Sudan University of Science and Technology

College of Graduate Studies

EXPERIMENTAL INVESTIGATION OF ETHANOL/WATER BLEND INJECTION ON EMISSION CHARACTERISTICS OF DIESEL ENGINES

دراسة مخبرية عن أثر حقن خليط الإيثانول/الماء على خصائص الإنبعاثات
فى ماكينات الديزل

A THESIS SUBMITTED IN FULFILLMENT FOR THE DEGREE OF DOCTOR OF PHILOSOPHY (PhD) IN MECHANICAL (POWER) ENGINEERING

Presented by **Mohammed Salih Mahjoub**

Supervisor I **Prof. Dr. Sabir Mohmmed Salih**

Supervisor II **Dr. Tagelsir Hassan Hussan**

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بِسْمِ اللَّهِ الرَّحْمَنِ الرَّحِيمِ

اقْرَأْ بِاسْمِ رَبِّكَ الَّذِي خَلَقَ (1) خَلَقَ الْإِنسَانَ مِنْ عَلَقٍ (2) اقْرَأْ وَرَبُّكَ الْأَكْرَمُ (3)

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(سورة العلق)

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Abstract

In the present study, a new technology controlled Ethanol/Water mix injection method is applied to a direct injection (DI) diesel engine to show effect on engine performance and emissions under steady operation. Ethanol/water mix is injected into the inlet manifold during inlet period. The experimental test matrix included six different fuels, namely Neat Diesel fuel and five secondary Ethanol/Water mix fuel with diesel fuel. The mixed fuels contain 0.55–2.79 by volume fraction of diesel fuel, corresponding to 0%, 25%, 50%, 75% and 100% by volume of ethanol in the mixes. Tests were performed in a single-cylinder naturally aspirated, four stroke, water cooled, and direct-injection diesel engine at variable engine speed (600-1600 rpm) on full engine load condition by using six-test fuels. It has been found that the application of Ethanol/Water mix leads to a significant reduction in the more environment concerning emissions of oxides of nitrogen (NOx) except increase at case pure ethanol, carbon dioxide (CO_2), hydrocarbon (UBHC) at case (EW50, EW75, Ethanol), and exhaust gas temperature. However, increase in carbon monoxide (CO), hydrocarbon (UBHC) at case (Water, EW25) and oxygen (O_2) emission have been found after use of Ethanol/Water. The results revealed that the brake specific fuel composition (BSFC), volumetric efficiency (VE) of both Ethanol/Water and diesel fuels reduces with increasing blend ratio, with a brake thermal efficiency (BTE) increase for both fuels, and a brake power (BP) increase. This can be attributed to the oxygen content of ethanol when compared with Neat Diesel. Ethanol/water injected diesel engine is modeled by using DIESEL-RK simulation software model for same experimental operation conditions. The obtained results are compared with conventional diesel engine in terms of NOx, CO_2 emissions, exhaust gas temperature, and performances. The simulation results agree with experimental data quite well. The results simulation show that Bosch smoke number, PM and summary emission of (PM and NOx) emissions increase after speed 1000 rpm with dual fuel water content decreasing, and the NO_2 emission under the simulated test decreases remarkably. With fuel ethanol content decreasing, brake means effective pressure (BMEP) descends distinctly comparing to the Neat Diesel fuel.

مُسْتَخْلَص

إتجهت كثير من الدول لتشريع قوانين تعمل على تقليل انبعاث غازات العادم من محركات الديزل إلى أدني مستوى لها، الهدف من هذه الدراسة معرفة مدى تأثير خليط إيثانول/ماء عن طريق مدخل الهواء مع الوقود الأساسي (الديزل) في محرك أحادى الإسطوانة رباعي الأشواط، بحيث تكون نسبة الخليط إلى الديزل (0.55-2.79) حجما"، ويستخدم الإيثانول/الماء بنسب 0%، 25%، 50%، 75%، 100% حجما". تمت هذه الدراسة في محرك ديزل أحادى الاسطوانة يتم التبريد للمحرك عن طريق الماء وضخ الديزل مباشرة لغرفة الاحتراق الرئيسية، عليه يتم تشغيل المحرك بالوقود الأساسي الديزل مع ضخ إيثانول/ماء عند مدخل الهواء للمحرك. اختيرت النقاط التشغيلية المعملية لتغطى المدى التصميمي للمحرك وبتحميل أقصي لكل سرعة. إذن الهدف الرئيسي من استخدام تكنولوجيا إضافة خليط إيثانول/ماء مع وقود الديزل لدراسة تأثيرها على الغازات الناتجة من الإحتراق وهي أكاسيد النتروجين و أول أكسيد الكربون والهيدركربونات الغير محترقة، الأهداف المصاحبة وهي القدرة النتاجة وكفاءة المحرك والاستهلاك النوعي للوقود ودرجة حرارة غازات العادم ومعامل الهواء الزائد من عملية الاحتراق، تم قياس الانبعاثات وأداء المحرك في حالة تشغيل الخليط مع الديزل ومقارنتها مع وقود الديزل. أظهرت النتائج المعملية انخفاض في اكاسيد النيتروجين و الهيدركربونات والاستهلاك النوعي للوقود مع زيادة في معامل الهواء الزائد وأول أكسيد الكربون. هذه التجارب تمت بتعديل لا يذكر في محرك الديزل مع تحقيق نتائج كبيرة جدا" لتقليل الإنبعاث من محرك الديزل وزيادة الأداء في المحرك.

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List of Abbreviations

$CV_{LHV,d}$	The Lower Heating Value of Diesel Fuel
$CV_{LHV,e}$	The Lower Heating Value Of Ethanol
$m°_d$	Mass Consumption Rates of Diesel Fuel
$m°_e$	Mass Consumption Rates of Ethanol Fuel
A	Operation by Diesel Fuel with Chemical Pump(0.96 L/hr.@6 bar)
ASCC	American Society of Concrete Contractors
B	Operation by Diesel Fuel with Chemical Pump(1.12 L/hr.@6 bar)
BMEP	Brake Mean Effective Pressure
Bosch	Bosch Smoke Number
BP	Brake Power
BSFC	Brake Specific Fuel Consumption
BTE	Brake Thermal Efficiency
C	Operation by Diesel Fuel with Chemical Pump(1.85 L/hr @10 bar)
CI	Compression-Ignition Engine
CO	Carbon Monoxide
CO_2	Carbon Dioxide
D	Operation by Diesel Fuel with Chemical Pump(2.16 L/hr. @10 bar)
DI	Direct Injection
EGR	Exhaust-Gas Recirculation
EGT	Exhaust Gas Temperature
EPA	Environmental Protection Agency
EW25	25/75 Ethanol/Water Mixture By Vol.
EW50	50/50 Ethanol/Water Mixture By Vol.
EW75	75/25 Ethanol/Water Mixture By Vol.
NAAQS	National Ambient Air Quality Standards
NO_2	Nitrogen Dioxide
NOx	Nitrogen Oxides
O_2	Oxygen

PM	Particulate Matter
ppm	Parts Per Million
SE	Summary Emission Of PM and NOx
UBHC	Unburnt Hydrocarbon
VE	Volumetric Efficiency
λ	Excess Air Coefficient

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Appendix A: Properties of Water

Appendix B: Properties of Fuels

Appendix C: Specific Exhaust Gas Analyzer SV-5Q

CHAPTER ONE

Introduction

1. Introduction

Diesel engines have been widely used as power of engineering machinery, automobile, and shipping equipment for its excellent drivability and thermal efficiency. At the same time, diesel engines are major contributors of various types of air pollutant emissions such as Carbon monoxide (CO), Oxides of Nitrogen (NOx), Particulate matter (PM), and other harmful compounds [1]. Alcohols can be used in compression ignition (CI) engines as pure or blended with conventional diesel fuel [2].

1.1. Background and Context

The advantages of alcohols as a fuel include:

– Low viscosity compared to diesel fuel, therefore it can easily be injected, atomized and mixed with air.

– Less emission because of its high stoichiometric fuel–air ratio, high oxygen content, high H/C ratio and low sulfur content.

– High evaporative cooling, which results in a cooler intake process and compression stroke. This raises the volumetric efficiency of the engine and reduces the required work input in the compression stroke.

– High laminar flame propagation speed, which may make combustion process finish earlier, thus improve engine thermal efficiency [2].

In addition, ethanol (C_2H_5OH) is a pure substance. It contains an oxygen atom and can thus be viewed as a partially oxidized hydrocarbon. As a fuel for CI engines, ethanol has certain advantages over diesel fuel including reductions of soot, carbon monoxide (CO) and unburned hydrocarbon (HC) emissions. Unfortunately, ethanol is currently unable to be used extensively due to limitations in technology and economic and regional considerations [3, 4].

In order to use the ethanol in diesel engines, various techniques were developed such as blending [5–11], emulsification [12–14], fumigation [15–19] and dual injection [20, 21]. Separate injection systems are required for each fuel in dual injection systems. In the fumigation systems, secondary fuel is added to the intake air. The advantages of injection are versatile of on-line variation of water quantity, increase of volumetric efficiency due to cooling effect, uniform or homogeneous water distribution in combustion chamber, etc. [22].

Fumigation is where liquid water is injected into the intake manifold upstream of the intake valve. The fumigation technique has been shown to reduce NOx emissions in DI Diesel applications but suffers from the drawback that the liquid water in the combustion chamber is typically in areas where it is less effective at reducing emissions. Therefore, fumigation requires approximately twice the liquid volume for the same reduction in engine out NOx when compared to direct water injection. Additionally, liquid water present after combustion can contaminate the oil and increase engine wear [23].

The use of ethanol–diesel blends in diesel engines without any modifications negatively affects the engine performance and NOx emissions. However, steam injection method decreases NOx emissions and improves the engine performance [24].

Lean ethanol-water/air mixtures have potential for reducing NOx and CO emissions in internal combustion engines, with little well-to-wheels CO_2 emissions. Conventional ignition systems have been unsuccessful at igniting such mixtures [25]. Ongoing work with methanol- and ethanol-fueled engines at the EPA's National Vehicle and Fuel Emissions Laboratory has demonstrated improved brake thermal efficiencies over the baseline diesel engine and low steady state NOx, UBHC and CO, along with inherently low PM emissions. In addition, the engine is expected to have significant system cost advantages compared with a similar diesel, mainly by virtue of its low-pressure port fuel injection (PFI) system. While recognizing the considerable challenge associated with cold start, the alcohol-fueled engine nonetheless offers the advantages of being a more efficient, cleaner alternative to gasoline and diesel engines [26].

Injection timing in a turbocharged six cylinder direct injection engine using direct injection of ethanol/water mixes of varying water contents into the inlet port of each cylinder would be fixed at a given speed, for all loads. At high load, thermal efficiency increased with increased ethanol flow rate, for all values of ethanol/water ratio. The maximum gain in thermal efficiency was about 5% of the diesel fuel, only efficiency at the highest load test point at 8 bar BMEP. At intermediate load, thermal efficiency was unchanged. At low loads, thermal efficiency decreased with increased ethanol addition, down by up to 25% of the diesel fuel only value, at maximum ethanol flow rate. The ethanol/water ratio had little effect on this trend. At 8 bar BMEP NO emissions decreased with increased water content and fumigation rates, but a small increase in NO emissions was measured for ethanol/water mixtures greater than 68% by mass [27].

1.2. Statement of the Problem

As a general conclusion from the study of the specialized literature, one could say that the published research on the performance and emissions of modern, diesel engines with medium-to high percentage of water blending is limited. This observation was the main motivation factor for the first part of this thesis.

As a second conclusion from the literature review, the published research on the performance and emissions of diesel engines run on ethanol blends is also limited. This observation was the main motivation factor for the second part of this thesis.

As a third conclusion from the literature search, the published research on the performance and emissions of diesel engines run on ethanol/water mixture is also limited. This observation was the main motivation factor for the third part of this thesis.

1.3. Objectives and Scope of Work

Overall, very little research has conducted to date on the effects of intake ethanol/water content variation caused by altitude level change and fuel ethanol/water content on diesel engine performance and emissions. The purposes of this work are to explore different effects of intake ethanol/water content caused by altitude variation and fuel ethanol/water content on diesel engine performance and emissions with the effects of different percentage of ethanol/water fuels on diesel engine under high altitude conditions, and to provide a theoretical foundation for the application and promotion of ethanol oxygenated alternative fuels in high-altitude regions.

1.4. Research Approach

As far as the above cause and strategy are concerned, this thesis was planned to consist of two parts:

[1] Investigate the engine combustion and emissions characteristics by Neat Diesel with intake manifold fuel blends of ethanol/water fuel, ethanol, and water on a diesel engine. The Brake power, Brake thermal efficiency, Brake specific fuel consumption, Volumetric efficiency, Excess air coefficient, Exhaust Gas Temperature, regulated emissions, including, NOx, CO, and UBHC, CO_2, O_2, are investigated and discussed. The results are compared to the relatively scarce works reported in the specialized literature for the specific engine category and similar intake manifold blending rates.

[2] The effect of the performance and emissions on the diesel engine is further investigated by means of ethanol, ethanol/water, and water on DIESEL-RK simulation software. The purpose of this is to evaluate the effect of the addition of different percentage ethanol/water on base diesel fuel properties, as well as the effect of these compounds on emissions (NOx, Specific NOx emission reduce to NO_2, PM, Summary emission of PM and NOx, CO_2, Bosch Smoke Number) from an indirect injection engine. Also effects of engine performance (Specific Fuel Consumption, Power Engine, Efficiency of Engine, Brake Mean Effective Pressure, Exhaust Manifold Gas Temperature) are investigated.

1.5. Thesis Structure

The thesis consists of six chapters including introduction. In the chapter 2, literature of exhaust gas characterization and emission control techniques, rules and regulations for diesel engines and its mechanism. In particular the chapter gives more attention to the methods of reductions emissions. In the chapter 3, research methodology of the diesel engine is introduced and explained on Thermodynamic Laboratory at the University of The King Saud. It covers description of the state of the art and the methods implemented in the thesis. Chapter 4 includes result and discussion of the experimental tests engine performance and emissions for the Neat Diesel fuel, Ethanol/Water. Chapter 5 discusses the engine test effects in the laboratory and the results that are compared to the simulations output. The last chapter is the conclusion and recommendation for future work.

CHAPTER TWO

Exhaust Gas Characterization and Emission Control Techniques

2. Literature of Exhaust Gas Characterization and Emission Control Techniques

2.1. Preface

The CO, NOx, UBHC, and PM emissions in the exhaust of internal combustion engines are of direct concern from a human health perspective. In addition, evaporative emissions of hydrocarbons and other volatile components of fuels have gained in relative importance as the exhaust emissions have been reduced by emission control technology. While these emissions are present with all fuels used in vehicles, the relative emission rates and their composition can change significantly from one fuel to another. Ethanol's introduction as a clean alternative fuel is mostly related to the reductions in CO and exhaust UBHC that can be realized. Ethanol has higher volatility than diesel so that it will tend to increase evaporative emissions when blended with diesel unless the composition of the diesel itself is changed to counteract this increase in volatility. This can be done by reducing the fraction of lighter hydrocarbons like butane in the diesel. On the other hand evaporative emissions of ethanol itself are not of particular concern at the ambient concentration that is encountered.

In terms of individual HC and other volatile compounds, attention has focused on the following list of compounds in the "Air Toxics" group, so named because they are known or suspected to be hazardous to human health. Benzene 1,3 butadiene, formaldehyde, acetaldehyde, acrolein, MTBE (methyl tertiary butyl ether) Formaldehyde and acetaldehyde are of particular interest from an Ethanol fuel perspective as these compounds form directly from the combustion of the Ethanol (formaldehyde from methanol, acetaldehyde from ethanol), resulting in higher emissions, while the emissions of the others are generally reduced relative to diesel. The CO_2 emitted from the combustion of ethanol fuel is not a direct human health hazard but like all other emissions of CO_2, contribute to global climate change effects. As indicated above, the use of ethanol from biomass (e.g. corn) can reduce the amount of CO_2 emitted per unit energy derived from the fuel [28].

2.2. Emission of Diesel Engine

2.2.1. Carbon Monoxide Emissions

CO results from incomplete combustion in rich air/fuel mixtures due to an air deficiency. Although carbon monoxide is also produced during operation with excess air, the concentrations are minimal, and stem from brief periods of rich operation or inconsistencies within the air/fuel mixture. Fuel droplets that fail to vaporize from pockets of Rich mixture that do not combust completely. CO is an odorless and tasteless gas. In humans, it inhibits the ability of the blood to absorb oxygen, thus leading to asphyxiation [29].

CO is a colorless, odorless and toxic gas. It blocks the lungs' ability to obtain oxygen. CO is produced by incomplete combustion of fossil fuels and is a major part of air pollution. Compression ignition (diesel) engines generate significantly lower CO emissions than spark ignited engines, but reductions are easy to achieve and should be pursued [30].

2.2.1.1. Carbon Monoxide Formation

CO emissions from Diesel Engines are a concern from toxicological effects on humans. The formation of CO from the combustion of the hydrocarbon radical, R, is as follows:

$$RH \rightarrow R \longrightarrow RO_2 \longrightarrow RCHO \longrightarrow RCO \longrightarrow CO \text{ ------- } (2.1)$$

Once formed, CO is slow to oxidize to CO, with water providing the primary oxidant source through the following steps:

$$CO + O_2 \longrightarrow CO_2 + O \text{ ----------------- } (2.2)$$

$$O + H_2O \longrightarrow 2OH \text{ --------------------- } (2.3)$$

$$CO + OH \longrightarrow CO_2 + H \text{ ----------------- } (2.4)$$

$$H + O_2 \longrightarrow OH + O \text{ --------------------- } (2.5)$$

The reaction in Equation (2.2) is slow, with primary oxidation of CO occurring through Equation (2.4), with Equation (2.5) producing OH radicals feeding back to Equation (2.4). Diatomic hydrogen (H_2) can also provide oxidation of CO through formation of H_2O, but H_2 is not formed in sufficient quantities in compression engines combustion to contribute to CO oxidation [31].

CO is a colorless, odorless, and (at much higher levels) poisonous gas, formed when carbon in fuels is not burned completely. It is a product of motor vehicle exhaust, which contributes about 60 percent of all CO emissions nationwide. High concentrations of CO generally occur in areas with heavy traffic congestion. In cities, as much as 95 percent of all CO emissions may emanate from automobile exhaust. Other sources of CO emissions include industrial processes such as carbon black manufacturing, non-transportation fuel combustion, and natural sources such as wildfires. Woodstoves, cooking, cigarette smoke, and space heating are sources of CO in indoor environments. Peak CO concentrations typically occur during the colder months of the year when CO automotive emissions are greater and nighttime inversion conditions are more frequent.

CO is a colorless, odorless gas emitted from combustion processes. Nationally and, particularly in urban areas, the majority of CO emissions to ambient air come from mobile sources. CO can cause harmful health effects by reducing oxygen delivery to the body's organs (like the heart and brain) and tissues. At extremely high levels, CO can cause death.

EPA first set air quality standards for CO in 1971. For protection of both public health and welfare, EPA set an 8-hour primary standard at 9 parts per million (ppm) and a 1-hour primary standard at 35 ppm. In a review of the standards completed in 1985, EPA revoked the secondary standards (for public welfare) due to a lack of evidence of adverse effects on public welfare at or near ambient concentrations. The last review of the CO NAAQS was completed in 1994 and the Agency chose not to revise the standards at that time [32].

The Clean Air Act requires EPA to set national ambient air quality standards for “criteria pollutants.” Currently, carbon monoxide and five other major pollutants are criteria pollutants. The others are ozone, lead, sulfur oxides, nitrogen oxides, and particulate matter. The law also requires EPA to review the standards periodically and revise them if appropriate to ensure that they provide the requisite amount of health and environmental protection and to update those standards as necessary [32]. Everywhere in the country has air quality that meets the current CO standards. Most sites have measured concentrations below the national standards since the early 1990s, since which time, improvements in motor vehicle emissions controls have contributed to significant reductions in ambient concentrations [32]. CO can cause harmful health effects by reducing oxygen delivery to the body's organs (like the heart and brain) and tissues. At extremely high levels, CO can cause death [32].

2.2.2. Carbon Dioxide Emissions

When CO_2 is inhaled in elevated concentrations may act to produce mild narcotic effects, stimulation of the respiratory system and asphyxiation depending on the concentration present and the duration of exposure. Other effects known to occur in humans at high exposures of CO_2 include changes in sensory perceptions, disturbed judgments and mood changes [33].

The current ASCC 8-hour time weighted average exposure standard for coal mines is 12500 ppm. It should be noted that this exposure standard is based on both health and practicability considerations and thus should be administered cautiously. For non-coalmines an exposure standard of 5000ppm has been adopted [33]. Carbon dioxide is a colorless, odorless, non-toxic gas that is one of main products of fossil-fuel combustion. It is a greenhouse gas that contributes to the global warming [30].

2.2.3. Oxides of Nitrogen Emissions

NOx, or oxides of nitrogen, are the generic term embracing chemical compounds consisting of nitrogen and oxygen. They result from secondary reactions that occur in all combustion processes where air containing nitrogen is burned. The primary forms that occur in the exhaust gases of internal-combustion engines are nitrogen oxide (NO) and nitrogen dioxide (NO_2), with dinitrogen monoxide (N_2O) also present in minute concentrations. NO is colorless and colorless. In atmospheric air, it is gradually converted to NO_2. Pure NO_2 is a poisonous, reddish-brown gas with a penetrating odor. NO_2 can induce irritation of the mucous membranes when present in the concentrations found in highly-polluted air. Nitrous oxides contribute to forest damage (acid rain) and also act in combination with hydrocarbons to generate photochemical smog [29].

NOx emissions in CI engines form from high cylinder temperatures, and air-rich conditions. Diffusion flames burn at the stoichiometric boundary of air and fuel, creating near-adiabatic flame temperatures close to air molecules [34]. NOx forms from the dissociation and re-association of nitrogen and oxygen molecules. Air is mostly composed of molecular nitrogen (N_2) and molecular oxygen (O_2). In simplified terms, at high temperatures, the O_2 dissociates into O, and the N_2 dissociates into N. Through different mechanisms, the O and N eventually re-associate to form NO. As the cylinder temperatures decrease, due to the expanding chamber volume and subsiding combustion, NO further oxidizes to NO_2. However, there is less time for the NO to fully oxidize to NO_2, so most engine emissions are largely composed of NO, with some amounts of NO_2. Once the NO enters the atmosphere, it eventually after time fully oxidizes to NO_2 (and can contribute to smog formation and acid rain formation) [34].

Several air-polluting gases composed of nitrogen and oxygen which play an important role in the formation of photochemical smog. Nitrogen oxides are collectively referred to as “NOx”, where “x” represents a changing proportion of oxygen to nitrogen. Internal combustion engines are significant contributors to the worldwide ambient NOx levels. For the purpose of emission regulations, NOx is composed of colorless nitric oxide (NO), and the reddish-brown, very toxic and reactive nitrogen dioxide (NO_2). Other nitrogen oxides, such as nitrous oxide N_2O (the anesthetic “laughing gas”), are not regulated emissions [30].

2.2.4. Oxides of Nitrogen Formation

NOx is referred to here as mixtures of nitric oxide (NO) and nitrogen dioxide (NO_2). NOx emissions are controlled because NO and NO_2 contribute to the formation chemistry of low-level ozone, or smog, an environmental and human health hazard. NO_2 is also directly of concern as a human lung irritant.

NOx can also be defined to include other oxides of nitrogen, including N_2O, NO_3, N_2O_4, and N_2O_5. These additional nitrogen oxide species are insignificant in the emissions from IC engines, and readily react to NO and NO_2. NO generally accounts for over 90 percent of the total NOx emissions from fossil fuel combustion, with the remainder being NO_2. The formation of NO can be explained by three different mechanisms [31]:

1) The Extended Zeldovich mechanism, or thermal NO, in which O, OH, and N_2 are in equilibrium concentrations.

2) Other mechanisms with NO formation rates above that predicted by the Extended Zeldovich mechanism, including:

- Fenimore CN and HCN pathways.
- N_2O-intermediate route.
- "Super-equilibrium" concentrations of (O) and (OH) in combination with the Extended Zeldovich mechanism.

3) Fuel nitrogen mechanism, in which fuel-bound nitrogen is oxidized to (NO).

The primary pathway for (NO) formation is oxidation of atmospheric molecular nitrogen (N_2) through the thermal or Zeldovich mechanism:

$$O + N_2 \leftrightarrow NO + N \text{ -------------- } (2.6)$$

$$N + O_2 \leftrightarrow NO + O \text{ -------------- } (2.7)$$

Extending the thermal NO formation mechanism to include the hydroxyl radical reaction with nitrogen was proposed by Lavoie et al (1970):

$$N + OH \leftrightarrow NO + H \text{ ------------- } (2.8)$$

Thermal NO formation rate is slow relative to combustion and is considered unimportant below 1800 K. Thermal NO formation attributed to Equations (2.6) through (2.8) is considered formed in the post-combustion exhaust gases.

Prompt NO, also referred to as the Fenimore mechanism, is NO that is quickly formed in the premixed laminar flame before thermal NO has formed. Hydrocarbon radicals react with molecular nitrogen to create hydrogen cyanide as an intermediate to NO formation in the following steps:

$$CH + N_2 \leftrightarrow HCN + N \text{ ----------- } (2.9)$$

$$HCN + O \leftrightarrow NCO + H \text{ --------- } (2.10)$$

$$NCO + H \leftrightarrow HN + CO \text{ ---------- } (2.11)$$

$$NH + H \leftrightarrow N + H_2 \text{ -------------- } (2.12)$$

$$N + OH \leftrightarrow NO + H \text{ ------------- } (2.13)$$

Prompt NO formation is also considered insignificant in internal combustion engines due to the thin flame fronts, short residence times, and high pressures in the combustion chamber.

The formation of NO through an N_2O intermediate mechanism is important in fuel-lean (ϕ<0.8), lower temperature conditions (T < 1800 K). The three steps are:

$$O + N_2 + M \leftrightarrow N_2O + M \text{ ------- } (2.14)$$

$$H + N_2O \leftrightarrow NO + NH \text{ --------- } (2.15)$$

$$O + N_2O \leftrightarrow 2NO \text{ ------------- } (2.16)$$

The M in Equation (2.14) represents a third body collision molecule. The significance of the N_2O intermediate mechanism can be seen in Equation (2.16) where two moles of NO are formed per mole of N_2O. While our subject engine operates at ϕ = 0.8, the flame temperature is slightly higher than 1800 K, so N_2O intermediate pathway formation of NO is probably not significant.

Fuel-bound nitrogen is another source of combustion NO emissions. This process is significant in coal combustion, where bituminous coal contains up to 2% by mass bound nitrogen. Nitrogen in the fuel is quickly reacted to HCN or ammonia,

NH_3, and follows the reaction steps beginning with Equation (2.10) for prompt NO formation. Kerosene and gasoline fuels contain trace to zero quantities of nitrogen, so fuel-bound nitrogen contribution to NO formation is not considered significant in internal combustion engines. The final reaction mechanism considered here for NOx formation is production of NO_2. Reactions contributing to the formation and destruction of NO_2 are as follows:

$$NO + HO_2 \leftrightarrow NO_2 + OH \text{ (Formation) ---------- (2.17)}$$

$$NO_2 + H \leftrightarrow NO + OH \text{ (Destruction) ----------- (2.18)}$$

$$NO_2 + O \leftrightarrow NO + O_2 \text{ (Destruction) ------------- (2.19)}$$

The HO_2 radicals form in low-temperature regions leads to NO_2 formation through Equation (2.17). NO_2 destruction via reaction with the H and O radicals are active at high temperatures [Equations (2.18) and (2.19)]. Thus NO formation from NO_2 would be preferred at high temperatures and NO_2 would only survive during low-temperature cooling of exhaust gases. This validates the previous statement that most of the NOx emitted from internal combustion engines is NO.

Heywood (1988) reports the following relationship for NO formation rate based upon empirical data and the assumption of equilibrium concentrations of O, O_2, OH, H, and N_2, by decoupling the NO formation from combustion (i.e., assuming NO formation in post-combustion gases always dominates NO produced in the flame):

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{\sqrt{T}} e^{[\frac{-69,090}{T}]} [N_2]e \sqrt{[O_2]e} \text{ ------- (2.20)}$$

Where [ ]e denotes equilibrium concentrations. The significant dependence of temperature on NO formation rate in Equation (2.20) is evident [31].

2.2.5. Unburnt Hydrocarbon Emissions

UBHC, is a generic designation for the entire range of chemical compounds uniting hydrogen H with carbons C. UBHC emissions are the result of inadequate oxygen being present to support complete combustion of the air/fuel mixture. The combustion process also produces new hydrocarbon compounds not initially present in the original fuel (by separating extended molecular chains, etc.). Aliphatic hydrocarbons (alkanes, alkenes, alkines and their cyclical derivatives) are virtually odorless. Cyclic aromatic hydrocarbons (such as benzol, toluol and polycyclic hydrocarbons) emit a discernible odor. Some hydrocarbons are considered to be carcinogenic in long-term exposure. Partially oxidized hydrocarbons (aldehydes, ketones, etc.) emit an unpleasant odor. The chemical products that result when these substances are exposed to sunlight are also considered to act as carcinogens under extended exposure to specified concentrations [29].

The presence of un-burnt fuel and partial combustion products in diesel emissions gives rise to numerous aliphatic and aromatic hydrocarbons in the exhaust. As the level of hydrocarbons present is dependent on engine tune and the completeness of combustion it is not possible to state with certainty which hydrocarbons will be present. It is known that some high molecular weight aromatic hydrocarbons (polynuclear or polycyclic aromatic hydrocarbons or PAHs) give rise to potentially serious adverse health effects however specific exposure standards for individual compounds have not been established. In this circumstance it is prudent to minimize employee exposures [33]. Incomplete combustion of fossil fuels results into an exhaust or evaporative pollutant of hydrogen and carbon atoms in various chain lengths [30].

2.2.6. Particulate Matter

The problem of particulate emissions is primarily associated with diesel engines. Levels of particulate emissions from gasoline engines with multipoint injection systems are negligible. Particulates result from incomplete combustion. While exhaust-gas composition varies as a function of the combustion process and engine operating condition, these particulates basically consist of hydrocarbon chains (soot) with an extremely extended specific surface ratio. Un-combusted and partially combusted hydrocarbons form deposits on the soot, where they are joined by aldehydes, with their penetrating odor. Aerosol components (minutely dispersed solids or fluids in gases) and sulfates bond to the soot. The sulfates result from the sulfur contents in the fuel. Consequently, these pollutants do not occur if sulfur-free fuel is used [29].

Soot emissions (particulate matter) in CI engines form from poorly aerated fuel droplets not able to fully oxidize during the cycle. While increased temperatures lead to higher NOx emissions, they also lead to lower soot emissions. There is a tradeoff. Soot forms when the liquid fuel (injected before the piston reaches TDC) is not able to completely vaporize (and thus oxidize) during the cycle. A vapor barrier forms around each fuel droplet. At high temperatures, the fuel vapor burns, slowly vaporizing the liquid fuel droplet. However, the surrounding temperatures are quickly dropping due to the expanding combustion volume, and the flame around the liquid droplet quenches. The quenched flame leaves a very small, charred, spherical drop of unburned fuel. This charred droplet leaves the chamber as a soot particle [34].

2.2.6.1. Soot Formation

The high level of soot formation during the combustion process is a severe problem associated with Diesel engines. The Diesel engine, because of its heterogeneous combustion, produces more soot than a SI engine. Reported soot formation in a Diesel engine to be 50 to 100 times more than that of a comparable SI engine. Heterogeneous combustion produces a diffusion flame and across any plane through diffusion flame there is a wide variation of the fuel oxidizer ratio from very fuel-rich to very fuel-lean. Thus in a diffusion flame there is always a zone very close to the flame that is at a high temperature which has a very high carbon to oxygen ratio. This characteristic of a diffusion flame is the reason that they always have some luminosity and form soot relatively easily. In all flames, the lower the pressure, the lower the tendency to soot. In a review of existing data indicates the temperature range of interest for soot formation in the flame is approximately 2000 K to 2400 K. The peak concentration of soot in the flame occurs near 2100 K.

At both ends of the range, i.e. 2000 K and 2400 K, the soot concentration is negligible. He also mentions that fuels with high H/C ratios produce less exhaust soot than fuels with low H/C ratios. Usually the required conditions to form soot in a flame are at least two: **(1)** rich fuel-oxygen mixture and **(2)** a temperature of at least

2000 K. In addition to these two factors, showed fuel properties have significant effect on the total particulate emissions and its soluble organic fraction (SOF). They showed higher distillation temperature, lower API gravity and higher aromatics content cause higher levels of particulate matter emissions.

Formation of soot in a Diesel engine has an effect on measuring NO because of the physical and chemical adsorption of NO by the soot. Therefore, a shorter sample line and also a high flow rate which reduce the retention time for NOx minimize the error in the measuring of NOx [35].

2.3. Mechanism of Emissions Formation

The majority of exhaust gas, 99.9% (by volume) from a typical diesel fuelled CI engine consists of harmless components, such as nitrogen, carbon dioxide, water vapor and oxygen. Gaseous components, such as UBHC, CO, and NOx, as well as PM, are regulated. All other exhaust components, individual components and particle properties, except for CO, are not regulated by law. Emissions can be composed of various individual exhaust gas compounds, which can be more or less harmful to the environment and to health, depending on conditions, such as fuel, engine setting and exhaust gas after treatment [36].

According to Heywood, the combustion process in a CI engine can be divided into four phases. The first phase, an ignition delay period, starts after the initial injection of the fuel and continues until the initiation of combustion. The delay period is governed by the rate of fuel and air mixing, diffusion, turbulence, heat transfer, chemical kinetics, fuel vaporization, and fuel composition. Fuel cetane rating is an indication of ignition delay.

The second phase extends over the rapid premixed burning of the mixture between fuel and air from the ignition delay period. The third phase, diffusion-controlled burning, where the fuel burns as it is injected and diffuses into the cylinder. The final phase includes a small amount of rate-controlled burning during the expansion stroke after the end of injection. During these phases, the chemical energy stored in the fuel is released as heat of oxidation to ideally, yield CO_2, water and energy. This is not an ideal process, therefore, particles, NOx and other exhaust gas components are formed. Initial particle formation occurs primarily during the diffusion-burn phase of combustion and is highest during high load and other conditions consistent with high fuel-air ratios. Depending on the time available for combustion and the availability of oxygen, the fuel droplets are either completely or partially oxidized. At high temperatures, a large part of the unburned fuel that is not oxidized is pyrolised, i.e., stripped of hydrogen, to form carbonaceous particles. Depending on the kind of fuel, pyrolysis may be facilitated as ordinary diesel fuels are composed of hydrocarbons that are easily cracked.

The conversion of a fuel into carbonaceous particles, involving millions of carbon atoms in a few milliseconds, is a complex process. The accepted model for this is that the pyrolysis and partial oxidation of the carbon-containing fuel result in hydrocarbon radicals, from which small hydrocarbons are formed, e.g., acetylene, benzene and others.

Larger aromatic rings are proposed to be formed mainly by acetylene addition mechanisms. These larger aromatic rings are then suggested to condense and form primary particles. Larger particles are formed when primary particles pick up molecules from the gas phase, promoting surface growth. The size increases further when particles, soot, are formed as a result of coagulation of primary particles to larger agglomerates.

These physical/chemical processes, nucleation, coagulation, condensation, absorption and adsorption are responsible for the final particle sizes, when the exhaust is cooled and diluted. An overview of a typical particle-formation process, from the start of combustion and out to the atmosphere, all according to Kittelson. It must be pointed out that the processes involved in particle formation are extremely sensitive to dilution conditions and that nanoparticle formation, which constitutes about 90% of the number of particles emitted from diesel fuelled CI engines, is also dependent upon the ratio between solid accumulation mode carbon mass to the mass of volatile precursor material as a driving force for gas-to-particle conversion. Higher nanoparticle emissions, i.e., more nanoparticles, are believed to be a consequence of lowering the mass of carbonaceous particles with respect to the mass of volatile material likely to become solid or liquid by homogeneous nucleation or condensation/adsorption, i.e., when carbonaceous particles are absent or reduced, condensation/adsorption cannot occur to the same extent, therefore, nucleation becomes the favored reaction and, hence, more nanoparticles are formed. **Figure (2.1)** shows an image of diesel particles in which primary particles can be distinguished as they make up larger particle agglomerates.

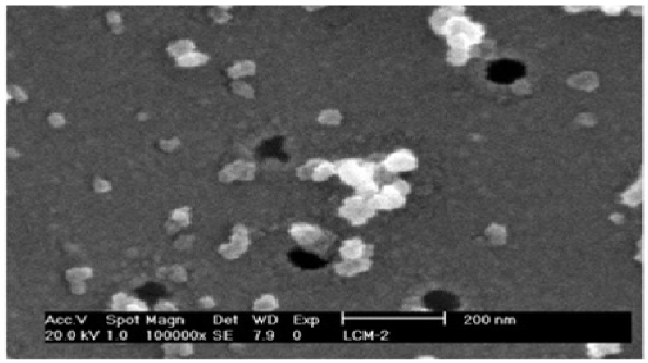

Figure 2.1: Diesel particles in high magnification showing single particles and particle agglomerates

The combustion process and its adjustment play a key role in the diesel engine when it comes to achievable performance, fuel consumption, and emissions. Engine performance is limited by the black-smoke emission value (maximum permitted exhaust-gas opacity at full load), the maximum permitted exhaust-gas temperature and the material properties of the turbine inlet. Combustion in the diesel engine can be divided into three phases:

- Ignition lag, i.e. the time between start of injection and start of ignition
- Premixed combustion
- Diffusion flame (mixture-controlled combustion)

Ignition lag, and thus a small quantity of injection fuel, is required during the first phase to limit combustion noise. After combustion starts, a good mixture formation is needed to achieve low soot and NOx emissions. The following factors have a decisive influence on the combustion phases:

- Pressure and temperature states within the combustion chamber.
- Mass, composition, and movement of the charge.
- Injection pressure process.

These parameters are adjustable firstly by engine-specific parameters, and secondly by variable operating parameters. The following fixed, engine-specific parameters are important for a given cylinder displacement [29]:

- Compression ratio.
- Stroke/bore ratio.
- Shape of piston recess.
- Intake port geometry.
- Intake and exhaust valve timing.

CI engines use fuels of lower volatility, with compression ratios from 11:1 to 22:1 and compression pressures between approximately 2700 and 4800 kPa. As the name implies, the high compression pressures of the CI engine ignite the fuel/air mixture, so no ignition source (e.g., spark plug) is required. Advantages of the CI engine over the SI engine include a lower specific fuel consumption, slightly higher thermal efficiency, relatively cheaper fuel costs, lower CO and hydrocarbon emissions at low and medium loads, lower capital costs, and higher durability.

Disadvantages include higher noise of operation, higher engine weight required to withstand the higher pressures, and excess oxygen in the exhaust preventing use of standard catalysts for air pollutant control. CI engines can be characterized by the injection type-either direct injection (DI) or indirect ignition (IDI). The Detroit Diesel 4-71N test engine is DI, implying that the fuel is injected directly into the combustion cylinder to mix with the intake air. IDI engines mix the fuel and air prior to entering the combustion cylinder in an attempt to improve mixing and therefore combustion. CI engines can additionally be characterized by the number of strokes required per power cycle, discussed below.

The fuel-injection system plays a key role in the combustion process since it defines the point and rate-of-discharge curve. In turn, the last two parameters are the key factors controlling emissions and efficiency. Besides the fuel-injection system, development focus is increasing on the air-flow system since compliance with ever more stringent NOx emission limits requires very high exhaust-gas recirculation rates [31].

2.3.1. Fuel-Injection System

On the air-intake side, mixture formation is influenced by movement of the charge in side of the cylinder. This, in turn, depends on intake-duct geometry and combustion chamber shape. As injection pressures have risen, the function of mixture formation has gradually shifted to the fuel-injection system. As a result, this has led to the development of the low-whirl combustion process.

On the fuel-injection side, extremely small nozzle holes with flow-optimized geometries promote good mixture formation as the injected fuel is then well prepared. At the same time this shortens ignition lag, and only small quantities of fuel are injected. During diffusion combustion that follows, optimized atomization results in high EGR compatibility, and this produces less NOx and soot [31].

2.3.2. Air- Flow System

Besides the fuel-injection system, more attention is also focusing on the air-flow system, since compliance with ever more stringent NOx emission limits requires very high EGR compatibility of the combustion process. This minimizes the formation of NOx so that the particulate filter (now fitted in ever greater numbers) can cope with the quantity of particulate emissions produced. It requires a system that is capable of combining comparatively high charge-air pressures at high, precise EGR rates identical for all cylinders, and at the lowest possible intake temperatures [31].

2.3.3. Cylinder Charge

Other measures carried out on the engine have an impact on cylinder charge from peripheral system, and ultimately on the concentration of pollutants in the exhaust gas. The most important measure for minimizing pollutants here is exhaust-gas recirculation. Exhaust gas recirculation to the intake manifold raises the proportion of inert gas, thus causing a drop in peak combustion temperature. It also reduces the production of nitrogen oxides [31].

2.3.4. Combustion Temperature

Together with the excess-air factor, the combustion temperature has a significant influence on the formation of NOx, High temperatures and excess air (λ>1) promote the formation of nitrogen oxides. In heterogeneous diffusion combustion, local, lean zones are inevitable, thus increasing the formation of nitrogen oxides. The aim of optimizing the combustion process, therefore, is to lower peak temperatures in the combustion chamber by raising the inert-gas component (EGR) and optimizing mixture formation at the same time in order to lessen the slight increase in soot production. In poor combustion conditions and at low casing the CO and UBHC are rose strongly which are the products of incomplete combustion. To counteract this, EGR coolers are bypassed when the engine is running cold. These coolers normally have a high cooling capacity required to reduce NOx emissions when the engine is operating at normal running temperatures.

NOx forms at high temperatures with excess air. Localized peak temperatures and localized, high excess-air factors must then be lowered. This is only achievable by retarding the start of injection at high injection rates during diffusion combustion. Combustion starts shortly before top dead center. This avoids almost any compression of combustion products that could increase the temperature. The high injection rate results in rapid turnover with 50% mass fraction burnt and high EGR compatibility. High combustion-chamber temperatures promote the formation of NOx [31].

2.3.5. Engine Speed

A high engine speed means greater friction losses in the engine and a higher power input by the ancillary assemblies (e.g. water pump). Engine efficiency, therefore, drops as engine speed increase. If a specific performance is produced at high engine speed, it requires a greater fuel quantity than if the same performance is produced at low engine speed. It also produces more pollutant emissions [31].

2.4. Emissions Effect on Health and Environment

Diesel engines emit pollutants into the environment. Underground environments are confined and have restricted ventilation and enclosed areas such that the pollutants cannot readily escape as the pollutants would if the diesel engine were operating in an open atmosphere. Thus, health risks to people from excessive exposure to the diesel engine pollutants are increased when a diesel engine is operating in an underground environment. Additional controls should be implemented [33].

2.4.1. Health Impacts

- Aggravated asthma & allergy symptoms.
- Chronic bronchitis.
- Heart & lung disease.
- Cancer.
- Premature death [37].

2.4.2. People Facing the Greatest Risk

- Children.
- Asthmatics.
- Occupationally exposed workers.
- People with existing respiratory problems [37].

2.4.3. Environmental Impacts

- Crop & forest damage.
- Acid rain.
- Eutrophication of waterways.
- Smog.
- Preventable death.
- Climate change [37].

As already mentioned in the background section there are indications that particles are a severe threat to both the environment and people´s health [38-47]. DNA damage and cardiovascular diseases have been found in rats exposed to particles [48-50]. Besides particles and particle-bound compounds, diesel exhaust is also making up of several hundred gaseous compounds. The impact on the environment and people´s health from many of these individual compounds, both particle-bound and gaseous, is known. Exposure to benzene is known to increase the risk of leukemia, [51]. Alkenes, such as ethene and propene, are converted through metabolism in the human body to their corresponding epoxides, which may react in the cells and, thus, initiate a mutagenic effect [52]. Furthermore, all hydrocarbons, except for methane, participate, more or less, in the formation of ground level ozone [53]. Methane, N_2O, CO_2 and other compounds found in the exhaust gas contribute to global warming [53]. Formaldehyde and acetaldehyde are classified as probable carcinogenic by the National Institute for Occupational Safety and Health, NIOSH, the World Health Organization, WHO, and the European Union, EU.

EPA classifies both individual compounds in the exhaust and diesel exhaust as a whole as carcinogenic. As an example, EPA classifies individual compounds, such as formaldehyde, as a probable human carcinogen, (Group B1), acetaldehyde as a probable human carcinogen (Group 2b), and acrolein as a possible human carcinogen (Group C).

Furthermore, aldehydes, as a group, are, after nitrogen oxides, one of the most powerful agents for the formation of ground level ozone, besides NOx, in reaction with hydrocarbons [53]. The maximum incremental reactivity, MIR, is the tendency of an organic gas to contribute to the formation of ozone in specific atmospheres containing NOx [54, 55].

2.5. Reduce Emissions

The 1990s have been a time for major technological advances to reduce emissions from diesel engines by diesel engine manufacturers. Driven first by the U.S. Environmental Protection Agency’s 1991 requirement that diesel engines meet a PM emission standard of 0.25 g/bhp-hr combined with a 6.0 g/bhp-hr NO emission standard, and followed by the Agency's 1994 standard of 0.1 (0.05 for urban buses) g/bhp-hr for PM emissions along with a 5.0 g/bhp-hr NOx requirement and the 1998 standard requiring that NO emissions be further reduced to 4.0 g/bhp-hr, engine manufacturers have focused on [56]:

- Improved fuel injection techniques,
- Improved air management methods,
- Improved combustion chamber design, and
- Improved oil control.

2.5.1. Fuel Injector Design

Significant research and development has taken place on fuel injector design and placement to help manufacturers meet the U.S. on road 1991 standards. Injector inclination, the number of holes and their diameters, sac volumes, and spray patterns have all been optimized for low emissions. Also in some instances, valve covered orifices (VCO) have been incorporated into injector designs to minimize residual fuel from entering the combustion chamber. Hydraulically actuated electronic unit injectors have allowed manufacturers to control the rate of fuel injection which also has resulted in lower emissions of PM and NO for those engines which use them.

2.5.2. Fuel Injection Pressure

Increased fuel injection pressure has been used to increase atomization of the fuel in the combustion chamber, which in turn has resulted in lower PM emissions. Injection pressures in excess of 20000 psi can be found on some diesel engines today. These engines are characterized by decreased swirl in order to minimize the NO formation which otherwise could occur due to the enhanced combustion resulting from the higher injection pressures. Manufacturers have had to make more robust fuel system components because of the increased pressures which have in turn increased the cost of the engines. Another strategy that has been used by manufacturers-especially manufacturers of lower cost light and medium heavy-duty engines is using medium fuel injection pressure in combination with increased swirl [56].

2.5.3. Turbocharging and Air Cooling

A turbocharger is used to extract energy from a diesel engine's exhaust flow by using of an air compressor attached to an exhaust gas turbine located in the exhaust stream. The turbine is used to compress air to be fed to the intake air manifold. The increased mass of air to the combustion chamber allows for more fuel delivery and hence, increases engine power. Better combustion also results from turbocharging which in turn decreases PM emissions. Cooling the compressed air supplied to the intake air manifold reduces the NO emissions which otherwise would result from increased combustion temperatures [56].

In order to meet current on road emissions standards, manufacturers have optimized turbocharger operation to match engine operating conditions more precisely, thereby avoiding over boosting which causes combustion to deteriorate, as well as making the turbocharger more responsive to transient conditions. Both of these techniques have resulted in lower PM emissions [56].

Employing after cooling, which results in lower combustion temperatures, has allowed manufacturers to optimize injection timing to minimize PM emissions while off-setting the increase in NO emissions that otherwise would occur [56].

2.5.4. Intake Manifold and Port Design

Intake manifolds and port configurations have been designed for better in-cylinder air distribution, eliminating fuel-rich spots. Rich areas during combustion result in incomplete combustion of some of the injected fuel and increase UBHC and PM emissions. The designs also insure proper fuel penetration into the cylinder and minimize cylinder wall wetting which both serve to decrease UBHC and PM emissions [56].

2.5.5. Combustion Chamber Design

Medium-duty diesel engines generally use re-entrant piston bowl designs. The re-entrant bowl causes in-cylinder turbulence and better fuel/air mixing. The better mixing improves combustion and decreases both PM and UBHC emissions [56].

2.5.6. Oil Control

Oil control on 1991 and newer on-road diesel engines has improved significantly compared to pre-1991 engines where as much as 30 percent of the PM emitted could be attributed to the combustion of lubricating oil. This improvement has decreased PM emissions by 10 percent [56].

2.5.7. Exhaust Gas Recirculation

EGR is a highly effective internal engine measure to lower NOx emissions on the diesel engines; a distinction is made between [56]:

- Internal EGR, which is determined by valve timing and residual gas.
- External EGR, which is routed to the combustion chamber through additional lines and a control valve.

The NOx-reducing effect is mainly due to the following causes [56]:

- Reduction in exhaust-gas mass flow.
- Drop in the rate of combustion, and thus local peak temperatures due to an increase in the inert-gas component in the combustion chamber.
- Reduction in partial oxygen pressure or local excess-air factor.

Diesel exhaust emission controls were first used in work environments when diesel oxidation catalysts were used in underground mines and on forklift trucks over twenty-five years ago, primarily for CO and UBHC control. Early on in the use of catalyst technology for diesel vehicles, manufacturers recognized the potential of catalyst technology to possibly increase the mutagenic activity of diesel exhaust. Consequently, attention was paid to properly formulate the catalyst to not only eliminate this potential but to reduce the mutagenic activity of the exhaust [56].

More recently because of the U.S. EPA's urban bus rebuild/retrofit requirements for the reduction of diesel PM emissions, diesel oxidation catalyst technology has become recognized as an effective means of reducing PM emissions from diesel engines by greater than 25 percent [56]. In the late 1970s, considerable attention was given to the development of diesel particulate filter (DPF) technology, which was capable of reducing over 90 percent of diesel PM emissions [56].

In 1986, the first diesel particulate filter systems were commercialized for underground production vehicles. Although the filters have witnessed limited use since first commercialized, their use has been highly effective where appropriately applied [56]. In the mid to late 1980s and into the 1990s with several regulatory initiatives underway in the U.S., emission control manufacturers have continued to refine and develop advanced diesel oxidation catalysts and filter systems. Also, much progress has been made in developing other advanced technologies like lean-NO catalysts and absorbers among other technologies which will also play a role in reducing emissions from diesel engines. Applying traditional stationary source NO control technologies, like selective catalytic reduction (SCR), to mobile sources has begun [56]. Compression ignited, CI, engines may be broadly identified as being either fuelled by diesel or alternatively fuelled, two or four-stroke, injected directly or indirectly, naturally aspirated or supercharged, with catalyst or other exhaust gas after-treatment. They are also classified according to service requirements, such as light-duty (LD) or heavy-duty (HD) automotive/truck, small or large industrial, construction, rail or marine engine. The engines treated in this thesis are mainly engines intended for heavy-duty applications, on-road and non-road [36].

EGR reduces cylinder NOx formation by three mechanisms [34]:

1. Dilution of intake air, due to EGR reduces pre-mix combustion and prolongs diffusion-burn combustion, both reduce cylinder temperatures.

2. Increased water and CO_2 in the combustion chamber, due to increases the mixture's specific heat, leading to reduced combustion temperatures.

3. Increased dissociation of these water and CO_2 molecules, due to dissociation of these (and other complex molecules) leads to reduced combustion temperatures.

Not only does EGR reduce NOx formation, it also Reduces UBHC since the unburned vapor fuel has a “second chance” to return. Increases fuel consumption since the reduced cylinder temperatures also reduce cylinder pressures, which reduces power output (and thus increases fuel consumption for any given power output). Increases CO_2 is released, as a direct consequence of increased fuel consumption. Increases CO release since the combustion quality is poor with EGR (complete oxidation does not occur as well when operating EGR). Additionally, the increased levels of CO_2 in the intake dissociates into CO during the combustion process. Some of this CO may oxidize to CO_2, but with reduced levels of O_2, not all of re-associates. Increases H_2O release, again as a direct consequence of increased fuel consumption increases soot formation, since as was already presented, decreased cylinder temperatures leads to increases levels of soot [34].

Since high local temperatures (>2000k) and a sufficiently high partial oxygen pressure are required to from NOx, the measures listed above result in a drastic reduction in the formation of NOx as the EGR rate rises. Reducing the reactive components in the combustion chamber also leads to a rise in black smoke, which limits the quantity of recirculated exhaust gas [29].

In order to enhance the effects of EGR, the recirculated exhaust-gas quantity is cooled in the heat exchanger cooled by engine coolant. This raises gas density in the intake manifold and causes a lower final compression temperature. In general, the effect of higher localized excess-air factors cancel each other out as a result of increased charge density and reduced peak temperature. At the same time, however, EGR compatibility rises to produce possibly higher exhaust-gas recirculation rates at much lower NOx emissions [29].

Since diesel-engine exhaust gas already has a low temperature at very low load points anyway, cooling the recirculated exhaust gas at the high EGR rates required to reduce NOx emissions leads to unstable combustion. This then results in a significant rise in UBHC and CO emissions. A switchable EGR cooler is very effective to increase combustion chamber temperature, stabilize combustion, reduce untreated HC and CO emissions, and raise exhaust-gas temperature. In particular, this occurs in the cold start phase of the car emission test, during which the oxidation-type catalytic converter has not reached its light-off temperature. It also helps the oxidation-type catalytic converter to reach its operating temperature much faster [29]. Exhaust gas recirculation is an effective method for NOx control. The exhaust gases mainly consist of inert carbon dioxide, nitrogen and possess high specific heat. When recirculated to engine inlet, it can reduce oxygen concentration and act as a heat sink. This process reduces oxygen concentration and peak combustion temperature, which results in reduced NOx. EGR is one of the most effective techniques currently available for reducing NOx emissions in internal combustion engines. However, the application of EGR also incurs penalties. It can significantly increase smoke, fuel consumption and reduce thermal efficiency unless suitably optimized. The higher NOx emission can be effectively controlled by employing EGR [57].

2.5.8. Biodiesel

The use of biodiesel in diesel engines does not require any engine modification. Biodiesel gives considerably lower emissions of PM, CO and UBHC without any fuel consumption or engine performance penalties. Many researchers have found that with biodiesel fueled engine produces higher NOx emissions compared to diesel [58].

Biodiesel is an oxygenated fuel and after combustion gives higher NOx emissions. The reason behind this is higher boiling point, higher bulk modulus, and inherent oxygen content. However this presence of oxygen reduces CO and UBHC emission. Bulk modulus is another important property, which results in a dynamic advance of injection timing in bio-diesel fuelled engine. Bulk modulus of biodiesel is higher than the diesel fuel, which leads to a more rapid transfer of the pressure waves from fuel pump to lift the needle of the injector much earlier. This advance results in more fuel accumulation before the start of combustion leading to higher peak temperature and pressure in premixed phase and subsequently higher NOx. The following NOx reduction techniques can be used in a biodiesel fuelled diesel engine [57]. Environment and health concerns have resulted in stringent emission standards which require diesel engines to meet a 5.0 g/ kWh NOx standard in EURO-III. There is a critical need for cost effective technologies to meet these mandates and to clean the environment. One promising approach towards meeting these standards with minimal changes in the present infrastructure of the transportation industry, is the use of fuel additives that when injected into the cylinder, along with the diesel fuel, can result in substantial NOx reduction [57].

The use of micro emulsions containing scab-Engen additives offers potential solution for substantial reduction in NOx. The addition of water was limited to 10 wt. % as microemulsions containing less than 10wt% water were found to be effective in carrying scavenger additives. A typical composition containing 10wt% water, less than 10wt% surfactant/co-surfactant/neutralizing agent and 1wt% scavenger additive reduced NOx by almost 30% across the load range of the engine. The addition of quantities greater than 1wt% of the scavenger additive did not have a beneficial effect on NOx reduction. This may be due to the additive being destroyed in the flame and not making it past the flame front for reaction with NOx. The use of scavenger additives leads to a severe depression in the cetane number of the fuel. Selection of a cetane improver is critical from both cost and NOx reduction perspectives. Moreover, most of the additives are expensive and can promote auto-oxidation in bio-diesel [57].

2.5.9. Selective Catalytic Reduction

Selective catalytic reduction technique is most versatile technique for NOx control in diesel engines. Catalysts used in SCR are manufactured from various ceramic materials used as a carrier, such as titanium oxide, and active catalytic components are usually oxides of base metals (such as vanadium and tungsten), zeolites, and various precious metals. The two most common designs of SCR catalyst geometry used today are honeycomb and plate. The honeycomb form usually is an extruded ceramic applied homogeneously throughout the ceramic carrier or coated on the substrate. Like the various types of catalysts, their configuration also has advantages and disadvantages. Plate type catalysts have lower pressure drops and are less susceptible to plugging and fouling than the honeycomb types, however plate type configurations are significantly larger and more expensive. Honeycomb configurations are significantly smaller than plate types, but have higher pressure drops and plug much more easily. Technical difficulties with SCR units are [57]:-

- Contamination of catalyst.
- Tuning of SCR system with engine operating cycle.
- Low exhaust gas temperature (below the optimal range of catalyst).

2.5.10. Retarded Injection Timing

Injection timing is another well-studied mechanism for controlling NOx emissions. Advancing injection timing causes higher NOx emissions since combustion starts earlier, and thus the residence time of the burning mixture in the cylinder is increased. This allows the NOx formation reactions to proceed. An advance in injection timing for biodiesel relative to that of petroleum diesel is caused by its higher bulk modulus of compressibility. Since pump–line–nozzle (PLN) injection systems generally start fuel injection upon reaching a certain fuel pressure, a higher bulk modulus leads to this pressure requirement being met more quickly, and thus fuel is injected earlier. Engines equipped with high-pressure common rail fuel injection systems do not rely on the transfer of a pressure wave to initiate injection; bulk modulus is therefore not thought to alter injection timing in these types of diesel engines. Retarded injection leads to increased fuel consumption, reduced power, increased UBHC and excess smoke. Monyem et al. observed a reduction in NOx emissions of 35% to 43% for 6-degree retardation in injection timing [57].

2.5.11. Ethanol

Ethanol Volume % Minimum: Specifying the minimum ethanol content is essential to minimize the presence of impurities. The minimum ethanol content of denatured ethanol plus the denaturant make up at least 96.86% of the total volume thereby, limiting impurities to fewer than 3%. Impurities typically found in commercially produced fuel ethanol include such compounds as methanol and fuel oils such as amyl and isoamyl alcohols [59].

2.5.12. Ethanol-Water/Air Mixtures

Lean ethanol-water/air mixtures have potential for reducing NOx and CO emissions in internal combustion engines, with little well-to-wheels CO_2 emissions. Conventional ignition systems have been unsuccessful at igniting such mixtures. An alternative catalytic ignition source is being developed to aid in the combustion of aqueous ethanol. The operating principle is homogeneous charge compression ignition inside a catalytic pre-chamber, which causes torch ignition and flame propagation in the combustion chamber. Ignition timing can be adjusted by changing the length of the catalytic core element, the length of the pre-chamber, the diameter of the pre-chamber, and the electrical power supplied to the catalytic core element. To study engine operation, a 1.0L 3-cylinder Yanmar diesel engine was converted for ethanol-water use, and compared with an unmodified engine. Comparing the converted Yanmar to the stock engine shows an increase in torque and power, with improvements in CO and NOx emissions. Hydrocarbon emissions from the converted engine increased significantly, but are largely due to piston geometry not well suited for homogeneous charge combustion. No exhaust after treatment was performed on either engine configuration. Applying this technology in an engine with a combustion chamber and piston design suited for homogeneous mixtures has the potential to lower emissions to current standards, with a simple reduction catalytic converter [25].

Diesel engines are now preferred due to its high fuel economy in various applications. However they have unsatisfactory emission characteristics. Hence there is a constant search for alternate fuels, which will meet the present emission norms. Researchers are more concentrating on the efficient combustion by employing [60]:

- Highly pressurized fuel injection like CRDI.
- Advanced engine and exhaust gas management systems.
- Enriched fuel with additives like ignition accelerators, antiknock agents.
- Alternate fuels to have more colorific value.

The reasons for opting alternate fuels are [60]:

- High cost of petroleum products and surge the cost of hydrocarbon fuels.
- Increased demand for petroleum products.
- Strict emissions norms like EURO NORMS, BHARAT NORMS, KYOTO PROTOCOL, etc.
- Global warming and adverse environment effect due to pollution from the automobiles.

Many alternate fuels are being considered for automotive vehicles and ETHANOL of the best alternate fuels. Ethanol is produced from molasses, which is a by-product of sugarcane. Ethanol can be produced in large quantities at low cost from these molasses. It is a renewable fuel and its high oxygen content improves the combustion characteristics. Ethanol reduces harmful emissions from I.C engines of Sulphur-di-oxide, oxides of nitrogen and particulate emissions.

2.5.13. Properties of Ethanol [60]:

- Viscosity of ethanol is less.
- Specific gravity of ethanol is 0.794.
- Boiling temperature of ethanol is 78ºC.
- Ethanol has a low cetane rating.
- It is inflammable and its vapour form explosive mixtures with air.
- It is an excellent solvent for fuels, oils, fats etc.
- It is miscible with water in all proportionate mixing being attended by a concentration of volume.

