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AUTOMOBILE RADIATOR DESIGN AND VALIDATION

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Abstract:- *The purpose of this research is to primarily design a cooling system for a formula styled race-car which is entirely designed, manufactured and tested by graduate students from colleges across the world. The primary requirements of the vehicle are to have high acceleration, low weights and an ability to endure. Keeping in mind the following requirements and some governing rules made by the sanctioning bodies, the students begin their design procedure. The engine used by us for the vehicle is a 4-stroke single cylinder 390 cc one, opted for its excellent power to weight ratio. The approach towards the design of the cooling system has not only been a theoretical one, but also a pragmatic one. This research begins with experimentally determining some important data of the engine. This data is used to determine the quantified requirements of the system during the various modes of vehicle operation. Based on the targeted values, the sizing of the radiator is done using basic heat transfer concepts. Further optimization is done using CFD analysis. The reason for not opting for a conventional wind tunnel is that it does not holistically validate the system's capability. This research involves the development of a procedure to validate the system in its operational state on the vehicle. Various sensors are deployed around the vehicle to acquire real time data which is further processed to determine the actual efficiency of the system. Based on the above approach, a fruitful way of validating the cooling system has been established and the co-relation between the expected and acquired results showed that both the values commensurate. The theoretical and CFD results were able to accurately predict the actual heat transfer at higher RPM and at lower RPM the predicted values were more than the actual heat rejection.*

KeyWords: *Heat transfer, CFD simulation, wind tunnel, on-track validation, data acquisition.*

1. Introduction

Formula Student is an SAE affiliated student level design competition, where students across the world design and fabricate a formula styled race car. The car is required to be light weight, quick and enduring. The official team of our college has been participating in these competitions since 2014. A combustion power-train with a single cylinder 390 cc engine has been used to power the vehicle. The students are required to abide to the rule book made by the respective sanctioning bodies of the competitions. As per the rules, the coolant to be used is distilled water and hence this paper is based on the research carried out by us to design a custom cooling system for the vehicle. Basic heat transfer concepts along with C.F.D analysis will be used for the optimization of the same. The inputs for the calculations and further analysis have been experimentally obtained and the complete validation of the design has been physically verified using a standalone D.A.Q system.

2. Methodology:

Various experiments were performed to determine the engine characteristics. The data obtained from these experiments was then used for theoretical calculations, CFD and validation. Initially, the radiator was sized considering the engine load to cooling system by theoretical calculations. These calculations were also used to determine radiator performance at various air velocities passing through radiator and mass flow rate of coolant. These values were cross verified using CFD simulations in Star CCM+. The radiator was then manufactured and tested practically with various sensors installed and data was logged in dynamic condition. Comparison is made between the above results to determine variation in actual performance and designed performance.

3. Flowchart:

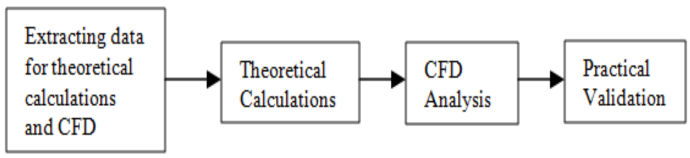

Fig. 1: Methodology Flowchart

4. Experiments:

4.1. Experiment to determine engine load to cooling system:

The engine used for powering the vehicle had values of stock power and torque as 32 kW and 35 Nm respectively. Before starting the fundamental design and calculations, certain engine parameters are required. Hence, an experiment was performed for calculating the change in the temperature of the coolant across the engine at different crankshaft rotation speeds. A system comprising of the engine, two thermistors and a pressure cap in series was used as shown in the figure below for calculating the change the temperature of the coolant. The thermistor readings were logged with the help of a data logger. The ΔT has been used for calculating the cooling load of the engine.

