CHAPTER 6

EQUIPMENT DESIGN

(A) DESIGN OF REACTOR:

1. PROCESS DESIGN:

The nitration of toluene is an important heterogeneous liquid-phase reaction. It is generally considered that the function of the sulfuric acid is to maintain the nitric acid in a "dehydrated" state as a result of its strong for water. If the nitration is carried out with strong nitric acid alone the rate of reaction is rapid at the start but drops off quickly as the water formed by the reaction dilutes the acid. This reduction in rate as the reaction proceeds is greatly lessened by the presence of sulfuric acid or some other "dehydrator".

Rates of mono nitration when mixed nitric and sulfuric acid is employed have been measured by McKinley and White on toluene. The investigators made measurements in continuous equipment under steady-state conditions with high degrees of agitation to ensure distribution equilibrium.

Thermodynamic and experimental data indicate that the equilibrium constant is very large for the nitration of aromatic under liquid- phase conditions. Accordingly the reverse reaction may be neglected and the rate reaction equation below can be written as follows if it is assumed that agitation is sufficient to approximate equilibrium distribution between the phases. *(Ref. – 2)*

$$r = k\, x_{Aa}\, \gamma_{Aa}\, x_{Bb}\, \gamma_{Bb}\, (V_a + V_b k') \quad \text{---------------------- (1)}$$

where $k = k_a K_B$

$k' = k_b K_A / k_a K_B$

V_a = fractional volume of phase a

Suffix a - acid layer and

b - nitration product (organic phase)

A - nitric acid

B - Toluene

x_{Aa} = mole fraction of HNO_3 in acid phase.

x_{Bb} = mole fraction of toluene in organic phase (nitration product)

γ_{Aa} = activity coefficient of HNO_3 in acid layer.

γ_{Bb} = activity coefficient of toluene in nitration product

The overall reaction velocity constant k is a function of temperature, which may be expressed by the Arrhenius equation.

$$\ln k = - E / RT + A \qquad \text{---------------------- (2)}$$

In applying equation (1) the mutual solubility of the phases may be neglected and x_{Aa} is based on the total HNO_3, H_2SO_4 and H_2O content of the system while x_{Bb} is based on the total toluene plus nitrotoluene.

The data of McKinley and White indicate that for the mono nitration of the toluene the activity coefficient of the toluene γ_{Bb} may be taken as *unity* over a wide range of conditions. It was also found that $K' = 0$, indicating that the reaction taking place in the organic phase is negligible. With these terms evaluated, rate data at a constant temperature may be used for empirical evaluation of the relative activity coefficients of the nitric acid γ_{Aa}

With the assumption that the activity coefficients of Fig. 210 are independent of temperature, the other constants of equation s (7) and (8) for the mono nitration of benzene and toluene in the temperature range from 15 to 50 °C are as follows:

γ_{Bb}	1.0
K'	0
E cal per g-mole	14,000
A (rates in g-mole per hr per liter of combined phases)	27.58

$\Rightarrow$ $\ln k = -E / RT + A$

$= -14000 / (1.987 \times (273 + 35)) + 27.58$

$= -22.876 + 27.58$

$\Rightarrow$ $k = 110.39$ g mol/ hr lt

The following tables indicate the mole fractions of the various components in both the inlet and outlet streams of the reactor:

Inlet to the reactor:

Mixed Acid:

	Tons/day	Kg/hr	Kgmol/hr	Weight %	Mol %
HNO_3	13.094	545.58	8.66	5 %	3.82 %
H_2SO_4	188.75	77864.58	80.251	72.13 %	35.43 %
H_2O	59.381	2474.21	137.456	22.69 %	60.69 %
MNT	0.457	19.04	0.139	0.18 %	0.06 %
TOTAL	261.682	10903.41	226.506	100 %	100 %

Organic Phase:

	Tons/day	Kg/hr	Kgmol/hr	Weight %	Mol %
TOLUENE	18.105	754.38	8.1998	99 %	99.2 %
PARAFFIN	0.183	7.62	0.067	1 %	0.8 %
TOTAL	18.288	762	8.2668	100 %	100 %

Assuming 97 % conversion in the reactor, the composition of mixed acid and organic phases in the outlet stream are as follows.

Outlet from the reactor:

Acid Layer:

	Tons/day	Kg/hr	Kgmol/hr	Weight %	Mol %
HNO3	0.585	24.383	0.387	0.23 %	0.17 %
H2SO4	188.696	7862.316	80.228	74.71 %	35.51 %
H2O	62.710	2612.916	145.162	24.83 %	64.24 %
MNT	0.585	24.384	0.178	0.23 %	0.08 %
TOTAL	252.570	10524	225.955	100 %	100 %

<u>Nitration Product:</u>

	Tons/day	Kg/hr	Kgmol/hr	Weight %	Mol %
MNT	26.262	1094.25	7.987	96.51 %	94.20 %
NITRO CRESOL	0.293	12.192	0.080	1.08 %	0.94 %
TOLUENE	0.183	7.62	0.083	0.67 %	0.98 %
PARAFFINS	0.183	7.62	0.067	0.67 %	0.79 %
H_2SO_4	0.219	9.144	0.093	0.81 %	1.10 %
H_2O	0.073	3.048	0.169	0.26 %	1.99 %
TOTAL	27.213	1133.874	8.479	100 %	100 %

From equation (1),

$r = k\, x_{Aa}\, \gamma_{Aa}\, x_{Bb}\, \gamma_{Bb}\, (V_a + V_b k')$

Since $k' = 0$,

$r = k\, x_{Aa}\, \gamma_{Aa}\, x_{Bb}\, \gamma_{Bb}\, (V_a)$

where, γ_{Aa} = activity coefficient of HNO_3 in acid layer.

