Performance Study of a Compression Ignition Engine Fuelled with Biodiesel and Ethanol

Steffen Oberstein

Submitted in fulfillment of the requirements for the degree of
Master of Engineering Science (MEngSc)

University of Tasmania

April 2007

Declaration of originality

I declare that the thesis contains no material which has been accepted for a degree or diploma by the University or any other institution, except by way of background information and duly acknowledged in the thesis, and to the best of my knowledge and belief no material previously published or written by another person except where due acknowledgement is made in the text of the thesis, nor does the thesis contain any material that infringes copyright.

Steffen Oberstein

S. O.

Signature

17.07.08

Date

Statement of authority of access (copyright)

This thesis is available for loan and limited copying in accordance with the *Copyright Act 1968*.

Steffen Oberstein

S. O.

Signature

17.07.08

Date

Dedicated

To

My Loved Parents, Grandparents and Sister

Gewidmet

Zu

Meinen Geliebten Eltern, Grosseltern und Schwester

Abstract

Biodiesel and ethanol are recognized as a potential fuel of the future with several environmental advantages. While several published literature details the practical uses and applications of ethanol, little or no evidence is available in the public domain on the dual fuel mixture with biodiesel and ethanol and associated engine performance.

There is a large established diesel infrastructure on remote islands powered by generators. A good understanding of exhaust gas emissions by these generators can provide useful information on the environmental implication of emissions. There is an established knowledge on the quantitative reduction of harmful emissions when using biodiesel and ethanol. This knowledge can forecast the state of the engine performance and the other detrimental health effects it can have on the general population. A good understanding of the quantitative and qualitative trends is available in the literature, for CI engines run on biodiesel, as established knowledge. However, information of the reduced emissions and fuel consumption, using biodiesel and ethanol mixture is not extensively available in the public domain. Manufacturers of diesel generators have specific data available for use of biodiesel but the use and performance of their generators using dual fuels is not discussed.

In this thesis, rigorous design and modifications for conversion of a Kubota generator to run on biodiesel-ethanol dual fuel system is proposed and built from first principles. The test rig development associated with the calculations for fuel flow rates and associated engine management systems will be integral part of this overall systematic design. As part of this investigation an innovative fuel injection system, to accommodate biodiesel and ethanol, is designed and incorporated. Data acquisition systems to measure on-line measurements of engine performance such as the Brake Specific Fuel Consumption (BSFC) and emissions will be developed as part of this work.

In this investigation a comprehensive range of engine operating conditions will tested using both biodiesel-ethanol dual fuel. Over the range of engine operating conditions, emissions will be measured using on-board gas analyzer for systematic injection and increase of ethanol mixture. In this work, emissions such as Hydro Carbons (HC), carbon dioxide, carbon monoxide and No_x are measured. The qualitative and quantitative comparison of harmful emissions for B100 biodiesel and various ratios of ethanol mixture in the blends will be carried out.

The discussions will highlight the specific benefits, if any, of injecting ethanol and biodiesel. This work is a step towards understanding the levels of decreased emissions using bio-fuels and establishing qualitative and quantitative trends of engine performance on a sound mathematical and quantitative basis.

Acknowledgements

This study was carried out at the School of Engineering of the University of Tasmania during the year 2006 – 2008. I wish to express my deepest gratitude to everyone who has contributed to my study and supported my work towards this thesis. Especially, I wish to mention the following persons.

First of all I would like to issue my warm thanks to my supervisor, Associate Professor Vishy Karri. He has supported and inspired me throughout this project. I am very grateful for the opportunity given to me to work under his dynamic guidance.

To those who supported this project in the workshop and at the University, thanks; especially Nathan Smith, Jon McCulloch, Peter Seward, Bernard Chenery and David Morley, your assistant in setting up the test rig and developing the electronic control unit (ECU) was indispensable.

I would like to use this opportunity to express my special thanks to the beautiful island Tasmania. This time has been a very important and impacting chapter of my life. It has changed me as a person and opened my mind to other cultures and aspects in life. I met one of the warmest and happiest people on this island and I am glad to call few of them my best friends. It is difficult to name friends as I love them all but I would like to send thanks to Rob, David and Sushil who all supported and encouraged me in difficult time. I also want to mention James who has spend many hours prove reading my thesis. Without you guys this thesis would still be a work in progress; thanks.

Finally, I want to say thanks to my parents, my grandparents and my sister who continuously supported all my decisions even when it was hard for them times as my chosen path has not always been theirs. I love and miss you all.

Table of Contents

List of Figures

List of Tables

Chapter 1

Introduction

In the current global climate fossil fuels are quickly running out. The effect of CO2 emissions on the planet is creating increasing concern about global warming. Due to these issues companies from around the world have began striving to reduce CO2 and NO2 emissions and produce cheaper fuel. One of the fastest moving areas in alternative energies is the biodiesel industry. Biodiesel is the product of processing oils and fats from plants and animals (feed stocks) to produce ethylesters which can be used to fuel automobiles.

The realization around the world that renewable energy is an integral part of a sustainable earth has lead to action from many countries. Some countries have focused on alternative fuel production, while others still only have fledgling industries. Whatever the case there is no doubt that fuels such as biodiesel will develop into a large industry in most parts of the globe. This report aims to see the current state of development in Brazil, the United States, Germany, Japan and Australia. Australia can learn from these countries to continue to develop its own biodiesel industry. An attempt is made to make an overall global economic prediction for biodiesel

The demand for diesel in Australia is very large. With approximately 60% of all vehicles in Australia powered by diesel, a market for biodiesel certainly exists. The other main feature of the Australian landscape is the vast amounts of are not available, suitable for growing feedstock for biodiesel production. The feedstocks currently used in the world include castor beans, African Oil Palm, Sunflower, Babassu Palm, Soybeans, rapeseed and Cotton [1]. Australia also has major ethanol production facilities in Queensland together with increased efforts to commercialize biodiesel production.

Today's diesel engines require a clean-burning, stable fuel that performs well under a variety of operating conditions. Biodiesel is the only alternative fuel that can be used directly in any existing, unmodified diesel engine. Because it has similar properties to petroleum diesel fuel, biodiesel can be blended in any ratio with petroleum diesel fuel. Many federal and state fleet vehicles in Europe and the USA are already using biodiesel blends in their existing diesel engines. Over 30 million miles of testing in fleet engines proves biodiesel blend is the optimum choice of alternative fuel.

The low emissions of biodiesel make it an ideal fuel for use in marine areas, national parks and forests, Antarctica, and cities. Biodiesel has many advantages as a transport fuel. For example, biodiesel can be produced from domestically grown oilseed plants such as canola. It can also be produced from waste products such as tallow, the animal fat byproduct from freezer houses. Producing biodiesel from domestic crops reduces the Australia's dependence on foreign petroleum, increases agricultural revenue, and creates jobs. Biodiesel is the only alternative fuel in the US to complete EPA Tier I Health Effects Testing under section 211(b) of the Clean Air Act, which provide the most thorough inventory of environmental and human health effects attributes that current technology will allow. Biodiesel is the only alternative fuel that runs in any conventional, unmodified diesel engine. It can be stored anywhere that petroleum diesel fuel is stored. Biodiesel can be used alone or mixed in any ratio with petroleum diesel fuel. The most common blend is a mix of 20% biodiesel with 80% petroleum diesel, or "B20." Because B20 reduces emission gases exponentially more than 20%, it provides five times the benefit. The lifecycle production and use of biodiesel produces approximately 80% less carbon dioxide emissions, and almost 100% less sulphur dioxide. Combustion of biodiesel alone provides over a 90% reduction in total unburned hydrocarbons, and a 75-90% reduction in aromatic hydrocarbons. Biodiesel further provides significant reductions in particulates and carbon monoxide than petroleum diesel fuel. Biodiesel provides a slight increase or decrease in nitrogen oxides depending on engine family and testing procedures. Based on Ames Mutagenicity tests, biodiesel provides a 90% reduction in cancer risks. Biodiesel is 11% oxygen by weight and contains no sulphur. The use of biodiesel can extend the life of diesel engines because it is more lubricating than petroleum diesel fuel, while fuel consumption, auto ignition, power output, and engine torque are relatively unaffected by biodiesel. Biodiesel is safe to handle and transport because it is as biodegradable as sugar, 10 times less toxic than table salt, and has a high flashpoint of about 125°C compared to petroleum diesel fuel, which has a flash point of 55°C. Biodiesel can be made from domestically produced, renewable oilseed crops such as soybeans, canola, poppy seed, and animal fat or tallow. Biodiesel is a proven fuel with over 30 million successful US road miles, and over 20 years of use in Europe. When burned in a diesel engine, biodiesel replaces the exhaust odor of petroleum diesel with the pleasant smell of popcorn or french fries [2]. While biodiesel is gaining popularity in the rest of the world, in Brazil for example, ethanol from sugarcane is becoming popular as an alternative fuel. The commercial use of biodiesel starting with B2 creates a potential internal market of at least 800 million litres per year, in the next 3 years. This translates to a potential saving of US$160million per year, by not using imported diesel[1]. Currently, Brazil imports 10% of the diesel consumed. By mixing the diesel used with a 2% blend of Biodiesel, Brazil stand to make a saving of US$160million per year. If that is increased to 10%, the total amount of imported diesel can be eliminated, which would save the country around US$800million.The production of Biodiesel in Brazil can potentially reach levels where not only demands are met, but export opportunities exist.

Brazil is currently the leader in ethanol fuel production [3]. With 43.8% of all energy consumption in Brazil coming from renewable sources, such as hydro power, Ethanol and Biodiesel, they are currently leading the way with regards to a cleaner use of energy. The main issue for the long term future is finding enough fats and oils for biofuel production, without a dramatic impact on the food industry, and research on more viable crops will be required [4].

In this thesis, rigorous design and modifications for conversion of a Kubota generator to operate on biodiesel-ethanol dual fuel system is proposed and built from first principles. The test rig development associated with the calculations for fuel flow rates and associated engine management systems will be integral part of this overall systematic design. As part of this investigation an innovative fuel injection system, to accommodate biodiesel and ethanol, is designed and incorporated. Data acquisition systems using LabView programming techniques to measure on-line measurements of engine performance such as the Brake Specific Fuel Consumption (BSFC) and emissions will be developed as part of this work.

In this investigation a comprehensive range of engine operating conditions will tested using both biodiesel-ethanol dual fuels. Over the range of engine operating conditions, emissions will be measured using on-board gas analyzer for systematic injection and increase of ethanol mixture. In this work, emissions such as Hydro Carbons (HC), carbon dioxide, carbon monoxide and No_x are measured. The qualitative and quantitative comparison of harmful emissions for B100 biodiesel and various ratios of ethanol mixture in the blends will be carried out.

Chapter 2

Internal Combustion Engines

A dual-fuel engine is an internal combustion engine in which two fuels are injected separately. A gaseous fuel mixed with air is introduced into the cylinder during the induction stroke and is compressed during the compression stroke. Diesel injected into the combustion chamber near the Top Dead Center of the piston movement provides the ignition source for the gaseous fuel-air mixture. Therefore, dual-fuel engines have many features of operation and combustion process in common with both spark ignition and compression ignition engines.

This chapter gives the reader a fundamental understanding of internal combustion engines emphasizing operation, combustion process and emitted emissions. The first section outlines internal combustion engines in particular spark ignition and compression ignition engines to provide the framework to understand the complexity of dual-fuel engines. The last section describes operation and combustion process principle of dual-fuel engines by using a combustion process model developed by Liu and Karim [5].

2.1 Introduction and Classification

Internal combustion engines play a significant role in our every day life. The most common applications are for mobile propulsion in automobiles, ships and aircraft. This is due to their high power-weight ratio and the excellent fuel-energy, which provides many advantages to mobile applications.

Internal combustion engines are heat engines. The main feature of heat engines is the conversion of thermal energy into mechanical energy by exploiting the properties of a working substance, usually a liquid or a fluid. The thermal energy is produced by an exothermic reaction of the working substance and an oxidizer under high pressure and temperature. The high compressed gas then performs work during expansion and converts the work into usable motion. The occurrence of this thermodynamic process classifies heat engines into internal and external combustion engines. External combustion engines, such as steam and sterling engines, use an external combustion chamber. Hence, the combustion of the working substance takes place external from the mechanism providing the work. Common applications for steam engines are steam turbines to generate electricity.

This contrasts with internal combustion engines in which the combustion of the fuel-air mixture and the conversion of the thermal energy into mechanical energy occur in a confined space. This confined space is called an internal combustion chamber. The various types of heat engines are shown in figure 2-1.

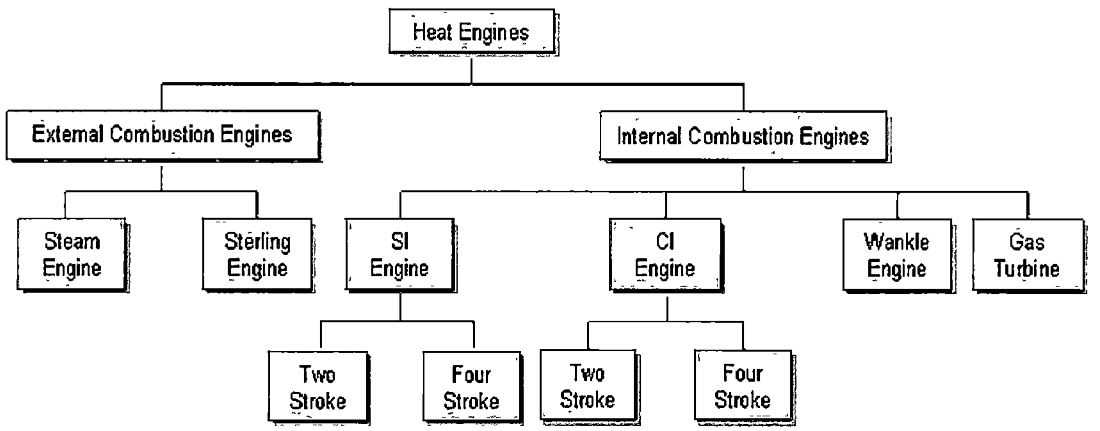


Figure 2-1: Heat engines [6]

Internal combustion engines are classified according to the number of strokes to complete one operation cycle and the mode of combustion [7]. The two types of operations are the two stroke and four stroke cycle. However, four stroke cycle is more commonly used in modern engines. The four strokes refer to intake, compression, combustion and exhaust strokes that occur during two crankshaft rotations per working cycle, which is also known as the Otto Cycle.

Each upwards or downwards movement of the piston is called a stroke. The cycles describing how many strokes are needed to complete one power stroke of an engine. One power stroke can be subdivided into two processes:

1. Gas exchange process: the product of the combustion is exhausted and replaced by fresh gases
2. Power process: the fuel-air mixture is compressed, ignited and the hot and high pressure gas expands and is translated into work by the engine

The fundamental difference between two cycle engines and four cycle engines is their gas exchange process. The two stroke engine completes one power stroke with each revolution of the crankshaft. The exhaust and charging process occurs simultaneously as the piston moves through the bottom dead center [8].

In a four stroke engine a full rotation of the crank is used to first force out the exhaust gases during the upwards movement of the piston and then charge the engine with fresh air during the downwards movement of the piston. Hence one power stroke needs 720° rotation of the crankshaft whereas two stroke engines operate in only 360° of the crankshaft rotation. The two stroke engine combines the gas exchange process and the power process, whereas a four stroke engine separates the intake and exhaust process.

The two modes of combustion are characterized by the ignition method used, spark ignition (SI) and compression ignition (CI). The Spark ignition engine uses a spark to ignite the air-fuel mixture and is often referred to as the gasoline or petrol engine according to the fuel being used. In the compression engine the charge auto-ignites due to the rise of temperature and due to the increased pressure in the combustion chamber.

2.2 Principle of Internal Combustion Engine Operation

A reciprocating engine, also known as a piston engine, utilizes a piston to convert pressure into a rotating motion. Figure 2.2 shows the main components of a typical 4 stroke engine.

E: Exhaust camshaft

I: Intake camshaft

V: Valves

S: Spark (glow plug in diesel engines)

P: Piston

R: Connecting rod

C: Crankshaft

W: Water jacket for coolant flow

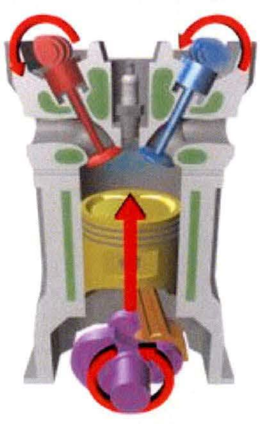


Figure 2-2: Components of a typical 4 stroke cycle engine [9]

A gaseous or liquid fuel is injected into the cylinder and mixed with air. The charge ignites, the hot gases expand and forcing the piston to the bottom of the cylinder, the Bottom Dead Center (BDC). Afterwards, the piston returns to the top of the cylinder, the Top Dead Center (TDC).

The linear movement of the piston is converted to a circular movement via the connecting rod and crankshaft. The upwards and downwards movement of the piston in the cylinder causes a frequent changing volume of an enclosed space. The space is called the combustion chamber, where the air-fuel mixture ignites due to compression or spark and the combustion occurs.

The valves on top of the cylinder allow the exchange of the gases within the combustion chamber. In particular, the inlet valve allows the inflow of the new air-fuel mixture into the cylinder, whilst the exhaust valve allows combusted gases to escape the cylinder. The valves are governed by the exhaust and intake camshaft. The camshaft and the piston are both driven by the crankshaft. The piston is driven at the same motion as the crankshaft, whereas the camshaft rotates in a ratio of 1:2 to the crankshaft. Therefore the camshaft completes two revolutions while the crankshaft completes only one revolution, which ensures the separation of the gas exchange process from the power process. The operation cycle of a 4-stroke internal combustion engine will be described in detail by the following section, compression engine.

2.3 Compression Ignition Engine

Compression engines are commonly known as Diesel engine named after the German inventor Dr. Rudolf Diesel, who was born in Paris on the 18th of March in 1858. He invented the prototype of a Diesel Engine in early 1892. Diesel intended to use a variety of different fuels such as coal dust and peanut oil [10].

Modern diesel engines are the most commonly used power source for many motor vehicles requiring high power output, especially buses and trucks. This is owing to the fact that diesel engines can run at a much higher compression ratio compared to spark ignition engines, which leads to higher efficiency and power output. However diesel engines are more expensive to manufacture due to their robust construction and sophisticated injection system.

Compressing any gas raises its temperature. The compression engine uses this principle for ignition of the diesel fuel. In Diesel engines only air is charged into the cylinder through the inlet valve. The compression ratio compares the volume of air in the cylinder before compression with the volume of air after compression [6].The air in the combustion chamber is compressed at a volume ratio from 12 to 20 by of the piston in the engine cylinder [11]. However, the compression ratios differ from engine to engine.

The diesel fuel is injected into the cylinder near the TDC position of the piston. The ignition occurs as soon as the first droplet of fuel is injected into the combustion chamber, since both the pressure and temperature of the air in the cylinder is very high. During the compression process the air can be heated up to ~500°C. Due to this high temperature the fuel ignites spontaneously without applying a flame or spark. An exception is the glow plug, used in some compression engines to improve the cold start characteristics.

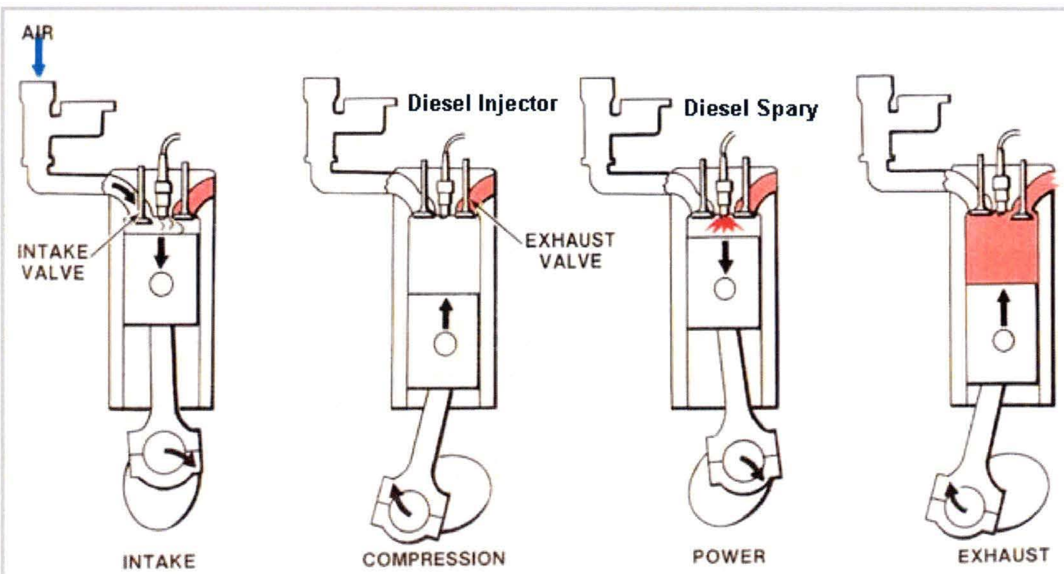


Figure 2-3: Operation cycle of a 4 stroke diesel engine [8]

I. **Induction stroke:** The piston moves from Top Dead Center (TDC) to Bottom Dead Center (BDC), the inlet valve is open and due to the downward movement of the piston a partial vacuum is created in the cylinder above the piston. Air is sucked in and the inlet valve closes.

II. **Compression stroke**: The piston moves from the BDC to TDC and the air trapped in the cylinder becomes compressed. When the Piston reaches TDC the air temperature reaches 500° to 650°. Diesel fuel is sprayed in fine atomized form into the combustion chamber and auto ignites, due to the high temperature of the air in the combustion chamber (the auto ignition point of diesel fuels is between 350°C and 450°C). Both valves remain closed during this stroke.

III. **Expansion stroke**: Due to the initiated combustion the gas expands and causes a downwards movement of the piston from the TDC to the BDC. The exhaust valve opens.

IV. **Exhaust stroke**: Due to the upwards movement of the piston, the combustion gases are pushed out. The inlet valve opens [6].

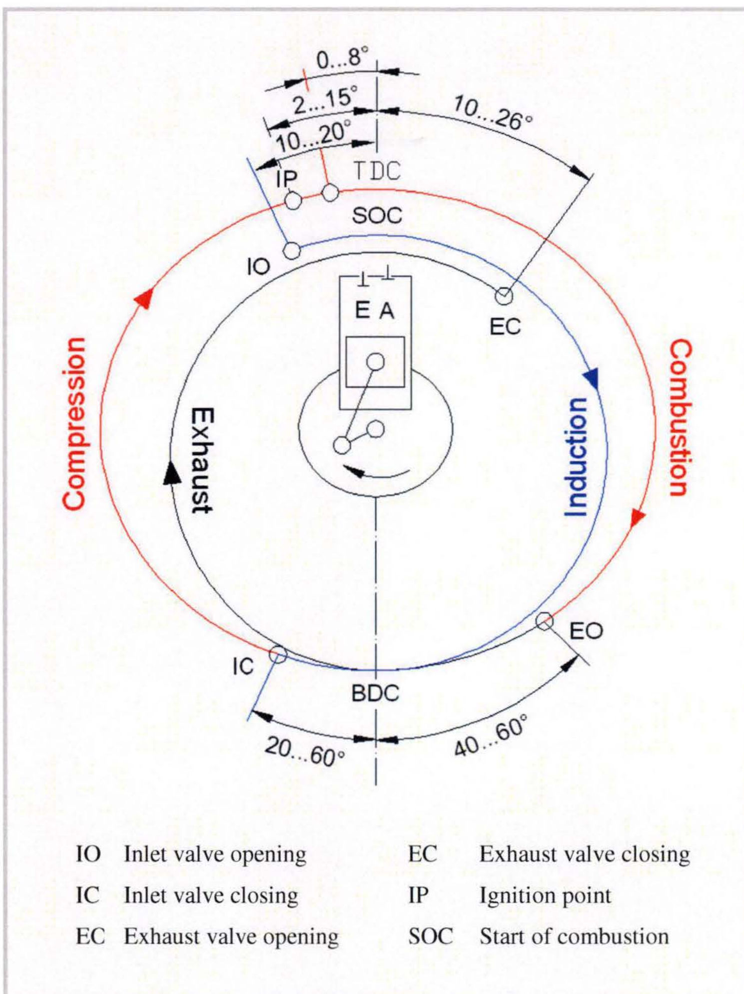


Figure 2-4: Valve timing of a four stroke diesel engine [12]

As displayed in Figure 2-4, the inlet valve opens already 10° to 20° before TDC during the exhaust stroke. The early opening of the inlet valve ensures that the valve is fully open when the piston reaches TDC and starts moving down. Also the inlet valve remains open for 20° to 60° after BDC which ensures that sufficient time is given for air to get sucked into the combustion chamber. The exhaust valve opens 40° to 60° before BDC in the power stroke and closes 10° to 26° after TDC. The early opening of the exhaust valve reduces pressure on the piston top and gives enough time for the combustion gas to discharge. Secondly due to the late closing of the exhaust valve the inducted fresh air pushes the exhaust gases out.

