Complex Engineering Problem

Subject:

Refrigeration Lab

Submitted By:

Syed Adil Hasan Naqvi (ME07B)

Abdul Wasay Ikhlaq (ME07B)

Hamza Munir (ME07A)

Submitted To:

Sir Hamza Khursheed

Date:

May 18th, 2018

Department of Mechanical Engineering
School of Mechanical and Manufacturing Engineering
NUST Islamabad

Cabin cooling load determination

The cabin cooling load is the amount of heat energy to be removed to maintain the vehicle cabin temperature at a human comfort level when worst case outdoor design temperatures are being experienced. This total heat can be broken down further smaller heating loads due to factors such as ambient temperature, solar radiations, engine heat, human metabolism etc.

The mathematical formulation of the model that accounts for all such loads can be summarized as follows:

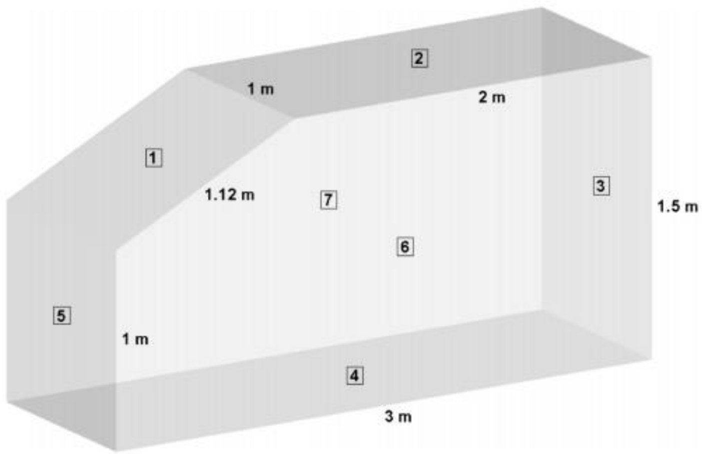
$$Q_{cabin} = Q_{met} + Q_{dir} + Q_{dif} + Q_{ref} + Q_{amb} + Q_{exh} + Q_{eng} + Q_{ven}$$

Where Q_{cab} is net overall thermal load encountered by the cabin, Q_{met} is metabolic load, Q_{Dir} is direct load, Q_{Dif} is diffuse load, Q_{ref} is reflected radiation load, Q_{Amb} is ambient load, Q_{Exh} is exhaust load due to high temperature, Q_{Eng} is engine load due to high temperature, Q_{Ven} is load generated due to ventilation. The goal here in to determine values of all these factors at the worst exterior and interior conditions to determine the total cabin cooling load.

Humans have a normal body temperature of around 37°C. The comfort temperature is taken to be 23°C [1] for our load calculations.

Cabin geometry and properties

A close approximation of the interior of a vehicle was taken from [4]. The schematic of the cabin geometry is as following:



The properties of the cabin geometry along with the driving conditions were also sourced from [4] and are as following:

Property	Glass	Vehicle Body
Conductivity (*W/mK*)	1.05	0.2
Density (*kg/m³*)	2500	1500
Transmissivity	0.5	0
Absorptivity	0.3	0.4
Specific Heat (*J/kgK*)	840	1000
Thickness (*mm*)	3	10

Specification	Value
Date	July 21, 2012
Local Time	13:00 to 16:00
Location	Houston, Texas
Driver Height, Weight	1.7 m, 70 kg
Passenger Height, Weight	1.6 m, 55 kg
Ventilation Flow	0.01 m^3/s (21.2 CFM)
Ground Reflectivity	0.2
Ambient Temperature	34.4°C
Initial Cabin Temperature	80°C
Ambient Relative Humidity	70%
Cabin Relative Humidity	50%
Comfort Temperature	23°C
Pull-Down Time	600 s
Deep Thermal Mass	5600 J/K

Metabolic load

The assumption in this study was considered taking into account 1 driver and 3 passengers to calculate the heat in the cabin due to human metabolism. The metabolism load is given as:

$$Q_{met} = \sum_{i=0}^{4} M_i A_{du,i}$$

	Weight (kg)	Height (m)	Metabolic heat prod. rate (W/m^2)
Driver	70	1.77	85
Passengers	70	1.77	55

