DESIGN OF A SHELL AND TUBE HEAT EXCHANGER

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ABSTRACT

Often, in process industries the feed stream has to be preheated before being sent to the process unit (reactor or distillation column) and the product stream has to be cooled before being sent for storage. In order to minimize heat losses and maximize energy efficiency, the heat removed from the product stream is provided to the feed stream as preheat. This is done with the help of heat exchangers in process industries. Many different types of heat exchangers are available depending on the need of operation; however, shell and tube heat exchangers are by far the most commonly used heat exchangers.

In this work, the design of a counter current shell and tube heat exchanger used in nitric acid plants has been presented. The design work has been done considering the desired capacity of the plant, which is 100 tons/day of nitric acid. Two different methods, Kern's method and Bell's method, have been used for the design. Bell's method was found to be more accurate as the overall heat transfer coefficient calculated by Bell's method was close to the assumed value. Further, the design of auxiliary parts of the heat exchanger such as flanges, gaskets, bolts, supports and saddles have also been presented.

Keywords: *heat exchanger, design, process industries*

NOMENCLATURE

R_e Reynolds number

Pr Prandtl number

U_o Overall heat transfer coefficient

D_b Bundle diameter

D_s Shell diameter

d_i Tube diameter

h_i Tube side heat transfer coefficient

A_s Cross flow area

G_s Mass velocity

d_e Equivalent diameter

k Thermal conductivity of equipment material

Nu Nusselt number

h_s Shell side heat transfer coefficient

ΔP_s	Shell side pressure drop
ΔP_t	Total pressure drop
ΔP_c	Cross flow pressure drop
f	Allowable stress
r_o	Knuckle radius

1. INTRODUCTION

Heat exchangers are typically used in process industries for the transfer of heat from a hot fluid to a cold fluid. This helps in minimizing energy losses and maximizing energy efficiency, which further results in higher economic returns. Many different types of heat exchangers are available such as shell and tube heat exchangers, plate heat exchangers, spiral heat exchangers, etc. But, shell and tube heat exchangers are the most commonly used because of their versatility, robust nature and reliability.

A shell and tube heat exchanger is an indirect contact heat exchanger in which the hot and cold fluid streams do not come in direct contact but the heat transfer occurs through a dividing wall. There are three different classifications of shell and tube heat exchangers such as parallel flow, cross flow and counter-current flow. Out of these three different types, counter current flow provides the highest heat transfer rate for a given amount of time. In this work, the design of a counter current shell and tube heat exchanger used in nitric acid manufacturing plants has been presented. The design involves a large number of operating and geometric variables in order to match the heat duty requirement for the corresponding plant capacity, which is 100 tons per day of nitric acid. Two different iterative procedures, Kern's method and Bell's method, have been used in the design work.

2. METHODOLOGY

A reference geometric configuration of the equipment is chosen at first and a desired overall heat transfer coefficient is chosen. The values of the design parameters are evaluated based on the initial assumptions and the assumed design specifications. An iterative procedure is followed to arrive at a satisfactory heat transfer coefficient and a reasonable design specification.

3. CALCULATIONS

3.1 Kern's Method

Heat capacity of nitric oxide = 1.066 kJ/kg.K

Nitric oxide flow rate = 0.827 kg/s

Inlet temperature of nitric oxide = 150 °C

Outlet temperature of nitric oxide = 50 °C

Heat load = 0.827 x 1.066 x (150 - 50) KW

= 88.1582 KW

Heat capacity of water = 4.195 kJ/kgK

Water flow rate = 0.744 kg/s

ΔT (water) = 88.1582/(0.744x4.195)

= 28.246 °C

LMTD = ((150-63.246)-(50-35))/ ln(86.754/15)

