

Module #1

PROCESS DESIGN OF HEAT EXCHANGER: TYPES OF HEAT EXCHANGER, PROCESS DESIGN OF SHELL AND TUBE HEAT EXCHANGER, CONDENSER AND REBOILERS

- 1. PROCESS DESIGN OF SHELL AND TUBE EXCHANGER FOR SINGLE PHASE HEAT TRANSFER**
 - 1.1. Classification of heat exchangers**
 - 1.2. Thermal design considerations**
 - 1.2.1. Shell**
 - 1.2.2. Tube**
 - 1.2.3. Tube pitch, tube-layout and tube-count**
 - 1.2.4. Tube passes**
 - 1.2.5. Tube sheet**
 - 1.2.6. Baffles**
 - 1.2.7. Fouling Considerations**
 - 1.2.8. Selection of fluids for tube and the shell side**
 - 1.3. Process (thermal) design procedure**
 - 1.4. Design problem**

- 2. PROCESS DESIGN OF SHELL AND TUBE EXCHANGER FOR TWO PHASE HEAT TRANSFER**
 - 2.1. Condenser**
 - 2.1.1. Types of condensers**
 - 2.1.2. Condenser design**
 - 2.1.2.1. Mean temperature difference**
 - 2.1.2.2. Calculation of heat transfer co-efficient during condensation**
 - 2.1.2.3. Pressure drop calculation**
 - 2.1.3. De-superheating and sub-cooling**

 - 2.2. Reboilers**
 - 2.2.1. Classification of reboilers**
 - 2.2.2. Design of Kettle reboiler**

Lecture 1: Heat Exchangers Classifications

1. PROCESS DESIGN OF SHELL AND TUBE EXCHANGER FOR SINGLE PHASE HEAT TRANSFER

1.1. *Classification of heat exchangers*

Transfer of heat from one fluid to another is an important operation for most of the chemical industries. The most common application of heat transfer is in designing of heat transfer equipment for exchanging heat from one fluid to another fluid. Such devices for efficient transfer of heat are generally called Heat Exchanger. Heat exchangers are normally classified depending on the transfer process occurring in them. General classification of heat exchangers is shown in the **Figure 1.1**.

Amongst of all type of exchangers, shell and tube exchangers are most commonly used heat exchange equipment. The common types of shell and tube exchangers are:

Fixed tube-sheet exchanger (non-removable tube bundle): The simplest and cheapest type of shell and tube exchanger is with fixed tube sheet design. In this type of exchangers the tube sheet is welded to the shell and no relative movement between the shell and tube bundle is possible (**Figure 1.2**).

Removable tube bundle: Tube bundle may be removed for ease of cleaning and replacement. Removable tube bundle exchangers further can be categorized in floating-head and U-tube exchanger.

- Floating-head exchanger: It consists of a stationery tube sheet which is clamped with the shell flange. At the opposite end of the bundle, the tubes may expand into a freely riding floating-head or floating tube sheet. A floating head cover is bolted to the tube sheet and the entire bundle can be removed for cleaning and inspection of the interior. This type of exchanger is shown in **Figure 1.3**.
- U-tube exchanger: This type of exchangers consists of tubes which are bent in the form of a 'U' and rolled back into the tube sheet shown in the **Figure 1.4**. This means that it will omit some tubes at the centre of the tube bundle

depending on the tube arrangement. The tubes can expand freely towards the ‘U’ bend end.

The different operational and constructional advantages and limitations depending on applications of shell and tube exchangers are summarized in **Table 1.1**. TEMA (USA) and IS: 4503-1967 (India) standards provide the guidelines for the mechanical design of unfired shell and tube heat exchangers. As shown in the **Table 1.1**, TEMA 3-digit codes specify the types of front-end, shell, and rear-end of shell and tube exchangers.

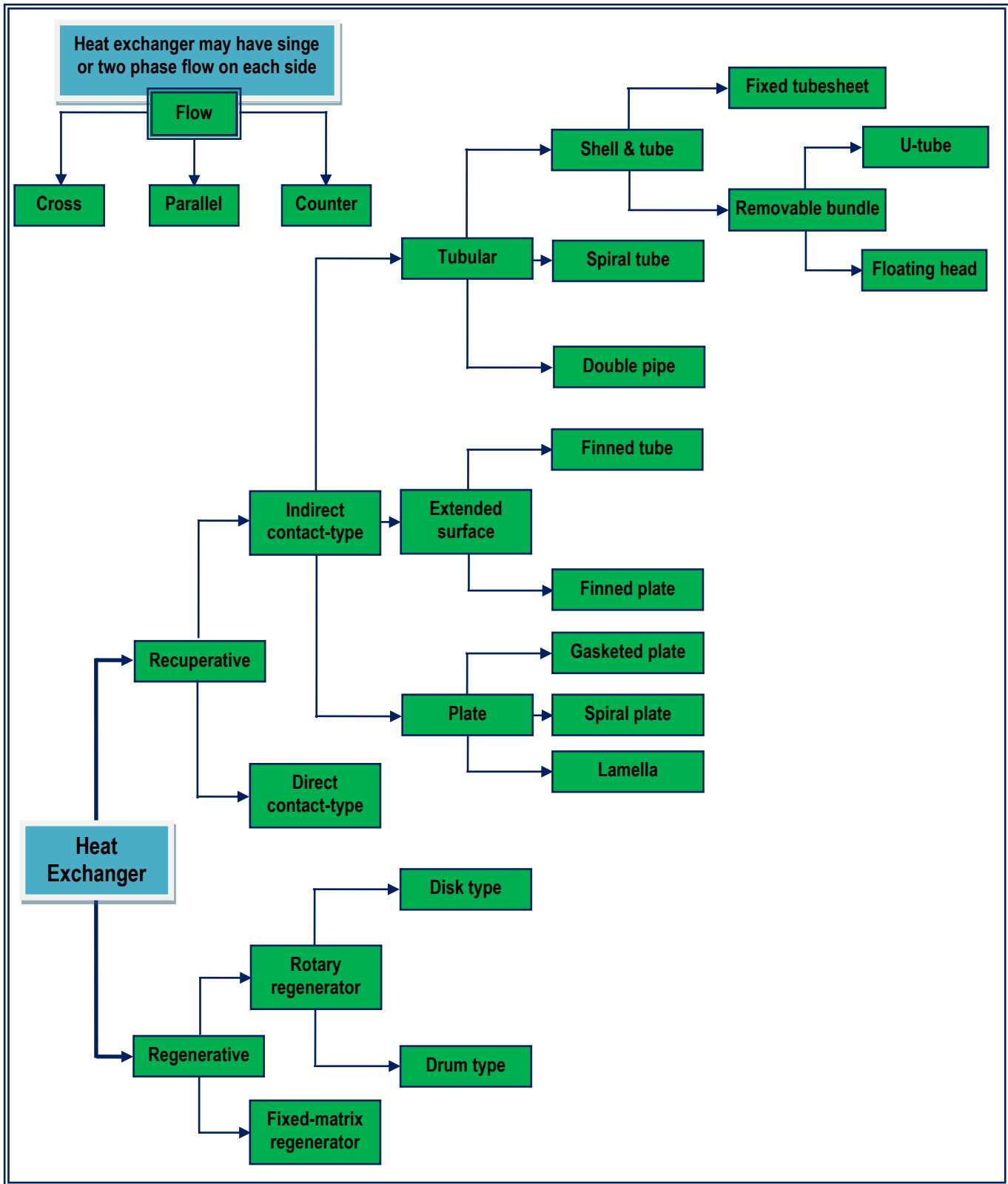


Figure 1.1. Classification of heat exchangers depending on their applications.

Table 1.1. Features of shell and tube type exchangers.

| <i>Shell and Tube Exchangers</i> | <i>Typical TEMA code</i> | <i>Advantages</i> | <i>Limitations</i> |
|----------------------------------|------------------------------|---|---|
| Fixed tube sheet | BEM, AEM, NEN | <p>Provides maximum heat transfer area for a given shell and tube diameter.</p> <p>Provides for single and multiple tube passes to assure proper velocity.</p> <p>Less costly than removable bundle designs.</p> | <p>Shell side / out side of the tubes are inaccessible for mechanical cleaning.</p> <p>No provision to allow for differential thermal expansion developed between the tube and the shell side. This can be taken care by providing expansion joint on the shell side.</p> |
| Floating-head | AEW, BEW, BEP, AEP, AES, BES | <p>Floating tube sheet allows for differential thermal expansion between the shell and the tube bundle.</p> <p>Both the tube bundle and the shell side can be inspected and cleaned mechanically.</p> | <p>To provide the floating-head cover it is necessary to bolt it to the tube sheet. The bolt circle requires the use of space where it would be possible to place a large number of tubes.</p> <p>Tubes cannot expand independently so that huge thermal shock applications should be avoided.</p> <p>Packing materials produce limits on design pressure and temperature.</p> |
| U-tube | BEU, AEU | <p>U-tube design allows for differential thermal expansion between the shell and the tube bundle as well as for individual tubes.</p> <p>Both the tube bundle and the shell side can be inspected and cleaned mechanically.</p> <p>Less costly than floating head or packed floating head designs.</p> | <p>Because of U-bend some tubes are omitted at the centre of the tube bundle.</p> <p>Because of U-bend, tubes can be cleaned only by chemical methods.</p> <p>Due to U-tube nesting, individual tube is difficult to replace.</p> <p>No single tube pass or true countercurrent flow is possible.</p> <p>Tube wall thickness at the U-bend is thinner than at straight portion of the tubes.</p> <p>Draining of tube circuit is difficult when positioned with the vertical position with the head side upward.</p> |

