PROC 5071: Process Equipment Design I

Heat Exchangers

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1 Preliminaries

1.1 What do you need to know about heat exchangers?

Typical questions that you might have to answer about heat exchangers:

- What type of heat exchanger do I need in a new process?
- What size heat exchanger do I need?
- What are the heat transfer coefficients?
- Is an existing heat exchanger operating properly?
- How much fouling can the exchanger withstand before the performance is significantly degraded? How do I know if this has occurred with an operating heat exchanger?

1.2 What is a heat exchangers?

- A heat exchangers does exactly what its name refer to - exchanges heat between two streams and thus heating one and cooling the other.

1.3 What causes heat transfer?

- Thermal energy is related to the temperature of a matter.
- For a given material and mass, the higher the temperature, the greater its thermal energy.
- When two bodies are at different temperatures, thermal energy transfers from the one with higher temperature to the one with lower temperature.

1.4 The modes of heat transfer

Conduction : transfer of heat through solids or stationery fluids.

Convection : uses the movement of fluids to transfer heat.

Radiation : does not require a medium for transferring heat; this mode involves the electromagnetic radiation emitted by an object for exchanging heat.

Any energy exchange between bodies occurs through one of these modes or a combination of two or all.

2 Types of heat exchangers

There are three basic types of heat exchangers

1. Direct contact heat exchanger
2. Regenerator
3. Recuperator

2.1 Direct contact heat exchanger

- The hot and cold streams are brought into direct contact.
- These are particularly common when one stream is solid or entrained with a solid (air dryers, etc.) or for vapor-liquid systems (spray dryers, cooling towers, etc.).

- Use of liquid-liquid systems is limited to immiscible pairs.

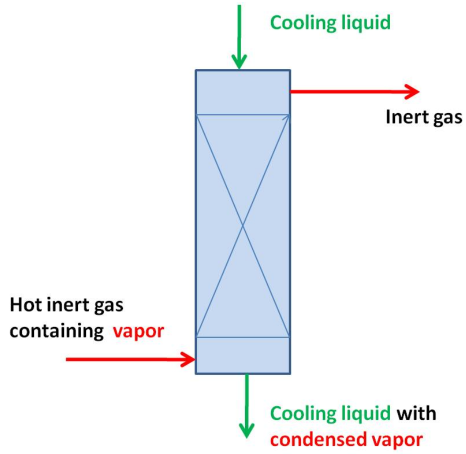

Figure 1: Schematic of a direct contact heat exchanger.

2.2 Regenerator

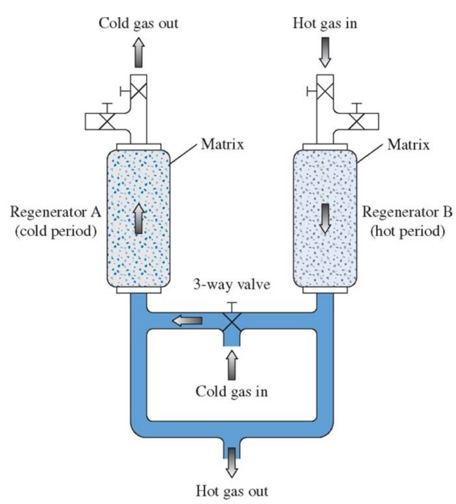

Figure 2: Schematic of regenerating heat exchanger.

- A regenerating exchanger transfers heat in steps: first from the hot fluid to a storage medium and subsequently from the storage medium to the cold fluid.

- A sand tank or rotary slab may be used as the storage phase.

2.3 Recuperator

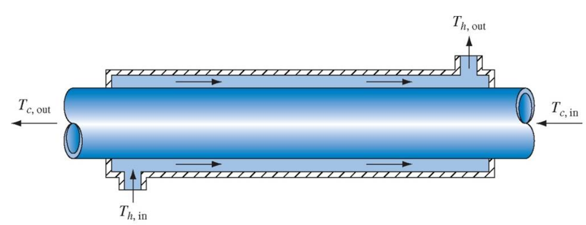

Figure 3: Schematic of a recuperator.

- Hot and cold fluids are separated by a wall and heat is transferred by conduction through the wall.
- Most common heat exchangers in industries.

3 Factors considered for choice of heat exchangers

Different factors affect the design of heat exchangers.

- Cost
- Efficiency
- Space
- Materials
- Maintenance
- Ease of construction

4 Commonly used industrial heat exchangers

- Double pipe
- Spiral
- Finned
- Compact
- Shell and tube

4.1 Double pipe heat exchanger

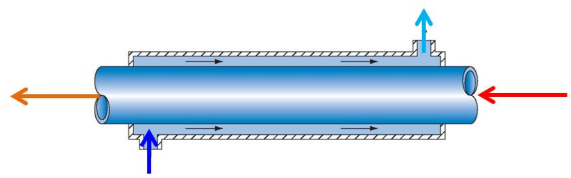

Figure 4: Schematic of a double pipe heat exchanger.

