University of Southern Queensland

Faculty of Engineering and Surveying

COMPRESSED NATURAL GAS AS AN ALTERNATIVE FUEL IN DIESEL ENGINES

A dissertation submitted by

WONG, Wei Loon

in fulfillment of the requirements of

Courses ENG 4111 and ENG 4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: October, 2005

ABSTRACT

This project aims to investigate the engine performance and exhaust emission of dual-fuel operation on a single cylinder compression ignition engine. Natural gas is used as the main gaseous fuel with diesel as the pilot fuel to provide a source of ignition. The high compression ratios of diesel engines can be achieved without loss in power together with substantial cost reduction in fuel and conversion kits.

Comparative results of dual-fuel operation with conventional diesel fuel through experimental results demonstrated its benefits both in the fields of performance and emission. The engine torque and brake power output show vast improvement and dual-fuel operation is able to achieve a higher thermal efficiency under all operating conditions. The emission levels of polluting gases such as carbon monoxide, oxides of nitrogen (NO_x) and carbon dioxide also records an enormous decrease.

University of Southern Queensland

Faculty of Engineering and Surveying

ENG4111 & ENG4112 *Research Project*

Limitations of Use

Certification

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are extremely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been preciously submitted for assessment in any other course or institution, except where specifically stated.

Wong Wei Loon

Student Number : 0050027398

Signature

Date

ACKNOWLEDGEMENTS

I would like to express my gratitude to my supervisors -Dr. Harry Ku who guided me through this project as well as providing a lot of handy tips, Dr. Fok Sai-Cheong who provided assistance and help to me regarding the research work during his time here and Dr. Talal Yusaf from UNITEN who is my local supervisor back in Malaysia, who gave me permission to access the labs to conduct the experiments for this research.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

λ	-	Air to fuel ratio
p	-	Cylinder pressure
V	-	Cylinder volume
η_f	-	Engine thermal efficiency
τ	-	Engine torque
Q_{ch}	-	Gross heat release rate
Q	-	Heat transfer
Q_{ht}	-	Heat transfer rate to the cylinder walls
Q_{HV}	-	Lower calorific value of fuel
$\dot{m}_f$	-	Mass flow rate of fuel
Q_n	-	Net heat release rate
n_R	-	Number of crank revolutions for each power stroke per cylinder
n	-	Number of engine cylinders
γ	-	Ratio of specific heats
h_f	-	Sensible enthalpy of fuel
U	-	Sensible internal energy of cylinder contents
c_p	-	Specific heat at constant pressure
c_v	-	Specific heat at constant volume
W_c	-	Work produced per cycle

GLOSSARY

A	-	Area of engine bore
bmep	-	Brake mean effective pressure
BP	-	Brake power
bsfc	-	Brake specific fuel consumption
C_2H_6	-	Ethane
C_3H_8	-	Propane
CH_4	-	Methane
CI	-	Compression Ignition
CNG	-	Compressed Natural Gas
CO	-	Carbon monoxide
CO_2	-	Carbon dioxide
CR	-	Compression ratio
DC	-	Direct Current
DPM	-	Diesel Particulate Matter
ECU	-	Electronic Control Unit
H	-	Hydrogen
IP	-	Input power
L	-	Length of engine stroke
LNG	-	Liquefied Natural Gas
N	-	Engine speed
NGV	-	Natural Gas Vehicles
NO	-	Nitric oxide
NO_2	-	Nitrogen dioxide
NO_x	-	Nitrogen oxides
O_2	-	Oxygen
OH	-	Hydroxide
P	-	Power developed by engine
PM	-	Particulate Matter
rpm	-	Revolutions per minute

SCR	-	Selective catalytic reduction
sfc	-	Specific fuel consumption
SO_4	-	Sulfate
SOF	-	Soluble Organic Fractions
THC	-	Total unburnt hydrocarbon
V	-	Voltage

CHAPTER 1

INTRODUCTION

1.1 PROJECT BACKGROUND

Diesel fuel in compression ignition (CI) engines produces a high level of toxicity in emission gases (Kojima, 2001) which leads to a health and environmental hazard. The high level of nitrous oxides (NO_x), carbon monoxide (CO), carbon dioxide (CO_2) and particulate matter 10 (PM10) emission associated with diesel fuel has long been an issue. Although the use of diesel is favorable in fleet vehicles since it produces a high compression ratio to enable generation of more power, Kelley (undated) reported that the higher compression ratio causes a significant problem in starting the engine at low temperatures. Wills (2004) supported the findings and mentioned that fuel type plays an important role in the ease to start the engine.

Natural gas has been considered as a potential substitute to conventional fuels in vehicles due to its lower emission of greenhouse gases and safety properties. A frequent report by natural gas vehicle (NGV) owners is that CNG powered vehicles have less power and shorter driving range (Graham, 2000). In fact, this is due to the lower compression ratio when spark ignition dedicated CNG engines are used. The emission and reduced performance problems of both diesel and dedicated CNG engines can be eliminated by the use of dual fuel diesel-CNG engines.

1.2 PROJECT OBJECTIVES

The main object of this project is to research the effects of using CNG as an alternative fuel as a replacement for diesel in compression ignition engines. This means that dual-fuel engine must be used to utilize diesel as the pilot fuel to ignite CNG. The engine performance and emission qualities are to be investigated by running the engine at different speeds with varying set of loads. The sub-objectives of the project are:

a) Research the history of CNG usage worldwide and a literature review on the engine performance using CNG as the main fuel supply inclusive of the advantages and limitations.

b) Conversion of the current CI engine to install the CNG fuel system to enable the use of dual fuel diesel-CNG engine.

c) Study on the effect of using CNG as fuel in terms of power, torque, brake specific fuel consumption (BSFC), and thermal efficiency. Perform a comparison analysis on the dual fuel combustion and conventional diesel fuel.

d) Examine the emission data collected for both fuels and conduct feasibility study on CNG as a fuel alternative in terms of pollution and economy.

1.3 PROJECT METHODOLOGY

Initially, literature review of CNG usage worldwide is compiled and commented. Information on usage of CNG fuel in vehicles worldwide and locally was gathered from online sources, journals, magazines and newspaper articles. A concise summary of properties of natural gas is presented and critically documented. At the same time, reviews are made on the advantages and disadvantages of CNG as fuel compared to conventional fuels such as diesel and gasoline.

Next, the engine performance of the dual fuel CNG-diesel engine will be analyzed with properties such as engine torque, brake power, brake specific fuel consumption, brake mean effective pressure and engine efficiency being emphasized. The gaseous emission types are also explained by examining the formation and causes of pollutants such as nitrous oxides, carbon monoxide, carbon dioxide, and total unburnt hydrocarbons.

The experimental procedures and setup will then be explained to illustrate the method of measuring engine performance and emission in the laboratory. Experimental data will be recorded systematically for different engine speeds and varying loads to enable comparisons of pure diesel and dual-fuel to be made. The data collected will be tabulated and relevant graphs plotted. Next, the results will be critically analyzed and finally a conclusion is made based on the experimental results.

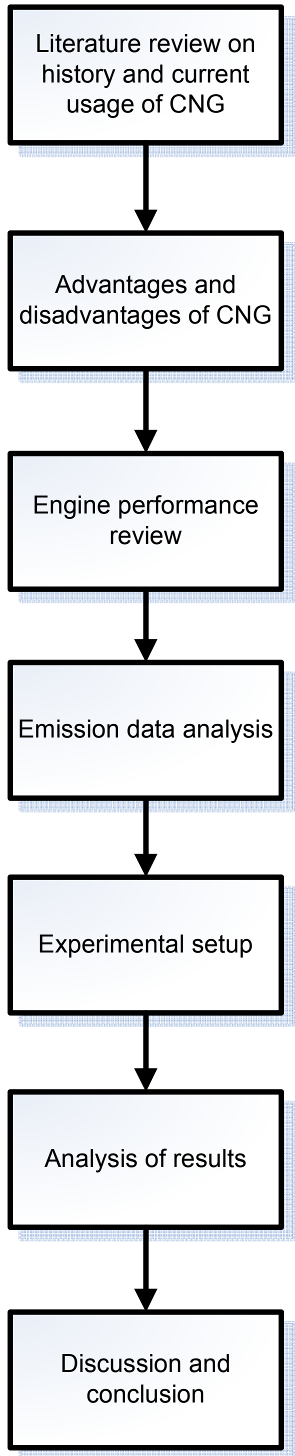


Figure 1.1: *Project Methodology*

CHAPTER 2

LITERATURE REVIEW

2.1 COMPRESSED NATURAL GAS (CNG)

2.1.1 Introduction

Compressed Natural Gas is composed primarily of methane (CH_4), and other hydrocarbons such as ethane, propane and butane. According to Alternative Fuel Data Centre (2004), the composition of CNG is further enriched with other gases such as carbon dioxide, hydrogen sulphide, nitrogen, helium and water vapour. The content of CNG was believed to originate from plants and animal remains which had decomposed for millions of years (Info Comm, 2005). Natural gas is formed deep underground trapped between layers of rock and sand in reservoirs underneath the Earth, like other fossil fuels. Due to its lower density characteristics, CNG will float above other trapped substances such as crude oil and water.

A drilling rig is used to penetrate the Earth surface to draw out the natural gas. The extract is then refined to remove impurities and transmitted through a series of pipelines to processing plants and then sent to transmission companies before reaching the end-user (NaturalGas.org, 2004).

Natural gas was first used as fuel to boil water, light street lamps and gained worldwide acceptance to be used in residential homes as water heaters, clothes dryer and in cooking. The Pacific Gas and Electric Company (2003) reported that the natural gas boom period began in 1950s in the United States where a huge network of facilities and distribution pipes were constructed for the purpose of promoting the use of natural gas. The use of natural gas in the transportation sector began as early as the 1930s and had little development since then (NaturalGas.org, 2004). Without the support of the public, the use of Natural Gas Vehicles (NGV) is limited mostly to the public transportation sector.

Natural gas is compressed as CNG to be used as fuel in the vehicles with the alternative being Liquefied Natural Gas (LNG). The former is most widely used in alternative fuel vehicles. It promotes environmental friendliness with its low emission of harmful gases and comparable engine performance (U.S. Environmental Protection Agency, 2002).

2.1.2 Usage of Compressed Natural Gas

NaturalGas.org (2004) reported that natural gas was originally used by the Chinese as a fuel to process seawater to separate its salt contents and make it drinkable. In Europe, Britain was the first country to commercialize the use of natural gas although its use was limited only to street lighting (Gas-Lite Manufacturing, 2004). The method then spread to other parts of the world including the United States. After the Second World War, improvements were made to the transportation and storage of natural gas with extensive use of the technology available then (NaturalGas.org, 2004). As predicted, with the increased area coverage of natural gas supplies, it became an increasingly popular source of energy for the public.

Intense research was performed to analyze the feasibility of using natural gas as a substitute for conventional fuel like gasoline and diesel. According to Shamsudin & Yusaf (1995), Italy became the leading country in research of natural gas as an alternative fuel and had around 235 000 vehicles converted to be powered by the fuel. The United States is now the world's highest consumer of natural gas with 28.8% consumption followed by the Russian Federation (BP Global, 2005). Almost 130,000 NGVs are operational in the United States with an approximation of 2.5 million vehicles worldwide (NaturalGas.org, 2004). The Pacific Gas and Electric

Company (2003) reported that there are more than 40 natural gas vehicles manufacturing companies available worldwide. This shows a significant expansion in the field of research of NGVs.

CNG are primarily used as fuel for transit buses, taxi cabs, heavy duty trucks and other public vehicle fleet worldwide. The high fuel usage of these vehicles makes conversion to CNG fuel more economical and decreases the payback period of the conversion cost. Due to the environmental benefits of CNG as an alternative fuel, the American Government is investing heavily in the field of Research & Development and providing subsidies to support NGVs in order to encourage the use of natural gas (Natural Gas Vehicle Coalition, 2005).

In Malaysia, the popularity of NGV is limited and the majority of vehicles are still running on conventional fuel due to the inadequate fuelling stations and long fuelling time for natural gas. Although Malaysia has an abundant natural gas reserve estimated at around 82.5 trillion cubic feet (Autoworld, 2004), the market penetration remains minimal and the majority of natural gas is exported. However, steps are being implemented to promote the use of NGVs through publications and education to instill customer awareness on its vast benefits. PETRONAS Company, through its subsidiary, PETRONAS NGV Sdn. Bhd., has been promoting the use of CNG to Malaysian motorists by providing services and fuelling facilities (Autoworld, 2004). Apart from that, Malaysia has been selected as the host nation for the Asia Pacific Natural Gas Vehicles Association (ANGVA) conference in 2005 to discuss the technological developments in the use of NGVs.

2.1.3 Natural Gas Composition

Natural gas generally consists of a mixture of hydrocarbons with methane (CH_4) as the main constituent. Ethane, propane, butane, nitrogen and carbon dioxide gases contribute to the remaining composition while traces of water vapour and hydrogen sulphide may be present in some natural gases. The properties of natural gas will vary depending on the location, processing and refining facilities. Usually, the maximum and minimum compositions are specified to enable comparisons to be made.

Compound	Typical	Maximum	Minimum
Methane	87.3%	92.8%	79.0%
Ethane	7.1%	10.3%	3.8%
Propane	1.8%	3.3%	0.4%
Butane	0.7%	1.2%	0.1%
Nitrogen	2.2%	8.7%	0.5%
Carbon Dioxide	0.9%	2.5%	0.2%

Table 2.1: *Typical Composition of Natural Gas in Percentage (Questar Gas, undated)*

Current research on the natural gas vehicles found that the engine performance and emission are greatly affected by varying compositions of natural gas (Ly, 2002). It was also reported that the heating value, efficiency, and concentration of unburnt hydrocarbon and other emission particles would highly depend on the source of supply of natural gas as the main fuel. Ly (2002) also mentioned that this effect is especially dominant in heavy-duty engines with high compression ratio applications due to the increased amount of engine "knocking". Engine knocks are caused by the pre-mature ignition of the air-fuel mixture in the combustion cylinder, causing the engine to overheat and run inefficiently.

According to Natural Gas.org (2004), the raw natural gas is processed to remove impurities such as oil, condensate and water particles. The presence of these particles may obstruct the smooth flow of fuel into the engine when in use and may even bring the engine to a halt. 'Dry' natural gas, which consists of almost entirely methane, is then obtained by distilling the other hydrocarbons.

2.1.4 Natural Gas Properties

Natural gas in its original form is non-toxic, colorless and odorless (Questar Gas, undated). A chemical substance called Mercaptan is added to natural gas to add a scent of rotten egg as a safety precaution so that leakage may be detected by the human olfactory sense (Info Comm, 2005). Inhalation of natural gas will not interfere with the body functions or cause detrimental health damage to our body. Barbotti CNG (2002) mentioned that the natural gas does not emit any aldehydes and other air toxins, which may be an issue with other fuel types.

Apart from that, natural gas is also lighter than air due to its low density. According to Clean Air Technologies Information Pool (2005), a natural gas spill would be less dangerous compared to a gasoline or diesel oil spill since the natural gas vapor would dissipate into the air and not accumulate on the ground.

The non-corrosive characteristic of natural gas is favorable to prevent oxidation of storage tanks and hence will reduce the possibility of contamination. Table 2.2 provides a comparison on the physical properties of compressed natural gas (CNG) and conventional diesel fuel. As can be seen, natural gas comprises primarily of CH_4 while the hydrocarbon chains in diesel are longer and more complex. CNG also shows a lower molecular weight and specific gravity compared to diesel.

Research has shown that natural gas has a narrow combustion limit between 5 to 15 percent (Questar Gas, undated). This implies that combustion of natural gas will only take place when concentration of natural gas in the air lies in the range mentioned. Combined with its high ignition temperature, natural gas can be safely used without the high risk of accidental explosion.

Property	Compressed Natural Gas (CNG)	Conventional Diesel
Chemical Formula	CH_4	C_3 to C_{25}
Molecular Weight	16.04	≈200
Composition by weight,%		
Carbon	75	84-87
Hydrogen	25	13-16
Specific Gravity	0.424	0.81-0.89
Density, kg/m^3	128	802-886
Boiling temperature, °C	-31.7	188-343
Freezing point, °C	-182	-40-34.4
Flash point, °C	-184	73
Autoignition temperature, °C	540	316
Flammability limits, % volume		
Lower	5.3	1
Higher	15	6
Specific Heat, J/kg K	-	1800

Table 2.2: *Properties of Natural Gas and Diesel (Alternative Fuels Data Centre, 2004)*

According to P.C. McKenzie Company (undated), the octane number of CNG is 120 compared to 87-93 of gasoline. The octane number measures the potential of "knocking" in the engine due to fuel selection. A high octane number signifies a higher resistance to engine "knocking" and increased efficiency of a smooth power transmission. A direct comparison cannot be made with diesel fuel as it operates in a compression-ignition engine and is measured with a property called cetane number.

2.2 ADVANTAGES AND LIMITATIONS OF CNG

2.2.1 Advantages

Barbotti CNG (2002) reported that CNG is the world's cleanest operating fuel in engines due to its low emission levels of nitrous oxides (NO_x), carbon monoxide (CO) and carbon dioxide (CO_2) which contributes to the overall greenhouse effect and global warming. Lewis (2005) added that the CNG is free of benzene and therefore eliminates the health risk of consumers who may be directly exposed to the carcinogenic material.

According to NGV.org (2001), the amount of total hydrocarbon (THC) and Particulate matter 10 (PM10) are greatly reduced with the use of NGVs. The environmental benefits are one of main reasons why most governments around the world are promoting the use of CNG as fuel in consumer vehicles (Gwilliam, 2000). Currently, the Malaysia government is also promoting the use of NGVs by providing a 25% deduction in road tax for all vehicles running on CNG (Petronas Dagangan Berhad, 2005).

Another main advantage of NGV is from the economics point of view. The present price of natural gas is RM0.565 (AUD $0.195) per litre compared to RM1.20 (AUD $0.414) per litre for petrol and RM0.881 (AUD $0.304) per litre for diesel (Petronas Dagangan Berhad, 2005). This indicates a substantial 53% and 27% savings in fuel cost respectively. NGV.org (2001) reported that the price of natural gas is also more stable than other fuels. The massive cost savings of CNG fuel will definitely encourage transportation companies and end users to consider purchasing dedicated NGVs or switching to the alternative fuel.

With an abundant reserve of natural gas and network of dedicated piping systems, it is convenient for NGV users to gain access to natural gas and refuel their vehicles by just installing a home refueling system (NGV.org, 2001). Wide usage of natural gas will also help reduce the dependence on finite petroleum fuels and avoid a steep price increase in fuels.

Kojima (2001) researched that the use of natural gas in buses produces less noise and vibrations compared to conventional fuel. This will lead to longer service life and reduced maintenance costs. Fleet operators also reported a 40% savings on maintenance costs since interval between vehicle check-ups is lengthened (Barbotti CNG, 2002). Engine performance are also claimed to be superior to gasoline engines since NGVs encounter less knocking and has a wide range of temperature tolerances (Barbotti CNG, 2002).

2.2.2 Limitations

NGVs are mostly used in the fleet transportation industry compared to private vehicle owners due to its high initial cost of engine conversion. According to Natural Gas Vehicle Coalition (2005), this is caused by low production volumes of NGVs to accommodate economies of scale. Although the government is paving the way to encourage CNG usage, customer awareness still remains low due to vague marketing strategies which lack focus (Natural Gas Vehicle Coalition, 2005).

Researches have shown a slight decrease in engine performance- around 10-15% in CNG fuelled vehicles (Indian Energy Sector, 2000). Graham et al. (2000) researched that the lower compression ratios with dedicated CNG engines compared to diesel engines is the main reason for this power decrease. The spark ignition (SI) engine in dedicated NGVs will not operate above a 11.5:1 ratio (Clean Air Power, undated) but the problem may be resolved by using a dual-fuel engine which will be discussed later.

Murray et al. (2000) revealed that another factor which causes NGVs to be unpopular among consumers is the lack of refueling station available in most countries. For instance in Malaysia, the availability of CNG refueling stations are limited as only a subsidiary company, PETRONAS NGV is currently offering the facility. From personal communication with NGV vehicle owners, the long decompression and fill time of CNG fuel usually causes an outstretched queue in refueling stations, much to the inconvenience of NGV owners.

Due to the gaseous form of CNG fuel, an accelerated wear of exhaust valves are also encountered due to the drying effect (Indian Energy Sector, 2000).

2.3 SAFETY

In addition to its excellent emission quality where toxic gases are reduced and pose a lower health hazard to the public, Barbotti CNG (2002) reported that the fuel cylinders used to store CNG in vehicles are designed to withstand impact through several tightly scrutinized tests. A survey conducted in the United States also showed a 37% decrease in injury rate for NGVs compared to gasoline-powered fleet vehicles and no fatality rates (Barbotti CNG, 2002).

The favorable physical property of CNG which enables it to dissipate into the air in case of a leakage and thus avoiding contamination is also a safety advantage. However, Graham et al. (2000) argued that CNG vapors formed at low temperatures from leaks will generate large clouds of flammable vapor and increase the potential of an explosion coupled with a spark.

Due to its high storage pressure at a range of 20-25 MPa, the refueling process is a safety issue since 0.2-0.3 kWh of energy per cubic meter of natural gas is required to compress it (Kojima, 2001). Kojima (2001) further exemplified a recent incident in India where five people were injured in a NGV during refueling due to inferior gas cylinder condition.

2.4 DIESEL ENGINE CONVERSION

2.4.1 Introduction to conventional diesel engine

Diesel engines are mostly used in heavy-duty applications and in fleet transportations due to its higher engine efficiency achieved through the higher compression ratio (CR). During operation, the compression ratios of diesel engines can reach up to 20:1, compared to compression ratios of 8:1 for spark ignition engines utilizing gasoline (Kojima, 2001).

