6
Design of Equipment

PACKED BED ABSORBER

Absorber system MEA is used as the absorber and its 14.5% in solution .

Amount of gas components present in absorber before entering the packed column is given by:

Gas composition	vol%	kmol
CO_2	33.86	905.36
CO	1.60	42.69
H_2	63.73	1703.97
CH_4	0.42	11.3
N_2	0.39	10.40

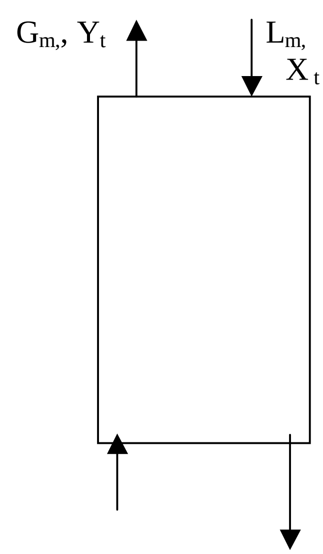
Assumption made in this type of absorber is that only co_2 is absorbed and all other gases act as a inert in 14.5% MEA solution

Gas flow rate of inerts (G_m) = 1768.35 kmol/hr

Also the mole ration of carbon dioxide and inerts at top and bottom is given by

Y_b (kmole of CO_2/kmole of inerts) = 0.51

Y_t (kmole of CO_2/kmole of inerts) =0.014



By carbon dioxide balance we get

$G_m(Y_b - Y_t) = L_m(X_b - X_t)$

Assuming a pure MEA solution is used for absorbtion

Therefore X_t=0

We get

L_mX_b = 877.10 kmol/hr

Now from graph

$(L_m/G_m)_{min}$ = 0.508

also $(L_m/G_m)_{actual}$ = (1.1 to 1.5 times)$(L_m/G_m)_{min}$

now assuming $(L_m/G_m)_{actual}$ = 1.25X$(L_m/G_m)_{min}$

$(L_m/G_m)_{actual}$ = 1.25X0.508 =0.6375

also G_m =1768.35 kmol/h

L_m=1768.35X0.6375=1127.32 kmol/hr

But the amount obtained is only of MEA , thus amount of solution is given by

L_m=1127.32/14.5 X10^{-2} =7774.64 kmol/hr

From above equation we get

X_b =0.113

Column diameter calculation:

G_b = 44910.82 kg/hr

G_b = 12.48 kg/sec

L_b= L_m X 24.25 + 877.10 X 44

L_b =227127.42 kg/hr = 63.09 kg/sec

Also calculated density of gases and liquids are

For **liquid**

ρ_{liq} =992.06 kg/m^3 μ =0.9 cp(from perry)

for **gases**

ρ_{gas}=0.626 kg/m^3

Let us choose ,**Intalox saddles ,ceramic** as packing material

From table 18-5 ,page 18-23, of perry

We get

D_p= 38 mm ε =0.80

Specific surface area =195 m^2/m^3 F_p=170

Now we have

$L/G \times (\rho_g/\rho_l)^{1/2} = (63.09/12.48) X (0.626/992.06)^{0.5} = 0.126$

Where
L =liquid mass rate ,kg/(m^2.s)

G =Gas mass rate,kg/(s.m^2)

Also from fig 18- 38, page 18-22 of perry

We get

$$\frac{G^2\ Fp\ \Psi\ \mu^{0.2}}{\rho_g\ \rho_l\ g} = 0.14$$

where
G =Superficial mass flow rate of the gas kg/s.m^2

U =superficial gas velocity,m/s

A_p= Total area of packing ,m^2/m^3(bed)

ε =fractional void in dry packing

ρ_l and ρ_g=liquid and gas density,kg/m^3

μ_l= liquid viscosity ,cP

therefore we get

G_f =2.275 kg/(s.m^2)

Gas flow rate of bottom is fixed so that the cross section can be calculated.

For this we have to operate below the flooding limit, thus G we choose should

Be 60-85% of G_f

Also we have

A_c= G_b/(0.85XG_f) m^2

A_c = 6.45 m^2

D_c = 2.866m

Also to ensure, there is proper wetting column diameter should be at least 10 times greater then packing diameter and above value of column diameter satisfies the given condition.