γ_{Bb} = activity coefficient of toluene in nitration product

From graph $\gamma_{Aa} = 150$, $\gamma_{Bb} = 1$.

And,

$x_{Aa} = 0.0017$, $x_{Bb} = 0.0098$

Density of Acid Layer: $(\rho)_{AL} = 1506.4\ kg/m^3$

Density of Nitration Product: $(\rho)_{NP} = 1146.9\ kg/m^3$

$\Rightarrow$ Volume of the acid layer: $(V)_{AL}$ = 10524/1506.4 = 6.98624 m3/hr

= 6986.24 lt/hr

Volume of the nitration product: $(V)_{NP}$ = 1133.874/1146.9 = 0.98863 m3/hr

= 988.63 lt/hr

Total Volume = $V_{Total} = (V)_{AL} + (V)_{NP}$

= 6986.24 + 988.63 = <u>7974.87 lt/hr</u>

Now, $V_a = (V)_{AL} / V_{Total} = 6986.24 / 7974.87$

$= 0.876$

In equation (1);

$\Rightarrow$ $r = 110.39 \times 0.0017 \times 0.0098 \times 150 \times 1 \times 0.876$

$r = 0.24166$ gmol / lt hr.

Toluene feed rate = F_{AO} = 8.1998 kgmol/hr

Therefore Volume of the Reactor:

$V_R = F_{AO} (X_{Af} - X_{AO}) / (-r)$

$= 8.1998 (0.97 - 0) / 0.24166$

$V_R =$ <u>$33\ m^3$</u>

2. MECHANICAL DESIGN OF REACTOR:

Volume of the reactor = V = 33 m^3

Assuming L/D = 3

$V = (\pi D^2/4) \times L = (\pi/4) \times 3(D)^3$

$\Rightarrow$ Diameter, $D = (4V/3\pi)^{1/3} = [4 \times 33 / (3 \times \pi)] = 2.4$ m

$\Rightarrow$ Height, $L = 33/(\pi \times 2.4^2/4) = 7.3$ m

(Ref. – 8,10,11)

Turbine Agitator Design:

We use 6 blade turbine agitator .The diameter of impeller varies from 30 to 50 % of tank diameter.

Assuming that turbine operates at 200 rpm .

Diameter of reactor = 2.4 m

Diameter of agitator = 0.34 x 2.4

= 0.816 m

i.e using 34% of diameter of reactor as impeller diameter.

Density of mixed acid and mononitrotoluene, ρ =1461.66 kg/m^3

Viscosity of mixture, μ = 4 cP

$N_{Re} = \rho N da^2/\mu = [1461.66 \times 200/60 \times (0.816)^2] / 4 \times 10^{-3} = 811045.9 > 10{,}000$

From M.V. Joshi, Process Equipment Design, 2nd edition

From power curve, N_p = 6.1 for Reynolds number greater than 10,000

From equation 14.1

Power, $P = N_P \rho N^3 Da^5/(g_c \times 75)$

$= (6.1 \times 1461.66 \times (200/60)^{3)} \times (0.816)^5) / (9.81 \times 75)$

= 162.38 hp

Gland losses (10%) = 16.238 hp

Power input = 162.38 + 16.238 = 178.62 hp

Transmission system losses (20%) = 178.62 x 0.2

= 35.724 hp

Total hp = 178.62 + 35.724 = 214.344 hp

This will be taken as 220 hp to allow for fitting losses

∴ It is advisable to use 220 hp motor.

Shaft design

Continuous average rated torque on the agitator shaft,

Tc= (hp x 75 x 60)/ (2 π N)

= (220 x 75 x 60)/ (2 π x 200)

= 787.82 Kg m

Polar modulus of the shaft cross section is,

$Z_p = T_m/f_s$

Where T_m is maximum torque = 1.5Tc = 1181.73 kg m

f_s – shear stress – 550 kg/cm^2

Z_p =(1.5 x 787.82 x 100) /550

= 214.86 cm^3

$\pi d^3/16$ = 214.86

d = 10.3 cm

Torque T_m is resisted by force F_m acting at a radius of $0.75R_b$ from the axis of agitator shaft.

Where Rb is radius of blade.

Force, $F_m = T_m/0.75R_b$

F_m = (1.5 x 787.82 x 100) / (0.75 x 25)

= 6302.6 kg

Maximum bending momentum

M = F_m x l

Where 'l' is length of overhang of agitator shaft between bearing and agitator.

Let 'l'= 1500mm

$\therefore$ M = 6302.6 x 1.5 = 9453.9 Kg-m

Equivalent bending moment

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T_m^{\ 2}}\ \right]$$

= ½ [9453.9 + $(9453.9^2 + 1181.73^2)^{1/2}$]

= 9490.7 Kg-m

The stress due to equivalent bending

F = M_e/Z

Z = $\pi(10.3)^3/32$ (Modulus of section of the shaft cross section)

Z = 107.3 cm^3

$\therefore$ f = (9490.7 x 100)/107.3

= 8845 kg/cm^2

Stress f is higher than the permissible elastic limit (2560 Kg/Cm^2). Therefore use a *16 cm* diameter shaft for which the stress will be,

f = 2360 Kg/cm^2

Deflection of shaft

$\delta = (Wl^{3)}/(3\ EI)$ [$W = F_m$]

Where E is the modulus of elasticity = 19.5 x 10^5 Kg/cm^2

$\therefore \delta = [6302.6\ (150)^3\ ]/[3 \times 19.5 \times 10^5 \times (\pi \times 16^4)/64]$

= 1.13 cm

Critical speed , $N_c = (4.987\ \ x\ 60)\ /\sqrt{\delta}$

= 282 rpm

Since actual shaft speed is 200 rpm which is 71% of the critical speed. Therefore it is necessary to increase the value of critical speed by decreasing the deflection.

