Module # 2

MECHANICAL DESIGN OF HEAT EXCHANGER: MECHANICAL DESIGN OF SHELL AND TUBE HEAT EXCHANGER

Lecture 1: Mechanical Design Standards

1. MECHANICAL DESIGN STANDARDS OF SHELL AND TUBE HEAT EXCHANGERS

Mechanical design of heat exchangers includes design of various pressure and non-pressure parts. The structural rigidity and satisfactory service of heat exchangers depends on the appropriate mechanical design. Mechanical design is generally performed according to the design standardsand codes. Some mechanical design standards used in heat exchanger design are: TEMA (United States), IS:4503-1967 (India);BS: 3274 (United Kingdom) and BS: 20414 (United Kingdom). The design structure of IS: 4503-1967 is provided in **Table 2.1 [1]**.

Most countries of the world follow the TEMA (Tubular Exchanger Manufacturers Association) standards for the mechanical design of **unfired shell and tube heat exchangers**. The TEMA standards are applicable for the maximum shell ID and wall thickness of 60 and 2 inch, a maximum design pressure of 3000 psi and a maximum nominal diameter (inch) × design pressure (psi) of 60000 lb/in, respectively**[2].**

Three basic classes of TEMA standards are: '**C**', '**B**' and '**R**'.

- The class '**C**' specifies the standards for general service exchangers.
- The class '**B**' specifies the standards of heat exchangers for chemical services.
- The class '**R**' specifies the standards of heat exchangers for more severe application in petroleum and related processes.

Seven types of shells are standardized by the TEMA. The TEMA standards also specify the types of front-end, shell, and rear-end of shell and tube exchangers as shown in **Figure 2.1**. For example, a fixed tube-sheet type BEM exchanger is illustrated in **Figure1. 2 of module #1**.

Table 2.1. Structure of IS: 4503-1967[1].

Part 1	Scope	Part 13	Baffles and support plates
Part 2	Types and nomenclature	Part 14	Floating heads
Part 3	Terminology	Part 15	Channels and channel covers
Part 4	Classification and permitted pressures	Part 16	Nozzles and branch pipes
Part 5	Material of constructions	Part 17	Gaskets and gland packings
Part 6	Design stress	Part 18	Flanges
Part 7	General design	Part 19	Supports
Part 8	Corrosion and corrosion allowance	Part 20	Inspection
Part 9	Shells	Part 21	Testing
Part 10	Tubes	Part 22	Marking
Part 11	Tube sheet	Part 23	Preparation for dispatch
Part 12	Shell covers	Part 24	Certificates of compliance

