DESIGN OF A TRIPPLE EFFECT FORWARD FEED EVAPORATOR FOR THE CONCENTRATION OF GLYCEROL SOLUTION:

Design: Glycerol solution is to be concentrated from 7Wt% to a final concentration of 80Wt% The feed is available at 50993 Kg/Hr and 27°C. Steam is supplied at 103.66Kpa (Abs.) to the first effect and a vacuum of 74.2(Abs.) mm of Hg is maintained in the last effect. Heat losses by radiation and by entertainment are neglected. Condensate are assumed to enter each effect at its saturation temperature.

Glycerol in feed = 50993×0.07

= 3569.51 Kg/Hr

Feed rate W_F = 50993/3600

= 14.16 Kg/Sec

Bottom outlet from third effect = $3569.51/(0.8\times3600)$

= 1.24 Kg/Sec

Total evaporation =14.16 – 1.24

=12.9 Kg/Sec

Assuming equal evaporation in all the three effects.

W_1– Evaporation rate in first effect

W_2- Evaporation rate in second effect

W_3– Evaporation rate in the third effect

$W_1 = W_2 = W_3 = (12.9/3) = 4.3$ Kg/sec

Outlet from first effect = $W_F - W_1$

= 14.16 – 4.3

= 9.86 Kg/sec

Outlet from second effect = $W_F – W_1 - W_2$

= 14.16 – 4.3 – 4.3

= 5.56 Kg/sec

Outlet from third effect = $W_F – W_1 – W_2 – W_3$

= 14.16 – 4.3 – 4.3 – 4.3

= 1.26 Kg/sec

Concentration of glycerol in outlet from first effect = $(14.16 \times 0.07)/ 9.86$

= 10 wt%

Concentration of glycerol in outlet from second effect = $(14.16 \times .07)/5.56$

$= 0.178$

$= 17.8$ wt%

Concentration of glycerol in outlet from third effect = $(14.16 \times .07)/1.26$

$= 0.79$

$= 79$ wt%

Saturation temperature of inlet steam $T_S = 100°C$

Boiling point of water in the third effect = 12°C

Temperature difference =100 – 12

= 88°C

BOILING POINT RISE FOR GLYCEROL SOULTION

1) BPR for the third effect at 79 Wt% glycerol = 10.4°C
2) BPR for the second effect at 17.8 Wt% glycerol = 1.5°C
3) BPR for the first effect at 10 Wt% glycerol = 0.5°C

Sum of the BPR of the glycerol solution = 10.4 + 1.5 + 0.5

= 12.4°C

Effective temperature difference = 88 – 12.4

= 75.6°C

Now we assume that the amount of heat transfer from steam to the solution is the same in all the three effects i.e. Q (heat transfer in the first effect) = Q (heat transfer in the second effect) = Q (heat transfer in the third effect)

OVERALL HEAT TRANSFER COEFFICIENT

Since it is a vertical effect evaporator, the overall heat transfer coefficient can safely be assumed as follows:

In the first effect $U_1 = 2325$ W/m°C

In the second effect $U_2 = 1275$ W/m°C

In the third effect $U_3 = 1031$ W/m°C

Now as we have assumed heat transfer rate's to be equal , we have

$$Q_1 = Q_2 = Q_3$$

$$(U_1 \times A_1 \times \Delta t_1) = (U_2 \times A_2 \times \Delta t_2) = (U_3 \times A_3 \times \Delta t_3)$$

We design the triple effect evaporator such that the heating area in all the three effect is the same

$A_1 = A_2 = A_3$

$U_1 \times \Delta t_1 = U_2 \times \Delta t_2 = U_3 \times \Delta t_3$

$\Delta t_2 / \Delta t_1 = U_1 / U_2$

$= 2352 / 1275$

$= 1.8$

$\Delta t_3 / \Delta t_2 = U_2 / U_3$

$=1275 / 1031$

$= 0.6$

$\Delta t_1 + \Delta t_2 + \Delta t_3 = 75.6°C$

$0.6\ \Delta t_2 + \Delta t_2 + 0.6\ \Delta t_2 = 75.6°C$

$\Delta t_2 = 32.3°C$

$\Delta t_1 = 21.54°C$

$\Delta t_3 = 21.64°C$

ACTUAL BOILING POINTS IN EACH EFFECT

First effect:

$T_1 = T_S - \Delta t_1$

$= 100 - 21.54$

$= 78.54°C$

Second effect:

$T_2 = T_1 - (BPR)_1 - \Delta t_2$

$= 78.5 - 0.5 - 32.3$

$= 45.7°C$

Third effect:

$T_3 = T_2 - (BPR)_2 - \Delta t_3$

$= 45.7 - 1.5 - 21.64$

$= 22.56°C$

Effect 1 (ºC)	Effect 2 (ºC)	Effect 3 (ºC)	Condenser (ºC)
T_S = 100	T_1 = 78	T_2 = 44.2	T_3 = 12.16
T_1 = 78.5	T_2 = 45.7	T_3 =22.56	

HEAT BALANCE

FIRST EFFECT:

