

PROCESS EQUIPMENT DESIGN

FAT SPLITTER

The Fat Splitter is a tall tower reactor where the reaction takes place. The reaction is continuous but the process is a batch process.

From literature,



The reaction proceeds to a state of equilibrium. But at high temperature the endothermic nature of Fat hydrolysis results in displacement of equilibrium in favor of split products.

The rate at which equilibrium is reached corresponds approximately to a first order reaction

$$dc/dt = k c$$

$$k = \text{Rate constant (hr}^{-1}\text{)}$$

$$t = \text{Reaction time (hr)}$$

$$c = \text{Concentration of Triglyceride (\%)}$$

From Fig. 4.1

$$dc/dt = 2.62 \% C \text{ hr}^{-1}$$

From Literature, we have

Reaction time = 3 hours

$$\text{i.e. } \tau = V/V_o$$

V = Reactor volume (m^3)

V_o = Volumetric flow rate (m^3/s)

Fat inlet = 1826.86 kg

Water inlet = 559.53 kg

Density of fat = $870.23 \text{ kg} / \text{m}^3$

Density of water at 245°C and 5 Mpa = $823.37 \text{ kg} / \text{m}^3$

Volume of fat = 2.1 m^3

Volume of Water = 0.68 m^3

Total Volume = 2.78 m^3

$$V = 3 * (V_o)$$

$$= 3 * (2.78)$$

$$= 8.34 \text{ m}^3$$

Let $L/D = 10$

Therefore $L = 10.61 \text{ m}$

$$D = 1.02 \text{ m}$$

FLOODED CHILLER—SHELL AND TUBE HEAT EXCHANGER

SHELL SIDE

On the Shell side we use the refrigerant R-22 at a temperature 2°C . The thermodynamic and thermophysical properties of R-22 at 2°C are

$$H_{fg} = 203.7 \text{ KJ/Kg}$$

$$\rho_f = 1279 \text{ KG/Kg}$$

$$\mu_f = 0.231 * 10^{-3}$$

$$K_f = 0.1 \text{ w/m k}$$

$$C_f = 1.177 \text{ KJ/Kg}$$

$$\rho_g = 22.573 \text{ Kg/m}^3$$

$$\mu_g = 1.2 * 10^{-5} \text{ Kg/ m s}$$

$$\sigma = 0.0115 \text{ N/m}$$

Let the temperature drop across the outside refrigerant film be

$$\Delta T_o = 7^{\circ}\text{C}$$

Refrigerant side Pool – boiling heat transfer co-efficient from Rosenhow Correlation is

$$h_o \Delta T = \mu_f h_{fg} \left[\frac{g (\rho_f - \rho_g)}{\sigma} \right]^{1/2} \left[C_f \Delta T_o / 0.013 h_{fg} Pr_f^{1.7} \right]^3$$

$$h_o (7) = (0.231 * 10^{-3}) (203.7 * 10^3) * \left[\frac{9.81 (1279 - 22.53)}{0.0115} \right]^{1/2} * \\ \left[\frac{1.179 * 10^3 * 7}{0.013 (203.7 * 10^3) (3.07)^{1.7}} \right]^3$$

$$h_o = 687.47 \text{ w / m}^2 \text{ k}$$

TUBE SIDE

Let's take tube side 48 tubes of 0.0131 m diameter each, with number of passes 2

Area of flow of (Methanol + Fatty acids) solution

$$= 24 * \pi / 4 * (0.0131)^2$$

$$= 0.00324 \text{ m}^2$$

$$\text{Mass flow rate (M}_w) = 5486.71 / 3600$$

$$= 1.52 \text{ kg / s}$$

$$\text{Mass velocity of water (G) = M}_w / A$$

$$= 1.52 / 0.00324 = 469 \text{ kg / s m}^2$$

$$\text{Velocity of solution} = G / \rho = 469 / 807 = 0.58 \text{ m/s}$$

$$\begin{aligned}\text{Reynolds number} &= (G * D_i) / \mu \\ &= (469 * 0.0131) / 0.001 \\ &= 61439.9\end{aligned}$$

Tube side forced convection heat transfer coefficient inside tubes using Dittus Boelter equation

$$\begin{aligned}h_i D_i / k &= 0.023 (Re)^{0.8} (Pr)^{0.3} \\ h_i (0.0131) / 0.16 &= 0.023 (61439.9)^{0.8} (65)^{0.3} \\ h_i &= 1054.88 \text{ w / m}^2\end{aligned}$$

INTEGRAL FIN AREA CALCULATIONS

Refrigerant side heat transfer coefficient is lower than tube side coefficient. Therefore to increase heat transfer integral fins on the outside is used.

