

An investigation into a nano-fuel technology combustion enhancer for diesel engines to achieve increased combustion efficiency

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Dissertation submitted in partial fulfilment of the requirements
for the degree *Magister* in *Mechanical Engineering* at the
Potchefstroom Campus of the North-West University

Supervisor:

Mr CP Kloppers

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**AN INVESTIGATION INTO A NANO-FUEL TECHNOLOGY
COMBUSTION ENHANCER FOR DIESEL ENGINES TO
ACHIEVE INCREASED COMBUSTION EFFICIENCY.**

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ABSTRACT

Keywords: fuel additive, internal combustion engine, dynamometer, engine test bench, GCV_v.

Modern day life would not be possible without the internal combustion engine and the fuel that provides them with the necessary energy. Unfortunately, there are a number of negative side effects to using these engines, such as the harmful greenhouse gases produced by the combustion process as well as the continuous increase in fuel price over the past 25 years. One solution to both of these issues would be to use a fuel additive designed to reduce fuel consumption and decrease the negative effects that the combustion of fossil fuels have on the environment.

The purpose of this study is to determine the effects that a fuel additive called Naf-Tech has on the engine performance, fuel consumption and efficiency of an engine and whether or not it works primarily as a combustion enhancer as stated by its manufacturers. In addition to these requirements, a mathematical model will be designed to simulate the engine under consideration at the conditions used during tests.

The method used to achieve the objectives of this study required the setting up of an engine test bench where it was possible to run tests on the engine both with and without the use of Naf-Tech. During these tests, measurements of parameters such as power, torque, fuel consumption as well as thermal, combustion and volumetric efficiency were taken to determine how the additive affected the various aspects of the engine's performance.

The mathematical model was written in a software package called Engineering Equation Solver (EES). The model was designed to simulate the thermodynamic cycle of the diesel engine using the same input values experienced by the engine on the test bench. From this, it was able to predict the required engine parameters.

The results obtained from the tests carried out on the test bench indicated that although there was no difference in the output power and torque of the engine with the use of Naf-Tech, there was a 12.4% reduction in fuel consumption. It was also found that by using the additive, there was a 12.27% and 0.1% increase in thermal and combustion efficiency respectively.





The results of the mathematical model were compared to those of the test bench as well as a separate Microsoft Excel model. By doing this, it was found that the model was able to predict the required parameters of the engine with a great degree of accuracy.

From the results obtained by this research, it can be concluded that Naf-Tech does, in fact, work as a combustion enhancer that is capable of improving the fuel efficiency of an internal combustion engine by the amount stated by its manufacturers. The results also indicate that with the use of Naf-Tech, significant reductions in harmful exhaust gases will be experienced.



DECLARATION

I Mark Gareth Sinclair (Identity Number: 9007025050085) hereby declare that the work contained in this dissertation is my own work. Some of the information contained in this dissertation has been gained from various journal articles; textbooks etc., and has been referenced accordingly.

Initial & Name

Witness





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NOMENCLATURE

Abbreviation or Acronym	Definition
AF	Air/fuel ratio
BDC	Bottom Dead Center
bmep	Brake mean effective pressure
CO	Carbon monoxide
CO ₂	Carbon dioxide
ECU	Engine control unit
EGR	Exhaust gas recirculation
GCV	Gross calorific value
HC	Hydrocarbon
NCV	Net calorific value
NT	Naf-Tech
NO _x	Nitrogen oxides
O ₂	Oxygen
SAPIA	South African Petroleum Industry Association
TDC	Top Dead Center
VOC	Volatile organic compounds





Symbol	Description	Unit
b_{mep}	Brake mean effective pressure	kPa
FC	Fuel consumption	L/h
GCV_v	Gross calorific value at constant volume	MJ/kg
H_P	Enthalpy of products of combustion process	kJ/kg
H_R	Enthalpy of reactants of combustion process	kJ/kg
m_a	Mass of air entering each cylinder per cycle	kg
m_{AF}	Mass of Air-Fuel mixture	kg
m_f	Mass of fuel injected per cycle	kg
$\dot{m}_a$	Mass flow rate of air	kg/s
$\dot{m}_f$	Mass flow rate of fuel	kg/s
N	Engine speed	rpm
NCV_p	Net calorific value at constant pressure	MJ/kg
P_{low}	Pressure at BDC	kPa
P_{high}	Pressure at TDC	kPa
Q_{in}	Heat transfer	kJ
$\dot{Q}_{in}$	Heat transfer rate	kW
r_v	Compression ratio	-
SFC	Specific fuel consumption	g/kW.h
T_A	Ambient temperature	°C
T_C	Temperature of AF mixture before compression	°C
T_H	Temperature of gas immediately after combustion	°C



Symbol	Description	Unit
V_1	Volume at TDC	m^3
V_2	Volume at BDC	m^3
V_d	Displacement volume	L or cm^3
W	Work done by one cycle of the engine	kJ
$\dot{W}$	Power	kW
W_b	Brake work	kJ
W_c	Work per cycle	kJ
W_{in}	Work required to compress Air-Fuel mixture	kJ/kg
W_{out}	Work created by combustion	kJ/kg
W_{NET}	Work produced by one cycle	kJ/kg
η_c	Combustion efficiency	%
η_f	Fuel conversion efficiency	%
η_{th}	Thermal efficiency	%
η_v	Volumetric efficiency	%
$\rho_{a,i}$	Density of air at inlet	kg/m^3
τ	Torque	$N.m$



Chapter 1: Introduction

1.1 Project Background

Since the invention of the internal combustion engine, continuous research and development has been done to improve engine performance and efficiency. Over the past few decades, the focus of research has not only been on improving the performance of engines, but also on the fuel that provides the energy to these engines (Guru, 2002). As the price of crude oil increases, the cost of each of the products that are refined from it increase as well (R.A. Ratti, 2013). This has a negative impact on the transportation industry as the price of petrol and diesel will rise. As this happens, the public finds it more difficult to survive as the cost of travelling increases as well as necessities such as food and clothing. With global warming being an additional negative side effect from the exhaust gas produced by internal combustion engines (C.Y. Lin, 2003), large amounts of research and funding have gone into the development of fuel additives. These additives are aimed at improving vehicle's engine efficiency and reducing harmful exhaust gases (C.Y. Lin, 2003) (Guru, 2002) (Kasper, 1999).

The purpose of these fuel additives is to alter or improve a number of the fuel's properties by simply adding them to the fuel. These properties include increasing the engine lubrication, combustion efficiency, octane/cetane number or just to clean out the carbon deposits left behind in the engine due to incomplete combustion and years of use. The octane rating is the measure of petrol's ability to resist detonating under pressure, where the cetane rating is the measure of diesel fuel's ignition delay (Pulkrabek, 2004).

There are many fuel additives on the market today, but for the purpose of this research, an additive called Naf-Tech, created by Naf-Tech Energy, will be under consideration.

Naf-Tech Energy is a South African company based in Pretoria. The creation of Naf-Tech aims at being a combustion enhancer, which means that it makes the combustion process more efficient, thus releasing more of the potential energy contained in the fuel to be converted to useful work. This would affect the fuel efficiency of the engine and would naturally have an effect on other performance aspects of the engine as well (Naf-Tech-Energy, 2014).





Tests previously conducted on Naf-Tech involve adding it to diesel fuel used by large transportation trucks that usually travel long distances across the country. The results of these tests indicated fuel savings between 8-12% (Naf-Tech-Energy, 2014). These tests only indicate the fuel savings obtained by the trucks but do not state whether or not there were any other performance variances with the use of Naf-Tech. It is for this reason that the research is being carried out.

A diesel engine in good running condition is needed to fulfil the requirements set out for this study. A GWM 2.5 TCi engine with the turbocharger system and ECU was found and purchased for this purpose. It was mounted on the dynamometer test bed and connected to the dynamometer where it will be put to test to obtain all the necessary data.

1.2 Problem Statement

Before any product can be put on the market, the manufacturer would need to thoroughly test its every aspect, i.e. to characterise the product. The benefits, as well as any negative effects of the product, have to be determined to ensure that it is actually beneficial to the end user.

The aim is to fully test the effects of the fuel additive, Naf-Tech, in diesel-fuelled engines. This includes monitoring and recording any variations while using standard diesel as well as the diesel/Naf-Tech mixture, for the following parameters:

- Output torque
- Output power
- Fuel efficiency
- Thermal efficiency
- Exhaust gas content
- Operating temperature

The above mentioned parameters will be fully discussed and explained in Chapter 2.

In order to reliably test all the necessary performance parameters, the energy flow through the engine will need to be determined to find the exact location and magnitude of each of the energy losses.



All claims from Naf-Tech Energy, the manufacturers of the fuel additive, such as a fuel savings of 10-20% and the fact that it works by improving the combustion efficiency will be put to test to determine whether or not they are accurate (Naf-Tech-Energy, 2014).

1.3 Scope

The scope of this project is to determine the effects that the diesel fuel additive called Naf-Tech has on engine performance. This is to be done by conducting tests on a diesel engine for sufficient periods of time for a full characterisation. The project is limited to the following for an engine when conducting tests using both 500 ppm commercial diesel as well as the diesel containing Naf-Tech fuel additive:

- Performance parameters:
 - Torque
 - Power
- Efficiencies:
 - Combustion efficiency
 - Thermal efficiency
- Fuel consumption
- Exhaust gas analysis with the aid of an opacity meter as well as a five gas analyser

A mathematical model specifically created for the engine being tested will simulate the above performance parameters. The simulation model will be able to accurately produce the same parameters measured during the tests. The experimental testing, as well as the mathematical model, will be used to determine the flow of energy through the engine when using standard diesel as well as a diesel/Naf-Tech mixture.

In addition to the above-mentioned components of the project, the combustion efficiency of the engine with both the commercial diesel, as well as the diesel/Naf-Tech mixture will be determined by the exhaust gas analysis. This will aid in determining why there are, or are not any performance variations between the two tests. Along with this, the Gross Calorific Value (GCV_v) will also be determined for the diesel fuel, the Naf-Tech fuel additive as well as the diesel/Naf-Tech mixture with the aid of a bomb calorimeter. These tests will help to determine the reasons for any differences in the results of the tests with and without the addition of the fuel additive.



1.4 Project Objectives

1.4.1 Main Objectives

The main objectives of this project are to run tests and analyse a diesel internal combustion engine. The purpose for this is to compile a complete comparison on how the engine performs with commercial diesel as the fuel, as well as with a diesel/Naf-Tech mixture. In order to do a thorough comparison, it is necessary to have a full understanding of how the energy of the fuel is divided and flows through the engine to each component.

To successfully complete the project at hand, the following main objectives need to be completed:

- Fully characterise the diesel engine with the commercial diesel as the fuel.
- Fully characterise the diesel engine with the correct ratio of diesel/Naf-Tech mixture as the fuel.
- Create a mathematical model to accurately simulate the same engine under the same conditions as tested.
- Parameters such as power, torque, fuel consumption, combustion efficiency and thermal efficiency need to be recorded in the final results for both the experimental tests as well as the mathematical model.
- A high-quality engineering report, capturing the method employed to solve the problems, simulations including assumptions, test procedures, results and conclusions.

1.4.2 Secondary Objectives

There are a few secondary objectives that do not form part of the scope of the project that are needed for the successful completion of this study. These are as follows:

- Obtain a new generation diesel engine with an ECU and common rail system.
- Set the engine up on the dynamometer test bench.
- The gross calorific value (GCV_v) for the diesel/Naf-Tech mixture, pure diesel as well as the Naf-Tech will need to be measured separately from the engine tests in order to calculate the energy released during the combustion process.



- Install a separate cooling system for both the engine and dynamometer that are both sufficient enough to keep the operating temperatures at a constant value for the full duration of the tests.
- Measure the air and fuel flow through the engine.
- Create a fuel dosing system to automatically add the correct amount of Naf-Tech to the diesel according to the mixing ratio.

1.5 Research methodology

Upon completion of chapter one, there would be a clear understanding of the problem and scope of the project at hand, research regarding the topic can begin for the literature study. Background literature for each aspect of this study will be researched to gather a wide range of knowledge to find a solution to the given problem. The research information gained in this chapter will be used at a later stage as it is implemented into the setting up and running of the experimental tests and simulation model.

Once the literature study has been completed, the detail design phase will begin. This will include the design of the mathematical model and detailed steps describing the manner in which it will solve the problem. The setting up and installation of the testing facility will be thoroughly explained and completed according to the knowledge gained from the literature study.

The next step in the process will be to run tests on the engine to fulfil all the requirements that were set out in chapter one. To successfully gather all the necessary results from the tests, the flow of energy through the system will need to be known. To do this, firstly the air and fuel flow through the engine will be measured. Along with this, the inlet and outlet temperatures for each of the energy flows, such as the air flow, water cooling system as well as the oil flow will be measured. The engine's driveshaft will be coupled to the dynamometer for the purpose of measuring the output torque and power produced by the engine. An opacity meter will be inserted into the exhaust system of the engine to measure the levels of each type of gas produced by the combustion process. This will also make it possible to determine the engine's combustion efficiency. For the purpose of this study, the tests with the above setup will be conducted for two instances i.e. with standard diesel and thereafter with the diesel/Naf-Tech mixture. By doing this, it is possible to determine the effects that the fuel additive has on the engine's performance as stated in the problem statement.



The mathematical model for this engine will be created to accurately produce all the results obtained from the experimental tests.

The results from both tests i.e. with standard diesel and diesel/Naf-Tech mixture will be compared to one another to find what effects the fuel additive has on the performance of the engine and whether or not there are any advantages or disadvantages to using it. The experimental results would then be compared to the results gained from the mathematical model to determine the accuracy of the simulation.

Upon completion of all comparisons, the necessary conclusions regarding the effects of the fuel additive on engine performance can be made.

1.6 Dissertation layout

- Chapter 1: Introduction to the project, including the problem statement, scope and objectives.
- Chapter 2: Literature survey regarding all relevant research on topics on or related to aspects of this project.
- Chapter 3: Mathematical model design includes the method followed to solve the simulation model as well as describing the formulae used.
- Chapter 4: Detail design phase will include all the instrumentation setup, calibration thereof as well as the experimental procedures for each of the tests.
- Chapter 5: This chapter will present all the results gained from the various tests carried out to fulfil the outcomes of this study. The results discussed in this chapter will include the GCV_v, Aggreko, CSIR, NWU GWM tests as well as the results from the mathematical model.
- Chapter 6: The verification and validation of the mathematical model takes place in this chapter. The EES model's results will be compared with the results obtained from the engine test bench as well as an Excel model.
- Chapter 7: This chapter concludes all the research findings made throughout the duration of the study.



Chapter 8: This chapter includes all the recommendations and improvements that would be beneficial to this research in the future.

1.7 Planning

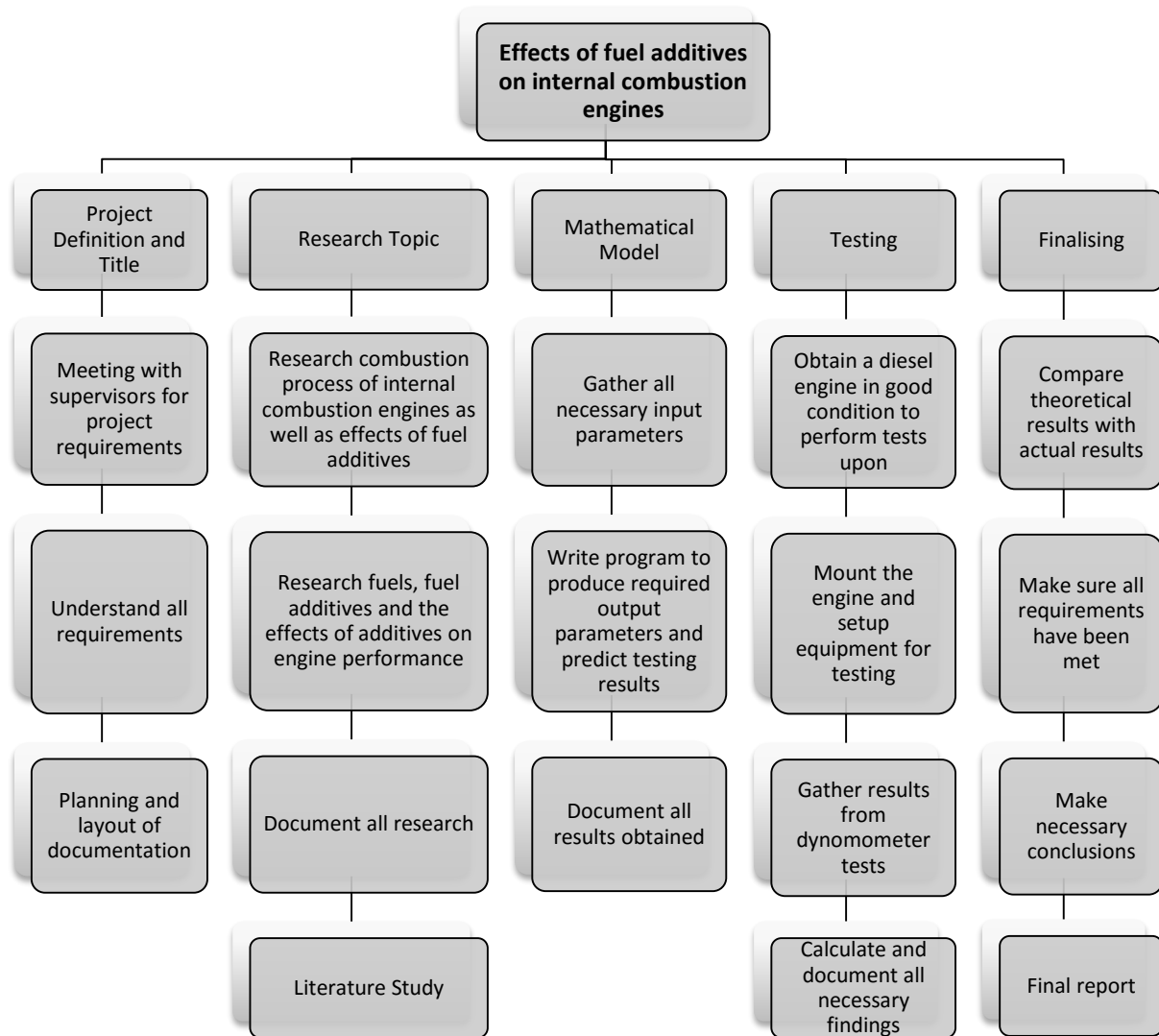


Figure 1 - Research planning



Chapter 2: Literature Survey

Chapter 2 covers all the research regarding diesel fuel, diesel engines and their performance characteristics. Discussions regarding important background knowledge necessary for understanding how an internal combustion engine operates, where it gets its energy from and how to measure their performance characteristics are presented in this chapter. In addition to this, fuel additives for improving engine performance will also be included in this chapter along with methods for testing these additives.

2.1 Background History

2.1.1 Early Internal Combustion Engines

The internal combustion engine is considered an energy converting unit as it converts the chemical energy contained in fuel to thermal energy, by means of combustion. The energy gained from this process is then converted to mechanical energy that propels a vehicle forward (Schwarz, 2009).

Creation of the first true four stroke engine took place in 1876, by Nicolaus August Otto and Eugen Langen. This was later referred to as the Otto cycle engine (Scoltock, 2010). In 1894, in an attempt to improve the internal combustion engine, a German engineer by the name of Rudolf Diesel, designed and created the first working diesel (compression ignition) engine. This engine was able to produce a much greater thermal efficiency of 26% when compared to the Otto cycle engine's 12% of the time (Shrinivasa, 2012). The two main reasons for the great improvement were that this engine operated with a much larger compression ratio than the conventional Otto cycle engine. The second reason being that the combustion process occurred over a longer period of time which meant that the engine was able to convert more of the combustion heat to mechanical energy. (Scoltock, 2010). Today's modern diesel engines used in motor vehicles usually produce an efficiency of between 40-45% (Goldensrein, 2011). With the addition of certain fuel additives, their efficiencies can be increased to the range of 50-60% (Naf-Tech-Energy, 2014), (Blue Sea Biotech), (Corporation, 2014), (Neptune Products).

With more and more vehicles on the roads each year in South Africa, internal combustion engines have become an extremely important aspect of everyday life (SAPIA, 2013). They enable effortless transportation between destinations in a fraction of the time it used to take and thus leaving more time to complete daily tasks. Over the past 25 years, the price of fuel has been on the rise (SAPIA, 2013) which makes it important for internal combustion engines to be as efficient as possible to cut down the cost of transportation.

2.1.2 Diesel Engine

In recent times, compression ignition engines have become extremely efficient, and with the worldwide “green” trend to reduce the global warming effect, these engines have become more popular than in past years (SAPIA, 2013). The increase in diesel consumption can be seen in Figure 2 below. These statistics show that diesel engines are slowly becoming more popular than the petrol engines.

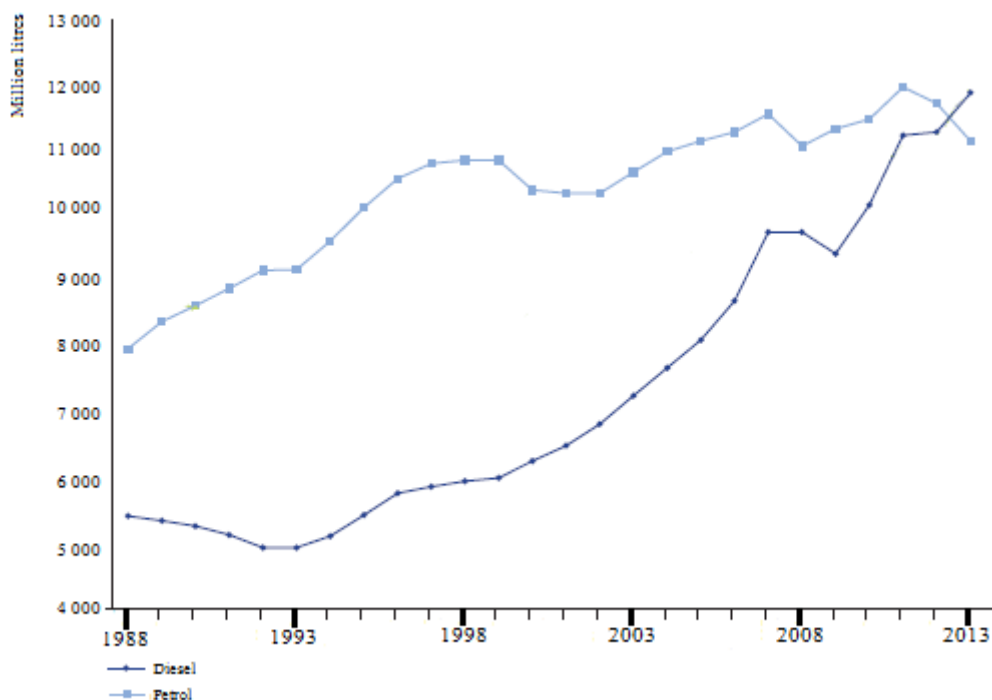


Figure 2 - Increase in petrol and diesel consumption in South Africa (SAPIA, 2013)

Diesel engines produce a higher amount of torque than petrol engines. For this reason, they are widely used in larger transportation modes such as trucks, trains and ships where torque is needed over power in order to pull the massive loads (Haywood, 1988).



2.2 Performance

The performance of any piece of machinery is of utmost importance as it indicates the capabilities of the machine as well as being a measure of how efficiently it operates. The performance of an internal combustion engine is affected by aspects such as fuel quality, atmospheric air pressure, atmospheric air temperature, air/fuel ratio and compression ratio to name a few. Engine performance is measured from output parameters such as torque, power and thermal efficiency.

To correctly measure the performance of any engine, it would need to be connected to a dynamometer to measure the power and torque produced. To obtain a full range of data for engine performance, the engine would also need to be connected to the necessary measuring equipment such as thermocouples, pressure sensors, exhaust gas analysis system as well as water, fuel, oil and air flow devices. In addition to this necessary equipment, modern engines will need to be connected to their corresponding electronic control unit (ECU) (Usman Asad, 2011).

2.2.1 Torque

Torque is defined as a force acting at a certain distance from a pivot point which tends to cause a rotational effect. The torque produced by an engine is an important indicator of the engine's ability to do work (Pulkrabek, 2004). Torque (τ) is measured in Newton-meter [N-m] and can be determined from the following formula (Pulkrabek, 2004):

$$\tau = \text{Force} \times \text{Distance} \quad \text{Equation 1}$$

The torque produced by a four stroke engine is determined as follows (Pulkrabek, 2004), (Haywood, 1988):

$$\tau = \frac{(bmep) V_d}{4\pi} \quad \text{Equation 2}$$

or

$$\tau = \frac{\eta_f \eta_v N V_d GCV_v \rho_{a,i} (F/A)}{4\pi} \quad \text{Equation 3}$$



where:

- bme_p - Brake mean effective pressure
- V_d - Cylinder displacement volume (not including combustion chamber)
- η_f - Fuel conversion efficiency
- η_v - Volumetric efficiency
- N - Engine speed
- GCV_v - Gross calorific value
- $\rho_{a,i}$ - Density of air at inlet
- $\frac{F}{A}$ - Fuel / Air ratio

The brake mean effective pressure (bme_p) is the average pressure that would be exerted on each of the pistons in the engine from top dead center (TDC) to bottom dead center (BDC) for a specific known brake power output (Pulkrabek, 2004). This can be calculated as follows (Pulkrabek, 2004):

$$bme_p = \frac{W_b}{\Delta v} \quad \text{Equation 4}$$

$$\Delta v = v_{ol\ BDC} - v_{ol\ TDC}$$

where: W_b - Brake work

The conventional method for measuring the torque output of an engine is to set it up on a dynamometer test bench. This makes it possible to measure torque over the full rev range of the engine and from this, a torque curve such as Figure 3 below can be drawn.

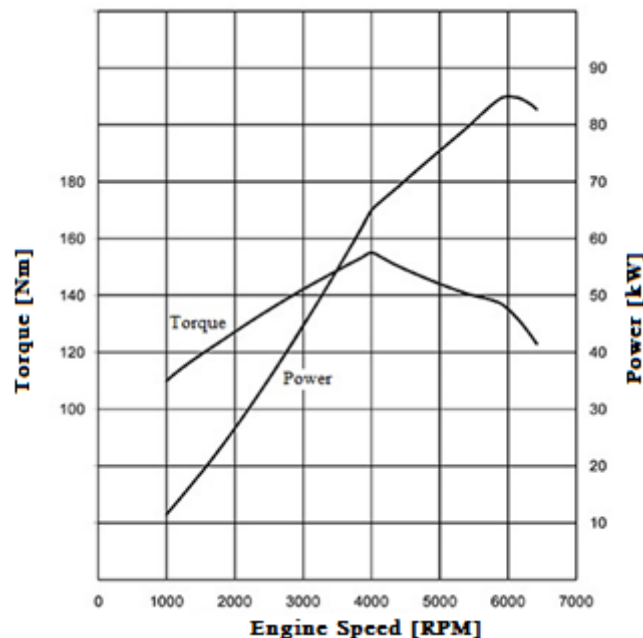


Figure 3 - Torque/Power curve for 1.6L VVT Chevrolet Aveo (General-Motors, 2014)

Engine manufacturers try as much as possible to get engines to produce a “flatter” torque curve in order to obtain a high torque output at both low and high speeds. This is where the advantages of using a diesel engine come into play. Diesel engines produce their maximum torque at a low RPM and gradually decrease as the engine speed rises.

The addition of a forced induction system such as a turbocharger would maintain the maximum torque for an extended rev range, as in Figure 4 below, before it begins to decrease (Pulkrabek, 2004).

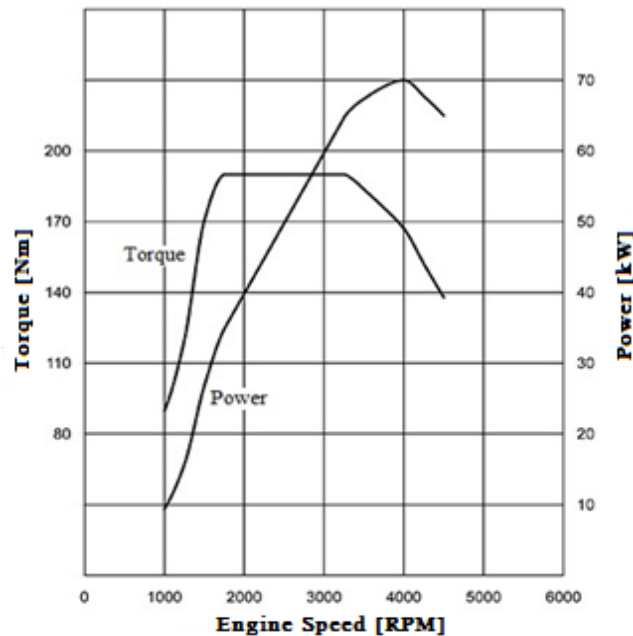


Figure 4 - Torque/Power curve for 1.3L turbo diesel Opel Corsa (General-Motors, 2014)

2.2.2 Power

The power, measured in kilowatts [kW], which is produced by an engine is defined as the rate of work that the engine is able to deliver over a specific time (Pulkrabek, 2004). For example, an engine that produces a larger amount of power would be able to move a specific load from position A to position B faster than an engine that produces less power. When comparing peak power and torque outputs, the peak power is reached at a higher engine speed than the peak torque output (General-Motors, 2014). This is because the torque output of an engine is not only dependent on engine speed, but other aspects as well, whereas power is directly proportional to the product of the engine speed and the torque produced. It is for this reason that the engines power continues to increase past the point of maximum torque in the RPM range as seen in the image above (Pulkrabek, 2004).

The output power in [kW] of an engine can be determined from the following formula if the engines torque is known (Pulkrabek, 2004):



$$\dot{W} = \frac{N 2\pi \tau}{60} \div 1000 \quad \text{Equation 5}$$

or

$$\dot{W} = \frac{N \tau}{9549.305}$$

where: $\dot{W}$ - Power

τ - Torque

In some cases, the torque value is not known. The following formula can be used to determine the power of an engine without torque (Haywood, 1988):

$$\dot{W} = \frac{\eta_f m_a N GCV_v (F/A)}{n_R} \quad \text{Equation 6}$$

or

$$\dot{W} = \frac{\eta_f \eta_v N V_d GCV_v \rho_{a,i} (F/A)}{2} \quad \text{Equation 7}$$

$$\eta_f = \frac{W_c}{m_f GCV_v} \quad \text{Equation 8}$$

$$\frac{F}{A} = \frac{\dot{m}_f}{\dot{m}_a} \quad \text{Equation 9}$$

$$\eta_v = \frac{2 \dot{m}_a}{\rho_{a,i} V_d N} \quad \text{Equation 10}$$

or

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} \quad \text{Equation 11}$$

where: m_a - Mass of air entering each cylinder per cycle

n_R - Number of crank revolutions per power stroke

W_c - Work per cycle

m_f - Mass of fuel injected per cycle

$\dot{m}_f$ - Fuel mass flow rate

$\dot{m}_a$ - Air mass flow rate



2.2.3 Thermal Efficiency

The thermal efficiency of an engine is a ratio of how much energy is produced by the engine's combustion process in a specific time period compared to the amount of energy provided to the engine by the fuel over the same period. In other words, it is the percentage of useful energy extracted from the fuel (Judge, 1967). Internal combustion engines have never been extremely efficient, but since the invention of the four stroke engine, the diesel engine has been able to produce a higher efficiency than that of the petrol engine. Today's modern diesel engines have been improved greatly since the first one was created in 1894. Petrol engines are capable of realistically producing efficiencies around 35-40% while diesel engines are able to produce efficiencies of 40-45% (Goldensrein, 2011).

Although the efficiencies of modern engines are improvements over the original engines, they are still extremely inefficient. The problem is that internal combustion engines produce heat that in turn generates entropy. Entropy is described as a measure for indicating how much chaos exists in a system (Sonntag, 2009). From this definition, it can be seen that the entropy generated in a system decreases the amount of useful energy extracted from the combustion process (Goldensrein, 2011).

The thermal efficiency of any engine is affected by numerous factors. One of these factors being the engines compression ratio. The thermal efficiency is directly proportional to the engine's compression ratio (Judge, 1967). Thus, the higher the compression ratio of an engine, the greater the thermal efficiency will be, such as is the case with compression ignition engines. This is due to the fact that a larger compression ratio would generally mean that the expansion stroke of the engine would be longer and would, therefore, transfer more of the gases energy as useful work to the piston. In other words, the greater expansion results in less heat being rejected to the exhaust system, making the engine more efficient (Rogowski, 1953). This is true until a point where the compression ratio becomes too high, and the increase in thermal efficiency becomes progressively smaller until it begins to decrease as the compression ratio increases (Judge, 1967). The graph in Figure 5 below clearly shows that there is an increase in efficiency as the compression ratio rises.

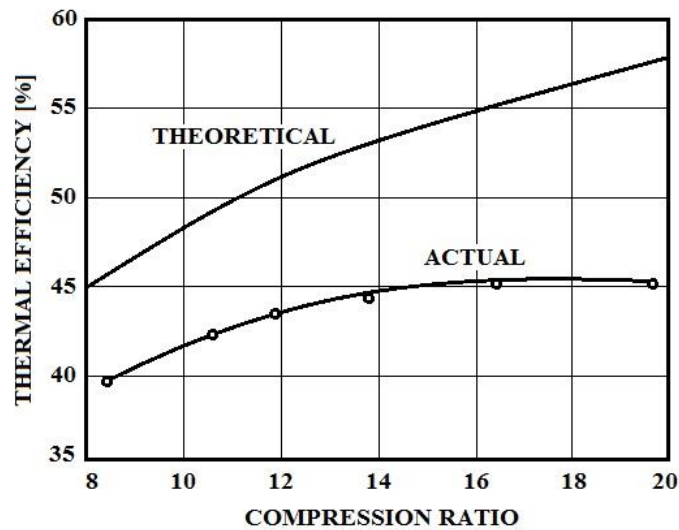


Figure 5 - Ideal and actual thermal efficiency vs. compression ratio (Judge, 1967)

The compression ratio of an engine is the difference in volume from BDC to TDC. The following formula can be used to determine the compression ratio (Pulkrabek, 2004):

$$r_v = \frac{V_2}{V_1} \quad \text{Equation 12}$$

where: r_v - Compression ratio

V_1 - Volume at TDC

V_2 - Volume at BDC

Typical values for compression ratios in diesel engines range from 12:1 to 24:1 (Pulkrabek, 2004).

As can be seen in Figure 5 above, there is a great difference between the actual thermal efficiency and the theoretical thermal efficiency. The theoretical values indicate what efficiencies the engine would be able to produce without the effect of any losses in the engine. These values are the maximum efficiencies that the engine could ever produce.



The following formula can be used to determine the theoretical efficiency of an engine (Goldensrein, 2011):

$$\eta_{th} = 1 - \frac{T_C}{T_H} \quad \text{Equation 13}$$

where:

- η_{th} - Thermal efficiency
- T_C - Temperature of AF mixture before compression
- T_H - Temperature of gas immediately after combustion

The actual efficiency simulates the real world conditions more accurately than the theoretical efficiency. To calculate the actual thermal efficiency, the following formula can be used (Pulkrabek, 2004):

$$\eta_{th} = \frac{\dot{W}}{\dot{Q}_{in}} \quad \text{Equation 14}$$

$$\dot{Q}_{in} = \dot{m}_f GCV_V \eta_c \quad \text{Equation 15}$$

where:

- $\dot{Q}_{in}$ - Heat transfer rate
- $\dot{m}_f$ - Mass flow rate of fuel
- GCV_V - Gross calorific value of the fuel
- η_c - Combustion efficiency

Some other factors that have an influence on the efficiency of an engine are the physical design of the engine. This includes the actual shape and volume of the combustion chamber, the fuel injection method and location in the cylinder as well as the intake and exhaust systems of the engine. The air-fuel ratio of the engine also plays an important role when considering the thermal efficiency (Judge, 1967) as can be seen in the Figure 6 below:

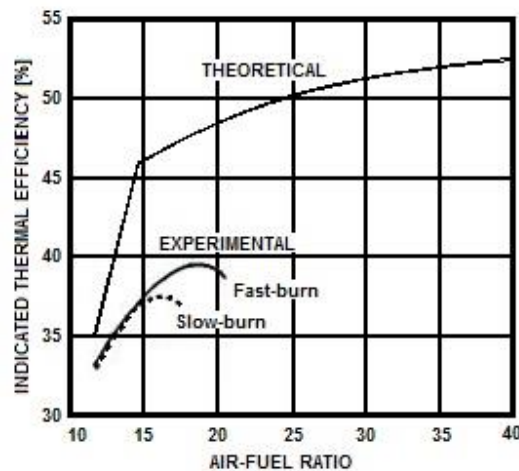


Figure 6 - Thermal efficiency vs. air-fuel ratio (Judge, 1967)

2.2.4 Combustion Efficiency

The combustion efficiency of an internal combustion engine indicates the percentage of the fuel that has been completely combusted during the combustion process i.e. the completeness of combustion. This also refers to the amount of energy that can be extracted from the fuel during the combustion process (Haywood, 1988). Combustion efficiency is an important factor in internal combustion engines since as little as 0.5% of unburnt fuel in the exhaust can have a dramatic effect on engine performance. The combustion efficiency of most modern engines is above 98% (Haywood, 1988).

The best method to determine the combustion efficiency of an engine is to analyse the exhaust gas contents as these are a direct link to what happens during the combustion process. This will indicate any unburnt fuel particles along with the percentage of each of the exhaust gases such as carbon monoxide (CO), carbon dioxide (CO₂), nitrogen oxides (NO_x) as well as aldehydes (Judge, 1967). Once the percentages of each of the exhaust gases are known, it is possible to calculate the combustion efficiency of the engine for a specific instance. The following methods can be used to determine the combustion efficiency (Storm, 2013):



$$\eta_c = \left(1 - \frac{CO}{CO+CO_2}\right) \times \frac{1-C_xH_y}{1} \quad \text{Equation 16}$$

where: η_c - Combustion efficiency

CO - Carbon monoxide

CO₂ - Carbon dioxide

C_xH_y - Unburnt fuel

The above formula uses the percentages of each gas from the engine exhaust, whereas the formula below makes use of the enthalpies of the combustion process to calculate combustion efficiency (Haywood, 1988).

$$\eta_c = \frac{H_R(T_A) - H_P(T_A)}{\dot{m}_f \times GCV_f} \quad \text{Equation 17}$$

where: H_R - Enthalpy of reactants of the combustion process

H_P - Enthalpy of products of the combustion process

T_A - Ambient temperature

2.3 Measuring engine performance

As new technology is developed, internal combustion engines become increasingly advanced and more efficient. This means that engines are able to operate at a higher performance level, while using less energy (Shrinivasa, 2012) (Pulkrabek, 2004). With these advancements, it is necessary to be able to measure the engine's performance levels, and how efficient they actually are.

Engine performance is usually measured with the aid of a dynamometer (Haywood, 1988). This is a device that is used to measure engine output torque and power over the engines full range of load and speed (Pulkrabek, 2004). It primarily measures the torque produced by the engine and by using this value along with the engine's speed in the following formula, the power developed by the engine can be determined (Pulkrabek, 2004).



$$\dot{W} = \frac{N 2\pi \tau}{60} \quad \text{Equation 18}$$

There are two main types of dynamometers which can be used to measure the performance of an engine namely, chassis dynamometers and engine dynamometers.

The chassis dynamometer or “rolling road” is used to test the performance of a vehicle which includes all the losses such as from transmission, differential and gearbox. These types of dynamometers require the vehicle to park with its driving wheels on the rollers. These rollers are turned by accelerating the vehicle, thus applying a force on the rollers by the wheels. The performance of the vehicle is then measured and displayed on a torque/power curve (Martyr, 2012) such as in Figure 3 and Figure 4.

Engine dynamometers operate in a different manner to chassis dynamometers. These are connected directly to the engine’s crankshaft or flywheel. The engine, therefore, needs to be removed from the vehicle before it can be tested. By measuring engine performance in this manner, all the vehicle losses such as the transmission and gearbox will be eliminated and, therefore, makes it possible to measure the engine's full capabilities (Martyr, 2012).

For purposes of this study, only the engine dynamometer will be considered. The manner in which engine dynamometers operate is by applying a braking force to the engine being tested, this force then resists the turning direction of the engine. The energy that has been produced by the engine is absorbed using various methods in the dynamometers. By observing how much force is required to resist the engine turning at all operating speeds, it is possible to calculate the torque and power developed and thus draw a power-torque graph (Pulkrabek, 2004).

