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

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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.
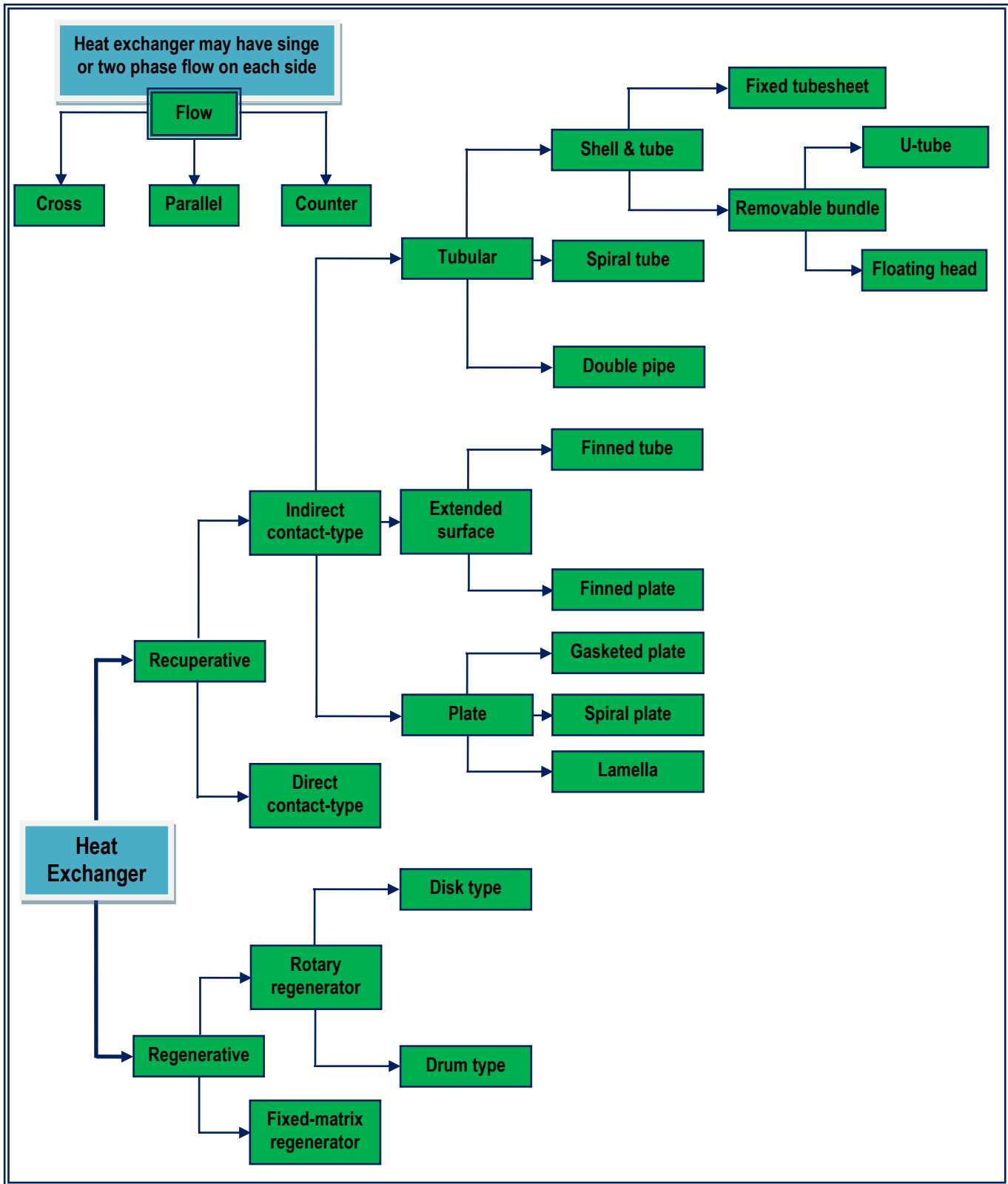
Heat exchanger may have singe or two phase flow on each side

Flow

- Cross
- Parallel
- Counter

Heat Exchanger

Recuperative
- Indirect contact-type
  - Tubular
    - Shell & tube
      - Fixed tubesheet
      - Removable bundle
        - U-tube
        - Floating head
    - Spiral tube
    - Double pipe
  - Extended surface
    - Finned tube
    - Finned plate
  - Plate
    - Gasketed plate
    - Spiral plate
    - Lamella
- Direct contact-type

Regenerative
- Rotary regenerator
  - Disk type
  - Drum type
- Fixed-matrix regenerator

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

Table 1.1. Features of shell and tube type exchangers.

