Process design of Absorber

The gases entering the absorber per day are :

Carbon disulfide, CS_2 = 459.92 Kmols

Hydrogen disulfide, H_2S = 1314.06 Kmols

Methane, CH_4 = 73.0 Kmols

Total Kmols of gases entering = 1846.98 Kmols

Gases leaving the absorber from the top are :

Hydrogen disulfide, H_2S = 1307.49 Kmols

Methane, CH_4 = 73.0 Kmols

Carbon disulfide, CS_2 = 2.30 Kmols

For liquid phase,

At top of tower,

X_t= mols of carbon disulfide/mols of oil

At bottom of tower,

X_b= mols of carbon disulfide/mols of oil

For Gas phase,

At top of tower,

Y_t = mols of carbon disulfide / mols of inert gases

At bottom of tower,

Y_b = mols of carbon disulfide / mols of inert gases

We take pure oil at the inlet

hence,

$X_t = 0$

We have to find X_b

$Y_t = 2.3/(1307.49 + 73.0)$

$= 0.001666$

$Y_b = 459.92/(1314.06 + 73.0)$

$= 0.33158$

Taking the heat of solution of carbon disulfide in hydrocarbon oil as negligible, the equilibrium curve is plotted by :

X :	0.00989	0.0909	0.2307	0.333	0.412
Y :	0.0052	0.0538	0.222	0.475	0.866

From the equilibrium curve, $X^*_b = 0.280$

$(L_m'/G_m')_{min} = (Y_b - Y_t)/(X^*_b - X_b) = 1.20$

Taking (L_m' / G_m') as 1.2 times that of $(L_m'/G_m')_{min}$

Hence, $(L_m' / G_m') = 1.44$

$L_m' = 1.44 \times G_m'$

$G_m' = 1387.06$ Kmols

Hence, $L_m' = 1.44 \text{ X } 1387.06 = 1997.32$ Kmols

And $X_b = 0.235$

$G_{mb} = G_m' (1 + Y_b)$

$= 1387.06 \text{ X } (1+ 0.33157)$

$= 1846.96$ Kmols

$L_{mb} = L_m' (1 + X_b)$

$= 1997.3 \text{ X } (1 + 0.235)$

$= 2466.74$ Kmols

$G_{mt} = G_m' (1 + Y_t)$

$= 1387.06 \text{ X } (1 + 0.001666)$

$= 1389.37$ Kmols

$L_{mt} = L_m'(1 + X_t)$

$= 1997.36$ Kmols

Average molecular weights :

$(M_G)_{Top} = (73.0/1382.8) \times 16.0 + (1307.49/1382.8) \times 34.05$
$+ (2.30/1382.8) \times 76.1$

$= 33.167$ Kg/Kmol

$(M_G)_{Bottom} = (459.92/1846.98) \times 76.1 + (1314.06/1846.98) \times 34.05$
$+ (73.0/1846.98) \times 16.0$

$= 43.80$ Kg/Kmol

Density of gases = PM_{avg}/RT

The column is operating at 40ºC and 1 atm

Density of gases :

$(\rho_g)_{top} = (PM_{avg}/RT)_{top} = [(1.013 \times 10^5 \times 33.8167)/(8314 \times 313)]$

$= 1.2911$ kg /m^3

$(\rho_g)_{bottom} = (PM_{avg}/RT)_{bottom} = [(1.013 \times 10^5 \times 43.80)/(8314 \times 313)]$

$= 1.705$ kg /m^3

$G_b = 1846.96 \times 43.80$

$= 80,896.84$ Kgs

$G_t = 1389.31 \times 33.167$

$= 46081.23$ Kgs

$L_b = 2466.96 \times 160.26$

$= 3,95,319.75$ Kgs

$L_t = 1997.36 \times 180.0 = 3,59,524.80$ Kgs

$[(L_b / G_b)(\rho_g / \rho_l)^{1/2}]_{top} = 7.8019/(1.2911/810)^{1/2}$

$= 0.3115$

$[(L_b / G_b)(\rho_g / \rho_l)^{1/2}]_{bottom} = 4.8867/(1.705/160.26)^{1/2}$

$= 0.2242$

Since, the ratio is more at the top of the tower, we will consider the flooding conditions at the top of the tower.

For 80% flooding conditions and $[(L_b / G_b)(\rho_g / \rho_l)^{1/2}] = 0.3115$

The value of $(G_f^2 F_p \psi \mu^{0.2}) / (\rho_g \rho_l g) = 0.044$

Where, $\psi = (\rho_{water}/\rho_l) = 1.2345$

μ is viscosity of liquid = 1.7 cP

Choosing 75mm Ceramic Intalox saddles as packing material.

