DESIGN OF EQUIPMENTS

ABSORBER

PROCESS DESIGN OF ABSORBER:

BASIS:

1 HOUR OF OPERATION

COMPOSITION OF THE INCOMING GAS:

COMPONENT	AMOUNT IN KMOLES	MOLE FRACTION
N_2	3298.32	0.8301
SO_3	403.732	0.1016
SO_2	13.777	0.00346
O_2	257.395	0.06478
TOTAL	3973.23	1.0

Avg Mol Wt = [28 x 3298.32 + 32 x 257.395 + 64 x 13.778 + 80 x 403.7] / 3973.23
= 33.6 Kg/Kmoles

Inlet temperature of the gas =110 $^{\circ}$C

Density of the gas = [33.6 x 273] / { 22.4 x 383 }
= 1.0711 Kg/m^3

Sulfur Trioxide is absorbed in 98% sulphuric acid and the gases after absorption are returned back to the converter. The exit concentration of the acid is assumed to be 103% (3% free oleum)

Assuming negligible absorption of the other gases and at the average temperature of the gas inside the tower at 95 $^{\circ}$C,

Moles of SO_3 at the exit = 158.014

SO_3 to be absorbed = 403.732 - 158.014
= 245.718 Kmoles
= 19657.5 Kgs

Water required to absorb SO_3 = 245.72 x 18
= 4422.96 Kgs

Water present in incoming gas = 110.98 Kmoles
Sulfuric acid formed = 110.98 Kmoles

= 10876.04 Kgs

Free SO_3 with it = 326.3

Total SO_3 absorbed by water = 110.98 x 80 + 326.3

= 9204.7 Kgs

SO_3 to be absorbed in 98% acid = 19657.5 – 9204.7

= 10452.8 Kgs

Let 'W' be the weight of 98% acid used in the tower,

Then,

SO_3 absorbed by it = [W x 0.02 x 80] / 18

Total weight of 100% acid = W + { [W x 0.02 x 80] / 18 }

= 1.0889 W

Free SO_3 associated with it = 0.03 x 1.0889 W

= 0.03267 W

Total Weight = (1.0889 + 0.03267) W

= 1.12157 W

Kgs of SO_3 absorbed by W Kgs = 1.12157 W – W

= 0.12157 W

Now,

For 0.12157 W $\longrightarrow$ =10452.8 Kgs

then, For W $\longrightarrow$ =85982 Kgs acid / Hr

= 23.88 Kgs/s

Thus, Liquid flow rate is given as,

L = 23.88 Kgs/s

Density (ρ_L) = 1850 Kg/m^3

Gas Flow Rate = [3973.23 x 33.6] / 3600

= 37.08 Kgs/s

G = 37.08 Kgs/s

Density (ρ_G) = 1.0711 Kg/m^3

DIAMETER CALCULATION:

Adopting the methodology as given in **RICHARDSON AND COULSON,VOLUME 6**,

First we calculate,

$= [L/G] \times \{ (\rho_G / \rho_L) \}^{0.5}$

= 0.0155

In the Literature given by **RICHARDSON & COULSON, Pg 544**

From the Plot of K_4 Vs [L/G] x { (ρ_G / ρ_L) }$^{0.5}$

K_4 at flooding line = 6.1

Lets choose the following packing, as given in **RICHARDSON & COULSON, Pg 533**

Material	= 3" Ceramic, Raschig Rings
Nominal Size	= 76 mm
Bulk Density	= 561 Kg/m^3
Surface Area	= 69 m^2/m^3
Packing Factor	= 65 m^{-1}
Voidage	= 75%

Then,

$$G^* = [\ \{K_4 \times \rho_G (\rho_L - \rho_G)\} / \{13.1 \times F_p \times (\rho_L / \mu_L)^{-0.1}\}]^{0.5}$$
$$= [\ \{6.1 \times 1.0711 (1850 - 1.0711)\} / \{13.1 \times 65 \times (6 \times 10^{-3} / 1850)^{0.1}\}]^{0.5}$$
$$= 7.08 \ Kg/m^2\text{-s}$$

Designing for a Pressure Drop of 42 mm water per m of packing, we have

K_4 = 1.9

Then,

% Loading = $\{1.9 / 6.1\}^{0.5}$ x 100%
= 56 %

And,

G^* = 3.95 Kg/m^2-s

Then, Cross Section Area Required,

A = [Mass Flow Rate] / G^*
= 37.08 / 3.95
= 9.38 m^2

Thus,

D_i = $[\{4 \times 9.38\}/\pi]^{0.5}$
= 3.45 m

Hence the Diameter which is calculated from this approach is 3.45 m

HEIGHT OF PACKING CALCULATION:

L	= 23.88 Kgs/s
Density (ρ_L)	= 1850 Kg/m^3
G_V	= 37.08 Kgs/s
Density (ρ_G)	= 1.0711 Kgs/m^3

Volumetric Flow rate of the entering gas is given by,

G_v = [37.08 / 1.0711]
= 34.62 m^3/s

Gas Velocity at the bottom of the tower is given by,

V_{bg} = 34.62 / 9.38
= 3.69 m/s

Mass Flow Rate at the top of the tower is given by,

G_T = [3727.51 x 33.6] / 3600
= 34.79 Kgs/s

Volumetric Flow rate at the top of the tower is given by,

G_t = [34.79 / 1.0711]
= 32.48 m^3/s

Gas Velocity at the top of the tower is given by,

V_{bg} = 32.48 / 9.38
= 3.46 m/s

Then Average Gas Velocity is given as,

V_{avg} = 3.57 m/s

And, Average Gas Velocity in the Packing,

V_P = 3.57 / 0.75
= 4.76 m/s

Liquid Flow = 23.88 / 9.38
= 2.54 Kgs/m^2-s

Given that,
Surface Area of Packing = 69 m^2/m^3
Liquid Density = 1850 Kg/m^3

Then,

Wetting Rate = 2.54 / [1850 x 69]
= 1.9898×10^{-5} m^3/m-s

The Above wetting rate is greater than the required minimum limit and this is adequate for wetting the packing.

The Methodology adopted for the calculation of Height of the Packing is referred from the literature by **NORMAN W.S (ABSORPTION, DISTILLATION AND COOLING TOWERS), Pg 214.**

The Average Properties of the gas at the temperature are given as follows,

Density of the gas mixture (ρ_G) = 1.0711 Kg/m^3
Viscosity of the gas mixture (μ_{mix}) = 2.772×10^{-5}

Diffusivity of the gas (D) $= 8.2 \times 10^{-6}\ m^2/s$

Schimidt Number (N_{Sc})
$$= (\mu_{mix}) / [(\rho_G) \times (D)]$$
$$= 2.772 \times 10^{-5} / [1.0711 \times 8.2 \times 10^{-6}]$$
$$= 3.15$$

As given in the literature,

The Reynolds number is calculated for the Standard Wetted Wall Column having the diameter,

$$d = 0.083 \text{ ft}$$
$$= 0.0253 \text{ m}$$

Reynolds Number (N_{Re})
$$= [\rho_G \times d \times V_P] / (\mu_{mix})$$
$$= [1.0711 \times 4.76 \times 0.0253] / (2.772 \times 10^{-5})$$
$$= 4654$$

Cited in the Reference **NORMAN W.S (ABSORPTION, DISTILLATION AND COOLING TOWERS), Pg 212,** the co-relation is,

$$k_G \times (RT/ V_P) \times (P/p_{BM}) \times \{(\mu_{mix}) / [\rho_G \times D]\}^{0.5} = 0.04 \times \{[\rho_G \times d \times V_P]/(\mu_{mix})\}^{-0.25}$$

Now,

With $(P/p_{BM}) = 1$(approx), we have,

$$k_G = [\ 0.04 \times (4654)^{-0.25} \times (3.15)^{-0.5} \times 15.61 \times 3600\ ] / \{1.318 \times 368\}$$
$$= 0.316 \text{ lb mole / hr-ft}^2\text{-atm}$$

Also given in the table of **NORMAN W.S (ABSORPTION, DISTILLATION AND COOLING TOWERS), Pg 210 & Pg 211**

For the conditions specified above the partial pressure of SO_3 in equilibrium with the acid is extremely small and it may be assumed that the absorption is controlled by gas film.