= 40.88

R = $T_1 - T_2/(t_2 - t_1)$

= 100/28.246

= 3.54

S = $t_2 - t_1/(T_1 - t_1)$

= 28.246/115

= 0.2456

From graph, F_t = 0.8

ΔT_m = 0.8 x 40.88

= 32.704 °C

We assume the value of U (overall heat transfer coefficient) to be 300 W/m^2 K

Total area of heat transfer A = Q/(UxΔT_m) = 8.98 m^2

We take o.d. 20 mm, i.d. 16 mm and length 4.83 m of tube

Area of 1 tube = 3.14 x d_o x L = 0.303 m^2

No. of tubes= A/(Area of 1 tube)

= 29.62 tubes

≅ 30 tubes

We use 1.25 triangular pitch

Bundle diameter $D_b = 20 * (30/K_1)^{1/n_1}$

$= 20 * (30/0.249)^{1/2.207}$

= 175.349 mm

We use split ring floating head type

Bundle diametrical clearance = 50 mm

Shell diameter D_s = 175.349 + 50

= 225.349 mm

Tube side coefficient

Mean water temperature = (35 + 63.246)/2

= 49.123 ℃

Tube cross sectional area = $\pi/4 \times d_i^2$

$= 3.14/4 \times (16^2)$

$= 201\ mm^2$

Tubes per pass = 30/2 = 15 tubes

Total flow area = $15 \times 201 \times 10^{-6}$

$= 3.015 \times 10^{-3}$

$= 0.003015\ m^2$

Water mass velocity = 0.744/(0.003015)

$= 246.766\ Kg/sm^2$

Density of water = $995\ kg/m^3$

Water linear velocity = 246.766/995

= 0.248 m/sec

$h_i = 4200 \times (1.35+0.02t) \times u_t^{0.8}/(d_i^{0.2})$

$= 4200 \times (1.35+ 0.02x49.1) \times 0.248^{0.8}/(16^{0.2})$

$= 1844.169\ W/m^2K$

Shell side coefficient

Baffle spacing = $D_s/5$ = 225.349/5

= 45.0698 mm

Tube pitch = 1.25 x 20 = 25 mm

Cross flow area $A_s = ((25-20)/25)x(225.349x45.0698x10^{-6})$

$= 2.03 \times 10^{-3}\ m^2$

Mass velocity, Gs = 0.827/0.00203

$= 407.389\ kg/sm^2$

Equivalent diameter, de = $1.1/20*(25^2 - 0.917 \times 20^2)$

= 14.4 mm

Mean shell side temperature = (150 + 50)/2 = 100 ℃

Viscosity = 24×10^{-6} Pa.s

$= 0.024 \times 10^{-3}$ Pas

k = 0.04 W/m.K

$R_e = G_s d_e/\mu = 407.389 \times 14.4 \times 10^{-3}/0.024 \times 10^{-3} = 244433.4$

$Pr = \mu C_p/k = 0.024 \times 10^{-3} \times 1.066 \times 10^3/ (0.04)$

$= 6.396 \times 10^{-4} \times 10^3$

$= 0.6396$

From graph, $j_h = 2.8 \times 10^{-3}$ (Page no. 665, Coulson & Richardson [1])

$Nu = h_s d_e/k_f = j_h R_e Pr^{1/3}$

$h_s = 0.04/0.0144 \times (2.8 \times 10^{-3} \times 244433.4 \times 0.6396^{1/3})$

$= 1638.01\ W/m^2K$

Estimate wall temperature

Mean temperature difference across all resistances = 100 - 49.123

$= 50.877\ ^{\circ}C$

Film temperature difference = $U/h_o \times \Delta T$

$= 300/1638.01 \times 50.877$

$= 9.318\ ^{\circ}C$

Mean wall temperature = 100 – 9.318

$= 90.681\ ^{\circ}C$

Overall coefficient

Thermal conductivity of cupro-nickel alloy = 50 W/m.K

Fouling coefficient for water = $3000 W/m^2K$

Fouling coefficient for nitric oxide = $5000\ W/m^2K$

$1/U_0 = 1/h_o + 1/h_{od} + (d_o/d_i) \times 1/h_{id} + (do/di) \times 1/h_i$

$1/U_o = 1/1638.01 + 1/5000 + (20\times10^{-3} \times \ln(20/10))/2\times50 + (20/16)\times(1/3000) + (20/16) \times (1/1844.169)$

$U_o = 513.05\ W/m^2K$

This value is well above the assumed value of $300W/m^2K$. Thus, the design is satisfactory.

