DESIGN CONSIDERATION OF DRIERS

DESIGN OF DRYER

Design of a rotary dryer only on the basis of fundamental principle is very difficult. Few of correlations that are available for design may not prove to be satisfactory for many systems. The design of a rotary dryer is better done by using pilot plant test data and the full scale operating data of dryer of similar type if available, together with the available design equations. A fairly large number of variables are involved such as solid to be dried per hour, the inlet and exit moisture contents of the solid, the critical and equilibrium moisture contents, temperature and humidity of the drying gas. The design procedure based on the basic principles and available correlations is discussed below. In this case we assume that the solid has only unbound moisture and as shown in fig 2.7 in stage II the solid is at the wet bulb temperature of the gas.

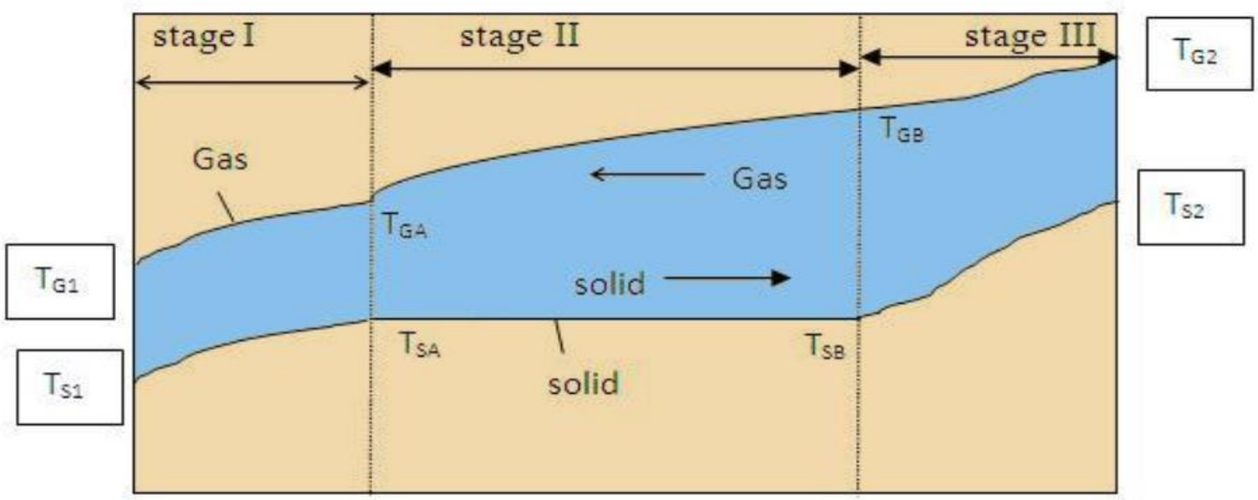

Figure 2.7: Temperature profile for solid and gas in a counter current rotary dryer

1. Heat losses from dryer surfaces are neglected.
2. Once the capacity of the dryer is known, the drying gas flow rate, its temperature and humidity are decided considering a number of factors. And the following moisture & enthalpy balances need to be satisfied.

$$Gs\ (Y_1 - Y_2) = Ms\ (X_1 - X_2)$$

$$Gs\ (Hg_2 - Hg_1) = Ms\ (H_{S2} - Hs_1)$$

Here, G_s = flow rate of air (dry basis, kg/h), Ms = flow rate of solid (kg/h, dry basis), H_s = humidity of air (kg/H_2O/kg dry air)

3. The gas and solid temperatures at the stage boundaries are obtained by moisture and energy (enthalpy) balances. The number of heat transfer unit for each zone is calculated. for the stage II. The number of heat transfer units is given by

$$(N_{tG})_{h,II} \times \Delta T_m = (T_{GB} - T_{GA})$$

4. The total length of dryer is given by

   $L = (L_T)_1\ (N_{tG})_1 + (L_T)_{II}\ (N_{tG})_{II} + (L_T)_{III}\ (N_{tG})_{III}$

5. The shell diameter is calculated from the dry gas flow rate (from step I) and suitable gas flow velocity or gas mass flow rate

   Some useful correlations for the design of a rotary dryer are given below.

   Volumetric gas-solid heat transfer coefficient.