2.3.1 Types of dynamometers

Numerous different types of engine dynamometers exist, where each type would have its own advantages and disadvantages for different purposes. Although each type of dynamometer works differently, the basic principles remain the same.

Water Brake Dynamometer (Hydraulic Dynamometer)

The tests required for the purpose of this project will be conducted with the aid of a water brake dynamometer, specifically the Froude type DPX dynamometer.

Water brake dynamometers work by rotating a cylindrical rotor inside a watertight case. When the rotor is in motion due to the engine's rotation, the centrifugal force creates a toroidal flow in recesses formed in the casing and rotor.

This effect causes momentum to transfer from the rotor to the casing (stator) which then develops a torque that resists the direction of rotation of the rotor. The torque applied to the engine is varied by moving sluice gates between the rotor and stator. This increases or decreases the development of the toroidal vortices (Martyr, 2012).

The image below shows a cross sectional diagram of the Froude type DPX dynamometer:

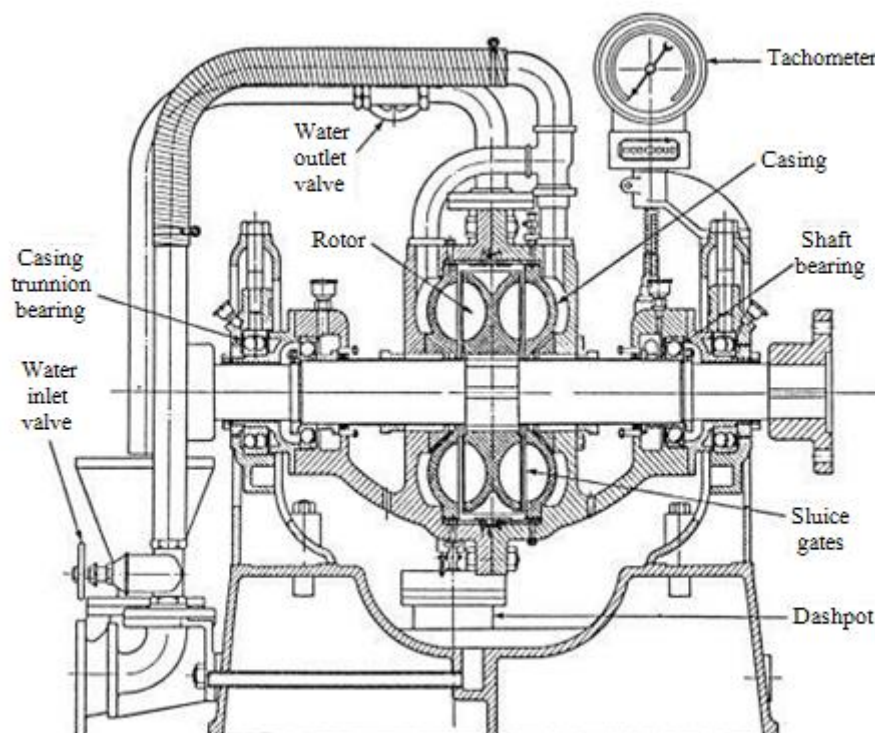


Figure 7 - Froude type DPX hydraulic dynamometer (Martyr, 2012)

The image below was taken during the restoration of the water brake dynamometer being used for this study. The recesses in the rotor are clearly seen along with the sluice gates that are used to control the load applied to the engine.



Figure 8 - Water brake dynamometer rotor with sluice gates

Some types of hydraulic dynamometers include:

- Constant fill machines: These operate by moving thin sluice gates between the stator and rotor, this controls the amount of torque the dyno absorbs from the engine (Martyr, 2012).
- Variable fill machines: These types of dynamometers use a variable water mass control in order to absorb the engine torque (Martyr, 2012).

Eddy Current Dynamometers

Eddy current dynamometers work by applying an excitation current to field coils in the dynamometer housing to create a magnetic field. This magnetic field acts upon a high magnetic permeability toothed rotor disc between the field coils. This rotor disc is connected to, and driven by the engine's driveshaft in the same manner as the water brake

dynamometer. As the engine turns the rotor disc, it causes changes in the magnetic flux in the loss plates which create a force that opposes the rotors turning. A computer controls the amount of force that is needed to brake the engine being tested where the value is manually or automatically changed (Martyr, 2012).

During operation, the eddy current dynamometers generate large amounts of heat as the dynamometer resists the rotation created by the engine i.e. dissipating the engine's power into heat. The greater the engine's power output, the greater the resisting force and thus the greater the heat production. This makes it necessary for the dynamometer to run with a cooling system to keep it within its operating temperature range (Martyr, 2012).

A cross-sectional diagram of a typical eddy current dynamometer can be seen in Figure 9 below:

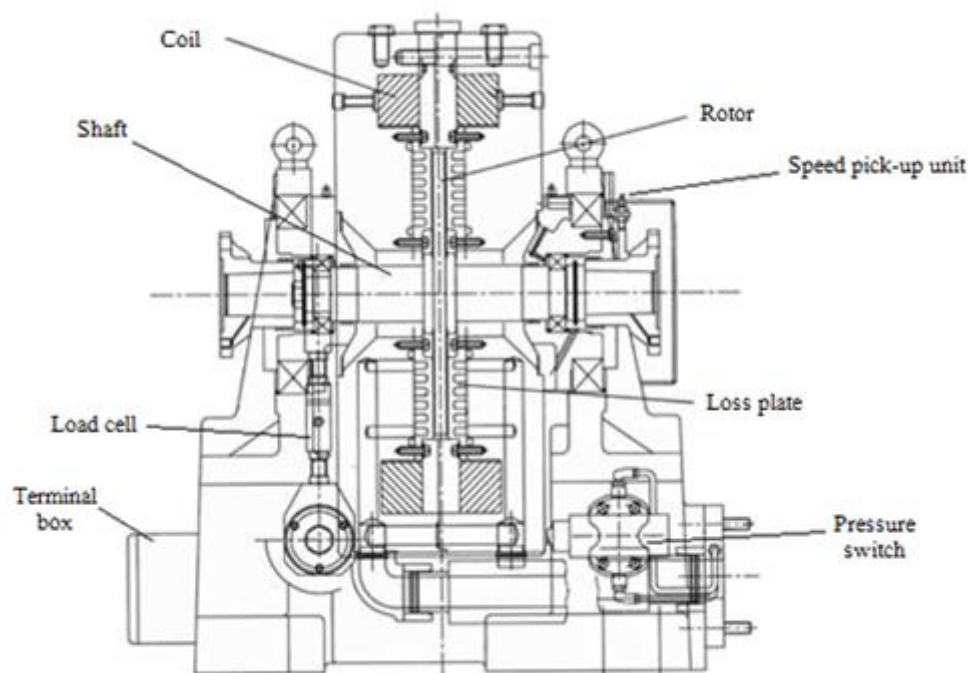


Figure 9 - Eddy current dynamometer (Martyr, 2012)



Alternating Current (AC) Dynamometers

This type of dynamometer converts the power absorbed from the engine and converts it into electrical energy. The load applied to the engine can be controlled by the electrical load applied to the electric circuit connected to the generator. Another feature of these dynamometers is that the generator is also capable of running as an electric motor that can drive the engine unit. This function is useful when measuring the amount of friction resistance present in the engine together with any other losses (Martyr, 2012).

Direct Current (DC) Dynamometers

DC dynamometers are very similar to AC dynamometers, but make use of a DC motor instead of an AC motor. There are a few disadvantages of this type of dynamometer when compared to the AC dynamometer.

Listed below are the features of each of the above-mentioned dynamometers:

Table 1 - Various dynamometer features (Dyne_Systems_Inc., 2014)

Features	Water Brake	Eddy Current	AC	DC
Drive engine	No	Yes	Yes	No
Inertia	Variable	Low	High	Low
Control stability	Good	Excellent	Very good	Low
Speed ability	High	High	Moderate	High
Electricity grid regeneration	No	Yes	Yes	No
Control response	Moderate	Fastest	Fast	Slow
Cooling	High	Low	Low	High
Electrical requirements	Low	High	High	None
Technology	Old	New	Old	Old
Power range	High	Moderate	Moderate	High

There are a number of dynamometers that have not been mentioned as they are not as widely used in the industry.

2.4 Air-Fuel flow

2.4.1 Air flow

An intake manifold aids air flow into each of the cylinders in both petrol and diesel engines. This component is designed to deliver air to each cylinder via separate pipes called runners. The runners must have an adequate inside diameter to maximise the air flow into each cylinder without any restrictions (Pulkrabek, 2004).

Unlike the petrol engine which is controlled by the amount of air that is let into the system, diesel engines take in a full charge of air for each inlet stroke. This is done due to the fact that there needs to be enough air in each cylinder to create the necessary compression temperature needed for the mixture to self-ignite. It is for this reason that depending on the desired output from the engine, the fuel is the variable that is regulated and not the air flow (Judge, 1967). This means that there is no set air-fuel ratio for a specific diesel engine.

As the air-fuel ratio varies, it has an effect on the thermal efficiency of the engine. The graph in Figure 10 below shows the relation between thermal efficiency and the air-fuel ratio used:

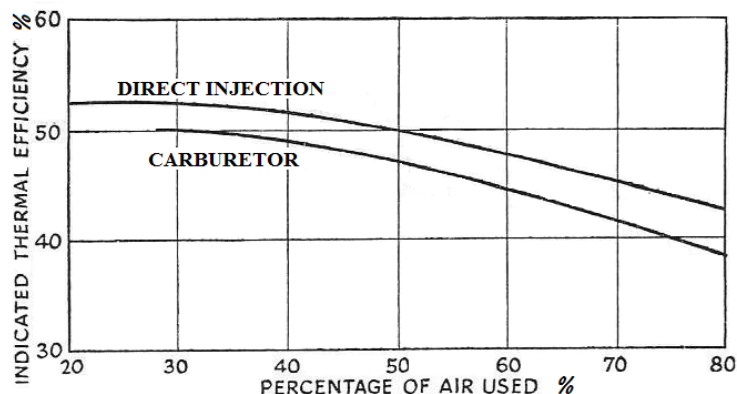


Figure 10 - Thermal efficiency vs. AF ratio (Judge, 1967)

Exhaust gas recirculation (EGR) is an air flow technique used in engines for the purpose of reducing the NO_x emissions. It works by recirculating a small percentage of the exhaust gas (inert to combustion) back into the engine's inlet manifold. By doing this, the exhaust gas replaces a portion of the excess air that a diesel engine always operates on and dilutes the air which is already in the cylinder, thus lowering the temperature by absorbing energy during

combustion (Pulkrabek, 2004). Since NO_x is formed mainly due to O_2 and N_2 being subjected to high temperatures, the reduction in combustion temperature produces less NO_x emissions to the atmosphere with the negative side effect of a loss in the engine's volumetric efficiency (Pulkrabek, 2004) (Chrysler, 1973).

Turbocharging

One of the best methods to increase the output performance of an engine is by turbocharging it. This method produces a significant increase in power and torque without adding large amounts of weight (Pulkrabek, 2004).

It can be seen by looking at the equations for power in section 2.2.2 that an increase in mass air flow into the engine would result in an increase in performance. The principal that turbochargers operate on is by utilising the exhaust gas to create a forced induction system to aid the engine in increasing its performance (Judge, 1967). This is done by routing the exhaust gases through turbine blades that cause it to spin. The exhaust gases then pass through the exhaust pipe and are then released into the atmosphere. The turbine shaft connects directly to compressor blades on the air inlet side of the turbo. The compressor is a type of centrifugal pump which compresses the air that is entering the engine and, therefore, forces a larger volume of air into each cylinder. The increase in air volume entering the cylinders means that additional fuel can be added to the mixture to result in a more powerful combustion of fuel which in turn releases energy (Pulkrabek, 2004).

The compression of a gas such as in the case of turbocharging an engine causes its temperature to rise. To reduce the air temperature flowing into the engine, it is first passed through an intercooler. An intercooler is a heat exchanger that allows the warm compressed inlet air to flow through sealed passages as cool air flows past the cooling fins either with the aid of the engine's fan or from the wind created by a vehicle at speed. The added advantage of an intercooler is that as it cools the air down, it increases its density allowing additional air to enter each cylinder (Judge, 1967).

With the addition of a turbine in the exhaust system, a higher back pressure is created which would result in a slight power loss. This power loss is more than made up with the additional power that the turbo produces. To reduce the pressure in the exhaust pipe, the blades of the turbine and compressor rotate at speeds of up to 130000 RPM (Pulkrabek, 2004).

These speeds create typical boost pressure ratios of 1.5:1, which mean that the air being forced into the cylinders would be slightly over 150 kPa (Judge, 1967).

The additional air into each cylinder is perfect for generating more power and torque, but if the boost pressure rises too high, it could cause “knocking” in the engine which is the pre-detonation of fuel due to high compression temperatures. It is for this reason that a component called a wastegate is used in turbochargers. A wastegate is a valve that operates according to the boost pressure at the inlet manifold. If the pressure rises too much, the valve opens up to allow only a portion of the exhaust gas to pass through the turbine while the remainder is bypassed straight into the exhaust system. By doing this, the boost pressure will be kept from increasing too much (Haywood, 1988). Another device called a blow-off valve is used to reduce excess boost pressure at the inlet manifold in instances such as gear changes or reducing engine speed to prevent pressure surges by releasing some compressed air to the atmosphere (Allard, 1982) (Gorla, 2003) (Garrett Turbos, 2015).

The basic layout of a turbocharged engine is illustrated below in Figure 11.

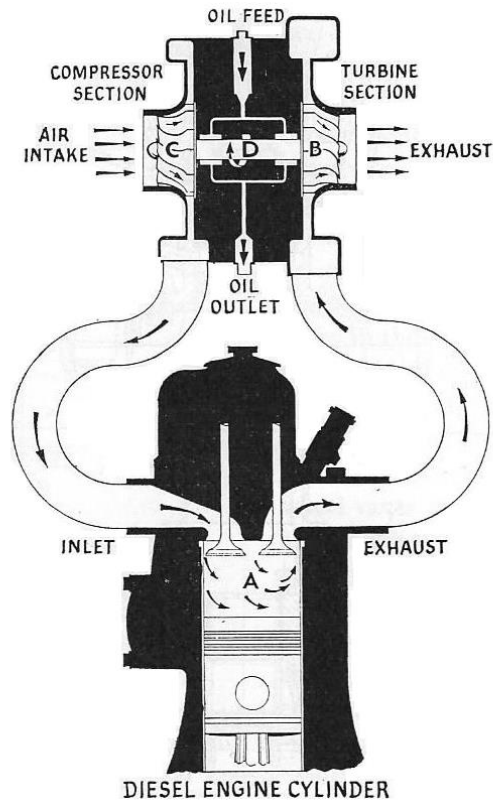


Figure 11 - Layout of a turbocharged internal combustion engine (Judge, 1967)

The graph below illustrates the effect that incorporating a forced induction system such as a turbocharger to an engine has on fuel consumption. As seen from the data in the graph below, there is a clear drop in fuel consumption at just 50% forced induction.

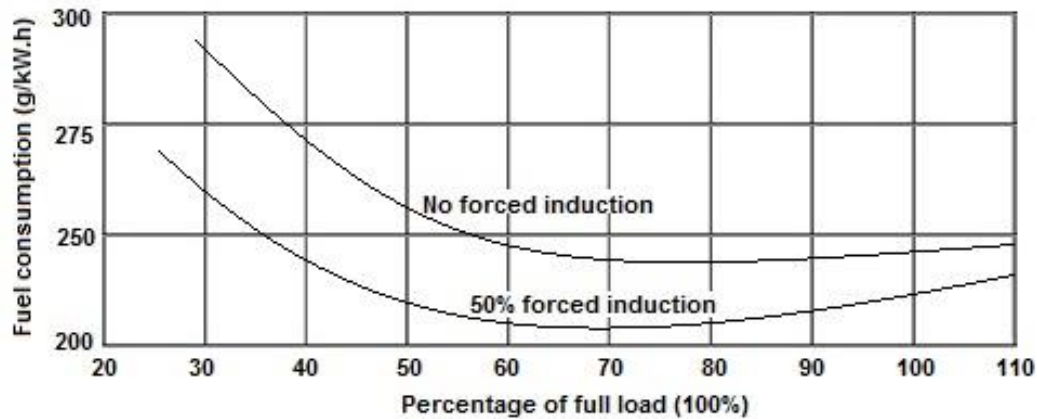


Figure 12 - Effect of forced induction on fuel consumption (Judge, 1967)

2.4.2 Fuel flow

To control the fuel flow in modern day diesel engines, electronically controlled high-pressure common rail systems are used. These systems can improve the overall efficiency of engines due to the fact that parameters such as fuel pressure, timing and quantity can be controlled and altered to provide the optimum performance. This method in turn improves the vehicle's fuel consumption as well as lowering exhaust emissions (Xu-Guang, 2012).

Common rail systems work by using a fuel pump located along the fuel line before the injectors to pressurize the fuel to pressures that can exceed 200 MPa. The same high-pressure line feeds each of the injectors in the fuel system, hence the term "common rail" (Armas, 2011). The engine's ECU usually controls the high-pressure fuel pump. The same ECU typically controls the injectors as well in modern engines which provides a highly accurate method of adding fuel to the compressed air in each cylinder. Together, this system makes it possible for the engine to produce a higher torque at lower engine speeds as well as increasing fuel efficiency and decreasing harmful emissions. Older common rail systems provided injection pressures that were directly proportional to the engine speed. This effect, along with the use of a camshaft method to open and close the injectors would result in less accurate fuel mixing and lower efficiency when compared to ECU controlled systems (Jääskeläinen, 2014).



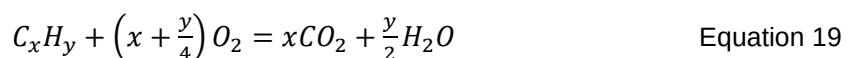
The high pressure that modern common rail systems operate at means that the fuel mixes with the air at a faster rate and is more uniformly distributed, this shortens the ignition delay of the fuel thus improving the engine's performance (Xu-Guang, 2012).

2.5 Emissions

Emissions of an engine are the products of the combustion of the air and fuel that flow through the engine to produce the necessary power. Capturing engine emission data during tests on an internal combustion engine is extremely important as it reveals what is happening inside the combustion chamber. The exhaust emissions can indicate whether the engine is running lean or rich, what the combustion efficiency is, as well as if there are any other problems with the engine such as worn piston rings or leaking gaskets or seals (Martyr, 2012) (Merker, 2012).

It is also important for an engine to produce as little harmful emissions as possible. These harmful emissions pose a threat not only to humans but the environment as well. In many countries, there are emissions standards that vehicles are required to meet. Vehicles also get taxed on the amount of emissions they produce, which makes it extremely vital to be able to reduce the harmful gases as much as possible (Martyr, 2012).

During combustion of fuel in an internal combustion engine, two chemical processes occur. These are the complete and incomplete combustion processes. Complete combustion of fuel produces Nitrogen (N_2), water (H_2O), Carbon Dioxide (CO_2) and Oxygen (O_2) from the following reaction (Judge, 1967):



This formula indicates that C_xH_y amount of fuel needs $\left(x + \frac{y}{4}\right)$ amount of oxygen to produce x amount of carbon dioxide and $\frac{y}{2}$ amount of water (Judge, 1967). All these gases except CO_2 are relatively harmless and naturally form part of the atmospheric air (James Lents). CO_2 is regarded as a greenhouse gas and harmful to the environment.

This gas will always be a by-product of combustion of any hydrocarbon fuel, but can be decreased by increasing the efficiency of the engine or by increasing the combustion efficiency. By doing this, the fuel consumption of the engine would also decrease (James Lents).





Incomplete combustion of the fuel produces harmful gases that need to be decreased as much as possible. These gases include carbon (C) i.e. smoke, carbon monoxide (CO), hydrocarbons (HC) i.e. formed by unburnt fuel, as well as nitric oxide (NO) and nitrogen dioxide (NO₂), together, referred to as NO_x (James Lents).

Diesel engines produce different emissions to petrol engines. The composition of exhaust emissions can vary between engines due to engine design, operating conditions as well as the type of fuel used. The composition of diesel fuel varies greatly around the world, such as in Europe a Sulphur content of ≤10 ppm is required whereas in South Africa 500 ppm is still available (Martyr, 2012) and this sulphur content in diesel fuels produces Sulphur Dioxide (SO₂) in the exhaust gas. Because diesel engines operate with excess air, there are no significant CO emissions as there are in petrol engines, but there are significantly larger amounts of particulate and NO_x emissions produced (Martyr, 2012).

Volatile organic compounds or VOCs are organic chemicals that are produced from the burning of a fuel source such as diesel. VOCs are described as any organic compound that has a low boiling point at standard air pressure (Chemical Manufacturers Association). While VOCs are not directly responsible for any health or environmental issues, they do react with various nitrogen oxides (NO_x). This reaction, in the presence of heat and sunlight, forms ground level ozone which is the main component of smog (European Union, 2004).

Due to the high production of particulate in diesel exhaust emissions an opacity meter is required to work alongside an exhaust gas analyser. An opacity meter or smoke meter works by emitting light from a probe placed in the engine's exhaust pipe. The unit then measures the amount of light blocked by the smoke that passes over the probe as the gas exits the pipe and displays it as a percentage value where 0% refers to no smoke, and 100% is thick smoke (Martyr, 2012). This would then provide an indication of how the engine is performing under the set conditions of the test.

To accurately test the emissions of an engine, "real world" driving would need to be simulated as closely as possible. This method is not a problem for a fully assembled vehicle as it would just need to be connected to an exhaust gas analyser and driven down the street. In the case that an engine has not yet been placed into a vehicle, it would need to be put under a specific load for certain amounts of time to simulate driving conditions according to a driving cycle graph (Martyr, 2012). These driving cycles are usually drawn up by the local

environmental protection agency to simulate driving conditions such as urban or highway driving (Emissions testing, 2013). Many of these driving cycles are divided into three sections namely, cold start phase, transient phase and hot start phase. Emissions in each of these phases are recorded separately as the engine would perform differently for each section (Martyr, 2012).

Below is an example of an urban driving cycle:

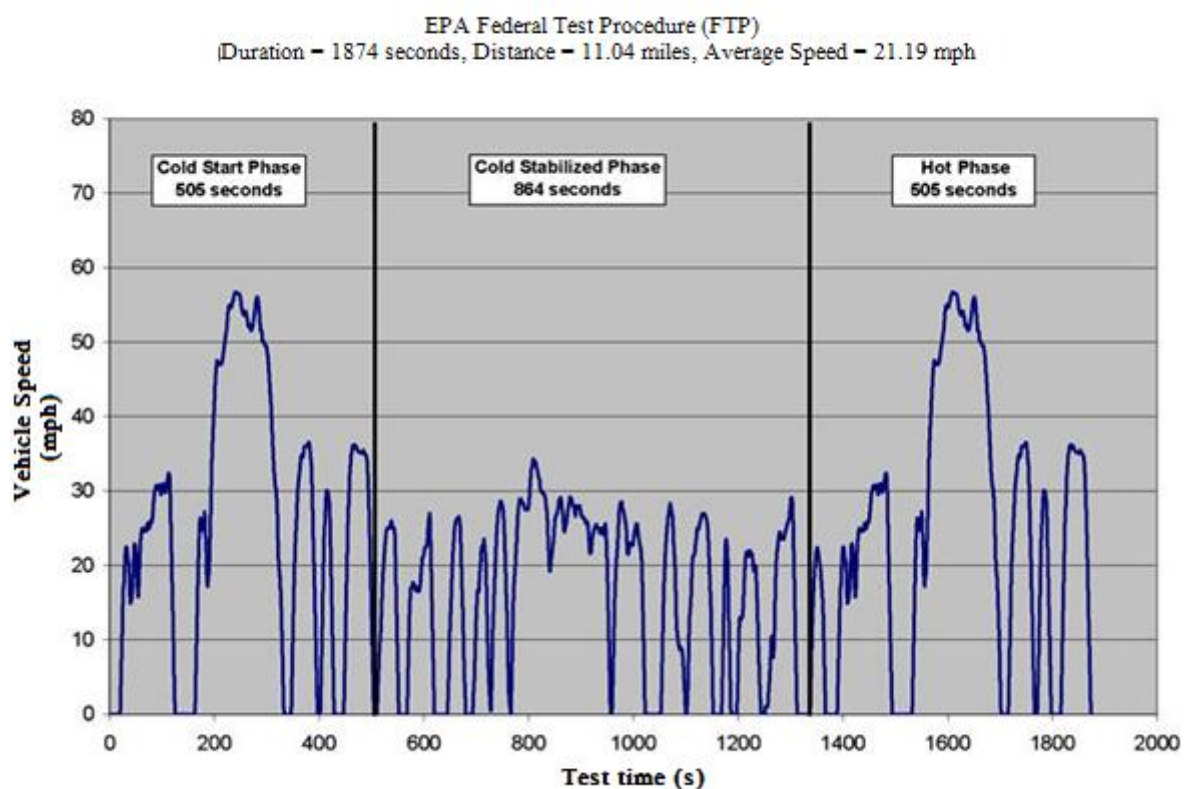


Figure 13 - Driving cycle for emissions testing (Emissions testing, 2013)

Large vehicle testing facilities determine the emission values for vehicles typically by using a Constant Volume Sampling system (CVS). This method works by pulling in a constant volume of clean air that mixes with the exhaust gas. The volume of clean air needs to be larger than that of the exhaust gas. By pulling in a large volume of clean air, the exhaust gases from the vehicle are cooled down enough to be handled by the measuring equipment, it also has the advantage of simulating more realistic atmospheric conditions as the exhaust gas from any vehicle is emitted into the atmosphere and reacts with the air.



The disadvantage of using this method is that as modern vehicles become “cleaner” and emit less harmful gases, the addition of the clean air dilutes the exhaust gas to a point where instrument accuracy could cause errors in the measurements (James Lents) (Martyr, 2012).

For the purpose of this project, the CVS method will not be used due to the costs of such a facility. Instead, a five gas analyser along with a smoke (opacity) meter will be used to measure exhaust emissions during all experimental tests. This gas analyser is capable of measuring all the gases necessary for this study, i.e.: CO, CO₂, O₂, HC and NO_x.

2.6 Cooling system

The main purpose of an internal combustion engine is to convert chemical energy into thermal energy to create mechanical power. This process creates temperatures inside the combustion chamber that can exceed 2400°C. The materials used to construct the engine and its components are not able to handle these temperatures for prolonged periods of time and need to be cooled down. On the other hand, for the best thermal efficiency, the engine needs to operate at the highest possible temperature (Pulkrabek, 2004).

This is where the engine’s cooling system comes into play, by reducing the high temperatures that the materials are not able to handle, but also controlling the operating temperature for the best possible thermal efficiency (Pulkrabek, 2004).

Depending on the type of engine, only about 30-40% of the energy in the fuel is converted into useful shaft work, approximately 30% is lost to exhaust heat and the remainder, approximately one third, is removed by the engine’s cooling system (Pulkrabek, 2004).

The energy converted from the fuel is distributed through the engine as follows (Pulkrabek, 2004):

$$\text{Power generated} = \dot{W}_{\text{shaft}} + \dot{Q}_{\text{exhaust}} + \dot{Q}_{\text{loss}} + \dot{W}_{\text{ancil}} \quad \text{Equation 20}$$

where: $\dot{W}_{\text{shaft}}$ - brake power output from the crankshaft
 $\dot{Q}_{\text{exhaust}}$ - energy lost to exhaust heat
 $\dot{Q}_{\text{loss}}$ - all energy lost to heat transfer (see Equation 21 below)
 $\dot{W}_{\text{ancil}}$ - power lost to run engine ancillaries

The energy losses in heat transfer can be divided up as follows (Pulkrabek, 2004):

$$\dot{Q}_{\text{loss}} = \dot{Q}_{\text{cooling}} + \dot{Q}_{\text{oil}} + \dot{Q}_{\text{ambient}} \quad \text{Equation 21}$$

where: $\dot{Q}_{\text{cooling}}$ - energy lost to cooling system
 $\dot{Q}_{\text{oil}}$ - energy lost to engine oil
 $\dot{Q}_{\text{ambient}}$ - energy lost to surrounding atmosphere through radiation

There are two main types of cooling systems used by vehicle engines, namely, water cooling and air cooling. The majority of vehicles have a water cooling system as it is more effective than the air cooled systems.

2.6.1 Air cooled systems

Air-cooled systems are not generally used in motor vehicles as they are not as efficient as the water cooled systems. The main use of these systems is to cool smaller engines such as lawn mowers, model cars and planes and chainsaws to save costs. Many motorcycles and airplanes make use of this system as well to save weight (Pulkrabek, 2004).

Air cooling systems are much simpler than water cooled systems as they only rely on the air flow across external fins on the engine's surfaces. These fins are used for the purpose of increasing the total surface area of the engine that is in contact with air (Pulkrabek, 2004). Figure 14 below is an example of an air cooled engine.

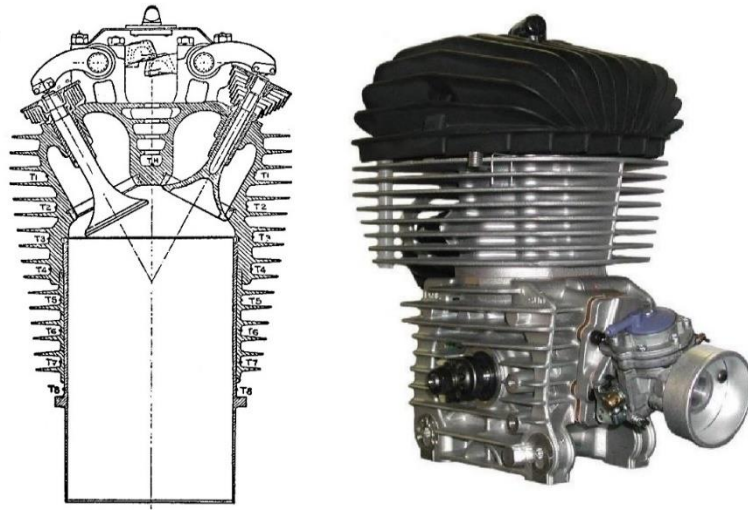


Figure 14 - Air cooled engine (Genevro, 2014)

2.6.2 Water cooled systems

The water cooling systems operate by circulating and cooling a water/antifreeze mixture through the engine. Water is an excellent coolant as its specific heat capacity is approximately four times greater than air which means that it can remove a larger amount of heat than air is capable of (Pulkrabek, 2004).

Engine cooling systems make use of water pumps to circulate the cooling water. The coolant begins by being pulled into the engine by the water pump. The coolant enters at the bottom of the engine and exits at the top for the reason that water vapour rises in water, and if vapour bubbles were to form in the engine block, they would be able to be carried out by the flow of coolant as it is in an upward direction. Once the coolant has entered the engine block, it flows into cavities surrounding the cylinders called water jackets, which allows for much better removal of unwanted heat. The coolant then flows through the cylinder head and out the engine at the top, past the thermostat and into the top of the radiator heat exchanger. The thermostat is a temperature sensitive valve that controls the flow of the coolant through the cooling system. When the engine is below the optimum operating temperature, the thermostat remains closed and prevents the coolant from passing through the radiator and cooling the engine even further. As the engine warms up, so does the thermostat. As this happens, the thermostat valve begins to open up allowing larger amounts of water to flow past the radiator until it is fully open, typically at 83°C (Pulkrabek, 2004).

The radiator is designed to be a fin and tube heat exchanger. It makes use of the air flowing past the fins, either by the movement of the vehicle or by the engine fan, to cool the hot water from the engine (Incopera, 2007)

Water is a good standalone coolant, but certain properties of water such as its freezing point, boiling point and the fact that it oxidises metal are undesirable. These properties are altered by adding ethylene glycol also known as antifreeze. Antifreeze decreases the freezing point and raises the boiling point of water as well as providing protection against rust formation in the engine block (Pulkrabek, 2004).

2.7 Generators

Generators are important pieces of equipment to have and can be used for many applications that require power generation.

A generator is a device that converts electrical energy from a mechanical power source such as an internal combustion engine. Generators work on the principle of magnetic induction which means that electric charges can be produced by moving a cable or wire through a magnetic field. This movement through the magnetic field creates a potential difference between the opposite ends of the wire which results in the electric charges to flow through the wire thus generating a current (Diesel Supply and Service, 2013). The basic workings of a generator are shown in Figure 15 below:

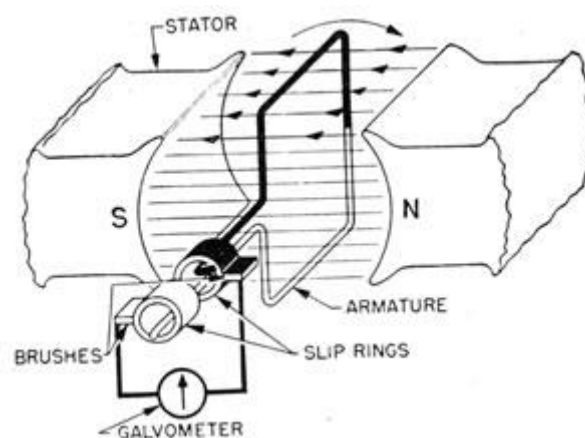


Figure 15 - Operating principal of a generator (Historic Naval Ships Association, 2007)

A typical generator consists of a number of components. The figure below illustrates each of the components that form part of the generator system.

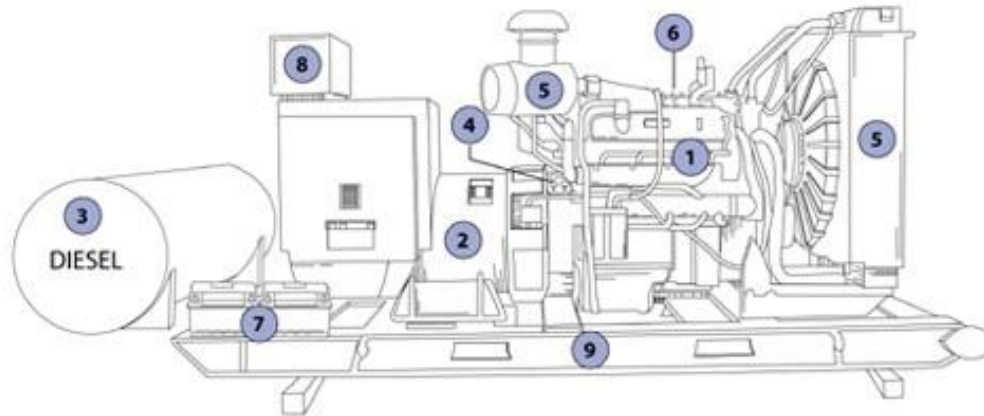


Figure 16 - Generator components (Diesel Supply and Service, 2013)

The numbered components from Figure 16 above are:

1. Engine
2. Alternator
3. Fuel tank
4. Voltage regulator
5. Cooling and exhaust systems
6. Lubricating system
7. Battery charger
8. Control panel
9. Main assembly/frame (Diesel Supply and Service, 2013)

2.8 Fuel additives

According to the South African Petroleum Industry Association (SAPIA), the consumption of petrol and diesel in South Africa has been increasing each year. This increase has a large impact on air pollution and global warming. In addition to this, the price of these fuels has also been on the rise over the years (SAPIA, 2013). The images below illustrate the increase in the South African fuel consumption and fuel prices respectively over the past 25 years.

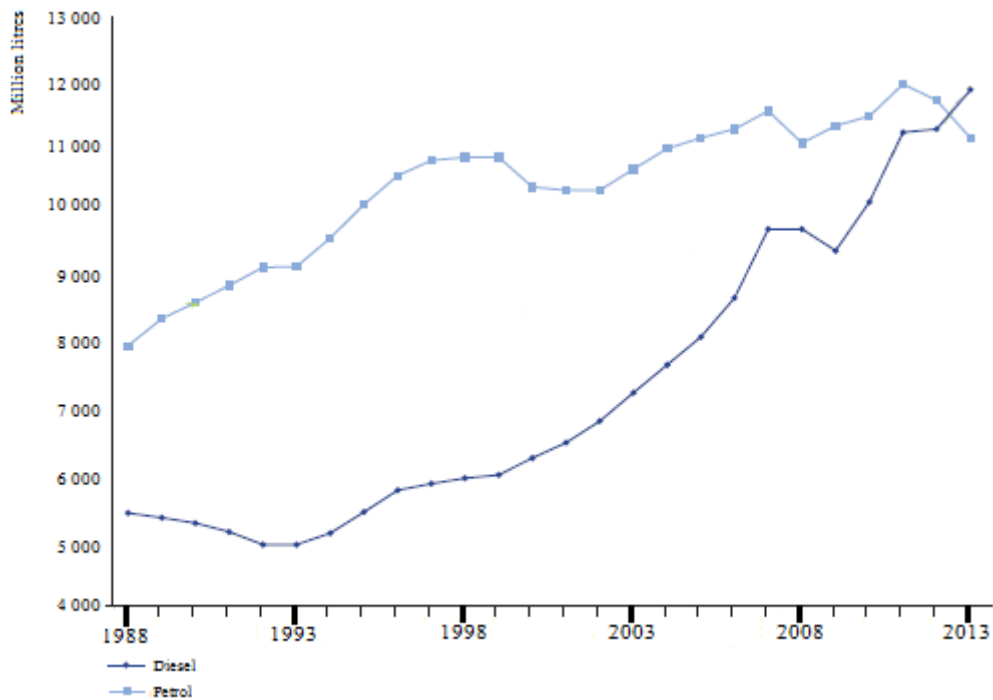


Figure 17 - Fuel consumption in South Africa over last 25 years (SAPIA, 2013)

Price trends

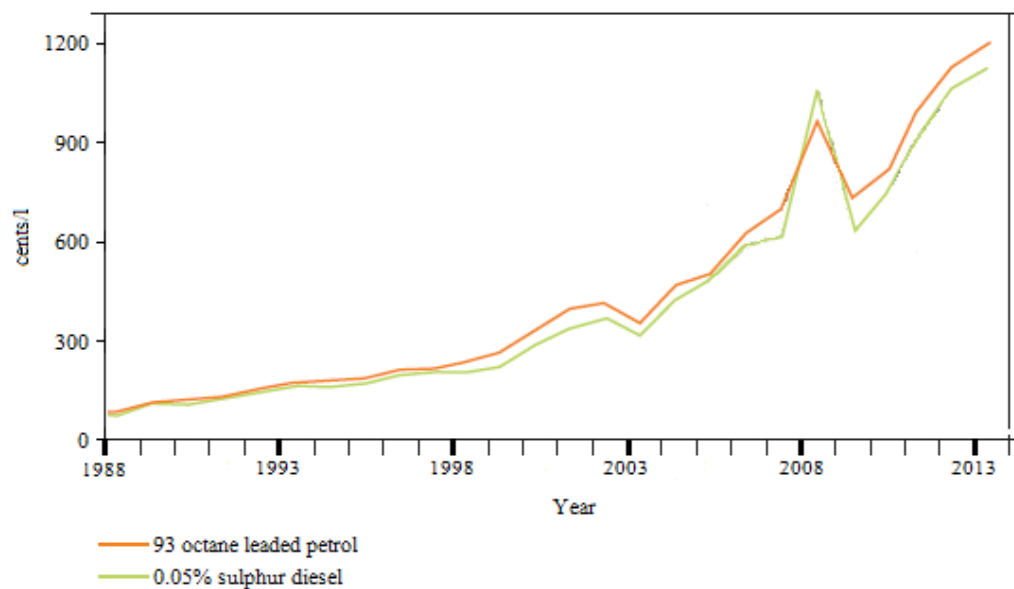


Figure 18 - Price of fuel in South Africa over last 25 years (SAPIA, 2013)

A solution that could see a decrease in the negative effects of internal combustion engines as well as save money on fuel is to use certain additives in combination with the fuel.

Fuel additives have the general purpose of making an engine run more efficiently by improving or altering one or more of the properties of the fuel. These benefits could be improved engine lubrication, better combustion efficiency or boost the fuel's octane or cetane number. One example of this was the lead compounds added to petrol for the purpose of lubricating the valves and reducing knocking in internal combustion engines. The lead additives were later restricted in fuels as they were causing harmful effects towards human health (Kasper, 1999).

2.8.1 Naf-Tech

Naf-Tech is the fuel additive that will be under observation for this study and will be tested for its performance and efficiency properties in internal combustion compression ignition engines.



