PROCESS DESIGN

ROTARY DRIER

The rotary drier forms an important part of the plant. The drier has a capability of removing about 60% of the water entering the feed. The drier is a counter current drier, with air as the heating medium. The RH of the air entering is 10%. The air entering is heated to a temperature of 156ºC. The slurry is fed into the drier at a temperature of 80ºC.

Feed to the drier:

Water in feed = 1722.38 lb/Hr

Dry Solid in feed = 68205.51 lb/Hr

Water Content in Product = 688.94 lb/Hr

Water removed by the drier = 1033.44 lb/Hr

Condition of inlet air:

Ambient temperature of air = 30ºC

RH = 10%

Wet bulb temperature = 22ºC

Humidity W_G = 0.002 lb water/lb dry air.

Inlet temperature of air = 156ºC

To find the wet bulb temperature of inlet air:

$W_G - W_H = h_G (t_G - t_w)/(m\lambda_W Pk_G)$

W_G Humidity of air at temperature t_G ºF

Ww Humidity of air at temperature t_w ºF

t_G Temperature of inlet air ºF

t_W Wet bulb temperature ºF

M Molecular wt of air

λ_w Latent Heat of vaporization at t_wºF

$h_G/(m Pk_G) = 0.26$ for air at t_w.

Trial 1: assume wet bulb temperature is 90ºC = 194ºF

$W_w = 0.046$

$W_G - W_w = 0.044$

$\Phi = (0.26 (313 - 194)/547.3) = 0.056$

$\Phi > W_G - W_w$

Therefore the temperature assumed is high

Trial 2:

Assume a wet bulb temperature of 180ºF

W_w = 0.065

$W_G – W_w$ = 0.063

Φ = (0.26 (313-180)/532) = 0.063

Therefore wet bulb temperature assumed is true.

Therefore t_w = **180ºF**

To Calculate the Outlet Temperature of Air:

Emperically it is found that the drier operates economically when total number of transfer units (NTU) is between 1.5 to 2.5 (Badger Pg 508)

NTU = $\ln ((t_{G1} – t_W)/(t_{G2} - t_W))$

Take NTU = 2.5

t_{G2} = **190.92ºF**

Heat Balance:

C_p(PTA) = 0.2871 Btu/lbºF , C_p(Water) = 1 Btu/lbºF

Product discharge temperature = (313 + 190)/2 = 251.5ºF

Temperature of the feed = 176ºF

Heat required to raise the product to discharge temp. = 68205.51x 0.2871 (251.5-176) + 688.94 (251.5 – 176)

= 1.53 x 10^3 Btu/Hr

Heat reuired to remove water = 1033.44((180-176) + 0.45 (190-180) + 550

= 5.771 x 10^5 Btu/Hr

Total Heat = **2.107 x 10^6 Btu/hr**

Air Required:

S_1 - Humid Heat of inlet air

S_1 = 0.24 = 0.45 x 0.002

= 0.2409

Take average humid heat = 0.242

G_G^1 Sx Humid Heat of air x Temp difference = Total Heat

$G_G^1\ S\ (0.242)(313 - 190) = 2.107 \times 10^6$

$$G_G^1\ S = \frac{2.107 \times 10^6}{0.242\ (313 - 190)}$$

G_G^1 S = **70785.46 lb /Hr**

$$\text{Humidity of outlet air} = \frac{1033.4}{70785.46} + 0.002$$

= 0.0145 lb water /lb dry air

Humid heat = 0.24 +0.45 x 0.014

= 0.247

$$S_{av} = \frac{0.2409 + 0.246}{2} = 0.244$$

Therefore the average humidity taken above is valid

To Calculate the mean temperature difference acreoss the Rotary Drier:

Let q_p = Heat required to preheat the feed from inlet to wet bulb temperature.

q_s = Heat required to heat product from wet bulb temperature to discharge temperature.

q_v = Heat required to evaporate water at wet bulb temperature.

Preheating period:

q_p = 68205.51 x 0.2871 (180-176) + 1722.38 (180 – 176)

= **8.52 x 10^4 Btu/Hr**

$$\text{Change in air temperature} = \frac{8.527 \times 10^4}{2.107 \times 10^6} \times (313 - 190)$$

= 51.85 ºF

Air temperature at the end of preheat = 190 + 51.85

= 241.85 ºF

$$(\Delta T)_p = \frac{(190 - 158) - (241.85 - 180)}{\ln\left[\dfrac{190 - 158}{241.85 - 180}\right]}$$

= **45.29 ºF**

Heating period:

q_s = 68205.51 x 0.2871 (251.5 – 180) + 688.94 (251.5 – 180)

= **1.449 x 10^6 Btu/Hr**

Change in air temperature = $\dfrac{1.449 \times 10^6}{2.107 \times 10^6} \times (313 - 190)$

