MAJOR EQUIPMENT DESIGN

PACKED BED ABSORBER

Equilibrium data for CO_2 and Monoethanolamine solution

X mol CO_2 / mol MEA sol	Y mol CO_2 / mol inerts
0.0038	0.026
0.0047	0.034
0.0057	0.044
0.0067	0.056
0.0076	0.067
0.0086	0.078
0.0095	0.087
0.0105	0.099
0.0115	0.109
0.0124	0.120
0.0133	0.133
0.0145	0.153
0.0160	0.180
0.0170	0.202

Gas flow rate at bottom G_b = 10559.1 kmol / hr

= 41.68 kg / s

Gas flow rate at the top G_t = 8726.9 kmol / hr

= 20.85 kg / s

Yt =0.005

Yb =0.216

G_{in} = 8683.5 kmol / hr

PROPERTIES

Gas density ρ_g = 0.487 kg / m^3

Liquid density ρ_l = 934.4 kg / m^3

Gas viscosity μ_g= 0.0175 cP

Liquid viscosity μ_l = 0.299 cP

Gas diffusivity D_g = 1.65X10^{-5} m^2 / s

Liquid diffusivity D_l = 1.96X10^{-5} m^2 / s

Gas heat capacity C_{pg} = 2.094 kJ / kg K

Liquid heat capacity C_{pl} = 4.145 kJ / kg K

From Graph

$(L_{in} / G_{in})_{min}$ = 11.99

(L_{in} / G_{in}) = 1.1×11.99 = 13.189

L_{in} = 13.189x 8683.5

= 99652.5 kmol / hr

Top Section

L_t = Ls $(1+X_t)$

= 99652.5 kmol / hr

Bottom Section

L_b = 99652.5$(1+X_b)$

= 101117.4 kmol / hr

14.5 wt% MEA souliton has molecular wt.= 20.08 kg / kmol

L_t = 198026.6 kg / hr = 555.00 kg/s

L_b = 92452.82 kg / hr = 577.85 kg / s

Calculation of coloumn diameter

Choosing 90 mm pall rings (metal)

Void fraction ε = 0.97

Packing factor F_p = 53

Surface area a = 92 m^2 / m

$$\frac{L}{G}\left(\frac{\rho_g}{\rho_l}\right) = 0.61$$

At the bottom

$$\frac{L}{G}\left(\frac{\rho_g}{\rho_l}\right) = 0.56$$

At the top

Hence choosing the larger value of 0.61

From Graph

$$\frac{G_f^2 F_p \psi \mu_l^{0.2}}{\rho_g \rho_l g} = 0.038$$

Where

G_f = gas superficial velocity

F_p = packing factor = 53 m^{-1}

ψ = correction factor for density = 1.029

μ_l = viscosity of liquid in cp = 0.299

ρ_g = density of gas = 0.487 kg/m^3

ρ_l = density of liquid = 934.4 kg/m^3

g = acceleration due to gravity

On substituting we obtain

G_f =2.085 $kg/m^2\,s$

Operating G = $0.85\underline{G_f}$

= 1.772 kg/m^2s

A_c = G_b/G

A_c = 20.85 / 1.772

A = 9.948 m^2

Di = 3.20 m

$\boxed{\textbf{Design dia =5m}}$

Pressure drop

$$\Delta P = C_2 10^{C_3 U_{tl}} \rho_g U_{tg}^2$$

ΔP = pressure drop expressed in terms of water

C_2, C_3 are constants for specific type of packing

U_{tg}, U_{tl} = superficial velocities of gas and liquid expressed in ft / s

ρ_g = gas density in lb / ft^3

$U_{tg} = G_{avg} / \rho_g A_c$ = 4.226 m / s = 14.108 ft / s

$U_{tl} = L_{avg} / \rho_l A_c$ = 0.0596 m / s =0.197 ft / s

Hence ΔP = 27.43 in of H_2O / ft of packing

= 2285.4 mm of H_2O / m of packin

Degree of wetting from Nuris & Jackson criterion (Weeping Check)

