COMPARISON OF ENGINE PERFORMANCE AND EMISSIONS FOR CONVENTIONAL PETROLEUM DIESEL FUEL AND DIESEL-ETHANOL BLENDS

A THESIS SUBMITTED TO
THE GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES
OF
MIDDLE EAST TECHNICAL UNIVERSTY

BY

GÜL ERKAL

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR
THE DEGREE OF MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

APRIL 2010

Approval of the thesis:

COMPARISON OF ENGINE PERFORMANCE AND EMISSIONS FOR CONVENTIONAL PETROLEUM DIESEL FUEL AND DIESEL-ETHANOL BLENDS

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ABSTRACT

COMPARISON OF ENGINE PERFORMANCE AND EMISSIONS
FOR
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BLENDS

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January 2010, 127 pages

Ethanol is an environmental friendly alternative diesel fuel that has received significant attention both as a possible renewable alternative fuel and as an additive to existing petroleum-based fuels. Beyond simply representing an additional fuel supply, ethanol exhibits several advantages when compared to existing petroleum fuel. The objective of this work is to investigate experimentally the effects of using different blends of specified percentages of ethanol on the engine performance and emissions and to compare it with that of conventional diesel fuel. Tests will be done on the ‘‘Engine Test Laboratory’’ of the Turkish Tractor Factory (TTF) using a four-cylinder, turbocharged and naturally aspirated, DI diesel engines. Engine performance parameters such as engine speed, torque, power, fuel consumption will be measured. At the same time, the engine emissions including particulate matter, unburned hydrocarbons, carbon monoxide, and NO_X will also be recorded.

Keywords: Diesel Engine, Ethanol, Performance Parameters, Emission Parameters, Alternative fuel

ÖZ

MEVCUT DİZEL YAKITLARI VE DİZEL-ETANOL KARŞIMLARI İÇİN MOTOR PERFORMANS VE EMİSYONLARININ KARŞILAŞTIRILMASI

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Ocak 2010, 127 sayfa

Etanol yakıt, çevre dostu ve yenilenebilir önemli bir alternatif yakıttır ve şu an kullanılan petrol bazlı yakıtlara katkı maddesi olarak kullanılır. Basit bir katkı maddesi olarak biliniyor olsa da, etanol , şu an kullanılan yakıtlara göre oldukça önemli avantajlara sahiptir. Bu çalışmanın amacı, belirlenen yüzdelerle oluşturulacak karışımlarla, motor performans ve emisyon değerlerini incelemek ve diğer yakıtlarla karşılaştırmaktır. Testler, Türk Traktör Fabrikası "Motor Test Laboratuarında" ve 4 silindirli, turbo ve doğal emişli dizel motorlar kullanılarak yapılacaktır. Hız, tork, güç ve yakıt tüketimi gibi motor performans değerleri ölçülecek, aynı zamanda motor emisyon değerleri de incelenecektir.

Anahtar Kelimeler: Dizel motor, Etanol, Performans Parametreleri, Emisyon Parametreleri, Alternatif Yakıtlar

To My Beloved Parents

ACKNOWLEDGEMENTS

I would like to express my gratitude to all those who gave me the possibility to complete this thesis. I am deeply indebted to my supervisor Assist. Prof. Dr. Ahmet Yozgatlıgil whose, stimulating suggestions and encouragement helped me in all the time of research of this thesis.

I want to thank Dr. Remzi Şahin for his great help and for his precious comments during all my studies.

I want to thank the Turkish Tractor Factory for giving me permission to commence this thesis in the first instance, to do the necessary research work and to use laboratory for all tests. I have furthermore to thank to Mr. Friedrich Wirleitner, Mr. Ergün Gültekin, Mr. Hasan Haldun Özgümüş, Mr. Aydın Gültekin, Mr. Soner Ateş and Mr. Tahir Yıldırım who gave and confirmed this permission and encouraged me to go ahead with my thesis.

I want to thank to TETRA Technological Systems Ltd. to use laboratory for all analyzes and thank to my sister Miss. Sibel Erkal and Mr. Gürsel Ulutuncel for their support and help.

I want to thank my colleagues from Turkish Tractor Factory and ORS supported me in my research work. I want to thank them for all their help, support and interest.

Especially, I would like to give my special thanks to my parents for their support to complete this work.

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SYMBOL:

(A/F):	Air Fuel Ratio
A_n:	Nozzle tip Opening Area
B:	Fuel consumption per hour (kg/h)
be:	Specific Fuel Consumption (kg/kWh)
BSFC:	Brake specific Fuel Consumption
BTE:	Brake Thermal Efficiency
Conc:	Concentration (ppm)
FSN:	Fuel Smoke Number
G_{AIRW}:	Intake Mass Flow Rate (kg/h)
$G_{EDFW,i}$:	Equivalent Diluted exhaust Gass Mass Fow Rate (kg/h)
G_{EXH}:	Exhaust Gas Mass Flow Rate (kg/h)
G_{FUEL}:	Fuel Mass Fuel Rate (kg/h)
G_{TOTW}:	Diluted Exhaust Gas mass Flow Rate
Ha:	Absolute Humidity of the Intake Air g/kg
Hu:	Lower thermal Value of the Fuel (kj/kg)
Kp:	Humidity Correction Factor for Particulate
n:	Motor Rotation Speed (1/min)
Ne:	Effective Power (kW)
Md:	Rotation Momentum (Nm)
$M_{SAM,i}$:	Mass of Diluted Exhaust Sample Passed Through the Particulate Sampling Filter
p_A:	Saturation Vapor Pressure (kPa)
P_B:	Degree of filter paper blackened
p_B:	Total Barometric Pressure (kPa)
P_L:	Pollution Level (%)
PTmass:	Particulate Mass Flow Rate (g/h)
R_A:	Absolute lightness of sample

Ra: Relative Humidity of Intake Air (%)

R_B: Power of reflection of blackened filter paper

R_W: Power of reflection of white filter paper

$WF_{,i}$: Weight Factor "i"

Symbols for the chemical Components:

CO: Carbon Monoxide

CO_2: Carbon Dioxide

HC: Hydrocarbons

NOx: Nitrogen Oxide

NO: Nitric Oxide

NO_2: Nitrogen dioxide

O_2: Oxygen

PT: Particulate

PM: Particulate Matter

H_2O: Water

C2H5OH: Ethanol

Abbreviations:

CLD: Chemiluminescent Detector

FID: Flame Ionization Detector

HFID: Heated Flame Ionization Detector

NDIR: Non Dispersive Infra Red type Analyzer

TGA: Thermo Gravimetric Analyzer

Greek Letters:

ρair **:** Atmospheric air Density (kg/m^3)

$\rho fuel$ **:** Fuel Density (kg/m^3)

ηt **:** Thermal Efficiency

λ **:** Excess air Coefficient

ϕ **:** Equivalence Ratio

CHAPTER 1

INTRODUCTION

Energy is the basic element of daily and industrial life. Only 30% of the energy in the world is produced without combustion in hydraulic and nuclear plants. The rest 70% is obtained from burning fossil fuels such as coal, petroleum, gas or their synthetic derivatives [1]. The rapidly increasing world population, industrialization, and rapid urbanization increase the consumption of these fossil fuel sources. On the Table 1.1 the duration of the use of fossil fuel reserves of the world is given.

Table 1.1 The duration of the use of fossil fuels reserves of the world (2002) [2]

	Petrol	Natural Gas	Coal
Location	Year	Year	Year
North America	14	11	239
Central and South America	38	66	474
Western Europe	8	18	161
Eastern Europe	24	82	500
Central Europe	87	100	175
Africa	28	98	268
Asia and Oceania	16	40	164

According to this table, in 50 years this source will have a dramatic decrease with the effect of increasing world population. Recently, the basic source of the automotive industry is fossil fuels. Since the fossil fuel reserves are located in a particular region,

for time to time some economical and political crises occur in the world. Due to these crises and economical conflicts, petrol prices increase.
The increase in prices of diesel fuel and foreseeable future depletion of world wide petroleum reserves provide strong encouragement for the search of alternative fuels [3].

On the other hand diesel engines are the major source of air pollution with its toxic exhaust emissions like nitrogen oxide, carbon monoxide, sulfur dioxide, and other volatile organic compounds, soot, and PM. It is commonly known that these compounds are harmful to human and ecological health. Also, the combustion of petroleum products is the major cause of global warming and the ecological imbalance [4].

1.1 Fuels

Diesel fuel is the physical mixture produced by refined petroleum. Up to 150 ºC raw fuel, up to 150-250 ºC kerosene, jet fuel, up to 250-350 ºC diesel fuel and after 350 ºC heavy oils are obtained from petroleum. Although petroleum is known to be a specific kind of fuel, it is actually a raw and natural material extracted from underground. Because of the mixture of hydrocarbons petroleum does not always have a fixed chemical composition.

1.2 Alternative Fuels

Biofuels, used in transportation, are helpful and environment-friendly alternative fuels. Besides, while evaluating the factors affecting the environment as a whole, development of renewable energy and their effects on utilization density of agricultural lands should also be considered.

Moreover, the economy of the alternative fuels used in engines has a significant importance. Especially as a result of the growing economic crisis, it is desired to obtain maximum performance from fuels which are also expected to be economical.

Because of its economical characteristic much interest is shown to the engines operating with diesel. As an example for its economy, the LPG that is once only used by taxi drivers and has become widespread in the whole country. Therefore, a research to obtain the most economical fuel has still being carried on.

Alcohol and its mixtures, which are evaluated as alternative fuels, have reached a significant level of usage in many countries. Alcohol can be produced from agricultural products. The facts that our country is an agricultural country, it has rich lignite reserves and it is one of the leading petroleum importer countries show that alcohol can become an important source of energy for our country.

It is possible to examine alcohol based fuels in two groups: Methanol and ethanol can be used in gasoline and diesel engines purely or by being mixed in different rates.

Methanol, of which chemical formula is CH_3- OH, is a toxic form of alcohol, colorless and odorless. It has a wider range of ignition when compared to gasoline and diesel fuel. Thus, the saturated vapor in the fuel tank can be explosive according to the environmental temperature. Oxygen content in methanol is 49,9 % .Though its oxygen content is an advantage and its lower heating value, which is 2.2 times lower than gasoline, is a disadvantage.

Ethanol is a colorless and transparent liquid of which chemical formula is C_2H_5 - OH, can be considered as the same with methanol in terms of engine characteristics. Its latent evaporation temperature is 1.3 times lower than methanol. Its energy density which is higher than methanol provides opportunity to use smaller fuel tanks in vehicles.

1.3 Ethanol in the World

Ethanol is the alternative fuel which becomes widespread in the fastest way among renewable energy sources. Its production rate and its trade increase extensively. The most significant reason why biofuels are rapidly becoming widespread and why they constantly increase their economical value is that they reduce the environmental effects caused by other fuels.

Ethanol usage decreases CO_2 emission values. It will reduce the foreign dependency of the country. The significance of ethanol is once more emphasized by the price obscurities caused by the instabilities in supply demand balances of other fuels and the fact that other fuel sources have started to be exhausted. As its raw material is agricultural the facts that it improves agricultural development and it's supporting sustainable agriculture by directing agricultural surplus product risk to energy agriculture are among its significant advantages for an agricultural country like Turkey.

For the vehicles operating with alternative fuels, the models that have been developed most in the world are known to be those operating with bioethanol. The number of the vehicles that use ethanol, which is more powerful and less harmful to the environment, is increasing day by day. Today, the mixture of gasoline and 5% - %10 ethanol in volume in gasoline engines is a common application. In diesel engines, as in gasoline, low rates of bioethanol-diesel fuel mixtures can be used without any modification.

The usage of ethanol in engines is more common in countries with wider agricultural fields. In the states of the USA where agriculture is common E80 fuel with 80% ethanol 20% composition is used as fuel in vehicles for years. In Brazil, where there is hardly any petroleum reserve but lots of sugar cane, vehicles operate with ethanol for more than 15 years.

Ethanol is a kind of alcohol obtained from the fermentation of sugar, cellulose and starch that can be converted to sugar. Ethanol can be obtained from agricultural products such as potato, cereals, sugarcane and sugar beet.

Regarding the ethanol usage in the world: The USA is the second country in the world to have the highest capacity production facilities and various promotions are still applied in order to reduce production costs. Germany, with its legal regulations, has enabled the usage of 100% biodiesel and has exempted biodiesel from consumption taxes. In France, tax promotions are applied for biodiesel and mixtures up to 5% are permitted in the refineries. In Italy, tax exemptions for a certain period of time are applied to the facilities with a capacity up to 125,000 tons. In Belgium, 100% biodiesel usage is permitted. There are also tax promotions for some certain experimental projects. In Spain, tax reduction is applied for the biodiesel used in experimental projects. In Austria, 100% biodiesel usage is permitted and there are tax exemptions. In England, except for some areas tax exemptions are applied.

In Brazil, the biggest ethanol manufacturer of the world, one third of all vehicles work with ethanol obtained from sugar cane. Brazil exports 1,890,000 tons of ethanol to Western countries per year. As biological fuel production is based on labor force as much as agriculture ethanol sector is known to employ nearly 1 million people in 1997.

Sweden, one of the countries that pay utmost importance to biodiesel, plans to remove petroleum completely.

All of the vehicles in the country will operate with biodiesel; and to this end, the government has put R&D projects in cooperation with the two leading vehicle manufacturers of Sweden, Saab and Volvo.

1.4 Ethanol in Turkey

The ethanol Production Capacity of Turkey:	144,000 m3/year
1. Çumra Bioethanol Facilities	84,000 m3/year
2. Other Bioethanol Facilities	60,000 m3/year

If the ethanol usage rate in our country increases from 2% to 5%, the amount that is paid to petroleum import will be decreased by 670 million $. Thus, agricultural products will be supported, employment rate will increase, the additive value will reach 304 million $. Ethanol, obtained from local products that are added to gasoline such as wheat, corn and especially sugar beet, is becoming widespread rapidly.

Turkey needs to pay more attention to ethanol which has a usage rate between 10% and 30% in countries such as the USA, Brazil, EU countries and China. The ethanol if it is produced in our country will increase growth rate, employment rate and the income of our country by providing additive value together with supply security.

1.5 Alcohol Fuels

The main factor for alcohol usage becoming widespread is that petroleum production rate in the world has reached a level which is unable to meet the demand. It is possible to state the advantages and the disadvantages of using alcohol as an alternative fuel in engines as follows [5]. Alcohol can replace the fuels obtained from crude oil, the countries that do not have crude oil reserves can meet their energy necessities without having to be dependent on petroleum. Alcohol productions (especially ethyl alcohol) can constitute a significant income source for the farmers of the countries with developed agricultural conditions.

Alcohol can achieve high compression rates and increase engine performance with their higher octane number. In other words it can be possible to obtain higher performances from smaller engines with using alcohol. Alcohol increases the octane value of the mixture when it is mixed with gasoline. The octane value of the mixture consisting of 10% methanol and 90% gasoline is 95. Alcohol can cause obstruction in the filters of fuel systems that is converted from gasoline to alcohol. If the alcohol – gasoline mixture contains a small amount of water, phase separation ensues especially in cold weather. Phase separation mostly occurs in methanol mixtures. Some parts of gasoline fuel systems are not compatible with alcohol. Although important for small amounts of ethanol – gasoline mixtures, pure methanol causes serious damages in fuel systems. In order to prevent this kind of problems additives must be used with blends with alcohol.

Another reason why alcohol is an alternative to engine fuels is that its emission is low [6].

1.6 Physical and Chemical Properties of Ethanol

The researches show that ethanol is successfully used in engines and operate with petroleum without any necessity for additives. While researchers search for ways to make ethanol an effective alternative they have to study about the chemical and physical properties of ethanol.

1.6.1 Chemical Properties of Ethanol

Chemical composition of the ethanol affects the engine performance and emissions.

1.6.2 Blend Stability

Temperature and water content of ethanol are the most effected factors of ethanol solubility [7]. At lower temperatures and higher water contents ethanol diesel blend shows a phase separation. In this study, in order to eliminate the phase separation, a mixer is mounted on the fuel tank to mix the blend during the tests. To prevent phase separation and instability of the blend two different alternatives can be chosen: Adding an emulsifier which acts to suspend small droplets of ethanol within the diesel fuel, or adding a co-solvent that acts as a bridging agent through molecular compatibility and bonding to produce a homogeneous blend [8].

Another specification of ethanol affecting the miscibility of ethanol is its aromatic content. Reducing the aromatic content of diesel fuels will affect the miscibility of ethanol in diesel fuel and the amount of additive required to achieve a stable blend [7].

1.6.3 Cetane Number - Octane Number

Chemical composition of the fuel affects the engine performance and emissions. The cetane number is the most important specification of diesel fuel and it is the most important factor that affects combustion. Cetane number is the tendency of the fuel to self ignition in engine combustion chamber. In other words a very small cetane number means a very longer ignition delay, and a late and incomplete combustion which cause the following: decrease in efficiency, increase in fuel consumption, emission of air pollutant gases, and difficult cold starting.

Ethanol has high octane number and this means that it can be a good fuel for spark ignition engines. Octane number of ethanol increases when used with gasoline and maintains a complete combustion by reducing the emissions of hazardous gases such as CO and hydrocarbons. It has a high octane number and this provides an advantage for being used in internal combustion engines. Also, its high cetane number compared to other alternative fuels like methanol ensures that ethanol can be considered as a good diesel engine fuel too. But it is a fact that the cetane number of ethanol is lower that diesel fuel and blending ethanol with diesel will reduce the cetane level of diesel fuel. Cetane improver is readily available and suitable and it can be used in ethanol diesel blends

A high cetane number ensures good cold starting ability, low noise, and long engine life [7]. Cetane numbers of blended fuel depend on the amount and type of self ignition improver additives used in the blends and the percentage of ethanol. Because the cetane number of ethanol is extremely low, the ethanol–diesel blend fuel cetane level reduces significantly.

1.6.4 Viscosity and Lubricity

Lubricity is a well known potential problem when ethanol is used with diesel fuel. Fuel viscosity and lubricity are very important parameters for fuel systems. Lower fuel viscosities lead to greater pump and injector leakage which reduce maximum fuel delivery and power output [7]. And also, these leakages increase the fuel consumption of the engine and reduce the maximum fuel delivery and ultimately power output.

1.6.5 Energy Content

Ethanol has less energy content than diesel fuel which means for every percent of ethanol added to a blend reduces energy content of the blend. This property of ethanol is the basic reason of the power output reduction in the engine. Also it causes higher fuel consumption rates for ethanol diesel blends.

1.6.6 Flash Point

The flashpoint of the fuel affects the shipping and storage classification of fuels and the precautions that should be used in handling and transporting the fuel. The flashpoint of ethanol–diesel blend fuels is mainly dominated by ethanol. [7]

1.7 History of Diesel Engines

The history of diesel engines dates back until the years in which the automobile was born. This engine, discovered by Rudolf Diesel towards the end of 1800s, has started to be used in ships and trains in years and diesel engines have become available for vehicles.

Diesel engines, which were developed by Rudolf Diesel, were first investigated in around 1824 by the French engineer Carnot, and by Herbert Akroyd Stuart from 1885-1990. These were followed by Captaine's semi-diesel engine in 1890.

Finally, Rudolf Diesel made the research continuing from1892 to 1897 and obtained patent for them. As it was stated previously, the working principle of the diesel engines was first introduced by the French engineer Nicholas Leonard Sadi Carnot in 1824. Carnot cycle is a special thermodynamic cycle introduced in 1820s by Sadi Carnot and it was developed by Emile Clapeyron in 1830s and 1940s.

The reason why Carnot cycle is the most productive cycle available is that it consists of fully reversible steps. No heat transfer occurs between the two systems among which there is a temperature difference in any of these steps. Therefore, the entropy changes in every step and in total it is zero.

In 1895, three years after the first trial, Dr. Diesel introduced his machine which operated in accordance with four-cycle, cooled with water, in which fuel was ejected to the cylinder with high pressure air and of which thermic productivity rate was 24%. Afterwards, as a result of the cooperation between Dr. Diesel and MAN, the first example of the diesel engines used today was produced in 1897. In 1923, the first diesel vehicles were built and shown at Berlin Motor Fair. Caterpillar Company introduced the Diesel engine tractor in 1931. Mercedes Benz Company built the first automobile with a diesel engine (Type 260D) in 1936. M.A.N. produced the first four stroke engine reaching 45% efficiency in 1950.

1.8 Burning and Emissions of Diesel Engines

The high temperature levels in the combustion chamber of diesel engines cause nitrogen and oxygen turn into hazardous nitrogen oxides. Nitrogen oxides are gases hazardous to health and to environment. In diesel engines, the engine works with a bigger amount of air which increases the potential of formation of nitrogen oxides. The combustion rate in diesel engines is low in other words, the fuel - air mixture entering the combustion chamber does not burn homogeneously and greenhouse gases are revealed.

In order to cope with these gases which are hazardous to human health and to environment emission norms have been set. Emission norms are the rules that have to be obeyed in order to cope with air pollution which is among significant problems in our country, too.

In motor vehicles, which are among the primary sources of air pollution, the pollutants are CO, HC, NO_x and PM. The pollutants which are of more importance in diesel engines are NO_x and PM. Recent researches aim to reduce the level of the nitrogen oxides NO_x and CO emissions. Using alternative fuels serves this purpose which is an important advantage.

The volumetrically composition of the air can be summarized as follows: 21% O_2, 78% N_2, 1% various gases such as Argon (Ar), carbon dioxide (CO_2), methane (CH_4) and water vapor (H_2O). The fact that even minor changes in the composition of the atmosphere can result in huge climatic changes puts an emphasis on the significance of air pollution. Among the pollutant emissions in atmosphere, 93% of CO_2, 57% of HC, 39% of NO_x and 1% of SO_2 (diesel) are released by motor vehicles.

The first thing necessary for the improvement of engine exhaust emissions and performances at the same time is the effective combustion. Obtaining a good performance from the engine can be possible by using the fuel effectively and by constantly controlling the emissions under all working conditions.

These relationships between engine efficiency and emissions have directed the researchers to seek ways of obtaining high levels of efficiency from the engine economically and without polluting the environment. Increasing the engine performance and reducing the emissions by minimizing the power losses of the engine are the main subjects in the studies.

1.9 Combustion and Exhaust Emissions

Diesel engine emissions firstly depend on the engine design and operating conditions. Of course fuel property is very affective on the composition of exhaust emissions. [9]

The diesel engines always operate with much more air than spark ignition engines and more air in the combustion chamber means minimum CO emissions. So, CO emission is not a significant problem for diesel engines. However, for diesel engine formation of particulates and NO_x are real problems. Figure 1.1 represents the formation of emissions for complete and incomplete combustions. The emission of gases and particulates covered by legislation are produced by incomplete combustion which is:

- Carbon monoxide (CO),
- Carbon (C) (form of smoke)
- Hydrocarbons (THC)
- Nitric oxide (NO) and nitrogen dioxide (NO_2), considered as NO_x. [9]

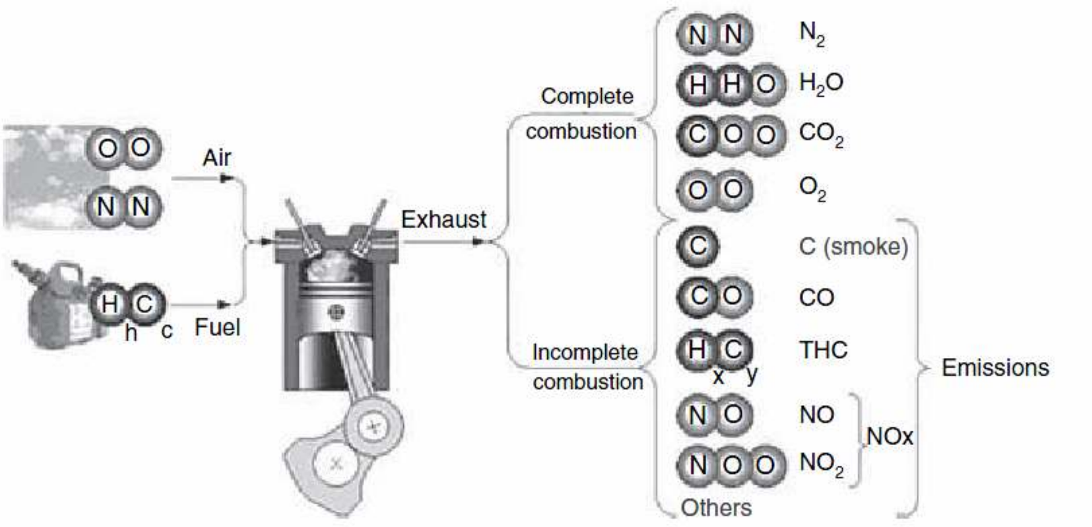


Fig.1.1 Gaseous components of combustion processes

Individual diesel exhaust particles may be highly complex coagulations of compounds, as shown in Fig. 1.2. Around 90 percent of particles emitted by a modern automotive diesel may be below 1_m in size which challenges the tools used to measure their presence [9].

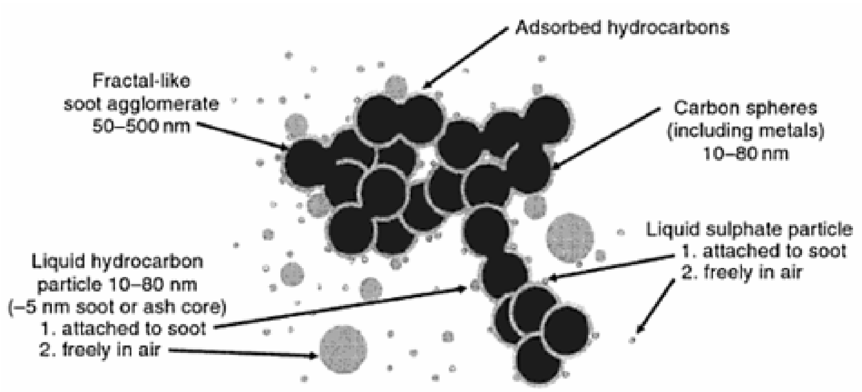


Fig. 1.2 Schematic Representation of Diesel Particles

Emission formation also depends on the design of combustion chamber and injectors. Soot and unburned hydrocarbons are formed during premixed and mixing control phases. Formation of NO_x is depended on combustion temperature and mixture ratio. Combustion temperature should exceed 1800K to formation of NO_x. The mixture has much more oxygen concentration which means that PM and THC formation rate is reduced.

1.10 The Scope of the Study

The purpose of the study is to use ethanol as an alternative fuel for tractor engine without any engine modification and using any additives. The aim of this study is to investigate the optimum ethanol percentage can be used with turbocharged and naturally aspirated diesel engines. For this aim performance and emission tests with using different blends were performed. With these tests effects of ethanol to emissions and performance parameters are observed. All experiments were performed according to European Union Directive (97/68/EC) which is describing the emission limitations for off-road diesel engines.

Another aim of this study is to investigate the effects of turbocharger unit on emissions with or without using ethanol. Firstly all emission and performance tests were performed with naturally aspirated engine and secondly, all tests repeated with a turbocharged engine.

In this study the effects of blending ethanol with diesel fuel not only investigated for CO, HC and NOx emissions but also observed PM components using Thermo Gravimetric Analyzer.

