7. PROCESS EQUIPMENT DESIGN

1. ROTARY DRIER

Amount of water infeed = 212.5 kg/hr
Dry solid infeed = 10417 kg/hr
Water content in product = 105.25 kg/hr
Hence water dried in drier = 107.25 kg/hr
Inlet air temperature = 150º C
Outlet air temperature = 85º C
Inlet temperature of feed = 30º C
Discharge temperature = 80º C

Assuming wet bulb temperature of 80º C, 70% humidity of air.

The temperature of the air leaving the drier should be selected on the basis of an economic balance between drier cost and fuel cost. It has been that rotary driers are most economically operated when the total number of transfer units (NTU) range from 1.5 to 2.0. Assuming NTU = 1.5.

\[ NTU = \ln \left[ \frac{[tg_1- t_w]}{[tg_2- t_w]} \right] \]

\[ 1.5 = \ln \left[ \frac{[150- 80]}{[tg_2- 80]} \right] \]
\[ tg_2 = 95.62 ^\circ C \]
**Heat balance**

\[ C_p \text{ of } (NH_4)_2SO_4 = 1.63 \text{ kJ/kg } ^\circ C \]
\[ C_p \text{ of water} = 4.187 \text{ kJ/kg } ^\circ C \]

**Temperature detail**

<table>
<thead>
<tr>
<th></th>
<th>Feed</th>
<th>Air</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet</td>
<td>30 °C</td>
<td>150 °C</td>
</tr>
<tr>
<td>Outlet</td>
<td>80 °C</td>
<td>85 °C</td>
</tr>
</tbody>
</table>

Heat required to rise the feed to 45 °C,
\[ 10417 \times 1.63 \times (45-30) + 105.25 \times (45-30) = 261326.4 \text{ kJ} \]

Considering 55 kg of water of evaporation,

Heat required to evaporate 55 kg of water = mλ
\[ = 55 \times 2400 \]
\[ = 132000 \text{kJ} \]

Heat required to super heat the product to 80°C,
\[ 10417 \times 1.63 \times (80-45) + 27 \times 1.9 \times (80-45) = 596085 \text{ kJ} \]

Total heat required to raise the product to discharge temperature,
\[ Q_t = 261326.4 + 132000 + 596085 \]
\[ = 989412 \text{ kJ} \]

LMTD across the dryer, \( \Delta t_m \)

\[ \Delta t_m = \frac{[(150-30) - (85-80)]}{\ln[(150-30)/(85-30)]} \]

The minimum velocity of air is set based on the particle size. Air flow rate of 100 lb/hr .ft\(^3\) is sufficient for 420 microns. Hence this will be used in application. The minimum velocity is used since it gives the smallest possible size of drier.
Amount of air required:

\[ M = \frac{Q_t}{C_p \Delta t} \]

\[ = \frac{989412}{(150-85)} \]

\[ = 15221.72 \text{ kg / hr} \]

The maximum amount of water present in this amount water is 60\% i.e. 9133.03 kg / hr

An extra amount of 10\% of this quantity to account the heat losses.

\[ 1.1 \times 15221.72 = 16743.89 \text{ kg / hr.} \]

If the velocity of air is 1000lb/hr.ft = 4880 kg / hr. m\(^3\)

Area of drier = \( \frac{16743.89}{4880} \) 3.431 m\(^2\)

Diameter of the dryer = \( \sqrt{3.431 \times 4/\pi} \)

\[ = 2.09 \text{ m.} \]

Diameter of dryer = 2.09 m.

Length of transfer unit has been related to mass velocity and diameter by following relation,

\[ L_{tu} = 0.0064 \times C_p \times (G)^{0.84} \times 2.04 \]

\[ = 7.36 \text{ m.} \]

Length of the drier = \( L_{tu} \times NTU \)

\[ = 7.36 \times 1.5 \]

\[ = 11.05 \text{ m.} \]

Following dimensions for the drier are chosen.

