Thermal Systems ENGR-UH 4710

Fall 2022

Assignment	: **Project 1: Heat Exchanger Design**
Date	: **06/10/2022**
Section Number	: **01**
Instructor	: **Prof. Sunil Kumar**

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1. Problem Statement and Design Concept

1.1 Problem Statement

As a coolant for an automobile engine, Ethylene Glycol (50% diluted with deionized water) exits the engine at 90°C and flows through a heat exchanger that should cool the fluid to 60°C. The coolant flows at 1.2kg/s and is cooled using ambient air at 30°C. The Glycol side is susceptible to fouling, and the flow inlet and outlet headers of the heat exchanger have been ignored for the purpose of this design.

1.2 Design Concept

The chosen heat exchanger is the tube-fin type with plate fins as seen from Fig.1 below. This type of heat exchanger was chosen because of its efficiency with air as a coolant. The tubes are in-line, and single-pass flow is used to reduce the pumping power to move the coolant. The heat exchanger is to be compact (area density > $700m^2/m^3$, D_h <6.35mm) in order to fit inside the car but provide good heat transfer from the ethylene glycol to air. The two most used materials for fin-tube heat exchangers are copper and aluminum, but copper was chosen since it is more conductive and cost-effective than aluminum and would provide better heat transfer.

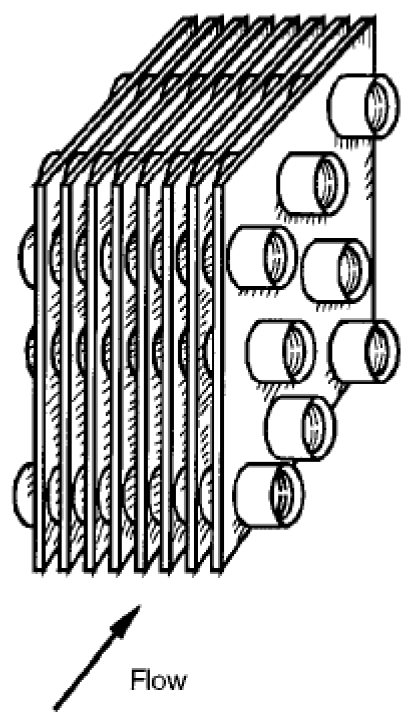


Figure 1. Tube-fin heat exchanger used

2. Final Design

Our final design is presented here to provide context for the calculations in the next section. The design resolved to be an in-line single-pass tube-fin heat exchanger with circular tubes and rectangular fins with cooling air in a crossflow arrangement. The tubes are each equally spaced according to the longitudinal pitch and transverse pitch. The dimensions for our final tube-fin heat exchanger (car radiator) design is shown below in Table 1:

Table 1: Final Design Dimensions

	Dimension	Unit	Value
Tube Dimensions	Number of Tubes (n)		600
	Number of Tube Rows (n_L)		20
	Tube Diameter	mm	6.35
	Tube Length	m	1.5411
Fin Dimensions	Number of Fins (Plates)		650
	Fin Length	mm	590.55
	Fin Width	mm	266.7
	Fin Thickness	mm	1
Pitch (Spacing)	Longitudinal Pitch	mm	19.05
	Transverse Pitch	mm	25.40
Sizing	Volume	m^3	0.10238
	Nominal Surface Area	m^2	189.770
	SA:V	m^2/m^3	1853.67

The material for the proposed heat exchanger was selected to be copper for its high thermal conductivity properties heat exchanger, though it is relatively more expensive than our initial proposed material, aluminum.

As the fins are assumed to have negligible thickness, the volume of the heat exchanger can be found to be the volume of the fins involved. The nominal surface area exposed to air was also calculated and the area density derived. The volume, surface area and surface area to volume ratio (SA:V) of our heat exchanger was calculated and also shown in Table 1 above. Typical compact heat exchangers should have a surface area to volume ratio upwards of 700 m^2/m^3 [1]. Thus, our heat exchanger design satisfies the condition of being a compact heat exchanger.

The schematics of our final heat exchanger can be seen in Appendix 7.2.

3. Assumptions and Calculations

Some assumptions were made in order to simplify the problem and the resulting calculations. The tube thickness is assumed to be negligible since they are thin-walled and highly conductive, so the thermal resistance of the tube is negligible. Flow is assumed to be steady and fully developed with constant properties and fouling factors. Fouling exists only on the Ethylene Glycol side. Heat loss to the surroundings is negligible as are the changes in the kinetic and potential energies of fluid streams.

The design principle was based on finding the heat transfer rate using the mathematical methods of calculating heat transfer, and then using the physics of the problem to match that value with the appropriate design. This method will be demonstrated below in depth. We chose to fix the temperature difference of air as 20°C.

Given ethylene glycol (50:50 water) as the hot fluid and air as the cold fluid, we obtain the below properties for each fluid at the bulk mean temperature (average of inlet and outlet temperature). The properties of air are obtained from the textbook, *Heat and Mass Transfer* while the properties of ethylene glycol are obtained from "The Engineering Toolbox" website [2].

Ethylene Glycol (T_{bm} = 75°C):

$$T_{ethyl,in} = 90°C$$

$$T_{ethyl,out} = 60°C$$

$$C_{p,ethyl} = 3641.5\ J/Kg°C$$

$$\mu_{ethyl} = 9.06 \times 10^{-4}\ Kg/ms$$

$$K_{ethyl} = 0.3947\ W/mK$$

$$Pr_{ethyl} = 9.82$$

$$\rho_{ethy;} = 1045\ Kg/m^3$$

$$\dot{m}_{ethyl} = 1.2\ Kg/s$$

$$\dot{m}_{ethyl/tube} = \frac{1.2}{Number\ of\ Tubes} Kg/s$$

Air (T_{bm} = 40°C):

$$T_{air,in} = 30°C$$

$$T_{air,out} = T_{air,in} + 20°C = 50°C$$

$$C_{p,air} = 1008\ J/Kg°C$$

$\mu_{air} = 1.918 \times 10^{-5}$ Kg/ms

$K_{air} = 0.02662$ W/mK

$Pr_{air} = 0.7255$

$\rho_{air} = 1.127$ Kg/m^3

$\dot{m}_{air} = 6.5027\ Kg/s$

$$\boldsymbol{\dot{Q}_{math} = 131.094\ KW}$$

To find the mass flow rate of air, the below equation for conservation of energy was used,

$$\dot{Q}_{math} = \dot{m}_{ethyl} \times c_{p,ethyl} \times (T_{ethyl,out} - T_{ethyl,in})$$

$$= \dot{m}_{air} \times c_{p,air} \times (T_{air,out} - T_{air,in}) = 1.2 \times 3641.5 \times (90 - 60) = 131.094 \text{ KW}$$

$$\dot{m}_{air} = \frac{\dot{Q}_{math}}{c_{p,air} \times (T_{air,out} - T_{air,in})} = \frac{131094}{1008 \times (50 - 30)} = 6.5027\ Kg/s$$