2nd iteration

We assume length to be 2.39 m

Area of 1 tube= $3.14 \times d_o \times L = 3.14 \times 20 \times 10^{-3} \times 2.39$

= 0.1539 m

No. of tubes = 60 tubes

Bundle diameter, $Db = 20 \times (60/0.249)^{1/2.207}$

= 240.049 mm

Using fixed tube type

Bundle diametrical clearance = 11 mm

Shell diameter, D_s = 240.049 + 11

= 251.049 mm

Tube side coefficient

Mean water temp. = (35+63.246)/2

= 49.123 ℃

Tube cross sectional area = 3.14/4 x (16^2)

= 201 mm^2

Tubes per pass = 60/2 =30 tubes

Total flow area = 30x 201 x10^{-6}

= 6.03 x10^{-3}

= 0.00603 m^2

Water mass velocity = 0.744/(0.00603)

= 123.38 Kg/m^2.s

Density of water = 995 kg/m^3

Water linear velocity = 123.38/995

= 0.124 m/sec

h_i = 4200 x (1.35 + 0.02t) x $u_t^{0.8}$/($d_i^{0.2}$)

= 4200 x (1.35+ 0.02 x 49.1) x $0.124^{0.8}$/($16^{0.2}$)

= 1059.197 W/m^2K

Shell side coefficient

Baffle spacing = D_s/5 = 251.049/5

= 50.2098 mm

Tube pitch = 1.25 x 20 = 25 mm

Cross flow area A_s = ((25-20)/25) x (251.049 x 50.2098 x 10^{-6})

= 2.521 x 10^{-3} m^2

Mass velocity, Gs = 0.827/0.002521

= 328.044 kg/sm^2

Equivalent diameter, de = $1.1/20 \times (25^2 - 0.917 \times 20^2)$

$= 14.4$ mm

Mean shell side temperature = (150 + 50)/2 = 100 ^oC

Viscosity = 24×10^{-6} Pa.s

$= 0.024 \times 10^{-3}$ Pas

k = 0.04 W/mK

$R_e = G_s d_e/\mu = 328.044 \times 14.4 \times 10^{-3}/0.024 \times 10^{-3}$

$= 196826.4$

$Pr = \mu C_p/k = 0.024 \times 10^{-3} \times 1.066 \times 10^3/(0.04)$

$= 6.396 \times 10^{-4} \times 10^3$

$= 0.6396$

From graph, $j_h = 3.4 \times 10^{-3}$ (Page no. 665, Coulson & Richardson [1])

$Nu = h_s d_e/k_f = j_h R_e Pr^{1/3}$

$h_s = 0.04/0.0144 \times (3.4 \times 10^{-3} \times 196826.4 \times 0.6396^{1/3})$

= 1601.63 W/m^2K

Estimate wall temperature

Mean temperature difference across all resistances = 100 - 49.123

= 50.877 ℃

Film temperature difference = $U/h_o \times \Delta T$

= 300/1601.63 x 50.877

= 9.529 ℃

Mean wall temperature = 100 – 9.529

= 90.471℃

Overall coefficient

Thermal conductivity of cupro-nickel alloy = 50 W/mK

Fouling coefficient for water = 3000 W/m^2K

Fouling coefficient for nitric oxide = 5000 W/m^2K

$1/U_0 = 1/h_o + 1/h_{od} + (d_o/d_i) \times 1/h_{id} + (do/di) \times 1/h_i$

$1/U_o = 1/1601.63 + 1/5000 + (20 \times 10^{-3} \times \ln(20/10))/2 \times 50 + (20/16) \times (1/3000) + (20/16) \times (1/1059.107)$