Details about this study are explained in the following chapters. Results and details in previous studies which are performed with using ethanol with diesel are given in Chapter 2. Experimental set up is presented in Chapter 3. Emission and performance test procedures and performed calculations and formulations are given in Chapter 4. Chapter 5 is representing the test results and comments for all experiments and in Chapter 6 conclusion and future works are presented.

CHAPTER 2

LITERATURE SURVEY

The importance of the world's environmental pollution and the strict governmental regulations on exhaust emissions has led us to seek alternative fuels to automotive manufacturers. For these purposes a number of studies on the blends of diesel fuel and alternative fuels have been performed.

Ethanol is an important alcohol-based alternative fuel used to reduce air pollution level and consuming petroleum fuels. To reduce the dependency of petroleum fuels ethanol has been received much attention in recent years by many countries. Moreover, ethanol is recognized as an environmentally friendly alternative fuel because previous studies have shown that there is a substantial reduction of CO, unburned hydrocarbons and particulate matter emission in ethanol compared to conventional diesel engines. [10-11]

İçöz et al. (2009), investigated the production of ethanol from sugar beet and use ethanol as a fuel in Turkey. They stated in their study that using ethanol will reduce to consumption of fossil fuels and emissions of greenhouse gases. Turkey imports approximately 60% of its total energy demand. The advantages of blending %5 ethanol with fuel improve Turkish rural economy, lead to grow 10–15% more agricultural products than that at present, reduce foreign dependency for fossil fuels and emitted hazardous gases. Besides these advantages show that Turkey's agricultural potential is sufficient to grow agricultural products to produce ethanol [12].

Kenneth R. et al. (2010), perform their researches about the technology and economics of blending ethanol with gasoline. The first limitation using ethanol with gasoline is the lower vapor pressure of ethanol. Vapor pressure is an important property for engine cold start. Higher octane number of ethanol is an advantage for blending with gasoline. And the second barrier is ethanol's lower energy. Lower energy causes power reductions and more fuel consumptions. Kenneth R. et al. suggested that to produce cheaper gasoline with lower octane number and blend it with ethanol to increase its octane number can be a good solution for expensive fuel prices. Ethanol's water content is another restriction because presence of water in ethanol prevents it from mixing homogenously with gasoline. And of course its oxygen content is an advantage to obtain complete combustions [13].

2.1 Effects of Ethanol in Diesel Engine

Reviewed papers dealing with the ethanol diesel fuel blends for diesel engines are presented in Table 2.1. Test engine, test conditions, emissions, and performance data are given in this table. Following coding is used on the table:

R: Reduce **I:** Increase, ***:** Slightly Increase or Decrease, **-:** Not Change, **X:** Not Measured

Rakopoulos et al. (2008) performed experiments with ethanol as an alternative fuel to understand its effects on performance and emissions. A six cylinder turbocharged and inter-cooled Mercedes Benz mini bus engine are operated with 5% and 10% ethanol by volume. Tests are performed without any engine modifications and any cetane improving additives but an emulsifying agent is used to obtain homogeny mixture. Referance diesel fuel and blends are tested at three loads (20%, 40% and 60%) for 1200 rpm and 1500 rpm.

They reported that by using ethanol, NOx level stayed the same or it decreased slightly and THC level increased. Bsfc was a little higher than the reference diesel case and brake thermal efficiency was equal to or slightly higher than the diesel fuel case. As a conclusion, it can be said that ethanol can be safely used for mini bus engines up to 10% level [14].

Table 2.1 Available Emission and Performance results for several studies with Ethanol

Reference	Author	Ethanol Level	Engine Description	Operating Conditions	Perform. Measur.			Emissions Measured				
					Power	BSFC	BTE	HC	CO	NO_x	PM	SMOK
14	Rakopoulos, et al.	5%, 10%	Mercedes Benz, Turbocharged and Intercooled Direct Injection 6 cylinder	Various Speed and Load Points	X	I	I	I	R*	R*	X	R
15	Lapuerta, et al.	10%	European cars Turbocharged and Intercooled Direct Injection 4 Cylinder	5 Mode Test Cycle	X	X	X	I*	X	I*	R	X
16	Kim and Choi	15%	Turbocharged and Intercooled Direct Injection 4 Cylinder	ECE R49 13 Mode Test Cycle	X	I*	I*	I	I	I	R	R
3	Ajav, et al.	5%, 10% 15%, 20%	Kirloskar 1 Cylinder Water Cooled	Constant Speed Various Load	-	I	R	X	R	R	X	X
17	Satge de Caro, et al.	10%, 15% 20%	Hatz DI engine 1 Cyl. Renault IDI engine 4 Cyl.	Various Speed and Load Points	I	X	X	I	R*	-	X	X
					R	X	X	I	I	X	X	R
18	Can et al.	10%, 15%	Ford XLD 418T Turbocharged 4 Cylinder Indirect Injection Water Cooled	Constant load and Variable Speed Points	R	X	X	X	R	X	R	R
19	Chong-Lin et al.	5%, 10% 15%, 20%	Dongfeng Chaoyang Engine Turbocharged and Intercooled Direct Injection 6 cylinder	ECE R49 13 Mode Test Cycle	X	X	X	I	I	I*	R	R
20	Dominiguez et. al.	5%, 10% 15%	Air Cooled, 2 Cyl. DI Naturally Aspirated Diesel Engine	8 Mode Test Cycle (97/68/EC)	R	I	X	I	I	-	R*	X
21	Chen, et al.	10%, 20% 30%	Cummins 4B 4 Cylinder Diesel Engine	Constant Speed Variable Load and Constant Load Variable Speed	R	I	X	X	I	I*	R	R
22	AbuQudais et al.	20%	Naturally Aspirated 4 Cylinder Water Cooled	Various Speed and Load Points	X	X	I	I	R	X	R	X
23	Bang Quan et al.	10% 30%	Naturally Aspirated 2 Cylinder DI	Various Speed and Load	X	I	X	X	X	I	X	R

Lapuerta et al. (2008) studied the effects of ethanol up to 10% on an automotive diesel engine with no engine modifications and no additives. They also used partial flow dilution tunnel to collect PM and they observed the change of particulate size distribution using ethanol. Torque, bsfc, and thermal efficiency were increased and smoke opacity and particulate matter emissions were reduced by adding ethanol to diesel. They measured a reduction in the mean particle diameter and a balance in NOx emissions. They suggested in their study that using additives obtains more homogeny blends and increases solubility of ethanol. Also, with some modifications in fuel pump the performance of the engine can be better. [15].

Kim and Choi (2008) used ethanol due to its oxygenated contains with common rail DI diesel engine. In their study they used three different fuels: reference diesel fuel, ethanol-diesel blend and blend with cetane improver to investigate the engine performance and emissions with these fuels. They performed 13-mode test cycle and they reported that THC and CO emissions were slightly increased and smoke and particulate matter was decreased. They stated that ethanol is a good alternative fuels for spark ignition engine thanks to its high octane number but its low cetane number resists self-ignition in diesel engines [16].

Ajav et al.(1999) studied with 5%, 10%, 15% and 20% ethanol levels with diesel fuel and investigate the effect on engine performance and emissions parameters. There was not any significant difference between the power developed on the blends and the diesel alone. As a result up to 20% ethanol can be used safely with diesel without any power reduction. The more ethanol percentage increases, the more Bsfc increases Max brake thermal efficiency was observed with diesel fuel due to higher fuel consumption with ethanol. Because of ethanol's lower calorific value exhaust gas temperature decreased while the percentage of ethanol in the blend was increased. Ajav et al. stated that due to ethanol's cooling effect the lubricating oil temperature was reduced by using ethanol. They also measured emissions data to understand the effects of ethanol. They observed that ethanol causes a reduction in CO and NOx emissions. [3]

Satge de Caro et al. (2001) investigated the effect of ethanol with different additives on two different DI and IDI engines. They studied about the compatibility of diesel ethanol mixture in the presence of additive. Due to its water content ethanol has limited miscibility. To prevent cloudiness and phase separation, diesel fuel additives has to be used. This will make the stability of blend increase. Viscosity is an important physical property of ethanol that has to be improved. Additive must be used to improve lubrication. As it is known ethanol has very lower cetane number that reduces blends cetane level.

Satge de Caro et al. also used this additive to keep cetane level under control for good cold starting and suitable ignition. They tested DI engine with up to 10% ethanol and IDI engine up to 20% ethanol with and without additives to see the effects of ethanol and also additives. Variable speed variable load test cycle was selected and tests were performed for all engines. For DI engine ethanol leads to a power reduction and a loss of heat content. NOx does not show any specific trend using ethanol but CO level is reduced and HC level is increased. It was observed that using additives reduces the increase in HC emissions. For IDI engine, with the same test cycle, due to loss of heat content using ethanol leads to 11% reduction in engine power. While HC and CO emission is increased NOx emission is decreased. Smoke is reduced when ethanol is added to diesel. They concluded that using additive improves ignition delay and cetane level. It controls HC and CO emissions and reduces NOx emissions [17].

Can et al. (2004) evaluated isopropanol as a possible additive to prevent phase separation and to obtain a homogeny mixture. Their test results show that the ethanol addition reduces CO, soot, and SO2 emissions but it caused a reduction in power. Can et al. explains this reduction with lower heating value and density of ethanol. Due to lower cetane number longer ignition delay and inefficient combustion occur and these reduce power and torque. They also tested engine with different fuel injection pressures as 150, 200 and 250 bar. As a result higher injection pressures cause a reduction in engine power. NOx emissions increase with higher injection pressure and CO emissions decrease [18].

Chong-Lin et al. (2007) examined ethanol-diesel blended fuel as an alternative fuel for diesel engine up to 20% level of ethanol. They measured the regulated emissions as THC, CO, Ox, and PM and investigate the PM extracts. They tested the engine according to ECE R49 13-Mode tests cycle and they observed that CO and THC emissions increased with the increase of the ethanol volume percent and NOx emissions slightly increased. PM consists of dry soot and SOF. By using ethanol dry soot was reduced and SOF was increased because of more incomplete combustion and unburned hydrocarbon [19].

Dominguez et al. (2005) investigated the effects of ethanol on performance and emissions with two different engines which were on-road and off-road. They prepared containing up to 15% ethanol and compared results for these two engines. The 8-mode test cycle (According to 97/68/EC) for off-road engine and 13-mode test cycle (according to 1999/96/EC) for on-rod engine were conducted. As a result 5% power reduction was observed with 15% ethanol content for off-road engine and 4% power reduction was observed with 10% ethanol content for on-road engine. For the emissions, both engines showed the similar results that THC and CO emissions rose and NOx and PM showed no general tendency [20].

Chen et al. (2008) added vegetable methyl ester to prevent phase separation of the mixture. They prepared four kinds of fuel with diesel up to %30 ethanol and up to 10% ester. They stated that oxygenated fuels were helpful to reduce PM emissions and they proved that with their tests. According to their test results PM decreased while the percentage of ethanol in the blend increased. Chen et al. also investigated the effect of ethanol on PM compounds like dry soot, and SOF and sulfate. It was observed that adding ethanol to diesel fuel increased SOF and sulfate and decreased dry soot level [21].

Abu-Quidas et al. (2000) studied on an alternative blending method called fumigation which is the addition of ethanol to the intake manifold. They explained the advantages of fumigation technique are as a minimum modification in engine and the facility of using ethanol or only diesel in the engine if wanted. The basic limit for direct blending of ethanol to diesel is the ethanol's limited solubility but with fumigation method homogeny mixtures can be obtained. They also tested three different blends to see the effects of fumigation method comparing to direct blending method. As a result, fumigated ethanol reduced smoke and soot emissions much more than blending. Brake thermal efficiency, CO and THC emission levels increased compared to the fuel prepared by blending method. [22].

Bang-Quan et al. (2003) conducted exhaust emission and performance tests with ethanol and analyze both regulated and unregulated emissions like formaldehyde, acetaldehyde and unburned ethanol emissions. They stated that gasoline engine emitted unburned ethanol level was higher than those diesel engines. According to their test results formaldehyde emissions do not show any general tendency. Unburned ethanol and acetaldehyde emissions were increased by using ethanol and soot formation was reduced [23].

2.2 Methanol and Biodiesel as an Alternative Fuel

Alcohols are very popular alternative fuels for diesel engines. Both ethanol and methanol are used as alternative fuels. And also biodiesel is a very common alternative fuel. Several studies were performed with not only adding only ethanol, methanol and biodiesel to diesel fuel but also with their mixtures to diesel fuel.

Chunde Y et al. (2008) performed experiments using methanol-diesel fuel. In this paper, a compound combustion system for diesel methanol (DMCC) used on a naturally aspirated engine with and without the oxidation catalytic converter. As a result of these experiments using DMCC method with oxidation catalyst helps reduce all emissions emitted from engine like soot, NOx, CO, and THC [24].

Chueung C. S et al. (2008) studied with biodiesel as an alternative fuel and used biodiesel with fumigation method instead of blending to investigate the effects of emissions and performance. Constant speed variable load tests were performed with 4 cylinder naturally aspirated engine. Test results showed that no significant change were observed in carbon dioxide (CO_2) emission and brake thermal efficiency, while carbon monoxide (CO) and hydrocarbon (HC) emissions were increased NOx and particulate matter (PM) emissions were decreased. Also an increase in emitted nitrogen dioxide (NO_2) from the exhaust was recorded. Chueung C. S. et all reported their comparative test results with using fumigation technique and direct blending. Test results showed that using fumigation method all emission levels reduced much more than direct blending [25].

Sayın (2010) performed a study using ethanol and methanol in the same engine. He summarized the advantages of alcohols as their lower viscosity, less emissions due to their high H/C ratio and high oxygen content and lower sulfur content. Different kinds of fuels were prepared and to prevent the phase separation a mixer was mounted on the fuel tank and also an additive was used. After performing constant load variable speed tests the results were reported. According to results it is clear that methanol has much more effect to reduce CO and THC emissions than ethanol. NOx emissions are very dependent to combustion temperature and Oxygen ratio. Methanol has higher oxygen content, and lower cetane number than ethanol. Due to these specifications of methanol, increase in NOx emissions is more than that of ethanol and diesel. The BTE decreases while the methanol and ethanol content increase. The lowest BTE is obtained with methanol due to its Lower Heating Value [26].

CHAPTER 3

EXPERIMENTAL SET-UP AND MEASURING SYSTEMS

In this chapter experimental set-up will be defined beside a brief description of the measurement equipment. All the experiments were performed in the engine test laboratory in the Türk Traktör Factory (TTF). The TTF engine test Laboratory contains a complete system for measuring all the parameters relating to the diesel engine performance and exhaust gas emissions analysis. The experimental set-up contains mainly a dynamometer to load the engine, a gaseous emission sampling system to measure gaseous emissions and a dilution tunnel for particulate sampling and measurement with a complete data acquisition system to record all measured data. Figure 3.1 gives a schematic layout of the experimental system used.

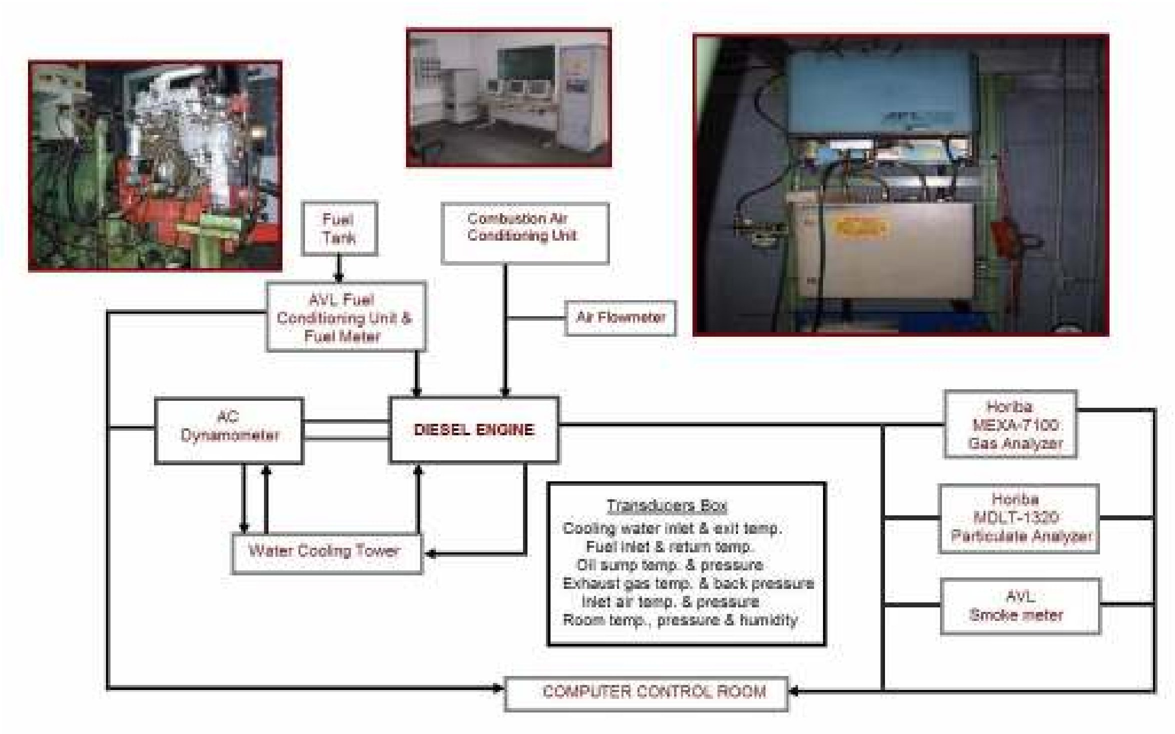


Fig 3.1 Schematic Design of the Experimental Mechanism

3.1. The Test Engines

Two different diesel engines out of the 8000 Series tractor engines produced in TTF were used for all performance and emission tests. The specifications of these two engines are given in Table 3.1. The main differences of these two engines are turbocharger unit. Two closer ranged engines were selected to observe the turbo unit effects on emissions and particulates. The tested engine is normally installed with all the required accessories needed for proper operation like the inlet air cleaner, the starter motor, the cooling water system, etc.

Fig 3.1.1 A general view of the engine tested in the TTF Engine Laboratory

Table 3.1 Specifications of test engines

Engine	TTF 8000s Engine	TTF 8000s Engine
Model	75HP NA	85HP TC
Type	4 Stroke	4 Stroke
Cylinder Number	4 Cylinder	4 Cylinder
Combustion Chamber	Direct injection	Direct injection
Bore	104 mm	104 mm
Stroke	115 mm	115 mm
Volume	3908 cm3	3908 cm3
Compression Ratio	18:1	18:1
Nominal Revolution	2500 rpm	2500 rpm
Max. Power	55 kW (2500 rpm)	63 kW (2500 rpm)
Max. Torque	340 Nm (1500 rpm)	340 Nm (1500 rpm)
Cooling System	Water cooled	Water cooled
Fuel Pump	BOSCH Distributor type	BOSCH Distributor type
Weight	430 kg	430 kg

3.2. Description of the Measuring Systems

3.2.1. Engine Performance System Measuring equipment

Engine performance testing system consisted primarily of the test engine coupled with a electronic brake dynamometer. Other components of this system can be defined as fuel consumption measuring device, air flow meter and other devices mounted to setup like air filter and fuel tank and their connections.

3.2.1.1. The Dynamometer

An electrical dynamometer that includes a rotor mounted for rotation on a shaft and a stator surrounding the rotor was used to measure power and torque directly from the crankshaft or the flywheel of the engine. An electronic dynamometer brake control system ensured to measure and to control the engine speed and torque for the performance tests. The technical specifications of the dynamometer are given in Table 3.2.

Table 3.2. Technical specifications of the dynamometer

Brand	Mc Clure
Model	GD 4000-AC 215
Production Date	1997
Type	Water cooled AC motor
Maximum Power	200 kW (4500 rpm)
Maximum Torque	450 Nm
Range	0 – 4500 rpm

The dynamometer comprises a high speed, low inertia, water cooled AC electric motor, mounted in low friction rolling contact trunnion bearings and torque reacted by a strain gauge load cell. The motor is powered by a flux vector AC inverter derive giving full four quadrant control. The motor is rated at 0/200/200 kW; 0/4500/8000 rev/min, as shown on the Figure 3.3 performance curve. Maximum torque is 450Nm. The dynamometer is cooled by water and operation with insufficient water or without water will cause severe damage. Because of this machine is protected by a water temperature switch and a water pressure switch. These safety devices should not be bypassed.

A pair of load cell calibration arms is provided, each having an effective radius of 1019.72 mm. Thus an applied load of 10 kg induces a torque of 100Nm.

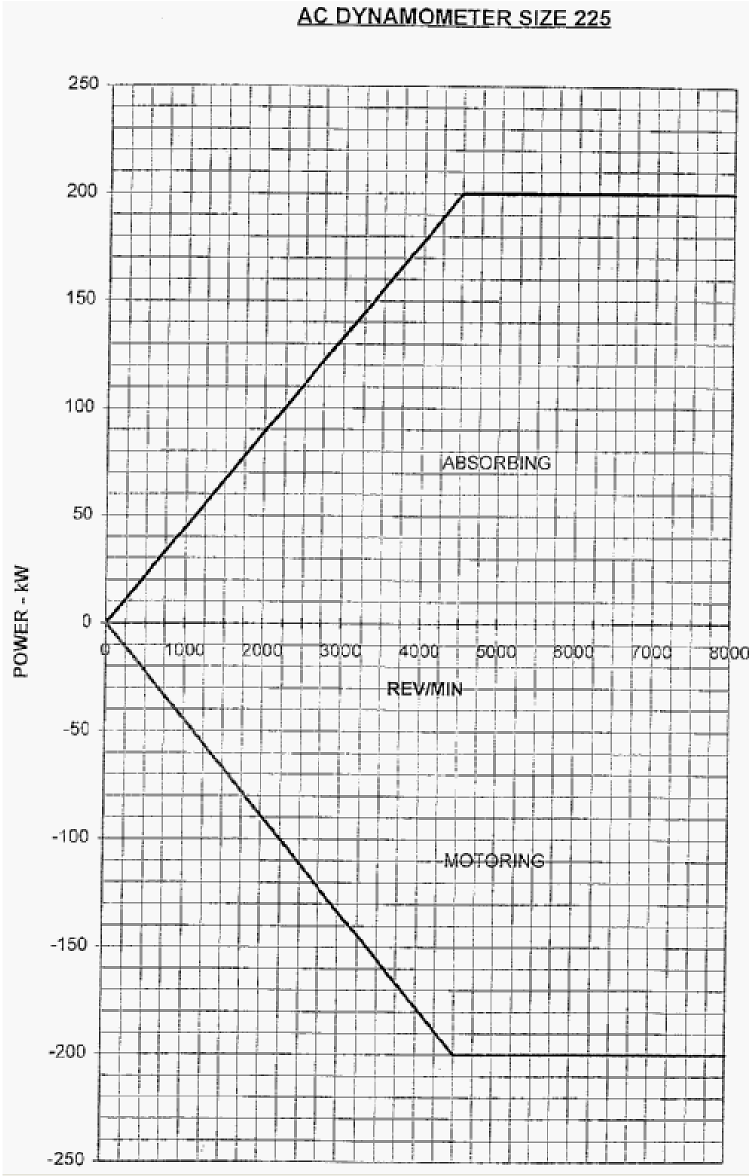


Fig 3.2.1 Performance Curve of the Dynamometer

Two engines were tested from idle to their maximum speeds to collect data and to show the effects of blending ethanol to their performance. During all tests variable load is applied by the dynamometer to measure the engines performance. This system was connected to a computer to record the applied braking torque and to calculate the power output of the engine. A load cell and rpm speed sensor is used to collect these data.

Maximum capacity or allowed speed of used dynamometer was 4500 rpm and maximum torque is defined as 450 Nm. But for this study only 2500 rpm was used due to the maximum speed of the engine.

Fig 3.2.2 Engine Connection of the Dynamometer

3.2.1.2. The Fuel Consumption Measuring Device

Fuel consumption measuring device was mounted between fuel tank and fuel pump for experimental set-up. As seen from Figure 3.5, measuring tank is suspended on a balance system. Fuel consumption of the engine is measured by using the weight loss data of the tank related to time.

For this fuel flow measurement device, while fuel mass is measured, measuring accuracy is not effected by fuel density.

The measuring tank is mounted with a return connection of the engine and venting line. Fuel consumption measuring device has a system with a sensitive sensor that records the registers the weight of the measuring tank and calculates fuel consumption. This system calculates the fuel consumption with an accuracy of

0.15% at 25g or even 0.12% at 75 g as defined in the standard (DIN1319). And also a calibration system is integrated with this device [27].

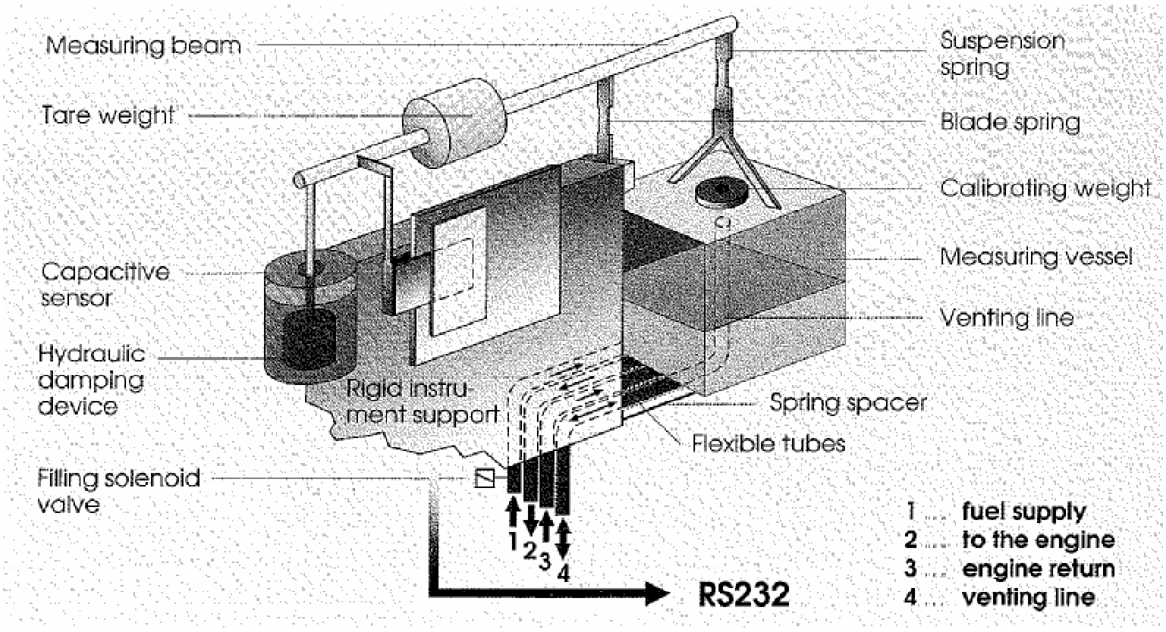


Fig 3.2.3 Fuel Flow measurements System [27]

The specifications of the device were given in Table 3.5. While its photo was given in Figure 3.5.

Table.3.3 Technical specifications of Fuel Consumption Measuring Device

Producer	AVL List GmbH / Austria
Brand	AVL
Type	733 S Dynamic Fuel meter
Range	0 – 75 kg/h

Fig 3.2.4 Fuel Consumption Measuring Device

3.2.1.3. The Smoke Meter

AVL 415 Variable Sampling Smoke Meter was used in this study to measure smoke level. Before explaining the measurement principle of this device, some definitions will be explained. The dry soot content of the exhaust gas emitted from the engine is measured by the help of smoke meter. Dry soot concentration can be monitored in three different unit as: in mg/m^3, in % pollution level or in FSN (Bosch Filter Smoke Number). The soot concentration means the soot content in mg/m^3 at 1 bar and 25°C. The soot is the measured value of the graphitic carbon contained and the soot content in mg/m^3 at 1 bar and 25°C is called as the soot concentration.