Where M is the passenger metabolic heat production rate. It is found from the tabulated values in ISO 8986 [2]. For a driver and a sitting passenger, the values can be estimated as 85 W/m^{2}and 55 W/m^2, respectively. The Dubois is area A_{Du} which is an estimation of the body surface area as a function of height and weight is calculated by:

$$A_{du} = 0.202 W^{0.425} H^{0.725}$$

Where W and H are the passenger weight and height respectively and are taken to be average [3].

$$A_{du} = 0.202(70)^{0.425}(1.77)^{0.725}$$

$$= 1.85\ m^2$$

Now to find metabolic load,

$$Q_{met} = (85 + 55 + 55 + 55) \times 1.85$$

$$= \mathbf{464.77\ W}$$

Direct load

$$Q_{dir} = \sum_{surface} S\tau I_{dir} \cos\theta$$

Where I_{dir} the direct radiation heat gain per unit area and θ is the angle between the surface normal and the position of sun in the sky. τ is the surface element transmissivity and S is the surface area, respectively. Before local sunrise and after local sunset, simply no radiation loads are considered. The direct radiation heat gain per unit area is found by:

$$I_{dir} = A/exp(B/sin\beta)$$

Where A and B are constants tabulated in ASHRAE Handbook of Fundamentals [1] for different months. β is the altitude angle that is calculated based on position and time. The direct radiation loads are important air-conditioning loads that influence the temperature in the cabin. The direct radiation load increases due to the increase in the sun altitude angle which is happening in the simulation.

From [1],

$$A = 1093\, W/m^2$$

$$B = 1.486$$

$$h = 30° \; (at\ 2pm)$$

$$l = 29.7°N \; (for\ Houston)$$

$$d = 23.47 sin \frac{360(284+N)}{365} = 20.459°$$

$$\beta = sin^{-1}[\cos(l)\cos(h)\cos(d) + \sin(l)\sin(d)] = 61.40°$$

$$\Rightarrow I_{dir} = 884.33\, W/m^2$$

$$\Rightarrow \gamma = sin^{-1}\left[\frac{\cos(d)\sin(h)}{\cos(\beta)}\right] = 101.86°$$

$$\Rightarrow \alpha = F[\pi - \gamma - \xi] = 78.14°$$

$$\Rightarrow \theta = cos^{-1}[\cos(\alpha)\cos(\beta)] = 84.354°$$

Putting in all values yields,

$$\boldsymbol{Q_{dir} = 925.94\, W}$$

Diffuse load

Diffuse radiation is the part of solar radiation, which results from indirect radiation of daylight on the surface.

$$Q_{dif} = \sum_{surface} S\tau I_{dif}$$

Where I_{dif} is the diffuse radiation heat gain per unit area which is calculated from:

$$I_{dif} = CI_{dir}\frac{1+\cos(\xi)}{2}$$

Diffuse radiation distribution inside the cabin is independent of the solar angle. For other orientations the effect of latitude is more evident, but not so influential. The diffuse load increases due to the increase in the sun altitude angle.

From [1] & [4],

$$C = 0.138$$

$$\Rightarrow I_{dif} = 61.01\, W/m^2$$

Putting values in original equation gives,

$$\Rightarrow \boldsymbol{Q_{dif} = 281.39\, W}$$

Reflected load

The value of reflectivity obviously depends on the surface property on the ground. The reflectivity of the ground or a horizontal surface from where the solar radiation is reflected on to a given surface was 0.02.

$$Q_{ref} = \sum_{surface} S\tau I_{ref}$$

I_{ref} is the reflected radiation heat gain per unit area, is calculated form:

$$I_{ref} = \rho g(I_{dir} + I_{dif})\frac{1+\cos(\xi)}{2}$$

whereρg is the ground reflectivity coefficient. Based on the absorptivity of each particular surface element, a percentage of incident radiation load can be absorbed by that surface, hence increasing its temperature.