- The exchangers are made of concentric pipes
- One fluid flows through the outer pipe, other through the inner
- Two types based on the flow direction
  - Parallel flow: both fluid flow in the same direction
  - Countercurrent flow: fluids flow in opposite direction

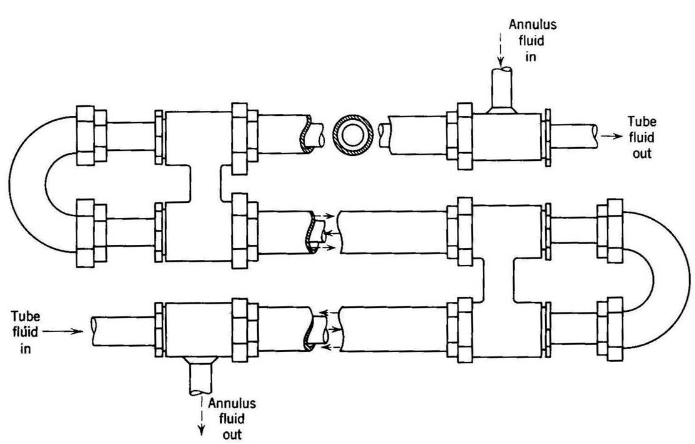

Figure 5: Typical design of a double pipe heat exchanger.

4.2 Spiral heat exchangers

- Constructed from sheets of metal wound in a circular fashion.
- The fluids flow in adjacent chambers between the sheets of metal.
- Easy to fabricate

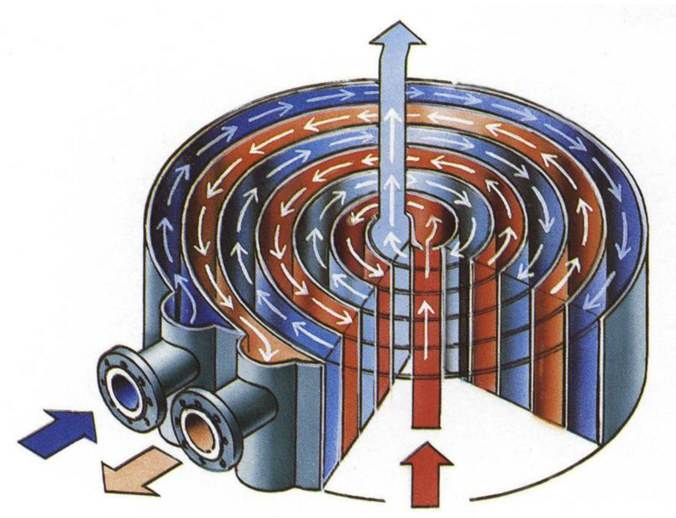

Figure 6: Schematic of a spiral heat exchanger.

4.3 Finned and compact heat exchangers

- contain fins on one heat exchange surface to increase the heat exchange surface.
- Home hot water heating system is an example of this type.

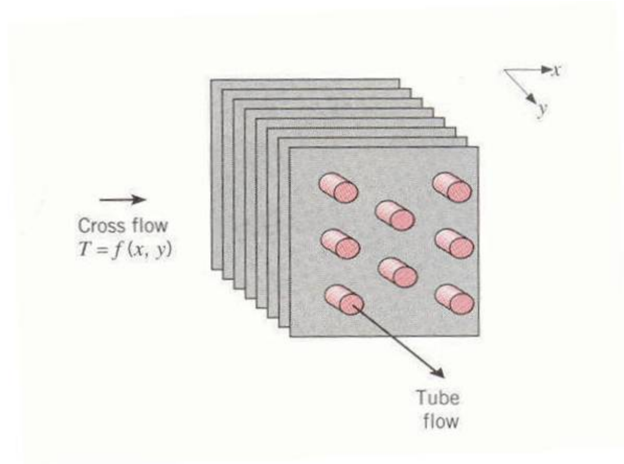

Figure 7: Schematic of a finned heat exchanger.

4.4 Shell and tube heat exchangers

- Most common in industries
- Provide large heat transfer area in small space
- Can be operated at high pressure
- Can be made of various materials

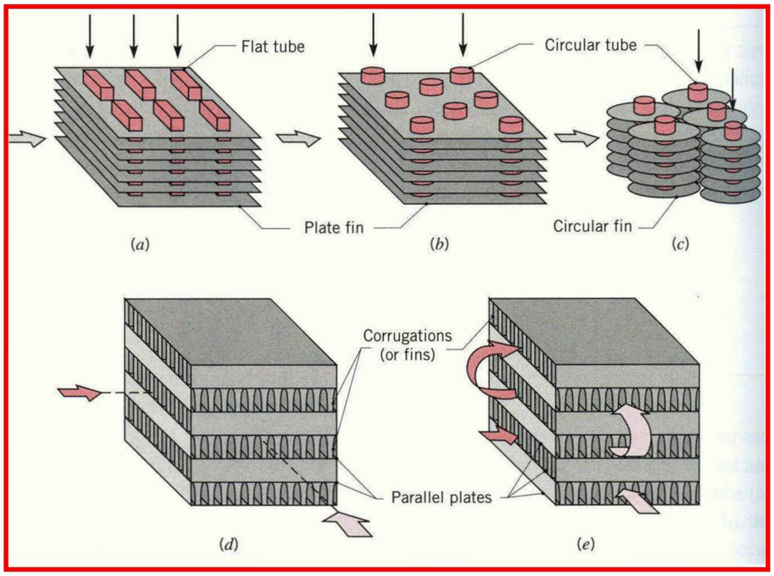

Figure 8: Various design of compact heat exchangers.

