5-2008

Measurements and Control of Noise Pollution from Dual Fuel Engine

Abdullah Ahmed Jawi Al Aseery

United Arab Emirates University
Deanship of Graduate Studies
M. Sc. Program in Environmental Sciences

Measurements and Control of Noise Pollution from Dual Fuel Engine

By

Abdullah Ahmed Jawi Al Aseery

A Thesis Submitted to
United Arab Emirates University
In Partial Fulfillment of the Requirements
For the Degree of M.Sc. in Environmental Sciences

Supervisors

Dr. Mohamed Younes E. Selim Associate Professor Department of Mechanical Engineering College of Engineering UAE University	Dr. Salah Al-Omary Associate Professor Department of Mechanical Engineering College of Engineering UAE University

2008

The Thesis of **Abdullah Ahmed Al Aseery** for the Degree of Master of Science in Environmental is approved.

....

Examining Committee Member, **Dr. Mohamed Younes**

M. A. Gadalla

....

Examining Committee Member, **Dr. Mohamed Gadalla**

May 7, 2008

....

Examining Committee Member, **Dr. Emad Elnajjar**

....

Program Director, **Dr. Tarek Youssef**

....

Assistant Chief Academic Officer for Graduate Studies, **Prof. Ben Bennani**

United Arab Emirates University
2007/2008

DEDICATION

To the persons who supported me without limitations.

To the persons who means everything

To my great parents

To the source of my power and pride

To my wife

ACKNOWLEDGEMENTS

First of all, gratitude and thanks to Allah who always helps and guides me.

I would like to extend my sincere thanks and graduate to my country and the great university that I have studied in for several years under the leadership Sheikh Nahyan Bin Mubarak Al- Nahyan

The author is greatly indebted to Dr. Mohamed Younes E. Selim Department of Mechanical Engineering, College of Engineering, for suggesting this line of research, kind supervision and systematic guidance. With his generous and fruitful discussions this work has been completed.

The author is deeply grateful to Dr. Salah Al-Omary , Department of Mechanical Engineering, College of Engineering, UAEU, for his supervision, valuable advices and paternal encouragement.

I would like to express my thanks to Eng. Mohamed El-Saeed Department of Mechanical Engineering, College of Engineering, for his great help and discussions he offered during the preparation of this thesis and for his valuable encouragement and follow up through this work.

ABSTRACT

This study examines the effects of adding the water to the intake air of a dual fuel engine on the performance and combustion noise. Dual fuel engine is a diesel engine using small amount (5%) liquid diesel fuel as a pilot fuel and burns a liquefied petroleum gas as the main fuel. The addition of water to the combustion chamber has known effects on reducing the dangerous emission of nitrogen oxides. Previous research has attributed the observed reduction of nitrogen oxide emissions to a suppression of flame temperature due to quenching effects from the water, thereby reducing thermal NO_x formation.

The thesis highlights the effects on the performance, CO/HC/Smoke exhaust emission and combustion noise of the dual fuel engine. Experimental procedures conducted using a Ricardo diesel version variable compression research engine are discussed. Results from testing dual fuel engine with varying the added water to fuel ratios and other design and operating parameters will be presented and discussed.

The data shows slight decrease in the power output with increasing the amount of water added. This drop can be recovered by reducing the compression ratio of the engine. The addition of water also shows an

increase in the combustion noise, however the noise can be decreased by increasing the engine speed, reducing the pilot fuel mass, retarding the injection timing or reducing the compression ratio. CO and HC emissions have shown slight increase but still within the accepted range and can be reduced further by a catalytic converter. Exhaust opacity has shown slight increase with adding more water; however it can be also reduced by reducing the mass of pilot fuel.

Recommendations are given for further studies, including the use of CFD simulation and the use of more water amounts.

LIST OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

Symbol	Meaning	Units
EGR	*Exhaust Gas Recycle*	
NO_x	*Nitrogen Oxides*	
NO	*Nitrogen Oxide*	
NO_2	*Nitrogen Dioxide*	
LPG	*Liquefied Petroleum Gas*	
CFR	*SEE PAGE 19*	
CO	*Carbon Monoxide*	
CO_2	*Carbon Dioxide*	
CI	*Compression Ignition*	
IDI	*In-Direct Injection*	
CR	*Compression ratio*	
ACU	*Automatic Control Unit*	
KTK	*Type of Thyristor converter unit*	
PM	*Particulate Matter.*	
SO	*Sulfur oxide*	
SO_2	*Sulfur Dioxide*	
HC	*Hydrocarbons*	
DAS	*Data Acquisition System*	
DE	*Diesel Emissions*	
PAHs	*Polycyclic Aromatic Hydrocarbons*	
bsfc	*Brake Specific Fuel Consumption*	
VOCs	*Volatile Organic Compounds*	
GAC	*Granular Activated Carbon*	
Pt	*Platinum*	
Rh	*Rhodium*	
Pd	*Palladium*	
SCR	*Selective Catalytic Reduction*	

NSCR	*Non-Selective Catalytic Reduction*	
VOCs	*Volatile Organic Compounds*	
HCCI	*Homogeneous Charge Compression Ignition*	
GAC	*Granular Activated Carbon*	
NTPD	*Non-thermal Plasma Discharge*	
HNO_3	*Nitric Acid*	
NaOH	*Sodium Hydroxide*	
$NaNO_3$	*Sodium Nitrate*	
μMAC	*Mu Mac Basic*	
BTDC	*Before Top Dead Center*	
WOS	*Work Station Operating System*	
CR	*Compression Ratio*	
$\dot{m}_w$	*Mass Flow Rate of Water*	*kg/sec*
$\dot{m}_{air}$	*Mass Flow Rate of Air*	*kg/sec*
$\dot{m}_{fuel}$	*Mass Flow Rate of Fuel*	*kg/sec*
N	*The rev./sec of the EngineSpeed*	*Rps*
Pi	*The Combustion Pressure*	*bar*
T	*Torque*	*N.m*
P	*Power*	*Watt*
ACU	*Automatic Control Unit*	
RCU	*Remot Central Unit*	
PSU	*Power Supply Unit*	
TDC	*Top Dead Center*	
CFD	*Computational Fluid Dynamics*	
θ	*Angle*	*Degree*

CHAPTER I

INTRODUCTION

1.1 History of internal combustion engines exhaust emission

During the course of the past six decades, the internal combustion engine has been under ever tighter scrutiny, due to its role as a major source of air pollution. The smog problem due to automobile traffic was first observed in the early 1940s in the Los Angeles basin. It has been shown that the smog problems in Los Angeles were caused by reactions between oxides of nitrogen and hydrocarbon compounds in the presence of sunlight. It was later concluded that the oxides of nitrogen and the hydrocarbons, causing the photochemical smog, originated from combustion in automotive engines.

In 1959, the state of California legislature took, as a consequence of the smog problems in Los Angeles, the first legislative steps towards a reduction of the automotive air pollution, by setting emissions standards and regulations for automotive engines. The state of California initiative was followed, first by the federal government of the United States, then by Japan, Europe, and the rest of the world. Ever since this first initiative, the state of California has been leading the way, in terms of setting ever stricter standards on vehicle pollutant emissions.

The two-way catalytic converter (oxidizing catalytic converter) was introduced in 1975 as a means of reducing tailpipe emissions of hydrocarbons and carbon monoxide. In 1976 Volvo introduced its "smogfree" 1977 model year car, as the first car featuring a three way catalytic converter (oxidizing/reducing catalytic converter), able to drastically reduce tailpipe emissions of all three controlled pollutants; hydrocarbons, carbon monoxide, and oxides of nitrogen.

There have been many methods to control the air pollution problem from diesel engines. The use of exhaust gas recycle (EGR), adding water to the combustion chamber with any method, and adjusting the engine design parameters helped to reduce the nitrogen oxides NO_x problem.

The use of gaseous fuels e.g. natural gas, liquefied petroleum gas (LPG), etc. is a promising approach for lowering the dependence on petroleum liquid fuels and to reduce the emissions of CO_2 and other pollutants from the diesel engines. For a country like the United Arab Emirates, it is advantageous to utilize gaseous fuels in diesel engines. The advantages of using gaseous fuels in such engines stem from the economical and environmental benefits. Economical benefits are to reduce the dependence on liquid fuels and utilizing the gaseous fuels locally so that liquid petroleum oil can be exported abroad. Environmental benefits include the reduction of CO_2 green house gas, the hydrocarbons and diesel odor. However, the NO_x emission is still very high and needs to be reduced if such engines are to be commercialized.

The use of gaseous fuels in diesel engines can be achieved by two ways. The first way is to replace the injection system with spark ignition while the other is to use dual fuel concept. In dual fuel engine, the injected diesel fuel is kept at minimum amount; say 5-10% of the maximum, and the gaseous fuel is admitted as the main fuel in the intake manifold with special nozzles.

It has been concluded from diesel engines combustion that the addition of water in the combustion chamber effectively reduces the NO_x emission but there is a concern that it may increase the noise level from such engines.

1.2 Objectives of the thesis

There have been many efforts to reduce the NO_x emission problem from dual engines, but the addition of water to the combustion chamber has not been tried before. Therefore it is the main objective of the current project to examine the effect of adding water to intake of the dual fuel engine on the performance and combustion noise of such engine. The diesel engine used here is converted to run on gaseous fuel such as liquefied petroleum gas (LPG) and the water has been added as a water vapor in the intake air admitted to the engine; hence increasing the absolute humidity of the air. The absolute humidity is then increased compared to the base atmospheric case and the effects of adding water on the performance and exhaust emissions of the dual fuel engine have been tested. An optimization has been carried out to select the most appropriate water to air ratio, water to fuel ratio, operating and design parameters for the engine to produce the best conditions. The best conditions here mean the engine should produce the highest power output, with minimum exhaust emission and minimum combustion noise.

1.3 Outline of the thesis

The following chapters of the thesis are: chapter two which covers the literature review, chapter three which presents the experimental data and engine test rig description, the results and analysis is presented in chapter four. Finally the thesis is ended with chapter five which covers the conclusions and recommendations for future works.

CHAPTER II

LITERATURE REVIEW

2.1 Introduction

A literature review is given in this chapter which covers the topics discussed in this thesis. The topics discussed in the thesis include the importance of the internal combustion engines to engineering applications. We also present exhaust emission problem from the internal combustion engines utilizing the diesel fuels in that chapter. In this chapter we also highlight the importance of utilizing the gaseous fuels in these engines and emissions from these engines. The methods of controlling these pollutants are also covered. Finally, the approach used in this thesis of utilizing steam injection to the admitted air to reduce the emissions (or adding moisture in the intake air) is explored.

2.2 Exhaust emissions from diesel engines

Compression ignition (CI) engines, commonly known as diesel engines, are considered to be the units of the transportation and energy sectors in the world. Millions of units are used on a daily basis and the emission of CI engines is an important aspect of pollution source. CI engines are designed to operate on a less refined petroleum distillate than gasoline which contains less energy content. Yet CI engines are more efficient at translating fuel energy content into output than gasoline (engine Spark Ignition). CI engines also have fewer overall emissions than gasoline engines and are preferred for emissions reductions [1]. Unfortunately, lagging technological improvements in CI engines and fuel have hindered emissions reductions, performance and acceptance of these engines. In order to reduce vehicular pollution a change in engine type and design for the majority of the transportation sector is needed. By switching to new, efficient diesel engines, substantial reductions in ambient pollution may be achieved. Compared to gasoline engines, diesel engines produce 30% as much hydrocarbon, 5% as much carbon monoxide and about 50% as much particulate emissions. Utilizing diesel engines in a hybrid electric configuration can double these reductions in emissions. Exhaust equipment and fuel modification or substitution can further these reductions to an even greater extent. All of these technologies and improvements are currently available and proven. Increased initial costs can easily be offset by substantially decreased operating costs, improved environmental and health benefits and mass production. By optimizing fuel type, engine design, exhaust equipment and power-train systems vehicular pollution can be effectively regulated and minimized.

The major emissions components of compression ignition engines are particulate matter (PM) and nitrogen oxides (NO_x). Other emissions include carbon monoxide (CO), sulfur oxides (SO), hydrocarbons (HC) and noise. Unburned hydrocarbons are responsible for many of the severe health problems and are linked to caner, mutations and toxicity. While PM is responsible for smog and respiratory problems. One group of hydrocarbons, Aldehydes, are responsible for the odor constituent from diesel engines. Minimization of PM and NO_x is the primary focus of emissions reduction work, yet reducing all emissions is important.

NO_x contribute to acid rain and reduce aesthetic air quality. CI engines operate at a much leaner air fuel ratio than gasoline engines which facilitates lower CO and volatile organic compounds. However, due to higher operating temperature and pressure in diesels, NO_x and PM emissions are higher [2].

There has been probably more health research on diesel emissions (DE) than on any other single source, possibly excepting cigarette smoke. Emissions from diesels are a mixture of gases, vapors, semi-volatile organic compounds and particles. The particles are all respirable and fall into two general chemical classes: 1) "soot" or elemental carbon particles coated with condensed organic and inorganic compounds, and 2) ultra fine particles of condensed organic material and sulfur compounds having little or no elemental carbon content [3]. Marine diesel engine emissions such as particulate matter and black smoke carry carcinogen components that significantly impact the health of human beings [4].

2.2.1 Engine Design and Tuning

Engine design and tuning are the primary variables in emission control. There are two types of CI engines that operate in a similar manner. but have different exhaust emissions profiles. Direct injection (DI) is the newer. more powerful Cl engine design used for high speed engines. The traditional. more w [illegible] s [illegible] s indirect injection (IDI). The primary physical difference in these engines is the IDI engine has a pre-combustion chamber and typically is designed to use glow-plugs to facilitate cold starting conditions.

The main benefit of the in-direct injection design is that it facilitates better fuel air mixture. Air and fuel are injected into the pre-combustion chamber of an IDI engine. The downward movement of the piston during the inlet stroke and the upward movement of the piston during the compression stroke force the air and fuel to mix in a swirl pattern that is created by the changing volume of the combustion chamber. The velocity of the swirl has been measured at over 21 times the velocity of the engine speed [2]. The resulting through mixing of the fuel and air in the pre-combustion chamber results in more complete combustion. The result is that there is less smoke (particulate matter). carbon monoxide [illegible] s [illegible] s [illegible] Aldehydes produced than in an equivalent direct injection engine.

IDI and DI engines also employ different fuel injection systems. IDI engines use a split rail system to deliver fuel from the fuel injection pump to the fuel injectors. DI engines use a common rail system that allows more precise and higher pressure fuel control and injection. Fuel in IDI engines is typically injected once per compression stroke at pressures up to 3.000 psi. While DI engines often inject fuel at anywhere from 5.000 to

25,000 psi and some of the more sophisticated fuel injection systems inject the fuel multiple times per compression stroke to enhance fuel mixing and flame characteristics. This facilitates fewer emissions from incomplete combustion.

The design and placement of the fuel injectors are also important components of exhaust emission control. The smaller the opening of the injector, the more the fuel atomizes upon injection. Atomization of fuel affects the mixing of fuel and air as well and the duration of the flame, ultimately affecting the completeness of fuel burn. The fuel injection nozzle width and length also affect atomization. For optimum atomization of fuel, it is necessary to inject the fuel at a high pressure as possible to maximize the spray area and air saturation, but it is necessary to minimize the amount of fuel contacting the cylinder wall as it reforms into droplets where it is less prone to vaporization.

The timing and rate of fuel injection are also very important factors of emissions control. As the injection is increased the ignition delay (the time between the fuel is being injected and combustion begins) is getting higher. This increases the total amount of fuel that is injected before ignition begins. The longer ignition delay also heats the fuel to higher temperatures which ends the combustion process earlier. This increases the residence time of the fuel and exhaust gases in the combustion chamber, which effects the emissions formation. The longer the exhaust gases are trapped in the combustion chamber the hotter they become. This allows for the recombination and formation of polycyclic aromatic hydrocarbons. Smoke intensity is also reduced but combustion noise, higher mechanical and thermal stresses and increased NO emissions result. Delayed injection timing has the opposite effect as exhaust gases are cooler when released. The rate of injection is also an important factor. High initial rates of injection reduce exhaust smoke because the injection process is ended earlier which improves elimination reactions, through better fuel air mixing.

Combustion in a compression ignition engine is initiated by a sharp increase in pressure which creates high temperatures and leads to ignition. The nuclei of fuel particles ignite first, followed by the rest of the fuel particle. Some fuel is not burnt because it is not mixed with oxygen. Once the fuel is injected it is compressed until it ignites. This is called the ignition delay. The ignition delay affects the rest of the combustion process, the mechanical and thermal stresses, noise, and exhaust emissions. Many reactions take place during the ignition delay and are broken into two categories, physical and chemical processes. The physical processes include:

1) Spray disintegration and droplet formation.

2) Heating of the liquid and fuel evaporation and

3) Diffusion of the vapor into the air to form a combustible mixture.

The chemical processes include:

1) Decomposition of the heavy hydrocarbons into lighter components and

2) Chemical reactions between the decomposed particles and oxygen.

The chemical and physical processes occur simultaneously as the fuel vapor makes contact with the air. The chemical processes are actually the rate controlling processes for combustion. Peroxides and Aldehydes are formed during the ignition delay and reach their peak concentrations just before combustion [5].

2.2.2 Constituents of Exhaust Emissions

Unburned hydrocarbons in the diesel exhaust consist of either original or decomposed fuel molecules, or recombined intermediate compounds. Some hydrocarbons are the result of lost lubricating oil. At high loads the hydrocarbon emissions originate from the fuel molecules in the core and on the cylinder walls. Under these conditions the temperatures reached are fairly high and cause decomposition of some of the original fuel molecules. Since the fuel/air ratio in the core and near the walls is generally rich, there is a great possibility that some recombination reactions may occur between the hydrocarbon radicals and the intermediate compounds. The result is higher concentrations of the heavier hydrocarbons. The process of recombination of the hydrocarbon compounds and radicals may also result in compounds having a different structure than the original fuel [6].

Carbon monoxide is formed during intermediate combustion stages of hydrocarbon fuels. During combustion, CO is oxidized to CO2 through combination reactions between CO and different oxidants. If these recombination reactions are incomplete, CO will be left. Higher temperatures reduce the amount of CO that is not oxidized. However, as combustion temperatures increase, available oxygen decreases and CO is left unoxidized. Therefore, CO levels typically are low at low loads, but become higher at higher loads, due to failure to be oxidized.

Smoke is the most visible type of emission emitted from CI engine exhaust. Smoke consists of different types of particulate matter, which vary with load. They can be divided into two categories:

1) Liquid particles appearing as white clouds of vapor emitted under cold starting, idling and low loads. These particles consist of mainly unburned fuel and lubricating oil and may be accompanied by partial oxidation products and disappear as load is increased.
2) Soot or black smoke is emitted as a product of the incomplete combustion process, particularly at maximum loads [7].

Smoke is measured as the opacity of the exhaust. The maximum opacity for new diesel engines is regulated by the EPA at 0.01 grams per brake horsepower per hour (g/bhp-h) for particulate matter less than 10 microns in size (PM10) (U.S. EPA, 2007). Smoke is affected by the ignition delay. Later ignition leads to greater amounts of smoke in the emissions. The Cetane number of fuel is a measure of the ignition delay. The higher the Cetane number, the shorter the ignition delay and less smoke is produced. Regulation of particulate matter is important because it causes health problems. PM from 0.1 microns to 10 microns can penetrate deep into the lungs where they are deposited in the respiratory bronchioles or alveolar sacs. These deposits have been shown to increase cancer formation in humans and lab animals [8]. An interesting factor that occurs in the exhaust of CI engines is that the amount of nitrogen and sulfur absorbed by the particulate matter fluctuates depending on the exhaust temperature. Higher exhaust temperatures, which correlate with higher engine loads, will absorb more sulfur and less nitrogen than lower temperatures [6].

Nitric oxide (NO_X) is formed during the combustion process at various concentrations in all the spray regions. NO results from the disassociation of oxygen molecules into atomic oxygen as a result of high combustion temperatures. Nitrogen attaches to atomic oxygen after combustion has occurred and forms NO and other Nitrogen-Oxygen isomers (NO_X) [9]. NO_X emissions increase with cetane number due to longer ignition delay and higher combustion temperature [10]. NO emissions are regulated by the EPA at 0.02 g/bhp-h [11].

Sulfur dioxide (SO_2) is formed by the oxidation and combustion of sulfur throughout the combustion process. SO_2 is a result of the amount of sulfur in the fuel. New diesel fuel standards require that sulfur be limited to 15ppm to minimize SO_2 pollution [11]. SO_2 causes acid rain and low level smog formation. It is also linked to respiratory problems. Polycyclic aromatic hydrocarbons (PAHs) are byproducts of the incomplete combustion of organic matter. They are of major health concern, due to their well-known carcinogenic and mutagenic properties. The presence of PAHs in engine emissions is not determined solely by the presence of PAHs in fuel. PAHs can be formed by a mixing of various exhaust gases in the atmosphere, a catalytic reaction with exhaust equipment, or from the combustion of lubricants and other fluids present in the engine [12]. PAHs are normally associated with small particles that generally have long residence times in the atmosphere and therefore have the potential to be transported quite long distances [13]. While the exhaust may not contain a certain PAH, such as nitrobenzo(a)pyrene, a carcinogen, it may contain benzo(a)pyrene, which is not a known carcinogen. However, when exposed to NO_2 and trace amounts of nitric acid in the atmosphere, the benzo(a)pyrene forms nitrobenzo(a)pyrene [12]. Another interesting change that takes place is that the benzo(a)pyrene that is emitted as part of the exhaust is a promutagen, it needs activation to cause mutation. However, the nitrobenzo(a)pyrene is a direct mutagen and requires no chemical activation to cause mutations on the Ames test (a test used to determine mutagenic activity) [14]. Typical PAH constituents of diesel exhaust are chrysene(21%), pyrene (18%), and benzo[a]anthracene (17%); all of which are considered mutagenic [13]. Surprisingly, PAH emissions are not specifically regulated, emissions of nonmethane hydrocarbons are limited by the EPA at .14 g/bhp-h [11].

2.2.3 Fuel Variables

The components of fuel are one of the main determinants of the amount and type of pollutants that will be present in exhaust emissions. Sulfur is one of the most problematic compounds found in diesel fuel currently. Sulfur is oxidized in the combustion process and forms various isomers. These isomers, such as sulfur dioxide, are responsible for acid rain, respiratory problems and other and environmental problems. Sulfur and sulfates are detrimental to catalytic emissions reduction equipment. They bind to the surfaces of the exhaust equipment and render it useless for the catalytic reduction of other pollutants. New EPA standards for on-road diesel require a reduction from .05 g/bhp-h to .01 g/bhp-h. However, sulfur is responsible for most of the lubricating quality of diesel fuel, which is essential for certain engine components such as the fuel injection pump. Olefins and aromatics are being used for lubrication, replacing sulfur content in diesel fuel. As noted earlier olefins and PAHs are known to be mutagenic and carcinogenic [15]. Essentially, a pollutant is being replaced by cleaner, but more dangerous compounds.

Nitrogen, carbon, phosphorus and other elements as well as trace amounts of many metals are often constituents of diesel fuels. By limiting the amount of such compounds in fuels, overall pollution from CI emissions can be controlled.

There are many different pieces of equipment that have an impact on emissions formation from compression ignition engines. Equipment attaches to the engine or the exhaust and modifies the way that exhaust gases are formed. Most of these technologies are proven, but not widely used for various reasons. A change in fuels with a focus on emissions reduction may include the use of some or all of these technologies. Also, some of these technologies are not used because they are not compatible with current fuels, but may be compatible with WVO.

2.3 Control Technologies

Technologies to control air pollutant emissions from ICEs can be categorized as process or post-combustion controls. Process controls include changes and improvements to the combustion chamber, fuel/air delivery, and engine components aimed at reducing air pollutant emissions. For example, Computational Fluid Dynamics (CFD) analysis and design has been used to improve and air intake valve ports to enhance fuel-air mixing. Post-combustion controls include catalytic converters and other technologies applied to react with combustion exhaust constituents including NO_x, CO, HCs, and particulate.

2.3.1 Combustion Modifications

Exhaust gas recirculation (EGR) is commonly employed to reburn combustion by-products, especially CO and particulate. EGR also dilutes the intake air oxygen concentration, increasing the heat capacity of the combustion products per unit of heat release, lowering the combustion flame temperature [16]. The authors of [17] reported 50% NO_x reduction at 20% EGR, but CO emissions doubled and fuel economy decreased by 8%. Exhaust gas recirculated on diesel engines must be well-filtered to prevent fuel sulfur and exhaust particulate from eroding and corroding engine intake valves, cylinders, and pistons. It has been also found that when hot EGR is used the amount of NO was decreased, but this is also increases the amount of CP, HC and increases the brake specific fuel consumption (BSFC) [8].

Adjusting fuel injection timing is an effective method for decreasing NO_x emissions. Traditional practice is to delay fuel injection into the combustion chamber to lower the final flame temperature, but this generally results in higher unburned hydrocarbon emissions. Yanagihara [19] demonstrated that the reduced NO_x production by shortening the injection duration while advancing the fuel injection. His results are attributed to improved fuel—air mixing prior to combustion, which both improves combustion efficiency and reduces unburned hydrocarbons.

2.3.2 Selective and Non-Selective Catalytic Reduction

Selective catalytic reduction (SCR) involves the use of a catalyst generally requiring an additive, such as ammonia, to initiate NO_x reduction chemistry. A common application of SCR in internal combustion engines is the Pt, Rh, and Pd three-way catalytic converters used on spark ignition engines combusting gasoline. Unburned hydrocarbons act as the selective reducing agent for the catalysts. Non-selective catalytic reduction (NSCR) also involves the use of a catalyst but without the need for an additive to reduce NO_x. The application of SCR or NSCR catalysts in a diesel exhaust is severely complicated (1) primarily by the higher excess oxygen content of diesel exhaust, resulting in a net oxidizing environment, and (2) by the presence of sulfur in diesel fuel and resulting catalyst poisoning by sulfur dioxide in the exhaust gases. Significant research is ongoing using secondary injection of small quantities of fuel in the exhaust stream to act as the reducing agent, with demonstrated NO_x reductions of approximately 45 percent at reasonable space velocities and high fuel metering rates [20].

Catalyzing exhaust emissions has also been employed in spark ignition engines as a way to reduce emissions. This technology has recently begun to be used with the exhaust system of CI engines. By passing exhaust gasses over a heated platinum/rhodium coated aluminum surface several chemical reactions are promoted. CO and volatile organic compounds (VOCs) are oxidized into CO_2 and H_2O. NO_x is also decomposed and oxidized into NO and elemental nitrogen and oxygen, which is easier to absorb in the environment [18].

2.3.3 Water Injection

Four major approaches for introducing water into the combustion zone have been reported in the literature:

1. Fumigating the water into the engine intake air
2. Direct injection into the engine through separate injectors
3. In-line mixing of water and fuel prior to injection (unstabilized emulsion)
4. Mixtures of stabilized emulsions treatable as a single-phase drop-in replacement fuel

The authors of [21] demonstrated that water mist injection into the bell housing of diesel-fueled turbine engines can lead to promising results. Water mist introduced to the intake of reciprocating compression ignition engines, particularly two-stroke engines with the intake air passing through the crankcase, poses significant corrosion potential. Separate water injecting valves in the engine avoids intake system contact with the mist [22]. Several authors have evaluated all or most of the options for introducing water into the combustion process and have primarily determined that water-in-fuel emulsions, stabilized or unstabilized, are most effective in reducing NO_x, BSFC, and result in lower increases of CO and UHC emissions [23].

An emulsion is defined as a mixture of two or more generally insoluble liquids. A permanent emulsion exists when sufficient droplet sizes have been reached to prevent the separation of the insoluble materials. Unstabilized emulsions are generated through the high-speed, high-shearing of particles and solids in a liquid. A limiting concern with emulsions is the high capital costs of emulsification mixers and pumps, which are used extensively in the food and agriculture industries.

Unstabilized emulsions require high shear to suspend small droplets of water in the fuel [24]. An advantage of unstabilized emulsions are reduced fuel costs, due to lack of additives needed, and reduced emissions from not combusting surfactants or other emulsifying agents [25] propose a complex fueling system including a vortex chamber to provide in-line mixing of water and diesel fuel without requiring the addition of an emulsifying agent. Diesel fuel pumps, including the unit on our Detroit Diesel 4-71 engine, operate at high volume and high pressure, with a recirculation loop back to the fuel tank. This serves several purposes:

1. A high-volume pump can create the high-pressure needed for the fuel injectors at less expense
2. Recirculating warms the fuel and helps resist gelling at low temperatures
3. The warmed fuel improves combustion

All of these factors contribute to the effectiveness of an in-line fuel—water emulsifying system, assuming that retrofitting the fuel system is acceptable.

Stabilized emulsions use an emulsifying agent to suspend the water in the fuel and reduce the energy required for a permanent emulsion The Air Force preferred a drop-in replacement fuel. without the requirement for modifying the engines or fueling system. The current research thus uses a surfactant to create a permanent. stabilized emulsion that can be treated as a single-phase fuel. A drawback to water-fuel emulsions is the amount of air bubbles reportedly contained in the emulsion mixture. In Ref. [27]. the authors evaluated the effect of water-fuel emulsions on diesel engine performance and emissions under transient conditions. They conclude that air bubbles in the fuel and its variability contributes to fluctuation in the injection timing and a poorer performance under transient conditions. They recommend removal of bubbles from the water-fuel emulsion.