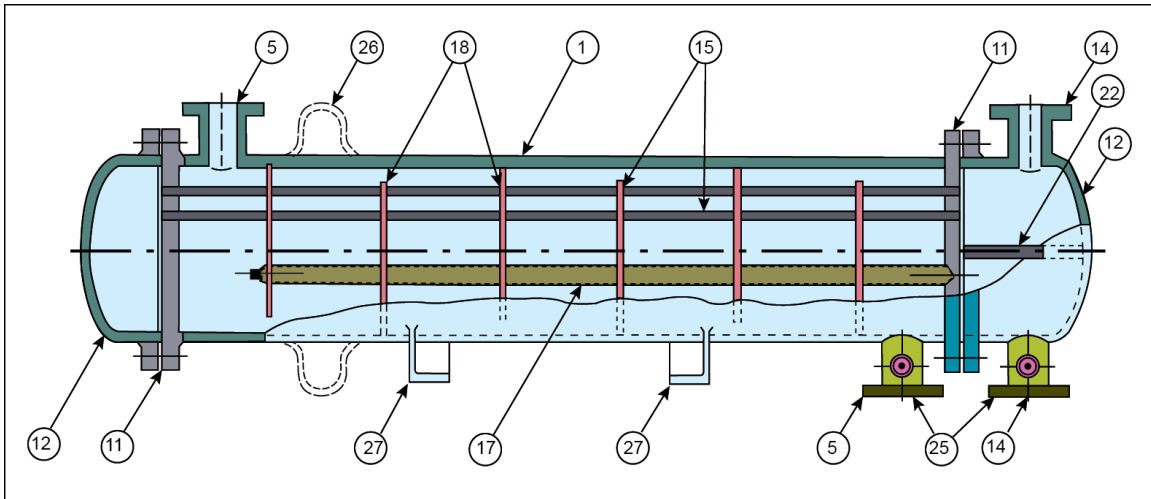


Figure 1.2. Fixed-tube heat exchanger ([1]).

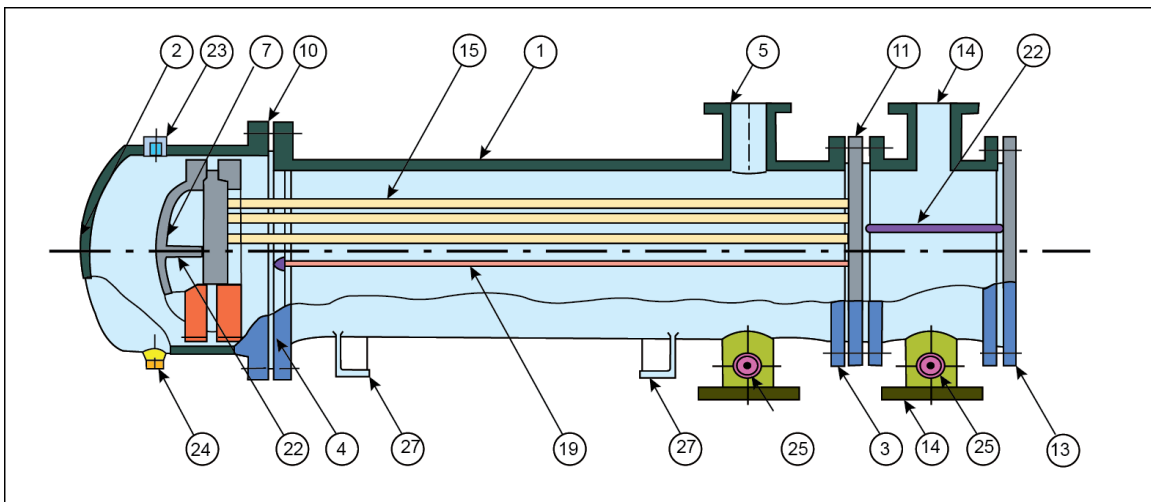


Figure 1.3. Floating-head heat exchanger (non-pull through type) [1].

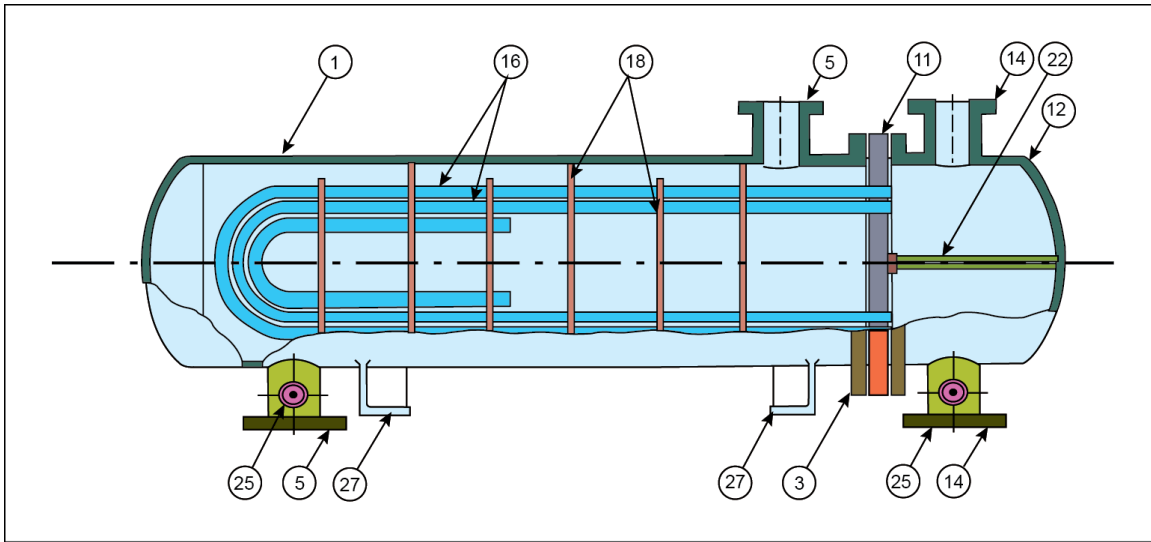


Figure 1.4. Removable U-tube heat exchanger [1].

Typical parts and connections shown in **Figures 1.2, 1.3 and 1.4 (IS: 4503-1967)** are summarized below.

| | |
|--------------------------------|---|
| 1. Shell | 16. Tubes (U-type) |
| 2. Shell cover | 17. Tie rods and spacers |
| 3. Shell flange (channel end) | 18. Transverse (or cross) baffles or support plates |
| 4. Shell flange (cover end) | 19. Longitudinal baffles |
| 5. Shell nozzle or branch | 20. Impingement baffles |
| 6. Floating tube sheet | 21. Floating head support |
| 7. Floating head cover | 22. Pass partition |
| 8. Floating head flange | 23. Vent connection |
| 9. Floating head gland | 24. Drain connection |
| 10. Floating head backing ring | 25. Instrument connection |
| 11. Stationary tube sheet | 26. Expansion bellows |
| 12. Channel or stationary head | 27. Support saddles |
| 13. Channel cover | 28. Lifting lugs |
| 14. Channel nozzle or branch | 29. Weir |
| 15. Tube (straight) | 30. Liquid level connection |

Lecture 2: Thermal Design Considerations

The flow rates of both hot and cold streams, their terminal temperatures and fluid properties are the primary inputs of thermal design of heat exchangers.

1.2. Thermal design considerations

Thermal design of a shell and tube heat exchanger typically includes the determination of heat transfer area, number of tubes, tube length and diameter, tube layout, number of shell and tube passes, type of heat exchanger (fixed tube sheet, removable tube bundle etc), tube pitch, number of baffles, its type and size, shell and tube side pressure drop etc.

1.2.1. Shell

Shell is the container for the shell fluid and the tube bundle is placed inside the shell. Shell diameter should be selected in such a way to give a close fit of the tube bundle. The clearance between the tube bundle and inner shell wall depends on the type of exchanger ([2]; page 647). Shells are usually fabricated from standard steel pipe with satisfactory corrosion allowance. The shell thickness of 3/8 inch for the shell ID of 12-24 inch can be satisfactorily used up to 300 psi of operating pressure.

1.2.2. Tube

Tube OD of 3/4 and 1'' are very common to design a compact heat exchanger. The most efficient condition for heat transfer is to have the maximum number of tubes in the shell to increase turbulence. The tube thickness should be enough to withstand the internal pressure along with the adequate corrosion allowance. The tube thickness is expressed in terms of BWG (Birmingham Wire Gauge) and true outside diameter (OD). The tube length of 6, 8, 12, 16, 20 and 24 ft are preferably used. Longer tube reduces shell diameter at the expense of higher shell pressure drop. Finned tubes are also used when fluid with low heat transfer coefficient flows in the shell side. Stainless steel, admiralty brass, copper, bronze and alloys of copper-nickel are the commonly used tube materials:

1.2.3. Tube pitch, tube-layout and tube-count

Tube pitch is the shortest centre to centre distance between the adjacent tubes. The tubes are generally placed in square or triangular patterns (pitch) as shown in the **Figure 1.5**.

The widely used tube layouts are illustrated in **Table 1.2**.

The number of tubes that can be accommodated in a given shell ID is called tube count. The tube count depends on the factors like shell ID, OD of tube, tube pitch, tube layout, number of tube passes, type of heat exchanger and design pressure.

1.2.4. Tube passes

The number of passes is chosen to get the required tube side fluid velocity to obtain greater heat transfer co-efficient and also to reduce scale formation. The tube passes vary from 1 to 16. The tube passes of 1, 2 and 4 are common in application. The partition built into exchanger head known as partition plate (also called pass partition) is used to direct the tube side flow.