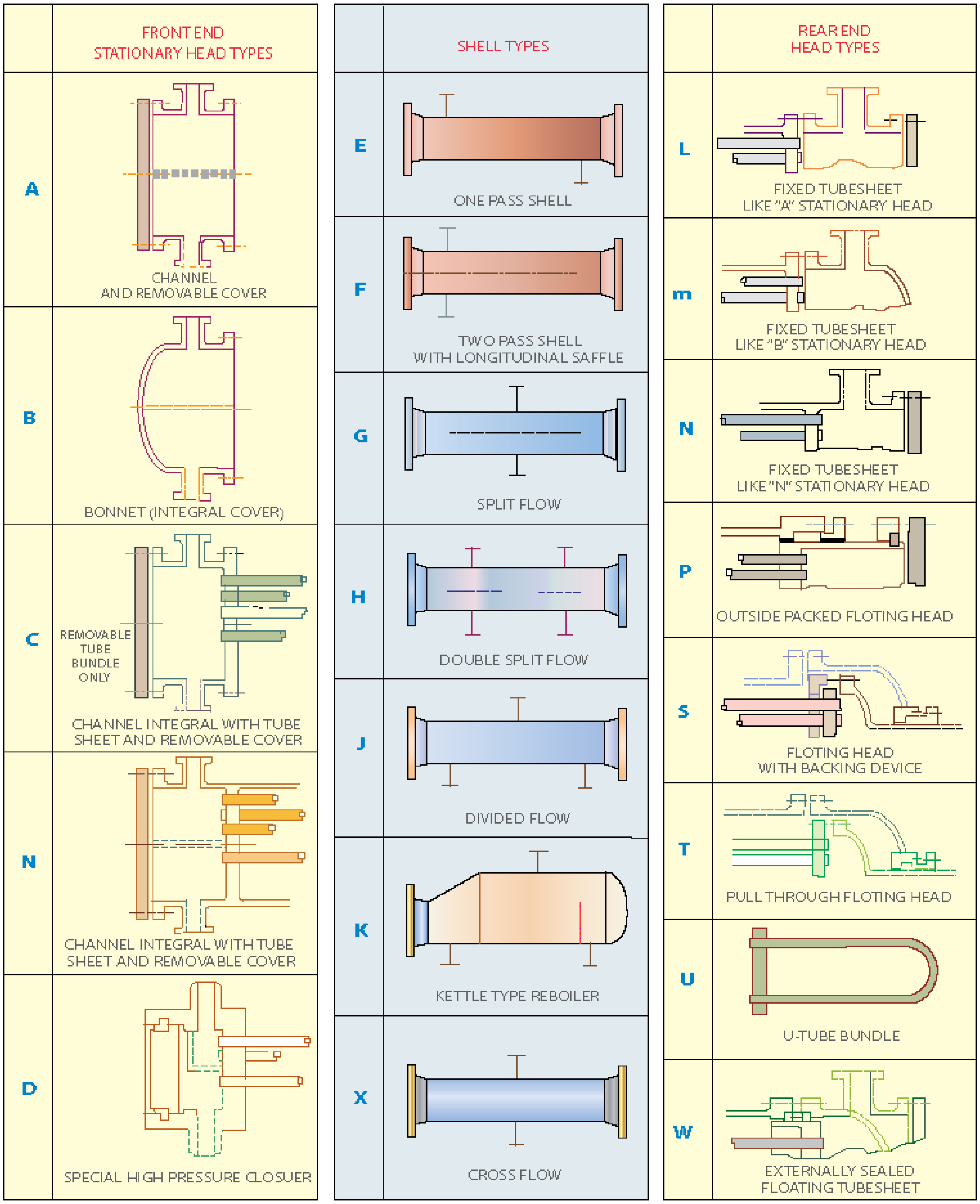

Figure 2.1. Types of shells, front end and rear ends (TEMA classifications) [3].

2. DESIGN CONSIDERATIONS

2.1. Design pressure and temperature

Design pressure of a heat exchanger is the gage pressure at the top of the vessel. This pressure is used to determine the minimum wall thickness of the various pressure parts. The IS: 4503 species that the design pressure should at least 5% greater than the maximum allowable working pressure. Usually a 10% higher value is used. The maximum allowable working pressure is the gage pressure for a specified operating temperature that is permitted for the service of the exchanger units. According the IS: 4503, the shell and tube sides pressure should be specified individually. The design pressure specification is at 250, 120 and 65ºC for carbon steel, stainless steel and non-ferrous metals respectively. The maximum permissible stresses for various heat exchanger components should not be exceeded at the allowable pressure.

The design temperature is used to determine the minimum wall thickness of various parts of the exchanger for a specified design pressure. It is normally 10ºC greater than the maximum allowable temperature.

2.2. Materials of construction

All materials used construction of shell and heat exchangers for pressure parts must have the appropriate specification as given in IS: 4503 Appendix C. The materials of construction should becompatible with process fluids and others parts of materials and also should be cost effective. The maximum permitted operating fluid temperatures should not exceeds for the various pressure-retaining components as specified by IS:4503 given in**Table 2.2 [1]**.High chrome-Mo-Ni alloys (Cr content 12-27%)can be used for high temperature services upto 2100ºC. Useof any carbon or low alloy steel is not recommended for the construction of heat exchangers forthe service below 0ºC.

Table 2.2. Materials of constructions [1].

Materials of construction	**Allowable fluid temperature, °C (°F)**
Carbon steel	540 (1004)
C-Mo steel	590 (1094)
Cr-Mo steel	650 (1202)
Low alloy steel (< 6 % Cr)	590 (1094)
Alloy steel (<17 % Cr)	590 (1094)
Austenitic Cr-Ni steel	650 (1202)
Cast iron	200 (392)
Brass	200 (392)

Lecture 2: Design Components

2.3 Design components

The major mechanical design components of shell and tube heat exchangers are: shell andtube-sheet thickness, shell cover, flanges, nozzles, gaskets, stress calculations and design of supports.