2.5.14. Ethanol Properties Compared To Diesel [60]:

S.NO	Property	Ethanol	Diesel
1	Density kg/m^3	785	840
2	Volume lower heating value kg/liter	21000	36500
3	Latent heat of vaporization kJ/kg	840	251
4	Cetane number	8	50
5	Enthalpy of vaporization kJ/kg	837	225-600
6	Auto ignition temperature	365- 425	204-260
7	Stoichiometric Air fuel ratio	9	14.5

2.5.15. Use of Ethanol in Diesel Engines

The various techniques by which the ethanol can be used as a fuel for compression ignition engines are [60]:

- Blend formation.
- Fumigation.
- Dual injection.
- Spark ignition.
- Ignition improvers.
- Surface ignition.

2.5.15.1. Blend Formation

The easiest method by which ethanol could be used is in the form of diesel ethanol blend. But ethanol has limited solubility in diesel; hence ethanol/diesel solutions are restricted to small percentages (typically 20%). This problem of limited solubility has been overcome by emulsions, which have the capability of accommodation larger displacement of diesel up to 40% by volume. But the major drawbacks of emulsions are the cost of emulsifiers and poor low temperatures physical properties [60].

2.5.15.2. Dual Injection

Dual injection is a method by which nearly 90% Displacement of diesel by ethanol is possible. The drawback of this method includes the complexity and expense of a second injection system and a second fuel tank and system [60].

2.5.15.3. Spark Ignition

Spark ignition of neat ethanol in diesel engines provides a way of displacing 100% of diesel. A spark plug and the associated ignition system components must be added to the engine. Space must be available for spark plugs in the cylinder head and it's also important for sparkplugs in the cylinder head and it's also important for proper plug cooling [60].

2.5.15.4. Ignition Improvers

Another method of using neat ethanol is to increase their cetane numbers sufficiently with ignition improving additives to ensure that compression ignition will occur. This method saves the expense and complexity of engine components changes, but adds fuel costs [60].

2.5.15.5. Surface Ignition

This is another method of using ethanol 100% ethanol in diesel engines. Surface ignition occurs when the temperature of the air-fuel mixture adjacent to a hot surface exceeds its self-ignition limit [60].

2.5.15.6. Fumigation

Fumigation is a method by which ethanol is introduced in to engine by carbureting or vaporizing the ethanol into the intake stream. This method requires addition of a carburetor or vaporizer along with a separate fuel tank, lines and controls. But with the emergence of electronic injection techniques the fumigation technique has been made possible by using an injector in the intake manifold [60].

2.5.16. Diesel – Water Emulsion

Increasing environmental issues and growth of global warming day by day are the driving forces for the researchers to arrive at a clean burning fuel. Fossil fuels cause various environment problems such as acid precipitation, Ozone depletion, global warming etc. As discussed elsewhere, for many years, it is recognized that for simultaneous reduction of NOx and smoke, water in fuel emissions are effective [61-63]. The effect of mixing water with diesel to make an emulsified fuel considers the needs for the vehicle performance and its cleanest possible operation. Two test fuels have been chosen for the experimental research work and they are termed EM 1 (Emulsion 1) and EM 2 (Emulsion 2). EM 1 is prepared in the ratio of 91/8/1 which represents 91% diesel, 8% water, 0.5% surfactant (Span 20) and 0.5 % co-surfactant (Tween 20) with continuous stirring. EM 2 has a composition of 94% diesel, 5% water, 0.5% Span 20 and 0.5% Tween 20. EM 1 and EM 2 are used as the test fuels and compared with petroleum diesel for its performance and emission characteristics. The test fuels EM1 and EM2 have an increase in brake thermal efficiency for a percentage of 4.59 and 2.48 respectively. Hydrocarbon emission decreased for a percentage of 10.41 and 6.25 with the fuels EM1 and EM2 respectively. The smoke opacity decreased for a percentage of 7.54 and 3.92 with the fuels EM 1 and EM 2 respectively. A drastic decrease of oxides of nitrogen was found. The decreased values were 28.52% and 24.48% with the emulsified fuels EM 1 and EM 2 respectively [64]. The application of an emulsification technique to prepare the fuel has been considered to be one of the possible approaches to reduce the production of diesel engine pollutants, as well as the rate of fuel consumption. Water-in-diesel oil emulsified formulations are reported to reduce the emissions of NOx, SOx, CO, black smokes and PM without compensating the engine performance. Emulsified diesel fuels of 0, 5, 10, 15 and 20 water/diesel ratios by volume, were used in single cylinder, direct injection diesel engine, operating at 1500-2700 rpm. The results show that there was a significant benefits associated with the addition of water contents in diesel oil. The obtained experimental results indicate that the addition of water in the form of emulsion improves the brake thermal efficiency. On the whole it is concluded that BSFC, exhaust gas temperature, NOx, UBHC, CO, CO_2 and black smoke opacity decrease as the percentage of water in the emulsions increases [65]. The proper mixing technique and emulsifying agent were used to produce stable emulsions of 10% to 30% water by volume in diesel. The stability of these emulsions ranges from one week up to 4 weeks. The physical properties of stable water-diesel emulsions such as density, viscosity and pour point were observed. The effect of water-diesel concentrations, on the performance of a single cylinder diesel engine in terms of engine speed, torque, brake power output, brake specific fuel consumption, brake thermal efficiency, exhaust gas temperature and emissions such as NOx and PM were studied. The results showed that the water emulsification has a potential to improve the diesel engine performance and to reduce gas pollutants [66]. Although the experiments were conducted on a single cylinder four stroke cycle direct injection diesel engine at constant speed with a fuel injection pressure of 200 bars. Tests were conducted using commercial diesel fuel and diesel fuel with 10% and 20% water by volume. From the test results, it was found that the water emulsification has a potential to improve brake thermal efficiency and brakes specific fuel consumption. The NOx and hydrocarbon emissions were found to decrease with increase in water percentage in the emulsified diesel [67]. The combustion of water-diesel emulsion in diesel engine was simulated using a computer program to estimate the heat release rate, cylinder pressure, brake thermal efficiency, brake specific fuel consumption and NO formation. The numerical simulation was performed at different equivalence ratios, engine speeds and water percentages. The numerical simulation was preferred to study the combustion behavior and emission of diesel engine because the experimental investigations were time consuming and costly affair [68]. The effect of water–oil emulsions on the engine performance and on the main pollutant emissions, NOx, total hydrocarbons (THC), soot, PM and its composition, was studied. A turbocharger intercooler indirect injection (IDI) Diesel engine was tested under five different steady state operating conditions, selected from the transient cycle for light duty vehicles established in the European Emission Directive 70/220. Tests were performed using a commercial fuel as a reference and an emulsified fuel for each operating condition. Results reported here suggest that the water emulsification has a potential to slightly improve the brake efficiency and to significantly reduce the formation of thermal NO, soot, hydrocarbons and PM in the Diesel engine [69]. Oxygen-enriched combustion of diesel engine can reduce smoke emission and improve thermal efficiency, but also lead to the increase of NO emission. In this research, experiments were conducted on a turbo-charged direct injection diesel engine under the two conditions of 2000 rpm and 180 Nm equivalent powers (57% of the original max load at 2000 rpm) as well as 100% load of this speed. The combination of intake oxygen enrichment and water emulsified diesel was used to improve the NO smoke emissions without serious penalty in (BSFC). The results showed that when engine load was 180 Nm with the conditions of 0%-20% water emulsion ratio and 21%-21.5% intake oxygen concentration, as well as under the condition of 100% load with 10%-15% water emulsion ratio and 21%-22% intake oxygen concentration, the NO-Smoke emissions were lower than that of original engine and BSFC was not exceeding 5% of the original engine by optimized combination of water emulsion ratio and oxygen concentration [70].

Another study examines the effects of combusting a mixture of diesel fuel, water, and surfactant on NOx emissions from a compression ignition diesel engine. Previous research has attributed the observed reduction of nitrogen oxide emissions to a suppression of flame temperature due to quenching effects from the water, thereby reducing thermal NOx formation. Experimental procedures conducted using a Detroit Diesel 4-cyclinder diesel engine are discussed. Results from testing diesel fuel with varying ratios of water balanced with a surfactant to stabilize the emulsion will be presented and discussed. The data shows significant NOx emission reduction with up to 45 percent water, by volume, in the fuel [71].

2.5.17. Diesel Fuel with Water Injection

Water injection was by means of a Bosch pump and various pencil-type nozzles installed, adjacent to the fuel injector in the cylinder head. Port injection and port induction were also briefly investigated. A five-hole, 90^o included angle nozzle was used, as was a three-hole, 30^o included angle unit. For comparison, a nozzle directing one spray obliquely at the cylinder wall was also tested. Firing pressure was monitored using a piezoelectric transducer; both pressure-time and pressure-volume (indicator) records were obtained. In order to determine timing of both fuel and water injection, needle lift was monitored using a differential transformer pickup [72].

The results of this study indicate [72]:

- Optimum total engine cooling by direct water injection was accomplished over a wide range of water injection timings (from 450 to 720 CA degrees after TDC power stroke) at water/fuel ratios of 2.9 to 3.7 with output power and brake specific fuel consumption improved 5 to 20%, respectively, over that with the standard jacket-cooled CLR engine.
- Emissions are affected in an expected manner by the presence of water: NOx is decreased, sometimes substantially, while the other emissions (UBHC, CO) tend to increase.
- When cooling the exhaust, the condensate becomes an effective scrubber of sulfur oxides. NOx was not significantly reduced by scrubbing, but if the condensate is made sufficiently alkaline (pH :> 8), CO_2 was unintentionally scrubbed out.

Fumigation is where liquid water is injected into the intake manifold upstream of the intake valve. The fumigation technique has been shown to reduce NOx emissions in DI Diesel applications but suffers from the drawback that the liquid water in the combustion chamber is typically in areas where it is less effective at reducing emissions. Therefore, fumigation requires approximately twice the liquid volume for the same reduction in engine out NOx when compared to direct water injection. Additionally, liquid water present after combustion can contaminate the oil and increase engine wear. The effects of in-cylinder water injection on a direct injection (DI) Diesel engine were studied using a computational fluid dynamics (CFD) program based on the Kiva-3v code. The spray model is validated against experimental bomb data with good agreement for vapor penetration as a function of time. It was found that liquid penetration increased approximately 35% with 23% of the fuel volume replaced by water, due mostly to the increase in latent heat of vaporization [73].

Engine calculations were compared to experimental results and showed very good agreement with pressure, ignition delay and fuel consumption. Trends for emissions were accurately predicted for both 44% and 86% load conditions. Engine simulations showed that the vaporization of liquid water as well as a local increase in specific heat of the gas around the flame resulted in lower Nitrogen Oxide emissions (NOx) and soot formation rates. Using stratified fuel-water injection increases soot at 86% loads due in part to late injection. Because NOx decreased at all loads, the injection timing can be advanced to minimize fuel consumption and soot [73].

A system for injection of diesel fuel and water with real time control, or real-time water injection (RTWI), was developed and applied to a heavy-duty diesel engine. The RTWI system featured electronic unit pumps that delivered metered volumes of water to electronic unit injectors (EUI) modified to incorporate the water addition passages. The water and diesel mixed in the injector tip such that the initial portion of the injection contained mostly diesel fuel, while the balance of the injection was a water and diesel mixture. With this hardware, real-time cycle-by-cycle control of water mass was used to mitigate soot formation during diesel combustion. Using RTWI alone, NOx emissions were reduced by 42% [74].

Three water injection timing strategies were used, at the suction stroke, at the compression stroke and at the expansion stroke. The quantity and time of water injection were varied during every strategy. Also the effect of water injector sprays location inside the combustion chamber relative to the spark plug position was studied. The results showed that the reduction of NOx emissions was most strongly dependent on the water injection timing. The optimum water injection timing, for maximum NOx reduction, was depended on the change in the water injection quantity and this optimum water injection timing was advanced with the increase of water quantity. The indicated thermal efficiency (ITE) improved when water was injected in the compression stroke. The ITE improvement was only through range of water injection timing at compression stroke and this range mainly dependent on the water quantity. The range of water injection timing which ensures an improvement in the ITE was increased as the water quantity increased. The water injection timing at the expansion stroke has little effect on the NOx emissions which has been already formed [75].

Steam injected diesel engine is modeled by using zero-dimensional single-zone combustion model for 20% steam ratio at full load condition. The obtained results are compared with conventional diesel engine in terms of performance and NO, CO, CO_2, UBHC emissions. The simulation results agree with experimental data quite well. In the experimental results, it is determined that the engine torque and the effective power increase up to 2.5% at 1200 rpm, specific fuel consumption (SFC) and effective efficiency improves up to 6.1% at 2400 rpm, NO emissions reduce up to 22.4% at 1200 rpm, CO_2 emissions decrease up to 4.3% at 1800 rpm, smoke density increases from 44% to 46% at 2200 rpm [76].

In the another study, the effects of water injection (WI) into intake air on the performance and exhaust emissions were experimentally investigated in a Renault K9K 700 type turbocharged common-rail DI automotive diesel engine. Experiments were performed at different loads and engine speeds, with various water ratios (WRs). The water was injected into intake air by a carburetor, which main nozzle section is adjustable; at approximately 2%, 4%, 6%, 8% and 10% (by vol.) water ratios. It was determined that, WI into intake air at specified ratios decreases significantly smoke index K and NOx emission and improves somewhat the engine performance. Smoke index K decreases by increasing of WRs at 2000, 2500 and 3000 rpm. Its maximum reduction ratio was obtained as 41.75% for 11.71% WR at 3000 rpm. At chosen loads and engine speeds, as WRs increases, NOx emission decreases. More significant reduction of NOx has been obtained after 6% WR. For full load (149 N m) at 2500 rpm, maximum decrement of NOx emission was attained as 12.489% for 9.400% WR. Water addition results insignificant effect on BSFC at 2000, 2500 and 3000 rpm. At these engine speeds BSFC takes values close to the Neat Diesel Fuel (NDF) or increases slightly. However, BSFC decreases approximately 4% with water addition for selected loads at 3500 rpm. WI does not show any significant change in-cylinder pressure and indicated power [77].

Finally, experiments were conducted to compare the effects of water–diesel emulsion and water injection into the intake manifold on performance, combustion and emission characteristics of a DI diesel engine under similar operating conditions. The water to diesel ratio for the emulsion was 0.4:1 by mass. The same water–diesel ratio was maintained for water injection method in order to assess both potential benefits. All tests were done at the constant speed of 1500 rpm at different outputs. The static injection timing of 23^{o} BTDC was kept as constant for all experimental tests. In the first phase, experiments were carried out to assess the performance, combustion and emission characteristics of the engine using the water–diesel emulsion. The emulsion was prepared using the surfactant of HLB: 7. The emulsion was injected using the conventional injection system during the compression stroke. The second phase of work was that water was injected into the intake manifold of the engine using an auxiliary injector during the suction stroke. An electronic control unit (ECU) was developed to control the injector operation such as start of injection and water injection duration with respect to the desired crank angle. The experimental result indicates the both methods (emulsion and injection) could reduce NO emission drastically in diesel engines. At full load, NO emission decreased drastically from 1034 ppm with base diesel to 645 ppm with emulsion and 643 ppm with injection. But, NO emission reduction is lesser with injection than emulsion at part loads. Smoke emission is lower with the emulsion (2.7 BSU) than with water injection (3.2 BSU) as compared to base diesel (3.6 BSU). However, CO and UBHC levels were higher with emulsion than water injection. As regards NO and smoke reduction, the emulsion was superior to injection at all loads. Peak pressure, ignition delay and maximum rate of pressure rise were lesser with water injection as compared to the emulsion. It is well demonstrated through this comparative study that the emulsion method has higher potential of simultaneous reduction of NO and smoke emissions at all loads than injection method [78].

2.5.18. Ethanol-Diesel Blends

The present work reviews the literature concerning the effects of alcohol/diesel blends on the exhaust emissions of diesel engines. The addition of ethanol to diesel fuel simultaneously decreases cetane number, high heating value, aromatics fractions and kinematic viscosity of ethanol blended diesel fuels and changes distillation temperatures. An additive used to keep the blends homogenous and stable, and an ignition improver, which can enhance cetane number of the blends, have favorable effects on the physicochemical properties related to ignition and combustion of the blends with 10% and 30% ethanol by volume. The emission characteristics of five fuels were conducted on a diesel engine. At high loads, the blends reduce smoke significantly with a small penalty on CO, acetaldehyde and unburned ethanol emissions compared to diesel fuel. NOx and CO_2 emissions of the blends are decreased somewhat. At low loads, the blends have slight effects on smoke reduction due to overall leaner mixture. With the aid of additive and ignition improver, CO, unburned ethanol and acetaldehyde emissions of the blends can be decreased moderately, even total hydrocarbon emissions are less than those of diesel fuel. The results indicate the potential of diesel reformation for clean combustion in diesel engines [79].

Ethanol is an attractive alternative fuel because it is a renewable bio-based resource and it is oxygenated, thereby providing the potential to reduce particulate emissions in compression–ignition engines [80]. The properties of ethanol–diesel blends have a significant effect on safety, engine performance and durability, and emissions. An increase in fuel consumption approximately equivalent to the reduction in energy content of the fuel can be expected when using ethanol–diesel blends. With ethanol percentages of 10% or less, operators have reported no noticeable differences in performance compared to running on diesel fuel. While there is considerable value in being able to use the fuel directly in an unmodified engine, small adjustments to fuel injection characteristics may result in further gains in reducing emissions [80]. The test results show that it is feasible and applicable for the blends with n-butanol to replace pure diesel as the fuel for diesel engine; the thermal efficiencies of the engine fuelled by the blends were comparable with that fuelled by diesel, with some increase of fuel consumptions, which is due to the lower heating value of ethanol. The characteristics of the emissions were also studied. Fuelled by the blends, it is found that the smoke emissions from the engine fuelled by the blends were all lower than that fuelled by diesel; (CO) were reduced when the engine ran at and above its half loads, but were increased at low loads and low speed; the (UBHC) emissions were all higher except for the top loads at high speed; (NOx) emissions were different for different speeds, loads and blends [81].

Ethanol blended diesel fuel has a shorter spray tip penetration when compared to pure diesel fuel. In addition, the spray cone angle of ethanol blended fuels is larger. It is believed that the lower fuel density of ethanol blended fuels affects the spray characteristics. When the ethanol blended fuels are injected around top dead center (TDC), they exhibit unstable ignition characteristics because the higher ethanol blending ratio causes a long ignition delay. An advance in the injection timing also induces an increase in the combustion pressure due to the sufficient premixed duration. In a four-cylinder diesel engine, an increase in the ethanol blending ratio leads to a decrease in NOx emissions due to the high heat of evaporation of ethanol fuel, however, CO and UBHC emissions increase. In addition, the CO and UBHC emissions exhibit a decreasing trend according to an increase in the engine load and an advance in the injection timing [82].

Engine experiments were carried out to compare the effects of different ethanol–diesel blend fuels on regulated emissions (THC, CO, NOx, PM) and PAH emissions. The experimental results indicated that under the ECE R49-13 test mode, the BSTHC and BSCO emissions tended to increase with the addition of ethanol, and the maximum increment could be up to 53.1% and 70.5% relative to E0, respectively. The BSNOx and BSPM emissions were observed little variation. But, the ethanol–diesel blends showed significant benefit in terms of smoke reduction. The more ethanol was added, the less smoke emitted. For PAHs emissions, it presented an increasing trend with a growth of ethanol content in the ethanol–diesel blends. Comparing with E0, only E5 showed the advantage of reducing BSPAH emissions by 19.1% [83].

An experiment on the application of diesel and ethanol blends as fuel in diesel engine was carried out at various engine loads and ethanol percentages. The experiments were performed using solar, E2.5%, E5%, E7.5%, and E10% ethanol-diesel blends and 0, 10, 20, 30, 40, 50 and 60 Nm engine loads. Several engine parameters i.e. power, brake specific fuel consumption, brake thermal efficiency, the exhaust gas temperature, and lubricating oil temperature were investigated. As a complement of the experiment, the exhaust emission characteristic of CO, UBHC and smoke were also investigated. The results indicate that the engine power and the indicated mean effective pressure increase with increasing of ethanol percentage. The brake specific fuel consumption and exhaust gas temperature decrease, meanwhile the lubricating oil temperature increase with increasing the ethanol content. From the experiments, as the increase of ethanol percentage content, the emission of CO, UBHC and smoke decrease [84].

Ethanol is an alternative renewable fuel produced from different agricultural products. The ethanol–Diesel emulsion technique is one of the techniques to use ethanol in Diesel engines. The most important advantage of this technique is to be able to use ethanol without any modification in Diesel engines. In this study, the effects of ethanol addition (10% and 15% in volume) to Diesel No. 2 on the performance and emissions of a four stroke cycle, four cylinder turbocharged indirect injection Diesel engine having different fuel injection pressures (150, 200 and 250 bar) at full load were investigated. 1% isopropanol was added to the mixtures to satisfy homogeneity and prevent phase separation. Experimental results showed that the ethanol addition reduces CO, soot and SO_2 emissions, although it caused an increase in NOx emission and approximately 12.5% (for 10% ethanol addition) and 20% (for 15% ethanol addition) power reductions. It was also found that increasing the injection pressure of the engine running with ethanol–Diesel fuel decreased CO and smoke emissions, especially between 1500 and 2500 rpm, with respect to Diesel fuel, while it caused some reduction in power [85].

Another experimental investigation is conducted to evaluate and compare the performance and exhaust emission levels of ethanol as supplement to the conventional diesel fuel, at blend ratios (by volume) of 5/95 and 10/90, in a fully instrumented, six-cylinder, turbocharged and after-cooled, direct injection (DI), Mercedes–Benz, mini-bus diesel engine installed at the authors' laboratory. The series of tests are conducted using each of the above fuel blends, with the engine working at two speeds and three loads. In each test, exhaust smokiness and exhaust regulated gas emissions such as (NOx), (CO) and total (UBHC) are measured. BSFC and BTE are computed from the measured fuel volumetric flow rate and calorific values [86].

The smoke density was significantly reduced with the use of the ethanol–diesel fuel blends with respect to that of the Neat Diesel fuel, with this reduction being higher the percentage of ethanol in the blend.

The NOx emissions remained the same or very slightly reduced with the use of the ethanol–diesel fuel blends with respect to those of the neat diesel fuel.

The CO emissions were equal or slightly reduced with the use of the ethanol–diesel fuel blends with respect to those of the neat diesel fuel, with this reduction being higher the percentage of ethanol in the blend.

The (UBHC) emissions were increased with the use of the ethanol–diesel fuel blends with respect to those of the neat diesel fuel, with this increase being higher the higher the percentage of ethanol in the blend.

Concerning the engine performance with the ethanol–diesel fuel blends against the neat diesel fuel case, a little higher specific fuel consumption was observed with increasing percentage of the ethanol in the blends, and a corresponding very slight increase of brake thermal efficiency.

Theoretical aspects of diesel engine combustion combined with the widely differing physical and chemical properties of the ethanol against those of the normal diesel fuel, are used to aid the correct interpretation of the observed engine behavior.

A practical conclusion is that the tested ethanol–diesel blends can be used safely and advantageously in the present bus diesel engine, at least in these small blending ratios [86].

Engine-out emissions from a Volkswagen model TDI engine were measured for three different fuels: neat diesel fuel, a blend of diesel fuel and additives containing 10% ethanol, and a blend of diesel fuel and additives containing 15% ethanol. The test matrix covered five speeds from 1320 to 3000 rpm, five torques from 15 Nm to maximum plus the 900-rpm idle condition, and most of the points in the FTP-75 and US-06 vehicle tests. Emissions of (PM), (NOx), (UBHC), and (CO) were measured at each point, as were fuel consumption, exhaust oxygen, and carbon dioxide output. PM emissions were reduced up to 75% when ethanol-diesel blends were used instead of neat diesel fuel. Significant reductions in PM emissions occurred over one-half to two-thirds of the test matrix. NOx emissions were reduced by up to 84%. Although the regions of reduced NOx emissions were much smaller than the regions of reduced PM emissions, there was considerable overlap between the two regions where PM emissions were reduced by up to 75% and NOx emissions were reduced by up to 84%. Such simultaneous reduction of both PM and

NOx emissions would be difficult to achieve by any other means. UBHC and CO emissions were also reduced in the regions of reduced PM and NOx emissions that overlapped. Because the ethanol-diesel blends contain less energy on both a per-unit-mass basis and a per-unit-volume basis, there was a reduction in maximum torque of up to 10% and an increase in BSFC of up to 7% when these blends were used [87].

The following essay investigates the behavior of farm tractor engine under full and real load conditions, by using fuels as Diesel, Diesel-20% ethanol and Diesel-30% ethanol mixtures. Concretely, it examines the farm Tractor with Diesel engine from the viewpoint of power and gas emissions and consumption. A series of laboratory instruments were used for the realization of the experiments. The results of full load tests using Diesel and Diesel - ethanol mixtures in Diesel Tractor engine, indicate that the CO exhaust gas content tends to decrease in case of using ethanol as fuel. This is probably caused by the presence of oxygen in the ethanol that participates in the combustion process. In the case of full load tests the combustion temperatures are sufficient for efficient combustion in all the cases of engine rpm. Therefore, the reduction of exhaust emissions with the ethanol content is natural. However, the power decreases as far as the ethanol is increasing [88].

2.5.19. Diesel Fuel with Direct Injection Ethanol

Renewable feedstock's and high octane rating make ethanol a promising alternative fuel. In contrast to the conventional approach of applying ethanol-gasoline blends in spark-ignition engines, this study investigates the potential of ethanol fuelling in a diesel engine to achieve higher efficiency. Experiments are performed using a single-cylinder version of a common-rail diesel engine that is widely used in passenger cars. A dual-fuelling technology is implemented such that ethanol is introduced into the intake manifold using a port-fuel injector while diesel is injected directly into the cylinder. The main focus is the effect of ethanol energy fraction and diesel injection timing on engine efficiency and tailpipe emissions. While these two parameters are varied, in-cylinder pressure measurement and subsequent analysis of indicated mean effective pressure, apparent heat release rate, and ignition delay, combustion phasing, and burn duration are performed. From the ethanol energy variation tests at fixed diesel injection timing, it is found that increased ethanol energy fraction increases the engine efficiency until the operation is limited by misfiring associated with over-retarded combustion phasing. By energy fraction, up to 60% of diesel is replaced by ethanol, which achieves 10% efficiency gain compared with diesel-only operation. Detailed analysis of the results reveals that the decreased burn duration is the primary cause for the efficiency gain, i.e. the fast burning of ethanol improves the combustion. However, the burn duration appears to increase with advancing the diesel injection timing at affixed ethanol energy ratio despite the fact that the highest indicated mean effective pressure of 1020 kPa is measured when the diesel injection timing is set at eight crank angle degrees before top dead centre, the most advanced diesel injection timing of this study. This is due to optimized combustion phasing such that the main heat release occurs near top dead centre, which out performs the increased burn duration. Therefore, both burn duration and combustion phasing should be considered to explain trends in the indicated mean effective pressure or efficiency of dual-fuel combustion engines. The tailpipe emissions suggest that UBHC, CO and NOx emissions increase with increasing ethanol fraction, which raises a question on the advantages of utilizing ethanol in a diesel engine. However, negligible smoke emissions are measured at ethanol energy ratio of 20% or higher suggesting that optimization of these emissions would be much easier compared with conventional diesel combustion [89].

The effects of ethanol fumigation on the inter-cycle variability of key in-cylinder pressure parameters in a modern common rail diesel engine have been investigated. Specifically, maximum rate of pressure rise, peak pressure, peak pressure timing and ignition delay were investigated. A new methodology for investigating the start of combustion was also proposed and demonstrated which is particularly useful with noisy in-cylinder pressure data as it can have a significant effect on the calculation of an accurate net rate of heat release indicator diagram. Inter-cycle variability has been traditionally investigated using the coefficient of variation. However, deeper insight into engine operation is given by presenting the results as kernel density estimates; hence, allowing investigation of otherwise unnoticed phenomena, including: multi-modal and skewed behavior. This study has found that operation of a common rail diesel engine with high ethanol substitutions

(> 20% at full load, > 30% at three quarter load) results in a significant reduction in ignition delay. Further, this study also concluded that if the engine is operated with absolute air to fuel ratios (mole basis) less than 15, the inter-cycle variability is substantially increased compared to normal operation [90].

Conventional diesel engines with ethanol as fuel are associated with problems due to high self-ignition temperature of the fuel. The hot surface ignition method, wherein a part of the injected fuel is made to touch an electrically heated hot surface (glowplug) for ignition, is an effective way of utilizing ethanol in conventional diesel engines. The purpose of the present study is to investigate the effect of thermal insulation on ethanol fueled compression ignition engine. One of the important ethanol properties to be considered in the high compression ratio engine is the long ignition delay of the fuel, normally characterized by lower cetane number. In the present study, the ignition delay was controlled by partial insulation of the combustion chamber (low heat rejection engine) by plasma spray coating of yttria stabilized zirconia for a thickness of 300 lm. Experiments were carried out on the glowplug assisted engine with and without insulation in order to find out the possible benefits of combustion chamber insulation in ethanol and diesel operation. Highest brake thermal efficiency of 32% was obtained with ethanol fuel by insulating the combustion chamber. Emissions of the UBHC, NOx and CO were higher than that of diesel. But the smoke intensity and was less than that of diesel engine. Volumetric efficiency of the engine was reduced by a maximum of 9% in LHR mode of operation [91].

Fossil fuels are the most imperative parameters to flourish the every sphere of modern civilization including industrial development, transportation, and power generation and easing the accomplishment of works. The rapid increase in usage of fossil fuel has unavoidable deleterious effect on environment. The international consciousness for environment protection is growing and ever more strict emission legislations are being enacted. Simultaneously the storage of fossil fuel is depleting. Hence, the above situations promote the scientists to find alternative sustain able fuels along with their suitable using technique which will reduce the pollutant emission and will be applicable for gaining satisfactory engine performance. In these perspectives, alcohol fumigation is getting high demand as an effective measure to reduce pollutant emission from diesel engine vehicles. Alcohol fumigation is a dual fuel engine operation technique in which alcohol fuels are premixed with intake air. The aim of this paper is to identify the potential use of alcohols in fumigation mode on diesel engine. In this literature review, the effect of ethanol and methanol fumigation on engine performance and emission of diesel engine has been critically analyzed. A variety of fumigation ratios from 5% to 40% have been applied in different types of engines with various types of operational mode. It has been found that the application of alcohol fumigation technique leads to a significant reduction in the more environment concerning emissions of (CO_2) up to7.2%, (NOx) up to 20% and (PM) up to 57%. However, increase in (CO) and (UBHC) emission have been found after use of alcohol fumigation. Alcohol fumigation also increases the BSFC due to having higher heat of vaporization. BTE decreases at low engine load and increases at higher engine load [92].

Other experiments were conducted on a four-cylinder direct-injection diesel engine with methanol or ethanol injected into the air intake of each cylinder, to compare their effect on the engine performance, gaseous emissions and particulate emissions of the engine under five engine loads at the maximum torque speed of 1800 rev/min. The methanol or ethanol was injected to top up 10% and 20% of the engine loads under different engine operating conditions. The experimental results show that both fumigation methanol and fumigation ethanol decrease (BTE) at low engine load but improves it at high engine load; however the fumigation methanol has higher influence on the BTE. Compared with Euro V diesel fuel, fumigation methanol or ethanol could lead to reduction of both NOx and particulate mass and number emissions of the diesel engine, with fumigation methanol being more effective than fumigation ethanol in particulate reduction. The NOx and particulate reduction is more effective with increasing level of fumigation. However, in general, fumigation fuels increase the UBHC, CO and NO_2 emissions, with fumigation methanol leading to higher increase of these pollutants. Compared with ethanol, the fumigation methanol has stronger influence on the in-cylinder gas temperature, the air/fuel ratio, the combustion processes and hence the emissions of the engine [93].

Also other study is to investigate the suitability of neat ethanol (95%) as an alternative fuel in an engine converted from diesel to ethanol, and experimentally determine the effect of different main fuel jets at carburetor on engine exhaust emissions and fuel consumption at high engine compression ratio (CR) of 14. For this purpose, three different diameter main fuel jets (1.40 mm, 1.50 mm, 1.60 mm) were used for ethanol fuel tests in a naturally aspirated four-stroke pre-chamber diesel engine. Tests were made at full-load conditions at speeds between 1200 and 2600 rpm at intervals of 400 rpm. In addition, cold-start system was developed and engine could be operated at reliable idle speed and medium speed down to 0°C ambient temperatures. The results show that UBHC emissions increase with increasing fuel jet diameter and CO, PM and UBHC emissions reduce significantly at idle speed and medium engine speeds [94].

Finally, the effects of ethanol fumigation (i.e. the addition of ethanol to the intake air manifold) and ethanol-diesel fuel blends on the performance and emissions of a single cylinder diesel engine have been investigated experimentally and compared. An attempt was made to determine the optimum percentage of ethanol that gives lower emissions and better performance at the same time. This was done by using a simple fumigation technique. The results show that both the fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better than when using blends. The optimum percentage for ethanol fumigation is 20%. This percentage produces an increase of 7.5% in BTE, 55% in CO emissions, 36% in UBHC emissions and reduction of 51% in soot mass concentration. The optimum percentage for ethanol-diesel fuel blends is 15%. This produces an increase of 3.6% in BTE, 43.3% in CO emissions, 34% in UBHC and a reduction of 32% in soot mass concentration [95].

To diversify the mix of domestic energy resources and to reduce dependence on imported oil, ethanol is widely investigated for applying in combination with Diesel fuel to reduce pollutants, including smoke and NOx. Present work aims at developing a fumigation system for introduction of ethanol in a small capacity Diesel engine and to determine its effects on emission. Fumigation was achieved by using a constant volume carburetor. Different percentages of ethanol fumes with air were then introduced in the Diesel engine, under various load conditions. Ethanol is an oxygenated fuel and lead to smooth and efficient combustion. Atomization of ethanol also results in lower combustion temperature. During the present study, gaseous emission has been found to be decreasing with ethanol fumigation. Results from the experiment suggest that ethanol fumigation can be effectively employed in existing compression ignition engine to achieve substantial saving of the limited Diesel oil. Results show that fumigated Diesel engine exhibit better engine performance with lower NOx, CO, CO_2 and EGT. Ethanol fumigation has resulted in increase of (UBHC) emission in the entire load range. Considering the parameters, the optimum percentage was found as 15% for ethanol fumigation [96].

Another study was undertaken to evaluate and compares the effects of ethanol fumigation and ethanol diesel fuel blends on the performance and exhaust emissions of an agriculture diesel engine. Ethanol fumigation was achieved by using a simple carburetor whereas ethanol diesel blend was prepared by inline mixing. 10%, 20%, 30% ethanol was supplemented in diesel in both the techniques. The series of test were conducted using each of the above mentioned ethanol based fuels to test an agriculture diesel engine and performance and emission characteristics were evaluated. The BTE showed an upward trend and BSFC exhibited a downward trend on ethanol substation than diesel fuel operation. This may be mainly due to increase in the ignition delay upon ethanol substitution, so a rapid rate of energy is released which reduces the heat loss from the engine because there is not enough time for this heat to leave the cylinder through heat transfer to the coolant Exhaust temperature, in general, were found lower for ethanol based fuel than diesel as large amount of latent heat absorbed by combustion chamber to evaporate ethanol and thus EGT decreases. As far as CO, UBHC emissions are concerned, they were found to increase with ethanol substitution than diesel operation. The fumigation showed lower increase in these emissions as compared to blending. A thickened quench layer created by the cooling effect of vaporizing alcohol or increase in ignition delay could have played a major role in the increased CO production. The NOx emissions were found lower for ethanol based fuels than the diesel fuel. This may be due to reduction in combustion temperature to reduce due to introduction of ethanol as vaporization of ethanol takes heat from combustion chamber. The smoke opacity was found lower for blended or fumigated ethanol as the engine is running ‘leaner’, with the combustion being now assisted by the presence of the fuel-bound oxygen of the ethanol even in locally rich zones. Also, diesel fuel has a high tendency to smoke due to its low H/C ratio and the nature of its combustion process [97].

Limited crude oil resources and growing environmental awareness have raised a huge led to a huge motivation to seek new fuel alternatives for combustion engines. In this study the goal was to research the potential of ethanol as a compression ignition engine fuel utilizing dual-fuel combustion technology. In the studied DF (dual-fuel) concept, the primary fuel is injected into the intake manifold and it is ignited by injecting diesel fuel into the cylinder near the TDC (top-dead centre). A similar concept has been widely used in natural gas dual-fuel engines. By using ethanol as a primary fuel instead of gaseous fuel, the dual-fuel concept could be a true alternative for on-road and off-road heavy-duty vehicles without the obstacles of energy density and fuel distribution challenges of the gaseous fuels. The research engine in this study was a high speed heavy duty diesel engine modified for dual fuel combustion. The engine was equipped with a common-rail diesel injection system, which gave flexibility to modify the diesel injection to control the combustion. Especially the relationship between pilot and main diesel injection was investigated. The gathered results were promising. A maximum ethanol/diesel mass ratio of around 90% was achieved at high load conditions [98].

2.5.20. Diesel Fuel with Ethanol-Water Blends

Three groups of n-butanol–diesel blends with 0, 0.5, and 1.0 wt.% water-content were investigated to simulate the hydrated butanol produced by acetone–butanol–ethanol fermentation and a simple distillation treatment. Both 30-day standing and centrifugal test results showed that 15 wt.% n-butanol (BT) was the minimum additive ratio to stabilize the 1.0 wt.% water content diesel blend, while those blends that contained 0 or 0.5 wt.% water could remain as stable one-phase clear liquids by adding just 5 wt.% BT. These stable diesel blends were further examined in a heavy-duty diesel-fueled engine generator (HDDEG). Using BT-diesel blends increased the indicated specific fuel consumption (ISFC) because of the lower heating value of n-butanol, while the micro-explosions that occurred could reduce the ISFC when using 0.5 wt.% water-containing BT-diesel blends. NOx emissions increased with the increasing BT content at a low additive ratio (5–15 wt.%), and reduced when adding a higher amount of BT (>15 wt.%). PM, total-PAHs, and total-BaPeq emissions were all significantly reduced when the increasing BT additive ratio contained either 0, 0.5, or 1.0 wt.% water because of the lower sulfur and higher oxygen fuel contents. On the other hand, the CO emission level went up with the addition of BT. Notably, the diesel blends with 0.5 wt.% water only slightly increased ISFC when low fractions of BT were added, i.e. 0.40% and 0.81% ISFC increases with the addition of 5 and 10 wt.% BT, respectively. In addition, there were significantly lower NOx, PM, Total-PAHs and Total-BaPeq, emissions with the blends than with regular diesel. With the aim of achieving both good energy performance and less pollutant emissions, the 5 and 10 wt.% BT additive with 0.5% water content blends were the most suitable for practical use in an HDDEG without any engine modifications or changes in controls [99].

2.5.21. Diesel Fuel with Ethanol–Water Mixtures into the Inlet Air

A heavy duty common rail marine diesel engine operating with two stage injection is tested under load on a test bench with vaporized ethanol–water mixtures mixed into the inlet air at various rates. Ethanol/water mixture strengths of 93%, 72% and 45% by mass are tested. Results are presented for two engine loads at 1800 rpm, with brake mean effective pressure (BMEP) 17 bar and 20 bars. At each test point, constant engine speed and brake torque are maintained for various rates of aqueous ethanol addition. Small increases in brake thermal efficiency are measured with moderate rates of ethanol addition at a BMEP of 20 bars. Exhaust emissions of oxides of nitrogen, carbon monoxide, hydrocarbons, oxygen and carbon dioxide, and exhaust opacity are measured. CO emissions and exhaust opacity tend to increase with increased ethanol addition. NOx emissions tend to decrease with increased ethanol addition and with increased water content. Hydrocarbon emissions remain low, near the detection limit of the analyzer. Cylinder pressure and the electronically controlled two stage liquid fuel injection timing are recorded with a high speed data acquisition system. Apparent heat release rate is calculated from the measured cylinder pressure. The apparent heat release rate and fuel injection timing together allow analysis of the mechanism of the combustion process with ethanol fumigation. Two stage injections involve a small pre-injection of diesel fuel to reduce early pressure rise rates in normal diesel engine combustion. Even though injection timing is retarded by the Engine Control Unit as more ethanol is added, combustion timing effectively advances due to the effect of two stage injection. Where the ethanol/air mixture strength is above the lower flammability limit at compression temperatures, the mixture is ignited by the pre-injection and begins to burn rapidly by flame propagation and/or autoignitive propagation before the main liquid fuel injection begins. This occurs for ethanol energy substitution rates greater than 30%. Two distinct peaks in heat release rate appear at the higher ethanol rates. Severe knock becomes apparent for 34% ethanol. Two stage injections may be disadvantageous in these circumstances [100].

CHAPTER THREE

Methodology

3. Research Methodology

3.1. Preface

The experimental setup was carried in a single-cylinder naturally aspirated, four-stroke, water cooled, and direct-injection diesel engine with a hemispherical bowl in piston combustion chamber. **Table (3.1)** shows the specification of the engine. The engine and associated dynamometer were fully instrumented to record engine performance data.

Experimental was setup to integrate the measurement of engine performance parameters with control level of Ethanol/Water DI fuel. These will be explaining later.

Table 3.1: Engine Specifications

Engine type		GKW
Bore (cm) × stroke (cm)		8.75 x 11
Displacement (cm3)		631.56
Compression ratio		17.5:1
Fuel		Diesel, Ethanol or Water
Ignition source		Diesel injection
Fuel injection		Direct injection
Injection system		Pencil-type, three-nozzle
Combustion chamber		Hemispherical bowl in piston
Number of valves per cylinder		Two
Rated power (HP)		7.5@ 1600 rpm

3.2. Experimental Method and Materials

3.2.1. Engine Modification

The principal modification to the diesel engine was the insert of a nozzle, which was located on the inlet manifold; **Figure (3.1)** and **Figure (3.2)**. The ethanol/water was mixed (by vol.) outside these will be explaining later. The ethanol/water was injected by external pump; see **Table (3.2)** specification of pump.

Table 3.2: Pump Specifications

Type		CNPA1002PPE200A01
Power Supply		230V-50/60Hz 11W-0.12A
Dosing Rate		2.1 l/h-10bar

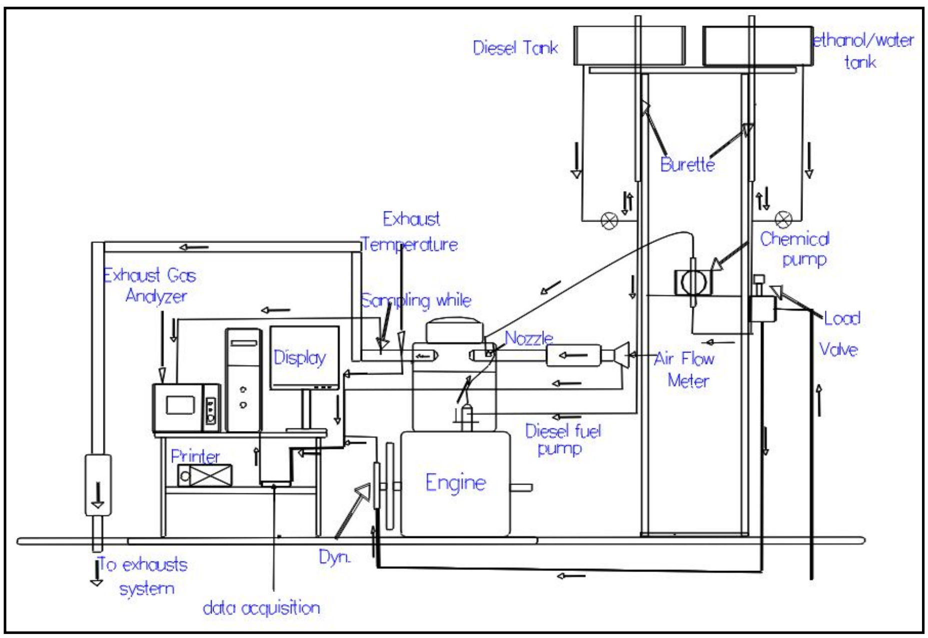

Figure 3.1: Schematic Arrangement of the Engine Test Bed, Instrumentation and Data Logging System

Figure 3.2: The Experimental Setup

The DI or spray controlling through rotation two keys; **Figure (3.3)**, one for capacity flow rate and another for pressure which points to different settings on around keys for indication purpose.

Figure 3.3: The DI or Spray Controlling Through Rotation Two Keys

The inlet air was mixed with ethanol/water inside manifold; **Figure (3.4)**. As position of nozzle, ethanol/water after pumped return liquid with friction wall manifold, ethanol/water disadvantage with regard to their high latent heat of vaporization, and positing intake air will be improve the vaporization the injected ethanol/water in the inlet manifold. This will be improve hydraulic block engine, and delay combustion of Ethanol/Water.

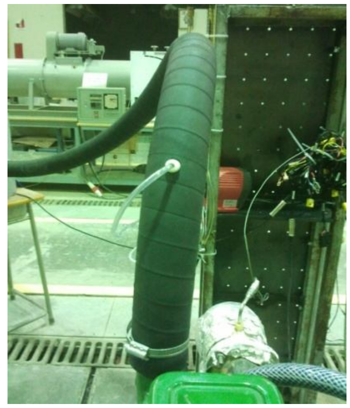

Figure 3.4: Position of the Nozzle

A proper selection of a nozzle type and size is essential for correct and accurate pesticide application, a more homogeneous mixture; **Figure (3.5)**. It is also increase flame speed, improves ignitability, and wall friction. The physical and chemical natural even during the delay period are speed up, resulting in similar combustion ethanol/water-diesel blends and higher peak temperature. Since the nozzle type, size, and insert location less dense, the component of the mixture is producing reduced. This is due to the perfect fuel/air being more normal into cylinder; an improvement in combustion efficiency is resulted.

Figure 3.5: Nozzle

3.2.2. Fuel Preparation

For the chemical preparation of the ethanol/water three different blends are blended by volume in the laboratory:

3.2.2.1. Water

Limit the duration of the cleaning to one day or at the most two days. The properties of water see appendix (A)

3.2.2.2. EW25

For the preparation of these mixing, 250 ml of ethanol, 750 ml of water were mixed in flask on volume basis to prepare 1 liter of mixing.

3.2.2.3. EW50

For the preparation of these mixing, 500 ml of ethanol, 500 ml of water were mixed in flask on volume basis to prepare 1 liter of mixing.

3.2.2.4. EW75

For the preparation of these mixing, 750 ml of ethanol, 250 ml of water were mixed in flask on volume basis to prepare 1 liter of mixing.

The properties of the diesel, and ethanol, fuel used see appendix (B).

3.2.3. Fuel Consumption

❖ **Diesel Fuel**

The volume of fuel consumed over given time period was measured using burette accuracy ± 0.1 ml, and +0.1 second for the time over which the fuel was consumed. It is repeated three times due to inaccuracy. The fuel flow rate records manually, see **Figure (3.6)** the average reading.

❖ **Ethanol/Water Fuel**

This was similar diesel fuel measurement see **Figure (3.6)**. The accuracy for the fuel consumed was ±0.1 ml, and +0.1 second for the time over which the fuel was consumed.

Figure 3.6: Diesel Tank, Burette for Diesel, Ethanol/Water Tank, Burette for Ethanol/Water

3.2.4. Emissions Measurement

SV-5Q automobile exhaust may, in accordance with non-light-disguising infrared absorption method, via micro-computer analysis, directly measure the thickness of UBHC, CO and CO_2 in the exhaust gas of vehicles and inspect the density of NO and O_2 via Electrochemical sensor so as to calculate excess air coefficient (λ) see **Figure (3.7)**. The analyzer introduces advanced foreign technology. When used Neat Diesel, Ethanol–Diesel DI, Water–Diesel DI, and different blends Ethanol/Water–Diesel. Appendix (C) shows the environment conditions, measurement range, resolution, and allowed error for SV-5Q. It is composed of complete imported machinery cores, boasts the advantages of accuracy measurement, high endurance and speed. As an Intelligent Equipment, it is equipped with Microprocessor, and LCD. Besides, it also has equipment's such as, temperature sensor, inner Micro Printer.

Figure 3.7: Gas Analyzes, Sampling Tube in Exhausts Manifold, Display Gas Analyzes

Using, after setup gas analyzes, insert sampling tube in exhausts manifold see **Figure (3.7)**, Make sure that fronted filter, "left time: 10minitues", which means the left time of warming up by countdown is 10 minutes. After preheating, the analyzer will enter sub menu "leaking check" automatically to check if gas lines system leaks. When the analyzer enters condition of zeroing, it will show on the bottom of screen: "zeroing". After that, the notice will disappear. When it finishes zeroing, the analyzer will inspect UBHC remainder automatically. When using a period of time, Sometimes this change might influence the measuring results. Therefore, I have to adjust zero after a certain time (Generally once per half an hour). After the record is selected, the menu will be as followed. After enter measurement menu, press "OK" to start the analyzer. The air pump will work. Then insert sampling probe head to the exhaust pipe of the vehicle to be measured to the length of 400cm or so, the screen will show the real time values of UBHC, CO, CO_2, and O_2 in gas. The analyzer is equipped with micro-printer. To print data, press "LEFT" key, then the measurement data will be printed immediately.

3.2.5. Emissions Testing Selection

A conducted of the Neat Diesel (without ethanol/water DI) were level of emissions, performance establish power, torque, Brake Specific Fuel Consumption, and Brake Thermal Efficiency. Engine speed chosen for this study were 600, 800, 1000, 1200, 1400, and 1600 rpm to covered full range engine. At each speed, the engine was loaded to maximum torque.

The Ethanol/Water DI studied in this research involved five experiential settings:

1. Experimental setting involved of Water DI, into the air intake of the engine. The flow rates and pressure of injected was 0.96 L/hr. @6bar, 1.12 L/hr. @6bar, 1.85 L/hr. @10bar, and 2.16 L/hr. @10bar. Diesel supply at all test time by original engine fuel pump. The experiment matched the targeted engine speeds chosen from the Neat Diesel test.

2. Experimental setting involved of EW25 DI, into the air intake of the engine. The flow rates and pressure of injected was similar Water DI. Diesel supply at all test time by original engine fuel pump. The experiment matched the targeted engine speeds chosen from the Neat Diesel test.

3. Experimental setting involved of EW50 DI, into the air intake of the engine. The flow rates and pressure of injected was similar Water DI. Diesel supply at all test time by original engine fuel pump. The experiment matched the targeted engine speeds chosen from the Neat Diesel test.

4. Experimental setting involved of EW75 DI, into the air intake of the engine. The flow rates and pressure of injected was similar Water DI. Diesel supply at all test time by original engine fuel pump. The experiment matched the targeted engine speeds chosen from the Neat Diesel test.

5. Experimental setting involved of Ethanol DI, into the air intake of the engine. The flow rates and pressure of injected was similar Water DI. Diesel supply at all test time by original engine fuel pump. The experiment matched the targeted engine speeds chosen from the Neat Diesel test.

3.3. Specification of Experimental Equipment

3.3.1. Overall Lay-out of Test Laboratory

The diesel engine is situated in cell (3) of Thermodynamic Laboratory at the University of The King Saud. It is hardwearing engine, **Figure (3.8)**. This engine was designed to ability of research purposes. The test laboratory is semi-soundproofed allows the engine. Exhaust fumes are drawn down an exhaust pipe and into a main chimney. There is central air-conditioning and a fan on the ceiling for heating and ventilating of the test laboratory. Personnel were available for instrumentation and troubleshooting needs. System supply water used for cooled engine and hydraulic dynameters.

3.3.2. DYNOmiteTM Dynamometer

The diesel engine coupled to hydraulic dynameters. DYNO-MAX allows a Personal Computer to monitor and control dynamometer. It communicates, via the PC's port(s), with one or more DYNOmite Computer (or third-party) Data Acquisition boards. Because the DYNOmite computer handles most of the high-speed data acquisition and control, it reduces the load on PC. The PC primarily handles just the data display, storage, post processing, and printing. DYNO-MAX leverages the PC's (and Windows operating system's) flexibility, permitting expanding and adapting the dynamometer system as needs change. “Pro” Console and computer provide a professional dyno operating station. This combination allows full remote operation of any engine and high-end dyno cell’s equipment (optional controls and DYNO-MAX 2000 software). Each “Pro” console is equipped with a genuine Dell PC (running Intel’s dual-core Hyper-Threaded Pentium processors) and a high resolution LCD flat panel monitor. A billet aluminum, adjustable range, manual throttle control is installed (to the operator’s left), while provisions to mount the DYNOmite load valve control are on the right.

Backup analog gauges (such as water temperature, oil pressure, fuel pressure, and tachometer) plus manual override switches, are ready for wiring into engine and test cell see **Figure (3.8)**.

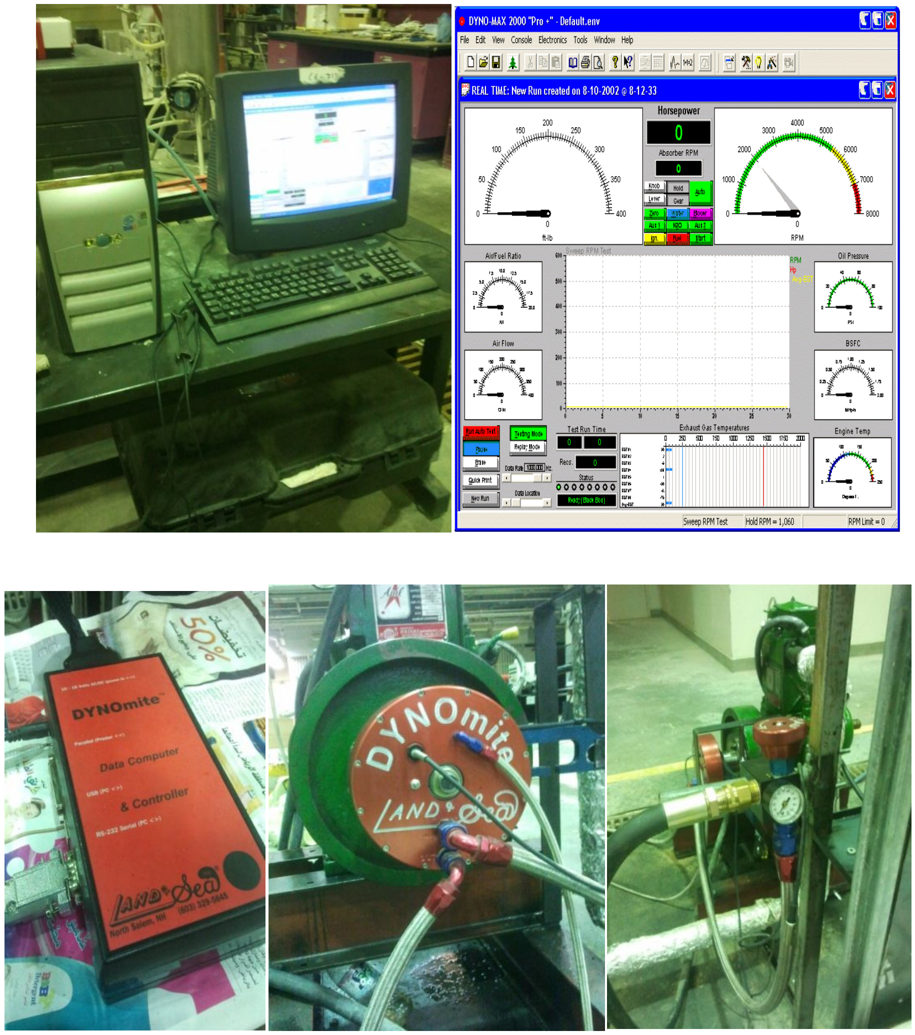

Figure 3.8: Hydraulic Dynameters. And A Personal Computer to Monitor and Control Dynamometer

DynoMATE and dragMATE Relay Control Modules both add several extra output lines to DYNOmite dynamometer system. A“dynoMATE” module adds eight additional DYNO-MAX 2000 “Pro” digital to analog switched outputs – per module. A single “dragMATE” module provides 5-volt logic control for all 14 lights on a standard NHRA style Xmas tree. Each module requires one RS-232 serial port (or USB to RS-232 adapter) see **Figure (3.8)**. Full Function Data Harness allows the

DYNOmite Data Acquisition Computer to record EGTs, fuel flow, pressures, etc. and control other advanced functions when used in conjunction with the appropriate sensors, controls, and software upgrades. Load Servo Valve is one of the most popular DYNOmite accessories. One-handed throttle operation during even dynamometer test sessions. Just open the throttle to have the computer take over loading the engine, millisecond response outperforms. PWC and other 9" DYNOmite Absorber Adapters for mounting the 9" DYNOmite water brake on many personal water crafts or specific snowmobiles, without engine removal from the boat or sled see **Figure (3.8)**.

PWC adapters utilize a short torque arm and special coupling that bolts onto the pump housing (after first removing the jet's nozzle). For sleds, special taper arbor/rotor combinations fit the 9" absorber to applications not served by the standard DYNOmite dual-taper arbor. DYNO-MAX 2000TM“Pro” Version 9.38 is the industry's ultimate professional dynamometer data collection and analysis software. Its cutting edge, user friendly 32-bit interface and depth of advanced features make this the best dynamometer control package available. Jack Shaft RPM Pick-Up Transducer Kit provides MPH, shift-ratio, and output shaft RPM. By monitoring both the output shaft and crankshaft RPM, the computer can calculate shift ratio and MPH. When using DYNO-MAX software.

Accelerometer monitors acceleration and braking .The DYNO-MAX software option can automatically calculate real-time delivered rear wheel or engine Hp. for DYNOmite water brake absorbers feature full-bridge strain gauges for maximum signal-to-noise ratios and minimum temperature sensitivity. The arm's patented design is immune to moment-arm stop's radius. This feature eliminates torque reading errors from geometry changes. Balanced four-wire configuration provides 15 millivolts of output at 100% of full scale load. Each arm comes prewired and all ready to plug into the torque-arm harness lead of compatible DYNOmite Data Acquisition Computers.

3.3.3. Exhaust Gas Temperature

Thermocouples are a must accessory for jetting two strokes, setting up SV-5Q systems, or just safely dynoing high output engines see **Figure (3.9)**. While power is often gained by leaning out, this can take the engine to the point of melting pistons. The fast acting thermocouple reading per DYNOmite-Pro board set.

Figure 3.9: Sensor Exhaust Gas Temperature

3.3.4. Air Flow Meter Turbine

Air flow rates for measuring the intake airflow of diesel engine. These low inertia turbines measure the intake airflow either directly through a plenum see **Figure (3.10)**. Turbines are available in sizes to accommodate a wide range of air horn sizes and CFM requirements. (Requires Data Channel Expansion Unit part #430-454 and DYNO-MAX software equipped PC.)

Figure 3.10: Air Flow Meter Turbine

3.4. Experimental Procedure

The testing procedure is as follows. The engine is single-cylinder, four-stroke, water-cooled, direct injection (DI) compression-ignition engine, handle start; handle controlled. The engine has 87.5mm Bore, 110mm Stroke with a compression ratio of 17.5, and diesel engine rated at 5.9 kW (8 bhp) at 1600 rpm. A gas analyzer is used which consists of SV- 5Q exhaust analyzer was used to measure the CO, CO_2, UBHC, O_2, NOx and λ.

After completion of standard warm up procedure, the engine was tested on a matrix of six speeds started in Neat Diesel-only mode during the engine warm-up period. The whole experimental plan was realized in two stages: (i) running engine with Neat diesel; and (ii) running engine with Ethanol/Water–Diesel. Each test was repeated three times to make sure the data were reliable. The relative standard deviations of the repeated test results are: <3.1% for fuel consumption, <3.2% for thermal efficiency (get), <24% for CO, <29% for UBHC, <9% for NOx and <30% for CO_2, O_2, and λ.

First stage running engine with Neat Diesel at six speed (600, 800, 1000, 1200, 1400, 1600) rpm, i.e. when operation 600 rpm measure after engine stability horsepower, torque, air flow rate, diesel fuel consumption, and exhaust temperature. All of this is parameter saving and print by DYNO-MAX software. Through stability insert sampling probe head to the exhaust pipe of the engine to be measured to the length of 400cm or so, the screen will show the real time values of UBHC, CO, CO_2, and O_2 in gas. Then printing result by micro-printer.

Second stage includes some setup as follows:

- Water DI.
- EW25 DI.
- EW50 DI.
- EW75 DI.
- Ethanol DI.

By chemical pump four times setting, 0.96 L/hr. @6bar, 1.12 L/hr. @6bar, 1.85 L/hr. @10bar, and 2.16 L/hr. @10bar for each speed. **Table (3.3)** shows (Ethanol/Water)–Diesel ratio (by vol.). Mixing ratios Ethanol/Water (0%, 25%, 50%, 75% and 100%) (by vol.).

Table 3.3: Ratio Ethanol/Water–Diesel (by Vol.)

Setting Pump	Abbreviation	600 rpm	800 rpm	1000 rpm	1200 rpm	1400 rpm	1600 rpm
0.96 L/hr. @6 bar	A	1.25	0.99	0.82	0.70	0.62	0.55
1.12 L/hr. @6 bar	B	1.45	1.16	0.96	0.82	0.72	0.64
1.85 L/hr. @10 bar	C	2.39	1.90	1.58	1.35	1.18	1.05
2.16 L/hr. @10 bar	D	2.79	2.22	1.84	1.58	1.38	1.22

Each setup the chemical pump same manner as described above into the intake manifold. After that adjust speed 600rpm and start dynamotor. The observation made during the test for measure horsepower, torque, air flow rate, diesel fuel consumption, and exhaust temperature. All of this is parameter saving and print by DYNO-MAX software. Through stability insert sampling probe head to the exhaust pipe of the engine to be measured to the length of 400cm or so, the screen will show the real time values of UBHC, CO, CO_2, O_2, NOx and λ in gas. Then printing result by micro-printer. Repeat for speed (600-1600) rpm sequence 200 rpm.