$$\overline{L_W} = \frac{L_{\min}}{\rho_l a A_c}$$

Where

L_{min} = minimum flow rate of liquid = 555.00 kg / s

ρ_l = density of liquid = 934.4 kg / m^3

a = surface area of packing per unit volume = 92 m^2 / m^3

wetting rate is greater than 0.85 ft^3 / ft hr hence no weeping will occur

Evaluation of tower height

$$Z = H_{OG} N_{OG}$$

$$H_{OG} = H_g + m\frac{G_m}{L_m} H_l$$

For ring type of packings

$$H_g = \frac{0.017\psi D^{1.24} Z^{0.33} Sc_g^{0.5}}{(Lf_1 f_2 f_3)^{0.5}}$$

ψ = parameter for a given packing = 75

D = column diameter if diameter is less than 0.6m else 0.6m

Sc_g = Schmidt no for gas phase =2.1778

L = liquid rate = 55.58 kg / m^2s

$f_1 = (\mu_l / \mu_w)$ = 0.8243

$f_2 = (\rho_l / \rho_w)$ =1.0885

$f_3 = (\sigma_w / \sigma_l)$ = 1.0611

substituting we get

$H_g = 0.25Z^{0.33}$

Liquid phase transfer unit

$$H_l = \frac{\varphi C}{3.28} Sc_l^{0.5} \left(\frac{Z}{3.05}\right)^{0.15}$$

φ = correlation parameter for a given packing = 0.11

C = correlation parameter for high gas flow rates = 0.5

Sc_g = Schmidt no for liquid phase = 163.6

Z = tower height

On substitution

$H_l = 0.181Z^{0.15}$

m = slope of the equilibrium curve =11.25

L_m = 100449.7 kmol / hr

G_m = 9643.0 kmol / hr

Overall number of transfer units in the gas phase

$$N_{OG} = \int_{Y_t}^{Y_b} \frac{dY}{Y - Y^*} - \frac{1}{2}\ln\left(\frac{1+Y_b}{1+Y_t}\right)$$

Y	Y^*	$1/(Y-Y^*)$
0.005	0	200
0.015	0.005	100
0.025	0.010	66.67
0.035	0.015	50
0.045	0.020	40
0.055	0.025	33.33
0.065	0.032	30.30
0.075	0.039	27.78
0.085	0.047	26.32
0.095	0.057	26.32
0.105	0.065	25
0.115	0.075	25
0.125	0.083	23.81
0.135	0.095	25
0.145	0.100	22.22
0.155	0.107	20.83
0.165	0.116	20.41
0.175	0.128	21.17
0.185	0.138	21.17
0.195	0.151	22.73
0.205	0.164	24.49
0.210	0.171	25.64
0.215	0.180	28.57

$$\int_{0.005}^{0.215} \frac{dY}{Y - Y^*} = 7.284 + 0.259 = 7.543$$

Computing by using Simpsons 1/3 rule for numerical integration

Hence

$$Z = N_{OG} H_{OG} = 1.862 Z^{0.33} + 1.452 Z^{0.15}$$

computing the value of Z by trial and error we have

$$\boxed{\mathbf{Z \;=\; 5.0\ m}}$$

The total pressure drop is **ΔP = 11.427 m of H_2O**

MINOR EQUIPMENT

Shell and tube heat exhanger

The gases coming out of the NH_3 converter is at a very high temperature of 778K These gases are used to heat the inlet gases to the converter. The gases then are cooled to a temperature of 313 K before it enters the condenser where Ammonia is condensed. Because the heat load is very high five HE are used in parallel.

Amount of heat to be removed Q	=	4989.1 kW
Mass flow rate of gas m_h	=	23.8 kg/s
Inlet temperature T_1	=	405 K
Outlet temperature T_2	=	313 K

It is assumed that cooling water inlet temperature is 293 K and exit temperature is 308 K.