Figure 19 - Naf-Tech Energy (Naf-Tech-Energy, 2014)

Engineers have always assumed that the reason for black soot in diesel exhaust was from unburnt fuel particles reaching the cylinder walls where they would cool down and enter the exhaust system. Recent studies at the Combustion Research Facility (CRF) have shown that this is not the case. What actually occurs is that fuel and air do not completely mix with each other inside the cylinders (Czerwinski, 2011). The reason being that standalone diesel has a very high surface tension that causes incomplete vaporisation of the fuel. This, in turn, creates a loss in thermal efficiency for the engine and, therefore, a higher fuel consumption and lowered performance output. With the addition of Naf-Tech in the fuel, it lowers the surface tension of the diesel which greatly improves the vaporisation. This makes the combustion process more efficient and would increase the engines thermal efficiency. (Naf-Tech-Energy, 2014). The visual effects that Naf-Tech has on diesel combustion can be seen in Figure 20 below. The improved vaporisation of fuel results in a more efficient and complete combustion (Naf-Tech-Energy, 2014).

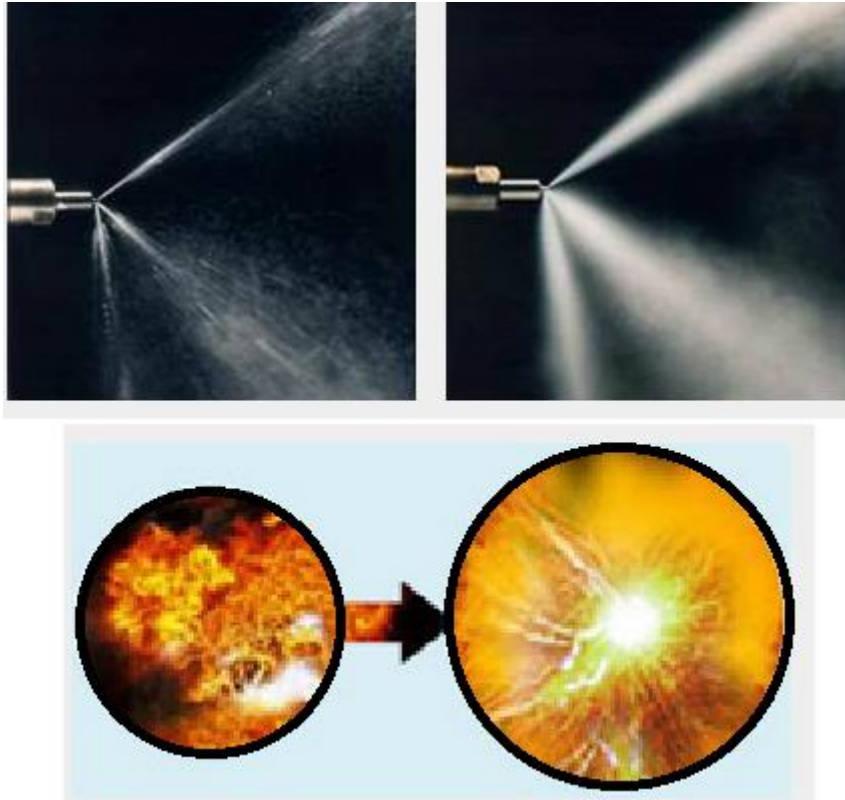


Figure 20 - Naf-Tech effect on injector spray (Naf-Tech-Energy, 2014)

The table below compares some of the various fuel additives available on the market including Naf-Tech. It describes the benefits of each additive, where they originate from as well as the approximate time for availability. From this table, it can be seen how Naf-Tech compares to the other fuel additives on the market.

Table 2 - Fuel additive comparison (Corporation, 2014), (Naf-Tech-Energy, 2014), (Neptune Products), (Blue Sea Biotech)

Fuel Additive	Super21	Naf-Tech	FTC Catalytic Engine Decarbonizer	Dragon Fire 888	SuperSaver Diesel fuel enhancer D106
Manufacturer	Eniva Corporation	Naf-Tech Energy	Neptune Products	Green Star Group	Blue Sea BioTech
Description	Petrol/Diesel combustion enhancer	Combustion enhancer	Petrol/Diesel Engine Decarboniser	Fuel combustion enhancer	Diesel combustion enhancer
Country of origin	USA	South Africa	New Zealand	USA	Australia
Availability	> 1 month	± 4 days	> 1 month	> 1 month	> 1 month
Benefits	Cleans engine Increases power Reduces toxic emissions	Improves combustion and thermal efficiency Increases power Improves fuel usage Engine runs cooler	Cleans engine Improves fuel economy Reduces greenhouse gases	Reduce harmful gases Improves fuel consumption	Improves combustion efficiency
Claimed savings	25% increase fuel economy 10% increase in power	Up to 21.43% fuel savings	Improves fuel consumption by 6-8%	Improves fuel consumption by 7.5%	Saves 6-20% fuel

2.8.2 Testing of fuel additives

Before any product, including fuel additives, can be sold on the market, they need to be tested to determine what their benefits would be and to ensure that there are no negative effects of using the product.

Fuel additives are generally tested in a very similar manner to how the engines themselves are tested i.e. on a dynamometer, but with the addition to the test procedure of a number of runs with the fuel additive under different conditions (Muncrief, 2009) (Kalam, 2008).



A few examples were looked at to get an idea of what procedures were followed for different fuel additive tests.

One example of a testing method for fuel additives comes from (Muncrief, 2009). In this method, a computer-controlled electric generator is used instead of a dynamometer. The first step done was to run the engine for 20 minutes on the base diesel fuel. This would establish the baseline results. Thereafter the additive is accurately added to the fuel at specified concentrations via a dosing pump. The tests ran for 20 minutes each time the concentration was changed. During each of the tests the vital data is monitored and logged. Data such as the engines performance and exhaust emissions are important as they will indicate if the additive has the desired effect on the engine.

Another example of additive testing from (Kalam, 2008) also makes use of a computer-controlled dynamometer. In this test procedure, the engine is run from 1000-4000 RPM for the base fuel and then repeated with the additive included. For each of these tests, the engine is run at 50% throttle to measure the power as well as specific fuel consumption. Thereafter the dynamometer is set to a constant 50 Nm of torque and the engine to 2250 RPM to measure the exhaust emissions.

The above examples of testing methods can be used to determine the effects that fuel additives have on internal combustion engines and their performance. However, fuel additives are designed to alter different properties of engine performance such as combustion efficiency, wear properties, cetane numbers, etc. (Kasper, 1999). For it to be possible to determine the effects that a certain additive has on some of these properties, specialised tests are needed.

In the case of this project, the fuel additive that is being tested is a combustion enhancer. This means that the combustion characteristics of diesel would need to be accurately monitored with and without the addition of the additive. For this to be accomplished, tests were conducted at the CSIR on a diesel burner where the temperatures of the burning diesel were carefully monitored both with base diesel as well as a mixture of diesel and the fuel additive.

The basic procedures used in the examples above will also be used to gather the results of this study. Due to time and money constraints, a computer-controlled dynamometer is not a



viable option for this project. The load on the dynamometer that is required will need to be controlled manually but will be able to provide the accuracy that is necessary.

2.9 Energy in fuel

The amount of energy that a fuel contains is extremely important as it affects the performance outputs of an engine. By knowing how much energy the fuel contains it is possible to determine how efficient an engine is at converting that energy into useful work. The amount of energy contained in diesel or petrol is usually referred to as the gross calorific value (GCV_v) or net calorific value (NCV_p). (Bech, 2009).

The GCV_v or higher heating value of a fuel is defined as the amount of heat that is released from the combustion of a specific quantity of fuel in pure oxygen while taking the latent heat of vaporization of water into account. The GCV_v of a fuel is determined by using a bomb calorimeter which is done by placing a precise amount of fuel into a sealed combustion chamber that is filled with oxygen. The combustion chamber is submerged in water of a known temperature (usually 25°C). The fuel sample is then ignited, and the temperature of the water changes. These temperature variations are measured and recorded to determine how much heat was released. The water surrounding the combustion chamber is allowed to return to its original temperature for the latent heat of vaporization of water to be taken into account. The NCV_p or lower heating value of a fuel is defined as the amount of heat released from the combustion of a specific quantity of fuel in oxygen. In this case, the latent heat of vaporization of water is not taken into account and is for this reason that the GCV_v is always higher than the NCV_p of a fuel (Furey, 2013) (Bech, 2009).

Typical net and gross calorific values for diesel are 42.612 [MJ/kg] and 45.575 [MJ/kg] respectively. These values can differ slightly from different regions around the world and between various manufacturers (US Department of energy, 2011).

Chapter 3: Mathematical Simulation Detail Design

Chapter 3 will discuss the method followed to create the EES (Engineering Equation Solver) mathematical model of the internal combustion engine test bench as well as the Excel model that is used to verify the EES model. The model was created to simulate one full revolution in the Diesel cycle with all the specifications for the specific engine being tested. The conditions used in the models are the same as those used for the actual test bench. See Appendix A1 for the EES code.

By simulating one revolution for one cylinder of a diesel engine, it is possible to obtain all the necessary data to compare to the actual test results. Data collected from the technical service manual for the engine was used for the models input values. Any information that was not included in the manual was obtained through tests and measurements conducted on the engine. These input values need to be as accurate as possible to ensure that the results from the model are comparable to actual test data. The input data used for the mathematical model can be found in Appendix A2.

The mathematical model was created to simulate one full revolution of the Diesel cycle. This is done by determining the conditions and properties of the working fluid at each component of the intake and exhaust systems as well as at each of the four strokes in the cylinder of the engine.

For this to be done, properties such as temperature, pressure, volume, density and mass must be known at each point to ensure the greatest possible accuracy for the simulated cycle. The formulae used to calculate these values are explained below.

For the air intake of the engine or point 1, the temperature and pressure is measured from the ambient conditions. The volume of air used is gained from working out the displacement capacity of a single cylinder, and the density of the air is obtained from a lookup table that was verified by the built in EES function to calculate air density. From these, the mass of the charge air can be determined by multiplying volume by air density.

Due to the loss of pressure caused by the air cleaner, the properties at point 2 are recalculated in the same manner as before but with the lower pressure.

To determine the air properties after it has passed through the turbocharger, additional calculations are needed. Measurements are taken during the tests conducted on the engine at various conditions to determine the pressure created by the turbo's compressor.

These measurements are then inserted into lookup tables for the model to read. The model will select the pressure value based on the input values for that specific run. The efficiency of the turbo will also be determined using temperatures measured from the engine tests. These values will be inserted into the following formulae to determine the air properties at that point.

$$\frac{T_{3 \text{ isentropic}}}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma_{air}-1}{\gamma_{air}}} \quad \text{Equation 22}$$

$$\eta_{turbo, \text{ isentropic}} = \frac{T_{3 \text{ isentropic}} - T_2}{T_3 - T_2} \quad \text{Equation 23}$$

Once the air has passed the turbo compressor, it reaches the intercooler. At this point, the temperatures before and after are measured during the tests to determine the intercooler efficiency. This efficiency will be used to theoretically calculate the parameters for the air at this point. Intercooler efficiency is calculated by using the formula shown below.

$$\eta_{intercooler} = \frac{T_3 - T_4}{T_3 - T_{ambient}} \quad \text{Equation 24}$$

At the inlet manifold, the charge air experiences another pressure drop due to the geometries and design of the manifold itself as well as the inlet valves.

When the piston is at TDC, there remains a small volume that is occupied by exhaust gas after the exhaust stroke. This remaining gas from the previous stroke mixes with the clean air being forced into the cylinder during the inlet stroke. An initial guess value needs to be chosen to determine the properties of the remaining gas in the cylinder. An iterative process will then determine the values for the following points in the cycle. The pressure of this gas is determined by adding the exhaust and turbo back pressure to the ambient pressure. The volume used at this point is the combustion chambers volume.

By using the same gas from the previous point, the values must be recalculated with the piston at BDC, i.e. just after the inlet stroke. This means that the volume used at this point will include the piston displacement as well as the combustion chamber volume. The properties of the charge air will also need to be recalculated for this new volume.



The charge air and the residual gas are mixed together inside the cylinder which means that the mass and the pressure of the two gases are added together to create the new conditions in the cylinder.

The temperature of the new gas mixture is gained as follows:

$$T_9 \times m_9 = T_7 \times m_7 + T_8 \times m_8 \quad \text{Equation 25}$$

where:

- T_9 - Temperature of air/gas mixture in the cylinder
- m_9 - Mass of air/gas mixture in cylinder
- T_7 - Temperature of residual gas in cylinder
- m_7 - Mass of residual gas in cylinder
- T_8 - Temperature of charge air in cylinder
- m_8 - Mass of charge air in cylinder

To simulate the compression stroke, the isentropic temperature first needs to be determined by using the air/gas mixture properties:

$$\frac{T_{10 \text{ isentropic}}}{T_9} = (r_v)^{\gamma_{mixture} - 1} \quad \text{Equation 26}$$

$$\eta_{compression, isentropic} = \frac{T_{10, isentropic} - T_9}{T_{10} - T_9} \quad \text{Equation 27}$$

where:

- $T_{10 \text{ isentropic}}$ - Isentropic temperature after compression
- T_9 - Air/ gas mixture temperature before compression
- r_v - Compression ratio
- γ - Specific heat ratio for air/ gas mixture
- $\eta_{compression}^*$ - Isentropic compression efficiency
- T_{10} - Temperature of air/ gas mixture after compression

* Isentropic compression efficiency values range from 80-90%. It is closer to 80% when piston ring friction losses are included in this value and closer to 90% when piston ring friction losses are added as a separate (mechanical losses) value in the calculations.



Dual cycle diesel engines have two stages after the compression stroke i.e. a constant volume stage and a constant pressure stage. These are simulated by using the pressure ratio (α) and the cut-off ratio (β), where the pressure ratio is the increase in pressure during combustion, and the cut-off ratio is the change in volume that occurs during combustion (Pulkrabek, 2004).

To calculate the pressure for the constant volume stage, the pressure before combustion is multiplied by the pressure ratio. The mass of the gas inside the cylinder at this point is the mass at the previous point with the addition of the fuel.

The constant pressure stage is simulated in a similar manner to the constant volume stage. The volume for this point obtained from the product of the volume at the constant volume stage and the cut-off ratio.

During the power stroke of the engine, the gases in the cylinder expand which cause the piston to move to BDC. This means that to calculate the temperature of the gas after combustion, the isentropic expansion efficiency is needed. As is the case with the isentropic compression efficiency, the expansion efficiency ranges between 80-90% where 80% includes the friction losses from the piston rings in the value and 90% neglects the friction losses. Shown in the formulae below is the method to obtain the temperature after combustion:

$$\frac{T_{13 \text{ isentropic}}}{T_{12}} = \left(\frac{1}{r_v}\right)^{\gamma_{gas}-1} \quad \text{Equation 28}$$

$$\eta_{\text{expansion, isentropic}} = \frac{T_{12} - T_{13}}{T_{12} - T_{13, \text{ isentropic}}} \quad \text{Equation 29}$$

The pressure after combustion can be calculated as follows:

$$\frac{P_{13}}{P_{12}} = \left(\frac{1}{r_v}\right)^{\gamma_{gas}} \quad \text{Equation 30}$$

where:	$T_{13, \text{ isentropic}}$	- Isentropic temperature of gas after power stroke
	T_{12}	- Temperature of gas before power stroke
	γ	- Specific heat ratio for gas
	$\eta_{\text{expansion}}$	- Isentropic expansion efficiency
	T_{13}	- Temperature of gas after power stroke



The volume at this point will be the entire cylinder volume, i.e. at BDC.

The blowdown phenomenon occurs during the power stroke as the exhaust valve begins to open and exhaust gas escapes past the valve before the piston has reached BDC.

Because the gases in the cylinder are open to the exhaust system, the pressure is as follows:

$$P = P_{ambient} + \Delta P_{exhaust} + \Delta P_{turbo \text{ back pressure}} \quad \text{Equation 31}$$

By first obtaining the isentropic temperature, it is possible to calculate the blowdown gas temperature at this point:

$$\frac{T_{14 \text{ isentropic}}}{T_{13}} = \left(\frac{P_{14}}{P_{13}}\right)^{\frac{\gamma_{gas}-1}{\gamma_{gas}}} \quad \text{Equation 32}$$

$$T_{14} = -((T_{13} - T_{14, \text{ isentropic}} - T_{13})) \quad \text{Equation 33}$$

Due to the exhaust valve being slightly open at this point, the volume needs to be calculated.

$$\frac{P_{13}}{P_{14}} = \left(\frac{V_{14}}{V_{13}}\right)^{\gamma_{gas}} \quad \text{Equation 34}$$

The final point to complete the Diesel cycle simulation is after the exhaust stroke at TDC. At this point, the temperature and pressure are the same as the values determined from the previous step. The volume at this point is equal to the combustion chamber volume as the piston is at TDC.

The following calculations are needed to determine the necessary parameters from the simulated cycle.

$$W_{in} = \frac{P_{compression} \times Vol_{compression} - P_{intake} \times Vol_{intake}}{(\gamma_{air/gas} - 1) \times m_{intake}} \quad \text{Equation 35}$$

$$W_{out} = \frac{P_{combustion} \times Vol_{combustion} - P_{power} \times Vol_{power}}{(\gamma_{gas} - 1) \times m_{power}} \quad \text{Equation 36}$$

$$W_{net} = W_{out} - W_{in} \quad \text{Equation 37}$$



The net work done by the engine along with the specific energy that enters the engine through the fuel are used to determine the thermal efficiency produced by the engine. This is important as it will indicate how efficient the engine is at converting the fuel's energy into useful mechanical power.

$$q_{in, total} = \frac{GCV \times 1000}{AF_{ratio}} \quad \text{Equation 38}$$

$$\eta_{thermal} = \frac{w_{net}}{q_{in, total}} \times 100 \quad \text{Equation 39}$$

It is now possible to calculate the power and torque output of the engine with the aid of the following formulae:

$$P_{power} = w_{net} \times \dot{m}_{air} \quad \text{Equation 40}$$

$$\tau = \frac{P_{power} \times 60}{2 \times \pi \times RPM} \times 1000 \quad \text{Equation 41}$$

Energy losses play a large role in internal combustion engines.

Energy lost through heat in the exhaust system accounts for approximately one-third of the total energy input.

$$q_{out, exhaust} = c_{v, blowdown} \times (T_{power} - T_{ambient}) \quad \text{Equation 42}$$

The second major energy loss in an internal combustion engine is the heat that the cooling system needs to extract. This loss is also usually one-third of the total energy available.

$$q_{out, cooling} = c_{p, water} \times (T_{cool, out} - T_{cool, in}) \quad \text{Equation 43}$$

Additional losses are all the ancillary components such as the alternator and cooling fans, as well as the radiated heat through the engine walls to the atmosphere.

To obtain the combustion efficiency of the engine, the exhaust emissions need to be studied as they are a direct indication of what happens during the combustion process in the cylinders.



The mathematical model uses data from a lookup table obtained from the tests conducted on the engine to calculate the combustion efficiency as follows.

$$\eta_{combustion} = \left(\left(1 - \frac{CO}{CO+CO_2} \right) \times \left(\frac{1-H_x C_y}{1} \right) \right) \quad \text{Equation 44}$$

For this mathematical model to more accurately represent the GWM engine under observation, it needs to incorporate a simulated EGR system. This was done by determining the percentage of exhaust gas returned to the cylinders during the tests.

The EGR valve determines the amount of exhaust gas recirculated. The engine conditions such as speed or load applied, determines the valve position at that specific moment. The position of the valve at each test condition was obtained from an ECU reader and inserted into a lookup table in the mathematical model.

The volume of exhaust gas entering the inlet manifold is used to occupy space in each cylinder and replace a percentage of the charge air. Because this gas is nonreactive, the volume of exhaust gas in each cylinder is subtracted from the displacement volume to decrease the amount of combustible gas present in the cylinders. The mass of the recirculated gas is also added to the total mass of the charge air already inside the cylinder while the mass of the clean air that had been displaced was subtracted.

The Excel model that is used to verify the EES simulation model is set up using the same method described in the above steps.



Chapter 4: Experimental Setup

Chapter 4 describes each of the steps that have been fulfilled to obtain a fully functioning engine test bench facility to complete this study. These include the installation of the GWM engine, instrumentation and calibration thereof as well as the modifications needed to increase the accuracy of measurements for all the necessary parameters. A full explanation of the testing procedure that was designed to achieve all the required results, along with the safety measures and guards necessary for testing will be described in this section as well.

4.1 Engine test bench setup

Once the engine had been installed and correctly aligned on the dyno frame, it was connected to the dynamometer via a prop shaft connected directly to its flywheel. A new exhaust system was designed and built to fit the positioning of the engine on the stand to safely extract the exhaust gases outside the building.

Figure 21 below illustrates a schematic representation of the test bench setup. This image shows the engine as well as the water brake dynamometer to which it is connected. The air flow and fuel delivery systems are also indicated in the image along with the data acquisition unit which will be described later in this chapter.

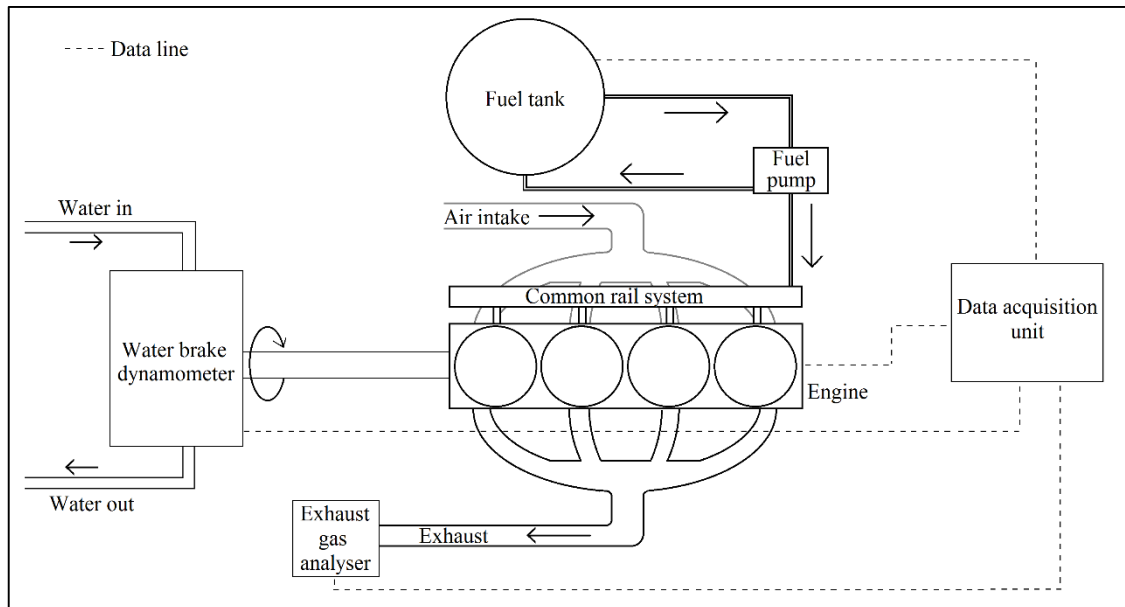


Figure 21 - Experimental setup



Figure 22 - Engine test bench

It is important to know what happens to the available energy from the fuel during and after the combustion process. For this to be possible, the necessary instrumentation needs to be installed in the appropriate positions. The following parameters need to be measured for the duration of each test to be conducted on the engine:



- Engine cooling water in temperature
- Engine cooling water out temperature
- Engine cooling water mass flow
- Dynamometer water in temperature
- Dynamometer water out temperature
- Dynamometer water mass flow
- Air temperature going into engine
- Air temperature exiting engine
- Air mass flow through engine
- Air pressure at engine inlet manifold
- Air pressure at engine exhaust manifold
- Air temperature before turbo intercooler
- Ambient air temperature
- Load applied to engine
- Fuel weight
- Engine speed
- Emission analysis which includes
 - CO
 - CO₂
 - O₂
 - NO_x
 - HC

By analysing the data from the above-mentioned measurements, it is possible to determine the parameters that form part of the scope of this study such as the fuel consumption, power/torque output, combustion efficiency and thermal efficiency.

The next step was to install all the instrumentation to measure the above parameters of the system. By knowing what parameters need to be measured, installing the necessary instrumentation could be planned, so nothing would be neglected. Carefully drilled holes were made for each of the thermocouples that needed to be inserted into the engine. Care was taken not to let any metal filings fall into the engine. An example of a thermocouple inserted into the engine is shown below.



Figure 23 - Exhaust thermocouple in manifold

At the end of 2014 the first set of tests was performed on the test bench for the purpose of identifying any parameters that would still need to be measured and to sort out any problems. This minimises the possibility of any issues occurring during testing at a later stage. One of the main concerns from the tests was that the current cooling system was not able to keep the engine at a constant operating temperature let alone from overheating.

To overcome this issue, a large turbofan, shown in Figure 24 below was acquired and a stand built for it to force air directly through the engine's radiator. This fan is capable of producing a wind speed of approximately 80 km/h. The addition of the fan had a large improvement in the operating temperature but was still not sufficient for the duration of tests required. To further improve the cooling capacity, a second radiator was installed in the system. Once again, the addition of the radiator had a great improvement in the operating temperature of the engine, and from additional tests it was found that the engine was able to operate for the full test duration without the temperature continuously increasing.



Figure 24 - Cooling fan for radiator

At both the inlet and outlet for the air system in the engine, pressure transducers needed to be installed. Specifically at the exhaust, the air needed to be cooled down to 100°C before the transducer could safely handle it for accurate readings. The temperature reduction was accomplished by creating a coil and cooling fins for the unit as indicated below.

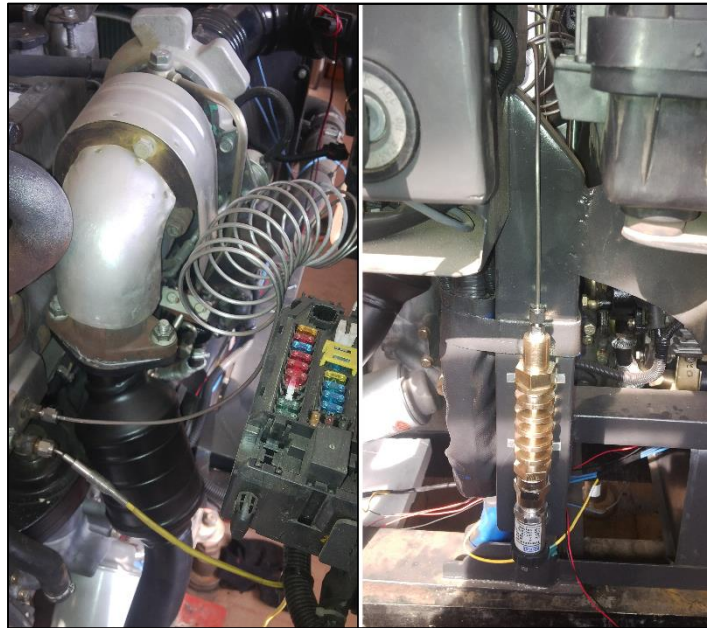


Figure 25 - Exhaust pressure transducer with cooling fins and tube

Once each of the thermocouples and pressure transducers had been placed in the ideal position for the measurements, the fuel measuring system needed to be designed and built.

The way in which the fuel is measured is by suspending a 25 litre container from a 50 kg Vishay load cell. This measurement required high accuracy which this specific unit met as it has an error of 0.05%.

A fuel manifold was manufactured for the purpose of running the engine from the main tank's fuel until each test started. The valve positions were then altered to use the fuel from the secondary tank suspended from the load cell. Figure 26 shows this setup below.



Figure 26 - Fuel measurement system

A component of the test bench that needed repairs was the dynamometer. More specifically, the oil dashpot. The purpose of the dashpot is to dampen out all the vibration caused by the engine to give a steady, constant torque reading. The problem with it was a worn oil seal and, therefore, did not retain any oil. This caused problems when trying to read the torque produced by the engine.

The solution to this was to bypass the dash pot completely by removing the counter balance weights and installing a 200 kg load cell in their place. This addition meant that the dashpot and weights were no longer needed and could be removed. A major benefit to this upgrade was that the torque produced by the engine could be measured extremely accurately and, in addition, could be logged to the computer. For this solution to work correctly, a mounting point was needed to keep the load cell steady and simulate the force acting upon the engine. This mount would need to withstand at least 86 kg to handle the 300 Nm of torque produced by the engine. A SolidWorks simulation was performed on the mount design before it was manufactured to ensure that it would be able to handle the force. The figure below shows the results from the stress analysis which revealed that the maximum stress that the mount would experience under an upward force of 86 kg would be 105.2 MPa. This value is far below the yield stress for the selected SAE 1010 steel and is therefore adequate for the purpose. The results of the simulation for the maximum deformation and strain of the mount can be seen in Appendix B.

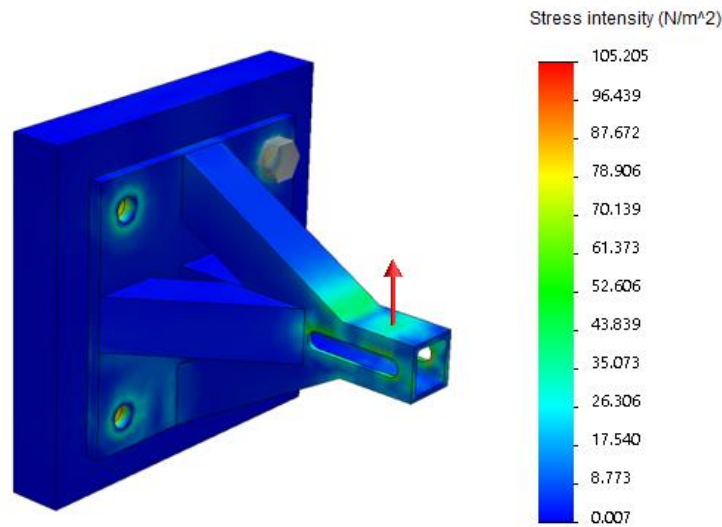


Figure 27 - SolidWorks stress simulation for dynamometer load cell upgrade

The load cell alteration made to the dynamometer can be seen in Figure 28 below. The original counter weights are shown to the left of the picture whereas the load cell upgrade is depicted on the right hand side.

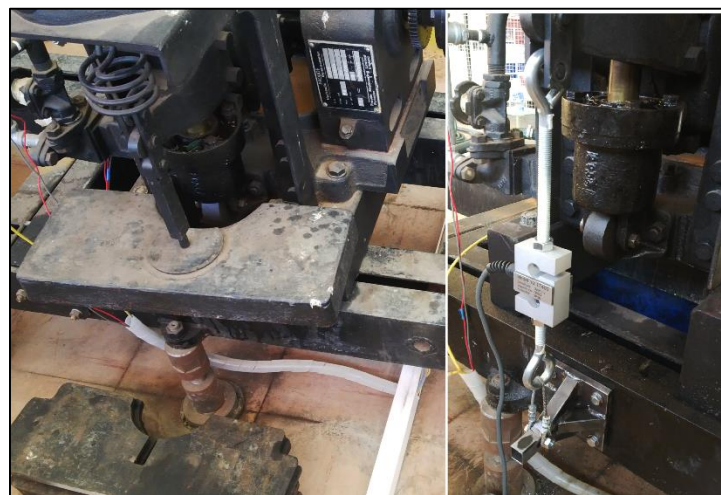


Figure 28 - Dynamometer load cell modification

The addition of an electronic, high-temperature water mass flow meter (Figure 29 below) to the engine cooling system was a great benefit when it came to accuracy and data acquisition. The original mass flow meter was not able to adjust its readings according to the temperature of water flowing through it which has a large effect on accuracy. This problem along with the fact that the engine produced a mass flow rate larger than what the old meter could read meant that a more sophisticated unit was required.

This water mass flow meter works by sensing the velocity at which a paddle wheel spins as a result of the water flowing through the unit.



Figure 29 - Electronic water mass flow meter

One of the final improvements made to the test bench was the addition of a handheld optical tachometer. This modification adds the ability to continuously log the engine speed throughout each of the tests. This unit adds increased accuracy when compared to the analogue tachometer dial on the engine's control panel.

As can be seen in Figure 30 below, the optical tachometer was mounted on the dynamometer in such a way that it could be easily removed if needed elsewhere. The tachometer was mounted in front of a reflective sticker on the hub of the prop shaft. This specific mounting position is required as the tachometer works by emitting light that is reflected from the sticker and read by an optical sensor in the unit. A voltage is then produced which is dependent on the engine speed and can be read by a data logger.



Figure 30 - Optical tachometer setup

4.2 Calibration

The calibration of test equipment is extremely important for any experiment. If the equipment accuracy is not reliable then neither are the results obtained from the tests which would lead to an incorrect conclusion being made.

Testing the accuracy of all the measuring devices such as thermocouples, pressure transducers, mass flow meters, tachometers and load cells needs to be done to know whether calibration is needed.

4.2.1 Thermocouples

A thermocouple works by joining two different strands of wire together, each being a different metal. At this join, a small potential difference is created. The value of the voltage is directly proportional to the temperature at this point.

Two different types of thermocouples that were used for tests; these are type T and type K thermocouples. The main difference between the two is that type K thermocouples have a wide temperature range from -200°C to 1250°C , whereas the type T thermocouples have a narrow range of -200°C to 350°C , which makes the type K thermocouples ideal for exhaust gas temperature measurements.

The figure below illustrates the specifications of each type of thermocouple.

ANSI Code	Alloy Combination		Thermocouple Color Coding		Maximum Temperature Useful Range	Limits of Error
	+Lead	-Lead	Thermocouple Grade	Extension Grade		Standard
K	CHROMEGA NICKEL- CHROMIUM Ni-Cr	ALOMEGA NICKEL- ALUMINIUM Ni-Al (magnetic)			-200 to 1250°C	-200 to 1250°C 2.2°C or 0.75%
T	COPPER Cu	CONSTANTAN COPPER- NICKEL Cu-Ni			-200 to 350°C	-200 to 350°C 1.0°C or 0.75%

Figure 31 - Thermocouple specifications (OMEGA ENGINEERING, 2013)

Two set points are needed to calibrate the thermocouples as the relationship between voltage produced, and temperature are linear. The set points that were used were 0°C and 80°C. For the 0°C point, all the thermocouples were placed into a container with a mixture of crushed ice and water. This slurry of ice and water would be at 0°C since ice will remain at a constant temperature until it has been completely melted. A highly accurate thermometer was also placed in the ice mixture to ensure that the temperature was, in fact, 0°C. Any thermocouples that deviated from 0°C were adjusted on the data logger until they displayed the correct value. The figures below illustrate the process.



Figure 32 - Thermocouple calibration at 0°C

To ensure the greatest accuracy from the thermocouples, the readings were continuously adjusted until there was no more than a 0.3°C error.

As can be seen below in Figure 33 for the 80°C set point, the same procedure was followed as the 0°C point. Water in a container was heated up to 80°C with the aid of a heating element. The thermocouples along with the thermometer were then placed into the water, and the values were compared and corrected if necessary.



Figure 33 - Thermocouple calibration at 80°C

4.2.2 Pressure

Pressure is the amount of force applied to a certain area.

$$P = \frac{F}{A} \quad \text{Equation 45}$$

Pressure transducers were used to measure the engine air inlet pressure as well as the exhaust pressure before the turbo's turbine.

These pressure transducers produce a current that ranges from 4-20 mA. This current is converted to a pressure value in kilopascals [kPa] in the data logger ranging from 0-1000 kPa.

To verify the accuracy of the pressure transducers, each one was connected to a pressure regulator which controlled a pressurised air supply. From the regulator, the pressurised air was fed to the pressure transducer as well as to a second calibrated pressure gauge. In a similar manner to the thermocouple calibration, the transducers needed to be set at two separate points. The pressure values that were used were 0 kPa (gauge) and 300 kPa (gauge). The images below show the calibration setup.

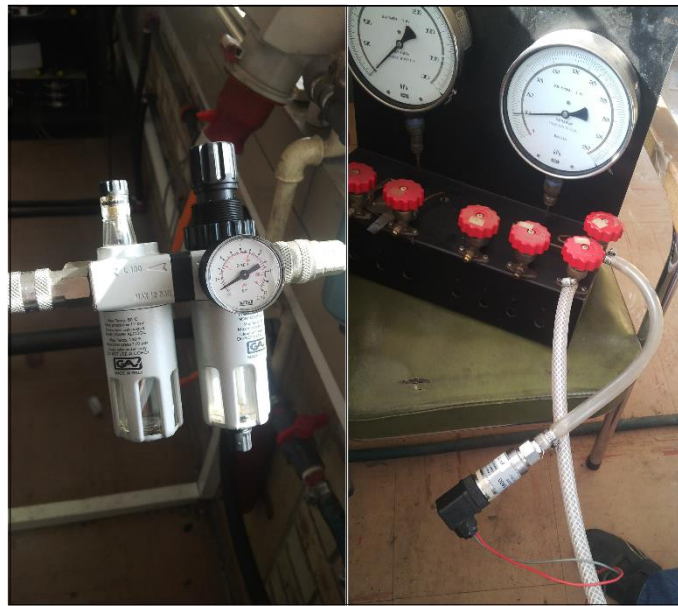


Figure 34 - Pressure transducer calibration setup

The calibration results for the pressure transducers at 0 and 300 kPa can be seen in the figures below.

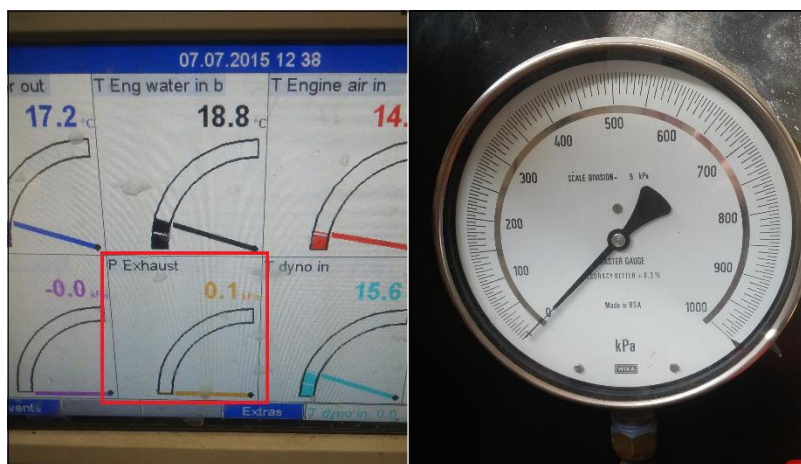


Figure 35 - Pressure transducer calibration at 0 kPa



Figure 36 - Pressure transducer calibration at 300 kPa

The calibration procedure was repeated until the measurement error was within an acceptable range as seen in the two images above.

4.2.3 Flow meters

Flow meters are used to determine the volume of water passing through the dynamometer and engine systems at any specific time. They are usually marked in litres per minute or kg per second.

For purposes of this experiment, two flow meters are to be used. The first one is to measure the volume of water flowing through the dynamometer and the second is to measure the volume of water moving through the engine's cooling system.

Calibration of the dynamometer's flow meter was done by pumping water through the flow meter and into a bucket for 10 seconds. The bucket was then suspended from a pre-calibrated digital scale and load cell to measure the mass of the water in the bucket. This procedure was repeated three times to ensure a consistent reading. For each test, the reading on the flow meter was noted and compared to the mass of water in the bucket as shown in Figure 37 below.

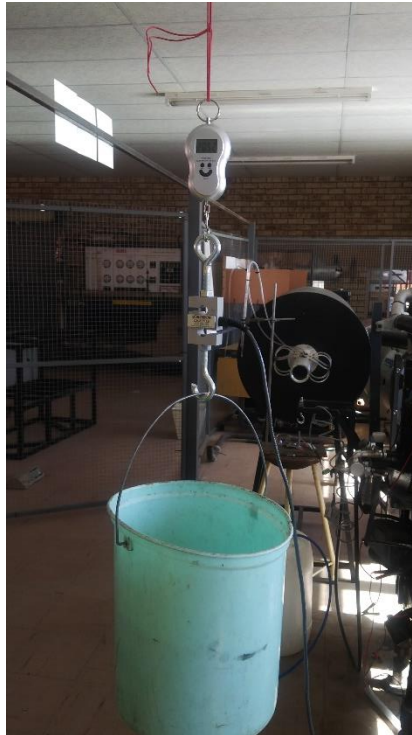


Figure 37 - Measuring mass of water for mass flow meter calibration

The results for the dynamometers flow meter are shown in the table below.

Table 3 - Dynamometer flow meter calibration results

Method	Test 1	Test 2	Test 3
Flow meter [L/min]	33	33	33
Bucket [L/min]	32.8	32.5	33.3

As can be seen from the above results, the flow meter has been calibrated previously at the working temperature and is, therefore, suitable for the experiment.

Due to the engine's cooling water being at a high temperature during operation, the flow meter needed to be calibrated with heated water to compensate for the change in density. From previous tests on the engine, it was found the average temperature of water flowing into the engine after it had been cooled down was 60°C.



