1. QUADRUPLE EFFECT EVAPORATOR

From material balance,
Feed to the first effect = F = 238 tons/hr
$= (238 \times 1000) \div 3600$
$= 66.11$ kg/s
Fraction of solids in this feed = $X_f = 0.146$
Liquid output from the last effect = $L_4 = 57.12$ tons/hr
$= (57.12 \times 1000) \div 3600$
$= 15.865$ kg/s
Fraction of solids in this output = $X_4 = 0.6084$
Therefore evaporator load = 66.11-15.865 = 50.245 kg/s
Assume equal amount of water is vapourized in each effect.
Therefore $V_1 = V_2 = V_3 = V_4 = 50.245 \div 4 = 12.5613$ kg/s

Material balance in each effect is as follows:

I Effect

Liquid outlet from first effect = $L_1 = F - V_1$
$= 66.11 - 12.5613$
$= 53.5487$ kg/s
Now the solid balance is, $F \times X_f = L_1 \times X_1$
Therefore $X_1 = (66.11 \times 0.146) \div 53.5487 = 0.1803$
In this liquid outlet, the solids per 100 parts of water = $(18.03 \times 100) \div (100 - 18.03)$
$= 22$
Therefore boiling point rise[1] = $BPR_1 = 0.5^oC$

II Effect

Liquid outlet from second effect = $L_2 = L_1 - V_2$
$= 53.5487 - 12.5613$
$= 40.9874$ kg/s
Now the solid balance is, $L_1 \times X_1 = L_2 \times X_2$
Therefore $X_2 = (53.5487 \times 0.1803) \div 40.9874 = 0.2355$
In this liquid outlet, the solids per 100 parts of water = $(23.55 \times 100) \div (100 - 23.55)$
$= 30.80$
Therefore boiling point rise[1] = $BPR_2 = 0.7^oC$

III Effect
Liquid outlet from third effect = $L_3 = L_2 - V_3$
$= 40.9874 - 12.5613$
$= 28.4261$ kg/s
Now the solid balance is, $L_2 \times X_2 = L_3 \times X_3$
Therefore $X_3 = (40.9874 \times 0.2355) \div 28.4261 = 0.3396$

In this liquid outlet, the solids per 100 parts of water = $(33.96 \times 100) \div (100 - 33.96)$
$= 51.42$

Therefore boiling point rise[1] = $BPR_3 = 1.2^oC$

IV Effect

Liquid outlet from fourth effect = $L_4 = 15.865$ kg/s
Now the solid balance is, $L_3 \times X_3 = L_4 \times X_4$
Therefore $X_4 = 0.6084$
In this liquid outlet, the solids per 100 parts of water = $(60.84 \times 100) \div (100 - 60.84)$
$= 156.36$

Therefore boiling point rise[1] = $BPR_4 = 3.6^oC$

<u>To calculate Δt</u>

Steam is available to first effect at 9 psig.
i.e. $P_s = 9 + 14.695$
$= 23.695$ psia
But 1 psia = 6894.8 N/m^2
Thus $P_s = 23.695 \times 6894.8 = 163.372\ kN/m^2 = 1.634$ bar
Therefore $T_{1s} = 387^oK = 113.89^oC$

The pressure in the vapour space of the 4th effect is at a vacuum of 26 in Hg.
i.e. $P_4 = 30 - 26 = 4$ in Hg absolute
$= 1.96$ psia
$= 13.56$ kPa
$= 0.1356$ bar
Thus $T_{5s} = 325^oK = 51.85^oC$
Therefore $\Delta t = T_{1s} - T_{5s} = 113.89 - 51.85 = 62.04^oC$
Effective $\Delta t = \Delta t - (BPR_1 + BPR_2 + BPR_3 + BPR_4)$
$= 62.04 - (0.5 + 0.7 + 1.2 + 3.6)$
$= 56.04^oC$

<u>To calculate Δt in each effect</u>

Neglecting the sensible heat necessary to heat the feed to the boiling point, approximately all the latent heat of condensing steam appears as latent heat in the vapour.
Hence $q_1 = q_2 = q_3 = q_4$
$U_1A_1\Delta t_1 = U_2A_2\Delta t_2 = U_3A_3\Delta t_3 = U_4A_4\Delta t_4$
Usually, the areas in all effects are equal.
Therefore $U_1\Delta t_1 = U_2\Delta t_2 = U_3\Delta t_3 = U_4\Delta t_4$
According to Hugot, the overall heat transfer coefficients in each effect are given as follows:

Effect	1	2	3	4
U [$Btu/hrft^{2o}F$]	400-500	275-375	200-275	125-150

Assuming overall heat transfer coefficient in each effect as follows:

Effect	1	2	3	4
U [Btu/hrft^{2o}F]	450	325	250	140
U (w/m^{2o}K)	2555	1845	1420	795

Therefore $(\Delta t_2 \div \Delta t_1) = U_1 \div U_2 = 2555 \div 1845 = 1.385$
$\Delta t_2 = 1.385 \times \Delta t_1$
$\Delta t_3 \div \Delta t_2 = U_2 \div U_3 = 1845 \div 1420 = 1.299$
$\Delta t_3 = 1.299 \times \Delta t_2 = 1.299 \times 1.385 \times \Delta t_1$
$\Delta t_4 \div \Delta t_3 = U_3 \div U_4 = 1420 \div 795 = 1.7862$
$\Delta t_4 = 1.7862 \times \Delta t_3 = 1.7862 \times 1.299 \times 1.385 \times \Delta t_1$
But $\Delta t_1 + \Delta t_2 + \Delta t_3 + \Delta t_4 = 56.04$
$\Delta t_1 (1 + 1.385 + 1.299 \times 1.385 + 1.299 \times 1.385 \times 1.7862) = 56.04$
Therefore $\Delta t_1 = 7.5753 = 7.58$
$\Delta t_2 = 10.4918 = 10.49$
$\Delta t_3 = 13.6289 = 13.63$
$\Delta t_4 = 24.3439 = \underline{24.34}$
56.04

<u>Boiling point of solution in each effect</u>

To calculate the actual boiling point of the solution in each effect

<u>I Effect</u>

Boiling point of solution = $t_1 = T_{1s} - \Delta t_1$
Where T_{1s} = saturated temperature of steam to first effect = 113.89^{o}C
Therefore $t_1 = 113.89 - 7.58 = 106.31^oC$

<u>II Effect</u>

T_{2s} = saturated temperature of steam to second effect = $t_1 - BPR_1$
= 106.31 – 0.5
= 105.81^{o}C
Boiling point of solution = $t_2 = T_{2s} - \Delta t_2$
= 105.81 – 10.49
= 95.32^{o}C

<u>III Effect</u>
T_{3s} = saturated temperature of steam to third effect = $t_2 - BPR_2$
= 95.32 – 0.7
= 94.62^{o}C
Boiling point of solution = $t_3 = T_{3s} - \Delta t_3$
= 94.62 – 13.63

$= 80.99^oC$

IV Effect

T_{4s} = saturated temperature of steam to fourth effect = $t_3 - BPR_3$

$= 80.99 - 1.2$

$= 79.79^oC$

Boiling point of solution = $t_4 = T_{4s} - \Delta t_4$

$= 79.79 - 24.34$

$= 55.45^oC$

T_{5s} = saturated temperature of steam to condenser = $t_4 - BPR_4$

$= 55.45 - 3.6$

$= 51.85^oC$

Effect 1	Effect 2	Effect 3	Effect 4	Condenser
$T_{1s} = 113.89^oC$	$T_{2s} = 105.81^oC$	$T_{3s} = 94.62^oC$	$T_{4s} = 79.79^oC$	$T_{5s} = 51.85^oC$
$t_1 = 106.31^oC$	$t_2 = 95.32^oC$	$t_3 = 80.99^oC$	$t_4 = 55.45^oC$	

Heat Balance

Heat capacity of sugar solution

The heat capacity of sugar solution in each effect is calculated from equation

$C_p = 4.19 - 2.35 \times X$ (kJ/kg^{o}K)

Weight fraction of dissolved solids	Heat capacity (kJ/kg^{o}K)
$X_f = 0.146$	$C_{pf} = 3.8469$
$X_1 = 0.1803$	$C_{p1} = 3.7663$
$X_2 = 0.2355$	$C_{p2} = 3.6366$
$X_3 = 0.3396$	$C_{p3} = 3.3919$
$X_4 = 0.6084$	$C_{p4} = 2.7603$

Enthalpy data

Temperature (^{o}C)	Enthalpy of saturated water vapour (H_s kJ/kg)	Enthalpy of saturated liquid water (h_s kJ/kg)
$T_{1s} = 113.89$	$H_{1s} = 2698.0$	$h_{1s} = 479.8$
$T_{2s} = 105.81$	$H_{2s} = 2638.8$	$h_{2s} = 438.8$
$T_{3s} = 94.62$	$H_{3s} = 2667.5$	$h_{3s} = 396.7$
$T_{4s} = 79.79$	$H_{4s} = 2643.0$	$h_{4s} = 329.7$
$T_{5s} = 51.85$	$H_{5s} = 2595.0$	$h_{5s} = 217.0$

Heat capacity of steam in each effect (from steam table)

Temperature in each effect (^{o}C)	Specific heat of steam (kJ/kg^{o}K)
$t_1 = 106.31$	2.0480
$t_2 = 95.32$	2.0090
$t_3 = 80.99$	1.9630
$t_4 = 55.45$	1.9075

I Effect

The heat balance for first effect is as follows:

$S\lambda_{1s} + Fh_f = V_1H_1 + L_1h_1$

$\lambda_{1s} = H_{1s} – h_{1s} = 2698.0 – 479.8 = 2218.2$ kJ/kg

F = 66.11 kg/s

Taking reference temperature as 0^oC and feed to the first effect is at 96^oC.