2.3.1. Shell diameter and thickness

The nominal diameter (outside diameter in millimeters rounded is to the nearest integer) of the heat exchanger is specified in**IS: 2844-1964** in case of shells manufactured from flat sheet. The following diameters (in mm) should be preferably used in the case of cylindrical pipe shell: 159, 219, 267, 324, 368, 419, 457, 508, 558.8, 609.6, 660.4, 711.2,762, 812.8, 863.6, 914.4 and 1016.

The shell thickness (t_s) can be calculated from the equation below based on the maximum allowable stress and corrected for joint efficiency **[2]:**

$$t_s = \frac{pD_s}{fJ - 0.6p} + c \qquad (2.1)$$

t_s=shell thickness

p= design pressure

D_s= Shell ID

f =Maximum allowable stress of the material of construction

J=Joint efficiency (usually varies from 0.7 to 0.9)

The minimum shell thicknesses should be decided in compliance with the nominal shell diameter including the corrosion allowance as specified by IS: 4503. Usually the minimum shell thicknesses are in order for various materials for the same service:Cast iron$>$ Carbon steel$\geq$ Al and Al-alloys (up to 700°C)$>$ Cu and Cu-alloys$\geq$ Ni$\geq$ Austenitic stainless steel$=$ Monel inconel.

2.3.2. Shell cover

There are different types shell covers used in shell and tube heat exchangers: flat, torispherical, hemispherical, conical and ellipsoidal. Out of various types of head covers, torispherical head is the most widely used in chemical industries for operating pressure up to 200psi.The thickness of formed head is smaller than the flat for the same service **[2].**Accordingthe IS: 4503, the minimum thickness of the shell cover should be at least equal to the thickness of the shell.

The required thickness of a torispherical head (t_h) can be determined by:

$$t_h = \frac{pR_iW}{(2fJ-0.2p)} + c \tag{2.2}$$

$$W = \frac{1}{4}\left(3+\sqrt{\frac{R_i}{r_i}}\right) \tag{2.3}$$

R_i= Crown radius, r_i=Knucle radius, c=corrosion allowance

2.3.3. Channel coversdiameter and thickness

The outside diameter of the channel shall be the same as that of the shell. The thickness of the channel shall be greater of the two values: (i) shell thickness or (ii) thickness calculated on the basis of the design shown below pressure.

The effective channel cover thickness (t_{cc}in mm) is calculated from the formula (IS:4503 section 15.6.1)**[1]:**

$$t_{cc} = \frac{D_c}{10}\frac{\sqrt{C_1 p}}{f} \tag{2.4}$$

D_c= diameter of the cover [mm] usually same as the outside shell diameter

C_1= a factor which is 0.25 when the cover is bolted with fullfaced gaskets and 0.3 when bolted with narrow faced or ring type gaskets

p= design pressure in **kgf/cm²**and

f= allowable stress value in **kgf/mm²** at design temperature

2.3.4. Pass partition plate

IS: 4503, specifies that the minimum thickness of channel pass partition plates including corrosion allowance should be 10 mm for both carbon steel and alloy upto channel size of 600 mm. For higher channel size, the same should be 13 mm carbon steel and 10 mm for alloy.

2.3.5. Tube sheet thickness

Tube sheet is a circular flat plate with regular pattern drilled holes according to the tube sheet layouts. The open end of the tubes is connected to the tube sheet. The tube sheet is fixed with the shell and channel to form the main barrier for shell and tube side fluids. The tube sheet is attached either by welding (called integral construction) or bolting (called gasketed construction) or a combination of both types. The typical tube sheet construction is in **Figure 2.2**.

The minimum tube-sheet thickness (TEMA standard) to '<u>resist bending'</u>can be calculated by [2]:

$$t_{ts} = \frac{FG_p}{3}\sqrt{\frac{P}{kf}} \tag{2.5}$$

Where, F=1 for fixed tube and floating type tube sheet; F=1.25 for U-tube tube sheet

G_p=diameter over which pressure is acting (for fixed tube sheet heat exchanger G_p=D_s, shell ID; $G_p$is port inside diameter for kettle type, for floating tube sheet$G_p$shall be used for stationery tube sheet).

f= allowable stress for the tube sheet material

Mean ligament efficiency (k):

$$k = 1 - \frac{0.907}{\left(\frac{P_T}{d_o}\right)^2} \text{for triangular pitch} \tag{2.6}$$

$$k = 1 - \frac{0.785}{\left(\frac{P_T}{d_o}\right)^2} \text{for square or rotated pitch} \tag{2.7}$$

The effective pressure, $P = P_s + P_b$ or $P = P_t + P_b$ when the tube sheet is extended as a flange for bolting heads.

