

**ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES**

PhD THESIS

Mustafa Kaan BALTACIOĞLU

**HYDROXY (HHO) AND HYDROGEN (H₂) ENRICHED COMPRESSED
NATURAL GAS (CNG) USAGE IN BIODIESEL FUELLED COMPRESSION
IGNITION (CI) ENGINES**

DEPARTMENT OF MECHANICAL ENGINEERING

ADANA, 2016

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ABSTRACT

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Reduction of diesel fuel dependency is intended with more environmentally friendly and economical methods in this experimental thesis. For this purpose, canola oil methyl ester (COME) and palm oil methyl ester (POME) are produced and blended with standard diesel fuel at various volumetric ratios (25%, 50%, 75% and 100%). Beside, pure hydrogen, hydroxy (HHO) and compressed natural gas (CNG) fuels are used with different volumetric flow rates. The amount of liquid fuel injected into the cylinders is reduced by 15%-30% via fuel pump plunger pin without structural changes on the diesel engine with controlled by stepping motor and devices. Instead of the reduced liquid fuel, 15%, 25% and 35% (by volumes) Hydrogenated (H₂ and HHO) gases mixed with CNG; and supplied by using mixture chamber before intake manifold of the test engine. Engine performance and exhaust emission graphs and terms are included in order to avoid the difficulties on fuel mixtures definitions. Results were compared and discussed with optimum amount of diesel injection and optimum mixture percentages. As a result; HHOCNG is firstly examined with pilot biodiesel injection and results were explored in-depth.

Key Words: Hydrogen, HHO, CNG, Biodiesel, pilot diesel injection,

ÖZ

DOKTORA TEZİ

HİDROKSİ (HHO) VE HİDROJEN (H₂) İLE ZENGİNLEŞTİRİLMİŞ SIKIŞTIRILMIŞ DOĞAL GAZIN (CNG) BİYODİZEL YAKITLI DİZEL MOTORLARDA KULLANIMI

Mustafa Kaan BALTACIOĞLU

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Bu deneysel tezde, dizel yakıt bağımlılığının azaltılması daha çevre dostu ve ekonomik yöntemlerle tasarlanmıştır. Bu amaç için, kanola yağı metil ester (COME) ve palm yağı metil ester (POME) üretilmiş ve çeşitli hacimsel oranlarında standart dizel yakıt (%25,%50,%75 ve%100) ile karıştırılmıştır. Buna ek olarak, saf hidrojen, hidroksi (HHO) ve sıkıştırılmış doğal gaz (CNG) yakıtları, farklı hacimsel oranlarda karıştırılıp emme havası ile birlikte kullanılarak zenginleştirme yapılmıştır. Silindirlere enjekte edilen sıvı yakıt miktarı, dizel motorunda yapısal değişiklikler olmadan, yakıt pompası piston pimi ve adım motoru kullanılması vasıtasıyla % 15 - %30 arasında düşürülmüştür. Kısılmış dizel akaryakıt yerine,%15,%25 ve % 35 hacimsel oranlarda CNG ile karıştırılmış hidrojen türevi (H₂ ve HHO) yakıtları; test motorun emme manifoldundan önce hazırlanmış olan karışım odasını kullanarak deneysel değerlendirmeler sağlanmıştır. Motor performansı ve egzoz emisyon grafikleri ve terimler yakıt karışımları tanımlamalarındaki zorlukları önlemek amacıyla açıkça izah edilmiştir. Karşılaştırmalı ve en iyi sonuçların elde edildiği karışım türleri ve oranları detaylı bir biçimde irdelenmiştir. Genel sonuç olarak; HHOCNG yakıt karışımı, pilot biyodizel enjeksiyonu ile ilk defa denenmiş ve sonuçlar derinlemesine incelenmiştir.

Anahtar Kelimeler: Hidrojen, Hidroksi, Doğalgaz, Biodizel, Pilot püskürtme

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LIST OF ABBREVIATIONS AND NOMENCLATURE

ASTM	: American Society for Testing and Materials
BSFC	: Brake Specific Fuel Consumption
CA	: Crank Angle
CI	: Compression Ignition
CN	: Cetane Number
CNG	: Compressed Natural Gas
CO	: Carbon Monoxide
CO ₂	: Carbon Dioxide
COME	: Canola Oil Methyl Ester
DI	: Direct Injection
DME	: Dimethyl Ester
EGR	: Exhaust Gas Recirculation
EN	: European Norm
H ₂	: Hydrogen
HCNG	: Hydrogenated Compressed Natural Gas
HHO	: Hydroxy, BrownGas, Oxyhyrdogen, Aquygen
HHOCNG	: Hydroxy-Compressed Natural Gas
HOME	: Honge Oil Methyl Ester
ICE	: Internal Combustion Engine
LHV	: Lower Heaitng Value
LPM	: Liter per Minute
NO _x	: Nitrogen Oxide
POME	: Palm Oil Methyl Ester
TDC	: Top Dead Centre
THC	: Total Hydrocarbon
WPOME	: Waste Palm Oil Methyl Ester
η_{bth}	: Brake Thermal Efficiency
η_v	: Volumetric Efficiency

1. INTRODUCTION

1.1. Overview

Modern diesel engines power much of the world's equipment and, most notably, are prime movers commonly available today. Among personal and commercial vehicles, the diesel engines hold a significant market share worldwide. And their share and popularity are both increasing. Traditionally, diesel engines run on mineral diesel, which is produced from crude oil. This fact comes with several consequences, giving rise to various concerns, which cannot be put aside. In fact, they have to be, and in many areas they already are, addressed decisively in order to ensure long-term and sustainable use of these excellent machines in the future (B. Kegl et al, 2013).

- The first concern is the limited crude oil reserves. Crude oil covers about 37% of world's energy demands (Fig. 1.1) (Asif and Muneer 2007; Dorian et al. 2006; Kegl 2012; Kja'rstad and Johnsson 2009).
- The second concern is the immense quantity of fuel, consumed by diesel engines all over the world. A vast majority of diesel engines are engaged in road transport, which accounts for about 81 % of total energy used for transportation (Fig. 1.2) (Chapman 2007). Besides the road transport, diesel engines are also engaged in railway transport, naval transport, electricity generation, and so on. Anyhow, even if we manage to develop efficient alternative fuel production, the immense fuel volume, consumed daily by world's diesel engines, will still be a problem. For this reason, trying to reduce the diesel engine fuel consumption to the limits of possible should be worth of every effort.
- The third concern is related to the chemical process of transformation of internal fuel energy into mechanical work or, more precisely, to the exhaust emissions of this process.

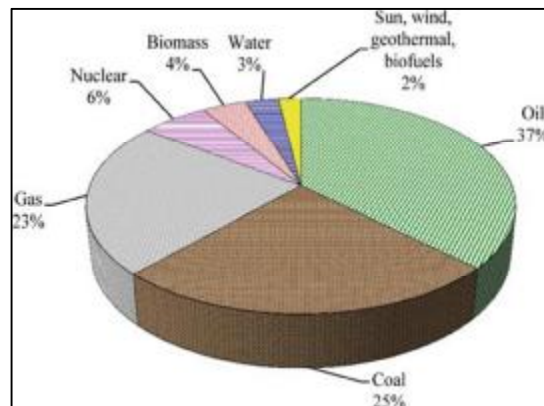


Figure 1.1. Shares of global energysources

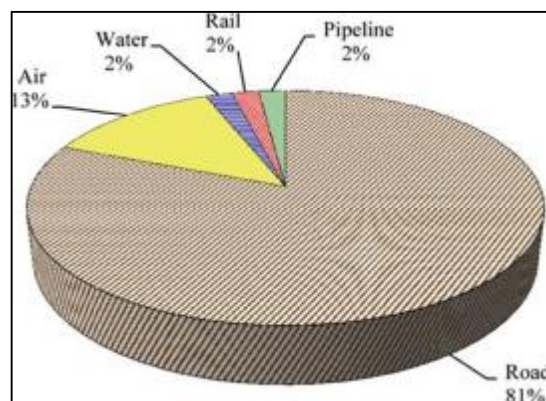


Figure 1.2. Energy consumption shares in the transport sector

A diesel engine produces mainly CO_2 , NO_x , CO, unburned HC, and PM/smoke emissions. These emissions contribute negatively to:

- Global climate changes
- General pollution of air, water, and soil
- Direct health effects (cancer, cardiovascular and respiratory problems, ..)

1.2. Diesel Engines Characteristics and Green Diesel Engines

The diesel engine is a compression-ignition engine in which the fuel and air are mixed inside the engine. The air required for combustion is highly compressed inside the combustion chamber. This generates high temperatures which are

sufficient for the diesel fuel to spontaneously ignite when it is injected into the cylinder. The diesel engine thus uses heat to release the chemical energy contained within the diesel fuel and convert it into mechanical force (Reif, 2014).

Diesel engine is a compression ignition engine of a 2- or 4-stroke type. From the $p - V$ diagrams (Fig. 1.3), it can be seen that the duration of the whole diesel cycle is 360°CA for the two-stroke engine and 720°CA for the four-stroke engine (B. Kegl, 2013). The whole cycle consists of the following phases: intake of air, compression of air, fuel injection, mixture formation, ignition, combustion, expansion, and exhaust. The intake phase begins with the intake valve opening and lasts till the intake valve closing. After that the intake air is compressed to a level corresponding to compression ratios from 14:1 to 25:1 or even more. In automotive application, diesel engines are practically always of the 4-stroke type (Bauer 1999), either naturally aspirated or turbocharged.

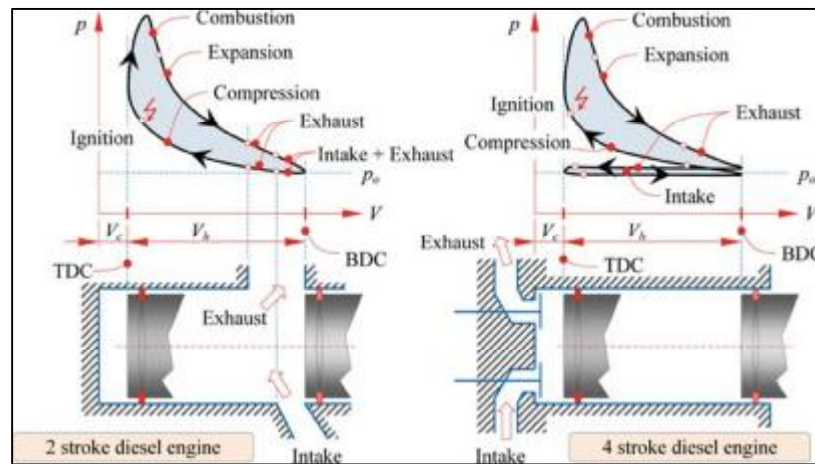


Figure 1.3. The p - V diagrams of two-stroke and four-stroke diesel engine

In a naturally aspirated diesel engine (Fig. 1.4), the pressure p_k in the intake tubes is smaller than the ambient pressure p_o . Since at the end of the exhaust process the pressure p_r of the residual gases is higher than p_k , this means that the intake of air starts only after the piston travels a significant distance towards the bottom dead center. This means that the in-cylinder pressure p_a at the end of the intake process is always lower than p_k . In fact, in a naturally aspirated diesel engine the following

relations are always valid: $p_r > p_k > p_a$ and $V_F < V_h$, where the symbol V_F denotes the actual volume of fresh intake air.

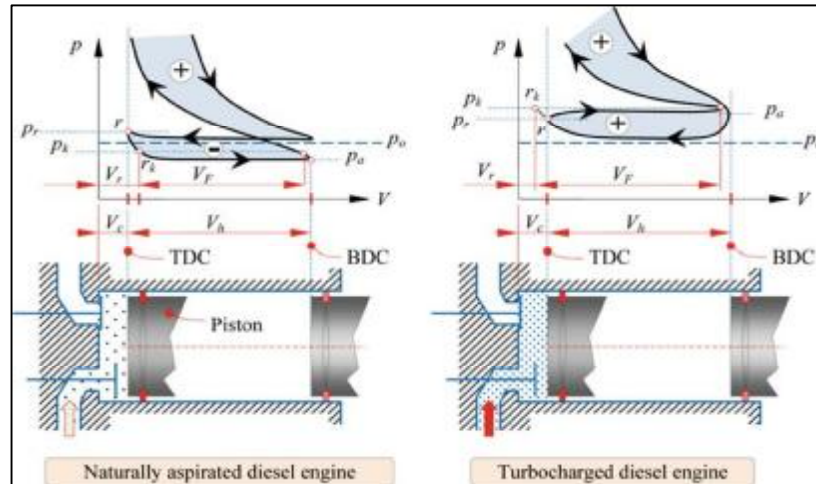


Figure 1.4 The p-V diagrams of naturally aspirated and turbocharged diesel engine (B. Kegl, 2013).

For many years, diesel engine emissions have not been considered to be a major problem. Luckily, this has changed substantially. Nowadays, respectable efforts are put into the development of diesel engines and new supplemental technologies with the aim to reduce these harmful emissions. Special attention is focused on engine management, fuel injection and combustion control, exhaust gas recirculation, catalytic after treatment and filtering of exhaust emission, and alternative fuel usage (B. Kegl, 2013).

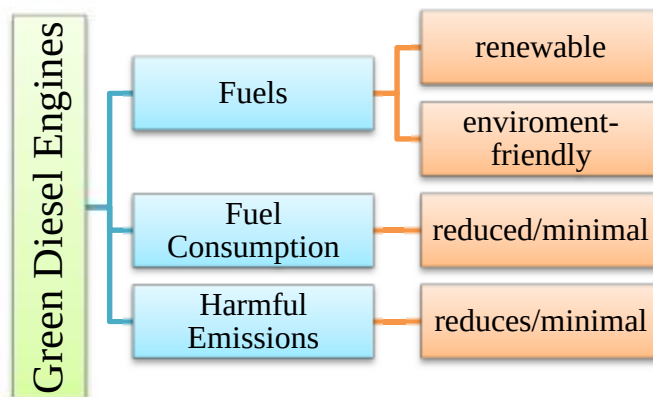


Figure 1.5 Green diesel engine features

All conditions in Figure 1.5. have been discussed in this experimental investigation. With a more explanatory approach, renewable and environmental-friendly fuels which are canola and palm biodiesel as liquid, additionally, compressed natural gas, hydrogen and hydroxy gas as gaseous fuel form. Also semi-pilot injection method, that will be explained more detailed under related topics, is used to reduce fuel consumption. As a result of these conditions, reduction on harmful exhaust emissions, such as carbon-monoxide, carbon dioxide and nitrogen oxide, is obtained.

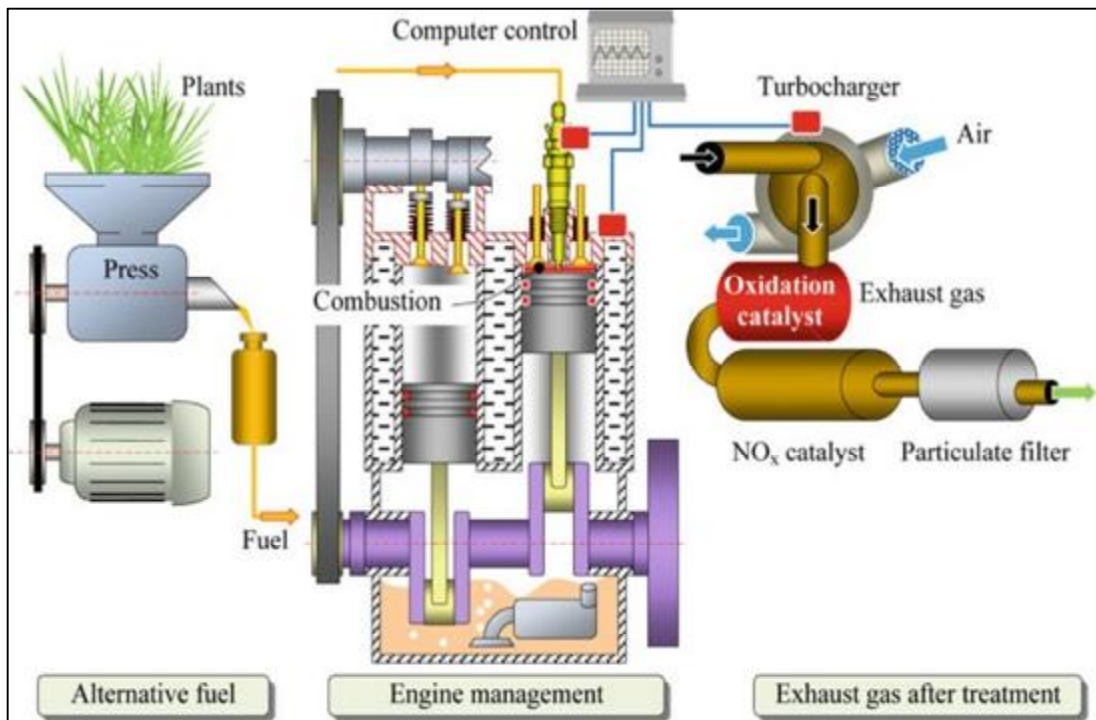


Figure 1.6 Techniques to improve engine characteristics (B. Kegl et al, 2013)

Representation of three optional techniques to improve engine characteristics is given in Figure. 1.6above. Investigation of using alternative fuels without modification on the test engine is one of the base objectives in this study.

1.3. Alternative Fuels

Most of the initial interest in alternative fuels started after the oil crisis in the 1970s. It has been grown more recently by concerns about supply interruptions, high prices, air quality and greenhouse gases. The concern of the air quality in many areas around the world makes finding solutions more urgent. As the price of oil rises, alternate fuels become more competitive. Major questions remain to be answered on which fuel or fuels will emerge and to what extent alternative sources will replace gasoline as the main product of crude oil (Hordeski, 2008).

Engine management and alternative fuels usage offer a possibility to reduce the formation of harmful emissions. Diesel engine characteristics depend significantly on the engine type. But, even for a given engine type, the engine characteristics can still be varied in a wide range in dependence on engine management, exhaust gas after treatment, and usage of alternative fuels (Fino et al. 2003; Gray and Frost 1998; Maiboom et al. 2008; Peng et al. 2008; Stanislaus et al. 2010; Twigg 2007).

Previously, a number of fuels have been investigated by other researchers as possible alternatives for diesel engines. Today's most frequently investigated and used alternative fuels are:

- Water in diesel emulsion
- *Natural gas* and liquefied petroleum gas
- Methane and propane
- Dimethyl ether and dimethyl carbonate
- Fischer–Tropsch diesel
- *Hydrogen*
- Alcohols
- Vegetable oils, bioethanol, and *biodiesel*

This experimental study focuses on usage of alternative fuels and combinations of them in diesel engines. Experimental fuel mixtures consist of canola and palm biodiesel as liquid, additionally, compressed natural gas, hydrogen and hydroxy gas (HHO) as gaseous fuel form.