Table 1.2. Common tube layouts.

| Tube OD, in | Pitch type | Tube pitch, in |
|-------------|------------|----------------|
| 3/4 | Square | 1 |
| 1 | | 1 1/4 |
| 3/4 | Triangular | 15/16 |
| 3/4 | | 1 |

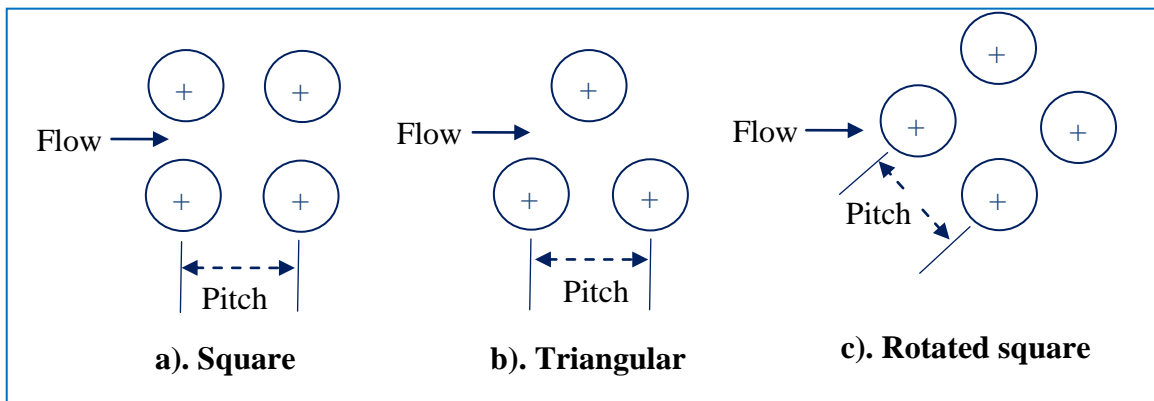


Figure 1.5. Heat exchanger tube-layouts.

1.2.5. Tube sheet

The tubes are fixed with tube sheet that form the barrier between the tube and shell fluids. The tubes can be fixed with the tube sheet using ferrule and a soft metal packing ring. The tubes are attached to tube sheet with two or more grooves in the tube sheet wall by ‘tube rolling’. The tube metal is forced to move into the grooves forming an excellent tight seal. This is the most common type of fixing arrangement in large industrial exchangers. The tube sheet thickness should be greater than the tube outside diameter to make a good seal. The recommended standards (IS:4503 or TEMA) should be followed to select the minimum tube sheet thickness.

1.2.6. Baffles

Baffles are used to increase the fluid velocity by diverting the flow across the tube bundle to obtain higher transfer co-efficient. The distance between adjacent baffles is called baffle-spacing. The baffle spacing of 0.2 to 1 times of the inside shell diameter is commonly used. Baffles are held in position by means of baffle spacers. Closer baffle spacing gives greater transfer co-efficient by inducing higher turbulence. The pressure drop is more with closer baffle spacing. The various types of baffles are shown in **Figure 1.6**. In case of cut-segmental baffle, a segment (called baffle cut) is removed to form the baffle expressed as a percentage of the baffle diameter. Baffle cuts from 15 to 45% are normally used. A baffle cut of 20 to 25% provide a good heat-transfer with the reasonable pressure drop. The % cut for segmental baffle refers to the cut away height from its diameter. **Figure 1.6** also shows two other types of baffles.

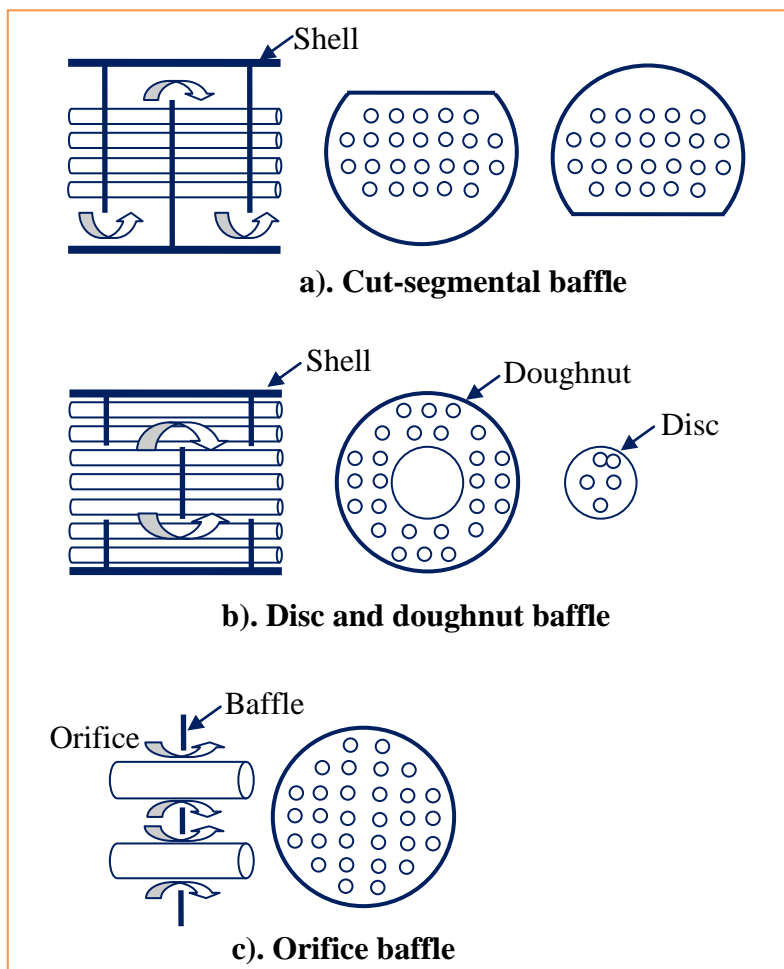


Figure 1.6. Different type of heat exchanger baffles: a). Cut-segmental baffle, b). Disc and doughnut baffle, c). Orifice baffle

1.2.7. Fouling Considerations

The most of the process fluids in the exchanger foul the heat transfer surface. The material deposited reduces the effective heat transfer rate due to relatively low thermal conductivity. Therefore, net heat transfer with clean surface should be higher to compensate the reduction in performance during operation. Fouling of exchanger increases the cost of (i) construction due to oversizing, (ii) additional energy due to poor exchanger performance and (iii) cleaning to remove deposited materials. A spare exchanger may be considered in design for uninterrupted services to allow cleaning of exchanger.

The effect of fouling is considered in heat exchanger design by including the tube side and shell side fouling resistances. Typical values for the fouling coefficients and resistances are summarized in **Table 1.3**. The fouling resistance (fouling factor) for petroleum fractions are available in the text book ([3]; page 845).

Table 1.3. Typical values of fouling coefficients and resistances ([2]; page 640).

| Fluid | Coefficient ($\text{W}\cdot\text{m}^{-2}\cdot^{\circ}\text{C}^{-1}$) | Resistance ($\text{m}^2\cdot^{\circ}\text{C}\cdot\text{W}^{-1}$) |
|--------------------------|--|--|
| 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 |

1.2.8. Selection of fluids for tube and the shell side

The routing of the shell side and tube side fluids has considerable effects on the heat exchanger design. Some **general guidelines** for positioning the fluids are given in **Table 1.4**. It should be understood that these guidelines are not ironclad rules and the optimal fluid placement depends on many factors that are service specific.

Table 1.4. Guidelines for placing the fluid in order of priority

| <i>Tube-side fluid</i> | <i>Shell-side fluid</i> |
|------------------------|---|
| Corrosive fluid | Condensing vapor (unless corrosive) |
| Cooling water | Fluid with large temperature difference ($>40^{\circ}\text{C}$) |
| Fouling fluid | |
| Less viscous fluid | |
| High-pressure steam | |
| Hotter fluid | |

Lecture 3: Process (Thermal) Design Procedure

1.3. Process (thermal) design procedure

Shell and tube heat exchanger is designed by trial and error calculations. The main steps of design following the **Kern method** are summarized as follows:

Step #1. Obtain the required thermophysical properties of hot and cold fluids at the **caloric temperature or arithmetic mean temperature**. Calculate these properties at the caloric temperature if the variation of viscosity with temperature is large. The detailed calculation procedure of caloric temperature available is in reference [3] (**page 93-99**).

Step #2. Perform energy balance and find out the heat duty (Q) of the exchanger.

Step #3. Assume a reasonable value of overall heat transfer coefficient ($U_{o,assm}$). The value of $U_{o,assm}$ with respect to the process hot and cold fluids can be taken from the books ([3] **page 840 Table 8**; [4] **page 297 Table 8.2**.)

Step #4. Decide tentative number of shell and tube passes (n_p). Determine the LMTD and the correction factor F_T ([3] **page 828-833 Figs. 18-23**; [4] **page 292 Figs. 8.10a & 8.10b**). F_T normally should be greater than 0.75 for the steady operation of the exchangers. Otherwise it is required to increase the number of passes to obtain higher F_T values.

Step #5. Calculate heat transfer area (A) required:
$$A = \frac{Q}{U_{o,assm} \cdot LMTD \cdot F_T} \quad (1.1)$$

Step #6. Select tube material, decide the tube diameter (ID= d_i , OD= d_o), its wall thickness (in terms of BWG or SWG) and tube length (L). Calculate the number of tubes

(n_t) required to provide the heat transfer area (A):
$$n_t = \frac{A}{\pi d_o L} \quad (1.2)$$

Calculate tube side fluid velocity,
$$u = \frac{4m(n_p/n_t)}{\pi \rho d_i^2} \quad (1.3)$$

If $u < 1$ m/s, fix n_p so that,
$$Re = \frac{4m(n_p/n_t)}{\pi d_i \mu} \geq 10^4 \quad (1.4)$$

Where, m , ρ and μ are mass flow rate, density and viscosity of tube side fluid. However, this is subject to allowable pressure drop in the tube side of the heat exchanger.

Step #7. Decide type of shell and tube exchanger (fixed tubesheet, U-tube etc.). Select the tube pitch (P_T), determine inside shell diameter (D_s) that can accommodate the calculated number of tubes (n_t). Use the standard tube counts table for this purpose. Tube counts are available in standard text books ([3] page 841-842 Table 9; [4] page 308 Table 8.3).

Step #9. Assign fluid to shell side or tube side (a general guideline for placing the fluids is summarized in Table 1.4). Select the type of baffle (segmental, doughnut etc.), its size (i.e. percentage cut, 25% baffles are widely used), spacing (B) and number. The baffle spacing is usually chosen to be within $0.2 D_s$ to D_s .