Pressure drop calculation

This is calculated using the formula's from perry

$\Delta P = C_2 \times (10^{C_3 U_{tl}}) \times \rho_g \times (Utg)^2$

where

ΔP=in H_2O/ft packing

ρ_g = gas density ,lb/ft^3

U_t and u_t =superficial velocities of gas and liquid respectively

C_2 and C_3 = constant given in table

now we have

L =63.09/6.45 =9.77kg/sec.m^2 =7204.398 lb/hr.ft^2

G =12.48/6.45 = 1.93 =1423.18 lb/hr.ft^2

U_{tg}=1.93/0.626 =3.08 m/s =10.10 ft/sec

U_{tl} = 9.77/992.06 =9.85X10^{-3}=0.0323 ft/sec

Also from table 18-7we get

C_2=0.14

C_3=0.0181

$\Delta P = (0.14)X(10)^{0.0181x0.0323}X(10.10)^2X0.039$

ΔP =0.5577 in water/ft packing = 46.44 mm Hg water/m packing

The height of tower is not known therefore total pressure drop cannot be calculated

Degree of wetting :

We have to calculate the degree of wetting rate

$L_w = L/A_g \rho_l a^1$

$L_w = 63.09/6.45 x 992.06 x 195$

$L_w = 1.24\ ft^3/hr.ft$

Thus wetting is under the specified limit and proper distribution of liquid is taking place.

Tower height calculation

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

$H_{OG} = H_G + m.(G_m/L_m).H_l$

$N_{OG} = \int (1-Y)_{cm} / (1-Y)(Y-Y^*)$

$N_{OG} = 1/2(\ln(1-Y_t)/(1-Y_b)) + \int dy / Y-Y^*$

Calculation of H_G

$H_G = (0.029 * \Psi\ D_c^{1.11} Z^{0.33}\ SC_g^{0.5})/(L\ f_1\ f_2\ f_3)^{0.5}$-----------(perry)

Sc_G = gas-phase Schimidt number (dimensionless number)

$= \mu_G/\rho_g D_g$

D =column diameter, m

Z = packed height, m

L =liquid rate, $kg/s.m^2$

ψ =Correlation parameter

$f_1 = (\mu_L/\mu_w)^{0.16}$, with $\mu_W = 1.0$ mPa.s

$f_2 = (\rho_w/\rho_L)^{1.25}$ with $\rho_w = 1000\ kg/m^3$

$f_3 = (\sigma_w/\sigma_l)^{0.8}$, with $\sigma_w = 72.8$ mN/m

From calculation we get

f_1= 1.0016 f_3 = 1.06

f_2 = 1.01 μ_l =.09 cP

also for %flood = 85 we have ψ = 65

and D_{mix}=0.640 cm^2/sec (calculated by assuming a binary mix of CO_2 and H_2)

Sc_G =0.2496

Substituting in above equation we get

$H_G = (Z)^{0.33}$ x 0.977

Calculation of H_L

$H_L =(\phi C/3.28)X(\mu_L/\rho_L D_L)^{0.5}X(Z/3.05)^{0.15}$

ϕ = Correlation parameter for given packing, m

C = correlation factor for high gas rates (fig 18-59)

μ_L =liquid viscosity , Pa.s

ρ_L =Liquid density , kg/m^3

D_L =liquid –diffusion coefficient, m^2/s

Z =height of packing,m

From property calculation we get :

ϕ = 0.023 μ_L =0.90 cP

ρ_L =992.06 C = 0.48 (from fig 18-59, perry)

we get

$H_L = 0.061(Z)^{0.15}$

Calculation of N_{OG}

We have taken

$L_m/G_m = 0.6375$

Also we have

$N_{OG} = \int dy/(Y-Y^*) - 1/2(\ln(1+Y_b/1+Y_t)$

Now from X –Y Graph we get Y*

Which is tabulated here

Y	**Y^***	**$1/(Y-Y^*)$**
0.51	**0.065**	**2.24**
0.35	**0.0425**	**3.25**
0.25	**0.025**	**4.44**
0.23	**0.020**	**4.76**
0.2	**0.0175**	**5.48**
0.10	**0.005**	**10.53**
0.014	**0**	**71.42**