= 81.87 ºF

Air temperature at the start of heating = 313 - 81.87

= 228.13 ºF

$$(\Delta T)_s = \frac{(228.13 - 180) - (313 - 251.5)}{\ln\left[\dfrac{228.13 - 180}{313 - 251.5}\right]}$$

= **54.54 ºF**

Evaporating Period:

q_p = 2.107 x 10^6 – 1.449 x 10^6 - 8.52 x 10^4

= **5.658 x 10^5 Btu/Hr**

$$(\Delta T)_v = \frac{(228.13 - 180) - (241.85 - 180)}{\ln\left[\dfrac{228.13 - 180}{241.85 - 180}\right]}$$

= **54.7 ºF**

We have the mean temperature difference given by (Page 510 Badger)

$$\frac{1}{(\Delta T)_m} = \frac{1}{q_t}\left[\frac{q_p}{(\Delta T)_p} + \frac{q_s}{(\Delta T)_s} + \frac{q_v}{(\Delta T)_v}\right]$$

$$= \frac{1}{2.49 \times 10^6}\left[\frac{8.52 \times 10^4}{45.29} + \frac{1.449 \times 10^6}{54.54} + \frac{5.658 \times 10^5}{54.7}\right]$$

$$= 0.0202$$

$$(\Delta T)_m = \mathbf{64.18\ ^{o}F}$$

NTU Check:

$$NTU = \frac{(T_1 - T_2)}{(\Delta T)_m}$$

$$= (313 - 190)/64.18$$

$$= \mathbf{1.916}$$

According to the condition NTU should be between 1.5 and 2.5. Therefore the above mean temperature value can be accepted.

TRIAL 1:

To Calculate the Diameter of the Drier:

The Design of the drier depends majorly on the amount of air entering the drier and the velocity with which it enters the drier. The air entering the drier is 70785 lbs. for designing the air is taken in excess so that the loss of heat from the drier is compensated.

So the air entering the drier can be taken as 71000 lb/Hr

Assume that the maximum superficial air mass velocity to be = 1000 lb/(Hr ft^2)

$$G_G\ S = 71000\left[1 + 0.0165 \times \frac{70785}{71000}\right]$$

$$= 72168 \text{ lb/Hr}$$

$$S = \frac{72168}{1000}$$

$= 72.168\ ft^2$

$D = (4 \times 72.168 / \Pi)^{0.5}$

$= 9.58\ ft = 2.92\ m$

To Calculate the overall heat transfer coefficient:

We have by empirical equation

$$U_a = \frac{15\,(G_G)^{0.16}□}{D}$$

G_G = Maximum superficial air mass velocity, lb/ft^2 Hr

U_a = The Overall heat transfer coefficient (volumetric), Btu/Hr ft^3 ºF

D = Diameter of the Drier in ft.

$$U_a = \frac{15\,(1000)^{0.16}□}{9.58}$$

$= 4.75$ Btu/Hr ft^3 ºF

To calculate the Length of the drier:

$Q = U_a SZ (\Delta T)_m$

Where,

Q = Total heat,Btu/Hr

Z = Length of the drier, ft

S = Heat transfer area,ft^2

$$Z = \frac{2.107 \times 10^6}{4.73 \times 72.2 \times 64.18}$$

Z = 29.2 m

Z/D ratio Check:

The Z/D ratio for a drier should be between 3 – 10. The Z/D ratio in the above trial is 10. for the drier to work efficiently it is better to reduce the ratio.

TRIAL 2:

To Calculate the Diameter of the Drier:

Assume that the maximum superficial air mass velocity to be = 800 lb/(Hr ft^3)

$$G_G\,S = 71000\left[1 + 0.0165 \times \frac{70785}{71000}\right]$$

$$= \mathbf{72168\ lb/Hr}$$

$$S = \frac{72168}{800}$$

$$= \mathbf{90.21\ ft^2}$$

$$D = (4 \times 90.21/\Pi)^{0.5}$$

$$= \mathbf{10.71\ ft = 3.26\ m}$$

To Calculate the overall heat transfer coefficient:

We have by empirical equation

$$U_a = \frac{15\,(G_G)^{0.16}}{D}$$

$$U_a = \frac{15\,(800)^{0.16}}{10.71}$$

$$= \mathbf{4.08\ Btu/Hr\ ft^3\ ^oF}$$

To calculate the Length of the drier:

$$Q = U_a\,SZ\,(\Delta T)_m$$

$$Z = \frac{2.107 \times 10^6}{4.08 \times 90.21 \times 64.18}$$

$$Z = \mathbf{88.9\ ft = 27.07m}$$

Z/D ratio Check:

$$Z/D = \frac{27.07}{3.26}$$

$= \mathbf{8.3}$

Which checks the condition that the Z/D ratio is between 3 and 10

Therefore the above diameter and length can be taken

To Calculate the Speed of rotation of the drier:

Assume the peripheral speed of rotation to be 30 ft/min

Revolutions per minute = Peripheral Speed/Diameter

RPM = 30/10.29 = 2.8

= **3**

The revolution of a drier varies between 2 – 5

Therefore the above value can be accepted.

