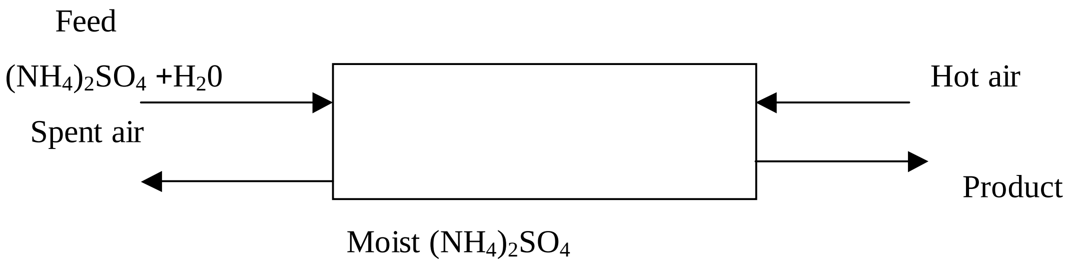
7. PROCESS EQUIPMENT DESIGN

1. ROTARY DRIER



Amount of water infeed = 212.5 kg/hr

Dry solid infeed = 10417 kg/hr

Water content in product = 105.25 kg/hr

Hence water dried in drier = 107.25 kg/hr

Inlet air temperature = 150º C

Outlet air temperature = 85º C

Inlet temperature of feed = 30º C

Discharge temperature = 80º C

Assuming wet bulb temperature of 80º C, 70% humidity of air.

The temperature of the air leaving the drier should be selected on the basis of an economic balance between drier cost and fuel cost. It has been that rotary driers are most economically operated when the total number of transfer units (NTU) range from 1.5 to 2.0. Assuming NTU = 1.5.

$$NTU = \ln \; [\,[t_{g1} - t_w] / [t_{g2} - t_w]\,]$$

$$1.5 = \ln \; [\{150 - 80\} / \{t_{g2} - 80\}]$$

$$t_{g2} = 95.62\ ^\circ C$$

Heat balance

C_p of $(NH_4)_2SO_4$ =1.63 kJ/kg ºC

C_p of water = 4.187 kJ/kg ºC

Temperature detail

	Feed	Air
Inlet	30 ºC	150 ºC
Outlet	80 ºC	85 ºC

Heat required to rise the feed to 45 ºC,

10417 x 1.63 x(45-30)+ 105.25 x (45-30) = 261326.4 kJ

Considering 55kg of water of evaporation,

Heat required to evaporate 55 kg of water = mλ

= 55 x2400

=132000kJ

Heat required to super heat the product to 80ºC,

10417 x 1.63 x (80-45) + 27 x1.9 (80-45) = 596085 kJ

total heat required to raise the product to discharge temperature,

Q_t = 261326.4+ 132000 + 596085

= 989412 kJ

LMTD across the dryer, Δt_m

$$\Delta t_m = \frac{[(150 - 30) - (85\text{-}80)]}{\text{Ln}[(150\text{-}30) \div (85\text{-}30)]}$$

The minimum velocity of air is set based on the particle size. Air flow rate of 100 lb/hr .ft^3 is sufficient for 420 microns. Hence this will be used in application. The minimum velocity is used since it gives the smallest possible size of drier.

Amount of air required:

$M = (Q_t / C_p \Delta t)$

$= 989412/ (150- 85)$

$= 15221.72$ kg / hr

The maximum amount of water present in this amount water is 60% i.e.

9133.03 kg / hr

An extra amount of 10% of this quantity to account the heat losses.

1.1 x 15221.72 = 16743.89 kg / hr.

If the velocity of air is 1000lb/ hr.ft = 4880 kg / hr. m^3

Area of drier = (16743.89/ 4880) 3.431 m^2

Diameter of the dryer = $\sqrt{}$ (3.431 x4/ л)

= 2.09 m.

$\boxed{\text{Diameter of dryer = 2.09 m.}}$

Length of transfer unit has been related to mass velocity and diameter by following relation,

$Ltu = 0.0064 \, x Cp \, x \, (G)^{0.84} \, x 2.04$

$= 7.36$ m.

