MAJOR EQUIPMENT DESIGN

Process Design Of Chlorobenzene Distillation Column

For simplicity of calculation let us assume that the feed entering the distillation or chlorobenzene column is a binary mixture of MCB and p-dichlorobenzene and the o-dichlorobenzene is not entering as feed.

Vapour Pressure Data (Pressure =48269.00 Pa) :

Temperature (K)	**Vapour pressure of MCB (mm Hg)**	**Vapour pressure of DCB (mm Hg)**
405	700	190
408	810	210
411	900	240
415	1000	260
418	1050	300
422	1150	330
426	1300	370
429	1400	400
433	1500	450
437	1700	500
441	1800	575
444	2000	625
448	2200	680
453	2400	740

Vapour-liquid equilibrium data:

Temperature (K)	x_a (mole fraction of MCB in liquid)	y_a (mole fraction of MCB in vapor)
419.637	0.00	0.00
416.329	0.05	0.135
413.256	0.10	0.251
410.394	0.15	0.351
407.722	0.20	0.436
405.220	0.25	0.511
402.873	0.30	0.575
400.665	0.35	0.632
398.584	0.40	0.683
396.617	0.45	0.727
394.755	0.50	0.766
392.989	0.55	0.803
391.311	0.60	0.833
389.714	0.65	0.862
388.190	0.70	0.888
386.735	0.75	0.911
385.344	0.80	0.932
384.011	0.85	0.951
382.733	0.90	0.969
381.506	0.95	0.985
380.326	1.00	1.00

We have,

F = 27.04 kmols / hr

D = 23.52 kmols / hr

W = 3.52 kmols / hr

X_F = 0.865

X_D = 0.865

X_W = 0.865

Average molecular weight of Feed = 117.15 kg / kmol

Average molecular weight of Distillate = 112.84 kg / kmol

Average molecular weight of Residue = 145.96 kg / kmol

D = 2653 kg / hr

W = 513.77 kg / hr

Assume that the feed is a saturated liquid at its boiling point.

So $q = \frac{Hv - Hf}{Hv - Hl}$;

q = 1.

Slope = (q \ q-1) = ∞

From the graph ;

minimum (X_d \ R_m +1) = 0.73

minimum reflux R_m = 0.3561 mol reflux / mol D

Operating reflux ratio (1.5 R_m) = 0.5341

Intercept = 0.64

No. of ideal stages = 11

No. of stages in the enriching section = 4

No. of stages in the stripping section = 7

L = RD = 12.56 kmol / hr

G = L+D = 36.08 kmol / hr

$\overline{L}$ = L +qF = 39.6 Kmols \ hr

$\overline{G}$ = G +(q-1)F = 36.08 Kmols \ hr

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PROPERTIES :

	ENRICHING SECT		STRIPPING SECT	
	Top	Bottom	Top	Bottom
Temp. liq, °K	380.56	382.99	382.99	445.27
Temp. vap, °K	380.12	382.67	382.67	446.46
Liq. Flowrate,kmol/hr	12.56	12.56	39.6	39.6
Vap. Flowrate,kmol/hr	36.08	36.08	36.08	36.08
Vap density,kg/m3	1.723	1.733	1.733	1.898
Liq density,kg/m3	1012.0	1031.7	1031.7	1105.2
Avg. mol.wt. (vap.)	112.84	114.23	114.23	145.96
Avg. mol wt. (liq.)	112.84	116.29	116.29	145.96
Mole fraction, x	0.99	0.89	0.89	0.03
Mole fraction, y	0.99	0.95	0.95	0.03
L/G *(ρG/ρL)0.5	0.014	0.015	0.046	0.045
Liq. flow rate ,kg/hr	1417.27	1495.49	4605.08	5780.02
Vap. flow rate , kg/hr	4071.27	4121.42	4121.42	5266.24

AVERAGE PROPERTIES

	Enriching Section	Stripping Section
Liq. flowrate : kmol/hr.	12.56	39.6
Liq. flow rate : kg/hr.	1456.38	5192.55
Vap. flow rate : kmol/hr	36.08	36.08
Vap. flow rate : kg/hr.	4096.34	4693.83
$\bar{\rho}_L$ (kg/m^3)	1021.85	1068.45
$\bar{\rho}_V$ (kg/m^3)	1.728	1.8155
$\bar{\bar{T}}_{liq}$ (^oK)	381.77	414.13
$\bar{\bar{T}}_{vap}$ (^oK)	381.39	414.565
$\bar{\mu}_{liq}$, Pas	3.5415×10^{-4}	3.3155×10^{-4}
$\bar{\mu}_{vap}$, cP	1.44×10^{-5}	1.244×10^{-5}
D_L , cm^2/s	5.6961×10^{-5}	6.49050×10^{-5}
D_G , cm^2/s	8.169×10^{-2}	9.45×10^{-2}
σ (liquid,dynes/cm)	28.4914	22.3476

ENRICHING SECTION

TRAY HYDRAULICS :

(1) Plate spacing , t_s = 305 mm

(2) Hole diameter , d_h =5 mm

(3) Hole pitch , l_p =15 mm

(4) Tray thickness , t_T =3 mm

(5) $$\frac{\text{Hole area}}{\text{Perforated area}} = \frac{A_h}{A_p} = 0.10$$

Assume equilateral triangular pitch

(6) Column dia ,D_c :

Based on entrainment flooding.

All relations from Perry's handbook, 6th edition.

Fig. 18-10 , $C_{sb} = U_{nf} \left[\frac{20}{\sigma}\right]^{0.2} \left[\frac{\rho_g}{\rho_l - \rho_g}\right]^{0.5}$ ft/s.

$(L/G) \times (\rho_g / \rho_l)^{0.5} = 0.015$

C_{sb} =0.23 m\s

U_{nf} = 5.9982 ft\s

Assume U_n =0.8 U_{nf} = 1.4511 m/s.