Choose therefore 17cm diameter shaft.

Then,

δ = 0.887 cm

And N_c = 60 x 4.987/ 0.887 = 338 rpm

Actual speed is 60% of the critical speed

Blade design

Using blade width, W = 75mm

Blade thickness, t = 8mm

Number of blades = 6

Stress in the blade, F =(maximum torque)/(tw^2/n)

=(10.74*10)/(0.8 x 7.5^2/6)

= 143 Kg/cm^2

The value of stress is well within the limit for carbon steel.

Hub and key design

Hub diameter of agitator = 2 x shaft diameter

= 2 x 17

= 34 cm

Length of the hub = 1.5 x shaft diameter = 34 cm

Length of key = 1.5 x shaft diameter = 34 cm

T_{max}/(d/2)= lbf_s = (lt/2)f_c = (1181.73 x 100)/(17/2)

f_s- shear stress in key (for carbon steel it is = 650 kg/cm^2)

f_c – stress in crushing of key(for carbon steel it is = 1300 kg/cm^2)

34 x b x 650 = 34 x t/2 x 1300 = 13902.7

on solving,

b = 6 mm

t = 6 mm

Use 6mm x 6mm x 34 cm key

Stuffing box and gland

Internal pressure = 1.1*10^5 Pa

= 1.2 kg/cm^2

Internal diameter of stuffing box (in cm)

B = d+√d

= 17 + √17 = 21.12 cm

Thickness of stuffing box in mm, t = PB/2f + C

Permissible stress in the material of stuffing box, 'f' = 950 kg/ cm^2

t = (1.12 x 21.12 x 10) /(2 x 950) + 17

= 17.12 mm

Load on gland,

$F = (\pi/4)\ p(b^2 - d^2)$

$= (\pi/4)(21.12^2 - 17^2)1.12$

= 138 Kg

Size of the stud

$F = (\pi d_o^2/4)\ nf$

n is the no of studs = 4

f is the permissible stress for stud = 587 Kg/cm^2

$d_0^2 = 0.0748$ cm

$d_0 = 0.27$ cm

But minimum stud diameter of 15 mm should be provided.

∴Stud diameter = 15 mm

Flange thickness = 1.75 x d

= 1.75 x 15 = 27mm

<u>Coupling: -</u>

A clamp coupling of cast iron is used

Force per bolt = 2 $T_{max}/(\pi\mu d\ (n/2))$

No of bolts, n= 8 (for shaft diameter of greater than 5cm)

μ - Coefficient of friction = 0.25

Force per bolt= (2 x 1181.73 x 100) / (π x 0.25 x 17 x (8/2))

= 4425.36 kg

Area of bolt = (force on bolt)/(shear stress on bolt)

=4425.36/587

= 7.5 cm^2

Diameter of bolt= (7.5 x 4)/π

= 9.5 cm

Overall diameter of coupling = 2 x shaft diameter

= 2 x 17

= 34 cm

Thickness of reactor vessel

Allowable stress value, f = 1130 kg/cm^2 (upto 150^0 C for carbon steel)

Thickness of reactor, t = (PD)/[(2fj)-P] + C

Where

'j' is joint efficiency = 0.85

'P' is the design pressure

$P = 1.1\ P_{actual}$

= 1.1 x 1.05

= 1.156 kg/cm^2

∴t = (1.156 x 2400)/[(2 x 1130 x 0.85)-1.156] + 3

= 4.5 mm with corrosion allowance

∴Thickness of reactor vessel = t = 6 mm

Design of reactor head

Using Flat head

Thickness of head, t = (CD/10) x (√(P/f))

Where 'C' is taken as 0.5

(from table 3.27, IS 2825 – 1969)

∴ t = (0.5 x 2400/10) x (1.156 x 100/1130)$^{0.5}$

= 38 mm

= 40 mm with 2mm allowance for corrosion.

Conical bottom thickness

T_h =PD /(2 x f x J x (cosα)

Where

$\alpha = 60^0 / 2$

$= 30^0$

$\therefore T_h$ = 1.156 x 2400 / (2 x 1130 x 0.85 x cos30) + C

= 1.7 + 3

= 4.7 mm with corrosion allowance.

Nozzle diameter for Toluene inlet:

Feed rate of toluene = 762 Kg/hr.

= 0.212 Kg/s

Assuming velocity, v_1 = 0.01 m/s

Area of nozzle = m_t /(v_1 x $\rho_{1)}$

Where m_t is mass flow rate of toluene

v_1 is the velocity of toluene

ρ_1 is density of toluene = 884.5 kg/m^3

$\therefore$ Area of the nozzle, A = 0.212 /(0.01 x 884.5)

= 0.024 m^2

Diameter of Nozzle = $[(0.024 \text{ x } 4) / \pi]^{1/2}$

D_n = 17.5 cm (using diameter of 45 cm)

Thickness of the nozzle

$T_h = PD_n$ / (2Fj-P)

= 1.156 x 450/[(2 x 1130 x 0.85) – 1.156]

= 0.27 + 3 mm = 3.27 mm

Nitrating acid mixture inlet nozzle Diameter:

Mass flow rate of acid mixture:

m_{am} = 10903.5 kg/hr = 3.03 kg/s

Velocity, v_1 = 0.01 m/s assuming

Density of mixture, ρ = 1520.6 Kg/m^3

$\therefore$ Area of nozzle = $m_{am} / (\rho\ v_1)$

= 3.03 / 1520.6 x 0.01

= 0.199 m^2

Diameter of nozzle = $(0.199 \times 4 / \pi)^{1/2}$

D_w = 50 cm

Thickness of the nozzle

$T_h = PD_w / (2fJ-P) + C$

=1.156 x 500 / (2 x 0.85 x 1130 – 1.156) + C

= 0.3 + 3

= 3.3 mm

Support Design

Bracket or lug support for a reactor:

Diameter of reactor = 2.4 m

Height of reactor = 7.3 m

Assuming, Clearance from vessel bottom to foundation = 1 m

Wind pressure = 128.5 kg/cm^2

Number of Brackets to be provided for reactor of diameter 2.4 m, n_b = 4 numbers.