We have all the variables (from material properties and previously determined values) for the above equation. Plugging in values yields,

$$\boldsymbol{Q_{ref} \approx 0}$$

This is because the transmittivity for our base material is zero and for the glass it's 0.5. This significantly reduces Q_{ref}. Moreover, the value for ground reflectivity is very low (about 0.02) which reduces the value of I_{ref} which further reduces the value of Q_{ref}. Hence, it is safe to ignore Q_{ref} entirely as its significance in total load is close to zero.

Ambient load

The increasing in the heat indicates that the weather data effect is very important as the change in the ambient temperature affecting the calculation of the external and internal cooling loads. According to these results, the maximum cooling load is above the cooling capacity of the system occurred in high ambient temperature.

$$Q_{amb} = \sum_{surface} SU(T_s - T_i)$$

Where U is the overall heat transfer coefficient of the surface element. T_s and T_i are the average temperature and average cabin temperature, respectively. U has different components consisting of the inside convection, conduction through the surface, and outside convection that can be written in the form:

$$U = \frac{1}{R} \; where \; R = \frac{1}{h_o} + \frac{\lambda}{\mathrm{k}} + \frac{1}{h_i}$$

Where R is the net thermal resistance for a unit surface area. ho and hi are the outside and inside convection coefficients, k is the surface thermal conductivity, and λ is the thickness of the surface element. The thermal conductivity and thickness of the vehicle surface thermal can be measured rather easily. The convection coefficients ho and hi depend on the orientation of the surface and the air velocity. Here, the following estimation is used to estimate the convection heat transfer coefficients as a function of vehicle speed.

$$h = 0.6 + 6.64\sqrt{v}$$

Where h is the convection heat transfer coefficient in W/m^2K and v is the vehicle speed in m/s. Despite its simplicity, this correlation is applicable in all practical automotive instances. The cabin air is assumed stationary and the ambient air velocity is considered equal to the vehicle velocity.

$$for\ v = 60km/s$$

$$\lambda_{glass} = 3\text{mm}$$

$$\lambda_{base\ material} = 10\text{mm}$$

$$\text{k}_{glass} = 1.05\ \text{W/mK}$$

$$k_{base\ material} = 0.2\text{mm W/mK}$$

$$h = 52.03\ \text{W}/m^2K$$

$$T_s = 34.4°C$$

$$T_i = 23°C$$

Putting values in the original equation yields,

$$\boldsymbol{Q_{amb} = 233.4\ W}$$

Exhaust load

Assumed Engine RPM = 2000 rpm

Exhaust gasses are at a significantly higher temperature so they also contribute to cabin heating. The temperature of the exhaust gasses in Celsius can be calculated as:

$$T_{exh} = 0.138RPM - 17$$

When there is no acceleration or there is a normal stable drive, the engine RPM stays at around 2000. So the temperature becomes,

$$T_{Exh} = 619℃$$

The surface area at bottom in contact with exhaust pipe is,

$$A = 1.5 * 0.0508 * \pi/2$$

The overall heat transfer coefficient is,

$$U = 2.5W/mC$$

So, the overall exhaust load comes out to be,

$$Q_{Exh} = A_{Exh}U(T_{Exh} - T_i)$$

$$\mathbf{Q_{Exh} = 140.5\ W}$$

Engine load

Due to high temperature of engine, some heat is also added to the cabin and hence contributes to the overall heat load. The engine temperature is given by,

$$T_{Eng} = -2 \times 10^{-6}RPM^2 + 0.0355RPM + 77.5$$

$$T_{Eng} = 140.5\ ℃$$

Surface area is,

$$A = 0.5 * 0.5 * 6$$

Overall heat transfer coefficient,

$$U = 1.5W/mC$$

So, the total engine load can be calculated as follows:

$$Q_{Eng} = A_{Eng}U\left(T_{Eng} - T_i\right)$$

$$\mathbf{Q_{Eng} = 263.33\ W}$$

Ventilation load

It is not efficient to use hot ambient air for ventilation when cooling is required. Therefore, a small amount of ventilation flow rate is assumed to account for the minimum fresh air requirement.