Research has also been conducted extensively on the use of additives to improve the lubrication. reduce the corrosive effect of water in the fuel. and improve the emulsion stability. Nitrate-containing ignition improvers are recommended to reduce exhaust emissions [28] Lubricity additives composed of dimer or trimer acides. phosphate esters. sulfurized castor oils are recommended by Peter-Hoblyn [29] and catalysts can also be used in situ in the fuel to reduce NO_x [29]. In addition to providing lubrication improvement. additives to water—fuel emulsions can be employed for antifreeze characteristics [31] obviously important when significant volume percents of water are present in a fueling system in freezing climates. The authors of [32] demonstrated that surfactants added to diesel fuels can clean up fuel injectors as well as prevent further injector deposits. They also reported a slight increase in NO_x from the combustion of the surfactants. as we will experience without also adding water to the fuel.

Grookes. et al.. [33] attribute water—fuel emulsions with improved combustion and lower particulate and NO_x emissions to the secondary atomization of the water. often designated as microexplosions. In Ref. [34] the authors extensively examined the micro-explosions of emulsified fuels and determined that there is a minimum percent water content in the emulsion required for micro-explosions to occur. and that the percent increases with the kinematic viscosity of the fuel.

Table 1 provides a summary of water injection and water-in-fuel emulsion research in the literature related to the present study. Note that Montagne et. al. in Ref. [32] reported a 5 percent NO_x *increase* when adding surfactants only to diesel fuel for cleaning fuel injectors. Grookes. et al.. In Ref. [33] were comparing diesel and vegetable oil fuels. both dry and as an emulsion with 10% water. by volume Their results were included to demonstrate that small ratios of water provide negligible reductions in NO_x emissions. Small quantities of water are effective in proving BSFC. which could have merit for fuel savings. In [23]. the authors considered macro- and micro-emulsions to determine if water droplet size has a significant effect on NO_x reduction. BSFC. and other combustion parameters. They define micro-emulsion as emulsions with water droplet sizes smaller than the wavelength of visible light. approximately 555 nm. whereas macro-emulsions are characterized with water droplet sizes larger than the wavelength of visible light. Thus. micro-emulsions are reported to visually appear clear. while macro-emulsions appear cloudy. Our fuel mixture would be characterized in this manner as a macro-emulsion As shown in Table 2.1. the reported NO_x reductions were 25 and 23 percent for macro- and micro-emulsions. respectively. They reported significantly-higher (unspecified) ratios of surfactant were required to establish a micro-emulsion. although they reported longer stability lifetime for the micro-emulsion. In Ref [25] the authors showed that using the minimum surfactant required to establish a stable macro-emulsion is justified.

Table 2.1: Summery of Diesel engine NO_x, reduction using water and other additives

Reference	Method	Water, Vol%	Nox, -%
Crookes et al., 1990 [33]	Unstabilized	10	4
Greeves et al., 1976 [24]	Unstabilized	80	60
Montagne et l., 1987 [32]	Surfacant only		-5
O'Neal et al., 1981 [23]	Macro-emulsion	20	25
	Micro-emulsion	20	23
Sawa and Kajitani, 1992 [27]	Stabilized	40	64

The data of the references mentioned in, Table 2.1, except for that of [32], are plotted in Figure 2.1.

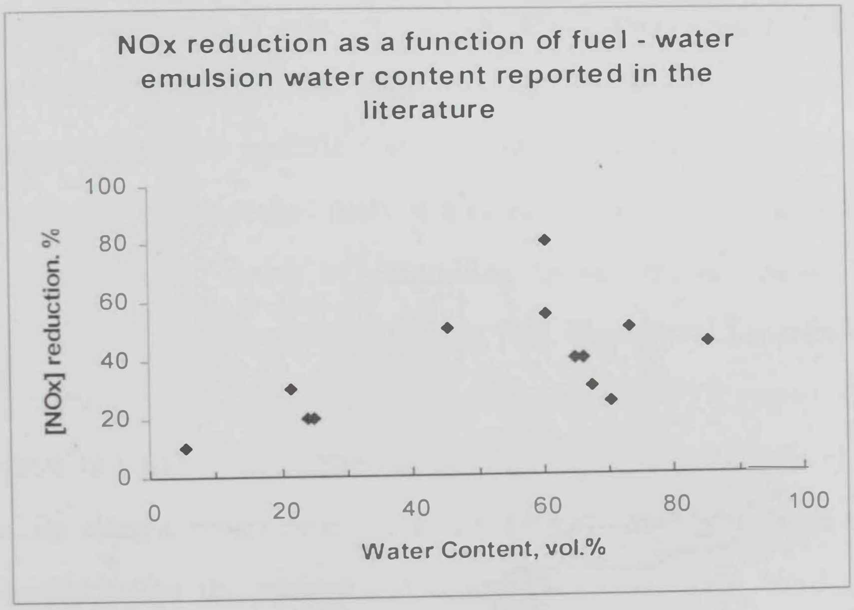


Figure 2.1: Plot of the water content verses NO_x reduction for the data presented in references of Table 2.1

Trend is visible for reduced NO_x emissions with increased water content in the fuel. As shown in Table 2.1, this data represents the five independent research programs conducted across a span of 22 years.

Adding water to the fuel air mixture to decrease the NO formation also has the trade off of increasing the CO and HC emissions under most circumstances. Essentially the water is just lowering the gas and combustion temperatures, which has the same effect as advancing the injection timing or intercooling [8].

Nazha et. al. [35] used the inlet manifold water induction and Exhaust Gas Recirculation (EGR) with four cylinder diesel engine. The engine was modified to incorporate a water injection system into the air intake and an EGR system. Water-in-fuel emulsion was prepared by mixing and circulating the mixture for an appropriate length of time prior to injection into the engine. It has been shown that while EGR can produce a substantial reduction in NO_x emissions; this is achieved at the expense of increased smoke. Water induction on the other hand can result in up to 60% reduction in NO_x at no real cost in terms of engine performance or other pollutants. The authors indicated that using of emulsified fuel leads to reductions in both NO_x and smoke at no cost to the engine performance (as indicated by specific fuel consumption and thermal efficiency). It is concluded that the use of emulsified fuels or a combination of EGR and water addition could prove beneficial in terms of controlling diesel engine emissions. Similar conclusions have been reached by G.H. Abd-Alla [36]. Brusca and Lanzafame [37] used a continuous injection system to supply water a single cylinder CFR engine. Results have shown that water injection really represents a new way to control NO_x formation in Diesel Engine. By using a humid air motor system (HAM) that is connected to an eleven liters diesel engine under the various speed and load conditions. Nord. *et al* [38] explored the reduction of the NO_x. The reduction was directly related to the humidity of the inlet air and a further reduction can be anticipated with higher humidity. The influence of the system on the emissions of hydrocarbons (HC) was negligible while a moderate increase in the emission of carbon monoxide (CO) was noticed [38]. Increasing the water to fuel ratios as a way to reduce the NO_x using a new Mitsubishi water injection system [39] or port water injection [40] have been also employed.

Lin and Huang [4] found that the addition of an oxygenating agent into fuel oil is one of the possible approaches for reducing engine emissions. Ethylene glycol monoacetate was found to be a promising candidate primarily due to its low poison and oxygen-rich composition properties. The experimental results show that an increase in the inlet air temperature caused an increase in brake specific fuel consumption (BSFC). Carbon Monoxide. Carbon Dioxide emission. and exhaust gas temperature. while decreasing the excess air. Oxygen and Nitrogen Oxide emission concentrations. Increasing the inlet air humidity increased the Carbon Monoxide concentration while the decreased excess air. oxygen and nitrogen oxide emission concentrations.

Papagiannakis and Hountalas [41] tried to study the effect of gaseous fuel as a supplement for liquid diesel fuel. and they selected the natural gas which has a relatively high auto ignition temperature and moreover is an economical and clean burning fuel. The effect of liquid fuel percentage replaced by natural gas on engine performance and emissions has been studied.

Christensen and Johansson [42] reached to same result by using water injection in a Homogeneous Charge Compression Ignition (HCCI) engine. The NO_x emissions. which are very low for HCCI. decreased even more when water injection was applied. To reduce the peak flame temperatures associated with NO formation. Mello and Mellor [43] used an inert fluid injection. such as water/steam.

2.3.4 Other Control Technologies

The authors of [44] described a filter cart designed for capture of NO_x, CO, VOCs, and particulate from the A/M32A-86 diesel generator. The device is a series of sub-systems, including a vermiculite filter to capture particulate, air-to-air heat exchanger and demister for cooling and dewatering, and granular activated carbon (GAC) filters to adsorb NO_x, CO, and VOCs. This device requires a large footprint.

After the GAC filters are saturated and adsorption rates begin to decline, the filters are thermally regenerated. Adsorbed gases are desorbed, and can either be compressed, bottled, and reused, or destroyed on-line via selective catalytic reduction. The vermiculite particulate filters are replaced and discarded after excessive increase in pressure drop. The filter cart requires a large footprint and would not be feasible to mobilize to a war-fighting theatre. Advantages include that one filter cart can service multiple generators, depending on the capacity of the filter cart, and it can be used to control emissions from other combustion sources.

Non-thermal plasmas have also been applied to diesel exhaust, and specifically applied to reduce NO_x emissions from the A/M32A-86 [45]. This application also required the use of a series of subsystems, including a ceramic particulate filter, nonthermal plasma discharge (NTPD) reactor tube with alcohol injection, and a wet gas scrubber. The particulate filter captures particulate and would be cleaned in-line using the hot exhaust gases. The NTPD essentially uses high-voltage, low amperage, high-pulse rate electrical discharges to generate reactive, oxidative species in the exhaust gases. The addition of alcohols is reported to increase the reaction efficiency, lowering electron volts required to oxidize NO to NO_2. The wet scrubber is then used to adsorb and react the NO_2 with water to form nitric acid, HNO_3, and then with sodium hydroxide, NaOH, to form sodium nitrate, $NaNO_3$, useful as a fertilizer. This system would also require a large footprint.

2.4 Combustion Noise Problem

The study of engine noise has been carried out since the early stages of engine development. In 1931, Ricardo first found a descriptive relationship between the combustion pressure rise and the noise produced [46]. Later, a number of parameters in determining the noise developments were investigated which include the first and second derivatives of cylinder pressure. These methods were effective in revealing the relationship between engine combustion and noise. Some of these methods still play an important role in identifying the sources of engine noise [47].

Although there are a number of engine noise sources, one of the most fundamentals is the combustion-induced noise [48]. It occurs towards the end of the compression stroke and subsequent expansion stroke. The rapid pressure change due to the combustion transmits through engine structures and forms a part of the airborne noise. This pressure change also causes the vibration of the engine components such as the cylinder head, pistons, connecting rods and engine body. The vibration of these components then provides another part of the overall engine noise. Together these noise sources account for over 80% of total engine noise. The combustion-induced noise is however the dominant source. It occurs around the top dead centre (TDC).

Other noise sources are due to engine functions such as the injection of fuel and the operation of inlet and exhaust valves. These sources usually produce low level noise and make up a fraction of the overall noise. Yet all have designated times of occurrence in terms of crank angles. For instance, fuel injection is usually performed around 8-10^o before the TDC in the compression stroke. The exact instances of these events depend on the individual design of the diesel engines.

Increasing road traffic in the world has brought about legislation aimed at reducing noise from vehicle engines. Noise is transmitted throughout the engine block and other rigidly attached components as vibration. These vibrations can exist across the entire spectrum of frequencies. When they are in the range at which a healthy human ear can hear they radiate from the engine structure as audible. This noise and vibration can create severe problems for the engine structure, its operators, and the surroundings. Engine noise can be grouped into one of three categories: air flow, mechanical and combustion noise [49].

2.5 Methods of controlling the engine noise problem

For the dual fuel engines using diesel and gaseous fuel, G H. Abd Alla, et. al. [50] found that increasing the amount of pilot fuel at high loads led to early knocking. The combustion noise is found to be related to the type of gaseous fuels and to the engine design and operating parameters of the dual fuel engine. The engine tested used dual fuels of Diesel and three gaseous fuels separately. [51, 52]. Effects of various working fluid compositions on combustion noise in diesel engines are studied by Galinsky *et al.* [49] and provided a better understanding of the relationship between the combustion process and noise production. Combustion noise levels, rate of pressure rise, intake mixture composition and load are related.

2.6 Dual Fuel Engines

The availability of alternative gaseous fuels has lead to a worldwide spread of internal combustion engines running on dual fuel concept. Gaseous fuels also promise to be suitable for higher compression engines since it is known that they resist knock more than conventional liquid fuels, as well as producing less polluting exhaust gases, if appropriate conditions are satisfied for its mixing and combustion. Therefore it is more economical and of environmental advantage to use gaseous fuel in diesel engines which use dual fuel concept. There have been many published works on the use of gaseous fuels in dual fuel engines. Natural gas use in dual fuel engines has been studied from the combustion duration and ignition delay point of view as indicated in [1] and from performance and emissions point of view as presented in [3]. Combustion and thermal loading and temperature distribution have also been studied for dual engines [9]. Pure methane has also been studied in dual fuel engines from flame spread limits point of view [12] and performance and emissions point of view [13]. LPG has been studied from the point of view of performance and emissions; [13].

2.7 Scope of the current work

There have been many solutions to the NO_x and noise emissions problem for diesel engines. However, there has not been any work that deals with the solution of these problems for dual fuel engines. When diesel engine is converted to run on a diesel fuel as pilot fuel and when it is converted to run on gaseous fuel as the main fuel, then the noise and NO_x are still high and must be reduced. Therefore, the objective of the present work is to examine the effects of adding water to the intake air of the engine on the engine performance, combustion noise and exhaust emissions of a dual fuel engine running on diesel – LPG. The water has been added in the present work to the intake air of the engine in the form of vaporized water produced by a special test rig. The water addition effects on the performance and exhaust emissions have been also investigated.

CHAPTER III

ENGINE AND EXPERIMENTAL TEST RIG: DESCRIPTION AND PROCEDURE

3.1 Introduction

The research engine used in the present study is the research engine Ricardo E6 single cylinder variable compression indirect injection diesel engine. Among the other engines exist in the Heat Engine laboratory of the UAE University, the Ricardo E6 engine is very flexible since the compression ratio, ignition timing and other factors are all controllable (see next section for details). This flexibility makes it a very useful piece of equipment for research. It can be used in many educational and research studies related to engine performance, fuel additives, new fuel blends, and their air pollution.

The test bed used has several accessories such as the intake steam system, liquid and gaseous fuel system, and gas analyzer. All the instruments are connected to a Data Acquisition System (DAS) to store the data and facilitate the research studies. This chapter considers the engine test bed and all the instruments used in this research. It gives a brief description of each component along with its principle of operation.

3.2 The Ricardo E6 Engine

The Ricardo E6 engine is a single cylinder, indirect injection, water cooled, four-stroke unit of 507 cc (31 in^3) swept volume, having a bore of 76 mm (3.0 in) and a stroke of 111 mm (4.375 in). The cylinder, cylinder head and cam box are bolted together forming an integral unit which is attached to the crankcase through a large nut around the lower end of the cylinder retained by the cylinder housing as shown in Fig.3.1. Full details of the engine are given in Table 3.1 and a complete picture is shown in Fig.3.2 and Fig. 3.3. The engine can be operated as spark or compression ignition one. The compression ratio of the petrol/gas version is continuously variable between 4.5 - 20.0 while the engine running. The ratio can also be varied for the diesel version but, as the performance of any diesel combustion system is very sensitive to piston-to-cylinder head clearance, the ratio is, in practice, usually set at about 22. The cylinder position is measured by means of a micrometer and the compression ratio (CR) can be obtained according to the formula:

CR = (Clearance volume + Swept volume)/(Clearance volume)

The corresponding clearance volume for the compression ignition engine with the cylinder in the normal fully wound down position is 25.26 cc, which gives a compression ratio of 21.07. The engine valves are operated from the overhead camshaft by means of swinging cam followers. In the diesel version, a head having a Ricardo Comet Mk V compression swirl combustion chamber is fitted. This type of combustion system consists of two parts. The swirl chamber in the head has a top half of spherical form and the lower half is a truncated cone which communicates with the cylinder by means of a narrow passage or throat. The second part consists of special cavities cut into the crown of the piston as shown schematically in Figs. 3.3 and 3.4. The lower half of the combustion chamber containing the throat is thermally insulated from the rest of the cylinder head. On the compression stroke of the engine, air is forced into the Comet chamber and is given a rotating motion by the shape of the chamber and by the position of the communicating passage.

The liquid fuel is injected into the Comet chamber via a pintle type nozzle, and the organized swirl of the air in the chamber ensures that there is supplied. The American Bosch fuel injection pump, type APE 1B using a cam form 6/1, and a 6 mm plunger is driven from camshaft extension at the rear of the engine. The injection, timing can be altered over a limited range (25° to 45° BTDC) while the engine is running. The injector body is fitted with a 4 mm spring which is adjusted to give an opening pressure of 150 atmospheres to the nozzle.

Table 3.1 Ricardo Engine specifications

Number of cylinders 1
Bore 76.2 mm (3")
Stroke .... 111.1 mm (4.375").
Swept Volume ....0.507 liters (31 in^3)
Max. Speed ....50 rev/s (3000 rpm)
Min. Speed .... 15 rev/s (900 rpm)
Max. Power .... Diesel (CR = 20.93)
Naturally Aspirated),.................................... 9 kW
Supercharged (0.5 bar),.................................. 14.0 kW
Max. Cylinder Pressure .... 150 bar
Compression Ratio (CR) Max. CR 21.07
Injection Timing Varied over 25°- 45°
BTDC.

3.2.1 The Coolant and Oil System

The coolant system is a pressurized closed circuit. The coolant is drawn from the bottom of a header tank and pumped through a heat exchanger before passing out to the engine cooling jacket. The outlet pipe to the engine contains an electric heater used to maintain the coolant temperature during motoring tests.

The lubricating oil is contained in the engine sump. The oil is drawn from the sump and pumped through the heat exchanger, oil filter and temperature sensor pocket, before returning to the main oil system on the engine. The oil pressure is limited to 2.0 bar by a relief valve mounted on the engine. The recommended operating temperatures for both the engine coolant and oil are $70^{\circ} C$, which are maintained via the use of heat exchangers with normal lab water as the cooling agent.

3.2.2 Dynamometer

Dynamometer is an electrical dc machine rated at 22 kW-420 volts, mounted on trunnion bearings supported by pedestals. The system is self ventilated by a shaft driven fan and is operated through a KTK thyristor converter unit, which enables it to act as an ac motor to drive the engine during starting and motoring operations, or a dc generator when used to load the engine. During loading the dc power from the dynamometer is inverted to 3-phase ac via the thyristor unit, and then fed back into the mains. Load is measured using a torque arm of 390 mm (mounted on the dynamometer frame) which operates on a strain gauge load cell, the output from which provides a continuous display of torque in (Nm) at the control console. The dynamometer is fitted with two identical torque arms to maintain static balance. A tachogenerator mounted on the dynamometer shaft provides a speed signal to the closed loop speed control system.

3.2.3 Control Console

The control console is a free standing unit which houses the following:

a) Automatic Control Unit (ACU): It is a safety trip system designed to protect the equipment and/or operator. System fault, speed, oil pressure, flame, stop switch, water level, water temperature and oil temperature are the monitored parameters. In the event of any of the above conditions being unsatisfactory, the appropriate amber warning lamp will illuminate, and the system (diesel version) will shut down as follow: Fuel solenoid valve closes (on service frame), cut off solenoid de-energized to stop, dynamometer drive disconnected and last the engine coasts to stop.

b) Torque and Speed Indicators.

c) Remote Control Unit.

d) Power Supply Unit.

3.2.4 Service Frame

The services frame is associated with the fuel system and has mounted on it the following items:

- A compression/ratio chart related to the particular E6 engine supplied.
- A liquid fuel tank.
- A multi-slope manometer.
- Fuel pump and fuel cooler.

- An oil pressure gauge.
- A fuel burette *(replaced by the Compuflow meter).*
- A fuel regulator.
- A gas Rotameter

The exhaust system provided on the engine consists of a flexible stainless steel pipe (to tolerate high temperatures and absorb vibration) which connects the engine manifold to a short steel pipe with two silencers and one sample orifice.

The engine is mounted on a 450 kg flexibly supported cast iron block (to reduce the excited vertical vibratory forces) which is attached to a rigid welded base frame by (unseen) four rubber mountings which minimize transmitted disturbance to the base frame.

3.3 Fuel System

The present project deals with Dual Fuel Compression Ignition Engine, which means two fuels are being used simultaneously. The diesel liquid fuel is injected regularly through the existing fuel injection system and mass of liquid diesel is kept at minimum value to ignite the other main gaseous fuel. The gaseous fuel used is Liquefied Petroleum Gas (LPG) which is introduced to the engine through the gas system at the intake pipe of the engine just ahead of the intake valve. The gaseous fuel forms them main fuel admitted as it forms more than 90% of the fuel used. It is admitted at a pressure slightly higher than atmospheric in the intake pipe using a proper nozzle facing the intake air flow downward with many holes provided to homogenize the gas into the intake air. Thus, the fuel system can be divided to Diesel (Fig. 3.5) and Gas Fuel Systems.

3.3.1 Diesel Fuel System

In general, the fuel system of a Diesel-engine installation is defined as the equipment which is necessary to handle fuel oil from the point where it is delivered to the plant and until it reaches the fuel-injection pump. This equipment consists of strainers, filters, transfer pumps, tanks, piping and fuel meters. In delivering the fuel to the combustion chamber a diesel fuel injection system must fulfill five main requirements:

1) Meter or measure correct quantity of fuel injected

2) Time fuel injection

3) Control rate of fuel injection

4) Atomize fuel into fine droplets

5) Properly distribute fuel in combustion chamber

The diesel fuel system of the Ricardo E6 engine is shown schematically in Figures 3.3 and 3.5. The following related components will be discussed briefly:

i) Compuflow System.

ii) Fuel Injection Pump.

iii) Injection Nozzle.

iv) The Control and Display Unit: It executes the signal from the load cell, controls the electromagnetic valve in the fuel supply line, displays the measuring parameters and indicates fault conditions.

Measuring Technique

When the system power is ON and the fuel container is empty, the load cell sends a signal to the microprocessor in the Control and Display Unit which, in turn, sends a signal to the electromagnetic valve in the fuel supply line to open and thus allowing the container to be filled. The electromagnetic valve is closed in a similar way when the container is filled with the preset value. During filling the last rate is displayed, therefore we do not record the rates till the RUN button is illuminated; otherwise the last flow rate is recorded independent of the current flow rate. Over a selective time period, the rate of fuel consumption value is calculated to get the average.

3.3.1.1 The Compuflow System

The Compuflow is a multi-function micro processor-based fuel measurement system. It measures the weight of fuel consumed in a preset time, the time to consume a preset amount of fuel, the instantaneous fuel flow rate, the engine speed and total engine revolutions. The system is capable of producing an output of all displayed parameters (0-5 Volt each).

Description of the system

The major parts of the Compuflow system are:

I) The Measuring Head Unit, as shown in Figures 3.3-3.6, consists of two parts:

1) *An Aluminum fuel container* to which the fuel supply, the fuel feed and fuel return lines are connected.

2) *A very sensitive load cell* that senses any change in fuel weight in the fuel container and transmits a corresponding electrical signal to the microprocessor in the Control and Display Unit via the Weigh Rig Control Unit.

II) The Weigh Rig Control Unit: Its function is to interface the control signals from the Control and Display Unit to the Measuring Head Unit.

3.3.1.2 The Injection Pump

The plunger-type injection pump, shown in Fig. 3.7 (left), has an engine-driven camshaft which rotates at one-half engine speed. Roller cam followers, riding on the camshaft lobes, operate the plungers to supply high-pressure fuel through the *delivery valve* to the injection nozzle. The pumping element is a plunger and barrel, one set to supply an engine cylinder. The plunger has a constant stroke. However, in order to vary the amount of fuel delivered per stroke for satisfying varying load demands, the upper part of the plunger is provided with a vertical channel extending from its top face to an annular groove, the top edge of which is milled in the form of a helix (called the *control edge)* as shown in Fig. 3.7. The barrel has either a single or double control port, depending on the design (since the actual design is not provided with the engine). The top of the barrel is closed by a spring-loaded valve called the delivery valve.

3.3.1.3 Injection Nozzle

The injection Nozzle is a simple device used to atomize the fuel for better combustion and spread the fuel spray to full mix it with air. In the present injection nozzle (pintle type) the fuel pressure acts on the lower end of the valve, moving it inward to release a fuel spray as indicated in Fig. 3.8. A spring at the upper end of the valve is normally adjusted to set the opening pressure (opening pressure equals to 150 bars). A small amount of fuel leaks past the nozzle valve and lubricates the working parts. The excess lubricating fuel is removed from the top of the nozzle at the fuel leak-off and returns to the Compuflow system.

3.4 Gaseous Fuel Measurements

The LPG gas fuel flow rate is measured by a variable area meter (rotameter). The rotameter consists of a gradually tapered glass tube mounted vertically in a frame with the large end up. The gas flows up through the tube and suspends freely a float. As the flow varies, the float rises or falls thus varying the area of the annular space between it and the tube, so that the head loss across this annulus is equal to the weight of the float. The tube is marked in divisions and the reading of the meter is taken from the scale reading and the reading edge of the float, which is taken at the largest cross section of the float. The rotameter has a calibration chart to convert observed scale readings to flow rate. The rotameter float is constructed of stainless steel.

3.5 Exhaust Gas Measurements

3.5.1 Exhaust Gas Analysis

The exhaust gas module is portable gas analyzers used to perform vehicle diagnostic tests and to measure emission gas levels found in the exhaust of all internal combustion engines. Figure 3.9 show a photo for the exhaust gas analyzer. The gas analyzer measures the emission levels of carbon dioxide, carbon monoxide, hydrocarbons, and oxygen. The exhaust gases enter the gas analyzer through the sampling hose/probe assembly. The gas analyzer then analyzes the gases and sends analyzed data to the tool. The tool, with the Gas HG-520 software installed, enables reading the analyzed data and controls the gas analyzer functions.

3.5.2 The Opacimetter Shady X2000

The instrument shown in Figure 3.10 is used to measure the opacity of diesel engine exhaust. It is made up of detecting unit inside a measuring cell, optic systems, computer controlled data processing and display units and control keypad. The gas from the engine's exhaust system is taken along a hose to the measurement cell heated to 70°C where it forms a uniform 20 cm thick column. After measuring the exhaust gas is vented to the outside.

The instrument displays the opacity settings by means of a 20 character, 5-line display. Two scales are variable: linear absorption % from 0 to 99 and logarithmic in 3 figures for absolute light absorption K expressed in (m^{-1}). Both scales go from 0 for total light flow to full-scale for complete absorption.

3.6 Measuring of Combustion Pressure and Crank Angle

Another data acquisition system is used to collect the cylinder combustion pressure and crank angle data. The pressure signal is collected by a high pressure water cooled piezo-electric pressure transducer and fed into a charge amplifier then to a data acquisition card linked to the personal computer. The crank angle signal is fed into a degree marker shaper channel and the output is fed into the acquisition card. The acquisition card could collect data at the rate of 250 kHz. A Labview program has been written to collect the data from the two channels; shown in Figs. 3.11 and 3.12, at a sampling rate of 10000 points per second and store the pressure and crank angle data in the computer disk for offline analysis. A computer program is written in MS Excel to find the maximum combustion pressure and the pressure rise rate data at all cycle points from mid compression stroke to mid expansion stroke. The maximum value of pressure rise rate is then obtained and recorded. This value will be used to represent the noise level at that operating condition. Experiments have been carried out after running the engine for some time until it reaches steady state and oil temperature is at $60^{\circ}C \pm 5$, and cooling water temperature is at $70^{\circ}C \pm 5$

3.7 Data Acquisition System (DAS)

The objective of a data acquisition system is to monitor, record and control various processes in case of continuous operation. In the present research the DAS is employed only to collect the important performance parameters of the Ricardo engine using specially developed software.

The measured and stored parameters are:

- Temperature (air, fuel, oil, exhaust, coolant in and coolant out), Torque and engine speed. All signals are taken from the Ricardo Engine Control Console.
- Liquid fuel mass flow rate from the Compuflow System.

In the current case there have been some troubles with Gas Analysis System so an external Gas Analysis System has been used. Figure 3.13 and Figure 3.14 shows the general connection between the mother board of the DAS system and the measuring instruments with their type and range of output signals. In addition to an IBM computer, an Epson printer, and a HP plotter, the system consists of two parts. The first part is a μMAC-5000 master (mother) board, and the second part is one casing containing six expander boards (one μMAC-4030 board, one μMAC-4050 board and four μMAC-4015 boards). A brief description of the different boards is given below:

3.7.1 The Mother Board (μMAC-5000)

This board can be used independently as a single board measurement and control system as shown in Fig. 3.13. It has a built-in microcomputer using a measurement and control language called *μMACBASIC*. The board consists of 12 analog input channels, 8 analog output channels, 8 digital input channels, and 8 digital output channels. It is provided with an 8088-based microcomputer (5 MHz CPU) with standard memory of 80 KB of ROM and 32 KB RAM.

