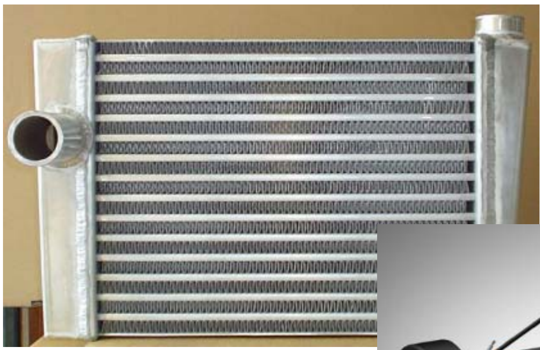
# Graduate Project 1

# University Racing Eindhoven



# Cooling system URE05

# Report

Author: J.P.A. Lubbinge

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Design of a cooling system for the University Racing Eindhoven

racecar 2008-2009


| | |
|---|---|
| Institution: | University of Applied Sciences |
| | Campus 2-6 |
| | Postbus 10090 |
| | 8000 GB, Zwolle |
| | Phone: +31 38 469 99 11 |
| Department: | School of Engineering & Design |
| Education: | Mechanical Engineering |
| Autor: | Mr. J.P.A. Lubbinge |
| | 18, Het Nieveen |
| | 7948 CH, Nijeveen |
| | The Netherlands |

| | |
|---|---|
| Company: | University Racing Eindhoven<br>2, De Wielen<br>W-laag -1.25<br>Eindhoven, 5612 AZ |
| Supervisor: | Mr. L.M.T. Somers, Prof. Msc. |
| First thesis supervisor: | Mr. A.F.C.M. Jansen, Msc. |
| Second thesis supervisor: | Mr. D.J. Struik, Msc. |

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# Preface

This rapport has been written within the scope of a graduation project at Windesheim, University of Applied Sciences, department Engineering and Design. This graduation project has been enabled by University Racing Eindhoven which is the Formula Student team of the Eindhoven University of Technology.
The graduation project has been supervised by Mr. L.M.T. Somers, Msc. (supervisor Eindhoven University of Technology), Mr. Jansen A.F.M.C., Msc. (first thesis supervisor) and Mr. Struik, D.J., Msc. (second thesis supervisor).
Thanks is owed to anyone who devoted time to make this graduation project possible.

Eindhoven, December the 12th, 2008

J.P.A. Lubbinge

# Contents

# Summary


The goal of this graduate project is to design a cooling system for the race car of University Racing Eindhoven for 2008-2009. The radiator will be produced by a sponsor of University Racing Eindhoven which is *ERF* (Eindhoven manufacturer of radiators). In consultation with *ERF* an aluminium type of radiator will be used which is very suitable for this automotive application. The specifications of this radiator are arranged below.

- Material: aluminium
- Thickness: 40mm
- Time of delivery: 2 months

The position and the required area of the radiator should be determined. The radiator should be kept as small as possible because this reduces weight. The additional parts comprising a potential fan, water pump, of the cooling should be determined also.

Several measurements have been carried out to determine the efficiency of the radiator. Hence the required area can be calculated based on an average speed of the car during the toughest prove the cooling system has to stand which is the endurance event. The required area amounts $0{,}108m^2$. The dimensions of the radiator which will be used are arranged below.

- Frontal area: $0{,}108m^2$
- Thickness: $40mm$
- Length: $410mm$
- Width: $300mm$

In this dimensions the dimensions of the connections and some packaging space is included. The radiator will be mounted at an angle of $40°$. This will not harm the efficiency of the radiator. Mounting the radiator at an angle lowers the centre of gravity and reduces the height necessary for building-in.

To ensure sufficient cooling while standing still an driving with low speed a electrical fan will be mounted which is available from a previous year. The capacity of the fan is abundantly enough to ensure the same air flow which is will be generated during the endurance event (at average speed). The fan will be mounted behind the radiator and will pull the air trough the radiator since this is more efficient than pushing the air through the radiator.

The fan will be mounted at a certain distance of the radiator and will be ducted. By ducting the fan it will cover the complete area of the radiator which benefits the efficiency of the fan.

The fan will be ducted by carbon plates which will mounted by aluminium profiles mutually since these materials are very light. The leakages will be finished off by using a suitable glue.

The water pump which will be used is an electrical water pump since the amount of revolutions per minute of the electrical pump does not depend on the amount of revolutions per minute of the engine. Therefore sufficient cooling is ensured during standing still and driving with low speed. The used pump is comparable to the pump of last year.

Where possible the pipes will be made of aluminium since the thermal conductivity of aluminium is very high. Close to the engine there are some narrow spaces. Therefore several narrow corners cannot be avoided. The flexible material which will be used in this case is heat resistant rubber.

# Introduction

University Racing Eindhoven is the Formula Student Team of the Eindhoven University of Technology. Formula Student is a project for technical students which comprises of designing, building, testing and racing a single seater race car. Several times a year teams from all over the world come together to compete. Teams will be judged on technical design, cost analyses, team management and presentation.

University Racing Eindhoven does exist from 2003 and joined the Formula Student Competition in Great Britain, Germany and Italy. The team consist of 60 team members who design, build and test a new race car every year. Next summer the team will join the European races in Great Britain (Silverstone), Germany (Hockenheim) and Italy (Fiorano) with the race car URE09.

The objective of this graduation project is to design a cooling system for the URE09. The determining factors for the cooling system of the modified engine in the race car in relation to the standard engine are very different. Therefore the cooling system should be calculated. The most important item is the size and position of the radiator because this influence the weight and road handling (centre of gravity) of the car.

First the general fundamentals of heat transfer will be discussed in chapter 1. Chapter 2 will discuss the heat transfer processes occurring within an internal combustion engine and also the heat transfer process of the cooling system. For designing a cooling system many measurements have been carried out. The measurements methods will be discussed in chapter 3 and the management results will be discussed in chapter 4. Chapter 5 will discuss the position of the radiator and chapter 6 will discuss the additional parts of the cooling system. The conclusion and recommendations are arranged in chapter 7.

# 1 General introduction engineering heat transfer

## 1.1 Introduction

Heat will be transferred when there is a temperature difference between two media. In accordance with the second law of thermodynamics, heat will be transferred from a medium of higher temperature to a medium of lower temperature. There are three different physical mechanisms by which heat can be transferred:

- Conduction
- Convection
- Radiation

These three mechanisms will be discussed in this chapter. In particular conduction and convection will be discussed because radiation can be neglected (see paragraph 2.4.2 for substantiated calculations).

## 1.2 Conduction

Heat transfer by conduction occurs in solid body's and liquids and gasses in rest, by interaction between adjacent molecules. When a solid body is heated at one side, heat transfer is established. The heat transfer through this solid body is schematically shown in figure 1.1 below.

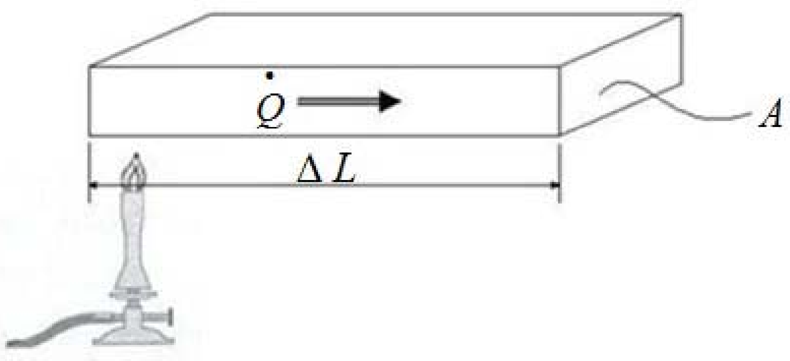



*Figure 1.1 Schematic rendition heat transfer*

In which: $\dot{Q}$ = heat transfer in $W$

$\Delta L$ = length with respect to the hottest spot in $m$

$A$ = area in $m^2$

Plotting out the temperature within the material against the length results in the graph as shown in figure 1.2 below.

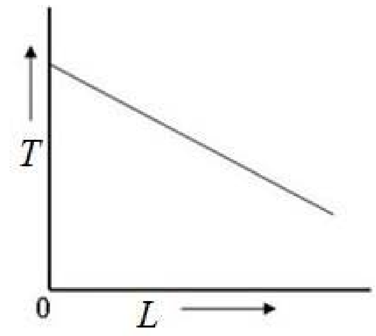



*Figure 1.2 Graph with temperature plotted out against length*

The equation of heat conduction is given by the law of Fourier:

$$\dot{Q} = \lambda \cdot A \cdot \frac{dT}{dx} \qquad [1.1]$$

With: $\lambda$ = coefficient of thermal conductivity in $\frac{W}{m \cdot K}$

When the solid body is considered as a wall with thickness $s$, the heat transfer process will take place as shown in figure 1.3 below. To avoid inconsequence, the wall is considered as a dividing wall by which the left side is defined as the inner side and the right side is defined as the outside.

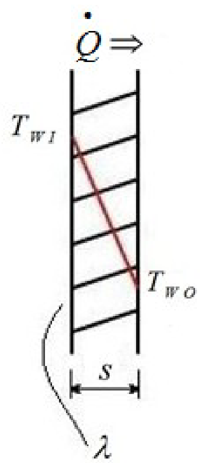


*Figure 1.3 Scematic rendition conduction through a wall*

With: $T_{wi} =$ Temperature wall inside in $K$

$T_{wo} =$ Temperature wall outside in $K$

Hence equation 1.1 can be written as:

$$\dot{Q} = \frac{\lambda}{s} \cdot A \cdot (T_{wi} - T_{wo}) \qquad [1.2]$$

## 1.3 Convection

Convection is heat transfer on a wall caused by a moving liquid or gas. A schematic rendition of the temperature process is given in figure 1.4 below.

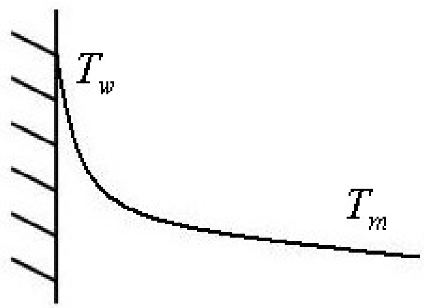



*Figure 1.4 Schematic rendition temperature process on al wall*

With: $T_w =$ Temperature wall

$T_m =$ Temperature medium (gas or liquid)

The equation for the heat transfer is:

$$\dot{Q} = \alpha \cdot A \cdot (T_w - T_m) \qquad [1.3]$$

With: $\alpha =$ coefficient of heat transfer in $\frac{W}{m^2 \cdot K}$

## 1.4 Heat passage

When heat conduction and heat convection appear in one combined system (heat convection on the inner side of the wall, heat conduction through the wall and heat convection on the outside of the wall), the situation appears which is schematically shown if figure 1.5 below.

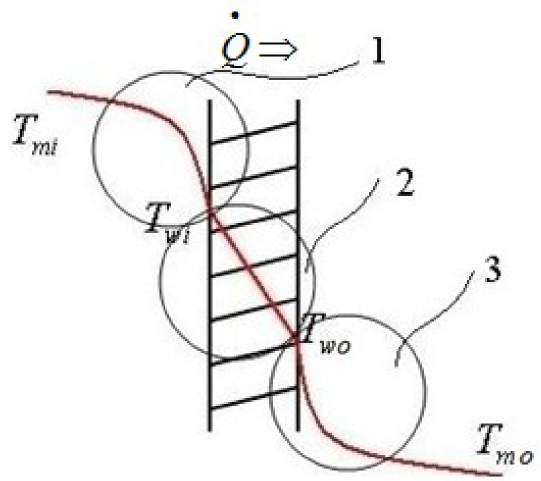



*Figure 1.5 Schematic rendition combined heat transfer system*

The equation for the complete system as shown in figure 1.5 follows by summation of the equations for the dividend systems 1, 2 and 3 (respectively equation 1.3, 1.2 and 1.3). Summation results in:

| | | |
|---|---|---|
| Dividend system 1 | equation 1.3 | $\dot{Q} = \alpha \cdot A \cdot (T_{mi} - T_{wi})$ |
| Dividend system 2 | equation 1.2 | $\dot{Q} = \frac{\lambda}{s} \cdot A \cdot (T_{wi} - T_{wo})$ |
| Dividend system 3 | equation 1.3 | $\dot{Q} = \alpha \cdot A \cdot (T_{wo} - T_{mo}) \quad +$ |

$$\dot{Q} = \frac{1}{\frac{1}{\alpha_i} + \frac{s}{\lambda} + \frac{1}{\alpha_o}} \cdot A \cdot (T_{mi} - T_{mo}) \qquad [1.4]$$

With: $\alpha_i$ = coefficient of heat transfer medium inside

$\alpha_o$ = coefficient of heat transfer medium outside

The first part of equation 1.4 is known as the $k$-value (coefficient of heat passage).

$$\frac{1}{k} = \frac{1}{\alpha_i} + \frac{D}{\lambda} + \frac{1}{\alpha_o} \qquad [1.5]$$

By implementing equation 1.5 in equation 1.4 the following equation appears:

$$\dot{Q} = k \cdot A \cdot \Delta T \qquad [1.6]$$

## 1.5 Radiation

Heat transfer by radiation occurs by emission and adsorption of electromagnetic waves. Radiation occurs between bodies which are not in contact with each other and does not require a medium in contrast to conduction and convection.

The energy emitted by radiation equals:

$$\dot{Q} = \varepsilon \cdot \sigma \cdot A \cdot T^4 \qquad [1.7]$$

With: $\varepsilon =$ coefficient of emission

$\sigma =$ Stefan-Boltzmann constant $= 5{,}67 \cdot 10^{-8} \frac{W}{m^2 K^4}$

Heat transfer by radiation occurs within the engine from the hot gasses to the walls of the combustion chamber and from the hot surfaces to the ambient air. The magnitude of the heat transfer by radiation compared to the heat transfer by convection can be neglected (the magnitude of the radiation compared to the convection within diesel engines cannot be neglected).