Now from graph we get

$N_{OG} = 4.75$

Also

$H_{OG} = H_G + m(G_m/L_m)H_L$

Where

m = slope of linear equlibrium relationship therefore

$H_{OG} =\{ (Z)^{0.33}(0.977) +(0.05)(Z)^{0.15}\}$

Also we get

$Z = (4.75)\{ (Z)^{0.33}(0.977) +(0.05)(Z)^{0.15}\}$

Calculating above equation we get

Z =10.4 m

Height of Absorber = 10.4 m

MECHANICAL DESIGN OF ABSORBER

Material for shell is **Carbon Steel**

THICKNESS OF SHELL

Thickness of shell = t_s

t_s = [p D / 2f J – p] + c

Where,

Inner Diameter of vessel = D_i = 2.866 m

Working Pressure = 1.013 $*10^5$ N/m^2

Design Pressure = p = 1.10 * 1.013 $x10^5$ N/ m^2

Permissible Stress = 95 $*10^5$ N/m^2 =95 N/mm^2

J= Joint Efficiency = 0.85

Corrosion allowance =2mm

Hence,

t_s =3.95mm

We take thickness as **8mm**(Including corrosion allowance)

So outer diameter of shell Do = 2.866 m + 2 x 0.008m = 3.02 m

Axial Stress Due to Pressure

Axial stress due to pressure =f_{ap}

f_{AP} = P* D_i / 4* (t_S – c)

= $1.1*1.013*10^5*2.866/4*(8-2)$

= $13.31*10^6$ N/m^2

Stress due to Dead Load

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

$D_o = D_i + 2\ t_s$

$=2.86 + 2*2*10^{-3} = 3.02$ m

Density of Shell material = ρ_s = 7700 kg /m3

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

$=7.7*10^3*$ X N/m^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 =575 kg / m^3(from bhattacharya)

Mean diameter of vessel = D_m

For large diameter column

$D_{ins} = D_o$

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

$= (100*10^{-3}*575*3.02)*\mathbf{X}/(6*10^{-3}*2.94)$

$= 9.844*10^3 \mathbf{X}$ N /m^2

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

Density of liquid =ρ_l = 1000 kg/m^3

$f_{dliq} = [(\pi /4) D_i^2 X \rho_l] / \pi D_m (t_s - c)$

= 14.868 x10 5 N/m^2

d) Compressive stress due to attachment

i. Packing weight

ii. Head weight

iii. Ladder

Density of packing (Intalox saddles ceramic) =670 kg /m^3

Packing Weight = $(\pi /4) D_i^2 X* \rho_p*9.81$

$= (\pi/4)*(2.866)^2*670*9.81 X$

=42401.97 X N

Head weight (approximately) = 35000 N

Weight of Ladder = 1600 X N ----------------(4)

Total compressive stress due to attachments f_d is given by

$f_{d(attachments)}$ = (Packing Weight + Head weight + Ladder) / [$\pi D_i (t_s - c)$]

= $(44001.97X + 35000)/(\pi*2.86*6*10^{-3})$ N

=816215.7379 X +649233.45 N

Stress due to Wind

Stress due to wind is given by

$f_{wx} = M_w / Z$

Where,

Bendnig Moment = $M_w = (0.7* p_w* D_o* X^2)/2$

$Z = (\pi /4) *D_o^2 *(t_s - c)$

Pressure due to wind = $p_w = 0.05 \times v_w^2$

Considering velocity of wind be 100 Mph(assumed)

V_w =44.7m/s

p_w = 1197 N /m^2(from table)

$f_{wx} = 1.4* p_w *X^2 /\pi D_o (t_s - c)$

$= 29.44*10^3*X^2$

To determine the value of X

$f_{t\,max} = 95 *10^6\ N/m^2$

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

$95 * 10^6 (0.85) = (29.44*10^3X^2 +13.31*10^6) - (816215.74X +649233.45)$

X = 127.82 m

DESIGN OF GASKET AND BOLT SIZE

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 =6.25 mm

Inner diameter = D_i = 2.866 m

Outer diameter = D_o = 2.866m + 2 x 8 x 10 –3

Flange inner diameter =D_{fi} = 3.02 m

Flange outer diameter = D_{fo} = 3.08 m

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

= 3.05 m

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

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

$=\pi * 6.25*10^{-3}*3.05*11.2*10^{6}$ N

$= 670.73*10^{3}$ N

Design pressure = P = $1.013*10^{5}*1.10$ (10% of allowance is given)