Paper blackening (P_B) is an important parameter for the measurement of the smoke. P_B is a measurement of the degree to which the filter paper is blackened. The reflectometer head measures the paper blackening. The measurement range can be defined as: The value for white filter papers measurement value is zero and completely blackened paper is 10. The linear graduation between black and white can be given as eq. 3.1.

$$P_B = 10.(1 - \frac{R_B}{R_W}) \qquad (3.1)$$

R_B = Power of blackened filter paper reflection
R_W = Power of white filter paper reflection

For the determination of the soot content therefore in addition to paper blackening, the volume of exhaust gas passed through the filter paper must be considered.

In accordance with the definition (ISO Draft 10054) the following applies:
$FSN=P_B$ for Leff=405 mm (length of column of gas at 1 bar and 25°C)
Effective length can be explained as, the length of the column of gas which is drawn through the filter paper. For other effective lengths, the FSN is calculated from the paper blackening and effective length of with the aid of a table which describes the relation between filter load (mg soot/m^2) and paper blackening. [28]

Pollution level (%)
$P_L = 100\text{-}11.5*R_A$ in %
R_A absolute lightness of sample can be calculated as,

R_A= 100*(reflectometer value of sample/ reflectometer value of standard whiteness value)

To determine the pollution level of the sample therefore the pollution level of the unblackened filter paper must be known and parameterized as the negative offset of the pollution level (%). [29]

The measuring method of the smoke meter can be summarized as;

- The smoke meters samples a volume of exhaust gas by means of a probe in the exhaust line and suck it through a clean piece of filter paper.
- The gas volume sucked through the filter paper is measured in a flow meter.
- The effective length is calculated
- The blackness of the filter paper due to soot (paper blackening) is determined by a reflectometer head

- The soot content of the exhaust gas is determined from the paper blackening and effective length
- Soot content displayed as FSN or soot concentration in mg/m3 or pollution level in %. [29]

3.2.1.4. The Transducer Box: Temperature and Pressure Measurement

The transducer box is mounted on the frame work above the engine drive shaft. The box is carried on rubber anti vibration mounts to isolate equipment which is sensitive to vibration.

There a number of quick-connect pressure and temperature sensors provided. These consist of a tee piece (Figure 3.7) carrying a thermocouple for temperature measurement and a flexible hose for pressure sensing. The thermocouple cable and the pressure hose are protected by a heavy duty plastic spiral wrap. The tee piece is fitted with the female half of a ¼". Mating male halves are either fitted to various engines rigging station.

The pressure sensing hose is connected to a pressure measurement transducer mounted in the transducer box. The thermocouple cable terminates in a standard connector which is plugged into the transducer box. The fuel inlet has provision for temperature measurement only and not pressure measurement.

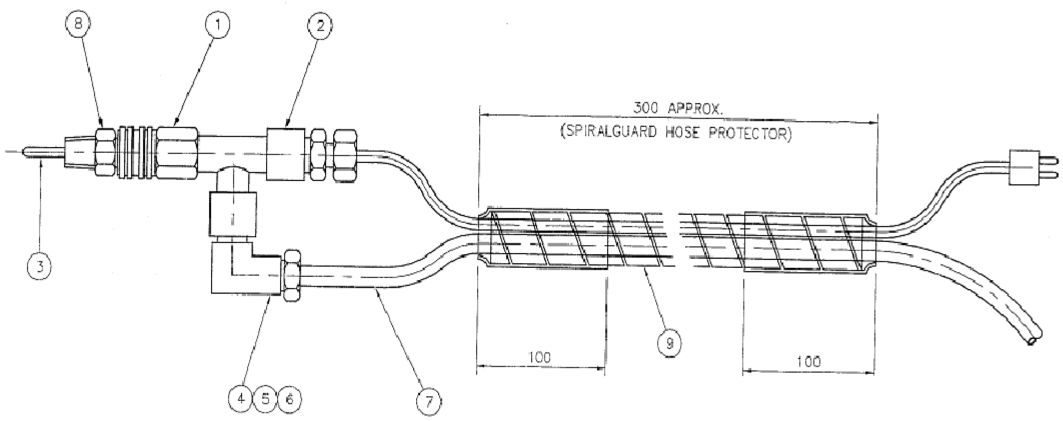


Fig 3.2.5 Temperature/Pressure Fitting

And also pt-100 temperature detector is used to measure room temperature, oil temperature, fuel temperature and air input temperature a Ni-Cr-Ni high temperature detector is used to measure engine water input and output temperatures (0-100^oC) exhaust temperature (0-100^oC)) a pressure detector is used to measure oil pressure, exhaust return pressure and air input pressure and a barometric pressure detector is used to measure the pressure of testing room (800-1200 mbar) relative humidity detector is used to measure for the relative humidity of the testing room (0-100%))

3.2.2. Exhaust Gaseous Emissions Measuring System

Emissions of the gaseous components were measured using gas analyzers Horiba, MEXA 7000. Concentration determination of the gaseous samples is performed as defined in the standard (97/68/EC). The standard stated that, to measure CO and CO_2 level, non dispersive infra red (NDIR) type analyzer was used. This analyzer consists of a light source an interferometer and an IR detector. And also to measure HC emissions, heated flame ionization detector (HFID) type analyzer was used. Chemiluminescent detector (CLD) was used to measure NO_X emissions. Chemiluminescence analyzer measure the NOx emissions with help of chemical reaction that produces light [30].

3.2.3. Particulate Emissions Measuring system

Diesel Particle measurement is normally made using a suitable dilution tunnel system. The size and shape of these systems varies considerably according to the tested engine.

The concept of mini dilution tunnel where only a fraction of the raw exhaust gas is being diluted has become a practical solution to the sizing problem of the conventional large dilution tunnels. Many investigators used this concept and the results obtained agree to a large extent with the results obtained from a large dilution tunnels.

A Horiba, MDLT- 130 Mini Dilution Tunnel used in this study for particulate emission measurement. This measurement system consists of following units: Main control unit, mini dilution unit and particulate sampling unit as seen from Figure.3.11

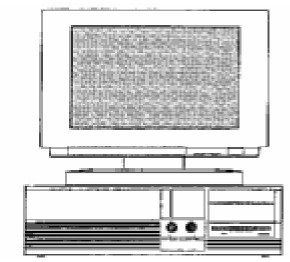
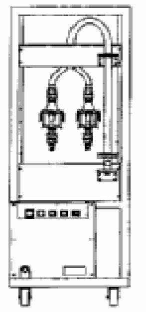
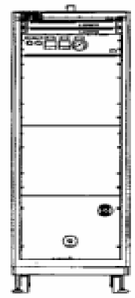
Main control unit	Mini dilution unit	Particulate sampling unit
		

Fig 3.2.6 Dilution Tunnel Control Units

Main Control Unit, is the central control unit which controls the tunnel and outputs various types of data. User can define various types of commands and conditions using buttons on the screen communication. Function of Mini Dilution Unit can be specified as, by filtering total dilution exhaust gas which is a compound of the sample gas collected from the engine exhaust pipe and the dilution air to collect PM samples. And the third unit is the particulate sampling unit, mounts basic facilities and also behaves as a conditioning unit. These basic facilities are, a pump unit to collect dilution sample at constant flow rate, the dilution air blower which increases or decreases flow rate of sample in proportion to the flow rate of the exhaust gas from the engine, the venture flow meter which measures flow rate of the dilution sample exhaust gas and the dilution air and the power supply unit of the system.

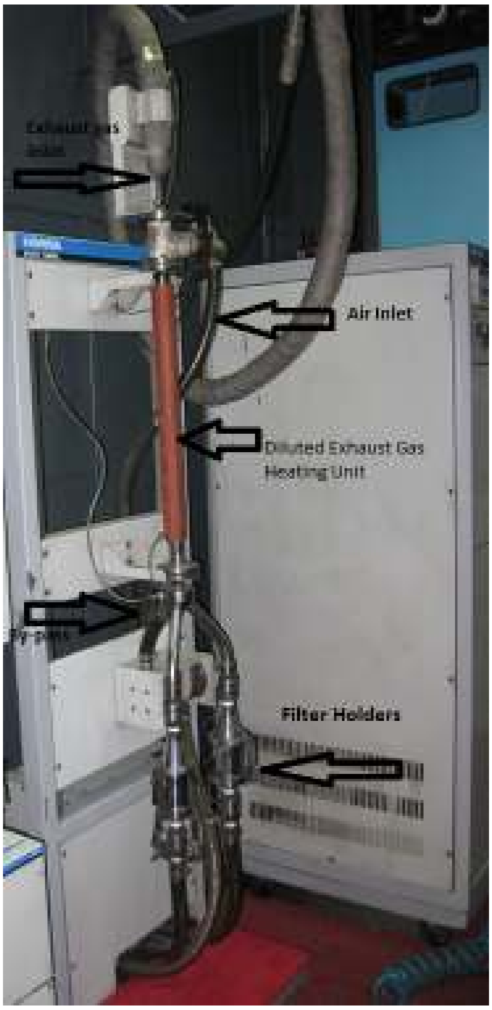


Fig 3.2.7 Dilution Tunnel

Dilution tunnel comprises a flow controller, a vacuum pump, a shutoff valve, and two filter holders. Each filter holder can hold two Ø70 mm filters. The filters specifications defined in section 3.2.3. In this study two different filters, having different outer diameters, are used so a new part is designed and produce to hold smaller filter, as seen from the Figure 3.13 (a).

Two sets of filter holders can be quickly connected and removed from the sampling system, thus allowing changing the filters for every mode easily with multiple filter method as seen from the Figure 3.13 (b).

The replacing procedure for the filter element can be defined as: to disconnect the coupler junction, to remove the couple junction and place the filter unit on the filter stay, to release the four locks to replace the filter element.

Fig 3.2.8 (a) Filter holder and spacer for smaller filters (b) Filter Holder

During an emission test, the shutoff valves are opened. Exhaust gases are passed through the bay-pass passage during the mode but only last 2 minute of the mode, gases are passed through their respective filter. The filtered gas then passes through the flow conditioning unit before passes through the filters. The flow conditioning unit, OVN, maintain a constant flow and a constant filtering temperature for exhaust gas throughout the emission test. And also in order to calculate the particulate mass concentration volume flow rate is recorded during the emission test.

Particulate dilution system collects diluted internal combustion engine exhaust gas in a filter at a constant flow rate, after the exhaust gas is thoroughly mixed with clean air from a dilution tunnel. In this system some portion of exhaust gases from exhaust pipe are diluted with the air from environment. The dilution ratio for this study is ¼ for exhaust gases. Air and exhaust gases are mixed and than they passed through a tunnel to prepare their sampling conditions. Partial exhaust dilution system is used.

Isokinetic Sampling

The basic requirement for exhaust gas sampling is that the sample must be represent of total gas stream successively which means that, size distribution and the particulate concentration should correspond to that of the main engine exhaust. For isokinetic sampling systems, gas velocities are the same for inlet of sampling nozzle and main gas stream.

An important behavior that should extract to a representative sample is an inertial behavior of the aerosol particles of PM. If, (A_n) is the nozzle tip opening area and (Q) is sample volume flow rate the velocity $V_n=Q/A_n$ must be equal to gas stream velocity (V_s) for the sample. Isokinetic or equal velocity sampling constraint can be defined as $V_n = V_s$. Simply this constraint can be explained as, for isokinetic sampling; the velocity of the exhaust gas entering the sampling probe is equal in velocity of the main engine exhaust stream.

The equal velocities are very important parameter because, if the suction is too fast, it will create a convergent gas stream at the nozzle face and also with this stream excessive amount of smaller particles entering the probe and also, if the suction is too slow, while the heavier particles continue into the probe because of their inertial behavior, smaller particles follow the deflected gas stream and do not enter the probe as seen from the Figure 3.14

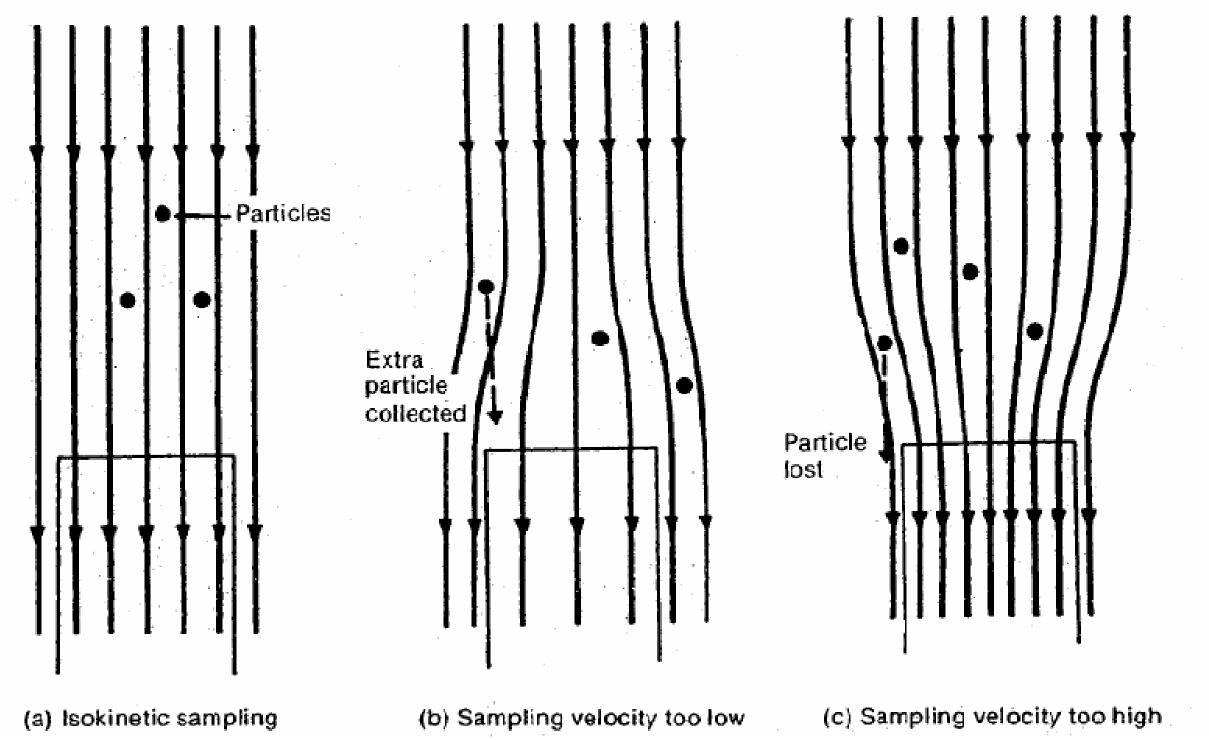


Fig 3.2.9 Isokinetic Sampling System [30]

There are some additional devices or equipments are necessary for the determination of particle emissions with dilution tunnel:

1. Sampling filters: Two types of filters are used for this study. Fluorocarbon based membrane filter for 75 HP NA engine and Glass Microfiber Filters for 85 HP TC engine. Specifications of these filters are:

* Type	: Teflon-coated fluorocarbon based membrane filters
* Manufacturer Company	: Pall German Sciences
* Diameter	: 70 mm

* Type	: Glass Microfiber Filters
* Manufacturer Company	: Whatman
* Diameter	: 70 mm

2. Weighing cabinet:

The temperature and the relative humidity of the cabinet in which particle filters stabilization and weighing procedures are carried out are kept in a specific range.

3. Analytic balance:

* Type	: M5P Electronic scale
* Manufacturer Company	: Sartorius / Germany
* Range	: 0 - 3000 mg

3.3. Data Acquisition System

The computer software named Task Master 2000 is composed by RICARDO Test Automation Ltd/England is used for this study. Using this software all tests are performed, all data are collected and all experiments are controlled. Figure 3.15 shows the user interface of this program.

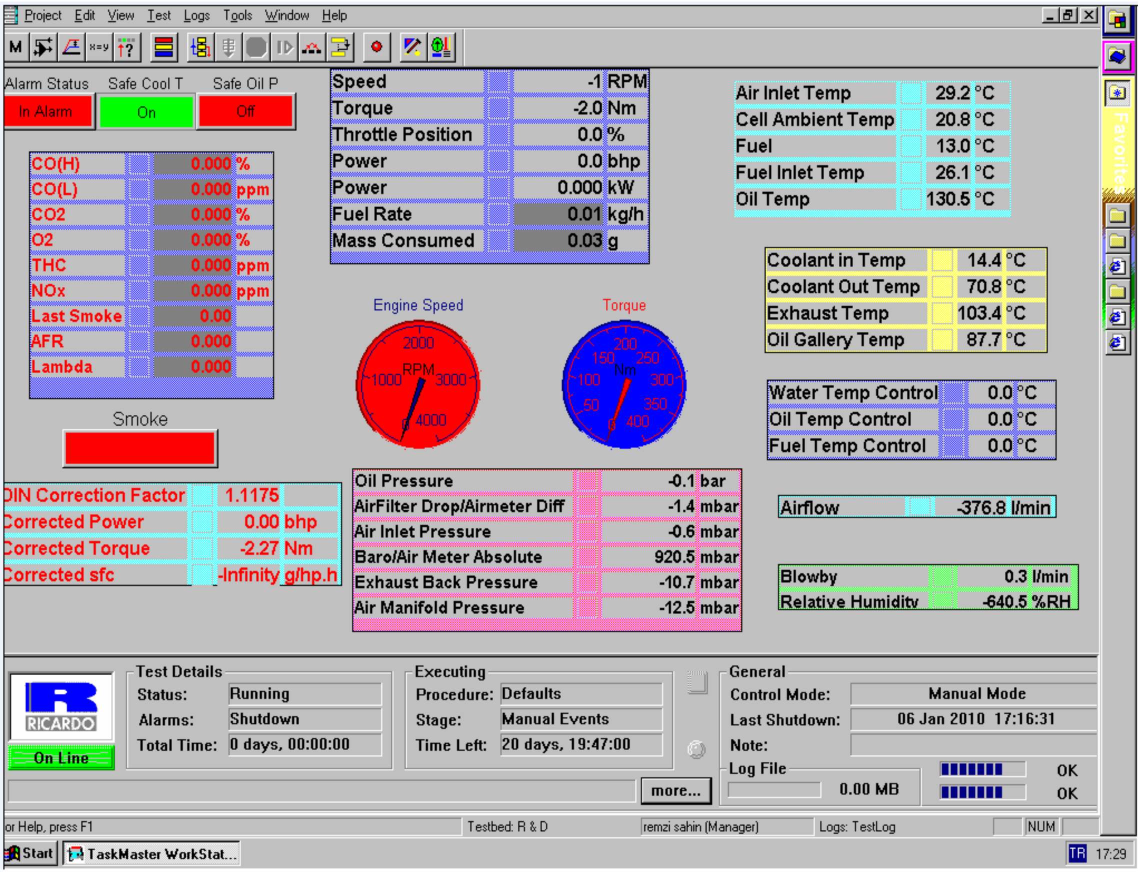


Fig 3.3.1 Screen shot of data acquisition program

During the experiments, interface of the software shows, all measured and calculate parameters. These parameters are oil fuel and coolant temperatures, air and oil pressures, speed, fuel rate, power, torque and corrected power and torque values. And also some other information can be monitored in the software like, exhaust back pressure and blow-by which are not used in this study. This computer program is set according to 8-mode test. All modes durations and power and loads can be monitored during the tests. Engine bench also can be used manually.

3.4. Thermal Gravimetric Analyzer (TGA) System

Thermo gravimetric Analyzer or TGA measures the change in temperature in relation to changes in weight. It can be said that three basic measurements performed with this device: weight, temperature, and temperature change. As a result of these measurements, weight loss curves for samples are obtained.

The weight changes as a function of time or temperature can be measured by Thermo Gravimetric Analyzer in a controlled atmosphere.

The analyzer usually consists of a high-precision balance with a pan (generally platinum) loaded with the sample. (Figure 3.16)

Fig 3.4.1 Pan loaded with sample

The pan is placed in a small electrically heated oven with a thermocouple to accurately measure the temperature. The atmosphere may be purged with an inert gas to prevent oxidation or other undesired reactions. A computer is used to control the instrument [31].

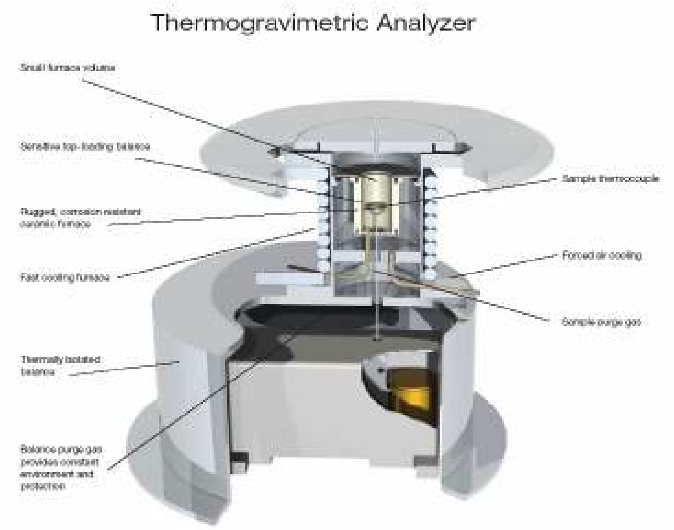


Fig 3.4.2 TGA

Analysis is carried out by raising the temperature gradually and plotting weight vs. temperature. The temperature in many testing methods routinely reaches 1000°C or greater, but for this study max temperature was 750°C used. After the data is obtained, curve smoothing and other operations done such as to find the exact points of inflection.

3.4.1. Sample preparation and TGA measurement

Analyzing steps can be defined by user interface as seen from the Figure 3.18. For this study, analysis internal was defined from 30° C to 750°C. The heating rate was defined as 5 °C/min for 30 °C to 250°C and 20 °C/min for 250 °C to 750°C using this interface. The inert gas (N_2) was purged through the TGA at 20ml/min.

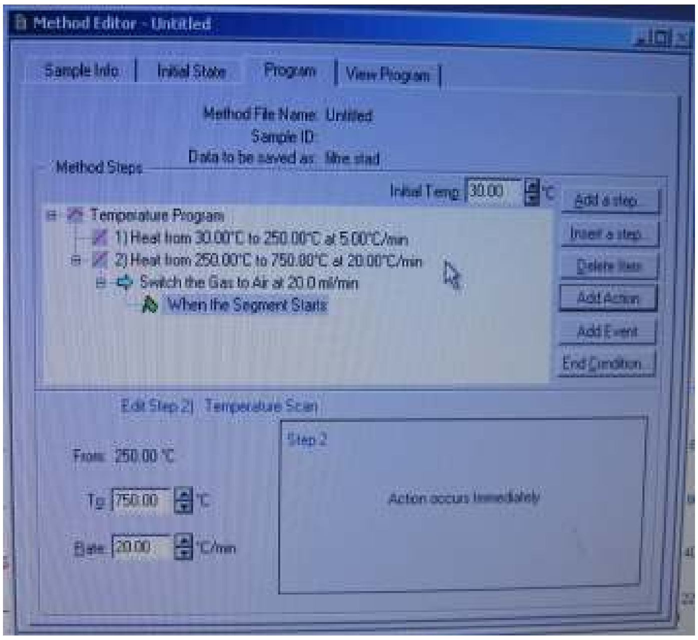


Fig 3.4.3 TGA user interface

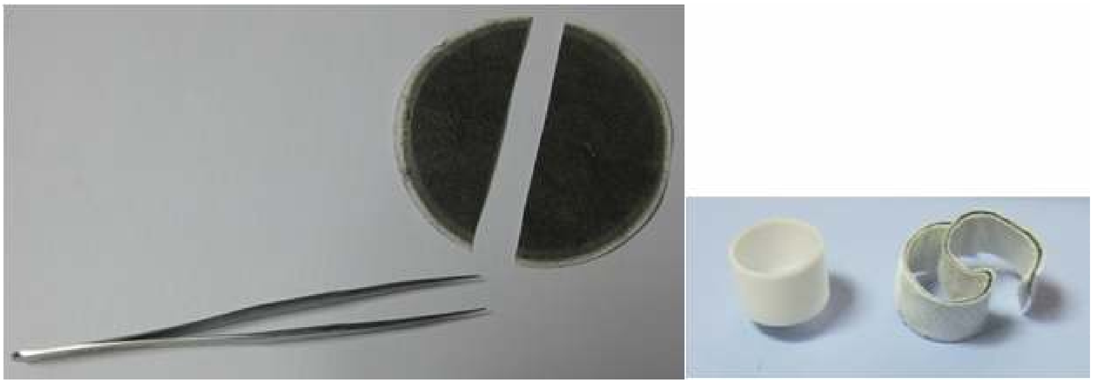


Fig 3.4.4 Sampling methods for TGA

To obtain information about the diesel particulate matter composition it is necessary to know the thermo gravimetric behavior of the blank filter it means that selected analysis method has to be used for blank filter.

Selected method to describe particulate matter by TGA can be explained with the Figure 3.19. After collecting particles in the dilution tunnel, the filter removed from the filter holder of the tunnel and a small sized portion from the middle of filter is cut and placed into sample holder of TGA.

CHAPTER 4

EXPERIMENTAL METHOD AND CALCULATION

This chapter describes the method of determining emissions of gaseous and particulate pollutants from the engines and performance parameters of the engine being tested. To inspect the effects of ethanol to diesel fuel two different types of tests are applied to the engines. Two test cycles are described in this chapter. The first one can be called as performance test which is constant load variable speed test and the second one can be called as emission test which is constant speed variable load test. Performance and emission tests procedures are explained step-by-step in this chapter.

General calculations made by evaluating the data regarding the performance and emission values obtained from the experiments carried out with both diesel and ethanol mixed diesel were given in this chapter. Torque values are automatically saved using the detector in electronically braking system. Power, specific fuel consumption, and total efficiency are calculated using the data that is obtained.

Since this work was completely an experimental work, normally, for all tests there are some unavoidable sources of error due to the nature of the measuring devices. For example, a pronounced and important error was observed in the fuel consumption rate values obtained from AVL 733 fuel consumption rate measuring device. Due to the automatic nature of the measurement technique that is used in this device, this device has a big chance to record wrong values and this may be reflected in some results that can not be explained. To compensate for the systematic errors that may appear in the calculated parameters, error analysis was performed to evaluate the average probability of the error for all measurements and given in Appendix F.

The first tests, for all performance and emission tests are made by using reference diesel fuel in order to determine the performance and emission values of the engine with only diesel and to be able to compare and to understand the effects of ethanol. After the first experiment is completed the second experiment is carried out by blending 5% ethanol into diesel. The following codes are used for diesel and ethanol mixed diesel as they are thought to provide ease for the graphical illustration of experimental data:

Diesel	: E0
Diesel + 5% Ethanol	: E5
Diesel + 10% Ethanol	: E10
Diesel + 15% Ethanol	: E15
Diesel + 20% Ethanol	: E20

4.1 Experimental Method

4.1.1 Properties of Tested Fuel

In the experiments diesel and a mixture of ethanol-diesel are used. Ethanol is provided by Delta Kimya from Ostim Organized Industry. The specifications of the fuels used in the experiments are given in Table 4.1.