Because of the neglectable magnitude (also see paragraph 2.4.2 for substantiated calculations), radiation will not further be discussed.

# 2 Heat transfer internal combustion engines

## 2.1 Introduction

Heat transfer plays an important role in an internal combustion engine – from now on called IC engine. The maximum temperature of the burned gas can increase to approximately 2500 $K$ at the spark plug. The fuel heating value will not completely be converted into useful power. A massive part of the energy will be converted into (technical useless) heat which disappears in several ways. The energy balance of an IC engine will be discussed in paragraph 2.2. A part of the useless heat has to be dissipated by a cooling system which will be discussed in paragraph 2.3 (the necessariness of a cooling system) and paragraph 2.4 (the thermal process of the cooling system). In the last paragraph the general heat transfer fundamentals of the previous chapter will be applied to the thermal processes taking place in the cooling system.

## 2.2 Energy Balance IC engine

Figure 2.1 below shows a Sankey diagram of the energy flow in an IC engine.

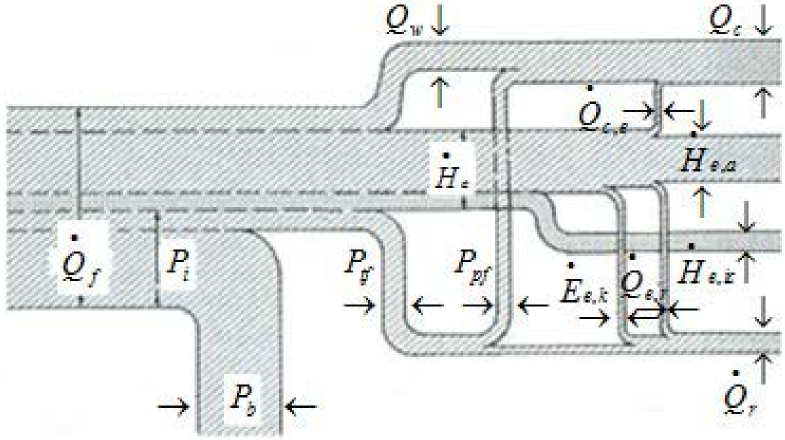



*Figure 2.1 Sankey diagram energy flow IC engine* [1]

[1] Source: *'Internal combustion engine fundamentals'*

To ensure the readability of this rapport figure 2.1 is shown in a small version without the explanation of all used symbols. Appendix A at the back of this report shows a enlargement of figure 2.1 included an explanation of all used symbols.

The branch $\dot{Q}_c$ at the upper right of figure 2.1 above represents the maximum heat load rejected to the coolant, which is the heat to be rejected by forced heat transfer when the engine delivers maximum power. Looking at the energy converted into useful brake power $P_b$ at the lower left, the value of $\dot{Q}_c$ is slightly lower then $P_b$.

Table 2.1 below shows the energy balance for a small spark ignition engine. All the final branches out of figure 2.1 are given as an average percentage of the total energy of the fuel (given by the mass flow of the fuel multiplied by the caloric value of the fuel).

| | $P_b$ | $\dot{Q}_c$ | $\dot{H}_{e,a}$ | $\dot{H}_{e,ic}$ | $\dot{Q}_r$ |
|---|---|---|---|---|---|
| **Spark ignition engine** | 28% | 18% | 41% | 5% | 8% |

*Table 2.1 Energy balance spark ignition engine* [2]

The very complex process occurring within the engine is defined by table 2.1 above. It is very hard to calculate the maximum heat transfer which has to be dissipated by the cooling system because the coefficient of heat transmission medium inside, the coefficient of heat conduction and the coefficient of heat transmission medium outside are very hard to determine. Therefore the maximum heat transfer should be determined on the engine test bed. Due to a lack of time and an engine test bed which was not ready for such measurements, this could not be carried out. Therefore the heat which has to be eliminated by the cooling system has been calculated with the aid of the amount of brake power and the proportions in table 2.1. Figure 2.2 below shows the maximum brake power of the engine of last year (URE04) amounts approximately $59kW$.

[2] Source: *'Internal combustion engine fundamentals'*

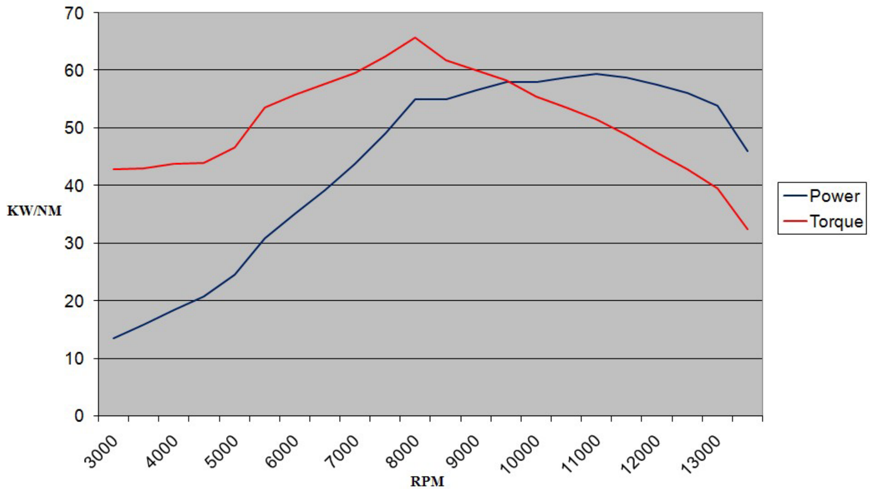



*Figure 2.2 Torque/rpm curve engine URE04* [3]

The engine for the URE09 will be improved and should deliver approximately 10% more power than the engine of the URE04. This results in

$$P_b = 59 \cdot 1,10 = 64,9 \approx 65kW \qquad [2.1]$$

Using table 2.1 gives for the maximum heat load of the engine for the URE09

$$\dot{Q}_c = P_b \cdot Z_{Q,P} = 65 \cdot 10^3 \cdot \left(\frac{18}{28}\right) = 41,721... \cdot 10^3 \approx 41,7kW \qquad [2.2]$$

With: $Z_{Q,P} =$ proportion between the maximum heat load and brake power

## 2.3 Forced heat transfer IC engine

As explained in the introduction of this chapter a massive part of the energy will be converted into technical useless heat. The temperature of the metal on the inside of the combustion chamber space is limited to a much lower value and therefore forced heat transfer (cooling) is necessary. The temperature of the inside of the combustion chamber space should have a much lower value for a number of reasons:

- The temperature of the gas-side surface of the cylinder wall should not reach values above approximately 500 K [4]. This to prevent deterioration of the lubricating oil film, which eventually can result in deformation and heavily engine damage.

- The spark plugs and valves have to be kept cool to prevent overheated spark plug electrodes or exhaust valves. Overheated spark plug electrodes and valves causes problems with the ignition and combustion process.

[3] Source: Tests with engine test bed 2007-2008
[4] Source: *'Internal combustion engine fundamentals'*

Although the temperature of the engine should not be too high, it is just as bad to establish more cooling then needed. By using the Carnot-cycle the highest amount of efficiency can be estimated. The efficiency of a Carnot-cycle $\eta_c$ is given by the proportion between the lowest ($T_{low}$) and the highest ($T_{high}$) temperature in the process:

$$\eta_c = 1 - \frac{T_{low}}{T_{high}} \qquad [2.1]$$

This implies the temperature of the engine should be as high as possible to keep the efficiency as high as possible. More cooling then needed also results in dilution of the lubricating oil and increase of wear.

## 2.4 Thermal process cooling

### 2.4.1 Heat transfer process

The thermal process concerning cooling can be divided into two parts:

- The heat transfer at the engine
- The heat transfer at the radiator

The heat transfer at the engine takes places from the hot gasses to the combustion chamber wall by convection; through the combustion wall by conduction and eventually from the combustion chamber wall to the coolant by convection. This process equals the process of convection-conduction-convection on a wall discussed in paragraph 1.4 with the wall in figure 1.5 considered as the combustion chamber wall.

The heat transfer at the radiator equals the process described above: heat transfer from the coolant to the radiator by convection; heat transfer through the radiator by conduction and eventually heat transfer from the radiator to the circulation air. Also this process equals the process of convection-conduction-convection on a wall discussed in paragraph 1.4 with the wall in figure 1.5 considered as the radiator).

Assumed is 100% of the heat transfer within the engine reaches the radiator. Actually some of the heat will be eliminated by heat transfer on the ducts and radiation. The heat transfer by convection on the ducts and by radiation has a very small amount and will be neglected during future calculations. This neglectable amount will be proved in the next paragraph.

### 2.4.2 Neglectable heat transfer

The heat transfer occurring in the ducts is comparable to the process discussed in paragraph 1.4. (convection on the inside of the duct, conduction trough the duct and convection on the outside of the duct). Figure 2.2 below shows a schematic rendition of the cross section of a duct.

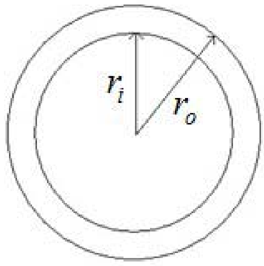



*Figure 2.2 Cross section duct*

With: $r_i$ = radius inside

$r_o$ = radius outside

The heat transfer equation for a duct is comparable with equation 1.4. The equation for a duct has a logarithmic term because of the round shape.

$$\dot{Q} = \frac{2 \cdot \pi \cdot l \cdot (T_{mi} - T_{mo})}{\frac{1}{\alpha_i \cdot r_i} + \frac{\ln \frac{r_o}{r_i}}{\lambda} + \frac{1}{\alpha_o \cdot r_o}} \qquad [2.2]$$

Considering a aluminum duct with a diameter of $25mm$ and a wall thickness of $2mm$, all the terms can be reasonably estimated. The amount of the coefficient of heat transfer is in the order of magnitude of dozens concerning gas and between 1000 and 2000 concerning water. Fill in all the terms results in

$$\dot{Q} = \frac{2 \cdot \pi \cdot 1 \cdot (85 - 20)}{\frac{1}{1500 \cdot 0,021} + \frac{\ln\left(\frac{25}{21}\right)}{237} + \frac{1}{50 \cdot 0,025}} = 490,589... \approx 500W \qquad [2.3]$$ [5]

For each meter duct $(l = 1)$ the heat transfer amounts approximately $500W$. The use of rubber for connections and tight corners cannot be avoided. The coefficient of thermal conductivity for rubber amounts $0,16 \frac{W}{m \cdot K}$. This results in a heat transfer of approximately $200W$. The temperature for the inside medium used in equation 2.3 amounts $85°C$ which is of course only valid for half of the system (the coolant after the radiator has been significance cooled down). The total heat transfer occurring in the ducts will not exceed $1500W$ assuming 2 meter ducting and some connections.

[5] The unit of temperature in this equation is $°C$ instead of $K$ which is the SI unit. This is allowed because the temperature in this equation concerns a temperature difference (one degree Celsius temperature difference equals on degree Kelvin temperature difference).

The heat transfer occurring by radiation follows from equation 1.7. The coefficient of emission for aluminum amounts $0,09$. Handling the same temperature as in equation 2.3 above the equation for the amount of heat transfer by radiation results in

$$\dot{Q} = 0,09 \cdot 5,67 \cdot 10^{-8} \cdot 1 \cdot (85 + 273,15)^4 = 83,963... \approx 100W \qquad [2.4]$$

For each square meter aluminum $(A = 1)$ the heat transfer by radiation amounts approximately $100W$. The total heat transfer occurring by radiation will not exceed $200W$ assuming 2 square meter aluminum).

Summation of the two types of heat transfer (heat transfer in the ducts and heat transfer by radiation) will not exceed $1700W$. At the most this equals a few percent of the maximum heat load of the engine which has to be dissipated by the cooling system (paragraph 2.2). Therefore these two types can be neglected during further calculations.

To clarify: The heat transfer occurring by the two types of heat transfer discussed above is very desirable. The heat dissipated certainly does not have to be dissipated by the radiator. In case of big amounts this benefits the size of the radiator which reduces weight. Neglecting these amount could be considered as 'safe calculating'. The calculations made above are coarse approximations but delivers enough grip to consider the amounts as neglectable.

# 3 Testing

## 3.1 Introduction

As already explained the heat transfer processes occurring within an IC engine is very complicated. This also applies to the heat transfer process occurring within the radiator. It is very hard to determine the coefficients of convection and conduction. In paragraph 3.2 the background of the used methods for calculation will be explained. In paragraph 3.3 this will merge into the used methods for calculation.

## 3.2 Radiator efficiency

Also the industry does not calculate with separated calculations using the coefficients of convection and conduction. The *'Eindhovense Radiateuren Fabriek'* (radiator manufacturer Eindhoven, ERF) uses Graphs which obviate the separated coefficients. These graphs for a particular kind of radiator have been established by experiments. Figure 3.1 below shows a graph for a particular (not existing) radiator.

