Heat Exchanger Design for Car Radiator Application

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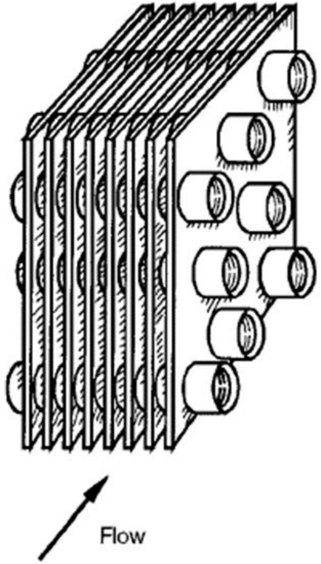
Problem Statement and Design Concept



- Ethylene Glycol (50:50) at 1.2Kg/s needs to be cooled from 90℃ to 60℃ through a heat exchanger with coolant of ambient air.
- The chosen exchanger is tube-fin with plate fins due to its efficiency with air as a coolant.
- The tubes are in-line and the flow is single-pass to reduce the pumping power needed.
- The heat exchanger is compact (area density > 700 m^2/m^3, D_h <6.35mm), final calculations gave [area density= 1853.67 m^2/m^3 and D_h= 6.35mm]
- Material chosen is Copper due to its higher conductivity and cost-effectiveness

Design Assumptions

1. Flow is steady and fully developed with constant properties and fouling factors.
2. Fouling exists only on the Ethylene Glycol side.
3. Heat loss to the surroundings, changes in kinetic and potential energies are negligible.
4. Since the tubes are thin-walled and highly conductive, their thickness and thermal resistance was assumed to be negligible.

Calculations: Design Principle

- The design principle was based on:
  1. Finding the heat transfer rate using mathematical methods of calculating heat transfer. This was proven to be **131.094 kW**.

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

  2. Then using the physics of the problem to match that value with the appropriate design through thermal resistance and LMTD.
- Fixed design parameter: the temperature difference of air is 20°C.

Mass flow and LMTD

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

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

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

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

Log Mean Temperature Difference (LMTD)

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

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 $F = 0.95$

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

Inner Heat Transfer Coefficient

<u>Convective Heat Transfer Coefficient Inside the Tube</u>

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},

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

Outer Heat Transfer Coefficient

Convective Heat Transfer Coefficient Outside the Tube

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

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

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

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

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

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

Total Thermal Resistance and Overall h

Total Thermal Resistance

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

$$R_{total} = \frac{1}{h_{inside} \times A_{inside}} + \frac{1}{h_{outside} \times A_{total\ surface}} + \frac{R_f}{A_{inside}}$$

$$= \frac{1}{271.0 \times 18.446} + \frac{1}{161.4042 \times 188.3679} + \frac{0.00035}{18.446} = 0.6667$$

Thus,

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

From the first step, we know that

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

So,

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

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

NTU and Effectiveness

Calculating NTU

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

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

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

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

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

$$\boxed{= 0.500}$$

Pressure Drop and Pumping Power

Calculating Pressure Drop

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 $V_{air,max}$.

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

Calculating Pumping Power

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

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

Preliminary Designs: Physical Dimensions

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

Preliminary Design: Specifications

Specification	Value	Specification	Value
Mass Flow Rate of Air (kg/s)	6.503	*Effective Heat Transfer Area (m^2)*	80.797
Required Air Velocity (m/s)	10.123	*Fin Efficiency*	**0.7738**
Inlet Air Temperature (℃)	30	*Log Mean Temperature Difference (℃)*	33.0226
Outlet Air Temperature (℃)	50	*Number of Transfer Units (NTU)*	0.05816
Physics Heat Transfer Rate (kW)	**8.407**	*Effectiveness*	**0.03201**
Overall Heat Transfer Coefficient ($W/m^2℃$)	3.145		

Optimization Process

Our design team settled on a heat exchanger with a tube length ranging between 1 to 1.75m.

The following values were altered until our derived heat transfer rate matched our expected value of **131.094 kW**:

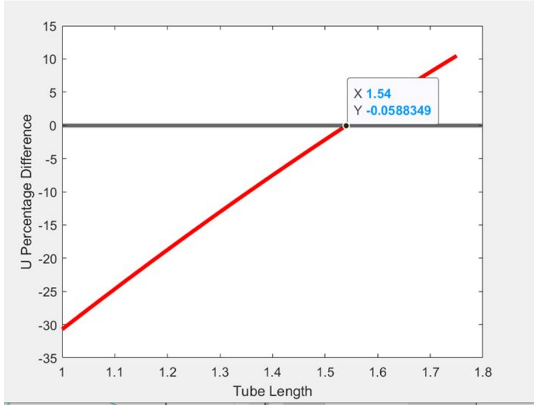
- the longitudinal pitch (which directly affects the fin width)
- the transverse pitch (which directly affects the fin length)
- fin thickness
- number of tubes
- number of fins
- number of rows
- tube diameter