3.7.2 Expander Boards

Expander Boards are used to increase the channel capacity of the Data Acquisition System. These boards are compatible with the μMAC-5000 master board and they are interfaced to it via an expansion bus. A brief description of the different expander boards utilized is given below:

a) **μMAC-4030 Analog Output Expander Board:** It has eight channels for analog output signals which may be used to operate control elements such as valves or actuators. The *μMAC-4030* board includes a microprocessor for controlling analog converters and communication with the *μMAC-5000* board, the ROM (software storage), and RAM (data storage).

b) **Four μMAC-4015 Analog Input Expander Boards:** The *μMAC-4015* is used for data collection purposes and it has 12 channels accepting analog input signals of the same type.

c) **μMAC-4050 Pulse/Frequency, I/O Expander Board** : It contains eight frequency input and/or output channels. Each independent channels can be used for event counting, frequency measurement, and time proportional output.

3.7.3 The μMACBASIC Language

This language is an extension of the well-known BASIC high level programming language, mixed with some statements from FORTRAN and PASCAL, optimized for measurement and control applications. The μMACBASIC is specially designed to perform measurement and control functions in any application requiring analog or digital input and/or output signals *(I/O)*. A personal computer is used as a host computer for the *μMAC* system. Work station Operating System (WOS) has been developed for popular microcomputers. The wos provides many functions such as:

1) Storage of the μMACBASIC program in the PC's disk.

2) File management (name, copy, etc.) without losing communications with the μ*MAC* system.

3) Use of the IBM's disk for reading and writing simple data files under μMACBASIC program control.

4) Use of an attached printer via the PC.

More details about μMACBASIC program see Appendix A. Photos of the cards are shown in Fig. 3.15.

3.8 The Digital Storage Oscilloscope and Accessories

The digital storage oscilloscope (Gould 4041) is a computer interfaceable type operates both as a conventional real time oscilloscope with a 25 MHz band width and also as a digital storage instrument with a 10 MHz maximum sampling rate. The digital oscilloscope operates according to the principle that analog signals may be converted to digital signals through a sampling process Instead of displaying the analog signal directly; it first performs an analog-to-digital conversion and then stores the digital signals in a buffer memory. The signal may then be displayed on the CRT screen as points. Because the digital signal is stored, it may be recalled and reexamined on an expanded scale. In addition, the signal may be stored on inexpensive auxiliary disks for later study or possible manipulation with a computer. In the present project, the oscilloscope was just used to detect the onset of knocking, to know when the ignition starts and to observe the pressure levels generated within the cylinder for the sake of comparison of the different cases, as shown in Fig. 3.16.

3.9 Steam Generation System

The steam injection unit has been designed and built to change the moisture content in the moving stream of air. It may be seen in Fig. 3.17 in a complete picture. It is equipped with a variable speed fan blows air through a 254 mm square ducting. The air then passes through the steam area where steam is injected into the air flow by using the boiling water tank. Three electric heaters are fitted in the water tank to evaporate the water at different rates. Two thermometers were used to measure the wet and dry bulb temperature and the air humidity ratio. The engine intake moisture can be changed by injecting different amount of steam from the water tank which is equipped with three electrical heaters as follows:

1) First water heater; 2 kW capacity.
2) Second water heater; 2 kW capacity.
3) Third water heater; 1 kW capacity.

The speed of the blower fan can be varied by a variable transformer. The engine air intake pipe has been connected to the steam generation system by a 80 mm hose. The intake air humidity ratio can be changed then by operating more water heaters which enables more water to evaporate and to increase the humidity ratio of the engine intake air. The variation of the system fan speed also enables the control of the humidity ratio as the experiments have to be carried out at fixed humidity ratio. The system enabled the relative humidity to change from atmospheric value to 100% (saturated air).

3.10 Experimental Procedure

The following parameters have been varied according to the shown levels:

- The water specific humidity has been varied at the levels: 6, 14 and 24 g/kg

This water/air gives the following ratio of water/fuel:

$$\frac{m^{\cdot}_{w}}{m^{\cdot}_{fuel}} = \frac{m^{\cdot}_{w}}{m^{\cdot}_{air}} \times \frac{m^{\cdot}_{air}}{m^{\cdot}_{fuel}}$$

Minimum water/fuel ratio = $$\left(\frac{m^{\cdot}_{w}}{m^{\cdot}_{fuel}}\right)_{min} = \frac{6}{1000} \times \frac{20}{1} = \frac{12}{100}$$

Maximum water/fuel ratio = $$\left(\frac{m^{\cdot}_{w}}{m^{\cdot}_{fuel}}\right)_{max} = \frac{24}{1000} \times \frac{20}{1} = \frac{48}{100}$$

- The engine speed and it is varied from 18 to 30 rev./sec
- The pilot diesel fuel mass injected and it is varied from 0.00004 to 0.0002 kg/s
- The engine injection timing for the pilot fuel and it is varied from 20 to 45° BTDC
- The engine compression ratio and it is varied from 14 to 22 (for the diesel engine)
- The mass of gaseous fuel and it is varied to give from 2 Nm torque output till the onset of engine knocking and strong pressure waves observed

The following parameters have been collected:

- The engine output torque, Nm
- The Liquid and gaseous fuel flow rate, kg/s
- The combustion pressure over many cycles, bar
- CO emission level in exhaust gases, %
- HC emissions level in exhaust gases, ppm
- Opacity of the exhaust gases, %

The following parameters have been calculate and presented:

- Engine output power

Power = Torque x angular speed

$$BP = Tx2\pi N \quad ,$$

where T is the torque measured in Nm, and N is the rev./sec of the engine

- Engine brake specific fuel consumption

$$bsfc = \frac{\dot{m}_f}{BP}$$

where $\dot{m}_f$ is measured fuel flow rate in kg/s, and BP is brake power in kW

- The maximum combustion pressure

$$P_{\max} = \max(P_i),$$

where Pi is the combustion pressure, bar

- The maximum combustion pressure rise rate (measure of combustion noise emissions)

$$(\frac{dP}{d\theta})_{\max} = \max(\frac{dP_i}{d\theta})$$

where this is calculated in MS Excel for all the cycle points by getting the slope of the pressure against the crank angle, then the maximum value is estimated.

- The onset of engine knocking; when the pressure rise rate becomes very high and engine roughness is observed.

A sample of the combustion pressure data as well as the pressure rise rate is shown in Fig. 3.18.

3.11 Common values for parameters in experiments

If not specified differently, then the common values of the experiment parameters are as follows: $N = 20\ rps$, $IT = 35^{\circ}$ BTDC, Compression Ratio = 22

3.12 Error analysis of measured data

3.12.1 Errors

The error is the difference between the (unknown) true value and the measured value (best experimentally determined value).

3.12.2 Error Classification

Based on the above mentioned sources, errors are classified as follows:

(I) Systematic or Fixed Errors

Systematic error is of an insidious nature: it is completely unobtrusive. As the term indicates, it is repetitive and of a fixed value, recurring consistently every time the measurement is made. Such errors may result from:

A. Calibration errors

B. Certain types of consistently recurring human error

C. Errors of technique

D. Uncorrected loading errors

(II) Random or Accidental Errors

Random errors are distinguishable by their lack of consistency. An observed quantity may not be consistent when estimating reading or the process may include certain uncontrolled or poorly controlled variables causing changing conditions. We have different errors. Errors stemming from environmental variations, errors resulting from variations in definition and errors derived from insufficient sensitivity of the measuring system.

(III) Illegitimate errors

Illegitimate errors: As their name implies, they should not exist. They include outright mistakes that can be eliminated through exercise of care and repetition of the measurement.

A. Blunders or mistakes.

B. Computational Errors

C. Chaotic Errors.

3.12.3 Uncertainty

3.12.3.1 Uncertainty Analysis

Suppose a set of measurements is made and the uncertainty in each measurement may be expressed with the same odds. These measurements are then used to calculate some desired result of the experiments. We wish to estimate the uncertainty in the calculated result on the basis of the uncertainties in the primary measurements. The result R is a given function of the independent variables $X1$, x_2, x_3, ... ,Xn . Thus, $R = R(X1, x_2, x_3 \ldots, Xn)$

Let W_R be the uncertainty in the result and $W_1, W_2, \ldots W_n$ be the uncertainties in the independent variables. If the uncertainties in the independent variables are all given with the same odds, then the uncertainty in the result is given as:

$$W_R = [(\frac{\delta R}{\delta x_1} W_1)^2 + (\frac{\delta R}{\delta x_2} W_2)^2 + ... + (\frac{\delta R}{\delta x_n} W_n)^2]^{1/2}$$

Particular notice should be given to the fact that the uncertainty propagation in the result WR predicted by the equation above depends on the squares of the uncertainties in the independent variables W_n. This means that if the uncertainty in one variable is significantly larger than the uncertainties in the other variables, say, by a factor of 5 or 10, then it is the largest uncertainty that predominate and the others may probably be neglected. In the previous discussion we noted that an uncertainty analysis may aid the investigator in selecting alternative methods to measure a particular experimental variable. It may also indicate how one may improve the overall accuracy of a measurement by attacking critical variables in the measurement process.

3.12.3.2 Uncertainties in Some Important Parameters

As mentioned earlier, the error in any quantity (dependent parameter) is a function of the independent variables included in its calculated value.

The independent parameters involved in our experiments with their approximate uncertainties are presented in Table 3.2. The uncertainties calculated in Table 3.2 are based on the maximum scattering percentage around a mean reading (for example a typical mean reading of the fuel pressure drop across the orifice meter is 50 mV (this is the reading on the digital pressure meter). It is found that the reading fluctuates between 46 and 55 mV, so, the maximum scatter is ± 4.5 mV from which we can find the maximum scattering percentage as shown in the Table (9 %).

An example of the experimental scattering may be seen in figures 3.19, 3.20 and 3.21 for the torque, maximum pressure and maximum pressure rise rate. It may be noticed that the repeatability of the experimental results is acceptable and the maximum uncertainties are given in Table 3.2.

Table 3.2 Uncertainties in some independent variables

Independent Variable	Uncertainty (%)
The water specific humidity, g	0.83
The engine speed, rpm	2.5
The engine output torque, Nm	4
The pilot diesel fuel volume injected, kg/s	6
The engine injection timing for the pilot fuel, °BTDC	5
The engine compression ratio	1
The mass of gaseous fuel, kg/s	1.25

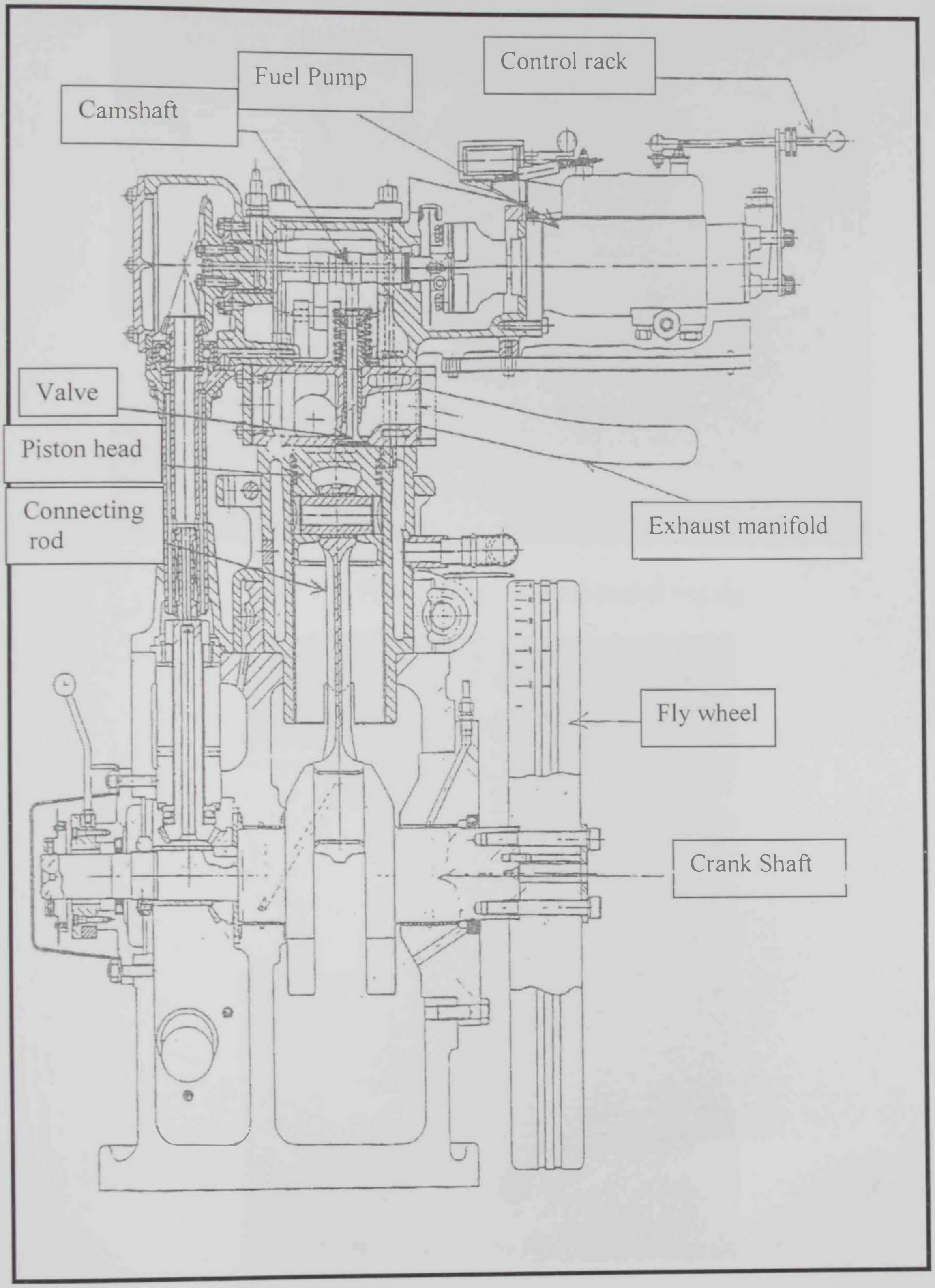


Figure 3.1: Longitudinal arrangement of the Ricardo E6 Engine Diesel version [53]

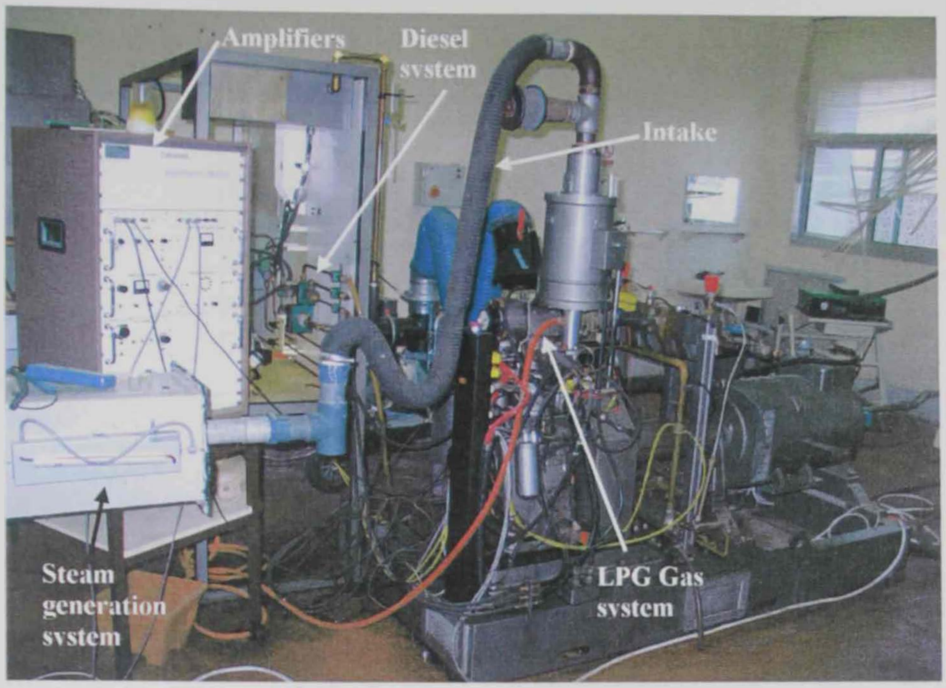


Figure 3.2: Photo of the complete engine test rig

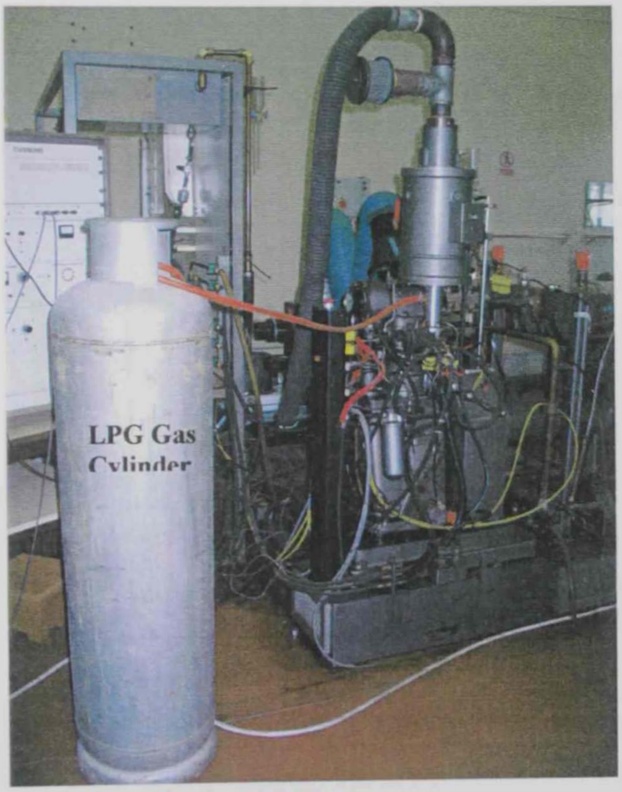


Figure 3.3: Photo of the engine and gas system test rig

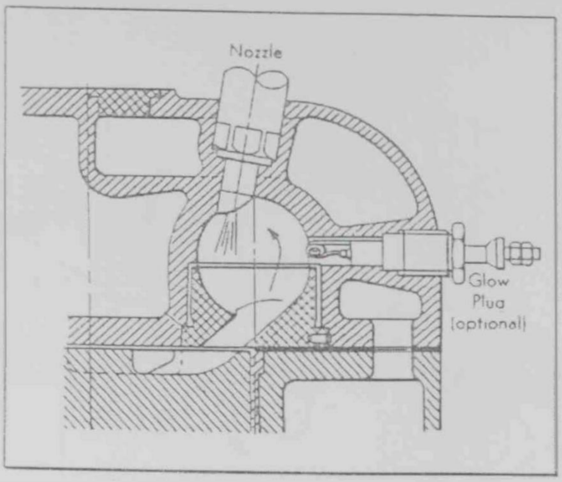


Figure 3.4: Combustion chamber [53]

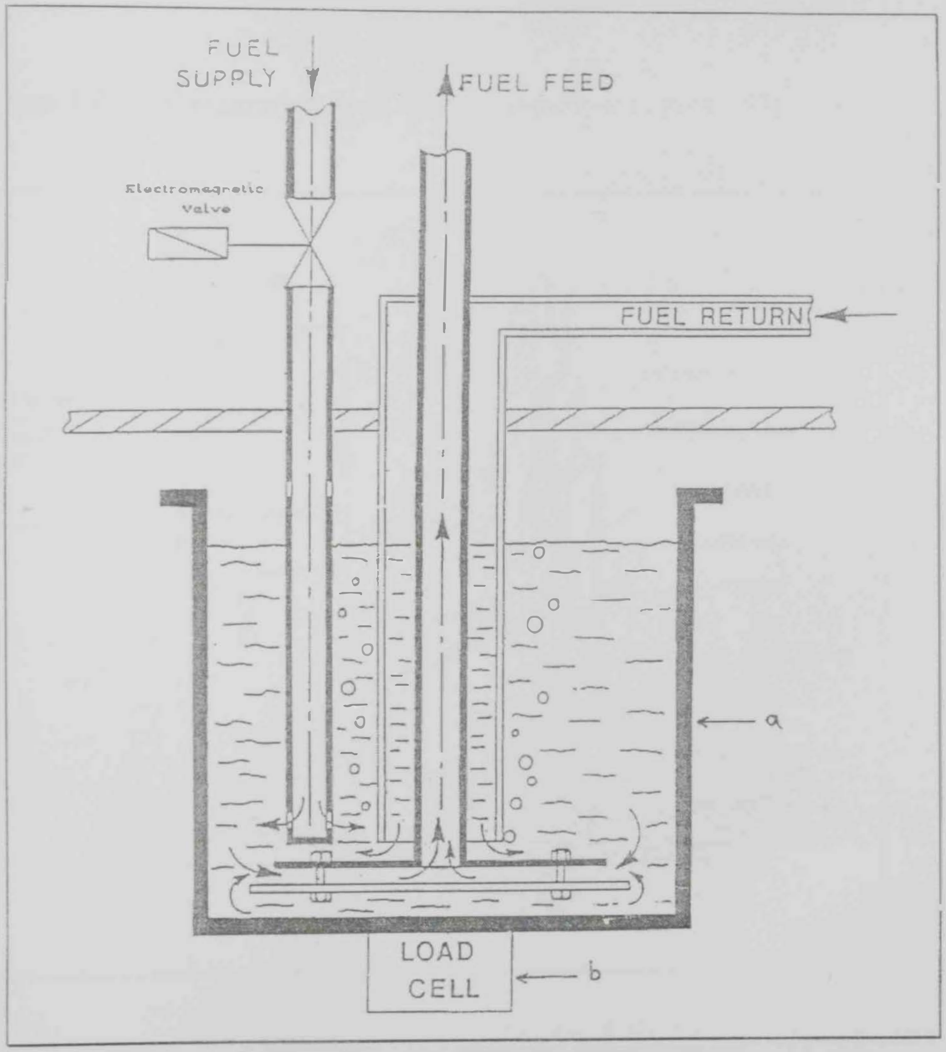


Figure 3.5: Fuel injection system with single barrel pump.
Left: system layout. Right: section through fuel injection pump.[53]

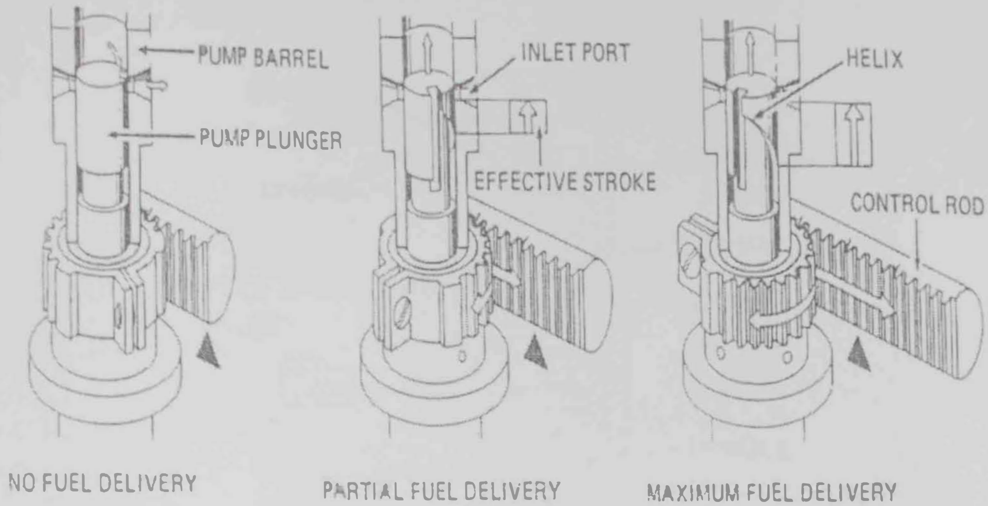


Figure 3.6: Fuel measuring head of the compuflow system [53]

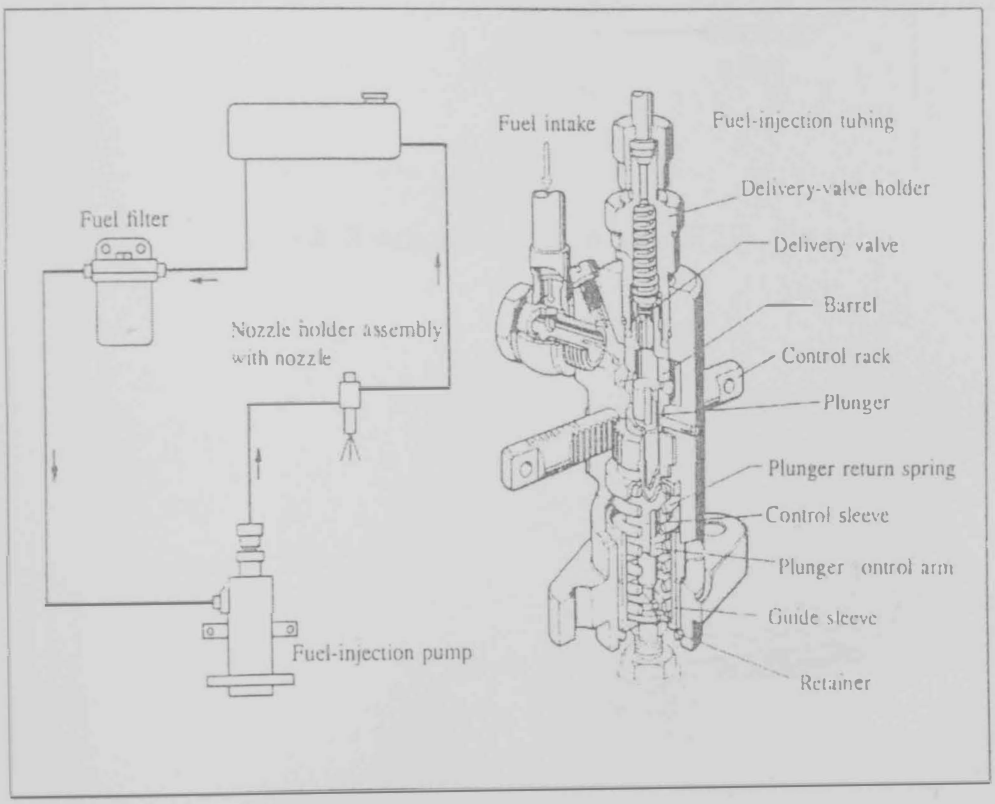


Figure 3.7: Movement of the control rod turns the pump plunger [53]

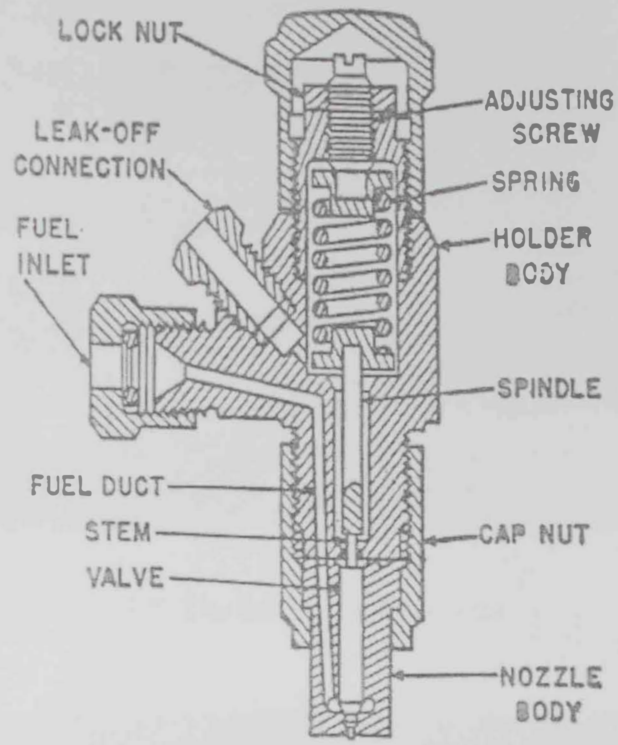


Figure 3.8: Boach fuel injection nozzle [53]