U_o = 405.62 W/m^2K

This value is well above the assumed value of 300W/m^2K

Thus, the design is satisfactory

Pressure drop

$R_e = \rho u d_i/\mu$ = 995x0.124x16x10^{-3}/0.001

= 1974.08

$Pr = \mu C_p/k$

= 0.001x4.195x1000/0.59

= 7.11

$j_F = 3.9x10^{-2}$

$\Delta P_t = N_p(8j_F(L/d_i) + 2.5)\rho u_t^2/2$

$\Delta P_t = 2 \text{ x } (8 \text{ x } 3.9 \text{ x } 10^{-2} \text{ x } (2.39 \text{ x } 10^3/16) + 2.5) \text{ x } 995 \text{ x } 0.124^2/2$

= 751.26 N/m^2 = 0.751 kPa

As this value is low we could consider increasing the number of passes to 6

Tubes per pass = 60/6 = 10

Total flow area = 10 x 201 x 10^{-6}

= 201 x 10^{-5} m^2

Water mass velocity = 0.744/201 x 10^{-5}

= 370.149 Kg/m^2s

Water linear velocity = 370.149/995 = 0.372 m/s

R_e = 16x10^{-3} x 0.372 x 995/0.001 = 5922.24

j_F = 1.5 x 10^{-2}

$\Delta P_t = 6 \text{ x } (8 \text{ x } 1.5 \text{ x } 10^{-2} \text{ x } 2.39\text{x}10^3/16 + 2.5) \text{ x } 995 \text{ x } 0.372^2/2$

= 8437.08N/m^2

= 8.4 kPa

$1/U_o$ = 1/1601.63 + 1/5000 +20x10^{-3} x ln(20/10)/2 x 50 + 20/16 x 1/3000 + 20/16 x 1/2550.18

U_o = 563.27 W/m^2C

h_i = 4200 x (1.35 + (0.02 x 491.23)) x $0.372^{0.8}/16^{0.2}$

= 2550.7849 W/m^2 K

Shell side

Re = 196826.4

$j_F = 2.9 \times 10^{-2}$

$\Delta P_s = 8 \times j_{Fx}(Ds/de) \times (L/l_B) \times (\rho u_s^2/2)$

$\Delta P_s = 8 \times 2.9 \times 10^{-2} \times (251.049/14.4) \times (2.39 \times 10^3/50.2098) \times 1.249 \times 26.264^2/2$

= 82.93 kPa

This value is acceptable

3.2 Bell's Method

No of tubes = 60

Bundle diameter, D_b = 240.049 mm

Tube outer diameter, d_o = 20 mm

Tube length = 2.39 m or 2390 mm

Tube pitch = 25 mm

Baffle cut = 25 %

Shell diameter, D_s = 251.049 mm

Gas velocity in shell = 26.264 m/sec

For heat transfer coefficient (h_{oc})

$A_s = ((P_t - d_o) \times D_s \times l_b) / P_t)$

$= ((25-20) \times 251.049 \times 100.418) / 25)$

$= 5041.96\ mm^2$ or $5.041 \times 10^{-3}\ m^2$

Density of nitric oxide, $\rho = 1.249\ kg/m^3$

Viscosity of nitric oxide, $\mu = 0.024 \times 10^{-3}\ N\ sec/\ m^2$

$G_s = 0.827 / (5.041 \times 10^{-3}) = 164.05\ kg/\ m^2\ sec$

Reynolds number, $R_e = G_s d_o / \mu$

$= (164.05 \times 20)/ 0.024$

$= 1.367 \times 10^5$

$j_h = 3.4 \times 10^{-3}$ (from the graph) (Page no. 694, Coulson and Richardson [1])