Diesel engines in most heavy-duty vehicles are compression ignition engines, which predominantly operates under a cycle comprising of four strokes:

i. Induction stroke

   The inlet valve is opened and air is forced into the combustion chamber when the piston moves outwards through atmospheric pressure. As the piston reaches its bottom dead centre, the intake valve closes.

ii. Compression stroke

   The piston then moves inwards to compress the air present in the chamber. The air is heated to a temperature as high as 550°C (Shell Canada Limited, undated) which is above the flashing point of the diesel fuel. Just before the end of the stroke, fuel is injected into the heated and compressed air.

iii. Power stroke

   The diesel fuel is ignited and the pressure created pushes the piston outwards, providing power to the engine via the connecting rod and crankshaft.

iv. Exhaust stroke

   The piston moves outwards and burnt gases are pushed out of the combustion chamber through the exhaust valve. As the piston reaches the top dead centre position, the cycle is repeated.

For direct injection diesel engines where diesel fuel is directly injected into the combustion chamber at the end of the compression stroke, the combustion rates can be calculated by applying the first law of thermodynamics for the quasi-static (uniform pressure and temperature) control system (Heywood, 1988):

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \dot{m}_f h_f = \frac{dU}{dt} \qquad (2.1)$$

where

$\frac{dQ}{dt}$ = heat transfer rate across system boundary into the system

$p\frac{dV}{dt}$ = rate of work transfer done by the system due to boundary displacement

$\dot{m}_f$ = mass flow rate of diesel fuel into the system

h_f = sensible enthalpy of diesel duel

$\frac{dU}{dt}$ = rate of change of sensible internal energy of cylinder contents

For heat-release analysis, equation (2.2) applies:

$$\frac{dQ_n}{dt} = \frac{dQ_{ch}}{dt} - \frac{dQ_{ht}}{dt} = p\frac{dV}{dt} + \frac{dU}{dt} \qquad (2.2)$$

where

$\frac{dQ_n}{dt}$ = net heat-release rate

$\frac{dQ_{ch}}{dt}$ = gross heat-release rate

$\frac{dQ_{ht}}{dt}$ = heat-transfer rate to the walls

$p\frac{dV}{dt}$ = rate of work transfer done by the system due to boundary displacement

$\frac{dU}{dt}$ = rate of change of sensible internal energy of cylinder contents

If the contents of the cylinder are modeled to be an ideal gas, equation (2.2) becomes:

$$\frac{dQ_n}{dt} = p\frac{dV}{dt} + mc_v\frac{dT}{dt} \tag{2.3}$$

Using ideal gas law, $pV = mRT$, with R to be constant, it follows that:

$$\frac{dp}{p} + \frac{dV}{V} = \frac{dT}{T} \tag{2.4}$$

Substitution of Equation (2.4) into (2.3) can be used to eliminate T:

$$\frac{dQ_n}{dt} = \left(1 + \frac{c_v}{R}\right)p\frac{dV}{dt} + \frac{c_v}{R}V\frac{dp}{dt}$$

or

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1}p\frac{dV}{dt} + \frac{1}{\gamma - 1}V\frac{dp}{dt} \tag{2.5}$$

where γ = ratio of specific heats

$$= \frac{c_p}{c_v}$$

The range for γ for diesel-heat release is usually 1.3 to 1.35 for analysis.

2.4.2 Engine conversion types

2.4.2.1 Bi-fuel engine

A Bi-fuel engine utilizes two fuel systems, usually consisting NGV and gasoline. In general, spark-ignition engines are easily converted into bi-fuel engines by retrofitting a NGV kit to the engine system (Autoworld, 2004). A fuel selector allows the user to choose which fuel to use.

According to Equitable Gas (undated), when natural gas is selected as the running fuel, the compressed gas is passed through the master manual shut-off valve and enters the engine compartment. A regulator in the engine compartment reduces the gas pressure to 1 bar before the gas passes into the fuel-injection system through a solenoid valve. When gasoline is selected as fuel, the natural gas system is shut off to avoid any mixture of fuel.

Murray et al. (2000) reported that vehicles using the bi-fuel engine suffer from power loss of around 10-15% when natural gas is used during wide open throttle.

2.4.2.2 Dedicated NGV engine

Dedicated NGV engines require more modification compared to a vehicle operating with bi-fuel. Most of the components of a diesel engine need to be replaced as NGV use a spark ignition engine to ignite its fuel. The gas supply system and ignition system need to be changed and an electronic control unit (ECU) fitted to control the operations for a dedicated NGV engine.

Autoworld (2004) outlined that the number of manufactured dedicated NGV in Malaysia is still very low compared to retrofitted bi-fuel engines. According to Alternative Fuels Data Centre (2004), dedicated NGV show better performance and superior emissions since the engine system is optimized to run solely on natural gas. Dedicated NGV only need to carry a single fuel load if compared with other types of engines and this weight reduction increases the fuel efficiency.

2.4.2.3 Dual-fuel engine

In dual-fuel engines, natural gas and diesel fuel is used simultaneously in the combustion chamber to produce power. Approximately 80% of natural gas is consumed while the remainder comes from diesel which acts as a pilot fuel to ignite the gas in the combustion chamber (Clean Air Power, undated).

To convert existing diesel engines to run on dual-fuel, an electronic control unit (ECU) needs to be installed. The function of the ECU is to control engine speeds while monitoring engine temperature and pressure by incorporating electronic and mechanical sensors to ensure safe operation of the dual-fuel system. Clean Air Power (undated) added that, in Caterpillar engines which are common in most heavy duty applications, the ECU installed will communicate with the Caterpillar Advanced Diesel Management System (ADEM) to determine the quantity and timing of diesel pilot fuel according to the engine's RPM signal. Apart from that, a gas mixer needs to be fitted to the air manifold to allow complete mixture between air and natural gas (Hybrid Fuel Systems Inc., 2004). In the Garretson fuel system, the venturi principle is used to obtain the proper gas/air mixture with a sensitive and properly calibrated fuel controller (Alternate Fuels Technologies Inc., undated).

A rack limiter is also installed to monitor the engine's load and speed so that the accurate amount of pilot diesel fuel can be supplied. Sensors and solenoids are added together with diesel and natural gas injectors so that the injectors are controlled by the ECU through pulse width modulated signals to maximize efficiency.

The pistons and cylinder heads of the combustion chamber are also modified to allow proper natural gas and air mixture so that the high compression ratio of the original diesel engine can be maintained with dual-fuel. The cylinder head is optimized to allow both diesel and natural gas injectors to operate for a standard cycle.

Dual-fuel engines have the advantage of providing the same power as a conventional diesel engine since it retains the high compression ratio and produces lower amounts of emissions such as NO_x and particulate matters. Hybrid Fuel Systems Inc. (2004) further stated that dual-fuel NGVs have better fuel economy and lower maintenance costs compared to a dedicated NGV engine.

CHAPTER 3

ENGINE PERFORMANCE

3.1 Torque

An engine's torque is a measure of its rotational force exerted to transmit power from the engine to the wheels of the vehicle through the drive train. The torque and power produced by an engine can be measured using a dynamometer which is mounted to the engine as a separate component. Torque can be improved by addition of engine cylinders or increasing the capacity of the engine although an increase in fuel consumption would be significant. The product of torque and angular speed gives the power developed by the engine:

$$P = \frac{2\pi N\tau}{60} \tag{3.1}$$

Or

$$\tau = \frac{60P}{2\pi N} \tag{3.2}$$

where

τ	=	torque (N. m)
P	=	power developed by engine (W)
N	=	engine speed (rpm)

3.2 Input Power

The input power of the engine refers to the maximum rate at which energy is supplied to the engine. It corresponds with the indicated power calculated from a *p-V* diagram based on the work done during compression and expansion process of the diesel cycle, less the heat loss to exhaust and coolants. The heat of combustion of fuel is supplied to the engine and assuming the cycle efficiency as unity where all the chemical energy of the fuel is converted into useful work, the engine input power is given by

$$IP = \dot{m}_f \times Q_{HV} \times 10^3 \qquad (3.3)$$

where

IP	=	input power (kW)
$\dot{m}_f$	=	mass flow rate of fuel (kg/s)
Q_{HV}	=	lower calorific value of fuel (MJ/kg)

For diesel fuel, $Q_{HV,Diesel}$ = 42.5 MJ/kg

For CNG fuel, $Q_{HV,CNG}$ = 45 MJ/kg

The lower calorific value of fuel is used in (3.3) since all water compounds in the fuel are assumed to be in vapour phase without any condensation.

3.3 Brake Power

The brake power is the power output delivered by the engine shaft. It is less than the indicated power since heat is lost to overcome the total friction generated in the engine which is summed as friction power. Friction power consists of pumping friction during intake and exhaust, mechanical friction in bearings, valves and components such as oil and water pumps. Brake power refers to the rate at which work is done and shows a maximum value when engine speed is increased close to maximum before decreasing since friction becomes very significant at high engine speeds.

$$\text{Brake power} = \text{Indicated power} - \text{Friction power} \tag{3.4}$$

In a diesel engine, the brake power can be varied by changing the fueling rate or air-fuel ratio to produce the desired power for an application. In the experimentation, brake power is obtained from:

$$BP = \frac{2\pi N\tau}{60\times10^3} \tag{3.5}$$

where

BP	=	brake power	(kW)
N	=	engine speed	(rpm)
τ	=	torque	(N. m)

3.4 Specific Fuel Consumption

Specific fuel consumption is the measure of fuel flow rate per unit power output and relates to the fuel efficiency of an engine. It is inversely proportional to efficiency of the engine as lower values of specific fuel consumption are favorable for higher performance. Specific fuel consumption is defined as:

$$sfc = \frac{\dot{m}_f}{P} \times 3.6 \times 10^6 \quad (3.6)$$

where

sfc	=	specific fuel consumption	(g/kW. hr)
$\dot{m}_f$	=	mass flow rate of fuel	(kg/s)
P	=	power output	(kW)

In the performance measurement and comparison between engines running on diesel and dual-fuel for the experimentation, the power output measured is the brake power. Therefore, brake specific fuel consumption is:

$$bsfc = \frac{\dot{m}_f}{BP} \times 3.6 \times 10^6 \quad (3.7)$$

where

bsfc	=	brake specific fuel consumption	(g/kW. hr)
$\dot{m}_f$	=	mass flow rate of fuel	(kg/s)
BP	=	brake power	(kW)

The brake specific fuel consumption varies with the compression ratio and fuel equivalence ratio. A higher compression ratio will produce lower bfsc since more power can be extracted from the burning fuel. Bsfc decreases as engine size becomes progressively smaller since heat losses from the combustion gas to the cylinder wall are reduced. Generally, compression ignition engines with diesel fuel produce a higher amount of energy per unit fuel compared to spark ignition engines.

3.5 Brake Mean Effective Pressure

The brake mean effective pressure is a useful measure of the relative performance of an engine. It refers to the mean pressure to be maintained in the pistons of the cylinder to produce a power output during each power stroke. The brake mean effective pressure can be calculated from the torque and is defined as:

$$bmep = \frac{BP \times n_R \times 60}{A \times L \times N \times n} \tag{3.8}$$

where

bmep	=	brake mean effective pressure (kPa)
BP	=	brake power (kW)
n_R	=	number of crank revolutions for each power stroke per cylinder (one for two-stroke cycle and two for four-stroke cycle)
A	=	area of engine bore (m^2)
L	=	length of engine stroke (m)
N	=	engine speed (rpm)
n	=	number of cylinders

It indicates the work done per cycle for every unit of cylinder volume displaced and is the direct measure of brake torque, not engine power. A higher bmep corresponds to a higher engine output since more pressure is transmitted through the connecting rods to the crankshaft. However, engine wear increases with increasing bmep and leads to high mechanical stresses on engine components and imposing high thermal stresses on combustion chambers.

The maximum value of bmep for a compression ignition engine is obtained at the engine speed where maximum torque is obtained. The bmep of the same engine is measured to be slightly lower at maximum rated power.

3.6 Engine Thermal Efficiency

Generally, an IC engine loses almost 42% of its energy to the exhaust system and a further 28% to the cooling system (The Concept IC Engine). The engine thermal efficiency refers to the ratio of work produced per cycle to the amount of fuel input to the engine per cycle.

$$\eta_f = \frac{W_c}{m_f Q_{HV}} = \frac{\dot{W}_c}{\dot{m}_f Q_{HV}} = \frac{P}{\dot{m}_f Q_{HV}} \qquad (3.9)$$

where

η_f	=	engine thermal efficiency	
P	=	output power produced per cycle	(kW)
$\dot{m}_f$	=	mass flow rate of fuel per cycle	(kg/s)
Q_{HV}	=	lower calorific value of fuel	(MJ/kg)

In the experimentation, the desired output power per cycle is the brake power. Therefore, incorporating equation (3.3) into (3.9),

$$\eta_f = \frac{BP}{IP} \qquad (3.10)$$

Another method of obtaining the engine thermal efficiency is by utilization of the specific fuel consumption. Substituting equation (3.6) into (3.9) gives:

$$\eta_f = \frac{3600}{sfc \times Q_{HV}} \tag{3.11}$$

where

sfc	=	specific fuel consumption	(g/kW. hr)
Q_{HV}	=	lower calorific value of fuel	(MJ/kg)

We can see that the specific fuel consumption is inversely proportional to the total engine thermal efficiency, as mentioned earlier.

CHAPTER 4

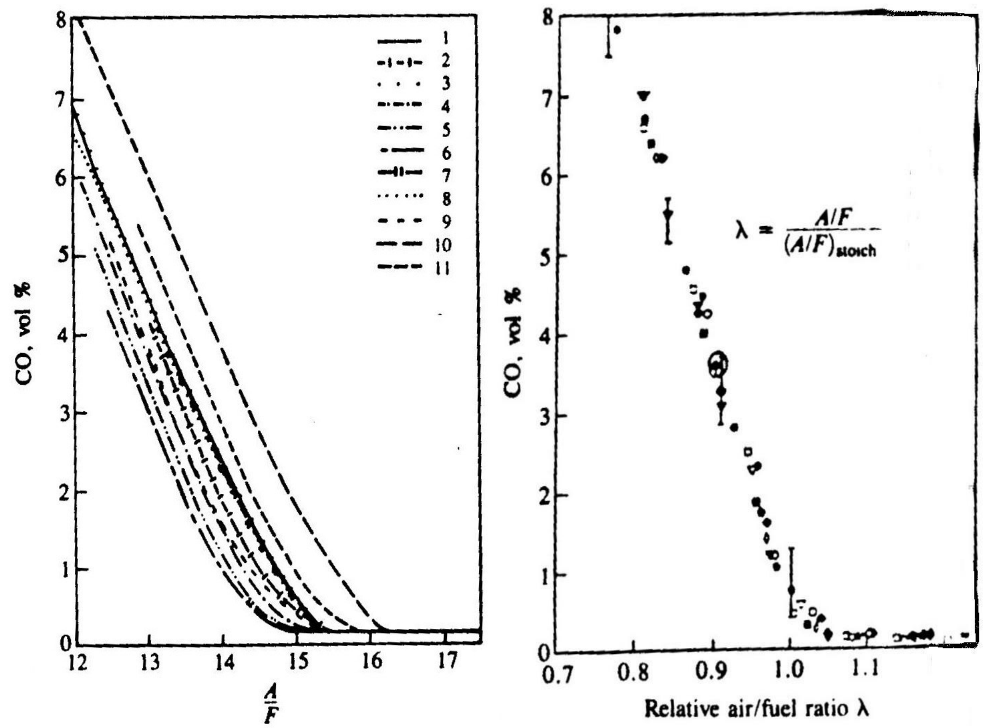
EMISSION

4.1 Carbon Monoxide (CO)

Carbon monoxide (CO) is a colourless, odorless, flammable and highly poisonous gas which is less dense than air. Inhalation of carbon monoxide can be fatal to humans since a small concentration as little as 0.1% will cause toxication in the blood due to its high affinity to oxygen carrying hemoglobins. Exposure levels must be kept below 30 ppm to ensure safety (Environmental Centre, undated). Apart from that, carbon monoxide also helps in the formation of greenhouse gases and global warming by encouraging the formation of NO_x.

Carbon monoxide forms in internal combustion engines as a result of incomplete combustion when a carbon based fuel undergoes combustion with insufficient air. The carbon fuel is not oxidized completely to form carbon dioxide and water. This effect is obvious in cold weathers or when an engine is first started since more fuel is needed.

Carbon monoxide emission from internal combustion engines depend primarily on the fuel/air equivalence ratio (λ). Figure 4.1(a) shows the variation of CO emission for eleven fuels with different hydrocarbon contents. A single curve may be used to represent the data when using the relative air/fuel or equivalence ratio as represented in Figure 4.1(b).



(a) (b)

Figure 4.1: *Variation of CO emission for different fuels (a) with air/fuel ratio; (b) with relative air/fuel ratio (Heywood, 1988)*

Both the graphs clearly show that the amount of CO emitted increases with decreasing air to fuel ratio. Spark ignition gasoline engines which normally run on a stoichiometric mixture at normal loads and fuel-rich mixtures at full load shows significant CO emissions. On the other hand, diesel engines which run on a lean mixture only emit a very small amount of CO which can be ignored. Ferguson (1986) researched that additional CO may be produced in lean-running engines through the flame-fuel interaction with cylinder walls, oil films and deposits. Direct-injection diesel engines also emit more CO than indirect-injection engines. However, the CO gas emission increases with increasing engine power output for both engines.

CO formed from hydrocarbon radicals can be oxidized to form carbon dioxide in an oxidation reaction, in an equilibrium condition (Turns, 1996):

$$CO + OH = CO_2 + H \tag{4.1}$$

The emission of CO is a kinetically-controlled reaction since the measured emission level is higher than equilibrium condition for the exhaust. Three-body radical recombination reactions such as

$$H + H + M = H_2 + M \tag{4.2}$$

$$H + OH + M = H_2O + M \tag{4.3}$$

$$H + O_2 + M = HO_2 + M \tag{4.4}$$

are found to be rate-controlling reactions for emission of CO gas.

Reduction of carbon monoxide in internal combustion engines can be achieved by improving the efficiency of combustion process or utilization of oxidation catalysts to oxidize carbon monoxide to carbon dioxide. Engine modifications such as improved cylinder head design, controlled air intake and electronic fuel injection can help to maintain a lean air/fuel mixture which is favorable.

4.2 Total Hydrocarbon (THC)

Total hydrocarbon (THC) is used to measure the level of formation of unburnt hydrocarbons caused by incomplete combustion in the engine. The hydrocarbons emitted may be inert such as methane gas or reactive to the environment by playing a major role in the formation of smog. The types hydrocarbons emitted from the exhaust greatly depend on the type and composition of fuel used. Heywood (1988) added that fuels with a greater concentration of aromatics and olefins compounds will result in a higher percentage of reactive hydrocarbons.

In diesel engines, hydrocarbon emission can be significant under two normal operating conditions due to the complex nature of fuel-air and burned-unburned gas mixing in the combustion chamber. Firstly, a mixture of fuel leaner than the lean combustion limit in the chamber during the ignition delay period will cause incomplete combustion and hence formation of unburnt hydrocarbons. The locally overlean mixture of fuel will not autoignite or support a propagating flame, causing a slow reaction to develop.

On the other hand, undermixing of fuel which occurs when the fuel mixture is too rich to ignite or support a flame causes hydrocarbon formation during the combustion cycle. The injector sac volume provides an important contribution to the hydrocarbon emission in direct-injection engines. The diesel fuel left at the tip of the injector enters the cylinder at low velocity and does not have enough time to achieve a standard mixture with air. Experimental results from Heywood (1988) showed that the amount of hydrocarbon emission is proportional to the injector sac volume. Generally, hydrocarbon emission in diesel engines are higher during engine idling and at light loads when the amount of excess air is great (Ferguson, 1986).

Suitable diesel catalysts can be used to oxidize hydrocarbons to carbon dioxide and water molecules. According to Nett Technologies Inc. (undated), hydrocarbon traps are also used to capture hydrocarbon emissions especially at low temperatures when the oxidation catalysts are not functional during engine idling times.

4.3 Oxides of Nitrogen (NO_x)

Nitrogen oxides consist primarily of nitric oxide (NO) and nitrogen dioxide (NO_2) as a product of oxidation of atmospheric nitrogen in the combustion chamber. Diesel fuel contains a significant amount of nitrogen compounds and acts as an additional source of NO. Formations of nitric oxides from molecular nitrogen are described by the following equations (Heywood, 1988):

$$O + N_2 \leftrightarrow NO + N \tag{4.5}$$

$$N + O_2 \leftrightarrow NO + O \quad (4.6)$$

$$N + OH \leftrightarrow NO + H \quad (4.7)$$

Nitric oxides (NO) formed in the combustion chamber can be rapidly oxidized to form NO_2 through the following reaction:

$$NO + H_2O \rightarrow NO_2 + OH \quad (4.8)$$

At the same time, NO_2 will be subsequently converted back to NO via:

$$NO_2 + O \rightarrow NO + O_2 \quad (4.9)$$

A considerable amount of NO_2 is found in diesel engines especially during light loads or engine idling times. At lower temperatures, the transformation of NO_2 back to NO in reaction (4.9) is quenched by the cooler regions of the chamber and the ratio of NO_2 to NO can go as high as 30% (Heywood, 1988).

