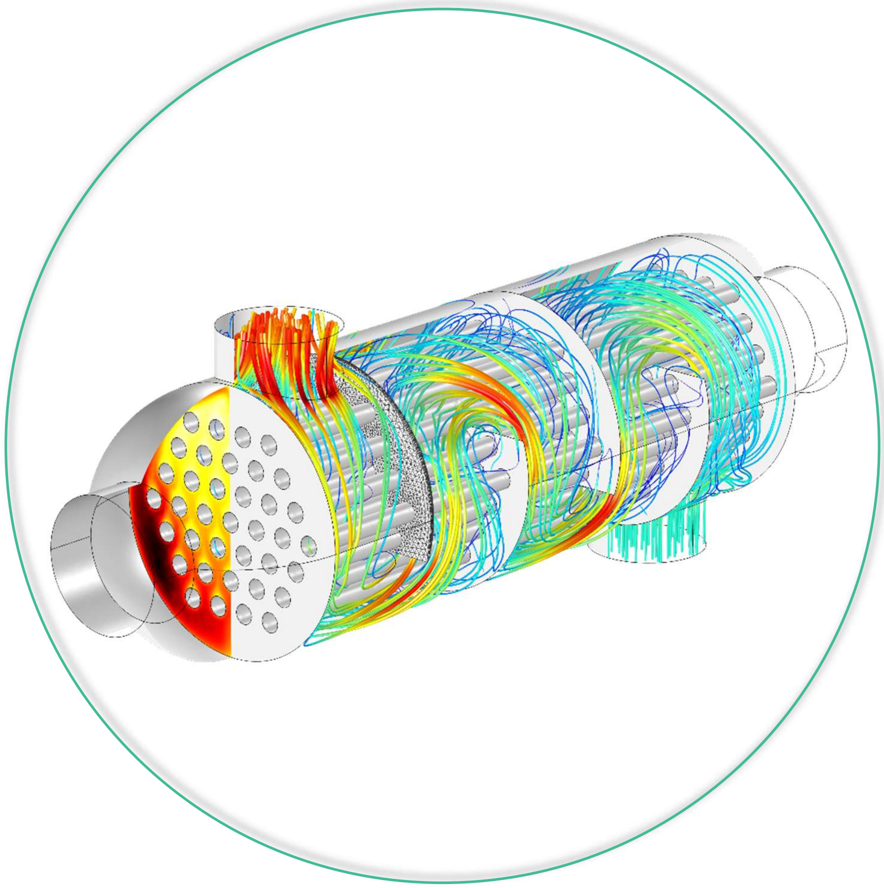
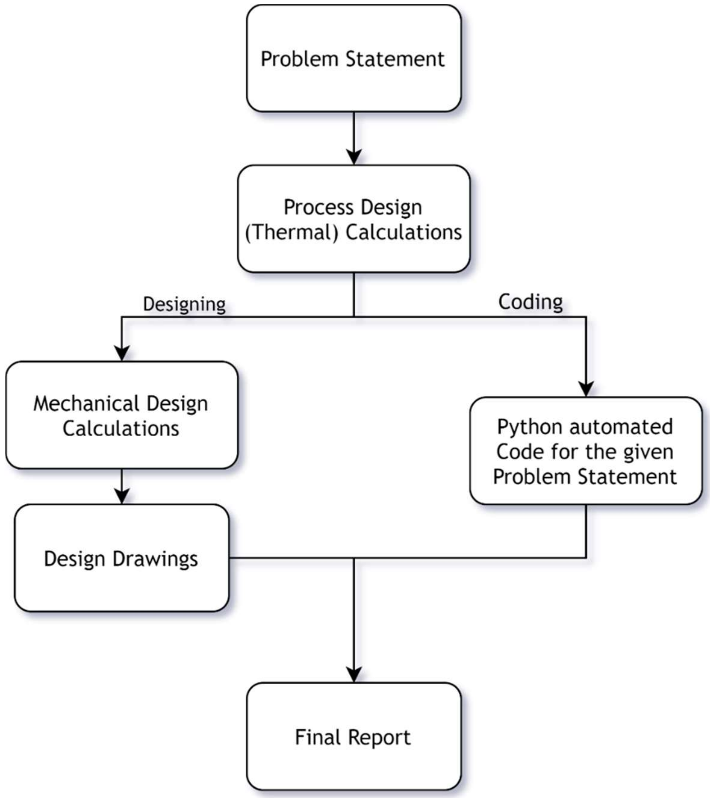
CH39023: PROCESS EQUIPMENT DESIGN-I

LAB REPORT: SHELL AND TUBE HEAT EXCHANGER



Group Number: 4		
Sl.No.	**Name**	**Roll Number**
1	Ashad Ahmad	18CH10012
2	Aravind Barla	18CH10016
3	Chaitanya Sai Sahu	18CH10019
4	Suvesh Bansal	18CH10057
Professors In charge	Prof. Sourav Mondal	
	Prof. Bhaskar Bhaduri	

CONTENTS & THE FLOW CHART



Portion	Name	Contribution	Pg
Process Design Calculations	Chaitanya	Standard Considerations, LMTD, LMTD Correction Factor, Caloric Temperatures, Enthalpy Balance	3-17
	Ashad	Assumed Heat coefficient, number of tubes, Shell Dimensions, Tube side Reynolds number, Tube side heat coefficient	
	Aravind	Shell Side Reynolds Number, Shell side heat coefficient, Overall Heat coefficient, further Iterations	
	Suvesh	Tolerance, Tube side Pressure Drop, Overdesign limits	
Mechanical Design Calculations & Drawing	Ashad	Design Considerations, Shell Cover, Channel Cover, Tube sheet thickness; Tori-spherical Head and Shell Flange drawings	18-31
	Aravind	Impingement Plates/ Baffles, Nozzle, Gasket, Bolts, Flange; Side View of Overall HE, Tube sheet Layout drawings	
Python Code	Chaitanya	Initial Code containing Plot fitting and Regression Curves	Python File
	Suvesh	Later part of the Code with Reynolds and Heat coefficients	
Final Report	Chaitanya	Final Handwritten Calculations (Process & Mechanical)	
	Suvesh	Verifying the Handwritten Calculations with sample Code Output	
	Aravind	Final Drawings (HE Side view & Solidworks drawing sheets)	

PROBLEM STATEMENT

Design a Shell and Tube Heat Exchanger (stripped Heavy Naphtha Trim Cooler) to cool the Heavy Naphtha stream of

$$\left[4.0 + 0.03 \times (\text{Group Number})\right] \frac{kg}{s}$$

at 65°C to 45°C using water at 35°C, as the coolant. The temperature difference in the coolant is 10°C.

→ The design tolerance for the overall heat transfer coefficient should be max. 5%.

→ The tube side pressure drop should be less than 10 psi.

→ The overdesigned area tolerance should not be beyond 10%.