Flight Design:

Number of flights in the drier = 3 x D

Where D is the diameter of the drier in ft

Number of flights = 3 x 10.69 = **32**

Radial height of the Flight:

The radial height of the flight is taken as 1/8th of the diameter of the drier.

The radial height of the flight = (1/8) x 10.69

= **16.03 inches**

DRIER DETAILS:

Drier Type : Counter Current Rotary Drier

Diameter of the drier = 10.69 ft = 3.26 m

Length of the Drier = 88.9 ft = 27.07 m

RPM of the drier = 3 rpm

Number of Flights = 32

Radial Height of the flights = 16.03 inches

Temperature of the inlet air = 156 ºC

RH (%) of the inlet air 10%

Temperature of inlet wet solid = 90 ºC

Mean Temperature Difference = 17.88 ºC

Air mass flow rate = 71000 lb/hr

Moisture removed by the Drier = 2812.5 Kg/Hr

The Volumetric Heat transfer Coefficient of drier = 4.08 Btu/Hr ft^3 ºF

CONDENDER DESIGN

The condenser is used to condense the vapor leaving the residue still. Part of the vapor condensed is sent back to the still as reflux. Assume that the vapor entering is saturated and the condenser removes only the latent heat. i.e. the liquid leaving is a saturated liquid. The vapor comprises of acetic acid and water with a saturation temperature of 101.4 ºC. The cooling fluid used is treated water in the tube side.

Feed to the condenser = 10.903×10^3 Kg/Hr

$\underline{m}$ = 2.84 Kg/s

$\overline{C_P}$ = 1.162 KJ/Kg K

$\overline{\lambda}$ = 2001.37 KJ/Kg K

Heat of vapor = $2001.37 \times 10.224 \times 10^3$

Q = 20.462×10^6 KJ/Hr

= **6062.81 KW**

To calculate the amount of Cooling water Required:

Cooling water is treated water and assume that the water leaves the condenser at a temperature of 40 ºC.

$Q = mC_P \Delta T$

$$m = \frac{6062}{4.187\,(40 - 25)}$$

The Cooling water required = **96.52 Kg/s**

To find the LMTD:

$$(\Delta T)_{lmtd} = \frac{(101.4 - 40) - (101.4 - 25)}{\ln\left[\dfrac{101.4 - 40}{101.4 - 25}\right]}$$

$(\Delta T)_{lmtd}$ = **68.62 ºC**

To Calculate the Heat Transfer Area:

From the table assume the heat transfer coefficient = 576.8 W/m^2K

$Q = U_a A (\Delta T)_{lmtd}$

$$A = \frac{6062.81 \times 10^3}{576.8 \times 68.62}$$

= **153.178 m^2**

To Calculate the Number of Tubes:

Take the pipe to be a 16 BWG pipe with 0.75”

O.D. = 0.75” , I.D. = 0.745” , Length = 3.66 m , a = 0.0598 m^2

Number of tubes N_t = A/(a x L)

N_t = (153.178/(0.0598 x 3.66))

= **699**

To find the dimensions of the Shell:

From the table 11-3 (Perry – 11-14)

Triangular pitch 1”, 1-2 pass Heat Exchanger

N_t = **716**

Shell ID = **787 mm**

Corrected Heat transfer Area = 716 x 0.5987 x 3.66

= **156.7 m^2**

Corrected Heat transfer coefficient = $$U_o = \frac{Q}{A \times \Delta T} = \frac{6062.81 \times 10^3}{156.7 \times 68.62}$$

$(U_o)_{assumed}$ = **563.8 W/m^2K**

Calculation Of Film Transfer Coefficient:

T_w = (101.4 + (40+25)/2)/2 = 66.95 ºC

T_f = (101.4 + 66.95)/2 = 84.13 ºC

Properties:

ρ_l = 962.81 Kg/m^3

μ_l = 0.778 x 10^{-3} N-s/m^2

k = 0.1979 W/m^{2}K

Shell Side FTC:

$$N_{Re} = \frac{4\ W}{\mu N_t\ \Pi\ D_o}$$

$$= \frac{4 \times 2.84}{(0.778 \times 10^{-3} \times 716 \times \Pi \times 1.905 \times 10^{-2})} = \mathbf{340.95}$$

$$h_o = 1.51\left[\frac{k^3\ \rho^2\ g}{\mu^2}\right]^{1/3} (N_{Re})^{-1/3}$$

$$h_o = 1.51\left[\frac{0.1979^3\ \ 962.81^2\ 9.81}{(0.778 \times 10^{-3})^2}\right]^{1/3} (340.95)^{-1/3}$$