Packing Factor, $F_p = 70\ m^{-1}$

Approximate surface area, a = 92 (m^2/m^3)

Percent void space, ε = 80%

Nominal Size, d_p = 75 mm

Hence, $G_f = 2.1674\ Kg/m^2\text{-s}$

And, $G = 0.8 \times G_f = 1.73393\ Kg/m^2\text{-s}$

Cross sectional area of the absorber, A_c :

$A_c = G_b / G = 0.30759\ m^2$

Hence, the diameter of the column, D_c :

$D_c = 0.630$ m

Check : $D_c/d_p = 0.63/0.075 = 8.4$,

This is less than the required minimum ratio of 10.

This packing is not suitable for the column.

Choosing 25 mm Ceramic Intalox saddles as packing material.

Packing Factor, F_p = 320 m^{-1}

Approximate surface area, a = 255 (m^2/m^3)

Percent void space, ε = 77%

Nominal Size, d_p = 25 mm

We get G_f = 1.013169 Kg/m^2-s

G = 0.81095 Kg/m^2-s

A_c = 0.65768 m^2

D_c = 0.9150 m

Check : D_c/d_p = 36.60, which is greater than the required minimum ratio of 10.

Hence, the packing material is suitable.

Wetting rate calculation :

Wetting Rate = L_w = L'/(A_c ρ_l a')

L_w = [359524.8/(3600 X 24)]/[0.65768 X 810 X 255]

= 3.063 X 10^{-5} m^3/m-s

= 1.18696 ft^3/ft-hr

As, this wetting rate is greater than the required minimum of 0.85 ft^3/ft-hr
the wetting rate is acceptable.

Pressure Drop calculation :

$\Delta P = C_2 \times 10^{(C3Utl)} \times \rho_g \times U_{tg}^2$

Where, u_{tl} : superficial liquid velocity

And, U_{tg} : superficial gas velocity

L = 6.327 Kg/m^2-s = 4655.40 Lb/ft^2-hr

For this value of L,
And for saddles of 1" : $c_2 = 0.31$, $c_3 = 0.0222$

Also, the range of L : 2,520 – 14,440 Lb/ft^2-hr

The calculated L lies safely in this range.

G = 0.810953 Kg/m^2-s = 596.707 Lb/ft^2-hr

Hence, the superficial liquid and gas velocities :

U_{tl} = 7.811 X 10 m/s = 0.029744 m/s

U_{tg} = 0.6281 m/s = 2.607 ft/s

ΔP = 0.0678 in water/ft packing

= 5.6499 mm water/m packing

Tower Height Calculations :

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

Where,

Z : height of tower

H_{OG} : height of overall gas transfer unit

N_{OG} : number of overall gas transfer unit

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

Cornell's Method :

The generalized equation of Cornell et al, for gas phase transfer units, H_G

$H_G = (0.029\, \psi\, D^{1.11}\, Z^{0.33}\, Scg^{0.5}) / (L\, f_1\, f_2\, f_3)^{0.5}$

ψ obtained from graph for 80% flooding conditions = 50 m

$Sc_g = \mu_g / \rho_g D_g$

$f_1 = (\mu_l/\mu_w) = 1.00886$

$f_2 = (\rho_l/\rho_w) = 1.30134$

f_3 = (surface tension of the liquid/surface tension of water) = 0.9379

D_g : diffusion coefficient for mixture of gases

For multi component gas phase diffusivity,

Blanc's law :

$D_{i,mix} = 1/[\ \Sigma\ (x_j/D_{ij})]$

$D_{AB} = [10^3\ T^{1.75}\{(M_A+M_B)/(M_AM_B)\}^{1/2}]/[P\{(\ \Sigma V_A)^{1/3}+(\Sigma V_B)^{1/3}\}^2]$

$D_{CS2\text{-}H2S} = 0.1153\ cm^2/s$

$D_{CS2\text{-}CH4} = 0.1471\ cm^2/s$

Hence, $D_{g\ mix} = 0.1169\ cm^2/s$

$Sc_g = 0.89445$

$H_G = [(0.029)(50)0.915^{1.11}\ Z^{0.33}\ 0.98445^{0.5}]/[6.327(1.00886)(1.301348)(0.9379\ ]^{0.5}$

$H_G = 0.44518\ X\ Z^{0.33}$

The generalized equation of Cornell et al, for liquid phase transfer units, H_L

$H_L = (\Phi\ C\ /\ 3.28)\ (\mu_l\ /\rho_l\ D_l\ )^{½}\ (\ Z\ /3.05\ )^{0.15}$

Where, C : correction factor

Φ : correlation parameter = 0.06, for L = 4655.4 Lb/ft^2-hr

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

Wilke Chang equation :