Using these properties and the specifications of the final design in section 2 above, the procedure below was used to validate the design's operation and validity by comparing $\dot{Q}_{physics}$ with $\dot{Q}_{math}$.

Log Mean Temperature Difference (LMTD)

$$LMTD = \frac{(T_{ethyl,in} - T_{air,out}) - (T_{ethyl,out} - T_{air,in})}{\log\left((T_{ethyl,in} - T_{air,out}) - (T_{ethyl,out} - T_{air,in})\right)} = 34.7606$$

To get the corrected LMTD,

$$P = \frac{T_{ethyl,out} - T_{ethyl,in}}{T_{air,in} - T_{ethyl,in}} = 0.50$$

$$R = \frac{T_{air,in} - T_{air,out}}{T_{ethyl,out} - T_{ethyl,in}} = 0.6667$$

From these values, we can read the table and find the correction factor to be $F = 0.95$

$$LMTD_{corrected} = F \times LMTD = 0.95 \times 34.7606 = 33.0226$$

Convective Heat Transfer Coefficient Inside the Tube

To calculate the convective heat transfer coefficient inside the tube, we first find Reynolds number for glycol inside the tube, Re_{ethyl}. We need V_{ethyl} first,

$$V_{ethyl} = \frac{4 \times \dot{m}_{ethyl/tube}}{\rho_{ethyl} \times \pi \times {D_{tube}}^2} = 0.0604 \text{ m/s}$$

$$Re_{ethyl} = \frac{\rho_{ethyl} \times V_{ethyl} \times D_{tube}}{\mu_{ethyl}} = \frac{1045 \times 0.0604 \times 0.00635}{9.06 \times 10^{-4}} = 442.63$$

Since Re_{ethyl} is less than 2300, the formula for Nusselt number for constant surface heat flux is

$$Nu = \frac{h_{inside} D_{tube}}{k_{ethyl}} = 4.36$$

Rearranging to solve for h_{inside},

$$h_{inside} = \frac{0.3947 \times 4.36}{0.00635} = 271.0 \text{ W/m}^2\text{K}$$

Convective Heat Transfer Coefficient Outside the Tube

$$V_{air} = \frac{\dot{m}_{air}}{\rho_{air} \times Fin\ Length \times Tube\ Length} = \frac{6.5027}{1.127 \times 0.5905 \times 1.5411} = 6.34 \text{ m/s}$$

$$V_{air,max} = \frac{S_T}{S_T - D} \times V_{air} = \frac{0.0254}{0.0254 - 0.00635} \times 6.34 = 8.4532 \text{ m/s}$$

Then, Re_{air} can be found,

$$Re_{air} = \frac{\rho_{air} \times V_{air} \times D_{tube}}{\mu_{air}} = \frac{1.127 \times 8.4532 \times 0.00635}{1.918 \times 10^{-5}} = 3154.1$$

The corresponding equation for this Reynolds number range is

$$Nu_{air} = 0.27 \times {Re_{air}}^{0.63} \times {Pr_{\mathrm{air}}}^{0.36} = 38.502$$

$$Nu_{air,corrected} = Row\ number\ correction\ factor \times Nu_{air} = 1 \times 38.502 = 38.502$$

$$h_{outside} = \frac{38.502 \times 0.02662}{0.00635} = 161.4042\ W/m^2K$$

Total Thermal Resistance Calculation

A. <u>Area Calculations</u>

To calculate the total thermal resistance, we first need to find the relevant areas which are the inner area and the surface area of the heat exchanger.

$$A_{inside} = N_{tubes} \times \pi \times D_{tube} \times Tube\ length = 600 \times \pi \times 0.00635 \times 1.5411 = 18.446\ m^2$$

Then, the surface area is shown below where η is the fin efficiency and N_{tubes} is the number of tubes and N_{fins} is the number of fins,

$$A_{total\ surface} = A_{unfinned} + (\eta \times A_{finned})$$

$$A_{unfinned} = N_{tubes} \times \left((\pi \times D_{tube} \times tube\ length) - (N_{fins} \times \pi \times D_{tube} \times tube\ thickness)\right) = 10.666\ m^2$$

$$A_{finned} = N_{fins} \times \big(2 \times (fin\ length \times fin\ width) + (fin\ width \times fin\ thickness) + (fin\ length \times fin\ thickness)\big) - \frac{(2 \times N_{fins} \times \pi \times D_{tube}{}^2)}{4}) = 179.1035\ m^2$$

Next, we need the efficiency to calculate the total surface area,

$$\eta = \tanh\left(\frac{mL_c}{mL_c}\right)$$

$$m = \sqrt{\frac{2 \times h_{outside}}{K_{Cu} \times fin\ thickness}} = \sqrt{\frac{2 \times 161.4042}{401 \times 1.0 \times 10^{-03}}} = 28.3727$$

Taking one unit-cell to be the area around one tube and knowing that the tubes are evenly distributed from the sides and from each other, we can define the area of one unit-cell to be:

$$A_{unit\ cell} = \frac{fin\ length\ \times fin\ width}{N_{tubes}} = \frac{0.5905 \times 0.2667}{600} = 2.625e^{-04}\ m^2$$