Thus $T_f = 96^oC$

$h_f = C_{pf} \times (T_f – 0) = 3.8469 \times 96 = 368.504$ kJ/kg

$H_1 = H_{2s} + C_p \times BPR_1$

Where C_p is heat capacity of steam.

$H_1 = 2638.8 + 2.048 \times 0.5 = 2684.82$ kJ/kg

$h_1 = C_{p1} \times t_1 = 3.7663 \times 106.31 = 400.4$ kJ/kg

$L_1 = 66.11 – V_1$

Therefore $S \times 2218.2 + 66.11 \times 368.504 = V_1 \times 2684.82 + (66.11 – V_1) \times 400.4$

$2218.2 \times S – 2284.42 \times V_1 = 2042.53$

$S – 1.0299 \times V_1 = 0.9208$ --(1)

II Effect

The heat balance for second effect is as follows:

$V_1\lambda_{2s}+ L_1h_1 = V_2H_2 + L_2h_2$

$\lambda_{2s} = H_1 – h_{2s} = 2684.82 – 438.8 = 2246.02$ kJ/kg

$H_2 = H_{3s} + C_p \times BPR_2$

Where C_p is heat capacity of steam.

$H_2 = 2667.5 + 2.009 \times 0.7 = 2668.91$ kJ/kg

$h_2 = C_{p2} \times t_2 = 3.6366 \times 95.32 = 346.641$ kJ/kg

$L_2 = 66.11 – V_1 – V_2$

Therefore

$V_1 \times 2246.02 + (66.11 – V_1) \times 400.4 = V_2 \times 2668.91 + (66.11 – V_1 – V_2) \times 346.641$

$V_1 – 1.0591 \times V_2 = – 1.6212$ ---(2)

III Effect

The heat balance for third effect is as follows:

$V_2\lambda_{3s}+ L_2h_2 = V_3H_3 + L_3h_3$

$\lambda_{3s} = H_2 – h_{3s} = 2668.91 – 396.7 = 2272.21$ kJ/kg

$H_3 = H_{4s} + C_p \times BPR_3$

Where C_p is heat capacity of steam.

$H_3 = 2643.0 + 1.963 \times 1.2 = 2645.36$ kJ/kg

$h_3 = C_{p3} \times t_3 = 3.3919 \times 80.99 = 274.71$ kJ/kg

$L_3 = 66.11 – V_1 – V_2 – V_3$

Therefore

$V_2 \times 2272.21 + (66.11 – V_1 − V_2) \times 346.641 = V_3 \times 2645.36 + (66.11 − V_1 − V_2 − V_3) \times 274.71$

$V_3 = 2.0059 - 0.0303 \times V_1 + 0.9281 \times V_2$ --(3)

IV Effect

The heat balance for third effect is as follows:
$V_3\lambda_{4s} + L_3h_3 = V_4H_4 + L_4h_4$
$\lambda_{4s} = H_3 - h_{4s} = 2645.36 - 329.7 = 2315.66$ kJ/kg
$H_4 = H_{5s} + C_p \times BPR_4$
Where C_p is heat capacity of steam.
$H_4 = 2595.0 + 1.9075 \times 3.6 = 2601.867$ kJ/kg
$h_4 = C_{p4} \times t_4 = 2.7603 \times 55.45 = 153.06$ kJ/kg
$L_4 = 66.11 - V_1 - V_2 - V_3 - V_4$
Therefore
$V_3 \times 2315.66 + (66.11 - V_1 - V_2 - V_3) \times 274.71 = 2601.867 \times V_4 + (66.11 - V_1 - V_2 - V_3 - V_4) \times 153.06$
$V_4 = 3.2842 - 0.04968 \times V_1 - 0.04968 \times V_2 + 0.8960 \times V_3$ --------------------------------(4)

But $V_1 + V_2 + V_3 + V_4 = 50.245$ ---(5)

By solving equations (1) to (5), we will get

$V_1 = 11.3015$ kg/s
$V_2 = 12.2015$ kg/s
$V_3 = 12.9880$ kg/s
$V_4 = 13.7540$ kg/s
$S = 12.5602$ kg/s

To calculate areas in each effect

$q_1 = S \times \lambda_{1s} = 12.5602 \times 2218.2 = 27861.04$ kW
$q_2 = V_1 \times \lambda_{2s} = 11.3015 \times 2246.02 = 25383.4$ kW
$q_3 = V_2 \times \lambda_{3s} = 12.2015 \times 2272.21 = 27724.4$ kW
$q_4 = V_3 \times \lambda_{4s} = 12.9880 \times 2315.66 = 30075.79$ kW

$A_1 = q_1 \div (U_1 \times \Delta t_1) = (27861.04 \times 10^3) \div (2555 \times 7.58) = 1438.59\ m^2$
$A_2 = q_2 \div (U_2 \times \Delta t_2) = (25383.4 \times 10^3) \div (1845 \times 10.49) = 1311.53\ m^2$
$A_3 = q_3 \div (U_3 \times \Delta t_3) = (27724.4 \times 10^3) \div (1420 \times 13.63) = 1432.44\ m^2$
$A_4 = q_4 \div (U_4 \times \Delta t_4) = (30075.79 \times 10^3) \div (795 \times 24.34) = 1554.28\ m^2$

The mean area = $A_m = (A_1 + A_2 + A_3 + A_4) \div 4 = 1434\ m^2$

TRAIL 2

Now for this trail, taking the calculated values of vapour evaporated in each effect in last trail as follows:
$V_1 = 11.3015$ kg/s

V_2 = 12.2015 kg/s
V_3 = 12.9880 kg/s
V_4 = 13.7540 kg/s
Material balance in each effect is as follows:

I Effect

Liquid outlet from first effect = $L_1 = F - V_1$
$= 66.11 - 11.3015$
$= 54.8085$ kg/s

Now the solid balance is, $F \times X_f = L_1 \times X_1$
Therefore $X_1 = (66.11 \times 0.146) \div 54.8085 = 0.1761$
In this liquid outlet, the solids per 100 parts of water = $(17.61 \times 100) \div (100 - 17.61)$
$= 21.37$

Therefore boiling point rise = $BPR_1 = 0.5^oC$

II Effect

Liquid outlet from second effect = $L_2 = L_1 - V_2$
$= 54.8085 - 12.2015$
$= 42.6070$ kg/s

Now the solid balance is, $L_1 \times X_1 = L_2 \times X_2$
Therefore $X_2 = (54.8085 \times 0.1761) \div 42.6070 = 0.2265$
In this liquid outlet, the solids per 100 parts of water = $(22.65 \times 100) \div (100 - 22.65)$
$= 29.28$

Therefore boiling point rise = $BPR_2 = 0.7^oC$

III Effect

Liquid outlet from third effect = $L_3 = L_2 - V_3$
$= 42.6070 - 12.9880$
$= 29.6190$ kg/s

Now the solid balance is, $L_2 \times X_2 = L_3 \times X_3$
Therefore $X_3 = (42.6070 \times 0.2265) \div 29.6190 = 0.3259$
In this liquid outlet, the solids per 100 parts of water = $(32.59 \times 100) \div (100 - 32.59)$
$= 48.35$

Therefore boiling point rise = $BPR_3 = 1.15^oC$

IV Effect

Liquid outlet from fourth effect = L_4 = 15.865 kg/s
Now the solid balance is, $L_3 \times X_3 = L_4 \times X_4$
Therefore $X_4 = 0.6084$
In this liquid outlet, the solids per 100 parts of water = $(60.84 \times 100) \div (100 - 60.84)$
$= 156.36$

Therefore boiling point rise = $BPR_4 = 3.6^oC$

To calculate Δt

Steam is available to first effect at 9 psig.
i.e. $P_s = 9 + 14.695$
$= 23.695$ psia
But 1 psia = 6894.8 N/m^2
Thus $P_s = 23.695 \times 6894.8 = 163.372$ kN/m^2 = 1.634 bar
Therefore $T_{1s} = 387^oK = 113.89^oC$

The pressure in the vapour space of the 4th effect is at a vacuum of 26 in Hg.
i.e. $P_4 = 30 - 26 = 4$ in Hg absolute
= 1.96 psia
= 13.56 kPa
= 0.1356 bar
Thus $T_{5s} = 325^oK = 51.85^oC$
Therefore $\Delta t = T_{1s} - T_{5s} = 113.89 - 51.85 = 62.04^oC$
Effective $\Delta t = \Delta t - (BPR_1 + BPR_2 + BPR_3 + BPR_4)$
$= 62.04 - (0.5 + 0.7 + 1.15 + 3.6)$
$= 56.09^oC$