L = 12m ; D = 2m

\( \frac{L}{D} = 12/2 \)

\[ = 6 \]

\( \frac{L}{D} \) should range in between 4 – 10. Hence the design is safe.
2. SHELL AND TUBE HEAT EXCHANGER

1) **Temperature detail:**

<table>
<thead>
<tr>
<th>Cold fluid</th>
<th>Hot fluid 15(^0)c</th>
</tr>
</thead>
<tbody>
<tr>
<td>In let</td>
<td>15(^0)c</td>
</tr>
<tr>
<td>Outlet</td>
<td>35(^0)c</td>
</tr>
</tbody>
</table>

2. **Heat load**

Hot fluid: (Aq. Ammonia)

\[
Q_H = m \times Cp \times \Delta t
\]

Where \( m = 1453 \text{ kg/hr} = 3.98 \text{ kg/sec.} \)

\( Cp = 2.57 \text{ KJ/kg} \)

\[ Q_H = 3.98 \times 2.57 \times [97-20] \]

\( Q_H = 799.99 \text{ kJ} \)

\( B_a + Q_H = Q_c = Q = 799 \text{ KJ} \)

Where \( Q_H = \text{heat load of hot fluid} \)

\( Q_c = \text{heat load of cold fluid.} \)
Cold fluid : (Water)

\[ Q_c = M_C \times C_p \times \Delta t \]

Where \( M_C \) = to be determined.

\( C_p = \) Sp heat of water.

\[ \therefore 799 = M_C \times 4.184 \times (35.15) \]

\( M_C = 9.548 \text{ kg/sec}. \)

Mass of cold water required to remove the heat associated \( \{ \} = 9.548 \text{ kg/sec}. \)

2) **LMTD Calculation \( \Delta t \)**

\( T_1 = 97^\circ \text{c} \quad t_2 = 35^\circ \text{c} \)

\( T_2 = 20^\circ \text{c} \quad t_1 = 15^\circ \text{c} \)

\[ \Delta T_{\text{LMTD}} = \frac{(97^\circ - 35) - (20 - 18)}{\ln \left( \frac{97^\circ - 35}{20 - 15} \right)} \]

**Correction factor Fr**

\[ R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \quad S = \frac{(t_2 - t_1)}{(T_1 - t_1)} \]

\[ R = \frac{(97 - 20)}{(35 - 15)} \quad S = \frac{(35 - 15)}{(97 - 15)} \]

\[ R = 3.85 \quad S = 0.2439 \]
From perry 6th cd. page 10.27
Considering 2-shell pass, 4 table pass i.e., 2.4 exchanger.

\[ F_T = 0.8 \]

Corrected LMTD = \( 22.63 \times 0.8 = 18.10 \)°
At \( \text{LMTD} = 18.10^\circ \)

3) Rounting of fluid:

- Cleaner fluid is water -------- Shell side.
- Unclean fluid is Aq.Ammonia ---- Tube side

4) Heat Transfer Area:

Reference perry, page (10-44) \( U_d \) for water in shell side, inorganic solvent in tube side is ranging between (100-250) BTU/ (F.Ft².hr)

\[ \text{Range is } = (567.83 - 1419.57) \text{ J} \]
\[ ^\circ\text{C m}^2.\text{s} \]
Total Heat Transfer Area (HTA) = \( 799 \times 10^3 \) = 73.57 m²
(600 x 18.10)

Choose l6 BWG tubes.

OD = (5/8)" = 0.01587 m
ID = 0.495" = 0.01257 m
Length of tube = 16ft = 4.8768 ft
Heat transfer area = 0.1636 ft² / ft² length
= 0.04986 m² / m.length.
Heat transfer area of over tube = 0.04968 x 4.8768
= 0.2431 m²

∴ No of tube = 73.57 = 302

Nearest tube count from perry, page 11-13 is 274. and corresponding shell diameter (inner) = 438 mm.

∴ Shell ID = 438mm.

Corrected heat transfer area = 274 x 0.243
= 66.60 m²

\[
\text{corrected } U_d = \frac{799 \times 10^3}{66.60 \times 18.1} = 662.0 \text{ W/m}^2\text{°k}
\]

5) **Fluid velocity check**

a) Tube side: (aq ammonia)

Number of passes = 4

Available flow area
\[
= \frac{\pi \times d_i^2 \times N_T}{4} \times N_p
\]

\[
= \frac{\pi \times (0.01257)^2 \times 274}{4} \times 4
\]

\[a_t = 0.0085 \text{ m}^2\]

∴ Velocity of fluid in the tube \(V_t = \frac{M_t}{a_t}\)
\[ f \times d_t \]

\[ V_t = 3.98 \times \frac{1}{832} \times \frac{1}{0.0085} \]
\[ V_t = 0.563 \text{ m/s} \]

b) Shell side: (water)

Shell ID, \( D_s = 438 \text{ mm} \)

Lc, baffle cut = 0.25 \( \times \) \( D_s \)

