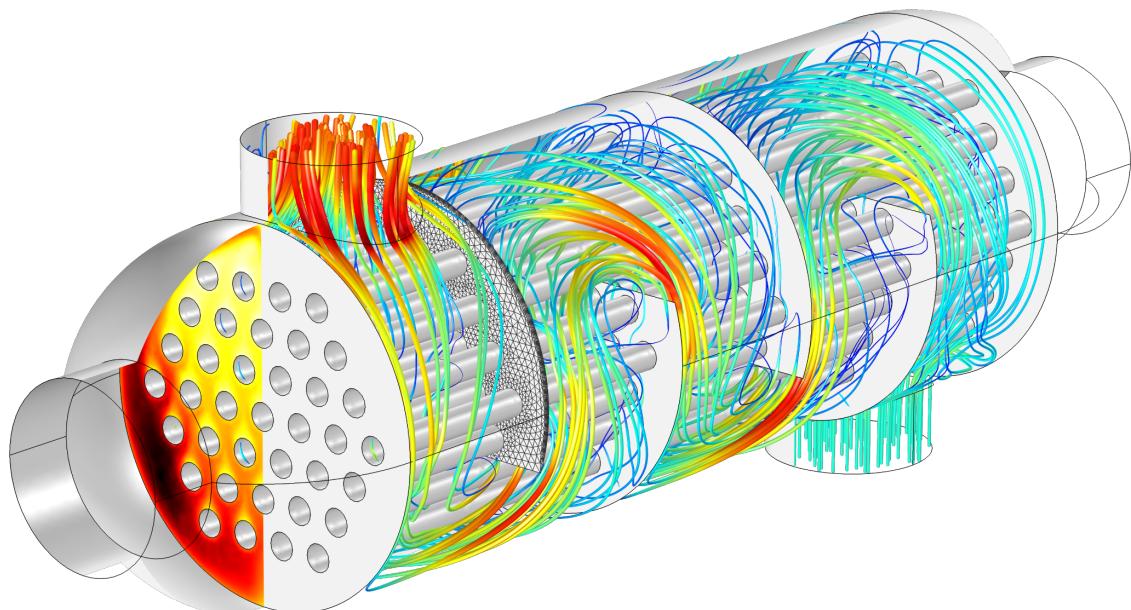
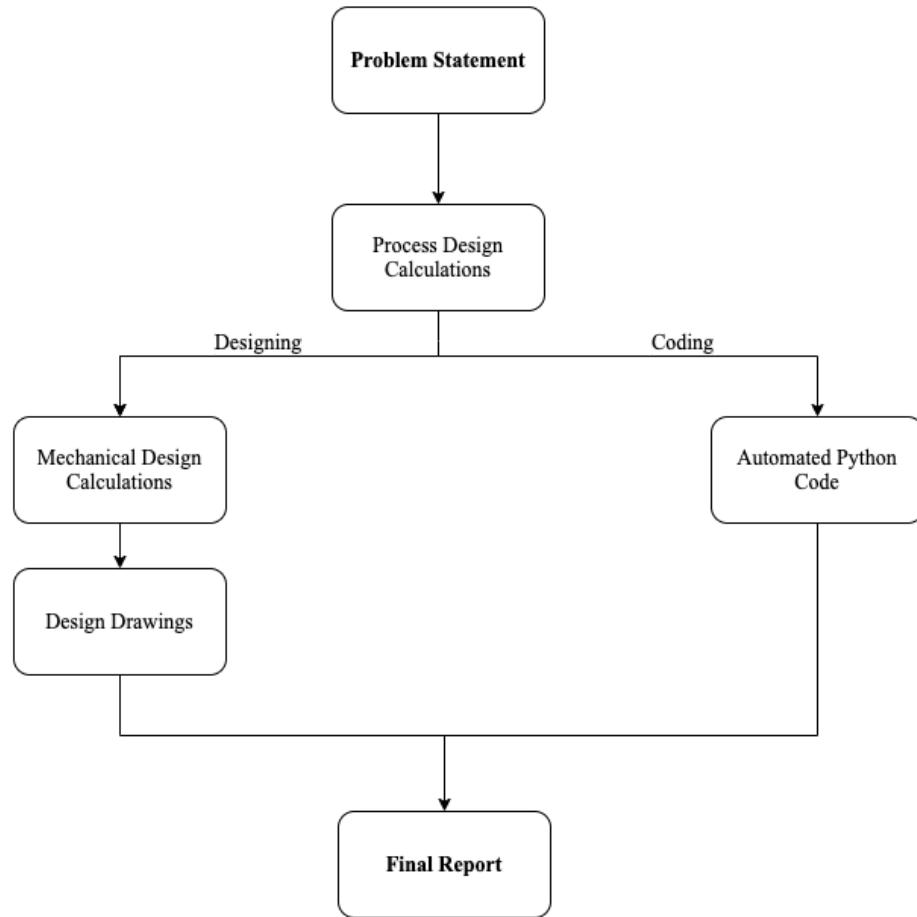


# Shell and Tube Heat Exchanger



Group - 29  
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# Flow Chart



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## **Problem Statement**

Design a Shell and Tube Heat Exchanger (stripped Heavy Naptha Trim cooler) to cool heavy naphtha of  $1.5 + 0.03 * (\text{Group Number}) \text{ kg/s}$  at  $85^{\circ}\text{C}$  to  $55^{\circ}\text{C}$  using water at  $35^{\circ}\text{C}$ , as the coolant

## Shell and Tube Heat Exchanger

### Fluid Allocation

Heavy naptha is considered on the tube side, since the viscosity of heavy naptha is more than that of water. Heavy naptha is more corrosive than water, hence chosen as the tube side liquid, as shell is more expensive.

Tube Side fluid : Heavy Naptha  
Shell Side fluid : Water

### Process requirements

- ① Design Tolerance < 5%.
- ② Tube side Pressure Drop < 10 psi
- ③ Overdesign Area < 10%.

Given, mass flow rate of heavy naptha =  $1.5 + (0.03)(29)$   
 $= 2.37 \text{ kg/s}$

For water,  $T_{c,in} = 35^\circ\text{C}$       For Heavy Naptha,  $T_{h,in} = 85^\circ\text{C}$   
 $T_{c,out} = 45^\circ\text{C}$                            $T_{h,out} = 55^\circ\text{C}$

API of heavy naptha = 47.3°  
 API of water = 10°

### Calorofic Temperatures

$$r = \frac{\Delta t_c}{\Delta T_h} = \frac{T_{h,out} - T_{h,in}}{T_{h,in} - T_{c,out}} = \frac{55 - 35}{85 - 45}$$

$$\therefore r = 0.75$$

From D.Q. Kern Pg: 827,

For water, at  $r = 0.75$ ,  $K_c = 1.24$  (By extrapolation)

For heavy naptha, at  $r = 0.75$ ,  $K_c = 0.1$

$$\therefore F_c \text{ for heavy naptha} = 0.44$$

$$F_c \text{ for water} = 0.38$$

$$T_h^* = T_{h,out} + F_c (T_{h,in} - T_{h,out}) = 55 + 0.44(85 - 55) \\ \therefore 67^\circ\text{C}$$

$$T_c^* = T_{c,in} + F_c (T_{c,out} - T_{c,in}) = 35 + 0.38(45 - 35) \\ = 38.8^\circ\text{C}$$