$W_S \lambda_S + W_F H_F = W_1 H_1 + (W_F - W_1) h_1$

latent heat of steam λ_S = 2257.86 KJ/ kg

H_F – Enthalpy of feed at inlet temperature (27ºC) $= C_{pf} \times (T_f - 0)$

$= (0.576 \times 4.18) \times 27$

= 65 KJ / Kg

H_1- Enthalpy of vapor leaving the first effect $= H_{2S} + (C_p)_{steam} \times (BPR_1)_{superhea}$

$= 2640 + (1.884 \times 0.5)$

= 2487 KJ/ Kg

H_{2s}- Enthalpy of steam at 78ºC = 2640 KJ / Kg

$(C_p)_{steam}$ at 78ºC = 1.884 KJ/ Kg

h_1 - enthalpy of outlet from first effect at 78.5ºC $= C_{p1} \times (t_1 – 0)$

$= 0.65 \times 4.18 \times 78.5$

= 213.28 KJ/Kg

$$W_S \times (2257.86) + (14.16 \times 65) = (W_1 \times 2487) + (14.16 – W_1) \times 213.28$$

$$W_S \times (2257.86) = 2273.7 \times W_1 + 2099.6$$

$$W_S = W_1 + 0.93 \quad \text{------------------------}(1)$$

SECOND EFFECT:

$$W_1 \lambda_1 + (W_F – W_1) h_1 = W_2 H_2 + (W_F – W_1 – W_2) h_2$$

Latent heat of steam at 78ºC $\lambda_1 = H_1 - h_{2S}$

$= 2487 - 325$

$= 2162$ KJ /Kg

H_{3S} – Enthalpy of steam vapor at 44.2ºC = 2580 KJ /Kg

H_2 - Enthalpy of vapor leaving the second effect $= H_{3S} + (C_p)_{steam} \times(BPR_2)_{supreheat}$

$= 2580 + (1.884 \times 1.5)$

$= 2583$ KJ/Kg

h_2-Enthalpy of outlet from the second effect at 45.7ºC $= C_{p2} \times (t_2 - 0)$

$= 0.6 \times 4.18 \times 45.7$

$= 114.62$ KJ /Kg

$$W_1 \times 2162 + (14.16 – W_1) \times 213.28 = (W_2 \times 2583) + (14.16 – W_1 – W_2)\times 114.62$$

$$2063.34 \times W_1 = 1397 + 2468.38 \times W_2$$

$$W_1 + 0.667 = 1.196 \times W_2 \text{ -------------------(2)}$$

THIRD EFFECT:

$$W_2 \lambda_2 + (W_F – W_1 –W_2) \times h_2 = W_3 H_3 + (W_F – W_1 - W_2 - W_3) \times h_3$$

Latent heat of steam at 44.2ºC $\lambda_2 = H_2 - h_{3S}$

$= 2583 - 190$

$= 2393$ KJ / Kg

H_{4S} -Enthalpy of steam vapor at 12 .16ºC = 2523 KJ/Kg

H_3 -Enthalpy of vapor leaving the third effect $= H_{4S} + (C_p)_{steam} \times (BPR_3)_{ssuperheat}$

$= 2523 + (1.884 \times 10.4)$

$= 2543$ KJ/ Kg

h_3 – Enthalpy of outlet from third effect at 22.56ºC $= C_{p3} \times(t_3 - 0)$

$= 0.57 \times 4.18 \times 22.56$

$= 53.75$ KJ /Kg

$$W_2 \times 2393 + (14.16 – W_1 – W_2) \times 114.62 = W_3 \times 2543 + (14.16 – W_1 – W_2 – W_3) \times 53.75$$

$$2332.13 \times W_2 - 60.87 = 2489.25 \times W_3 + 861.92$$

$$W_2 + 0.370 = 1.067 \times W_3 + 0.026 \times W_1 \text{ --------------------(3)}$$

$$W_1 + W_2 + W_3 = 12.9 \text{ Kg/ Sec -------------------------(4)}$$

Solving equations' (1),(2),(3) and (4), we get:

$W_S = 5.153$ Kg/sec

$W_1 = 4.223$ Kg/sec

$W_2 = 4.089$ Kg/sec

$W_4 = 4.588$ Kg/sec

Now, $Q_1 = W_S\lambda_S$

$=11634.75$ KJ/sec

But $Q_1=U_1A_1\Delta t_1$

Therefore $A_1= (11634.75\times10^3)/(21.54\times2325)$

$= 232.32\ m^2$

$Q_2 = W_1\lambda_1$

$= 9130.13$ KJ/sec

But $Q_2 = U_2A_2\Delta t_2$

Therefore $A_2 = (9130.13\times10^3)/(32.3\times1275)$

$= 228\ m^2$

$Q_3 = W_2\lambda_2$

$= 9784.98$ KJ/sec

But $Q_3 = U_3A_3\Delta t_3$

Therefore $A_3 = (9784.98\times10^3)/(21.64\times1031)$

$= 235\ m^2$

Thus the obtained areas are within the acceptable range of 5% difference

Therefore the average area per effect of the evaporator is 232 m^2.

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 5/4-in. nominal diameter, 80 schedule, brass tubes of 10-ft length.