For this reason, the temperature of the water used for the calibration was first heated up to 60°C before pumping it through the flow meter into the bucket. The results of this test can be seen below.

Table 4 - Engine flow meter results before calibration

Before calibration	Test 1	Test 2	Test 3
Flow meter [L/min]	55.2	55.3	55.2
Bucket [L/min]	51.3	50.8	51.1

After the K-factor of the flow meter had been altered to correct the reading, the following results were obtained.

Table 5 - Engine flow meter results after calibration

After calibration	Test 1	Test 2	Test 3
Flow meter [L/min]	51.2	51.1	51.2
Bucket [L/min]	50.9	51.2	51.4

After calibration of the mass flow meters had been done, the readings that each of them produced were within acceptable ranges of the true value. The slight variations in the readings for the bucket tests are due to human error during the timing and filling.

4.2.4 Load cells

The two load cells used in this setup are extremely important as they record the fuel consumption as well as the load applied to the engine. It is thus vital for them to be as accurate as possible.

To calibrate the load cells, it was necessary to apply known weights of various masses to them. The calibration was started by hanging the load cell so that it would hang under its own mass which was set to 0 kg. Before any of the weights were applied to the load cells, they were first placed on an accurate, calibrated laboratory scale for the mass to be known. For the greatest accuracy, the total mass placed on each of the load cells needed to be larger than what the load cells would bear during the experiment.



Figure 38 - Load applied to load cell for calibration

The figure above shows the weights placed on the hanging load cell as well as the mass readout after being calibrated. This load was applied and removed three times to ensure that the reading remained constant.

4.2.5 Engine speed

The engine speed will be monitored and logged with the aid of a handheld optical tachometer. This device emits light that is aimed at the hub of the engine's output shaft. The hub is painted black with the addition of a reflective silver patch at a single point. The tachometer picks up the light reflected from the silver patch as it rotates and produces a 0-10 volt output according to the rate at which it senses a reflection of light. The data logger converts this voltage into a reading of 0-10 000 RPM.

A comparison of the data produced by this unit to the analogue tachometer mounted on the engine's control panel as well as the digital readout from the engine's ECU was made to determine if this unit was accurate or not.

All of the readouts displayed extremely similar values to one another and, therefore, indicated that the optical tachometer was correctly calibrated.

4.2.6 Air flow

For the engine's air flow to be known during the experiment, an air flow meter would be needed. For this purpose, a large drum and venturi were connected to the air intake system. The reason for the drum in Figure 39 is to act as a baffle for the pulses of air that the engine sucks into each cylinder. This drum would produce a stable reading on the venturi manometer. To ensure that the venturi was producing accurate readings, it was compared to two separate digital manometers. The results indicated that the venturi was accurate enough to be used.

Shown below is an image of the air drum used to baffle out the air flow pulses created by the engine.



Figure 39 - Air drum used to minimise pulses in air flow measurements

In order for logging of the air flow during tests to be possible, the output wires from the engine's built in mass air flow meter were connected to the data logger. The meter produces a voltage output of 0-5 volts which is air flow dependent. For it to be possible to convert these voltages to mass air flow, the engine was run at different speeds ranging from idle to WOT. The voltages were compared to the venturi readings so that a formula could be derived to automatically convert the voltage to mass flow. The graph and equation are shown below.

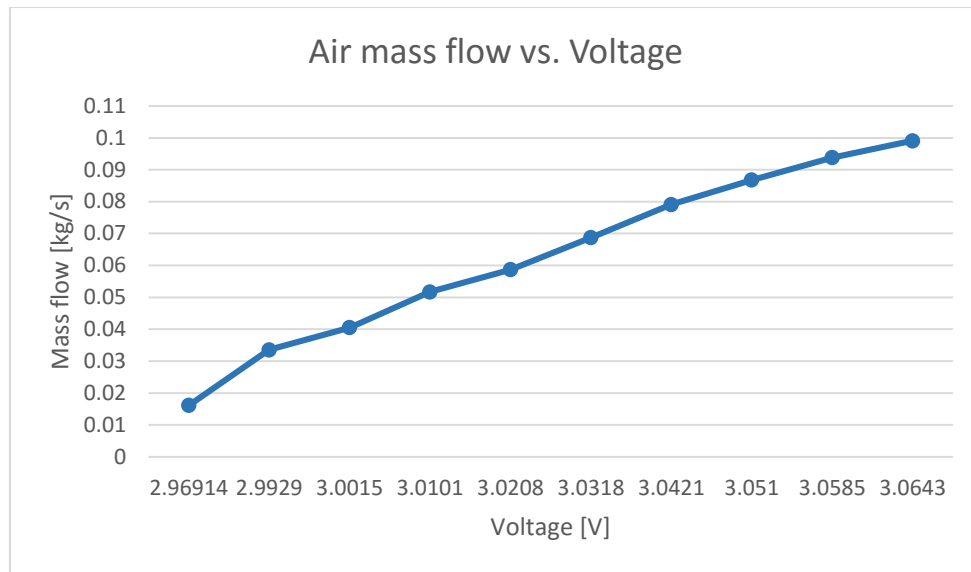


Figure 40 - Air mass flow vs. voltage graph

$$\dot{m}_{air} = Voltage^2 \times 0,952195 - Voltage \times 4,86117 + 6,05456 \quad \text{Equation 46}$$

4.3 Safety

Safety is always extremely vital when working around machinery. As can be seen Figure 41 below, personal protective equipment (PPE) is required before anyone is allowed to work near the engine test bench while it is in operation. This equipment includes safety shoes, safety glasses and ear protection.



Figure 41 - PPE safety signs



One of the major safety hazards on the test bench is the open prop shaft that connects the engine to the dynamometer. A shield was manufactured to completely surround the shaft to address this issue that would prevent any part of the shaft from being thrown around during operation and possibly injuring someone in the event that something went wrong.

The engine exhaust system is designed so that the silencer box is installed under the dynamometer stand. This is done to prevent anyone from falling over or onto the box and resulting in serious burn injuries.

Another precaution that has to be taken is to ensure that there is always an easily accessible fire extinguisher at all times in case of a fire.

Other general safety rules that need to be followed when operating the engine test bench are as follows:

- Don't wear any loose hanging clothing or jewellery when working near any moving parts.
- Long hair must be tied back when operating the engine
- Don't leave tools or any loose objects on or near the engine and dynamometer.
- Make sure all the necessary engine and dynamometer guards are in place before starting the engine.
- Clean up any oil, water or diesel spills immediately.
- Never operate the test bench alone or without supervision in case of an emergency.
- Don't touch any moving parts.
- Certain parts get extremely hot during operation, make sure to avoid contact with these parts.

4.4 Test procedure

Due to the specific outcomes of this study, a number of different tests needed to be completed. This meant that a unique test procedure had to be compiled for each situation.

4.4.1 Gross calorific value for diesel/additive

Since this study focuses on combustion efficiency of internal combustion engines, it was important to know how much energy was entering the engine at all times. For this to be possible, the GCV_v of both the diesel fuel and Naf-Tech needed to be determined.

These tests are done with the aid of a bomb calorimeter at the chemical engineering department at North West University which can be seen in Figure 42 below. This instrument has been professionally calibrated beforehand.



Figure 42 - Bomb calorimeter used for GCV_v test

Before the test began, it was found that Naf-Tech evaporates at a rate too fast to be accurately weighed and tested. Gelatine capsules were purchased and used to overcome this issue. Since the capsules would also be burnt during the tests, their GCV's needed to be determined in a separate test without the diesel and additive. This value would then be subtracted from the test value obtained with the fuel and additive. The cotton thread used for ignition has a known calorific value that also needs to be subtracted from the test value before the final value can be determined.



Taking the above into consideration, the testing procedure is as follows:

1. Weigh three empty, dry capsules to determine their mass.

Repeat steps 2 and 3 at least three times.

2. Input the mass value of the empty capsule into the bomb calorimeter's computer, and place the capsule into the combustion crucible of the unit.
3. Attach a single cotton thread (wick) from the electrode to the capsule in the unit and start the combustion process.
4. Add no more than 1g of diesel to the capsules with a syringe. At least three capsules need to be setup for each type of diesel.
5. Add no more than 1g of Naf-Tech to each of 3 capsules.
6. Mix three different ratios of diesel and Naf-Tech in three separate capsules. The first must contain a smaller amount of Naf-Tech than what the manufacturers recommend, the second needs to be the recommended mixing ratio. The last capsule must contain a larger amount of Naf-Tech than the recommended ratio.

Repeat steps 7 to 9 for each of the capsules containing diesel and/or Naf-Tech at least three times for repeatability:

7. Weigh the capsule with its contents.
8. Input the mass value into the bomb calorimeter's computer, and place the capsule into the combustion crucible of the unit and attach a cotton thread from the electrode to the capsule. Start the combustion process.
9. Once the test has been completed, thoroughly wash and dry the crucible for the next test.

The energy readouts from the empty capsules and the cotton threads are subtracted from the readout obtained from the fuel and additive tests to find the correct GCV_v of each.

Below is an example of the readout from one of the experiments.

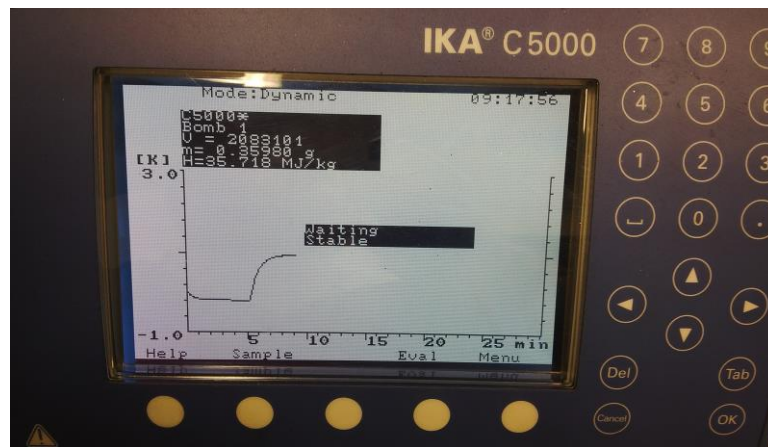


Figure 43 - Bomb calorimeter digital readout

4.4.2 Aggreko Rental Energy generator

Tests were conducted at Aggreko in Olifantsfontein to compare the efficiency and exhaust emissions of an 800 kVA generator upon the introduction of Naf-Tech to the fuel at different ratios. The procedure followed to conduct the test is listed below:

1. Fill the generator fuel tank before starting the tests and take note of the fuel reading.
2. Set up the data logging equipment for all the tests.
3. For a cold start, turn the generator on and run at idle with no load for 5 minutes to warm the engine up.
4. Once the engine is warm, set the load bank control unit to 528 kW at 100% load for safety reasons and close the circuit breaker to the load bank.
5. Initially, repeat steps 6 to 9 at least three times under the same conditions to determine the repeatability of the tests.
6. Take load, voltage and current readings from the generators control panel as well as the load bank control unit at the start of each test.
7. Let the generator run at the set condition for 55 minutes, allowing 5 minutes to take readings and refill the fuel tank before the next test.
8. Take load, voltage and current readings from the generators control panel as well as the load bank control unit at the end of each test.
9. Refill the generator fuel tank before the next test.

10. Repeat steps 6 to 9 at 50% (264 kW), 75% (396 kW) and 100% (528 kW) load. The tests at each load need to be done at least three times to ensure the results are repeatable.
11. Add Naf-Tech to the diesel of the generator in the ratio of 3ml Naf-Tech per litre fuel.
12. Once again, repeat steps 6 to 9 at 50% (264 kW), 75% (396 kW) and 100% (528 kW) load at least three times each while adding additional Naf-Tech to the fuel tank to maintain the mixing ratio before each test.
13. Once the tests in step 12 have been completed, fill the tank without adding additional Naf-Tech. These tests will indicate the effect of Naf-Tech at lower concentration levels.
14. After the last test has been completed, remove the load and let the generator idle for 5 minutes before shutdown.

4.4.3 CSIR diesel burner

Tests were performed at the CSIR in Pretoria on a diesel burner test rig to determine the difference in combustion properties of the fuel additive and whether or not the additive introduces additional energy to the base diesel. This will be determined by monitoring the steady-state flame temperature produced from the combustion of the fuel. A schematic representation of the test rig is shown in Figure 45 below. Three sets of tests will be conducted with both the diesel and the diesel/additive mixture i.e. a rich AF ratio, an intermediate AF ratio and a lean AF ratio. During the tests, the outlet temperature will be measured at five different radii. The thermocouples will be rotated around the circumference of the burner outlet, as indicated in Figure 44, to obtain a full range of temperature values.

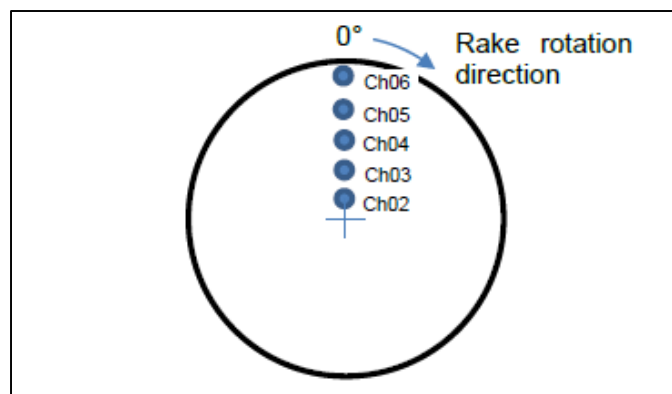


Figure 44 - Temperature measurement points and rotation direction (Meyers, 2015)

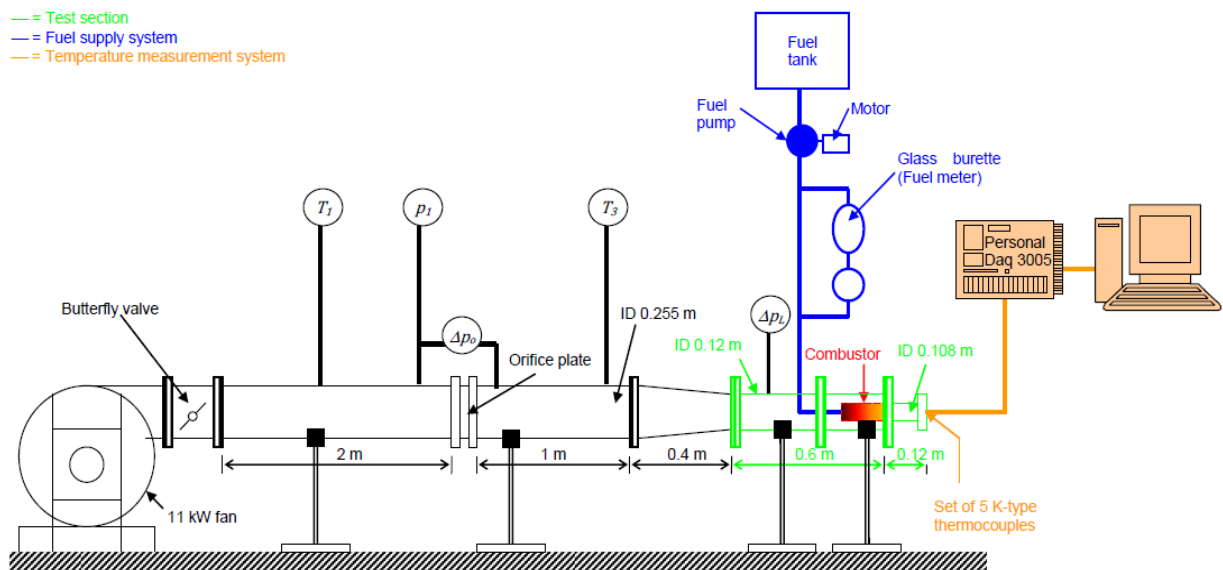


Figure 45 - Schematic diagram of the combustor test rig (Meyers, 2015)

The image below shows how the test rig was set up.



Figure 46 - Combustor test rig



The procedure to be followed during the tests was as follows:

1. Ensure all extraction fans are correctly positioned and operating.
2. Start the blower fan to provide a constant air supply to the burner.
3. Monitor the inlet temperature until a steady value has been reached.
4. Note the atmospheric pressure and temperature.
5. Begin the test by turning on the fuel pump and igniting the air/fuel mixture.
6. Insert the thermocouples to the first position, as in Figure 44 above.
7. Reposition the thermocouples after 10 minutes.
8. Once measurements have been taken at each of the positions, the lean blowout air/fuel ratio must be determined by increasing the air flow rate until the mixture is too lean for combustion to take place.
9. Note the atmospheric pressure and temperature.

Repeat steps 1 to 9 for each air-fuel ratio, i.e. lean, intermediate and rich.

4.4.4 GWM 2.5 TCi diesel engine (North-West University)

The main objective of this study is to perform tests on the GWM 2.5 TCi diesel engine situated on the North West University - Potchefstroom campus. These tests will measure the performance, fuel consumption and emissions of the engine in order to determine the effect that Naf-Tech has on the combustion efficiency. For it to be possible to obtain the above-mentioned results, the following procedure needs to be followed during testing:

Repeat the steps below for each of the different fuel requirements i.e. with and without Naf-Tech.

1. Turn the dynamometer water pump on.
2. Turn the ignition switch to start the engine and let it warm up to and stabilize at 80°C before conducting any tests.
3. Turn the radiator fans on and make sure all the parameters have stabilized before starting the tests.

Repeat steps 4 to 8 at least three times each for repeatability at the following engine speeds: 3400, 3000, 2600, 2200 and 1800 RPM for the additive concentrations of 0 ml Naf-Tech/L diesel, 2.5 ml Naf-Tech/L diesel, 3 ml Naf-Tech/L diesel and 3.5 ml Naf-Tech/L diesel.



4. Open the throttle fully until the engine speed cannot be increased anymore for wide open throttle tests.
5. Turn the hand wheel on the dynamometer to increase the load applied to the engine until engine speed drops to the desired point and wait for all parameters to stabilize.
6. Switch the fuel supply from the main tank to the fuel tank on the load cell and start the stopwatch for 10 minutes.
7. Stop the stopwatch after 10 minutes and switch the fuel supply back to the main tank.
8. Remove the load from the engine, set it to idle and refill the load cell tank with fuel for the next test.

All tests conducted on the GWM engine will be done with the use of 500 ppm diesel. Each of the tests performed on the engine will use diesel from the same batch for the purpose of eliminating any variations in fuel quality or composition or pollutants.



Chapter 5: Results

Chapter 5 presents and discusses all the results for each of the test procedures described in the previous chapter. These results have been analysed and sorted into their respective sections in this chapter. Each section of results obtained from the various tests contributes towards the outcome of this study.

The results gained from the gross calorific value test done on the fuel and additive are discussed in section 5.1. The results gathered from the tests completed at Aggreko in Olifantsfontein on the 800 kVA generator are presented in section 5.2. Section 5.3 describes the results of the tests undertaken at the CSIR on the diesel burner. Section 5.4 will present all the results obtained from the tests carried out on the GWM engine at NWU with and without the use of Naf-Tech and finally, section 5.5 will discuss the results gained from the mathematical model and where possible, a comparison will be made to the manufacturers specification sheet from the technical manual.

5.1 Gross calorific value for diesel/additive

The raw data from the bomb calorimeter tests was analysed to obtain the following results in Table 6 below.

For the purpose of the tabulated results in this chapter and the next, Naf-Tech will be referred to as “NT”.

Table 6 - GCV test results

Gross Calorific Value [MJ/kg]						
Test no.	Capsule	Naf-Tech	Diesel	2 ml NT/ litre diesel	3 ml NT/ litre diesel	7 ml NT/ litre diesel
1	15,181	33,79	46,951	45,626	46,942	46,137
2	17,462	32,626	46,186	46,015	46,466	46,757
3	16,756	32,484	46,426	45,457	46,986	45,856



The bar graph below is a graphical representation of the data in the above table. This gives a better indication of the comparison between each of the tests.

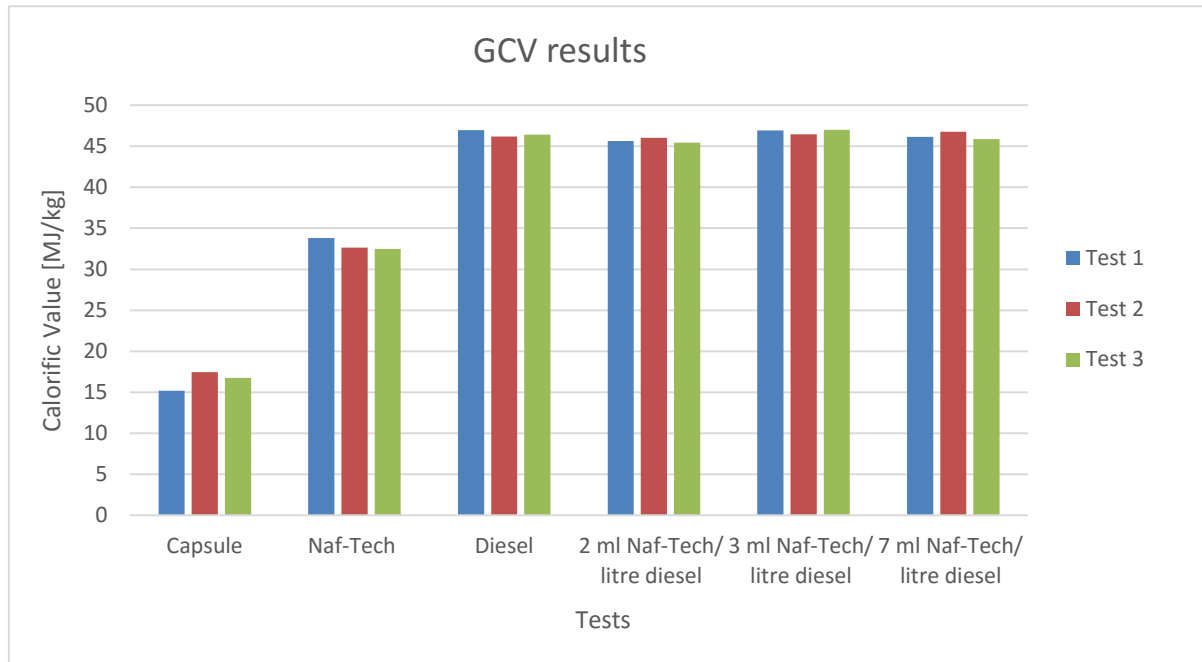


Figure 47 - Graph of GCV results

As can be seen from these results, there is negligible effect on the energy contained in the fuel with the various concentrations of Naf-Tech.

5.2 Aggreko tests

5.2.1 Fuel consumption and performance

The tests conducted on the generator revealed that at 50 and 100% load, the fuel consumption remained relatively constant before and after the introduction of the additive to the fuel. At 75% load, there is a reduction of 9.82% in fuel consumption with the use of the additive.



Table 7 below is a summary of the results gained from the tests regarding fuel usage.

Table 7 - Fuel consumption results

Fuel consumption [l/h]			
	50% load	75% load	100% load
No Naf-Tech [l/h]	70,5996	114,8703	136,4190
With Naf-Tech [l/h]	71,3877	103,3726	140,5231

Figure 48 is a graphical representation of the above table.

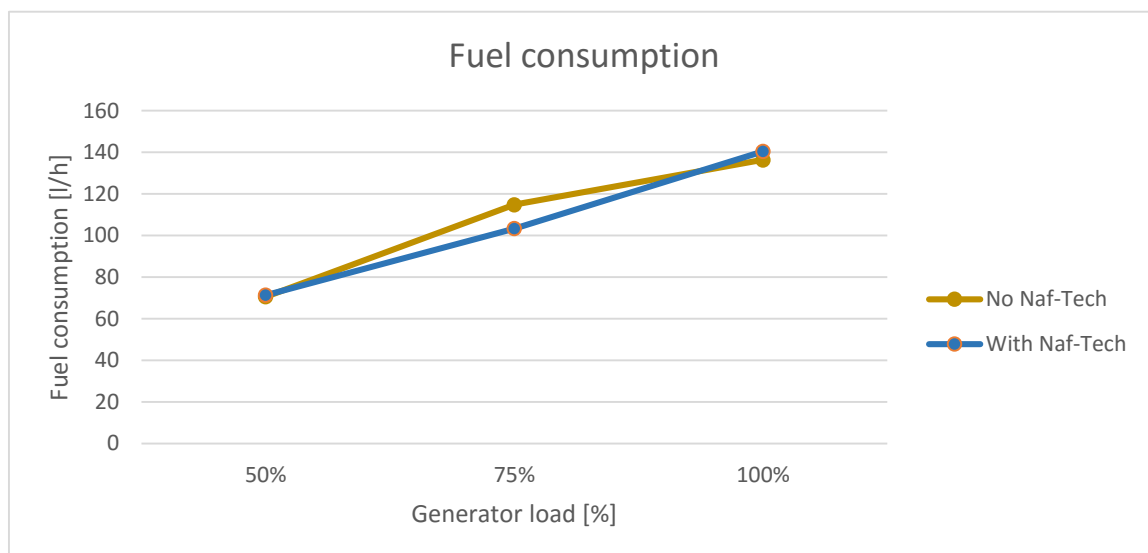


Figure 48 - Fuel consumption vs. load

The results below show the effect of Naf-Tech on the fuel consumption per kWh produced by the generator.

Table 8 - Fuel consumption per kWh results

Fuel consumption per kWh [l/kWh]			
	50% load	75% load	100% load
No Naf-Tech	0,2818	0,3114	0,2819
With Naf-Tech	0,2931	0,2808	0,2874

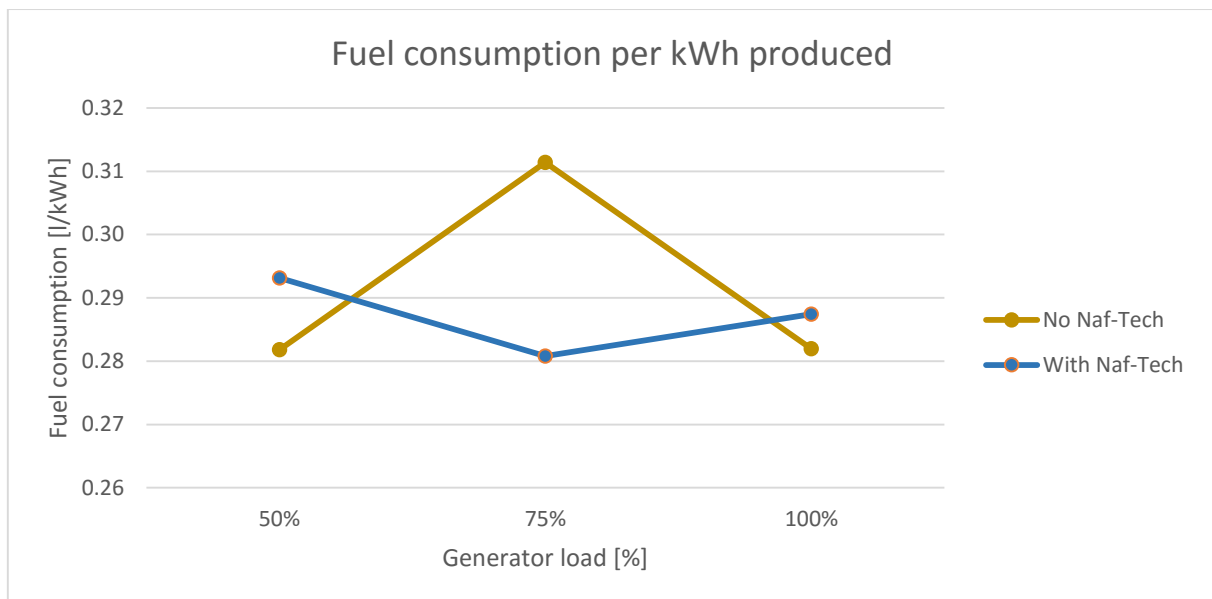


Figure 49 - Fuel consumption per kWh produced

The results above indicate that with the use of Naf-Tech the generator was 9.82% more efficient at 75% load, but at 50 and 100% load, a slight drop in efficiency is seen.

5.2.2 Emissions

Levego was contracted for the purpose of emission testing on the generator unit. Levego is an independent company that specialises in air quality analysis and are suppliers of the necessary equipment to measure the exhaust gases during the tests.

The gases measured during the test were CO, CO₂, O₂, NO_x and VOCs.

The table below was taken from the Levego report and shows the results of the VOC emission tests carried out at Aggreko.

Table 9 - VOC's Aggreko emission test (Woollatt, 2015)

Aggreko VOCs emission results [ppm]			
	50% load	75% load	100% load
No Naf-Tech	-	602,00	484,00
With Naf-Tech	172,00	129,00	97,00

The results in the above table indicate that the volatile organic compounds (VOCs) produced by the combustion process are reduced by 79% at each of the load conditions while using Naf-Tech with the fuel.

Unfortunately, no data was recorded for the tests conducted at 50% load without the additive as the gas analyser unit malfunctioned during that specific test but the results at the 75 and 100% load instances are enough to make appropriate conclusions with regards to the emissions.

Figure 50 below shows a graphical representation of the above table. From this graph, the effect of Naf-Tech on the VOCs emissions is clearly seen.

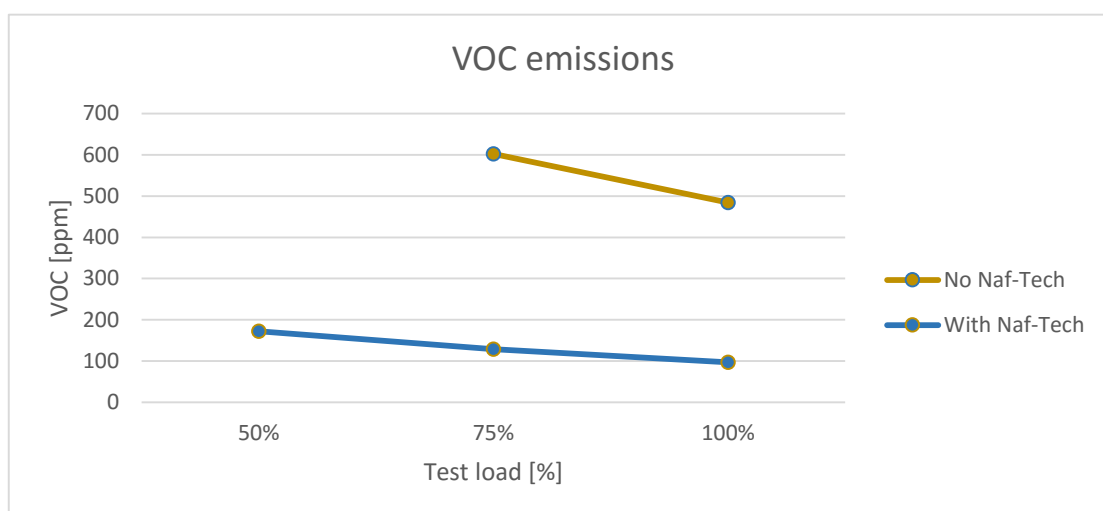


Figure 50 - VOC's Aggreko emissions test graph

As with the combustion of any fuel, the formation of combustion gases will be present. Displayed in the tables below are the summarised results for these emissions.

Table 10 - Aggreko emission testing results without Naf-Tech (Woollatt, 2015)

Combustion gas analysis results (No Naf-Tech)

	CO [mg/Nm ³]	CO ₂ [%]	O ₂ [%]	NO _x [mg/Nm ³]
50 % load	241,05	5,54	13,38	1418,53
75 % load	318,69	7,24	11,04	1691,37
100 % load	457,61	9,25	8,27	2014,34



Table 11 - Aggreko emission testing results with Naf-Tech (Woollatt, 2015)

Combustion gas analysis results (With Naf-Tech)				
	CO [mg/Nm ³]	CO ₂ [%]	O ₂ [%]	NO _x [mg/Nm ³]
50 % load	179,48	7,25	10,95	1830,49
75 % load	321,21	8,66	9,05	1973,39
100 % load	454,83	7,77	10,30	1725,55

As seen from the results in the above two tables, there is no distinct difference in the exhaust emissions after Naf-Tech has been introduced to the fuel. As stated in the report written by Levego, the differences seen in the results are of normal variation for a combustion process such as this the one in this instance (Woollatt, 2015).

5.3 CSIR diesel burner tests

Due to the large amount of data obtained from the diesel burner tests at the CSIR, only the results from the outermost thermocouple, CH06, (refer to Figure 44) at the burner exhaust was chosen for the purpose of this chapter. This choice was made due to thermocouple CH06 being directly in the path of the flame produced by the diesel combustion. For the full set of results of these tests, please see Appendix C.

The results indicated in the graphs below represent the temperature data obtained from the thermocouple mentioned above, both before and after the use of the fuel additive.

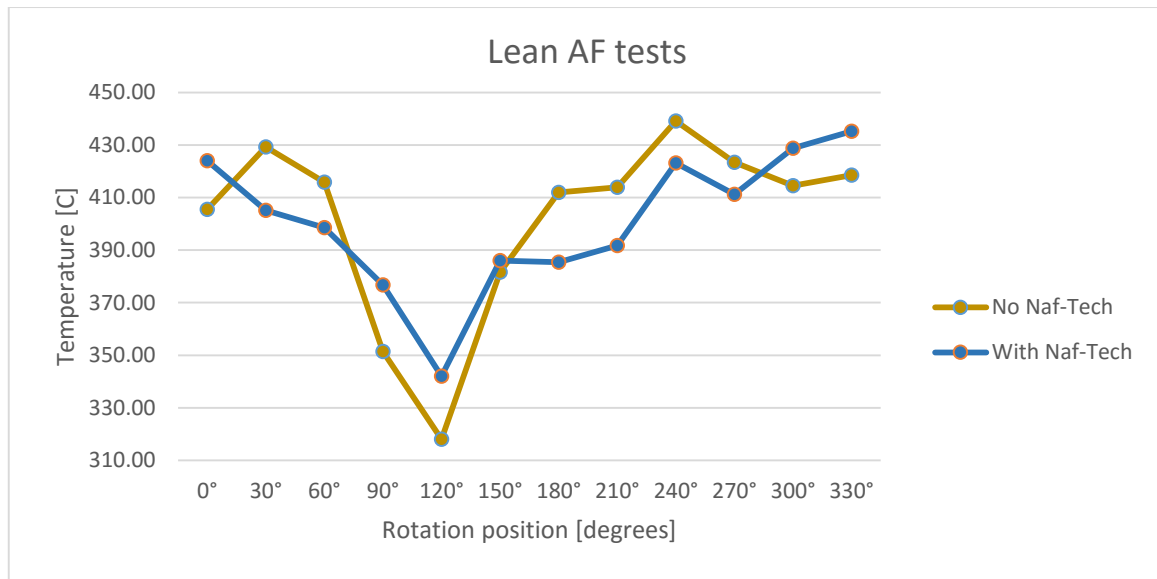


Figure 51 - Lean AF test

The first set of tests run was to determine the effects of the additive with a lean AF ratio. The ratio used was 82:1. By observing the results in Figure 51 above, it is seen that there is no distinct increase or decrease in the flame temperature with the addition of the additive to the fuel.

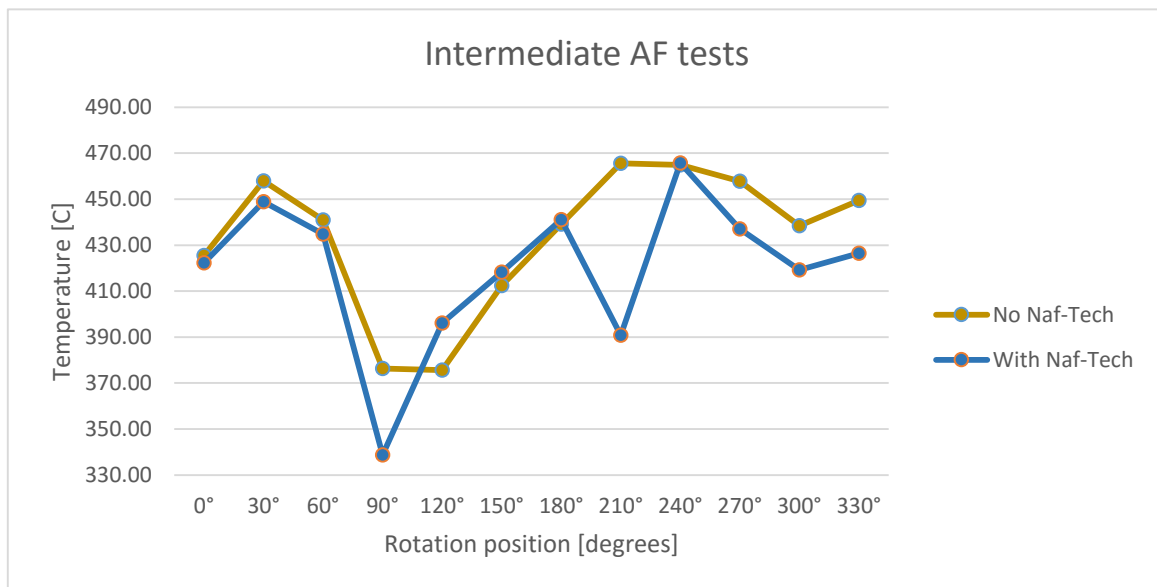


Figure 52 - Intermediate AF test

The second set of tests used an intermediate AF ratio of 75:1. The results from this set revealed a similar outcome to the initial lean test set.

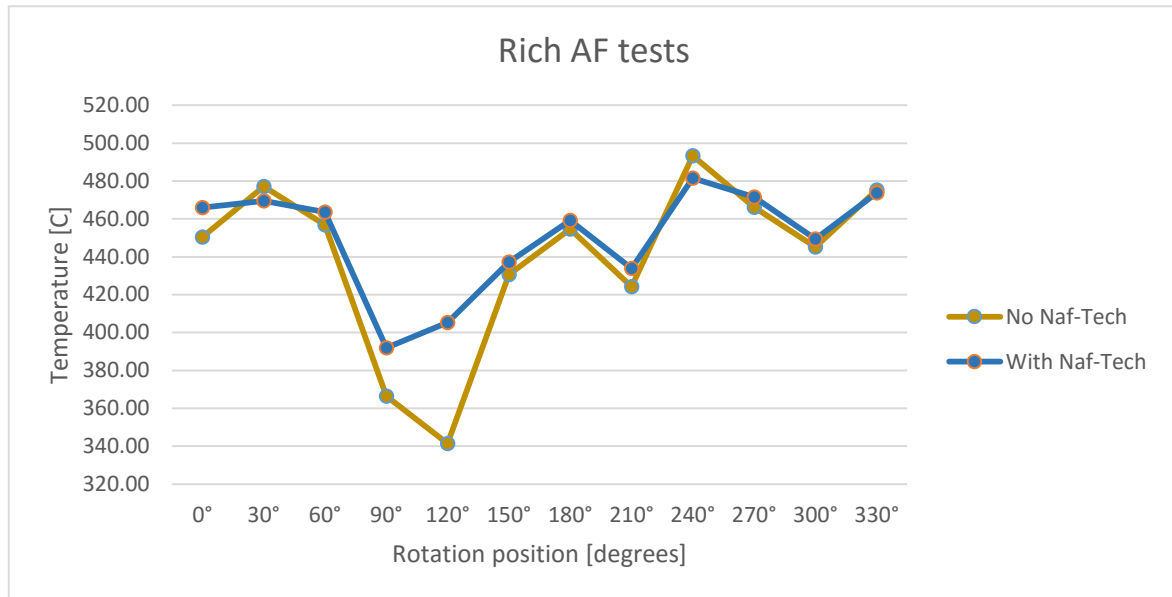


Figure 53 - Rich AF test

The final set of tests was conducted using the richest AF ratio of 66:1. As seen in Figure 53, the results gained from this set indicate, yet again, that there is no effect on the flame temperature when using Naf-Tech.

With the exception of a few outliers that occur due to carbon deposits breaking off the combustion walls and settling on a thermocouple, it can be seen that there is no definite trend that indicates any difference in flame temperature before and after the use of the additive.



5.4 NWU GWM 2.5 TCI diesel engine tests

The results of the tests in this section are the main focus of the study and are therefore described in greater detail than the results in the previous sections.

Two separate sets of tests were done on the GWM engine. These two sets, from here onwards referred to as test set one and test set two, followed the same procedures with the exception of running at different test RPM intervals. The main reason the first set was done was to determine if all instrumentation was in place and if all the procedures were correct. The results from this set of tests were determined to be reliable although not as accurate as the second set. The increased accuracy in the second set was due to the instrumentation and logging equipment being upgraded. This eliminated the need to take manual measurements by hand as done during test set one.