## Properties of Fluids at Calorific Temperatures\*

### Heavy Naptha

Specific Heat Capacity @  $T = T_h^* (C_p) = 2036.9947 \text{ J/kg K}$   
 Viscosity @  $T = T_h^* (\mu_h) = 0.733 \text{ cP}$   
 Density @  $T = T_h^* (\rho) = 791.387 \text{ kg/m}^3$   
 Thermal Conductivity @  $T = T_h^* (k_h) = 0.15 \text{ W/m K}$

### Water

Specific Heat Capacity @  $T = T_c^* (C_p) = 4178.5761 \text{ J/kg K}$   
 Viscosity @  $T = T_c^* (\mu_c) = 0.6681 \text{ cP}$   
 Density @  $T = T_c^* (\rho) = 992.6812 \text{ kg/m}^3$   
 Thermal Conductivity @  $T = T_c^* (k_c) = 0.598 \text{ W/m K}$

\* All the values have been computed using the python code using, by fitting a polynomial and finding the relation between property and temperature using data found online (Refer to "find-prop" for more, all the data references have been attached in the code)

### Enthalpy Balance

$$\dot{m}_h C_{p_h} (T_{h,in} - T_{h,out}) = \dot{m}_c C_{p_c} (T_{c,out} - T_{c,in})$$

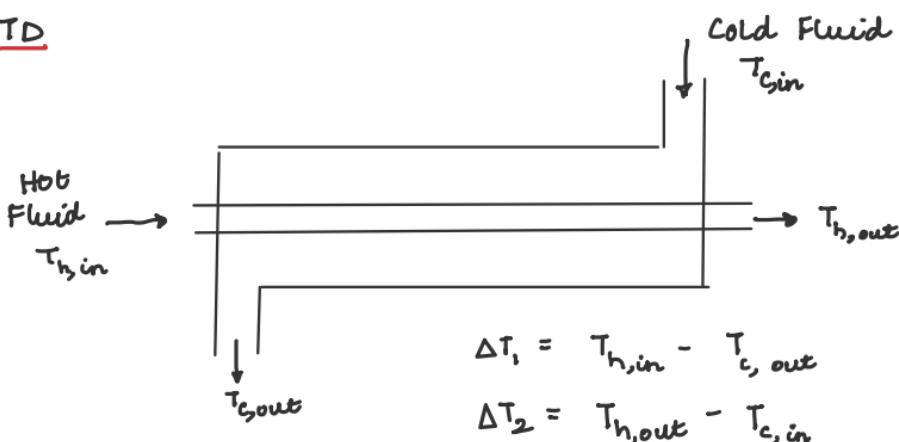
$$\Rightarrow 2.37 \times 2036.9947 \times 30 = \dot{m}_c \times 4178.5761 \times 10$$

$$\dot{m}_c = \frac{2.37 \times 2036.9947 \times 30}{4178.5761 \times 10}$$

$$\therefore \dot{m}_c = 3.466 \text{ kg/s}$$

$$\text{Heat duty : } Q = \frac{2.37 \times 2036.9947 \times 30}{144830.3232 \text{ J/s}}$$

### LMTD



$$\therefore LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{40 - 20}{\ln(2)} = 28.8539^{\circ}C$$

### LMTD Correction Factor

$$R = \frac{(T_{h,in} - T_{h,out})}{(T_{c,out} - T_{c,in})} = \frac{20}{10} = 2$$

$$S = \frac{(T_{c,out} - T_{c,in})}{(T_{h,in} - T_{h,out})} = \frac{10}{30} = \frac{1}{3}$$

For 1-2 heat exchanger :  $F_T = 0.935$  (D.Q. Kern, Pg: 828)

We proceed with this value as  $F_T > 0.9$

### Assumed U

Overall  $U_D \in (50, 125) \text{ Btu/hr ft}^2 \text{ of}$   
(Since heavy naphtha is a medium organic)

$$\text{let } U_{assm} = \left(\frac{50 + 125}{2}\right) = \frac{87.5 \text{ Btu/hr ft}^2 \text{ of}}{= 496.848 \text{ Btu/hr ft}^2 \text{ of}}$$

$$Q = UA F_T (LMTD)$$

$$A = \frac{Q}{UF_T LMTD} = \frac{144830 \cdot 3232}{496.848 \times 0.935 \times 28.8539}$$

$$\therefore A = 10.8043 \text{ m}^2$$

### Number of Tubes

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

Consider, Tube length ( $L$ ) = 20 ft = 6.096 m

Consider the standard tube diameters,

$$\begin{aligned} d_i &= \text{Tube ID} = 0.67 \text{ inch} = 0.017018 \text{ m (BWG}_8\text{)} \\ d_o &= \text{Tube OD} = 1 \text{ inch} = 0.0254 \text{ m} \end{aligned}$$

$$\therefore n_t = \frac{10.8043}{\pi \times 0.0254 \times 6.096} \approx 23 \text{ tubes}$$

Standard Number of Tubes available = 32 (D.Q. Kern, Pg: 841)

Number of Tube passes ( $N_t$ ) = 2  
Number of shell passes = 1

We select a square pitch over triangular pitch, as,

- (i) Cleaning is easier
- (ii) Lower Pressure drop
- (iii)  $Na$  isn't too high

$$\text{Shell ID} = 10 \text{ inch} = 0.254 \text{ m}$$

$$\text{Square Pitch} = 1\frac{1}{4} \text{ inch} = 0.03175 \text{ m}$$

### Tube Side Reynold's Number

$$Re_{\text{tube}} = \left( N_u \times m_h \right) / \left( \frac{\pi}{4} \times d_i \times m_h \times n_b \right)$$

$$= \frac{2 \times 4 \times 2.37 \times 1000}{0.017018 \times 0.733 \times 32} = 15112.8485$$

### Tube Side heat transfer co-efficient

$$\text{Dittus Boelter Eqn: } N_u = 0.023 Re^{0.8} Pr^{1/3} \left( \frac{m_w @ T = T_c^+}{m_w @ T = T_h^+} \right)$$

$$\frac{h_i d_i}{k_h} = 0.023 \left( \frac{4.4}{k_h} \right)^{1/3} Re^{0.8} \left( \frac{m_w @ T_c^+}{m_w @ T_h^+} \right)$$

$$\Rightarrow h_i = 0.023 \times \left( \frac{2036.9947 \times 0.733}{0.1500 \times 1000} \right)^{1/3} (15112.8485)^{0.8} \times \left( \frac{m_w}{m_w} \right)$$

$$\therefore h_i = 890.915 \text{ W/m}^2\text{K}$$

### Shell side heat transfer co-efficient,

Important Considerations,

- a) 25% Cut Segmental Baffles
- b) Baffle spacing,  $B = D_s/2$

$$D_s = 0.254 \text{ m}, P_t = 0.03175 \text{ m}$$

$$\text{Tube clearance, } (C) = P_t - d_o = 0.00635 \text{ m}$$

$$A_s = \frac{CB D_s}{P_t} = 6.415 \times 10^{-3} \text{ m}^2$$

$$D_{eq} = \frac{4 \left( C P_t^2 - \frac{\pi}{4} d_o^2 \right)}{\pi d_o} = 0.0251 \text{ m}$$

$$Re_{\text{shell}} = \frac{m_o D_{eq}}{A_s \mu_c} \quad (\text{i.e. } Re_s = \frac{v_s d_e}{\mu}, v_s = \frac{m}{A_s} \left( \frac{1}{d_e} \right))$$

$$= \frac{3.466 \times 0.0251 \times 1000}{6.415 \times 10^{-3} \times 0.6681} = 20298.50941$$

$$\therefore Re_{\text{shell}} = 20298.50941$$

$R_{do} \equiv$  Tube Outside Dirt Factor,  $R_{do} = 0.00075 \text{ m}^2\text{K/W}$