P_s = shell side pressure, P_t = tube side pressure, P_b = equivalent bolting pressure

For fixed tube sheet and U-tube tube sheet, P is **effective shell side** or **effective tube side pressure** as defined by TEMA standards**[3]**.

The effective tube sheet to '<u>resist shear</u>' is given by:

$$t_{ts} = \frac{0.31 D_L}{\left(1 - \frac{d_o}{P_T}\right)} \left(\frac{P}{f}\right) \qquad (2.8)$$

Where, $D_L \left(= \frac{4A}{C}\right)$ is the equivalent diameter of the perforated tube sheet

C is the perimeter measured by connecting the center to center of the outermost tubes of tube layout.

A = Total area enclosed by C

The shear formula does not control the tube sheet thickness when:

$$\frac{P}{f} < 1.6 \left(1 - \frac{d_o}{P_T}\right)^2 \qquad (2.9)$$

The effective thickness of the tube sheets also can be calculated by the method given in Appendix Eof IS:4503, by trial and error approach. IS:4503 specifies that the minimum tube sheet thickness should be between 6 and 25.4 mm based on the outside tube diameter.

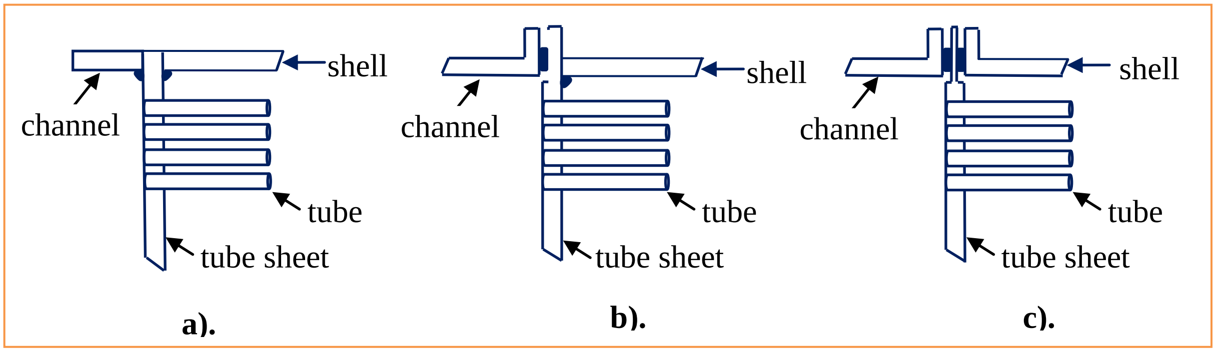

Figure 2.2. Tube sheet connections: a) Integral construction on both sides, b). one side integral construction and other side gasketed construction, c). both side gasketed construction.

2.3.6. Impingement plates or baffles

Impingement plates are fixed on the tube side between the tube bundle and inlet nozzle to deflect the liquid or vapor-liquid mixture to protect the tubes from erosion. According to the IS:4503, the protection against impingement may not be required for the services involving non-corrosive, non-abrasive, single phase fluids having entrance line values of $\rho u^2 <$125, where $u$is the linear velocity of the fluid in m/s and ρ is the density in g/cm^3. In all other cases, the tube bundle at the entrance against impinging fluids should be protected. Usually a metal plate about ¼ inch (6 mm) thick is used as the impingement plate.

2.3.7. Nozzles and branch pipes

The wall thickness of nozzles and otherconnections shall be not less than that defined for the applicableloadings, namely, pressure temperature, bending and static loads (IS:4503). Butin no case, the wall thickness of ferrous piping,excluding the corrosion allowance shall be less than $(0.04d_{oc} + 2.5)$ mm,where $d_{oc}$is the outside diameter of the connection. The typical nozzle size with shell ID is provided in **Table 2.3**.

Table 2.3. Nozzle size with shell ID.