Net area for flow , $A_n = A_c - A_d$

Vapour flow rate = 0.6583 m^3/s

A_n = 0.6585 / 1.4511 = 0.4538 m^2.

A_d = 0.0988 m^2

Assume $\frac{L_w}{D_c} = 0.77$

$$\text{Sin}\left(\frac{\theta_c}{2}\right) = \frac{L_w/2}{D_c/2}$$

$\theta_c = 100.7^0$

$A_c = \frac{\pi D_c^2}{2}$ = 0.785 D_c^2

A_d = 0.0988 D_c^2

A_n = 0.785 D_c^2 – 0.0988 D_c^2

D_c = 0.81 m , Corrected A_d = 0.0622 m^2

A_c = 0.5153 m^2

L_w= 0.62 m

Active area, $A_a = A_c - 2A_d$ =0.3909 m^2.

(7)Perforated area,A_p

corrected $\theta_c = 99.9^0$

$\alpha = 180 - \theta_c = 80.1^0$

Area of calming + distribution zone, A_{cz}

$A_{cz} = 2(L_w * t)$, t=thickness

$A_{cz} = 0.0351\ m^2$ (that is 6.81% of A_c)

$A_{wz} = 0.0327\ m^2$.

let w = 20 mm

$A_w = 0.0221\ m^2$ (that is 4.3% of A_c)

$A_d = 0.0622\ m^2$

$A_p = A_c - 2\ A_d - A_{cz} - A_{wz}$

$= 0.3231\ m^2$

$A_h = 0.1\ A_p = 0.03231\ m^2$

(8) No. of holes, n_h = 1646 holes.

(9) Weir height , h_w = 12 mm

(10) Weeping check :

$$h_d = K_1 + K_2 \frac{\rho_g U_h^2}{\rho_l}$$

Assume sieve plates

$K_1 = 0$, $K_2 = 50.8 / C_v^2$

$A_h / A_a = 0.083$

$t_T / d_h = 0.6$

$C_v = 0.73$

$\therefore h_d = 66.95$ mm

Frances Weir Equation :

$$\therefore h_{ow} = F_w \text{ x } 664 \text{ x } \left[\frac{q}{L_W}\right]^{2/3}$$

q(liquid load) = 3.459 x 10^{-4} m^3/s = 6.275 gal/min

F_w = 1.005

h_{ow} = 4.95 mm

Head loss due to bubble formation ,

$$h_\sigma = 409\left[\frac{\sigma}{\rho_l d_n}\right]$$

∴ h_σ = 2.28 mm liq.

Now, h_d + h_σ = 59.23 mm liq.

h_w + h_{ow} =16.95 mm

Since from the graph; actual design > minimum design there is no weeping (from fig. 18-11)

(11) Downcomer flooding :

h_{dc} =h_t + h_w+ h_{ow}+ h_{da}+ h_{hg}

Dynamic Seal; h_{ds} = h_w + h_{ow} +h_{hg}/2

q= 4.02165 x 10^{-4} m^3/s = 6.382 gal/min

L_w = 0.62 m = 2.05 ft ; F_w = 1.005

h_{ow} =5.00 mm

h_{ds} = 12 +5+0.15\2 =17.075 mm

h_t = h_d+h_l^1

h_l^1 = βh_{ds}

U_a = 1.6845 m/s = 5.5707 ft/s

$\rho_a = 0.1079\ lb / ft^3$

$F_a = U_a \rho_g^{0.5}$

$= 1.83$

$\beta = 0.59$

$\phi_t = 0.2$

$h_l' = 0.59 \times 17.075 = 10.074$ mm

$h_t = 66.95 + 10.074 = 77.024$ mm.

Loss under downcomer, h_{da}

$h_{da} = 165.2\ (q / A_{da})^2$

Assume clearance C = 13 mm.

$h_{ap} = 17.075 – C = 4.075$ mm

$A_{da} = L_w h_{ap} = 2.5265 \times 10^{-3}\ m^2$

$h_{da} = 3.92$ mm

$h_{dc} = h_t + h_w + h_{ow} + h_{da} + h_{hg}$

$=77.024+12+5+3.92+0.15$

$= 98.094$ mm

$t_s = 305$ mm

As $h_{dc} < t_s$,there is no downcomer flooding.

Summary of tray calculations

$D_c = 0.81$m

$L_w = 0.62$m

$h_w = 12$ mm

$t_s = 305$ mm

$d_h = 5$ mm

$l_p = 15$ mm, triangular pitch.

$t_t = 3$ mm

$n_h = 1646$

% flooding = 80

(12) Column Efficiency

(a)Point Efficiency, E_{OG}

$$N_g = \frac{0.776 + 0.2285 h_W - 0.238 U_a \rho_g^{0.5} + 105 W}{N_{scg}^{0.5}}$$