Thus, one side of the unit cell would be

$$unit\ cell\ side = \sqrt{A_{unit\ cell}} = \sqrt{2.625e^{-04}} = 0.016202\ m$$

$$L_c = \frac{unit\ cell\ side - D_{tube}}{2} + \frac{fin\ thickness}{2} = \frac{0.016202 - 0.00635}{2} + \frac{1.0\times10^{-03}}{2} = 0.00543\ \text{m}$$

$$\eta = \tanh\left(\frac{28.3727 \times 0.00543}{28.3727 \times 0.00543}\right) = 0.992$$

Thus, the total surface area becomes,

$$A_{total\ surface} = 10.666 + (0.992 \times 179.1035) = 188.3679\ m^2$$

B. <u>Total Thermal Resistance</u>

Considering the fouling factor for ethylene glycol to be $R_f = 0.00035$,

$$R_{total} = \frac{1}{h_{inside} \times A_{inside}} + \frac{1}{h_{outside} \times A_{total\ surface}} + \frac{R_f}{A_{inside}}$$
$$= \frac{1}{271.0 \times 18.446} + \frac{1}{161.4042 \times 188.3679} + \frac{0.00035}{18.446} = 0.6667$$

<u>Overall Heat Transfer Coefficient</u>

$$R_{total} = \frac{1}{U_{physics} \times A_{total\ surface}}$$

Thus,

$$U_{physics} = \frac{1}{R_{total} \times A_{total\ surface}} = 21.075\ W/m^2K$$

From the first step, we know that

$$\dot{Q}_{math} = 131.094\ KW$$

So,

$$U_{math} = \frac{\dot{Q}_{math}}{A_{total\ surface} \times LMTD_{corrected}} = \frac{131094}{188.3679 \times 33.0226} = 21.075 W/m^2K$$

Therefore, we have achieved the required design since $\boldsymbol{U_{physics} = U_{math} = 21.075 W/m^2K}$.

<u>Calculating NTU</u>

To calculate NTU, we first find C_{min}, which is the smaller value between C_{ethyl} and C_{air} calculated below.

$$C_{ethyl} = \dot{m}_{ethyl} \times c_{p,ethyl} = 4.3698e^{+03}\ kW/K$$

$$C_{air} = \dot{m}_{air} \times c_{p,air} = 6.5547e^{+03}\ kW/K$$

Thus, $C_{min} = C_{ethyl} = 4.3698e^{+03}\ kW/K$

$$NTU = \frac{U_{physics} \times A_{total\ surface}}{C_{min}} = \frac{21.075 \times 188.3679}{4.3698e^{+03}} = 0.9085$$

<u>Calculating Effectiveness</u>

Two methods for calculating effectiveness were used and both gave similar results, so we will present only one method. The first is the general formula (which is shown below), while the second is specified for cross-flow heat exchangers (single-pass) with both fluids unmixed. Both gave values of around 0.5.

$$\varepsilon = \frac{\dot{Q}_{physics}}{\dot{Q}_{max}} = \frac{U_{physics} \times A_{total\ surface} \times LMTD_{corrected}}{C_{min} \times \Delta T_{max}} = \frac{21.075 \times 188.3679 \times 33.0226}{4.3698e^{+03} \times (90-30)} = 0.500$$

<u>Calculating Pressure Drop</u>

Pressure drop can be calculated using the formula below, where $f = 0.17$ is the friction factor and $\chi = 0.8$ is the correction factor. Both factors are obtained from their plots against Reynolds number based on $V_{air,max}$.

$$\Delta P = N_{rows} f \chi \frac{\rho_{air} {V_{air,max}}^2}{2} = 20 \times 0.17 \times 0.8 \times \frac{1.1270 \times 8.4532^2}{2} = 109.522\ Pa$$

<u>Calculating Pumping Power</u>

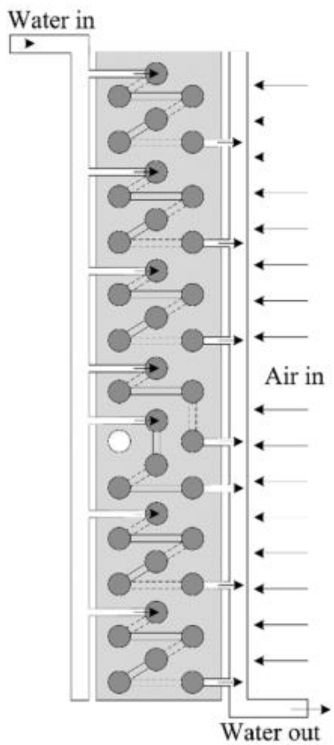
The pumping power is shown below and is dependent on the pressure drop above.

$$\dot{W}_{pump} = \frac{\dot{m}_{air} \times \Delta P}{\rho_{air}} = \frac{6.5027 \times 109.522}{1.1270} = 631.9321W$$

4. Preliminary Design and Optimization

4.1 Preliminary Design

The inspiration for the original design was initially drawn from literature review. As such, our initial dimensions were slight modifications to the HEX:3R-7C tube-fin heat exchanger designed by the University of Rijeka, Croatia [3]. The schematics for the HEX:3R-7C heat exchanger can be observed in Figure 2 below.



Heat exchanger	HEX:3R-7C
Overall height, mm	840
Overall length, mm	760
Overall depth, mm	155
Tube fluid flow length, mm	700
Fin depth, mm	90
Fin height, mm	780
Number of fins, -	260
Fin pitch, mm	2.6
Fin thickness, mm	0.2
Fin type	flat plain
Fin material	aluminum
Air-side heat transfer area, m^2	40
Surface compactness, m^2/m^3	814
Number of rows	3
Transverse tube pitch, mm	60
Longitudinal tube pitch, mm	30
Outer tube diameter, mm	15.9
Inner tube diameter, mm	14.8
Tube material	copper

Figure 2: Dimensions and Specifications of HEX:3R-7C Heat Exchanger

Thus, the dimensions and specifications for the initial tube-fin heat exchanger (car radiator) design are shown below in Table 2 and 3 respectively. The initial diameter was set based on the standard copper tube sizes for heat exchangers [4].