1.3.1. Compressed Natural Gas (CNG)

Natural gas (NG) is one of the most important energy carriers today, because it is available in large quantities and its reserves are of the same magnitude as the crude oil reserves. Typical compositions of NG are approximately 97.2 % methane, 3.3 % ethane, 2.2 % nitrogen, 0.5 % carbon dioxide, 0.7 % propane, 0.1 % isobutane, 0.2 % N-butane, 0.1 % pentane, and 0.1 % hexane (Nwafor, 2000). NG offers several advantages such as clean combustion, high availability, and an attractive price (Poornipatpong and Cheenkachorn 2011). Additionally, its relatively high auto-ignition temperature is suitable for higher compression engines (Selim 2001). Due to a low cetane number, an engine using natural gas requires injection of mineral diesel fuel as the pilot ignition fuel. Such engines are dual fuel engines, which need two fuel systems (Kowalewicz and Wojtyniak 2005). Previous studies investigated the characteristics of dual fuel operation in unmodified or slightly modified diesel engines (Selim 2001; Baltacioglu 2015; Arat 2015). For dual fuelling, the in-cylinder pressure and heat release rate are lower than for neat diesel fuelling. By increasing the pilot diesel fuel injection quantity, the pressure in the cylinder and the rate of its rise also increase. Furthermore, an increase of pilot fuel quantity extends the lean burning limit and decreases HC and CO emissions, which are generally higher than for diesel fuelling. The ignition delay is longer for dual fuelling (Cordiner et al. 2008; Kowalewicz and Wojtyniak 2005). The combustion characteristics of a diesel engine with natural gas/diesel fuels show positive effects on thermal efficiency, total specific fuel consumption, soot, and NOx emissions (Selim 2001). However, the reports indicate that such dual operation system cannot reach high speed operation that is obtainable by a diesel-only engine. Therefore, adequate modifications might be necessary to mitigate this negative effect (Kegl B. et al, 2013). The engineering

properties of hydrogen based gaseous fuels and natural gas mixtures are unique parameters for engine design and reducing the exhaust emissions. Ma (2010), reported the importance of these mixture with these considerations. This new blended fuel is known as HCNG, or Hythane, for which the use will create a basic infrastructure for the use of hydrogen in the future(Ma,2010).

The actual composition varies, depending on the region of the source. As an engine fuel, natural gas may be used either in a compressed form, compressed natural gas (CNG) or in a liquid form, liquefied natural gas (LNG). Compressed natural gas is only used in about 30,000 vehicles in the United States, which includes school buses, delivery trucks, and fleet vehicles. Worldwide, about a million vehicles in thirty-five countries operate on natural gas. Some of the countries where natural gas is widely used include New Zealand, Italy and countries of the former Soviet Union. Natural gas has several advantages over gasoline. It emits at least 40% less hydrocarbons and 30% less carbon dioxide per mile compared to gasoline. It is also less expensive than gasoline per gallon-equivalent. Maintenance costs can also be less than those for gasoline engines since natural gas causes less corrosion and engine wear. Although natural gas is a plentiful fossil fuel, it is nonrenewable. There is also a range limitation and natural gas vehicles can cost more due to the need to keep the fuel under pressure. The weight and size the pressure tank reduces storage space and affects fuel economy. Most gasoline-powered engines can be converted to dual-fuel engines with natural gas. The conversion does not require the removal of any of the original equipment. A natural gas pressure tank is added along with a fuel line to the engine through special mixing equipment. A switch selects either gasoline or natural gas/propane operation. Diesel vehicles can also be converted to a dual-fuel configuration. Natural gas engines may use lean-burn or stoichiometric combustion. Lean-burn combustion is similar to that which occurs in diesel engines, while stoichiometric combustion is more similar to the operation of a gasoline engine. Compressed natural gas has a high octane rating of 120 and produces 40 to 90% lower hydrocarbon emissions than gasoline. There are also 40 to 90% lower carbon monoxide emissions and 10% lower carbon dioxide emissions than gasoline. A larger, heavier fuel tank is needed and the driver must refill about every

161kilometers. Refilling can takes two to three times longer than refilling gasoline. Some slow fill stations take several hours and the limited availability of filling stations can be a problem (Hordeski, 2008).

1.3.2. Hydrogen and Hydroxy Gas (HHO)

Hydrogen is abundant, being the most common element in the universe. The sun consumes 600 million tons of it each second. But unlike oil, vast reservoirs of hydrogen are not to be found on earth. As an energy carrier, it could increase our energy diversity and security by reducing our dependence on hydrocarbon-based fuels. But it cannot be harvested directly. It must be extracted from another material. A wide variety of materials contain hydrogen, which is one reason it has attracted widespread support. Hydrogen is often called a secondary energy carrier, instead of a primary energy source. However, finding, extracting and delivering these so-called primary energy sources require energy and major investments before they can be utilized. Coal and natural gas come closer to true primary energy sources since they can be burned directly with little or no refining, but energy is still needed to extract these resources and deliver them where the energy is needed. Some tests have shown that the air coming out of a hydrogen fueled engine is cleaner than the air entering the engine. Acid rain, ozone depletion and carbon dioxide accumulations could be greatly reduced by the use of hydrogen. Hydrogen can be stored as a gas, liquid, or as a part of a solid metal, polymer or liquid hydride. Although it is possible to store hydrogen as a high pressure gas in steel containers, disadvantages exist because of the weight of the storage containers and the safety hazard in the event of an accident (Hordeski, 2008).

Figure 1.7. proposes various paths through which the four kindsenergies to drive hydrogen productioncan be obtained fromgreen energy sources. The electrical and thermal energy canbe derived from renewable energies (like solar, wind, geothermal, tidal, wave, ocean thermal, hydro, biomass), orfrom nuclear energy, or from recovered energy. The biochemicalenergy is that stored in organic matter (in form of carbohydrates, glucose and sugars etc.) and can be manipulated bycertain

micro-organisms that can extract hydrogen from various substrates or it can be chemically converted to thermal energy (Dincer, 2012).

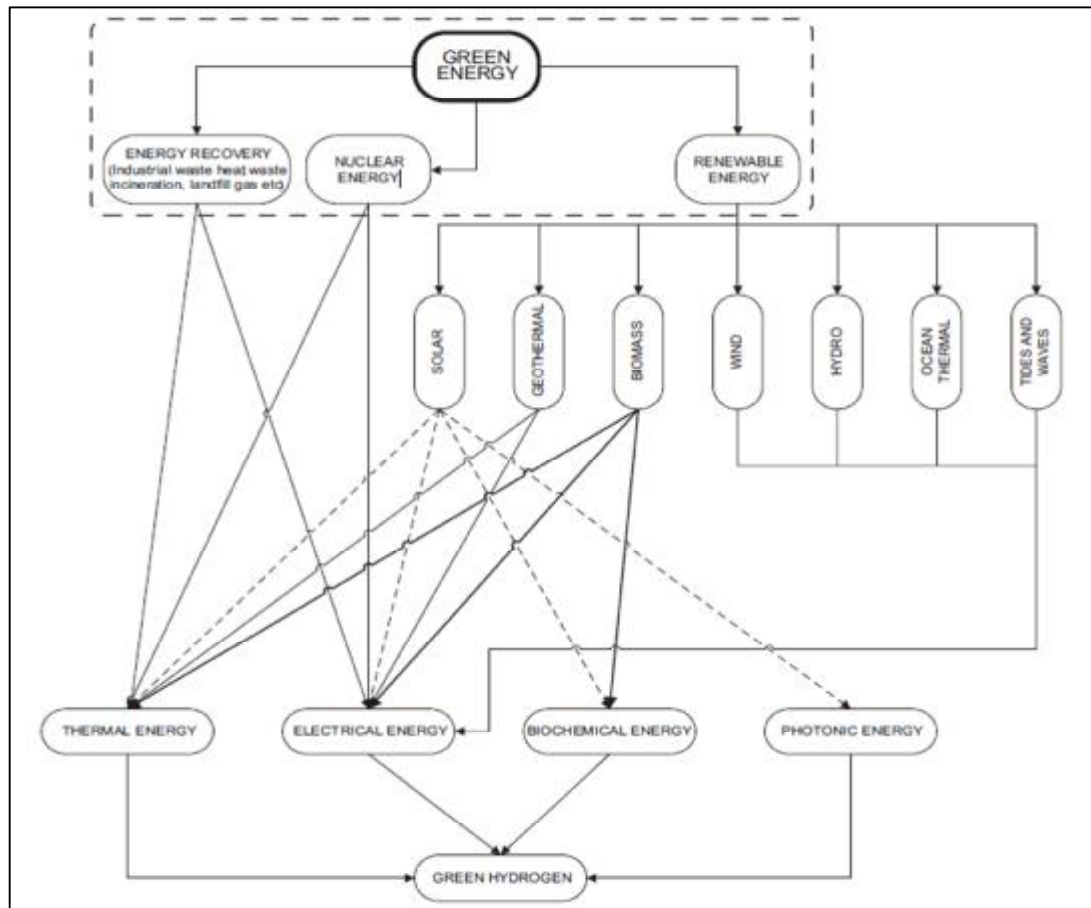


Figure 1.7. Generation of basic form of energy from primary green energy sources (Dincer, 2012)

Electrolysis is the most common method used to split H_2 from water. Currently, 12-15% of hydrogen production globally source is presented by electrolyses. Early studies were initiated by Yull Brown in 1977 via equipment generally referred to as electrolyzers and the resulting gas is known as “brown’s gas” or HHO (Al-rousan, 2010). (In literature, scientists can be named of this gas differently- Aquygen (registered trademark owned by HTA inc.(Santilli,2006)), HHO, hydroxy, oxyhydrogen, hydroxygen, knallgas, etc.). HHO is a trademark and comes from the separation of water molecules $H-OH$ that contains (theoretically) 66% H_2 and 33% O_2 . It has high calorific value and 1 kg of HHO is three times as

potent as gasoline and eight times as potent as diesel. Also achieving of HHO gas under water electrolyses, several item affected the end product. This affected the efficiency of the HHO generators. One another method is the steam reforming of methane from natural gas. Steam reforming converts the methane and other hydrocarbons in natural gas into hydrogen and carbon monoxide using the reaction of steam over a nickel catalyst. Electrolysis uses an electrical current to split water into hydrogen at the cathode (+) and oxygen at the anode (–). Steam electrolysis uses heat, instead of electricity, to provide some of the energy needed to split water and can make the process more energy efficient (Hordeski, 2008). Additionally, favorable characteristics and some limitations to use hydrogen for internal combustion engines can be listed as below (Karim 2015):

- It is a renewable fuel that can be manufactured from widely available sources, such as water, through the expenditure of energy.
- It is a clean fuel that produces much less exhaust emissions on combustion than other fuels. Its exhaust gas produces water and, on condensation, energy through cogeneration. However, its combustion in air produces oxides of nitrogen.
- Hydrogen is catalytically sensitive and has some very attractive combustion characteristics, such as clean combustion, fast burning rates, and a wide flammable mixture range. It requires low ignition energy, but with relatively high auto-ignition temperatures. It has a very high heating value on mass bases with high flame temperatures. It is highly buoyant, diffusive, and remains a gas down to extremely low cryogenic temperatures.

However, at present there are some limitations to its wide application as a fuel, especially in engines:

- It requires much useful energy for its manufacture and lacks at present the infrastructure for its wide distribution, resulting in its high cost on an energy basis in comparison to other fuel resources.
- It has potential problems relating to safety, material compatibility, portability, storage, handling, and transport. It is extremely difficult to liquefy and keep as a liquid, requiring much useful work.
- It has a very low heating value on volume and liquid bases, with its flames having very low luminosity.

Important fuel properties of hydrogen, CNG and HCNG (vol 18%) are shown in Table 1.1. Overview for HHO is given in Table 1.2 and additionally, “Hydrogen & HHO” fuel potential benefits and challenges are compared in Table 1.3 below.

Table 1.1. Important fuel properties of hydrogen, CNG and HCNG(vol 18%)

Properties	H ₂	CNG	HCNG
Stoichiometric AFR	34	17.2	17.8
Limits of flammability in air, (vol %)	4-75	5-15	5-35
Auto ignition temp. K	858	813	825
Flame temp in air K	2318	2148	2210
Max. energy for ignition in air, mJ	0.02	0.29	0.21
Burning velocity in NTP air, cm s ⁻¹	325	45	110
Quenching gap in NTP air, cm	0.064	0.203	0.152
Diffusivity in air, cm ² s ⁻¹	0.63	0.2	0.31
Equivalence ratio	0.1-7	0.7-4	0.5-5.4
Cetane number	-	-	-
Lower Heating Value (Mj/kg)	119.9	46.28	48.26

Table 1.2. Overview of HHO properties

Name	HHO, Hydroxy, Brown's Gas
Chemical form.	HHO; 66.6% Hydrogen and 33.3% Oxygen
Inventor	Dr. Yule Brown
Purpose	On-demand H ₂ fuel; to be consumed immediately after production. Future replacement fuel for petroleum fuels Supplemental role in efficient burning of gasoline fuel in vehicles

Production	Common ducted electrolyzer; parallel plate and series connected cells
Advantages	Flame doesn't burst, is sootless due to the absence of carbon Suitable for indoor work; cleanest energy
Application	Thermal processing industry: gas welding, jewelry process, etc. HHO generators installed with IC Engines for improve the power

Table 1.3. "Hydrogen & HHO" fuel potential benefits and challenges (Jordan, 2012)

Property	Benefits	Challenges
<i>Wide Flammability Range</i>	Enables lean or more dilute combustion	Lean combustion reduces power.
<i>Low Ignition Energy</i>	Allows lean yet rapid ignition	Can prematurely ignite on hot spots.
<i>Small Quenching Distance</i>	Flame can travel closer to walls	Can lead to flashback
<i>High Auto-Ignition Temp</i>	Enables higher compression ratio	Hard to ignite in CI engine.
<i>High Diffusivity</i>	Facilitates more uniform mixture	Can leak to undesirable areas.
<i>High Flame Speed</i>	More closely approaches ideal Otto engine cycle	N/A
<i>High Octane Rating</i>	Enables higher compression ratio, Reduces knock & MEP variation	Hard to ignite in CI engine.
<i>Low Density</i>	N/A	Lower energy density per volume.

1.3.3. Biodiesel

The first public demonstration of vegetable oil based diesel fuel was at the 1900 World's Fair, when the French government commissioned the Otto Company to build a diesel engine to run on peanut oil. The French government was interested in vegetable oils as a domestic fuel for their African colonies. Rudolph Diesel later did extensive work on vegetable oil fuels and became a leading proponent of such a concept, believing that farmers could benefit from providing their own fuel. However, it would take almost a century before such an idea became a widespread

reality. Shortly after Dr. Diesel's death in 1913 petroleum became widely available in a variety of forms, including the class of fuel we know today as 'diesel fuel'. With petroleum being available and cheap, the diesel engine design was changed to match the properties of petroleum diesel fuel. The result was an engine which was fuel efficient and very powerful. For the next 80 years diesel engines would become the industry standard where power, economy and reliability are required (Pacific Biodiesel).

The two most common sets of regulatory standards for biodiesel fuels are ASTM D6751 in the USA and EN 14214 in Europe. Some of the specifications comprising these standards are directly related to the chemical composition of biodiesels, for example, viscosity, cetane number, cloud point, distillation, etc. Other specifications are related to the purity of biodiesel and address issues pertaining to production processes, transport, and storage, such as flash point, methanol content, metals content, acid level, and filter plugging tendency. Oxidative stability is also an important property of biodiesel that is influenced by both biodiesel chemical composition and by storage and handling conditions. Biodiesel is a renewable fuel, consisting of various fatty acid methyl esters with the exact composition depending on the feedstock. This is a distinctly different composition than the hydrocarbon content of mineral diesel. In spite of that, biodiesel has many properties very close to those of mineral diesel. Consequently, the required biodiesel-related modifications of the diesel engine are typically rather minor. On the other hand, because of its different chemical character, biodiesel has several properties, which differ from those of mineral diesel just enough to offer an opportunity to reduce harmful emissions without worsening other economy and engine performances. It should be noted, however, that biodiesel properties may depend heavily on its raw materials (B. Kegl et al, 2013).

Some of the raw materials for biodiesel are oils from *canola*, *palm*, *soybean*, *sunflower*, rapeseed, beef and sheep tallow, poultry, fish, jatropha, almond, barley, camelina, coconut, copra, groundnut, karanja, laurel, oat, coffee beans, poppy seed, okra seed, rice bran, sesame, sorghum, wheat, and microalgae (Altıparmak et al. 2007; Arkoudeas et al. 2003; Balat and Balat 2008; Cetinkaya et al. 2005; Chen and

Chen 2011; Chisti 2007; Demirbas 2006, 2008; Giannelos et al. 2005; Gomez et al. 2002; Goodrum and Eitchman 1996; Hu et al. 2005; Jain and Sharma 2010; Jha et al. 2008; Kalam and Masjuki 2002; Lapinskiene et al. 2006; Lebedevas et al. 2006; Misra and Murthy 2011; Murugesan et al. 2009; Oliveira et al. 2008; Park et al. 2008; Peng et al. 2006; Peterson and Hustrulid 1998; Senzikiene et al. 2006; Sharma et al. 2008; Shumaker et al. 2008; Smith et al. 2010; Tang et al. 2008; Yuan et al. 2005). Additionally, more than 400 oil-bearing crops have been identified and mostly soybean, palm, sunflower, safflower, cottonseed, rapeseed (canola), and peanut oils are considered potential alternative fuels for diesel engines by reseachers.

Common way to produce biodiesel is generally transesterification of vegetable oils and animal fats as shown in Fig. 1.8 below. Transesterification is a chemical reaction between triglycerides and a short-chain alcohol in the presence of a catalyst to produce monoesters (Demirbas 2009; West et al. 2008). The commonly used catalysts in the ester reaction are lipase catalyst, acid catalyst, and alkali catalyst. The most of the commercial biodiesel is produced from plant oils, by using very effective alkali catalysts such as sodium or potassium hydroxides, carbonates, or alkoxides. The long- and branched-chain triglyceride molecules are transformed to monoesters and glycerin. Commonly used short-chain alcohols are methanol, ethanol, propanol, and butanol. Methanol is the most used commercially because of its low price (Lin et al. 2011; Sinha et al. 2008). Additionally, the one of the products which is glycerin is used in pharmaceuticals.

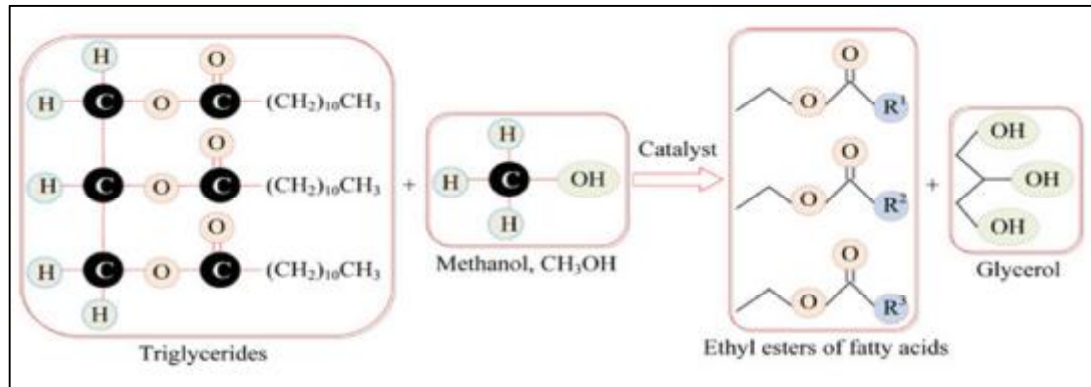


Figure 1.8. Transesterification process of biodiesel fuels (B. Kegl et al, 2013).

The raw materials for biodiesel production are quite different in various parts of the world and they depend greatly on climate, local soil conditions, and availability (Fig. 1.9) (Lin et al. 2011). In the USA, soybean oil is the most commonly biodiesel feedstock, whereas the rapeseed (canola) oil and palm oil are the most common source for biodiesel in Europe and in tropical countries, respectively (Singh and Singh 2010). World biodiesel production has risen from 1.8 billion liters in 2003 (Bozbas, 2008) to 19 billion liters in 2010. The largest producer of biodiesel is the European Union, which generated 53 % of all biodiesel in 2010. The top three biodiesel producing nations in Europe are Germany, France, and Italy.

Many various methods for biodiesel production from various sources have been developed so far. Thus, nowadays, biodiesel can be produced by using technologies such as ultrasonic cavitation, hydrodynamic cavitation, microwave irradiation, response surface technology, two-step reaction process, etc.

