, it is desirable to have fewer but longer tubes (reduced flow area and increased flow length).
  - Long tubes are accommodated in a short shell exchanger by multiple tube passes.
- The number of tube passes in a shell generally range from 1 to 10
  - The standard design has one, two, or four tube passes.
  - An odd number of passes is uncommon and may result in mechanical and thermal problems in fabrication and operation.

Tube Pitch

- **The selection of tube pitch** is a compromise between a
  - **Close pitch** (small values of P_t/d_o) for increased shell-side heat transfer and surface compactness, and an
  - **Open pitch** (large values of P_t/ d_o) for decreased shell-side plugging and ease in shell-side cleaning.

- Tube pitch P_t is chosen so that the **pitch ratio** is $1.25 < P_T/d_o < 1.5$.
- When the tubes are to close to each other (P_T/d_o less than 1.25), the header plate (tube sheet) becomes to weak for proper rolling of the tubes and cause leaky joints.

Tube Pitch

- Tube layout and tube locations are standardized for industrial heat exchangers.
- However, these are general rules of thumb and can be "violated" for custom heat exchanger designs.

$$\boxed{\frac{P_T}{d_o} = 1.33}$$

Tube pitch is defined as the shortest distance between two adjacent tubes.

For a triangular pattern, TEMA specifies a minimum tube pitch of 1.25 times the tube O.D. Thus, a 25-mm tube pitch is usually employed for 20-mm O.D. tubes.

Designers prefer to employ the minimum recommended tube pitch, because it leads to the smallest shell diameter for a given number of tubes. However, in exceptional circumstances, the tube pitch may be increased to a higher value, for example, to reduce shellside pressure drop. This is particularly true in the case of a cross-flow shell.

For square patterns, TEMA additionally recommends a minimum cleaning lane of ¼ in. (or 6 mm) between adjacent tubes. Thus, the minimum tube pitch for square patterns is either 1.25 times the tube O.D. or the tube O.D. plus 6 mm, whichever is larger. For example, 20-mm tubes should be laid on a 26-mm (20 mm + 6 mm) square pitch, but 25-mm tubes should be laid on a 31.25-mm (25 mm × 1.25) square pitch.

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.

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

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

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

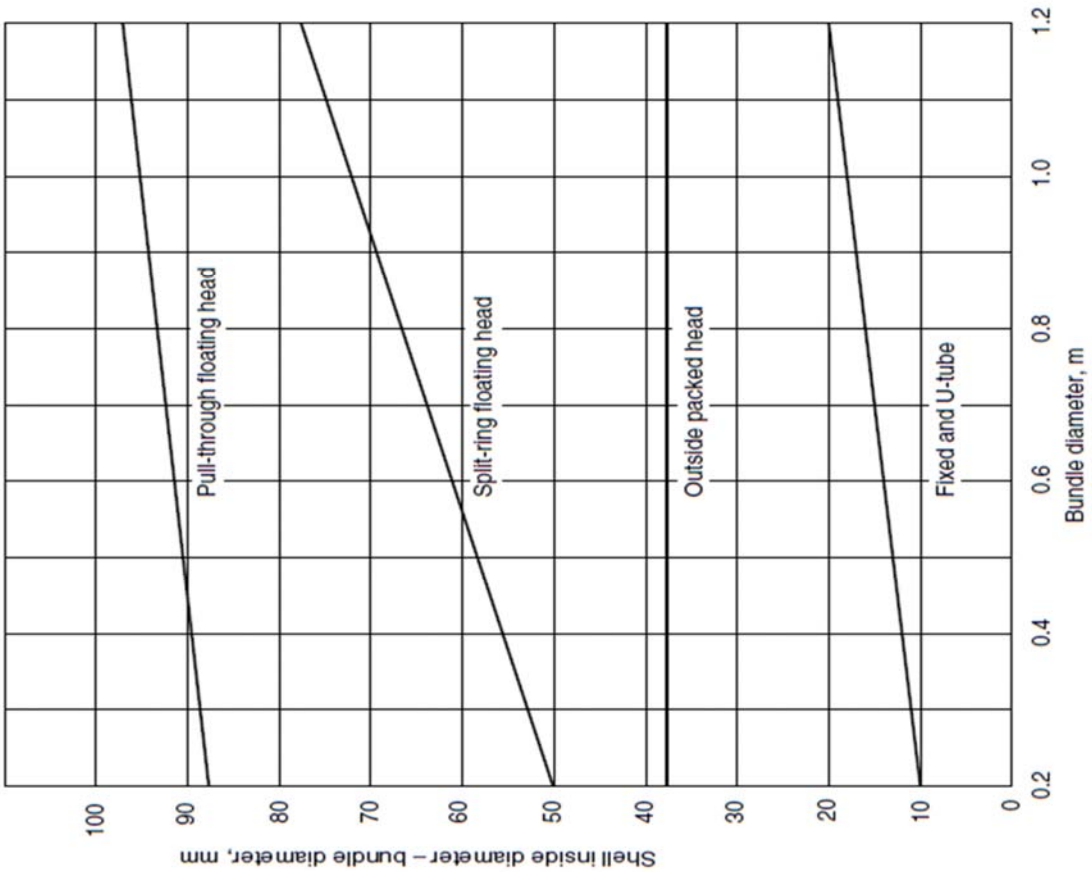


Figure 12.10. Shell-bundle clearance

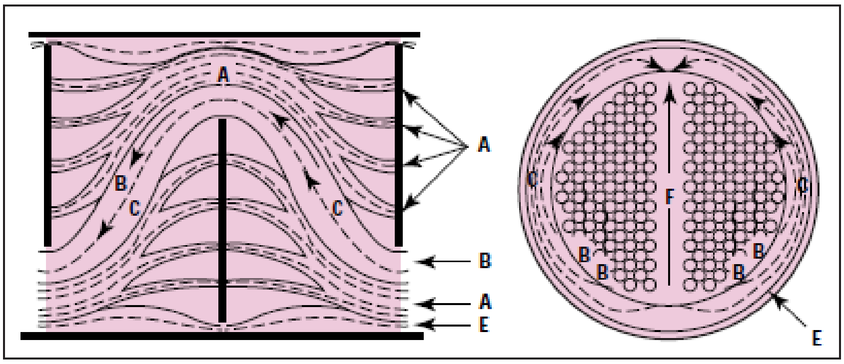
Shells

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.

Since the flow fractions depend strongly upon the path resistances, varying any of the following construction parameters will affect stream analysis and thereby the shell-side performance of an exchanger:

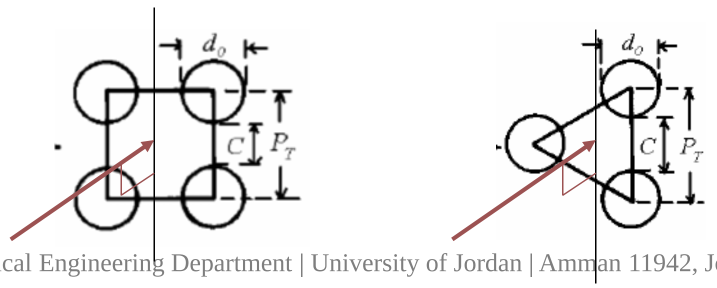
- baffle spacing and baffle cut;
- tube layout angle and tube pitch;
- number of lanes in the flow direction and lane width;
- clearance between the tube and the baffle hole;
- clearance between the shell I.D. and the baffle; and
- location of sealing strips and sealing rods.



Equivalent Counter Flow : Hydraulic or Equivalent Diameter

- The shell-side equivalent diameter is calculated along (instead of across) the long axes of the shell and therefore is taken as four times the net flow area as layout on the tube sheet (for any pitch layout) divided by the wetted perimeter.

$$D_e = 4\frac{\text{Net Free-flow area}}{heattransferperimeter}$$



Equivalent diameter for square layout:

$$D_{e-square} = \frac{4A_{flow}}{P_e} = \frac{4\left\{P_T^2 - \frac{\pi}{4} d_O^2\right\}}{\pi d_O}$$

Equivalent diameter for Triangular layout:

$$D_{e-triangular} = \frac{4A_{flow}}{P_e} = \frac{4\left\{\frac{\sqrt{3}P_T^2}{4} - \frac{\pi}{8} d_O^2\right\}}{\pi d_O / 2}$$

Minimum Shell Thickness

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

- The shell diameter must be selected to give as close a fit to the tube bundle as is practical, to reduce bypassing round the outside of the bundle;
- 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.

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.

Type of baffles. Baffles are used to support tubes, enable a desirable velocity to be maintained for the shell-side fluid, and prevent failure of tubes due to flow-induced vibration. There are two types of baffles: plate and rod. Plate baffle single-segmental, double-segmental, or triple-segmental, as shown in Figure 7.

The TEMA standards specify the minimum baffle spacing as one-fifth of the shell inside diameter or 2 in., whichever is greater. Closer spacing will result in poor bundle penetration by the shellside fluid and difficulty in mechanically cleaning the outsides of the tubes. Furthermore, a low baffle spacing results in a poor stream distribution as will be explained later.

Baffles serve two functions:

- **Support the tubes** for structural rigidity, preventing tube vibration and sagging
- **Divert the flow** across the bundle to obtain a higher heat transfer coefficient.

FIGURE 8.4. Sketch of segmental cut baffles and location in shell.

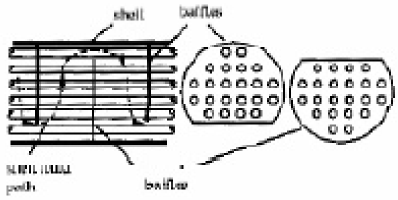


FIGURE 8.5. Sketch of doughnut and disc baffles and location in shell.

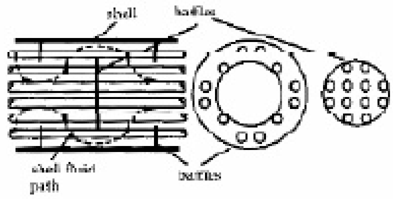
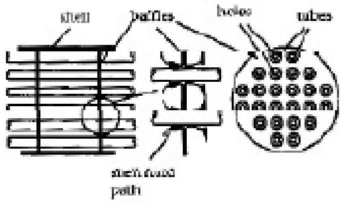
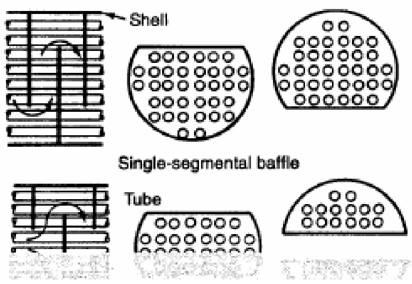


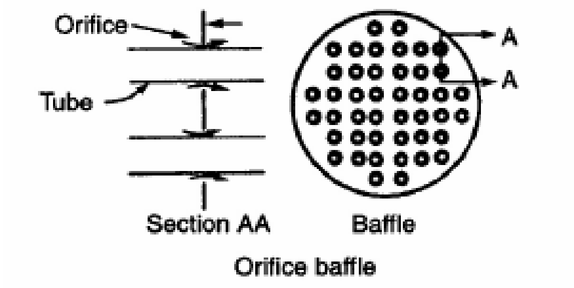
FIGURE 8.6. Sketch of orifice baffle and location in shell.