→ Design the following HE components:-

- Shell cover
- Channel cover
- flange
- tube-sheet
- baffles
- pass partition plate
- support
- Bolts
- Nozzle

PART - II

PROCESS DESIGN

→ **Fluid Allocation:-**

We have chosen Heavy Naphtha to be the shell side fluid as

- It has to undergo a larger temp. differ
- It is the more viscous of the two.

and water has a greater fouling resistance than Heavy Naphtha, hence Water is chosen as the tube side fluid.

→ **Process Requirements:-**

1) Design Tolerance < 5%
2) Tube Side Pressure Drop < 10psi
3) Overdesigned Area < 10%

→ **GIVEN DATA:-**

Hot Fluid Inlet Temperature ($T_{h,i}$) = 65°C

Desired Outlet Temperature of Hot Fluid ($T_{h,o}$) = 45°C

Cold Fluid Inlet Temperature ($T_{c,i}$) = 35°C

Cold Fluid Outlet Temperature ($T_{c,o}$) = 45°C

$$\dot{m}_h = 4 + 0.03(4) = 4.12 \ \frac{kg}{s}$$

I) Caloric Temperature:-

$$r = \frac{\Delta t_c}{\Delta t_h} = \frac{T_{h,out} - T_{c,i}}{T_{h,i} - T_{c,o}}$$

API gravity of Heavy Naphtha = 47.3

$$\Delta T_{shell} = T_{h,i} - T_{h,o} = 20°C = 68°F$$

For the given API & ΔT of shell side fluid,

$$K_c = 0.0886$$

$$r = \frac{45°C - 35°C}{65°C - 45°C} = \frac{1}{2}$$

For the given r & Kc,

$$\boxed{F_c = 0.4357}$$

$$T_h^* = T_{h,o} + F_c (T_{h,i} - T_{h,o})$$

$$= 45 + 0.4357 \times (65 - 45)$$

$$\boxed{T_h^* = 53.7137°C}$$

$$T_c^* = T_{c,i} + F_c (T_{c,o} - T_{c,i})$$

$$= 35 + 0.4357 \times (45 - 35)$$

$$\boxed{T_c^* = 39.3569°C}$$

II) Enthalpy Balance:-

$$\dot{Q} = \dot{m}_h \, C_{p_h}\big|_{T_h^*} \times (T_{h,in} - T_{h,o})$$

$$C_{ph}\big|_{T_h^*} = 2170.0844 \, \frac{J}{kg \cdot K}$$

$$\dot{Q} = 4.12 \, \frac{kg}{s} \cdot 2170.0844 \, \frac{J}{kg \cdot K} \cdot 20K$$

$$\boxed{\dot{Q} = 178814.9573 \, \frac{J}{s}}$$

$$\dot{m}_c \, C_{pc}\big|_{T_c^*} (T_{c,o} - T_{c,i}) = \dot{Q}$$

$$\Rightarrow \dot{m}_c = \frac{\dot{Q}}{C_{pc}\big|_{T_c^*} \times (T_{c,o} - T_{c,i})}$$

$$C_{pc}\big|_{T_c^*} = 4178.759 \, \frac{J}{kg \cdot K}$$

$$\Rightarrow \dot{m}_c = \frac{178814.9573 \, \frac{J}{s}}{4178.759 \, \frac{J}{kg \cdot K} \cdot 10K}$$

$$\Rightarrow \boxed{\dot{m}_c = 4.2791 \, \frac{kg}{s}}$$

III) LMTD:-

$$\Delta T_1 = T_{h,i} - T_{c,o}$$

$$\Rightarrow \Delta T_1 = 65°C - 45°C = 20K$$

$$\Delta T_2 = T_{h,o} - T_{c,i}$$

$$\Rightarrow \Delta T_2 = 45°C - 35°C = 10K$$

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{20-10}{\ln(2)}$$

$$\boxed{LMTD = 14.427\,K}$$

IV) LMTD Correction Factor F_T

$$R = \frac{T_{h,i} - T_{h,o}}{T_{c,o} - T_{c,i}} = \frac{20}{10} = 2$$

$$S = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} = \frac{10}{30} = \frac{1}{3}$$

For 1-2, 1-4 & 1-6 heat exchangers:-

$$F_{T(1-2)} = 0.8052$$

For 2-4, 2-6 heat exchangers:-

$$F_{T(2-4)} = 0.9583$$

For 3-6 heat exchangers:-

$$F_{T(3-6)} = 0.98$$

Since, 1-2 & 1-4 HE don't have an $F_T >= 0.9$, 2-4 heat exchanger is the best option with $F_T = 0.9583$

V) Assumed U

Overall $U_D \in (50, 125) \frac{BTU}{hr.ft^2.°F}$

$$U_D = \frac{50+125}{2} = 87.5 \left(\frac{BTU}{hr.ft^2.°F}\right) = 496.848 \frac{W}{m^2K}$$

$$\boxed{U_{assum} = 496.848 \ \frac{W}{m^2K}}$$

$$(U_{assum})\, A \times (LMTD \times F_T) = \dot{Q}$$

$$\Rightarrow \text{Area } A = \frac{\dot{Q}}{U_{assum} \times LMTD \times F_{T(2-4)}}$$

$$\Rightarrow A = \frac{178814.9573}{496.848 \times 14.427 \times 0.9583}$$

$$\Rightarrow \boxed{A = 26.0311 \ m^2}$$

VI) Number of Tubes, n_t

$L = 10ft$ ✗ $\Rightarrow$ not suitable as n_t becomes large in further iterations.

tube length $(L) = 20ft = 6.096m$

~~BWG~~ Considering the standards,

d_i = tube ID = $\frac{3}{4}$ inch = 0.01905m

d_o = tube OD = 1 inch = 0.0254m

BWG = 10.6428

$$\because (\pi d_o L)\, n_t = A$$

$$\Rightarrow n_t = \frac{A}{\pi d_o L}$$

$$= \frac{26.0311}{\pi \times 0.0254 \times 6.096}$$

$$n_t = 53.5135$$

$$\Rightarrow \boxed{\lceil n_t \rceil = 54}$$

VII) Number of Available Set of Tubes (n_a) & Shell Dimensions:-

Since, we have considered 2 shell passes,

tube passes ≥ 4.