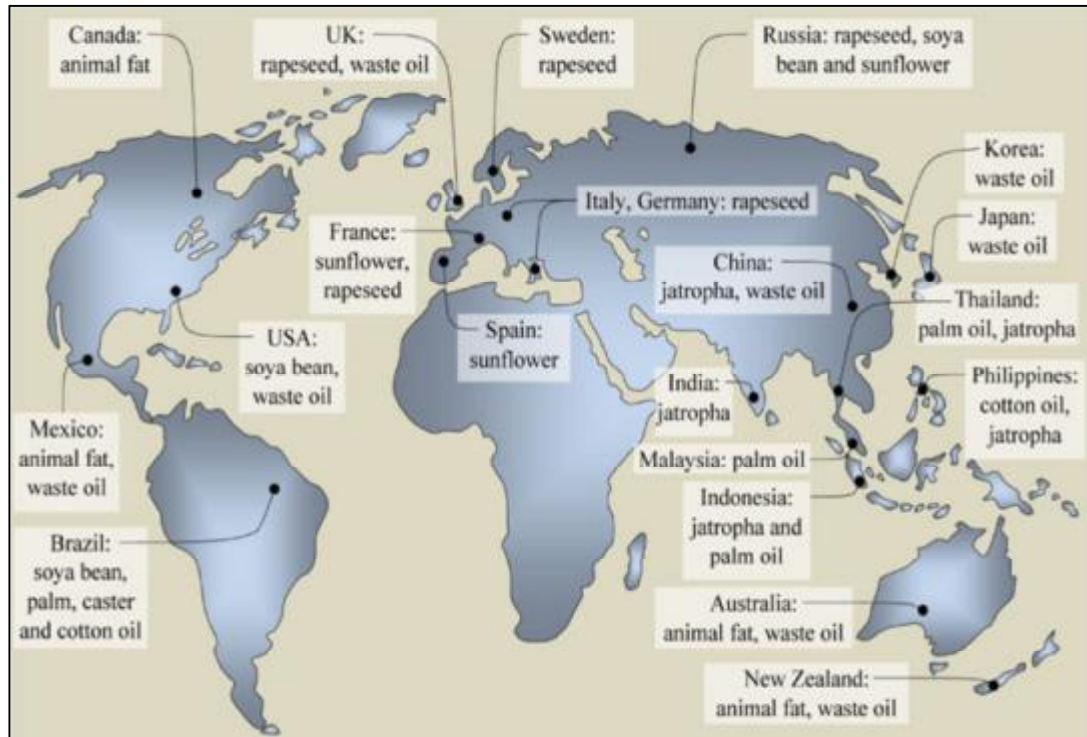


Figure. 1.9. Biodiesel sources around the world (B. Kegl et al, 2013).

Some of the general advantages and disadvantages are outlined below by the strength–weakness–opportunities–threat (SWOT) analysis (Misra and Murthy 2011; Russo et al. 2012).

Strengths

- Biodiesel is environmentally friendly, biodegradable, and sustainable; compared to mineral diesel, it has a lower toxicity; in Europe and USA its quality is regulated by corresponding standards.
- Biodiesel reduces net carbon dioxide emissions up to 78 % on a lifecycle basis when compared to mineral diesel.
- Biodiesels exhibits low sulfur and aromatic content; this results in almost no sulfur dioxides emissions.

- The properties of biodiesel (viscosity, ignition properties) and mineral diesel fuels are quite similar; therefore, biodiesel is an alternative fuel that can be used in existing, virtually unmodified, diesel engines.
- Biodiesels exhibit high combustion efficiency and high cetane numbers, which corresponds to short ignition delays and greater efficiency.
- Biodiesels exhibit good lubrication properties that reduce wear in the engine.
- The coproduct glycerine of the transesterification process can be commercially exploited.
- The raw materials for biodiesel production are renewable and widely available.
- The side products and agricultural residues can be used as fodder, fertilizers, or raw materials for second-generation biofuels.

Weaknesses

- Long-term storage of biodiesel fuels may cause degradations in certain fuel properties; biodiesel is hygroscopic; additives are often necessary.
- There are problems related to fuel solidification at low temperatures.
- There are problems related to situations when biodiesels are first introduced into equipment that has a long history of neat mineral diesel usage; mineral diesel fuel typically forms a layer of deposits on the inside of tanks and fuel tubes; biodiesels loosen these deposits, causing the block of fuel filters; proper filter maintenance during some period, following the introduction of biodiesels, is therefore required.
- Due to high oxygen content, biodiesels produce relatively high NO_x emission levels during combustion.
- The kinematic viscosity is higher than found in mineral diesel fuel; this affects fuel atomization during injection and requires a somewhat modified fuel injection system.

- Biodiesel fuels are relatively aggressive toward fuel pipes, sealings, and filters; these elements need to be adapted adequately.
- Oxidative stability is lower than that of mineral diesel so that, under extended storage conditions, it is possible that oxidation products may cause harm to engine components.
- The production of biodiesel is not sufficiently regulated; biodiesels that do not conform to the European or US standards can cause corrosion, fuel system blockage, seal failures, filter clogging, and deposits in the injection systems.

Opportunities

- Currently, biodiesels are the dominant biofuels in Europe.
- Broader biodiesel usage can improve energy supply and consequently strengthen the economy.
- Energy policy that provides tax reductions/exemptions and biofuel obligations could largely increase the use of pure plant oil and biodiesels.
- Waste oil can be used as cheap feedstock for biodiesel.
- Oil plants other than rapeseed can be cultivated in Europe (e.g., sunflower, soya).

Threats

- Biodiesel fuels may become toxic and polluting in case that they contain harmful solvents and other chemical additives.
- In Europe, rapeseed is mainly used as the feedstock source; rapeseed can be cultivated only every 4 years on the same field.
- A standard for biodiesel feedstock does not exist; variations in feedstock properties may lead to undesirable variations of biodiesels.

1.4. Dual Fuel Diesel Engines and Pilot Diesel Injection

Technological changes in the manufacture of power sources are required if they are to run on alternative fuels. It is likely that more power sources will move away from single-fuels to several fuels which would compete. This is done in many power plants today. Dual-fuel or flexible fuel are now used to some degree around the world (B. Kegl, 2013).

The term, dual-fuel diesel operation, describes compression ignition engines that burn *simultaneously* two entirely different fuels in varying proportions. These two fuels are usually made up of a gaseous fuel, which supplies much of the energy released through combustion, and a second fuel, which is a liquid employed mainly to provide the energy needed for ignition and the remaining fraction of the energy release by the engine. Another term sometimes used in gas-fueled engine applications is the *gas-diesel engine*. These have tended to relate to dual-fuel engines where the gaseous fuel is injected directly into the cylinder either during the early stages of compression or sometimes after the injection of the liquid fuel toward the end of compression. The gaseous fuel does not auto ignite on its own via compression ignition, but usually burns with the assistance of the injected liquid-fueled ignition processes. Accordingly, these engines are dual-fuel engines that employ different forms of introduction of the gaseous fuel that does not undergo auto ignition on its own (Karim, 2015).

Some alternative fuels can be used pure, while others have to be mixed with mineral diesel. In any case, the diesel engine has to be modified to some extent. The engines, which use mineral diesel and some gaseous fuel, are referred to as dual fuel engines. Natural gas and bio-derived gas appear to be quite attractive alternative fuels for dual fuel engines in view of their environment-friendly nature. In dual fuel gas diesel engine a mixture of air and gaseous fuel is prepared in an external mixing device and compressed in the cylinder. The compressed mixture is then ignited by energy from the combustion of the diesel fuel spray, which is called the pilot fuel. The amount of pilot fuel needed for this ignition is between 10 and 20 % of the amount needed for diesel-only operation at normal working loads. This amount

varies with the point of engine operation and engine design parameters (B. Kegl, 2013). Beside, for older CI engines, Miyake M. et al (1983) mentioned that, “natural gas and pilot liquid diesel fuel were mixed prior to injection and found that 40% pilot diesel fuel was needed for stable ignition”.

The operation of dual-fuel engines depends in a large way on how the gaseous fuel is introduced. The common approach is to introduce the gaseous fuel into the incoming air well ahead of the intake valve, as commonly accomplished in gasoline-fueled spark ignition engines. The gas may also be introduced at the beginning of the compression process so that much of the gaseous fuel becomes thoroughly mixed with the air before pilot fuel injection. These engines can be described as premixed dual-fuel engines, and this mode of gas introduction as fumigation. The relatively small quantity of the liquid diesel fuel injected to provide controlled ignition in dual-fuel engines is commonly described as the *pilot* (Karim, 2015).

A pilot injection phenomenon is a critic issue in dual fuel diesel engines. It's characteristics (quality of fuel, CN, and spray), size, mass, time and pressure is very important for the optimum steady state combustion occurs. Pilot diesel injection phenomenon clearly summarized by Papagiannakis R.G. et al (2003,2007) “In dual fuel CI engines operating with gaseous fuels as primary fuel and a ‘*pilot*’ amount of liquid diesel fuel as an *ignition source*, the gaseous fuel is inducted along with the intake air and is compressed like in a conventional diesel engine. The mixture of air and gaseous fuel does not auto ignite due to its high autoignition temperature. A small amount of liquid diesel fuel is injected near the end of the compression stroke to ignite the gaseous mixture”.

As similarly, pilot diesel injection strategy exhibits the factors of total economy as lower specific fuel consumption and does not necessary to modify the engine; environmental issues as lower undesirable exhaust emissions can be obtained with dual-fuel operation with pilot diesel injection. There is a very substantial reduction in the emissions of carbon and particulates, with a significant reduction in the production of oxides of nitrogen. This is mainly since the bulk of the energy release is produced by the combustion of very lean gaseous fuel–air mixtures and the

associated reduction in the size of the very hot combustion zone, which mainly involves the liquid diesel pilot fuel. Additional benefits as; relevant pilot quantity can lead to improvements in engine durability, reduced maintenance costs, and quieter running. There will also be less heat transfer, lubricant degradation, and surface deposition of the liquid fuel, leading to its surface ignition and combustion (Karim, 2015).

The combustion process during diesel-only operation can be identified on a heat release rate diagram presented in Figure 1.10 (KeglB, 2013). As it is shown in the figure, combustion processes in CI engines running on pure diesel fuel can be described by dividing into stages. Injection rate history is a very important characteristic. It strongly depends on the delivery rate history and it has the most important influence on the heat release rate (Fig. 1.10). The delivery rate is the quantity of fuel pushed per unit of time through the delivery valve into the high pressure tube. It is mainly influenced by the pump plunger diameter and pump plunger velocity, which depend on the cam profile. For example, a concave cam profile results in pump plunger velocity being higher than that obtained by the tangential or convex profile (Ishiwata et al. 1994; Kegl and Muller 1997; Kegl 1999, 2004). The actual start of fuel delivery depends on injection pump timing, which is usually given in crankshaft angle before TDC and indicates the moment when the pump plunger begins compressing the fuel. The delivery rate history is a well-controllable characteristic. For this reason, injection rate history, injection start, and injection end are rather difficult controllable quantities. After injection start the fuel atomizes, evaporates, and mixes with air until the mixture ignites. The time span between the injection start and combustion start is termed the ignition delay. The ignition delay, injection rate, and combustion chamber design influence strongly the heat release rate history. Therefore, heat release rate is also a rather difficult controllable quantity.

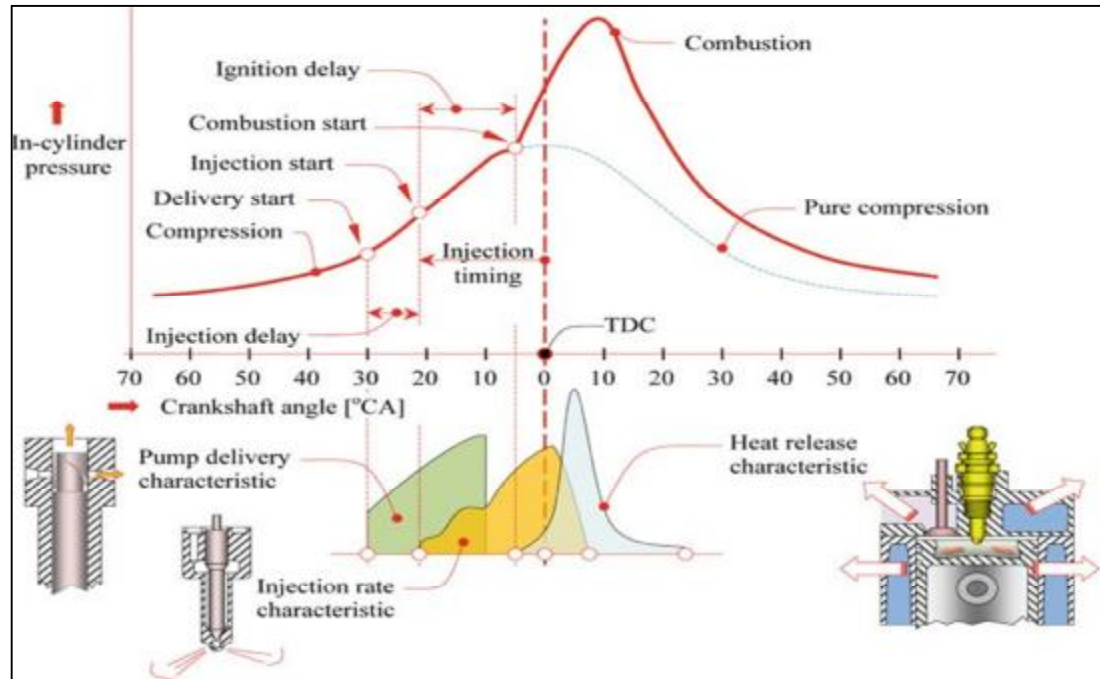


Figure 1.10. Delivery rate history, injection rate history, heat release, and in-cylinder pressure

However, the combustion processes in gas-fumigated dual-fuel engines using pilot injection have been identified to take place in five stages as shown in figure 1.11. (Sahoo, 2009) . They are the pilot ignition delay (AB), pilot premixed combustion (BC), primary fuel ignition delay (CD), rapid combustion of primary fuel (DE) and the diffusion combustion stage (EF) (Sahoo, 2009).

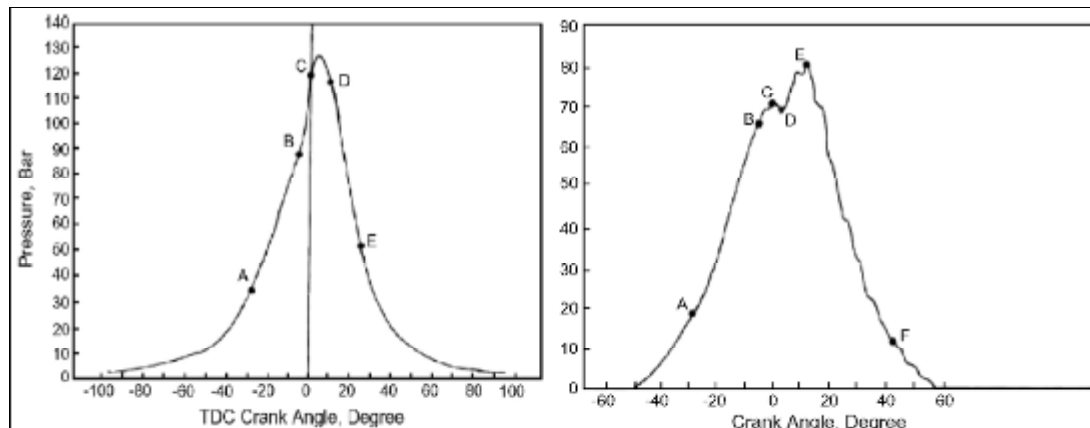


Figure 1.11. Combustion process in CI and Dual-fuel pilot injection in CI engines

1.5. Principles and objectives of the thesis

Elimination or reduction of diesel fuel dependency is intended and investigated opportunities with more environmentally friendly and economical methods in this thesis. For his purpose, canola oil and palm oil methyl ester (COME, POME) are produced and blended with standard diesel fuel at various volumetric ratios (25%, 50%, 75% and 100%). Additionally, pure hydrogen, hydroxy (HHO) and compressed natural gas (CNG) fuel mixtures are used with different volumetric flow rates. The amount of liquid fuel injected into the cylinders is reduced by 15%-30% via fuel pump plunger pin and replaced with gaseous fuel mixtures (HCNG and HHOCNG) without structural changes on the diesel engine. Liquid fuel injection amount is substituted with step motor and driver devices. Gaseous fuel mixtures are supplied by using mixture chamber before intake manifold of the test engine. Results were illustrated, compared and discussed with optimum amount of diesel injection via graphs. To sum up the main objectives of this thesis can be listed as below:

- Investigate and improve the non-modified diesel engine performance and exhaust emission characteristics by using alternative fuel mixtures.
- Optimization of the pilot injection amount with stepping motor and driver devices.
- Replacing diesel fuel with HHOCNG, HCNG and biodiesel fuel mixtures to investigate engine characteristics experimentally. Additionally, HHOCNG+biodiesel results will be added to literature as a preliminary experimental study.
- Compensate the disadvantages of biodiesel and CNG single fuel usage on performance results by enriching the test fuels with hydrogen and hydroxy gaseous fuels by reducing the dependence on fossil fuels.
- Clarify the most promising liquid and gaseous test fuel mixture combination for performance and emission criteria.

- Provide a guideline for future works on the implementation of the technology (controlling of dual fuel engines with semi-pilot injection by using alternative fuels).

2. PRELIMINARY WORK

This part is the literature survey on similar alternative fuels usage and dual-fuel pilot injection diesel engines operations. This section consists from two subsections, which are experimental fuels (biodiesels (COME-POME), natural gas (NG), hydrogen (H_2), hydroxy (HHO), hydrogenated-compressed natural gas (HCNG-HHOCNG)) and dual-fuel engines with pilot diesel injection. Researchers have deep and several investigations on dual-fuel diesel engines, hydrogen enrichment and CNG induction to the diesel engine with different methods. However, using combinations of these fuels with similar method to this thesis is a new area and open for research and developments. There is a gap of literature especially on HHOCNG and HCNG usage with pilot biodiesel injection method. Effects of these fuels and methodologies are presented in this section one by one under related subtitles. Similar and related studies with the present study are presented below.

2.1. Experimental Fuels

2.1.1. Canola and palm biodiesel in diesel engines

Wirawan et. al., (2005) evaluated the use of palm biodiesel blended with diesel fuel for certain amount (e.g. up to 30% or B30) and reported that has significantly improved emission quality, in the result of a 20,000 km road test carried out in Indonesia. Increment of the biodiesel content provided the emission reduction up to 25.35% for CO, 10.82% for NO_x + THC, 42.02% for particulate and 23.50% for opacity compared with diesel fuel.

Mofijur et al. (2014) reported the properties, performance, and emissions of 5% and 10% palm biodiesel blends (P5, P10) in a multi-cylinder diesel engine at various engine speeds. P5 and P10 produced 1.38%, 3.16%, and lower brake power and 0.69% and 2.02% higher BSFC, respectively, than diesel. P5 and P10 also

showed 1.96% and 3.38% increases in NO_x; 14.47% and 18.42% decreases in HC; and 13.17% and 17.36%, and decrease in CO, respectively, compared with diesel.

Liaquat et al. (2013) investigated the effects of P20 and diesel fuel during an endurance test and found 3.88% higher BSFC, 11.71% lower HC, 11% lower CO, and 3.31% higher NO_x emissions than diesel.

Ndayishimiye and Tazerout (2011) evaluated the engine performance and emission characteristics of a palm oil-based biodiesel in a diesel engine and found that a high percentage of palm biodiesel blended with diesel fuel decreases the heating value while increasing brake thermal efficiency. NO_x emissions and CO emissions were higher at a low load, but lower at a full load for a high percentage of palm oil methyl ester.