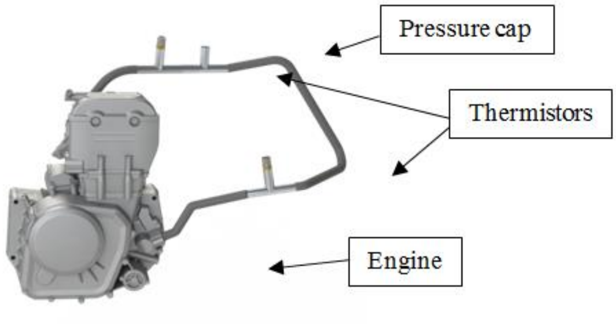

Fig 1: Experimental Setup

Table 1: Temperature difference across engine at different engine RPM

Engine Speed	Coolant Temperature Sensor	(T1)	(T2)
2500	70	Engine	68.5
3500	76	76.3	72.6
4500	78	77.8	73.7
5500	80	79.85	75.1
6500	82	82	76.6
7500	82	82.4	77.5

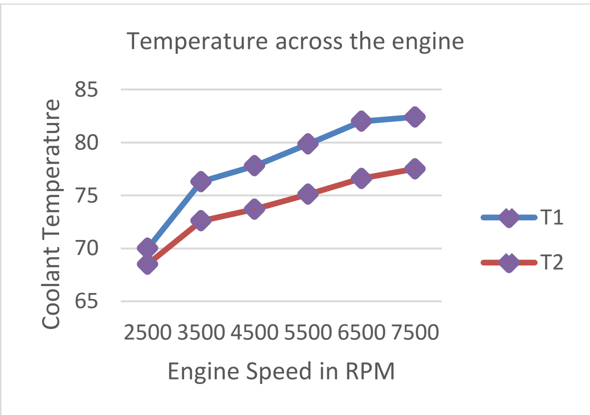

Chart 1: Temperature difference across engine at different engine RPM

4.2. Experiment to determine mass flow rate of coolant at various engine RPM:

This experiment was conducted to determine the mass flow rate of the water as a function of engine crankshaft rotation speed. The mass flow rate of water is varied with the pump which is geared to the engine crankshaft. This test was hence performed to find mass flow rate of the water at different crankshaft rotation speed. The engine speeds at which the experiment was performed were from 2000 rpm to 7500 rpm. A digital flow-meter was used in series with the engine, radiator, two thermistors and a pressure cap. Following is the set-up of the experiment performed.

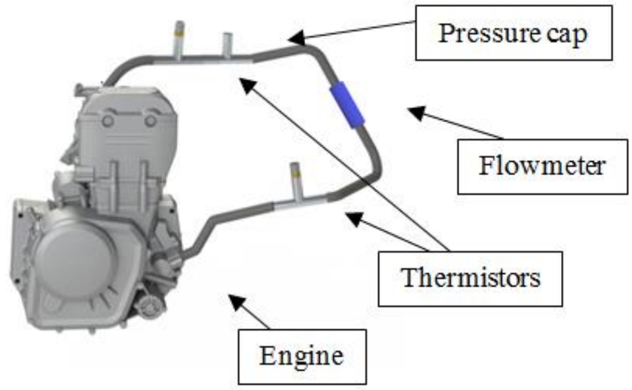

Fig 2: Experimental Setup

Table 2: Mass flow rate of coolant at various engine RPM

RPM	Mass flow rate (LPM)	Mass flow rate (kg/s)
2500	10.78	0.18326
2800	13.76	0.23392
3500	14.2	0.24157
4500	19.79	0.33643
5000	21.65	0.36805
5500	22.575	0.38378
6000	23.5	0.3995
7500	27.57	0.46869

5. Theoretical Calculations to determine radiator sizing and performance at different load:

5.1. Formulae:

$$L_r = \frac{n_t}{n_C} \times 10 \times 10^{-3}$$

$$W_r = W_t \times n_t + 0.004 \times (n_t + 1)$$

$$Ar = Hr \times Lr$$

$$Asr = 2 \times Hr \times Lr + 2 \times H_r \times W_r + 2 \times L_r \times W_r$$