Before each of the test sets, the engine oil was replaced to prevent any unnecessary wear on engine components. The oil used was Shell Helix-Diesel 15W-40. The diesel fuel used for the purposes of each test was Shell 500 ppm.

5.4.1 Test set one

The first set of tests that produced the following results was performed towards the end of 2014 after all the instrumentation, and measuring apparatus had been installed. These tests were conducted using the same procedure as the second set of tests.

To obtain baseline results for this set, commercial diesel was used without the addition of Naf-Tech. The tests that followed were conducted with the manufacture's recommended mixing ratio of 3ml Naf-Tech/ litre diesel.

The results below represent the power produced by the engine with and without the addition of the additive.



Table 12 - Power – tabulated results (test set one)

Power [kW]

RPM	Without NT	With NT
1800	46,97	47,10
2400	64,35	64,20
3000	71,65	71,90
3500	73,91	74,50

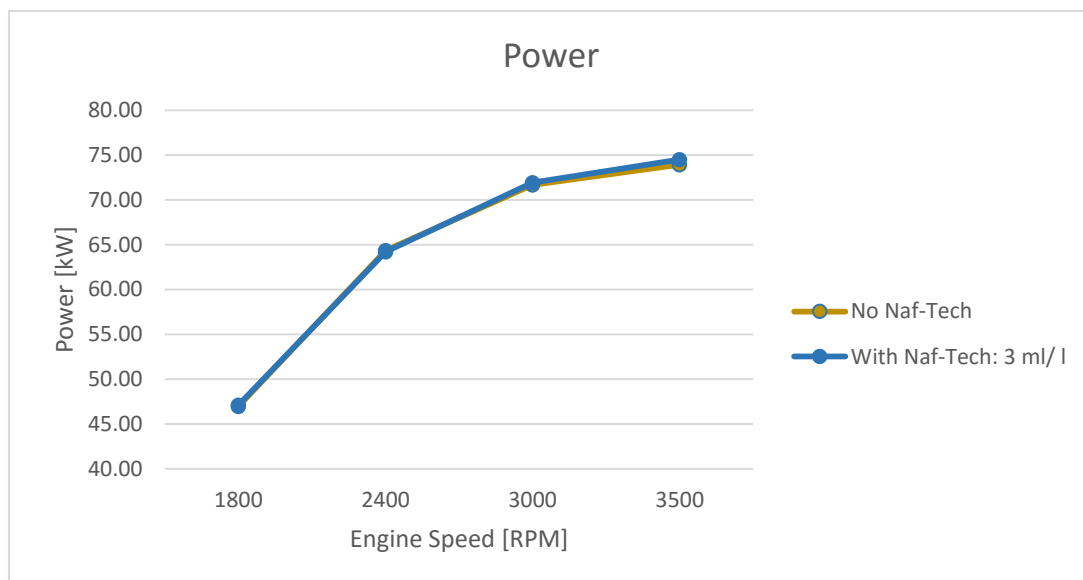


Figure 54 - Power – graphical results (test set one)

Table 13 and Figure 55 below summarise the results obtained for the torque of the engine over the two tests in test set one.



Table 13 - Torque – tabulated results (test set one)

Torque [Nm]		
RPM	Without NT	With NT
1800	249,18	249,87
2400	256,04	255,44
3000	228,07	228,86
3500	201,65	203,26

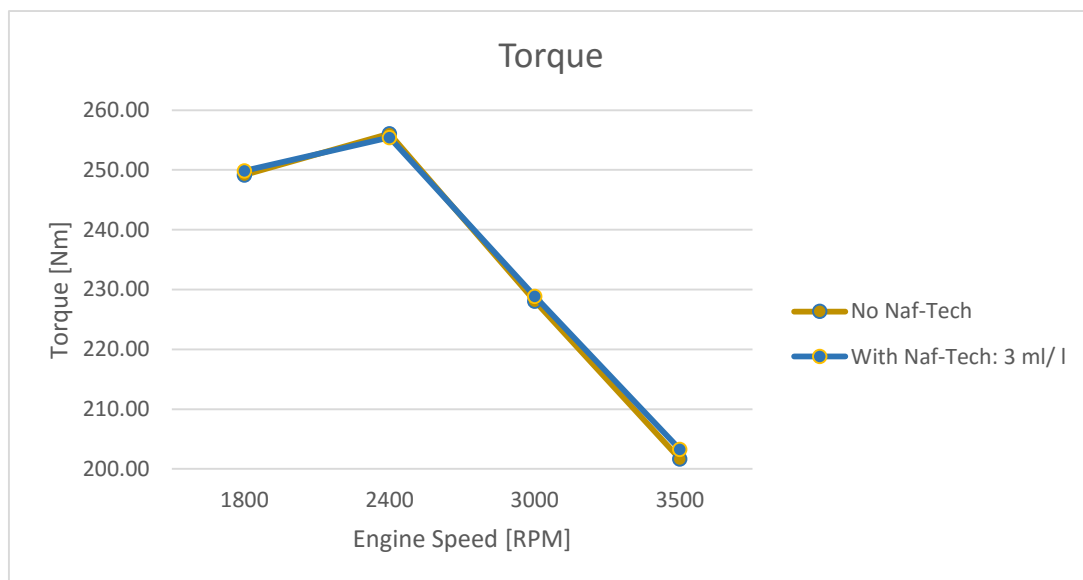


Figure 55 - Torque – graphical results (test set one)

By looking at the above results, it is seen that the power and torque produced by the engine before and after the addition of the fuel additive are exactly the same. This result was expected as the ECU is designed to always deliver a consistent power and torque output for a specific condition. With this being said, the parameter that would change to keep a constant performance curve would be the amount of fuel delivered to the engine.

Fuel consumption

The fuel consumption recorded for the duration of the tests was as follows:



Table 14 - Fuel consumption – tabulated results (test set one)

Fuel consumption [l/h]		
RPM	Without NT	With NT
1800	21,846	17,4581
2400	24,197	22,7013
3000	27,435	21,4606
3500	30,336	29,92

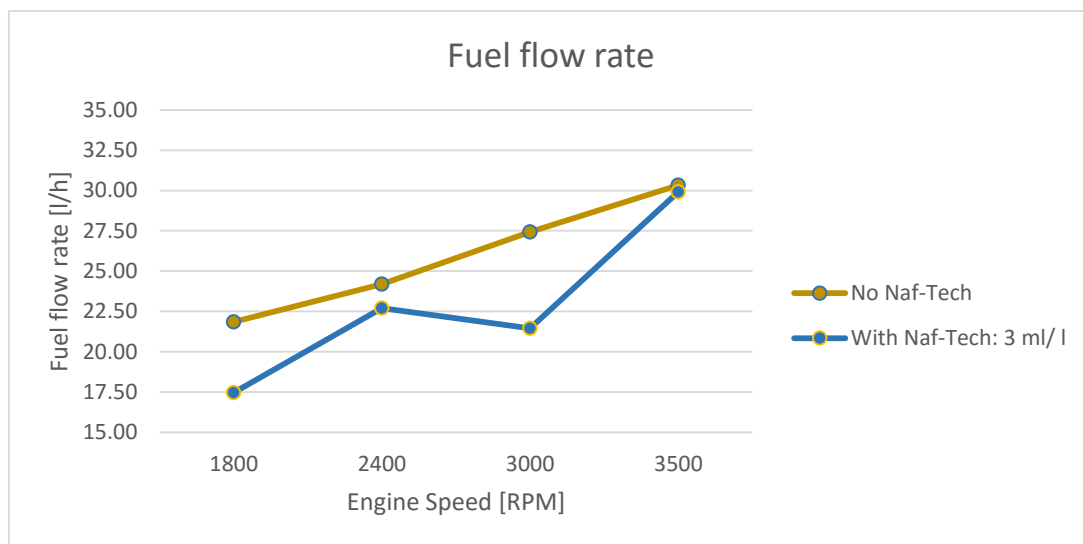


Figure 56 - Fuel consumption – graphical results (test set one)

It is clear from the results above that there is a reduction in fuel consumption with the use of Naf-Tech. The fuel efficiency increase obtained at each RPM during the tests was calculated and is shown in the table below:

Table 15 - Fuel efficiency increase

RPM	%
1800	20,09
2400	6,18
3000	21,78
3500	1,37

The average gain in fuel efficiency over the full range of engine speeds was calculated to be 12.4 %.



Efficiency

Combustion efficiency

Table 16 below summarises the combustion efficiency results obtained from the tests conducted on the engine.

Table 16 - Combustion efficiency – tabulated results (test set one)

Combustion efficiency [%]		
RPM	Without NT	With NT
1800	98,36	98,38
2400	99,11	99,33
3000	99,57	99,69
3500	99,59	99,57

The graph in Figure 57 is a graphical representation of the results above.

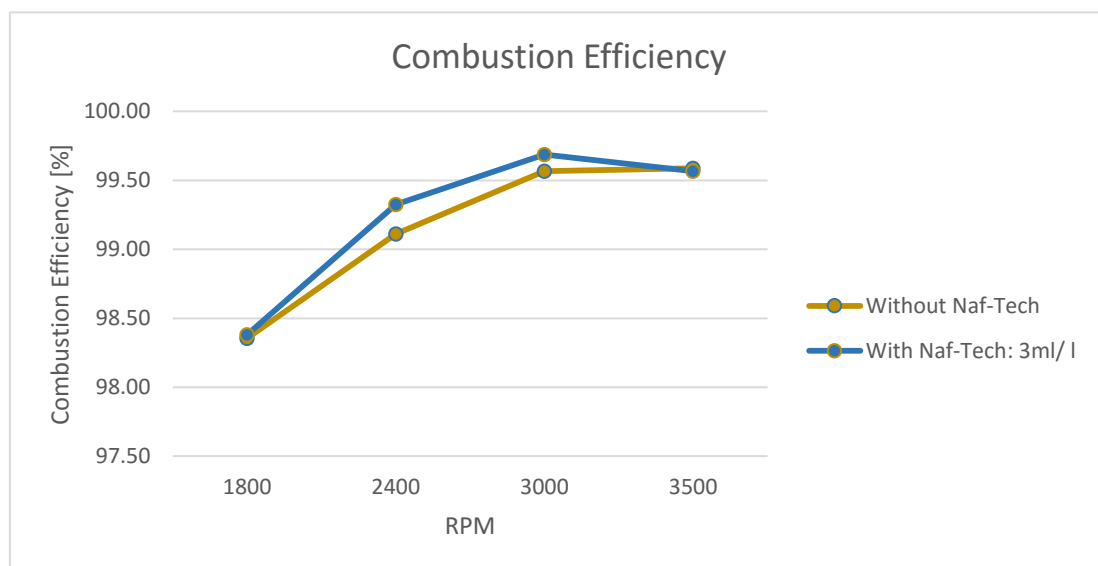


Figure 57 - Combustion efficiency – graphical results (test set one)

As can be seen in the graph in Figure 57 above, the overall combustion efficiency achieved during the tests has been increased by 0.1% with the usage of Naf-Tech. Although it's not a large increase, it would have a significant effect on the engine's performance. This increase means that the combustion process was extracting a higher percentage of the fuel's potential energy with the addition of the additive.



Thermal efficiency

The thermal efficiency of each test is summarised in the table below.

Table 17 - Thermal efficiency – tabulated results (test set one)

Thermal efficiency [%]		
RPM	Without NT	With NT
1800	20,21	25,37
2400	24,9	26,39
3000	24,32	31,12
3500	22,83	23,12

The graph below represents the results from the table above.

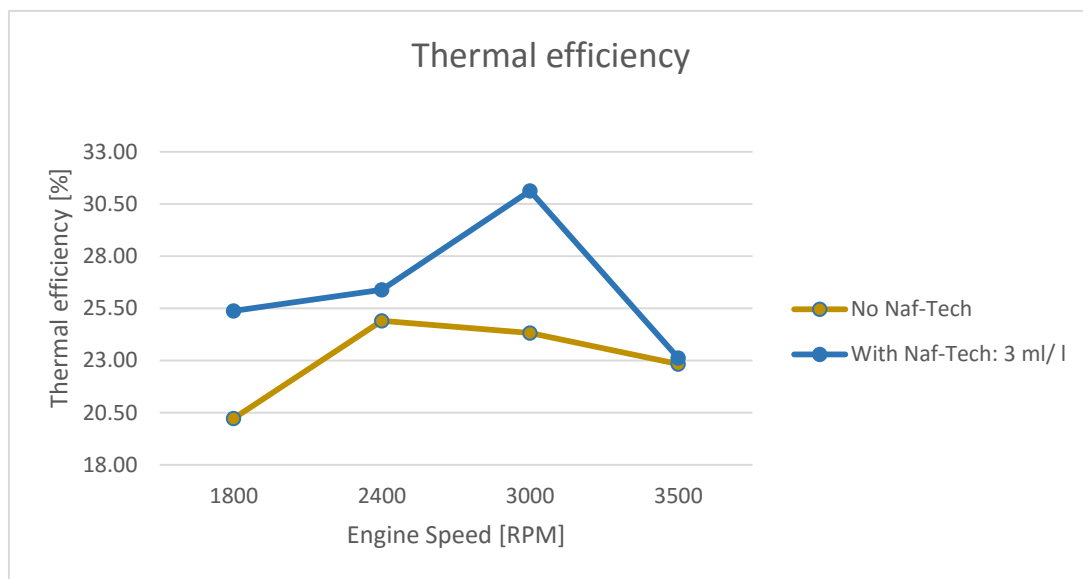


Figure 58 - Thermal efficiency – graphical results (test set one)

The engine's thermal efficiency increased by 12.27% with the use of Naf-Tech which means that the engine was able to extract more of the energy present in the fuel. This result coincides with the results for the combustion efficiency.



Volumetric efficiency

The results obtained from the tests regarding volumetric efficiency are all listed in the table below.

Table 18 - Volumetric efficiency – tabulated results (test set one)

Volumetric efficiency [%]		
RPM	Without NT	With NT
1800	170,34	172,35
2400	166	168,35
3000	157,74	156,95
3500	148,31	145,8

From these results, the graph in Figure 59 was drawn.

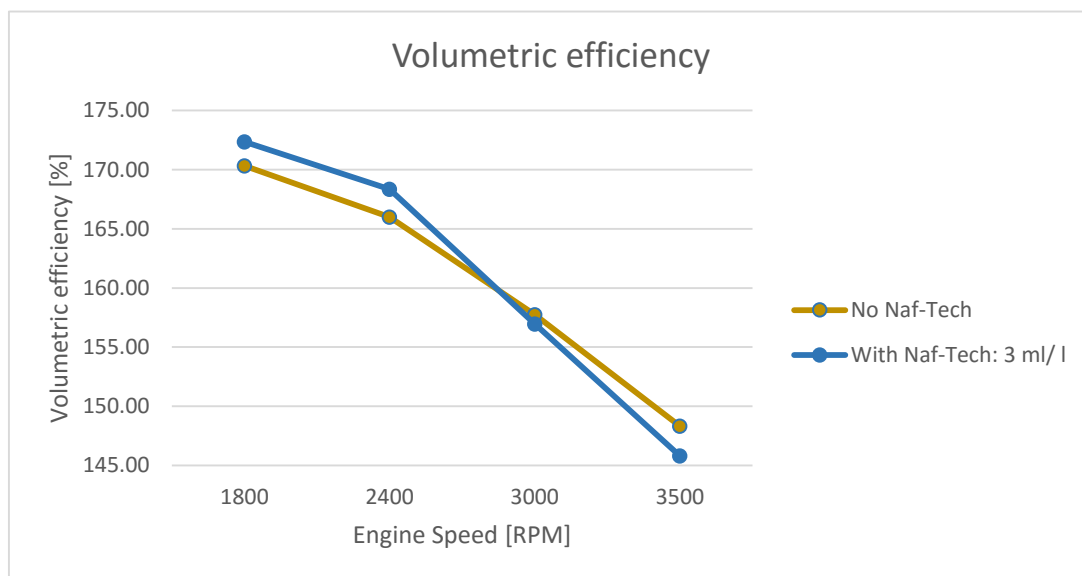


Figure 59 - Volumetric efficiency – graphical results (test set one)

As seen from the above results and graph, the use of Naf-Tech had little to no effect on the volumetric efficiency of the engine. This result was expected as a diesel engine always operates with the same volume of air at a certain speed. In addition to this, Naf-Tech is a fuel additive and would thus only have an effect on aspects that the fuel flow would affect such as thermal and combustion efficiency.



Emission results

By analysing the emissions produced by the engine, it was possible to determine what the effect of Naf-Tech is on the combustion process. Shown below are the results of the exhaust gas analysis.

Table 19 - Carbon monoxide results – tabulated results (test set one)

CO [%]			% Difference
RPM	Without NT	With NT	[%]
1800	0,127	0,115	9,21
2400	0,09	0,065	27,78
3000	0,027	0,015	43,75
3500	0,023	0,022	3,57

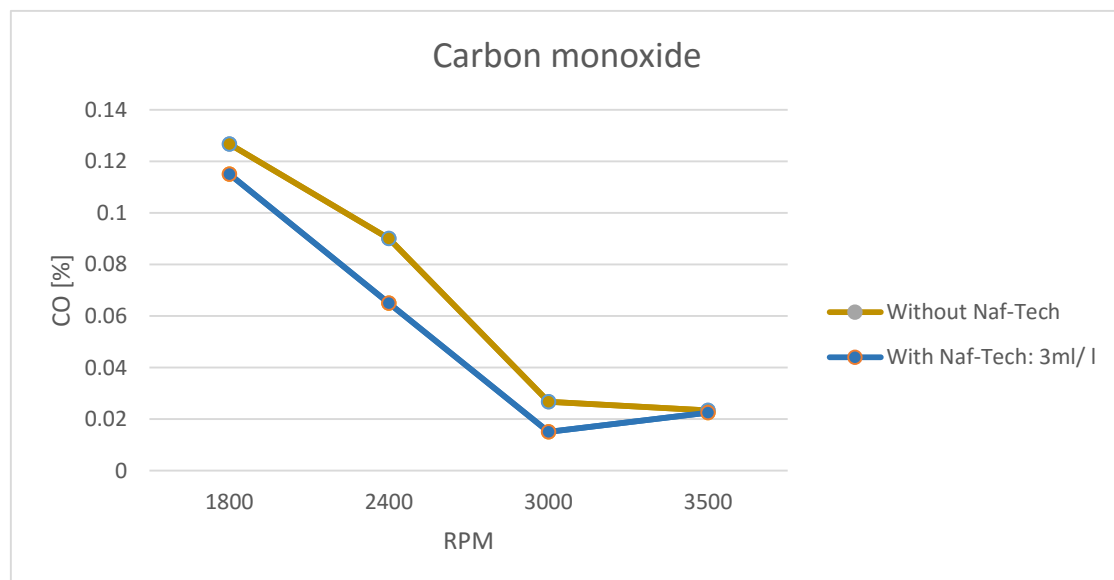


Figure 60 - Carbon monoxide – graphical results (test set one)

With the use of Naf-Tech in the fuel, the CO emissions produced by the engine have been reduced on average by 21%.

The CO₂ results from the gas analyser are shown below.

Table 20 - Carbon dioxide results – tabulated results (test set one)

CO ₂ [%]			% Difference
RPM	Without NT	With NT	[%]
1800	9,5	9,1	4,210
2400	10,05	9,575	4,726
3000	9,967	8	19,732
3500	9,467	7,25	23,415

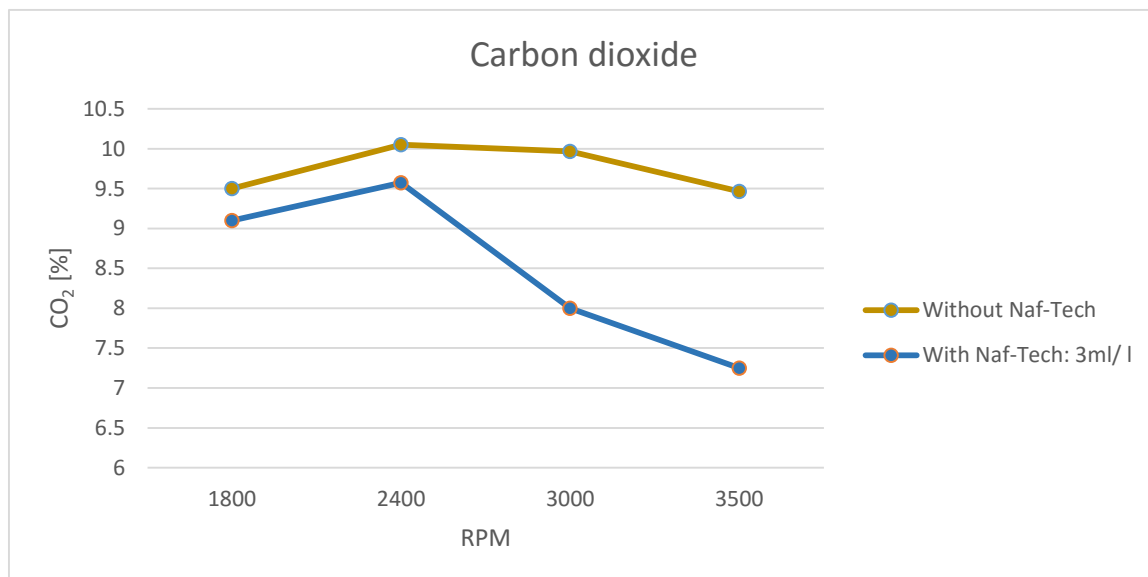


Figure 61 - Carbon dioxide – graphical results (test set one)

There is a significant reduction in CO₂ emissions with an overall decrease of 13% when compared to running the engine without the additive.

The presence of hydrocarbons in the exhaust gas means that a percentage of the fuel is not being burnt during combustion. A reduction in this would mean that a higher percentage of fuel is burnt, and therefore, more of the fuel's energy can be released to provide power to the engine, effectively making the combustion process more efficient.



Table 21 - Hydrocarbon results – tabulated results (test set one)

HC [ppm]			% Difference
RPM	Without NT	With NT	[%]
1800	0,667	0,75	-12.5
2400	0	0	0
3000	0,333	0,25	25
3500	0,333	0,25	25

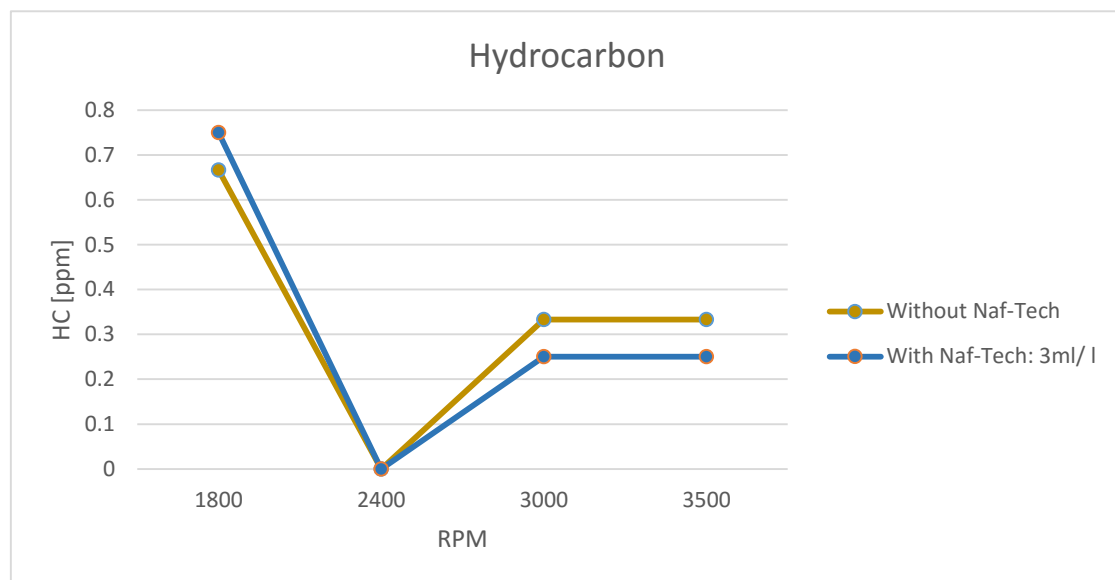


Figure 62 - Hydrocarbon – graphical results (test set one)

As can be seen from Table 21 and Figure 62 above, the fuel additive has a greater effect on the HC at the higher engine speeds. At 1800 RPM, there is a slight increase in unburnt fuel in the exhaust system, but overall, there is a 9.3% reduction in HC emissions between the two tests.

The results below represent the oxygen emissions from the tests.

Table 22 - Oxygen results – tabulated results (test set one)

O ₂ [%]			% Difference
RPM	Without NT	With NT	[%]
1800	6,750	7,610	-12,741
2400	6,035	6,993	-15,866
3000	6,140	9,365	-52,524
3500	7,077	10,668	-50,742

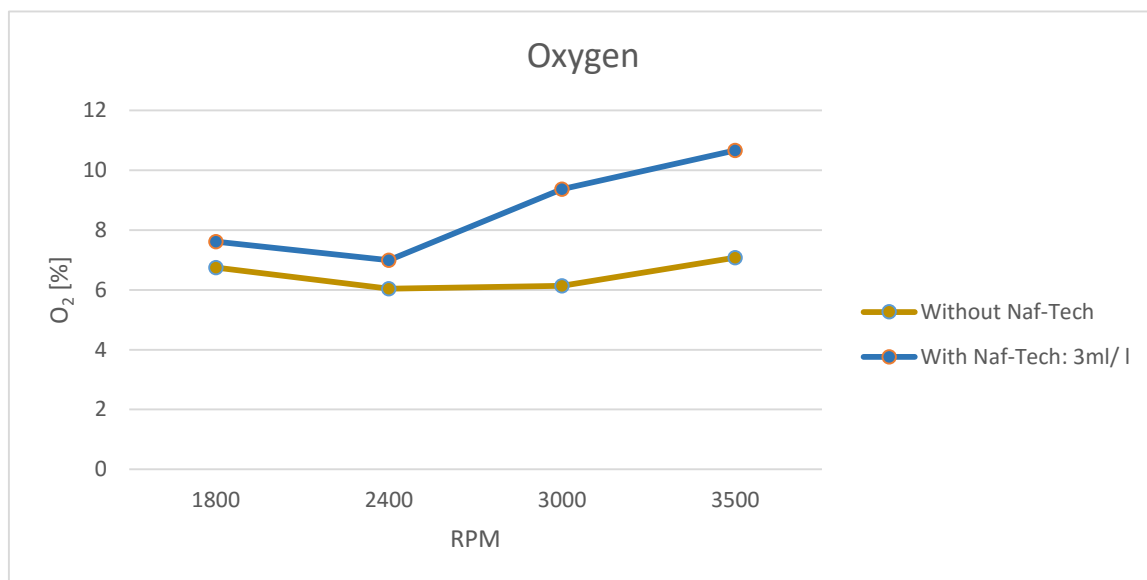


Figure 63 - Oxygen – graphical results (test set one)

As can be seen from the above results, the percentage O₂ released to the exhaust system has increased by 32.9% with the use of the additive. This is due to more of the fuels energy being extracted with the use of the additive, as seen in the combustion and thermal efficiency results thus far, the engine would need less fuel to produce the same output performance. This means that the ECU will automatically reduce the mass flow rate of the fuel to the engine. While less fuel is used, the same volume of air is flowing into the cylinders due to diesel engines always using a full charge of air for compression which would result in a larger amount of O₂ remaining in the exhaust gases.



The NO_x emissions produced throughout the tests are summarised below.

Table 23 - NO_x results – tabulated results (test set one)

NO _x [ppm]			% Difference
RPM	Without NT	With NT	[%]
1800	578,333	522,250	9,697
2400	485,500	447,500	7,827
3000	410,333	288,250	29,752
3500	305,000	196,750	35,492

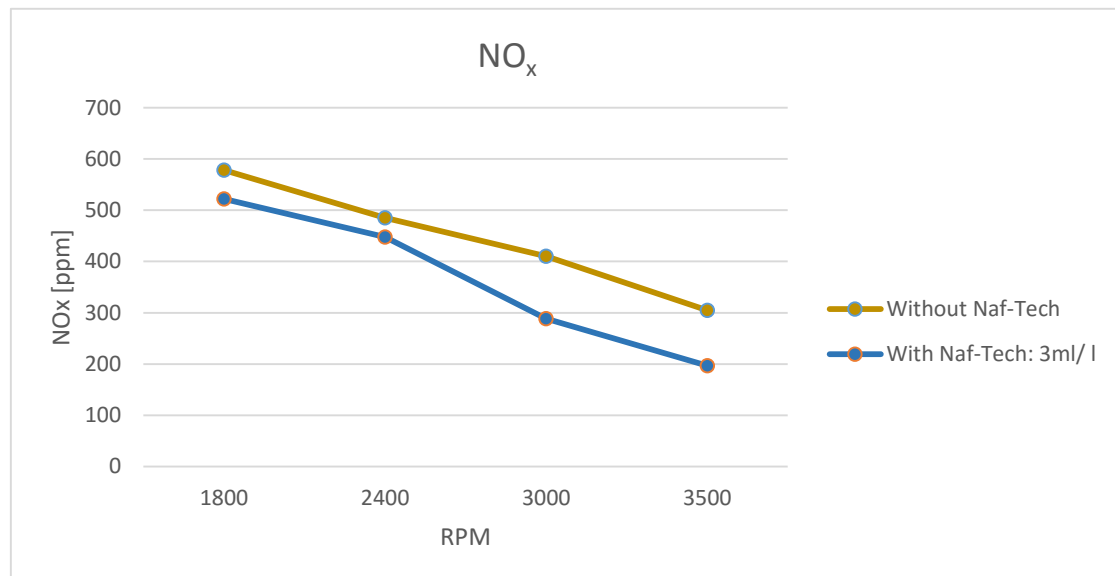


Figure 64 - NO_x – graphical results (test set one)

With an overall decrease of 20.69% in NO_x emissions, it is clear that the additive has a positive effect on the formation of nitrogen oxides during combustion.



Energy losses/ transfers

Of the total amount of energy put into the engine from the fuel, the majority of it is lost to heat that is always removed from the system. The energy losses in the results below were converted to the units of power [kW] for the purpose of easily comparing them to the performance (power output) of the engine.

The table and graph below represent the losses to the engine's cooling system.

Table 24 - Power transferred to cooling system – tabulated results (test set one)

Power transferred to cooling system [kW]		
RPM	Without NT	With NT
1800	32,65	38,30
2400	33,04	36,35
3000	32,42	35,28
3500	36,83	30,93

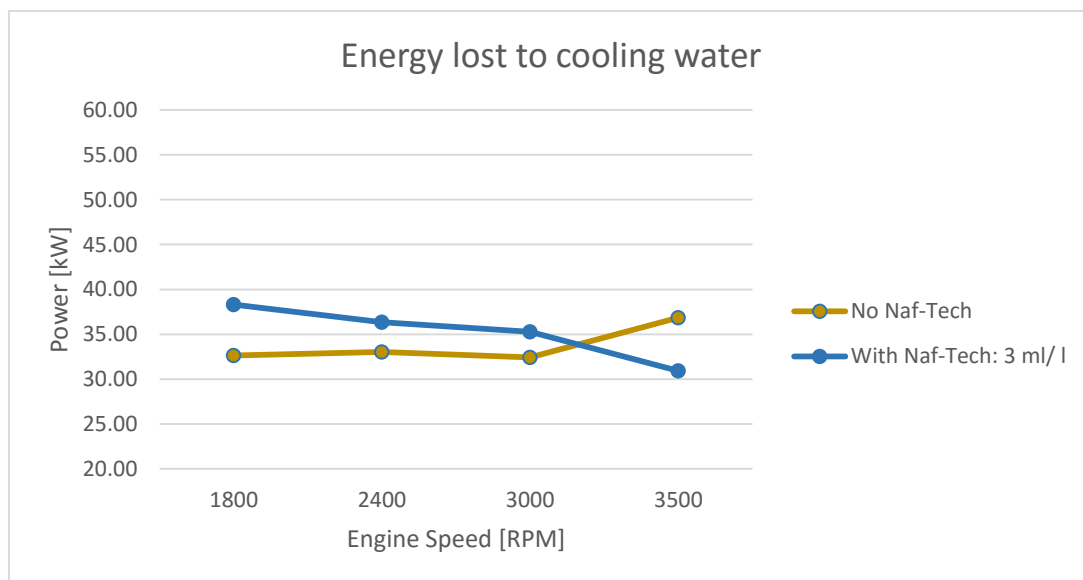


Figure 65 - Power transferred to cooling system – graphical results (test set one)

The results above indicate that the energy transferred to the cooling system did not change by much at various engine speeds.



The results below indicate the power lost through heat transfer to the exhaust gas.

Table 25 - Power transferred to exhaust heat – tabulated results (test set one)

Power transferred to exhaust heat [kW]		
RPM	Without NT	With NT
1800	21,46	28,13
2400	30,42	37,27
3000	38,01	40,89
3500	38,08	39,70

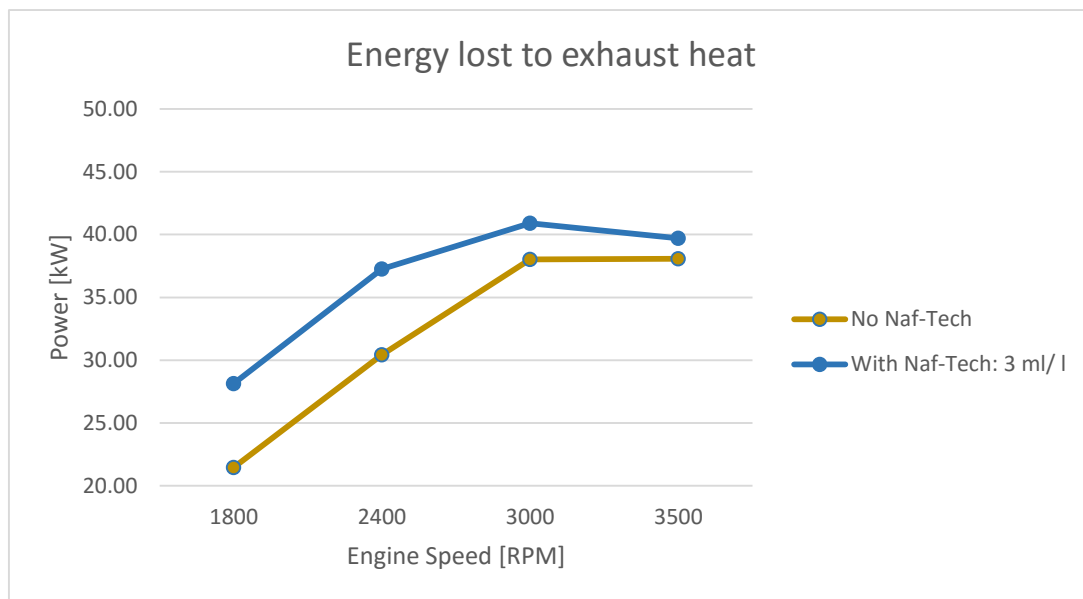


Figure 66 - Power transferred to exhaust heat – graphical results (test set one)

By observing the results in Figure 66 above, it is seen that the energy loss increases as the engine speed rises. It can also be seen that the energy lost to the exhaust is 13% higher with the use of Naf-Tech.

The power that was dissipated to the dynamometer water is depicted below. The values indicated here are similar to the output power produced by the engine. This is because the dynamometer that was used is a water brake that converts all the engines shaft power to heat which is removed by the cooling water.

Table 26 - Power dissipated in dynamometer water– tabulated results (test set one)

Power dissipated to dyno water [kW]		
RPM	Without NT	With NT
1800	47,23	49,50
2400	64,84	66,89
3000	70,95	67,34
3500	73,42	69,82

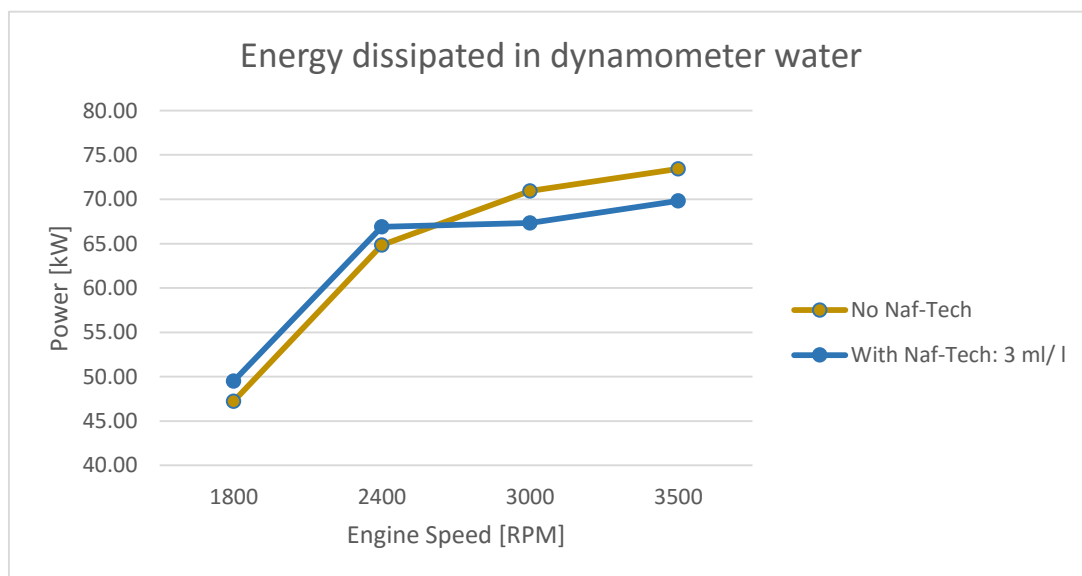


Figure 67 - Power dissipated in dynamometer water – graphical results (test set one)

If the results in Figure 67 above are compared to the graph in Figure 54, it can be seen that there is a close resemblance to one another as both sets of results represent the engine's shaft power. The results shown in Figure 54 are obtained from a load cell readout that mechanically measures the torque produced by the engine while the results in the graph above are calculated from the heat dissipated in the dynamometer's water by resisting the engine's rotational force.



5.4.2 Test set two

The second set of tests conducted on the engine was the same as the first, with the addition of upgraded electronic logging equipment. These were necessary for increased accuracy as well as to remove the human error caused by manual readings. The only difference in the test procedure between the two test sets was the engine speed conditions at which the tests were run. The first set of tests was run at 1800, 2400, 3000 and 3500 RPM, where the second set's tests were conducted at the following RPM conditions: 1800, 2200, 2600, 3000 and 3400 RPM.

Various concentrations of the additive were investigated for the tests in the second set i.e.: 0ml Naf-Tech/ l diesel, 2.5 ml Naf-Tech/ l diesel, 3 ml Naf-Tech/ l diesel and 3.5 ml Naf-Tech/ l diesel. Each of the curves on the graphs below represents one of the above-mentioned Naf-Tech concentrations.

Unfortunately, shortly after the tests had begun, it was noticed that the engine was not performing as it should. The condition of the engine gradually deteriorated as the tests continued which can be seen when comparing the results of this section to the previous set of tests. An expert's opinion regarding the cause of the engine problems is discussed fully in the conclusion.

Considering the above issues with the engine, the following results, although extremely accurate, cannot be used to test the effect that Naf-Tech has on performance when compared to the baseline diesel runs.



Torque and Power

The results obtained for the power produced by the engine during the tests are depicted in Table 27 and Figure 68 below.

Table 27 - Power – tabulated results (test set two)

Power [kW]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	50,47	49,50	49,40	48,41
2200	60,30	60,59	60,43	57,89
2600	68,40	70,83	69,67	66,20
3000	73,28	72,54	72,41	69,24
3400	73,36	73,70	72,73	70,26

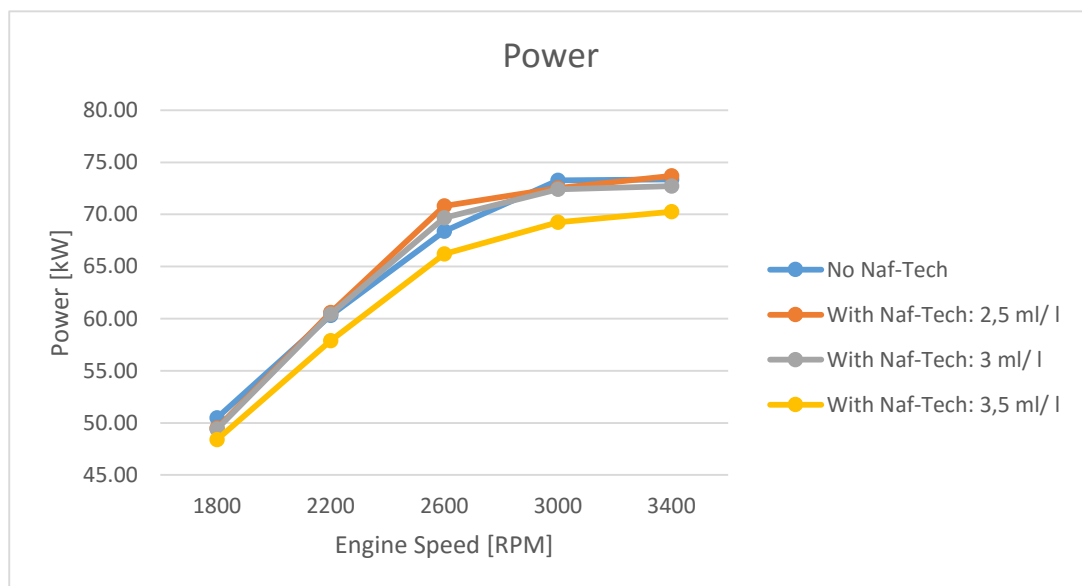


Figure 68 - Power – graphical results (test set two)

The torque results produced by the engine for each of the Naf-Tech concentrations are shown below.