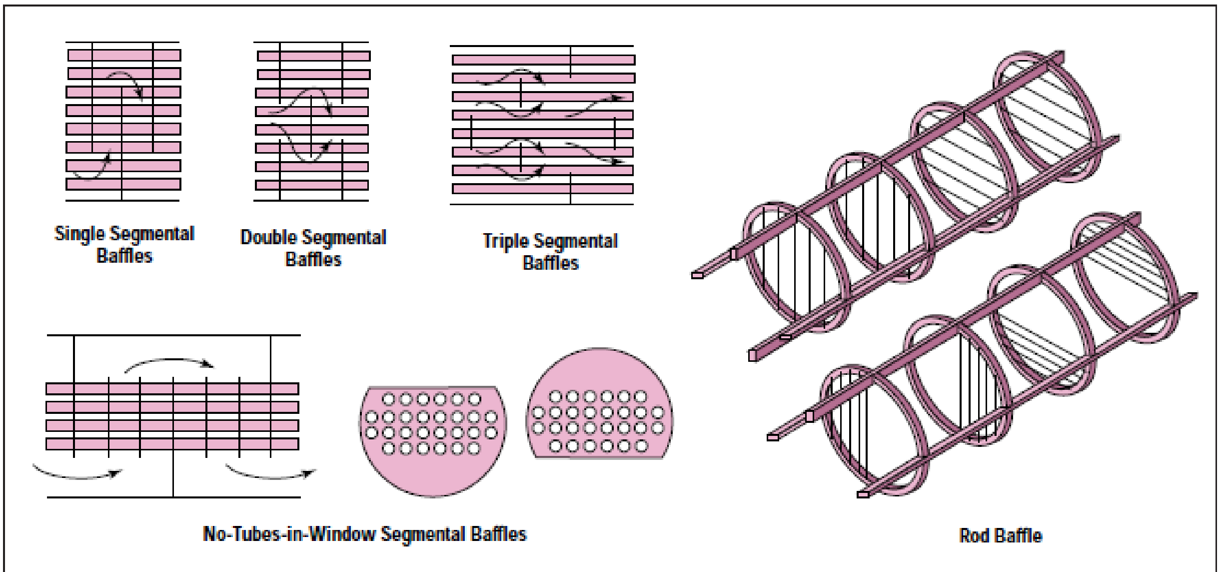


- The single and double segmental baffles are most frequently used. They divert the flow most effectively across the tubes.
- The baffle spacing must be chosen with care.
  - Optimal baffle spacing is somewhere between 40% - 60% of the shell diameter.
  - Baffle cut of 25%-35% is usually recommended.
- The triple segmental baffles are used for low pressure applications.



- In an orifice baffle shell-side-fluid flows through the clearance between tube outside diameter and baffle-hole diameter.





The maximum baffle spacing is the shell inside diameter. Higher baffle spacing will lead to predominantly longitudinal flow, which is less efficient than cross-flow, and large unsupported tube spans, which will make the exchanger prone to tube failure due to flow-induced vibration.

Optimum baffle spacing. For turbulent flow on the shellside (Re > 1,000), the heat-transfer coefficient varies to the 0.6–0.7 power of velocity; however, pressure drop varies to the 1.7–2.0 power. For laminar flow (Re < 100), the exponents are 0.33 for the heat-transfer coefficient and 1.0 for pressure drop. Thus, as baffle spacing is reduced, pressure drop increases at a much faster rate than does the heat-transfer coefficient.

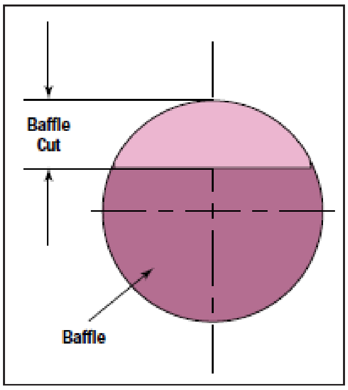
This means that there will be an optimum ratio of baffle spacing to shell inside diameter that will result in the highest efficiency of conversion of pressure drop to heat transfer. This optimum ratio is normally between 0.3 and 0.6.

Baffle cut.

Baffle cut can vary between 15% and 45% of the shell inside diameter.

Both very small and very large baffle cuts are detrimental to efficient heat transfer on the shellside due to large deviation from an ideal situation, as illustrated in Figure 9. It is strongly recommended that only baffle cuts between 20% and 35% be employed. Reducing baffle cut below 20% to increase the shellside heat-transfer coefficient or increasing the baffle cut beyond 35% to decrease the shellside pressure drop usually lead to poor designs. Other aspects of tube bundle geometry should be changed instead to achieve those goals. For example, double-segmental baffles or a divided-flow shell, or even a cross-flow shell, may be used to reduce the shellside pressure drop.

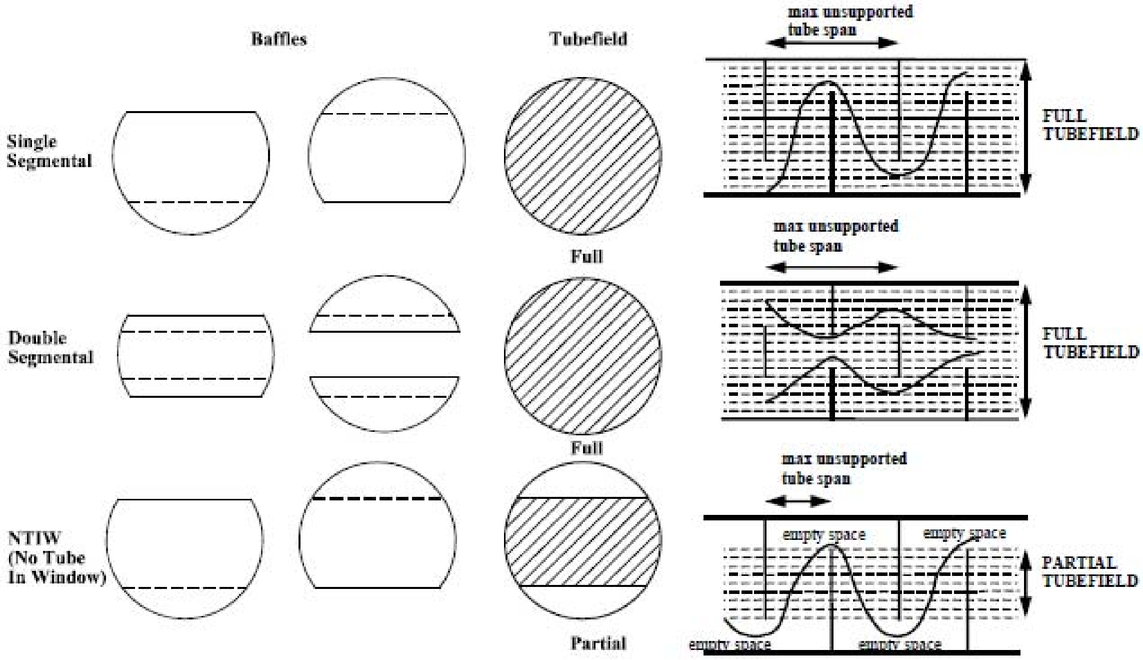
The term “baffle cut” is used to specify the dimensions of a segmental baffle. The baffle cut is the height of the segment removed to form the baffle, expressed as a percentage of the baffle disc diameter. Baffle cuts from 15 to 45 per cent are used. Generally, a baffle cut of 20 to 25 per cent will be the optimum, giving good heat-transfer rates, without excessive drop.



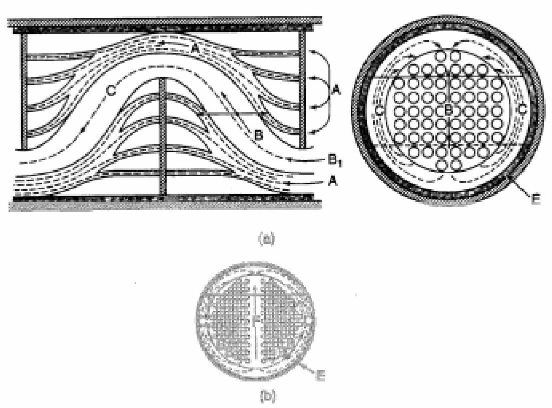
Another leakage path occurs through the clearance between the tube holes in the baffle and the tubes. The maximum design clearance will normally be $\frac{1}{32}$ in. (0.8 mm).

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.

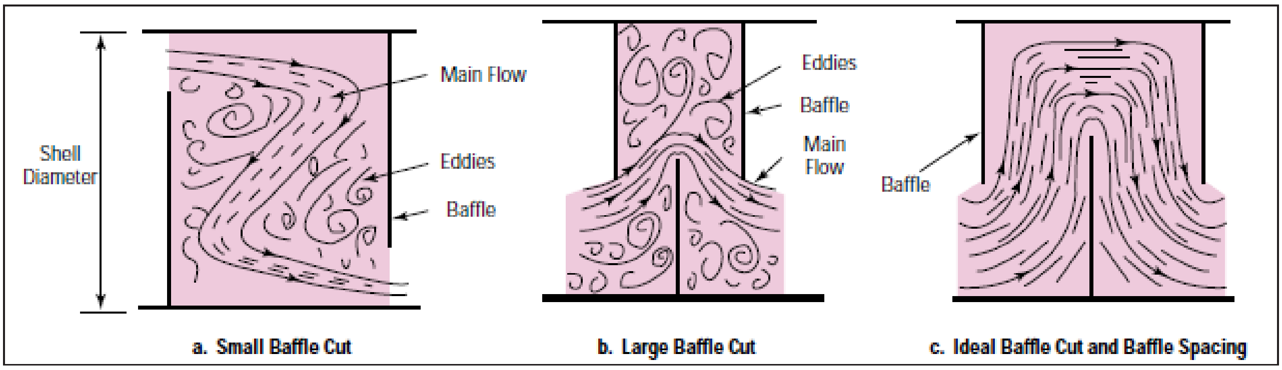
Baffle Types & Shell Flow Patterns



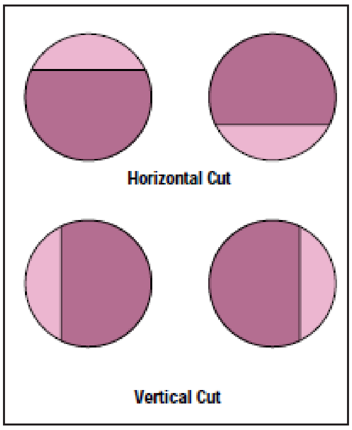
- There are five different shell side flow streams in a baffled heat exchanger
  - **Stream A** is the leakage stream in the orifice formed by the clearance between the baffle tube hole and the tube wall.
  - **Stream B** is the main effective cross-flow stream, which can be related to flow across ideal tube banks.



  - **Stream C** is the tube bundle bypass stream in the gap between the tube bundle and shell wall.
  - **Stream E** is the leakage stream between the baffle edge and shell wall.
  - **Stream F** is the bypass stream in flow channel partitions due to omissions of tubes in tube pass partitions.



. *Effect of small and large baffle cuts.*



Baffle cut orientation.