| Shell and Tube Exchangers	Typical TEMA code	Advantages	Limitations
Fixed tube sheet	BEM, AEM, NEN	Provides maximum heat transfer area for a given shell and tube diameter. Provides for single and multiple tube passes to assure proper velocity. Less costly than removable bundle designs.	**Shell side / out side of the tubes are inaccessible for mechanical cleaning.** **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.**
Floating-head	AEW, BEW, BEP, AEP, AES, BES	Floating tube sheet allows for differential thermal expansion between the shell and the tube bundle. Both the tube bundle and the shell side can be inspected and cleaned mechanically.	**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.** **Tubes cannot expand independently so that huge thermal shock applications should be avoided.** **Packing materials produce limits on design pressure and temperature.**
U-tube	**BEU, AEU**	**U-tube design allows for differential thermal expansion between the shell and the tube bundle as well as for individual tubes.** **Both the tube bundle and the shell side can be inspected and cleaned mechanically.** **Less costly than floating head or packed floating head designs.**	**Because of U-bend some tubes are omitted at the centre of the tube bundle.** **Because of U-bend, tubes can be cleaned only by chemical methods.** **Due to U-tube nesting, individual tube is difficult to replace.** **No single tube pass or true countercurrent flow is possible.** **Tube wall thickness at the U-bend is thinner than at straight portion of the tubes.** **Draining of tube circuit is difficult when positioned with the vertical position with the head side upward.**
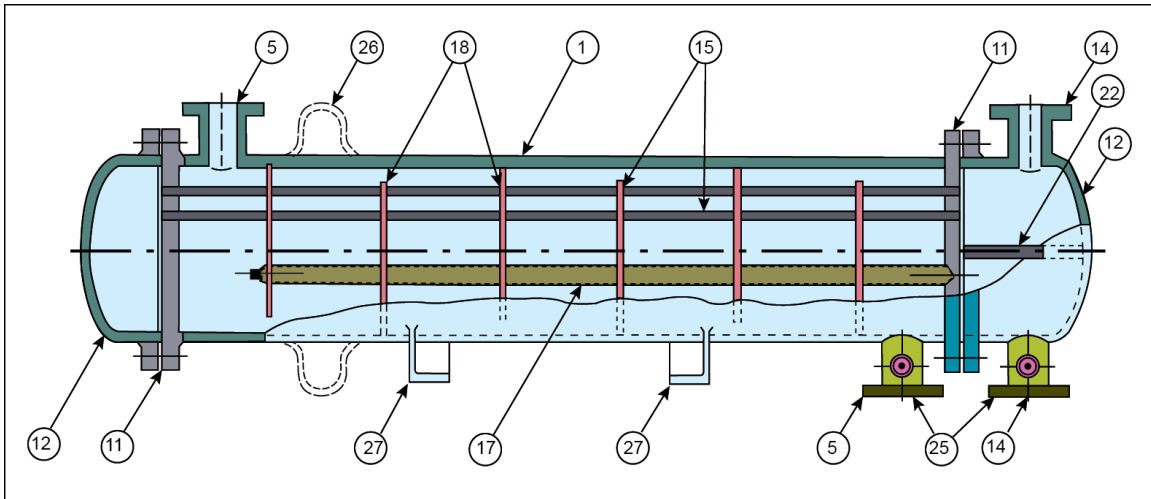
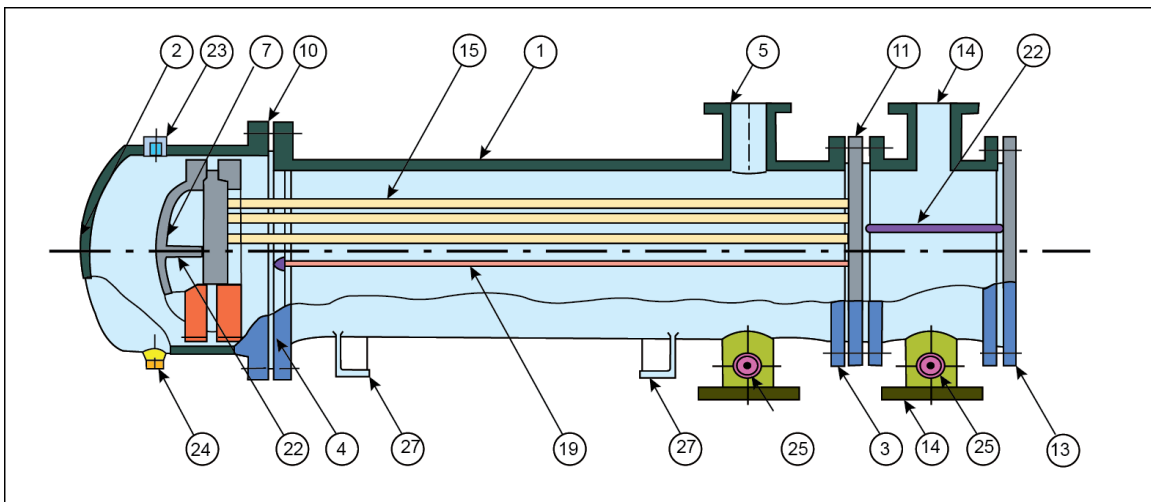**Figure 1.2. Fixed-tube heat exchanger ([1]).**