= **1055.7 W/m^{2}K**

Tube Side FTC:

Properties:

ρ = 1000 Kg/m^3

μ = 1 x 10^{-3} N-s/m^2

k = 0.578 W/m^{2}K

C_p = 4.187 KJ/Kg °C

Flow area = 281.24 x 10^{-6} m^2/m

a_t = (716 x 281.24 x 10^{-6})/2

= **0.1006**

G_t = 96.52/0.1006) = **959.4 kg/m^2 s**

$$N_{Pr} = C_P\ \mu / k$$

$$= \frac{4.187 \times 10^{3} \times 1 \times 10^{-3}}{0.578}$$

$$= \mathbf{7.244}$$

$$N_{Re} = D\ G_t / \mu$$

$$= \frac{959.4 \times 1.8923 \times 10^{-2}}{1 \times 10^{-3}}$$

$$= \mathbf{18151.85}$$

$$\frac{h_i\ D_i}{k} = 0.023 \times (N_{Re})^{0.8}\ (N_{Pr})^{0.3}$$

$$h_i = \frac{0.023 \times (18151.85)^{0.8} \times (7.244)^{0.3} \times 0.578}{1.8923 \times 10^{-2}}$$

$$= \mathbf{3249.34\ W/m^2K}$$

Overall Outside Heat Transfer Coefficient:

$$\frac{1}{U_o} = \frac{D_o}{D_i}\frac{1}{h_i} + \frac{1}{h_o} + \frac{1}{h_{dirt}}$$

$$\frac{1}{U_o} = \frac{0.75}{0.745}\frac{1}{3249.34} + \frac{1}{1055.7} + 5.3 \times 10^{-3}$$

$$= 0.00176$$

$$U_o = \mathbf{568.18\ W/m^2K}$$

$(U_o)_{calculated} > (U_o)_{assumed}$

Therefore the above value of shell and tube dimension can be accepted

Pressure Drop Calculations:

Shell Side:

$T_{vap} = 101.4\ ^{o}C$

$\mu_{vap} = 0.0118 \times 10^{-3}\ N\text{-}s/m^2$

$\rho_{vap} = 0.595\ Kg/m^3$

$$a_s = \frac{(D_s \times C^1 \times B)}{P_T}$$

$B = (0.15 \text{ to } 1)\ D_s$

Let us assume the baffle spacing is equal to D_s

Therefore $B = 787 \times 10^{-3}$ m

$C^1 = P_T – D_o$

$C^1 = 1” – 0.75”$

$= 0.00635$ m

$$a_s = \frac{(787 \times 10^{-3} \times 0.00635 \times 787 \times 10^{-3})}{2.54 \times 10^{-2}}$$

$= \mathbf{0.19675\ m^2}$

Equivalent Diameter:

$$D_e = \left[\frac{\frac{P_T}{2} \times 0.86 \times P_T - \frac{1}{2}\left[\frac{\Pi D_o^2}{4}\right]}{\frac{\Pi D_o}{2}} \right]$$

$$D_e = \left[\frac{\frac{2.54\text{x}10^{-2}}{2} \times 0.86 \times 2.54\text{x}10^{-2} - \frac{1}{2}\ \frac{\Pi \times (1.905\text{x}10^{-2})^2}{4}}{\frac{\Pi\ 1.905\text{x}10^{-2}}{2}} \right]$$

$= \mathbf{18.03 \times 10^{-3}\ m}$

G_s = (2.84)/0.1967

= **14.438 Kg/m²s**

$$(N_{Re}) = \frac{G_s \times D_e}{\mu_{vap}}$$

$$= \frac{18.03 \times 10^{-3} \times 14.438}{0.0118 \times 10^{-3}}$$

= **22080.35**

$f = 1.87\,(N_{Re})^{-0.2}$

= $1.87\,(22080.35)^{-0.2}$

= **0.2529**

$N_b + 1 = 3.66/0.787 = 4.65$

$N_b = 3.65 =$ **4**

$$\Delta P_s = \left[\frac{4\, f\, (N_b + 1)\, D_s\, G_s^2}{2\, D_e\, \rho_{vap}}\right] \times 0.5$$

$$\Delta P_s = \left[\frac{4\ 0.2529 \times 5 \times 0.787 \times 14.438^2}{2 \times 0.01803 \times 0.595}\right] \times 0.5$$

= **19.1482 KPa**

Which is slightly greater than he allowed of pressure drop in the shell side which is 14 KPa value. The above value can be accepted since the number of baffles used is minimum and the pressure drop is not very high when compared o the allowed value.

