A COMPARATIVE STUDY OF THE COMBUSTION CHARACTERISTICS OF A COMPRESSION IGNITION ENGINE FUELLED ON DIESEL AND DIMETHYL ETHER

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Declaration

I, Paulo Miguel Pereira Lopes, hereby declare that the content of this report is of my own work unless otherwise stated. This report is being submitted for the degree of Master of Science in the University of the Witwatersrand. This report has not been submitted before for any degree or examination in any other university.

PMP Lopes

Date

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I would like to thank Dr D Cipolat for his time and guidance during the course of the project, as well as his patience and dedication to the field of alternative fuel research. I would also like to thank all the technicians in the workshop, who were always more than willing to help with any problem I encountered.

Abstract

This research is an investigation into the performance and combustion characteristics of a two-cylinder, four-stroke compression ignition engine fuelled on diesel and then on dimethyl ether (DME). Baseline tests were performed using diesel. The tests were then repeated for dimethyl ether fuelling. All DME tests were performed at an injection opening pressure of 210 bar, as recommended for diesel fuelling. The tests were all carried out at constant torque with incremental increases in speed and an improved method of measuring the DME flow rate was devised. It was found that the engine's performance characteristics were very similar, regardless of whether the engine was fuelled on diesel or DME. Brake power, indicated power and cylinder pressure, during the highest loading condition of 55 Nm, were virtually identical for diesel and DME fuelling, with the most significant finding being that the engine was more efficient when fuelled on DME than when fuelled with diesel. Another interesting finding was that the energy release of diesel decreases with increasing load, whilst the energy release of DME increases with increasing load. At the highest loading condition of 55 Nm, the energy release of DME was approximately 210 joules higher than that of diesel. This investigation concluded that DME may definitely be a suitable substitute fuel for diesel.

Contents

List of Figures

List of Tables

1. Introduction

Human society in its modern form is heavily dependent on the provision, conversion and use of non-biological forms of energy and it is reasonable to assert that current world population levels would not be sustainable without this output. Energy is thus one of the most critical of the world's resources.

Biological systems, such as fuel cells, operate by directly converting the chemical energy of a fuel to work, while non-biological systems convert chemical energy firstly to raise the temperature of the working substance to a high level before generating work output. It is likely that market introduction of biological systems may take a very long time before they have high enough work outputs and compact sizes. Combustion engines are likely to dominate for a long time, thus the development of fuels to produce high-temperature working substances by combustion should therefore be carefully studied [1].

For a number of years, emissions from transportation engines have been studied widely because of the contribution of such engines to atmospheric pollution [2]. The single greatest contributor of air pollution is the diesel or compression ignition engine, which dates back to initial conception by Rudolf Diesel in 1892 [3]. Since that time, these engines have continued to develop due to our progression in knowledge in the field of engine development, but for the last twenty years, engine development has focused on developing cost effective and cleaner fuels.

Due to limited fossil fuel reserves, and the detrimental effects that diesel fumes have on the earth's atmosphere, the development of renewable, alternate fuels that can be manufactured from non-fossil sources would inevitably have to be introduced. Interest in this field has created market awareness in fuels such as methanol, ethanol and dimethyl ether (DME), the latter being the fuel of concern in this report.

The overall aim of this research report is to study the performance and combustion characteristics of an engine fuelled on an alternative fuel, with as little modification to the engine as possible. Studies on DME fuelled compression ignition engines, inside and outside the University, have shown that the characteristics of DME make it a favourable alternate fuel to diesel.

2. Objectives

The objectives of this project were to:

1) improve the fuel delivery and measuring systems for DME fuelling;
2) compare the performance of the engine fuelled on DME and diesel during constant torque and increasing speed testing;
3) compare the energy release of both fuelling systems; and
4) compare the results with those of other research done on this fuel.

3. Literature Survey

3.1. Background

The emphasis on the background of this report is to develop an understanding on the basic operation of compression ignition engines in terms of performance, efficiency, details of the combustion process, fuel requirements and emissions formation. It is also required that a sound understanding of alternative fuelling is established, in order to deduce useful engine operational differences produced from alternative fuelling.

3.1.1. Basic operation of compression ignition engine

The compression ignition engine that was used is of a four-stroke, direct injection type. The basic operation of this engine is illustrated below.

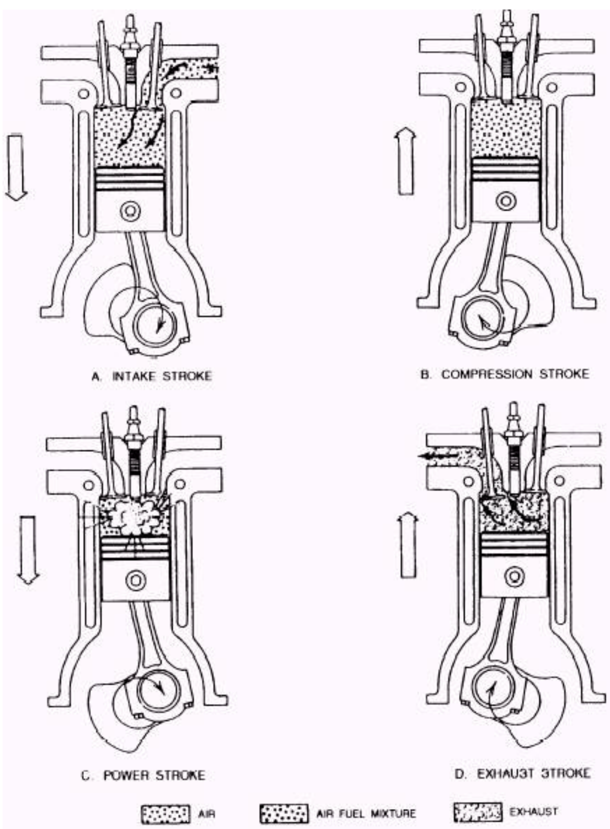

Figure 3-1: Four-stroke cycle of a compression ignition engine [4]

The four-stroke compression ignition cycle consists of four keystrokes:

- Intake Stroke;
- Compression Stroke;
- Power Stroke; and
- Exhaust Stroke.