Table 28 - Torque – tabulated results (test set two)

Torque [Nm]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	270,23	263,11	263,02	256,36
2200	265,52	264,46	264,55	251,01
2600	263,11	261,92	257,29	243,53
3000	237,17	233,69	231,82	220,47
3400	208,88	207,13	208,12	199,31

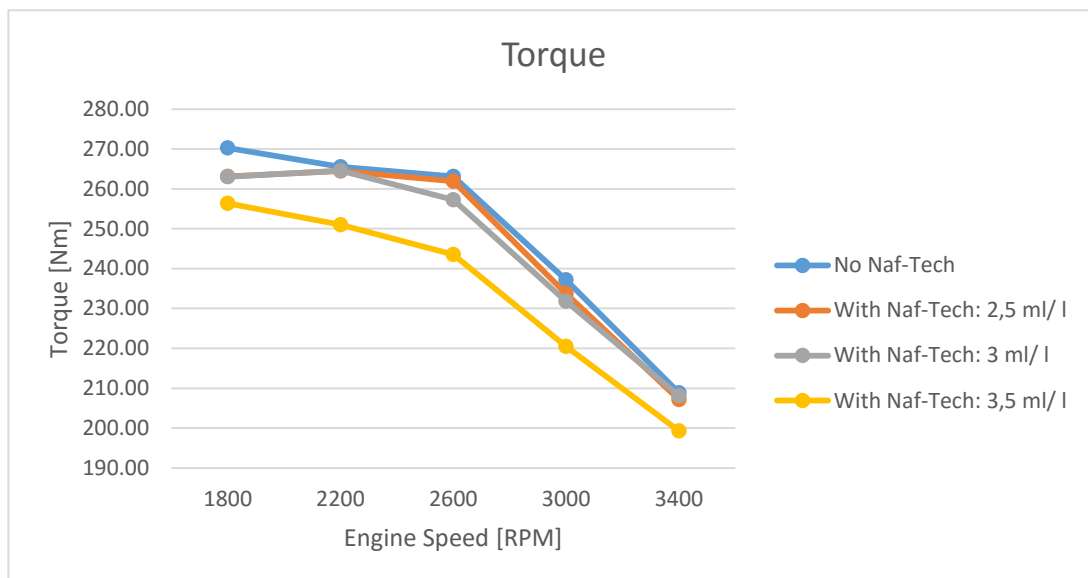


Figure 69 - Torque – graphical results (test set two)

By looking at the above graphs, it is seen that the performance of the final test i.e. the 3,5 ml/l (yellow) test, has been reduced by approximately 4% due to the extent of the engine's rapid deterioration at this point. It can also be seen from the graphs that, although the engines condition had been gradually worsening from the initial test, the first three tests once again produced similar performance results regardless of the Naf-Tech mixing ratio. The similarities in these results are due to the ECU altering the input parameters of the engine to counter the effect Naf-Tech as well as those caused by the faults.

Due to a rapid decline in the condition of the engine after the third test, the ECU was not able to keep the output parameters constant any longer.

Fuel consumption

The table below includes the fuel consumption results for each of the Naf-Tech mixing ratios under all the conditions tested.

Table 29 - Fuel consumption – tabulated results (test set two)

Fuel consumption [l/h]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	14,49	14,14	14,45	14,26
2200	17,81	18,04	18,17	17,56
2600	20,71	21,95	21,55	21,04
3000	23,41	23,43	23,47	22,89
3400	25,25	25,87	25,25	25,28

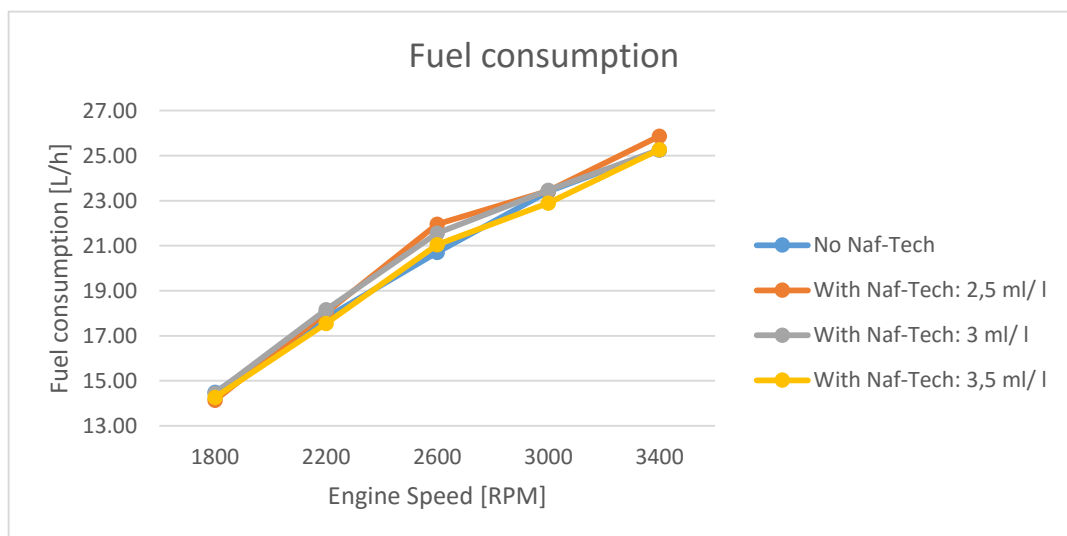


Figure 70 - Fuel consumption – graphical results (test set two)

As seen in Figure 70 above, the fuel consumption remained relatively constant for each of the four tests. The reason for this being that on the one hand, the ECU would be decreasing the fuel consumption as a result of adding Naf-Tech to the fuel as seen in Figure 56 where a 12% reduction in fuel consumption was observed with the use of the fuel additive. On the other hand, the ECU would be increasing the fuel consumption to counter the negative effects caused by the engines deteriorating condition. This would result in the fuel consumption remaining relatively constant for each of the tests conducted.



Efficiency

Combustion efficiency

The combustion efficiencies for each of the Naf-Tech concentrations tested are summarised in the table and graph below.

Table 30 - Combustion efficiency – tabulated results (test set two)

Combustion efficiency [%]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	97,76	98,37	98,39	94,11
2200	98,37	98,76	98,45	98,72
2600	98,26	99,32	99,33	99,29
3000	97,70	98,71	99,65	99,41
3400	99,78	95,23	99,55	98,42

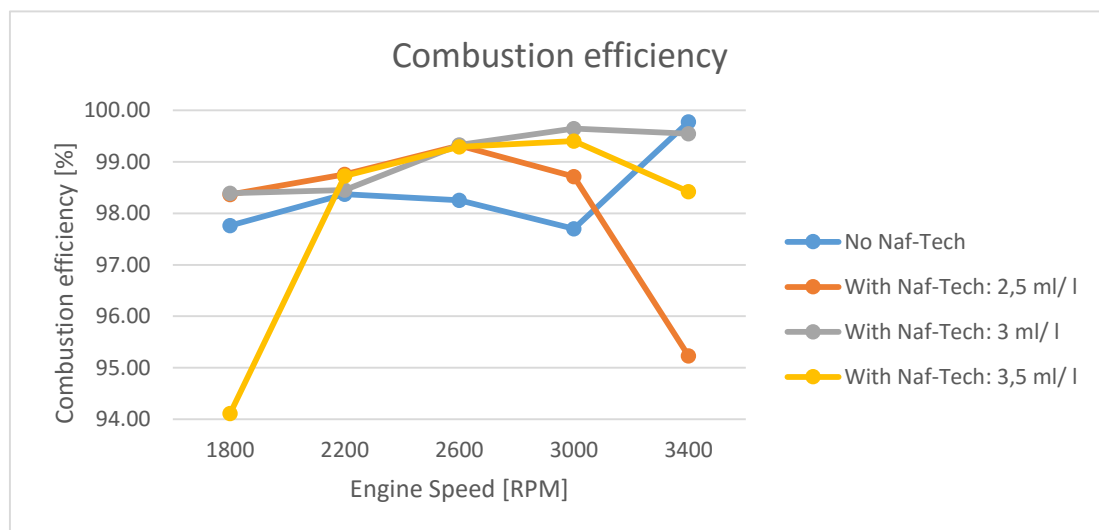


Figure 71 - Combustion efficiency – graphical results (test set two)

As shown in Figure 71 above, the combustion efficiency curve is not as consistent as the curve produced by the results from test set one (Figure 57).



Thermal efficiency

The thermal efficiency results gathered from the tests can be seen in Table 31 and Figure 72 below.

Table 31 - Thermal efficiency – tabulated results (test set two)

Thermal efficiency [%]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	32,95	32,90	32,14	33,41
2200	31,83	31,46	31,25	31,16
2600	31,09	30,05	30,09	29,30
3000	29,65	29,01	28,63	28,14
3400	26,93	27,72	26,75	26,12

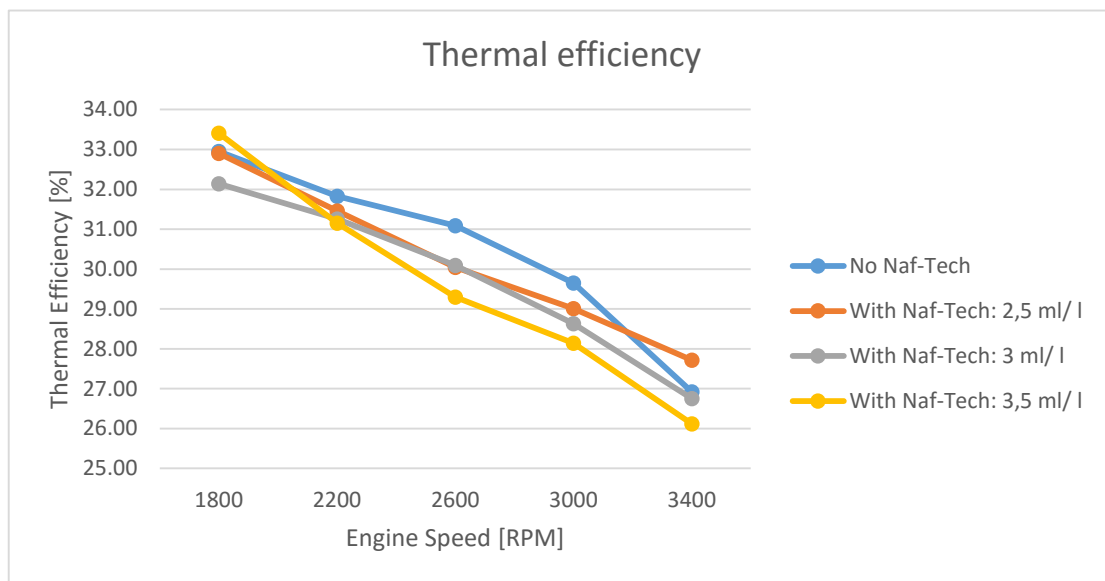


Figure 72 - Thermal efficiency – graphical results (test set two)

The above thermal efficiency results remain very similar for each test. This is caused by the same reason the fuel consumption in Figure 70 remains constant for each of the tests.



Volumetric efficiency

The volumetric efficiency results for test set two are shown below.

Table 32 - Volumetric efficiency– tabulated results (test set two)

Volumetric efficiency [%]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	147,83	147,15	147,64	146,30
2200	149,38	150,62	151,08	145,43
2600	146,91	148,54	146,82	143,39
3000	142,74	142,85	142,20	139,19
3400	135,96	133,98	135,00	133,50

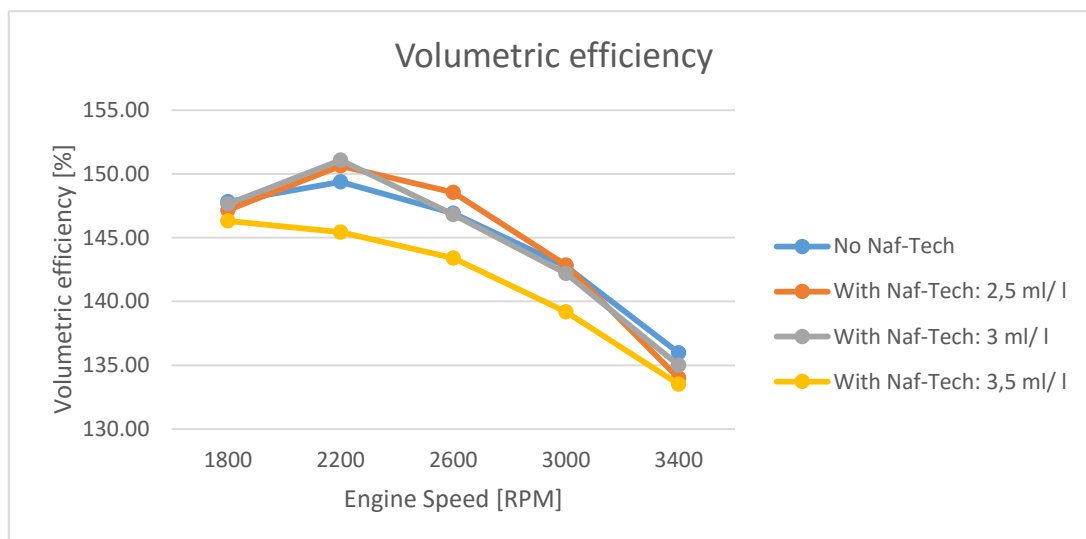


Figure 73 - Volumetric efficiency – graphical results (test set two)

The volumetric efficiency of a diesel engine will always remain similar, as depicted in the graph above, unless the air flow has been affected for some reason, as was the case with the final test i.e. 3.5 ml/l (yellow). In the case of this engine, there were many leaks at the exhaust gasket as well as the turbo casing which means that a large volume of gas is being lost to the atmosphere before it passes through the turbo's turbine blades. This prevents the turbo from reaching the speed it was designed to reach for a specific engine condition. The reduction in turbo compressor speed means that it would not be able to force as much air into the cylinders as what it normally would, thus reducing the volumetric efficiency and power of the engine.



Emissions

All the emission data obtained from the tests are summarised in the tables and graphs below.

The carbon monoxide results gathered from test set two can be seen in Table 33 and Figure 74 below.

Table 33 - Carbon monoxide – tabulated results (test set two)

CO [%]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	0,02	0,03	0,02	0,03
2200	0,03	0,03	0,03	0,02
2600	0,07	0,06	0,06	0,06
3000	0,10	0,10	0,11	0,12
3400	0,14	0,12	0,14	0,15

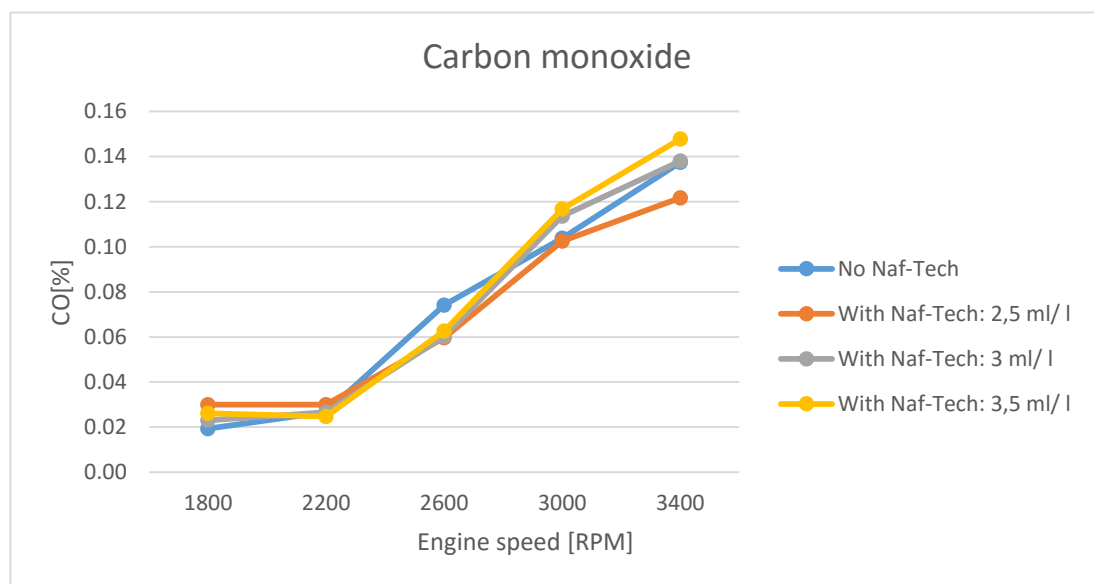


Figure 74 - Carbon monoxide – graphical results (test set two)

The results indicate that the percentage CO produced by the combustion process remained similar for each of the tests conducted. The production of CO increases at the higher engine speeds.



Table 34 and Figure 75 below represent the results of the carbon dioxide emissions.

Table 34 - Carbon dioxide – tabulated results (test set two)

CO ₂ [%]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	8,71	9,44	9,53	9,45
2200	9,04	9,27	9,28	9,41
2600	9,14	9,26	9,44	9,70
3000	8,79	8,76	8,95	9,23
3400	8,55	8,41	8,45	8,72

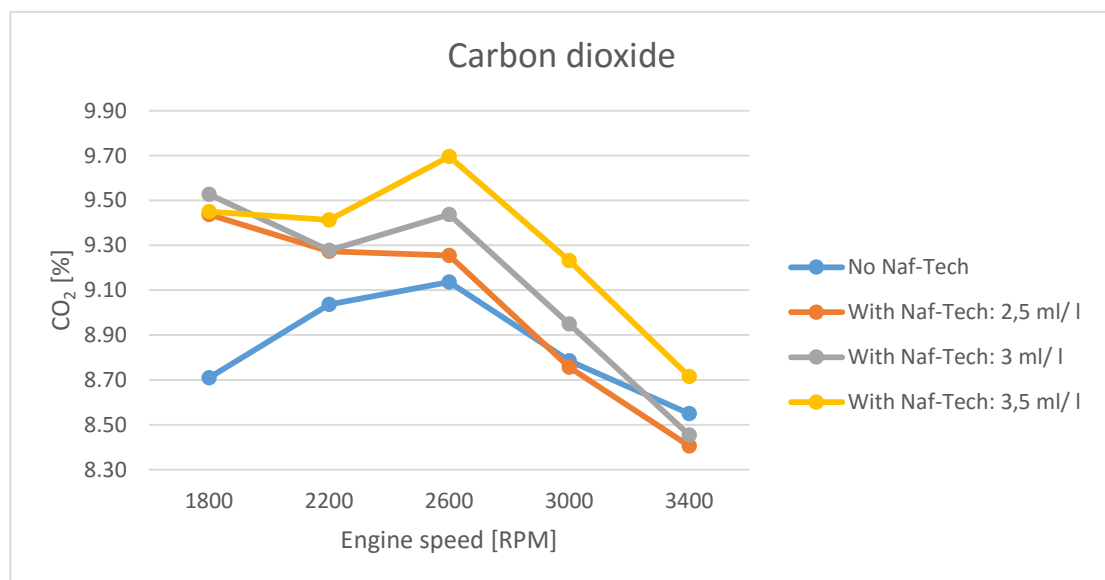


Figure 75 - Carbon dioxide – graphical results (test set two)

From the results above it can once again be seen that as the tests progressed, the condition of the engine deteriorated, thus increasing the formation of CO₂ emissions. It is also seen that as the speed of the engine increases, the levels of CO₂ gradually increase before decreasing at about 2600 RPM

The results for the hydrocarbons present in the exhaust gas are shown below.

Table 35 - Hydrocarbon emissions – tabulated results (test set two)

HC [ppm]				
RPM	0 ml Naf-Tech	2,5ml Naf-Tech	3 ml/ Naf-Tech	3,5 ml Naf-Tech
1800	0,00	13,50	0,42	2,61
2200	4,03	1,93	0,13	0,67
2600	1,89	0,08	0,07	0,13
3000	0,93	0,17	0,60	0,06
3400	1,33	0,42	0,00	8,58

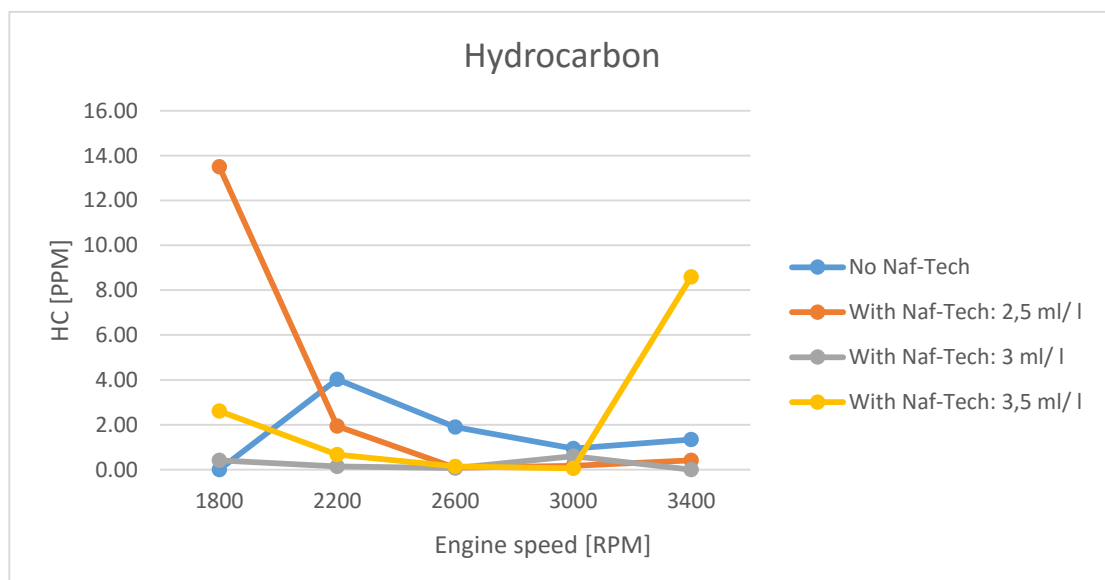


Figure 76 - Hydrocarbon emissions – graphical results (test set two)

The level of HCs in the exhaust system is relatively low when considering that 200 ppm HC is equivalent to $\approx 1\%$ of fuel remaining unburnt and being passed right through the engine.

The percentage of oxygen remaining after the combustion process during each of the tests is summarised in the table and graph below.

Table 36 - Oxygen emissions – tabulated results (test set two)

O ₂ [%]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	8,31	7,41	7,09	7,18
2200	7,81	7,47	7,36	7,15
2600	7,59	7,45	7,03	6,67
3000	8,09	8,02	7,80	7,35
3400	8,41	8,56	8,37	8,13

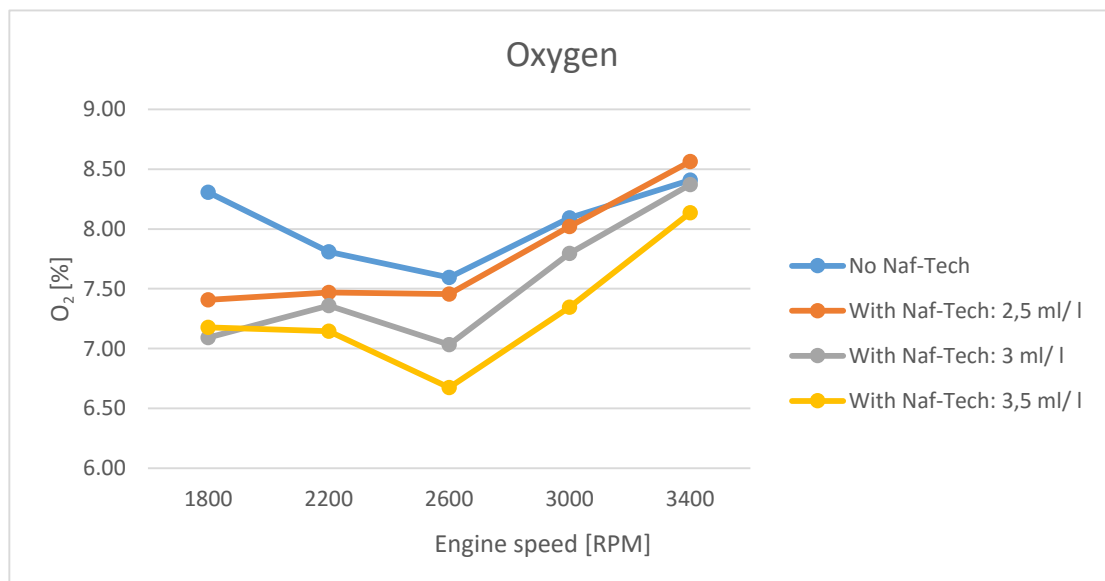


Figure 77 - Oxygen emissions – graphical results (test set two)

As seen from the graph in Figure 77, the quantity of oxygen being released into the exhaust system has decreased from test to test as time continued. When comparing the above graph to the CO₂ results (Figure 75) it is seen that the two gases are indirectly proportional to one another.

The NO_x emissions that were formed during the combustion process for each of the tests are displayed below.

Table 37 - NO_x emissions – tabulated results (test set two)

NO _x [ppm]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	296,56	357,13	349,28	371,19
2200	402,97	418,68	422,81	408,14
2600	439,28	472,88	475,33	487,42
3000	497,75	508,52	487,67	530,79
3400	533,00	563,00	526,70	531,02

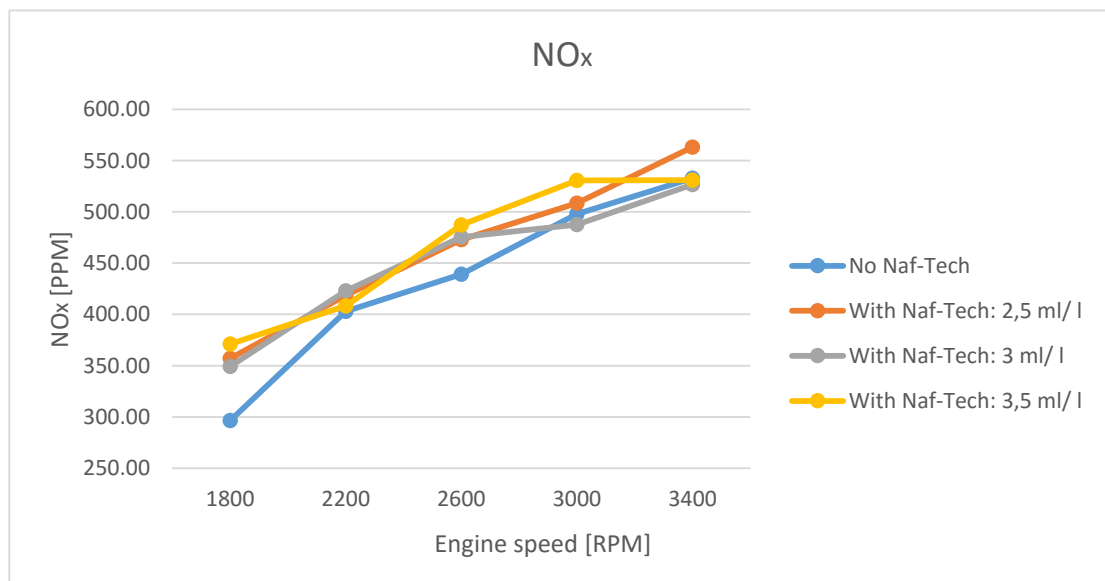


Figure 78 - NO_x emissions – graphical results (test set two)

The above results indicate that the NO_x emissions were gradually increasing from one test to the next. The amount of NO_x gas produced is much higher at the higher engine speeds.



Energy losses/ transfers

The results in Table 38 and Figure 79 below summarise the losses due to heat extraction by the engine's cooling system.

Table 38 - Power transferred to cooling system – tabulated results (test set two)

Energy transferred to cooling system [kW]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	37,72	36,19	38,11	37,40
2200	43,88	44,71	44,74	44,72
2600	47,59	51,31	51,47	50,78
3000	52,85	55,98	55,35	54,72
3400	55,00	56,37	58,23	58,98

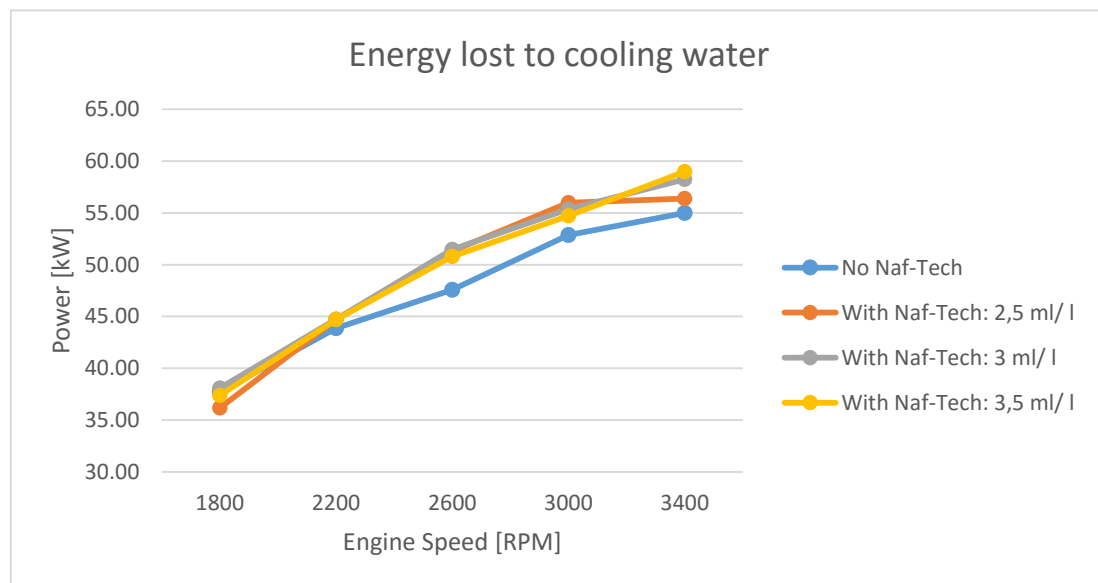


Figure 79 - Power lost to cooling system – graphical results (test set two)

In contradiction to Figure 65 (the equivalent to the above results from test set one) where the power transferred to the cooling water remains relatively constant throughout the engine's speed range, Figure 79 above indicates that the cooling system absorbs a much larger percentage of the heat produced by the engine as the speed increases.



This increase in heat absorbed is due to the cooling system upgrade by including a second radiator as well as a large turbofan to force cool air through the main radiator. These upgrades made it possible for the cooling system to remove a larger amount of the heat produced by the engine when compared to test set one and thereby keeping it well within its safe operating temperature and allowing much longer test durations.

The table and graph below represent the results obtained from test set two for the power lost to the exhaust gas.

Table 39 - Power lost to exhaust heat – tabulated results (test set two)

Power lost to exhaust heat [kW]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	22,93	21,97	22,91	23,03
2200	30,16	30,94	31,00	29,87
2600	37,54	40,42	40,13	39,21
3000	46,72	46,49	46,54	46,16
3400	52,74	53,82	52,43	53,85

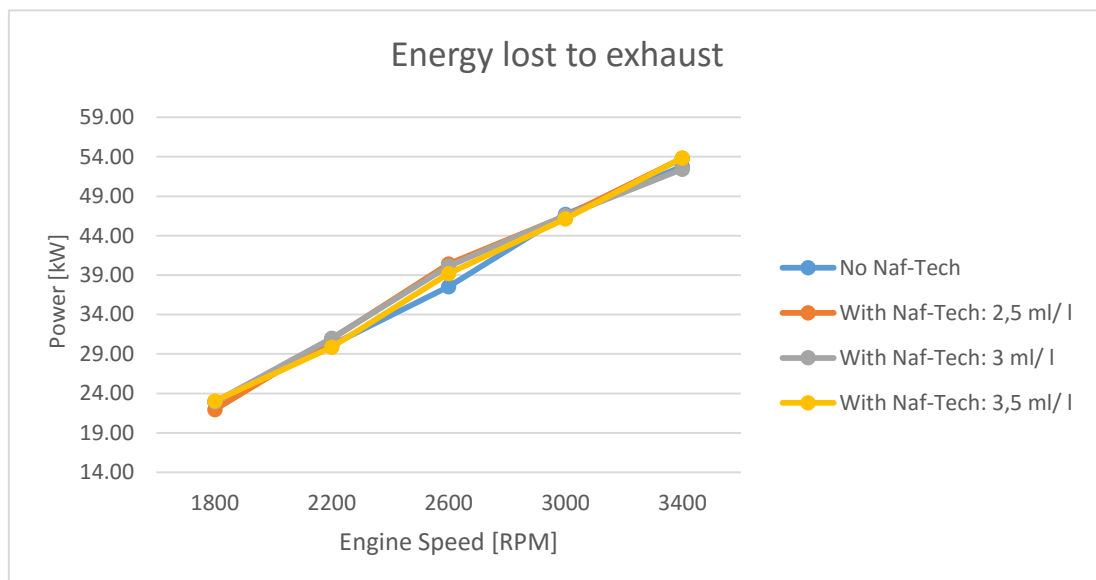


Figure 80 - Power lost to exhaust heat – graphical results (test set two)



Figure 80 above shows that, regardless of problems with the engine or the mixing ratio of Naf-Tech, there is no variation with regards to the energy lost to the exhaust gases.

The results regarding the power transferred to the dynamometer's cooling water are provided below.

Table 40 - Power dissipated to dynamometer water – tabulated results (test set two)

Power dissipated to dyno water [kW]				
RPM	0 ml NT	2,5ml NT	3 ml/ NT	3,5 ml NT
1800	49,73	46,91	51,68	52,47
2200	59,78	59,92	61,45	55,96
2600	68,71	72,36	69,01	65,07
3000	75,01	73,92	73,06	69,69
3400	76,90	79,60	75,14	73,52

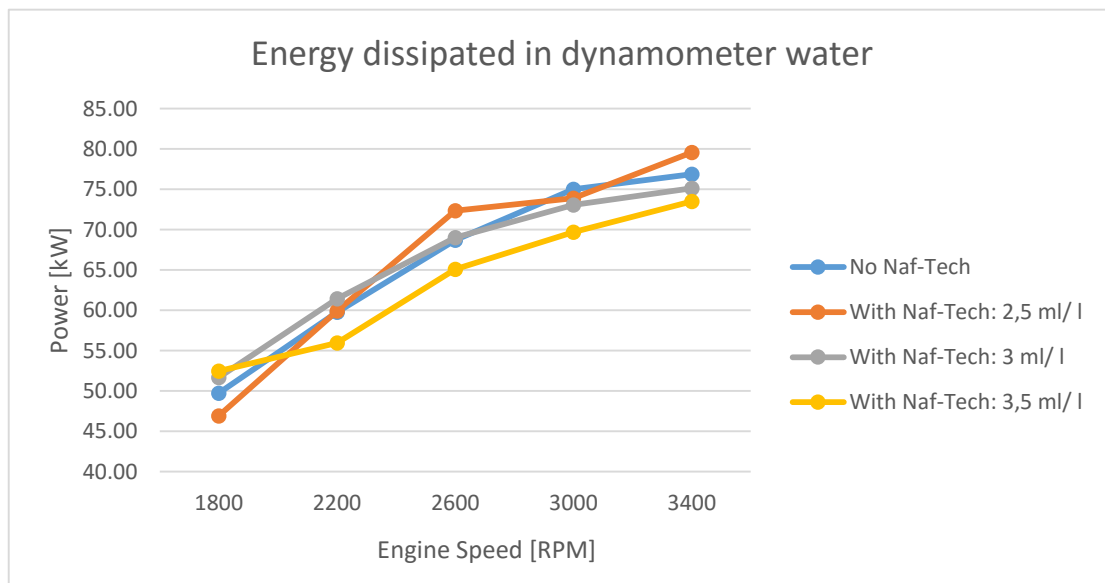


Figure 81 - Power dissipated to dynamometer water – graphical results (test set two)

As can be seen in the figure above, the curves for each of the Naf-Tech mixing ratios are similar with the exception of the final 3.5 ml/l test. These results correspond to the shaft power produced by the engine in Figure 68 for the same set of tests.

As mentioned in the beginning of subsection 5.4.2, shortly after the second set of tests had begun on the GWM test bench, it was noticed that the engine was not performing as it should. The operating temperature was gradually increasing for each of the tests and eventually the engine would not run under full load at 1800 RPM anymore. It was found that there was a large leak in the turbo casing as well as the exhaust manifold's bolts were loose and falling out which caused many leaks in the exhaust system. These leaks had a negative effect on the speed at which the turbo's turbine spun thus reducing the power that the engine was able to produce at a specific engine speed. Once the bolts were refitted and tightened to the recommended torque setting, the engine's performance increased slightly although there was still the leak in the turbo casing as well as past the exhaust gasket as the seal had been broken when the bolts became loose. It was, however, possible to complete the full set of tests before the engine deteriorated to an extent where it would no longer be usable.

Upon conclusion of the tests, the engine was removed from the site for inspection and repairs. The result of the inspection revealed that the exhaust manifold gasket was damaged, and a leak was present at the joint of the turbo casing. Evidence of overheating was found upon removal of the cylinder head and was ascribed to the failure of the engine. In addition to this, scoring was also found on the wall of the second cylinder.

Once the cause of the engine's problems were known to be due to overheating, all the data regarding the engine temperatures was analysed to determine when the overheating occurred. It was found that the engine was frequently run above normal operating temperature during the tests conducted in 2014 shortly after it had been commissioned at NWU. As a result of the prolonged use of the engine with these issues, its condition was increasingly deteriorating as the tests commenced which eventually caused the ECU to enter "limp mode". This means that the ECU limits the performance and speed of the engine to protect itself from further harm.

Certain results from the second set of tests coincide with the above opinion of the expert such as the volumetric efficiency that verifies the presence of the leaks in the turbo casing and exhaust system. It is seen from the results that the volumetric efficiency had decreased for the final test which was due to the turbo's reduction in speed. This meant that a smaller volume of air was entering the engine, thus reducing the power output. These faults will prevent any conclusions to be made specifically regarding any effects that Naf-Tech has on engine performance.



5.5 EES Simulation

The results obtained from running the mathematical model are summarised in this section. Only the significant results corresponding to the engine's performance are displayed, whereas the remainder of the results along with the EES code can be found in Appendix A.

Where possible, the simulation results are compared to the engine specs that are available from its technical manual. By doing this, it gives a better idea of the accuracy of the model

5.5.1 Power and Torque

The power and torque values obtained from the simulated EES model along with the same parameters from the engine's technical manual are shown below in the two tables and Figure 82.

Table 41 - Simulated power and torque – tabulated results

Simulated power and torque		
RPM	Power [kW]	Torque [Nm]
1800	53,44	283,50
2200	65,12	282,70
2600	73,68	270,60
3000	77,88	247,90
3400	76,74	215,50

Table 42 - Spec sheet power and torque – tabulated results

Spec sheet power and torque		
RPM	Power [kW]	Torque [Nm]
1800	56,50	300,00
2200	69,10	300,00
2600	77,60	285,00
3000	80,40	256,00
3400	79,00	222,00

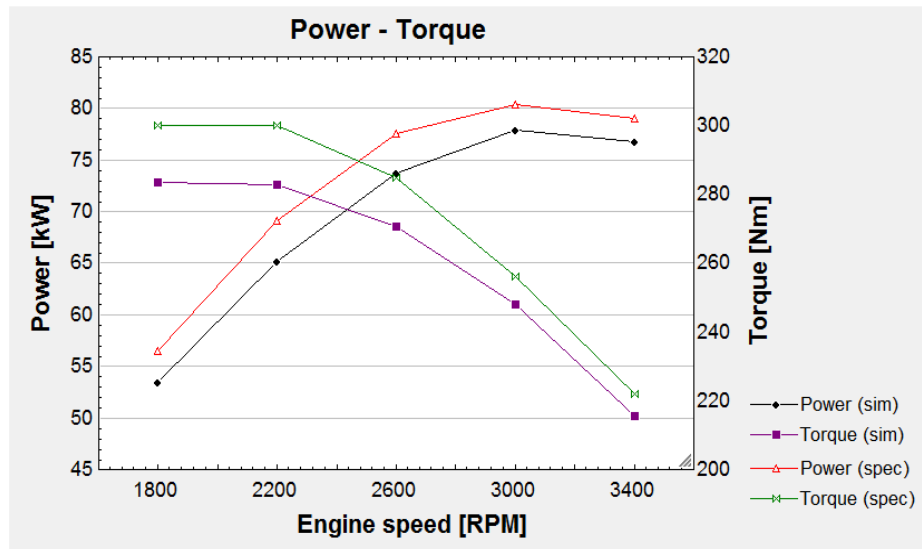


Figure 82 - Simulated power and torque – graphical results

When comparing the simulated values to the values taken from the manual, it is seen that the model is relatively accurate with a maximum error between the two power curves of 5.4% and 5.5% for the torque curves.