Figure 3.9 The exhaust gas analyzer

Figure 3.10: The Opacimetter Shady X2000

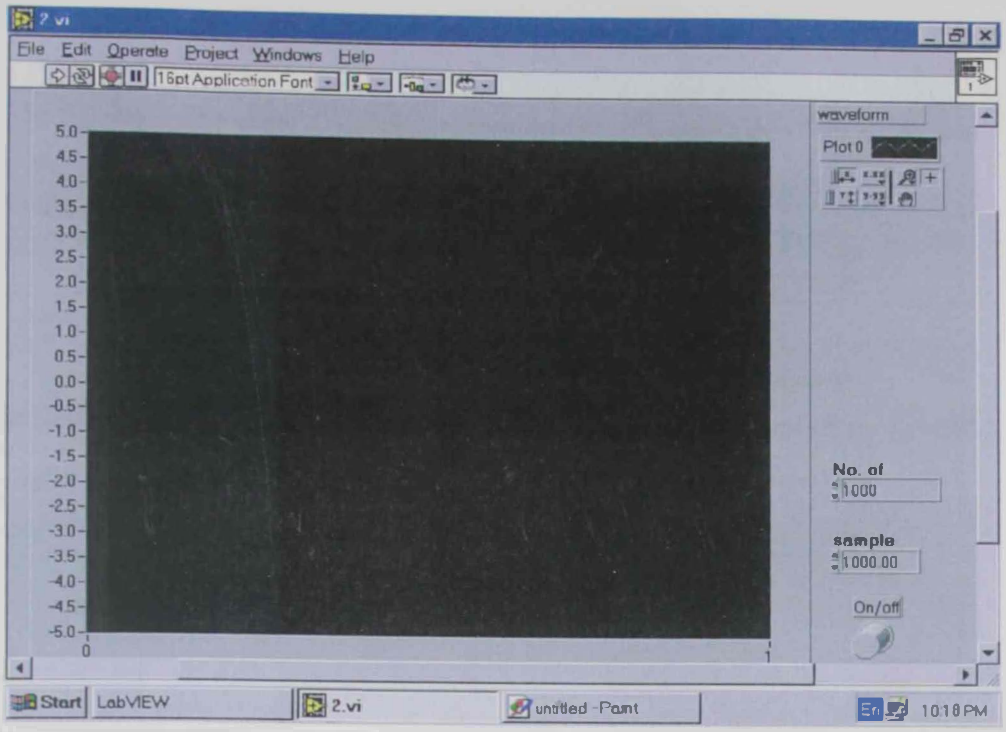


Figure 3 11: The Labview display screen

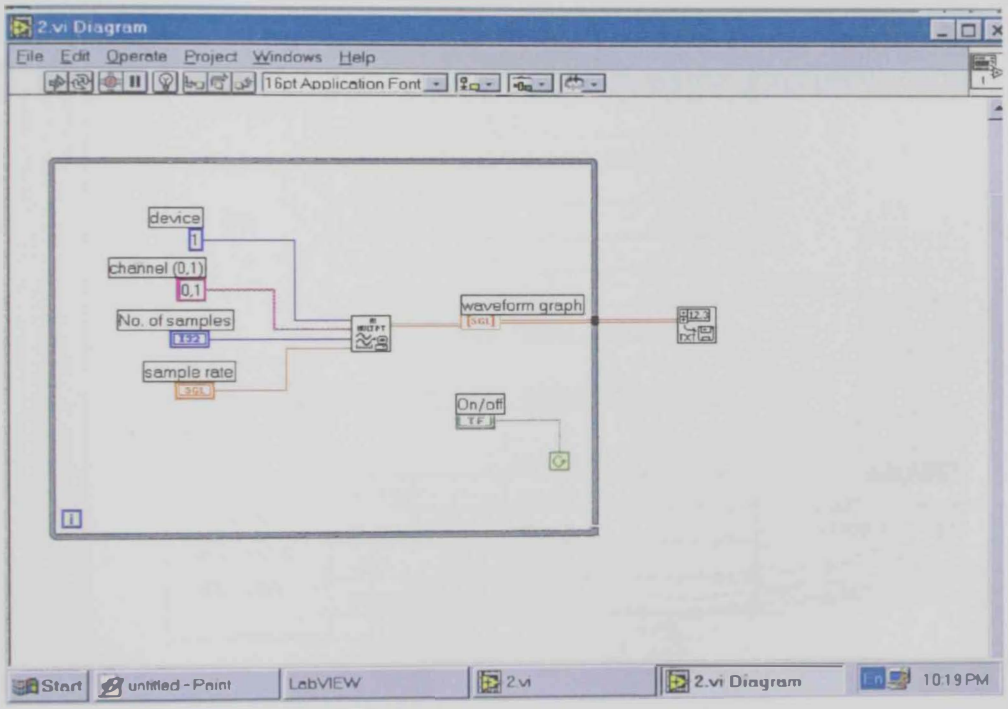


Figure 3.12: The Labview program

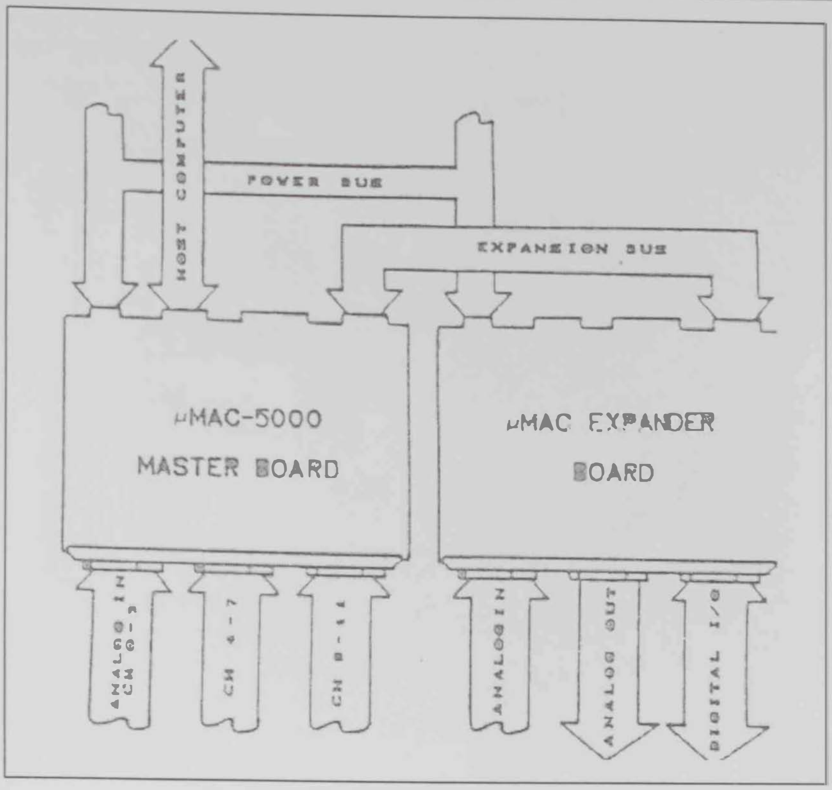


Figure 3.13: µMAC-5000 measurement and control system

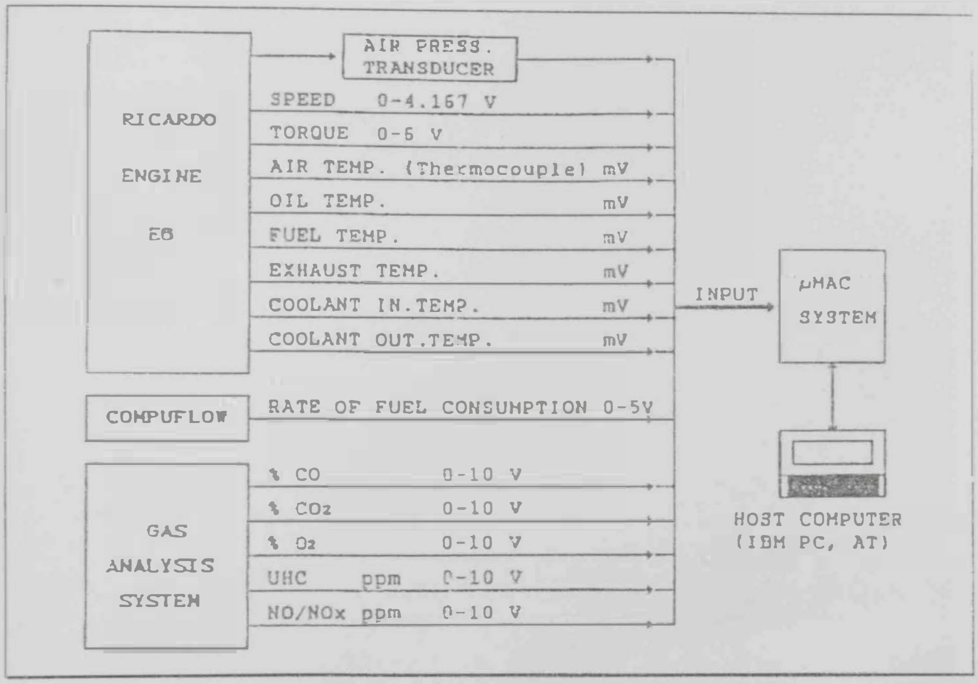


Figure 3.14: General connections between the µMAC system and the measuring instrument

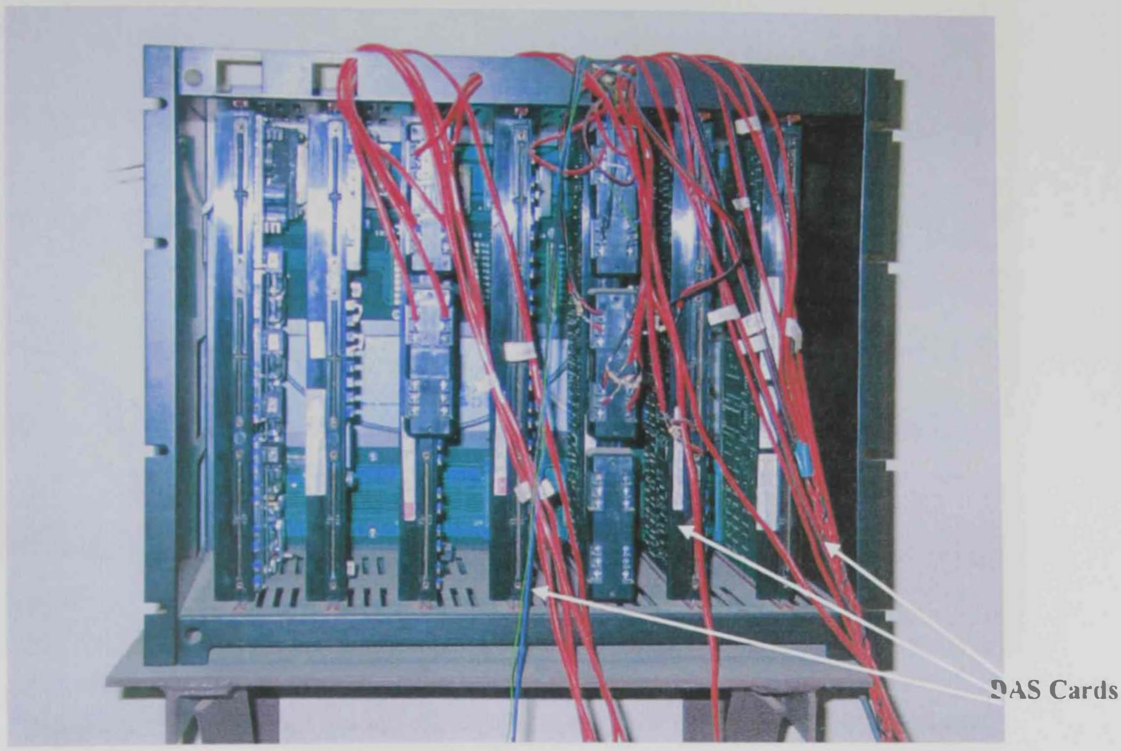


Figure 3.15. Photo of the DAS Cards and connections to sensors

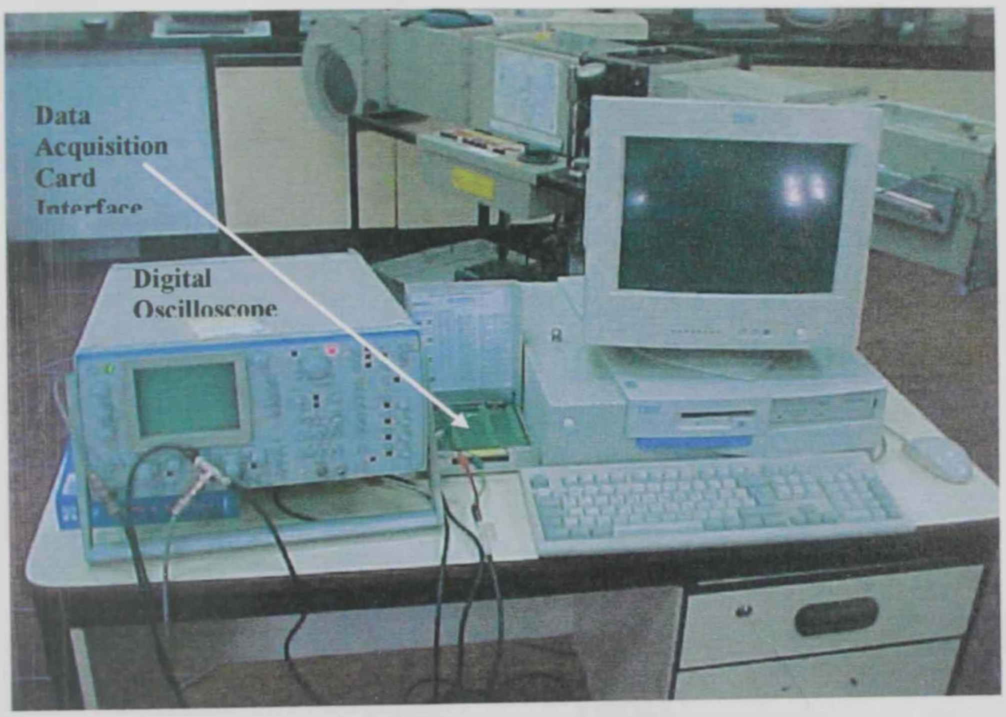


Figure 3.16: Photo of the storage oscilloscope and pressure-crank angle acquisition card

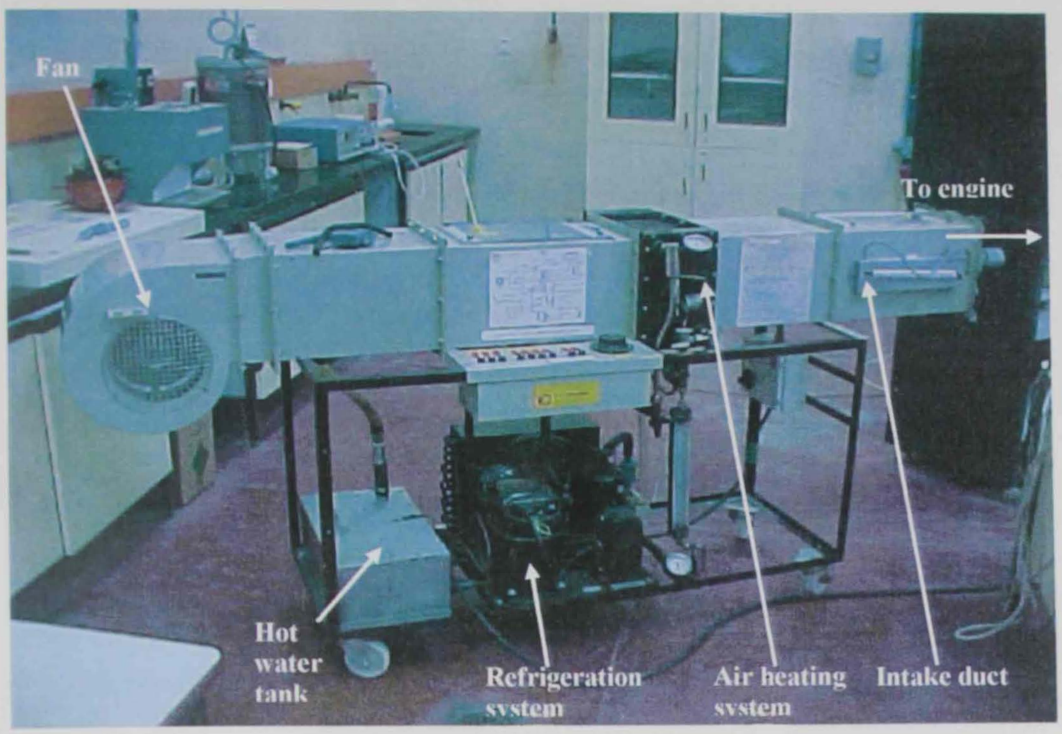


Figure 3.17: Photo of Air conditioning unit with steam generation system

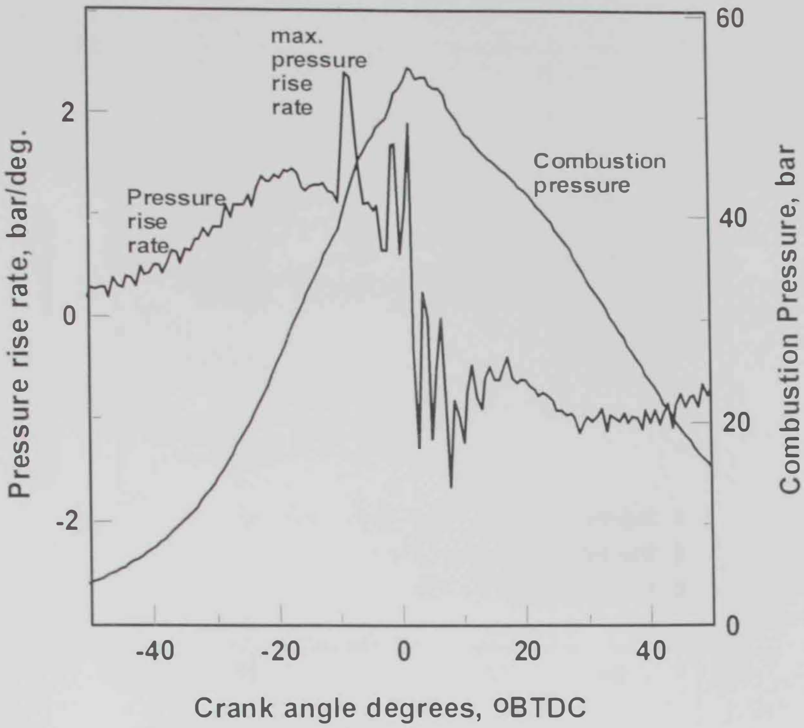


Figure 3.18 Sample of combustion pressure and pressure rise rate

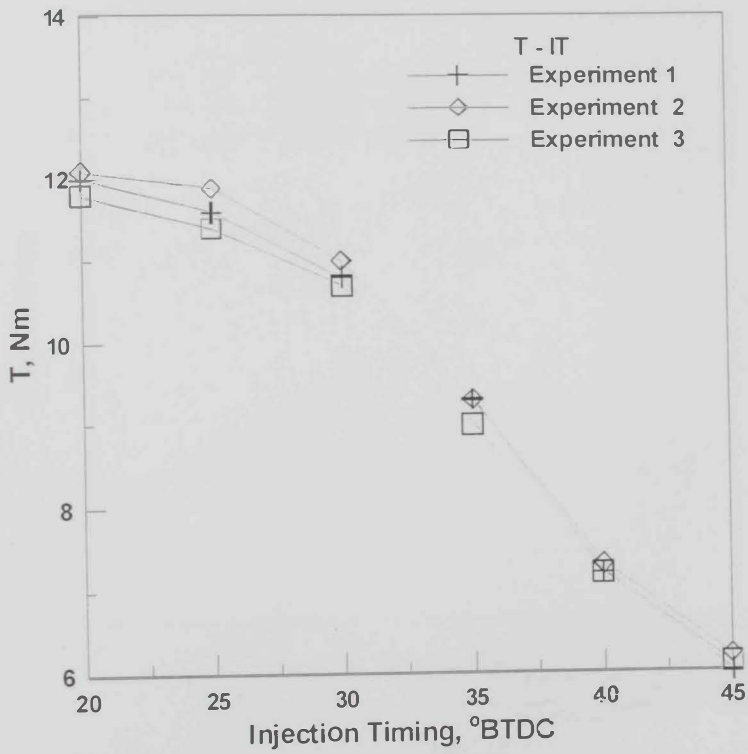


Figure 3.19 Repeatability of torque readings at different injection timing

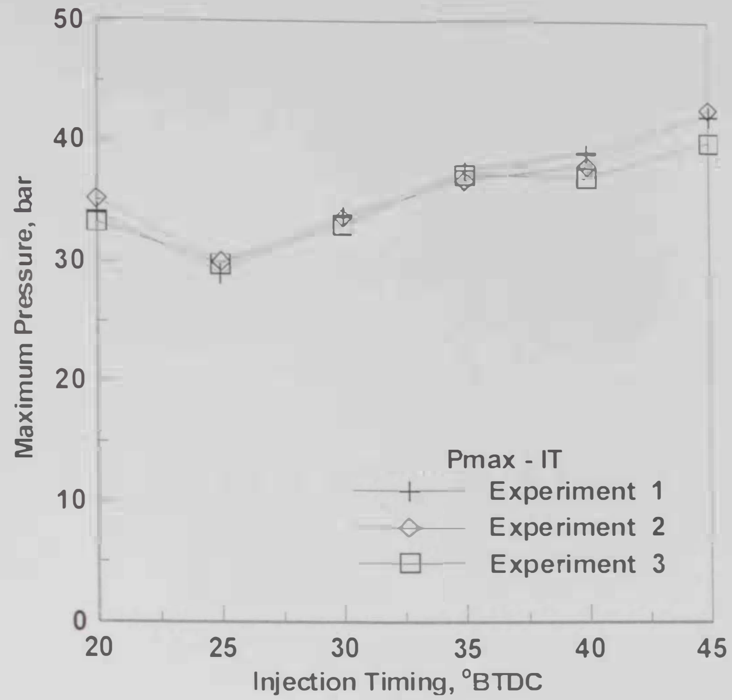


Figure 3.20 Repeatability of maximum pressure readings at different injection timing

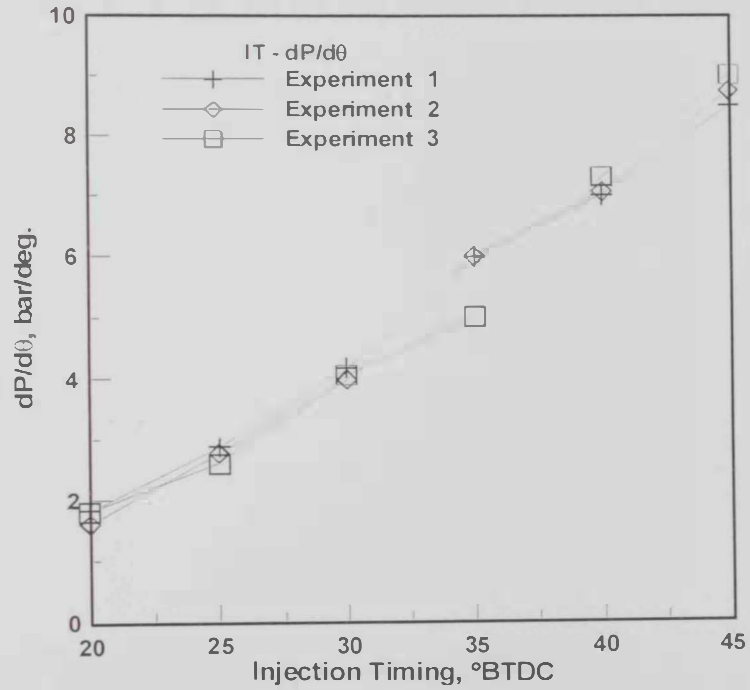


Figure 3.21 Repeatability of maximum pressure rise rate readings at different injectio

CHAPTER IV

RESULTS AND DISCUSSIONS

4.1 Introduction

A discussion is given in this chapter about the results taken from the dual fuel engine running on diesel and LPG, and the effects of water addition and some design and operating parameters on the engine performance and emissions are presented. The studied parameters are the engine speed, the engine load, the pilot fuel mass injected the injection timing of the pilot fuel and the compression ratio. The measured parameters included the brake horsepower output of the engine, the brake specific fuel consumption, the maximum combustion pressure, the maximum combustion pressure rise rate, CO, HC, and smoke emissions.

4.2 Effect of water addition on brake power output

4.2.1. Effect of engine speed

Figure 4.1 shows the experimental results of the brake power output of the engine as a function of the engine speed and water addition. It can be seen form the figure that increasing the engine speed generally increases the brake power output. The increase in the engine speed increases the liquid fuel flow rate as well as the gaseous fuel flow rate which increase the heat release rate and this increases the brake power output of the engine. It may be seen also from the same figure that increasing the water addition level from 6 g/kg to 14g/kg then to 24 g/kg generally reduce the brake power output of the engine, however this reduction in the brake power output is not much compared to the benefit of using the water in the engine as the NOx is proved to reduce by other researchers. For example the brake power output form the engine at speed of 30 rps (1800 rpm) and at 6g/kg water (atmospheric conditions) is 1775 Watt, while for the highest specific humidity of 24g/kg it is 1650 Watt. The percentage drop for 14 g/kg is 4.2 % and for 24 g/kg is 7%.The drop in the brake power output as the water addition increases may be attributed to the reduction in the maximum combustion temperature and pressure which reduces the brake mean effective pressure.

4.2.2. Effect of mass of pilot fuel

Figure 4.2 Shows the effect of the mass of pilot fuel injected on the brake power output for the different levels of water addition. The first line is for specific humidity 6g/kg, the second line is for 14g/kg, while the third line is for 24g/kg. It may be seen from the figure that increasing the mass of pilot fuel generally increases the brake power output of the engine for all ratios of water level. This increase is due to the increase of the mass pilot fuel which increases the heat released from the injected fuel. However any increase in the water addition ratio causes a drop in the brake power output, and this drop is also expected as the mean effective pressure is reduced duel to the drop in the maximum combustion temperature and pressure. However it may be seen that the drop in the brake power output is less for small amounts of pilot fuel, and bigger for higher amounts of pilot fuel. It maybe mentioned here that the dual fuel engine generally uses small amount of pilot fuel which means the drop in the brake power output will be very small, and there is no need to go to higher amount of pilot fuel.

4.2.3. Effect of injection timing

Figure 4.3 shows the effect of water addition and injection timing on brake power output. It may be seen from the figure that increasing the injection timing advance from 20 to 45 degree before top dead center generally reduces the brake power output. Increasing the injection advance causes the combustion to start earlier, which leads to an increase in the combustion temperature and pressure. This increase occurs in the compression stroke which decreases the mean effective pressure and reduces the net work of the engine and the brake power output. It may be also seen from the figure that the water addition decreases the brake power output slightly.

4.2.4. Effect of compression ratio

Figure 4.4 shows the effect of compression ratio and water addition on brake power output of the dual fuel engine. It can be noticed from the figure that the increase in the compression ratio from 14 up to 22 generally increases the brake power output of the engine. Increasing the compression ratio of the engine increases the thermal efficiency of the ideal cycle which in turn increases the brake power output of the engine. It may be noticed that the water addition decreases the brake power output of the engine at most of the compression ratios except for compression ratio 22. It may be noticed also that for compression ratio of 22 the brake power output is slightly decreased. This may be due to the fact of running the dual fuel engine at high compression ratio of 22 and because of the engine knocking which tends to reduce the brake power output. It's not recommended, therefore, that the dual fuel engine uses compression ratio of 22 – to avoid the engine knocking. The use of slightly lower compression ratios e.g. 20 would produce higher brake power output and even in the case of using high amount of water in the admitted air (24 g/kg) the engine would give the same power as of compression ratio of 22. This is favorable in the addition also of the knocking-free run of the engine.

4.3 Effect of water addition on brake specific fuel consumption

4.3.1. Effect of engine speed

Figure 4.5 shows the effect of the engine speed and the water addition on brake specific fuel consumption. It may be seen from the figure that increasing the engine speed generally increases the brake specific fuel consumption (bsfc) for all water addition ratios. The brake specific fuel consumption is calculated from the mass of the pilot fuel divided by the brake power output of the engine. Increasing the engine speed increases both the mass of pilot fuel and mass of gaseous fuel as well as the brake power output of the engine. It seems that the mass of the fuel increases more than the increase in the brake power output that's why the brake specific fuel consumption tends to increase. It may be also noticed from the figure that the increase in the water addition form 6 g/kg to 14g /kg increases the brake specific fuel consumption. This increase is due to one parameter. This parameter is the drop of the engine in the brake power output with the water addition as may be seen from figure 4.1.

4.3.2. Effect of pilot fuel mass flow rate

Figure 4.6 shows the effect of the water addition and the mass of pilot fuel on the brake specific fuel consumption. The dual fuel engine is running on both fuels at the same time; pilot and LPG. It may be seen from this figure that the increase in the mass pilot fuel generally decreases the brake specific fuel consumption. The decrease in the specific fuel consumption is due to the increase in the brake power output of the engine, as may be seen in figure 4.2. However the mass of the gaseous fuel (LPG) is constant. It may be also seen from the figure that adding more water will increase the specific fuel consumption at all masses of pilot fuel.

4.3.3. Effect of injection timing

Figure 4.7 depicts the effect of the water addition and injection timing on brake specific fuel consumption for the three water levels tested. It may be seen that advancing the injection timing form 20 to 45 degrees before the top dead center generally increases the brake specific fuel consumption. The increase in the bsfc is due to the drop in the brake power output of the engine as a result of increasing the pressure and temperature earlier in the compression stroke which decreases mean effective pressure. Also increasing the water addition level from 6 to 14 g/kg then to 24 g/kg increases the brake specific fuel consumption slightly for the same reasons mentioned above.

The increase in the brake specific fuel consumption is almost the same for all injection timing. For the brake power output and brake specific fuel consumption it may be concluded to run the engine at early injection timing for higher brake power output and less brake specific fuel consumption at all water addition ratios while increasing the injection timing to 45 for example will reduce the brake power output of the engine and increases the specific fuel consumption.