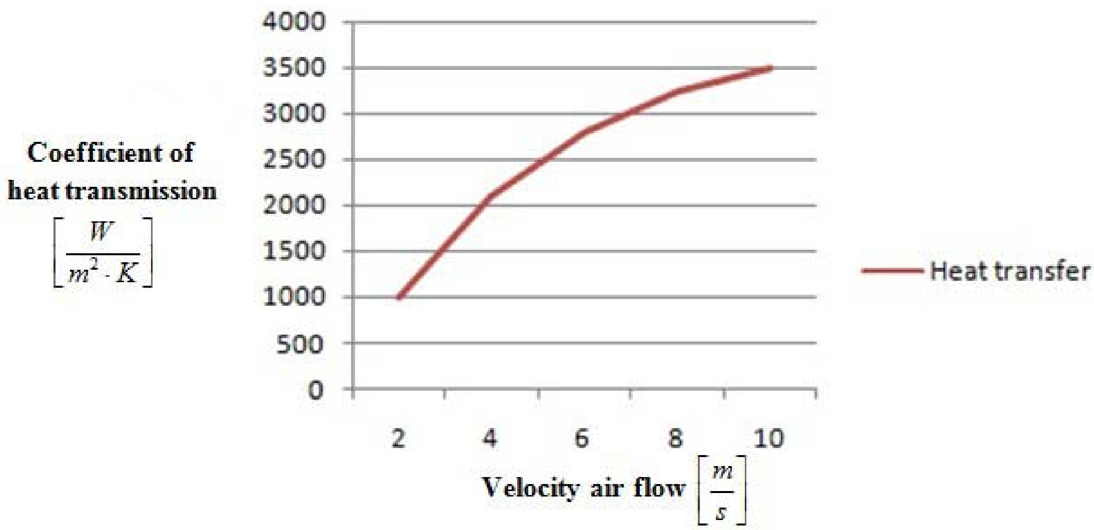



*Figure 3.1 Graph fictive radiator*

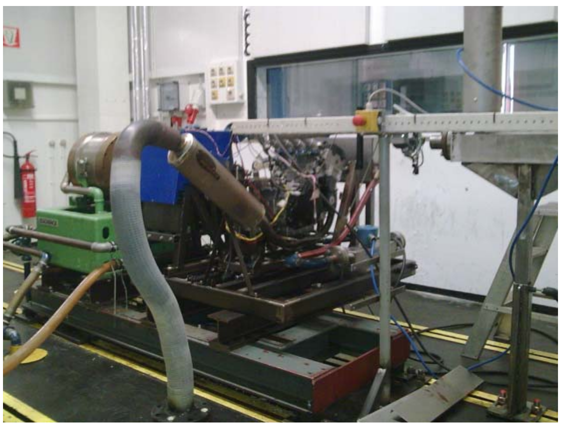
The graph shows the efficiency of the radiator by different airflows. The unit of the heat transfer coefficient plotted on the vertical axis is $\frac{W}{m^2 \cdot K}$. The wattage in the numerator represents the heat transfer dissipated by the radiator. This amount equals the heat transfer $\dot{Q}_c$ from the engine to the coolant which has an average value of $\dot{Q}_c = 41,7kW$ (paragraph 2.2). The area in the denominator represents the frontal area $A_r$ of the concerned radiator. The temperature difference in the denominator represents the difference $\Delta T_{r,a}$ between the average temperature within the radiator and the temperature of the circulating air. Appendix B gives an arrangement of the term and conditions concerning the cooling system. One of the terms and conditions prescribes the manufacturer. The radiator will be produced by *'Eindhovense Radiateuren Fabriek'*. In consultation with ERF an aluminum radiator will be produced which is very suitable for this automotive application.

## 3.3 Calculation methods

### 3.3.1 Introduction

The same type aluminum radiator which will be used this year has been used last year. This radiator has been mounted on the engine test bed to determine the efficiency of the radiator. This engine test bed is situated at the Eindhoven University of Technology and is in use by University Racing Eindhoven for testing and adjusting the engine. Figure 3.2 below shows the engine test bed.

Measurements has been carried out for different air flows, different angles of the radiator, different setups concerning the fan for the radiator and different types of surface finishing. The engine test bed is interfaced to LabView (graphical program for measurements and automation) which gathers all the data (figure 3.3).



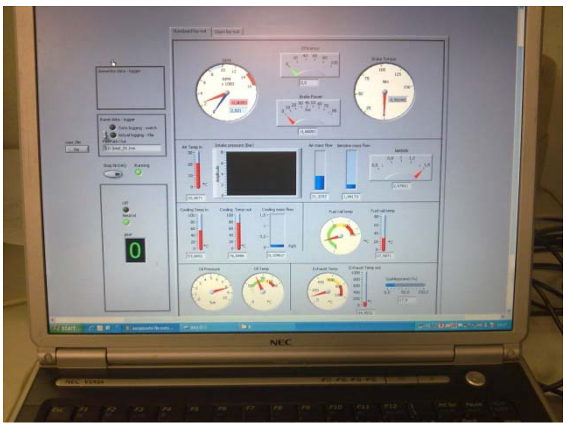


*Figure 3.2 Engine test bed* *Figure 3.3 LabView running*

## 3.3.2 Air flow

To simulate the air flow through the radiator a fan is mounted. This fan can be adjust by a frequency controller. The air flow can be adjudged up till $10\frac{m}{s}$. Measurements have been carried out with $4\frac{m}{s}$, $6\frac{m}{s}$, $8\frac{m}{s}$ and $10\frac{m}{s}$. The proportion between the frequency of the fan and the velocity of the airspeed has been determined with an air speed indicator at a distance of approximately $500mm$ before the radiator. By measuring in front of the radiator the values are comparable to the velocity of the air flow while driving (the velocity will be influenced by turbulence by measuring directly in front of the radiator). The data obtained from LabView is digest with Matlab (a mathematical program) to set up graphs for the several air flows.

## 3.3.3 Angle radiator

The angle of the radiator on the engine test bed can be adjudged. First of all the most efficient angle should be determined. In case the angle of the radiator makes no difference for its efficiency, mounting the radiator at an angle is desirable. When the radiator is mounted at an angle this lowers the center of gravity. This is schematically shown in figure 3.4 below in which $CG$ represents the center of gravity.

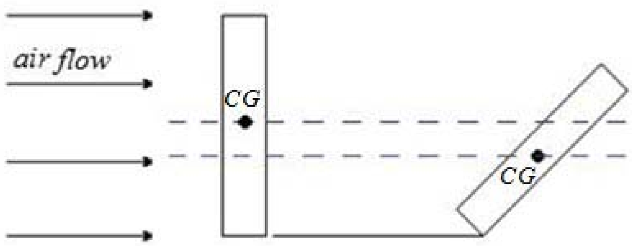



*Figure 3.4 Center of gravity (CG) radiator*

Besides an angle lowers the center of gravity is also lowers the height of the radiator. This is desirable because this reduces the required height of the space for building-in.

### 3.3.4 Setup radiator

While driving the airflow through the radiator takes care of efficient cooling. The higher the velocity of the airflow, the higher the coefficient of heat convection on the outside of the radiator (paragraph 1.3). When the race car is not driving (for example during driver changes) this results in very poor cooling (no airflow).

Therefore a fan is desirable. For mounting an electrical fan there are two options: pushing or pulling the air through the radiator. These two options are schematically shown in figure 3.5 below in which the left side shows a pushing fan and the right side shows a pulling fan.

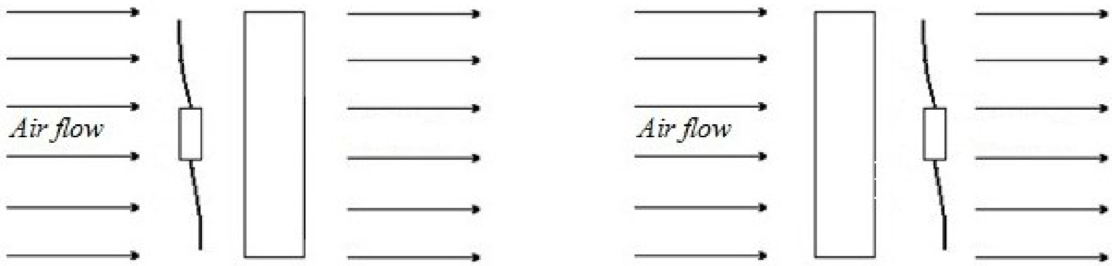



*Figure 3.5 Pushing (right) and pulling (left) fan*

Pulling the air through the radiator is more efficient because of the centrifugal force. Because the fan turns around at high speed a certain amount of air will be thrown out the fan instead of being pushed through the fan. Therefore a lot of air will not flow through the radiator which is very harmful towards the efficiency.

A fan mounted directly  at the back of the radiator will have a harmful effect on the airflow through the radiator. Figure 3.6 below gives an schematic rendition of this effect (left). The right side shows the fan at a certain distance behind the radiator.

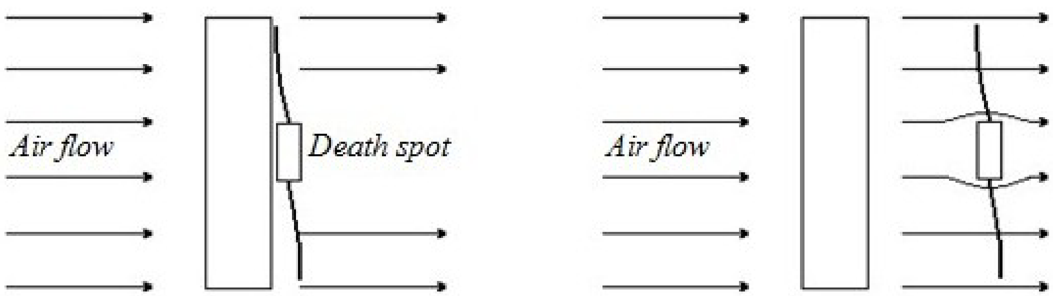



*Figure 3.6 Fan behind the radiator*

Besides the harmful effect the fan will not cover the whole area of the radiator which makes it less efficient. The fan at a certain distance towards the radiator can be ducted. When ducted the fan will cover the whole area of the radiator. The difference concerning efficiency has been tested by using two setups. Figure 3.7 below shows the fan directly mounted at the back of the radiator. Figure 3.8 below shows the ducted fan at a certain distance of the radiator.

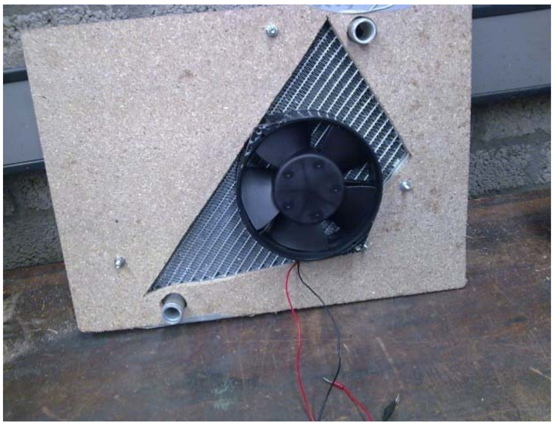


*Figure 3.7 Fan mounted at radiator*

*Figure 3.8 Fan behind radiator (ducted)*

The duct has a flap on top of the duct. This flap should be closed while the fan is running (so the fan will cover the whole area) and this flap should be open while the fan is off. The flap will open by the air flowing through the duct. Therefore the air will not be disturbed by the fan.

### 3.3.5 Coating

One of the sponsors *(Ceradure)* of University Racing Eindhoven offers a special coating for radiators which should make the radiator more efficient. This coating will be applied in combination with a modification concerning the material structure. This results in a finer crystal structure. From last year two (identical) radiators are available. Therefore one of the radiators has been coated by *'Ceradure'* to determine whether the coating benefits the efficiency of the radiator or not.

# 4 Results of measurements

## 4.1 Introduction

The results of the measurements described in chapter 3 will be discussed in this chapter. All the measurements results will be arranged. This chapter will clarify the efficiency of the radiator.

All the measurements data has been assimilated with Matlab and all the measurements are converted into graphs to determine the efficiency. In all these graphs the heat transmission coefficient has been plotted out on the y-axis against the time (which the measurement toke) on the x-axis (instead of velocity of the airflow in figure 3.1). By combining all the results of the measurements a graph like figure 3.1 can be plotted to determine the efficiency of the radiator for several velocities of the air flow. Hence the required frontal area can be calculated. To ensure the readability of this rapport only the average values for all the measurements are given in this chapter. In each paragraph the average values concerning the considered situation are arranged in that paragraph. The average values which are given are the exact values Matlab calculated. All the values are arranged in a table which is shown in Appendix D. The inaccuracy of the measurements is pretty high. Therefore the values in the table are rounded off to hundreds. The average values are calculated by dividing the area under the graph by the time. This results in the average height which equals the average amount of the coefficient for heat convection. This calculation is given by the following equation.

$$\frac{\int_0^{t_m} \frac{\dot{Q}_c}{A_r \cdot \Delta T_{r,a}}}{t_m} \qquad [4.1]$$

All the measurements graphs are included as appendixes at the back of the rapport.

Three major different situations has been tested:

- Radiator without coating without ducted fan
- Radiator without coating with ducted fan

- Radiator with coating without ducted fan

These tree major situations will be discussed in this chapter. The cooling system will be designed based on the toughest proof the car has to stand which is the endurance race. This race will demand the most of the car for the longest time. Appendix C explains all the proves the car has to stand. From previous race data the average speed has been determined on $36\frac{km}{h}$ which corresponds to $10\frac{m}{s}$ . While testing different angles the velocity of the air flow has been set at $8\frac{m}{s}$ because $10\frac{m}{s}$ verged the most of the fan which could not last the time this measurements toke.

## 4.2 Radiator without coating without ducted fan

### 4.2.1 Radiator with fan mounted at different angles

The fan has been directly mounted at the back of the radiator and the radiator has been mounted at four different angles $\varphi$ which are $\varphi = 0°$, $\varphi = 20°$, $\varphi = 40°$ and $\varphi = 50°$. This angles are relative to the vertical, so a $20°$ angle means $20°$ reverse rollover relative to the vertical. This is schematically shown in figure 4.1 below.