$= 1.11*10^{5}$ 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

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

$= [\pi*2*6.25*10^{-3}*3.05*2 + \pi*(3.05)^{2}]1.11*10^{5}$

$= 8.375*10^{5}$ 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 x 10 6 N/m2

f_b = 54.5 x 10 6 N /m2

A_m is the area of bolt

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

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

$A_{m1} = 670.73*10^3/58.7*10^5$

$= 0.011\ m^2$

$A_{m2} = 0.0154\ m^2$

Number of bolts = mean diameter $/b_o$ x 2.5

=244 bolts

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

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

$=[(0.0154/244)*(4/\pi)]^{1/2}$

=0.90 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 = $H = (\pi /4)\ G^2\ P$

$=(\pi/4)*(3.05)^2*1.11*10^5$ N

$=0.78*10^6$ N

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

$h_G = (\ B - G\ )/\ 2$

$= 3.11-3.05/2$

$= 0.03m$

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

$= 3.08 +2*0.90*10^{-2} +12*10^{-3}$

$= 3.11$ m

$W_m = 8.375*10^5$ N

$K= 1/(0.3+(1.5*8.375*10^5*0.03/0.78*10^6*3.05))$

K= 3.166

Hence the thickness of flange = 59.93 mm

HEAD DESIGN: FLANGED & SHALLOW

Material stainless steel

Permissible stress = f= 130 N/mm^2

Design pressure = p = 1.064 x 10 5 N/m^2

Stress identification factor W is given by

$W = (¼) [3 + (R_c/R_1)^{1/2}]$

Crown Radius = R_c= 2.866m

Knuckle radius = R_1 = 0.172 m

So,

Stress identification factor W is 1.77

Thickness of head = t_h = (p x R_c W)/(2f)

t_h= 2.07 mm

So , we can take thickness of head as that of thickness of shell

NOZZLE THICKNESS

Material **Carbon steel**

Considering diameter of nozzle = D_n =0.5 m

Thickness of nozzle =t_n

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

Permissible stress =130 *10^6 N/m^2

J=0.85

t_n=P*D_n /(2 f x J –P)

t_n = 0.24

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 =2.866 m

Outer Diameter of the vessel = D_o =3.02 m

Height of the vessel = H = 10.4 m

Density of carbon steel = ρ_s = 7700 kg /m^3

Density of water = ρ_l =1000 kg /m^3

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

$= (\pi/4)\ (D_o^2 - D_i^2) * H\ \rho_s * 9.81 + (\pi/4)\ D_i^2 * H * \rho_L * 0.8*9.81 + (\pi/4)\ D_i^2 * H * \rho_p*9.81 + 35000N + 1600 * H$

$= 3.133*10^6$ N

Diameter of Skirt is 2.866 m

Considering the height of Skirt is 4m

Wind Pressure is 1197N/m^2

Stress due to Dead Weight

Thickness of the skirt support is t_{sk}

Stress due to dead load

f_d = Total Weight $/\pi\ D_s\ t_{sk}$

$= 3.133*10^6/\pi*2.866*\ t_{sk}$ N/m^2

$= 3.48*10^5/t_{sk}$ N/m^2

Due to wind load

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

$p_{lw} = k\ p_1\ h_1\ D_o$ (for Height less then 20m)

for Height less then 20m

Where K is coefficient depending on the shape factor.

k=0.7 for cylindrical surface

P is wind pressure for the vessel.

P_1 = 700 N /m^2 { 40-100 Kg/m^2}

H=10.4m

$p_{lw} = k\ P_1\ h_1\ D_o$

= 15389.92

Bending moment due to wind at the base of the vessel is determined by

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

$=80027.584$ N- m

$f_{wb} = 4 \times M_w / \pi D_o\ t_{sk}$

$=4*80027.584 / \pi*2.866*t_{sk}$

$=35552.78/t_{sk}$

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

$= 2.69*10^5/t_{sk}$ N

Maximum Compressive Stress

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

$= (77847.09 / t_{sk})$ N /m^2

Yield point = 200 N / mm^2

1/3) Yield point $\geq f_c$ permissible

$= 66.6$ N/mm^2

$t_{sk} = 9.26$ mm

Process Design of Heat exchanger

Heat exchanger used is **shell and tube**. In these exchanger synthesis gas is coming from the cooler of CO conversion unit. In exchanger the temperature of gas mixture is reduced from 250^oC to 25^oC . Cold water is available at 20^0C.