2.4 Spark Ignition Engine

A spark ignition engine, also known as gasoline or petrol engine is an internal combustion engine designed to operate on gasoline (petrol) or similar volatile fuels. It differs from a diesel engine in the method of fuel ignition and injection.

The fuel enters the combustion chamber as a homogeneous mixture with air via the air intake during the induction stroke, and it uses a spark to initiate the combustion. The spark is generated by a spark plug (Figure 2-5). In compression engines, the fuel is injected directly into the burning hot and compressed air at the end of the compression stroke, whereas in petrol engines, the fuel is premixed with air before the compression.

The operation principles of a four stroke internal combustion engines still apply as discussed in the section 2.3 Compression Engine. The main difference between diesel engine and petrol engine is in the actual combustion process, which is discussed in the following section 2.5 Combustion Process.

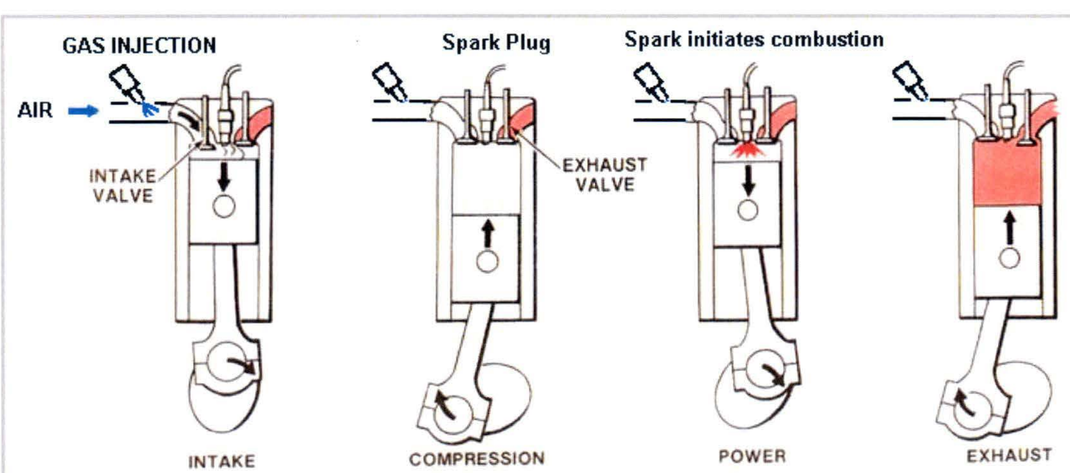


Figure 2-5: Operation cycle of a 4 stroke gasoline engine [8]

2.5 Principle of the Combustion Process

The following section introduces the fundamental principles of the combustion process of compression ignition and spark ignition engines. This knowledge will help to understand the combustion process occurring in dual-fuel engines, as it combines features of both engines.

While the chemical reactions during combustion are undoubtedly very similar in compression ignition and spark ignition engines, the physical aspects of the two combustion processes are quite different [11].

In the spark ignition engine the fuel is ideally in gaseous form and homogeneously mixed with air. The time of ignition is accurately controlled by an ignition spark at a specific crank angle. The charge ignites instantly from the spark. Due to the homogeneous mixture of fuel and air a definite flame front moves rapidly through the mixture in the combustion chamber. This generally accepted theory is based on chain reactions within the mixture since the energy is transferred from particle to particle in measurable velocities until the combustion is complete. Thus the pressure in the combustion chamber rises steady until it reaches its peak. This combustion behavior is called normal combustion and is due to flame propagation.

A complete combustion can only be achieved, if sufficient oxygen is available to burn all of the fuel injected. The following chemical principle applies to determine the theoretically required air mass for the combustion:

$$\lambda = \frac{Air\,mass}{Fuel\,mass \quad \times \quad Stoichiometric\,ratio}$$

λ = 1: The intake air mass is equal to the air mass theoretically required to combust all of the fuel injected

λ <1: The intake air mass is less than required. The mixture is considered rich.

λ >1: The intake air mass is more than required. The mixture is considered lean.

The stoichiometric ratio is represented by λ and specifies the air mass in kg required to completely burn 1 kg of fuel. The engine can theoretically operate in three different ways:

A basic rule that applies is that complete combustion does not occur inside the cylinder of internal combustion engines. This rule still remains valid even when the mixture inside the combustion chamber contains more air than required for combustion under stoichiometric conditions. Theoretically a lean fuel mixture enhances the combustion and therefore the thermal efficiency. However spark ignition engines operate less efficiently with lean air-fuel mixtures. The lean mixtures causes a much slower flame reaction and therefore the combustion process extends far into the expansion process and uncombusted fuel is discharged during the exhaust valve opening. Consequently, for maximum efficiency spark ignition engines typically operate with stoichiometric air-fuel mixtures.

Due to the compression of the gas and expansion of the burned part of the charge, other parts of the gas can reach a temperature where the charge starts to auto-ignites. This part of the charge is known as the end-gas part. This part of the gas has not been involved by the normal flame front reactions at this point of the combustion process. If a sufficient amount of end-gas autoignites, a significant pressure wave can be observed. This phenomenon is called detonation. It is generally accepted that detonations are caused by autoignition of the end gas [11]. It is often recognized as ping or spark knock. Detonation reduces the output and efficiency of the engine and causes stress on the cylinder and piston, which can lead to serious damage as shown in Figure 2.6.

In the diesel engine the fuel is sprayed under high pressure into the burning hot and compressed air. The fuel auto-ignites as the first droplet enters the cylinder. There is a short delay between the moment when the fuel starts to vaporize and the actual ignition associated by a measurable pressure rise. This time period is called the ignition delay period.

However, it must be noted that the droplet is surrounded by vapor immediately after entering the combustion chamber. Therefore the reactions must start in the vapor surrounding the droplet. The mixture concentration in the center of the droplet is too rich to ignite at this point of the process [11].

Figure 2-6: Damage on piston caused by pre – ignition and detonation [11]

Distinct from spark ignition engines, the fuel is sprayed directly into the combustion chamber, which causes an extremely inhomogeneous mixture of fuel and air. Also in diesel engines the fuel is not dependent on chain reactions to transfer the energy from one point to another as the pressure and temperature at the point of ignition are well above the auto-ignition point of the fuel. The local circumstances of temperature, pressure and the mixture of fuel and air control the combustion in the cylinder. The combustion process in diesel engines occurs in three stages [11]:

1. **Delay period** of the diesel fuel
2. **Period of rapid combustion**, burning of the fuel which evaporated and mixed with air during the delay period of the diesel fuel.
3. **Period of diffusion combustion**, burning of the remainder of the fuel, which has not found the sufficient oxygen to combust during the prior combustion phase

When the third period of combustion is extended into the expansion stroke, due to an excessively long delay period or poor injection characteristics, the result is a reduced output in power and hence low engine efficiency. When the delay period is longer or as long as the injection duration, sufficient time is allowed for the fuel to evaporate and mix with air before the actual ignition occurs. Consequently, all of the fuel charge burns at a rate similar to the end gas burning observed during detonations in spark ignition engines. This results in a rapid pressure rise in the combustion chamber and causes high stresses and heavy vibration of the cylinder and its associated parts. This knocking phenomenon can be observed in Diesel engines, similar to the detonation in spark ignition engines. Another cause for diesel knocking can be early fuel injection resulting in the appearance of pressure peaks before the TDC position of the piston.

As discussed, the delay period and the tendency of the fuel to autoignite play an important role for the combustion process. Consequently the ignition qualities of the fuel are an important factor for the tuning of the fuel injection to achieve optimum engine efficiency. The following fuel characteristics specify the ignition qualities of fuels in internal combustion engines.

- The **autoignition point** is the minimum temperature at which the vapor-air mixture of a liquid catches fire without the application of a flame or spark [13]. The autoignition temperature of a fuel will decreases as the pressure increases or the oxygen concentration increases in the combustion chamber.
- The **cetane number** is a measure of the ignition delay of a fuel used in compression ignition engines. The ignition delay represents the period between the start of injection and start of combustion of the fuel. In given diesel engines, fuel with a high cetane rating will have shorter ignition delay periods than lower cetane rating fuels.
- The **octane rating** is a measure of the resistance to autoignition of gasoline (petrol) and other fuels used in spark-ignition internal combustion engines. It is a measure of anti-detonation of a gasoline or fuel [14].

In summary, a high tendency to autoignite, or low octane rating, is undesirable in a spark ignition engine but desirable in a diesel engine.

2.6 Dual-Fuel Engines

The concept of dual-fuel-engine is not a novel development and the first attempts at operating compression engines on gaseous fuels can be traced back to the beginning of the twentieth century. However, there has been little interest in dual-fuel engine applications in the early twentieth century mostly due to the performance problems near the light load and full load regions. In the late second half of the twenty century the interest increased in dual-fuel engines as a result of the increasing fuel costs and the increasing awareness of the effect of greenhouse gases on the climate [15]. However, dual-fuel compression ignition engines have been employed in a wide range of applications such as railroad locomotives, mine trucks and diesel power generating systems. This concept allows the use of various gaseous resources in conventional compression engines without excessive engine modification costs.

Dual-fuel engines have characteristics of spark ignition and compression ignition engines. In dual-fuel engine mode, a gaseous fuel mixed homogeneously with air is introduced through the air intake into the combustion chamber of a diesel engine. The common gaseous fuels used are natural gas (NG) and liquefied petroleum gas (LPG). More recent research has attempted to inject oxygenated hydrocarbons into diesel engines, such as ethanol and alcohol. These fuels are an excellent alternative for high compression ratio engine applications because of their high knock resistant qualities [16]. However the most research has been devoted to natural gas applications in diesel engines. The engine operation and the combustion process features using natural gas are very similar to operating a diesel engine using other gaseous fuels. The findings can be therefore used to understand the operation and combustion principles of dual-fuel engines [17; 18; 19; 20; 21].

Diesel engines rely on the compression ignition of the fuel. Gaseous fuels used in diesel engines cannot ignite without an ignition source due to their low cetane number and high auto-ignition point. Consequently, the injection of a small amount of diesel is necessary to initiate combustion. In dual-fuel operation, the gaseous air-fuel mixture is injected during the induction stroke into the cylinder, as in spark ignition engines (Figure 2-7). The mixture is compressed during the compression stroke but unlike in spark ignition engines, diesel is injected towards the end of the compression stroke. The diesel ignites due to the heat of compression, just as it would in a normal diesel engine. The combustion of the diesel spray initiates the ignition of the gaseous air-fuel mixture in the cylinder [22; 15]. Due to the function of the diesel to start the combustion, it is referred to as pilot fuel. The engine operation can be changed between dual-fuel to pure diesel mode while the engine is running.

Due to the presence of a gaseous fuel, the combustion characteristics of dual-fuel engines are different from common diesel engines. The combustion process displays a complex combination of both compression and spark ignition engine operation.

Dual-fuel engines are therefore also known as hybrid engines. Dual-fuel engines generally control the power out put by the fuel quantity injected into the cylinder, unlike spark ignition engines which are using the throttle position to control the power output. Depending on the energy portion of gaseous fuel to the total energy, the engine operates closer to the diesel or Otto cycle. Hence when the bulk of the energy is provided by the preformed air-fuel mixture, the engine operates closer to the Otto cycle as the combustion is controlled by flame propagation through the mixture.

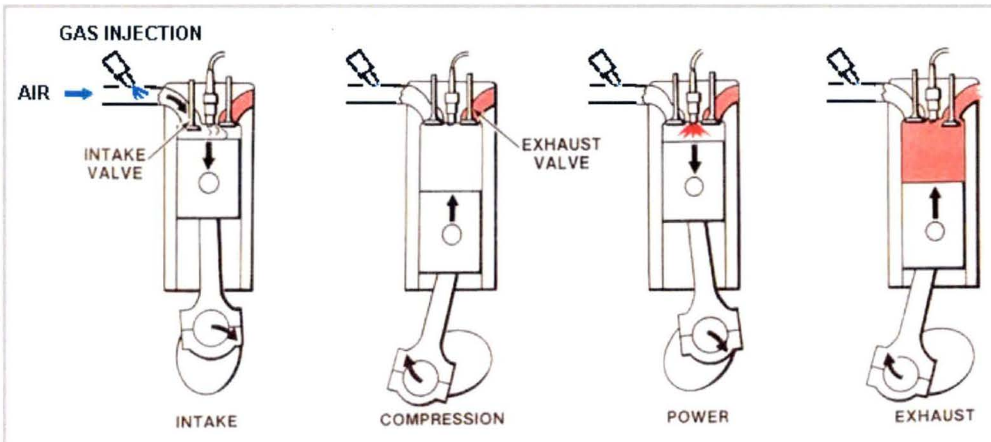


Figure 2-7: Operation cycle of a dual-fuel engine [8]

Many studies have been conducted to analyze the combustion process of dual-fuel engines in order to predict performance, emissions and the onset of knocks. To reduce prohibitive development time and cost associated with engine modification and test procedures, computer-based analytical models were developed to describe the complex combustion process in dual-fuel engines[23; 24; 25].

One of the most accepted models was developed by Liu and Karim [5]. They developed a multi-zone thermodynamic model that was able to describe the combustion process of dual-fuel engines and predict the onset of knock, performance and emission formations over the whole range of engine operation conditions [20]. The model uses a kinetic scheme to describe the oxidation of the gaseous fuel from the start of the compression stroke to the end of the expansion stroke and the interaction between gaseous and diesel fuels during combustion.

According to Liu and Karim, it is assumed that the combustion process occurs in four zones (Figure 2-8):

1. Unburned pilot fuel zone
2. Diffusion burned zone
3. Unburned gaseous fuel zone
4. Propagation zone

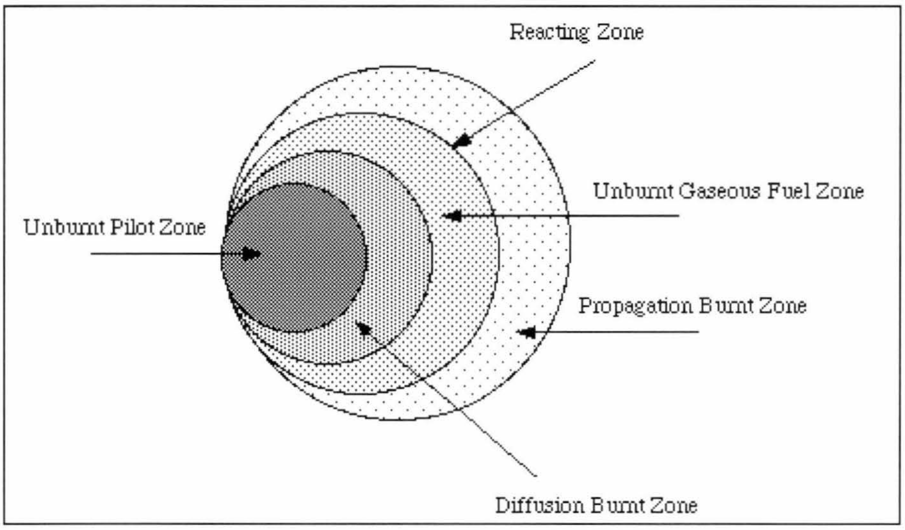


Figure 2-8: Combustion zones of dual-fuel operation [5]

Gaseous fuel is injected into the cylinder during the induction stroke and creates a homogeneous mixture with air in the cylinder. Diesel is injected in atomized form into the combustion chamber at the end of the compression stroke.

As discussed in section 2.5 Principle of the Combustion Process, the diesel droplet is surrounded by vapor immediately after entering the combustion chamber. Therefore the vapor mixes with the surrounding gaseous mixture and rapidly starts to burn and initiates the combustion. This zone is called the **Diffusion burned zone**. It must be noted that the delay period is slightly extended, due to diesel vapor mixing with the gaseous fuel. The cetane number of alcohols or other gaseous fuels is significantly lower than the cetane number of diesel. Consequently, the mixture of both fuel has a reduced cetane number compared to diesel [26].

The mixture in the center of the droplet is the **Pilot fuel unburned zone**, because it is too rich to ignite immediately. The **Unburned gaseous fuel zone** is the gaseous air-fuel mixture, which is compressed and heated during the compression stroke. During the combustion, more diesel forms new mixtures with gaseous fuel and starts to burn due to the diffusion combustion of the diesel vapor. The flame propagates through the charge towards the **Propagation burned zone.** The energy release is assumed to be at the edge of the burned zone due to the flame propagation through the mixture.

Combustion behavior changes according to the mixture concentration. In summary, high concentration of gaseous fuel in the combustion chamber results predominantly in flame propagation combustion but high diesel concentration in the combustion chamber results predominately in diffusion combustion.

However, many investigators report that dual-fuel engines operate best under moderate to high load conditions. Under these conditions, dual-fuel engines can reach or exceed the thermal efficiency of common diesel engines. They also can achieve lower CO_2, NO_x and particulate matter emissions [24; 27]. The reduced carbon dioxide emissions are simply due to the smaller carbon content of the gaseous fuel compared to diesel. The lower NO_x emissions can be explained as follows. The formation of NO_x emissions is favored by high combustion temperatures under the abundance of oxygen [28]. Therefore the effect of the temperature can be assumed as predominate in dual-fuel engines, as they operate with lean mixtures. Many investigators stated that a slight decrease in combustion temperature, as the delay period extended causes more of the combustion process to occur during the expansion stroke. Therefore a decrease in NO_x emissions is measurable for high concentrations of gaseous fuel in the combustion chamber [29].

The most concerning problems are the knock tendency and the low load efficiency of the engine. A dual-fuel engines operates with a lean air-fuel mixture. Due to the very lean mixture nature of the charge, the flame reaction is much slower under low load conditions than in moderate to high load conditions. Therefore the fuel burning is extended far into the expansion process, which results in a high emission content of unconverted gaseous fuel and carbon monoxide. However studies show that dual-fuel engines can operate with very lean mixtures and achieve higher thermal efficiencies than spark ignition engines. This is due to the high energy source of the pilot spray and their widespread diffusion burning within the cylinder [24].

Due to the hybrid nature of the combustion process in dual-fuel engines, the knock tendency of dual-fuel engines can be explained by two types of knock problems. The autoignition of the end gas causes the type of knocking, which usually can be detected in spark ignition engines. Knocking can also be caused by too early injection of the diesel fuel and the extended delay period. Due to the extended delay period, the conditions in the cylinder are insufficient to ignite the diesel fuel immediately and the diesel vapor has more time to create a mixture with the gaseous fuel an air. When the circumstances for autoignition are sufficient, the mixture burns in an undesirable rapid rate and consequences in high pressure peaks. This type of knock can usually be observed in diesel engine and is therefore referred to as diesel knock. Both types of knocking can occur within the same cycle [16]. However, using gaseous fuel in diesel engines is generally a good alternative.

2.7 Exhaust-Gas Emissions

Alternative fuels for internal combustion engines have become a significant issue during the last decades, which is the result of increasing concern for the environmental impact of engines, in particular green house and toxic component emissions. Another reason is the limitation of the primary fossil energy sources which society has to face in the near future.

The following section provides details of exhaust emissions produced by an internal combustion engine emphasising on their mechanisms of formation and act on human health and the environment.

2.7.1 Major Components

Complete combustion does not occur inside the cylinder of internal combustion engines, even when the combustion mixture contains excess air. Consequently, the engine's exhaust gas contains a number of toxic components in addition to a high level of non-toxic gases. Less efficient combustion leads to an increase of theses toxic components, which represent potential sources for damage on the environment and human health.

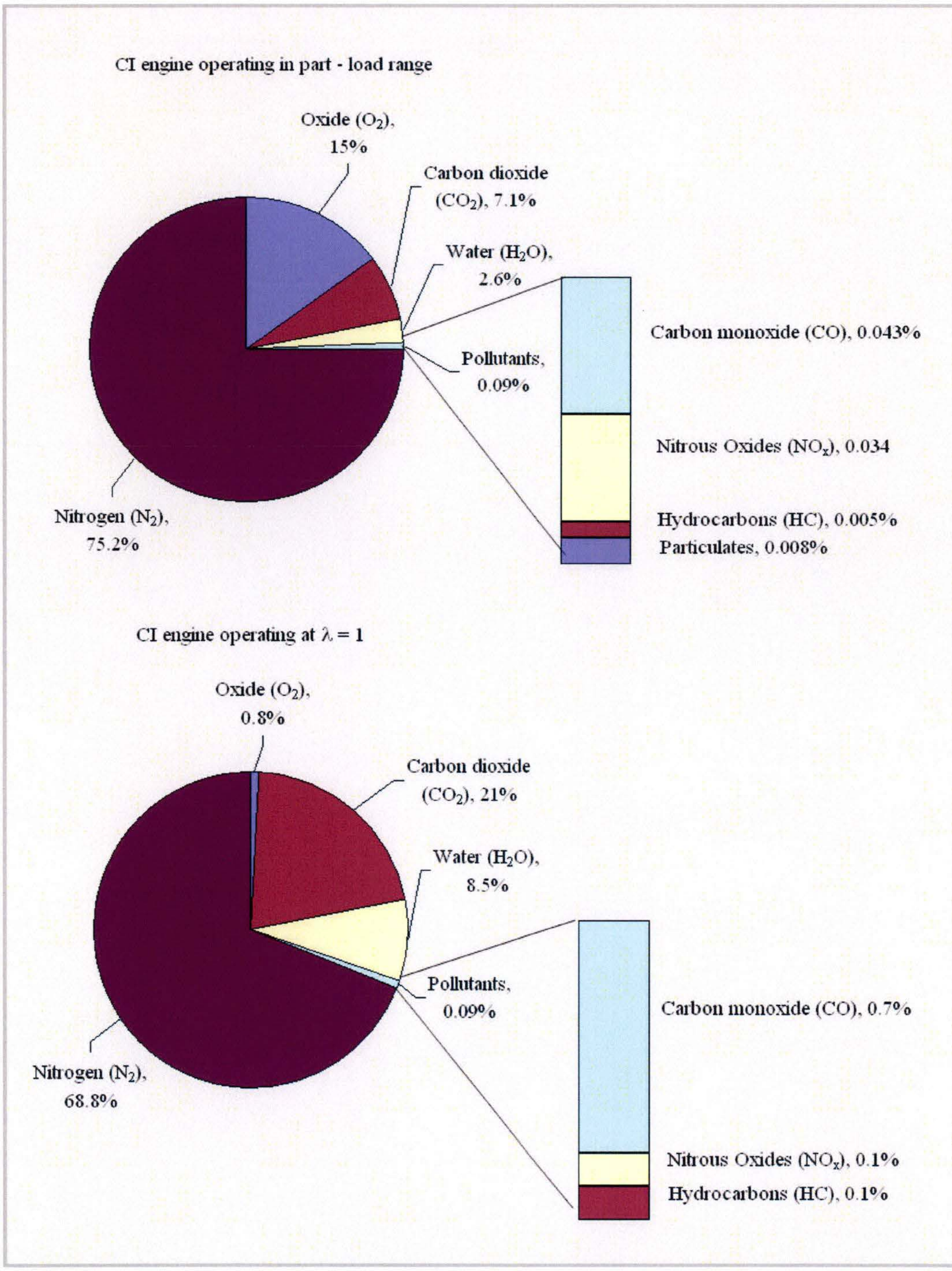


Figure 2-9: Exhaust gas composition of internal combustion engines [12]

The following gases are classified as pollutants:

- Carbon monoxide
- Hydrocarbons
- Nitrous oxides
- Sulfur dioxide
- Particulates

Engine modifications, exhaust-gas treatment systems and the less carbon containing fuels can reduce the amount of pollutants produced. Assuming the presence of sufficient oxygen, a possible complete combustion would occur according to the following chemical reaction [12]:

$$n_1\, C_xH_y + m_1\, O_2 \rightarrow n_2\, H_2O + m_2\, CO_2$$

In addition to the primary combustion products water (H_2O) and Carbon Dioxide (CO_2), the exhaust gases contain a number of toxic components as discussed above. Figure 2-9 represents the typical exhaust gas composition of internal combustion engines in part and high load engine conditions.