Amount of cooling water $m_c = Q/C_p\,\Delta T$	= 79.4 kg/s
Inlet temperature t_1	= 293 K
Outlet temperature t_2	= 308 K

$$LMTD = \frac{(T_1 - T_2) - (t_2 - t_1)}{\ln\left(\frac{T_1 - T_2}{t_1 - t_2}\right)}$$

= 50.38 K

$$R = \frac{T_1 - T_2}{t_2 - t_1} = 6.46$$

$$S = \frac{t_2 - t_1}{T_1 - t_1} = 0.12$$

For these values the LMTD correction factor $F_T = 1$

$(LMTD)_c = 50.38$ K

Properties

	Gas stream at 359 K	Water at 300.5 K
ρ (kg / m^3)	20.03	996.4
μ (Pa s)	4.876×10^{-5}	9×10^{-4}
k (W / m K)	0.125	0.359
C_p (kJ / kg K)	1.991	4.187

Routing of Fluids

The gases are taken on shell side. Cooling water is taken on the tube side.

AREA OF HEAT TRANSFER

Assuming U_d = 350 W / m^2K

Area of heat transfer is given by

$$A = \frac{Q}{U_d (LMTD)_C} = 242.83 m^2$$

For the exchanger 1 inch 16 BWG tubes are chosen

d_o = 25.4 mm

d_i = 22.1 mm

area of tube per unit length = 0.2168 ft^2 / ft

Assumption is made that the tubes are of length 16 ft

Hence the area per tube a = 4.1888 ft^2 = 0.3333 m^2

No of tubes

$$N_t = \frac{A}{a} = 730$$

From the tube count table we have for 1 inch o.d tubes on 1.25 inch square pitch

N_t =738 N_p = 4 D_s = 1067 mm

Hence corrected area for heat transfer A =N_t x a = 245.97 m^2

Thus corrected coefficient of heat transfer U_{ac} = 341 W / m^2 K

Velocity

Tube side

Tube cross sectional area for flow of fluid in one tube pass

$$a_T = \frac{\pi}{4} d_i^2 \frac{N_t}{N_p} = 0.0708 m^2$$

where

d_i is the tube inside diameter =22.1 mm

N_t is the number of tubes =738

N_p is the number of tube passes =4

Tube side velocity

$$v_T = \frac{m_C}{a_T \rho_T} = 1.125 \, m/s$$

where

m_c is the mass flow rate of cold water

ρ_t is the density of tube side fluid

The velocity of a liquid should lie between 1 and 2 m/s which is satisfied.

Shell Side

Shell side cross sectional area for fluid flow

$$a_S = \frac{(P_0 - d_o) l_s D_s}{P_o} = 0.1423 m^2$$

Where

P_o = pitch = 31.75 mm

d_o = outside diameter = 25.4 mm

D_s = shell inside diameter = 1067 mm

l_s = baffle spacing assumed to be 0.5 D_s = 534mm

Shell side velocity is given by

$$v_s = \frac{m_s}{a_s \rho_s} = 10.325 \, {}^{m}\!/_{s}$$

Where

m_s = shell side mass flow rate

ρ_s = shell side fluid density

The velocity of flow for gases must lie between 10 and 40 m / s which is satisfied

Film Transfer Coefficient

Tube side

Reynolds number

$$\mathrm{Re} = \frac{\rho v d_i}{\mu} = 27524.8$$

Where

μ is the viscosity of fluid

ρ is the density of fluid

d_I is the inside diameter of tube

Prandtl number

$$\mathrm{Pr} = \frac{\mu C_p}{k} = 10.75$$

Where

C_p is the specific heat capacity

k is the thermal conductivity

Nusselts equation is given as

$$Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.4}$$

$$\frac{h_t d_i}{k} = 167$$

$$h_t = 2646.1 \, {}^{W}\!/_{m^2 s}$$

Where

h_t is the tube side heat transfer coefficient

Shell side

Reynolds number

$$\mathrm{Re} = \frac{\rho v_s d_o}{\mu} = 53759$$

Prandtl number

$$\mathrm{Pr} = \frac{\mu C_p}{k} = 0.94$$

By Deitus Bolter equation Nusselts number

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3}$$

$$\frac{h_s d_o}{k} = 137.6$$

$$h_s = 681.64 \,{}^{W}\!/_{m^2 s}$$

Where

h_s is the shell side heat transfer coefficient

Overall heat transfer coefficient

$$\frac{1}{U} = \frac{1}{h_t} + \frac{1}{h_{td}} + \frac{d_i}{d_o}\left(\frac{1}{h_s} + \frac{1}{h_{sd}}\right) + d_i\,\frac{\ln(d_o/d_i)}{k}$$

Where

h_d represents the dirt coefficient. It is assumed to be 2839 W / m² K for both shell and tube side

On substitution we obtain

U = 366.4 W / m² s

The design value is larger than the assumed value. Hence design is safe.