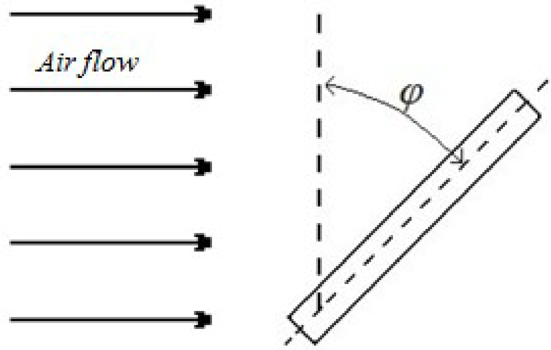



*Figure 4.1 Schematic rendition angle radiator*

The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix E.

- For $0°$ → $\dot{K}_r = 6371{,}2\frac{W}{m^2 \cdot K}$

- For $20°$ → $\dot{K}_r = 6597{,}1 \frac{W}{m^2 \cdot K}$
- For $40°$ → $\dot{K}_r = 6357{,}9 \frac{W}{m^2 \cdot K}$
- For $50°$ → $\dot{K}_r = 6204{,}3 \frac{W}{m^2 \cdot K}$

The radiator efficiency at different angles does not fluctuate very much. The efficiency is slightly higher for $20°$. For $50°$ it is slightly lower which is plausible because the border of $45°$ has been passed. Therefore most of the air will flow along the radiator instead of through the radiator.

### 4.2.2 Radiator with fan mounted at different velocities of air flow (0° angle)

The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix F.

- For $4 \frac{m}{s}$ → $\dot{K}_r = 6177{,}1 \frac{W}{m^2 \cdot K}$
- For $6 \frac{m}{s}$ → $\dot{K}_r = 5754{,}6 \frac{W}{m^2 \cdot K}$
- For $8 \frac{m}{s}$ → $\dot{K}_r = 5740{,}6 \frac{W}{m^2 \cdot K}$
- For $10 \frac{m}{s}$ → $\dot{K}_r = 5724{,}9 \frac{W}{m^2 \cdot K}$

The radiator efficiency at different velocities of the air flow does not fluctuate very much. When the fan is mounted directly at the back of the radiator, more air flow cannot increase the efficiency. The value for $4 \frac{m}{s}$ is slightly higher which indicates low velocity of the air flow experiences less harm from the fan. However the efficiency does not decrease gradually it also can indicate some measuring inaccuracy.

### 4.2.3 Radiator without fan mounted at different angles

The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix G.

- For $0°$ → $\dot{K}_r = 6472,6 \frac{W}{m^2 \cdot K}$
- For $20°$ → $\dot{K}_r = 6700,8 \frac{W}{m^2 \cdot K}$
- For $40°$ → $\dot{K}_r = 6912,7 \frac{W}{m^2 \cdot K}$
- For $50°$ → $\dot{K}_r = 6846,4 \frac{W}{m^2 \cdot K}$

When the radiator is more angled the coefficient of heat transmission increases significantly. When the border of $45°$ has been passed the efficiency slightly decreases which is plausible once more. When the fan is not mounted directly at the back of the radiator it is desirable to mount the radiator at an angle.

### 4.2.4 Radiator without fan mounted at different velocities of air flow (0° angle)

The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix H.

- For $4\frac{m}{s}$ → $\dot{K}_r = 6450{,}8\frac{W}{m^2 \cdot K}$
- For $6\frac{m}{s}$ → $\dot{K}_r = 6306{,}7\frac{W}{m^2 \cdot K}$
- For $8\frac{m}{s}$ → $\dot{K}_r = 6472{,}6\frac{W}{m^2 \cdot K}$
- For $10\frac{m}{s}$ → $\dot{K}_r = 6364{,}4\frac{W}{m^2 \cdot K}$

The radiator efficiency at different velocities of the air flow without the fan mounted does not fluctuate very much but is significantly higher than the efficiency of the radiator with the fan mounted directly at the back of the radiator (paragraph 4.2.2)

### 4.2.5 Radiator with fan mounted and running (0° angle)

The value for the coefficient of heat transmission $\dot{K}_r$ is given below. The graph is included as appendix I.

- Fan running → $\dot{K}_r = 4743{,}9\frac{W}{m^2 \cdot K}$

By using an air speed indicator the developed air velocity of the fan has been determined. The velocity amounts approximately $8\frac{m}{s}$. The efficiency of the radiator with the fan running is significantly lower than the efficiency of the radiator with the fan mounted (not running) at the same velocity of the air (paragraph 4.2.2). This clarifies the efficiency of the fan is very poor which is caused by the different sizes of the fan and the radiator. The fan does not cover the complete area of the radiator.

## 4.3 Radiator without coating with ducted fan

## 4.3.1 Radiator at different velocities of the air flow (0° angle)

The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix J.

- For $4\frac{m}{s}$ → $\dot{K}_r = 6046{,}0\frac{W}{m^2 \cdot K}$
- For $6\frac{m}{s}$ → $\dot{K}_r = 5874{,}3\frac{W}{m^2 \cdot K}$
- For $8\frac{m}{s}$ → $\dot{K}_r = 6238{,}3\frac{W}{m^2 \cdot K}$
- For $10\frac{m}{s}$ → $\dot{K}_r = 6391{,}4\frac{W}{m^2 \cdot K}$

The efficiency of this set-up is significantly higher than the efficiency of the radiator with the fan directly mounted at the back of the radiator. The values even are comparable to the efficiency of the radiator with no fan mounted at all. This proves ducting the fan is very desirable. The flap on top of the duct which should be closed while the fan is running and should be open while the fan is off performs as expected. Figure 4.2 below shows the opened flap at $8\frac{m}{s}$.

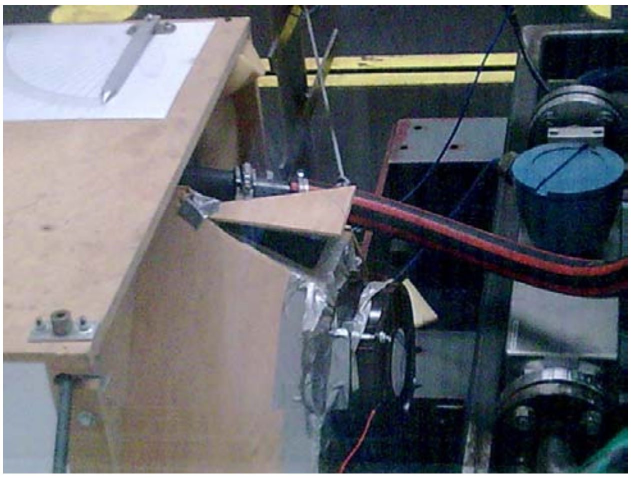


*Figure 4.2 Flap opened*

## 4.3.2 Radiator with the fan running (0° angle)

The value for the coefficient of heat transmission $\dot{K}_r$ is given below. The graph is included as appendix K.

- Fan running → $\dot{K}_r = 5508,7 \dfrac{W}{m^2 \cdot K}$

This efficiency of the fan ducted and at a certain distance behind the radiator is much higher than the efficiency of the fan directly mounted at the back of the radiator (paragraph 4.2.5). Ducting the fan makes it $\dfrac{5508,7 - 4743,9}{4743,9} \cdot 100 = 16,122... \approx 16\%$ more efficient. Therefore ducting the fan is very desirable for the efficiency of the fan. Paragraph 4.3.1 already clarified ducting the fan also benefits the efficiency concerning the air flow.

# 4.4 Radiator with coating without fan

## 4.4.1 Radiator at different velocities of the air flow (0° angle)

The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix L.

- For $4 \dfrac{m}{s}$ → $\dot{K}_r = 6253,6 \dfrac{W}{m^2 \cdot K}$
- For $6 \dfrac{m}{s}$ → $\dot{K}_r = 6409,3 \dfrac{W}{m^2 \cdot K}$
- For $8 \dfrac{m}{s}$ → $\dot{K}_r = 7103,3 \dfrac{W}{m^2 \cdot K}$
- For $10 \dfrac{m}{s}$ → $\dot{K}_r = 6991,2 \dfrac{W}{m^2 \cdot K}$

The efficiency of the coated radiator at high velocity is much higher than the efficiency of the radiator without the coating under the same conditions (paragraph 4.2.4). Concerning the velocities $8 \dfrac{m}{s}$ en $10 \dfrac{m}{s}$ the efficiency of the coated radiator increases with approximately $10\%$. The difference at lower velocities can be neglected.

## 4.4.2 Radiator with lower mass flow coolant (0° angle)

During the last test which has been carried out the mass flow of the coolant has been decreased. The values for the coefficient of heat transmission $\dot{K}_r$ are arranged below. The graph is included as appendix M.

- For $4\frac{m}{s}$ → $\dot{K}_r = 4241,8\frac{W}{m^2 \cdot K}$
- For $6\frac{m}{s}$ → $\dot{K}_r = 4994,3\frac{W}{m^2 \cdot K}$
- For $8\frac{m}{s}$ → $\dot{K}_r = 5110,8\frac{W}{m^2 \cdot K}$

These values clarify the mass flow should not be that low because the efficiency is lower than the efficiency of the radiator with high mass flow. Besides the efficiency is lower also the cooling is less steady. This can be conducted from the graph which is oscillating more than the graphs with high mass flow of the coolant.

## 4.5 measurements processing

### 4.5.1 Quantities

The most efficient setup concerning the variables angle of the radiator the and the setup of the fan has been estimated. The measurements values concerning the most efficient setup has been used to arrange a graph comparable to the graph shown in figure 3.1. The coefficient of heat transmission plotted on the vertical axis has the equation

$$\dot{K}_r = \frac{\dot{Q}_c}{A_r \cdot \Delta T_{r,a}} \qquad [4.2]$$

In this equation $\dot{K}_r$ can be determined out of the graph by assuming the average velocity of the air flow. As mentioned in paragraph 4.1 from previous race data the average speed has been determined on $36\frac{km}{h}$ which is corresponding to $10\frac{m}{s}$. Also the maximum heat load $\dot{Q}_c$ is known (paragraph 2.2) and amounts $\dot{Q}_c = 41,7kW$. The temperature difference $\Delta T_{r,a}$ can be assumed. For assuming this temperature difference the worst case scenario should be handled which is Italy (highest temperature which results in the lowest temperature difference). For an ambient air temperature of $30°C$ and an average water temperature within the radiator of $85°C$ the temperature difference amounts

$$\Delta T_{r,a} = T_r - T_a = 90 - 35 = 55°C = 55K \quad [6] \qquad [4.3]$$

The only unknown quantity left is the area of the radiator $A_r$ which can be determined.

[6] The unit of temperature in this equation is $°C$ instead of $K$ which is the SI unit. This is allowed because the temperature in this equation concerns a temperature difference (one degree Celsius temperature difference equals on degree Kelvin temperature difference).

## 4.5.2 Graph efficiency radiator

Because of the increase of the efficiency of the radiator and the fan at higher velocities ($8\frac{m}{s}$ and $10\frac{m}{s}$ the fan will be mounted at a certain distance from the radiator and will be ducted. Also the coating for these velocities is very desirable and will be used. The increase of efficiency can be calculated by proportions between the measurements results of paragraph 4.2.4 and paragraph 4.4.1 because these two set-ups are identical (except the coating). The calculated increase are arranged below.

- For $4\frac{m}{s}$ → increase $= \frac{6253,6-6450,8}{6450,8} \cdot 100 = -3,057... \approx -3,1\%$
- For $6\frac{m}{s}$ → increase $= \frac{6409,3-6306,7}{6450,8} \cdot 100 = 1,591... \approx 1,6\%$
- For $8\frac{m}{s}$ → increase $= \frac{7103,3-6472,6}{6472,6} \cdot 100 = 9,744... \approx 9,7\%$
- For $10\frac{m}{s}$ → increase $= \frac{6991,2-6364,4}{6364,4} \cdot 100 = 9,849... \approx 9,8\%$

These increase amounts can be used to determine the efficiency of the radiator which will be ducted. Therefore the calculations which are arranged below are based on the efficiency values of paragraph 4.3.1.

- For $4\frac{m}{s}$ → $\dot{K}_r = 6046,0 \cdot \frac{1}{1,031} = 5864,210... \approx 5864 \frac{W}{m^2 \cdot K}$
- For $6\frac{m}{s}$ → $\dot{K}_r = 5874,3 \cdot 1,016 = 5968,289... \approx 5968 \frac{W}{m^2 \cdot K}$
- For $8\frac{m}{s}$ → $\dot{K}_r = 6238,3 \cdot 1,097 = 6843,415... \approx 6843 \frac{W}{m^2 \cdot K}$
- For $10\frac{m}{s}$ → $\dot{K}_r = 6391,4 \cdot 1,098 = 7017,757... \approx 7018 \frac{W}{m^2 \cdot K}$

With these values the graph can be plotted. A remark concerning the angle of the radiator has to be made. The measurements of paragraph 4.2.1 proved the values for the efficiency of the radiator at different angles with the fan mounted do not fluctuate very much. Paragraph 4.2.3 proves the efficiency of the radiator without a fan mounted increases significantly when the radiator is more angled. Because of a lack of time and a lack of abilities concerning the engine test bed, the setup of the radiator which will be used (ducted fan) could not be tested at different angles. Therefore is very hard to make valid calculations concerning the efficiency of the angled radiator. The efficiency of the set-up which will be used probably lies between these two measurements. This implies mounting the radiator at an angle will benefit the efficiency of the radiator but this cannot be proven. Therefore these likely efficiency increases are neglected by setting-up the graph. One conclusion can be made out of the measurements results concerning the angle of the radiator: mounting the radiator at an angle will not have a bad influence on the efficiency as long as the angle does not exceed $40°$. The graph for the radiator is shown in figure 4.3 below.