Pollutant emissions reduction from diesel engines requires detailed knowledge of the combustion process. However, the complex nature of the combustion process in a diesel engine makes it difficult to understand the events occurring in the combustion chamber which determine the emissions of exhaust gases including CO, CO_2, O_2, UBHC and NO. Several studies have reported on the effects of fuel and engine parameters on diesel exhaust emissions.

Engine performance and exhaust emission tests are very important to observe effects of a fuel on the performance and emissions of the engine. Since, these test results indicate an idea whether the fuel is used in an engine efficiently and without any problem or not. For that reason, it is necessary to determine performance parameters of an engine. There are several performance parameters, such as; BP, BSFC, BTE, VE. It is necessary to find these parameters which can be obtained using measurement values of fuel and air consumptions, heating capacity of the fuel, torque and speed.

CHAPTER FOUR

Results and Discussion

4. Results and Discussion of Experimental

4.1. Preface

This chapter illustrates results of engine emissions and performance for the Neat Diesel fuel, Ethanol/Water–Diesel by two sections. The emissions and performance characteristics will be studied and compared with the base fuel.

1st section investigation the differences in the measure exhaust emissions of the two water/ethanol–diesel fuel blends from the baseline operation of the engine, i.e. when working with Neat Diesel fuel, are determined and compared. Theoretical aspects of diesel engine combustion combined with the widely differing physical and chemical properties of the ethanol/water–diesel against those for the diesel fuel, are used to aid the correct interpretation of the observed engine behavior.

2nd section of the investigation was the assessment of engine performance of Neat Diesel along the ethanol/water–diesel different mixing by volume for different injection pressures and flow rate compared to the data obtained for base diesel fuel.

4.2. Effect of Ethanol/water addition to diesel fuel on exhaust emissions

4.2.1. Effect of Water−Diesel DI on Nitrogen Oxides Emissions

Figure (4.1) shows the wide range ratio Water–Diesel DI effect on NOx emissions vs. engine speed. NOx emission values that measured at full load condition in the case of Water at different flow rates. NOx emissions decrease with increase engine speed, and the NOx emissions of the engine using the Water–Diesel showed a similar tendency. It is well known that NOx formation rate strongly depends on peak temperature and duration of combustion at peak temperature in the cylinder. Thus, when the Water–Diesel is performed, peak temperatures decreased compared to that of Neat Diesel engine. The minimum NOx emission is 75 ppm at

1600 rpm in case Neat Diesel condition. According to Neat Diesel NOx emission values, the highest change is 83% at 800 rpm in case (A) condition, the lowest change is 13.9% at 600 rpm in case (B) condition. However, NOx emissions in case (A) are the minimum values due to the fact that water in the form of micrometer sized droplets exerts some positive effects on the combustion of the fuel and exhaust emissions, frequently NOx.

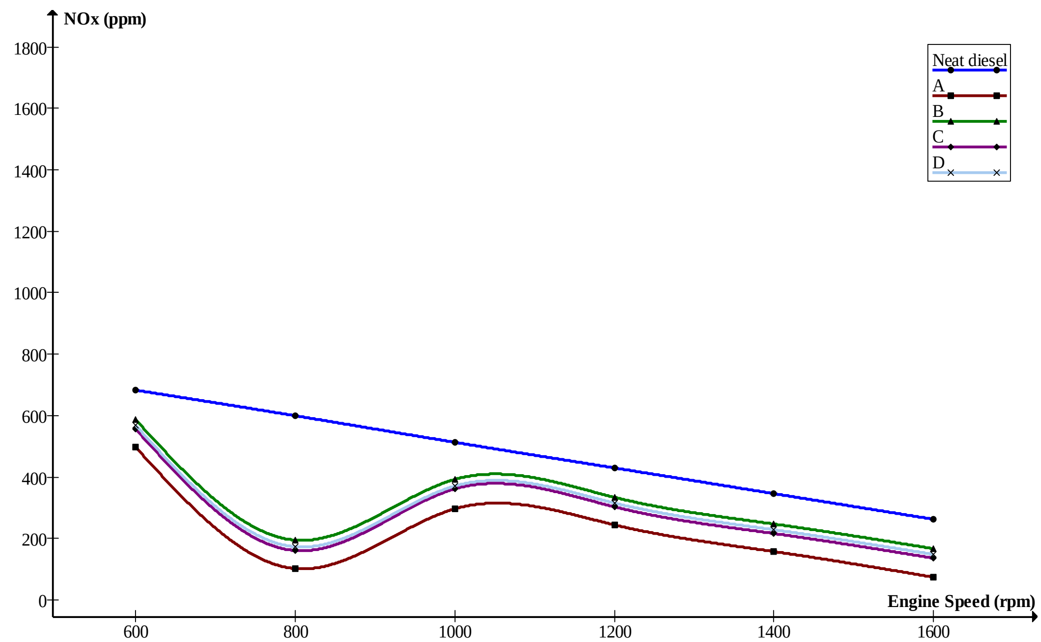

Figure 4.1: Wide Range Ratio Water–Diesel DI Effect on NOx Emissions vs. Engine Speed

4.2.2. Effect of EW25–Diesel DI on Nitrogen Oxides Emissions

Figure (4.2) shows the wide range ratio EW25–Diesel DI effect on NOx emissions vs. engine speed. NOx emission values that measured at full load condition in the case of EW25 at different flow rates. NOx emissions decrease with increase engine speed, and the NOx emissions of the engine using the EW25–Diesel showed a similar tendency. The two reasons for reduction in NOx emissions using, first injection water in diesel engines decreased the flame temperatures in the combustion chamber, second ethanol combustion production water; this is amount of water desorbing flame temperature. EW25–Diesel DI is lower NOx emissions than Neat Diesel fuel. No significant change in NOx at operation (1000 – 1600) rpm, in case (B, C), the reason effective of ethanol and water the same, first one increase and second decrease. The minimum NOx is 205 ppm at 1600 rpm in case (A) condition. According to Neat Diesel NOx emission values, the highest change is 61.3% at 800 rpm in case (A) condition, the lowest change is -0.19% at 1600 rpm in case (C) condition. However, NOx emissions in case (A) are the minimum values due to the fact that EW25 in the form of micrometer sized droplets.

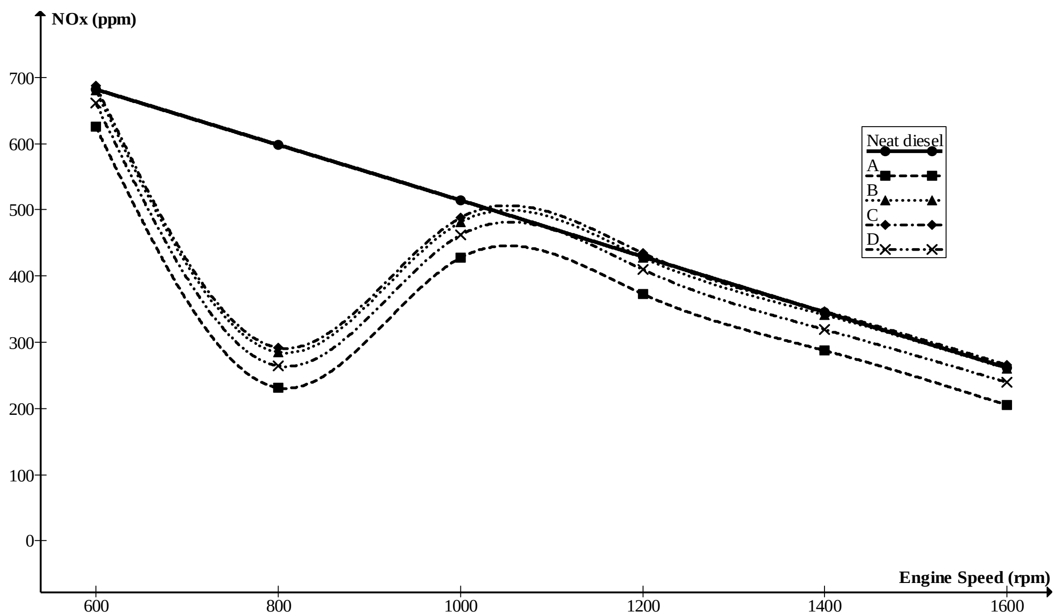

Figure 4.2: Wide Range Ratio EW25–Diesel DI Effect on NOx Emissions vs. Engine Speed

4.2.3. Effect of EW50–Diesel DI on Nitrogen Oxides Emissions

Figure (4.3) shows the wide range ratio of EW50–Diesel DI effect on NOx emissions vs. engine speed. NOx emissions values were measured at full load condition in the case of EW50 at different flow rates. NOx emissions decrease with increase engine speed for Neat Diesel, using EW50 DI on case (A, B, and C). NOx emissions of the engine using the EW50–Diesel DI on case (D) showed contradicted tendency. Because the burn time for the reactant mixture was shortened due to the increase in engine speed. Since the EW50 fuel was mixed with a larger quantity of excess air, the equivalent ratio of the fuel mixture decreased. Exhaust gas dilution by the excess air is another cause for the decrease in NOx.EW50–Diesel DI is lower NOx emissions than Neat Diesel fuel. This is because EW50 is an oxygen-rich fuel. When the oxygen content in the fuel blends increases, the ignition delay is shortened. The amount of premixed fuel and peak burning temperature were lowered, leading to the drop in NOx emissions

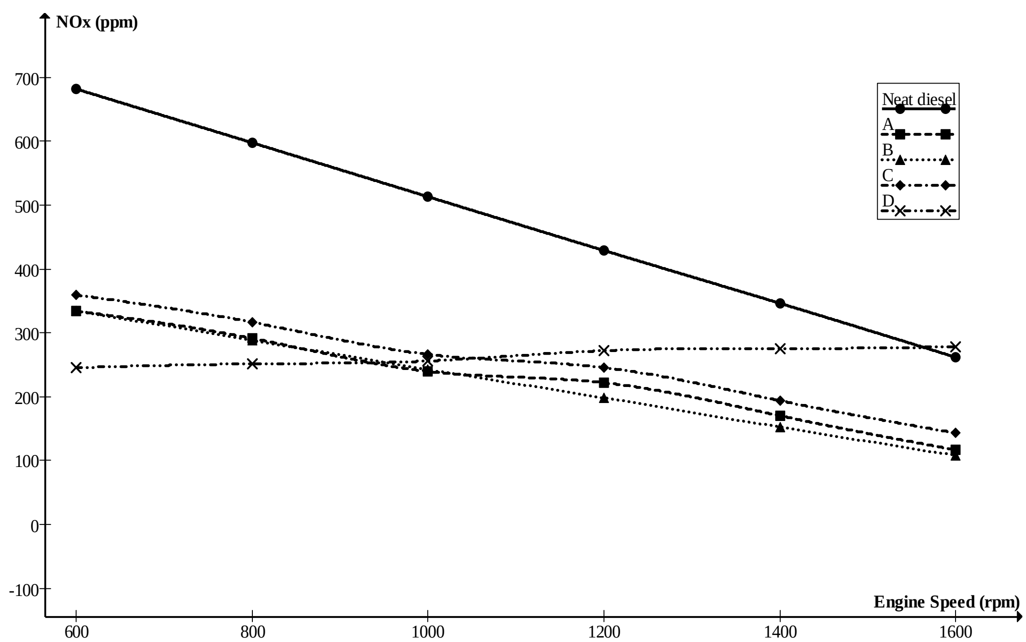

Figure 4.3: Wide Range Ratio EW50–Diesel DI Effect on NOx Emissions vs. Engine Speed

The minimum NOx is 108 ppm at 1600 rpm in case (B) condition. According to Neat Diesel NOx emission values, the highest change is 64.1% at 600 rpm in case (D) condition, the lowest change is -0.65% at 1600 rpm in case (D) condition. However, NOx emissions in case (D, B) are the minimum values, when the engine speed increased from (600-900 rpm, and 900-1600 rpm) respectively. The lower cetan number of the ethanol fuel causes to longer ignition delay, and so leads to higher combustion temperature in the premixed combustion mode. Whereas, ethanol fuel generally have lower flame temperature, due to lower energy content and higher heat of evaporation. As a result of these conflicting factors, the emitted NOx emission of EW50–Diesel DI fuel blends is lower than that reference Neat Diesel fuel.

4.2.4. Effect of EW75–Diesel DI on Nitrogen Oxides Emissions

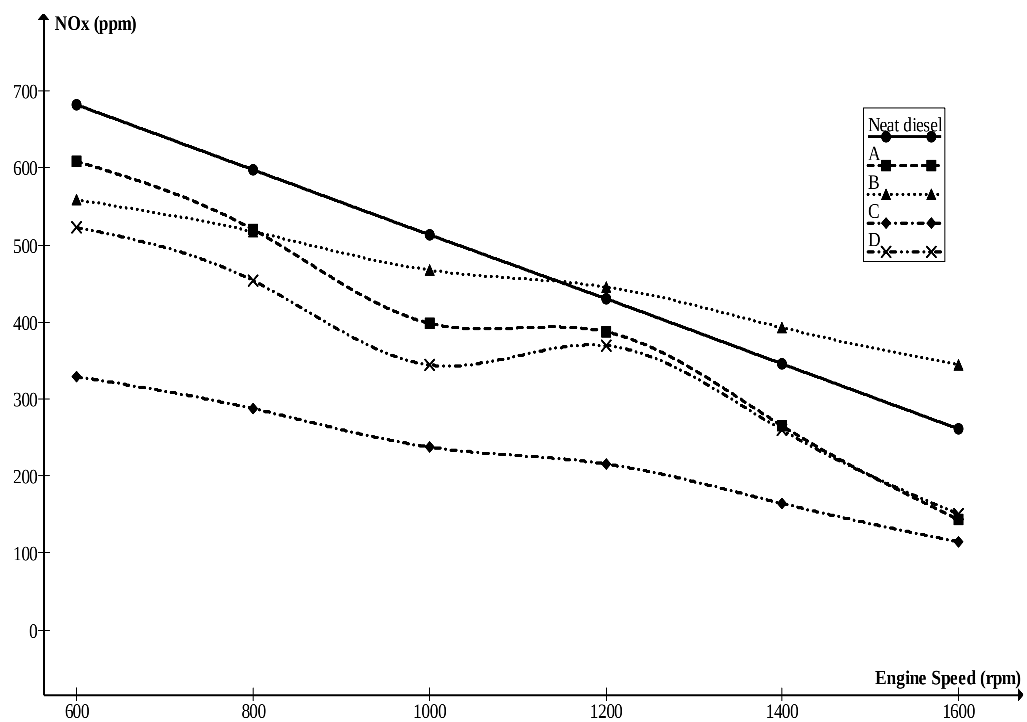

Figure 4.4: Wide Range Ratio EW75–Diesel DI Effect on NOx Emissions vs. Engine Speed

Figure (4.4) shows the wide range ratio EW75–Diesel DI effect on NOx emissions vs. engine speed. NOx emissions values that measured at full load condition in the case of EW75 blend at different flow rates. NOx emissions decrease with increase engine speed for Neat Diesel, using EW75–Diesel on case (A, C, and D). NOx emissions of the engine using the EW75–Diesel on case (B) showed contradicted tendency. Suggest that the local oxygen availability has the dominant effect. The higher values of the fuel-bound oxygen for the ethanol blends against the Neat Diesel fuel may be bringing locally the 'prepared' mixture nearer to stoichiometry (towards the lean) during the premixed combustion phase (when NOx is mainly formed), thus leading to the relative increase of NOx formation.

In any case, the leanness and the combustion temperature of the mixture on a local basis form a delicate balance on NOx formation, weighting more or less on the one or the other side, depending on the type of blends, and the specific engine calibration and operating conditions. The minimum NOx is 115 ppm at 1600 rpm in case (C) condition. According to Neat Diesel NOx emission values, the highest change is 56% at 1600 rpm in case (C) condition, the lowest change is -31.8% at 1600 rpm in case (B) condition. However, NOx emissions in case (C) are the minimum values. The decrease in NOx with the use of EW75–Diesel DI case (C) is attained through their lower combustion temperatures, which may be caused by the shorter ignition delay for EW75–Diesel DI case (C) in comparison to diesel. Furthermore, aromatic content and density exhibit a good correlation with NOx emissions, so lower NOx emissions are also owing to lower aromatics content and lower density for EW75–Diesel DI case (C) fuels.

4.2.5. Effect of Ethanol–Diesel DI on Nitrogen Oxides Emissions

Figure (4.5) shows the wide range ratio of Ethanol–Diesel DI effect on NOx emissions vs. engine speed. NOx emissions values were measured at full load condition in the case of ethanol DI at different flow rates. NOx emissions decrease with increase engine speed for Neat Diesel, using ethanol DI on case (A, B, C and D). The effect of oxygenated fuel blends on NOx emissions is complex and is not conclusive. However, cetane number, fuel density or aromatic fuel composition can influence on NOx emissions. The minimum NOx is 157 ppm at 1600 rpm in case (A) condition. According to Neat Diesel NOx emission values, the highest change is

39.9% at 1600 rpm in case (A) condition, the lowest change is -122.4% at 600 rpm in case (D) condition. However, NOx emissions in case Neat Diesel are the minimum values, when the engine speed increased from 600 to 1200 rpm.

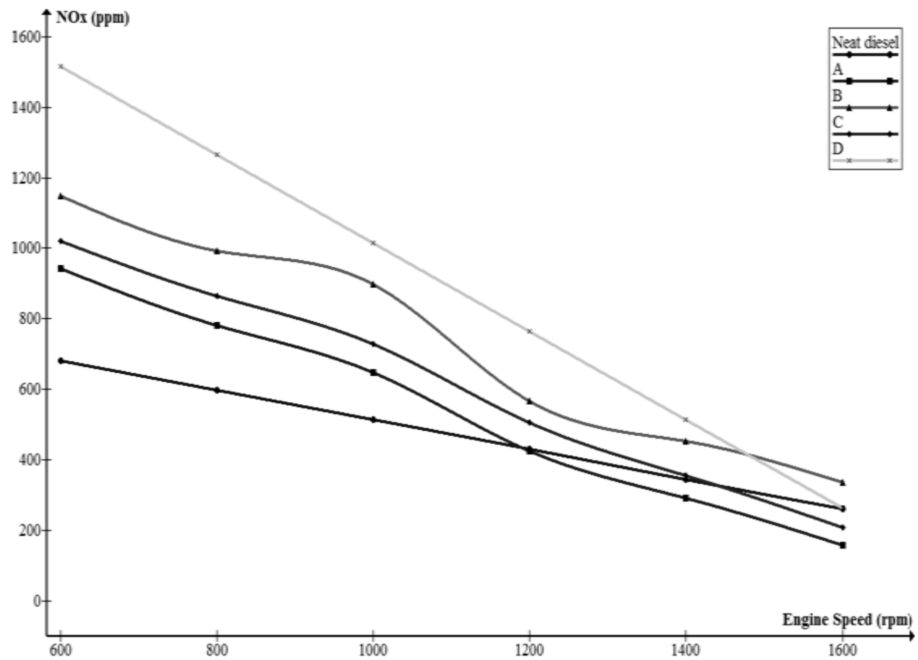

Figure 4.5: Wide Range Ratio Ethanol–Diesel DI Effect on NOx Emissions vs. Engine Speed

Our results also demonstrated a few percent increase in NOx emissions at most operation conditions when the diesel engine was fueled with ethanol. When the engine speed increased from 1200 to 1600 rpm the minimums values of NOx in case (A). NOx emissions was increased when a diesel engine was fueled with ethanol, and this is attributed it to the higher in-cylinder gas temperature by using the oxygenated fuels in comparison with that of Neat Diesel. It is known that NOx formation is strongly related to the in cylinder gas temperature as well as the oxygen content in the fuel, which are influenced by many parameters, such as fuel structure and injection timing. The lower cetane number of the blended fuel could result in longer ignition delay and hence more complete mixing of fuel vapors with air before ignition occurs, which will lead to higher NOx emissions. However, the larger amount of fuel to be evaporated could reduce the in cylinder gas temperature to some extent. These conflicting effects on NOx emissions lead to the results obtained in this study.

4.2.6. Effect of Different Dual-Fuelling On Nitrogen Oxides Emissions at Same Operation Conditions

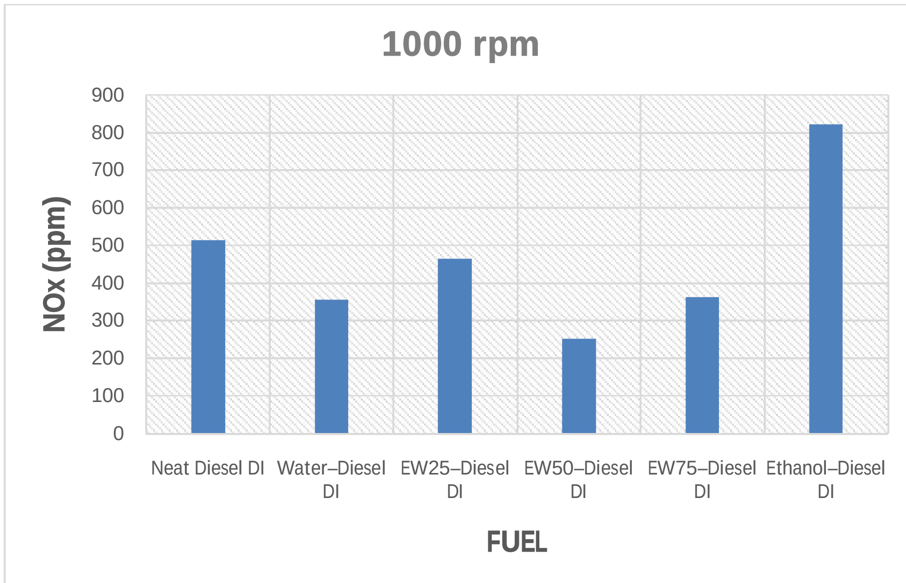

Figure 4.6: NOx Emissions vs. Different Dual-Fuelling For Constant Operation Conditions

The literature shows that the main mechanisms of NOx formation are: thermal NOx (or Zel´dovich. Considering the burn in diesel engines, it can be shown that NOx formation comes from the thermal mechanisms. **Figure (4.6)** shows the NOx emissions vs. different dual-fuelling for constant operation conditions. Fixed speed 1000 rpm and average results NOx from different flow rates, consider different results NOx emissions for Ethanol/Water–Diesel DI. Water–Diesel DI produced 30.9% lower NOx emissions compared to Neat Diesel, the reason for reduction using water decreased of flame temperatures in the combustion chamber.

EW25–Diesel DI produced 9% lower NOx emissions compared to Neat Diesel, because using ethanol in blends it has oxygen content. EW50–Diesel DI produced 51% lower NOx emissions compared to Neat Diesel, because it the optimum Ethanol/Water blends. EW75–Diesel DI produced 30% lower NOx emissions compared to Neat Diesel, due to the latent heat of vaporization of ethanol which decreases peak temperature in the cylinder. Ethanol–Diesel DI produced 60% higher NOx emissions compared to Neat Diesel, except in the misfiring conditions, the NOx emission increases due to increased combustion temperature via thermal Zel'dovich mechanism.

The most successful method of reducing nitrogen oxides emissions is by lowering the peak cylinder temperature through retarded injection timing (this may, however, affect engine efficiency) or, more successfully, applying Ethanol/Water–Diesel DI. The latter method has been rendered very popular in recent years as an efficient means for reducing the emitted NOx from compression ignition engines on account of the imposed, increasingly stringent, emission regulations. Introduction of cooled (exhaust) gas into the combustion chamber results in dilution of the air-charge by replacing O_2 with the non-reacting CO_2 and H_2O. Consequently, the in cylinder gas mixture and the gas temperatures of the cycle are reduced. As a result, NOx emissions are reduced too, aided by the lower oxygen availability since ethanol in a diesel engine replaces oxygen, and so promotes as light enrichment of the mixture.

4.2.7. Effect of Water–Diesel DI on Carbon Monoxide Emissions

Figure (4.7) shows the wide range ratio of Water–Diesel DI effect on CO emissions vs. engine speed. CO emissions values were measured at full load condition in the case of Water at different flow rates. CO emissions decreases with increase engine speed for tests Neat Diesel and with Water–Diesel DI in case (A). CO increases with increase in engine speed for the tests Water–Diesel DI in case (B, C, and D). Micro explosion of water droplets decreases the combustion zone temperature by adding the effect of oxygen for better combustion. It is in accord with the previous research which states that the latent heat of water will cool the charge due to the evaporation of water and also the cylinder average temperature following injection and before ignition. The effect of adding water to the fuel would increase the oxygen availability in the fuel and result in lower CO emissions, i.e.

case (A). Injected large amount of micro explosions leads to a large of mixing of unburned mixture. Less conversion of unburned fuel case (B) to CO emission is resulted. After speed 900 rpm in case (B, C, and D) CO emissions more than Neat Diesel due to the daily time ignition. The minimum CO emission is 0.05% at 1200 rpm in case (A) condition. According to Neat Diesel CO emission values, the highest change is 81.9% at 1600 rpm in case (A) condition, the lowest change is -170.9% at 1600 rpm in case (D) condition. However, CO emissions in case (A) are the minimum values. This happens because with increase in water the temperature inside the cylinder decreases slowing down the combustion of carbon, as a result of which incomplete combustion occurs.

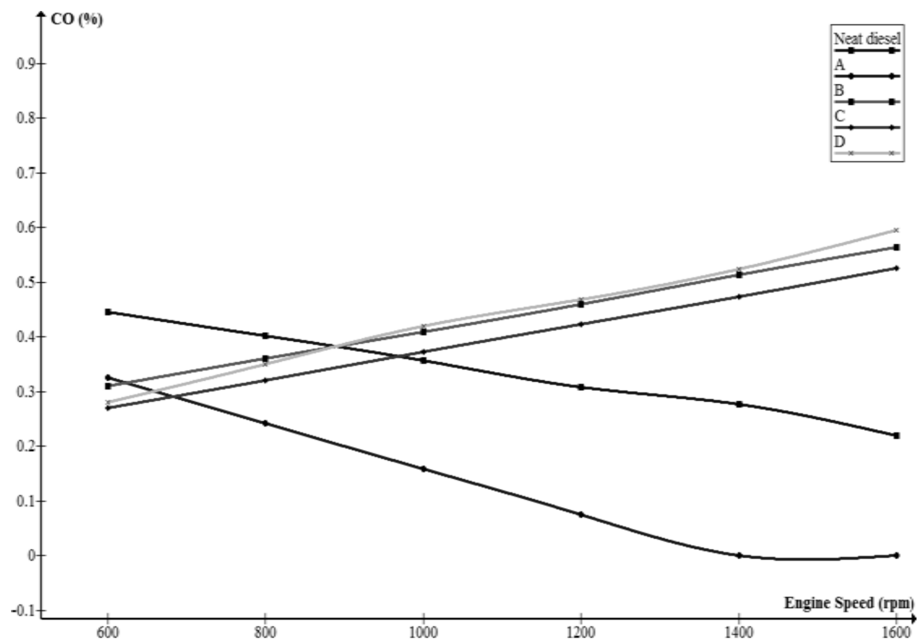

Figure 4.7: Wide Range Ratio Water–Diesel DI Effect on CO Emissions vs. Engine Speed

4.2.8. Effect of EW25–Diesel DI on Carbon Monoxide Emissions

Figure (4.8) shows the wide range ratio EW25–Diesel DI effect on CO Emissions vs. engine speed. CO emissions values were measured at full load condition in the case of EW25 DI at different flow rates. CO emissions decreases with increase engine speed for tests Neat Diesel and with EW25–Diesel DI in case (A, B). CO increases with increase in engine speed for the tests EW25–Diesel DI in case (C, D). At low speeds the high latent heat of vaporization of ethanol causes a temperature reduction in the cylinder, which prevents the oxidation of CO, resulting in increased CO emissions as the EW25–Diesel DI is increased, in case (A, B) and decreased in case (C, D), suggest the resulting pressure injection pump.

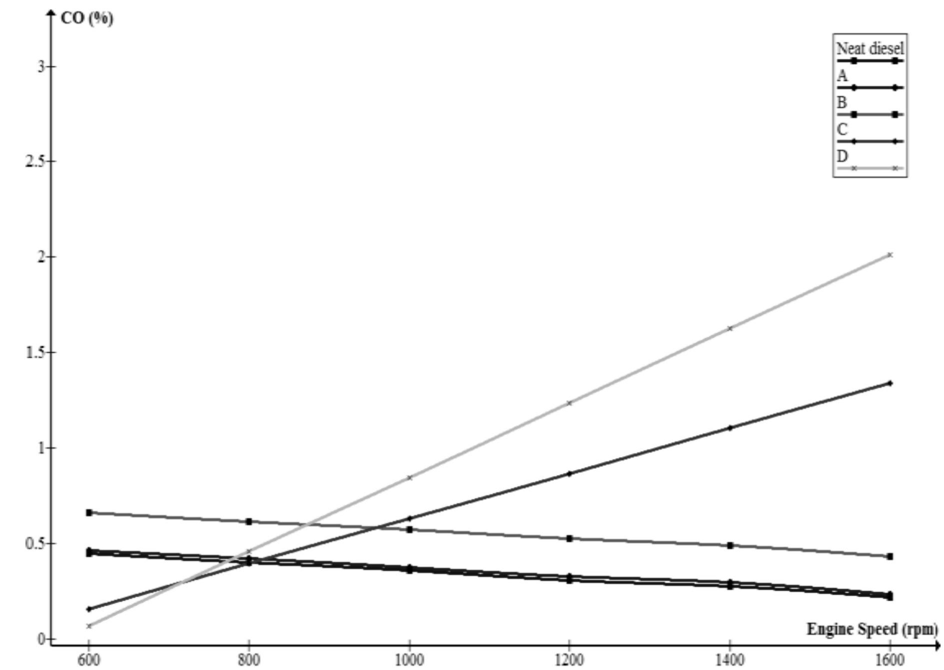

Figure 4.8: Wide Range Ratio EW25–Diesel DI Effect on CO Emissions vs. Engine Speed

The minimum CO emission is 0.07% at 600 rpm in case (D) condition. According to Neat Diesel CO emission values, the highest change is 85% at 600 rpm in case (D) condition, the lowest change is -816% at 1600 rpm in case (D) condition. The increase of CO emission with EW25–Diesel DI may be due to lower heating value of ethanol that leads to a lower in cylinder temperature, causing incompleteness in combustion.

4.2.9. Effect of EW50–Diesel DI on Carbon Monoxide Emissions

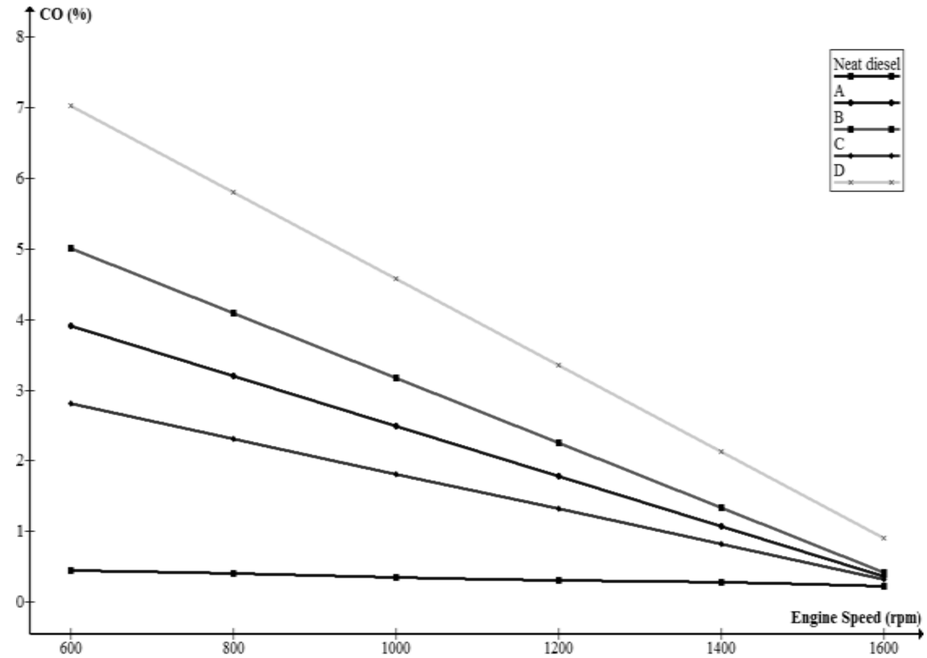

Figure 4.9: Wide Range Ratio EW50–Diesel DI Effect on CO Emissions vs. Engine Speed

Figure (4.9) shows the wide range ratio EW50–Diesel DI effect on CO emissions vs. engine speed. CO emissions values that measured at full load condition in the case of EW50 at different flow rates. CO emissions decreases with increase engine speed for tests Neat Diesel and with EW50–Diesel DI. CO emissions are the result of improper mixing and incomplete combustion and are controlled primarily by the global or local air/fuel equivalence ratio. With lower flame temperature, a thickened quench layer is formed and a larger fraction of ethanol may be found in a rich air/fuel ratio range or even liquid state in the quench layer, as the flame front spreads too slowly to reach them. Another reason for the increase in CO emission is the increase in ignition delay. Also amount of water in mixing increased in CO emission. This results in combustion of a proportion of the fuel in the expansion stroke, which lowers the gas temperature and reduces the CO oxidation reaction rate, resulting in incomplete combustion and causing relatively high CO emissions. The minimum CO emission is 0.33% at 1600 rpm in case (A) condition. According to Neat Diesel CO emission values, the highest change is -49% at 1600 rpm in case (C) condition, the lowest change is -1477% at 600 rpm in case (D) condition. The increase of the CO emissions may be explained by the fact that the secondary injection is expected from the higher residual pressure and oscillations encountered in the fuel system when using EW50–Diesel DI.

4.2.10. Effect of EW75–Diesel DI on Carbon Monoxide Emissions

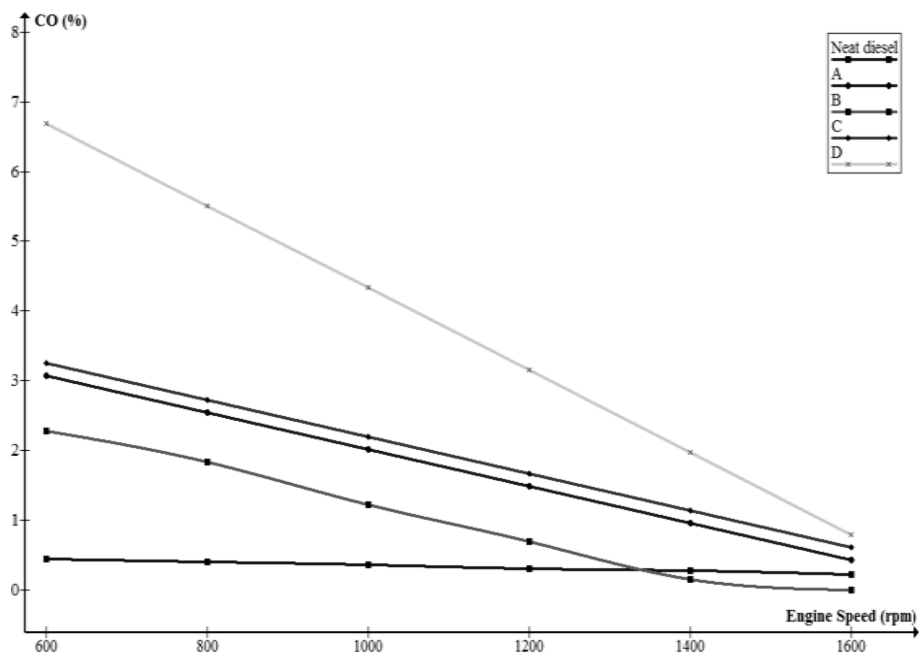

Figure 4.10: Wide Range Ratio EW75–Diesel DI Effect on CO Emissions vs. Engine Speed

Figure (4.10) shows the wide range ratio EW75–Diesel DI effect on CO emissions vs. engine speed. CO emissions values were measured at full load condition in the case of EW75 at different flow rates. CO emissions decreases with increase engine speed for tests Neat Diesel and with EW75–Diesel DI. The reason is that, when operated with a fuel-rich equivalence ratio at high engine speed, there is not enough oxygen to convert the entire carbon to carbon dioxide; as a result, some fuel does not get fully burned, and some carbon ends up emitted as CO. At low engine speed, over-lean mixture areas, low in-cylinder temperatures, and bad atomization conditions influenced by the high viscosity of EW75 at low temperatures can lead to higher CO emissions when using EW75–Diesel DI fuel. Under high engine speed, the molecular oxygen in ethanol fuel improves the combustion for local rich mixtures, and the low cetane number of ethanol fuel leads to less fuel-rich zone formation; consequently, the CO emission decreases. The minimum CO emission is 0.05% at 1600 rpm in case (B) condition. According to Neat Diesel CO emission values, the highest change is 77.25% at 1600 rpm in case (B) condition, the lowest change is -1400% at 600 rpm in case (D) condition. Because of to the ethanol's oxygen content, hence lower energy content and higher heat of evaporation, when an engine that is tuned for diesel combustion operates with EW75–Diesel, lower exhaust gas temperatures are experienced.

4.2.11. Effect of Ethanol–Diesel DI on Carbon Monoxide Emissions

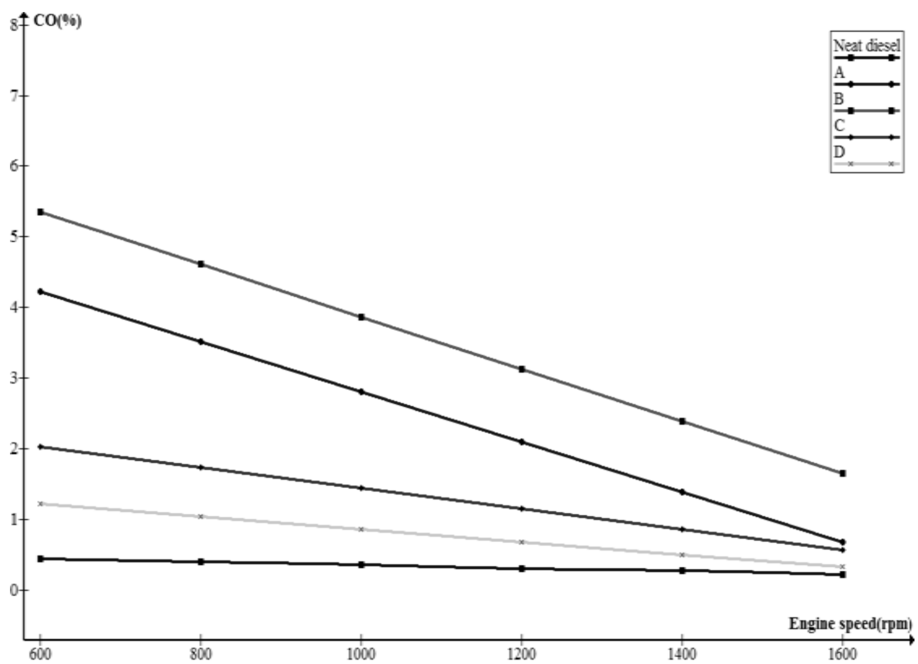

Figure 4.11: Wide Range Ratio Ethanol–Diesel DI Effect on CO Emissions vs. Engine Speed

Figure (4.11) shows the wide range ratio Ethanol–Diesel DI effect on CO emissions vs. engine speed. CO emissions values that measured at full load condition in the case of Ethanol at different flow rates. CO emissions decreases with increase engine speed for all tests. This fact can be collectively attributed to the same physical and chemical mechanisms affecting almost in the same way, at least qualitatively, the net formation of this is pollutants. More precisely, it is the excess oxygen in the fuel blend that aids the in-cylinder oxidation of CO to CO_2 that is most probably responsible for this behavior. The minimum CO emission is 0.33% at 1600 rpm in case (D) condition. According to Neat Diesel CO emission values, the highest change is -48.2% at 1600 rpm in case (D) condition, the lowest change is -1099% at 600 rpm in case (B) condition. This phenomenon or trend is due to that ethanol contains oxygen element in it.

4.2.12. Effect of Different Dual-Fuelling on Carbon Monoxide Emissions at Same Operation Conditions

CO could increase or decrease depending on the engine type and operating conditions. Generally CO increases with increasing substitution rate for all ethanol/ water mixes. Any air/gas mixture that is not entrained into the burning diesel fuel spray will remain un-reacted or partially reacted, unless the gas–air mixture strength is sufficient to support high temperature combustion in its own right.

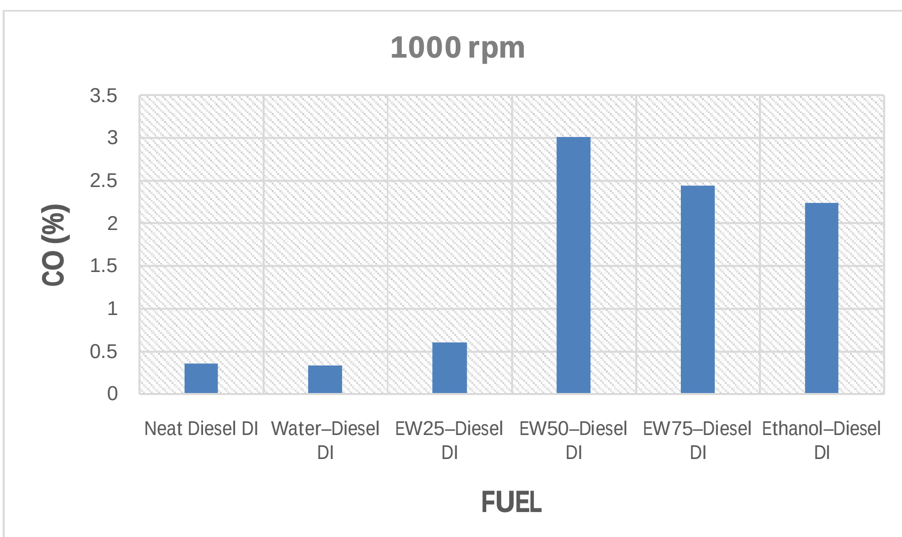

Figure 4.12: CO Emissions vs. Different Dual-Fuelling For Constant Operation Conditions

Figure (4.12) fixed speed 1000 rpm and average results CO from different flow rates, consider different results CO emissions for Ethanol/Water–Diesel DI. Water–Diesel DI produced 5% lower CO emissions compared to Neat Diesel, the reason for reduction using micro explosion of water droplets decreases the combustion zone temperature by adding the effect of oxygen for better combustion. It is in accord with the previous research which states that the latent heat of water will cool the charge due to the evaporation of water and also the cylinder average temperature following injection and before ignition. EW25–Diesel DI produced 70% higher CO emissions compared to Neat Diesel, generally incomplete combustion leads to higher CO levels. This process occurs in conditions that are locally rich, have insufficient oxidizers, or have low combustion temperatures. EW50–Diesel DI produced 744% higher CO emissions compared to Neat Diesel, result can be explained by the fact that the oxidation of CO is active due to the high combustion temperature, which is caused by the large injection quantity of ethanol. It is believed that larger injection quantities may lead to rich running conditions in the combustion chamber. EW75–Diesel DI produced 584% higher CO emissions compared to Neat Diesel, due to insufficient oxygen and low temperature, CO emissions increased. Ethanol–Diesel DI produced 528% higher CO emissions compared to Neat Diesel, while the ethanol blending supplied more molecular oxygen to the combustion chamber, the high heat of evaporation of ethanol fuel leads to a lower temperature in the combustion cylinder. The lowered combustion temperature inhibited the oxidation of CO to CO_2. CO emissions also exhibited an increasing trend with a retardation of the injection timing.

4.2.13. Effect of Water–Diesel DI on Unburnt Hydrocarbon Emissions

Figure (4.13) shows the wide range ratio Water–Diesel DI effect on UBHC emissions vs. engine speed. UBHC emissions values were measured at full load condition in the case of Water at different flow rates. UBHC emissions increase with increase engine speed for tests Neat Diesel and with Water–Diesel DI. When the engine speed is relatively high, there is higher peak pressure inside the combustion chamber. As the engine speed increases the fuel consumption rate also increases thereby the temperature increases, as a result higher amount of UBHC. The minimum UBHC emission is 30 ppm at 600 rpm in case (D) condition. According to Neat Diesel UBHC emission values, the highest change is 37% at 600 rpm in case (D) condition, the lowest change is -65% at 800 rpm in case (D) condition. From **Figure (4.13)** it has been seen that Neat Diesel has higher UBHC emissions than only case (A, D). At this case during the micro explosion there was grater physical injection of water due to their volatility difference, further enhanced better fuel–air mixing, leading to reduction of UBHC.

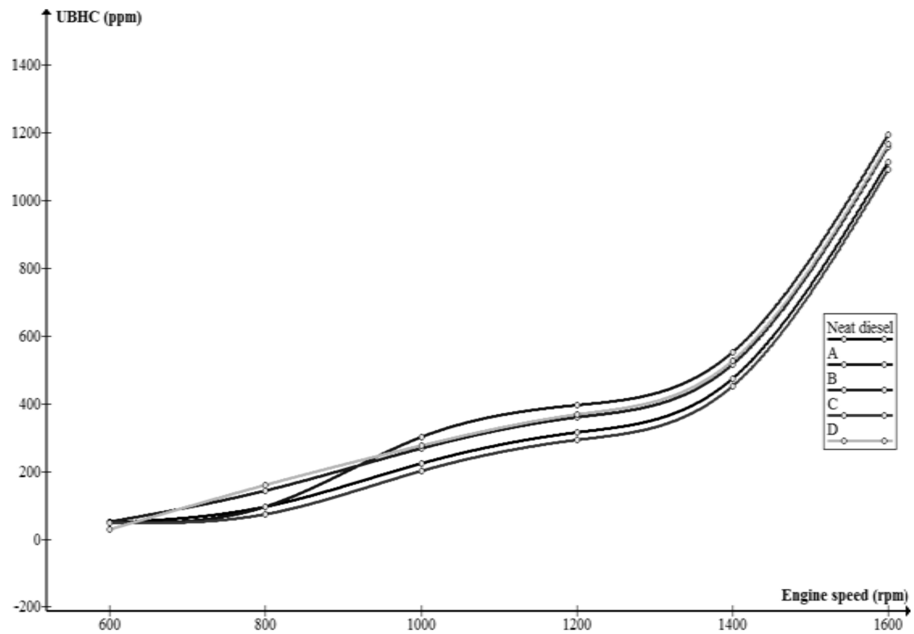

Figure 4.13: Wide Range Ratio Water–Diesel DI Effect on UBHC Emissions vs. Engine Speed

4.2.14. Effect of EW25–Diesel DI on Unburnt Hydrocarbon Emissions

Figure (4.14) shows the wide range ratio EW25–Diesel DI effect on UBHC emissions vs. engine speed. UBHC emissions values were measured at full load condition in the case of EW25 at different flow rates. UBHC emissions increase with increase engine speed for tests Neat Diesel and with EW25–Diesel DI. This effect was thought to be due to the low cetane number fuel having a long ignition delay causing a high maximum heat release rate and shortened combustion duration.

However for high speed, lower cetane fuels produced higher UBHC due to local over lean mixtures caused by the ignition delay, hence resulting in an incomplete combustion. The minimum UBHC emission is 48 ppm at 600 rpm in case Neat Diesel condition. According to Neat Diesel UBHC emission values, the highest change is 2% at 800 rpm in case (C) condition, the lowest change is -111% at 600 rpm in case (A) condition. The significantly lower UBHC emissions with the case Neat Diesel is likely to be a result of the shorter ignition delays and the better vaporization characteristics, as the energy requirement for complete vaporization of the fuel is lower.

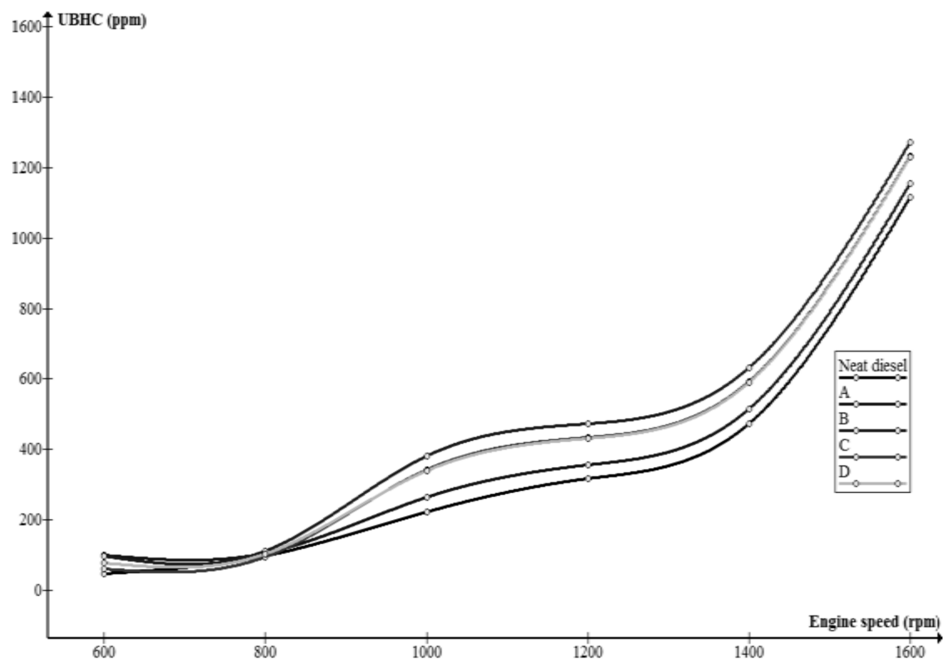

Figure 4.14: Wide Range Ratio EW25–Diesel DI Effect on UBHC Emissions vs. Engine Speed

4.2.15. Effect of EW50–Diesel DI on Unburnt Hydrocarbon Emissions

Figure (4.15) shows the wide range ratio EW50–Diesel DI effect on UBHC emissions vs. engine speed. UBHC emissions values were measured at full load condition in the case of EW50 at different flow rates. UBHC emissions increase with increase engine speed for tests Neat Diesel. The UBHC emissions contain partially or completely unburned fuel produced in locations where combustion takes place under fuel-rich conditions, due to incomplete air-fuel mixing. UBHC increases with increase engine speed to 1000 rpm after that decreases to full engine speed for case EW50–Diesel DI. The UBHC reduction may be related to the lower cetane number for ethanol than Neat Diesel. A lower UBHC of EW50–Diesel than that of Neat Diesel indicates that EW50 has less heavy distillation, which suggests that EW50 is easy to evaporate and to mix with air, forming a more combustible charge. The minimum UBHC emission is 0 ppm at 600 rpm in case (C) condition. According to Neat Diesel UBHC emission values, the highest change is 100% at 600 rpm in case (C) condition, the lowest change is -141% at 600 rpm in case (D) condition. Therefore, reductions in UBHC emission are partially attributed to the lower (D) of EW50–Diesel. Because ethanol has a short ignition delay period, the over-rich and over-lean mixture regions formed during the ignition delay period might be smaller, resulting in significantly reduced UBHC emissions. Ethanol, oxygenated fuel containing 34.78 % oxygen by mass, has good mixing characteristics, and is a superheated vapor after entering the engine cylinder. The volume of fuel-rich regions existing during the combustion period could be less, resulting in reduced UBHC emissions.

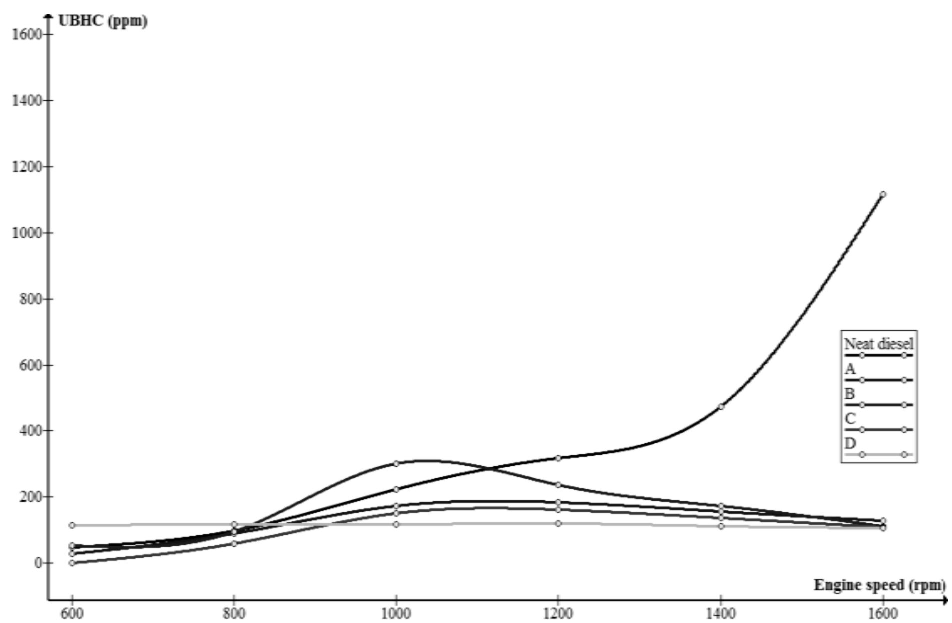

Figure 4.15: Wide Range Ratio EW50–Diesel DI Effect on UBHC Emissions vs. Engine Speed

4.2.16. Effect of EW75–Diesel DI on Unburnt Hydrocarbon Emissions

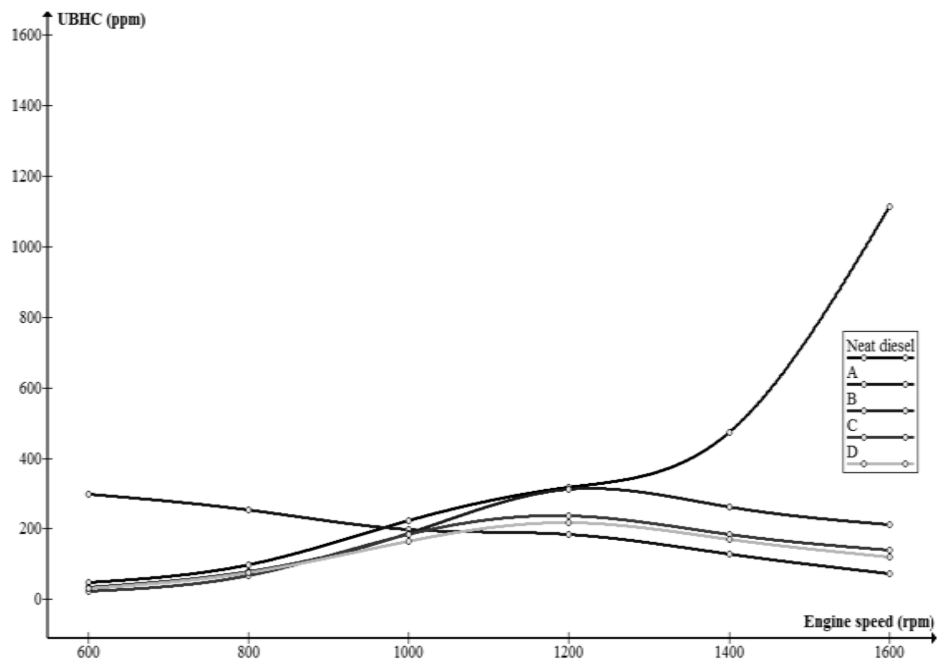

Figure 4.16: Wide Range Ratio EW75–Diesel DI Effect on UBHC Emissions vs. Engine Speed

Figure (4.16) shows the wide range ratio EW75–Diesel DI effect on UBHC emissions vs. engine speed. UBHC emissions values were measured at full load condition in the case of EW75 at different flow rates. UBHC emissions increase with increase engine speed for tests Neat Diesel. The oxygen content in the A/F diesel leads at low speeds to a more complete and cleaner combustion. UBHC increases with increase engine speed to 1200 rpm after that decreases to full engine speed for EW75–Diesel DI. Although EW75 is less volatile than diesel fuel, higher final distillation points have been reported for diesel fuel. This final fraction of the diesel may not be completely vaporized and burnt, thereby increasing UBHC emissions. The minimum UBHC emission is 24 ppm at 600 rpm in case (C) condition. According to Neat Diesel UBHC emission values, the highest change is 94% at 1600 rpm in case (A) condition, the lowest change is -521% at 600 rpm in case (A) condition. At higher engine speeds, the amount of fuel injection is small.

Lean fuel-air mixture regions in cylinders may be more prevalent because of poor fuel distribution, large amounts of excess air, and low cylinder temperatures. Because of this, UBHC emissions are high at high engine speeds.

4.2.17. Effect of Ethanol–Diesel DI on Unburnt Hydrocarbon Emissions

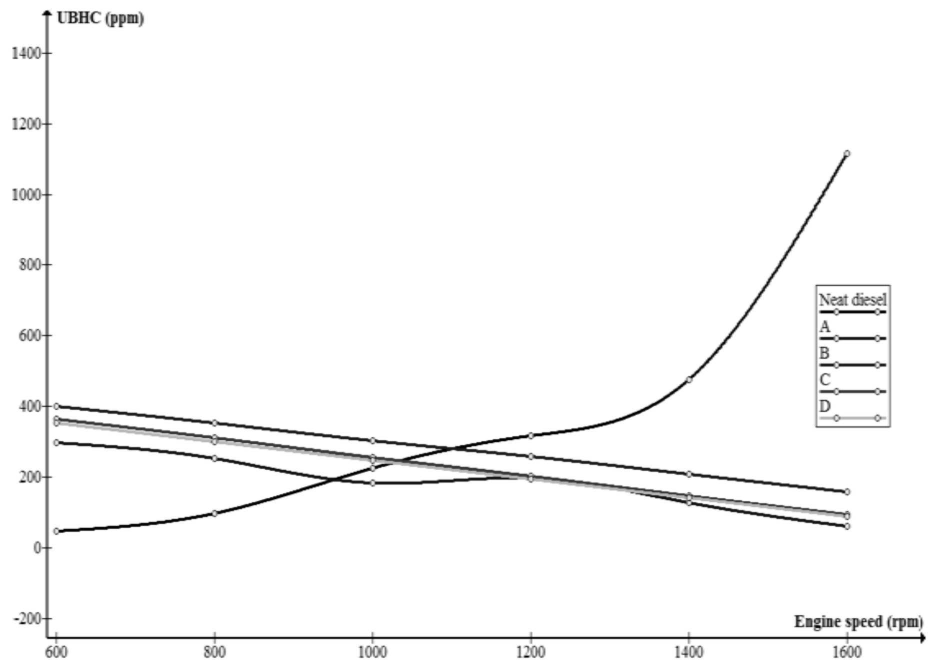

Figure 4.17: Wide Range Ratio Ethanol–Diesel DI Effect on UBHC Emissions vs. Engine Speed

Figure (4.17) shows the wide range ratio Ethanol–Diesel DI effect on UBHC emissions vs. engine speed. UBHC emissions values that measured at full load condition in the case of Ethanol at different flow rates. UBHC emissions increase with increase engine speed for tests Neat Diesel. It can be underlined that the lower heating power of the Neat Diesel implies higher fuel consumptions and therefore it can produce high local fuel-to-air ratios that can cause an increase in UBHC emissions. UBHC decreases with increase engine speed for case Ethanol–Diesel DI. The UBHC by diesel engines are complex mixtures of unburned and partially burned hydrocarbon fuel components partitioned in the gaseous and liquid phases. At lower engine speeds, due to the higher UBHC emissions caused by higher values of the excess air-to-fuel ratio or lambda. The minimum UBHC emission is 48 ppm at 600 rpm in case Neat Diesel condition. According to Neat Diesel UBHC emission values, the highest change is 95% at 1600 rpm in case (A) condition, the lowest change is -738% at 600 rpm in case (B) condition. A general reduction in UBHC was obtained under the operation conditions. Combined results on Ethanol–Diesel suggest that the use of Neat Diesel leads to a slight increase of ethanol and pressure injection.