A_1 = Tip areas

A_2 = Base areas

A_3 = Fin areas

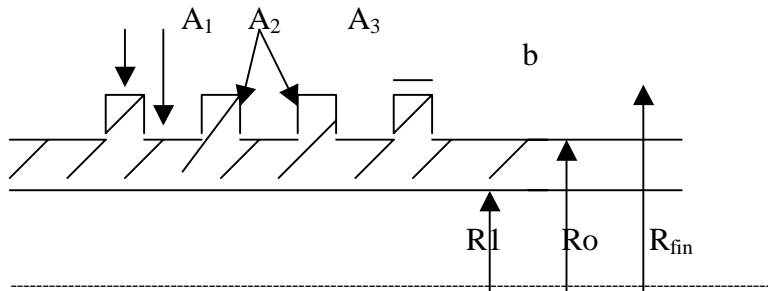


Fig 4.2 Integral Fin Tube

Calculations for per meter

$$A_1 = \pi D_1 L = \pi (0.0131) = 0.041 \text{ m}^2$$

$$A_o = \pi D_o L = \pi (0.01704) = 0.0548 \text{ m}^2$$

$$A_{fin} = 748 \pi D_{fin} b = 748 \pi (0.01905) (0.008) = 0.0358 \text{ m}^2$$

$$A_2 = A_o - A_1$$

$$= 0.190 \text{ m}^2$$

$$2A_3 = [2 \pi (\Delta_{fin}^2 - \Delta_o^2) 748] / 4$$

$$= \pi (0.1905^2 - 0.01704^2) 748 / 2$$

$$= 0.0852 \text{ m}^2$$

$$\text{Fin Area} = A_{fin} = A_2 + 2A_3$$

$$= 0.121 \text{ m}^2$$

$$A_b = A_2 = 0.019 \text{ m}^2$$

$$\begin{aligned} A_{\text{total}} &= A_b + \eta_f A_{\text{fin}} \\ &= 0.14 \text{ m}^2 \quad (\eta_f = 1) \end{aligned}$$

OVERALL HEAT TRANSFER COEFFICIENT

$$\begin{aligned} 1 / (U * A) &= 1 / (h_i * A_i) + 1 / (h_o * A_{\text{total}}) \\ &= R_i + R_o \end{aligned}$$

$$U * A = 29.84 \text{ w / m k}$$

$$\text{LMTD} = 24.3^\circ \text{ C}$$

$$\text{Heat transfer rate} = 29.84 * 24.3$$

$$= 725.12 \text{ w / m length.}$$

$$\text{Estimated Capacity} = 2 * 24 * 6 * 725.12$$

$$= 208.84 \text{ kw}$$

$$\text{Heated to be transferred} = 1.52 * 3375.92 * 40$$

$$= 205.25 \text{ kw} < \text{Estimated Capacity.}$$

Checking for ΔT_o assumption

$$\Delta T_o = R_o (725.12)$$

$$= 7.53^\circ \text{ C}$$

Assumption is good. It is only 7.5% away.

PRESSURE DROP (TUBE SIDE)

$$F = 0.079 (\text{Re})^{-1/4}$$

$$= 0.079 (6153.9)^{-1/4}$$

$$= 7.05 * 10^{-4}$$

$$\Delta H_p = (4 F L V^2) / (2 g D_i)$$

$$= (4 * 7.05 * 10^{-4} * 6 * 2 * 0.58^2) / (2 * 0.0131^2 * 9.81)$$

$$= 3.4 \text{ m}$$

$$\Delta P = \rho * \Delta H_p * g$$

$$= 807 * 3.4 * 9.81$$

$$= 26.92 \text{ kPa} < 70 \text{ kPa}$$

SECTION – 5

MECHANICAL DESIGN OF REACTOR :

The material chosen is: IS 2002 – 1962 , Grade 1.

CYLINDRICAL SHELL:

- 1) TO FIND SHELL THICKNESS t_s

P= Design pressure.

$$= 50.9 \text{ kgf} / \text{cm}^2$$

$D_i = 1020 \text{ mm}$.

F = Allowable stress

$$= 9.5 \text{ kgf} / \text{mm}^2$$

J = Joint efficiency

$$= 0.8$$

$$t_s = (P * D_i) / (200 * F * J - P)$$

$$= (50.9 * 1020) / (200 * 9.5 * 0.8 - 50.9)$$

$$t_s = 34.34 \text{ mm}$$

Corrosion allowance = 1/8 inch

$$= 3.175 \text{ mm}$$

$$t_s = 34.34 + 3.175$$

$$= 37.515 < 38 \text{ mm}.$$

Hence the thickness is allowed.