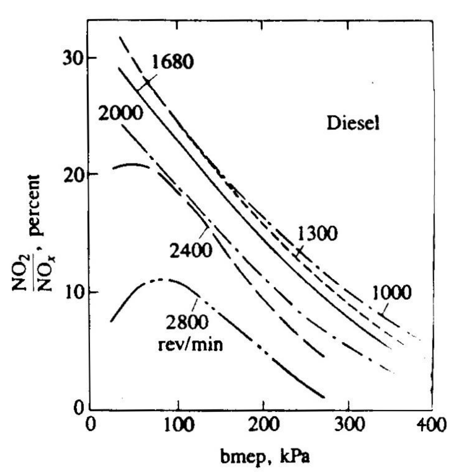


Figure 4.2: *Ratio of NO_2 to NO in diesel exhaust for varying engine load and speed (Heywood, 1988)*

Figure 4.2 clearly shows that the maximum amount of NO_2 is emitted in a diesel engine at low engine speed and minimum loads. This can be damaging to the atmosphere as NO_2 formed will contribute to formation of ground-level ozone or smog and reacts in the air to form corrosive nitric acid.

At the same engine speed, Ferguson (1986) reported that the amount of NO_x produced increases with engine load for a direct-injection engine and the maximum amount occurs when the fuel mixture is slightly lean. With higher loads, the peak pressures of the cylinder and temperature distribution are higher and coupled with enhanced mixing of the diesel fuel, NO_x levels are increased. As a rule of thumb, the emission of NO are roughly proportional to the mass of fuel injected.

Selective catalytic reduction (SCR) can be used to convert NO_x emitted to form oxygen and nitrogen through the use of reducing agents such as ammonia or urea. It is combined with an oxidation catalyst to oxidize any traces of ammonia which may escape the system into the atmosphere.

4.4 Particulate Matters (PM)

Particulates are defined as a complex aggregate of solid and liquid material other than water that can be collected in an exhaust filter. Diesel particulate matters (DPM) are generally divided into three categories which include dry carbon particles or soot, Soluble Organic Fractions (SOF) resulting from incomplete combustion, adsorption and condensation of heavy hydrocarbon onto carbon particles, and sulfate fraction (SO_4).

In addition to these elements, small amounts of zinc, sulphur, phosphorus, calcium, iron, chromium and silicon are found in particulates. The organic fractions of particulates are serious hazards to the health of the public and environment. The primary carbon particles have a nuclei size of around 0.04 to 1 µm which have adverse health effects when respirated.

Generally, the density of PM depends on the engine load, speed and exhaust conditions. Heywood (1988) mentioned that the highest PM concentration occurs at the core region of diesel fuel spray in direct-injection engines when the local equivalence ratio is very rich. Soot concentration decreases with increasing distance from the centerline.

A compromise must be considered when designing ways to reduce the amount of PM formation during combustion process since a higher combustion level and increased in-cylinder temperatures will result in a higher concentration of NO_x formation. A trap oxidizer is used to filter diesel particulate matters in the exhaust as a method to reduce emission after the combustion process. The trapped particulates are oxidized to remove them and refresh the filter. Catalysts are used to improve the filter efficiency by increasing the regeneration capacity of the filters.

4.5 Carbon Dioxide

Carbon dioxide emissions in diesel engines are products of direct combustions or by-product of oxidizing other unwanted emission gases with the aid of catalysts. Although diesel engines generally produce low amounts of CO_2 compared to other emission gases, the emission of carbon dioxide must be regulated and controlled to reduce negative impacts on the environment such as accumulation of greenhouse gases and global warming.

4.6 Oxides of Sulphur (SO_x)

Conventional diesel fuel contains sulphur compounds in its composition and results in the emission of sulphur oxides in the form of SO_2 and SO_3 as the products of combustion. According to Turns (1996), the average content of sulphur compounds in diesel fuel lies within the range of 0.1 to 0.8% which is very high compared to gasoline. Sulphur trioxide (SO_3) reacts readily with water to form sulfuric acid which accumulates in the exhaust system. On the other hand, sulphur dioxide (SO_2) will be oxidized in the atmosphere to form SO_3 before reacting to form sulfuric acid. Apart from assisting in the corrosion of exhaust metal, the sulfuric acid present destroys the effectiveness of catalysts used in the exhaust system to reduce other emission pollutants.

The smog produced from sulphur content in diesel fuel generally cause environmental and health hazards, and has been liked with respiratory diseases and illnesses. The sulphur content in the exhaust can be reduced by using low sulphur diesel fuels which reduces soot emission without affecting the engine performance. The oxides of sulphur present in the exhaust can be reduced by using limestone ($CaCO_3$) or lime (CaO) to produce calcium precipitates and carbon dioxide according to the following equations:

For limestone,

$$CaCO_3 + SO_2 + 2H_2O \rightarrow CaSO_3 \cdot 2H_2O + CO_2 \tag{4.10}$$

For lime,

$$CaO + SO_2 + 2H_2O \rightarrow CaSO_3 \cdot 2H_2O \tag{4.11}$$

According to Ly (2002), the sulphur content of natural gas is much lower than ultralow sulphur diesel levels of 10-50 ppm. Natural gas also does not contain toxic benzene or 1,3-butadiene compounds. This greatly reduces any oxides of sulphur from being produced in the combustion process and eliminates the irritating odor of sulfuric gases.

CHAPTER 5

EXPERIMENTAL SETUP

5.1 Engine Preparation

A four stroke, single cylinder air cooled diesel engine is used for the experiment and the full engine specifications are provided in Appendix A. The engine speed runs up to 3600 rpm. The engine is attached at one end to the electrical heating element dynamometer with a drive shaft coupling flange. A throttle is used to control and increase the speed of the engine as the control variable. The dynamometer and engine is cooled by an attached cooling tower with cooling fan and heat sump to dispose generated heat. The other end of the dynamometer is hooked up to the digital readout system which contains the digital RPM meter, flowmeter, oil sump temperature and torque meter so that experimental readings can be obtained.

5.2 Preparation of Load Banks

The load bank for the engine which consists of electrical heaters is installed underneath the digital readout system and connected to the engine. The load bank is rated at 320 V and is used to provide a 1 kW load increment so that the response of the diesel engine can be measured. The electrical heaters are immersed in water so that heat power output from the engine is used to increase the temperature of the water and care must be taken so that no water spills occur which may short-circuit the wiring system.

5.3 Calibration of Digital Thermocouple

A set of digital thermocouple is used to obtain the temperatures of the engine both at the body and exhaust. The probe of the thermocouple is inserted into the exhaust of the engine and on the surface of the engine body respectively to measure the steady-state temperature when the engine runs in normal operating conditions for different values of loads. Since temperatures fluctuate slightly, a range of readings are taken over a period of time before finalizing the mean value.

5.4 Gas Analyzer Setup

The emission gas levels are tested and measured using the Autologic Autogas Emission Analyzer as shown in Figure 5.1. A portable 5 Gas Emission Analyzer with PC software unit is used for this experiment to measure levels of HC, CO, O_2, CO_2, and NO_x. Additional units for measurements of RPM, Oil temperature and Diesel Smoke Meters can be combined to the main gas analyzer unit. The analyzer performance exceeds standards such as the ASM/BAR 97, OIML, and BAR 90.

Other accessories for the gas analyzer includes a 25 foot sampling hose attached to a sensor probe, durable high strength aluminium casing for protection of all the connections, automatic water removal system so that water vapors that may get trapped will not interfere with the results, and a compatible software for all version of Windows operating system to allow data to be recorded. The full range of specifications of the gas analyzer is listed in Appendix C.

The typical warm-up time for the gas analyzer is 2 minutes and it possesses a great level of accuracy. Initially, the gas analyzer settings are configured and reset to zero for both diesel and dual-fuel so that consistent results are obtained. The probe of the gas analyzer is inserted into the exhaust duct of the engine and all connections are tightened before measurements are taken. The first filter unit for the gas analyzer is cleaned after every test run for standardization of results.

Fig 5.1: *Autologic Autogas Gas Analyzer* **(Autologic Company)**

5.5 Natural Gas Conversion Kit

A natural gas conversion kit system is available in the laboratory to enable compressed natural gas to be used as fuel in the test engine. The conversion kit is of model TMB Tartarini Natural Gas Conversion Kit and consists of components such as the gas pressure regulator, solenoid valve, air mixer, fuel select switch and gauges as shown in Figure 5.2.

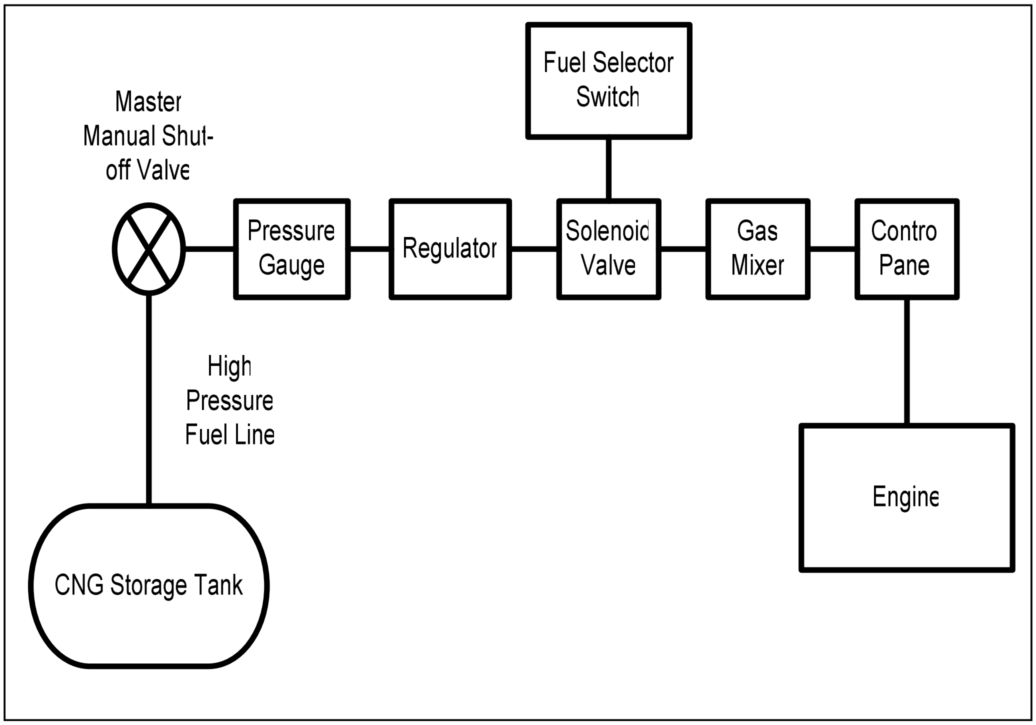


Fig 5.2: *Natural Gas Converter Kit*

5.5.1 Natural Gas Pressure Regulator

The pressure regulator decreases the CNG pressure to near atmospheric pressure from 20MPa in the storage tank to allow natural gas to flow into the gas mixer. Apart from that, the regulator also acts as a control to modulate the flow of natural gas to the gas mixer.

5.5.2 Natural Gas Solenoid Valve

The solenoid valve is used as a main control switch to allow natural gas to flow from the pressure regulator to the gas mixer and engine. The solenoid valve is electronic timer controlled and contains a built-in gas filter and attached pressure gauge. It also acts as an emergency shut off device to stop the flow of natural gas into the system when a leak is detected or when other devices malfunction. Apart from that, the solenoid valve improves ignition during cold temperature start up of the engine.

5.5.3 Natural Gas Mixer

Natural gas is mixed with air in the gas mixer to obtain the optimum ratio for combustion before being transferred into the combustion chamber of the engine through the control panel which measures the flow rate of the mixture.

5.5.4 Fuel Selector Switch and Gauge

A fuel selector switch is used for the user to switch between diesel and dual-fuel system in the experiment. The amount of natural gas in the system can also be monitored through the attached measurement gauge.

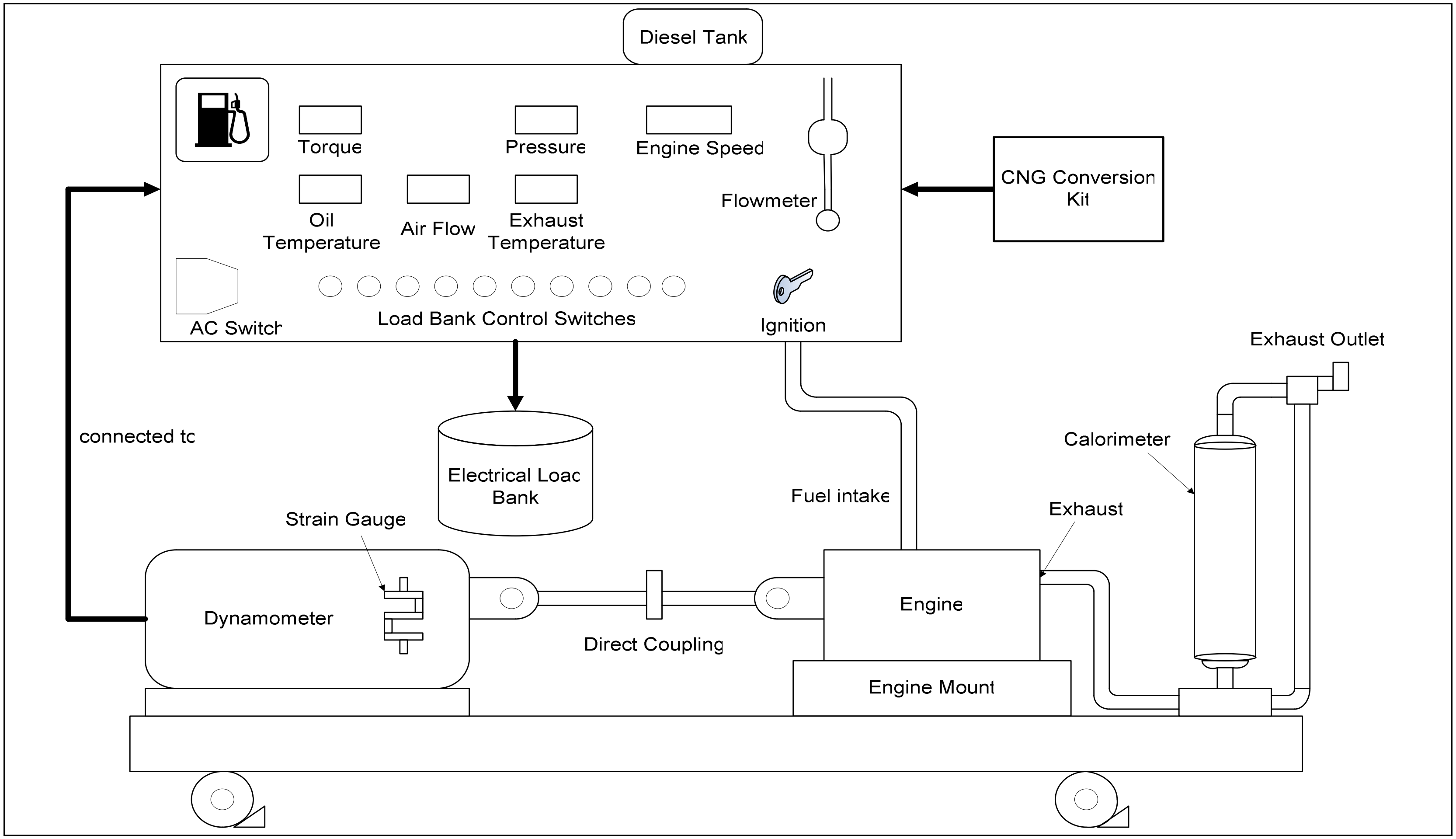


Fig 5.3: *Experimental Setup Overview*

5.6 Experimental Procedures

Figure 5.3 shows the experimental setup to perform the test on diesel and dual-fuel operating conditions. The engine is connected to the dynamometer through a direct coupling. The dynamometer is then connected to the main control board which controls the intensity of the electrical loads applied to the engine.

The effect of diesel and dual-fuel natural gas with pilot diesel fuel on the engine performance and emission levels are to be investigated. Initially, diesel fuel is used and the engine is started and no loads are applied for five minutes to allow the engine to reach a steady-state operating condition. Next, the throttle of the engine is adjusted so that the engine speed is maintained at 1600 RPM. The measurements for torque, volume flow rate, body and exhaust temperature of the engine are recorded when the values reach a steady-state condition. The volume flow rate is calculated by recording the time taken for the engine to use 10ml of fuel in the flowmeter attached to the control panel. After calibration, the gas analyzer probe is inserted into the exhaust duct of the engine and measurements of the levels of emission gases are recorded. The probe is removed from the exhaust duct after measurements are taken and cleaned.

After that, the electrical load bank is applied to the engine at 1kW using the selector switches on the control panel. The same readings are taken from the digital readout system and gas analyzer. The engine is loaded with additional 1kW loads progressively and measurements are recorded at the same engine speed until maximum load occurs, in which the engine fails to support the applied load and stalls.

The above procedure is repeated for engine speeds of 1800, 2000, 2200, and 2600 RPM so that effective comparisons can be made. For each speed, the engine is loaded until the maximum condition is reached. Before the performance and emission test for dual-fuel is conducted, the engine is allowed to cool to room temperature to obtain standardized results.

For the dual-fuel experiment, diesel pilot fuel is supplied to warm the engine for five minutes. After that, the master shut-off valve of CNG storage tank is opened to allow natural gas to travel to the gas regulator through the high pressure fuel line. Natural gas is then supplied to the engine through the on-board control panel and the mass flow rate of natural gas is recorded using a flow meter. The amount of diesel pilot fuel is kept constant while the engine speed is controlled by increasing the flow rate of natural gas to the engine until the specified engine speeds are obtained.

All experimental data for engine speeds of 1600, 1800, 2000, 2200, and 2600 rpm are recorded and tabulated.

CHAPTER 6

RESULTS AND DISCUSSION

6.1 Introduction

The results from the experiments performed on the four-stroke engine for maximum load operating conditions are shown below in graph form and discussed. The results are comparable with Talal et al. (2003). For engine performance, the graphs of engine body temperature, exhaust gas temperature, engine torque, brake power, brake specific fuel consumption (bsfc) and engine thermal efficiency against varying engine speeds from 1600 rpm to 2600 rpm are plotted.

On the other hand, the graphs of emission of carbon monoxide (CO), hydrocarbon (HC), oxides of nitrogen (NO_x), carbon dioxide (CO_2), and excess oxygen (O_2) against the same range of engine speed are shown.

6.2 Engine Performance for Maximum Load Operating Conditions

The experimental data was taken at an ambient temperature of 29°C after the engine was started for five minutes to achieve a stable operating condition. Electrical loads of 1KW each were progressively added and the results below show the comparison graphs for both conventional diesel fuel and dual-fuel readings and their respective explanations under maximum loading conditions for the respective engine speeds. The body and exhausts temperatures were measured with a thermocouple when the temperatures become stable.

6.2.1 **Engine Body Temperature**

Figure 6.1 shows the measured engine body temperatures for both diesel and dual-fuel against engine speeds for maximum load operating conditions. The engine body temperature for diesel fuel is typically higher for all the engine speeds tested compared to dual-fuel.

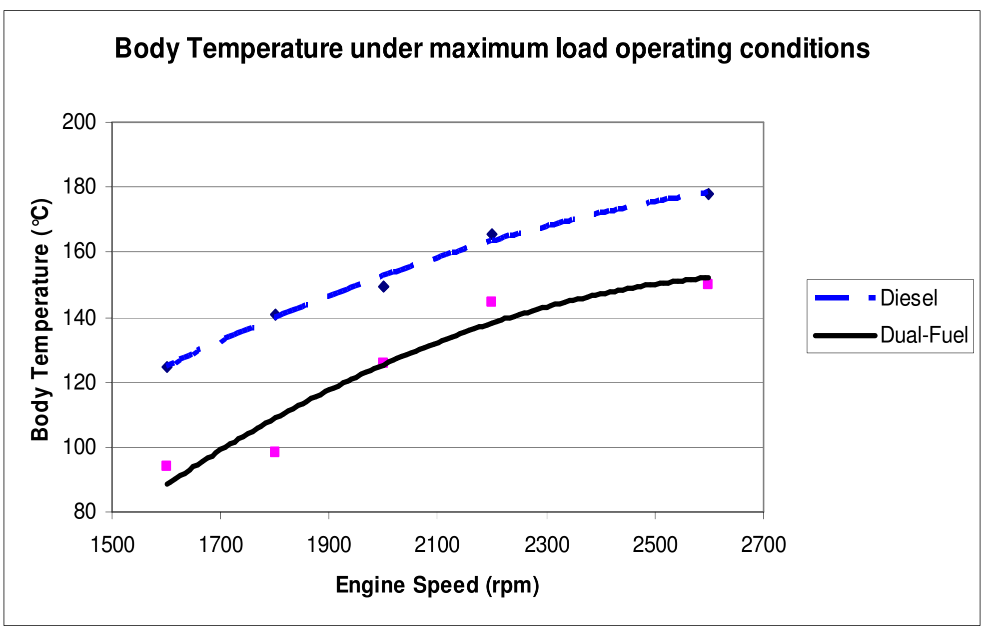


Figure 6.1: *Engine body temperatures under maximum load operating conditions*

The maximum difference of temperature occurs at an engine speed of 1800 rpm where the body temperature for diesel fuel is approximately 42.4°C higher than the corresponding temperature recorded for dual-fuel under the same loading conditions and engine speed. On average, the engine body temperature of diesel is 29.0°C higher than dual-fuel.

It is observed from Figure 6.1 that an increase in engine speed from 1600 rpm to 2600 rpm will effectively increase the body temperature as well. The amount of fuel injected into the combustion chamber is increased when engine speed is increased by adjusting the throttle of the engine. As more fuel enters the chamber, combustion takes place at a higher temperature resulting in an increase in measured engine body temperature.

The low temperatures measured for dual-fuel body temperature compared to diesel indicates a higher level of oxygen concentration for combustion since lower temperatures will cause an increase in density of air. With more oxygen gas present, dual-fuel can undergo a better combustion and produce higher engine efficiency.