4.2.18. Effect of Different Dual-Fuelling on Unburnt Hydrocarbon Emissions at Same Operation Conditions

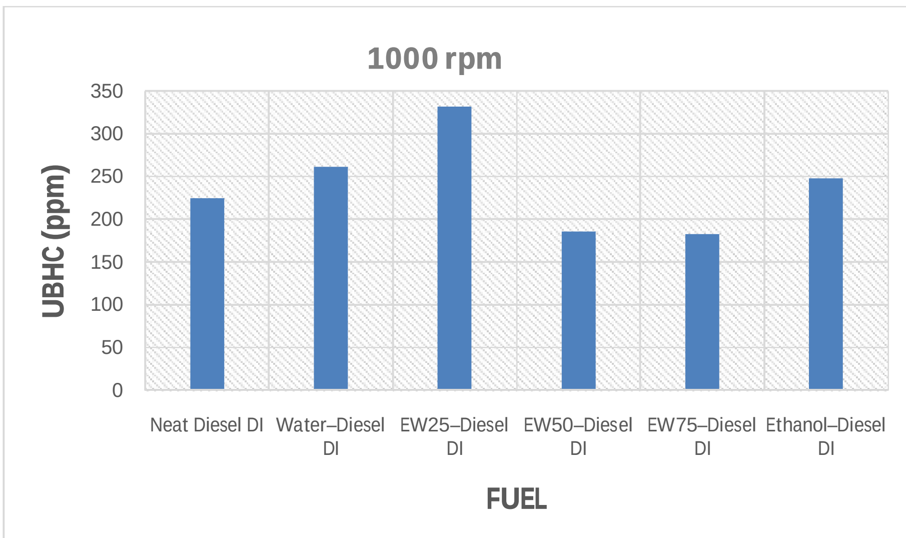

Figure 4.18: UBHC Emissions vs. Different Dual-Fuelling For Constant Operation Conditions

UBHC emissions could increase or decrease depending on the engine type and operating conditions. Generally, it is known that UBHC emissions decrease obviously with the reduction of fuel aromatic content. With fuel aromatic content decreasing, the ignition delay period of the engine shortens, and unburned hydrocarbons in the ignition delay period descend, which helps reducing UBHC emissions. **Figure (4.18)** shows the UBHC emissions vs. different dual-fuelling for constant operation conditions. Fixed speed 1000 rpm and average results UBHC emission from different flow rates, consider different results UBHC emissions for Ethanol/Water–Diesel DI. Water–Diesel DI produced 17% higher UBHC emissions compared to Neat Diesel, the reason for increases may be addition, and the water increased the ignition delay and rate of pressure rise.

EW25–Diesel DI produced 48% higher UBHC emissions compared to Neat Diesel, generally The UBHC results can be explained as both a higher latent heat value and a lower cetane number of EW25–Diesel, resulting in a slower evaporation, and leading to a higher fraction of UBHC surviving in the engine exhaust. EW50–Diesel DI produced 17% lower UBHC emissions compared to Neat Diesel, when the additive and ignition improver are used, UBHC emissions of EW50–Diesel reduce to quite a low level and even are less than of Neat Diesel in most cases. Homogeneity of EW50–Diesel and increased oxidation of combustion intermediates by the use of additive and ignition improver would be expected to be the main reasons for UBHC reduction in this engine control system. EW75–Diesel DI produced 18% lower UBHC emissions compared to Neat Diesel, due to the higher oxygen content of the blended fuels. For EW75–Diesel, the higher oxygen content of the blended fuels could improve the combustion process while the lower viscosity and density of the blended fuels could lead to better air-fuel mixing, resulting in lower UBHC emissions. Ethanol–Diesel DI produced 10% higher UBHC emissions compared to Neat Diesel, the formation of unburned hydrocarbons originates from various sources in the engine cylinder and its theoretical study is still difficult. These sources, in the present case, for the increased UBHC emissions with ethanol addition may be due to the higher heat of evaporation of the ethanol causing slower evaporation and so slower and poorer fuel-air mixing, the increased spray life causing unwanted fuel impingement on the combustion chamber walls and so flame. Mainly the increase of the so called 'lean outer flame zone' where flame is unable to exist. The latter one refers to the envelop of the spray boundary where the fuel has already mixed beyond the lean flammability limit during the ignition delay period and, thus, will not be able to auto-ignite or sustain a fast reaction front.

4.2.19. Effect of Water–Diesel DI on Carbon Dioxide Emissions

Figure (4.19) shows the wide range ratio Water–Diesel DI effect on CO_2emissions vs. engine speed. CO_2 emissions values that measured at full load condition in the case of Water at different flow rates. CO_2 emissions increase with increase engine speed for tests Neat Diesel and with Water–Diesel. The increase in engine speed causes a rise in fuel consumption rate, a drop in the equivalent ratio, and a rise in the fuel rich burning condition. All of these factors cause the CO_2 concentration to grow with the increase in engine speed. At time of burning of the water micro explosions occur, resulting better mixing occurs leading to more complete combustion. CO_2 production is directly proportional to fuel consumption. The maximum CO_2 emission is 9.82 (%) at 1600 rpm in case (D) condition. According to Neat Diesel CO_2 emission values, the highest change is 14% at 600 rpm in case (C) condition, the lowest change is -21% at 600 rpm in case (D) condition. In fact, the Water–Diesel may be used in Neat Diesel engines to reduce the most important greenhouse gas CO_2.

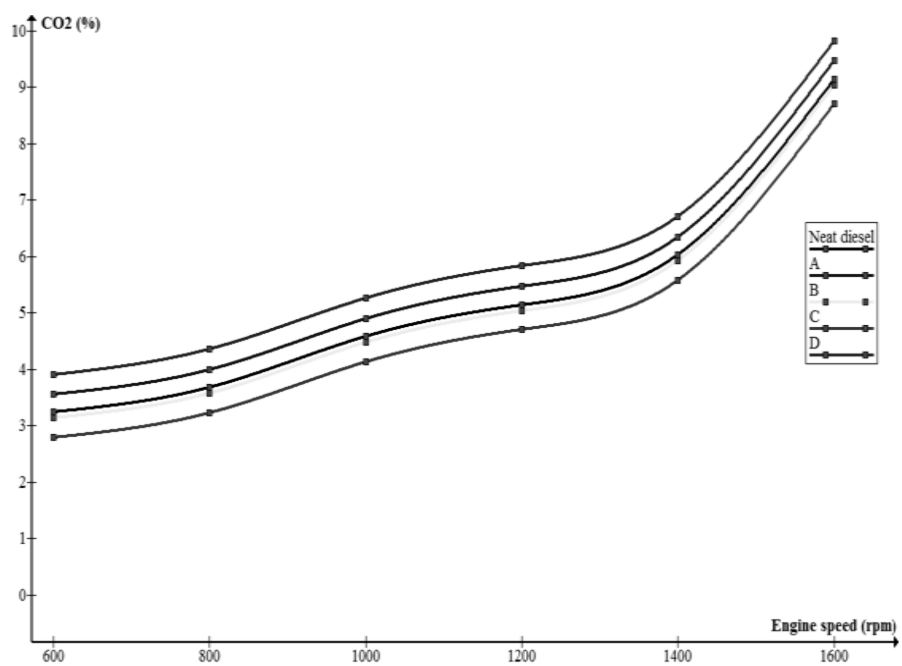

Figure 4.19: Wide Range Ratio Water–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

4.2.20. Effect of EW25–Diesel DI on Carbon Dioxide Emissions

Figure (4.20) shows the wide range ratio EW25–Diesel DI Effect on CO_2 emissions vs. engine speed. CO_2 emissions values were measured at full load condition in the case of EW25 at different flow rates. CO_2 emissions increase with increase engine speed for tests Neat Diesel and with EW25–Diesel. Ideally, combustion of a hydrocarbon fuel should produce only CO_2 and water (H_2O). Poor combustion characteristics of EW25–Diesel at all condition increased fuel consumption to get the same power and thus CO_2 emissions increased. The maximum CO_2 emission is 10 (%) at 1600 rpm in case (C) condition. According to Neat Diesel CO_2 emission values, the highest change is -2% at 1600 rpm in case (B) condition, the lowest change is -27% at 600 rpm in case (B) condition. However, increased CO_2 with respect to advanced injection for case (C) could have noticed with decrease in CO emissions.

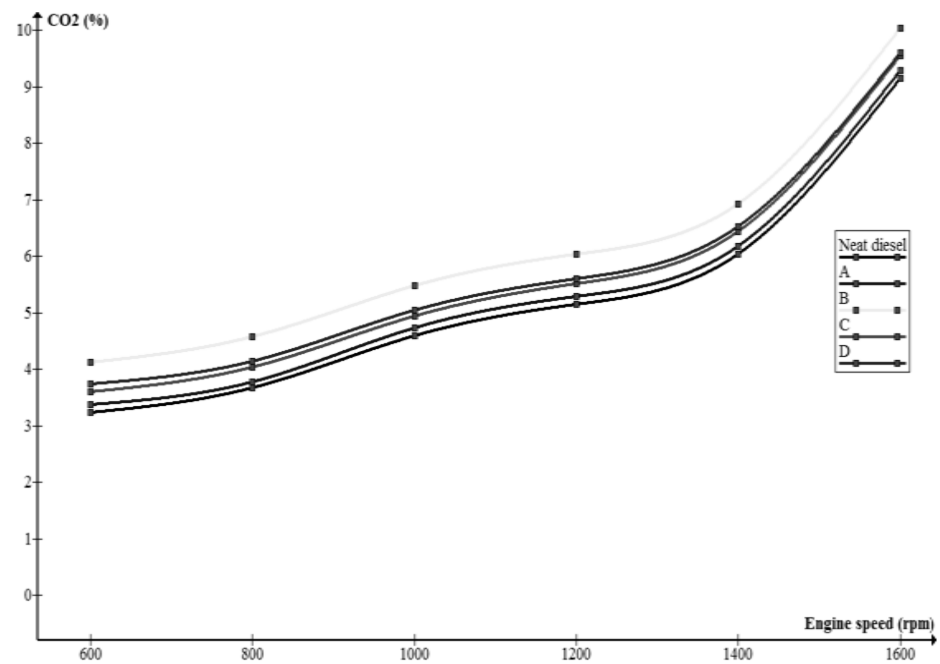

Figure 4.20: Wide Range Ratio EW25–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

4.2.21. Effect of EW50–Diesel DI on Carbon Dioxide Emissions

Figure (4.21) shows the wide range ratio EW50–Diesel DI effect on CO_2 emissions vs. engine speed. CO_2 emissions values were measured at full load condition in the case of EW50 at different flow rates.CO_2 emissions increase with increase engine speed for tests Neat Diesel and with EW50–Diesel DI. The CO_2 emissions generally reduce with EW50–Diesel fuelled diesel engine at all the engine speeds. The reason of reduction of EW50–Diesel fuelled diesel engine is that ethanol includes 34.78 % by mass oxygen in its chemical structure. The maximum CO_2 emission is 9.15 (%) at 1600 rpm in case Neat Diesel condition. According to Neat Diesel CO_2 emission values, the highest change is 73% at 600 rpm in case (C) condition, the lowest change is 4% at 1600 rpm in case (A) condition. The main reason of CO_2 reduction is low C/H ratio and high oxygen content of the blends.

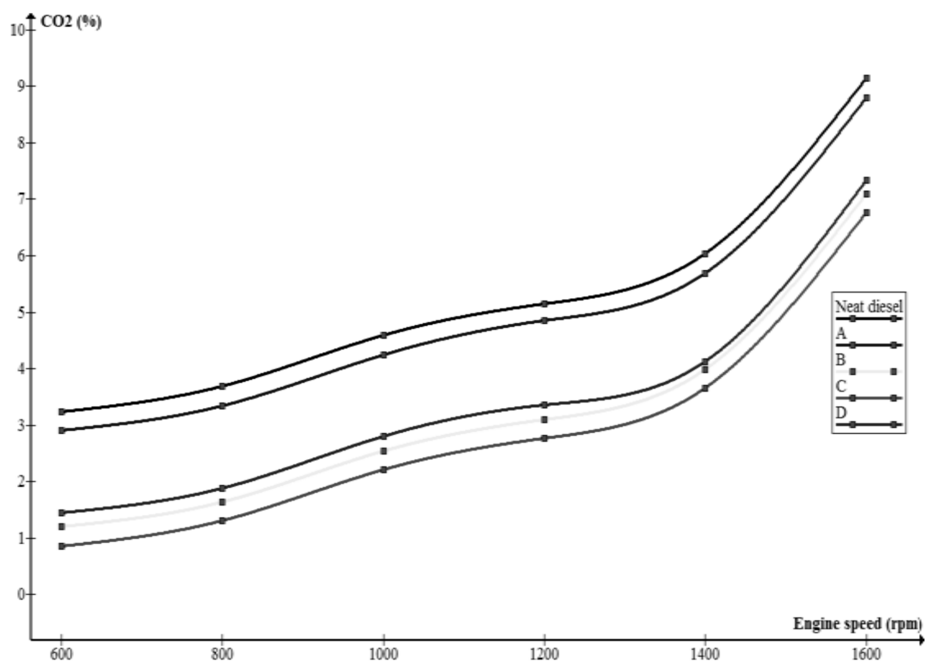

Figure 4.21: Wide Range Ratio EW50–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

4.2.22. Effect of EW75–Diesel DI on Carbon Dioxide Emissions

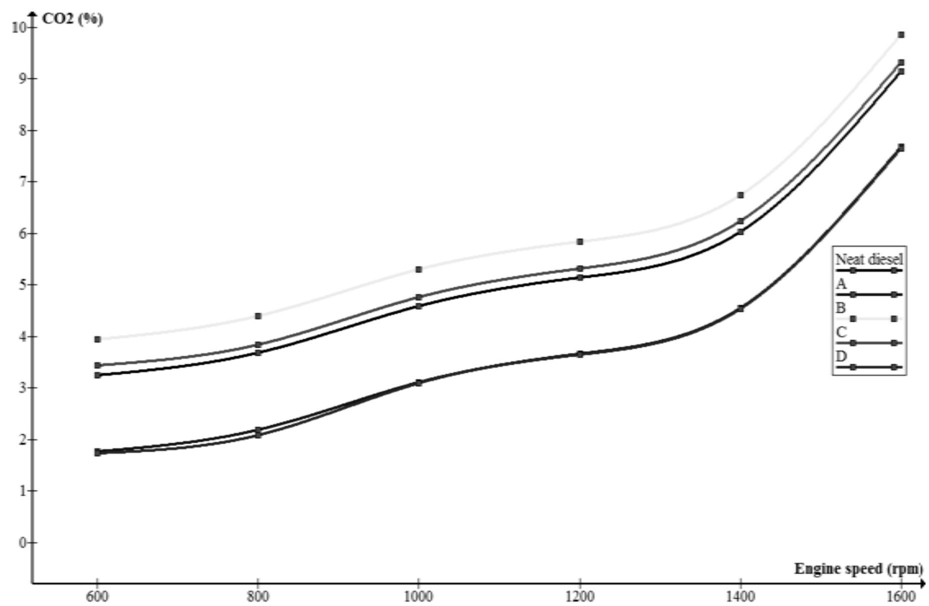

Figure 4.22: Wide Range Ratio EW75–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

Figure (4.22) shows the wide range ratio EW75–Diesel DI effect on CO_2 emissions vs. engine speed. CO_2 emissions values that measured at full load condition in the case of EW75 at different flow rates. CO_2 emissions increase with increase engine speed for tests Neat Diesel and with EW75–Diesel. Formation of CO during combustion process strongly depends on two things, combustion temperature and availability of oxygen. CO_2 percentage decreased as ethanol substitution was increased. Found case EW75–Diesel (A, D) as optimum level of injection. The maximum CO_2 emission is 9.85 (%) at 1600 rpm in case (D) condition. According to Neat Diesel CO_2 emission values, the highest change is 46% at 600 rpm in case (D) condition, the lowest change is -22% at 600 rpm in case (B) condition. The combustion process consists of two stages, at first stage, CO is formed and at second stage, if in cylinder temperature is sufficient to support the complete combustion and excess oxygen is available then CO reacts with additional oxygen to form CO_2.

4.2.23. Effect of Ethanol–Diesel DI on Carbon Dioxide Emissions

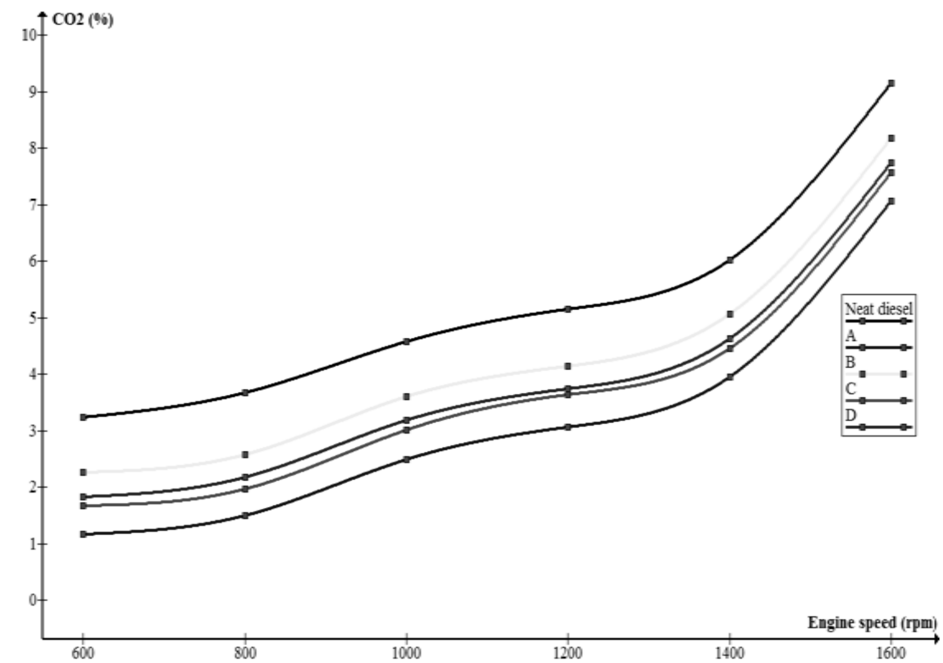

Figure 4.23: Wide Range Ratio Ethanol–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

Figure (4.23) shows the wide range ratio Ethanol–Diesel DI effect on CO_2 emissions vs. engine speed. CO_2 emissions values were measured at full load condition in the case of Ethanol at different flow rates. CO_2 emissions increase with increase engine speed for tests Neat Diesel and with Ethanol–Diesel. Also the observation shows lower CO_2 emission using Ethanol–Diesel compared to Neat Diesel fuel. CO_2 emission greatly depends on the CO emission. The maximum CO_2 emission is 9.15 (%) at 1600 rpm in case Neat Diesel condition. According to Neat Diesel CO_2 emission values, the highest change is 64% at 600 rpm in case (A) condition, the lowest change is 11% at 1600 rpm in case (B) condition. This increase in CO_2 is to be expected, as a decrease in CO emissions usually coincides with an increase in CO_2 emissions.

4.2.24. Effect of Different Dual-Fuelling On Carbon Dioxide Emission at Same Operation Conditions

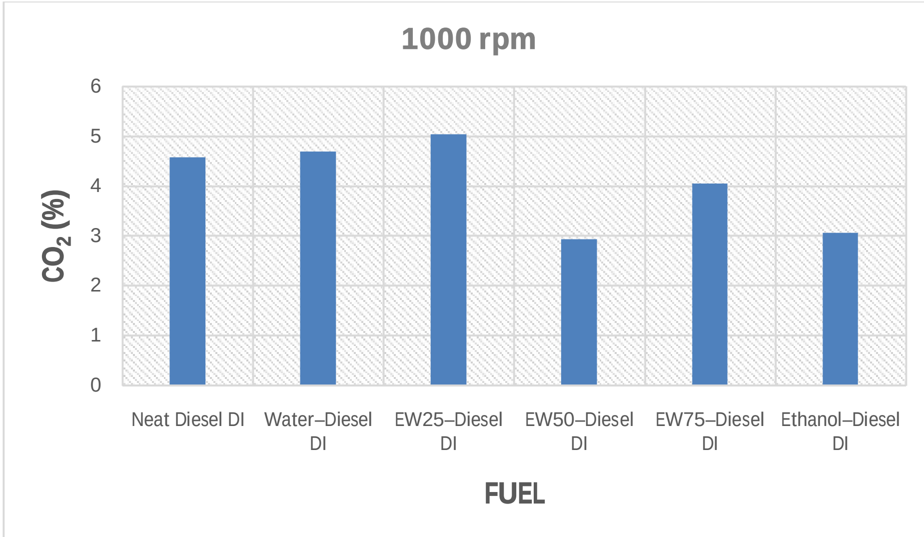

Figure 4.24: CO_2 Emissions vs. Different Dual-Fuelling For Constant Operation Conditions

Ideally, combustion of a hydrocarbon fuel should produce only CO_2 and water (H_2O).Although, the diesel engine is an attractive solution for carbon dioxide (CO_2) reduction, there remains a challenge to control simultaneously (NOx) and CO emissions to a level required by prevailing regulations. The CO_2 emissions and fuel consumption were not degraded with the optimized engine. **Figure (4.24)** shows the CO_2 emissions vs. different Dual-Fuelling for constant operation conditions. Fixed speed 1000 rpm and average results CO_2 emission from different flow rates, consider different results CO_2 emissions for Ethanol/Water-Diesel. Water-Diesel DI produced 2% higher CO_2 emissions compared to Neat Diesel, as a result, CO_2 emissions increases because BSFC increases with the Water-Diesel. EW25-Diesel DI produced 10% higher CO_2 emissions compared to Neat Diesel, as the C/H ratio for both EW25-Diesel and conventional diesel are similar, it can be assumed that more fuel mass was consumed. EW50-Diesel DI produced 36% lower CO_2 emissions compared to Neat Diesel, due to more complete combustion and increased combustion temperature, which is caused by the presence of more oxygen in the fuel. EW75-Diesel DI produced 11% lower CO_2 emissions compared to Neat Diesel, the use of ethanol allows a higher relative concentration of oxygen to exist in the combustion gases and this achieves a greater conversion of CO to CO_2. Ethanol-Diesel DI produced 33% lower CO_2 emissions compared to Neat Diesel, ignition improver is very high.

4.2.25. Effect of Water–Diesel DI on Oxygen Emissions

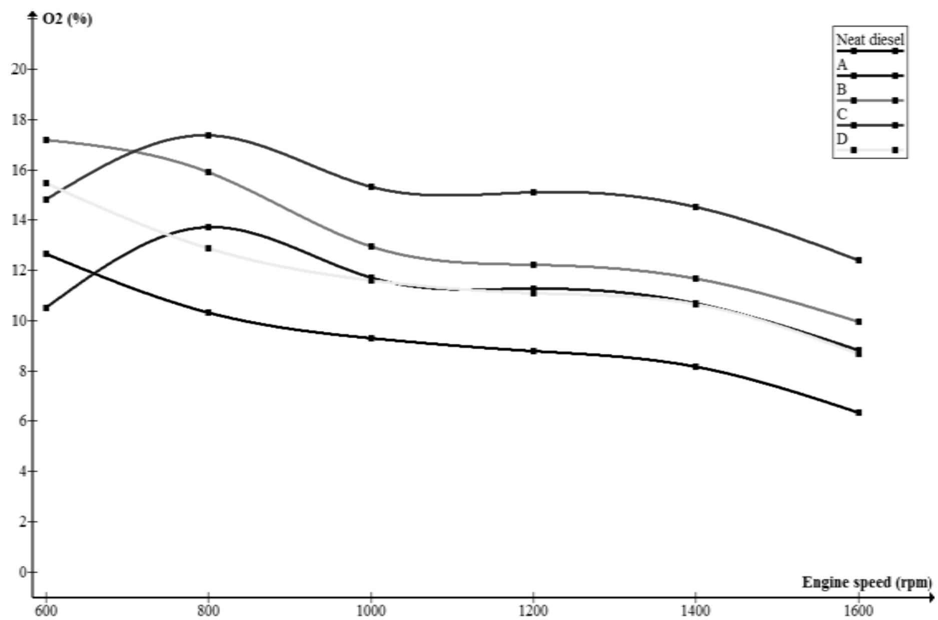

Figure 4.25: Wide Range Ratio Water–Diesel DI Effect on O_2 Emissions vs. Engine Speed

Figure (4.25) shows the wide range ratio Water–Diesel DI effect on O_2 emissions vs. engine speed. O_2 emissions values were measured at full load condition in the case of Water at different flow rates. O_2 emissions decreases with increase engine speed for tests Neat Diesel and with Water–Diesel, except special Water–Diesel in case (A, C) O_2 increases between (600-800) rpm. As the time available for fuel injection is very small there is a very little time available for fuel to uniformly mix with oxygen present in the combustion chambers.O_2 emission appears in engine exhaust because of heterogeneous mixing of O_2 with diesel. The minimum O_2 emission is 6.34 (%) at 1600 rpm in case Neat Diesel condition. According to Neat Diesel O_2 emission values, the highest change is 17% at 600 rpm in case (A) condition, the lowest change is -96% at 1600 rpm in case (C) condition.

At Water–Diesel DI to the combustion chamber was small and the mixture remained lean resulting in more O_2 emission in the exhaust. As the engine speed was increased more oxygen was consumed due to higher fuel supplied resulting in decrease in O_2 emission.

4.2.26. Effect of EW25–Diesel DI on Oxygen Emissions

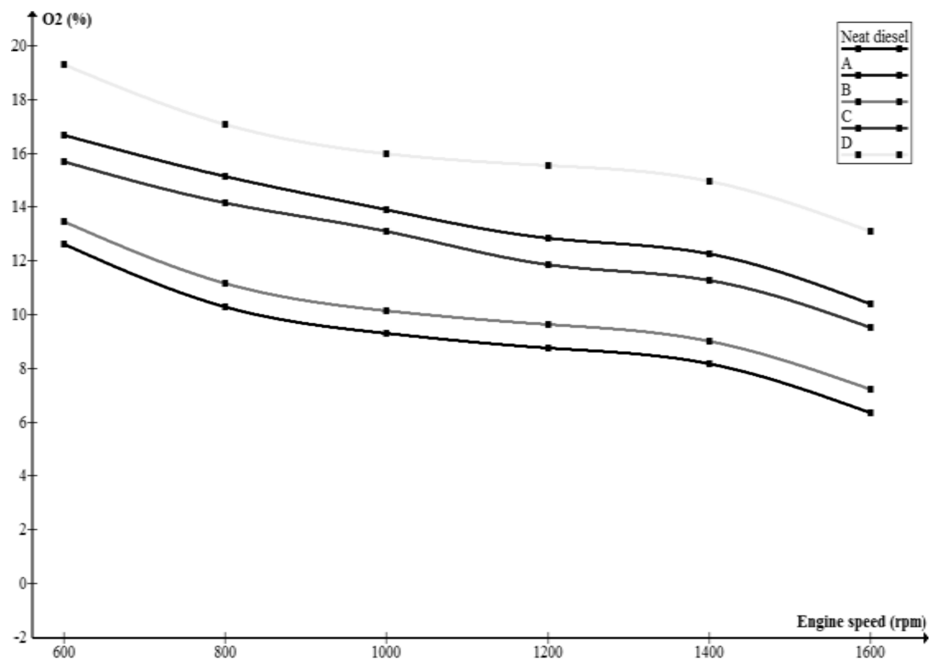

Figure 4.26: Wide Range Ratio EW25–Diesel DI Effect on O_2 Emissions vs. Engine Speed

Figure (4.26) shows the wide range ratio EW25–Diesel DI effect on O_2 emissions vs. engine speed. O_2 emissions values that measured at full load condition in the case of EW25 at different flow rates. O_2 emissions decreases with increase engine speed for tests Neat Diesel and with EW25–Diesel. Suggested that at higher engine speed, with the higher amount of burning fuel mass, combustion took place with lower excess oxygen, which Air/EW25 mixture, further increases in O_2 emission was observed in exhaust due to further improvement in combustion because mixing of air with EW25 started during inlet manifold and better air-fuel contact ratio was achieved resulting, contributed to NOx and CO the decreased of the emission, leading to the formation of a small number of emission, in almost large amount of oxygen emission. The minimum O_2 emission is 6.34 (%) at 1600 rpm in case Neat Diesel condition. According to Neat Diesel O_2 emission values, the highest change is -7% at 600 rpm in case (B) condition, the lowest change is -107% at 1600 rpm in case (D). This causes slow vaporization and mixing of fuel and air. Another reason for ethanol has oxygen content increases the O_2 emission to fuel ratio in the fuel rich regions, the O_2 emissions are generally reduced at full engine speed because of the increased air-fuel ratio and more complete combustion.

4.2.27. Effect of EW50–Diesel DI on Oxygen Emissions

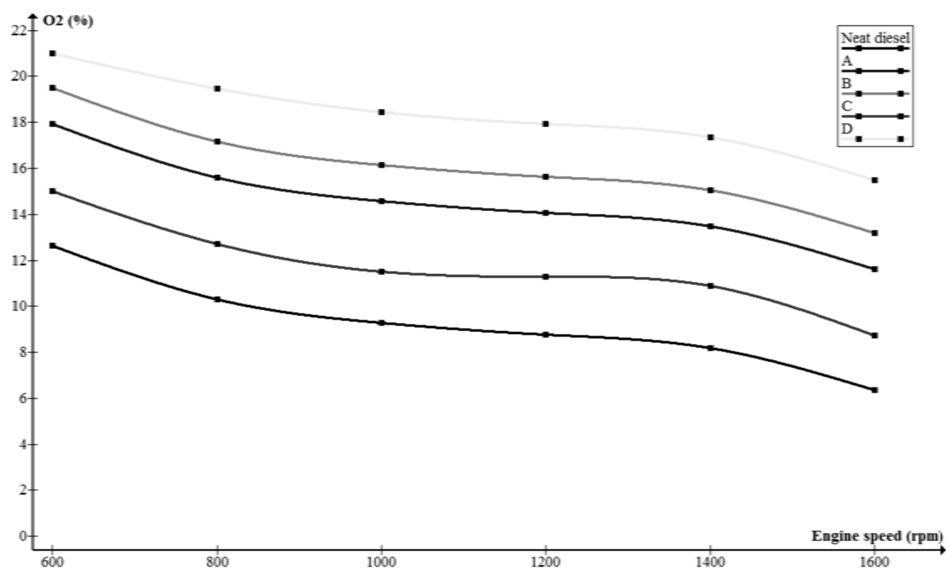

Figure 4.27: Wide Range Ratio EW50–Diesel DI Effect on O_2 Emissions vs. Engine Speed

Figure (4.27) shows the wide range ratio EW50–Diesel DI effect on O_2 emissions vs. engine speed. O_2 emissions values that measured at full load condition in the case of EW50 at different flow rates. O_2 emissions decrease with increase engine speed for tests Neat Diesel and with EW50–Diesel, which is due to the fact that more fuel is supplied and injected to larger engine speed, leading to the lower air fuel ratio. Then more precursors of O_2 emissions could be unused in the period of diffusion combustion. The minimum O_2 emission is 6.34 (%) at 1600 rpm in case Neat Diesel condition. According to Neat Diesel O_2 emission values, the highest change is -19% at 600 rpm in case (C) condition, the lowest change is -144% at

1600 rpm in case (D) condition. Higher O_2 concentration in EW50–Diesel operation is due to higher rate of ethanol, also due to the dissociation of water droplet and CO_2 at high combustion temperature produces more O_2 concentration at high temperature combustion zone in the cylinder. In lean mixture region, greater amount of dissociation occurs to produce half mole of O_2 from each of 1 mole CO_2 and 1 mole H_2O. Furthermore, lean mixture can accommodate more O_2 from this dissociation process than that of rich mixture region which introduces more complete combustion of the cylinder charge during mixing controlled and late combustion phases. The O_2 emission will be increased as exhaust gas during exhaust stroke.

4.2.28. Effect of EW75–Diesel DI on Oxygen Emissions

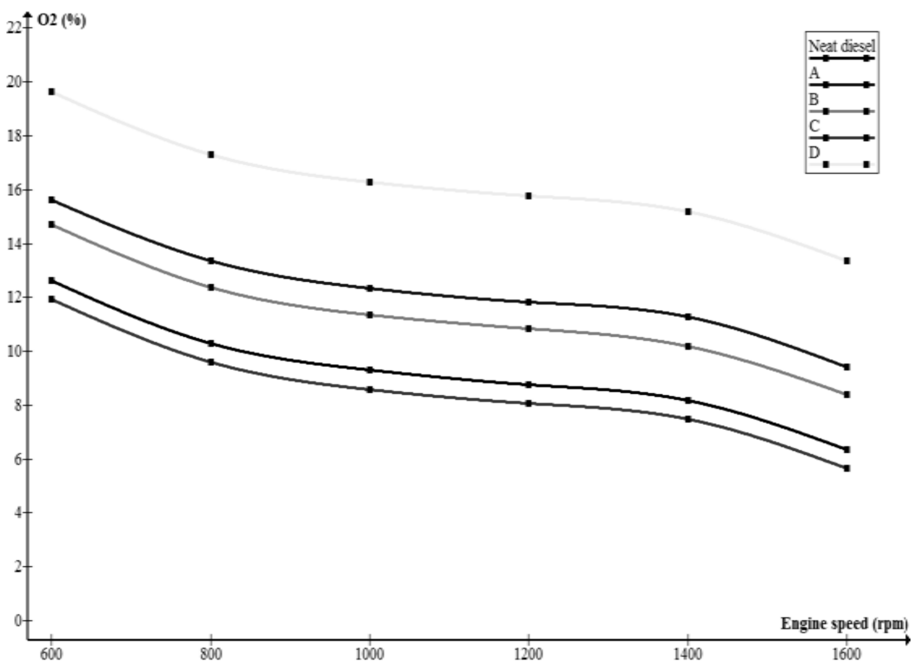

Figure 4.28: Wide Range Ratio EW75–Diesel DI Effect on O_2 Emissions vs. Engine Speed

Figure (4.28) shows the wide range ratio EW75–Diesel DI effect on O_2 emissions vs. engine speed. O_2 emissions values were measured at full load condition in the case of EW75 at different flow rates.O_2 emissions decreases with increase engine speed for tests Neat Diesel and with EW75–Diesel. O_2 emissions depend on oxygen content of ethanol and excess air of mixture. Furthermore, high oxygen content of fuel combined with low C/H ration also help to increase O_2 emission. The minimum O_2 emission is 5.64 (%) at 1600 rpm in case (C) condition. According to Neat Diesel O_2 emission values, the highest change is 11% at 1600 rpm in case (C) condition, the lowest change is -110% at 1600 rpm in case (D) condition. O_2 emission reduction in diesel engine with EW75–Diesel DI case (B) occurred due to the oxygen content of the ethanol in the fuels blended would help to increase the oxygen-to-fuel ratio in the fuel regions. With supported that the fact that ethanol has less carbon than diesel, therefore, the complete combustion will obtained and the O_2 emissions is reduced. Cetane number and oxygen content are more effective than the lower heating value and latent heat of vaporization for the peak temperature increase in the cylinder. Therefore, the concentration of O_2 emission increased when EW75–Dieselis used as the test fuel.

4.2.29. Effect of Ethanol–Diesel DI on Oxygen Emissions

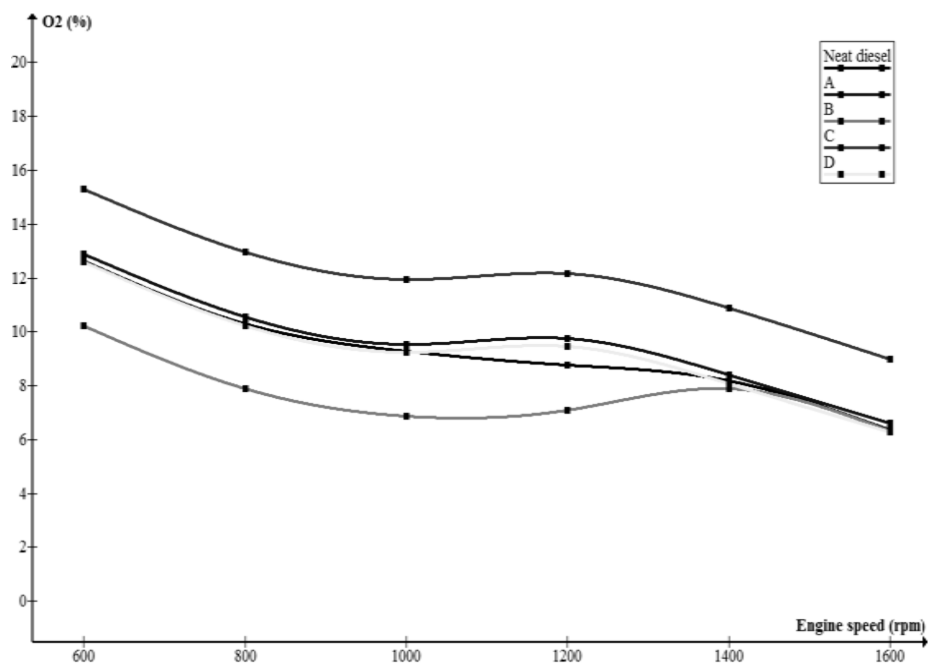

Figure 4.29: Wide Range Ratio Ethanol–Diesel DI Effect on O_2 Emissions vs. Engine Speed

Figure (4.29) shows the wide range ratio Ethanol–Diesel DI effect on O_2 emissions vs. engine speed. O_2emissions values that measured at full load condition in the case of Ethanol at different flow rates. O_2 emissions decreases with increase engine speed for tests Neat Diesel and with Ethanol–Diesel. This is due to the ethanol provides more oxygen component in the fuel and at higher engine speed the ratio of fuel-air is high and produce lower of O_2 emissions. On the other hand, the evaporation latent heat of ethanol is much higher than diesel, thus increases the temperature in the combustion chamber during the mixing. Therefore, it will benefit for O_2 emissions reduction. The minimum O_2 emission is 6.29 (%) at 1600 rpm in case (D) condition. According to Neat Diesel O_2 emissions values, the highest change is 26% at 1000 rpm in case (B) condition, the lowest change is -42% at 1600 rpm in case (C) condition. It is observed that O_2 emissions decreases for case (B) compared to Neat Diesel fuel, especially at high engine speeds of all cases not change. The presence of atomic bound oxygen in ethanol satisfies positive chemical control over O_2 emissions.

4.2.30. Effect of Different Dual-Fuelling on Oxygen Emissions at Same Operation Conditions

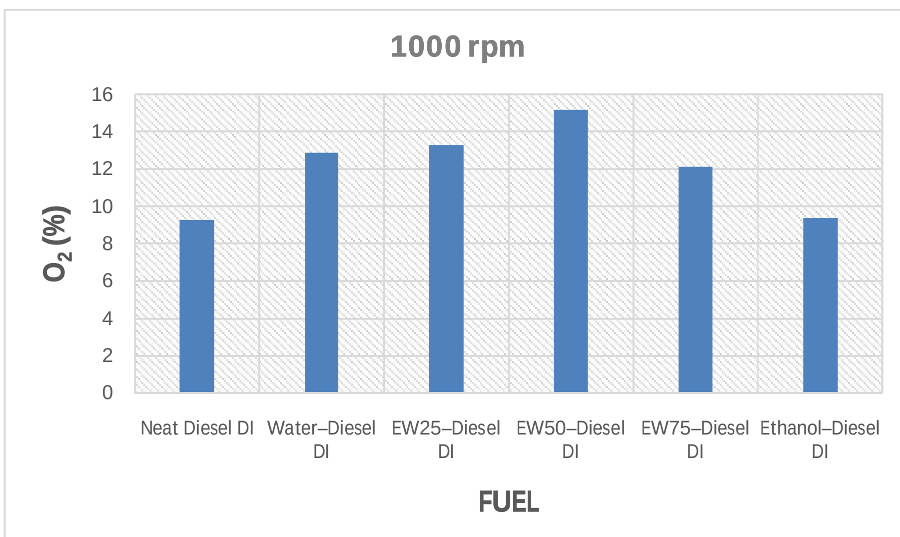

Figure 4.30: O_2 Emissions vs. Different Dual-Fuelling For Constant Operation Conditions

In the present study, water is introduced into the intake air during intake stroke and it partially evaporates and mixes up with the fresh air. Thus, the presence of water vapor causes a decrease in the temperature and a rise in the specific heat of the intake charge. This usually results in lower combustion temperatures and lower NOx emission. Furthermore, adding water into air charge will reduce the concentration of oxygen per unit volume of charge in the combustion chamber. Since ethanol has less carbon than diesel fuel and its oxygen content increases the oxygen to fuel ratio in the fuel rich regions.

Figure (4.30) shows the O_2 Emissions vs. different Dual-Fuelling for constant operation conditions. Fixed speed 1000 rpm and average results O_2 emissions from different flow rates, consider different results O_2 emissions for Ethanol/Water–Diesel DI. Water–Diesel DI produced 39% higher O_2 emissions compared to Neat Diesel, the water molecules that formed in this state were vaporized because of high temperature of combustion chamber. For this reason, temperature of combustion chamber was partly attracted and so the peak temperature of combustion state decreased. In these conditions, we can say that the engine operates slightly cooler with rich oxygenated fuels. EW25–Diesel DI produced 43% higher O_2 emissions compared to Neat Diesel, the EW25–Diesel evaporated liquid fuel then burn rapidly in combination, with pre-mixed combustion in the ethanol air mixture depleting the region around the fuel spray of oxygen so that after the first peak in heat release rate there is a significant dwell in the combustion rate until the remaining injected liquid fuel vapor finds increases oxygen for emissions. EW50–Diesel DI produced 63% higher O_2 emissions compared to Neat Diesel, this combustion of the EW50/Air mixture ahead of the amount ethanol could lead to local depletion of oxygen, resulting in reducing heat release rate after the first peak, even though liquid fuel is still being injected. The last phase of heat release would then involve combustion of both evaporated liquid fuel and any remaining ethanol, with a significant delay until the late injected liquid fuel can find increases oxygen for emissions. EW75–Diesel DI produced 31% higher O_2 emissions compared to Neat Diesel, this can be explained by the fact that oxygen content in the ethanol is effective in improving combustion in a mixture. The reason may be the faster combustion rate due to high temperature as a result of longer ignition delay and oxygen enhanced combustion can find increases oxygen for emissions. Ethanol–Diesel DI produced 1% higher O_2 emissions compared to Neat Diesel, due to combustion process is mixing controlled the use of ethanol injection has a beneficial effect, while if the combustion process is evaporation controlled the effect is adverse.

4.2.31. Effect of Water–Diesel DI on Exhaust Gas Temperature

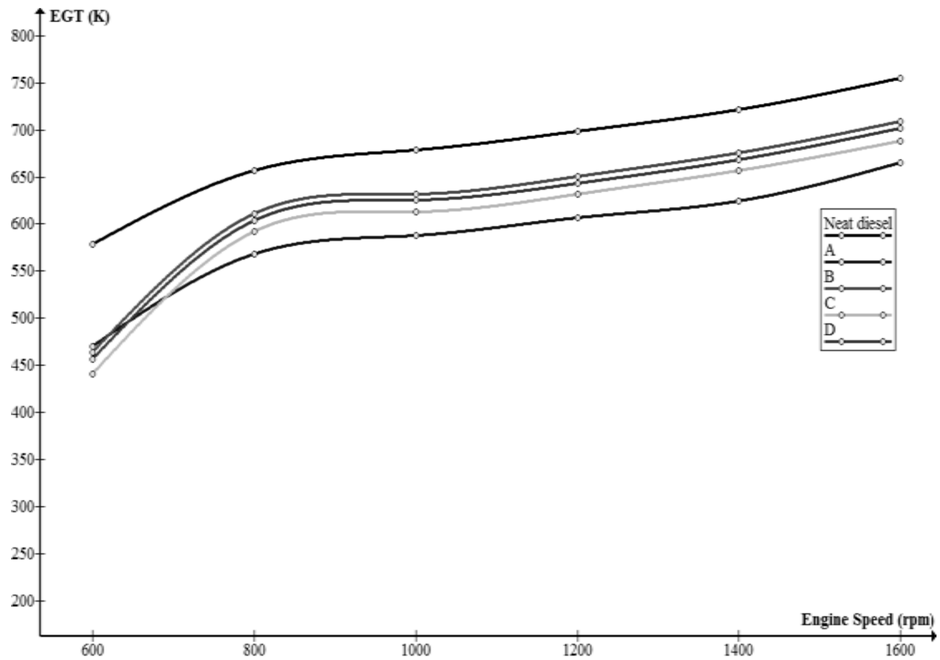

Figure 4.31: Wide Range Ratio Water–Diesel DI Effect on EGT vs. Engine Speed

Figure (4.31) shows the wide range ratio Water–Diesel DI effect on EGT vs. engine speed. EGT values that measured at full load condition in the case of Water at different flow rates. EGT increases with increase engine speed for tests Neat Diesel and with Water–Diesel. The latent heat of water will cool the charge due to the evaporation of water and also the cylinder average temperature following injection and before ignition. Water–Diesel DI that micro explosion of water reduced peak combustion temperature. The minimum EGT is 441 K at 600 rpm in case (C) condition. According to Neat Diesel EGT values, the highest change is 24% at 600 rpm in case (C) condition , the lowest change is 6% at 1600 rpm in case (B) condition. The burning of Neat Diesel fuel appears to have a larger EGT, primarily because of its higher heating value and lower amount of oxygen than the other

Water–Diesel DI. The heat absorbed by water can explain the decrease in the exhaust temperature. The latent heat of water will cool the charge due to the evaporation of water, and the cylinder average temperature following injection and before ignition becomes lower as the water percentage increases.

4.2.32. Effect of EW25–Diesel DI on Exhaust Gas Temperature

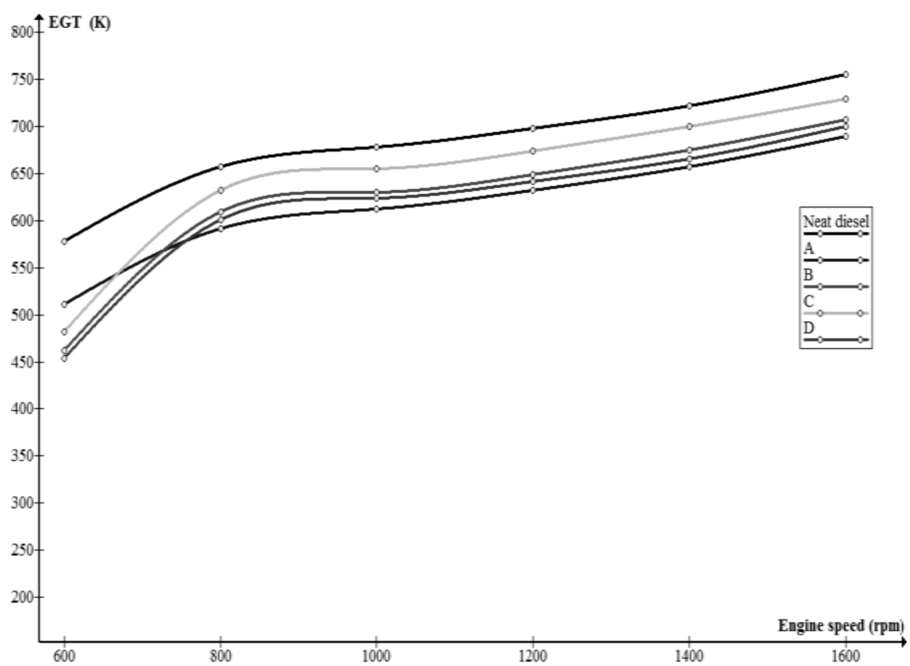

Figure 4.32: Wide Range Ratio EW25–Diesel DI Effect on EGT vs. Engine Speed

Figure (4.32) shows the wide range ratio EW25–Diesel DI effect on EGT vs. engine speed. EGT values that measured at full load condition in the case of EW25 at different flow rates. EGT increases with increase engine speed for tests Neat Diesel and with EW25–Diesel. This is because the engine speed increase was accompanied with an increase in the diesel fuel consumption rate. Conversely, because of the lower heating value of EW25–Diesel DI, the increase in the EW25 proportion in the diesel fuel reduced the EGT. The minimum EGT is 453 K at 600 rpm in case (D) condition. According to Neat Diesel EGT values, the highest change is 22% at 600 rpm in case (D) condition , the lowest change is 3% at 1400 rpm in case (C) condition. This is because EW25–Diesel is an oxygen-rich fuel. When the oxygen content in the ethanol fuel increases the ignition delay is shortened. The amount of premixed fuel and peak burning temperature were lowered.

4.2.33. Effect of EW50–Diesel DI on Exhaust Gas Temperature

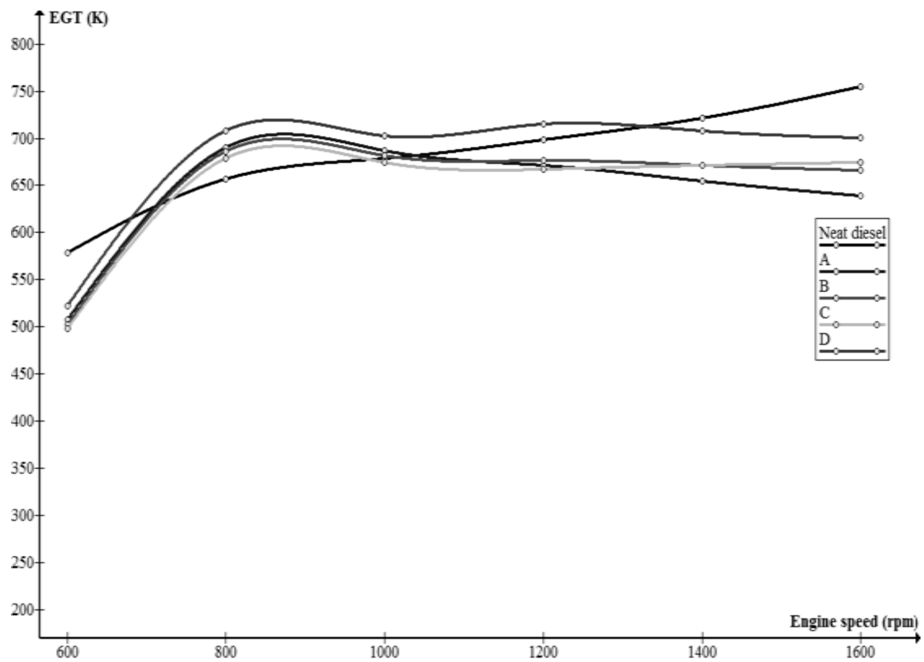

Figure 4.33: Wide Range Ratio EW50–Diesel DI Effect on EGT vs. Engine Speed

Figure (4.33) shows the Wide Range Ratio EW50–Diesel DI Effect on EGT vs. engine speed. EGT values that measured at full load condition in the case of EW50 at different flow rates. EGT increases with increase engine speed for tests Neat Diesel. EGT increases with increase engine speed to 800 rpm after that decreases to full engine speed for case EW50–Diesel DI. It was observed that all EW50–Diesel fuel blends slightly reduced the EGT. This may be attributed to the lower energy content, higher oxygen and latent heat of evaporation content of the EW50–Diesel fuel blends. Increasing molecular oxygen content of the fuel blends increases the combustion temperature and decreases the energy content of the fuel. EW50–Diesel fuels blend supplies more oxygen to the combustion chamber, hence elevates the combustion temperature. However, the higher heat of evaporation of EW50–Diesel fuel blends cause to lower temperature in the combustion chamber. As a result of these conflicting factors, the variation of ethanol content in the fuel blends has a detractive effect on the EGT. The minimum EGT is 498 K at 600 rpm in case (C) condition. According to Neat Diesel EGT values, the highest change is 15% at 1600 rpm in case (A) condition , the lowest change is -8% at 800 rpm in case (D) condition. The lower EGT of the EW50–Diesel engine makes it difficult to burn completely, especially when the amount of ethanol injected is low, which results in a lean air/ethanol mixture to burn at low engine speeds.

4.2.34. Effect of EW75–Diesel DI on Exhaust Gas Temperature

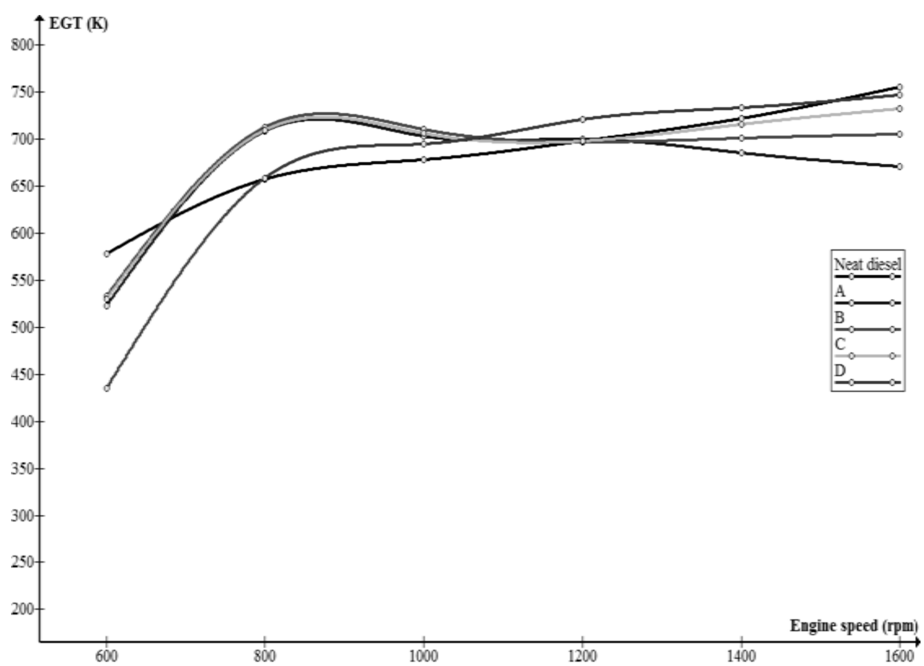

Figure 4.34: Wide Range Ratio EW75–Diesel DI Effect on EGT vs. Engine Speed

Figure (4.34) shows the wide range ratio EW75–Diesel DI effect on EGT vs. engine speed. EGT values that measured at full load condition in the case of EW75 at different flow rates. EGT increases with increase engine speed for Tests Neat Diesel and EW75–Diesel DI case (D). EGT increases with increase engine speed to

800 rpm after that decreases to full engine speed for EW75–Diesel DI case (A). EGT increases with increase engine speed to 800 rpm, after that decreases to 1200 rpm, and increases to full load for case (B, C). It was observed that all EW75–Diesel DI in all cases, this lower temperature may be attributed to both its lower calorific value and its higher heat of evaporation. However, the latter can be offset by the opposing effect of the lower cetane number (and thus longer ignition delay) of the EW75–Diesel, leading possibly to higher local temperatures during the premixed part of combustion. The minimum EGT is 435 K at 600 rpm in case (D) condition. According to Neat Diesel EGT values, the highest change is 25% at 600 rpm in case (D) condition , the lowest change is -8% at 800 rpm in case (B) condition. During the EW75–Diesel phase, fuel and air are well-mixed prior to combustion to produce a more complete combustion and higher combustion temperature.

4.2.35. Effect of Ethanol–Diesel DI on Exhaust Gas Temperature

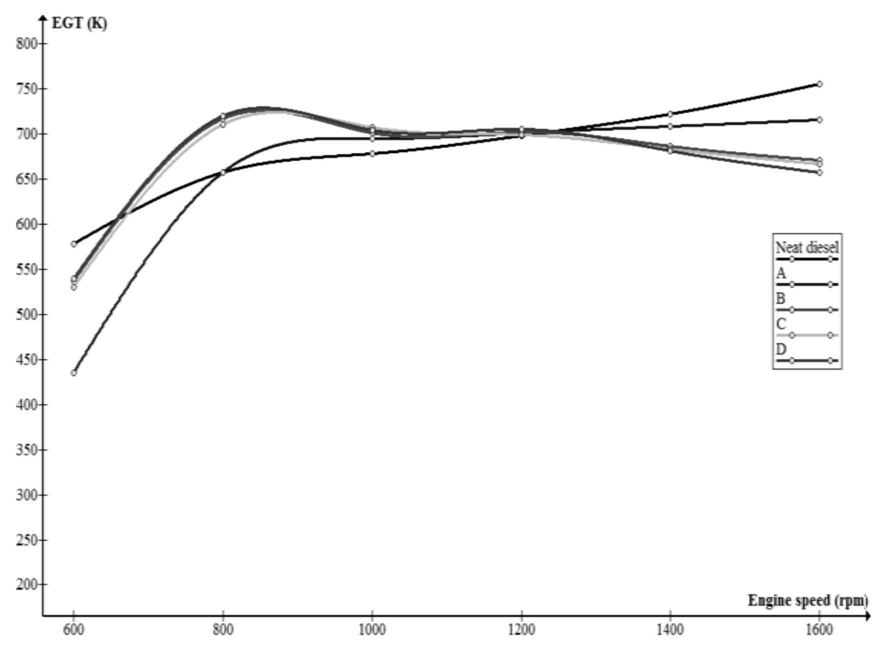

Figure 4.35: Wide Range Ratio Ethanol–Diesel DI Effect on EGT vs. Engine Speed

Figure (4.35) shows the wide range ratio Ethanol–Diesel DI effect on EGT vs. engine speed. EGT values that measured at full load condition in the case of Ethanol at different flow rates. EGT increases with increase engine speed for tests Neat Diesel. EGT increases with increase engine speed to 800 rpm after that decreases to full engine speed for case Ethanol–Diesel. The lower gas temperature can be attributed to two reasons. Firstly, evaporation of ethanol in the intake system lowers the intake mixture temperature. Secondly, the latent heat of vaporization of ethanol is four times that of diesel, but the heating value is about half that of diesel. Thus, more heat is needed for ethanol than for diesel for fuel vaporization, while the energy released by ethanol is lower than that from the same mass of diesel. The minimum EGT is 435 K at 600 rpm in case (A) condition. According to Neat Diesel EGT values, the highest change is 25% at 600 rpm in case (A) condition, the lowest change is -10% at 800 rpm in case (D) condition. Ethanol has high latent heat of vaporization hence less amount of heat is released during combustion process which reduces the combustion temperature.

4.2.36. Effect of Different Dual-Fuelling on Exhaust Gas Temperature at Same Operation Conditions

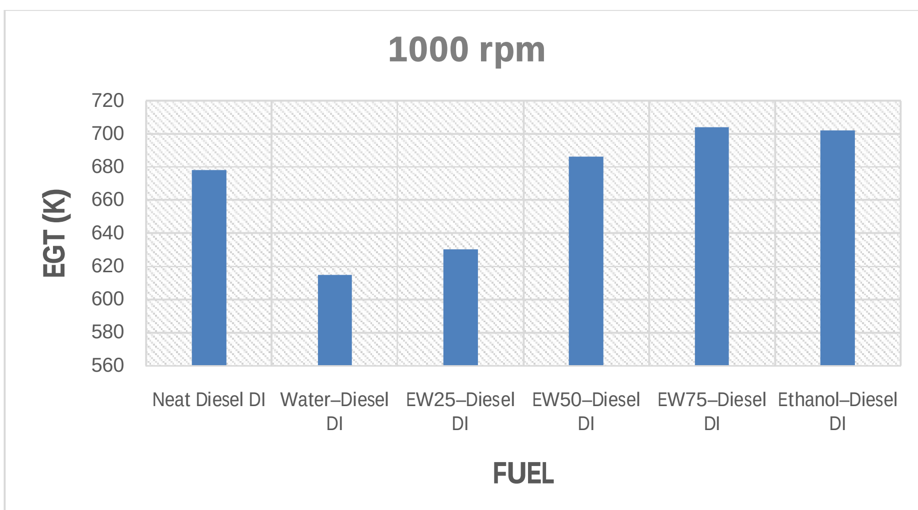

Figure 4.36: EGT vs. Different Dual-Fuelling For Constant Operation Conditions

Figure (4.36) shows the EGT vs. different Dual-Fuelling for constant operation conditions. Fixed speed 1000 rpm and average results EGT from different flow rates, consider different results EGT for Ethanol/Water–Diesel. Water–Diesel DI produced 9% lower EGT compared to Neat Diesel. Peak temperatures in the domain are reduced by two localized phenomena. First, vaporization of liquid water decreases the internal energy proportionally to the vaporization enthalpy of the liquid water. Secondly, higher concentrations of water vapor increase the specific heat of the gas. EW25–Diesel DI produced 7% lower EGT compared to Neat Diesel, increasing ethanol fraction suggesting increasing combustion temperature. EW50–Diesel DI produced 1% higher EGT compared to Neat Diesel, reason amount of ethanol and water by volume. EW75–Diesel DI, Ethanol–Diesel DI were produced 4% higher EGT compared to Neat Diesel. This is due to the ethanol as partially oxidized (OH radicals), and while burning has higher burning temperature, increased EGT were observed when running on blends.

4.3. Effect of Ethanol/Water Addition to Diesel Fuel on Performance Engine

BSFC is the ratio between mass fuel consumption and brake power and it is inversely proportional to thermal efficiency for a given fuel. BSFC is computed by following equation:

$$BSFC = \frac{m°_d + m°_e}{BP}$$

Where

BP is the brake power in KW.

$m°_d$ And $m°_e$ are the mass consumption rates of diesel fuel and ethanol, respectively, in g/h.

BTE is defined as the brake power divided by the fuel energy supplied through fuel injection. BTE is calculated by the following formula:

$$BTE = \frac{BP}{(m^{\circ}_{d} \times CV_{LHV,d}) + (m^{\circ}_{e} \times CV_{LHV,e})} \times 100\%$$

Where

BP is the brake power, KW.

m°_{d} Is the mass consumption rate of diesel fuel, kg/s.

m°_{e} Is the mass consumption rate of ethanol, kg/s.

$CV_{LHV,d}$ Is the lower heating value of diesel fuel, kJ/kg.

$CV_{LHV,e}$ Is the lower heating value of ethanol, kJ/kg.

In this work, literatures illustrated the effect of ethanol dual fuels on the BTE have been surveyed. Most authors have reported around same results after investigating ethanol dual fuel technology method on diesel engine.