Shell ID, inch	Nozzle ID, inch
<12	2
12 to 17.25	3
19.25 to 21.25	4
23.23 to 29	6
31-38	8
>39	10

Lecture 3: Design Components

2.3.8. Gaskets

Gaskets are used to make the metaltometal surfaces leak-proof. Gaskets are elasto-plastic materials and relatively softer than the flange materials. Deformation of gaskets under load seals the surface irregularities between metal to metal surfaces and prevents leakage of the fluid.For design pressures<16 kgf/cm^2 and when there is no contact with oil or oil vapor, the compressed asbestos fiber, natural or synthetic rubber or other suitable gasket and packing materials having the appropriate mechanical and corrosion resisting properties may be used (IS:4503).

A preliminary estimation of gaskets is done using following expression:

$Residual\ gasket\ force\ = (Gasket\ seating\ force) - (Hydrostatic\ pressure\ force)$

The residual gasket force should be greater than that required to prevent the leakage of the internal fluid. This condition results the final expression in the form of:

$$\frac{D_{OG}}{D_{IG}} = \sqrt{\frac{Y-pm}{Y-p(m+1)}} \qquad (2.10)$$

D_{OG}=outside gasket diameter [mm]

D_{IG}=inside gasket diameter [mm]; usually, D_{IG}=D_s+0.25

p=design pressure

Y= minimum design seating stress (**Table 2.4**)

m= gasket factor (**Table 2.4**)

Calculate the width of the gasket width, $N = (D_{OG} - D_{IG})/2$ (2.11)

[**The IS:4503 specifies that the minimum width of peripheral ring gaskets for external joints shall be 10 mm for shell sizes up to 600 mm nominal diameter and 13 mm for all larger shell sizes**]

Table 2.4. Gasket factors and minimum gasket seating force [4].

Gasket materials		Gasket factor $(\boldsymbol{m})$	Maximum design seating stress $(\boldsymbol{Y})$, kgf/mm^2
Flat metal jacketed, asbestos fill	Soft Al	3.25	3.87
	Soft Cu or brass	3.50	4.57
	Iron or soft steel	3.75	5.35
	Monel	3.50	5.62
	Chrome 4-6%	3.75	6.33
	Stainless steel	3.75	6.33
Solid flat metal	Soft Al	4.00	6.19
	Soft Cu or brass	4.75	9.14
	Iron or soft steel	5.50	12.65
	Monel	6.00	15.32
	Chrome 4-6%	6.00	15.32
	Stainless steel	6.50	18.28
Corrugated metal with asbestos fill	Soft Al	2.50	2.04
	Soft Cu or brass	2.75	2.60
	Iron or soft steel	3.00	3.16
	Monel	3.00	3.87
	Chrome 4-6%	3.25	3.87
	Stainless steel	3.50	4.57

2.3.9. Bolts design

The bolt design procedure is as follows:

The minimum initial bolt load (W_{m1}) at atmospheric pressure and temperature is given by:

$$W_{m1} = \pi b G Y \tag{2.12}$$

The gasket is compressed under tight pressure. The required bolt load (W_{m2}) is given by:

$$W_{m2} = H + H_P = 2\pi b G m p + \frac{\pi}{4} G^2 p \tag{2.13}$$

Where, mean gasket diameter, $G = \frac{D_{OG} + D_{IG}}{2}$ (2.14)

Total hydrostatic end force, $H = \frac{\pi}{4} G^2 p$ (2.15)

Total joint contact surface compression load, $H_P = 2\pi b G m p$ (2.16)

Effective gasket seating width, b=b_o for $b_o < \frac{1}{4}$inch (6 mm) and b=$0.5\sqrt{b_o}$ for $b_o > \frac{1}{4}$inch (6 mm)

Basic gasket seating width $b_o = N/2$ for flat flange (2.17)

Determine the controlling load: the greater value of W_{m2} or W_{m1}

Calculate the **required (minimum) bolt crosssectional area**, A_m based on the controlling load:

$$A_m = \frac{W_{m2}}{f_b} \; or \; \frac{W_{m1}}{f_a} \tag{2.18}$$

f_b = allowable bolt stress at design temperature, f_a = allowable bolt stress at ambient temperature

Select the number ofbolts (usually a multiple of 4 is used), bolt circle diameter (C_b), root diameter (d_{br}) and bolt edge distance (E)(follow IS: 4864-1968, to select bolts details).

From the number ofbolts chosen, find out the **actual bolt area (**A_b**). Always** $\boldsymbol{A_b}$ **should be greaterthan** $\boldsymbol{A_m}$.