Therefore Outer diameter d_o = 42.164 mm

Inner diameter d_i = 32.46 mm

Length L = 10 ft

= 3.048 m

Tube pitch $(\Delta)P_T = 1.25 \times d_o$

$= 1.25 \times 42.164$

= 52.705 mm

Surface area of each tube $a = \pi d_o L$

$= \pi \times 52.705 \times 10^{-3} \times 3.048$

$= 0.4037\ m^2$

Number of tubes required $N_t = A / a$

= 619

Area occupied by tubes $= N_t \times (1/2) \times P_T \times P_T \times \sin\propto$

$= 619 \times 0.5 \times (52.705 \times 10^{-3})^2 \times 0.866$

$= 0.7445\ m^2$

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 choose this factor as 0.9.

Therefore actual area required = 0.7445/ 0.9

$= 0.827\ m^2$

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

$= 0.5 \times [619 \times (\pi/4) \times (0.04216)^2]$

$= 0.432\ m^2$

Downcomer diameter $= \sqrt{(4 \times 0.432) / \pi}$

= 0.742 m

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

$= 0.432 + 0.827$

$= 1.259\ m^2$

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

$= 1.27$ m

MECHANICAL DESIGN OF EVAPORATOR:

Take standard vertical short tube evaporator (calendria type)

Data:

Evaporator drum operating under vacuum at 0.4163 bar

Amount of water to be evaporated = 15480 kg/hr

Heating surface required A = 232 m^2

Steam is available to first effect at pressure of 1.03 bar.

Density of liquid (10 wt% glycerol) = 1019 kg/m^3

Density of water vapour = PM/RT

$= (0.4163 \times 10^5 \times 18) / (8314 \times 351)$

$= 0.258 \ kg/m^3$

Design pressure = 5% extra of maximum working pressure

$= 1.05 \times 1.03$

$= 1.082$ bar

$= 1.103 \ kgf/cm^2$

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

$t_s = (PD_o) / (2fJ + P)$

$= (1.103 \times 1270) / [(2 \times 980 \times 0.85) + 1.103]$

$= 0.84$ 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 = 10 mm

2) Tube sheet thickness:

To find tube sheet thickness

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

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 $K = [19 \times 10^5 \times 12 \times (1270 - 10)] / [9.5 \times 10^5 \times 619 \times 4.85 \times (42.164 - 4.85)]$

$= 0.027$

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

$= \sqrt{[(2 + 0.027) / (2 + 3 \times 0.027)]}$

$= 0.99$

The effective tube sheet thickness is given by

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

$= 0.99 \times 1270 \times \sqrt{[(0.25 \times 1.103) / 980]}$

$= 21$ mm

with corrosion allowance the thickness may be taken as 25 mm.

4) **Check for tube thickness:**

The tube thickness is given by

$t_t = Pd_i / (2fJ - P)$

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

Therefore $t_t = (1.103 \times 32.46) / [(2 \times 538 \times 1) - 1.103]$

$= 0.033$ mm

But provided thickness is 4.85 mm.Therefore chosen tubes have enough strength to withstand within operating conditions.

5) 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.

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

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

A – cross sectional area of drum

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

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

$= [15480 /(3600 \times 0.258)] / [1.3 \times 0.0172 \times \sqrt{\{(1019 – 0.258)/0.258\}}]$

$= 11.87\ m^2$

Therefore drum diameter = $\sqrt{\{(4 \times 11.87) /3.14\}}$

3.89 m

Which is very large and therefore taking the drum diameter same as that of calendria.i.e 1.27 m

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

Thus drum height = 2×1.27

= 3.81 m

5) Drum thickness:

Drum is operating at 0.4163 bar. Design is therefore based on an external pressure of 1.7856 kg/cm^2 (Design pressure).

Assume thickness of 15 mm.

The critical external pressure is given by

$Pc = [2.42 \times E \times (t /Do)^{2.5}] / [(1 - \mu^2)^{3/4} \times \{(L /Do) – 0.45 \times (t / Do)^{0.5}\}]$

$= [2.42 \times 19 \times 10^5 \times (10 /1290)^{2.5}] / [(1 – 0.32^2)^{3/4} \times \{(3810 /1290) – 0.45 \times (10 /1290)^{0.5}\}]$

$= 8.96$ kg/cm^2

Pa = Pc / 4

$= 2.24$ kg/cm^2

According to IS – 2825 (Appendix F)

L /Do =3.81/ 1.27

= 3

Do / t = 1270 /10

= 85

Therefore factor B = 10878

P_a= B / [14.22 × (Do / t)]

= 10878 / [14.22 ×85]

= 9 kg/cm^2

Here P_a is greater than design pressure.

Assume thickness as 15 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 = 1290 mm

Calendria sheet thickness = 10 mm

Inside diameter of calendria = 1270 mm

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

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

Determination of gasket width

$do / di = [(y - Pm) /\{y - P(m + 1)\}]^{0.5}$

Assuming gasket thickness of 3.2 mm

Therefore y = 11.0, m = 2, P = 0.11 MN/m^2

$do / di = [(11 - 0.11 \times 2) /\{11 - 0.11 \times (2 + 1)\}]^{0.5}$

= 1.005

Let di of gasket equals 1300 mm i.e. 10 mm larger than calendria diameter, then

d_o= 1.3065 m

Minimum gasket width 2b = (1.3065 –1.300) /2

= 3.25×10^{-3} m

Basic gasket seating width, bo = 12 / 2

= 6.0 mm

Diameter at location of gasket load reaction is, G = di + N

= 1300 + (3.25×10^{-3})