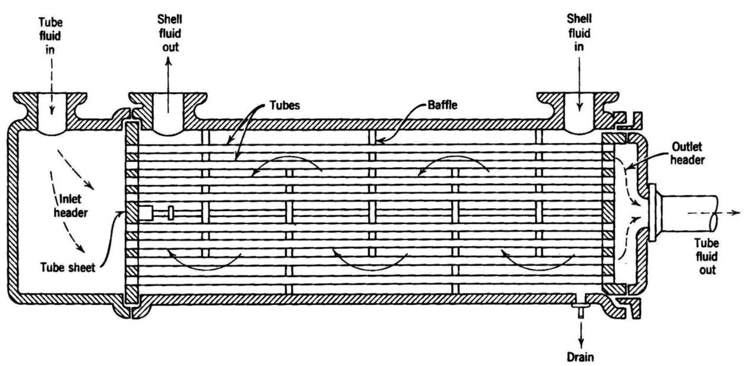

Figure 9: Schematic of a shell and tube heat exchanger.

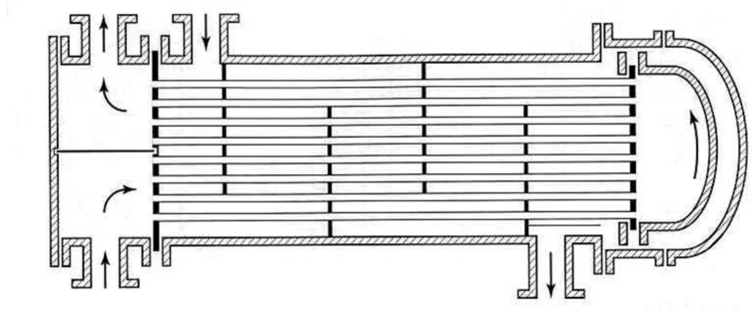

Figure 10: 1-2 Shell and tube heat exchanger.

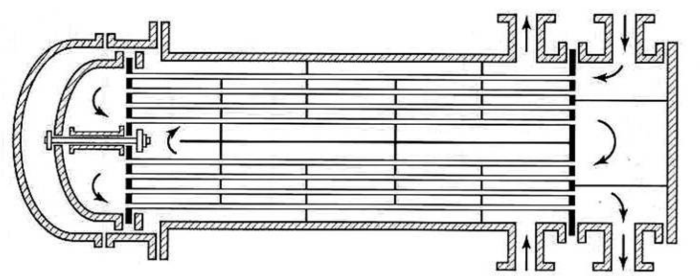

Figure 11: 2-4 Shell and tube heat exchanger.

5 Structure of shell and tube heat exchangers

5.1 Single and multi pass exchangers

5.2 Baffle types

1. Segmental
2. Segmental with strip
3. Disk and doughnut

4. Orifice

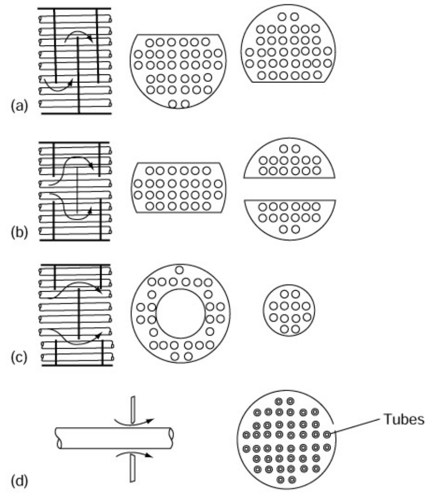

Figure 12: Baffle configurations for shell and tube heat exchangers.

6 Heat exchanger design basics

6.1 Heat transfer theories

$$Q = UA\Delta T_m$$

where,

- Q : heat transferred per unit time (W)
- U : the overall heat transfer coefficient $(W/m^2.^oC)$
- A : heat transfer area (m^2)
- ΔT_m : the mean temperature difference, the

temperature driving force (^oC)

6.2 Required calculations

Case 1: For a new HE design

- Heat required, Q, and ΔT is given
- Heat transfer coefficient, U and size, A of the HE to be calculated

Case 2: For an existing HE

- Size, A is given
- May require to re-estimate U
- For a given Q, achievable ΔT to be calculated

6.3 An iterative procedure

- Design of a shell and tube heat exchanger is an iterative procedure. Why?
- Heat transfer coefficients and pressure drop depend on many geometric factors that are de-

termined as part of the design process. These factors include

- ○ shell and tube diameters
- ○ tube length and layout
- ○ baffle type and spacing
- ○ number of tube and shell passes

6.4 Given and assumed conditions

- Inlet conditions
  - ○ Temperatures, pressures, compositions, flow rates and phase conditions of the two inlet streams
- If a heating or cooling utility is to be selected, it is selected from standard table along with its inlet and exit temperatures.

7 Design steps

7.1 Allocation of the streams

- Based on the properties and conditions of the two streams, decision is made on which stream goes to the tube side and which to the shell side.

7.2 Overall energy balance

- Based on the information about the streams, an overall energy balance is carried out to calculate the heat duty and the remaining exiting conditions of the streams.
- If a utility stream is used, its flow rate is calculated from an overall energy balance.

7.3 Checking for temperature crossover

- Assuming a 1-1 exchanger with countercurrent flow, a check is made that the second law of thermodynamics is not violated and a reasonable ΔT exists at the ends of the exchanger.

- If a phase change occurs on either side of the exchanger a heating or cooling curve is calculated to check for any temperature crossover.

7.4 Initiation

- An overall heat transfer coefficient is assumed from the standard range of values for the given system.
- The LMTD is calculated.
- A preliminary estimate of the heat exchanger area is made.
- If the estimated area is too large ($> 8000\ ft^2$), multiple heat exchanger with the same area are used in parallel.