Pressure Drop

Tube side

Re = 27524.8

$$f = 0.079\,\mathrm{Re}^{-0.25} = 0.00613$$

$$\Delta P_L = \left(\frac{4\,fLv_t^2}{2gd_i}\right)\rho g = 3384.38\,N/m^2$$

where

f is the friction factor

l is length of pipe

v_t is the tube side velocity

g is the acceleration due to gravity

d_i is the tube inside diameter

$$\Delta P_c = 2.5\left(\frac{\rho v_t^2}{2}\right) = 1576.25\,N/m^2$$

$$\Delta P_t = N_p\left(\Delta P_L + \Delta P_c\right) = 19.842kPa$$

Where

N_p is the no of tube passes =4

The pressure drop for liquids must lie between 14 and 70 kPa. Hence the design is safe from view of pressure drop on the tube side.

Shell Side

Pressure drop in the cross flow section

$$\Delta P_{bk} = b\frac{f_k W^2 N_c}{\rho S_m^2}\left(\frac{\mu_w}{\mu_l}\right)^{0.14}$$

f_k is the ideal tube bank friction factor =0.104

W is the mass flow rate = 23.8 kg/s

$$N_c = \frac{D_s\left[1-2\left(l_c/D_s\right)\right]}{P_p} = 16.8$$

N_c is no of tube rows crossed in one cross flow section

l_c is the baffle cut =0.25 D_s

P_p is pitch parallel to flow = 31.75 mm

D_s is the shell inside diameter =1067 mm

ρ is the fluid density = 20.3 kg / m^3

S_m is cross flow near centerline for one cross flow section = 0.1423 m^2

ΔP_{bk} = 0.4555 kPa

Pressure drop for an ideal window section

$$\Delta P_{wk} = b\frac{W^2\left(2+0.6N_{cw}\right)}{S_m S_w \rho}$$

b =2 x 10^{-3}

S_w is for flow through window =0.1598 m^2

ΔP_{wk} = 0.3657 kPa

$$\Delta P = \left[\left(N_b - 1\right)\Delta P_{bk} + N_b \Delta P_{wk}\right] + 2\Delta P_{bk}\left(1+\frac{N_{cw}}{N_c}\right)$$

$$N_b = \frac{L}{l_s} - 1 = 8$$

ΔP = 7.845 kPa

The pressure drop for gases should be between 2 and 20 kPa. Hence the design is safe on shell side also

MECHANICAL DESIGN OF ABSORBER

Material for shell is Carbon Steel

THICKNESS OF SHELL

Thickness of shell = t_s

$$t_s = \frac{pD}{2fJ + p} + c$$

Where,

D_i = Inner Diameter of vessel = 3.2m

Working Pressure = 1 atm = 1.054kgf / cm^2

p = Design Pressure = 1.1 x 1.054 = 1.137 kgf / cm^2

f =Permissible Stress = 950 kgf / cm^2

J= Joint Efficiency = 0.85

c= Corrosion allowance =2mm

t_s = 6mm

thickness of shell is 6mm

HEAD DESIGN: FLANGED & SHALLOW

Material stainless steel

Permissible stress = f= 130 N / mm2

Design pressure = p = 1.064 x 10 5 N / m2

Stress identification factor W is given by

W = (¼) [3 + (R_c / R_1)$^{1/2}$]