Take standard thickness of 38 mm

$$D_o = 1020 + t_s + C$$

$$= 1058 \text{ mm}$$

2) CHECKING FOR TOWER HEIGHT

Let the height of tower be X ft

$$\begin{aligned}\text{Diameter} &= D_o + (D_o / 24) + 2S_f + 2/3 \text{ icr} \\ &= 41.6 + (41.6 / 24) + 2(4) + 2/3 (21/8) \\ &= 52.5'' = 1.33 \text{ m}\end{aligned}$$

a) WEIGHT OF HEAD

$$\begin{aligned}&= (\pi d^2 t / 4) (\rho / 1728) \\ &= (\pi * 52.5^2 * (7/8) * 490) / (4 * 1728) \\ &= 537.02 \text{ lb} \\ &= 13.6 \text{ kg}\end{aligned}$$

b) AXIAL STRESS

$$\begin{aligned}f_{ap} &= (pd) / 4(t_s - l) \\ &= (691.4 * 52.5) / (4 * 0.875) \\ &= 10.371 \text{ psi} \\ &= 75041.59 \text{ kPa}\end{aligned}$$

C) CALCULATION OF DEAD WEIGHT

Since shell is of constant thickness

$$F_{\text{dead wt shell}} = 3.4 x$$

Weight of ladder = 25 lb per ft.

$$\text{Weight of 12 " Schedule 30 pipe} \quad \left| \quad = 43.8 \text{ lb per ft} \right.$$

$$\begin{aligned}\text{Weight of pipe insulation} & \quad \left| \quad = 39.3 \text{ lb per ft} \right. \\ &= \pi/4 [(1.5)^2 - (1.0)^2] 40\end{aligned}$$

Weight of top head = 537.02 lb

Total weight W = 537.02 + 107.48 lb

F_{dead weight attachments}

$$\begin{aligned}&= (W) / \pi d (t_s - C) \\ &= (537.02 + 107.48 x) / [\pi(52.5) (0.875)] \\ &= 3.72 + 0.745 X\end{aligned}$$

D) CALCULATION OF STRESS DUE TO WIND LOADS

Design wind pressure = 26 psf.

$D_{\text{eff}} = D_{\text{ia}} + \text{insulation thickness}$

$$= 52.5 + 6$$

$$= 58.5''$$

$$= 1.4859 \text{ m}$$

$$F_{\text{wx}} = (15.89 (d_{\text{eff}})^2) / (d_o^2 (ts-C))$$

$$= (15.89 \times 58.5)^2 / (52.5)^2 \times 0.875$$

$$= 0.385x^2 \text{ psf}$$

e) COMBINED STRESS UNDER OPERATING CONDITIONS

UPWIND SIDE

$$F_{t(\text{max})} = F_{\text{wx}} + F_{\text{ap}} - F_{\text{dx}}$$

$$= 0.385 x^2 - 0.745 x + 10,367.28$$

For an allowable stress of 121650 psi and joint efficiency of 0.8.

$$0.385 x^2 - 0.745x - 10367.28 = 121650$$

$$x = 127.32 \text{ ft} > 35 \text{ ft.}$$

DOWNWIND SIDE

$$F_{c(\text{max})} = F_{\text{wx}} - F_{\text{ap}} + F_{\text{dx}}$$

$$= 0.385 x^2 + 0.745x - 103678.28$$

From elastic stability

$$F_c = 1.5 \times 10^6 (0.875 / 16.2)$$

$$= 8101.85 \text{ psi}$$

$$0.385 x^2 + 0.745 x - 18469.13 = 0$$

$$x = 170.27 \text{ ft} > 35 \text{ ft}$$

The calculated height is greater. Therefore the calculated thickness is safe to take the load.

3) FLANGE DESIGN:

For internal pressure vessels with pressure greater than 300 psi, Internal flanges are designed.

DESIGN CONDITIONS

Operating Pressure - 662.3 psi

Operating temperature - 473 F

Atmospheric temperature - 100 F

Flange material - A-105 Grade 1

Bolting material - A - 193 grade B7

Corrosion allowance - 0

Allowable stress } Operating temperature $F_b = 15,000$ psi
 } Atmospheric temperature $F_a = 20,000$ psi

Allowable Flange stress } Operating temperature $F_{Fo} = 12,950$ psi
 } atmospheric temperature $F_{Fa} = 15,000$ psi

ASKET AND BOLTING CONDITIONS

Material = Asbestos.

Let gasket thickness = 1/16"

$Y = 3700$

$M = 2.75$

$$\frac{do}{di} = \sqrt{\frac{y - Pm}{y - P(m+1)}}$$

$$\frac{do}{di} = \sqrt{\frac{3700 - (662.3)2.75}{3700 - (662.3)3.75}}$$

$$= 1.04$$

$$\begin{aligned}
 d_o &= d_i (1.04) \\
 &= (40.15) (1.04) \\
 &= 41.76''
 \end{aligned}$$

$$\begin{aligned}
 \text{Minimum gasket width} \\
 &= (41.76 - 40.15) / 2 \\
 &= 0.805''
 \end{aligned}$$

Use 1" Gasket width.

$$\begin{aligned}
 \text{Mean gasket dia } G &= 41.76 + 1 \\
 &= 42.76''
 \end{aligned}$$