The compression ignition engine differs from a spark ignition engine in that it only draws in air during the intake stroke rather than an air-fuel mixture. Also, the compression ignition engine has no spark plug. The air from the intake stroke is compressed, and thus its temperature increases significantly. The fuel is then directly injected into the hot pressurised air and ignites due to the combustion characteristics of the fuel. The ignition process causes expansion, forcing the piston down the cylinder through the power stroke. As the piston returns up, the exhaust valve is opened, and the burnt gases are expelled.

3.1.2. Torque and power

Engine torque, as in the case of this research report, is measured using a dynamometer. The dynamometer rotor is coupled electro magnetically (Wits research), hydraulically, or by mechanical friction to a stator. The stator is balanced with the rotor stationary.

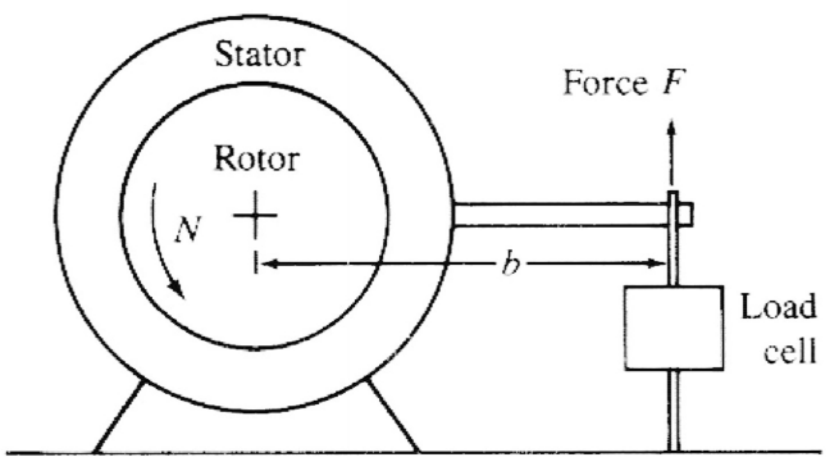

Figure 3-2: Schematic of the principle of operation of a dynamometer [3].

The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs or pneumatic means [3]. Using notation in figure 3.2, the torque, in Newton Metres (Nm), exerted by an engine T is calculated using:

$$T = Fb \qquad (1)$$

Where: F = Force (N)

b = Moment arm (m)

The power delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed, as is given by the following equation:

$$P = 2\pi NT \qquad (2)$$

Where: N = Crankshaft rotational speed (rpm)

It should be noted that torque is defined as the measure of an engine's ability to do work, and power is the rate at which work is done [2], and is known as brake power due to the method of dynamometer measurement described above.

3.1.3. Efficiency

In order to calculate the mechanical efficiency of any engine, the indicated power must be determined. The definition of indicated power is the rate of work transfer from the gas within the cylinder to the piston, which differs from brake power in that brake power takes into account the power lost in overcoming engine friction, driving engine accessories and pumping power. Indicated power (P_I) is calculated using the following equation:

$$P_i = \frac{W_i N}{n_R} \qquad (3)$$

Where: W_i = Indicated work per cycle (J)

n_R = Number of crank revolutions for each power stroke per cylinder

Thus, the mechanical efficiency can now be calculated, and is defined as the ratio of brake power delivered by the engine, to the indicated power:

$$\eta_m = \frac{P_b}{P_i} \qquad (4)$$

The air filter, intake manifold, intake port and intake valve together restrict the amount of air which an engine of given displacement can induct. Volumetric efficiency is the parameter used to measure the effectiveness of an engine's induction system and is defined as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston, and is calculated using the following equation:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \qquad (5)$$

Where: m_a = Mass of air (kg)

$_{a,\, i}$ = Inlet air density (kg/m^3)

V_d = Displacement volume (m^3)

3.1.4. Fuel consumption

Fuel consumption is measured as a flow rate, i.e. a mass flow per unit time ($\dot{m}_f$) [2]. A more useful parameter is the specific fuel consumption (sfc) and is defined as the fuel flow rate per unit power output:

$$sfc = \frac{\dot{m}_f}{P} \qquad (6)$$

3.2. Combustion

As explained above, a compression ignition engine experiences spontaneous ignition as fuel is sprayed at high velocity into hot compressed air at the end of the engine's compression stroke. As the fuel is injected through orifices at the injector tip, it atomises into small droplets and penetrates into the combustion chamber. The fuel vaporises and mixes with the high-temperature, high-pressure air. At this point spontaneous ignition of portions of the already mixed fuel and air occurs due to the air temperature being above the fuel's ignition point, however this only happens after a delay period of a few crank angle degrees. Thus the combustion process in a compression ignition engine is dependent on fuel spray behaviour, ignition delay and fuel characteristics [3].

3.2.1. Fuel Spray Behaviour

As explained above, fuel is introduced into the combustion chamber of a diesel engine through nozzles or orifices with a large pressure differential between the fuel supply line and the cylinder. The liquid column leaving the nozzle disintegrates within the cylinder over a finite length called the break-up length into droplets of different sizes. As the column moves away from the nozzle, the mass of the air within the spray increases, the spray diverges, its width increases, and the velocity decreases, as can be seen in the schematic below:

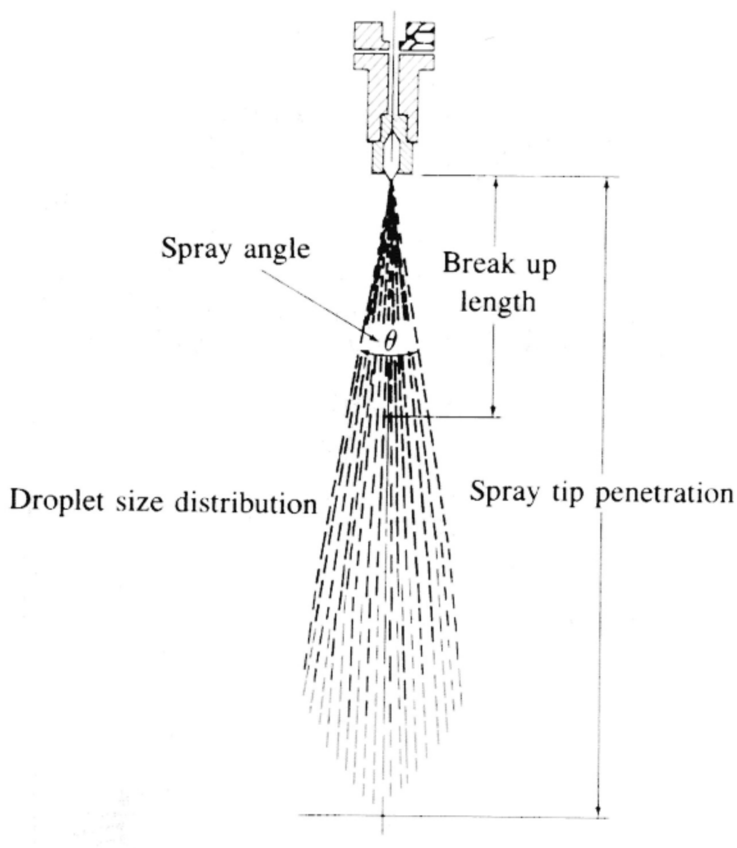

Figure 3-3: Schematic of diesel fuel sprays, defining its major parameters [3].

In direct injection engines used in the transportation industry, fuel is injected into swirling air, to increase fuel-air mixing rates. The schematic below illustrates the spray pattern resulting from a fuel jet injected radially outward into a swirling flow.

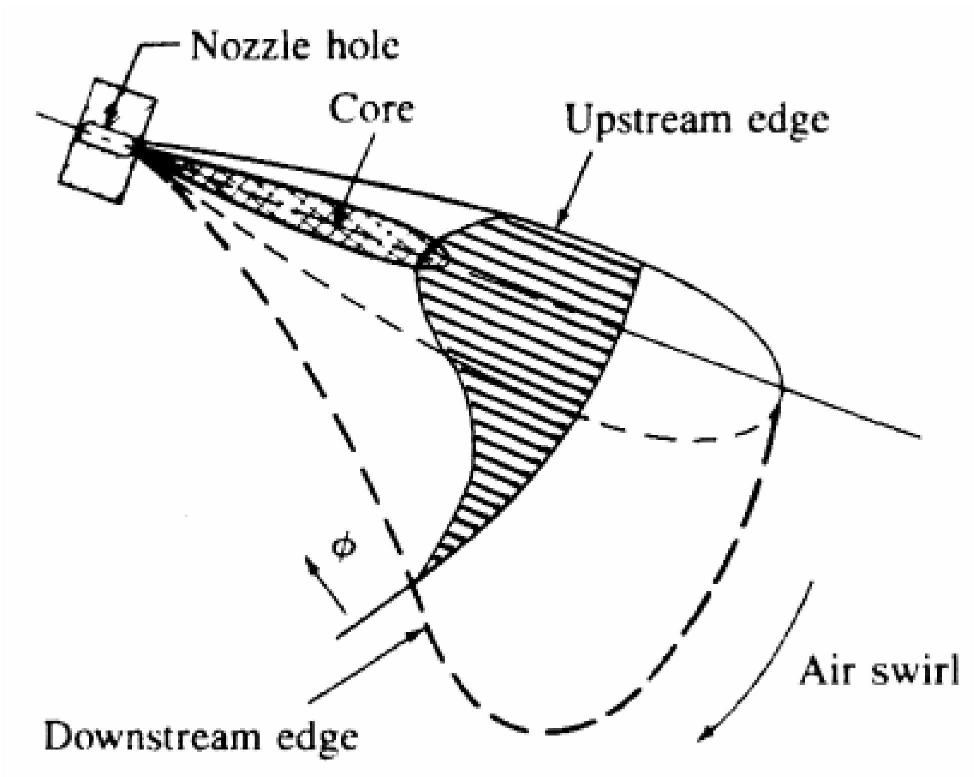

Figure 3-4: Schematic fuel spray injection into swirling airflow [3].

To couple the spray development process with the ignition phase of the combustion, it is essential to know which regions of the spray contain the fuel injected at the beginning of the injection process, since it is these regions of the spray that are likely to auto ignite first. The shaded area, labelled , represents equivalence ratio. Equivalence ratio is defined as the ratio of actual fuel/air ratio divided by the stoichiometric fuel/air ratio, and from the diagram it can be seen to be zero at the downstream edge and increases towards the core.

Also, droplet size distribution at a given location in the spray may also change with time during the injection period. Since the details of the atomisation process are different in the spray core than at the spray edge, and the trajectories of individual drops depend on their size, initial velocity, and the location within the spray, the drop size distribution will vary with position within the spray [3]. Droplet size distribution is also highly dependent on injector nozzle length/diameter ratio, air and fuel characteristics, such as viscosity and surface tension, as well injection pressure.

Atomisation

Under diesel engine injection conditions, the fuel jet usually forms a cone-shaped spray at the nozzle exit. This type of behaviour is classified as the atomisation break-up regime, and it produces droplets with sizes very much less than the nozzle exit diameter [3]. The most significant variables affecting atomisation are gas/liquid density ratio and nozzle geometry, which include jet spreading or spray angle, and the length/diameter ratio of the injector. For diesel injection, typical density ratios range between 15 x 10^{-3} and 30 x 10^{-3}, and it has been found that the length/diameter ratio is the most significant ratio.