   $\bar{U}_a = (W/m^3.K) = 237\ (G')^{0.67}/d$

   Here, G' = gas mass flow rate ($kg/m^2.h$) and d, dryer diameter

   Length of transfer unit $L_T = G'_{CH} / \bar{U}_a$

$$L_T = 0.0063\ _{CH}\ .\ d.\ G_S^{0.84}$$

   Here, c_H = average humid heat, and d = dryer diameter

Solid retention time:

$$\theta = \frac{0.23\ L}{S\ N^{0.9}\ d} \pm 1.97\ \frac{B\ L\ G'}{F}$$ (+ve sign is for counter flow; –ve sign is for parallel flow of the gas and solid)

Where,

θ = retention time (min); L = dryer length (m)

S = slope of the dryer (m/m); N = speed (rpm)

G' = gas mass flow rate ($Kg/m^2.h$)

F = feed rate (Kg/m^2. h) dry basis

$B = 5\ (d_p)^{-0.5}$

d_p = weight average particle diameter (micron)

d = dryer diameter (m)

Example 2.1: Size of the rotary dryer can be estimated for the following case. A moist non hygroscopic granular solid at 26^0C is to be dried from 20% initial moisture to 0.3% final moisture in a rotary dryer at a rate of 1500 kg/h. The hot air enters the dryer at 135^0C with a humidity of 0.015. With condition that the temperature of the solid leaving the dryer must not exceed 110^0C and the air velocity must not exceed 1.5 m/s in order to avoid dust carry over. C_{ps} = 0.85 kJ/kg.K. Recommend the diameter, length and other parameters of the dryer.

Solution:

Basis of calculation is 1 hr operation

Solid contains 20% initial moisture

Mass of dry solid $= M_S = 1500\,(1\text{-}0.2) = 1200$ kg/hr

Moisture in the wet solid $= X_1 = 20/80 = 0.25$

Moisture in the dry solid $= X_2 = 0.3/99.7 = 0.00301$

Water evaporated, $m_{S,\,evaporated} = M_S\,(X_1 - X_2)$

$$= 1200\,(0.25 - 0.00301) = 296.4 \text{ Kg}$$

Given data:

$T_{S1} = 26^0C$; $T_{G2} = 135^0C$; $Y_2 = 0.015$

Let us assume that the exit temperature of the gas is $T_{G1} = 60^oC$ and for solid $T_{S2} = 100^oC$

Now enthalpy of different streams (suppose ref temp = 0^oC)

$H_{S1} = [C_{PS} + (4.187)\,X_1]\,[T_{S1} - 0]$

$= [0.85 + (4.187)\,0.25]\,[26 - 0] = 49.31$ KJ/kg dry air

$H_{S2} = [C_{PS} + (4.187)\,X_1]\,[T_{S1} - 0]$

$= [0.85 + (4.187)\,0.0.00301]\,[100 - 0] = 86.2$ KJ/kg dry solid

$H_{g2} = [1.005 + (1.88)\,0.015]\,[135 - 0] + (0.015)\,(2500) = 177$ KJ/kg

$H_{g1} = [1.005 + (1.88)\,Y_1]\,[60 - 0] + Y_1\,(2500) = 60.3 + 2613\,Y_1$

Overall mass balance

$G_S\,(Y_1 - Y_2) = M_S\,(X_1 - X_2) \Longrightarrow G_S\,(Y_1 - 0.015) = 296.4$

$$\Longrightarrow \quad G_S = 296.4/(Y_1 - 0.015)$$

$$M_S\,[H_{S2} - H_{S1}] = G_S\,[H_{g2} - H_{g1}]$$

$$\Longrightarrow \quad 1200\,[86.2 - 49.31] = 296.4/(Y_1 - 0.015) \times (177 - 60.3 - 2613Y_1)$$

$$Y_1 = 0.04306 \text{ and } G_s = 296.4/(Y_1 - 0.015) = 10560 \text{ Kg/h}$$

Shell Diameter

Volume of humid inlet gas (135^0C and $Y_2 = 0.015$)

$$V_{H2} = 1.183 \text{ m}^3\text{/Kg dry air}$$

Volume of humid exit gas (60^0C and $Y_1 = 0.04306$)

$$V_{H1} = 1.008 \text{ m}^3\text{/Kg dry air}$$

The max. volumetric gas flow rate $= G_s.V_{H2}$

$$= 10560 \times 1.183 = 12490 \text{ m}^3\text{/h}$$

The working velocity i.e. superficial velocity $= 1.5 - 0.2 \times 1.5$

$$= 1.2 \text{ m/s}$$

$$\therefore \; \pi / 4 \times d^2 \, (1.2) = d = 1.98 \text{ m, } \textit{say 2.0 m}$$

Heat Transfer Unit

Dryer is divided into three zones and therefore, the stage wise calculation of temperature and humidity of the stream can be obtained by material and energy balance.