Crown Radius = R_c= 3.2m

Knuckle radius = R_1 = 0.192m

Stress identification factor W is 1.77

Thickness of head = t_h = $\frac{pR_cW}{2fJ}$

t_h= 6mm

Axial Stress Due to Pressure

Axial stress due to pressure $=f_{ap}$

$$f_{ap} = \frac{pD_i}{4(t_s - c)}$$

$= 355.4 \text{ kgf/cm}^2$

Stress due to Dead Load

a) Compressive Stress due to weight of shell up to a distance X

$D_o = D_i + 2\,t_s = 3.212\text{m}$

Density of Shell material = ρ_s = 7700 kg / m3

$$f_{ds} = \frac{\frac{\pi}{4}\left(D_o^2 - D_i^2\right)\rho_s X}{\frac{\pi}{4}\left(D_o^2 - D_i^2\right)}$$

$=.77X \text{ kgf/cm}^2$

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

Insulator used is asbestos

Thickness of insulation = t_{ins} =100mm

Diameter of insulation = D_{ins}

Density of insulation =2300 kg / m^3

Mean diameter of vessel = D_m

For large diameter column

$D_{ins} = D_m$

$$f_{ins} = \frac{\pi D_{ins} t_{ins} \rho_{ins} X}{\pi D_m (t_s - c))}$$

$= 1.925 \text{ kgf/cm}^2$

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

Density of liquid $=\rho_l$ = 934.4 kg / m^3

$$f_{liq} = \frac{\frac{\pi}{4} D_i^2 \rho_l \varepsilon}{\pi D_m (t_s - c)}$$

$= 23.29 \text{ kgf/cm}^2$

d) Compressive stress due to attachment

i. Packing weight

ii. Head weight

iii. Ladder

Density of packing (pall ring) = 270 kg / m^3

Packing Weight = $\dfrac{\frac{\pi}{4} D_i^2 \rho_p X}{\pi D_m (t_s - c)}$

Head weight (approximately) = 2000kg

Weight of Ladder = 140 kg / m

Total compressive stress due to attachments is given by

f_{attach} = (Packing Weight + Head weight + Ladder) / [π D_i (t_s – c)]

= (3.1972 + 0.2225 X) kg / cm^2

Stress due to Wind

Stress due to wind is given by

$$f_{ws} = \frac{1.4 P_w X^2}{\pi D_m (t_s - c)}$$

Where,

Pressure due to wind = P_w = 125 kgf / m^2

=0.2778 X^2 kgf / cm^2

To determine the value of X

f_{max} = 950 kgf / cm^2

Upwind side

Jf_{tmax} = f_{wx} + f_{ap} +f_{dx}

807.5 = 365.4+3.1792+39.6345 X +0.2778 X^2

Solving for X we have

$X = 10.331m$

Downwind side

$Jf_{tmax} = -f_{wx} - f_{ap} - f_{dx}$

$807.5 = -365.4-3.1792-39.645\ X + 0.2778\ X^2$

Solving for X for compressive stresses

$X=167$ m

Hence the column will support a height of 5m as the calculated height is much higher than 5 m in both cases