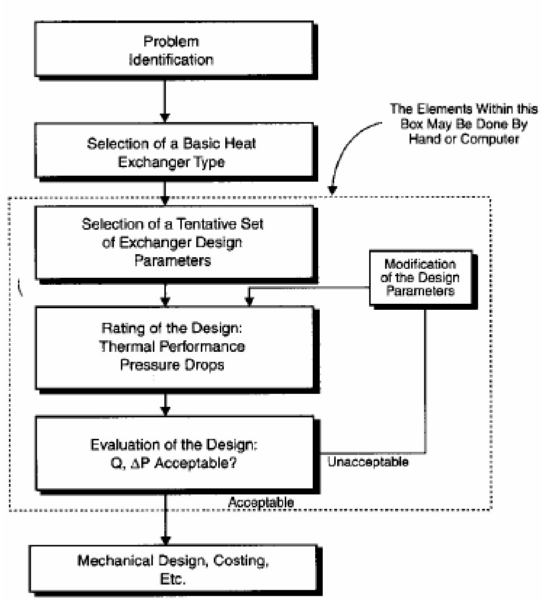
- **When the tube bundle employs baffles**, the heat transfer coefficient is higher than the coefficient for undisturbed flow around tubes without baffles.
- For a baffled heat exchanger
  - the higher heat transfer coefficients result from the **increased turbulence**.
  - the **velocity of fluid fluctuates** because of the constricted area between adjacent tubes across the bundle.
- **Only part of the fluid takes the desired path** through the tube bundle (Stream B), whereas a potentially substantial portion flows through the 'leakage' areas (Streams A, C, E & F)
  - However, these clearances are inherent to the manufacturing and assembly process of shell-and-tube exchangers, and the flow distribution within the exchanger must be taken into account.

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)

Basic Design Procedure

- Heat exchanger must satisfy the
  - Heat transfer requirements (design or process needs)
  - Allowable pressure drop (pumping capacity and cost)
- Steps in designing a heat exchanger can be listed as:
  - Identify the problem
  - Select an heat exchanger type
  - Calculate/Select initial design parameters
  - Rate the initial design
    - Calculate thermal performance and pressure drops for shell and tube side
  - Evaluate the design
    - Is performance and cost acceptable?



- Heat exchange design must:
  - Provide required area
  - Contain process pressure
  - Prevent leaks from shell to tubes or tubes to shell
  - Allow for thermal expansion
  - Allow for cleaning if fouling occurs
  - Allow for phase change (some cases)
  - Have reasonable pressure drop

- S&T heat exchangers are built to standards set by the Thermal Exchanger Manufacturers Association (TEMA)

Design Procedure-Kern Method

1. Define the duty: heat-transfer rate, fluid flow-rates, temperatures. $Q = (m\ C_p\ \Delta T) + (\delta m\ \Delta H_{vap})$
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 $Q = U\ A\ F\ \Delta T_{lm}$
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.

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.

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

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.

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.

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

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

Heat transfer

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

Heat-transfer factor, j_h

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.

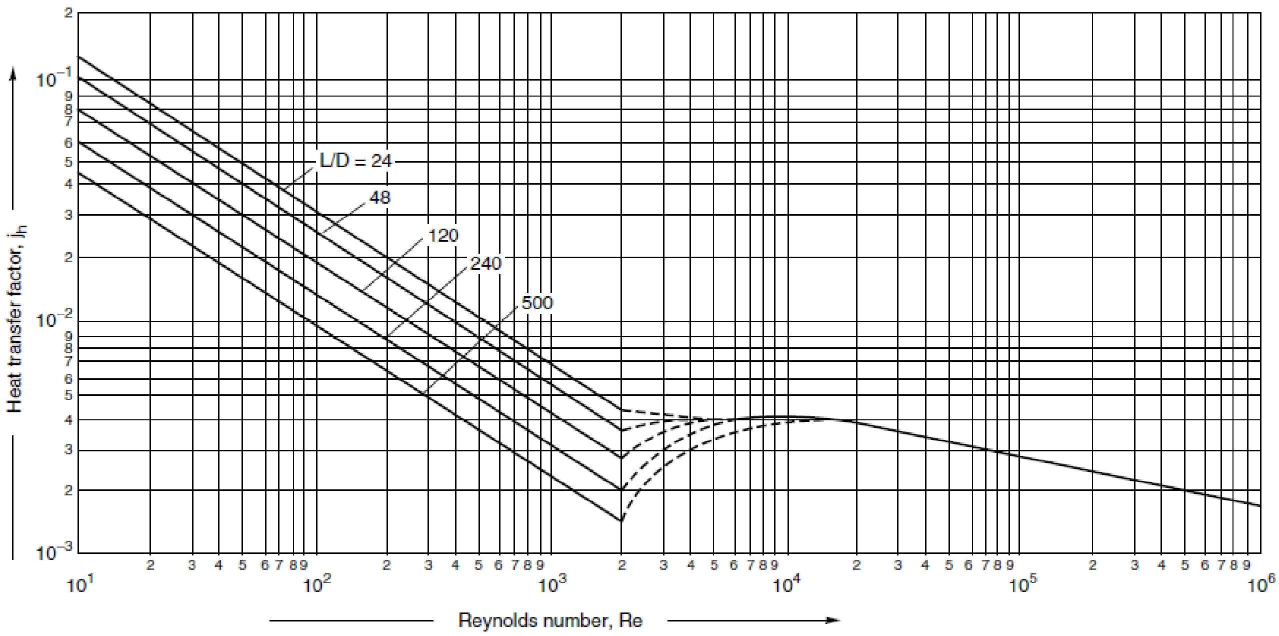


Figure 12.23. Tube-side heat-transfer factor

Tube-side pressure drop

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

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.

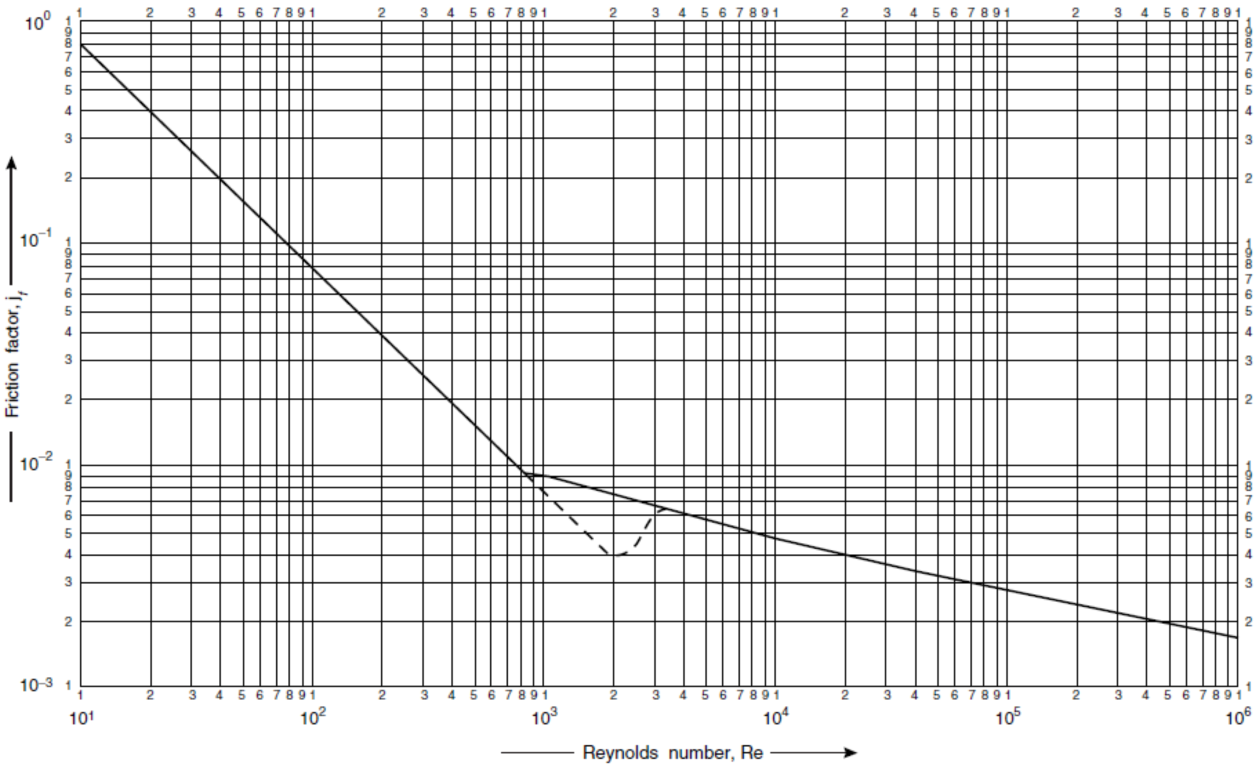


Figure 12.24. Tube-side friction factors

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

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

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.

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:

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

For an equilateral triangular pitch arrangement:

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

where d_e = equivalent diameter, m.

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

$$Re = \frac{G_s d_e}{\mu} = \frac{u_s d_e \rho}{\mu} \quad (12.24)$$

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

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

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.