7.5 Configuration

- The temperature correction factor, F_T is calculated.
- The configuration of the heat exchanger (number of shell and tube passes) is selected to get a desired F_T.
- It is desirable to have $F_T > 0.85$.
- $F_T < 0.75$ is generally unacceptable.
- For a given number of shell pass, the value of F_T is not affected significantly on the number of tube passes.
- The more shell passes, the higher is the value of F_T.

7.6 Tube side calculation

- A tube velocity in the range of 1 to 10 ft/s is selected, with a typical value being $4ft/s$.
- The total required inside tube cross-sectional area is then calculated to maintain the desired velocity.

- A tube size is selected and the total number of tubes per pass is calculated to obtain the required area.
- A tube length is selected and the number of tubes per exchanger is calculated based on the required heat transfer area.
- The tube side velocity and the tube length may be required to adjust to obtain an integer number of tube passes.
- Based on the fluid velocity and flow conditions, the tube side heat transfer coefficient is calculated.

7.7 Shell side calculation

- Shell side calculation requires determining the shell dimensions based on the number and length of tubes.
- Shell side calculations also involve baffle configuration.
- Minimum baffle spacing is 20% of the shell inside diameter and maximum 100%.

- Segmental baffle is the most common, with a segment height equal to 75% of the shell inside diameter; this is referred to as 25% baffle cut.
- Maximum baffle cut is 45%.
- Once the shell dimensions and baffle configuration are selected, the shell side heat transfer coefficient can be calculated from the flow and liquid properties.

7.8 Overall heat transfer coefficient

- From the calculated tube and shell side heat transfer coefficients, the overall heat transfer coefficient can be calculated using standard fouling factor values for the fluids under consideration.
- If the estimated overall heat transfer coefficient matches with the initially assumed value, the calculation is terminated.
- If not, the calculated values is used as the guess and the entire calculation is repeated.

7.9 Summary of the design steps

1. Specify fluid flow rates, temperatures and required heat duty.
2. Select the type of exchanger to be used.
3. Choose a trial value for the overall coefficient, U.
4. Calculate the mean temperature difference, ΔT_m.
5. Calculate the area required.
6. Decide the exchanger layout.
7. Calculate the individual heat transfer coefficients.
8. Calculate U and compare with the trial value. If there is a difference, substitute the calculation for the estimated value and return to step 4.
9. Calculate ΔP; if unsatisfactory, return to steps 6 or 2, in that order of preference.
10. Optimize the design: repeat steps 3 to 10, as necessary, to determine the cheapest exchanger that will satisfy the duty.

8 Estimation of the tube side heat transfer coefficient

The approach described below is applicable for

- turbulent flow in straight, smooth ducts, pipes and tubes of circular cross section.
- shell and tube as well as double pipe heat exchangers.
- convective heat transfer with no phase changes.

8.1 A generalized equation

A general correlation of the following form is used for estimating the tube side heat transfer coefficient.

$$Nu = CRe^aPr^b\left(\frac{\mu}{\mu_w}\right)^c$$

with

$$
\begin{aligned}
Nu &= \frac{h_t d_i}{k_f} \\
Re &= \frac{d_i u_t \rho}{\mu} \\
Pr &= \frac{c_p \mu}{k_f}
\end{aligned}
$$

8.2 Commonly used values of the constants

The values of the constants depend on flow regime and fluid viscosity. A set of commonly used values are

$$
\begin{aligned}
a &= 0.8 \\
b &= \begin{cases} 0.3 & \text{for cooling} \\ 0.4 & \text{for heating} \end{cases} \\
c &= 0.14 \\
C &= \begin{cases} 0.021 & \text{for gases} \\ 0.023 & \text{for nonviscous liquids} \\ 0.027 & \text{for viscous liquids} \end{cases}
\end{aligned}
$$

8.3 Expression using the j-factor

To express the correlation using a general equation for both laminar and turbulent flow, the heat transfer j-factor is used. In terms of the j-factor

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

The advantage of using the j-factor is that its values are expressed as a function of Re for a wide range of Re covering both the laminar and the turbulent flow.

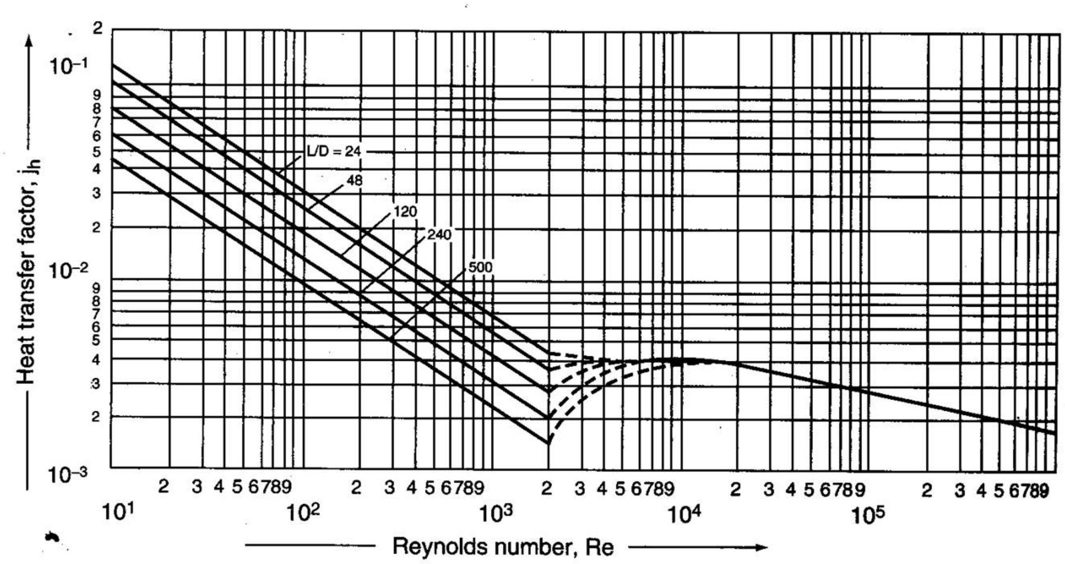

Figure 12.23. Tube-side heat transfer factor.

