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Analytical Design and Verification of Automotive Radiator using 1-D Simulation

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Abstract: The demand for greater compactness & higher performance is increased along with vehicle comfort & fuel economy. After introducing with necessity of engine cooling systems in the automobile &components of engine cooling system analytical calculations are made for sizing of radiator for given heat load &application. Aerodynamic drag reduction is desirable to minimize vehicle fuel usage, yet a practical requirement of vehicle design is adequate airflow through the radiator to ensure adequate engine cooling under all operating conditions. Therefore, it is important to optimize the cooling air flow through a vehicle's radiator without detriment to aerodynamic drag. These results are compared more sophisticated simulation software result which is customized for the sizing of heat exchangers. The comparison of results shows good agreement in between. This paper introduces an analytical approach for design and sizing of a Heat exchanger and also offers a starting point for future research in the design of Automotive cooling system.
Keywords: Heat exchanger, Radiator, fuel economy, automotive cooling system

I. INTRODUCTION

Virtually all passenger cars produced in the world today have at least two compact heat exchangers as important components. The first is the radiator that serves to reject the waste heat from the engine cooling fluid to the atmosphere, the second transfer's thermal energy from the same source to the passenger compartment to provide thermal comfort. A significant & increasing number of automobiles are fitted with air conditioning for which a further two compact heat exchangers are required; a refrigerant evaporator to extract the heat from the passenger compartment air & a condenser to reject the cycle waste heat from the refrigerant to the environment. In vehicles with turbo-charged engines, charge air coolers are frequently used to cool the compressed air before it passes to engine. The final category of automotive compact heat exchanger covers oil coolers that can be required to provide cooling for engine or automatic transmission oil.

All the heat produced by the combustion of fuel in the engine cylinders is not converted into the useful work at the crankshaft. A typical distribution for the fuel energy is given in the figure

In this paper, analytical design procedure of a Radiator is described along with verification technique using 1-D simulation to ensure the Radiator performance for calculated Tube-fin matrix size.

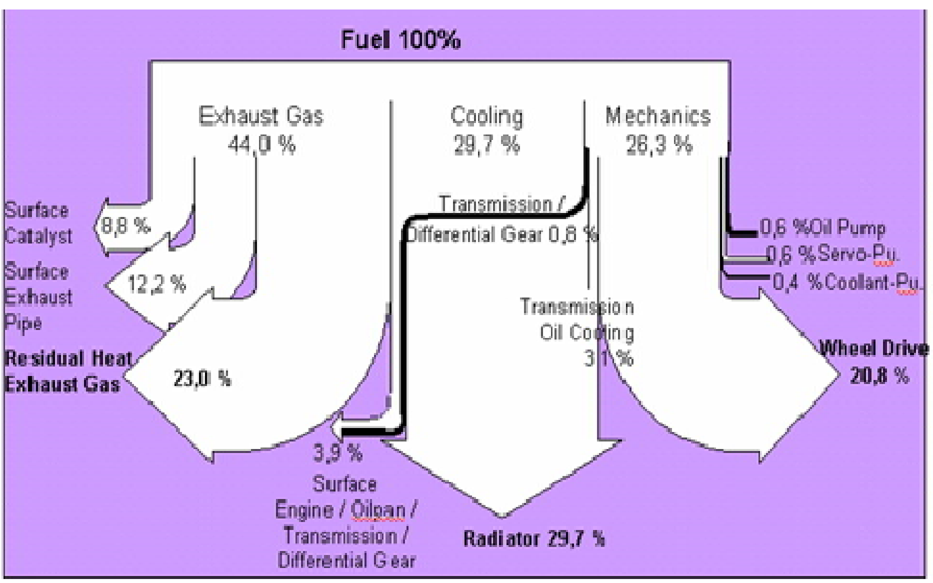

Fig.1: A typical distribution for the fuel energy

II. SIGNIFICANCE OF ENGINE COOLING

A. Necessity of Engine cooling

From figure 1 typical energy distribution from Total 100% Fuel Input is :

26.3%	Power
29.7%	Coolant
44%	Exhaust

Itisseenthatthequantityofheatgiventothecylindricalwallisconsiderableand soitisrequiredtoprovidesuitablemeanstodissipatetheexcessheatfromthe walls so as to maintain temperature below certain limits. Following are some reasons why cooling is required at all:

1) The lubricating oil used determines the max. Engine temperature. Depending upon the type of lubricating oil used, this temperature ranges from 160 to 200 deg. above this temperature the lubricating oil diorites& causes engine seizes.
2) If the cylinder head temperature is high, the volumetric efficiency and hence power output of the engine is reduced.
3) High engine temperature may results in pre-ignition & detonation.
4) Strength of materials used for various engine parts usually decreases with an increase in temperature. Oil seals & gaskets my burn out which can cause leakage & also increased temperature leads in excessive thermal stresses may results in cracking.