Shell side:

Feed is the mixture of gas

We have

H_2 =0.94665 kg/sec

CO_2 = 11.066 kg/sec

CO = 0.33 kg/sec

CH_4 = 0.05kg/sec

N_2 = 0.081 kg/sec

Therefore from above

Mass flow rate of gas is given by

M_g = 12.47 kg/sec

Inlet temperature (T1)= 250^0C

Outlet temperature(T2)= 25^0C

Tube side :

Inlet temperature (t1)= 20^0C

Outlet temperature(t2)= 40^0C

Heat balance

Heat supplied by gas is given by

$Q_h = m_h \mathbf{C_p} * (T_2 - T_1)$

Therefore we have

= (0.94665*14.644+ 11.066*0.96 + 1.088*0.33 +0.05*3.01 +1.008)(250-25)

= 5643.84 KW/h

At steady state.

$Q_h = Q_c = m_c C_P (t_2 - t_1)$

5643.84=mc*4.18*(30-20)

m_c=67.44 kg/sec

Mass flow rate of gases is 12.7 kg/sec

Mass flow rate of cold water is 67.44 kg/sec

LMTD

LMTD = 54.85^{o}C

We have

$R = T_1 - T_2/t_2 - t_1$

= 225/20 =11.25

$S = t_2 - t_1/T_1 - t_1$ =0.08

F_T=LMTD correction factor.

From graph of F_T Vs S

F_T =0.95

LMTD(corrected)=0.95*54.85=52.11^{0}C

Heat transfer area:

We have U range from

U = 10 –50 Btu/^{o}Fft^{2}hr

Choose overall heat transfer coefficient = 283.9 W/(m^{2}K) =50 Btu/Fft^{2}hr

Q = UA(LMTD)

A=5643.85*10^3/52.86*0.95*5.678*50

A=388.68m^2

Tube selection

Let us choose

¾ in OD ,10 BWG Tubes

OD=3/4 in=19.05 mm

ID=0.62 in=15.75 mm

Length of tube =L=16ft=4.88m

Heat transfer area per tube =0.0598 m^2/m length

Heat transfer of one tube = 0.2892

Number of tubes = 388.68/0.2892 = 1344

Let us choose 1-4 pass and U type Heat exchanger

We have

Nearest tube count from tube count table

N_T= 1378

¾ in tubes arranged in triangular pitch

shell ID(D_f)=1067mm=42in

Corrected heat transfer area=1378*0.2892 m^2

=401.41 m^2

Corrected over all heat transfer coefficient (U)=274.90 W/(m^{2}K)

Average properties of fluids

a) shell side (gas mixture) at 137.5^0C

$\rho=0.51\ kg/m^3$

$\mu=0.014*10^{-3}\ Ns/m^2$

Cp=2.008KJ/kg.K

k=0.149 w/m.k

b)tube side (water) at 30^0C

$\rho=995.647$ kg/m3

$\mu=8.5*10^{-4}Ns/m^2$

Cp=4.18 KJ/kg.K

k=0.621w/m.k

Tube side velocity

Number of passes $N_P=6$

Flow area $=(\prod*ID^2/4)*N_T/N_P$

$=(3.14*0.01575^2/4)*1388/4$

$A_a\ =0.068\ m^2$

$V_t=m_c/\ (A_a\ \rho)$

$=67.44/(0.068*995.647)$

$=0.996$ m/s

Velocity is within the range.