CALCULATION OF BOLT LOADS:

LOAD TO SEAT GASKET

$$\begin{aligned}
 W_{m2} &= H_y \\
 &= b\pi Gy.
 \end{aligned}$$

$$b_0 = n/2 = 0.5 / 2 = 0.25''$$

$$b = b_0 \text{ if } b_0 \leq 0.25 = 0.25''$$

$$\begin{aligned}
 H_y &= 0.25 (\pi) (42.76) 3700 \\
 &= 124235.97 \text{ lb.}
 \end{aligned}$$

LOAD TO KEEP JOINT TIGHT UNDER OPERATIONS

$$\begin{aligned}
 H_p &= 2b\pi GM P \\
 &= 2(0.25) \pi (42.76) (2.75) (662.3) \\
 &= 122310.32 \text{ lb.}
 \end{aligned}$$

Load from internal pressure.

$$\begin{aligned}
 H &= [(\pi G^2) / (4)] P \\
 &= [\pi (42.76)^2 / 4] (662.3) \\
 &= 950907.11 \text{ lb}
 \end{aligned}$$

Total operating load

$$\begin{aligned}
 W_{m1} &= H + H_p \\
 &= 1073217.42 \text{ lb}
 \end{aligned}$$

$$W_{m1} > W_{m2}$$

Therefore controlling load = W_{m1}

Calculation of minimum bolting area.

$$\begin{aligned} A_{m1} &= 1073217.42 / 20,000 \\ &= 53.66 \text{ in}^2 \end{aligned}$$

Take 60 bolts of 3/4" diameter on a bolt circle of 45.23"

Bolt circle diameter.

$$\begin{aligned} C &= ID + 2(1.415 + R) \\ &= 40.15 + 2(1.415 + 9/8) \\ &= 45.23'' \end{aligned}$$

CALCULATION OF FLANGE OD

Flange OD = Bolt circle diameter + 2ϵ

$$\begin{aligned} &= 45.23 + 2 (13/16) \\ &= 46.855'' \\ &= A. \end{aligned}$$

Check of gasket width:

$$\begin{aligned} A_{b \text{ actual}} &= 60 \times (0.302) \\ &= 18.12 \text{ in}^2 \end{aligned}$$

$$\begin{aligned} \text{Min gasket width} &= (A_{b \text{ actual}} * f_{\text{allow}}) / 2y\pi G. \\ &= (18.12 * 15000) / (2 * 3700 * \pi * 42.76) \\ &= 0.273 \text{ (compared with 0.5'')} \end{aligned}$$

$$\begin{aligned} W &= 0.5 (A_m + A_b) f_a \\ &= 0.5(18.12 + 53.66) 15,000 \\ &= 538350 \text{ lb} \end{aligned}$$

$$W_{m1} = 1073217.42 \text{ lb.}$$

FLANGE LOADS (OPERATING CONDITIONS)

$$H_D = \pi/4 B^2 P$$

B = Shell outside diameter

$$\begin{aligned} H_D &= \pi/4(41.64)^2 (662.3) \\ &= 901745.83 \text{ lb} \end{aligned}$$

$$\begin{aligned}
H_G &= Wm_1 - H = H_p. \\
&= 1073217.42 - 950907.11 \\
&= 122310.31 \text{ lb.}
\end{aligned}$$

$$\begin{aligned}
H_T &= H - H_D \\
&= 950907.11 - 901745.83 \\
&= 49161.28 \text{ lb}
\end{aligned}$$

LEVER ARM

$$\begin{aligned}
h_D &= r + 0.5g_1 \\
&= 9/8 + (0.5)1.75 \\
&= 2 \text{ “} \\
h_G &= 0.5(C-G) \\
&= 0.5(45.23 - 42.76) \\
&= 1.235 \text{ “} \\
h_T &= 0.5 (R + g_1 + h_g) \\
&= 0.5(9/8 + 1.75 + 1.235) \\
&= 2.055 \text{ “}
\end{aligned}$$

FLANGE MOMENTS (OPERATING CONDITIONS)

$$\begin{aligned}
M_D &= H_D * h_D \\
&= 901745.83 \times 2 \\
&= 1803491.66 \text{ lb-in}
\end{aligned}$$

$$\begin{aligned}
M_G &= H_G * h_G \\
&= 122310.31 \times 1.235 \\
&= 151053.23 \text{ lb - in}
\end{aligned}$$

$$\begin{aligned}
M_T &= H_T * h_T \\
&= 49161.28 \times 2.055 \\
&= 101026.43 \text{ lb - in}
\end{aligned}$$

$$\begin{aligned}
M_O &= M_D + M_G + M_T \\
&= 2055568.32 \text{ lb - in}
\end{aligned}$$