$$Q_{ven} = m_{ven}(e_o - e_i)$$

Where m_{ven} is the ventilation mass flow rate and e_o and e_i are the ambient and cabin enthalpies, respectively. Enthalpies are calculated from:

$$e\ = 1006T + \ (2.501 \times 10^6 + 1770T)X$$

Where T is air temperature and X is humidity ratio in gram of dry air. Humidity ratio is calculated as a function of relative humidity by:

$$X = 0.62198 \times \left(\frac{\emptyset P_s}{100P} - \ \emptyset P_s\ \right)$$

Where $\emptyset$ is relative humidity, P is pressure and P_s is the water saturation pressure at temperature T.

$$m_{ven} = 0.1\ kg/s$$

$$\emptyset_{amb} = 0.7 \qquad \emptyset_{cabin} = 0.5$$

$$P_{s,amb} = 5.446\ kPa \qquad P_{s,cabin} = 2.812\ kPa$$

$$X_{amb} = 2.372 \qquad X_{cabin} = 0.874$$

Plugging values in the original equation yields,

$$\boldsymbol{Q_{ven} = 322.5\ W}$$

AC Load Calculations

All the loads summed up have to be taken care of by the air conditioning unit. The sum of the loads mentioned above give the total heat load of the table which the air conditioning unit must remove.

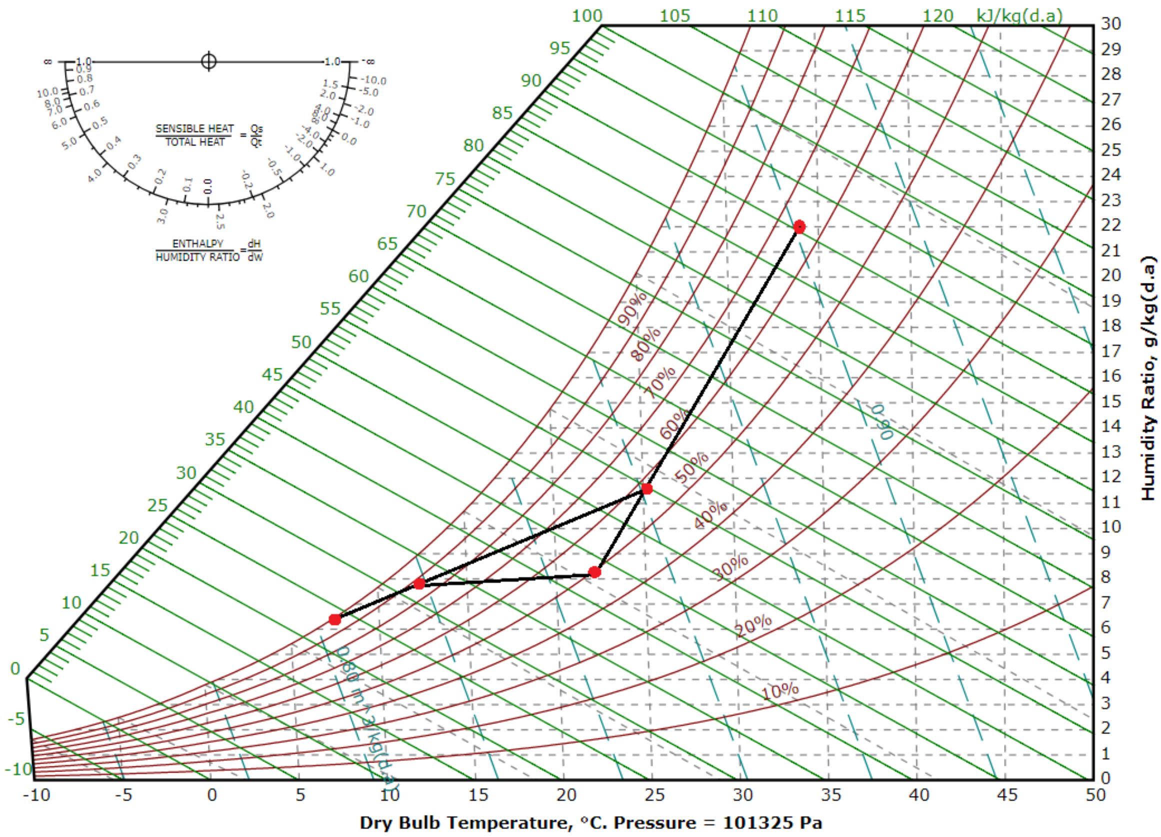
$$\text{Total Sensible Heat Load} = \dot{Q} = 2631.83\text{W}$$