6.2.2 **Exhaust Gas Temperature**

The exhaust gas temperatures measured with a thermocouple for both diesel and dual-fuel are plotted in Figure 6.2. Similar to the engine body temperature, the exhaust temperature follows a similar trend where the temperature increases as the engine speed is increased. The exhaust gas temperature of diesel fuel is also higher than dual-fuel, with an average difference of 20.3°C. The measured values also shows that the temperature difference between diesel and dual-fuel is almost constant at approximately 7% higher for diesel fuel for engine speeds ranging from 1600 rpm to 2600 rpm.

The high exhaust temperature of diesel engines compared to dual-fuel will result in unfavorably higher NO_x output. The higher exhaust temperature for diesel fuel operation also indicates that combustion of dual-fuel is leaner than diesel since less heat is usually produced during lean combustion in a compression ignition engine.

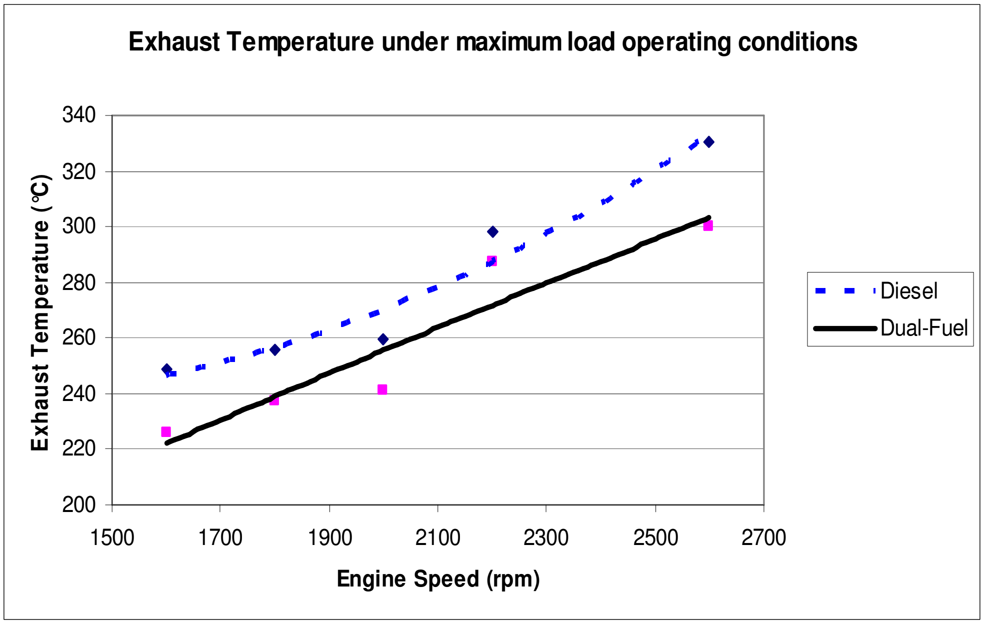


Figure 6.2: *Engine exhaust temperatures under maximum load operating conditions*

6.2.3 **Engine Torque**

Figure 6.3 shows the measured engine torque for diesel and dual-fuel operations. The torque increases for engine speeds ranging from 1600 rpm to 2100 rpm to a peak before dropping in magnitude for higher speeds up to 2600 rpm for both diesel and dual-fuel. It is observed from Figure 6.3 that the measured value from the dynamometer shows a higher torque for dual-fuel operation for all engine speeds. Analytically, the average percentage increment using dual-fuel is 12.4% compared to diesel fuel with the latter displaying a decrease of 1 Nm torque at all the engine speeds taken.

As engine speed is increased from 1600 rpm to 2100 rpm, the measured torque for diesel and dual-fuel increases indicating a higher fuel flow rate into the combustion chamber and an increased energy input available to the engine. The combustion efficiency must also be taken into account when measuring the output torque generated by the engine.

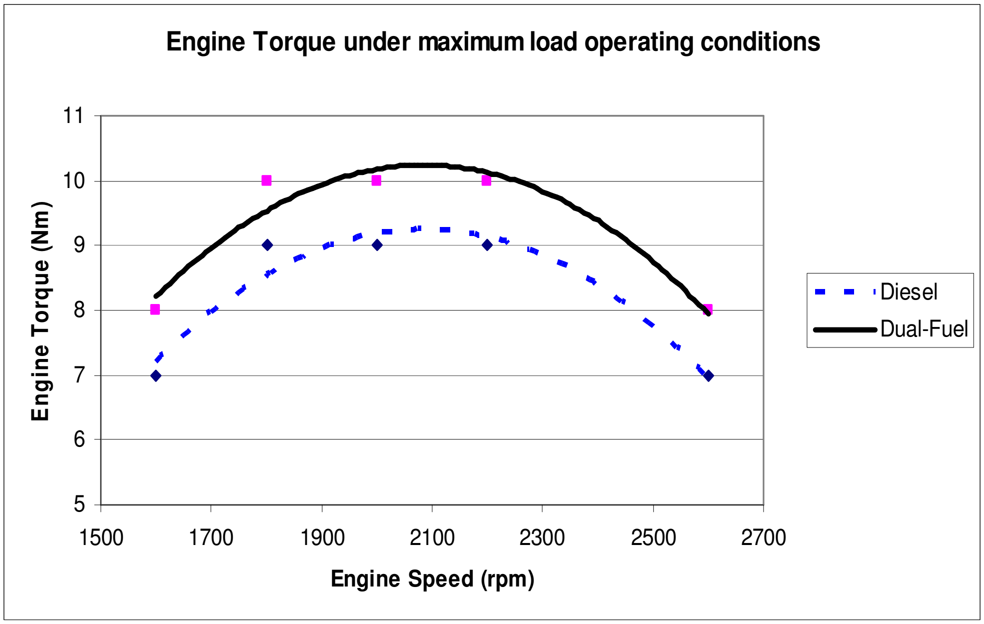


Figure 6.3: *Engine torque output under maximum load operating conditions*

The engine output torque drops tremendously for engine speed of 2600 rpm due to engine knocking. The incorrect ignition timing which results in engine knocks results in uncontrolled combustion and vibrations which will decrease the overall torque output available. The high pressure in the combustion chamber due to the compression stroke will cause CNG to self-ignite to the design temperature and pressure with high compression ratio. In addition, as the engine speed increases, the relative air to fuel ratio decreases causing more fuel to be injected into the system. Accumulation of excess fuel in a high pressured combustion chamber with high temperatures favors engine knocking, especially at very high loading conditions.

6.2.4 **Engine Brake Power**

From Figure 6.4, the engine brake power produced for dual-fuel operation is higher than diesel fuel operation with a maximum value difference of 0.272 kW recorded at the engine speed of 2600 rpm. In average, the dual-fuel engine produces a higher engine brake power by as much as 0.213 kW compared to conventional diesel fuel.

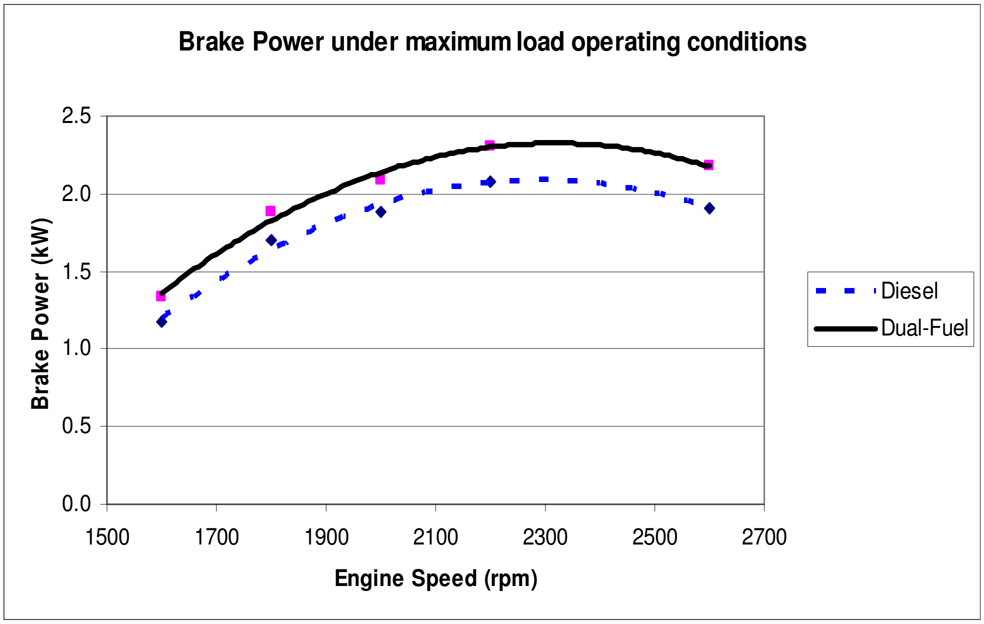


Figure 6.4: *Engine brake power under maximum load operating conditions*

As explained earlier, the engine brake power produced decreases slightly due to engine knocks at the speed of 2600 rpm for both diesel and dual-fuel. The dual-fuel operation achieved a maximum brake power output of 2.304 kW at speed of 2200 rpm, which is almost 83% of the rated engine output power of 2.8 kW.

6.2.5 **Brake Specific Fuel Consumption**

Figure 6.5 shows the comparison of brake specific fuel consumption under maximum load conditions for diesel and dual-fuel operations. The fuel consumption per unit power output for dual-fuel is lower than diesel fuel with the maximum occurring at 2600 rpm with 17.1% reduction. On an average basis, dual-fuel operation reduces the brake specific fuel consumption by 15.6% for all the engine speeds tested. This phenomenon can be attributed to the chemical properties of natural gas where the higher octane value of CNG compared to diesel decreases the amount of fuel required for combustion to drive the engine to support the same amount of loading.

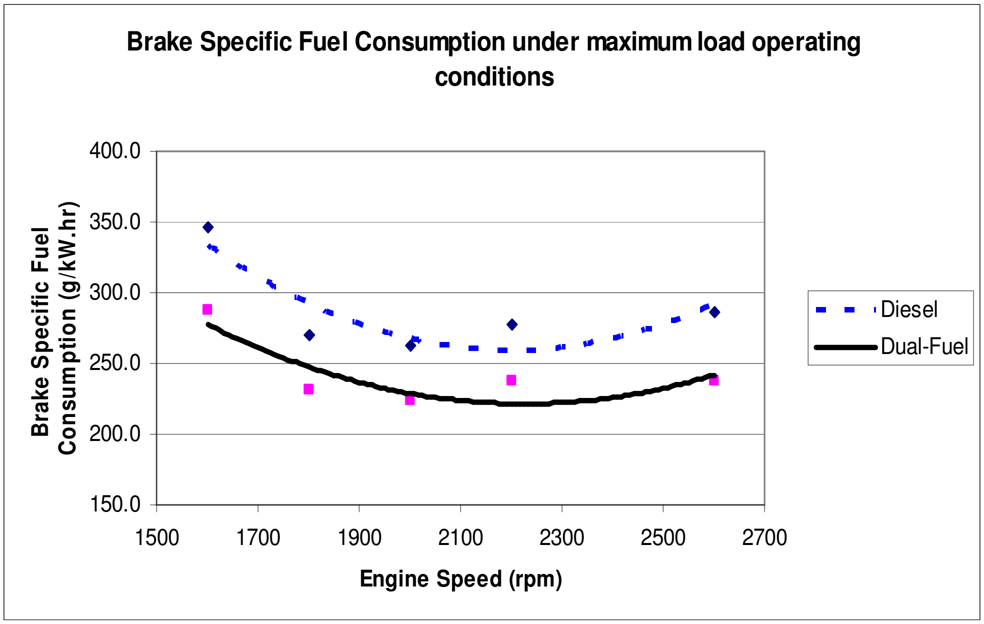


Figure 6.5: *Brake specific fuel consumption under maximum load operating conditions*

It is also observed that the specific fuel consumption decreases when the engine speed is increased from 1600 rpm to around 2200 rpm, and then shows a slight increment for speeds above 2200 rpm. At low speeds, the higher values of brake specific fuel consumption can be explained by the higher frictional forces acting on the piston and thus consuming more fuel since more heat energy is lost to friction. The fuel conversion efficiency improves when the engine speed is increased by reference to the decrease in brake specific fuel consumption for engine speeds from 1600 rpm to 2200 rpm.

By analyzing Figure 6.5, the increase in brake specific fuel consumption for engine speeds higher than 2200 rpm is mainly due to the engine knocks at 2200 rpm and 2600 rpm. Engine knocks generally decreases the combustion efficiency of the engine and greatly increases the friction since fuel ignitions are not synchronized, causing more fuel to be consumed.

6.2.6 **Engine Thermal Efficiency**

The engine thermal efficiency for diesel and dual-fuel operation is depicted in Figure 6.6 below. From the graph, it can be concluded that the thermal efficiency of the engine running on dual-fuel is higher compared to diesel fuel for engine speeds ranging from 1600 rpm to 2600 rpm. This is similar with the result obtained from Talal et al. (2003). From Figure 6.6, the maximum engine efficiency achieved for dual-fuel and diesel occurs at the engine speed of 2200 rpm at approximately 36.0% and 32.5% respectively.

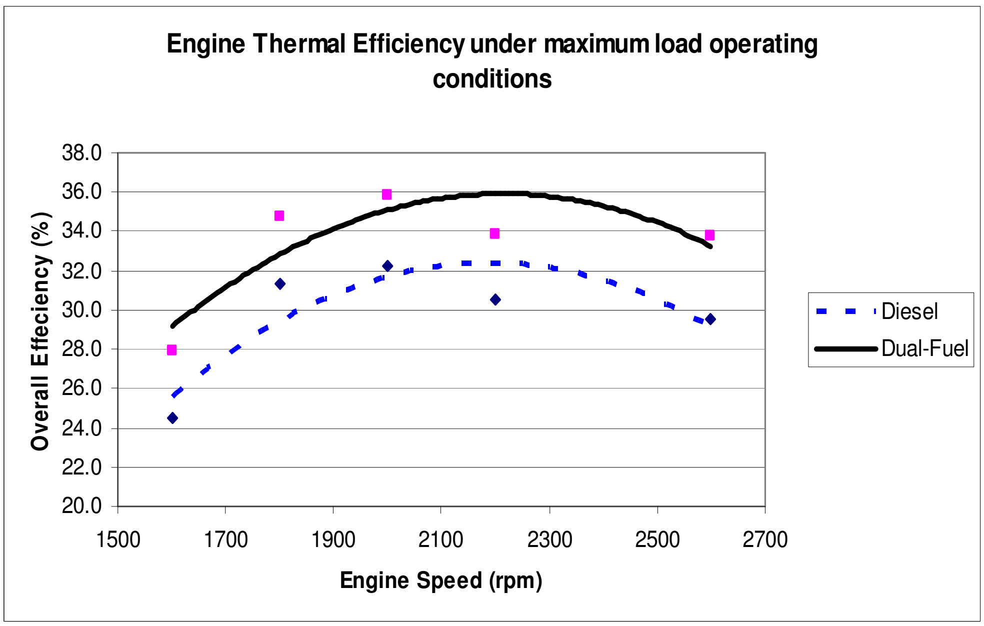


Figure 6.6: *Engine thermal efficiency under maximum load operating conditions*

The improvement in engine thermal efficiency for dual-fuel at an average of 3.6% can be explained by the higher heating value of natural gas which produces more heat for combustion for an equivalent mass flow rate compared to diesel.

Apart from that, according to Petronas Dagangan (2005), the current price of natural gas fuel stands at RM 0.565 (AUD $0.185) per litre compared to the diesel price of RM 1.281 (AUD $0.442) per liter. Neglecting the small amount of pilot diesel fuel used in dual-fuel operation, the use of natural gas as a substitute for diesel fuel offers a huge savings of almost 56% coupled with beneficial performance characteristics such as increased overall torque and brake power output as well as providing an increase in engine thermal efficiency.

At higher loads where more torque and power is generated, the engine thermal efficiency improves for both diesel and dual-fuel operations since higher mechanical efficiency is achieved at higher combustion temperatures. Apart from that, the higher rate of utilization of fuel and higher air to fuel ratio greatly increases the combustion rate of the engine since more air and fuel is introduced to the system.

At engine speeds higher than 2200 rpm, the engine efficiency thermal begin to decrease slightly due to the effects of engine knocks which not only causes output power losses but also damages the engine. The design of the engine needs to be modified to prevent engine knocking and substantially improve fuel economy. Early fuel detonation can be prevented by the design of a smaller and fast-burn motion combustion chambers and decreasing the air to fuel ratio of the mixture since combustions with rich mixture produces less heat.

6.3 Exhaust Emission for Maximum Load Operating Conditions

A portable gas analyzer which is capable of measuring the amounts of CO, HC, NO_x, CO_2, and O_2 is used to measure the exhaust emission of the engine running under diesel fuel and dual-fuel for varying engine speeds. The gas analyzer was calibrated after each measurement to obtain a high accuracy of the readings taken.

6.3.1 Carbon Monoxide

Figure 6.7 shows the emission characteristics of diesel fuel and dual-fuel operation. As can be seen, a significant reduction in carbon monoxide emission can be achieved by running the engine with dual-fuel operation. At lower engine speeds ranging from 1600 to 2000 rpm, the reduction is more apparent with an improvement of up to 70% when the engine is running at 1600 rpm.

Generally, the amount of CO present in the exhaust indicates incomplete combustion in the combustion chamber, mostly due to insufficient air or cold engine temperatures. The level of CO increases with decreasing air to fuel ratio which indicates that a rich mixture of fuel will produce more CO pollutants. Under maximum load operating conditions, diesel fuel produces a high level of CO gas since the combustion of diesel fuel is richer than dual-fuel.

For diesel fuel, it is observed that a general decrease in CO emission for engine speeds ranging from 1600 rpm to 2300 rpm. This result is in agreement with the increasing body and exhaust temperature recorded for diesel fuel operation. A higher body and exhaust temperature at high levels of engine speed indicates a higher combustion temperature and hence a more complete combustion and produces less CO gas together with increased power output. At a high speed of 2600 rpm, the CO emission level becomes higher due to engine knocking which results in lower combustion efficiency. This is also due to the fact that engine knocks become apparent when the engine is operating under high load conditions. The mixing time of diesel fuel and air inside the chamber is greatly reduced when engine knocking occurs.

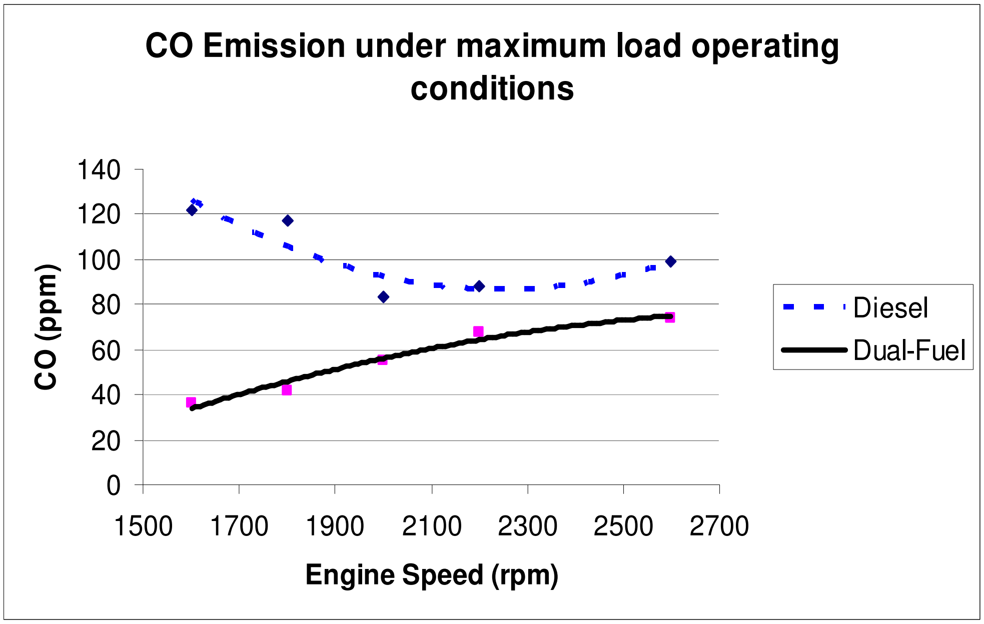


Figure 6.7: *Emission of CO under maximum load operating conditions*

For dual-fuel, the recorded CO emission is lower if compared to diesel fuel operation for all engine speeds. Natural gas produces a greater combustion efficiency leading to lower amounts of CO since natural gas in its gaseous state usually contain less contaminants than diesel fuel. Apart from that, turbulent mixing between natural gas and air in the engine produces a higher quality mixture since both are in gaseous phase, thus producing a lower carbon monoxide level generally. CO emission for dual-fuel operation increases slightly with increasing engine speed since the residence times of fuel in the combustion chamber is decreased at high engine speeds, causing higher CO formation. This result shows that dual-fuel operation is able to achieve a better combustion compared to diesel fuel under maximum load conditions.

Although the level of emission of CO is greatly reduced by using dual-fuel, the quantity produced is still above the safety level of 30 ppm. This problem can be solved by using catalytic converters to oxidize the poisonous CO into CO_2 and water or by providing regular maintenance on the engine particularly in cleaning the air filters and the fuel delivery system.

6.3.2 Unburnt Hydrocarbons

Similar to CO, the amount of unburnt HC present in the exhaust indicates a poor combustion and excess unburnt fuel which usually occurs when the fuel mixture is rich. From Figure 6.8, the emission level of HC for dual-fuel operation is substantially higher than diesel fuel. At a low engine speed of 1600 rpm, the level of HC emission recorded for dual-fuel is almost five times the amount for diesel. The reason for this difference is in the composition of natural gas which consists primarily of methane. The uncontrolled flow of natural gas into the combustion chamber produces an excess of fuel and the unburnt methane gas accounts for the majority of HC emissions collected at the exhaust. Overfueling of engine with natural gas produces a high level of HC since the oxygen gas is insufficient for complete combustion with a low air to fuel ratio.