Partial Pressure of SO_3 in the gas at inlet, $p_1 = 0.1016$

Partial Presure at the Outlet, $p_2 = [0.1016 \times 0.0423]/[0.8984+0.1016 \times 0.0423]$
$$= 4.76 \times 10^{-3}$$

Mean Driving Force
$$= [\Delta p_1 - \Delta p_1] / \ln [\Delta p_1 / \Delta p_1]$$
$$= 0.0316 \text{ atm}$$

SO_3 absorbed
$$= 19657.5 / (80 \times 0.454) \text{ lbmoles}$$
$$= 541.71 \text{ lbmoles/ hr}$$

Area of Packing
$$= 541.71 / (0.31 \times 0.0316)$$
$$= 55300\ ft^2$$
$$= 5140\ m^2$$

Area of Packing/ft height $= 69 \times 9.38$

= 647.22

Height of Packing Required = 5140 / 647.22
= 7.94 m

Therefore the height of the packing required is 8m

MECHANICAL DESIGN OF ABSORBER

Inner Diameter of vessel, D_i = 3.45 m
Height of the packing required = 8m
Skirt height = 2m
Density of material column = 7700 Kg/m^3
Wind pressure = 130 Kg/m^2

MATERIAL :

Carbon Steel
Permissible tensile stress (f)= 950kg/cm^2

THICKNESS OF SHELL:

Thickness of shell, t_s = [p D / (2f J – p)] + c
Where,
Inner Diameter of vessel, D_i = 3.45 m
Working Pressure = 1.013 $x10^5$ N/m^2
Design Pressure, p = 1.05 x 1.013 $x10^5$ N/m^2
= 0.10635 N/mm^2
Permissible Stress = 95 N/mm^2
Joint Efficiency(J) = 0.85
Corrosion allowance = 3mm
Hence,
t_s =2.25 mm

We take thickness as 8mm

So outer diameter of shell Do = 3.45 m + 2 x 0.008m = 3.466 m

STRESS ANALYSIS AND SHELL THICKNESS AT DIFFERENT HEIGHTS:

Let X be the distance in “m” from the top of the shell, then

1. AXIAL STRESS DUE TO PRESSURE

Axial stress due to pressure, f_{ap} = p D_i / 4 (ts – c)
= 184 Kg/cm^2

2. STRESS DUE TO DEAD LOAD
a) Compressive Stress due to weight of shell up to a distance X

Outer Diameter Of shell = D_i + 2 t_s

$= 3.466$ m

Density of Shell material, ρ_s $= 7700$ Kg /m^3

f_{ds} $= \pi/4 \, (D_o^2 - D_i^2)\rho_s X] / \pi /4 \, (D_o^2 - D_i^2)$

$= 0.77$ X Kg/cm^2

b) Compressive stress due to weight of insulation at height X

Material for Insulation = Asbestos

Thickness of insulation, t_{ins} = 100 mm

Density of insulation = 575 Kg/m^3

Let,

D_{ins} = Diameter of insulation,

D_m = Mean diameter of vessel

And,

For large diameter column,

$D_{ins} = D_m$

$$f_{dins} = [\pi D_{ins} t_{ins} \rho_{ins} X] / \{\pi D_m (t_s - c)\}$$
$$= 1.15 \text{ X Kg /cm}^2$$

c) Compressive stress due to liquid in column up to height X

Density of liquid, ρ_l = 1850 Kg/m^3

$$f_{dliq} = [(\pi /4) D_i^2 X \rho_l]/ \pi D_m (t_s - c)$$
$$= 3.19 \times 10^6 \text{ N/m}^2$$
$$= 31.9 \text{ Kg/cm}^2$$

d) Compressive stress due to attachment

We have the following attachments in the absorber column

Piping weight

Head weight

Ladder

Head weight (approximately) = 2500 Kgs

Weight of Ladder = 160 X Kgs

Total compressive stress due to attachments f_d is given by,

$$f_{d(attachments)} = (\text{Piping Weight} + \text{Head Weight} + \text{Ladder})/[\pi D_i (t_s - c)]$$
$$= (2500 + 160X) / (\pi \times 0.5 \times 345)$$
$$= 4.613 + 0.295 \text{ X Kg/cm}^2$$

e) Stress due to Wind

f_w = [1.4 x 130 x X^2 x 1] / [π x 3450 x 5]
= 0.3358 X^2 Kg/cm^2

To determine the value of X

Permissible Stress = 95 N/mm^2
And,
f_{tmax} = $f_{wx} + f_{ap} - f_{dx}$

Or, 0.3358 X^2 – (36.513 + 2.215X) + 184 – 950 x 0.85 = 0

Or, 0.3358 X^2 – 2.215 X - 660.01 = 0

Solving the above equation,
We get,
X = 47.752 m

SUPPORT FOR ABSORBER

Skirt support is used to support the absorber column.

Material to be used = Structural steel (IS 800)
Inner Diameter of the vessel, D_i = 3.45 m
Outer Diameter of the vessel, D_o = 3.466 m
Height of the Packing, = 8 m
Density of carbon steel, ρ_s = 7700 kg /m^3

Total weight = Weight of vessel + Weight of Attachments

= (π/4) ($D_o^2 - D_i^2$) x H x ρ_s x 9.81 +
(π /4) D_i^2 x H x ρ_l x 0.6 +
(π /4) D_i^2 x H x ρ_p + 35000N + 1600 x H
= 1.422 x10 7 N

Diameter of Skirt = 3.45 m
Considering the height of Skirt is 8m
Wind Pressure is 1285 N/m^2
Stress due to Dead Weight
Thickness of the skirt support is t_{sk}
Stress due to dead load
f_d = Total Weight /π D_s t_{sk}
= 9.302 x 10 5 N/m2

Due to wind load
The forces due to wind load acting on the lower and upper parts of the vessels are determined as

$p_{lw} = k\ p_1\ h_1\ D_o$
$p_{uw} = k\ p_2\ h_2\ D_o$
Where K is coefficient depending on the shape factor.
k=0.7 for cylindrical surface
p_1 is wind pressure for the lower part of the vessel.
p_2 is wind pressure for the upper part of the vessel
p_1 = 700 N /m2
p_2 = 2000 N /m2
h_1=20m
h_2 =14 m
$p_{lw} = k\ p_1\ h_1\ D_o$
= 47686.8
$p_{uw} = k\ p_2\ h_2\ D_o$
= 149872.8
Bending moment due to wind at the base of the vessel is determined by
$M_w = p_{lw}\ (h_1/2) + p_{uw}\ (h_1 + h_2/2)$
=4.09 x 10 6 Nm
$f_{wb} = 4 \times M_w / \pi D_o\ t_{sk}$
=10.71 x10 7 /t_{sk}

Stress due to Seismic Load
Load F= CW
W is total Weight of vessel
C is Seismic Coefficient
C=0.08
$f_{sb} = (2/3)[\ CWH/\pi\ R_{ok}^2\ t_{sk}]$
Where,
R_{ok} is radius of skirt
= 4.159 x 10 6 /t_{sk} N/m2

Maximum Stress at bottom of Skirt
$f_{tmax} = (f_{wb}\ \text{or}\ f_{sb}) - f_{db}$
= (3.229 x 10 6 / t_{sk}) N / m^2

Permissible tensile Stress for structural steel = 140 N/mm^2
t_{sk} = 0.023m
Hence thickness of skirt is 23 mm

Maximum Compressive Stress
$f_{cmax} = (f_{wb}\ \text{or}\ f_{sb}) + f_{db}$
= (5.089 x 10 6 / t_{sk}) N /m^2
Yield point = 200 N / mm^2
f_c permissible <or = 1/3) Yield point
= 66.6 N/mm^2
t_{sk} =76mm

Maximum Compressive Stress between bearing plate and foundation

f_c= Total Weight /A + M_w /2

D_{sko} = 5 .018m

D_{ski} = 4.866m

A = (π/4) ($D_{sko}^2 - D_{ski}$ 2)

f_c = 6.911 x 10 6 N / m^2

$F = (3 \times f_c \times L^2) / t_b^2$

Permissible stress F in bending is 157.5 N/mm^2

t_b = 118 mm

Anchor Bolt,

W_{min}=7.86 x 10 5 N (assumed)

F_c = (W_{min}/ A) – (M_w /Z)

= - 9.9 x 10 5 N /m2

F_c is '- ve' , vessel skirt must be anchored to the concrete foundation by anchor bolt

Number of bolt = 4.866 x 10 3 / 600

= 32

P_{bolt} = (f_c)$_{min}$ x A) / N

= 3.11 x 10 5 N

HEAT EXCHANGER

PROCESS DESIGN OF HEAT EXCHANGER:

BASIS:

1 HOUR OF OPERATION

GIVEN:

THE FLUIDS ARE:

WATER:

INLET TEMPERATURE = 25 $^{\circ}$C
OUTLET TEMPERATURE = 40 $^{\circ}$C

SULFURIC ACID:

INLET TEMPERATURE = 112 $^{\circ}$C
OUTLET TEMPERATURE = 30 $^{\circ}$C

The Sulfuric acid coming out from the absorption towers are cooled from a high temperature to a lower temperature in a Shell and Tube Type Heat Exchanger. Water which enters the Heat Exchanger at room temperature is heated to 40 $^{\circ}$C and comes out of the system.