Step #10. Determine the tube side film heat transfer coefficient (h_i) using the suitable form of Sieder-Tate equation in laminar and turbulent flow regimes.

Estimate the shell-side film heat transfer coefficient (h_o) from:

$$j_H = \frac{h_o D_e}{k} \left(\frac{c\mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_w} \right)^{-0.14} \quad (1.5)$$

You may consider, $\frac{\mu}{\mu_w} = 1.0$

Select the outside tube (shell side) dirt factor (R_{do}) and inside tube (tube side) dirt factor (R_{di}) ([3] page 845 Table 12).

Calculate overall heat transfer coefficient ($U_{o,cal}$) based on the outside tube area (you may neglect the tube-wall resistance) including dirt factors:

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{do} + \frac{A_o}{A_i} \left(\frac{d_o - d_i}{2k_w} \right) + \frac{A_o}{A_i} \left(\frac{1}{h_i} \right) + \frac{A_o}{A_i} R_{di} \right]^{-1} \quad (1.6)$$

Step #11. If $0 < \frac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} < 30\%$, go the next **step # 12**. Otherwise go to **step #5**,

calculate heat transfer area (A) required using $U_{o,cal}$ and repeat the calculations starting from **step #5**.

If the calculated shell side heat transfer coefficient (h_o) is too low, assume closer baffle spacing (B) close to $0.2 D_s$ and recalculate shell side heat transfer coefficient. However, this is subject to allowable pressure drop across the heat exchanger.

Step #12. Calculate % overdesign. Overdesign represents extra surface area provided beyond that required to compensate for fouling. Typical value of 10% or less is acceptable.

$$\% \text{ Overdesign} = \frac{A - A_{reqd}}{A_{reqd}} \times 100 \quad (1.7)$$

A = design area of heat transfer in the exchanger; A_{reqd} = required heat transfer area.

Step #13. Calculate the tube-side pressure drop (ΔP_T): (i) pressure drop in the straight section of the tube (frictional loss) (ΔP_t) and (ii) return loss (ΔP_{rt}) due to change of direction of fluid in a ‘multi-pass exchanger’.

$$\text{Total tube side pressure drop: } \Delta P_T = \Delta P_t + \Delta P_{rt} \quad (1.8)$$

Step #14. Calculate shell side pressure drop (ΔP_S): (i) pressure drop for flow across the tube bundle (frictional loss) (ΔP_s) and (ii) return loss (ΔP_{rs}) due to change of direction of fluid.

$$\text{Total shell side pressure drop: } \Delta P_S = \Delta P_s + \Delta P_{rs} \quad (1.9)$$

If the tube-side pressure drop exceeds the allowable pressure drop for the process system, decrease the number of tube passes or increase number of tubes per pass. Go back to **step #6** and repeat the calculations steps.

If the shell-side pressure drop exceeds the allowable pressure drop, go back to **step #7** and repeat the calculations steps.

Step #15. Upon fulfillment of pressure drop criteria, go mechanical design. Refer **module # 2** for the details of mechanical design.

1.4. Design problem

The above design procedure is elaborated through the calculation of the following example

Lecture 4: Design Problem

Problem Statement:

150000 lb per hour of kerosene will be heated from 75 to 120°F by cooling a gasoline stream from 160 to 120°F. Inlet pressure will be 50 psia for each stream and the maximum pressure drop of 7 psi for gasoline and 10 psi for kerosene are permissible. Published fouling factors for oil refinery streams should be used for this application. Design a shell and tube heat exchanger for this service.

PART 1: THERMAL DESIGN:

(PART 2: Mechanical design provided in module #2)

Given data:

Hot fluid inlet temperature (T_1) = 160°F

Hot fluid outlet temperature (T_2) = 120°F

Cold fluid inlet temperature (t_1) = 75°F

Cold fluid outlet temperature (t_2) = 120°F

Fouling factor of hot fluid (R_{dg}) = 0.0005 (for gasoline)

Fouling factor of cold fluid (R_{dk}) = 0.001 (for kerosene)

P_{inlet} (for hot fluid) = 50 psia

P_{inlet} (for cold fluid) = 50 psia

Δp_{max} (for hot fluid) = 7 psi

Δp_{max} (for cold fluid) = 10 psia

Mass flow rate of cold fluid (\dot{m}_k) = 150000 lb.h⁻¹

(Subscripts 'k' for kerosene and 'g' for gasoline)

I. Calculation of caloric temperature

For the calculation of caloric temperature please refer [3] (page 827).

$$r = \frac{\Delta t_c}{\Delta t_h} = \frac{T_2 - t_1}{T_1 - t_2} = \frac{120 - 70}{160 - 120} = 1.25$$

°API of hot fluid = 76°; Therefore $K_c = 1$; $F_c = 0.455$

(The caloric temperature factor, F_c with °API as a function K_c is available in reference [3] (page 827).

Caloric temperature of the hot fluid, $T_{hc} = T_2 + F_C(T_1 - T_2)$

$$= 120 + 0.455 \times (160 - 120)$$

$$= 138.2^\circ\text{F}$$

Caloric temperature of the cold fluid, $T_{cc} = t_1 + F_C(t_2 - t_1)$

$$= 75 + 0.455 \times (120 - 75)$$

$$= 95.475^\circ\text{F}$$

II. Fluid properties at caloric temperature

Viscosity:

76° API gasoline, $\mu_g = 0.2 \text{ cp}$ ($0.484 \text{ lb.ft}^{-1}.\text{h}^{-1}$)

46° API kerosene, $\mu_k = 1.6 \text{ cp}$ ($3.872 \text{ lb.ft}^{-1}.\text{h}^{-1}$)

Density:

$\rho_g = 685 \text{ kg.m}^{-3}$ (42.7 lb.ft^{-3})

$\rho_k = 800 \text{ kg.m}^{-3}$ (49.8 lb.ft^{-3})

Thermal conductivity:

$k_g = 0.075 \text{ Btu h}^{-1}\text{ft}^{-1} \text{ }^\circ\text{F}^{-1}$

$k_k = 0.083 \text{ Btu h}^{-1}\text{ft}^{-1} \text{ }^\circ\text{F}^{-1}$

Specific heat capacity:

$C_g = 0.57 \text{ Btu lb}^{-1}\text{ft}^{-1}$

$C_k = 0.48 \text{ Btu lb}^{-1}\text{ft}^{-1}$

Specific gravity:

$S_g = 0.685$

$S_k = 0.80$

III. Energy balance

Assume no heat loss to the surrounding.

$$Q_g = Q_k = \dot{m}_k C_k (t_2 - t_1) = \dot{m}_g C_g (T_1 - T_2) = 3240000 \text{ Btu/h}$$

$$\Rightarrow 150000 \times 0.48 \times (120 - 75) = \dot{m}_g \times 0.57 \times (160 - 120)$$

$$\Rightarrow \dot{m}_g = 142105 \text{ lb h}^{-1}$$

IV. Calculation of heat transfer area and tube numbers

Iteration #1:

The first iteration is started assuming 1 shell pass and 2 tube pass shell and tube exchanger with following dimensions and considerations.

- Fixed tube plate
- 1'' OD tubes (d_o) (14 BWG) on 1¼'' square pitch (P_T)
- Outer diameter of tube= 1''
- Tube length (L_t) = 16'
- Tube ID (d_i) = 0.834''
- Fluid arrangement: Kerosene is placed in tube side because it has the higher fouling tendency

The log mean temperature correction factor (F_T) for 1-2 shell and tube exchanger:

$$\begin{aligned}
 F_T &= \frac{\frac{\sqrt{R^2 + 1} \ln(1 - S)}{1 - RS}}{(R - 1) \ln \left(\frac{2 - S \left(R + 1 - \sqrt{(R^2) + 1} \right)}{2 - S \left(R + 1 + \sqrt{(R^2) + 1} \right)} \right)} \\
 &= \frac{\sqrt{0.29^2 + 1} \ln \left[\frac{(1 - 0.529)}{(1 - 0.89 * 0.529)} \right]}{(0.89 - 1) \ln \left[\frac{2 - 0.529 \left(0.89 + 1 - \sqrt{(0.89)^2 + 1} \right)}{2 - 0.529 \left(0.89 + 1 + \sqrt{(0.89^2) + 1} \right)} \right]} \\
 &= 0.802
 \end{aligned}$$

$$\text{where, } R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{160 - 120}{120 - 75} = 0.889; \quad S = \frac{t_2 - t_1}{T_1 - t_1} = \frac{120 - 75}{160 - 75} = 0.529$$

$$\begin{aligned}
 LMTD &= \frac{(T_2 - T_1) - (t_2 - t_1)}{\ln \left(\frac{T_2 - T_1}{t_2 - t_1} \right)} \\
 &= \frac{(160 - 120) - (120 - 75)}{\ln \left(\frac{160 - 120}{120 - 75} \right)}
 \end{aligned}$$

$$= 42.75 \text{ }^{\circ}\text{F}$$

Determining the heat transfer area ('A'):

The value of overall heat transfer coefficient ($U_{o,assm}$) of $45 \text{ Btu h}^{-1}\text{ft}^{-2}\text{ }^{\circ}\text{F}^{-1}$ is assumed to initiate the design calculation for the kerosene and gasoline heat exchanger. The approximate range of overall heat transfer coefficient depending on the hot and cold fluid can be found out from text books ([3] page 845).

$$\begin{aligned} A &= \frac{Q}{U_{assm} LMTD \times F_T} \\ &= \frac{\dot{m}_g C_g (T_1 - T_2)}{U_{assm} \times LMTD \times F_T} \\ &= \frac{142105 \times 0.57 \times (160 - 120)}{45 \times 42.75 \times 0.802} \\ &= 2100 \text{ ft}^2 \end{aligned} \tag{1.1}$$

Calculating no. of tubes (n_t):

$$\begin{aligned} n_t &= \frac{A}{\pi d_o L_t} \\ n_t &= \frac{2100}{\pi \times \left(\frac{1}{12}\right) \times 16} = 502 \end{aligned} \tag{1.2}$$

$n_t = 518$ is taken corresponding to the closest standard shell ID of 35'' for fixed tube sheet, 1-shell and 2-tube pass exchanger with 1'' tube OD on 1¼'' square pitch. You may refer to standard heat transfer books ([3] page 841-842) for the selection of suitable shell ID.