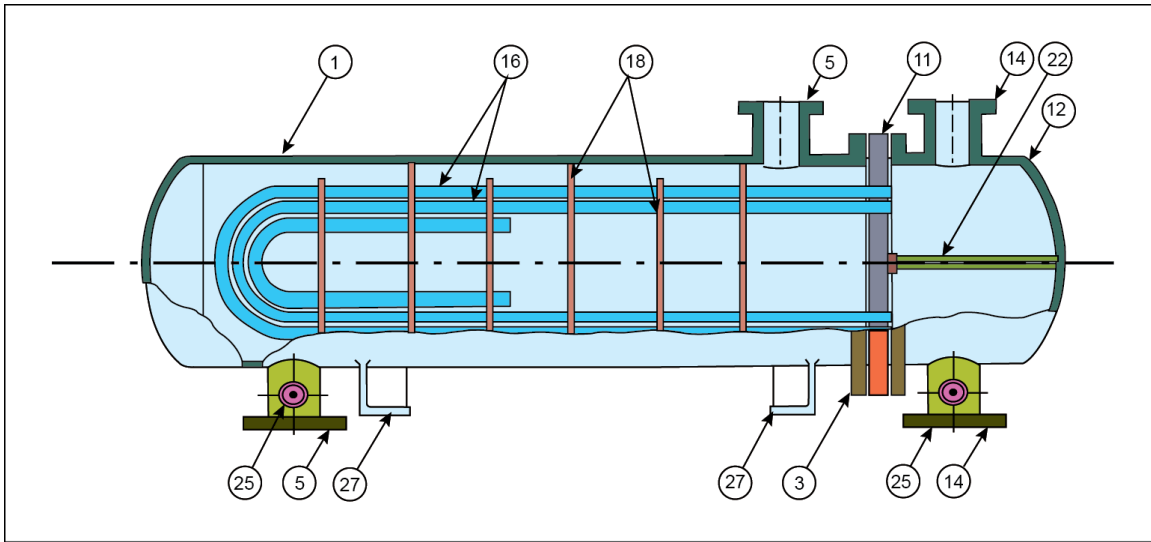**Figure 1.3. Floating-head heat exchanger (non-pull through type) [1].**

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 ¾ 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.

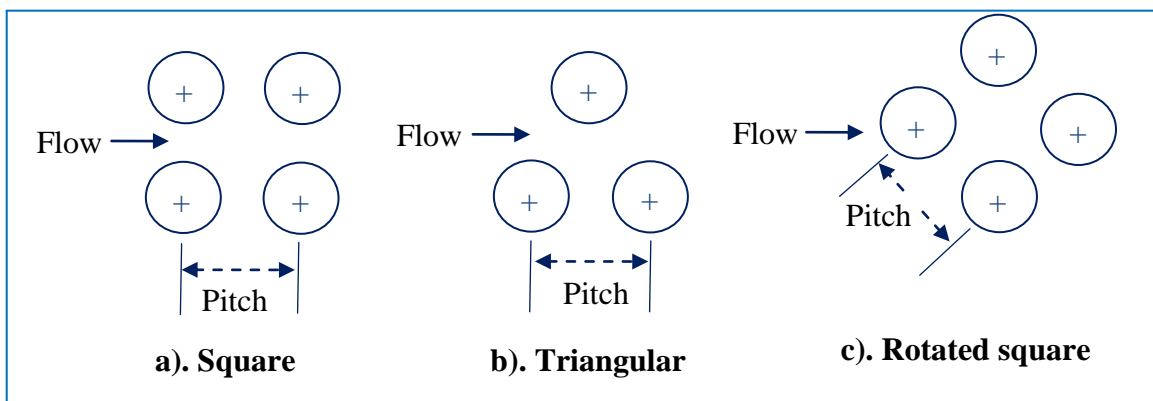| Tube OD, in	Pitch type	Tube pitch, in
3/4	Square	1
1		$1\frac{1}{4}$
3/4	Triangular	$\frac{15}{16}$
3/4		1