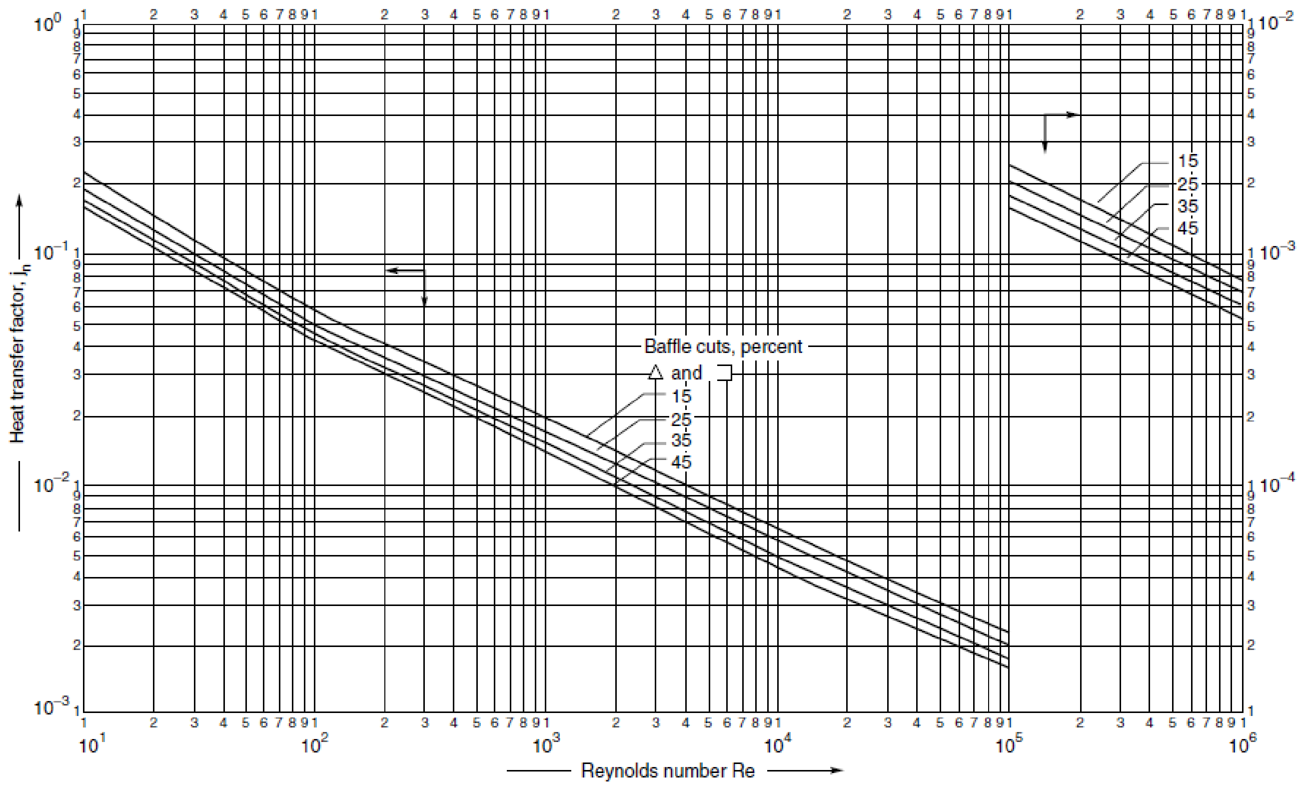


Figure 12.29. Shell-side heat-transfer factors, segmental baffles

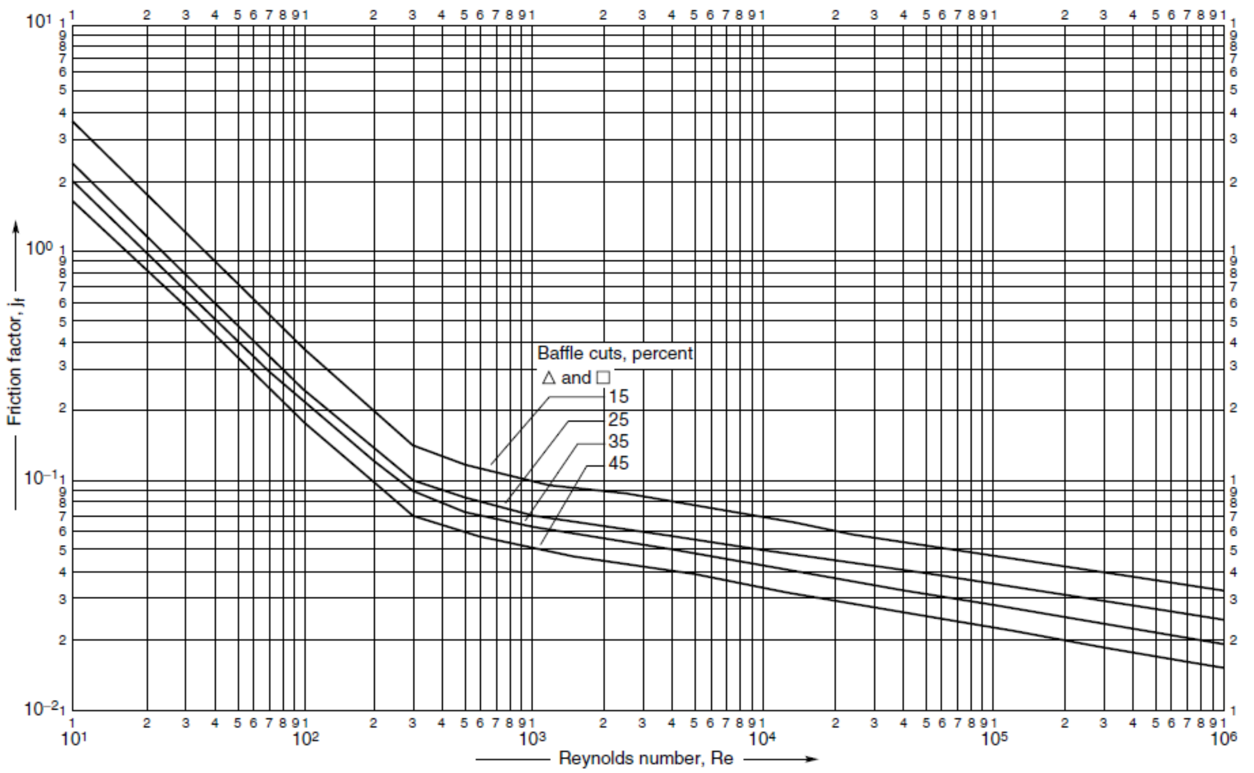


Figure 12.30. Shell-side friction factors, segmental baffles

Approximate Heat Transfer Coefficients

More examples (in metric units) in Chapter 12

Fluid	h (Btu/(hr.ft^2.F)) Shell-side	Tube-side
Liquids		
Water solutions, 50% water or more	300	300
Alcohols, organic solvents	200	200
Light Hydrocarbons (naphtha, gasoline)	190	190
Medium Hydrocarbons (kerosene, diesel)	130	120
Heavy oils (gas oils, crude oil)	30	20
Vapors		
Air, 10 psig	10	10
Hydrogen, 50 psig	100	100
Hydrogen, 300 psig	300	300
Hydrogen, 500 psig	400	400
Hydrocarbon vapor, 50 psig	60	60
Noncondensible gas, 2 psig	5	5

Note: Coefficients are based on 3/4 inch diameter tubes. For Tube side flows, correct by multiplying by 0.75/Actual OD. Estimated accuracy is 25%. For 50% hydrogen in vapor, reduce h to 2/3 of pure H_2 value.

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.

Only the thermal design will be considered.

$$\text{Heat capacity methanol} = 2.84\ \text{kJ/kg}^\circ\text{C}$$

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

$$\text{Heat capacity water} = 4.2\ \text{kJ/kg}^\circ\text{C}$$

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

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

Use one shell pass and two tube passes

$$R = \frac{95 - 40}{40 - 25} = 3.67$$

$$S = \frac{40 - 25}{95 - 25} = 0.21$$

From Figure 12.19

$$F_t = 0.85$$

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

From Figure 12.1

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

Provisional area

$$A = \frac{4340 \times 10^3}{26 \times 600} = 278\ \mathrm{m^2}$$

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

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

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

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

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

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

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

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

(*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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Shell-side coefficient

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

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

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

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

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

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

$$\text{Methanol density} = 750 \text{ kg/m}^3$$

$$\text{Viscosity} = 0.34 \text{ mNs/m}^2$$

$$\text{Heat capacity} = 2.84 \text{ kJ/kg}^{\circ}\text{C}$$

$$\text{Thermal conductivity} = 0.19 \text{ W/m}^{\circ}\text{C}$$

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

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

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 $= 68 - 33 = 35^\circ$C
across all resistances

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

Mean wall temperature $= 68 - 8 = 60^\circ$C

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

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

Overall coefficient

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

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

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

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

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

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

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

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

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

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

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

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 $(1/2)^2$

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

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.

Pumping Power

Assume pump efficiency as 0.80.

$\eta_p = 0.80$

Tube-side pumping power is:

$$P_{tube} = \frac{\dot{m}_{hot-oil} \times \Delta p_{tube}}{\eta_p \times \rho_{hot-oil}}$$

Shell-side pumping power:

$$P_{shell} = \frac{\dot{m}_{crude-oil} \times \Delta p_{shell}}{\eta_p \times \rho_{crude-oil}}$$

Results of Kern Method

Tube-side pressure drop (kPa)	
Shell-side pressure drop (kPa)	
Tube-side pumping power (kW)	
Shell-side pumping power (kW)	

Roadmap To Increase Heat Transfer

- Increase heat transfer coefficent
- Tube Side
  - o Increase number of tubes
  - o Decrease tube outside diameter
- Shell Side
  - o Decrease the baffle spacing
  - o Decrease baffle cut
- Increase surface area
  - o Increase tube length
  - o Increase shell diameter à increased number of tubes
  - o Employ multiple shells in series or parallel
- Increase LMTD correction factor and heat exchanger effectiveness
  - o Use counterflow configuration
  - o Use multiple shell configuration

Roadmap To Reduce Pressure Drop

- Tube side
  - o Decrease number of tube passes
  - o Increase tube diameter
  - o Decrease tube length and increase shell diameter and number of tubes
- Shell side
  - o Increase the baffle cut
  - o Increase the baffle spacing ✓
  - o Increase tube pitch
  - o Use double or triple segmental baffles

$\frac{B}{D_s} = 1$

$\frac{P_T}{d_o} = 1.5$

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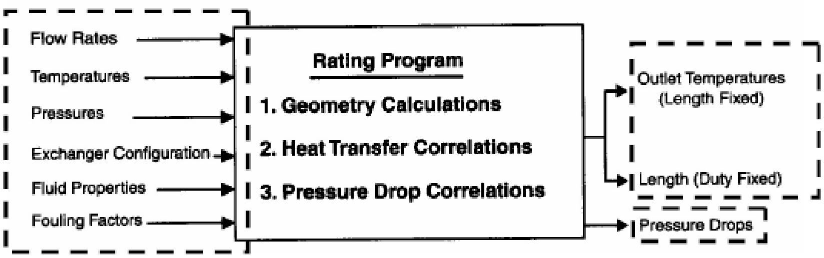
Tel. +962 6 535 5000 | 22888

- Study the effect of baffle spacing on size of heat exchanger.
- Study the effect of baffle spacing on total pumping power.

$$\frac{1}{15} < \frac{D_s}{L} < \frac{1}{5}$$

Rating of the Heat Exchanger Design

- **Rating an exchanger means** to evaluate the thermo-hydraulic performance of a **fully specified** exchanger.
- Input to the rating process is heat exchanger **geometry** (constructional design parameters), **process conditions** (flow rate, temperature, pressure) and **material/fluid properties** (density, thermal conductivity)
- **First output** from the rating process is either the outlet temperature for fixed tube length or the tube length itself to meet the outlet temperature requirement.
- **Second output** from the rating process is the pressure drop for both fluid streams hence the pumping energy requirements and size.



Insufficient Thermal Rating

- If the output of the rating analysis is not acceptable, a geometrical modification should be made
- If the required amount of heat cannot be transferred to satisfy specific outlet temperature, one should find a way to increase the heat transfer coefficient or increase exchanger surface area
  - One can increase the tube side heat transfer coefficient by increasing the fluid velocity - Increase number of tube passes
  - One can increase the shell side heat transfer coefficient by decreasing baffle spacing and/or baffle cut
  - One can increase the surface area by
    - Increasing the heat exchanger length
    - Increasing the shell diameter
    - Multiple shells in series

Insufficient Pressure Drop Rating

- If the pressure drop on the tube side is greater than the allowable pressure drop, then
  - the number of tube passes can be decreased or
  - the tube diameter can be increased which may result to
    - decrease the tube length – (Same surface area)
    - increase the shell diameter and the number of tubes
- If the shell side pressure drop is greater than the allowable pressure drop then baffle spacing, tube pitch, and baffle cut can be increased or one can change the baffle type.

The Trade-Off

Between Thermal Balance & Flow Loss

- Heat transfer and fluid friction losses tend to compete with one another.
- The total energy loss can be minimized by adjusting the size of one irreversibility against the other .
- These adjustments can be made by properly selecting physical dimensions of the solid parts (fins, ducts, heat exchanger surface).
- It must be understood, however, that the result is at best a thermodynamic optimum.
  - Constraints such as cost, size, and reliability enter into the determination of truly optimal designs.