W = liq. flow rate = $(3.9589 \times 10^{-4} / 0.715)$

= $8.13 \times 10^{-3}\ m^3 / s$

$U_a = 1.6845$ m / s

$h_w = 12$ mm, $N_{Scg} = (\mu_g \backslash \rho_g D_g) = 1.02$

$N_g = 0.3548$

$N_L = K_L a \theta_L$

$K_L a = (3.875 \times 10^8 D_L)^{0.5} (0.4\ U_a \rho_g^{0.5} + 0.17)$

= 1.568 / s

$\theta_L = (h_f A_a \phi_t) / (1000q)$

$h_f = 193.90$ mm

$\theta_L = 38.29$ s

$N_L = 60.038$

$$\lambda = M \frac{G_m}{L_m}$$

$$\frac{G_m}{L_m} = \frac{36.08}{12.56} = 2.886\ ;\ M = 0.385$$

$\lambda = 1.111$

$$N_{og} = \frac{1}{\frac{1}{N_g} + \frac{\lambda}{N_L}} = 0.974$$

$E_{OG} = 1 - e^{-NOG} = 0.2970$

(b) Murphree Plate Efficiency, E_{mv}

$\lambda E_{OG} = 0.32967$

$$N_{Pe} = \frac{Z_L^2}{D_E \theta_L}$$

$Z_L = D_c \cos(\theta_L / 2) = 0.52$ m

$D_E = 6.675 * 10^{-3} (U_a)^{1.44} + 0.922 * 10^{-4} h_L - 0.00562$

$= 0.0120$ m / s

$N_{Pe} = 2.2669$

From fig. 18.29a $\frac{E_{mv}}{E_{OG}} = 1.09$

$\therefore E_{mv} = 0.3237$

(c) Overall column efficiency, E_{oc}

$$E_{oc} = \frac{N_T}{N_A} = \frac{\log[1 + E_a(\lambda - 1)]}{\log \lambda}$$

$$\frac{E_a}{E_{MV}} = \frac{1}{1 + E_{MV}\left[\frac{\psi}{1-\psi}\right]}$$

Considering 80% flooding,

From fig, $\Psi = 0.2$

$\therefore E_a = 0.29946$

$E_{oc} = 0.30789$

$E_{oc} = N_t \backslash N_A$

$N_A = 12.66 \approx 12$ trays

$N_A = 12$ trays

Tower height, $= t_s * N_A$

$= 305 * 12 = 3660$ mm

$\therefore H = 3.66$ m

STRIPPING SECTION

TRAY HYDRAULICS :

(1) Plate spacing , $t_s = 305$ mm

(2) Hole diameter , $d_h = 5$ mm

(3) Hole pitch , $l_p = 15$ mm

(4) Tray thickness , $t_T = 3$ mm

(5) $\dfrac{\text{Hole area}}{\text{Perforated area}} = \dfrac{A_h}{A_p} = 0.10$

Assume equilateral triangular pitch

(6) Column dia ,D_c :

Based on entrainment flooding.

All relations from Perry's handbook, 6th edition.

Fig. 18-10 , $C_{sb} = U_{nf} \left[\dfrac{20}{\sigma}\right]^{0.2} \left[\dfrac{\rho_g}{\rho_l - \rho_g}\right]^{0.5}$ ft/s.

$(L/G) \times (\rho_{g/}\ \rho_{l})^{0.5} = 0.046$

$C_{sb} = 0.21$ m\s

U_{nf} = 5.204 ft\s

Assume U_n =0.8 U_{nf} = 1.2590 m/s.

Net area for flow , $A_n = A_c - A_d$

A_n = 0.6121 m^2.

A_d = 0.0988 m^2

Assume $\frac{L_w}{D_c} = 0.77$

$$\text{Sin}\left(\frac{\theta_c}{2}\right) = \frac{L_w/2}{D_c/2}$$

$\theta_c = 100.7^0$

$$A_c = \frac{\pi D_c^2}{2} = 0.785\ D_c^2$$

A_d = 0.0988 D_c^2

A_n = 0.785 D_c^2 – 0.0988 D_c^2

D_c = 0.94 m , Corrected A_d = 0.0883 m^2

A_c = 0.6939 m^2

L_w= 0.73 m

Active area, $A_a = A_c - 2A_d$ =0.5173 m^2.

(7)Perforated area,A_p

corrected $\theta_c = 99.9^0$

$\alpha = 180 - \theta_c = 80.1^0$

Area of calming + distribution zone, A_{cz}

A_{cz} =2(L_w * t) , t=thickness

A_{cz} = 0.05446 m^2 (that is 7.85% of A_c)

A_{wz} = 0.0251 m^2 (that is 3.6 % of A_c)

let w = 40 mm

A_d = 0.0883 m^2

$A_p = A_c - 2\ A_d - A_{cz} - A_{wz}$

$= 0.4377\ m^2$

$A_h = 0.1\ A_p = 0.043774\ m^2$

(8) No. of holes, n_h = 2230 holes.

(9) Weir height , h_w = 12 mm

(10) Weeping check :

$$h_d = K_1 + K_2 \frac{\rho_g U_h^2}{\rho_l}$$

Assume sieve plates

$K_1 = 0$, $K_2 = 50.8 / C_v^2$

$A_h / A_a = 0.0846$

$t_T / d_h = 0.6$

$C_v = 0.73$

$\therefore h_d$ =50.75 mm

Frances Weir Equation :

$$\therefore h_{ow} = F_w \times 664 \times \left[\frac{q}{L_W}\right]^{2/3}$$

q(liquid load) = $1.239 \times 10^{-3}\ m^3/s$ = 19.68 gal / min

$F_w = 1.02$

h_{ow} = 9.616 mm

Head loss due to bubble formation ,

$$h_\sigma = 409\left[\frac{\sigma}{\rho_l d_n}\right]$$

$\therefore h_\sigma$ = 1.7109 mm liq.

Now, $h_d + h_\sigma$ = 52.54 mm liq.

$h_w + h_{ow}$ =21.616 mm

Since from the graph; actual design > minimum design(10 mm) there is no weeping (from fig. 18-11)

(11) Downcomer flooding :

$h_{dc} = h_t + h_w + h_{ow} + h_{da} + h_{hg}$

Dynamic Seal; $h_{ds} = h_w + h_{ow} + h_{hg}/2$

q= 1.4527 x 10^{-3} m^3/s = 22.5973 gal/min

L_w = 0.73 m = 2.395 ft ; F_w = 1.025

h_{ow} =10.74 mm

h_{ds} = 12 +10.74+0.15\2 =22.815 mm

$h_t = h_d + h_l^1$

$h_l^1 = \beta h_{ds}$

U_a = 1.388 m/s = 4.5548 ft/s

ρ_a = 0.1133 lb/ft^3

$F_a = U_a \rho_g^{0.5}$

= 1.5331

β = 0.6

ϕ_t = 0.22

h_l'= 0.6 x22.815 = 13.689 mm

h_t = 50.75 + 13.689 = 64.439 mm.