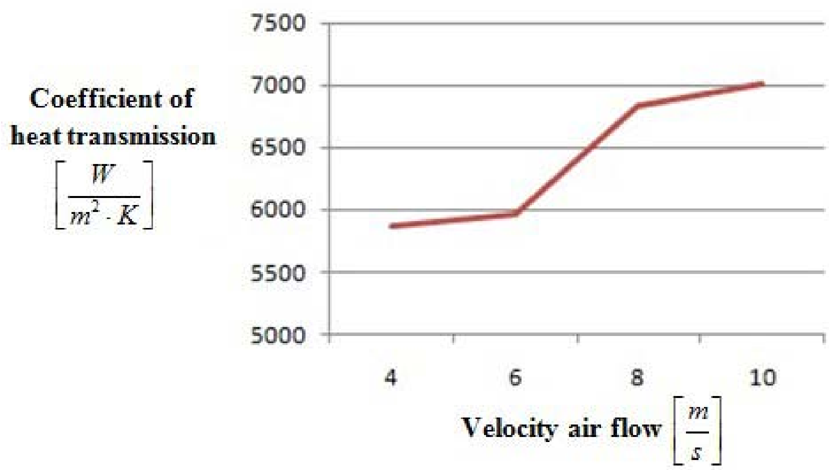



*Figure 4.3 Graph efficiency radiator*

## 4.6 Calculations measurements

Using the graph shown in figure 4.3 above the required area of the radiator can be calculated. For the average speed of $10\frac{m}{s}$ the coefficient of heat transmission amounts approximately $7000\frac{W}{m^2 \cdot \Delta K}$. Equation 4.1 can be rewritten to

$$A_r = \frac{\dot{Q}_c}{\dot{K}_r \cdot \Delta T_{r,a}} \qquad [4.4]$$

Fill in all the quantities results in

$$A_r = \frac{\dot{Q}_c}{\dot{K}_r \cdot \Delta T_{r,a}} = \frac{41,7 \cdot 10^3}{7000 \cdot 55} = 0,10831... \approx 0,108m^2 \qquad [4.5]$$

# 5 Position radiator

## 5.1 Introduction

Several possible positions for placing the radiator will be considered in this chapter. All reasonably possible positions are schematically shown in paragraph 5.2. The advantages and the disadvantages are tabulated in paragraph 5.3 together with the definitive position.

## 5.2 Possible positions

The reasonably possible positions are schematically shown in figure 5.1 below.

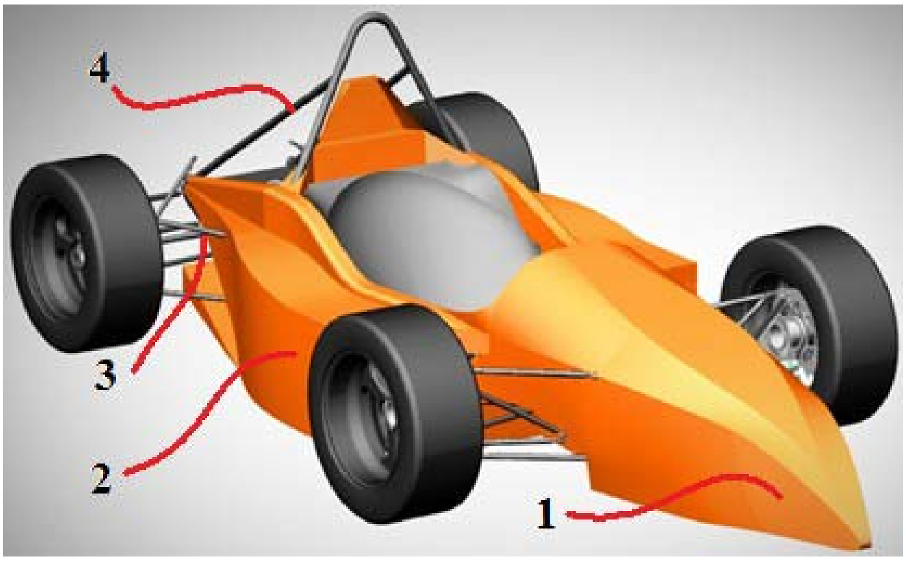



*Figure 5.1 Possible positions radiator*

Description positions:

1. In the front of the car (nose)
2. In the sidepods
3. At the bottom of the frame close to the engine
4. At the top of the frame above the engine

## 5.3 Considerations possible positions

The possible positions are judged on several items are arranged below (Appendix B terms and conditions). The numbers between the brackets represent the importance and are explained below.

- Efficiency concerning the airflow (4)
- Efficiency concerning ambient air (4)
- The height of the position (3)
- The space and possibilities for construction/mounting (3)
- The consequences concerning aerodynamics (2)
- The consequences concerning ducting (1)

A simple summation of the advantages and disadvantages for each position will not result in a proper arrangement of the possible positions because certain items are far more important than others. Therefore all items to be judged have a number on the scale 1-4 (the more important, the higher the number). This are the numbers between the brackets in the summation above.

For a positive judgment on a particular item the position receives the attendant points. For a negative judgment on a particular item the attendant points are subtracted.
The most important item is the efficiency of the radiator concerning the airflow and the ambient air. The efficiency of the cooling system defines for the biggest part the size of the radiator which has to be kept as small as possible to reduce weight. This explains the high amount of points adjudged to the first two items in the summation above.
The height of the position and the space and possibilities for construction/mounting are also very important, although marginally less important than the first two items. The height of the position influences the center of gravity which has to be kept as low as possible. The lower the center of gravity, the better the road holding of the car. Also the space and possibility for construction/mounting should not cause any trouble. In case of repair/servicing the radiator should have the possibility to be easily removed and mounted.
The last two items should not be overlooked although these are not the most important items. For example: a little bit more ducting will not cause as much weight as low efficiency does because the amounts and weight of ducting are not that high.
The advantages en disadvantages of the 4 positions mentioned in the previous paragraph together with the score are tabulated in table 5.1 on the next page.

## 5.4 Classification possible positions

| Position | Advantages | Disadvantages | Score |
|---|---|---|---|
| **1** | - Good air flow → efficiency<br>- Low position → low center of gravity | - Little space for construction/difficult mounting<br>- Disturbance aerodynamics nose<br>- Much and difficult ducting → weight, pressure drop (heavy pump) | 4+3-3-2-1<br>=<br>1 |
| **2** | - Good air flow → efficiency<br>- Low position → low center of gravity<br>- Easy mounting | | 4+3+2<br>=<br>9 |
| **3** | - Low position → low center of gravity<br>- Easy ducting | - Bad air flow → efficiency<br>- Close to the engine → hot ambient air → efficiency<br>- Little space for construction/difficult mounting | 3+1-4-4-3<br>=<br>-7 |
| **4** | - Easy mounting<br>- Easy ducting | - Bad air flow → efficiency<br>- Near to engine → hot ambient air → efficiency<br>- High position → high center of gravity | 3+1-4-4-3<br>=<br>-7 |

*Tabel 5.1 Consideration possible positions*

The consideration shows that position 3 (at the bottom of the frame close to the engine), together with position 4 (at the top of the frame above the engine), are the worst options. Position 1 (in the front of the car at the nose) receives an average score and the best position is position 2 (in the sidepods). This result in the following classification (arranged from best to worst option):

1. In the sidepods
2. In the front of the car

3. At the bottom or at the top of the back frame

The best position for the radiator is position 2, the sidepods of the car.

# 6 Additional parts

## 6.1 Introduction

The most important part of the cooling system is the radiator. Besides the radiator there are some additional parts concerning the cooling system. Due to the lack of time caused by trouble with the engine test bed less measurements have been carried out than planned. Therefore a massive part of the considerations in this chapter are based on experience and available parts of last year.

In paragraph 6.2 the used fan will be discussed and paragraph 6.3 will discuss the water pump. In paragraph 6.4 the ducting of the fan behind the radiator will be discussed and eventually the piping will be discussed in paragraph 6.5.

## 6.2 Fan

From last year several fans are available. The fan which has been mounted on the URE04 overloaded the electrical system. Therefore a slightly lighter type of fan will be used. This fan is able to establish an volume flow rate of the air of $\dot{V}_a = 0,7\frac{m^3}{s}$. Dividing this amount by the area of the fan results in the established velocity of the airflow $v_a$ which amounts $v_a = 17\frac{m}{s}$. This is more than the average speed during the endurance. Therefore the capacity of the fan is abundantly enough to ensure sufficient air flow while standing still and driving with low speed.

## 6.3 Ducting fan and radiator

The radiator and fan will be ducted by carbon plates which will be mounted by aluminum profiles mutually. Carbon will be used since it is very light. Leakage will be finished off by using a suitable glue. Flaps will be used on top of the duct so the air flow will not be disturbed to much while the fan is not running. When the fan is running the flaps will be closed. Therefore the fan will cover the whole area of the radiator.

## 6.4 Water pump

Like last year an electrical water pump will be mounted. The amount of revolutions per minute of an electrical water pump does not depend on the amount of revolutions per minute of the engine which is a big advantage. The cooling mass flow can be controlled at all times. This ensures sufficient cooling while the care is standing still or driving very slow during certain proves. The coolant mass flow during the tests (Chapter 4) was comparable to the coolant mass flow of last year. One test concerning the coolant mass flow has been carried out which is described in paragraph 4.4.2. Therefore the coolant flow will not be changed. The specifications of the water pump are arranged below.

- Nominal voltage: $12V$
- Maximum capacity: $22\frac{l}{min}$
- Amperage: $1,2A$
- Delivery pressure: $12,4kPa$

## 6.5 Piping

Because of narrow spaces several narrow corners cannot be avoided. This has to be flexible material which will be heat resistant rubber. Were possible aluminum ducts will be used. Aluminum is a light and cheap material with another massive advantage: the coefficient of thermal conductivity is very high compared to other materials. The coefficients of thermal conductivity of prevalent materials are arranged in table 6.1 below.

| **Material** | $\lambda$ |
|---|---|
| Aluminum | $237\frac{W}{m \cdot K}$ |
| Copper | $390\frac{W}{m \cdot K}$ |
| Iron | $79\frac{W}{m \cdot K}$ |

| | |
|---|---|
| Stainless steel | $25 \frac{W}{m \cdot K}$ |
| Steel | $50 \frac{W}{m \cdot K}$ |

*Table 6.1 Coefficient of thermal conductivity of prevalent materials*

The only material in table 6.1 above which has a higher coefficient of thermal conductivity is copper. Copper is not suitable for piping because it is very vulnerable and besides it has more than three times the weight of aluminum.

# 7 Conclusions and recommendations

## 7.1 Area radiator

The equation used for calculating the required frontal area is

$$A_r = \frac{\dot{Q}_c}{\dot{K}_r \cdot \Delta T_{r,a}} \qquad [6.1]$$

The heat load has been determined by using the energy balance of the engine and amounts $41,7kW$. The efficiency of the radiator (given by the coefficient of heat transmission $\dot{K}_r$) has been determined on $7000\frac{W}{m^2 \cdot K}$. The temperate difference is given by the difference between the temperature of the ambient air and the average temperature in the radiator. This difference has been calculated taking the worst case scenario for granted which is Italy. Italy will have the hottest temperature which will result in the lowest temperature difference. Assuming an ambient air temperature of $30°C$ and a coolant temperature of $85°C$ the temperature difference amounts $55°C$.

Hence the required frontal area of the radiator amounts $0,108m^2$. The space available in the sidepods is very narrow. Therefore the radiator will be mounted at an angle of $40°$. Mounting the radiator at an angles which does not exceed $40°$ will not have a bad influence toward the efficiency of the radiator. The dimensions of the radiator will be $length \cdot width \cdot dept =$ $410 \cdot 300 \cdot 40mm$. In this dimensions the dimensions of the connections and some surrounding packaging are included.

## 7.2 Additional parts

The fan which will be used is a fan available from a previous season. The fan will be mounted behind the radiator and will pull the air through the radiator. Pulling the air through the radiator is more efficient than push the fan trough the radiator because of the centrifugal force caused by the fan. This centrifugal force will trough a lot of air out of the fan. Therefore a lot of air will not make it through the radiator at all by pushing the air trough the radiator. The fan is able to establish a velocity of the air flow of approximately $17\frac{m}{s}$. The calculations are based on a average speed of $10\frac{m}{s}$. Therefore the capacity of the fan is abundantly enough to ensure sufficient air flow while standing still and driving with low speed.

An electric water pump will be mounted. The amount of revolutions per minute of an electric water pump does not depend on the amount of revolutions per minute of the engine which is a big advantage since this ensures sufficient cooling during standing still an driving with low speed. The electric water pump which will be used is the same type used last year since less coolant mass flow will have a harmful effect towards the efficiency of the cooling.

The ducting between the radiator and fan will be made of carbon plates mounted by aluminum profiles mutually. Flaps will be mounted on top of the duct.

## 7.3 Recommendations for future research

### 7.3.1 Heat load engine

The measurements and calculations deliver good grip to determine the required area of the radiator although there are some constrains. Considering the cooling process the major inaccuracy is the heat load of the engine which has to be dissipated. The amount of this heat load has been determined by using a general energy diagram. To calculate the required area of the radiator more accuracy the heat load which has to be dissipated should be known. This amount can be determined using the engine test bed.

### 7.3.2 Aerodynamics

The efficiency of the radiator could be improved by researching the aerodynamics aspects. The measurements concerning the radiator mounted at different angles could be repeated more extensively to determine which angle makes the radiator the most efficient.

### 7.3.3 Coolant mass flow

The coolant mass flow could be tested more extensively to determine which amount of coolant mass flow is the most efficient.

### 7.3.4 Additional parts

The additional parts comprising the fan, the water pump and the piping deserve more attention during future researches. In particular the fan can be improved by testing. Measurements concerning an angle of the fan, the position of the fan and also the dimensions of the fan can improve the efficiency of the fan and therefore the efficiency of the cooling system.