$R_{di} \equiv$  Tube Inside Dirt Factor,  $R_{di} = 0.00018 \text{ m}^2\text{K/W}$

Tube material : Carbon steel,  $k_{tube} = 54 \text{ W/mK}$

Colbourn Factor,  $j_H = 79.6518$  (At  $Re = Re_{shell}$ )

$$j_H = \frac{h_0 D_e}{k_c} (Pr)^{-1/3} \left( \frac{\mu_{\infty}}{\mu_{wall}} \right)^{-0.14}$$

$$\Rightarrow h_0 = 1134 \cdot 0369 \text{ W/m}^2\text{K}$$

$$\text{Using, } \frac{1}{U_{cal}} = \left[ \frac{1}{h_0} + R_{do} + \frac{A_o}{A_i} \left( \frac{d_o - d_i}{2k_{wall}} + R_{di} + \frac{1}{h_i} \right) \right]$$

$$\text{Substituting and evaluating, } U_{cal} = 244.7841 \text{ W/m}^2\text{K}$$

$$\begin{aligned} \text{Relative Error} &= \left| \frac{U_{cal} - U_{assm}}{U_{assm}} \right| \times 100 \\ &= \left| \frac{244.7841 - 496.848}{496.848} \right| \times 100 \\ &= 50.7326\%. \end{aligned}$$

But the tolerance is 5%. since our rel error > tolerance  
we perform further iterations

### Iteration - 1

$$U_{assm} = U_{cal} = 244.7841 \text{ W/m}^2\text{K}$$

$$A = Q / (U_{assm} \times F_T \times LMTD) = 21.93 \text{ m}^2$$

$$n_t = A / \pi d_{oL} = 44$$

$$\text{Standard Number of Tubes available (n_t)} = 45$$

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

$$D_s = 0.3048 \text{ m}$$

$$R_{tube} = 10746.9145$$

$$h_i = \frac{k_n}{d_i} \times 0.023 \times (R_{tube})^{0.8} (Pr)^{1/3} \left( \frac{\mu_{\infty}}{\mu_{wall}} \right)^{0.14}$$

$$\therefore h_i = 678.24 \text{ W/m}^2\text{K}$$

$$B = D_s/2 = 0.1524 \text{ m}, \quad A_s = \frac{\pi B D_s}{P_T} = 0.00929 \text{ m}^2$$

$$Re_{shell} = 14034.8241$$

Colbourn Factor,  $j_H = 65.54$  (at  $Re = Re_{shell}$ )

$$j_H = \frac{h_0 D_e}{k_c} (Pr)^{-1/3} \left( \frac{\mu_{\infty}}{\mu_{wall}} \right)^{0.14} \Rightarrow h_0 = 1185.542 \text{ W/m}^2\text{K}$$

$$\therefore U_{cal} = 214.0128 \text{ W/m}^2\text{K}$$

Relative error = 12.5708% ( $> 5\%$ , hence we re-iterate)

### Iteration - 2

$$U_{assm} = U_{cal} = 214.0128 \text{ W/m}^2\text{K}$$

$$A = \dot{Q}/(U_{assm} \times F_T \times LMTD) = 25.083 \text{ m}^2$$

$$n_t = A/\pi d_o L = 52$$

Standard Number of Tubes available ( $n_t$ ) = 56

$$D_s = 0.3365 \text{ m}$$

$$\text{On solving, } Re_{tube} = 8635.91$$

$$h_i = \frac{k_b}{d_i} \times 0.023 \times (Re_t)^{0.8} \times (Pr)^{1/6} \times \left(\frac{\mu_0}{\mu_{wall}}\right)^{0.14}$$

$$\therefore h_i = 569.3847 \text{ W/m}^2\text{K}$$

$$a_s = \frac{C_B D_s}{P_t} = 0.0113 \text{ m}^2 \Rightarrow Re_{shell} = 11511.6535$$

Colburn Factor,  $j_H = 58.95$  (At  $Re = Re_{shell}$ )

$$\therefore h_o = 1066.4834 \text{ W/m}^2\text{K}$$

$$\text{On solving, } U_{cal} = 189.83 \text{ W/m}^2\text{K}$$

Relative error = 11.3% (> 5%, hence we re-iterate)

### Iteration - 3

$$U_{assm} = U_{cal} = 189.83 \text{ W/m}^2\text{K}$$

$$A = \dot{Q}/(U_{assm} \times F_T \times LMTD) = 28.2787 \text{ m}^2$$

$$n_t = A/\pi d_o L = 59$$

Standard Number of Tubes available ( $n_t$ ) = 56

$$D_s = 0.3878 \text{ m}$$

$$\text{On solving, } Re_{tube} = 6363.3046$$

$$h_i = 445.9697 \text{ W/m}^2\text{K}$$

$$a_s = \frac{C_B D_s}{P_t} = 0.15 \text{ m}^2 \Rightarrow Re_{shell} = 8690.2$$

Colburn Factor,  $j_H = 50.74$  (At  $Re = Re_{shell}$ )

$$\therefore h_o = 918.0437 \text{ W/m}^2\text{K}$$

$$\text{On solving, } U_{cal} = 159.1453 \text{ W/m}^2\text{K}$$

Relative error : 16.1636% (> 5%, hence we re-iterate)

### Iteration - 4

$$U_{asm} = U_{cal} = 159.83 \text{ W/m}^2$$

$$A = Q / (U_{asm} \times F_T \times LMTD) = 33.7308 \text{ m}^2$$

$$h_t = A / \pi d_o L = 70$$

Standard Number of Tubes available ( $n_t$ ) = 76

$$D_s = 0.3873 \text{ m}$$

$$\text{On solving, } Re_{tube} = 6363.3046, h_i = 445.9697 \text{ W/m}^2\text{K}$$

$$Re_{shell} = 8690.2, h_i = 918.0437 \text{ W/m}^2\text{K}$$

$$\therefore U_{cal} = 159.1453 \text{ W/m}^2\text{K}$$

Relative error = 0%.