Ls, baffle spacing = 0.5 \( D_s = 0.219 \text{ m} \)

\[ S_m = \left[ \left( p^1 - D_0 \right) \times L_s \right] \times D_s \]

\[ p^1 = 13 \text{ inches square pitch} = 0.0206 \text{ m} \]

\[ S_m = \left( (0.0206 - 0.0158) \times 0.219 \right) \times \frac{0.438}{0.0206} \]
\[ S_m = 0.02235 \text{ m}^2 \]

Shell side velocity, \( V_s = \frac{M_s}{S_s \times S_m} \)

\[ = 9.548 \]

\[ = 997.04 \times 0.02235 \]

\[ = 0.4284 \text{ m/s} \]

No. of baffles

\[ N_b + 1 = \text{Total length of tube} \]
Baffle spacing

\[ = \frac{4.8768}{0.2} \]
\[ = 22.26 \approx 23 \]

\[ \therefore N_b = 22 \]

6) Film transfer co-efficient

a) Tube side

Richardson & coulson (Page no: 270 – 297 )

\[ A_t 55^0 c \]

\[ S = 832 \text{ kg/m}^3 \]
\[ C_p = 2.57 \text{ KJ/kg}^0 \text{k} \]
\[ M = 1.26 \text{ mN}_s / \text{m}^2 = 1.26 \times 10^{-3} N_s \]
\[ K = 0.219 \text{ w/m}^0 \text{k m}^2 \]

\[ N_{RC} = \frac{fV_s D}{M} \]
\[ = \frac{832 \times 0.563 \times 0.01257}{1.26 \times 10^{-3}} \]
\[ = 4673 \]

\[ N_{Pr} = \frac{MC_p}{K} \]
\[ = \frac{2.57 \times 10^{-3} \times 1.26 \times 10^{-3}}{14.78} \]
\[ j_H = 0.02 \]

\[ \therefore N_{Nh} = j_H \times N_{RC} \times (N_{Pr})^{\frac{1}{3}} \]

\[ N_{Nh} = 0.02 \times 4673 \times 14.78^{\frac{1}{3}} \]

\[ N_{Nh} = 229.35 \]

But, \( \frac{h_d}{k} = N_{Nh} \)

\[ \therefore h_i = 229.35 \times 0.219 \]

\[ 0.01257 \]

\[ = 3995.83 \text{ w/m}^2\text{k} \]

\[ h_i = 3995.83 \text{ w/m}^2\text{k} \]

(b) Shell side

at 25°C

\[ C = 995.045 \text{ kg / m}^3 \]

\[ C_p = 4.184 \text{ kJ / kg \degree k} \]

M = 0.95 \times 10^{-3} \text{ poise} \]

k = 1.42 w / m k

\[ N_{RC} = \frac{f \times V_2 \times D}{M} \]  

\[ \text{(D = tube outside dia )} \]

\[ M = 995.04 \times 0.4254 \times 0.01587 \]

\[ 0.095 \times 10^{-3} \]

\[ = 7620 \]

\[ N_{Pr} = \frac{M \times C_p}{K} \]
= \frac{0.95 \times 10^{-3} \times 4.18 \times 10^3}{1.42} = 2.8

From perry, page (10-29), \( j_h = 1 \times 10^{-3} \)

\[ \therefore N_{Nh} = 1 \times 10^{-3} \times 7620 \times (2.8)^{1/4} \]

\[ = 10.74 \]

but, \( h_0 d_0 = N_{Nh} \)

\[ \therefore h_0 = \frac{10.74 \times 1.42}{0.01587} = 960 \text{ w/m}^2\text{k} \]

\[ h_0 = 960 \text{ w/m}^2\text{k} \]

Overall heat transfer coefficient

\[ \frac{1}{U_0} = \frac{1}{h_0} + \frac{D_0}{D_i} + \frac{1}{h_i} + \frac{D_0 \ln (D_0/D_i)}{2k_w} + \frac{1}{h_{\text{dirt}}} \]