Diameter of anchor belt circle

(from table 13.2, M.V. Joshi, 2nd edition) = 2.65 m

Height of bracket from foundation = 2.25 m (assuming)

Permissible stresses for structural steel

(IS – 800) Tension = 1400 Kg / cm^2

Compression = 1238 Kg/cm^2

Bending = 1575 Kg/cm^2

Permissible bearing pressure for concrete = 35 Kg/cm^2

Calculation of weight of reactor vessel with its content:

Length of reactor = 7.3 m

Outside diameter of reactor = 2.412 m

Inner diameter of reactor = 2.4 m

Density of structure steel = 7880 kg/m^3

Weight of reactor vessel,

$W_1 = 7880 \times \pi/4\ (2.412^2 - 2.4^2) \times 7.3$

$= 2609$ Kg.

Weight of tolune + Mixed acid,

$W_2 = \pi/4\ (D_i^2)\ l\ \rho$

$= \pi/4 \times 2.4^2 \times 7.3 \times 1462$

$= 43860$ Kg

Weight of the head

$W_3 = t \times l \times b \times \rho$

$= 40 \times 10^{-3} \times 2.4 \times 2.4 \times 7880 = 1816$ Kg

Weight of 4 nozzles

$W_4 = 100$ Kg.

$\therefore$ **Total weight** $W = W_1 + W_2 + W_3 + W_4$

$= 48385$ Kg

Design weight $= 1.3 \times 48385$

$= 62900$ Kg.

Load due to wind pressure

Wind pressure, $P_w = k\ p\ h\ Do$

where, 'k'-Coefficient depending on the shape factor = 0.7

'h' is the height of reactor vessel = 7.3 m

'p' is given wind pressure =128.5kg/cm^2

'Do' is the outside diameter of vessel = 2.412 m

$\therefore P_w = 0.7 \times 128.5 \times 7.3 \times 2.412 = 1583.8$ Kg.

Maximum total compressive load in the support is

$$P = \frac{4P_w (H - F)}{nD_b} + \frac{\sum W}{n}$$

H – Height of the vessel above the foundation

F – Vessel clearance from foundation to vessel bottom.

$\sum W$ – Maximum weight of the vessel

n = number of brackets = 4

D_b = Diameter of bolt circle = 2.65 m

$$\Rightarrow P = \frac{4 \times 1583.8 \times (7.3 - 1)}{4 \times 2.65} + \frac{48385}{4}$$

= 15862 kg

Bracket:

(a)Base plate:

Suitable base plate size, a = 140 mm

B = 150 mm

Average pressure on the plate, $P_{av} = P/(aB)$

$P_{av} = (15862)/(14x15)$

$= 75.5\ kg/cm^2$

Maximum stress in a rectangular plate subjected to a pressure P_{av} and fixed at the edges is given by

$f = 0.7\ P_{av}\ (B^2/T^2)\ [\ a^2 / (B^2 + a^2)]$

$f = 0.7 \times 75.5 \times (15^2/T^2) \times [\ 14^2 / (15^2 + 14^2)\ ]$

$= 5536/T^2$

For structural steel value of ‘f’ =1575 kg/cm^2

$\therefore T_1^2 = 5536 \times 100 / 1575$

$\therefore T_1 = 18.7$ mm

$\therefore$ Using 19 mm thick plate.

(b) Web plate.

Bending moment of each plate = $(P/2) \times (D_b – D)/2 \times 100$

$= (15862/2) \times (2.65 – 2.4)/2 \times 100$

= 99137 kg cm

Stress at the edge, f = (bending moment of each plate)/(T_2 x a x a x 0.707)

Where T_2 is web plate thickness

$\therefore f = 99137/(T_2 \times 14 \times 14 \times 0.707)$

$= 715.4/T_2$

f = 1575 kg/cm^2 for structural steel

Therefore $T_2 = 715.4/1575$

= 4.5 mm

Column support for bracket:

It is proposed to use a channel section as column.

The size chosen is ISMC 150.

Size – 150 x 75

Area of cross section – 20 .88 cm^2

Modulus of section, z – 19.4 cm^3

Radius of gyration, r – 2.21 cm

Weight – 16.4 kg/m

Height from foundation ‘l’- 2.25 m

Equivalent length for fixed ends l_e =1/2

= 2.25/2

= 1.125 m

Slenderness ratio = l_e/r

= 1.125 x 100 / 2.21 = 51

For the load acting eccentric on a short column, the maximum combined bending and direct stress is given by,

$$f = \sum W/An + \sum W_e/nz$$

where,

$\sum w$ = Load on column

A – area of cross section

e – eccentricity

z – modulus of section of cross – section

n – number of columns = 1

f = 15862/20.88 + (15862 x 7.5)/19.4

= 6890 kg/cm^2

The calculated value of ‘f’ is less than the permissible compressive stress and hence the Channel selected is correct.