9 Estimation of the shell side heat transfer coefficient

- Accurate predictions of shell side heat transfer coefficient is difficult
- Geometry is complex and so is the flow pattern
- A number of correlations are available, none are as accurate as those for tube flow
- All are based on crossflow past an ideal tube bank
- Corrections are made for flow distortions due to baffles, leakage and bypassing

9.1 A general correlation

There are a number of correlations; we will us the following general correlation for the shell side.

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

with

$$Nu = \frac{h_s d_e}{k_f}$$
$$Re = \frac{d_e u_s \rho}{\mu} = \frac{d_e G_s}{\mu}$$
$$Pr = \frac{c_p \mu}{k_f}$$

9.2 Equivalent diameter

$$d_e = \frac{4 \times cross\ sectional\ area\ for\ flow}{wetted\ perimeter}$$

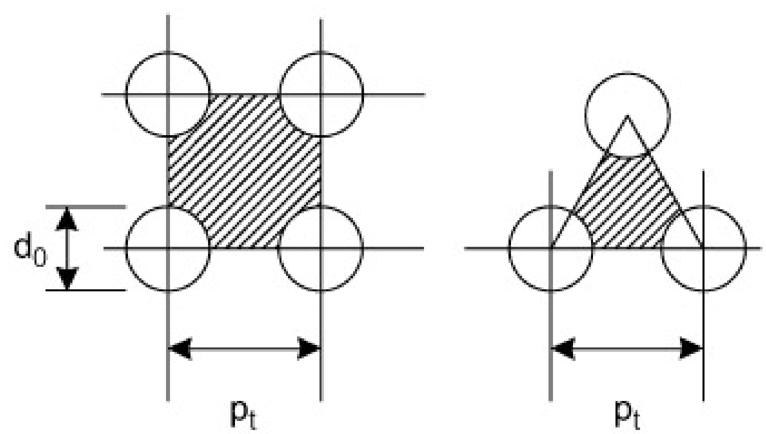

Figure 13: Tube configuration: square pitch (left), triangular pitch (right).

For square pitch

$$d_e = \frac{1.27}{d_o}(p_t^2 - 0.785d_o^2)$$

For triangular pitch

$$d_e = \frac{1.1}{d_o}(p_t^2 - 0.907d_o^2)$$

9.3 Shell side mass flux

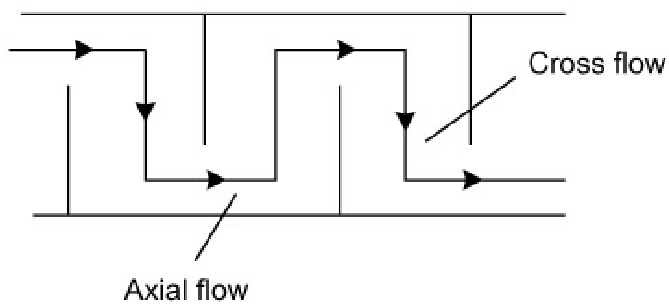

Figure 14: Simplified flow pattern of liquid in the shell.

The cross flow area is given by

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

Shell side mass flux

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

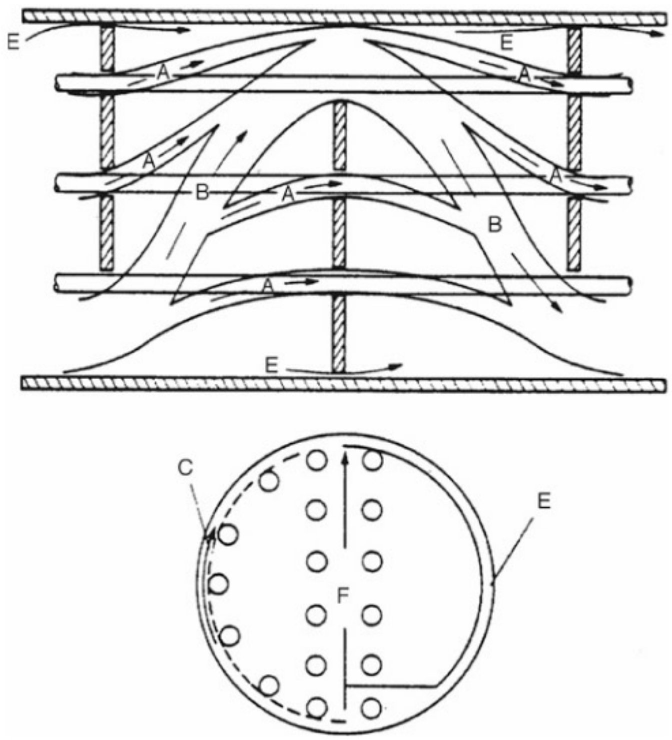

Figure 15: Flow pattern of liquid in the shell.