B. Effects of overcooling

However cooling beyond certain limits is not desirable because it decreases the overall efficiency due to following reasons:

1) At low temperatures, the sulphurous & sulphuric acids resulting from the combustion of fuel attacks the cylinder barrel. The dew points of these acids vary with pressure and hence the critical temperature, at which the corrosion assumes significant proportions, varies along the cylinder barrel. To avoid the condensation of acids the coolant temperature should be greater than 70 °C.
2) When the fuel is burnt in the engine water is formed as one of the by- product of the combustion. Most of this water vapor passes out from exhaust & some gets leaked through the piston rings into the crankcase. If the surrounding parts are cooled enough, this result in condensed liquid causes to churn in the oil &forms a thick sludge which further blocks the oil pump & restrict oil circulation.
3) Tests have shown that wear on cylinder walls is eight times greater with a coolant operating temperature of 38°C compared with 82°C.
4) Thermal efficiency is decreased due to more loss of heat to the cylinder walls.
5) The vaporization of fuel is less; this results in fall of combustion efficiency.
6) Low temperature increases viscosity of lubricant and hence more friction encountered, thus decreasing mechanical efficiency.

In the paper presented by D. Ganga Charyulu, Gajendra Singh, J.K. Sharma performance evaluation of over designed radiator is done and the design is then optimized by changing no of tubes as well as the air flow velocities.
This reference also specifies effect of material for fins and tubes on the performance radiator.
The objective of this study is to assess the performance of the existing radiator under the following conditions of operation:

1) The characteristics of the radiator core, for any number of tube rows, water flow rate and air flow rate. It will help the designer selecting the number of tube rows for a given application.
2) The performance evaluation with varying fouling factors. Performance is evaluated on the basis of fouling factors given in the TEMA standards. Since, the quality of fluids viz. air, water and oil in India are different from TEMA standards and also varies from region to region, it is, therefore, thought desirable that the performance of heat exchanger should be evaluated with different fouling factors.

In another reference, Peter Ambrose & Peter Leu, of, Behr GmbH & Co., Stuttgart future of engine cooling systems has been described. In next generation engine cooling system the mass flows of the participating media can be proportioned by electronically controllable coolant pumps, control valves, fans, and radiator shutters in order to achieve optimum temperatures for engine cabin and components.

By combining various different measures fuel consumption can be reduced by about 5% as can exhaust gas and acoustic emissions to a certain extent. The warm-up period can be shortened significantly, thus contributing to a higher level of comfort in the passenger compartment.

An appropriately designed and controlled engine cooling system can contribute significantly towards meeting or at least supporting the development targets of the automotive industry. Such benefits include:

1) Reduction of fuel consumption by optimum thermal boundary conditions for combustion engine, gear box and components , reduced power consumption of pumps and fans
2) Reduction of exhaust emissions
3) Reducing the thermal load on the engine, components and fluids and therefore extending service life by avoiding temperature peaks and reducing medium operating temperatures, reduction of temperature fluctuations and pressure stresses and their gradients.
4) Reduction of weight and space requirements by using compact and more efficient heat exchange.

In the paper presented by Mr. Katkar & Mr. Agrawal , very important aspect of radiator design is put forward that is the use of simulation methods like CFD to optimize the airflow distribution through the radiator to improve radiator performance. This study has been done for the small truck radiator optimization. This study is more related to the shroud design (radiator cover) than the radiator design but shows the importance of the airflow distribution through the radiator.

Increasing demands on engine power and performance to meet ever increasing customer requirements to meet increased load carrying capacity for trucks have necessitated improving the heat management system of the vehicle. Manufacturers of commercial vehicles are facing a substantial increase of heat release into the cooling system.

The main sources for this increase are: more stringent emissions leading to new combustion technologies and increased power of the engines. The total increase in cooling requirement may be up to 20% over the current level. At the same time the noise levels have to decrease and the fuel economy have to be improved.

A clear understanding of the airflow pattern inside the radiator cover is essential for optimizing the radiator cover shape (shroud) to increase the flow through the radiator core, thereby increasing the thermal efficiency of the radiator. The paper presents different solutions used for the optimization and there comparative studies to finalize the best one.

In the paper presented by Mr. Changhua Lin (Delphi Automotive Systems), Jeffrey Saunders (Monash University) & Simon Watkins (RMIT University), the effect of ambient and coolant temperature and coolant flow rate on the performance of the radiator is studied. This is done through the consideration of specific dissipation. These results indicate that the effect of ambient and coolant inlet temperature variation on SD is small(less than 2%) as compared to the effect of coolant flow rate.

Aerodynamic drag reduction is desirable to minimize vehicle fuel usage, yet a practical requirement of vehicle design is adequate airflow through the radiator to ensure adequate engine cooling under all operating conditions. Therefore, it is important to optimize the cooling air flow through a vehicle's radiator without detriment to aerodynamic drag. In the design and development stage of a vehicle engine cooling system, the final evaluation of the system is usually made by either on-road testing or placing the vehicle in an environmental wind tunnel (EWT) and simulating some of the climatic properties that exist on the road. However, environmental wind tunnels have tended to evolve in a rather pragmatic way. Initially dynamometers were installed to test the car operation. An air supply was needed to cool the engine, so a jet of air was directed at the radiator.

