Delft University of Technology

Equipment for Heat and Mass Transfer
ME45165

Equipment for Heat Transfer - Assignment 1

Design of shell and tube heat exchanger
E351 - Water-cooled ammonia condenser

Authors

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Design Procedure

The flowchart followed to design a shell and tube heat exchanger is as below.

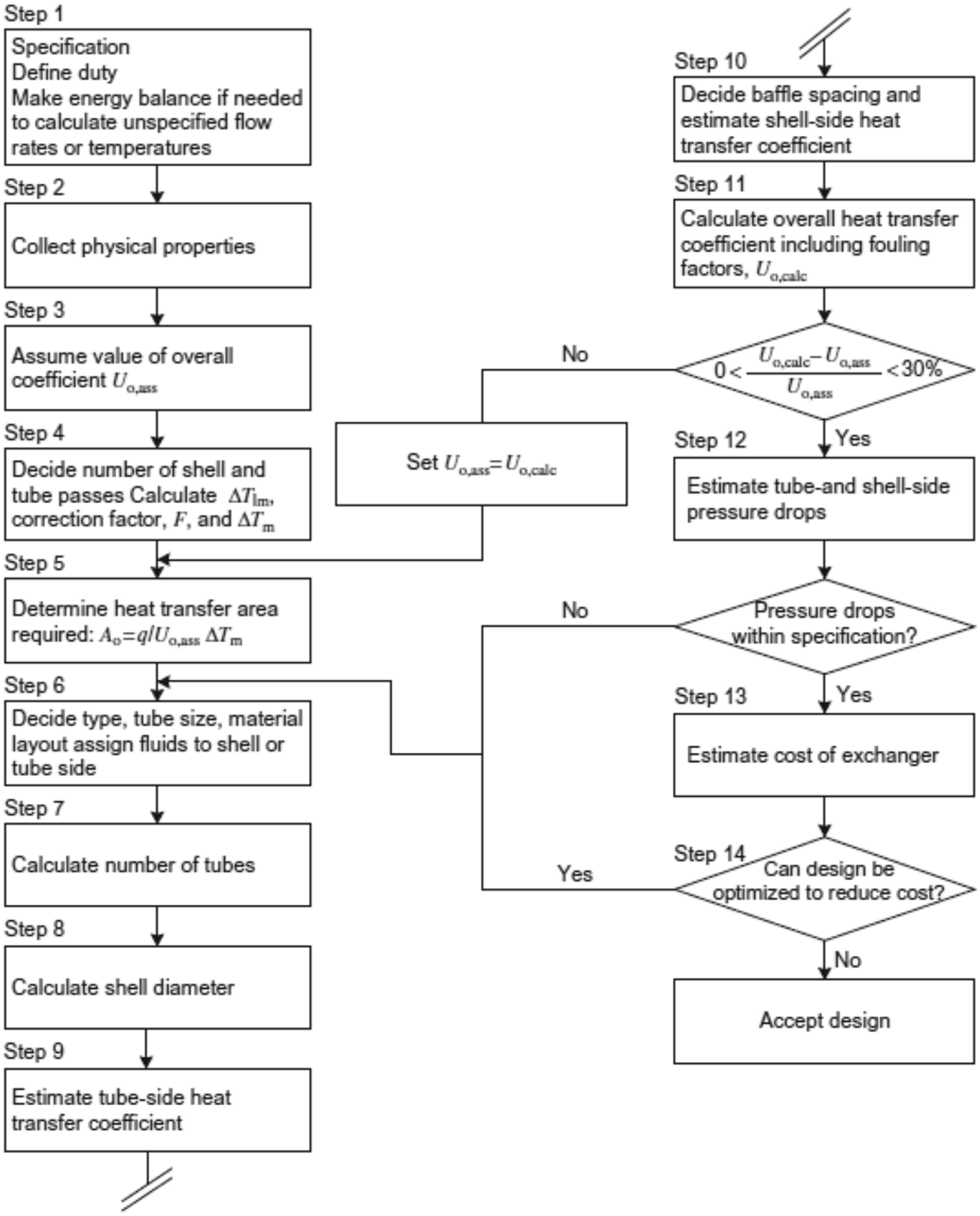


Figure 1: Design Procedure for Shell and Tube Heat Exchanger

Step 1

1.1 Specification

The given specifications for the design of E351 shell and tube heat exchanger are as follows:

S.No.	Fluid	Phase Change	Flow Rate (kg/s)	Temperature In (°C)	Temperature Out (°C)
1	Ammonia	Condensing	14	12	12
2	Water	1-Phase	680	5	-

Table 1: Given Fluid Properties

1.2 Determination of Duty

An estimate of the heat duty (total heat transferred) can be determined from the following equation. Due to condensation (phase change) in the fluid, the duty can be determined by the product of mass flow rate and change in enthalpies of the fluid undergoing temperature change.

$$\dot{Q} = \dot{m}_{amm}(h_{amm,g} - h_{amm,l}) \tag{1}$$

From the property table for ammonia, we determine $h_{amm,g} = 1617.0$ kJ/kg, $h_{amm,l} = 399.11$ kJ/kg. Substituting the obtained values in equation (1), we get,

$$\boxed{\dot{Q} = 17.05 \text{ MW}} \tag{2}$$

Performing an energy balance on the system, we can determine the outlet temperature of water as follows.

$$\begin{aligned} \dot{Q} &= \dot{m}_w \; C_{p,w} \; (T_{out,w} - T_{in,w}) \\ \implies \quad T_{out,w} &= T_{in,w} + \frac{\dot{Q}}{\dot{m}_w \; C_{p,w}} \\ \therefore \quad & \boxed{T_{out,w} = 284.1125 \text{ K} = 10.9625 \text{ °C}} \end{aligned} \tag{3}$$

Step 2

Physical Properties

The physical properties of the working fluids, ammonia and water, are determined at their arithmetic mean temperatures and tabulated below.

S. No.	Property	Symbol	Units	Fluid	
				Ammonia	Water
1	Liquid Density	ρ_l	kg/m^3	621.79	999.8
2	Vapor Density	ρ_v	kg/m^3	5.1983	-
3	Liquid Dynamic Viscosity	μ_l	Pa s	1.4992e-4	1.3857e-3
4	Vapor Dynamic Viscosity	μ_v	Pa s	9.4259e-6	-
5	Liquid Thermal Conductivity	k_l	W/mK	0.5232	0.57618

Table 2: Physical Properties

Step 3

Assumption of Overall Heat Transfer Coefficient

The initial value of the overall heat transfer coefficient is assumed from the figure given below. Ammonia is the process fluid in this case, and water, the service fluid. Joining their respective typical heat transfer coefficients, we get the following initial assumption for the overall heat transfer coefficient.

$$\boxed{U_{0,ass} = 900\ \mathrm{W/m^{2}{}^{\circ}C}} \tag{4}$$

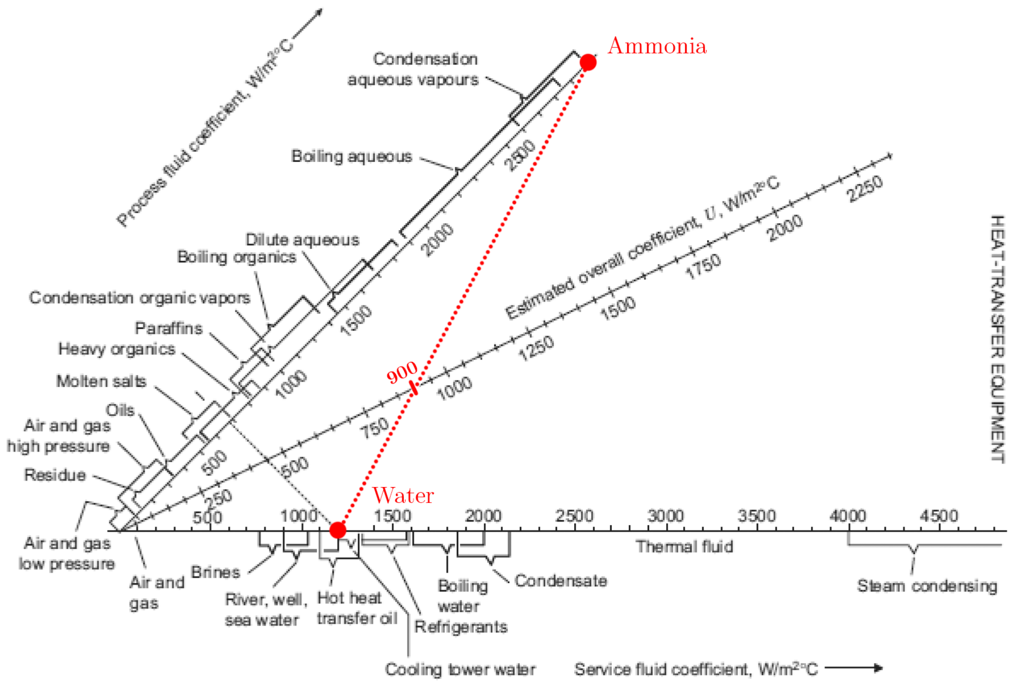


Figure 2: Typical values of Overall heat transfer coefficients

Step 4

4.1 Number of shell and tube passes

The number of tube passes (n_p) and shell passes are assumed to be 6 and 1 respectively.