Check for fluid velocity:

$$\begin{aligned} \text{Re} &= \frac{4 \dot{m}_k (n_p / n_t)}{\pi d_i \mu} \\ \text{Re} &= \frac{4 \times (150000) \times \frac{2}{518}}{\pi \times 0.834 / 12 \times 3.872} \\ &= 2740.2 < 10^4 \end{aligned} \tag{1.4}$$

As $Re \ll 10^4$, the design parameters and considerations needs to be revised to meet the Reynolds number criteria subject to allowable pressure drop in the tube side of the heat exchanger.

Iteration #2:

Assumptions:

- Fixed tube plate type
- 1'' OD tubes (14 BWG) on 1¼'' square pitch (P_T)
- Tube length (L_t) = 24' (the tube length is increased from 16')
- 1 shell pass-6 tube pass (tube passes is increased to 6 from 2)
- Tube ID=0.834''
- Flow area per tube=0.546 inch²

No. of tubes:

$$n_t = \frac{A}{\pi d_o L_t} \quad (1.2)$$

$$n_t = \frac{2100}{\pi \times \left(\frac{1}{12}\right) \times 24} = 335$$

$n_t = 368$ is taken corresponding to the closest standard shell ID of 31'' for fixed tube sheet, 1-shell and 6-tube pass exchanger with 1'' tube OD on 1¼'' square pitch. The tube-counts are available in heat transfer text book ([3] Table 9 & 10 page 841-843).

Fluid velocity:

$$Re = \frac{4 \dot{m}_k (n_p / n_t)}{\pi d_i \mu} \quad (1.4)$$

$$Re = \frac{4 \times (150000) \times \frac{6}{368}}{\pi \times \frac{0.834}{12} \times 3.872}$$

= 11571.4 > 10⁴ corresponding to $n_p=6$.

$$u = \frac{Re \mu_k}{d_i \rho_k} \quad (1.3)$$

$$= \frac{11571.4 \times 3.872}{\frac{0.834}{12} \times 49.8}$$

$$= 12945.15 \text{ ft/h (3.59 ft/s)}$$

= 1.04 m/s (so the design velocity is within the acceptable range).

V. Determination of heat transfer co-efficient

Tube side heat transfer co-efficient (h_i):

$$j_H = \frac{h_i d_i}{k} \left(\frac{\mu_k C_k}{k_k} \right)^{-1/3} \left(\frac{\mu}{\mu_w} \right)^{-0.14} \quad (1.5)$$

$j_H=42$ for the tube side fluid at $Re=11571.4$ ([3] page 834)

(Let's consider $\phi_t = \frac{\mu}{\mu_w} = 1$, μ = viscosity of the tube side fluid; μ_w = viscosity of tube

side fluid at wall temperature)

$$42 = \frac{h_i \left(\frac{0.834}{12} \right)}{0.083} \left(\frac{0.48 \times 3.872}{0.083} \right)^{-1/3}$$

$$h_i = 141.3 \text{ Btu h}^{-1} \text{ft}^{-1} \text{ } ^\circ\text{F}^{-1}$$

Shell side heat transfer co-efficient (h_o):

Assumptions:

- 25% cut segmental baffles
- Baffles spacing, $B=0.5D_s=15.5''$ (half of the shell ID is selected)

$$\text{Equivalent diameter for the shell side: } D_e = \frac{4 \left(P_T^2 - \frac{\pi}{4} d_o^2 \right)}{\pi d_o} \text{ for square pitch}$$

$$= 0.082 \text{ ft}$$

$$\left[\text{For triangular pitch, } D_e = \frac{4 \left(\frac{1}{2} P_T \times 0.86 P_T - \frac{1}{2} \frac{\pi}{4} d_o^2 \right)}{\frac{1}{2} \pi d_o} \right]$$

Shell side cross flow area, $a_s = \frac{C B D_s}{P_T}$ (please refer to **Figure 1.6**).

C= Tube clearance

$$= P_T - d_o$$

$$= 1\frac{1}{4} - 1 = 0.25''$$

$$a_s = \frac{\left(\frac{0.25}{12}\right)\left(\frac{15.5}{12}\right)\left(\frac{31}{12}\right)}{\left(\frac{1.25}{12}\right)} = 0.675 \text{ ft}^2$$

$$\text{Mass velocity, } G_s = \frac{\dot{m}_g}{a_s} = \frac{142105}{0.675}$$

$$= 210526 \text{ lb. h}^{-1} \cdot \text{ft}^{-2}$$

$$\text{Re} = \frac{D_e G_s}{\mu_g}$$

$$= \frac{0.082 \times (210526)}{0.484}$$

$$= 35668$$

$$\text{Now for the shell side, } j_H = \frac{h_o D_e}{k_g} \left(\frac{\mu_g C_g}{k_g} \right)^{-1/3} \left(\frac{\mu}{\mu_w} \right)^{-0.14} \quad (1.5)$$

$j_H=110$ for the shell side fluid at $\text{Re}=35668$ with 25% cut segmental baffles ([3] page 838)

$$110 = \frac{h_o(0.082)}{(0.075)} \left(\frac{0.57 \times 0.484}{0.075} \right)^{-1/3}$$

($\phi_s = \frac{\mu}{\mu_w} = 1$ is considered for the shell side fluid)

$$h_o = 155.3 \text{ Btu h}^{-1} \text{ ft}^{-2} \text{ }^\circ\text{F}^{-1}$$

Overall heat transfer co-efficient ($U_{o,cal}$):

Fouling factor, $R_{dk}=0.001 \text{ h ft}^2 \text{ }^\circ\text{F Btu}^{-1}$ for kerosene and $R_{dg}= 0.0005 \text{ h ft}^2 \text{ }^\circ\text{F Btu}^{-1}$ for gasoline is taken for this service.

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{dg} + \frac{A_o}{A_i} \left(\frac{d_o - d_i}{2k_w} \right) + \frac{A_o}{A_i} \left(\frac{1}{h_i} \right) + \frac{A_o}{A_i} R_{dk} \right]^{-1} \quad (1.6)$$

Let select, Admiralty brass as tube material with thermal conductivity, $k_w=70 \text{ Btu h}^{-1} \text{ ft}^{-1} \text{ }^\circ\text{F}^{-1}$.

$$U_{o,cal} = \left[\frac{1}{155.3} + 0.0005 + \frac{\pi(1)^2}{\pi(0.834)^2} \left(\frac{\frac{1}{12} - \frac{0.834}{12}}{2 \times 70} \right) + \frac{\pi(1)^2}{\pi(0.834)^2} \left(\frac{1}{141.3} \right) + \frac{\pi(1)^2}{\pi(0.834)^2} \times 0.001 \right]^{-1}$$

$$U_{o,cal} = 53.5 \text{ Btu h}^{-1} \text{ ft}^{-2} \text{ }^\circ\text{F}^{-1}$$

$$\begin{aligned} \text{Now, } \frac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} &= \frac{53.5 - 45}{45} \times 100 \\ &= 18.9\% < 30\% \end{aligned}$$

Therefore, the calculated overall heat transfer co-efficient is well within the design criteria.

VI. Pressure drop calculation

VI.1. Tube side pressure drop:

Friction factor $f = 0.00028 \times 144 = 0.04032 \text{ ft}^2/\text{ft}^2$ for $\text{Re}=11571.4$ ([3] page 836]

$a_t = (\text{no. of tubes}) \times (\text{flow area per tube}) / (\text{no. of passes})$

$$= \frac{368 \times 0.546}{6 \times 144} \text{ ft}^2$$

$$= 0.232 \text{ ft}^2$$

$$\text{Tube side mass velocity: } G_t = \frac{\dot{m}_k}{a_t} = \frac{150000}{0.232}$$

$$= 646552 \text{ lb. h}^{-1} \cdot \text{ft}^{-2}$$

Frictional pressure drop: $\Delta P_t = \frac{fG_t^2 L_t n_p}{7.5 \times 10^{12} \times d_i S_k \phi_t}$

$$= \frac{0.04032 \times 646552^2 \times 24 \times 6}{7.5 \times 10^{12} \times \frac{0.834}{12} \times 0.8 \times 1}$$

=5.81 psi

Return loss ΔP_r : (due to change in flow direction of the tube side fluid)

$$\Delta P_{tr} = 1.334 \times 10^{-13} (2n_p - 1.5) \frac{G_t^2}{S_k}$$

$$= 1.334 \times 10^{-13} (2 \times 6 - 1.5) \frac{(646552)^2}{(0.8)}$$

=0.73 psi

Total tube side drop neglecting nozzle loss:

$$\Delta P_T = \Delta P_t + \Delta P_r \tag{1.8}$$

=5.81+0.73

=6.54 psi < 10 psi

Therefore the tube side pressure drop is within the maximum allowable pressure drop of 10 psi.

VI.2. Shell side pressure drop calculation

Tube clearance, $C=0.25''$

Spacing, $B=15.5''$

$$a_s = 0.675 \text{ ft}^2$$

Mass velocity, $G_s = 210526 \text{ lb. h}^{-1} \cdot \text{ft}^{-2}$

Re=35668

$$\text{No of baffles, } n_b = \frac{\text{tube length}}{\text{baffle spacing}} = \frac{24}{15.5/12} = 18.6 \approx 19$$

Friction factor, $f = 0.0017 \times 144 = 0.2448 \text{ ft}^2/\text{ft}^2$ with 25% cut segmental baffles ([3]

page 839)

Shell side frictional pressure drop ΔP_s :

$$\Delta P_s = \frac{f G_s^2 D_s (n_b + 1)}{7.5 \times 10^{12} \times D_e S_k \phi_k}$$

$$= \frac{0.2376 \times 210526^2 \times (19 + 1) \times \frac{31}{12}}{7.5 \times 10^{12} \times 0.082 \times 0.685 \times 1}$$

=1.4 psi < 7 psi

$\Delta P_{rs} = 0$ (in case of single shell pass flow)

Total shell side drop neglecting nozzle loss:

$$\Delta P_S = \Delta P_s + \Delta P_{sr} = 1.4 \text{ psi} \quad (1.9)$$

Therefore the shell side pressure drop is within the maximum allowable pressure drop of 7 psi.