# References

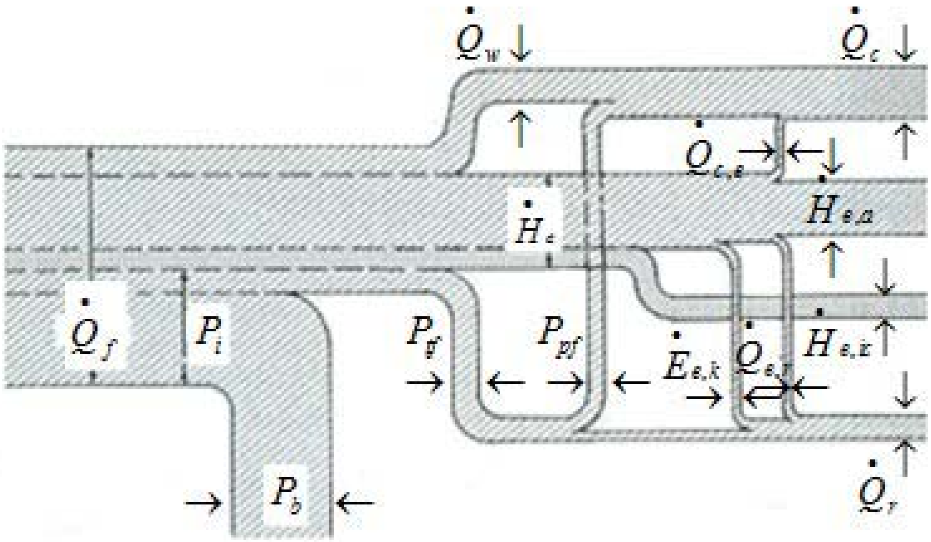

[1] Haberman, William L. and James E.A. John, *'Engineering Thermodynamic with Heat Transfer'*, Allyn and Bacon (1989)

[2] Heywood, John B. *'Internal combustion engine fundamentals'*, McGraw-Hill Science Engineering (1988)

[3] Feijen, J., *'Inleiding tot de warmtetechniek'*, HB Uitgevers (2003)

[4] Turns, Stephen R., 'Thermodynamics, Concepts and applications', Cambridge university press (2006)

# Appendix A Energy flow diagram IC engine




## Explanation used symbols

$\dot{E}_{e,k}$ = Exhaust gas kinetic energy flux

$\dot{H}_e$ = Exhaust gas enthalpy flux

$\dot{H}_{e,a}$ = Exhaust gas enthalpy flux rejected to atmosphere

$\dot{H}_{e,ic}$ = Exhaust gas enthalpy flux due to incomplete combustion

$P_b$ = Brake power

$P_i$ = Indicated power

$P_{pf}$ = Piston friction power

$P_{tf}$ = Total friction power

$\dot{Q}_c$ = Heat transfer rate to coolant

$\dot{Q}_{c,e}$ = Heat transfer rate to coolant in exhaust ports

$\dot{Q}_{e,r}$ = Heat transfer rate radiated from exhaust system

$\dot{Q}_f$ = Energy fuel flow

$\dot{Q}_r$ = Remaining energy transfers

$\dot{Q}_w$ = Heat transfer rate to combustion chamber wall

# Appendix B Terms and conditions

- The cooling system have to go through testing and racing. Therefore the cooling system have to endure at least one year.
- *'Eindhovense Radiateuren Fabriek' (ERF)* is sponsor of University Racing Eindhoven. Therefore the radiotor will be produced by ERF. In consultation with *'ERF'* a aluminum radiator will be used which have the following specifications:
    - Material: aluminium
    - Thickness: 40 mm
    - Time of delivery: approximately 2 months
- Schedule: at the end of October the concept has to be finished. At the end of January the cooling system has to be finished.
- *Water-cooled engines must only use plain water, or water with cooling system rust and corrosion inhibitor at no more than 0.015 liters per liter of plain water. Glycol based antifreeze or water pump lubricants of any kind are strictly prohibited.*[7]
- The temperature of the engine has to be approximately $90°C$ (source: experience previous year). This temperature has to be ensured during the proves described in Appendix C.
- In case a fan will be used, the fan has to be dust- and water resistance.
- In case a fan will be used, the fan may not overload the electrical system. The electrical system always have to be able to recharge.
- Considering the available space for the radiator and mounting the radiator, there will be used one radiator.
- The center of gravity of the car has to be as low as possible. Therefore the center of gravity of the radiator has to be as low as possible.
- The weight of the radiator should be as low as possible to keep the weight of the car as low as possible.
- The efficiency of the radiator has to be as high as possible.

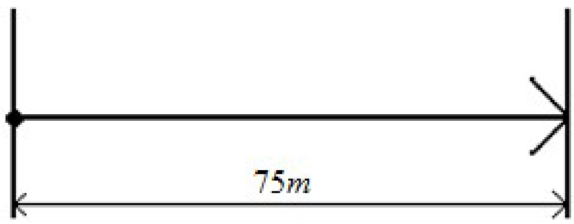
[7] Literally copied from the FSAE rules 2008-2009

# Appendix C Dynamic competition proves

The formula student competition is separated in two major parts which are the static events ant the dynamic events. The static events consists of technical inspection, cost and manufacturing event, presentation and design event. The four dynamic events are arranged below. The descriptions after 'Goal' and the figure under Ski-Pad event are literally copied from the FSAE rules 2009.

## Acceleration event

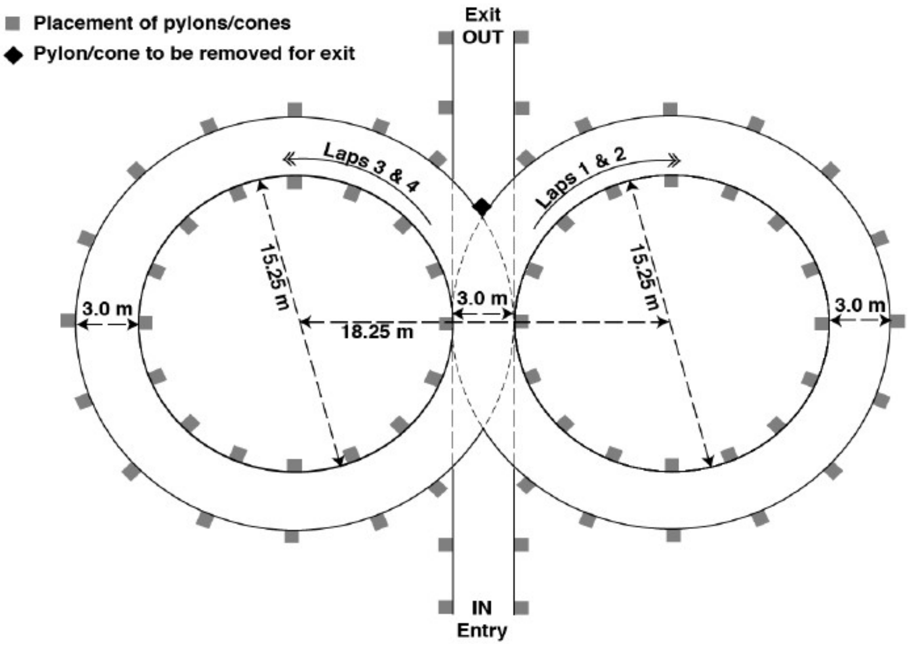
Goal: *The cars will accelerate from a standing start over a distance of 75 m (82 yards) on a flat surface.*




## Ski-Pad event

Goal: *The objective of the skid-pad event is to measure the car's cornering ability on a flat surface while making a constant-radius turn.*

FSAE SKIDPAD LAYOUT




Auto cross event

Goal: *The objective of the autocross event is to evaluate the car's maneuverability and handling qualities on a tight course without the hindrance of competing cars. The autocross course will combine the performance features of acceleration, braking, and cornering into one event. The length of each run will be approximately 0.805 km (1/2 mile) and the driver will complete a specified number of runs.*

***Straights:*** *No longer than 60 m (200 feet) with hairpins at both ends (or) no longer than 45 m (150 feet) with wide turns on the ends.*

***Constant Turns:*** *23 m (75 feet) to 45 m (148 feet) diameter.*

***Hairpin Turns:*** *Minimum of 9 m (29.5 feet) outside diameter (of the turn).*

***Slaloms:*** *Cones in a straight line with 7.62 m (25 feet) to 12.19 m (40 feet) spacing.*

***Miscellaneous:*** *Chicanes, multiple turns, decreasing radius turns, etc. The minimum track width will be 3.5 m (11.5 feet).*

## Endurance event

Goal: *The Endurance Event is designed to evaluate the overall performance of the car and to test the car's durability and reliability. The event will be run as a single heat approximately 22 km (13.66 miles) long.*

***Straights:*** *No longer than 77.0 m (252.6 feet) with hairpins at both ends (or) no longer than 61.0 m (200.1 feet) with wide turns on the ends. There will be passing zones at several locations.*

***Constant Turns:*** *30.0 m (98.4 feet) to 54.0 m (177.2 feet) diameter.*

***Hairpin Turns:*** *Minimum of 9.0 m (29.5 feet) outside diameter (of the turn).*

***Slaloms:*** *Cones in a straight line with 9.0 m (29.5 feet) to 15.0 m (49.2 feet) spacing.*

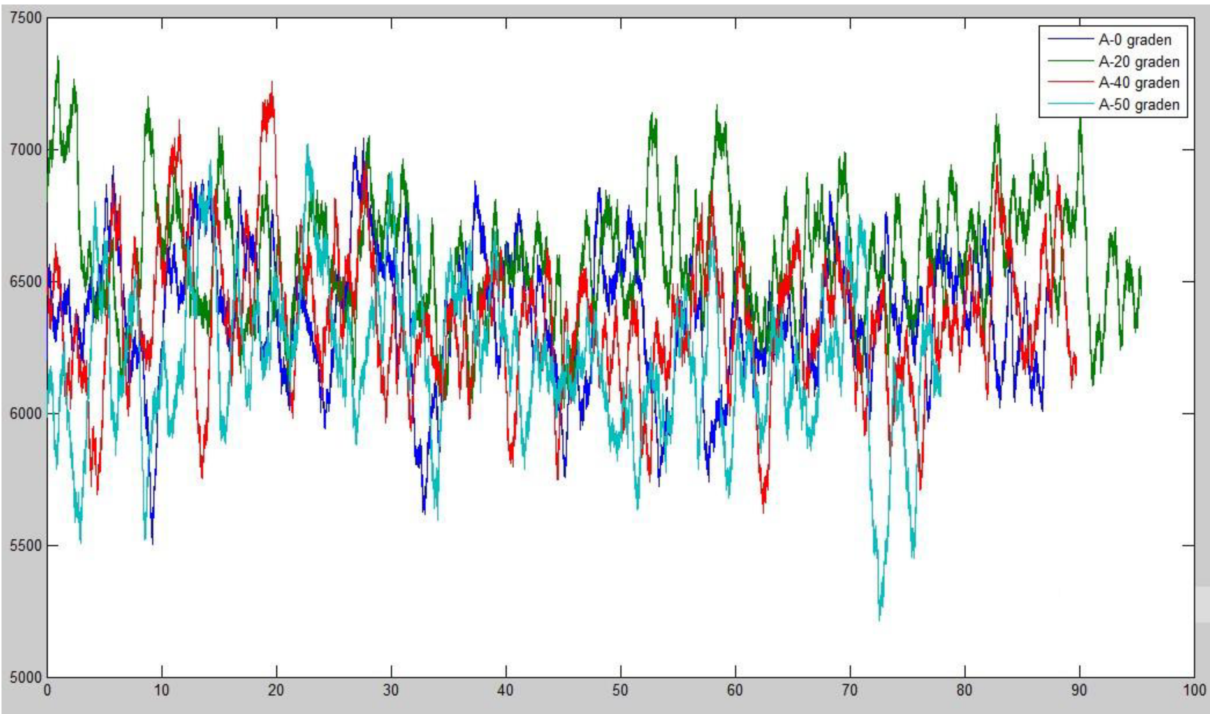
***Miscellaneous:*** *Chicanes, multiple turns, decreasing radius turns, etc. The standard minimum track width is 4.5 m (14.76 feet).*

# Appendix D Table average amount of efficiency

| Situation | Setup | Velocity air flow $\left[\frac{m}{s}\right]$ | Angle radiator $[°]$ | Coefficient of heat transmission $\left[\frac{W}{m^2 \cdot \Delta K}\right]$ |
|---|---|---|---|---|
| **Radiator without coating without ducted fan** | **Fan mounted not running** | 8 | 0 | 6400 |
| | | 8 | 20 | 6600 |
| | | 8 | 40 | 6400 |
| | | 8 | 50 | 6200 |
| | **Fan mounted not running** | 4 | 0 | 6200 |
| | | 6 | 0 | 5800 |
| | | 8 | 0 | 5700 |
| | | 10 | 0 | 5700 |
| | **No fan mounted** | 8 | 0 | 6500 |
| | | 8 | 20 | 6700 |
| | | 8 | 40 | 6900 |
| | | 8 | 50 | 6800 |
| | **No fan mounted** | 4 | 0 | 6500 |
| | | 6 | 0 | 6300 |
| | | 8 | 0 | 6500 |
| | | 10 | 0 | 6400 |
| | **Fan mounted and running** | 0 | 0 | 4700 |