Length of the drier = Ltu x NTU

= 7.36 x 1.5

=11.05 m.

Following dimensions for the drier are chosen.

L = 12m ; D= 2m

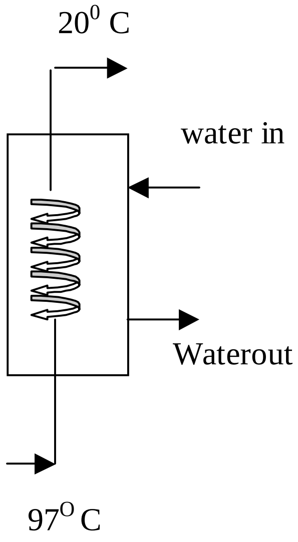
L/D = 12/2

= 6

L/D should range in between 4 – 10. Hence the design is safe.

2. SHELL AND TUBE HEAT EXCHANGER

1) **Temperature detail**:



	Cold fluid	Hot fluid 15^0c
In let	15^0c	97^0c
Outlet	35^0c	20^0

2. Heat load

Hot fluid: (Aq. Ammonia)

$$Q_4 = m \text{ x } Cp \text{ x } \Delta t$$

Where m = 1453 kg/nr = 3.98 kg/sec.

Cp = 2.57 KJ /kg

$\therefore Q_H = 3.98 \text{ x } 2.57 \ [97- 20\}$

$Q_H = 799.99$ k.j

$B_a + Q_H = Q_c = Q = 799KJ$

Where Q_H = heat load of hot fluid

Q_c = heat load of cold fluid.

Cold fluid : (Water)

$Q_c = M_C \times C_p \times . \Delta t$

Where M_C = to be determined.

C_p = Sp heat of water.

$\therefore 799 = M_C \times 4.184\ (35.15)$

M_C = 9.548 kg/ sec.

Mass of cold water required to remove the heat associated } = 9.548 kg/sec.

2) <u>LMTD Calculation ,Δt</u>

$T_1 = 97°c$ $t_2 = 35°c$

$T_2 = 20°c$ $t_1 = 15°c$

$\underline{\Delta}T_{lmtd} = (97° - 35) - (20 - 18) / \ln\ (\ (97° - 35) / (20 - 15)$

<u>Correction factor Fr</u>

$R = \frac{(T_1 - T_2)}{(t_2 - t_1)}$ $S = \frac{(t_2 - t_1)}{(T_1 - t_1)}$

$R = \frac{(97 - 20)}{(35 - 15)}$ $S = \frac{(35 - 15)}{(97 - 15)}$

$R = 3.85$ $S = 0.2439$

From perry $6^{th.}$cd. page 10.27

Considering 2-shell pass, 4 table pass i.e., 2.4 exchanger.

$\boxed{F_T = 0.8}$

Corrected LMTD = 22.63 x 0.8 = $\underline{18.10°}$

$\underline{\Delta t_{lmtd} = 18.10^0}$

3) Rounting of fluid:

Cleaner fluid is water -------- Shell side.

Unclean fluid is Aq.Ammonia .---Tube side

4) Heat Transfer Area:

Reference perry, page (10-44)U_d for water in shell side, inorganic solvent in tube side is ranging between(100-250) BTU/ (F.Ft^2.hr)

$$\text{Range is} = (567.83 - \frac{1419.57)\ J}{°C\ m^2.s}$$

$$\text{Total Heat Transfer Area (HTA)} = \frac{799 \times 10^3}{(600 \times 18.10)} = \underline{73.57\ m^2}$$

Choose l6 BWG tubes.

OD = (5/8)″ = 0.01587 m

ID = 0.495″= 0.01257 m

Length of tube = 16ft = 4.8768

Heat transfer area = 0.1636 ft^2 / ft^2 length

= 0.04986 m^2 / m.length.

$$\text{Heat transfer area of over tube} = 0.04968 \times 4.8768 = 0.2431 \frac{m^2}{\text{Tube}}$$

$$\therefore \text{No of tube} = \frac{73.57}{0.2431} = 302$$

Nearest tube count from perry, page 11-13 is 274. and corresponding shell diameter (inner) = 438 mm.