$\therefore$  The overall heat transfer coefficient of 1-2 shell and tube heat exchanger is  $U_d = 159.1453 \text{ W/m}^2\text{K}$

### Pressure Drop

$$n_t |_{final} = 76$$

$$Re_{tube} = 6363.3046$$

From moody diagram,  $f = 0.045$

$$\text{Tube side velocity, } v_t = \frac{N_t \dot{m}}{n_t (\rho A)} = 0.3465 \text{ m/s}$$

Tube side pressure drop,

$$\Delta P_f = \frac{1}{2} f \rho u^2 \left( \frac{L}{d_t} \right) = \frac{1}{2} \times 0.045 \times 791.39 \times 0.35^2 \times \left( \frac{20}{1} \right)$$

$$= 759.182 \text{ Pa}$$

$$\Delta P_r = \frac{4 N_t}{S} \frac{v^2}{2g} = \frac{4 \times 2}{791.39} \times \frac{0.35^2}{2 \times 9.81} \times 1000$$

$$= 0.06187 \text{ Pa}$$

$$\Delta P_{total} = \Delta P_f + \Delta P_r$$

$$= 759.2446 \text{ Pa}$$

$$\therefore \Delta P_{total} = 0.1101 \text{ psi} (< 10 \text{ psi})$$

### Overdesign

$$\% \text{ Overdesign} = \left| \frac{A - A_{eq}}{A_{eq}} \right| \times 100 \\ = \left| \frac{70 - 76}{76} \right| \times 100 = 8.5714\%$$

$$\% \text{ Overdesign} = 8.5714\% (< 10\%)$$

Hence, the above design is acceptable and cost-effective.

### Mechanical Design

Shell side passes = 1, Tube side passes = 2

No. of tubes = 76

Tube dimensions,  $L = 20 \text{ ft} (= 6.096 \text{ m})$

$d_i = 17.02 \text{ mm} (= 0.017 \text{ m})$

$d_o = 25.4 \text{ mm} (= 0.0254 \text{ m})$

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

Pitch : Square Pitch

Shell diameter, ID = 387.3 mm

Shell and Head : Carbon Steel Material, Torispherical Head

Carbon Steel Corrosion Allowance in Petroleum/chemical industries where severe condns. are expected ( $C_c$ ) = 3 mm

Design Temperature =  $1.1 \times T_{max}$

$$T_{max} = T_{h,i} = 85^\circ C = 185^\circ F$$

$$T_{design} = 1.1 \times T_{max} = 1.1 \times 185^\circ F = 203.5^\circ F$$

$$\therefore T_{design} = 203.5^\circ F$$

Design Pressure =  $1.1 \times P_{operating}$

$$P_{op} = 50 \text{ psia} \Rightarrow P_{design} = 1.1 \times 50 = 55 \text{ psia} \\ = 379.212 \text{ kPa} = 0.38 \text{ N/mm}^2$$

Permissible Stress,  $f = 100.6 \text{ N/mm}^2$  (Carbon steel)

## Shell Thickness Calculations

$$P = 0.38 \text{ N/mm}^2$$

$$D_s = 387.3 \text{ mm}$$

$$f = 100.6 \text{ N/mm}^2$$

$$J = 0.8 \text{ (Joint efficiency)}$$

$$t_s = \frac{PD_s}{fJ - 0.6P} + C = \frac{0.38 \times 387.3}{100.6 \times 0.8 - 0.6 \times 0.38} + 3 = 4.833 \text{ mm}$$

$$\therefore t_s = 5 \text{ mm} \quad [ \text{IS: 4503-1967, Pg: 22, Chemical / Petroleum Ind.} ]$$

## Shell Cover

$$\text{Crown Radius, } R_i = D_s = 387.3 \text{ mm}$$

$$\text{Knuckle Radius, } r_i = 6\% R_i = 23.238 \text{ mm}$$

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

$$h_i = 387.3 - \sqrt{(387.3)^2 - (387.3/2)^2 + 2(23.238)} \\ = 51.8191 \text{ mm}$$

$$\begin{aligned} \text{Effective exchanger length} &= L_b + 2h_i \\ &= 6.096 + 2(51.8191/1000) \\ &= 6.1996 \text{ m} \end{aligned}$$

$$\therefore L_{eff} = 6.1996 \text{ m}$$

$$W = \frac{1}{4} \left( 3 + \sqrt{\frac{R_i}{r_i}} \right) = 1.771$$

For head design,  $J = 1$

$$t_h = \frac{PR_iW}{2fJ - 0.2P} + C = \frac{387.3 \times 0.38}{2 \times 1 \times 100.6 - 0.2 \times 0.38} + 3 \\ = 3.731 \text{ mm}$$

Standard available thickness,  $t_h = 5 \text{ mm}$

$$D_c = \text{Outside Shell dia} = D_s + 2t_h = 397.3 \text{ mm}$$

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

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

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

$$t_{co} = \frac{D_c}{10} \sqrt{\frac{C_1 P}{f}} + C = \frac{397.3}{10} \sqrt{\frac{0.3 \times 3.87}{10.258}} + 3 \\ = 7.1782 \text{ mm}$$

Standard available thickness,  $t_{cc} = 10 \text{ mm}$

### Thickness of Pass Partition Plate

Since,  $D_c < 600 \text{ mm}$

Thickness of the pass partition plate = 10 mm  
(including corrosion allowance)  
 $\therefore t_{ppp} = 10 \text{ mm}$

### Tube Sheet Thickness

$F = 1$  (Fixed tube sheet)  $\rightarrow$  supported tube sheets  
TEMA Pg: 55

$$G_{tp} = D_s = 387.3 \text{ mm}$$

$$f = 100.6 \text{ N/mm}^2$$

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

$$P = \text{Effective Pressure} = (P_s + P_b)^\circ \text{ or } (P_s + P_t)^\circ \\ = P_s \text{ or } P_t = 0.38 \text{ N/mm}^2$$

$$\frac{P}{f} = 3.77 \times 10^{-3} - ①$$

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

$$\Rightarrow 1.6 \left(1 - \frac{d_o}{P_t}\right)^2 > \frac{P}{f}$$

Shear stress won't control the tube sheet thickness.  
 → Only 'Resist Bending' will be considered.

Min. Tube-sheet thickness for resist bending,

$$t_{ts} = \frac{FG_{IP}}{3} \sqrt{\frac{P}{kf}} = \frac{387.3}{3} \sqrt{\frac{0.38}{0.4976 \times 100.6}}$$

$$t_{ts} = 11.248 \text{ mm}$$

Shell dia  $\leq 500$  mm

$$T_{\text{design}} = \max(t_{ts}, D_s/10)$$

$$\therefore T_{\text{design}} = \max(11.248, 38.73) \\ = 38.73 \text{ mm}$$

$$\text{Corrosion Allowance} = 38.73 + 3 = 41.73 \text{ mm}$$

Standard available Thickness = 45 mm

### Impingement Plate

$$\rho = \rho_n l_{T_c} = 791.387 \text{ kg/m}^3 \\ = 0.791 \text{ g/cm}^3$$

$$\text{Nozzle Diameter} = D_n = 76.2 \text{ mm} \quad (\text{Nptel Table 2.3}) \\ = 3 \text{ inch} \quad P_g : 11$$

$$\text{Linear Velocity of} \quad u_{\text{tube}} = \frac{m_n}{\left(\frac{\pi D_n^2}{4}\right) P_n} = \frac{2.37}{\left(\frac{\pi \times 0.076^2}{4}\right) 791.387} \\ \text{Tube Fluid} \quad = 0.6567 \text{ m/s}$$

$$\rho u^2 = 0.3413 \text{ g/cm}^3 \text{ s} \quad << 125 \text{ g/cm}^3 \text{ s}$$