DESIGN OF GASKET AND BOLT

Width of gasket =N = 10 mm

Gasket material is asbestos

Gasket factor = m =2

Minimum design seating stress = $Y_{a=}$ 11.2 N / mm2

Basic gasket seating width $b_o = N/2$

b_o = 10 mm / 2 = 5mm

Effective gasket seating width

$b = 2.5\ (b_o)^{1/2}$

Inner diameter = D_i = 4.85 m

Outer diameter = D_o = 4.85m + 2 x 8 x 10 –3

Flange inner diameter =D_{fi} = 4.866 m

Flange outer diameter = D_{fo} = 4.926 m

Mean diameter = $G = (D_{fi} + D_{fo}) / 2$

= 4.896 m

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

$W_{m1} = \pi\ b\ G\ Y_a$

=962.98 x10 3 N

Design pressure = p

$= 1.064 \times 10^5$ N / m2

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

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

$= 2.039 \times 10^6$ N

Bolt used is hot rolled carbon steel

f_a is permissible tensile stress in bolts under atmospheric condition

f_b is permissible tensile stress in bolts under operating condition

$f_a = 58.7 \times 10^6$ N / m2

$f_b = 54.5 \times 10^6$ N / m2

A_m is the area of bolt

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

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

$A_{m1} = 0.016$ m2

$A_{m2} = 0.037$ m2

Number of bolts = mean diameter / b_o x 2.5

=392 bolts

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

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

=1.096 cm

FLANGE THICKNESS

Thickness of flange = t_f

$t_f = [G\sqrt{(p / K f)}\,] + c$

Where,

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

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

$=2.003 \times 10^6$ N

h_G is radial distance from gasket load reaction to bolt circle,

$h_G = (B – G) / 2$

$= 0.0155$

B = outside diameter of gasket + 2 x diameter of bolt + 12mm

= 4.927 m

$W_m = 2.039 \times 10^6$ N

K= 3.28

Hence the thickness of flange = 90.47 mm

NOZZLE THICKNESS

Material Carbon steel

Considering diameter of nozzle = D_n =150 mm

Thickness of nozzle =t_n

Material is Stainless steel (0.5 cr 18 Ni 11 Mo 3)

Permissible stress $=130 \times 10^6$ N / m2

J=0.85

$$t_n = \frac{pD}{2fJ - p}$$

t_n = 1.24mm

No corrosion allowance , since the material is stainless steel.

We can use thickness of 3mm

SUPPORT FOR ABSORBER

Material used is structural steel (IS 800)

Skirt support is used.

Inner Diameter of the vessel = D_i =3.2m

Outer Diameter of the vessel = D_o =3.212m

Height of the vessel = H = 5.0m

Density of carbon steel= ρ_s = 7700 kg / m3

Density of liquid = ρ_l = 934.4 kg / m^3

Total weight = Weight of vessel+ Weight of Attachments (liquid+ packing + head + ladder)

=59430 kg

total stress due to dead loads

$$f_{ds} = \frac{\sum w}{\pi D_m t} = \frac{59.00}{t} \text{ kgf/cm}^2$$

Stress Due to wind load

The force due to wind load acting on vessel is

p_{lw} = k p_1 h_1 D_o

.

k=0.7 for cylindrical surface

p_1 is wind pressure for the part of the vessel upto a height of 20 m.

p_1 = 128.5 kgf / m^2

h_1=8m

$$f_{wb} = \frac{1.4 p_1 H^2}{\pi D_m t} = \frac{0.054}{t} \text{ kgf/cm}^2$$

Stress due to Seismic Load

Load F= CW

W is total Weight of vessel

C is Seismic Coefficient =0.08

$$f_{sb} = \frac{8CWH}{3\pi D_o t} = \frac{0.2246}{t} \text{ kgf/cm}^2$$

Maximum Stress at bottom of Skirt

$f_{tmax} = (f_{wb} \text{ or } f_{sb}) - f_{db}$

$$= \frac{58.775}{t} \text{ kgf/cm}^2$$

Permissible tensile Stress for structural steel = 950 kgf/cm^2

Equating the two we have

$t \geq 0.74mm$

therefore the thickness of skirt is taken as 6mm

Maximum Compressive Stress

$f_{cmax} = (f_{wb} \text{ or } f_{sb}) + f_{db}$

$$= \frac{59.2446}{t} \text{ kgf/cm}^2$$

permissible tensile stress for structural steel is 950 kgf/cm^2

Equating the two we have

$t \geq 0.8mm$

Skirt Bearing Plate

$$f_c = \frac{\sum w}{A} + \frac{M_s}{Z}$$

$\sum$w = sum of all the dead loads

A = $(\pi/4)$ (D_{sko}^2 – D_{ski} 2) is the area of contact between skirt plate and the concrete.

M_s = moment due to wind load

Z = section modulus

f_c =5.689 kgf/cm^2

Permissible stress f in bending is 950 kgf/cm^2

$$f = \frac{6M_{\max}}{bt_B^2} = \frac{3f_c l^2}{t_B^2}$$

t_B= 63mm

Anchor Bolt,

W_{min}=45000 kg (assumed)

f_c = (W_{min}/ A) – (M_w /Z)

= 1.115 kgf/cm^2

F_c is positive ,hence anchor bolts are not required if the safety factor is above 1.5

$$j = \frac{W_{\min} R}{M_s}$$

= 7.45

The safety factor is above 1.5. Hence anchor bolts are not required.