FLANGE LOAD (Bolting up condition)

$$H_G = W = 538350 \text{ lb.}$$

LEVER ARM

$$\begin{aligned} h_g &= 0.5(C-G) \\ &= 0.5(45.23-42.76) \\ &= 1.235 \text{ "} \end{aligned}$$

FLANGE MOMENT (bolting up condition)

$$\begin{aligned} M_a &= H_G * h_g \\ &= 538350 * 1.235 \\ &= 664862.25 \text{ lb} \end{aligned}$$

M_{\max} = the greater of M_o or $M_a * (f_{Fo}/f_{Fa})$

$$\begin{aligned} M_a * (f_{Fo}/f_{Fa}) &= 664862.25 * (12,950 / 15,000) \\ &= 573997.74 \text{ lb - in} \end{aligned}$$

$$M_o = 2055568.32 \text{ lb-in}$$

Operating moment is controlling.

$$\begin{aligned} M_{\max} &= M_o \\ &= 2055568.32 \text{ lb - in.} \end{aligned}$$

THICKNESS OF FLANGE:

$$\begin{aligned} T &= 0.72 [(M_o y) / (B f_{\text{allow}})]^{1/2} \\ &= 0.72 [(2055568.32 * 8) / (45.23 * 15000)]^{1/2} \\ &= 4.9 \text{ "} \end{aligned}$$

Shape constant $K = A/B$

$$\begin{aligned} &= \text{Flange OD} / \text{Shell OD} \\ &= 46.855/42.76 \\ &= 1.095 \end{aligned}$$

h = hub length

h_o = thickness of hub at small end

Let $g_1/g_o = 1.5$

$$g_1 = 1.75$$

$$g_o = 1.17$$

$$h_o = \sqrt{B g_o}$$

$$= \sqrt{42.76 * 1.17}$$

$$= 7.07''$$

$$h/h_o = 0.8$$

$$h = 5.65''$$

We have,

$$T = 1.8$$

$$F = 0.82$$

$$Z = 4.3$$

$$V = 0.284$$

$$Y = 8$$

$$f = 1.00$$

$$U = 8.9$$

$$e = F/h_o$$

$$= 0.82/5.65$$

$$= 0.145$$

$$d = U/V h_o g_o^2 = 303.29$$

$$\alpha = t_e + 1$$

$$= 4(0.145) + 1$$

$$= 1.58$$

$$\beta = (4/3 * t_e) + 1$$

$$= 4/3 * 0.145 + 1$$

$$= 1.193$$

$$\gamma = \alpha / T$$

$$= 1.58 / 1.8$$

$$= 0.87$$

$$\delta = t^3/d$$

$$= (5.65)^3/303.39$$

$$= 0.594$$

$$\begin{aligned}\lambda &= \nu + \delta \\ &= 0.87 + 0.594 \\ &= 1.464\end{aligned}$$

STRESS CALCULATIONS

Longitudinal hub stress

$$f_H = f^* M / \lambda g_1^2$$

$$\begin{aligned}M &= M_{\max} / B = 2055568.32 / 42.76 \\ &= 48072.22 \text{ lb}\end{aligned}$$

f^* = hub-stress correction factor = 1

$$\begin{aligned}f_H &= (1) (48072.22) / [1.464 * (1.75)^2] \\ &= 10722.03 \text{ psi}\end{aligned}$$

$$\begin{aligned}\text{Allowable stress} &= 1.5 (f_{F0}) \\ &= 1.5 (12,950) \\ &= 19,425 \text{ psi}\end{aligned}$$

Radial flange stress

$$\begin{aligned}f_R &= \beta M / \lambda t^2 \\ &= (1.193 * 48072.22) / (1.464 * 5.65^2) \\ &= 1227.15 \text{ psi}\end{aligned}$$

Allowable stress = 121950 psi.

Tangential flange stress

$$\begin{aligned}f_T &= My/t^2 - z f_R \\ &= [(48072.22 * 8) / 5.65^2] - 4.3 (1227.15) \\ &= 6770.48 \text{ psi}\end{aligned}$$

SKIRT DESIGN

Tall vessels are usually supported by cylindrical shells or skirts. The cross section of skirt is uniformly distributed from axis. This gives a large value of section modulus and helps to increase resistance to bending action. The skirt is therefore a suitable supporting structure for tall vessels which are subjected to wind, seismic and other loads which cause a bending moment at base of the vessel.