To calculate Δt in each effect

$\Delta t_1^| = \Delta t_1 \times A_1) \div A_m = (7.58 \times 1438.59) \div 1434 = 7.60^oC$
$\Delta t_2^| = (\Delta t_2 \times A_2) \div A_m = (10.49 \times 1311.53) \div 1434 = 9.59^oC$
$\Delta t_3^| = (\Delta t_3 \times A_3) \div A_m = (13.63 \times 1432.44) \div 1434 = 13.61^oC$
$\Delta t_4^| = (\Delta t_4 \times A_4) \div A_m = (24.34 \times 1554.28) \div 1434 = 26.38^oC$
But $\Delta t = \Delta t_1| + \Delta t_2^| + \Delta t_3^| + \Delta t_4^| = 56.09^oC$
Let us take $\Delta t_1^| = 7.52^oC$
$\Delta t_2^| = 9.00^oC$
$\Delta t_3^| = 13.48^oC$
$\Delta t_4^| = \underline{26.09^oC}$
56.09^oC

Boiling point of solution in each effect

To calculate the actual boiling point of the solution in each effect

I Effect

Boiling point of solution = $t_1 = T_{1s} - \Delta t_1^{|}$
Where T_{1s} = saturated temperature of steam to first effect = 113.89°C
Therefore t_1 = 113.89 − 7.52 = 106.37°C

II Effect

T_{2s} = saturated temperature of steam to second effect = $t_1 - BPR_1$
= 106.37 − 0.5
= 105.87°C
Boiling point of solution = $t_2 = T_{2s} - \Delta t_2^{|}$
= 105.87 − 9.00
= 96.87°C

III Effect

T_{3s} = saturated temperature of steam to third effect = $t_2 - BPR_2$
= 96.87 − 0.7
= 96.17°C
Boiling point of solution = $t_3 = T_{3s} - \Delta t_3^{|}$
= 96.17 − 13.48
= 82.69°C

IV Effect

T_{4s} = saturated temperature of steam to fourth effect = $t_3 - BPR_3$
= 82.69 − 1.15
= 81.54°C
Boiling point of solution = $t_4 = T_{4s} - \Delta t_4^{|}$
= 81.54 − 26.09
= 55.45°C
T_{5s} = saturated temperature of steam to condenser = $t_4 - BPR_4$
= 55.45 − 3.6
= 51.85°C

Effect 1	Effect 2	Effect 3	Effect 4	Condenser
T_{1s} = 113.89°C	T_{2s} = 105.87°C	T_{3s} = 96.17°C	T_{4s} = 81.54°C	T_{5s} = 51.85°C
t_1 = 106.37°C	t_2 = 96.87°C	t_3 = 82.69°C	t_4 = 55.45°C	

Heat Balance

Heat capacity of sugar solution
The heat capacity of sugar solution in each effect is calculated from equation

$C_p = 4.19 - 2.35 \times X$ (kJ/kg^{o}K)

Weight fraction of dissolved solids	Heat capacity (kJ/kg^{o}K)
$X_f = 0.146$	$C_{pf} = 3.8469$
$X_1 = 0.1761$	$C_{p1} = 3.7662$
$X_2 = 0.2265$	$C_{p2} = 3.6577$
$X_3 = 0.3259$	$C_{p3} = 3.4241$
$X_4 = 0.6084$	$C_{p4} = 2.7603$

Enthalpy data

Temperature (^{o}C)	Enthalpy of saturated water vapour (H_s kJ/kg)	Enthalpy of saturated liquid water (h_s kJ/kg)
$T_{1s} = 113.89$	$H_{1s} = 2698.0$	$h_{1s} = 479.8$
$T_{2s} = 105.87$	$H_{2s} = 2685.5$	$h_{2s} = 443.6$
$T_{3s} = 96.17$	$H_{3s} = 2669.3$	$h_{3s} = 401.8$
$T_{4s} = 81.54$	$H_{4s} = 2647.0$	$h_{4s} = 342.7$
$T_{5s} = 51.85$	$H_{5s} = 2595.0$	$h_{5s} = 217.0$

Heat capacity of steam in each effect (from steam table)

Temperature in each effect (^{o}C)	Specific heat of steam (kJ/kg^{o}K)
$t_1 = 106.37$	2.0550
$t_2 = 96.87$	2.0170
$t_3 = 82.69$	1.9710
$t_4 = 55.45$	1.9100

I Effect

The heat balance for first effect is as follows:

$S\lambda_{1s} + Fh_f = V_1H_1 + L_1h_1$

$\lambda_{1s} = H_{1s} - h_{1s} = 2698.0 - 479.8 = 2218.2$ kJ/kg

$F = 66.11$ kg/s

Taking reference temperature as 0^{o}C and feed to the first effect is at 96^{o}C.

Thus $T_f = 96^o$C

$h_f = C_{pf} \times (T_f - 0) = 3.8469 \times 96 = 368.504$ kJ/kg

$H_1 = H_{2s} + C_p \times BPR_1$

Where C_p is heat capacity of steam.

$H_1 = 2685.5 + 2.055 \times 0.5 = 2686.53$ kJ/kg

$h_1 = C_{p1} \times t_1 = 3.7662 \times 106.37 = 400.61$ kJ/kg

$L_1 = 66.11 - V_1$

Therefore $S \times 2218.2 + 66.11 \times 368.504 = V_1 \times 2686.53 + (66.11 - V_1) \times 400.61$

$2218.2 \times S - 2285.92 \times V_1 = 2056.42$

$S - 1.0305 \times V_1 = 0.9271$ --(1)

II Effect

The heat balance for second effect is as follows:

$V_1\lambda_{2s} + L_1h_1 = V_2H_2 + L_2h_2$

$\lambda_{2s} = H_1 - h_{2s} = 2686.53 - 443.6 = 2242.93$ kJ/kg

$H_2 = H_{3s} + C_p \times BPR_2$

Where C_p is heat capacity of steam.

$H_2 = 2669.3 + 2.017 \times 0.7 = 2670.71$ kJ/kg

$h_2 = C_{p2} \times t_2 = 3.6577 \times 96.87 = 354.32$ kJ/kg

$L_2 = 66.11 - V_1 - V_2$

Therefore

$V_1 \times 2242.93 + (66.11 - V_1) \times 400.61 = V_2 \times 2670.71 + (66.11 - V_1 - V_2) \times 354.32$

$V_1 - 1.0545 \times V_2 = - 1.3931$ ---(2)

III Effect

The heat balance for third effect is as follows:

$V_2\lambda_{3s} + L_2h_2 = V_3H_3 + L_3h_3$

$\lambda_{3s} = H_2 - h_{3s} = 2670.71 - 401.8 = 2268.91$ kJ/kg

$H_3 = H_{4s} + C_p \times BPR_3$

Where C_p is heat capacity of steam.

$H_3 = 2647.0 + 1.971 \times 1.15 = 2649.27$ kJ/kg

$h_3 = C_{p3} \times t_3 = 3.4241 \times 82.69 = 283.14$ kJ/kg

$L_3 = 66.11 - V_1 - V_2 - V_3$

Therefore

$V_2 \times 2268.91 + (66.11 - V_1 - V_2) \times 354.32 = V_3 \times 2649.27 + (66.11 - V_1 - V_2 - V_3) \times 283.14$

$V_3 = 1.9888 - 0.0301 \times V_1 + 0.9288 \times V_2$ ---(3)

IV Effect

The heat balance for third effect is as follows:

$V_3\lambda_{4s} + L_3h_3 = V_4H_4 + L_4h_4$

$\lambda_{4s} = H_3 - h_{4s} = 2649.27 - 342.7 = 2306.57$ kJ/kg

$H_4 = H_{5s} + C_p \times BPR_4$

Where C_p is heat capacity of steam.

$H_4 = 2595.0 + 1.910 \times 3.6 = 2601.876$ kJ/kg

$h_4 = C_{p4} \times t_4 = 2.7603 \times 55.45 = 153.06$ kJ/kg

$L_4 = 66.11 - V_1 - V_2 - V_3 - V_4$

Therefore

$V_3 \times 2306.57 + (66.11 - V_1 - V_2 - V_3) \times 283.14 = 2601.876 \times V_4 + (66.11 - V_1 - V_2 - V_3 - V_4) \times 153.06$

$V_4 = 3.5117 - 0.05310 \times V_1 - 0.05310 \times V_2 + 0.8888 \times V_3$ --------------------------------(4)

But $V_1 + V_2 + V_3 + V_4 = 50.245$ --(5)

By solving equations (1) to (5), we will get

V_1 = 11.0969 kg/s
V_2 = 12.3980 kg/s
V_3 = 13.0216 kg/s
V_4 = 13.7285 kg/s
S = 12.3625 kg/s

<u>To calculate areas in each effect</u>

$q_1 = S \times \lambda_{1s} = 12.3625 \times 2218.2 = 27422.50$ kW
$q_2 = V_1 \times \lambda_{2s} = 11.0969 \times 2242.93 = 24889.57$ kW
$q_3 = V_2 \times \lambda_{3s} = 12.3980 \times 2268.91 = 33461.09$ kW
$q_4 = V_3 \times \lambda_{4s} = 13.0216 \times 2306.57 = 30035.23$ kW

$A_1 = q_1 \div (U_1 \times \Delta t_1') = (27422.50 \times 10^3) \div (2555 \times 7.52) = 1427.24\ m^2$
$A_2 = q_2 \div (U_2 \times \Delta t_2') = (24889.57 \times 10^3) \div (1845 \times 9.00) = 1498.90\ m^2$
$A_3 = q_3 \div (U_3 \times \Delta t_3') = (33461.09 \times 10^3) \div (1420 \times 13.48) = 1470.00\ m^2$
$A_4 = q_4 \div (U_4 \times \Delta t_4') = (30035.79 \times 10^3) \div (795 \times 26.09) = 1448.00\ m^2$

The mean area = $A_m = (A_1 + A_2 + A_3 + A_4) \div 4 = 1461\ m^2$
All areas are within 5% of mean area. Thus this is acceptable.