Buyukkaya (2010) studied the effect of turbocharger on the performance and exhaust emissions of a four-stroke, six cylinder, direct injection engine, fueled with biodiesel from rapeseed oil (canola oil) and its blends (B5, B20, B70, B100) at full load condition and speeds ranging between 1000 rpm to 2100 rpm. At 2000 rpm, the BTE values were 0.427, 0.425, 0.425, 0.424 and 0.423 for B20, B5, B70, B100 and diesel fuels, respectively. It was observed that the BTE for all biodiesel blends were higher than that of diesel. That may attribute to the additional lubricant provided by biodiesel. Similarly, a study done by Labeckas and Slavinskas (2006) with 4750 cc engine by using rapeseed (canola) oil biodiesel and obtained higher BTEs with 5–10% blends. Additionally to Buyukkaya (2010), reported that the BSFCs diesel and B100 fuels were 232 and 251 g/kWh (at 1400 rpm), respectively. BSFCs of the B5, B20, B70 and B100 fuels were observed to be higher by 2.5%, 3%, 5.5% and 7.5% than that of the diesel fuel, respectively. Also it is mentioned in the study that, at low engine speeds, the CO emissions of the B5, B20, B70 and B100 were 12%, 25%, 31% and 35% lower than that of diesel fuel respectively, on the other hand NO_x emissions were 5-10 % higher than neat diesel fuel.

A. N. Ozsezen and M. Canakci (2011) presented, the performance, combustion and injection characteristics of a direct injection diesel engine when it was fueled with canola oil methyl ester (COME) and waste (frying) palm oil methyl ester (WPOME) in their experimental study. In order to determine the performance

and combustion characteristics, the experiments were conducted at constant engine speeds under the full load condition of the engine. The results indicated that when the test engine was fueled with WPOME or COME instead of petroleum based diesel fuel (PBDF), the brake power reduced by 4–5%, while the brake specific fuel consumption increased by 9–10%. On the other hand, methyl esters caused reductions in carbon monoxide (CO) by 59–67%, in unburned hydrocarbon (HC) by 17–26%, in carbon dioxide (CO₂) by 5–8%, and smoke opacity by 56–63%. However, both methyl esters produced more nitrogen oxides (NO_x) emissions by 11–22% compared with those of the petroleum based diesel fuel over the speed range.

2.1.2. Natural gas in diesel engines

Eshank D.D. et. al (2013) examined the effect of CNG gas induction as well as CNG gas injection on the performance of CNG biodiesel operated dual fuel engine for both CNG and Honge oil methyl esters (HOME) on combustion of both manifold inducted and injected compressed natural gas (CNG) blended with HOME in a dual-fuel engine. From the experimental evidence it was found that combustion of HOME with CNG in a dual fuel engine operated with optimized parameters of injection timing of 27° TDC and compression ratio of 17.5 resulted in acceptable combustion emissions and improved brake thermal efficiencies. The implementation of injection strategies for both CNG and HOME fuels resulted in increased brake thermal efficiency and considerably reduce combustion emissions such as smoke, HC, CO and NO_x. They reported that; from the injection method, it was observed that, BTE was increased by 0.5 % compared to induction method. The obtained emission levels which were smoke, HC, CO decreased by 7.69%, 13.33%, 31.81%, respectively, beside NO_x increased by 1.45%.

Kyunghyun Ryu (2013) investigated the combustion and emissions characteristics of a compression ignition engine with a dual fuel (biodiesel–CNG) combustion system in his study. Experiments utilized a biodiesel pilot injection to ignite a main charge of compressed natural gas. The pilot injection pressure was maintained at approximately 120 MPa while the pilot injection timing was varied

across the range 11–23 crank angle degrees before top-dead-center to investigate the characteristics of engine performance and exhaust emissions in a single cylinder diesel engine. Results showed that performance can be optimized for biodiesel–CNG dual fuel combustion by advancing the pilot injection timing for low loads and delaying the injection timing for high loads. However, overall performance of diesel single fuel combustion still exceeds that of biodiesel–CNG dual fuel combustion. Compared to the diesel single fuel combustion, however, smoke emissions were significantly reduced over the range of operating conditions and NO_x emissions were also reduced except for the full load condition. Dual fuel combustion yields lower CO₂ emissions compared to diesel single fuel combustion over all engine conditions. Biodiesel–CNG dual fuel combustion resulted in relative high CO and HC emissions at low load conditions due to the low combustion temperature of CNG.

A.M. Namasivayam et. al (2010), reported in their study that in terms of performance and emissions they perform fairly similarly. This was because the physical, chemical and combustion properties of various methyl esters were comparable to those of conventional diesel. In order to reduce these emissions of HC and CO, alternative pilot fuels needed to be considered. As fuels employed during normal CI engine operation, both dimethyl ether (DME, a gaseous CI engine fuel) and water-in-fuel emulsions (conventional biodiesel mixed with varying concentrations of water) have shown that they reduce smoke and NO_x emissions significantly, while improving combustion quality. In this work, the performance of DME and water-in-biodiesel emulsions as pilot fuels was assessed. It was seen that the water-in-biodiesel emulsions did not perform as well as expected, as increased HC and CO emissions coupled with a mild change in NO_x levels was encountered (compared to conventional pilot fuel, in this case neat biodiesel). The emulsions performed very poorly as pilot fuels below a certain BMEP threshold. DME, while producing higher levels of HC and CO than neat biodiesel, managed to reduce NO_x significantly compared to neat biodiesel. Emissions of HC and CO, while higher than neat biodiesel, were not as high as levels seen with the emulsions. Thermal efficiency levels were generally maintained with the liquid pilot fuels, with the DME pilot producing comparatively lower levels. Dimethyl ether (DME), a

gaseous high-cetane fuel and twowater-in-fuel emulsions with different water concentrations (5%and 10% water by volume) were tested as pilot fuels duringdual–fuel combustion with natural gas. It was seen that at 1000r/min for low loads, pilot fuel emulsification reduced NO_x emissionsby about 20%.

Korakianitis et.al (2011) added a useful review paper to literature about natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions. They compared the open literature for SI and CI engines. In conclusion part, they expressed that, there is no significant loss in power in dual-fuel operation compared to conventional CI-engine operation provided a sufficient amount of natural gas-air mixture can be admitted in the chamber. In some of the literature the induction method employed prevents a premixed natural gas-air mixture to form in the inlet manifold (by keeping the natural gas supply separate from the incoming air until very close to the intake valve). This minimizes the volumetric efficiency penalty of natural gas induction or injection in the intake manifold, but it also results in a loss of power at higher speeds because comparatively lower amounts of natural gas are inducted per cycle. Failure of the pilot fuel to ignite the entire natural gas-air charge at the low and intermediate loads (as a result of low charge temperatures) causes lower thermal efficiencies. Low engine operating temperatures at these loads also result in lower NO_x emissions compared to normal CI-engine operation. Conversely, HC and CO emissions are significantly increased, while at high loads NO_x, HC and CO emissions are comparable to normal CI-engine levels. A variety of alternative high-cetane fuels can be used as pilot fuels, while water-in-fuel pilot fuel emulsions and water injection can be used to reduce emissions at select equivalence ratios. And they had attracted attention in the suggested future work that, the effects of various additions such as EGR or water (to reduce NO_x) or hydrogen (to accelerate combustion progress) are even less well known. A few computational studies are available in the open literature, but quality experimental data to provide basic insight or to validate the numerical work are currently not available.

Papagiannakis and team (2003, 2004, 2007, 2010 and 2012), has several experimental and theoretical studies on the dual fuel compression ignition engine

operated with pilot diesel fuel and natural gas. Their general approach was on a single cylinder, naturally aspirated, four-stroke, air-cooled, direct injection, high speed, having a bowl-in-piston combustion chamber Lister LV1 diesel engine, that; dual fuel combustion using natural gas as a supplement for liquid diesel fuel is a promising technique for controlling both NO and soot emissions. They mentioned the observed disadvantages concerning brake efficiency, HC and CO can be possibly mitigated by applying modifications on the engine tuning, e.g. injection timing of liquid diesel fuel mainly at part loads.

Moreover, in the Advancing Technology for America's Transportation report (chp.2) (2012), heavy-duty engines with perspective of 2015-2050; CNG vehicles are the strongest economic competitor to liquid ICE vehicles. As natural gas supplies have increased and prices have dropped, automakers and truck manufacturers have begun taking steps to introduce new vehicle lines fueled by natural gas, principally compressed natural gas (CNG) and liquefied natural gas (LNG). Several factors have referred from Ackerman (2015) and contributed to this interest:

- Price differential: On an energy-equivalent basis, oil has been more expensive than natural gas in recent years. Moreover, while oil prices are set in a global market, U.S. natural gas prices are largely determined domestically, and the discovery of large domestic reserves suggests that prices may remain relatively low.¹
- Environmental preference: Natural gas, while also a fossil fuel, generally produces lower emissions per vehicle mile than diesel and heavier oil.
- Energy diversity and security: Increased use of domestic natural gas for transportation may mean that less oil will be imported. The United States might be more insulated from global petroleum price volatility if more forms of transportation were based on natural gas.
- Growth potential: Only 2.9% of U.S. natural gas production is currently used in transportation, mainly to move gas through the pipelines, and expanded use of natural gas vehicles would likely lead to increased demand for natural gas.

Due to its high auto ignition temperature, natural gas needs an ignition source when used in engines, and two paths are in use today: spark ignition and compression ignition, using diesel fuel as the ignition source.

Both approaches are typically built of existing diesel engine blocks to retain the structural robustness and durability required in this market. The choice of ignition approach drives many of the subsequent technical requirements. Some of the key technical characteristics of the two approaches are summarized in Figure 2.1. (Advancing Technology for America's Transportation report (chp.14) (2012).

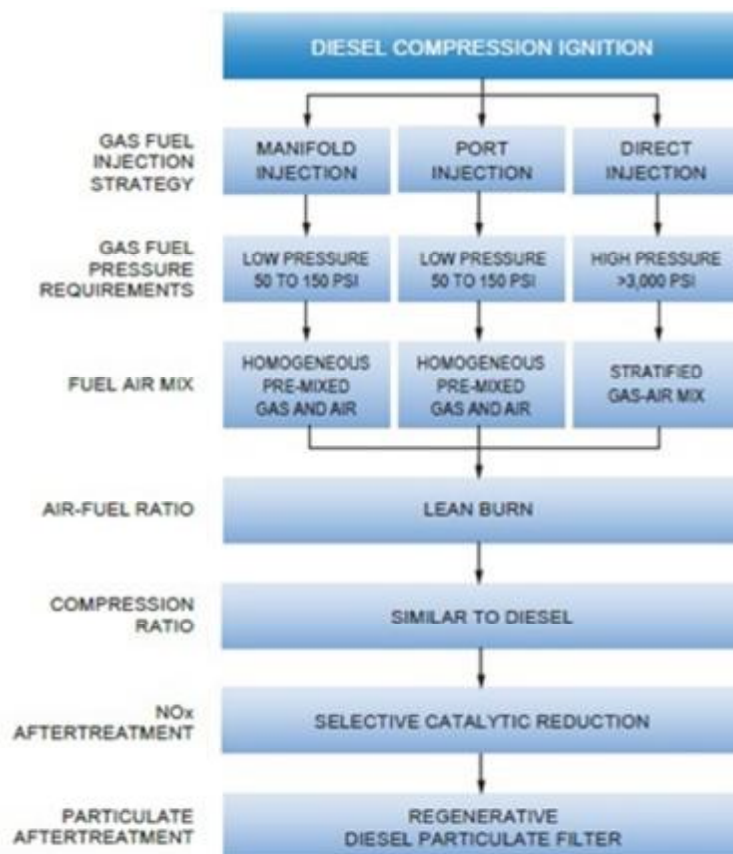


Figure 2.1. Technical approaches to NG Heavy-Duty engines. (ATAT report, 2012)

Kavtaradze et.al (2005) from Munich Technical University, presented a study on the ignition delay parameters in a diesel engine utilizing different fuels. They have suggested a theoretical-and-experimental method of determining the time of ignition delay, based on the chain theory of combustion of Semenov. And they

focused that, Formulas have been derived for the first time for the calculation of the time of ignition delay for advanced gas fuels (natural gas, various options of synthesis gas), as well as for conventional diesel fuel in view of recirculation of exhaust gases. Additionally, the suggested method may be employed to determine the ignition delay for almost any fuel used in diesel engines.

Carlucci et.al (2011) studied the combustion and emissions control in diesel–methane dual fuel engines: The effects of methane supply method combined with variable in-cylinder charge bulk motion. In the tests, diesel common rail research monocylinde engine used with 17.1:1 compression ratio. In the results, they demonstrated that; the port injection, as a methane supply method for dual fuel engines, is a very effective strategy to reduce unburned hydrocarbons and nitric oxides concentrations, especially when implemented with variable intake geometry systems to produce the suitable in-cylinder bulk motion and turbulence intensity for different engine operating parameters.

Bakar et.al (2012), presented the study of application of natural gas for internal combustion engines, which they compared the diesel engine and conversion to natural gas engine from the same engine. They introduced that, the original diesel engine cylinder pressure is higher than the modified diesel engine and CNG engine. It caused the compression ratio of NG engine is lower than the original diesel engine and the combustion energy output of diesel fuel has to produce the highest power than natural-gas fuel. Decreasing engine speed of NG engine has increase maximum temperature in engine. Engine torque, power, mean effective pressure and efficiency performance of original direct injection diesel engine cylinder pressure is higher than the modified diesel engine and sequential port injection dedicated NG engine. It meant that the thermodynamics energies were resulted from the diesel fuel combustion is higher than the NG fuel. It caused by the hydrocarbon chain, density and energy of diesel fuel of the diesel engine is higher than the gas fuel of NG engine were ignited using to spark assistant. The fuel consumption of NG engine is higher than the diesel engine. Fuel consumption was increased because of the unburned fuel was increased in the medium to the highest speed of NG engine. The effect of the increasing unburned fuel could decrease the engine brake torque and brake power.

The effect of lower brake torque and brake power increases the brake specific fuel consumption.

Karabektaş et al. (2013), reported the effects of using diethyl ether as additive on the performance and emissions of a diesel engine fueled with CNG. A single-cylinder, water cooled, direct injection, naturally-aspirated, Super Star diesel engine was used as a test bench. They concluded that, using natural gas as a dual fuel had a significant negative effect on the engine performance at low and medium loads. However, the engine performance improves at high engine loads. The use of natural gas as a dual fuel caused lower NO emissions at low and medium loads, while yielding higher NO emissions at high loads compared with the use of dual fuel. CO and HC emissions increased considerably with the use of natural gas at low and medium loads, while showing a decreasing trend at high loads. The addition of DEE to pilot fuel in a diesel engine using dual fuel caused lower specific energy consumption and higher brake thermal efficiency.

Mbarawa and Milton (2005) added theirs study in to the literature with the aspects of an examination of the maximum possible natural gas substitution for diesel fuel in a direct injected diesel engine. The engine used in their study was a Cummins Model 519, a six cylinder, naturally aspirated, four stroke DI diesel engine. As results drawn with; stable engine operation could be maintained with as much as 86 % NG energy substitution for diesel fuel. At low speeds of 1200 rpm to 1300 rpm, the maximum substitution of NG to 86% could provide torque levels equivalent to those for full load diesel fuel only operation; the overall equivalence ratios for DF exceeded those for straight diesel fuel operational mode at all loads and emissions of CO were higher for dual fuel than those for diesel fuel only operation.

2.1.3. Hydrogen in diesel engines

Baltacioglu et. al (2016), reported from their results that; hydrogen addition to biodiesel fuel is more environmentally friendly solution. On the other hand, HHO addition provided higher engine performance which can help to solve one of the biggest handicap of high level biodiesel blends usage. Additionally, biodiesel is a

naturally environmentally friendly and a good solution to decrease exhaust gas emissions. Unfortunately, high level of biodiesel fuel blends results with losses of performance for internal combustion engines. Under the guidance of these results, it was decided to use hydrogen addition to 25% and 50% biodiesel fuel blends. Because 25% and 50% biodiesel fuel blends need to support emission aspects than performance compared to 75% and 100% biodiesel fuel blends. So HHO became preferable addition to 75% and 100% biodiesel fuel blends which tolerates losses of the engines performances and provided better results than hydrogen addition.

Ozgur T. et. al. (2014) performed a simulation on performance, exhaust emission characteristics and combustion process of the engine fueled with hydrogen-diesel blends were compared to diesel fuel. Hydrogen was blended with diesel fuel at the volumetric ratios of 5%, 10% and 20%. AVL BOOST software was dedicated to simulate the performance and emission values for various blends of hydrogen with diesel fuel. The simulation results showed that hydrogen addition to diesel fuel improved both engine performance and exhaust emissions. More specifically, using hydrogen reduce the BSFC of the engine. NO_x and CO emissions are reduced by using hydrogen blend.

Tang et.al (2014) mentioned a review which titled with Progress in combustion investigations of hydrogen enriched hydrocarbons. More dominantly goal of hydrogen expressed as; hydrogen is a versatile energy carrier that can be made from a variety of primary energy sources such as natural gas, coal, and biomass, and non-carbon energy sources such as solar, nuclear, hydroelectric, and wind. Hydrogen can burn with high efficiency and produce essentially zero emissions. However, attractive as it is, a hydrogen economy is challenged by technical, economical and infrastructural barriers. For instance, there are significant difficulties associated with hydrogen storage due to its high flammability limits, low ignition energy (which causes safety problems), and its low volumetric energy content, which requires extra energy for high pressure storage. Additionally, there exist some unresolved issues for pure H_2 combustion, such as knock, detonation, pre-ignition, and flash-back. For these reasons, fossil fuels still dominate the current primary energy supply. To bridge this situation, a more rational approach is to use

hydrogen/hydrocarbon fuel blends, which synergistically resolve storage and combustion problems associated with burning pure hydrogen or pure hydrocarbons.

Ciniviz and Köse (2012)'s review study mentioned the hydrogen usage on diesel engines increases the H/C ratio of the entire fuel. Secondly, injecting small amounts of hydrogen to a diesel engine could decrease heterogeneity of a diesel fuel spray due to the high diffusivity of hydrogen which makes the combustible mixture better premixed with air and more uniform. However hydrogen cannot be used as a sole fuel in a compression ignition (CI) engine, since the compression temperature is not enough to initiate the combustion due to its higher self-ignition temperature.

In another study of Ciniviz and Köse (2005), they reported the pilot fuel quantity may be in the range of 10–30% while the rest of the energy is supplied by the main fuel. Hydrogen operated dual fuel engine has the characteristics to operate at leaner equivalence ratios at part loads, which results in NO_x reduction, and increase in thermal efficiency thereby reducing the fuel consumption. Oxides of nitrogen (NO_x) are the major problem in hydrogen operated dual fuel engine.

Lilik et.al (2010) explained the sole fuel usage of hydrogen situation with “Hydrogen has an autoignition temperature of 858 K requiring an ignition source to burn in an IC engine. Diesel fuel, which has an auto-ignition temperature of 525 K, can be used as a pilot fuel to ignite hydrogen. The literature contains many reports in which hydrogen was used in conjunction with diesel fuel to power CI engines. This “dual-fuel” combustion is often called “diesel pilot-ignited hydrogen combustion.” Diesel pilot-ignited hydrogen combustion at low quantities of hydrogen is beneficial since the diesel fuel is being replaced by hydrogen, which may stretch the supply of hydrocarbon fuels, but a range of emissions impacts have been reported for this process”.

Saravanan and Nagarajan (2008) performed a study which introduced the experimental investigation of hydrogen-enriched air induction in a diesel engine system. They used a four-stroke, water-cooled, single cylinder stationary diesel engine developing power of 3.78kW at the rated speed of 1500 rpm. Besides, they mentioned the system of study with the process of mixing air and fuel is called enrichment. By varying the percentage of diesel (10%, 30%, 50%, 70%, 80% and

90%) by volume (l/min), they studied the performance and emission characteristics of the hydrogen-enriched engine.

The pilot quantity of diesel was set and the pump rack locked and hydrogen flow varied simultaneously to maintain a constant speed. Researchers concluded; brake thermal efficiency outputs increased to 29.1% with 90% hydrogen enrichment, but results in knocking. Best results are obtained with 30% hydrogen: an efficiency of 27.9% is achieved without knocking over the entire load range. Specific energy consumption decreases with increase in hydrogen percentage over the entire range of operation. PM decreased significantly from 4 to 1 g/kWh with 90% hydrogen enrichment. Figure 2.2.(Saravanan, 2008), is illustrated below to presents the variation of heat release with crank angle at 30% hydrogen enrichment mixture at full load condition.