DESIGN

The cylindrical shell of skirt is designed for a combination of stresses due to vessel dead weight, wind load and seismic load. The skirt thickness is uniform and is designed to withstand the maximum values of tensile or compressive stresses.

a) STRESS DUE TO WEIGHT OF TOWER

$$f = W / (\pi * d_o * t_{sk})$$

$$W = (f_d + f_{head} + f_{\text{weight of liquid in shell}})$$

$$= 69.94 \text{ tonnes.}$$

$$f = 69944.16 / (\pi * 1.02 * t_{sk})$$

$$= 21838.45 / t_{sk} \text{ kg/m}^2$$

b) WIND LOAD

$$f_w = M * W / Z$$

$$= 4 * M * W / (\pi * 1.02^2 * t_{sk})$$

Z = Modulus of section of skirt cross-section

$$\text{Let } P = 110 \text{ kg/m}^2$$

$$P_{lw} = K * P * H * d_o$$

$$= 0.7 * 110 * 10.61 * 1.02$$

$$= 833.30 \text{ kg}$$

$$M_w = P_{lw} * H / 2$$

$$= 833.30 * 10.61 / 2$$

$$= 4420.65 \text{ kg-m}$$

$$f_w = 4 * 4420.65 / (\pi * d_o^2 * t_{sk})$$

$$= 5517.64 / t_{sk} \text{ kg/m}$$

$$\begin{aligned}\text{Maximum tensile stress at bottom} &= (21838.45 - 5517.64) / t_{sk} \\ &= 16320.81 / t_{sk} \text{ kg/m}\end{aligned}$$

$$\text{Maximum permissible stress} = 95 \text{ N/mm}^2$$

$$\begin{aligned}t_{sk} &= (16320.81 * 10^4) / (1400 * 10^3) \\ &= 1.165 \text{ mm}\end{aligned}$$

$$\begin{aligned}\text{Maximum Compressive stress at bottom} &= (21838.45 + 5517.64) / t_{sk} \\ &= 27356.09 / t_{sk}\end{aligned}$$

$$\begin{aligned}t_{sk} &= 27356.09 / 6660000 \\ &= 4.1 \text{ mm}\end{aligned}$$

$$\text{Permissible stress} = 66.6 \text{ N/mm}^2$$

MECHANICAL DESIGN FOR SHELL AND TUBE HEAT EXCHANGER

SHELL SIDE

TUBE SIDE

Shell diameter = 540 mm

Working Pressure = 0.027

N/mm²

Material = Carbon Steel

Design Pressure = 0.0297

N/mm²

Working Pressure = 0.534 N/ mm²

Design Pressure = 0.5874 N/ mm² □

Permissible stress for Carbon steel = 95 N/ mm²

SHELL THICKNESS

$$T_s = (P D) / (2f\sigma + P)$$

$$= (0.5874 * 540) / (2 * 95 * 0.85 + 0.5874)$$

$$= 1.96 \text{ mm}$$

Corrosion allowance = 2 mm

Minimum shell thickness = 6 mm

Therefore Shell thickness = 6+2 = 8 mm

NOZZLE DIAMETER FOR REMOVAL OF R-22 VAPOUR

$$\begin{aligned}\text{Vapour formation rate} &= 1.0133 \text{ kg/s} \\ &= 0.045 \text{ m}^3/\text{s}\end{aligned}$$

$$\text{Volume available for vapour occupation} = 0.707 \text{ m}^3$$

$$\begin{aligned}\text{Say the valve opens when it is 50\% filled} &= 0.5 (0.707) \\ &= 0.3535 \text{ m}^3\end{aligned}$$

$$\text{Mass flow rate} = 1.0133 \text{ kg/s}$$

$$\text{Density} = 22.573 \text{ kg/m}^3$$

$$\text{Velocity} = 0.35 \text{ kg/s}$$

$$\text{Area} = (\pi * d^2) / 4 = \text{Mass flow rate} / (\text{density} * \text{velocity})$$

$$\text{Diameter of nozzle} = 0.404 \text{ m}$$

HEAD THICKNESS (SHALLOW AND TORISPHERICAL HEAD)

$$T_h = (P R_c W) / (2 f J)$$

$$R_c = \text{Crown radius}$$

$$W = \text{Stress intensification factor}$$

$$W = 1 / 4 [3 + \sqrt{(R_c / R_k) }]$$

$$= 1 / 4 [3 + \sqrt{ (1 / 0.006) }]$$

$$= 1.77$$

$$R_c = 6\% (R_k)$$

$$J = 1$$

$$T_h = (0.5874 * 1.77 * 540) / (2 * 95)$$

$$= 2.95 \text{ mm}$$

Use the same thickness as shell.

$$T_h = 8 \text{ mm}$$

FLANGES

Shell thickness = 8 mm

Flange material – IS:2004-1962 class 2

Gasket material – Asbestos

Bolting steel – 5% Cr Mo steel

Allowable stress for Flange material = 100 MN / m²

Allowable stress for bolting material = 138 MN / m²

Outside diameter = B = 540 + 2*8

$$= 556 \text{ mm}$$

GASKET WIDTH

$$d_o / d_I = [(y - p * m) / (y - p (m + 1))]^{1/2}$$

m = Gasket factor = 2.75

$y = \text{minimum design seating stress} = 25.5 \text{ MN} / \text{m}^2$

Let gasket thickness = 1.6 mm

$$\begin{aligned} d_o / d_i &= [(25.5 - 0.5784 * 2.75) / (25.5 - 0.5874 * 3.75)]^{1/2} \\ &= 1.025 \end{aligned}$$