2.7.2 *Carbon Dioxide (CO_2)*

In complete combustion, the hydrocarbons in the fuel's chemical bond are transformed to carbon dioxide. The level of carbon dioxide in the exhaust gas is dependent on the operation point and condition of the engine. Its amount is directly proportional to the fuel consumption and can therefore be only reduced by the reduction of fuel consumption and the use of less carbon containing fuels.

Carbon dioxide is a natural product of a combustion process and in the past was classified as a pollutant. However, it is generally accepted that the rising CO_2 level in the atmosphere is one of the causes for the greenhouse effect and the associated global climate change [12].

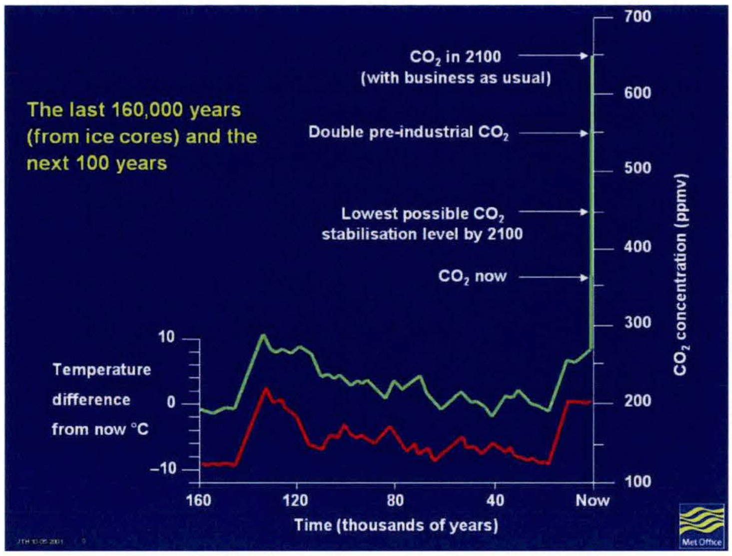


Figure 2-10: Relation between atmospheric temperature and CO_2 concentration [30]

Figure 2-10 represents the relation between the rising atmospheric temperature and the CO_2 concentration in the atmosphere over a long period, including the last ice age which is twenty thousand years ago. Projections of the carbon dioxide concentrations are shown on the right of the diagram. The global average surface temperatures and the CO_2 concentration for these curves are determined from an ice core bored out by Russian scientists from the Antarctic icecap. This fact justifies the efforts undertaken by many researchers and industry to reduce carbon dioxide emissions and fuel consumptions.

2.7.3 *Hydrocarbons (HC)*

Hydrocarbons represent a chemical compound uniting hydrogen with carbon. HC emissions are the result of inadequate oxygen being present to support a complete combustion and simply exhausted without being burned [14]. Exhaust emissions of diesel engines contain significantly less unburned hydrocarbons than in spark ignition engines. This is due to the fact that diesel engines operate on excess air rather than a spark ignition engine, which operates at stoichiometric air-fuel conditions. Therefore more air is available for the combustion with the result of less incomplete combustion products.

The level of unburned hydrocarbons is strongly influenced by the interaction of the fuel spray and the air in the combustion chamber as well as the properties of the fuel. A fine atomized fuel spray avoids formation of rich mixture areas in the combustion chamber as the droplets are widely distributed within the combustion chamber. That is the reason for a minimal presence of incomplete combustion products such as hydrocarbons in the exhaust gas of diesel engines, although diesel engines operate with excess air. Fuel droplets that fail to vaporize immediately from areas of rich mixtures in the combustion chamber do not combust completely [12]. The cetane number is another property of the fuel that can predict incomplete combustion products. A high cetane number provides a shorter ignition delay and shortens therefore the combustion duration of the fuel [31]. Consequently, more of the combustion process occurs during the compression stroke and as a result the combustion takes place more completely and hydrocarbon emissions decrease. The same principle applies for carbon monoxide as it is the result of incomplete combustion.

2.7.4 Carbon Monoxide (CO)

Carbon monoxide is a toxic, odorless and tasteless gas. Inhaled by humans, it can lead to asphyxiation, as it inhibits the ability of the blood to absorb oxygen. Carbon monoxide is the result of incomplete combustion when carbon is burned with a deficiency of air [14].

Carbon monoxide is formed during the intermediate combustion stages of hydrocarbon fuels. As combustion occurs the oxidation of CO to CO_2 takes place by reactions between CO and various oxidants. However, if the residence time is too short, insufficient oxygen or too low temperatures are present, more carbon monoxide will remain un-reacted and emitted as an exhaust gas by the engine [28; 32].

2.7.5 Nitrogen Oxides (NO_x)

In the exhaust gases emitted from internal combustion engines, NO_x refers to a class of components called nitrous oxides. Nitrogen oxides usually refer to a binary compound between nitrogen (N) and oxygen.

In internal combustion engines, the most significant nitrogen oxides are nitric oxide (NO) and nitrogen dioxide (NO_2). Nitric oxide (NO) usually represents the most abundant nitrogen oxide with 70-90 percent of the total nitrogen oxides [28]. Nitric oxide is colorless and odorless and coverts to nitrogen dioxide in atmospheric air. Nitrogen dioxide is a reddish brown gas with a penetrating odor and is poisonous in pure form. In highly polluted air, nitrogen oxides in form of acid rain can contribute to forest damage. It also reacts with hydrocarbons in combination with sun light and generates photochemical smog [12].

Most nitrogen oxides formed during the combustion are produced by a thermal mechanism and are therefore also known as thermal NO_x. Thermal NO_x is formed due to high temperature oxidation of the nitrogen contained by the combustion air. This thermal mechanism, also known as the "extended Zeldovich mechanism", is strongly dependant on temperature and the residence time of nitrogen at that temperature. Typically combustion temperatures must exceed temperatures of 1100°C to support this mechanism [28]. Due to this fact, high NO_x emissions are mainly a problem of compression ignition engines as their peak combustion temperature is significantly higher as in spark ignition engines.

The three important chemical reactions producing nitrogen oxides are:

$$O + N_2 \rightarrow NO + N$$

$$N + O_2 \rightarrow NO + O$$

$$N + OH \rightarrow NO + H$$

2.7.6 *Sulfur Dioxide (SO_2)*

Sulfur dioxide is a chemical bond between the elements sulfur (S) and oxygen. It is colorless and has a pungent, unpleasant odor. Sulfur dioxide is produced by combustion of fuels containing sulfur. However, the proportion of this pollutant is relatively small in exhaust gases of internal combustion engines. The main source of sulfur dioxide emissions are thermal power plants, industrial boilers and metal smelters.

Sulfur oxide emissions cause major damage to vegetation including forest and agriculture crops. Studies have shown that the exposure of high concentration of sulfur dioxide to plants can cause the loss of their foliage and become less productive or even die prematurely. On Humans, the exposure of sulfur dioxide in the ambient air has been associated with infections of the respiratory tracts, irritations of eyes, nose and throat and premature mortality [33].

2.7.7 Particulates

The term particulate, also known as particulate matter (PM) or emission smoke is primarily used for fine particles of solids or liquids suspended in the emission gas. Particulates discussed in this section are primarily a problem of compression ignition engines and the result of incomplete combustion of internal combustion engines. The levels of particulate emission of spark ignition engines are negligible.

Particulate formations are basically partly combusted or unburned hydrocarbon chains with a large surface ratio. Partly combusted and uncombusted hydrocarbons are joined by aldehydes and form these hydrocarbon chains which are also known as soot. Sulfur components contained in the fuel also often bond to the soot which consequently does not occur in sulfur free fuel. The soot usually has a penetrating odor [12; 28].

Smoke obstructs, reflects or refracts light and affects plant growth. Particulate precipitation can cause to damage buildings and vegetation. Many studies in the USA and other countries have shown that diesel emissions in particular particulate matter contains more than 40 known cancer causing substances. These toxic components contaminate the ambient air and potentially can cause cancer, infections in respiratory systems and premature death [34].

2.8 Concluding Remarks

Dual-Fuel engines display a complex combination of operation and combustion process features of both spark compression ignition engines and spark ignition engines. In this chapter operation, combustion process and emitted exhaust gas emissions of internal combustion engines were discussed to provide the framework to understand the operation of dual-fuel engines.

As discussed in Section 2.6, dual-fuel engines from compression ignition type represent an attractive option to utilize various gaseous fuel resources and minimize exhaust gas emissions while engine efficiency can be maintained at a similar level to those obtained with straight diesel fueling. At part load conditions dual-fuel engines suffer from low efficiency due to the lean gaseous fuel-air mixture. However, a sufficient operation of a dual-fuel engine technology still requires more complete analysis of various performance and operating parameters.

In the following chapter, the conversion concept of a diesel generator to operate on dual-fuel mode will be proposed. This involved minor engine modifications comprising the design and installation of fuel delivery system, injection control sensor system and air intake.

Chapter 3
Engine Conversion to Dual-Fuel Mode

This chapter highlights the conversion of a diesel engine to a dual-fuel engine fueled with biodiesel and ethanol. Biodiesel provides the source of ignition for ethanol and therefore is considered as the pilot fuel. The diesel injection system remains unmodified and ensures that the exact amount of fuel is delivered to maintain constant engine operation conditions.

A fuel delivery system and an injection control system were designed to enable the injection of ethanol into the air intake of the diesel engine (Figure 3-1). The engine conversion also involved the design of an intake manifold to install the fuel injector. Before installation of the ethanol injection system to the engine, a system test was conducted to ensure that the system operated according to given specification.

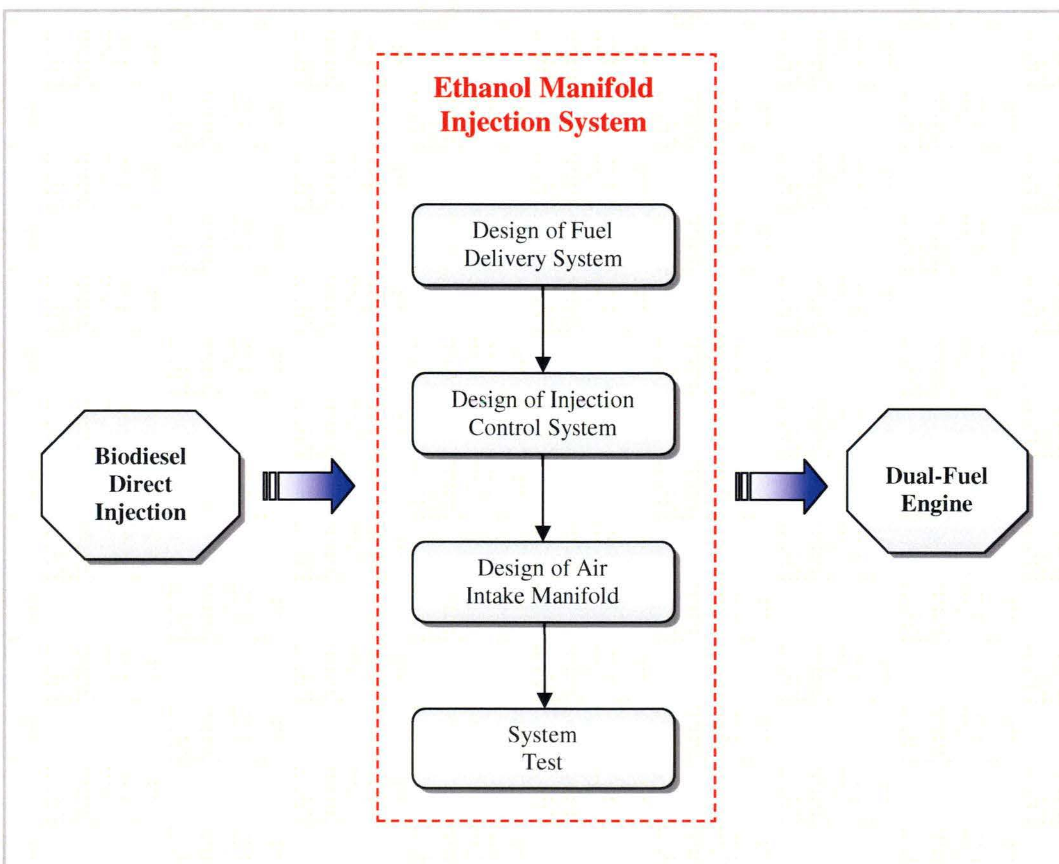


Figure 3-1: Design approach for engine conversion

3.1 Fuel Delivery System

The design of the fuel delivery system is based on systems used in conventional spark ignition engines (Figure 3-2). The fuel delivery system stores the fuel required, filters the fuel to remove suspended particles and to delivers the fuel to the injection system. The fuel is supplied to the injection system at a specific supply pressure under all engine operation condition to achieve the optimum fuel spray atomization by the fuel injector. The fuel delivery system presented is a return fuel system. This means that the fuel overflow is returned to the system via a fuel overflow line connecting the injector and the fuel injection supply line.

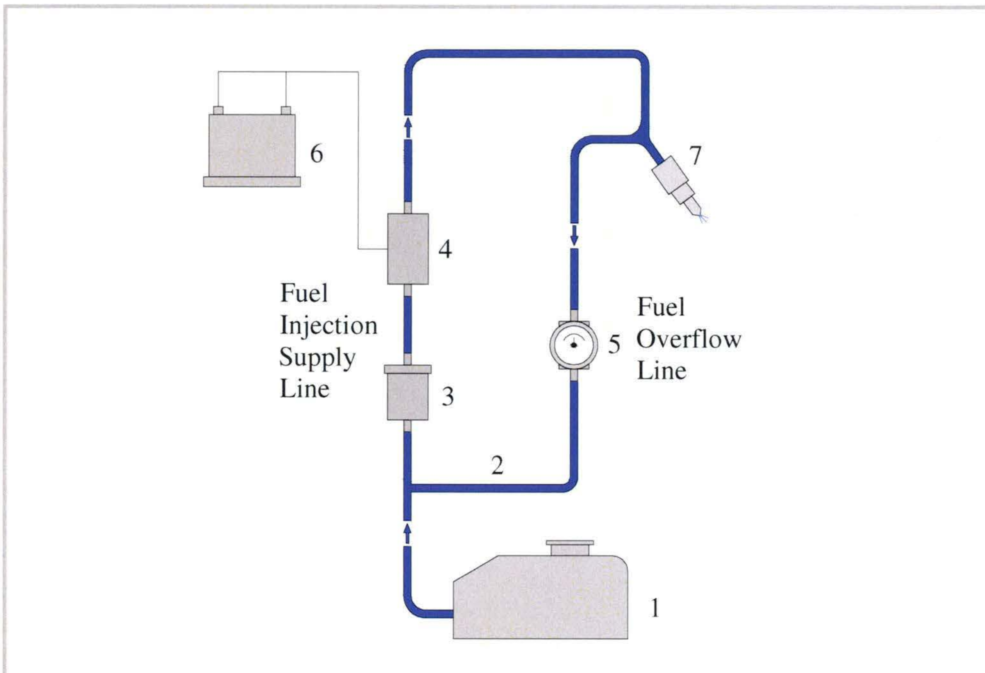


Figure 3-2: Fuel delivery system

The fuel delivery system shown in figure 4-2 consists of the following components:

1. Fuel tank
2. Fuel lines
3. Fuel filter
4. Fuel supply pump
5. Fuel pressure regulator
6. Power source
7. Fuel injector (component of injection control system)

All components are used in conventional low pressure fuel injection systems in the automotive industry complying with the Australian standard safety regulations such as fire resistance and pressure compatibility.

Many material compatibility studies have shown that ethanol has a corrosive effect on several parts of the engine and in particularly in fuel injection systems. Magnesium and lead is vulnerable to corrosion by ethanol. Non-metallic components are been affected by ethanol in particular elastomeric components such as rubber seal and o-rings. These materials have the tendency to harden and swell, which can lead to leaking of components [35; 36]. Consequently, material compatibility was an important factor for the component selection of the fuel injection system.

3.1.1 Fuel Supply Pump and Fuel Pressure Regulator

The fuel pump is responsible for maintaining a sufficient supply of fuel at the specific supply pressure to the fuel injector. Common manifold fuel injection systems operate with a fuel pressure between 2 and 4 bar depending on the type of injector. The fuel pump and fuel regulator attached to the test rig are shown in Figure 3-3.

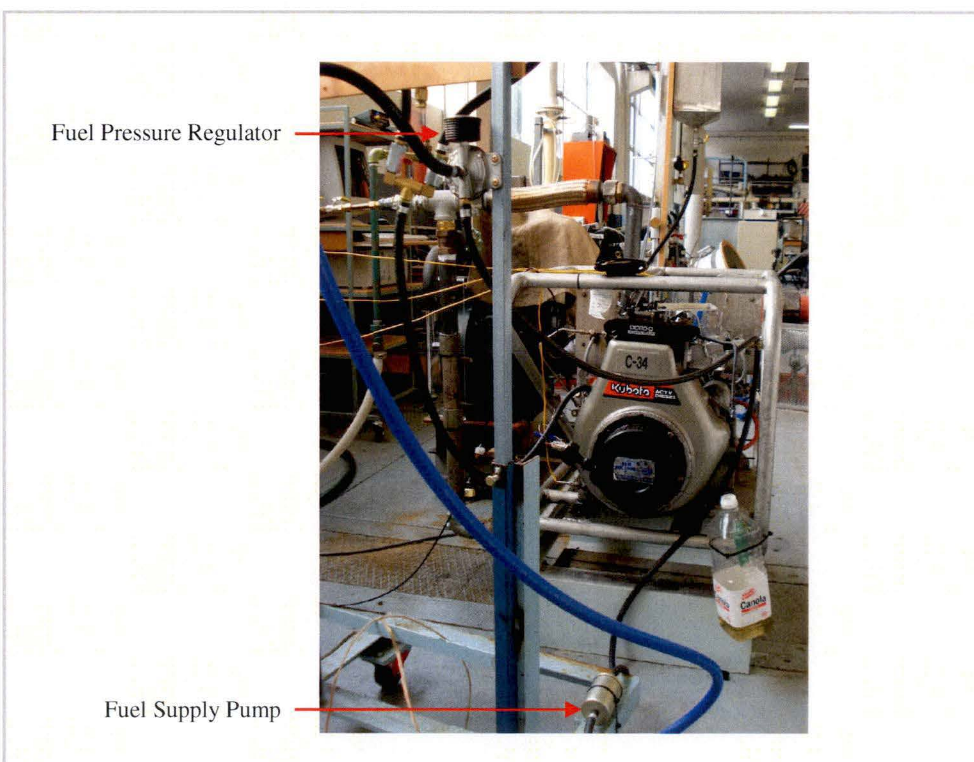


Figure 3-3: Supply pump and fuel pressure regulator assembly

The fuel electric fuel pump from the manufacturer Bosch circulates 148 liters per hour of fuel at a pressure of 5 bar. It is a positive displacement type pump; this means that the pump cannot produce pressure unless it is acting upon a restriction. The only restriction in this system is the fuel pressure regulator, also from the manufacturer Bosch.

The regulator controls the system pressure without effecting the pressure ability or flow volume capacity of the pump. The fuel regulator permits enough fuel to return to the tank so that the pressure drop across the injectors remains constant. Following installation instructions by the manufacturer, the fuel regulator is installed at the end of the fuel supply system to ensure that the system is flushed efficiently [37].

3.2 Injection Control System

Dual-fuel engines are hybrid engines combining features of compression engines and spark ignition engines. During this study ethanol mixed with air is injected into the cylinder during the induction stroke and is compressed during the compression stroke. Biodiesel injected into the combustion chamber provides the ignition source for the gaseous fuel-air mixture.

The operation of this type of engine requires two individual injection systems. The biodiesel injection requires a high pressure direct injection to overcome the high pressure occurring in the combustion chamber. The injection pulse is provided by a high pressure injection pump, which is controlled by mechanical governor. The diesel injection acts as the pilot injection and governs the fuel mass required relative to the crankshaft rotation and engine to maintain constant engine speed and operating conditions. The injected ethanol provides an additional energy source for the combustion process, which results in an adjustment of the fuel metering of the diesel injection system. Consequently, the diesel injection delivers the precisely metered amount of fuel to maintain constant engine operation conditions.

Figure 3-4 displays system used to control the injection of ethanol into the engine cylinder. The combustion process occurring in the combustion chamber is strongly depended on the way the fuel is injected into the cylinder.

To achieve an efficient combustion and a high engine performance the following criteria must be carefully balanced by the injection control system:

- Injection timing (start of injection)
- Injection duration (duration of injection)
- Degree of fuel atomization
- Fuel distribution inside the combustion chamber

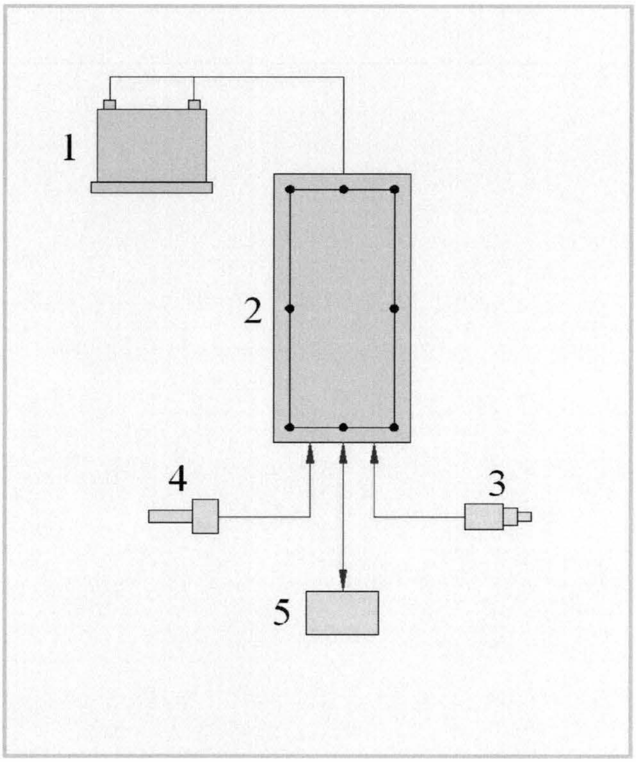


Figure 3-4: Injection control system

The injection control system shown in Figure 3-4 consists of following components:

1. Power source
2. Electronic control unit (ECU)
3. Fuel injector
4. Camshaft sensor
5. PC interface (control panel)

The camshaft sensor (4) detects the TDC of the piston movement and transmits this signal to the ECU. The ECU (2) processes this information and controls the injection timing and injection duration of the fuel injector (3) and therefore the amount of fuel injected into the combustion chamber. The injection control unit can be updated via the PC interface (5), with the required operation parameters for the ethanol injection.

3.2.1 Camshaft Sensor

The camshaft rotates at half the crankshaft speed. Considering that the engine operates in four cycles, the camshaft's rotational position can be used to identify the present operation cycle of the engine. The TDC of the piston movement is an indication as to whether the piston is in the compression or exhaust stroke. The camshaft sensor provides the ECU with this information.

Due to the extreme heat occurring during operation of internal combustion engines, sensors installed into the engine need to be heat and stress resistant. The senor selected to identify the TDC position of the camshaft is a Hall Effect sensor from type A3240 produced by the manufacturer Allegro. It is an extremely temperature stable and stress resistant sensor suitable foe operation over extended temperature ranges to +150° and therefore especially suitable for the use in internal combustion engines. A further advantage for this application is the relative wide air gap range between the trigger element and the sensor. As the name Hall Effect senor implies, such sensors use the Hall Effect for measurements.

For system installation, a permanent magnet is mounted to the camshaft, which generates a magnetic field strength perpendicular to the Hall element while it passes the camshaft sensor. As a result a voltage signal (Hall voltage) is generated by the sensor, which is in the millivolt range [12], displayed in figure 3-5. The signal is a digital signal and has only two stages, which are either "high" or "low".