= 1300 mm

Estimation of bolt loads

Load due to design pressure

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

$= (\pi/4) \times (1.3)^2 \times 0.11$

$= 0.15$ MN

Load to keep joint tight under operation

$Hp = \pi G(2b)mP$

$= \pi \times 1.3 \times (3.25 \times 10^{-3}) \times 2 \times 0.11$

$= 2.92 \times 10^{-3}$ MN

Total operating load $Wo = H + Hp$

$= 0.15 + 2.92 \times 10^{-3}$

$= 0.153$ MN

Load to seat gasket under bolting up condition $Wg = \pi Gby$

$= \pi \times 3.046 \times (3.25 \times 10^{-3}/2) \times 11$

$= 0.17$ MN

Here W_g is larger than W_o and therefore, controlling load = 0.17 MN

Calculation of minimum bolting area:

$A_m = A_o = W_g/S_o = 0.17/138$

$= 1.23 \times 10^{-3}$ m^2

Consider the bolt size as M 20×2

Therefore root area = 2×10^{-4} m^2

Number of bolts required = $(1.23 \times 10^{-3}) / (2 \times 10^{-4})$

$= 8$

Flange thickness:

An approximate value of flange thickness may be given by

$t_F = G\sqrt{\{P/(kf)\}}$

Where,

$k = 1/ [0.3 + \{(1.5W_m h_G) /(HG)\}]$

W_m - total bolt load = 1.2894 MN

$h_G = (B–G) / 2$

Where B is minimum pitch circle diameter.

$B = G + 12 + (2 \times 20)$

$= 1300 + 12 + 40$

$= 1352$ m

Therefore $h_G = (1352 - 1300) / 2$

$= 26$

H = 1.25 MN

= 127421 kg

G = 1300 mm

Therefore $k = 1 / [0.3 + \{(1.5 \times 131437.31 \times 26) / (127421 \times 1300)\}]$

$= 3.02$

Hence $t_f = 1300\sqrt{\{1.103/(3.02 \times 1019)\}}$

$= 24.61$ mm

Therefore use thickness of 30 mm including corrosion allowance.

7) Bracket design:

Data:

Diameter of vessel = 1270 mm

Height of vessel = 3.81 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 = $\pi dLt\rho$

$= \pi \times 1.27 \times 3.81 \times 0.015 \times 7820$

$W_1 = 1783$ kg

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

$= (\pi/4) \times 619 \times (0.0422^2 - 0.03246^2) \times 3.048 \times 8450$

$= 9150$ kg

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

$= (\pi/4) \times 1.27^2 \times 0.025 \times 7820 \times 2$

$= 495$ kg

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

$= 10728$ kg

a) Base plate:

Taking suitable base plate size, a = 140 mm, B = 150 mm

Maximum total compressive load is given by

$P = [\{4P_w(H - f)\} / (nD_b)] + [W / n]$

Where Pw - total force acting on vessel due to wind = $kPhD_o$

k = 0.7

P - wind pressure = 128.5 kg/m^2

h - height = 3.81 m

Thus $P_w = 0.7 \times 128.5 \times 3.81 \times 1.27$

$= 435.2\ kg/m^2$

H - height of vessel above foundation =1.27 m

f - vessel clearance from foundation = 0

n - number of brackets = 4

D_b diameter of bolt circle = 1.65 m

Hence compressive load,

$P = [\{4 \times 435.2 \times (1.27 - 0)\}/ (4 \times 3.15)] + [10728/ 4]$

$= 2857$ kg

Average pressure on the plate:

$P_{av} = P / (a\ B)$

$= 2857 / (14 \times 15)$

$= 13.6\ Kg/cm^2$

But $f = 0.7\ Pav\ (B^2 / T_1^2) \times \{a^4 / (a^4 + B^4)\}$

$= 0.7 \times 13.6 \times (15^2 / T_1^2) \times \{14^4 \div (14^4 + 15^4)\}$

$= 997.2 / T_1^2$

Therefore $T_1^2 = 997.2 / 1575$

$= 0.63$mm

$T_1 = 0.8$ mm

Use 6 mm thick plate.

b) Web plate:

Bending moment of each plate = $(19532.25 / 2) \times \{(3.15 - 3) / 2\} \times 100$

= 73246 kg cm

Stress at the edge = $f = (3 \times P \times C) / (T_2 \times h_2)$

$= (73246 / 0.707) / (T2 \times 14 \times 14)$

$1575 = 528.58 / T_2$

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 cm2

Modulus of section = Z_{yy} = 19.4 cm2

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 = le = $l \div 2 = 2.25 \div 2 = 1.125$ m

Slenderness ratio = $le \div r = (1.125 \times 100) \div 2.21 = 51.0$

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/cm2

The permissible compressive stress is

$$f = [W \div (A \times n)]\ [1 + a \times (le \div r)2] + [(W \times e) \div (n \times Z)]$$

$$= [71861 \div (20.88 \times 4)]\ [1 + (51^2 \div 7500)] + [(71861 \times 7.5) \div (4 \times 19.4)]$$

$$= 8104.12 \text{ kg/cm2}$$

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 $= 0.8 \times 75 + 2 \times 20$

$= 100$ mm

Length $= 0.95 \times 150 + 2 \times 20$

$= 182.5$ mm

Bearing pressure on each plate $P_b = P /(B \times C)$

$= 19532.25 / (10 \times 18.25)$

$= 107.03$ kg/cm^2

This is less than the permissible bearing pressure for concrete.