4.3.1. Effect of Water–Diesel DI on Brake Specific Fuel Consumption

Figure (4.37) shows the Wide Range Ratio Water–Diesel DI Effect on BSFC vs. engine speed. BSFC values that measured at full load condition in the case of Water at different flow rates. BSFC decreases with increase engine speed for tests Neat Diesel and Water–Diesel at case (D). BSFC decreases with increase engine speed to 1000 rpm after that increases to full engine speed for Water–Diesel at case (A, B, and C). When the Water–Diesel results in lower heat values and hence increases fuel consumption at high engine speed for the same power output. The absorption of energy for vaporization of the Water–Diesel also causes a higher BSFC than the Neat Diesel at higher speed. The minimum BSFC is 294 (g/KWh) at 1400 rpm in case (D) condition. According to Neat Diesel BSFC values, the highest change is 40% at 600 rpm in case (D) condition , the lowest change is -75% at 1600 rpm in case (B) condition. Since water has no calorific value at all. The reduction in BSFC with Water–Diesel may be attributed to formation of a finer spray due to rapid evaporation in the water, longer ignition delay results in more fuel burning in premixed combustion and suppression of thermal dissociation due to lower cylinder average temperature. The evaporation and additional mass of water cause the cylinder average temperature to become lower as the water amount and pressure were increased.

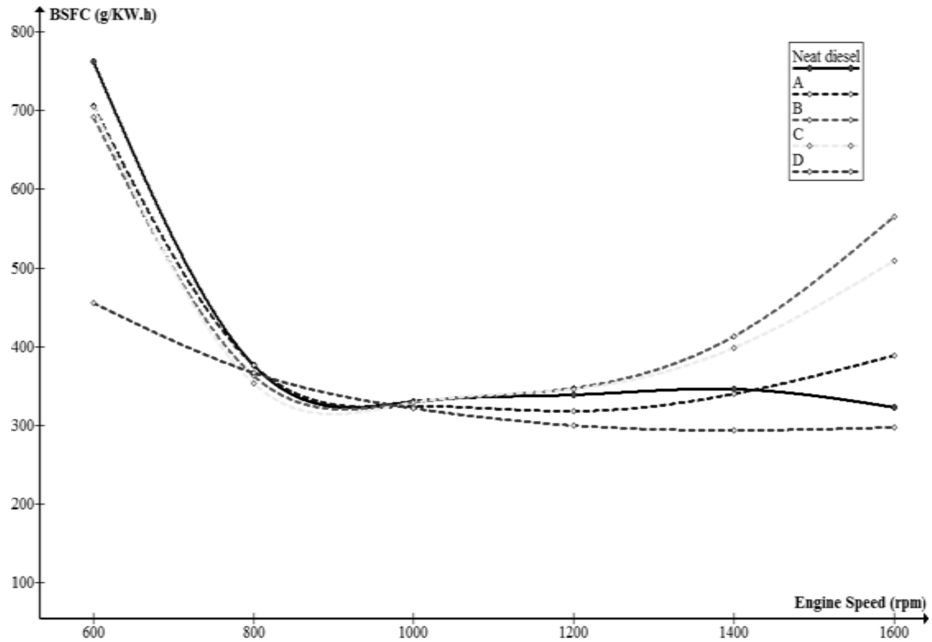

Figure 4.37: Wide Range Ratio Water–Diesel DI Effect on BSFC vs. Engine Speed

4.3.2. Effect of EW25–Diesel DI on Brake Specific Fuel Consumption

Figure (4.38) shows the wide range ratio EW25–Diesel DI effect on BSFC vs. engine speed. BSFC values that measured at full load condition in the case of EW25 at different flow rates. BSFC decreases with increase engine speed for tests Neat Diesel and EW25–Diesel. The most possible reason to reduce in BSFC and increase in torque, effective power and efficiency by means of EW25–Diesel DI could be explained with the improvement in vaporization and mixing processes which leads to a shorter combustion reaction. Generally, the engine consumes less EW25–Diesel fuel than with reference Neat Diesel fuel to generate the same engine output torque because of the heat content of the EW25–Diesel fuel. The minimum

BSFC is 297 (g/KWh) at 1600 rpm in case (D) condition. According to Neat Diesel BSFC values, the highest change is 52% at 600 rpm in case (C) condition , the lowest change is -2% at 600 rpm in case (A) condition. As would expected, the BSFC decreases with the increasing EW25 content in the fuel because of the increased energy content.

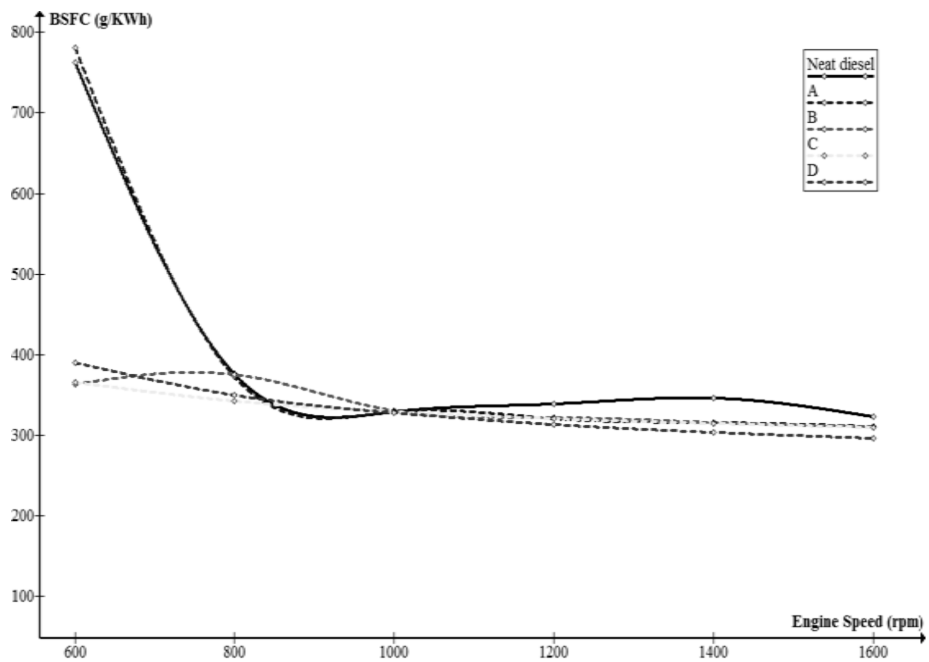

Figure 4.38: Wide Range Ratio EW25–Diesel DI Effect on BSFC vs. Engine Speed

4.3.3. Effect of EW50–Diesel DI on Brake Specific Fuel Consumption

Figure (4.39) shows the wide range ratio EW50–Diesel DI effect on BSFC vs. engine speed. BSFC values that measured at full load condition in the case of EW50 at different flow rates. BSFC decreases with increase engine speed for tests Neat Diesel and EW50–Diesel. When the engine speed is less than 1600 rpm, BSFC for EW50–Diesel is lower than that Neat Diesel. This can be explained by the following reason: when the speed increases, the fuel supply increases, so the injection duration and combustion were enlarged. Therefore, the BSFC for EW50–Diesel is lower than that Neat Diesel. The minimum BSFC is 313 (g/KWh) at 1600 rpm in case (D) condition. According to Neat Diesel BSFC values, the highest change is 54% at 600 rpm in case (C) condition , the lowest change is 0% at 1000 rpm in case (D) condition. Thus of the expected fuel consumption is the oxygen content in the EW50–Diesel fuel, with that of the Neat Diesel fuel, thus more fuel is needed to maintain the same power output when the blended fuel is in use.

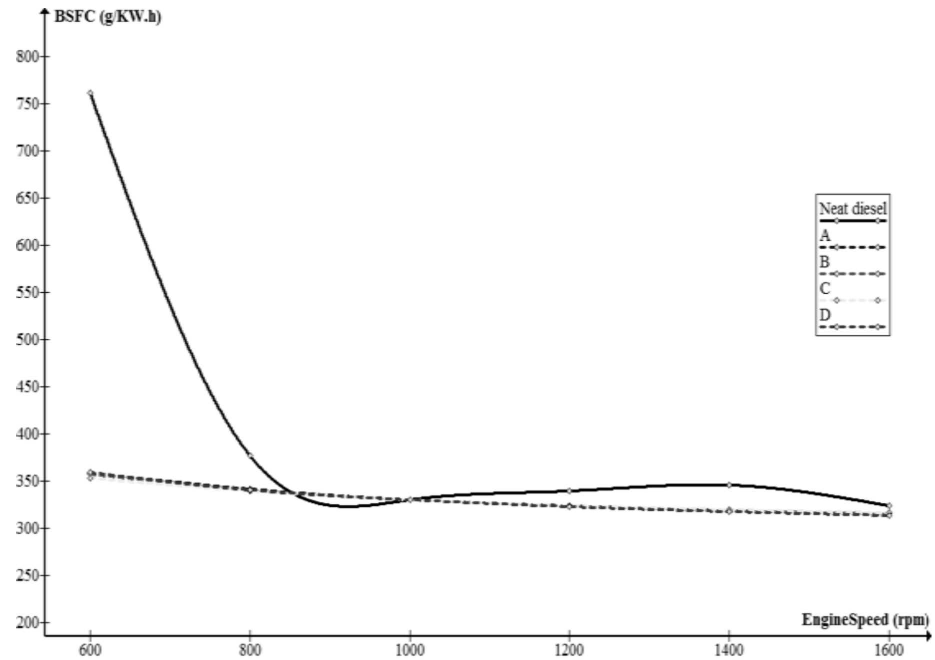

Figure 4.39: Wide Range Ratio EW50–Diesel DI Effect on BSFC vs. Engine Speed

4.3.4. Effect of EW75–Diesel DI on Brake Specific Fuel Consumption

Figure (4.40) shows wide range ratio EW75–Diesel DI effect on BSFC vs. engine speed. BSFC values that measured at full load condition in the case of EW75 at different flow rates. BSFC decreases with increase engine speed for tests Neat Diesel and EW75–Diesel. The higher BSFC values of the Neat Diesel against the corresponding ones of the EW75–Diesel DI may be attributed to the finer atomization and the lower heat losses due to lower temperatures. The minimum BSFC is 311 (g/KWh) at 1600 rpm in case (A) condition. According to Neat Diesel BSFC values, the highest change is 52% at 600 rpm in case (D) condition , the lowest change is 0% at 1000 rpm in case (D) condition. As can be observed the change of the flow rate of EW75–Diesel have less influence in the BSFC concentrations than other percentage Ethanol/Water–Diesel. The strong influence of EW75–Diesel was one result already expected, considering that EW75 dilution in the inlet mixture increases both the oxygen concentration and the combustion rate significantly.

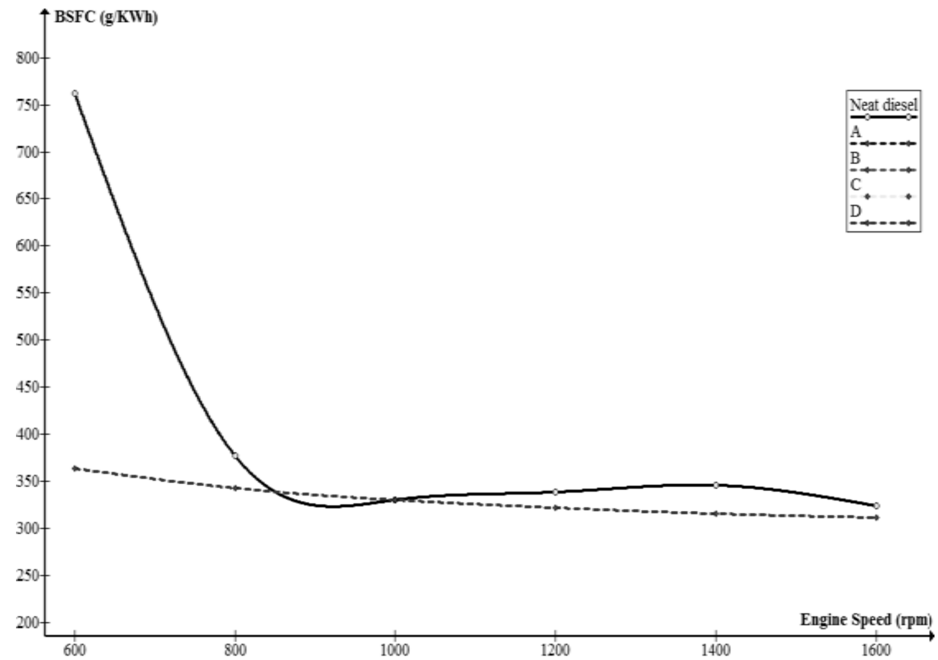

Figure 4.40: Wide Range Ratio EW75–Diesel DI Effect on BSFC vs. Engine Speed

4.3.5. Effect of Ethanol–Diesel DI on Brake Specific Fuel Consumption

Figure (4.41) shows the wide range ratio Ethanol–Diesel DI effect on BSFC vs. engine speed. BSFC values that measured at full load condition in the case of Ethanol at different flow rates. BSFC decreases with increase engine speed for tests Neat Diesel and Ethanol–Diesel at case (A, C, and D). The decrease in BSFC could be explained by the fact that, as the engine speed increases, the rate of increasing BP is much more than that of the increased fuel consumption owing to a rise in the combustion temperature. BSFC increases with increase engine speed for Ethanol–

Diesel at case (B). This is result may be depend operation condition of ethanol. The minimum BSFC is 271 (g/KWh) at 600 rpm in case (B) condition. According to Neat Diesel BSFC values, the highest change is 65% at 600 rpm in case (B) condition , the lowest change is -16% at 1600 rpm in case (B) condition. The results showed that increasing ethanol decreased the BSFC for diesel fuel. This behavior is attributed to heating value per unit mass of the ethanol, which is noticeably replacement of the diesel fuel.

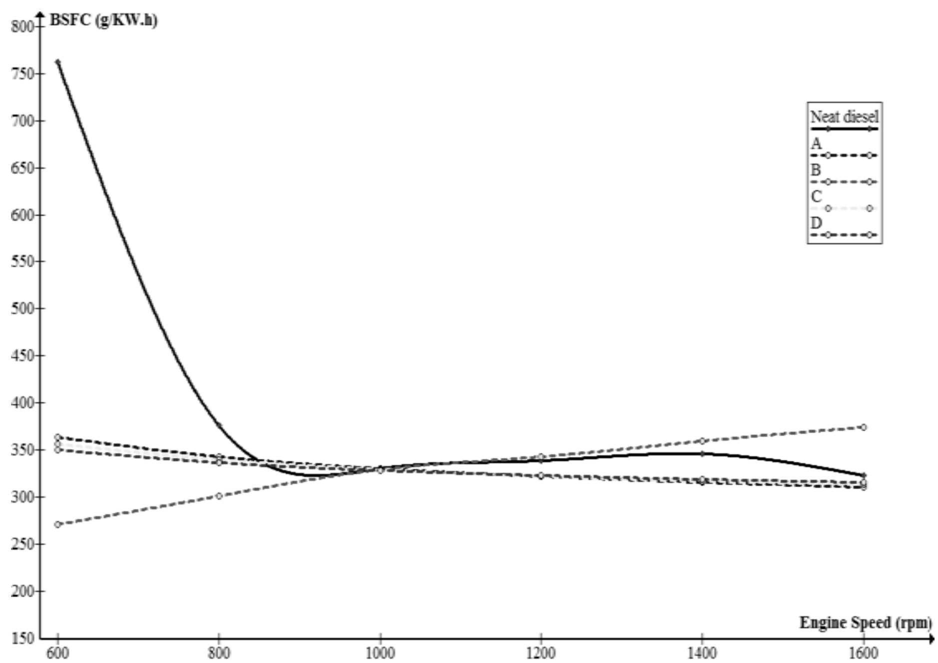

Figure 4.41: Wide Range Ratio Ethanol–Diesel DI Effect on BSFC vs. Engine Speed

4.3.6. Effect of Different Dual-Fuelling On Brake Specific Fuel Consumption at Same Operation Conditions

In addition, proposed that the improved energy efficiency from these oxygenated fuels was caused by their more complete combustion (a greater fraction of the fuel oxygen supply) and the premixed combustion region that resulted from the ignition delay caused by the lower cetane number of the Ethanol/Water.

Figure (4.42) shows the BSFC vs. different Dual-Fuelling for constant operation conditions. Fixed speed 1200 rpm and average results BSFC from different flow rates, consider different results BSFC for Ethanol/Water–Diesel. Water–Diesel DI produced 3% lower BSFC compared to Neat Diesel, the EGT decreases. The ignition delay increase with Water–Diesel, however, it was found that strong micro-explosion occur in the bottom region of the luminous flames near the spray tip. There are numerous small, round regions due to explosion of superheated water in the droplets. These spherical regions may grow bigger, collapse with new flames or connect with the mean flow motion. There is a range of the sizes, from small ones that are barely identifiable to those of the diameters of a few millimeters. The luminous flames of the diesel fuel are more homogeneous, brighter and yellow in color with no micro-explosion observed. Micro-explosions of a group of droplets of the water are strong enough to eject fragments of torn droplets to expand the tip and angle of the spray, enhancing mixing of fuel with surrounding air for faster and more efficient combustion. The combustion of Water–Diesel droplet is characterized by the micro-explosion, which is caused by the volatility difference between the Water–Diesel and the Neat Diesel fuel. EW25–Diesel DI produced 6% lower BSFC compared to Neat Diesel, due to the low viscosity and boiling point of ethanol, the temperature and pressure of fuel in the cylinders passes through the gas–liquid phase region of EW25–Diesel during the high-temperature compression process. And then the ethanol is vaporized, which promotes the atomization of fuel and the formation of an air–fuel mixture and improves the ignition condition. Additionally, the high oxygen content of ethanol can increase the air–fuel ratio and improve the BSFC. EW50–Diesel DI produced 5% lower BSFC compared to Neat Diesel, the reduction in the BSFC may be caused from the improvement of the air–fuel mixture formation due to ethanol addition. Ethanol needs more heat to evaporate in cylinder during intake and compression strokes. So in-cylinder temperature levels decreases which cause the reduced compression work. EW75–Diesel DI produced 5% lower BSFC compared to Neat Diesel, can be explained by the incomplete combustion resulting from the lower temperature as the injection timing is retarded. Ethanol–Diesel DI produced 3% lower BSFC compared to Neat Diesel, caused by the increased heating value for ethanol.

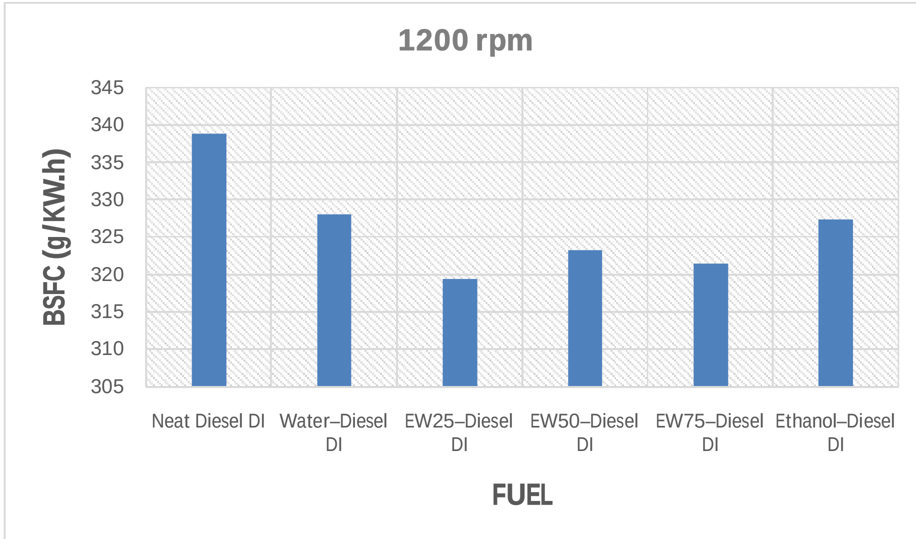

Figure 4.42: BSFC vs. Different Dual-Fuelling For Constant Operation Conditions

4.3.7. Effect of Water–Diesel DI on Brake Thermal Efficiency

Figure (4.43) shows the wide range ratio Water–Diesel DI effect on BTE vs. engine speed. BTE values that measured at full load condition in the case of Water at different flow rates. BTE increases with increase engine speed for tests Neat Diesel and Water–Diesel at case (D). BTE increases with increase engine speed to 1200 rpm after that decreases to full engine speed for Water–Diesel at case (A, B, and C), due to the knock. Advancing the injection timing improves the BTE due to longer time available for proper mixing and combustion. Water in small concentrations had a tendency to increase the combustion efficiency because of more complete oxidation of UBHC and CO. The maximum BTE is 28.2 (%) at 1400 rpm in case (D) condition. According to Neat Diesel BTE values, the highest change is 43% at 1600 rpm in case (B) condition , the lowest change is -67% at 600 rpm in case (D) condition. However, this is compensated by the increase in the delay time induced by the Water–Diesel and therefore the beginning of the heat release in the main chamber is detected approximately at the same time as with the reference fuel. From this point on, the combustion itself is faster for the Water–Diesel, which explains the increased BTE. Also result decreased BTE at high engine speed, the spray boundary where the Water–Diesel has already mixed beyond the lean flammability limit during the ignition delay period.

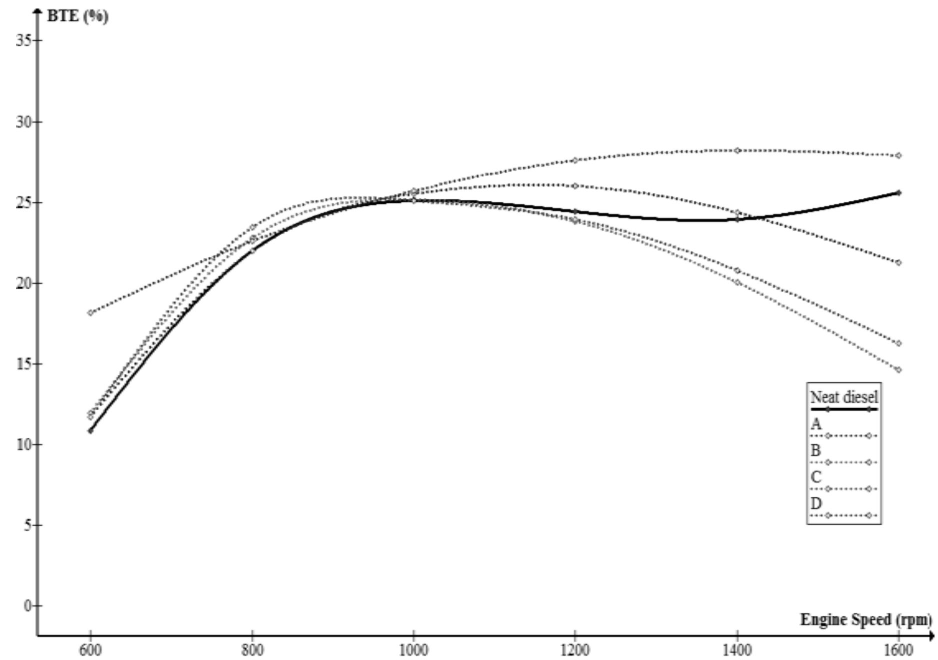

Figure 4.43: Wide Range Ratio Water–Diesel DI Effect on BTE vs. Engine Speed

4.3.8. Effect of EW25–Diesel DI on Brake Thermal Efficiency

Figure (4.44) shows the wide range ratio EW25–Diesel DI effect on BTE vs. engine speed. BTE values that measured at full load condition in the case of EW25at different flow rates.BTE increases with increase engine speed for tests Neat Diesel and EW25–Diesel. The higher BTE values of the EW25–Diesel blends may be attributed to the finer atomization and the lower heat losses due to lower temperatures. The maximum BTE is 27.9 (%) at 1600 rpm in case (D) condition. According to Neat Diesel BTE values, the highest change is 2% at 600 rpm in case (A) condition , the lowest change is -110% at 600 rpm in case (B) condition. The observation of the BTE of EW25–Diesel shows a potential for combined emissions reduction and efficiency improved through engine optimization by refining the injection timing.

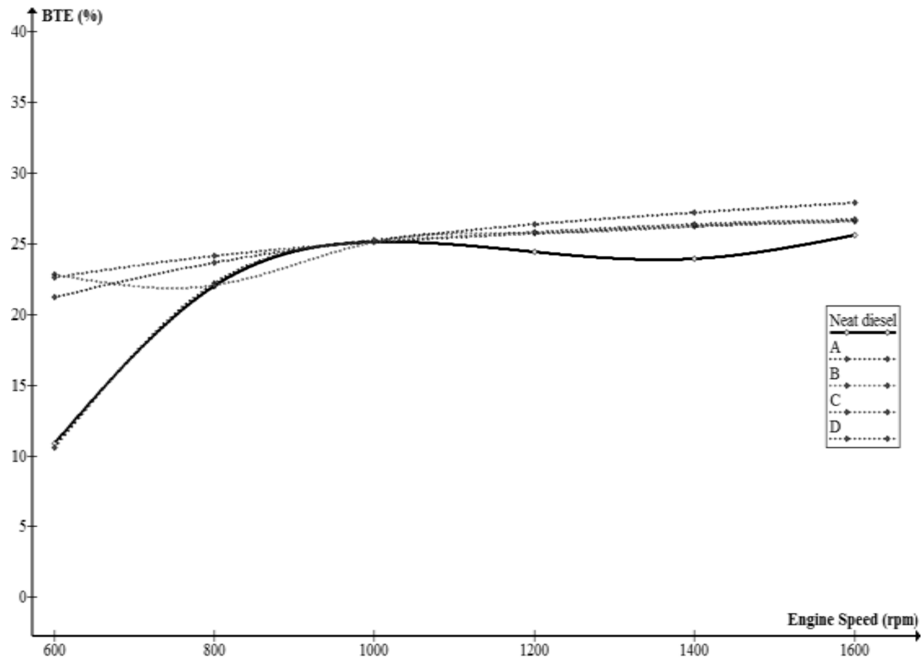

Figure 4.44: Wide Range Ratio EW25–Diesel DI Effect on BTE vs. Engine Speed

4.3.9. Effect of EW50–Diesel DI on Brake Thermal Efficiency

Figure (4.45) shows the wide range ratio EW50–Diesel DI Effect on BTE vs. engine speed. BTE values that measured at full load condition in the case of EW50 at different flow rates. BTE increases with increase engine speed for tests Neat Diesel and EW50–Diesel. This effect can be explained by the abrupt modification of the combustion process due to a lack of oxidant to the combustion reaction, generating a high concentration of BTE. The maximum BTE is 26.5 (%) at 1600 rpm in case (D) condition. According to Neat Diesel BTE values, the highest change is 0% at 1000 rpm in case (B) condition , the lowest change is -116% at 600 rpm in case (C) condition. BTE is slightly increased for EW50–Diesel compared to Neat Diesel fuel at all condition. This can be attributed to the rapid premixed combustion part possessed by ethanol mixing because of improved mixing during ignition delay, oxygen enrichment, leading to higher percentage of combustion and to the lower heat losses and ‘leaner’ combustion. The improvement of diffusive combustion phase would have also resulted due to oxygen enrichment. In addition, the total combustion duration is shortened for EW50–Diesel.

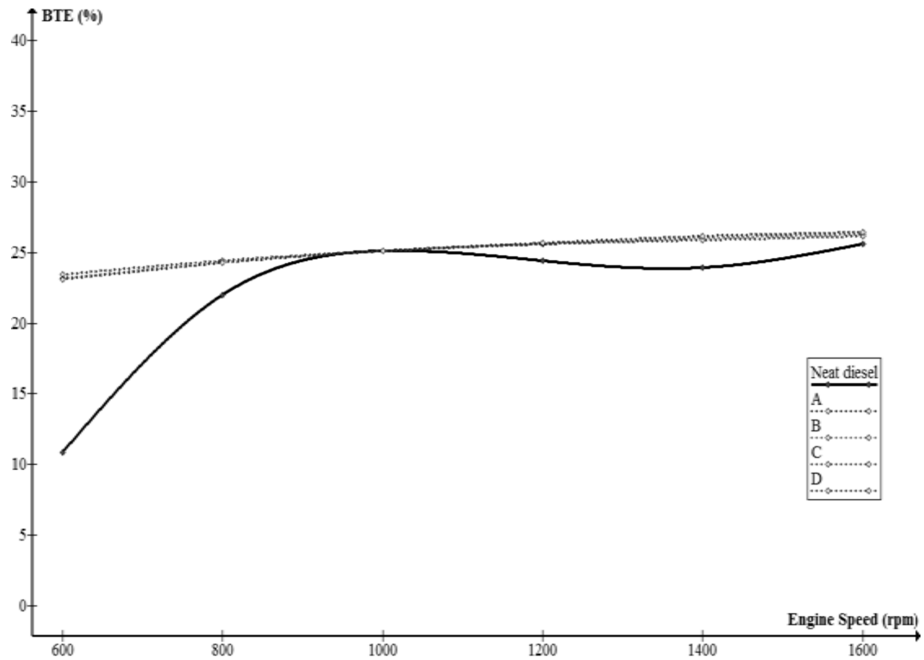

Figure 4.45: Wide Range Ratio EW50–Diesel DI Effect on BTE vs. Engine Speed

4.3.10. Effect of EW75–Diesel DI on Brake Thermal Efficiency

Figure (4.46) shows the wide range ratio EW75–Diesel DI effect on BTE vs. engine speed. BTE values that measured at full load condition in the case of EW75 at different flow rates. BTE increases with increase engine speed for tests Neat Diesel and EW75–Diesel. The improved energy efficiency from these oxygenated fuels was caused by their more complete combustion (a greater fraction of the fuel oxygen supply) and the premixed combustion region that resulted from the ignition delay caused by the lower cetane number of the ethanol. The maximum BTE is 26.5 (%) at 1600 rpm in case (D) condition. According to Neat Diesel BTE values, the highest change is 0% at 1000 rpm in case (D) condition , the lowest change is -110% at 600 rpm in case (D) condition. Diesel engine fueled with EW75–Diesel which was comrade from ethanol and water can effectively suppress NOx emissions by burning oxygenated ethanol. Oxygen-enriched combustion leading to the increase of cylinder temperature was overcome by water endothermic reaction. On the other hand, the extended ignition delay caused by EW75–Diesel can be improved by advancing ignition timing through intake oxygen-enrichment. There should be some complementarities between intake oxygen-enrichment and with appropriate water content. The NOx and CO_2 can be simultaneously reduced without serious penalty in brake specific fuel consumption by careful deployment of oxygen content and water in EW75–Diesel.

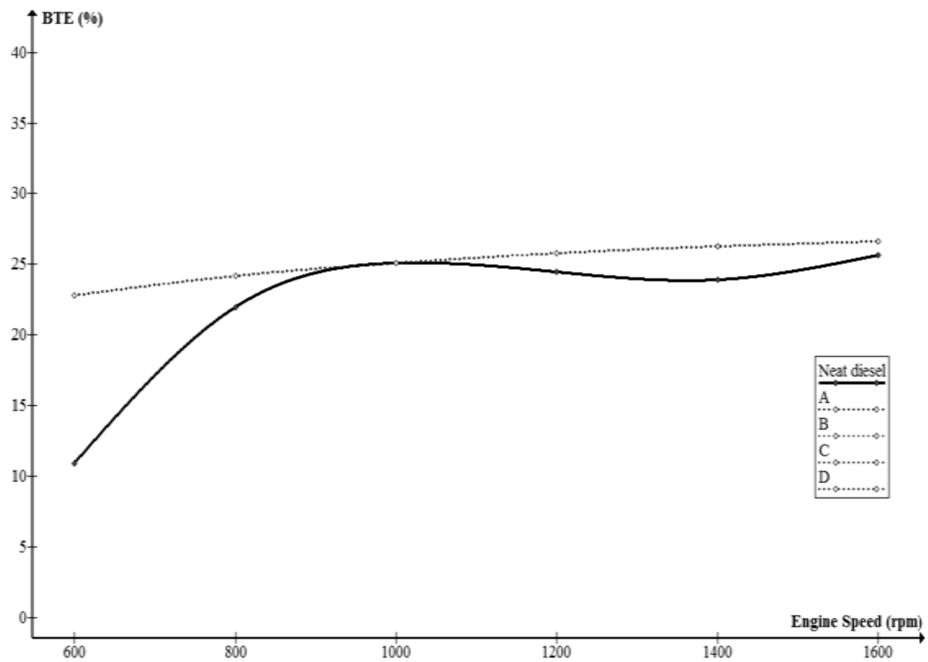

Figure 4.46: Wide Range Ratio EW75–Diesel DI Effect on BTE vs. Engine Speed

4.3.11. Effect of Ethanol–Diesel DI on Brake Thermal Efficiency

Figure (4.47) shows the wide range ratio Ethanol–Diesel DI effect on BTE vs. engine speed. BTE values that measured at full load condition in the case of Ethanol at different flow rates. BTE increases with increase engine speed for tests Neat Diesel and Ethanol–Diesel at case (A, C, and D). BTE decreases with increase engine speed for Ethanol–Diesel at case (B). Ethanol has high laminar flame propagation speed, which may complete the combustion process earlier. This improves engine BTE. Ethanol–Diesel DI is promising alternative transportation fuel because of its properties which allow its utilization in existing diesel engine with minor hardware modifications. Ethanol has high octane ratings. Therefore, higher compression ratios can be achieved before engine starts knocking which ensures more power supply efficiently and economically from engine. The maximum BTE is 30.6 (%) at 600 rpm in case (B) condition. According to Neat Diesel BTE values, the highest change is 14% at 1600 rpm in case (B) condition, the lowest change is -182% at 600 rpm in case (B) condition. The increase of BTE can be explained by attributing the following reasons:

(1) Homogeneous air/ethanol mixture burns faster hence provides more premixed combustion which tends to increase the BTE.

(2) Ethanol has lower cetane number which increases the ignition delay hence energy is released within a very short time, resulting reduction in the heat loss from the engine as there is no sufficient time for transferring heat through the cylinder wall to the coolant.

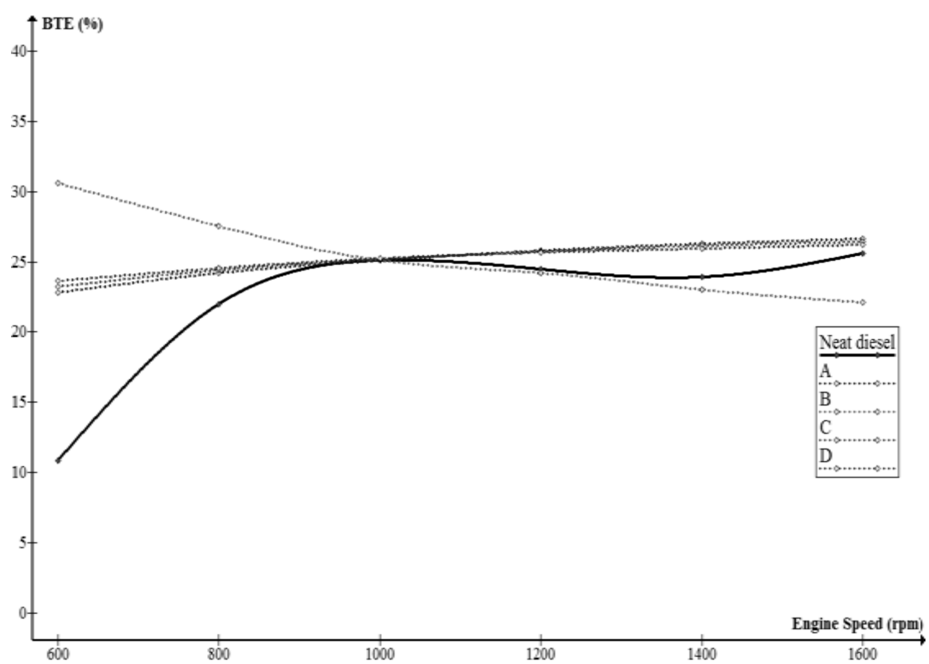

Figure 4.47: Wide Range Ratio Ethanol–Diesel DI Effect on BTE vs. Engine Speed

4.3.12. Effect of Different Dual-Fuelling On Brake Thermal Efficiency at Same Operation Conditions

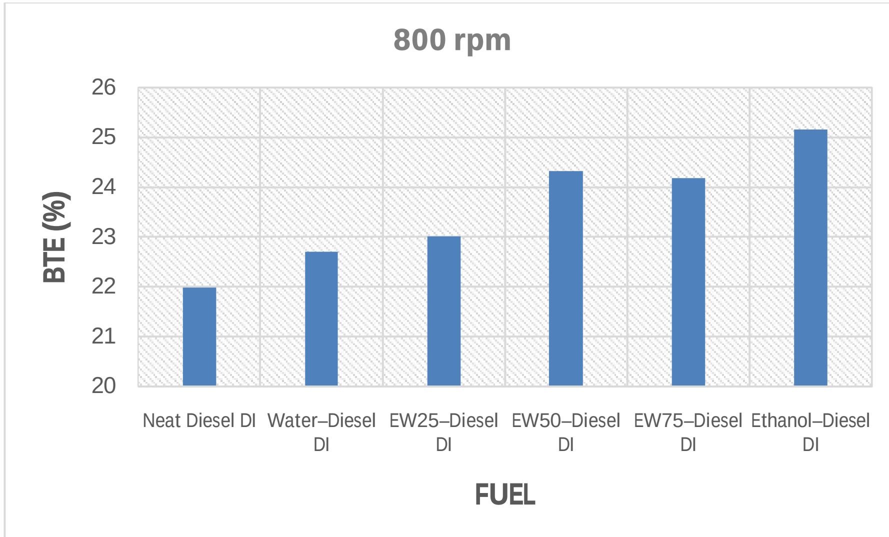

Figure 4.48: BTE vs. Different Dual-Fuelling For Constant Operation Conditions

This method of introduction has the advantage of providing apportion of the total fuel supply premixed with the intake air thus improving air utilization. This method requires minor modification of engine which is done by adding low pressure fuel injector, separate fuel tank, lines and controls but allows a large percentage of Ethanol/Water–Diesel fuels to be used in engine operation since no additives are required for stabilizing the miscibility of Ethanol/Water–Diesel fuel. As a result, the efficiency of engine will be better in DI mode. **Figure (4.48)** shows the BTE vs. different Dual-Fuelling for constant operation conditions. Fixed speed 800 rpm and average results BTE from different flow rates, consider different results BTE for Ethanol/Water–Diesel DI. Water–Diesel DI produced 3% higher BTE compared to Neat Diesel, expressed that combustion process would be much faster, hence effective efficiency increases and adiabatic flame temperature decreases. Heat release rate begins to decrease leading to reduce in-cylinder gas temperature. Thus, NOx emissions reduce due to lower temperature and heat release after that period. It can be concluded that as maximum pressure comparing to Neat Diesel. Effective BTE increased compared to Neat Diesel. EW25–Diesel DI produced 5% higher BTE compared to Neat Diesel, the difficulty of dissolving ethanol in diesel and the stability of blends is influenced by the temperature and water content, especially high percentages of ethanol are used. The use of EW25–Diesel improves the solubility of ethanol in diesel. It was found that the solubility of ethanol in diesel was affected by aromatic hydrocarbons, the temperature of middle distillates and the paraffin content of diesel. EW50–Diesel DI produced 11% higher BTE compared to Neat Diesel, due to the oxygen content in the blend fuels. EW75–Diesel DI produced 10% higher BTE compared to Neat Diesel, which indicates that the improvement of BTE would not continue with increasing additions of ethanol. Usually there is an optimal ratio of ethanol and diesel and the optimal ratio will change according to the concrete combustion system of engines. Ethanol–Diesel DI produced 14% higher BTE compared to Neat Diesel, which promotes the atomization of fuel and the formation of an air–fuel mixture and improves the ignition condition. Additionally, the high oxygen content of ethanol can increase the air–fuel ratio and improve the combustion efficiency.

4.3.13. Effect of Water–Diesel DI on Brake Power

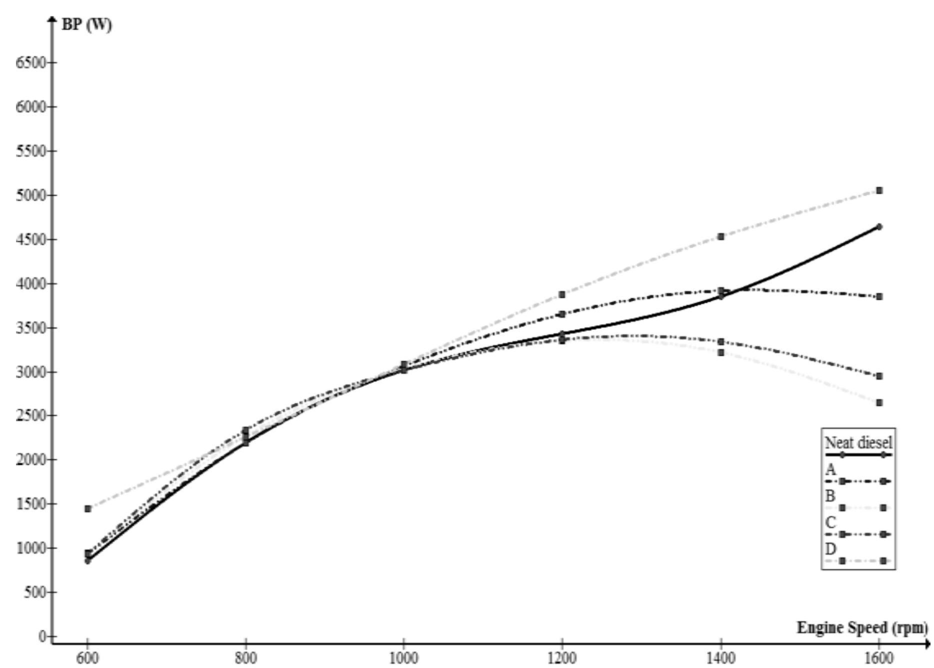

Figure 4.49: Wide Range Ratio Water–Diesel DI Effect on BP vs. Engine Speed

Figure (4.49) shows the wide range ratio Water–Diesel DI effect on BP vs. engine speed. BP values that measured at full load condition in the case of water at different flow rates. BP increases with increase engine speed for tests Neat Diesel and Water–Diesel at case (A, D). Also BP increases with increase engine speed to 1200 rpm after that decreases to full engine speed for Water–Diesel at case (B, C). The test engine consumed Water–Diesel DI in order to retain the same BP. This implies that the Water–Diesel DI has the largest extent of complete combustion to cause the largest BP among the fuel. This occurs due to micro explosion followed by water for which better air/fuel mixing and complete combustion occurs in the combustion chamber. The maximum BP is 5049 (W) at 1600 rpm in case (D) condition. According to Neat Diesel BP values, the highest change is 43% at 1600 rpm in case (B) condition, the lowest change is -67% at 600 rpm in case (D) condition. Finally, on the optimized combustion chamber configuration, when the Water–Diesel is used, the most probable reason to obtain improvement in BP is the reduction of heat losses. The ignition delay period is when the fuel that has been injected into the cylinder is undergoing chemical and physical preparation for combustion. Thus, the Water–Diesel requires less compression (negative) work than the Neat diesel due to the longer ignition delay during the compression stroke. This helps to reach a higher peak pressure after top dead centre (TDC) to produce more power output during the expansion stroke. In addition, when the ignition delay increases, more diesel would be physically prepared (evaporation, mixing) for chemical reaction, which increases the amount of diesel burned and the rate of heat release in the premixed burning. This results in enhancement of combustion and improvement of combustion efficiency. As a result of these changes, cylinder pressures and temperatures are lower.

4.3.14. Effect of EW25–Diesel DI on Brake Power

Figure (4.50) shows the wide range ratio EW25–Diesel DI effect on BP vs. engine speed. BP values that measured at full load condition in the case of EW25 at different flow rates. BP increases with increase engine speed for tests Neat Diesel and EW25–Diesel. The operation EW25–Diesel much better fuel conversion efficiencies both full and part engine speed, originating from the bulk, lean combustion and control by quantity of fuel injected paid at the price of a reduced power density. The EW25–Diesel DI also permits to explore the operation mixed two injection events to modulate the pre-mixed and the diffusion combustion phases and achieve better fuel conversion efficiencies or better power densities. The maximum BP is 5053 (W) at 1600 rpm in case (D) condition. According to Neat Diesel BP values, the highest change is 2% at 600 rpm in case (A) condition, the lowest change is -110% at 600 rpm in case (B) condition. This could be due to better mixing of air and EW25 and improved combustion efficiency.

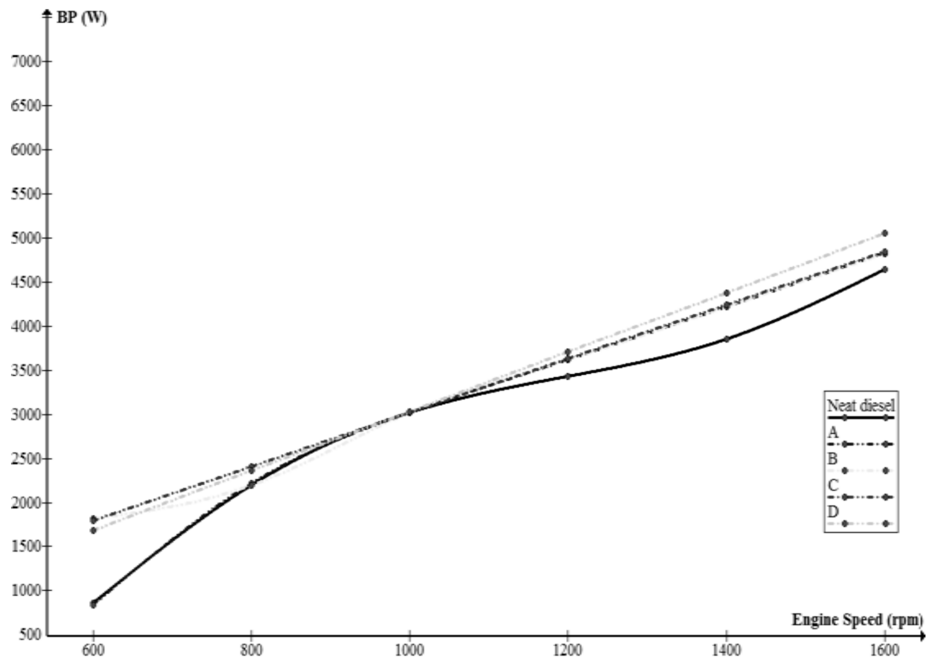

Figure 4.50: Wide Range Ratio EW25–Diesel DI Effect on BP vs. Engine Speed

4.3.15. Effect of EW50–Diesel DI on Brake Power

Figure (4.51) shows the wide range ratio EW50–Diesel DI effect on BP vs. engine speed. BP values that measured at full load condition in the case of EW50 at different flow rates. BP increases with increase engine speed for tests Neat Diesel and EW50–Diesel. Due to higher thermodynamics conditions, 1000 rpm operating points are less sensitive to the formulation (flash point improver) and fuel ignitability compare to the lower and higher speeds operating points. The maximum BP is 4792 (W) at 1600 rpm in case (D) condition. According to Neat Diesel BP values, the highest change is 0% at 1000 rpm in case (B) condition, the lowest change is -116% at 600 rpm in case (C) condition. This better BP obtained for all EW50–Diesel fuels could be explained by the high oxygen content which maintains a fast end of the combustion in rich condition and low propensity of these fuels to emit smoke, as discussed previously.

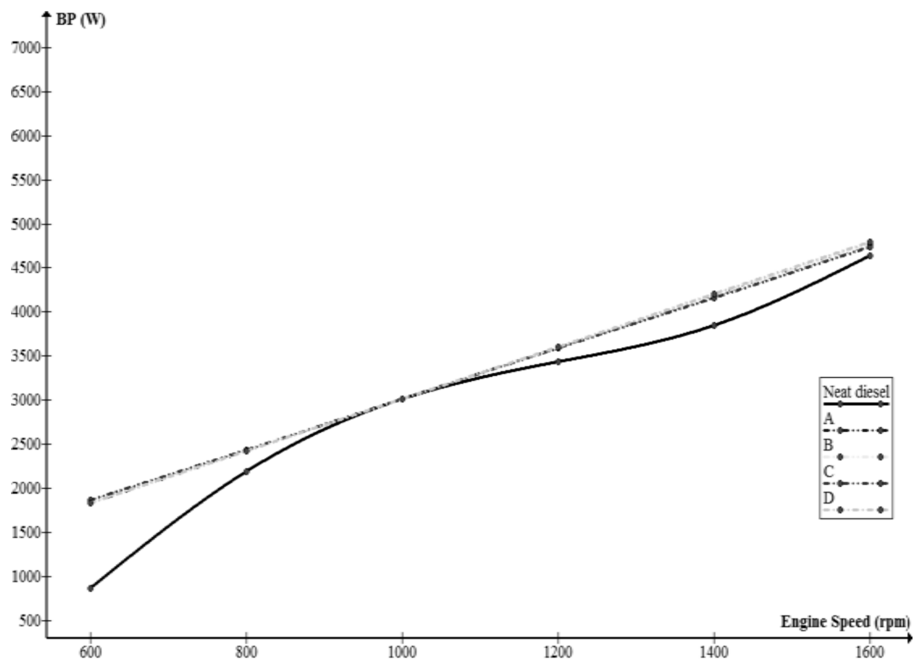

Figure 4.51: Wide Range Ratio EW50–Diesel DI Effect on BP vs. Engine Speed

4.3.16. Effect of EW75–Diesel DI on Brake Power

Figure (4.52) shows the wide range ratio EW75–Diesel DI effect on BP vs. engine speed. BP values that measured at full load condition in the case of EW75 at different flow rates. BP increases with increase engine speed for tests Neat Diesel and EW75–Diesel. One way to attest the EW75–Diesel for each combustion stages of ignition delay, premixed combustion, and late-cycle combustion is calculating the burn durations. For affixed combustion phasing, shorter burn duration corresponds to a higher average rate of heat release resulting in increased BP. The maximum BP is 4826 (W) at 1600 rpm in case (D) condition. According to Neat Diesel BP values, the highest change is 0% at 1000 rpm in case (D) condition, the lowest change is -

110% at 600 rpm in case (D) condition. It is suggested that the observed trend of increasing BP with increasing ethanol energy fraction is not a result of the combustion phasing that does not change with the ethanol energy fraction but the faster burning of the premixed EW75–Diesel–air mixture that increases the rate of heat release.

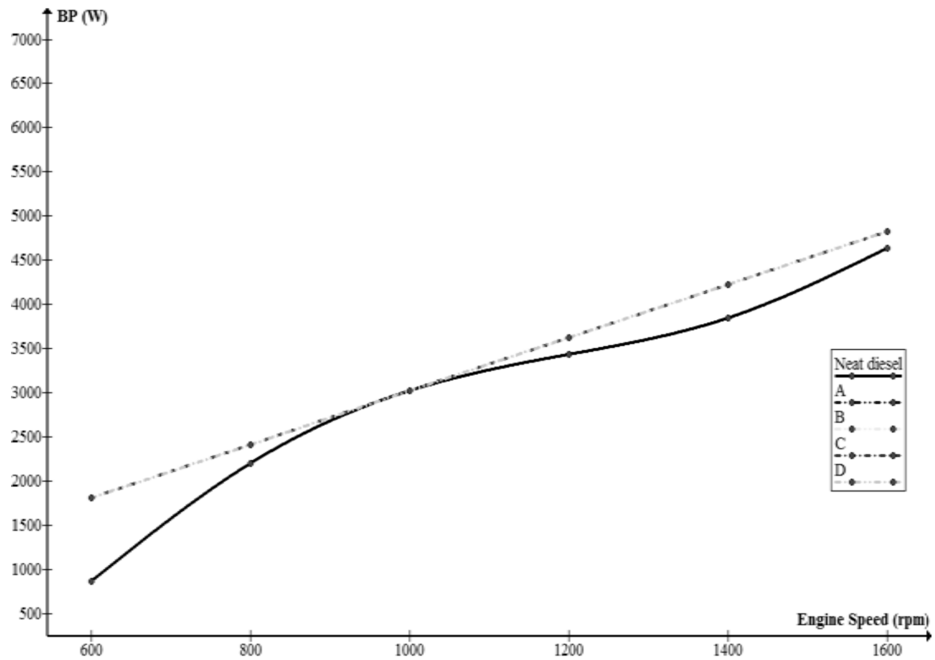

Figure 4.52: Wide Range Ratio EW75–Diesel DI Effect on BP vs. Engine Speed

4.3.17. Effect of Ethanol–Diesel DI on Brake Power

Figure (4.53) shows the wide range ratio Ethanol–Diesel DI effect on BP vs. engine speed. BP values that measured at full load condition in the case of Ethanol at different flow rates. BP increases with increase engine speed for tests Neat Diesel and Ethanol–Diesel. Benefits such as higher BP and may be realized with ethanol their high octane number gives the ability to operate at higher compression ratio without pre-ignition; their greater latent heat of vaporization gives a higher charge density; and their higher laminar flame speed allows them to be run with leaner, or more dilute, air/fuel mixtures. The maximum BP is 4825 (W) at 1600 rpm in case (A) condition. According to Neat Diesel BP values, the highest change is 14% at

1600 rpm in case (B) condition, the lowest change is -182% at 600 rpm in case (B) condition. Diesel fuelling shows an ignition delay trend, which decreases with increasing speed, as expected. This occurs as a result of the increasing combustion chamber temperature as the speed is increased. The ignition delay for Ethanol–Diesel fuelling is shorter than that of Neat Diesel fuelling. As indicated above, ethanol releases energy before top dead centre, with the result that ethanol is injected in a high temperature environment, thus evaporating rapidly. At the higher speeds, the combustion chamber temperature increases with the result that as Ethanol, it burns at the intake valve. This results in a zero ignition delay.

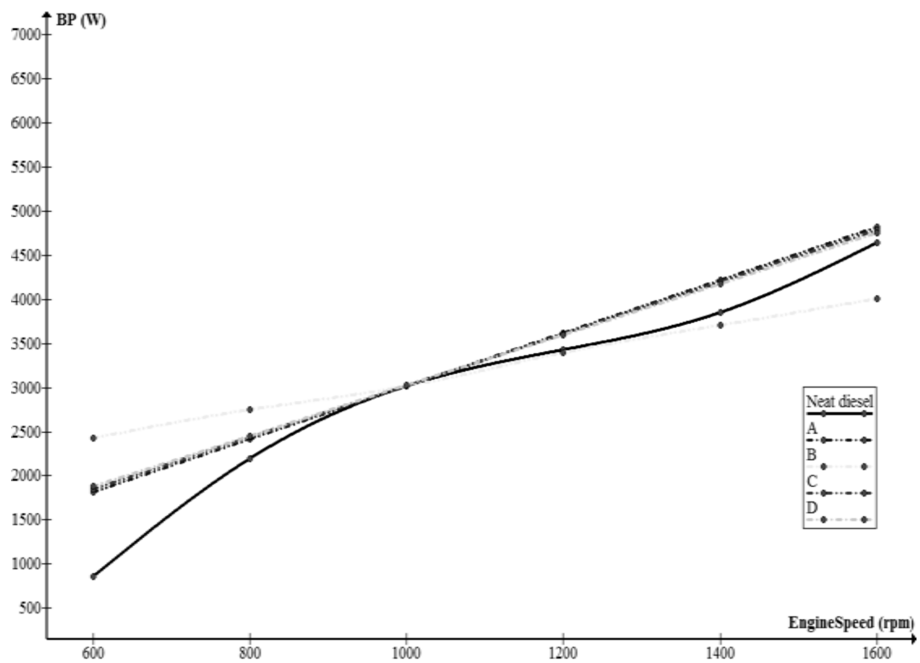

Figure 4.53: Wide Range Ratio Ethanol–Diesel DI Effect on BP vs. Engine Speed

4.3.18. Effect of Different Dual-Fuelling on Brake Power at Same Operation Conditions

The diesel engine is one of the internal combustion engines that can convert the chemical energy of fuel into mechanical work. **Figure (4.54)** shows the BP vs. different Dual-Fuelling for constant operation conditions. Fixed speed 800 rpm and average results BP from different flow rates, consider different results BP for Ethanol/Water–Diesel.

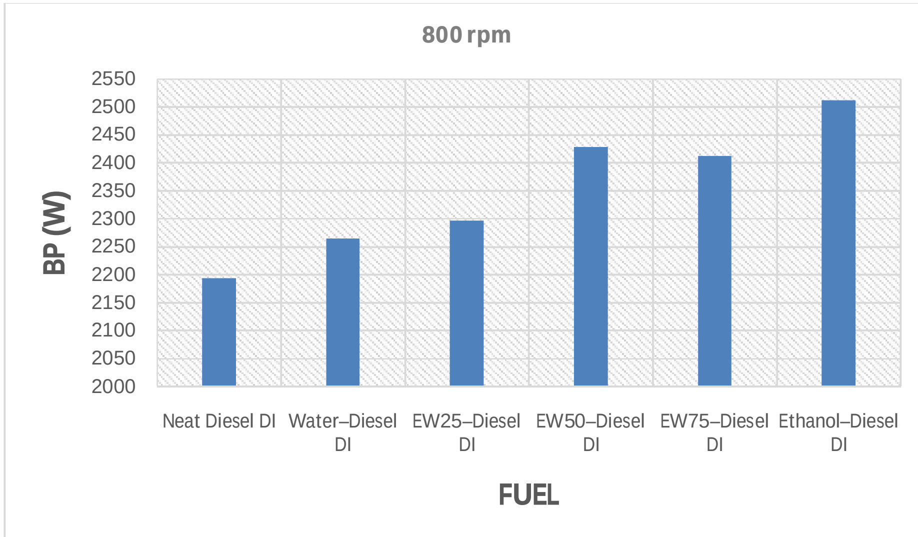

Figure 4.54: BP vs. Different Dual-Fuelling For Constant Operation Conditions

Water–Diesel DI produced 3% higher BP compared to Neat Diesel, due to the faster combustion of the Water–Diesel, a prerequisite to explain the previous observations, can be explained by the improvement in the spray atomization due to the water drops micro-explosions. EW25–Diesel DI produced 5% higher BP compared to Neat Diesel, showed that higher oxygen content in the spray (from oxygen enriched air or oxygenated fuel) reduced pyrolysis and increased oxidation, thus shortening the combustion duration. EW50–Diesel DI produced 11% higher BP compared to Neat Diesel, the complete homogeneity of the fuel lead to complete combustion and increased BP. These results occurred due to the EW50 cooling effect and from misfiring. EW75–Diesel DI produced 10% higher BP compared to Neat Diesel, can be explained on the basis of probable enhancing of EW75 the combustion process. As stated above, EW75 and air mixing could be improved by additional gas motions, which occurs by instantaneous burning of EW75–air mixtures. Thus, combustion process would get better BP which could be improved. In addition, EW75 that is introduced into the intake manifold partly vaporizes and cools the air during intake stroke. Ethanol–Diesel DI produced 14% higher BP compared to Neat Diesel, shows an increasing trend of the apparent heat release rate or properly positioned combustion phasing. The conversion of heat energy to mechanical work increases with rise in combustion temperature and that leads increase of BP with respect to ethanol.

4.3.19. Effect of Water–Diesel DI on Volumetric Efficiency

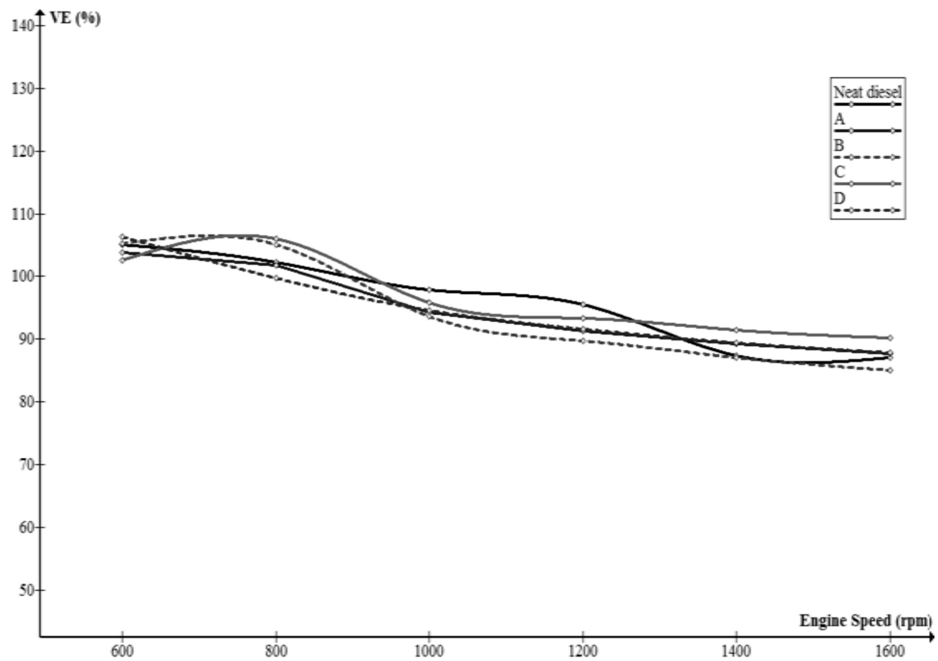

Figure 4.55: Wide Range Ratio Water–Diesel DI Effect on VE vs. Engine Speed

Figure (4.55) shows the wide range ratio Water–Diesel DI effect on VE vs. engine speed. VE values that measured at full load condition in the case of water at different flow rates. VE decreases with increase engine speed for tests Neat Diesel and Water–Diesel. Artificially improving the VE increases the amount of air that the diesel engine has available for combustion, which increases the power output of the engine. The advantages of injection are versatile of on-line variation of water quantity, increase of VE due to cooling effect, uniform or homogeneous water distribution in combustion chamber. The maximum VE is 106 (%) at 600 rpm in case (D) condition. According to Neat Diesel VE values, the highest change is 6% at

1200 rpm in case (B) condition, the lowest change is -5% at 1400 rpm in case (C) condition. Neat diesel has high VE compared to all other Water–Diesel at medium engine speeds. This is due to higher EGT released after the combustion process.