4.2 Determination of LMTD and correction factor

Assuming counter current flow (as it creates a more uniform temperature difference between the fluids, over the entire length of the fluid path), the logarithmic mean temperature difference is determined using the relation given below.

$$\Delta T_{lm} = \frac{(T_{in,amm} - T_{out,w}) - (T_{out,amm} - T_{in,w})}{\ln\left(\dfrac{T_{in,amm} - T_{out,w}}{T_{out,amm} - T_{in,w}}\right)} \tag{5}$$

Substituting the required values, we get,

$$\boxed{\Delta T_{lm} = 3.1232\ {}^{\circ}\mathrm{C}} \tag{6}$$

Step 5

Heat transfer Area

The area of heat transfer can be computed from the assumed heat transfer coefficient using the following relation.

$$A_0 = \frac{Q}{U_{0,ass}\ \Delta T_{lm}} \tag{7}$$

Substituting the values of $U_{0,ass}$ and ΔT_m, we get,

$$\boxed{A_0 = 6065.9 \text{ m}^2} \tag{8}$$

The condensation occurs at a constant temperature (isothermally) at a constant pressure for a pure saturated vapor and hence, no correction factor is required for the multipass condensor used in this case.

Step 6

6.1 Selection of Tube Sizes

The material of the tube chosen is *stainless steel* as it is cheaper and more prone to corrosion when compared to several other materials like aluminium and copper. The diameter, thickness and length of the tube are determined from the standard design values.

Outer diameter, $d_0 = 0.019$ m
Wall thickness, $t_w = 0.0017$ m
Inner diameter, $d_i = d_0 - 2t_w = 0.0156$ m
Tube length, $L = 7.32$ m

6.2 Selection of Tube Layout

A square arrangement is chosen, the reason being such an arrangement would be better when compared to triangular arrangement for fouling fluids.

Pitch, $P_t = 1.25d_0 = 0.0238$ m

6.3 Assignment of Fluids

As water is generally more corrosive than ammonia, it has a higher fouling rate and hence, the following assignment is done since the tubes are easier to clean and replace.

Tube side - Water
Shell side - Ammonia

Step 7

Number of tubes

The number of tubes can be computed from the equation given below. The area of heat transfer in one tube can be determined using the formula $\pi d_0 L$.

$$N_t = \frac{A_0}{\pi d_0 L} \tag{9}$$

$$\boxed{N_t = 13883} \tag{10}$$

The tube velocity and Reynolds number are measured and checked if we have the required value. The tube side velocity can be determined from the following relation.

$$v_w = \frac{4\dot{m}_w}{\rho_w \pi d_i^2}\left(\frac{N_p}{N_t}\right) \tag{11}$$

$$\boxed{v_w = 1.538 \text{ m/s}} \tag{12}$$

$$Re_w = \frac{\rho_w v_w d_i}{\mu} \tag{13}$$

$$\boxed{Re_w = 1.7310\text{e+}4} \tag{14}$$

The velocity obtained above is within the permissible limits for the velocity in the tube side for water. Also to be noted is that the determined Reynolds number is in the *turbulent* regime.

Step 8

Calculation of shell diameter

The bundle diameter can be obtained from the expression given below.

$$D_b = d_0\left(\frac{N_{tx}}{K_1}\right)^{1/n_1} \tag{15}$$

where N_{tx} is the number of tubes per heat exchanger

Total number of tubes, $N_t = 13883$
Number of heat exchangers, $x = 11$
Number of tubes per heat exchanger, $N_{tx} = 1262$

The values of K_1 and n_1 for 6 tube passes can be obtained from the table given below.

Square pitch, $p_t = 1.25d_o$

No. passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

Figure 3: Values of K_1 and n_1

$$\boxed{D_b = 0.993 \text{ m}} \tag{16}$$

Considering fixed and U-tube design, the value of tolerance between shell and tube bundle can be obtained by interpolation of the linear curve from Figure 4.

The tolerance value is obtained from Figure 4 as 0.0179 m.
The inner shell diameter, $D_i = D_b$ + tolerance = 1.0112 m

Note that the ratio of tube length to shell diameter ratio is 7.24 which lies in the optimum range of 5 to 10.

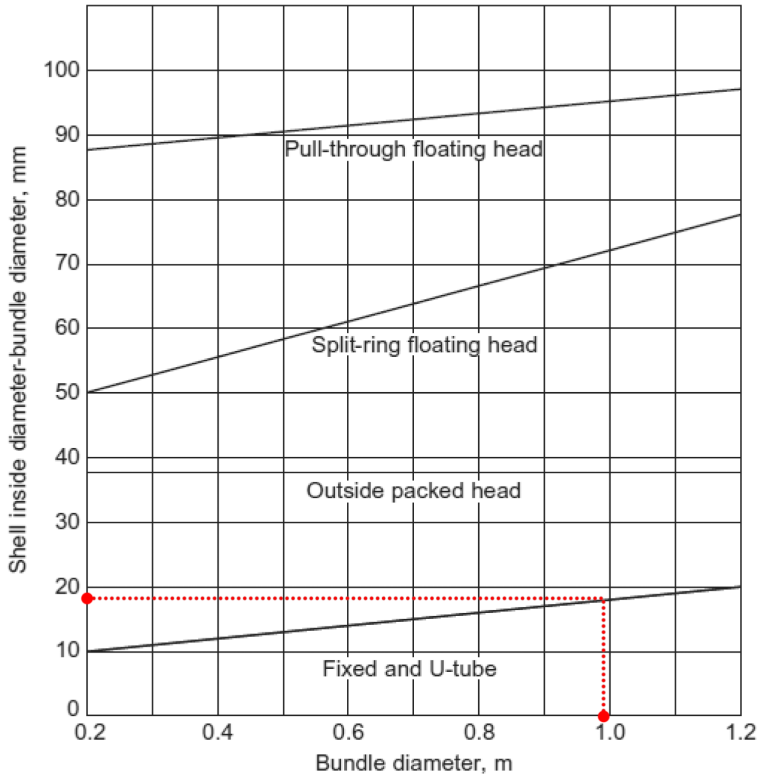


Figure 4: Values of tolerance

Step 9

Estimation of tube-side heat transfer coefficient

The tube-side heat transfer coefficient can be accurately estimated for water using the adapted equation given by Eagle and Ferguson.

$$h_t = \frac{4200(1.35 + 0.02t)u_t^{0.8}}{d_i^{0.2}} \tag{17}$$

where u_t is the velocity of water and t is the mean temperature of water

$$\therefore \quad \boxed{h_t = 5164.6 \text{ W/m}^2\text{K}} \tag{18}$$

The tube-side heat transfer coefficient can also be calculated using the equation given below.

$$h_t = \frac{k_l}{d_i} j_h Re Pr^{0.33} \left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{19}$$

where j_h is the heat transfer factor and μ_w is the wall viscosity at mean wall temperature.

The mean wall temperature can be determined using the expression given below.

$$h_t(t_w - t) = U_{0,ass}(T - t) \tag{20}$$

In the above expression, t is the tube-side mean temperature, t_w is the estimated wall temperature and T is the shell-side mean temperature.

Substituting the required values, the wall temperature can be determined as $t_w = 281.83$ K $= 8.68$ °C. For the calculated wall temperature, the dynamic viscosity at the wall is obtained from REFPROP as 1.3571e−3 Pa s. Accordingly, (μ/μ_w) is found to be 1.017 which signifies that the viscosity at the wall is almost same as the viscosity of water at mean temperature and hence (μ/μ_w) can be neglected from equation (19).

The other parameters are calculated using their respective expressions and substituted in the equation (19) to obtain h_t as 5438 W/m^{2}K. This value is comparable to the one obtained from equation (17) and hence the *lower h_t value* from equation (18) is considered for further calculations.

Step 10

Estimation of shell-side heat transfer coefficient

Considering horizontal arrangement with condensation in the shell, the mean shell-side heat transfer coefficient using Kern's method is determined using the following equation.

$$h_s = 0.95 k_L \left[\frac{\rho_L(\rho_L - \rho_v)g}{\mu_L \Gamma_h}\right]^{1/3} N_r^{-1/6} \tag{21}$$

where $\Gamma_h = \dfrac{W_c}{L n_t}$ is the tube loading, with W_c being the total condensate flow and N_r the average number of tubes in a vertical tube row (taken to be two-thirds of the number in central tube row). Substituting the known values, we determine the shell side heat transfer coefficient as below.

$$\therefore \quad \boxed{h_s = 7276.3 \text{ W/m}^2\text{K}} \tag{22}$$

Step 11

Calculation of overall heat transfer coefficient

The overall heat transfer coefficient U_0 can be calculated from the below given equation.

$$\frac{1}{U_0} = \frac{1}{h_s} + \frac{1}{h_{od}} + \frac{d_0 \ln\left(\dfrac{d_o}{d_i}\right)}{2k_w} + \frac{d_o}{d_i} \times \frac{1}{h_{id}} + \frac{d_o}{d_i} \times \frac{1}{h_t} \tag{23}$$

where

h_s is the shell-side fluid film coefficient
h_t is the tube-side fluid film coefficient
h_{od} is the shell-side fouling factor
h_{id} is the tube-side fouling factor

The fouling factors of water and ammonia are taken from the tables as 1000 W/m^{2}K and 5000 W/m^{2}K respectively. Substituting the required values in the above equation, we get,

$$\boxed{U_0 = 547.6765 \text{ W/m}^2\text{K}} \tag{24}$$

The following condition is checked in order to justify our assumption of U_0.

$$0\% < \left|\frac{U_{0,calc} - U_{0,ass}}{U_{0,ass}}\right| \times 100 < 30\% \tag{25}$$

The error between the calculated and assumed value of overall heat transfer coefficient is found to be 39%.