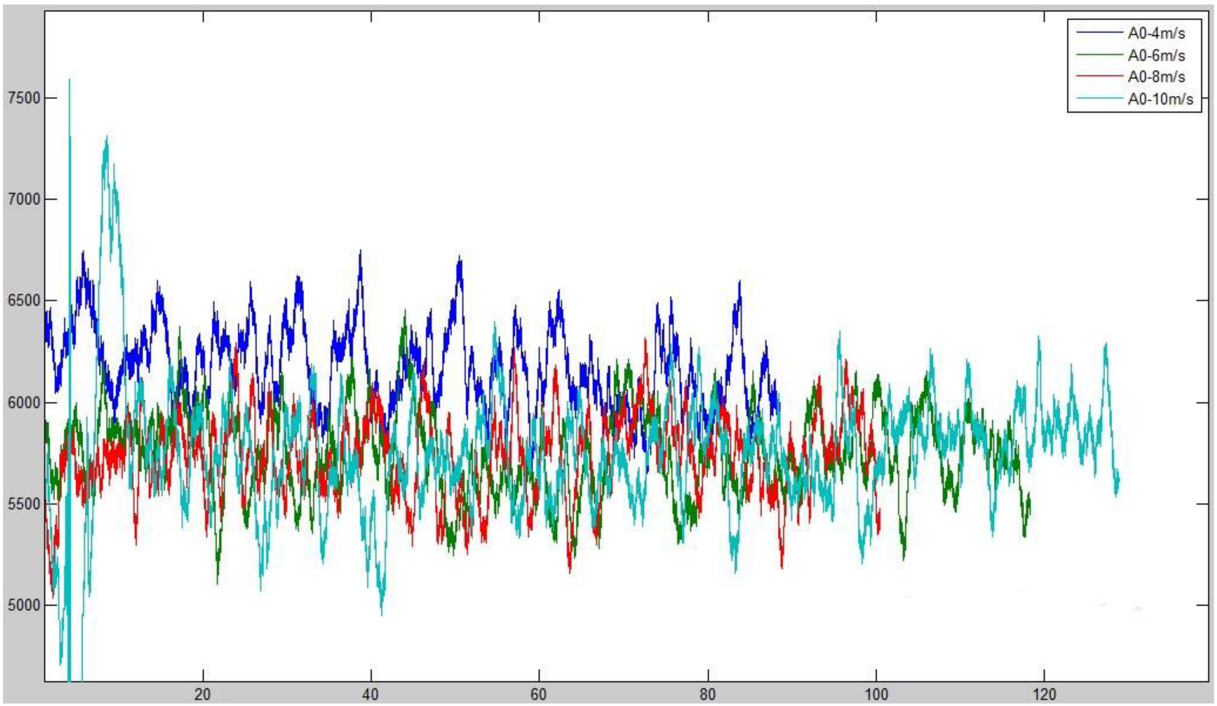
| | | | | |
|---|---|---|---|---|
| **Radiator without coating with ducted fan** | **Fan not running** | 4 | 0 | 6000 |
| | | 6 | 0 | 5900 |
| | | 8 | 0 | 6200 |
| | | 10 | 0 | 6400 |
| | **Fan running** | 0 | 0 | 5500 |
| **Radiator with coating without fan** | **Normal mass flow** | 4 | 0 | 6300 |
| | | 6 | 0 | 6400 |
| | | 8 | 0 | 7100 |
| | | 10 | 0 | 7000 |
| | **Lower mass flow** | 4 | 0 | 4200 |
| | | 6 | 0 | 5000 |
| | | 8 | 0 | 5100 |