Table 4.1 Specifications of Diesel and Ethanol

Parameters	**Diesel**	**Ethanol**
Density (20 ^{o}C) (kg/m3)	830	788
Cetane number	50	5-8
Kinematic Viscocity (40 °C) (mm2/s)	2.6	1.2

Lower calorific value (MJ/kg)	43	26.8
Specific heat capacity (J/kg°C)	1850	2100
Boiling point	180-360	78
Oxygen weight %	0	34.8
Latent heat of evaporation (kJ/kg)	250	840
Stoichiometric air-fuel ratio	15	9

As explained in Chapter 1, ethanol has some advantages and disadvantages as a fuel blending with diesel. The presence of ethanol generates different physicochemical modifications on fossil fuel, notably reduces the cetane number, viscosity, and gross heat as well. The basic disadvantage of ethanol is lower miscibility especially at lower temperatures. Moreover, adding ethanol to diesel fuel can reduce lubricity and create potential wear problems in sensitive fuel pump designs. Ethanol possesses also lower viscosity and calorific value as seen from the Table 4.1, the latter imposes minor changes on the fuel delivery system to make the engine achieve maximum power. In this study no engine modifications or changes are performed for the test engines. Lower calorific value of diesel fuel is 43 MJ/kg, while ethanol has 26.8 MJ/kg. And also, all blends calorific values are calculated to find the thermal efficiencies of the blend fuels as seen from Table 4.2.

Table 4.2 Lower calorific value (MJ/kg) for all fuels

Fuel	**Lower calorific value (MJ/kg)**
E0	43
E5	42.19
E10	41.38
E15	40.57
E20	39.76

Another disadvantage of the ethanol is that ethanol has a very low cetane number that reduces the cetane level of the diesel– ethanol blend. It requires normally the use of cetane enhancing additives that improve ignition delay. Ethanol has much lower flash point than the diesel fuel and higher vapor formation potential in confined spaces. Thus, it requires extra precautions in its handling. Various techniques have been developed to make diesel engine technology compatible with the properties of ethanol-based fuels. Broadly speaking, they can be divided into the following three classes: (a) ethanol fumigation to the intake air charge, by using carburetion or manifold injection, which is associated with limits to the amount of ethanol that can be used in this manner, due to the incipience of engine knock at high loads, and prevention of flame quenching and misfire at low loads, (b) dual injection system that is not considered very practical as it requires an extra high-pressure injection system and a related major design change of the cylinder head, and (c) blends (emulsions) of ethanol and diesel fuel by using an emulsifier to mix the two fuels for preventing their separation, requiring no technical modifications on the engine side.

4.1.2 Fuel Delivery System

Various techniques have been developed to make diesel engine technology compatible with the properties of ethanol-based fuels. Broadly speaking, they can be divided into the following three classes: (a) ethanol fumigation to the intake air charge, by using carburetion or manifold injection, which is associated with limits to the amount of ethanol that can be used in this manner, due to the incipience of engine knock at high loads, and prevention of flame quenching and misfire at low loads, (b) dual injection system that is not considered very practical, as requiring an extra high-pressure injection system and a related major design change of the cylinder head, and (c) blends (emulsions) of ethanol and diesel fuel by using an emulsifier to mix the two fuels for preventing their separation, requiring no technical modifications on the engine side.

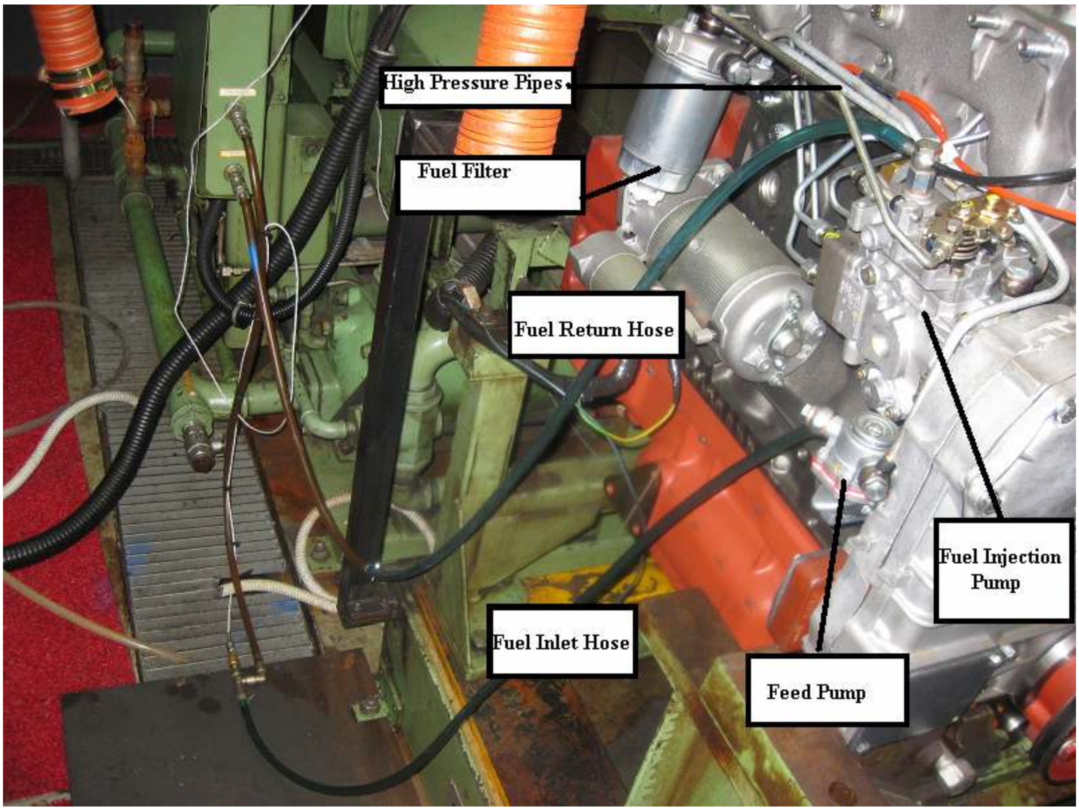


Fig 4.1.1 Fuel Delivery System

4.1.3 Performance Tests

Constant Load - Variable Speed test is performed to measure all performance parameters. For these tests recorded performance parameters are power, torque and fuel consumption.

Test Cycle can be explained step by step as follows:

- Preparing the blend for specified ethanol and diesel ratios and filling the fuel tank with the blend or only diesel.
- Checking the electrical connections of the engine
- Checking the fuel tank and fuel pump connections
- Checking the air filter connections

- Checking the water pump connections
- Turning the PC on and monitoring the software
- Operating the engine at idle speed for 5 minutes to warm the engine
- After five minutes test begins at 2500 rpm with full load.
- The engine speed is gradually reduced (10 rpm for every 5 minutes) by dynamometer until 1500 rpm (max Torque speed).
- Setting the throttle to full open throttle position .
- Operating the engine at max. speed and and max. load. (2500 rpm)
- Gradually decreasing the speed for 100 rpm for every 10 minutes until 1500 rpm .
- Power, torque, and fuel consumption values are recorded during the tests and brake specific fuel consumption and thermal efficiency are calculated using these results.
- Recording the engine speed, torque, fuel flow rate parameters for each mode.
- Controlling the oil, coolant, and fuel temperatures during the test.

Rotation speed is reduced by dynamometer by gradual loading while the engine is operated at full speed; all measurement values of engine rotations taken as reference are saved automatically in the computer.

4.1.4 Emission Tests

Two test engines are tested using a steady state 8-mode test cycle as ISO 8178 test requirements (97/68/EC). This test cycle is used world wide for off-road and industrial equipment. Test modes and weighting factors for this test cycle are given in Table 4.3 and Figure 4.2. The average integrated value for the 8-mode test cycle is calculated using the weight factor for each mode.

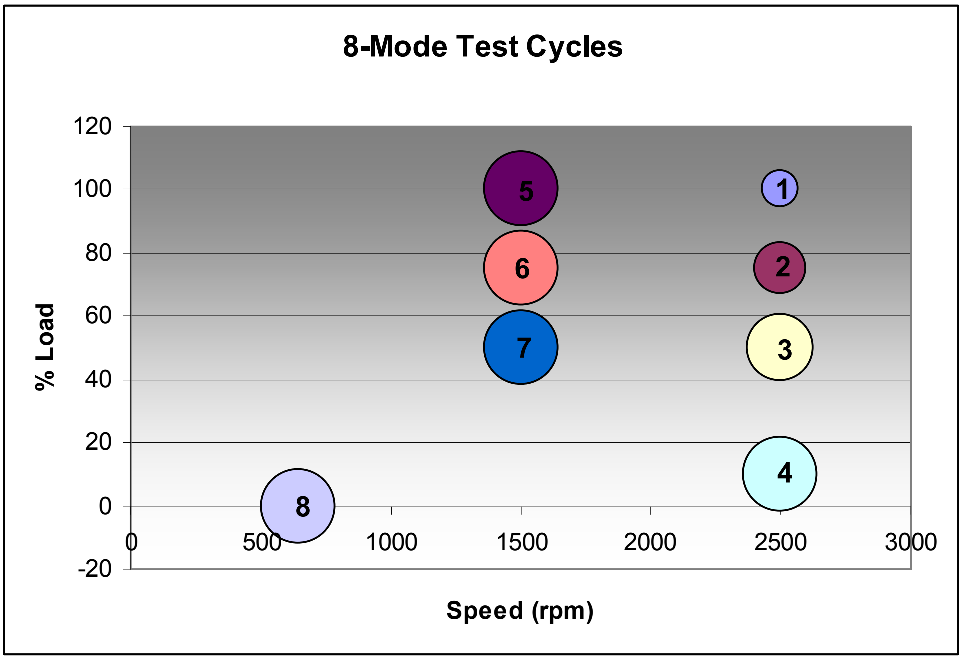


Fig 4.1.2 (8 Mode) Test Cycle According to (Directive 97/68/EC)

Table 4.3. 8 Mode Test Specification according to (Directive 97/68/EC)

Mod No	Speed (rpm)	Torque (%)	Weighting factor
1	2500	100	0.15
2	2500	75	0.15
3	2500	50	0.15
4	2500	10	0.10
5	1500	100	0.10
6	1500	75	0.10
7	1500	50	0.10
8	650	0	0.15

Constant speed - variable load tests are performed for all tests in order to measure emission parameters. Step by Step performance test procedure can be explained as follows:

- Preparing the blend for specified ethanol and diesel ratios and filling the fuel tank with the blend or only diesel.

- Checking the electrical connections of the engine.
- Checking the fuel tank and fuel pump connections.
- Checking the air filter connections.
- Checking the water pump connections.
- Checking the exhaust connections to emissions and measuring device and dilution tunnel.
- Turning the PC on and monitoring the software.
- Setting the throttle position.
- Running the engine at idle speed with no load for 5 minutes to warm the engine.
- Operating the engine at 2500rpm speed and max. Load. (%100)
- Gradually decreasing the load for every 10 minutes as %75, %50 and until %10 for first 4 mode.
- Setting engine speed at 1500 rpm and max load case. (%100) (Mode 5)
- Gradually decreasing the load for every 10 minutes as %75, %50 for Mode 6 and Mode 7.
- Recording the engine speed, torque, fuel flow rate parameters for each Mode.
- Changing the filters for all modes after collecting particulates at the last 2 min. for Mode 1 and 5 and last 1 min. for Mode 2,3,4,6 and 7.
- Controlling the oil, coolant, and fuel temperatures during the test.

4.1.5 Preparation of the Sampling Filters for Dilution Tunnel

At least an hour before the test each filter is placed in a closed, petri dish as seen from the Figure 4.3 and in a weighing chamber for stabilization. At the end of the stabilization period, each filter is weighed and the tare weight is recorded. The filter then, is stored in a closed petri dish or filter holder until needed for testing.

Fig 4.1.3 Each filter placed in a closed, Petri dish and in a weighing chamber for stabilization.

4.2 Calculation of Experimental Data

4.2.1 Calculation of Performance Parameters:

The simple calculation method to determine the performance and emission parameters are explained in detail below.

4.2.1.1 Effective Power

$$N_e = \frac{M_d \cdot n}{9550} \tag{4.1}$$

In the equation, the symbols refer to:

N_e : Effective power (kW)

M_d : Rotation momentum (Nm)

n : Motor rotation speed (1/min)

4.2.1.2 Specific Fuel Consumption

The data obtained from fuel consumption measurement device on specific engine rotation levels are used while calculating specific fuel consumption rate in the Equation 4.2.

$$b_e = \frac{B}{N_e} \tag{4.2}$$

In this equation the symbols refer to:

b_e : Specific fuel consumption (kg/kWh)
B : Fuel consumption per hour (kg/h)
N_e : Effective power (kW)

4.2.1.3. Thermal Efficiency

Thermal efficiency is stated as the rate of the effective power obtained in the engine from the energy of the fuel. It is determined as in Equation 2.3. considering the lower thermal values and fuel consumption rates of diesel and biodiesel.

$$\eta_t = \frac{3600}{b_e \cdot H_u} \tag{4.3}$$

In this equation the symbols refer to:

η_t : Thermal efficiency (%)
b_e : Fuel consumption per hour (kg/kWh)
H_u : Lower thermal value of the fuel (kJ/kg)

4.2.2 Emission Calculations

The exhaust gas flow should be determined as defined below:

$$G_{EXHW} = G_{AIRW} + G_{FUEL} \tag{4.4}$$

The emission mass flow rates for each mode shall be calculated as follows:

$$Gas_{mass} = u \text{ x } conc \text{ x } G_{EXHW} \tag{4.5}$$

Table.4.4. Values of the coefficients u-wet for various exhaust components

Gas	u	Conc
NOx	0.001587	ppm
CO	0.000966	ppm
HC	0.000479	ppm
CO_2	15.19	Percent

4.2.2.1 Calculation of the Specific Emissions:

The specific emission (g/kWh) calculated for all individual components in the following way:

$$Individual\,gas = \frac{\sum_{i=1}^{n} Gas_{mass,i}\, xWF_i}{\sum_{i=1}^{n} P_i\, xWF} \tag{4.6}$$

The weighting factors and the number of modes (n) used in the above calculation are according to Table. 4.4

4.2.2.2 Calculation of the Particulate Emission

The particulate emission calculated in the following way:

Humidity Correction Factor for Particulates

As the particulate emission of diesel engines depends on ambient air conditions, the particulate mass flow rate shall be corrected for ambient air humidity with the factor Kp given in the following formula

$$K_p = 1/(1 + 0.0133x(H_a - 10.71)) \tag{4.7}$$

where:

Ha: humidity of the intake air, gram of water per kg dry air;

$$H_a = \frac{6.220xR_a xp_a}{p_B - p_a xR_a x10^{-2}} \tag{4.8}$$

Calculation of the particulate mass flow rate: The particulate mass flow rate calculated as follows:

For the multiple filter method:

$$PT_{mass} = \frac{M_{f,i}}{M_{SAM,i}} x \frac{(G_{EDFW,i})_{aver}}{1000} \tag{4.9}$$

where i = 1, . . . n

The specific emission of particulates PT (g/kWh) calculated in the following way for multiple filter method:

$$PT = \frac{\sum_{i=1}^{n} PT_{mass,i} xWF_i}{\sum_{i=1}^{n} P_i xWF} \tag{4.10}$$

CHAPTER 5

EXPERIMENTAL RESULTS

In this chapter, engine performance and emission test results are presented with the help of detailed graphs. The measured data during all tests are given in Appendix A. Firstly, engine performance results and afterwards engine emission results are discussed for Naturally Aspirated and Turbocharged Engines.

5.1 Engine Performance Results

Engine Performance tests were performed on constant load (full load) and variable speeds from 2500 to 1500. Performance tests begun with 2500 rpm and the engine speed are gradually decreased to 1500 rpm. The engine performance at various speeds using diesel and ethanol-diesel blends is observed in terms of brake horsepower (BHP), brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) and smoke level. Performance parameters are measured during performance tests.

5.1.1 Engine Power

The engine power developed by the engine operating on reference diesel fuel and ethanol at various engine speed conditions is presented in Figure 5.1.1. The change in brake horsepower at various diesel-ethanol blend percentages can be observed from this graph.

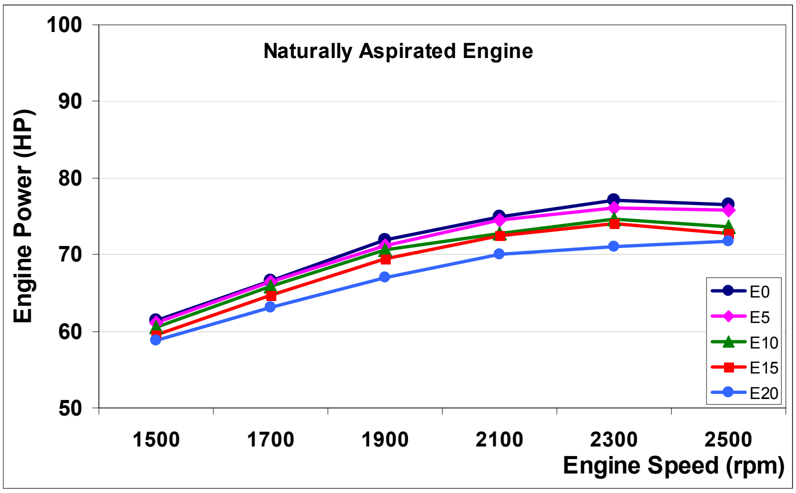


Fig 5.1.1 Engine power vs. speed for different fuel blends under full load condition for Naturally-Aspirated engine

The engine power developed by the turbocharged engine which was operated on the same conditions with naturally aspirated engine is presented in Figure 5.1.2. Comparing these two graphs, two different results can be obtained. First one is the effect of turbocharger on engine performance. It is apparent from the figures that by using turbocharger unit, the engine performance increases. The second result is about the ethanol effect on engine performance. Adding ethanol to diesel fuel for both turbo and naturally aspirated engines results in a decrease in engine power. This decrease level is much more in turbo engine than naturally aspirated engine.

This decrease in engine performance can be explained with lower energy level of ethanol. The lower energy level of the ethanol–diesel fuel blend causes some reductions in the engine power when it is used in diesel engines without any modifications. The heating value and density of ethanol are approximately 35% and 3.5% less than the values of diesel fuel, respectively [32]. And it proofs the fact that the reduction mainly depends on the percentage of ethanol. It can be seen from the figures that with increasing ethanol level in the blend, performance is decreased.

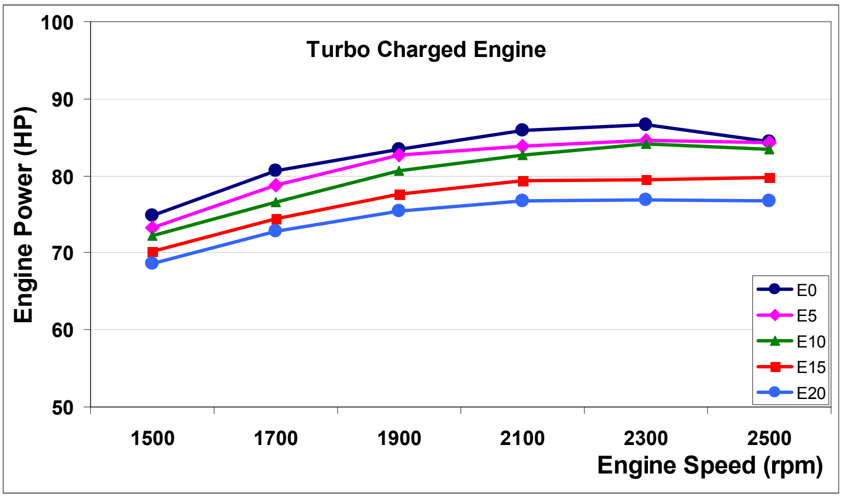


Fig 5.1.2 Engine power vs. speed for different fuel blends under full load condition for Turbo-Charged Engine

5.1.2 Brake Specific Fuel Consumption

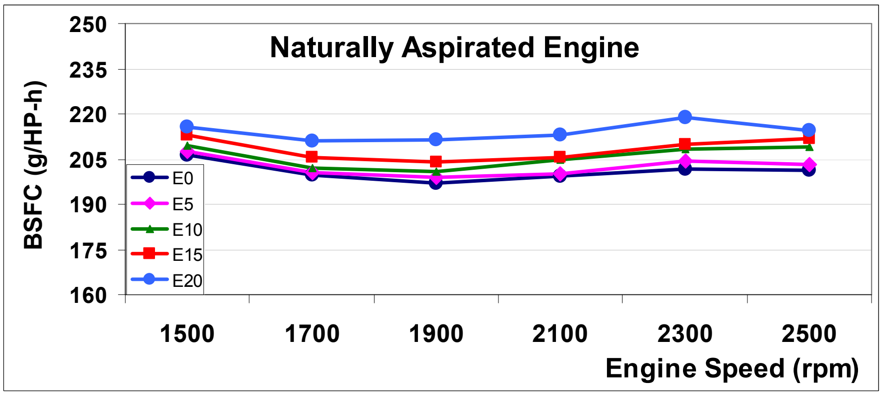


Fig 5.1.3 Engine BSFC vs. speed for different fuel blends under full load condition for Naturally-Aspirated engine

The brake specific fuel consumption is an important parameter to compare engines and a measure of fuel efficiency of an engine. The relationship between brake specific fuel consumption and speed for diesel-ethanol blends for naturally aspirated engine is shown in Fig. 5.1.3. The brake specific fuel consumption increases with an increase in engine speed and then it slightly decreases after 2300 rpm. For turbocharged engine the more the engine speed increases the more BSFC does as seen on the Figure 5.1.4. From the figures it can be said that adding ethanol to both turbocharged and naturally aspirated engines increases the fuel consumption. This result is an expected result due to the decrease in power output of the engines. Brake specific fuel consumption rate for an engine can be calculated using Equation 5.1

$$b_e = \frac{B}{N_e} \tag{5.1}$$

In this equation b_e refers to Fuel Consumption per Hour (kg/kWh), N_e refers to Effective Power (kW) and B refers to Fuel Consumption per Hour (kg/h). According to this equation, decrease in engine power leads to an increase in fuel consumption. This increment is much more for turbocharged engine than the naturally aspirated one. For both engines the brake specific fuel consumption has increased and the power output has decreased with the increase of ethanol content in the blended fuel at overall operating conditions. As explained before this is due to the fact that the lower heating value of ethanol is about 2/3 of that of diesel [33]. To increase the output power some engine modifications can be performed and also some additives can be used with ethanol.

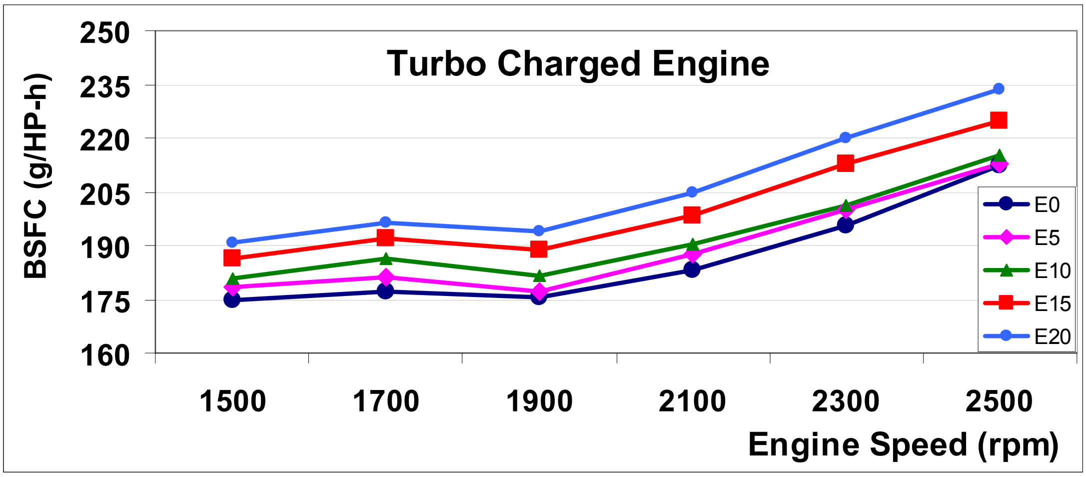


Fig 5.1.4 Engine BSFC vs. speed for different fuel blends under full load condition for Turbo-Charged engine

5.1.3 Brake Thermal Efficiency

The relationship between brake thermal efficiency and ethanol diesel fuel blends fuel is presented in Fig. 5.1.5. and Fig. 5.1.6 for Naturally Aspirated and Turbocharged engines respectively. It can be seen from the figures that for both engines, for the same load conditions the brake thermal efficiency decreases with increasing ethanol content in the blend.

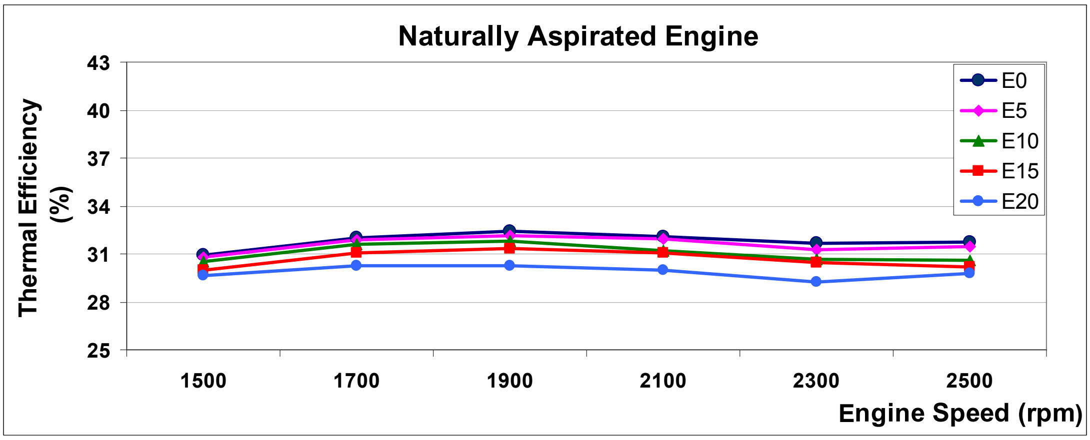


Fig 5.1.5 BTE vs. speed for different fuel blends under full load condition for Turbo-Charged engine

This decrease can be explained by the formulation of the BTE. It is similar to BSFC. BTE can be calculated from equation 5.2

$$\eta_t = \frac{3600}{b_e \cdot H_u} \tag{5.2}$$

In this equation η_t refers to Thermal efficiency (%) and, b_e refers to fuel consumption per hour (kg/kWh) and H_u refers to lower thermal value of the fuel (kJ/kg). It is well known that ethanol's lower heating value is about 2/3 of that of diesel and as seen from the figure 5.1.3 and 5.1.4 that BSFC increases with adding ethanol. As a result it can be said that the loss of heating value and the increase of brake specific fuel consumption (bsfc) using ethanol-diesel blends lead to a decrease in brake thermal efficiency.