VII. Over surface and over design

$$\text{Over surface} = \frac{U_C - U_{o,cal}}{U_C}$$

The clean overall heat transfer co-efficient: $U_C = \frac{h_o \times h_{i_o}}{h_o + h_{i_o}}$

$$h_{i_o} = h_i \times \frac{d_i}{d_o} = 141.3 \times 0.834 = 117.8 \text{ Btu h}^{-1} \text{ ft}^{-2} \text{ } ^\circ\text{F}^{-1}$$

$$U_C = 66.98 \text{ Btu h}^{-1} \text{ ft}^{-2} \text{ } ^\circ\text{F}^{-1}$$

$$\% \text{ Over surface} = \frac{66.98 - 53.5}{66.98} \times 100$$

=20% (acceptable)

Over design:

$$\% \text{ Overdesign} = \frac{A - A_{reqd}}{A_{reqd}} \times 100 \quad (1.7)$$

The design area of heat transfer in the exchanger ($n_t = 318$):

$$A = \pi d_o L_t n_t = \pi \times \frac{1}{12} \times 24 \times 368 = 2312 \text{ ft}^2$$

The required heat transfer area (where, $n_t=335$):

$$A_{reqd} = \pi d_o L_t n_t = \pi \times \frac{1}{12} \times 24 \times 335 = 2105 \text{ ft}^2$$

% Overdesign = 9.8% which is within the acceptable limit.

Refer module # 2 for the mechanical design of shell and tube heat exchanger.

Lecture 5: Shell and Tube Exchanger for Two Phase Heat Transfer

2. PROCESS DESIGN OF SHELL AND TUBE EXCHANGER FOR TWO PHASE HEAT TRANSFER

2.1. Condenser

The change from liquid phase to vapor phase is called vaporization and the reverse phase transfer is condensation. The change from liquid to vapor or vapor to liquid occurs at one temperature (called saturation or equilibrium temperature) for a pure fluid compound at a given pressure. The industrial practice of vaporization and condensation occurs at almost constant pressure; therefore the phase change occurs isothermally.

Condensation occurs by two different physical mechanisms i.e. **drop-wise condensation** and **film condensation**.

The nature of the condensation depends upon whether the condensate (liquid formed from vapor) wets or does not wet the solid surface. If the condensate wets the surface and flows on the surface in the form of a film, it is called film condensation. When the condensate does not wet the solid surface and the condensate is accumulated in the form of droplets, is drop-wise condensation. **Heat transfer coefficient is about 4 to 8 times higher for drop wise condensation.** The condensate forms a liquid film on the bare-surface in case of film condensation. The heat transfer coefficient is lower for film condensation due to the resistance of this liquid film.

Dropwise condensation occurs usually on new, clean and polished surfaces. The heat exchanger used for condensation is called condenser. In industrial condensers, film condensation normally occurs.

2.1.1. Types of condensers

There two general types of condensers:

i. Vertical condenser

Downflow vertical condenser: The vapor enters at the top of condenser and flows down inside tubes. The condensate drains from the tubes by gravity and vapor induced shear (**Figure 1.7**).

Upflow vertical condenser: In case of upflow condenser, the vapor enters at the bottom and flows upwards inside the tubes. The condensate drains down the tubes by gravity only.

ii. Horizontal condenser: The condensation may occur inside or outside the horizontal tubes (**Figure 1.8**). Condensation in the tube-side is common in air-cooled condensers. The main disadvantage of this type of condenser is that the liquid tends to build up in the tubes. Therefore the effective heat transfer coefficient is reduced significantly.

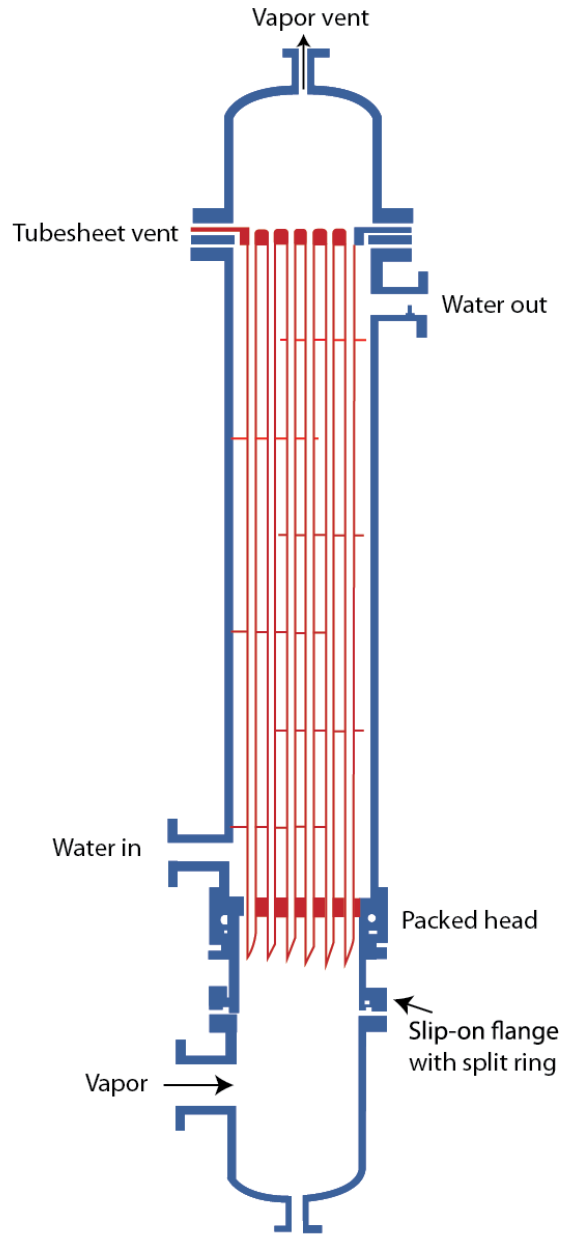


Figure 1.7. Downflow vertical condenser with condensation inside tube [5].

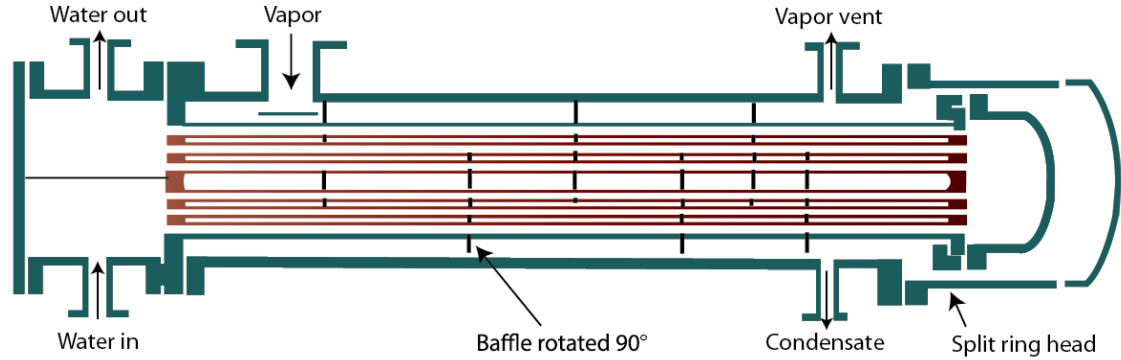


Figure 1.8. Horizontal condenser with condensation outside horizontal tubes [5].

2.1.2. Condenser design

The design of condenser is similar to a typical shell and tube exchangers. A condenser must have a vent for removal of non-condensable gas. The non-condensable gas decreases the heat transfer rate. Condenser usually use a wider baffle spacing of $B = D_s$ (ID of shell) as the allowable pressure drop in shell side vapor is usually less. Vertical cut-segmental baffles are generally used in condensers for side-to-side vapor flow and not for top to bottom. An opening at the bottom of the baffles is provided to allow draining of condensate.

2.1.2.1. Mean temperature difference

The condensation occurs almost at a fixed temperature (isothermally) at constant pressure for a pure saturated vapor compound. The logarithmic mean temperature difference can be used for condenser design. **No correction factor for multiple pass condensers is required.** The logarithmic mean temperature difference:

$$LMTD = \frac{(T_{sat} - t_1) - (T_{sat} - t_2)}{\ln \frac{(T_{sat} - t_1)}{(T_{sat} - t_2)}} = \frac{(t_2 - t_1)}{\ln \frac{(T_{sat} - t_1)}{(T_{sat} - t_2)}} \quad (1.10)$$

Where, T_{sat} = Saturation vapor temperature

t_1 = Coolant inlet temperature

t_2 = Coolant outlet temperature

2.1.2.2. Calculation of heat transfer co-efficient during condensation

Calculation of tube side heat transfer co-efficient (h_i): The calculation of heat transfer co-efficient for the cold fluid (coolant) can be performed similarly as discussed in design of shell and tube heat exchanger (heat transfer without phase change). Here it is assumed that the coolant flows the in tube side and the condensing saturated vapor flows in the shell side. If the condensation occurs in the tube side, follow the procedure discussed in next section for shell side calculation.