Tube details

Most generally used diameters today ranges from 1.25 to 2.00 in. outer diameter and most generally used lengths of tubes ranges from 4 to 15 ft.

Let us choose 2 in. nominal diameter, 80 schedule, brass tubes of 12 ft length.

Therefore Outer diameter = d_o = 60.325 mm
Inner diameter = d_i = 49.2506 mm

Length = L = 12 ft = 3.6576 m

Tube pitch(Δ) = $P_T = 1.25 \times d_o = 1.25 \times 2 = 2.5$ in. =63.5 mm

Surface area of each tube = $a = \pi d_o L = \pi \times 60.325 \times 10^{-3} \times 3.6576$
$= 0.6932\ m^2$

Number of tubes required = $N_t = A \div a = 1461 \div 0.6932 = 2108$
Area occupied by tubes = $N_t \times (1/2) \times P_T \times P_T \times \sin\propto$
$= 2108 \times 0.5 \times (63.5 \times 10^{-3})^2 \times 0.866$
$= 3.6806\ m^2$
Where $\propto = 60^o$
But actual area is more than this. Hence this area is to be divided by factor which varies from 0.8 to 1.0.
Let us this factor as 0.9.

Therefore actual area required = 3.6806 ÷ 0.9 = 4.09 m^2

The central downcomer area is taken as 40 to 70% of the total cross sectional area of tubes. Let us take it as 50%.

Therefore Downcomer area = $0.5 \times [N_t \times (\pi/4) \times d_o^2]$

$= 0.5 \times [2108 \times (\pi/4) \times (0.060325)^2]$

$= 3.0125\ m^2$

Downcomer diameter = $\sqrt{(4 \times 3.0125)} \div \pi = 1.96$ m

Total area of tube sheet in evaporator = downcomer area + area occupied by tubes

= 3.0125 + 4.09

= 7.1025 m^2

Thus tube sheet diameter = $\sqrt{(4 \times 7.1025)} \div \pi = 3.00$ m

2. CONDENSER

The condenser is a horizontal condenser designed to condense 14.1295 kg/s of water vapour coming from the fourth effect of evaporator at a temperature of 55.45°C and at a pressure of 13.56 kPa. At this pressure the saturation temperature is 51.45°C. But the degree of superheat is very less; hence assuming water vapour is at saturated temperature only. The coolant used is water, which is supplied at an inlet temperature of 30°C and leaves at an outlet temperature of 40°C.

(1) Heat balance

Latent heat of vapourisation of water at 51.45°C = λ = 2378 kJ/kg
Amount of heat removed from the vapour = Q = $m_v \times \lambda$
= 14.1295 × 2378
= 33599.95 kW

Amount of water to be circulated (m_w)**:**
$m_w \times C_p \times \Delta t = m_v \times \lambda = Q$
At average temperature i.e. 35^{o}C, C_p of water = 4.182 kJ/kg^{o}K
$m_w \times 4.182 \times 10^3 \times (40 - 30) = 33599.95$ kW
$m_w = (33599.95 \times 1000) / (4.182 \times 10^3 \times 10)$
m_w = 803.44 kg/s
Amount of water required = 803.44 kg/s

(2) Logarithmic mean temperature difference (LMTD)

$\Delta T_{LMTD} = [(51.85 - 30) - (51.85 - 40)] \div [\ln \{(51.85 - 30) \div (51.85 - 40)\}]$
= 16.34°C

Since condensation occurs at constant temperature, hence correction factor is not needed.

(3) Routing of fluids

Allocating the fluids with the lowest flow rate to the shell side will normally give the most economical design and also the higher pressure stream should be allocated to the tube side. High pressure tubes will be cheaper than a high pressure shell.

Here water is available at atmospheric pressure and steam is available at vacuum. Hence take steam on shell side and water on tube side.

(4) Heat transfer area

From table 10-10 (Perry hand book), the designing overall heat transfer coefficient for steam-water system is in the range of 400 – 1000 Btu/(hrft^{2o}F) i.e. 2271.32 – 5678.3 W/(m^{2o}K) and it includes total dirt factor 0.0005 (hrft^{2o}F)/Btu i.e. $R_d = 8.81 \times 10^{-5}$ (m^{2o}K)/W

Assume U_d = 2300 W/m^2°K
Total heat transfer area = A = $Q/(U \times \Delta T_{LMTD})$
= (33599.95 × 1000) ÷ (2300 × 16.34)
= 894.04 m^2

Let us choose tubing characteristics as follows:
1" O.D, 20 BWG , length of 10 ft laid on a 5/4" Δ pitch.
Outer diameter of tube = d_o = 1" = 25.4 mm
Inner diameter of tube = d_i = 0.93" = 23.622 mm

External surface area = 0.2618 ft^2/ft = 0.0798 m^2/m
Allowing 50 mm thickness for tube sheets. Now length of tube is 10 ft i.e 3.048 m
Thus length available for heat transfer = L = 3.048 – 0.050 = 2.998 m
Area per tube = 0.0798 × 2.998 = 0.2392 m^2/tube

Therefore number of tubes = N_t = 894.04 ÷ 0.2392
= 3737
From tube count table,
For tube O.D. of 1" on 5/4" triangular pitch
TEMA L or M
No. of shell passes = 1
No. of tube passes = N_p = 4
Nearest tube count = N_t = 3784
I.D. of shell = 2134 mm
Corrected heat transfer area = 3784 × 0.2392
A = 905.13 m^2
Corrected $U_d = Q \div (A \times \Delta T_{LMTD})$
= (33599.95 × 1000) ÷ (905.13 × 16.34)
= 2271.82 W/m^{2}°K

(5) Film heat transfer coefficients

(a) Shell side – condensing vapours

Temperature of vapour coming in = T_s = 51.85ºC
Average temperature of water = (40 + 30)/2
= 35ºC
Wall temperature = T_w = (51.85 + 35)/2
= 43.425ºC
Film temperature = T_f = (Ts + Tw) ÷ 2 = (51.85 + 43.425) ÷ 2 = 47.64ºC
So the property of water are taken at film temperature i.e 47.64ºC
Viscosity = μ = 0.63 ×10^{-3} Ns/m^2
Thermal conductivity = k = 0.6371 W/mºK
Density = ρ = 989.103 kg/m^3
Specific heat = C_p = 4.184 kJ/kgºK
Mass flow rate per unit length = $m_v \div (N_t^{2/3} \times L)$
= $14.1295 \div (3784^{2/3} \times 2.998)$
= 19.41 × 10^{-3} kg/(m ×s)
Renold's number = N_{re} = (4 × 19.568 × 10^{-3}) ÷ (0.63 × 10^{-3}) = 123.23

Outer film coefficient = $h_o = 1.51(k^3 \times \rho^2 \times g / \mu^2)^{1/3}(N_{re})^{-1/3}$
Thus $h_o = 1.51[(0.6371^3 \times 989.103^2 \times 9.81) \div (0.63 \times 10^{-3})^2]^{1/3}(123.23)^{-1/3}$
= 1.51 × 18423.18 × 0.201
= 5590.32 W/m^{2}°K

(b) Tube side

Average temperature of water = 35ºC
Physical properties of water at 35ºC
Specific heat = C_p = 4.22584 kJ/kgºK
Viscosity = μ = 0.77×10^{-3} Ns/m^2
Thermal conductivity = k = 0.6256 W/mºK
Density = ρ = 994.032 kg/m^3
Prandtl number = $N_{pr} = (C_p \times \mu)/k$
$= (4.22584 \times 1000 \times 0.77 \times 10^{-3}) \div 0.6256$
$= 5.20$
Mass velocity of water = m_w = 803.44 kg/s
Flow area = $a_t = \{(\pi \times d_i^2) \div 4\} \times (N_t \div N_p)$
$= \{(\pi \times 0.0254^2) \div 4\} \times (3784 \div 4)$
$= 0.4146$ m^2/pass
Water velocity = $v = m_w \div (\rho \times a_t)$
$= 803.44 \div (994.032 \times 0.4146)$
$= 1.95$ m/s
Renold's number = $N_{Re} = (d_i \times \rho \times v) \div \mu$
$= (0.0254 \times 994.032 \times 1.95) \div (0.77 \times 10^{-3})$
$= 59464.93 > 10000$
Dittus Boletere equation can be used
$Nu = 0.023 \times (N_{Re})^{0.8} \times (N_{pr})^{0.3}$
$= 0.023 \times 59464.93^{0.8} \times 5.2^{0.3}$
$= 248.85$
$h_i = (248.85 \times 0.6256) \div 0.0254 = 6590.49$ $W/m^{2o}K$

(6) Overall heat transfer coefficient

$1/U_o = (1/h_o) + (1/h_i) \times (d_o/d_i) + R_d$
$= (1/5590.32) + (1/6590.49) \times (25.4/23.622) + (8.81 \times 10^{-5})$
$= 4.301 \times 10^{-4}$ $m^{2o}K/W$
Therefore U_o = 2324.85 $W/m^{2\circ}K$
Required area = $A = Q \div (U_o \times \Delta T_{LMTD})$
$= (33599.95 \times 1000) \div (2324.85 \times 16.34)$
$= 884.49$ m^2
Which is less than the corrected (available) heat transfer area, therefore this value of U_o is good enough.