Restricting tube passes to only 4:-

N_t = tube passes = 4

n_a = design number of tubes $\gtrsim n_t$

Layout	N_t	n_a	Shell ID (D_{is})
$1\frac{1}{4}$ inch square pitch	4	68	15.25 inch
$1\frac{1}{4}$ inch triangular pitch	4	58	13.25 inch

VIII) Tube Side Reynolds number:-

$$\rho_w|_{T^*c} = 992.5674 \ kgm^{-3}$$

$$\mu_w|_{T_c^*} = 0.00066072 \ Pa.s$$

i) tube side flow velocity, $V_t = \frac{\dot{m}_w}{\rho_w} \cdot \left(\frac{4}{\pi d_i^2}\right) \cdot \left(\frac{N_t}{n_a}\right)$

$$= \frac{4.2791}{992.5674} \cdot \frac{4}{\pi(0.01905)^2} \cdot \frac{N_t}{n_a}$$

$$\boxed{V_t = \left(15.1257 \times \frac{N_t}{n_a}\right) \frac{m}{s}}$$

ii) $Re_{tube} = \frac{(\rho_w)(V_t)(d_i)}{\mu_w} = \frac{992.5674(15.1257)0.01905}{0.00066072} \cdot \frac{N_t}{n_a}$

$$\boxed{Re_{tube} = 432861.81 \frac{N_t}{n_a}}$$

To obtain $\underbrace{Re_{tube} > 10^4}_{\text{Sufficient Turbulance}}$, N_t has to be 4 or more.

($P_T = 1\frac{1}{4}$ in) Square Pitch :- $N_t = 4$, $n_a = 68$ $\Rightarrow \boxed{Re_{tube} = 25462.46}$

$1\frac{1}{4}$ in triangular Pitch :- $N_t = 4$, $n_a = 58$ $\Rightarrow \boxed{Re_{tube} = 29852.54}$

Square Pitch is prefered over triangular Pitch :-

i) n_a is not too high

ii) cleaning is easier

iii) low pressure drop

iv) $Re_{Tube}|_{Triangular}$ is not significantly higher than $Re_{tube}|_{square}$

IX) Tube Side Heat Transfer Coefficient:-

$$K_W|_{T_c^*} = 0.62777 \frac{W}{K}$$

$$\mu_W|_{T_h^*} = 0.0005141 67 \text{ Pa.s}$$

i) $$Pr_w|_{T_c^*} = \left(\frac{C_p \mu}{k}\right)\Big|_{T_c^*} = 4.3981$$

ii) Since, a) $Pr_W \in (0.7, 16700)$

b) $$\frac{L}{d_i} = \frac{6.096}{0.01905} = 320 \; [\geq 60]$$

c) $Re_{tube} \geq 10^4$

We can use the slightly modified version of Dittus Boilter Equation:-

$$Nu = 0.027 \cdot Re^{0.8} Pr^{1/3} \left(\frac{\mu_{\infty}}{\mu_{wall}}\right)^{0.14}$$

$$\Rightarrow \frac{h_i d_i}{K_w|_{T_c^*}} = 0.027 \times (25462.46)^{0.8} (4.3981)^{1/3} \times \left[\frac{\mu_w @ T_c^*}{\mu_w @ T_h^*}\right]$$

$$\Rightarrow \boxed{h_i = 5054.3024 \ \frac{W}{m^2 K}}$$

X) Shell side heat transfer coefficient:-

i) Important considerations :- a) 25% cut segmental baffles

b) Baffles spacing, $B = \frac{D_s}{2}$

$D_s \equiv$ shell ID $= 15.25$ inch

$B = \frac{15.25}{2}$ inch

Tube clearance $(C) = P_T - d_o = 1\frac{1}{4} - 1 = 0.25$ inch

Shell side cross flow area, $a_s = \frac{C B D_s}{P_T} = 23.25625 \text{ inch}^2$

$$a_s = 0.015004 \ m^2$$

ii) Shell side cross flow velocity, $V_s = \frac{\dot{m}_h}{(\rho_h|_{T_h^*}) \cdot a_s}$

iii) Square Pitch Equivalent Diameter, $D_e = \frac{4\left(P_T^2 - \frac{\pi}{4} d_o^2\right)}{\pi d_o}$

$$\boxed{D_e = 0.025132 m}$$

iv) $K_h\big|_{T_h^*} = 1.04 \ \dfrac{Btu \cdot inch}{hr \cdot ft^2 \cdot °F}$

$$\Rightarrow K_h\big|_{T_h^*} = 0.149997 \ \frac{W}{mK}$$

Using the regression relation,

$$\mu = A e^{\left[\frac{B}{T - T_0} + CT\right]}$$

for viscosity of Naphtha,

$$\mu_h\big|_{T_h^*} = 0.001403 \ Pa \cdot s$$

$$\mu_h\big|_{T_c^*} = 0.0024764 \ Pa \cdot s$$

---

v) $$Re_{shell} = \frac{\left(\rho_h\big|_{T_h^*}\right) \times V_s \times D_e}{\mu_h\big|_{T_h^*}} = \frac{\dot{m}_h \times D_e}{a_s \times \mu_h\big|_{T_h^*}}$$

$$\boxed{Re_{shell} = 4918.21}$$

---

vi) Colburn Factor :-

From shell side heat transfer curve (bundles with 25% cut Segment Baffles)

$Re = 4918 \Rightarrow \underline{j_H = 37.5096}$

$$\Rightarrow 37.51 = \frac{h_o D_e}{K_h\big|_{T_h^*}} \cdot \left(Pr_h\big|_{T_h^*}\right)^{-1/3} \cdot \left(\frac{\mu_\infty}{\mu_{wall}}\right)^{-0.14}$$

$$\Rightarrow 37.51 = h_o \left(\frac{0.025132}{0.149997}\right) \cdot \left(\left(\frac{C_{ph}\mu_h}{k_h}\right)\bigg|_{T_h^*}\right)^{-1/3} \cdot \left(\frac{\mu_h @ T_h^*}{\mu_h @ T_c^*}\right)^{-0.14}$$

$$\Rightarrow \boxed{h_o = 564.022 \ \frac{W}{m^2 K}}$$

XI) Overall Heat Transfer coefficient

i) R_{do} ≡ tube outside dirt / Naphtha's fouling resistance

$$R_{do} = 0.00018 \frac{m^2K}{W}$$

R_{di} ≡ tube inside dirt / Water's fouling resistance

$$R_{di} = 0.00075 \frac{m^2K}{W}$$

Town's Water
(Hard)

ii) Tube Material ≡ Steel Carbon, 0.5% C

$$K_{wall} = 54 \frac{W}{mK}$$

iii) $$\frac{A_o}{A_i} = \frac{\pi d_o^2}{\pi d_i^2} = \frac{16}{9}$$

iv) $$\frac{1}{U_{cal}} = \left[\frac{1}{h_o} + R_{do} + \frac{A_o}{A_i} \left\{ \frac{d_o - d_i}{2K_{wall}} + R_{di} + \frac{1}{h_i} \right\} \right]$$

Substituting & evaluating,

$$U_{cal} = 267.1957 \frac{W}{m^2K}$$

v) relative error $= \left| \frac{U_{cal} - U_{assum}}{U_{assum}} \right|$

$$= \left| \frac{267.1957 - 496.848}{496.848} \right|$$

$$= 46.222\%$$

vi) tolerance = 5%

relative error < tolerance ⇒ reiterating...