III. PRODUCT OVERVIEW-RADIATOR

The engine-cooling radiator consists of an array of tubes that carry the hot engine coolant with a secondary surfaces attached to the outside of the tubes over which the cooling air flows. The tubes can be arranged horizontally in what is known as cross-flow radiator or vertically in a down-flow radiator. Flow is usually single pass on the tube side, although two or even three pass configurations do exist. A tank is fitted to each end of the tubes in the array to distribute the flow to tubes & to collect it the other end. These days most tanks are injection moldings, usually of glass filled nylon material. Details of radiator construction is as given in the below figure 2.

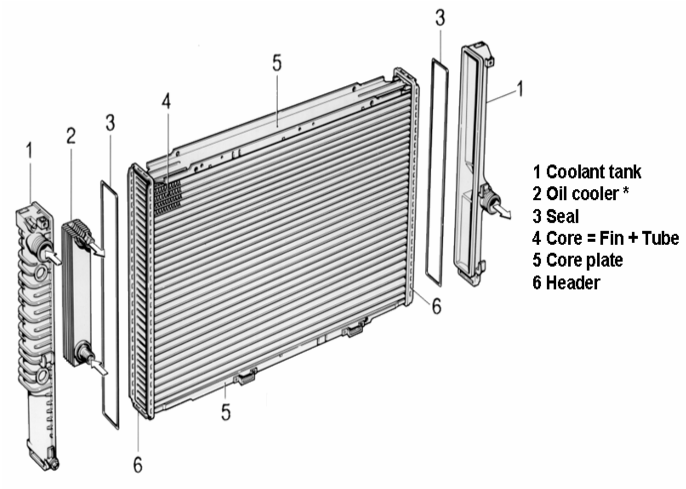

Fig. 2 :Radiator component descriptions

Radiator types based upon material are ,

Sr. No.	*Aluminum Radiator*	*Cu Brass radiator*
1	Less weight	More Weight
2	Uniform material of tube and fin give better heat exchange	A dissimilar material of fin & tube. Tin is joining material so heat transfer is less
3	Low cost	More cost
4	Not possible to repair	Possible to repair
5	Corrosive effect is more	Corrosion is less.

Table 1: Difference in radiators types based on material

Aluminum radiator performance when compare to Cu-Brass radiator performs 40% higher.

IV. PROBLEM DEFINITION

Design and development of modular components is a general trend in automobile industry to reduce size weight, cost , No. of parts and to protect the environment. For an automotive application , the efficient cooling system need to be designed which will be worth in terms of money as well.

Inputs and boundary conditions for design of Radiator ,

Basic Data:	Operation point		1
	Designation		Max P
	Horse power	kW	150kW
	RPM Engine	min^{-1}	4000
	RPM Fan	min^{-1}	4000
	vehicle velocity	km/h	30
	Ambient temperature (Max 50 degrees)/ Recirculation	°C	46
	vehicle pressure drop (zeta)		0.07
Radiator:	Composition Coolant: Water / Glysantine 50 / 50 Vol. %		
width = 450	Volume flow coolant	lpm	150
length= 685	entrance pressure coolant (absolute)	bar	1
depth= 26	Required cooling capacity	kW	104 kW
	Allowed max. coolant temperature X Max. value 115 C	C	115 C
	Max. allowed pressure drop through radiator	mbar	19.8
Further required elements:	e.g. Engine oil cooler, transmission oil cooler, steering oil cooler, etc.	kW	12
	Fan-Type : slip		
Cooling air mass flow:	Electric motor; Visco fan drive X; Diameter : 500; No. of blades : 9; ratio: i= 1:1.11	%	10.0

Table 2 : Technical inputs for design

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Before starting any radiator design we need to have some input data like:

1) Engine horse power & torque curve
2) Engine full load heat rejection to the coolant
3) Coolant pump & coolant system resistance curves
4) Automatic transmission heat rejection to the coolant
5) Radiator heat transfer required
6) Air side pressure drop
7) Coolant side pressure drop
8) Fan performance
9) Air flow required

V. PREDICTIVE PROCESS FOR SIZING OF ENGINE COOLING SYSTEM

A. Fundamentals for calculations

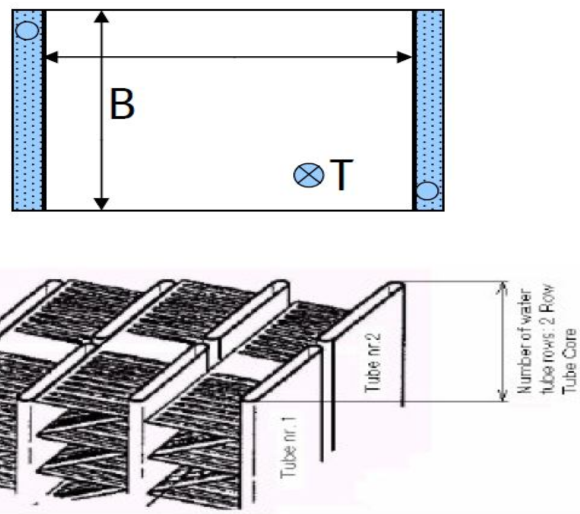


Fig. 3: Dimension nomenclatures

In coolant flow direction = H (Height)
In air flow direction = T (depth)
Perpendicular to H and T = B (width)

Constraints for selection:

✓Number of tubes (pitch)
✓Tube width (bore)
✓Tube depth
✓Fin density
✓Fin height (pitch)
✓Fin thickness
✓System depth