$D_{AB} = [7.4\ x10^{-8}\ (\phi\ M_B)^{½}\ T]\ /\mu_l\ V_A$

μ_l = 1.7 cP

$D_L = 1.534\ X\ 10^5\ cm^2/s$

$H_L = 0.10858\ X\ Z^{0.15}$

$H_{OG} = 0.44518\ Z^{0.33} + 0.10858\ Z^{0.15}\ mG_m/L_m$

N_{OG} Calculations :

$$N_{OG} = \int_{Y_t}^{Y_b} [dY / (Y-Y^*)] - [½ \ln \{(1+ Y_b) / (1+ Y_t)\}]$$

Y	Y*	1/(Y – Y*)
0.05	0.0425	133.3
0.10	0.085	66.6
0.15	0.125	40.0
0.20	0.1675	30.7
0.25	0.21	25.0
0.30	0.25	20.0
0.33157	0.2775	18.5
0.35	0.290	16.7

Plotting a graph of 1/(Y – Y*) Vs Y,

Area under the curve from Y_t to Y_b = $\int_{Y_t}^{Y_b} [dY / (Y-Y^*)] = 9.5$

And the value of $[\ln\{(1+ Y_b)(1+ Y_t)\}/2] = 0.142$

$N_{OG} = 9.5 - 0.142 = 9.358$

$(G_m / L_m)_t = 1389.37/1997.36 = 0.6956$

$(mG_m / L_m) = 0.83472$

$(G_m / L_m)_b = 0.75486$

$(mG_m / L_m)_b = 0.9058$

$(mG_m / L_m)_{avg} = 0.87028$

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

$\quad = [H_G + (m\ G_m / L_m)H_L]\ N_{OG}$

$Z = 4.16599\ Z^{0.33} + 0.88428\ Z^{0.15}$

By trial, Z is calculated to be,

Z = 10

Hence the height of the tower, Z = 10 mts

Mechanical Design of Absorber

Material used for making the shell is carbon steel.

Thickness of shell, t_s :

t_s = $[(p\ D) / (2f\ J – p)] + c$

Where,

Inner Diameter of vessel, D_i = 0.9150 m

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

Design Pressure, p = $1.05 \times 1.013 \times 10^5 = 1.063 \times 10^5\ N/m^2$

Permissible Stress, f = $95 \times 10^6\ N/m^2$

J= Joint Efficiency = 0.85

Corrosion allowance, c = 2mm

Hence, t_s = 2.6 mm

taking thickness as 8 mm

Thus, outer diameter of shell D_o = 0.915 m + 2 x 0.008 m = 0.931 m

Axial Stress Due to Pressure :

Axial stress due to pressure, f_{ap}

$f_{ap} = [(p\ D_i)/ \{4(t_s – c)\}]$

$= 4.052 \times 10^6\ N/m^2$

Stress due to Dead Load :

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

Density of Shell material = ρ_s = 7700 kg /m^3

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

$=7.7 \times 10^4\ X\ N/m^2$

b) Compressive stress due to weight of insulation at height x :

Insulator used is asbestos

Thickness of insulation t_{ins} = 3 cm

Diameter of insulation, D_{ins}

Density of insulation = 2200 kg / m^3

Mean diameter of vessel = D_m

For large diameter column

$D_{ins} = D_m$

$f_{dins} = [\pi D_{ins} t_{ins} \rho_{ins} x] / [\pi D_m (t_s - c)]$

$= 0.11 X 10^6 x N /m^2$

c) Compressive stress due to liquid in column up to height x :

Density of liquid, $\rho_l = 810 kg/m^3$

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

$= 30.3 x 10^3 N/m^2$

d) Compressive stress due to packing and attachments :

1) Packing weight
2) Head weight
3) Ladder

Density of packing (Ceramic Intalox saddles) = 705 kg/m^3

1) Packing Weight = $(\pi /4) D_i^2$ X x X density of packing X 9.81

= 4547.68 x N

2) Head weight = $[(\pi /4) D_i^2$ X x X 810 X 9.81] = 5225 x N

3) Weight of Ladder = 1600 x N

Total compressive stress due to attachments f_d is given by

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

= (263674.22 + 395713.1) x N

Stress due to Wind :

Stress due to wind is given by

$f_{wx} = M_w / Z$

Where,

$M_w = [(0.7 p_w D_o x^2)/2]$

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

Pressure due to wind = $p_w = 0.05 X v_w^2$

Considering velocity of wind to be 150 km/hr

$p_w = 1125\ N/m^2$

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

$= 89.75\ x10^3\ x^2\ N$

Determining the value of x :

$f_{tmax} = 95 \times 10^6\ N/m^2$

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

$95 X 10^6 (0.85) = 89.75\ x10^3\ x^2 + 4.052\ x10^6 - [7.7\ x10^4\ x + 0.11 X 10^6\ x + 30.3 X 10^3\ x + 263674.22\ x + 395713.1\ x]$

Gives, x = 32.1 m

As this value is greater than the design tower height, hence the shell thickness can withstand the tensile and compressive stresses.