BULK TEMPERATURE OF THE ACID = (30 + 112)/2
= 71.0 $^{\circ}$C

PROPERTIES OF **WATER** AT BULK TEMPERATURE OBTAINED FROM THE LITERATURE ARE AS FOLLOWS:

PROPERTIES	NUMERICAL VALUE
1. BULK TEMPERATURE OF WATER	32.5 $^{\circ}$C
2 DENSITY, ρ_w	994.86 Kg/m^3
3 SPECIFIC HEAT CAPACITY, C_{Pw}	4.184 KJ/Kg-K
4. THERMAL CONDUCTIVITY, K_w	0.623 W/m-K
5. VISCOSITY, μ_w	0.8 Centipoise

PROPERTIES OF **SULFURIC ACID** AT BULK TEMPERATURE OBTAINED FROM THE LITERATURE ARE AS FOLLOWS:

PROPERTIES	NUMERICAL VALUE
1. BULK TEMPERATURE OF WATER	71 °C
2 DENSITY, ρ_h	1850 Kg/m^3
3 SPECIFIC HEAT CAPACITY, C_{Ph}	1.4435 KJ/Kg-K
4. THERMAL CONDUCTIVITY, K_h	0.655 W/m-K
5. VISCOSITY, μ_h	6.83 Centipoise

1. HEAT LOAD:

HEAT INPUT = HEAT OUTPUT

m_h = 48572 Kg/Hr
= 13.49 Kg/s

$Q = m_h \times C_{Ph} \times [\Delta T]_h$
$= 13.49 \times 1.4435 \times 10^3 \times (112 - 30)$
$= 1.596 \times 10^6$ J/s

$Q = 1596.7 \times 10^3$ J/s

Overall Heat Balance gives,
$m_w \times 4187 \times 15 = 1596.7 \times 10^3$
$m_w = 25.42$ Kg/s

2. LMTD:

	ACID		WATER		ΔT
TEMPERATURES	112.0	↓	40	↑	72 °C
TEMPERATURES	30.0		25		5 °C

LMTD = [(112-40)-(30-25)] / ln [(112-40)-(30-25)] = 25.11 °C

R = [112-30] / {40-25} = 5.46
S = [40-25] / {112-25} = 0.172
From the graph of F_T against S at various R, we have
$F_T = 0.834$

Then, LMTD = 20.94 °C

3. ROUTING:

Shell Side = Sulfuric Acid
Tube Side = Cooling Water

4. DETERMINATION OF AREA:

Assume U_o = 630 W/m^2-K
Then, Area can be calculated as,

A = 1596.7×10^3 / [630 x 20.94]
= 121.03 m^2

5. CHOICE OF TUBES:

From the tubing characteristics as given in **PERRY**,
We choose the following dimensions of the tube,

1 inch Outer Diameter tubes with 1.25 inch Triangular Pitch,16BWG

D_o = 1.0 inch
= 25.4 mm
D_i = 0.87 inch
= 22.1 mm
P' = 31.75 mm

Let us assume the tube to be of length of 6m.
Number of tubes = 121.03 / (π x 0.0254 x 6)
= 253

6. CORRECTION OF HEAT TRANSFER AREA:

From the tube count table,
We have For TEMA P or S (1- 4 Exchanger)
1 Shell Pass and 4 Tube Passes
Diameter of Shell, D_s = 635 mm
Number of Tubes, N_t = 250

Corrected HT Area = (π x 0.0254 x 6) x 250
= 119.69 m^2

Corrected U_{oc} = 637.07 W/m^2-K

7. CALCULATION OF INSIDE HEAT TRANSFER COEFFICIENT

Area of the tubes, a_t = $[\pi \times d_i^2 \times N_t] / [4 \times N_P]$

$= 0.024\ m^2$

Mass velocity G_{st} $= 25.42 / 0.024$

$= 1059.16\ Kg/m^2\text{-}s$

Velocity inside the tubes, V_t $= m / [\ \rho \times a_t\ ]$

$= 1059.16 / 994.865$

$= 1.06\ m/s$

The above velocity is also within acceptable limits of 1 to 3 m/s.

Reynolds Number, N_{Re} $= [G_{st} \times d_i\ ]/\ \mu$

$= [1059.16 \times 22.1 \times 10^{-3}\ ] / 0.8 \times 10^{-3}$

$= 29260$

Prandtl Number, N_{Pr} $= 5.37$

$(h_i\ d_i / k) = j_H \times (N_{Re}) \times (N_{Pr})^{(1/3)}$

where,

$j_H = 0.0036$

Then,

$(h_i\ d_i / k) = 184.45$

$h_i = 5199\ W/m^2\ K$

8. CALCULATION OF OUTSIDE HEAT TRANSFER COEFFICIENT

Length of tube $L = 6\ m$

Baffle Spacing, $L_s = 0.266 \times Ds$

$= 168.9\ mm$

Number of baffles, $N_b + 1 = 6 / L_s$

$N_b = 35$

$S_m = [\ L_s\ (P' - D_o)\ D_s\ ] / P'$

$= [(0.03175 - 0.0254) \times 0.1689 \times 0.635 / 0.03175$

$= 0.02145\ m^2$

$v_s = m_h / (S_m \times \rho_h)$

$= \{48572/3600\} / (0.02145 \times 1850)$

$= 0.340\ m/s$

The above value of velocity is also in the range of 0.3 to 1m/s, so this is also acceptable.

Equivalent Diameter, d_e $= 1.1 \{31.75^2 - 0.917 \times 25.4^2 \}/ 25.4$
$= 18.04$ mm

$N_{Re} = [d_e\, G]/\mu$
$= [629 \times 18.04 \times 10^{-3}] / 6.83 \times 10^{-3}$
$= 1661$

$N_{Pr} = 42.9$

From the graph, we have
$j_H = 0.019$

$(h_o\, d_e / k) = j_H\, (N_{Re})\, (N_{Pr})^{(1/3)}\, (\mu / \mu_w)^{0.14}$
$= 103.65$

$h_o = 3763$ W/m^2 K

$[1 / U_o] = [1 / h_o] + [D_o / D_i] [1 / h_i] + [D_o \times \ln \{D_o/D_i\} / (2K_w)]$
$+ [1 / h_{od}] + [D_o / D_i] [1 / h_{id}]$

Taking,

$[1 / h_{od}] = 1 / 3000$ (m^2-K)/W
$[1 / h_{id}] = 1 / 5000$ (m^2-K)/W

$[1 / U_o] = 1.083 \times 10^{-3}$ (m^2-K)/W

$U_o = 923$ W/(m^2-K)

Note: As this value of U_o is greater than the corrected value of U_{oc}, so the design with the above specifications is accepted.

9. PRESSURE DROP CALCULATION:

For Tube side,

$f = 0.079 \times (N_{Re})^{-0.25}$
$= 0.00604$

$\Delta P_L = [(4fLV_t^2) \times \rho_f] / \{2 \times D_i\}$
$= 3666$ N/m^2

$\Delta P_t = 2.5 \times [\rho_f \times V_t^2 / 2]$
$= 1397$ N/m^2

ΔP_T $= N_p (\Delta P_L + \Delta P_t)$
$= 20252 \text{ N/m}^2$
$= 20.25$ KPa

Note: As the value of the pressure drop is less than 70KPa, the design is acceptable from the tube side pressure drop consideration.

For Shell Side,

Pressure Drop in the Cross Flow section is calculated by,

$$\Delta P_c = [\{b \text{ x } f_K \text{ x } W^2 \text{ x } N_c\} / (\rho_f \text{ x } S_m^{\ 2})] \text{ x } \{\mu_w / \mu_b\}^{0.14} \qquad \text{KN/m}^2$$

N_{Re} = 1668
b $= 2 \text{ x } 10^{-3}$
f_K = 0.25
$\rho_h (\rho_f)$ $= 1850 \text{ Kg/m}^3$
$m_h(W)$ = 13.49 Kg/s

N_c $= [D_s (1 - 2 \{L_c/D_s\}) / P_p]$
= 635 x (1-2 x 0.25) / 22
= 14.43
= 14.5

$$\Delta P_c = [\{2 \text{ x } 10^{-3} \text{ x } 0.25 \text{ x } 13.49^2 \text{ x } 15\}/(1850 \text{ x } 0.02145^2)] \qquad \text{KN/m}^2$$
= 1.55 KPa

Pressure Drop in End Zones is calculated as,

$$\Delta P_e = \Delta P_c (1 + \{N_{cw} / N_c\}) \qquad \text{KN/m}^2$$

N_{cw} $= 0.8\, l_c / P_p$
= [0.8 x 0.25 x 635] / 22
= 6

ΔP_e = 1.55 [1 + (6 / 14.5)]
= 2.191 KPa

Pressure Drop in Window Zones

$$\Delta P_w = [b \text{ x } W^2 \text{ x } (2 + 0.6 N_{cw}) / \{S_m \text{ x } S_w \text{ x } \rho_f\}] \qquad \text{KN/m}^2$$

b $= 5 \text{ x } 10^{-4}$
S_w $= S_{wg} - S_{wt}$

From the graph from **PERRY Fig. 10-18, Pg. 10-29**

S_{wg} = 100 inch^2
= 0.0645 m^2
S_{wt} = (N_t / 8) x (1 – F_c) x π x D_o^2

From the graph from **PERRY Fig. 10-16, Pg. 10-28**

F_c = 0.65
S_{wt} = (250 / 8) x (1 – 0.65) x π x 0.0254^2
= 0.0222 m^2

S_w = S_{wg} - S_{wt}
= 0.0645 – 0.0222
= 0.0423 m^2

ΔP_w = [5 x 10^{-4} x 13.49^2 x (2 + 0.6 x 6) / {0.02145 x 0.0423 x 1850}]
= 0.303 KPa

Therefore the total Pressure Drop on the shell side is calculated by the following relation

ΔP_s (TOTAL) = 2 x ΔP_e + (N_b – 1) x ΔP_c + N_b x ΔP_w
= 2 x 2.191 + 35 x 1.55 + 36 x 0.303
= 69 KPa

As this value of Pressure Drop on the shell side is less than the 70 KPa, the design is acceptable from the Pressure Drop Point of View.