5.2. Water-side calculations:

$$H_t = H_r$$

$$A_{cst} = (W_t - L_t) \times L_t + \pi \times \frac{{L_t}^2}{4}$$

$$P_t = 2 \times (W_t - L_t) + \pi \times \frac{L_t}{2} \times 2$$

$$A_{st} = P_t \times H_t \times n_t$$

$$D_{hh} = 4 \times \frac{A_{cst}}{P_t}$$

$$Q_h = \frac{m_h}{\rho_h}$$

$$V_h = \frac{Q_h}{n_t \times A_{cst}}$$

$$Re_h = \frac{V_h \times D_{hh} \times \rho_h}{\mu_h}$$

$$Nu_h = \frac{\frac{\alpha}{8} \times (Re_h - 1000) \times Pr_h}{1 + (12.7 \times (\frac{\alpha}{8})^{0.5} \times ((Pr_h^{\frac{2}{3}}) - 1))}$$

$$h_h = \frac{Nu_h \times k_h}{D_{hh}}$$

$$At = \frac{nt \times Lt \times Ht}{n_C}$$

5.3. Air-side calculations:

$$L_c = L_f + \frac{H_f}{2}$$

$$A_{s1} = 2 \times L_c \times W_f$$

$$A_b = 2 \times L_r \times \frac{W_r}{n_C} + H_f \times W_f \times \rho_f$$

$$A_{base} = \rho_f \times A_f + A_b$$

$$A_{sf} = A_{base} \times n_f$$

$$P_f = 2 \times (W_f + H_f)$$

$$A_{csf} = W_f \times H_f$$

$$Q_c = A_r \times V_c$$

$$V_{ra} = \frac{Q_a}{A_r - A_t}$$

$$Re_a = \frac{V_a \times W_f}{\nu_a}$$

$$Nu_a = 0.664 \times {Re_a}^{0.5} \times {Pr_a}^{\frac{1}{3}}$$

$$h_a = \frac{Nu_a \times W_f}{k_a}$$

$$M_a = Q_a \times S_a$$

$$m_f = \sqrt{\frac{h_a \times P_a}{A_a \times k_{Al}}}$$

$$n_f = \frac{\tanh m_f \times L_C}{m_f \times L_c}$$

$$\eta_o = 1 - \frac{\rho_f \times A_f \times (1 - \eta_f)}{A_{base}}$$

5.4. NTU Method:

$$\frac{1}{U_a} = \frac{1}{\eta_f \times h_a \times A_{sf}} + \frac{1}{h_h \times A_{sf}}$$

$$C_{min} = m_w \times Cp_w$$

$$C_{max} = m_w \times Cp_w$$

$$C = \frac{C_{min}}{C_{max}}$$

$$NTU = \frac{U_A}{C_{min}}$$

$$\varepsilon = 1 - e^{[\frac{1}{C} \times NTU^{0.22} \times e^{[(-NTU^{6.78} \times C) - 1]}]}$$

$$Q_{max} = C_{min}(T_{h1} - T_{c1})$$

$$Q = \varepsilon \times Q_{max}$$

$$T_{h2} = T_{h1} \times \frac{Q}{C_h}$$

$$T_{c2} = T_{c1} + \frac{Q}{C_c}$$

Where,

Height-H

Length-L

Width-W

Thickness-t

Cross-sectional area-Acs

Surface Area-As

Perimeter-P

Hydraulic diameter-Dh

Mass flow rate-m

Discharge-Qd

Velocity-V

Reynolds Number-Re

Factor for Nusselt number-α

Nusselt number-Nu

Convective Heat Transfer Coefficient-h

Frontal area-A

Fin constant-mf

Corrected length-Lc

Fin density-ρf

Base surface area-Ab

Total base surface area-Abase

Conductive heat transfer co-efficient-k

Efficiency-η

Overall Efficiency-ηo

Overall heat transfer co-efficient-U

Number of transfer units-NTU

Capacity ratio-C

Effectiveness-ε

Temperature-T

Density-ρ

Dynamic viscosity-μ

Kinematic viscosity-ν

Prantdl number-Pr

Specific heat at constant pressure-Cp

Number-n

Velocity Inside the radiator-Vr

Dynamic pressure drop-Pd

Subscripts:

t-tube

f-fin

h-hot fluid(water)

c-cold fluid(air)