Table 2: Initial Design Dimensions

	Dimension	Unit	Value
Tube Dimensions	Number of Tubes (n)		100
	Number of Tube Rows (n_L)		4
	Tube Diameter	mm	15
	Tube Length	m	0.5
Fin Dimensions	Number of Fins (Plates)		300
	Fin Length	mm	1140
	Fin Width	mm	195
	Fin Thickness	mm	0.2
Pitch (Spacing)	Longitudinal Pitch	mm	54.00
	Transverse Pitch	mm	58.85
Sizing	Volume	m^3	0.01334
	Nominal Surface Area	m^2	103.805
	SA:V	m^2/m^3	7782.65

Table 3: Initial Tube-Fin Heat Exchanger Design Specifications

Specification	Unit	Value
Mass Flow Rate of Air	kg/s	6.503
Required Air Velocity	m/s	10.123
Inlet Air Temperature (Temperature of Ambient Air)	°C	30
Outlet Air Temperature	°C	50
Inlet Engine Coolant Temperature	°C	90
Outlet Engine Coolant Temperature	°C	60
Overall Heat Transfer Coefficient	$W/m^2°C$	3.145
Total Heat Transfer Rate	W	8392.54
Effective Heat Transfer Area	m^2	80.797
Fin Efficiency		0.7738
Log Mean Temperature Difference (Corrected to Fit Crossflow Heat Exchangers)	°C	33.0226
Number of Transfer Units (NTU)		0.05816
Effectiveness		0.03201

4.2 Issues with Preliminary Design

The first issue with the preliminary design was the need to equate the rate of heat transfer gotten from the relationship $\dot{Q}_{expected} = \dot{m}c_p(T_{ethyl,in} - T_{ethyl,out}) = 131.094\ kW$ to the rate of heat transfer derived from our designed heat exchanger's overall heat transfer coefficient, $\dot{Q} = UA_sF\ \Delta T_{LM}$. Another way to represent this is to calculate our expected overall heat transfer coefficient, $U_{expected}$, using the expression: $U_{expected}A_sF\ \Delta T_{LM} = \dot{m}c_p(T_{ethyl,in} - T_{ethyl,out})$. Our heat exchangers derived value for U should thus be equal to $U_{expected}$. In our preliminary design, our $\dot{Q}$ value was found to be equal to 8.407 kW.

The second issue with our initial preliminary design was the low fin efficiency, effectiveness, overall heat transfer coefficient which all point towards a poor heat exchanger.

4.3 Optimization Techniques

Initially, certain values were randomly changed, and patterns were observed. The first condition to be altered was the length of the tube. For the above conditions in section 4.2 to be achieved, the tube had to be 359.23m long, which is an unreasonable value for a car radiator.

As such, the tube length was chosen to fit within range of 1m to 1.75 m, while other variables were changed. All calculations, optimization iterations and graphs were done using MATLAB R2022a. Within this fixed range for tube length, other dimensions on the heat exchanger were altered and the overall heat transfer coefficient, U, was derived. The parameters continuously changed including the longitudinal pitch (which directly affects the fin width), the transverse pitch (which directly affects the fin length), fin thickness, number of tubes, number of fins, number of rows and tube diameter. The physics derived U value was then compared to the expected U value. After each alteration, the percentage difference between the derived U and $U_{expected}$ was computed for the tube lengths of 1m to 1.75m and plotted with tube length on the x-axis and percentage difference between derived U and expected U on the y-axis. The point at which the graph cuts the x-axis shows the tube-length where U = $U_{expected}$, thus satisfying our first condition. If the graph did not cross the x-axis, our first condition was not satisfied within tube lengths of 1m to 1.75m. This design failure can be observed in Figure 3 below. Hence, another set of alterations was carried out on our variable design parameters until U = $U_{expected}$ was satisfied.

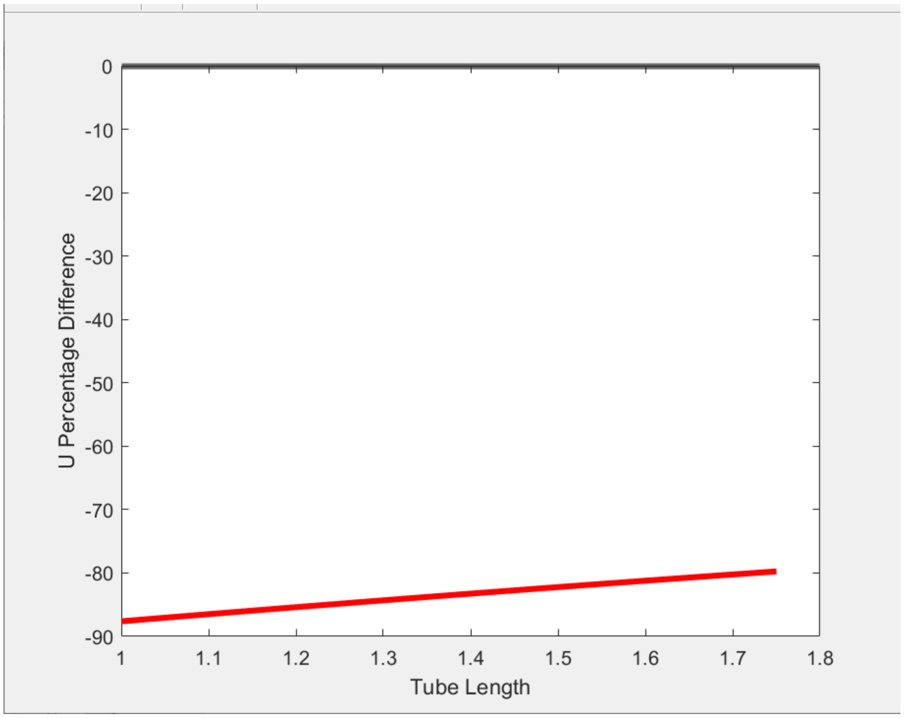


Figure 3: Design Failure as Observed in Preliminary Design Specifications within select Tube Length Range

Most parameters in the design were changed based on a trial-and-error method and by noticing trends. Some changes were inspired from a more extensive literature review. Based on research, the number of tubes were increased to 600 to suit the typical number of tubes found in a car radiator [5]. Another similar alteration based on research was the change of number of fins from 300 to 650 to maintain an optimal value of 16 fins per inch. Performance change in heat exchanger is shown to be insignificant or even detrimental with an increase in number of fins beyond 16 fins per inch [6]. The individual fin area was reduced by reducing the fin length and widths in order to improve fin efficiency. Larger fins sometimes prove inefficient as a large fraction of the external surface area might not be used for heat exchange. Other changes can be seen in Table 4 below.