Heat Exchanger Analysis: The Effectiveness–NTU Method

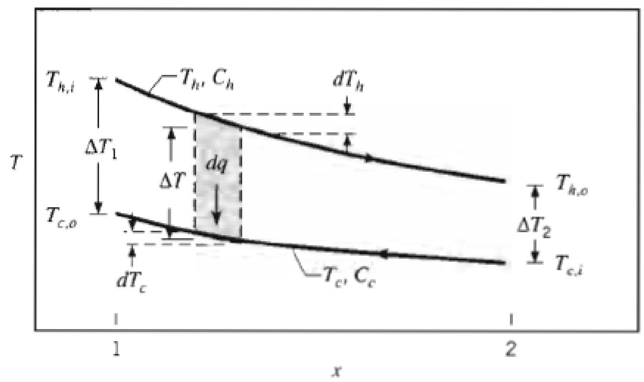
This method is based on a dimensionless parameter called the **heat transfer effectiveness** ε, defined as

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}}$$

the *maximum possible heat transfer rate*, $q_{\max}$, could, in principle, be achieved in a counterflow heat exchanger

i.e. the maximum temperature difference in a heat exchanger is the difference between the inlet temperatures of the hot and cold fluids.

$$\Delta T_{\max} = T_{h,\text{ in}} - T_{c,\text{ in}}$$



$$\dot{Q}_{\max} = C_{\min}(T_{h,\,\text{in}} - T_{c,\,\text{in}})$$

where $C_{\min}$ is the smaller of $C_h = \dot{m}_h C_{ph}$ and $C_c = \dot{m}_c C_{pc}$.

The determination of $\dot{Q}_{\max}$ requires the availability of the *inlet temperature* of the hot and cold fluids and their *mass flow rates,* which are usually specified. Then, once the effectiveness of the heat exchanger is known, the actual heat transfer rate $\dot{Q}$ can be determined from

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = \varepsilon C_{\min}(T_{h,\,\text{in}} - T_{c,\,\text{in}})$$

Therefore, the effectiveness of a heat exchanger enables us to determine the heat transfer rate without knowing the *outlet temperatures* of the fluids.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,\,\text{out}} - T_{c,\,\text{in}})}{C_{\min}(T_{h,\,\text{in}} - T_{c,\,\text{in}})} \longrightarrow \frac{T_{c,\,\text{out}} - T_{c,\,\text{in}}}{T_{h,\,\text{in}} - T_{c,\,\text{in}}} = \varepsilon \frac{C_{\min}}{C_c}$$

$$\varepsilon = \frac{C_c(T_{c,o} - T_{c,i})}{C_{\min}(T_{h,i} - T_{c,i})} \qquad \varepsilon = \frac{C_h(T_{h,i} - T_{h,o})}{C_{\min}(T_{h,i} - T_{c,i})}$$

$$\varepsilon_{\text{parallel flow}} = \frac{1 - \exp\left[-\dfrac{UA_s}{C_{\min}}\left(1 + \dfrac{C_{\min}}{C_{\max}}\right)\right]}{1 + \dfrac{C_{\min}}{C_{\max}}}$$

Again $C_{\min}$ is the *smaller* heat capacity ratio and $C_{\max}$ is the larger one, and it makes no difference whether $C_{\min}$ belongs to the hot or cold fluid.

$$c = \frac{C_{\min}}{C_{\max}} \qquad \textbf{capacity ratio}$$

Effectiveness relations of the heat exchangers typically involve the *dimensionless* group $UA_s/C_{\min}$. This quantity is called the **number of transfer units NTU** and is expressed as

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}C_p)_{\min}} \qquad \textbf{(13-39)}$$

TABLE 13–4

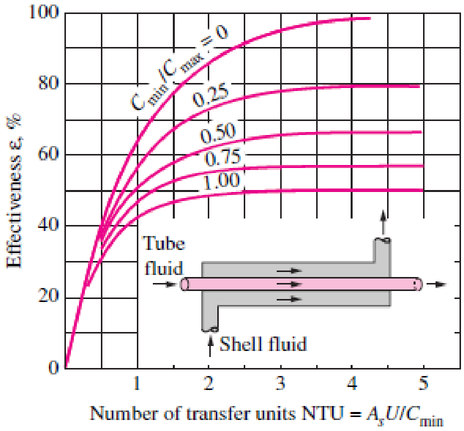
Effectiveness relations for heat exchangers: $NTU = UA_s/C_{min}$ and $c = C_{min}/C_{max} = (\dot{m}C_p)_{min}/(\dot{m}C_p)_{max}$ (Kays and London, Ref. 5.)

Heat exchanger type	Effectiveness relation
1 *Double pipe:*	
Parallel-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1+c)]}{1+c}$
Counter-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1-c)]}{1 - c\exp[-NTU(1-c)]}$
2 *Shell and tube:* One-shell pass 2, 4, . . . tube passes	$\varepsilon = 2\left\{1 + c + \sqrt{1+c^2}\,\dfrac{1+\exp[-NTU\sqrt{1+c^2}]}{1-\exp[-NTU\sqrt{1+c^2}]}\right\}^{-1}$
3 *Cross-flow (single-pass)*	
Both fluids unmixed	$\varepsilon = 1 - \exp\left\{\dfrac{NTU^{0.22}}{c}[\exp(-c\ NTU^{0.78}) - 1]\right\}$
C_{max} mixed, C_{min} unmixed	$\varepsilon = \frac{1}{c}(1 - \exp\{1 - c[1 - \exp(-NTU)]\})$
C_{min} mixed, C_{max} unmixed	$\varepsilon = 1 - \exp\left\{-\frac{1}{c}[1 - \exp(-c\,NTU)]\right\}$
4 *All heat exchangers with $c = 0$*	$\varepsilon = 1 - \exp(-NTU)$

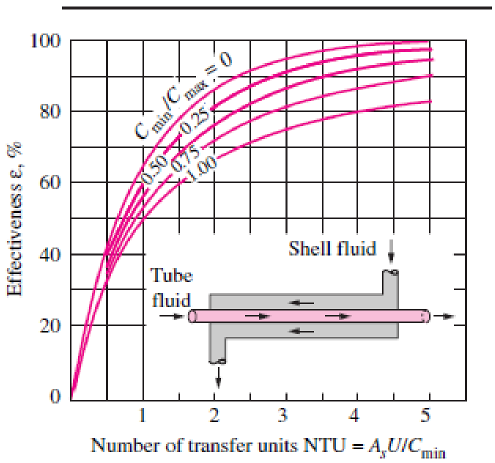
TABLE 13–5

NTU relations for heat exchangers $NTU = UA_s/C_{min}$ and $c = C_{min}/C_{max} = (\dot{m}C_p)_{min}/(\dot{m}C_p)_{max}$ (Kays and London, Ref. 5.)

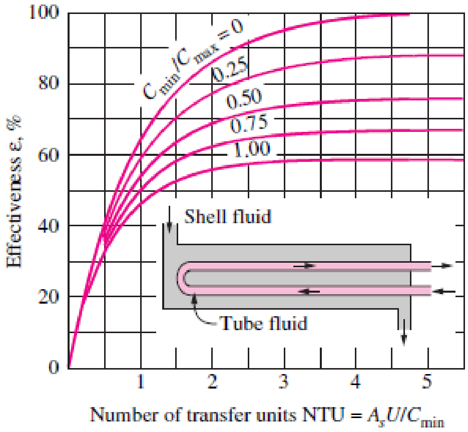
Heat exchanger type	NTU relation
1 *Double-pipe:*	
Parallel-flow	$NTU = -\dfrac{\ln[1 - \varepsilon(1+c)]}{1+c}$
Counter-flow	$NTU = \dfrac{1}{c-1}\ln\left(\dfrac{\varepsilon - 1}{\varepsilon c - 1}\right)$
2 *Shell and tube:* One-shell pass 2, 4, . . . tube passes	$NTU = -\dfrac{1}{\sqrt{1+c^2}}\ln\left(\dfrac{2/\varepsilon - 1 - c - \sqrt{1+c^2}}{2/\varepsilon - 1 - c + \sqrt{1+c^2}}\right)$
3 *Cross-flow (single-pass)* C_{max} mixed, C_{min} unmixed	$NTU = -\ln\left[1 + \dfrac{\ln(1 - \varepsilon c)}{c}\right]$
C_{min} mixed, C_{max} unmixed	$NTU = -\dfrac{\ln[c\ln(1-\varepsilon) + 1]}{c}$
4 *All heat exchangers with $c = 0$*	$NTU = -\ln(1 - \varepsilon)$



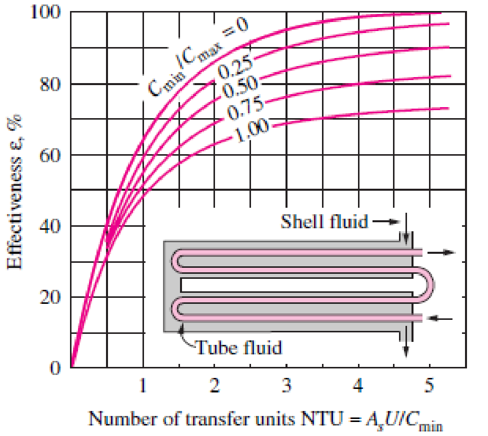
(*a*) Parallel-flow



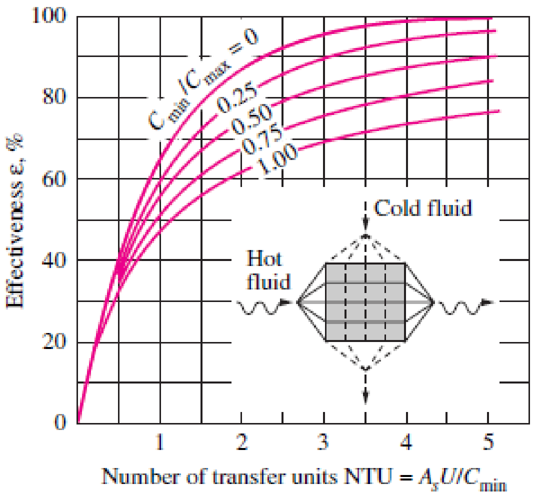
(*b*) Counter-flow



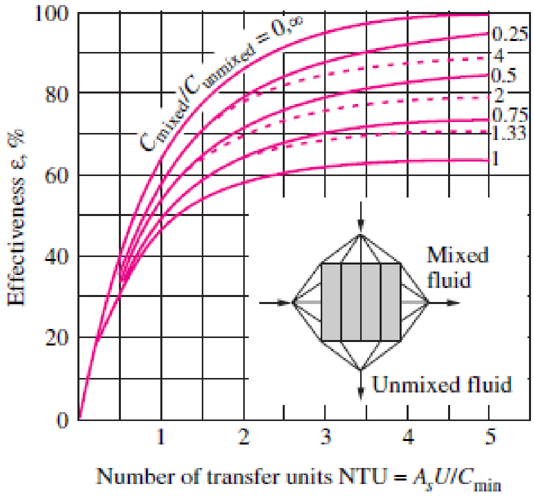
(*c*) One-shell pass and 2, 4, 6, … tube passes



(*d*) Two-shell passes and 4, 8, 12, … tube passes



(*e*) Cross-flow with both fluids unmixed



(*f*) Cross-flow with one fluid mixed and the other unmixed

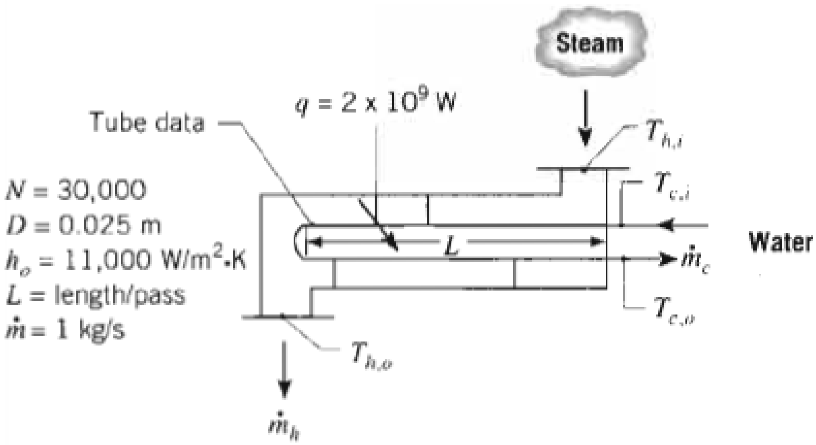
The condenser of a large steam power plant is a heat exchanger in which steam is condensed to liquid water. Assume the condenser to be a *shell-and-tube* heat exchanger consisting of a single shell and 30,000 tubes, each executing two passes. The tubes are of thin wall construction with $D = 25$ mm, and steam condenses on their outer surface with an associated convection coefficient of $h_o = 11{,}000\ \mathrm{W/m^2 \cdot K}$. The heat transfer rate that must be effected by the exchanger is $q = 2 \times 10^9$ W, and this is accomplished by passing cooling water through the tubes at a rate of 3×10^4 kg/s (the flow rate per tube is therefore 1 kg/s). The water enters at 20°C, while the steam condenses at 50°C. What is the temperature of the cooling water emerging from the condenser? What is the required tube length L per pass?