Check for the minimum gasket width, $N_{min} = \frac{A_b f_b}{2\pi Y G}$ (2.19)

N should be greater than N_{min} .

2.3.10. Design of flange

Calculation of flange forces:

Hydrostatic end force on area inside of the flange is given, $H_D = \frac{\pi B^2 p}{4}$ (2.20)

Where, B is the centre line to centre line bolt-spacing can be taken same as outside shell diameter)

Pressure force on the flange face, $H_T = H - H_D$ (2.21)

Gasket load under operating conditions, $H_G = W - H$ (2.22)

For gasket seating condition, $H_G = W$ (2.23)

Calculation of flange moment:

Calculate the summation of flange moments for the operating condition,

$$M_f = M_D + M_T + M_G \tag{2.24}$$

Moment due to H_D, $M_D = H_D h_D$; where $h_D = (C_b - B)/2$ (2.25)

Moment due to H_T, $M_T = H_T h_T$; where $h_T = (h_D + h_G)/2$ (2.26)

Moment due to H_G, $M_G = H_G h_G$; where $h_G = (C_b - G)/2$ (2.27)

The flange bolt load, $W = \frac{(A_m + A_b) f_a}{2}$ for gasket seating condition and, (2.28)

$W = W_{m2}$ forthe operating condition (2.29)

Calculate the flange moment for the gasket seating condition: $M_f^o = \frac{W(C_b - G)}{2}$ (2.30)

Calculate the flange thickness (t_f) based on the maximum value for the gasket seating condition or operating condition given by:

$t_f = \sqrt{\frac{M_f Y}{f_f B}}$ or $\sqrt{\frac{M_f^o Y}{f_{fa} B}}$ which one is greater (2.31)

f_f = allowable flange stress at design temperature, f_{fa} = allowable flange stress at ambient temperature.

You can determine Y as a function of K. The value K is available in standard pressure vessel design book. You may find from reference **[4]** (page 238, Figure 12.22).

$K = \frac{A}{B}$; where flange OD, A =bolt circle (C_b) diameter + 2E

[The standards of flanges, gaskets, and flange faces design are given in IS: 4864-1968 to IS: 4870-1968]

2.3.11. Design of supports

The selection of the type of support for a pressure vessel depends on various parameters like the vessel elevation from the ground, materials of construction, wall thickness, operating temperature, external loads (such as wind loads, seismic condition etc).Supports for The vertical pressure vessels units are supported generally by i). skirt supports, ii). ring supports and iii).lug supports. Whereas, the horizontal pressure vessels are supported by i). saddle supports, ii). leg supports and iii). ring supports. Saddle supports are widely used in horizontal heat exchanger units.

IS:4503 specifies that the horizontal heat exchanger units shall be provided with at least two supporting saddles with holes for anchor bolts. The holes in at least one of the supports shall be elongated to provide for expansion of the shell. The vertical units shall be provided with at least two supports of sufficient size to carry the unit in a supporting structure of sufficient width to clear shell flanges.

Lecture 4: Hand on Calculations

3. SOLVED EXAMPLE

Part 2: Mechanical design

(Part 1: Thermal design calculation is given in module #1)

The process design of shell and tube for single phase heat transfer solved in **module #1** is continued for the mechanical design.

The minimum information required for the mechanical design of some important components of shell and tube exchanger is summarized below:

a. Shell side and tube side passes: 1 shell pass and 6 tube passes.

b. Number, type, size, and layout of tubes: Number of tubes 318; tube length 20´ (6.096 m as per IS: 4503-1967 and IS:2844-1964 standards); tube OD 1´´ (25.4 mm); tube ID: 0.834´´ (21.2 mm); square pitch ($P_T = 1\frac{1}{4}$"); fixed tube sheet.

c. Shell diameter and head: Shell ID 31´ (787.4 mm); torispherical head is selected;carbon steel for both shell and head.

d. Corrosion allowance: Corrosion allowance of 3 mm for carbon steel is taken as per IS:4503 for the service in the petroleum industries.

e. Design temperature and pressure: design temperature 1.1×160=176°F (80°C) (10% greater than the highest process fluid temperature is taken); design pressure 0.38 N/mm^2 (55 psia) (10% higher than the inlet pressure of both the streams).

f. Permissible stress, f =100.6 N/mm^2 for carbon steel.