Design of Gasket and Bolt Size calculations :

Width of gasket = N = 10 mm

Gasket material is asbestos

Gasket factor = m = 2

Minimum design seating stress = $Y_a = 11.2\ N/mm^2$

Basic gasket seating width $b_o = N/2$

$b_o = 10/2 = 5mm$

Effective gasket seating width

$b = 2.5\ (b_o)^{1/2} = 5.6$ mm

Flange inner diameter $=D_{fi} = 0.915$ m

Flange outer diameter = $D_{fo} = 0.931$ m

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

$= 0.923$ m

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

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

$= 0.16238 X 10^6\ N$

Design pressure = p

$= 1.064 \times 10^5\ N/m^2$

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

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

$= 0.07868 \times 10^6$ 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

$f_a = 58.7 \times 10^6\ N/m^2$

$f_b = 54.5 \times 10^6\ N/m^2$

A_m is the area of bolt

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

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

$A_{m1} = 0.00276\ m^2$

$A_{m2} = 0.00144\ m^2$

Number of bolts = mean diameter $/b_o \times 2.5$

= 66 bolts

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

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

= 0.729 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 = $H = (\pi /4)\ G^2\ p$

= 71192.60 N

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

$h_G = (B - G)/2$

= 0.0133

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

= 0.949 m

W_m = 0.16238 X 10^6 N

K= 2.8628

Hence the thickness of flange = 20.25 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= 0.915 m

Knuckle radius, R_1 = 0.055 m

Stress identification factor, W = 1.77

Thickness of head = $t_h = (p \times R_c W)/(2f)$

t_h= 0.91 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 = 0.25 m

Thickness of nozzle = t_n

Material is Stainless steel.

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

J = 0.85

$t_n= (p D_n) /(2 f J – p)$

t_n = 0.12 mm

No corrosion allowance , since the material is stainless steel.

Thickness = 3mm

SUPPORT FOR ABSORBER :

Material used is structural steel (IS 800)
Skirt support is used.

Inner Diameter of the vessel, D_i = 0.915 m

Outer Diameter of the vessel, D_o = 0.931

Height of the vessel, H = 10 m

Density of carbon steel, ρ_s = 7700 kg /m3

Density of oil, ρ_l = 810 kg /m^3

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

$= [(\pi/4)\ (D_o^2 - D_i^2) \times H\ \rho_s \times 9.81 + (\pi/4)\ D_i^2 \times H \times \rho_l \times 0.77 + (\pi/4)\ D_i^2 \times H \times \rho_p + 52250 + 1600 \times H\ ]$

$= 77.174 \times 10^3$ N

Diameter of Skirt is 0.915 m

Considering the height of Skirt is 2 m

Wind Pressure is 1125 N/m^2

Thickness of the skirt support is t_{sk} .

Stress due to dead load :

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

$= 26.847 \times 10^3/t_{sk}$ N/m^2

Stress due to wind load :

The forces due to wind load are determined as :

$p_w = k\ p_1\ h_1\ D_o$

Where K is coefficient depending on the shape factor.

K = 0.7 for cylindrical surface

$p_1 = 89.75 \times 10^3\ x^2$

$= 8.975 \times 10^6$ N/m^2

$p_w = k\ p_1\ h_1\ D_o$

$= 59.49 \times 10^6$ N/m^2

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

$M_w = p_{lw} \ X \ (h_1/2)$

$= 292.45 \ X \ 10^6$ N-m

$f_{wb} = (4 \ X \ M_w) / (\pi D_o \ t_{sk})$

$= 39.99 \ X \ 10^6 / t_{sk}$

Stress due to Seismic Load

Load = C W

W is total Weight of vessel = $77.174 \ X \ 10^3$ N

C is Seismic Coefficient = 0.08

$f_{sb} = (2/3)[\ C\ W\ H/(\pi\ R_{ok}^2\ t_{sk})\ ]$

where,

R_{ok} is radius of skirt

$f_{sb} = 60.4617 \ X \ 10^3/t_{sk}$ N/m^2

Maximum Stress at bottom of Skirt :

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

$= (1.121 \ x \ 10^6/t_{sk})$ N/m^2

Permissible tensile Stress for structural steel = 140 N/mm^2

$t_{sk} = 0.008$ m

Hence thickness of skirt is 8 mm

Design of Condenser

The condenser is working with the refining column

Feed gas : CS_2 = 657.03 Kmols = 2083.33 Kg/hr

Entering at 80ºC, Pressure is 1 atm.