Stage III

Very less water left for vaporization in stage III. Consider solid is at T_{SB}, the wet bulb temperature of the air at location between III & II.

$$\text{assume } T_{SB} = T_{SA} = 41^0\text{C}$$

Enthalpy of solid at the inlet to stage III

$$H_{SB} = [0.85 + (0.00301)(4.187)](41-0)$$

$$= 35.37 \text{ KJ/kg dry solid}$$

Humid heat of gas entering stage III

$$C_{HB} = [1.005 + (1.88)(0.015)]$$

$$= 1.003 \text{ KJ/kg.K}$$

Heat balance over stage III

$$M_S \, [H_{S2} - H_{SB}] = G_S \, (C_{HB})_{III} \, (135 - T_{GB})$$

$$T_{GB} = 129^0\text{C}$$

Adiabatic saturation temperature of air entering stage II (129^0C & humidity of 0.015) is 41.3^0C.

At the boundary B, $\Delta T_B = 129 - 41 = 88^0\text{C}$

At end 2, $\Delta T_2 = 135 - 100 = 35^0\text{C}$

$$LMTD_{III} = (\Delta T)_m = 88\text{-}35/\ln(88/35) = 57.5^0C$$

$$(N_{tG})_{III} = T_2 - T_{GB}/(\Delta T)_m = 135 - 129/57.5 = 0.104$$

Stage II

$\Rightarrow$Use heat balance equation over stage II to calculate the value of T_{GA}. The calculated T_{GA} value can be use to estimate the number of transfer units.

Since $Y_B = 0.015$

$H_{GB} = [1.005 + 1.88\ Y_B]\ (129\text{-}0) + 2500\ (Y_B) = 170.8$ KJ/Kg

$H_{AS} = [0.85 + C_{PS}\ X_1]\ (T_{SA} - 0) \quad = [0.85 + (4.187)\ (0.25)]\ (41)$

$= 77.77$ KJ/(Kg dry solid)

Enthalpy balance:

$$M_S\ (H_{SB} - H_{SA}) = G_S\ (H_{GB} - H_{GA})$$

$$1200\ (35.37 - 77.77) = 10560\ (170.8 - H_{GA})$$

$$\therefore\ H_{GA} = 175.6 \text{ KJ/Kg}$$

Once H_{GA} value is known then T_{GA} can be calculated using the following equation

$$H_{GA} = 175.6 = [1.005 + 0.04306\ (1.88)]\ [T_{GA} - 0] + 0.04306\ (2500)$$

$$\Rightarrow\ T_{GA} = 63^0C$$

At section A temp diff. $\Delta T_A = 63 - 41 = 22^0C$ and $\Delta T_B = 88^0C$

$(\Delta T)_M = (88\text{-}22)/\ln(88/22) = 47.6^0C$

Number of transfer unit $\quad = (N_{tG})_{II} = T_{GB} - T_{GA}/(\Delta T)_M$

$= (129 - 63)/47.6 = 1.386$

To validate the assumed value of exit gas temperature i.e. $T_{G1} = 60^0C$, first do an energy balance over stage I.

$$G_S\ (H_{g2} - H_{g1}) = M_S\ (H_{S2} - H_{S2})$$

$$10560\ (175.6 - H_{g1}) = 1200\ (77.77 - 49.31)$$

$$\Rightarrow\ H_{g1} = T_{G1} = 59.6^0C$$

Stage I

$(\Delta T)_1 = 60\text{-}26 = 34^0C$

$(\Delta T)_A = 22^0C$

$(\Delta T)_M = 34\text{-}22/\ln(34/22) = 27.5$

Number of transfer unit, $N_{tG} = 0.104 + 1.386 + 0.109 = 1.53$

Length of Transfer Unit:

Avg. mass flow rate = [10560 (1.015) + 10560 (1.04306)]/2

= 10867 Kg/h

The gas mass flow rate, G' = $(10867/3600)/\ \pi / 4 \times (2)^2$

= 0.961 Kg/m^2.S

Volumetric heat transfer coeff. = $\overline{U_a} = (237\ (G')^{0.67})/d$

$$\therefore\ \overline{U_a} = (237 \times (0.961)^{0.67})/2 = 115\ W/m^3.K$$

Humid heat at the ends

$$C_{H2} = 1.005 + 1.88\ (0.015) = 1.033$$

$$C_{H1} = 1.005 + 1.88\ (0.04306) = 1.083$$

Avg. humid heat,

$$C_H = (1.033 + 1.083)/2 = 1.058\ KJ/Kg.\ K$$

Length of transfer unit, $L_T = G'.\ C_H/\overline{U_a} = (0.961 \times 1058)/115 = 8.84\ m$