B. Analytical calculations

The heat rejected to coolant greatly depends upon the engine type. It has been observed from the tests that for a small high speed engine the heat rejected to the coolant can be as high as 1.3 times the bhp developed, while for an open chamber engine it is only about 60 % of the bhp developed.
If only an engine power curve is available, a conservative estimates of the WOT heat rejection is,

$$Q\ (\text{BTU/Min}) = 0.65 * \text{BHP} * 42.2$$

The engine coolant picks up heat from the engine block & cylinder heads. The coolant, as it flows through the radiator, transfers this heat to the air. The different modes of heat transfer consist of forced and natural convection, radiation and conduction. The primary mode we are concerned with sizing the radiator is the forced convection of heat from the coolant to the air stream flowing through the radiator under steady state condition.

The heat transfer is expressed as,

$$Q = m\ Cp\ DT$$

Since the amount of heat given up by the coolant, as it flows through the radiator is equal to the amount of heat transferred to the air,

$$Q\ coolant = Q\ air$$

The heat transfer rate, Q, is also function of the radiator itself and is expressed by,

$$Q = Uo \times Ao \times LMTD$$

The overall heat transfer coefficient is a function of the inside & outside heat transfer coefficients of heat transfer and conduction,

$$1/Uo = 1/ho + Ao / (hi\ Ai) + R$$

The philosophy of how several design loads should be for radiator sizing varies between different vehicle and engine manufactures. One of the largest variables is the maximum engine coolant temperature allowed which can range between 107°C to 132°C. Other variables are allowable weight and frontal area. The first step in beginning the predictive process is to size the radiator at maximum engine horsepower or full load conditions.

Figure 4 shows the horsepower curve, from this curve one can find the maximum horse power to occur at 4000 RPM.

Engine heat rejection to the coolant = 91kW @4000 RPM

If the vehicle has an automatic transmission and it is cooled by an in tank cooler, the transmission heat loss must be added.

From figure 5, automatic transmission heat rejection = 13 kW

Therefore,

Total heat rejection = 104 kW

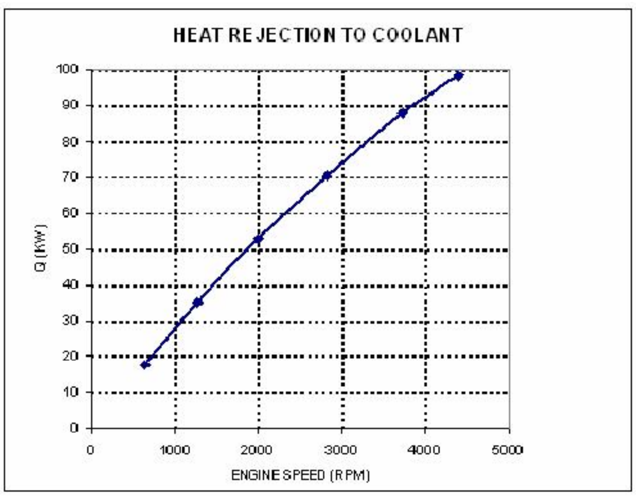


Fig. 4: Full load heat rejection to coolant

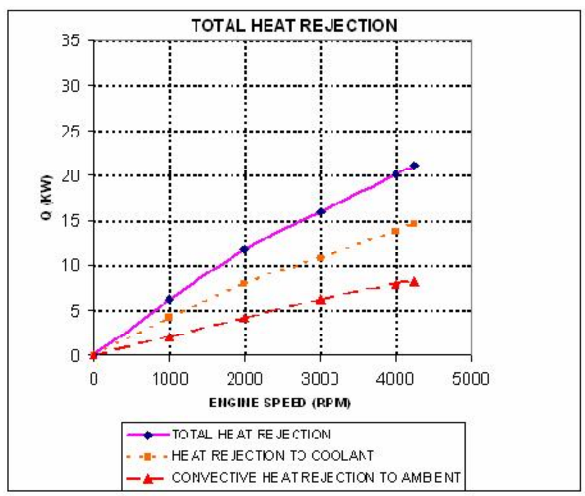


Fig. 5: Automatic transmission heat rejection to coolant

The coolant entering the radiator (top tank temp) and the ambient air temperature now needs to be specified. If the radiator is sized according to full load condition and WOT, it will be oversized for normal driving and will add cost and weight to the cooling system design. Therefore air and coolant temperatures must be selected realistically to fully protect the engine and allow low cost.

Engine durability is also dependent on the coolant temperatures so the engine design specifies the time at temperature histograms. For the coolant, ideally, at high speeds engine load is low and airflow is high so coolant temperature is low but in city traffics speed is low and load is high so air flow is low and hence coolant temperature is high. Higher the coolant temperature allowed lower the size of radiator since differential between coolant & air temperature is more.

Following is the condition given by the customer for maximum allowed air and coolant temperatures.

- ✓ Ambient temperature = 35°C
- ✓ Air temperature into radiator = 35°C
- ✓ Air temperature out of radiator = 88°C

✓ Coolant temperature into radiator = 121°C
✓ Max. Coolant temperature out of the radiator = 110°C

The next step is to specify the radiator frontal area available.