R-radiator

C-core

Known data:

Water parameters:

ρh=960.84kg/m3

μh=2.9×10-4kg/ms

kh=0.67862W/mK

Prh=1.7438

Cph=4.08×103J/kgK

Air Parameters:

ρc=1.1459kg/m3

kc=0.0267W/mK

Prc=0.71317

νc=1.65×10-5

μc=1.89×10-5

Cpc=1.01×103J/KgK

Tube parameters:

Wt=0.0166m

Lt=0.002m

tt=0.0003

Fin Parameters:

Hf=0.0001m

Lf=0.008m

ρf=189

Variable Parameters:

nt, nC, Hr, mh, Vc, nr

Using the above calculations, heat rejected by radiator is been calculated at different air velocities and engine RPM. The results of these calculations and engine load experimentation values were analyzed and accordingly radiator specifications were finalized.

6. CFD Analysis of the Radiator:

The 3D model of the radiator was created in Solid Works and was exported as a single surface in the form of an IGS file. The initial designs were modeled as one section and features that didn't take part in the internal flow were suppressed.

Meshing and CFD analysis were done using Star CCM+. Star CCM+ is a comprehensive engineering simulation package for solving problems involving flow (of fluids or solids), heat transfer and stress.

6.1. Geometry:

The core of the radiator was represented by a cuboid which was then given a porous media. The core is of the dimension from the theoretical calculations that is 255×210×30mm. The header above the radiator has an inlet pipe and the header below had the outlet pipe which directed the flow of water. Two cuboids were created in front and behind the radiator to represent the air region.

Before the meshing and selecting physical model, the inlet and outlet for the two fluids and Interfaces between different bodies were created. In order to set up different boundaries a new region was created.

6.2. Meshing:

The meshing models chosen for the flow analysis were

Trimmer and Surface Remesher

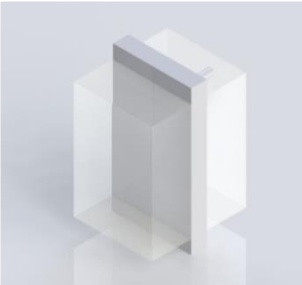

Fig 3: Geometry for CFD simulations.

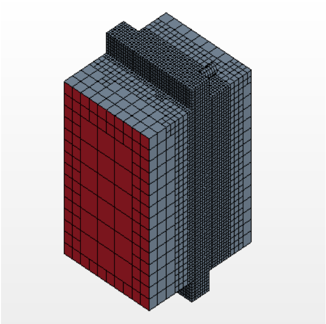

Fig 4: Final mesh

6.3. Mesh:

In order to reduce the CPU time and the load on the computer, the core section of the geometric model was given a finer mesh then the other regions of the geometry.

The base size for the core section was set to 0.64m

The base size for the air region was set to 2.56m

6.4. Physical models:

For the radiator, the physical model was selected deciding the type of flow.

As there are two fluids in the analysis, different models were created.