For stainless steel \( k_w = 45 \)

\[ h_{\text{dirt}} = 0.003 \]

\[ \frac{1}{U_0} = \frac{1}{960} + (0.01587/0.012257 \times 1/3995.53) + 0.01587 \ln \left(\frac{0.01587}{0.01257}\right)/(2 \times 45) + 1/0.003 \]

\[ U_0 = 243.096 \text{ w/m}^2\text{k} \]
7) **Pressure drop calculation:**

a) **Tube side**

\[ \Delta P_L = 4fL \frac{V^2}{gD_t} \]

but, \( f = 0.079 \times R_c^{-0.25} \)

\[ = 0.079 \times (4673)^{-0.25} \]

\[ = 0.0095 \]

\[ \Delta P_L = 4 \times 0.0095 \times 4.8768 \times (0.563)^2 \times 832 \]

\[ = 1.943 \text{ KPa} \]

\[ \Delta P_t = 2.5 (f \times Vt^2) \]

\[ \Delta P_t = 2.5 \times 832 \times (0.563)^2 \]

\[ = 0.3796 \text{ KPa} \]

\[ \Delta P_{\text{total}} = N_p \times (\Delta P_L + \Delta P_t) \]

\[ = 4 \times [1.9439 + 0.3796] \]

\[ \Delta P_{\text{total}} = 9.294 \text{ KPa} \]

\( \Delta P_{\text{total}} \) is less than 70 Kpa hence design is satisfactory.

b) **Shell side**

Shell side pressure drop is calculated using bell ‘s method

(Perry : page 10-26 to 10-31)

\( N_{RC} = 7620 \)
From figure 10.25 (a) page 10-31 friction factor \( f_k \)

\( f_k = 0.19 \)

(i) Pressure drop across cross flow section \( \Delta P_c \)

\[
\Delta P_c = b \times f_k \times w^2 \times N_c \times (Mw/Mb)^{0.4} \times f \times f_m
\]

\( b = 2 \times 10^{-3} \)

\( w = 9.54 \text{ kg/s} \)

\( S_m = 0.02235 \text{m}^2 \)

\( N_c = D_s \times \frac{1 - 2(Lc/D_s)}{P_p} \)

Where \( D_s = \text{shell 1D} = 0.438 \text{m} \)

\( Lc = 0.1095 \)

\( P_p = \text{pitch parallel (cross)flow} = \frac{13 \text{ in}}{16} = 0.0206 \text{m} \)

\[
N_c = 0.438 \times \frac{1 - 2(0.1096/0.438)}{0.0206}
\]

\[
N_c = 16
\]

\[
\therefore \Delta P_c = 2 \times 10^{-3} \times 0.19 \times 9.54^2 \times 16 \times [1]^{0.4} \]

\[
997.04 \times 0.02235 = 0.0252 \text{ KPa}
\]

\[
= 0.0252 \text{ KPa}
\]
(ii) **End zone pressure drop** \( \Delta P_c \)

\[
\Delta P_c = \Delta P_L + \left( \frac{N_{cw}}{N_c} \right)
\]

\( N_{cw} = 0.8l_s \)

\[
P_p = \frac{0.8 \times 0.1095}{0.0206} = 4
\]

\[
\therefore \quad \Delta P_c = 0.0252 \times 1 + 4
\]

\[
\Delta P_c = 0.0315 \text{ KPa}
\]

(iii) **Pressure drop in window zone** \( \Delta P_w \)

\[
\Delta P_w = b \times w^2 \left[ 2 + 0.6 k_{cw} \right] f_m \times S_w \times f
\]

\( b = 5 \times 10^{-4} \)

\( S_m = 0.02235 \text{ m}^2 \)

\( N_{cw} = 4 \)

\( w = 9.84 \text{ kg/s} \)

\( S = 997.045 \text{ kg/m}^2 \)

Area for flow though window \( S_w = S_{w_g} - S_{w_l} \)

\( S_{w_g} \), from fig (10-18), page (10-29), perry hand book.

\[
S_{w_g} = 0.029
\]
\[ Sw_i = \frac{N_t}{8} \left( 1 - F_c \right) \pi D_o \]
\[ = \frac{274 \times (1 - 0.68) \times \pi (0.0158)^2}{8} \]
\[ Sw_i = 0.0085 \]

\[ Sw = 0.029 - 0.0029 \]

\[ Sw = 0.0205 \text{ m}^2 \]

Pressure drop at window zone \( \Delta P_w \)
\[
= 5 \times 10^{-4} \times (9.54)^2 \times (2 + 0.6 \times 4) / 0.02235 \times 0.0205 \times 997.045
\]
\[ \Delta P_w = 0.386 \text{ Kpa} \]

Total pressure drop at shell side, \( \Delta P_T \) would be given by
\[
\Delta P_T = 2 \times \Delta P_c + \Delta (N_b - 1) \times \Delta P_c + N_b \Delta P_w
\]
\[
\Delta P_T = 2 \times 0.0315 + (22 - 1) \times 0.0252 + 22 \times 0.386
\]
\[ \Delta P_T = 9.08 \text{ Kpa} \]

Total pressure drop at shell side is less than 70 Kpa hence, shell & heat exchanger design is satisfactory.
1. MECHANICAL DESIGN OF ROTARY DRIER

1. **Flight design:**

   Number of flights = 3 x D.
   