Hence, according to the design flow chart, the $U_{0,calc}$ is considered as $U_{0,ass}$ for the next iteration for which the calculations are carried out from step 5. After running the iterations, the changes made and the final values of the parameters for which the error is less than 30% are tabulated as given below.

Parameters	**Symbols**	**Units**	**Values**
Design Parameters			
Tube Outer Diameter	d_o	mm	19
Tube Inner Diameter	d_i	mm	15.6
Tube Pitch	p_t	mm	23.8
Tube Wall Thickness	t	mm	1.7
Tube Length	L	m	7.32
Number of Tubes	N_t	-	22513
Number of Heat Exchangers	x	-	21
Number of Tubes per Heat Exchanger	N_{t_x}	-	1072
Number of Tube Passes	N_p	-	8
Bundle Diameter	D_b	m	0.9667
Shell Inner Diameter	D_i	m	0.9844
Operational Parameters			
Heat Transfer Area	A_0	m^2	9836.6
Velocity of Water (tube-side)	v_t	m/s	1.2645
Reynolds Number (tube-side)	Re_t	-	1.42e+4
Tube-side Heat Transfer Coefficient	h_t	W/m^2K	4416.0
Shell-side Heat Transfer Coefficient	h_s	W/m^2K	6918.9
Overall Heat Transfer Coefficient	U_0	W/m^2K	534.43
Error	-	%	3.7

Table 3: Values of parameters obtained after final iteration

It can be seen that the error between the acquired overall heat transfer coefficient for the iterated case and that of the assumed value, approximately 4%, is within the permissible limits.

Step 12

12.1 Calculation of tube-side pressure drop

The tube-side pressure drop is calculated from the equation given below.

$$\Delta P_t = N_p \left[8 j_f \left(\frac{L}{d_i} \right) \left(\frac{\mu}{\mu_w} \right)^{-m} + 2.5 \right] \frac{\rho u_t^2}{2} \tag{26}$$

where j_f is the friction factor determined from figure 5 and $m = 0.14$ for turbulent flows.

Substituting the known and computed values, we get,

$$\boxed{\Delta P_t = 123.8 \text{ kPa} = 1.238 \text{ bar}} \tag{27}$$

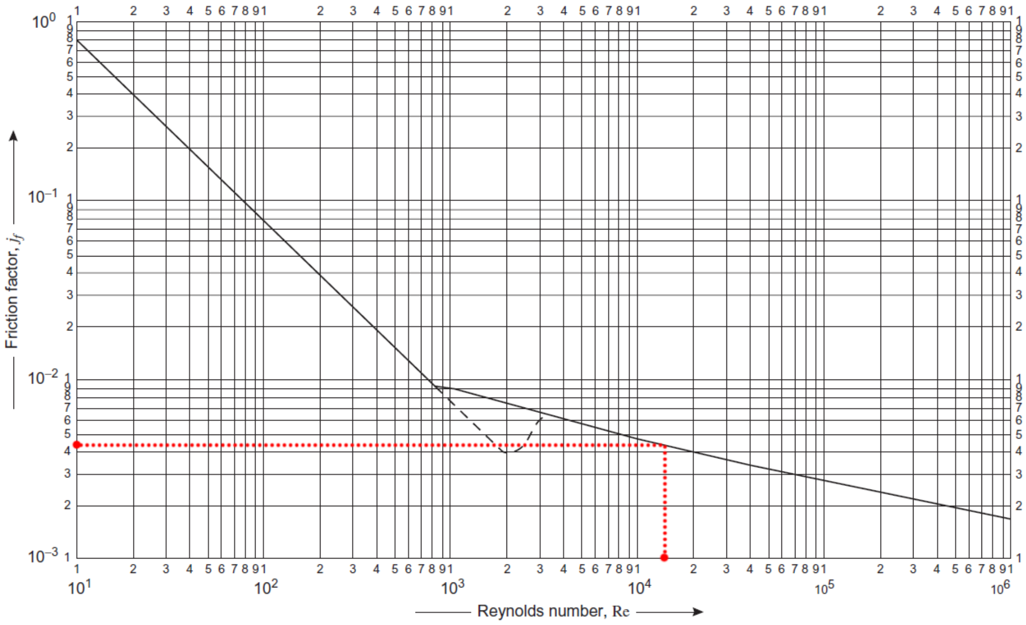


Figure 5: Tube-side friction factors

The obtained value for the pressure drop is higher than the allowable range of 50 - 70 kPa. Hence, we tune the design parameters to incorporate a lower pressure drop. After re-iterating, we get the final values of the design parameters that satisfy all the required criteria as listed in the table below.

Parameters	Symbols	Units	Values
Design Parameters			
Tube Outer Diameter	d_0	mm	19
Tube Inner Diameter	d_i	mm	15.6
Tube Pitch	p_t	mm	23.8
Tube Wall Thickness	t_w	mm	1.7
Tube Length	L	m	6.1
Number of Tubes	N_t	-	28451
Number of Heat Exchangers	x	-	25
Number of Tubes per Heat Exchanger	N_{t_x}	-	1166
Number of Tube Passes	N_p	-	8
Bundle Diameter	D_b	m	0.998
Shell Inner Diameter	D_i	m	1.016
Operational Parameters			
Heat Transfer Area	A_0	m^2	10359
Velocity of Water (tube-side)	v_t	m/s	1.0
Reynolds Number (tube-side)	Re_t	-	1.1262e+4
Tube-side Heat Transfer Coefficient	h_t	W/m^2K	3661.8
Shell-side Heat Transfer Coefficient	h_s	W/m^2K	6666.5
Overall heat transfer coefficient	U_0	W/m^2K	517.21
Error	-	%	1.86
Tube-side pressure drop	ΔP_t	kPa	66.24

Table 4: Final design parameter values

12.2 Calculation of shell-side pressure drop

As the flow across the shell is a two-phase flow, it is recommended that the pressure drop be determined for the liquid phase. An approximate estimate for the actual two-phase case is 50% of the determined value. To start with, the area of the cross flow inside the shell is given by the equation:

$$A_s = \frac{(p_t - d_0)D_i\ l_B}{p_t} \tag{28}$$

where l_B is the baffle spacing which is assumed to be same as inner diameter of the shell as condensers typically have wider spacing. Substituting the known parameters in the above equation, we get,

$$\boxed{A_s = 0.2064\ \mathrm{m}^2} \tag{29}$$

The shell side velocity is computed using the relation:

$$u_s = \frac{G_s}{\rho} \tag{30}$$

where the mass velocity $G_s = \dfrac{W_s}{A_s}$ with W_s indicating the shell-side flow rate, in this case, 14 kg/s.

$$\therefore \quad \boxed{u_s = 1.091\ \mathrm{m/s}} \tag{31}$$

The shell-side Reynolds number for the computed velocity is 8.488e+3 and hence, the flow can be regarded as *turbulent.*

The shell-side equivalent diameter for square pitch orientation is calculated as follows.

$$d_e = \frac{1.27}{d_0}(p_t^2 - 0.785 d_0^2) \tag{32}$$

$$\boxed{d_e = 0.0188\ \mathrm{m}} \tag{33}$$

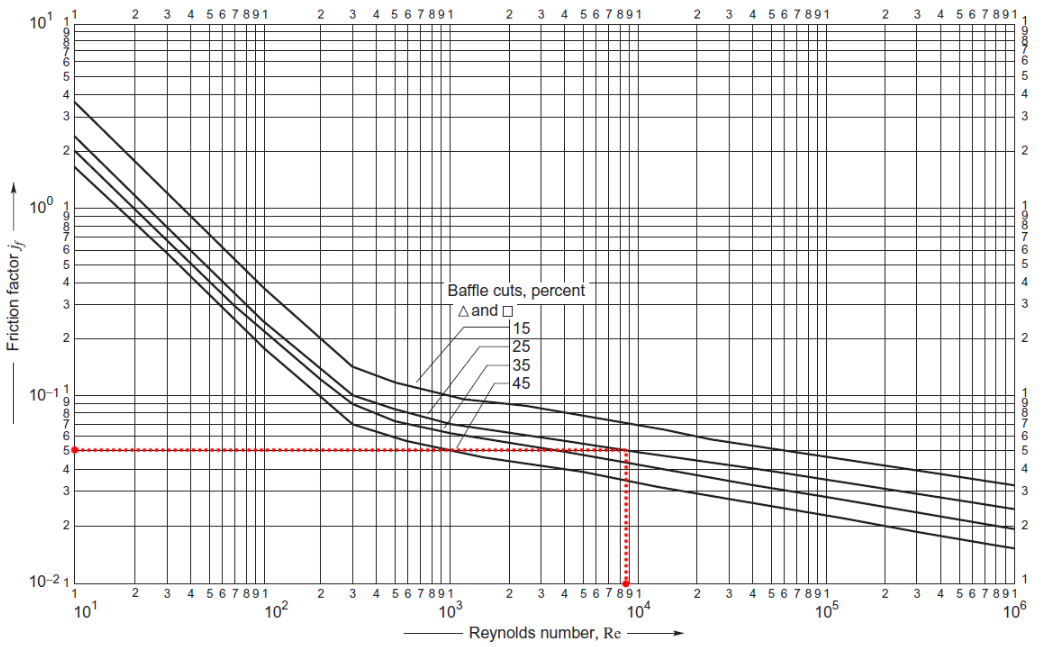


Figure 6: Shell-side friction factors

The shell-side pressure drop is calculated from the equation given below.

$$\Delta P_s = 8 j_f \left(\frac{D_i}{d_e}\right)\left(\frac{L}{l_B}\right)\frac{\rho u_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{34}$$

In the above equation, j_f is the friction factor determined from figure 6, D_i the shell inner diameter, L the tube length, l_B the baffle spacing, ρ the density of liquid ammonia at 12°C, μ the viscosity of liquid ammonia at 12°C.