Let d_i of Gasket equal 550 mm

$$\begin{aligned} d_o &= 1.025 (550) \\ &= 563.75 \text{ mm} \end{aligned}$$

$$\begin{aligned} \text{Mean Gasket width} &= (563.75 - 550) / 2 \\ &= 6.875 \text{ mm} \end{aligned}$$

Taking Gasket width of 12mm

$$\begin{aligned} d_o &= 0.55 + 0.012 * 2 \\ &= 0.574 \text{ m} \end{aligned}$$

Basic Gasket width $b_o = 5 \text{ mm}$

Diameter of location of Gasket load reaction is

$$G = d_I + N$$

$$= 0.55 + 0.12$$

$$= 0.562 \text{ m}$$

ESTIMATION OF BOLT LOADS

LOAD DUE TO DESIGN PRESSURE

$$H = (\frac{\pi}{4} G^2 P) / 4$$

$$= (\frac{\pi}{4} * 0.562^2 * 0.5874) / 4$$

$$= 0.146 \text{ MN}$$

LOAD TO KEEP JOINTS TIGHT UNDER OPERATION

$$H_p = \frac{\pi}{4} * G * 2b * m * P$$

$$= \frac{\pi}{4} * 0.562 * 2 * 5 * 10^{-3} * 2.75 * 0.5874$$

$$= 28.51 * 10^{-3} \text{ MN}$$

$$\text{Total operating load} = H + H_p$$

$$= 0.175 \text{ MN}$$

Load to seat Gasket under bolting conditions

$$\begin{aligned}W_g &= \pi * G * b * y \\ &= \pi * 0.562 * 0.005 * 25.5 \\ &= 0.255 \text{ MN}\end{aligned}$$

Therefore controlling load = 0.255 MN

$$\begin{aligned}\text{Minimum bolting area} &= W_g / S_g \\ &= 0.255 / 138 \\ &= 1.63 * 10^{-3} \text{ m}^2\end{aligned}$$

CALCULATION OF OPTIMUM BOLT SIZE

Bolt size M 18 * 2

Actual no of bolts = 44

R = 0.027 m

$g_1 = g_o / 0.707$

= 1.415 g_o for weld leg

$g_o = 8 \text{ mm}$

$C = B + 2 (g_1 + k)$

= 0.556 + 2 (1.415 * 0.008 + 0.027)

= 0.633 m

Using 66mm bolt spacing

$$C = (44 * 0.066) / \pi$$
$$= 0.9245$$

Bolt circle diameter = 0.9245 m

FLANGE OUTSIDE DIAMETER

$$A = C + \text{Bolt diameter} + 0.02$$
$$= 0.93 + 0.018 + 0.02$$
$$= 0.968 \text{ m}$$

CHECK GASKET WIDTH

$$(A_b * S_g) / (\pi * G * N)$$
$$= (1.56 * 10^{-4} * 44 * 138) / (\pi * 0.562 * 0.012)$$
$$= 44.71 < 2y$$

FLANGE MOMENT COMPUTATION

a) FOR OPERATING CONDITIONS

$$W_o = W_1 + W_2 + W_3$$

$$W_1 = (\pi * B^2 * P) / 4$$

$$= (\pi * 0.556^2 * 0.5874) / 4$$

$$= 0.142 \text{ MN}$$

$$W_2 = H - W_1$$

$$= 0.146 - 0.142$$

$$= 0.004 \text{ MN}$$

$$W_3 = W_o - H$$

$$= H_p \text{ (Gasket Load)}$$

$$= 23.1 * 10^{-3} \text{ MN}$$

TOTAL FLANGE MOMENT

$$M_o = (W_1 * a_1) + (W_2 * a_2) + (W_3 * a_3)$$

$$a_1 = (C - B) / 2$$

$$= (0.93 - 0.556) / 2$$

$$= 0.187 \text{ m}$$

$$a_3 = (C - G) / 2$$

$$= (0.93 - 0.562) / 2$$

$$= 0.184 \text{ m}$$

$$a_2 = (a_1 + a_3) / 2$$

$$= 0.1855 \text{ m}$$

$$M_o = 31.55 * 10^{-3} \text{ MN}$$

b) FOR BOLTING UP CONDITIONS

$$M_g = W a_3$$

$$W = (A_m + A_b) S_g / 2$$

$$A_b = 44 * 1.56 * 10^{-4}$$

$$= 6.76 * 10^{-3} \text{ m}^2$$

$$A_m = 1.63 * 10^{-3} \text{ m}^2$$

$$S_g = 138 \text{ MN} / \text{m}^2$$

$$W = (6.76 + 1.63) * 10^{-3} * 138 / 2$$

$$= 0.578 \text{ MN}$$

$$M_g = 0.578 * 0.184$$

$$= 0.107 \text{ MN}$$

M_g is the controlling moment.