i. Shell thickness calculation (refer to section 2.3.1)

$t_s = \frac{pD_s}{fJ - 0.6p} + c; J = 0.8$ (2.1)

=3.72 mm

Including corrosion allowance 6.72 mm, use 8 mm thickness

(This value is in accordance to IS:4503 corresponding to the shell diameter)

ii. Torispherical head (refer to section 2.3.2)

Crown radius, R_i = 787.4 mm (crown radius, $R_i = D_s$ is considered)

Knuckle radius r_i =0.06 of R_i = 47.24 mm (knuckle radius r_i =6% of D_s is taken)

Inside depth of the head (h_i) can be calculated as:

$$h_i = R_i - \left[\left(R_i - \frac{D_s}{2}\right)\left(R_i + \frac{D_s}{2} - 2r_i\right)\right]^{1/2}$$

= 105.4 mm

Effective exchanger length (L_{eff} **)** = $L_t + 2 \times h_i$ = 6.096 m + 2×0.1054 m

= 6.306 m

Thickness of head $t_h = \frac{pR_iW}{(2fJ - 0.2p)} + c$; J=1 is taken for head design (2.2)

$$W = \frac{1}{4}\left(3 + \sqrt{\frac{R_i}{r_i}}\right)$$ =1.77 for $r_i = 0.06 \times R_i$ (2.3)

$t_h = 2.63$; Including corrosion allowance 5.63 mm, use same thickness as for shell, i.e., 8 mm

iii. Channel cover thickness (refer to section 2.3.3)

Channel cover material: carbon steel

$$t_{cc} = \frac{D_c}{10}\frac{\sqrt{C_1 p}}{f} \qquad (2.4)$$

D_c =Outside shell diameter=803.4 mm; C_1 =0.3; p=3.88 kgf/cm^2 (0.38 N/mm^2)

f=10.26 kgf/mm^2 (100.6 N/mm^2)

t_{cc}=8.5 mm; Use 12 mm including the corrosion allowance

iv. Tube sheet thickness (refer to section 2.3.5)

The tube sheet thickness is calculated based on the bending and considering the design pressure only. It is assumed that shear does not control the design. Carbon steel is used for tube sheet material.

$$t_{ts} = \frac{FG_p}{3}\sqrt{\frac{P}{kf}} \qquad (2.5)$$

F=1 for fixed tube sheet; k =0.5 (square pitch)

t_{ts}=22.8 mm (satisfies the IS:4503 specification for 1′′ outside diameter tube)

v. Impingement plate (refer to section 2.3.6)

The density (ρ_k) of the tube side fluid (kerosene) =0.8g/cm^3 (800 kg/m^3);mass flow rate ($\dot{m}_k$) of kerosene =18.91 kg/s (150000 lb/h)

Kerosene velocity, $u_k = \frac{\dot{m}_k}{\left(\frac{\pi D_n^2}{4}\right)\rho} = \frac{18.91}{\left(\frac{\pi\times 0.2032^2}{4}\right)\times 800}$=0.73 m/s

Where, nozzle diameter,D_n= 203.2 mm=0.2032 m

The impingement parameter, $\rho v^2 = 0.8 \times 0.73^2$ =0.426<<125

Therefore the impingement protection is not required.

vi. Nozzle thickness (t_n) (refer to section 2.3.7)

Use carbon steel for the nozzle (same material)

Considering diameter of nozzle (D_n) to be 203.2 mm (8 inch) (**Table 2.3**); J=0.8

$$t_n = \frac{pD_n}{2fJ - p} + c \quad = 0.48 \text{ mm}$$

(2.1)

Use 6 mm thickness including the corrosion allowance.

The pressures at the entry point of both shell side and tube fluid are same. Therefore, the same nozzle specification can be used for tube side fluid also.

vii. Design of gaskets (refer to section 2.3.8)

$$\frac{D_{OG}}{D_{IG}} = \sqrt{\frac{Y - pm}{Y - p(m+1)}}$$

(2.10)

Gasket factor m=3.75, minimum design seating stress Y= 5.35 kgf/mm^2 (for flat iron jacketed, asbestos fill) (**Table 2.4**)

$\frac{D_{OG}}{D_{IG}} = 1.05$; D_{IG}=787.4 + 0.25=787.65 mm

D_{OG} ≈830 mm

Gasket width, $N = (D_{OG} - D_{IG})/2$=22 mm, Use 35 mm

(2.11)