Also, for jets in the atomisation regime, the spray angle was found to follow the relationship:

$$\tan\frac{\theta}{2}=\frac{1}{A}4\pi\,{\frac{\rho_g}{\rho_l}}^{\frac{1}{2}}\frac{\sqrt{3}}{6} \qquad (7)$$

Where: A = Constant for a given nozzle geometry

$_g$ = Gas density (kg/m^3)

$_l$ = Liquid density (kg/m^3)

Spray Penetration

The speed and extent to which the fuel spray penetrates across the combustion chamber has an important influence on air utilisation and fuel-air mixing rates [3]. In some combustion chambers, where walls are hot and high air swirl is present, fuel impingement on the walls is desired. However, this is not desired on multi-spray direct injection engines where over penetration on cool walls and little air swirl result in low mixing rates and increased emissions of unburned and partially burnt species. The penetration of liquid fuel sprays under typical diesel engine operation conditions has been extensively researched.

A formula developed by Dent best predicts the penetration (S) of the fuel spray tip across the combustion chamber for injection into quiescent air as a function of time:

$$S = 3.07 {\frac{\Delta p}{\rho_g}}^{\frac{1}{4}} (t d_n)^{\frac{1}{2}} {\frac{294}{T_g}}^{\frac{1}{4}} \qquad (8)$$

Where: p = Pressure drop across the nozzle (Pa)

t = Time after the start of injection (s)

$_g$ = Gas density (kg/m^3)

d_n = Nozzle diameter (m)

3.2.2. Ignition Delay

Ignition delay in a compression ignition engine is defined as the time (or crank angle) interval between the start of injection and the start of combustion. The start of injection is taken as the time when the needle lifts off its seat and is known as needle lift. The start of combustion is more difficult to determine and is in most cases identified from the change in slope of the heat-release rate determined from cylinder pressure data. The ignition characteristics of a fuel affect the ignition delay, and it is therefore important in determining diesel engine operating characteristics such as fuel conversion efficiency, smoothness of operation, misfire, smoke emissions, noise and ease of starting.

3.2.3. Fuel Characteristics

The ignition quality of a diesel fuel is defined by its cetane number. The cetane number is determined by comparing the ignition delay of the fuel with that of primary reference fuel mixtures in a standardised engine test [3], and is given by:

$$CN = \%\, n - \mathrm{cetane} + 0.15 \text{ x } \%\, \mathrm{HMN} \qquad (9)$$

Where: HMN = Heptamethylnonane

n = Hexadecane

For low cetane fuels with long ignition delays, most of the fuel is injected before ignition occurs, which results in very rapid burning rates once combustion starts with high rates of pressure rise and high peak pressures, resulting in incomplete combustion, reduced power output and poor fuel conversion efficiency. Under extreme conditions, when auto-ignition of most of the injected fuel occurs, an audible knocking sound is produced and is often referred to as diesel knock. Diesel knock is a result of a large amount of fuel reacting because of a very high rate of pressure rise at the beginning of the cycle.

For higher cetane number fuels, with shorter ignition delays, ignition occurs before most of the fuel is injected. The rates of heat release and pressure rise are then controlled primarily by the rate of injection and fuel-air mixing, and smoother engine operation results.

3.3. Emissions

Since the 1950's it has been recognised that transportation engines in the developed countries are the major sources of air pollution. One of the major drives in the research and development of alternative fuels is the need to discover fuels with less harmful emissions. In order to solve the problem of emissions reduction, a sound understanding of emission cause and effect need to be established.

3.3.1. Carbon Monoxide (CO)

The formation of carbon monoxide is attributed to a lack of oxygen during the combustion process, which theoretically should correspond to the chemical equilibrium equation that follows [2]:

$$H_2O + CO = H_2 + CO_2 \qquad (10)$$

At maximum flame temperatures, combustion yields insignificant quantities of CO relative to CO_2, but as the combustion gases become cooler, the equilibrium shifts in a direction favouring oxidation of CO to CO_2. The toxicity of carbon monoxide is well known, and it is a fact that its effects, such as the death of humans, are dependent upon both time and concentration.

3.3.2. Nitric Oxide (NO_x)

The principal oxide of nitrogen formed in combustion processes is nitric oxide, NO. Nitric oxide is a high-enthalpy species relative to N_2 and O_2 from the standpoint of basic thermodynamics. Therefore, its presence is favoured by the existence of high temperature [2]. The most undesirable toxic effect of nitric oxide (or any of the oxides of nitrogen) is the very small, but cumulative joining with moisture in the lungs causing respiratory problems. It is also a necessary component of photochemical smog [1].

3.3.3. Hydrocarbons

Unlike carbon monoxide or nitric oxide, hydrocarbons are not substances that one expects to find in high-temperature combustion gases [2]. The appearance of unburned hydrocarbons in the combustion process is a result of insufficient ignition. Ignition inhibition can be attributed to three engine conditions that can affect either the rate of internal heat generation or the rate of heat loss [2]:

1. Local mixture composition may be too rich or lean, causing slow oxidation reactions, resulting in ignition inhibition due to heat losses.
2. For very small isolated elements of fuel-air mixture the element surface-to-volume ratio may become so large and heat losses consequently so great that ignition cannot occur.
3. Heat losses from fuel-air mixture adjacent to a cool surface may be so great that ignition cannot occur.

Unburned hydrocarbons, combined with oxides of nitrogen and sunlight form photochemical smog, which causes watering and burning of the eyes and can also cause respiratory problems.

3.3.4. Particulates

Particulate material may be formed in solid, liquid or gaseous state. Liquid particulates are observed primarily in diesel engines at light load where ignition in very lean mixtures fails to occur. Particulate matter is made up from hydrocarbons, lead additives and sulphur dioxide [3]. Lead is well known as a toxic compound, and remains airborne in the atmosphere for appreciable times. Detrimental human effects from lead additives are inconclusive, and thus of less concern in comparison to the other harmful emissions.

3.3.5. Sulphur Dioxide

Sulphur dioxide from automotive vehicles is small in comparison to the content released due to coal burning. However, sulphur dioxides combined with fog were the primary ingredients responsible for the famous 1952 London Smog and the deaths at Donora, Pennsylvania in 1948 [2]. The combination of sulphur dioxide and moisture produce sulphuric acid, especially at high temperatures, which results in acid rain. They are also known to be particular harmful to human lungs and are related to the amount of soot formation and hence the amount of particulates introduced to the atmosphere during diesel combustion [7].