FLANGE THICKNESS

$$T^2 = (m * C_F * y) / B * S_T = (m * C_F * y) / B * S_{Fo}$$

$$K = A / B$$

$$= 0.968 / 0.556$$

$$= 1.74$$

Let C_F be 1

$$Y=4$$

$$T^2 = (0.107 * 1 * 3) / (0.556 * 100)$$

$$T = 0.075 \text{ m}$$

$$= 75 \text{ mm}$$

TUBE SHEET THICKNESS

$$T_{ts} = (F * G) (0.25 * P / f)^{1/2}$$

$$= (1 * 0.562) (0.25 * 0.5874 / 95)^{1/2}$$

$$= 22.1 \text{ mm}$$

$$T_{ts} = 25 \text{ mm (Including corrosion allowance)}$$

CHANNEL AND CHANNEL COVER

$$\begin{aligned} T_h &= G C (k * p / f)^{1/2} \\ &= 0.562 (0.3 * 0.0297 / 95)^{1/2} \\ &= 5.4 \text{ mm} \end{aligned}$$

But minimum thickness is 6 mm. Therefore including corrosion allowance the channel thickness is 8mm

NOZZLE THICKNESS

$$\begin{aligned} T_n &= (P * D) / (2f J - P) \\ &= (0.0297 * 68) / (2 * 95 * 1 - 0.0297) \\ &= 0.01603 \text{ mm} \end{aligned}$$

Inlet and outlet diameter = 68 mm

Vent = 34 mm

Drain = 34 mm

Opening for relief valve = 75 mm

Corrosion allowance = 3 mm

Considering the size of Nozzle and Pressure rating, it is necessary to provide for a reinforcing pad on channel cover. Area required to be compensated for each nozzle

$$A = t_h * d$$

$$= 68 * 8$$

$$= 544 \text{ mm}^2$$

SADDLE SUPPORT

Material – Low Carbon Steel

Diameter – 556 mm

Length of shell – 6 m

Knuckle radius = 6% (D)

$$= 33.36 \text{ mm}$$

TOTAL DEPTH OF HEAD

$$H = (D_o * r_o / 2)^{1/2}$$

$$= (556 * 33.36 / 2)^{1/2}$$

$$= 96.30 \text{ mm}$$

Weight of vessel and contents = 16667.88 kg

Distance of saddle centre line from shell end

$$\begin{aligned}A &= 0.5 * R \\ &= 0.5 * 278 \\ &= 139 \text{ mm}\end{aligned}$$

LONGITUDINAL BENDING MOMENTS

$$M_1 = Q A [1 - \{ 1 - (A/L) + (R^2 - H^2) / (2 * A * L) \} / \{ 1 + (4/3) * (H/L) \}]$$

Q = Load carried by each symmetrical support.

$$\begin{aligned}Q &= W / 2 [L + (4/3) H] \\ &= 16667.88 / 2 [6 + (4/3) 0.0963] \\ &= 51073.7 \text{ kg-m}\end{aligned}$$

$$\begin{aligned}M_2 &= (Q * L / 4) [\{ 1 + 2 * (R^2 - H^2) / L^2 \} / \{ 1 + (4/3) * (H / L) \} - (4 * A) / L] \\ &= (51073.7 * 6 / 4) [\{ 1 + 2 * (0.278^2 - 0.0963^2) / 6^2 \} / \{ 1 + (4/3) * (0.0963 / 6) \} - (4 * 0.139) / 6] \\ &= 67,927.86 \text{ kg-m}\end{aligned}$$

$$M_1 = (51073.7 * 0.278) [1 - \{ 1 - (0.139 / 6) + (0.278^2 - 0.0963^2) / (2 * 0.139 * 6) \} / 1 + (4/3)(0.0963/6)]$$

$$= 1186.31 \text{ kg-m}$$

STRESSES IN SHELL AT SADDLE

1) AT THE TOPMOST FIBRE OF CROSSSECTION

$$F_1 = M_1 / (k_1 * \pi * R^2 * T)$$

$$= 1186.31 / (1 * \pi * 0.278^2 * 0.008)$$

$$= 61.087 \text{ kg/cm}^2$$

$$F_2 = M_1 / (k_2 * \pi * R^2 * T)$$

$$= 1186.31 / (1 * \pi * 0.278^2 * 0.008)$$

$$= 61.087 \text{ kg/cm}^2$$

Stresses are well under permissible values.

2) STRESS IN SHELL AT MIDSPAN

$$F_3 = M_2 / (\pi * R^2 * T)$$

$$= 67927.86 / (\pi * 0.278^2 * 0.008)$$

$$= 34.97 \text{ kg/cm}^2$$

All the stresses are under permissible levels.