Base plate for column:

Size of the column 150 x 75

It is assumed that the base plate extends 25 mm on either side of channel

Side B – 0.8 x 75 + 2 x 25 =110mm

Side C – 0.95 x 150 + 2 x 25 = 192.5 mm

Bearing pressure, P_b = (p/4) x (1/(BC))

= (15862/4) x (1/(11x19.25))

= 18.73 kg/cm^2

This is less than the permissible bearing pressure for concrete.

Stress is the plate, $f = (P_b/2) \times h^2 / (t^2/6)$

$f = (6 \times 18.73 \times 2.5^2)/(2t^2)$

but f = 1575 Kg/cm^2

$\therefore t^2 = 351.2/1575$

and 't' = 4.7 mm. It is usual to select a plate 4 to 6 mm thick.

(B) DESIGN OF CONDENSER:

1. PROCESS DESIGN OF CONDENSER:

Here the design of a Vertical Condenser is carried out.

Inlet temperature of cooling water = 24 °C

Outlet temperature of cooling water = 40 °C

Condensing Temperature = T_C = 230 °C

Mononitrotoluene to be condensed, m = 25 tons/day

= 1041.67 Kg/hr

= 0.2894 Kg/s

Here condensing temperature is assumed to be 230 °C i.e the average boiling pont of MNT, since 100% composition of MNT is obtained from the batch still as distillate.

Latent heat of vaporization of MNT = $\lambda_{MNT} = 4.6 \times 10^7$ J/Kmol

= 335.766 KJ/Kg

(Ref – 6)

(A) **Preliminary Calculations:**

(a) *Heat Balance:*

$$Q_w = Q_{MNT}$$

$$m_w\, Cp_w\, (\triangle T)_w = m_{MNT}\, \lambda_{MNT}$$

$$\Rightarrow \quad m_w = \frac{0.2894 \times 335.766 \times 10^3}{4.187 \times 10^3 \times (40 - 24)} = 1.45 \text{ Kg/s}$$

$$\Rightarrow \quad Q = (m\lambda)_{MNT} = 0.2894 \times 335.766 \times 10^3 = 95827.73 \text{ W}$$

(b) *LMTD Calculations:*

$$\text{LMTD} = (T_1 - t_2) - (T_2 - t_1) / \ln [(T_1 - t_2)/(T_2 - t_1)]$$

$$\text{LMTD} = \frac{(230 - 24) - (230 - 40)}{\ln \{ (230 - 24) / (230 - 40)\}} = 197.89 \text{ °C}$$

(B) <u>Routing of Fluids:</u>

Shell side --- Condensing vapors of Mononitrotoluene

Tube side --- Cooling water

(C) **<u>Heat Transfer Area:</u>**

Assume $(U_o)_d = 200\ W/m^2K$

$$A = \frac{Q}{U \times (LMTD)} = \frac{95827.73}{200 \times 197.89} = \underline{2.4212}\ m^2$$

Using *5/8 inch OD, BWG tubes*:

$OD = d_o = 0.015875\ m$

$ID = d_i = 0.012573\ m$

Heat Transfer area = $a = \pi d_o = \pi \times 0.015875 = 0.04987\ m^2/m$ length

Assume Length, L = 6ft = 1.8288 m

Allowing 50 mm thickness for tube sheet.

Heat transfer area for one tube = $a^1 = aL = 0.04987 \times 1.7788 = \underline{0.0887}\ m^2/tube$

$\Rightarrow$ Number of tubes, $N_t = A/a^1 = 2.4212 / 0.0887 = 28$ tubes.

From Perry Hand Book; Table 11-3, for 5/8 inch OD, BWG tubes with 13/16 inch square pitch, TEMA - P or S:

Taking number of tubes = $N_t = \underline{48}$

Shell ID = 8 inch = 203 mm

Number of passes = $N_p = 2$

Therefore,

$(A)_{corrected} = 48 \times 0.0887 = 4.2576\ m^2$

$\mathbf{(U_o)_{d,\ corrected}} = 113.7\ W/m^2K$

(D) **<u>Film Transfer Coefficient:</u>**

Average temperature of water = [24 + 40] / 2 = 32 °C

Wall temperature = T_w = [230 + (24 + 40)/2] / 2 = 131 °C

Film temperature = T_f = [131 + 230] / 2 = 180.5 °C

Shell Side:

Properties of MNT liquid at 180.5 °C

Density = ρ_s = 1000 Kg/m^3

Viscosity = μ_s = 0.0005 Pa-s

Thermal conductivity = κ_s = 0.11 W/mK

Specific heat = C_{Ps} = 266 KJ/mol-K

$$Re = \frac{4W}{\mu_s N_t \pi d_o} = \frac{4 \times 0.2894}{5 \times 10^{-4} \times 48 \times \pi \times 0.015875} = \underline{967.13}$$

$h_o = 1.47\ (\kappa^3\rho^2 g/\mu^2)^{1/3}\ (Re)^{-1/3}$

$\mathbf{h_o} = 1.47 \times [\ 0.11^3 \times (10^3)^2 \times 9.81 / (5 \times 10^{-4})\ ]^{1/3} \times [\ 967.13\ ]^{-1/3} = \underline{555.64}$ W/m^{2}K

Tube side:

Properties of water at 32 °C

Density = ρ_t = 995.026 kg/m^3

Viscosity = μ_t = 8.57 x10^{-4} Ns/m^2

Thermal Conductivity = κ_t =0.614 W/mK

Specific heat = C_{Pt} = 4.18 KJ/kg K

Total flow area: $a_t = (N_t/N_p) \times \pi d_i/4 = (48/2) \times [\pi \times (0.012573)^2 / 4] = 0.0029797$ m^2

$G_t = m_w/a_t = 1.45 / 0.0029797 = 486.63$ Kg/m^{2}s

$(Re)_t = G_t\, d_i / \mu_t = 486.63 \times 0.012573 / 0.0029797 = \underline{7114.9}$