3.4. Comparison of Fuels

In order to establish whether a certain alternative fuel is suitable for fossil fuel replacement, their characteristics need to be compared before testing commences.

3.4.1. Fuel Properties

The table below shows a comparison of the properties of four different fuels, namely dimethyl ether (DME), diesel, propane and butane.

Table 3-1: Fuel Properties [8].

Property	DME	Diesel
Chemical formula	C_2H_6O	$C_{14.4}H_{24.9}$
Mole weight (g)	46	>100
Boiling Point (^oC)	-24.9	180 – 350
Vapour Pressure @ 25^OC (bar)	5.1	<0.01
Liquid Density (kg/m^3)	668	840
Liquid Viscosity (cP)	0.15	4.4 – 5.4
Low Heat Value (MJ/kg)	28.43	42.5
Explosion Limit in air (vol %)	3.4 - 17	0.6 – 6.5
Ignition Temperature (^oC)	235	250
Cetane Number	55 - 60	40 - 55

Table 3-1 continued

%wt of Carbon	52.2	86
%wt of Hydrogen	13.0	14
%wt of Oxygen	34.8	0

From the table it can be seen that DME is a favourable alternative fuel to diesel since it has a similar ignition temperature meaning that glow plugs for cold starting can be excluded.

DME has a cetane number higher than that of diesel, which results in more favourable combustion, and it also has a much lower weight percentage of carbon in comparison to diesel, resulting in cleaner emissions.

DME has the disadvantage, however, of possessing a boiling point of $-24.9^{o}C$, which makes fuelling a rather tedious task, requiring pressurising or cooling of the fuel delivery lines.

3.5. DME fuelling

Research and development of DME fuelling on standard diesel engines has been performed inside and outside of the University for a number of years. The section that follows outlines the findings of DME fuelled engine testing that has been completed at various institutions.

3.5.1. Previous Work

Outside of the University of the Witwatersrand

Sorenson and Mikkelsen presented an experimental study in which the use of neat DME in a small non-turbo charged diesel engine was demonstrated [9]. They found that with only minor fuel system modifications, such has having to pressurise the fuel line to prevent cavitations, DME gave very satisfactory combustion, performance and emissions.

It was concluded that DME was successfully tested as an alternative fuel for a small, non-turbo charged, direct-injection diesel engine, and obtained the following results [9]:

- Equivalent thermal efficiency
- Near-zero smoke
- NO_x emissions reduced by approximately 75%
- Lower engine noise

The engine, set at an injector opening pressure of 80 bar, operated for more than 500 hours without any lubricating additive and at 590 hours the conventional pump failed. It was also found that by incorporating 30% EGR (exhaust gas re-circulation), a 90% reduction of NO_x could be achieved, with no loss in thermal efficiency and no visible smoke, but with a 70% increase in CO emissions.

At the X'ian Jiaotong University, a light-duty direct–injection diesel engine with DME fuel was studied experimentally [10]. The effects of fuel injection parameters, in-cylinder air motion (such as a pump plunger diameter), nozzle type, fuel injection timing, nozzle tip protrusion, nozzle opening pressure and swirl ratio, on performance and emissions of the DME engine were investigated.

The investigations found that by installing a low-pressure pump, a fuel pressure regulator and a buffer in the fuel supply system, the vapour lock of DME in the fuel system is eliminated. Also, it was found that the engine ran smoothly on DME over a wide range of speeds and loads, with a thermal efficiency 3% higher than with diesel fuel. It was concluded that [10]:

- Pump plunger diameter, nozzle type, fuel delivery advance angle, vertical protrusion of nozzle tip into cylinder, nozzle opening pressure and swirl ratio greatly affected DME fuelled engine performance
- DME fuelled engines can significantly reduce NO_x emissions and achieve zero-smoke combustion
- With DME fuel, the injection delay was longer, the ignition delay was shorter, the peak pressure, maximum rate of pressure rise and the combustion noise were lower, in comparison with using diesel fuel
- DME is a prospective alternative fuel for compression ignition engines

A study performed at the National Traffic Safety and Environmental Laboratory by Li Jun, investigated DME fuelling on a direct injection single-cylinder diesel engine equipped with a test common rail fuel injection system to clarify how DME injection characteristics affect the heat release and exhaust emissions [11].

Firstly, to characterise the effect of DME, physical properties on the macroscopic spray behaviour, injection quantity, injection rate, penetration, cone angle and volume were measured using a high-pressure combustion chamber (4 MPa).

The following results were found [11]:

- Spray penetration using the 3-hole injector was a little longer, spray angle was wider and bigger spray volume was obtained compared with the 5-hole injection system. Fuel atomisation speed of the 3-hole injection nozzle was slow at the beginning of the injection process
- With the 5-hole injection nozzle, the rate of heat release in premixed combustion increases by faster atomisation and quicker vaporisation around the spray core. HC and CO emissions are sufficiently low when a small quantity of fuel is injected, but increases due to poor air introduction to the spray at a large quantity of fuel injection
- With the 3-hole injection nozzle, delay of fuel atomisation inhibits premixed combustion and results in an increase of unburned components at small quantities of fuel injection. However, when a large quantity of fuel is injected, diffusion combustion is active compared to that with the 5-hole injection nozzle and it is possible to suppress an increase in CO and HC emissions
- Atomisation speed of the 3-hole injection nozzle is low, and higher injection rates result with longer ignition delay, allowing spray to be distributed in a combustion chamber more extensively. Unburned components are decreased and an increase in CO emissions is inhibited at large quantities of fuel injection.
- When the 5-hole injection system was used, a higher swirl ratio results in a longer ignition delay with large quantity of injection fuel. During this time the spray is distributed in the combustion chamber, where mixture with air is enhanced. Therefore contributing to the reduction of HC and CO emissions with the 5-hole injection nozzle in particular
- Due to the differences in injection process of the nozzles, when a large quantity of fuel is injected, the 3-hole injection nozzle with an equivalent energy consumption rate to the 5-hole injection nozzle results in less NO_x emissions.