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 positioned 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.
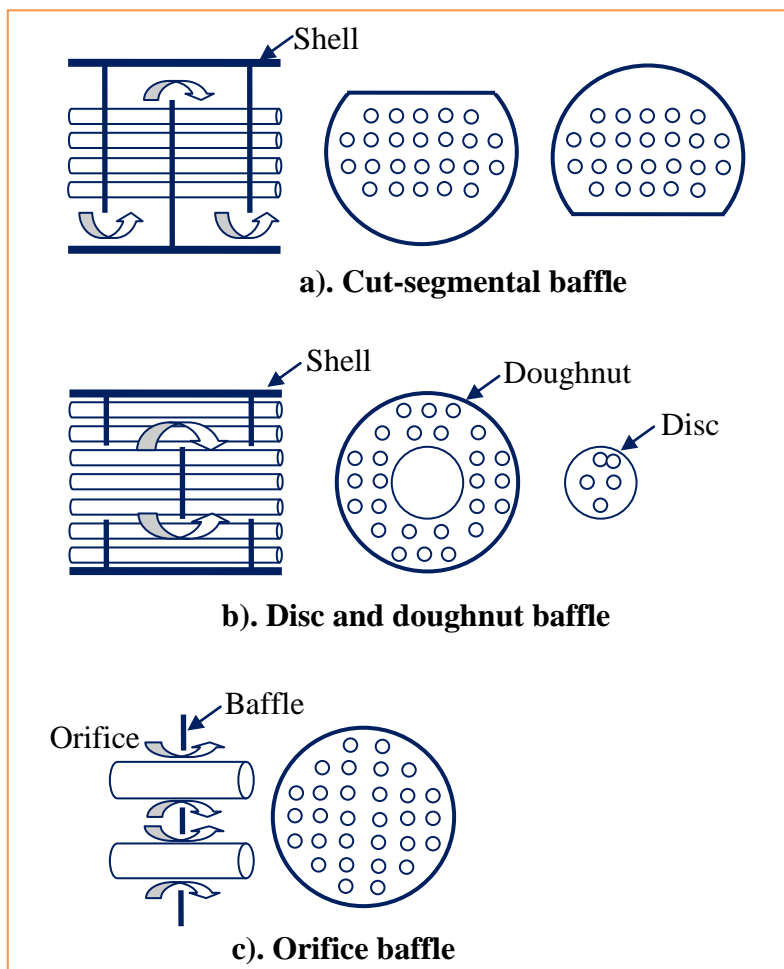

Figure 1.6. Different type of heat exchanger baffles: a). Cut-segmental baffle, b). Disc and doughnut baffle, c). Orifice baffle

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 $(W.m^{-2}.°C^{-1})$	Resistance $(m^2.°C.W^{-1})$
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	0.0002-0.000-1
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

Tube-side fluid	Shell-side fluid
Corrosive fluid	Condensing vapor (unless corrosive)
Cooling water	Fluid with large temperature difference (>40°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: $A = \dfrac{Q}{U_{o,assm}.LMTD.F_T}$ (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): $n_t = \dfrac{A}{\pi d_o L}$ (1.2)

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

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

Where, $\dot{m}$, 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:

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

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

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:

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{do} + \frac{A_0}{A_i}\left(\frac{d_0 - d_i}{2k_w}\right) + \frac{A_0}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_0}{A_i}R_{di}\right]^{-1} \tag{1.6}$$

Step #11. If $0 < \dfrac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} < 30\%$, 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.

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

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'.

Total tube side pressure drop: $\Delta P_T = \Delta P_t + \Delta P_{rt}$ $\qquad$ (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.

Total shell side pressure drop: $\Delta P_S = \Delta P_s + \Delta P_{rs}$ $\qquad$ (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 Kc 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×(160-120)

=138.2°F

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

=75+0.455×(120-75)

=95.475°F

II.   Fluid properties at caloric temperature

Viscosity:

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

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

Density:

ρ_g=685 kg.m^{-3} (42.7 lb.ft^{-3})

ρ_k=800 kg.m^{-3} (49.8 lb.ft^{-3})

Thermal conductivity:

k_g=0.075 Btu h^{-1}ft^{-1} °F^{-1}

k_k=0.083 Btu h^{-1}ft^{-1} °F^{-1}

Specific heat capacity:

C_g = 0.57 Btu lb^{-1}ft^{-1}

C_k = 0.48 Btu lb^{-1}ft^{-1}

Specific gravity:

S_g = 0.685

S_k = 0.80

III.   Energy balance

Assume no heat loss to the surrounding.