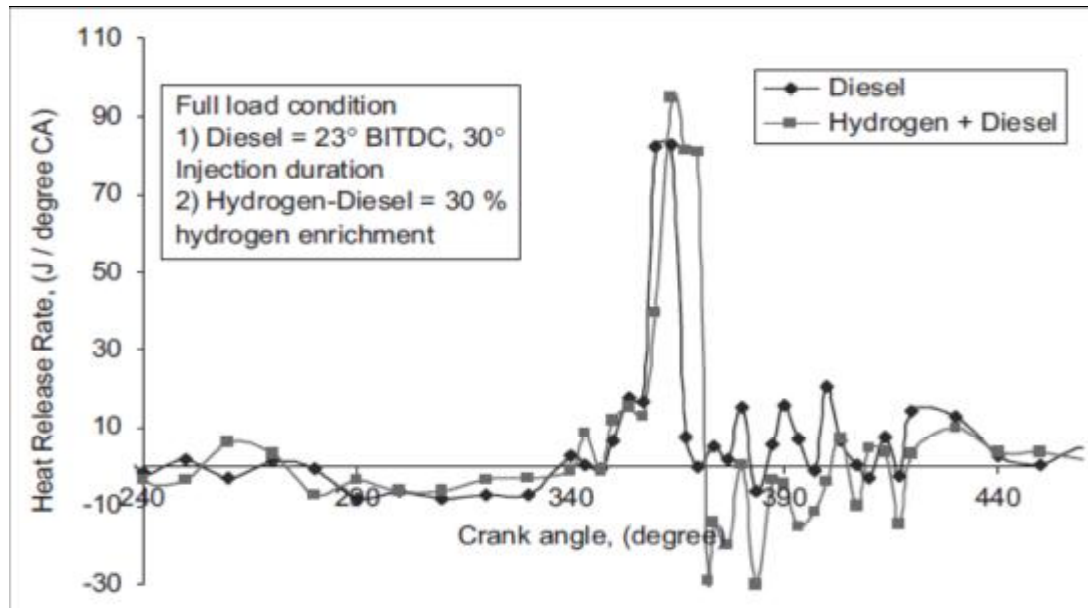


Figure 2.2. Variation of heat release with crank angle at 30% hydrogen enrichment mixture at full load condition

Stanislaw Sz waj and Karol Grab-Rogalinski (2009), investigated both pure hydrogen combustion under HCCI (homogeneous charge compression ignition) conditions and hydrogen–diesel combustion in a compression ignition (CI) engine. They concluded that, applying hydrogen as an extra fuel, which can be added to diesel fuel in the CI engine, seemed to be a reasonable measure to improve engine

performance because this measure approaches the combustion process to constant volume. It was achieved for a hydrogen dose of 25% or more but a heavier combustion knock is also generated in this case. Hydrogen added in small amounts to a diesel engine did not generate knock at a level harmful for the engine. The maximum limit for the hydrogen dose for the specific engine taken for these tests was approximately 16% with respect to energy percentage. A small dose of hydrogen added to a diesel engine did not make the engine run worse. Small amounts of hydrogen (e.g. about 5%) when added to a diesel engine shortens the diesel ignition lag and, in this way, decreases the rate of pressure rise. It provided better conditions for soft engine run and increased engine durability.

M. Senthil Kumar et. al (2003), mentioned that indicated an increase in the brake thermal efficiency from 27.3% to a maximum of 29.3% at 7% of hydrogen mass share at maximum power output. Smoke was reduced from 4.4 to 3.7 BSU at the best efficiency point. There was also a reduction in HC and CO emissions from 130 to 100 ppm and 0.26–0.17% by volume respectively at maximum power output. With hydrogen induction, due to high combustion rates, NO level was increased from 735 to 875 ppm at full output. Ignition delay, peak pressure and maximum rate of pressure rise were also increased in the dual fuel mode of operation. Combustion duration was reduced due to higher flame speed of hydrogen. Higher premixed combustion rate was observed with hydrogen induction. Comparison was made with diesel being used as the pilot fuel instead of vegetable oil. In the case of diesel the brake thermal efficiency was always higher. At the optimum hydrogen share of 5% by mass, the brake thermal efficiency went up from 30.3–32%. Hydrocarbon, carbon monoxide, smoke emission and ignition delay were also lower with diesel as compared to vegetable oil. Smoke level decreased from 3.9 to 2.7 BSU with diesel as pilot at the optimum hydrogen share. Peak pressure, maximum rate of pressure rise, heat release rate and NO levels were higher with diesel than Jatropa oil. On the whole, it was concluded that induction of small quantities of hydrogen can significantly enhance the performance of a vegetable (Jatropa) oil/diesel fueled diesel engine.

Maki and Prabhakaran (2011), investigated the performance and emissions of a multi cylinder diesel engine fueled with hydrogen-diesel blends experimentally. They used a test bench which consisted with a four cylinder, four stroke, water cooled, indirect injection, naturally aspirated, and compression ignition engine and running at varied speed equipped with a hydrogen induction system and an eddy current dynamometer with variable loading arrangement. Their result showed that; a continuous hydrogen induction into the inlet manifold is a unique way of addressing simultaneously issues related to thermal performance and pollutant emission from diesel engine operated with diesel as a fuel. There was a monotonous effect of continuous hydrogen induction rate on thermal performance parameters such as brake thermal efficiency, diesel fuel consumption rate and volumetric efficiency. And they mentioned that, it was seen that hydrogen induction rate about 7.5 lpm gives an optimum performance keeping the emissions level at a reasonable low levels. At 7.5 lpm, the levels of CO, CO₂ and HC are not increase significantly while the NO_x is reduced and same rate approximately reduced the diesel fuel consumption by 20% and increased the brake thermal efficiency by about 8~9%.

2.1.4. Hydroxy (HHO) in diesel engines

J. Allen Jeffrey and M.Subramanian (2014) conducted a research to examine the performance and emission parameters of biodiesel and HHO supplementing biodiesel. When HHO was made to supplement with biodiesel there was rise in brake thermal efficiency and reduced fuel consumption when compared to biodiesel blend. But in terms of emission CO and HC emissions were low or same at some points of engine operation. High burning velocity and oxygen content in HHO has reduced CO and HC emissions. But the NO_x emission were high in HHO supplementing biodiesel than biodiesel blend this was because of rise in temperature due to increasing quantity of oxygen.

Durairaj et.al (2012) focused on the production, characterization of biodiesel and opportunities to aid with oxy-hydrogen gas which produced by the electrolysis process from the water. Additionally, they had preheated HHO gas with the help of

waste heat recovered from the engine exhaust. As a result, they mentioned the use of HHO and biodiesel in conventional engines reduced the emissions of unburned hydrocarbon, CO, and particulate matter. Beside, preheating was resulted with less vibration of the engine.

Yılmaz et.al (2010) produced a hydrogen generator (plexiglas reactor) and used various ion conducting ions such as; KOH, NaOH, NaCl. Besides, they produced a hydroxy electronic control unit (HECU) to decrease HHO flow rate by decreasing voltage and current automatically by programming the data logger. They concluded the results with HHO system addition to the engine without any modification, an increasing engine torque output by an average of 19.1%, reducing CO emissions by an average of 13.5%, HC emissions by an average of 5% and SFC by an average of 14% (same test engine and dynamometer were used in their study which has used during the experiments of this thesis). Unlike present thesis, they used the enrichment of HHO to diesel engine method, the major differences is using pilot injection method and dual-fuel engine mode. Moreover, HHOCNG+biodiesel replacement is using for the first time in literature with this thesis.

Sankar T, (2014) examined the performance of four stroke diesel engines by using alternative fuels. The performance and emission levels were calculated for the different blended ratios of karanj oil methyl ester. Fuel mixtures were consisting, D100%, D+HHO, K100%, K+HHO, K25 % + HHO, K50% + HHO, K75% + HHO. They found out that the emission level was reduced by the combination of Karanj and HHO with diesel and the engine performance are maintained as per the diesel standards. It is recommended that the fuel combination of D75% + K25% + HHO for the biodiesel purpose without any change in diesel engine. In ratio D75% + K25% + HHO obtained results were CO=0.04%, CO₂=4.8%, HC=29ppm, NO_x=292ppm, SFC=1.81kg/sec.kw, η_m =63.92%, η_{bt} =34.61%, η_{it} =37.61%.

Birtas and Chiriac (2011) added a study to literature about injection timing for a diesel engine operating with gasoil and hydrogen rich gas with simulation method. They used a 3.8 L, naturally aspirated, direct injection and 17.5:1 compression ratio diesel engine to predict the simulation. They mentioned the results of the study as; the simulations performed with the AVL Boost program have showed that for

reduced HRG flow rates and optimized diesel fuel injection timing. Slight increase in the engine brake thermal efficiency and moderate increase of the maximum cylinder pressure was obtained. Significant smoke and CO emission were reduced, but important NO_x emission increase was performed. These aspects seem to be particularly interesting from the perspective of modifying the injection characteristic of the injection pump, in direction of advancing the static injection timing.

Bari and Esmaeil (2010) investigated the impacts of using a small amount of H_2/O_2 mixture as an additive on the performance of a four-cylinder diesel engine. The required amount of the mixture was generated using electrolysis of water considering on-board production of H_2/O_2 mixture. Hydrogen which has about nine times higher flame speed than diesel has the ability to enhance overall combustion generating higher peak pressure closer to TDC resulting in more work. The results of their study showed that with the introduction of 6.1% total diesel equivalent H_2/O_2 mixture into diesel, the brake thermal efficiency increased by 2.6% at 19 kW, 2.9% at 22 kW, and 1.6% at 28 kW. The brake specific fuel consumption of the engine reduced by 7.3%, 8.1%, and 4.8% at 19 kW, 22 kW, and 28 kW, respectively.

Zamit et.al (2012) had an experimental investigation about the effects of hydrogen enhanced combustion with comparisons between SI and CI engines on performance and emissions. They used two different HHO generator (additional HHO volume: 0.3 lpm and 1.67 lpm) and three different engines (Ford 1L and 1.4L petrol and Peugeot 1.8 L diesel). They suggested that, a comparison of operation at peak efficiency and peak power output with and without HHO addition showed that in most cases the increase in output power was still not enough to produce the hydrogen, even if a 75% efficient hydrogen generator was to be used. They referred that, test results showed, in agreement with the findings of other researchers, that hydrogen addition produced the best effects at lean mixture operation and HHO addition also extended the lean limit of the engine under many of the testing points.

Samuel and McCournic (2010) studied the hydrogen enriched diesel combustion. In experiments, they used a single cylinder research diesel engine under conducted 1500 rpm and 5.3 bar load with 2.8 lpm HHO addition. As a result, they introduced that, demonstrated that a device that can electrolyze the water using a

conventional battery system can be used to enrich the air with HHO without modifying the engine at all. Their system provided up to 5.4% fuel economy and a similar reduction in CO₂ values. Increased amount of oxy-hydrogen to the intake system reduced the duration of combustion and increased the rate of pressure rise without altering the location of combustion.

2.1.5. Hydrogenated compressed natural gas (H/HHO-CNG) in diesel engines

N.R. Banapurmath et al. (2014) attempted to study the behavior of some gaseous fuels such as CNG and HCNG in diesel engines on dual-fuel mode. Engine tests were conducted on a four-stroke single cylinder water-cooled DI compression ignition engine with a displacement volume of 0.662 liter, compression ratio of 17.5:1 and developing power of 3.7 kW at 1500 rev/min. The engine was always operated at a rated speed of 1500 rev/min. The flow rates for both CNG and HCNG were kept constant at 0.5 kg/h in order to compare their performance in dual-fuel engine. The liquid fuel of HOME was common. From the study the following conclusions were made.

- Single fuel operation resulted in higher BTE than dual-fuel operation.
- HOME-CNG/HCNG operation can partially substitute fossil fuels
- The soot and NO_x emission levels were found to be lower for dual-fuel operation than for neat liquid fuels operation.
- HCNG resulted in increased performance. Addition of hydrogen to CNG as a fuel in dual-fuel engines resulted in significant improvement on performance; HCNG makes it possible to run the engine on a leaner mixture, resulting in lower emissions of HC and CO and higher NO_x emissions.
- Use of HCNG in CI engines on dual-fuel mode helps to meet the Euro-V norms which can be enforced in the near future in India.

N.M. Gireesh et. al (2014), were conducted an experimental investigations on a single-cylinder four-stroke water cooled direct injection (DI) compression ignition (CI) engine operating in dual fuel mode using diesel/Honge oil methyl ester (HOME) as injected fuels and varying hydrogen percentage in CNG gas combinations. Percentage of hydrogen in CNG was varied from 5 to 20% in steps of 5% and the resulting mixture was inducted into the engine manifold through a mixing device suitably developed in-house. Results showed a considerable improvement in brake thermal efficiency with reduced smoke, hydrocarbon (HC), carbon monoxide (CO) emissions and marginally increased nitric oxide (NOx) emission levels with increased hydrogen flow rates. Furthermore, the ignition delay, combustion duration, pressure-crank angle and heat release rate were analyzed and compared with base data. Their combustion analysis showed that the rapid rate of burning of CNG-air mixture with the increased addition of hydrogen resulted in higher cylinder pressure and energy release rates. Hydrogen blended with CNG enabled leaner operation and showed an improvement in overall performance and environmental benefit.

2.2. Dual Fuel Engines with Pilot Diesel Injection

General approach to dual-fuelling in compression-ignition engines is a mode of combustion where a small pilot injection of high-cetane fuel (i.e. diesel, biodiesel) ignites a premixed high-octane fuel (i.e. CNG) and air mixture. This allows conventional CI engines to decrease their exhaust emissions such as smoke and nitrogen oxides while maintaining their high thermal efficiencies and performance outputs. However, poor ignitability of the main fuel-air charge results in increased emissions of unburnt hydrocarbons and carbon monoxide. Conventional pilot fuels such as diesel and biodiesel have been researched extensively in this experimental study. Dual fuel systems cover very important place in internal combustion engines literature to improve the performance and to reduce fuel consumption and harmful exhaust emissions. Most important and similar researches are summarized from literature and presented in this subsection.

In recent engines, the development of dual fuel technology is motivated by emissions regulations which require specific limits be met for pollutants released into the atmosphere. Development is also motivated by the potential savings on operations costs for dual fuel engines compared to diesel engines. For these reasons, many diesel engines are converted to run with gaseous fuels such as natural gas and retain positive features of diesel operation. Natural gas is preferred because it is cheap and widely available. It also leads to reduce emission problems in diesel engines (Mansor, 2014).

Euro emission regulation for on-road diesel engines was introduced in 1989. Euro 0, I and II used a test cycle according to ECE R49 which was a 13-mode steady state cycle. With the introduction of Euro III that cycle was replaced by two steady state cycles, ESC and ELR, and a new dynamic test was introduced, ETC. Euro III has been followed by Euro IV and Euro V. The latest emission regulation, Euro VI, was introduced in 2014. In figure 2.3 European and North American NO_x and PM Emission history is presented.

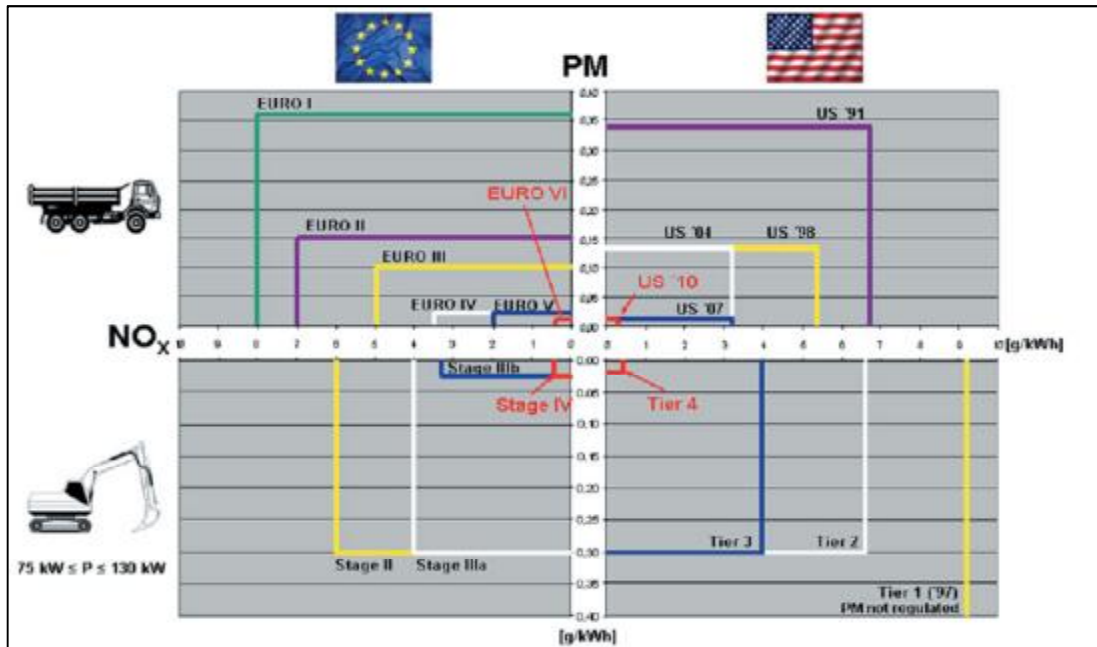


Figure 2.3. European and North American NO_x and PM Emission history (AVL, 2010)

L. Tarabet (2014) 's experimental investigation has been conducted to examine the effect of using eucalyptus biodiesel and natural gas to run a single cylinder DI diesel engine in dual fuel mode under various engine loads. Test engine modified properly to operate under this mode and its basic configuration had maintained. They concluded that; the analysis of combustion has shown that biodiesel in dual fuel mode improves combustion stability and particularly at high engine loads. Unlike diesel combustion in dual fuel mode, which improves the cylinder pressure peak only at higher loads, biodiesel combustion with NG gives higher pressure peak over the entire load range. Also, the ignition delay for NG–biodiesel indicates a shortened trend compared to diesel fuel in conventional and dual fuel modes due to the high cetane number of biodiesel. Overall, dual fuel combustion using natural gas as a supplement for eucalyptus biodiesel is an attractive technique for remedying to the deficiencies of using diesel fuel in conventional and dual fuel modes such as UHC, CO, CO₂ and particulate matter emission levels. This is particularly true at high engine load.

T. Korakianitis et. al. (2011) conducted an experimental study. Rapeseed methyl ester (RME) and diesel fuel used separately as pilot fuels for dual-fuel compression–ignition (CI) engine operation with hydrogen gas and natural gas (the two gaseous fuels were tested separately). During hydrogen dual-fuel operation with both pilot fuels, thermal efficiencies were generally maintained. Hydrogen dual-fuel CI engine operation with both pilot fuels increased NO_x emissions, while smoke, unburnt HC and CO levels remained relatively unchanged compared with normal CI engine operation. During hydrogen dual-fuel operation with both pilot fuels, high flame propagation speeds in addition to slightly increased ignition delay result in higher pressure-rise rates, increased emissions of NO_x and peak pressure values compared with normal CI engine operation. During natural gas dual-fuel operation with both pilot fuels, comparatively higher unburnt HC and CO emissions were recorded compared with normal CI engine operation at low and intermediate engine loads which were due to lower combustion efficiencies and correspond to lower thermal efficiencies. This could be due to the pilot fuel failing to ignite the natural gas–air charge on a significant scale. During dual-fuel operation with both gaseous

fuels, an increased overall hydrogen–carbon ratio lowers CO₂ emissions compared with normal engine operation. Power output (in terms of brake mean effective pressure, BMEP) as well as maximum engine speed achieved was also limited. This results from a reduced gaseous fuel induction capability in the intake manifold, in addition to engine stability issues (i.e. abnormal combustion). During all engine operating modes, diesel pilot fuel and RME pilot fuel performed closely in terms of exhaust emissions. Overall, CI engines can operate in the dual-fuel mode reasonably successfully with minimal modifications. However, increased NO_x emissions (with hydrogen use) and incomplete combustion at low and intermediate loads (with natural gas use) were concerns; while port gaseous fuel induction limits power output at high speeds.