→ ITERATION 1 with $U_{assum} = U_{cal}$

V) $U_{assum} = 267.1957 \frac{W}{m^2K}$

$$A = \frac{\dot{Q}}{(U_{Assum})\, LMTD \times F_T}$$

$$A = 48.4046\, m^2$$

VI) $n_t = \frac{A}{\pi d_o L} = 99.508$

$$\lceil n_t \rceil = 100$$

VII) $P_T = 1\frac{1}{4}$ inch, square layout

$N_t = 4 \Rightarrow n_a = 128, \quad D_s = 0.48895\, m = 19\frac{1}{4}$ inch

VIII) $Re_{tube} = (432861.81)\frac{N_t}{n_a}$

$\rightarrow Re_{tube} = 13526.93 \quad (\geq 10^4)$

IX) $h_i = \frac{Kw|_{T_c^*}}{d_i} \times 0.027 \times (13526.93)^{0.8} (Pr^{1/3}) \cdot \left(\frac{\mu_\infty}{\mu_{wall}}\right)^{0.14}$

$\Rightarrow h_i = 3047.196 \frac{W}{m^2K}$

X) i) $B = \frac{D_s}{2} = \frac{19.25 \text{ inch}}{2}$

$a_s = \frac{CBD_s}{P_T} = 0.023907 m^2$

ii) $Re_{shell} = \frac{\dot{m}_h \times D_e}{a_s \times \mu_h|_{T_h^*}} = 3086.635$

iii) $jH = 29.3323$

$\Rightarrow h_o = j_H * 15.037$

$\Rightarrow h_o = 441.062 \ \frac{W}{m^2K}$

XI) i) $\frac{1}{U_{cal}} = \left[\frac{1}{h_o} + \frac{1}{h_i} \cdot \frac{A_o}{A_i}\right] + (1.61786 \times 10^{-3})$

$\Rightarrow \boxed{U_{cal} = 223.7873 \ \frac{W}{m^2K}}$

ii) relative error $= \frac{267.1957 - 223.7873}{267.1957} \times 100$

$\boxed{\text{relative error} = 16.2459\%}$ ($> 5\%$)

$\Downarrow$

reiterate

→ ITERATION 2 :-

V) $U_{assum} = 223.7873 \ \frac{W}{m^2K}$

$A = \frac{\dot{Q}}{(U_{assum}) \times LMTD \times F_T}$

$A = 57.7937 m^2$

VI) $n_t = \frac{A}{\pi d_o L} = 118.81$

$\lceil n_t \rceil = 119$

VII) $P_T = 1\frac{1}{4}$ inch, square layout

$N_t = 4$, $\Rightarrow \boxed{n_a = 128}$, $D_s = 19\frac{1}{4}$ inch

$\because$ n_a didn't change, Re_{tube}, h_i, Re_{shell}, h_o, U_{cal} will also remain the same.

XI) $U_{cal} = 223.7873 \frac{W}{m^2K}$

relative error $= \left|\frac{U_{cal} - U_{assum}}{U_{assum}}\right| \times 100$

$\Rightarrow$ relative error $= 0\% < 5\%$ tolerance

$\Rightarrow$ The overall heat transfer coefficient of the 2-4 Shell & Tube Heat Exchanger will be

$$\boxed{U = 223.78727 \frac{W}{m^2K}}$$

XII) PRESSURE DROP :-

$n_a|_{final} = 128$ tubes

i) $Re_{tube} = 13526.93$

friction factor (f) $= 0.03641$

ii) tube side velocity,

$$V_t = \left(15.1257 \times \frac{N_t}{n_a}\right) \frac{m}{s}$$

$$V_t = 0.47268 \frac{m}{s}$$

iii) Frictional Pressure Drop (from Darcy-Weisbatch eq)

$$\Delta P_f = \frac{1}{2} f \left[\rho_w|_{T_c^*} V_t^2 \times \left(\frac{L}{d_i}\right)\right]$$

$$\boxed{\Delta P_f = 1291.906 \ Pa}$$

iv) Return loss due to change in flow direction in the tubes:-

$$\Delta P_r = \frac{4 N_t}{SG_w|_{T^\circ C}} \times \frac{V_t^2}{2g}$$

$$\boxed{\Delta P_r = 0.18363 \ Pa}$$

v) Neglecting nozzle loss,

total pressure loss in tube side,

$$\Delta P = \Delta P_f + \Delta P_r$$

$$= 1292.09 \ Pa$$

$$= 0.187402 \ psi$$

$$\boxed{\Delta P = 0.187402 \ psi} \ll 10 \ psi$$

since pressure drop on the tube side is very low, it is justified to use square pitch.

XIII) Overdesigned Area:-

$$\% \text{ overdesign} = \left(\frac{A_{actual} - A_{req}}{A_{req}}\right) \times 100$$

$$= \left(\frac{n_a - \lceil n_t \rceil}{\lceil n_t \rceil}\right) \times 100$$

$$= \frac{128 - 119}{119} \times 100$$

$$\boxed{\% \text{ overdesign} = 7.563\%} \ \boxed{< 10\%}$$

PART - II

MECHANICAL DESIGN

→ DATA CONSIDERED :-

i) Shell side passes = 2, Tube side passes = 4

ii) No. of tubes (n_a) = 128;

Tube dimensions :- $L = 6\,ft$ (= 6.096m)

d_i = 19mm (= 0.01905m)

d_o = 25.4mm (0.0254m)

Tube material : Carbon Steel (IS: 4503-1967, Pg 22)

Pitch : Square pitch 1¼ inch

iii) Shell diameter, ID = D_i = 488.95mm (= 19.25 inch)

Shell and head ; Carbon Steel material, Torispherical head

iv) Carbon steel corrosion allowance in petroleum/chemical industries where severe conditions are expected (C) = 3mm

v) Design Temperature = 1.1 × T_{max}

$T_{max} = T_{hi} = 65°C = 149°F$

$T_{design} = 1.1\,T_{max} = 1.1 \times (149°F) = 163.9°F$

$T_{design} = 73.2778°C$

Design Pressure = 1.1 × P_{inlet} = 1.1 × 50 Psia

= 55 psia = 0.38 N/mm²

vi) Permissible stress, f = 100.6 N/mm² [Carbon Steel]

I) SHELL Thickness Calculations:-

$p = 0.38\ N/mm^2$

$D_s = 488.95\,mm$

$f = 100.6\,N/mm^2$

$J = 0.8$ (Joint Efficiency)

$$t_s = \frac{pD_s}{fJ - 0.6p} + C = 6.31522\,mm$$

$\boxed{t_s = 10\,mm}$ [IS:4503-1967, Pg 22, Chemical/Petrol Industries]

II) Shell Cover:-

i) Crown Radius $R_i = D_s = 488.95\,mm$

Knuckle Radius $r_i = 6\% R_i = 29.337\,mm$

Inside depth of head $(h_i) = R_i - \sqrt{\left[\left(R_i - \frac{D_s}{2}\right)\left(R_i + \frac{D_s}{2}\right) + 2r_i\right]}$