Thus, the design is acceptable from process design consideration.

SUMMARY OF PROCESS DESIGN FOR HEAT EXCHANGER

Mass flow rate of acid	= 13.49 Kg/s
Mass flow rate of water	= 25.42 Kg/s
Shell outer diameter	= 635 mm
Number of tubes	= 250
Tube OD	= 1 inch
Pitch (Triangular)	= 1.25 inch
Tube length	= 6 m
Shell side pressure drop	= 69 KPa
Tube side pressure drop	= 20 KPa
Heat Exchanger type	= TEMA P or S type 1-4 Heat Exchanger

MECHANICAL DESIGN OF HEAT EXCHANGER

Working Pressure	= 1 atm
Inlet Temperature	= 110 $^{\circ}$C
Design pressure	= 1.1 atm
Number of tubes	= 250
Shell diameter	= 635 mm

The entire mechanical design is referred from the literature in **PROCESS EQUIPMENT DESIGN** by **M.V JOSHI.**

1. SHELL THICKNESS

Material: IS 2825-1969 Grade I plain Carbon steel.

Shell thickness , (t_S) $= [\{P \times D_i\} / (2fJ - P)]$
$= [\{635 \times 1.1\} / (2 \times 95 \times 0.85 - 1.1)]$
= 4.29 cm
= 5 mm

From the Table 9.2, its found that minimum shell thickness when severe conditions are not expected is 8mm, which includes the Corrosion Allowance.

2. NOZZLES

Take inlet and outlet nozzles as 100mm diameter.

Vent nozzle	= 25mm diameter
Drain nozzle	= 25mm diameter
Relief Valve	= 50 mm diameter.
Nozzle thickness	$= [P \times D_i] / \{2 f J - P\}$
	= 0.68 mm

Minimum nozzle thickness is 6mm and 8mm is choosen which includes the corrosion allowance.

Also only the inlet and outlet nozzles need compensation. The compensation required is minimum and is given by pads of 10mm thickness.

3. HEAD

Torispherical heads are taken for both ends.

R_c (Crown radius) = 635 mm

R_{nk} (knuckle radius) = 63.5 mm

Head thickness (t_h) = [P x R_c x W] / { 2 f J }
Where,

W = (1/4) x [3 + $(R_c / R_{nk})^{0.5}$]
= 1.54

Head thickness (t_h) = [P x R_c x W] / { 2 f J }
Head thickness = 6.66 mm

Therefore we take Head Thickness as that of the Shell Thickness = 8mm

4. TRANSVERSE BAFFLES

Number of Baffles = 35
Baffle cut = 25%
Baffle thickness = 6mm (standard)

5. TIE RODS AND SPACERS

Diameter of tie rods = 10mm
Diameter of Spacers = 8mm

6. FLANGE DESIGN

Flange is ring type with plain face.

Flange material: IS 2004-1962 Class 2 Carbon Steel
Bolting steel: 5% Chromium, Molybdenum Steel

Gasket Material: Asbestos

Shell OD = 0.635 m
Shell Thickness = 0.008 m (g)
Shell ID = 0.627 m

Allowable stress for flange material = 100 N/mm^2
Allowable stress of bolting material = 138 N/mm^2

6 (i). DETERMINATION OF GASKET WIDTH

Minimum design yield seating stress , y = 52.386 N/mm^2
Gasket factor, m = 3.75
Gasket Size:
Outer Diameter = 680 mm

Inner Diameter = 650 mm

Mean Gasket Diameter, G = 665 mm

Minimum gasket width,

Choose N = 30 mm.

Basic gasket seating width, b_o = 30/2

= 15 mm

Effective Gasket Seating Width, b $= 2.5 \times [b_o]^{0.5}$

= 9.7 mm

6 (ii). ESTIMATION OF BOLT LOADS

Under atmospheric conditions, the bolt load due to gasket reaction is given by

$W_{m1} = \pi \, b \, G \, y$

$= \pi \times 2 \times 665 \times 52.39$

= 1061 KN

Load due to design pressure

$W_{m2} = H + H_T$

$W_{m2} = \pi G^2 P/4 + \pi G(2b)mp$

= 549.23 KN

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

6 (iii). CALCULATION OF MINIMUM BOLTING AREA:

$A_m = A_o = W / S$

S = allowable stress for bolting material

$A_{m1} = A_o = 1061 \times 10^3/138 = 7688.4 \text{ mm}^2$

6 (iv). CALCULATION OF OPTIMUM BOLT SIZE.

Bolts are of 5% Cr Mo Steel

Number of bolts $= G / [b_o \times 2.5]$

= 665 / [15 x 2.5]

= 18 bolts

Diameter of bolts $= [(A_{m1} / \text{Number of bolts}) \times (4 / \pi)]^{1/2}$

= 24 mm

7. FLANGE THICKNESS

Thickness of flange , t_f $= [G\sqrt{(p/Kf)}] + C$

Where,
C is the Corrosion allowance
h_G is radial distance from gasket load reaction to bolt circle
Hydrostatic end force, H = $(\pi /4)\, G^2\, p$
= 38.69 KN
K = $1/[\,0.3 + \{(1.5\, W_m\, h_G) / (H \times G)\}]$
h_G = $(B - G)/2$

Where,
B = Outside diameter of Gasket + 2xDiameter of Bolt +12mm
= 680 + 2 x 24 + 12
= 740 mm
Then, h_G = $(B - G)/2$
= 37.5 mm

K = $1/[\,0.3 + \{(1.5\, W_{m1}\, h_G) / (H \times G)\}]$
= 0.382

t_f = $[G\sqrt{(p/Kf)}\,] + C$
= 35.8 + C
= 38 mm
Hence the thickness of flange = 38 mm

TUBE SIDE

Material:Stainless steel (IS- grade 10)

Thickness of tube = $t_f = \{P \times D_o\} / (2\, f\, J + P)$

Where,
Working pressure = 12 N/mm^2
Design pressure, P = 14 N/mm^2
Permissible Stress, f = 100.6 N/mm^2
Joint Efficiency, J = 1
Thickness of tube = 1.65 mm
Use tube with thickness of 2mm

No Corrosion allowance, since the tubes are of stainless steel.

1. TUBE SHEET

The tube sheet is held between shell flange and the channel. The joint on the shell flange side is of male and female facing and on the channel side of ring facing, since the pressure on the channel

Thickness of Tube Sheet , t_{ts} = $FG\sqrt{[(0.25\, P)/f]}$

Where,

F = 1.25

Thickness of tube sheet = 155 mm

2. CHANNEL AND CHANNEL COVER

Material :Carbon Steel

Permissible stress,f = 95 N/mm^2

For Ring type gasket

K =0.3

Thickness of channel, t_h $= G \sqrt{[(K \times P)/f]}$

=140 mm

4. GASKET SIZE

Width of ring gasket, N = 30 mm

Gasket material: Steel Jacketed Asbestos

Gasket factor, m = 5.5

Minimum design seating stress, Y_a = 126.6 N / mm^2

Basic gasket seating width, b_o = N / 8

= 30 / 8

= 3.75 mm

Effective gasket seating width,b = b_o

Mean diameter, G = 665 mm

Design pressure, P = 14 N/mm^2

Under atmospheric conditions, the bolt load due to gasket reaction is given by

$$W_{m1} = \pi\, b\, G\, Y_a$$
$$= 991.8 \text{ KN}$$

After the internal pressure is applied, the gasket which is compressed earlier, is released to some extent and the bolt load is given by

$$W_{m2} = \pi \times 2\, b \times G \times m \times P + (\pi / 4) G^2 P$$
$$= 6069 \text{ KN}$$

"f " is permissible tensile stress in bolts under atmospheric condition

Bolt Material: 5%Cr Mo Steel,

$f = 140.6\ N/mm^2$

A_m = area of bolt

$A_{m1} = W_{m1} / f_a$

$= 7075\ mm^2$

$A_{m2} = W_{m2} / f_b$

$= 43165\ mm^2$

Number of bolts = (mean diameter) / 10 x 2.5

= 26 bolts

To determine the size of bolts , the larger of above two areas should be considered