Shell side velocity

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

P^l =pitch=25.4 mm

$L_S=D_s$

$=[(25.4-19.05)*1067]*(1067/\ 25.4)$

$=0.285\ m^2$

$V_s = m_h/(\rho\ S_m)$

$= 12.70/(0.51*0.285)$

$= 89.12$ m/s

$N_b + 1 = L/L_S$

$= 4.83/1.067 = 5$ baffles

7) Shell side heat transfer coefficient

$N_{NU} = j_H\ N_{re}(N_{Pr})^{1/3}$

Where

N_{Nu} = Nusselt number

N_{Re}=Reynolds number

$N_{Re} = D_e G_s/\mu$

$G_s = 1886.85$

$D_e = 4*((P_t)^2*0.86/2 - \prod d^2/4)/(0.5*\prod*d_o)$

$D_e = 4.5*10^{-3}$ m

$N_{Re} = 60635.33$

N_{Pr}=Prandtl number

$= 0.74$

$j_h = 3*10^{-3}$

$N_{NU} = 10^{-2}*60635.33*(0.74)$

$= 164.53$

$h_o = 110.8*0.149/0.01905$

$= 1286.88$ w/m^2.K

Tube side heat transfer coefficient

$N_{Nu}=0.023(N_{Re})^{0.8}\ (N_{Pr})^{0.3}$

$N_{Re}=18374.96$

$N_{Pr}=5.73$

$N_{Nu}=0.023(18374.96)^{0.8}\ (5.73)^{0.3}$

$=106.13$

$h_i=4184.55\ w/m^2.K$

9) overall heat transfer coefficient

$h_o=1286.88\ w/m^2.K$

$h_i=4184.55\ w/m^2.K$

Also overall dirt factor is assumed to 0.005 $hft^2\ {}^oF/Btu$

$1/U_C = 1/h_o + 1/h_i*(D_o/D_i)$

$1/U_C = 1/1286.88 + (1/4184.55)*(19.05/15.75)$

$U_C = 906.79\ w/m^2k$

Also we have Heat transfer including dirt factor is given by

$1/U_d = 1/U_C + R_d$

$U_d = 1/906.79 + 0.005*0.1761$

U_d = 504.21 w/m^{2}k

Assumed value and design values are almost same.

Pressure drop calculation

a) Tube side pressure drop

Tube side Reynolds number=N_{Re}= 18374.96

friction factor=f=$0.079(N_{Re})^{-1/4}$

$=0.079(18374.96)^{-1/4}$

=0.0068

$\Delta P_L=(4fLv_t^2/2gD_i)*\rho_t g$

$= (4*0.0068*4.88*(0.996)^2/2*9.8*15.75*10^{-3})*999*9.81$

= 4162.0 N/m^2 =4.162 KN/m^2

$\Delta P_E= 2.5(\rho_t v_t^2/2)$

$= 2.5(995.647*(0.996)^2/2)$

$= 1.23*10^3$ N/m^2

= 1.23 KN/m^2

$(\Delta P)_T = N_p*(\Delta P_L+\Delta P_E)$

= 4*(4.162 +1.23)

= 21.56 KN/m^2

b) Shell side pressure drop (Kern's method)

In the calculation for the heating or cooling gas differs in only minor respects from the calculation for liquid –liquid system . The relationship between gas film gas film coefficients and allowable pressure drops are critically dependent upon the operating pressure of the system where as for incompressible fluids the operating pressure of the system .

Because of this reason KERN's method is used for the pressure drop calculation.

a_s =shell side flow area = (I.D) C^1*B/P_T

where

C^1=clearance =(15/16-3/4)inch =0.187=4.76mm

B=Baffle spacing =D_s

P_T =15/16inch =23.8 mm

From above we get

a_s = 0.2846 m2

also G_s =12.47/0.2846 =43.82 kg/m^2.sec

From above shell side reynold's number is calculated

Which is

Shell side Reynolds number = 60635.33

Also $f=1.87*(14085)^{-0.2}$

f =0.28

Also

Number of baffles = L/B =4.83/1.067 =5

$\Delta P_s = [4*f*(N_b +1)*D_s*G_s^2]/[2*g*D_c*\rho_g]$

$\Delta P_s = [4*0.28*6*1.067*(43.82)^2*9.81]/[2*9.81*4.5*10^{-3}*0.51]$

$\Delta P_s = 20.53\ kN/m^2$

Which is under limit so that we can proceed with our plant design .