Length of dryer, L = $N_{tG}.\ L_T$

= $1.56 \times 8.84 = 13.8$ m

d = 2 m and L = 14 m

SOLVED PROBLEMS

Example 2.2: (Process design**)**

A rotary drier using counter current flow is to be used to dry 25000 lb/hr of wet solid (PTA) containing 5 weight percent water to a water content of 0.10 weight per cent. The wet solid enters at 30°C (86^{0}F). Ambient air at 30°C (86^{0}F) will be heated to 156^{0}C (313^{0}F). Specific heat of solid is 0.2871. Estimate the length and diameter of the drier.

Feed to the drier:	**Condition of inlet air:**
Water content in the feed = 25000 × 0.05 = 1250 lb/Hr Dry solid in feed = 25000 – 1250 = 23,750 lb/Hr Water content in Product = 24 lb/Hr Water removed by the drier = 1226 lb/Hr Steam pressure = 150 psig	Ambient temp. of air (dry bulb) = 30^{0}C = 86^{0}F Wet bulb temp. (wet bulb)= 22^{0}C = 71^{0}F Heated Inlet temp. of air = 156^{0}C = 313^{0}F Humidity H_{mW} = 0.002 lb water/lb dry air.

$$\varphi = H_{mG} - H_{mW} = h_G(T_G - T_W)/(m\lambda_W P k_G)$$

H_{mG} Humidity of air at temperature T_G ^{0}F

H_{mW} Humidity of air at temperature T_W ^{0}F

T_G Temperature of inlet air ^{0}F; T_W Wet bulb temperature ^{0}F

M Molecular weight of air; λ_W Latent heat of vaporization at T_W ^{0}F

$h_G/(mPk_G) = 0.26$ for air at T_W and here m = 29.

First Trial: assume wet bulb temperature is 90^{0}C = 194°F

Hence at T_W = 194 ° F, H_{mW} = 0.046 ∴ $H_{mW} - H_{mG}$ = 0.046- 0.002 = 0.044; λ_w = 547.3

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

$\boxed{\varphi > H_{mW} - H_{mG}}$

$\boxed{\text{Since } \Phi > H_{mW} - H_{mG} \text{ therefore the temperature assumed is high}}$

Second Trial: Assume a wet bulb temperature of 180^0 F

$H_{mW} = 0.065$ $\therefore$ $H_{mW} - H_{mG} = 0.065 – 0.002 = 0.063$; $\lambda_w = 532$

$\varphi = (0.26\ (313\text{-}180)/532) = 0.063$

Therefore wet bulb temperature assumed is true i.e. $T_W =$ **180^0 F**

The temperature of the outlet air should be selected on the basis of an economic balance between dryer and the fuel costs. Empirically it is found that drier operates economically when total number of transfer units (NTU) is between 1.5 to 2. (Badger and Banchero, Pg 508)

$NTU = \ln(T_{G1} - T_W)/(T_{G2} - T_W)$

Take NTU = 1.5 = ln (313 -180)/(T_{G2}- 180) $\therefore$ **$T_{G2} = 209\ ^0F$**

Energy balance:

C_P (PTA) = 0.2871Btu/lb^{0}F; C_P (Water) = 1 Btu/lb^{0}F

Product discharge temperature = (313 + 209)/2 = 261^{0}F

Temperature of feed = 176^{0}F

Heat required to raise the product to discharged temp.

= 23705 x 0.2871(261-176) + 24 (261-176) = 5.8143 x 10^5Btu/Hr

Heat required to remove the water = 1226 [(180-176) + 0.45 (209-180) + 550]

= 6.952 x10^5 Btu/Hr

Total Heat = 1.27 x 10^6 Btu/Hr

Air Required:

S_H-Humid Heat of inlet air = 0.24 + 0.45 x 0.002 = 0.2409

Use average humid heat = 0.242

G_G' S x Humid heat of air x Temperature= Total Heat, here S = cross sectional area, sq ft

$G_G{}^{'} . S \times (0.242) \times (313\text{-}209) = 1.27 \times 10^6$

$G_G{}^{'} . S = 50723.27$ **lb/Hr**

$$\text{Humid heat of outlet air } = \frac{1226}{50723.27} + 0.002 = 0.02617 \text{ lb water/lb dry air}$$