$h_i = 65.4376\,mm$

ii) Effective exchanger length $(L_{eff}) = L_t + 2h_i$

$L_{eff} = 6.2269\,m$

iii) $W = \frac{1}{4}\left(3 + \sqrt{\frac{R_i}{r_i}}\right) = 1.77$

iv) For head design, $J = 1$ (taken)

$$\Rightarrow t_h = \frac{pR_iW}{2fJ - 0.2p} + C = 4.6351\,mm$$

$t_h = max(t_h, t_s) \Rightarrow \boxed{t_h = 10\,mm}$

III) Channel Cover Thickness:-

D_c = outside shell diameter $= D_s + 2t_h = 508.95\,mm$

$C_1 = 0.3$ (cover is bolted with narrow faced or ring type gaskets)

$P = 0.38\,N/mm^2 = \frac{0.38}{9.8} = 3.87492\,\frac{kgf}{cm^2}$

$f = 100.6\,N/mm^2 = 10.258\,\frac{kgf}{mm^2}$

$$\Rightarrow t_{cc} = \frac{D_c}{10}\sqrt{\frac{C_1 P}{f}} = 5.34919\,mm + \overset{C}{(3\,mm)}$$

$$= 8.34919\,mm$$

$t_{cc} = \max(t_{cc}, t_s) \Rightarrow \boxed{t_{cc} = 10\,mm}$

IV) Thickness of Pass Partition Plate:-

$\because D_c < 600\,mm$

$\Rightarrow$ thickness of channel pass partition plate $= 10\,mm$ (including corrosion allowance)

$$\boxed{t_{ppp} = 10\,mm}$$

V) Tube Sheet Thickness:-

$F = 1$ (fixed tube sheet) → supported tube sheets TEMA Pg. 55

$G_p = D_s = 488.95\,mm$

$f = 100.6\,N/mm^2$

$$K = 1 - \frac{0.785}{\left(\frac{P_T}{d_o}\right)^2} = 0.4976$$

p = effective pressure $= (p_s + \cancel{p_b}^{\,0})$ or $(p_t + \cancel{p_b}^{\,0})$

$= p_s$ (or) p_t

$= p = 0.38\ N/mm^2$

i) $\frac{p}{f} = 3.77 \times 10^{-3}$ — ①

$1.6\left(1 - \frac{d_o}{P_T}\right)^2 = 64 \times 10^{-3}$ — ②

$\because$ ② > ① i.e., $1.6\left(1 - \frac{d_o}{P_T}\right)^2 > \frac{p}{f}$

shear stress won't control the tube sheet thickness.

$\Rightarrow$ only 'resist bending' will be considered.

ii) Minimum Tube-sheet thickness to resist bending:-

$$t_{ts} = \frac{FG_p}{3} \times \sqrt{\frac{p}{Kf}} = \frac{488.95}{3} \times \sqrt{\frac{0.38}{(0.4976)(100.6)}}$$

$t_{ts} = 14.2002$.

$t_{ts} \geq 19mm$ for $d_o = 25.4mm$ IS: 4503-1967 Pg. 25

VI) Impingement Plate:-

$\rho = \rho_w|_{T_c^*} = 999.44676\ kgm^{-3}$

$= 0.992 g/cm^3$

Nozzle Diameter $= D_n = 101.6 mm$ Nptel Table 2.3

$= 4 inch$ Pg. 11

linear velocity of tube fluid $u_w = \frac{\dot{m}_w}{\left(\frac{\pi D_n^2}{4}\right)\rho_w} = \frac{4.279}{922.45 \times \left(\frac{\pi}{4} \times 0.1016^2\right)}$

$u_w = 0.57225 \frac{m}{s}$

$\rho u^2 = 0.32499 \frac{g}{cm^3 s} <<< 125 \frac{g}{cm^3 s}$

$\Downarrow$

$\boxed{\text{Protection against impingement isn't required.}}$

VII) Nozzle Thickness (t_n) :-

material of nozzle = Carbon Steel

$D_n = 101.6\, mm$,

$J = 0.8$

$f = 100.6 N/mm^2$, $p = 0.38 N/mm^2$

$t_n = \frac{p D_n}{2fJ - p} + c = 0.24043 + 3$

$\boxed{t_n = 3.24043 mm}$

$P_s = P_t \Rightarrow \boxed{(t_n)_{shell} = (t_n)_{tube}}$

VIII) DESIGN of Gaskets:-

i) Flat iron jacketed, asbestos fill

$\Rightarrow$ Gasket factor $(m) = 3.75$

Maximum design seating stress $(Y) = 5.35 \frac{kgf}{mm^2}$

$Y = 52.4656 \frac{N}{mm^2}$

$$\Rightarrow \frac{D_{OG}}{D_{IG}} = \sqrt{\frac{Y - pm}{Y - p(m+1)}} \quad \ni Y \text{ in } \frac{kgf}{mm^2}, \; p \text{ in } \frac{N}{mm^2}$$

$$\Rightarrow \frac{D_{OG}}{D_{IG}} = 1.05223$$

ii) Outside Gasket Diameter (D_{OG}):-

inside Gasket Diameter (D_{IG}) $\doteq D_s + 0.25$

$$\boxed{D_{IG} = 489.2mm}$$

$$\boxed{D_{OG} = 1.05223 \times D_{IG} = 514.752mm}$$

iii) Gasket Width (N) $= \frac{D_{OG} - D_{IG}}{2} = 12.776069mm$,

using 35mm

Mean Gasket Diameter (G) $= \frac{D_{OG} + D_{IG}}{2} = 501.976mm$

Basic Gasket Seating Width (b_0) $= \frac{N}{2} = 17.5mm$

Effective gasket steading width (b) $= \frac{\sqrt{b_0}}{2} = 2.092mm$

($\because b_0 > 6mm$)

IX) BOLTS:-

i) $G = 501.976069mm$

- Bolt Load due to Gasket reaction under atmospheric conditions:-

$$W_{m_1} = \pi b G Y = 173059.8641N$$

- Bolt load under tight pressure:-

$$W_{m_2} = \underbrace{2\pi b G m p}_{\text{total joint contact surface compression load } (H_p)} + \underbrace{\frac{\pi}{4} G^2 P}_{\text{Total hydrostatic end force } (H)} = 84604.59N$$

(ii) Controlling load = max $(W_{m_1}, W_{m_2}) = W_{m_1}$

Minimum bolt cross-sectional area,

$$\Rightarrow A_m = \frac{W_{m_1}}{f_a} = \frac{173059.8641}{100.6} = 1720.277\ mm^2$$

(iii) IS 4864-1968,

Bolt details for nominal diameter G ($\lesssim$ 508mm):