∴ Protection against impingement isn't required

### Nozzle Thickness ( $t_n$ ):

$$D_n = 76.2 \text{ mm} \\ J = 0.8 \\ f = 100.6 \text{ N/mm}^2, \quad P = 0.38 \text{ N/mm}^2$$

$$t_n = \frac{PD_n}{2fJ - P} + C = 0.18032 + 3$$

$\therefore t_n = 3.18032 \text{ mm}$ , Standard available thickness = 4mm

$$P_g = P_t \Rightarrow (t_n)_{\text{shell}} = (t_n)_{\text{tube}}$$

### Design of Gaskets

Flat iron Jacketed, asbestos fill

$$\Rightarrow \text{Gasket factor } (m) = 3.75$$

Maximum design seating stress ( $\gamma$ ) = 5.35 kgf/mm<sup>2</sup>

$$\gamma = 52.4656 \text{ N/mm}^2$$

$$\frac{D_{OG}}{D_{IG}} = \sqrt{\frac{4 - P_m}{4 - P(m+1)}} = 1.05223$$

Outside Gasket Diameter ( $D_{OG}$ )

$$\begin{aligned} \text{Inside Gasket Diameter } (D_{IG}) &= D_s + 0.25 \\ &= 387.55 \text{ mm} \end{aligned}$$

$$D_{OG} = 1.05223 \times D_{IG} = 407.792 \text{ mm}$$

$$\text{Gasket Width } (N) = \frac{D_{OG} - D_{IG}}{2} = 10.1209 \text{ mm} \quad (\text{Using } 35 \text{ mm})$$

$$\text{Mean Gasket Diameter } (r_i) = \frac{D_{OG} + D_{IG}}{2} = 397.671 \text{ mm}$$

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

$$\text{Eff. Gasket seating width } (b) = \frac{\sqrt{b_0}}{2} = 2.092 \text{ mm} \quad (b_0 > 6 \text{ mm})$$

### Bolts

$$G_i = 397.671 \text{ mm}$$

Bolt load due to gasket  $r_n$  under atm. conditions,

$$W_{m_1} = \chi_b G_i \gamma = 137122.9406 \text{ N}$$

Bolt load under tight pressure

$$W_{m_2} = \underbrace{2\pi b G_i m P}_{\text{Total joint contact surface compression load } (H_p)} + \underbrace{\frac{\pi}{4} G_i^2 P}_{\text{Total hydrostatic end force } (H_t)} = 54646.45 \text{ N}$$

Total joint contact surface compression load ( $H_p$ )      Total hydrostatic end force ( $H_t$ )

Controlling load = max (  $w_{m_1}, w_{m_2} \right) = w_{m_1}$

Min. bolt cross-sectional area,

$$A_m = \frac{w_{m_1}}{f_a} = \frac{137122.9406}{100.6} = 1363.05 \text{ mm}^2$$

Bolt details for nominal diameter G.C.  $\leq 508 \text{ mm}$ )

Thread nominal diameter = M16

Bolt Circle diameter ( $C_b$ ) = 470 mm

[ $d_3$  - column IS : 4864 - 4870 (R68)]

No. of bolts ( $n_b$ ) = 16

Root diameter ( $d_{br}$ ) = 18 mm

Actual Bolt cross sectional area,

$$A_b = (\pi/4) d_{br}^2 \times n_b = 4071.50 \text{ mm}^2 > A_m$$

Min. Bolt cross-sectional area,

$$N_{min} = \frac{A_b f_b}{2\pi G} = \frac{4071.50 \times 100.6}{2 \times \pi \times 52.46 \times 397.67} \times \frac{1}{0.101} = 30.94 \text{ mm} \\ \hookrightarrow \text{in kgf/mm}^2 \quad (N = 35 \text{ mm} \\ > 30.94 \text{ mm})$$

### Flange Thickness

Flange Bolt load ( $w$ ) =  $\left\{ \begin{array}{l} \left( \frac{A_m + A_b}{2} \right) f_a, \text{ for seating cond.} \\ w_{m_2}, \text{ for operating cond.} \end{array} \right.$

Gasket Seating cond. (no internal load applied)

$$w = \left( \frac{A_m + A_b}{2} \right) f_a = \left( \frac{4071.50 + 1816.48}{2} \right) \times 100.6 \\ = 296165.39 \text{ N}$$

Flange Moment (Gasket Seating cond.)

$$M_F^o = \frac{w(C_b - G)}{2} = \frac{296165.39 (470 - 397.67)}{2}$$

$$\therefore M_F^o = 10710678.25 \text{ Nmm}$$

For operating cond.,

$$B = \text{Center line to center line bolt spacing} \\ = D_c = 397.3 \text{ mm}$$

Hydrostatic end force  
on area inside of flange,  $H_p = \frac{\pi B^2 P}{4} = 47109.73 \text{ N}$

$$\text{So, } h_p = \frac{C_b - B}{2} = \frac{470 - 397.3}{2} = 36.35 \text{ mm}$$

Moment due to  $H_p$ ,  
 $M_p = H_p h_p = 1712433.686 \text{ N-mm}$

Gasket load under operating cond.,

$$H_G = W - H$$

$$H = \frac{\pi G_1^2 P}{4} = 47183.27 \text{ N}, \quad W = W_{m_2} = 54646.45 \text{ N} \\ \therefore H_G = 7468.18 \text{ N}$$

$$h_G = \frac{C_b - G}{2} = \frac{470 - 397.61}{2} = 36.195 \text{ mm}$$

Moment due to  $H_G$  is,  
 $M_G = H_G h_G = 270129.86 \text{ N-mm}$

Pressure force on flange face,  $H_T = |H - H_G| = 39646.55 \text{ N}$   
 $h_T = \frac{h_p + h_G}{2} = 36.273 \text{ mm}$

Moment due to  $H_T$ ,  
 $M_T = H_T h_T = 1438679.485 \text{ N-mm}$

Summation of flange moments for operating cond.,

$$M_F = M_p + M_G + M_T = 3420642.971 \text{ N-mm}$$

Under seating cond.,  $M_F^* = 10710673.25 \text{ N-mm}$

Since,  $M_F^* > M_F \Rightarrow$  Controlling moment  $M = M_F^*$

Flange Thickness

$$A = 510 \quad \text{"d<sub>2</sub> column"} \quad \text{IS: 4864 - 4870 (1968)}$$

$$K = \frac{C_p + 2E}{B} = \frac{A}{B} = \frac{510}{397.3} = 1.284$$

For flange,  $y = f(K)$

Fig 12.2  
[P&D Brownwell L.T.; G.H. Young]

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

On graph-fitting,  $y = 8$

$$t_F = \sqrt{\frac{M_F Y}{f_k B}} = \sqrt{\frac{167106.73 \cdot 25 \times 8}{(100 \cdot 6)(397.3)}} = 46.3 \text{ mm}$$

$$\therefore t_F = 46.3 \text{ mm}$$

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.