The frontal area is determined by a predictive model using similar vehicle configurations , vehicle system resistance and the mass flow through the radiator contributed by the fan and ram air. An outcome to the radiator frontal is the amount of free flow area required in front of the radiator for adequate air flow to cool engine from idle to low speed WOT.

Radiator frontal area to be considered as 0.3623 m^2 as it is depending on the space constraint provided by the customer.

Radiator dimensions = 711 mm x 508 mm x 18 mm

Fig. 6 :Radiator Dimensions

Radiator nomenclature is B x H x T. Thus,

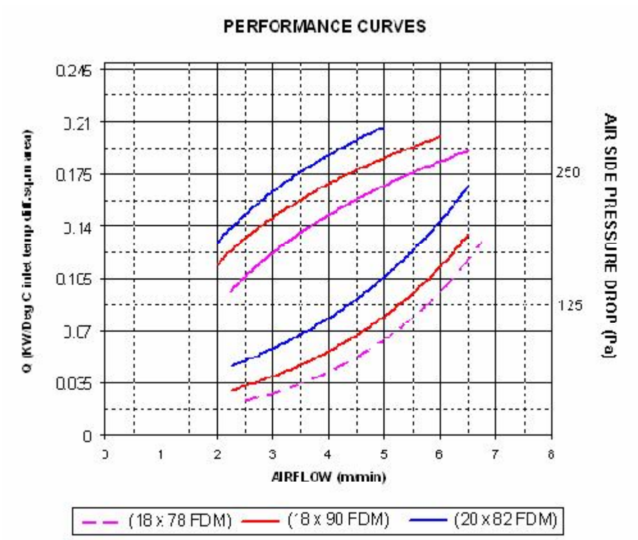
Width= 711 mm
Height = 508 mm
Depth = 18 mm

Radiator selection has three degrees of freedom

1) Radiator type (Cross flow , down flow)
2) Radiator depth
3) Number of fins (Fins per inches)

As radiator depth and fin density increases the fan requirement increases.

The radiator selected should be as thin as possible and have the least number of fins per inch. Radiator performance curve is as shown in figure 7.



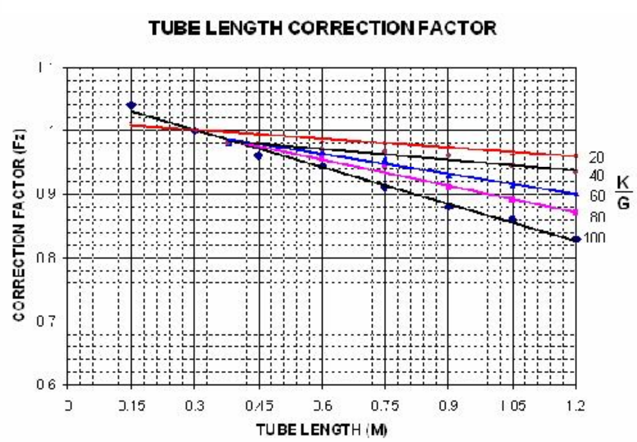


Fig. 7: Radiator heat transfer and air side resistanceFig. 8 : Radiator heat transfer correction for tube length

The radiator curve of figure is presented as heat transfer, Q* vs. air velocity. It must be normalized heat transfer quantity so that different radiators may be compared on equivalent basis.

$$Q^* = 104 / ((121-35) \times (0.3653))$$
$$Q^* = 48 \text{ kCal / hr m}^2 {}^\circ\text{C}$$

Since radiator performance varies with the tube length, tube length correction factor should be applied as shown in figure 8.

Tube length = 711 mm

Before the tube length correction can be used, however the coolant flow rate must be known. The coolant flow is obtained from the coolant system resistance and pump curve in figure 9.