MECHANICAL DESIGN OF SHELL AND TUBE HEAT EXCHANGER

Carbon Steel (Corrosion allowance 3 mm)

SHELL SIDE

Number of shell =1

Number of pass =4

Fluids in shell are Hydrogen, Carbon dioxide, carbon monooxide, Methane and Nitrogen

Design pressure =1.064 x10^5 N /m^2 =0.11N/mm^2

Temperature of inlet =250^{o}C

Temperature of outlet = 25^{o}C

Permissible Stress for carbon steel (f)= 95 N/mm^2

Segmental baffle cut with tie rods and spacers

TUBE SIDE

Tube and sheet material --------- Stainless steel

No. of tubes =1388

Outside Diameter =19.05mm

Inside Diameter =15.75mm

Length =4.88m

Pitch Δ^{lr} =1inch

Fluid = Water

Working pressure =1atm =0.1N/mm^2

Design pressure =0.11N/mm^2

Inlet temperature = 20^{0}C

Outlet temperature =40^{0}C

SHELL SIDE:

SHELL SIDE DIAMETER

Shell Diameter =1067mm

<u>Shell thickness</u>

t_s= pd/2fj +p

where j=85%

= 0.11*1067/(2*95*0.85 +0.11)

=0.73mm

From IS-4503 Table (4) gives a minnimum thickness of 6.3 including corrosion allowance .Use 10.0 mm thickness.

HEAD THICKNESS

Consider Shallow dished and torespherical head

t_h=P*R_c*W/(2*f*j)

Where j=0.85

R_c = Crown radius

W =stress intensification factor

R_K = knuckle radius

R_K = 6%R_c

W=1/4*(3+ $(R_C/R_K)^{1/2}$) = 1.77

t_h= 0.11*1067*1.77/2*95

== 1.09mm

Use thickness same as for shell , i.e. 10mm including corrosion allowance.

<u>TRANSVERSE BAFFLES</u>

Spacing baffles = D_S = 1067mm

Number of baffles = 4.88/1.067 =5

But in process design Number of baffles is assumed to be 3

So that pressure drops comes under the given limit

Thickness of baffles =6mm

TIE RODS AND SPACERS

Tie Rods and Spacers shall be provided to retain all cross baffles and tube support heater accurately.

Diameter of rod =15mm

No. of the rod =6mm

Following is from bhattacharya

FLANGES

Design Pressure =0.11MN/m^2

Flange material =IS:2004 ------1962 class2

Bolting Steel = 5% Cr Mo Steel

Gasket material = Asbestos composition

Shell diameter =1067mm

Shell thickness=10mm

Outside diameter = 1087mm

Allowable stress of flange material =100MN/m^2

Allowable stress for bolting material =138 MN/m^2

d_O/d_i =(y-Pm)/(y-pm)

where

m =gasket factor

y= min design seating stress MN/m^2

assuming gasket thickness of 1.6mm

y=25.5

m= 2.75…………… from IS 2825-1969

$d_o/d_i = [(25.5-0.11*2.75)/(25.5-0.11(2.75+1)]^{1/2}$

d_o/d_i =1.002

let d_I of the gasket equal to 1097mm , 10mm greater than Shell diameter.

Therefore

$d_o = 1.002 * d_i$

$= 1.002 * 1097$

$= 1099 mm$

Minimum gasket width $= d_o - d_i / 2 = (1099.1 - 1097) * 10^{-3} / 2$

$= 0.001 m = 1 mm$

Taking Gasket width = 0.012m

Diameter of location of gasket load reaction is

$G = d_i + N$

$= 1.097 + 0.010$

$= 1.107$ m

Estimation of bolt loads

Load due to design pressure

$H = \prod * G^2 * P / 4$

$= \prod (1.107)^2 * 0.11 / 4$

$= 0.106$ MN

Load to keep joint tight under operation

$H_P = \prod * G * (2 * b) * m * p$

$= \prod * (1.107)^2 * 2 * 5 * 10^{-3} * 2.75 * 0.11$

$= 0.012$ MN

Total operating load :