The fluid velocity is given by

$$u_S = \frac{G_S}{\rho}$$

The j_h factor is a function of Re and can be obtained in the literature in graphical forms.

10 Overall heat transfer coefficient

- Overall coefficient is obtained from the reciprocal of overall resistance

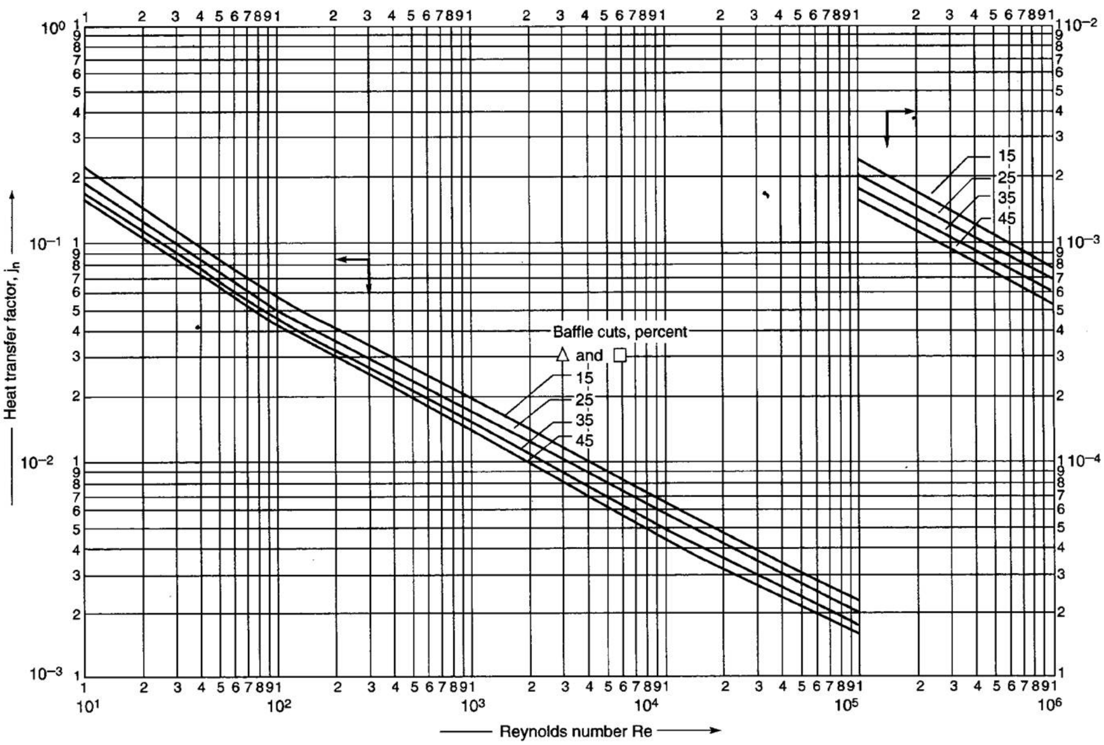

Figure 12.29. Shell-side heat transfer factors, segmental baffles.

- Overall resistance is obtained as the sum of individual resistances

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{x_w}{k_w}\frac{d_o}{d_{LM}} + \frac{1}{h_i}\frac{d_o}{d_i}$$

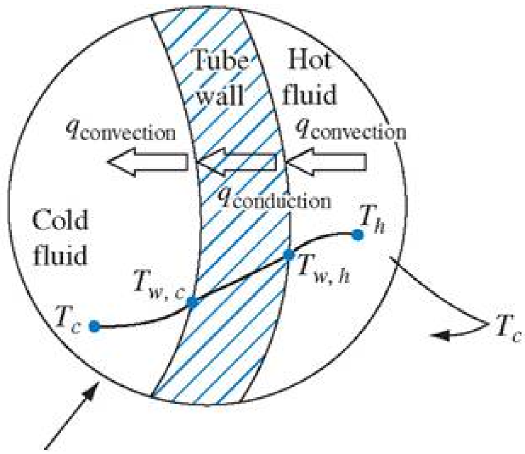

Figure 16: The heat transfer process and associated parameters.

10.1 Fouling resistance to heat transfer

Overall heat transfer coefficient considering the fouling factors

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{x_w}{k_w}\frac{d_o}{d_{LM}} + \frac{1}{h_i}\frac{d_o}{d_i} + \frac{1}{h_{id}}\frac{d_o}{d_i}$$

In terms of tube diameters: $x_w = \frac{d_0 - d_i}{2}$ and $d_{LM} = \frac{d_o - d_i}{\ln(d_o/d_i)}$

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

Table 12.2: Fouling Factors (Coefficients), Typical Values

Fluid	Coefficient (W/m² °C)	Factor (resistance) (m² °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 vapors	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

10.2 Fouling factor values

11 Pressure drop in shell and tube heat exchangers

11.1 Pressure drop in the tube side

- For flow of liquid or gas
- With no phase change

$$-\Delta P = \frac{f_D G^2 L}{2 g_c \rho D_i} \tag{1}$$

- $g_c = 32.17 lb_m/lb_f.s^2$
- $N_{Re} > 10,000$ with a smooth wall, f_D can be calculated or Fanning friction factor chart can be used.
- Eq. 1 accounts for only skin friction
- Pressure drop also occurs
  - by contraction as fluid enters the tube from header
  - by expansion as fluid leaves the the pipe to the header
  - by reversal in fluid flow direction for multiple passes
  - as fluid enters the heat exchanger from a nozzle
  - as fluid passes out through a nozzle
- For non-isothermal flow in a multiple tube pass exchanger

$$-\Delta P = K_P \frac{N_P f_D G^2 L}{2 g_c \rho D_i \phi} \tag{2}$$

where

- ◦ K_P; correction factor for contraction, expansion and flow reversal
- ◦ N_P: number of tube passes
- ◦ Φ: correction factor for non-isothermal turbulent flow

$$\phi = 1.02 \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3}$$

- ◦ A reasonable value for K_P is 1.2.