5.5.2 Fuel consumption

As with the power and torque results, the fuel consumption figures have been compared to that of the engine's technical manual. The fuel usage results have been represented as fuel consumption [L/h] and specific fuel consumption [g/kW.h] in the results below.

Table 43 - Simulated fuel consumption – tabulated results

Fuel consumption		
RPM	Fuel consumption [l/h]	SFC [g/kW.h]
1800	14,85	230,60
2200	18,75	239,00
2600	22,73	256,10
3000	25,28	269,40
3400	27,04	292,50



Table 44 - Spec sheet fuel consumption – tabulated results

Spec sheet fuel usage

RPM	Fuel consumption [l/h]	SFC [g/kW.h]
1800	15,06	218,00
2200	18,91	227,00
2600	21,93	234,00
3000	24,10	248,00
3400	25,30	261,00

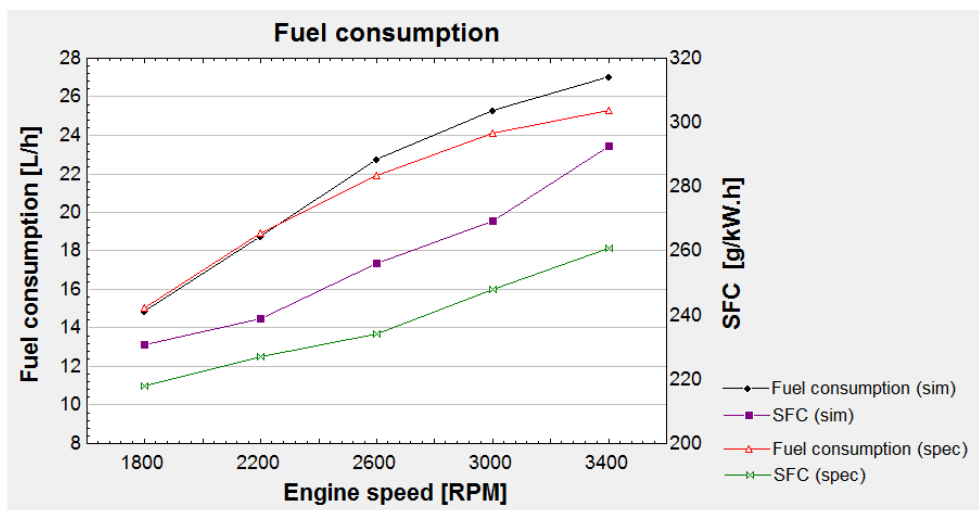


Figure 83 - Simulated fuel consumption – graphical results

The simulation model results are realistic and provide accurate results, especially at the lower end of the speed range. As the engine speed increases the simulated results deviate slightly, with a maximum error of 6% from the spec sheet values, but not to an extent that they would be considered unreliable.



5.5.3 Efficiencies

The combustion efficiency results calculated by the mathematical model are shown below.

Table 45 - Simulated combustion efficiency – tabulated results

Combustion efficiency [%]	
RPM	Combustion efficiency
1800	97,76
2200	98,37
2600	98,26
3000	97,69
3400	99,78

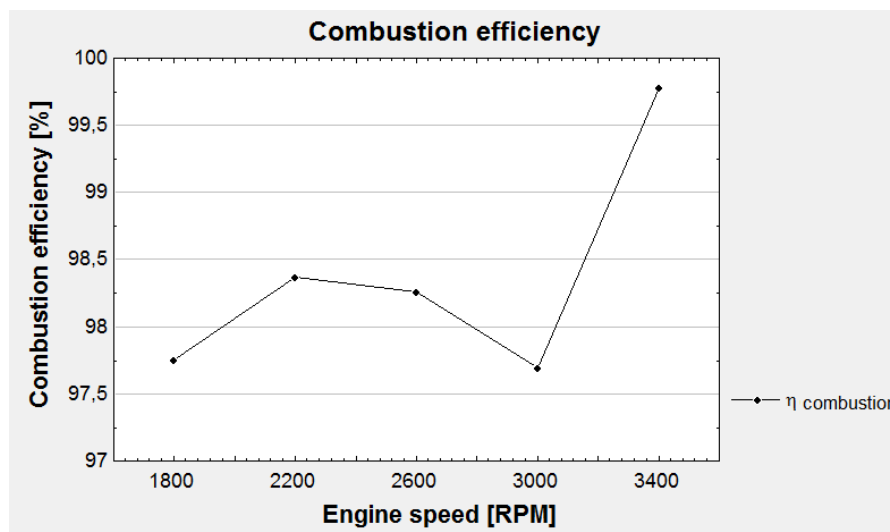


Figure 84 - Simulated combustion efficiency – graphical results

It is important to note that combustion efficiency is determined from an exhaust gas analysis. This therefore means that the test data from the gas analyser had to be inserted into the model to produce the required combustion efficiency. Because the same data from the gas analyser was used to calculate the combustion efficiency for the test results in Figure 57 and Figure 71, the efficiency results above are exactly the same as those found in section 5.4.2.



The simulated thermal efficiencies produced by the model can be seen in the table and graph below.

Table 46 - Simulated thermal efficiency – tabulated results

Thermal efficiency [%]	
RPM	Thermal efficiency
1800	26,98
2200	26,28
2600	24,59
3000	23,52
3400	21,51

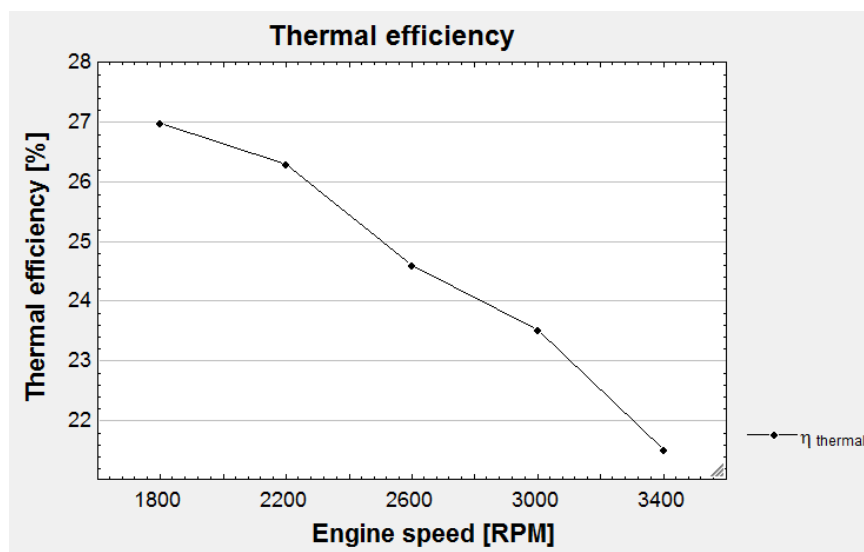


Figure 85 - Simulated thermal efficiency – graphical results

The results from the model regarding thermal efficiency display similar characteristics to those of the engine during the tests.



The simulated volumetric efficiency results are listed in the table below and displayed graphically in Figure 86.

Table 47 - Simulated volumetric efficiency – tabulated results

Volumetric efficiency [%]	
RPM	Volumetric efficiency
1800	142,80
2200	144,90
2600	144,90
3000	142,30
3400	140,30

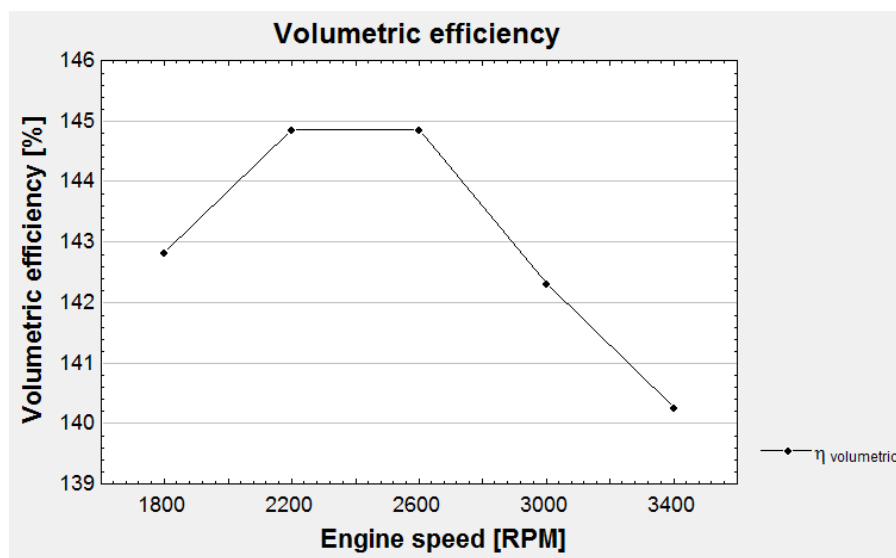


Figure 86 - Simulated volumetric efficiency – graphical results

The results obtained from the mathematical model for the volumetric efficiency are realistic for a turbocharged engine such as the one under consideration.



Chapter 6: Validation and verification of results

This chapter will discuss the method of validation and verification of the mathematical model along with the results of each. For validation purposes, the model's results are compared with the results obtained from the tests conducted on the engine test bench. Verification of the EES model was done by using an Excel model set up by Professor. C.P. Storm and Mr C.P. Kloppers and can be found in Appendix B. The Excel and EES models both work on the same basic principal with the exception of Lookup tables used in EES, that contain input data from tests conducted on the engine itself, to create a more realistic model. For this reason, the two model's results will differ slightly from one another.

The purpose of this chapter is to ensure that the model that has been created, accurately simulates real-life conditions and can be used to predict the performance of the engine under consideration with minimal error.

The mathematical model was designed to simulate the thermodynamic cycle of an internal combustion engine. Due to the additive working by altering the properties of the fuel used, the model is not able to simulate the engine's performance parameters with any concentrations of Naf-Tech in the fuel. For this reason, the simulation results need to be compared to a test that was run without the use of the additive.

After taking the above into consideration, the model's results were chosen to be compared to results of the 0 ml/l NT run from test set two. This choice was firstly made as the results from the second set of tests were more accurate than the first and secondly, the 0 ml/l NT run was completed before the faults had any influence on the results.

6.1 Power and Torque

In the table and graph below is the comparison of power produced by the EES simulation, test bench (validation) and Excel model (verification).

Table 48 - Power comparison – tabulated results

Power [kW]			
RPM	EES Simulation	Test bench	Excel model
1800	53,44	50,47	48,49
2200	65,12	60,30	60,36
2600	73,68	68,40	69,57
3000	77,88	73,28	74,89
3400	76,74	73,36	79,62

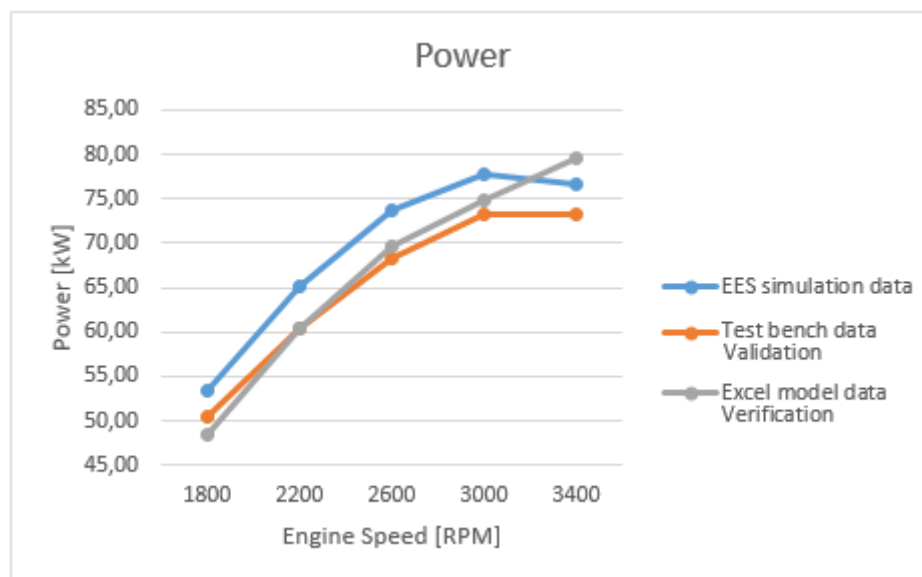


Figure 87 - Power comparison – graphical results

The comparison of the three sources of results indicates that although the curves do not precisely coincide with each other, they are extremely similar. It can, therefore, be stated that the EES model is able to predict the power output of the engine with a high degree of accuracy. As seen in Figure 87 above, the results from the EES model follow the same curve as the results for the test bench.



The average error between these two curves is 6.01% and between the two simulated curves, i.e. EES and Excel, is 4.45%. The error between each of the sets of results can be seen in Table 49 below.

Table 49 - Error margin – power results

Error [%]		
RPM	EES - Test bench	EES - Excel
1800	5,56	9,27
2200	7,41	7,31
2600	7,17	5,58
3000	5,91	3,83
3400	4,41	3,75

Simulation and actual data will never be 100% accurate when compared to one another. This is due to the fact that parameters for actual tests will always vary as time go by, whereas the simulated results will use constant inputs as set by the user.

Table 50 and Figure 88 below summarise the comparison results for torque from each of the three sources.

Table 50 - Torque comparison – tabulated results

Torque [Nm]			
RPM	EES Simulation	Test bench	Excel model
1800	283,50	270,23	257,24
2200	282,70	265,52	262,00
2600	270,60	263,11	255,52
3000	247,90	237,17	238,39
3400	215,50	208,88	223,61

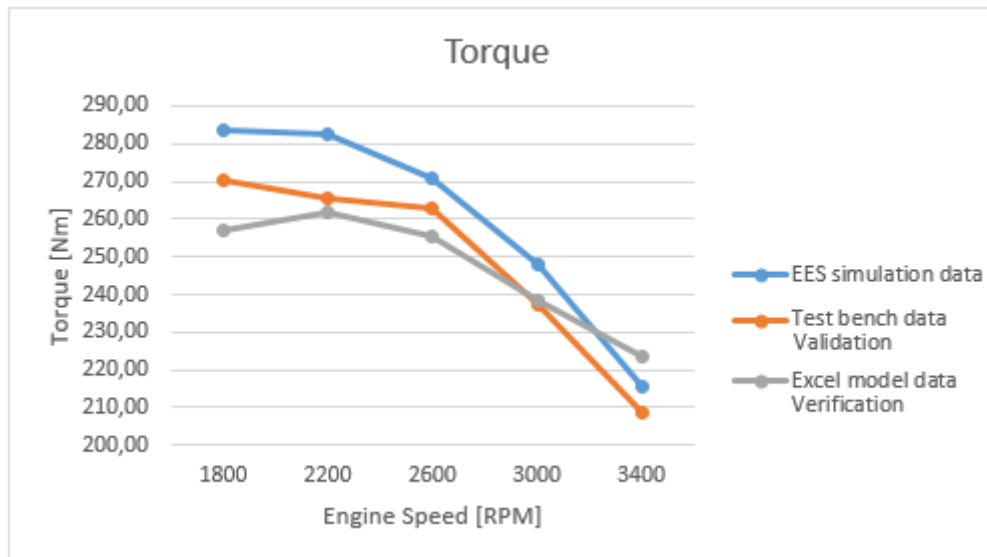


Figure 88 - Torque comparison – graphical results

Due to the power and torque of an engine being directly linked to each other, the accuracy of the torque results above would be similar to that of the power results in Figure 87.

The torque results from each of the sources as described above follow similar curves to one another as is the case with the power results. The average error between the EES and test bench results is 4.18% where the error between the EES and Excel models is 4.44%. This slight variation in the torque results from the three different sources is due to the same reason that the power results have small differences

The error margins for each of the engine speeds can be seen below.

Table 51 - Error margin – torque results

Error [%]		
RPM	EES - Test bench	EES - Excel
1800	4,68	9,26
2200	6,08	7,32
2600	2,77	5,57
3000	4,33	3,83
3400	3,07	3,76

6.2 Fuel consumption

The fuel usage of the engine from the EES simulation, test data and Excel model are all listed and displayed below for comparison purposes.

Table 52 - Fuel consumption comparison – tabulated results

Fuel consumption [l/h]			
RPM	EES Simulation	Test bench	Excel model
1800	14,85	14,49	16,28
2200	18,75	17,81	20,38
2600	22,73	20,71	24,52
3000	25,28	23,41	26,70
3400	27,04	25,25	28,24

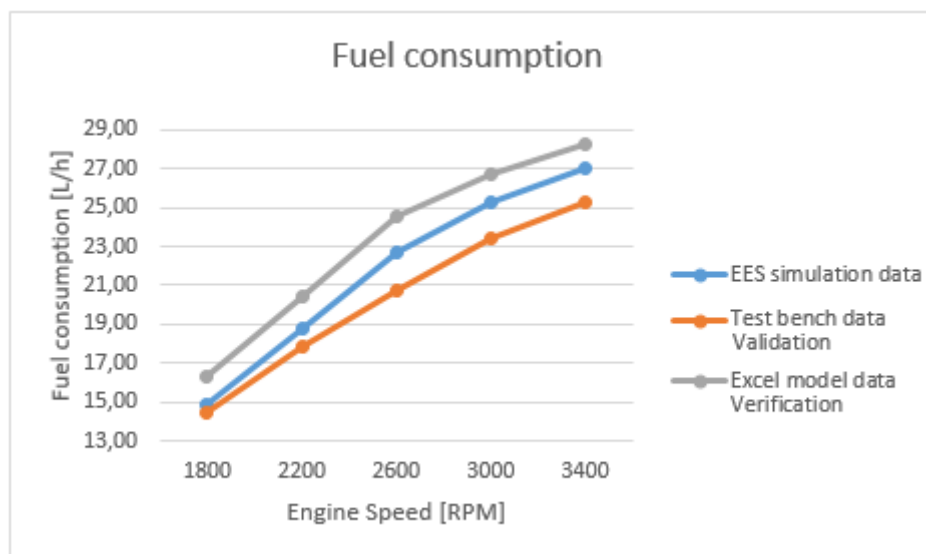


Figure 89 - Fuel consumption comparison – graphical results

Just as is the case with the power and torque curves, the three sets of results for the fuel consumption only deviate slightly from each other. At 1800 RPM the EES and test bench results differ by 2.42% with a maximum error of 8.88% at 2600 RPM which indicates that the EES model is reliable at predicting the fuel consumption. The percentage error for each set of results can be found in the table below.



Table 53 - Error margin – fuel consumption results

Error [%]		
RPM	EES - Test bench	EES - Excel
1800	2,42	9,64
2200	5,00	8,70
2600	8,88	7,88
3000	7,40	5,61
3400	6,61	4,45

In addition to the fuel consumption results presented above, a comparison of the specific fuel consumption was also done and is shown below.

Table 54 - Specific fuel consumption comparison – tabulated results

Specific fuel consumption [g/kW.h]			
RPM	EES Simulation	Test bench	Excel model
1800	230,60	238,30	278,71
2200	239,00	245,18	280,27
2600	256,10	251,31	292,56
3000	269,40	265,12	295,89
3400	292,50	285,77	294,45

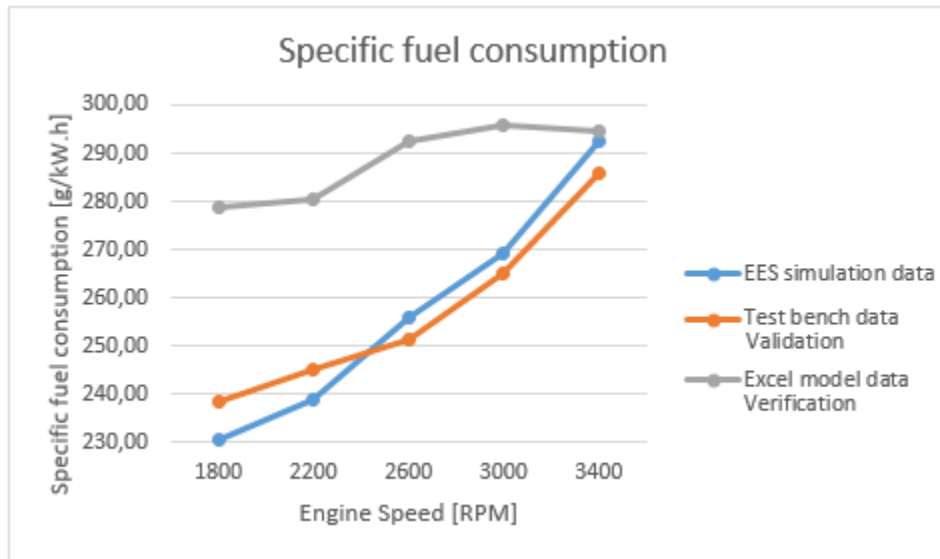


Figure 90 - Specific fuel consumption comparison – graphical results

While the EES model and test results are extremely similar, with a maximum error of 2.59% between the two at 2200 RPM, the results from the Excel model differ with a large error at the lower speeds when compared to the EES. This significant difference is due to the power results from Excel being slightly lower as well as the fuel consumption [l/h] being slightly higher than the same results from the EES model. This combination causes the SFC to be much higher than the EES and test bench results. A full list of errors for these results can be seen in Table 55 below.

Table 55 - Error margin – specific fuel consumption results

Error [%]		
RPM	EES - Test bench	EES - Excel
1800	3,34	20,86
2200	2,59	17,27
2600	1,87	14,24
3000	1,59	9,83
3400	2,30	0,67

6.3 Efficiencies

The efficiencies of an engine play a vital role in indicating how well it is operating and converting chemical energy into mechanical energy. For this reason, a comparison has been made regarding, the thermal, volumetric and combustion efficiencies from the EES model, test bench and Excel model.

The comparison of the thermal efficiency is provided below.

Table 56 - Thermal efficiency comparison – tabulated results

Thermal efficiency [%]			
RPM	EES Simulation	Test bench	Excel model
1800	26,98	32,95	24,39
2200	26,28	31,83	24,30
2600	24,59	31,09	23,27
3000	23,52	29,65	22,92
3400	21,51	26,93	22,97

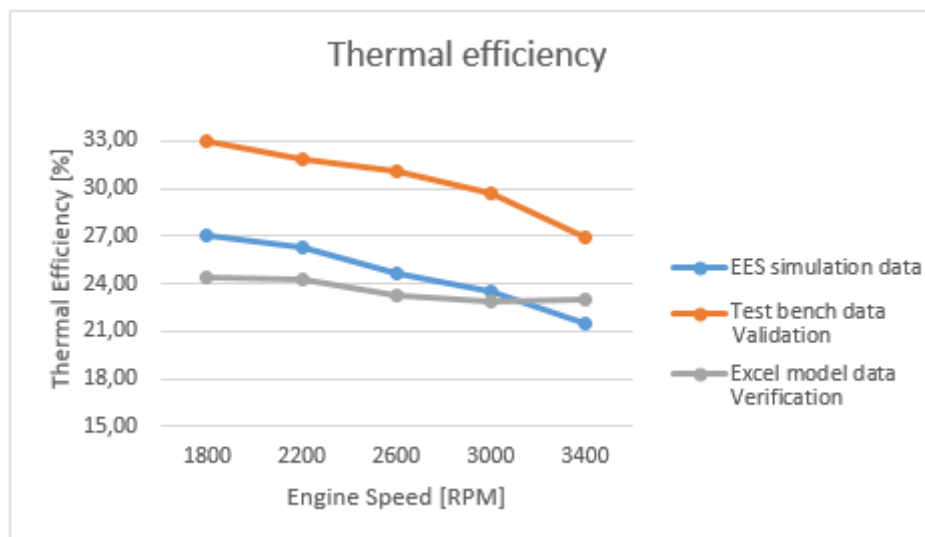


Figure 91 - Thermal efficiency comparison – graphical results

While the EES and Excel model have produced very similar thermal efficiency results, the data obtained from the test bench produce an efficiency curve that is 24.17% higher than that



of the EES simulation. The large difference between the actual efficiency and that of the simulated data could be due to an unknown parameter that was not taken into account while creating the mathematical models or errors in the accuracy of the measuring equipment during the tests.

The differences between each of the sets of results above are listed below.

Table 57 - Error margin – thermal efficiency results

Error [%]		
RPM	EES - Test bench	EES - Excel
1800	22,12	9,61
2200	21,11	7,53
2600	26,43	5,36
3000	26,05	2,54
3400	25,18	6,76

The volumetric efficiency results from the two models and the test bench are compared to one another in the table and figure below.

Table 58 - Volumetric efficiency comparison – tabulated results

Volumetric efficiency [%]			
RPM	EES Simulation	Test bench	Excel model
1800	142,80	147,83	156,61
2200	144,90	149,38	157,43
2600	144,90	146,91	156,24
3000	142,30	142,74	150,28
3400	140,30	135,96	146,49

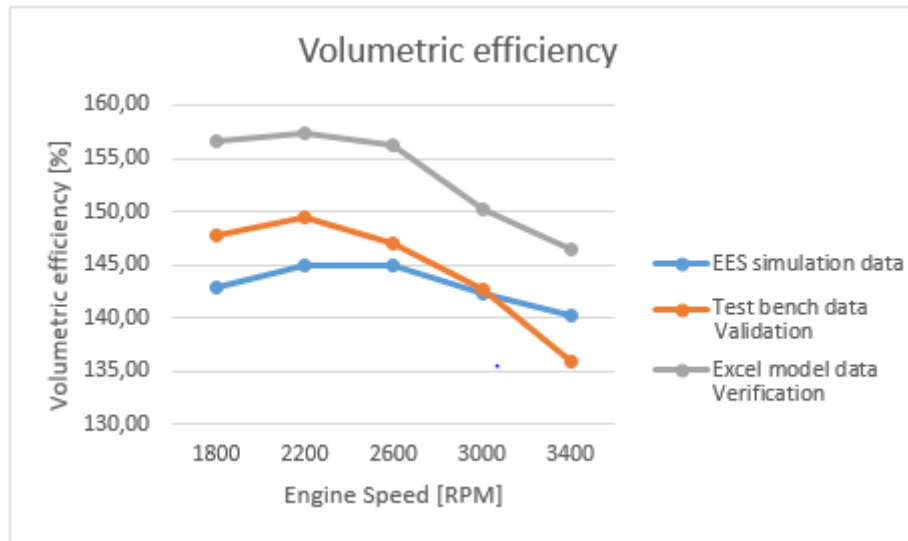


Figure 92 - Volumetric efficiency comparison – graphical results

The results from the EES simulation and test bench are very similar, with an average error between the two of 1.04%. There is a larger difference between the results of the EES and Excel models with an average error of 7.23%. The error between each of the sets of results at different engine speeds is listed in Table 59 below.

Table 59 - Error margin – volumetric efficiency results

Error [%]		
RPM	EES - Test bench	EES - Excel
1800	3,52	9,67
2200	3,09	8,65
2600	1,39	7,82
3000	0,31	5,61
3400	3,09	4,41

The larger variations in the Excel data is caused by it not being as accurate as the EES model. The EES model uses Lookup tables to obtain accurate input data for each test condition whereas the Excel model does not include these tables.

The combustion efficiency results from the three sets are the last to be compared and can be seen below.



Table 60 - Combustion efficiency comparison – tabulated results

Combustion efficiency [%]			
RPM	EES Simulation	Test bench	Excel model
1800	97,76	97,76	97,76
2200	98,37	98,37	98,37
2600	98,26	98,26	98,26
3000	97,70	97,70	97,70
3400	99,78	99,78	99,78

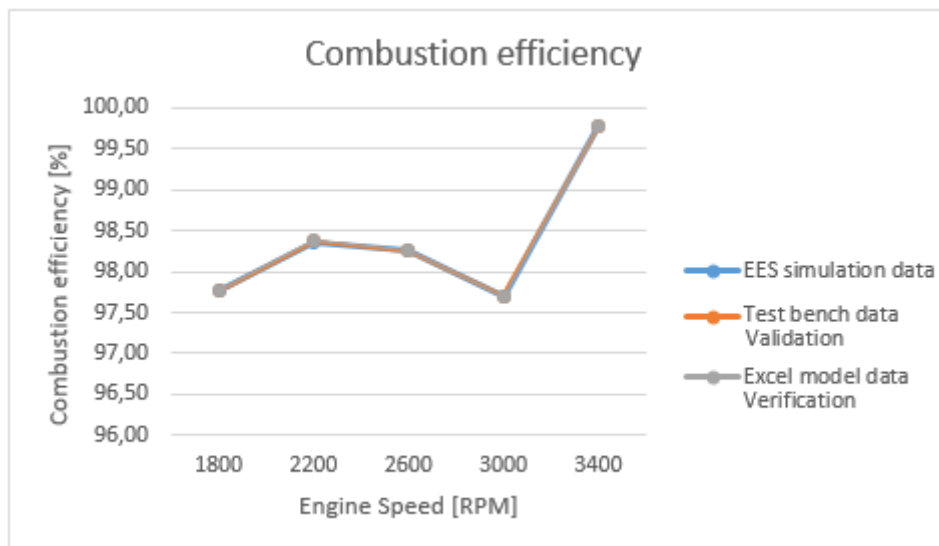


Figure 93 - Combustion efficiency comparison – graphical results

As stated before, the combustion efficiency from each of the sets of results would be exactly the same. This is because the method for calculating combustion efficiency in each of the three sets uses the same results from the exhaust gas analysis which means that there will be no variation between the EES, test bench and Excel results as seen above.

Modern diesel engines usually operate with a combustion efficiency of 98-100% which means that the values seen in Figure 93 from the three sets are realistic.



Chapter 7: Conclusions

The main aim of this dissertation is to investigate the effect that the combustion enhancing fuel additive called Naf-Tech has on the performance and fuel consumption of a diesel internal combustion engine. This is to be done by conducting a number of tests on a diesel engine both with and without the use of Naf-Tech in the fuel. In addition to this, a mathematical model needed to be created that is able to accurately simulate the engine under all the conditions used during the test procedure.

7.1 *Gross calorific value*

Before any tests could be carried out on the engine, it needed to be known if the additive had any effect on the gross calorific value of the diesel. Tests were therefore done with the aid of a bomb calorimeter where it was found that the GCV_v of Naf-Tech was much lower than that of commercial diesel, and when mixed together, had no effect on the energy contained in the base fuel.

Additional tests were conducted at CSIR in Pretoria on a diesel burner test rig to ensure that Naf-Tech indeed has no effect on the combustion process due to altering the amount of energy in the fuel. This set of tests was aimed at measuring the steady-state flame temperature produced from the combustion of the diesel. The results of these tests revealed that there was a negligible difference in the temperature before and after Naf-Tech was added to the fuel.

From the results of the above two sets of tests, it can be concluded that Naf-Tech does not in any way alter the GCV_v of the base fuel but rather works by improving the combustion efficiency as stated by the manufacturers.





7.2 Performance, fuel consumption and efficiency

Tests were performed at Aggreko on an 800 kVA generator for the purpose of determining the effect on performance, efficiency and emissions with the use of Naf-Tech in the fuel. The test results indicated that there was no difference in the performance output of the generator unit with the addition of the additive to the fuel. It was found that the unit experienced an increase in efficiency of 9.8% at 75% load, but a slight decrease at 50 and 100% load. With regards to the emission results, Naf-Tech has no effect on the four combustion gases that were analysed i.e. CO, CO₂, O₂ and NO_x whereas the VOC emissions produced by the generator with the use of Naf-Tech show a drastic decrease of 79% as opposed to the baseline diesel test.

The main focus of this study was the tests to be completed on the GWM engine at NWU. The results from the first set of tests carried out on the engine reveal that there was no difference in the power and torque produced by the engine before and after the addition of Naf-Tech, which was due to the ECU regulating the output parameters to remain consistent for each condition. The additive did, however, increase the overall fuel efficiency of the engine by 12.4%. In addition to the positive effect that Naf-Tech had on fuel consumption, there was a large reduction in all the harmful exhaust emissions which means that a more efficient combustion process is taking place. The thermal and combustion efficiencies measured during the tests show 12.27% and 0.1% increases respectively.

In conclusion to the two above sets of tests, it can be stated that although Naf-Tech has no effect on the output performance of an ECU controlled engine, it drastically improves the fuel consumption of the unit. The efficiencies measured during the tests prove that Naf-Tech does, in fact, work by increasing the combustion efficiency of the engine which can also be seen in the reduction of harmful exhaust emissions.

Unfortunately, for the final set of tests conducted on the engine, there were a number of faults that developed. The problems that occurred with the engine are fully described in paragraph 5.4.2 in chapter 5. Due to these faults, it is not possible to make any conclusions from the results of test set two regarding the effect that Naf-Tech had on the engine. It is, however, possible to state that, from the expert's opinion and by analysing the results, the deteriorating



condition of the engine can be ascribed to the large leaks in the exhaust and turbo system as well as the damage caused by overheating.

7.3 Mathematical model

The mathematical model was created to simulate the thermodynamic cycle of a diesel engine with an EGR system. By analysing the results of the model and comparing them to the engine's technical manual as well as validating and verifying them against the results from the test bench and Excel model, it can be concluded that the model is able to predict the necessary parameters of the engine at any condition with a great degree of accuracy.

7.4 Final conclusion

This research is aimed at determining whether or not the combustion enhancer called Naf-Tech has any effect on the performance and efficiency of an internal combustion engine.

From the results of each of the tests conducted with the fuel additive, it can clearly be concluded that Naf-Tech does, in fact, work as a combustion enhancer and improve the engine's fuel efficiency by the amount claimed by the manufacturers. It can also be stated that by using Naf-Tech in the fuel, harmful greenhouse gas emissions are reduced considerably as opposed to using commercial diesel without the additive.



Chapter 8: Future work and recommendations

An issue that had a large negative effect on the performance of the test bench during testing was the size of the cooling drum for the dynamometer water. The 250 L drum was not large enough to keep the dynamometer below 60°C for the full duration of the tests. It is therefore recommended that a larger tank be installed in its place. A 750 L container has already been purchased for this purpose and just needs to be installed.

To reduce the time used to reach each of the test conditions as well as to produce more accurate, consistent results for each test, it is recommended that the control of the dynamometer and engine throttle be controlled automatically eg, with the use of a PID controller.

For safety reasons, it is advisable that the engine's control panel be relocated to a position that is not directly alongside the test engine. Its current position is not safe for the operator.

It is recommended that the computer system used for capturing the electronic data be upgraded. The current PC crashed on multiple occasions during the tests which then required the test to be restarted.