Table 4: Changes in Dimensions from Initial to Final Design

Dimension	**Unit**	**Initial Design Value**	**Final Design Value**	**Nature of Change**
Number of Tubes (n)		100	600	Increase
Number of Tube Rows (n_L)		4	20	Increase
Tube Diameter	mm	15	6.35	Decrease
Number of Fins (Plates)		300	650	Increase
Fin Length	mm	1140	590.55	Decrease
Fin Width	mm	195	266.7	Increase
Fin Thickness	mm	0.2	1	Increase
Longitudinal Pitch	mm	54.00	19.05	Decrease
Transverse Pitch	mm	58.85	25.40	Decrease

Using these values, the optimal tube length was found to be 1.5411m as seen in Figure 4 below.

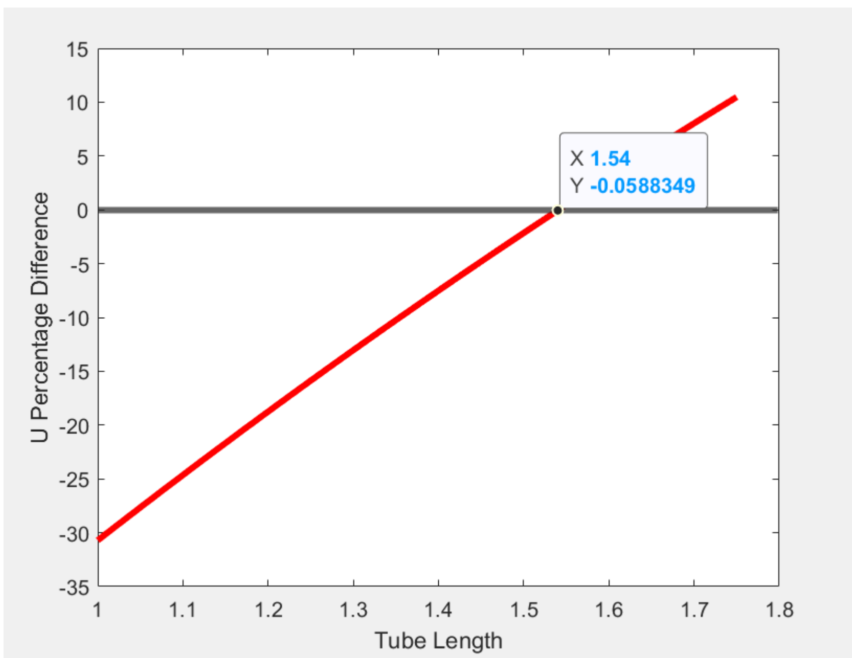


Figure 4: Derivation of Optimal Tube Length using Graphical Method (Results of Final Design)

The final design dimensions were listed earlier in Table 1 and again in Appendix 7.1. The specifications for our heat exchanger after the optimization process were calculated in Section 3 and are shown in Table 5 below.

Table 5: Final Tube-Fin Heat Exchanger Design Specifications

Specification	**Unit**	**Value**
Mass Flow Rate of Air	kg/s	6.503
Required Air Velocity	m/s	6.340
Inlet Air Temperature (Temperature of Ambient Air)	℃	30
Outlet Air Temperature	℃	50
Inlet Engine Coolant Temperature	℃	90
Outlet Engine Coolant Temperature	℃	60
Overall Heat Transfer Coefficient	W/m^2℃	21.075
Total Heat Transfer Rate	W	131092
Effective Heat Transfer Area	m^2	188.368
Fin Efficiency		0.9922
Log Mean Temperature Difference (Corrected to Fit Crossflow Heat Exchangers)	℃	33.0226
Number of Transfer Units (NTU)		0.9085
Effectiveness		0.5000

5. Performance Analyses and Discussion

5.1 Design and Off-Design Performance Evaluation

5.1.1. Design Point Performance Evaluation

Design Point Performance is the expected performance where efficiency is at its peak and where design specifications are matched perfectly. It helps assess the product's ability to do what it is expected to do.

Table 6: Design Point Performance Evaluation

Performance Evaluators	**Physics U**	**Theoretical/Math U**	**Percentage Difference (%)**
Overall Heat Transfer Coefficient, U ($\frac{W}{m^2°C}$)	21.074526	21.074881	0.001683
Heat Transfer Rate (W)	131090	131094	0.00305

Analysis:

The mathematically derived overall heat transfer coefficient matched the derivations from physics. Physics U takes into consideration the thermal resistance while Math U is compliant with the design expectations/ problem statement. With the percentage difference at zero, it is expected that the heat exchanger will perform as per the demands of the operation specifications. This is all at ambient air 30℃ .

5.1.2. Off-Design Performance Evaluation

In the real world, heat exchangers may not operate in the most optimal conditions but are expected to be able to operate with the best effectiveness regardless. Off-design performance can be assessed using different parameters such as loading, ambient temperature, or even fuel type. This report evaluates the performance of the heat exchanger at varying ambient temperatures ranging from 10℃ to 50℃ .

Table 7: Off-Design Performance Evaluation

Inlet Ambient Temperature ℃	Change from Inlet Ambient Temperature (off the design point) ℃	Exit Air Temperature (℃)	Exit Ethyl Temperature (℃)
10	-20	36.666218	50.00
20	-10	43.332941	55.00
30	0	49.999663	60
40	10	56.666386	65.00
50	20	63.333109	70.00

Analysis:

Irrespective of ambient temperatures, the proposed heat exchanger model maintains effectiveness. The expectations from the problem statement are that the coolant and as per the energy balance equation, it is expected that the outlet temperature of air should be higher as that is what is observed in Table 7 above.