- If the heat exchanger is vertical and the fluid flow upwards, the outlet pressure is further reduced by the height of the heat exchanger times the fluid density
- If the flow is downward, the outlet pressure is increased by the same amount.

11.2 Pressure drop in the shell side

- A preliminary estimate is obtained by the method by Grimison

- The equation is modified Fanning equation

$$-\Delta P = K_S \frac{2N_R f' G_S^2 L}{g_c \rho \phi} \tag{4}$$

- where
  - K_S is the correction factor for friction due to inlet and outlet nozzles, the presence of baffles causing flow reversal, recrossing of tubes and variation in cross-sectional area for flow
  - K_S can be approximately taken as $1.1 \times (1 + no.\ of\ baffles)$
  - N_R is the number of tube rows across which the shell fluid flows. For 25% baffle cut, N_R can be takes as 50% of the number of tubes at the centre plane.
- f' is the modified friction factor

$$f' = b \left(\frac{D_o G_S}{\mu_b} \right)^{-0.15} \tag{5}$$

where

- for triangular pitch

$$b = 0.023 + \frac{0.11}{(x_T - 1)^{1.08}} \tag{6}$$

- for tube in line, square pitch

$$b = 0.044 + \frac{0.08x_L}{(x_T - 1)^{0.43+1.13/x_L}} \tag{7}$$

- x_T is the ratio of the pitch traverse to flow to tube OD
- x_L is the ratio of the pitch parallel to flow to tube OD
- For square pitch $x_T = x_L$

11.3 Logarithmic mean temperature difference, $LMTD$

- Logarithmic mean temperature difference, $LMTD$, is used as ΔT

$$q = UA\Delta T_m$$

- $LMTD$ is defined as

$$\Delta T_{lm} = \frac{\Delta T_a - \Delta T_b}{\ln(\Delta T_a / \Delta T_b)}$$

$$\Delta T_a = T_1 - t_2 \quad \Delta T_b = T_2 - t_1$$

- Where ΔT_a and ΔT_b are the temperature differences at the two ends. When they are nearly equal, their arithmetic average can be used for ΔT_{lm}
- For heat exchangers with multiple passes

$$\Delta T_m = F \Delta T_{lm}$$

where F is a correction factor

11.4 Correction factor, F

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$S = \frac{t_2 - t_1}{T_1 - t_1}$$

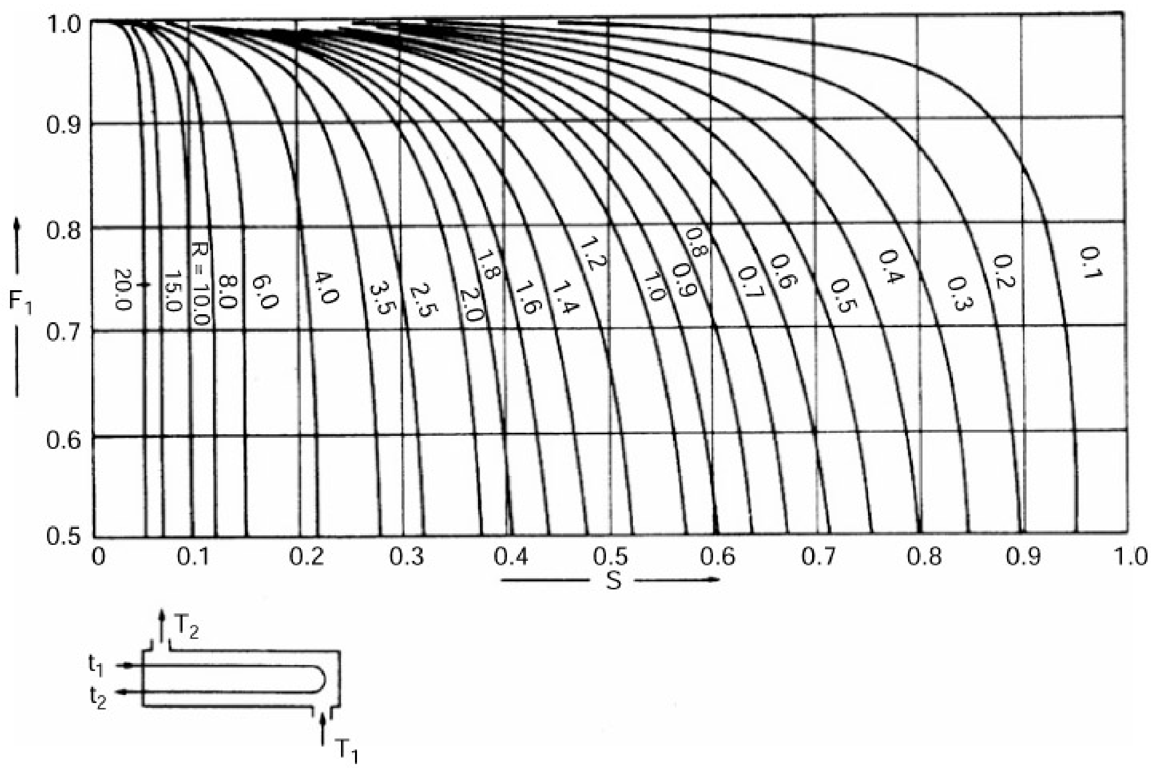

Figure 17: LMTD correction factor for 1-2x shell and tube heat exchangers.