   = 3 x2.09
   
   = 6.27 ≈ 7 flights are required using lip angle of 45°.

   Radial height is taken as 1/8 of diameter,
   
   Radial height = 2.09/8
   
   = 0.2615m.

2. **Thickness of dryer:**

   Let x be the thickness of drier.

   Mild steel can be used since it can withstand temperature up to 200°C.

   Density = 7688.86 kg/m³.

   D₂ – D₁ = 2x.

   Volume of mild steel = \((\pi D₂^2/4 - \pi D₁^2/4) \times L\)

   \[\begin{align*}
   &= (\pi(D₁+2x)^2/4 - \pi D₁^2/4) \times L \\
   &= \pi DLx.
   \end{align*}\]

   Weight of dryer = \(\pi \times 12.24 \times 2.09 \times x \times 7688.86\)

   = 0.626 x10⁶ kg.

   Assume holdup = 0.2

   Volume of drier filled with material = \(\pi D^3 \times 0.2\)
\[ \pi \times 2^2 \times 12 \times 0.2 \]
\[ \frac{4}{4} = 7.53 \text{ m}^3. \]

Weight of material at any time \[ = 7.53 \times 10^{49.2} \]
\[ = 11219.7 \text{ kg}. \]

The dryer is supported over two-tension roll assemblies, 20ft apart. It is uniformly distributed load.

Maximum bending moment \( = WL/8 = M. \)

\[ M = (0.626 \times 10^6 \times x/8 + 11219.7 /9) \times 12 \]
\[ = 0.939 \times 10^6x + 16829.5 \]

We know that

\[ M = f \times Z. \]

\[ Z = \pi \times (D_2^4 - D_1^4) / 32D_2. \]
\[ = 0.785x^3 + 12.59x^2 + 67.31x. \]

\[ f = 1800\text{psi}. \]

Take factor of safety \( = 5. \)

\[ f = 3.6 \times 10^5\text{lb/ft}^3. \]
\[ = 1.75767 \times 10^4\text{kg/m}^2. \]

Thus \( M = f \times Z \) on simplification becomes,

\[ 1.38 \times 10^6x^3 + 22.13 \times 10^6x^2 + 113.264 \times 10^6x - 0.819 = 0 \]

\[ x = 15 \text{ mm} \]
3. **Diameter of the feed pipe:**

Feed rate = 10417 + 212.9 = 10629.9 kg/hour  
Density of feed = 1410 kg/m³  
Hence volumetric feed rate = 10629.9 / 1410 = 7.534 m³/hr  
Assuming the velocity of air = 150 m/hr, for chute inclination of 60°  
Cross-section of feed chute = 7.53 / 150 = 0.050 m²  
Diameter of feed chute = \(\sqrt{\frac{C.S.A. \times 4}{\pi}}\) = \(\sqrt{0.050 \times 4 / \pi}\)  
= 0.252 m

4. **BHP to drive the drier:**

\[
BHP = r \times (4.75 \times d \times w + 0.1925 \times d \times w + 0.33 \times w) / 1000
\]

Where,  
\(w = \pi \times 12 \times 2 \times 0.01 \times 7688.86 + \pi / 4 \times (2^2 \times 12 \times 0.1 \times 1410)\)  
\(w = 28.6565 \times 10^3 \text{ kg}\)

**HP of blower:**  
Temperature of atmospheric = 30°C  
Humidity in air = 16743.89 kg/min = 915.5 ft³/min  
Volume of this air, \(Q = 279.05 / 29 \times 22.4 \times 303 / 298\)  
\(= 219.9 \text{ m}^3/\text{min}\)  
\(= 718.9 \text{ ft}^3/\text{min}\)
HP of blower = \(0.000157 \times Q \times \text{(head developed by water)}\)
\[= 0.000157 \times 718.9 \times 10\]
\[= 1.2 \text{ hp}\]

5. **HP of exhaust fan**

Outlet temperature of drier = 95.62 \(^{0}\) C
Humidity of outlet air = 0.65 \(\times 0.00726\)

Total quantity of air going out = 16743.9 kg/hr = 279.05 kg/min

Volume of this air = \((279.05/29) \times 22.4 \times (368.62/298)\)
\[= 406.9 \text{ m}^3/\text{min} \text{ or } 1335.13 \text{ ft}^3/\text{min}.\]