$\boxed{\text{Thread nominal diameter} \doteq M16}$

$\boxed{\text{Bolt circle diameter } (C_b) = 560mm}$

["d_8 column" IS : 4864-4870 (1968)]

$\boxed{\text{no. of bolts } (n_b) = 20}$

$\boxed{\text{root diameter } (d_{br}) = 18mm}$

Actual Bolt cross-sectional area,

$$A_b = (\pi/4)\, d_{br}^2 \times n_b = 5089.38\ mm^2 > A_m \checkmark$$

iv) min. gasket width, $N_{min} = \dfrac{A_b f_b}{2\pi \underset{\downarrow}{Y} G} = 30.34213975\ mm$

(kgf/mm²) $(N = 35\ mm > 30.34 = N_{min})$

X) Flange Thickness:-

$$\text{Flange Bolt Load } (W) = \begin{cases} \left(\dfrac{A_m + A_b}{2}\right) f_a, & \text{for seating condition} \\ W_{m_2}, & \text{for operating condition} \end{cases}$$

a) Gasket seating condition (no internal load applied):-

i) $W = \left(\dfrac{A_m + A_b}{2}\right) f_a = \left[\dfrac{1720.27698 + 5089.38}{2}\right] \times 100.6$

$$W = 342525.7511 N$$

ii) (a) $M_f^o = \frac{W(C_b - G)}{2} = \frac{(342525.75)(560 - 501.976)}{2}$

↓

flange moment (gasket seating condition)

$$M_f^o = 9937345.274 \text{ N.mm}$$

b) For Operating Condition:-

B = Centre line to centre line bolt spacing

$= D_c = 508.95 mm$

i) Hydrostatic end force on area inside of flange, $H_D = \frac{\pi B^2 p}{4} = \frac{\pi D_c^2 p}{4}$

$$H_D = 77307.87 N$$

So, $h_D = \frac{C_b - B}{2} = \frac{560mm - 508.95mm}{2} = 25.525 mm$

Moment due to H_D is:-

$$M_D = H_D h_D = 1973283.417 N\text{-mm}$$

(ii) Gasket load under operating conditions:-

$H_G = W - H$

W_{m_2} ; $H = \frac{\pi G^2 p}{4} = 75203.75N$

$\Rightarrow H_G = 84604.59345 - 75203.7513$

$H_G = 9400.84N$

& $h_G = \frac{C_b - G}{2} = \frac{560 - 501.976089}{2} = 29.0119655 mm$

$\Rightarrow$ moment due to H_G is,

$$M_G = H_G h_G = 272736.9081 \text{ N-mm}$$

iii) Pressure force on the flange face, $H_T = |H - H_0|$

$$H_T = 2104.12 \text{ N}$$

$$h_T = \frac{h_D + h_G}{2} = 27.268 \text{ mm}$$

moment due to H_T:-

$$M_T = H_T h_T = 57376.16183 \text{ N-mm}$$

iv) Summation of flange moments for operating conditions:

$$M_f = M_D + M_G + M_T = 2303396.487 \text{ N-mm}$$

But under seating conditions, $M_f^\circ = 9937345.274$ N-mm

$\because M_f^\circ > M_f \Rightarrow$ controlling moment $M = M_f^\circ$

c) Flange Thickness:-

$A = 600$ "d_2 column" IS: 4864-4870 (1968)

i) $K = \frac{C_b + 2E}{B} = \frac{A}{B} = \frac{600}{508.95} = 1.1788977$

ii) for flange, $Y = fn(K)$ Fig 12.22 [Process Equipment Design, Brownell. LE.; EH. Young]

$$Y = \frac{1}{K-1}\left[0.66845 - 5.71690 \cdot \frac{K^2 \log_{10} K}{K^2 - 1}\right]$$

Graph-fitting, $Y = 14.299$

iii) $t_f = \sqrt{\frac{M_f^\circ Y}{f_{fa} B}} = \sqrt{\frac{(9937345)(14.299)}{(100.6)(508.95)}} = 52.68069$ mm

$$\boxed{t_f = 52.68069 \text{ mm}}$$

XI) SUPPORTS:-

Horizontal Heat Exchanger Unit

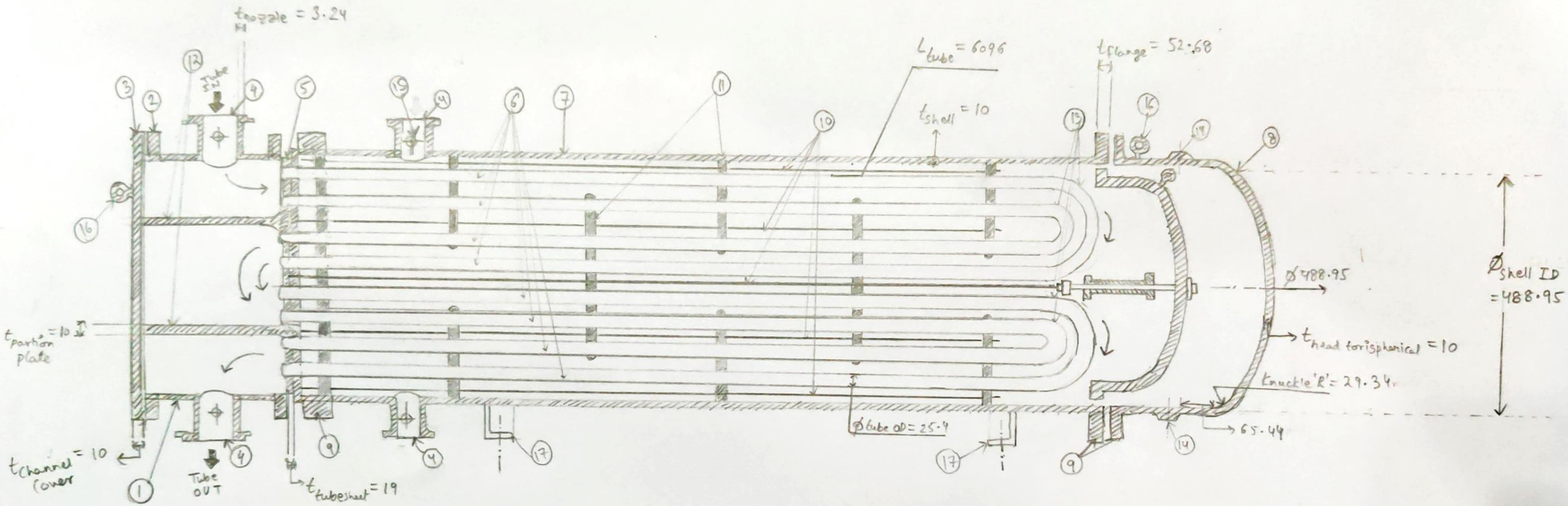
=> 2 supporting saddles with holes for anchor bolts.

=> Holes in at least one support shall be elongated to provide for expansions of shell.