(7) Pressure drop calculations

(a) Tube side

The Renold's number as calculated before for tube side is = N_{re} = 59464.93

Now $f = 0.079(N_{re})^{-1/4}$

$= 0.079 \times (59464.93)^{-1/4}$

$= 0.0051$

The pressure drop due to friction is given by

$\Delta P_f = [(4 \times f \times L \times v^2) \div (2 \times g \times d_i)] \times \rho \times g$

$= [(4 \times 0.0051 \times 2.998 \times 1.95^2) \div (2 \times 9.81 \times 0.0254)] \times 994.032 \times 9.81$

$= 4853.74\ N/m^2$

The pressure drop due to the velocity of fluid is given by

$\Delta P_v = (2.5/2) \times \rho \times v^2$

$= (2.5/2) \times 994.032 \times (1.95)^2$

$= 4724.76\ N/m^2$

Therefore total pressure drop is given by

$\Delta P_t = N_p \times (\Delta P_f + \Delta P_v)$

$= 4 \times (4853.74 + 4724.76)$

$= 38.313\ kPa < 70\ kPa$

This is within the permissible limit of a maximum pressure drop of 70 kPa in the tube side for no phase change.

So this pressure drop is acceptable.

(b) Shell side

Mass flow rate of water vapour = m_v = 14.1295 kg/s

Saturation temperature of vapour = T_{vap}= 51.85ºC

Viscosity of water vapour = $\mu_{vap} = 1.1 \times 10^{-5}\ Ns/m^2$

Density of vapour = ρ_{vap} = (PM/RT) = 0.7 kg/m^3

Flow area for shell side = $S_m = \{(P_t - d0) \times l_s \times D_s\} \div P_t$

Where P_t = pitch = 1.25" = 31.75 mm

d_0 = 25.4 mm

l_s = baffle spacing = 0.2 D_s to D_s

D_s = shell diameter = 2.134 m

Assume $l_s = 0.7\ Ds = 0.7 \times 2.134 = 1.4938$ m

Therefore $S_m = \{(31.75 - 25.4) \times 1.4938 \times 2.134\} \div 31.75$

$= 0.7969\ m^2$

Equivalent diameter = $D_e = 4 \times [\{(0.866/2)\ P_t^2\} - \{(1/2)(\pi \times d_o^2)/4\}] \div \{(\pi \times d_o)/2\}$

$= 0.01836$ m

$G_s = m_v \div S_m = 14.1295/ 0.7969 = 17.7306\ kg/sec\ m^2$

$(N_{re})_{vap} = (G_s \times D_e)/\mu_{vap}$

$= (17.7306 \times 0.01836)/(1.1 \times 10^{-5})$

$= 29593.95$

From fig 10-25(a), p 10-31 (Perry hand book)

$f_k = 0.16$

Number of baffles, now $N_b+1 = L/l_s$

$= 2.998/1.4938$

$= 2$

Therefore $N_b = 2$

To calculate pressure drop for shell side, we have to consider three zones.

1) End zones, ΔP_e (there are two end zones)

2) Cross flow zones, ΔP_c, $(N_b - 1)$ crosses

3) Window zones, ΔP_w (N_b zones)

Thus total pressure drop is

$$(\Delta P_s)_t = 2 \times \Delta P_e + (N_b - 1)\, \Delta P_c + N_b \times \Delta P_w$$

1) ΔP in cross flow section

$$\Delta P_c = \{(b \times f_k \times m_v^2 \times N_c) \div (\rho \times S_m^2)\} \times (\mu_w / \mu_b)^{0.14}$$

Where b = 0.002

$f_k = 0.16$

$N_c = \{D_s \times [1 - 2 \times (l_c/D_s)]\}/P_p$

l_c = baffle cut = $0.25 \times D_s = 0.5335$ m

P_p = pitch parallel to flow = $(\sqrt{3}/2) \times P_t = 0.866 \times 31.75 = 27.4963$ mm

Therefore N_c = number of tubes row crossed in one cross flow section

$= \{2.134 \times [1 - (2 \times 0.25)]\} \div 0.0274963$

$= 39$

Thus $\Delta P_c = [0.002 \times 0.16 \times (14.1295)^2 \times 39] \div [0.7 \times (0.7969)^2]$

$= 5.6048$ kPa

2) ΔP in end zones

$$\Delta P_e = \Delta P_c \times [1 + (N_{cw} \div N_c)]$$

Where N_{cw} = number of effective cross flow rows in each window

$= (0.8 \times l_c) \div P_p$

$= (0.8 \times 0.5335) \div 0.0275$

$= 15.52$ i.e. 16

Thus $\Delta P_e = 5.6048 \times [1 + (16/39)] = 7.904$ kPa

3) ΔP in window zones

$$\Delta P_w = [b \times m_v^2 \times (2 + 0.6\, N_{cw})] \div (S_m \times S_w \times \rho)$$

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

S_w = area of flow through windows = $S_{wg} - S_{wt}$

S_{wg} = gross window area

S_{wt} = area occupied by tubes

$= (N_t/8) \times (1 - F_c) \times \pi \times d_0^2$

F_c = fraction of tubes in each cross flow

Here $l_c/D_s = 0.25$ and $D_s = 84$"

From fig 10-18, p 10-29 (Perry hand book)

$S_{wg} = 1000\ in^2 = 0.6452\ m^2$

From fig 10-16, p 10-28

Fc = 0.66

Therefore $S_{wt} = (3784/8) \times (1 - 0.66) \times \pi \times (0.0254)^2$

$= 0.3259\ m^2$

Thus $S_w = 0.6452 - 0.3259 = 0.3193\ m^2$

Hence $\Delta P_w = [5 \times 10^{-4} \times (14.1295)^2 \times (2 + 0.6 \times 16)] \div [0.7969 \times 0.3193 \times 0.7]$

$= 6.5$ kPa

$(\Delta P_s)_t = 2 \times 7.904 + 0 + 1 \times 6.5 = 22.31$ kPa

But actual pressure drop is 40% of this value.

Therefore $(\Delta P_s)_t = 0.4 \times 22.31 = 8.92$ kPa

This is also within the permissible limit of a maximum pressure drop of 14 kPa.

So this is acceptable.

Mechanical Design of Two Equipments

1. EVAPORATOR

Take standard vertical short tube evaporator (calendria type)

Data

Evaporator drum operating at 1.2074 bar ($T_{2s} = 105.87^oC$)

Amount of water to be evaporated = 11.0969 kg/s = 43188.84 kg/hr

Heating surface required = A = 1461 m^2

Steam is available to first effect at pressure of 1.634 bar.

Density of liquid = 1400 kg/m^3

Density of water vapour = PM/RT = $(1.2074 \times 105 \times 18) \div [8314 \times (106.37 + 273.15)]$

$= 0.6890\ kg/m^3$

Design pressure = 5% extra of maximum working pressure

$= 1.05 \times 1.634$

$= 1.7157$ bar

$= 1.7856\ kgf/cm^2$

Material

Evaporator – low carbon steel

Tubes – brass

Permissible stress for low carbon steel = 980 kg/cm^2

Modulus of elasticity for low carbon steel = $19 \times 10^5\ kg/cm^2$

Modulus of elasticity for brass = $9.5 \times 10^5\ kg/cm^2$

Conical head at bottom

Cone angle – 120^o

Conical head at top

Cone angle – 120^o

(1) Calendria sheet thickness

The thickness is given by

$t_s = (PD_o) \div (2fJ + P)$

$= (1.7856 \times 3000) \div [(2 \times 980 \times 0.85) + 1.7856]$

$= 3.22$ mm

The actual thickness must be much more so as to allow for corrosion and give rigidity to the shell.

Therefore it may be taken as $t_s = 12$ mm

(2) Tube sheet thickness

To find tube sheet thickness

$K = [E_s \times t_s \times (D_o - t_s)] \div [E_t \times N_t \times t_t \times (d_o - t_t)]$

Where E_s = elastic modulus of shell

E_t = elastic modulus of tube

D_o = outside diameter of shell = 3 m

d_o = outside diameter of tube = 60.325 mm

t_s = shell thickness = 12 mm

t_t = tube wall thickness = 5.5 mm

N_t = number of tubes in shell = 2108

Therefore $K = [19 \times 105 \times 12 \times (3000 - 12)] \div [9.5 \times 105 \times 2108 \times 5.5 \times (60.325 - 5.5)$

$= 0.1128$

$F = \sqrt{[(2 + K) \div (2 + 3 \times K)]}$

$= \sqrt{[(2 + 0.1128) \div (2 + 3 \times 0.1128)]}$

$= 0.951$

The effective tube sheet thickness is given by

$t_{ts} = FD_o \sqrt{[(0.25 \times P) \div f]}$

$= 0.951 \times 3000 \times \sqrt{[(0.25 \times 1.7856) \div 980]}$

$= 60.89$ mm

with corrosion allowance the thickness may be taken as 64 mm.

(3) Check for tube thickness

The tube thickness is given by

$t_t = Pd_i \div (2fJ - P)$

The permissible stress for brass = 538 kg/cm^2 and J = 1

Therefore $t_t = (1.7856 \times 49.25) \div [(2 \times 538 \times 1) - 1.7856]$

$= 0.082$ mm

But provided thickness is 5.5 mm. Therefore chosen tubes have strength to withstand with operating conditions.

(4) Evaporator drum diameter

The following equations help to determine the drum diameter. The diameter of the drum may be same as that of the calendria. However it is necessary to check the size from the point of satisfactory entrainment separation.