Calculation of shell-side heat transfer coefficient (condensing film heat transfer coefficient) (h_o): The Kern method is discussed here to calculate the individual heat transfer co-efficient of the condensing fluid by trial and error calculation.

i. Assume, $h_{o(assm)}$ in the range from 100 to 300 $\text{BTU.h}^{-1}.\text{ft}^{-2}.\text{°F}^{-1}$. The film coefficient of condensing hydrocarbons generally varies in this range. Air-free condensing steam has a coefficient of 1500 $\text{BTU.h}^{-1}.\text{ft}^{-2}.\text{°F}^{-1}$.

ii. Calculate the tube wall temperature (T_w):

$$T_w = T_{C(avg)} + \frac{h_o(T_v - T_{C(avg)})}{(h_{io} + h_o)} \quad (1.11)$$

or

$$T_w = T_{cc} + \frac{h_o(T_v - T_{cc})}{(h_{io} + h_o)} \quad (1.12)$$

Where, $h_{io} = h_i \times \frac{d_i}{d_{io}}$ (d_i tube ID and d_o tube OD)

$T_{C(avg)}$ = Average temperature of the cold fluid

T_{cc} = Caloric temperature of the cold fluid

iii. Calculate condensate film temperature, $T_f = \frac{(T_w + T_v)}{2}$ (1.13)

T_v = Condensation temperature (For pure fluid compound T_v is the saturation temperature. Average of condensation over a temperature range also can be used for non-isothermal condensation).

iv. Calculate all thermophysical property of the condensing fluid at film temperature (T_f).

v. Recalculate, $h_{o(cal)}$ from j_H factor.

Now again set, $h_{o(assm)} = h_{o(cal)}$ and continue the calculation till $h_{o(assm)} \approx h_{o(cal)}$.

vi. Calculate the overall heat transfer-coefficient (U_d) including the dirt factors.

Lecture 6: Condenser and Reboiler Design

2.1.2.3. Pressure drop calculation

i. Tube side pressure drop

In case of tube side condensation:

For condensation in the tube side by taking one-half of the conventional pressure drop relation can be used.

$$\Delta P_t = \frac{1}{2} \left(\frac{f G_t^2 L_t n_p}{7.5 \times 10^{12} \times d_i S_t \phi_t} \right), \text{ psi} \quad (1.14)$$

Where,

f = friction factor

G_t = mass velocity [lb. h⁻¹.ft⁻²]

L_t = Tube length [ft]

n_p = Number of tube passes

d_i = Tube ID [ft]

S_t = Specific gravity of the tube side fluid

ϕ_t = Viscosity correction factor

($\phi_t = \frac{\mu}{\mu_w} = 1$, μ = viscosity of the tube side fluid; μ_w = viscosity of water)

ii. Shell side pressure drop

In case of shell side condensation: Similarly for condensation in the shell side:

$$\Delta P_s = \frac{1}{2} \left(\frac{f G_s^2 D_s (n_b + 1)}{7.5 \times 10^{12} \times D_e S_s \phi_s} \right), \text{ psi} \quad (1.15)$$

Subscript 's' indicates shell side fluid.

n_b = number of baffles

D_e = Equivalent diameter for the shell [ft]

Calculate all fluid property at film temperature T_f . No return loss calculation is required for the condensing fluid.

In case of non-condensing fluid (single phase flow), use the conventional pressure drop relation.

2.1.3. De-superheating and sub-cooling

De-superheating is different from condensation of a saturated vapor. The sensible heat should be removed first to de-superheat the vapor to obtain the saturated vapor. Similarly, the saturated liquid is to be further cooled down (sub-cooled) by extracting sensible heat below the boiling point. The temperature profile is shown in **Figure 1.9** for the condensation of superheated vapor to obtain the sub-cooled liquid from the same exchanger. The mean temperature difference and heat transfer coefficient should be calculated individually for each section if the degree of superheat/ sub-cool is large. The weighted mean temperature difference and overall transfer co-efficient can be used to design the condensers if heat load due to sensible heat transfer in each unit about 25% of latent heat transfer. Otherwise, it is convenient to design separate de-superheater and sub-cooling exchangers. The calculations for detail study can be found out in reference [3] (page 283-285).

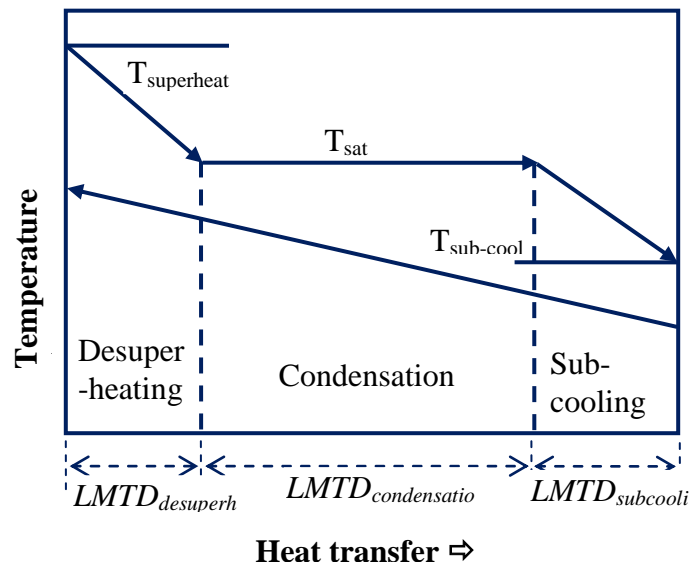


Figure 1.9. Condensation with de-superheating and sub-cooling [2].

Practice problem:

Design a horizontal condenser for the condensation of 45,000 lb/h of almost pure normal propyl alcohol available at 15 psig. At this pressure, the boiling point of n-propyl alcohol is 244°F. Water available in the temperature range of 95 to 120°F can be as the coolant. The maximum pressure drop of 2 psi and 10 psi is permissible for the vapor phase and water respectively.

2.2. Reboilers**2.2.1. Classification of reboilers**

There are three major types of reboilers:

- i. **Thermosyphon natural circulation reboiler:** The boiling occurs inside the tubes in vertical thermosyphon reboiler and inside shell in horizontal thermosyphon reboiler (**Figure 1.10**). In vertical thermosyphon reboiler, the liquid circulation occurs due to density difference between vapor-liquid mixture (two phase) in the exchanger from the reboiler and the liquid through the downcomer to the reboiler. Advantages: most economical because no pump is required. Limitations: not suitable for heavily viscous fluid; high construction cost for the installation of the column base at suitable elevation to get thermosyphon effect; not suitable for low temperature difference processes due to boiling point elevation imposed by static head.

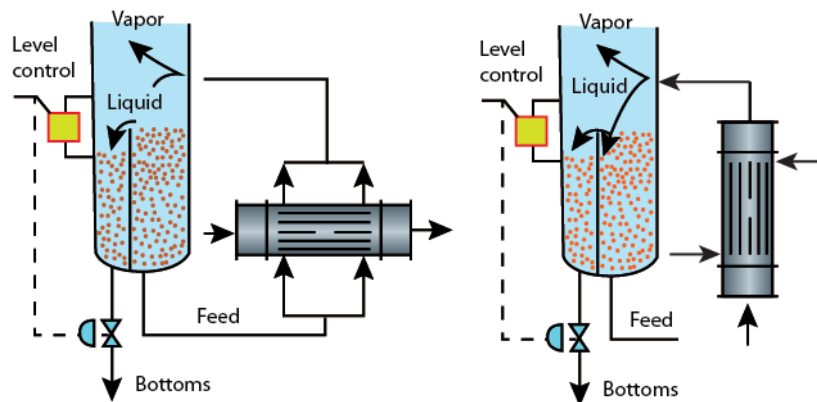


Figure 1.10. Thermosyphon reboiler [5]. (a) Horizontal thermosyphon reboiler. (b) Vertical thermosyphon reboiler

- ii. Forced circulation reboiler:** The liquid is fed by means of a pump. Forced circulation reboilers with vertical or horizontal tubes boiling may be designed. Forced circulation reboilers are similar to vertical thermosiphon reboilers, except the pump is used for the circulation of the liquid and the hot liquid flows inside column. To calculate the heat transfer coefficient it is generally assumed that, heat is transferred only by forced convection. The usual method of shell and tube exchanger design can be used.

Advantage: suitable for viscous and highly fouling fluids.

Disadvantage: high pumping and maintenance cost; pump is required to circulate the boiling liquid through the tubes and back into the column.

- iii. Kettle reboiler:** The tube bundle is immersed in a pool of liquid at the base of the column in an oversize shell (**Figure 1.11**). Kettle reboiler is also called a “submerged bundle reboiler”. The height of the tube bundle is usually 40-60% of the shell ID. The submergence of the tube bundle is assured by an overflow weir at height of typically 5-15 cm from the upper surface of topmost tubes.

Advantage: suitable for vacuum operation and high vaporization rate up to about 80% of the feed.

Limitations: low heat transfer rate than other types as there is no liquid circulation (low velocity); not appropriate for fouling fluids; kettle reboiler is not suitable for heat sensitive materials as it has higher residence time.

The bundle diameter D_b , can be obtained from the empirical equation ([2] page 647-649):

$$D_b = d_o \left(\frac{n_t}{K_1} \right)^{1/n_1} \quad (1.16)$$

where, D_b = bundle diameter [mm], n_t = number of tubes, d_o = tube outside diameter [mm]. The values of the constants K_1 and n_1 are in **Table 1.5**.