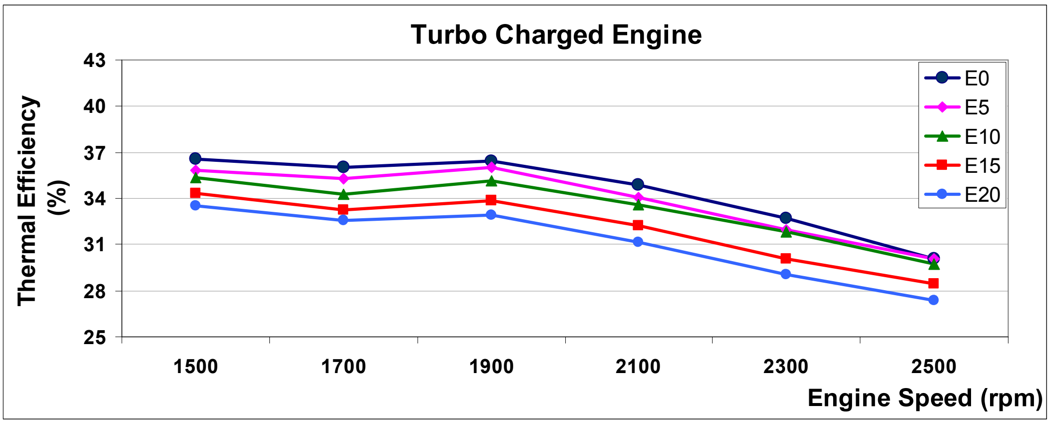


Fig 5.1.6 BTE vs. speed for different fuel blends under full load condition for Turbo-Charged engine

5.1.4 Smoke

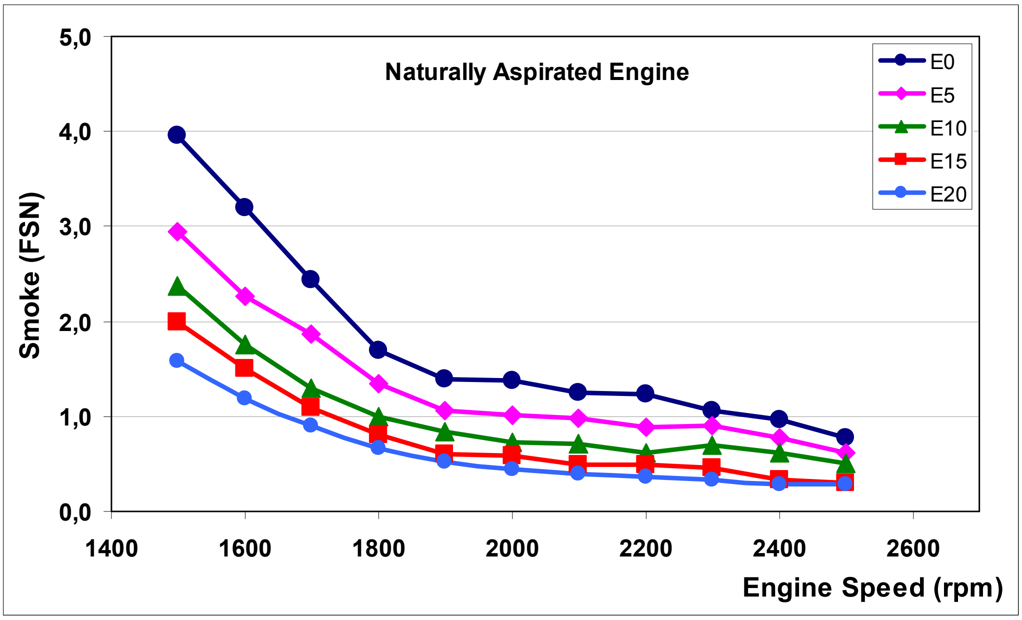


Fig 5.1.7 Smoke vs. speed for different fuel blends under full load condition for Turbo-Charged engine

Fig. 5.1.7 and Figure 5.1.8 show the emitted smoke level of the naturally aspirated and turbocharged engines respectively. It can be said that for both engines the smoke emitted by the ethanol–diesel fuel blends are significantly lower than the reference diesel fuel. This reduction increases with the increase of the rate of ethanol in the mixture.

This reduction can be explained by the presence of oxygen in the ethanol. Ethanol provides oxygen with rich zones in combustion chamber due to more effective combustion, smoke level is decreased. Adding ethanol to diesel reduces H/C ratio of the fuel and reduces smoke level. With more air provides by turbo unit smoke level reduces much more than Naturally Aspirated engine.

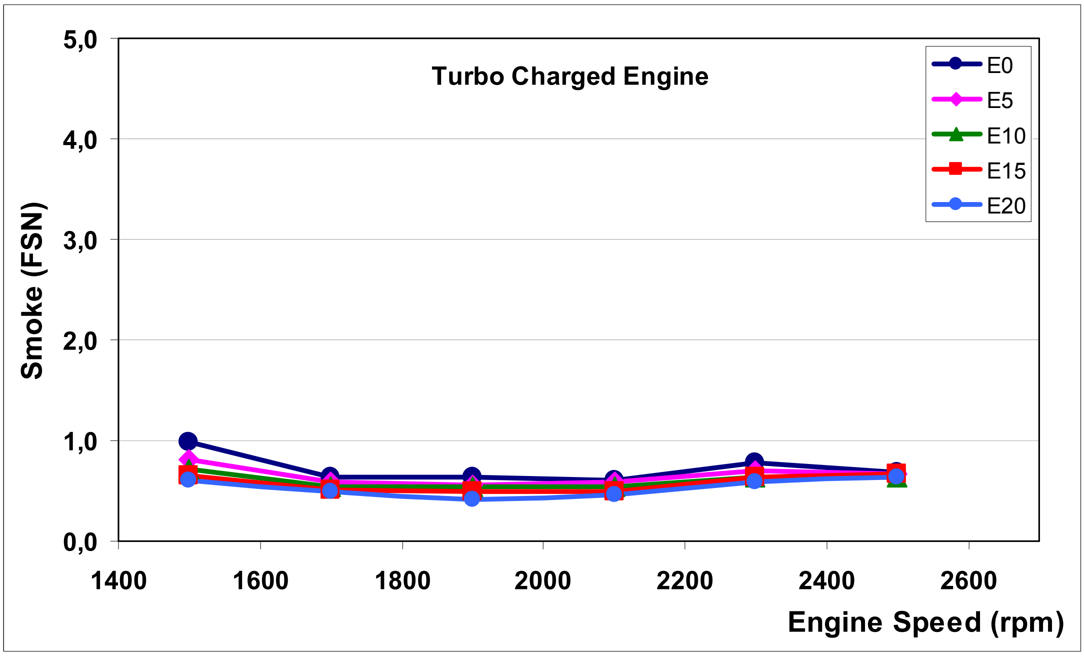


Fig 5.1.8 Smoke vs. speed for different fuel blends under full load condition for Turbo-Charged engine

5.2 Calculated measured emission levels (g/kW-h) according to ISO 8178

8-Mode test procedure is performed for each fuel, to measure non-road engine emissions on a steady-state test cycle that is the same as the ISO 8178, 8-mode steady-state test cycle. Two test engines are tested using a steady state 8-mode test cycle as ISO 8178 test requirements (97/68/EC).

5.2.1 CO Emissions

Figures 5.2.1 and 5.2.2 show the variations in the CO emission with diesel and ethanol as a function of the engine speed for NA and TC engine, respectively. Using ethanol with diesel fuel increases CO emissions for both NA and TC engines. CO emission increases with increasing ethanol level in blend. Main reason of this increase is the lower cetane number of the blended fuels induced an increase in the combustion delay. This delay was higher with the rise in ethanol content. The excessive overall combustion delay especially at low loads led to an increase in CO emissions.

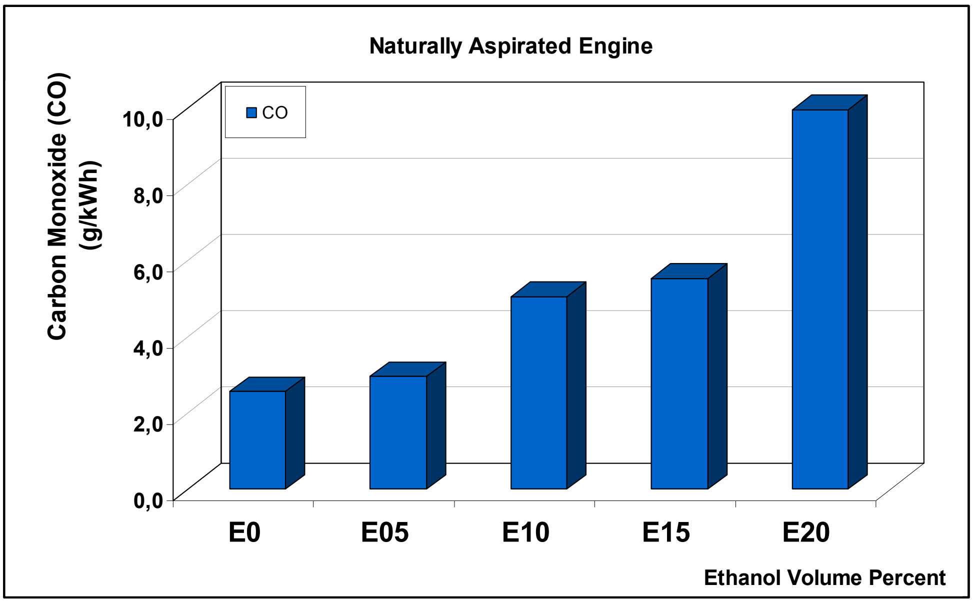


Fig 5.2.1 Carbon monoxide emissions for different fuel blends for Naturally Aspirated Engine

These figures clearly show the powerful reducing effect of TC unit on CO emissions. It is also observed that reduction in CO emissions is averagely 70% using TC unit with engine. This expecting result can be explained with the aim of Turbocharger unit. Turbocharger unit by increasing air density makes more oxygen sent to the intake manifold and this means more complete combustion period.

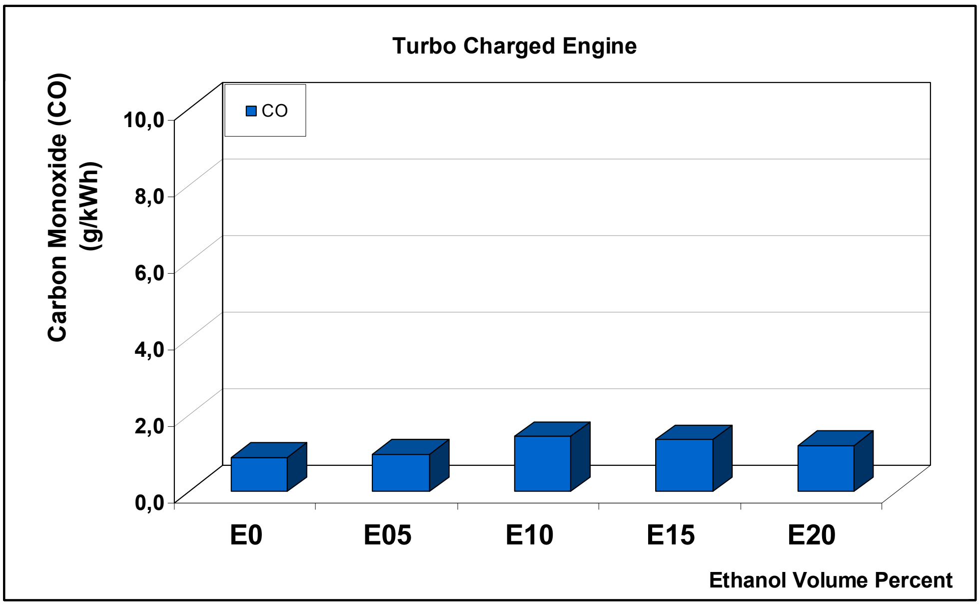


Fig 5.2.2 Carbon monoxide emissions for different fuel blends for Turbo Charged Engine

Effect of Turbocharger unit of CO emission can be observed in load vs emissions graphs in Figure 5.2.3 and 5.2.4. While max. CO emission level for naturally aspirated engine is measured as 2812 ppm, for Turbo Charged engine this value is measured as 757 ppm with 20% Ethanol and 10% load.

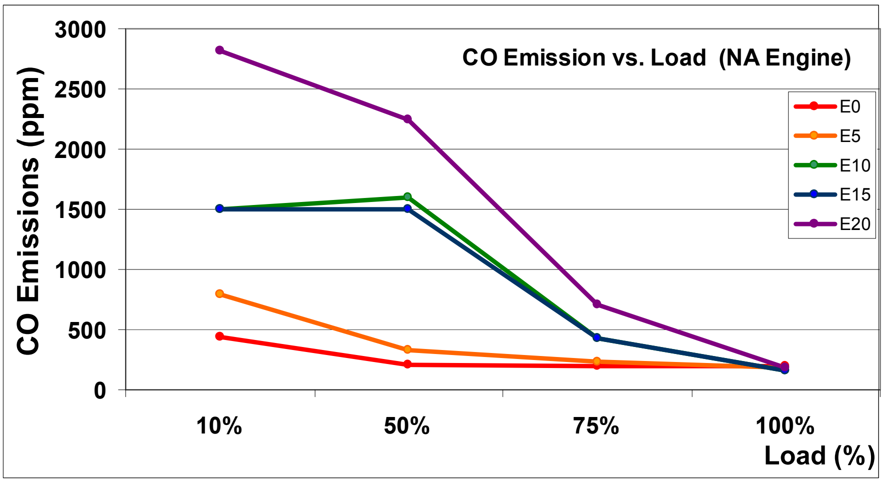


Fig 5.2.3 Carbon monoxide Emissions vs. Load for different fuel blends for Naturally Aspirated Engine

As mentioned before CO emissions are mostly independent from load or speed variations and dependent to fuel air ratio and oxygen content of the mixture. But it can be said that with the increasing load, lean mixture occurs and CO emission level will decrease. For lean mixtures Carbon molecules can find more Oxygen molecules to create CO_2 emissions.

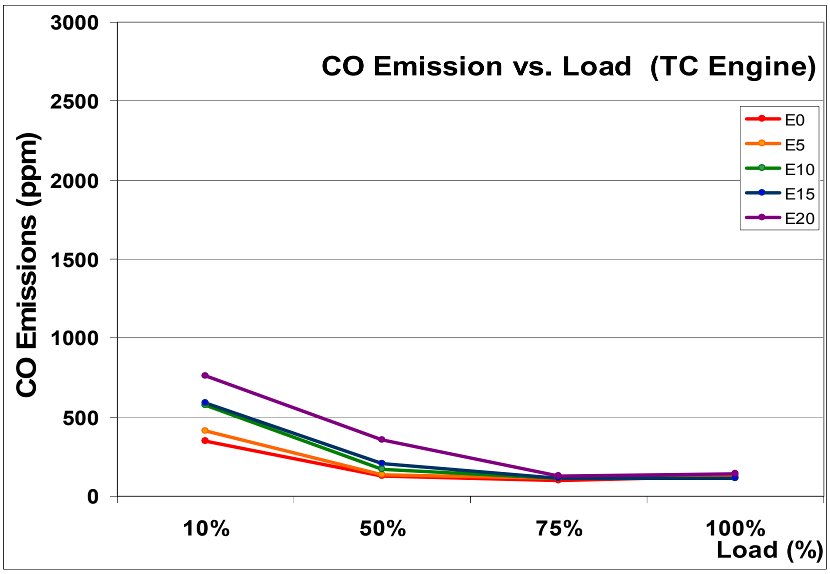


Fig 5.2.4 Carbon monoxide Emissions vs. Load for different fuel blends for Turbo Charged Engine

5.2.2 THC Emissions

Figure 5.2.5 and 5.2.6 show that the unburned hydrocarbon emissions are increased by the use of ethanol-diesel fuel blends comparing to diesel fuel and by the increase of emission being higher with the increase in the percentage of ethanol in the blend. This condition can be explained with higher heat of evaporation of ethanol. This causes slower evaporation and poorer fuel air mixing and this increases spray penetration and causes unwanted fuel impingement on the chamber walls. As a result of all these incomplete combustion occurs in the cylinder and THC emissions increase [14]. Other reason of the increase in THC emissions is the limited miscibility of ethanol with the diesel fuel. Due to the inhomogeneous ethanol-diesel blends leaner mixture is obtained in the chamber and results in more unburned fuels in combustion process [34].

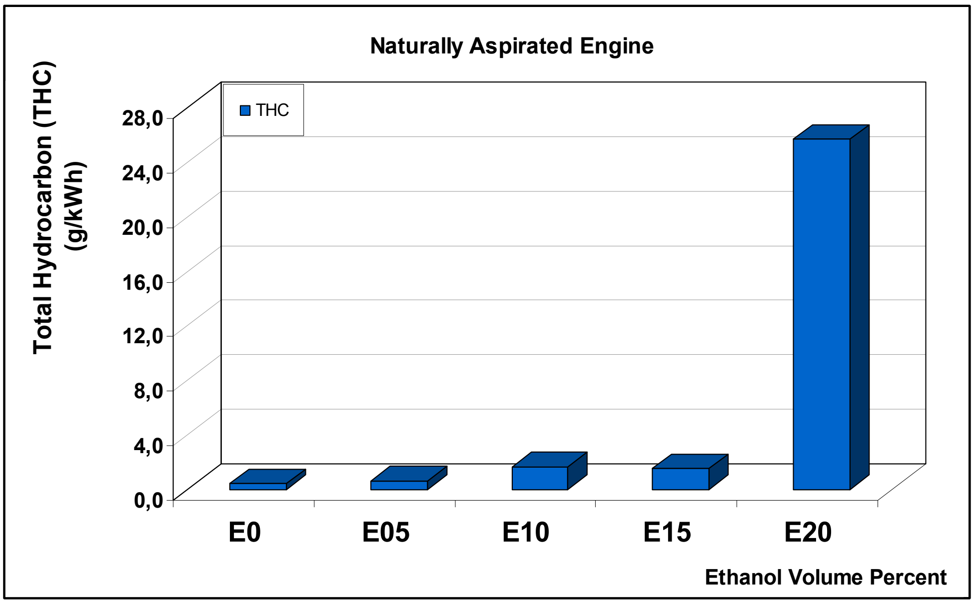


Fig 5.2.5 THC Emissions for different fuel blends for Naturally Aspirated Engine

Turbocharger unit feeds the engine with more air and as a result of this more complete combustion occurs. Figure 5.2.6 shows the results of this more complete combustion in terms of hydrocarbon emissions for turbo charged engine. It is apparent from the figure that THC emission with TC Engine is lower than NA Engine.

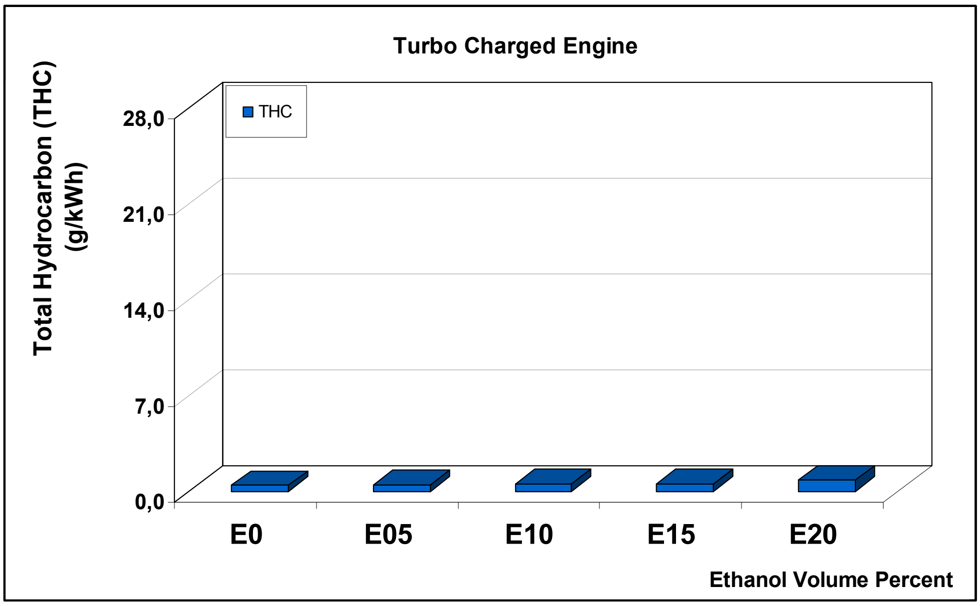


Fig 5.2.6 THC Emissions for different fuel blends for Turbo Charged Engine

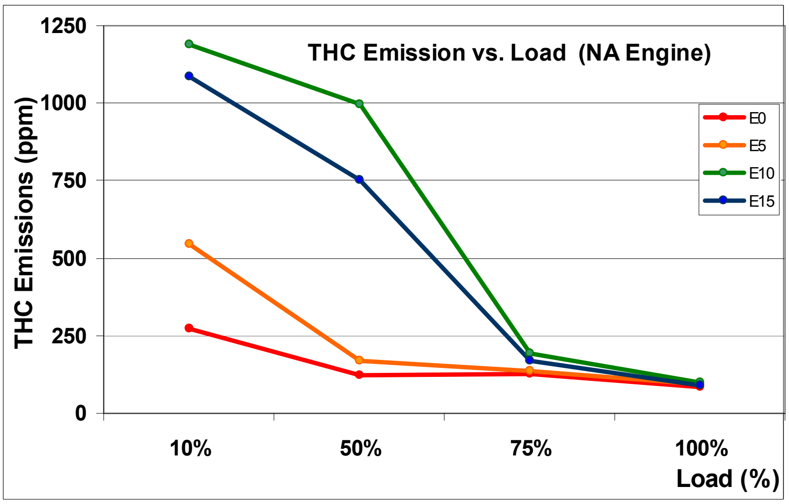


Fig 5.2.7 THC Emissions vs. Load for different fuel blends for Naturally Aspirated Engine

Similar to CO emissions with increasing load lean mixtures occur and THC emissions decrease. As a result it can be said that with increasing load CO and THC emissions decrease but NOx emissions increase. The effect of Turbocharger unit can be observed especially for E10 and E15 blends. It can be said that turbo unit can control THC emissions for higher ethanol percentages.

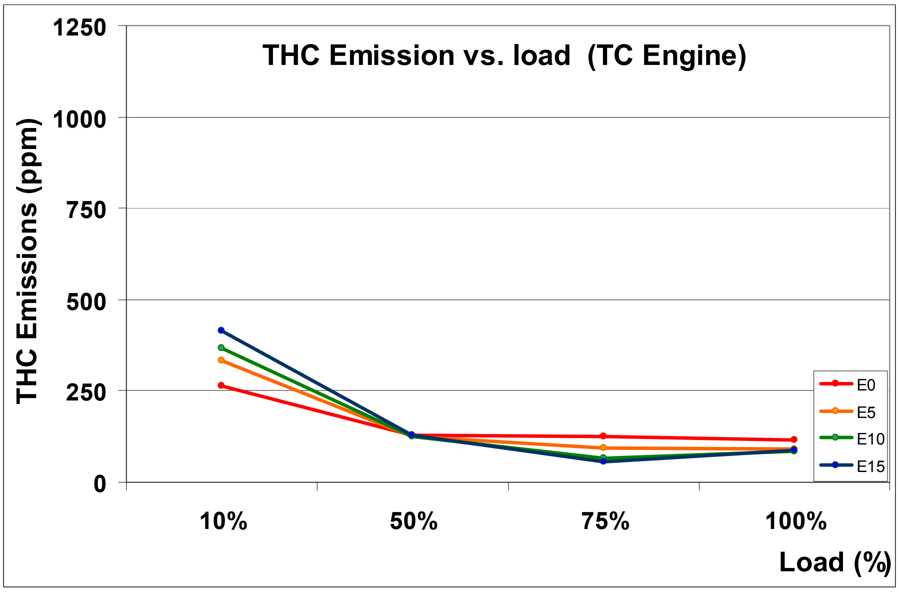


Fig 5.2.8 THC Emissions vs. Load for different fuel blends for Turbo Charged Engine

5.2.3 NOx Emissions

Higher oxygen content and lower cetane number of the ethanol are the most effective parameters on NOx emissions. Engine combustion temperature is another important parameter for NOx emissions. Lower cetane number of ethanol reduces combustion temperature and lower combustion temperature reduces NOx emission. Higher oxygen content of ethanol increases NOx formation. These two results balance each other and blends and diesel fuel show similar results as in this study. It is clear that NOx emissions are slightly higher or the same as diesel fuel comparing to ethanol blends as seen from the figure.

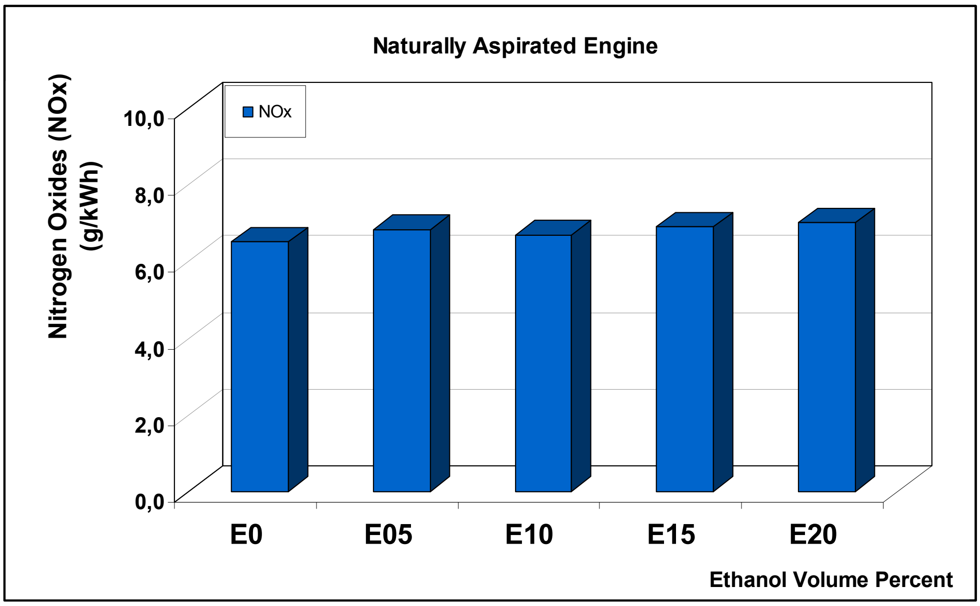


Fig 5.2.9 NOx Emissions for different fuel blends for Naturally Aspirated Engine

Oxygen content and combustion temperature are important parameters to formation of NOx emissions. Turbocharger unit sends more oxygen to intake manifold and it supports NOx formation.