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

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

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

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:

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

$$= \frac{\frac{\sqrt{0.29^2+1}\,\ln\left[\frac{(1-0.529)}{(1-0.89*0.529)}\right]}{}}{(0.89-1)\ln\left[\dfrac{2-0.529\left(0.89+1-\sqrt{(0.89)+1}\right)}{2-0.529\left(0.89+1+\sqrt{(0.89^2)+1}\right)}\right]}$$

$$= 0.802$$

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

$$LMTD = \frac{(T_2-T_1)-(t_2-t_1)}{\ln\left(\dfrac{T_2-T_1}{t_2-t_1}\right)}$$

$$= \frac{(160-120)-(120-75)}{\ln\left(\dfrac{160-120}{120-75}\right)}$$

= 42.75 °F

Determining the heat transfer area ('A'):

The value of overall heat transfer coefficient ($U_{o,assm}$) of 45 Btu $h^{-1}ft^{-2}$ °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).**

$$A = \frac{Q}{U_{assm} LMTD \times F_T}$$ (1.1)

$$= \frac{\dot{m}_g \, C_g \, (T_1 - T_2)}{U_{assm} \times LMTD \times F_T}$$

$$= \frac{142105 \times 0.57 \times (160 - 120)}{45 \times 42.75 \times 0.802}$$

$$= 2100 \text{ ft}^2$$

Calculating no. of tubes (n_t):

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

$$n_t = \frac{2100}{\pi \times \left(\frac{1}{12}\right) \times 16} = 502$$

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:

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

$$\text{Re} = \frac{4 \times (150000) \times \frac{2}{518}}{\pi \times \frac{0.834}{12} \times 3.872}$$

$$= 2740.2 < 10^4$$

As Re<<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 inch^2

No. of tubes:

$$n_t = \frac{A}{\pi d_o L_t} \tag{1.2}$$

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

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:

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

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

$= 11571.4 > 10^4$ corresponding to n_p=6.

$$u = \frac{\text{Re}\,\mu_k}{d_i \rho_k} \tag{1.3}$$

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

= 12945.15 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):

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

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

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

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

h_i= 141.3 Btu h^{-1}ft^{-1} °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)

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

=0.082 ft

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

Shell side cross flow area, $a_s = \dfrac{CBD_S}{P_T}$ (please refer to **Figure 1.6**).

C= Tube clearance

=P_T- d_o

=1¼ -1=0.25″

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

Mass velocity, $G_S = \dfrac{\dot{m}_g}{a_s} = \dfrac{142105}{0.675}$

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

$\text{Re} = \dfrac{D_e G_S}{\mu_g}$

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

$= 35668$

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

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

$$110 = \frac{h_0(0.082)}{(0.075)}\left(\frac{0.57 \times 0.484}{0.075}\right)^{-1/3}$$

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

h_o=155.3 Btu h^{-1} ft^{-2} °F^{-1}

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

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

$$U_{o,cal} = \left[\frac{1}{h_o} + R_{dg} + \frac{A_0}{A_i}\left(\frac{d_0 - d_i}{2k_w}\right) + \frac{A_0}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_0}{A_i}R_{dk} \right]^{-1} \quad (1.6)$$

Let select, Admirality brass as tube material with thermal conductivity, k_w=70 Btu h^{-1} ft^{-1} °F^{-1}.

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

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

Now, $\dfrac{U_{o,cal} - U_{o,assm}}{U_{o,assm}} = \dfrac{53.5 - 45}{45}\times 100$

$\qquad = 18.9\% < 30\%$

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 $f = 0.00028\times 144 = 0.04032$ ft^2/ft^2 for Re=11571.4 ([**3**] **page 836**]

a_t = (no. of tubes)×(flow area per tube)/(no. of passes)

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

$=0.232$ ft^2

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

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

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

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

=5.81 psi

Return loss ΔP_{rt} **:** (due to change in flow direction of the tube side fluid)

$$\Delta p_{tr} = 1.334 \times 10^{-13} \left(2n_p - 1.5\right) \dfrac{G_t^2}{S_k}$$

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

=0.73 psi

Total tube side drop neglecting nozzle loss:

$$\Delta P_T = \Delta P_t + \Delta P_{tr} \tag{1.8}$$

=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, $G_S = 210526$ lb. h^{-1}.ft^{-2}

Re=35668

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

Friction factor, $f = 0.0017 \times 144 = 0.2448$ ft^2/ft^2 with 25% cut segmental baffles (**[3] page 839**)

Shell side frictional pressure drop ΔP_s :

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

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

$$= 1.4 \text{ psi} < 7 \text{ psi}$$

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

Total shell side drop neglecting nozzle loss:

$$\Delta P_S = \Delta P_s + \Delta P_{sr} = 1.4 \text{ psi} \tag{1.9}$$

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

VII. Over surface and over design

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

The clean overall heat fransfer co-efficient: $U_C = \dfrac{h_o \times h_{io}}{h_o + h_{io}}$

$h_{io} = h_i \times \dfrac{d_i}{d_o} = 141.3 \times 0.834 = 117.8 \text{ Btu h}^{-1} \text{ ft}^{-2} \text{ °F}^{-1}$

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

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

$\qquad\qquad = 20\%$ (acceptable)

Over design:

% Overdesign $= \dfrac{A - A_{reqd}}{A_{reqd}} \times 100 \tag{1.7}$

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

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

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 co-efficient is reduced significantly.
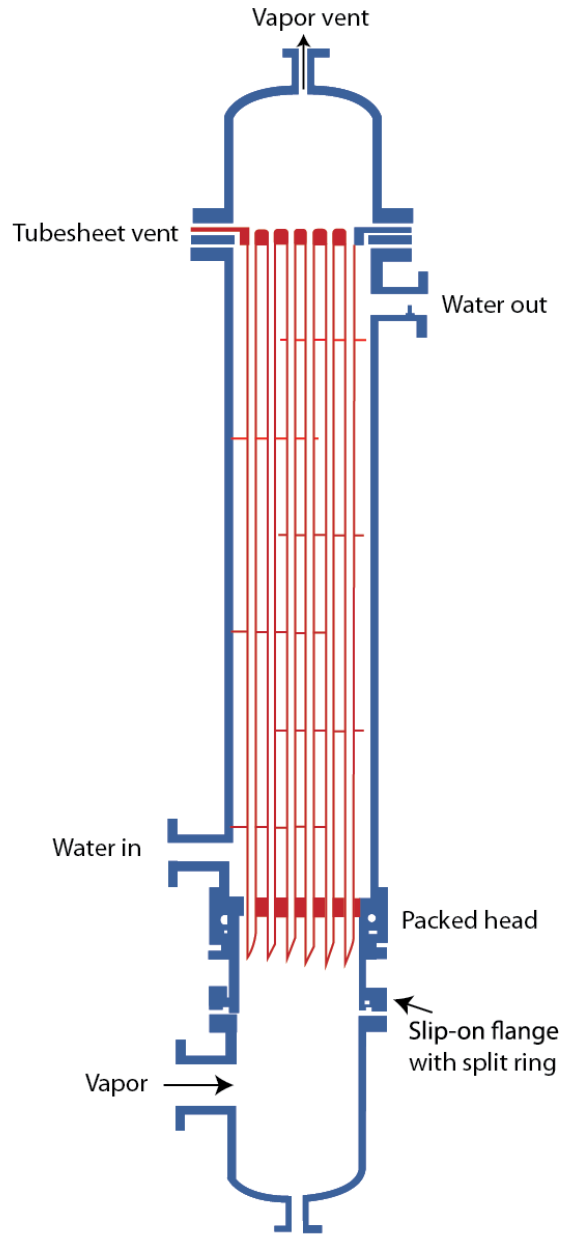
Vapor vent

Tubesheet vent

Water out

Water in

Packed head

Slip-on flange
with split ring

Vapor

Figure 1.7. Downflow vertical condenser with condensation inside tube [5].
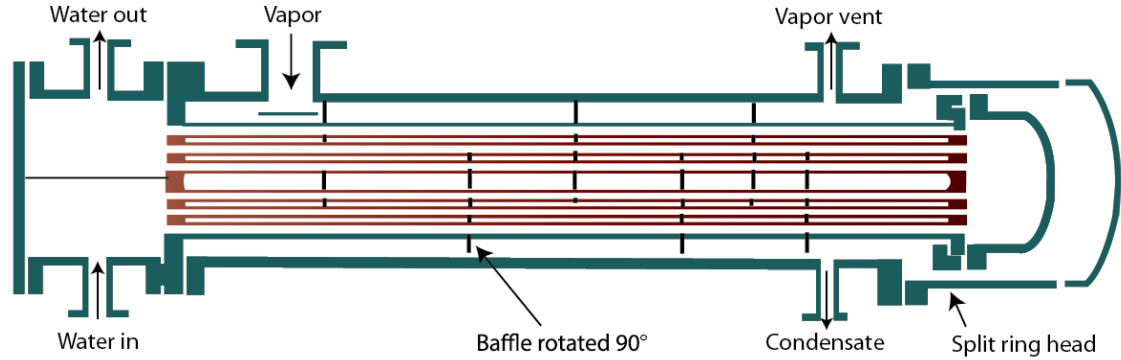
Figure 1.8. Horizontal condenser with condensation outside horizontal tubes [5].

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:

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

(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 BTU.h^{-1}.ft^{-2}.°F^{-1}. The film coefficient of condensing hydrocarbons generally varies in this range. Air-free condensing steam has a coefficient of 1500 BTU.h^{-1}.ft^{-2}.°F^{-1}.

ii. Calculate the tube wall temperature (T_w):

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

(1.11)

or

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

(1.12)

Where, $h_{io} = h_i \times \dfrac{d_i}{d_{io}}$ (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, $T_f = \dfrac{(T_w + T_v)}{2}$ (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

<u>In case of tube side condensation:</u>

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

$$\Delta P_t = \frac{1}{2}\left(\frac{fG_t^2 L_t n_p}{7.5\times10^{12} \times d_i S_t \phi_t} \right), \text{psi}$$

(1.14)