4.3.20. Effect of EW25–Diesel DI on Volumetric Efficiency

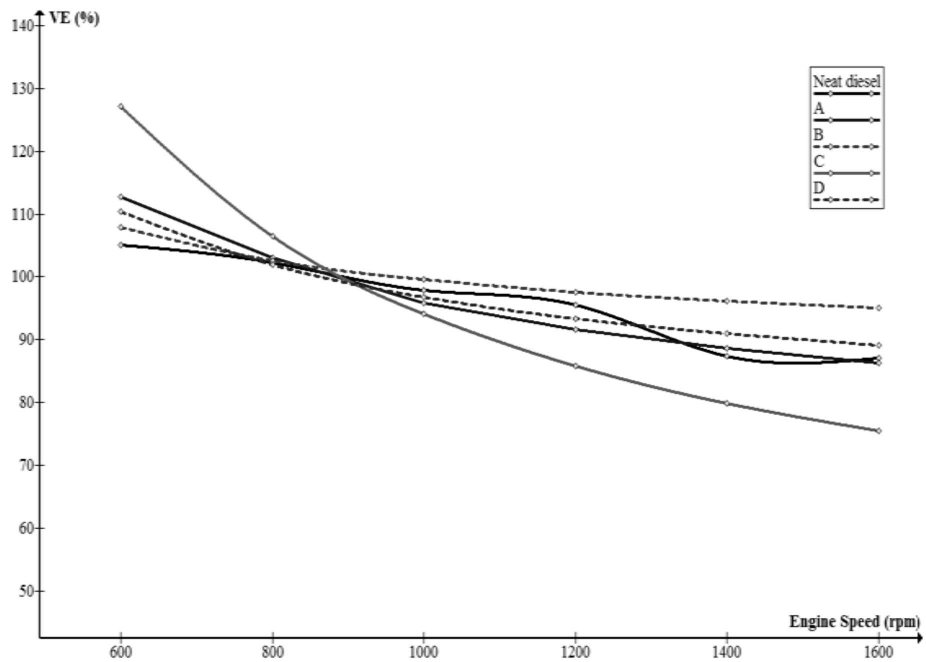

Figure 4.56: Wide Range Ratio EW25–Diesel DI Effect on VE vs. Engine Speed

Figure (4.56) shows the wide range ratio EW25–Diesel DI effect on VE vs. engine speed. VE values that measured at full load condition in the case of EW25 at different flow rates. VE decreases with increase engine speed for tests Neat Diesel and EW25–Diesel. This result due to increase incomplete combustion, increase in mechanical frictional losses and reduction in VE. The maximum VE is 127 (%) at 600 rpm in case (C) condition. According to Neat Diesel VE values, the highest change is 13% at 1600 rpm in case (C) condition, the lowest change is -21% at 600 rpm in case (C) condition. This is in line with the increase in CO_2 at the higher engine speed associated with deteriorated combustion and reduced VE. EW25–Diesel DI of has lower VE compared to Neat Diesel for range (900- 1300) rpm at case (A, D) and range (900- 1600) rpm at case (C). A high-retained EGT will heat the incoming fresh air and lowers the VE.

4.3.21. Effect of EW50–Diesel DI on Volumetric Efficiency

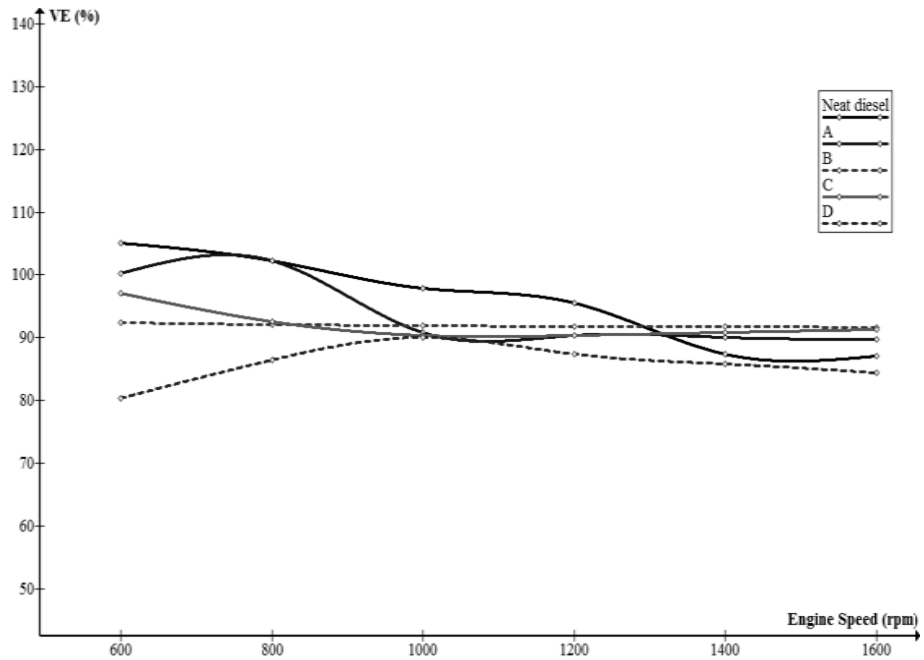

Figure 4.57: Wide Range Ratio EW50–Diesel DI Effect on VE vs. Engine Speed

Figure (4.57) shows the wide range ratio EW50–Diesel DI effect on VE vs. engine speed. VE values that measured at full load condition in the case of EW50 at different flow rates. VE decreases with increase engine speed for tests Neat Diesel and EW50–Diesel at case (A, B, and C). Also VE increases with increase engine speed for tests EW50–Diesel at case (D). A lower exhaust temperature leads to a higher VE. The maximum VE is 105 (%) at 600 rpm in Neat Diesel condition. According to Neat Diesel VE values, the highest change is 24% at 600 rpm in case (D) condition, the lowest change is -5% at 1600 rpm in case (B) condition. Increase heat release with the EW50–Diesel causes an increase in temperature of the combustion chamber walls of diesel engine. The VE drops in the EW50–Diesel condition at engine speed less than 1300 rpm compared to Neat Diesel condition, due to their different content. Reduction in the VE was occurred in EW50–Diesel because of the higher cylinder temperature.

4.3.22. Effect of EW75–Diesel DI on Volumetric Efficiency

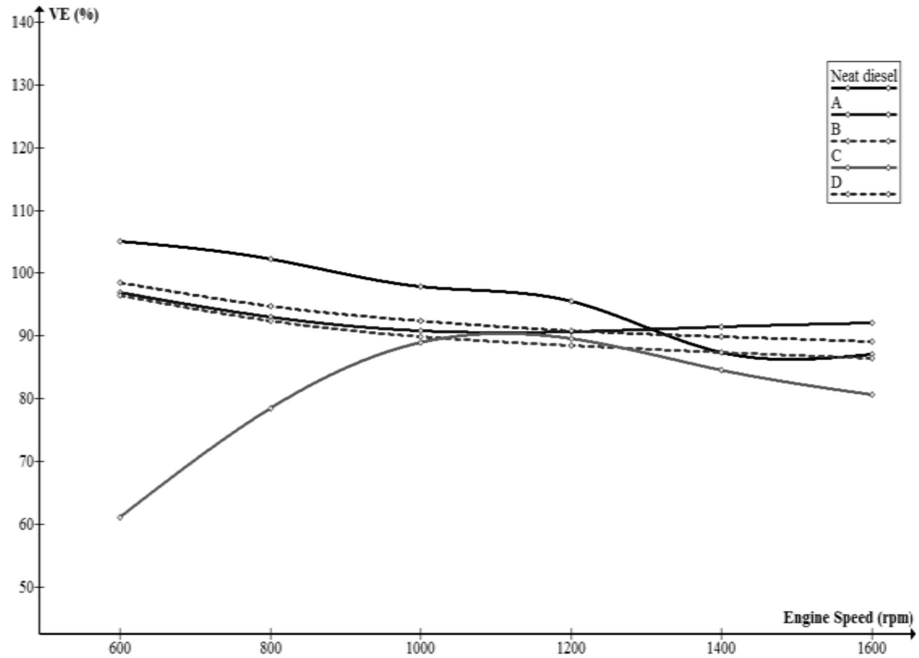

Figure 4.58: Wide Range Ratio EW75–Diesel DI Effect on VE vs. Engine Speed

Figure (4.58) shows the wide range ratio EW75–Diesel DI effect on VE vs. engine speed. VE values that measured at full load condition in the case of EW75 at different flow rates. VE decreases with increase engine speed for tests Neat Diesel and EW75–Diesel at case (A, B, and D). Also VE increases with increase engine speed for tests EW75–Diesel at case (C). The VE drops in the EW75–Diesel condition at engine speed less than 1300 rpm compared to Neat Diesel condition; this due to mass of EW75 into the fresh mixture thus will decrease the VE. The maximum VE is 105 (%) at 600 rpm in Neat Diesel condition. According to Neat Diesel VE values, the highest change is 42% at 600 rpm in case (C) condition, the lowest change is -6% at 1600 rpm in case (A) condition. In the operating zone represented in the map, the EW75 mass flow is controlled by means of the chemical pump. For a given speed the engine is able to admit a given volume of intake gas. Since the intake mixing contains both fresh air and EW75, the higher the EW75 the lower the air mass flow admitted. EW75 contributes to an increase in the combustion chamber temperature, in turn lowering the intake mass flow.

4.3.23. Effect of Ethanol–Diesel DI on Volumetric Efficiency

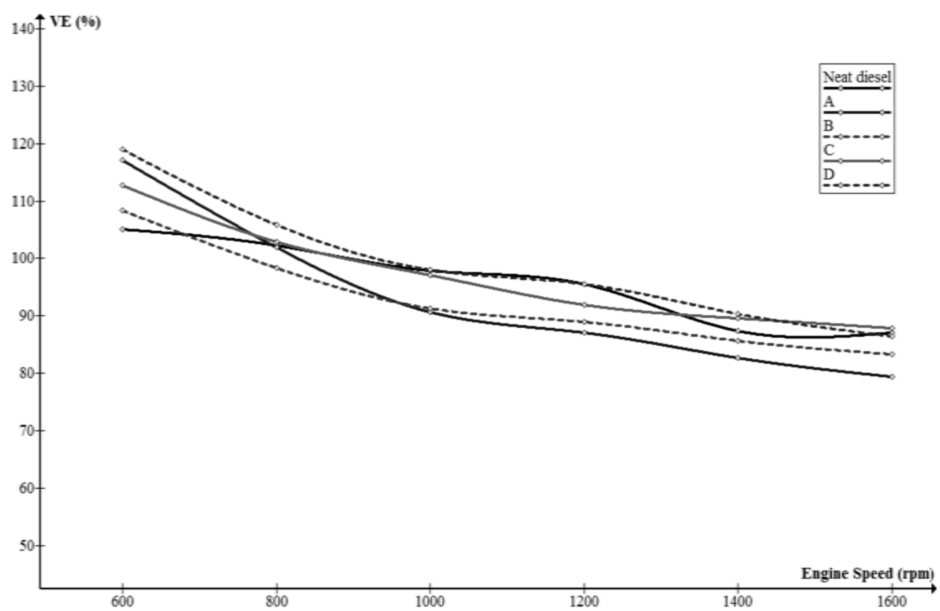

Figure 4.59: Wide Range Ratio Ethanol–Diesel DI Effect on VE vs. Engine Speed

Figure (4.59) shows the wide range ratio Ethanol–Diesel DI effect on VE vs. engine speed. VE values that measured at full load condition in the case of Ethanol at different flow rates. VE decreases with increase engine speed for tests Neat Diesel and Ethanol–Diesel. The in cylinder combustion temperature is higher in Ethanol–Diesel DI due to the higher heating value, that will be transferred to the engine parts, so the intake are temperature increases and decreases the VE. The maximum VE is 119 (%) at 600 rpm in case (D) condition. According to Neat Diesel VE values, the highest change is 9% at 1600 rpm in case (A) condition, the lowest change is -13% at 600 rpm in case (D) condition. Since ethanol has higher heat of vaporization, which results in cooling effect in the intake process and compression stroke. As a result the VE of the engine is increased and the required amount of the work input is reduced in the compression stroke. Also due to higher operating temperatures, with insulated components, the intake air is heated to a higher temperature and consequently the mass of air drawn in each cycle is lower, resulting in a decrease in VE.

4.3.24. Effect of Different Dual-Fuelling On Volumetric Efficiency at Same Operation Conditions

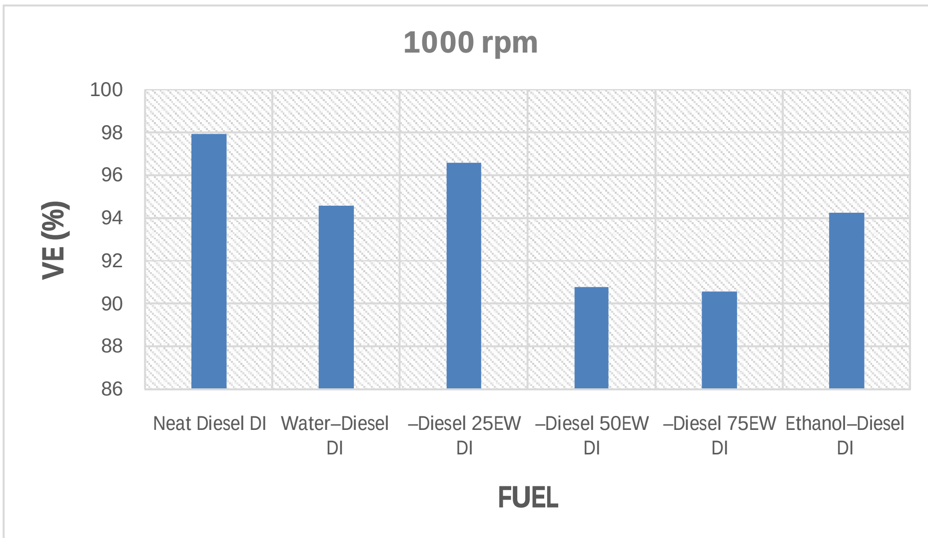

Figure 4.60: VE vs. Different Dual-Fuelling For Constant Operation Conditions

VE is an indication of breathing ability of the engine. It depends on the ambient conditions and operating conditions of the engine. **Figure (4.60)** shows the VE vs. different Dual-Fuelling for constant operation conditions. Fixed speed 1000 rpm and average results VE from different flow rates, consider different results VE for Ethanol/Water–Diesel. Water–Diesel DI produced 3% lower VE compared to Neat Diesel. This decrease in the VE is attributed to the decrease in the density of air entering the cylinder because of replacement amount air by water. The degree of degradation of VE depends on the degree of flow rate water. EW25–Diesel DI produced 1% lower VE compared to Neat Diesel. This is due to increase of gas temperature with the EW25. These are due to reduction of VE and the relatively lower turbulent flame speed of Water–Diesel-air combustion. EW50–Diesel DI produced 7% lower VE compared to Neat Diesel. Depending on the overall air-fuel ratio, the mixture can be homogeneous-stoichiometric or homogeneous lean. Early injection makes it possible to achieve a VE that is higher than EW50–Diesel. EW75–Diesel DI produced 8% lower VE compared to Neat Diesel. Reason for this is that the inner surface of engine cylinder is hot which makes the residual gases and fresh air to expand more thus reducing the flow rate of incoming air. Ethanol–Diesel DI produced 4% lower VE compared to Neat Diesel. This is due to higher EGT released after the combustion process. The VE has been depending and some other possible dentitions are discussed. The dentition used in this thesis includes mass of Ethanol/Water–Diesel DI in the same way as mass of fresh mixture. An experimental of how the heat transfer can be described as the total increase of temperature is discussed and how this will affect the VE. The VE is also affected by the amount of residual gas trapped in the combustion chamber. To understand VE it is good to have a sketch over the complete air path. This has been described and divided into three different parts: Inlet, Volumetric and Exhaust.

4.3.25. Effect of Water–Diesel DI on Excess Air Coefficient

Figure (4.61) shows the wide range ratio Water–Diesel DI effect on (λ) vs. engine speed. (λ) Values that measured at full load condition in the case of Water at different flow rates. As engine speed increases as (λ) decrease to 1000 rpm, after that increase to 1400 rpm, then decreases to maximum engine speed for all condition tests. The addition of Water causes the lower temperature of the combustion compared to Neat Diesel, which decreased vaporization and atomization in mixing with air and leads to not complete combustion. In addition, with the use of Water–Diesel increase λ. The minimum (λ) is 0.36 at 1000 rpm in case Neat Diesel condition. According to Neat Diesel (λ) values, the highest change is 20% at 600 rpm in case (A) condition, the lowest change is -574% at 1000 rpm in case (C) condition. The longer delay period in Water–Diesel operation is due to the cooling effect of water on the intake air temperature contributes toward lower charge temperature at the time of fuel injection. Water droplet in the cylinder charge retards the combustion process of the charge entrained into the burning diesel and affects its turbulent mixing resulting in extended combustion duration. This is in a good agreement with Arrhenius function stated that ignition delay is strongly dependent on the intake charge temperature. It is concluded that the delay period increases with the increase of water injection duration for all cases.

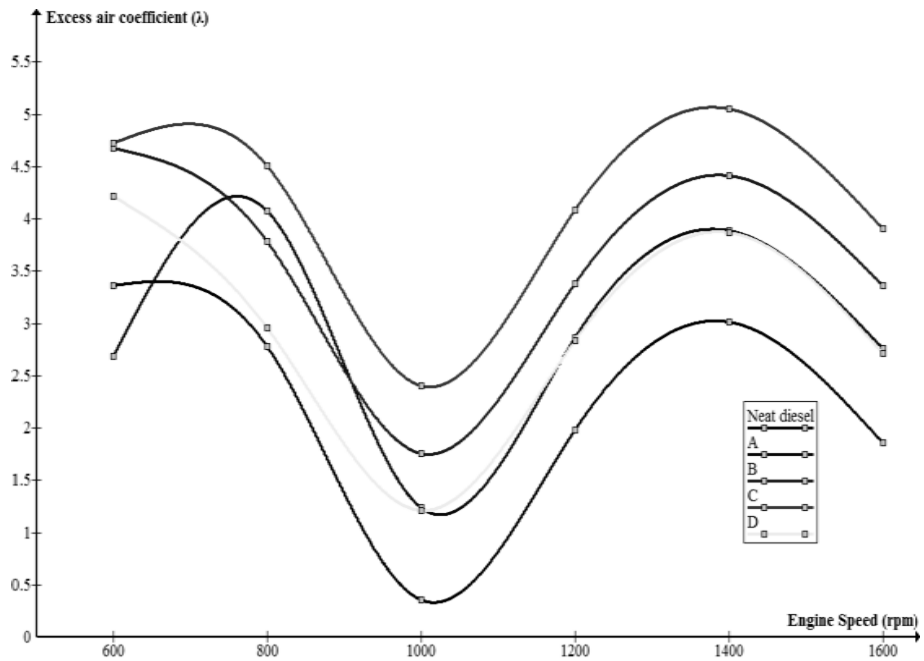

Figure 4.61: Wide Range Ratio Water–Diesel DI Effect on (λ) Vs. Engine Speed

4.3.26. Effect of EW25–Diesel DI on Excess Air Coefficient

Figure (4.62) shows the Wide Range Ratio EW25–Diesel DI effect on (λ) vs. engine speed. (λ) Values that measured at full load condition in the case of EW25 at different flow rates. As engine speed increases as (λ) decrease to 1000 rpm, after that increase to 1400 rpm, then decreases to maximum engine speed for all condition tests. This means that the influence of lower cetane number of ethanol on the ignition delay is inferior at medium engine speed. On the other hand, the addition of ethanol leads to decrease in the viscosity of the blended fuel which is positive to form more air–fuel mixture and result in a larger percentage of fuel burned in the premixed burning phase. The minimum (λ) is 0.36 at 1000 rpm in case

Neat Diesel condition. According to Neat Diesel (λ) values, the highest change is -25% at 800 rpm in case (A) condition, the lowest change is -441% at 1000 rpm in case (D) condition. This is results some advantages with these fuels such as improved atomization, better air/fuel mixing and enhanced engine characteristics, when used in a diesel engine without any modifications. Nonetheless, due to the limitations imposed on EW25–Diesel fuels, mixing lower proportions of them with diesel has been recommended.

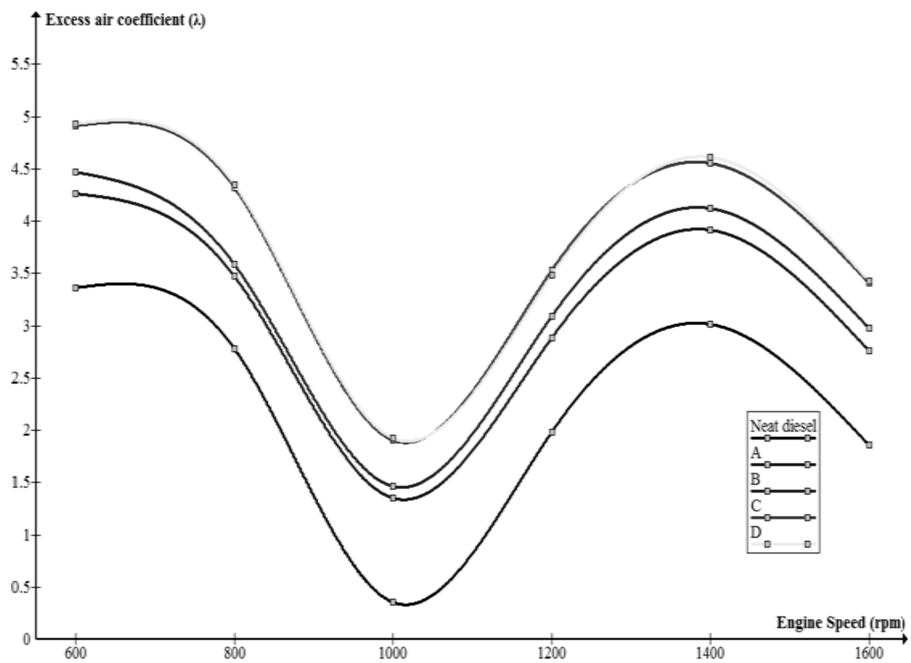

Figure 4.62: Wide Range Ratio EW25–Diesel DI Effect on (λ) vs. Engine Speed

4.3.27. Effect of EW50–Diesel DI on Excess Air Coefficient

Figure (4.63) shows the wide range ratio EW50–Diesel DI effect on (λ) vs. engine speed. (λ) Values that measured at full load condition in the case of EW50 at different flow rates. As engine speed increases as (λ) decrease to 1000 rpm, after that increase to 1400 rpm, then decreases to maximum engine speed for all condition tests. Therefore, the (λ) EW50–Diesel is more than that of Neat Diesel. Although the use of EW50 may have a little effect on intake air mass flow, the (λ) diesel fuel plays a leading role on the AFR in this case. This result can provide a prerequisite for the comparisons between the Neat Diesel and EW50–Diesel. The minimum (λ) is 0.36 at 1000 rpm in case Neat Diesel condition. According to Neat Diesel (λ) values, the highest change is -49% at 600 rpm in case (D) condition, the lowest change is -523% at 1000 rpm in case (C) condition. In the internal combustion engine, (λ) is one of the important economic performance parameters of a fuel because maximum engine performance and minimum amount of exhaust toxic gases can be only obtained in special range of (λ). If the appropriate ratio of air-fuel mixture is not produced, not only a partly of fuel energy will be wasted because of insufficient combustion, but also exhaust toxic emission will be increased.

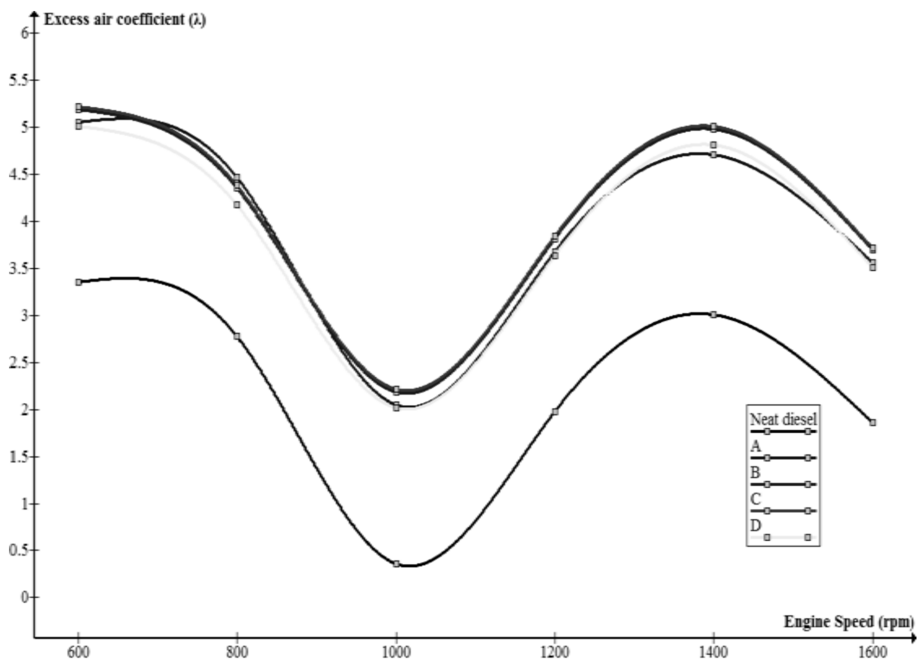

Figure 4.63: Wide Range Ratio EW50–Diesel DI Effect on (λ) vs. Engine Speed

4.3.28. Effect of EW75–Diesel DI on Excess Air Coefficient

Figure (4.64) shows the wide range ratio EW75–Diesel DI effect on (λ) vs. engine speed. (λ) Values that measured at full load condition in the case of EW75 at different flow rates. As engine speed increases as (λ) decrease to 1000 rpm, after that increase to 1400 rpm, then decreases to maximum engine speed for all condition tests. Showed increase in (λ) with increasing EW75–Diesel DI, which is again an indication towards incomplete combustion of the charge inside the cylinder due to absence of sufficient air as EW75 owing to its indirect injection, replaces a part of the air intake. Further absence of excess air due to EW75 replacing intake air also aids in lower NOx formation. The minimum (λ) is 0.36 at 1000 rpm in case Neat Diesel condition. According to Neat Diesel (λ) values, the highest change is -34% at 600 rpm in case (B) condition, the lowest change is -438% at 1000 rpm in case (D) condition. From the perspective of oxygen availability in the ethanol, this helps form lean combustion in the cylinder compared with conventional diesel combustion. As the ethanol content increases, the air excess ratio of the combustion increases with the blend ratio.

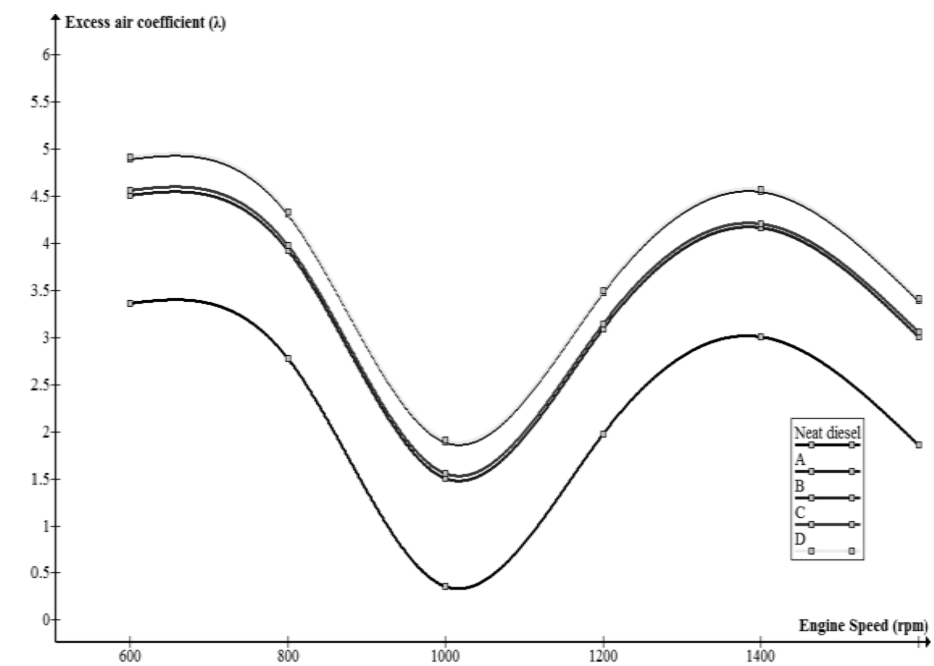

Figure 4.64: Wide Range Ratio EW75–Diesel DI Effect on (λ) vs. Engine Speed

4.3.29. Effect of Ethanol–Diesel DI on Excess Air Coefficient

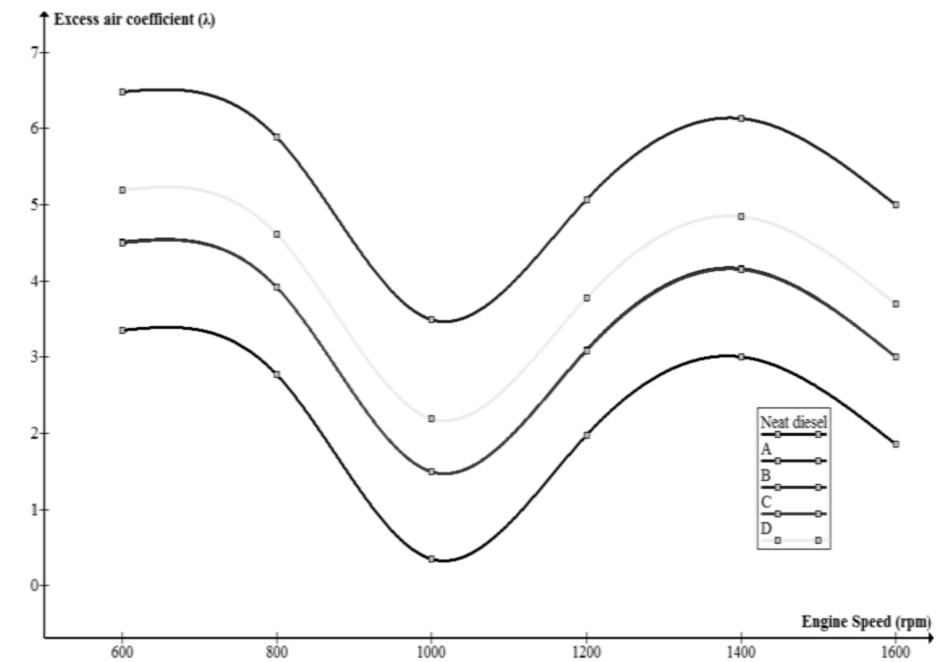

Figure 4.65: Wide Range Ratio Ethanol–Diesel DI Effect on (λ) vs. Engine Speed

Figure (4.65) shows the wide range ratio Ethanol–Diesel DI effect on (λ) vs. engine speed. (λ) Values that measured at full load condition in the case of Ethanol at different flow rates. As engine speed increases as (λ) decrease to 1000 rpm, after that increase to 1400 rpm, then decreases to maximum engine speed for all condition tests, due adding ethanol to fresh air entering diesel engine, appears to be a more beneficial way of utilizing ethanol. These properties strongly affect injection characteristics, air–fuel mixing characteristics and thereby combustion characteristics of ethanol in a diesel engine. The increased (λ) at high and lower engine speeds can be explained by the reasons that the use of ethanol is effectively introduced to the fuel-rich regions and suppress emissions formation in combustion chamber. The charge cooling increases ignition delay and thus, enhances the mixing of fuel with air which in turn makes better air utilization. The high oxygen content of the ethanol combined with low C/H ratio and aromatic fractions contributes to the increase of NOx. The minimum (λ) is 0.36 at 1000 rpm in case Neat Diesel condition. According to Neat Diesel (λ) values, the highest change is -34% at 600 rpm in case (A) condition, the lowest change is -882% at 1000 rpm in case (B) condition. High level of oxygen atoms present in the fuel also results in overall leaner mixture. All these factors result in overall increases in (λ). The difference in (λ) value at each operation experimental for all tested fuels was very slight, so that differences in emission results were caused by different characteristics and composition of tested fuels, but not because of differences in fuel/air equivalence ratio.

4.3.30. Effect of Different Dual-Fuelling on Excess Air Coefficient at Same Operation Conditions

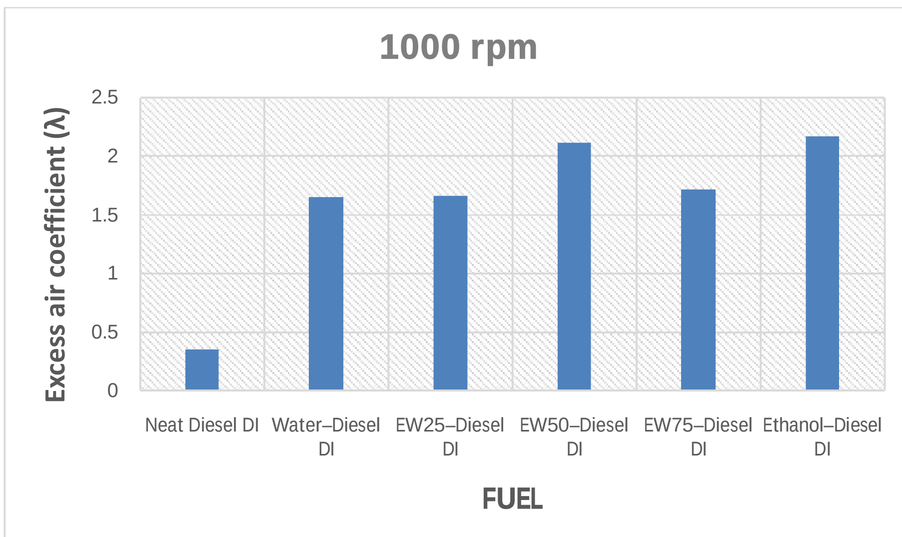

Figure 4.66: (λ) vs. Different Dual-Fuelling for Constant Operation Conditions

Fuel to air equivalence ratios (fuel to air ratio divided by its stoichiometric value) based on the evaporated fuel for the Neat Diesel fuel and Ethanol/Water–Diesel DI. (λ) Value is derived from the stoichiometric air/fuel ratio. Using this definition means that the mixture ratio, $\lambda < 1.0$ for fuel-rich flames and $\lambda > 1.0$ for fuel-lean flames. **Figure (4.66)** shows the (λ) vs. different Dual-Fuelling for constant operation conditions. Fixed speed 1000 rpm and average results (λ) from different flow rates, consider different results (λ) for Ethanol/Water–Diesel DI. Water–Diesel DI produced 346% higher (λ) compared to Neat Diesel, it was believed that lowering distillation characteristics improved atomization and dispersion of fuel spray, and that fast evaporation of lighter fuel accelerated the fuel mixing with air. When the engine is operation with water, the fuel injection quantity of the engine is less, and the temperature within the cylinder is lower, and it is helpful (λ) is high. EW25–Diesel DI produced 367% higher (λ) compared to Neat Diesel, significantly higher stoichiometric fuel/air ratio, accordingly with its oxygen content, leading to increases relative fuel/air ratio. General, higher excess of air (with respect to stoichiometric air) leads to cleaner combustion. EW50–Diesel DI produced 495% higher (λ) compared to Neat Diesel, due for ethanol, lower inlet air temperature was required in order to operate the engine and it was due to the higher enthalpy of vaporization of ethanol as compared to Neat Diesel. Overall, ethanol shows better combustion characteristics due to its lower enthalpy of vaporization and auto-ignition temperature. EW75–Diesel DI produced 383% higher (λ) compared to Neat Diesel. Higher (λ) for EW25–Diesel are potentially caused by various interactive effects. On one hand, lower cetane number of EW75 causes longer ignition delay, which provides more time for fuel to evaporate and mix with air leading to a more homogenous air fuel mixture. On the other hand, higher heat of evaporation of EW75 results in slower evaporation, and hence slower fuel–air mixing. Ethanol–Diesel DI produced 510% higher (λ) compared to Neat Diesel, due to the lower energy content from the ethanol fuels. This causes changes in the value of (λ) for the different fuel blends used. In a direct injection diesel engine, the fuel–air distribution is not homogeneous. This means that (λ) is controlled by mixing of air and fuel besides excess air ratio. In the premixed part of combustion the fuel that was vaporized and mixed with fresh air during the ignition delay interval is burned, because of the large amount of excess air.

CHAPTER FIVE

Computer Simulation

5. Computer Simulation of Single-Cylinder Engine for Diesel and Ethanol/Water Mixture Fuels

5.1. Preface For DIESEL-RK Software

The development of the single cylinder modeling in one-dimensional simulation for four-stroke direct-injection (DI) diesel engine was presented in this thesis. The details of the engine parameters used in this model are described in **Table (3.1)**. **Figure (5.1)** shows the diesel engine modeling using program DIESEL-RK software.

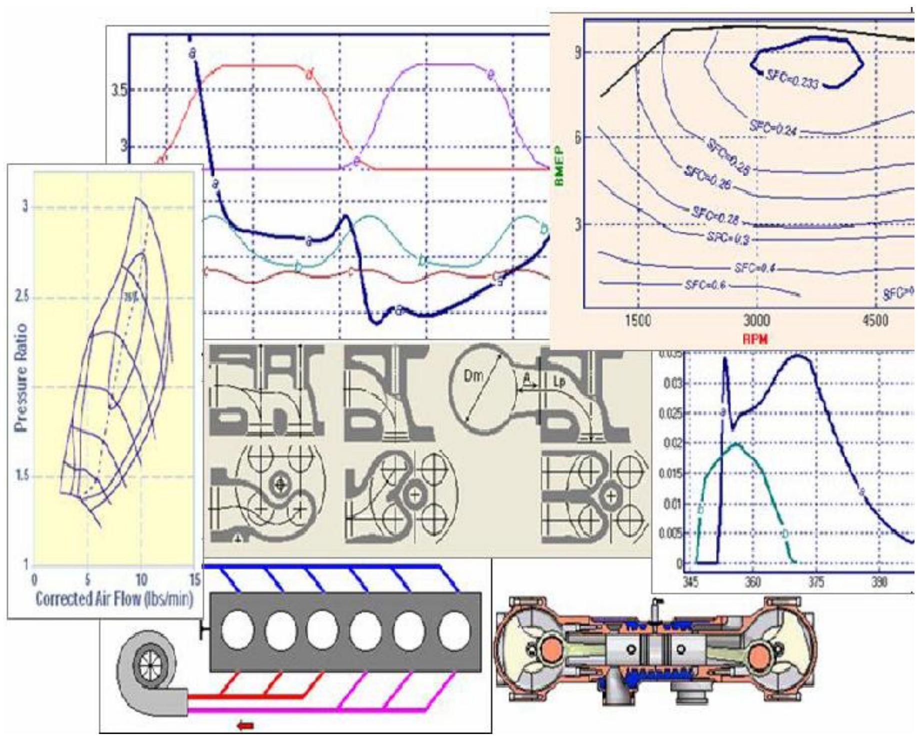

Figure 5.1: modeling DIESEL-RK Software [101]

The program DIESEL-RK makes it possible to simulate the working process of any type of internal combustion engines with high accuracy of predictions with the use of minimum empirical coefficients. The values of these coefficients are constant for any configuration and operating modes of engines and over the whole operating range including part load and idling [101]. The components in this system need a few data to complete the data form and running the model. Engine cylinder and fuel injection system is focused in engine cylinder performance were support diesel fuel from fuel injection system, fresh air intake system and exhaust gas to exhaust system. The components, size and data must be record and inserted to the DIESEL-RK form. All of the engine cylinder and fuel injection system component are injector, cylinder and engine. Exhaust system is the last system in the diesel engine. The DIESEL-RK software is used to simulate the commercial single cylinder diesel engine. The diesel engine is simulated to study the characteristic of engine performance when the engine is operating with different percentage ethanol/water fuel. The simulation results were compared with the data from the diesel engine operating with conventional diesel.

5.2. Effect of Ethanol/Water Addition to Diesel Fuel on Exhaust Emissions

5.2.1. Effect of Water–Diesel DI on Nitrogen Oxides and Nitrogen Dioxide Emissions

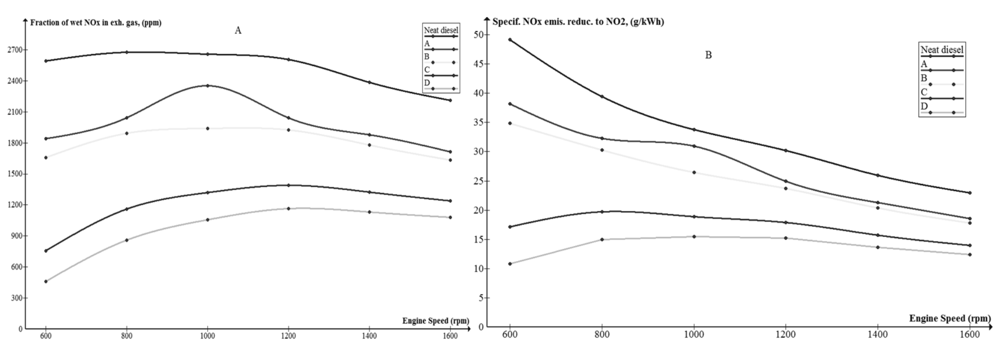

Figure 5.2: Wide Range Ratio Water–Diesel DI Effect on NOx and NO_2 Emissions vs. Engine Speed

Figures (5.2) show, for the fraction of wet NOx in exhaust gas and Specific NOx emission reduce to NO_2, respectively, for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that the NOx and NO_2 emissions by all Water–Diesel is lower than the ones for the corresponding Neat

Diesel Fuel case, with this reduction being higher the higher the percentage of the water. At speed 1000 rpm, the NOx emissions decreased by 11.5%, 27%, 50% and 60% with A, B, C, and D respectively. It can be seen that "A" has lower decreases in the NOx compared to other cases of Water–Diesel. This behavior was also observed in a similar experimental test see **Figure (4.1)** for NOx emissions same operation conditions. At speed 1000 rpm, the NO_2 emissions decreased by 8.5%, 21.7%, 44% and 54% with A, B, C, and D respectively. It can be seen that "A" has lower decreases in the NOx compared to other cases of Water–Diesel.

5.2.2. Effect of EW25–Diesel DI on Nitrogen Oxides and Nitrogen Dioxide Emissions

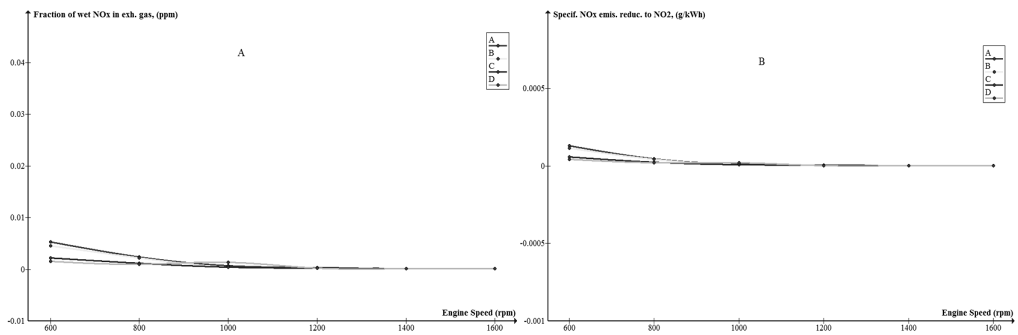

Figure 5.3: Wide Range Ratio EW25–Diesel DI Effect on NOx and NO_2 Emissions vs. Engine Speed

Figures (5.3) show, for the fraction of wet NOx in exhaust gas and Specific NOx emission reduce to NO_2, respectively, for the Neat Diesel fuel and EW25–Diesel DI, at the six speeds considered. One can observe that the NOx and NO_2 emissions by all EW25–Diesel DI is lower than the ones for the corresponding Neat Diesel fuel case, with this reduction being higher the higher the percentage of the EW25. At speed 1000 rpm, the NOx emissions decreased by 100% with A, B, C, and D. It can be seen that "D" has lower decreases in the NOx compared to other cases of EW25–Diesel. This behavior was also observed in a similar experimental test see **Figure (4.2)** for NOx emissions same operation conditions. At speed 1000 rpm, the NO_2 emissions decreased by 100% with A, B, C, and D. It can be seen that "D" has lower decreases in the NOx compared to other cases of EW25–Diesel DI.

5.2.3. Effect of EW50–Diesel DI on Nitrogen Oxides and Nitrogen Dioxide Emissions

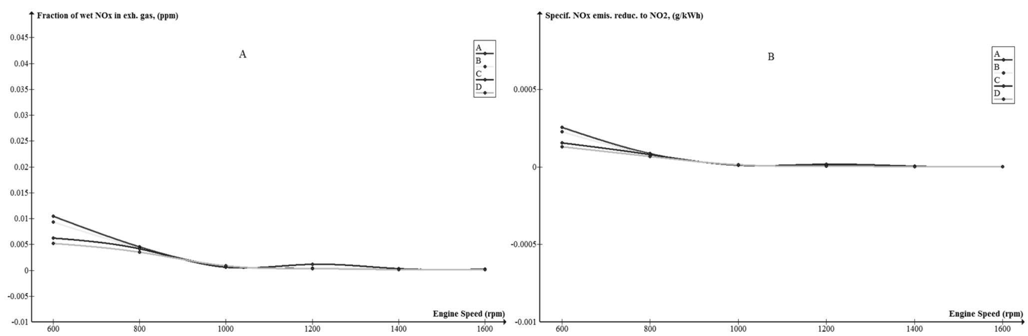

Figure 5.4: Wide Range Ratio EW50–Diesel DI Effect on NOx and NO_2 Emissions vs. Engine Speed

Figures (5.4) show, for the fraction of wet NOx in exhaust gas and Specific NOx emission reduce to NO_2, respectively, for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the NOx and NO_2 emissions by all EW50–Diesel DI is lower than the ones for the corresponding Neat Diesel fuel case, with this reduction being higher the higher the percentage of the EW50. At speed 1000 rpm, the NOx emissions decreased by 100%, for all cases EW50–Diesel DI. It can be seen that "D" has lower decreases in the NOx compared to other cases of EW50. This behavior was also observed in a similar experimental test see **Figure (4.3)** for NOx emissions same operation conditions. At speed 1000 rpm, the NO_2 emissions decreased by 100%, for all cases EW50–Diesel DI. It can be seen that "D" has lower decreases in the NOx compared to other cases of EW50–Diesel DI.

5.2.4. Effect of EW75–Diesel DI on Nitrogen Oxides and Nitrogen Dioxide Emissions

Figures (5.5) show, for the fraction of wet NOx in exhaust gas and Specific NOx emission reduce to NO_2, respectively, for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the NOx and NO_2 emissions by all EW75 is lower than the ones for the corresponding Neat Diesel fuel case, with this reduction being higher the higher the percentage of the EW75. At speed 1000 rpm, the NOx emissions decreased by 100%, for all cases EW75–Diesel DI. It can be seen that “C” has lower decreases in the NOx compared to other cases of EW75–Diesel. This behavior was also observed in a similar experimental test see **Figure (4.4)** for NOx emissions same operation conditions. At speed 1000 rpm, the NO_2 emissions decreased by 99.9%, for all case EW75–Diesel DI. It can be seen that “C” has lower decreases in the NOx compared to other cases of EW75–Diesel.

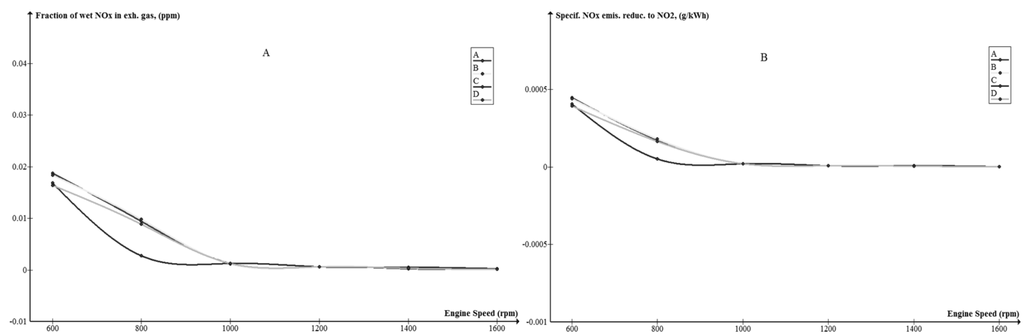

Figure 5.5: Wide Range Ratio EW75–Diesel DI Effect on NOx and NO_2 Emissions vs. Engine Speed

5.2.5. Effect of Ethanol–Diesel DI on Nitrogen Oxides and Nitrogen Dioxide Emissions

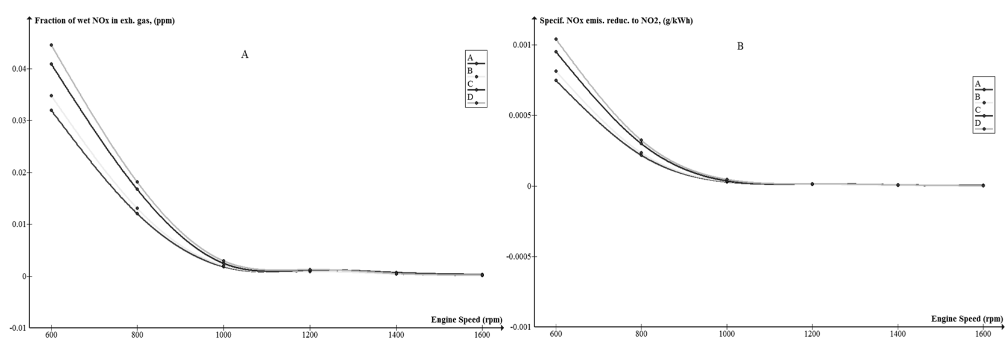

Figure 5.6: Wide Range Ratio Ethanol–Diesel DI Effect on NOx and NO_2 Emissions vs. Engine Speed

Figures (5.6) show, for the fraction of wet NOx in exhaust gas and Specific NOx emission reduce to NO_2, respectively, for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that the NOx and NO_2 emissions by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this reduction being higher the higher the percentage of the Ethanol. At speed 1000 rpm, the NOx emissions decreased by 99.9%, for all cases Ethanol. It can be seen that "A" has lower decreases in the NOx compared to other cases of Ethanol–Diesel. This behavior was also observed in a similar experimental test see **Figure (4.5)** for NOx emissions same operation conditions. At speed 1000 rpm, the NO_2 emissions decreased by 99.9%, for all case Ethanol–Diesel. It can be seen that "A" has lower decreases in the NOx compared to other cases of Ethanol–Diesel.

5.2.6. Effect of Water–Diesel DI on Particulate Matter Emissions

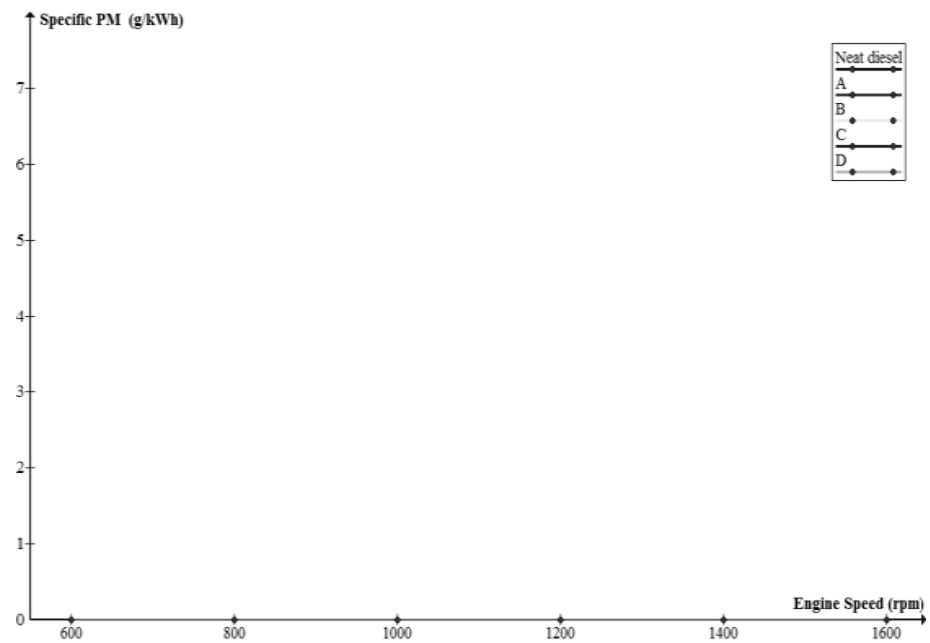

Figure 5.7: Wide Range Ratio Water–Diesel DI Effect on PM vs. Engine Speed

Figure (5.7) show the Specific PM in exhaust gas for the neat diesel fuel and with Water–Diesel, at the six speeds considered. At all conditions of simulation test, the value PM emissions is zero. The amount of PM emission depends on the quality of the fuel oil and the completeness of burning in the combustion chambers. PM is generated from incomplete hydrocarbon burning when the fuel oil is injected into a cylinder and mixes with its surrounding air imperfectly. PM is generally composed of three compounds: **(1)** solid carbon particles produced from the burning process, PM emitted from the diesel engines in the early burning stage consisting of 40–80% solid carbon particles; **(2)** soluble organic fractions (briefly termed as SOF), produced from the adsorption or condensation of hydrocarbons with heavy molecular weight onto the surface of the carbon particles. Most SOF comes from unburned lubricant (about 40% of the total); fuel oil (about 25% of the total); **(3)** sulfides, additives for fuel oil, etc. Hence, adequately controlling the burning process can effectively reduce the solid carbon particles and SOF, leading to a decrease in the exhausted PM. Expressed that combustion process would be much faster, hence effective efficiency increases and adiabatic flame temperature decreases, the formation of soot, NO, UBHC and PM emissions reduces with respect to the increased amount of OH radicals because of water dissociation when using emulsified fuel in diesel engine.

5.2.7. Effect of EW25–Diesel DI on Particulate Matter Emissions

Figure (5.8) show the PM in exhaust gas for the Neat Diesel fuel and EW25–Diesel DI, at the six speeds considered. One can observe that the PM by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this increase being lower the higher the percentage of the EW25. After speed 1000 rpm, the PM increased from zero, for all cases EW25–Diesel. It can be seen that "A" has lower increases in the PM compared to other cases of EW25–Diesel. Because the largest PM emission even at high cetane numbers. Unfortunately, if the diesel combustion system is not well controlled, it can produce higher levels of PM.

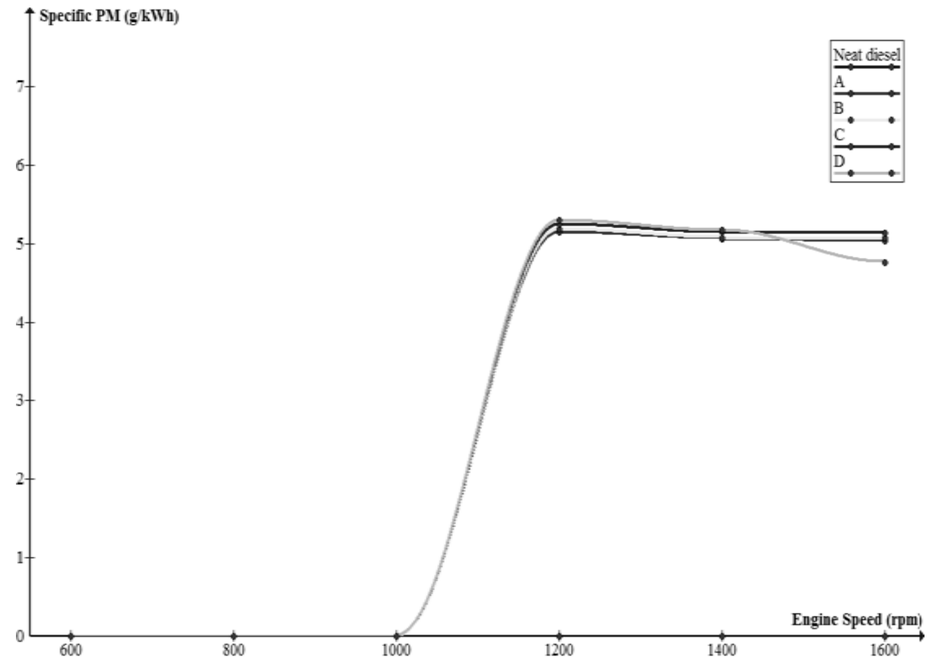

Figure 5.8: Wide Range Ratio EW25–Diesel DI Effect on PM vs. Engine Speed

5.2.8. Effect of EW50–Diesel DI on Particulate Matter Emissions

Figure (5.9) show the PM in exhaust gas for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the PM by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case. On the other hand, EW50–Diesel is found to significantly small change PM level in the EW50–Diesel conditions. After speed 1000 rpm, the PM increased from zero, for all cases EW50–Diesel. It can be seen that "C" has lower increases in the PM compared to other cases of EW50–Diesel. Regarding PM, the results are mixed: PM emission could increase or decrease, depending on the operating conditions. They concluded that the effect of the composition and structure on PM emissions is negligible as compared to the oxygen content, which was acknowledged as the main factor causing PM emission reductions.

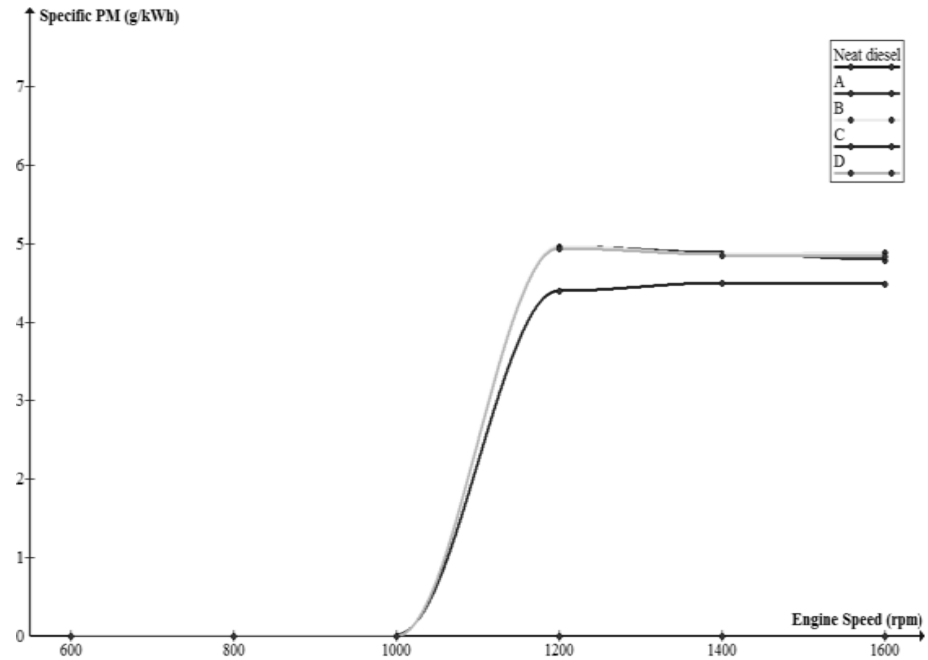

Figure 5.9: Wide Range Ratio EW50–Diesel DI Effect on PM vs. Engine Speed

5.2.9. Effect of EW75–Diesel DI on Particulate Matter Emissions

Figure (5.10) show the PM in exhaust gas for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the PM by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case. A certain increase in PM emissions of EW75–Diesel when compared to Neat Diesel fuel can be considered a normal trend, with the extent of increase of PM varying with EW75 blend ratio, engine technology, test cycle, etc. After speed 1000 rpm, the PM increased from zero, for all cases EW75–Diesel. It can be seen that “C” has lower increases in the PM compared to other cases of EW75–Diesel. This increase is mainly produced by oxygenated compounds contained in EW75. Oxygen contained in fuel contributes to an increase of the local oxygen–fuel ratio during combustion. In addition, the lack of aromatic hydrocarbons and sulphur compounds further contribute to increase PM emissions.

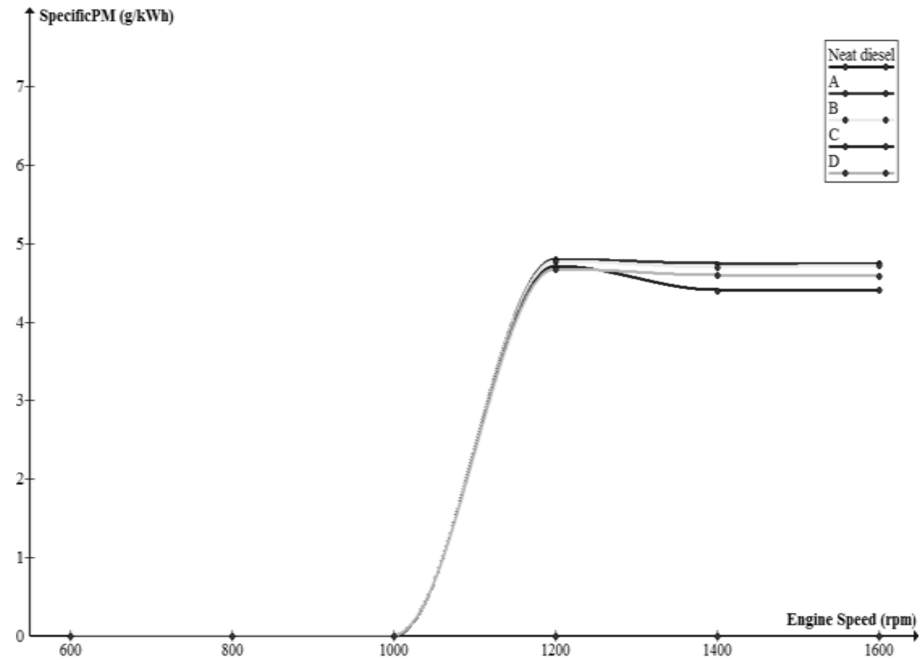

Figure 5.10: Wide Range Ratio EW75–Diesel DI Effect on PM vs. Engine Speed

5.2.10. Effect of Ethanol–Diesel DI on Particulate Matter Emissions

Figure (5.11) show the PM in exhaust gas for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that the PM by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case. The lower cetane number of the blended fuel could result in longer ignition delay and hence more complete mixing of fuel vapors with air before ignition occurs, which will lead to higher PM emissions and lower NOx emissions. However, the larger amount of fuel to be evaporated could increase the in cylinder gas temperature to some extent. It can be seen that “C” has lower increases in the PM compared to other cases of Ethanol–Diesel.