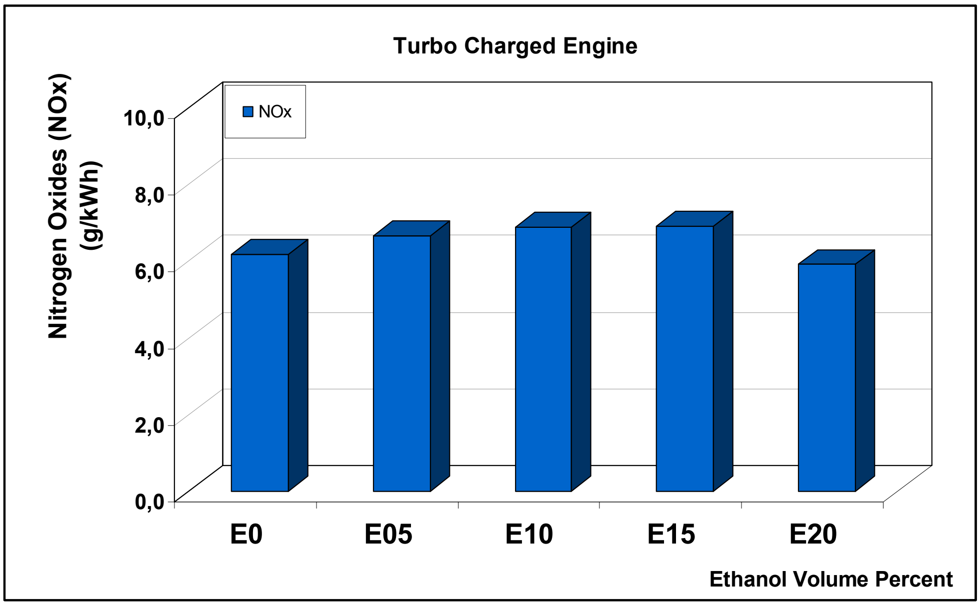


Fig 5.2.10 NOx Emissions for different fuel blends for Turbo Charged Engine

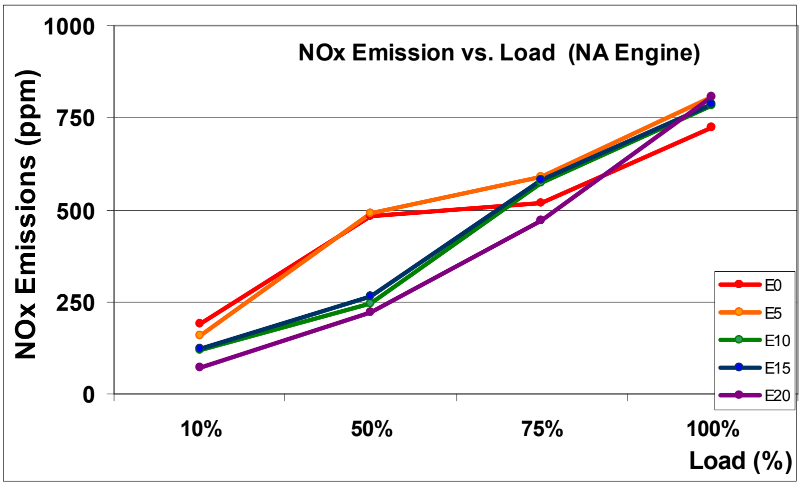


Fig 5.2.11 NOx Emissions vs. Load for different fuel blends for Naturally Aspirated Engine

Figure 5.2.11 and Figure 5.2.12 show the effect of load on NOx emissions. It is evident from the figures that NOx emissions increase by the increasing load and the max. increase in NOx emissions occur at full load conditions because of long ignition delay and rich oxygen circumstance from ethanol to the mixture. [35]

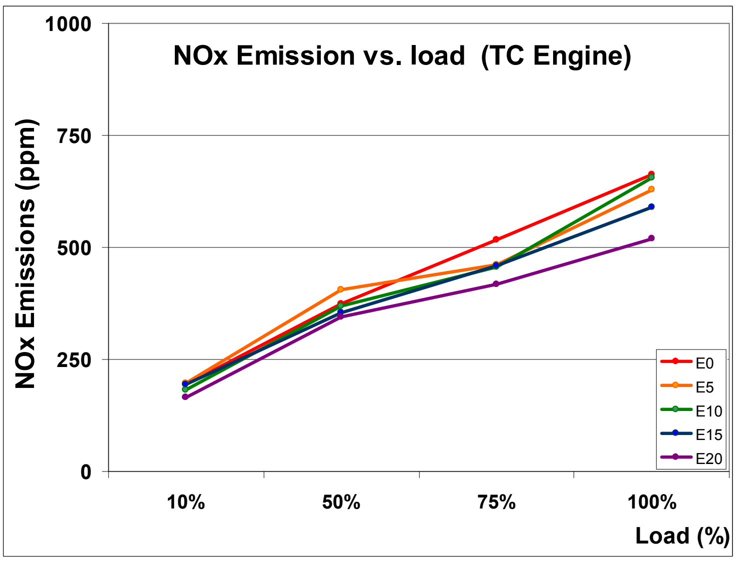


Fig 5.2.12 NOx Emissions vs. Load for different fuel blends for Turbo Charged Engine

5.2.4 PM Emissions

Ethanol and other oxygenated fuel blends are known to reduce total PM emissions. The presence of atomic bound oxygen in the fuel satisfies positive chemical control over soot formation. [18] Soot forms in the rich unburned-fuel-containing core of the fuel sprays [36]. It is commonly assumed that oxygenates blended with diesel fuel effectively deliver oxygen to the pyrolysis zone of the burning diesel spray resulting in reduced PM generation. [37] In this study an expected result is obtained. To analyze the particulate matter, particulate mass was measured for all filters before and after test for all modes.

Figure 5.2.12 and 5.2.13 show that the use of ethanol reduces PM emissions relative to diesel for both engines. Ethanol percentage is a very important parameter for reducing PM emissions. As percentage of ethanol in the mixture increases PM emission wich spread from the engine reduces. For Naturally aspirated engine with 20% Ethanol level, PM is increased unexpectedly. As a conclusion, it can be said that 20% ethanol level is a limit for adding ethanol to Naturally Aspirated Engine. However, for Turbocharged engines adding 20% ethanol to engine is still an acceptable level.

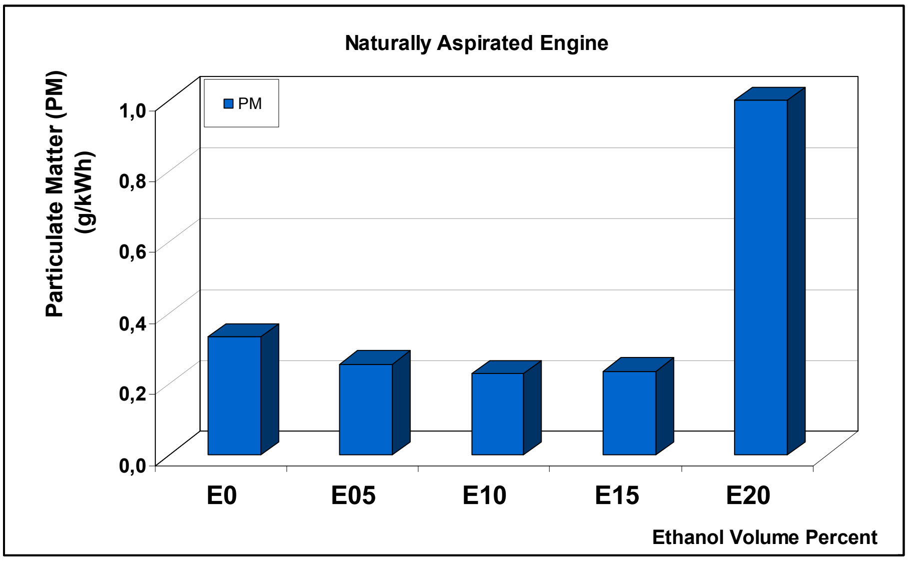


Fig 5.2.13 PM Emissions for different fuel blends for Naturally Aspirated Engine

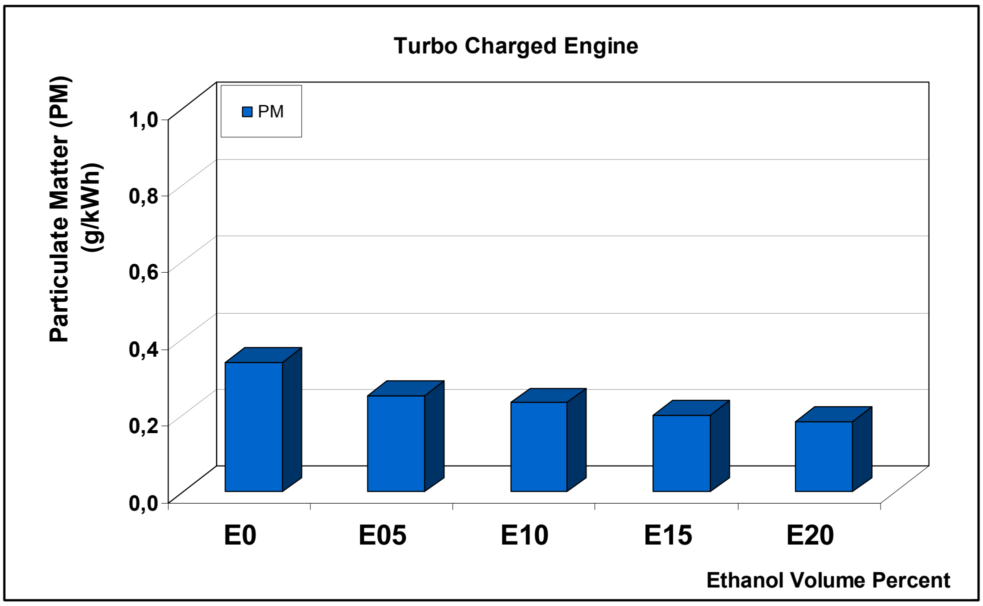


Fig 5.2.14 PM Emissions for different fuel blends for Turbo Charged Engine

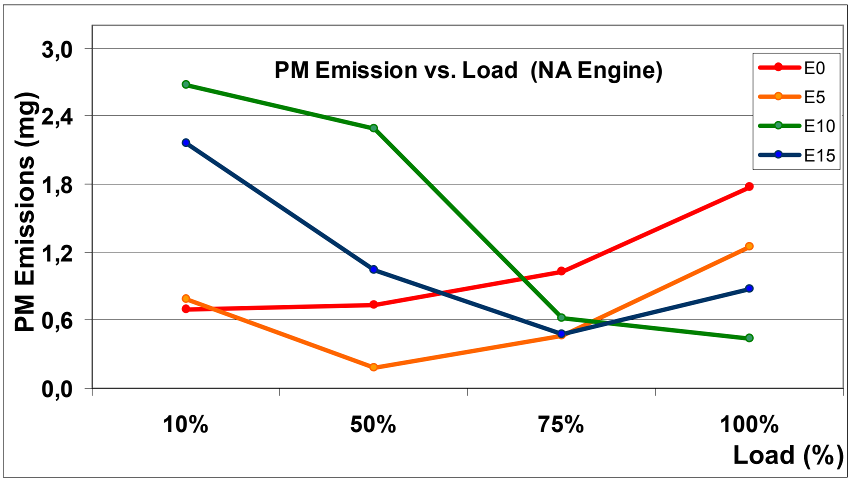


Fig 5.2.15 PM Emissions vs. Load for different fuel blends for Naturally Aspirated Engine

With the help of turbocharger unit and the oxygenated content of ethanol, as seen from figure 5.2.16, PM emissions can be controlled more than that of naturally aspirated engine. For both turbo charged and naturally aspirated engines, at high and low loads, engine emitted higher PM emissions than medium loads.

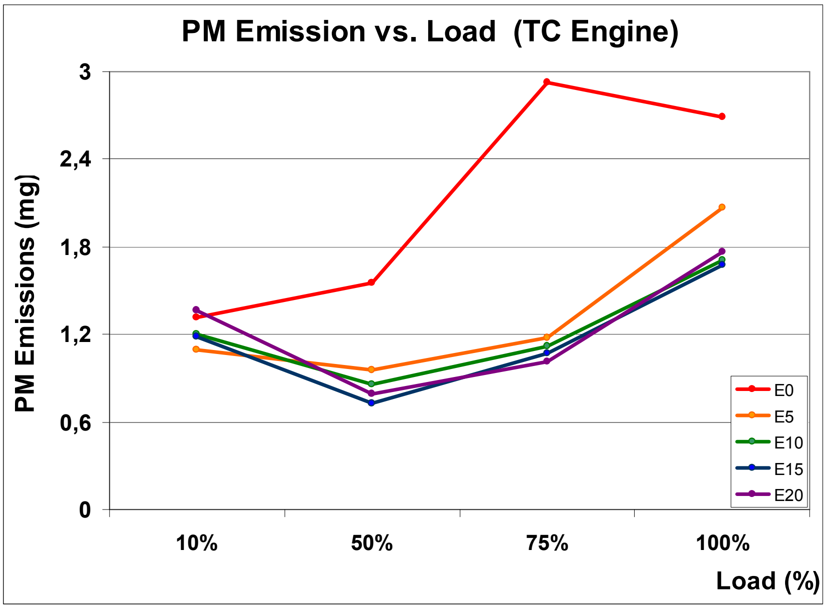
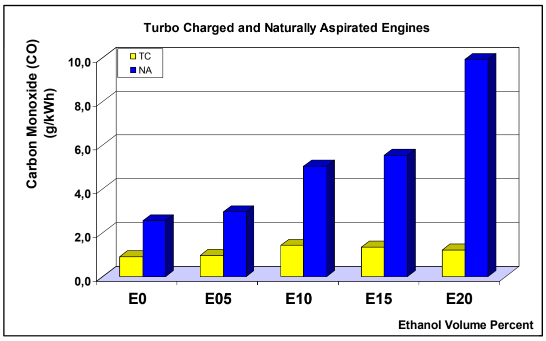
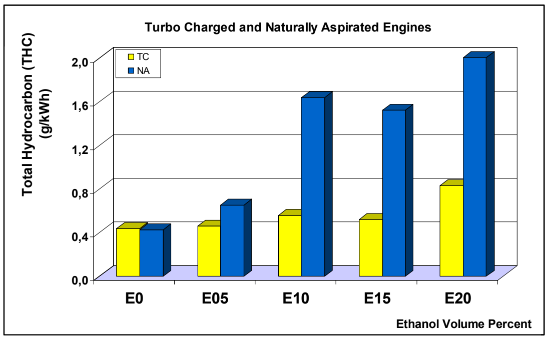


Fig 5.2.16 PM Emissions vs. Load for different fuel blends for Turbo charged Engine

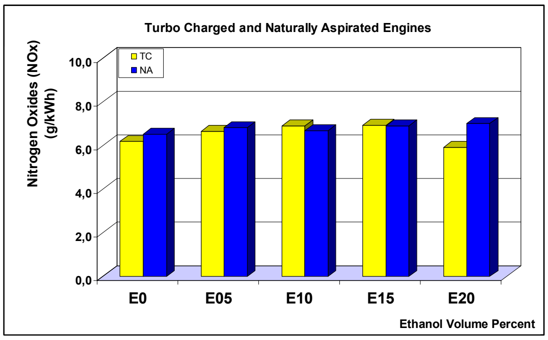
5.3 Turbocharger Effect on Emission



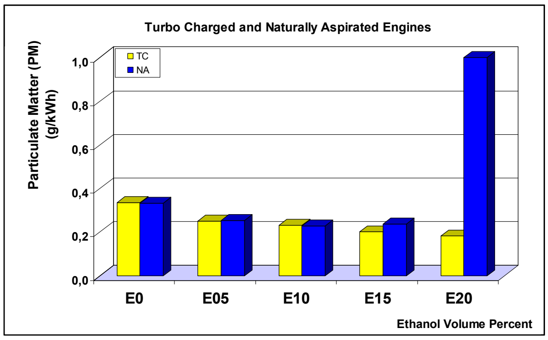
(a) CO



(b) THC



(c) NOx



(d) PM

Fig 5.3.17 Effect of Turbocharger unit on Exhaust Emissions (a) CO (b) THC (c) NOx (d) PM

As discussed in section 5.1 and 5.2, in this study, all performance and emission tests have been performed for the cases of both Turbo charged and Naturally Aspirated conditions to investigate the effect of Turbo Unit fuelled with ethanol. For turbo engines with turbocharger unit, some of the wasted heat energy is recovered and converted to useful work. Turbocharger is widely employed in current diesel engines [38]. The most important effect of Turbo unit on exhaust emissions is the increased air that provided by the Turbo. Increased air provides better air-fuel mixture and more complete combustion and more complete combustion means lower emission levels. As seen from the Figure 5.2.17 Turbo has a very strong effect on CO and THC emission levels due to more complete combustion. Turbo unit provides higher combustion temperature which is an important parameter for formation of NOx emissions and also more air and more oxygen increase NOx emissions.

5.4 Legislation for the Reduction of Exhaust Gas Emissions

Emissions from non-road mobile machinery are regulated by 97/68/EC. Engine manufacturers have to obey all regulations explained in this directive. Engine manufacturers have to produce engines with emission and particulate levels compatible with Tier 2 emission levels as given Table 5.4.1 for Turkey market. From the beginning of 2011 all off-road diesel engines have to be improved in accordance with the emission levels as given in the table 5.4.2. Tier 3.

Table 5.4.1 Max. Accepted Exhaust Emission levels for Tier 2 Engine According to (97/68/EC)

Net Power (P) (kW)	Carbon monoxide (CO) (g/kWh)	Hydrocarbons (HC) (g/kWh)	Oxides of Nitrogen (NOx) (g/kWh)	Particulates (PM) (g/kWh)
37≤ P <75	**5.0**	**1.3**	**7.0**	**0.4**

Table 5.4.2 Exhaust Emission levels for Tier 3 Engine According to (97/68/EC)

Net Power (P) (kW)	Carbon monoxide (CO) (g/kWh)	Hydrocarbons + Oxides of Nitrogen (NOx + HC) (g/kWh)	Particulates (PM) (g/kWh)
37≤ P <75	**5.0**	**4.7**	**0.4**

In this study all tests are performed with Tier 2 exhaust emission level diesel engine. Test results for reference fuel show that all emissions are within the specified emission limits. As seen from the figures 5.4.1 and 5.4.2, using ethanol is helpful to reduce emissions and all emission levels are within the legislation limits except 20%

ethanol percentage with naturally aspirated engine. According to these figures it can be said that ethanol up to 15% for naturally aspirated engines and up to 20% for turbocharged engines can be used safely.

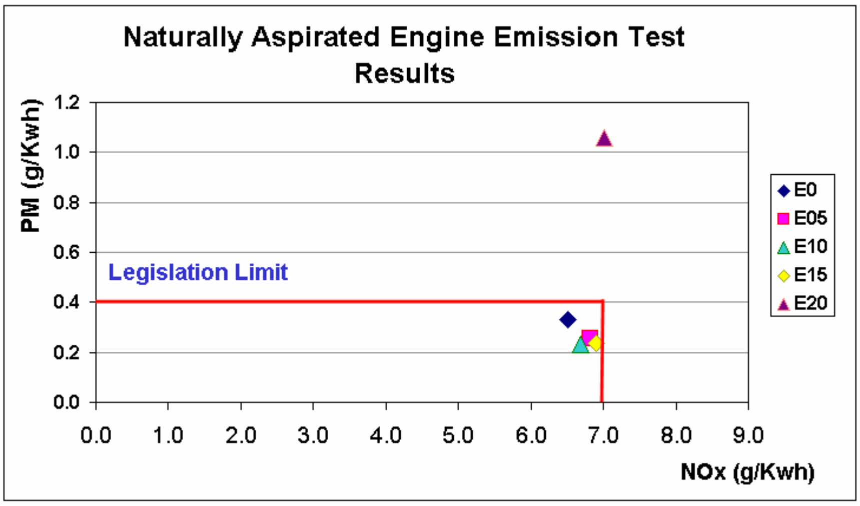


Fig 5.4.1 PM (g/Kwh) vs. NOx (g/Kwh) Emissions Regarding to Legislation Limits for Naturally Aspirated Engine

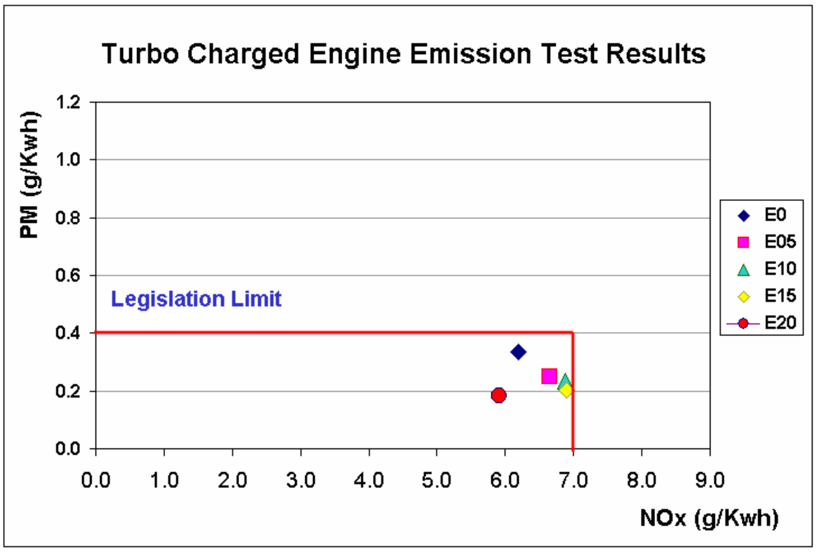


Fig 5.4.2 PM (g/Kwh) vs. NOx (g/Kwh) Emissions Regarding to Legislation Limits for Turbocharged Engine

5.5 TGA Analyses Results

The US EPA defines diesel particulate matter is the mass collected on a fiber glass filter from exhaust that has been diluted and cooled to 52°C or below. [39] Controlling particulate matter emitted by diesel engine exhausts is not easy like controlling NOx or CO emissions. In order to control PM emissions it is very important to know the components of the PM. There are some techniques to measure the components of PM. Instead of using time consuming and toxic materials techniques TGA is the best alternative to perform this study.

Several studies show that combustion chamber processes and heat release fuel and lubrication oil are the most important parameters of the formation of particulates. Particulate matter is a general name of some compounds like, ash, volatile organic sulfur and solid carbonaceous materials (dry soot). Dry soot is formed very rapidly during the first combustion phase in local rich regions of diesel heterogeneous mixture combustion [30]. Most of the soot is oxidized and the residue part is spread from exhaust. Dry soot can be formed between 1000 and 2800 K temperature and at 50 to 100 atm pressures. A tiny fraction of the fuel as well as atomized and evaporated lubrication oil are decomposed and appear as Volatile Organic Fraction (VOF) and if it includes water, it can be named as Volatile Fraction (VF). [40] Metal compounds in the fuel and lubrication oil lead to a small amount of inorganic ash. [41]. For TGA method it is assumed that particulates compounds are only VF, dry soot, and residue.

TGA method enables analysis of continuous weight loss of a sample filter at controlled heating rate in an inert atmosphere [40]. Particulates were collecting on the filter homogenously in dilution tunnel during 8-Mode tests for each mode. While analyzing filters, engine speed and load conditions are taken into consideration because the load and the speed of the engine are the most important parameters for the effect PM formation. For this reason, only Mode#1 filters are analyzed to see the PM components at max speed and max load in this study. The engine is run at 2500 rpm and full load for Mode #1 and it is investigated that ethanol has a significant effect on PM components.

Thermo-gravimetric sampling method allows the measurement of emissions of particulate compounds using filters loaded from dilution tunnel. Figure 5.5.1 shows the filter schema. The gray surface on the figure 5.5.1 shows the accumulated particulates on the filters. White surface on the filter is formed because of the shape of the filter holder. TGA pan allows analyzing only a small portion of the filter each time due to its small size. For all filters rectangular portion from the center of the filter are used to analyze.

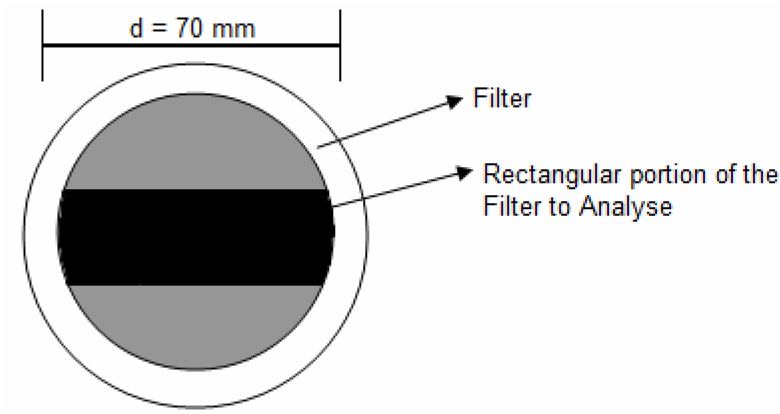


Fig 5.5.1 Sample Filter

To understand and to interpret the TGA curves, it must be known that the decrease in the TGA curves represents the amounts of the components of PM. The first weight drop region represents the VOF part. It occurs roughly at lower temperature between 100-350 °C. So, if the filter temperature exceeds approximately 350°C, VOF is vaporized and only dry soot remains accumulated on the filter wall. [29]. The second region represents the dry soot content of the PM. It occurs at the temperature ranging between 350-600 °C. The last region represents the residue part and a mass of material which is not burned in TGA. TGA curves for diesel fuel and ethanol diesel blends have been given in Appendix B.

PM components have an important effect in after diesel treatment. VOF can be easily eliminated by some catalysts such as diesel oxidation catalyst (DOC), but dry soot is more difficult to eliminate. [42] Due to its oxygen component ethanol can reduce the level of emitted dry soot. As seen from figure 5.5.1, adding ethanol to diesel fuel VOF is increased from 12,3 % to 34,5 % and also dry soot is decreased from 36,92 % to 24,21 %.

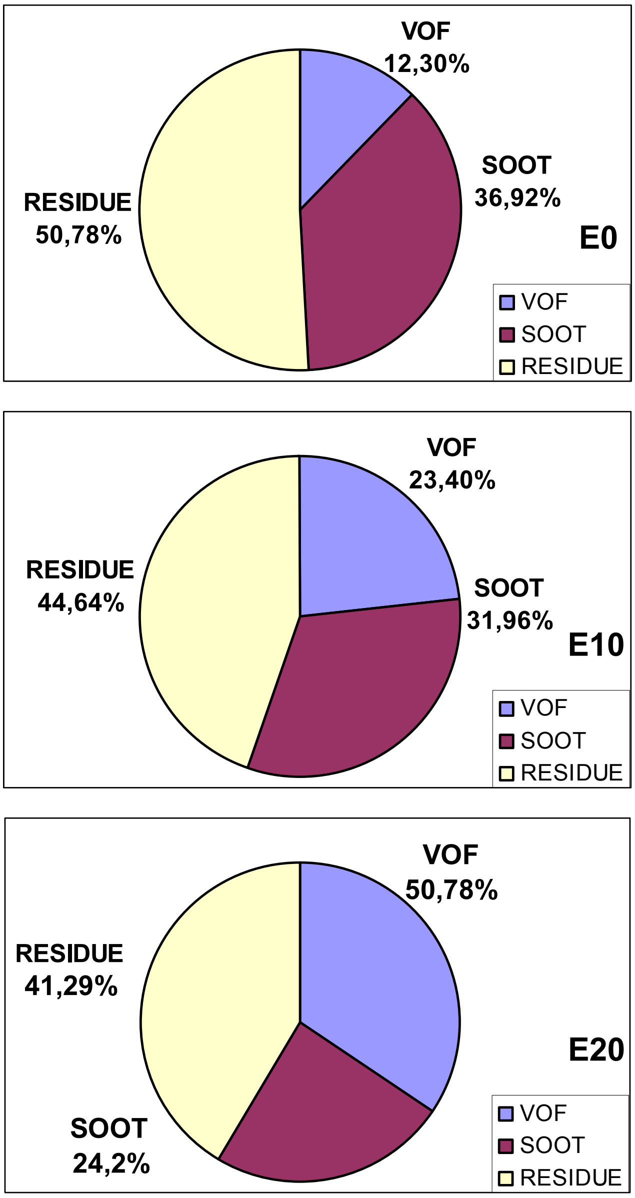


Fig 5.5.2 Effect of Ethanol on PM Components

Oxygenated fuels, like ethanol, are known to reduce PM level. Due to ethanol's oxygen content, oxygen rate in ethanol fuel is more than the diesel fuel in the rich zones. With more oxygen, more complete combustion can be performed and soot level is reduced. As seen from the figure 5.5.2 soot level in the PM decreases while the ethanol percentage increases.