The design expectations are that ethylene glycol, the coolant, should be cooled at design point specifications, to 60℃ . In off-design analysis, we explored the exit ethyl temperature at varying inlet ambient temperatures. The exit ethyl temperature increased proportionally, and this is likely because a higher temperature ambient air will not be able to cool the coolant as much as the lower temperature ambient air due to the energy balance equation.

5.2 Qualitative Discussion of Pressure Drop and Pumping Power

Since fluids are pumped through heat exchangers, it is important to know the amount of pumping power required for the system to perform at optimum rates as per design specifications. Another reason is that this determines the operational costs, and this advises the feasibility of the design proposal. Pressure drop is directly related to pumping power because as the pressure drop increases, the required power to pump the fluid increases as well.

The proposed model has a low pumping power requirement of 631.932W. This suggests that our proposed model consumes less power to operate and therefore is cost-efficient.

6. References

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7. Appendix

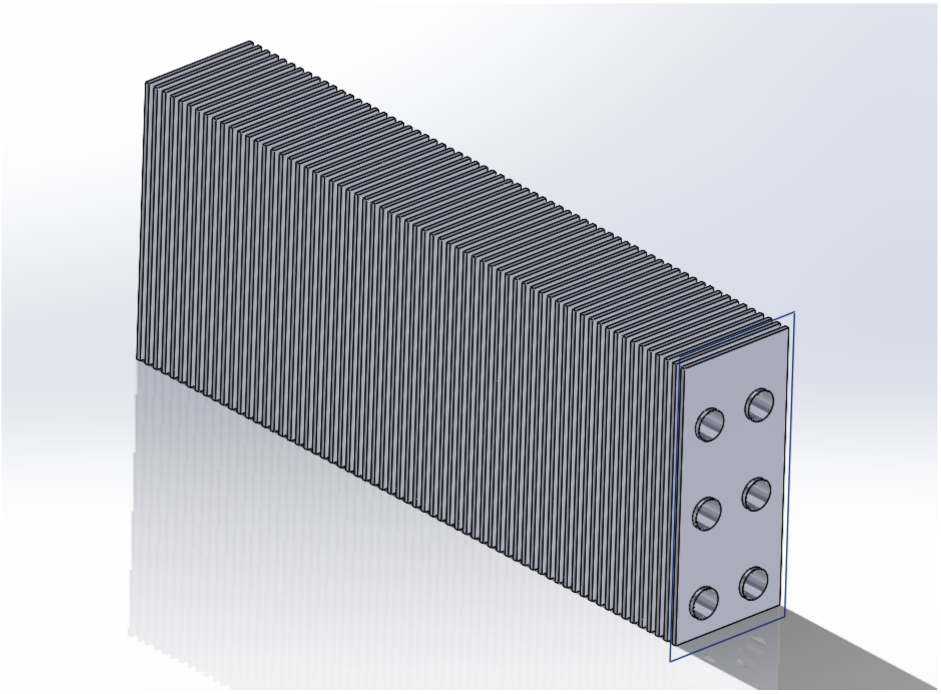
7.1. Heat Exchanger Datasheet

Parameters	**Air**	**Ethylene-Glycol (50:50)**	**Units**
Fluid type	Gas	Liquid	
Fluid Density	1.127	1045	kg/m^3
Fluid Specific heat capacity	1008	3641.5	J/(kg°C)
Fluid Thermal conductivity	0.0266	0.3947	W/mK
Dynamic Fluid viscosity	1.918e-05	9.060e-04	kg/ms
Mass Flow Rate	6.5027	1.2000	kg/s
Inlet temperature	30	90	℃
Outlet temperature	50	60	℃
Pressure drop	84.9352		Pa
Pumping Power	490.07		W
LMTD	33.0226		℃
Fouling	0.00035		m^{2}K/W
Number of fins	650		
Number of tubes	600		
Maximum Operating Temperature (Ambient Air)	90		℃
Fin Material	Copper		
Plate Material	Copper		
Tube Diameter	6.35		mm
Tube Length	1.5411		m
Number of tubes	600		
Number of Tube rows	20		
Number of fins (plates)	650		
Plate width	266.70		mm
Plate thickness	1.00		mm
Plate length	590.55		mm
Volume	0.10238		m^3
Nominal Surface Area	189.770		m^2
SA:V	1853.67		m^2/m^3

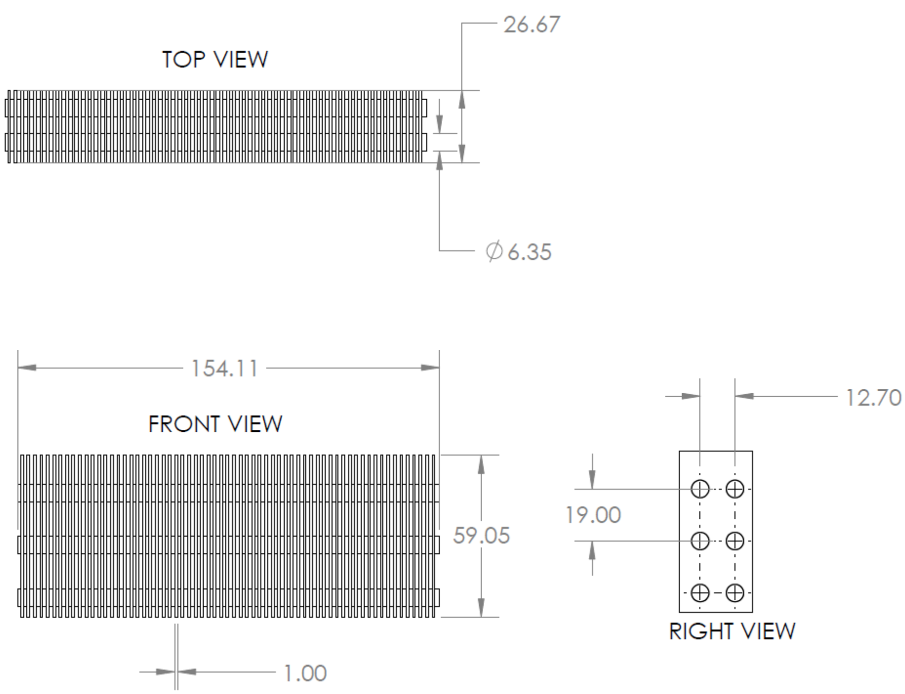
7.2. Schematic Drawing of Heat Exchanger (via SolidWorks)

This section represents 1/10th of total heat exchanger volume

I. 3D Model



II. 2D Schematic



7.3. Compliance with Codes and Standards

This project complies with the American Society for Testing and Materials (ASTM). The codes of standards were used to advise the coolant properties, product maintenance, calculations, design, and quantity of components.