4.3.4. Effect of compression ratio

Figure 4.8 gives the effect of the water addition and compression ratio on the brake specific fuel consumption for the three levels of water addition. It may be seen that increasing the compression ratio from 14 to 16, 18, 20 to 22 generally decreases the brake specific fuel consumption. This decrease is due to the increase in the brake power output of the engine. However it may be seen from the figure that at compression ratio of 22, the brake specific fuel consumption tends to increase. Increasing the water addition from 6g/kg to 14g/kg then to 24 g/kg specific humidity generally increases the brake specific fuel consumption and this is due to the drop in the brake power output. However this increase is slightly low at compression ratio of 20. It's recommended that the engine uses compression ratio of 20 to prevent the engine knocking.

4.4 Effect of water addition on maximum pressure rise rate

4.4.1. Effect of engine speed

Figure 4.9 shows the effect of the engine speed and the water addition on the maximum pressure rise rate for the three water addition levels of 6, 14 g/kg and 24g/kg specific humidity. It may be seen from this figure that the increase in the engine speed from 16 to 32 revolutions per second generally decreases the maximum pressure rise rate $(\frac{dP}{d\theta})_{\max}$.

The maximum pressure rise rate is a measure of the combustion noise and it

is directly related to it. Higher pressure rise rate produces higher combustion noise and vice versa. Increasing the engine speed increases the mixing between air and fuel which makes the combustion to start smoother and hence decreases the ignition delay period. This reduction in the ignition delay period causes the maximum pressure rise rate to decrease so the engine will be running more smoothly especially at higher engine speeds. It may be seen also from the same figure that increasing the water addition increases the maximum pressure rise rate in the engine. Adding more water to the engine decreases the maximum combustion temperature and increases the ignition delay period and this increases the maximum pressure rise rate so the engine will be nosier at a higher water level. However, at the highest speed of (30 rps) the maximum pressure rise rate is about 4 bars/degrees which is much lower than the value at low engine speed. Therefore running the engine at high speed with high water level would decreases the NO_x emission, because of the decrease in temperatures, and also decreases the combustion noise.

4.4.2. Effect of pilot fuel mass flow rate

Figure 4.10 shows the effect of water addition and mass of pilot fuel on maximum pressure rise rate $(\frac{dP}{d\theta})_{max}$. It may be seen from the figure that increasing the mass of pilot fuel increases maximum pressure rise rate. Increasing the mass of pilot fuel means that it will stay longer time for this amount to be ignited; or longer ignition delay ignition. The increase in the ignition delay period causes the maximum pressure rise to increase and the engine will run noisier at high amount of mass of pilot fuel used. For the dual fuel engine, it should be noted that the use of mass of pilot fuel should as low as possible as the main fuel here is the gaseous fuel and the function of the pilot fuel is just to ignite it. Therefore running the engine at small pilot fuel will reduce the maximum pressure rise rate. It may be seen also from the same figure that increasing the water addition level resulted in an increase in maximum pressure rise rate. The increase in the pressure rise rate is due to the increase in the ignition delay period as adding more water to the mixture lead to lower temperatures which in turn increases the ignition delay and produces more pressure rise rate. However at very small amount of pilot fuel mass this increase in pressure rise rate is very small, i.e. the engine will not be noisy even if we add more water.

4.4.3. Effect of injection timing

Figure 4.11 shows the effect of water addition and pilot fuel injection timing on maximum pressure rise rate. It may be seen from the figure that advancing the injection timing leads to higher pressure rise rate. Increasing the injection timing advance means injecting the pilot fuel earlier and earlier at low pressures temperatures. Reducing the pressure and temperature at the injection increases the ignition delay period which in turn increases the pressure rise rate.

It may be seen also from the figure that increasing the water addition form 6g/kg to 14g/kg to 24 g/kg lead to an increase in pressure rise rate stated above in the previous figure.

4.4.4. Effect of compression ratio

The effect of water addition and compression ratio on the maximum pressure rise rate may be depicted in Fig. 4.12. It may be noticed from this figure that increasing the compression ratio from 14 to 22 generally increases the maximum pressure rise rate or combustion noise. The increase in the compression ratio causes all pressures and temperatures to increase during the engine cycles. This increase may lead a more probability of self ignition of LPG before the pilot flame ignites it. The self ignition of the gaseous fuel (low ignition temperature for LPG) causes the pressure rise rate to increase. It is advised that for t he dual fuel engine when it runs on LPG, to reduce the compression ratio to avoid having the engine running at high combustion noise or even knocking.

Similar trend for the water addition may be noticed to the previous figures and the advantage of adding water may be greatly noticed when the engine runs at lower compression ratios as the maximum pressure rise rate is small.

4.5 Effect of water addition on maximum combustion pressure

4.5.1. Effect of engine speed

The effect of the engine speed and water addition on the maximum combustion pressure may be seen in Fig. 4.13. As may be noticed from the figure that increasing the engine speed from 18 to 30 revolutions per second generally decreased the maximum combustion pressure for the same amount of water exists in the inlet air. The increase in the speed has shown a decrease in the maximum pressure rise rate (Fig. 4.9) above. The decrease in the maximum pressure rise rate means a slow increase of the combustion pressure during the compression stroke while the piston is going inward. This slow increase in the pressure causes most of the combustion to occur during the expansion stroke and reduce the maximum pressure of the cycle as shown in Fig. 4.13. On the contrary, a decrease in the engine speed causes the pressure rise rate to increase or the pressure goes up fast during the compression stroke which causes the maximum pressure to increase.

Similar to previous effects, the increase in the water added in the air from 6 to 14 to 24 g kg caused the maximum combustion pressure to increase. This increase is due to the fact that adding more water increased the pressure rise rate (above) and it then increased the maximum pressure.

4.5.2. Effect of pilot fuel mass flow rate

Figure 4.14 shows the effect of the mass of pilot fuel and the amount of water on the maximum pressure. It may be noticed from this figure that increasing the mass of pilot fuel increases the maximum pressure of the cycle. This increase is a result of the increase of the maximum pressure rise rate as shown in fig. 4.10. The maximum pressure of the cycle increased from about 60 bars at the lowest amount of pilot to about 71 bars at the highest amount. Similar to previous effects, when the water added is increased the maximum pressure has been also increased as a result of the increase of the pressure rise rate.

4.5.3. Effect of injection timing

Figure 4.15 depicts the effects of increasing the injection timing advance and the water added on the maximum pressure of the cycle. It may concluded from this figure that advancing the injection of the pilot diesel fuel generally increases the maximum pressure of the cycle as the pressure rise rate becomes higher; Fig.4.11. The dual fuel engine becomes noisy when the injection is advanced and it may be recommended to retard the injection to limit its noise and prevent high knocking of the engine which may reduce the life of it as well as deteriorating the performance of the engine.

Adding more water in the inlet air slightly increases the maximum pressure of the cycle as may be seen from the figure.

4.5.4. Effect of compression ratio

The effects of increasing the compression ratio and adding more water on the maximum pressure of the cycle may be shown in Fig.4.16. It may be concluded from the figure that increasing the compression ratio from 14 to 22 generally increases the maximum pressure of the cycle. This increase is a result if two parameters which lead to the sharp increase from about 35 to about 70 bars. The first is the increase in the maximum pressure rise rate which causes the combustion to accelerate and increase the maximum pressure of the cycle. The second is the increase in the whole cycle pressures as a result of the increasing the compression ratio. This includes the pressure during the compression stroke before the injection of the pilot fuel which leads to an increase in the maximum pressure of the cycle. The increase in the water added also increased the maximum pressure of the cycle as seen in the figure. It is also recommended to reduce the compression ratio of such dual fuel engine to reduce the maximum pressure rise rate or the combustion noise. There has been like sharp increase in the pressure rise rate and maximum pressure when the engine compression ratio increased from 20 to 22.

4.6 Effect of water addition on CO emissions

4.6.1. Effect of engine speed

Figure 4.17 shows the effect of water addition and the engine speed on CO emissions. It may be seen from this figure that increasing the engine speed increases the CO emission up to a speed of 28 revolutions per second, then starts to fall down and this occurs at all water addition levels. Increasing the engine speed generally increases the mass of fuel used and this tends to increase the CO emission. It may be seen also from the same figure that increasing the water addition level from 6 to 14 g/kg to 24 g/kg increases the CO on the exhaust gasses as injecting more water in the inlet air tends to dilute the mixture more with water and this makes the mixture more diluted and it may give higher level of CO emissions in the exhaust.

4.6.2. Effect of mass of pilot fuel

Figure 4.18 shows the effect of water addition and mass of pilot fuel on CO emissions. It may be seen from the figure that increasing the mass of pilot fuel generally decreases the CO emissions. This is due to the more complete combustion that occurs at higher mass of pilot fuel injected, as higher mass of pilot fuel injected gives bigger and bigger flames, and this tends to burn completely the gaseous fuel that occur in the combustion chamber. However, it may be also seen that at very high level of mass of pilot fuel there is an increase in CO emissions and this may be due to some unburned fuel that comes out of the diffusion flame at higher mass of pilot fuel. It may be also seen from the same figure that increasing the water level in the inlet air increases in the CO emissions in the exhaust gasses. This is due to the increased dilution of the inlet air fuel mixture which gives more amounts of CO emissions in the exhaust gasses.

4.6.3. Effect of injection timing

Figure 4.19 shows the effect of water addition and the injection timing on CO emissions for the three tested water addition level of 6, 14, 24g/kg. It may be seen from the figure that the CO emissions in the exhaust gasses is maximum at injection timing of 25-30 degrees before Top Dead Centre (TDC) , and it is minimum at very late or very early injection timing. At very late injection timing, the CO emission is very low as the injection occurs very early during the compression stroke. Injecting the fuel during the compression stroke give more time for the combustion to occur and burn all the CO emissions that will result in the exhaust gasses, hence the CO emissions will decrease. Increasing the water addition from 6 to 14 to 24 g/kg generally increases the CO emission in the exhaust gasses because of the dilution effect as more water exists in the fuel air mixture and this tends to increase the CO emissions in the exhaust gasses. However the level of the CO emission is far below the allowable limits with exhaust gasses.

4.6.4. Effect of compression ratio

Figure 4.20 shows the effect of water addition and the compression ratio on the CO emission. It may be seen from the figure that increasing the compression ratio from 14 to 22 increases generally the CO emission in the exhaust gasses, however the CO emission level is far below the allowable limits for the current exhaust gasses limits. Increasing the water addition level from 6 to 14 then to 24 g/kg generally increases the CO emission because of the dilution effect as more water is injected in the exhaust gasses.

4.7 Effect of water addition on HC emissions

4.7.1. Effect of engine speed

Figure 4.21 shows the effect of water addition and the engine speed on CO emission. It may be noticed from the figure that increasing the engine speed generally decreases the HC emission for all water addition level tested. Increasing the engine speed increases the swirl and turbulence level in the combustion chamber and this improves the mixing between the liquid fuel injected, the gaseous fuel and the air. More mixing enhances the combustion and should reduce the HC emission.

Increasing the water additions from 6 to 14 to 24 g/kg increases the HC emission in the exhaust gasses as more dilution occurs and more incomplete combustion and this leads to higher levels of HC emission in the exhaust gasses. However, all are below the allowable limits for the HC emissions in the exhaust gasses. Moreover the technology of catalytic converter is very well advanced and it should be relatively easy to reduce the HC emission lower by using it.

4.7.2. Effect of mass of pilot fuel

The effect of water addition and the mass of pilot fuel on HC emission may be seen in figure 4.22. It may be noticed from the figure that increasing the mass of pilot fuel generally decreases the HC emission in the exhaust gasses. The reason for this is that increasing the mass of pilot fuel tends to make the flame bigger and bigger, and this tends to completely burn the gaseous fuel that exist in combustions chamber. This reduces the HC emission in the exhaust gasses. Increasing the water addition level generally increases the HC emission because of the dilution effect. Nonetheless, as mentioned above that using a catalytic converter may help to reduce the HC emission even when the water is increased.

4.7.3. Effect of injection timing

The effect of water addition and the injection timing on HC emission may be depicted in figure 4.23. It may be noticed form the figure that increasing the injection timing advance from 20 to 45 degrees before the Top Dead Centre generally decreases the HC emission in the exhaust gasses for the three tested water addition level. Starting the combustion earlier and earlier during the compression stroke gives higher temperature and the pressure in the combustion chamber. Higher temperature and pressure gives more chance for more complete combustion and gives less HC. Similar effect of adding the water to the combustion chamber has shown, that increased the HC emission because of the dilution effect. More water is injected in the combustion chamber gives higher level of emissions of HC.

4.7.4. Effect of compression ratio

Figure 4.24 shows the effect of water addition and compression ratio on HC emission. It may be seen from the figure that HC emission becomes higher and higher with increasing the compression ratio of the engine. The reason for this may be due to the higher knocking possibility of the engine at higher compression ratio which tends to increase the unburned Hydrocarbon in exhaust gasses. Similar trends of adding the water to the HC emission is noticed that resulted in increasing the HC emission in the exhaust gasses because of the more dilution effect.

4.8 Effect of water addition on Smoke emissions

4.8.1. Effect of engine speed

Figure 4.25 shows the effects of adding the water to inlet air and the engine speed on opacity or the smoke level of the exhaust gases. It may noticed from the figure that increasing the engine speed generally increases the opacity level of the exhaust gases for the three water addition levels tested. Increasing the engine speed means more fuel is injected in the combustion chamber and more fuel is burned. At the same time there is a less time for combustion of this high amount of fuel which tends to increase the opacity or the smoke level in the exhaust gases. Increasing the water addition level in the inlet air has resulted in increasing the opacity level due to more water exists in the combustion chamber that leads to more dilution effect and more unburned carbons atoms that lead to higher level of smoke in the exhaust gases.

4.8.2. Effect of mass of pilot fuel

The effect of the water addition and the mass of pilot fuel on opacity level is given in figure 4.26. It may be noticed form the figure that increasing the mass of pilot fuel injected increases generally the opacity level of the exhaust gasses. More pilot fuel injected means higher amount of diesel fuel injected to ignite the gaseous fuel. This increases the diffusion combustion for the diesel fuel injected which leads to bigger possibility of having unburned Carbon atoms and higher level of opacity. The increase in the water addition level from 6 to 14 to 24 g/kg gives similar effect as before of an increase in the opacity level at all values of mass of pilot fuel tested.

The good advantages of the dual fuel engine would be then to utilize as small amount of pilot fuel as possible to reduce the smoke emissions in the exhaust gases. Adding more water to inlet air at small amount of pilot fuel did not increase much the opacity level.

4.8.3. Effect of injection timing

The effect of injection timing of the pilot diesel fuel on the opacity for the three tested water addition level may be seen in Fig. 4.27. It may be noticed from this figure that for injection timing of around 30° BTDC the opacity level is minimum and increases both sides for the late and very early injection. At this injection timing, it seems that the combustion is most complete and produces the least smoke in the exhaust. It seem that the 30° BTDC is optimum for this engine.

4.8.4. Effect of compression ratio

The effect of compression ratio and the water addition level on the opacity level of the exhaust gasses may be seen in figure 4.28. It may be seen from this figure that increasing the compression ratio of the engine generally decreases the opacity level of the exhaust gasses or the smoke level. Increasing the compression ratio tends to increase all the cycle temperatures and pressure which gives more complete combustion for all the hydrocarbon fuel and the less opacity level or smoke level of the exhaust gasses. The same effect for the water addition to increase the opacity level of the exhaust gasses occurs.

4.9 Knocking limits of dual fuel engine

4.9.1 Effect of torque on maximum pressure rise rate and maximum pressure

The effect of load and water addition level on the dual fuel knocking limit is illustrated in figures 4.29 a, b and c. The figure shows the increase of the maximum pressure rise rate or the combustion noise when the load output is increased. The load (or output torque) is increased from minimum value close to zero and until the engine starts to knock at very high amount of gaseous fuel. The torque output has been increased by increasing the amount of LPG gaseous fuel while maintaining the pilot fuel constant. The end of the experiment is when there is heavy knocking and any further increase to the gaseous fuel amount would decrease the output power (or torque) of the engine. That point is recognized as the onset of ignition failure.

Figure 4.29a shows the knocking limit (or the point of highest torque) to be around 10 Nm, while it is around 15 Nm for compression ratio of 20 (Fig. 4.29b) and it is around 16 Nm for compression ratio of 18 (Fig 4.29c). It may be concluded that the decrease in the compression ratio from 22 to 20 and then to 18 enabled the engine to run at higher torque output without heavy knocking or ignition failure. The reduction in the compression ratios reduces the cycle pressure and temperature which reduces the possibility of self-ignition of the LPG gaseous fuel and having knocking in the dual fuel engine. It is recommended therefore to reduce the compression ratio of the dual fuel engine to reduce the knocking possibility and reduce the combustion noise and enables the engine to run at higher output torques.

It may be also noticed that increasing the output torque leads to an increase of the maximum pressure rise rate. This is a result of the increase in the amount of LPG gaseous fuel admitted in the intake air which –when burned – produces high pressure rise rate.

Similar trend may be found for the maximum combustion pressure as shown in fig. 4.30a, b and c. the increase in the maximum pressure with increasing the torque is a result of the increase in the pressure rise rate shown above in fig. 4.29. The same conclusion for the knock limit is also seen here for the maximum pressure.

4.9.2 Effect of torque on brake specific fuel consumption

Figures 4.31a, b and c show the effect of output torque on the brake specific fuel consumption (*bsfc*) for the three tested water addition level tested and at different compression ratios. Fig. 4.31a represents the effect of torque on *bsfc* at a compression ratio of 18, while Fig. 4.31b is at compression ratio of 20 and Fig. 4.31c is at compression ratio of 22. By comparing these three figures, it may be noticed that decreasing the compression ratio from 22 to 20 then to 18 has resulted in an increase in the knocking limit or the maximum torque that the engine can produce without knocking. It may be also shown in these figures that increasing the output torque results in a decrease in the *bsfc*. The increase in the output torque is a result of the increase on the mass of LPG gaseous fuel which also produces more power output, therefore it seems that the increase in the power is more than the increase in the fuel mass and also due to an improvement of the engine efficiency at high loads.

4.9.3 Effect of torque on opacity

Figure 4.32 depicts the effect of output torque, compression ratio and the water addition level on the opacity level of the exhaust gases. Each compression ratio is illustrated in a separate Figure. The compression ratios of 18, 20 and 20 are shown in Figures 4.32a, 4.32b and 4.32c respectively. It could be seen from these three figures that at all compression ratios, the increase in the output torque has resulted in the increase in the smoke level or opacity level of the exhaust gases. Even at higher amount of water addition levels the same trend is noticed. This increase in the opacity may be due to the increase in the amount of gaseous fuel admitted with the inlet air. The increase in the gaseous fuel may have resulted in the increase of fuel to air ratio and this reduces the amount of oxygen available for the complete combustion of the fuel. This may have resulted in the increase in the opacity level of exhaust gases.

The comparison of Fig. 4.32(a) with Fig. 4.32(b) and Fig. 4.32(c) gives similar trend of reducing the opacity with increasing the compression ratio.

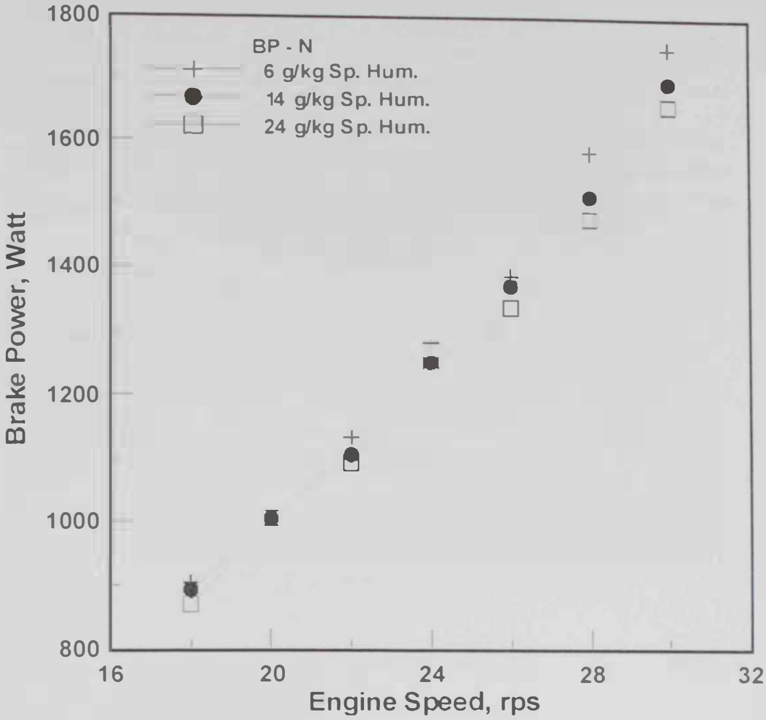


Figure 4.1. Effect of water addition and engine speed on brake power output

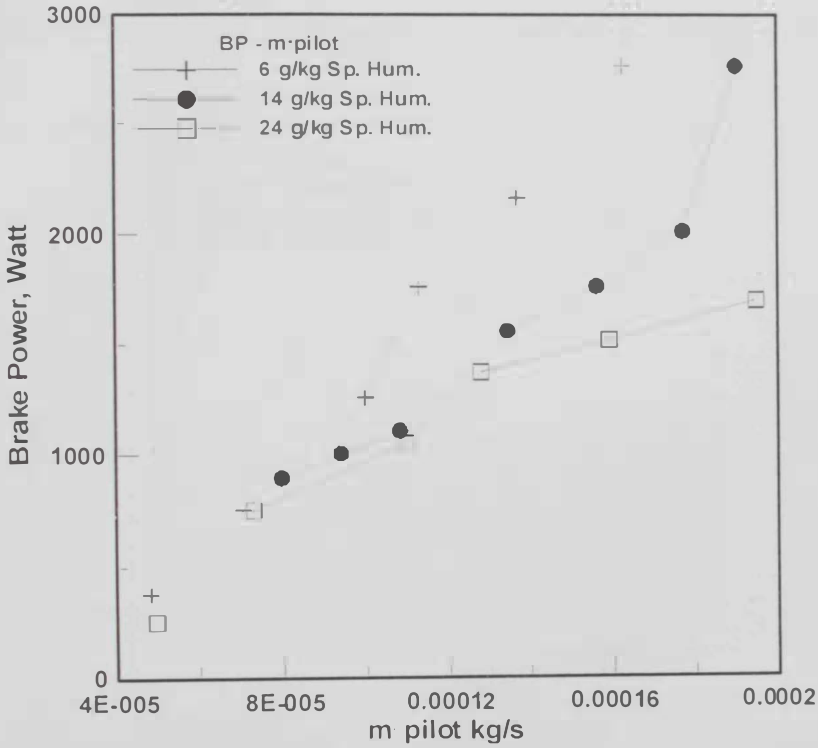


Figure 4.2. Effect of water addition and mass of pilot fuel on brake

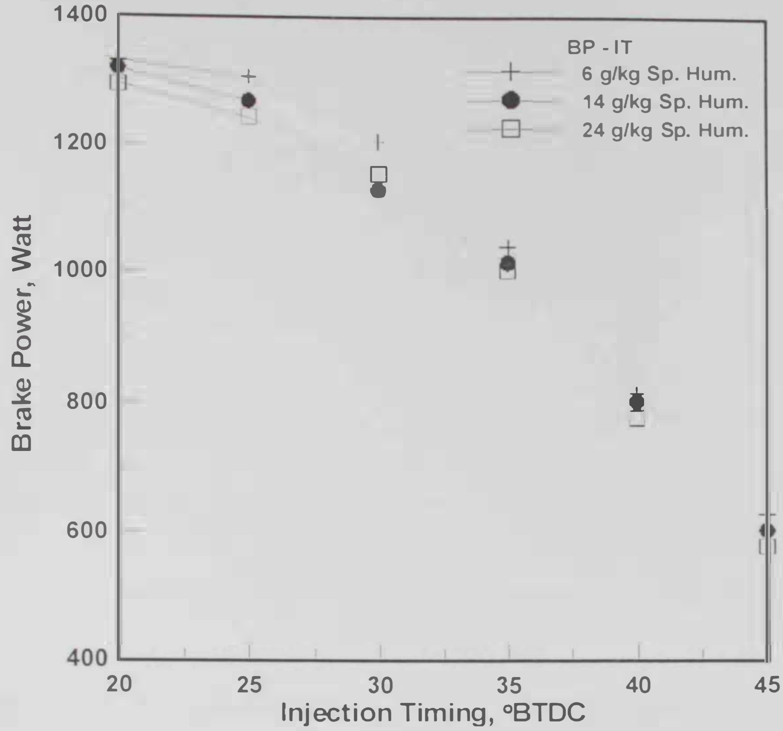


Figure 4.3. Effect of water addition and injection timing on brake power out

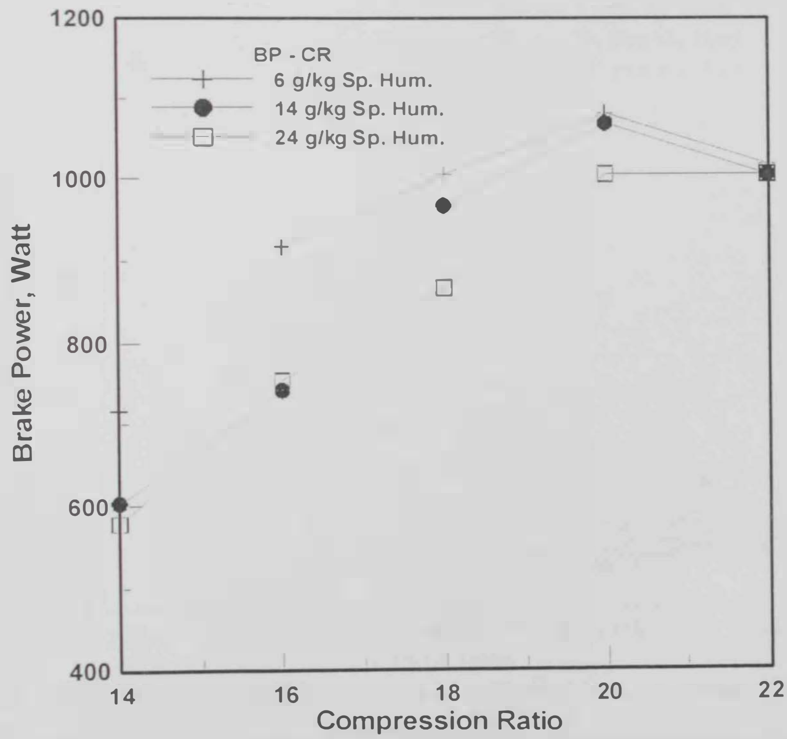


Figure 4.4. Effect of water addition and compression ratio on brake power output

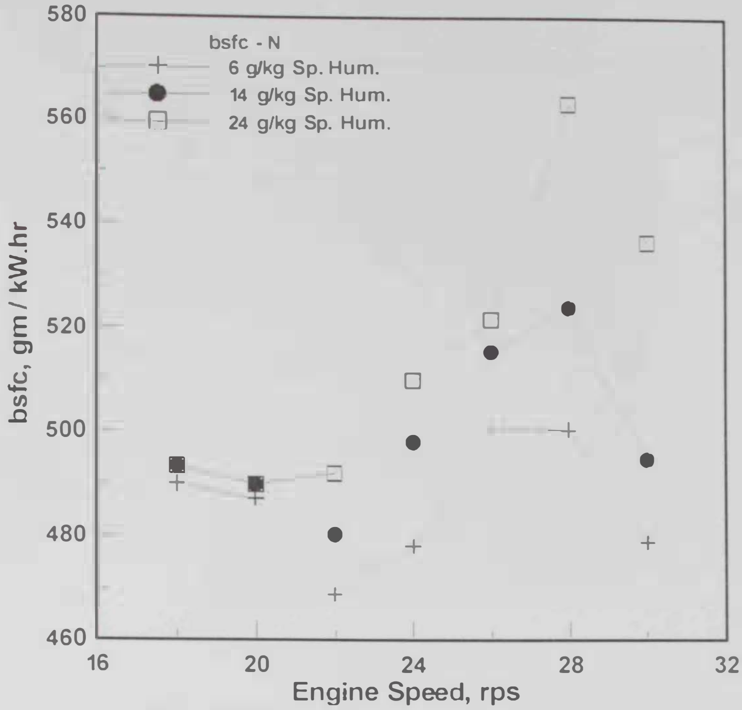


Figure 4.5 Effect of water addition and engine speed on brake specific fuel consumption

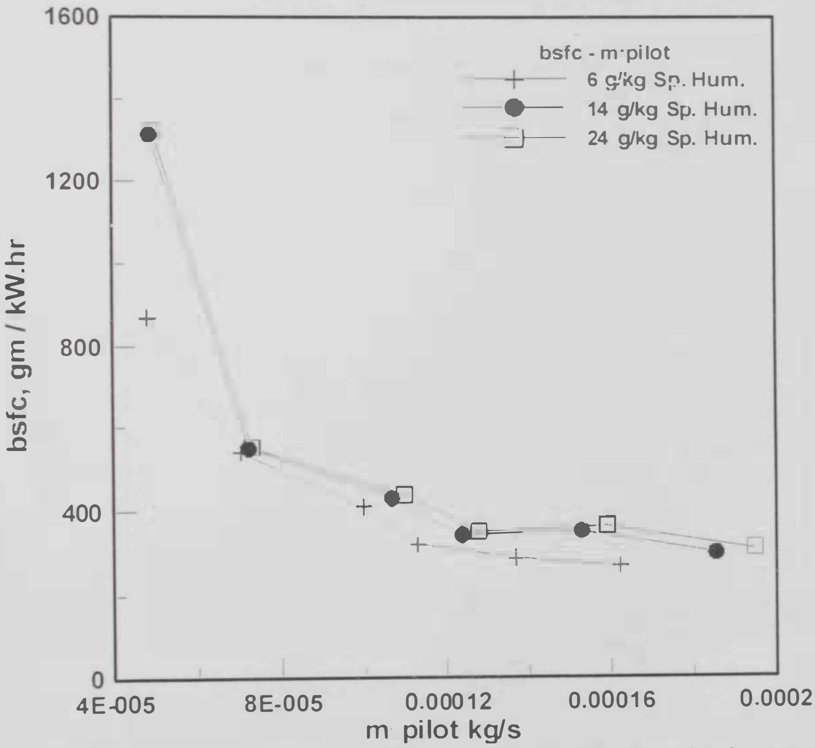


Figure 4.6. Effect of water addition and mass off pilot fuel on brake specific fuel consumption