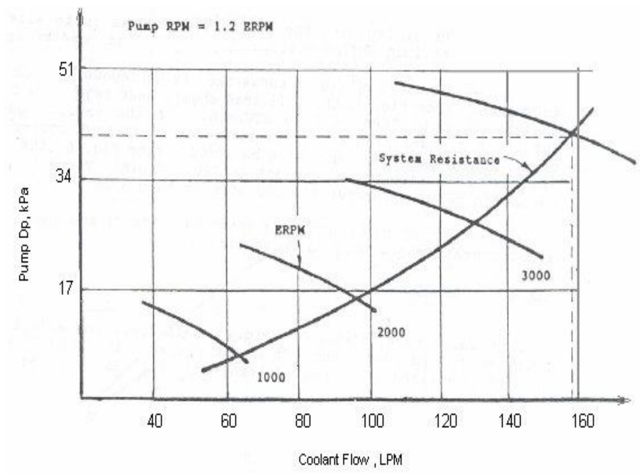


Fig. 9: Coolant pump characteristics

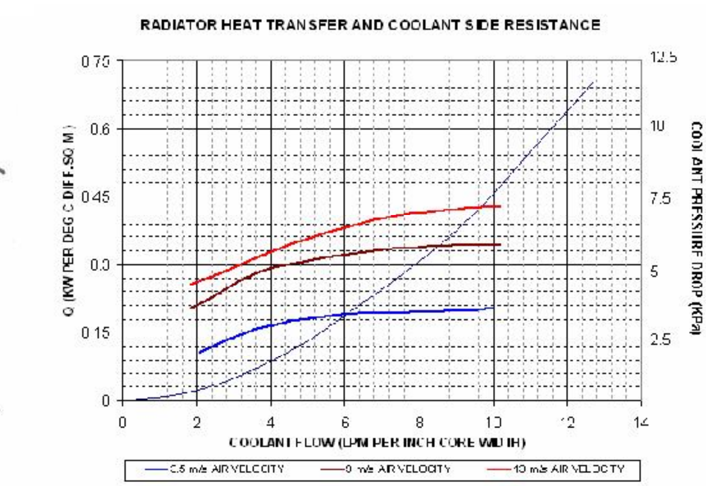


Fig. 10: Radiator heat transfer with coolant flow & air flow

The intersection of pump throttling or RPM curve with the system resistance curve defines the coolant flow. From figure 6.8 , the coolant flow at 4000 Engine RPM (4800 Pump RPM) is about 150 LPM.

Now we can calculate, tube length correction factor as follows,

$$K = 9.8 * 100$$
$$G = 40.7/ (28/12) = 17.4$$
$$K/G = 980 / 17.4 = 56$$

So from the figure 6.7, the tube length correction factor is about 0.96.

Now we have to calculate air flow across the radiator.

Radiator FDM = 90

The temperature of air leaving the radiator should not exceed 90 °C to ensure that the under hood global temperature remains in the area of about 107°C to protect against electronic device failures, fuel vapor lock & plastic components deformation.

The air flow required is calculated from,

$$Q = m_{air} * Cp * (Tao - Tai)$$
$$104 = m_{air} * 1.004 * (190-95)$$
$$m_{air} = 120 \text{ kg/min} = 2 \text{ kg/s}$$

The value of Q* should protect about a 2 % radiator tube side & air side fouling. The protected Q* including tube length correction factor is,

$$Q^* = 9.8/0.96 * 1.02$$
$$Q^* = 10.4 \text{ BTU /min per °F ft2}$$

From the graph,18 mm Deep, 90 FDM radiators just meet this requirement at 2 kg/s air velocity. Coolant velocity has to be checked to protect against the possible tube side & tube header erosion corrosion caused by high coolant velocity (normally when it is more than 3 m/s).

The density of 50-50 water glycol at 90°C is about 1015 kg/m3

Number of tubes = 47

Total tube cross sectional area = 47 * 0.09144 = 4.2 mm2

Tube side coolant velocity = 154 /1.015*60*4.2 = 1.88 m /sec

The tube side velocity is acceptable and any further increase in tube side flow beyond a given coolant flow rate will only results in a nominal increase in heat transfer with a large increase in tube side pressure drop.

Radiator effectiveness, Erad = (Tao-Tair) / (Tci -Tco)

Erad = (88-3 5) / (121-35)

Erad = 61 %

Now, temperature of the coolant out must first be determined.

The temperature differential of the coolant from basic heat transfer equation is,

(Tco -Tci) = 104*3600/156.5*60*3.684 = 11°C

Where,

Cp of 50-50 water glycol = 3.684 kJ /kg °C

Tco=121-11 =110 °C

LMTD = (110-35) - (log (110/75))

= 42 / 0.81

LMTD = 51 °C

Calculations for Overall heat transfer coefficient (Uo):

Heat rejected to the coolant,

Q = 104 KW

Ao = 10.38 Sq. m outside radiator surface

Uo = Q/ (Ao * LMTD)

Uo = 104 / (10.38 x 51)

Uo = 0.196 kW/m2° C

Now inside & outside heat transfer coefficient can be estimated.

Inside heat transfer coefficient can be calculated with the help of Dittus & Boelter equation,

Nu = 0.023 * (Re)0.8 (Pr)0.3

Also

Nu = hi x Dh / K

Thus,

hi x Dh / k = 0.023 * (Re) 0.8 (Pr) 0.3

Now each of the parameter in this equation has to be calculated,

Dh = 4 * tube cross section / wetted perimeter

Dh = (4 x 2.7 x 10 -5) / [(2) x (1.7+18)] x 10-3

Dh = 0.003 m

Also for water - glycol of 50 - 50 mixtures

σ = 1015 Kg / m3

μ = 0.00063 N.s/m = 6.31 x 10-4 kg/m.s

k = 0.42 w / m °C

Cp = 3625 J/ Kg°C

V = 1.88 m/s

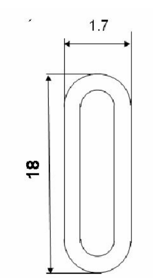


Fig.11 :Tube dimensions

Reynolds Number Re = Dh x V x σ / μ

Re = (0.003 x 1.88 x 1015) / 0.00063 = 9086

Prandtl Number, Pr = μ x Cp / k

Pr = 0.00063* 3625 / 0.42 = 6.38

Now inside heat transfer coefficient can be calculated,

$$hi = [(0.023 \times 0.42) / 0.0033] \times (9086)0.8 \times (6.38)0.3$$

$$hi= 7492 \ w /m^2 \ °C$$

An extended surface heat exchanger inside & outside coefficient per unit airside area should be about the same for good exchanger design.

i.e.1 / ho = Ao/ (Ai x hi)

The outside coefficient may now to be determined from the equation,

$$1/ho = 1 / Uo – Ao / (hi \times Ai) – R$$

Ai = Inside heat transfer area= 1.16 m2

R is metal resistance, assumed to be negligible

Thus,

$$1/ho = 1/196 – [10.38 / (7492 * 1.16)]$$

$$ho = 249 \ w / m2 \ ° C$$

Airside pressure can be directly taken from the graph:

Thus,

Air side pressure drop = 1050 pa at 4.7 m/s air velocity and 90 FDM core surface.