MECHANICAL DESIGN OF SHELL AND TUBE HEAT EXCHANGER

Carbon Steel (Corrosion allowance 3 mm)

SHELL SIDE

Number of pass =1

Fluids in shell are Hydrogen, Nitrogen Ammonia and inerts like Argon and Methane

Design pressure =50.9 kgf?cm^2

Shell diameter = 1067mm

Considering steel dished head (torispherical)

R_i= 1067 mm

r_i =0.06 of R_i

Inside depth of head can be calculated as

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

= 136 mm

Effective Length = L = 4.88 m + 2 x (0.136)

= 5.154 m

NOZZLE THICKNESS

Material used is carbon steel

Considering diameter of nozzle to be 150mm

Permissible stress = f = 950 kgf / cm^2

Corrosion allowance = 3mm

$$t_h = \frac{pD_i}{2fJ - p} + c$$

=5mm

HEAD THICKNESS

Thickness of head = $t_h = \frac{pR_iW}{2fJ} + c$

$W = \frac{1}{4}\left(3 + \sqrt{R_i / r_i}\right)$ =1.77

Hence t_h = 60 mm thickness

TRANSVERSE BAFFLE

Spacing between baffles as = 534 mm

FLANGE JOINT (BETWEEN SHELL AND TUBE SHEET)

This will be made to satisfy the requirements of the flange joint between tube sheet and channel. Gasket width (N) = 22 mm

Flange Thickness

Inner diameter of flange =D_{fi} = 1143 mm

Outer diameter of flange = 1167mm

Gasket Size G=1155 mm

Material is Steel Jacketed asbestos

Width of gasket =N = 24 mm

Gasket factor = m = 3.75

Minimum design seating stress = y_a= 5.34 kgf/cm^2

Basic gasket seating width b_o = N/2

b_o = 24 mm / 2 = 12 mm

Effective gasket seating width

b = 2.5 (b_o)$^{0.5}$ = 9 mm

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

$W_{m1} = \pi bGY_a$ = 100500 kg

Design pressure = p = 50.9 kgf/cm^2

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

$$W_{m2} = 2\pi bGmp + \frac{\pi}{4}G^2 p \qquad = 667719 \text{ kg}$$

Thickness of flange = t_f

$$t_f = G\sqrt{\frac{p}{kf}} + c$$

Where,

$$k = \frac{1}{0.3 + \frac{1.5 w_m h_g}{HG}}$$

Hydrostatic end force = $H = \frac{\pi}{4}G^2 p$

h_G is radial distance from gasket load reaction to bolt circle,

h_G = (B – G)/2

=37.5 mm

B = bolt circle diameter

w_m is the greater of the two loads in this case W_{m2}

Hence the thickness of flange = 138.8 mm

TUBE SIDE

Stainless steel (IS- grade 10)

<u>Tube thickness</u>

$$t_t = \frac{pD}{2fJ + p}$$

Where,

Working pressure = 1atm = 1.054 kgf/cm^2

Design pressure = p =1.1 times 1.054 = 1.136 kgf/cm^2

Permissible Stress =f = 950 kgf/cm^2

J=1 for seamless tubes

Thickness of tube =0.2 mm

The thickness of tubes is taken as ***2 mm*** with no corrosion allowance as the material is made of stainless steel.