$$\text{Total Latent Heat Load} = 220\text{W}$$

$$\text{Grand Heat Load} = 2851.83\text{W}$$

$$\text{RSHF} = 0.92$$

Assuming we want to maintain the cabin at 23℃ and 50% RH. So, these are our comfort conditions. The ambient relative humidity and ambient temperature are respectively 0.65 and 34℃. 20% of the cabin air is replaced with the outside air in each cycle. The supply air is to be no cooler than 12℃ because it would cause inconvenience for the passengers. These states can be represented on the psychrometric chart as follows.



So, from here, we can calculate the minimum mass flow rate of air needed to achieve this cooling as:

$$Q_{Grand} = \dot{m}(h_{supply} - h_{room})$$

$$-2.851 = \dot{m}(32.5 - 43)$$

$$\dot{m} = 0.271Kg/s$$

The equivalent volume of supply air is:

$$\dot{V} = 0.271 \times 0.819$$

$$\dot{V} = 0.222m^3/s$$

The coil ADP is 7.5℃. and the coil load can be calculated as follows:

$$Q_{coil} = 0.271(54 - 32.5)$$

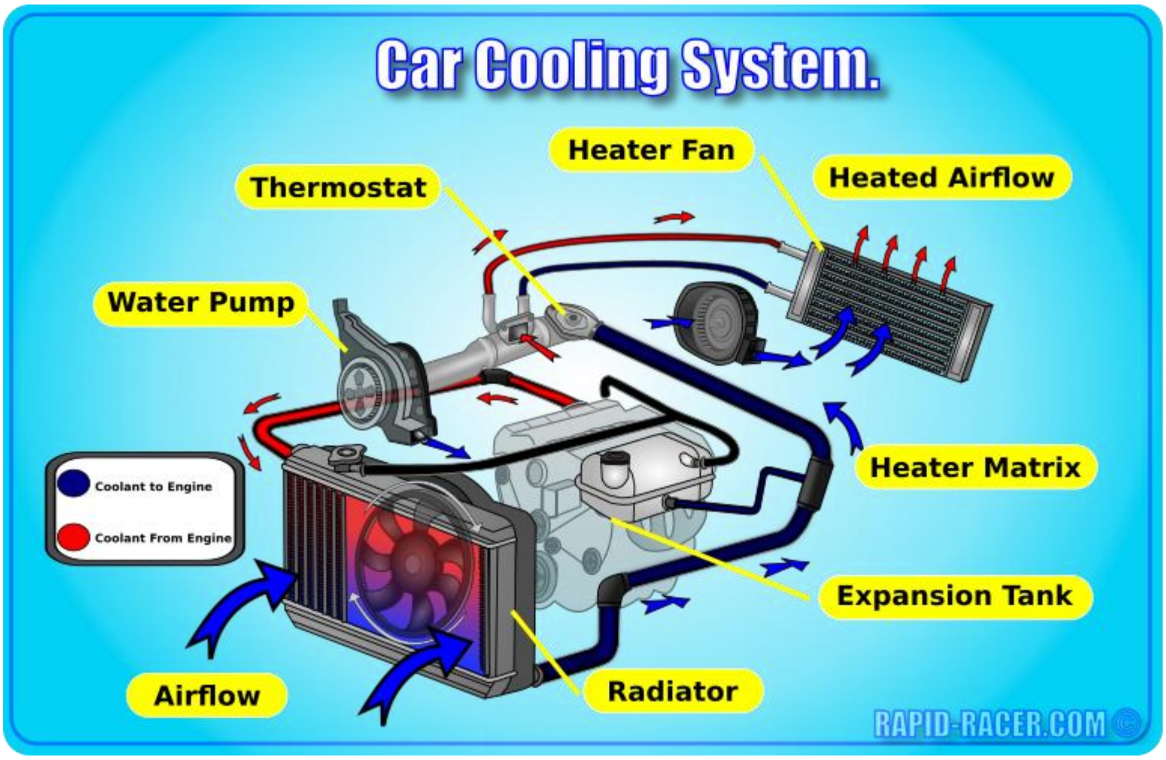
$$\boldsymbol{Q_{coil} = 5.8265kW}$$

So, the AC requirements are 5.8KW for the cooling coil and air circulation mass flow rate of 0.271 Kg corresponding to a supply volume flowrate of 0.308 m^3/s.

These values correspond to the maximum possible heating load during peak conditions. Such peak loads are unlikely to occur during normal operating conditions and hence these values are somewhat higher than those found in literature.

Engine Cooling System

The nomenclature and the general schematic of the cooling system of a car is shown below and its different parts are labelled:



For the analysis of engine cooling system, we will consider different factors as followed. And accordingly, the calculations are shown for the required engine cooling system. We will start with the cooling load.

Cooling Load

To start the analysis, we will the average temperature of engine body to be 85 Celsius. Also, about 70% of the energy developed from exploding gasoline is converted into heat energy which has to be removed by our cooling system.

$$\mathrm{Q_{Eng} = US_{Eng}(T_{Eng} - T_i)}$$

Where,
U is the overall heat transfer coefficient of the surface element in contact with the engine.
S_{Eng} is the surface area exposed to the engine temperate.

T_{Eng} is the engine temperature and found by,

$$TEng\ =\ -2 \times\ 106\, RPM^2\ +\ 0.0355RPM\ +\ 77.5$$

Taking some valid assumptions for the unknown constants, the engine heat comes out to be,

$$\mathbf{Q_{Eng} = 38000\ W}$$

Refrigerant

The choice of refrigerant is quite important as it must be able to handle engine temperatures in range of 200-300 degrees without boiling. We will use antifreeze or ethyl glycol mixed with water in a one to one ratio for our purposes as this is coolant not only meets the above-mentioned requirements but has also shown to be the most promising coolant as it is the most used in commercial cars. The following variants are available commercially.

Out of different available options, we will use the following mix:

- 70/30 mix of $C_2H_6O_2$/Water: -55 C / -67 F to 113 C / 235 F

We selected this mix because our engine needs to work in a range of 70 to 100 degrees and also needs to have capabilities to prevent freezing of water, therefore this is the most appropriate selection.