$R_d = (V/A) \div [0.0172 \times \sqrt{\{(\rho_l - \rho_v) \div \rho_v\}}]$

Where V – volumetric flow rate of vapour in m^3/s

A – cross sectional area of drum

For drums having wire mesh as entrainment separator device, Rd may be taken as 1.3.

$$A = V \div [R_d \times 0.0172 \times \sqrt{\{(\rho_l - \rho_v) \div \rho_v\}}]$$

$$= [43188.84 \div (3600 \times 0.689)] \div [1.3 \times 0.0172 \times \sqrt{\{(1200 – 0.689) \div 0.689\}}]$$

$$= 18.66 \text{ m}^2$$

Therefore drum diameter $= \sqrt{\{(4 \times 18.66) \div л\}}$

$= 4.87$ m

Which is very large and therefore taking the drum diameter same as that of calendria.

Drum height can be taken as 2 to 5 times of tube sheet diameter.

Thus drum height = $2 \times 3 = 6$ m

(5) Drum thickness

Drum is operating at 1.2074 bar. Design is therefore based on an external pressure of 1.7856 kg/cm^2 (Design pressure).
Assume thickness of 10 mm.
The critical pressure is given by

$$P_c = [2.42 \times E \times (t \div D_o)^{2.5}] \div [(1 - \mu^2)^{3/4} \times \{(L \div D_o) - 0.45 \times (t \div D_o)^{0.5}\}]$$
$$= [2.42 \times 19 \times 10^5 \times (10 \div 3020)^{2.5}] \div [(1 - 0.3^2)^{3/4} \times \{(6000 \div 3020) - 0.45 \times (10 \div 3020)^{0.5}\}]$$
$$= 1.588 \text{ kg/cm}^2$$

$P_a = P_c \div 4 = 0.397$ kg/cm^2
According to IS – 2825 (Appendix F)
$L \div D_o = 6000 \div 3020 = 1.99$
$D_o \div t = 3020 \div 10 = 302$

Therefore factor B = 1800
$P_a = B \div [14.22 \times (D_o \div t)] = 1800 \div [14.22 \times 302] = 0.419$ kg/cm^2
Here P_a is less than design pressure.
Assume thickness as 20 mm.
According to IS – 2825 (Appendix F)
$L \div D_o = 6000 \div 3020 = 1.99$
$D_o \div t = 3020 \div 20 = 151$
Therefore factor B = 5000
$P_a = B \div [14.22 \times (D_o \div t)] = 5000 \div [14.22 \times 151] = 2.33$ kg/cm^2
Here P_a is more than design pressure. Thus thickness 20 mm is acceptable.
t = 20 mm

(6) Flange calculation

Flange material = IS:2004 – 1962 class 2
Bolting steel = 5% Cr Mo steel
Gasket material = asbestos composition
Outside diameter of calendria = 3024 mm
Calendria sheet thickness = 12 mm
Inside diameter of calendria = 3000 mm
Allowable stress of flange material = 100 MN/m^2
Allowable stress for bolting material = 138 MN/m^2

Determination of gasket width

$$d_o \div d_i = [(y - P \times m) \div \{y - P \times (m + 1)\}]^{0.5}$$

Assuming gasket thickness of 3.2 mm
Therefore y = 11.0, m = 2, P = 0.17157 MN/m^2

$$d_o \div d_i = [(11 - 0.17157 \times 2) \div \{11 - 0.17157 \times (2 + 1)\}]^{0.5} = 1.008$$

Let d_i of gasket equals 3034 mm i.e. 10 mm larger than calendria diameter, then

$$d_o = 1.008 \times 3.034 = 3.059 \text{ m}$$

Minimum gasket width = (3.059 – 3.034) ÷2 = 0.012 mm
Basic gasket seating width, $b_o = 12 \div 2 = 6.0$ mm
Diameter at location of gasket load reaction is, $G = d_i + N$
$= 3.034 + 0.012$
$= 3.046$ m

Estimation of bolt loads

Load due to design pressure

$$H = (\pi/4) \times G^2 \times P$$
$$= (\pi/4) \times (3.046)^2 \times 0.1716$$
$$= 1.25 \text{ MN}$$

Load to keep joint tight under operation

$$H_p = \pi \times G \times 2b \times m \times P$$
$$= \pi \times 3.046 \times 0.012 \times 2 \times 0.1716$$
$$= 0.0394 \text{ MN}$$

Total operating load = $W_o = H + H_p = 1.25 + 0.0394 = 1.2894$ MN
Load to seat gasket under bolting up condition = $W_g = \pi \times G \times b \times y$
$= \pi \times 3.046 \times 0.006 \times 11$
$= 0.6316$ MN
Here W_o is larger than W_g and therefore, controlling load = 1.2894 MN
Calculation of minimum bolting area,

$$A_m = A_o = W_o \div S_o = 1.2894 \div 138 = 0.00934 \text{ m}^2$$

Consider the bolt size as M 20 × 2
Therefore root area = 2×10^{-4} m^2
Number of bolts required = $0.00934 \div (2 \times 10^{-4}) = 48$

Flange thickness

An approximate value of flange thickness may be given by

$$t_f = G\sqrt{\{P \div (k \times f)\}}$$

Where,

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

W_m = total bolt load = 1.2894 MN = 131437.31 kg
$h_G = (B – G) \div 2$
Where B is minimum pitch circle diameter.
B = G + 12 + (2 × 20) = 3046 + 12 + 40 = 3098 mm
Therefore $h_G = (3098 – 3046) \div 2 = 26$
H = 1.25 MN = 127421 kg
G = 3046 mm
Therefore $k = 1 \div [0.3 + \{(1.5 \times 131437.31 \times 26) \div (127421 \times 3046)\}] = 3.193$
Hence $t_f = 3046\sqrt{\{1.1716 \div (3.193 \times 1019)\}} = 69.94$ mm
Therefore use thickness of 75 mm including corrosion allowance.

(7) Bracket design

Data

Diameter of vessel = 3000 mm = 3 m
Height of vessel = 6 m
Clearance from vessel bottom to foundation = 1000 mm (assumed)
Density of carbon steel = 7820 kg/m^3
Density of brass = 8450 kg/m^3
Wind pressure = 128.5 kg/m^2
Number of brackets = 4
Diameter of bolt circle = 3.15 m
Height of bracket from foundation = 2.25 m
Permissible stress for structural steel (IS – 800)

Tension = 1400 kg/cm^2
Compression = 1233 kg/cm^2
Bending = 1575 kg/cm^2

To find weight of vessel with contents

Weight of vapour drum = $\pi dL \times t \times \rho$

$= \pi \times 3 \times 6 \times 0.02 \times 7820$

$W_1 = 8844.21$ kg

Weight of tubes = $W_2 = (\pi/4) \times N_t \times (d_o^2 - d_i^2) \times L \times \rho$

$= (\pi/4) \times 2108 \times (0.0603^2 - 0.0493^2) \times 3.6576 \times 8450$

$= 61690.27$ kg

weight of tube sheet = $W_3 = (\pi/4) \times D_s^2 \times t \times \rho \times 2$

$= (\pi/4) \times 3^2 \times 0.012 \times 7820 \times 2$

$= 1326.63$ kg

Therefore total weight = $W = W_1 + W_2 + W_3 = 71861$ kg

(a) Base plate

Taking suitable base plate size, a = 140 mm, B = 150 mm
Maximum total compressive load is given by

$$P = [\{4 \times P_w \times (H - f)\} \div (n \times D_b)] + [W \div n]$$

Where P_w = total force acting on vessel due to wind = $k \times P^| \times h \times D_o$
k = 0.7
$P^|$ = wind pressure = 128.5 kg/m^2
h = height = 6 m
D_o = diameter of vessel = 3 m
Thus $P_w = 0.7 \times 128.5 \times 6 \times 3 = 809.55$ kg
H = height of vessel above foundation = 6 m
f = vessel clearance from foundation = 0
n = number of brackets = 4
D_b = diameter of bolt circle = 3.15 m
Hence compressive load,

$P = [\{4 \times 809.55 \times (6 - 0)\} \div (4 \times 3.15)] + [71861 \div 4]$
$= 19532.25$ kg

Average pressure on the plate,

$P_{av} = P \div (a \times B) = 19532.25 \div (14 \times 15) = 93.01$ kg/cm^2

But $f = 0.7 \times P_{av} \times (B2 \div T_1^2) \times \{a^4 \div (a^4 + B^4)\}$
$= 0.7 \times 93.01 \times (15^2 \div T_1^2) \times \{14^4 \div (14^4 + 15^4)\}$
$= 6820 \div T_1^2$

Therefore $T_1^2 = 6820 \div 1575 = 23.09$ mm
$T_1 = 4.8$ mm

Use 6 mm thick plate.

(b) Web plate

Bending moment of each plate $= (19532.25 \div 2) \times \{(3.15 - 3) \div 2\} \times 100$
$= 73246$ kg cm

Stress at the edge $= f = (3 \times P \times C) \div (T_2 \times h^2)$
$= (73246 \div 0.707) \div (T_2 \times 14 \times 14)$
$1575 = 528.58 \div T_2$

Therefore $T_2 = 0.3356$ cm $= 3.356$ mm

T_2 may be taken as 4 to 6 mm.