In a study performed at the Hiroshima University, experiments were performed of ignition characteristics of the DME and diesel fuel sprays injected by a DI diesel injector into a high temperature, high-pressure vessel [12]. A Schlieren optical system was adopted for visualising the ignition process as well as the vaporisation process of the DME and diesel fuel sprays.

Ignition delay was measured using a photo-sensor of high sensitivity. Pressure and temperature of the ambient air and oxygen concentration of the ambient air were changed as experimental parameters. Under the specified conditions, the following was found [12]:

- A non-luminous flame is observed in the ignition process of the DME spray.
- In the case of the ordinary air (concentration Xo_2 = 20.6%), the ignition delay of the DME spray is almost the same as those of the diesel fuel spray under the high ambient pressure, whereas under the low ambient pressure the ignition delay of the DME spray is shorter than that of the diesel fuel spray
- The ignition delays of the DME and diesel fuel sprays in the low oxygen concentration air (Xo_2 = 10%) are longer than those in the ordinary air (Xo_2 = 20.6%). The ignition delays of the DME and diesel fuel sprays in the low oxygen concentration air are decreased as the ambient air pressure is increased, even in the high ambient air pressure range. In the case of the ordinary air, there is not much difference in the ignition delay with the ambient air pressure for the DME and diesel fuel sprays in the high ambient air pressure range
- In the case of the low oxygen concentration air (Xo_2 = 10%), the ignition delay of the DME spray is shorter than that of the diesel fuel spray under any ambient air pressure. An increase in the ignition delay with decreasing oxygen concentration of the ambient air from Xo_2 = 20.6% to Xo_2 = 10% is smaller for the DME spray than that for the diesel fuel spray
- The gradient of the ignition delay line of the diesel fuel spray in the high ambient air temperature range is smaller in the case of the low oxygen concentration air (Xo_2 = 10%) than that in the ordinary air (Xo_2 = 20.6%).

At the University of the Witwatersrand

Kaczor submitted a research dissertation in 2002 entitled “A comparative study of the combustion and emissions of a compression ignition engine fuelled on Diesel and Dimethyl Ether”.

His testing was conducted on the same engine as that used in the present investigation, but his methodology was different in that constant speed testing at an injection pressure of 100 bar was conducted. Kaczor reported the following findings [14]:

- Energy release in DME was higher than diesel at the same load and speed
- An additional phase of energy release was found during DME fuelling
- The engine was capable of producing 75% of the maximum torque at each of the speeds when fuelled on DME
- The maximum torque obtained during DME fuelling was approximately 50 Nm
- The fuel consumption during DME fuelling was, under certain speed and loading conditions, almost 300% greater than that of diesel
- DME was found to be a desirable alternative fuel to diesel

3.5.2. Applications and Characteristics of DME

The term 'Alternative Fuels' considers alternatives to conventional diesel and gasoline fuels [13]. Figure 3-5 below shows a schematic overview of commonly used fuels for internal combustion engines and a variety of substances, which are currently, discussed as alternative fuels [13].

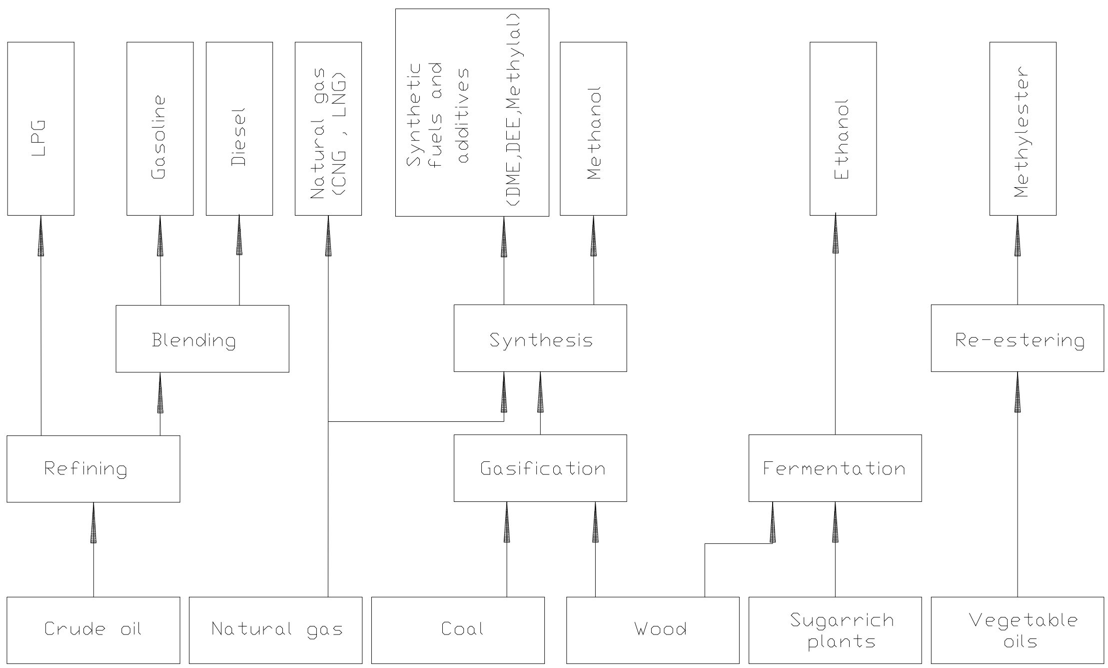

Figure 3-5: Overview of fuels for IC engines and their resources [13].

It is estimated that in terms of energy, the fossil gas resources of the world are equivalent to those of oil. Although widely used as automotive fuel, natural gas (CNG – Compressed Natural Gas and LPG – Liquefied Petroleum Gas) has the disadvantage of a cost intensive fuel storage and transportation.