Determining Tube Length where $Q_{rate} = Q_{rate\_expected}$

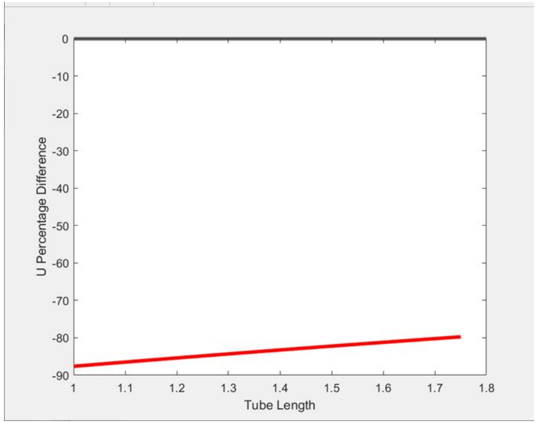
- Calculated $U_{expected}$ using:

$$U_{expected} \quad A_{Effective\ S.A.} \quad F\Delta T_{LM} = 131.094kW.$$

- Calculated the percentage difference (percentage error) between our physics derived U and $U_{expected}$
- Graph the U percentage error against Tube Length.
- Optimal Tube Length occurs where the graph cuts x-axis (i.e., U percentage error = 0)



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



Design Failure as Observed in Preliminary Design Specifications

Optimization Changes - Tubes

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 ↓
Longitudinal Pitch	mm	54.00	19.05	Decrease ↓
Transverse Pitch	mm	58.85	25.40	Decrease ↓

Optimization Changes - Fins

Dimension	**Unit**	**Initial Design Value**	**Final Design Value**	**Nature of Change**
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 ↑

Final Design: Specifications

Specification	Value	Specification	Value
Mass Flow Rate of Air (kg/s)	6.503	*Effective Heat Transfer Area (m^2)*	188.368
Required Air Velocity (m/s)	6.340	*Fin Efficiency*	**0.9922**
Inlet Air Temperature (℃)	30	*Log Mean Temperature Difference (℃)*	33.0226
Outlet Air Temperature (℃)	50	*Number of Transfer Units (NTU)*	0.9085
Physics Heat Transfer Rate (kW)	**131.092**	*Effectiveness*	**0.5000**
Overall Heat Transfer Coefficient (W/m^2℃)	21.075		

Performance Analyses and Discussion

Design Point Performance

Performance Evaluators	**Physics U**	**Theoretical/Math U**	**Percentage Diff. (%)**
Overall Heat Transfer Coefficient (W/m²K)	21.074526	21.1074881	0.001683
Heat Transfer Rate (W)	131090	131094	0.00305

Performance Analyses and Discussion

Off-Design Performance

Inlet Ambient Temperature (℃)	Inlet Ambient Temperature (Off the design point) (℃)	Exit Air Temperature (℃)	Exit Ethyl Temperature (℃)
10	-20	36.67	50.00
20	-10	43.33	55.00
30	0	50	60
40	10	56.67	65.00
50	20	63.33	70.00

Manufacturer’s Specifications - A

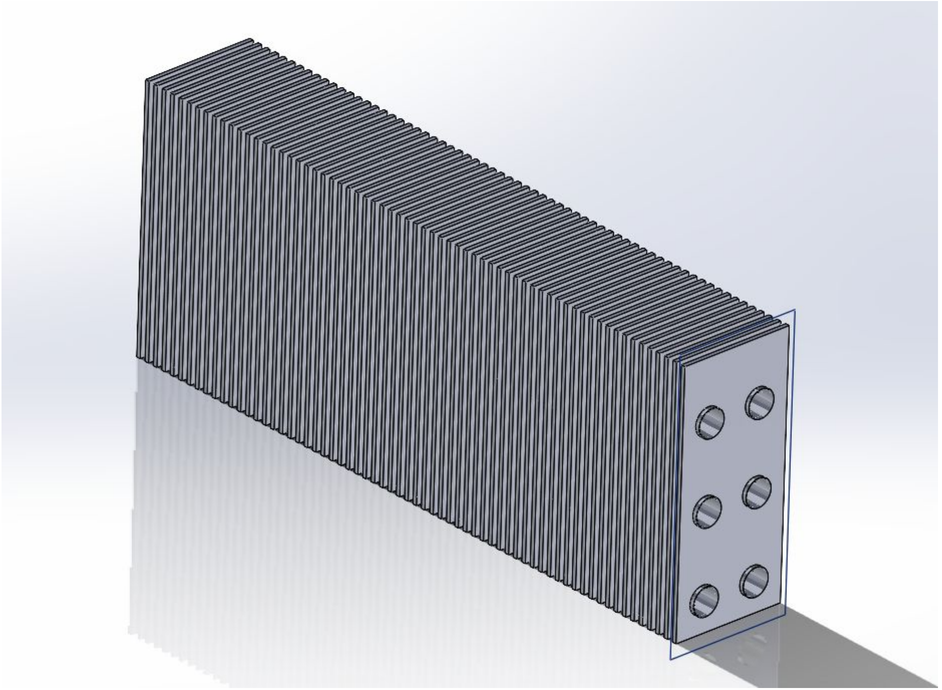
Parameters	Air	Ethylene-Glycol (50:50)	Units
Fluid type	Gas	Liquid	
Fluid Density	1.127	1045	kg/m^3
Fluid Specific heat capacity	1008	3641.5	J/(kg°C)
Fluid Thermal conductivity	0.0266	0.3947	W/mK
Dynamic Fluid viscosity	1.918e-05	9.060e-04	kg/ms
Mass Flow Rate	6.5027	1.2000	kg/s
Inlet temperature	30	90	℃
Outlet temperature	50	60	℃
Pressure drop	84.9352		Pa
Pumping Power	490.07		W
LMTD	33.0226		℃
Fouling	0.00035		m^2K/W
Number of fins	650		
Number of tubes	600		
Maximum Operating Temperature (Ambient Air)	90		℃