Flow rate, m = 0.5787 Kg/s

Q' = m λ = 195.92 KW

Using water as coolant, flow rate of water required,

m_w = 4.687 Kg/s

for a counter current operation, LMTD:

ΔT_{ln} = 44.81ºC

Calculation of heat transfer area :

Assuming heat transfer coefficient, U = 500 W/m^2-K

Dirt factor = 0.003 ft^2h-K/Btu

Heat transfer Area, A :

A = 8.744 m^2

Choosing ¾" OD, 8 ft long tubes,

H.T.A = 0.14593 m^2

N_t = 60 tubes

Choosing a 1-2 Heat exchanger, TEMA P or S type 15/16" triangular pitch,

Nearest tube count = 62 tubes

Shell I.D = 254 mm

Corrected heat transfer area, A = 9.008 m^2

Corrected U = 533.874 W/m^2-K

Fluid velocities :

a) Tube side : Number of passes, $N_p = 2$

Flow area, $a_t = 6.04 \times 10^{-3}\ m^2$

Tube side velocity, $v_t = 0.85219$ m/s

For the tube side the acceptable velocity range is 1-2 m/s. Hence, this velocity is unacceptable.

Trial 2 :

Choosing TEMA P or S type for 1 shell and 4 tube passes, 1" triangular pitch.

$N_t = 62$

Shell I.D = 305 mm

$v_t = 1.704$ m/s , which is within the acceptable limits.

$a_t = 3.02 \times 10^{-3}$

b) Shell side :

D_s : Shell I.D

Cross flow area at the center of the shell, S_m :

$S_m = [(P' – D_o)\ L_s]\ D_s/P'$

$= 0.01159\ m^2$

Taking 25% baffle cut, $L_c = 76.25$ mm

$L_s = 0.5\ D_i = 152.5$ mm

$N_b + 1 = 16$

$N_b = 15$ baffles

Summary of shell and tube details :

Number of tube passes	: 4
Number of shell passes	: 1
Tube I.D	: 15.75 mm
Tube O.D	: 19.05 mm
Shell I.D	: 305 mm
Number of tubes	: 62
Tube pitch	: 1” = 25.4 mm
Length of tube	: 2.348 m
Baffle pitch	: 152 mm
Number of baffles	: 15
P_p	: 21.99 mm
L_c	: 76.25 mm
$U_{corrected}$	: 533.874 W/m^2-K

Film Transfer Coefficients :

a) Shell side (condensing vapors) :

Cooling water is heated from 30ºC to 40ºC

Wall temperature, $T_w = (1/2)[80 +(30+40)/2] = 57.5$ºC

Film temperature, $T_f = (80+57.5)/2 = 68.75$ºC

Using properties of CS_2 at this temperature,

Thermal conductivity, K = 0.15427 W/m-K

Density, $\rho = 1225$ Kg/m^3

Viscosity, $\mu = 0.28$ cP

Reynold’s number (N_{Re}) = 209.41

$h = 1.51 [(K^3 \rho^2 g)/ \mu]^{1/3} Re^{-1/3}$

$= 2246.33$ W/m^2-K

b) Tube side (water) :

Flow area, $a_t = 3.02 \times 10^{-3}$ m^2

Density, $\rho = 1000$ Kg/m^3

Viscosity, μ = 0.85 cP

Prandtl 's number (N_{Pr}) = 5.995

$h_i \, d_i / K = 0.023 \, (N_{Re})^{0.8} (N_{Pr})^{0.3}$

$= 156.51$

hence, $h_i = 4876.86$ W/m^2-K

wall resistance = $(x_w/k_w) \times [d_o/(d_i - d_o)] \times \ln(d_o/d_i)$

$= 4.02646 \times 10^{-5}$

Overall Heat Transfer Coefficient :

$U_{od} = 792.55$ W/m^2-K

Pressure Drop calculations :

a) Tube side (water) :

$F = 0.079 \, (N_{Re})^{-1/4} = 5.926 \times 10^{-3}$

$\Delta P_L = [(4 \, F v_t^2)/(2 g d_i)] \, \rho \, g$

$= 5309.53$ N/ m^2

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

$= 3629.52$ N/ m^2

$(\Delta P_T)_{total} = Np \, (\Delta P_L + \Delta P_E)$

$= 35.756$ Kpa

This value is less than the maximum value of allowable pressure drop of 70 Kpa Hence, it is acceptable.

b) Shell side :

Bell's method:

$(N_{Re}) = 209.41$,
Friction factor, $f_K = 0.4$

(i) Pressure drop in cross flow sections,

There are $(N_b – 1)$ crosses.

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

$b = 2 \times 10^{-3}$

$N_c = D_s[1 – 2(L_c/D_s)]/P_p = 7$

$S_m = [(P' – D_o)L_s]\ D_s/P' = 0.011628$ m^2

$\Delta P_c = 0.0132$ KPa

(ii) Pressure drop in the end zones,

There are two end zones - entrance and exit.