For diesel fuel, the level of HC emission records a slight increase with engine speed but is still at a much lower level than dual-fuel. With increasing load, the amount of HC produced in the emission will decrease since greater combustion efficiency can be achieved with increased temperature. This explains the low level of HC for diesel fuel under maximum load operating conditions.

From the graph, it can be observed that the level of HC for dual-fuel decreases with engine speed. The level of HC drops because a more complete combustion is present at high engine speeds, which is in agreement with the exhaust temperature data measured. The high levels of unburnt hydrocarbon which contains mainly methane gas for dual-fuel operation poses a great risk since methane gas is a very powerful greenhouse gas with almost twenty times the global warming effect of carbon dioxide. Therefore, a suitable oxidation catalyst must be used to counter the effects of high HC emission of dual-fuel engine. The unburnt hydrocarbon can be easily converted to carbon dioxide and water by retrofitting an oxidation catalyst to replace the muffler of a vehicle.

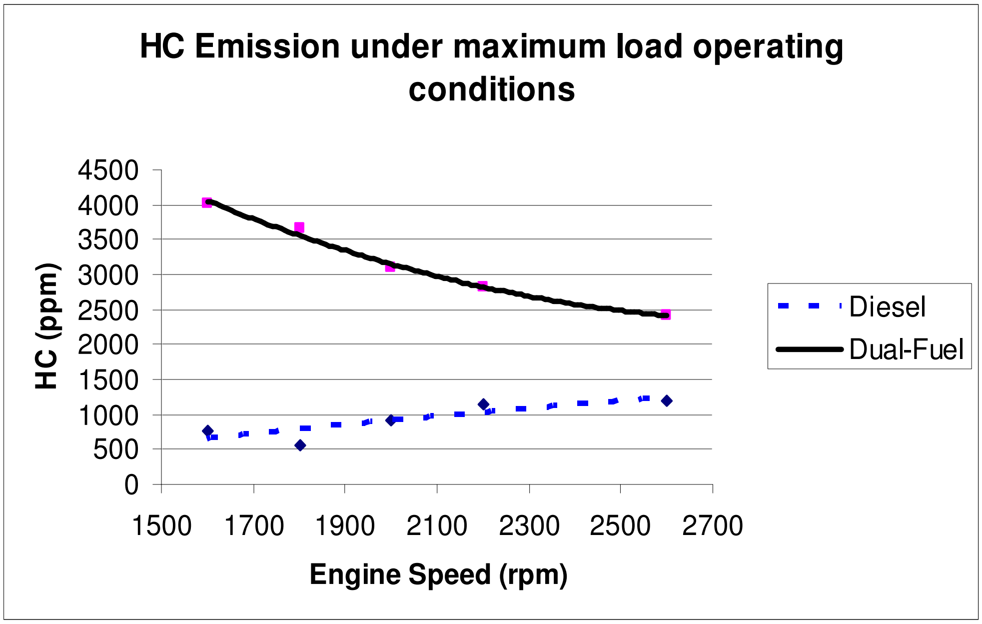


Figure 6.8: *Emission of HC under maximum load operating conditions*

Another factor which causes engines operating with dual-fuel to produce a higher level of hydrocarbons is over-scavenging of the cylinder. Diesel engines operate by igniting the fuel at the end of the compression stroke and usually the exhaust gas is forced out of the cylinder through blow scavenging, which is operated via an auxiliary pump to allow clean air to enter the chamber. Since the scavenging process involves only air, fresh natural gas and air may be forced out into the exhaust when over-scavenging occurs in dual-fuel operation. The longer time period during over-scavenging causes unburnt hydrocarbons to escape into the exhaust port and increases the overall level of recorded HC.

6.3.3 Oxides of Nitrogen (NO_x)

As observed from Figure 6.9, the NO_x level for dual-fuel operation is considerably lower than conventional diesel fuel for engine under maximum load conditions. The use of dual-fuel is favorable in reducing NO_x emission levels by an average of 58%. Generally, an increase in NO_x emission is observed when the engine speed is increased from 1600 rpm to 2600 rpm for both fuels.

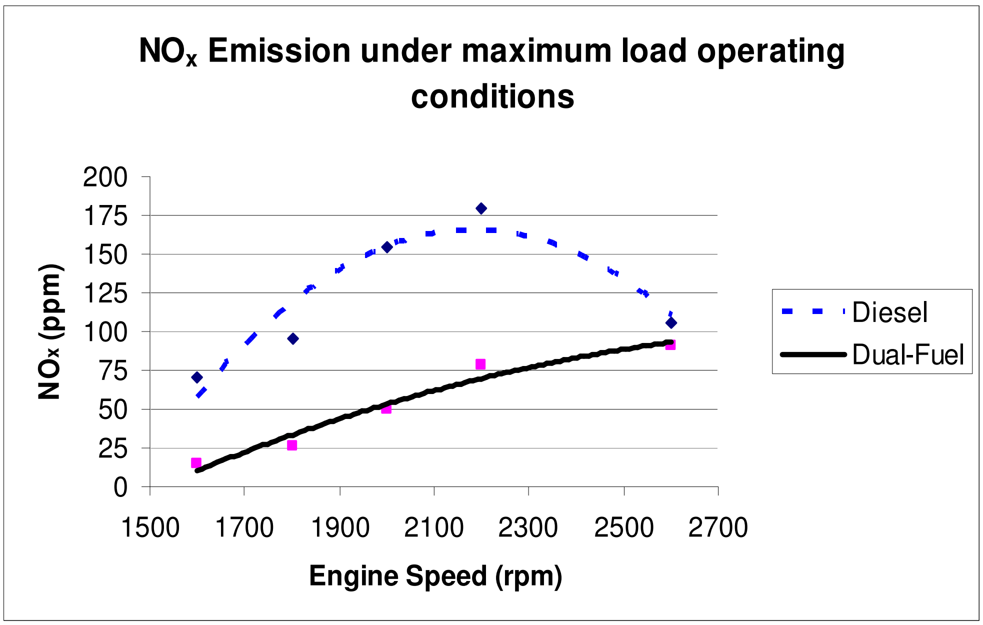


Figure 6.9: *Emission of NO_x under maximum load operating conditions*

The formation of NO_x is largely dependent on the peak temperature in the combustion chamber as well as the concentration of oxygen and nitrogen gas from the air intake. Both conditions increase the level of NO_x formation in the exhaust which leads to formation of smog and acid rains. The overall rise of NO_x as engine speed increases for both diesel and dual-fuel corresponds to the increase in body and exhaust temperature in Figure 6.1 and 6.2. The higher combustion temperature and pressure in the chamber favors the chemical reaction to form NO and NO_2 with enough oxygen.

Engines running with dual-fuel produce less NO_x than diesel since diesel fuel contains high volatile nitrogen compounds in their composition which contributes to a higher level of nitrogen concentration in the combustion chamber. Since diesel engines operate primarily in the lean region when diesel fuel is consumed, there is excess air and oxygen for the nitrogen compounds to form NO_x when the combustion temperature is high.

From Figure 6.9, a steep decline in NO_x level for diesel fuel operating at engine speed of 2600 rpm is observed. This may be caused by the low residence time for fuel in the combustion chamber at this speed. At high engine speeds, the time for NO and O_2 compounds to react is severely shortened and decreases the overall NO_x emission. Another reason for this occurrence is the low amount of O_2 present in the exhaust for diesel fuel at 2600 rpm. The low O_2 concentration of 10.5% at 2600 rpm indicates a richer combustion with less air in the chamber. Hence, the level of NO_x is greatly reduced due to insufficient air.

The low emission of NO_x for dual-fuel engines can be attributed to several factors. Firstly, the premixed combustion is less intense and produces less activation energy for nitrogen and oxygen compounds to disintegrate and form NO. The reduced mixing of air and fuel also lowers the oxidation rate of NO to NO_2 in the chamber. Apart from that, the lower exhaust temperatures present in the dual-fuel system as indicated in Figure 6.2 compared to diesel fuel reduces the NO_x production level. Finally, the concentration of O_2 is reduced in the chamber due to the presence of gaseous natural gas fuel, which will displace an equal amount of air.

6.3.4 Carbon Dioxide

From Figure 6.10, dual-fuel operation under maximum load operating conditions produces less CO_2 compared to diesel fuel by an average of 1.16%. The most significant reduction occurs at engine speed of 2200 rpm where dual-fuel provides reduction of 2.2% in CO_2 emission.

Generally, CO_2 emission levels indicate the quality and composition of fuel that is being used for combustion. The CO_2 emission levels are lower for dual-fuel since natural gas has a lower carbon content compared to the complex and longer hydrocarbon chains of diesel. Natural gas consists mainly of simple methane hydrocarbon in its composition and has less carbon molecules.

Apart from that, the ratio of carbon to hydrogen affects the amount of CO_2 production in a typical combustion. Almost 88% of natural gas comprises of methane with the chemical formula of CH_4 while the general formulation for diesel fuel is $C_nH_{1.8n}$. It is clearly seen that the ratio of carbon to hydrogen atoms in natural gas is 1 : 4 compared to 1 : 1.8 in diesel which indicates that the carbon content in diesel fuel is much higher. Therefore, the CO_2 concentration is higher for diesel operation for all range of engine speeds.

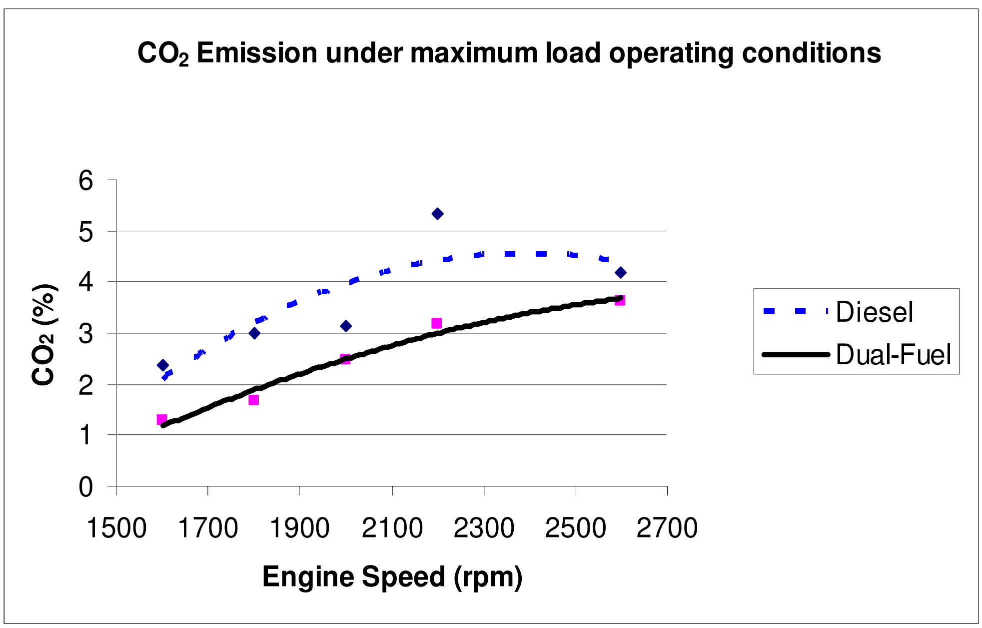


Figure 6.10: *Emission of CO_2 under maximum load operating conditions*

From Figure 6.10, a trend in CO_2 emission is also observed where an increase in engine speed will in turn cause the CO_2 level to rise gradually for both diesel and dual-fuel operations. This engine behaviour can be explained by the increase in fuel intake for combustion as the engine approaches higher speeds to support the load connected to it. The addition of fuel to the chamber causes the level of CO_2 emission to be higher at high engine speeds since more carbon molecules are introduced into the engine.

6.3.5 Excess Oxygen

The level of excess O_2 in the exhaust is shown in Figure 6.11. As can be seen, the level of excess O_2 for diesel fuel is higher than dual-fuel for maximum load conditions. A higher percentage of O_2 in the exhaust corresponds to leaner combustion where the air to fuel ratio is higher than 1. Diesel engines normally operate at a lean point of stoichiometric for diesel fuel and cause more air to be present in the combustion chamber and exhaust, producing a higher level of unused excess O_2.

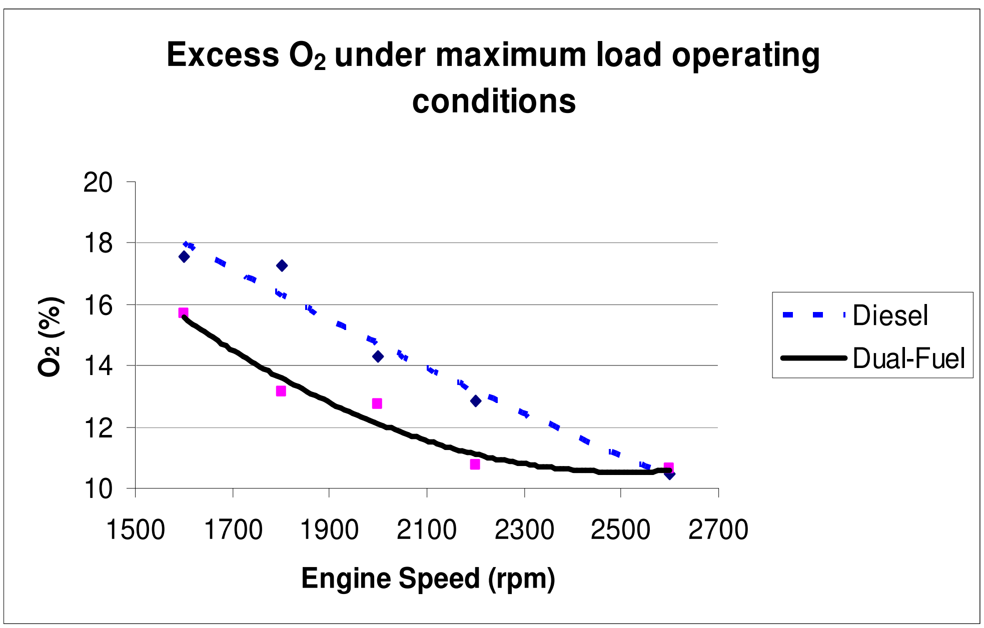


Figure 6.11: *Excess of O_2 under maximum load operating conditions*

When the engine operates using dual-fuel, the level of excess O_2 decreases by an average of 1.9% over the range of speeds from 1600 rpm to 2600 rpm compared to diesel. This result implicates that combustion of dual-fuel is better and leads to lower levels of excess O_2 left in the exhaust since majority of the oxygen supplied is used for the combustion process.

This result is in agreement with the increasing CO_2 emission level shown in Figure 6.10 for both diesel and dual-fuel operating conditions. At higher engine speeds, the combustion becomes more complete to support the higher torque and output power levels and produces more CO_2 and less excess O_2 in the exhaust.

Figure 6.11 also shows a general decline in O_2 percentage as engine speed increases.

6.4 Engine Performance for Moderate Load Operating Conditions

The engine performance data for moderate load operating conditions are taken at a load of 3 kW for every engine speed for both diesel and dual fuel to enable a fair comparison for the engine performance aspects. Similarly, the torque readings are taken from the attached dynamometer, the mass flow rate of diesel from the flow meter and a digital thermocouple is used to obtain the engine body and exhaust temperatures when the readings become constant.

6.4.1 Engine Body Temperature

From Figure 6.12, it is observed that the engine body temperature of dual-fuel operating system is generally lower than diesel fuel.

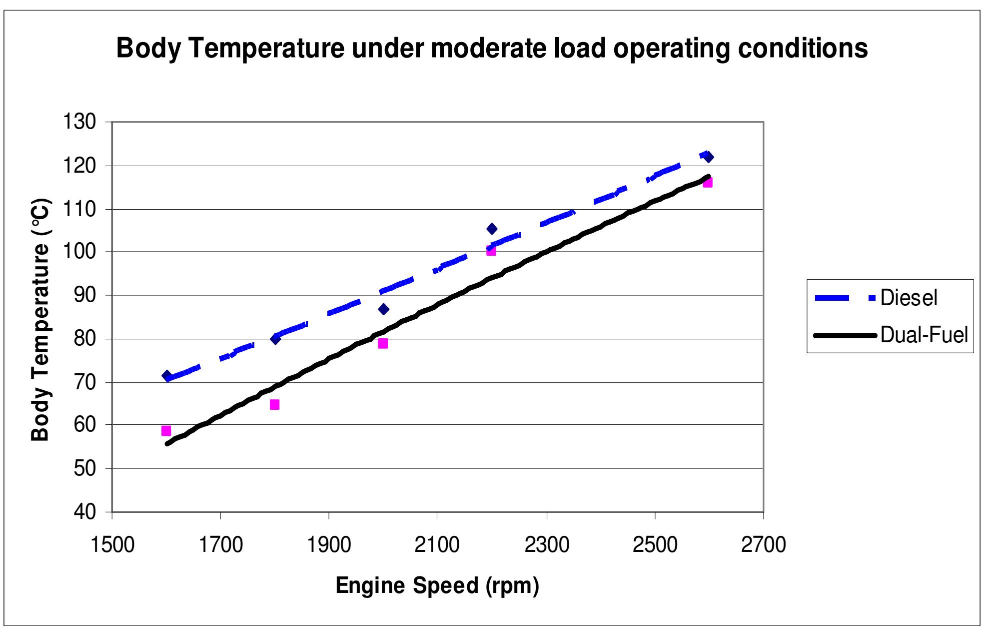


Figure 6.12: *Engine body temperatures under moderate load operating conditions*

On average, the body temperature of diesel fuel is 9.48 °C higher than dual-fuel for the range of engine speeds tested. The body temperature of dual-fuel increases at a higher rate than diesel fuel as the engine speed is increased.

The general increase in body temperature at higher engine speeds is caused by an increase in fuel intake when the throttle is adjusted to increase the engine speed. The additional fuel input to the engine produces combustion at a higher temperature albeit still lower than the maximum load condition. On average, the body temperature for moderate load is 58.6 °C lower for diesel fuel and 39.1 °C lower for dual-fuel compared to maximum loading conditions. This result shows that there is a substantial difference in combustion temperature and amount of fuel and air intake between a similar engine operating under moderate load at 3 kW and maximum load at a range of 7-8 kW.

6.4.2 Exhaust Gas Temperature

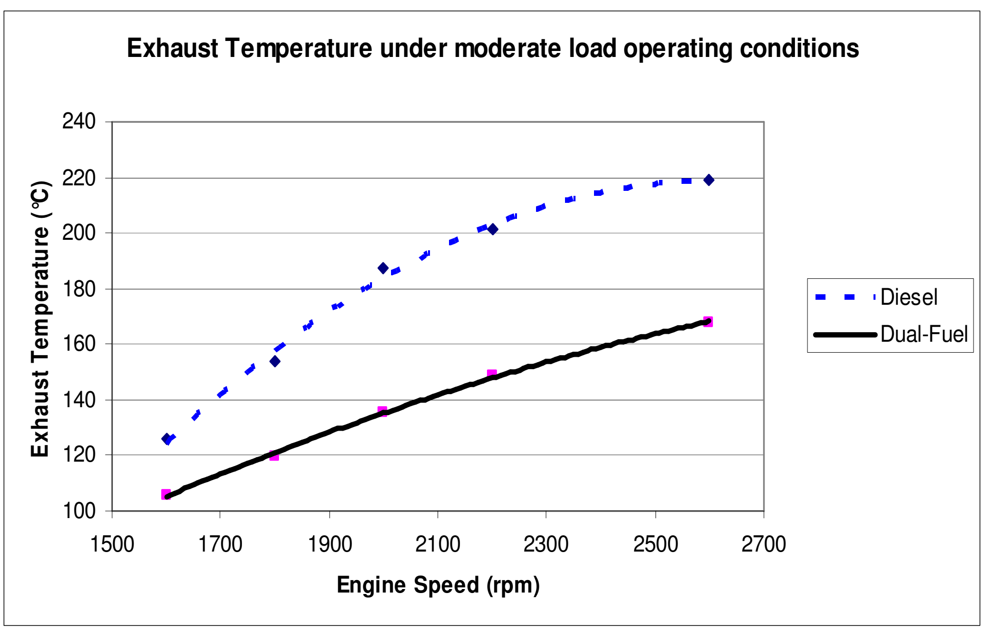


Figure 6.13: *Engine exhaust temperatures under moderate load operating conditions*

For the exhaust gas temperature, the general increase in temperature as the engine speed climbs is also observed. From Figure 6.13, it is shown that the rate of increase of exhaust temperature in diesel fuel operation is greater than dual-fuel as the engine speed is increased from 1600 rpm to 2600 rpm. The maximum temperature difference between diesel and dual-fuel occurs at an engine speed of 2200 rpm with 52.9 °C difference.

Similar to the engine body temperature, the exhaust temperatures for both fuels under moderate loading conditions are lower than the maximum load, with an average value of 101.1 °C lower for diesel and 122.8 °C lower for dual-fuel at all engine speeds.

As mentioned before, the high exhaust temperature for diesel fuel indicates a higher percentage of NOx production in the combustion products. It is also a direct result of lean combustion associated with diesel fuel in CI engines which generates more heat in the combustion process.

6.4.3 **Engine Torque**

From Figure 6.14, the engine torque measured from the dynamometer shows that rotational force exerted by the engine to provide power to the loads is higher for dual-fuel compared to diesel. An increase of 1 Nm of torque for every measured engine speed is recorded. This shows that the energy input for dual-fuel operation is higher than diesel fuel since more power is produced.