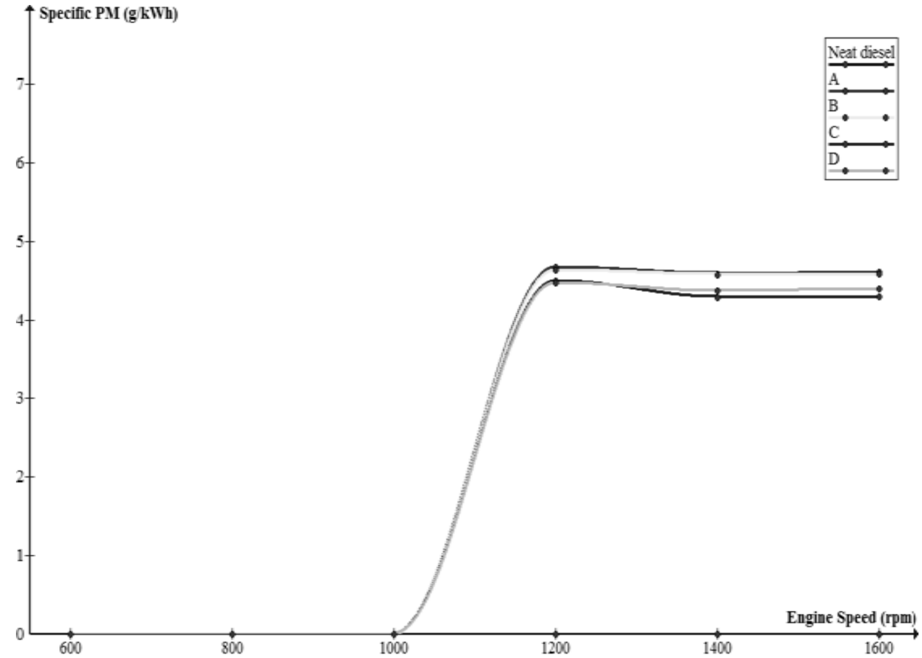

Figure 5.11: Wide Range Ratio Ethanol–Diesel DI Effect on PMvs. Engine Speed

5.2.11. Effect of Water–Diesel DI on Summary Emission of Particulate Matter and Nitrogen Oxides

Figure (5.12) show the Summary emission of PM and NOx (SE) in exhaust gas for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that the SE by all Water–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of the Water–Diesel. At speed of 1200 rpm, the SE is decreased by 17%, 21 %, 40.8% and 49.5% with A, B, C, and D respectively. It can be seen that “D” has lower decreases in the NOx compared to other cases of Water–Diesel. Because of this effect, a reduction is seen in engine efficiency, and also in PM emissions considerably. Investigated water injection into diesel engine and observed that whilst NOx, PM emissions.

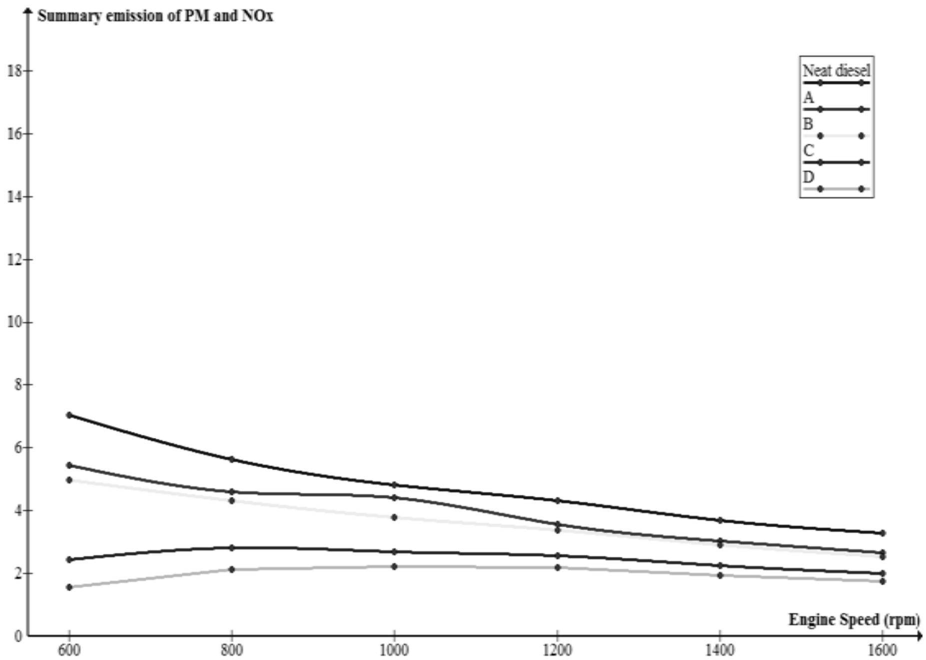

Figure 5.12: Wide Range Ratio Water–Diesel DI Effect on SE Emissions vs. Engine Speed

5.2.12. Effect of EW25–Diesel DI on Summary Emission of Particulate Matter and Nitrogen Oxides

Figure (5.13) show the Summary emission of PM and NOx (SE) in exhaust gas for the Neat Diesel fuel and EW25–Diesel, at the six speeds considered. One can observe that the SE by all EW25–Diesel less than speed 1000 rpm is lower than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of the EW25–Diesel. At speed of 1200 rpm, the SE is increased by 297%, 300 %, 305% and 310% with A, B, C, and D respectively. It can be seen that “D” has higher increases in the SE compared to other cases of EW25–Diesel. Diesel engines have the advantages of better fuel economy, lower emissions of UBHC and CO. However, diesel engines suffered from high emissions of PM and NOx, and it is hard to reduce them simultaneously. Methods to reduce PM and NOx emissions include high-pressure injection, turbocharging and exhaust after treatments, etc. Fuel additives are still investigated and thought to be one of the attractive solutions. Adding oxygenates in diesel can substantially reduce emission of PM without significant effects on NOx. Oxygenates are also full or partial substitutes for diesel.

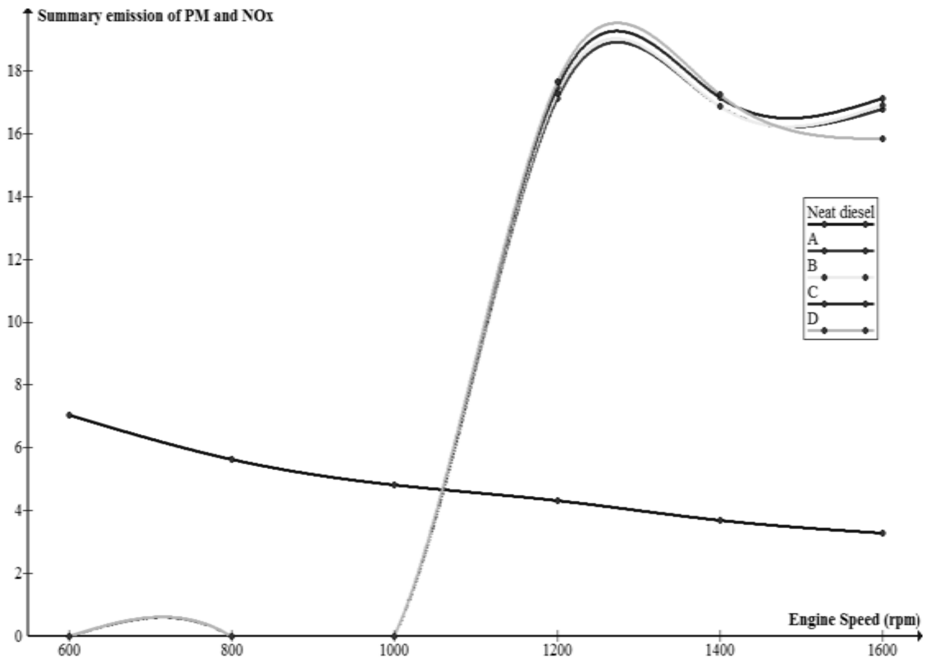

Figure 5.13: Wide Range Ratio EW25–Diesel DI Effect on SE Emissions vs. Engine Speed

5.2.13. Effect of EW50–Diesel DI on Summary Emission of Particulate Matter and Nitrogen Oxides

Figure (5.14) show the Summary emission of PM and NOx (SE) in exhaust gas for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the SE by all EW50–Diesel less than speed 1000 rpm is lower than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of the EW50–Diesel case (A, B, C). At speed of 1200 rpm, the SE is increased by 283%, 283 %, 240% and 281% with A, B, C, and D respectively. It can be seen that “D” has higher increases in the SE compared to other cases of EW50–Diesel. The explanation to this result could be the delayed combustion process (simulation without premixed combustion phase) combined with a flat high temperature distillation curve of EW50–Diesel. A delayed combustion process with the highest UBHC emission produced the highest PM emission as a consequence of UBHC condensation on the filter.

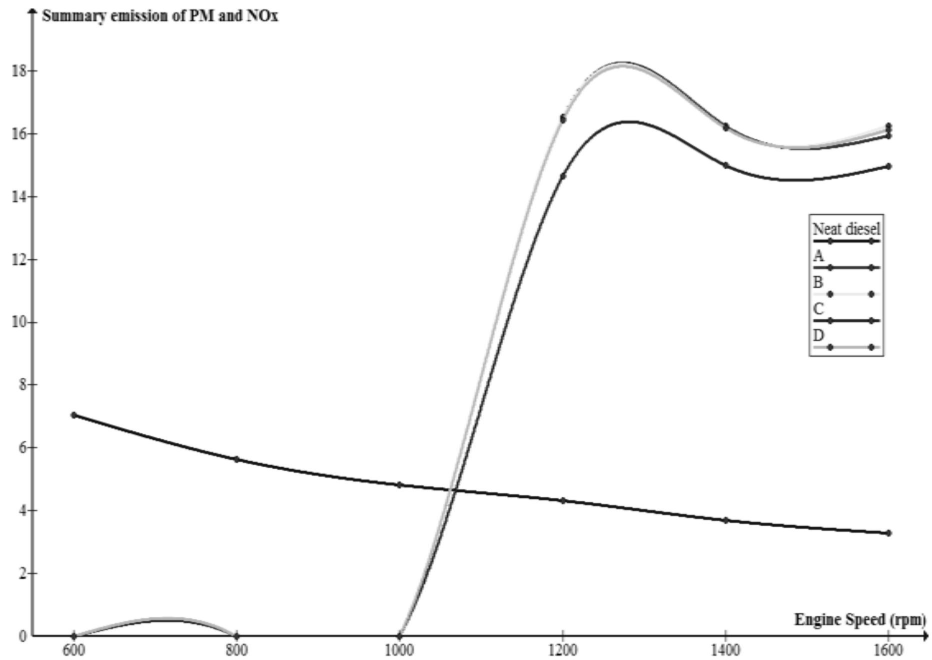

Figure 5.14: Wide Range Ratio EW50–Diesel DI Effect on SE Emissions vs. Engine Speed

5.2.14. Effect of EW75–Diesel DI on Summary Emission of Particulate Matter and Nitrogen Oxides

Figure (5.15) show the Summary emission of PM and NOx (SE) in exhaust gas for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the SE by all EW75–Diesel less than speed 1000 rpm is lower than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of EW75–Diesel case (A, B, C). At speed of 1200 rpm, the SE is increased by 271%, 269%, 264% and 261% with A, B, C, and D respectively. It can be seen that “A” has higher increases in the SE compared to other cases of EW75–Diesel. However, this operating condition can produce an increase in PM emissions, increasing as the EW75 is retarded. This trend can be explained by the longer injection duration as consequence of the lower LHV of EW75. The EW75–

Diesel fuel produced a significant decrease of both NOx and PM emissions independently of the low tested only.

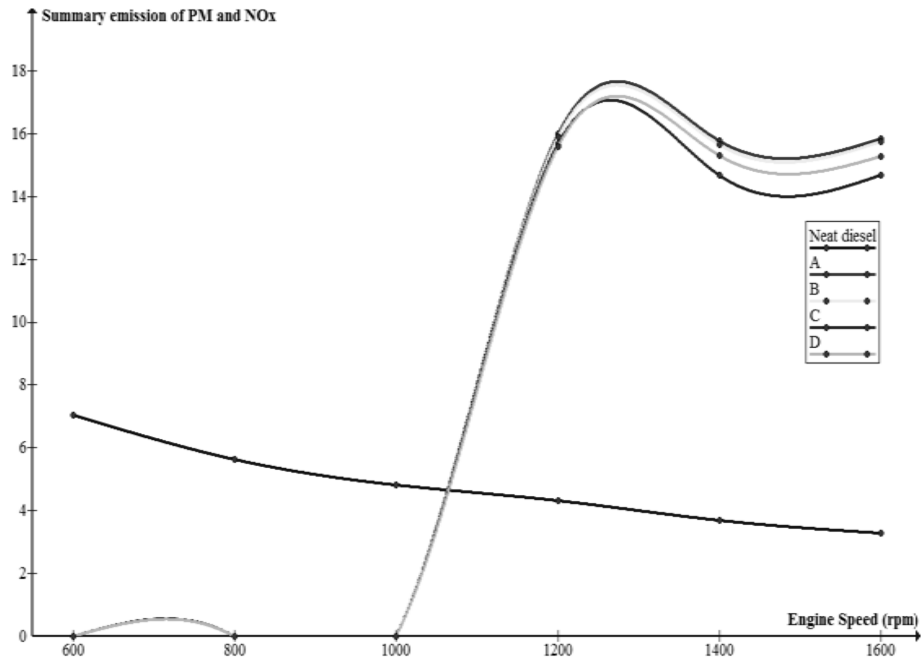

Figure 5.15: Wide Range Ratio EW75–Diesel DI Effect on SE Emissions vs. Engine Speed

5.2.15. Effect of Ethanol–Diesel DI on Summary Emission of Particulate Matter and Nitrogen Oxides

Figure (5.16) show the Summary emission of PM and NOx (SE) in exhaust gas for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that the SE by all Ethanol–Diesel less than speed 1000 rpm is lower than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of Ethanol–Diesel case (A, B, C). At speed of 1200 rpm, the SE is increased by 261%, 258 %, 247% and 245% with A, B, C, and D respectively. It can be seen that “C” has higher increases in the SE compared to other cases of Ethanol–Diesel. SE reduced remarkably for blends especially at medium speeds and all Ethanol–Diesel. Advancing injection reduced the SE for all blends and diesel fuel at low speeds. Significant reduction in SE is observed for high ethanol content blends; however reduction in SE does not indicate the reduction in

PM in same proportion. NOx emissions are decreased for blends at all working conditions. It is slightly increased at other speeds. This indicates latent heat of vaporization is more effective than Cetane index and oxygen content at these operating conditions. The injection amount is required to be advanced for use of high percentage Ethanol–Diesel DI.

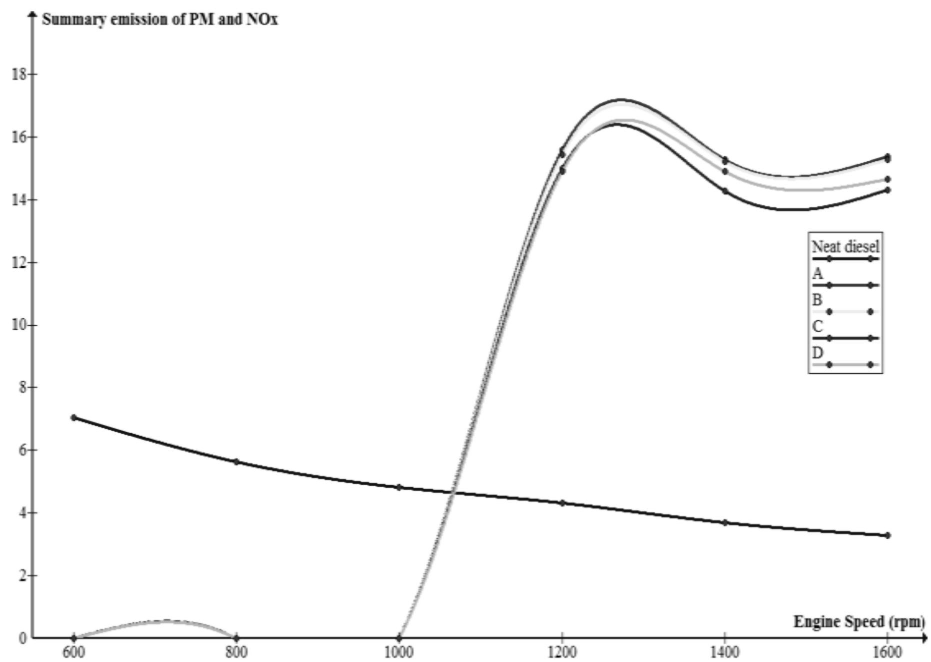

Figure 5.16: Wide Range Ratio Ethanol–Diesel DI Effect on SE Emissions vs. Engine Speed

5.2.16. Effect of Water–Diesel DI on Specific Carbon Dioxide Emissions

Figure (5.17) show of the Specific (CO_2) in exhaust gas for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that the CO_2 by all Water–Diesel is higher than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of Water–Diesel. At speed of 1200 rpm, the CO_2 is increased by 5.4%, 6.3%, 10.7% and 12.8% with A, B, C, and D respectively. It can be seen that “D” has higher increases in the CO_2 compared to other cases of Water–Diesel. CO_2 emissions increase depending on the increase of (BSFC) Water–Diesel fuelled. The reason of formation of CO_2 is the result of complete combustion. If the in-cylinder temperature during combustion process is optimum sufficient to support the complete combustion then transformation of CO to CO_2 is not occurred. Results from experimental test show that **Figure (4.19)** CO_2 concentrations Water–Diesel have small different effect.

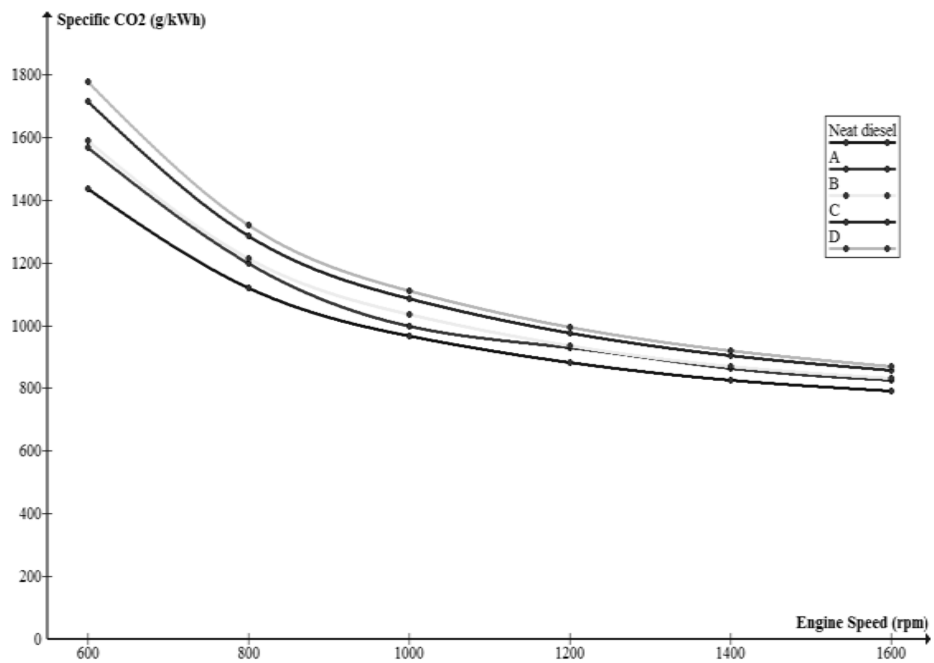

Figure 5.17: Wide Range Ratio Water–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

5.2.17. Effect of EW25–Diesel DI on Specific Carbon Dioxide Emissions

Figure (5.18) show of the Specific (CO_2) in exhaust gas for the Neat Diesel fuel and EW25–Diesel, at the six speeds considered. One can observe that the CO_2 by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW25–Diesel. At speed of 1200 rpm, the CO_2 is decreased by 75.90%, 75.95%, 76.17% and 76.28% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the CO_2 compared to other cases of EW25–Diesel. The result of incomplete combustion of the EW25–Diesel, factors causing combustion deterioration such as high latent heats of vaporization can be responsible for the poor oxidation reaction rate of CO and decreased CO_2 production. Dissimilar results have been reported in experimental studies shown in **Figure (4.20)**, because operation is further increased, the equivalence ratio approaches stoichiometric and a sharp increase in CO_2 are observed with diesel fuel.

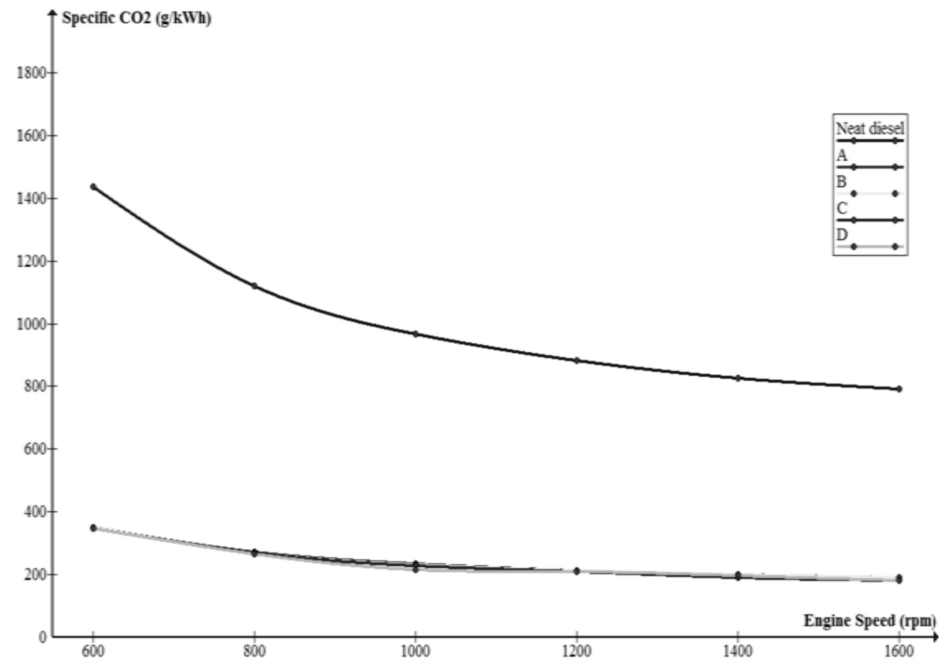

Figure 5.18: Wide Range Ratio EW25–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

5.2.18. Effect of EW50–Diesel DI on Specific Carbon Dioxide Emissions

Figure (5.19) show of the Specific (CO_2) in exhaust gas for the Neat Diesel fuel and with the EW50, at the six speeds considered. One can observe that the CO_2 by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW50–Diesel. At speed of 1200 rpm, the CO_2 is decreased by 76.97%, 77.21%, 83.07% and 78.01% with A, B, C, and D respectively. It can be seen that “D” has higher decreases in the CO_2 compared to other cases of EW50–Diesel. This is due to the lower carbon content of ethanol present in the EW50 fuel. These results have the same trends with varying used in the experimental are shown in **Figure (4.21)** CO_2 concentrations.

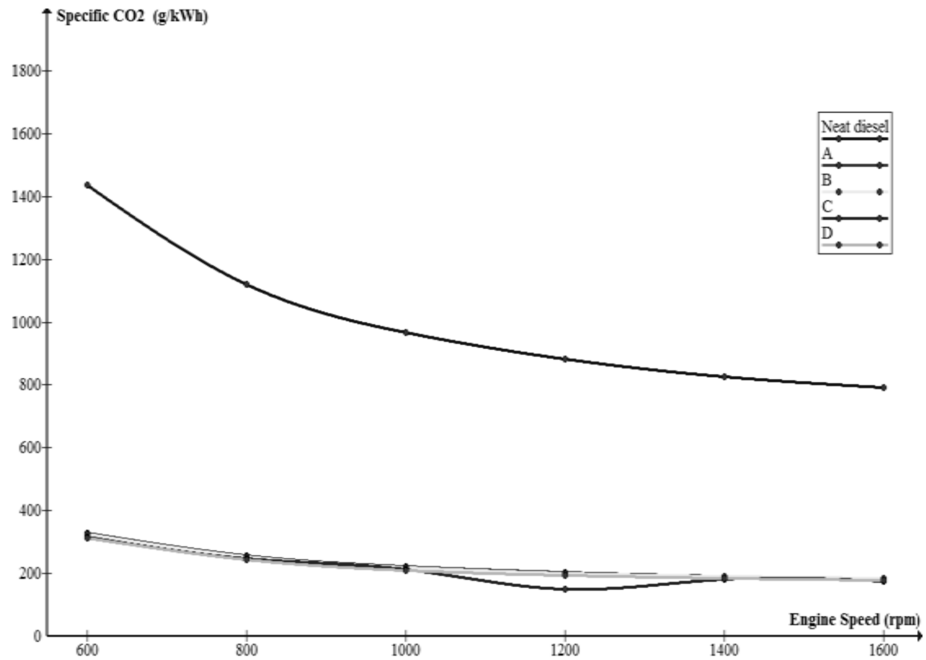

Figure 5.19: Wide Range Ratio EW50–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

5.2.19. Effect of EW75–Diesel DI on Specific Carbon Dioxide Emissions

Figure (5.20) show of the Specific (CO_2) in exhaust gas for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the CO_2 by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of EW75–Diesel. At speed of 1200 rpm, the CO_2 is decreased by 77.9%, 78.16%, 79.06% and 79.38% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the CO_2 compared to other cases of EW75–Diesel. This may be due to incomplete combustion, which results in a lower heat release rate and lower BTE at EW75–Diesel. The CO_2 emissions increase at Neat Diesel due to the partial and longer duration of combustion, which results in higher peak heat release rate, and higher peak pressure. Show experimental test **Figure (4.22)** CO_2 concentrations similar result case (A, D) only, due to the several of environment.

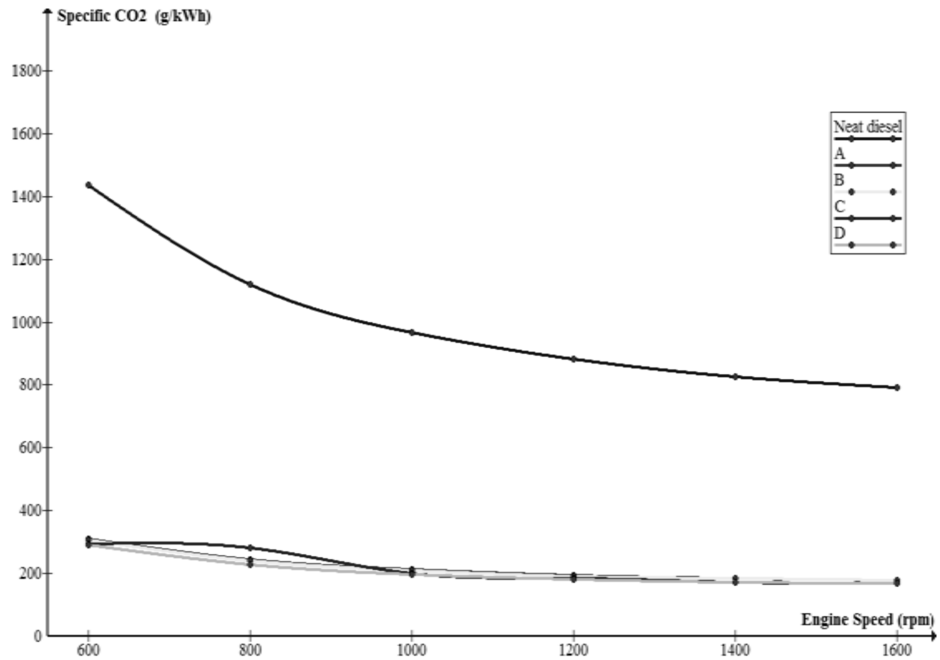

Figure 5.20: Wide Range Ratio EW75–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

5.2.20. Effect of Ethanol–Diesel DI on Specific Carbon Dioxide Emissions

Figure (5.21) show of the Specific (CO_2) in exhaust gas for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that the CO_2 by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Ethanol–Diesel. At speed of 1200 rpm, the CO_2 is decreased by 77.9%, 78.16%, 79.06% and 79.38% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the CO_2 compared to other cases of Ethanol–Diesel. The CO_2 emission for the dulling fuel is lower than Neat Diesel for almost all speeds and all blends. This may be due to late burning of fuel leading to incomplete oxidation of CO. Similar trends in experimental test show **Figure (4.23)** the variation of CO_2.

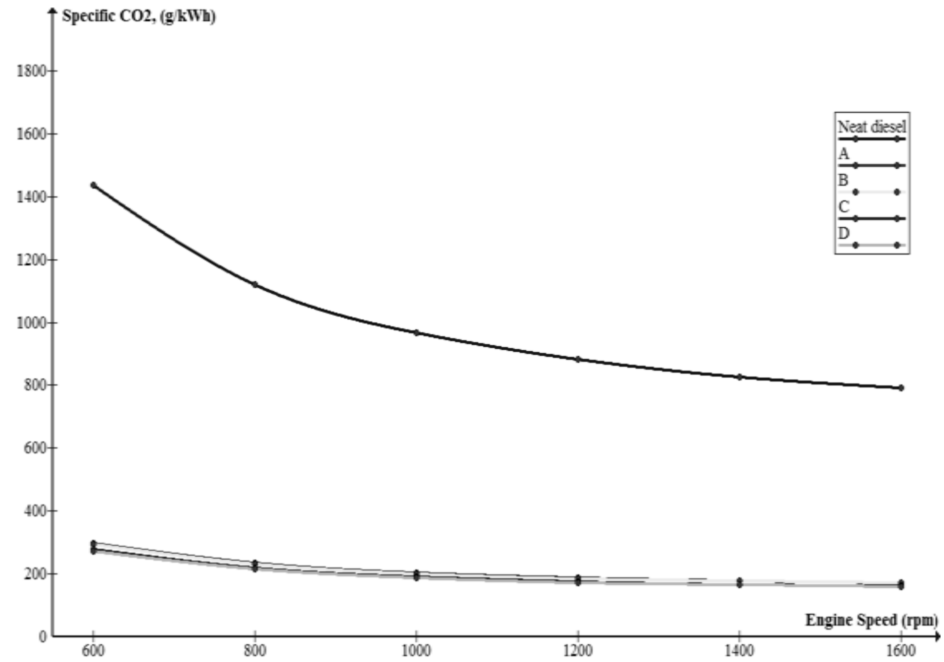

Figure 5.21: Wide Range Ratio Ethanol–Diesel DI Effect on CO_2 Emissions vs. Engine Speed

5.2.21. Effect of Water–Diesel DI on Bosch Smoke Number

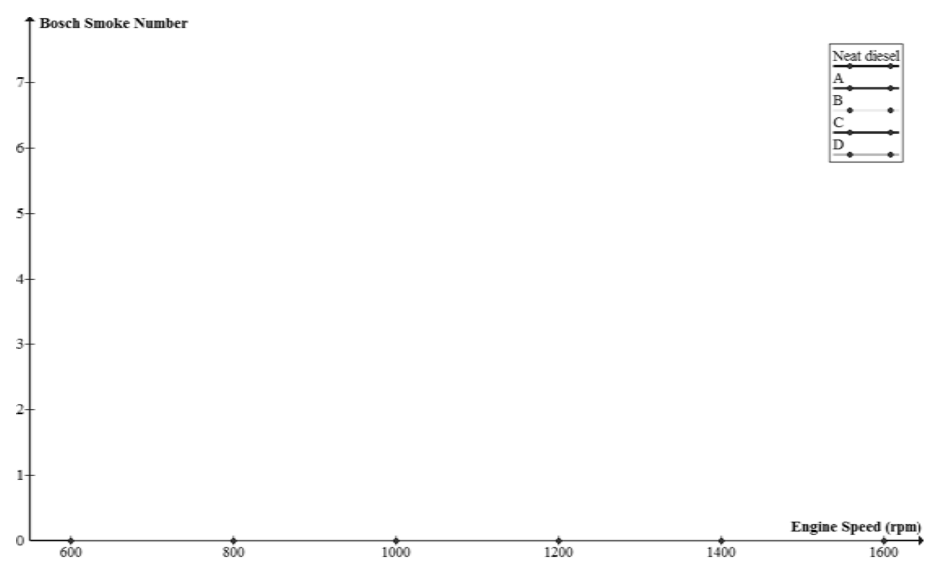

Figure 5.22: Wide Range Ratio Water–Diesel DI Effect on Bosch Emissions vs. Engine Speed

Figure (5.22) show the Bosch Smoke Number in exhaust gas for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that the Bosch Smoke Number by all Water–Diesel and Neat Diesel fuel equal zero, due to its oxygen content and small particle diameter of the injected fuel at high injection pressure about simulation software. Bosch Smoke Number is almost negligible, while NOx emission has been reduced to the minimum.

5.2.22. Effect of EW25–Diesel DI on Bosch Smoke Number

Figure (5.23) show the Bosch Smoke Number in exhaust gas for the Neat Diesel fuel and EW25–Diesel, at the six speeds considered. One can observe that the Bosch Smoke Number by all EW25–Diesel after 1000 rpm is higher than the ones for the corresponding Neat Diesel fuel case, with this negligible change the percentage of the EW25–Diesel. It is believed that the water of the EW25–Diesel fuel significantly enhances the evaporation of the fuel and improves the mixing with air; as a result, more smoke is formed.

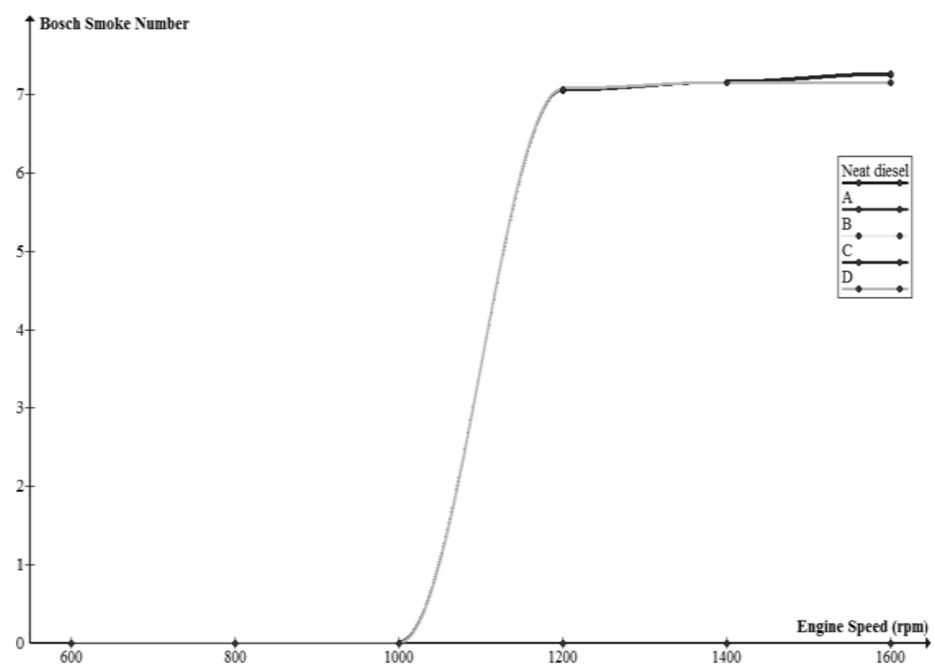

Figure 5.23: Wide Range Ratio EW25–Diesel DI Effect on Bosch Emissions vs. Engine Speed

5.2.23. Effect of EW50–Diesel DI on Bosch Smoke Number

Figure (5.24) show the Bosch Smoke Number in exhaust gas for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the Bosch Smoke Number by all EW50–Diesel after 1000 rpm is higher than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of the EW50–Diesel case (A, B, D). It can be seen that "C" has lower increases in the Bosch compared to other cases of EW50–Diesel. This is due to the high oxygen content and low sulfur content of Ethanol. The oxygen content of ethanol provides certain advantages like post flame oxidation and increases the flame speed during the air fuel interactions, particularly in the fuel-rich region. Indeed, it reveals the presence of the oxygen content of ethanol which enhances the hydrocarbon oxidation. The vapor fuels exhibit smoke reduction when compared to Neat Diesel fuel and EW50–Diesel.

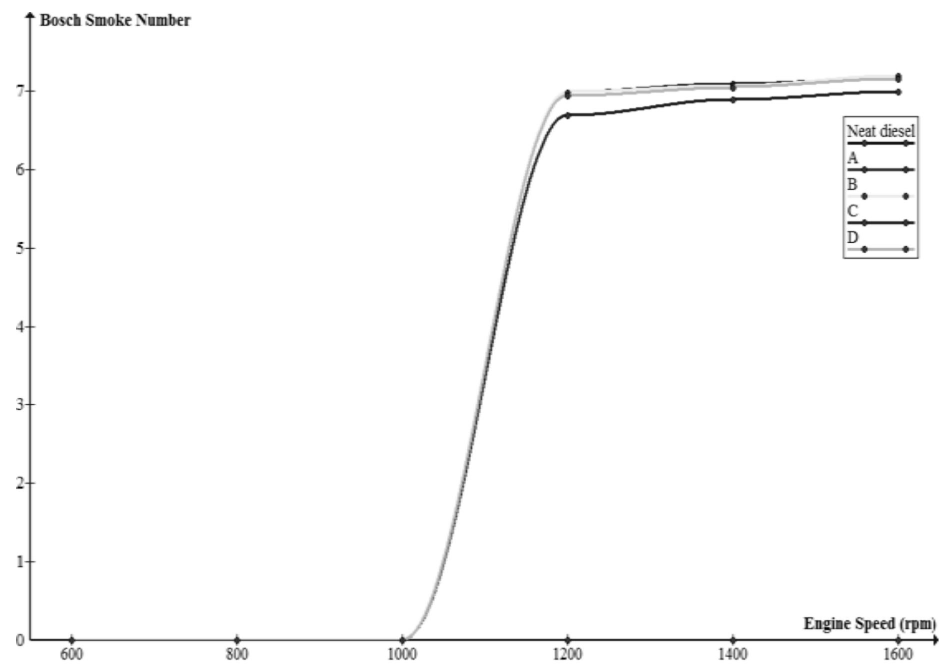

Figure 5.24: Wide Range Ratio EW50–Diesel DI Effect on Bosch Emissions vs. Engine Speed

5.2.24. Effect of EW75–Diesel DI on Bosch Smoke Number

Figure (5.25) show the Bosch Smoke Number in exhaust gas for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the Bosch Smoke Number by all EW75–Diesel after 1000 rpm is higher than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of the EW75–Diesel case (A, B, D). It can be seen that “C” has lower increases in the Bosch compared to other cases of EW75–Diesel. This may be reasoned to more complete combustion due to better air fuel mixing and the presence of oxygen in the case “C”. The above result indicates that if Bosch is greatly by means of a high ratio of EW75–Diesel, more attention can be focus on the control of NOx emissions. Therefore, it may be a good choice for diesel engines to decrease the use of expensive after-treatment system by addition of oxygenated ethanol.

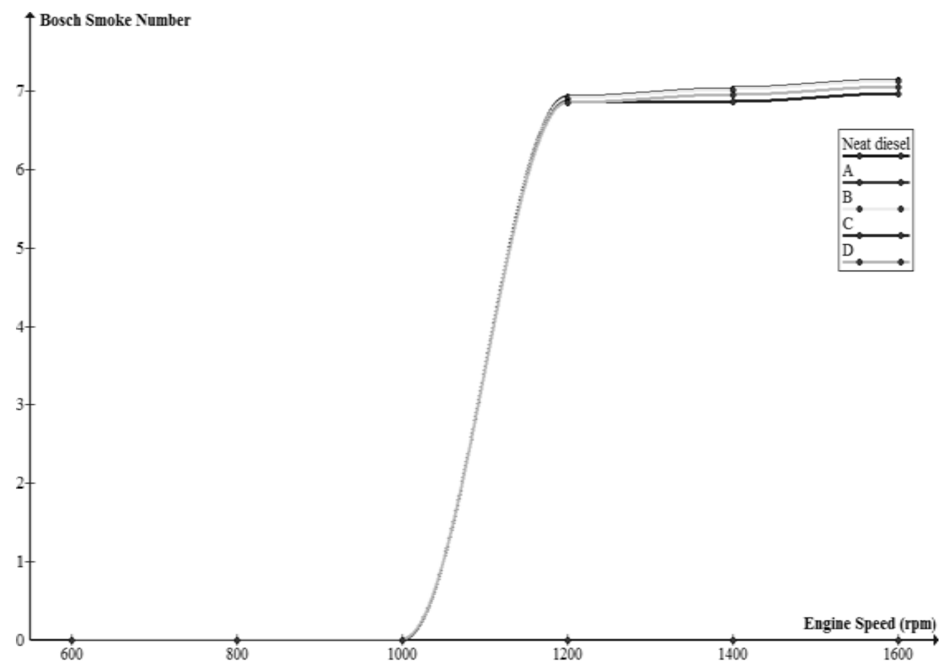

Figure 5.25: Wide Range Ratio EW75–Diesel DI Effect on Bosch Emissions vs. Engine Speed

5.2.25. Effect of Ethanol–Diesel DI on Bosch Smoke Number

Figure (5.26) show the Bosch Smoke Number in exhaust gas for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that the Bosch Smoke Number by all Ethanol–Diesel after 1000 rpm is higher than the ones for the corresponding Neat Diesel fuel case, with this increase being higher the higher the percentage of the Ethanol–Diesel case (A, B, D). It can be seen that “C” has lower increases in the Bosch Smoke Number compared to other cases of Ethanol–Diesel. This may be due to fuel being able to improve mixing with air throughout the combustion chamber which results in complete combustion. Because the mixing of fuel with air becomes bad through physical period, smoke emissions will be more. However, the smoke emissions of the both fuels are increased when the injection timing is retarded at high load condition with low injection pressure. This is because of the sluggish and diffusion combustion phase caused by a reduced rate of fuel-air mixing due to later injection.

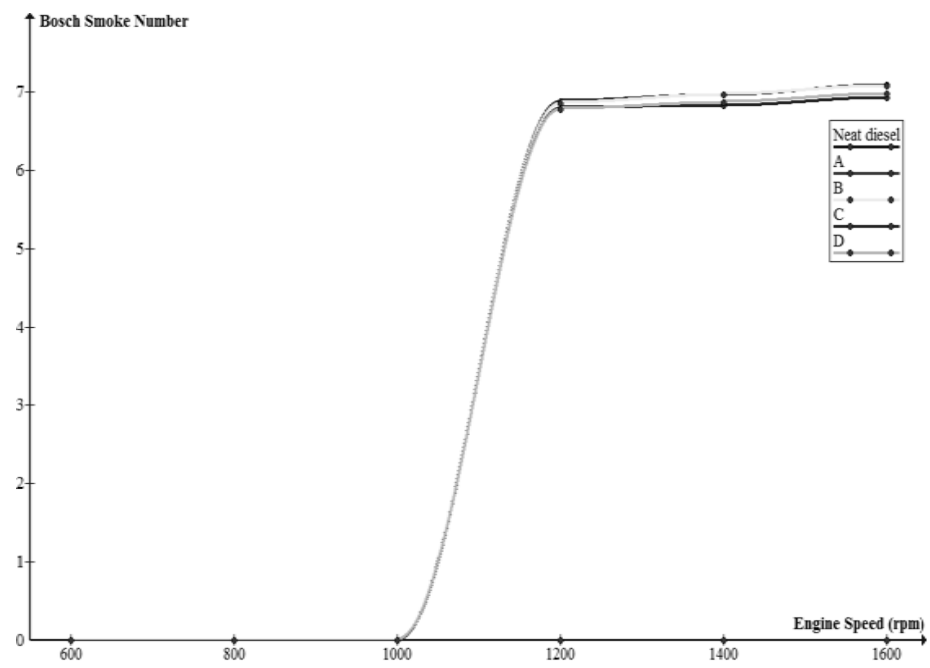

Figure 5.26: Wide Range Ratio Ethanol–Diesel DI Effect on Bosch Emissions vs. Engine Speed

5.2.26. Effect of Water–Diesel DI on Exhaust Gas Temperature

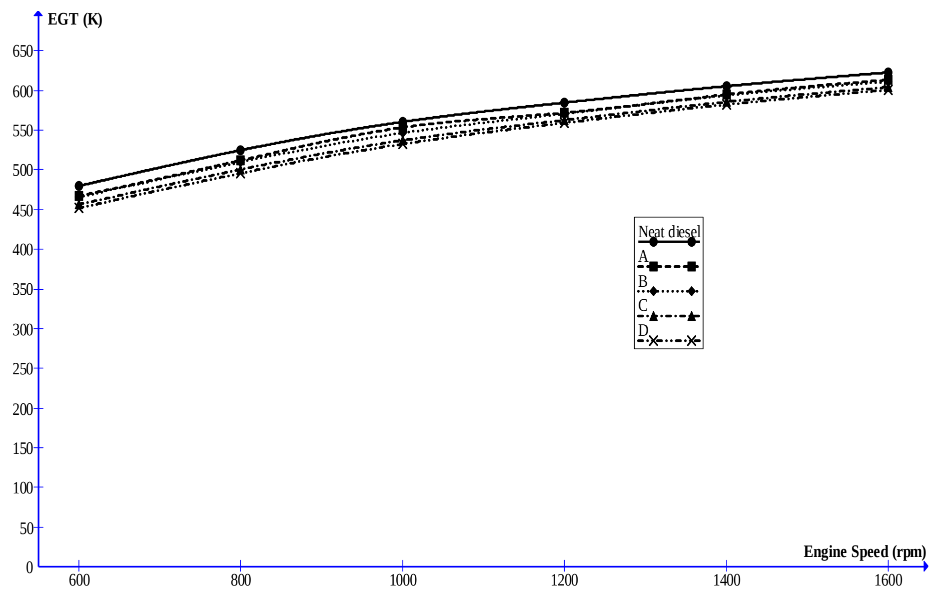

Figure 5.27: Wide Range Ratio Water–Diesel DI Effect on EGT vs. Engine Speed

Figure (5.27) show of EGT for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that EGT by all Water–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Water–Diesel. At speed of 1200 rpm, EGT is decreased by 2.1%, 2.4%, 3.7% and 4.4% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in EGT compared to other cases of Water–Diesel. It would be expected that the fuels with the highest in cylinder temperature levels would have the highest NOx emission. The lack of complete fuel combustion increases the EGT contributing to a higher heat loss. The results in a temperature drop in the burning zone due to a dilution effect, thermal, and chemical effects. Furthermore, the water may contribute to the reduction in combustion temperatures. These results have the same trends with varying used in the experimental are shown in **Figure (4.31)**wide range ratio Water–Diesel DI Effect on EGT vs. Engine Speed.

5.2.27. Effect of EW25–Diesel DI on Exhaust Gas Temperature

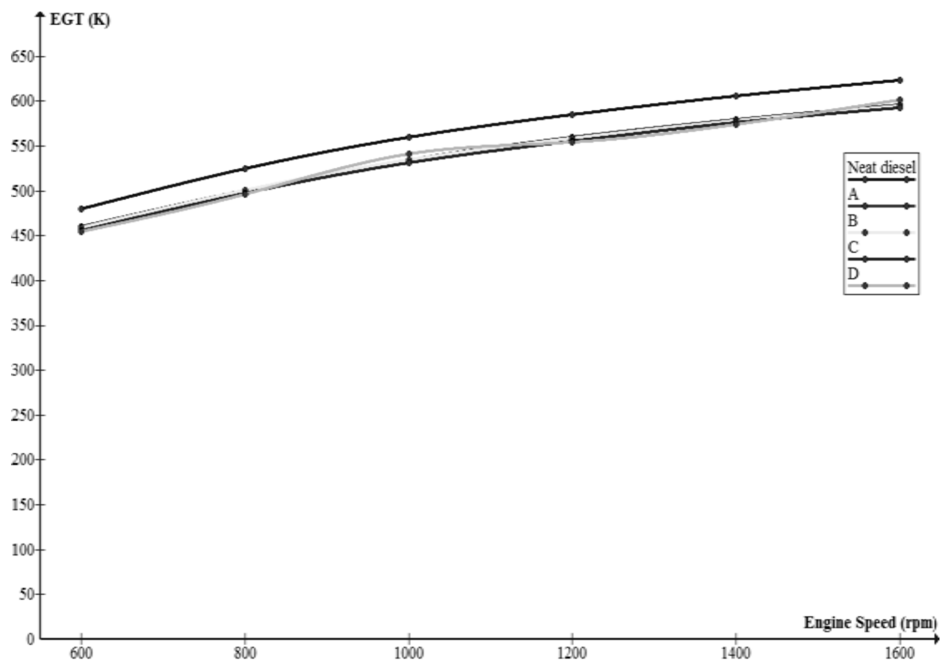

Figure 5.28: Wide Range Ratio EW25–Diesel DI Effect on EGT vs. Engine Speed

Figure (5.28) show of EGT for the Neat Diesel fuel and EW25–Diesel, at the six speeds considered. One can observe that EGT by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW25–Diesel. At speed of 1200 rpm, EGT is decreased by 4.3%, 4.5%, 5.0% and 5.2% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in EGT compared to other cases of EW25–Diesel. In the engine, the heat has to come from the air, so the enthalpy of vaporisation is more critical in determining how the spray evaporates and disperses than the distillation curve. The combined effects of the change in stoichiometric Air Fuel Ratio and the higher enthalpy of vaporisation means that, per unit mass of stoichiometric mixture, the enthalpy of vaporisation has increased. These trends are more pronounced with experimental test show on **Figure (4.32)** wide range ratio EW25–Diesel DI effect on EGT vs. engine speed.

5.2.28. Effect of EW50–Diesel DI on Exhaust Gas Temperature

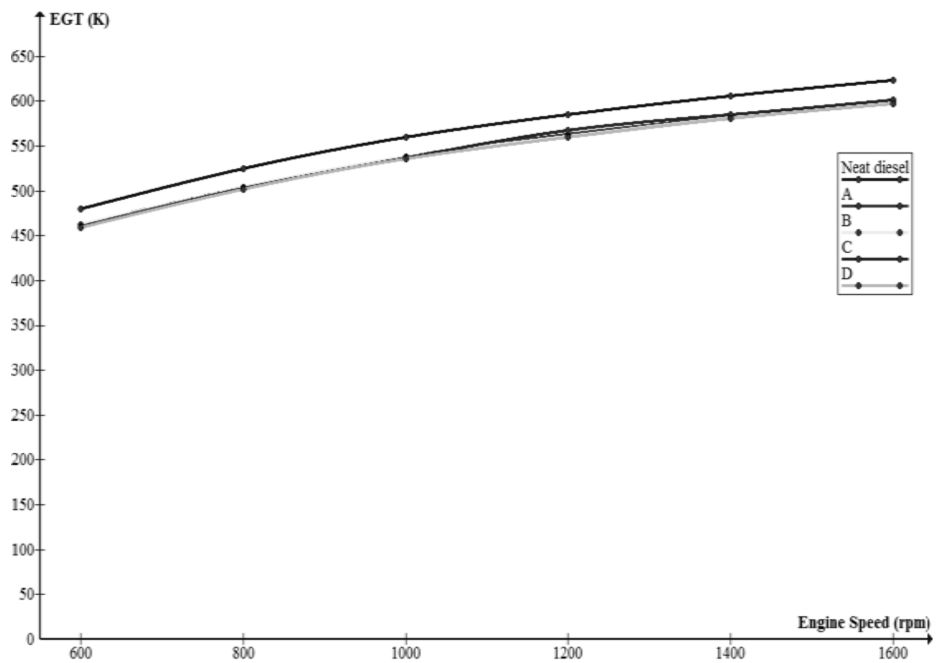

Figure 5.29: Wide Range Ratio EW50–Diesel DI Effect on EGT vs. Engine Speed

Figure (5.29) show of the EGT for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the EGT by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW50–Diesel. At speed of 1200 rpm, the EGT is decreased by 3.8%, 4.0%, 2.9% and 4.2% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the EGT compared to other cases of EW50–Diesel. However, EW50–Diesel has a tendency to oxidize and it also has poor low-temperature properties. A shorter ignition delay implies a lower rate of pressure rise and lower peak temperature, resulting in reduced NOx, and PM emissions. One of the more obvious effects of running on a low cetane number fuel is an increase in engine noise. On the other hand, the addition of EW50 leads to decrease in the viscosity of the blended fuel which is positive to form more air–fuel mixture and result in a larger percentage of fuel burned in the premixed burning phase. The trend of simulation are very similar to those of experiment, show on **Figure (4.33)** wide range ratio EW50–Diesel DI effect on EGT vs. engine speed.

5.2.29. Effect of EW75–Diesel DI on Exhaust Gas Temperature

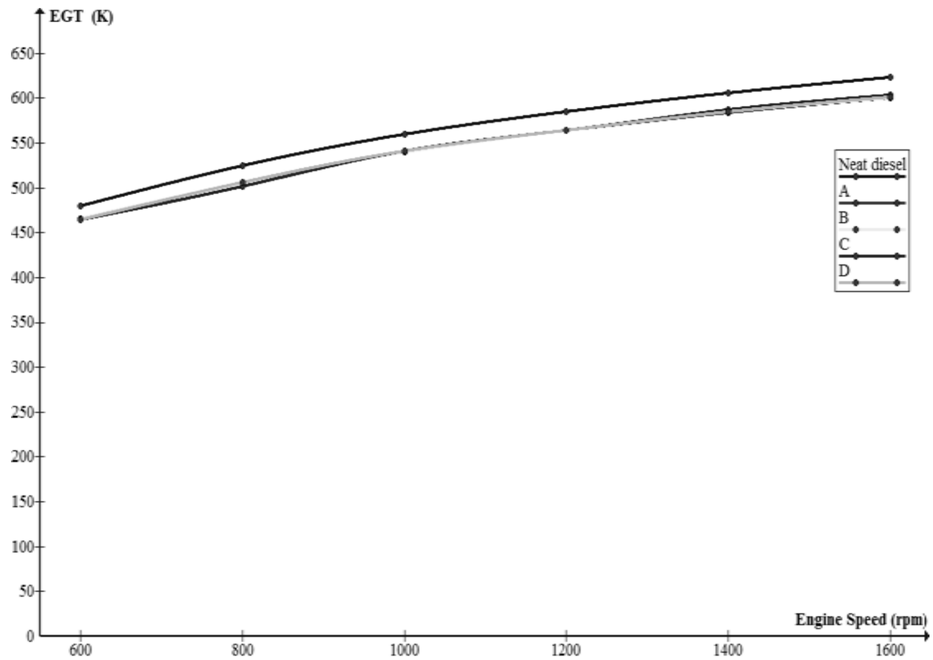

Figure 5.30: Wide Range Ratio EW75–Diesel DI Effect on EGT vs. Engine Speed

Figure (5.30) show of the EGT for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that EGT by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of EW75–Diesel. At speed of 1200 rpm, EGT is decreased by 3.5%, 3.5%, 3.5% and 3.4% with A, B, C, and D respectively. It can be seen that “D” has higher decreases in EGT compared to other cases of EW75–Diesel. The reason for the reduction in EGT is due to the lower calorific value of EW75–Diesel fuel as compared to the Neat Diesel and lower temperature, at the end of compression. Lower exhaust loss may be the possible reason for higher performance. The same study also pointed out lack of sufficient data on fumigation studies, when compared to experimental results **Figure (4.34)** of wide range ratio

EW75–Diesel DI effect on EGT vs. engine speed. Similarly, there have been also few reports on the dual fuel operation of EW75–Diesel, given the less viscous and lower cetane fuels are best suited to be premixed with inlet air for their effective operation in a diesel engine.

5.2.30. Effect of Ethanol–Diesel DI on Exhaust Gas Temperature

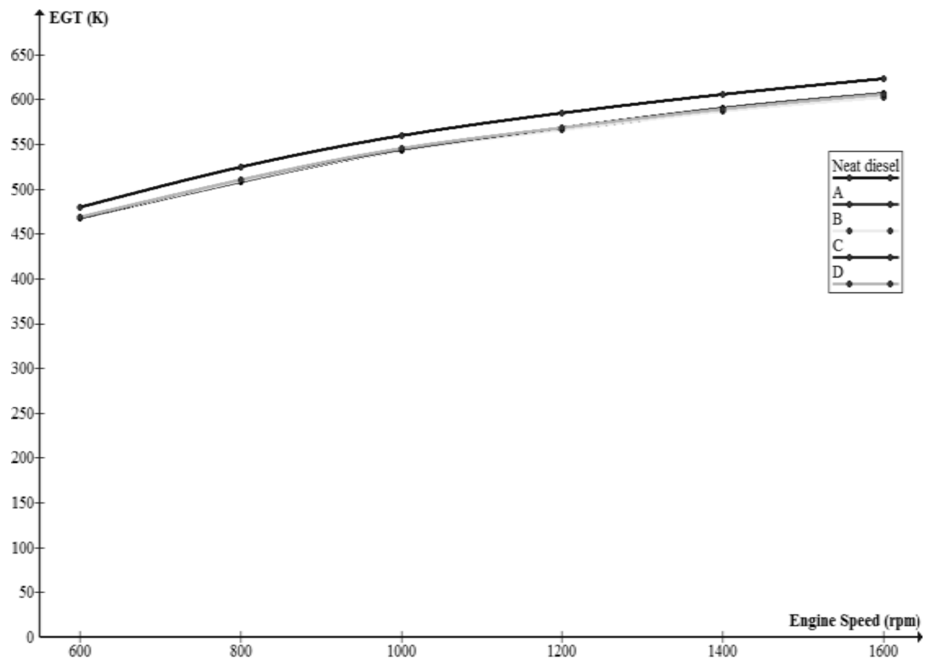

Figure 5.31: Wide Range Ratio Ethanol–Diesel DI Effect on EGT vs. Engine Speed

Figure (5.31) show of EGT for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that EGT by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Ethanol–Diesel. At speed of 1200 rpm, the EGT is decreased by 3.1%, 3.0%, 2.8% and 2.7% with A, B, C, and D respectively. It can be seen that “D” has higher decreases in the EGT compared to other cases of Ethanol–Diesel, which can be attributed to a lower cylinder gas temperature and lower combustion duration. **Figure (4.35)** shows the experimental results of wide range ratio Ethanol–Diesel DI effect on EGT vs. engine speed dissimilar simulation between speeds (800-1200) rpm.

5.3. Effect of Ethanol/Water Addition to Diesel Fuel on Performance Engine

5.3.1. Effect of Water–Diesel DI on Brake Specific Fuel Consumption

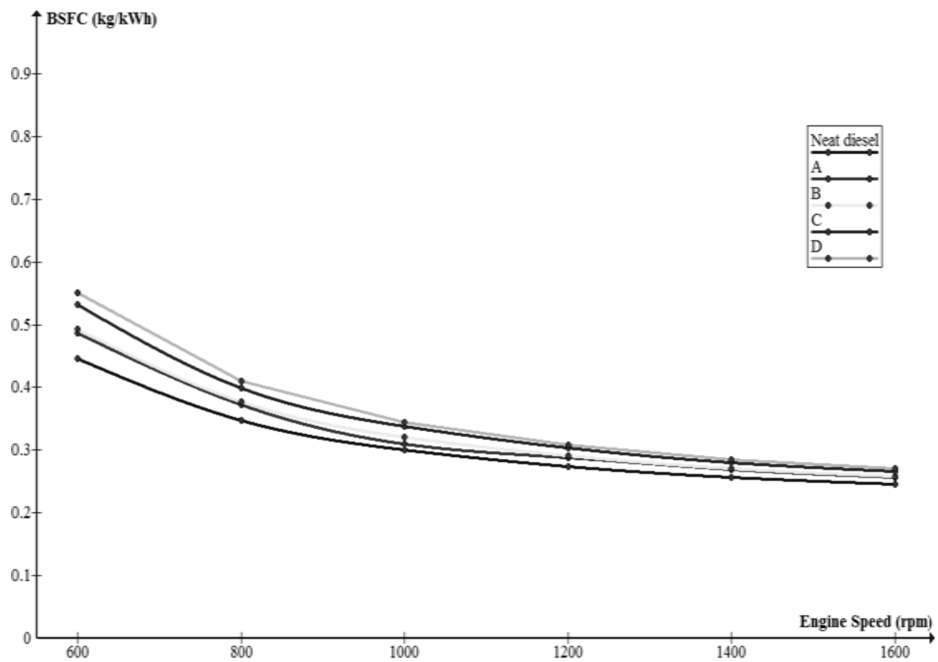

Figure 5.32: Wide Range Ratio Water–Diesel DI Effect on BSFC vs. Engine Speed

Figure (5.32) show of BSFC for the Neat Diesel fuel and Water–Diesel fuel, at the six speeds considered. One can observe that the BSFC by all Water–Diesel is higher than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Water–Diesel. At speed of 1200 rpm, the BSFC is increased by 5.4%, 6.3%, 10.7% and 12.8% with A, B, C, and D respectively. It can be seen that “D” has higher increases in the BSFC compared to other cases of Water–Diesel. It is generally accepted that fuel consumption is proportional to volumetric energy density of the fuel based upon the lower and net heating value. In another experimental study **Figure (4.37)** it was found that the BSFC of the engine obtained with Water–Diesel is so close to the results obtained with simulation.