∴ Shell ID = 438mm.

$$\text{Corrected heat transfer area} = 274 \times 0.243 = 66.60 m^2$$

$$\text{corrected } U_d = \frac{799 \times 10^3}{66.60 \times 18.1} = \underline{662.0\ W/m^{2} \cdot {}^\circ k}$$

5) <u>Fluid velocity check</u>

a) Tube side: <u>(aq ammonia)</u>

Number of passes = 4

$$\text{Available flow area} = \frac{\pi}{4} \times d_i^{\,2} \times \frac{N_T}{N_P}$$

$$= \frac{\pi}{4} \times (0.01257)^2 \times \frac{274}{4}$$

$$a_t = 0.0085 m^2$$

$$\therefore \text{Velocity of fluid in the tube } V_t = \underline{M_t}$$

$f \times d_t$

$V_t = 3.98 \times \frac{1}{832} \times \frac{1}{0.0085}$

$\underline{V_t = 0.563\ m/s}$

b) Shell side: (water)

shell I D , D_s = 438mm

L_c baffle cut = 0.25 x Ds

L_s , baffle spacing = 0.5 D_s = 0.219m

$S_m = [(p^1 – Do)L_s] \times \frac{D_s}{p^1}$

S_m = Gross sectional area at centre of shell

Nb = <u>No</u> of baffles , L = length of tube

$p^1 = \frac{13}{16}$ inches square pitch = 0.0206m

$S_m = ((0.0206 – 0.0158) \times 0.219) \times \frac{0.438}{0.0206}$

$\underline{S_m = 0.02235 m^2}$

Shell side velocity, $V_s = \frac{M_s}{S_s \times S_m}$

$= \frac{9.548}{997.04 \times 0.02235}$

= 0.4284 m/s

<u>No. of baffles</u>

$N_b + 1 = \underline{\text{Total length of tube}}$

Baffle spacing

$= \underline{4}.8768/ 0.2$

$= 22.26 \ \underline{\sim} \ 23$

$\therefore \qquad \boxed{N_b = 22}$

6) Film transfer co-efficient

a) Tube side

Richardson & coulson (Page no: 270 – 297)

$\underline{A_t\ 55\ ^0\ c}$

$S = 832\ kg / m^3$

$Cp = 2.57\ KJ/ kg^0 k$

$M = 1.26\ mN_s / m^2 \qquad = 1.26 \times 10^{-3}\ \underline{N_S}$

$K = 0.219\ w/m^0\ k\ \ m^2$

$$N_{RC} = \frac{fV_tD}{M}$$

$$= \frac{832 \times 0.563 \times 0.01257}{1.26 \times 10^{-3}}$$

$$= 4673$$

$$N_{Pr} = \frac{MC_p}{K}$$

$$= \frac{2.57 \times 10^{-3} \times 1.26 \times 10^{-3}}{14.78}$$

$= 14.78$

From perry page (10-29) $j_H = 0.02$

$\therefore N_{Nh} = j_H \text{ x } N_{RC} \text{ x } (N_{Pr})^{1/3}$

$N_{Nh} = 0.02 \text{ x } 4673 \text{ x } 14.78^{1/3}$

$N_{Nh} = 229.35$

But, $\dfrac{h_i d_i}{k} = N_{Nh}$

$\therefore h_i = \dfrac{229.35 \text{ x } 0.219}{0.01257}$

$= 3995.83 \quad w/m^{2\,0}k$

$h_i = \underline{3995.83 \quad w/m^{2\,0}k}$

(b) Shell side

at 25^0c

$C = 995.045 \text{ kg} / m^3$

$Cp = 4.184 \text{ kj} / kg\ ^0 k$

$M = 0.95 \text{ x } 10^{-3}$ poise

$k = 1.42 \text{ w} / \text{m k}$

$N_{RC} = \dfrac{f \text{ x } V_s \text{ x } D}{M}$ (D = tube outside dia)