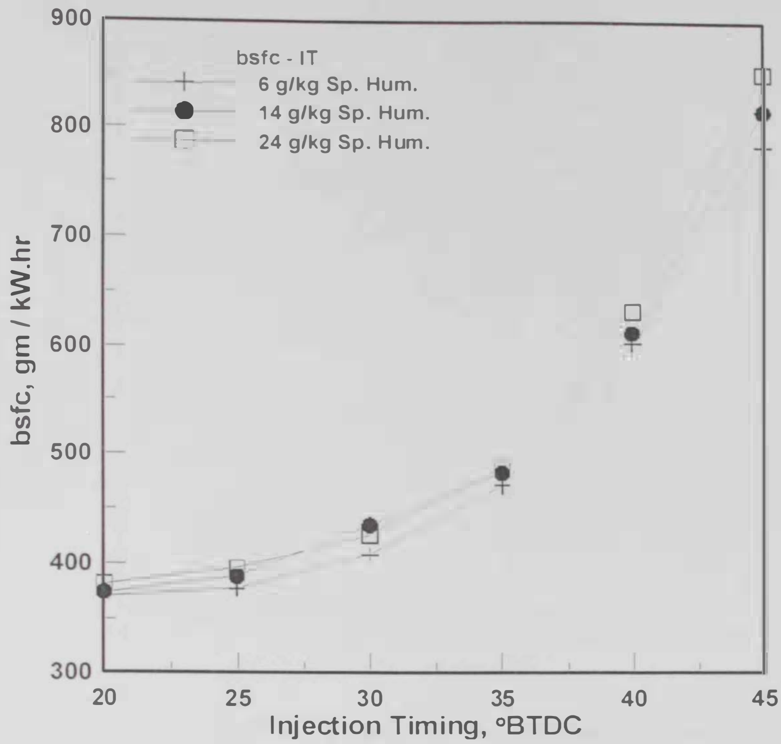


Figure 4.7. Effect of water addition and injection timing on brake specific fuel consumption

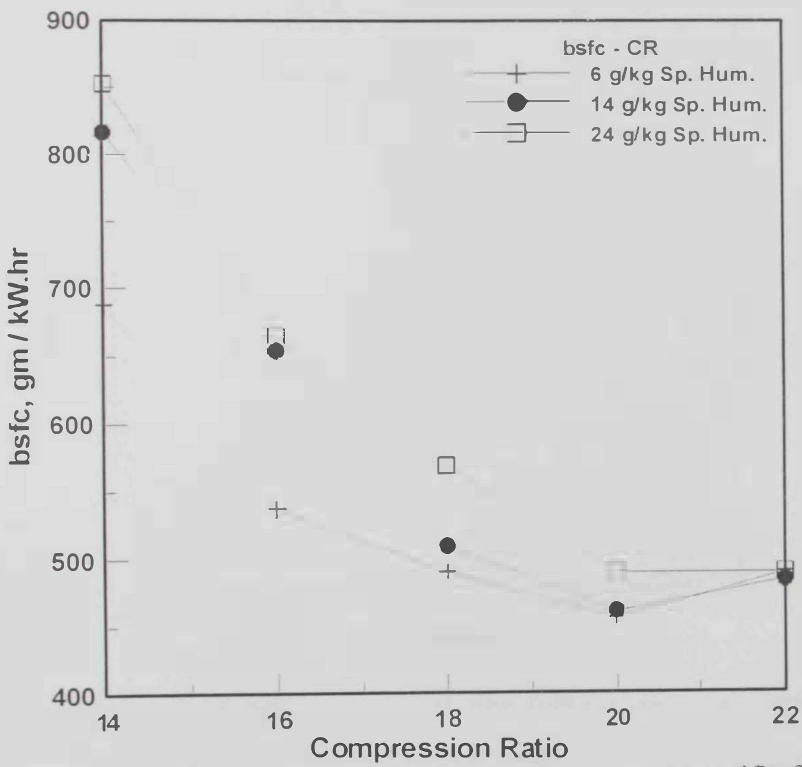


Figure 4.8. Effect of water addition and compression ratio on brake specific fuel consumptior

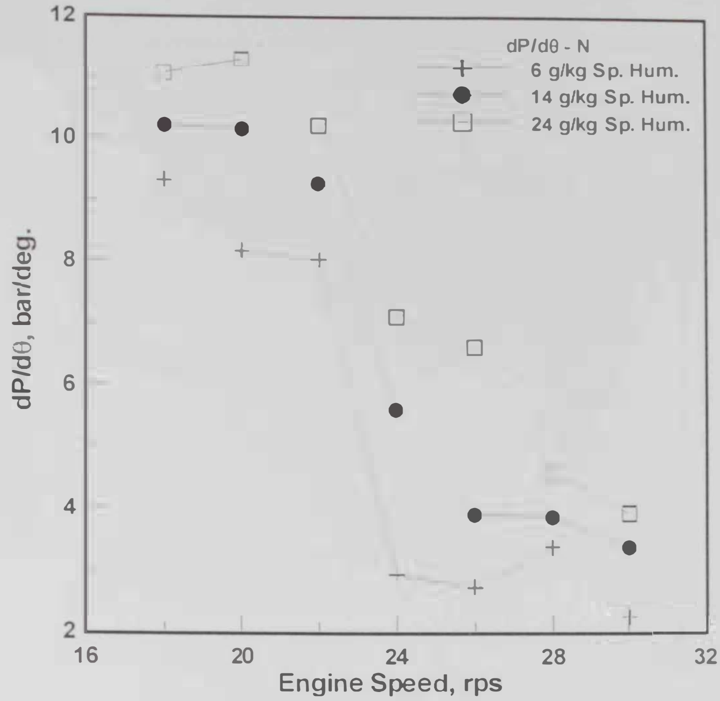


Figure 4.9 Effect of water addition and engine speed on maximum pressure rise rate

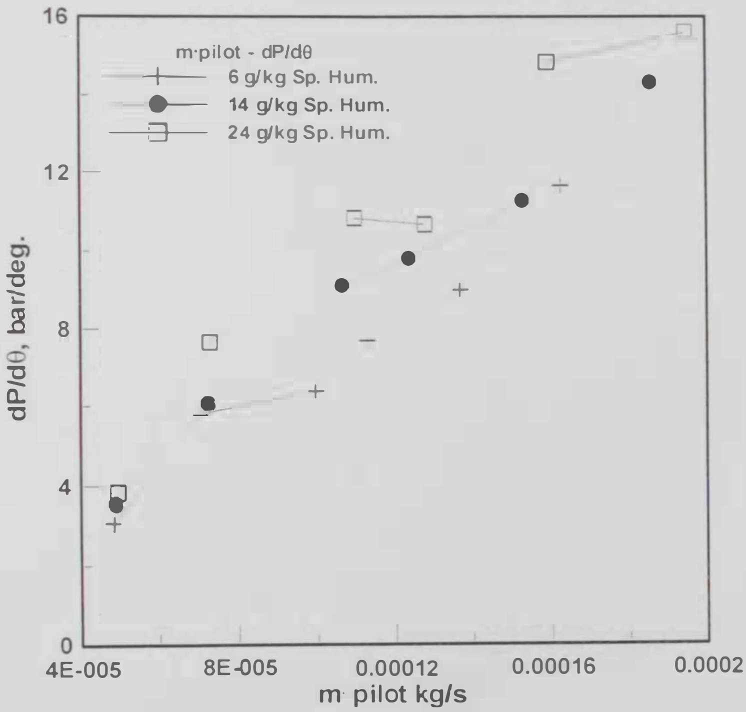


Figure 4.10 Effect of water addition and mass of pilot fuel on maximum pressure rise rate

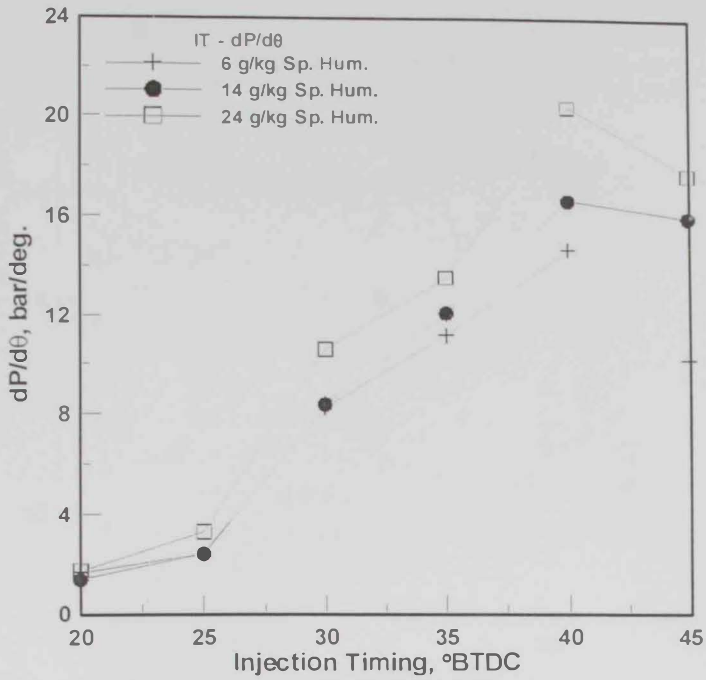


Figure 4.11 Effect of water addition and injection timing on maximum pressure rise rate

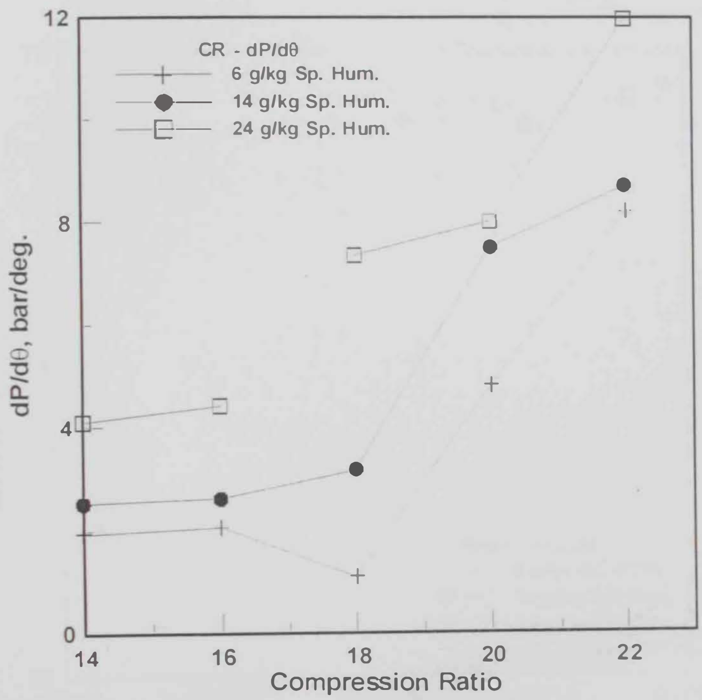


Figure 4.12 Effect of water addition and compression ratio on maximum pressure rise rat

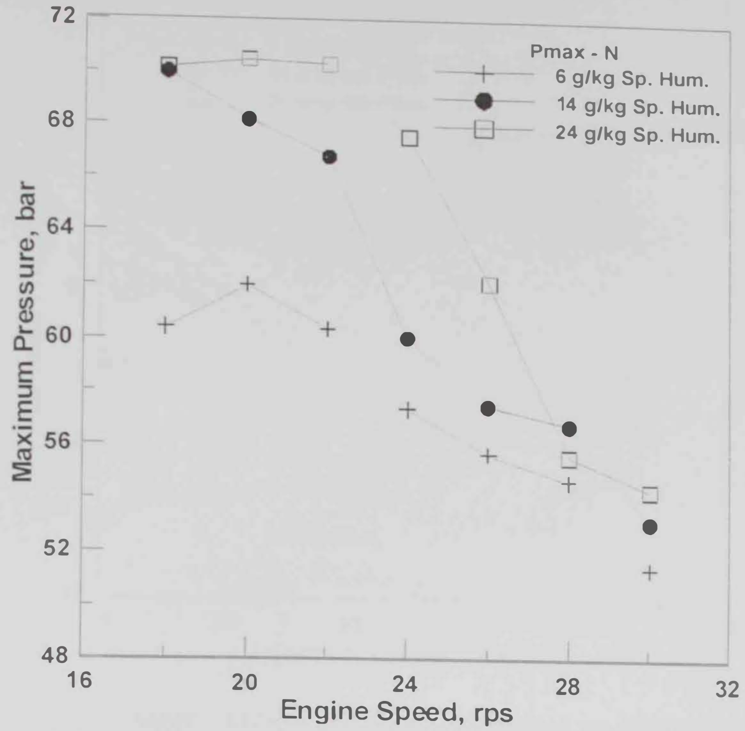


Figure 4.13 Effect of water addition and engine speed on maximum combustion pressure

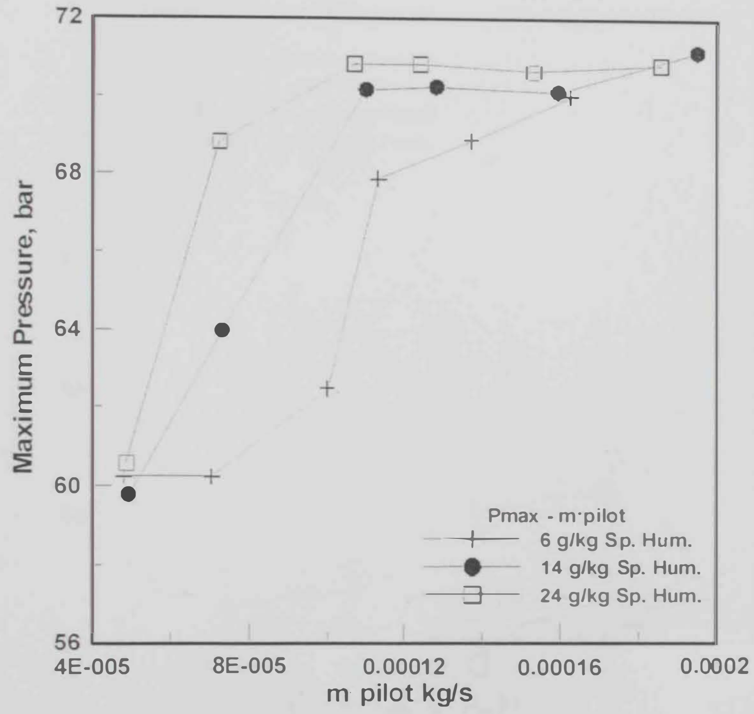


Figure 4.14 Effect of water addition and mass of pilot fuel on maximum combustion pressure

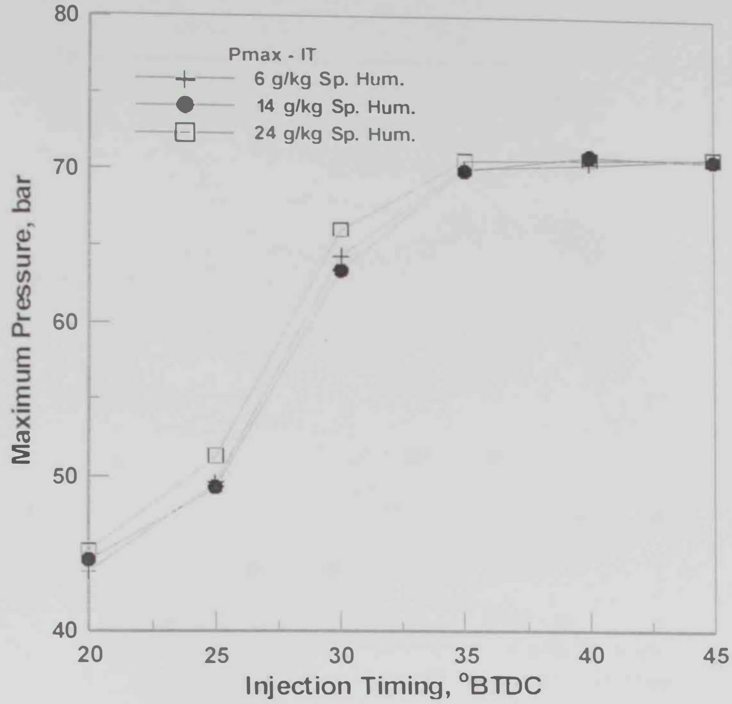


Figure 4.15 Effect of water addition and injection timing on maximum combustion pressure

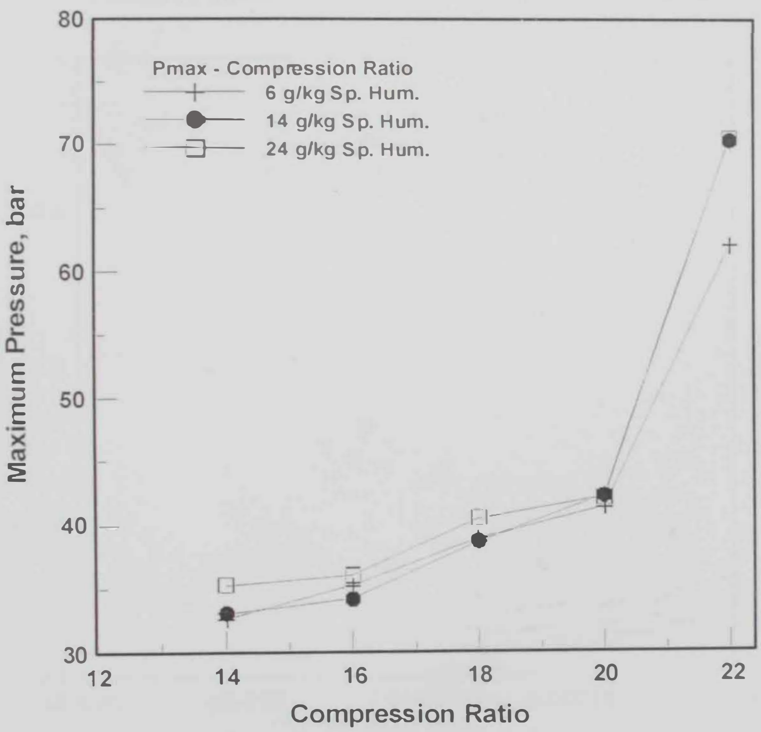


Figure 4.16 Effect of water addition and compression ratio on maximum combustion pressure