Humid heat $= 0.24 + 0.45 \times 0.02617 = 0.2517$ and $S_{Havg} = (0.2409 + 0.2517)/2 = 0.2463$

Therefore the average humidity taken above is valid

Mean temperature difference across the rotary drier can be calculated by using following formulae

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= 23705 x 0.2871(180-176) + 1250 (180-176) = $\mathbf{3.2274 \times 10^4}$ **Btu/hr**

Change in air temp. is = $[(3.2274 \times 10^4)/(1.27 \times 10^6)] \times (313 - 176) = 2.67\ ^0F$

Air temperature at the end of preheat = $209 + 2.67 = 212\ ^0F$

$$\Delta T)_P = \frac{(209-158)-(212-180)}{\ln\left[\dfrac{209-158}{212-180}\right]} = 40.76°F$$

Heating period:

$q_s = 23705 \times 0.2871\ (261\text{-}180) + 24\ (261\text{-}180) = 5.542 \text{ x } 10^5$ Btu/hr

$$\text{Change in temperature} = \frac{5.542 \text{ x } 10^5}{1.27 \text{ x } 10^6} \text{ x } (313\text{-}209) = 45.38°F$$

Air temperature at the start of heating = $313 - 45.38 = 267\ ^0F$

$$(\Delta T)_s = \frac{(267-180)-(313-261)}{\ln\left[\dfrac{267-180}{313-261}\right]} = 68°\text{F}$$

Evaporating period:

q_p= 1.27 x 10^6 – 5.542 x 10^5 – 3.2274 x 10^4 = **6.83 x 10^5 Btu/hr**

$$(\Delta T)_v = \frac{(267-180)-(212-180)}{\ln\left[\dfrac{267-180}{212-180}\right]} = 55°\text{F}$$

The mean temperature difference given as

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

$$\frac{1}{(\Delta T)_M} = \frac{1}{2.49 \text{ x } 10^6}\left[\frac{8.52 \text{ x } 10^4}{45.29} + \frac{1.499 \text{ x } 10^6}{54.54} + \frac{5.658 \text{ x } 10^5}{54.7}\right] = 0.0168$$

$(\Delta T)_M$ = **60^0F**

NTU Check:

$$\text{NTU} = \frac{(T_1 - T_2)}{(\Delta T)_m} = \frac{(313\text{-}209)}{60} = 1.73$$

According to the condition NTU should be between 1.5 to 2. Therefore the above mean temperature value can be accepted.

TRIAL 1: To Calculate the Diameter of the Drier

Air entering the drier is 50723.27 lbs/h. But for designing purpose air is taken in excess so that the loss of heat from the drier is compensated.

Air entering the drier can be taken as ~ 51000 lb/hr.

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

$$G_G\,S = 51000\left[1{+}0.0165 \text{ x } \frac{50723.27}{51000}\right] = 51836.93 \text{ lb/Hr}$$

$$S = \frac{51836.93}{1000} = 51.837\ ft^2$$

$$D = (4*51.837/\pi)^{0.5} = 8.07\ ft = 2.46\ m$$

Similarly length of the dryer can be calculated by using following equation

$$Q = U_a \times S \times Z \times (\Delta T)_M$$

Where,

Q = Total heat, Btu/Hr

Z = Length of drier, ft

S = cross sectional area, ft^2

Before that we need to calculate the overall heat transfer coefficient from:

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

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

U_a = Overall heat transfer coefficient (volumetric), Btu/Hr ft^3 ^0F

D = Diameter of the drier in ft.

$$Ua = \frac{15(1000)^{0.16}}{8.07} = 5.57\ Btu/hr\ \ ft^3\ \ {}^\circ F$$

∴ Length of the drier

$$Z = \frac{1.27 \text{ x } 10^6}{5.57 \text{ x } 51.837 \text{ x } 60} = 73.30\ ft = 22.34\ m$$

Z/D ratio check:

$$Z/D = \frac{22.34}{2.46} = 9$$

Which checks the condition that the Z/D ratio is between 3- 10. Therefore the above diameter and length can be accepted.