LPG has been introduced as an alternative automotive fuel for several decades but has gained importance in recent years because of its ultra clean combustion properties. LPG (propane and butane) is a by-product of both the crude oil refinery process and natural gas preparation. In contrast to natural gas, LPG is a liquid at moderate pressures.

Before DME was tested as a pure fuel, it was considered as an ignition promoter for other fuels, mainly methanol. The idea of using pure DME as a fuel was challenging until it was found that DME could be produced economically in large scale. Due to extensive research in this field, today it is obvious that DME's excellent ignition and combustion properties are best utilised by high-pressure direct injection of liquid DME into the cylinder.

DME is the fuel of concern in this report, and has the following chemical formula:

$CH_3 – O – CH_3$

It is currently used as a propellant for spray cans because of its favourable environmental characteristics [9]. The properties of DME are given in Table 3-1, where its properties are compared to those of diesel.

From table 3-1, it can be seen that one of DME's most important characteristics is its low ignition temperature, which is similar to that of diesel fuel [9]. There are also, however, problems with the use of DME. Firstly, its high vapour pressure may require fuel system attention and secondly its low viscosity, which is expected to give problems with injection system durability in a conventional type of fuel injection system.

4. Analysis

To improve the results obtained in previous testing conducted at the University, certain adjustments and modifications were made to the test rig.

4.1. Adjustments and Modifications

The first adjustment made to the engine was setting the injection timing. Previous testing resulted in advancement and retardation of the injection timing, and thus the correct timing position was uncertain.

At a load of 35 Nm and a speed of 1500 rpm under diesel-fuelled operation, the maximum cylinder pressure was approximately 6000 kPa prior to the injection timing adjustment. The timing on cylinder 1 (cylinder closest to the flywheel of the engine) was advanced by approximately 5^{o}, which resulted in the cylinder pressure increasing from 6000 kPa to 6800 kPa. The timing was also advanced on cylinder 2. This adjustment resulted in a further increase in cylinder pressure to 7500 kPa.

Preliminary testing showed that diesel and DME fuel flow rates recorded in previous testing may have been inaccurate due to fuel overflow from the injectors that was not accounted for. This was correct for diesel operation by rerouting the overflow fuel line from the overflow of the injectors to the fuel line, just after the fuel flow meter. For DME, a calibrated rotameter was fitted to the overflow fuel line. The reading obtained from the overflow rotameter was subtracted from the primary rotameter to give an accurate measurement of actual DME flow rate.

Measuring DME fuel flow has been a major obstacle in the past due to a positive displacement pump incorporated into the DME fuel line to ensure pressurised operation. The reason for this is that the positive displacement action of the pump causes fluctuation in any types of fuel measuring systems previously used. Thus, in the past cumbersome load cell measuring devices requiring counter weights were experimented with, and it was found that fluctuations in the load cell output voltages were too great to produce accurate data. In order to rectify this, a diaphragm pump, which is also a positive displacement pump, with a smaller and more rapid stroke, was used. This allowed DME mass flow measurement with a calibrated primary rotameter, fitted down stream of a non-return valve, directly out of a pressurised DME bottle.

Prior to the diaphragm pump being fitted to the system, a damping system was designed to try and dampen the pulsations of the pump. A T-shaped damper was manufactured from seamless steam pipe, but was found to be unsuccessful due to DME being capable of dissolving in air.

At early stages of testing, whilst calibrating the airflow measurement device, it was found that during acceleration of the engine from 1150 rpm to 2000 rpm, the airflow increased linearly, but dipped between engine speeds of 1600 – 1700 rpm. This was believed to be due to the air choking through a 38 mm orifice plate. A 52.4 mm orifice plate was then fitted and the calibration was repeated. It was found that a similar dipping characteristic was found but this time between 1700 – 1800 rpm. It was decided to revert to the small orifice plate to allow comparison between results obtained during this investigation and results obtained from past investigations.

An electronic throttle controller, with PI feedback control was fitted to the engine. This allowed, for the first time, performance of constant torque testing instead of constant speed testing. Also, the controller allowed the operator to adjust the engine speed from a knob on the controller, or to control the entire engine, both speed and load, from the speed \ torque controller unit in the computer room of the test cell.

The engine also underwent a few other minor modifications. Firstly the engine mountings were changed due to excessive vibration caused by worn and old engine mountings. Towards the end of testing the engine seized, which in all probability was caused by excessive heating of the engine due to poor cooling by the fans, which may have been placed too far from the engine, and poor lubrication characteristics of DME. The engine was stripped and rebuilt replacing gaskets, pistons, rings and having the cylinder sleeves honed due to scouring caused by the seizure.

Finally, the most important modification made was to allow the engine to run for extended periods while fuelled on DME. Previously, the average operating time for a DME fuelled run was approximately 15 minutes. The problem was thought to be vapour locking caused by the vaporisation of DME due to excessive heating at the engine. Numerous methods to solve the problem, such as water-jacketed fuel lines, ice packing and rerouting of fuel lines, all proved to be inadequate. After the diaphragm pump was fitted to the DME fuel line, operating runs for longer than 15 minutes were still impossible. A simple simulation was performed whereby the pumping system was isolated from the engine, and DME was allowed to circulate through the pumping system in a closed loop formation. It was found that whilst isolated from the engine, the pumping system held a pressure of 50 bar for approximately 15 minutes, and then suddenly experienced a complete pressure drop.

Pipe lines were shortened, needle valves were fitted and the pump was lowered from a high shelf to floor level. Unfortunately, this was not the solution. By circulating water rather than DME, in a closed loop system, it was found that a pressure of 50 bar could be maintained for as long as required. This meant that it was not a faulty pumping system but rather that DME was vaporising long before reaching the engine, and in fact at the pump (see Figure 5-14).

The pump has a return line so that the pump is able to deliver more fuel than the engine can consume at any given time. A pressure relief valve that was used to maintain a constant 50 bar pressure upstream of the pump and 4 bar pressure in the return line was fitted to the beginning of the return line. As the system operates, more fuel is circulated through the return than pumped to the engine. Thus, the pumping action of the pump does work on the pumping fluid, which in turn causes heating, and vaporisation of DME in the return line.