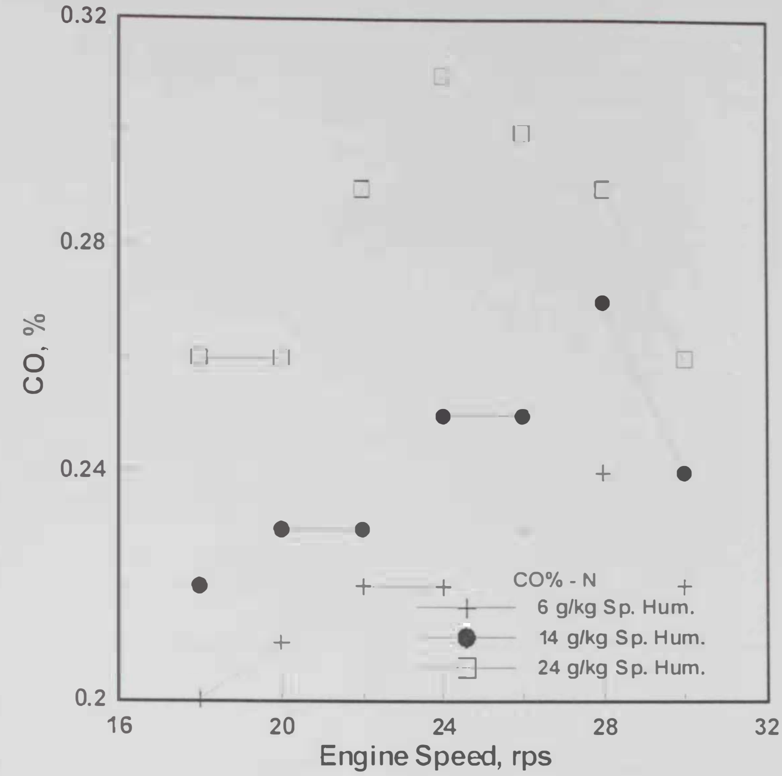


Figure 4.17 Effect of water addition and engine speed on CO emission

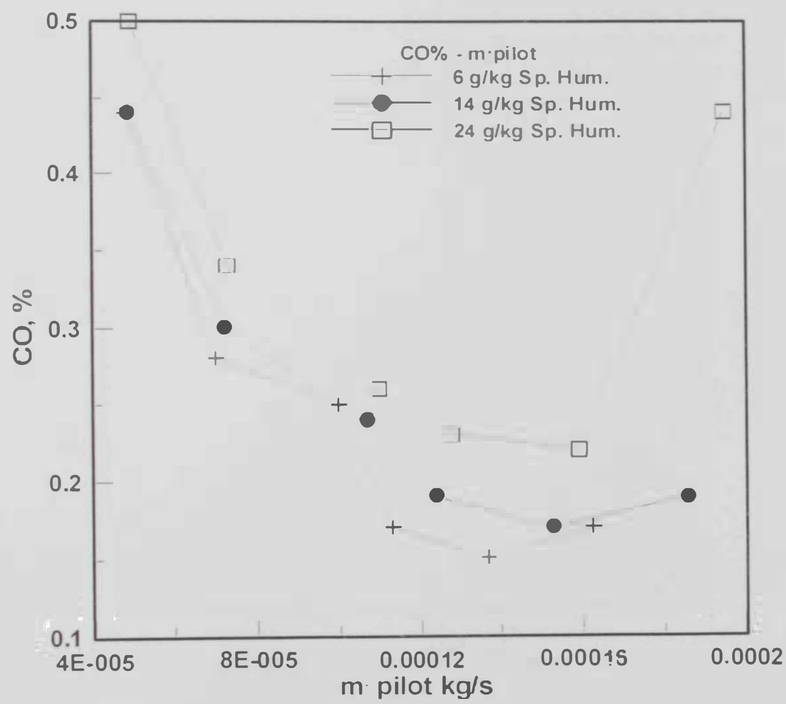


Figure 4.18 Effect of water addition and mass of pilot fuel on CO emission

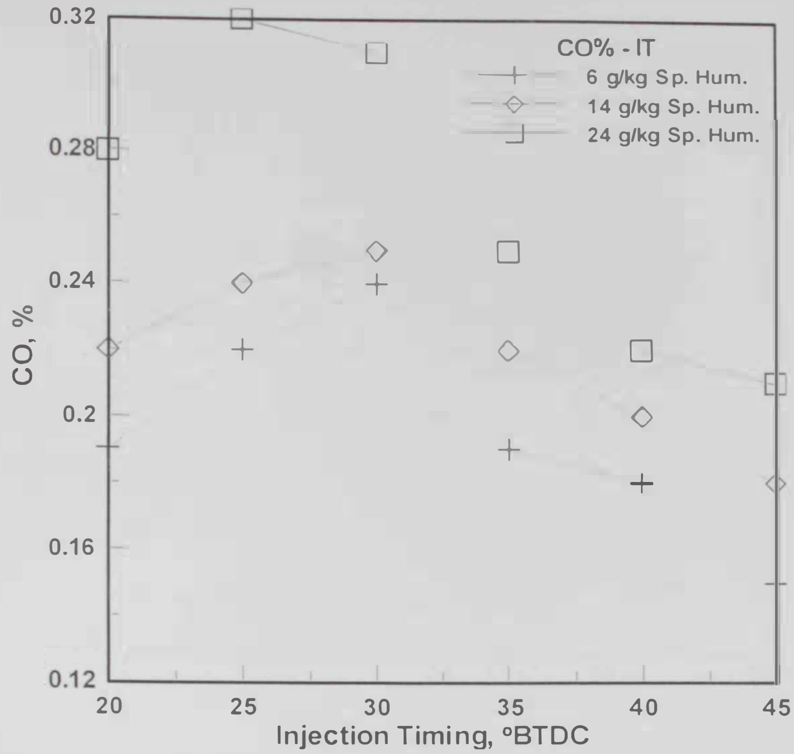


Figure 4.19 Effect of water addition and injection timing on CO emission

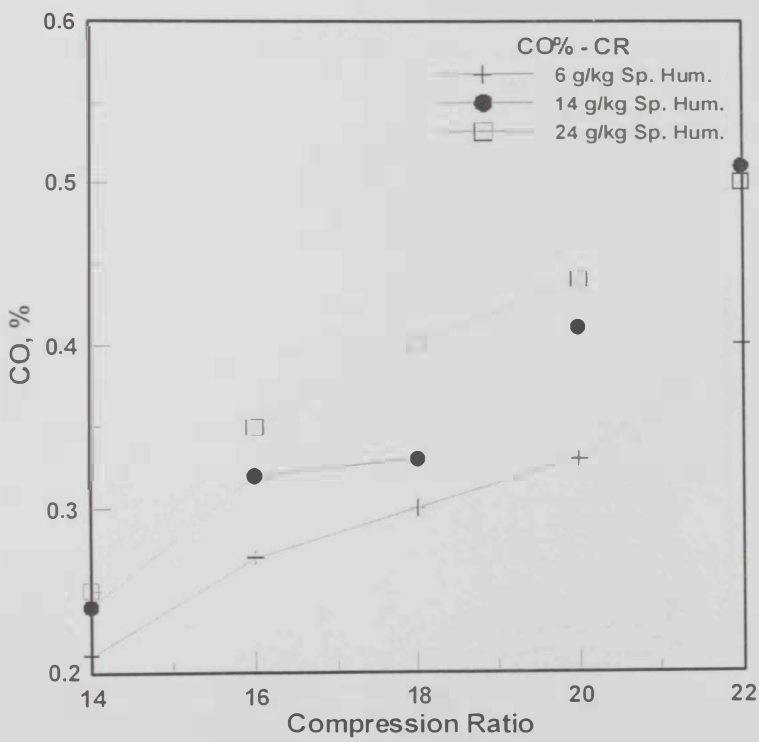


Figure 4.20 Effect of water addition and compression ratio on CO emission

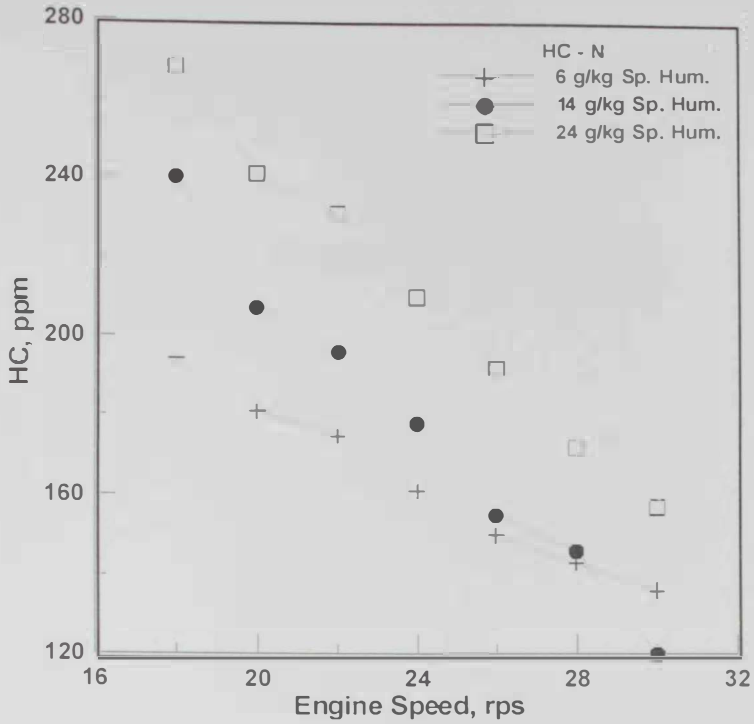


Figure 4.21. Effect of water addition and engine speed on HC emission

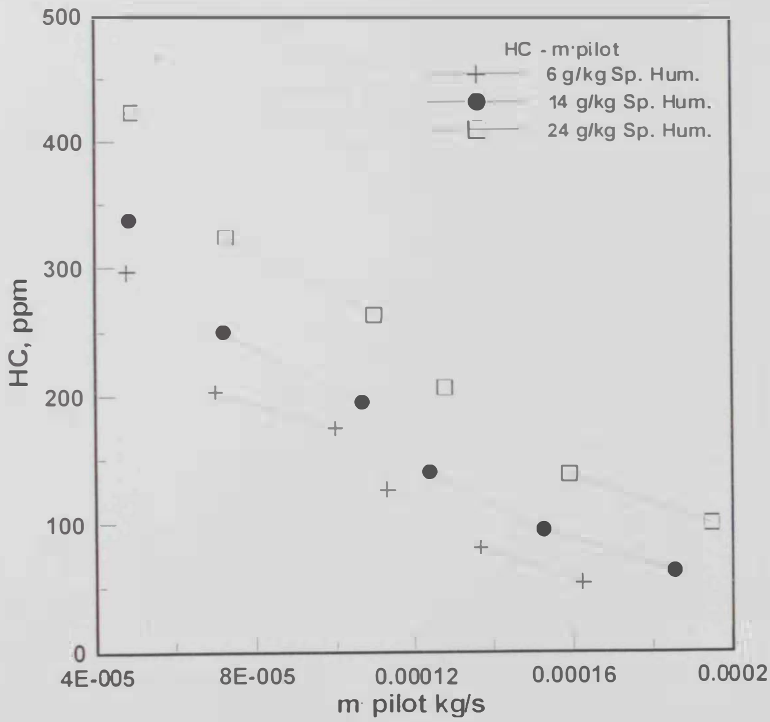


Figure 4.22 Effect of water addition and mass off pilot fuel on HC emission

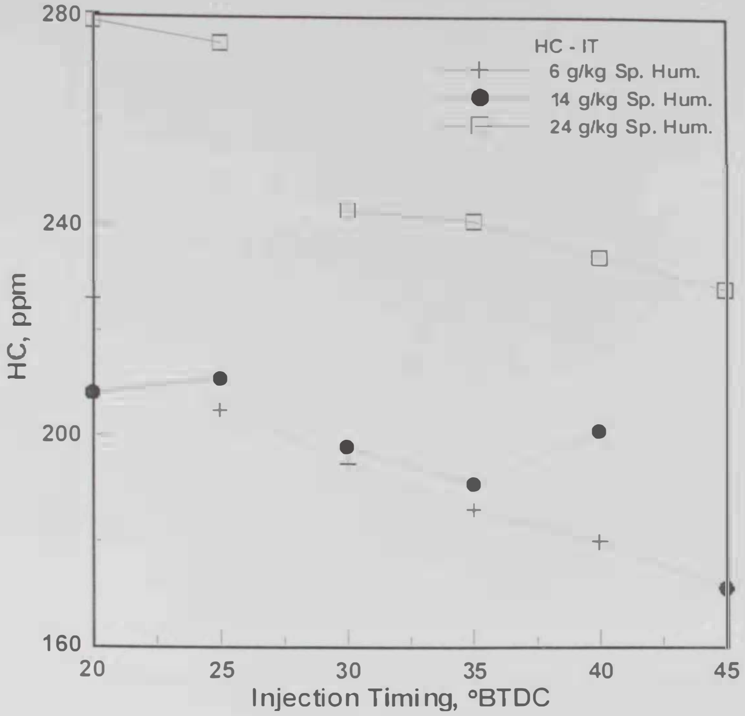


Figure 4.23. Effect of water addition and injection timing on HC emission

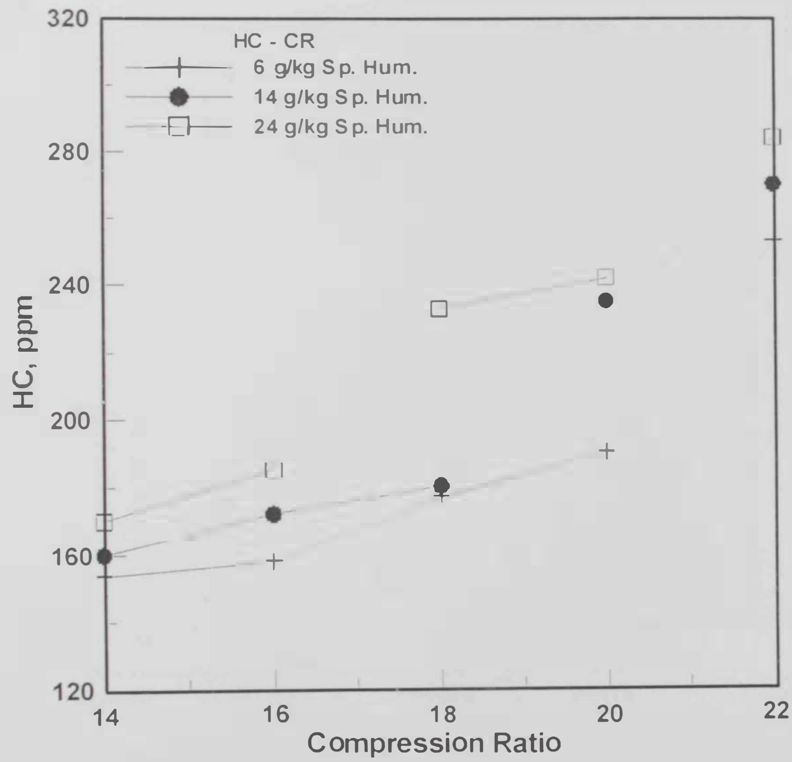


Figure 4.24. Effect of water addition and compression ratio on HC emission

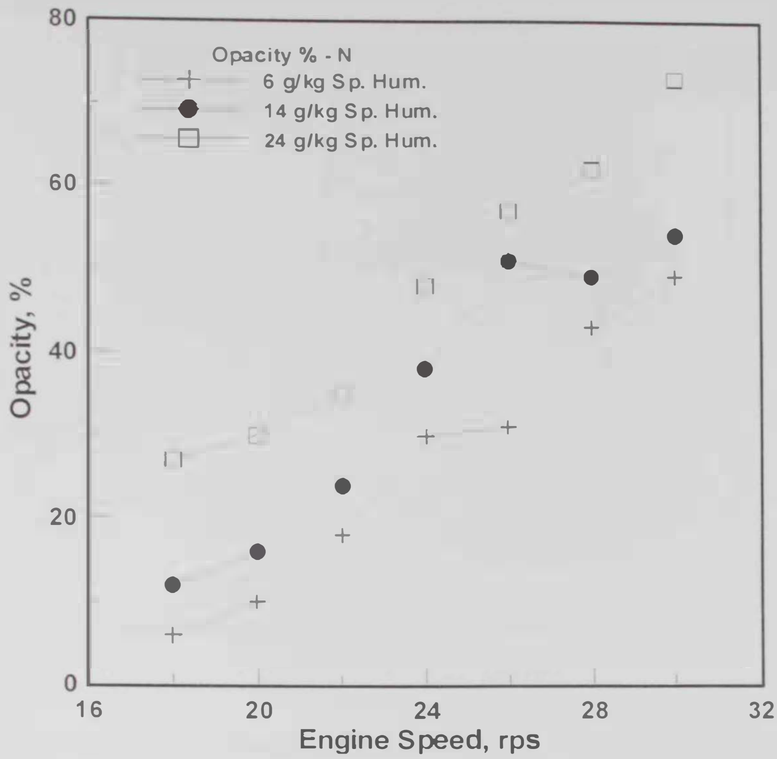


Figure 4.25 Effect of water addition and engine speed on opacity

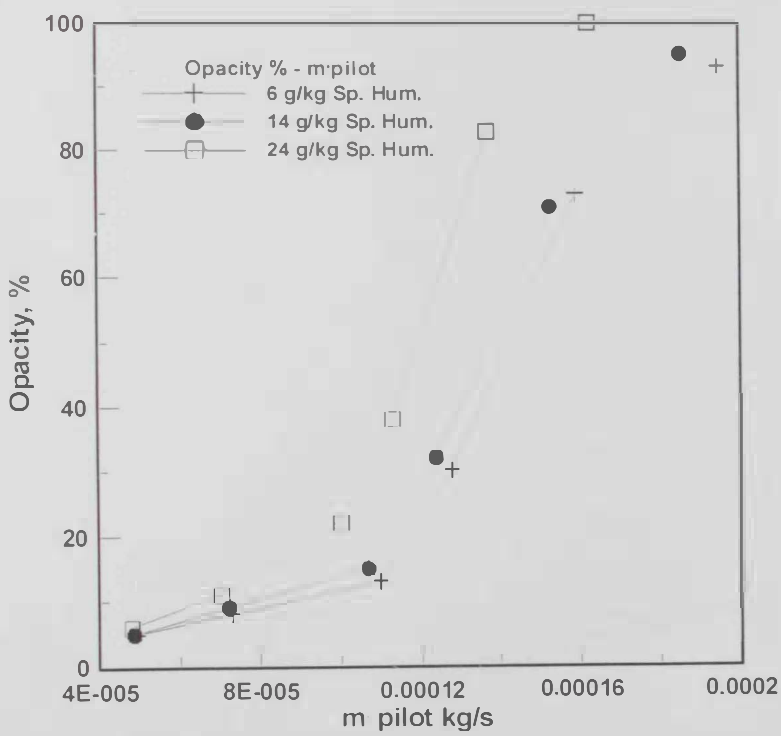


Figure 4.26 Effect of water addition and mass off pilot fuel on opacity

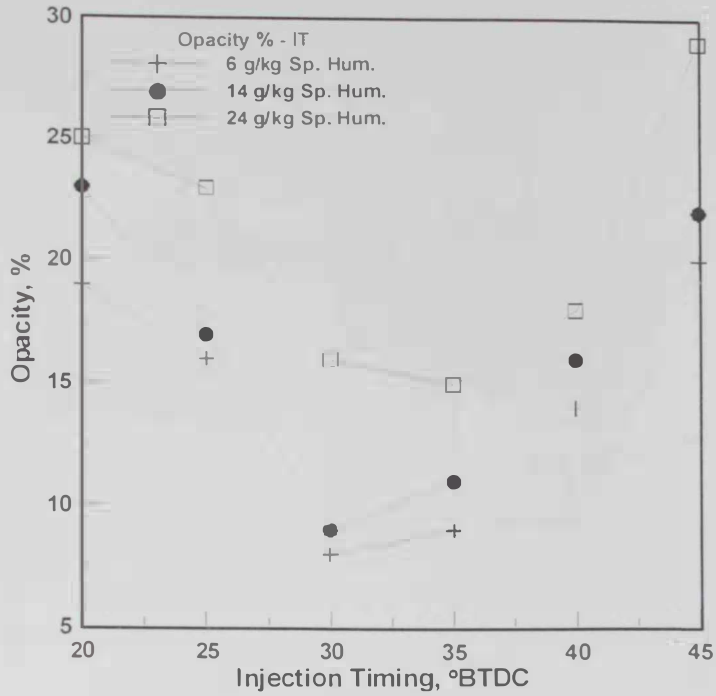


Figure 4.27 Effect of water addition and injection timing on opacity

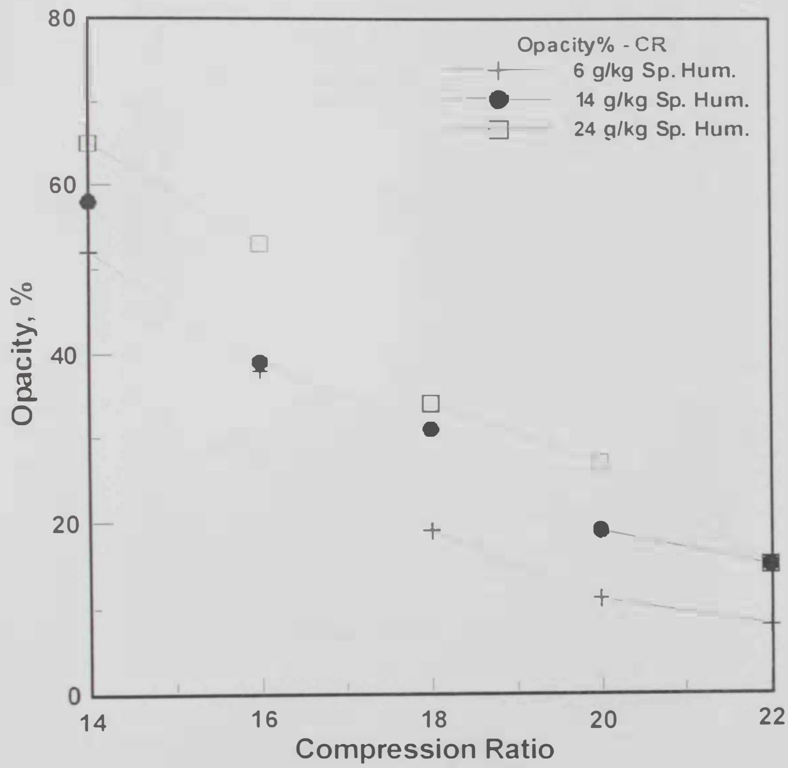
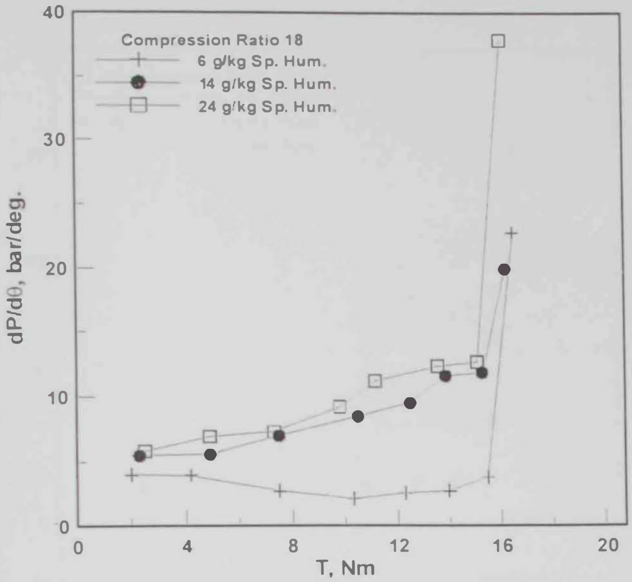
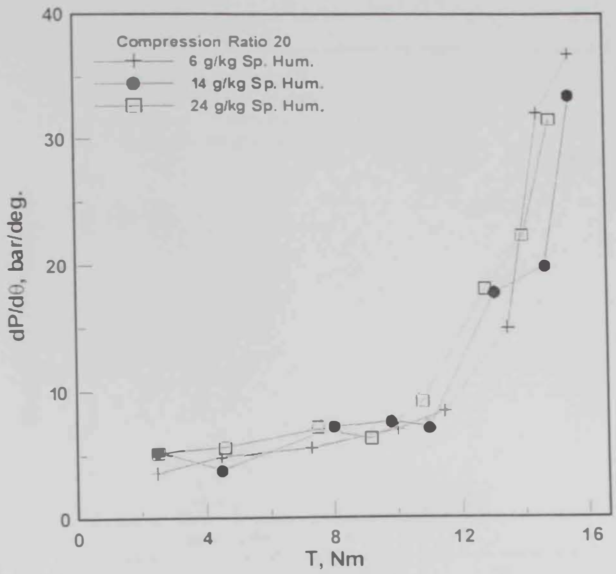


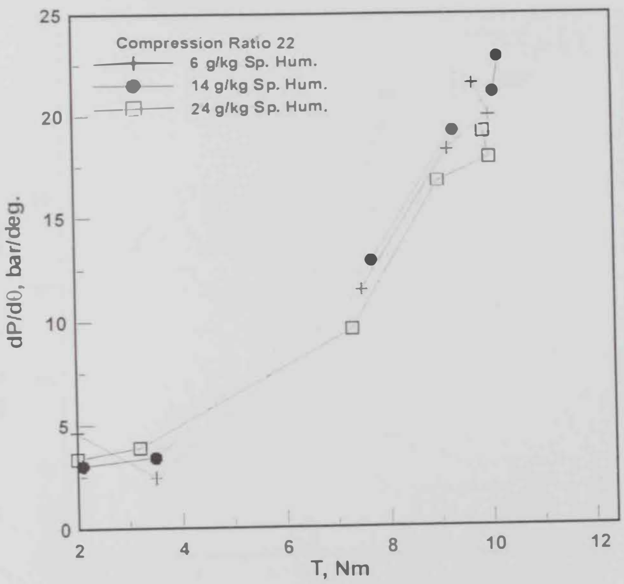
Figure 4.28 Effect of water addition and compression ratio on opacity



(a) Compression ratio = 18

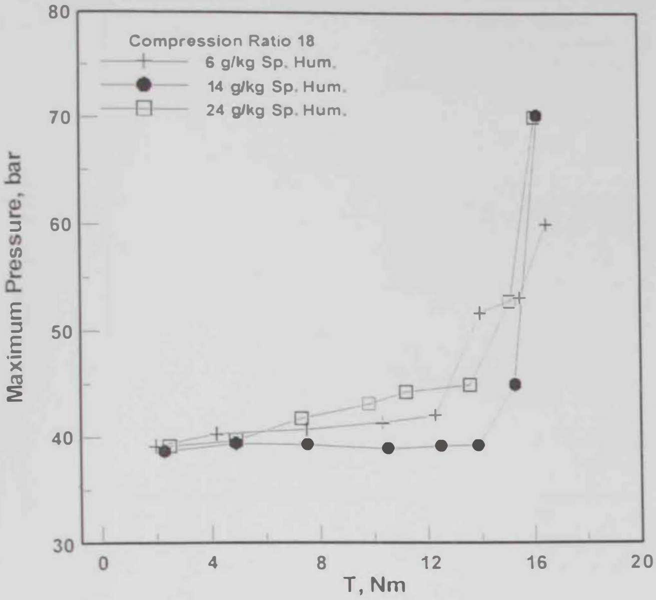


(b) Compression ratio = 20

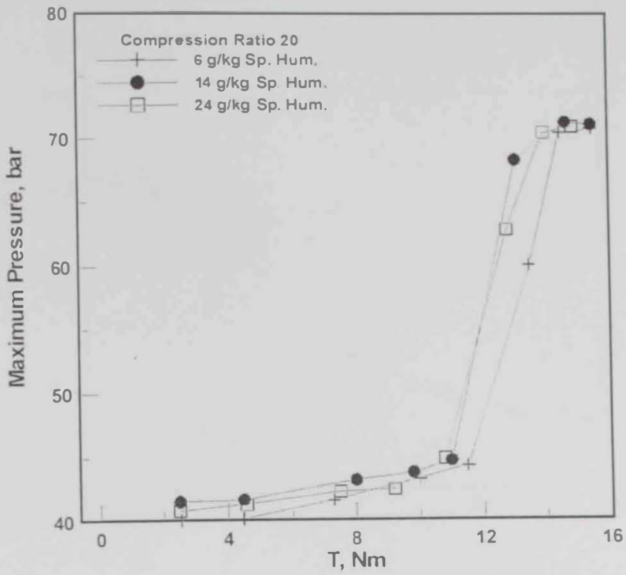


(c) Compression ratio = 22

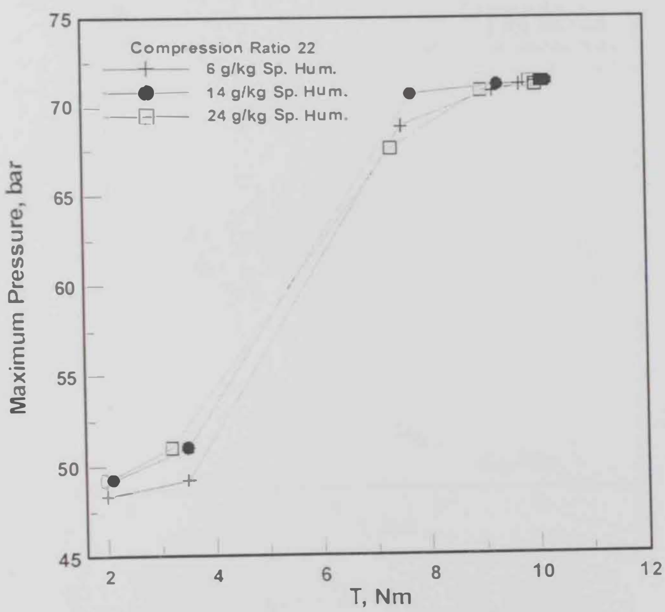
Figure 4.29: Effect of water addition and torque on maximum pressure rise rate for different compression ratios. (a) *CR=18; (b) CR=20, and (c) CR=22*



(a) *CR=18*

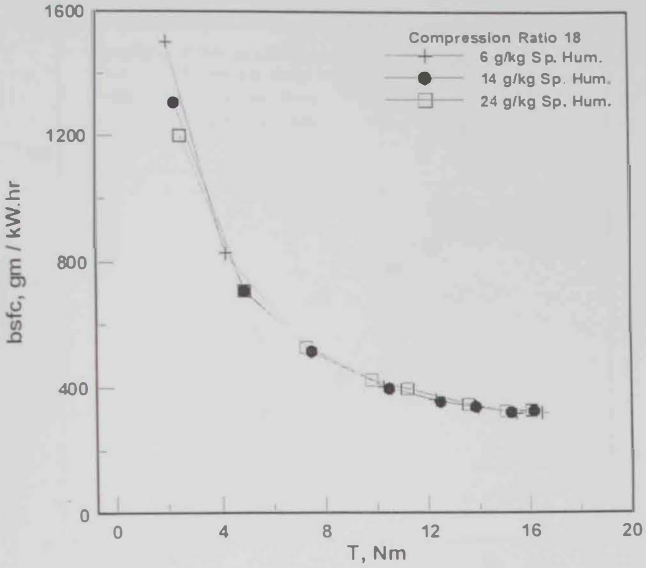


(b) *CR=20*

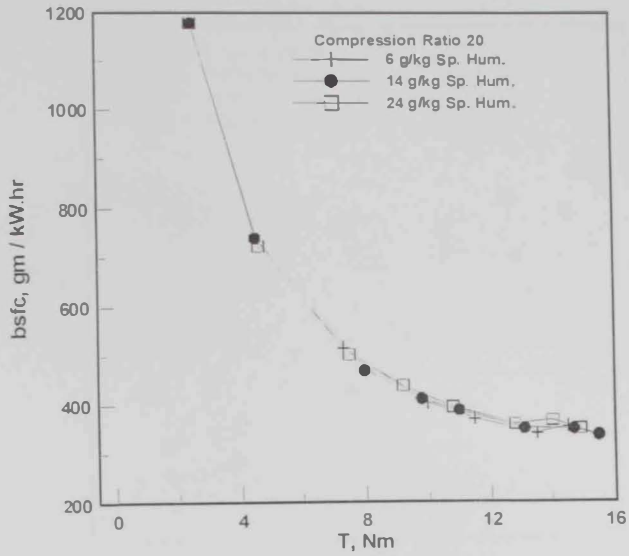


(c) *CR=22*

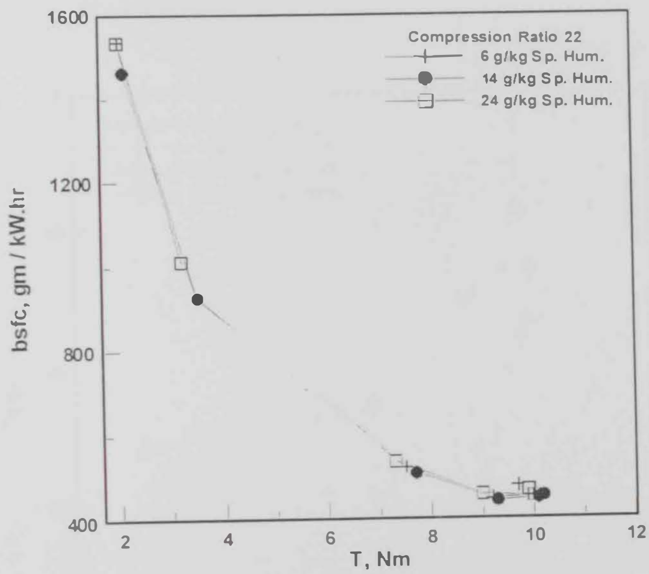
Figure 4.30 Effect of water addition and torque on maximum pressure for different compression ratios. (a) *CR=18; (b) CR=20, and (c) CR=22*



(a) *CR=18*

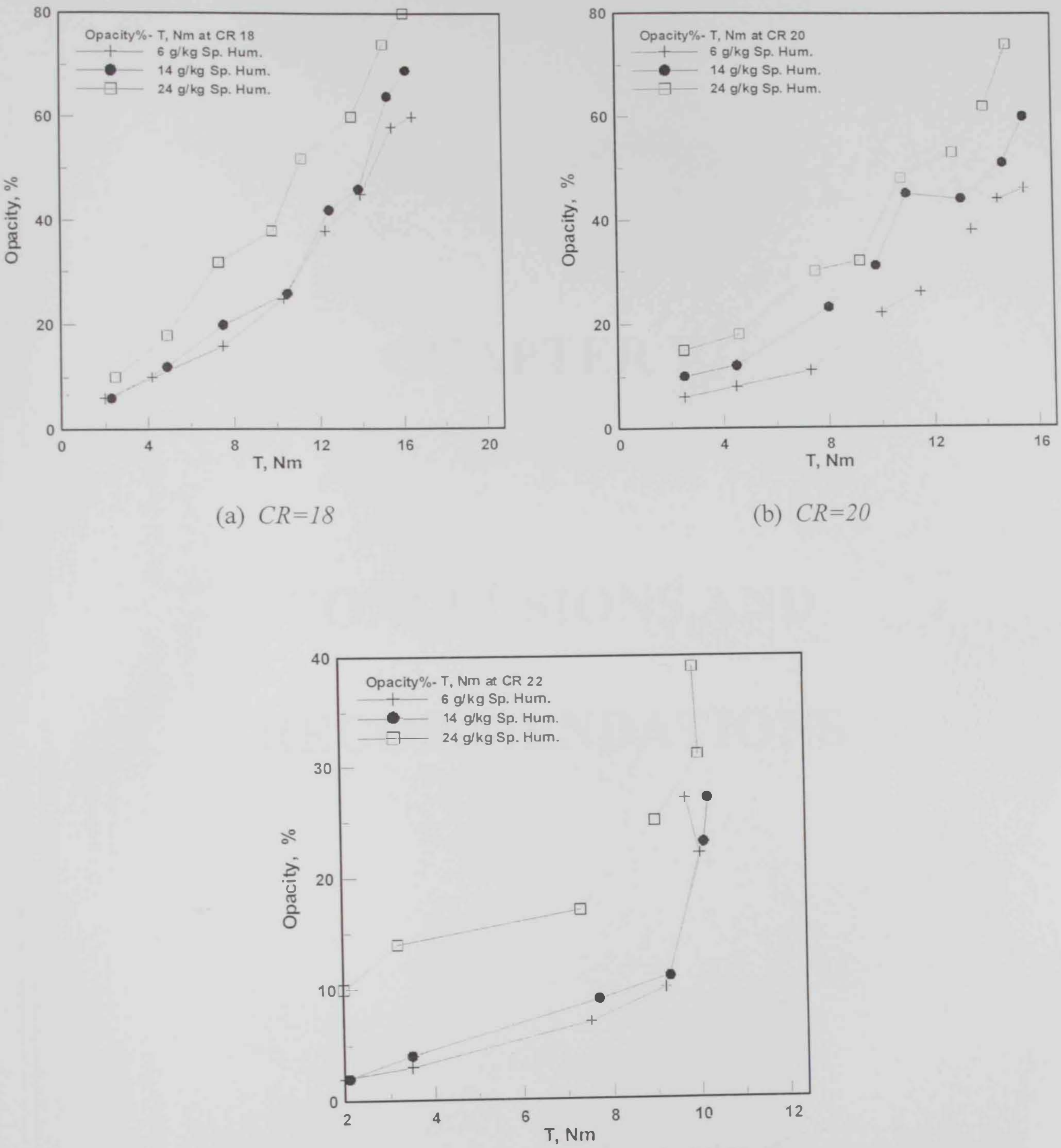


(b) *CR=20*



(c) *CR=22*

Figure 4.31 Effect of water addition and torque on brake specific fuel consumption for different compression ratios. (a) *CR=18; (b) CR=20, and (c) CR=22*



(a) *CR=18* (b) *CR=20*

(c) *CR=22*

Figure 4.32 Effect of water addition and torque on opacity for different compression ratios. (a) *CR=18; (b) CR=20, and (c) CR=22*

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

5.1 Introduction

A discussion and results have been explored in the previous chapter about the results taken from the dual fuel engine running on diesel and LPG, and the effects of water addition and some design and operating parameters on the engine performance and emissions are presented. The studied parameters are the engine speed, the engine load, the pilot fuel mass injected the injection timing of the pilot fuel and the compression ratio. From the experimental results discussed in the previous chapter, the following conclusions may be drawn.

5.2 Effect of water addition on brake power output

1. Increasing the engine speed generally increases the brake power output.
2. Increasing the mass of pilot fuel generally increases the brake power output of the engine for all ratios of water level.
3. Increasing the injection timing advance from 20 to 45 degree before top dead center generally reduces the brake power output.
4. The increase in the compression ratio from 14 up to 22 generally increases the brake power output of the engine.
5. Increasing the water addition level from 6 g/kg.air to 14g/kg then to 24 g/kg.air what generally reduce the brake power output of the engine, however this reduction in the brake power output is not much compared to the benefit of using the water in the engine as the NO_x is proved to reduce.
6. The use of slightly lower compression ratios e.g. 20 would produce higher brake power output and even in the case of using high amount of water in the admitted air (24 g/kg.air) the engine would give the same power as of compression ratio of 22.