HP of exhaust fan = \(0.000517 \times 1335.13 \times 16\)
\[= 6.90 \text{ hp}\]

6. **Diameter of outlet and inlet pipe**

At inlet conditions of 150 \(^{0}\) C and humidity of 0.002

the volume of air handled = \(219.2 \times 423/303\)
\[= 306 \text{ m}^3/\text{min} \text{ or } 5.1 \text{ m}^3/\text{sec}.\]

Assuming air velocity = 25 m / s,
C.S.A of inlet pipe = \(5.1/25 = 0.20 \text{ m}^2\)

Inlet pipe diameter = 0.504 m

At outlet conditions of 95.62\(^{0}\)C

The volume of air handled = \(219.2 \times 368/313\)
\[= 4.43 \text{ m}^3/\text{sec}\]

C.S.A of outlet pipe = \(4.43/25 = 0.178 \text{ m}^2\)

Outlet pipe diameter = 0.476 m
2. MECHANICAL DESIGN OF HEAT EXCHANGER

(a) Shell side details
Material: carbon steel
Number of shell passes: 2
Working fluid: water
Working pressure: 0.1 N/mm²
Design pressure: 0.11 N/mm²
Inlet temperature: 15°C
Outlet temperature: 35°C
Permissible stress for carbon steel: 95 N/mm²
Shell inner diameter: 438 mm

(b) Tube side details
Number tubes: 274
Number of passes: 4
Outside diameter: 15.87 mm
Inside diameter: 12.57 mm.
Length: 4.88 m
Pitch triangular: 13/16 inch
Working pressure: 0.1 N/mm²
Design pressure: 0.11 N/mm²
Inlet temperature: 97°C
Outlet temperature: 20°C
SHELL SIDE

1. Shell thickness

\[ t_s = \frac{PD}{2fJ+P} \]
\[ = 0.11 \times \frac{438}{(2 \times 95 \times 0.85 + 0.11)} \]
\[ = 0.31 \]

Minimum thickness of shell must be 6.0 mm
Including corrosion allowance shell thickness is 8mm

2. Head thickness.

Shallow dished and torispherical

\[ w = \frac{1}{4} \times (3 + \sqrt{\frac{R_c}{R_K}}) \]
\[ = \frac{1}{4} \times (3 + \sqrt{\frac{R_c}{0.06 R_c}}) \]
\[ = 1.77 \]

\[ t = \frac{PR_c W}{2fJ} \]
\[ = 0.11 \times 305 \times 1.77/(2 \times 95 \times 0.85) \]
\[ = 0.528 \text{ mm.} \]

Minimum shell thickness should be 10mm including corrosion allowance.

3. Transverse Baffles

Baffle spacing = 0.8 xDc
\[ = 350 \text{ mm} \]
number of baffles,

\[ N_{b} + 1 = \frac{L}{L_s} = \frac{4.88}{0.350} = 14 \]

\[ N_b = 13 \]

Thickness of baffles, \( t_b = 6 \text{mm} \)

4. **Tie Rods and spacers:**

Tie rods are provided to retain all cross baffles and take support plates accurately.

For shell diameter, 300-500mm

Diameter of Rod = 9mm

Number of rods=4

5. **Flanges**

Design pressure=0.11 N/mm²

Flange material IS:2004-1962,class 2

Bolting steel :5% Cr-Mo steel

Gasket material: asbestos composition

Shell inside diameter = 438mm.

Shell thickness: 8mm=\( g_o \)

Outside diameter of shell: 446 mm

Allowable stress of flange material : 100MN/m²

Allowable stress of bolting material = 138 MN/m²

Shell thickness of flange = 10 mm.

Outside diameter of flange = 325 mm.

6. **Determination of gasket width**

\[ \frac{d_o}{d_i} = \left(\frac{(y-P_m)/(y-P(m+1))}\right)^{0.5} \]
Assume a gasket thickness of 10 mm

\( y \) = minimum design yield seating stress = 25.5 MN/m²

\( m \) = gasket factor = 2.75

\[
d_0/d_i = \left(\frac{(44.85-0.11 \times 2.75)/(44.85-0.11(2.75+1))}{0.5}\right)
\]

\( d_0/d_i = 1.001 = 1.001 \)

\( d_0 = 1.001 \times 0.438 = 0.4385 \text{ m} \)

Minimum gasket width = (0.4385 - 0.438)/2 = 0.00075.