Loss under downcomer, h_{da}

$h_{da} = 165.2 (q / A_{da})^2$

Assume clearance C = 13 mm.

h_{ap} = 22.815 – C = 9.815 mm

$A_{da} = L_w h_{ap}$ = 7.164 x 10^{-3} m^2

h_{da} = 4.94 mm

$h_{dc} = h_t + h_w + h_{ow} + h_{da} + h_{hg}$

$=64.439+12+10.74+4.94+0.15$

$= 92.269$ mm

$t_s = 305$ mm

As $h_{dc} < t_s$,there is no downcomer flooding.

Summary of tray calculations

$D_c = 0.94$m

$L_w = 0.73$m

$h_w = 12$ mm

$t_s = 305$ mm

$d_h = 5$ mm

$l_p = 15$ mm, triangular pitch.

$t_t = 3$ mm

$n_h = 2230$

% flooding = 80

(12) Column Efficiency

(a)Point Efficiency, E_{OG}

$$N_g = \frac{0.776 + 0.2285 h_W - 0.238 U_a \rho_g^{0.5} + 105 W}{N_{scg}^{0.5}}$$

W = liq. flow rate = $(1.3499 \times 10^{-3} / 0.835)$

$= 1.6166 \times 10^{-3}\ m^3 / s$

$U_a = 1.3883$ m / s

$h_w = 12$ mm, $N_{Scg} = (\mu_g \backslash \rho_g D_g) = 0.725$

$N_g = 0.6522$

$N_L = K_L a \theta_L$

$K_La = (3.875 \times 10^8 D_L)^{0.5} (0.4\, U_a \rho_g^{0.5} + 0.17)$

$= 1.4562\ /s$

$\theta_L = (h_f A_a \phi_t) / (1000q)$

$h_f = 133.242$ mm

$\theta_L = 11.23$ s

$N_L = 16.353$

$$\lambda = M \frac{G_m}{L_m}$$

$$\frac{G_m}{L_m} = \frac{36.08}{39.6} = 0.911 \text{ ; } M = 1.635$$

$\lambda = 1.489$

$$N_{og} = \frac{1}{\frac{1}{N_g} + \frac{\lambda}{N_L}} = 0.6156$$

$E_{OG} = 1 - e^{-NOG} = 0.4596$

(b) Murphee Plate Efficiency, E_{mv}

$\lambda E_{OG} = 0.6843$

$$N_{Pe} = \frac{Z_L^2}{D_E \theta_L}$$

$Z_L = D_c \cos(\theta_L / 2) = 0.59$ m

$D_E = 6.675 * 10^{-3} (U_a)^{1.44} + 0.922 * 10^{-4} h_L - 0.00562$

$= 7.78838 \times 10^{-3}$ m / s

$N_{Pe} = 3.979$

From fig. 18.29a $\frac{E_{mv}}{E_{OG}} = 1.23$

$\therefore E_{mv} = 0.565$

(c) Overall column efficiency, E_{oc}

$$E_{oc} = \frac{N_T}{N_A} = \frac{\log\left[1 + E_a(\lambda - 1)\right]}{\log \lambda}$$

$$\frac{E_a}{E_{MV}} = \frac{1}{1 + E_{MV}\left[\frac{\psi}{1 - \psi}\right]}$$

Considering 80% flooding,

From fig, $\Psi = 0.22$

$\therefore E_a = 0.4873$

$E_{oc} = 0.5368$

$E_{oc} = N_t \setminus N_A$

$N_A = 13.04 \approx 13$ trays

$N_A = 13$ trays

Tower height, $= t_s * N_A$

$= 305 * 13 = 3965$ mm

$\therefore$ H = 3.965 m

Overall tower height = 3.66 + 3.965 = 7.625 m

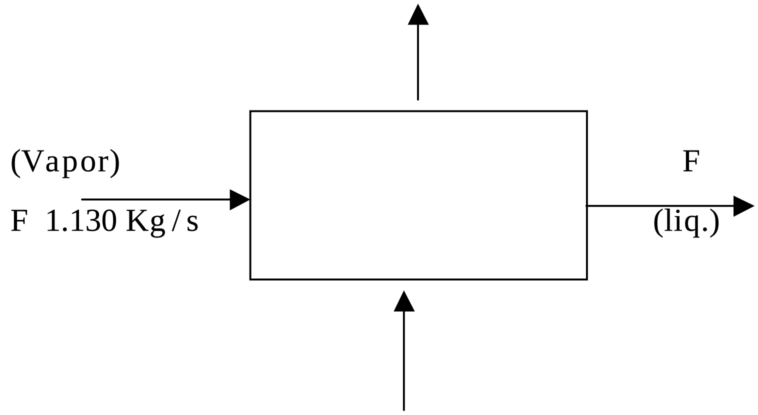
Total number of plates =12+13 = 25 plates.

PROCESS DESIGN OF CONDENSER

Hot fluid enters at 380.12 K and leaves at 380.12 K

Cold fluid enters at 298 K and leaves at 313 K.

PROCESS DESIGN :



Cooling

Condensation occurs at isothermal condition, correction factor F_t =1.

Assume feed is saturated vapor at 380.12 K

λ =Latent Heat =324.86 KJ/Kg

Vapor flow rate = 1.130 Kg/s

Heat load =mλ=1.130 x 324.86=367 x 10^3 J\sec

Properties are evaluated for cold fluid water

μ = Viscosity = 1 CP

ρ = Density = 1000 kg/m^3.

c_p = Heat capacity =4.187 KJ/kg K

k = 0.578 W/m K

Using this,

Mass of process water required = 5.849 kg/s.