Table 1.5. Constants used to calculate the tube bundle diameter.

| Pitch type | Constants | Number of tube passes (n_t) | | | | |
|--|-----------|---------------------------------|-------|-------|--------|--------|
| | | 1 | 2 | 4 | 6 | 8 |
| Triangular ($P_T = 1.25d_o$) | 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 ($P_T = 1.25d_o$) | K_1 | 0.215 | 0.156 | 0.158 | 0.0402 | 0.0331 |
| | n_1 | 2.207 | 2.291 | 2.263 | 2.617 | 2.643 |

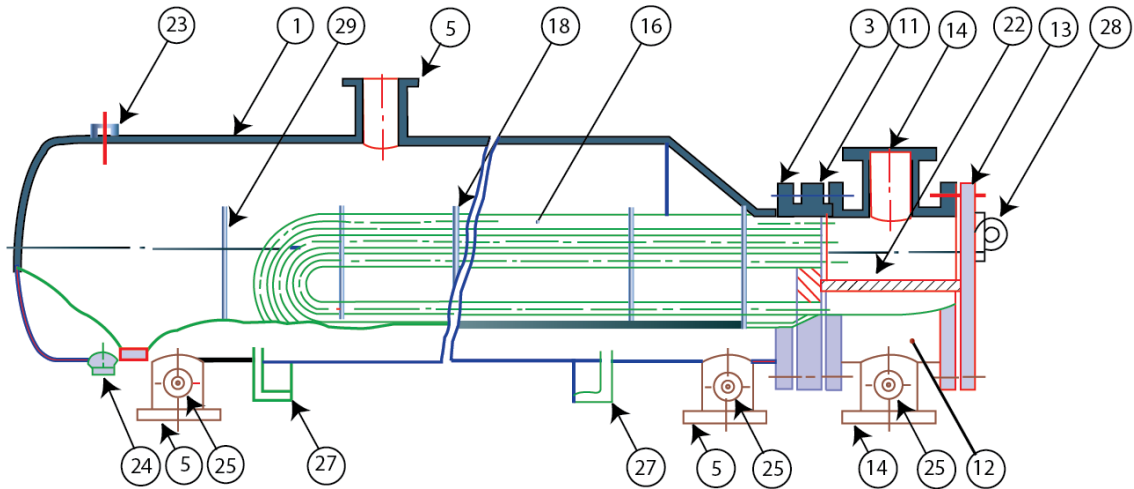


Figure 1.11. Kettle type reboiler [1].

2.2.2. Design of kettle reboiler

The Kern method for designing of Kettle reboiler for isothermal boiling is summarized below. It is assumed that the degree of sub-cooling and super-heating of the cold fluid is negligible i.e. vaporization of close boiling compounds with negligible super-heating of vapors formed.

- i. Make energy balance and determine the heat duty.
- ii. Calculate of fluid property at the caloric temperature (or at arithmetic mean temperature) as already shown.
- iii. Follow the same guideline and design requirements for shell containing the vaporizing liquid.
- iv. Calculation of heat transfer co-efficient

Calculation of individual heat transfer co-efficient hot fluid: The calculation of heat transfer co-efficient of the hot fluid can be performed similarly as in case of design of shell and tube heat exchanger for single phase.

Calculation of individual heat transfer coefficient of the boiling liquid: The Kern method is discussed here to calculate the individual heat transfer co-efficient of the boiling liquid by trial and error procedure.

Kern [2] recommends that the maximum allowable vaporizing film coefficients:

- 300 Btu/h.ft² °F for natural or forced circulation vaporizing organics.
- 1000 Btu/h.ft² °F for natural or forced circulation vaporizing aqueous solution of low concentration.

The maximum allowable heat flux:

- 20000 Btu/(h.ft²) for forced circulation reboilers and 12000 Btu/(h.ft²) for natural circulation reboilers vaporizing organics.
- 30000 Btu/(h.ft²) for both forced or natural circulation reboilers vaporizing aqueous solution.

Assume that $h_{(assm)} = 300$ Btu/h.ft² °F for organics or 1000 Btu/h.ft² °F for water.

With this assumed value, calculate the tube wall temperature (T_w):

$$T_w = T_{h(avg)} + \frac{h_{io}(T_{hc} - T_{h(avg)})}{(h_{io} + h_o)} \quad (1.17)$$

Where, $h_{io} = h_i \times \frac{d_i}{d_{io}}$ (d_i tube ID and d_o tube OD)

$T_{h(avg)}$ = Average temperature of the hot fluid

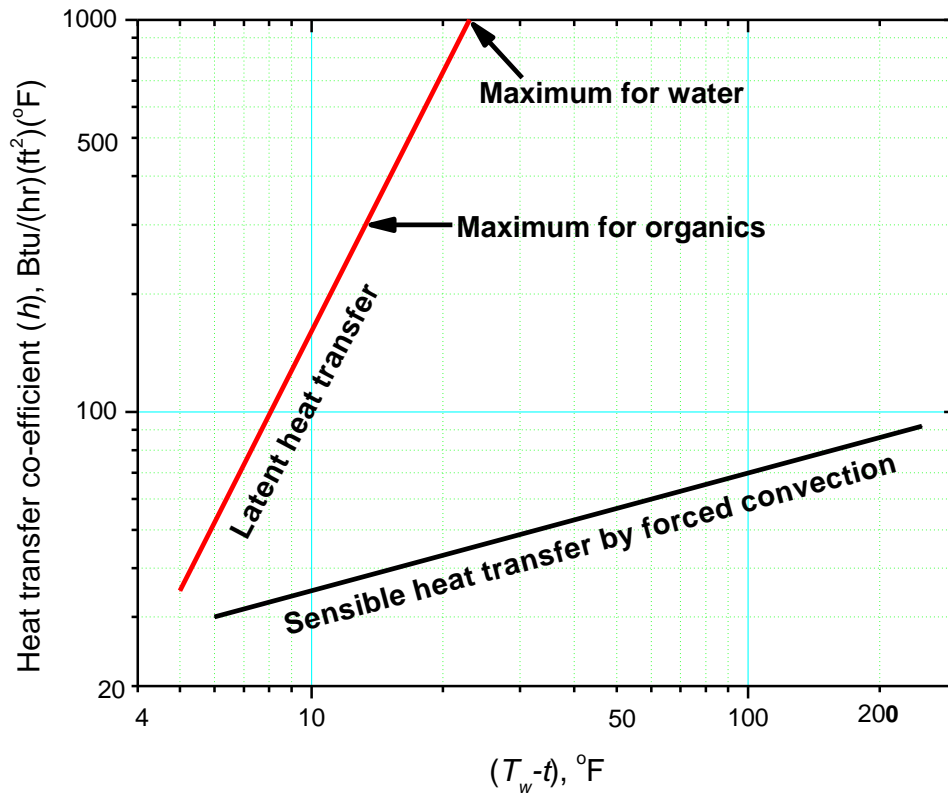
T_{hc} = Caloric temperature of the hot fluid

Now, re-determine h_{cal} (latent heat transfer) from the **Figure 1.12** corresponding to $(T_w - t)$. (t is the cold fluid boiling temperature).

Continue the calculation till, $h_{cal} \approx h_{(assm)}$.

If the calculated h_{cal} is greater than the maximum heat transfer co-efficient of 300 Btu/h.ft² °F for organics and 1000 Btu/h.ft² °F for water, take $h_{cal} = 300$ Btu/h.ft² °F for organics and $h_{cal} = 1000$ Btu/h.ft² °F for water.

Calculate the overall heat transfer-coefficient (U_d) including the dirt factors.



Temperature difference between tube wall and boiling liquid

Figure 1.12. Natural circulation boiling and sensible heat transfer [3].

- v. Decide type of exchanger i.e. fixed tube sheet or U- shell (use U-tube reboiler for large temperature difference), tube size (diameter, length, tube pitch), layout, effective tube length. A tube pitch of between 1.5 to 2 times the tubes OD should be used to avoid vapor blanketing.

- vi. Calculate exchanger area ($A = \frac{Q}{U_d(LMTD)}$) and number of tubes ($n_t = \frac{A}{\pi d_o L_t}$).

The number of tubes should be calculated based on the effective tube length for U-tube reboilers. The effective tube length is less than physical tube length due to U-bend.

- vii. Calculate the heat flux= $\frac{Q}{A}$ [Btu/(h.ft²)]. This value should be less than the maximum heat flux of 20000 Btu/(h.ft²) for forced circulation reboilers vaporizing organics and 30000 Btu/(h.ft²) for both forced or natural circulation

reboilers vaporizing aqueous solution. Otherwise, go to **step # v**, repeat the calculation until within the allowable limits.

viii. Check for allowable vapor velocity (u_v) ([3] page 749):

The maximum vapor velocity u_v (m/s) at the liquid surface should be less than that given by the expression below to avoid too much entrainment.

$$u_v < 0.2 \left(\frac{\rho_l - \rho_v}{\rho_v} \right)^{1/2} \quad (1.18)$$

where, ρ_l = liquid density and, ρ_v = vapor density

If this criterion is not satisfied, go to **step # v** and revise the calculation.

ix. Pressure drop calculation

Tube side pressure drop (hot fluid): The pressure drop calculation of the hot fluid can be carried out as already presented.

Shell side pressure drop (vaporizing liquid): There will be negligible hydrostatic head for the flow of liquid from the column to reboilers (low circulation velocity) if the liquid level above the tube bundle is not too high. Therefore, shell side pressure drop may be considered negligible.

x. Calculate over surface and over design

xi. Go for mechanical design

Design problem:

Gasoline (65°API gravity) flow rate of 60,000 lb/h with a small boiling range at 400°F is to be vaporized to form 37,050 lb/h vapor at an operating pressure of 200 psig. Use gas oil (30°API gravity) in the temperature range from 600 to 500°F at 120 psig operating pressure as the heating medium. A tube side pressure drop of 10 psi is allowable. Design a suitable Kettle reboiler to serve the purpose.

References

- [1]. Indian Standard (IS: 4503-1967): Specification for Shell and Tube Type Heat Exchangers, BIS 2007, New Delhi.
- [2]. R. K. Sinnott, Coulson & Richardson's Chemical Engineering: Chemical Engineering Design (volume 6), Butterworth-Heinemann, 3rd ed. 1999.
- [3]. D. Q. Kern, Process Heat Transfer, McGraw-Hill Book Company, Int. ed. 1965.
- [4]. Dutta B.K. 'Heat Transfer-Principles and Applications', PHI Pvt. Ltd., New Delhi, 1st ed. 2006.
- [5]. James R. Couper; W. Roy Penney, James R. Fair, Stanley M. Walas, Chemical Process Equipment: selection and design, Elsevier Inc., 2nd ed. 2005.