The sensor is glued into a metal bolt, which is mounted to the engine body opposite the camshaft. A small air gap is left between sensor and the magnet to avoid possible collision. The mechanical calibration between sensor and magnet during the installation is important for an accurate TDC identification of the piston movement. The signal needs to be triggered by the sensor at the exact point in time when the piston movement is in TDC position.

The error range for the presented sensor system resulting of mechanical calibration error and measurement error of the element is estimated ± 2 degrees of the piston movement, which is an acceptable error for the operation of the presented injection system.

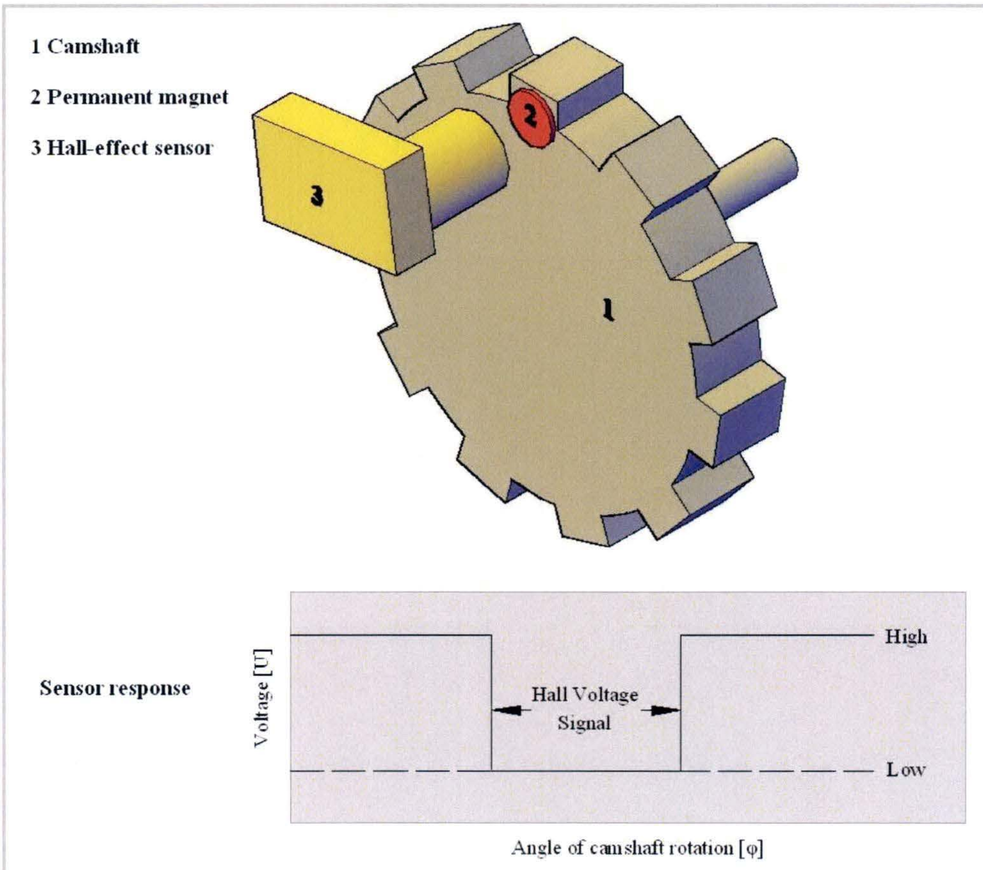
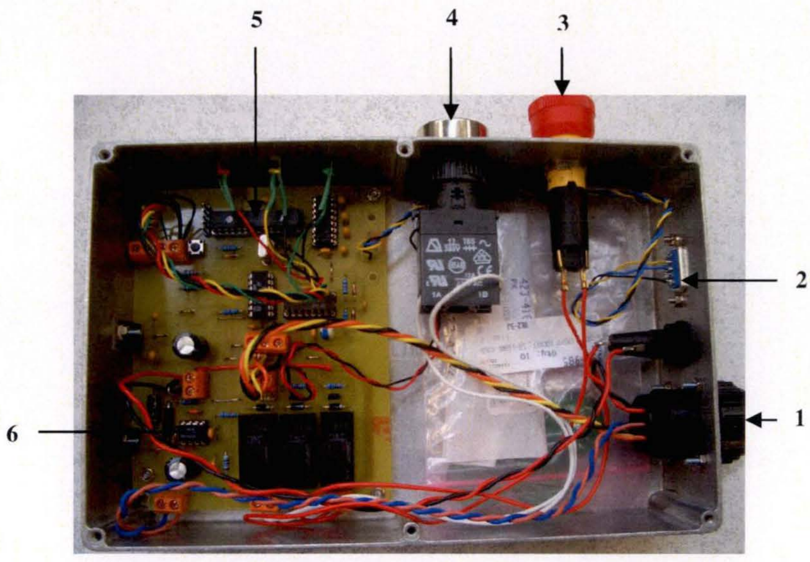


Figure 3-5: Camshaft senor assembly and senor response

3.2.2 Electronic Control Unit (ECU)

The electronic control unit receives the electrical signals from the sensors, evaluates them and then calculates the triggering signal for the fuel injector using specific mathematical calculation sequences. It controls the injection timing and injection duration of the fuel injector by means of an electrical output signal.

The ECU was programmed and built by Jon Mcallach at the University of Tasmania. Figure 3-6 displays the printed-circuit-board (pcb) with the electronic components installed into a metal case. The Power supply, sensor and injector are connected to the pcb through a multi-pole plug-in connector (1). The Microcontroller (5) and driver (6) are integrated into the case that in such a way that the heat dissipation to the case is ensured. The serial port provides an interface to a PC (2) with the installed software for the control panel. The run-switch (4) and the emergency-stop (3) control the power supply for the ECU.



1 Multi-pole plug-in connector
2 PC interface
3 Emergency-stop
4 Run-switch
5 Microcontroller (MIC)
6 Driver

Figure 3-6: Electronic control unit

Figure 3-7 describes the signal processing of the ECU. The peripheral components of the ECU are the injector and the camshaft. They represent the interface between the engine and the ECU in its role as the processing unit. The PC serves as the interface between the control panels transmits the calibrated operation parameters to the ECU and receives status information from the ECU. The microcontroller is the central component of the ECU. It processes the information and contains input and output channels and the serial interface to the pc. A safety function is integrated to prevent ethanol injection during engine over-speed or under-speed.

The input signals from the sensor and the PC interface serve as input variables for the microcontroller to calculate the output signal. With its output signal, the ECU triggers the driver and the driver directly operates the fuel injector. The software package "Lab View" was used to program the control panel. The project specified the requirements of the system software programmed by the School of Engineering Technician Jon Mcallach. The role of the control panel (Figure 3-8) is to update the ECU with the operation parameters and to monitor the ECU operation status, such as engine speed. Two different operation parameters can be calibrated using the control panel. The start angle sets the injection timing and the pulse angle the injection duration. The camshaft sensor is calibrated to trigger at the TDC between the compression and expansion stroke. Consequently, 360° needs to be added to the desired start angle to inject during the intake stroke.

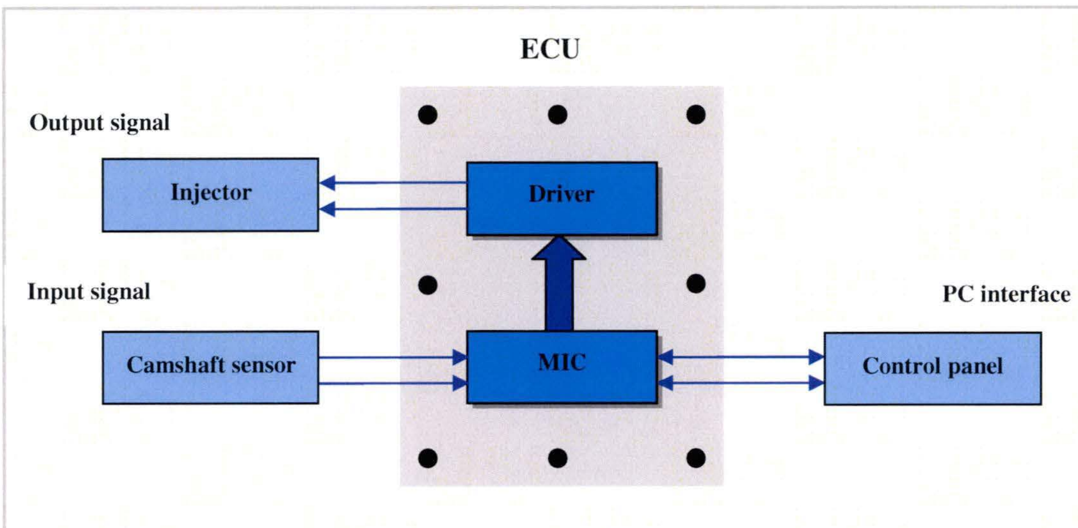


Figure 3-7: Signal processing in the ECU

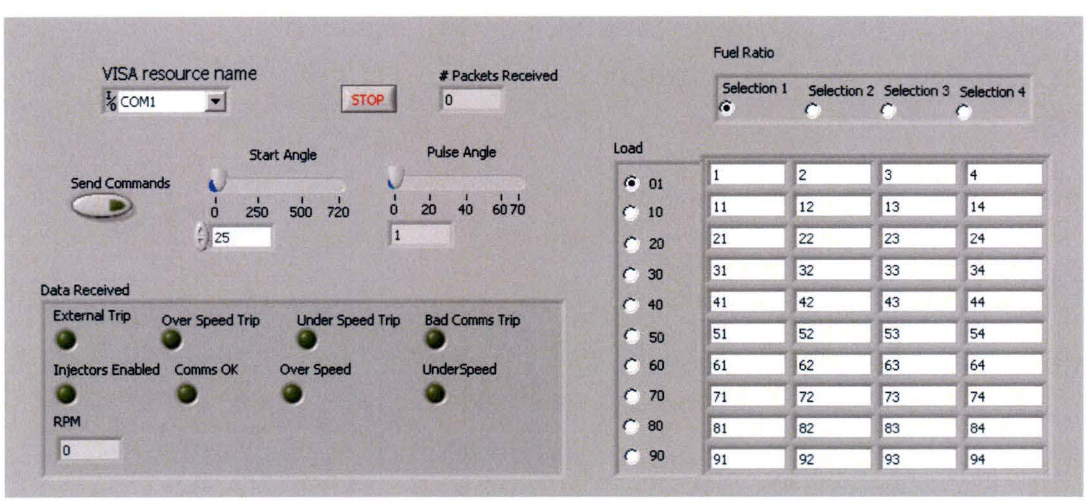


Figure 3-8: Lab view control panel

The matrix displayed at the right of the control panel in Figure 3-8 serves as input cells for the injection durations. It enables the operator to enter multiple injection duration into the system and to quickly switch in between them during engine operation without the need to enter the new parameter into the system. The system also can be upgraded with the integration of a mathematical algorithm, which selects the most efficient injection duration of ethanol depending on the current engine operation conditions.

3.2.3 Fuel Injector

The injector injects into the air intake the exact amount of fuel at the accurate pressure and precise instant in time as calculated by the ECU. The injector used to inject ethanol into the air intake is a Bosch solenoid valve injector, which has have a high frequency response, operates with low control power and is compact.

The fuel injector selected operates on a static flow rate of 120 g/min at an operating pressure of 3 bar. Bosch fuel injectors have their flow rates defined in n-heptane as part of their engineering specification. n-heptane is a pure chemical and does not have the same viscosity and density as ethanol. Hence the n-heptane flow rate figures stated can be used as a general guide only and an individual flow test needs to be conducted to determinate the actual flow rate for the injector operation with ethanol.

Solenoid valves operate with a dead time, which represents the time interval between the energization and the response of the solenoid valves. During this time period no fuel can be injected and therefore needs to be considered to determinate the injection duration for the ECU calibration. Hence, the value injection duration for the ECU calibration differs from the actual injection duration of the injector. For further discussions this value is defined as injection duration (ECU), which is the sum of the dead time of the injector and the injection duration of the injector. Consequently, two injector parameters need to be measured during testing, the injector dead time and the injector fuel flow rate. Figure 3-9 displays the test rig set up to conduct the injector fuel flow test. The test rig consists of the following devises:

- Power source
- Digital storage oscilloscope (DSO)
- ECU
- Function generator
- Fuel injector

To calculate operation parameters and engine speed the ECU requires a TDC signal from the camshaft sensor. Here, camshaft sensor signal is simulated by the function generator and therefore the test rig does not require the operation of engine. The function generator provides a stable signal eliminating possible fluctuations in engine speed, which would decrease the accuracy of the test results. A constant speed is required to precisely determine the injection duration and calculate the resulting fuel flow during a test cycle. TDC signal and the injector signal are displayed by the DSO to monitor the test procedure and calibrate the test rig.

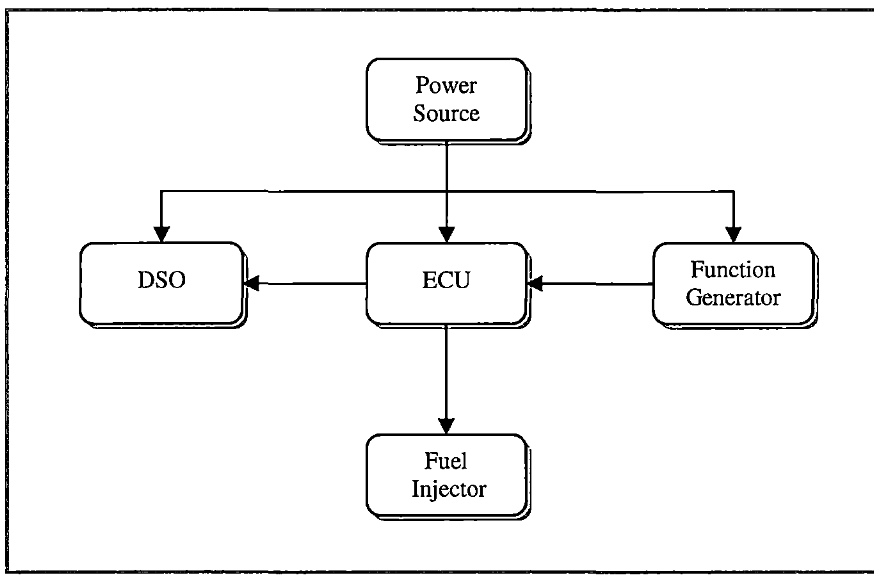


Figure 3-9: Test rig set up for injector fuel flow test

A sight glass enabled the measurement of the injector fuel flow. The measurement capacity of the sight glass is 20 milliliters with a resolution of 0.1 milliliter. To calculate the fuel flow the time for a particular displacement is noted as the fuel is consumed. The test was repeated for 10 different injection durations to ensure accuracy and consistency of the measurement. The following injector parameters represent the average of all test findings:

- Dead time = 1.22ms
- Injector fuel flow rate = 90 g/min

The tolerances of the glass vial resolution, the hand held time measurements and the fluctuations of the engine speed provide an accuracy of the flow rate of ±0.1 mL/s.

3.3 Intake Manifold Design

The intake manifold (air intake port) supplies the air-fuel mixture to the cylinder. It also is serves as a mount for flow meter, fuel injector and other automotive accessories.

The design of the manifold strongly influences the operation efficiency and the performance of the engine. The primary function of the manifold is to evenly distribute the air-fuel mixture to each cylinder to ensure that the fuel charge taken into each cylinder is from the same strength and quality. The design of the intake manifold is also an important factor for the volumetric efficiency of the engine. Abrupt changes in contours or rough transitions between segments of the manifold produce turbulences of the intake flow and consequent in pressure drops at these points. Therefore, smooth contours and transitions between segments are an important aspect for the design and manufacturing of the manifold.

The engine used for the experimental study is a single cylinder compression engine. Consequently, no fuel is introduced into the air intake during the induction stroke of the engine and the single function of the air intake is to conduct pure air to the intake valve of the engine. The modification of a compression engine to a dual-fuel engine requires the installation of a fuel injector to the air intake port, as ethanol is introduced into the air intake during the induction stroke. Further modifications of the air intake port are not necessary, as the experimental study is conducted under operation parameters given from the manufacturer.

The design of the air intake manifold is an assembly of two separate components:

- The air intake port
- The intake for the fuel injector

The air intake manifold design is orientated on the geometry given by the pre-existing manifold. The geometry of fuel injector intake complies with the given geometry of the fuel injector and allows a quick and easy removal or installation. The intake is sealed by an o-ring sitting near the nozzle end of the injector.

The position and angle of the fuel injector intake on the manifold strongly influences the evaporation rate of the fuel. To achieve a high efficient combustion, the air-fuel mixture has to be mixed homogeneously before entering the combustion chamber. Consequently, the fuel has to evaporate extremely rapidly to mix with air before entering the cylinder. This however requires that sufficient heat transfer occurs between the impinging point of the fuel spray on the air intake and the fuel to achieve acceptable mixture preparation. Martins and Finlay have shown that fuel spray should impinge directly on the back of the valve to achieve optimal mixture preparation and reported that any wetting of the intake walls results in lower quality of the mixture preparation [38]. Therefore, the injector intake sits close to the engine body and sprays with an angle of 25° at the center of the intake valve. Figure 3-10 displays the orientation and position of the injector.

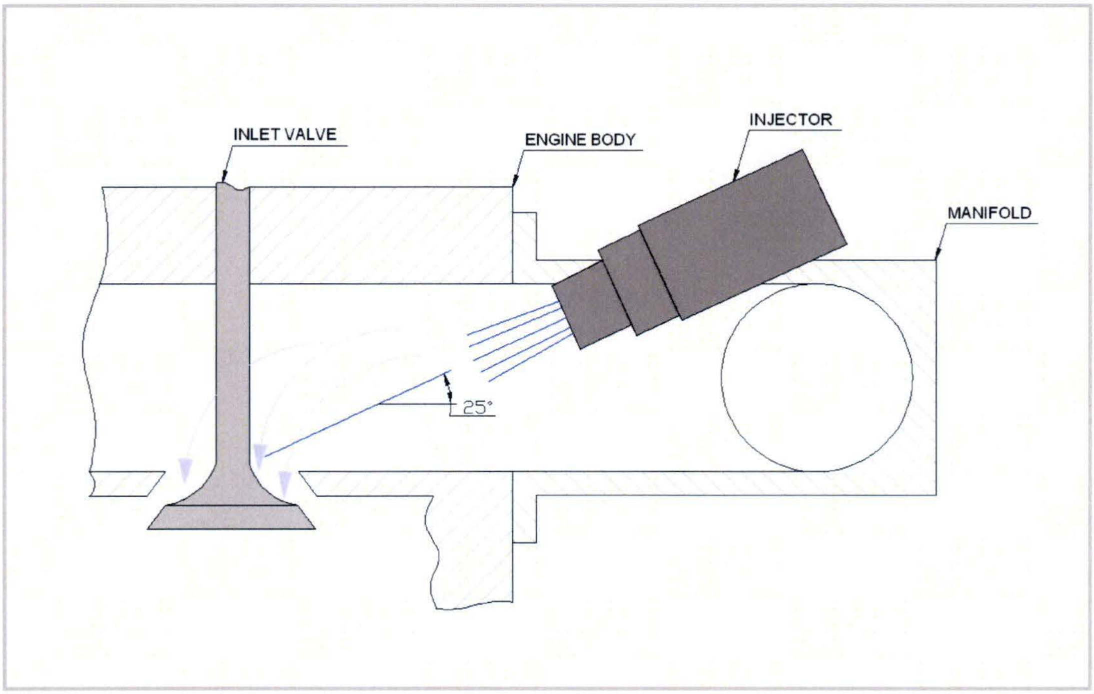
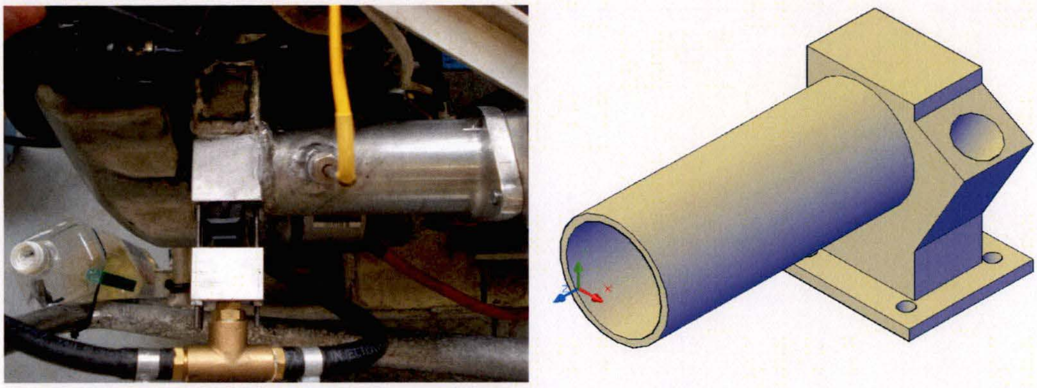


Figure 3-10: Injection diagram

The fuel injector intake and the air intake manifold are fabricated as two individual parts using a CNC machine with CAM technology using code generated by a AutoCAD drawing. Aluminum was used as a material for fabrication of the components. Fillet welding was used to assemble the two components. The fuel injector is clamped to the intake and the air intake manifold assembly is mounted by bolts to the engine block. The components have been manufactured and the assembled at the onsite workshop of the School of Engineering. Figure 3-11 displays the air intake manifold as an assembly and as a AutoCAD model.



(a) (b)

Figure 3-11: (a) Intake manifold assembly; (b) AutoCAD model of intake manifold

3.4 System Test

A system test is conducted on a complete designed system, to evaluate the system's performance and its compliance with given specifications. To ensure that the system operates as required, all input and output parameter need to be monitored and compared with the given specifications.

The test is conducted in two test cycles. The first cycle uses a function generator to simulate the TDC, whereas during the second cycle the signal is provided by the camshaft sensor (Figure 3-12). The test engine operates on 3000rpm and therefore the camshaft rotates with 1500rpm half the speed of the crankshaft. The two test cycles were conducted under the same operating conditions.

The following parameters were monitored and measured using a DSO:

- Camshaft sensor signal (TDC interval)
- Injection timing (start of injection)
- Injection duration (injector responds)

These parameters were compared to given specifications to ensure that the injection system is operating in compliance with the given specifications.

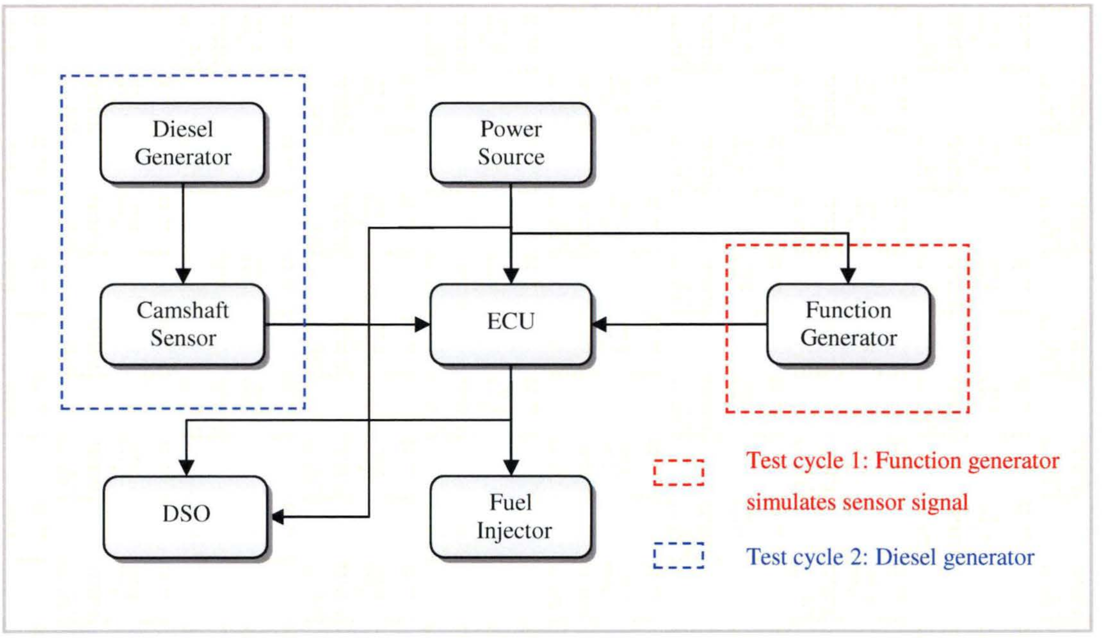


Figure 3-12: Process chart system test

During the test procedure the ECU operating parameters were set as followed:

- Injection timing: 400° after TDC
- Injection duration: 70°

Figure 3-13 represents the TDC voltage signal provided and the corresponding Injector voltage signal. The TDC is represented by the yellow line and separates the compression and expansion stroke and therefore the injection timing is set 400° after TDC to initiate the injection 40° after TDC during the intake stroke. One TDC interval represents a full engine operation cycle and therefore 720° of the crankshaft rotation. The crankshaft rotating at a speed of 3000rpm needs ~0.05 ms to fulfill one degree of a full rotation and therefore 40 ms to rotate 720°.