Prandtl number, Pr = 0. 6396

Thermal conductivity, k_f = 0.04 W/m.K

$h_{oc}.do= k_f.j_h.R_e.Pr^{0.33}$

$h_{oc} = (3.4 \times 10^{-3} \times 136712.29 \times (0.6396^{0.33}) \times 0.04) / (20 \times 10^{-3})$

$h_{oc} = 801.09$ W/ m^2 k

Tube row correction factor (F_n)

$P'_t = 0.87 \times P_t = 0.87 \times 25$ (for equilateral triangular pitch)

= 21.8 mm

Baffle cut height, $H_c = 0.25 \times 251.049$

= 62.762 mm

Height between the baffle tips = $D_b - 2 \times H_c$

= 251.049 – 2 x 62.726

= 125.524 mm

$N_{cv} = 125.524 / 21.8 = 5.757$

So, $F_n = 0.956$ (from the graph) (Page no. 695, Coulson and Richardson [1])

Window correction factor, F_w

$H_b = (D_b/2) - D_s (0.5- 0.25)$

= (240.049/2) – 251.049 x 0.25
= 57.26 mm

Bundle cut = 57.26/240.049 = 0.23 (23 %)

$R_a' = 0.18$ (from the graph) (Page no. 704, Coulson and Richardson [1])

Tubes in one window area, $N_w = 60 \times 0.18 = 10.8$

Tubes in cross-flow area, $N_c = 60 - (2 \times 10.8) = 38.4$

$R_w = (2 \times 10.8)/ 60 = 0.36$

$F_w = 1.02$ (from the graph) (Page no. 696, Coulson and Richardson [1])

Bypass correction, F_b

$A_b = l_b(D_s - D_b)$

$= 5.52 \times 10^{-4}\ m^2$

$A_b/A_s = (5.52 \times 10^{-4}/2.52 \times 10^{-3})$

= 0.2191

$F_b = \exp(-a \times (A_b/A_b)(1- (2N_s/N_{cv})^{0.33})$ $(N_s/N_{cv} = 1/5)$

$F_b = \exp(-1.35 \times 0.2191(1- 0.4^{0.33})$

= 0.925

Leakage correction, F_L

Tube to baffle = 0.4 mm

Baffle to shell = 2.4 mm

$A_{tb} = (C_t \times \pi \times d_o (N_t – N_w)) / 2)$

$= (0.4 \times 3.14 \times 20(60 - 10.8))/2$

$= 6.179 \times 10^{-4}\ m^2$

$A_{sb} = ((C_s \times D_s \times (2\pi - \Theta_b)/2)$

For 25% cut, $\Theta_b = 2.1$

$= (2.4 \times 251.049(6.28 - 2.1))/2$

$= 1.259 \times 10^{-3}\ m^2$

$A_L = A_{tb} + A_{sb}$

$= 6.179 \times 10^{-4} + 1.259 \times 10^{-3}$

$= 1.8769 \times 10^{-3}\ m^2$

Now, $A_L / A_s = (1.8769 \times 10^{-3} / 5.04 \times 10^{-3}) = 0.37$

From graph, $\beta_L = 0.26$ (Page no. 698, Coulson and Richardson [1])

$F_L = 1 – 0.26((6.179 \times 10^{-4} + 2 \times 1.259 \times 10^{-3})/ 1.876 \times 10^{-3})$

$= 0.56$

Shell- side coeeficient

$h_s = 801.09 \times 0.856 \times 1.02 \times 0.925 \times 0.56 = 359.77\ W/m^2.K$

This value is appreciably lower than predicted by Kern’s method.