Tube Side:

$f = 0.079\ (Re)^{-0.25}$

$= 0.079\ (18151.8)^{-0.25}$

$= \mathbf{6.806 \times 10^{-3}}$

$$\Delta P_l = \left[\frac{4\ f\ L\ V_t^2}{2\ g\ D_i} \right]$$

$V_t = 959.4/1000 =$ **0.959 m/s**

$$\Delta P_l = \left[\frac{4 \times 6.806 \times 10^{-3} \times 3.66 \times 0.959^2}{2 \times 9.81 \times 1.823 \times 10^{-2}} \right]$$

$=$ **2.424 KPa**

$$\Delta P_t = \frac{2.5}{2} (\ \rho_t\ V_t^2)$$

$$\Delta P_t = \frac{2.5}{2} (\ 1000 \times 0.959^2)$$

$=$ **1.149 KPa**

$\Delta P_{total} = N_b\ (\Delta P_l + \Delta P_t)$

$= 2\ (2.424 + 1.149)$

$=$ **7.14 KPa**

The Pressure drop in the tube side is also well within the allowed limit therefore the dimensions of the tube also can be accepted.

CONDENSER DETAILS

Type Of Condenser : 1-2 Pass Counter Current Floating Head Condenser

Heat Load on the Condenser: 6062.81 KW

Shell Side: (Acetic Acid + Water) vapors

Tube Side: Treated Cooling Water

Mass Flow Rate on the Shell Side: 2.84 Kg/s

Mass Flow Rate on the Tube Side: 96.52 Kg/s

Temperature Difference: 68.62 ºC

Heat Transfer Area: 156.7 m^2

Diameter of the Shell : 787 mm

Number of Tubes: 716

Type of Tube Used : 16BWG Tubes

Inner Diameter of the Tube: 0.745”

Outer Diameter of the Tube: 0.75”

Length of the Tube: 3.66 m

Pitch of the Tubes: 1” Δ

Heat Transfer Coefficient: 568.18 W/m^2K

Number of Baffles: 4

Pressure Drop on Shell Side: 19.1482 KPa

Pressure Drop on Tube Side: 7.14 KPa

MECHANICAL DESIGN

ROTARY DRIER

The drier at any point of time has a temperature of around 150 ºC. Therefore the material used to construct the drier should withstand the high temperature. The material used to construct the drier is mild steel. Since mild steel can withstand temperature of about 200ºC.

Working Pressure in the drier = 101.3 KPa = 0.1013 N/mm^2

Design pressure = 1.5 x WP

= 1.5 x 0.1013 = 0.152 N/mm^2

Permissible Stress of the material used = 124 N/mm^2

Inner Diameter of the drier = 3.258 m = 3258 mm

Length of the Drier = 27m

To find the thickness of the drier shell:

$$t_s = \frac{p\,D}{2\,f\,J + p}$$

p – Design pressure, N/mm^2

D – Diameter of the drier, mm

F – permissible stress, N/mm^2

J – 0.85

$$t_s = \frac{0.152 \times 3258}{2 \times 0.85 \times 124 + 0.152}$$

= 2.34 mm

But the minimum thickness is given as 8 mm for the shell

Therefore including corrosion allowance the thickness can be taken as **10 mm**

Therefore the outer Diameter = D_i + 2 x 10

= 3258 + 2 x 10

= **3278 mm = 3.278 m**

To Find the Thickness of the insulation:

From the heat balance it is clear that there is some amount of heat lost into the atmosphere. To limit the heat loss to the same figure an insulation is to be given to the drier. The insulating material can be chosen as asbestos.

Density of asbestos = 577 Kg/m^3

Thermal Conductivity of asbestos = 681.4 x 10^{-3} W/m^2 K

Material of the drier is mild steel

Thermal Conductivity = 147.6 W/m^2 K

Convective Heat transfer Coefficient = 56.78 W/m^2 K

From Heat Balance

Heat Loss from the Drier = 97.006 KW

D_1 = 3.258 m

D_2 = 3.278 m

t_1 = 10 mm

Let 'y' be the thickness of insulation.

$D_3 = D_2 + 2y$

T_1 = 122 ºC , T_2 = 76 ºC

We have from the Continuity equation

$$Q = \left[\frac{(T_1 - T_2)}{\frac{t_1}{k_1 A_1} + \frac{t_2}{k_2 A_2} + \frac{1}{H_c A_3}} \right]$$