The range of U for organic solvent and water is (289.3 to 567.8 w\$m^2$k)

Assuming counter current operation ΔT_{ln} = 74.36

Assuming U heat transfer coefficient = 300 W/m^2K

A = Q /U (ΔT_{ln}) =(367.38x10^3)\300x74.36 =16.468 m^2.

Assuming length of pipe to be 10 ft. and we take 3/4 inch O.D.,16 BWG tube,

I.D. of tube =0.01574 m

External area =0.05987 m^2\m

N_t =16.468\0.05487x3.048 = 90.19

From Perry for 1-4 TEMA P or S for 3/4 inch OD on 15/16 inch Triangular pitch

N_t = 106 for Shell diameter = 0.337 m.

So corrected area =19.34 m^2.

Corrected U =254.59 W/m^2K

Flow area = 3.14/4 X 0.01574^2 = 194.7 X 10^{-6} m^2 /tube.

a_t = 106 x 194.7 x10^{-6} /4 = 5.156 x 10^{-9} m^2 /pass

G_t = 5.589/5.156 x 10^{-9} = 1134.20 kg/m^2s velocity =1134.2 / 1000

=1.134 m \ sec

Therefore the velocity is above the minimum value.

Tube side heat transfer coefficient:

Prandtl number (N_{Pr}) = (c_p μ) \ k = 7.24

Reynolds's number (N_{Re}) =(D_ivρ) \ μ = 17858.23

For turbulent conditions Dittus Boelter equation.

$$\frac{h_i d_e}{k} = 0.023\,(N_{Re})^{0.8}\,(N_{Pr})^{0.33}$$

h_i = 4091.04 W / m^2K

FILM TRANSFER COEFFICIENT

Shell side – Distillate

Temp. of wall= 69.75 deg C

Film temp. = 88.375 deg C or 361.37 K

Property of condensing vapor at this temp.

ρ = 1050 kg / m^3

k = 0.115 W / mK

μ = 0.0004 Pas.

C_p= 1.549 KJ / kg K

$$\text{Reynold's number } (N_{Re}) = \frac{4 \times Mass\ \text{flow rate of condensate}}{\mu \times N_t^{2/3} \times Length\ \text{of tube}}$$

=168.97

$$h_s = 1.151[\ \ ]^{1/3}\,[N_{Re}]^{-1/3}$$

h_s =1279.27 W / m^2K

the dirt factor = $5.283x10^{-4}$ W / m^2K

Overall heat transfer cofficient :

$1 \backslash U = (D_o \backslash D_i) x 1 \backslash h_i + 1 \backslash h_s + 5.283x10^{-4}$

U = 622.79 W / m^2K

Calculated U = 622.79 W / m^2K   assuming U= 300 W / m^2K

∴ Design is okay.

PRESSURE DROP CALCULATION

TUBE SIDE

N_{Re} = 17858.23

$f = 0.079 (N_{Re})^{-1/4} = 6.833 \times 10^{-3}$

$H = (4 \times f \times v^2 \times L) \backslash (2gD)$

=03.378

$P = \rho g H$

=3.378 kN \ m^2

$\Delta P = (2.5\rho\ v^2) \backslash 2$

=1.596 kN \ m^2

ΔP_{Total} = 4(1.596+3.378)

=19.896 kN\ m^2

which is very less than permissible, therefore design is okay

SHELL SIDE

T_{Vapour} =380.12 K

$\delta_m = (p^1 - D) l_s D_s \backslash p^1$

here ,

p^1 = pitch =23.81 mm

l_s = 0.89 D_s = 300 mm

D_s = shell diameter = 337 mm

δm =0.021 m^2

PRESSURE DROP CALCULATION

End zones Δp_e ,two end zones.

Cross flow zones Δp_c ,(N_b –1) cross

Window zones Δp_w , N_b zones

$\Delta p_c = b \ f_k \ w^2 \ N_c \ (\mu_w \backslash \ \mu_b)$

$b = 5 \times 10^{-4}$

$w = 1.13$ Kg\s

$\delta m = 0.021 \ m^2$

$N_c = D_s (1 - 2(l_c \backslash \ D_s)) \backslash \ P_p$

$P_p = (1.732 \backslash 2) \ p^1$ mm

=20.61 mm

l_c = 30% of shell dia.

$N_c = 6.54$

Δp_c =13.1 KN \ m^2

$\Delta p_w = (b \times (W)^2 (2 + 0.6 \times N_{cw})) \backslash \ (S_w \times \delta m \times \rho)$

$N_{cw} = 0.8 \ l_c \ \backslash \ P_p$

= 3.92

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

$S_{wg} = 0.0258 \ m^2$

$S_{wt} = ((N_t) (1 - F_c) \Pi D_0^2) \backslash 8$

$S_{wt} = 4.22 \times \times 10^{-3} \ m^2$

Δp_w =3.57 KN \ m^2

$\Delta p_e = \Delta p_c \times 2$

= 13.1 x 2

= 26.2 KN \ m^2

Total pressure drop = 0 +3.57 +26.2

= 29.77 KN \ m^2

But actual pressure is 40% of this = 11.908 KN \ m^2

As $11.908 < 14$, so the design is okay.

MECHANICAL DESIGN OF DISTILLATION COLUMN

a) **SPECIFICATION**:

1. inside diameter : 0.94 m (design with the maximum dia for safety)
2. height of top disengaging section: 0.3 m
3. height of bottom separation section: 0.4 m
4. design pressure: 0.4925 kgf/cm^2

5. Since the vessel operated under vacuum,subjected to external pressure,external pressure:0.5405 kgf/cm^2
6. design pressure: 1.033 kgf/cm^2
7. design temperature: 120^{o}C
8. shell material: carbon steel(sp. Gr.=7.7) (IS:2002-1962, GRADE I)
9. permissible tensile stress: 950 kgf/cm^2
10. insulation material: asbestos
11. density of insulation: 2700 kg/m^3
12. insulation thickness: 50 mm
13. tray spacing: 305 mm
14. down comer plate material: stainless steel(sp. Gr.: 7.8)

SHELL THICKNESS CALCULATION:

Let the thickness of the shell= 6 mm

Using stiffener channels of C-60, 18x4, of CSA=18 in2

$Wt = 51.9\ lb/ft$

At a distance of 305 mm, (below each tray)

∴, D_o = 0.952 m

L = 0.305 m

∴, **L/D_o** = 0.3203

&, **D_o/t** = 158.66 ∴, B= 13100

∴, **$p_{allowable}$= B/(14.22*(D_o/t)),** t= 1.067 mm ≈ 6 mm

Which, suggest the thickness is allowable under the operating condition.