Stress in the plate $= [(107.03 / 2) \times (20^2 /10^2)] / (t^2 / 6)$

$= 12.84 / t^2$ kg/cm^2

But $f = 1575$ kg/cm^2

Therefore $t^2 = (12.84 / 1575) \times 100$ mm^2

$t = 0.8152$ mm

It is usual to select a plate of 4 to 6 mm thickness.

CONDENSER:

Design: The condenser is a horizontal condenser designed to condense 7652 Kg/Hr of 98% vapour of glycerine at 1 atm pr. and 190°C .The coolant used is water which is supplied in the tube side at an inlet temperature of 20°C and leaves at an outlet temperature of 35°C.

Heat of vapourisation of glycerine λ = 18170 cal./mole

$= 18170 \times 4.18 \times 92$ J/Kg

= 6987.46 KJ/Kg

Amount of heat removed from the vapour $Q = m \times \lambda$

$= (7652/3600) \times 6987.46 \times 1000$

= 14852.23 KJ/sec

AMOUNT OF WATER TO BE CIRCULATED:

$(m)_W \times C_p \times \Delta t = (m)_G \times \lambda$

$(m)_W \times 4.187 \times (35-20) = 14852.23$ KW

$(m)_W = (14852.23 \times 1000) / (4.187 \times 15)$

$(m)_W = 236.48$ Kg/Sec

Amount of water required = 236.48 Kg/sec

LOGARITHMIC MEAN TEMPERATURE DIFFERENCE:

LMTD = ((190-20)-(190-35))/(ln ((190-20)/(190-35)))

= (170 - 155)/ (ln (170/155))

= 162.36°C

OVERALL HEAT TRANSFER COEFFICIENT:

Assume U = 600 $W/m^{2\circ}k$

Total heat transfer area $A = Q/(U \times (\Delta t)_{LMTD})$

$= 14852.23/(600 \times 162.38)$

$= 152.45\ m^2$

Choose tubes of 5/4" O.D.,16 BWG ,length of 16 ft laid on a 25/16" square pitch.

Heat transfer area $A = 152.45\ m^2$

Number of tubes $N_t = 152.45/(\pi \times 1.25 \times 0.0254 \times (4.88-0.05))$

= 317

From tube count table,

For tube O.D. of 5/4" on 25/16" square pitch

TEMA P or S

No. of passes = 2

Nearest no. of tubes N_t = 310

I.D. of shell = 889 mm

Corrected heat transfer area = $n \times \pi \times d \times L$

$= 310 \times 3.14 \times 0.03175 \times 4.88$

$= 150 \text{ m}^2$

Corrected U = $(14852.23 \times 1000)/(150 \times 162.38)$

$= 614 \text{ W/m}^2{}^\circ\text{k}$

SHELL SIDE HEAT TRANSFER COEFFICIENT

CONDENSING VAPOUR SIDE:

Temperature of vapour coming in = 190ºC

Average temperature of water = (20 + 35)/2

= 27.5ºC

Wall temperature = (190 + 27.5)/2

=108.75ºC

Film temperature = (190 + 108.75)/2

=149.38ºC

So the property of water are taken at 149ºC

viscosity $\mu = 1.6\times10^3 \text{ Ns/m}^2$

Thermal conductivity K = 0.670 W/mºC

Density ρ =1175 Kg/m^3

Specific heat C_p = 3.05 KJ/KgºC

Mass flow rate per unit length = $W/(N_t^{2/3} \times L)$

$=7652/(3600\times310^{2/3}\times4.88)$

$=9.568\times10^3$ Kg/m sec

Renold's number $N_{re} = (4\times9.568\times10^3)/(1.6\times10^3)$

= 23.92

Outer film coefficient $h_c = 1.51(K^3\times\rho^2\times g/\mu^2)^{2/3}(N_{re}^{1/3})$

$h_c = 1.51(0.3008\times1380625\times9.81/2.56\times10^6)^{2/3}(23.92^{1/3})$

$= 1808.3 \text{ W/m}^2{}^\circ\text{k}$

TUBE SIDE HEAT TRANSFER COEFFICIENT:

Average temperature of water =27.5ºC

Physical properties of water at 27.5ºC

Specific heat C_p 4.18 KJ/KgºC

Viscosity $\mu = 0.9\times10^3$ Ns/m^2

Thermal conductivity K = 0.616 W/mºC

Prandtl number $P_r = (C_p \times \mu)/K$

$= 6.41$

Mass velocity of water = 236.48 Kg/sec

$a_t = (n\times\pi\times0.0284)/(4\times2)$

$= (310\times3.14\times0.0284)/8$

$= 0.0492$ m^2/pass

G_t = mass velocity/a_t

= 236.48/0.0492

= 2806.5 Kg /m^2sec

Inside diameter of the tube d_i = 1.12'

= 0.0284 m

$N_{Re} = (d_i \times G_t)/\mu$

$= (0.0284\times4806.5\times437)/(0.9\times10^3)$

$= 15192 > 10000$

Ditus boltern equation can be used

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

$= 0.023\times15192^{0.8}\times6.41^{0.3}$

$= 561.17$

h_i = 4518 w/m^2°k

CALCULATED OVERALL HEAT TRANFER COEFFICIENT:

$1/U_o = 1/h_o + 1/h_i\times(d_o/d_i) + 0.000528$

$= 1/1808.3 + 1/4518\times(1.25/1.12) + 0.000528$

$= 0.000553+0.000247+0.000528$

$=0.001328$

U_o = 753 W/m^2°k

Which is greater then the corrected U

Therefore this value of U is good enough.