$(Pr)_t = \mu_t CPt/\kappa_t = 8.57 \times 10^{-4} \times 4.18 \times 10^3 / 0.614 = \underline{5.83}$

$h_i d_i/\kappa_t = 0.023\ (Re)^{0.8}\ (Pr)^{0.4}$

$\Rightarrow \mathbf{h_i} = (0.614/0.012573) \times 0.023 \times (7114.9)^{0.8}\ (5.83)^{0.4} = \underline{2744.5}$ W/m^{2}K

Wall thickness, x = 0.015875 – 0.012573 = 3.3 x 10^{-3} m

κ_w = 45 w/mK (for steel)

d_w = [0.015875 + 0.012573] / 2 =0.014224 m

Dirt factor = 5.823×10^{-4} m/JsK

$$\frac{1}{(U_o)_d} = \frac{1}{h_o} + \frac{d_o}{d_i}\frac{1}{h_i} + \frac{x}{\kappa_w}\frac{d_o}{d_w} + \text{dirt factor}$$

$$= \frac{1}{555.64} + \frac{(0.015875)}{(0.012573 \times 2744.5)} + \frac{(3.3 \times 10^{-3} \times 0.015875)}{(45 \times 0.014224)} + 5.283 \times 10^{-4}$$

$(U_o)_{d,calculated}$ = <u>357.6 W/m²K</u>

This is acceptable.

(E) **<u>Pressure Drop Calculations:</u>**

<u>Shell side</u>: (Kern's method)

T_{vap} = 230 °C

Properties of MNT vapors at 230 °C

$\rho_{vap} = 4.1$ kg/m^3

$\kappa_{vap} = 0.022$ W/mK

$\mu_{vap} = 1.1 \times 10^{-5}$ Pa-s

Shell side flow area:

$a_s = (ID)_s C^1 B / P_T$

where C^1 = 13/16 inch – 5/8 inch = 3/16 inch = 4.7625×10^{-3} m

B = Baffle spacing = 0.203 m

P_T = Pitch = 13/16 inch = 20.6375×10^{-3}

$\Rightarrow$ $a_s = 0.203 \times 407625 \times 10^{-3} \times 0.203 / 20.6375 \times 10^{-3} = 9.51 \times 10^{-3}$ m^2

Equivalent diameter = $d_e = 4/(\pi d_o) \times [(P_T)^2 - \pi(d_o)^2/4]$

$= 4/(\pi \times 0.015875) \times [(20.6375 \times 10^{-3})^2 - (\pi/4) \times (0.015875)^2]$

$= 0.01828$ m

$G_s = m_s / a_s = 0.2894 / 9.51 \times 10^{-3} = 30.43$ kg/m^{2}s

$(Re)_s = G_s D_e / \mu_{vap} = 30.43 \times 0.01828 / 1.1 \times 10^{-5}$ = $\underline{50569}$

$f = 1.87 (Re)^{-0.2} = 1.87 \times (50569)^{-0.2} = 0.214$

$N_b + 1 = L/B = 1.8288/0.203 = 9$

$\triangle \mathbf{P_s} = [4 f (N_b + 1) D_s (G_s)^2 / (2 D_e \rho_{vap})] \times (0.5)$

$= [4 \times 0.214 \times 9 \times 0.203 \times 30.43^2 / (2 \times 0.01828 \times 4.1)] \times (0.5)$

$=$ $\underline{4830.55 \text{ N/m}^2}$

<u>*Tube side*</u>:

$(Re)_t = 7114.9$

$f = 0.079 (Re)^{-1/4} = 0.079 \times (7114.9)^{-0.25} = 0.0086$

$\triangle P_L = 4 f L (G_t)^2 / (2 d_i \rho_t) = 4 \times 0.0086 \times 1.8288 \times (486.63)^2 / [995.026 \times 0.012573]$

$=$ $\underline{595.4}$ N/m^2

$\triangle P_t = (2.5/2) \times [(G_t)^2/\rho_t] = (2.5/2) \times (486.63)^2 / 995.026 =$ $\underline{297.5}$ N/m^2

$\triangle \mathbf{P_{Total}} = N_p (\triangle P_L + \triangle P_t) = 2 \times (595.4 + 297.5) =$ $\underline{1785.8 \text{ N/m}^2}$

This is within the permissible pressure drop.

2. MECHANICAL DESIGN OF CONDENSER:

(a) Shell Side:

Material carbon steel (Corrosion allowance = 3mm)

Number of shells =1

Number of passes =2

Working pressure = 1 atm = 0.101 N/mm^2

Design pressure = 1.1 x 0.101 = 0.11 N/mm^2

Temperature of the inlet = 230 ^{0}C

Temperature of the outlet = 230 ^{0}C

Permissible Strength for Carbon steel = 95 N/mm^2

b) Tube side:

Number of tubes = 48

Outside diameter = 15.875 mm

Inside diameter = 12.573 mm

Length = 1.8288 m

Feed =Water

Pitch, □ ^{re} = 13/16 inch = 0.0206 m

Working Pressure =1 atm = 0.101 N/ mm^2

Design Pressure =0.11 N/mm^2

Inlet temperature = 24 ^{0}C.