The viscosity at the wall is calculated in a similar way as done before by determining the wall temperature using the viscosity correction relation and finding the viscosity at that temperature from the tables.

Substituting all the required values, we get,

$$\boxed{\Delta P_s = 0.241 \text{ kPa}} \tag{35}$$

12.3 Calculation of shell-side pressure drop using VDI method

The shell side pressure drop can be calculated by characterizing it into individual parts as per the relation:

$$\Delta P_s = (n_B - 1)\Delta P_Q + 2\Delta P_{QE} + n_B \Delta P_W + \Delta P_N \tag{36}$$

In the above equation, ΔP_Q is the pressure drop in a central cross flow section, ΔP_{QE} the pressure drop in an end cross flow section, ΔP_W the pressure drop in a window section and ΔP_N the pressure drop in both nozzles.

A detailed description of the procedure for determining the individual pressure drops is included in the Appendix. The values for the individual pressure drops obtained are as follows:

$$\begin{aligned} \Delta P_Q &= 3.1573 \text{ Pa} \\ \Delta P_{QE} &= 26.6652 \text{ Pa} \\ \Delta P_W &= 1.4263 \text{ Pa} \\ \Delta P_N &= 165.1703 \text{ Pa} \end{aligned} \tag{37}$$

Hence, the total pressure drop computed using the above method is

$$\boxed{\Delta P_s = 0.247 \text{ kPa}} \tag{38}$$

It can be seen that the determined pressure drop in the shell side matches with that determined from Kern's method with considerable accuracy. Also, the magnitude of 0.24 kPa is well within the permissible limits and hence, the design is considered fit.

12.4 Visualisation of temperature profile using cell method

The cell method can be used to visualise the variation in the temperature of the coolant in the tubes as it traverses through the shell. The shell is assumed to be singular and the number of rows has been determined to be $N_r = 28$.

The temperature at the exit of each row is computed since the shell-side heat transfer coefficient h_s varies with the number of rows. An energy balance is performed with the overall heat transfer coefficient as below:

$$UA\ \Delta T_{LM} = \dot{m}_w C_{p,w}(T_{out,w} - T_{in,w}) \tag{39}$$

In the above equation, the values for U and ΔT_{LM} depend on the values of $T_{out,w}$ and $T_{in,w}$. Hence, the system is solved using a simple loop iterating N_r times, with the outlet temperature of one cell taken as the inlet temperature of the succeeding cell.

Upon iterating, we obtain the temperature plot as shown in figure 7.

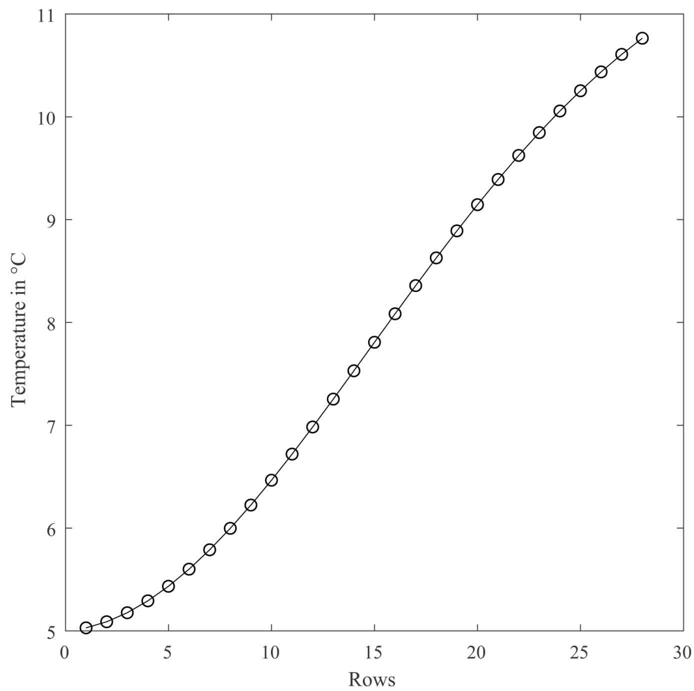


Figure 7: Temperature variation across the tubing rows

From the cell method, we obtain the temperature at the outlet of the tubing as:

$$\boxed{T_{out,w} = 283.9142 \text{ K}} \tag{40}$$

Recalling from Equation (3), we determined the required outlet temperature of water with an overall energy balance as $T_{out,w} = 284.1125$ K. Hence, the outlet temperature computed as per the cell method ranges within 2% approximately (in terms of °C) when compared with Kern's method, which is within the acceptable limits.

Step 13

Estimation of cost

The equipment cost for the shell and tube exchanger can be determined from the 2010 model from Sinnott and Towler as follows:

$$Cost_{equipment} = a + bS^n \tag{41}$$

The values of a, b and n are obtained from the table given in figure 8 as $a = 28000$, $b = 54$, $n = 1.2$ and S is the total area of heat transfer per heat exchanger.

The costs are then adjusted for inflation with an inflation factor of 1.14. Hence, the total predicted costs for the equipment is calculated to be 2.69 million Euros.

$$\boxed{Cost_{equipment} \approx € \, 2,692,500} \tag{42}$$

Additionally, the costs for installation and maintenance are to be accounted for. Sinnott and Towler recommends using a factor of 3.5 with the equipment costs to estimate the total costs associated with the heat exchanger unit. Hence, the total costs are calculated to be as below.

$$\boxed{Cost_{total} \approx € \ 9,423,800} \tag{43}$$

Equipment	Units for Size, S	S_{lower}	S_{upper}	a	b	n	Note
Dryers							
Direct contact Rotary	m^2	11	180	15,000	10,500	0.9	1
Atmospheric tray batch	area, m^2	3.0	20	10,000	7,900	0.5	
Spray dryer	evap rate kg/h	400	4,000	410,000	2,200	0.7	
Evaporators							
Vertical tube	area, m^2	11	640	330	36,000	0.55	
Agitated falling film	area, m^2	0.5	12	88,000	65,500	0.75	2
Exchangers							
U-tube shell and tube	area, m^2	10	1,000	28,000	54	1.2	
Floating head shell and tube	area, m^2	10	1,000	32,000	70	1.2	
Double pipe	area, m^2	1.0	80	1,900	2,500	1.0	
Thermosiphon reboiler	area, m^2	10	500	30,400	122	1.1	
U-tube Kettle reboiler	area, m^2	10	500	29,000	400	0.9	
Plate and frame	area, m^2	1.0	500	1,600	210	0.95	2

Figure 8: Parameters for determining equipment cost

Step 14

Can the design be optimized to reduce cost?

Costs can be reduced in the following ways:

- Reducing the number of heat exchangers will, in turn, increase the number of tubes per heat exchangers. Thereby, the velocity of water in the tubes reduces which lowers the shear inside the tubes and hence increases the rate of tube-side fouling.
- Choosing a cheaper material alternative like carbon steel will make the tubes more vulnerable to fouling, thereby reducing its service life. So, thicker walls might be required which adds to the weight of the heat exchanger and further costs associated with it.
- Choosing standard design values so as to reduce custom machining costs. However, the current design already matches the industrial design standards and all the design values are ISO standard values.

As the current design is optimized to design standards and all the typical ways to reduce costs have known negative consequences, it is assumed that the design is optimized to costs.

Hence, the current design is finalised. The parameter values of the final design of the shell and tube heat exchanger are tabulated below.