Where,

f = friction factor

G_t = mass velocity [lb. $\text{h}^{-1}.\text{ft}^{-2}$]

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

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

ii.   Shell side pressure drop

<u>In case of shell side condensation</u>: Similarly for condensation in the shell side:

$$\Delta P_s = \frac{1}{2}\left(\frac{fG_s^2 D_S \left(n_b +1 \right)}{7.5\times10^{12} \times D_e S_s \phi_s} \right), \text{psi}$$

(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)**.
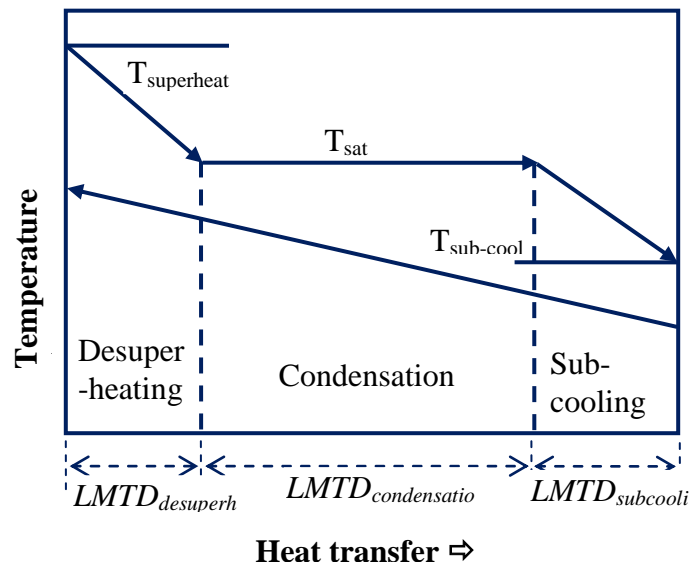

Figure 1.9. Condensation with de-superheating and sub-cooling [2].

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.
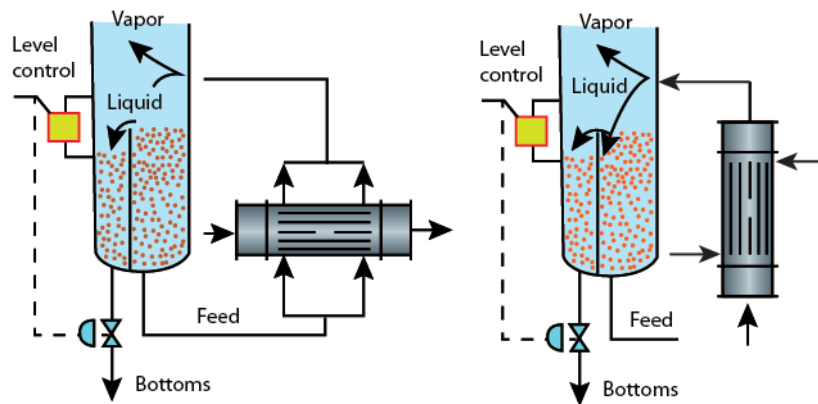

Figure 1.10. Thermosyphon reboiler [5]. (a) Horizontal thermosyphon reboiler. (b) Vertical thermosyphon reboiler

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**):

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

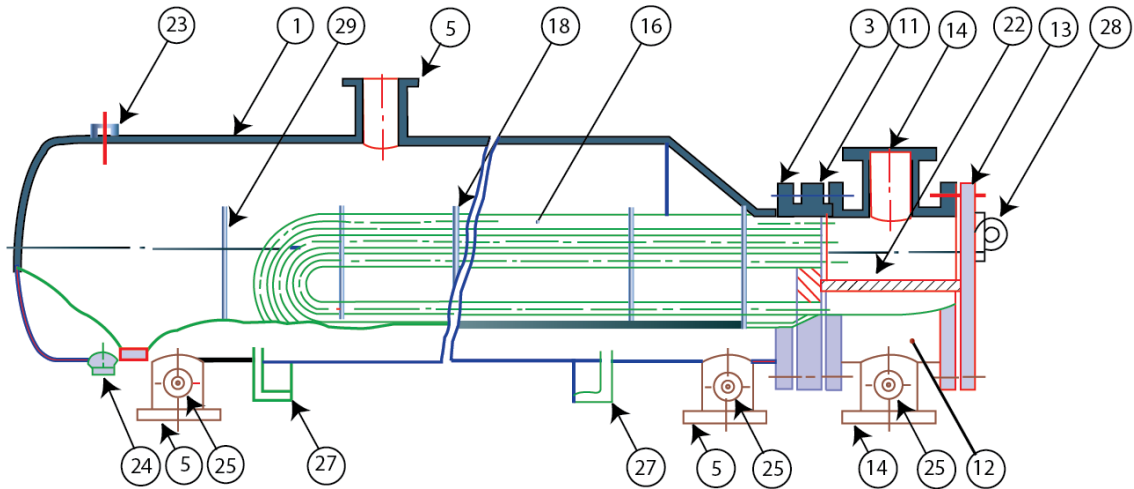| Pitch type	Constants	Number of tube passes (n_t)				
		1	**2**	**4**	**6**	**8**
Triangular ($P_T = 1.25d_o$)	K_1	0.319	0.249	0.175	0.0743	0.0365
	n_1	2.142	2.207	2.285	2.499	2.675
Square ($P_T = 1.25d_o$)	K_1	0.215	0.156	0.158	0.0402	0.0331
	n_1	2.207	2.291	2.263	2.617	2.643