Outlet temperature = 40 ^{0}C

(Ref – 7,8,10)

Shell Side:

$$t_s = \frac{PD_i}{2fJ-P}$$

t_s = Shell thickness

P = design pressure =0.11 N/ mm^2

Di = Inner diameter of shell = 0.203 m = 203 mm

f = Allowable stress value = 95 N/mm^2

J = Joint factor = 0.85

$$t_s = \frac{0.11 \text{ x } 0.203}{(2 \text{ x } 95 (0.85) - 0.11)} = 0.138 \text{ mm}$$

Minimum thickness = 6 + 3 = 9 mm (Including corrosion allowance)

$\therefore t_s$ = 10 mm

Head: (Torrispherical head)

$$t_h = \frac{PRcW}{2fJ}$$

t_h = thickness of head

$$W = ¼ \{3 + (Rc/R_k)^{1/2}\}$$

where, R_c = Crown radius = outer diameter of shell = 223mm

R_k = knuckle radius = 0.06 R_c

$$\Rightarrow W = ¼ \{3 + (R_c / 0.06 R_c)^{1/2}\} = 1.77$$

$$\Rightarrow t_h = \frac{0.11 \text{ x } 223 \text{ x } 1.77}{2 \text{ x } 95 \text{x } 0.85} = 0.27 \text{ mm}$$

Minimum shell thickness should be = 10 mm (Ref .7)

$\therefore t_h$ = 10mm

Since for the shell, there are no baffles, tie-nods & spacers are not required.

Flanges:

Loose type except lap-joint flange.

Design pressure (p) =0.11 N/mm2

Flange material : IS:2004 –1962 class 2

Bolting steel : 5% Cr-Mo steel.

Gasket material = Asbestos composition

Shell side diameter =203mm

Shell side thickness =10mm

Outside diameter of shell = 203 + 10 x 2 = 223 mm

Determination of gasket width:

$$\frac{d_o}{d_i} = \left[\frac{y - pm}{y - p(m+1)}\right]^{1/2}$$

where, y = Yield stress

m = gasket factor

Gasket material chosen is asbestos with a suitable binder for the operating conditions.

Gasket thickness = 10mm

m = 2.75

y = 2.60 x 9.81 = 25.5 N/mm^2

$$\frac{d_o}{d_i} = \left[\frac{25.5 - 0.11\,(2.75)}{25.5 - 0.11\,(2.75 + 1)}\right]^{1/2} = 1.002$$

Inside diameter of gasket

d_i = outside diameter of shell + 5 mm

= 223 + 5mm

= 228 mm

Outside diameter of the gasket

d_o = 1.002 (228) = 228.5 mm

Minimum gasket width = $\frac{228.5 - 228}{2} = 0.25$ mm

But minimum gasket width = 6mm

$\therefore$ G = 228 + 2 x 6 = 240 mm (diameter at the location of gasket load reaction)

Estimation of bolt loads:

Load due to design pressure (H) = $\frac{\pi}{4} G^2 P$

$$H = \frac{\pi}{4} (0.240)^2 (0.11) = 4.98 \times 10^{-3} \text{ MN}$$

Load to keep the joint tight under operating conditions.

$$H_p = \pi G (2b) m p$$

where, b= Gasket width = 6mm = 0.006m

$\Rightarrow$ $H_p = \pi (0.24) (2 \times 0.006) \times 2.75 \times 0.11 = 2.737 \times 10^{-3}$ MN

Total operating load (W_o) = H + H_p

$= 7.717 \times 10^{-3}$ MN

Load to seat gasket under bolt –up condition = W_g

$W_g = \pi G b y$

$= \pi \times 0.24 \times 0.006 \times 25.5$

$W_g = 0.115$ MN

$\Rightarrow$ $W_g > W_0$

$\therefore$ W_g is the controlling load

$\therefore$ Controlling load = W_g = 0.115 MN

Calculation of minimum bolting area:

Minimum bolting area (A_m) = $A_g = \frac{W_g}{S_g}$

where, S_g = Tensile strength of bolt material (MN/m^2)

Consider , 5% Cr-Mo steel, as design material for bolt

At 230 ^{0}C

$S_g = 138 \times 10^{6}$ N/m^2 = 138 MN/m^2

$$\Rightarrow \quad A_m = \frac{0.115}{138} = 8.33 \times 10^{-4}\ m^2$$

Calculation for optimum bolt size:

$g_1 = g_o/0.707 = 1.415\ g_o$

where, g_1 = thickness of the hub at the back of the flange

g_o = thickness of the hub at the small end = 10 + 2.5 =12.5mm

Selecting bolt size: M18x2

R = Radial distance from bolt circle to the connection of hub & back of flange

R = 0.027 m

C = Bolt circle diameter = ID + 2 (1.415 g_o + R)

= 203 +2 (1.415 (12.5) + 27) = 292 mm

= 0.292 m

Actual flange outside diameter (A) = C + bolt diameter + 0.02

= 0.292 + 0.018 + 0.02

Check for gasket width:

A_b = minimum bolt area = 44 x 1.54 x 10^{-4} m^2

$$\frac{A_b S_g}{\pi GN} = \frac{(44 \times 1.54 \times 10^{-4})\ 138}{\pi \times 0.24 \times 0.012} = 43.35\ N/mm^2$$

2y = 2 x 25.5 = 51 N/mm^2

$$\Rightarrow \quad \frac{A_b S_g}{\pi GN} < 2y$$

i.e., bolting condition is satisfied.