PRESSURE DROP CALCULATIONS:

TUBE SIDE:

N_{re} = 15192

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

$= 0.004$

$\Delta P_L=(4fLv^2/2gd_i)\times\rho\times g$

$v_t= G_t/\rho$

= 4.82m/sec

$\Delta P_L= (4\times0.004\times4.88\times4.82^2/2\times0.0284)\times997$

=15.920 Kpa

$\Delta P_C = (2.5/2)\times\rho\times v_t^2$

=14.476 Kpa

$\Delta P_T = 2(\Delta P_L+\Delta P_C)$

= 2(15.920+14.476)

= 60.8Kpa

This is within the permissible limit of a maximum pressure drop of 70Kpa in the tube side

So this pressure drop is acceptable.

SHELL SIDE:

Mass flow rate of glycerin =7652/3600

=2.13 Kg/sec

Saturation temperature of vapour T_{vap}=190ºC

Clearance C = pitch –O.D.

= 25/16”-5/4”

= 7.94 mm

Pitch P_t = 25/16”

= 0.04 m

$a_s = (I.D)\times C\times B/P_t$

Here I.D.=B=D_s-diameter of shell = 889 mm

$a_s = 0.889\times0.889\times7.94\times10^3/0.04$

$= 0.157\ m^2$

Equivalent diameter $D_e = 4[P_t^2-(\pi\times d_o^2)/4]/(\pi\times d_o)$

$= 4[0.04^2-(3.14\times0.03175^2)/4]/(3.14\times0.03175)$

$= 0.0324m$

$G_s = 2.13/0.157$

$=13.57\ kg/sec\ m^2$

At 190ºC vapor viscosity $\mu_{vap} =1.1\times10^5\ Ns/m^2$

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

$=13.57\times0.0324\times10^5/1.1$

$= 39970$

$f =1.87(N_{re}^{0.2})$

$=0.225$

Density of vapour $\rho_{vap}= 2.42\ Kg/m^3$

Number of baffles $N_b+1=L/D_s$

$=4.88/0.889$

$=5.5$ i.e. 6

Therefore $N_b =5$

$\Delta P_S =2[(f\times(Nb+1)Ds\times G_s^2\times g)/(g\times D_e\times\rho_{vap})]\times0.5$

$=(0.225\times6\times0.889\times13.57^2)/(0.0324\times2.42)$

$=3194.4$ pa

$=3.194$ Kpa

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

So this is acceptable.

MECHANICAL DESIGN OF CONDENSER:

SHELL SIDE:

Material : Carbon steel

No. of shells : 1

No. of passes : 2

Fluid : 80 % Glycerine vapour

Internal diameter : 889mm

Working pressure : 0.1 N/mm^2

design pressure : 0.11 N/mm^2

Inlet temperature : 190ºC

Outlet temperature: 190ºC

Allowable stress : 950 Kg/Cm^2

TUBE SIDE:

Material : Stainless steel (IS grade 10)

No. of tubes : 310

Outside diameter : 31.75 mm

Length : 4.88 m

fluid : water

pitch : 39.69 mm (square)

Allowable stress : 10.06 Kg/m^2

working pressure : 1.033 Kg/cm^2

Design pressure : 1.55 Kg/cm^2

Inlet temperature : 20ºC

Outlet temperature : 35ºC

SHELL SIDE:

SHELL THICKNESS:

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

$= (0.11\times889)/((2\times95\times0.85)-0.11)$

$= 0.61$ 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

$= 7652/3600$

$= 2.13$ Kg/sec

Density $\rho = 1175$ kg/m^3

Assume velocity to be 10 m/sec

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

$d_n^2 = (2.13\times4)/(10\times3.14\times1175)$

$d_n = 0.015$ m

NOZZLE THICKNESS

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

$= (0.11\times15)/(2\times95\times0.85-0.11)$

$= 0.10$ 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. = 711.2 mm

R_K - Knuckle radius 10% of shell I.D.= 88.9 mm

W = 1.46

$t_h = 0.71$ mm

Using same thickness as that of the shell = 8 mm

BAFFLE ARRANGEMENT:

Tansverse bafffles

Number of baffles =5

Baffle Spacing = D_s = 889 mm

Thickness of baffles = 6 mm

Height of baffle = 0.75 × D_s

= 666 mm

TIE RODS AND SPACERS:

For shell diameter D_s = 889 mm

No. of tie rods = 6

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 =889 mm

Shell thickness t_s = 8mm

Shell outside diameter = $(2\times t_s)+889$

= (2×8) +889

= 905 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 = 889 mm

G_i = 889 +(2×8)

= 905 mm

Mean gasket width = G_i+N

= 915 mm

therefore G = 915 mm

Estimation of bolt load:

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

= $(\pi \times 0.915^2 \times 0.11)/4$

= 0.072 MN

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

Load to keep joint tight under pressure $H_p = \pi(2b)GmP_d$

= 3.14×0.01×0.915×2.75×0.11

= 0.0087 MN

Total operating load W_o =H+H_p

=0.072+0.0087

= 0.0807 MN

Load to seat gasket under bolting up condition $W_b = \pi bGP_d$

=3.14×0.915×0.005×0.11

= 0.367 MN

Since W_b>W_o, controlling load =0.367 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 190ºC=138 MN/m^2

Therefore A_{m1} = 0.0807/138

= 0.000585 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} = 0.367/138

= 0.00266 m^2

Since $A_{m2} > A_{m1}$,$A_m = A_{m2}$ = 0.00266 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)

= 915/36

= 25

Actual number of bolts = 24

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 = 0.905 m

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

$g_1 = 0.008$

C = 2(0.027 + 0.008) + 0.905

= 0.975 m

Therefore bolt circle diameter = 0.975 m

Flange outside diameter:

A = C + bolt diameter + 0.02

= 0.975 + 0.018 + 0.02

= 1.013 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, $A_b S_g/\pi GN = 15.86$

Since 15.86 > 2y

Condition is satisfied.

Flange moment computations:

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

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

$= (3.14 \times 0.905^2 \times 0.11)/4$

= 0.071 MN

$W_2 = H - W_1$

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

$= (3.14 \times 0.915^2 \times 0.11)/4$

$= 0.0732$ MN

$W_2 = 0.0723 – 0.071$

$= 0.0013$ MN

$W_3 = W_o – H$

$= 0.0807 – 0.0723$

$= 0.0084$ MN

Total flange moment ,

$M_o = W_1a_1 + W_2a_2 + W_3a_3$

$a_1 = (C – B)/2$

$= 0.035$ m

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

$= 0.03$ m

$a_2 = (a_1+a_2)/2$

$= 0.0325$ m

$M_o = 0.071\times0.035 + 0.0013\times0.03 + 0.0084\times0.0325$

$= 2.78 \times 10^{-3}$ MNm

Bolting up condition:

Total flange moment $M_g = Wa_3$

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

$= 0.00472\times138$

$= 0.651$ MN

$a_3 = 0.03$ m

Therefore $M_g = 0.01953$ MNm

$= 19.5 \times 10^{-3}$ MNm

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

Therefore $M = M_g = 19.5\times10^{-3}$ MNm

Flange thickness:

$t^2 = (MC_FY)/BS_T$

$K = A/B$

= Outer diameter of flange/ inner diameter of flange

$= 1.013/0.905$

$= 1.12$

$Y = 14$

Assume $C_F = 1$

Therefore thickness 't' = 0.055 m

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

$= (3.14\times0.975)/24$

$= 0.128$ m

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

$= [0.128/(2\times0.018+0.055)]^{1/2}$

$= 1.19$

Therefore $C_F^{1/2} = 1.091$

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

$= 1.091\times0.055$

$= 0.06$ m

$= 60$ mm

TUBE SIDE:

TUBE THICKNESS:

$t_t = Pd_o/2fJ+P$

J =1 for seamless tube

Therefore $t_t = (1.55\times31.75)/(2\times1006+1.55)$

$= 0.024$ 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 =915 mm

Therefore $t_S = 915[(0.25\times1.55)/1006]^{1/2}$

$= 17.96$ 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 =$915[(0.3\times1.55)/950]^{1/2}$

= 20.24 mm

With corrosion allowance t = 25 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=905 mm

Length of shell =4.88 m

Torispherical head:

Crown radius =711.2 mm

Knuckle radius =88.9 mm

Working pressure =1Atm

Shell thickness = 8 mm

Head thickness = 8mm

Corrosion allowance = 2mm

permissible stress = 950 Kg/cm^2

R-Vessel radius = 425.5

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

= 203.63 mm < 0.2L

Longitudinal bending moment:

The bending moment at the support is;

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

$A = 203.63$ 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

$W_1 = [3.14(0.905^2-0.889^2)\times 4.88\times 7700]/4$

$= 847$ 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

$W_2 = 310[3.14(0.03175^2-0.0298^2)\times 4.88\times 7800]/4$

$= 1112$ Kg

Weight of tube sheet:

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

$= (2\times 3.14\times 0.905^2\times 0.0180\times 7800)/4$

$= 180$ Kg

Liquid load in the shell:

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

$= [(\pi D_s^2 L)/4 - (n\pi d_o^2 l)/4]\times 1175$

$= [(\pi\times 0.905^2\times 4.88)/4 - (310\times\pi\times 0.03175^2\times 4.88)/4]\times 1175$

$= (3.14 - 1.20)\times 1175$

$= 2280$ Kg

Liquid load in tubes:

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

$= (310\times 3.14\times 0.0298^2\times 4.88\times 997)/4$

$= 1052$ Kg

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

$= 847 + 1112 + 180 + 2280 + 1052$

= 5471 Kg

Hence, Q =5471/2[4.88+(4×0.257)/3]

= 14287 Kgm

M_1=(14287×0.20363)[1-{(1-0.20363/4.88)+(0.4525^2-0.257^2/2×0.203×4.88)/(1+4×0.257/3×4.88)}]

=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)]

=(14287×4.88/4)[{1+2(0.4525^2-0.257^2)/4.88^2}/{1+(4×0.257/3×4.88)}- (4×0.203/4.88)]

= 17430[1.312 - 0.1664]

= 19968 Kgm

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 m

t-thickness of shell = 8 mm

f_1 =115.6/(0.107×3.14×0.4525^2×0.008)

=20.93 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 m

f_2 =115.6/(0.192×3.14×0.4525^2×0.008)

=11.67 Kg/cm^2

Stress in the shell at mid point:

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

=19968/(3.14×0.4525^2×0.008)

=388 Kg/cm^2

Thus the values of stresses are within the limited range

Hence the designed support is acceptable.