The satisfactory operation of dual-fuel engines depends on numerous operating and design variables that tend to be greater in number than those controlling the performance of conventional spark ignition or diesel engines and refereed by Karim (2015).

- The type of fuel used, its composition and heating value, the physical and chemical properties of the fuel, and their variations with temperature and pressure; values of the effective flammability limits and burning rates, especially at the high temperatures and pressures encountered in engines at the end of compression and initiation of the combustion process.
- Intake and exhaust temperatures and pressures, the equivalence ratio values used, the presence of any diluents in the fuel or air supplied, the autoignition characteristics, ignition energy and limits, and their corresponding variations with equivalence ratio and temperature, and operational knock limits.
- The associated operational volumetric efficiency with throttling when employed, turbocharging, and the extent of exhaust gas recirculation (EGR) used; the nature and extent of any fuel and temperature stratification, the type of pilot fuel employed, its relative size, injection characteristics, and timing.

- Values of the engine compression ratio, bore, and stroke, combustion chamber shape, engine speed range, and any surface activity.

The modern high-compression-ratio diesel engine is imminently suited to dual-fuel operation, particularly when using a fuel such as methane, the main component of a wide range of natural and biogases. Largely by virtue of the provision of a high compression ratio with excess air and an adequate sized, consistently timed pilot liquid fuel injection, almost any gaseous fuel or vapor can be utilized to a varying degree of success. A wide range of pure gaseous fuels, such as methane, ethane, propane, butane, hydrogen, and ethylene, have been employed in dual-fuel engine applications.

Karim G.A.(2015) suggested that; some of the contributory factors to these trends could have included the following:

- The combustion process in the gas-fueled diesel engine of the dual-fuel type is quite complex. It displays some of the features and associated problems of the compression ignition diesel engine, as well as those of the premixed spark ignition type.
- The exhaust emissions of dual-fuel operation, until relatively recently, were not easy to deal with satisfactorily when in combination with the widely varying quality of the fuel systems possible, over the whole ranges of load and speed. However, many recent advances have been made, making it easier to deal with these difficulties.
- To obtain the potential benefits associated with gas-fueled operation, many influencing operational and design factors require careful optimization and control. Suitable computer controls of the many engine variables were not widely available in the past, while relying mainly on mechanical type control devices. This tended to make transport applications harder to implement satisfactorily and lacked the versatility and reliability required for the frequent transients in performance. The

need to provide two complex sets of controls and adequate portable storage for two sets of fuel systems of widely different properties has also contributed to the challenges.

- There has been a continuing need to retain the capacity of gas-fueled engines to revert to the diesel operational mode promptly and smoothly when needed. This facility is required to be retained without undermining the performance of the engine as a diesel. This sometimes represented a challenge since diesel engines are required to satisfy increasingly stricter and more challenging requirements of performance and controls.
- Until recently, relatively small diesel engines were less operationally flexible and tended to be more costly in comparison to their spark ignition counterparts. Accordingly, the exclusive conversion to fuel gas spark ignition operation was easier to accomplish and more satisfactorily performed, especially while retaining the bi-fuel operational capacity.
- Large output stationary engines, such as those employed exclusively for the generation of electric power, consume very large quantities of fuel. The need to lower fuel costs and maintain high efficiencies is an important consideration. These tended to favor employing the efficient and reliable liquid-fueled diesel engines, which in recent years have reached a very advanced stage of sophistication and reliability. They are readily turbocharged, and their performance is well optimized. However, only when relatively cheap and abundant fuel gas supplies were available were dual-fuel engines employed.
- A consequence of operating diesel engines in the premixed dual-fuel mode is that some of the intake air is displaced by the gaseous fuel induced. The amount of air displaced increases with the decrease in average molecular weight of the fuel. For example, for stoichiometric mixtures, the decrease by volume is 3.13% for butane, 4.03% for propane, 5.66% for ethane, 9.51% for methane, and 29.6% for hydrogen. This air displacement can have a significant influence on the volumetric efficiency, emissions, and associated power output of the engine,

requiring remedial measures to recoup the potential shortfall in engine power output and performance. Otherwise, relatively bigger engines are needed with increased capital and operational costs.

- The low efficiency and sensitivity to changes in the fuel gas composition of the low-compression-ratio spark ignition gas engine made it fall increasingly out of favor in comparison to the gas-fueled diesel engine. However, diesel engines, with their continued improvements in the control of their emissions, efficiency, and reliability, tended to relegate the dual-fuel engine to a secondary role, confining it increasingly to special applications where economic advantages can be assured through the exploitation of much cheaper gaseous fuels. The increased attention in recent years to reduce the emission of greenhouse gases is aiding in the widespread application of dual-fuel engines in transport applications.

Although dual fuel systems have been operating satisfactorily and economically, there is still room for reducing costs and enhancing their performance further, whether in terms of efficiency, power production, maximizing diesel fuel replacement, displaying a wider tolerance to changes in gaseous fuel composition, or ensuring additional improvements in exhaust gas emissions.

Since pilot diesel injection has been mentioned as one of the very important points of dual fuel engines, specific parameters have been explained carefully previously by researchers. This type of injection related with several parameters and detailed studies for an effective combustion presented below.

Nwafor (2000) examined the effect of choice of pilot fuel. It mentioned that, test series were designed to find means of replacing a diesel fuel pilot injection with an alternative environmentally friendly pilot fuel for combustion of natural gas in a diesel engine. Natural gas has a high self-ignition temperature (704°C) and cannot be used in C.I. engines without a means of initiating combustion since the temperature attained at the end of the compression stroke is relatively lower than the autoignition temperature of the gas. In this case, ignition is brought about by the injection of a small pilot liquid fuel. Baseline test results on the diesel fuel pilot were first

established. The gas flow rate for various pilot fuel operations was the same for a given speed and load, and the variable pilot fuel was controlled by the engine governor. Important result could be drawn that, the overall test results indicate that engine performance with the plant oil injection was satisfactory and comparable with the diesel fuel pilot operation.

Mbarawa et.al (2001) studied the experiments and modeling of natural gas combustion ignited by a pilot diesel fuel spray. The main of goal of the study expressed as; improve the understanding of how a pilot injection of diesel fuel affects the combustion of a natural gas–air mixture in an environment approximating that of a diesel cycle. Study included the combustion rate of mixture burning rates of natural gas and diesel and a three-dimensional (3D) numerical model incorporating complex interaction between the fuels during combustion was used. As the results of experiment conclusion, simulation results compare well with the experiments. It was shown that NG combustion in diesel environments can be improved by using an injector with a great number of small holes and by increasing the injection pressure of the pilot diesel fuel, in this case to about 60 Mpa.

Liu (2006) examined the combustion noise of dual fuel and reported that, the power output at constant pilot and speed varies essentially linearly with the extent of fuel gas admission. Also, the noise at very light load tends to be mostly associated with the rapid combustion of the diesel pilot, but decreases rapidly with the increased admission of methane. The noise later becomes much less dependent on the amount of gas burned which is presented in figure 2.4. below.

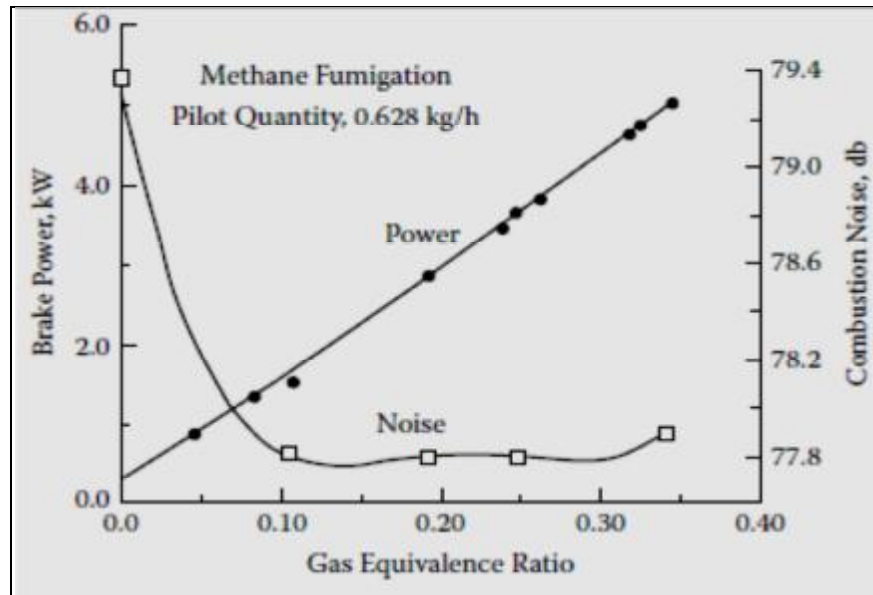


Figure 2.4. Variation of combustion noise output with gas equivalence ratio at constant pilot and speed (Adapted from Liu,2006).

Sahoo et.al (2009) performed a critical review for effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines. Study has a lot of useful parameters for understanding the dual fuel diesel engines. Paper is explained the effect of pilot fuel injection timing clearly. According to this explanation, the injection timing of the pilot fuel is an important factor that influences the performance of dual-fuel engines. For a fixed total equivalence ratio, advancing the injection timing increase the peak cylinder pressure because more fuel is burned before TDC and the peak pressure moves closer to TDC. Retarding the injection timing decreases the peak cylinder pressure because more of the fuel burns after TDC. This is because, the pilot fuel combustion is delayed and thus, the temperature of the mixture is not enough to propagate the flame in the whole gaseous fuel-air mixture; and consequently, incomplete combustion of the gaseous fuel mixture takes place. The charge temperature increases with advancing the injection timing of the pilot fuel and the associated higher energy release rates of the mixture. Similarly, the rates of pressure rise during the combustion of the gaseous fuel increases with advancing the injection timing of the pilot fuel. As s general results concluded by Sahoo et al (2009) consisting the effects of; engine load & speed, pilot fuel injection timing, mass of pilot fuel inducted, engine compression ratio, engine

intake manifold conditions and type of gaseous fuels. And mentioned, it seems that dual-fuel combustion using gaseous fuels is a promising technique for controlling both NO and soot emissions even on existing diesel engines with slight modification to the engine structure. The penalty in 'bsfc' experienced is partially compensated by the lower price of gaseous fuels. The observed disadvantages, at low engine load condition, concerning 'bsfc', HC and CO can be reduced by applying modifications in engine tuning, i.e. injection timing of the pilot fuel. Again, in diesel dual-fuel engines the ignition characteristics of the gaseous fuels are still to be understood and needs more research on it. Thus, in overall, the engine operating and design parameters, and selection of type of gaseous fuel has to be chosen accordingly for an existing diesel engine to run on dual-fuel concept. This can minimize the engine performance, combustion and emission characteristics divergences between the existing diesel engine and a dual-fuel diesel engine.

Papagiannakis et.al (2010) examined with a single cylinder, naturally aspirated, air-cooled, high speed, direct injection diesel test engine, located at the authors' laboratory, properly modified to operate under dual fuel (i.e. diesel and natural gas) mode. The results which are reported concern the brake thermal efficiency, exhaust gas temperature, and NO, CO, HC and soot emissions for various engine operating conditions, i.e. loads and speeds. Additionally, the maximum percentage of the liquid diesel fuel substituted by natural gas is up to 86%, while the least total relative air-fuel ratio goes down to 1.28 depending on engine operating conditions, i.e. load and speed.

Liu et.al (2013) introduced the results of effects of pilot fuel quantity on the emissions characteristics of a CNG/diesel dual fuel engine with optimized pilot injection timing experiments. They touched upon the emission characteristics of the CNG/diesel dual fuel engine and mentioned, reducing NO_x emissions averagely by 30%, unburned HC emissions are obviously higher, HC emissions occurs by unburned methane around 90% and PM was increased with the increase of pilot diesel quantity.

Torregrosa et.al (2013) studied the sensitivity of combustion noise and NO_x and soot emissions to pilot injection in PCCI diesel engines. Results showed that,

PCCI combustion with two injections reduces NO_x and soot but affects engine torque, PCCI combustion noise level and quality depend on the pilot injection, combustion noise must be improved before PCCI becomes a suitable alternative.

Yang et.al (2015) mentioned the parametric investigation of natural gas port injection and diesel pilot injection on the combustion and emissions of a turbocharged common rail dual-fuel engine at low load study. A common rail diesel research engine was converted to operate in dual-fuel mode and extensive experiments were conducted to investigate the effects of natural gas injection timing on the combustion and emissions performance under different pilot injection pressure and timing at low load conditions in experiment. Consequently they mentioned that, employing appropriate natural gas injection timing accompanied with reasonable pilot injection parameters is critical to further improve combustion performance and exhaust emissions of a dual-fuel engine at low loads. Additionally their experimental study showed that the advanced pilot injection timing and higher pilot injection pressure obtains better BTE and emissions except for NO_x.

3. MATERIAL AND METHOD

The objective of this study was to reduce diesel fuel dependency or totally eliminate it by HHOCNG and HCNG fuel replacement. Mixing ratios between hydrogen and CNG were 15%, 25%, 35% (by volumes). Another important issue was to follow nature of dual fuel engines by pilot fuel injection method. Additionally, biodiesel fuel blends are produced and used instead of diesel fuel for pilot injection. Main idea of this thesis was to success about harmful exhaust emission and maintains or improves the engine performance without modifications. All of these criteria are explained detailed in this section. Beside, this part of thesis consists parts of the author (2015)'s papers which were titled and also placed at references "Experimental comparison of pure hydrogen and HHO (hydroxy) enriched biodiesel (B10) fuel in a commercial diesel engine" and "Optimizing the quantity of diesel fuel injection by using 25HHOCNG gas fuel mixture"

3.1. Test Engine Rig, Hydraulic Dynamometer and Exhaust Emission Measurement Device

All tests were carried out in the Internal Combustion Engine (ICE) Laboratory of Automotive Engineering Department of Çukurova University at Adana City in Turkey. A schematic of experimental set up is illustrated in figure 3.1. Also technical specifications of the test engine and hydraulic dynamometer are given in table 3.1. Performance tests were conducted on a 3.6L, four cylinders, four-stroke, naturally aspirated, water-cooled direct injection with glow plug CI engine (Mitsubishi Canter) which connected to a Netfren mark hydraulic dynamometer for loading the test engine. Performance data from engine and hydraulic dynamometer was obtained with specific software program of Netfren. All experiments were conducted at full load condition in the range of 1300-2500 rpm and plotted by 200 rpm increments. Exhaust emissions were measured by MRU Delta 1600 V gas analyzer which was directly conducted to host computer and data were obtained with

specific software of MRU. (Baltacioglu et. al, 2015) MRU Delta 1600 V device is pointed with number ‘3’ as shown in figure 3.1.



Figure 3.1. Experimental rig set up

Table 3.1: Test engine and Hydraulic dynamometer specifications.

Brand	Mitsubishi CanterD.I. diesel	NetfrenHydraulic dynamometer	
Model	4D34 - In line 4	Torque range	0-1700 Nm
Type	D.I. diesel with glow plug	Speed range	0-7500 rpm
Displacement	3.6	Body diameter	250mm
Bore	104mm	Torque arm length	250mm
Stroke	105mm		
Power	89kW at 3200rpm		
Torque	295Nm at 1800rpm		

3.2. Determination of diesel fuel optimum substitution as pilot diesel injection

Substitution of liquid fuel is may be the most critical and sensitive point of this thesis. Fuel pump of the test engine has disassembled and transferred to the

Bosch Adana Service for calibration. Determination of optimum fuel injection limits and manual readjustment tests are shown in figures 3.2.a/b below.



Figure 3.2.a/b. Bosch fuel-injection pump test bench / Manual readjustment of fuel pump plunger

Optimizing the substituted diesel fuel quantities is one of the major goals of this study. Mechanical direct injection diesel fuel pump which used for fuel delivery to engine cylinders was re-set by Toshiba TA 8435 stepping motor driver to reduce amount of fuel. The stepping motor and stepping motor driver were used for substitution of diesel fuel to obtain more sensitive measurements. Plunger pin of fuel pump twisted clockwise to reduce the amount of diesel fuel injection to the cylinders. For this reason substitution of diesel fuel was obtained with the help of stepping motor and devices which mounted on mechanical diesel fuel pump plunger. A bipolar, 1.8 deg/step, 6.2 Nm holding torque stepping motor adjusted to plunger and

controlled with Toshiba TA 8435 stepping motor driver. It operated with software in main computer and obtained more sensitive measurement results. Optimizing the quantities of diesel fuel, three clockwise rotations were selected. Detailed information is adopted from author's previous publication below.

Pilot diesel injections occurred with twisting the plunger pin one tour (360°), two tours (720°) and three tours (1080°) of fuel pump plunger pin. Additionally middle degrees of rotations were examined with 540°, 900° and 1440° but half rotation of plunger, 180°, did not show a noticeable decrease in this mechanical fuel pump for using pilot diesel injection quantity. Because of this it was not found necessary for presentation of these data. As it is mentioned previously, minimum pilot quantities are changeable in diesel engines for different combustion chamber structures and operating conditions. Before the tests, the engine was operated for enough time with diesel fuel to reach the operation temperature. For optimization of the test results, separately three sets of experiments were performed and average test data were used for the results. The engine was operated with normal diesel fuel and 25HHOCNG gas fuel mixture as a sample with various substituted diesel fuel quantities as pilot diesel injection systematic. One of the main handicaps of gaseous fuels usage in diesel engines, created negative effects in performance parameters. Especially for the CNG fuel used in diesel engines, exhaust gas emissions decreased and affected positively, but in performance up to 15% reduction, CNG was required using in dual-fuel system with hydrogen-enrichment. HCNG improves the diesel performance nearly 10% better than CNG with diesel. In this paper, HHO is the major player for amelioration the performance of diesel test engine. HHO contains an extra oxygen atom compared to hydrogen that provides the engine more powerful by mixture of air-gas fuels and oxygen before intake manifold. As a general one set average (90 sec), diesel substitution observed with instantaneous fuel consumption for diesel, 360°, 720° and 1080° is 1.75 g/sec, 1.34 g/sec, 1.118 g/sec and 1.59 g/sec respectively. In other word substituted diesel fuel affected like pilot diesel injection reductions handled with 360°, 720° and 1080° is 21%, 30% and 18% respectively. 25HHOCNG mixture improved the torque values versus diesel engine can be reformed for 360°, 720° and 1080° is 10%, 15% and 13% respectively. Major reason

for these improvements is HHO addition into the cylinders. Maximum brake torque of 206.92 Nm is observed at 1403 rpm for 720°. Molar concentration of oxygen in air-gas mixtures occurred more flexible combustion and hydrogen atoms prevented the combustion phases faster. With this situation, CNG has playing critical role for the combustion and due to this faster combustion, CNG combust more willingly. Especially middle and high engine speeds. Similar to brake torque values, 13%, 19% and 15% increases presented for 360°, 720° and 1080° power values compared with neat diesel. Maximum brake power of 60.62 kW is observed at 2518 rpm for 720°. Once again, gas fuels combustion properties affected the power development. One of the major goals of this study, while improving performance values, decreasing the BSFC values with pilot diesel injection parameters helped of substituting diesel fuel amount. 25HHOCNG gas mixture with substituted diesel fuel, affected positively in BSFC and forms more lucrative fuel consumption. Decreasing values can be listed for 360°, 720° and 1080° compared to neat diesel is 21%, 29% and 15% respectively. Minimum BSFC is obtained with 720° by 125.67 g/kWh in 1512 rpm. As a general result, for this test engine, **720° clockwise twisted pin is the optimum point** for experimental fuel pump plunger while using 25HHOCNG fuel mixtures (Arat et. al,2016). These results clarified the next steps of the experimental investigations. After this determination, all of the experiments performed by using this important reference point.