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Appendix A – EES simulation code

A1. EES code

"INPUTS"

"Variables"

Speed_Engine = 3000 [RPM]

T_amb = 312,15 [K] "measure"

P_amb = 87,4 [kPa] "measure"

AF_ratio = Lookup('AF ratio'; LookupRow('AF ratio'; 'RPM'; Speed_Engine); '500ppm, no NT')

rho_air_in=Density(Air_ha;T=T[3];P=P[3])

m_dot_air = (Speed_Engine/(60*2))*Cylinders*rho_air_in*(Capacity_d/1000000)*0,9 "90% for friction losses"

m_dot_fuel = m_dot_air*(1/AF_ratio)

fuel_flow = (m_dot_fuel/rho_air)*((60*60)/1000)

Gasket_thick = 0,004876 [m]

DELTAP_ac = 5 [kPa]

DELTAP_inter = 6 [kPa]

DELTAP_inlet = 10 [kPa]

DELTAP_exh = 5 [kPa]

DELATP_Turbo_backpressure = 240 [kPa]

Turbo_pressure = Lookup('Full load'; LookupRow('Full load'; 'RPM'; Speed_Engine); 'Boost')

"Constants"

Eta_isen_turbo = 0,98

T_inter_in = (Lookup('Full load'; LookupRow('Full load'; 'RPM'; Speed_Engine); 'T_inter_in')) + 273,15

T_inter_out = (Lookup('Full load'; LookupRow('Full load'; 'RPM'; Speed_Engine); 'T_in')) + 273,15

Eta_intercooler = (T_inter_in - T_inter_out)/(T_inter_in - T_amb) "Values from testing"

Eta_isen_comp = 0,82

Eta_isen_expansion = 0,82

Eta_combust = comb "ehxhaust analysis"

GCV = 46,9 [MJ/kg] "Bomb calorie test"

Altitude = 1351 [m]

g = 9,81 [m/s]



Bore = 0,093 [m] "From User Manual"

Stroke = 0,092 [m] "From User Manual"

Cylinders = 4 [-]

$r_v = 17,2$ "Compression ratio"

$r_v = (\text{Vol_cylinder}) / \text{Vol_combustion}$

$\text{Capacity_d} = \pi * (\text{Bore}^2 / 4) * \text{Stroke} * 1000^2$ "displacement"
 $\text{Vol_cylinder} = \text{Capacity_d} + \text{Vol_combustion}$
 $\text{Capacity_tot} = \text{Capacity_d} * 4$

$\text{Piston_area} = \pi * (\text{Bore}^2 / 4)$

$\text{Piston_indentation} = 12,5 \text{ [cm}^3\text{]}$

"Fluid properties"

$C_{p_air} = 1,0035 \text{ [kJ/kg]}$
 $C_{v_air} = 0,7165 \text{ [kJ/kg]}$
 $R_{air} = C_{p_air} - C_{v_air}$
 $\gamma_{air} = C_{p_air} / C_{v_air}$
 $\rho_{air} = 1,184 \text{ [kg/m}^3\text{]}$

$C_{p_gas} = 1,05801 \text{ [kJ/kg]}$
 $C_{v_gas} = 0,77065 \text{ [kJ/kg]}$
 $R_{gas} = C_{p_gas} - C_{v_gas}$
 $\gamma_{gas} = C_{p_gas} / C_{v_gas}$

"Air-Gas properties"

$\text{Air_percent} = m_8 / m[9]$
 $\text{Gas_percent} = m[7] / m[9]$
 $\text{EGR_percentage} = m_{\text{EGR_cyl}} / m[9]$
 $C_{p_AG} = \text{Air_percent} * C_{p_air} + \text{Gas_percent} * C_{p_gas} + \text{EGR_percentage} * C_{p_gas}$
 $C_{v_AG} = \text{Air_percent} * C_{v_air} + \text{Gas_percent} * C_{v_gas} + \text{EGR_percentage} * C_{v_gas}$
 $R_{AG} = C_{p_AG} - C_{v_AG}$
 $\gamma_{AG} = C_{p_AG} / C_{v_AG}$



"Diesel cycle"

"Atmosphere - Point 1"

```
T[1] = T_amb
T_C[1] = T[1] - 273,15
P[1] = P_amb
P[1] * v[1] = R_air * T[1]
Vol[1] = Capacity_d/1000000
m[1] = (Vol[1])/v[1]
rho[1] = Density('Air density', 1:P[1])
rho[1] = Density('Air density', T=T[1], P=P[1])
```

"divide by 1000000 to convert from cm³ to m³"

"test"

"Air Cleaner out - Point 2"

```
T[2] = T_amb
T_C[2] = T[2] - 273,15
P[2] = P[1] - DELTAP_ac
P[2] * v[2] = R_air * T[2]
Vol[2] = Capacity_d/1000000
m[2] = (Vol[2])/v[2]
rho[2] = 1/v[2]
```

"Turbocharger out - Point 3"

```
r_p = (P[1]+Turbo_pressure)/P[1]
P[3] = P[2]*r_p
T_isen[3]/T[2] = (P[3]/P[2])^((gamma_air - 1)/gamma_air)
Eta_isen_turbo = (T_isen[3] - T[2])/(T[3] - T[2])
T_C[3] = T[3] - 273,15
P[3] * v[3] = R_air * T[3]
Vol[3] = Capacity_d/1000000
m[3] = (Vol[3])/v[3]
rho[3] = 1/v[3]
```

"Intercooler out - Point 4"

```
Eta_intercooler = (T[3] - T[4])/(T[3] - T_amb)
T_C[4] = T[4] - 273,15
P[4] = P[3] - DELTAP_inter
P[4] * v[4] = R_air * T[4]
Vol[4] = Capacity_d/1000000
m[4] = (Vol[4])/v[4]
rho[4] = 1/v[4]
```

"After inlet manifold - Point 5"

```
T[5] = T[4]
T_C[5] = T[5] - 273,15
P[5] = P[4] - DELTAP_inlet
P[5] * v[5] = R_air * T[5]
Vol[5] = Capacity_d/1000000
m[5] = (Vol[5])/v[5]
rho[5] = 1/v[5]
```



"Residual gas - Point 6"

$$T_{\text{exh}} = 1001,279 \text{ [K]}$$

$$T[6] = T_{\text{exh}}$$

$$T_{\text{C}}[6] = T[6] - 273,15$$

$$P[6] = P_{\text{amb}} + \text{DELATP_Turbo_backpressure} + \text{DELTA}P_{\text{exh}}$$

$$P[6] \cdot V[6] = R_{\text{gas}} \cdot T[6]$$

$$V[6] = V_{\text{combustion}}/1000000$$

$$m[6] = (V[6])/\lambda[6]$$

$$\rho[6] = 1/\lambda[6]$$

"Residual gas after intake - Point 7"

$$m[7] = m[6]$$

$$T[7] = T[6]$$

$$T_{\text{C}}[7] = T[7] - 273,15$$

$$V[7] = V_{\text{cylinder}}/1000000$$

$$P[7] \cdot V[7] = m[7] \cdot R_{\text{gas}} \cdot T[7]$$

$$P[7] \cdot \lambda[7] = R_{\text{gas}} \cdot T[7]$$

$$\rho[7] = 1/\lambda[7]$$

"EGR after intake"

$$D_{\text{exh}} = 0,075 \text{ [m]}$$

$$D_{\text{egr}} = 0,045 \text{ [m]}$$

$$A_{\text{exh}} = (\pi \cdot D_{\text{exh}}^2)/4$$

$$A_{\text{egr}} = (\pi \cdot D_{\text{egr}}^2)/4$$

$$\text{Area_reduction_EGR} = A_{\text{exh}}/A_{\text{egr}}$$

$$\text{EGR_}\% = \text{Lookup}(\text{'EGR'}; \text{LookupRow}(\text{'EGR'}; \text{'RPM'}; \text{Speed_Engine}); \text{'EGR_}\%')$$

$$m_{\text{dot_EGR}} = m_{\text{dot_air}}/\text{Area_reduction_EGR} \cdot \text{EGR_}\%$$

$$m_{\text{dot_EGR}} = m_{\text{EGR_cyl}}/(\text{Speed_Engine}/(60 \cdot 2)) \cdot \text{Cylinders} \cdot 0,9$$

$$m_{\text{EGR_ratio}} = m[8]/m_{\text{EGR_cyl}}$$

$$P_{\text{EGR}} \cdot V_{\text{EGR}} = m_{\text{EGR_cyl}} \cdot R_{\text{gas}} \cdot T_{\text{EGR}}$$

$$V_{\text{EGR_ratio}} = V[1]/V_{\text{EGR}}$$

$$V_{\text{EGR}1} = (V_{\text{cylinder}} \cdot \text{EGR}[8])$$

$$P_{\text{EGR}} = P_{\text{amb}} + \text{DELTA}P_{\text{exh}} + \text{DELATP_Turbo_backpressure}$$

$$T_{\text{EGR}} = 523,15 \text{ [K]} \quad \text{"guess value"}$$

"Charge air after intake - Point 8"

$$T[8] = T[5]$$

$$T_{\text{C}}[8] = T[8] - 273,15$$

$$m[8] = m[5]$$

$$\text{EGR}[8] = \text{Lookup}(\text{'EGR'}; \text{LookupRow}(\text{'EGR'}; \text{'RPM'}; \text{Speed_Engine}); \text{'EGR_}\%')$$

$$V_{\text{EGR}}[8] = (V_{\text{cylinder}} \cdot \text{EGR}[8])$$

$$V[8] = (V_{\text{cylinder}})/1000000$$

$$P[8] \cdot V[8] = m[8] \cdot R_{\text{air}} \cdot T[8]$$

$$P[8] \cdot \lambda[8] = R_{\text{air}} \cdot T[8]$$

$$\rho[8] = 1/\lambda[8]$$



"Inlet mixture after intake - Point 9"

```
Vol_9 = Vol[8] - Vol_EGR
P[8]*Vol_9 = m_8*R_air*T[8]
m[9] = m[7] + m_8 + m_EGR_cyl
T[9]*m[9] = T[7]*(m[7]) + T[8]*(m[8]) + T_EGR*m_EGR_cyl
T_C[9] = T[9] - 273,15
P[9] = P[7] + P[8]
Vol[9] = Vol[8]
Vol_9_act = Vol[8] - Vol_EGR
P[9]*V[9] = R_AG*T[9]
rho[9] = 1/V[9]
```

"After compression stroke - Point 10"

```
T_isen[10]/T[9] = (r_v)^(Gamma_AG - 1)
Eta_isen_comp = (T_isen[10] - T[9])/(T[10] - T[9])
T_C[10] = T[10] - 273,15
P[10]/P[9] = ((Vol_9_act)/(Vol_combustion""/1000000))^(Gamma_AG)

P_[10] = (Vol[9]/Vol[10])^(Gamma_AG)*P[9]
Vol_EGR_10 = ((Vol_combustion/1000000)/Vol[9])*(Vol_EGR[8]/1000000)
Vol[10] = Vol_combustion/1000000 - Vol_EGR_10""
P[10]*V[10] = R_AG*T[10]
m[10] = m[9]
rho[10] = 1/V[10]
```

"After combustion (constant volume) - Point 11"

```
beta1 = v[12]/V[10]

alpha = 2                                     "pressure ratio"
beta = 1,443                                  "cut off ratio"
P[11] = P[10]*alpha
m[11] = m[10] + m[8]/AF_ratio
Vol[11] = Vol[10]
P[11]*Vol[11] = m[11]*R_gas*T[11]
T_C[11] = T[11] - 273,15
P[11]*V[11] = R_gas*T[11]
rho[11] = 1/V[11]
```

"After combustion (constant pressure) - Point 12"

```
P[12] = P[11]
Vol[12] = Vol[11]*beta
m[12] = m[11]
P[12]*Vol[12] = m[12]*R_gas*T[12]
T_C[12] = T[12] - 273,15
P[12]*V[12] = R_gas*T[12]
rho[12] = 1/V[12]
```



"After power stroke - Point 13"

$$T_{isen[13]}/T[12] = (Vol[12]/Vol[13])^{(\Gamma_{gas} - 1)}$$

$$(T[12] - T[13])/(T[12] - T_{isen[13]}) = \eta_{isen_expansion}$$

$$T_{C[13]} = T[13] - 273,15$$

$$P[13]/P[12] = ((Vol[12]/Vol[13]))^{\Gamma_{gas}}$$

$$Vol[13] = (Vol_{cylinder}/1000000)$$

$$P[13]*V[13] = R_{gas}*T[13]$$

$$\rho[13] = 1/V[13]$$

$$m[13] = m[12]$$

"After blowdown - Point 14"

$$P[14] = P_{amb} + \Delta P_{exh} + \Delta P_{Turbo_backpressure}$$

$$T_{isen[14]}/T[13] = (P[14]/P[13])^{((\Gamma_{gas} - 1)/\Gamma_{gas})}$$

$$T[14] = -((T[13] - T_{isen[14]}) - T[13])$$

$$T_{C[14]} = T[14] - 273,15$$

$$P[14]*V[14] = R_{gas}*T[14]$$

$$\rho[14] = 1/V[14]$$

$$m[14] = m[13]$$

$$c_{v_blowdown} = c_{v_gas}$$

"After Exhaust Stroke - Point 15"

$$T[15] = T[14]$$

$$T_{C[15]} = T[15] - 273,15$$

$$P[15] = P[14]$$

$$Vol[15] = Vol_{combustion}/1000000$$

$$P[15]*V[15] = R_{gas}*T[15]$$

$$m[15] = Vol[15]*\rho[15]$$

$$\rho[15] = 1/V[15]$$

"Engine output parameters"

$$w_{in} = ((P[10]*Vol[10] - P[9]*Vol_{9_act})/(\Gamma_{AG} - 1))/m[10]$$

$$w_{out} = (((P[12]*Vol[12] - P[13]*Vol[13]))/(\Gamma_{gas} - 1))/m[13]$$

$$w_{NET} = w_{out} - w_{in}$$

$$q_{in_total} = (GCV*1000)/AF_{ratio}$$

$$q_{out_exh} = c_{v_blowdown}*(T[13] - T_{amb})$$

$$\eta_{thermal} = (w_{NET}/q_{in_total})*100$$

$$P_{indicated} = w_{NET}*m[9]*(Speed_{Engine})/(60*2)*4$$

$$\tau_{indicated} = (P_{indicated}*60)/(2*\pi*Speed_{engine})*1000$$

$$Power_{fuel} = GCV*1000*m_{dot_fuel}$$

$$Power_{loss_exh} = q_{out_exh}*m_{dot_air}$$



"Fuel Consumption"

$$FC = ((m_dot_air/AF_ratio)*3600)/0.83$$

$$SFC_1 = FC/(P_indicated*(1/830))$$

$$SFC = (FC*0.83)/(P_indicated*(1/1000))$$

$$\eta_{a_v} = m[8]/m[1]*100$$

$$\eta_{a_{vol}} = (m_dot_air/(Speed_engine/60))/((Vol_cylinder/1000000)*\rho[1])/2*100 \quad \text{"volumetric efficiency"}$$

"Gas analysis"

$$R_o = 8,31450 \text{ [kJ/kmole-K]}$$

"Universal gas constant"

"C_xH_y hydrocarbon fuel composition"

$$x = 12$$

"Carbon"

$$y = 23$$

"Hydrogen"

"Atomic mass"

$$MM_Hydrogen = 1,0079 \text{ [kg/kmol]}$$

$$MM_Carbon = 12,0110 \text{ [kg/kmol]}$$

$$MM_Nitrogen = 14,0067 \text{ [kg/kmol]}$$

$$MM_Oxygen = 15,9994 \text{ [kg/kmol]}$$

$$MM_Sulphur = 32,0600 \text{ [kg/kmol]}$$

"Molar mass of fuel"

$$Mol_mass_fuel = MM_Carbon*x + MM_Hydrogen*y$$

$$R_fuel = R_o/Mol_mass_fuel$$

////////////////////COMBUSTION////////////////////

"Fuel- C_xH_y"

$$C_x = 12$$

$$H_y = 23$$

"Emission analysis"

$$CO = \text{Lookup}(\text{'Exhaust emissions (no NT)'}; \text{LookupRow}(\text{'Exhaust emissions (no NT)'}; \text{'RPM'}; \text{Speed_Engine}); \text{'CO'})$$

$$CO_2 = \text{Lookup}(\text{'Exhaust emissions (no NT)'}; \text{LookupRow}(\text{'Exhaust emissions (no NT)'}; \text{'RPM'}; \text{Speed_Engine}); \text{'CO_2'})$$

$$HC = \text{Lookup}(\text{'Exhaust emissions (no NT)'}; \text{LookupRow}(\text{'Exhaust emissions (no NT)'}; \text{'RPM'}; \text{Speed_Engine}); \text{'HC'})$$

$$O_2 = \text{Lookup}(\text{'Exhaust emissions (no NT)'}; \text{LookupRow}(\text{'Exhaust emissions (no NT)'}; \text{'RPM'}; \text{Speed_Engine}); \text{'O_2'})$$

$$NO_x = \text{Lookup}(\text{'Exhaust emissions (no NT)'}; \text{LookupRow}(\text{'Exhaust emissions (no NT)'}; \text{'RPM'}; \text{Speed_Engine}); \text{'NO_x'})$$

$$Unburnt_fuel = HC/200$$



"Gas percentage in air"

O2_air% = 20,86840

N2_air% = 100 - O2_air%

Air_ratio = N2_air%/O2_air%

"Chemical balance"

a = (CO + CO_2*2 + O_2*2 + NO_x*2)/2

b = a*Air_ratio

c = Unburnt_fuel

d = CO

e = CO_2

f = O_2

h = NO_x/10000

i = (b*2 - h)/2

j = (H_y - c*H_y)/2

"Combustion efficiency"

comb = ((1 - (d/(d + e))))*((1 - c)/1))*100

Figure 94 - Mathematical model code (EES)



A2. EES simulation inputs

Table 61 - Mathematical model input data

Inputs	Values	Units
Engine speed	1800 - 3400	RPM
Gravitational acceleration	9.81	m/s ²
Ambient Temperature	39	°C
Ambient Pressure	87.4	kPa
Altitude	1351	m
Air-Fuel ratio	16:1	[-]
Gasket thickness	1	mm
Turbo boost	(106 - 116 gauge) RPM dependent	kPa
EGR valve	(10 - 45) RPM dependent	%
ΔP air cleaner	5	kPa
ΔP intercooler	6	kPa
ΔP inlet	10	kPa
ΔP exhaust	5	kPa
ΔP turbo backpressure	240 gauge	kPa
η_{turbo}	98	%
$\eta_{\text{intercooler}}$	(43 - 61) RPM dependent	%
$\eta_{\text{compression}}$	82	%
$\eta_{\text{expansion}}$	82	%
$\eta_{\text{combustion}}$	(98 - 99.5) RPM dependent	%
GCV _v	46.9	MJ/kg
Bore	0.093	m
Stroke	0.092	m
Cylinders	4	[-]
Residual volume	38.58	cm ³
Displacement volume	624.9	cm ³
Total volume per cylinder	663.5	cm ³
Total engine capacity	2500	cm ³
Compression ratio r_v	17.2	[-]

A3 EES simulation results

Main	Key Variables
$AF_{ratio} = 16,2$ $\eta_{combust} = 97,76 \text{ [%]}$ $\eta_{thermal} = 26,98 \text{ [%]}$ $\eta_{vol} = 142,8 \text{ [%]}$ $FC = 14,85 \text{ [l/h]}$ $\dot{m}_{air} = 0,05546 \text{ [kg/s]}$ $Power_{loss,exh} = 36,04$ $P_{indicated} = 53,44 \text{ [kW]}$ $r_v = 17,2$ $w_{in} = 381,6 \text{ [kJ/kg}_{air}]$ $SFC = 230,6 \text{ [g/kW.h]}$ $w_{NET} = 781,1 \text{ [kJ/kg}_{air}]$ $\tau_{indicated} = 283,5 \text{ [Nm]}$ $Speed_{Engine} = 1800 \text{ [RPM]}$	

Figure 95 - EES results (key variables)

Power ===== Torque								
1..5	1	2	3	4	5	6	7	8
	Speed _{Engine} [RPM]	P _{indicated} [kW]	τ _{indicated} [Nm]	η _{thermal} [%]	η _{vol} [%]	FC [l/h]	SFC [g/kW.h]	
Run 1	1800	53,44	283,5	26,98	142,8	14,85	230,6	
Run 2	2200	65,12	282,7	26,28	144,9	18,75	239	
Run 3	2600	73,68	270,6	24,59	144,9	22,73	256,1	
Run 4	3000	77,88	247,9	23,52	142,3	25,28	269,4	
Run 5	3400	76,74	215,5	21,51	140,3	27,04	292,5	

Figure 96 - EES parametric table

Appendix B – Excel model (Verification)

B1. Excel input parameters

INPUT PARAMETERS					
$T_{REFERENCE}$	[K]	150,000	Cylinders	In-line	4
$P_{REFERENCE}$	[kPa]	50,000	Displacement	[cm ³]	2500,000
$V_{REFERENCE}$	[m ³ /kg]	0,86118	$C_{p\ AIR}$	[kJ/kgK]	1,00400
Bore	[mm]	93,000	$C_{v\ AIR}$	[kJ/kgK]	0,71694
Stroke	[mm]	92,000	$C_{p\ GAS}$	[kJ/kgK]	1,05801
Gasket thickness	[mm]	0,000	$C_{v\ GAS}$	[kJ/kgK]	0,77065
Head $V_{RESIDUAL}$ (V_C)	[cm ³]	38,580	$C_{p\ MIXTURE}$	[kJ/kgK]	1,00672
Cylinder $V_{RESIDUAL}$	[cm ³]	0,000	$C_{v\ MIXTURE}$	[kJ/kgK]	0,71965

$T_{AMBIENT}$	[°C]	39,000	$\Delta P_{INTAKE / INLET\ V/V}$	[kPa]	10,000
$P_{AMBIENT}$	[kPa]	87,400	$\Delta P_{EXHAUST\ V/V}$	[kPa]	5,000
$\Delta P_{AIR\ CLEANER}$	[kPa]	5,000	$T_{RESIDUAL\ GAS}$	[°C]	728,129
$r_{p\ SUPER/TURBO\ CHARGER}$	[-]	2,270	$\Delta P_{EXHAUST}$	[kPa]	5,000
$\eta_{ISENTR\ CHARGER}$	[%]	98,000%	Turbo $EXHAUST\ BACKPRESSURE$	[kPa] Gauge	240,000
$\Delta P_{INTERCOOLER}$	[kPa]	6,000	$\eta_{ISENTR\ COMPR\ STROKE}$	[%]	82,000%
$HX_{EFFECT\ INTERCOOLER}$	[%]	60,550%	$\eta_{ISENTR\ EXPANS\ STROKE}$	[%]	82,000%
$T_{COOLING\ FLUID}$	[°C]	14,000	A : F	[kg _{AIR} / kg _{FUEL}]	17,200

GCV_V	[MJ/kg]	46,900
$\eta_{COMBUSTION}$	[%]	97,760%
β	[-]	1,443
α	[-]	2,000
Engine Speed	[r.p.m.]	1800,000
Torque _{DYNO}	[Nm]	256,000
Power _{DYNO}	[kW]	48,255

Figure 97 - Excel Input data

B2. Excel cycle calculations

CYCLE CALCULATIONS						
PROPERTIES	Reference for Air	Ambient a	Air cleaner a/c	Super/Turbo c	Intercooler i/c	Intake Manifold, Inlet v/v i
T_i [K]	-	-	-	394,601	287,150	-
T_1 [K]	150,000	312,150	312,150	396,284	345,341	345,341
T_1 [°C]	-123,150	39,000	39,000	123,134	72,191	72,191
P [kPa]	50,000	87,400	82,400	187,050	181,050	171,050
C_p WORK FLUID [kJ/kgK]	1,00400	1,00400	1,00400	1,00400	1,00400	1,00400
C_v WORK FLUID [kJ/kgK]	0,71694	0,71694	0,71694	0,71694	0,71694	0,71694
R WORK FLUID [kJ/kgK]	0,28706	0,28706	0,28706	0,28706	0,28706	0,28706
γ WORK FLUID	1,40040	1,40040	1,40040	1,40040	1,40040	1,40040
$(\gamma-1)/\gamma$	0,28592	0,28592	0,28592	0,28592	0,28592	0,28592
$(\gamma-1)$	0,40040	0,40040	0,40040	0,40040	0,40040	0,40040
V WORK FLUID [m³/kg]	0,86118	1,02524	1,08745	0,60817	0,54755	0,57956
ρ WORK FLUID [kg/m³]	1,16119	0,97538	0,91958	1,64428	1,82631	1,72544
$V_{(1 \times \text{CYL})}$ WORK FLUID [m³]	0,0006249	0,00062495	0,00062495	0,00062495	0,00062495	0,00062495
m WORK FLUID [kg]	0,0007257	0,00060956	0,00057469	0,00102759	0,00114135	0,00107831
Check: $PV = mRT$ [kg]	0,0007257	0,00060956	0,00057469	0,00102759	0,00114135	0,00107831
ΔS_V WORK FLUID [kJ/kgK]	0,00000	0,57546	0,59237	0,59665	0,46786	0,48417
ΔS_P WORK FLUID [kJ/kgK]	0,00000	0,57546	0,59237	0,59665	0,46786	0,48417
ΔS_T WORK FLUID [kJ/kgK]	0,00000	0,57546	0,59237	0,59665	0,46786	0,48417
ΔS_{AVE} WORK FLUID [kJ/kgK]	0,00000	0,57546	0,59237	0,59665	0,46786	0,48417

Residual Gas after Exhaust	Residual Gas after Intake V_1 @ $PV=mRT$	Air after Intake V_d @ $PV=mRT$ gas	Inlet Mixture V_1 @ $PV=mRT$	Compression 2	Combust @ constant Volume x
-	-	-	-	1221,9180	-
1001,279	1001,279	345,341	381,5031	1414,3261	1937,6886
728,129	728,129	72,191	108,3531	1141,1761	1664,5386
337,400	19,6177	151,432	171,0499	9151,0708	18302,1416
1,05801	1,05801	1,00400	1,00672	1,00672	1,05801
0,77065	0,77065	0,71694	0,71965	0,71965	0,77065
0,28735	0,28735	0,28706	0,28707	0,28707	0,28735
1,37287	1,37287	1,40040	1,39891	1,39891	1,37287
0,27160	0,27160	0,28592	0,28516	0,28516	0,27160
0,37287	0,37287	0,40040	0,39891	0,39891	0,37287
0,85276	14,66638	0,65464	0,64028	0,04437	0,03042
1,17266	0,06818	1,52755	1,56182	22,53869	32,87015
0,00003858	0,00066353	0,00066353	0,00066353	0,00003585	0,00003585
0,00004524	0,00004524	0,00101357	0,0011196	0,0011196	0,00117855
0,00004524	0,00004524	0,00101357	0,0010363	0,0008081	0,00117855
1,45989	2,27737	0,51914	0,58667	0,76331	1,01086
1,46018	2,27766	0,51914	0,58669	0,76332	1,01115
1,46097	2,27844	0,51914	0,58672	0,76335	1,01194
1,46035	2,27782	0,51914	0,58669	0,76333	1,01132

Figure 98 - Excel calculations (1)



Combust @ constant Pressure 3	Power 4	Blowdown 5	Exhaust 6
-	1079,7368	1215,374	-
2795,7270	1388,6150	1215,374	1215,3745
2522,5770	1115,4650	942,2245	942,224
18302,1416	551,0911	337,4000	337,4000
1,05801	1,05801	1,0580	1,0580
0,77065	0,77065	0,7707	0,7707
0,28735	0,28735	0,28735	0,28735
1,37287	1,37287	1,37287	1,37287
0,27160	0,27160	0,27160	0,27160
0,37287	0,37287	0,37287	0,37287
0,04389	0,72406	1,03510	1,03510
22,78195	1,38110	0,96609	0,96609
0,00005173	0,00066353	0,00094856	0,00003858
0,00117855	0,00117855	0,00117855	0,0000372719
0,0011786	0,00091640	0,00091640	0,0000372719
1,39872	1,66491	1,66491	1,66491
1,39902	1,66520	1,66520	1,66520
1,39980	1,66598	1,66598	1,66598
1,39918	1,66536	1,66536	1,66536

Figure 99 - Excel calculations (2)

B3. Excel results

RESULTS					
$r_{BORE / STROKE}$	-	1,01087	W_{IN}	[kJ/kg _{AIR}]	480,517
V_2	[cm ³]	38,58000	W_{OUT}	[kJ/kg _{AIR}]	1322,436
V_d	[cm ³]	624,94760	$Q_{IN FUEL TOTAL}$	[kJ/kg _{AIR}]	2726,744
V_1	[cm ³]	663,52760	$Q_{IN AIR ACTUAL}$	[kJ/kg _{AIR}]	2665,665
r_v	-	17,19875	$Q_{IN AIR CONST V}$	[kJ/kg _{AIR}]	403,332
Displacement check:	[cm ³]	2499,790	$Q_{IN AIR CONST P}$	[kJ/kg _{AIR}]	907,811

$W_{IN AUXILIARIES}$	[kJ/kg _{AIR}]	132,244	Q_{IN}	[kW]	165,825
W_{NET}	[kJ/kg _{AIR}]	709,675	$Q_{OUT EXHAUST}$	[kJ/kg _{AIR}]	829,583
i.m.e.p.	[kPa]	1190,908	$Q_{OUT AUXILIARIES}$	[kJ/kg _{AIR}]	633,354
Torque _{INDICATED}	[Nm]	257,244	e_{IN}	[kJ/kg _{AIR}]	3339,505
Power _{INDICATED}	[kW]	48,491	e_{OUT}	[kJ/kg _{AIR}]	2785,373
Massflow _{AIR}	[kg/s]	0,060814			

η_{CARNOT}	[%]	88,835%	$w_{IN check}$	[kJ/kg _{AIR}]	743,266
$\eta_{DUAL-DIESEL CYCLE IDEAL}$	[%]	66,536%	$w_{OUT check}$	[kJ/kg _{AIR}]	1084,397
$\eta_{DUAL-DIESEL CYCLE ACTUAL}$	[%]	24,390%			
$\eta_{DUAL-DIESEL ACTUAL DYNO}$	[%]	29,100%			
$\eta_{VOLUMETRIC}$	[%]	156,610%	156,6109		

Figure 100 - Excel results

Appendix C – SolidWorks simulations

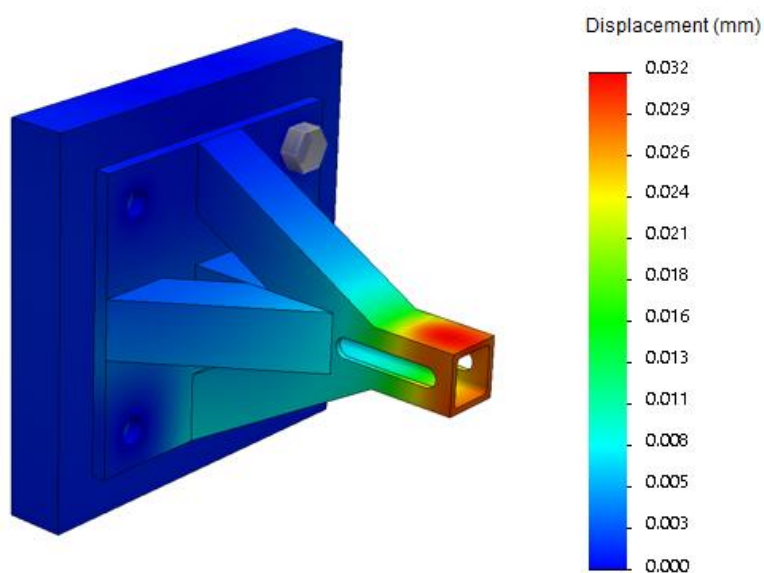


Figure 101 - SolidWorks simulation deformation analysis

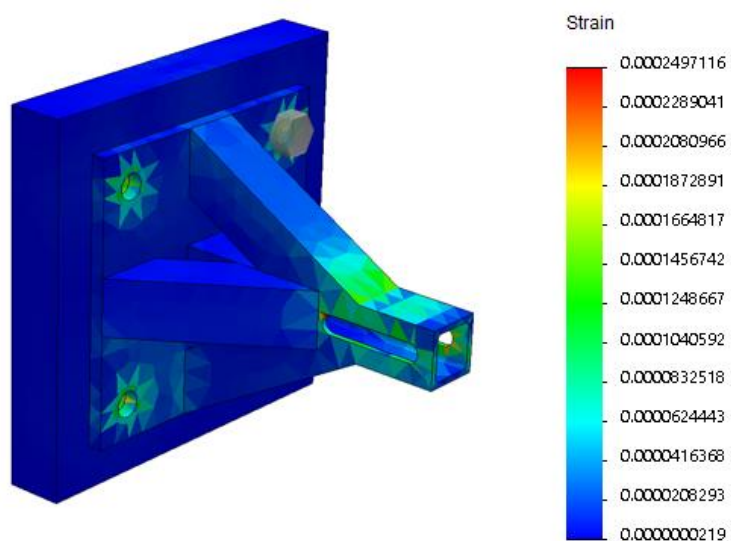


Figure 102 - SolidWorks simulation strain analysis

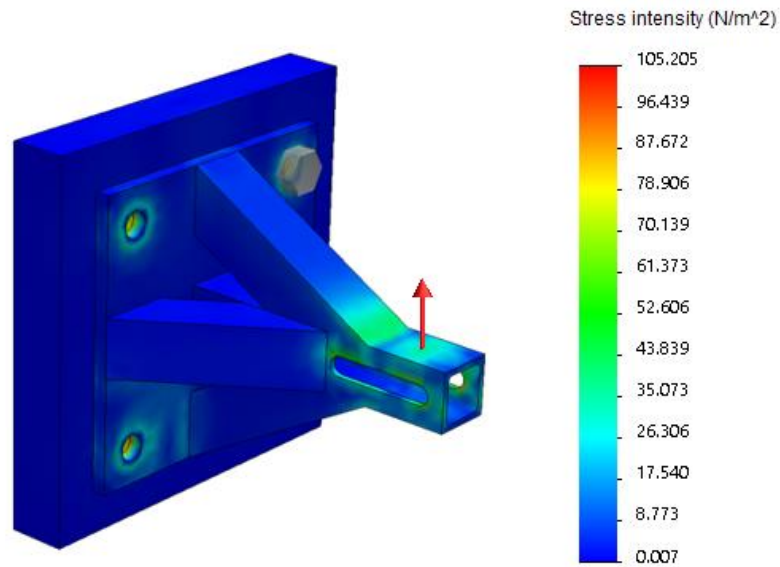


Figure 103 - SolidWorks simulation stress analysis



Appendix D – CSIR results

D1. Lean AF tests

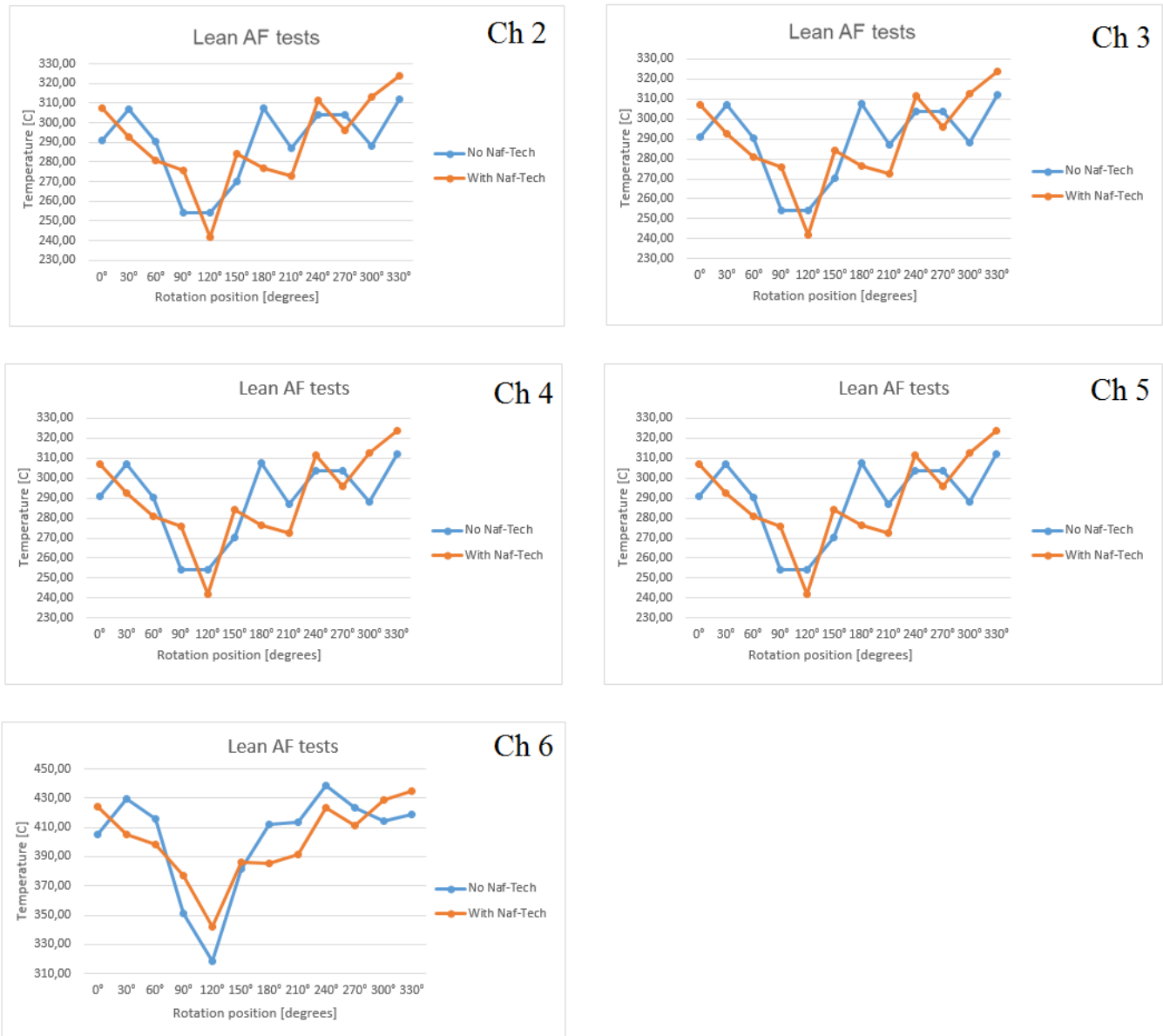


Figure 104 - CSIR diesel burner (lean tests)



D2. Intermediate AF tests

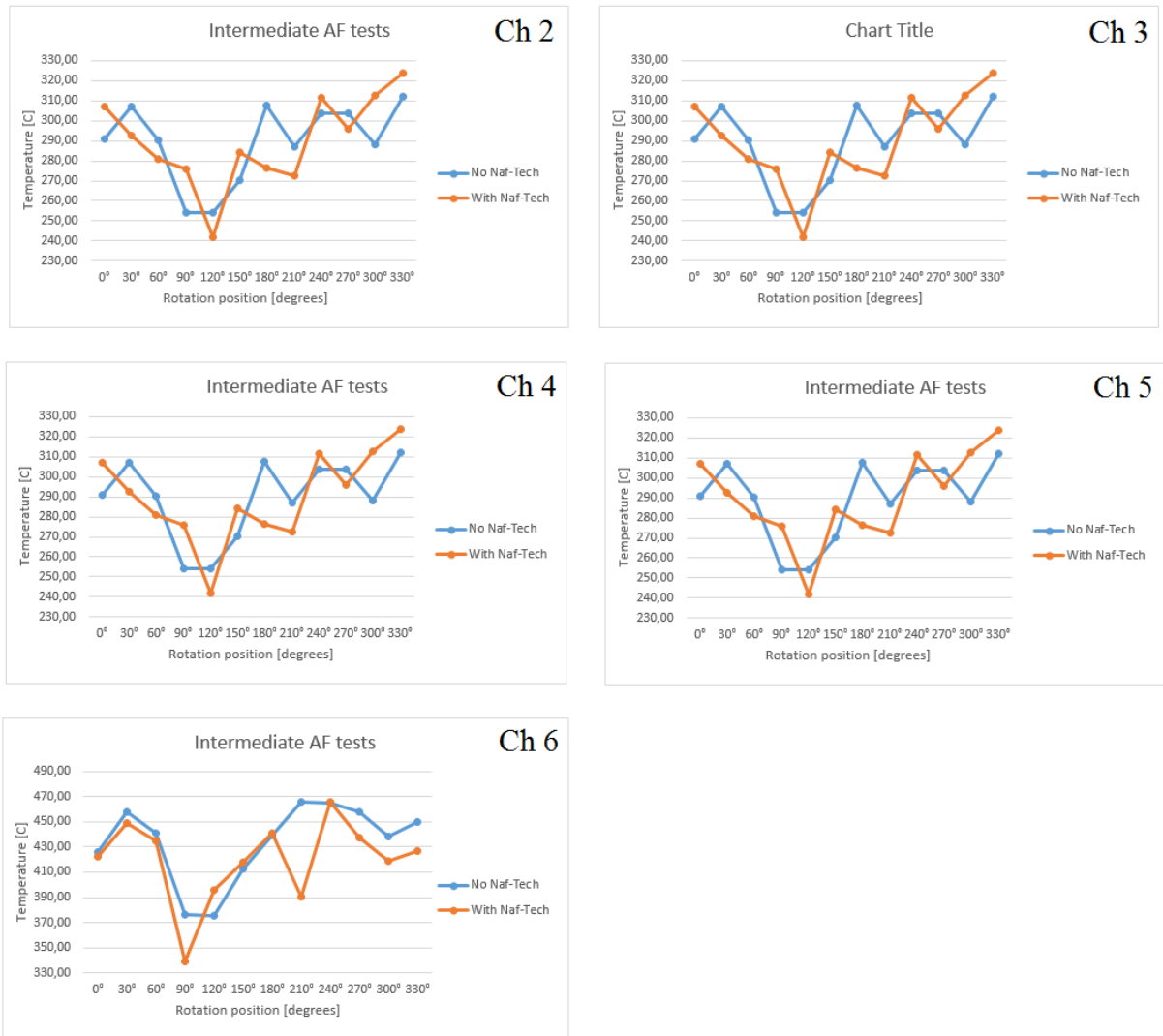


Figure 105 - CSIR diesel burner (intermediate tests)



D2. Rich AF tests

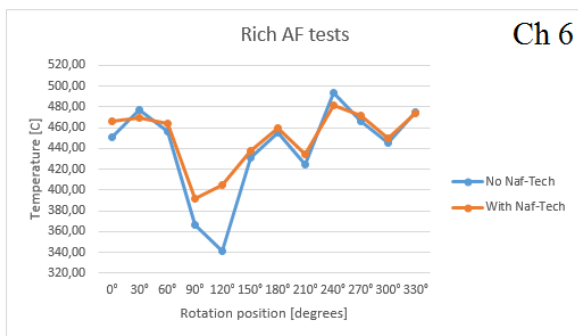
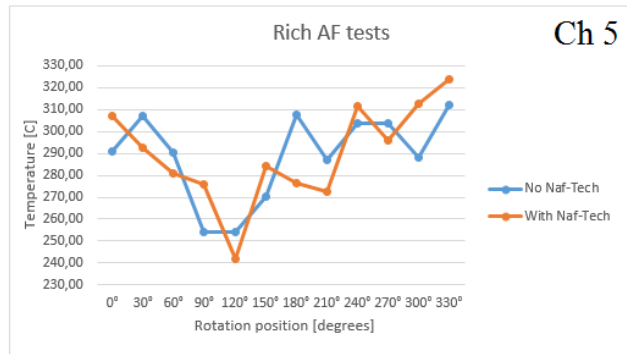
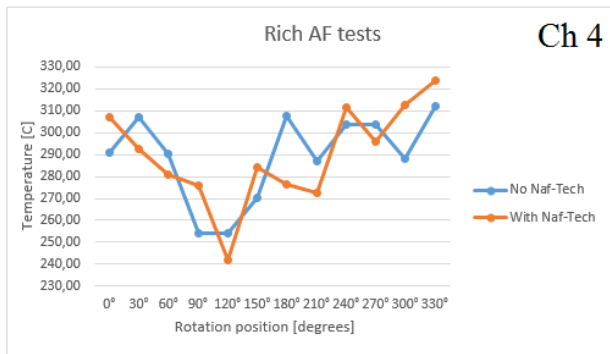
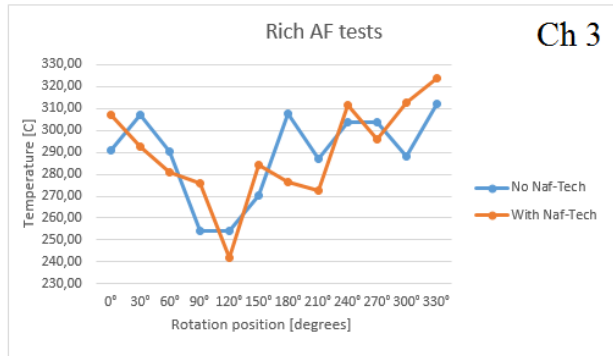
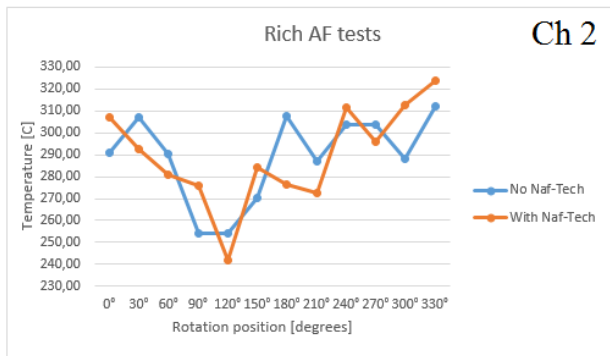


Figure 106 - CSIR diesel burner (rich tests)