Mean gasket diameter $G = \frac{D_{OG} + D_{IG}}{2} = 808$ mm

Basic gasket seating width $b_o = N/2$ =17.5 mm

Effective gasket seating width, b=0.5$\sqrt{b_o}$ = 2.09 mm

viii. Bolts (refer to section 2.3.9)

The bolt load due to gasket reaction under atmospheric conditions is given:

$$W_{m1} = \pi bGY = 278515 \text{ N}$$

(2.12)

The bolt load under tight pressure:

$$W_{m2} = 2\pi bGmp + \frac{\pi}{4}G^2 p = 15120 + 194848 = 209968 \text{ N}$$

(2.13)

Therefore, W_{m1} is the controlling load because $W_{m1}>W_{m2}$

The minimum bolt cross sectional area (bolt material carbon steel and $f_a = f_b$:

$A_m = \frac{W_{m1}}{f_a} = \frac{278515}{100.6} = 2769$ mm^2 (2.18)

M16 nominal thread diameter with bolt circle diameter (C_b) of 860 mm, 32 bolts and 18 mm root diameter (d_{br}) are selected from IS:4866-1968.

Corresponding actual bolt circle diameter, $A_b = \frac{\pi}{4} d_{br}^2 \times no.of\ bolts$=8143 mm^2

$A_b>A_m$; Therefore the selected bolts are suitable.

The minimum gasket width, $N_{min} = \frac{A_b f_b}{2\pi YG}$ (2.19)

$= \frac{8143\times100.6}{2\pi\times5.35\times808}$ =30.1 mm (compared to 35 mm selected gasket width)

ix. Flange thickness (refer to section 2.3.10)

i. For the gasket seating condition (no internal load applied)

$W = \frac{(A_m+A_b)f_a}{2} = \frac{(2769+8143)\times100.6}{2}$=548874 N (2.28)

$M_f^o = \frac{W(C_b-G)}{2} = \frac{529458\,(860-808)}{2}$ =14270714 N-mm (2.30)

ii. For operating condition

$H_D = \frac{\pi B^2 p}{4} = \frac{\pi\times803.4^2\times0.38}{4}$= 192635 N; $h_D = (C_b - B)/2$=1/2(860-803.4)=28.3 mm; (2.25)

$M_D = H_D h_D$ =5451570 N-mm

(B=Outside shell diameter=787.4+16=803.4 mm)

$H_G = W - H;\ H = \frac{\pi G^2 p}{4} = \frac{\pi\times808^2\times0.38}{4}$=194848; $W = W_{m2} =$ 207870 N (2.27)

$H_G = W_{m2} - \frac{\pi G^2 p}{4}$=207870-194848=13022 N

$h_G = \frac{(C_b-G)}{2} =$ (860-808)/2=26 mm; $M_G = H_G h_G$ =338572 N-mm

$H_T = H - H_D$ =194848-192635=2213 N, $h_T = (h_D + h_G)/2$=(28.3 +26)=27.15 mm (2.26)

$M_T = H_T h_T$=60083 N-mm

Summation moments under operating condition $M_f = M_D + M_T + M_G$=5850225 N-m

Therefore M_f^o is the controlling moment ($M_f^o > M_f$).

Flange thickness (carbon steel): $t_f = \sqrt{\frac{M_f^o Y}{f_{fa} B}}$ (2.31)

$K = \frac{A}{B} = \frac{900}{803.4}$=1.12 mm; Y =18

($A = \sim 900$mm for the chosen bolts: IS:4866-1968)

$t_f = \sqrt{\frac{14270714 \times 18}{100.6 \times 803.4}}$=56.4 mm

Practice problem: Perform the mechanical design of the condenser and Kettle type reboiler thermal design problem provided in the **module #1**.

References

[1]. Indian Standard (IS: 4503-1967): Specification for Shell and Tube Type Heat Exchangers, BIS 2007, New Delhi.

[2]. Kuppan T. Heat Exchanger Design Handbook, Marcel Dekker, Inc. 2000, New York.

[3]. Standards of the Tubular Exchanger Manufacturers Association (TEMA), Inc. 18 ed., 1999, New York.

[4]. Brownell L.E. and E.H. Young, Process Equipment Design, John Wiley and Sons, Inc. 1959. New York.