12 Fluid allocation

Following is a list of main fluid properties considered for fluid allocation.

- Corrosiveness
- Fouling tendency
- Viscosity

The operating conditions considered for fluid allocation are:

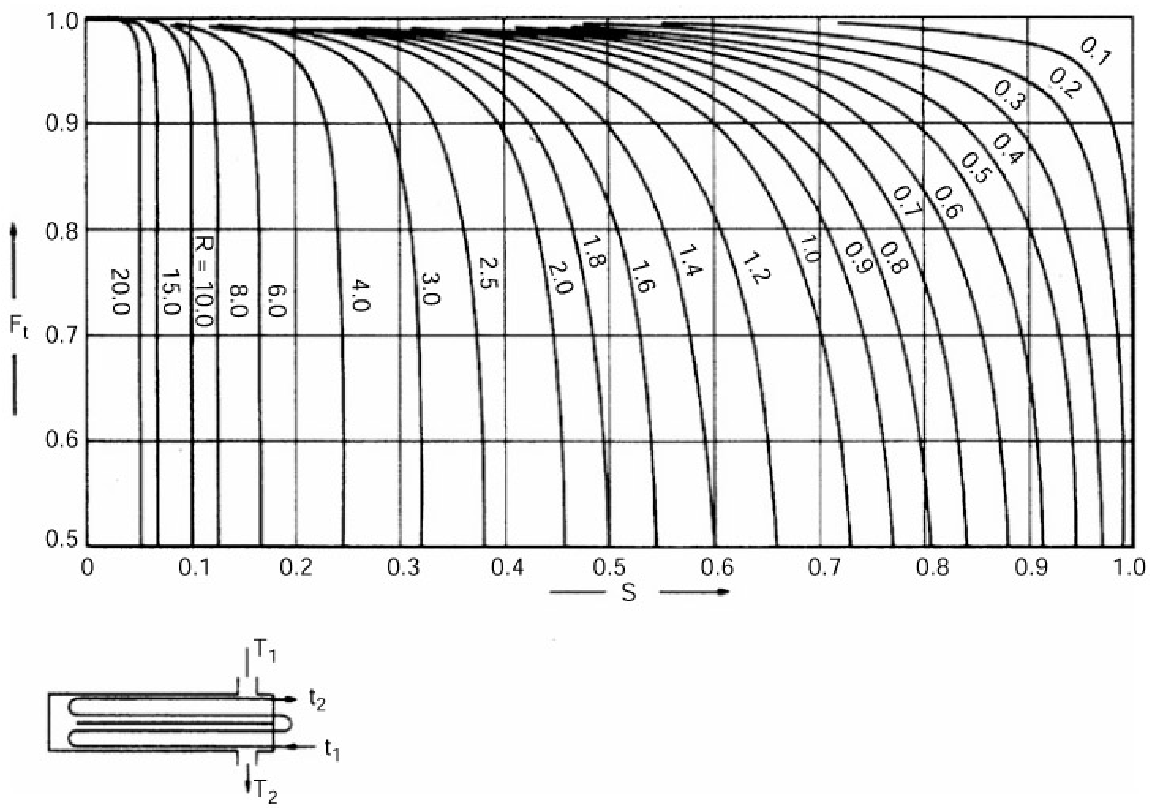

Figure 18: LMTD correction factor for 2-2x shell and tube heat exchangers.

- Fluid temperature
- Operating pressure
- Pressure drop
- Flow rates

12.1 Allocation based on corrosion

More corrosive fluid is allocated to the tube side.

- Reduce the cost of expensive alloy or clad component

- Less exposed area for corrosion
- Shell, baffles are not exposed

12.2 Allocation based on fouling tendency of fluid

Fluid having fouling tendency is placed in the tube side.

- Higher allowable velocity in the tubes reduce fouling
- Better control over deign fluid velocity
- Tube side is easier to clean
- Less exposed area for fouling

12.3 Allocation based on flow characteristic

- If the flow is turbulent, more viscous fluid goes to the shell side
  - In general, higher heat transfer coefficient can be obtained with the more viscous fluid in the shell side

- Critical Re for shell side is ≈ 200

- If turbulent flow cannot be achieved, more viscous fluid goes to the tube side
  - Prediction of heat transfer coefficient for the tube side is more certain

12.4 Allocation based on fluid temperature

For high temperature fluid, tube side is the better choice.

- If special alloys are required for fluid with higher temperature, putting it in the shell side will increase cost
- High temperature fluid in the shell side increases shell temperature
  - require proper arrangement to prevent heat loss
  - safety issues are needed to consider

12.5 Allocation based on fluid pressure and allowable pressure drop

Fluids at high pressure, lower allowable pressure drop and higher flow rate go to the tube side.

- High pressure tubes are cheaper to construct and operate than high pressure shell
- Tube side causes less pressure drop than the shell side
- For the same pressure drop, higher heat transfer coefficients are obtained for the tube side than the shell side
- Allocating the fluids with lower flow rate to the shell side normally gives the most economical design

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