$= \dfrac{995.04 \text{ x } 0.4254 \text{ x } 0.01587}{0.095 \text{ x } 10^{-3}}$

$= \underline{7620}$

$N_{Pr} = \dfrac{M \text{ x } C_p}{K}$

$$= \frac{0.95 \text{ x } 10^{-3} \text{ x } 4.18 \text{ x } 10^{3}}{1.42}$$

$= 2.8$

From perry, page (10-29) , $j_H = 1 \text{ x } 10^{-3}$

$\therefore\ N_{Nh} = 1 \text{ x } 10^{-3} \text{ x } 7620 \text{ x } (2.8)^{1/3}$

$= 10.74$

but, $\frac{h_o d_o}{k} = N_{Nh}$

$\therefore\ h_o = \frac{10.74 \text{ x } 1.42}{0.01587} = 960 \text{ w / m}^2\text{k}$

$$\boxed{h_o = 960 \text{ w / m}^2\text{k}}$$

Overall heat transfer coefficient

$$\frac{1}{U_o} = \frac{1}{ho} + \frac{D_o}{D_i} \text{ x } \frac{1}{hi} + \frac{D_o \ln (D_o/D_i)}{2k_w} + \frac{1}{h_{dirt}}$$

For stainless steel $k_w = 45$

$h_{dirt} = 0.003$

1/Uo= 1/ 960 + (0.01587/ 0.012257 x 1/ 3995.53)

+ 0.01587 ln (0.01587/ 0.01257)/ (2x 45) + 1/ 0.003

$U_o = 243.096 \text{ w / m}^{2\ 0}\text{k}$

7) Pressure drop calculation :

a) Tube side

$$\Delta P_L = \frac{4fLV^2}{2_g D_i} \text{ x f x g}$$

but, $f = 0.079 \text{ x } R_c^{-0.25}$

$= 0.079 \text{ x } (4673)^{-0.25}$

$= 0.0095$

$$\Delta P_L = \frac{4 \text{ x } 0.0095 \text{ x } 4.8768 \text{ x } (0.563)^2 \text{ x } 832}{2 \text{ x } 0.01257}$$

$\Delta P_L = 1.943a \ Kp_a$

$$\therefore \quad \Delta P_t = 2.5 \left(\frac{f \text{ x } Vt^2}{2} \right)$$

$$\Delta P_t = 2.5 \left(\frac{832 \text{ x } (0.563)^2}{2} \right) \qquad = 0.3796 \ KP_a$$

$$\Delta P_{total} = Np \text{ x } (\Delta P_L + \Delta P_t)$$

$$= 4 \text{ x } [1.9439 + 0.3796]$$

$$\boxed{\Delta P_{total} = 9.294 \text{ KPa}}$$

ΔP_{total} is less than 70 Kpa hence design is satisfactory.

b) shell side

Shell side pressure drop is calculated using bell 's method

(Perry : page 10-26 to 10-31)

$N_{RC} = 7620$

From figure 10.25 (a) page 10-31 friction factor f_k

$f_k = 0.19$

(i) Pressure drop across cross flow section $\triangle P_c$

$$\Delta P_c = \frac{b \text{ x } f_k \text{ x } w^2 \text{ x } N_c}{f \,.\, f^2_m} \text{ x } (\,Mw/\,Mb\,)^{0.4}$$

$b = 2 \text{ x } 10^{-3}$

$w = 9.54$ kg/s

$S_m = 0.02235m^2$

$N_c = D_s \text{ x } \; 1 - 2(Lc\,/\,D_s)\,/\,P_p$

Where Ds = shell 1D = 0.438m

Lc = 0.1095

P_p = pitch parallel (cross)flow = $\frac{13}{16}$ in = 0.0206m

$$N_c = 0.438 \; \frac{1 - 2\,(0.1096\;/0.438)}{0.0206}$$

$N_c = 16$

$$\therefore \; \Delta P_c = \frac{2 \text{ x } 10^{-3} \text{ x } 0.19 \text{ x } 9.54^2 \text{x } 16}{997.04 \text{ x } 0.02235} \; [1]^{0.4}$$