Diameter of bolts, D_b $= [(A_{m2} / \text{Number of bolts}) \times (4 / \pi)]^{1/2}$

= 46 mm

5. THICKNESS OF NOZZLE

Considering inlet and outlet diameter to be 100mm, then thickness of the nozzle is given by,

Thickness of nozzle, t_n $= (P \times D_n) / [2 \times f J – P]$

Permissible stress, f $= 95\ N/ mm^2$

Joint Efficiency, J = 0.85

Then,

Thickness of nozzles =10 mm

6. FLANGE THICKNESS:

Flange material: IS 2004-1962 Class 2 Carbon Steel

Thickness of the Flange , t_f $= [G\sqrt{(P/Kf)}] + C$

Where,

C is the Corrosion allowance

Allowable stress for flange material, $f = 100\ N/mm^2$

h_G is radial distance from gasket load reaction to bolt circle

Hydrostatic end force, H $= (\pi /4) G^2 P$

= 4863 KN

$K = 1/[0.3 + \{(1.5 W_m h_G) / (H \times G)\}]$

$h_G = (B – G)/2$

Where,

B = Outside diameter of Gasket + 2xDiameter of Bolt +12mm

= 680 + 2 x 46 + 12

= 784 mm

Then, h_G $= (B - G)/2$

$= 60$ mm

K $= 1/[\, 0.3 + \{(1.5\, W_{m2}\, h_G) / (H \times G)\}]$

$= 2.1326$

Then,

t_f $= [G\sqrt{(P/Kf)}\,] + C$

$= 170 + C$

$= 175$ mm

Hence the thickness of flange = 175 mm

SUPPORT FOR SHELL AND TUBE HEAT EXCHANGER

Length of the heat exchanger, L = 6000 mm
Inner diameter of Shell, D_i = 635 mm
Outer diameter of Shell, D_o = 643 mm
Thickness of Shell, t_s = 8 mm
Outer diameter of tube, d_o = 25.4 mm
Number of tubes, N_t = 250

Density of Steel, ρ_s = 7850 Kg /m^3
Density of Liquid in tubes, ρ_l = 1000 Kg /m^3

Volume of Shell body, V $= (\pi / 4)(D_o^2 - D_i^2) \times L$

$= 0.0482\ m^3$

Weight of Shell body, W_s $= V \times \rho_s$

$= 379$ Kgs

Volume of Tubes, V_t $= (\pi / 4)(d_o^2 - d_i^2) \times L \times N_t$

$= 0.185\ m^3$

Total Weight of Tubes, W_t $= V_t \times \rho_s$

$= 1453$ Kgs

Volume of Head, V_h $= 0.087\, D_i^3$

$= 0.022\ m^3$

Weight of Head, W_h $= V_h \times \rho_s$

$= 173$ Kgs

Weight of Liquid, W_l $= (\pi / 4)(d_i^2) \times L \times N \times \rho_l$

$= 577$ Kgs

Total Weight, W $= W_s + W_t + W_h + W_l$

$= 2582$ Kgs

= 25.82 KN

Depth of head, H = 250 mm

Q = (W/2)

= 12.91 KN-m

Now, we calculate,

Distance of saddle center line from shell end ,

$A = 0.4 \times R_i$

$= 0.4 \times (635/2)$

$= 127$ mm

1. LONGITUDINAL BENDING MOMENTS

Radius, R = 0.317 m

Depth of head, H = 0.250 m

The bending moment at the supports is

$$M_1 = QA\,[\,1 - \{(A/L) + (R^2 - H^2) / 2\,AL\}/\{1 + 4H/3L\}]$$
$$= 80.12 \text{ N-m}$$

The bending moment at the center of the span is given by

$$M_2 = (Q\,L / 4)[\{1 + 2\,(R^2 - H^2) / L^2\}/\{1 + 4H / 3L\} - (4A / L)\,]$$
$$= 16745 \text{ N-m}$$

2. STRESS IN SHELL AT THE SADDLE

For $\theta = 120^\circ$

$k_1 = 0.107$

$k_2 = 0.192$

Thickness of shell, t = 8 mm

$$f_1 = M_1/(k_1 \pi R^2 t)$$
$$= 297 \times 10^3 \text{ N/m}^2$$

$$f_2 = M_1/(k_2 \pi R^2 t)$$
$$= 165.2 \times 10^3 \text{ N/m}^2$$

3. STRESS IN THE SHELL AT MID- SPAN

The stress at the mid span is f_3, which is either tensile or compressive depending on the position of the fiber. The resultant tensile stresses (including the axial stress due to internal pressure) should not exceed the permissible stress, and the resultant compressive stress should not exceed the permissible compressive stress

$$f_3 = M_2 /(\pi R^2 t)$$
$$= 6.63 \times 10^6 \text{ N/m}^2$$

Axial Stress in Shell due to internal pressure

$$f_p = (P \times D_i)/(4\,t)$$
$$= 2.262 \times 10^6 \ N/m^2$$

All combined stresses ($f_p + f_1$) , ($f_p + f_2$) , and ($f_p + f_3$) are well within allowable limits. Hence, the given parameters can be considered for design.

Thus a shell and tube Heat Exchanger with the above specifications is designed.

COOLER

PROCESS DESIGN OF COOLER:

BASIS:

1 HOUR OF OPERATION

GIVEN:

THE FLUIDS ARE:

WATER:

INLET TEMPERATURE = 25 °C
OUTLET TEMPERATURE = 40 °C

PROCESS GAS:

INLET TEMPERATURE = 202.41 °C
OUTLET TEMPERATURE = 110 °C

The Process gas which consists of mixtures of Sulphur Dioxide, Sulfur Trioxide, Nitrogen (inert) and Oxygen are cooled from a high temperature to a lower temperature in a Shell and Tube Type Heat Exchanger. Water which enters the Heat Exchanger at room temperature is heated to 40 °C and comes out of the system.

BULK TEMPERATURE OF THE GAS MIXTURE = (202.41 + 110)/2
= 156.20 °C

PROPERTIES OF **WATER** AT BULK TEMPERATURE OBTAINED FROM THE LITERATURE ARE AS FOLLOWS:

PROPERTIES	NUMERICAL VALUE
1. BULK TEMPERATURE OF WATER	32.5 °C
2 DENSITY, ρ	994.86 Kg/m^3
3 SPECIFIC HEAT CAPACITY, C_P	4.187 KJ/Kg-K
4. THERMAL CONDUCTIVITY, K	0.623 W/m-K
5. VISCOSITY, μ	0.8 Centipoise

For the gas mixture, the only known property is its specific heat and other properties of the gas mixture are predicted using the Transport Phenomena considerations. The properties which have to be predicted are namely viscosity and thermal conductivity.

For the Gas Mixture,

SPECIFIC HEAT CAPACITY, C_{Pg} = 1.041 KJ/Kg-K

PROPERTIES OF **PROCESS GAS** ARE PREDICTED AS MENTIONED BELOW:

Based on the Literature on **TRANSPORT PHENOMENA BY BIRD,** we have the following :

CALCULATION OF VISCOSITY OF GAS MIXTURE:

The Critical constants data are given

COMPONENT	CRITICAL TEMPERATURE(K)	CRITICAL PRESSURE(ATM)
1. N_2	126.2	33.5
2. O_2	154.4	49.7
3. SO_2	430.7	77.8
4. SO_3	490.8	83.6

Bulk temperature = 154.29 + 273
= 427.29 K
Pressure = 2.13 atm

COMPONENT	T_R (K)	P_R(ATM)
1. N_2	3.380	0.06358
2. O_2	2.760	0.04285
3. SO_2	0.990	0.02730
4. SO_3	0.870	0.02550

μ_c for each component is then calculated by the equation as,

$$\mu_c = 7.7 \times (M)^{0.5} \times (P_C)^{(2/3)} \times (T_C)^{(-1/6)}$$

Where M is the Molecular weight

Tabulating the Data that are obtained from the graph as well as from the calculation,

COMPONENT	μ_r (poise)	μ_c (poise)	μ (poise)
1. N_2	1.6	1.89×10^{-6}	3.024×10^{-4}
2. O_2	1.2	254.2×10^{-6}	3.0504×10^{-4}
3. SO_2	0.45	408.5×10^{-6}	1.8380×10^{-4}
4. SO_3	0.38	468.8×10^{-6}	1.7810×10^{-4}

COMPONENT	MOLE FRACTION	MOLECULAR WEIGHT	μ (poise)

1. N_2	0.8301	28	3.024×10^{-4}
2. O_2	0.0647	32	3.0504×10^{-4}
3. SO_2	0.0034	64	1.8380×10^{-4}
4. SO_3	0.1016	80	1.7810×10^{-4}