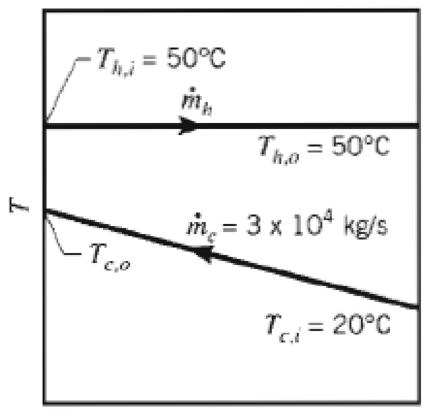
Known: Heat exchanger consisting of single shell and 30,000 tubes with two passes each.

Find:

1. Outlet temperature of the cooling water.
2. Tube length per pass to achieve required heat transfer.

Schematic:

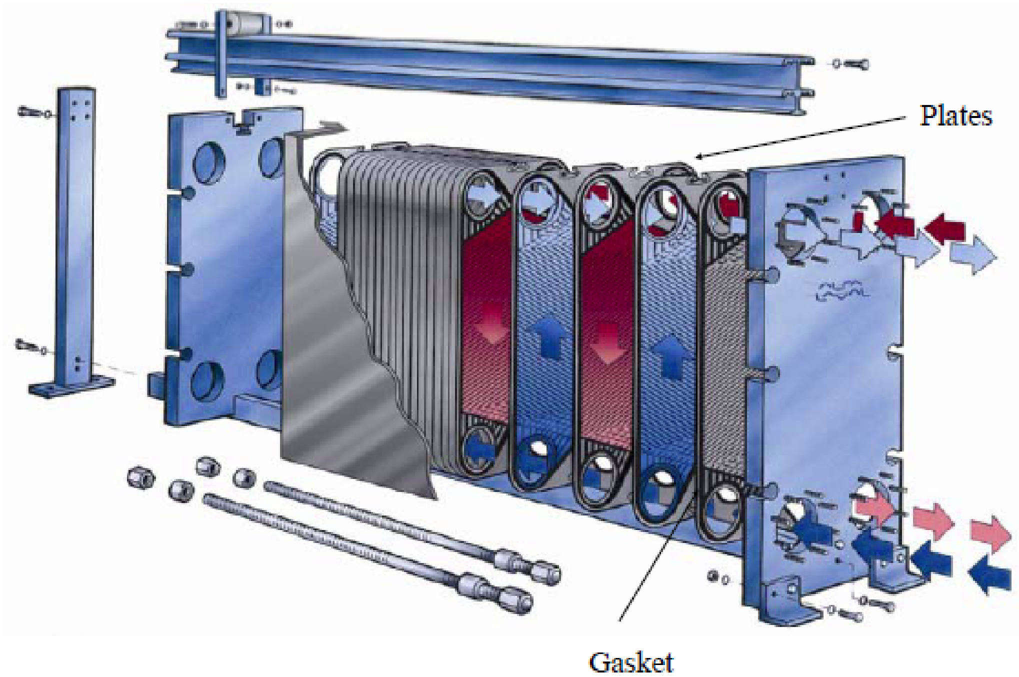


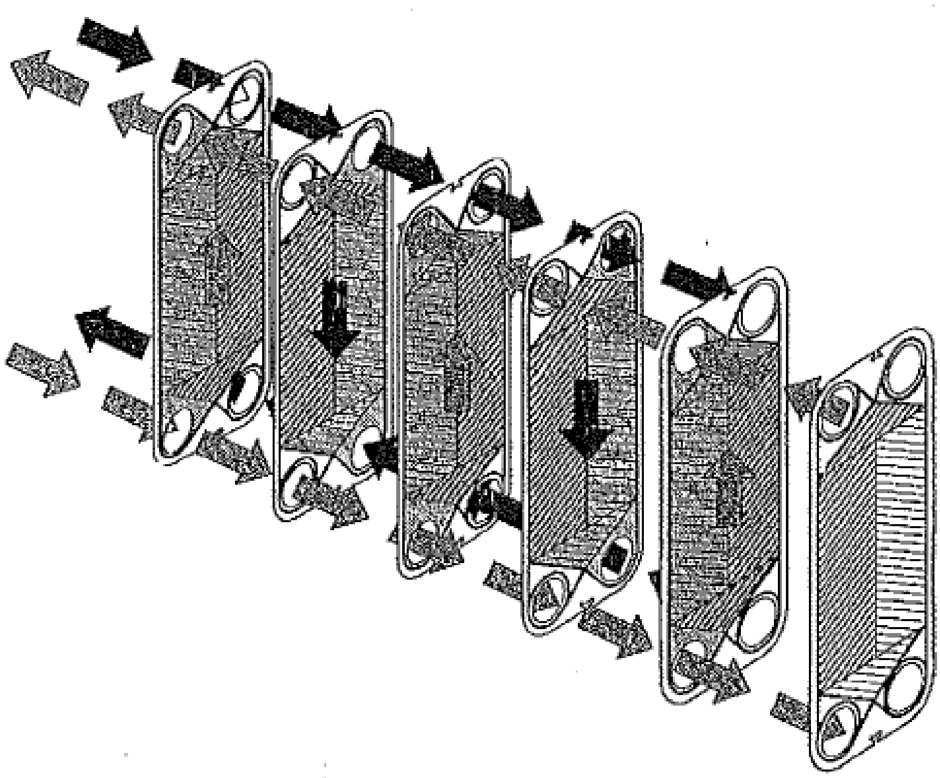


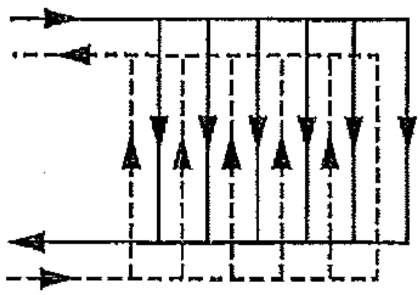
Properties: Table A.6, water (assume $\overline{T}_c \approx 27°\mathrm{C} = 300$ K): $\rho = 997\ \mathrm{kg/m^3}$, $c_p = 4179\ \mathrm{J/kg \cdot K}$, $\mu = 855 \times 10^{-6}\ \mathrm{N \cdot s/m^2}$, $k = 0.613\ \mathrm{W/m \cdot K}$, $Pr = 5.83$.

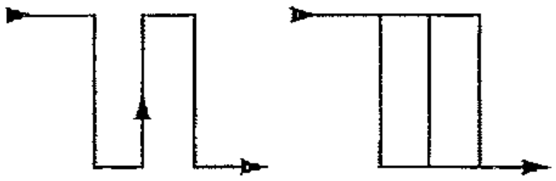
Compact Heat Exchangers

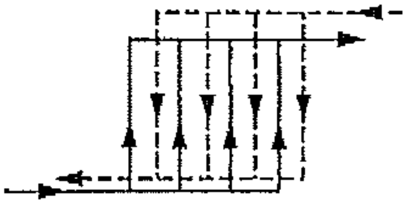
Plate & Frame Exchangers





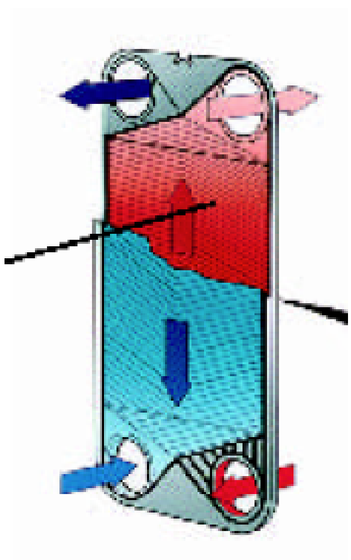






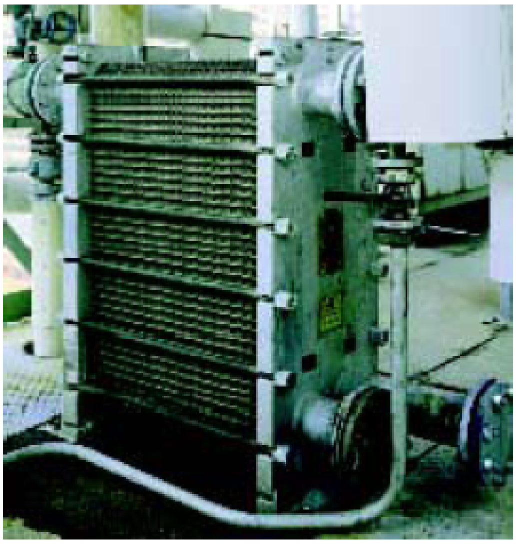
Flow pattern: (a) schematic of a U-type arrangement — counterflow, single-pass flow ($1\times6/1\times6$) (b) Z-arrangement ($1\times4/1\times4$ configuration).

Plate & Frame Exchangers

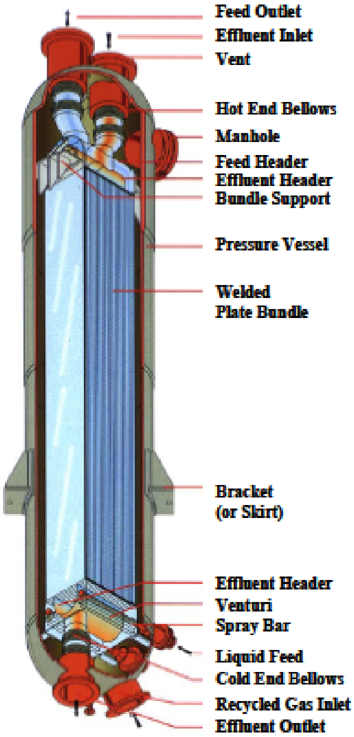


- Advantages
  - Close to counter-current heat transfer, so high F factor allows temperature cross and close temperature approach
  - Easy to add area
  - Compact size
  - Relatively inexpensive for high alloy
  - Can be designed for quick cleaning in place
- Disadvantages
  - Lots of gaskets
  - Lower design pressure, temperature
  - External leakage if gaskets fail
- Applications
  - Food processing, brewing, biochemicals, etc.