5.3.2. Effect of EW25–Diesel DI on Brake Specific Fuel Consumption

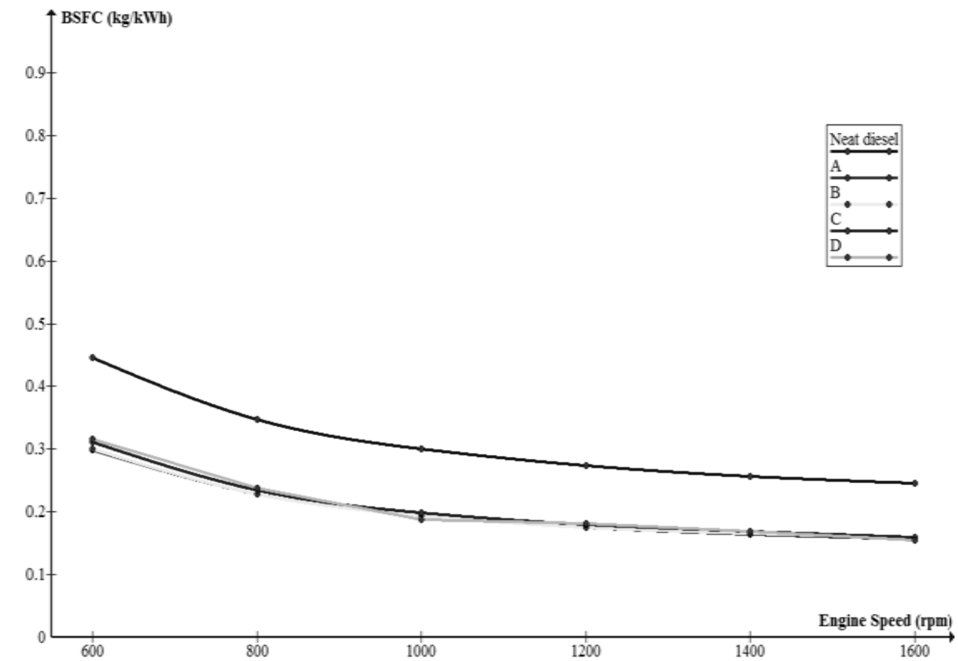

Figure 5.33: Wide Range Ratio EW25–Diesel DI Effect on BSFC vs. Engine Speed

Figure (5.33) show of the BSFC for the Neat Diesel fuel and EW25–Diesel fuel, at the six speeds considered. One can observe that BSFC by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of EW25–Diesel. At of speed 1200 rpm, BSFC is increased by 35.8%, 35.6%, 34.4% and 33.8% with A, B, C, and D respectively. It can be seen that approximate in BSFC compared to other cases of EW25–Diesel. The combustion modeling obtained similar results, **Figure (4.38)** shows that wide range ratio EW25–Diesel DI effect on BSFC vs. engine speed was obviously reduced.

5.3.3. Effect of EW50–Diesel DI on Brake Specific Fuel Consumption

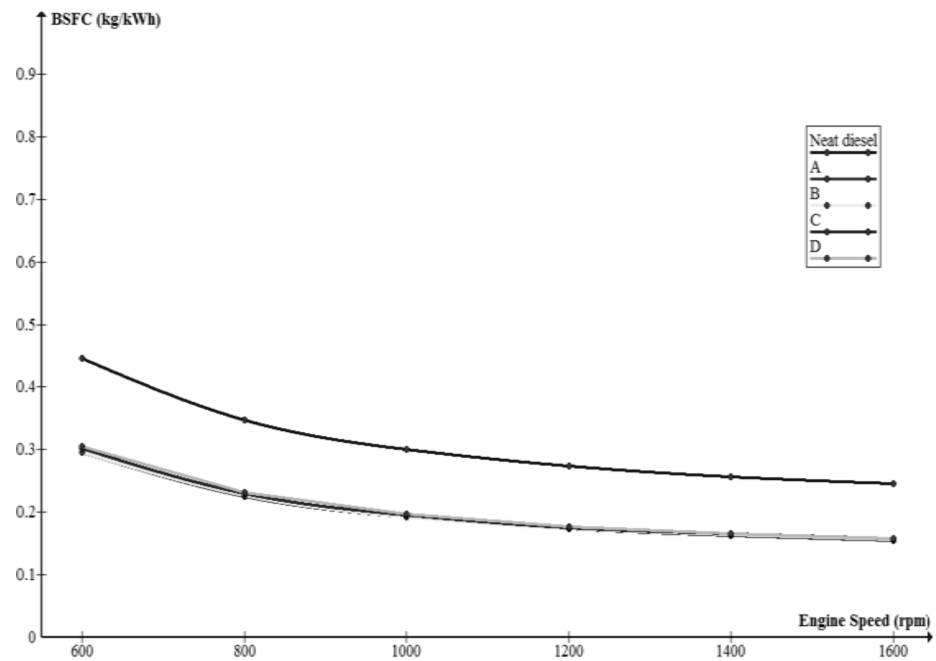

Figure 5.34: Wide Range Ratio EW50–Diesel DI Effect on BSFC vs. Engine Speed

Figure (5.34) show of BSFC for the Neat Diesel fuel and EW50–Diesel fuel, at the six speeds considered. One can observe that the BSFC by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW50–Diesel. At speed of 1200 rpm, the BSFC is increased by 36.4%, 36.2%, 35.7% and 35.2% with A, B, C, and D respectively. It can be seen that approximate in the BSFC compared to other cases of EW50–Diesel. To more clearly analyze the BSFC of each fuel blend, which has also been defined in chapter 4, is shown on **Figure (4.39)** wide range ratio EW50–Diesel DI effect on BSFC vs. engine speed. These trends could be investigation conducted by experimental test.

5.3.4. Effect of EW75–Diesel DI on Brake Specific Fuel Consumption

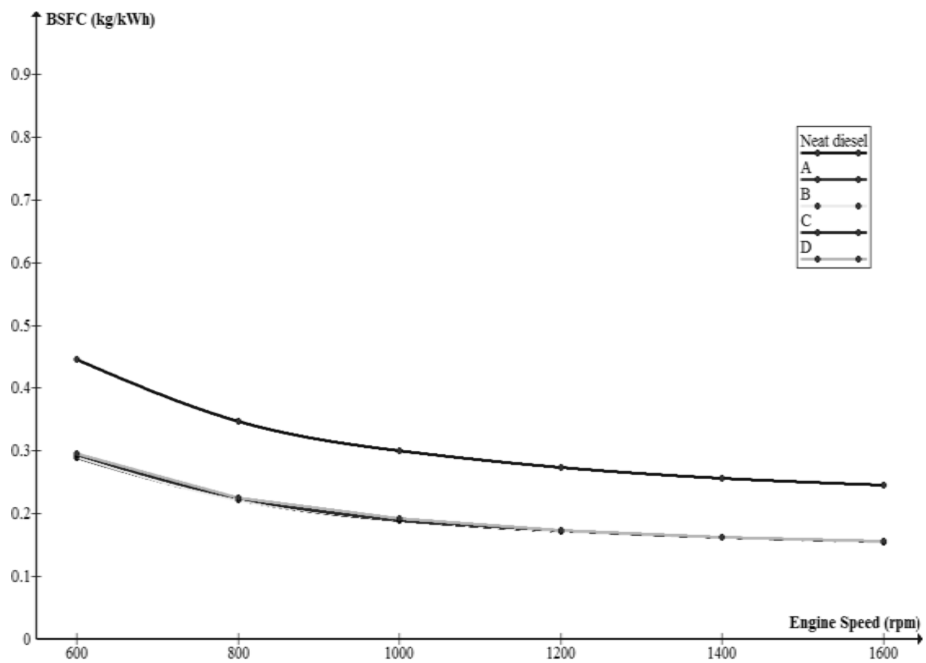

Figure 5.35: Wide Range Ratio EW75–Diesel DI Effect on BSFC vs. Engine Speed

Figure (5.35) show of BSFC for the Neat Diesel fuel and EW75–Diesel fuel, at the six speeds considered. One can observe that the BSFC by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW75–Diesel. At speed of 1200 rpm, the BSFC is increased by 37.0%, 36.9%, 36.5% and 36.4% with A, B, C, and D respectively. It can be seen that approximate in the BSFC compared to other cases of EW75–Diesel. Similar reduction ratios of BSFC were obtained at experimental tests as shown on **Figure (4.40)** wide range ratio EW75–Diesel DI effect on BSFC vs. engine speed.

5.3.5. Effect of Ethanol–Diesel DI on Brake Specific Fuel Consumption

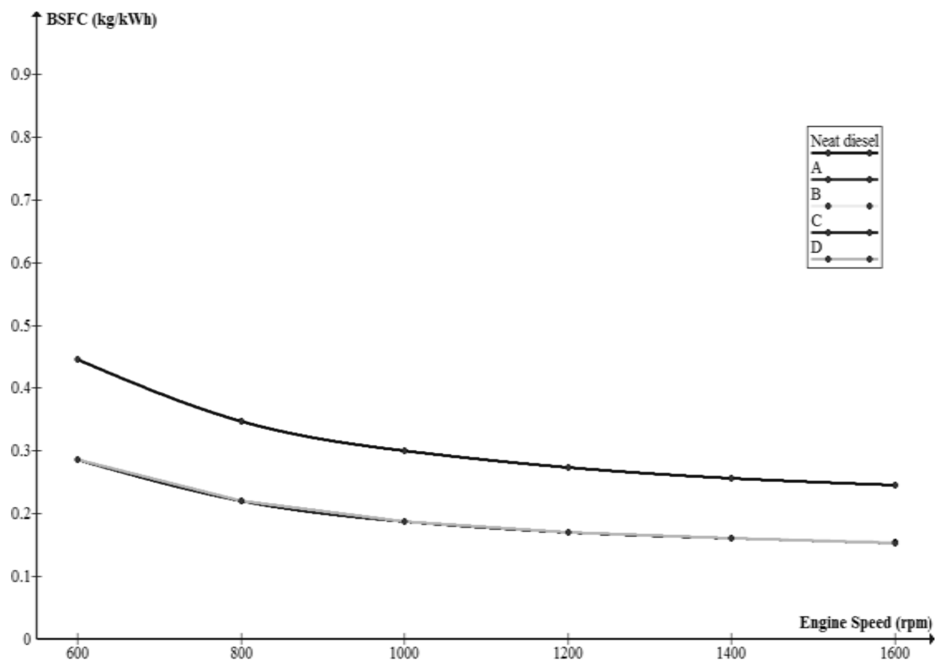

Figure 5.36: Wide Range Ratio Ethanol–Diesel DI Effect on BSFC vs. Engine Speed

Figure (5.36) show of BSFC for the Neat Diesel fuel and Ethanol–Diesel fuel, at the six speeds considered. One can observe that the BSFC by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Ethanol–Diesel. At speed of 1200 rpm, the BSFC is increased by 37.5%, 37.4%, 37.5% and 37.5% with A, B, C, and D respectively. It can be seen that approximate in the BSFC compared to other cases of Ethanol–Diesel. Similar results can be found in experimental studies as shown on **Figure (4.41)** wide range ratio Ethanol–Diesel DI effect on BSFC vs. engine speed.

5.3.6. Effect of Water–Diesel DI on Brake Power

Figure (5.37) show of the BP for the Neat Diesel fuel and Water–Diesel fuel, at the six speeds considered. One can observe that BP by all Water–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Water–Diesel. At speed of 1200 rpm, the BP is decreased by 5.0%, 5.8%, 9.5% and 11.2% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in BP compared to other cases of Water–Diesel. The general trend of decrease in BP with the Water–Diesel is observed for the simulation similar to the practical tests at all the operation condition show **Figure (4.49)** wide range ratio Water–Diesel DI effect on BP vs. engine speed.

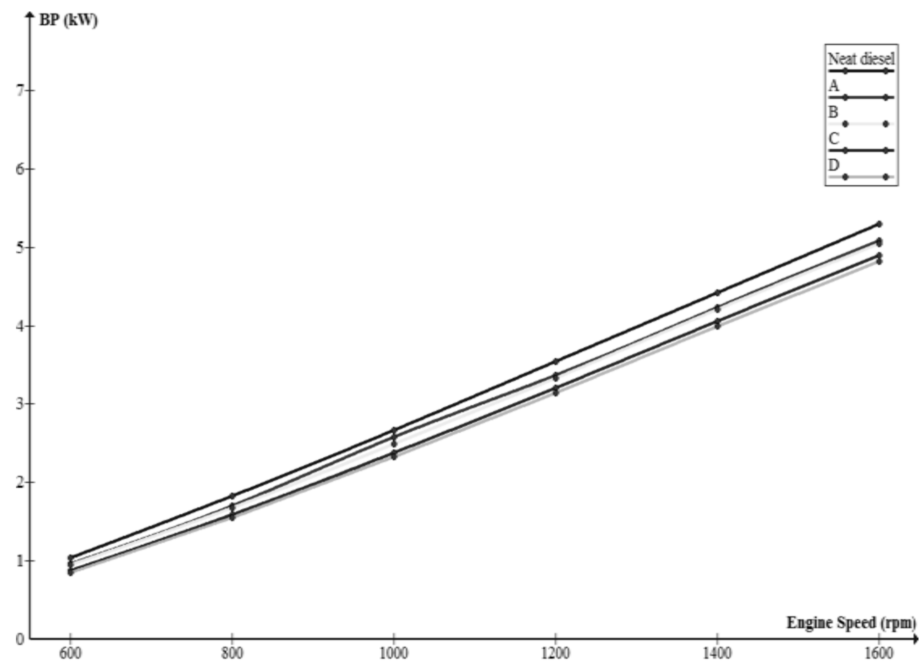

Figure 5.37: Wide Range Ratio Water–Diesel DI Effect on BP vs. Engine Speed

5.3.7. Effect of EW25–Diesel DI on Brake Power

Figure (5.38) show of BP for the Neat Diesel fuel and EW25–Diesel fuel, at the six speeds considered. One can observe that the BP by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW25–Diesel. At speed of 1200 rpm, the BP is decreased by 15.2%, 15.5%, 16.6% and 17.2% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the BP compared to other cases of EW25–Diesel. Dissimilar results were experimental tests show **Figure (4.50)** wide range ratio EW25–Diesel DI effect on BP vs. engine speed; due to setup has transmitters for air and fuel flow measurements, process indicator and engine indicator.

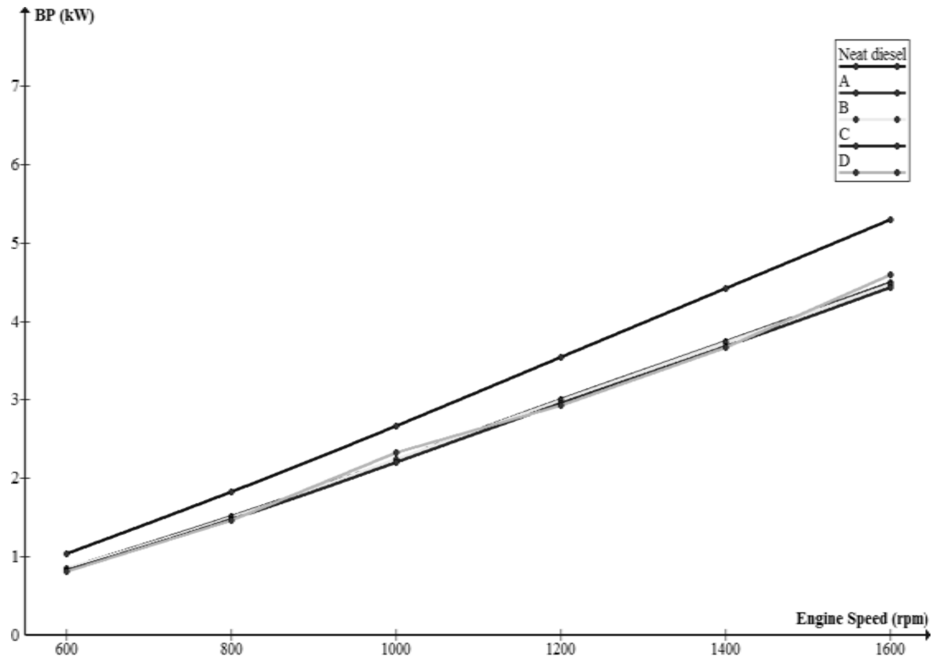

Figure 5.38: Wide Range Ratio EW25–Diesel DI Effect on BP vs. Engine Speed

5.3.8. Effect of EW50–Diesel DI on Brake Power

Figure (5.39) show of BP for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the BP by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW50–Diesel. At speed of 1200 rpm, the BP is decreased by 14.0%, 14.1%, 11.7% and 14.7% with A, B, C, and D respectively. It can be seen that “D” has higher decreases in the BP compared to other cases of EW50–Diesel. Different trend for practical test at same operation condition show **Figure (4.51)** wide range ratio EW50–Diesel DI effect on BP vs. engine speed, this due for cooling water and calorimeter water flow.

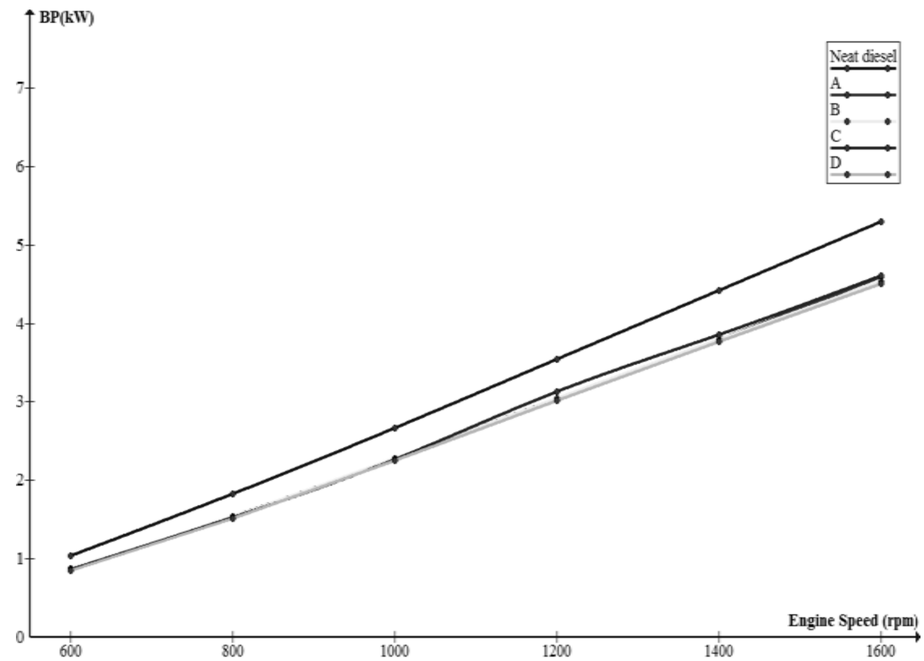

Figure 5.39: Wide Range Ratio EW50–Diesel DI Effect on BP vs. Engine Speed

5.3.9. Effect of EW75–Diesel DI on Brake Power

Figure (5.40) show of the BP for the Neat Diesel fuel and EW75–Diesel fuel, at the six speeds considered. One can observe that the BP by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW75–Diesel. At speed of 1200 rpm, the BP is decreased by 12.9%, 12.8%, 12.7% and 12.7% with A, B, C, and D respectively. It can be seen that “D” has higher decreases in the BP compared to other cases of EW75–Diesel. Various trends for practical test at same operation condition **Figure (4.52)** shows wide range ratio EW75–Diesel DI effect on BP vs. engine speed. This case indicates the accuracies of the measurements and the uncertainties in the calculated results.

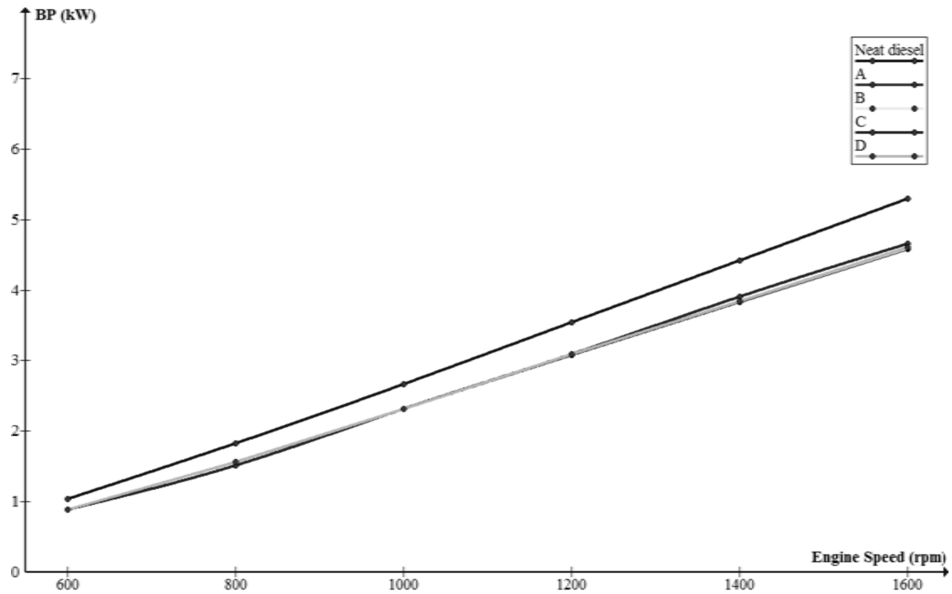

Figure 5.40: Wide Range Ratio EW75–Diesel DI Effect on BP vs. Engine Speed

5.3.10. Effect of Ethanol–Diesel DI on Brake Power

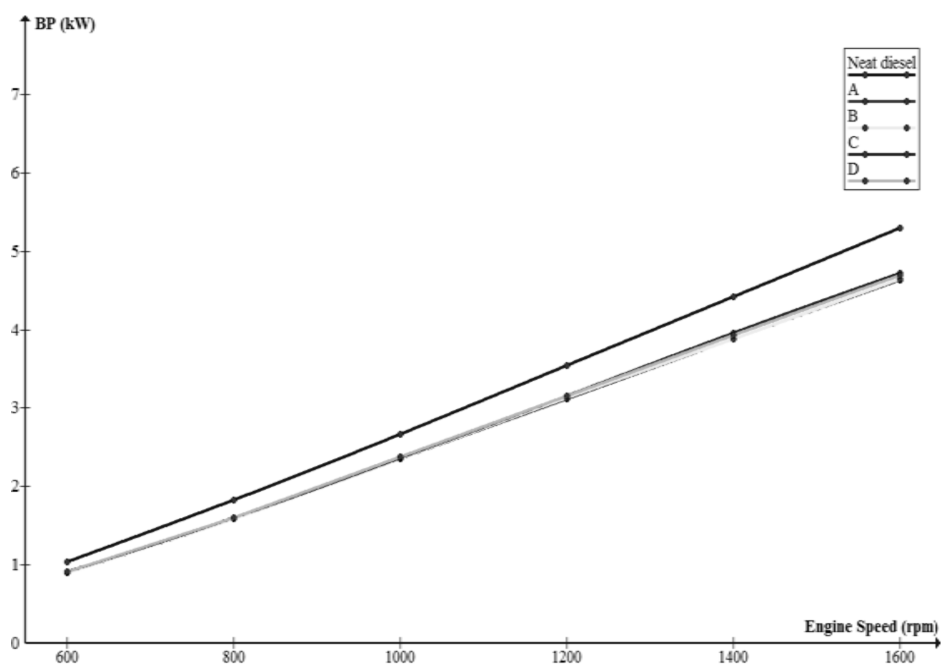

Figure 5.41: Wide Range Ratio Ethanol–Diesel DI Effect on BP vs. Engine Speed

Figure (5.41) show of BP for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that BP by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with very small change percentage of the Ethanol–Diesel. At speed of 1200 rpm, the BP is decreased by 11.9%, 11.7%, 11.0% and 10.8% with A, B, C, and D respectively. It can be seen that “C” has higher decreases in the BP compared to other cases of Ethanol–Diesel. It can be shows **Figure (4.53)** wide range ratio Ethanol–Diesel DI effect on BP vs. engine speed, that the experimental heat release analysis results for the relevant combustion mechanism, combined with the widely differing physical and chemical properties of the ethanol against the ones for the diesel fuel, were used to aid the correct interpretation of the observed simulation engine behavior.

5.3.11. Effect of Water–Diesel DI on Brake Thermal Efficiency

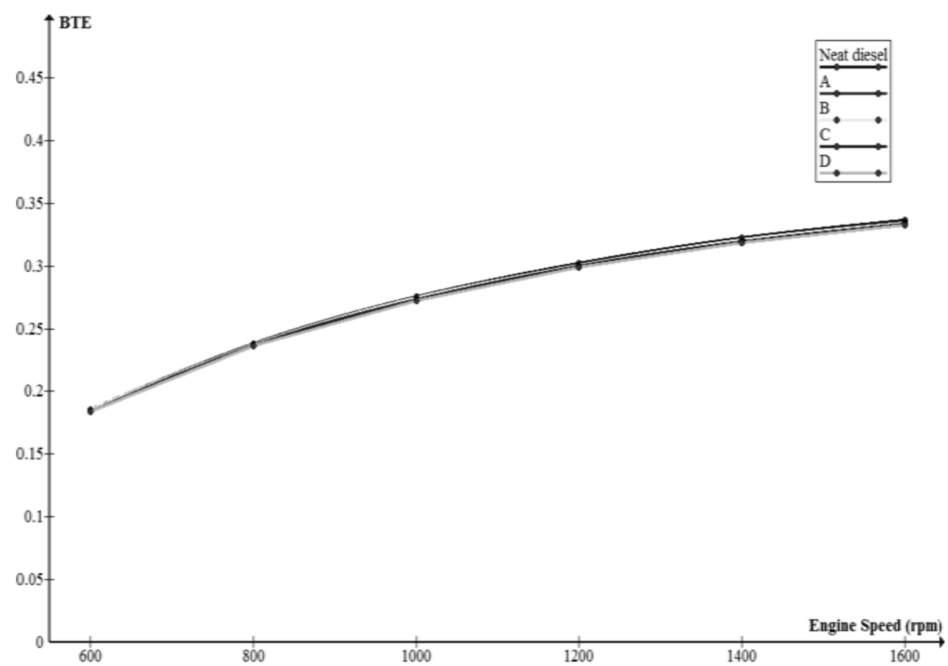

Figure 5.42: Wide Range Ratio Water–Diesel DI Effect on BTE vs. Engine Speed

Figure (5.42) show of BTE for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that BTE by all Water–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with very small change percentage of the Water–Diesel. At speed 1200 rpm, BTE decreased by 0.50%, 0.52%, 0.87% and 1.1% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the BTE compared to other cases of Water–Diesel. The BTE results of experimental and simulation varying with Water–Diesel flow rate at different conditions are shown on **Figure (4.43)** wide range ratio Water–Diesel DI effect on BTE vs. engine speed.

5.3.12. Effect of EW25–Diesel DI on Brake Thermal Efficiency

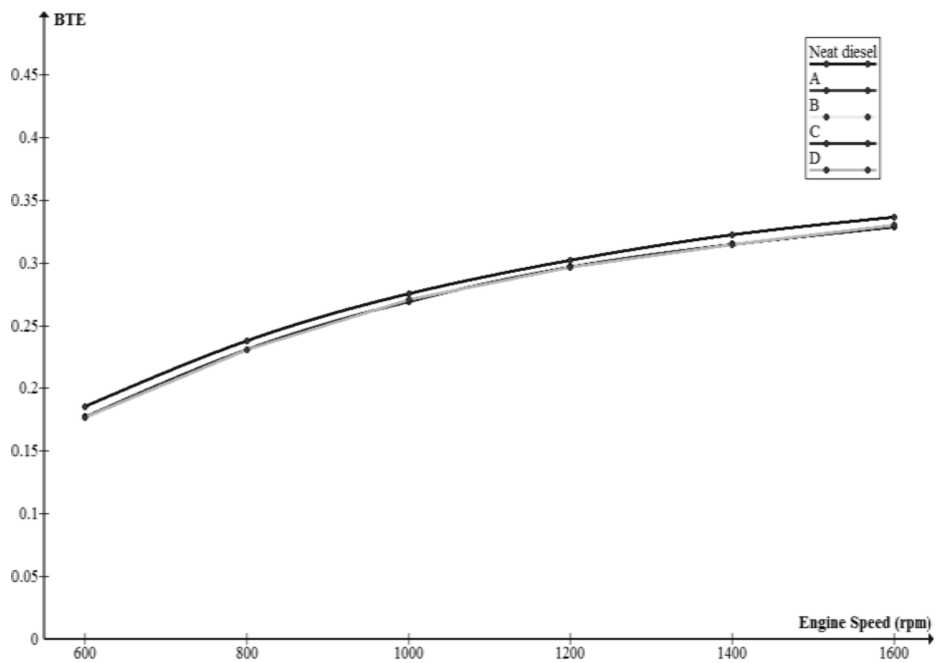

Figure 5.43: Wide Range Ratio EW25–Diesel DI Effect on BTE vs. Engine Speed

Figure (5.43) show of BTE for the Neat Diesel fuel and EW25–Diesel, at the six speeds considered. One can observe that the BTE by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with very small change percentage of the EW25–Diesel. At speed of 1200 rpm, the BTE is decreased by 1.5%, 1.5%, 1.8% and 1.9% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the BTE compared to other cases of EW25–Diesel.

Different tendencies are obtained for the experimental results operating points; it seems that the **Figure (4.44)** wide range ratio EW25–Diesel DI Effect on BTE vs. engine speed.

5.3.13. Effect of EW50–Diesel DI on Brake Thermal Efficiency

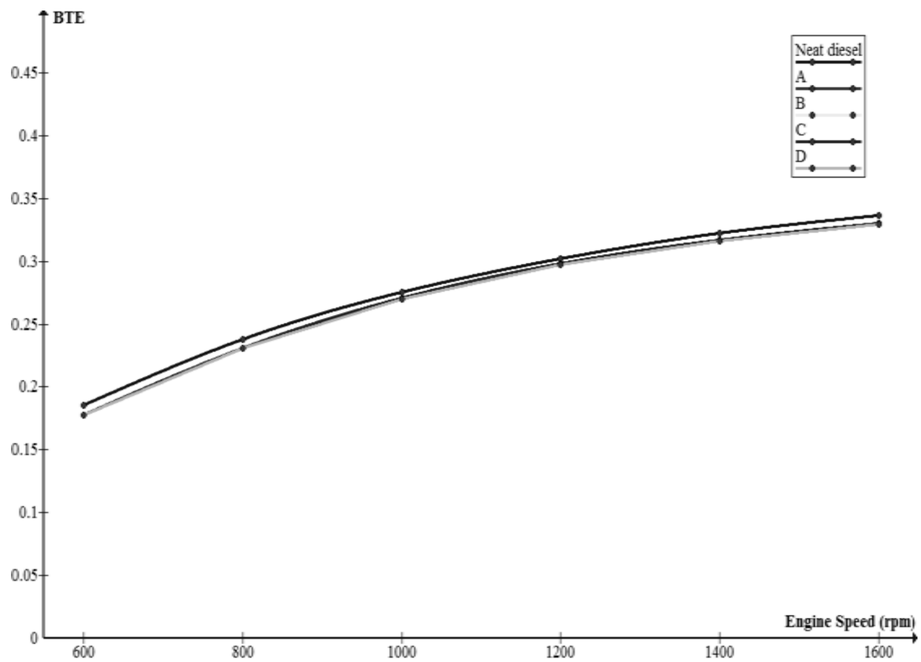

Figure 5.44: Wide Range Ratio EW50–Diesel DI Effect on BTE vs. Engine Speed

Figure (5.44) show of BTE for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that BTE by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with very small change percentage of the EW50–Diesel. At speed of 1200 rpm, BTE is decreased by 1.4%, 1.4%, 1.2% and 1.5% with A, B, C, and D respectively. It can be seen that “D” has higher decreases in the BTE compared to other cases of EW50–Diesel. This phenomenon can be interpreted as extending the experimental combustion **Figure (4.46)** shows wide range ratio EW50–Diesel DI effect on BTE vs. engine speed a complete combustion reaction when using the EW50–Diesel, which could be a result of a higher BTE.

5.3.14. Effect of EW75–Diesel DI on Brake Thermal Efficiency

Figure (5.45) show of the BTE for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the BTE by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with very small change percentage of the EW75–Diesel. At speed of 1200 rpm, BTE is decreased by 1.3%, 1.3%, 1.3% and 1.3% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the BTE compared to other cases of EW75–Diesel. Opposite trends were observed when comparing of wide range ratio EW75–Diesel DI effect on BTE vs. engine speed on practical tests, because of the presence of oxygen in the compound, **Figure (4.46)**.

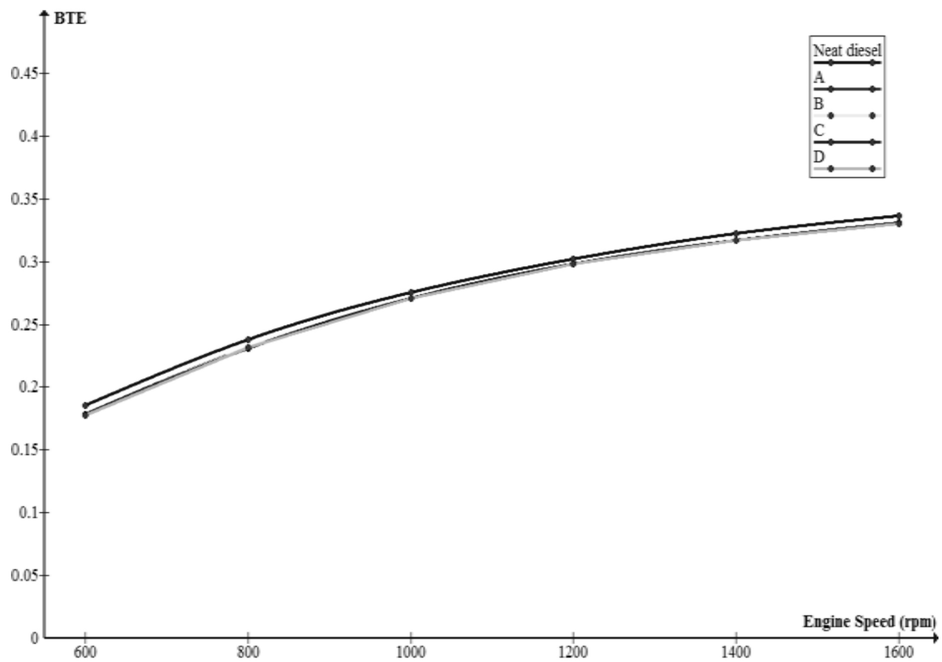

Figure 5.45: Wide Range Ratio EW75–Diesel DI Effect on BTE vs. Engine Speed

5.3.15. Effect of Ethanol–Diesel DI on Brake Thermal Efficiency

Figure (5.46) show of BTE for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that the BTE by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with very small change percentage of the Ethanol–Diesel. At speed 1200 rpm, BTE decreased by 1.2%, for all cases Ethanol–Diesel. It can be seen that "D" has higher decreases in the BTE compared to other cases of Ethanol–Diesel. The experimental results it seen **Figure (4.47)** wide range ratio Ethanol–Diesel DI effect on BTE vs. engine speed were not uniform in the computer simulation.

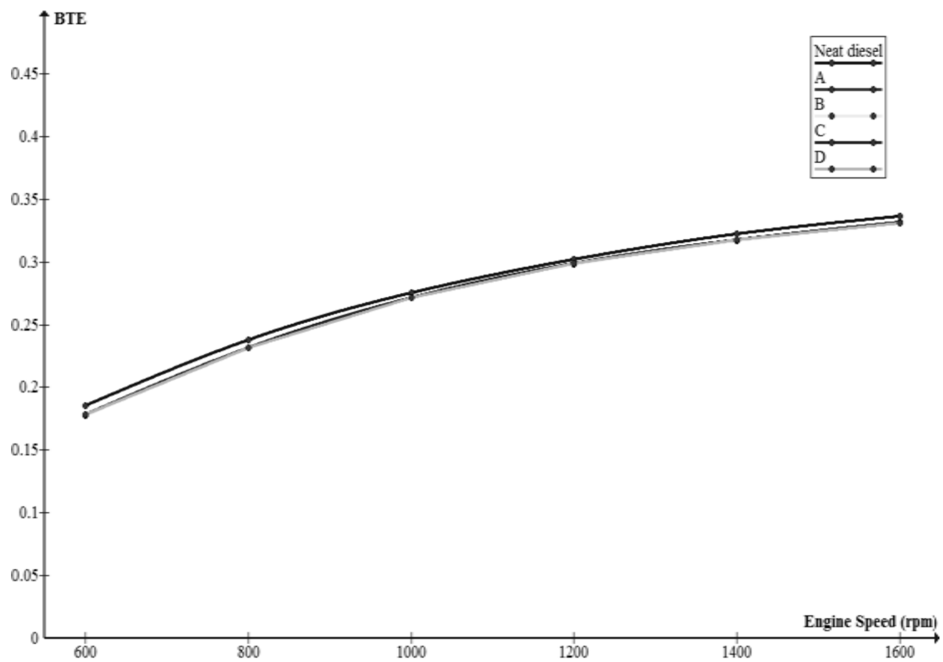

Figure 5.46: Wide Range Ratio Ethanol–Diesel DI Effect on BTE vs. Engine Speed

5.3.16. Effect of Water–Diesel DI on Brake Mean Effective Pressure

Figure (5.47) show of (BMEP) for the Neat Diesel fuel and Water–Diesel, at the six speeds considered. One can observe that the BMEP by all Water–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Water–Diesel. At speed of 1200 rpm, the BMEP is decreased by 5.0%, 5.8%, 9.5% and 11.2% with A, B, C, and D respectively. It can be seen that "D" has higher decreases in the BMEP compared to other cases of Water–Diesel. The main reason may be due to the higher volatility of Neat Diesel which speeds up the mixing velocity of air/fuel mixture, improves the combustion process and increases the combustion efficiency.

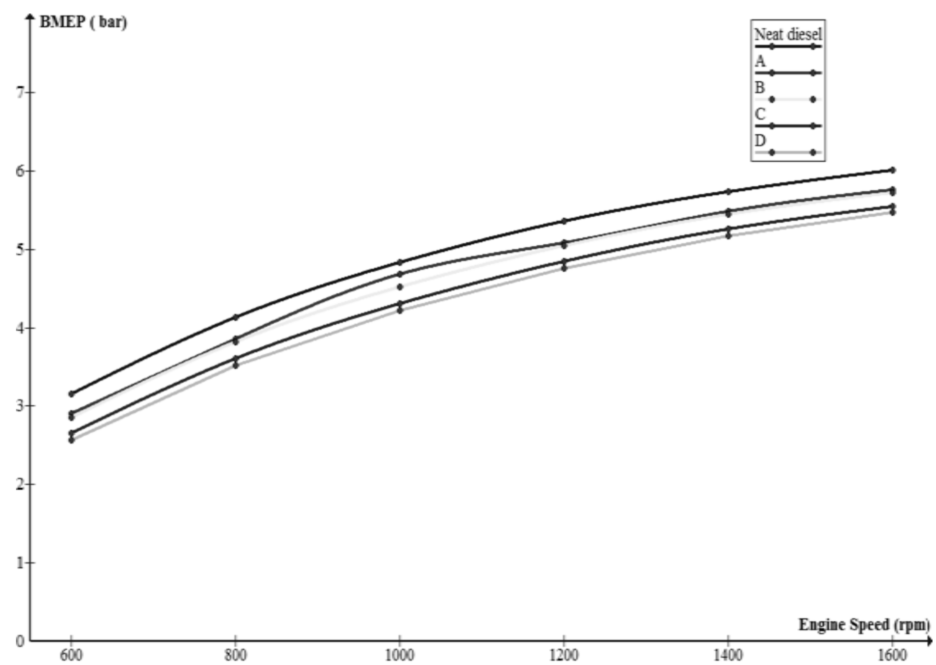

Figure 5.47: Wide Range Ratio Water–Diesel DI Effect on BMEP vs. Engine Speed

5.3.17. Effect of EW25–Diesel DI on Brake Mean Effective Pressure

Figure (5.48) show of (BMEP) for the Neat Diesel fuel and EW25–Diesel, at the six speeds considered. One can observe that the BMEP by all EW25–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the EW25–Diesel.At speed 1200 rpm, the BMEP decreased by 15.1%, 15.5%, 9.5% and 11.2% with A, B, C, and D respectively. It can be seen most conditions that "D" has higher decreases in the

BMEP compared to other cases of EW25–Diesel. Nevertheless, cetane number and vapor pressure are normally lower in organic solvents than in conventional Neat Diesel; therefore, the percentages of solvents in blends must also be limited. However, it is quite easy for pure acetone to absorb water because of the hydrogen bonding.

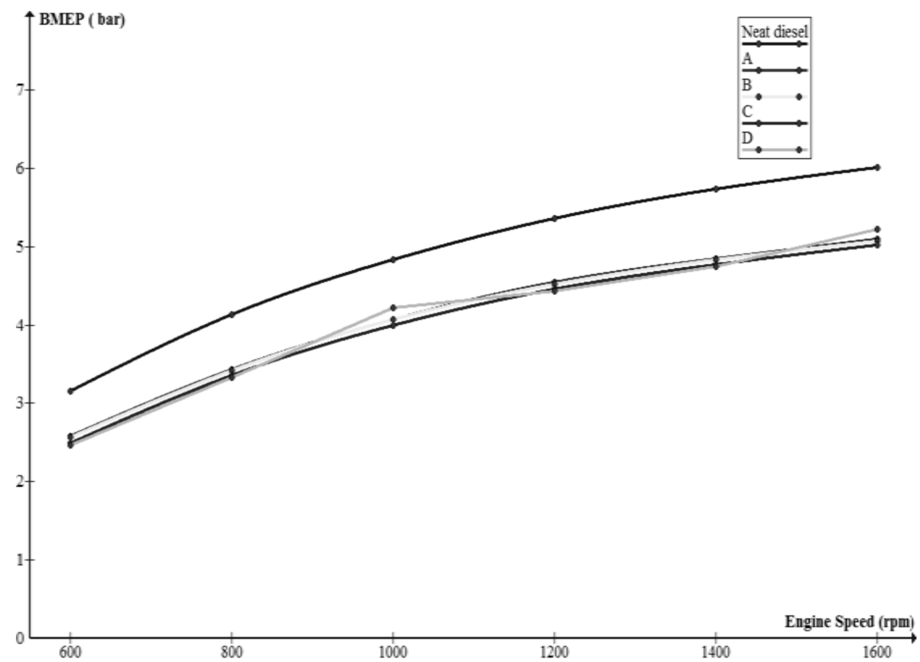

Figure 5.48: Wide Range Ratio EW25–Diesel DI Effect on BMEP vs.

Engine Speed

5.3.18. Effect of EW50–Diesel DI on Brake Mean Effective Pressure

Figure (5.49) show of (BMEP) for the Neat Diesel fuel and EW50–Diesel, at the six speeds considered. One can observe that the BMEP by all EW50–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of EW50–Diesel. At speed of 1200 rpm, the BMEP is decreased by 14.0%, 14.1%, 11.7% and 14.7% with A, B, C, and D respectively. It can be seen most conditions that “D” has higher decreases in the BMEP compared to other cases of EW50–Diesel, due to increase incomplete combustion, increase in mechanical frictional losses and reduction in VE.

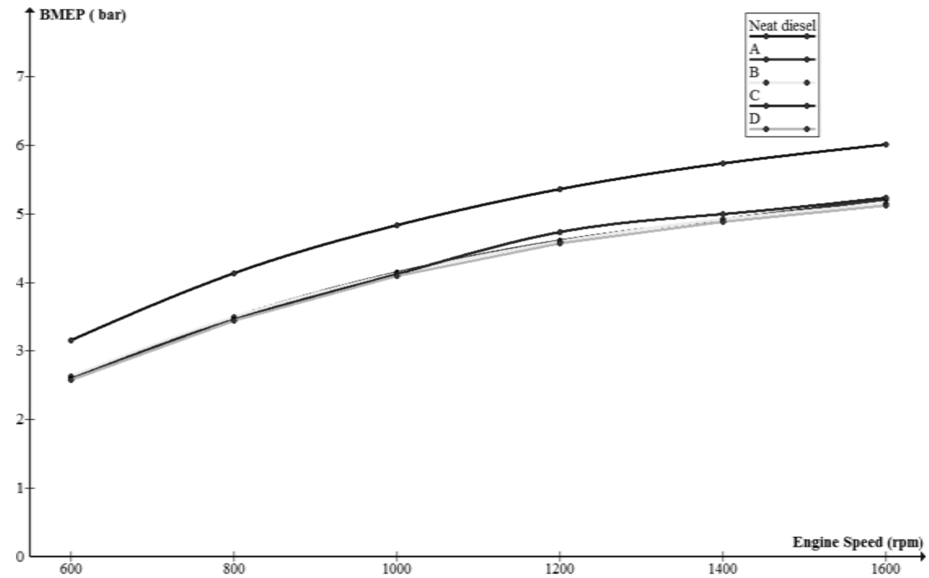

Figure 5.49: Wide Range Ratio EW50–Diesel DI Effect on BMEP vs. Engine Speed

5.3.19. Effect of EW75–Diesel DI on Brake Mean Effective Pressure

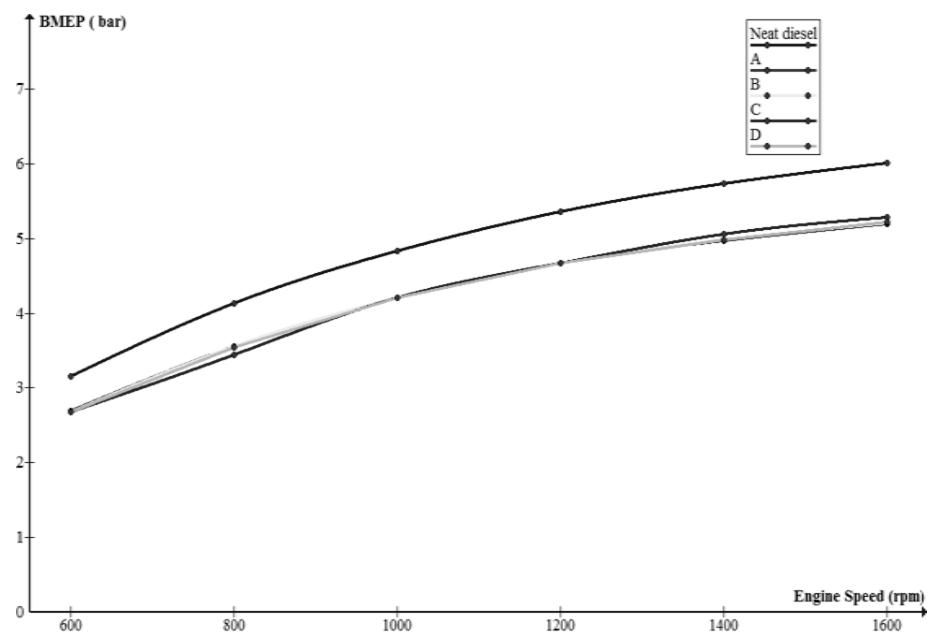

Figure 5.50: Wide Range Ratio EW75–Diesel DI Effect on BMEP vs. Engine Speed

Figure (5.50) show of (BMEP) for the Neat Diesel fuel and EW75–Diesel, at the six speeds considered. One can observe that the BMEP by all EW75–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of EW75–Diesel. At speed of 1200 rpm, the BMEP is decreased by 12.9%, 12.9%, 12.8% and 12.7% with A, B, C, and D respectively. It can be seen most conditions that "D" has higher decreases in the BMEP compared to other cases of EW75–Diesel. This may be due to increasing rate of the combustion pressure caused by an increase in ethanol blending slows because of the low cetane number and low heating values of fuel blends.

5.3.20. Effect of Ethanol–Diesel DI on Brake Mean Effective Pressure

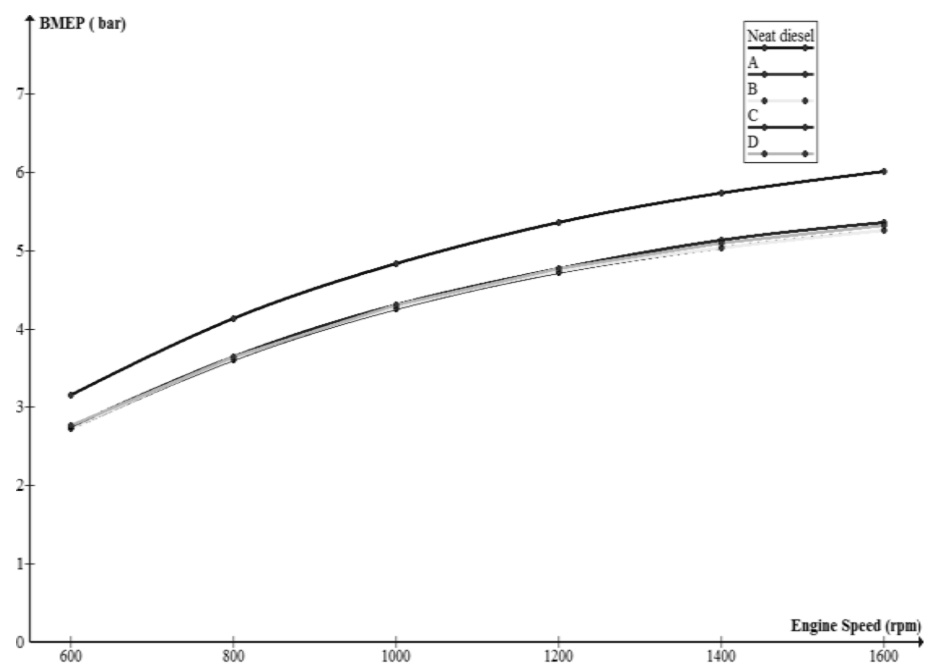

Figure 5.51: Wide Range Ratio Ethanol–Diesel DI Effect on BMEP vs. Engine Speed

Figure (5.51) show of (BMEP) for the Neat Diesel fuel and Ethanol–Diesel, at the six speeds considered. One can observe that BMEP by all Ethanol–Diesel is lower than the ones for the corresponding Neat Diesel fuel case, with this decrease being higher the higher the percentage of the Ethanol–Diesel. At speed of 1200 rpm, the BMEP is decreased by 11.9%, 11.7%, 11.0% and 10.8% with A, B, C, and D respectively. It can be seen most conditions that “A” has higher decreases in the BMEP compared to other cases of Ethanol–Diesel. The reason may be longer ignition delay, and shorter combustion duration at the same operation conditions.

CHAPTER SIX

Conclusions and Recommendations

6. Conclusions and Recommendations for Future Work

6.1. Conclusions

The thesis carried out employing both theoretical considerations and experimental investigations to study the effect of Ethanol/Water blends addition to diesel fuel on diesel emissions levels and engine performance. The combustion and emission fundamentals of Ethanol/Water ratio blend with (0-25-50-75-100)% (by vol.) ethanol (i.e., EW25) on a single-cylinder naturally aspirated, four-stroke, water cooled, and direct-injection diesel engine with a hemispherical bowl in piston combustion chamber were investigated by experiment and simulation. The engine characteristics were determined experimentally on an engine test bed and numerically by using the DIESEL-RK software and mixed controlled combustion model. All the measurements and simulations were made under full load and at various engine speeds, from 600 to 1600 rpm with sequence 200 rpm. The numerical results are in good agreement with the experimental data on the engine performance and combustion characteristics. Additionally, the effect of Ethanol/Water–Diesel was evaluated experimentally, and was compared with Neat Diesel fuel (i.e., EW25–Diesel); the mixed fuels contain 0.55–2.79 by volume fraction of diesel. The following conclusions can be drawn from the present study.

The use of Ethanol/Water–Diesel as a dual fuel causes lower NOx emissions at conditions (Water–Diesel, EW25–Diesel, EW50–Diesel, and EW75–Diesel), while yielding higher NOx emissions at Ethanol–Diesel compared with the use of Neat Diesel. In this case the simulated results of NOx and NO_2 emissions of Ethanol/Water–Diesel are lower than Neat Diesel results. However, the simulated and experimental results for NOx emissions of Ethanol/Water–Diesel are in good agreement compare with Neat Diesel.

The use of Ethanol/Water–Diesel as a dual fuel causes higher CO and O_2 emissions at all conditions experimental. On the contrary, at all conditions CO emissions are higher due to higher latent heat of vaporization and higher viscosity of ethanol fuels, and NOx emissions are lower at low loads due to lower flame temperature.

The use of Ethanol/Water–Diesel, as a dual fuel, causes lower UBHC emissions at conditions EW50–Diesel and EW75–Diesel, while yielding higher UBHC emissions at other (DI) compared with the use of Neat Diesel.

The CO_2 emissions for Ethanol–Diesel, EW50–Diesel, EW75–Diesel at case (A, D), and Water–Diesel at case (B, C) were reduced while comparing with Neat Diesel, but CO_2 emission was observed high, when EW75–Diesel at case (B, C), EW25–Diesel, and Water–Diesel at case (A, D) are used. As a result from numerical simulation showed that, the CO_2 emissions give lower values of the fuels containing ethanol than that of the Neat Diesel fuel, while Water–Diesel decrease CO_2 emissions. The Ethanol/Water–Diesel as a dual fuel blends also show a similar trend. In the comparison of CO_2 emission among the simulated and experimental test fuels, reductions of CO_2 emissions can be observed with increases in the ethanol blending ratio. In this investigation performing under the engine load condition, the cooling effect of the ethanol in the blended fuels reduces the temperature of the combustion cylinder and suppresses the formation of NOx.

The EGT of the blended fuels decreases with Ethanol/Water–Diesel dual fuel and Neat Diesel fuel due to the lower heat values of the Ethanol/Water–Diesel. The numerically obtained results agreed relatively well with the experimental results.

Emissions of PM, SE, and Bosch Smoke Number by a DIESEL-RK simulation software was estimated using Ethanol/Water–Diesel dual fuels stability blends and Neat Diesel. This indicates that, to some extent, numerical simulations may be used as a replacement for expensive experimental measurements.

The PM emissions from the engine fuelled by the Neat Diesel or Water–Diesel were all null value; but the PM emissions became more after speed 1000 rpm as the Ethanol/Water conditions. Similar results can be seen for Bosch Smoke Number.

The SE emissions from the engine fuelled by the Water–Diesel were all lower than that Neat Diesel, As the ethanol contents in the blended fuels increase, the SE also tends to increase after speed 1000 rpm.

The BSFC of the Ethanol/Water–Diesel fuels lower than Neat Diesel fuel due to the improve combustion. It seen this is result from experimental and simulation software, except results from software case Water–Diesel higher than Neat Diesel.

BSFC form the experimental results are lower for the Ethanol/Water–Diesel. However there is little change in the BTE. There is even a slight increase in BTE for the blended fuels.

The use of Ethanol/Water–Diesel, compared with Neat Diesel, form the simulation software results, leads to a decrease in the BSFC. Regarding the BTE of the engine, there are various factors affecting it. These factors lead to small decrease in the BTE.

The use of Ethanol/Water–Diesel usually increase BP compared to Neat Diesel fuel. This result at experimental conditions, which were caused by the BP of the blend increases it, becomes more stable and the vapor of ethanol in the air increases. This effect is also observed for different water and ethanol contents. Compared to the software results, Ethanol/Water–Diesel blends decreased the brake power at all operation conditions.

The Ethanol/Water addition into air did show increase change in (VE) and effective amount of air of this engine at selected engine speeds. However, it was observed that VE decreased approximately at medium speeds.

While (λ) increases for Ethanol/Water–Diesel fuel mixture for all engine operating regions of the exponential test engine. Main reason to explain this behavior can be related to increasing consumed fuel that was influenced by increasing of fuel's oxygen.

From the simulation software, the addition of Ethanol/Water to diesel fuel in a diesel engine using dual fuel causes lower mean effective pressure compared with the use of Neat Diesel, which is an indication that either Ethanol/Water is unable to provide sufficient energy density inside the cylinder or the charge might be chocking in absence of sufficient air as Ethanol/Water has replaced some of the intake air.

6.2. Recommendations for Future Work

The researcher recommends the followings:

- Investigate limits on intake temperature, flow rate and pressure for Ethanol/Water DI to determine a viable real world operating condition.
- Study performance engine and emissions for Ethanol/Water mix on multi-cylinder, four-stroke, water cooled, and direct-injection diesel engine.

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Appendix A: Properties of Water

Property	Level
pH	6.5 to 9.0
Hardness as $CaCO_3$	30 to 750 ppm
Alkalinity as $CaCO_3$	500 ppm maximum
Total Dissolved Solids (TDS)	1500 ppm maximum
Conductivity	2400 micromhos
Chlorides	250 ppm maximum Cl (410 ppm maximum as NaCl)
Sulfates	250 ppm maximum
Silica	150 ppm maximum

*Source: Chemically Treated Circulating Water Laboratory at the University of The King Saud.

- Limit the duration of the cleaning to one day or at the most two days.
- The temperature of the solution should never exceed (37.8°C).

Appendix B: Properties of Fuels

Property	No. 2 Diesel	Ethanol
Chemical Formula	C_3 to C_{25}	C_2H_5OH
Molecular Weight	≈200	46.07
Composition, Weight %		
Carbon	84–87	52.2
Hydrogen	33–16	13.1
Oxygen	0	34.7
Specific gravity, 15° C/15° C	0.81–0.89	0.796
Density, Kg/m^3 @ 15° C	802.8–886.7	792.05
Boiling temperature, °C	187.8–343.3	77.8
Reid vapor pressure, Kpa	1.3789	15.86
Octane no.[(1)]		
Research octane no.	--	108
Motor octane no.	--	92
(R + M)/2	N/A	100
Cetane no.[(1)]	40–55	--
Water solubility, @ 21° C		
Fuel in water, volume %	Negligible	100
Water in fuel, volume %	Negligible	100
Freezing point, °C	(-40)– (-1.11)[a]	-114
Viscosity		
Centipoise @ 15° C	2.6–4.1	1.19
Flash point, closed cup, °C	73.9	12.8
Autoignition temperature, °C	≈315.6	422.8
Flammability limits, volume %		
Lower	1	4.3
Higher	6	19
Latent heat of vaporization		
KJ / litre @ 15° C	≈195	0.663
KJ/Kg @ 15° C	≈232.6	921.096
KJ/Kg air for stoichiometric mixture @15° C	≈18.6	102.344

Property	No. 2 Diesel	Ethanol
Heating value (2)		
Higher (liquid fuel-liquid water) KJ/Kg	44659.2–46520	29772.80
Lower (liquid fuel-water vapor) KJ/Kg	41868–44194	26749.00
Higher (liquid fuel-liquid water) KJ/Kg	322616.20	195616.60
Lower (liquid fuel-water vapor) $KJ / litre @ 15.5° C$	35800	21200[b]
Heating value, stoichiometric mixture		
Mixture in vapor state, $kJ/cu.m @ 20° C$	3610[c]	3 440
Fuel in liquid state, KJ/Kg or air	–	2977.3
Specific heat, KJ/Kg °C	1.8	2.386
Stoichiometric air/fuel, weight	14.7	9
Volume % fuel in vaporized stoichiometric mixture	–	6.5

*Source: U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Alternative Fuels Data Center
http://www.afdc.energy.gov/afdc/fuels/properties.html

[a]Pour Point, ASTM D 97.

[b]Calculated.

[c]Based on Cetane.

Appendix C: Specific Exhaust Gas Analyzer SV-5Q

Parameter	Range(Environment Conditions)
Temperature	-5~40℃
Humidity	≤95%
Atmosphere Pressure	670~106kps
Power Supply	AC220V ±10%; 50Hz ±1Hz

Parameter	Measurement Range
HC	0~10000 10^{-6}(ppm)vol
CO	0~10.0 10^{-2} (%)vol
CO_2	0~20.0 10^{-2}(%)vol
O_2	0~25.0 10^{-2} (%)vol
NO	0~5000 10^{-6}(ppm)vol
speed	0~10000rpm
Oil temperature	0~120℃

Parameter	resolution
HC	1 ppm vol
CO	0.01% vol
CO_2	0.01% vol
O_2	0.01% vol
NO	1ppm
Oil temperature	0.1℃

Parameter	Allowed Error
HC	±12ppm vol (relative error: ±5%)
CO	±0.06%vol(relative error:±5%)
CO_2	±0.5 %(relative error: ±5%)
O_2	±0.1% vol (relative error: ±5%)
NO	±25ppm vol (relative error: ±5%)
speed	±10rpm (0~10000rpm) ±1% measurements(>10000rpm)

*Source: Most Success Technology Limited, http://ms-tech.en.made-in-china.com