Summary of conditions:

- ✓ Requested Radiator heat rejection = 104 KW
- ✓ Ambient temperature = 35 °C
- ✓ Air temp into the radiator = 35 °C
- ✓ Air temp out of radiator = 88 °C
- ✓ Coolant temp into radiator = 121 °C
- ✓ Coolant temperature out of radiator = 110 °C
- ✓ Radiator frontal = 0.36 m2
- ✓ Radiator core dimensions = 711mm wide x 508 mm high x 18 mm deep
- ✓ Radiator type = Cross flow
- ✓ Radiator = 90 FDM
- ✓ Mass air flow = 2 Kg/s
- ✓ Volume air flow = 1.7 m3/sec
- ✓ Air velocity = 4.7 m/s
- ✓ Radiator effectiveness = 61 %
- ✓ LMTD = 51°C
- ✓ Coolant flow = 150 LPM
- ✓ Coolant velocity = 1.88 m/s
- ✓ Uo = 196 w/ m2 °C
- ✓ hi = 7492 w/ m2 °C
- ✓ ho = 249 w/m2 °C
- ✓ Air side pressure drop = 1050 Pa

Block Dimensions (B x H xT)	711mm x 508mm x 18 mm
Frontal area	0.3623 m2
Fin Density	90 fins/dm
Number of tubes	47
Tube Pitch	10

Table 3:Tube-Fin Matrix dimensions

Simulation for the given boundary conditions:

✓ Boundary conditions:

Te (Coolant) = 121°C , Te (Air) = 35°C

Pe (Coolant) = 2.5 bar ,Pe (Air) = 1,0 bar

Coolant flow = 150 LPM
Air speed = 2 kg/s / 4.7 m/s
Coolant = 50 % water + 50% Glycerin

✓ Simulation and calculation:

Simplified 1-Dimensional simulation of engine cooling components is done using 1-D simulation software.
Coolant coming out from the radiant is divided into two parts.
One part goes through condenser and other through charger air cooler then again both meetand will pass through radiator.

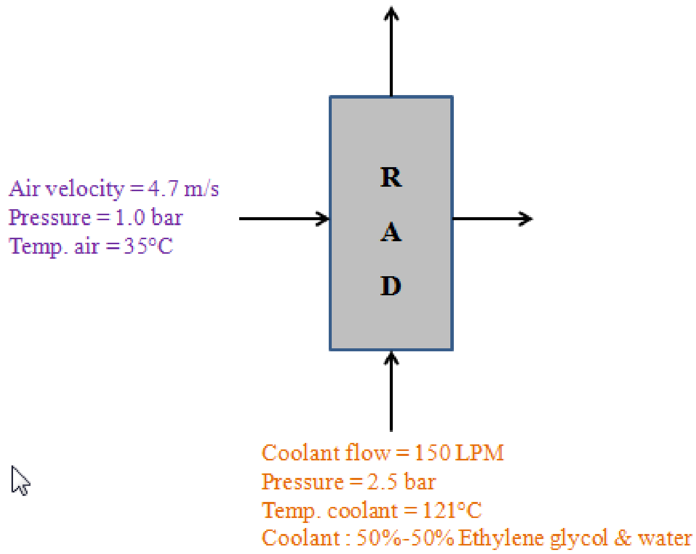


Fig.12 :1-D simulation of Radiator

Air Speed (m/s)	Heat rejection Kw	Air side pressure drop Pa	Coolant side pressure drop Pa
2	60.5	520.6	39856
4	103.42	1070.5	43528
6	135.8	2036	45235
8	165.43	3650	47897

Table 4:Tube-Fin performance table

Sr. No.	Description	Requirement	Simulation Results
1	Coolant flow rate	150 LPM	150 LPM
2	Air frontal velocity	5 m/s	4.7 m/s
3	Coolant inlet temperature	121 °C	121 °C
4	Air Inlet temperature	35 °C	35 °C
5	Total heat dissipation	104 KW	103.9 KW
6	Coolant outlet temperature	Max. 115 °C	110 °C
7	Air outlet temperature	90 °C	88 °C
8	Airside pressure drop	Max. 1240 Pa	1070 Pa
9	Coolant side pressure drop	Max. 50662 Pa	43996 Pa

Table 5 :Radiator performance comparison matrix

VI. CONCLUSION

The brief overview of this case study that has been given revealed several points.

1) Optimal design of cooling system leads to less fuel consumption & more cabin comfort but at the same time compactness needs to be ensured.
2) Downsizing of engine and less under hood package space are the key challenges in designing engine cooling system.