$\Delta P_e = \Delta P_c\ [1 + N_{cw}/N_c]$ KN/m^2

Number of effective cross flow zones in each window, $N_{cw:}$

$N_{cw} = 0.8\ L_c/P_p = 3$

Hence, $\Delta P_e = 0.01885$ KN/ m^2

(iii) Pressure drop in the window zones,

There are N_b zones

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

$b = 5 \times 10^4$

S_w= area for flow in the window zones (m^2)

$S_w = S_{wg} - S_{wt}$

Where, S_{wg} = gross window area (m^2)

S_{wt} = area occupied by tubes

S_{wg} = 23 sq. in = 0.01483 m^2 , for $L_c/D_s = 0.25$

$S_{wt} = (N_T/8)\ (1 - F_c) \times 3.14 \times D_o^2$ (m^2)

$= 3.04 \times 10^{-3} m^2$

hence, $S_w = 0.01179\ m^2$

$\Delta P_w = 0.003788$ KPa

Hence, the total pressure drop,

$(\Delta P_s)total = 2\Delta P_e + (N_b - 1)\ \Delta P_c + N_b\ \Delta P_w$

$= 0.2963$ Kpa

As, this pressure drop is lesser than the maximum allowable pressure drop in the shell side of 70 Kpa. It is acceptable.

MECHANICAL DESIGN OF CONDENSER:

SHELL SIDE:

Material : Carbon steel

No. of shells : 1

No. of passes : 4

Fluid : carbon disulphide

Internal diameter : 305 mm

Working pressure : 0.1 N/mm^2

design pressure : 0.11 N/mm^2

Inlet temperature : 80ºC

Outlet temperature: 80ºC

Allowable stress : 950 Kg/Cm^2

TUBE SIDE:

Material : Stainless steel (IS grade 10)

No. of tubes : 62

Outside diameter : 19.05 mm

Length : 2.438 m

fluid : water

pitch : 25.44 mm (triangular)

Allowable stress : 10.06 Kg/m^2

working pressure : 1.033 Kg/cm^2

Design pressure : 1.55 Kg/cm^2

Inlet temperature : 30ºC

Outlet temoerature : 40 ºC

SHELL SIDE

SHELL THICKNESS:

t_s = $(P_d \times D_S)/(2fJ-P_d)$

= (0.11×305)/((2×95×0.85)-0.11)

= 0.20 mm

But minimum thickness of shell is 6 mm

Therefore with corrosion allowance of 2mm

Thickness of shell = 8 mm

NOZZLE DIAMETER

M = Mass velocity/sec

= 2083.33/3600

= 0.5787 Kg/sec

Density ρ =1225 kg/m^3

Assume velocity to be 1 m/sec

$(\pi \times d_n^2 \times \rho \times v)/4 = M$

d_n^2= (0.5787×4)/(1×3.14×1225)

d_n =0.0245 m

NOZZLE THICKNESS

$t_n = (P_d \times d_n)/(2fJ-P)$

= (0.11×24.5)/(2×95×0.85-0.11)

= 0.016 mm

Nozzle thickness with corrosion allowance = 5 mm

HEAD THICKNESS:

$t_h = (P_d \times R_c \times W)/(2fJ)$

$W = (1/4)(3+(R_c/R_K)^{1/2})$

R_c - crown radius 80% of shell I.D. = 244 mm

R_K - Knuckle radius 10% of shell I.D.= 30.5 mm

W = 1.46

t_h = 0.242 mm

Using same thickness as that of the shell = 8 mm

BAFFLE ARRANGEMENT:

Tansverse bafffles

Number of baffles = 15

Baffle Spacing = D_s = 152 mm

Thickness of baffles = 6 mm

Height of baffle = $0.75 \times D_s$

= 114 mm

TIE RODS AND SPACERS:

For shell diameter D_s = 152 mm

No. of tie rods =

Diameter of rods = 13 mm

FLANGE CALCULATION:

Flange material = IS:2004-1962 class 2

Bolting steel =5% Cr M_o steel

Gasket material =asbestos composition

Shell inside diameter =305 mm

Shell thickness t_s = 8 mm

Shell outside diameter = $(2\times t_s)+305$

$= (2\times8) +305$

= 321 mm

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

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

GASKET WIDTH:

$G_o/G_i = [(y-P_d m)/(y-p_d(m+1))]^{1/2}$

m-Gasket factor =2.75

y-Minimum design seating stress =25.5 MN/m^2

$G_o/G_i = [25.5-(0.11\times2.75)/25.5-(0.11\times3.75)]^{1/2}$

= 1.0022

Minimum gasket width N =10 mm

Basic gasket seating width b_o = N/2

=5 mm< 6.3 mm

Let,inside diameter of gasket =inside diameter of shell =305 mm

G_i = 305 +(2×8)