Parameter Values of Final Design

Parameters	**Symbols**	**Units**	**Values**
Design Parameters			
Tube Outer Diameter	d_o	mm	19
Tube Inner Diameter	d_i	mm	15.6
Tube Pitch	p_t	mm	23.8
Tube Wall Thickness	t_w	mm	1.7
Tube Length	L	m	6.1
Number of Tubes	N_t	-	29150
Number of Heat Exchangers	x	-	25
Number of Tubes Per Heat Exchanger	N_{t_x}	-	1166
Number of Tube Passes	N_p	-	8
Bundle Diameter	D_b	m	0.998
Shell Outer Diameter	D_o	m	1.035
Shell Inner Diameter	D_i	m	1.016
Shell Wall Thickness	T_w	mm	9.5
Operational Parameters			
Water Outlet Temperature	$T_{out,w}$	°C	10.7642
Heat Transfer Area	A_0	m^2	10359
Velocity of Water (tube-side)	v_t	m/s	1.0
Tube Side Reynolds number (tube-side)	Re_t	-	$1.1262e+4$
Tube-side Heat Transfer Coefficient	h_t	W/m^2K	3661.8
Shell-side Heat Transfer Coefficient	h_s	W/m^2K	6666.5
Overall Heat Transfer coefficient	U_0	W/m^2K	517.21
Tube-side Pressure Drop	ΔP_t	kPa	66.24
Shell-side Pressure Drop (Kern method)	ΔP_s	kPa	0.241
Shell-side Pressure Drop (VDI method)	ΔP_s	kPa	0.247
Cost Parameters			
Tube Material	-	-	Stainless Steel
Shell Material	-	-	Stainless Steel
Estimated Cost per Heat Exchanger	-	€	107700
Inflation Factor	-	-	1.14
Estimated Total Equipment Costs	-	€	2,692,500
Total Estimated Costs	-	€	9,423,800

Table 5: Final Design Parameters

Appendix

E351 - Water-cooled Ammonia Condenser
MATLAB Live Script

Design of Shell and Tube Heat Exchanger

E351 - Water Cooled Ammonia Condenser

Group 12: Aswin Raghunathan (5128188) & Shyam Sundar Hemamalini (5071984)

1.1. Estimating heat flux from given specification

We are given the following specifications.

```
m_amm = 14; % kg/s
h_amm_l = 399.11*1000; % J/kg
h_amm_g = 1617.0*1000; % J/kg
```

Now, the duty can be determined as:

```
Q_dot = m_amm*(h_amm_g - h_amm_l) % W
```

```
Q_dot = 17050460
```

1.2. Performing energy balance to determine outlet temperature

The following specifications are also provided.

```
m_water = 680; % kg/s
T_in_water = 5 + 273.15; % K
Cp_water = 4205.3; % J/kgK
```

A simple substitution in the energy balance will give us the outlet temperature of water as:

```
T_out_water = T_in_water + Q_dot/(m_water*Cp_water) % K
```

```
T_out_water = 284.1125
```

2. Physical Properties

The physical properties of both water and ammonia are evaluated at the mean temperature as:

```
rho_amm_l = 621.79; % kg/m^3
rho_water_l = 999.8; % kg/m^3
rho_amm_g = 5.1983; % kg/m^3

visc_amm_l = 1.4992e-4; % Pa s
visc_amm_g = 9.4259e-6; % Pa s
visc_water_l = 1.3857e-3; % Pa s

k_amm_l = 0.5232; % W/mK
k_water_l = 0.57618; % W/mK
```

3. Initial Overall Heat Transfer Coefficient

The initial value has been assumed from the chart given in Sinnott and Towler as:

```
U_0_init = 900 % W/m^2K
```

```
U_0_init = 900
```

4.1. Shell and tube passes

The number of tube passes is initially assumed to be 6.

```
N_p = 6;
```

4.2. Determination of LMTD

```
T_in_amm = 12 + 273.15; % K
T_out_amm = 12 + 273.15; % K
T_mean_shell = (T_in_amm + T_out_amm)/2;
T_mean_tube = (T_in_water + T_out_water)/2;
```

Now, we have the formula for logarithmic mean temperature difference as:

```
T_lmtd = ((T_in_amm-T_out_water)-(T_out_amm-T_in_water))/...
    log((T_in_amm - T_out_water)/(T_out_amm - T_in_water)) % K
```

```
T_lmtd = 3.1232
```

5. Determination of Heat Transfer Area

From the assumed value of heat transfer coefficient, the heat transfer area can be approximated as:

```
A0 = Q_dot / (U_0_init*T_lmtd) % m^2
```

```
A0 = 6.0659e+03
```

6. Determination of Design Parameters

The following design assumptions are made:

```
d_out = 0.019; % m
t_wall = 0.0017; % m
L_tube = 7.32; % m
```

Depending on the above assumptions, we determine the following design values.

```
pitch_tube = 1.25*d_out % pitch in m
```

```
pitch_tube = 0.0238
```

```
d_in = d_out - 2*t_wall % inner diameter in m
```

```
d_in = 0.0156
```

7.1. Determination of Number of Tubes

For cylindrical tubes of the above design values, the heat transfer area translates as:

```
N_tubes = A0 / (pi*d_out*L_tube) % number of tubes
```

```
N_tubes = 1.3883e+04
```

```
N_hex = 11 % number of heat exchangers
```

```
N_hex = 11
```

```
N_t_hex = round(N_tubes/N_hex) % number of tubes per heat exchanger
```

```
N_t_hex = 1262
```

7.2. Determination of Tube Velocity

Now, we can determine an estimate of the velocity of water in the tubes as:

```
velocity_water = 4*m_water*N_p/...
    (rho_water_l*pi*(d_in^2)*N_tubes) % m/s
```

```
velocity_water = 1.5379
```

Also, the Reynolds number is given by:

```
reynolds = rho_water_l*velocity_water*d_in/visc_water_l
```

```
reynolds = 1.7310e+04
```

8. Calculation of Shell Diameter

For 6 tube side passes, we have the values of K1 and n1 and hence, the design values of the shell can be determined as:

```
K1 = 0.0402; % for 6 tube passes
n1 = 2.617;
d_bundle = d_out*(N_t_hex/K1)^(1/n1) % m
```

```
d_bundle = 0.9933
```

```
tolerance = (10 + (20-10)*(d_bundle - 0.2)/(1.2 - 0.2))/1000 % m
```

```
tolerance = 0.0179
```

```
d_shell_i = d_bundle + tolerance % m
```

```
d_shell_i = 1.0112
```

```
N_center = round(d_bundle / pitch_tube) % number of tubes in center row
```

```
N_center = 42
```

9.1. Estimation of Tube Side Heat Transfer Coefficient

The mean bulk temperature of water inside the tubes is given by:

```
T_water_mean = (T_in_water + T_out_water)/2; % K
T_wm_C = T_water_mean - 273.15; % °C
```

The above can be substituted in the correlation by Eagle and Ferguson for water as below.

```
h_i = (4200*(1.35+0.02*T_wm_C)*(velocity_water^0.8))/((d_in*1000)^0.2) % W/m^2K
```

```
h_i = 5.1646e+03
```

9.2. Estimation of Tube Side Heat Transfer Coefficient - Kern's Method

An alternative to the above method is the Kern's method. However, here, we need the correction for viscosity. We first determine the thermophysical properties for water at the mean bulk temperature.

```
prandtl = Cp_water*visc_water_l/k_water_l;
E = 0.0225*exp(-0.0225*(log(prandtl))^2);
stanton = E * (reynolds^-0.205) * (prandtl^-0.505);
jh = stanton*prandtl^0.67;
```

Now, we implement the viscosity correction and determine the viscosity of water near the wall.

```
T_wall_tube = T_mean_tube + U_0_init*(T_mean_shell - T_mean_tube)/h_i % K
```

```
T_wall_tube = 281.8316
```

```
visc_water_w = 1.3571e-3; % Pa s
```

This is then substituted in the correlation to determine the heat transfer coefficient.

```
h_i_kern1 = k_water_l*jh*reynolds*(prandtl^0.33)*((visc_water_l/... visc_water_w)^0.14)/d_in % W/m^2K
```

```
h_i_kern1 = 5.4380e+03
```

Since the above value is higher than the previously computed value, this is ignored.

10. Shell Side Heat Transfer Coefficient

The initial guess for shell side heat transfer coefficient is:

```
U_amm_init = 7275; % W/m^2K
```

The viscosity correction factor is then determined from the wall temperature.

```
T_wall = T_mean_shell - (T_mean_shell - T_mean_tube)*U_0_init/U_amm_init % K
```

```
T_wall = 284.6528
```

```
T_amm_con = (T_mean_shell + T_wall)/2 % K
```

```
T_amm_con = 284.9014
```

```
visc_amm_c = 1.503e-4; % Pa s
rho_amm_c = 622.15; % kg/m^3
k_amm_c = 0.52394; % W/mK
rho_amm_v = 5.1983; % kg/m^3
```

The tube loading is determined as below.

```
gamma = m_amm/(L_tube*N_t_hex) % kg/ms
```

```
gamma = 0.0015
```

```
N_r = 2*N_center/3
```

```
N_r = 28
```

As we employ a shell-side condensing horizontal axis heat exchanger, we use the following correlation to determine the heat transfer coefficient.

```
h_c_shell = 0.95 * k_amm_c *((rho_amm_c*(rho_amm_c-rho_amm_v)*9.81)/...
    (visc_amm_c*gamma))^(1/3) * (N_r^(-1/6)) % W/m^2K
```

```
h_c_shell = 7.2763e+03
```

11. Overall Heat Transfer Coefficient

Sea water, typically used in OTEC, is heavily fouling while ammonia is less fouling. Hence, the following values are assumed as the fouling factors. Also, as we consider the tube material to be low carbon steel, we have the following value for thermal conductivity.

```
fouling_water = 1000; % W/m^2K
fouling_ammonia = 5000; % W/m^2K
k_steel = 54; % W/mK
```

Computing the overall heat transfer coefficient is done as below:

```
val = (h_c_shell^-1) + (fouling_ammonia^-1) + d_out * log(d_out/d_in) / ... (2*k_steel) +
d_out/(d_in*fouling_water) + d_out/(d_in*h_i);
U1 = val^-1 % W/m^2K
```

```
U1 = 547.6765
```

Calculating Error

```
Error = (U_0_init - U1)/U_0_init*100 % in percentage
```

```
Error = 39.1471
```

The error, which is about 39%, exceeds the permissible limits. Hence, we take a new value for the overall heat transfer coefficient as the new guess and repeat.