Taking gasket width of \( N = 0.010 \text{ m} \)

\( d_0 = 0.458 \text{ m} \).

Basic gasket seating width, \( b_0 = 5 \text{ mm} \)

Diameter of location of gasket load reaction is

\( G = d_i + N \)

\( = 0.438 + 0.01 \)

\( = 0.448 \text{ m} \)

7. Estimation of Bolt loads.

Load due to design pressure

\[
H = \pi G^2 P/4
\]

\( = 3.14 \times 0.448^2 \times 0.11/4 \)

\( = 0.01756 \text{ MN} \)

Load to keep joint tight under operation

\( b = 2.5 \times (b_0)^{0.5} = 6.12 \text{ mm} \).

\[
H_p = \pi G(2b)mp
\]

\( = 3.14 \times 0.448 \times (2 \times 0.00612) \times 2.75 \times 0.11 \)

\( = 0.00656 \text{ MN} \)

Total operating load

\( W_o = H + H_p \)

\( = 0.01755 + 0.00656 \)
Load to seat gasket under bolting condition

\[ W_g = \pi Gby \]
\[ = 3.14 \times 0.448 \times 6.12 \times 10^{-3} \times 25.5 \]
\[ = 0.862 \text{MN.} \]

\[ W_g > W_o \text{, controlling load} = 0.8620 \text{ MN} \]

8. Calculation of optimum bolting area

\[ A_m = \frac{A_g}{S_g} = \frac{W_g}{S_g} \]
\[ = \frac{0.862}{138} \]
\[ = 6.246 \times 10^{-3} \text{ m}^2 \]

Calculation of optimum bolt size

Bolt size, M18 X 2

Actual number of bolts = 20

Radial clearance from bolt circle to point of connection of hub or nozzle and back of flange = \( R = 0.027 \text{ m} \)

\[ C = ID + 2(1.415g + R) \]
\[ = 325 + 2[1.41 \times 0.008 + 0.027] \]
\[ = 0.726 \text{m} \]

Bolt circle diameter = 0.40163 m.

Using bolt spacing \( Bs = 45 \text{mm} \)

\[ C = \frac{n Bs}{3.14} = 44 \times 0.045 / 3.14 = 0.63 \]

Hence \( C = 0.726 \)

Calculation of flange outside diameter

Let bolt diameter = 18 mm.

\[ A = C + \text{bolt diameter} + 0.02 \]
\[ = 0.716 + 0.018 + 0.02 \]

= 0.746 MN.
Check for gasket width

\[ A_b S_G / (\pi G N) = 1.54 \times 10^{-4} \times 44 \times 138 / (3.14 \times 0.4486^2) \]

\[ = 66.43 \text{ < 2 } \text{xy.} \]

where \( S_G \) is the Allowable stress for the gasket material

9. **Flange moment computation:**

(a) For operating condition

\[ W_o = W_1 + W_2 + W_3 \]

\[ W_1 = \frac{\pi B^2 P}{4} \]

\[ = \pi \times 0.446^2 \times 0.11 / 4 \]

\[ = 0.0173 \text{ MN} \]

\[ W_2 = H - W_1 \]

\[ = 0.01756 - 0.0173 \]

\[ = 1.6 \times 10^{-4} \text{ MN.} \]

\[ W_3 = W_o - H = H_p \]

\[ = 0.00672 \text{ MN.} \]

\[ M_o = \text{Total flange moment} \]

\[ M_o = W_1 a_1 + W_2 a_2 + W_3 a_3 \]

\[ a_1 = (C - B) / 2 = (0.726 - 0.446) / 2 \]

\[ a_1 = 0.14 \text{ m} \]

\[ a_3 = (C - G) / 2 = (0.726 - 0.448) / 2 \]

\[ a_3 = 0.1395 \text{ m} \]

\[ a_2 = (a_1 + a_3) / 2 = (0.14 + 0.139) / 2 \]

\[ = 0.139 \text{ m} \]

\[ M_o = 0.01739 \times 0.140 \times 1.6 \times 10^{-4} \times 0.1395 \times 0.00672 + 0.139 \]

\[ M_o = 3.391 \times 10^{-3} \text{ MN-m} \]
(b) For bolting condition

\[ M_g = W_{a3} \]

\[ W = (A_m + A_b) \times S_g / 2 \]

\[ W = (6.246 \times 10^{-3} + 6.76 \times 10^{-3}) \times 138 / 2 \]

\[ W = 0.897 \text{ MN} \]

\[ M_g = 0.897 \times 0.139 \]

\[ = 0.125 \text{ MN-m} \]