Three dimensional

Gradient

Steady flow

Constant density

Segregated fluid flow

Turbulent

K-epsilon turbulence

Two Layer All y + Wall Treatment

Realizable K- Epsilon two Layer

Reynolds- Average Navier- Stokes

Segregated Fluid Temperature

Air physical model:

Gas

Water physical model:

Liquid

6.5. Boundary Conditions:

Coolant inlet	Mass flow rate
Coolant outlet	Pressure outlet
Air inlet	Velocity inlet
Air outlet	Pressure outlet

6.6. Inertial and viscous resistance:

Water side resistance:

The pressure drop across the tube was calculated by Darcy-Wiesbach equation

Pressure drop = $f_D \times \frac{\rho}{2} \times \frac{v^2}{D_h}$

Where,

fD= coefficient of friction = 0.0014 (Moody's chart)

ρ = density = 960.84kg/m3

Dh = $\frac{2\times L_t \times W_t}{2\times(L_t+W_t)}$ = 1.45×10-3m

V= $\frac{m}{\rho \times L_t \times W_t}$

Ht = 0.3m

Mass flow rate of water from engine	Mass flow rate of water in a tube	Velocity of water in tube	Pressure drop
0.3995kg/s	0.01kg/s	0.39m/s	505.19Pa
0.24157kg/s	0.00603kg/s	0.2363m/s	885.6Pa

To compute porous media coefficients

Pressure drop = $x \times D \times \mu \times u + 0.5 \times x \times C \times \rho \times u^2$

X= heat exchanger thickness through which pressure drop takes place

D= viscous coefficient

μ= viscosity

C= inertial coefficient

P= Density

By substituting the values,

And solving simultaneously,

D= 86415557

C=110.7

Air side resistance:

This data was provided by the manufacturer

D=456

C=92

6.7. Results:

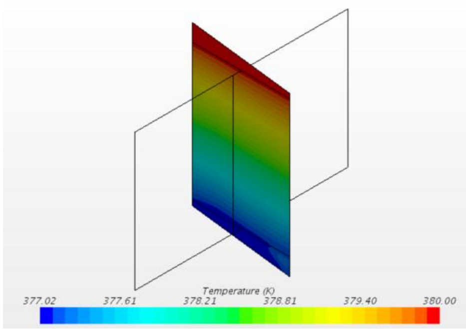

Fig 5: Temperature distribution on the water side

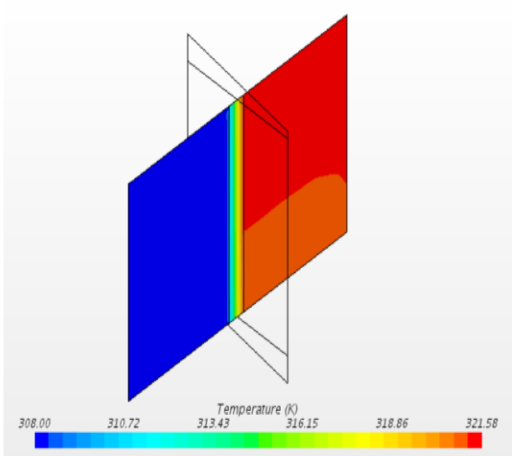

Fig 6: Temperature distribution on the air side

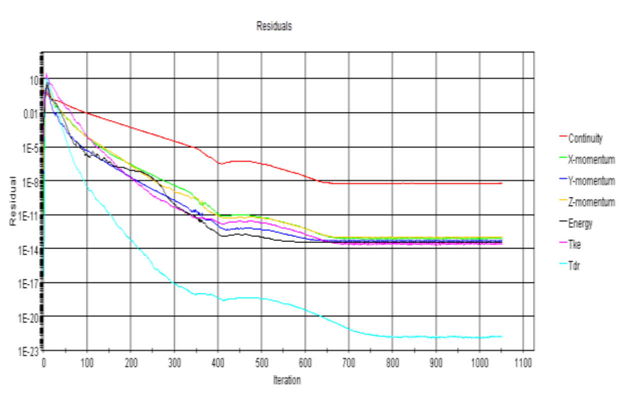

Fig 7: Residual Graph

The above results are used to cross verify the data acquired from theoretical calculations.