1. Coolant Expectations

This project worked with Ethylene Glycol 50:50 (50% diluted with deionized water) to ensure that the performance of the heat exchanger is consistent with industry standards through the Codes and Standards. Thus, we provide effective prevention against freezing boiling and corrosion.

Code 1: Cooling system fill should consist of coolant concentrate and water or prediluted glycol or glycol/glycerin blend base engine coolant (50 volume % minimum).

Code 2: The recommended coolant concentration range is 40 to 70 %.

Code 3: When concentrates are used at 50 to 60 % concentration by volume in water, or when prediluted glycerin base engine coolants (50 volume % minimum) are used without further dilution, they will function effectively to provide protection against freezing, boiling, and corrosion.

[Source: D3306.15030 (ASTM Standards, Glycol), p5]

2. Performance Evaluation

This project used the suggested parameters for performance analysis. These include log mean temperature difference, fouling factor and resistance, specific heat capacities, and overall convective heat transfer. This was used to calculate the "physics-derived" U and "mathematically-derived" U. This was also used to calculate the C-minimum and exit air temperatures for off-design performance.

[Source: ASME PTC 12.5-2000 (Single Phase Heat Exchanger)]

3. Log Mean Temperature Difference and Correction Factor

This project followed this approach to ensure that log mean temperature derivations were consistent with the standards. This was important because the LMTD influenced the physics-derived heat transfer rate and math-derived overall heat transfer coefficient. Additionally, the physics-derived U and the mathematically derived U were found to be the same, meaning that this product will function effectively in line with industry and production standards if produced.

Code 1: To account for the different flow arrangements, a correction factor, F is applied to the LMTD

[Source: ASTM PTC 12.5-2000, p75]

Code 2: Correction Factor should be extrapolated from ASTM standards for cross flow for single pass

[Source ASTM PTC 12.5-2000, p89]

4. Pressure Drop

Source: ASME PTC 12.5-2000

This project incorporated the friction factor in the derivation of the pressure drop. This project made use of the equations provided by the ASME to evaluate performance through fouling factor, thermal resistance, and overall heat transfer coefficient.