The DSO grid columns in Figure 4-14 represent the time scale with 10 ms per column in figure 4-14 9a) and 500µs in figure 4-14 (b). Figure 4-14 (a) confirms that the TDC interval is 40 ms and indicates that the injection pulse is initiated 400° after the TDC signal of the camshaft sensor. The wave form of the injector current signal shown in figure 4-14 (b) indicates injector pulse duration of 70°. The test findings confirm that the systems operate in compliance with the given specifications.

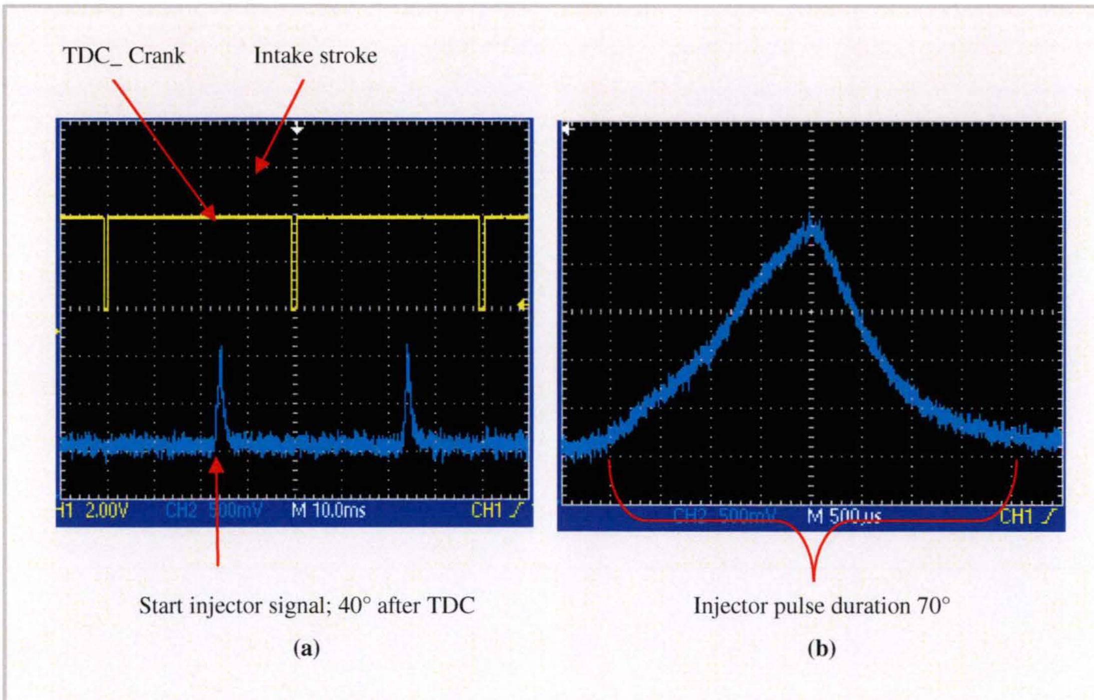


Figure 3-13: (a) TDC signal and corresponding injector signal; (b) Injector signal

3.5 Concluding Remarks

As outline in this chapter a novel injection system was developed to convert a compression ignition engine to operate on dual-fuel mode fuelled with biodiesel and ethanol. Integral part of the system was the design of a new air intake to enable the installation of the fuel injector and the development of the injection control system using LabView programming. As described in section 3.4, a system test was conducted confirming that the system operated according to given specifications. The following chapter describes the development of the experimental test rig incorporating the ethanol fuel injection system and the test measurement system.

Chapter 4
Experimental Test Rig Development

This chapter highlights development of the experimental test rig comprising experimental techniques, instrumentations and data acquisition methods. The aim was to analyze the performance of a small diesel generator operating on dual-fuel mode fuelled with biodiesel and ethanol.

The first section presents the test rig setup including the assembly of the ethanol injection system on the engine, the individual measurement instrumentation and methods, to enable the analysis of the given operation parameters. The second part of this chapter describes the test guidelines and procedure.

4.1 Test Rig Setup

This work investigates the benefits and detriments of the injection of ethanol into the air intake of a small diesel generator. The focus is on the performance and exhaust emissions of the engine. Various fractions of ethanol were injected into the diesel engine at varying load conditions to achieve comprehensive understanding of the effect of ethanol on the engine operation. Emissions, engine load, operation temperatures and fuel consumption of the individual-fuels were monitored during the test procedure (Figure 4-1).

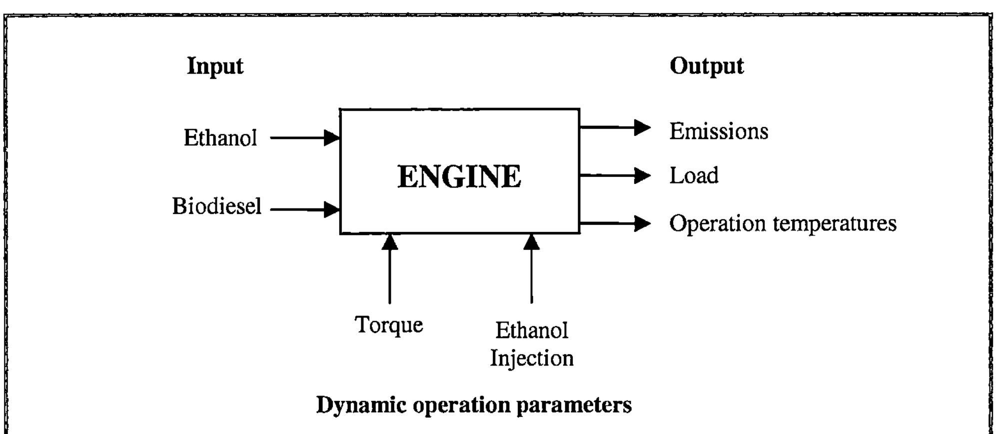
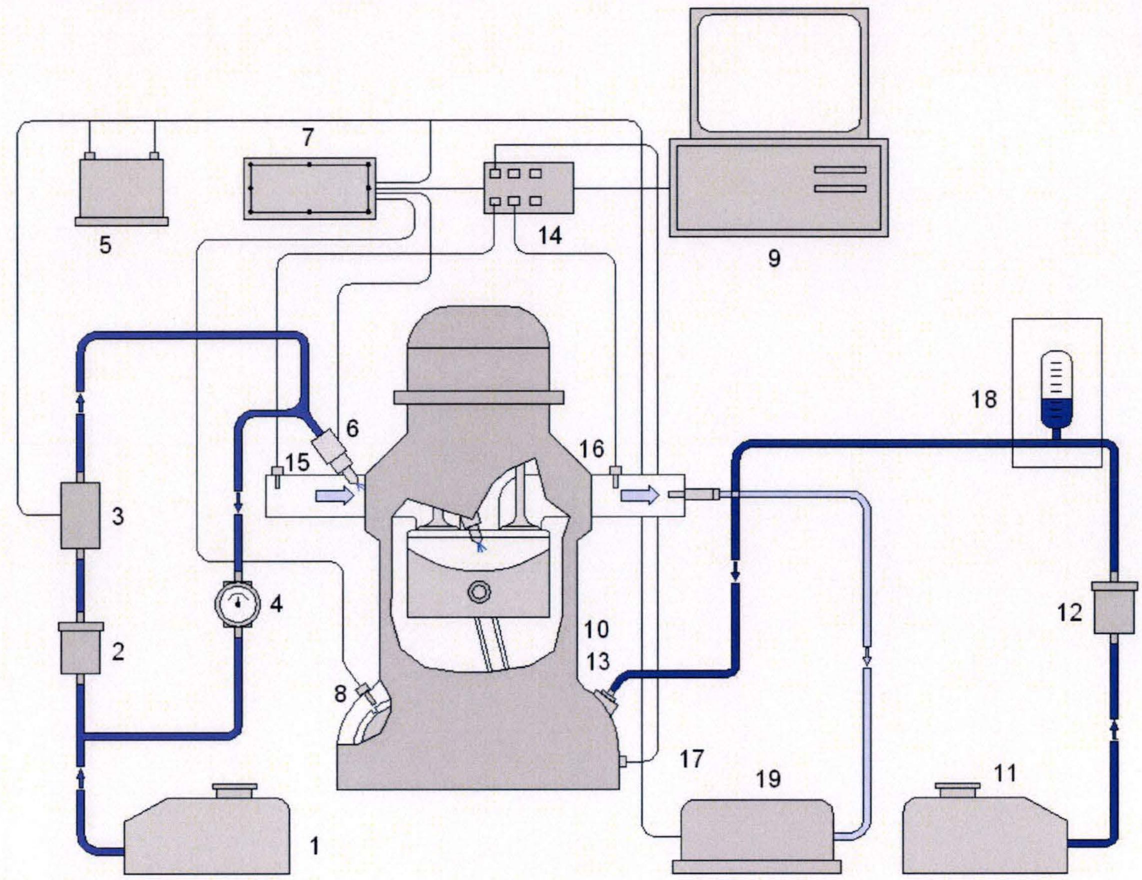


Figure 4-1: Test Parameters



Ethanol injection system

1 Fuel tank
2 Fuel filter
3 Fuel supply pump
4 Fuel supply pump
5 Power supply
6 Fuel injector
7 ECU
8 Camshaft sensor
9 Control panel

Diesel injection system

10 Diesel generator
11 Fuel tank
12 Fuel filter
13 Fuel supply pump

Measurement system

14 Data logger
15 Intake air temperature sensor
16 Exhaust air temperature sensor
17 Engine temperature sensor
18 Fuel flow meter
19 Emission analysis system

Figure 4-2: Test rig

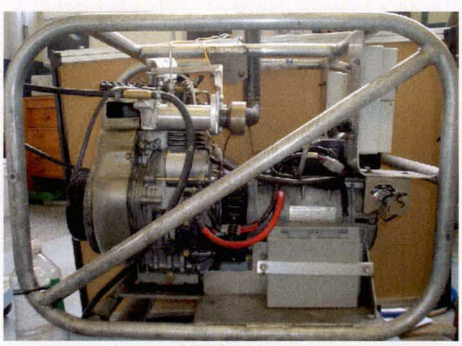
Figure 4-2 displays the test rig setup comprising ethanol injection system, diesel injection system and the measurement system. The test rig development involved the installation of the ethanol injection system and the measurement system to the test engine.

To measure the intake air temperature (15), exhaust temperature and engine temperature sensors are connected to a data logger (14), which provides the interface to the computer. The fuel flow meter (18) measures the biodiesel flow and the emissions analysis system measures gas emissions and opacity. The data acquisition was conducted using software provided by the measurement device manufacturer.

Fuel lines, fuel supply pump and fuel regulator are placed as far away from exhaust pipes, mufflers, and manifolds as possible, so that excessive heat will not cause vapor lock. All components are attached to the frame, the engine, and other units to minimize the effect of vibration and wear on sharp edges.

4.2 Test Engine

The main part of the test stand is an oil-cooled, 4 stroke, one cylinder diesel engine type OC80-D from the manufacturer Kubota. The Australian Antarctic Division, Tasmania has provided this specific engine for research and associated modification to the University of Tasmania. The Antarctic Division uses these engines a for power generation on Antarctic research bases. For this purpose the engine has been coupled to a synchronous alternator from the manufacturer Mecc Alte (Figure 4-3). The generator runs on a fixed engine speed of 3000 rpm and has a maximum output of 5 KW.



Model	Kubota OC80-D
Type	Air / Oil cooled 4 – stroke diesel engine
Swept volume	358 ml
Bore stroke	77 x 77 mm
Compression ratio	24.5
Engine speed	3000 r/min
Max. Power	5 KW
Governor	Centrifugal mechanical governor
Injection pressure	13.9 MPa to 14.7 MPa

Figure 4-3: Diesel generator OC80-D and specifications

4.3 Fuel Flow Meter

A sight glass was used to measure the volumetric fuel flow rate of the biodiesel. The sight glass was assembled on a movable frame and connected to the biodiesel delivery system between the fuel tank and engine (Figure 4-4).

The measurement volume of the sight glass is 80 milliliters with a resolution of 1 milliliter. To calculate the fuel flow, the time for a particular displacement is noted as the fuel is consumed. To ensure accuracy and consistency a minimum displacement of 20 milliliters of fuel is required for measurements. Including the tolerances of the glass vial resolution, the hand held time measurements and the fluctuations of the engine speed, the accuracy of the flow rate can be determined to ±1mL/s. The system is grounded to prevent possible electrostatic build up to reduce the explosion hazard.

Figure 4-4: Fuel flow meter

The flow measurement data is used to calculate the brake specific fuel consumption (BSFC) of the engine, a measurement for the engine efficiency. The BSFC is the ratio between the engine's fuel mass consumption and the crankshaft power produced [39]. The fuel flow of ethanol is measured indirectly by the injection timing. The fuel flow of the ethanol injection is precisely metered using the solenoid valve injector controlled be the ECU.

4.3.1 Load Bank

A variable load bank applies the load to the generator (Figure 4-4). The electricity provided by the generator is dissipated through the resistors as heat. The load bank provides a load range from 0 to 9 KW with increments of 0.25 KW and 1KW switches.

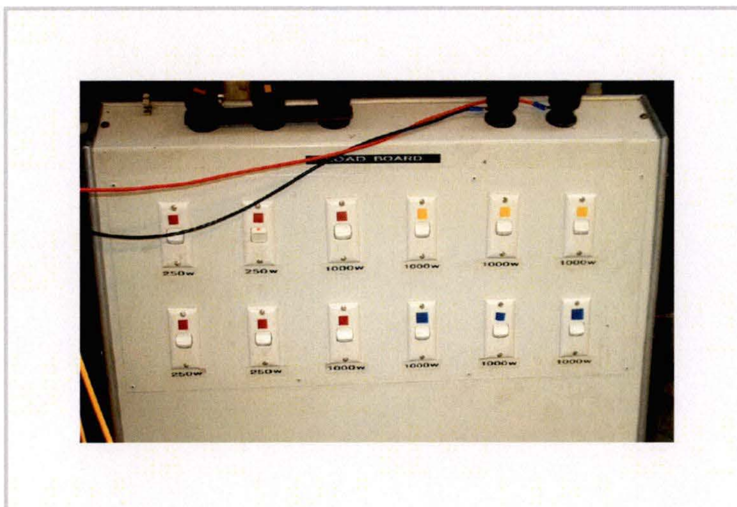


Figure 4-5: Load bank

A single phase 3184 Digital Power HI Tester from the manufacturer Hioki Corp. measures the load supplied by the generators. This data represents the brake power output of the engine (N_E). Measurements have shown that the load drawn is inconsistent with the specification of the resistor switches. The readings from the power meter varied ±15% from the specifications of the resistor switches. The error is allowed for by using a consistent combination of switches to obtain the required load and using the reading of the power meter for further calculations. Considering the error of the power meter of ±0.001 KW and load fluctuations a system error can be determined of ±0.01 KW.

4.4 Temperature Sensor System

Monitoring various temperatures is significant for the analysis of engine performance and the control of injection systems. Modern vehicles ECU's utilize the engine temperature to determine the optimum mass of fuel injected into the combustion chamber relative to the engine conditions. An example is the fuel enrichment for the cold start of an engine. In this project there is no temperature compensation for at ethanol injection.

Specific engine operation temperatures need to be monitored to ensure conditions remain within the specification limits given by the manufacturer. Exceeding those limits can shorten the life cycle of the engine. The following temperatures are monitored for the performance analysis:

- Air intake temperature
- Air exhaust temperature
- Sump temperature

The sensors used for the temperature measurement are thermocouples type K. Thermocouples are commonly used as temperature transducers. They are cheap and rugged and can operate over a wide range of temperatures. The main compromise is the precision since errors less than 1°C can be difficult to achieve. Thermocouples use the Seebeck effect to record temperatures by means of converting the temperature directly into voltage [40].

There is a wide range of different thermocouples classified by the material being used for the electrode. The selected type K is one of the most commonly used thermocouple (Figure 4-6). The material used for the positive electrode is Chromel, a nickel-chromium alloy and the negative electrode is made of Alumel, a nickel-aluminum-alloy. They are inexpensive and operate from 100°C to 1380°C with an expected bias error of ± 2.2°C or 0.75% [41].

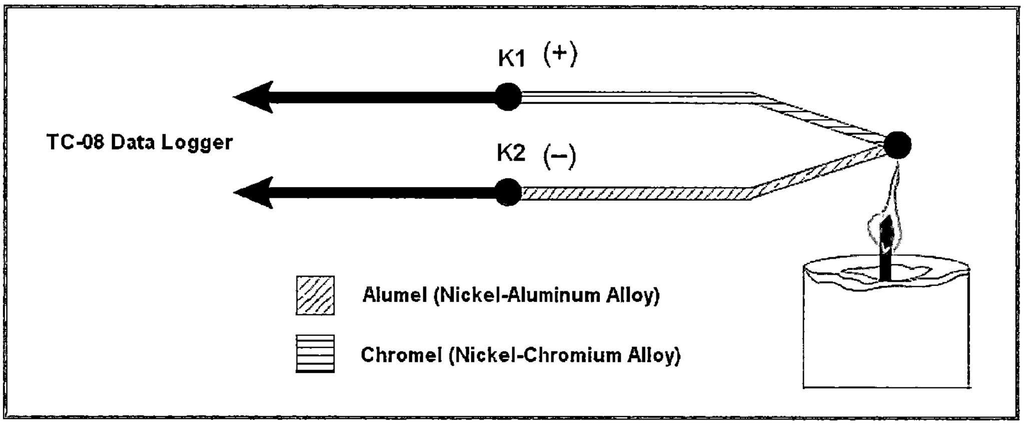


Figure 4-6: Thermocouple type K (adapted from [40])

The Thermocouples are connected to a TC-08 Thermocouple Data Logger from Pico Technology Lt. The data logger is linked via a USB connection to the computer and used for data acquisition (Figure 4-7).

Temperature range	-270°C to 1820°C
Temperature accuracy	±0.2 % or ±0.5 °C
Voltage accuracy	±0.2 % or ±10 μV
Thermocouple types supported	B, E, J, K, N, R, S, T
Number of channels	8

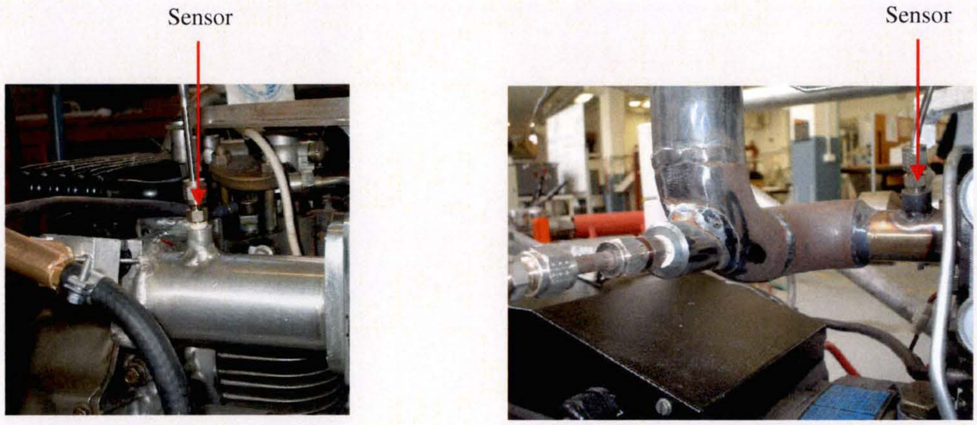
Figure 4-7: Data logger TC-08 and specifications [42]

4.4.1 Air intake and exhaust gas temperature

Air temperature is important in the combustion process in particular, the air intake temperature affects the evaporation of the injected fuel. Ethanol must evaporate and mix with air before entering the combustion camber. As discussed in chapter 2.5 "Principle of the Combustion Process", inhomogeneous mixtures can have a negative influence of the combustion process.

Higher temperatures in the air intake will promote the evaporation of ethanol droplets during the flight [38], which results in a more homogeneous mixture in the combustion chamber and higher engine efficiency. To monitor the air temperature in the air intake a thermocouple is mounted to the intake manifold (Figure 4-8 (a)).

As discussed in Section 2.7 "Exhaust Gas Emissions", NO_x formations in emissions are promoted by the high temperatures during the combustion process [43]. The exhaust temperature is monitored it is a good indicator of the combustion temperature. A thermocouple is mounted to the exhaust pipe to monitor the temperatures of the exhaust (Figure 4-8 (b)).



(a) (b)

Figure 4-8: Thermocouple mounted to (a) intake manifold; (b) exhaust pipe

4.4.2 Engine Temperature

The engine temperature is monitored for safety and performance analysis. The engine temperature must remain within the specification limits given by the manufacturer to avoid overheating and consequential possible damage of the engine.

The thermocouple is mounted in the sump plug of the engine (Figure 4-9. A hole has been drilled though the center of the sump plug and the thermocouple is placed through hole into to oil. The engine oil temperature represents a good indicator of the engine temperature since the engine uses circulation of oil to cool the combustion chamber.

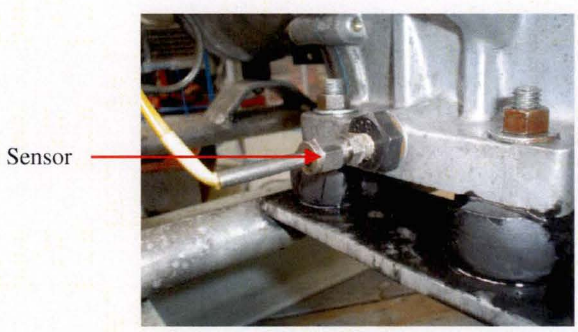


Figure 4-9: Thermocouple mounted into the sump plug

4.5 Emission Analysis System

The gas analyzer "TECPAC II" was used to measure the emissions of the engine (Figure 4-10). The analyzer is using a Non Dispersive Infrared method to measure HC, CO and CO_2 emissions and electrochemical cells for nitric oxide. Diesel emissions are usually associated with moisture and particulate matter, which need to be removed from the sample since they can affect the result. Therefore, a filtration system and a Delphi HDF296 water trap are installed on the analyzer. The system meets any current national specifications for emission inspections of vehicles on the road.

Emission	Range	Accuracy
HC	0 to 2000 ppm	±4.00 ppm absolute
CO	0 to 10 %	±0.02 % absolute
CO_2	0 to 16 %	±0.30 % absolute
O_2	0 to 25 %	±1.00 % absolute
NO	0 to 4000 ppm	±25.0 ppm absolute

Figure 4-10: Gas analyzer "TECPAC II" and specifications

The smoke rate of the exhaust was analyzed by the opacity meter "LCS 2100" from the manufacturer Sensors Inc. The opacity meter satisfies current national standards for smoke measurement of diesel vehicles on the road. The devise uses a partial stream measurement technique to monitor the smoke sample. The light source is a pulsed green LED with a peak of 560 nm. The method measures the particle of light being absorbed of the exhaust sample, represented by the opacity. On a scale of 0 to 100% opacity:

- Zero indicates no smoke in the exhaust
- 100% indicates the tube is completely blocked

Operational temperature	5°C to 40°C
Response time	1 msec
Humidity	0 to 95%
Pneumatic stability	± 1%
Opacity accuracy	± 2% relative
Pressure	0...10 kPa ± 0.3 kPa

Figure 4-11: Smoke meter "LCS 2100" and specifications [44]

4.6 Test Fuels

In this work, two fuels were injected into the diesel generator. Biodiesel was injected as pilot fuel directly into the combustion chamber and ethanol is injected into the air intake during the suction stroke.