$$\boxed{= 0.0252 \; K_{Pa}}$$

(ii) <u>End zone pressure drop</u> ΔP_c

$$\Delta P_c \;\; \Delta P_L \;\; 1 + (N_{cw} / \_N_c)$$

$$N_{cw} = \frac{0.8 l_c}{P_p} = \frac{0.8 \times 0.1095}{0.0206}$$

$$= 4$$

$$\therefore \Delta P_c = 0.0252 \times 1 + \frac{4}{16}$$

$$\boxed{\Delta P_c = 0.0315 \; K_{Pa}}$$

(iii) Pressure drop in window zone, ΔP_w

$$\Delta P_w = \frac{b \times w^2 \; [2 + 0.6 \; k l_{cw}]}{fm \times S_w \times f}$$

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

$S_m = 0.02235 \text{ m}^2$

$N_{cw} = 4$

$w = 9.84$ kg / s

$S = 997.045$ kg /m^2

Area for flow though window $Sw = Sw_g - Sw_t$

Sw_g, from fig (10-18) , page (10-29), perry hand book.

$$\boxed{Sw_g = 0.029}$$

$$Sw_t = \frac{N_T}{8} (1 - Fc)\, \pi\, D_o$$

$$= \frac{274}{8}\ x\ (1 - 0.68)\ x\ \pi\ (0.0158)^2$$

$$\boxed{Sw_t = 0.0085}$$

$\therefore S_w = 0.029 - 0.0029$

$\underline{S_w = 0.0205\ m^2}$

Pressure drop at window zone ΔP_w

$= 5\ x\ 10^{-4}\ x\ (9.54)^2\ x\ (2 + 0.6\ x\ 4) / 0.02235x0.0205x997.045$

$\Delta\ P_w = 0.386\ Kp_a$

Total pressure drop at shell side, ΔP_T would be given by

$\Delta P_T = 2\ x\ \Delta P_c + \Delta (N_b - 1)\ x\ \Delta P_c + N_b\ \Delta P_w$ $\triangle$

$\Delta P_T = 2\ x\ 0.0315 + (22 - 1)\ x\ 0.0252 + 22\ x\ 0.386$

$\Delta P_T = 9.08\ Kp_a$

Total pressure drop at shell side is less than 70 Kp_a hence, shell & heat exchanger design is satisfactory.

8. MECHANICAL DESIGN OF PROCESS EQUIPMENTS

1. MECHANICAL DESIGN OF ROTARY DRIER

1. **Flight design**:

Number of flights = 3 x D.

= 3 x2.09

=6.27 ≈ 7 flights are required using lip angle of 45°.

Radial height is taken as 1/8 of diameter,

Radial height = 2.09/8

= 0.2615m.

2. **Thickness of dryer:**

Let x be the thickness of drier.

Mild steel can be used since it can withstand temperature up to 200°C.

Density = 7688.86kg/m^3.

$D_2 - D_1 = 2x$.

Volume of mild steel =$(\pi D_2^2/4 - \pi D_1^2/4) \times L$

$=(\pi(D_1+2x)^2/4 - \pi D_1^2/4) \times L$

$= \pi DLx$.

Weight of dryer = π x12.24 x2.09 x x x 7688.86

= 0.626×10^6 x kg.

Assume holdup = 0.2

Volume of drier filled with material = $\underline{\pi}D^2L$ x0.2

$$= \frac{\pi}{4} \times 2^2 \times 12. \times 0.2$$

$= 7.53\ m^3$.

Weight of material at any time $= 7.53 \times 1049.2$

$= 11219.7$ kg.

The dryer is supported over two-tension roll assemblies, 20ft apart. It is uniformly distributed load.

Maximum bending moment = WL/8 = M.

$M = (0.626 \times 10^6 x/8 + 11219.7/9) \times 12$

$= 0.939 \times 10^6 x + 16829.5$

We know that

$M = f \times Z$.

$Z = \pi \times (D_2^4 – D_1^4) / 32D_2$.

$= 0.785x^3 + 12.59x^2 + 67.31x$.

$f = 1800$psi.

Take factor of safety = 5.