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

$$t_{ts} = FG\sqrt{\frac{kp}{f}}$$

Where F = 1

K=0.25

Thickness of tube sheet = 18 mm

Channel and Channel Cover

Carbon Steel

Thickness of channel

Permissible stress = f = 95 kgf/cm^2

K=0.3

Thickness of channel

$$t_h = G\sqrt{\frac{kp}{f}}$$

Substituting the values the thickness of the channel and channel cover is obtained as 19mm

Flange Joint

Gasket material is steel jacketed asbestos

Gasket width (N) = 22 mm

Gasket factor = m = 5.5

Minimum design seating stress = Y_a= 12.66 kgf/cm^2

Basic gasket seating width b_o = N / 8

b_o = 2.75 mm

Effective gasket seating width

b = b_o

Mean diameter = G = 1000mm

Design pressure = p = 1.136 kgf/cm^2

$$W_{m1} = \pi G b y_a$$

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

= 10002 kgf

After the internal pressure is applied, the gasket which is compressed

$$W_{m2} = 2b\pi Gmp + \frac{\pi G^2 p}{4}$$

earlier, is released to some extent and the bolt load is given by

= 109374 kgf

Bolt used is hot rolled carbon steel

f_a is permissible tensile stress in bolts under atmospheric condition

f_b is permissible tensile stress in bolts under operating condition

For Cr Ni Steel,

f_a= 1406 kgf/cm^2

f_b = 66 kgf/cm^2

$$A_{m1} = \frac{W_{m1}}{f_b}$$

=7.114 cm^2

$$A_{m2} = \frac{W_{m2}}{f_a}$$

= 16.572 cm^2

A_m is larger area of A_{m1} and A_{m2} =16.572 cm^2

Number of bolts = mean diameter / b_o x 2 = 40 bolts

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

$$d_b = \sqrt{\frac{A_m}{4\pi}}$$

= 36 mm

Using M 48 bolts

Pitch diameter = 44.68 mm

Minor diameter = 41.795 mm

$$A_b = \frac{\pi}{8}(4.4681 + 4.1795)^2$$

Actual bolt area is given by

= 19.66 cm^2

Minimum pitch diameter = G +N+2d_b = 1118mm

Hence Bolt diameter is chosen as B= 1150 mm

$$t_f = G\sqrt{\frac{p}{kf}} + c$$

$$k = \frac{1}{0.3 + \frac{1.5W_m h_G}{HG}}$$

Flange Thickness

=2.91

Hence t_f = 25 mm

THICKNESS OF NOZZLE

Considering inlet and outlet diameter to be 200mm

$$t_n = \frac{pD}{2fJ + p}$$

Thickness of nozzle

= 1 mm

Thickness of nozzles is taken as 6mm

SUPPORT FOR SHELL AND TUBE HEAT EXCHANGER

Length of the heat exchanger = 4.88m

InnerdiameterofShell=1067 mm

Outer diameter of Shell = 1137 mm

Inner diameter of tube = 22.1 mm

Outer diameter of tube = 25.4 mm

Number of tubes =738

Density of Steel $=\rho_s$ =7850 kg / m3

Density of Liquid $=\rho_l$ = 995.3 kg / m3

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

= $0.591 m^3$

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

= 4479 kg

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

= 0.44 m3

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

= 3360 kg

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

= $0.106 m^3$

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

= 801 kg

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

= 10845 kg

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

= 19485 kgf

Q =W / 2 =9742.5 kgf

Distance of saddle center line from shell end =A= $0.6 \times R_i$

=320 mm

Longitudinal Bending Moments

Radius = R =1.067m

Height of head =H = 0.136 m

The bending moment at the supports is

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

$$= 1842.6 \text{ kgf m}$$

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

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

$$= 2131 \text{kgf m}$$

Stress in Shell at the Saddle

For A > 0.5 R_i , the shell is not sufficiently stiffened by the end

For θ =120,

k_1= 0.107

k_2 = 0.192

Thickness of shell t = 35 mm

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

$= 13.97 \text{ kgf/cm}^2$

$f_2 = M_1 / (\ k_2 \pi\ R^2\ t)$

$=7.79 \text{ kgf/cm}^2$

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 .

$f_3 = M_2\ / (\pi R^2\ t\ )$

$= 18.62 \text{ kgf/cm}^2$

Axial Stress in Shell due to internal pressure

$$f_p = (p\ D_i) / (4\ t)$$

$$= 34.99\ kgf/cm^2$$

The combined stresses ($f_p + f_1$) , ($f_p + f_2$) , and ($f_p + f_3$) are will within the permissible values