CHAPTER 6

CONCLUSIONS

The engine performance and exhaust emission characteristics of ethanol and diesel fuel were experimentally investigated in two diesel engines operated at natural-aspirtaed and turbocharged conditions. The two engines were tested under two different test cycles: constant load variable speed and constant speed variable load test cycle.

Firstly, performance and emission tests were made by using reference diesel fuel in order to determine the base performance and emission values of the engine with only diesel and to be able to compare to understand the effects of ethanol. After that tests were carried out by blending ethanol up to 20% to diesel. The results obtained from this study can be summarized as follows:

1. The brake power and torque of the engine with diesel fuel are higher than those with blended ethanol-diesel fuel for both NA and TC engines. This means that adding ethanol to diesel fuel for both turbo and naturally aspirated engines results in a decrease in engine power. This decrease level is much more in turbo engine than naturally aspirated engine. This decrease in engine performance can be explained with lower energy level of ethanol. The lower energy level of the ethanol–diesel fuel blend causes some reductions in the engine power when it is used in diesel engines without any modifications.

2. Adding ethanol to both turbocharged and naturally aspirated engines increases the fuel consumption. This result is an expected result due to the decrease in power output of the engines. This increment is much more for turbocharged

engine than the naturally aspirated one. For both engines the brake specific fuel consumption has increased and the power output has decreased with the increase of ethanol content in the blended fuel at overall operating conditions.

3. The brake thermal efficiency decreases with increasing ethanol content in the blend. The loss of heating value and the increase of brake specific fuel consumption (bsfc) using ethanol-diesel blends lead to a decrease in brake thermal efficiency.

4. For both engines the smoke emitted by the ethanol–diesel fuel blends are significantly lower than the reference diesel fuel. This reduction increases with the increase of the rate of ethanol in the mixture. Ethanol provides more oxygen with rich zones in combustion chamber due to more effective combustion,

5. Using ethanol with diesel fuel increases CO emissions for both NA and TC engines. The main reason of this increase is the lower cetane number of the blended fuels induced an increase in the ignition delay. This delay was higher with the rise in ethanol content.

6. Unburned hydrocarbon emissions are increased by the use of ethanol-diesel fuel blends comparing to diesel fuel and by the increase of emission being higher with the increase in the percentage of ethanol in the blend. This condition can be explained with higher heat of evaporation of ethanol and the limited miscibility of ethanol with the diesel fuel.

7. NOx emissions are slightly higher or the same as diesel fuel comparing to ethanol blends. Lower cetane number of the ethanol and engine combustion temperature are the most important parameters for NOx emissions. And also

higher oxygen content of ethanol increases NOx formation. These results balance each other and blends and diesel fuel show similar results as in this study.

8. Ethanol and other oxygenated fuel blends are known to reduce total PM emissions. The presence of atomic bound oxygen in the fuel satisfies positive chemical control over soot formation. The use of ethanol reduces PM emissions relative to diesel for both engines. Ethanol percentage is a very important parameter for reducing PM emissions.

9. The effect of Turbocharger unit is investigated in this study. Turbo has a very strong effect on CO and THC emission levels due to more complete combustion. Turbo unit provides higher combustion temperature and more air means more oxygen which are the important parameters for formation of NOx emissions.

10. The use of Ethanol gives better performance and exhaust emissions of the turbocharged engine compared with the natural-aspirated one.

11. Thermo-gravimetric sampling method used to measure particulate compounds using filters loaded from dilution tunnel. Due to its oxygen component ethanol reduces the level of emitted dry soot. Adding ethanol to diesel fuel VOF is increased and dry soot is decreased.

Future Works

Over the past 20 years, ethanol has been evaluated as an additive to diesel fuel, but there still a lot of open points that are needed to be investigated. One can state some of them as the following:

1. In this study no additive is used with ethanol but, cetane number improver and emulsifier agents can be used with ethanol to investigate the effects.

2. In this study no engine modification is performed but to improve performance data, fuel injection pump modification and adjustments can be performed and investigated.

3. Tested TTF engines emission level is Tier 2 (According to 97/68/EC), all tests can be repeated with a Tier 3 engine to understand the compatibility of ethanol diesel blend to new emission regulations.

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APPENDICES A

EXPERIMENTAL DATA (PERFORMANCE TEST DATA)

PERFORMANCE TEST DATA - 75 HP- NA ENGINE

Power (HP)	E0	E5	E10	E15	E20
2500	**76,5**	**75,8**	**73,7**	**72,7**	**71,8**
2400	77,3	75,7	74,4	74,2	71,4
2300	77,0	76,1	74,6	74,0	71,1
2200	76,1	75,5	73,4	73,0	70,9
2100	74,9	74,5	72,8	72,5	70,0
2000	73,2	72,4	71,8	71,3	68,8
1900	71,9	71,2	70,6	69,5	67,0
1800	69,7	68,9	67,4	67,0	65,5
1700	66,6	66,4	65,9	64,8	63,0
1600	63,8	63,1	62,9	62,1	60,9
1500	**61,4**	**61,1**	**60,5**	**59,5**	**58,7**

Torque (Nm)	E0	E5	E10	E15	E20
2500	**218,9**	**215,8**	**207,1**	**212,6**	**204,6**
2400	229,3	224,5	220,5	219,8	210,5
2300	239,0	235,4	230,7	231,8	222,6
2200	245,8	243,9	237,0	235,5	229,3
2100	253,3	252,2	245,6	245,2	237,1
2000	260,5	256,7	251,2	252,9	244,6
1900	266,5	265,7	259,8	260,2	250,4
1800	274,1	272,1	265,6	266,6	258,1
1700	278,7	277,7	270,3	270,8	262,8
1600	289,2	283,0	277,8	279,3	269,2
1500	**295,5**	**292,4**	**285,0**	**286,1**	**276,9**

Fuel Rate (kg/h)	E0	E5	E10	E15	E20
2500	**12,50**	**12,34**	**13,04**	**12,43**	**11,31**
2400	12,80	12,60	12,36	12,56	13,24
2300	12,64	12,39	12,47	12,23	12,48
2200	12,33	11,98	12,08	12,23	11,94
2100	11,95	12,05	12,01	11,90	11,75
2000	11,77	11,57	11,74	11,69	11,44
1900	11,50	11,16	11,36	11,32	11,34
1800	11,11	10,98	11,12	11,13	10,94
1700	10,72	10,71	10,75	10,61	10,44
1600	10,38	10,30	10,50	10,41	10,15
1500	**9,98**	**10,25**	**10,26**	**10,14**	**10,06**

Smoke (FSN)	E0	E5	E10	E15	E20
2500	**0,77**	**0,61**	**0,51**	**0,31**	**0,28**
2400	0,97	0,77	0,62	0,33	0,29
2300	1,06	0,90	0,70	0,45	0,33
2200	1,23	0,88	0,62	0,49	0,37
2100	1,24	0,98	0,71	0,50	0,40
2000	1,37	1,02	0,72	0,59	0,45
1900	1,40	1,06	0,84	0,60	0,53
1800	1,69	1,34	0,99	0,81	0,67
1700	2,44	1,87	1,30	1,09	0,90
1600	3,19	2,27	1,75	1,51	1,19
1500	**3,95**	**2,95**	**2,37**	**2,00**	**1,58**

PERFORMANCE TEST DATA -85 HP-TC ENGINE

Power (HP)	E0	E5	E10	E15	E20
2500	**84,5**	**84,3**	**83,4**	**79,9**	**76,8**
2300	86,6	84,6	84,2	79,6	76,9
2100	85,9	83,9	82,7	79,3	76,8
1900	83,4	82,6	80,6	77,7	75,5
1700	80,6	78,8	76,7	74,4	72,8
1500	**74,8**	**73,3**	**72,3**	**70,2**	**68,6**

Torque (Nm)	E0	E5	E10	E15	E20
2500	**240,0**	**237,6**	**235,3**	**227,5**	**218,7**
2300	267,6	260,7	256,1	246,3	235,0
2100	287,5	280,3	275,9	269,1	260,5
1900	308,4	302,3	294,4	291,2	283,0
1700	328,5	321,7	316,0	311,9	305,3
1500	**345,1**	**340,8**	**335,5**	**330,7**	**325,7**

Fuel Rate (kg/h)	E0	E5	E10	E15	E20
2500	**14,00**	**14,30**	**14,28**	**14,40**	**14,79**
2300	13,28	13,39	13,59	13,80	13,63
2100	12,64	12,66	12,03	12,96	12,66
1900	11,71	12,00	11,72	11,59	11,56
1700	11,42	11,65	11,46	11,32	11,32
1500	**10,75**	**10,04**	**10,16**	**10,88**	**10,47**

Smoke (FSN)	E0	E5	E10	E15	E20
2500	**0,69**	**0,67**	**0,64**	**0,66**	**0,63**
2300	0,78	0,70	0,64	0,63	0,59
2100	0,61	0,58	0,54	0,49	0,45
1900	0,63	0,56	0,54	0,49	0,42
1700	0,64	0,58	0,54	0,52	0,49
1500	**0,98**	**0,81**	**0,71**	**0,65**	**0,61**

APPENDICES B

EXPERIMENTAL DATA (EMISSION TEST DATA)

EMISSION TEST DATA - 85 HP- TC ENGINE

E0

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2507	213,7	56,0	75,1	13,75	0,71
2	2498	164,9	43,3	58,0	10,84	0,42
3	2507	111,6	29,3	39,3	8,01	0,34
4	2508	26,0	6,8	9,2	3,88	0,07
5	1505	301,2	47,4	63,6	10,56	1,00
6	1499	230,3	36,3	48,6	7,77	0,34
7	1505	153,6	24,2	32,4	5,34	0,19
8	749	105,7	8,3	11,1	1,95	0,11

Tamb	Pamb	Humidity
°C	mbar	%
25,0	894,6	52,2
25,5	897,4	55,4
23,9	902,1	55,6
23,3	908,6	51,1
23,4	916,4	54,7
22,7	918,4	53,8
22,2	918,0	51,9
23,3	938,2	50,1

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1420	85,8	244,0	159,3
2	1,1394	66,1	187,9	165,7
3	1,1305	44,4	126,1	180,4
4	1,1213	10,3	29,2	369,9
5	1,1119	70,7	334,9	150,2
6	1,1080	53,9	255,2	144,5
7	1,1077	35,9	170,1	148,6
8	1,0857	12,1	114,8	157,9

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,017	8,32	129	0,02	117	662
2	0,014	7,04	102	0,01	124	516
3	0,016	5,47	128	0,02	130	375
4	0,042	2,85	350	0,04	263	194
5	0,022	10,99	182	0,02	93	952
6	0,011	8,50	85	0,01	114	814
7	0,013	6,09	103	0,01	119	575
8	0,013	4,66	104	0,01	115	588

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	22,44	1,54	21,7	93,9	21,8	91,5	90,64	68,26
2	25,84	1,78	21,2	93,6	21,7	91,2	192,82	144,97
3	32,18	2,21	20,8	93,9	21,4	91,3	199,76	150,12
4	57,54	3,96	20,1	93,7	21,0	91,2	186,72	140,17
5	17,93	1,23	19,8	94,0	20,6	90,8	92,48	69,97
6	21,91	1,51	19,8	93,5	20,5	90,4	192,10	144,54
7	29,00	1,99	19,5	93,7	20,2	90,9	199,16	149,65
8	36,73	2,53	19,2	93,6	19,9	90,7	199,36	149,81

Log No	P.M.	F. Empty	F. Full	
[1, 8]	mg	mg	mg	mg
1	2,686	205,1	207,786	2,686
2	2,929	209,202	212,131	2,929
3	1,55	208,726	210,276	1,55
4	1,312	207,758	209,07	1,312
5	2,087	210,055	212,142	2,087
6	1,114	209,162	210,276	1,114
7	1,236	204,937	206,173	1,236
8	0,674	207,326	208	0,674

E5

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2507	208,1	54,6	73,2	14,47	0,53
2	2511	158,5	41,5	55,7	10,81	0,41
3	2505	110,1	28,9	38,7	8,49	0,29
4	2504	24,7	6,5	8,7	4,05	0,14
5	1503	316,2	49,8	66,7	11,22	0,92
6	1502	232,6	36,6	49,1	8,01	0,34
7	1503	159,5	25,1	33,7	5,54	0,20
8	751	149,8	11,8	15,8	2,78	0,01

Tamb	Pamb	Humidity
°C	mbar	%
25,1	888,7	47,8
23,7	892,2	47,2
25,8	896,4	47,2
25,0	901,6	49,2
23,4	912,5	45,8
21,9	912,3	46,3
24,9	912,4	48,7
22,1	933,0	49,1

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1497	84,1	239,3	155,4
2	1,1425	63,6	181,1	184,7
3	1,1413	44,2	125,7	165,3
4	1,1331	9,8	28,0	387,2
5	1,1166	74,5	353,1	151,0
6	1,1141	54,7	259,1	147,2
7	1,1194	37,7	178,5	147,3
8	1,0896	17,2	163,2	159,6

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,017	8,14	118	0,02	91	628
2	0,015	6,85	113	0,02	93	462
3	0,018	5,39	138	0,02	127	405
4	0,050	2,90	413	0,05	334	197
5	0,022	11,25	180	0,02	108	1189
6	0,012	8,46	92	0,01	134	845
7	0,014	6,22	108	0,01	148	599
8	0,010	6,19	71	0,01	128	890

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	22,83	1,57	20,6	93,1	20,6	90,8	99,46	74,89
2	26,45	1,82	21,4	93,2	21,6	90,7	199,01	149,61
3	32,51	2,23	21,8	93,1	22,1	90,6	191,53	143,89
4	57,18	3,93	21,7	93,0	22,2	90,6	200,03	150,17
5	17,63	1,21	21,9	93,3	22,2	90,2	99,33	75,11
6	22,06	1,52	22,0	93,2	22,4	90,3	199,90	150,41
7	28,59	1,97	21,9	93,1	22,3	90,3	198,95	149,56
8	28,75	1,98	21,8	93,1	22,3	90,3	198,97	149,57

P.M.	F. Empty	F. Full	
mg	mg	mg	mg
2,068	206,887	208,955	2,068
1,175	208,468	209,643	1,175
0,955	210,908	211,863	0,955
1,092	209,67	210,762	1,092
2,282	206,838	209,12	2,282
1,494	208,759	210,253	1,494
0,596	206,575	207,171	0,596
0,598	208,323	208,921	0,598

E10

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2502	210,3	55,1	73,9	14,29	0,57
2	2498	162,2	42,5	57,0	10,86	0,33
3	2503	109,9	28,8	38,6	10,29	0,22
4	2501	20,6	5,4	7,2	5,77	0,08
5	1499	303,2	47,6	63,8	10,59	0,78
6	1496	229,4	36,0	48,3	7,84	0,28
7	1498	153,9	24,1	32,4	5,27	0,13
8	669	6,9	0,5	0,6	0,77	0,03

Tamb	Pamb	Humidity
°C	mbar	%
25,2	903,1	52,3
23,0	906,2	51,9
25,1	910,9	51,7
25,6	915,7	55,7
24,0	926,7	54,2
25,6	928,2	56,1
24,3	927,3	54,3
24,1	932,9	55,9

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1317	83,6	238,0	146,4
2	1,1144	63,5	180,8	295,4
3	1,1217	43,3	123,3	292,0
4	1,1168	8,1	23,0	-184,7
5	1,0932	69,7	331,5	155,5
6	1,1018	53,2	252,8	152,5
7	1,1004	35,6	169,3	150,3
8	1,0935	0,7	7,6	887,1

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,016	8,30	112	0,02	85	655
2	0,016	6,79	113	0,02	67	456
3	0,022	5,50	168	0,02	125	368
4	0,067	2,92	575	0,07	366	181
5	0,020	10,77	156	0,02	105	1021
6	0,012	8,52	84	0,01	128	849
7	0,016	6,06	117	0,02	140	600
8	0,017	1,44	126	0,02	136	193

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	23,05	1,58	17,4	94,8	17,7	92,2	99,46	75,19
2	27,26	1,87	18,8	94,6	19,3	92,2	199,56	150,20
3	32,67	2,25	19,9	94,6	20,4	92,1	199,83	150,36
4	57,84	3,98	21,0	94,5	21,3	92,1	199,25	149,76
5	18,61	1,28	21,8	94,9	22,0	91,7	99,87	75,51
6	22,51	1,55	21,5	94,7	22,0	91,8	199,82	150,51
7	30,14	2,07	21,7	94,6	22,3	92,1	199,00	149,74
8	117,11	8,05	21,8	94,6	22,3	91,8	199,55	150,16

P.M.	F. Empty	F. Full	
mg	mg	mg	mg
1,705	206,177	207,882	1,705
1,117	206,924	208,041	1,117
0,861	207,571	208,432	0,861
1,204	207,577	208,781	1,204
2,168	207,933	210,101	2,168
0,846	207,774	208,62	0,846
0,507	209,734	210,241	0,507
0,156	207,515	207,671	0,156

E15

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2500	204,4	53,5	71,7	14,80	0,57
2	2502	157,2	41,1	55,2	11,77	0,27
3	2503	106,4	27,9	37,4	9,12	0,15
4	2501	26,4	6,9	9,3	3,85	0,03
5	1497	296,2	46,5	62,3	10,64	0,63
6	1499	223,5	35,1	47,0	7,93	0,19
7	1497	152,3	23,9	32,0	5,11	0,09
8	673	7,0	0,5	0,7	0,56	0,03

Tamb	Pamb	Humidity
°C	mbar	%
25,0	908,5	61,0
26,7	912,2	58,4
25,4	916,3	59,6
22,7	923,1	57,5
24,0	933,9	57,8
21,7	932,6	57,8
23,0	935,5	58,2
24,5	950,3	59,2

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1154	80,0	227,9	218,8
2	1,1232	61,9	176,5	227,4
3	1,1157	41,7	118,7	166,6
4	1,1025	10,2	29,1	481,7
5	1,1014	68,6	326,2	155,9
6	1,0894	51,2	243,5	146,4
7	1,0883	34,9	165,7	178,7
8	1,0741	0,7	7,6	1221,1

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,017	7,99	117	0,02	87	589
2	0,016	6,91	112	0,02	57	459
3	0,026	5,43	204	0,03	128	354
4	0,076	2,84	588	0,08	414	194
5	0,019	10,94	142	0,02	109	1056
6	0,014	8,32	93	0,01	139	776
7	0,019	6,00	138	0,02	156	582
8	0,021	1,45	153	0,02	146	196

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	23,95	1,65	15,7	95,0	16,4	92,7	99,66	75,83
2	27,01	1,86	16,0	95,4	17,1	92,8	193,95	145,96
3	33,55	2,31	15,9	95,1	17,4	92,7	189,29	142,37
4	59,83	4,11	15,7	95,1	17,3	92,6	199,62	150,03
5	18,55	1,28	16,8	95,4	17,8	92,2	199,40	150,43
6	23,10	1,59	17,8	95,2	18,7	92,4	199,56	150,29
7	30,59	2,10	18,5	95,1	19,2	92,4	192,74	145,01
8	117,52	8,08	18,7	95,1	19,5	92,3	200,57	150,90

P.M.	F. Empty	F. Full	
mg	mg	mg	mg
1,677	209,807	211,484	1,677
1,071	207,258	208,329	1,071
0,725	208,916	209,641	0,725
1,183	205,211	206,394	1,183
1,842	208,726	210,568	1,842
0,768	204,623	205,391	0,768
0,683	205,662	206,345	0,683
0,419	207,208	207,627	0,419

E20

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2501	193,4	50,6	67,9	14,04	0,76
2	2500	150,1	39,3	52,7	12,62	0,32
3	2521	95,4	25,0	33,5	3,43	0,20
4	2499	25,9	6,8	9,1	1,40	0,08
5	1498	291,2	45,7	61,3	10,72	0,72
6	1498	221,4	34,7	46,5	8,03	0,23
7	1496	149,3	23,4	31,4	5,27	0,07
8	670	7,5	0,5	0,7	0,64	0,01

Tamb	Pamb	Humidity
°C	mbar	%
25,6	894,4	61,4
22,0	897,9	58,7
24,8	900,7	55,1
24,4	908,1	53,2
22,3	917,5	54,9
21,9	919,9	54,2
23,0	918,9	53,9
25,4	937,7	51,2

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
100,00%	1,1432	77,6	221,1	216,8
75,00%	1,1301	59,5	169,6	215,9
50,00%	1,1338	38,0	108,2	112,9
10,00%	1,1239	10,2	29,1	371,8
%100	1,1085	67,9	322,8	161,0
%75	1,1048	51,4	244,6	146,3
%50	1,1012	34,6	164,4	147,9
%0	1,0902	0,8	8,2	702,8

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
100,00%	0,019	7,91	140	0,02	88	520
75,00%	0,017	6,63	130	0,02	67	417
50,00%	0,044	5,29	356	0,04	185	345
10,00%	0,089	2,92	757	0,09	1382	165
%100	0,021	10,44	160	0,02	139	909
%75	0,014	7,97	106	0,01	145	746
%50	0,025	5,80	197	0,02	161	611
%0	0,023	1,41	180	0,02	169	195

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
100,00%	23,75	1,63	19,0	94,0	19,3	91,3	99,36	75,61
75,00%	27,64	1,90	19,6	94,0	20,0	91,4	215,90	149,37
50,00%	33,56	2,31	20,1	93,6	20,5	91,2	217,07	150,05
10,00%	57,45	3,95	20,3	93,6	20,9	91,3	216,08	149,35
%100	18,89	1,30	21,4	93,5	21,5	90,4	193,06	147,95
%75	23,61	1,62	21,5	93,5	21,9	90,5	200,01	150,42
%50	30,81	2,12	21,7	93,7	22,1	91,0	199,86	150,15
%0	117,75	8,09	21,5	93,6	22,0	91,1	193,36	147,85

P.M.	F. Empty	F. Full	
mg	mg	mg	mg
1,767	206,299	208,066	1,767
1,011	204,741	205,752	1,011
0,792	208,19	208,982	0,792
1,362	209,722	211,084	1,362
1,803	208,557	210,36	1,803
0,77	209,35	210,12	0,77
0,551	208,132	208,683	0,551
0,368	210,12	210,488	0,368

EMISSION TEST DATA - 75 HP- NA ENGINE

E0

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2501	197,6	51,7	69,3	12,61	0,70
2	2499	150,4	39,3	52,8	9,81	0,25
3	2500	99,7	26,1	35,0	6,85	0,16
4	2499	18,0	4,7	6,3	3,44	0,07
5	1504	261,7	41,2	55,2	10,12	3,66
6	1504	194,5	30,6	41,1	6,74	0,35
7	1503	129,1	20,3	27,2	4,69	0,13
8	700	1,1	0,1	0,1	0,50	0,05

Tamb	Pamb	Humidity
°C	mbar	%
22,2	925,1	47,0
22,8	923,4	48,5
24,3	920,7	48,5
23,5	925,2	49,0
23,4	934,2	47,0
23,7	935,4	47,2
23,9	930,0	48,7
22,9	935,8	47,5

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,0990	76,2	217,1	166,1
2	1,1022	58,1	165,8	168,0
3	1,1084	38,8	110,5	178,4
4	1,1014	6,9	19,8	491,6
5	1,0906	60,2	285,4	168,6
6	1,0897	44,7	212,0	151,5
7	1,0964	29,9	141,5	158,0
8	1,0878	0,1	1,2	5376,5

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,026	10,26	195	0,03	83	722
2	0,026	7,75	195	0,03	129	517
3	0,028	5,30	211	0,03	123	482
4	0,053	2,49	444	0,05	271	188
5	0,260	13,20	2461	0,26	24	904
6	0,014	8,48	86	0,01	114	971
7	0,017	5,68	115	0,02	109	664
8	0,020	1,40	145	0,02	104	218

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	19,50	1,34	21,7	94,8	22,1	92,3	99,41	74,96
2	24,93	1,71	21,9	94,8	22,4	92,3	199,60	150,04
3	34,93	2,40	21,9	94,6	22,4	92,0	167,54	125,82
4	70,54	4,85	22,1	94,7	22,5	92,3	199,57	149,77
5	15,62	1,07	22,8	96,7	22,9	91,8	97,74	75,29
6	22,81	1,57	22,9	94,9	23,1	91,8	199,67	150,40
7	32,61	2,24	22,5	94,7	22,8	91,8	198,99	149,61
8	125,76	8,65	22,2	94,7	22,6	92,0	200,03	150,31

P.M.
mg
1,771
1,027
0,733
0,699
7,898
1,372
1,002
0,384

E5

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2502	199,7	52,4	70,2	12,48	0,67
2	2502	149,3	39,1	52,5	10,00	0,13
3	2504	98,9	25,9	34,8	6,90	0,08
4	2500	19,2	5,0	6,7	3,63	0,02
5	1504	261,4	41,2	55,3	10,23	3,07
6	1506	194,5	30,7	41,1	6,88	0,35
7	1506	129,2	20,3	27,3	4,80	0,13
8	737	0,4	0,0	0,0	0,56	0,06

Tamb	Pamb	Humidity
°C	mbar	%
22,2	911,7	41,8
22,6	914,6	41,5
23,9	909,8	42,4
20,2	913,6	41,6
25,1	926,7	41,2
24,2	919,6	41,1
24,2	924,8	40,3
22,0	921,7	40,0

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1154	78,3	222,8	164,1
2	1,1125	58,4	166,1	163,5
3	1,1207	39,0	110,9	196,1
4	1,1091	7,5	21,2	600,9
5	1,1025	60,9	288,2	165,3
6	1,1094	45,6	215,8	152,8
7	1,1031	30,1	142,5	160,1
8	1,1029	0,0	0,4	12632,0

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,026	10,09	182	0,03	94	805
2	0,031	7,73	235	0,03	137	587
3	0,042	5,29	332	0,04	171	490
4	0,093	2,46	788	0,09	545	158
5	0,231	12,90	2094	0,23	46	908
6	0,015	8,38	86	0,01	149	953
7	0,019	5,65	125	0,02	134	674
8	0,027	1,32	187	0,03	166	199

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	19,69	1,35	20,5	93,5	20,8	90,9	78,61	59,27
2	24,89	1,71	21,0	93,5	21,5	90,8	182,72	137,22
3	35,05	2,41	21,0	93,7	21,6	91,1	199,91	150,08
4	71,46	4,91	21,3	93,6	21,8	91,2	199,23	149,58
5	15,76	1,08	22,2	94,5	22,4	90,7	92,64	70,87
6	23,05	1,58	22,2	93,8	22,6	90,7	195,92	147,55
7	32,81	2,26	22,7	93,5	23,0	90,3	200,84	150,98
8	134,63	9,25	22,3	93,5	22,8	90,4	190,68	143,33

P.M.
mg
1,247
0,467
0,185
0,781
5,122
1,314
1,215
0,077

E10

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2502	190,5	49,9	66,9	12,35	0,43
2	2503	142,2	37,2	49,9	10,12	0,10
3	2504	95,4	25,0	33,5	8,45	0,08
4	2499	19,3	5,1	6,8	3,66	0,00
5	1505	259,1	40,8	54,7	10,29	2,02
6	1504	191,7	30,2	40,5	6,98	0,27
7	1502	127,6	20,1	26,9	4,79	0,09
8	745	0,6	0,0	0,1	0,58	0,07