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 vaporing 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.ft^2 °F for natural or forced circulation vaporizing organics.

- 1000 Btu/h.ft^2 °F for natural or forced circulation vaporizing aqueous solution of low concentration.

The maximum allowable heat flux:

- 20000 Btu/(h)ft^2) for forced circulation reboilers and 12000 Btu/(h)ft^2) for natural circulation reboilers vaporizing organics.

- 30000 Btu/(h)ft^2) for both forced or natural circulation reboilers vaporizing aqueous solution.

Assume that $h_{(assm)}$ = 300 Btu/h.ft^2 °F for organics or 1000 Btu/h.ft^2 °F for water.

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

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

(1.17)

Where, $h_{io} = h_i \times \dfrac{d_i}{d_{io}}$ (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.ft^2 °F for organics and 1000 Btu/h.ft^2 °F for water, take h_{cal} = 300 Btu/h.ft^2 °F for organics and h_{cal} = 1000 Btu/h.ft^2 °F for water.

Calculate the overall heat transfer-coefficient (U_d) including the dirt factors.
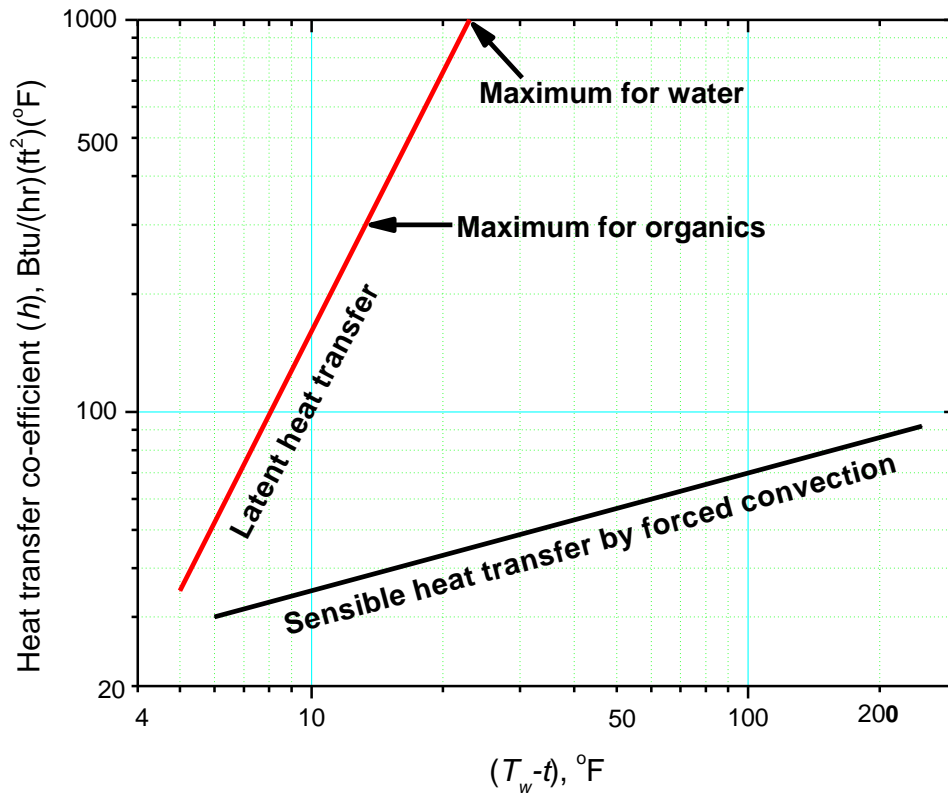
Figure 1.12. Natural circulation boiling and sensible heat transfer [3].

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 ($A = \dfrac{Q}{U_d(LMTD)}$) and number of tubes ($n_t = \dfrac{A}{\pi d_o L_t}$).

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=$\dfrac{Q}{A}$ [Btu/(h.ft^2)]. This value should be less than the maximum heat flux of 20000 Btu/(h)ft^2) for forced circulation reboilers vaporizing organics and 30000 Btu/(h)ft^2) 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.

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

(1.18)

where, ρ_l = liquid density and, ρ_l = 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

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