Therefore, allowing corrosion correction of 2mm, thickness choosen= 6+2=8 mm

HEAD:

Design for torispherical head.

The head is under external pressure(design pressure).

Let, t_h = 6 mm

$R_c = D_o$ = 0.952 m

∴, **$R_c/(100*t_h)$** = 1.586

∴, B= 10500

∴,$p_{allowable}$= B/(14.22*R_c/t_h), ∴, t_h= 1.3318

Since t_h < 6 mm ,so a design thickness of 6mm is sufficient for the head.

The design thickness chosen with a corrosion allowance :t_h= 6 mm

The approximate weight of the head is calculated= 50 kg

CHECK FOR SHELL THICKNESS:

<u>Material specification:</u>

Carbon steel (sp. Gr. =7.7) (IS: 2002-1962, GRADE I)

Tensile strength(R_{20})= 37 kgf/mm^2

Yield stress (E_{20})= 0.55R_{20}

Since, the vessel operated under vacuum, compressive axial stress:

$\mathbf{f_{ap} = pd/(4*(t_s\text{-}c))}$ = 40.975 kgf / cm^2

i) <u>Dead wt calculation:</u>

Total dead load can be calculated as:

∑W = head wt+ liquid wt(X)+ wt of the attachment(X)

head wt= 50 kg

liquid hold up in each tray= $\mathbf{\rho l*(A_a*h_l + A_d*h_{dc})}$

= 20 kg

wt of attachment per plate= 110 kg (approx.)

∴,∑W= (50+ 426X) kg

Where, X is the distance in meter from the top tangent.

Further, the wt of the insulation and shell also exerts a compressive stress:

$\mathbf{\pi*d_i*X*t*\rho_s + \pi/4*(D_{0,ins}^2 - d_0^2)*X*\rho_{ins}}$ = 561.39X kg

∴, total compressive stress on the shell due to dead wt:

$\mathbf{f_{dsx} = \sum W/(\pi*d_i*(t_s\text{-}c))}$ = (0.282+5.578X) kg / cm^2

ii) Wind pressure calculation:

Assume P_w = 130 kg / m^2 (maximum)

∴Moment at a distance X from the top tangent:

$\mathbf{M_{wx} = ½*P_w*X^2*d_{eff}}$ = 68.38X^2 kg-m

Where, d_{eff}= effective outer diameter of the vessel including insulation= 1.052 m

∴, f_{wx}= tensile stress on the upwind side

= $M_{wx}/(\pi*r_o^2*(t_s-c))$ = 1.6422 X^2 kg/cm^2

STRESS BALANCE FOR THE UPWIND SIDE:

$F_{t,max} = f_{wx} - f_{dsx} - f_{ap}$

Where, $F_{t,max}$= 50% of the maximum allowable stress

= 475 kg/cm^2

there for upwind side solution gives: X= 19.5 m(>7.625 m)

STRESS BALANCE FOR THE DOWNWIND SIDE:

$F_{C,max} = f_{wx} + f_{dsx} + f_{ap}$

Where, $F_{C,max}$= maximum allowable compressive stress.

= 1/3* yield stress

= 1/3* 2035 kgf/cm^2

Therefore, the solution to the quadratic equation:

X= 18.07086 m (> 7.625 m)

N.B. the wind moment on the downwind side act as a compressive force on the tower.

Since, the thickness of 8 mm with corrosion allowance is enough to with stand the load of the tower of 7.625 m height, the thickness of the shell is maintained 8 mm through out the entire tower length.

SKIRT SUPPORT

Diameter of column = 1.056 m

Height of vessel = 8.325 m

Height of skirt = 2 m

Diameter of skirt (straight) = 1.056 m

Dead weight of empty vessel = 5080 kgs (assumed)

Total dead weight = 5080 + 180 = 5260 kgs

(a) Due to dead weight :

$f_d = \sum w / (\pi \times D_{ok} \times t_{sk})$

$= 5260 / (3.14 \times 105.6 \times t_{sk})$

$= (15.855 / t_{sk})$ kg / cm^2

(b) Stress due to the wind load :

$f_{wb} = (M_{w/z}) = (4 \times M_w) / (\pi \times D_o^2 \times t_{sk})$

$M_w = p_{bw} \times (H / 2)$ for $H <= 20$ m

$= k \times p_1 \times h_1 \times D_{o\,x} (h_1 / 2)$

$= 3329.99 \times 100$

$f_{wb} = (38.02 / t_{sk})$ kg / cm^2

(c) Stress due to seismic load :

$f_{hb} = (M_{sb} / \pi \times R_{ok} \times t_{sk}) = (2 / 3) \times (CWH / \ / \pi R_{ok} t_{sk})$

$= (0.032 / t_{sk})$ kg / cm^2

(d) Tensile stress at bottom of skirt

$= f_{wb\,-} f_d$

$= (38.02 / t_{sk}) - (15.855 / t_{sk})$

$f_{tmax} = (22.165 / t_{sk})$ kg / cm^2

Assume skirt material ; IS- 2002-1962; GRADE I

Permissible stress = 950 kgf / cm^2

Yield Stress = 0.55 x 37 = 2035 kgf / cm^2

So,

t_{sk} = (22.165 / 950) = 0.23 mm

(e) Maximum compressive stress :

$f_{cmax} = f_{wb} + f_d$

= (53.875 / t_{sk}) kgf / cm^2

f_c permissible <= (1 / 3) x yield stress

<= 678.33 kgf / cm^2

t_{sk} = (53.875 / 678.33) = 0.79 mm

So, we use a minimum thickness of 6 mm.