3*. Initial Overall Heat Transfer Coefficient

```
U_0_init = 555; % W/m^2K
```

5*. Determination of Heat Transfer Area

```
A0 = Q_dot / (U_0_init*T_lmtd) % m^2
```

```
A0 = 9.8366e+03
```

6*. Determination of Design Parameters

```
d_out = 0.019; % m
```

```
t_wall = 0.0017; % m
L_tube = 7.32; % m
N_p = 8; % increasing the passes to increase the velocity in tubes
pitch_tube = 1.25*d_out % m
```

```
pitch_tube = 0.0238
```

```
d_in = d_out - 2*t_wall % m
```

```
d_in = 0.0156
```

7.1*. Determination of Number of Tubes

```
N_tubes = A0 / (pi*d_out*L_tube) % number of tubes
```

```
N_tubes = 2.2513e+04
```

```
N_hex = 21
```

```
N_hex = 21
```

```
N_t_hex = round(N_tubes/N_hex)
```

```
N_t_hex = 1072
```

7.2*. Determination of Tube Velocity

```
velocity_water = 4*m_water*N_p/(rho_water_l*pi*(d_in^2)*N_tubes) % m/s
```

```
velocity_water = 1.2645
```

```
reynolds = rho_water_l*velocity_water*d_in/visc_water_l
```

```
reynolds = 1.4233e+04
```

Both the velocity and the Reynolds number are within the permissible limits after changes to design.

8*. Calculation of Shell Diameter

```
K1 = 0.0331; % for 8 tube passes
n1 = 2.643;
d_bundle = d_out*(N_t_hex/K1)^(1/n1) % m
```

```
d_bundle = 0.9667
```

```
tolerance = (10 + (20-10)*(d_bundle - 0.2)/(1.2 - 0.2))/1000 % m
```

```
tolerance = 0.0177
```

```
d_shell_i = d_bundle + tolerance % m
```

```
d_shell_i = 0.9844
```

```
N_center = round(d_bundle / pitch_tube) % tubes in center row
```

```
N_center = 41
```

9.1*. Estimation of Tube Side Heat Transfer Coefficient

```
T_water_mean = (T_in_water + T_out_water)/2; % K
T_wm_C = T_water_mean - 273.15; % °C
h_i = (4200*(1.35+0.02*T_wm_C)*(velocity_water^0.8))/((d_in*1000)^0.2) % W/m^2K
```

```
h_i = 4.4160e+03
```

The changes to the tube side heat transfer coefficient is not pronounced.

9.2*. Estimation of Tube Side Heat Transfer Coefficient - Kern's Method

```
prandtl = Cp_water*visc_water_l/k_water_l;
E = 0.0225*exp(-0.0225*(log(prandtl))^2);
stanton = E * (reynolds^-0.205) * (prandtl^-0.505);
jh = stanton*prandtl^0.67;
T_wall_tube = T_mean_tube + U_0_init*(T_mean_shell - T_mean_tube)/h_i % K
```

```
T_wall_tube = 281.6363
```

```
visc_water_w = 1.365e-3; % Pa s
h_i_kern = k_water_l*jh*reynolds*(prandtl^0.33)*((visc_water_l/visc_water_w)^0.14)/d_in %
W/m^2K
```

```
h_i_kern = 4.6505e+03
```

10*. Shell Side Heat Transfer Coefficient

```
U_amm_init = 7275; % W/m^2K
T_wall = T_mean_shell - (T_mean_shell - T_mean_tube)*U_0_init/U_amm_init % K
```

```
T_wall = 284.8434
```

```
T_amm_con = (T_mean_shell + T_wall)/2 % K
```

```
T_amm_con = 284.9967
```

```
visc_amm_c = 1.503e-4; % Pa s
rho_amm_c = 622.15; % kg/m^3
k_amm_c = 0.52394; % W/mK
rho_amm_v = 5.1983; % kg/m^3
gamma = m_amm/(L_tube*N_t_hex) % kg/ms
```

```
gamma = 0.0018
```

```
N_r = 2*N_center/3
```

```
N_r = 27.3333
```

```
h_c_shell = 0.95 * k_amm_c *((rho_amm_c*(rho_amm_c-rho_amm_v)*9.81)/...
    (visc_amm_c*gamma))^(1/3) * (N_r^(-1/6)) % W/m^2K
```

```
h_c_shell = 6.9189e+03
```

Whereas, the heat transfer coefficient increases slightly in the shell side.

11*. Overall Heat Transfer Coefficient

```
fouling_water = 1000; % W/m^2K
fouling_ammonia = 5000; % W/m^2K
k_steel = 57; % W/mK
val = (h_c_shell^-1) + (fouling_ammonia^-1) + d_out * log(d_out/d_in) / (2*k_steel) +...
    d_out/(d_in*fouling_water) + d_out/(d_in*h_i);
U1 = val^-1 % W/m^2K
```

```
U1 = 534.4312
```

The newly computed value varies slightly from the assumed value.

Calculating Error

```
Error = (U_0_init - U1)/U_0_init*100 % in percentage
```

```
Error = 3.7061
```

The error between the values is about 3.7% which is well within the permissible limits. Hence, we can proceed to determine the pressure drop in the tube and shell side.

12. Tube Side Pressure Drop

Estimating the tube side pressure drop is done taking in account the viscosity correction factor as below:

```
jf = 4.5e-3; % friction factor
m = 0.14; % turbulent flow
P_tube = N_p * (8*jf*L_tube/d_in*(visc_water_l/visc_water_w)^(-m) + 2.5) ...
    * rho_water_l * velocity_water^2 /2 % Pa
```

```
P_tube = 1.2378e+05
```

The obtained value for the pressure drop, approximately 124 kPa, is higher than the permissible limit of 70 kPa. Hence, we change the design values to incorporate a lower pressure drop.

3**. Initial Overall Heat Transfer Coefficient

We use a better guess for the overall heat transfer coefficient as:

```
U_0_init = 527; % W/m^2K
```

5**. Determination of Heat Transfer Area

```
A0 = Q_dot / (U_0_init*T_lmtd) % m^2
```

```
A0 = 1.0359e+04
```

6**. Determination of Design Parameters

We reduce the length of tubing, thereby reducing the pressure drop.

```
d_out = 0.019; t_wall = 0.0017; % m
```

```
L_tube = 6.1; % m
N_p = 8;
pitch_tube = 1.25*d_out % pitch in m
```

```
pitch_tube = 0.0238
```

```
d_in = d_out - 2*t_wall % inner diameter in m
```

```
d_in = 0.0156
```

7.1**. Determination of Number of Tubes

However, we see an increase in the number of tubes and hence, the number of heat exchangers.

```
N_tubes = A0 / (pi*d_out*L_tube) % number of tubes
```

```
N_tubes = 2.8451e+04
```

```
N_t_hex = 1166;
N_hex = ceil(N_tubes/N_t_hex)
```

```
N_hex = 25
```

7.2**. Determination of Tube Velocity

```
velocity_water = 4*m_water*N_p/...
    (rho_water_l*pi*(d_in^2)*N_tubes) % m/s
```

```
velocity_water = 1.0006
```

The determined value of velocity is very close to the recommended minimum bound for liquid flow in tubes.

```
reynolds = rho_water_l*velocity_water*d_in/visc_water_l
```

```
reynolds = 1.1262e+04
```

8**. Calculation of Shell Diameter

```
K1 = 0.0331; % for 8 tube passes
n1 = 2.643;
d_bundle = d_out*(N_t_hex/K1)^(1/n1) % m
```

```
d_bundle = 0.9979
```

The assumed values of design align almost perfectly with the industrial standards for shell sizes.

```
tolerance = (10 + (20-10)*(d_bundle - 0.2)/(1.2 - 0.2))/1000 % m
```

```
tolerance = 0.0180
```

```
d_shell_i = d_bundle + tolerance % m
```

```
d_shell_i = 1.0159
```

```
N_center = floor(d_bundle / pitch_tube)
```

```
N_center = 42
```

9.1**. Estimation of Tube Side Heat Transfer Coefficient

```
T_water_mean = (T_in_water + T_out_water)/2; % K
T_wm_C = T_water_mean - 273.15; % °C
```

The corrected value for the tube side heat transfer coefficient is:

```
h_i = (4200*(1.35+0.02*T_wm_C)*(velocity_water^0.8))/((d_in*1000)^0.2) % W/m^2K
```

```
h_i = 3.6618e+03
```

9.2**. Estimation of Tube Side Heat Transfer Coefficient - Kern's Method

```
prandtl = Cp_water*visc_water_l/k_water_l;
E = 0.0225*exp(-0.0225*(log(prandtl))^2);
stanton = E * (reynolds^-0.205) * (prandtl^-0.505);
jh = stanton*prandtl^0.67;
```