Plate & Frame Exchangers

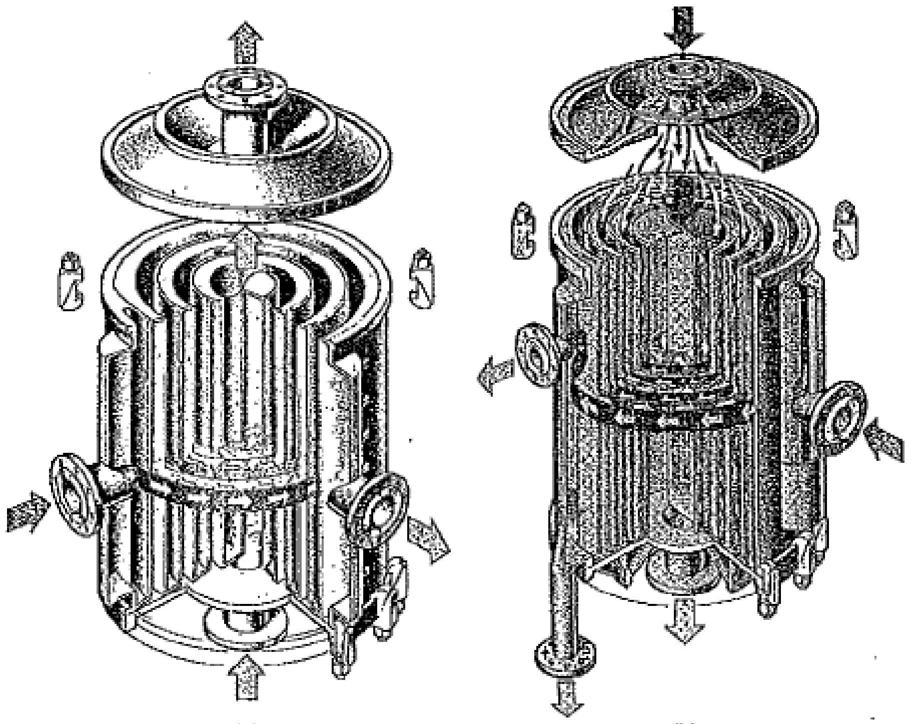


Welded Plate Heat Exchangers



Source: Alfa-Laval Packinox

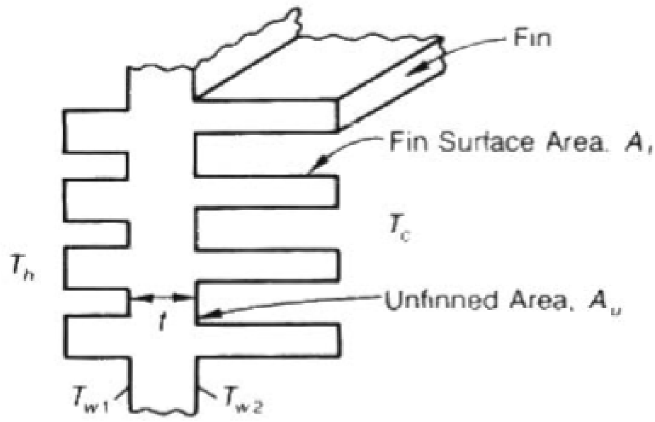
- Advantages
  - Higher thermal efficiency
  - Single unit can replace multiple shell & tube units
  - Closer approach to hot inlet temperature
  - Low pressure drop
  - Little chance of vibration problems
  - Excellent distribution of two phase flows
- Disadvantages
  - Single alloy material for plates
  - Difficult to clean
  - Few manufacturers at large scale (Alfa Laval Packinox)
- Used in large scale clean services that need close temperature approach



(a) (b)

Spiral plate heat exchangers:

Finned wall heat exchanger



$$Q = (\eta_f A_f h_f + A_u h_u)\,\Delta T$$

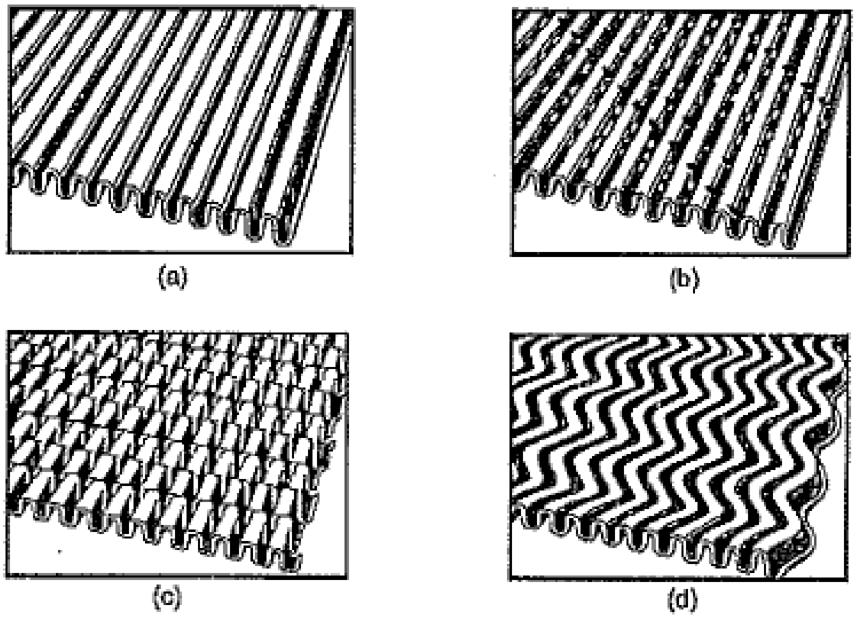
Taking $h_u = h_f = h$

$$Q = hA\left[1 - \frac{A_f}{A}(1 - \eta_f)\right]\Delta T$$

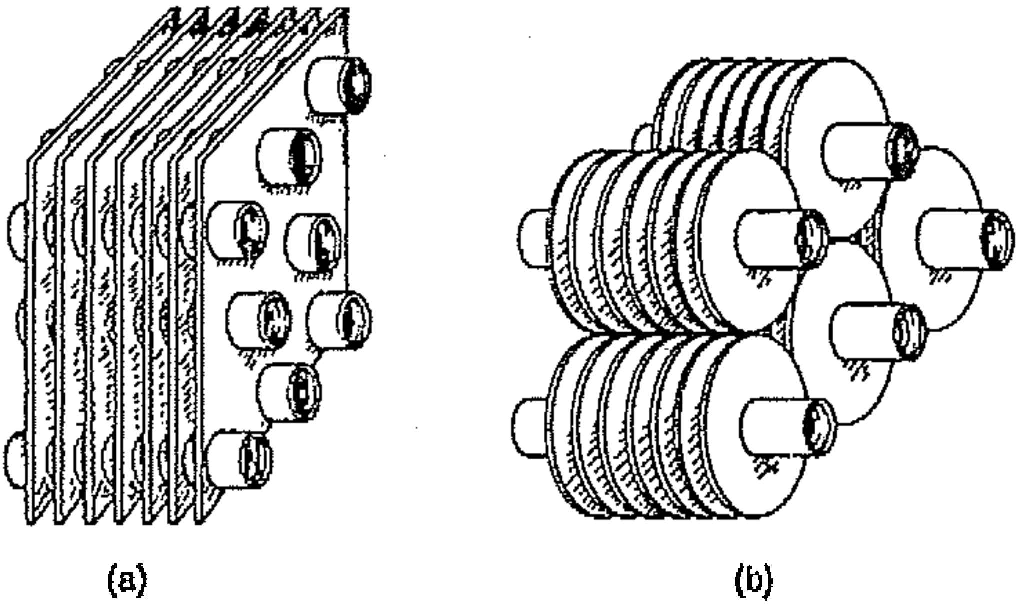
$$Q = \eta_o hA\,\Delta T$$

where $\eta_o = [1 - (1 - \eta_f)A_f/A]$ is called the overall surface efficiency and $A = A_u + A_f$.

Finned wall heat exchanger



Fin types in plate-fin exchangers: (a) plain; (b) perforated; (c) serrated; (d) herringbone. (Adapted from Butterworth, D. [1991] In *Boilers, Evaporators and Condensers*, John Wiley & Sons, New York.)



Finned-tube geometries used with circular tubes: (a) plate fin-and-tube used for gases; (b) individually finned tubes; (c) plain-fin and offset-fin. (From Webb, R. L., *Principles of Enhanced Heat Transfer*, John Wiley & Sons, New York, 1994.)

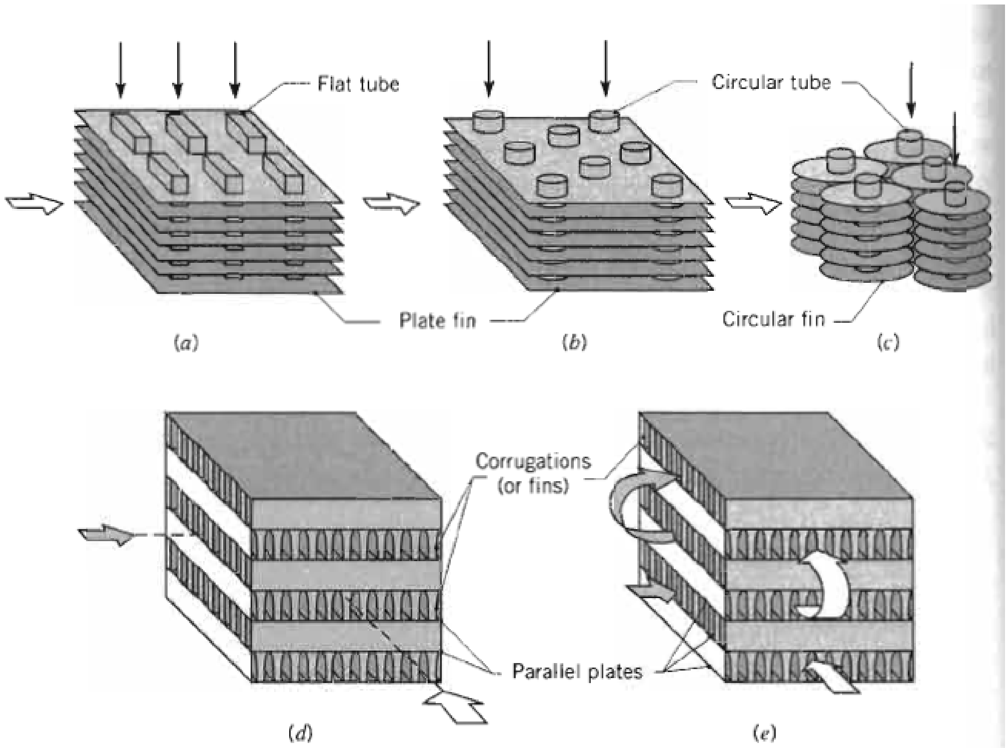
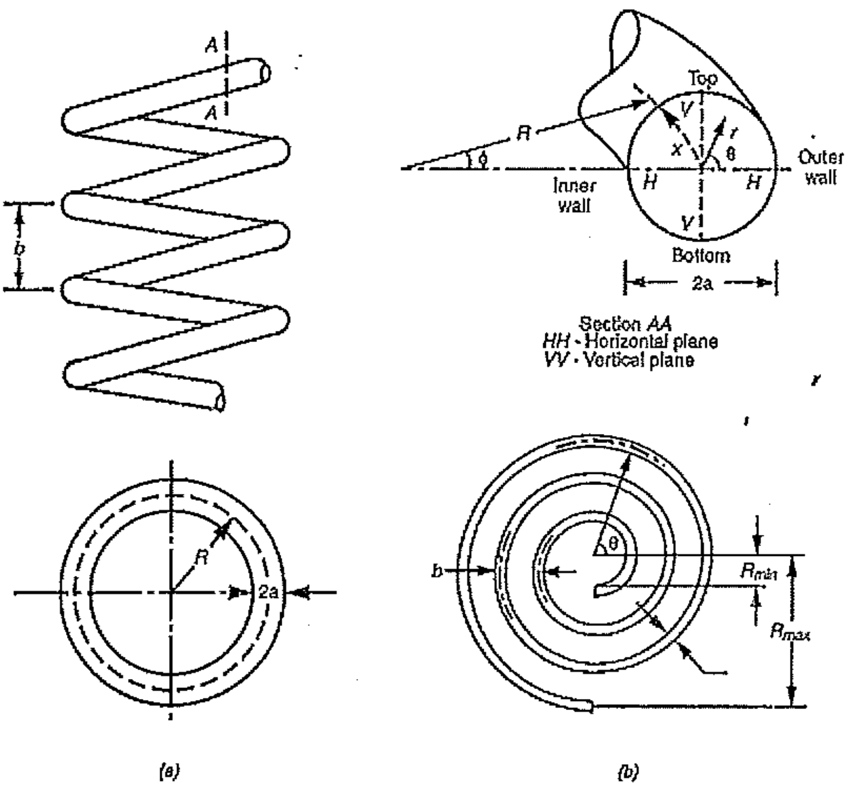