Similar to the maximum load operating condition, the engine torque for moderate load increases for engine speeds ranging from 1600 rpm to 2200 rpm for both operating conditions as the fuel intake is increased and higher combustion temperature is achieved. Engine knocks results in a slight drop of torque values at the engine speed of 2600 rpm for both diesel and dual-fuel, although the effect is not as apparent as the maximum load operating condition.

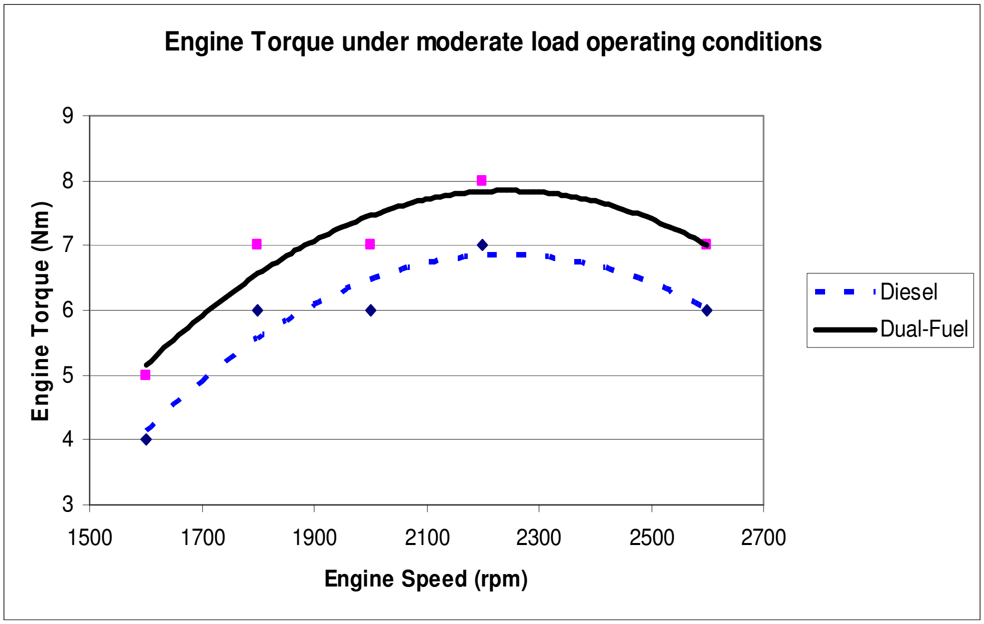


Figure 6.14: *Engine torque output under moderate load operating conditions*

In comparison, the engine torque produced for moderate loading is lower by an average of 29.2% for diesel fuel and 26.0% for dual-fuel for all engine speeds compared to the maximum load operating condition, mainly due to the reduced fuel intake at lower engine loads.

6.4.4 **Engine Brake Power**

The engine brake power comparison for both diesel and dual-fuel is shown in Figure 6.15. Proportional to the engine torque, the brake power produced by dual-fuel is higher than diesel for engine speeds ranging from 1600 rpm to 2600 rpm. The maximum brake power achieved is 1.634 kW for diesel fuel and 1.906 kW for dual-fuel, both at an engine speed of 2600 rpm. The 14.3 % increment in engine brake power is significant together with its lower specific fuel consumption and emissive gases to be a suitable alternative fuel to replace diesel. The engine brake power shows a typical slight decrease compared to maximum load conditions for both fuels.

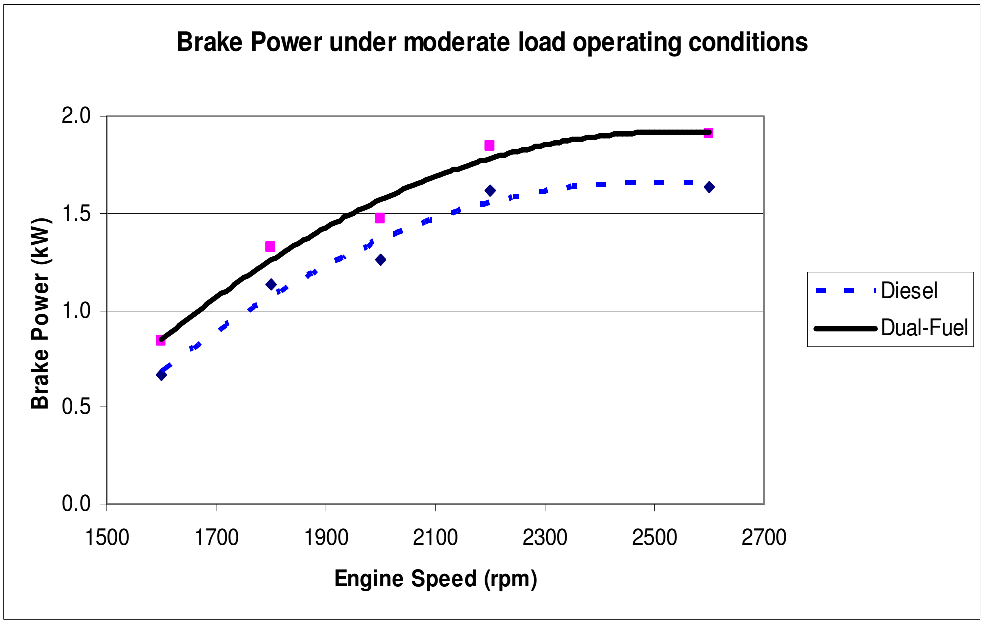


Figure 6.15: *Engine brake power under moderate load operating conditions*

6.4.5 **Brake Specific Fuel Consumption**

Figure 6.16 shows the brake specific fuel consumption for moderate load conditions for diesel and dual-fuel. The bsfc for dual-fuel is lower than diesel from low to high engine speeds by an average of 19.5%. The higher octane rating of CNG fuel compared to diesel decreases the amount of fuel needed for combustion, and hence lower brake specific fuel consumption for dual-fuel is noticed.

The bsfc for both the fuels decrease gradually when the engine speed is increased. At lower engine speeds, the high frictional forces acting on the piston and cylinder walls results in a greater fuel consumption rate since the heat of combustion is lost to friction. This effect is greatly reduced when the engine speed rises above 2000 rpm, which is the conventional engine speed for the engine of any conventional consumer vehicle during motion.

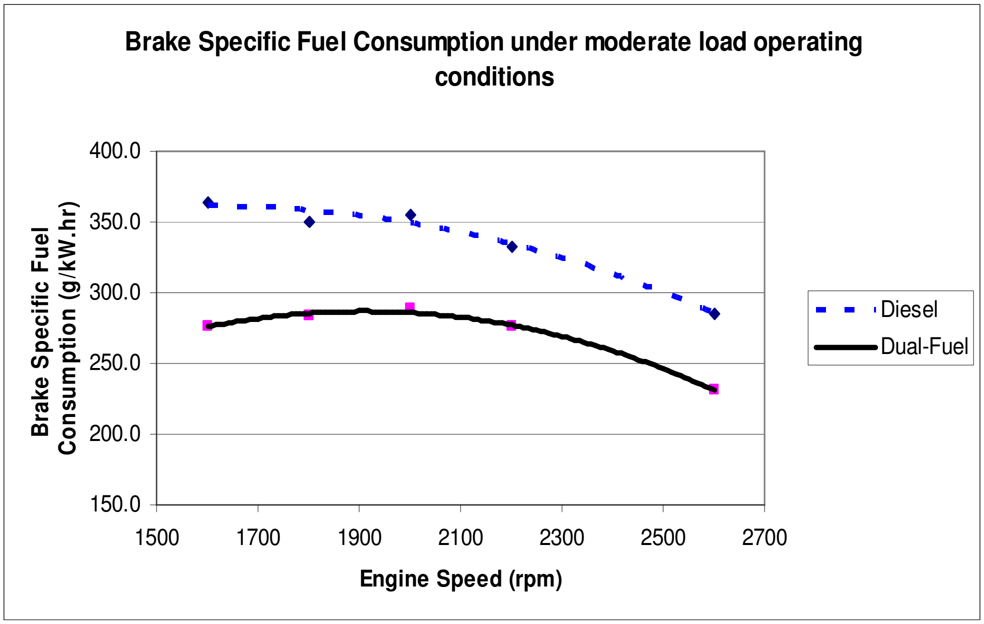


Figure 6.16: *Brake specific fuel consumption under moderate load operating conditions*

By comparing with the maximum load condition, the bsfc for moderate load is generally higher by an average of 17.8% for diesel and 12.3% for dual-fuel taking into account for all the engine speeds tested. The increase in bsfc can be explained by the lower mechanical efficiency of the engine for lower load conditions where there is a higher percentage of heat loss due to the lower utilization of fuel. Apart from that, at lower loading conditions, the lower combustion temperatures justified by the lower body and exhaust temperatures, and lower air to fuel ratio (λ) results in a lower combustion rate and therefore a lower fuel conversion efficiency.

6.4.6 **Engine Thermal Efficiency**

Similar to the engine at maximum loading conditions, Figure 6.17 shows that the engine thermal efficiency for dual-fuel in generally higher than diesel. On average, the thermal efficiency for dual-fuel is 4.48% higher for moderate load compared to 3.60% for maximum load. This difference can be explained by the lower rate of engine knocks at lower engine loads since the engine is not burdened to operate at its maximum potential.

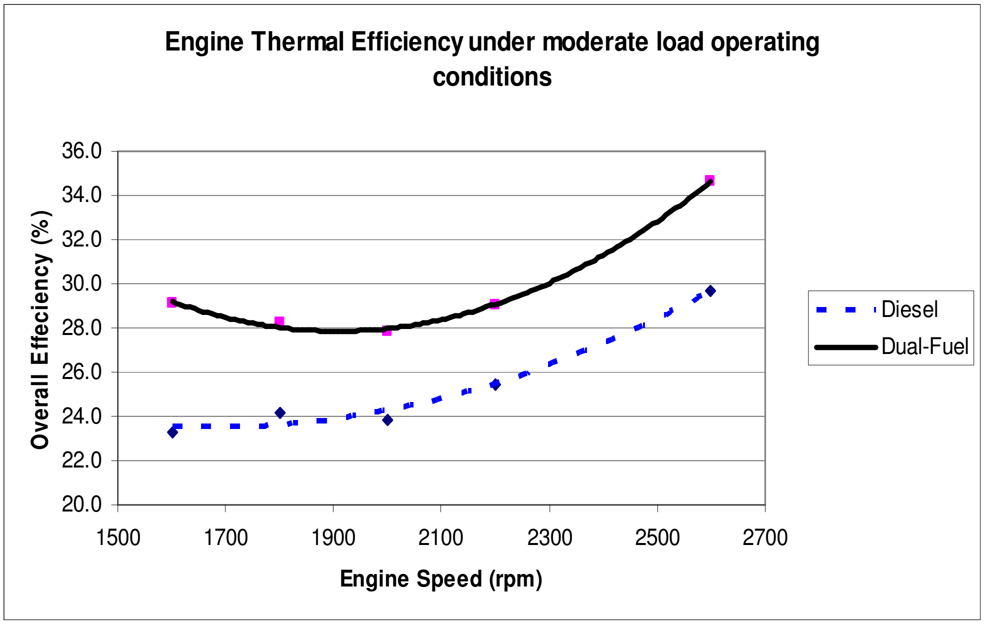


Figure 6.17: *Engine thermal efficiency under moderate load operating conditions*

The maximum engine efficiency achieved for moderate load conditions for both diesel and dual-fuel is generally lower than the maximum load operating conditions. Calculations show that the average drop in thermal efficiency associated with moderate loading to the engine is 4.32% for diesel and 3.47% for dual-fuel. This phenomenon can be described by the lower fuel utilization and air to fuel ratio when the engine load is low, causing the combustion rate to be greatly reduced in comparison.

6.5 Exhaust Emission for Moderate Load Operating Conditions

The discussions for the exhaust emission measured using the gas analyzer is categorized into different pollutants for both type of fuels. The emission levels of CO, HC, NO_x, CO_2, and O_2 are compared and discussed between diesel and dual-fuel and the effects of load intensity on the emission are also included.

6.5.1 Carbon Monoxide

The emission of carbon monoxide for diesel and dual-fuel when the engine is running under moderate loading is shown in Figure 6.18. Resembling the maximum load conditions, the CO emission levels for dual-fuel is considerably lower than diesel due to the cleaner contents of natural gas fuel. The average reduction level of CO emission by using the dual-fuel operation stands at 58.0% with the maximum occurring at the minimum engine speed of 1600 rpm where the reduction level goes up as high as 79.4%.

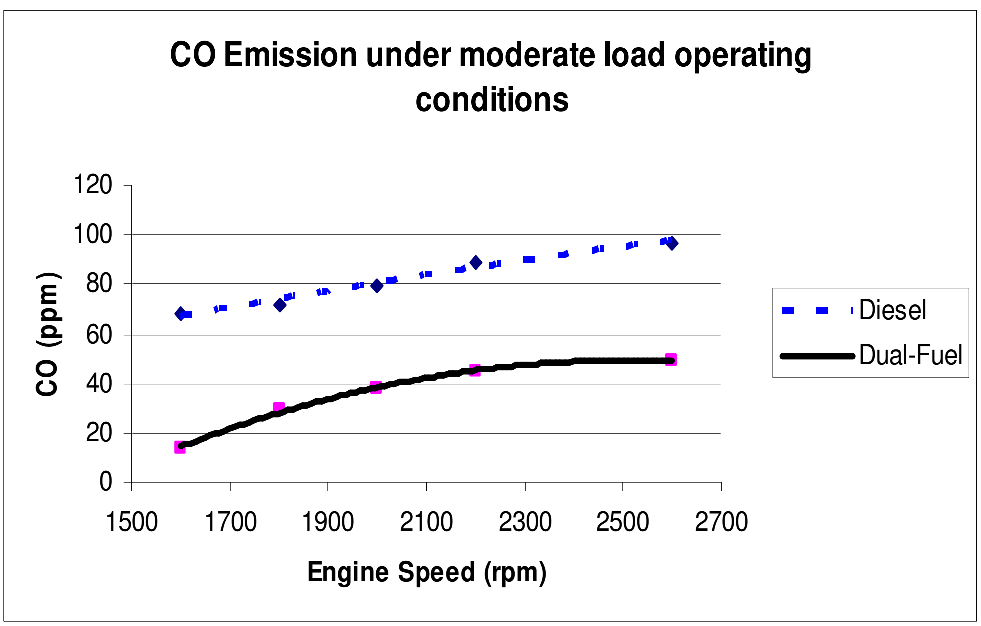


Figure 6.18: *Emission of CO under moderate load operating conditions*

The initial high levels of CO for diesel fuel can be related to the engine body and exhaust temperatures from Figures 6.12 and 6.13. The lower engine temperatures indicate lower combustion efficiency since combustion takes place at a lower temperature. This can be related to the enormous amount of CO production for diesel fuel during engine idling or cold-startup.

In comparison, the CO emission levels of both the fuels are lower compared to the maximum loading conditions as the combustion process takes place in the leaner region. The excess amount of air present in the chamber during moderate load conditions prevents carbon monoxide buildup since fuel combustion is complete with sufficient oxygen levels.

The richer mode of combustion when the engine speed increases causes a small increase in CO emission for both the fuels since less air is being supplied to the same volume of fuel.

6.5.2 Unburnt Hydrocarbons

The level of unburnt hydrocarbon is also associated with incomplete combustion and excess of unburnt fuel. From Figure 6.19, the level of HC emission for dual-fuel operation is extremely high compared to diesel fuel especially at low engine speeds below 1800 rpm. At a speed of 1600 rpm, dual-fuel produces a vast increase of HC, at around 145% higher than diesel. The uncontrolled natural gas flow input into the combustion chamber explains this occurrence. At lower loads and engine speeds, most of the natural gas is not completely utilized for the combustion process and most of the methane in the natural gas escapes into the exhaust to produce a high level of unburnt hydrocarbon detection.

At moderate engine loads, the level of HC produced is by average 96.0% higher for diesel and 10.5% higher for dual-fuel compared to the maximum load operating conditions. This result is justified by the lower combustion temperature present for moderate loading which causes incomplete combustion and excess fuel to be forced into the exhaust system.

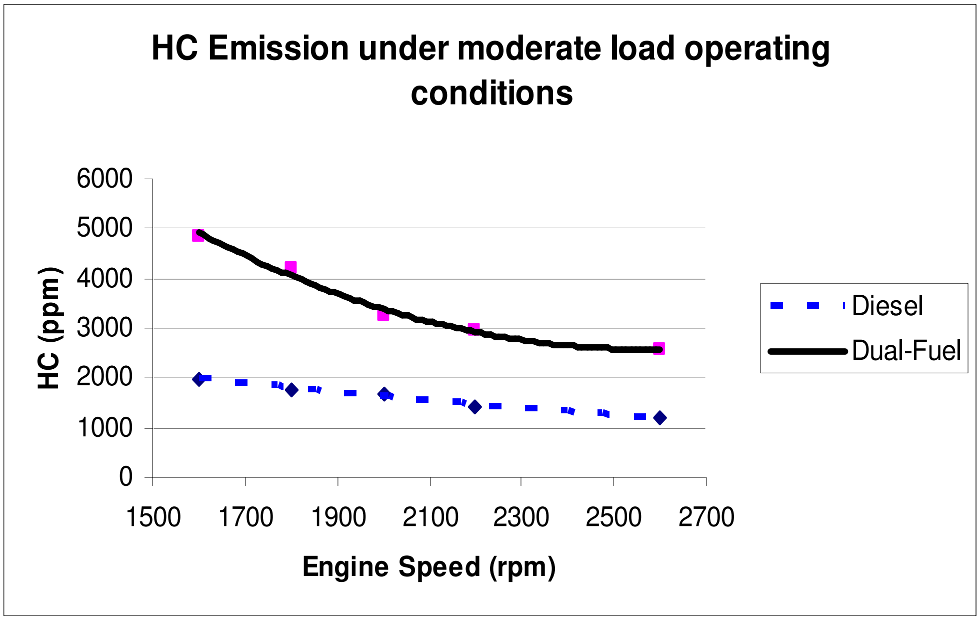


Figure 6.19: *Emission of HC under moderate load operating conditions*

As mentioned earlier in the maximum load operating conditions discussions, the level of HC emission reduces with increasing engine speed due to the increase in quality of combustion for higher engine speeds due to the increasing combustion temperature. It is thus shown that the level of unburnt hydrocarbon is heavily dependent on the combustion temperature compared to the air to fuel ratio.

6.5.3 Oxides of Nitrogen (NO_x)

Figure 6.20 shows the level of NO_x produced by both the fuels under moderate loading conditions. It is clear that the use of dual-fuel significantly reduces the harmful NO_x gases due to its lower nitrogen composition compared to diesel. The reduction by an average of 68.2% is a direct result of lower combustion temperatures for dual-fuel which impedes NO_x production.

The steady increase in NO_x emission as the engine speed increases can also be attributed to the increase in engine combustion temperature. A higher ignition temperature favors the chemical reactions to form NO_x when enough air and time is present. The emission levels increase by 110% for diesel and 228% for dual-fuel when then engine speed is increased from 1600 rpm to 2600 rpm.

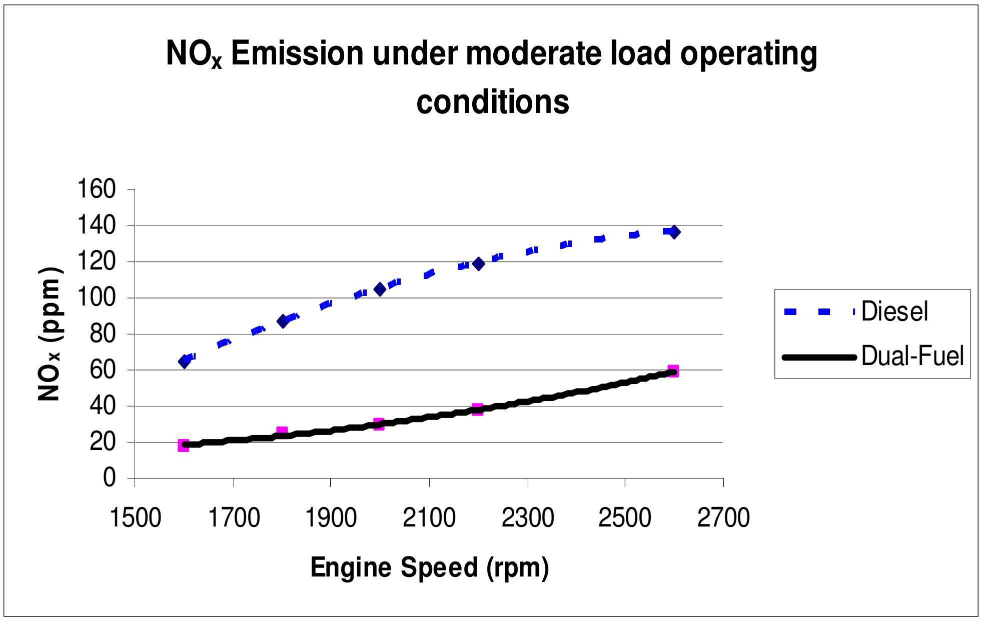


Figure 6.20: *Emission of NO_x under moderate load operating conditions*

By making comparisons with the same engine speeds, the average percentage of NO_x emission under moderate loading is 10.7% lower for diesel and 22.5% lower for dual-fuel compared to the maximum loading conditions, due to the influence of combustion temperature as well.

6.5.4 Carbon Dioxide

The level of CO_2 emission for diesel fuel is generally higher than dual-fuel from Figure 6.21. The maximum level of carbon dioxide emission for diesel fuel occurs at the engine speed of 2600 rpm with 4.12% emission while the dual-fuel operation recorded 2.29% maximum at the same engine speed. The highest CO_2 emission is recorded at the maximum engine speed measured because the fuel intake is at its highest.