3.3. Experimental Fuels

One of the main objectives of this present experimental study is to present a comparison between effects of pure hydrogen and HHO to performance and emissions of a diesel engine while reducing dependency to fossil fuels by using biodiesel.

3.3.1. Liquid Fuels

Three types of liquid fuels which are euro diesel, canola and palm biodiesels have used in this thesis. Euro diesel fuel has supplied from official petroleum stations. On the other hand, required amount of canola (COME) and palm (POME) biodiesels are produced from commercial canola and palm oil by transesterification method in fuel laboratory of automotive engineering. Methanol (CH_3OH) was used as alcohol (20% by volume) and sodium hydroxide (NaOH) was used as catalyst (3,5% by mass).

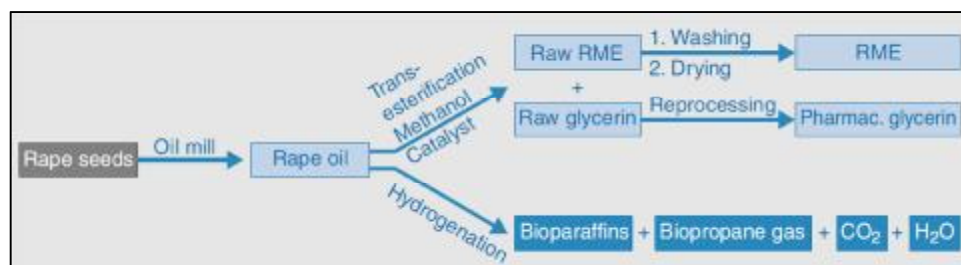


Figure 3.3. Manufacturing path for RME (rape oil methyl ester; biodiesel from rape oil) and by-products

The catalyst was dissolved in the alcohol. Methanol – sodium hydroxide mixture was poured into the canola and palm oils which were heated up to 60 °C and mixed meanwhile. During the esterification (around 1 hour), the temperature and the mixing speed of the canola and palm oils, alcohol and catalyst mixtures were kept constant (60 °C). After the transesterification the mixtures were taken to tanks to settle (10-12 hours), and then the glycerin was drained. Biodiesels were washed with pure warm water to remove alcohol and catalyst residue. Before drain the water, it was waited enough time to obtain two different phases between biodiesels and water. Canola and palm biodiesels were dried by heating up to 105 °C for an hour to eliminate the water which remains from washing. Biodiesel fuel blends were prepared by mixing commercial diesel fuel. Liquid test fuels were prepared by mixing COME and POME with diesel fuel at volumetric ratios of 25% (B25), 50% (B50), 75% (B75) and 100% (B100). Fuels were mixed in Yellowline OS 10 Basic circular shaker for half an hour (Figure 3.4).



Figure 3.4. Biodiesel-diesel blends

The fuel quality measurements of the blends were done according to EN 14214 and EN 590. The testing equipment and their measuring methods are briefly explained below. Also fuel specifications of diesel and biodiesel blends such as kinematic viscosity, cetane number, pour point, heating value, flash point and density were determined and presented in table 3.2.

Table 3.2. Test fuel specifications

	kinematic viscosity (mm ² /s)	density (g/cm ³)	cetane number	heating value (cal/gr)	flash point (°C)	pour point
Diesel	3.06	0.855	56	10589	76	-16
Canola Oil	5.31	0.914	47.80	9547	>140	-
C25	3.223	0.849	54.099	10306	98.5	-13
C50	3.602	0.862	52.918	10002	103.1	-9
C75	4.745	0.881	49.839	9502	109.9	-5
C100	4.291	0.883	46.042	9376	150	-12
Palm Oil	5.40	0.916	57.757	-	>140	+7
P25	2.939	0.861	49.190	10437	81.5	-6
P50	3.568	0.867	60.666	10366	88.5	-1
P75	4.153	0.871	52.335	10301	105	+7
P100	4.408	0.876	47.670	10220	>140	+10

Density measurements were done by Kyoto Electronics DA-130 type densimeter (Figure 3.5). This densimeter uses the resonant frequency method to measure the densities. The measurement interval of the device is 0 to 2 g/cm³ and 0 to 40 °C. Sensitivity range of device is ± 0.001 g/cm³, and a stability of 0.0001 g/cm³. TS 6311, ASTM D 4052-96 are measuring standards of the device.



Figure 3.5. Kyoto Electronics DA-130 type densimeter and Zeltex ZX440 type device

Cetane numbers and indexes were measured by Zeltex ZX440 type device (Figure 3.5.) that works with close infrared spectrometer (NIR) principal. By this principal, cetane number measurements became very fast and cheap with only 2-3% error compared to the time consuming expensive motor tests.



Figure 3.6. Tanaka AKV-202 kinematic viscometer and Tanaka MPC-102 pour point

The viscosities of the fuels were measured with Tanaka AKV-202 (Figure 3.6) brand kinematic viscometer device. The measurement range of the device is 1 to 10,000mm²/s. ISO 3104, ASTM D445, IP71, etc. are measuring test methods of the device. Tanaka MPC-102 (Figure 3.6) was used to determine pour point of liquid fuels. The device can measure according to the ASTM D6749/D97, ISO 3016 (PP), ASTM D2500, ISO 3015 (CP). Measuring range is +51°C to – 40 °C with tap water of 20 °C and +51 °C to – 65 °C with cooling liquid of -35 °C. Amount of applied air pressure for pour point detection, to accommodate different sample types: L (low) for diesel fuels, H (high) for lube oils, VH (very high) and UH (ultra high) for residual fuels and similar samples.



Figure 3.7. Tanaka Automated Pensky-Martens Closed Cup Flash Point Tester and IKA Werke C2000 bomb calorimeter

To measure the flash point of liquid test fuels, Tanaka Automated Pensky-Martens Closed Cup Flash Point Tester (Figure 3.7), model APM-7 was used. Flashpoints are measured with ASTM D93 standards and measuring range is between ambient temperature (20 °C) to 370 °C. IKA Werke C2000 bomb calorimeter (Figure 3.7) was used to determine the heating value of euro diesel, canola and palm biodiesel blend fuels. Related standards are standards ISO 2719, ASTM D93, IP 34, etc.

3.3.2. Gas Fuels

Apart from liquid fuels, gas fuels such as HHO (Hydroxy gas, Oxy-Hydrogen, Brown's gas), Hydrogen, and CNG (Compressed Natural Gas) were used as experimental fuels to achieve the goals of this thesis. CNG fuel supplied from a bundle tank which has capacity of 12.5 m^3 and fixed on 200 bar pressure. CNG used in this thesis contains 97.372% Methane, 2.327% Nitrogen, 0.230% Ethane and 0.068% Propane. Hydrogen (purity 99.999 %) tank which has gas capacity of 12.5 m^3 and fixed on 170 bar. HHO supplied from HHO generator. CNG, hydrogen tank and HHO generator are shown in figure 3.8.



Figure 3.8. CNG tanks, HHO generator and hydrogen tank

A back fire eliminator for hydrogen and CNG check valve used behind the checkout for hinders the reverse flow through the tanks. Regulators were used for reducing the pressures of hydrogen and CNG. Hydrogen regulator that adapted after back fire eliminator regulated the high pressure to atmospheric pressure. CNG pressure was reduced with the help of 2- stage regulator that brand of LOVATO. This kit reduced the pressure from 200 bar to 12 than 12 to 1, step by step. With the short term HHO generator provide the HHO system that is electrolysis of distilled water with electrolyte to any substance containing free ions that make it electrically conductive. For this study, KOH (potassium hydroxide) is chosen as an ion conductivity support. System contains double parallel connected dry cells with 14 plate total, water reservoir, bubbler, solenoid relay switch, constant current Pulse

Width Modulator (PWM) with liquid crystal display, fittings and electrical wires. HHO generator runs with engine battery (24V), 8-12 amps and 500 hertz frequency.

At the experiments, HHO passed through the bubbler system before the needle valve and flow meter. In case of Hydrogen & CNG usage, fuels supplied from regulators and set to required level by needle valves. After controlled and measured the flow of gaseous fuels with needle valves and volumetric flow meters, measurement data were collected through the data logger.

Alicat mark volumetric flow meters were used to set and measure gas flows before gas fuels get in to the mixing chamber (figure 3.9) and then in-cylinder. Maxtor needle vanes were used to control volumetric gas flows in the pipe lines. Flowmeters were conducted to data logger and acquired data were then transferred to host computer from the data logger. Various rates of hydrogen (purity: 99,999%) and HHO mixed with intake air and CNG by using mixing chamber which was produced and placed before the intake manifold entrance as shown in figure 3.9. “A, B, C and D”, point the hydrogen entrance from the tank (A), CNG entrance (B), HHO entrance (C) and “D” represents the intake manifold of the test engine respectively.

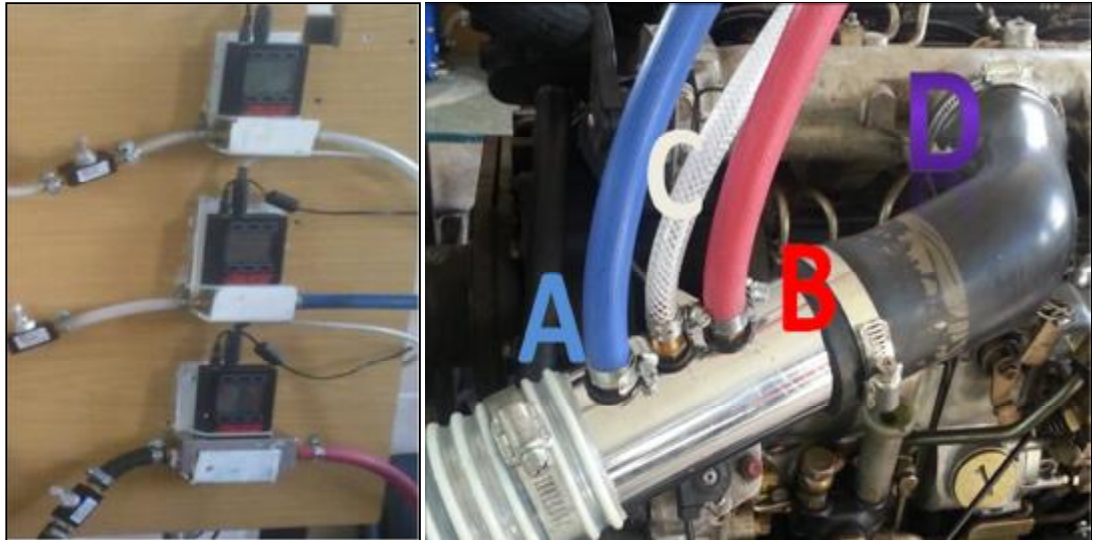


Figure 3.9. Alicat flow meters and Mixing chamber.

Specifications and accuracy of Alicat mark volumetric flow meter is represented in Table 3.3. below.

Table 3.3. Specifications of Alicat Flow Meters

PERFORMANCE	ALICAT M SERIES FLOW METER
Accuracy at calibration conditions after tare	$\pm (0.8\% \text{ of Reading} + 0.2\% \text{ of Full Scale})$
High Accuracy at calibration conditions after tare	$\pm (0.4\% \text{ of Reading} + 0.2\% \text{ of Full Scale})$
Repeatability	$\pm 0.2\% \text{ Full Scale}$
Typical Response Time	10 ms
Mass Reference Conditions (STP)	25°C & 14.696 Pisa
Operating Temperature	-10 to +50 °Celsius

3.3.3. Determination of Liquid and Gaseous Fuel Mixtures

Since the beginning of this experimental study, more than 500 tests were performed and results have been accepted by international scientific journals. One of these was examinations of hydrogen and HHO addition effects to biodiesel fuel (Baltacioglu, 2016). In that paper, Experimental results were presented under engine performance and exhaust gas emissions subsections. Main point of that experimental study was to investigate the effect of pure hydrogen, hydroxy (HHO) gas and biodiesel addition to the diesel fuel in diesel engines. Intake air was enriched with 10 L/min of H₂ and HHO before intake manifold without modification on the test engine. Effects of B10, HHO+B10 and H₂+B10 fuels on engine performance and exhaust emissions were compared with neat diesel fuel results. Substitution of diesel fuel via fuel pump plunger was also a part of that study. Experimental results are summarized and simply presented in table 3.4 to discuss and get a decision about future of this thesis.

Table 3.4. Comparison of experimental results HHO+B10 versus H₂+B10

Output		Preference (HHO+B10/H ₂ +B10)	Effect (%)
Emissions	CO (ppm)	H ₂ +B10	5.80 less
	CO ₂ (%)	H ₂ +B10	8.72 less
	NO _x (ppm)	H ₂ +B10	9.70 less
Performance	Brake Power (kW)	HHO+B10	4.33 more
	Brake Torque (Nm)	HHO+B10	2.15 more
	BSFC (g/kWh)	HHO+B10	7.60 less

Study has achieved its key objective and has fulfilled lack of comparison between effects of pure hydrogen and HHO to performance and emissions of a diesel engine while reducing dependency to fossil fuels by using biodiesel. Another important point of this study was performing the experiments with reduction of diesel engine by re-set of fuel pump plunger.

On the basis of the experimental investigation on diesel engine, conclusions can be drawn below;

- Enrichment of intake air with pure hydrogen or hydroxy gas gives promising results compared to neat diesel fuel according to the experiments.
- All other performance and exhaust emissions have improved except NO_x exhaust emissions. Both of the additives which are biodiesel and hydrogen gases predisposition to increase NO_x emissions
- It is interpreted that since HHO gas contains more oxygen, higher combustion efficiency is obtained and yields better combustion performance outputs than pure hydrogen as an additive fuel.
- According to neat diesel fuel, reductions of CO emission were 29% and 22% via H₂+B10 and HHO+B10 usage respectively. These improvements

mostly depend on carbon-free and more oxygenated fuel usage such as biodiesel, pure (H_2) and oxy (HHO) hydrogen fuels.

- Similar to the CO results, both of the hydroxy gas and hydrogen enriched fuels have provided better CO_2 exhaust emissions than neat diesel fuel. Lesser CO_2 exhaust emissions results have been measured with H_2 +B10 fuel usage compared to the HHO+B10 and neat diesel fuels (8.72% and 22.3% respectively).
- On the other hand, HHO+B10 fuel has performed higher engine performance results compared to the H_2 +B10 and neat diesel. Specification of hydrogen such as low ignition energy and faster flame speed than neat diesel improved brake power and torque outputs. Additionally, BSFC may reduce via wide flammability range and fast burning of hydrogen which is also helpful to obtain better combustion for diesel fuel. Eventually better engine economy and exhaust emissions are obtained.
- As an overall result, HHO+B10 usage in diesel engines is preferable because of easy and safety assembles to commercial transportation vehicles. Additionally low operation and consumables cost according to H_2 +B10.

As a result of this paper, it has proofed that hydrogen addition to biodiesel fuel is more environmentally friendly solution. On the other hand, HHO addition provided higher engine performance which can help to solve one of the biggest handicap of high level biodiesel blends usage. Additionally, biodiesel is a naturally environmentally friendly and a good solution to decrease exhaust gas emissions. Unfortunately, high level of biodiesel fuel blends results with loses of performance for internal combustion engines. Under the guidance of these results, it is decided to use hydrogen addition to 25% and 50% biodiesel fuel blends. Because 25% and 50% biodiesel fuel blends need to support emission aspects than performance compared to 75% and 100% biodiesel fuel blends. So HHO becomes preferable addition to 75%

and 100% biodiesel fuel blends which tolerates loses of the engines performances and provides better results than hydrogen addition.

After this point, another important topic of this thesis is to determine the mixture amount of gaseous percentages. Various researchers examined the several percentages (both mass and volume) for mixed the CNG and hydrogen. Previous studies in literature showed that, the percentage of hydrogen blends in CNG fuel mixture varying between 5-30% (that means ~1.5-10% by energy). Hythane is a 18% blend of hydrogen in CNG by energy content, which was patented by Frank Lynch of Hydrogen Components Inc, USA. A typical 20% blend of hydrogen by volume in CNG is 3% by mass or 7% by energy.

In this thesis, the volumetric ratios of HHO and hydrogen percentages in CNG fuel mixture were defined 15, 25, 35% by (vol/vol). This approach adhered to general literature and similar perspectives of previous studies. Additionally, it has examined the 35% which is out of general literature range. One of the major reason of this is to tolerate performance loses of high level blends of biodiesel fuels by increasing amount of hydrogen in gaseous fuels.

The amount of Hydrogen, HHO and CNG gases supplied to the engine is measured by the flow meters and the amount is controlled by needle vanes. Certain amounts of gases are mixed in a mixing chamber. The volume of gas mixtures are calculated by Equations (3.1,2). All experiments were repeated three times and the mixture's percentages were collected by the average of these three tests. For all test of this thesis, the total average of hydrogen, HHO and CNG fuel mixtures percentages can be found below. Additionally this value has a similarity to literature for optimum usage (20-30% HCNG) of these gas fuels.(Nanthagopal et.al (2011)).

$$H_2, HHO[\%vol] = \frac{V_{H_2, HHO}}{V_{CNG} + V_{H_2, HHO}} \quad (3.1)$$

$$CNG [\%vol] = \frac{V_{CNG}}{V_{CNG} + V_{H_2, HHO}} \quad (3.2)$$

Total amount of gaseous fuels: (sum of HCNG or HHOCNG) = 20 LPM

15HCNG = 3.04 lpm H₂ + 17.20 CNG 15HHOCNG = 3.13 lpm HHO + 17.57 CNG

25HCNG = 5.09 lpm H₂ + 15.32 CNG 25HHOCNG = 5.11 lpm HHO + 15.59 CNG

35HCNG = 7.10 lpm H₂ + 13.11 CNG 35HHOCNG = 7.27 lpm HHO + 13.32 CNG

General consideration for fuel usage ratios was separating 50% gas (HHO or H and CNG) and 50% liquid fuels (Diesel, COME and POME). For 15HCNG and 15HHOCNG, HHO or hydrogen consist 15% of all gas fuels which means 7.5% of all amount of fuel used for each experiments. With same calculation for CNG becomes 42.5%. Same calculation is applied to 25 and 35HCNG and HHOCNG fuels. There are similar situations for liquid fuels between COME, POME and neat diesel which effects the name of the liquid fuels (C-P25, C-P50, C-P75 or C-P100) related by volumetric ratio of COME and POME mixed with neat diesel. Hence, experimental fuels are referred as below to prevent typing complexity and visualized in Figure 3.10.

For 15HCNG and 15HHOCNG:

Diesel: 100% Diesel (standard fuel pump injection)

C25+15HCNG: 7.5% H + 42.5% CNG + 50% C25 (subs. liquid fuel pump injection)

C50+15HCNG: 7.5% H + 42.5% CNG + 50% C50 (subs. liquid fuel pump injection)

C75+15HHOCNG: 7.5% HHO + 42.5% CNG + 50% C75 (subs. liquid fuel pump inj.)

C100+15HHOCNG: 7.5% HHO + 42.5% CNG + 50% C100 (subs. liquid fuel pump inj.)

P25+15HCNG: 7.5% H + 42.5% CNG + 50% P25 (subs. liquid fuel pump injection)

P50+15HCNG: 7.5% H + 42.5% CNG + 50% P50 (subs. liquid fuel pump injection)

P75+15HHOCNG: 7.5% HHO + 42.5% CNG + 50% P75 (subs. liquid fuel pump inj.)

P100+15HHOCNG: 7.5% HHO + 42.5% CNG + 50% P100 (subs. liquid fuel pump inj.)