Taking into account the redefined value for wall film viscosity as:

```
T_wall_tube = T_mean_tube + U_0_init*(T_mean_shell - T_mean_tube)/h_i % K
```

```
T_wall_tube = 281.7096
```

```
visc_water_w = 1.362e-3; % Pa s
```

Hence, we obtain a better estimate for the tube side heat transfer coefficient.

```
h_i_kern = k_water_l*jh*reynolds*(prandtl^0.33)*((visc_water_l/visc_water_w)^0.14)/d_in %
W/m^2K
```

```
h_i_kern = 3.8620e+03
```

10**. Shell Side Heat Transfer Coefficient

```
baffle_s = d_shell_i; % m
U_amm_init = 7275; % W/m^2K
```

For the redefined design parameters and hence, the recalculated temperatures, we define the shell side fluid mean properties.

```
T_wall = T_mean_shell - (T_mean_shell - T_mean_tube)*U_0_init/U_amm_init % K
```

```
T_wall = 284.8589
```

```
T_amm_con = (T_mean_shell + T_wall)/2 % K
```

```
T_amm_con = 285.0044
```

```
visc_amm_c = 1.5014e-4; % Pa s
rho_amm_c = 622.00; % kg/m^3
k_amm_c = 0.52363; % W/mK
rho_amm_v = 5.1736; % kg/m^3
gamma = m_amm/(L_tube*N_t_hex) % kg/ms
```

```
gamma = 0.0020
```

```
N_r = 2*N_center/3
```

```
N_r = 28
```

Hence, we evaluate the shell side heat transfer coefficient as below.

```
h_c_shell = 0.95 * k_amm_c *((rho_amm_c*(rho_amm_c-rho_amm_v)*9.81)/...
    (visc_amm_c*gamma))^(1/3) * (N_r^(-1/6)) % W/m^2K
```

```
h_c_shell = 6.6665e+03
```

11**. Overall Heat Transfer Coefficient

```
fouling_water = 1000; % W/m^2K
fouling_ammonia = 5000; % W/m^2K
k_steel = 57; % W/mK
```

We substitute the recalculated values to determine the value for the overall heat transfer coefficient.

```
val = (h_c_shell^-1) + (fouling_ammonia^-1) + d_out * log(d_out/d_in) / (2*k_steel) +
d_out/(d_in*fouling_water) + d_out/(d_in*h_i);
U1 = val^-1 % W/m^2K
```

```
U1 = 517.2173
```

The obtained value is very close to the assumed value.

Calculating Error

```
Error = (U_0_init - U1)/U_0_init*100 % in percentage
```

```
Error = 1.8563
```

The error between the assumed value and the guess is about 1.8% which is again well within the permissible limits.

12.1. Tube Side Pressure Drop

```
jf = 4.5e-3; % friction factor
m = 0.14; % turbulent flow
```

However, recalculating the tube side pressure drop gives us:

```
P_tube = N_p * (8*jf*L_tube/d_in*(visc_water_l/visc_water_w)^(-m) + 2.5) ...
    * rho_water_l * velocity_water^2 /2 % Pa
```

```
P_tube = 6.6235e+04
```

The obtained value for the tube side pressure drop is about 66.2 kPa which is acceptable. Hence, we can proceed to determine the shell side pressure drop.

12.2. Shell Side Pressure Drop - Kern

A few geometrical and flow properties of the shell side are calculated beforehand.

```
A_s = (pitch_tube - d_out)*d_shell_i*baffle_s/(pitch_tube); % m^2
```

```
mass_velocity_shell = m_amm/A_s; % kg/s
velocity_shell = mass_velocity_shell/rho_amm_l % m/s
```

```
velocity_shell = 0.1091
```

```
d_e = 1.27*(pitch_tube^2 - 0.785*(d_out^2))/d_out; % equivalent diameter in m
reynolds_shell = velocity_shell*d_e*rho_amm_l/visc_amm_l
```

```
reynolds_shell = 8.4877e+03
```

```
Cp_amm = 4.6886e3; % J/kgK
prandtl_amm = Cp_amm*visc_amm_l/k_amm_l;
```

From the determined flow properties, the friction and heat transfer factors are determined from the charts as:

```
jh_shell = 5.7e-3;
jf_shell = 5e-2;
```

Before estimating the shell-side pressure drop, we estimate the viscosity correction factor as below:

```
h_i_shell = k_amm_l*jh_shell*reynolds_shell*(prandtl_amm^0.33)/d_e; % W/m^2K
T_wall_tube = T_mean_tube + U_0_init*(T_mean_shell - T_mean_tube)/h_i_shell % K
```

```
T_wall_tube = 282.5553
```

```
visc_amm_w = 1.5397e-4; % Pa s
```

Now that we have the correction term, we can estimate the shell side heat transfer coefficient as:

```
h_i_shell = k_amm_l*jh_shell*reynolds_shell*(prandtl_amm^0.33)*...
    ((visc_amm_l/visc_amm_w)^0.14)/d_e; % W/m^2K
```

From the above value, the pressure drop can be calculated as below:

```
P_shell = 4*jf_shell*(d_shell_i/d_e)*(L_tube/baffle_s)*(rho_amm_l*...
    (velocity_shell^2)/2)*((visc_amm_l/visc_amm_w)^(-0.14)) % Pa
```

```
P_shell = 241.4533
```

It can be seen that the pressure drop is well within the permissible limits.

12.3 Shell Side Pressure Drop - VDI

The VDI method is an alternate method to estimate the pressure drop, taking into account a lot of geometric factors. The values required for this method are first initialised.

Known values

```
mu = visc_amm_l; % Pa s
mu_w = visc_amm_w; % Pa s
rho = rho_amm_l; % kg/m^3
Re = reynolds_shell;
```

The above are the values for the thermodynamic properties of ammonia at the mean temperatures. Now, we calculate the design parameters.

```
S = baffle_s; % m
L = L_tube; % m
```

```
p_t = pitch_tube; % m
D_B = d_bundle; % m
D_i = d_shell_i; % m
d_o = d_out; % m
n_T = N_t_hex;
```

The tolerance between the shell and the tube bundle is given by a slightly larger value than the pitch and hence, it can be estimated as:

```
D_1 = d_shell_i - 9.5e-3/2; % m
d_B = d_out + 8e-3; % m
```

As we have condensation taking place, we utilise differently sized inlet and outlet nozzles, with the subscript 1 referring to the inlet nozzle and 2 the outlet nozzle.

```
d_N1 = 0.6; % m
d_N2 = 0.2; % m
```

Also, we assume a 25% baffle cut which gives us the following cut height.

```
H = 0.25*D_1; % m
```

The number of tube rows at various configurations is calculated as below:

```
n_W = ceil(0.39*n_T);
n_S = 2;
n_B = floor(L/S + 1);
n_d = floor(D_B/p_t);
n_MR = floor((D_1 - 2*H)/p_t)
```

```
n_MR = 21
```

Pressure Drop in Central Cross Flow Section

The pressure drop in the central cross flow section is determined as below.

```
e = p_t - d_o; % m
e_1 = (D_i - n_d*d_o -(n_d-1)*e)/2; % m
n_MRE = floor((D_1-2*e_1-d_o)/p_t - 2);
L_E = 2*e_1 + (n_d-1)*e; % m
A_E = S*L_E; % m^2
V = m_amm/rho; % m^3
w_e = V/A_E; % m/s
```

The equivalent Reynolds number in the central region is

```
Re_1 = w_e*d_o*rho/mu;
```

We estimate the effect of laminarity on the drag coefficient as:

```
a = p_t/d_o;
b = p_t/d_o;
f_alf = 280*pi*((b^0.5 - 0.6)^2 + 0.75)/((4*a*b - pi)*(a^1.6));
xi_lam = f_alf/Re_1;
```

Then, we do the same for turbulence as:

```
f_atf = (0.22+1.2*((1-(0.94/b))^0.6)/((a-0.85)^(1.3)))*10^(0.47*((b/a)-1.5)) ...
    + (0.03*(a-1)*(b-1));
xi_turb = f_atf/(Re_1^(0.1*b/a));
```

We also incorporate a correction factor for both the drag coefficients.

```
f_zl = (mu_w/mu)^(0.57/(((4*a*b/pi)-1)*Re)^0.25);
f_zt = (mu_w/mu)^0.14;
```

Hence, we have the equivalent drag coefficient as:

```
xi = xi_lam*f_zl + xi_turb*f_zt*(1-exp((- Re_1 - 1000)/2000));
```

Now, we can estimate the pressure drop in the absence of leaking and bypass as:

```
p_Q0 = xi*n_MR*rho*(w_e^2)/2; % Pa
```

However, to take the effect of leaking and bypass into consideration, we first require different cross-sectional areas given below:

```
gamma = 2*360*(acos(1-(2*H)/D_1)/pi);
A_GSB = (360-gamma)*pi*(D_i^2 - D_1^2)/(4*360); % m^2
A_GTB = (n_T - n_W/2)*pi*(d_B^2 - d_o^2)/4; % m^2
A_SG = A_GTB + A_GSB; % m^2
```

With the above, we can estimate the leakage correction factor as:

```
R_L = A_SG / A_E;
R_M = A_GSB / A_SG;
r = (-0.15*(1+R_M)+0.8);
f_L = exp(-1.33*(1+R_M)*(R_L^r))
```

```
f_L = 0.2143
```