$f = 3.6 \times 10^5 lb/ft^3$.

$= 1.75767 \times 10^4 kg/m^2$.

Thus $M = f \times Z$ on simplification becomes,

$$1.38 \times 10^6 x^3 + 22.13 \times 10^6 x^2 + 113.264 \times 10^6 x – 0.819 = 0$$

x = 15 mm

3. Diameter of the feed pipe:

Feed rate =10417+212.9= 10629.9 kg/hour

Density of feed = 1410 kg/m^3

Hence volumetric feed rate = 10629.9/1410 = 7.534 m^3 /hr

Assuming the velocity of air = 150 m/hr , for chute inclination of 60^0

Cross-section of feed chute = 7.53/ 150 = 0.050 m^2

Diameter of feed chute = $\sqrt{}$ (C.S.A. x4 /π) = $\sqrt{}$ (0.050 x 4 /π)

= 0.252 m

4. BHP to drive the drier:

BHP = r x (4.75 x d x w + 0.1925 x d x w + 0.33 w) /1000

Where,

w= weight of the drier + weight of the material + weight of the insulation.

w = π x12 x2 x 0.01 x 7688.86 + π /4 x (2^2 12 x0.1 x1410)

w = 28.65.65x 10^3 kg

HP of blower:

Temperature of atmospheric = 30^0 C

Humidity in air= 16743. 89 kg/ min = 915. 5 ft^3 /min

Volume of this air , Q = 279.05 / 29 x 22. 4 x 303/ 298

= 219.9 m^3/min

= 718.9 ft^3 /min

HP of blower = 0.000157 x Q x (head developed by water)

= 0.000157 x718.9 x10

=1.2 hp

5. HP of exhaust fan

Outlet temperature of drier = 95.62 ^{0} C

Humidity of outlet air = 0.65 x 0.00726

Total quantity of air going out = 16743.9 kg/hr = 279.05 kg/min

Volume of this air = (279.05/29) x 22.4 x (368.62 /298)

= 406.9 m^3/min or 1335.13 ft^3/min.

HP of exhaust fan = 0.000517 x 1335.13 x16

= 6.90 hp

6. Diameter of outlet and inlet pipe

At inlet conditions of 150 ^{0} C and humidity of 0.002

the volume of air handled = 219.2 x 423 / 303

=306 m^3/min or 5.1 m^3/sec.

Assuming air velocity = 25 m / s,

C.S.A of inlet pipe = 5.1/25 =0.20 m^2

Inlet pipe diameter = 0.504 m

At outlet conditions of 95.62^{0}C

The volume of air handled = 219.2 x 368 /313

= 4.43 m^3/sec

C.S.A of outlet pipe = 4.43 /25 = 0.178 m^2

Outlet pipe diameter = 0.476 m

2. MECHANICAL DESIGN OF HEAT EXCHANGER

(a) Shell side details

Material : carbon steel
Number of shell passes: 2
Working fluid: water
Working pressure: 0.1N/mm^2
Design pressure : 0.11N/mm^2
Inlet temperature: 15^{0}C
Out let temperature: 35^{0}C
Permissible stress for carbon steel: 95N/mm^2
Shell inner diameter: 438 mm

(b) Tube side details

Number tubes: 274
Number of passes: 4
Outside diameter: 15.87mm
Inside diameter : 12.57 mm.
Length: 4.88m
Pitch triangular:13/16 inch
Working pressure: 0.1 N/mm^2
Design pressure: 0.11N/mm^2
Inlet temperature : 97^{0}C
Outlet temperature: 20 ^{0}C

SHELL SIDE

1.Shell thickness

t_s= PD/(2fJ+P)

= 0.11 x438/(2 x95 x0.85+0.11)

= 0.31

Minimum thickness of shell must be=6.0 mm

Including corrosion allowance shell thickness is 8mm

2.Head thickness.

Shallow dished and torispherical

$w = 1/4 \ x\ (3 + \sqrt{(}R_c / R_K)$

$= 1/4 \ x\ (3 + \sqrt{(}R_c / .06\ R_c)$

$= 1.77$

t = $PR_cW/2fJ$

= 0.11 x305 x1.77/(2 x95 x0.85)

= 0.528 mm.

minimum shell thickness should be 10mm including corrosion allowance.