$A_1 = \Pi\ (D_1 + D_2) \text{ x } L/\ 2$

$= \Pi\ (3.258 + 3.278) \text{ x } 27\ /2$

$= 277.2\ m^2$

$A_2 = \Pi\ (D_2 + D_3) \text{ x } L\ /\ 2$

$= \Pi\ (3.278 + 3.278 + 2y) \text{ x } 27/2$

$= 278 + 84.82\ y\ m^2$

$A_3 = \Pi \times D_3 \times L$

$= \Pi\ (3.278 + 2y) \times 27$

$= 278 + 169.65\ y\ m^2$

$$97.006 \times 10^3 = \frac{(122 - 76)}{\dfrac{10 \times 10^{-3}}{47.6 \times 277.2} + \dfrac{y}{681.4 \times 10^{-3}(278 + 84.82\ y)} + \dfrac{1}{56.78\ (278 + 169.65\ y)}}$$

$$97.006 \times 10^3 = \frac{46\ (2.99 \times 10^6 + 2.736 \times 10^6\ y + 5.56 \times 10^5\ y^2)}{9.63 \times 10^3\ y^2 + 15.84 \times 10^3\ y + 190.16}$$

$908.59 \times 10^6\ y^2 + 1.41 \times 10^9\ y - 119.1 \times 10^6 = 0$

$y^2 + 1.551y - 0.131 = 0$

$y = 0.08$ m

Therefore the thickness of the insulation should be **80mm.**

To find the power to drive the Drier:

From Eqn (20- 44) Perry.

$$\text{Power} = \frac{r\ (4.75\ d\ w + 0.1925\ D\ W + 0.33\ W)}{100000}$$

Where r – rpm of the drier

d – shell diameter, ft

W – total rotating load, lb

w – live load, lb

D – riding ring diameter, ft (d+2)

To calculate the live load and the rotating load:

Density of mild steel = 480 lbs/ft^3

We have D_2 = outer diameter of the drier shell

D_1 = inner diameter of the drier shell

$$\text{Volume of shell material} = \frac{(\Pi D_2^2 - \Pi D_1^2)\ L}{4}$$

$$= \frac{(\Pi\ 10.72^2 - \Pi\ 10.69^2)\ 88.6}{4}$$

$$= \mathbf{44.695\ ft^3}$$

Weight of the drier = volume of the shell material x density

= 44.695 x 480

= **21453.73 lbs**

Assume Hold up = 0.1

$$\text{Volume of the drier filled with material} = \frac{\Pi D_1^2}{4} L \times 0.1$$

$$= \Pi\ 10.69^2\ 88.6 \times 0.1/4$$

$$= \mathbf{799.26\ ft^3}$$

Weight of the material in drier at any time = Volume x density

= 799.26 x 94.07 lbs

= **75.18 x 10^3 lbs**

$$\text{Volume of the insulating material} = \frac{(\Pi D_3^2 - \Pi D_2^2)\ L}{4}$$

$$= \frac{(\Pi\ 10.98^2 - \Pi\ 10.72^2)\ 88.6}{4}$$

$$= \mathbf{392.61\ ft^3}$$

Weight of the insulating material = volume x density

= 392.61 x 36

= **14133.81 lbs**

W = weight of the drier + weight of the insulation + weight of the material

= 21453.73 + 75.18 x 10^3 + 14133.81 = **1.1076 x 10^5 lbs** w = weight of the material

= **75.18×10^3 lbs**

D = d + 2

= 10.69 + 2 = **12.69 ft**

r = 3 rpm

$$BHP = \frac{3x(4.75x10.69 \times 75.18x10^3 + 0.1925 \times 12.69 \times 1.1076x10^5 + 0.33 \; x1.1076x10^5)}{100000}$$

= **124 BHP**

= **92.5 KW**

To Calculate the power required by the Blower:

Temperature of inlet air = 30 ºC

Humidity = 0.002 kg/kg

Total Quantity of air handled = 32734.68 Kg/hr

$$\text{Volume of the inlet air} = \frac{32734.68 \times 22.4 \times 303}{29 \times 298}$$

$= 25.70 \times 10^3 \; m^3/hr$

From equation (6 – 34a) Perry

Power = 2.72×10^{-5} Qp

Q - fan volume, m^3/hr

p – fan operating pressure, cm water column

p = 20 cm water column

Power = $2.72 \times 10^{-5} \times 25.70 \times 10^3 \times 20$

= **13.98 KW**

To Calculate the power required by the Exhaust fan:

Temperature of the outlet air = 87 ºC

Humidity of the outlet air = 0.065 kg/kg

Total Quantity of air handled = 34862.43 Kg/hr

$$\text{Volume of the inlet air} = \frac{34862.43 \times 22.4 \times 363}{29 \times 298}$$

$$= 32.80 \times 10^3\ m^3/hr$$

$$\text{Power} = 2.72 \times 10^{-5} \times 32.80 \times 10^3 \times 20$$

= **17.84 KW**

To Find the diameter of the feed pipe:

Feed Rate = (68205 +1722.38) lb/hr

Volumetric feed rate = 743 ft^3/hr = 21 m^3/hr

Assume the velocity of the feed to be 100 m/hr

Cross sectional area of the feed pipe = (21/100) = 0.21 m^2

Diameter of the feed pipe = 0.52 m = **21”**

To Find the Diameter of the air inlet and outlet pipe:

INLET:

Temperature of air = 156 ºC

Humidity = 0.002

Volumeof air = 7.14 m^3/s

Assume the Velocity of the air entering = 20 m/s

Cross sectional area of the inlet air pipe = (7.14/20) = 0.357 m^2

Diameter of the feed pipe = 0.674 m = 26.5”

With Corrosion allowance Diameter = **28”**

OUTLET:

Temperature of air = 122 ºC

Humidity = 0.065

Volumeof air = 9.11 m^3/s

Assume the Velocity of the air entering = 20 m/s

Cross sectional area of the inlet air pipe = (9.1/20) = 0.455 m^2

Diameter of the feed pipe = 0.761 m = 29.96”

With Corrosion allowance Diameter = **32”**

DRIER DETAILS

Length of the drier = 27 m

Inner Diameter of the Drier = 3.258 m

Outer Diameter of the Drier = 3.278 m

Thickness of the shell = 10mm

Thickness of insulation = 80 mm

Power Required to Drive the Drier = 92.5 KW

Power of the Blower = 13.98 KW

Power of the Exhaust Fan = 17.84 KW

Diameter of the Feed pipe = 21”

Diameter of the air inlet pipe = 28”

Diameter of the air Outlet pipe = 32”

Rotation of the Drier = 3 rpm

CONDENSER

The material chosen to build the condenser is carbon steel.

SHELL SIDE:

Number of Shells – 1

Number of passes -2

Fluid – (Acetic Acid + water) Vapors

Working Pressure – 0.1013 N/mm^2

Design Pressure – 0.152 N/mm^2

Temperature of inlet – 101.4 ºC

Temperature of outlet – 101.4 ºC

Segmental Baffles With 25% with tie rod and Spacers.

Permissible stress – 95 N/mm^2

To calculate the Shell Thickness:

$$t_s = \frac{p\,D}{2\,f\,J + p}$$

p – Design pressure, N/mm^2

D – Diameter of the Condenser, mm

F – permissible stress, N/mm^2

J – 0.85

$$t_s = \frac{0.152 \times 787}{2 \times 0.85 \times 95 + 0.152}$$

$$= 0.74 \text{ mm}$$

But the minimum thickness is given as 8 mm for the shell

Therefore including corrosion allowance the thickness can be taken as **10 mm**

Therefore the outer Diameter = $D_i + 2 \times 10$

$= 787 + 2 \times 10$

$=$ **807 mm**

Head of the Shell:

The Crown Radius is taken as the diameter of the shell = 787 mm

Knuckle radius = 0.1 x Crown Radius = 78.7 mm

Shell flange – female facing

Gasket – flat metal jacketed asbestos filled

Bolts – 5% Cr Mo Steel

To Calculate Head Thickness:

$$t_h = \frac{pR_c W}{2 f J}$$

Where W = 0.25 $(3+\sqrt{(R_c/R_i)}) = 1.55$

R_c = Crown Radius,mm

J = 1

$$t_h = \frac{0.152 \times 787 \times 1.55}{2 \times 9.5 \times 1}$$

$$= 1.488 \text{ mm}$$

But the minimum thickness should be 8 mm with corrosion allowance

Therefore the thickness of head t_h = **8mm**

Nozzle:

Assume the velocity of the streams – 10m/s

Inlet dia -75mm

Vent -25 mm

Drain – 25 mm

Opening relief valve -50mm

Nozzle Thickness:

$$t_s = \frac{p D}{2 f J - p}$$

$$t_s = \frac{0.152 \times 75}{2 \times 95 - 0.152} = 0.060 \text{ mm}$$

But the minimum is **4mm**

Therefore the nozzle thickness = 4mm

Transverse Baffles:

Baffle Spacing = 787 mm

Thickness of baffles – 6mm

Number of tie rods – 10

Diameter of the Tie Rod – 10mm

Flange Thickness:

Shell OD – 807mm

Gasket width can be determined using

$(d_o/ d_i) = \sqrt{[(y-pm)/(y-p(m+1))]}$

Where d_o – gasket outer dia, mm

d_i - gasket inner dia, mm

y – min design yield stress = 126.6 N/mm^2

m – gasket factor = 4.5

The Ratio is found to be – 1.0006

Let the inner radius of the gasket be 797 mm

Gasket width is 12mm

d_o = 821 mm

d_i = 797 mm

Mean Gasket Diameter = G = (821+797)/2 = 809 mm

Basic Gasket seating width = b_o = N/2 = 6mm

Effective Gasket seating width = b = b_o = 6mm

Bolt Load due to gasket reaction is given by

Under atm condition:-

$W_{m1} = \Pi\ b\ G\ y$

Y = 53.4 N/mm^2 (seating Stress)