To calculate the speed of the rotation of the drier;

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

Revolution per min = peripheral speed / diameter

RPM = 30/8.07 = 3.7

The revolution of the drier varies between 2 and 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 feet

Number of flights = 3 x 8.07 = 24.21, say 24

Radial height of the flight:

The radial height of the flight taken as $1/8^{th}$ of the diameter of the drier

The radial height of the flight = (1/8) x 8.07 = 12.18 inches, say 12.25”

DRIER DETAILS:

Drier Type: Counter Current Rotary Drier

Diameter of the drier = 8.07 ft = 2.46 m

Length of the Drier = 73.3 ft = 2.34 m

RPM of the drier = 3.7 rpm

Number of Flights = 24

Radial height of the flights = 12.25 inches

Temperature of the inlet air = 156°C = 313^{0}F

Temperature of the inlet wet solid = 90°C = 194^{0}F

Mean temperature Difference = 60^{0}F

Air mass flow rate = 51000 lb/hr

Moisture removed by the drier = 1226 lb/hr

The volumetric heat transfer coefficient of drier = 5.57 Btu/Hr ft^{3}°F

SOLVED PROBLEM

Example 2.3: (*Mechanical Design*) The drier has a uniform temperature of around 150 °C at any point of time (working pressure in the drier is 0.1013 N/mm^2). So the material used for the construction of the dryer should withstand the high (operating) temperature. Since mild steel withstand high temperature of 200 ^{0}C. The material used to construct the dryer is mild steel and permissible pressure of material used is 124 N/mm^2.

Length of drier = 22.34 m; Inner diameter of the drier = 2.46 m

$\therefore$ *Design pressure* = *1.5 x W P* = *1.5 x 0.1013* $\Rightarrow$ *0.152* $\mathrm{N}/\,mm^2$

Thickness of the drier shell:

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

P → Design pressure, $\mathrm{N}/\,mm^2$

D → Diameter of the drier, mm^2

F → Permissible stress, $\mathrm{N}/\,mm^2$

J → 0.85

$$t_s = \frac{0.152 \times 2460}{2 \times 0.85 \times 124 + 0.152} = 1.77 \text{ mm}$$

For the shell minimum thickness is given as 8 mm. Consider corrosion allowance of 2 mm therefore, including the C.A. the thickness can be taken as 10 mm.

Therefore the outer Diameter $= D_i + 2 \text{ x } 10 = 2460 + 2 \times 10$

$$D_0 = 2480 \text{ mm} = \mathbf{2.480 \text{ m}}$$

The thickness of the insulation:

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

Density of asbestos $= 577\ Kg/\,\mathrm{m}^3$

Thermal conductivity of asbestos $= 681.4 \text{ x } 10^{-3}\ W/\,\mathrm{m}^2 K$

Thermal conductivity of mild steel $= 147.6\ W/\,\mathrm{m}^2 K$

Convective heat transfer coefficient $= 56.78\ W/\,\mathrm{m}^2 K$

From heat balance,

Heat loss from the drier = 97.006 *KW*

Inner diameter of the drier shell, D_1 = 2.46 m

Outer diameter of the drier shell, D_2 = 2.48 m and t_1 = 10 mm

Let 'y' be the thickness of insulation.

$$D_3 = D_2 + 2y$$

$$T_1 = 122^oC \text{ and } T_2 = 76^oC$$

We have from continuity equation,

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

$$\begin{aligned} A_1 &= \pi\,(D_1 + D_2) \times L/2 \\ &= \pi\,(2.46 + 2.48) \times 22.34/2 \\ &= 174.24 \text{ m}^2 \end{aligned}$$

$$\begin{aligned} A_2 &= \pi\,(D_2 + D_3) \times L/2 \\ &= \pi\,(2.48 + 2.48 + 2y) \times 22.34/2 \\ &= (174.42 + 70\,.26y) \text{ m}^2 \end{aligned}$$

$$\begin{aligned} A_3 &= \pi \times D_3 \times L \\ &= \pi\,(2.48 + 2y) \times 22.34 \\ &= (174.04 + 140.36\,y) \text{ m}^2 \end{aligned}$$

$$97.006 \times 10^3 = \frac{122.0 - 76.0}{\frac{10 \times 10^{-3}}{147.6 \times 174.24} + \frac{y}{681.4 \times 10^{-3}\,(174 + 70.24\,y)} + \frac{1}{56.78\,(174.84 + 140.36\,y)}}$$

$$97.006 \times 10^3 = \frac{46}{\frac{10 \times 10^{-3}}{25717.82} + \frac{y}{(118.6 + 47.87y)} + \frac{1}{(9927.4\, +\, 7969.6\,y)}}$$

After solving the final equation obtained as follows

$y^2 + 1.42\,y - 0.0654 = 0 \qquad \therefore\ y = 0.04$ m

Therefore the thickness of the insulation should be **40 *mm***

To find the power to drive the Driver; Use equation (20- 44) from 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 → live load, Ib W → total rotating load, Ib

D → riding ring diameter, ft (d + 2)

To calculate the live load and the rotating load;