Flange Moment calculations :

(a) For operating conditions :

$W_Q = W_1 + W_2 + W_3$

$W_1 = \frac{\pi}{4} B^2 P$ = Hydrostatic end force on area inside of flange

$W_2 = H - W_1$

W_3 = gasket load = $W_Q - H = H_p$

B = outside shell diameter = 223 mm

$\Rightarrow W_1 = \frac{\pi}{4} (0.223)^2 \times 0.11 = 4.296 \times 10^{-3}$ MN

$W_2 = H - W_1 = (4.98 – 4.296) \times 10^{-3} = 0.684 \times 10^{-3}$ MN

$W_3 = H_p = 2.737 \times 10^{-3}$ MN

$\Rightarrow W_o = (4.296 + 0.684 + 2.737) \times 10^{-3}$ MN

$= 7.717 \times 10^{-3}$ MN

M_o = Total flange moment = $W_1 a_1 + W_2 a_2 + W_3 a_3$

where,

$a_1 = (C – B)/2$; $a_2 = (a_1 + a_3)/2$; $a_3 = (C - G)/$

and C = 0.292 ; B = 0.223 ; G = 0.24

$\Rightarrow a_1 = (0.292 – 0.223) / 2 = 0.0345$

$a_3 = (0.292 – 0.24) / 2 = 0.026$

$a_2 = (0.0345 + 0.026) / 2 = 0.03025$

$\Rightarrow M_o = [4.296 (0.0345) + 0.684 (0.3025) + 2.737 (0.026)] \times 10^{-3}$

$= 0.24 \times 10^{-3}$ MJ

(b) For bolting up condition:

M_g = Total bolting Moment = $W a_3$

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

where, $A_m = 8.33 \times 10^{-4}$

$A_b = 44 \times 1.54 \times 10^{-4} = 67.76 \times 10^{-4}$

$S_g = 138$ MN

$\Rightarrow W = \left(\frac{8.33 \times 10^{-4} + 67.76 \times 10^{-4}}{2}\right) \times 138 = 0.525$ MN

$M_g = 0.525 \times 0.026 = 0.01365$ MJ

$\Rightarrow \quad M_g > M_o$

$\therefore M_g$ is the moment under operating conditions

$M = M_g = 0.01365$ MJ

Calculation of the flange thickness:

$$t^2 = (MC_FY)/(BS_{FO})$$

Now,

$$K = \frac{A}{B} = \frac{\text{Flange diameter}}{\text{Inner Shell diameter}} = \frac{0.246}{0.223} = 1.103$$

K =1.103 Y = 15

Let $C_F = 1$

S_{FO} = Nominal design stresses for the flange material at design temperature.

$S_{FO} = 100$ MN

M = 0.01365 MJ

B = 0.223 m.

$t^2 = (0.01365 \times 1 \times 17) / (0.223 \times 100) = 0.0104 \text{ m}^2$

$\Rightarrow t = 0.102$ m

B_s = Bolt spacing = $\frac{\pi C}{n} = \frac{\pi(0.292)}{44} = 0.0208$ m

where, n = number of bolts = 44

C_F = Bolt pitch correction factor

$= [B_s / (2d + t)]^{1/2} = [0.0208 / (2 \times 0.018 + 0.102)]^{1/2}$

$= 0.388$

$\sqrt{C_F} = 0.623$

$\therefore$ t = 0.102 x 0.623 = 0.0635 m

Let us take t = 65 mm

Tube sheet thickness: (Cylindrical Shell)

$t_{1s} = G_c \sqrt{KP / f}$

Where, G_c = mean gasket diameter for cover.

P = design pressure.

K = factor = 0.25 (when cover is bolted with full faced gasket)

F = permissible stress at design temperature.

$\Rightarrow$ t_{1s} = 0.24 x [(0.25 x 0.11) / (95)]$^{1/2}$ = 4.083 x 10^{-3} m = 4.083 mm

Considering corrosion allowance of 3 mm,

t_{1s} = 8 mm

Channel and channel Cover:

$t_h = G_c\ (KP/f)^{1/2}$ (K = 0.3 for ring type gasket)

= 0.24 x (0.3 x 0.11/ 95)$^{1/2}$

= 4.47 x 10^{-3} m = 4.5 mm

Consider corrosion allowance = 3 mm.

t_h = 8 mm.

Saddle support:

Material: Low carbon steel

Total length of shell = 1.8288 m

Diameter of shell = 0.203 m

Knuckle radius = r_o = 0.06 x 0.203 = 12.18 mm

Total depth of head (H) = $(D_o r_o/2)^{1/2}$

= (0.203 x 0.01218 /2)$^{1/2}$

= 0.0356 m

Longitudinal Bending Moment:

Weight of the shell and its contents = W = 4753 kg

R = D/2 = 101.5 mm

Distance of saddle center line from shell end = A = 0.5R = 0.05075 m.

$Q = (W/2)(L+4H/3)$

$= 4753\ (1.8288 + 4 \times 0.0356/3)/2$

$= 4459$ kg m

$M_1 = QA[1-(1-A/L+(R^2-H^2)/(2AL))/(1+4H/(3L))]$

$$M_1 = 4459 \times 0.05075 \left\{ 1 - \frac{(1-0.05075/1.8288+(0.1015^2-0.0356^2)/(2\text{x}1.8288\text{x}0.05075))}{(1+4\text{x}0.0356/(3\text{x}1.8288))} \right\}$$

$= 1.11$ kg-m

Bending moment at center of the span

$M_2 = QL/4[(1+2(R^2-H^2)/L)/(1+4H/(3L))-4A/L]$

$M_2 = 1846.7$ kg-m

Stresses in shell at the saddle

(a) At the topmost fibre of the cross section

$f_1 = M_1/(k_1\pi R^2 t)$ $\qquad k_1=k_2=1$

$=1.11/(\pi \times 0.1015^2 \times 0.01)$

$= 3430$ kg/m^2

The stresses are well within the permissible values.

Stress in the shell at mid point

$f_2 = M_2/(k_2\pi R^2 t)$

$= 570.6$ kg/cm^2

Axial stress in the shell due to internal pressure

$f_p = PD/4t$

$= 0.11 \times 0.203 /(4 \times 0.01)$

$= 5.69$ kg/cm^2

$f_2 + f_p = 576.29$ kg/cm^2

The sum f_2 and f_p is well within the permissible value.