Using a basic cooling system rectified this problem. A copper brake line was bent into a 50 mm diameter, 120 mm long cylindrical coil. This coil was fitted to the return line and placed into a polystyrene box of ice and water. Using this new cooling method on the DME fuel line, the engine ran at 1600 rpm at a load of 60 Nm for 40 minutes before it was voluntarily turned off. Never before had the engine run at a load of 60 Nm while fuelled on DME.

5. Experimental Facilities

5.1. General Layout

The test cell in which this particular alternative fuel research was performed is test cell No. 3, in the Applied Thermodynamics Laboratory at the University of the Witwatersrand. The cell is divided into two rooms, one containing the data acquisition system, comprising of a computer, amplifiers, meters, an analogue to digital converter and an engine speed / torque controller unit. The other room contains an engine test bed, comprising of a 1.33 litre Lister Petter compression ignition engine coupled to an eddy current dynamometer, fuel tanks for both DME and diesel fuel, cooling fans and a pump for feeding DME from the fuel tank to the engine.

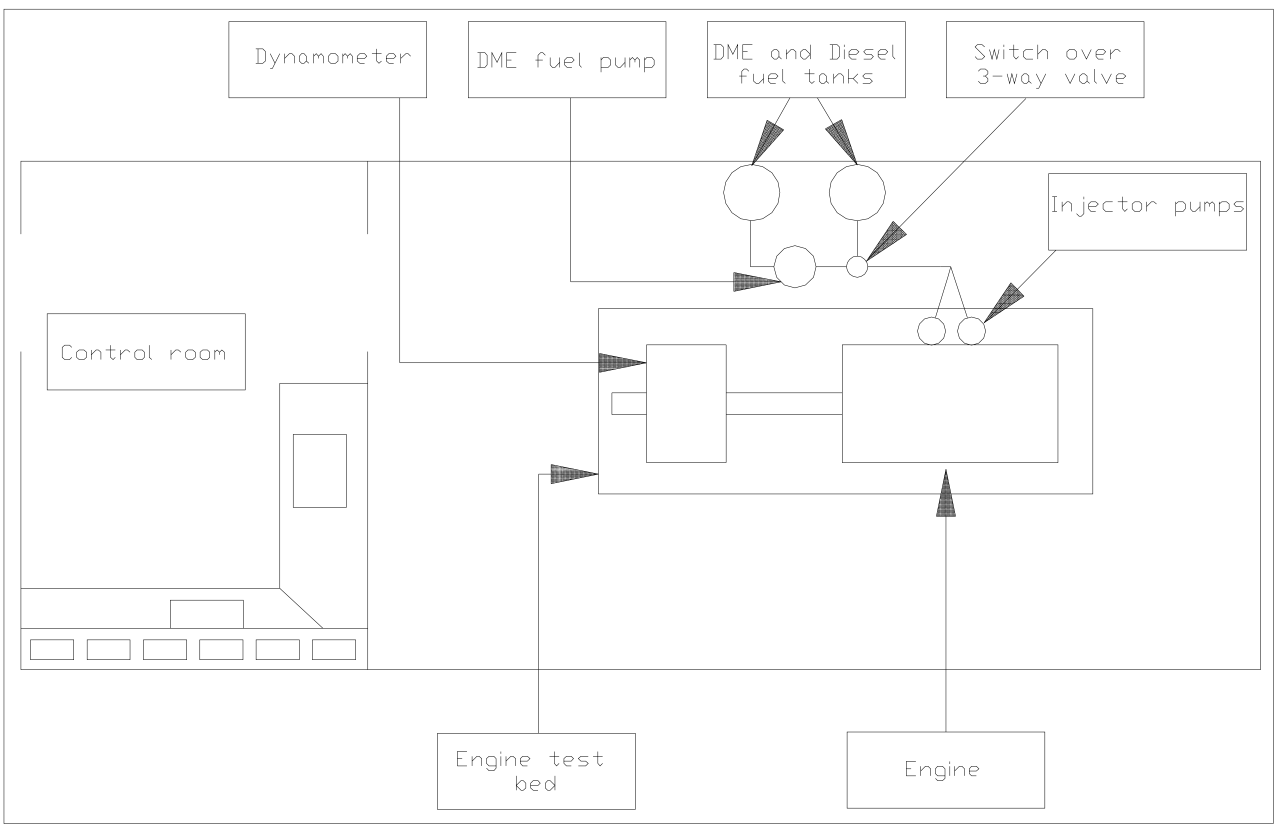

Figure 5-1: Test cell floor plan

5.2. Apparatus

5.2.1. Engine

The engine utilised for the purpose of this research project was a 4-stroke, 1.33 litre, direct-injection, two cylinder, air-cooled compression ignition engine. The engine was coupled to an eddy current dynamometer, which is used to simulate loading conditions on the engine.

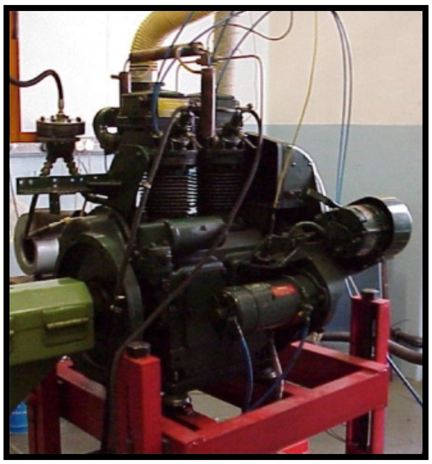

Figure 5-2: Lister Petter engine

As described in the chapter 4, modifications made to the engine set up were the rerouting of the injector overflow fuel line and the replacement of the old engine mountings by an engine cradle that was fixed to the steel test bed via new engine mountings.

The overflow line configuration for diesel and DME testing are slightly different and will be described and illustrated later in the report.