The biodiesel used was developed by Associate Professor Vishy Karri as part of a project within the School of Engineering at the University of Tasmania. The recipe of the fuel uses oils derived from the locally grown crops canola (75%) and poppy seed (25%). ethanol is used as the alcohol and potasium hydroxide as the catalyst. The biodiesel was produced at the laboratories facilities of the University of Tasmania. The biodiesel production plant consists of following components [2]:

- Catalyst reactor CR-102
- Biodiesel reactor BR-103
- Settling tank ST-105
- Washing tank WT-107

The biodiesel was tested by an external company, Oilcheck Pty Ltd based in New South Wales, to ensure its adherence to the Australian standards. The results of the test confirmed that most specifications matched the standards except carbon residue.

A performance and emission study was conducted on a diesel generator operating on UTAS biodiesel by Master Student Rebecca Crosthwait. To achieve a comprehensive analysis of the UTAS biodiesel performance the test was repeated using other biodiesel blends and straight diesel. The results were compared with the results of UTAS biodiesel. The analysis showed that the Tasmanian biodiesel performed well concerning the emitted emissions and the engine performance. The results of this investigations showed that all emissions and the fuel economy are competitive with other biodiesel blends [2].

Properties	Ethanol (contain 6% water)	Biodiesel
Density at 20 °C	0.789 kg/L	0.886 kg/L
Calorific Value	26 MJ/kg	40 MJ/kg
Temperature of Self Ignition	425 °C	174 °C
Kinematic Viscosity at 40°C	4.66 m^2/s	4.66 m^2/s
Cetane number	8	48 -60
Octane number	121-130	-
Carbon content	~52%	~77%
Hydrogen content	~13%	~12%
Oxygen content	~35%	~10%

Table 4-1: Properties of ethanol [45] and biodiesel

4.7 Testing Procedure

The developed test procedure is based on the standard ISO 8178-4. The International Standard Organization (ISO) develops testing guidelines to ensure the comparability of results between different testing bodies.

The standards ISO 8178-4 specifies the test cycles for the measurement and evaluation of gaseous and particulate exhaust emissions of reciprocating internal combustion engines. To simplify further discussions the following terms need to be defined:

- **Test cycle** is a sequence of engine test modes each with defined speed, torque.
- **Test mode** is an engine operation condition characterized by speed and a torque [46].

The standard specifies several test cycles depending on the operation condition and the application of the engine. The test procedure tailored according to the on the "5-

mode" test cycle D1 for engine operating on constant speed. Table 4-2 represents the load applied for each test mode.

Mode No.	**1**	**2**	**3**	**4**	**5**
Torque %	100	75	50	25	10
Load [KW]	5	3.75	2.5	1.25	0.5

Table 4-2: Test cycle according to ISO 8178 (Test cycle D1) [46]

During a test cycle the load was kept constant and the proportion of ethanol to biodiesel was changed from 0% to 50 % or to the value at which diesel knock occurred. The test cycle was repeated for each given load during the test cycle. The complete test procedure is displayed in Table 4-3.

The amount of ethanol injected into the cylinder during engine operation is calculated based on the total energy required in the combustion chamber to meet a specified load. The total energy in the combustion required to maintain stable operating conditions, is represented by the fuel consumption of the engine fueled with biodiesel. Consequently, a preliminary test conducted on to determine the biodiesel consumption. Based on these results, the amount of Ethanol was calculated for each test mode and test cycle.

Test Cycle No.	**Load [KW]**	**Test Mode No. [Fraction of ethanol energy to total energy in %]**					
		1	**2**	**3**	**4**	**5**	**6**
1	5	0	10	20	30	40	50
2	3.75	0	10	20	30	40	50
3	2.5	0	10	20	30	40	50
4	1.25	0	10	20	30	40	50
5	0.5	0	10	20	30	40	50

Table 4-3: Test procedure adapted from ISO 8178

The standards specify the minimum test mode length to 10 min while each test parameter is measured and recorded, during the last 3. The time can be shorten or extended for small or large engines, if the time is sufficient to achieve a stable operation condition. Preliminary tests have shown that the test engine achieve stable operation condition after 2 min and therefore the test mode length was 5 min while the measurement and recording of the test parameter was conducted during the last 3 min.

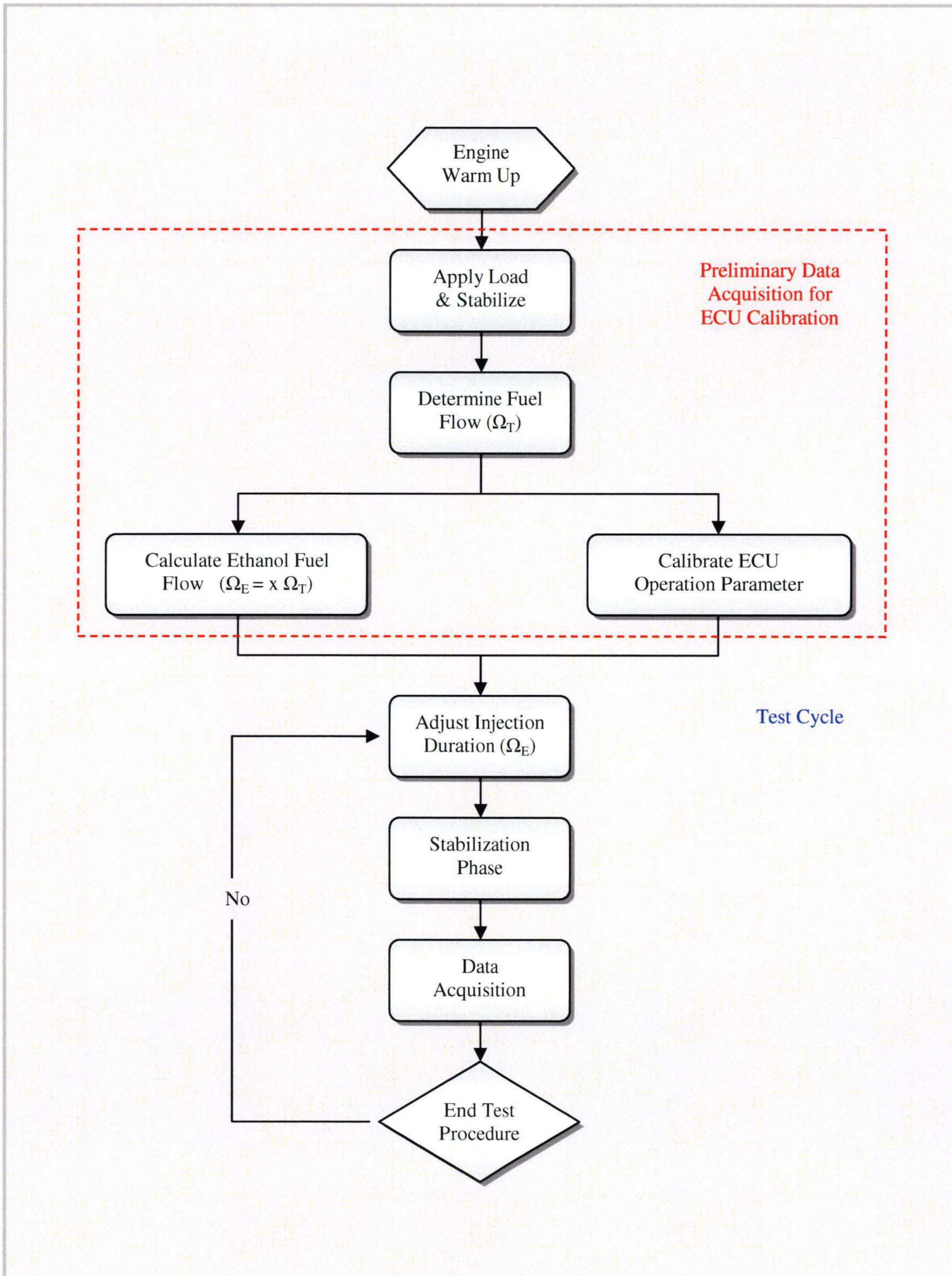


Figure 4-12: Test Procedure

Figure 4-12 shows that the test procedure is subdivided into the following 3 procedures:

1. **Engine warm up phase:** 15 min until engine achieving stable operating conditions
2. **Preliminary data acquisition:** During this phase the consumption was measured of the test engine operating on biodiesel. The result was used to calculate the required amount of ethanol and the resulting injection durations for each test mode. This step was followed by of the ECU calibration.
3. **Test cycle:** The test cycle involved the control of the ethanol injection and the data acquisition. The ethanol injection was adjusted for each test mode to inject the required amount of ethanol into the cylinder. The emissions, fuel consumption and engine operation temperatures were measured and recorded simultaneously.

4.8 ECU Calibration

The ECU controls the injection timing and injection duration of ethanol and therefore ensures that the required amount of ethanol is injected at the exact time into the combustion chamber. Volumetric analysis is used to determine the required amount of ethanol injected into the cylinder during an engine cycle.

It is assumed that the energy required on the piston is constant at a specific load and speed to maintain stable engine operating conditions. This energy can be determined by measuring the fuel consumption. Based on this assumption the injection duration for each test mode was calculated as follows:

$$\Omega_T = \Omega_B + \Omega_E \qquad [MJ]$$

$$\Omega_T = x_B \times \Omega_B + x_E \times \Omega_E \qquad [x_B + x_E = 1]$$

$$\Omega_E = x_E \times \Omega_B \qquad [\Omega_T = \Omega_B]$$

$$\Omega_B = \frac{C_B\ [L/\min] \times CV_B\ [MJ/kg] \times \rho_B\ [kg/L]}{3000\ [\min]} \qquad [MJ]$$

$$m_E = \frac{\Omega_E \ [MJ]}{CV_E \ [MJ/kg]} \qquad [kg]$$

$$Injection\,duration = \frac{m_E \ [kg]}{IFR \ [g/\min]} \times 60 \qquad [ms]$$

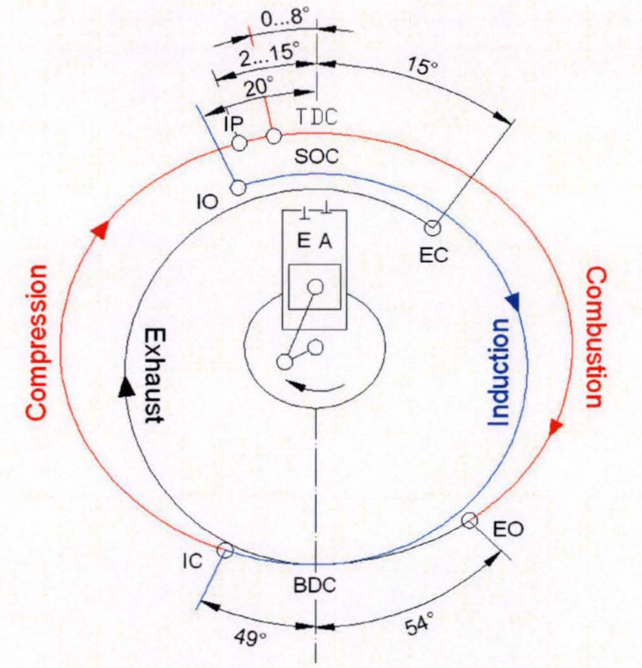
$$Injection\,duration\,[ECU] = Injection\,duration + injector\,dead\,time$$

where is:

Ω_T:	Total fuel energy	CV:	Calorific value
Ω_E:	Fuel energy ethanol	r:	Density
Ω_B:	Fuel energy biodiesel	IFR:	Injector flow rate
C:	Fuel consumption	x	Fuel Ratio
m	Mass		

The fuel-air mixture preparation process has a significant influence on engine performance and exhaust emissions and therefore the start of injection is an important engine operation parameter. Injection timing is also restricted by the valve timing. The fuel is injected during intake valve opening and exhausts valve closing, which ensures that no fuel can escape through the exhaust valve.

Movahednejad [47] has shown during an experimental study that the optimum point for the start of injection is the end of the exhaust valve opening and beginning of the intake valve opening. This injection strategy uses the backflow of the hot gases to the intake port, which causes better atomization and vaporization of the liquid fuel. Based on these findings and the valve timing of the test engine [Figure 4-13], the injection timing was calibrated at 20° ATDC. The injection timing remains the same for the entire test procedure and ensures that ethanol is injected into the cylinder during intake valve opening and exhaust valve closing. The fuel is injected shortly after the closing of the exhaust valve to achieve a better atomization and evaporation of the ethanol droplets caused by the heat of the gas backflow from the combustion chamber.



IO	Inlet valve opening	EC	Exhaust valve closing
IC	Inlet valve closing	IP	Ignition point
EC	Exhaust valve opening	SOC	Start of combustion

Figure 4-13: Valve timing test engine

4.9 Concluding Remarks

In this chapter an experimental test rig was developed to conduct a comprehensive performance analysis on a diesel generator operating on dual-fuel mode. Experimental techniques, instrumentation and data acquisition were selected to suit measurements of engine load, operation temperatures and fuel consumption.

As described in section 4.7, various proportions of ethanol at various loads were injected into the diesel engine to study the effect of ethanol on dual-fuel operation in detail. The results of the engine performance analysis as discussed in the following chapter are based on the measured operation data including brake power output, fuel consumption, the level of exhaust emissions of carbon monoxide, hydrocarbons, carbon dioxide, nitrogen oxides and the engine operation temperatures of the air intake and the exhaust.

Chapter 5
Experimental Results and Discussions

The tests as outlined in chapter 4 "Experimental Test Rig Development" were conducted and the measured data was collected for each fuel. This data was processed using statistical analysis tools and complied into an experimental matrix. The matrix represents the average values of all data obtained including brake power output, fuel consumption, the level of exhaust emissions of carbon monoxide, hydrocarbons, carbon dioxide, nitrogen oxides and the engine operation temperatures of the air intake and the exhaust.

The test results for each test cycle are represented as a function of the ratio of ethanol energy to the total fuel energy for a constant engine brake power output. The ratio of ethanol varies from zero to 50% or to the value at which diesel knock occurred. This procedure enables a comparison between emissions and efficiency at the same engine operation condition. To visualize the effect of ethanol at different engine load condition, biodiesel was used as a baseline fuel and change of test results were represented relative to the baseline at the given proportions of ethanol as a function of the brake power output.

The following discussions based on the test results will analyze the effect of ethanol on the operation conditions and performance of a compression ignition engine. Combustion process principles of dual-fuel engines (as discussed in chapter 2 "Internal Combustion Engines") are used to analyze and discuss the test data obtained.

5.1 Brake specific fuel consumption (BSFC)

Break specific fuel consumption (BSFC) is a widely used engineering measure to describe the fuel efficiency of internal combustion engines with respect to its output power. The BSFC is the amount of energy required to provide the specific net thrust for a given period.

$$BSFC = \frac{C\,[g/hr]}{N_E\,[kW]} \qquad [g/kwh]$$

where is:

C: *Fuel consumption*

N_E *Brake power output*

The brake specific fuel consumption was computed with the measurement of the fuel consumption, the brake power output and calorific value of both fuels. The BSFC depends strongly on load and speed of the engine. Figure 5-1 shows the break specific fuel consumption for the entire operation range of the engine at 3000 rpm for the various proportions of ethanol. Figure 5-2 illustrates the change of the BSFC relative to the biodiesel operation at various load conditions. It can be seen that the BSFC is:

- higher at high loads than at low loads for any proportion of ethanol (Figure 5-1);
- strongly depends on engine load and reaches its maximum at a load of 3.75 kWh (Figure 5-1);
- not strongly influenced by the addition of ethanol at moderate loads. The influence of ethanol increases at low and high loads (Figure 5-2);
- decreased at low loads proportional to the addition of ethanol. At high loads the BSFC slightly exceeds the BSFC value of biodiesel (Figure 5-2).

The combustion process is strongly dependant on combustion temperature and level of oxygen in the combustion chamber. At low loads oxygen is available in abundance due to the nature of the diesel engine and the influence of the combustion temperature is therefore predominating. Higher temperatures promote the burning of the fuel charge resulting in a more complete combustion which is represented in a lower BSFC. Figure 5-1 shows that the lowest BSFC was obtained at engine operation with 3.75kW for any proportions of ethanol. At maximum load conditions the fuel-air ratio is very and insufficient oxygen is present for the combustion, which results in a slight decrease of BSFC at the maximum brake power out put.

The influence of ethanol on the combustion process is more complex. The existence of a gaseous fuel-air mixture in the combustion chamber changes the nature of the diesel engine. Diesel vapor mixes with the surrounding gaseous fuel-air mixture; ignites when the flammability limit is reached and burns in a diffusion flame near stoichiometric conditions. The diesel pilot flame acts as ignition source for the ethanol-air mixture which burn due to flame propagation.

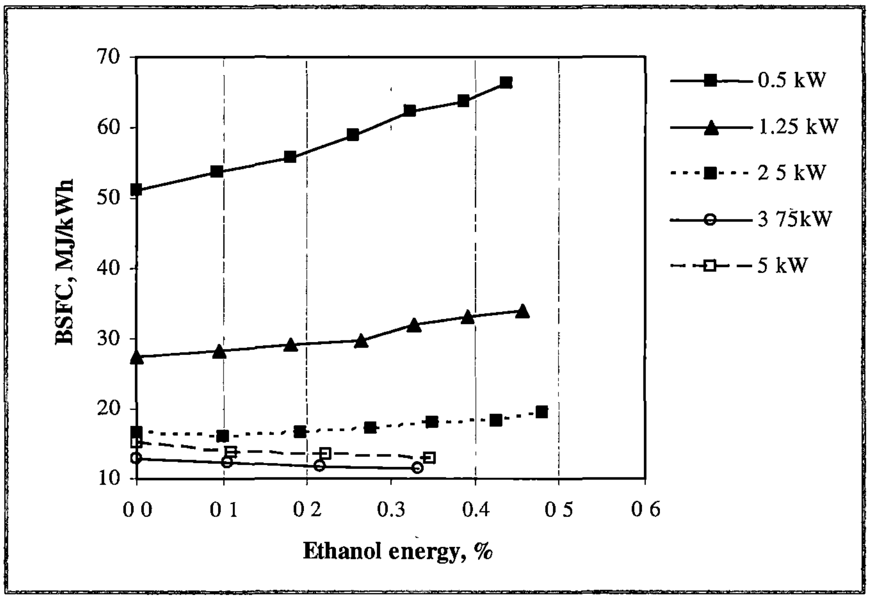


Figure 5-1: Break specific fuel consumption as a function of ethanol fueling

At low loads the dual-fuel operation gives higher energy consumption when compared with biodiesel operation (Figure 5-2). This is due to ignition extension, the cooling effect of ethanol and the nature of the combustion process in the dual-fuel engine [26; 5]. At low loads the combustion of the ethanol occurs at very lean conditions with low flame temperatures and low flame speed. As a result the flame fronts propagates at low speed and do not extend to the combustion chamber walls before the exhaust valve opening. Unburned ethanol escapes through the exhaust valve, which results in incomplete combustion and therefore a higher BSFC. The highest BSFC obtained was 66.2 MJ/kg at 0.5 kW with 45% ethanol proportions, which represents an increase of 29% when compared with biodiesel operation (51.26MJ/kg).

Figure 5-3 shows that the air intake temperature reduces with the increase of ethanol proportions for all engine operation conditions. This can be explained by the high heat of evaporation required by ethanol. The low intake air temperature reduces the combustion temperature which is unfavorable at low loads. Figure 5-4 shows the cooling effect of ethanol on the combustion temperature. It can be seen that the exhaust gas temperature increases for each load at any given proportion of ethanol. At high loads the reduced intake air temperature can be compensated by the high combustion temperature due to the rich fuel-air mixture.

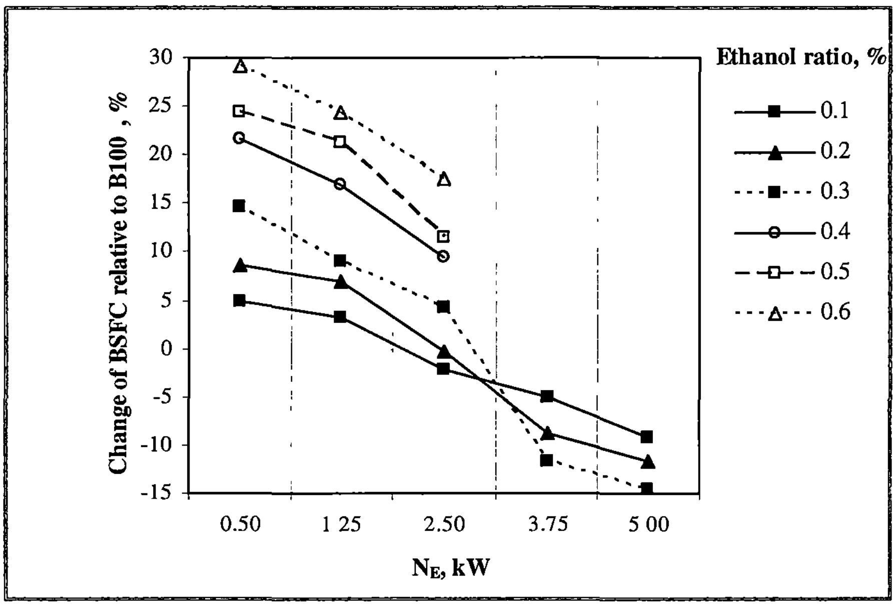


Figure 5-2: Change of break specific fuel consumption relative to biodiesel as a function of brake power output

For higher loads the efficiency is slightly better than those obtained for biodiesel operation. This can be explained by the fact that the flame propagation of the gaseous fuel-air mixture is improved due overall increase of the equivalent fuel-air ratio and the higher combustion temperature. The flame propagates with high speed from the diesel ignition center through the combustion chamber and propagates the burning of the diesel droplets. As a result the combustion process is more complete and BSFC decreases (Figure 5-2). The lowest BSFC (11.33 MJ/kg) was obtained at a load of 3.75 kW with 33% ethanol proportion, which represents a decrease of 12 % when compared with diesel operation (Figure 5-1).

Kowalewicz [48] investigated the influence of ethanol fueling on the combustion parameters of diesel engine operation. It was found that that at high loads the cylinder pressure increased and the combustion period decreased with higher ethanol a proportion which proves the hypothesis that the rapid combustion of the ethanol-air mixture promotes the combustion process of diesel droplets, which results in shorter combustion period and higher efficiency.