SKIRT BEARING PLATE

Assume bolt circle diameter = skirt dia +32.5 cm

=105.6 + 32.5

=138.1 cm

Compressive stresses between bearing plate and concrete foundation :

$f_c = (\sum w / A) + (M_w / z)$

where $z = (\pi \times D_o^2 \times t_{sk})$

f_c = 2.8015 kgf / cm^2

Permissible value for concrete is 35 kgf / cm^2.

Thus 2.8015 kgf / cm^2 is less than the permissible value for concrete.

Maximum bending moment in bearing plate :

Stress $f = (6 \times f_c \times l^2) / (t_b^2 \times 20$

Where l is the difference between the outer radius of the bearing plate and the outer radius of shirt.

= (138.1 / 2) –(105.6 / 2)

= 16.25 cm

$f = (2219.31\ /\ t_b^2)$ kgf/cm^2

Permissible stress in bending = 1125 kgf/cm^2 (assume)

t_b^2 = 2219.31/1125

t_b= 19.72 mm

As t_b>18 so a bolting chair is necessary.

ANCHOR BOLTS :

The minimum stress between the bearing plate and the concrete foundation will be

$f_{cmin} = (w_{min} / A) – (M_w / z)$

where w_{min} =5080 kgs=is the minimum weight of the empty vessel.

f_{cmin} = -1.139 kgf/cm^2

Since f_{cmin} is found to be negative ,the vessel skirt must be anchored to the concrete foundation by bolts.

Assuming 24 bolts ;

$P_{bolt} = f_{cmin}$ x A/24

= 93.64 kgs or considered approximate to 100 kgs.

MECHANICAL DESIGN OF CONDENSER

(a) Shell Side:

Material carbon steel (Corrosion allowance = 3mm)

Number of shells passes =1

Working pressure = 48269 Pa = 0.0483 N / mm^2

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

Temperature of the inlet = 107.12 ^{0}C

Temperature of the outlet = 107.12 ^{0}C

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

(b)Tube side :

Number of tubes = 106

Outside diameter = 19.05 mm

Inside diameter = 15.75 mm

Length = 4.88 m

Pitch, $\triangle^{lar}$ = 15 / 16 inches = 23.8 x 10^{-3} m

Feed =Water.

Working Pressure = 48269 Pa = 0.0483 N / mm^2

Design Pressure =0.101 N / mm^2

Inlet temperature =25 ^{0}C.

Outlet temperature = 40 ^{0}C

Shell Side :

Let the assumed thickness of shell be 6 mm

D_o = 349 mm

L = 4876 mm

(D_o / t) = 58.166

(L / D_o) = 13.97

From IS : 2825-1969, table F2, B = 14000

$P_{au} = B / (14.22 X (D_o / t))$

t = 0.366 mm

As 0.366 mm < 6 mm ,so we can design a shell of thickness 6 mm

t_S = 6+3(corrosion allowance) = 9 mm. We assume a thickness of 10 mm.

Head : (Torispherical head)

Let us assume a torispherical dished head is to be designed under external pressure or the design pressure. Let the assumed thickness of shell be 6 mm

$D_o = r_c$ = 349 mm

L= 4876 mm

(D_o / t) = 58.166

(L / D_o) = 13.97

From IS : 2825-1969, table F2, B = 12200

$P_{au} = B / (14.22 X (D_o / t))$

t = 0.420 mm

As 0.420 mm < 6 mm ,so we can design a head of thickness 6 mm

t_h = 6+3(corrosion allowance) = 9 mm. We assume a thickness of 10 mm.

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

Flanges :

Loose type except lap-joint flange.

Design pressure (p) =0.101 N/mm^2

Flange material : IS:2004 –1962 class 2

Bolting steel : 5% Cr Mo steel.

Gasket material = Asbestos composition

Shell side diameter =337 mm

Shell side thickness =10 mm

Outside diameter of shell =337 + 10x 2 = 357mm

<u>Determination of gasket width :</u>

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

y= Yield stress

m= gasket factor

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

Thickness = 10mm

m= 2.75

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

$$\frac{d_o}{d_i} = \left[\frac{25.5 - 0.101\,(2.75)}{25.5 - 0.101\,(2.75 + 1)} \right]^{1/2} = 1.004$$

d_i = inside diameter of gasket = outside diameter of shell

= 357 + 5mm

=362 mm

d_o = outside diameter of the gasket

= 1.004 (362)

= 364 mm

$$\text{Minimum gasket width} = \frac{364 - 362}{2} = 1\text{ mm}$$

But minimum gasket width = 6mm

∴ G= 362 + 2 (0.006) = 0.374 m

Where G = diameter at the location of gasket load reaction

Estimation of bolt loads :

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

$$H = \frac{\pi}{4} (0.374)^2 (0.101 \times 10^6) = 11095.69\ N$$

Load to keep the joint tight under operating conditions.

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

b= Gasket width = 6mm = 0.006m

$H_p = \pi\ (0.374)\ (2 \times 0.006)\ 2.75 \times 0.101 \times 10^6 = 7161.7\ N$

Total operating load (W_o) = $H+H_p$

= 15012.39 N

Load to seat gasket under bolt –up condition =W_g.

$W_g = \pi\ g\ b\ y$

$= \pi \times 0.374 \times 0.006 \times 25.5 \times 10^6$

W_g = 179768.21 N

$W_g > W_0$

∴ W_g is the controlling load

∴ Controlling load = 179768.21 N

Calculation of minimum bolting area :

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

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

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

At 107.12 ^{0}C.