3.Transverse Baffles

Baffle spacing =0.8 xD_c

= 350 mm

number of baffles,

N_b+1=L/L_S=4.88/0.350=14

N_b =13

Thickness of baffles, t_b=6mm

4.Tie Rods and spacers:

Tie rods are provided to retain all cross baffles and take support plates accurately.

For shell diameter, 300-500mm

Diameter of Rod = 9mm

Number of rods=4

5.Flanges

Design pressure=0.11 N/mm^2

Flange material IS:2004-1962,class 2

Bolting steel :5% Cr-Mo steel

Gasket material: asbestos composition

Shell inside diameter = 438mm.

Shell thickness: 8mm=g_o

Outside diameter of shell: 446 mm

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

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

Shell thickness of flange = 10 mm.

Outside diameter of flange = 325 mm.

6. Determination of gasket width

$d_O/d_i = [(y-Pm)/(y-P(m+1))]^{0.5}$

Assume a gasket thickness of 10 mm

y = minimum design yield seating stress = 25.5 MN/m^2

m = gasket factor = 2.75

$d_O/d_i = [(44.85-0.11 \times 2.75)/(44.85-0.11(2.75+1))]^{0.5}$

d_O/d_i = 1.001=1.001

d_O = 1.001 x 0.438= 0.4385 m

Minimum gasket width = (0.4385 -0.438)/2 = .00075.

Taking gasket width of N= 0.010m

d_o= 0.458 m.

Basic gasket seating width, b_o = 5mm

Diameter of location of gasket load reaction is

G= d_i + N

=0.438+0.01

= 0.448 m

7.Estimation of Bolt loads.

Load due to design pressure

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

= 3.14 x0.448^2 x0.11/4

= 0.01756 MN

Load to keep joint tight under operation

$b = 2.5 (b_0)^{0.5}$ = 6.12mm.

$H_p = \pi G(2b)mp$

= 3.14 x0.448 x (2 x0.00612) x2.75 x0.11

= 0.00656 MN

Total operating load

$W_o = H + H_p$

=0.01755 + 0.00656

=0.0241 MN.

Load to seat gasket under bolting condition

$W_g = \pi Gby$

$= 3.14 \text{ x } 0.448 \text{ x } 6.12 \text{ x } 10^{-3} \text{ x } 25.5$

= 0.862MN.

$W_g > W_o$,controlling load=0.8620 MN

8.Calculation of optimum bolting area

$A_m = A_g = W_g / S_g$

= 0.862 /138

$= 6.246 \times 10^{-3} \text{ m}^2$

Calculation of optimum bolt size

Bolt size,M18 X 2

Actual number of bolts =20

Radial clearance from bolt circle to point of connection of hub or nozzle and back of flange = R = 0.027 m

C =ID + 2(1.415g + R)

=325 +2[1.41 x0.008+0.027]

= 0.726m

Bolt circle diameter = 0.40163 m.

Using bolt spacing Bs = 45mm

C = n Bs / 3.14 =44 x 0.045 / 3.14 = 0.63

Hence C = 0 .726

Calculation of flange outside diameter

Let bolt diameter = 18 mm.

A= C+ bolt diameter +0.02

=0.716 +0.018+0.02

= 0.764m.

Check for gasket width

$A_bS_G / (\pi GN) = 1.54\times10^{-4}$ x44 x138/(3.14 x0.4486^2)

= 66.43 < 2 xy.

where S_G is the Allowable stress for the gasket material

9.Flange moment computation:

(a) For operating condition

$W_o = W_1+W_2+W_3$

$W_1 = \prod B^2P/4$

$= \prod$ x0.446^2 x0.11/4

= 0.0173 MN

W_2=H-W1

=0.01756-0.0173

= 1.6×10^{-4} MN.

$W_3=W_o-H=H_p$

= 0.00672MN.