We tabulate the calculations,

i	j	(M_i / M_j)	(μ_i / μ_j)	Φ_{ij}	$\Sigma(x_j \Phi_{ij})$
1	1	1.000	1.000	1.000	
	2	0.875	0.990	1.062	1.129
	3	0.437	1.640	1.954	
	4	0.350	1.690	2.202	
2	1	1.142	1.008	0.938	
	2	1.000	1.000	1.000	1.0625
	3	0.500	1.659	1.850	
	4	0.400	1.710	2.089	
3	1	2.285	0.607	0.521	
	2	2.000	0.602	0.557	0.5871
	3	1.000	1.000	1.000	
	4	0.800	1.032	1.134	
4	1	2.857	0.588	0.455	
	2	2.500	0.583	0.488	0.5142
	3	1.250	0.968	0.878	
	4	1.000	1.000	1.000	

In the above tabular column, the value of Φ_{ij} is calculated by the equation,

$$\Phi_{ij} = (1/\sqrt{8}) \times [\, 1 + (M_i / M_j)^{-0.5} \,] \times [\, 1 + (\mu_i / \mu_j)^{0.5} \times (M_j / M_i)^{0.25} \,]^{2}$$

$$\begin{aligned}
\mu_{mix} &= \Sigma \; [\, (x_i\, \mu_i) / \Sigma\, (x_j\, \Phi_{ij}) \,] \\
&= [\{0.8301 \times 3.024 \times 10^{-4}\}/1.129] + [\{0.06479 \times 3.0504 \times 10^{-4}\}/1.0625] \\
&\quad + [\{0.003467 \times 1.838 \times 10^{-4}\}/0.5871] + [\{.1016 \times 1.781 \times 10^{-4}\}/0.5142] \\
&= 2.772 \times 10^{-4} \text{ g-cm}^{-1}\text{-s}^{-1} \\
&= 2.772 \times 10^{-5} \text{ Kg/ms}
\end{aligned}$$

CALCULATION OF THERMAL CONDUCTIVITY:

COMPONENT	P_r	T_r	K_c	K_r	K
1. N_2	0.0635	3.380	1.024×10^{-4}	1	1.024×10^{-4}
2. O_2	0.4285	2.760	9.459×10^{-5}	0.78	7.370×10^{-5}
3. SO_2	0.0273	0.990	3.893×10^{-5}	0.33	1.284×10^{-5}
4. SO_3	0.0255	0.870	4.025×10^{-5}	0.28	1.127×10^{-5}

K_c is calculated from the relation given by,

$$K_c = [\, C_P + (5R/4M)\,] \times \mu$$

Where value of R is 1.987

$$
\begin{aligned}
K_{mix} &= \Sigma\ [\, (x_i K_i) / \Sigma (x_j \Phi_{ij})\,] \\
&= [\{0.8301 \times 1.024 \times 10^{-4}\}/1.129] + [\{0.06479 \times 7.370 \times 10^{-5}/1.0625] \\
&\quad + [\{0.003467 \times 1.285 \times 10^{-5}\}/0.5871] + [\{.1016 \times 1.127 \times 10^{-5}\}/0.5142] \\
&= 8.21 \times 10^{-5} \text{ Cal / s-cm-K} \\
&= 344.8 \times 10^{-4} \text{ J/s-m-K}
\end{aligned}
$$

1. HEAT LOAD:

With,

$$
\begin{aligned}
m_g &= 131518.5 \text{ Kg/Hr} \\
&= 36.5 \text{ Kg/s}
\end{aligned}
$$

$$
\begin{aligned}
Q &= m_g \times C_{Pg} \times [\Delta T]_{gas} \\
&= 131518 \times 1.041 \times 10^3 \times (202.41 - 110) \\
&= 12.65 \times 10^9 \text{ J/Hr} \\
&= 3.514 \times 10^6 \text{ J/s}
\end{aligned}
$$

As the values of Mass Flow Rate (m_g) and Heat Load (Q) are on higher side, we split the entire flow rate into 4 equal parts so that we have 4 equal area heat exchangers operating in parallel and which are handling equal heat load.

So, we have

$m_g = 9.13$ Kg/s

$Q = 878.5 \times 10^3$ J/s

Overall Heat Balance gives,

$m_w \times 4187 \times 15 = 878.5 \times 10^3$

$m_w = 13.98$ Kg/s

2. LMTD:

	GAS MIXTURE ↓	WATER ↑	ΔT
TEMPERATURES	202.41	40	162.4 °C

TEMPERATURES 110.0 25 85 °C

LMTD =[(202.41-40)-(110-25)] / ln[(202.41-40)/(110-25)] = 119.5 °C

R = [202.41-110] / {40-25} = 6.16
S = [40-25] / {202.41-25} = 0.0846
From the graph, we have
$F_T = 0.98$

Then, LMTD = 116.6 °C

3. ROUTING:

Shell Side = Process Gas
Tube Side = Cooling Water

4. DETERMINATION OF AREA:

Assume $U_o = 200$ W/m^2-K
Then, Area can be calculated as,
A = 878.5 x 10^3 / [116.6 x 200]
=37.67 m^2

5. CHOICE OF TUBES:

From the tubing characteristics as given in **PERRY**,
We choose the following dimensions of the tube,

0.75 inch Outer Diameter tubes with 1 inch Triangular Pitch

D_o = 0.75 inch
= 0.01905m
D_i = 0.62 inch
= 0.01575m
P' = 0.0254m

Let us assume the tube to be of length of 2m.
Number of tubes = 37.67 / (π x 0.01905 x 2)
= 314.728

6. CORRECTION OF HEAT TRANSFER AREA:

From the tube count table,
We have For TEMA L or M (1-6 Exchanger)
1 Shell Pass and 6 Tube Passes
Diameter of Shell, D_s = 540mm

Number of Tubes, N_t = 320

External Area = $0.0598\ m^2/m$ length
Corrected HT Area = $0.0598 \times (2 – 0.05) \times 320$
= $37.31\ m^2$

Corrected U_{oc} = $878.5 \times 10^3 / (37.31 \times 116.6)$
= $201.93\ W/m^2\text{-}K$

7. CALCULATION OF INSIDE HEAT TRANSFER COEFFICIENT

Flow area available per pass, a_t = $[\ \pi \times d_i^2 \times N_t\ ] / [\ 4 \times N_P\ ]$
= $0.01039\ m^2$

Velocity inside the tubes, V_t = $m / [\ \rho \times a_t\ ]$
= $13.98 / [\ 994.86 \times 0.01039\ ]$
= $1.3524\ m/s$

Reynolds Number, N_{Re} = $[\ 994.86 \times 0.01575 \times 1.3524\ ] / 0.8 \times 10^{-3}$
= 26488

Prandtl Number, N_{Pr} = 5.37

$(h_i\ d_i / k)$ = $0.023\ (N_{Re})^{0.8}\ (N_{Pr})^{(1/3)}$
= 139
h_i = $5498\ W/m^2\ K$

8. CALCULATION OF OUTSIDE HEAT TRANSFER COEFFICIENT

Length of the tube, L = 2m
Let, the number of baffles, N_b = 1

$N_b + 1 = L / L_s$
$L_s = 1$

S_m = $[\ L_s\ (P' - D_o)\ D_s\ ] / P'$
= $[(0.0254 – 0.01905) \times 0.540] / 0.0254$
= $0.135\ m^2$

G = W_s / S_m
= $9.13 / 0.135$
= $67.64\ Kg/m^2\text{-}s$

ρ_s = PM / RT
= $(2.13 \times 33.6) / (0.082 \times 429)$
= $2.044\ Kg/m^3$

$N_{Re} = DG/\mu$
$= [\,0.01905 \times 67.64\,] / 2.772 \times 10^{-5}$
$= 46484$

$N_{Pr} = 0.8368$

From the graph, we have
$j = 4 \times 10^{-3}$

$(h_o\, d_o / k) = j\,(N_{Re})\,(N_{Pr})^{(1/3)}$
$= 175.1$

$h_o = 316.7\ W/m^2\ K$

$$[\,1 / U_o\,] = [\,1 / h_o\,] + [\,D_o / D_i\,]\,[\,1 / h_i\,] + [D_o \times \ln\{D_o/D_i\} / (2K_w)] + [\,1 / h_{od}\,] + [\,D_o / D_i\,]\,[\,1 / h_{id}\,]$$

Taking,

$[\,1 / h_{od}\,] = 1 / 3000\ (m^2\text{-}K)/W$
$[\,1 / h_{id}\,] = 1 / 5000\ (m^2\text{-}K)/W$

$[\,1 / U_o\,] = 3.99 \times 10^{-3}\ (m^2\text{-}K)/W$

$U_o = 250.47\ W/(m^2\text{-}K)$

Note: As this value of U_o is greater than the corrected value of U_{oc}, so the design with the above specifications is accepted.