# Code Output

## **Design parameters of a Shell & Tube Heat Exchanger**

Hot Fluid: Heavy Naptha

Cold Fluid: Water

Enter the mass flow rate of Heavy Naptha in kg/s(if you want to enter in  
1.5+0.03\*G format enter 'G'): G

Group Number: 29

-----

Mass Flow Rate of Heavy Naptha: 2.37 kg/s

Please input the following values in degree Celsius.

Initial Temperature of the Hot Fluid: 85

Desired Final Temperature of the Hot Fluid: 55

Coolant initial temperature: 35

Coolant final temperature: 45

-----

Caloric Fraction of Hot Fluid: 0.4348

Caloric Temperature Of Hot Fluid: 68.0429 C

Caloric Fraction of Cold Fluid: 0.3774

Caloric Temperature Of Cold Fluid: 38.7737 C

-----

## **Properties of the Fluids at Tc**

Hot Fluid: Heavy Naptha

Specific heat Capacity of Hot Fluid: 2036.9947 J/kg ·K

Viscosity of Hot Fluid: 0.7333032907813091 cP

Density of Hot Fluid: 791.3870 kg/m<sup>3</sup>

Thermal Conductivity of Hot Fluid: 0.1500 W/m ·K

Cold Fluid: Water

Specific Heat Capacity of Water: 4178.5761 J/kg ·K

Viscosity of Cold Fluid: 0.6681 cP

Density of Cold Fluid: 992.6812 kg/m<sup>3</sup>

Thermal Conductivity of Cold Fluid: 0.5980 W/m ·K

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## **Enthalpy Heat Balance**

Heat Rate: 144830.3237 J/kg·s  
Mass Flow Rate of Water: 3.4660 kg/s

---

## **LMTD and LMTD Correction Factor**

LMTD considering counter-current flow: 28.8539 C  
R: 3.0 S: 0.2  
LMTD correction Factor for 1 shell pass and 2 or more tube passes  
Ft = 0.9350  
LMTD correction Factor for 2 shell pass and 4 or more tube passes  
Ft = 0.9847  
1-2 Heat Exchanger is the better option. As Ft>=0.9. The code will proceed calculating parameters for 1-2 HE.

---

## **Shell Side and Tube Side Parameters**

### Iteration 0

U(assumed): 496.8480 W/m<sup>2</sup> ·K  
Area: 10.8043 m<sup>2</sup>  
Required number of Tubes: 23  
Standard Number of tubes: 32  
Ds\_sq: 0.2540 m

Tube Side Fluid: Heavy Naptha  
Tube Side Reynolds Number (Re|tube): 15112.8485  
Tube Side Heat Transfer Coefficient (ho): 890.9150 W/m<sup>2</sup> ·K

Shell Side Fluid: Water  
Shell Side Reynolds Number (Re|shell): 20210.1467  
Shell Side Heat Transfer Coefficient (ho): 1134.0369 W/m<sup>2</sup> ·K

Ucal: 244.7841 W/m<sup>2</sup> ·K  
Relative error = 50.7326 %  
Since the relative error is >5% we proceed to do further iterations

### Iteration 1

Required number of Tubes: 46  
Standard Number of tubes: 45  
Ds\_sq: 0.3048 m

Tube Side Reynolds Number (Re|tube): 10746.9145

Tube Side Heat Transfer Coefficient (hi): 678.2444 W/m<sup>2</sup> ·K  
Shell Side Reynolds Number (Re|shell): 14034.8241  
Shell Side Heat Transfer Coefficient (ho): 1185.5418 W/m<sup>2</sup> ·K  
Ucal: 214.0128 W/m<sup>2</sup> ·K  
Relative error = 12.5708 %

#### Iteration 2

Required number of Tubes: 52  
Standard Number of tubes: 56  
Ds\_sq: 0.3365 m

Tube Side Reynolds Number (Re|tube): 8635.9134  
Tube Side Heat Transfer Coefficient (hi): 569.3847 W/m<sup>2</sup> ·K  
Shell Side Reynolds Number (Re|shell): 11511.6535  
Shell Side Heat Transfer Coefficient (ho): 1066.4834 W/m<sup>2</sup> ·K  
Ucal: 189.8284 W/m<sup>2</sup> ·K  
Relative error = 11.3004 %

#### Iteration 3

Required number of Tubes: 59  
Standard Number of tubes: 76  
Ds\_sq: 0.3873 m

Tube Side Reynolds Number (Re|tube): 6363.3046  
Tube Side Heat Transfer Coefficient (hi): 445.9697 W/m<sup>2</sup> ·K  
Shell Side Reynolds Number (Re|shell): 8690.2002  
Shell Side Heat Transfer Coefficient (ho): 918.0437 W/m<sup>2</sup> ·K  
Ucal: 159.1453 W/m<sup>2</sup> ·K  
Relative error = 16.1636 %

#### Iteration 4

Required number of Tubes: 70  
Standard Number of tubes: 76  
Ds\_sq: 0.3873 m

Tube Side Reynolds Number (Re|tube): 6363.3046  
Tube Side Heat Transfer Coefficient (hi): 445.9697 W/m<sup>2</sup> ·K  
Shell Side Reynolds Number (Re|shell): 8690.2002  
Shell Side Heat Transfer Coefficient (ho): 918.0437 W/m<sup>2</sup> ·K  
Ucal: 159.1453 W/m<sup>2</sup> ·K  
Relative error = 0.0000 %

-----

#### **Pressure Drop and Overdesign**

Tube side velocity: 0.3465 m/s

Friction Factor: 0.04462  
delta Pf:759.1828 Pa , delta\_Pr: 0.061871 Pa  
Tube Side Pressure Drop: 0.1101 psi ( 759.2446 Pa )  
Required Number of Tubes: 70  
Standard Number of Tubes: 76  
%Overdesign: 8.5714 %

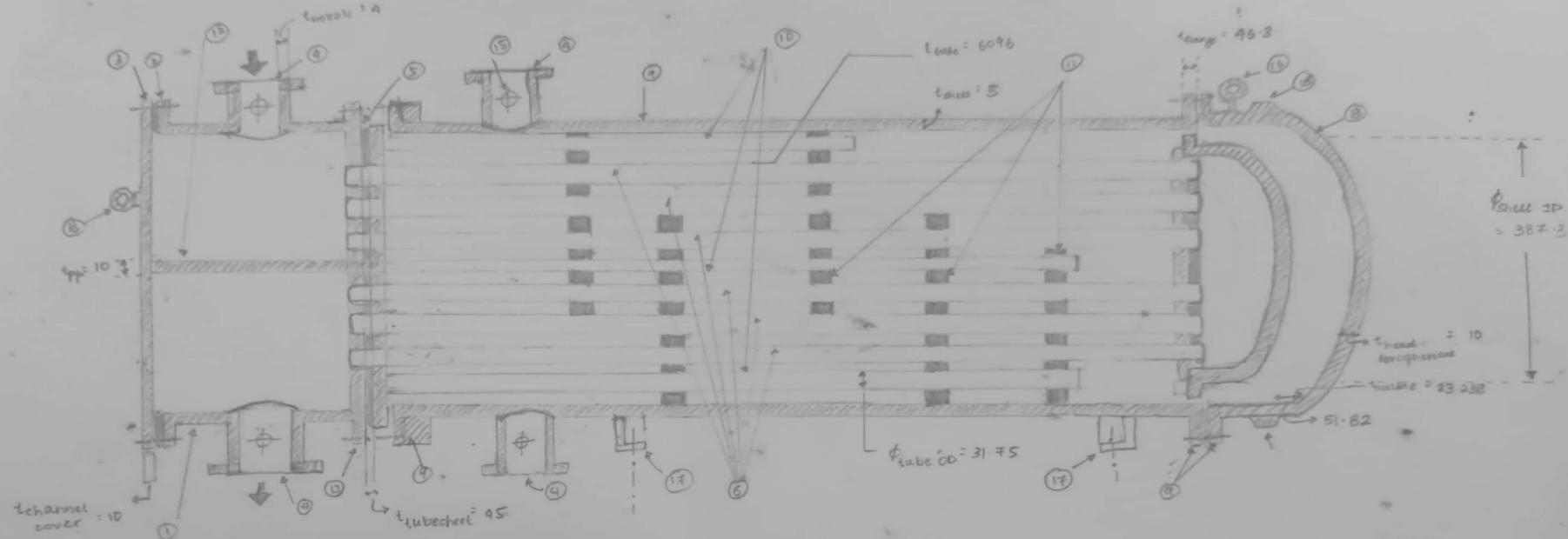
**Design is acceptable and cost-effective**