Manufacturer’s Specifications - B

Parameters	**Air**	**Ethylene-Glycol (50:50)**	**Units**
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

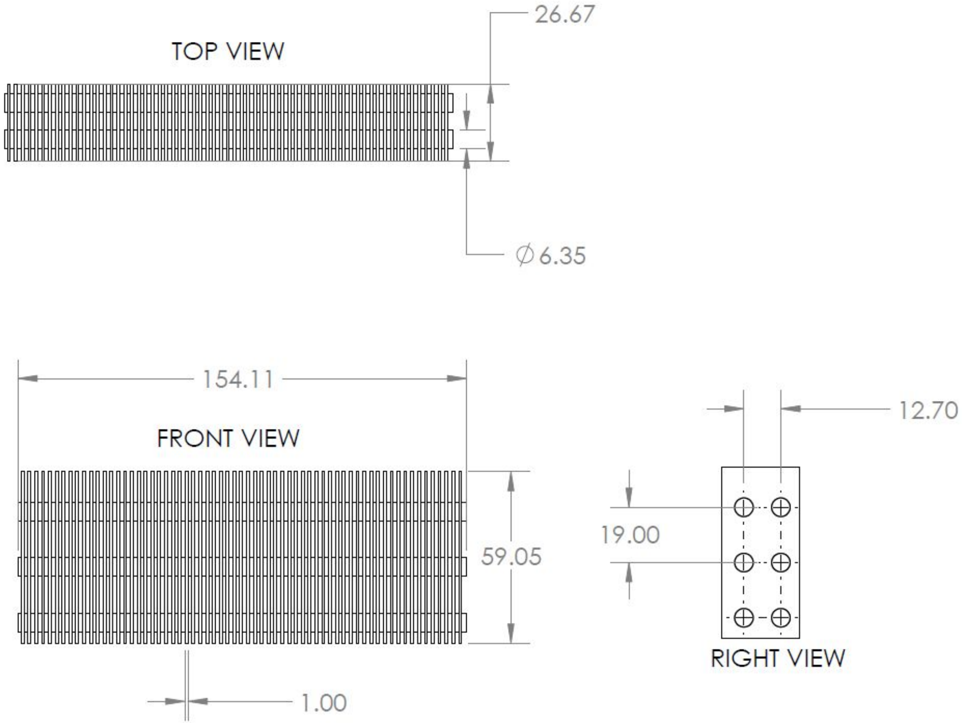
Final Product (Scale 1:10)

Air in



Air out

Schematics



- **Scale (1:10)**
- **Top, Front, Right View**
- **Dimensions in mm**

Applicable Codes and Standards

Compliance with Codes and Standards

I. Coolant Expectations

Source: ASTM DD306, p5
X1.1.2 Cooling system fill should consist of coolant concentrate and water or prediluted glycol or glycol/glycerin blend base engine coolant (50 volume % minimum).
X1.1.4 The recommended coolant concentration range is 40 to 70 %.
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

II. Log Mean Temperature Difference and Correction Factor

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

Source: ASTM PTC 12.5-2000, p75

2. Correction Factor extrapolated from ASTM standards for cross flow for single pass

Source ASTM PTC 12.5-2000, p89

II. Glycol Properties

Source: ASTM E1177.26690

Codes and Standards

TABLE 3.2
TYPICAL HEAT EXCHANGER THERMAL MODEL PARAMETERS NEEDED FOR PERFORMANCE ANALYSIS

Convective heat transfer coefficients for hot and cold streams	Thermal conductivity of the wall (tube or plate)
Mean temperature difference	Fouling resistance
Specific heats of the hot and cold streams	Thermal conductivity of fin and contact resistance between fin and tube
Cold side, hot side, and reference heat transfer areas	Surface effectiveness of enhancements such as fins