(c) Column support

It is proposed to use a channel section as column.
Size – 150 × 75
Area of cross section = A = 20.88 cm^2
Modulus of section = Z_{yy} = 19.4 cm^2
Radius of gyration = r_{yy} = 2.21 cm
Weight of section = 16.4 kg/m
Height from foundation = l = 2.25 m
Equivalent length for fixed ends = $l_e = l \div 2 = 2.25 \div 2 = 1.125$ m
Slenderness ratio = $l_e \div r = (1.125 \times 100) \div 2.21 = 51.0$

Now if the load is acting eccentric on a short column, the maximum combined bending and direct stress is given by

$f = [W \div (A \times n)] + [(W \times e) \div (n \times Z)]$
$= [71861 \div (20.88 \times 4)] + [(71861 \times 7.5) \div (4 \times 19.4)]$
$= 7805.73$ kg/cm^2

The permissible compressive stress is

$f = [W \div (A \times n)] [1 + a \times (l_e \div r)^2] + [(W \times e) \div (n \times Z)]$
$= [71861 \div (20.88 \times 4)] [1 + (51^2 \div 7500)] + [(71861 \times 7.5) \div (4 \times 19.4)]$
$= 8104.12$ kg/cm^2

The calculated values are less than the permissible compressive stress and hence the channel selected is satisfactory.

(d) Base plate for column

The size of column is 150 × 75. It is assumed that the base plate extends 20 mm on either side of the channel.
Width = 0.8 × 75 + 2 × 20 = 100 mm
Length = 0.95 × 150 + 2 × 20 = 182.5 mm
Bearing pressure on each plate = P_b = P ÷ (B × C) = 19532.25 ÷ (10 × 18.25)
= 107.03 kg/cm^2
This is less than the permissible bearing pressure for concrete.
Stress in the plate = $[(107.03 \div 2) \times (20^2 \div 10^2)] \div (t^2 \div 6) = 12.84 \div t^2$ kg/cm^2
But f = 1575 kg/cm^2

Therefore $t^2 = (12.84 \div 1575) \times 100$ mm^2
t = 0.8152 mm
It is usual to select a plate of 4 to 6 mm thickness.

2. CONDENSER

Material:

Carbon and low alloy steel
Carbon steels: up to 540°C
Low alloy steels: up to 590°C
Design pressure: by adding a minimum of 5% to the maximum working pressure.
Design temperature: 10°C higher than the maximum temperature that any part of the exchanger is likely to attain in course of operation.
Corrosion allowance: 1.5 mm for carbon and cast iron.

(A) Shell Side:

Material of construction: Carbon steel
Design pressure = 1 atm = $(1.01325 \times 10^5) \div 9.81$ = 10328.75 kgf/m^2 = 1.033 kgf/cm^2
Design temperature = 51.85 + 8.15 = 60°C
Shell inside diameter = D_s = 2134 mm = 2.134 m
No. of shell passes = 1
No. of tube passes = 4
Permissible stress for carbon steel = 12.8 kgf/mm^2 (IS: 3609 – 1966)
Fluid: water vapour
Inlet temperature = 51.85°C
Outlet temperature = 51.85°C

(1) Shell thickness

Minimum thickness of shell plates excluding of corrosion allowance in mm (IS: 2825)
$t_s = (P_d \times D_S)/\{(200 \times f \times J) - P_d\}$

Where J = joint efficiency = 0.85
Therefore t_s = (1.033 × 2134) ÷ (200 × 0.85 × 12.8) – 1.033}
= 1.0135 mm
But minimum thickness of shell is 6 mm.
Therefore with corrosion allowance of 2 mm, use thickness of shell as 8 mm.

(2) Nozzle diameter

Mass flow rate of vapour = m_v = 14.1295 kg/s
Density = ρ = 0.7 kg/m^3
Assume velocity to be 50 m/s
$(\pi \times d_n^2 \times \rho \times v)/4 = m_v$
$d_n^2 = (14.1295 \times 4)/(50 \times \pi \times 0.7)$
Therefore nozzle diameter = d_n =0.072 m
Take it as 75 mm.

(3) Nozzle thickness

The thickness of nozzle is given by

$t_n = (P_d \times d_n) \div \{(200 \times f \times J) - P\}$ (J = 1 for seamless pipes)

$= (1.033 \times 75) \div \{(200 \times 12.8 \times 1) - 1.033\}$

$= 0.0303$ mm

Nozzle thickness with corrosion allowance = 9 mm

(4) Head thickness

Torispherical head: the thickness of such heads is given by

$$t_h = (P_d \times R_c \times C) \div (200 \times f \times J)$$

Where $C = (1/4) \times [3 + (R_c/R_K)^{1/2}]$

J = 1 for ends made from one plate and attached to shell with a straight flange.

R_c = crown radius (it is equal to or less than the diameter of shell)

Choose R_c = shell diameter = 2134 mm

R_K = knuckle radius (at least 6% of shell I.D) = 0.10 × 2134 = 213.4 mm

Thus C = 1.54

Hence $t_h = (1.033 \times 2134 \times 1.54) \div (200 \times 12.8 \times 1)$

$t_h = 1.33$ mm

Using same thickness as that of the shell = 8 mm

(5) Transverse baffles

Baffle Spacing = $l_s = 0.7\ D_s = 0.7 \times 2134 = 1493.8$ mm

Number of baffles = $N_b = (L/l_s) - 1 = (2998/1493.8) - 1 = 1$

Thickness of baffles = 6 mm

Height of baffle = $0.75 \times D_s$

= 666 mm

(6) Tie rods and spacers

The baffles are supported independently of the tubes by the tie rods and positioned by spacers.

For shell diameter D_s = 2134 mm

No. of tie rods = 6

No. of spacers = 7

Diameter of rods = 13 mm

(7) Flange calculation

Design pressure = 1 atm = 0.101325 MN/m^2

Design temperature = 60^oC

Flange material = IS:2004-1962 class 2

Bolting steel = 5% Cr Mo steel

Gasket material = asbestos composition

Shell inside diameter = 2134 mm
Shell thickness = t_s = 8mm
Shell outside diameter = $(2 \times t_s) + 2134$
$= (2 \times 8) + 2134$
= 2150 mm
Allowable stress of flange material = 10.2 kgf/mm^2 = 100 MN/m^2
Allowable stress of bolting material = 10.2 kgf/mm^2 = 138 MN/m^2

(I) Determination of gasket width

$d_o/d_i = [(y - P_d \times m)/\{y - P_d(m+1)\}]^{1/2}$
d_o and d_i are outer & inner diameters of the gasket
Assume gasket thickness of 1.6 mm.
Therefore from literature,
M = gasket factor = 2.75
y = minimum design yield stress = 25.5 MN/m^2
$d_o/d_i = [\{25.5 - (0.101325 \times 2.75)\}/\{25.5 - (0.101325 \times 3.75)\}]^{1/2}$
= 1.004
Let inside diameter of gasket equal outside diameter of shell.
Therefore d_i = 2150 mm = 2.15 m
$d_o = 1.004 \times 2.15 = 2.1587$ m
Gasket width = (2.1587 – 2.15) ÷ 2 = 0.00434 m = 4.34 mm
But minimum gasket width is 10 mm for asbestos material.
Therefore N =10 mm
Thus d_o = 2170 mm = 2.17 m
Basic gasket seating width = b_o = N/2 = 10/2 = 5 mm
Diameter of location of gasket load reaction is
$G = d_i + N$
= 2134 + 10
= 2144 mm

(II) Estimation of bolt load

Load due to design pressure = $H = (\pi \times G^2 \times P_d)/4$
$= (\pi \times 2.144^2 \times 0.101325)/4$
= 0.3658 MN
Load to keep joint tight under pressure = $H_p = \pi \times (2b) \times G \times m \times P_d$
$= 3.14 \times 0.01 \times 2.144 \times 2.75 \times 0.101325$
= 0.0188 MN
Total operating load = $W_o = H + H_p$
= 0.3658 + 0.0188
= 0.3846 MN
Load to seat gasket under bolting up condition = $W_g = \pi \times b \times G \times y$
$= 3.14 \times 0.005 \times 2.144 \times 25.5$
= 0.8588 MN
Since $W_g > W_o$, controlling load = 0.8588 MN

Calculation of minimum bolting area:
Total cross sectional area of bolt required for gasket seating = $A_m = A_g = W_g \div S_g$
S_g = allowable bolt stress at ambient temperature (30ºC) = 138 MN/m^2
Therefore A_m = 0.8588/138
= 0.006223 m^2

Calculation of optimum bolt size:
Actual number of bolts = 44
C = bolt circle diameter = $\{2 \times (R + g_1)\} + B$
Choosing bolt M-18 × 2
Actual number of bolts = 44
R = radial clearance from bolt circle to point of connection of hub and back of flange
= 27 mm
B = inside diameter of flange = inside diameter of shell = 2.134 m
$g_1 = g_o \div 0.707$, let g_o= 8 mm
$g_1 = 1.415 \times g_o$
C = {2 × (0.027 + 1.415 × 0.008)} + 2.134
= 2.2106 m
Therefore bolt circle diameter = 2.2106 m
Now assume bolt spacing as 60 mm = B_s
But C can also be calculated as = $(n \times B_s)/\pi = (44 \times 0.06)/\pi = 2.2$ m
Therfore take bolt circle diameter as = C = 2.21 m

(III) Calculation of flange outside diameter

A = C + bolt diameter + 0.02
= 2.21 + 0.018 + 0.02
= 2.248 m
Check of gasket width**:**
To prevent damage to the gasket during bolting up, following condition is to be satisfied.

$$\{A_b S_g/(\pi G N)\} < 2y$$

Where A_b = root area of bolt (m^2) = $44 \times 1.54 \times 10^{-4}$ m^2
S_g = allowable stress for bolting material at atmospheric temperature = 138 MN/m^2
Therefore $A_b S_g/(\pi G N) = (44 \times 1.54 \times 10^{-4} \times 138)/(\pi \times 2.144 \times 0.01) = 13.88 < 2y$
Hence condition is satisfied.