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

$A_m = 1.32182 \times 10^{-3} m^2$

Calculation for optimum bolt size :

$g_1 = \frac{g_o}{0.707} = 1.415 \; g_o$

g_l = thickness of the hub at the back of the flange

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

Selecting bolt size M18x2

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

R= 0.027

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

C= 0.337 +2 (1.415 (0.0125) + 0.027) = 0.4263 m

Actual flange outside diameter (A) = C+ bolt diameter + minimum left out

$= 0.4263 + 0.018 + 0.02$

$= 0.4643$

Check for gasket width :

A_b = minimum bolt area = $44 \times 1.54 \times 10^{-4}\ m^2$

$$\frac{A_b S_g}{\pi GN} = \frac{(44 \times 1.54 \times 10^{-4})138}{\pi \times 0.628 \times 0.016} = 49.740 \quad N/mm^2$$

$2y = 2 \times 25.5 = 51 \quad N/mm^2$

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

i.e., bolting condition is satisfied.

Flange Moment calculations : (loose type flange)

(a) For operating conditions :

$W_Q = W_1 + W_2 + W_3$

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

$W_2 = H - W_1$

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

B= outside shell diameter = 0.357 m

$W_1 = \frac{\pi}{4} (0.357)^2 \times 0.101 \times 10^6 = 10109.91$ N

$W_2 = H - W_1 = 11095.69 – 10109.91 = 985.78$ N

$W_3 = 3916.7$ N

$W_o = 10109.91 + 3916.7 + 985.78 = 15012.39$ N

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

$a_1 = \frac{C - B}{2}$; $a_2 = \frac{a_1 + a_3}{2}$; $a_3 = \frac{C - G}{2}$

C=0.6804; B=0.611; G=0.628

$a_1 = \frac{0.6804 - 0.611}{2} = 0.0347$

$a_3 = \frac{C - G}{2} = \frac{0.6804 - 0.628}{2} = 0.0262$

$a_2 = \frac{a_1 + a_3}{2} = \frac{0.0347 + 0.0262}{2} = 0.0305$

$M_o = 10109.91 \times 0.0347 + 985.78\ (0.0305) + 3916.7\ (0.0262) = 483.49$ J

(b) For bolting up condition :

M_g = Total bolting Moment = $W a_3$

$W = \frac{(A_m + A_b)}{2} S_g$.

$A_m = 1.3812 \times 10^{-3}\ m^2$

$A_b = 44 \times 1.5\ 4 \times 10^{-4} = 67.76 \times 10^{-4}\ m^2$

$S_g = 138 \times 10^6$

W=562846.8 J

$M_g = 562846.8 \times 0.0262 = 14746.58$ J

$M_g > M_o$

$\therefore M_g$ is the moment under operating conditions

$M = M_g = 14766.58$ J

Calculation of the flange thickness:

$$t^2 = \frac{M C_F Y}{B S_{FO}}$$

C_F= Bolt pitch correction factor = $\sqrt{B_s / (2d + t)}$

B_s = Bolt spacing = $\frac{\pi C}{n} = \frac{\pi(0.6804)}{44} = 0.0486$

n= number of bolts.

Let $C_F = 1$

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

$S_{FO} = 100 \times 10^6$ N

M = 14766.58 J

B = 0.357

$$K = \frac{A}{B} = \frac{\text{Flange diameter}}{\text{Inner Shell diameter}} = \frac{0.4643}{0.357} = 1.3005$$

Y = 24

$$t = \sqrt{\frac{14766.58 \times 1 \times 24}{0.357 \times 100 \times 10^6}} = 0.0996 \text{ m}$$

d = 18 x 2 = 36mm

$$C_F = \sqrt{\frac{0.0486}{2(36 \times 10^{-3}) + 0.0996}} = 0.2883$$

$C_F^{1/2} = 0.53218$

$t = 0.0996 \times 0.53218 = 0.0530$ m

Let $t = 60$mm = 0.06m

<u>Tube sheet thickness</u> : (Cylindrical Shell) .

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

G_c = mean gasket diameter for cover.

P = design pressure.

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

F = permissible stress at design temperature.

$t_{1s} = 0.374 \sqrt{(0.25 \times 0.101 \times 10^6) / (95 \times 10^6)} = 0.0061$ m

<u>Channel and channel Cover</u>

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

= 0.00667 m =6.67mm

Consider corrosion allowance = 4 mm.

$t_h = 0.004 + 0.00667 = 0.0106$ m.

<u>Saddle support</u>

Material: Low carbon steel

Total length of shell: 4.88 m

Diameter of shell: 0.357 m

Knuckle radius = 0.06 x 0.357 = 0.02142 m = r_o

Total depth of head (H)= $\sqrt{(D_o r_o / 2)}$

$= \sqrt{(0.357 \times 0.02142 / 2)}$

= 0.0618 m

Weight of the shell and its contents = 1077.004 kg = W

R=D / 2=0.1785 mm

Distance of saddle center line from shell end = A =0.5R=0.0892 m.

Longitudinal Bending Moment

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

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

= (1077.004 / 2) X (4.88 + 4 x 0.106 / 3)

= 2703.99 kg m

M_1= 5.519 kg-m

Bending moment at center of the span

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

M_2 =2970.37 kg-m

Stresses in shell at the saddle

(a) At the topmost fibre of the cross section

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

$= 5.519 / (3.14 \times 0.1785^2 \times 0.008)$

= 6891.98 kg / m^2

The stresses are well within the permissible values.

Stress in the shell at mid point

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

$= 3709319.558\ kg / m^2$

Axial stress in the shell due to internal pressure

$f_p = PD / 4t$

$= 0.101 \times 10^6 \times 0.337\ / 4 \times 0.008$

$= 1063656.2\ kg / m^2$

$f_2 + f_p = 4772976.108\ kg / m^2$

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