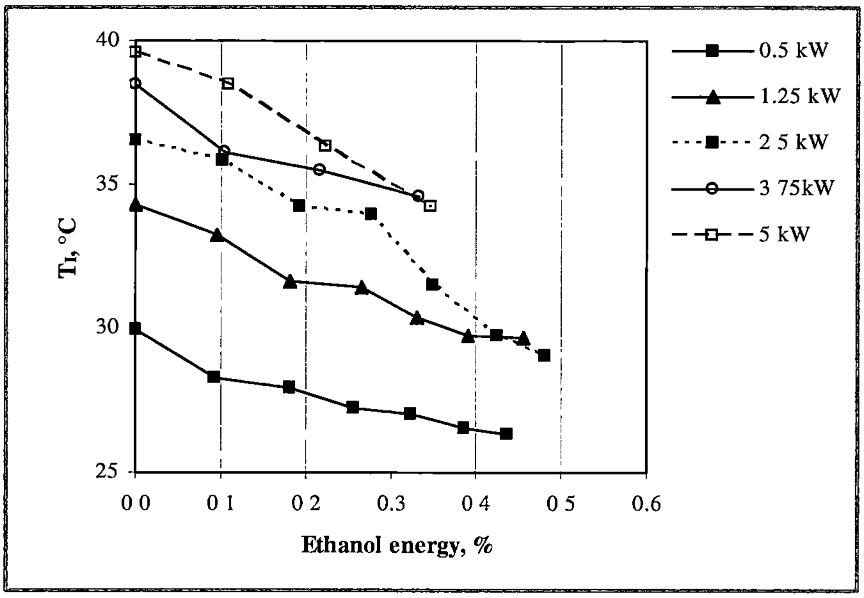


Figure 5-3: Air intake temperature as a function of ethanol fueling

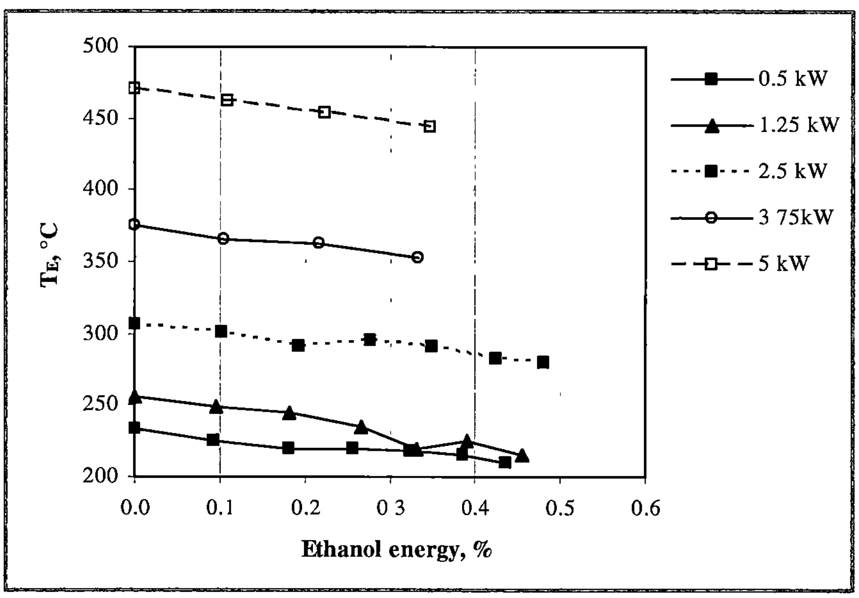


Figure 5-4: Exhaust gas temperature as a function of ethanol fueling

5.2 Carbon Monoxide

Carbon monoxide emissions are the result of incomplete combustion. Figure 5-5 shows measured CO concentrations at different load conditions for dual-fuel operation on various ethanol proportions. The change of CO emissions during dual- compared to biodiesel operation for various load conditions is represented by Figure 5-6. The following effects of dual-fuel operation on CO emissions can be seen:

- The higher the load, the higher the CO emissions (Figure 5-4).
- CO emissions drastically increase at low load conditions with increasing ethanol proportion.(Figure 5-4, 5-5).
- The effect on CO emission decreases with decreases with increasing brake power output (Figure 5-5).
- At higher loads CO emissions slightly decrease with increasing ethanol proportions.

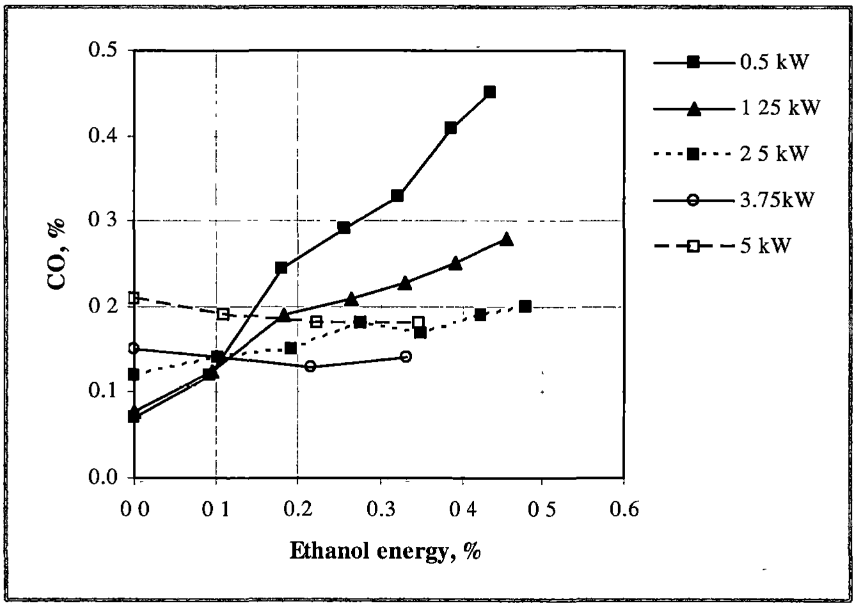


Figure 5-5: Carbon monoxide as a function of ethanol fueling

Carbon monoxide is one of the compounds formed during intermediate combustion stages. During the last stage of combustion oxidation of CO to CO_2 takes place. If the combustion is incomplete due to insufficient oxygen, low gas temperature or short residence time, CO remains in the exhaust emissions at the end of the combustion [28; 14].

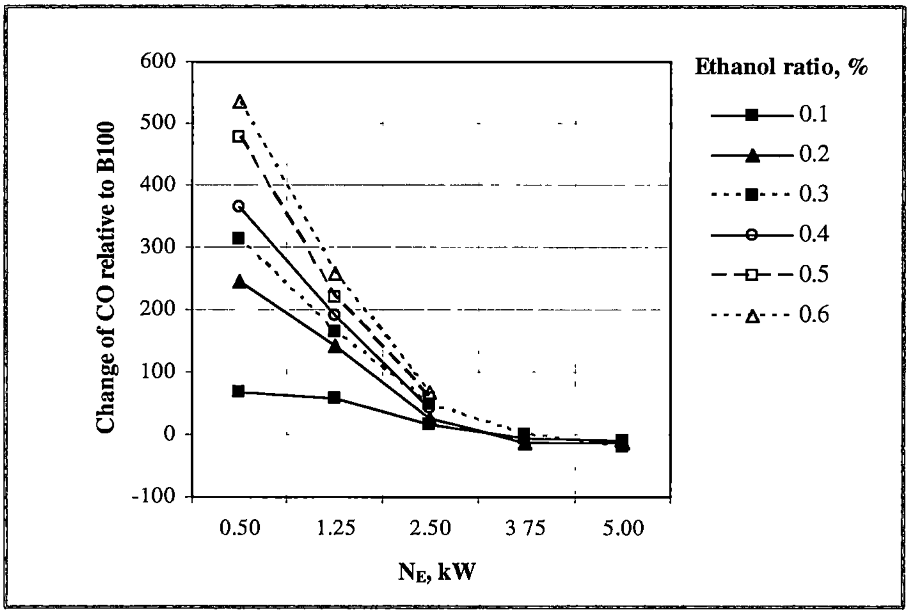


Figure 5-6: Change of carbon monoxide relative to biodiesel as a function of brake power output

In the lean flame region of the diesel vapor mixture, carbon monoxide formed immediately oxidizes to carbon dioxide due to the high oxygen concentration and the adequate gas temperatures. The oxidation of CO formed in the surrounding ethanol-air mixture is strongly depended on the gas temperature. At low load conditions the gas temperature is low and very little oxidation occurs. This effect is pronounced for increasing ethanol proportions as shown in Figure 5-5. The highest CO concentration obtained at 0.5 kW was 0.45% with 45% ethanol proportion, which represents an increase of 537% when compared with biodiesel operation (0.07%). At high loads CO emissions start slightly to decrease with increasing proportions of ethanol shown in Figure 5-6. At high loads combustion is more efficient with increasing amount of ethanol and more oxidation take place.

At high load gas temperatures are adequate for combustion and the availability of oxygen is the predominate factor. Ethanol is an oxidant and provides additional oxygen for CO oxidation resulting in a slight overall decrease of CO emissions. The highest decrease of CO emissions relative to biodiesel operation obtained was 19% at a brake power output of 5 kW and an ethanol fraction of 35% (Figure 5-6).

5.3 Hydrocarbons

Hydrocarbons emissions formed during intermediate combustion stages are a symptom of incomplete combustion caused by the present of inadequate oxygen and low gas temperatures. As a result hydrocarbons are exhausted at the end of the combustion process without being burned. Therefore the presence of hydrocarbons and carbon monoxides can be considered together and same principles as discussed in section 5.2 “Carbon monoxide” apply.

The influence of ethanol fueling on hydrocarbons emissions is shown in Figure 5-7. The change of hydrocarbon emissions relative to biodiesel fueling for the entire engine operation condition is shown in Figure 5-8. The following effect of dual-fuel operation on hydrocarbon emissions can be seen:

- Hydrocarbon concentrations are higher at high loads than at low loads (Figure 5-7).
- Emissions drastically increase at low loads with the increasing ethanol proportions. The effect decreases for moderate loads but hydrocarbon emissions still increase (Figure 5-7, 5-8).
- At high loads hydrocarbons emissions slightly decrease with increasing ethanol fueling (Figure 5-8).

At low loads gas temperatures are inadequate for complete combustion, which results in high concentration of hydrocarbon in the exhaust gases and low efficiency or high BSFC. This effect is more pronounced with increasing ethanol fueling as the flame regions of the gaseous fuel-air mixturc is vcry lean and gas temperatures to low (Figure 5-7). The highest concentration obtained at 0.5 kW was 53.3 ppm for ethanol fueling of 45 %, which represents an increase of 123 % when compared with biodiesel operation.

With increasing load, gas temperatures and the overall fuel-air ratio increase, resulting in better combustion and lower hydrocarbon emissions (Figure 5-8). At high loads with increasing ethanol fueling HC emissions slightly decrease when compared with biodiesel. Additionally oxygen molecules of the oxygen compound provide additional oxygen for a more efficient combustion. This effect is more pronounced during high load operation as oxygen is a predominant factor for insufficient combustion. The highest decrease of HC emissions relative to biodiesel operation (37.5 ppm) obtained was 18 % at a brake power output of 5 kW and an ethanol fueling of 35 % (Figure 5-8).

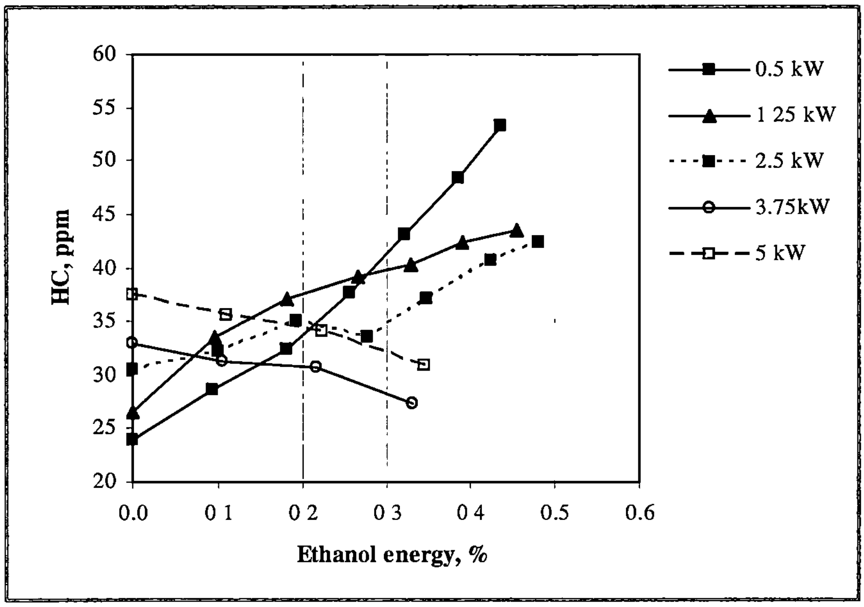


Figure 5-7: Hydrocarbons as a function of ethanol fueling

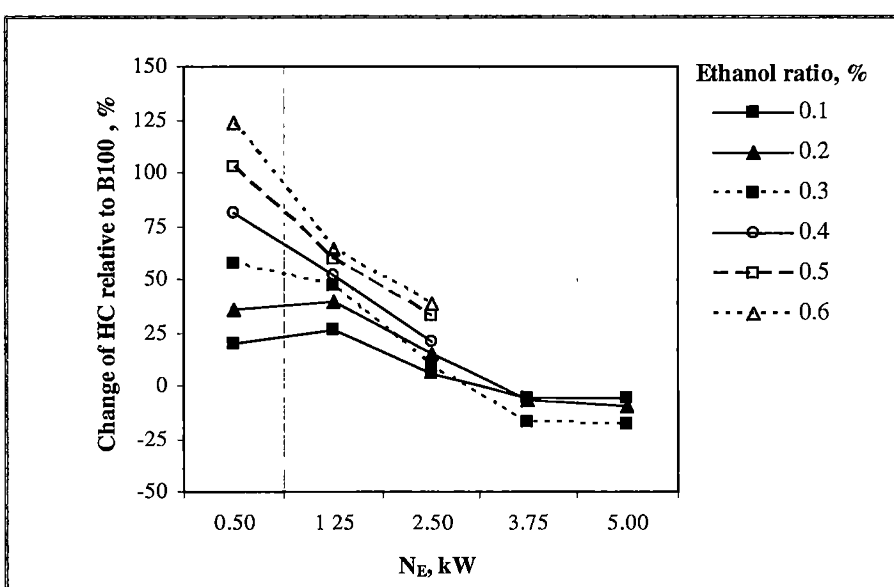


Figure 5-8: Change of hydrocarbons relative to biodiesel as a function of brake power output

5.4 Opacity

Opacity is a measure for the level of particulate matter in the exhaust gases. Particulate matters are fine particles of solids or liquids suspended and the result of incomplete combustion. Figure 5-9 shows the level of opacity at different load conditions as a function of ethanol fueling. The change opacity when compared to biodiesel operation for the entire operation conditions are shown in Figure 5-10. The following behavior for the level of opacity during dual-fuel operation can be observed:

- Opacity increases with increasing load conditions (Figure 5-9)
- The opacity tends to slightly decrease at low load conditions but overall dual fuel operation does not significantly affect the opacity at low load conditions (Figure 5-9, 5-10).
- At high loads opacity significantly decrease during dual-fuel operation

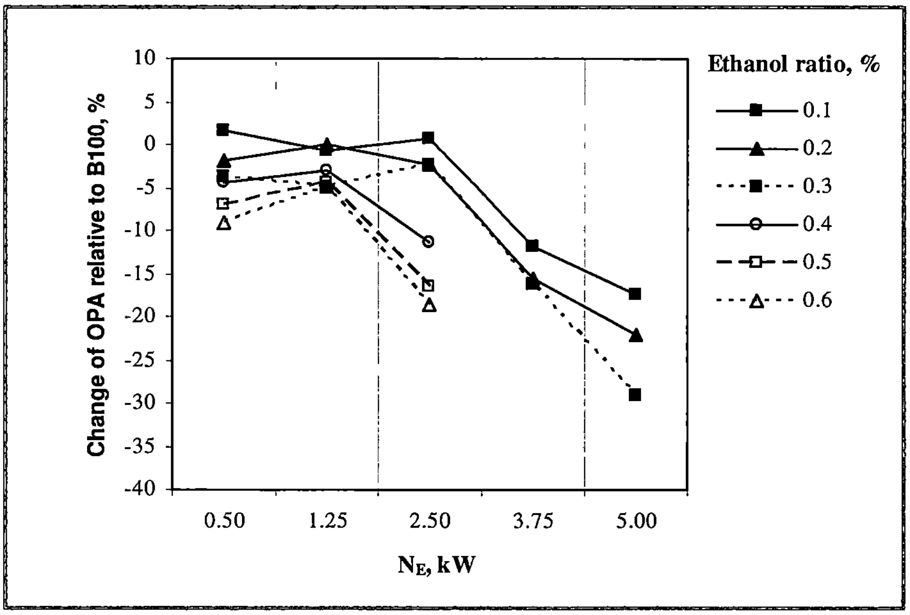


Figure 5-9: Opacity as a function of ethanol fueling

At high loads oxygen high gas temperature and a rich fuel-air mixture are present. Due to the rich the mixture, insufficient oxygen is present for combustion in some regions of the combustion chamber. As a result particulate matter remains in the exhaust gases. Burning ethanol produces no smoke and consequently increasing the proportion of ethanol in dual-fuel operation reduces opacity. Particulate matter is primarily produced during the combustion of diesel fuels [28; 49]. The highest decrease of opacity when compared with biodiesel operation (12.4 %) obtained was 29 % at a brake power output of 5 kW, which represents an ethanol fueling of 33%. At low load conditions gas temperature and fuel-air ratio is inadequate for complete combustion. The fact that no particulate matter is produced during the combustion of ethanol compensates for the higher level of produced particulate matter and results in a nearly constant level of opacity during dual-fuel operation.

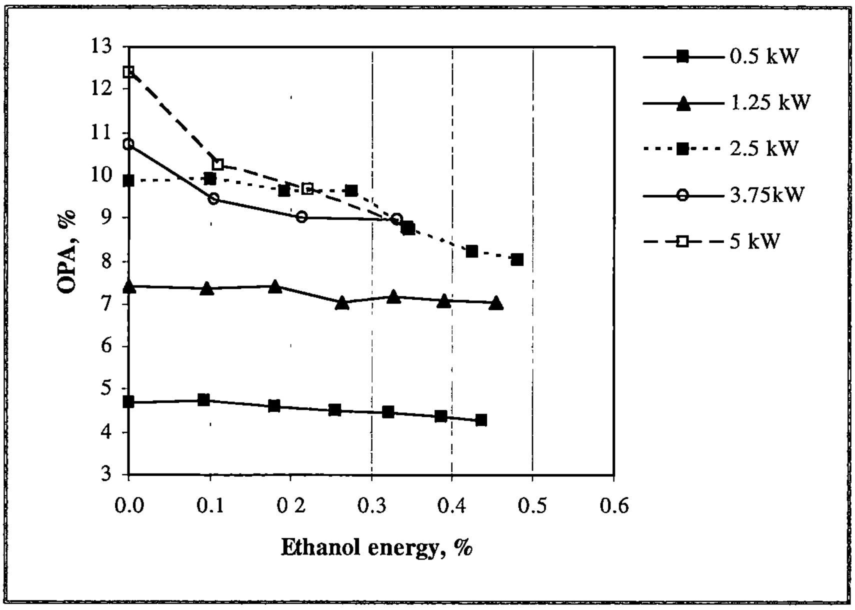


Figure 5-10: Change of opacity relative to biodiesel as a function of brake power output

5.5 Carbon Dioxide

Carbon monoxide together with Water (H_2O) is a primary combustion product of hydrocarbon fuel. Figure 5-11 shows measured CO_2 concentrations at different load conditions for dual-fuel operation on various ethanol proportions. The change of CO_2 emissions during dual-fuel operation compared to biodiesel operation for various load conditions is represented by Figure 5-12. The following effects of dual fuel-operation on CO_2 emissions can be seen:

- CO_2 increases with increasing brake power output (Figure 5-11).
- CO_2 emissions decrease for the entire operation conditions with increasing ethanol fueling (Figure 5-11, 5-12).
- At high loads a higher CO_2 emissions decrease more drastically with increasing ethanol fueling (Figure 5-12).

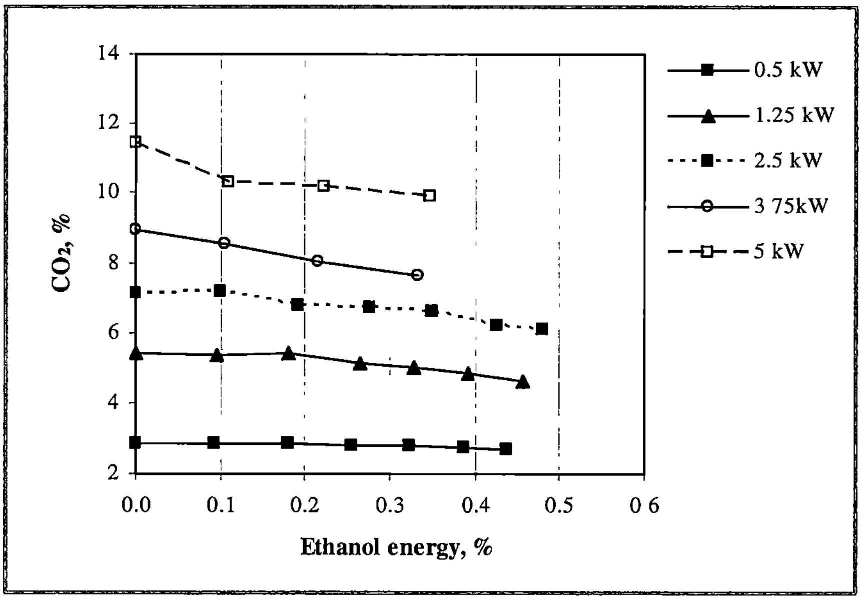


Figure 5-11: Carbon dioxide as a function of ethanol fueling

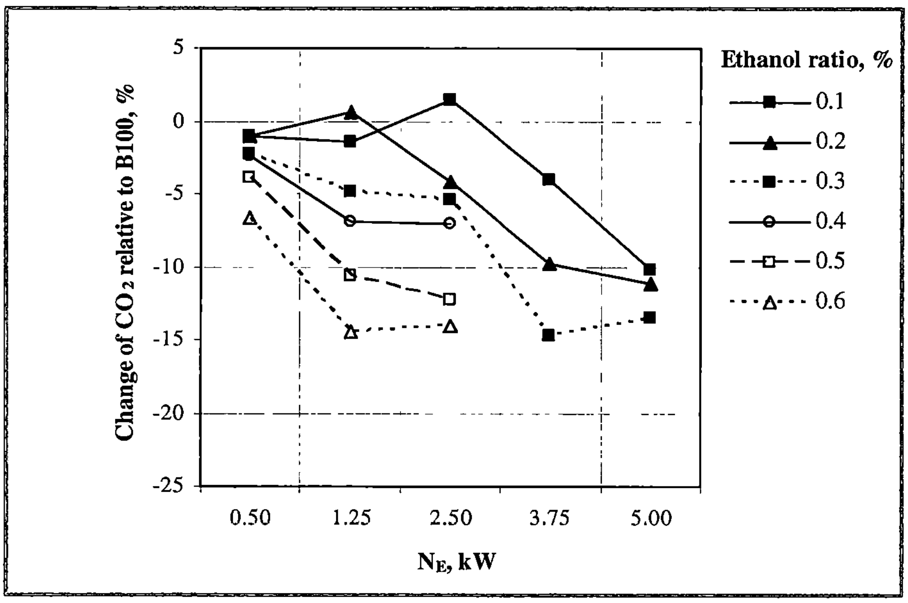


Figure 5-12: Change of carbon dioxide relative to biodiesel as a function of brake power output

At higher loads more fuel is burned resulting in higher carbon dioxide emissions. The decrease of carbon dioxide emissions with increasing proportions of ethanol during dual-fuel operation is a result of the presence of a higher amount of ethanol. Products of ethanol combustion contain less CO_2 and more H_2O. The effect is more pronounced during high load conditions as result more complete combustion and higher efficiency. The highest decrease of CO_2 emissions when compared with biodiesel (CO_2 8,9 %) operation obtained at 3.75 kW was 15 % with, which represents an ethanol fueling of 35 % (Figure 5-12).

5.6 Nitrogen oxides

The effect of dual-fueling on nitrogen oxides is illustrated in Figure 5-13. Figure 5-14 shows the change of NO_x concentrations relative to biodiesel operation as a function of the brake output power. It can be seen that NO_x concentration are:

- Higher at high loads than at low loads for any proportion of ethanol
- strongly decreased at low loads for an proportion of ethanol
- slightly increased at high loads for any proportion of ethanol

Formation of NOx under lean conditions predominately occurs through the Zeldovich mechanism [28] and is direct function of the availability of oxygen and increases exponential with temperature. At high loads more fuel is burned resulting in gas higher temperatures and therefore higher NO_x emissions.