## References

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1 - SHELL PASS , 2 - TUBE PASS, SHELL & TUBE HEAT EXCHANGER  
 CHE Type - reference by TEMA : Tubular Exchanger Manufacturers Association, (C) 1988)

SIDE VIEW



- ① Stationary (Front) Head - Channel
- ② stationary (Front) Head - Flange
- ③ channel cover
- ④ stationary Head Nozzle, shell nozzle
- ⑤ stationary tube sheet
- ⑥ Tubes ( $\frac{7}{16}$ ,  $1\frac{1}{4}$  square pitch)
- ⑦ shell
- ⑧ shell cover (Torispherical)
- ⑨ Flanges - shell stationary Head End, shell rear Head End, shell cover

- ⑩ Rods and Spacers
  - ⑪ Rafflers or support plates
  - ⑫ Pass Partition Plate
  - ⑬ Channel fixed tip
  - ⑭ Vent/Drain connection
  - ⑮ Instrument connection
  - ⑯ Lifting Lug
  - ⑰ Support saddle
- Shell & Tube  
GEOMETRIC  
TECHNOLOGY
- $t$ : thickness      } DIMENSIONS  
 $\phi$ : diameter      }  
 $R$ : Radius

All dimensions are in "mm"

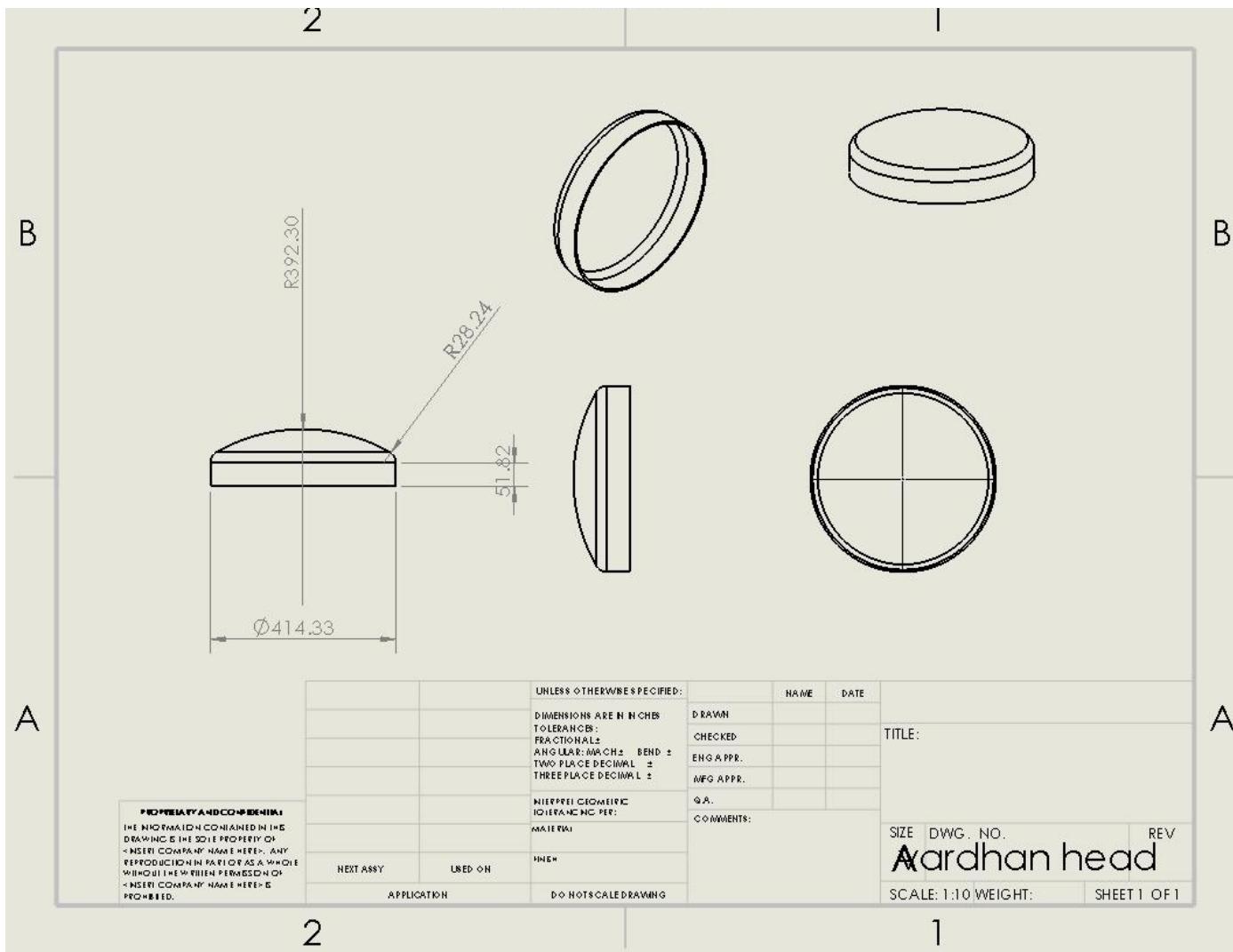
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Sheet : A4