Pressure drop

Cross-flow zone

Reynolds number, $R_e = 1.367 \times 10^5$ so, $j_f = 3.4 \times 10^{-2}$

$u_s = 26.2624$ m/ sec

$\Delta P_i = 8 j_f N_{cv} ((\rho \mu_s^2)/2)$

$= (8 \times 3.4 \times 10^{-2} \times 5.757 \times 1.249 \times (26.264^2))/2$

$= 674.55\ N/m^2$

$F_b' = \exp(-4 \times 0.219 \times (1 – 0.4^{0.333})$

$= 0.524$

From graph, β_L' = 0.47 (Page no. 701, Coulson and Richardson [1])

$F_L' = 1 - 0.47((6.179\times10^{-4} + 2 \times 1.259 \times 10^{-3})/ 1.876\times10^{-3})$

$= 0.214$

$\Delta P_c = 674.55 \times 0.524 \times 0.214 = 74.76\ N/m^2$

Window zone

For 25% baffle cut, R_a= 0.19 (from the graph)

$A_w = (\pi D_s^2 R_a/4) - (N_w \pi d_o^2/4)$

$= 9400.268 - 3391.2$

$= 6.009 \times 10^{-3}\ m^2$

$u_w = W_s / A_w \rho$

$= 0.827/(6.009\times10^{-3} \times 1.249)$

$= 110.18$ m/ sec

$u_z = (u_w u_s)^{0.5}$

$= (110.18 \times 26.264)^{0.5}$

$= 53.79$ m/sec

$N_{wv} = H_b / P_t' = 57.26/21.8 = 3$

$\Delta P_w = ((F_L'(2 + 0.6 N_{wv})\rho U_z^2) / 2)$

$= (0.217 (2 + 0.6x3)1.249 \times 2893.36)/2$

$= 140.6245 \times 2893.36$

$= 1489.97\ N / m^2$

End zone

$\Delta P_e = \Delta P_i((N_{wv} + N_{cv})/N_{cv})F_b'$

$= 674.55 \times (1.52) \times 0.524$

$= 537.59\ N/ m^2$

Total pressure

$N_b = 2390/100.418 = 23$

$\Delta P_s = 2\Delta P_e + \Delta P_c(N_b - 1) + N_b\Delta P_w$

$= (2 \times 537.55) + (74.76 \times 22) + (23 \times 1489.97)$

$= 36.989$ kPa

In case of fouled condition = $\Delta P_s = 1.55 \times 36.989 = 57.33$ kPa

3.3 Mechanical Design

Shell side

Material of construction = Cupro – nickel alloy

Permissible stress = 110 N/mm^2

Fluid: Liquid mixture from oxidation tower

Working pressure = 2.448 bar

= 2.448×10^5 N/m^2

= 0.2448 N/mm^2

Design Pressure, P_d = 0.26928 N/mm^2

Inlet temperature = 150 $^\circ$C

Outlet temperature = 50 $^\circ$C

Shell diameter, D_s = 251.049 mm

Length = 2.39 m

Shell thickness $t_s = P_d D_s/(2fJ + P_d)$

= 0.26928 x 251.049/(2 x 110 x 0.85 + 0.26928)