Tamb	Pamb	Humidity
°C	mbar	%
24,0	920,8	53,6
24,1	921,3	52,5
22,7	917,4	52,6
21,0	920,3	53,1
22,8	924,3	52,8
21,6	924,9	53,5
22,8	931,5	52,8
22,5	929,9	51,1

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1077	74,1	211,0	180,2
2	1,1071	55,3	157,4	187,2
3	1,1092	37,1	105,8	203,7
4	1,1027	7,5	21,3	671,8
5	1,1013	60,2	285,3	169,3
6	1,0984	44,4	210,5	156,7
7	1,0926	29,4	139,5	160,9
8	1,0940	0,1	0,6	9603,5

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,023	9,67	156	0,02	100	782
2	0,053	7,81	431	0,05	191	573
3	0,180	5,80	1601	0,18	998	246
4	0,169	2,45	1495	0,17	1187	117
5	0,089	12,50	791	0,09	90	934
6	0,016	8,17	96	0,02	150	925
7	0,024	5,53	167	0,02	139	676
8	0,035	1,35	271	0,04	189	205

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	20,32	1,40	17,3	94,3	17,8	91,6	99,72	75,02
2	24,30	1,67	18,4	94,2	19,0	91,6	199,45	149,72
3	31,02	2,13	19,6	94,2	20,1	91,7	199,92	150,10
4	68,24	4,69	20,4	94,2	20,9	91,7	200,40	150,42
5	16,42	1,13	20,4	94,0	21,0	90,8	85,43	65,06
6	23,39	1,61	20,6	94,4	21,3	91,5	182,73	137,49
7	33,07	2,27	20,9	94,2	21,6	91,3	200,02	150,38
8	126,77	8,71	21,1	94,2	21,7	91,4	200,03	150,33

P.M.
mg
0,435
0,611
2,284
2,669
2,849
0,702
0,467
0,362

E15

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2504	192,6	50,5	67,7	12,43	0,35
2	2506	142,6	37,4	50,1	10,42	0,01
3	2503	96,2	25,2	33,8	8,56	0,00
4	2507	18,1	4,7	6,4	4,39	0,01
5	1507	262,3	41,3	55,4	10,38	2,39
6	1506	190,5	30,0	40,2	6,95	0,22
7	1503	127,1	20,0	26,8	4,78	0,08
8	764	0,5	0,0	0,1	0,63	0,02

Tamb	Pamb	Humidity
°C	mbar	%
24,0	913,5	52,1
24,8	912,5	52,9
21,9	914,1	52,1
22,7	913,9	52,9
21,8	925,5	53,1
23,1	917,2	52,6
23,2	926,5	52,1
23,0	916,6	51,8

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1163	75,6	215,0	163,3
2	1,1191	56,1	159,6	178,4
3	1,1117	37,6	106,9	226,1
4	1,1136	7,1	20,2	475,8
5	1,0980	60,9	288,0	170,1
6	1,1103	44,7	211,5	152,3
7	1,0993	29,5	139,7	162,9
8	1,1108	0,1	0,6	10569,5

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,020	9,80	162	0,02	91	788
2	0,046	7,87	422	0,05	170	583
3	0,161	5,67	1498	0,16	751	265
4	0,149	2,52	1501	0,15	1087	122
5	0,112	12,53	1207	0,11	82	921
6	0,013	8,21	100	0,01	153	939
7	0,020	5,51	168	0,02	152	677
8	0,032	1,31	268	0,03	195	201

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	20,26	1,39	17,9	93,7	18,3	90,9	100,51	75,63
2	24,43	1,68	19,3	93,8	19,7	91,4	199,27	149,59
3	32,08	2,21	20,3	93,8	20,7	91,4	192,07	144,21
4	68,55	4,71	20,8	93,8	21,3	91,3	199,90	150,03
5	16,32	1,12	21,6	94,5	21,9	90,8	98,69	75,14
6	23,55	1,62	22,3	93,9	22,6	90,8	185,34	139,50
7	33,67	2,31	22,4	93,9	22,9	91,0	199,83	150,20
8	133,13	9,15	22,4	93,7	22,8	90,6	201,36	151,31

P.M.
mg
0,871
0,478
1,046
2,161
3,923
0,964
0,696
0,368

E20

Log No	SPEED	TORQUE	POWER	POWER	Fuel rate	Smoke
[1, 8]	RPM	Nm	kW	bhp	kg/h	
1	2500	184,5	48,3	64,7	12,30	0,21
2	2499	135,6	35,5	47,6	11,85	0,05
3	2498	87,0	22,7	30,5	11,07	0,01
4	2500	21,9	5,7	7,7	11,55	0,01
5	1503	251,7	39,6	53,1	9,99	1,53
6	1504	182,4	28,7	38,5	6,86	0,24
7	1503	120,0	18,9	25,3	4,81	0,07
8	734	0,4	0,0	0,0	0,60	0,07

Tamb	Pamb	Humidity
°C	mbar	%
21,7	917,7	51,6
23,4	917,1	52,4
21,8	913,1	52,0
23,9	915,9	49,1
24,4	924,0	49,7
27,6	921,9	49,3
22,6	922,0	49,6
23,9	928,2	49,8

Log No	Correction	Corrected		
[1, 8]	factor	bhp	Nm	g/hp.h
1	1,1071	71,7	204,3	189,7
2	1,1109	52,8	150,6	204,2
3	1,1128	33,9	96,9	342,0
4	1,1133	8,5	24,3	1336,4
5	1,1046	58,7	278,0	168,6
6	1,1130	42,8	203,0	162,5
7	1,1036	27,9	132,4	173,3
8	1,0986	0,0	0,4	13744,6

Log No	CO_H	CO2	CO_L	O2	THC	NOx
[1, 8]	%	%	ppm	%	ppm	ppm
1	0,025	9,32	177	0,03	99	808
2	0,083	8,37	706	0,08	200	469
3	0,249	7,16	2240	0,25	6833	221
4	0,316	4,57	2812	0,32	23479	70
5	0,060	12,06	513	0,06	112	925
6	0,016	7,95	108	0,02	169	930
7	0,029	5,35	224	0,03	173	688
8	0,040	1,34	321	0,04	224	202

Log No	AFR	λ	Tsamp	Psamp	Td.air	Pd.air	Vsamp	Vd.air
[1, 8]			°C	kPa	°C	kPa	litres	litres
1	21,00	1,44	20,1	93,9	20,2	91,4	99,76	75,02
2	22,89	1,57	20,4	93,9	20,8	91,3	200,08	150,26
3	25,74	1,77	21,0	93,9	21,5	91,3	200,51	150,61
4	36,93	2,54	21,0	93,8	21,6	91,9	100,00	75,00
5	16,86	1,16	22,0	94,5	22,3	90,9	98,48	74,84
6	24,05	1,65	22,6	94,1	22,9	91,0	200,07	150,53
7	34,43	2,37	22,5	94,0	22,9	91,1	199,18	149,72
8	129,34	8,89	22,5	94,0	22,8	91,4	199,36	149,83

P.M.
mg
0,286
0,363
12,994
10
2,322
1,08
0,729
0,5

APPENDIX C

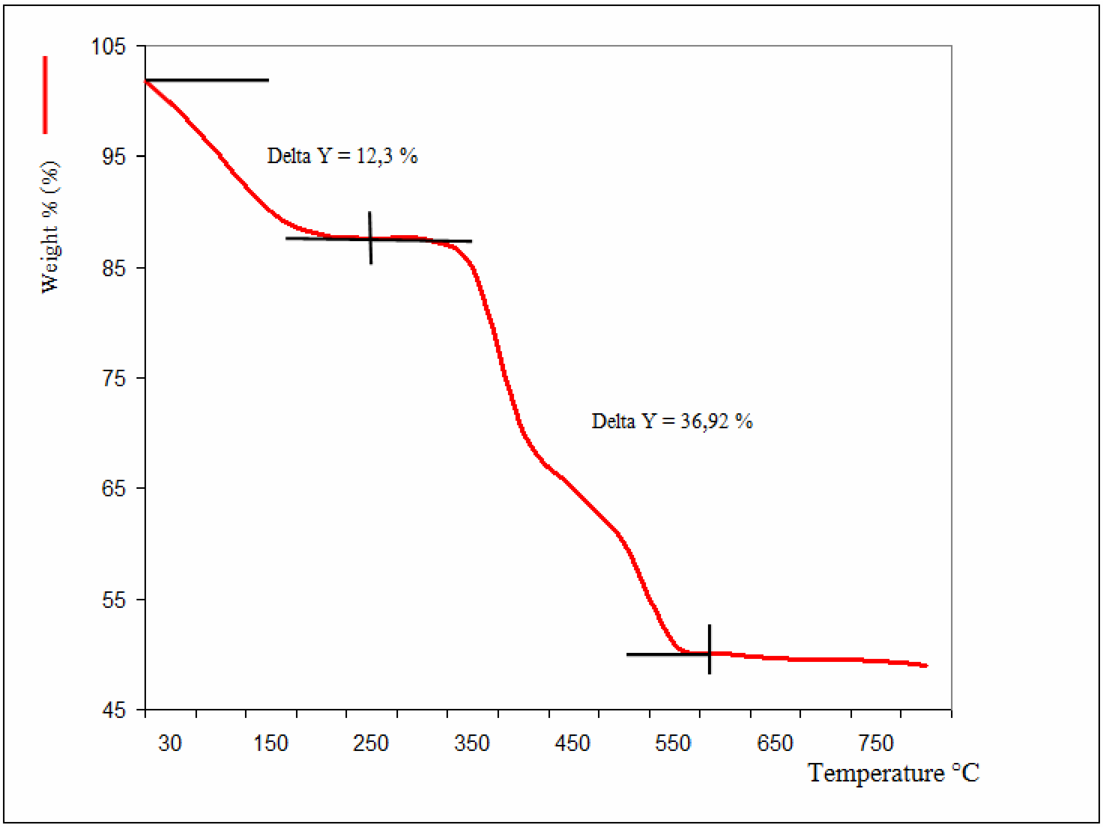
FILTER PHOTOS

1	**2**
	
3	**4**
	
5	**6**
	

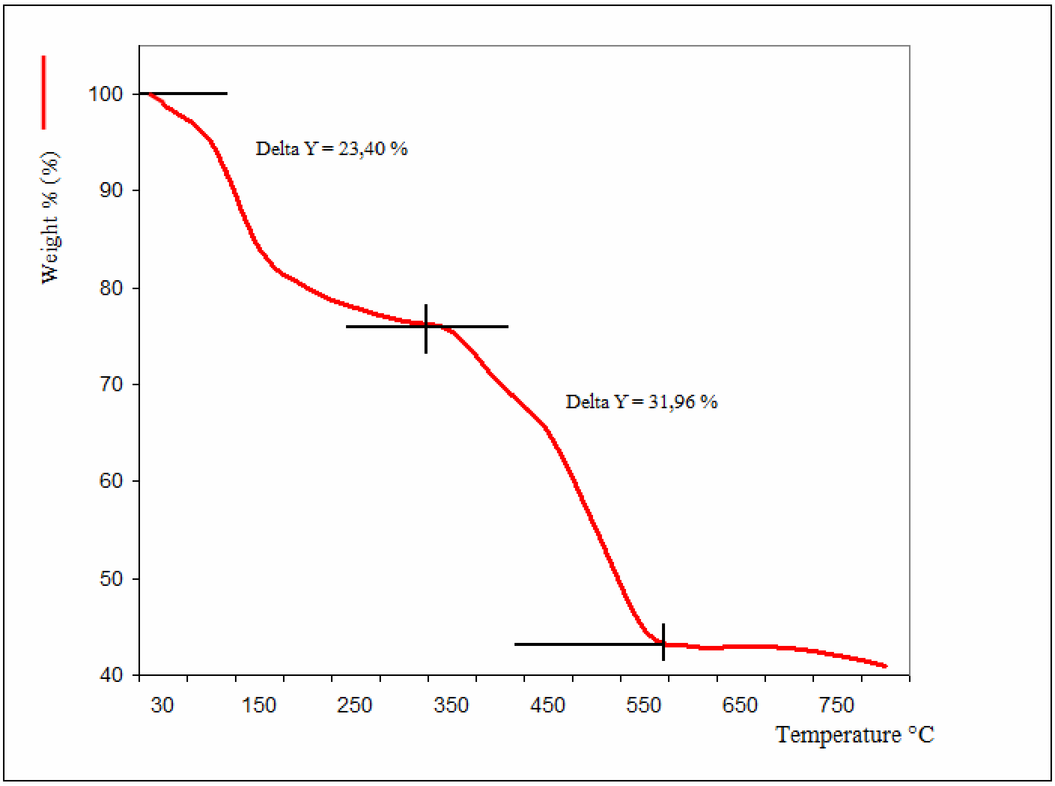
7	8
	

APPENDIX-D

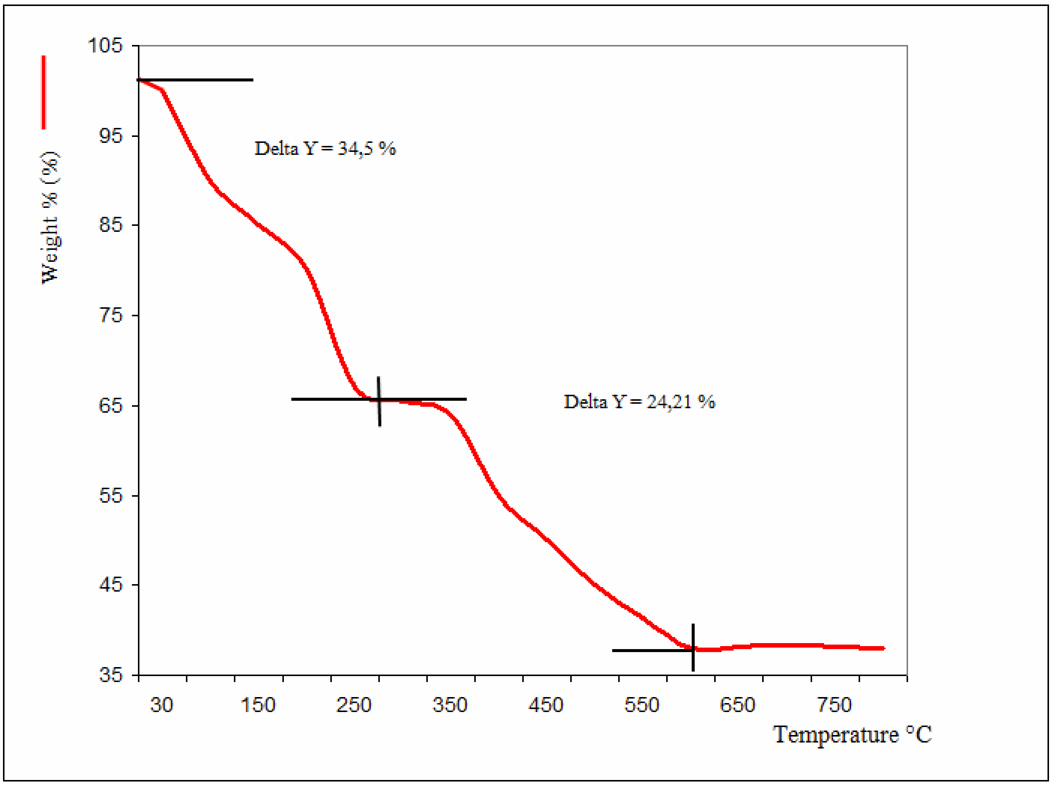
TGA RESULTS OF SAMPLES



TGA of DPF with diesel fuel for full load at 2500 rpm (Mode#1)



TGA of DPF with diesel + 10% ethanol fuel for full load at 2500 rpm (Mode#1)



TGA of DPF with diesel + 20% ethanol fuel for full load at 2500 rpm (Mode#1)

APPENDIX E

SAMPLE CALCULATION

Sample calculation is performed for first test with %100 Diesel fuel for Mod 1.

Ambient Temperature	: 22.2 °C ±0.01°C
Ambient Pressure	: 925.1 mbar
Relative Humidity	: %31.8
Engine Speed	: 2500 rpm
Engine Load	: %100
Torque	: 197.6 Nm
Fuel Consumption Rate	: 12.6 kg/h
Power	: 69.3 kW
Power	: 76.5 HP
Air/Fuel	: 19.5
Exh. Sample Inlet Pressure	: 94.8 kPa
Exh. Sample Temperature	: 21.7 °C
Exh. Sample Volume	: 99.4 liter
Dilution Air Temperature	: 22.1 °C
Dilution Air Inlet Presure	: 92.3 kPa
Dilution Air Volume	: 75 liters
CO	: 195 ppm
NOx	: 722 ppm
THC	: 83 ppm
PM	: 1.771 mg

Brake Power:

$$N_e = \frac{M_d \cdot n}{9550}$$

$$N_e = \frac{197.6x2500}{9550}$$

$$N_e = 51.727$$

Thermal Efficiency:

$$\eta_{th} = \frac{3600P_b}{Q_L \cdot m_{fuel}.1.341}$$

$$\eta_{th} = \frac{3600x76.5HP}{42000x1.341x12.6kg/h}$$

η_{th}= 38%

Brake Specific Fuel Consumption: (kg/HP-h)

$$BSFC = \frac{m_{fuel}}{N_e}$$

$$BSFC = \frac{12.6kg/h}{51.727}$$

$$BSFC = 0.243kg/KW-h$$

Emission Calculations

The exhaust gas flow

$G_{EXHW} = G_{AIRW} + G_{FUEL}$

G_{EXHW} =12.6+12.6x19.5

G_{EXHW}= 258.3 kg/h

The emission mass flow rates

Gas_{mass}= u x conc x G_{EXHW}

(NOx) mass = 0.001587x 722 ppm x 258.3 kg/h

(NOx) mass = 295.96

(CO) mass = 0.000966 x 195 ppm x 258.3 kg/h

(CO) mass = 48.655

(THC) mass = 0.000479 x 83 ppm x 258.3 kg/h

(THC) mass = 10.268

NO_X correction factor :

$A = 0.309 G_{FUEL} / G_{AIRD} - 0.0266$

$A = 0.309(1/19.5) - 0.0266$

A = -0,01075

$B = -209 G_{FUEL} / G_{AIRD} + 0.00954$

$B = -209(1/19.5) + 0.00954$

B = -0,00118

$$K_H = \frac{1}{1 + Ax(Ha - 10.71) + Bx(Ta - 298)}$$

$$K_H = \frac{1}{1 - 0.01075x(5.76 - 10.71) - 0.00118x(22.2 - 298 + 273)}$$

K_H = 0,946537

Calculation of the Specific Emissions

After calculating the emission mass flow rates for every mode,

$$Individua \lg as = \frac{\sum_{i=1}^{n} Gas_{mass,i} \, xWF_i}{\sum_{i=1}^{n} P_i xWF}$$

CO = 2.57 g/ kWh

THC = 0.43 g/ kWh

NO_X = 6.51 g/ kWh

Calculation of the Particulate Emission

Humidity Correction Factor for Particulates:

$$H_a = \frac{6.220 x R_a x p_a}{p_B - p_a x R_a x 10^{-2}}$$

$$H_a = \frac{6.220 x 31.8 x 2.672015}{92.5 - 2.672015 x 31.8 x 10^{-2}}$$

$$H_a = 5.76$$

$$K_p = 1/(1 + 0.0133 x (H_a - 10.71))$$

$$K_p = 1/(1 + 0.0133 x (5.76 - 10.71))$$

$$K_p = 1.070388$$

$$Msam = \frac{PV}{RT}$$

$$Msam = \frac{94.8 x (99.4/1000)}{0.287 x (21.7 + 273)}$$

Msam = 0,111475

$$Mair = \frac{PV}{RT}$$

$$Mair = \frac{92.3 x (75/1000)}{0.287 x (22.1 + 273)}$$

Mair = 0,08167

$$G_{TOTW,i} = \frac{Msam}{Vsam} + \frac{Mair}{Vair}$$

$$G_{TOTW,i} = \frac{0.111475}{99.4} + \frac{0.08167}{75}$$

$$G_{TOTW,i} = 2.21085$$

$$G_{DILW,i} = \frac{Mair}{Vdair}$$

$$G_{DILW,i} = \frac{0.08167}{75}$$

$$G_{DILW,i} = 0{,}000001$$

$$q_i = \frac{G_{TOTW,i}}{(G_{TOTW,i} - G_{DILW,i})}$$

$$q_i = \frac{2.21085}{(2.21085 - 0{,}000001)}$$

$$q_i = \frac{2.21085}{(2.21085 - 0{,}000001)}$$

$$q_i = 1{,}00000049$$

$$G_{EDFW,i} = G_{EXHW,i} x q_i$$

$$G_{EDFW,i} = 258.3 x 1{,}00000049$$

$$G_{EDFW,i} = 258.3 x 1{,}00000049$$

$$G_{EDFW,i} = 258.3$$

After completed all calculations for all modes; PT_{mass} can be calculated.

$$PT_{mass} = \frac{M_{f,i}}{M_{SAM,i}} x \frac{(G_{EDFW,i})_{aver}}{1000}$$

$$PT_{mass} = \frac{1.771}{0.111475} x \frac{258.5367}{1000}$$

$PT_{mass} = 4.107357$

$$PT = \frac{\sum_{i=1}^{n} PT_{mass,i} \, xWF_i}{\sum_{i=1}^{n} P_i \, xWF}$$

PT= 0.33

APPENDIX F

ERROR ANALYSIS

Measured Values:

• Ambient Temperature	: 22.2 °C	±0.1 °C
• Ambient Pressure	: 925.1 mbar	±0.1kPa
• Relative Humidity	: %31.8	0.3%
• Engine Speed	: 2500 rpm	±10rpm
• Engine Load	: %100	
• Torque	: 197.6 Nm	±1%
• Fuel Consumption Rate	: 12.6 kg/h	±2%
• Power	: 69.3 kW	±1%
• Power	: 76.5 HP	±1%
• Air/Fuel	: 19.5	0.3
• Exh. Sample Inlet Pressure	: 94.8 kPa	±0.1kPa
• Exh. Sample Temperature	: 21.7 °C	±0.1 °C
• Dilution Air Temperature	: 22.1 °C	±0.1 °C
• Dilution Air Inlet Presure	: 92.3 kPa	±0.1kPa
• CO	: 195 ppm	2 %
• NOx	: 722 ppm	2 %
• THC	: 83 ppm	3 %

Calculated Values:

- Brake Power:
- Thermal Efficiency:
- Brake Specific Fuel Consumption: (kg/HP-h)
- The exhaust gas flow
- The emission mass flow rates

(NOx) mass

(CO) mass

(THC) mass

- Specific Emissions

CO

THC

NO_X

Particulate Emission

Error Analysis:

Using, $z = f(x_1, x_2, x_3, \ldots x_n)$

and

$$\delta_z = \sqrt{\left(\frac{\partial z}{\partial x_1}\partial x_1\right)^2 + \left(\frac{\partial z}{\partial x_2}\partial x_2\right)^2 + \left(\frac{\partial z}{\partial x_3}\partial x_3\right)^2 + \ldots\ldots + \left(\frac{\partial z}{\partial x_n}\partial x_n\right)}$$

Error in Brake Power

$$N_e = \frac{M_d \cdot n}{9550}$$

$$\delta_{N_e} = \frac{1}{9550} x \sqrt{\left(nx\delta_{M_d}\right)^2 + \left(M_d x \delta_n\right)^2}$$

$$\delta_{N_e} = \frac{1}{9550} x \sqrt{(2500x0.2)^2 + (197.6x10)^2}$$

$\delta_{N_e} = 0.213$ for 51.727; Error: 0.41 %

Error in Brake Specific Fuel Consumption

$$\delta_{bsfc} = \sqrt{\left(\frac{\delta_{mfuel}}{N_e}\right)^2 + \left(\frac{-mfuelx\delta_{N_e}}{(N_e)^2}\right)^2}$$

$$\delta_{bsfc} = \sqrt{\left(\frac{0.03}{51.727}\right)^2 + \left(\frac{-12.6x0.213}{(51.727)^2}\right)^2}$$

$\delta_{bsfc} = 0.00116$ for 0.243 <u>Error: 0.47 %</u>

Error in Brake Thermal Efficiency

$$\delta_{bsfc} = 0.0639x\sqrt{\left(\frac{\delta_{N_e}}{m_{fuel}}\right)^2 + \left(\frac{-N_e x \delta_{mfuel}}{\left(N_{fuel}\right)^2}\right)^2}$$

$$\delta_{bsfc} = 0.0639x\sqrt{\left(\frac{0.213}{12.6}\right)^2 + \left(\frac{-51.727x0.03}{\left(12.6\right)^2}\right)^2}$$

$\delta_{bsfc} = 0.000605$ for 38 % ; <u>Error: 0.0015 %</u>

Error in Exhaust Gas Flow Rate

$$\delta_{GEXHW} = \delta mfuel + \sqrt{(A/Fx\delta_{mfuel}) + (mfuelx\delta_{A/F})}$$

$$\delta_{GEXHW} = 0.03 + \sqrt{(19.5x0.03)^2 + (12.6x0.3)^2}$$

$\delta_{GEXHW} = 3.855$ kg/h for 258.3 kg/h; <u>Error: 1.49 %</u>

Error in Emission Mass Flow Rate

$$\delta_{(NOx)mass} = 0.001587x\sqrt{(\delta_{GEXHW} xconc)^2 + (G_{EXHW} x\delta conc)^2}$$

$$\delta_{(NOx)mass} = 0.001587x\sqrt{(3.855x722)^2 + (258.3x0.03)^2}$$

$\delta_{(NOx)mass} = 4.42$ For (NOx) mass = 295.96; Error: %1.49

$$\delta_{(CO)mass} = 0.000966x\sqrt{(\delta_{GEXHW} xconc)^2 + (G_{EXHW} x\delta conc)^2}$$

$$\delta_{(CO)mass} = 0.000966x\sqrt{(3.855x195)^2 + (258.3x0.03)^2}$$

$\delta_{(COx)mass} = 0.725$ For (CO) mass = 48.655; Error: %1.48

$$\delta_{(THC)mass} = 0.000479x\sqrt{(\delta_{GEXHW}\,xconc)^2 + (G_{EXHW}\,x\delta conc)^2}$$

$$\delta_{(THC)mass} = 0.000479x\sqrt{(3.855x83)^2 + (258.3x0.03)^2}$$

$\delta_{(THC)mass} = 0.153$ For (THC) mass = 10.268; Eror: %1.49

Error in Specific Emissions

$$\delta_{(NOx)mass} = \sqrt{\left[\frac{\delta_{NOx}}{N_e}\right]^2 + \left[\frac{-NO_x\,x\delta_{Ne}}{(N_e)^2}\right]^2}$$

$$\delta_{(CO)mass} = \sqrt{\left[\frac{\delta_{CO}}{N_e}\right]^2 + \left[\frac{-COx\delta_{Ne}}{(N_e)^2}\right]^2}$$

$$\delta_{(THC)mass} = \sqrt{\left[\frac{\delta_{HC}}{N_e}\right]^2 + \left[\frac{-HCx\delta_{Ne}}{(N_e)^2}\right]^2}$$