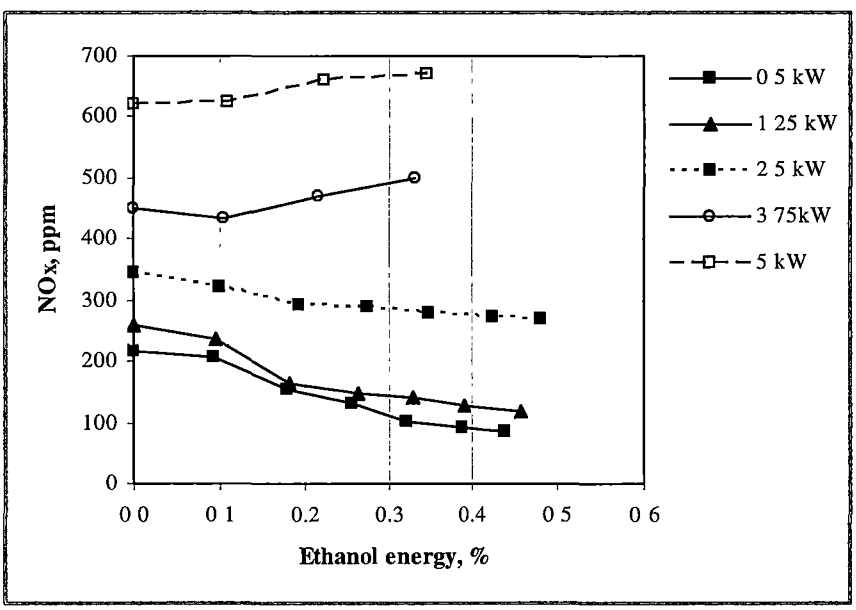


Figure 5-13: Nitrogen oxides as a function of ethanol fueling

At low load the result is clear, combustion temperature decrease at dual-fuel operation due to the cooling effect of ethanol (Figure 5-4, discussed in section 5.1 "Brake Specific Fuel Consumption") and results in a decrease of NO_x concentration (Figure 5-13). The lowest NO_x concentration obtained at 0.5 kW was 85.3 %, which represents a decrease of 60 % when compared with biodiesel operation (Figure 5-13, 5-14).

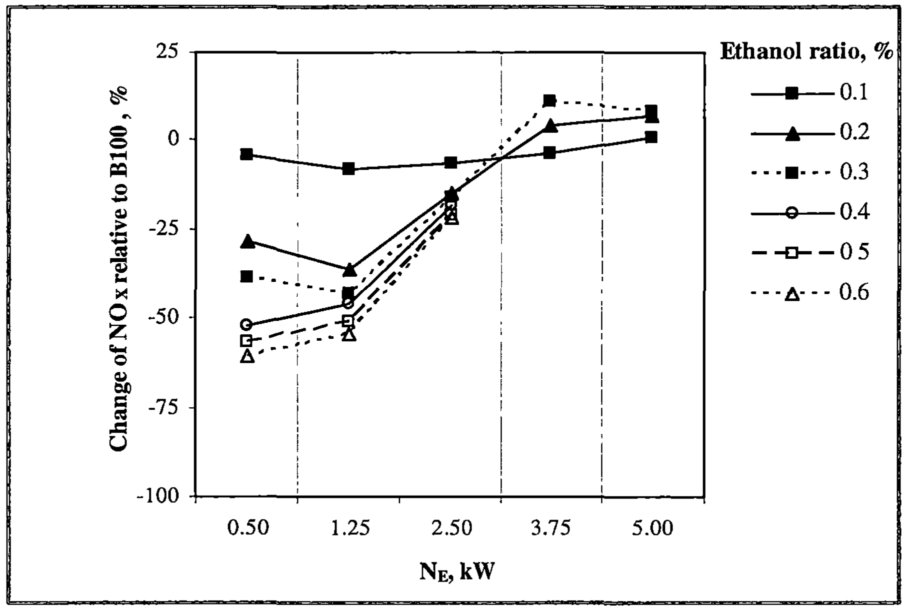


Figure 5-14: Change of Nitrogen oxides relative to biodiesel as a function of brake power output

At high load the flame front of gaseous ethanol-air mixture propagates at high speed and promotes thee combustion of the diesel droplets resulting in shorter combustion period and higher efficiency. The high gas temperature compensates the cooling effect of ethanol and NO_x concentrations slightly increase. The highest concentration of NO_x obtained at 5kW was 670 ppm with 33 % ethanol fueling, which represents an increase of 8% when compared with biodiesel operation.

5.7 Concluding remarks

The results of the experimental investigation presented in this chapter were obtained with biodiesel and dual-fuel operation with various proportions of ethanol at various loads and constant speed. The test engine run smoothly and satisfactory with up to 40% ethanol fueling at high loads whereas at low loads the knock limit was never reached. The limitation of ethanol fueling was due to the occurrence of diesel knock as discussed in more detail in chapter 2 "Internal Combustion Engine".

The effect of ethanol fueling on the engine performance can be summarized as follows:

- At high loads dual-fueling with biodiesel and ethanol results in a significant decrease in hydrocarbon emissions at similar or slightly improved efficiency compared with biodiesel operation.
- At low loads dual-fueling with increasing proportions of ethanol fueling leads to a drastic increase of incomplete combustion products in the exhaust gases at low efficiency. Carbon dioxide and smoke emissions still slightly decrease due to the chemical composition of ethanol.
- Injection of ethanol into the air inlet improves NO_x emissions in all operating conditions. This tendency is more pronounced at low to moderate load conditions.

The results are consistent with current studies on dual-fuel engines, which all report an increase of incomplete combustion products and decrease of efficiency compared with pure diesel operation. Exhaust emissions and regardless of the type of gaseous fuel injected into the cylinder [50; 29; 18; 19; 49].

Chapter 6

Final conclusions, recommendations and proposed future work

Dual-Fuel engines display a complex combination of operation and combustion process features of both spark compression ignition engines and spark ignition engines. In this chapter operation, combustion process and emitted exhaust gas emissions of internal combustion engines were discussed to provide the framework to understand the operation of dual-fuel engines.

From an extensive literature survey, it has been shown that dual-fuel engines from compression ignition type represent an attractive option to utilize various gaseous fuel resources and minimize exhaust gas emissions while engine efficiency can be maintained at a similar level to those obtained with straight diesel fueling. At part load conditions dual-fuel engines suffer from low efficiency due to the lean gaseous fuel-air mixture. However, a sufficient operation of a dual-fuel engine technology still requires more complete analysis of various performance and operating parameters.

The conversion concept of a diesel generator to operate on dual-fuel mode was discussed. The intricate development involved minor engine modifications comprising the design and installation of fuel delivery system, injection control sensor system and air intake. Integral part of the system design of the injection control system using LabView programming. A system test evaluated that the system operated according to given specifications. This is a reassurance on the reliability of the equipment used and the data gathered.

Having built an experimental test rig, a comprehensive experimental investigation was carried out covering a range of engine operating conditions. The performance analysis on a diesel generator operating on dual-fuel mode was reported. Experimental techniques, instrumentation and data acquisition were selected to suit measurements of engine load, operation temperatures and fuel consumption. Ethanol at various loads was injected into the diesel engine to study the effect of ethanol on dual-fuel operation in detail. The results of the engine performance analysis as discussed in the following chapter are based on the measured operation data including brake power output, fuel consumption, the level of exhaust emissions of carbon monoxide, hydrocarbons, carbon dioxide, nitrogen oxides and the engine operation temperatures of the air intake and the exhaust.

The results of the experimental investigation were obtained with biodiesel and dual-fuel operation with various proportions of ethanol at various loads and constant speed. The test engine run smoothly and satisfactory with up to 40% ethanol fueling at high loads whereas at low loads the knock limit was never reached. The limitation of ethanol fueling was due to the occurrence of diesel knock as discussed in more detail in chapter 2 "Internal Combustion Engine".

At high loads dual-fueling with biodiesel and ethanol results in a significant decrease in hydrocarbon emissions at similar or slightly improved efficiency compared with biodiesel operation.

- At a load 5 kW CO concentrations decreased 19% and HC concentrations decreases 18% with 33% ethanol fueling
- BSCF increased 12% at a load of 3.75kW with 35% proportions of ethanol

At low loads dual-fueling with increasing proportions of ethanol fueling leads to a drastic increase of incomplete combustion products in the exhaust gases at low efficiency.

- At a load of 0.5 kW CO concentrations increased 537% and HC concentrations 123% with 45% ethanol fueling
- BSFC decreased 29% at a load of 0.5 kW with 45% ethanol fueling

Injection of ethanol into the air inlet improves CO_2, NO_x and smoke emissions in all operating conditions. This tendency for NO_x is more pronounced at low to moderate load conditions.

- The highest decrease of NO_x emissions obtained at 0.5kW was 60% with 45% ethanol fueling
- CO_2 decrease 15% at a load of 3.75 kW with 35% ethanol fueling
- At a load 5 kW smoke emissions decrease 29% with 35% ethanol fueling

The pure load operation performance is a result the very lean gaseous ethanol-air mixture with low flame temperatures and low flame speed. At high load the rapid combustion of the ethanol-air mixture promotes the combustion process of diesel droplets, which results in shorter combustion period and higher efficiency.

From a future work point of view, the Performance of dual-fuel engines is strongly dependent on the engine operation conditions which require the need for the development of flexible and effective controls. These controls will allow optimum performance of dual-fuel engines over a wide range operating conditions. This will enable the optimum use of ethanol in relation to biodiesel. Therefore, to optimize the engine performance for minimum emissions, lower brake specific fuel consumption and power, an advanced controller to control on-line engine input parameters may be developed. Finally, the wear effect of dual-fuel operations on engine component durability can also be studied as an extension to this project.

Bibliography

1 *Biodiesel, the new fuel from brazil*, 2005, viewed 25. September 2006, <http://www.biodiesel.gov.br/docs/cartilha_ingles.pdf> >.

2 Crosthwait, R 2007, 'Performance appraisal of a diesel generator powered by biodiesel', Master thesis, UTAS.

3 Lynch, D, J. 2006, *Brazil hopes to build on its ethanol success*, USA Today, viewed 26. September 2006, <http://www.usatoday.com/money/world/2006-03-28-brazil-ethanol-cover_x.htm>.

4 Lake, A 2006, *BAA response to biofuel task force*, viewed 26. September 2006, <http://www.biodiesel.org.au/Taskforce2005/BAA_Biofuels_taskforce_submission.pdf>.

5 Liu, Z & Karim, GA 1997, 'Simulation of combustion process in gas-fuelled diesel engines', *Proc. Instn Mech. Engrs, Part A: J. Power and Energy*, vol. 211, pp. 159-169.

6 Nakra, CP 1990, *Basic Automobile Engineering*, 5 edn, Dhanpat Rai Publishing Company Ltd.

7 Benson, S 1982, *The termodynamics and gas dynamics of internal combustion engines*, vol. 1, Clarendon Press.

8 Amsoil 2001, *Two-cycle Engine Applications and Lubrication Needs*, viewed 5.August 2007, <http://www.amsoil.com/articlespr/article_2cycleapplications.aspx>.

9 Wikipedia 2007, *Reciprocating engine*, viewed 1. September 2007, <http://en.wikipedia.org/wiki/Reciprocating_engine>.

10 NNDB 2007, *Rudolf Diesel*, viewed 1. September 2007, <http://www.nndb.com>

11 Taylor, CF 1985, *The Internal Combustion Engine in Theory and Practice*, vol. 2, The M.I.T. Press.

12 Bauer, H, Dietsche, KH & Jaeger, T 2004, *Diesel engine management*, 3 edn, Robert Bosch GmbH.

13 Diracdelta 2006, *Science & Engineering Encyclopedia*, viewed 6.July 2007, <http://www.diracdelta.co.uk/science/source/h/o/home/source.html>.

14 Heywood, JB 1988, *Internal combustion engines Fundamentals*, McGraw-Hill Science Engineering.

15 Karim, GA 1983, 'The dual fuel engine of the compression ignition type-prospects, problems and solutions- a review', *SAE Paper No. 831073.*

16 Karim, GA & Klat, SR 1966, 'The knock and autoignition characteristics of some gaseous fuels and their mixtures', *Journal of the Institution of Fuel*, vol. 39, pp. 109-119.

17 Abu-Qudais, M, Haddad, O & Qudaisat, M 2000, 'The effect of alcohol fumigation on diesel engine performance and emissions', *Energy Conversion and Management*, vol. 41, no. 4, pp. 389-399.

18 Houser, KR, Lestz, SS, Dukovich, M & Yasbin, RE 1980, 'Methanol fumigation of a light duty automotive diesel engine', *SAE Paper No. 801379.*

19 Ilango, K 1983, 'Dual-fueling of diesel engines', *Journal of the Institution of Engineers (India): Mechanical Engineering Division*, vol. 64, pp. 40-44.

20 Pirouzpanah, V & Saray, RK 2007, 'Enhancement of the combustion process in dual-fuel engines at part loads using exhaust gas recirculation', *Proceedings of the Institution of Mechanical Engineers -- Part D -- Journal of Automobile Engineering*, vol. 221, no. 7, pp. 877-888.

21 Stewart, J, Clarke, A & Chen, R 2007, 'An experimental study of the dual-fuel performance of a small compression ignition diesel engine operating with three gaseous fuels', *Proceedings of the Institution of Mechanical Engineers -- Part D -- Journal of Automobile Engineering*, vol. 221, no. 8, pp. 943-956.

22 Karim, GA 1980, 'A review of combustion processes in the dual fuel engine-the gas diesel engine', *Progress in Energy and Combustion Science*, vol. 6, pp. 277-285.

23 Abdalla, GH, Soliman, HA & Badr, OA 2001, 'Combustion quasi-two-zone predictive model for dual-fuel engines', *Energy Conversion and Management*, vol. 42, pp. 1477-1498.

24 Karim, GA & Yan, ZD 1991, 'Modeling of autoignition and knock in a compression ignition engine of the dual fuel type', *IMechE Conference paper*, pp. 141-148.

25 Liu, Z & Karim, GA 1995, 'A predictive Model for the Combustion Process in Dual Fuel Engines', *SAE Paper No.952435*.

26 Karim, GA & Jones, W 1989, 'An examination of the ignition delay period in a dual-fuel engine', *SAE Paper 892140*.

27 Kowalewicz, A 2007, 'Performance, emissions and combustion parameters of eco-diesel fuelled with rapeseed oil methyl ester and ethanol.' *Combustion Science and Technology*, vol. 179, no. 11, pp. 2415 - 2435.

28 DieselNet 2006, *Emission formation in diesel engines*, viewed 18. July 2007, <http://www.dieselnet.com/tech/diesel_emiform.html>.

29 Galal, MG, Aal, MMA & Kady, MAE 2002, 'A comparitive study between diesel and dual-fuel engines: performance and emissions', *Combustion Science and Technology*, vol. 174, no. 11, pp. 241 - 256.

30 Houghton, J 2001, *What is global warming*, viewed 18. January 2008, <http://www.st-edmunds.cam.ac.uk/CiS/houghton/>.

31 Xing-cai, L, Jian-guang, Y, Wu-gao, Z & Zhen, H 2004, 'Effect of cetane number improver on heat release rate and emissions of high speed diesel engine fueled with ethanol-diesel blend fuel', *Fuel*, vol. 83, no. 14-15, pp. 2013-2020.

32 Patterson, G 1973, *Engine emissions pollutant formation and measurement*, Plenum Press New York, New York.

33 Worldbank 1998, *Sulfur Oxides*, viewed 15. January 2008, <http://lnweb18.worldbank.org/essd/envext.nsf/51ByDocName/SulfurOxides/$FILE/HandbookSulfurOxides.pdf>.

34 ARB 2006, *Health effects of diesel exhaust particulate matter*, viewed 23. January 2008, <http://www.arb.ca.gov/research/diesel/dpm_draft_3-01-06.pdf>.

35 Hansen, AC, Zhang, Q & Lyne, PWL 2005, 'Ethanol–diesel fuel blends—a review', *Bioresource Technology*, vol. 96, no. 3, pp. 277-285.

36 Ribeiro, NM, Pinto, AC & Quintella, CM 2005, 'The Role of Additives for Diesel and Diesel Blended (Ethanol or Biodiesel) Fuels: A Review', *Proceed-*

ings of the Institution of Mechanical Engineers: Part D : Journal of automobile engineering. London, vol. 219, no. D5, pp. 715-723
37 Bosch 2007, *Product fact sheet*.
38 Martins, JJG & Finlay, IC 1992, 'Fuel preparation in port-injected engines', paper presented to SAE Technical Paper Series.
39 Makartchouk, A 2002, *Diesel engine engineering*, Marcel Dekker, INC., New York.
40 Potter, D 2007, *Measuring temperature with thermocouples – a Tutorial*, viewed 29.August 2007, <http://www.noise.physx.u-szeged.hu/DigitalMeasurements/Sensors/Thermocouples.pdf>.
41 Figliola, RS & Beasley, DE 2000, *Theory and design for mechanical measurements*, 3rd edn, Leghigh Press INC.
42 Technology, P 2007, *USB TC-08 Thermocouple Data Logger*, viewed 26.August 2007, <http://www.picotech.com/thermocouple.html>.
43 Kuo, KK 1990, *Principles of combustion*, 2nd edn, John Wiley & Sons Inc., New Jersey.
44 Sensors, I 2007, *LCS*, viewed 27.August 2007, <http://www.sensors-inc.com/pdfs/LCS.pdf>.
45 Frith, M 2007, Ethanol & Biodiesel Discussion, 5. March.
46 ISO8178-4. 1996,*Reciprocation internal combustion engines - exhaust emission measurement, in part 4: test cycle for different engine applications*, British Standard,.
47 Movahednejad, E, Hosseinalipour, M, Ommi, F & Samimi, O 2006, 'Experimental and theoretical study of injection timing on performance and exhaust emissions in a port-injected gasoline engine', *Spring Technical Conference of ASME Internal Combustion Engine Division*, vol. 1405, no. 1, pp. 137-145.
48 Kowalewicz, A 2006, 'Eco-diesel engine fuelled with rapeseed oil methyl ester and ethanol. Part 3: combustion processes ', *Proceedings of the Institution of Mechanical Engineers: Part D : Journal of automobile engineering. London*, vol. 220, no. D9, pp. 1283-1291.
49 Kowalewicz, A 2005, 'Eco-diesel engine fuelled with rapeseed oil methyl ester and ethanol. Part 1: Efficiency and emission ', *Proceedings of the Institution of Mechanical Engineers: Part D : Journal of automobile engineering. London*, vol. 219, no. D5, pp. 715-723
50 Barata, J, M. 1995, 'Performance and emissions of a dual fueled Di diesel engine', paper presented to SAE Technical Paper Series.

Appendix

Test Results

Opacity [%]

Ethanol [%]	**0.5 kW**	**1.25 kW**	**2.5 kW**	**3.75 kW**	**5 kW**
0	4.67	7.41	9.84	10.69	12.38
0.1	4.75	7.36	9.91	9.44	10.22
0.2	4.58	7.41	9.62	9.02	9.65
0.3	4.50	7.05	9.61	8.94	8.79
0.4	4.47	7.19	8.72		
0.5	4.35	7.08	8.21		
0.6	4.25	7.05	8.01		

CO [%]

Ethanol [%]	**0.5 kW**	**1.25 kW**	**2.5 kW**	**3.75 kW**	**5 kW**
0	0.07	0.08	0.12	0.15	0.21
0.1	0.12	0.12	0.14	0.14	0.19
0.2	0.24	0.19	0.15	0.13	0.18
0.3	0.29	0.21	0.18	0.15	0.17
0.4	0.33	0.23	0.17		
0.5	0.41	0.25	0.19		
0.6	0.45	0.28	0.20		

HC [ppm]

Ethanol [%]	**0.5 kW**	**1.25 kW**	**2.5 kW**	**3.75 kW**	**5 kW**
0	23.89	26.56	30.57	33.00	37.53
0.1	28.69	33.49	32.30	31.18	35.50
0.2	32.45	37.10	35.10	30.80	34.00
0.3	37.66	39.13	33.50	27.33	30.95
0.4	43.15	40.23	37.00		
0.5	48.39	42.27	40.75		
0.6	53.30	43.50	42.30		

CO_2 [%]

Ethanol [%]	**0.5 kW**	**1.25 kW**	**2.5 kW**	**3.75 kW**	**5 kW**
0	2.86	5.40	7.10	8.92	11.46
0.1	2.83	5.33	7.20	8.57	10.30
0.2	2.83	5.44	6.80	8.05	10.18
0.3	2.79	5.14	6.72	7.61	9.91
0.4	2.79	5.02	6.60		
0.5	2.75	4.82	6.23		
0.6	2.67	4.62	6.10		

NO_x [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	215.89	258.62	343.47	451.72	619.82
0.1	207.16	237.52	321.00	433.94	623.40
0.2	154.76	163.84	292.00	469.69	662.10
0.3	132.84	147.66	288.00	500.00	669.54
0.4	103.05	139.91	279.62		
0.5	93.27	126.59	271.48		
0.6	85.30	118.50	268.23		

BSFC [MJ/kg]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	51.26	27.34	16.47	12.83	15.05
0.1	53.82	28.24	16.11	12.18	13.66
0.2	55.74	29.24	16.44	11.69	13.29
0.3	58.78	29.79	17.18	11.33	12.86
0.4	62.34	31.95	18.03		
0.5	63.77	33.17	18.38		
0.6	66.22	34.03	19.36		

Change of opacity relative to biodiesel operation [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	0	0	0	0	0
0.1	2	-1	1	-12	-17
0.2	-2	0	-2	-16	-22
0.3	-4	-5	-2	-16	-29
0.4	-4	-3	-11		
0.5	-7	-4	-17		
0.6	-9	-5	-19		

Change of CO relative to biodiesel operation [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	0	0	0	0	0
0.1	68	59	17	-7	-10
0.2	245	142	25	-13	-14
0.3	312	165	50	0	-19
0.4	366	191	42		
0.5	478	220	58		
0.6	537	257	67		

Change of HC relative to biodiesel operation [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	0	0	0	0	0
0.1	20	26	6	-6	-5
0.2	36	40	15	-7	-9
0.3	58	47	10	-17	-18
0.4	81	51	21		
0.5	103	59	33		
0.6	123	64	38		

Change of CO_2 relative to biodiesel operation [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	0	0	0	0	0
0.1	-1	-1	1	-4	-10
0.2	-1	1	-4	-10	-11
0.3	-2	-5	-5	-15	-14
0.4	-2	-7	-7		
0.5	-4	-11	-12		
0.6	-7	-14	-14		

Change of NOx relative to biodiesel operation [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	0	0	0	0	0
0.1	-4	-8	-7	-4	1
0.2	-28	-37	-15	4	7
0.3	-38	-43	-16	11	8
0.4	-52	-46	-19		
0.5	-57	-51	-21		
0.6	-60	-54	-22		

Change of BSFC relative to biodiesel operation [%]

Ethanol [%]	**0.5**	**1.25**	**2.5**	**3.75**	**5**
0	0	0	0	0	0
0.1	5	3	-2	-5	-9
0.2	9	7	0	-9	-12
0.3	15	9	4	-12	-15
0.4	22	17	9		
0.5	24	21	12		
0.6	29	24	18		

Fuelconsumption Biodiesel [L/hr]

Ethanol [%]	**0.50**	**1.25**	**2.50**	**3.75**	**5.00**
0.00	0.72	0.96	1.16	1.36	2.12
0.10	0.70	0.91	1.03	1.16	1.71
0.20	0.65	0.87	0.97	0.99	1.47
0.30	0.63	0.80	0.91	0.82	1.20
0.40	0.61	0.78	0.87		
0.50	0.57	0.75	0.78		
0.60	0.55	0.69	0.75		

Fuel consumption Ethanol [L/hr]

Ethanol [%]	**0.50**	**1.25**	**2.50**	**3.75**	**5.00**
0.00	0	0	0	0	0
0.10	0.12	0.17	0.20	0.24	0.37
0.20	0.25	0.33	0.40	0.47	0.73
0.30	0.37	0.50	0.60	0.71	1.10
0.40	0.50	0.67	0.81		
0.50	0.62	0.83	1.01		
0.60	0.75	1.00	1.21		

Exhaust gas temperature [°C]

Ethanol [%]	**0.50**	**1.25**	**2.50**	**3.75**	**5.00**
0.00	233.20	255.96	305.92	375.44	471.15
0.10	225.69	249.31	300.69	365.03	462.30
0.20	220.01	244.76	291.72	362.75	454.28
0.30	219.07	234.93	295.76	352.15	443.78
0.40	218.69	220.12	291.21		
0.50	214.83	225.66	282.61		
0.60	210.20	215.10	280.10		

Air intake temperature [°C]

Ethanol [%]	**0.50**	**1.25**	**2.50**	**3.75**	**5.00**
0.00	29.95	34.31	36.50	38.46	39.60
0.10	28.30	33.20	35.80	36.11	38.47
0.20	27.96	31.61	34.20	35.46	36.30
0.30	27.24	31.44	33.92	34.57	34.19
0.40	27.00	30.39	31.50		
0.50	26.54	29.77	29.74		
0.60	26.35	29.64	29.02		