= 0.36 mm

Corrosion allowance = 3 mm

Minimum thickness of 8 mm is chosen

Flanges, gasket, bolts

Flange material used – ASTM A-201 grade B

Allowable stress f = 100 N/mm^2

Material used for bolts – 5 % Cr Mo steel

Allowable stress f_b = 138 N/mm^2

Gasket material – Asbestos composition

Tube side

Tube and tube sheet material – Cupro-nickel alloy

Number of tubes = 60

Tube outer diameter = 20 mm

Tube inner diameter = 16 mm

Length = 2.39 m

Working pressure = 1 atms

= 0.1013 N/mm^2

Design pressure = 0.1114 N/mm^2

Inlet temperature = 35 ℃

Outlet temperature = 63.246 ℃

Tube thickness

$t_h = P_d Do/(2fJ + P_d)$

$= 0.1114 \times 20/(2 \times 110 \times 0.85 + 0.1114)$

= 0.0119 mm

Saddle support

Material of construction – Carbon steel

Density – 7800 kg/m^3

r_o – knuckle radius = 6 % of shell diameter

= 15.06 mm

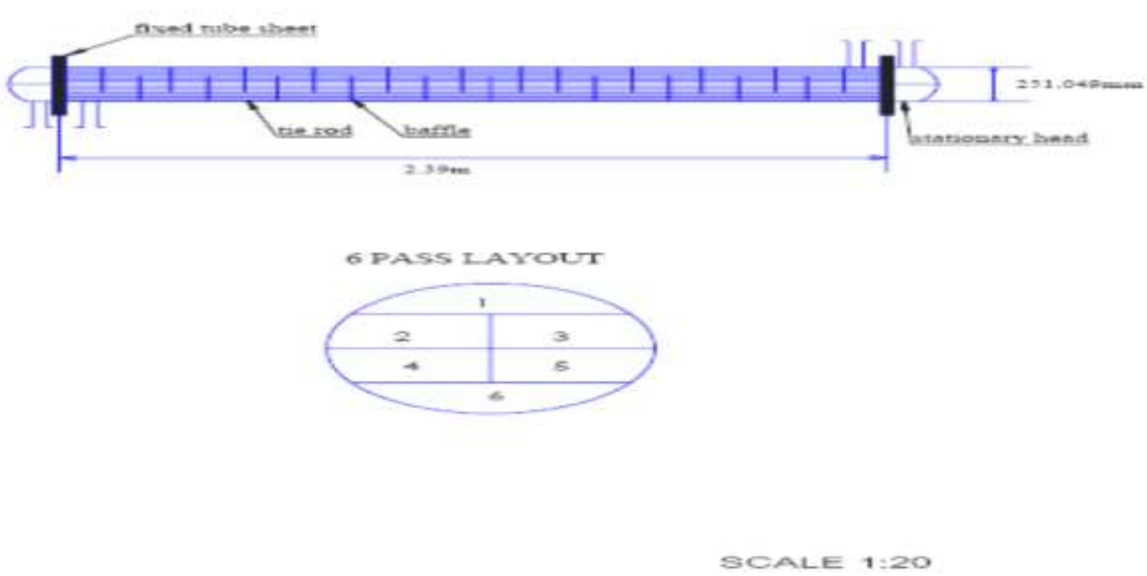

Fig. 1 Sketch showing the shell and tube heat exchanger according to design

4. DESIGN SUMMARY

4.1 Kern's Method	
Number of tubes	60
Bundle diameter	240.049 mm
Bundle diametrical clearance	11 mm
Shell diameter	251.049 mm
Tube side coefficient	1059.197 W/m^2K
Baffle spacing	50.209 mm
Tube pitch	25 mm
Equivalent diameter	14.4 mm
Shell side coefficient	1601.63 W/m^2K
Mean wall temperature	90.471 ℃
Overall heat transfer coefficient	405.62 W/m^2K
Shell side pressure drop	82.93 kPa
Number of passes	6
4.2 Bell's Method	
Shell side area	5041.96 mm^2
Height between baffle tips	125.524 mm
Tubes in one window area	11
Heat transfer coefficient	359.77 W/m^2K
Cross flow pressure drop	74.76 N/m^2
Window zone pressure drop	1489.97 N/m^2
End zone pressure drop	537.59 N/m^2
Shell side pressure drop	36.989 kPa
4.3 Mechanical Design	
Shell thickness	8 mm
Tube and sheet material	Cupro-nickel alloy
Tube outer diameter	20 mm
Tube inner diameter	16 mm
Length of heat exchanger	2.39 m
Tube thickness	0.0119 mm
Saddle support material	Carbon steel
Knuckle radius	15.06 mm

Table 1 Summary of the design work

5. REFERENCES

[1] Coulson and Richardson's Chemical Engineering, Volume 2, 5th, edition, J.M. Coulson and J.F. Richardson, Butterworth-Heinemann (2002)