3) Analytical design procedure for industrial design of automotive radiator system is gives coarse estimations for core sizing but to make it more optimize we need to use simulation programs.
4) Simulation of cooling system helps design engineers to ease the work of design by using different configurations.
5) This work can be continued for studying NTU method for radiator design. In depth study of more simulation techniques are necessary.
6) The Key design considerations to design a optimal Radiator are, Compactness, Low Pressure drop , Low cost , High volume , Durabilty requirement , Heat transfer and fluid flow.

VII. ACKNOWLEDGMENT

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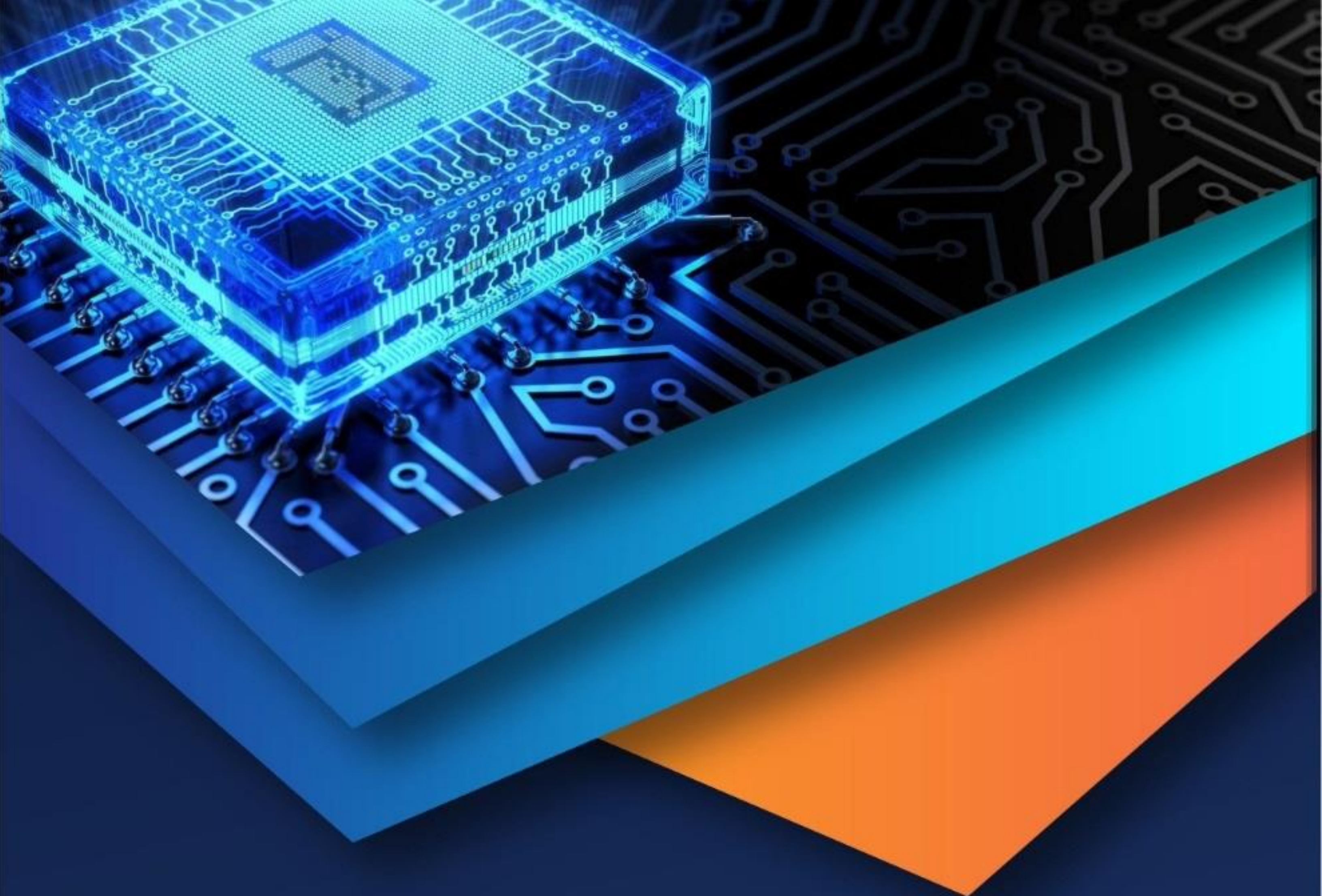
NOMENCLATURE

Q = Total amount of heat transfer
m = Mass flow rate
DT = Temperature difference
U_0=Overall Heat transfer Co-efficient
h_0 = Outside convection heat transfer coefficient
h_i = Inside convection heat transfer coefficient
R = Thermal Resistance
Cp = Specific heat capacity
m_{air} = Mass flow rate of air
E = Effectiveness
Tao = Ambient air outlet temperature
Tai = Ambient air inlet temperature
Tco = Coolant outlet temperature
Tci = Coolant inlet temperature
LMTD = Log mean temperature difference
A_0 = Outside Area
Nu = Nusselt number
Re = Reynolds number
Pr = Prandtl number
Dh = Hydraulic diameter
V = Velocity
$P_{e(Coolant)}$ = Coolant Pressure
$P_{e(Air)}$ = Air Pressure



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