(IV) Flange moment computations

For operating conditions:

$$W_o = W_1 + W_2 + W_3$$

Where $W_1 = (\pi \times B^2 \times P_d)/4$
$= (3.14 \times 2.15^2 \times 0.101325)/4$
= 0.36 MN
$W_2 = H - W_1$
= 0.3658 – 0.3600
= 0.0058 MN

$W_3 = W_o – H$
$= H_p$
$= 0.0188$ MN
Total flange moment is,
$M_o = W_1a_1 + W_2a_2 + W_3a_3$
The values of a_1, a_2 and a_3 for loose type except lap joint flange is given by
$a_1 = (C – B)/2 = (2.21 – 2.15)/2 = 0.03$ m
$a_3 = (C - G)/2 = (2.21 – 2.144)/2 = 0.033$ m
$a_2 = (a_1 + a_3)/2$
$= 0.0315$ m
$M_o = 0.36 \times 0.03 + 0.0058 \times 0.0315 + 0.033 \times 0.0188$
$= 1.16 \times 10^{-2}$ MJ
For bolting up condition**:**
In this case the total flange moment is given by
$M_g = W \times a_3$
Where $W = \{(A_m + A_b)S_g\}/2$
$= \{(6.223 \times 10^{-3} + 44 \times 1.54 \times 10^{-4}) \times 138\}/2$
$= 0.8969$ MN
$a_3 = (C – G)/2 = 0.033$ m
Therefore $M_g = 0.8969 \times 0.033$
$= 0.0296$ MJ
Since $M_g > M_o$, hence moment under bolting-up condition, M_g is controlling.
Therefore $M = M_g = 0.0296$ MJ

(V) Calculation of flange thickness

$$t^2 = (M \times C_F \times Y) \div (B \times S_T)$$

K = A/B
= outer diameter of flange/ inner diameter of flange
= 2.248/2.15
= 1.046
Therefore Y = 38
Assume $C_F = 1$ (for initial calculation C_F may be taken as unity and finally its magnitude can be checked.
Therefore $t^2 = (0.0296 \times 1 \times 38) \div (2.15 \times 100)$
Thickness t = 0.072 m
Actual bolt spacing = $B_S = (\pi \times C)/n$
$= (3.14 \times 2.21)/44$
$= 0.068$ m
Bolt pitch correction factor = $C_F = [B_S/2d + t)]^{1/2}$
$= [0.068/2 \times 0.018 + 0.072)]^{1/2}$
$= 0.7935$
Therefore $C_F^{1/2} = 0.891$
The flange thickness calculated above is to be multiplied by $C_F^{1/2}$
Hence actual flange thickness = $C_F^{1/2} \times t$
$= 0.891 \times 0.072$

$= 0.064$ m
$= 64$ mm

(B) Tube Side:

Tube and tube sheet material: stainless steel
Number of tubes = 3784
Outside diameter of tube = 25.4 mm
Length of tube = 10 ft = 3.048 m
Fluid: water
Pitch (Δ) = 31.75 mm
Allowable stress = 10.06 kgf/mm^2
Working pressure = 1 atm = 1.033 kgf/cm^2
Design pressure = 1.1 × 1.033 = 1.1363 kgf/cm^2
Inlet temperature = 30°C
Outlet temperature = 40°C

(1) Thickness of tube

$$t_t = (P \times d_o) \div \{(2 \times f \times J) + P\}$$

J =1 for seamless tube
Therefore $t_t = (1.1363 \times 25.4) \div \{(2 \times 1006 \times 1) + 1.1363\}$
$= 0.014$ mm
No corrosion allowance since the tube is made of stainless steel.
Therefore thickness of tube = 1mm

(2) Tube sheet thickness

A simpler equation for the effective tube sheet thickness is as follows:

$$t_S = F \times G \times [(0.25 \times P)/f]^{1/2}$$

F = the value of F varies according to type of heat exchanger, in most cases it is taken as one.
G = mean diameter of gasket = 2144 mm
Therefore $t_S = (1 \times 2144) \times [(0.25 \times 1.1363)/1006]^{1/2}$
$= 36.03$ mm

(3) Channel and channel cover

The effective thickness of the flat channel cover is calculated from the formula

$$t = G \times [(K \times P)/f]^{1/2}$$

K = 0.3 for ring type gasket
Material of construction is carbon steel.
So allowable stress f = 950 kgf/cm^2
Therefore $t = 2144 \times [(0.3 \times 1.1363)/950]^{1/2}$
$= 40.61$ mm

Use 45 mm thickness including corrosion allowance.

(4) Nozzle thickness

Assume inlet and outlet diameter = 75 mm
Thickness of nozzle

$t_n = (P \times d)/\{(2 \times f \times J) - P\}$
$= (1.1363 \times 75)/\{(2 \times 1 \times 950) - 1.1363\}$
$= 0.045$ mm

With corrosion allowance, thickness = 4 mm

(5) Saddle support for horizontal vessels

Material: Low carbon steel
Vessel diameter = 2150 mm
Length of shell = 3.048 m
Torispherical head:
Crown radius = 2134 mm
Knuckle radius = 213.4 mm (10% of shell diameter)
Working pressure = 1 atm
Shell thickness = 8 mm
Head thickness = 8 mm
Depth of head = 257 mm
Corrosion allowance = 3 mm
Permissible stress = 950 kgf/cm^2
Distance of saddle centre line from shell end = A = this distance is usually 0.4 to 0.5R and less than 0.2L.
Therefore A = 0.45 × R = 0.45 × (2134/2) = 480.15 mm < 0.2L

(I) Longitudinal bending moment

The bending moment at the support is
$M_1 = Q \times A \times [1 - \{(1 - A/L) + (R^2-H^2)/2AL\}/ (1 + 4H/3L)]$
Q = load carried by each support
= (W/2) × [L + (4 × H/3)]
Where W = Weight of fluid and vessel

Weight of shell material:

$W_1 = [\pi(D_o^2 - D_i^2) \times L \times \rho_{shell\ material}]/4$
$\rho_{shell\ material} = 7700\ kg/m^3$
$W_1 = [3.14(2.15^2 - 2.134^2) \times 3.048 \times 7700]/4$
$= 1263.47$ kg

Weight of tubes:

$$W_2 = [N_t \times \pi \times (d_o^2 - d_i^2) \times L \times \rho_{tube\ material}]/4$$

$\rho_{tube\ material} = 7800\ kg/m^3$

$W_2 = [3784 \times 3.14 \times (0.0254^2 - 0.023622^2) \times 3.048 \times 7800]/4$

$= 6158.48$ kg

Weight of tube sheets:

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

$= (2 \times 3.14 \times 2.15^2 \times 0.03603 \times 7800)/4$

$= 2040.60$ kg

Liquid load in the shell:

$W_4 =$ (shell volume - tube volume) $\times \rho_{liquid}$

$= [(\pi D_s^2 L)/4 - (N_t \pi d_o^2 L)/4] \times 989$

$= [\{(\pi \times 2.15^2 \times 3.048)/4\} – \{(3784 \times \pi \times 0.0254^2 \times 3.048)/4\}] \times 989$

$= (10.902 – 5.844) \times 989$

$= 5002.19$ kg

Liquid load in tubes:

$W_5 = (N_t \times \pi \times di^2 \times L \times \rho_{liquid})/4$

$= (3784 \times 3.14 \times 0.023622^2 \times 3.048 \times 994)/4$

$= 5024.30$ kg

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

$= 19489.04$ kg

Hence, $Q = (19489.04/2) \times [3.048 + (4 \times 0.257)/3]$

$= 33040.42$

$M_1 = 33040.42 \times 48.02 \times [1 - \{(1 – 48.02/3.048) + (106.7^2 – 25.7^2)/ (2 \times 48.02 \times 3.048)\}/ (1 + 4 \times 25.7/3 \times 3.048)]$

Thus $M_1 = 137475.13$ kg cm

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

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

$M_2 = (33040.42 \times 3.048/4) [\{1+2(106.7^2-25.7^2)/304.8^2\}/ \{1+ (4 \times 25.7/3 \times 304.8)\} - (4 \times 48.02/304.8)]$

Therefore $M_2 = 1199158.58$ kg cm

(II) Stress in shell at the saddle

At the top most fiber of the cross-section:

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

For an angle of 120º, $K_1 = 0.107$

t = thickness of shell = 8 mm

$f_1 = 137475.13/ (0.107 \times 3.14 \times 106.7^2 \times 0.8)$

$= 436.85\ kg/cm^2$

At the bottom most fiber of the cross-section:

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

For an angle of 120º, $K_2 = 0.192$

$f_2 = 137475.13 / (0.192 \times 3.14 \times 106.7^2 \times 0.8)$

$= 25.02$ kg/cm^2

(III) Stress in the shell at mid point

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

$= 1199158 / (3.14 \times 106.7^2 \times 0.8)$

$= 41.91$ kg/cm^2

Thus the values of stresses are within the limited range; hence the designed support is acceptable.