$W_d = H + H_P$

=0.106 + 0.012

=0.118 MN

load to seat gasket under bolting up condition

$W_g = \prod * G * b * y$

$= \prod * 1.107 * 0.005 * 25.5$

= 0.44 MN

Also **$W_g > W_d$**

Therefore

Controlling load =0.44MN

Minimum bolting area $= A_m = W_g/S_g$

= 0.44/138

$= 3.19*10^{-3}\ m^2$

S_g =138 from bhattacharya pg no.10

Calculation for optimum bolt size

Let us choose Bolt as M 18X12

Min no. Of bolts = 44

Also

R= 0.027m

We have

$G^1 = B + 2(g_1+R)$

$G^1 = 1.087 + 2[g_1 + R]$

$g_1 = g_o/0.707 = 1.415 g_o$ for weld leg

$G^1 = 1.087 + 2 (1.415*10*10^{-3} + 0.027)$

$G^1 = 1.424$

Using 75 mm bolt spacing

$C^1 = 44*0.075/\prod = 1.05m$

From the above calculation the minimum bolt circle is 1.424 m when M18 Bolt

44 bolts of 18 mm diameter on 1.424 m bolt circle are specified.

Bolt circle diameter =1.42 m

A = C +Bolt diameter +0.02

= 1.42 +0.018 + 0.02

= 1.458m = 1.46m

Check of gasket width

$= A_b*S_g/\prod*G*N$

$= 44*138*1.54*10^{-3}/\prod*1.107*0.01$

= 26.88

also 26.88<2*y

Flange moment computation

$W_0 = W_1 + W_2 + W_3$

$W_1 = \prod *B^2*P/4$

Hydrostatic end force on area inside of flange .

$W_1 = \prod*(1.087)^2*0.11/4$

= 0.102 MN

$W_2 = H - W_1$

= 0.106 – 0.102 =0.004 MN

$W_S = W_o – H = H_P$=gasket load

= 0.012MN

$M_o = W_1a_1 + W_2a_2 + W_3a_3$

Where M_o = Total Flange moment

$a_1 = C – B/2 = 1.42 – 1.0187/2 = 0.17$

$a_3 = C^1 – C_1/2 = 1.424 – 1.107/2 = 0.1585$

$a_2 = a_1 + a_3/2 = 0.164m$

$M_o = 0.102\ 0.17 + 0.004*0.164 + 0.012*0.164$

=0.020 MN-m

For bolting up condition

$M_g = Wa_3$

$W = A_m + A_b/2 * S_g$

$A_b = 44*1.54*10^{-4}\ m^2$

$S_g = 138\ MN/m^2$

$A_m = 3.19*10^{-3}\ m^2$

$A_m + A_b/2 = 4.983*10^{-3}\ m^2$

W = 0.687

$M_g = 0.108$

$M_g > M_o$.

Hence moment under operating condition

M_g is controlling

Therefore

$M_g = M$

Calculation of flange thickness

$t^2 = M*C_f*Y/B*S_T = M*C_f*Y/B*S_{fo}$

where

k= A/B =1.46/1.087 =1.34

Assuming $C_f = 1$

Y= 6 from graph

$t^2 = 0.108*1*6/1.087*100$

t = 0.0792 m

Actual bolt spacing

$B_S = \prod*C/n = \prod*1.42/44 = 0.101$

Bolt correction factor

$C_f = (B_S/2*d + t)^{1/2}$

$C_f = [(0.101)/(2*0.018 + 0.101)]^{1/2}$

$C_f = (0.737)^{1/2} = 0.858$

Actual Bolt thickness = $(C)^{1/2} *t$

$= 0.93* 0.0792$

$= 0.0796 = 73.66$ mm = 75mm

Tube Sheet Thickness

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

$t_{ts} = 1*1.107* (0.25*0.11/95)^{1/2}$

$= 0.0188$m

$t_{ts} = 18.83 +3 = 22$ mm (Includes corrosion allowance)

Channel and Channel cover

$T_h = G_C*(K*P/f)^{1/2}$

$= 1.107*(0.3*0.11/95)^{1/2}$

$= 0.0206$ m = 22mm (includes corrosion allowance)

Saddle Support

Material : **low carbon steel**

Vessel diameter = 1087mm

Length of shell = 4.88 m

Knuckle Radius = 6*1087/100 = 65.22mm

Total depth of Head = $(D_o*R_o/2)^{1/2}$

=(1087*65.22/2)

= 188.27 mm

R_i= 0.838 m

r_i =0.1 x0.838

Inside depth of head can be calculated as

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

= 0.136m

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 0.5m

Permissible stress = f = 95 x 10 6 N/ m2

Corrosion allowance = 3mm

$t_n = p D_n/(2f J - p)]+c$

=3.26 mm