M_o=Total flange moment

$M_o=W_1a_1 + W_2a_2 + W_3a_3$

a_1=(C-B)/2=(0.726-0.446)/2

a_1=0.14 m

a_3=(C-G)/2=(0.726-0.448)/2

a_3=0.1395m

$a_2=(a_1 + a_3)/2$= (0.14 +0.139 0.)/2

=0.139m

M_o=0.01739 x0.140 +1.60×10^{-4} x0.1395 x0.00672 +0.139

M_o= 3.391×10^{-3} MN-m

(b) For bolting condition

$M_g = Wa_3$

$W = (A_m + A_b)\ xS_g/2$

$W = (6.246\times10^{-03} + 6.76\times10^{-3})\ x138/2$

W= 0.897 MN

$M_g = 0.0.897\ x\ 0.139$

$= 0.125$ MN-m

$M_g > M_o$,Hence moment under operating condition M_g is controlling, $M_g = M$

10.<u>Calculation of flange thickness</u>

$t^2 = M\ C_F\ Y / (B\ S_F)$, S_F is the allowable stress for the flange material

K =A/B = 0.764/0.446 = 1.71

For K = 1.71 Y = 4.4

Assuming $C_F = 1$

$t^2 = 0.125\ x\ 1\ x4.4/ (0.446\ x\ 100)$

t= 0.11m

Actual bolt spacing $B_S = \pi C/n = (3.14)(0.776)/(44) = 0.052$m

11.<u>Bolt Pitch Correction Factor</u>

$C_F = [B_s / (2d+t)]^{0.5}$

$= (0.052/(2\ x0.018+0.11)^{1/2}$

$= 0.596$

$\sqrt{C_F} = 0.772$

Actual flange thickness = $\sqrt{C_F}$ xt

= 0.11 x0.772

= 0.085 m

= 85mm.

12.Channel and channel Cover

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

= 0.446 x$\sqrt{(0.25 \times 0.11/95)}$

= 0.00767m =7.67mm

t_h= 8mm including corrosion allowance

13.Tube sheet thickness

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

= 1 x0.448$\sqrt{(0.3 \times 0.11/95)}$

= 0.0084=8.84 mm

t_{ts}= 9mm including corrosion allowance.

14. Saddle support

Material: low carbon steel

Total length of shell: 4.88 m

Diameter of shell: 4.38 mm

Knuckle radius : 6% of crown radius =26.28 mm

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

= $\sqrt{(438 \times 26.28/2)}$

H = 75.86mm

A= 0.5 R = 0.5 x 438/2 = 109.5 mm.

weight of vessel and contents = weight of (shell + tube)

weight of the steel = 7600 kg/m^3.

Weight of shell = π D x 0.008 x 7600 x L

= 83.67 x 4.88

= 408.30 kg

Weight of tube = π (19.05 $\times 10^{-03}$ - 12.27 $\times 10^{-03}$) x 4.88 x 7600 x 274

= 1480 kg

weight of water = π 0.01224^{2} x 4.88 995 x 274

= 626.19 kg

weight of vessel and contents W = 2215.69 kg

15.<u>Longitudinal Bending Moment</u>

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

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

= 2215.6/2 x(4.88 +4 x0.07586/3)

= 5518kg m

M_1 = 598.05 kg-m kg-m

16. <u>Bending moment at center of the span</u>

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

M_2 =6014. 4 kg-m

17.<u>Stresses in shell at the saddle</u>

(a) At the topmost fibre of the cross section

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

$= 598.06 / (3.14 \text{ x}0..219^2 \text{ x}0.01)$

= 35.22 kg/cm^2

The stresses are well within the permissible values.

Stress in the shell at mid point

$f_2 = M_2/(k_2 \pi R^2 t)$ = 6014.4 / (1 x π 0.219^2 0.01)

$= 399.07\ \text{kg/cm}^2$

Axial stress in the shell due to internal pressure

$f_p = PD/4t$

$= 0.11\ \text{x}438/(\ 2\ \text{x}\ 10)$

$= 225.53\ \text{kg/cm}^2$

$f_2 + f_p = 624.64\ \text{kg/cm}^2$

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