Density of mild steel $= 480\ Ibs/\ ft^3$

We have,

D_2 = Outer diameter of the drier shell

D_1 = Inner diameter of the drier shell

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

$$= \frac{\pi \times 73.30 \times (8.13^2 - 8.07^2)}{4}$$

$$= 56\ ft^3$$

Weight of the drier = Volume of shell material × density

= 56 × 480

= 26859.71 lbs

Assume Hold up = 0.1

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

$$= \frac{\pi\ (8.07)^2}{4} \times 73.30 \times 0.1$$

$$= 374.92\ ft^3$$

Weight of material in drier at any time, w = Volume x Density

$= 628.41 \times 94.07$

$= 35.268 \times 10^3$ lbs

Volume of the insulating materials $= \dfrac{\pi L (D_3^2 - D_2^2)}{4}$

$= \dfrac{\pi \times 73.30 \times (8.26^2 - 8.13^2)}{4}$

$= 122.21\ ft^3$

Weight of the insulating material = Volume x Density

$= 122.21 \times 36$

$= 4399.62$ lbs

Total weight,

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

$= 26859.71 + 35.268 \times 10^3 + 4399.62 = \mathbf{6.652 \times 10^4}$ **lbs**

W = weight of the material

$w = 35.268 \times 10^3$ lbs

Riding ring diameter, $D = d + 2$

$= 8.07 + 2 = 10.07$ ft

The rpm of the drier, r = $3\ rpm$

$$BHP = \frac{3 \times (4.75 \times 8.07 \times 35.268 \times 10^3 + 0.1925 \times 10.07 \times 6.652 \times 10^4 + 0.33 \times 6.652 \times 10^4)}{100000}$$

= 45.08 BHP

= 33.62 KW

To calculate the power required by the Blower:

Temperature of the inlet air = 30 °C

Humidity of inlet air $= 0.002\ Kg\ of\ H_2O/Kg\ of\ air$

Total quantity of air handled = 23512.83 Kg/hr

$$\text{Volume of the inlet air} = \frac{23512.83 \times 22.4 \times 303}{29 \times 298}$$

$$= 18.466 \times 10^3\ m^3/hr$$

Use equation (6-34a) from Perry,

$\text{Power} = 2.72 \text{ x } 10^{-5}\ Q\ p$

Where

Q → Fan volume, m^3/hr

p → Fan operating pressure, cm water column

p = 20 cm water column

Power = $2.72 \times 10^{-5} \times 18.466 \times 10^3 \times 20 = 10$ KW

To calculate the power required by the Exhaust fan:

Temperature of outlet air = $87\ ^oC$

Humidity of the outlet air $= 0.065\ Kg\ of\ H_2O/Kg\ of\ air$

Total quantity of air handled =

$$\text{Volume of the inlet air} = \frac{24628.7 \times 22.4 \times 363}{29 \times 298} = 42.64 \times 10^3\ m^3/hr$$

Power = $2.72 \times 10^{-5} \times 42.64 \times 10^3 \times 20$

= 23.19 KW

To find the diameter of the feed pipe:

Feed Rate = 25000 lb/hr

Volumetric feed rate $= 743\ ft^3/hr$

$= 21\ \mathrm{m}^3/\ hr$

Assume the velocity of the feed to be 100 $\mathrm{m}/\ hr$

Cross sectional area of the feed pipe = $(21/100)$

$= 0.21\ \mathrm{m}^2$

Diameter of the feed pipe = $0.52\ \mathrm{m} = \mathbf{21''}$

To find the diameter of the air inlet and outlet pipe:

INLET:

Temperature of air = 156 ^oC

Humidity of inlet air $= 0.002\ Kg\ of\ H_2O/Kg\ of\ air$

Volumetric flow rate of air $= 7.14\ \mathrm{m}^3/s$

Assume the velocity of the air entering to be 20 $\mathrm{m}/\ s$

Cross sectional area of the inlet air pipe = $(7.14/20)$

$= 0.357\ \mathrm{m}^2$

Diameter of the inlet pipe $= 0.674\ \mathrm{m}$

$= 26.5''$

With corrosion allowance diameter $= \mathbf{28''}$

OUTLET:

Temperature of air = 156 ^oC

Humidity of outlet air $= 0.065\ Kg\ of\ H_2O/Kg\ of\ air$

Volumetric flow rate of air $= 9.11\ \mathrm{m}^3/s$

Assume the velocity of the outlet air to be 20 $\mathrm{m}/\ s$

Cross sectional area of the outlet air pipe $= (9.11/20)$

$= 0.455\ m^2$

Diameter of the outlet pipe $= 0.761$ m

$= 29.96''$

With corrosion allowance diameter $= \mathbf{32''}$

DRIER DETAILS

Length of the Drier = 22.34 m

Inner diameter of the drier = 2.46 m

Outer diameter of the drier = 2.48 m

The thickness of the shell = 10 *mm*

The thickness of the insulation = 40 *mm*

Power required to drive the Drier = 33.62 *KW*

Power of the Blower = 10 *KW*

Power of the Exhaust fan = 23.19 *KW*

Diameter of the feed pipe $= 21''$

Diameter of the inlet pipe $= 28''$

Diameter of the outlet pipe $= 32''$

Rotation of the Drier = 3 rpm

Exercise problems:

<u>Example 1</u>: A double drum drier is to be designed for drying a paste with a capacity of 100 kg/hr. The drier is heated with indirect stream available at atmospheric pressure (100°C). The following data is available:

Temperature of the paste = 30°C.

Initial moisture content of paste = 60°C (wet basis).

Final moisture content of paste = 10°C (wet basis).

Heat transfer from the condensing steam to steam wall = 8500 W/m^2k.

Heat capacity of the paste material = 3400 J/kgk.

Thermal conductivity of the paste material = 0.8 W/mk.

The thickness of layer of material = 1.5mm.

The thickness of iron drum wall = 8mm.

Thermal conductivity of iron drum = W/mk.

Air is blown over the surface of material at a velocity of 1.5 m/sec.

Temperature of the air is 40°C.

Relative humidity of air is 40%.

Latent heat of vaporization of water at atmospheric pressure = 2240 kJ/kg.

Maximum temperature of the outer surface of the material being dried is 70°C.

Vapour pressure of water at 70°C = 350 mmHg.

Partial pressure of water vapour in air at 40°C and relative humidity 40% id = 22 mmHg.

Rate of flow of moisture being evaporated can be estimated by the correlation.

$$G = 1.14 \times 10^{-5} u^{0.8} (\Delta P)$$

u – velocity of air flow over the surface, m/sec.

*(**Ans**: U = 210W/m^2k; heating surface area A = 3.02 m^2; Actual surface area = 4.368m^2; Area of each drum =2.184m^2; drum diameter = 562mm)*

Example 2: Salicyclic acid crystals are to be dried in a pneumatic dryer at a rate of 200 kg/h of dry product. Initial moisture content of the crystals is 20% while the final moisture content should be 1%. Temperature of the crystals supplied to the drier is 10°C while the temperature of the crystals discharged from the dryer is 50°C.

Temperature of the air entering the heater = 10°C

Temperature of the air leaving the heater and entering the drier.

Relative humidity of air entering the heater is 70%.

Temperature of the air leaving the dryer = 60°C.

Specific heat of dry crystals = 1160 J/kgk.

Equivalent diameter of crystals = 0.001 m.

Density of material = 1480 kg/m^3.

Wet bulb temperature = 30°C.

Estimate the diameter, and length of the pneumatic dryer and the time needed to dry salicyclicstals.

Moisture content of air initially = 0.0065 kg/kg dry air.

Moisture content of air finally = 0.020 kg/kg dry air.

Enthalpy of air at the inlet of air heater = 33.5 kJ/kg.

Enthalpy of air at the outlet of air heater h_1 = 111 kJ/kg.

Thermal conductivity of air = 0.0285 w/mk.

Density of air = 1.03 kg/m^3.

Kinematic viscosity of air = 2420 kj/kg.

*(**Ans**: flow rate of dry air required, M= 2787 kg/hr; Heat transferred to air, Q =59991Watts; number of particles passing through the dryer per second, n = 71691.4 / sec; velocity of deposition of the particles, V= 3.814 m/sec; Diameter of the pneumatic dryer, D = 0.45m)*

Example 3: **(Sizing of a rotary dryer)** A fine granular solid to be dried at a rate of 600 kg/h from 22% to 0.2% moisture (all wet basis) in a countercurrent rotary dryer using hot air at 110°C of humidity 0.012 kg/(kg dry air). The moist solid fed to the dryer is at 25°C and the dried solid leaves at 80-oC. The moisture in the solid is unbound. In order to avoid dusting, the gas velocity should not exceed 1.7 m/s. The specific heat of the dry solid is 0.9 kJ/kg. °C, Suggest a dryer size.

*(**Ans**: Diameter of dryer, D =1.8 m; Length of dryer = 25 m)*