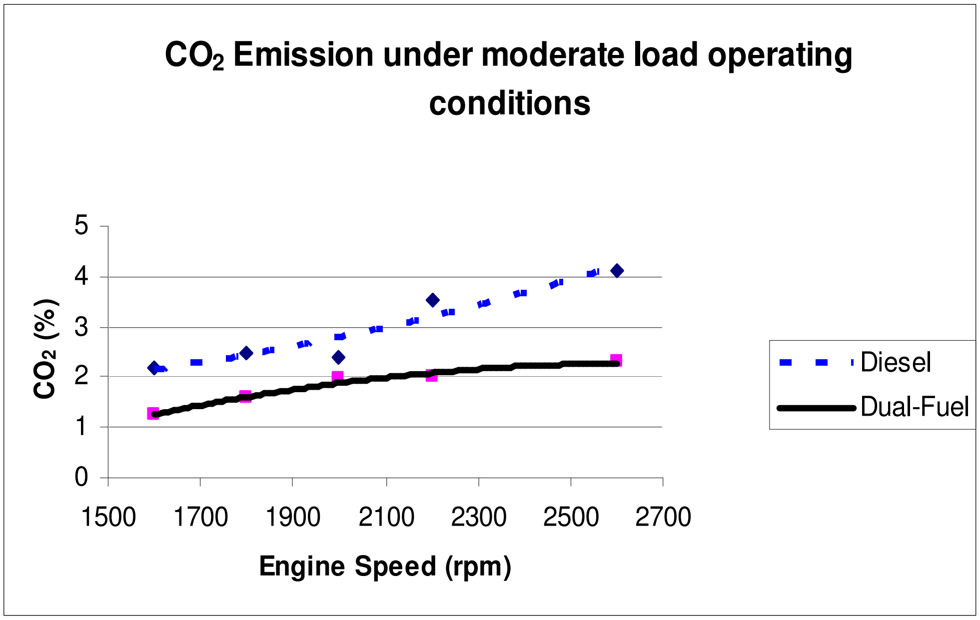


Figure 6.21: *Emission of CO_2 under moderate load operating conditions*

The higher carbon content and percentage for diesel fuel accounts for its higher CO_2 emission at all range of engine speeds. In comparison, the level of CO_2 emission is reduced by an average of 0.65% when running the engine at moderate loading conditions compared to the maximum load where the fuel intake is maximum.

6.5.5 Excess Oxygen

Figure 6.22 shows the level of excess oxygen present in the exhaust after combustion for both fuels under 3 kW loading condition. The higher level of excess O_2 detected implies a lower combustion efficiency since most of the oxygen is not used up when there is an excess of fuel from the unburnt hydrocarbon readings.

The data of excess oxygen for diesel and dual-fuel records only a slight difference between them, with only a 0.13% percentage difference on average. Generally, as the combustion temperature rises with engine speed, the level of excess oxygen decreases since more air is being used for combustion. The higher level of excess oxygen present during moderate loading also proves that the combustion rate is lower compared to maximum loading.

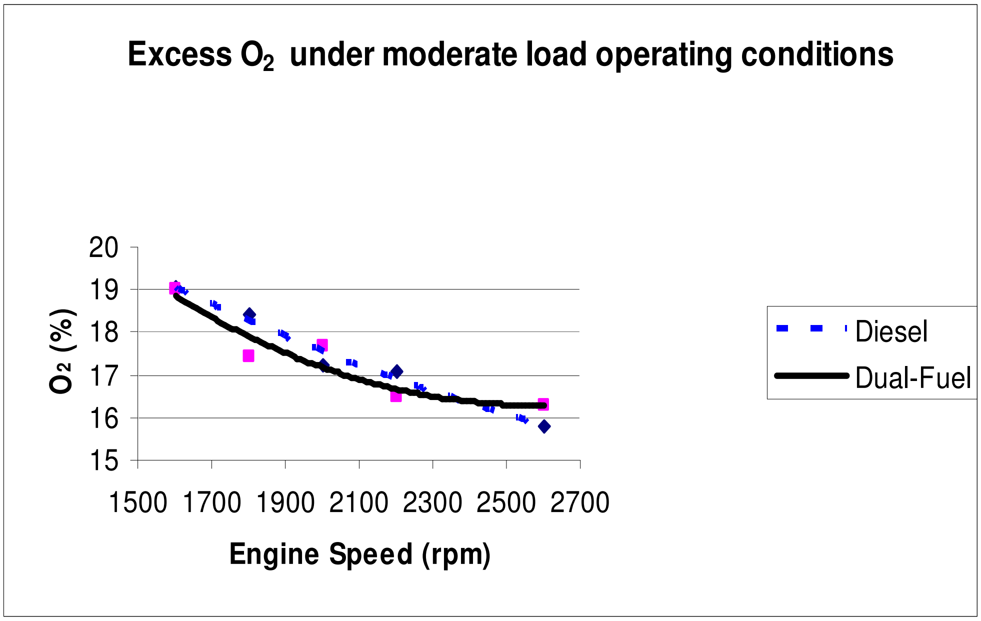


Figure 6.22: *Excess of O_2 under moderate load operating conditions*

6.6 Concluding Discussion

The average data values for engine speed ranging from 1600 rpm to 2600 rpm for both maximum and moderate load operating conditions are tabulated below for comparison between diesel and dual fuel.

Table 6.1 shows the engine performance summary for maximum loading which shows an improvement in engine torque, engine brake power and thermal efficiency while a reduction of specific fuel consumption is observed. On the other hand, Table 6.2 sums up the emission characteristics for maximum operating load which shows a reduction in carbon monoxide, oxides of nitrogen and carbon dioxide. It is also observed that the level of unburnt hydrocarbon emission increased by almost two and a half fold when dual-fuel is used.

Engine Performance	Diesel	Dual-Fuel	Percentage Increase
Engine Torque (Nm)	8.20	9.20	12.20
Engine Brake Power (kW)	1.75	1.96	12.23
Brake Specific Fuel Consumption (g/kW.hr)	288.75	243.43	-15.70
Engine Thermal Efficiency (%)	29.62	33.25	3.63

Table 6.1: *Engine Performance for maximum load*

Exhaust Emission	Diesel	Dual-Fuel	Percentage Decrease
Carbon Monoxide (ppm)	102	55	46.08
Unburnt Hydrocarbon (ppm)	918	3200	-248.58
Oxides of Nitrogen, NO_x (ppm)	121	52	57.02
Carbon Dioxide (%)	3.61	2.45	32.13
Excess Oxygen (%)	14.49	12.59	13.11

Table 6.2: *Exhaust Emission for maximum load*

On the other hand, Table 6.3 shows the engine performance summary for the experiment conducted with moderate loading conditions. The increase in engine torque, brake power and engine efficiency is observed as well, similar to the maximum load operating condition. There is more improvement in the engine performance for the moderate load operating condition as the engine thermal efficiency is improved by an amount as high as 4.48%. Apart from that, the exhaust emission for moderate loading conditions records a better outcome with a further reduction in the toxic and greenhouse gases while the unburnt hydrocarbon level is only increased by 122% when dual-fuel is used.

Engine Performance	Diesel	Dual-Fuel	Percentage Increase
Engine Torque (Nm)	5.80	6.80	17.24
Engine Brake Power (kW)	1.26	1.47	16.94
Brake Specific Fuel Consumption (g/kW.hr)	337.35	271.11	-19.64
Engine Thermal Efficiency (%)	25.30	29.78	4.48

Table 6.3: *Engine Performance for moderate load*

Exhaust Emission	Diesel	Dual-Fuel	Percentage Decrease
Carbon Monoxide (ppm)	81	35	56.79
Unburnt Hydrocarbon (ppm)	1608.4	3570.4	-121.98
Oxides of Nitrogen, NO_x (ppm)	102.6	33.8	67.06
Carbon Dioxide (%)	2.94	1.82	38.10
Excess Oxygen (%)	17.51	17.38	0.74

Table 6.4: *Exhaust Emission for moderate load*

In general, for both maximum and moderate load, the engine thermal efficiency is greater for dual-fuel operation compared to diesel and more engine torque and brake power is produced. Apart from that, there is a hefty decrease in major pollutant such as carbon monoxide, oxides of nitrogen (NO_x) and carbon dioxide. A lower level of excess oxygen signifies a higher combustion rate for dual-fuel. However, the level of unburnt hydrocarbon in dual-fuel operation for both maximum and moderate load record an incredibly 250% and 120% increase respectively. This is a direct result of uncontrolled natural gas flow into the engine which produces methane gas accumulation in the exhaust.

6.7 Catalytic Aftertreatment

The high level of unburnt hydrocarbon produced for dual-fuel operation for both moderate and maximum loading can be controlled by using catalytic aftertreatment techniques which uses noble metals such as platinum, palladium and rhodium to allow reactions to take place that oxidizes carbon monoxide and hydrocarbons to a safe level (Turns 1996). Washcoat such as alumina and promoters are added to increase the reaction level and efficiency of oxidation. The absence of sulphur in natural gas fuel allows the use of catalytic aftertreatments since sulphur compounds are known to interfere with the catalytic action and reduce the efficiency.

A three-way catalyst which controls the pollutant levels of three emission gases namely carbon monoxide, hydrocarbon and oxides of nitrogen can be used in the aftertreatment. The main criterion which governs the feasibility of this three-way catalyst is the air-fuel ratio which must be near stoichiometric value for an effective reduction in all three pollutant gases as shown in Figure 6.23.

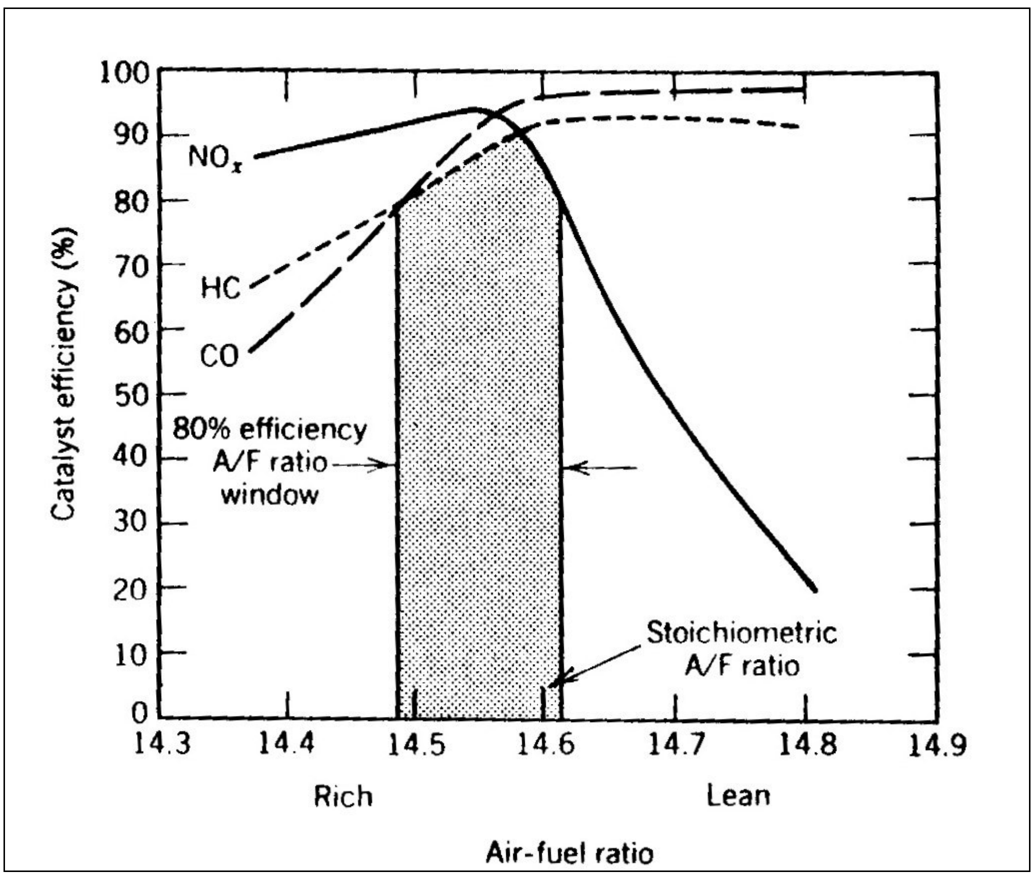


Figure 6.23: *Conversion efficiency of three-way catalyst as a function of air-fuel ratio (Ferguson 1986)*

Since CNG fuel produces a low level of oxides if nitrogen (NO_x) emission due to its low nitrogen content in its fuel, the lean operating condition of diesel engines will not cause a major problem since the catalyst conversion efficiency for both hydrocarbon and carbon monoxide is still high for lean air-to-fuel ratio as shown in Figure 6.23. The reduction of unburnt hydrocarbon and carbon monoxide remain the primary concern in using CNG fuel to reach the safety emission level imposed by the authorities.

Ammonia (NH_3) and hydrogen sulphide (H_2S) are both by-products of three-way catalysts generated by the reduction of oxides of nitrogen and sulphur dioxide respectively which are common in normal diesel engines. A high catalyst temperature and rich fuel mixture promotes the formation of both the by-products which causes irritation and discomfort to humans. The low level of NO_x emission coupled with the zero sulphur content of natural gas fuel causes the emission of ammonia and hydrogen sulphide gases to be low. The lower exhaust temperature for dual-fuel operation further reduces the catalyst by-products to a negligible level.

Apart from that, the high accumulation level of methane in the engine due to the uncontrolled intake of natural gas must be controlled through an automated valve which comprise of sensors of the gaseous fuel level required for combustion so that the amount of hydrocarbon in the exhaust can be reduced. Hydrocarbon filters and coalescers in stand alone vessels can also be set up in the natural gas fuel line supply to separate solid hydrocarbon and moisture in the air to allow efficient operation of the engine and its lubrication systems. Good engine maintenance and tune-up is also necessary to reduce the level of emission gases in the exhaust of the CNG fuel system.

CHAPTER 7

CONCLUSION

7.1 Achievement of Objectives

The main objectives of the Project Specification in Appendix A were achieved. A compression ignition standard diesel engine was successfully converted to utilize CNG fuel by retrofitting a conversion kit. Dual-fuel operation which utilizes diesel as a pilot fuel as a source of ignition for natural gas proves to be an excellent substitute for conventional diesel fuel with its superior engine performance and exhaust emission characteristics.

The results obtained from the experiments conducted on both the maximum and moderate load operating conditions show substantial reduction in carbon monoxide, oxides of nitrogen, carbon dioxide, and excess oxygen levels in the exhaust. Apart from that, the engine performance is also improved when using dual-fuel operation with increased engine torque, brake power and engine efficiency alongside an improvement in specific fuel consumption. The extreme levels of unburnt hydrocarbons produced when utilizing dual-fuel operation due to the uncontrolled fuel input to the engine is controlled using catalytic aftertreatment process which reduces the level of carbon monoxide and oxides of nitrogen simultaneously through application of a three-way catalyst.

The price of natural gas which costs less than half the price of conventional diesel fuel provides massive savings in terms of fuel costs and the payback period for the initial CNG conversion kit cost is reduced with these huge savings.

7.2 Recommendation and Future Work

The engine performance characteristics comparison between conventional diesel fuel and dual-fuel can be improved by consideration of the heat loss of combustion to the cylinder walls and also the friction generated which contribute to the total energy loss of the system through use of measurement instruments such as a calorimeter. The corrected engine output power can then be obtained to achieve a more accurate value of the overall efficiency of the engine.

Devices which are able to detect other emission gases such as particulate matter (PM), oxides of sulphur, ammonia, hydrogen sulphide and other chemical compounds in the exhaust of the engine can be used to monitor the emission levels of both fuels so that a more comprehensive study on the emission characteristics can be performed to ensure that no unusually high pollutant levels are associated with dual-fuel operation.

Apart from that, the catalytic aftertreatment chosen can be installed on the exhaust of the engine to allow emission data to be collected so that the reduction of the high level of unburnt hydrocarbon can be monitored and the apparatus can be modified if necessary to ensure that the safety standards are reached. Software simulation on the engine performances and the process of combustion in the engine can be used to design the optimum engine to incorporate dual-fuel by varying parameters such as the air-fuel ratio, compression ratio and other engine parameters.

LIST OF REFERENCES

1) Alternative Fuels Data Centre 2004, U.S. Department of Energy, United States of America, viewed 10 March 2005, <http://www.eere.energy.gov/afdc/altfuel/whatis_gas.html>

2) Alternate Fuels Technologies Inc. n.d., '*Propane and Natural Gas Conversions*', viewed 23 May 2005, <http://www.propanecarbs.com/home.html>

3) Autologic Company n.d, ' *Autogas Emissions Analyzer*', Sussex, Wisconsin, United States of America, viewed 10 June 2005, <http://www.autologicco.com/AllProducts/GasAnalyzer.shtm#GasPortable>

4) Autoworld 2004, '*NGV…What is it?'*, Autoweb Sdn. Bhd., viewed 12 March 2005, <http://www.autoworld.com.my/motoring/review/viewarticle.asp?awMID=83&awCatID=RT.ATC.TIP.GEN>

5) Barbotti CNG 2002, Barbotti CNG, Santa Fe, Argentina, viewed 17 March 2005, <http://www.barbotti.com/cng/advantages.htm>

6) BP Global 2005, '*Natural Gas'* , BP Public Limited Company, United Kingdom, viewed 2 March 2005, <http://www.bp.com/sectiongenericarticle.do?categoryId=2012413&contentId=2018342>

7) Clean Air Power n.d. , Clean Air Power Limited, San Diego, United States of America, viewed 18 March 2005, <http://www.cleanairpower.com/technology/df_technology.html>

8) Clean Air Technologies Information Pool 2005, Clean Air Initiative, Washington, United States of America, viewed 19 March 2005, <http://www.cleanairnet.org/infopool/1411/propertyvalue-17753.html>

9) Environmental Centre n.d., '*Air Quality: Airborne Illness*', viewed 23 May 2005, <http://www.nutramed.com/environment/monoxide.htm>

10) Equitable Gas n.d., Pennsylvania, United States of America, viewed 3 May 2005, <http://www.equitablegas.com/tech/ngv.htm>

11) Ferguson, C.R. 1986, *Internal Combustion Engines: Applied Thermosciences*, John Wiley and Sons, New York

12) Gas-Lite Manufacturing 2004, '*The History of The Gas Light*', Gas-Lite Manufacturing Company, Pittsburgh, United States of America, viewed 15 March 2005, <http://www.gaslite.com/history.html>

13) Graham, J., Hammit, J., Toy, E. 2000, 'Risk in Perspective: Fueling Heavy Duty Trucks: Diesel or Natural Gas?', *Harvard Centre for Risk Analysis*, Volume 8, Issue 1, viewed 18 March 2005, <http://www.cleanairnet.org/infopool/1411/articles-35700_risk_perspective.pdf>

14) Gwilliam, K. 2000, '*The Role of Natural Gas in the Transport Sector*', viewed 20 March 2005, <http://www.cleanairnet.org/infopool/1411/articles-35703_role_naturalgas.pdf>

15) Heywood, J. B. 1988, *Internal Combustion Engine Fundamentals*, McGraw Hill, Singapore

16) Hybrid Fuel Systems Inc. 2004, '*Achieving Optimum Fuel Economy and Efficiency with Fuel 2^{TM} Dual-Fuel Conversion Systems*', Florida, United States of America, viewed 20 May 2005, <http://www.hybridfuelsystems.com/pdf/White-Paper-Dec-1-2004.pdf>

17) Indian Energy Sector 2000, '*Compressed Natural gas: one for the road*' , viewed 7 March 2005, <http://www.teriin.org/energy/cng.htm>

18) Info Comm 2005, United Nations Conference on Trade and Development, viewed 14 March 2005, <http://r0.unctad.org/infocomm/anglais/gas/characteristics.htm>

19) Kelley, M. & Coatsworth, J. n.d. , '*Cold Weather Problems f Diesel Engines*', Marine Design Center, United States of America, viewed 13 March 2005, <http://www.nap.usace.army.mil/mdc/newsletters/engine_wet.htm>

20) Kojima, M. 2001, 'Breathing Clean: Considering the Switch to Natural Gas Buses', World *Bank Technical Paper No. 516*, p.1-50, viewed 12 March 2005, <http://www.cleanairnet.org/infopool/1411/articles-33913_breathingclean_kojima.pdf>

21) Lewis, G. 2005, '*Compressed Natural Gas Vehicles*', Surrey Health Borough Council, viewed 18 March 2005, <http://www.surreyheath.gov.uk/surreyheath/environment.nsf/webPages/gasvehicles.html>

22) Ly, H. 2002, 'Effects of Natural Gas Composition Variations on the Operation, Performance, and Exhaust Emissions of Natural Gas-Powered Vehicles', International Association for Natural gas Vehicles Inc., viewed 13 March 2005, <http://www.iangv.org/jaytech/files/IANGVGasCompositionStudyDecember2002.pdf>

23) Murray, J., Lane, B., Lillie, K., McCallum, J. 2000, '*An Assessment of the Emission Performance of Alternative and Conventional Fuels*', viewed 15 March 2005, <http://www.cleanairnet.org/infopool/1411/articles-35613_assessment_emission.pdf>

24) Natural Gas Vehicle Coalition 2005, '*Natural Gas Vehicle Industry Strategy*', Natural Gas Vehicle Coalition, viewed 13 March 2005, <http://www.ngvc.org/ngv/ngvc.nsf/attached/strategy.pdf/$FILE/strategy.pdf>

25) Natural Gas.org 2004, viewed 21 March 2005, <http://www.naturalgas.org/overview/overview.asp>