\( M_g > M_o \), Hence moment under operating condition \( M_g \) is controlling, \( M_g = M \)

10. Calculation of flange thickness

\[ t^2 = M \times C_F \times Y / (B \times S_F) \text{, } S_F \text{ is the allowable stress for the flange material} \]

\[ K = A / B = 0.764 / 0.446 = 1.71 \]

For \( K = 1.71 \), \( Y = 4.4 \)

Assuming \( C_F = 1 \)

\[ t^2 = 0.125 \times 1 \times 4.4 / (0.446 \times 100) \]

\[ t = 0.11 \text{ m} \]

Actual bolt spacing \( B_S = \pi C / n = (3.14)(0.776) / (44) = 0.052 \text{ m} \)

11. Bolt Pitch Correction Factor

\[ C_F = [B_s / (2d+t)]^{0.5} \]

\[ = (0.052 / (2 \times 0.018 + 0.11))^{1/2} \]

\[ = 0.596 \]

\[ \sqrt{C_F} = 0.772 \]
Actual flange thickness = $\sqrt{CF \times t_x}$

$= 0.11 \times 0.772$  

$= 0.085 \text{ m}$  

$= 85\text{ mm}$.

12. **Channel and channel Cover**

$t_{ch} = G_c \sqrt{(KP/f)}$

$= 0.446 \times \sqrt{(0.25 \times 0.11/95)}$  

$= 0.00767\text{ m} = 7.67\text{ mm}$

$t_{ch} = 8\text{ mm}$ including corrosion allowance.

13. **Tube sheet thickness**

$t_{ts} = FG \sqrt{(0.25P/f)}$

$= 1 \times 0.448\sqrt{(0.3 \times 0.11/95)}$  

$= 0.0084 = 8.84\text{ mm}$

$t_{ts} = 9\text{ mm}$ including corrosion allowance.

14. **Saddle support**

Material: low carbon steel

Total length of shell: 4.88 m

Diameter of shell: 4.38 mm

Knuckle radius: 6% of crown radius = 26.28 mm

Total depth of head (H) = $\sqrt{(D_0 r_v/2)}$

$= \sqrt{(438 \times 26.28/2)}$

$H = 75.86\text{ mm}$

$A = 0.5 \times R = 0.5 \times 438/2 = 109.5\text{ mm}$.

weight of vessel and contents = weight of (shell + tube)

weight of the steel = 7600 kg/m$^3$. 
Weight of shell = \(\pi D \times 0.008 \times 7600 \times L\)
= 83.67 \times 4.88
\[= 408.30 \text{ kg}\]

Weight of tube = \(\pi (19.05 \times 10^{-03} - 12.27 \times 10^{-03}) \times 4.88 \times 7600 \times 274\)
= 1480 kg

Weight of water = \(\pi 0.01224^2 \times 4.88 \times 995 \times 274\)
= 626.19 kg

Weight of vessel and contents \(W = 2215.69 \text{ kg}\)

15. **Longitudinal Bending Moment**

\[M_1 = QA\left[1-(1-A/L+(R_2-H_2)/(2AL))/(1+4H/(3L))\right]\]

\[Q = W/2(L+4H/3)\]

\[= 2215.6/2 \times (4.88 + 4 \times 0.07586/3)\]

\[= 5518 \text{ kg m}\]

\[M_1 = 598.05 \text{ kg-m kg-m}\]

16. **Bending moment at center of the span**

\[M_2 = QL/4[(1+2(R_2-H_2)/L)/(1+4H/(3L))-4A/L]\]

\[M_2 = 6014.4 \text{ kg-m}\]

17. **Stresses in shell at the saddle**

(a) At the topmost fibre of the cross section

\[f_1 = M_1/(k_1\pi R^2 t)\]

\[= 598.06 / (3.14 \times 0.219^2 \times 0.01)\]

\[= 35.22 \text{ kg/cm}^2\]

The stresses are well within the permissible values.

Stress in the shell at mid point

\[f_2 = M_2/(k_2\pi R^2 t) = 6014.4 / (1 \times \pi 0.219^2 0.01)\]
Axial stress in the shell due to internal pressure

\[ f_p = \frac{PD}{4t} \]

\[ = 0.11 \times \frac{438}{(2 \times 10)} \]

\[ = 225.53 \text{ kg/cm}^2 \]

\[ f_2 + f_p = 624.64 \text{ kg/cm}^2 \]

The sum \( f_2 \) and \( f_p \) is well within the permissible values.