# Appendix E Graph paragraph 4.2.1



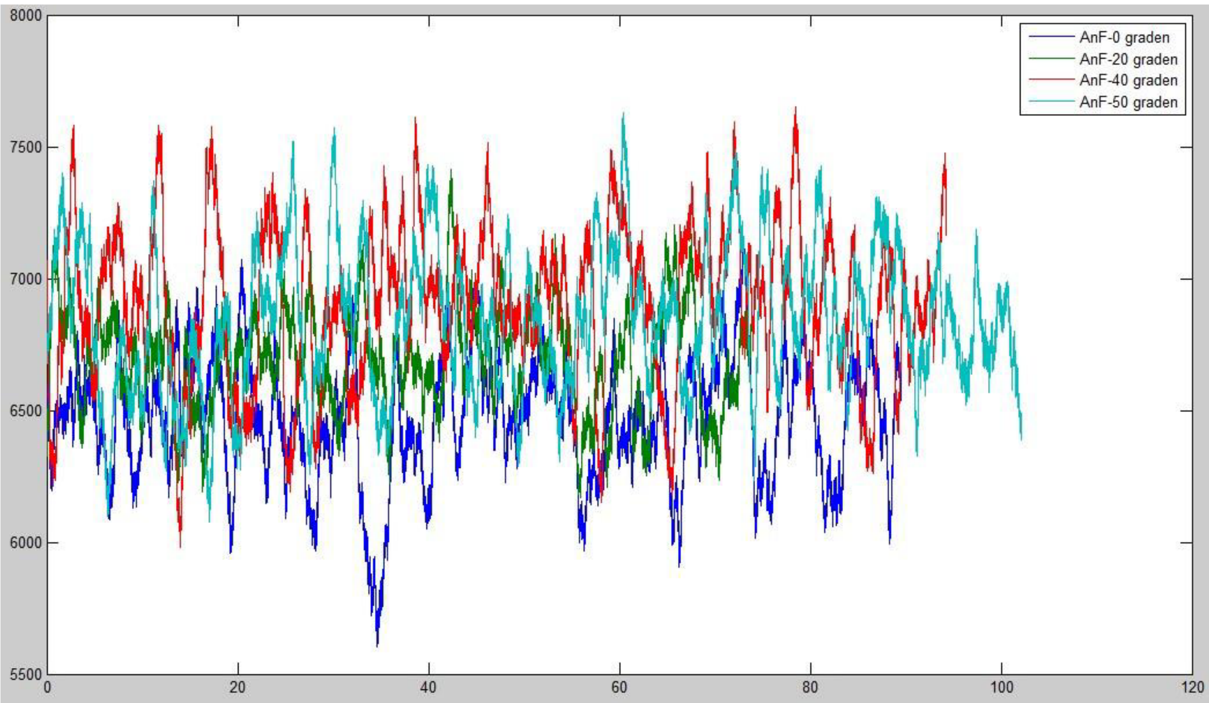

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix F Graph paragraph 4.2.2



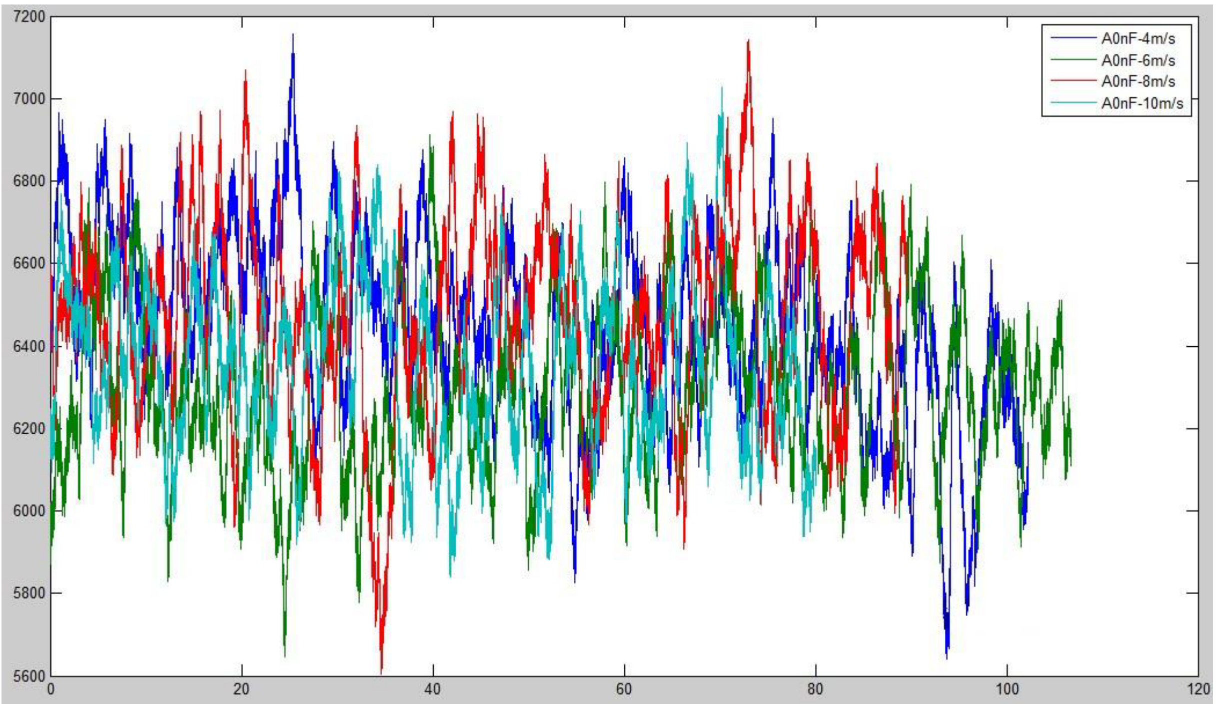

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix G Graph paragraph 4.2.3



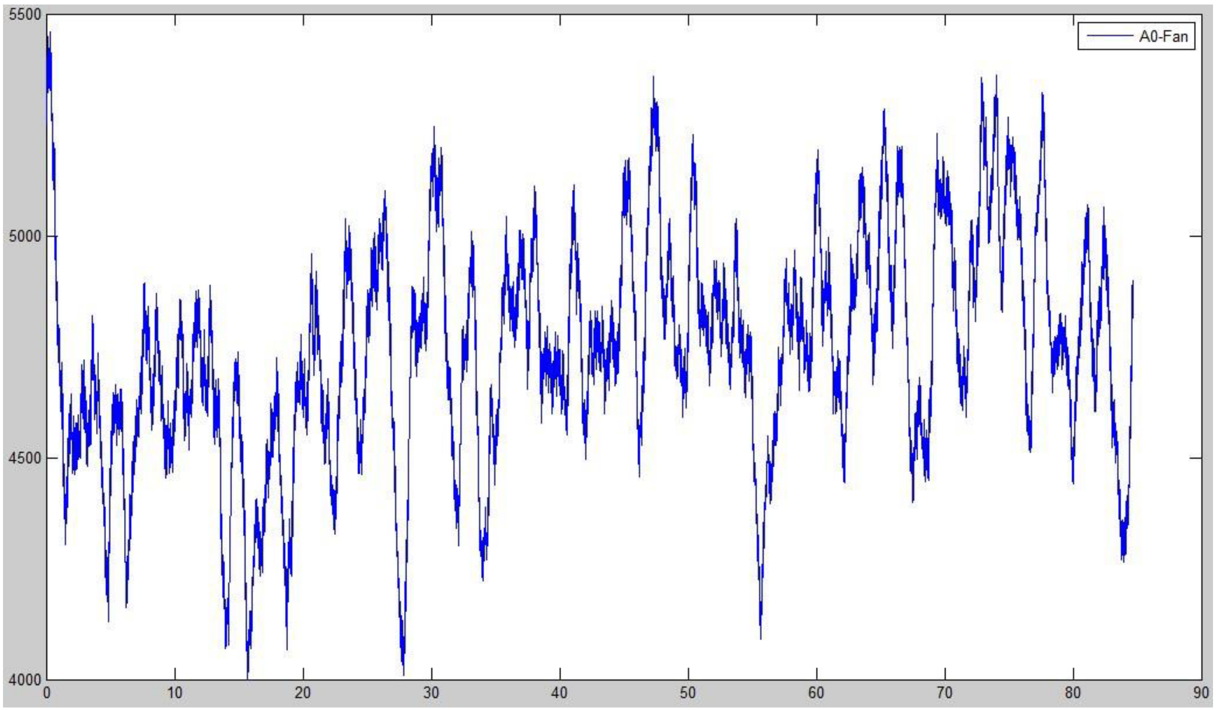

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix H Graph paragraph 4.2.4



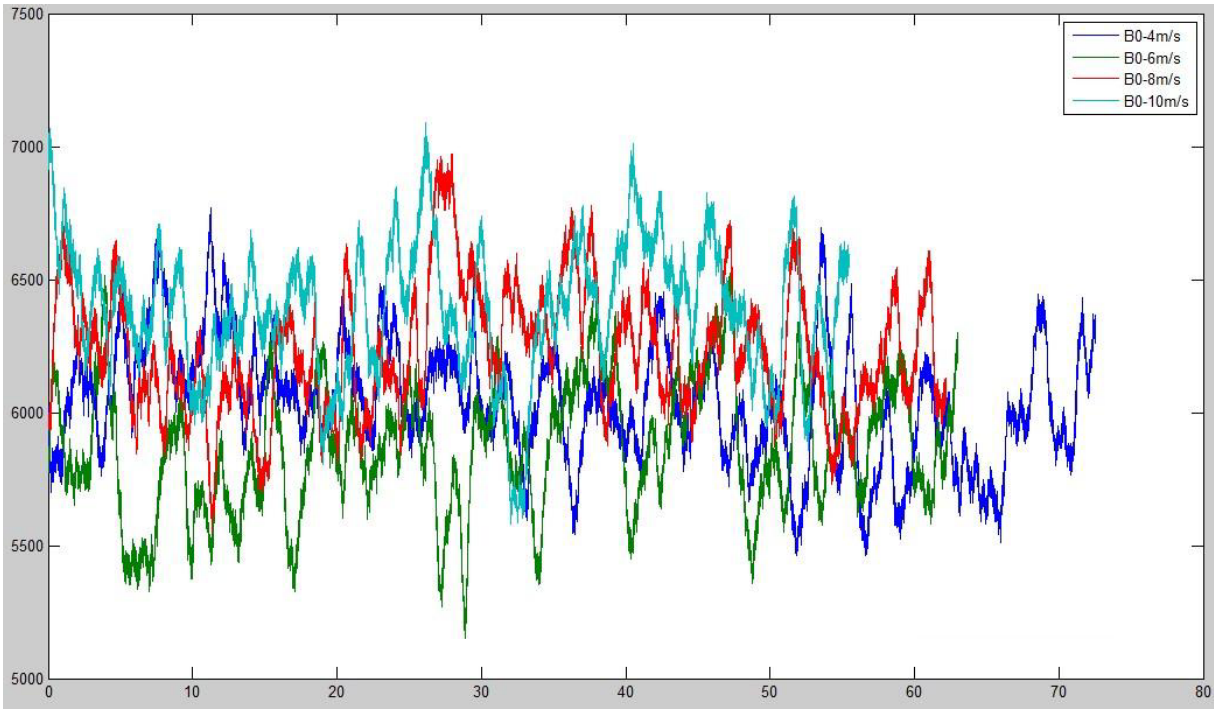

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix I Graph paragraph 4.2.5



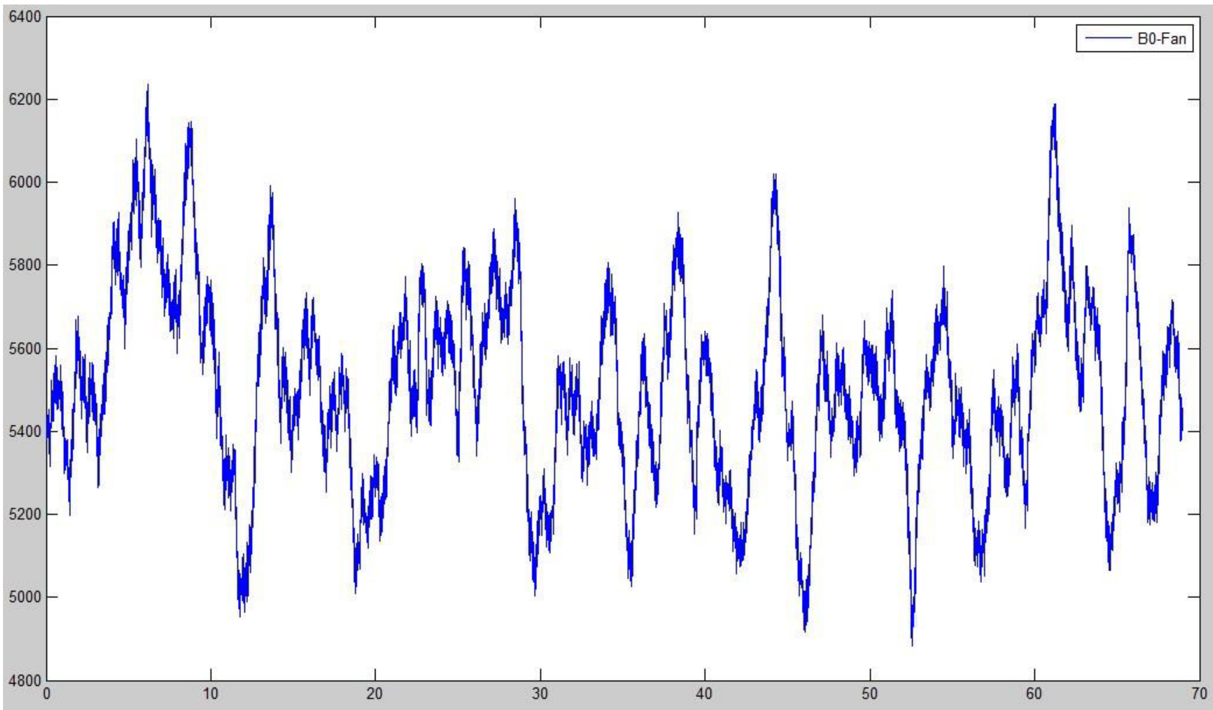

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix J Graph paragraph 4.3.1



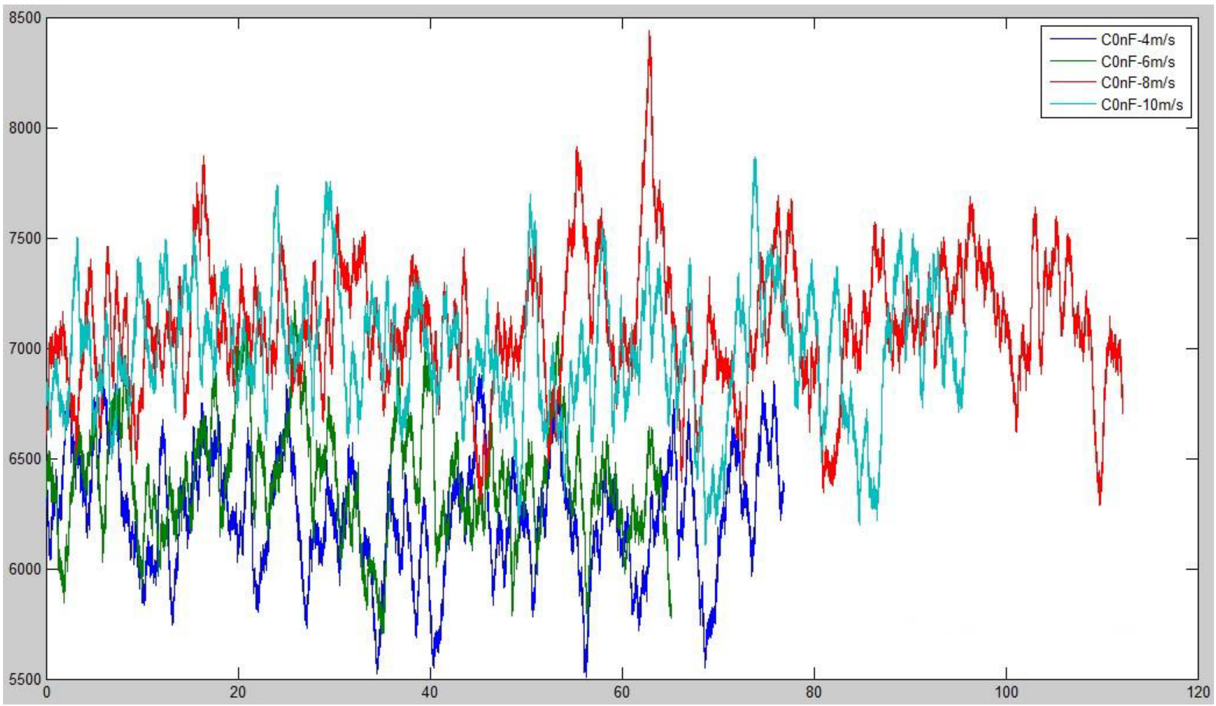

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix K Graph paragraph 4.3.2



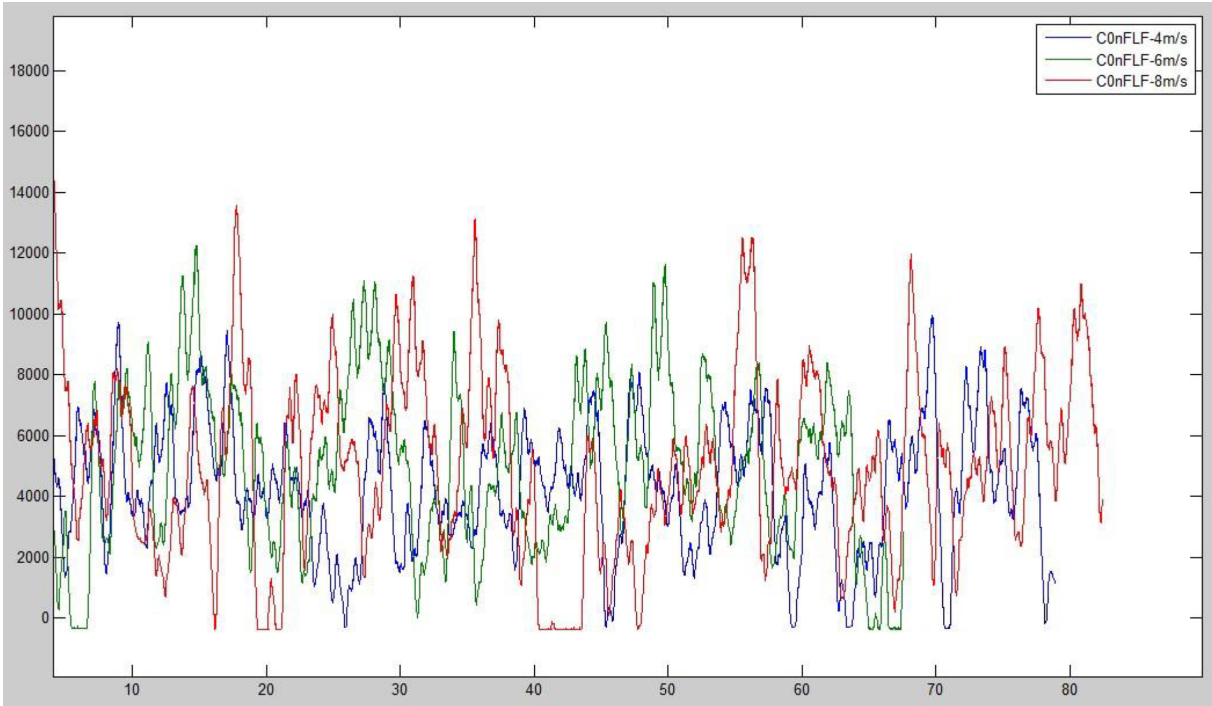

In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix L Graph paragraph 4.4.1




In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Appendix M Graph paragraph 4.4.2




In this graph the efficiency of the radiator (the coefficient of heat transmission given in $\frac{W}{m^2 \cdot K}$) has been plotted out against the time given in seconds. The graph clarifies the measurements inaccuracy.

# Nomenclature

## Normal characters

| | | | | |
|---|---|---|---|---|
| $A$ | - | Area | - | $m^2$ |
| $A_r$ | - | Frontal area radiator | - | $m^2$ |
| $\dot{E}_{e,k}$ | - | Exhaust gas kinetic energy flux | - | $W$ |
| $\dot{H}_e$ | - | Exhaust gas enthalpy flux | - | $W$ |
| $\dot{H}_{e,a}$ | - | Exhaust gas enthalpy flux rejected to atmosphere | - | $W$ |
| $\dot{H}_{e,ic}$ | - | Exhaust gas enthalpy flux due to incomplete comb. | - | $W$ |
| $\dot{K}_r$ | - | Coefficient of heat transmission radiator | - | $\frac{W}{m^2 \cdot K}$ |
| $L$ | - | Length | - | $m$ |
| $P_b$ | - | Brake power | - | $W$ |
| $P_i$ | - | Indicated power | - | $W$ |
| $P_{pf}$ | - | Piston friction power | - | $W$ |
| $P_{tf}$ | - | Total friction power | - | $W$ |
| $\dot{Q}$ | - | Heat transfer | - | $W$ |
| $\dot{Q}_c$ | - | Heat transfer rate to coolant | - | $W$ |
| $\dot{Q}_{c,e}$ | - | Heat transfer rate to coolant in exhaust ports | - | $W$ |
| $\dot{Q}_{e,r}$ | - | Heat transfer rate radiated from exhaust system | - | $W$ |

| | | | | |
|---|---|---|---|---|
| $\dot{Q}_f$ | - | Energy fuel flow | - | $W$ |
| $\dot{Q}_r$ | - | Remaining energy transfers | - | $W$ |
| $\dot{Q}_w$ | - | Heat transfer rate to combustion chamber wall | - | $W$ |
| $r_i$ | - | Radius inside | - | $m$ |
| $r_o$ | - | Radius outside | - | $m$ |
| $s$ | - | Thickness | - | $m$ |
| $t_m$ | - | Time of measurement | - | $s$ |
| $T$ | - | Temperature | - | $K$ |
| $T_{hoog}$ | - | Higher temperature Carnot cycle | - | $K$ |
| $T_{laag}$ | - | Lower temperature Carnot cycle | - | $K$ |
| $T_m$ | - | Temperature medium | - | $K$ |
| $T_{mi}$ | - | Temperature medium inside | - | $K$ |
| $T_{mo}$ | - | Temperature medium outside | - | $K$ |
| $T_w$ | - | Temperature wall | - | $K$ |
| $T_{wi}$ | - | Temperature wall inside | - | $K$ |
| $T_{wo}$ | - | Temperature wall outside | - | $K$ |
| $\Delta T_{r,a}$ | - | Temperature difference radiator-air | - | $K$ |
| $\dot{V}_a$ | - | Volume flow rate air | - | $\frac{m^3}{s}$ |
| $v_a$ | - | Velocity air flow | - | $\frac{m}{s}$ |

## Greece symbols

| | | | | |
|---|---|---|---|---|
| $\alpha$ | - | Coefficient of heat transfer | - | $\frac{W}{m^2 \cdot K}$ |
| $\alpha_i$ | - | Heat convection coefficient inside | - | $\frac{W}{m^2 \cdot K}$ |
| $\alpha_o$ | - | Heat convection coefficient outside | - | $\frac{W}{m^2 \cdot K}$ |
| $\sigma$ | - | Stefan-Boltzmann constant ($\sigma = 5,67 \cdot 10^{-8}$) | - | $\frac{W}{m^2 \cdot K^4}$ |
| $\lambda$ | - | Coefficient of heat conduction | - | $\frac{W}{m \cdot K}$ |

## Symbols without unit

| | | | | |
|---|---|---|---|---|
| $\varepsilon$ | - | Coefficient of emission | - | * |
| $k$ | - | Coefficient of heat passage | - | * |
| $\eta_c$ | - | Efficiency Carnot cycle | - | * |
| $Z_{Q,P}$ | - | Proportion heat load and brake power | - | * |