7. On-track validation:

The experiment was performed to find the temperature and pressure drop across the radiator at different vehicle and engine speeds. The temperature drop helped us determine the actual heat rejected by the radiator and the dynamic pressure difference was used to calculate the air velocity passing through the radiator.

The radiator was equipped with two thermistors, one at coolant inlet pipe and the other at coolant outlet pipe, both pipe having same diameter. Two differential pressure sensors one for static pressure drop and one for dynamic pressure drop were also installed in the radiator. To measure the dynamic pressure drop, the probes were positioned perpendicular to the airflow at the upstream and the downstream face of the radiator and to measure static pressure drop, the probes were positioned parallel to the airflow.

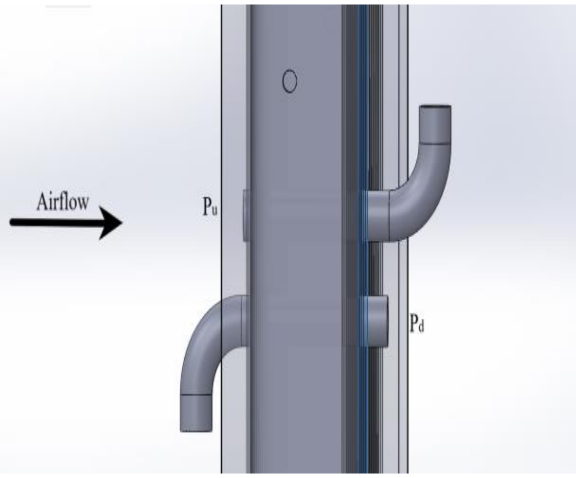

Fig 8: Probes position in radiator to measure dynamic and static pressure drop

The air velocity was calculated by dynamic pressure drop using the following equation,

$$Pd = \frac{1}{2}\rho V^2$$

Pd - Dynamic pressure drop

Pu – Static pressure drop

V - Velocity

ρ - Density

The sensors were connected to a data logger system and logged the data in the dynamic condition. The engine was initially fired at idle and then the car was made to follow a circular path of constant radius while keeping the engine speed constant. The procedure was repeated for different radii while keeping the RPM same so as to achieve various air velocity across the radiator. Then, the engine speed was changed and the procedure was redone. Hence, we obtained pressure and temperature drop values at various engine speeds and air velocities.

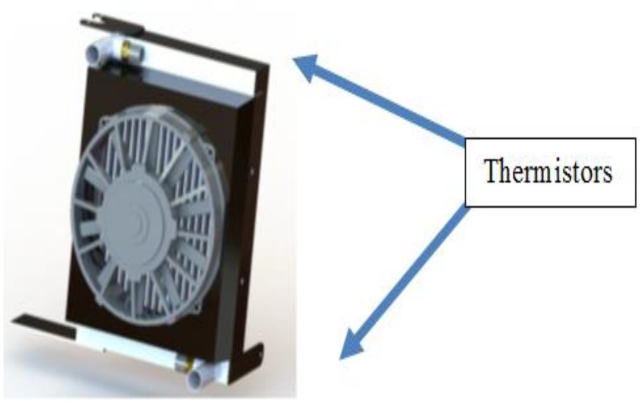

Fig 9: Thermistors' position on radiator

Fig 10: Pressure sensors installed on the radiator.

8. Result and Conclusion:

Mass flow rate of coolant and air velocity through radiator, from the data acquired, were compared to that deduced by different theoretical methods. All the data was calculated, simulated and recorded at specific engine RPM and vehicle velocities.

It was observed that at lower vehicle speeds, the heat rejected by the radiator during practical experiment was almost similar to the calculated value of heat rejected and it was 5% lower when compared to the CFD results. As the vehicle velocity increased, the actual heat rejected by the radiator reduced and the difference between results acquired by different methods increased up to 25%. Therefore, the theoretical and CFD results were able to accurately predict the actual heat transfer at lower vehicle speeds and at higher vehicle speeds, the predicted values were more than the actual heat rejection. This was probably due to the separation of flow in the sidepod.