2 - SHELL PASS, 4 - TUBE PASS, SHELL & TUBE HEAT EXCHANGER

(HE Type - reference by TEMA : Tubular Exchanger Manufacturers Association, © 1988)

SIDE VIEW



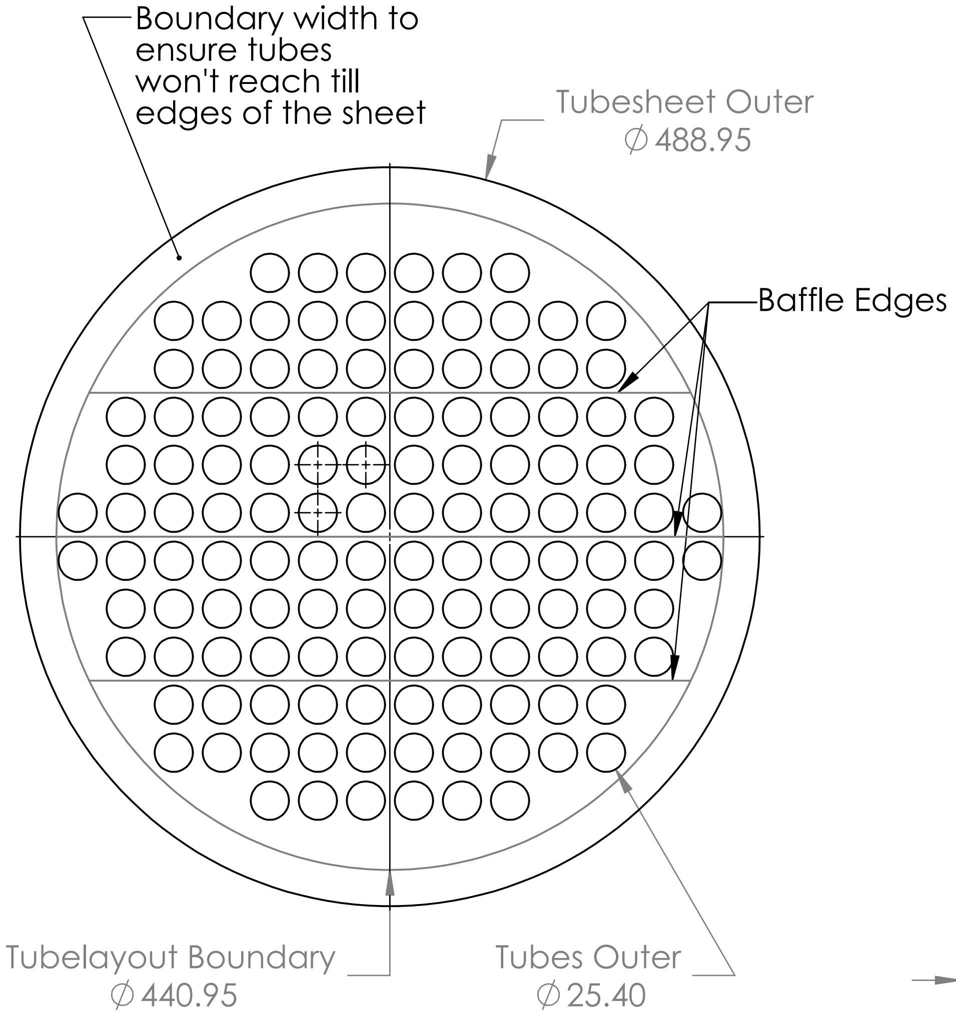
1. Stationary (Front) Head - Channel
2. Stationary (Front) Head - Flange
3. Channel Cover
4. Stationary Head Nozzle, Shell Nozzle
5. Stationary Tubesheet
6. Tubes (128, 1¼ inch Square pitch)
7. Shell
8. Shell Cover (Torispherical)
9. Flanges - Shell Stationary Head End, Shell Rear Head End, Shell Cover
10. Tie Rods and Spacers
11. Baffles or Support plates
12. Pass Partition Plates
13. U-Tube Bundles
14. Vent/Drain Connection
15. Instrument Connection
16. Lifting Lug
17. Support Saddle

Shell & Tube GEOMETRIC TERMINOLOGY

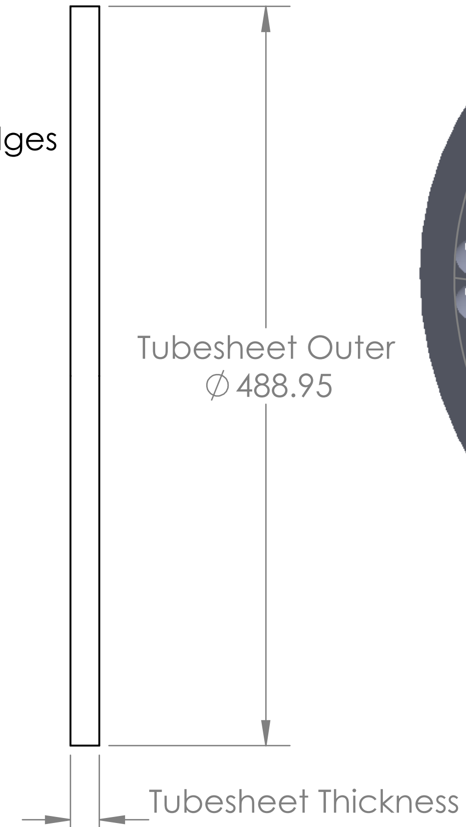
t : thickness
Ø : Diameter
R : Radius

DIMENSIONS

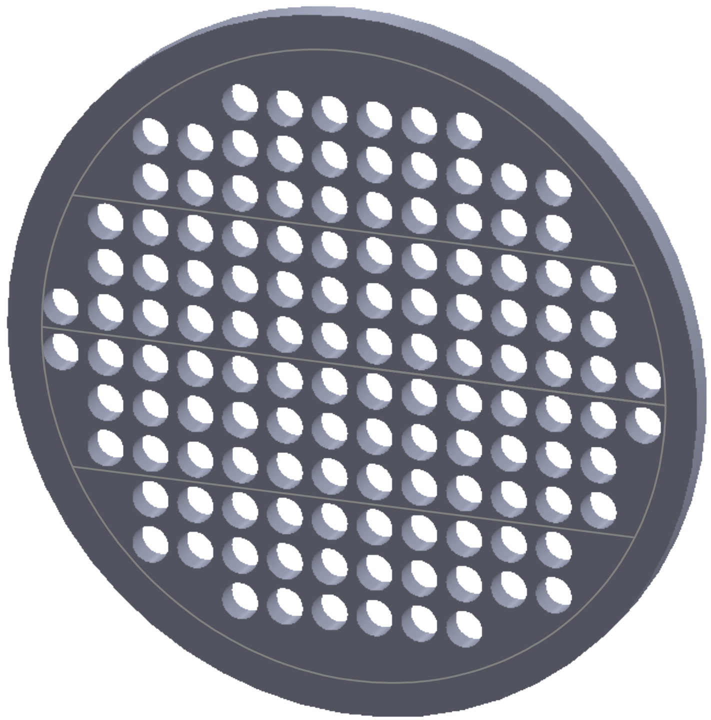
All dimensions are in 'mm'
Scale : ~1:10
Sheet : A4