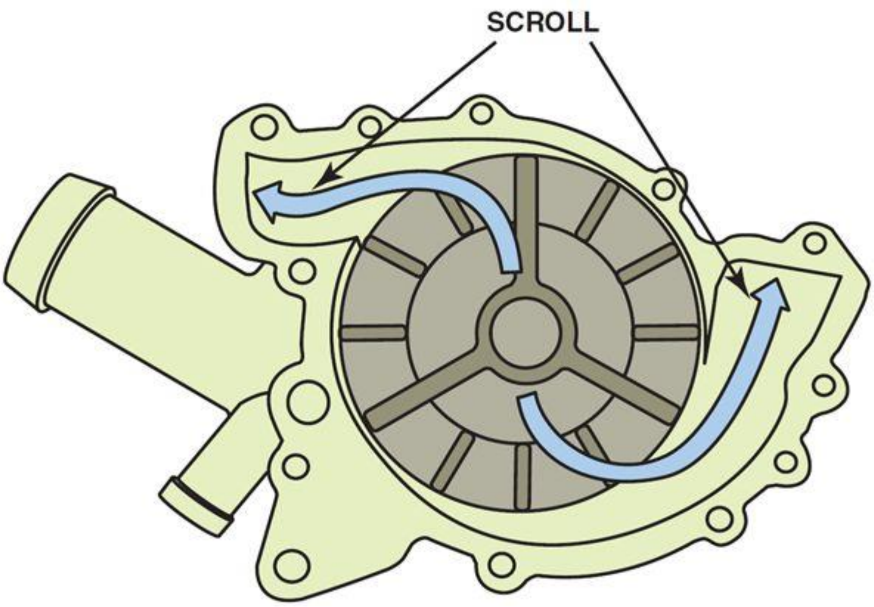
Water Pump

We will use a radial impeller pump which is run by the engine itself. Most cars have a pressure limit of 14 to 15 pounds per square inch (psi) or 100kPa which raises the boiling point another 45 F (25 C) so the coolant can withstand the high temperatures.

When the pressure reaches 15 psi, the pressure pushes the valve open, allowing coolant to escape from the cooling system. This coolant flows through the overflow tube into the bottom of the overflow tank. This arrangement keeps air out of the system.

The advantages of our pump are:

1. The smoothness of bearing, which supports the rotor on one side and the pulley on the other side.
2. The quality of profiles of the rotor, which can be in synthetic material reinforced with glass fiber, steel or cast iron, Quality is important for correct constant flow and to prevent cavitation effects.
3. The quality of seal, fitted between the rotor and the bearing.



Radiator Type

In most of today's vehicles you will find either a cross flow radiator or a down flow radiator. The primary difference between a cross flow radiator and a down flow radiator is where the tanks that hold the antifreeze coolant are located. A cross flow radiator design uses tanks that are located on the sides of the radiator core and cycle across the radiator core from side to side. On a down flow radiator, the tanks are located above and below the radiator core.

There are different types of radiators depending upon the type of material. Following are the materials:

1. Aluminum
2. Copper
3. Plastics
4. Brass

Aluminum has a high thermal conductivity, meaning it conducts heat very well, absorbing it faster. This state allows hot coolant passing aluminum tubes to be cooled instantly, which is beneficial when it returns for another cycle to cool the car engine. Depending on the manufacturer, a radiator made entirely from aluminum can conduct as much as 2,000 BTU per hour. Also, these radiators have aluminum cores that save money and weight by using plastic headers. This construction is more prone to failure and less easily repaired than traditional materials. That’s a big increase in efficiency when compared to its brass counterpart.

Considering the advantages of aluminum radiators over other radiators, we selected the aluminum radiator.

Calculations and Analysis

Following features are considered for a conventional car radiator:

Height = 350 mm

Width = 550 mm

Depth = 20mm

Thickness of fin = 0.30 mm

Pitch of fin = 1.50 mm

Separation between two adjacent tubes = 8 mm

Tube wall thickness = 0.4 mm.

$T_{H,avg}$ (hot water average temperature) = 85°C

Q_d as discharge rate fluid = 1.30 x 10^3 m^3/sec

Q_{req}= 38000 W

Wind velocity (V_{air}) = 150 km/hr.

Then,

Rate of heat transfer over the surface of radiator is given by:

$$Q = U \times A \times \Delta T_m$$

$$\Delta T_m = T_{H,avg} - T_{C,avg}$$

$$\Delta T_m = 85 - 40 = 45^o C$$

$$UA = \frac{1}{R_{Total}}$$

Where,

U is overall heat transfer coefficient between two fluids

R_{Total} total thermal resistance between water and air

θ_m is AMTD (arithmetic mean temperature difference)

$$R_{Total} = R_{in} + R_{cond} + R_{out}$$

Where,

$$R_{in} = \frac{1}{h_{in} \times A_{Total,in}}$$

Defining Nu as Nusselt number, k_{air} as thermal conductivity of air, Re as Reynolds number and Pr as Prandtl number

$$Re = \frac{\rho \times v \times L_C}{\mu} = 9.8 \text{ x} 10^3$$

$$Pr = \frac{\mu \times C_p}{k_{air}} = 1.8936$$

$$Nu = 3.66 + \frac{0.668 \left(\frac{D}{W}\right) \times Re \times Pr}{(1 + 0.04 \left(\frac{D}{W}\right) \times Re \times Pr)^{\frac{2}{3}}}$$