COST ESTIMATION:

Glycerin plant size =100 T/day

Fixed capital investment for cost index of 130 = Rs 10^8

Cost index for 2002 = 402

Therefore present fixed capital investment $=10^8\times(402/130)$

=Rs 30,92,30,769

Estimation of total investment cost:

1) Direct cost:

a) Purchased equipment cost:(15 – 40% of FCI)

Assume 40% of FCI

=Rs 12,36,92,307

b) Installation cost:(35 – 45% of PC)

Assume 45%

=Rs 13,91,53,846

c) Instrument and control installed:(6 –30% of PEC)

Assume 30% of PEC

=Rs 9,27,69,230

d) Piping installation cost:(10 –80% of PEC)

Assume 75%

=Rs.23,19,23,076

e) Electrical installation cost:(10 – 40% of PEC)

Assume 40% of PEC

=Rs 12,36,92,307

f) Building process and auxilliary:(10-70% of PEC)

Assume 65%

=Rs 20,09,99,999

g) Service facilities:(30-80% 0f PEC)

Assume 75%

=Rs 23,19,23,076

h) Yard improvement:(10-15% of PEC)

Assume 15%

=Rs 4,63,84,615

i) Land:(4-8% of PEC)

Assume 8%

=Rs 2,47,38,461

Therefore direct cost =Rs 1,21,52,76,917

2) Indirect cost:

Expenses which are not directly involved with material and labour of actual installation or complete facility

a) Engineering and supervision:(5-30% of DC)

Assume 30%

=Rs 36,45,83,075

a) Construction expenses:(10% of DC)

=Rs 12,15,27,691

b) Contractors fee:(2-7% 0f DC)

Assume 7%

=Rs 8,50,69,584

c) Contingency:(8-20% of DC)

Assume 10%

=Rs 12,15,27,691

Therefore total indirect cost =69,27,08,042

3) Fixed capital investment:

Fixed capital investment(FCI) = DC+IC

= Rs 1,90,79,84,959

4) Working capital investment:

10 –20% of FCI

Assume 15%

=Rs 28,61,97,743

5) Total capital investment:

= FCI + WC

=Rs 2,19,41,82,703

Estimation of total product cost(TPC):

Fixed charges:

a) Depreciation:(10% of FCI for machinery)

=Rs 2,19,41,827

b) Local taxes:(3-4% of FCI)

Assume 3%

=Rs 6,58,25,481

c) Insurances:(0.4-1% of FCI)

Assume 1%

=Rs 2,19,41,827

d) Rent:(8-12% of FCI)

Assume 9%

=Rs 4,57,97,372

Therefore total fixed charges =Rs 15,55,06,507

But, Fixed charges = (10-20% of TPC)

Assume 10%

Therefore Total product cost =155506507/0.1

=Rs 1,55,50,65,070

Direct production:

a) Raw material:(10-50% 0f TPC)

Assume 30%

=Rs 46,65,19,521

b) Operating labour(OL):(10-20% of TPC)

Assume 15%

=Rs 23,32,59,760

c) Direct supervisory and electric labour:(10-25% of OL)

Assume 15%

=Rs 3,49,88,964

d) Utilities:(10-20% of TPC)

Assume 15%

=Rs 23,32,59,760

e) Maintainence:(2-10% of FCI)

Assume 6%

=Rs 9,33,03,904

e) Operating supplies (OS):(10-20% of maintainence)

Assume 15%

=Rs 1,39,95,585

f) Laboratory charges:(10-20% of OL)

Assume 15%

=Rs 3,49,88,964

g) Patent and royalties:(2-6% of TPC)

Assume 4%

=Rs 6,22,02,602

Plant overhead cost:

50-70% of (OL+OS+M)

Assume 60%

=Rs 20,43,35,549

General expenses:

a) Administration cost:(40-60% of OL)

Assume 55%

=Rs 12,82,92,868

b) Distribution and selling price:(2-30% of TPC)

Assume 15%

=Rs 23,32,59,760

c) Research and development cost:(3% of TPC)

=Rs 4,66,51,952

Therefore general expenses(GE) =Rs 40,82,04,580

Therefore manufaacturing cost(MC)= Product cost+fixed charges+Plant overhead expenses

= Rs 1,91,49,07,126

Total production cost:

Total production cost =MC + GE

=Rs 2,32,31,11,706

Gross earnings and rate of return:

The plant is working for say 300 days a year

Selling price =Rs. 97/kg

Total income =100×300×1000×97

=Rs 2,91,00,00,000

Gross income =Total income – total product cost(1555065070)

=Rs 1,35,49,34,930

Tax =50%

Net profit =Rs 67,74,67,465

Rate of return = net profit/total capital investment

=30.87%