9. PRESSURE DROP CALCULATION:

For Tube side,

$f = 0.079 \times (N_{Re})^{-0.25}$
$= 6.193 \times 10^{-3}$

$\Delta P_L = [(4fLV_t^2) \times \rho_f\,] / \{2 \times D_i\}$
$= 2851.08\ N/m^2$

$\Delta P_t = 2.5 \times [\rho_f \times V_t^2 / 2]$
$= 2266.06\ N/m^2$

$$\Delta P_T = N_p (\Delta P_L + \Delta P_t)$$
$$= 30702.8 \text{ N/m}^2$$
$$= 30.7 \text{ KPa}$$

Note: As the value of the pressure drop is less than 70KPa, the design is acceptable from the tube side pressure drop consideration.

For Shell Side,

Pressure Drop in the Cross Flow section is calculated by,

$$\Delta P_c = [\{b \text{ x } f_K \text{ x } W^2 \text{ x } N_c\} / (\rho_f \text{ x } S_m^{\ 2})] \text{ x } \{\mu_w / \mu_b\}^{0.14} \qquad \text{KN/m}^2$$

N_{Re} = 46484
b = $2 \text{ x } 10^{-3}$
f_K = 0.12
$\rho_v (\rho_f)$ = 2.044 Kg/m^3
$m_g(W)$ = 9.13 Kg/s

$$N_c = [D_s (1 - 2 \{L_c/D_s\}) / P_p]$$
$$= 540 \text{ x } (1\text{-}2 \text{ x } 0.25) / 22$$
$$= 12.27$$

$$\Delta P_c = [\{2 \text{ x } 10^{-3} \text{ x } 0.12 \text{ x } 9.13^2 \text{ x } 12.27\}/(2.044 \text{ x } 0.135^2)] \qquad \text{KN/m}^2$$
$$= 6.6 \text{ KPa}$$

Pressure Drop in End Zones is calculated as,

$$\Delta P_e = \Delta P_c (1 + \{N_{cw} / N_c\}) \qquad \text{KN/m}^2$$

$$N_{cw} = 0.8 \, l_c / P_p$$
$$= [0.8 \text{ x } 0.25 \text{ x } 540] / 22$$
$$= 5$$

$$\Delta P_e = 6.6 [1 + (5 / 12.27)]$$
$$= 9.28 \text{ KPa}$$

Pressure Drop in Window Zones

$$\Delta P_w = [b \text{ x } W^2 \text{ x } (2 + 0.6 N_{cw}) / \{S_m \text{ x } S_w \text{ x } \rho_f\}] \qquad \text{KN/m}^2$$

b = $5 \text{ x } 10^{-4}$
S_w = $S_{wg} - S_{wt}$

From the graph from **PERRY Fig. 10-18, Pg. 10-29**

S_{wg} $= 75\ inch^2$
$= 0.04838\ m^2$
S_{wt} $= (N_t / 8) \times (1 – F_c) \times \pi \times D_o^2$

From the graph from **PERRY Fig. 10-16, Pg. 10-28**

F_c $= 0.65$
S_{wt} $= (320 / 8) \times (1 – 0.65) \times \pi \times 0.01905^2$
$= 0.01596\ m^2$

S_w $= S_{wg} - S_{wt}$
$= 0.04838 – 0.01596$
$= 0.03242\ m^2$

ΔP_w $= [5 \times 10^{-4} \times 9.13^2 \times (2 + 0.6 \times 5) / \{ 0.135 \times 0.03242 \times 2.044 \}]$
$= 23.29$ KPa

Therefore the total Pressure Drop on the shell side is calculated by the following relation

ΔP_s (TOTAL) $= 2 \times \Delta P_e + (N_b – 1) \times \Delta P_c + N_b \times \Delta P_w$
$= 2 \times 9.28 + 1 \times 23.29$
$= 41.856$ KPa

As this value of Pressure Drop on the shell side is less than the 70 KPa, the design is acceptable from the Pressure Drop Point of View.

Thus, the design is acceptable from process design consideration.

<u>SUMMARY OF PROCESS DESIGN FOR SINGLE COOLER</u>

Mass flow rate of process gas	= 9.13 Kg/s
Mass flow rate of water	= 13.98 Kg/s
Shell outer diameter	= 540 mm
Number of tubes	= 320
Tube OD	= 0.75 inch = 0.01905 m
Pitch (Triangular)	= 1 inch = 0.0254 m
Tube length	= 2 m
Shell side pressure drop	= 41.86 KPa
Tube side pressure drop	= 30.7 KPa
Cooler type	= TEMA L or M type 1-6 Heat Exchanger

MECHANICAL DESIGN OF COOLER

Working Pressure	$= 0.101\ N/mm^2$
	$= 1.03\ Kg/cm^2$
Design Temperature	$= 150\ ^{\circ}C$
Design pressure	$= 0.1084\ N/mm^2$
	$= 1.105\ Kg/cm^2$
Number of tubes	= 320
Shell diameter	= 540 mm

The entire mechanical design is referred from the literature in **PROCESS EQUIPMENT DESIGN** by **M.V JOSHI.**

1. SHELL THICKNESS

Material: IS 2825-1969 Grade I plain Carbon steel.

Shell thickness , (t_S) $= [\{P \times D_i\} / (2fJ - P)]$

$= [\{540 \times 1.105\} / (2 \times 950 \times 0.85 - 1.105)]$

= 0.37 cm

= 3.7mm

From the Table 9.2, its found that minimum shell thickness when severe conditions are not expected is 8mm, which includes the Corrosion Allowance.

2. NOZZLES

Take inlet and outlet nozzles as 100mm diameter.

Vent nozzle	= 25mm diameter
Drain nozzle	= 25mm diameter
Relief Valve	= 50 mm diameter.
Nozzle thickness	$= [P \times D_i] / \{2 f J - P\}$
	= 3.72mm

Minimum nozzle thickness is 6mm and 8mm is choosen which includes the corrosion allowance.

Also only the inlet and outlet nozzles need compensation. The compensation required is minimum and is given by pads of 10mm thickness.

3. HEAD

Torispherical heads are taken for both ends.

R_c (Crown radius) = 540 mm
R_{nk} (knuckle radius) = 54 mm

Head thickness (t_h) = [P x R_c x W] / { 2 f J }
Where,
W = (1/4) x [3 + $(R_c / R_{nk})^{0.5}$]
= 1.54

Head thickness = 4.836 mm
Therefore we take Head Thickness as that of the Shell Thickness = 8mm

4. TRANSVERSE BAFFLES

Number of Baffles = 1
Baffle cut = 25%
Baffle thickness = 6mm (standard)

5. TIE RODS AND SPACERS

Diameter of tie rods = 10mm
Diameter of Spacers = 8mm

6. FLANGE DESIGN

Flange is ring type with plain face.
Design pressure = 0.1084 N/mm^2 (external)

Flange material: IS 2004-1962 Class 2 Carbon Steel
Bolting steel: 5% Chromium, Molybdenum Steel

Gasket Material: Asbestos

Shell OD = 0.540 m
Shell Thickness = 0.008 m (g)
Shell ID = 0.532 m

Allowable stress for flange material = 100 N/mm^2
Allowable stress of bolting material = 138 N/mm^2

6 (i). DETERMINATION OF GASKET WIDTH

$$d_O/d_i = [(y-Pm)/(y-P(m+1))]^{0.5}$$

Assume a gasket thickness of 1.6mm

Minimum design yield seating stress , y = $25.5\ N/mm^2$

Gasket factor, m = 2.75

d_O/d_i = 1.002 m

Let, d_i = B+10

= 0.550 m

Minimum gasket width, N = 0.550(1.002-1)/2

= 0.00055 m

= 0.55 mm

Choose N = 40 mm.

d_o = 0.630 m

Basic gasket seating width, b_o = 40/2

= 20 mm

Effective Gasket Seating Width, b = $2.5 \times [b_o]^{0.5}$

= 11.18

Diameter at location of gasket load reaction G = d_i + N = 0.590 m

6 (ii). ESTIMATION OF BOLT LOADS

Under atmospheric conditions, the bolt load due to gasket reaction is given by

$$W_{m1} = \pi\, b\, G\, y$$

$$= \pi \times 2 \times 590 \times 25.5$$

$$= 94.53\ KN$$

Load due to design pressure

$$H = \pi G^2 P/4$$

$$= 29.64\ KN$$

where P is the design pressure

Load to keep joint tight under operation:

$H_p = \pi G(2b)mp$
$= \pi \times (590) \times (4) \times (2.75) \times (0.1084)$
$= 2.210$ KN

Total Operating Load, $W_{m2} = H + H_T$
$= 31.85$ KN

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

6 (iii). CALCULATION OF MINIMUM BOLTING AREA:

$A_m = A_o = W / S$
$= 94.53 \times 10^3 / S$

S = allowable stress for bolting material

$A_m = A_o = 94.53 \times 10^3/138 = 685 mm^2$

6 (iv). CALCULATION OF OPTIMUM BOLT SIZE.