Croos-sectional View



Side View



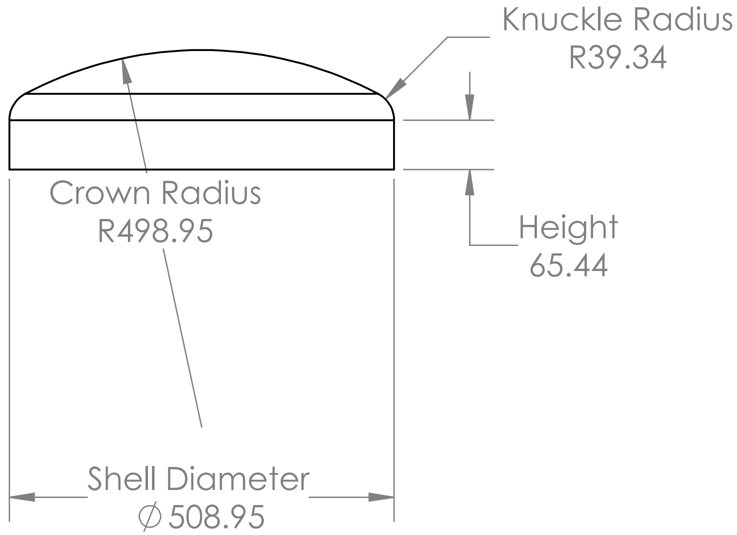
Dimetric View

Tubesheet Layout (4 pass, 128 tubes)

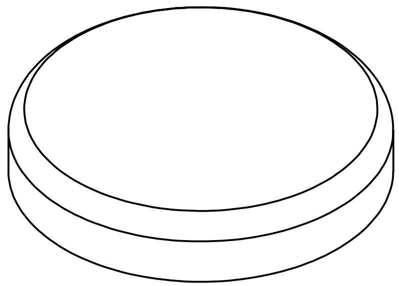
TITLE: Tubesheet

All dimensions in *mm*
Scale 1:5

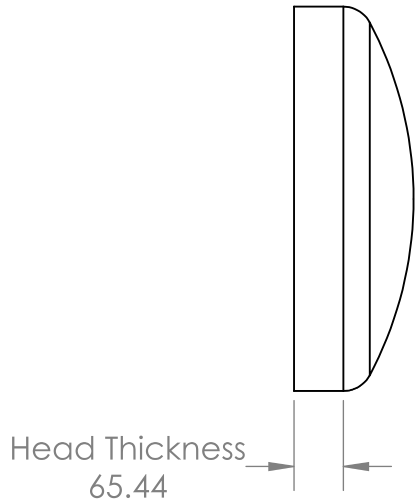
A4



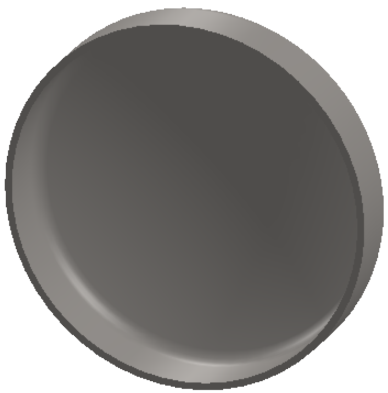
Top VIew



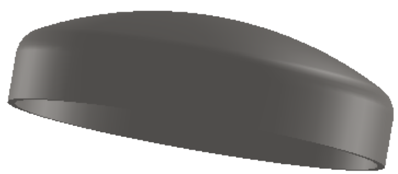
Front View



Side View



Isometric View

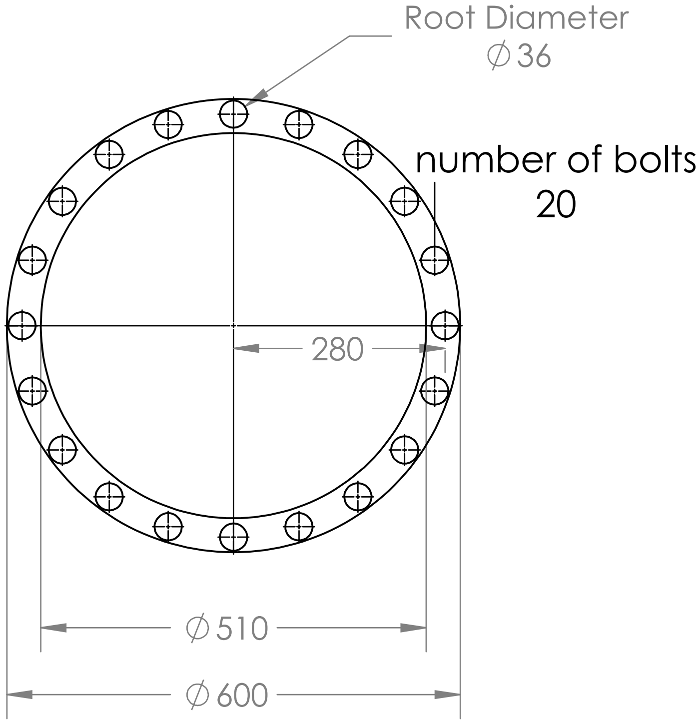


Front View - Inclined

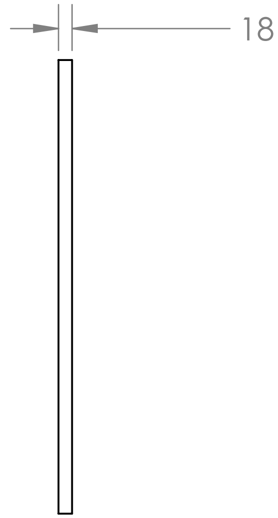
Heat Exchanger Torispherical Head

A4

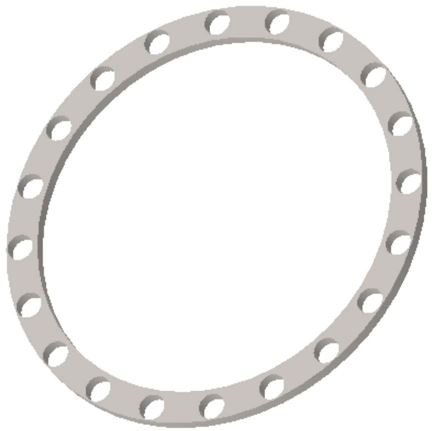
All dimensions in *mm*
Scale: 1:10



Cross Sectional View



Side View


Cross Sectional View - Inclined

TITLE:

Flange

All dimensions in *mm*
SCALE:1:10

A4