Table 3: Temperature difference of coolant across radiator and cooling load values at different engine speeds and air velocities

RPM	Mass Flow Rate	Air Velocity	Theoritical		CFD		Practical Expirement	
			Temperature Difference	Load (kW)	Temperature Difference	Load (kW)	Temperature Difference	Load (kW)
7500	0.46869	4	2.6	4959	3	5722	2.53	4826
7500	0.46869	8	3.66	6981	4.1	7820	3.42	6523
7500	0.46869	12	4.4	8392	4.8	9155	4.13	7877
7500	0.46869	16	4.97	9480	5.4	10300	5.09	9709
6500	0.401	4	2.99	4879	3.5	5712	2.31	3770
6500	0.401	8	4.19	6838	4.8	7833	3.57	5826
6500	0.401	12	5	8160	5.7	9302	3.89	6348
6500	0.401	16	5.63	9188	6.1	9955	4.39	7164
5500	0.38378	4	3.11	4857	3.4	5310	2.27	3545
5500	0.38378	8	4.35	6794	4.7	7341	3.21	5013
5500	0.38378	12	5.19	8106	5.8	9059	4.37	6825
5500	0.38378	16	5.83	9105	6.6	10308	4.75	7419

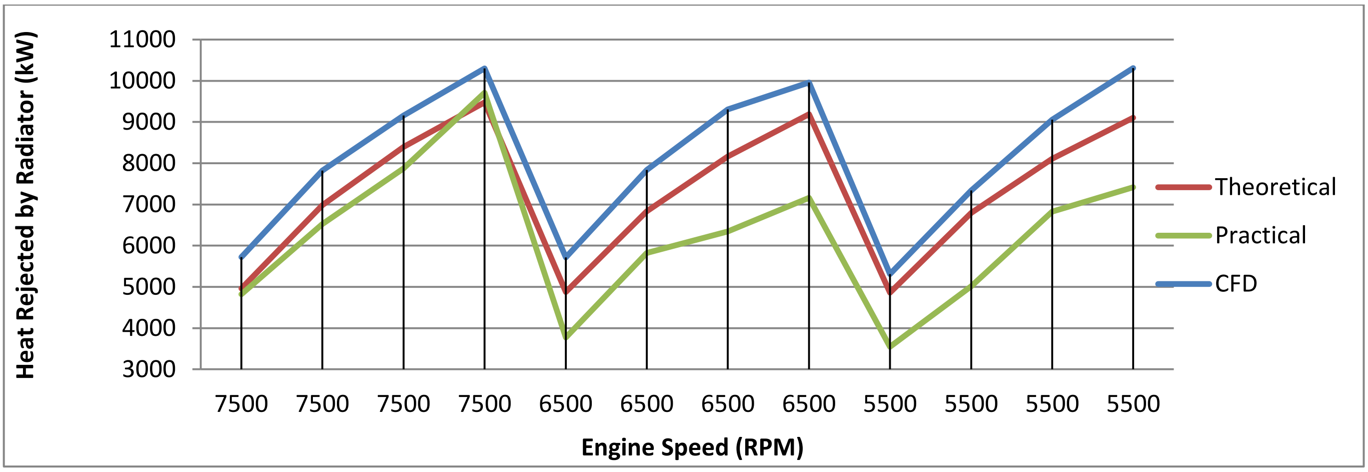

Chart 2: Comparison of heat rejected by the radiator calculated by different methods.

9. Applications:

This research involved the development of a procedure to validate the system in its operational state on the vehicle. The use of this kind of dynamic wind tunnel testing allowed us to test the cooling system without the use of a traditional wind tunnel hence, getting more accurate results according to the relevant conditions.

This methodology can also be used for passenger cars, trucks as well as for cooling system design of electrical components such as motors, high power circuits, etc.

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