$$Nu = 6.246$$

$$Nu = \frac{h_{in} \times L_c}{k}$$

Where L_c is characteristic length. Therefore:

$$h_{in} = 2101 \text{ W/ m}^2 \text{ K}$$

Also,

$$A_{Total,in} = .9674 \, m^2$$

Therefore,

$$\boldsymbol{R_{in} = 4.92 \times 10^{-4} \, {}^oC \, W^{-1}}$$

Now,

$$R_{cond} = \frac{Th}{k_{tube} \times A_{total,in}}$$

$$\boldsymbol{R_{cond} = 0.0185 \times 10^{-4} \, W^{-1}}$$

Now,

$$R_{out} = \frac{1}{\eta_o \times h_{out} \times A_{total,out}}$$

$$A_{total,out} = 6.13 \, m^2$$

$$\eta_o = 1 - \frac{A_{s,fin\,total}}{A_{tot}(1 - \eta_{fin})}$$

Therefore

$$\boldsymbol{\eta_o = 0.985}$$

Let V is velocity of air flow, υ kinematic viscosity of air, $A_{s,\text{ fin total}}$ is total fin surface area, As un$_{\text{fin}}$ is total un-finned surface area

$$Re = \frac{\rho \times \upsilon \times L_{out}}{\mu} = 5.4 \times 10^4$$

$$Pr = \frac{\mu \times C_p}{k_{out}} = 0.7046$$

$$Nu = 0.036 \times Re^{5/4} \times Pr^{1/3}$$

$$Nu = 195.67$$

$$Nu = \frac{h_{out} \times L_{out}}{k_{out}}$$

Therefore

$$h_{out} = 225 \text{ W/m}^2\text{K}$$

And

$$\boldsymbol{R_{out} = 7.15 \times 10^{-4} \, {}^oC \, W^{-1}}$$

Putting the values, we get R_{Total}

$$R_{Total} = 1.21 \times 10^{-3}\, {}^{o}C\, W^{-1}$$

$$UA = \frac{1}{R_{Total}}$$

$$UA = 760\, W/C$$

Therefore,

The rate of heat transfer for the radiator:

$$Q = U \times A \times \Delta T_m$$

$$\boldsymbol{Q = 34200\, W}$$

The difference in heat rate:

$$\Delta Q = Q_{req} - Q_{act}$$

$$\Delta Q = 38000 - 34200$$

$$\boldsymbol{\Delta Q = 3800\, W}$$

References

[1] ASHRAE Handbook of Fundamental, “American Society of Heating, Refrigerating, and Air Conditioning”, Atlanta, GA,1988.

[2] ISO 8996, “Ergonomics of Thermal Environments – Determination of Metabolic Heat Production. ISO”, Geneva,1989

[3] halls.md/average-height-men-height-weight/

[4] Mohammad Ali Fayazbakhsh and Majid Bahrami “Comprehensive Modeling of Vehicle Air Conditioning Loads Using Heat Balance Method” Simon Fraser University

[5] Oum Saad Abdulsalam, Budi Santoso, Dwi Aries “Cooling Load Calculation for Vehicle” Sebelas Maret University

[6] Valerie H. Johnson “Heat-Generated Cooling Opportunities in Vehicles” National Renewable Energy Laboratory

[7] B. Paul Vinofer, S.Rajakumar “Cooling load calculations and design of air conditioning system in automobile” Regional Office Anna University

[8] Zheng, Yinhua & Mark, Brown & Youmans, Harry. (2011). A Simple Method to Calculate Vehicle Heat Load. 10.4271/2011-01-0127

[9] Choosing and Using Alternative Refrigerants for Vehicle Air Conditioning EPA-130-F-97-065`

[10] thebalancesmb.com/different-types-of-water-pumps-844599

[11] “Designing a More Effective Car Radiator” Waterloo Maple Inc.