7.4. Project Code

```
close all;
clear all;
clc;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% PREVIOUS OPTIMAL SPECIFICATIONS
% num_fins= 600;
% fin_length = 1;
% fin_width = 0.2;
% tube_length = 1.5504;
% fin_thickness = 0.001;
% num_tubes= 600;
% num_rows = 5;
% num_columns = num_tubes/num_rows;
% tube_diam = 0.00635; %meters (m)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%


%for j = 1.53:0.0001:1.55
num_fins= 650;
tube_length = 1.5411;
fin_thickness = 0.001;
num_tubes= 600;
num_rows = 20;
num_columns = num_tubes/num_rows;
tube_diam = 0.00635; %meters (m)
transverse_pitch = 4*tube_diam;
longitudinal_pitch=  3*tube_diam;

tube_spacing_lengthwise = transverse_pitch - tube_diam ;
tube_spacing_widthwise = longitudinal_pitch - tube_diam;
fin_length = tube_spacing_lengthwise*(num_columns+1);
fin_width = tube_spacing_widthwise*(num_rows+1);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%defining fluid properties for ethylene glycol
mass_flow_ethyl = 1.2; %kg/s
mass_flow_ethyl_per_tube = mass_flow_ethyl/num_tubes;

T_ethyl_in = 90;
T_ethyl_out = 60;

cp_ethyl = 3641.5; %J/KgC;
dyn_visc_ethyl = 0.000906;
K_ethyl = 0.3947; %thermal conductivity of fluid
Pr_ethyl = 9.82;
density_ethyl = 1045;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

math_Q_rate = mass_flow_ethyl*cp_ethyl*(T_ethyl_in-T_ethyl_out);
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%defining fluid properties for air
T_air_in = 30;
Temp_diff = 20;
T_air_out = T_air_in + Temp_diff;
cp_air = 1008;
dyn_visc_air = 1.918E-05;
K_air = 0.02662;
Pr_air = 0.7255;
density_air = 1.127;
mass_flow_air = math_Q_rate/(cp_air*(T_air_out - T_air_in)); %kg/s
%mass_flow_air = 53.5;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%Constants to Define
%LTMD Correction Factor
P= (T_ethyl_out-T_ethyl_in)/(T_air_in-T_ethyl_in);
R = (T_air_in-T_air_out)/(T_ethyl_out-T_ethyl_in);
F=0.95;
% Graph is on page 664

%No of Rows Correction Factor
row_number_correction_factor = 1;
% 3 rows = 0.86, 4 rows = 0.9, 5 rows = 0.93
% 7 rows = 0.96, 10 rows =0.98, 13 rows = 0.99

%Pressure drop and Pumping Power Constants
friction_factor = 0.17;
correction_factor = 0.6;
% Data can be found on pg 448

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Inside the tube: convective heat transfer coefficient
v_inside = ((4*mass_flow_ethyl_per_tube)/(density_ethyl*pi*(tube_diam)^2));
Re_inside = (density_ethyl*tube_diam*v_inside)/dyn_visc_ethyl;

if Re_inside < 2300
    Nu_inside = 4.36;
elseif Re_inside>2300 && Re_inside<10000
    f= ((0.79*log(Re_inside))-1.64)^-2;
    Nu_numerator = (f/8)*(Re_inside-1000)*Pr_ethyl;
    Nu_denominator= 1+(12.7*((f/8)^0.5)*((Pr_ethyl^(2/3))-1));
    Nu_inside = Nu_numerator/Nu_denominator;
else
    Nu_inside = 0.023*((Re_inside)^0.8)*(9.82^(1/3));
end

h_inside = (K_ethyl*Nu_inside)/tube_diam;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%Outside the tube: convective heat transfer coefficient
```

```
v_outside = mass_flow_air/(density_air*tube_length*fin_length);
v_outside_max = (transverse_pitch*v_outside)/(transverse_pitch-tube_diam);
Re_outside = (density_air*tube_diam*v_outside_max)/dyn_visc_air;
Nu_outside = 0.27*(Re_outside^0.63)*(Pr_air^0.36);
Nu_corrected = row_number_correction_factor*Nu_outside ;
h_outside = (Nu_corrected*K_air)/tube_diam;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%calculating relevant areas
A_inside = num_tubes*pi*tube_diam*tube_length;
A_unfin = num_tubes*((pi*tube_diam*tube_length)-(num_fins*pi*tube_diam...
    *fin_thickness));
A_fin= num_fins*((2*((fin_length*fin_width)+(fin_width*fin_thickness)...
    +(fin_length*fin_thickness))-(2*num_fins*pi*(tube_diam^2)*0.25)));
A_nofin= A_inside;
%A_outside = A_unfin + A_fin;

%Calculating efficiency of the fin
K_copper = 401;
a=(2*h_outside)/(K_copper*fin_thickness);
m = sqrt(a);
area_of_unit_cell = (fin_length*fin_width)/num_tubes;
side_of_unit_cell = sqrt(area_of_unit_cell);

Lc= ((side_of_unit_cell-tube_diam)/2)+(fin_thickness/2);


fin_efficiency = tanh(m*Lc)/(m*Lc);

%Effective External Surface Area
A_s = A_unfin + (fin_efficiency*A_fin);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%Calculating R
R_f=0.00035;
R_thermal= (1/(h_inside*A_inside))+(1/(h_outside*A_s))+(R_f/A_inside);

U = 1/(A_s*R_thermal);
%U_inside = (1/(A_inside*R_thermal));
%U_outside = (1/(A_s*R_thermal));

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Calculating NTU

C_ethyl= mass_flow_ethyl*cp_ethyl;
C_air = mass_flow_air*cp_air;

if C_ethyl<C_air
    C_min= C_ethyl;
    C_max=C_air;
else
    C_min = C_air;
    C_max = C_ethyl;
end
```

```
NTU = (U*A_s)/C_min;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Calculating LMTD

LMTD = ((T_ethyl_in-T_air_out)-(T_ethyl_out-T_air_in))/...
    (log((T_ethyl_in-T_air_out)/(T_ethyl_out-T_air_in)));



LMTD_corrected = F*LMTD;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Calculating effectiveness

delta_T_max = T_ethyl_in-T_air_in;
Q_dot_max = C_min * delta_T_max;
physics_Q_rate = U*A_s*LMTD_corrected;
Effectiveness_1 = physics_Q_rate/Q_dot_max;

c = C_min/C_max;
exp_term = ((NTU^0.22)/c)*((exp(-c*(NTU^0.78)))-1);
Effectiveness_2 = 1 - exp(exp_term);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Finding U math

U_math = math_Q_rate/(A_s*LMTD_corrected);
U_perc_diff = abs(((U-U_math)/U_math)*100);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Pressure drop and Pumping Power
P_L = longitudinal_pitch/tube_diam;
P_T = transverse_pitch/tube_diam;
correction_value_finder=(P_T-1)/(P_L-1);

pressure_drop = 0.5*num_rows*friction_factor*correction_factor*...
density_air*((v_outside_max)^2);
pumping_power = ((mass_flow_air*pressure_drop)/density_air);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Changing Parameter (Pick a parameter to change)
%fprintf('Tube Length = %f, U percentage diff = %f \n', tube_length,U_perc_diff);

%end

fprintf('Values for Calculations (Not results)\n');
fprintf('_________________________________________________\n');
fprintf('P_L= %f \n', P_L);
fprintf('Correction Value Finder = %f \n', correction_value_finder);
fprintf('Reynolds Number Outside = %f \n', Re_outside);
fprintf('P = %f \n', P);
fprintf('R= %f \n', R);
fprintf('\n');
```

```
fprintf('Results under Normal Ambient Air \n');
disp('___________________________________________________');
fprintf('Dimensions of Heat Exchanger \n');
fprintf('Length of Fin (m) = %f \n', fin_length);
fprintf('Width of Fin (m)= %f \n', fin_width);
fprintf('Thickness of Fin (m)= %f \n', fin_thickness);
fprintf('Tube Length (m)= %f \n', tube_length);
fprintf('Tube Diameter (m)= %f \n', tube_diam);
fprintf('\n');
%fprintf('Reynolds Number Inside = %f \n', Re_inside);
%fprintf('Reynolds Number Outside = %f \n', Re_outside);
%fprintf('Internal h (h_i) = %f W/(m2°C)\n', h_inside);
%fprintf('External h (h_o) = %f W/(m2°C)\n', h_outside);
fprintf('Performance Results \n');
fprintf('Physics U= %f W/(m2°C) \n', U);
fprintf('Theoretical/Math U= %f W/(m2°C) \n', U_math);
fprintf('U percentage diff = %f \n', U_perc_diff);
fprintf('Effectiveness1= %f \n', Effectiveness_1);
fprintf('Effectiveness2= %f \n', Effectiveness_2);
fprintf('Pressure Drop= %f Pa \n', pressure_drop);
fprintf('Pumping Power= %f kW \n', pumping_power/1000);
fprintf('NTU= %f \n', NTU);
fprintf('Fin Efficiency= %f \n', fin_efficiency);
fprintf('\n');

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Off-design performance - different code
fprintf('Off-design Performance Evaluation \n');
fprintf('_________________________________________________\n');

for i = 10:10:50
    %Initialization
    T_air_in = i;
    Q_dot_max2 = C_min*(T_ethyl_in-T_air_in);
    Q_dot2 = Effectiveness_1*Q_dot_max2;
    T_air_out = (Q_dot2/(C_air))+T_air_in;

    %Printing Exit Air Temp
    fprintf('Ambient Air Temperature In: %f C \n', T_air_in);
    fprintf('Exit Air Temperature: %f C \n', T_air_out);
    disp('___________________________________________________');

end

disp("Done")
```