26) Nett Technologies Inc. n.d., '*Frequently Asked Questions About Diesel Emission*', Toronto, Canada, viewed 23 May 2005, <http://www.nett.ca/faq_diesel.html>

27) NGV.org 2005, viewed 21 March 2005, <http://www.ngv.org/ngv/ngvorg01.nsf/bytitle/BenefitsofNGVs.htm>

28) P.C. McKenzie Company n.d. , P.C. McKenzie Company, Pittsburgh, US, viewed 20 March 2005, <http://www.mckenziecorp.com>

29) Pacific Gas and Electric Company 2003, Culver Company, Massachusetts, United States of America, viewed 14 March 2005, <http://www.pge.com/microsite/safety_esw_ngsw/ngsw/more/vehicles.html>

30) Petronas Dagangan Berhad 2005, viewed 17 March 2005, <http://www.mymesra.com.my/index.php?ch=24&pg=324&ac=2006>

31) Questar Gas n.d., Questar Corp., Salt Lake City, United States of America, viewed 16 March 2005, <http://www.questargas.com/AboutNaturalGas/AboutNaturalGas.html>

32) Shamsudin, A.H. & Yusaf, T. 1995, '*A Review on the Use of Natural Gas in Internal Combustion Engine*', ASEAN Journal Science Technology Development, p. 101-113

33) United States Environmental Protection Agency 2002, '*Clean Alternative Fuels: Compressed Natural Gas'*, viewed 16 March 2005, <http://www.epa.gov/otaq/consumer/fuels/altfuels/420f00033.pdf>

34) Shell Canada Limited n.d., '*Diesel Engines*', viewed 1 May 2005, <http://www.shell.ca/code/motoring/encyclopedia/diesel/diesel_engines.html>

35) Talal F. Yusaf, Mushtak Talib Ali Al-Atabi, D. R. Buttsworth (2003) Small Scale Power Generation Using A Dual Fuel System. *RERIC International Energy Journal, Vol. 19, No. 2*, *December*

36) The Concept IC Engine n.d., viewed 5 June 2005, <http://conceptengine.tripod.com/conceptengine/id3.html>

37) Turns, S.R. 1996, *An Introduction to Combustion: Concepts and Applications*, McGraw-Hill, Singapore

38) Wills, J. 2004, '*Cold Weather Engine Starting Tips*' University of Tennessee, United States of America, viewed 14 March 2005, <http://bioengr.ag.utk.edu/Extension/ExtProg/machinery/Articles/coldstrt.htm>

BIBLIOGRAPHY

1) ‘*CNG engine performance’* n.d. , pp.1-7 , viewed 29 March 2005, <http://www.arb.ca.gov/regact/cng-lpg/appe.doc>

2) Dursbeck, F., Erlandsson, L., Weaver, C. 2001, ‘*Status of Implementation of CNG as a Fuel for Urban Buses in Delhi*’, New Delhi, India, viewed 21 April 2005, <http://www.cleanairnet.org/infopool/1411/articles-35706_status_implementation.pdf>

3) ESI International 1999, ‘*Diesel Emission Control Strategies Available to the Underground Mining Industry*’, Washington DC, United States of America, viewed 4 June 2005,<http://www.deep.org/reports/esi_final_report.pdf>

4) Exxon Mobil Corporation 2002, viewed 29 March 2005, <http://www.exxon.com/USA-English/GFM/Products_Services/Fuels/Diesel_Fuels.asp>

5) French, T. M. 1990, ‘*Compressed Natural Gas Vehicles*’, Department of Natural Resources, Loisiana, United States of America, viewed 30 March 2005, <http://www.dnr.state.la.us/SEC/EXECDIV/TECHASMT/data/alternative/cng.htm>

6) JongWoo, K. n.d., ‘*Development of Direct Injection CNG Engine*’, Inha Technical College, Korea

7) National Transportation Library 2004, ‘*Clean Air Program: Summary Assessment of the Safety, Health, Environmental and System Risks of Alternative Fuel*’, viewed 21 April 2005, <http://ntl.bts.gov/DOCS/afrisks.html#toc>

8) Nicor Gas Inc. n.d., ‘*How does an NGV operate?*’, Nicor Inc., Illinois, United States, viewed 3 May 2005, <http://www.nicorinc.com/en_us/commercial/products_and_services/appliances/ngv_works.htm>

9) Ogawa, H., Miyamoto, N., Li, C., Nakazawa, S., Akao, K. 2001, ‘*Smokeless and Low NO_x Combustion in a Dual Fuel Diesel Engine with Induced Natural Gas as Main Fuel*’, Hokkaido University, Japan, viewed 20 May 2005, <http://powerlab.mech.okayama-u.ac.jp/~esd/comodia2001/3-06.pdf>

10) Papagiannakis, R.G., Hountalas, D.T. 2003, "*Combustion and Exhaust Emission Characteristics of a Dual Fuel Compression Ignition Engine Operated with Pilot Diesel Fuel and Natural Gas*', Athens, Greece, p. 2972-2987

11) Pimagro n.d., '*Dual-fuel Truck Fleet- Start Up Experience*', viewed 30 March 2005, <http://www.eere.energy.gov/afdc/pdfs/pimagro.pdf>

12) Rashidi, M n.d., '*Compressed Natural Gas Vehicles*', University of Shiraz, Iran, viewed 12 March 2005, <http://succ.shirazu.ac.ir/~motor/ngv1.htm>

13) Sigall, J 2000, '*Analysis of Alternative Fuel Technologies for New York City Transit Buses*', New York, United States of America, viewed 20 April 2005, <http://www.cleanairnet.org/infopool/1411/articles-35609_analysis_alternative.pdf>

14) Traver, M.L. n.d., '*Emission Formation in Compression Ignition Engines*', College of Engineering and Mineral Resources, West Virginia, United States of America, viewed 22 May 2005, <http://www2.cemr.wvu.edu/~englab/Tutorials/EmissTut/diesel.html>

15) Union Gas 2004, Union Gas Limited, Ontario, Canada, viewed 29 March 2005, <http://www.uniongas.com/aboutus/aboutng/ngv/ngvhistory.asp>

16) United States Environmental Protection Agency 2002, '*Clean Alternative Fuels: Compressed Natural Gas'*, viewed 29 March 2005, <http://www.epa.gov/otaq/consumer/fuels/altfuels/420f00033.pdf>

APPENDIX A

PROJECT SPECIFICATION

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

ENG 4111/4112 Research Project
PROJECT SPECIFICATION

FOR:	WONG, Wei Loon
TOPIC:	COMPRESSED NATURAL GAS AS AN ALTERNATIVE FUEL IN DIESEL ENGINES
SUPERVISORS:	Dr. Harry Ku (USQ) Dr. Fok Sai Cheong (USQ) Dr. Talal Yusaf (UNITEN)
ENROLMENT:	ENG 4111 – S1, X, 2005 ENG 4112 – S2, X, 2005
PROJECT AIM:	This project aims to investigate the effects of using a dual-fuel mixture of compressed natural gas (CNG) and diesel in a compression ignition (CI) engine performance
PROGRAMME:	**Issue B, 23rd April 2005**

a) Research the history of CNG usage worldwide and a literature review on the engine performance using CNG as the main fuel supply inclusive of the advantages and limitations.

b) Conversion of the current CI engine to install the CNG fuel system to enable the use of dual fuel diesel-CNG engine.

c) Study on the effect of using CNG as fuel in terms of power, torque, brake specific fuel consumption (BSFC), and thermal efficiency. Perform a comparison analysis on the dual fuel combustion and conventional diesel fuel.

d) Examine the emission data collected for both fuels and conduct feasibility study on CNG as a fuel alternative in terms of pollution and economy.

As time permits,

e) Conduct a Matlab program to effectively analyze the heat and mass transfer and emission data for comparison with experimental results

f) Develop a CFD FLUENT analysis on the CNG flow rate and its effects on engine performance

AGREED:

________________ (Student) ______________, ______________ (Supervisors)

__ / __ / __ __ / __ / __ __ / __ / __

APPENDIX B

ENGINE SPECIFICATIONS

Model	Y170F Vertical 4 Stroke Diesel Engine
Cooling	Air cooled single cylinder
Combustion System	Direct Injection
Bore	70 mm
Stroke	55 mm
Displacement	211 cc
Maximum Engine Speed	3600 rpm
Maximum Output	2.8 kW (3.755hp)
Continuous Output	2.5 kW (3.353hp)
Net Weight	27 kg
Dimensions	324 x 410 x 416 mm

Table B.1 *Engine Specifications*

APPENDIX C

GAS ANALYZER SPECIFICATIONS

Model	Autologic Autogas Portable 5 Gas Emission Analyzer with software
Part No	310-0120
Measuring Item	HC, CO, O_2, CO_2, and NO_x
Method	Non-dispersive Infrared (NDIR)
Range	HC 0-30000 ppm CO 0-15% O_2 0-25% CO_2 0-25% NO_x 0-5000 ppm
Resolution	HC 1 ppm CO 0.001 vol% O_2 0.01 vol% CO_2 0.01 vol% NO_x 1 ppm
Response Time	0-90 % of 8 seconds for NDIR measurements
Warm-up Time	Less than 2 minutes
Operating Temperature	0-50 °C
Humidity Level	Up to 95% (non-condensing)
Altitude	-300 to 2500 m
Vibration	1.5G Sinusoidal 5-1000 Hz
Power Supply	AC 90-230 V, 50-60 Hz

Table C.1 *Gas Analyzer Specifications*

APPENDIX D

EXPERIMENTAL DATA

Engine Speed	1600	1800	2000	2200	2600
Time for 10ml fuel (s)	78	69	64	55	58
Volume Flow Rate (m^3/s)	1.28×10^{-7}	1.45×10^{-7}	1.56×10^{-7}	1.82×10^{-7}	1.72×10^{-7}
Body Temp (℃)	124.6	140.8	149.6	165.3	177.9
Exhaust Temp (℃)	248.8	255.5	259.6	298.1	330.7
Torque (Nm)	7	9	9	9	7
Brake Power (kW)	1.1730	1.6967	1.8852	2.0737	1.9061
Mass flow rate (kg/s)	1.13×10^{-4}	1.28×10^{-4}	1.38×10^{-4}	1.60×10^{-4}	1.52×10^{-4}
Input Power (kW)	4.7949	5.4203	5.8438	6.8000	6.4483
BSFC (g/kW.hr)	346.2483	270.6052	262.5716	277.7617	286.5503
BMEP (kPa)	415.5844	534.3228	534.3228	534.3228	415.5844
Efficiency	24.4639	31.3024	32.2601	30.4959	29.5606

Table D.1 *Engine Performance Data for Diesel Fuel under Maximum Load Operating Conditions*

Engine Speed	1600	1800	2000	2200	2600
Time for 10ml fuel (s)	290	249	230	202	272
Diesel Mass flow rate (kg/s)	3.03×10^{-5}	3.53×10^{-5}	3.83×10^{-5}	4.36×10^{-5}	3.24×10^{-5}
Body Temp (℃)	93.9	98.4	125.9	144.7	150.2
Exhaust Temp (℃)	225.8	237.4	240.9	287.5	299.8
Torque (Nm)	8	10	10	10	8
Brake Power (kW)	1.3406	1.8852	2.0947	2.3041	2.1785
CNG Mass flow rate (kg/s)	1.35×10^{-4}	1.54×10^{-4}	1.66×10^{-4}	1.92×10^{-4}	1.74×10^{-4}
Input Power (kW)	4.7949	5.4203	5.8438	6.8000	6.4483
BSFC (g/kW.hr)	287.3534	231.0534	224.2024	237.1270	237.4353
BMEP (kPa)	474.9536	593.6920	593.6920	593.6920	474.9536
Efficiency	27.9588	34.7804	35.8446	33.8843	33.7835

Table D.2 *Engine Performance Data for Dual-Fuel under Maximum Load Operating Conditions*

Engine Speed	Diesel					Dual Fuel				
	CO (ppm)	HC (ppm)	NO_X (ppm)	CO_2 (%)	O_2 (%)	CO (ppm)	HC (ppm)	NO_X (ppm)	CO_2 (%)	O_2 (%)
1600	122	758	70	2.37	17.56	36	4015	15	1.29	15.68
1800	117	566	96	2.99	17.25	42	3657	26	1.68	13.14
2000	83	922	155	3.15	14.32	55	3094	50	2.46	12.74
2200	88	1150	180	5.35	12.87	68	2816	78	3.17	10.78
2600	99	1196	106	4.18	10.47	74	2420	91	3.64	10.62

Table D.3 *Exhaust Data Comparison for Diesel and Dual-Fuel under Maximum Load Operating Conditions*

Engine Speed	1600	1800	2000	2200	2600
Time for 10ml fuel (s)	130	80	71	59	68
Volume flow rate (m^3/s)	7.69×10^{-8}	1.25×10^{-7}	1.41×10^{-7}	1.69×10^{-7}	1.47×10^{-7}
Body Temp (℃)	71.3	79.8	86.8	105.4	121.9
Exhaust Temp (℃)	125.7	153.6	187.5	201.5	218.8
Torque (Nm)	4	6	6	7	6
Brake Power (kW)	0.6703	1.1311	1.2568	1.6129	1.6338
Mass flow rate (kg/s)	6.77×10^{-5}	1.10×10^{-4}	1.24×10^{-4}	1.49×10^{-4}	1.29×10^{-4}
Input Power (kW)	2.8769	4.6750	5.2676	6.3390	5.5000
BSFC (g/kW.hr)	363.5607	350.0955	355.0264	332.9105	285.1456
BMEP (kPa)	237.4768	356.2152	356.2152	415.5844	356.2152
Efficiency	23.2990	24.1951	23.8590	25.4440	29.7062

Table D.4 *Engine Performance Data for Diesel Fuel under Moderate Load Operating Conditions*

Engine Speed	1600	1800	2000	2200	2600
Time for 10ml fuel (s)	462	354	340	322	300
Diesel Mass flow rate (kg/s)	1.90×10^{-5}	2.49×10^{-5}	2.59×10^{-5}	2.73×10^{-5}	2.93×10^{-5}
Body Temp (℃)	58.7	64.5	78.6	100.1	115.9
Exhaust Temp (℃)	105.9	119.6	135.4	148.6	168.0
Torque (Nm)	5	7	7	8	7
Brake Power (kW)	0.8379	1.3196	1.4663	1.8433	1.9061
CNG Mass flow rate (kg/s)	8.19×10^{-5}	1.27×10^{-4}	1.42×10^{-4}	1.67×10^{-4}	1.50×10^{-4}
Input Power (kW)	2.8769	4.6750	5.2676	6.3390	5.5000
BSFC (g/kW.hr)	275.9697	284.2620	288.1396	275.6569	231.5298
BMEP (kPa)	296.8460	415.8544	415.8544	474.9536	415.8544
Efficiency	29.1237	28.2276	27.8355	29.0789	34.6572

Table D.5 *Engine Performance Data for Dual-Fuel under Moderate Load Operating Conditions*

Engine Speed	Diesel					Dual Fuel				
	CO (ppm)	HC (ppm)	NO_X (ppm)	CO_2 (%)	O_2 (%)	CO (ppm)	HC (ppm)	NO_X (ppm)	CO_2 (%)	O_2 (%)
1600	68	1984	65	2.19	19.04	14	4857	18	1.25	18.99
1800	72	1776	87	2.47	18.41	29	4215	25	1.58	17.45
2000	79	1682	105	2.38	17.24	38	3259	29	1.97	17.69
2200	89	1402	119	3.55	17.09	45	2940	38	2.01	16.47
2600	97	1198	137	4.12	15.78	49	2581	59	2.29	16.30

Table D.6 *Exhaust Data Comparison for Diesel and Dual-Fuel under Moderate Load Operating Conditions*

APPENDIX E

SAMPLE ANALYSIS CALCULATIONS

Calculations for engine performance with engine running at maximum operating conditions:

A. FOR DIESEL FUEL

At 1600 rpm,

Time for 10 ml of diesel = 78s

Volume flow rate $= \dfrac{10\times10^{-3}\times10^{-3}\,m^3}{78s}$

$= \underline{1.282\times10^{-7}\,m^3/s}$

Density of diesel = 880 kg/m^3

Mass flow rate $= 1.282\times10^{-7}\,m^3/s$ x 880 kg/m^3

$= \underline{1.128\times10^{-4}\,kg/s}$

a) Input Power

$$IP = \dot{m}_f \times Q_{HV} \times 10^3$$

where

IP	=	input power	(kW)
$\dot{m}_f$	=	mass flow rate of fuel	(kg/s)
Q_{HV}	=	lower calorific value of fuel	(MJ/kg)

For diesel fuel, $Q_{HV,Diesel}$ = 42.5 MJ/kg

For CNG fuel, $Q_{HV,CNG}$ = 45 MJ/kg

Input Power $= 1.128\times10^{-4}\times42.5\times10^3$

= **4.794 kW**

b) Torque

The torque is directly measured from the electronic control board mounted to the dynamometer.

$$\text{Torque},\ \tau = \underline{\mathbf{7.0\ Nm}}$$

c) Brake Power

$$\text{Brake Power, BP} = \frac{2\pi N\tau}{60}$$

where

τ	=	torque (N. m)
BP	=	power developed by engine (W)
N	=	engine speed (rpm)

$$\text{BP} = \frac{2\pi \times 1600 \times 7.0}{60}$$

$$= \underline{\mathbf{1.173\ kW}}$$

d) Specific Fuel Consumption

$$sfc = \frac{\dot{m}_f}{P} \times 3.6 \times 10^6$$

where

sfc	=	specific fuel consumption (g/kW. hr)
$\dot{m}_f$	=	mass flow rate of fuel (kg/s)
P	=	power output (kW)

$$\text{sfc} = \frac{1.128\times10^{-4}}{1.173}\times3.6\times10^{6}$$

$$= \underline{\mathbf{346.2\ g/\ kW.hr}}$$

e) <u>Brake Mean Effective Pressure</u>

$$bmep = \frac{BP\times n_R\times60}{A\times L\times N\times n}$$

where

bmep	=	brake mean effective pressure (kPa)
BP	=	brake power (kW)
n_R	=	number of crank revolutions for each power stroke per cylinder
A	=	area of engine bore (m^2)
L	=	length of engine stroke (m)
N	=	engine speed (rpm)

$$\text{bmep} = \frac{1.173\times2\times60}{\frac{\pi}{4}\times(0.07)^2\times0.055\times1600\times1} \quad \text{(for single cylinder)}$$

$$= \underline{\mathbf{415.63\ kPa}}$$

f) <u>Overall Efficiency</u>

$$\text{Efficiency} = \frac{BP}{IP}$$

$$= \frac{1.173}{4.794}$$

$$= 0.245$$

$$= \underline{\mathbf{24.5\%}}$$

B. **<u>FOR DUAL FUEL</u>**

At 1600 rpm,

Time for 10 ml of diesel pilot fuel = 290 s

Volume flow rate of diesel $= \dfrac{10\times10^{-3}\times10^{-3}\,m^3}{290s}$

$= \underline{3.448\times10^{-8}\,m^3 / s}$

Density of diesel = 880 kg/m^3

Mass flow rate of diesel $= 3.448\times10^{-8}\,m^3 / s$ x 880 kg/m^3

$= \underline{3.034\times10^{-5}\,kg / s}$

Diesel ratio $= \dfrac{\text{Diesel mass flow rate in dual fuel}}{\text{Diesel mass flow rate in pure diesel}}$

$= \dfrac{3.034\times10^{-5}\,kg / s}{1.128\times10^{-4}\,kg / s}$

= <u>0.269</u>

CNG Ratio = 1 - 0.269

= <u>0.731</u>

a) <u>Input Power</u>

The input power is taken to be the same for both diesel fuel and dual fuel since the engine is operating under the same speed and load conditions.

$\therefore$ Input power = **<u>4.794 kW</u>**

From the input power, the mass flow rate of CNG can be calculated:

$$\left[\left(\text{Diesel ratio}\times \dot{m}_{f,Diesel}\times Q_{HV,Diesel}\right)+\left(\text{CNG Ratio}\times \dot{m}_{f,CNG}\times Q_{HV,CNG}\right)\right]\times 10^{3}=4.794$$

$$\left[\left(0.269\times 3.034\times 10^{-5}\times 42.5\right)+\left(0.731\times \dot{m}_{f,CNG}\times 45\right)\right]\times 10^{3} = 4.794$$

$$\dot{m}_{f,CNG} = \underline{1.352\times 10^{-4}\, kg/s}$$

b) <u>Torque</u>

The torque is directly measured from the electronic control board mounted to the dynamometer.

Torque, τ = <u>**8.0 Nm**</u>

c) <u>Brake Power</u>

$$BP = \frac{2\pi N\tau}{60}$$

$$= \frac{2\pi\times 1600\times 8.0}{60}$$

= <u>**1.340 kW**</u>

d) <u>Specific Fuel Consumption</u>

$$sfc = \frac{\dot{m}_f}{P}\times 3.6\times 10^{6}$$

$$sfc = \left[\left(\frac{0.269\times 3.034\times 10^{-5}}{1.340}\right)+\left(\frac{0.731\times 1.352\times 10^{-4}}{1.340}\right)\right]\times 3.6\times 10^{6}$$

= <u>**287.44 g/kW.hr**</u>

e) <u>Brake Mean Effective Pressure</u>

$$\text{bmep} = \frac{BP \times n_R \times 60}{A \times L \times N \times n}$$

$$\text{bmep} = \frac{1.340 \times 2 \times 60}{\frac{\pi}{4} \times (0.07)^2 \times 0.055 \times 1600 \times 1} \quad \text{(for single cylinder)}$$

= **<u>474.81 kPa</u>**

f) <u>Overall Efficiency</u>

$$\text{Efficiency} = \frac{BP}{IP}$$

$$= \frac{1.340}{4.794}$$

= 0.2795

= **<u>27.95%</u>**