The bypass correction factor is determined in a similar fashion as below:

```
A_B = S*(D_i - D_B - e) % m^2
```

```
A_B = 0.0134
```

```
R_S = n_S/n_MR;
R_B = A_B / A_E;
f_B = exp(-3.7*R_B*(1-(2*R_S)^(1/3)));
```

Hence, we have the net pressure drop in the central section as:

```
p_Q = p_Q0*f_L*f_B/2 % Pa
```

```
p_Q = 3.1573
```

Pressure Drop in End Cross Flow Section

The pressure drop in the end cross flow depends on the ratio of the number of rows at the end section and the total number of rows in the central cross section as below. We also incorporate a similar bypass correction factor and we get:

```
p_QE0 = p_Q0*n_MRE/n_MR; % Pa
p_QE = p_QE0*f_B/2 % Pa
```

```
p_QE = 26.6652
```

Pressure Drop in Window Section

We start the estimation of this section by determining the corresponding cross-sectional areas as below:

```
n_MRW = 0.8*H/p_t;
A_WT = pi*(D_i^2)*gamma/(4*360) - (D_1 - 2*H)*D_1*(sin(gamma/2))/4 % m^2
```

```
A_WT = 0.4662
```

```
A_T = pi*(d_o^2)*n_W/(8); % m^2
A_W = A_WT - A_T; % m^2
```

We also require a value for the wetted perimeter for which we use:

```
U_W = pi*D_i*gamma/360 + pi*d_o*n_W/2; % m
```

Now, the equivalent diameter for the flow in the window section is:

```
d_g = 4*A_W/U_W; % m
```

Now, we can estimate the flow properties as below:

```
w_p = V/A_W; % m/s
w_z = (w_e*w_p)^(0.5); % m/s
```

Considering both the laminar and the turbulent effects, we get:

```
p_W_lam = ((56*n_MRW*mu)/(e*w_z*rho) + (52*mu*S)/((d_g^2)*w_z*rho) + 2)*rho*(w_z^2)/2 % Pa
```

```
p_W_lam = 4.1417
```

```
p_W_turb = (0.6*n_MRW + 2)*rho*(w_z^2)/2 % Pa
```

```
p_W_turb = 12.6006
```

We can then estimate the net pressure drop in the window section as:

```
p_W = ((p_W_lam^2 + p_W_turb^2)^(0.5))*f_L*f_zt/2 % Pa
```

```
p_W = 1.4263
```

Pressure Drop in Inlet and Outlet Nozzles

Ammonia enters the condenser as a gas and exits as a liquid. Hence, we incorporate a differing nozzle diameter.

```
rho_1 = rho_amm_v; % kg/m^3
rho_2 = rho_amm_l; % kg/m^3
```

The flow velocities can be determined from the flow rate as:

```
w_N1 = 4*V/(pi*(d_N1^2)); % m/s
w_N2 = 4*V/(pi*(d_N2^2)); % m/s
```

The cross-sectional area ratios are given by:

```
A_NF1 = (d_N1^2)/((D_i^2)-n_T*(d_o^2));
A_NF2 = (d_N2^2)/((D_i^2)-n_T*(d_o^2));
```

Now, we determine the nozzle drag coefficients individually as below:

```
xi_N1 = 3.308*(A_NF1^1.14)*(d_N1/D_i)*(D_B/d_N1)^2.4;
xi_N2 = 2.482*(A_NF2^1.14)*(d_N2/D_i)*(D_B/d_N2)^2.4;
```

Hence, we can estimate the total pressure drop at the nozzles as:

```
p_N = xi_N1 * rho_1 * (w_N1^2) /2 + xi_N2 * rho_2 * (w_N2^2) /2 % Pa
```

```
p_N = 165.1703
```

Total Pressure Drop

As condensation occurs at the shell side, we take 50% of the pressure drop as determined for a single-phase system considering ammonia as a liquid. However, since we incorporated the difference in densities for the pressure drop at the nozzles, we can take the corresponding value as it is. Hence, the total pressure drop is the sum of 50% of the pressure drop in the central section, window section and the end section and the pressure drop at the nozzles.

```
P_tube_VDI = (n_B - 1)*p_Q + 2*p_QE + n_B*p_W + p_N % Pa
```

```
P_tube_VDI = 247.4287
```

It can be seen that the determined pressure drop is very close to the value determined in the previous section.

13. Cell Method to Verify Temperatures

The outlet temperature of coolant water is to be verified for this design. Hence, we try to determine the temperature in the tubing by dividing the shell into control volumes divided into tube rows, in this case, 28 rows.

```
m_dot = 680; % kg/s
Cp = 4205.5; % J/kgK
A = pi*d_out*L_tube; % m^2
```

The film thickness will vary as we go down the rows. Hence, the heat transfer coefficient for the film is to be redetermined in each case, keeping the rest constant.

```
U_constant = (fouling_ammonia^-1) + d_out * log(d_out/d_in) / (2*k_steel) +...
    d_out/(d_in*fouling_water) + d_out/(d_in*h_i);
hcs_constant = 0.95 * k_amm_c*(rho_amm_c*(rho_amm_c - rho_amm_v)*9.81)/...
    (visc_amm_c*gamma)^(1/3);
```

We know that the temperature of ammonia is constant at 12°C and the inlet temperature of water is constant at 5°C. Also, we know the number of center rows as 28.

```
T_in = 278.15; % Water inlet temperature in K
T_in_amm = 285.15; % K
T_out_amm = 285.15; % K
N_r = 28;
```

We iterate through the rows, determining the temperature at each cell.

```
for i=1:N_r
hcs = hcs_constant*(i^(-1/6)); % W/m^2K
U = 1/((1/hcs) + U_constant); % W/m^2K
T_out_guess = 273; % K
T_out = fzero(@(T_out) U*i*(13883/232)*A*((T_in_amm - T_out) -...
    (T_out_amm - T_in))/log((T_in_amm - T_out)/(T_out_amm - T_in)) -...
    m_dot*Cp*(T_out - T_in), T_out_guess);
T_in = T_out;
To(i) = T_out - 273.15;
end
```

```
T_out % K
```

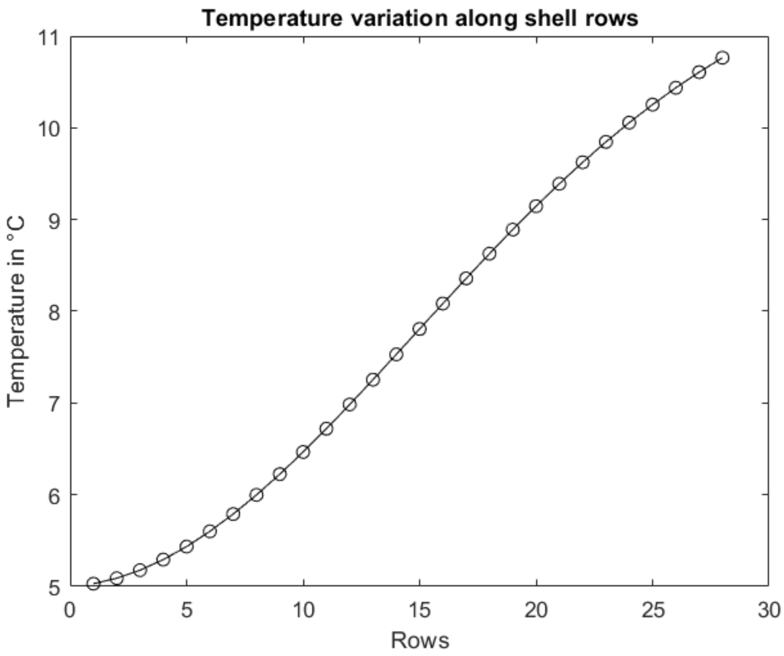
```
T_out = 283.9142
```

```
error_temp = abs(T_out-T_out_water)/(T_out_water-273.15)*100 % in percentage
```

```
error_temp = 1.8087
```

We visualize the variation in temperature along the rows using the plot function.

```
x = linspace(1,N_r,N_r);
figure();
plot(x,To,'-ok');
xlabel('Rows');
ylabel('Temperature in °C');
title('Temperature variation along shell rows');
```



14. Cost Estimation

The 2010 model is used for estimating total costs. The coefficients are taken from the table in Sinnott and Towler. We get the following:

```
a = 28000; b = 54; n = 1.2;
S = A0/N_hex;
```

The total cost per heat exchanger is proportional to the heat transfer area, and hence, a simple power law is employed.

```
C = a + b * S^n; % in USD
C_T = C*N_hex;
C_T = C_T*0.92; % in EUR
```

The obtained cost is multiplied by the inflation factor of 1.14 to obtain the cost of equipment in euros in 2020 as

```
Cost_equipment = C_T * 1.14 % in EUR
```

```
Cost_equipment = 2.6925e+06
```

The total capital costs associated with the heat exchanger includes the installation and maintenance costs which have a multiplying factor of 3.5. Hence, the total cost is

```
Cost_total = Cost_equipment*3.5 % in EUR
```

```
Cost_total = 9.4238e+06
```

The above costs are nominal considering the heat exchangers are industrial grade and heavy duty.