$W_{m1} = \Pi \times 6 \times 809 \times 53.4$ = 814.31 KN

Under operating condition:-

W_{m2} = Π 2b G mp +(Π/4) G^2 p

= 20861 + 78132

= 98.99 KN

$$\text{Factor k} = \left[0.3 + \frac{1.5\ W_m\ h_G}{H\ G} \right]^{-1}$$

W_m = total bolt load – 814.31

H – 78.132

h_G = (B-G)/2

= (890 – 809)/2 = 40.5

$$\text{Factor k} = \left[0.3 + \frac{1.5 \times 814.31 \times 40.5}{78.132 \times 809} \right]^{-1}$$

= 0.9236

Flange Thickness = 809 √(.152/(95 x 0.9236)) = 34 mm

With Corrosion allowance of 2 mm

Flange thickness = **36 mm**

Tube Side:

Tube & Tube Sheet Material – Stainless steel

Number of tubes -716

Outside diameter – 1.905 cm

Length – 3.66 m

Pitch – 25.4 mm

Fluid- Treated water

Working pressure – 5000 KPa – 5 N/mm^2

Design Pressure – 7.5 N/mm^2

Inlet Temperature – 25 ºC

Outlet Temperature – 40 ºC

Permissible Stress – 100 N/mm^2

Tube Sheet:

The tube sheet is held between shell flange and channel.

Based on the design pressure of 7.5 N/mm^2 and mean gasket diameter as 375 mm

Tube Sheet Thickness

$t_{ts} = FG\sqrt{(0.25p/f)}$

$F = 1.25$

$t_{ts} = 1.25 \times 375\sqrt{(0.25 \times 7.5/100)}$

$= 64$ mm

The Tube Sheet thickness can be taken as **70mm**

Channel Cover:

Material – Carbon Steel

Permissible stress – 95 N/mm^2

$t_c = G_c\sqrt{(KP/f)}$

for ring type K = 0.3

$t_c = 375\sqrt{(0.3 \times 7.5/95)}$

$= 58$mm

The Channel Cover thickness is taken as = **60mm**

Flange Joint (Tube Sheet And Chennel):

G = 375 mm

Ring Gasket width = 24mm

$b_o = W/8 = 2.75\ mm^2$

$y_a = 126.6\ N/mm^2$

m = 5.5 , b =2.75

$W_{m1} = \Pi \times 375 \times 2.75 \times 126.6 = 410$ KN

$W_{m2} = \Pi\ 2b\ G\ mp + (\Pi/4)\ G^2\ p$

$= (267.28 + 828.35) \times 10^3$

$= 1095.63$ KN

CSA of the bolt:

A_{m1} = 410000/140.6 = 29.26 cm^2

A_{m2} = 1095.63 /140.6 = 77.92 cm^2

Number of Bolts = 375/(10 x 2.5) = **15**

Diameter of the bolt = √(77.92 x 4/(16 x Π)) = **2.5 cm**

M 48 bolts are used: Pitch diameter = 44.68 mm

Minor diameter = 41.80 mm

Actual Bolt area = **294cm²**

Minimum pitch Circle diameter = 375 + 22 + 2 x 48 = **493 mm**

B = 525

Flange Thickness:

$$\text{factor } k = \left[0.36 + \frac{1.5 \times 410 \times 75}{267.28 \times 75}\right]^{-1}$$

= 0.384

Flange Thickness = 375 √(7.5/(95 x 0.384)) = 170 mm

With Corrosion allowance

Flange thickness = **180 mm**

Saddle Support:

Material: Carbon Steel

Shell diameter = 787mm

R = D/2

l = 3660mm

Torispherical Head: Crown radius = 787, knuckle radius = 78.7mm

Total Head Depth = 152mm = H

Shell Thickness = Head Thickness = 8mm

f_t = 95 MN/m^2

Weight of the vessel = 574 Kg

Weight of Tube = 62.13 Kg

Weight of head = 648.2 Kg

Weight of Liquid = 484.96Kg

Weight of the shell and its contents = 1800 Kg

W = 1800 x 9.81 = 18000 N

Distance of saddle center line from shell end = A = 0.4 x D/2

= 158 mm

Q = **9000 N**

Longitudinal Bending Moment:

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

$M_1 =$ **6.1057×10^5 Nmm**

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

= **7.145×10^6 Nmm**

Stresses in shell at the saddle:

$f_1 = M_1/(\pi R^2 t) =$ **0.284 N/mm**

$f_2 =$ **0.284 N/mm**

$f_3 = M_2/(\pi R^2 t) =$ **19.99 N/mm**

$f_p = p\, d/(4(t_s - c)$

= **4.9 N/mm^2**

$f_p + f_1 =$ **5.184 N/mm^2**

$f_p - f_2 =$ **4.616 N/mm^2**

$f_p + f_3 =$ **24.89 N/mm^2**

All stresses are within allowable limits. Hence, the given parameters can be considered for design.