FIGURE 11.5 Compact heat exchanger cores. (*a*) Fin–tube (flat tubes, continuous plate fins). (*b*) Fin–tube (circular tubes, continuous plate fins). (*c*) Fin–tube (circular tubes, circular fins). (*d*) Plate–fin (single pass). (*e*) Plate–fin (multipass).

Spiral and helical heat exchangers



- Spiral heat exchanger can be considered as a plate heat exchanger in which the plates are formed into a spiral. The fluids flow through the channels formed between the plates.

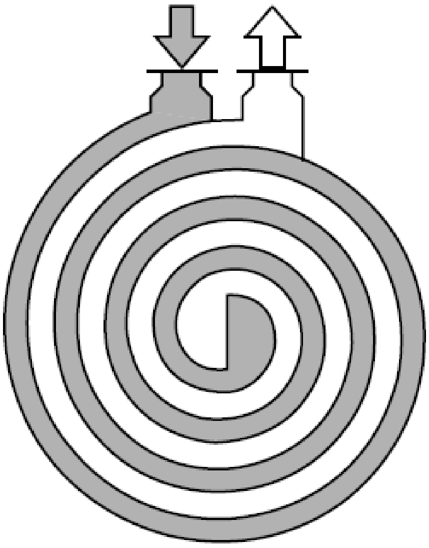


Figure 12.64. Spiral heat exchanger

Dimensional Data For Commercial Tubing

TABLE 8.1

Dimensional Data for Commercial Tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.²)	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Length, Steel (lb.)	ID Tubing (in.)	OD/ID
1/4	22	0.028	0.0295	0.0655	0.0508	0.066	0.194	1.289
1/4	24	0.022	0.0333	0.0655	0.0539	0.054	0.206	1.214
1/4	26	0.018	0.0360	0.0655	0.0560	0.045	0.214	1.168
3/8	18	0.049	0.0603	0.0982	0.0725	0.171	0.277	1.354
3/8	20	0.035	0.0731	0.0982	0.0798	0.127	0.305	1.233
3/8	22	0.028	0.0799	0.0982	0.0835	0.104	0.319	1.176
3/8	24	0.022	0.0860	0.0982	0.0867	0.083	0.331	1.133
1/2	16	0.065	0.1075	0.1309	0.0969	0.302	0.370	1.351
1/2	18	0.049	0.1269	0.1309	0.1052	0.236	0.402	1.244
1/2	20	0.035	0.1452	0.1309	0.1126	0.174	0.430	1.163
1/2	22	0.028	0.1548	0.1309	0.1162	0.141	0.444	1.126
5/8	12	0.109	0.1301	0.1636	0.1066	0.602	0.407	1.536
5/8	13	0.095	0.1486	0.1636	0.1139	0.537	0.435	1.437
5/8	14	0.083	0.1655	0.1636	0.1202	0.479	0.459	1.362
5/8	15	0.072	0.1817	0.1636	0.1259	0.425	0.481	1.299
5/8	16	0.065	0.1924	0.1636	0.1296	0.388	0.495	1.263
5/8	17	0.058	0.2035	0.1636	0.1333	0.350	0.509	1.228
5/8	18	0.049	0.2181	0.1636	0.1380	0.303	0.527	1.186
5/8	19	0.042	0.2298	0.1636	0.1416	0.262	0.541	1.155
5/8	20	0.035	0.2419	0.1636	0.1453	0.221	0.555	1.136
3/4	10	0.134	0.1825	0.1963	0.1262	0.884	0.482	1.556
3/4	11	0.120	0.2043	0.1963	0.1335	0.809	0.510	1.471
3/4	12	0.109	0.2223	0.1963	0.1393	0.748	0.532	1.410
3/4	13	0.095	0.2463	0.1963	0.1466	0.666	0.560	1.339
3/4	14	0.083	0.2679	0.1963	0.1529	0.592	0.584	1.284
3/4	15	0.072	0.2884	0.1963	0.1587	0.520	0.606	1.238
3/4	16	0.065	0.3019	0.1963	0.1623	0.476	0.620	1.210
3/4	17	0.058	0.3157	0.1963	0.1660	0.428	0.634	1.183
3/4	18	0.049	0.3339	0.1963	0.1707	0.367	0.652	1.150
3/4	20	0.035	0.3632	0.1963	0.1780	0.269	0.680	1.103
7/8	10	0.134	0.2892	0.2291	0.1589	1.061	0.607	1.441
7/8	11	0.120	0.3166	0.2291	0.1662	0.969	0.635	1.378
7/8	12	0.109	0.3390	0.2291	0.1720	0.891	0.657	1.332
7/8	13	0.095	0.3685	0.2291	0.1793	0.792	0.685	1.277
7/8	14	0.083	0.3948	0.2291	0.1856	0.704	0.709	1.234
7/8	16	0.065	0.4359	0.2291	0.1950	0.561	0.745	1.174
7/8	18	0.049	0.4742	0.2291	0.2034	0.432	0.777	1.126
7/8	20	0.035	0.5090	0.2291	0.2107	0.313	0.805	1.087
1	8	0.165	0.3526	0.2618	0.1754	1.462	0.670	1.493
1	10	0.134	0.4208	0.2618	0.1916	1.237	0.732	1.366
1	11	0.120	0.4536	0.2618	0.1990	1.129	0.760	1.316
1	12	0.109	0.4803	0.2618	0.2047	1.037	0.782	1.279
1	13	0.095	0.5153	0.2618	0.2121	0.918	0.810	1.235
1	14	0.083	0.5463	0.2618	0.2183	0.813	0.834	1.199
1	15	0.072	0.5755	0.2618	0.2241	0.714	0.856	1.167
1	16	0.065	0.5945	0.2618	0.2278	0.649	0.870	1.119

Dimensional Data For Commercial Tubing

TABLE 8.1 (CONTINUED)

Dimensional Data for Commercial Tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.²)	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Length, Steel (lb.)	ID Tubing (in.)	OD/ID
1	18	0.049	0.6390	0.2618	0.2361	0.496	0.902	1.109
1	20	0.035	0.6793	0.2618	0.2435	0.360	0.930	1.075
1-1/4	7	0.180	0.6221	0.3272	0.2330	2.057	0.890	1.404
1-1/4	8	0.165	0.6648	0.3272	0.2409	1.921	0.920	1.359
1-1/4	10	0.134	0.7574	0.3272	0.2571	1.598	0.982	1.273
1-1/4	11	0.120	0.8012	0.3272	0.2644	1.448	1.010	1.238
1-1/4	12	0.109	0.8365	0.3272	0.2702	1.329	1.032	1.211
1-1/4	12	0.095	0.8825	0.3272	0.2773	1.173	1.060	1.179
1-1/4	14	0.083	0.9229	0.3272	0.2838	1.033	1.084	1.153
1-1/4	16	0.065	0.9852	0.3272	0.2932	0.823	1.120	1.116
1-1/4	18	0.049	1.042	0.3272	0.3016	0.629	1.152	1.085
1-1/4	20	0.035	1.094	0.3272	0.3089	0.456	1.180	1.059
1-1/2	10	0.134	1.192	0.3927	0.3225	1.955	1.232	1.218
1-1/2	12	0.109	1.291	0.3927	0.3356	1.618	1.282	1.170
1-1/2	14	0.083	1.398	0.3927	0.3492	1.258	1.334	1.124
1-1/2	16	0.065	1.474	0.3927	0.3587	0.996	1.370	1.095
2	11	0.120	2.433	0.5236	0.4608	2.410	1.760	1.136
2	13	0.095	2.573	0.5236	0.4739	1.934	1.810	1.105
2-1/2	9	0.148	3.815	0.6540	0.5770	3.719	2.204	1.134

Source: Courtesy of the Tubular Exchanger Manufacturers Association.

Tube Materials

- **Materials selection** and compatibility between construction materials and working fluids are important issues, in particular with regard to **corrosion** and/or operation at **elevated temperatures**.
- Requirement for low cost, light weight, high conductivity, and good joining characteristics often leads to the selection of aluminum for the heat transfer surface.
- On the other side, stainless steel is used for food processing or fluids that require corrosion resistance.
- In general, one of the selection criteria for exchanger material depends on the corrosiveness of the working fluid.
- A summary Table is provided as a reference fo rcorrosive and non-corrosive environments

Materials for Corrosive & Noncorrosive Service

Material	Heat Exchanger Type or Typical Service
Noncorrosive Service	
Aluminum and austenitic chromium–nickel steel	Any heat exchanger type, $T < -100°C$
$3\frac{1}{2}$ Ni steel	Any heat exchanger type, $-100 < T < -45°C$
Carbon steel (impact tested)	Any heat exchanger type, $-45 < T < 0°C$
Carbon steel	Any type of heat exchanger, $0 < T < 500°C$
Refractory-lined steel	Shell-and-tube, $T > 500°C$
Corrosive Service	
Carbon steel	Mildly corrosive fluids; tempered cooling water
Ferritic carbon–molybdenum and chromium–molybdenum alloys	Sulfur-bearing oils at elevated temperatures (above 300°C); hydrogen at elevated temperatures
Ferritic chromium steel	Tubes for moderately corrosive service; cladding for shells or channels in contact with corrosive sulfur bearing oil
Austenitic chromium–nickel steel	Corrosion-resistant duties
Aluminum	Mildly corrosive fluids
Copper alloys: admiralty, aluminum brass, cupronickel	Freshwater cooling in surface condensers; brackish and seawater cooling
High nickel–chromium–molybdenum alloys	Resistance to mineral acids and Cl-containing acids
Titanium	Seawater coolers and condensers, including PHEs
Glass	Air preheaters for large furnaces
Carbon	Severely corrosive service
Coatings: aluminum, epoxy resin	Exposure to sea and brackish water
Linings: lead and rubber	Channels for seawater coolers
Linings: austenitic chromium–nickel steel	General corrosion resistance

Source: Data from Lancaster (1998).