5.3 Effect of water addition on brake specific fuel consumption

1. Increasing the engine speed generally increases the brake specific fuel consumption (bsfc) for all water addition ratios.
2. The increase in the mass pilot fuel generally decreases the brake specific fuel consumption.
3. Adding more water will increase the specific fuel consumption at all masses of pilot fuel.
4. Advancing the injection timing form 20 to 45 degrees before the top dead center generally increases the brake specific fuel consumption.
5. Increasing the compression ratio from 14 to 16. 18, 20 to 22 generally decreases the brake specific fuel consumption.

5.4 Effect of water addition on maximum pressure rise rate

1. The increase in the engine speed from 16 to 32 revolutions per second generally decreases the maximum pressure rise rate $\left(\frac{dP}{d\theta}\right)_{max}$. or the combustion noise.
2. Increasing the mass of pilot fuel increases maximum pressure rise rate.
3. Advancing the injection timing leads to higher pressure rise rate.
4. Increasing the compression ratio from 14 to 22 generally increases the maximum pressure rise rate or combustion noise.

5.5 Effect of water addition on maximum combustion pressure

1. Increasing the engine speed from 18 to 30 revolutions per second generally decreases the maximum combustion pressure for the same amount of water exists in the inlet air.
2. Increasing the mass of pilot fuel increases the maximum pressure of the cycle.
3. Advancing the injection of the pilot diesel fuel generally increases the maximum pressure of the cycle.

4. Increasing the compression ratio from 14 to 22 generally increases the maximum pressure of the cycle.

5.6 Effect of water addition on CO emissions

1. Increasing the engine speed increases the CO emission up to a speed of 28 revolutions per second. then starts to fall down and this occurs at all water addition levels.
2. Increasing the mass of pilot fuel generally decreases the CO emissions.
3. The C O emissions in the exhaust gasses is maximum at injection timing of 25-30 degrees before top dead centre (TDC), and it is minimum at very late or very early injection timing.
4. Increasing the compression ratio from 14 to 22 increases generally the CO emission in the exhaust gasses , However the CO emission level is far below the allowable limits for the current exhaust gasses limits.

5.7 Effect of water addition on HC emissions

1. Increasing the engine speed generally decreases the HC emission for all water addition level tested.
2. Increasing the mass of pilot fuel generally decreases the HC emission in the exhaust gasses.
3. Increasing the injection timing advance from 20 to 45 degrees before the top dead centre generally decreases the HC emission in the exhaust gasses for the three tested water addition level.
4. HC emission becomes higher and higher with increasing the compression ratio of the engine.

5.8 Effect of water addition on Smoke emissions

1. Increasing the engine speed generally increases the opacity level of the exhaust gases for the three water addition levels tested.
2. Increasing the mass of pilot fuel injected increases generally the opacity level of the exhaust gasses.
3. For injection timing of around 30° BTDC the opacity level is minimum and increases both sides for the late and very early injection.
4. Increasing the compression ratio of the engine generally decreases the opacity level of the exhaust gasses or the smoke level.

5.9 Knocking limits of dual fuel engine

1. The increase of the maximum pressure rise rate or the combustion noise when the load output is increased.
2. The decrease in the compression ratio from 22 to 20 and then to 18 enabled the engine to run at higher torque output without heavy knocking or ignition failure.
3. The increase in the maximum pressure with increasing the torque.
4. Decreasing the compression ratio from 22 to 20 then to 18 has resulted in an increase in the knocking limit or the maximum torque that the engine can produce without knocking.
5. Increasing the output torque results in a decrease in the *bsfc*.
6. At all compression ratios, the increase in the output torque has resulted in the increase in the smoke level or opacity level of the exhaust gases.

5.10 Recommendations for future work

Based on the results of the current research the following recommendtions are suggested.

1. The current research project deals with the experimental data collected fir the diesel engine running on dual fuel of diesel liquid fuel and LPG gaseous fuel as the main fuel. it is recommended to repeat the same experiment using a

Computational Fluid Dynamics (CFD) calculations as the software exists at the UAE University (KIVA3-verrel3).

2. The water addition proved to be effective in changing the performance, noise emissions and NO_x emissions. The proposed project deals with testing the use of the CFD code KIVA3-verrel3. It would be relatively easy to add some water H_2O at the beginning of the compression stroke with the existing fuel-air mixture.
3. The current research changed the water addition level from 6 g/kg air (atmospheric air) to 14 then to 24 g/kg air, however it would be advantageous to test higher amount of water injected in the inlet air using different methods of water addition.

References

References

[1] Korbitz, W. Biodiesel production in Europe and North America an encouraging prospect. *Renewable Energy*. Volume 16. 1999. pp 1078-1083.

[2] Hohmann, S., Renz, U. Numerical simulation of fuel sprays at high ambient pressure: the influence of real gas effects and gas solubility on droplet vaporisation. *International Journalof Heat and Mass Transfer*. 46, Issue 16, pp 3017-3028, July 2003.

[3] Mauderly, Joe, L. Diesel Emissions: Is more health research still needed? *Toxicological Sciences*. 62, pp6-9. 2001.

[4] Lin C.-Y. and Huang J.-C., The An oxygenating additive for improving the performance and emission characteristics of marine diesel engines. Ocean Engineering 30 (2003) 1699–1715, 981389.

[5] Cooper, D. C., Alley, F. C. Air pollution control: a design approach. Waveland Press, Prospect Heights, Illinois. 1994.

[6] Duran, A., Carmona, M., Ballesteros, R. Competitive diesel engine emissions of sulfur and nitrogen species. *Chemoshpere*. 2003.

[7] Duleep, K. G. Quantifying the emission benefits of opacity testing and repair of heavy-duty diesel vehicles. *Environmental Science and Technology*. 37, Issue 3, pp 630-637, 2003.

[8] Zvirin, Y., Gutman, M., Tartakovosky, L. Fuel effects on emissions. **In:** Sher, E. eds., Handbook of air pollution from internal combustion engines; pollutant formation and control. Academic press: San Diego. 1997. pp 575-603.

[9] Springer, G. S., Patterson, D. J. Engine emissions: pollutant formation and measurement. Plenum Press, New York. 1973.

[10] Cngur, Y., Altiparmak, D. Effect of fuel Cetane number and injection pressure on a DI Diesel engine performance and emissions. *Energy and Conservation Management*. 44, Issue 3, pp 389-397, 2003.

[11] U.S. EPA, Office of Air and Radiation. Highway Diesel Progress Review. EPA420-R-02-016, June 2002. Available at: www.epa.gov/air/caac/dieselreview.pdf.

[12] Pitts, J. An overview of diesel emissions on air quality. Plenum Press, New York. 1973.

[13] Perola, V. C, Zacarias. D., Pires, A. F., Pool, C. S., Carvlho, R. F. Measurements of polycyclic aromatic hydrocarbons in airborne particles from the metropolitan area of Sao Paulo City, Brazil. *Atmoshperic Environment*. 37, Issue 21, pp 3009-3018, July 2003.

[14] Lanni, T. Fine urban precursor emissions control for diesel urban transit buses. *Environmental Pollution*. Vol. 123, Issue 3, June 2003. pp 427-437.

[15] Baines, M. T., Somers, J. H., Hellman, K. H., Effects of fuel variables on diesel emissions. *Journal of the Air Pollution Control Association*. 32, Issue 8, pp 810-813, 1982.

[16] Turns, S.R., 1996. *An Introduction to Combustion: Concepts and Applications*. McGraw-Hill, Inc., New York, NY.

[17] Larsen, C., Oey, F., and Levendis, Y.A., 1996, "An Optimization Study of the Control of NO_x and Particulate Emissions from Diesel Engines," SAE Paper 960473.

[18] Kureti, S., Weisweiler, W., Hizbullah, K. Simultaneous conversion of nitrogen oxides and soot into nitrogen and carbon dioxide over iron containing oxide catalysts in diesel exhaust gas. *Applied Catalysis B: Environmental.* 43, Issue 3, July 2003. pp 281-291.

[19] Yanagihara, H., 1997, "Simultaneous Reduction of NO_x and Soot in Diesel Engines Using a New Mixture Preparation Method," *JSME International Journal*, Series B, Vol. 40, No. 4, pp. 592- 598.

[20] Nakatsuji, T., Yasukawa, K., Tabata, K., Ueda, K., and Niwa, M., 1998, "Catalytic Reduction System of NOx in Exhaust Gases from Diesel Engines with Secondary Fuel Injection," *Applied Catalysis B: Environmental*, Vol. 17, pp. 333-345.

[21] Urbach, H.B., Knauss, D.T., Emory, J., Wallace, B.L, Wasser, J.A., Sexton, M.R., and Frese, J., 1997, "The Reduction of NOx Emissions from Marine Power Plants," Paper No. 97-MP5.08, Proceedings of the Air & Waste Management Association's 90th Annual Meeting & Exhibition, June 8-13, 1997, Toronto, Ontario, Canada.

[22] Yoshihara, S., Okabe, M., and Nakamaura, T., 1996, "Water Injecting Type Diesel Engine," U.S. Patent 5,522,349, United States Patent Office.

[23] O'Neal, G.B., Storment, J.O., and Waytulonis, R.W., 1981, "Control of Diesel Exhaust Emissions in Underground Coal Mines—Single Cylinder Engine Experiments with Modified and Non- Conventional Fuels," SAE SP-495, Diesel Combustion and Emission Part 3, Int. Off-Highway Meeting & Exposition, Milwaukee, WI, pp. 13-23.

[24] Greeves, G., Khan, I.M., and Onion, G., 1976, "Effects of Water Introduction on Diesel Engine Combustion and Emissions," Proceedings, 16th Annual Symposium (Int.) on Combustion, Williams & Wilkins Co., pp. 321-336.

[25] De Vita, A., 1989, "Multi-Cylinder D.I. Diesel Engine Tests with Unstabilized Emulsion of Water and Ethanol in Diesel Fuel," SAE Paper 890450.

[26] Ulrich, A. and Kessler, A., 1992, "Method and Apparatus for Producing a Water-in-Fuel- Emulsion and Emulsifier-Free Water-in-Fuel-Emulsion," U.S. Patent 5,125,367, United States Patent and Trademark Office.

[27] Sawa, N., and Kajitani, S., 1992, "Physical Properties of Emulsion Fuels (Water/Oil-Type) and Its Effect on Engine Performance under Transient Operation," SAE Paper 920198, International Congress & Exposition, Detroit MI.

[28] Schwab, S.D., 1997, "Emulsion Diesel Fuel Composition with Reduced Emissions," U.S. Patent 5,669,938, United States Patent and Trademark Office.

[29] Peter-Hoblyn, J.D., and Valentine, J.M., 1996, "Reduction of Nitrogen Oxides Emissions from Vehicular Diesel Engines," U.S. Patent 5,584,894, United States Patent and Trademark Office.

[30] Peter-Hoblyn, J.D., Valentine, J.M., and Dubin, L., 1998, "Enhanced Lubricity Diesel Fuel Emulsions for Reduction of Nitrogen Oxides," U.S. Patent 5,743,922 United States Patent and Trademark Office.

[31] Marelli, E., 1995, "Diesel Fuel Emulsion," U.S. Patent 5,445,656, United States Patent and Trademark Office.

[32] Montagne, X., Herrier, D., and Guibet, J.-C., 1987, "Fouling of Automotive Diesel Injectors-Test Procedure, Influence of Composition of Diesel Oil and Additives," SAE Paper 872118.

[33] Crookes, R.J., Nazha, M.A.A., and Kiannajad, F., 1990, "A Comparison of Ignition and Emissions Characteristics for Alternative Diesel Fuels and Emulsions," *I. Mech. E. Seminar*, pp.47-52.

[34] Yoshimoto, Yasufumi, Tsukarhara, Minoru, Muryama, and Tadashi, 1989, "Studies on the Microexplosion of Emulsified Fuels," *Nippon Kikai Gakkai Ronbunshun B Hen*, Vol. 55, No. 519, pp. 3538-3543.

[35] Nazha M A A, Rajakaruna H and Wagstaff S A. The Use of Emulsion, Water Induction and EGR for Controlling Diesel Engine Emissions, 2001-01-1941.

[36] Abd Alla G.H., Soliman H.A., Badr A. and Abd Rabbo M.F.. The Effect of pilot fuel quantity on the performance of a dual fuel engine. Energy Conversion & Management 41 (2000) 559-572.

[37] Brusca S. and Lanzafame R. The Evaluation of the Effects of Water Injection in a Single Cylinder CFR Cetane Engine, 2001-01-2012.

[38] Kent Nord, Dan Haupt and Karl-Erik Egebäck. The Particles and Emissions from a Diesel Engine Equipped with a Humid Air Motor System, 2001-01-3616.

[39] Psota M. A., Easley W. L., Fort T. H., and Mellor A. M.. The Water Injection Effects on NOx Emissions for Engines Utilizing Diffusion Flame Combustion, SAE 971657.

[40] Masahiro Ishida, Hironobu Ueki, and Daisaku Sakaguchi. The Prediction of NOx Reduction Rate Due to Port Water Injection in a DI Diesel Engine, SAE 972961.

[41] R.G. Papagiannakis, D.T. Hountalas. The Experimental investigation concerning the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine, Applied Thermal Engineering 23 (2003) 353-365.

[42] Magnus Christensen and Bengt Johansson. The Homogeneous Charge Compression Ignition with Water Injection, 1999-01-0182.

[43] J. P. Mellol and A. M. Mellor. The NOx Emissions from Direct Injection Diesel Engines with Water/Steam Dilution, 1999-01-0836.

[44] Canfield, C.A., Babyak, R.A., and Wander, J.D., 1997, "Demonstration of a Filter Cart for NOx Removal from Ground Support Equipment," Technical Report No. AL/EQ-TP-1997-0001, U.S. Air Force Research Laboratory, Tyndall AFB, FL.

[45] Akridge, R.J., Donegan, B.M., Rolader, G.E., Bienvenue, D.R., Verge, R.E., and Elliot,D., 1997, "Reduction of Nitrogen Oxides Emitted by Aerospace Ground quipment at March ARB," Paper No. 97-TA36A.01, Proceedings of the Air & Waste Management Association's 90th Annual Meeting & Exhibition, June 8-13, 1997, Toronto, Ontario, Canada.

[46] T. PRIEDE 1980 SAE 800534, 2039-2069. In search of origins of engine noise - an historical review.

[47] SCHABERG P. W., PRIEDE T. and DUTKIEWICZ R. K. 1990 SAE 900013, 1-13. Effects of rapid pressure rise on engine vibration and noise.

[48] PRIEDE T. 1979 SAE 790205, 783-797. Problems and developments in automotive engine noise research.

[49] G Galinsky, Reader G T., Potter I J., and Gustafson R W. The EFFECT OF VARIOUS WORKING FLUID COMPOSITIONS ON COMBUSTION NOISE IN DIESEL ENGINES, AIAA-94-3996-CP.

[50] Abd Alla G.H. Soliman H.A.. Badr O.A, Abd Rabbo M.F.. The Effect of injection timing on the performance of a dual fuel engine, Energy Conversion and

Management 43 (2002) 269-277.

[51] Mohamed Y.E. Selim, Thermal Loading and Temperature Distribution of a Pre Combustion Chamber Diesel Engine Running on Gasoil/Natural Gas, ICE Division ASME International Spring Conference, April 1998.

[52] Mohamed Y.E. Selim. The Sensitivity of dual fuel engine combustion and knocking limits to gaseous fuel composition, Article in Press, Energy Conversion and Management, Volume 45, issue 3, pp. 411-425 (2004).

[53] Variable Compression Engine Ricardo E6, Ricardo Consulting plc., Operating Manual, 1985.

Appendix

Appendix

Data acquisition program (in Mu-Mac-Basic)

```
REAL: I,J,TA,TF,OIL,TG,TCN,TCT,SPD,TRQ,SPDF,TRQF,MF
REAL: TAV,TFV,OILV,TGV,TCNV,TCTV,SPDV,TRQV,SPDFV,TRQFV,MFV
REAL: FA,AMA,PHI,DENSITY,VD,PDA,TCF,FLAG,COUNT
REAL: CO,CO2,O2,UHC,NOX
REAL: COV,CO2V,O2V,UHCV,NOXV
REAL: SCO,SO2,SUHC,SNOX,MAX1
INTEGER: FILE1,FILE2,FILE3
STRING: A$[16],B$[16],S$[16]
10 REM ************************************************************
20 REM **  THIS PROGRAM (Written In UMACBASIC) COLLECTS MEASURED  **
30 REM **  DATA FROM THE DATA ACQUISITION SYSTEM AND SAVES THEM  **
40 REM **  IN DATA FILES.                                **
50 REM **                                                **
60 REM **%%%  MODEFIED & IMPROVED By.....ALAA ISKANDARANI   %%%**
70 REM **                ON 14/12/1994                  **
80 REM **                B.Sc. Project                  **
90 REM **                                                **
100 REM **  Program is available in the WOS directory under the  **
110 REM **               name ....DIESEL                  **
120 REM ************************************************************
130 REM
140 REM * Open Data Files For Output Data. *
150 REM ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
160 PRINT #WOS CHR$(163);
170 FILE1 = 1: FILE2 = 2: FILE3 = 3
180 OPEN("O","DFN3",FILE1)
190 OPEN("O","DTN3",FILE2)
200 OPEN("O","DPN3",FILE3)
210 REM
220 REM * Configurate The Input Channels. *
230 REM ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
 240 INTYPE(12,41,0.1)
 250 INTYPE(13,41,0.1)
 260 INTYPE(14,41,0.1)
 270 INTYPE(15,41,0.1)
 280 INTYPE(16,41,0.1)
 290 INTYPE(17,41,0.1)
 300 INTYPE(36,2,0.1)
 310 INTYPE(37,2,0.1)
 320 INTYPE(38,2,0.1)
 340 INTYPE(40,2,0.1)
 350 INTYPE(41,2,0.1)
 360 INTYPE(42,2,0.1)
 370 INTYPE(43,2,0.1)
 380 INTYPE(44,2,0.1)
390 REM
400 REM * Engine Loading Test. *
410 REM ~~~~~~~~~~~~~~~~~~~~
```

```
420 INPUT "IS THE ENGINE READY FOR LOADING TEST (Y/N) OR (C) FOR GAS ANA. SCALES
"; B$
421 REM THE ORIGINAL GAS ANALYSER SCALES OF O2,CO,UHC,NOX GASES ARE
25,0.5,4000,4000 RESPECTIVELY.
422 IF B$ <> "C" AND B$ <> "c" THEN SO2 = 25: SCO = .5: SUHC = 4000: SNOX = 4000
425 IF B$ ="C" OR B$="c" THEN B$="y":INPUT"INSERT THE SALES OF CO,O2,UHC,NOX " SCO,
SO2, SUHC, SNOX
430 IF B$ <> "Y" AND B$ <> "y" THEN GOTO 870
435 PRINT
437 MAX1 = 100
438 FLAG = 0
439 COUNT = 0
440 PRINT "  TA  | TG  |  SPD  | TRQ  |  MF  | AMA  | PHI |  O2  | CO  | CO2 | UHC | NOx"
450 PRINT "  C   |  C  |  rpm  |  Nm  | Kg/hr| Kg/hr|     |  %   |  %  |  %  | PPM | PPM"
460 PRINT " ---- |-----|-------|------|------|------|-----|------|-----|-----|-----|-----"
470 FOR I = 1 TO MAX1
480   REM * Averaging The Input Data 5 Times. *
490   REM ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
500   TA = 0: OIL = 0: TG = 0: TCN = 0: TCT = 0: TRQ = 0: SPD = 0
510   TF = 0: MF = 0: CO = 0: CO2 = 0: O2 = 0: UHC = 0: NOX = 0
520   FOR J = 1 TO 5
530     REM * Collecting Measured Data And Converting Them To S.I. Units *
540     REM * ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
550     TA = AIN(12) + TA:   REM INDUCED AIR TEMPERATURE
560     TF = AIN(13) + TF:   REM LIQUID FUEL TEMPERATURE
570     OIL = AIN(14) + OIL: REM OIL TEMPERATURE
580     TG = AIN(15) + TG:   REM EXHAUST GASES TEMPERATURE
590     TCN = AIN(16) + TCN: REM TEMP. OF COOLING WATER IN
600     TCT = AIN(17) + TCT: REM TEMP. OF COOLING WATER OUT
610     TRQ = (((AIN(36) * 100) * 5.0795) - .8139) + TRQ
620     SPD = (((AIN(37) * 100) * 723.32) + 49.31) + SPD
630     MF = (((AIN(38) * 100) * 7.536) - .0912) + MF
650     CO = (((AIN(40) * 200) * SCO) / 10) + CO
660     CO2 = (((AIN(41) * 200) * 20) / 10) + CO2
670     O2 = (((AIN(42) * 200) * SO2) / 10) + O2
680     UHC = (((AIN(43) * 200) * SUHC) / 10) + UHC
690     NOX = (((AIN(44) * 200) * SNOX) / 10) + NOX
700   NEXT
710   TAV = TA / 5: OILV = OIL / 5: TGV = TG / 5: TCNV = TCN / 5: TFV = TF / 5
720   TCTV = TCT / 5: TRQV = TRQ / 5: SPDV = SPD / 5: MFV = MF / 5
730   COV = CO / 5: CO2V = CO2 / 5: O2V = O2 / 5: UHCV = UHC / 5: NOXV = NOX / 5
731   REM FLAG JUST TELL US WHEN SOMETHING WRONG WITH GAS ANALYSER SCALES
732   REM THE GIVE MASSAGE IF THE SCALE IS NOT RIGHT
733   REM """"""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""""
735   IF (COV > SCO) OR (O2V > SO2) THEN FLAG = FLAG + 1: PRINT "** BELOW VALUES
BEYOND SCALES OF CO OR/AND O2, FLAG= "; FLAG
737   IF (NOXV > SNOX) OR (UHCV > SUHC) THEN FLAG = FLAG + 1: PRINT "** BELOW
VALUES BEYOND SCALES OF NOX OR/AND UHC, FLAG= "; FLAG
740   PDA = -5E-09 * SPDV ^ 3 + .00003 * SPDV ^ 2 - .0287 * SPDV + 32.856
745   TCF = 1.055203 - .002882 * TAV + 6.1316E-06 * TAV ^ 2 + 1.1495E-09 * TAV ^ 3
747   VD = .000119 * PDA * TCF
748   DENSITY = 100 / (.287 * (TAV + 273))
750   AMA = DENSITY * VD * 3600
755   FA = MFV / AMA
757   PHI = FA / .069
```

```
760  PRINT USING "##.# ###.# ####  ##.#  #.##  ##.#  #.##  ##.## #.## ##.# #### ####"; TAV; TGV;
SPDV; TRQV; MFV; AMA; PHI;O2V; COV; CO2V; UHCV; NOXV
761  REM print "O2 = ";O2V
762  REM TO GROUP THE DATA POINTS
764  COUNT = COUNT + 1
766  IF COUNT = 5 THEN COUNT = 0: PRINT "---------------------------------------------------------------------------
----------"
770  REM
780  REM * Saving The Collected Data (Loading) In Their Data Files *
790  REM * ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
800  PRINT #FILE2, USING "####.#"; SPDV; TAV; OILV; TGV; TCNV; TCTV
810  PRINT #FILE3, USING "####.###"; SPDV; TRQV; MFV; AMA; COV; CO2V; O2V; UHCV;
NOXV; PHI
812  S$ = INPUT$(1)
813  IF S$ = "F" OR S$ = "f" THEN FLAG = 0
814  IF S$ = "Q" OR S$ = "q" THEN I = MAXI
816  IF S$ = "R" OR S$ = "r" THEN SO2 = 25: SCO = .5: SUHC = 4000: SNOX = 4000
818  IF S$ = "C" OR S$ = "c" THEN INPUT "INSERT THE SCALES OF CO,O2,UHC,NOX IN ORDER
" SCO, SO2, SUHC, SNOX
830 NEXT
840 PRINT:PRINT:PRINT #WOS CHR$(163);
850 REM * FRICTION TEST (MOTORING).*
860 REM  ~~~~~~~~~~~~~~~~~~~~~~~~~~~
870 INPUT "IS THE ENGINE READY FOR FRICTION TEST (Y/N)"; A$
880 IF A$ <> "Y" AND A$ <> "y" THEN GOTO 1090
890 PRINT : PRINT
900 PRINT "    SPEED        TORQUE"
910 PRINT "   -------        ------"
920 FOR I = 1 TO MAXI
930  REM * Averaging The Input Data 5 Times. *
940  REM  ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
950  TRQF = 0: SPDF = 0
960  FOR J = 1 TO 10
970   REM * Collecting Measured Data And Converting Them To S.I. Units *
980   REM  ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
990    TRQF = (ABS(((AIN(36) * 100) * 5.0795) - .8139)) + TRQF
1000    SPDF = (ABS(((AIN(37) * 100) * 723.32) + 49.31)) + SPDF
1010  NEXT
1020  SPDFV = SPDF / 10: TRQFV = TRQF / 10
1030  PRINT USING "   ####.#        ##.#"; SPDFV; TRQFV
1040  REM * Saving The Collected Data (Motoring) in Its Data File *
1050  REM  ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
1060  PRINT #FILE1, USING "####.#"; SPDFV; TRQFV
1070  S$ = INPUT$(1)
1075  IF S$ = "Q" OR S$ = "q" THEN I = MAXI
1080 NEXT
1090 REM * Close The Data Files. *
1100 REM  ~~~~~~~~~~~~~~~~~~~~~~
1110 CLOSE (FILE1)
1120 CLOSE (FILE2)
1130 CLOSE (FILE3)
ENDFILE
```

تظهر النتائج المعملية الى نقص بسيط فى قدرة المحرك عند استعمال الماء علما بأن هذا النقص يمكن تعويضه بالتحكم فى عوامل تصميم المحرك الأخرى. وأدى أيضا إضافة الماء إلى المحرك إلى زيادة ضوضاء الاحتراق علما بأنه يمكن تقليلها بالتحكم فى عوامل التصميم والتشغيل الأخرى مثل زيادة سرعة المحرك, وتقليل كمية وقود الديزل المستعمل وتأخير توقيت حقن الوقود وتقليل نسبة انضغاط المحرك, وعند زيادة كمية الماء المستعملة زادت نسبة غازات أول أكسيد الكربون وغازات الهيروكربونات بدرجة بسيطة علما بأنه يمكن تقليلها باستعمال محول حفاز. كما زادت نسب عتامة غازات العادم مما يدل على زيادة نسبة الدخان الأسود ولكن أمكن تقليل هذا الدخان بتقليل كمية وقود الديزل المستعملة فى احتراق الوقود الغازى.

وفى نهاية الرسالة تم إقتراح خطوات مستقبلية فى نفس خط البحث مثل استعمال كمية أكبر من الماء وإستعمال برنامج حاسب الى للحسابات.

الخلاصة

الهدف من هذه الدراسة هو إختبار تأثير إضافة الماء إلى الهواء الجوى الداخل إلى محرك احتراق داخلى يعمل بنظام الوقود المشترك (ديزل و غاز البترول المسال) على أداء المحرك وعلى ضوضاء الاحتراق الناتجة. محرك الوقود الثنائى هو محرك ديزل عادى يستعمل جزء بسيط من وقود الديزل لبدء الإحتراق ويستعمل فى نفس الوقت وقود غازى وهو غاز البترول المسال (غاز الطبخ) كوقود أساسى يدخل مع الهواء.

إضافة الماء إلى غرفة احتراق المحرك لها تأثير معروف على محرك الديزل من تقليل غازات أكاسيد النيتروجين فى غازات العادم, وتم إرجاع نقص هذه الاكاسيد الضارة بصحة الإنسان إلى نقص درجة حرارة الاحتراق عند استعمال الماء مما يقلل تكونها وانبعاثاتها.

وتهتم هذه الرسالة بالتأثيرات الناتجة عن إضافة الماء على أداء المحرك وعلى انبعاثات غازات أول أكسيد الكربون وغازات الهيدروكربونات غير المحترقة ونسبة الدخان الأسود و كذلك شدة الضوضاء الناتجة عن الاحتراق فى المحرك الثنائى الوقود.

تم عمل برنامج عملى لإجراء التجارب باستعمال محرك أبحاث "ريكاردو" يعمل بوقود الديزل بحيث يمكن تغير والتحكم فى كافة عوامل التشغيل والتصميم, وتم عرض نتائج للمحرك الثنائى الوقود عند نسب مختلفة من الماء بالنسبة لكمية الوقود وكذلك عند ظروف تصميم وتشغيل مختلفة.

جامعة الإمارات العربية المتحدة
عمادة الدراسات العليا
برنامج علوم البيئة

دراسة معملية لقياس والتحكم في ضوضاء الاحتراق لمحرك ثنائي الوقود

رسالة مقدمة إلى

جامعة الإمارات العربية المتحدة
استكمالا لمتطلبات الحصول على درجة الماجستير في العلوم
(علوم البيئة)

مقدمة من الطالب
عبدالله أحمد جاوي العسيري
بكالوريوس هندسة ميكانيكية
جامعة الإمارات العربية المتحدة - 2000م

تحت إشراف

د.محمد يونس سليم استاذ مشارك قسم الهندسة الميكانيكية كلية الهندسة جامعة الإمارات العربية المتحدة	د.صلاح العمري أستاذ مشارك قسم الهندسة الميكانيكية كلية الهندسة جامعة الإمارات العربية المتحدة

مايو - 2008