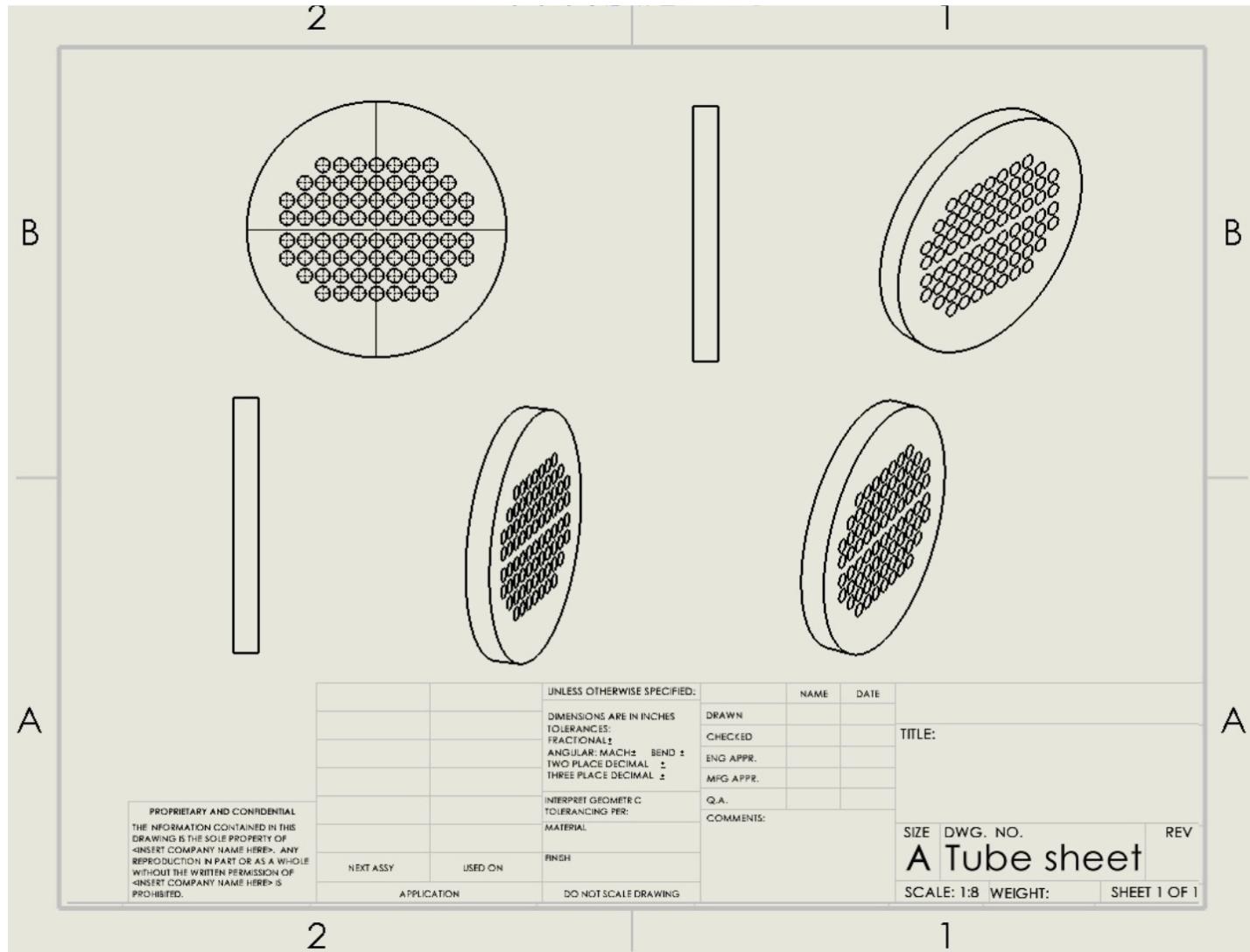
## SOLID WORKS DESIGN FILES

TORISPHERICAL HEAD  
TUBESHEET LAYOUT  
SHELL AND NOZZLE  
FLANGE

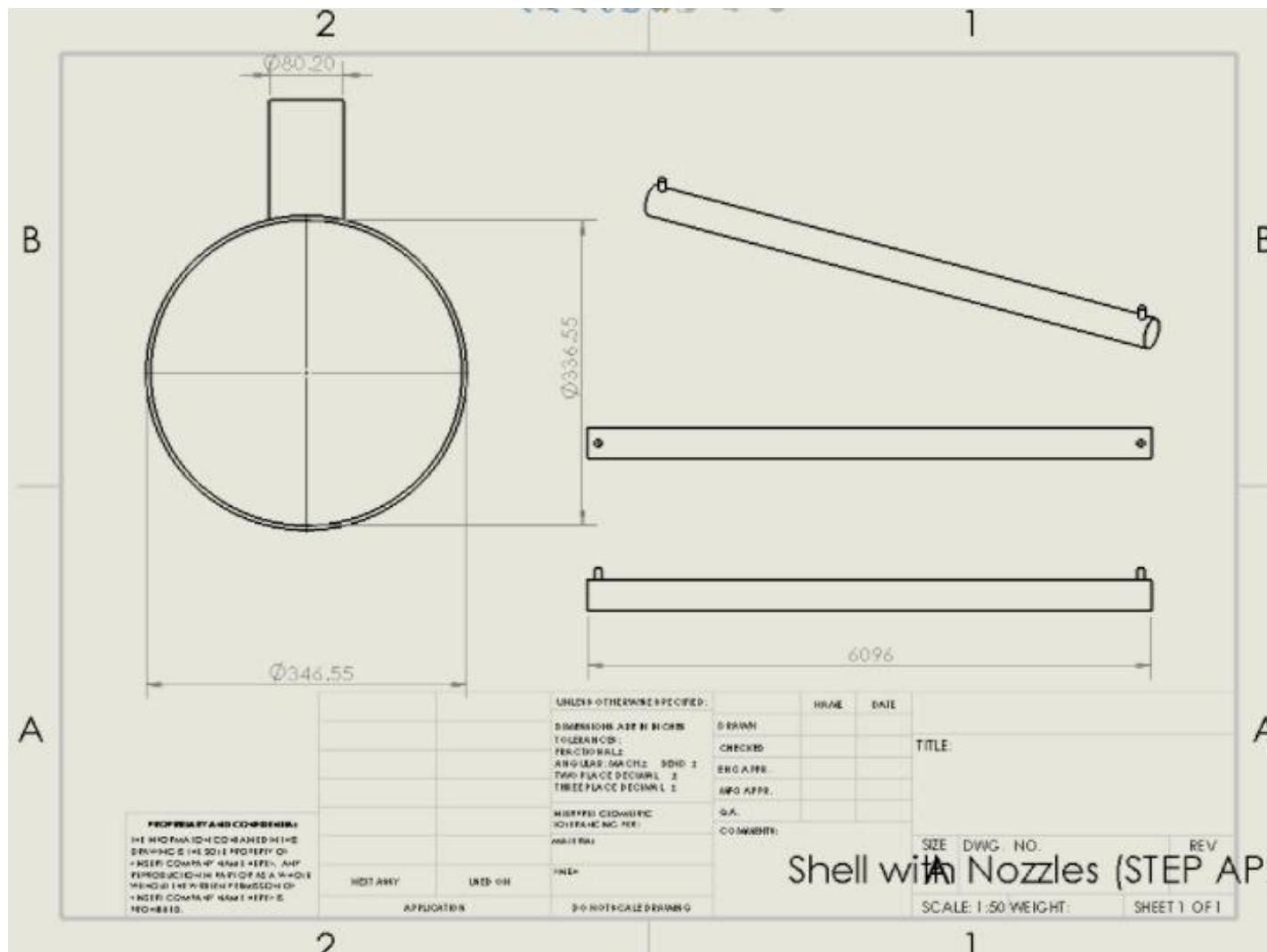
# TORISPHERICAL HEAD



# TUBESHEET LAYOUT



# SHELL AND NOZZLE



# FLANGE