=321 mm

Mean gasket width =G_i+N

=331 mm

therefore G = 331mm

Estimation of bolt load:

Load due to design pressure H =($\pi G^2 P_d$)/4

=(π×0331^2×0.11)/4

= 0.0094 MN

Effective gasket sitting width b =b_o=5 mm since b<6.3 mm

Load to keep joint tight under pressure H_p = π(2b)GmP_d

= 3.14×0.01×0.331×2.75×0.11

= 0.0031MN

Total operating load W_o =H+H_p

=0.0094+0.0031

= 0.0125 MN

Load to seat gasket under bolting up condition W_b = πbGy

=3.14×0.331×0.005×25.5

= 0.133 MN

Since W_b>W_o, controlling load =0.133 MN

Minimum bolting area:

Total cross sectional area of bolt under operating condition A_{m1}=W_o/S_b

S_b-nominal bolt stress at design temperature of 80ºC=138 MN/m^2

Therefore A_{m1} =0.0125/138

=0.000091 m^2

Total cross sectional area of bolt required for gasket seating A_{m2} =W_b/S_a

S_a-nominal bolt stress at ambient temperature (30ºC)=138 MN/m^2

Therefore A_{m2} =0.126/138

$=0.000913\ m^2$

Since $A_{m2} > A_{m1}$, $A_m = A_{m2} = 0.000913\ m^2$

Calculation of optimum bolt size:

$C = 2(R+g_1)+B$

Choosing bolt M-18 × 2

Total number of bolts = G/(18 × 2)

= 331/36

= 9

Actual number of bolts = 12

R-radial clearance from bolt circle to point of connection of hub and back of flange = 27 mm

B-inside diameter of flange =outside diameter of shell =0.331 m

$g_1 = g_o/0.707$, let $g_o = 8$ mm

$g_1 = 0.008$

C =2(0.027 + 0.008) + 0.331

= 0.441 m

Therefore bolt circle diameter = 0.441 m

Flange outside diameter:

A = C + bolt diameter + 0.02

= 0.441 + 0.018 + 0.02

= 0.479 m

Check of gasket width:

A_b-root area of bolt (m^2)-$1.54 \times 10^{-4}\ m^2$

S_g-allowable stress for bolting material at atmospheric temperature =138 MN/ m^2

Therefore, $nA_bS_g/\pi GN = 25.77$

Since 25.77 < 2y

Condition is satisfied.

Flange moment computations:

$W_o = W_1 + W_2 + W_3$ (under operating condition)

$W_1 = (\pi B^2 P_d)/4$

$= (3.14 \times 0321^2 \times 0.11)/4$

= 0.0090 MN

$W_2 = H - W_1$

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

$= (3.14 \times 0.331^2 \times 0.11)/4$

$= 0.0095$ MN

$W_2 = 0.0095 - 0.0090$

$= 0.0005$ MN

$W_3 = W_o - H$

$= 0.0125 - 0.0095$

$= 0.003$ MN

Total flange moment,

$M_o = W_1 a_1 + W_2 a_2 + W_3 a_3$

$a_1 = (C - B)/2$

$= 0.032$ m

$a_3 = (C\text{-}G)/2$

$= 0.027$ m

$a_2 = (a_1 + a_2)/2$

$= 0.0295$m

$M_o = 0.0090 \times 0.032 + 0.0005 \times 0.0295 + 0.003 \times 0.027$

$= 3.838 \times 10^{-4}$ MNm

Bolting up condition:

Total flange moment $M_g = W a_3$

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

$= 0.0736$ MN

$a_3 = 0.027$ m

Therefore $M_g = 0.01953$ MNm

$= 1.987 \times 10^{-3}$ MNm

Since $M_g > M_o$ for moment under operating condition, M_g is controlling.

Therefore $M = M_g = 1.987 \times 10^{-3}$ MNm

Flange thickness:

$t^2 = (M C_F Y)/B S_T$

$K = A/B$

= Outer diameter of flange/ inner diameter of flange

= .479/0.331

= 1.447

Y =14

Assume C_F =1

Therefore thickness 't' = 0.055 m

Actual bolt spacing $B_S = \pi C/n$

$= (3.14 \times 0.441)/12$

= 0.115 m

Bolt pitch correction factor $C_F = [B_S/(2d+t)]^{1/2}$

$= [0.115/(2 \times 0.018+0.055)]^{1/2}$

=0.381

Therefore $C_F^{1/2}$ = 0.617

Actual flange thickness = $C_F^{1/2}$ t

$= 0.617 \times 0.055$

= 0.0339 m

= 33.9 mm

TUBE SIDE:

TUBE THICKNESS:

$t_t = Pd_o/2fJ+P$

J =1 for seamless tube

Therefore $t_t = (1.55 \times 19.05)/(2 \times 1006+1.55)$

= 0.0146 mm

No corrosion allowance since the tube is made of stainless steel

Thickness of tube = 1mm

TUBE SHEET:

$t_S = FG[0.25P/f]^{1/2}$

F-The value of F varies according to type of heat exchanger, for most cases it is taken as 1

G = 331 mm

Therefore $t_S = 331[(0.25 \times 1.55)/1006]^{1/2}$

= 6.49 mm

CHANNEL AND CHANNEL COVER:

$t = G[KP/f]^{1/2}$

K =0.3 for ring type gasket

Material of construction is carbon steel

So allowable stress f =950 Kg/cm^2

Therefore $t = 331[(0.3\times1.55)/950]^{1/2}$

= 7.323 mm

With corrosion allowance t =10 mm

NOZZLE THICKNESS:

Assume inlet and outlet diameter = 75 mm

Thickness of nozzle t_h =Pd/2fJ-P

=(1.55×75)/(2×0.85×950-1.55)

=0.0612 mm

with corrosion allowance thickness = 6mm

SADDLE SUPPORT DESIGN:

Material -Low carbon steel

Vessel diameter= 305 mm

Length of shell = 2.438 m

Torispherical head:

Crown radius =244 mm

Knuckle radius = 30.5 mm

Working pressure =1 Atm

Shell thickness = 8 mm

Head thickness = 8mm

Corrosion allowance = 2mm

permissible stress = 950 Kg/cm^2

R-Vessel radius = 152.5

Distance of saddle centre line from shell end A =0.45×R

= 68.63 mm < 0.2L

Longitudinal bending moment:

The bending moment at the support is;

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

A = 68.63 mm

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

W-Weight of fluid and vessel.

Weight of shell material:

$W_1 = [\pi(D_o^2-D_i^2)L\rho_{\text{shell material}}] /4$

$\rho_{\text{shell material}}$ - 7700 Kg/m^3

$W_1 = [3.14(0.331^2-0.305^2)\times2.438\times7700] /4$

$= 244$ Kg

Weight of tubes:

$W_2 = n[\pi(D_o^2-D_i^2)L\rho_{\text{tube material}}] /4$

$\rho_{\text{tube material}}$ -7800 Kg/m^3

$W_2 = 62[3.14(0.01905^2-0.01705^2)\times2.438\times7800] /4$

$= 66.8$ Kg

Weight of tube sheet:

$W_3 = (2\pi D^2 t\rho)/4$

$= (2\times3.14\times0.331^2\times0.0180\times7800)/4$

$= 24.15$ Kg

Liquid load in the shell:

W_4 =(shell volume -tube volume)ρ_{liquid}

$=[(\pi D_s^2 L)/4 - (n\pi d_o^2 l)/4]\times1175$

$=[(\pi\times0.331^2\times2.438)/4 - (62\times\pi\times0.01905^2\times2.438) /4]\times1175$

$=(0.2098 – 0.043)\times1175$

$= 196$ Kg

Liquid load in tubes:

$W_5 = n\pi d_i^2 l\rho_{\text{liquid}} /4$

$=(62\times3.14\times0.01705^2\times2.438\times997)/4$

$= 215.9$ Kg

Therefore total weight $W_T = W_1 + W_2 + W_3 + W_4 + W_5$

= 746 Kg

Hence, $Q = 746/2[2.438+(4\times0.257)/3]$

$= 1037.19$Kgm

$M_1=(1037.19\times0.06863)[1-\{(1-0.06863/2.44)+(0.1525^2-0.257^2/2\times0.203\times4.88)/(1+4\times0.257/3\times4.88)\}]$

$=2909.261[1-\{(0.958+0.0698)/1.0702\}]$

$=115.26$ Kgm

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

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

$=(1037.19\times2.438/4)[\{1+2(0.4525^2-0.257^2)/2.438^2\}/\{1+(4\times0.257/3\times2.438)\}- (4\times0.203/2.438)]$

$= 1234[1.312 - 0.1664]$

$= 1414$ Kgm

Stress in shell at the saddle:

At the top most fiber of the cross-section,

$f_1 =M_1/(K_1\pi R^2t)$

For an angle of 120º ,$K_1 =0.107$ m

t-thickness of shell = 8 mm

$f_1 =115.6/(0.107\times3.14\times0.152^2\times0.008)$

$=10.37$ Kg/cm^2

At the bottom most fiber of the cross-section

$f_2 =M_1/(K_2\pi R^2t)$

For an angle of 120º ,$K_2 =0.192$ m

$f_2 =115.6/(0.192\times3.14\times0.152^2\times0.008)$

$=8.47$ Kg/cm^2

Stress in the shell at mid point:

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

$=1414/(3.14\times0.152^2\times0.008)$

$=248$ Kg/cm^2

Thus the values of stresses are within the limited range

Hence the designed support is acceptable.