Bolts are of 5% Cr Mo Steel

Number of bolts $= G / [b_o \times 2.5]$
$= 590 / [20 \times 2.5]$
$= 12$ bolts

Diameter of bolts $= [(A_m / \text{Number of bolts}) \times (4 / \pi)]^{1/2}$
$= 9$ mm

7. FLANGE THICKNESS

Thickness of flange, $t_f = [G\sqrt{(p/Kf)}] + C$

Where,
C is the Corrosion allowance
h_G is radial distance from gasket load reaction to bolt circle
Hydrostatic end force, $H = (\pi /4) G^2 p$
$= 29.63$ KN
$K = 1/[0.3 + \{(1.5 W_m h_G) / (H \times G)\}]$
$h_G = (B – G)/2$

Where,
B = Outside diameter of Gasket + 2xDiameter of Bolt +12mm
$= 630 + 2 \times 9 + 12$
$= 660$mm
Then, $h_G = (B – G)/2$
$= 0.035$ m

$K = 1/[\ 0.3 + \{(\ 1.5\ W_{m1}\ h_G) / (H \times G)\}]$
$= 1.713$

$t_f = [G\sqrt{(p/Kf)}\ ] + C$
$= 14.8 + C$
$= 20$ mm

Hence the thickness of flange = 20 mm

TUBE SIDE

Material:Stainless steel (IS- grade 10)

Thickness of tube = $t_f = \{P \times D_o\} / (\ 2\ f\ J + P)$

Where,

Working pressure = 12 N/mm^2
Design pressure, P = 14 N/mm^2
Permissible Stress, f = 100.6 N/mm^2
Joint Efficiency, J = 1.0
Thickness of tube = 1.24mm
Use tube with thickness of 2mm

No Corrosion allowance, since the tubes are of stainless steel.

1. TUBE SHEET

The tube sheet is held between shell flange and the channel. The joint on the shell flange side is of male and female facing and on the channel side of ring facing, since the pressure on the channel

Thickness of Tube Sheet , t_{ts} = $FG\sqrt{[(0.25\ P)/f]}$

Where,

F = 1.25
Thickness of tube sheet = 140 mm

2. CHANNEL AND CHANNEL COVER

Material :Carbon Steel

Permissible stress,f = 95 N/mm^2

For Ring type gasket

K =0.3

Thickness of channel, t_h $= G\sqrt{[(K \times P) / f\ ]}$
=125 mm

4. GASKET SIZE

Width of ring gasket, N = 22 mm
Inner diameter, D_i = 0.550 m
Outer diameter, D_o = 0.630 m

Gasket material: Steel Jacketed Asbestos

Gasket factor, m = 5.5
Minimum design seating stress, Y_a = 126.6 N / mm^2
Basic gasket seating width, b_o = N / 2
= 22 / 2
= 11 mm

Effective gasket seating width,b = $2.5 \times (b_o)^{0.5}$
= 8.3 mm
Mean diameter, G = $(D_i + D_o) / 2$
= 0.590 m

Design pressure, P = 14 N/mm^2

Under atmospheric conditions, the bolt load due to gasket reaction is given by

$$W_{m1} = \pi \, b \, G \, Y_a$$
$$= 1948 \text{ KN}$$

After the internal pressure is applied, the gasket which is compressed earlier, is released to some extent and the bolt load is given by

$$W_{m2} = \pi \times 2 \, b \times G \times m \times P + (\pi / 4) G^2 P$$
$$= 6197 \text{ KN}$$

"f " is permissible tensile stress in bolts under atmospheric condition

Bolt Material: 5%Cr Mo Steel,
f = 140.6 N/mm^2

A_m = area of bolt
A_{m1} = W_{m1} / f_a
= 13855 mm^2
A_{m2} = W_{m2} / f_b
= 44075 mm^2

Number of bolts = (mean diameter) / b_o x 2.5

= 22 bolts

To determine the size of bolts , the larger of above two areas should be considered

Diameter of bolts,D_b = $[(A_{m2} / \text{Number of bolts}) \times (4 / \pi)]^{1/2}$
= 51 mm

5. THICKNESS OF NOZZLE

Considering inlet and outlet diameter to be 100mm, then thickness of the nozzle is given by,

Thickness of nozzle, t_n = $(P \times D_n) / [2 \times f J - P]$
Permissible stress, f = 95 N/ mm^2
Joint Efficiency, J = 0.85
Then,
Thickness of nozzles =10 mm

6. FLANGE THICKNESS:

Flange material: IS 2004-1962 Class 2 Carbon Steel

Thickness of the Flange , t_f = $[G\sqrt{(P/Kf)}] + C$
Where,
C is the Corrosion allowance
Allowable stress for flange material, f = 100 N/mm^2

h_G is radial distance from gasket load reaction to bolt circle
Hydrostatic end force, H = $(\pi /4) G^2 P$
= 3827 KN
K =$1/[0.3 + \{(1.5 W_m h_G) / (H \times G)\}]$
h_G = $(B – G)/2$
Where,
B = Outside diameter of Gasket + 2xDiameter of Bolt +12mm
= 630 + 2 x 51 + 12
= 744 mm
Then, h_G = $(B – G)/2$
= 77 mm
K = $1/[0.3 + \{(1.5 W_{m2} h_G) / (H \times G)\}]$
= 1.620
Then,
t_f = $[G\sqrt{(P/Kf)}] + C$
= 158 + C
= 160 mm

Hence the thickness of flange = 160 mm

SUPPORT FOR SHELL AND TUBE HEAT EXCHANGER

Length of the heat exchanger, L	= 2000 mm
Outer diameter of Shell, Do	= 556 mm
Inner diameter of Shell, D_i	= 540 mm
Thickness of Shell, t_s	= 8 mm
Outer diameter of tube, d_o	= 19.05 mm
Inner diameter of tube, d_i	= 15.75 mm
Number of tubes, N_t	= 320

Density of Steel,ρ_s	= 7850 Kg /m^3
Density of Liquid in tubes ,ρ_l	= 1000 Kg /m^3

Volume of Shell body, V
$= (\pi / 4) (D_o^2 - D_i^2) \times L$
$= 0.0275\ m^3$

Weight of Shell body, W_s
$= V \times \rho_s$
= 216 Kgs

Volume of Tubes, V_t
$= (\pi / 4) (d_o^2 - d_i^2) \times L \times N_t$
$= 0.0577\ m^3$

Total Weight of Tubes, W_t
$= V_t \times \rho_s$
= 453 Kgs

Volume of Head, V_h
$= 0.087\ D_i^3$
$= 0.013\ m^3$

Weight of Head, W_h
$= V_h \times \rho_s$
= 102 Kgs

Weight of Liquid, W_l
$= (\pi / 4) (d_i^2) \times L \times N \times \rho_l$
= 124.6 Kgs

Total Weight, W
$= W_s + W_t + W_h + W_l$
= 900 Kgs
= 9.0 KN

Depth of head, H = 220 mm

Q $= (W/2) \times (L+4H/3)$
= 10.32 KN-m

Now, we calculate,
Distance of saddle center line from shell end ,
A $= 0.45 \times R_i$
$= 0.45 \times (0.540/2)$

= 121.5 mm

1. LONGITUDINAL BENDING MOMENTS

Radius,	R	= 0.270 m
Depth of head,	H	= 0.220 m

The bending moment at the supports is

$$M_1 = QA [1 – \{(A/L)+ (R^2 - H^2) / 2 AL\}/\{1 + 4H/3L\}]$$
$$= 171.68 \text{ N-m}$$

The bending moment at the center of the span is given by

$$M_2 = (Q L / 4)[\{1+ 2 (R^2 - H^2) / L^2 \}/\{1+ 4H / 3L\} - (4A / L)]$$
$$= 3302 \text{ N-m}$$

2. STRESS IN SHELL AT THE SADDLE

For θ $=120^\circ$

$k_1 = 0.107$

$k_2 = 0.192$

Thickness of shell, t = 8 mm

$$f_1 = M_1/(k_1\pi R^2 t)$$
$$= 876 \times 10^3 \text{ N/m}^2$$

$$f_2 = M_1/(k_2\pi R^2 t)$$
$$= 488 \times 10^3 \text{ N/m}^2$$

3. STRESS IN THE SHELL AT MID- SPAN

The stress at the mid span is f_3, which is either tensile or compressive depending on the position of the fiber. The resultant tensile stresses (including the axial stress due to internal pressure) should not exceed the permissible stress, and the resultant compressive stress should not exceed the permissible compressive stress

$$f_3 = M_2 /(\pi R^2 t)$$
$$=1.80 \times 10^6 \text{ N/m}^2$$

Axial Stress in Shell due to internal pressure

$$f_p = (P \times D_i)/ (4 t)$$
$$= 1.829 \times 10^6 \text{ N/m}^2$$

All combined stresses ($f_p + f_1$) , ($f_p + f_2$) , and ($f_p + f_3$) are well within allowable limits. Hence, the given parameters can be considered for design.

Note: This Cooler is fabricated 4 in number and are operated in parallel to take care of the Cooling duty required in the Process.