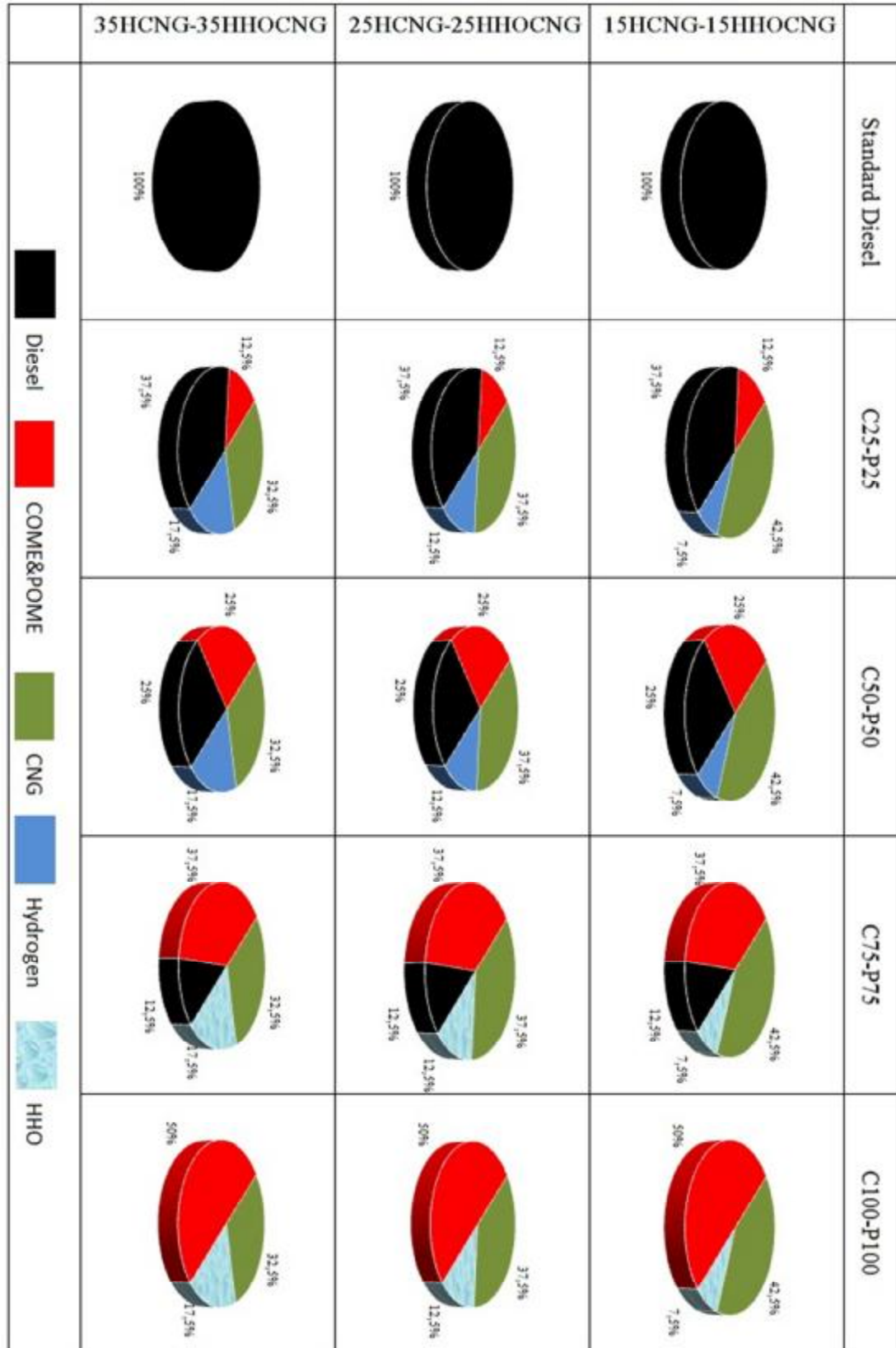


Figure 3.10. Description of test fuels and mixture percentages

For 25HCNG and 25HHOCNG:

Diesel: 100% Diesel (standard fuel pump injection)

C25+25HCNG):12.5% H+37.5% CNG + 50% C25 (subs. liquid fuel pump injection)

C50+25HCNG):12.5% H+37.5% CNG + 50% C50 (subs. liquid fuel pump injection)

C75+25HHOCNG):12.5% HHO+37.5% CNG+50% C75 (subs. Liq. fuel pump inj.)

C100+25HHOCNG):12.5%HHO+37.5%CNG+50%C100 (subs.Liq. fuel pump inj.)

P25+25HCNG): 12.5% H+37.5% CNG + 50% P25 (subs. liquid fuel pump injection)

P50+25HCNG): 12.5% H+37.5% CNG + 50% P50 (subs. liquid fuel pump injection)

P75+25HHOCNG):12.5% HHO+37.5% CNG+50% P75 (subs. liquid fuel pump inj.)

P100+25HHOCNG):12.5% HHO+37.5% CNG+50% P100 (subs.Liq. fuel pump inj.)

For 35HCNG and 35HHOCNG:

Diesel: 100% Diesel (standard fuel pump injection)

C25 (35HCNG): 17.5% H+32.5% CNG+50% C25 (subs. liquid fuel pump injection)

C50 (35HCNG): 17.5% H+32.5% CNG+50% C50 (ssubs. liquid fuel pump inj.)

C75 (35HHOCNG): 17.5% HHO+32.5% CNG+50% C75 (subs.Liq. fuel pump inj.)

C100 (35HHOCNG):17.5%HHO+32.5% CNG+50%C100 (subs.Liq. fuel pump inj.)

P25 (35HCNG): 17.5% H+32.5% CNG + 50% P25 (subs. liquid fuel pump injection)

P50 (35HCNG): 17.5% H+32.5% CNG+50% P50 (subs. liquid fuel pump injection)

P75 (35HHOCNG): 17.5% HHO+32.5% CNG+50% P75 (subs.Liq. fuel pump inj.)

P100 (35HHOCNG):17.5%HHO+32.5% CNG+50% P100 (subs.Liq. fuel pump inj.)

4. RESULTS AND DISCUSSION

Results of this experimental study are presented in this section. Effects of the 15, 25, 35% HCNG and HHOCNG combined gas fuel mixtures usage on engine performance and exhaust emissions are examined with semi pilot biodiesel injection. Comparisons of canola and palm biodiesel fuel mixtures are used as liquid fuel instead of standard diesel fuel. Liquid fuels are supported by hydrogen enriched CNG gas fuels via intake manifold.

Before starting the experiments, test engine was operated for enough time (10-15 minutes) with standard diesel fuel to reach the steady state operation conditions.

All tests were implemented with the engine speed between 1300 to 2500 rpm with an interval of 200 rpm at WOT and full load condition. Each experiment repeated three times and took their mean values of these three experiments to optimize experimental errors and uncertainty of the system devices. Additionally, the temperature of the laboratory kept between 24-26°C. All gaseous fuels induction pressures are atmospheric pressure conditions.

One of the important goals of the thesis is using alternative gas and liquid fuel mixtures in an original diesel engine without any modifications. In this experimental study, pilot diesel injection procured with adjusting the mechanical fuel pump plunger with the help of stepping motor and devices; stepping motor driver was used for adjustment of the mechanical fuel pump plunger pin.

This system ensured less fuel consumption without any structural modification. As it is mentioned in previous section of the thesis, adjustment of fuel pump plunger pin 720° clockwise provided approximately 15-30% reduction in fuel consumption. This method called as semi-pilot diesel injection in this study.

Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graphs are representing only liquid test fuels consumption, not the gaseous test fuels.

4.1. Performance Results

In this subsection contains graphs of engine performance data that presented after detailed examination in Figure 4.1. to 4.8. Performance results are separated in to section to discuss up to biodiesel usage in fuel mixtures. Brake specific fuel consumption, brake torque and power versus engine speed graphs are combined in single figures to avoid complexity and provide easier comparison. Additionally, to be more informative about combustion, volumetric and thermal efficiencies of C25 + (15-25-35) HCNG fuel mixtures are calculated with experimental data and presented at the end of this subsection as an sample in Figure 4.9.

4.1.1 Performance of canola biodiesel blends with HCNG and HHOCNG

Performance results of using C25 + 15-25-35 HCNG with semi pilot biodiesel injection are illustrated in figure 4.1. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake power are shown in figure 4.1.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of Figure 4.1. It is clearly seen from the illustrated BSFC results that all mixtures have improved according to diesel fuel. Single C25 usage decreased 8.26%, overall mean improvements of C25 + 15-25-35 HCNG are 14.69%, 13.1% and 19.99% respectively; which means best reduction for BSFC is obtained by using C25 + 35HCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.1. When compared all fuel mixtures with neat diesel, the brake power results are similar with Brake Torque data. Except semi pilot injection of C25, all mixtures gave better results than diesel fuel. Very similar results are obtained between combined fuel mixtures versus neat diesel. Overall improvements are 4.90%, 1.94% and 7.42%, for mixtures C25 + 15HCNG, C25 + 25HCNG and C25 + 35HCNG, respectively. Unlike combined fuel mixtures; single fuel usage with semi pilot biodiesel injection method decreased brake power 8.45%. The Brake Power data shows that C25 + 35HCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of C25 + (15-25-35) HCNG is placed at the bottom of Figure 4.1. All fuel mixtures performed better performance results than neat diesel except semi pilot injection of C25. It is clear that C25 + 35HCNG has the best overall mean value about producing 5.79% more brake torque when compared with diesel fuel. Brake torque improvements of C25 + 15HCNG and C25 + 25HCNG are 4.30% and 1.72%, respectively.

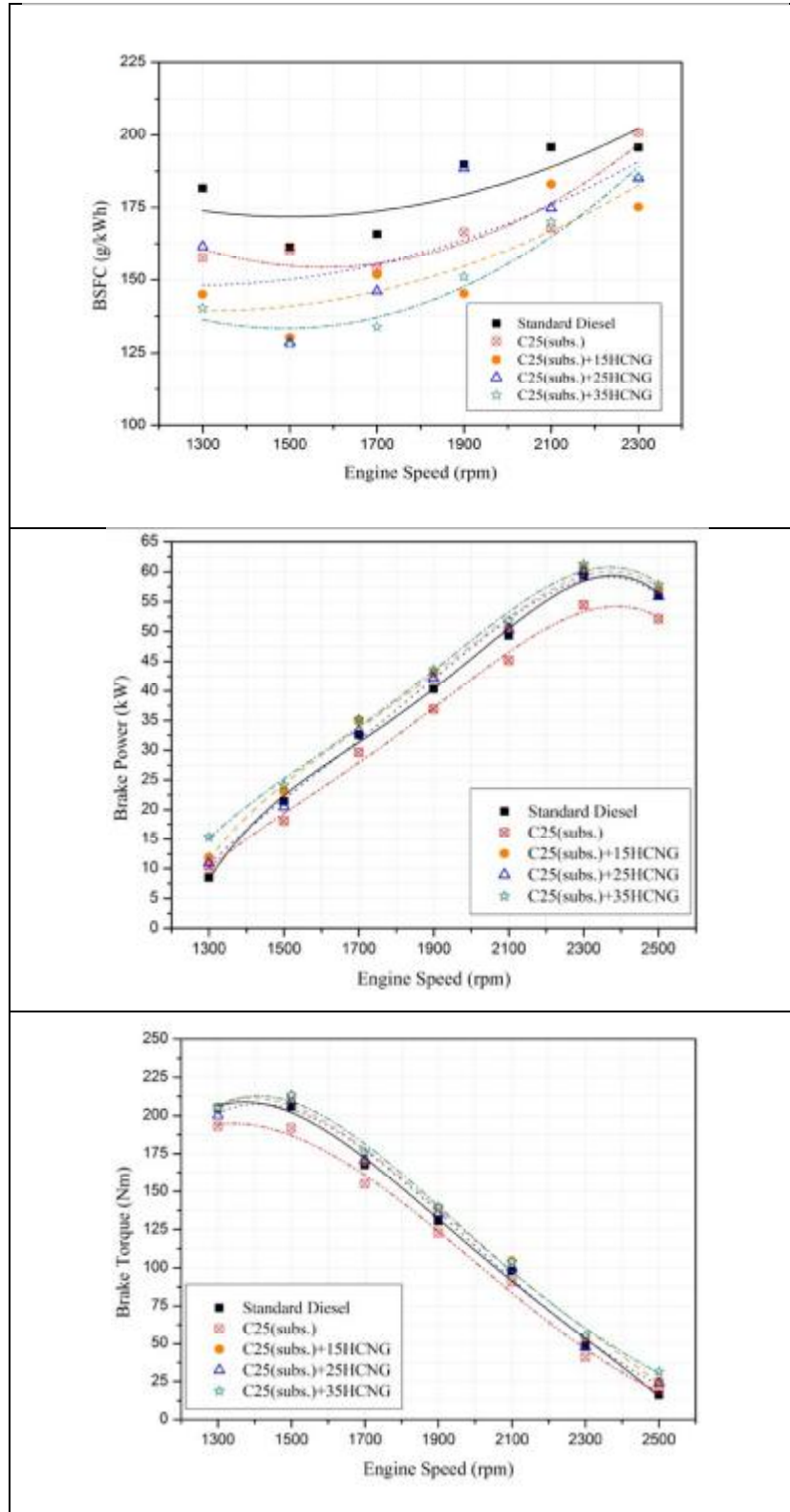


Figure 4.1. BT, BP and BSFC Compressions of C25 with 15-25-35 HCNG mixtures

Performance results of using C50 + 15-25-35 HCNG with semi pilot biodiesel injection are illustrated in figure 4.2. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake are evidently shown in figure 4.2.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of Figure 4.2. It is clearly seen from the illustrated BSFC results that all mixtures have improved results according to diesel fuel. Single C50 usage decreased 4.90%, overall mean improvements of C50 + 15-25-35 HCNG are 20.68%, 17.41%, and 17.76% respectively; which means best reduction for BSFC is obtained by using C50 + 15HCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.2. When compared all fuel mixtures with neat diesel, the brake power results as the similar with Brake Torque data. Except semi pilot injection of C50, all mixtures gave better results than diesel fuel. Very similar results are obtained between combined fuel mixtures C50 + 25HCNG and C50 + 35HCNG versus neat diesel. Overall improvements are 10.79%, 0.97%, and 3.63%, for mixtures C50 + 15HCNG, C50 + 25HCNG and C50 + 35HCNG, respectively. Unlike combined fuel mixtures; single fuel usage with semi pilot biodiesel injection method decreased brake power 4.21%. The Brake Power data shows that C50 + 15HCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of C50 + 15-25-35 HCNG is placed at the bottom of Figure 4.2. All fuel mixtures performed better performance results than neat diesel except semi pilot injection of C50. It is clear that C50 + 15HCNG has the best overall mean value about producing 7.75% more brake torque when compared with diesel fuel. Brake torque improvements of C50 + 25HCNG and C50 + 35HCNG are 1.25% and 2.91%, respectively.

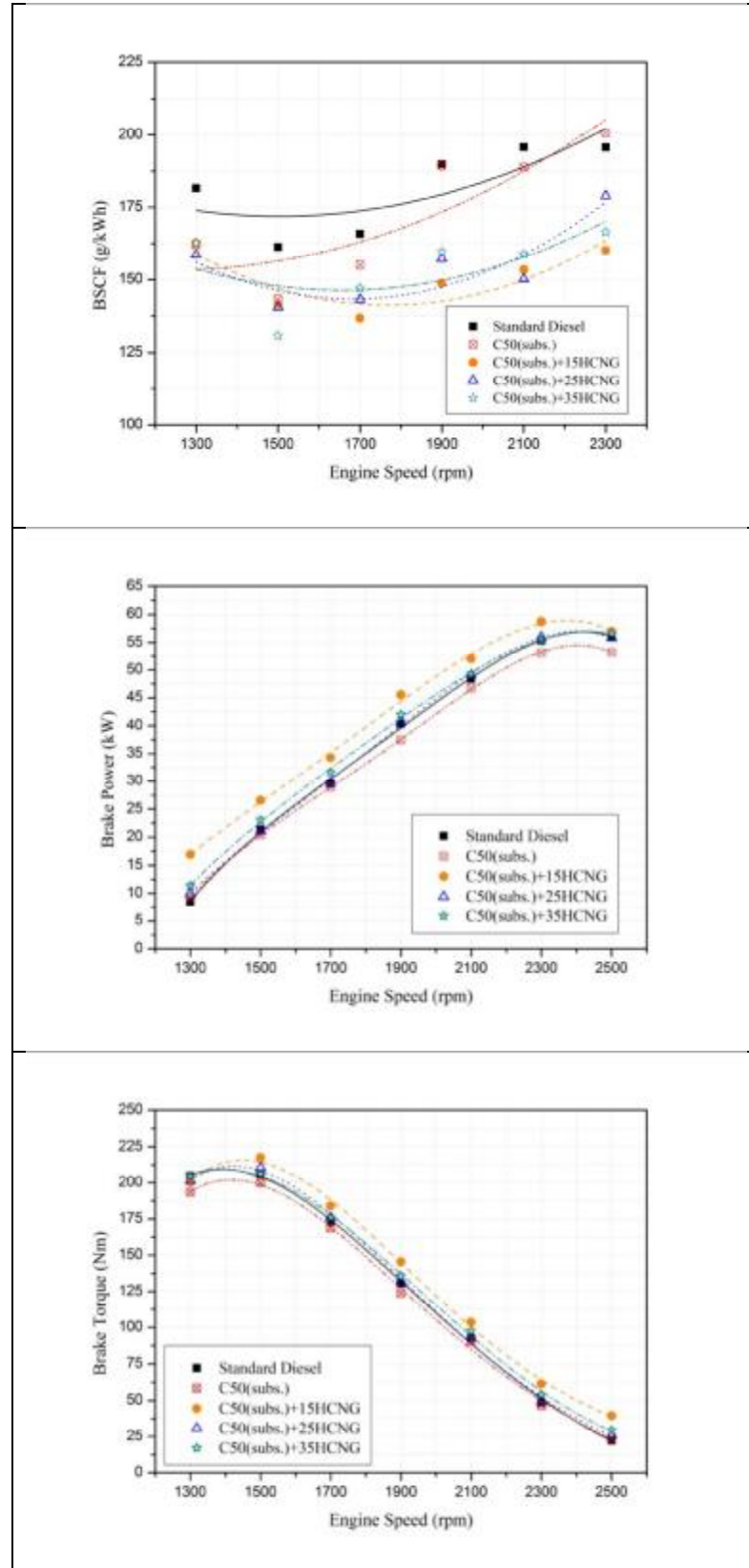


Figure 4.2.BT, BP and BSFC Compressions of C50with 15-25-35 HCNG mixtures

Performance results of using C75 + 15-25-35 HHOCNG with semi pilot biodiesel injection are illustrated in figure 4.3. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake are evidently shown in figure 4.3.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of Figure 4.3. It is clearly seen from the illustrated BSFC results that all mixtures have improved results according to diesel fuel, except semi pilot injection of C75 fuel. Overall mean improvements are 7.73%, 3.11%, 2.46%, respectively; which means best reduction for BSFC is obtained by using C75 + 15HHOCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.3. When compared all fuel mixtures with neat diesel, the brake power results as the similar with Brake Torque data. Overall improvements are 2.25%, 1.59% for mixtures C75 + 15HHOCNG, and C75 + 25HHOCNG, respectively. Unlike C75 + 15HHOCNG, and C75 + 25HHOCNG mixtures; semi pilot injection of C75 and C75 + 35HHOCNG mixture decreased brake power 7.32%, 4.35%, respectively. The Brake Power data shows that C75 + 15HHOCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of C75 + 15-25-35 HHOCNG is placed at the bottom of Figure 4.3. It is clear that C75 + 15HHOCNG has the best overall mean value about producing 1.57% more brake torque when compared with diesel fuel. Overall improvements are 1.57%, 1.47% for mixtures C75 + 15HHOCNG, and C75 + 25HHOCNG, respectively. Unlike C75 + 15HHOCNG, and C75 + 25HHOCNG mixtures; semi pilot injection of C75 and C75 + 35HHOCNG mixture decreased brake power 5.48%, 3.73%, respectively.

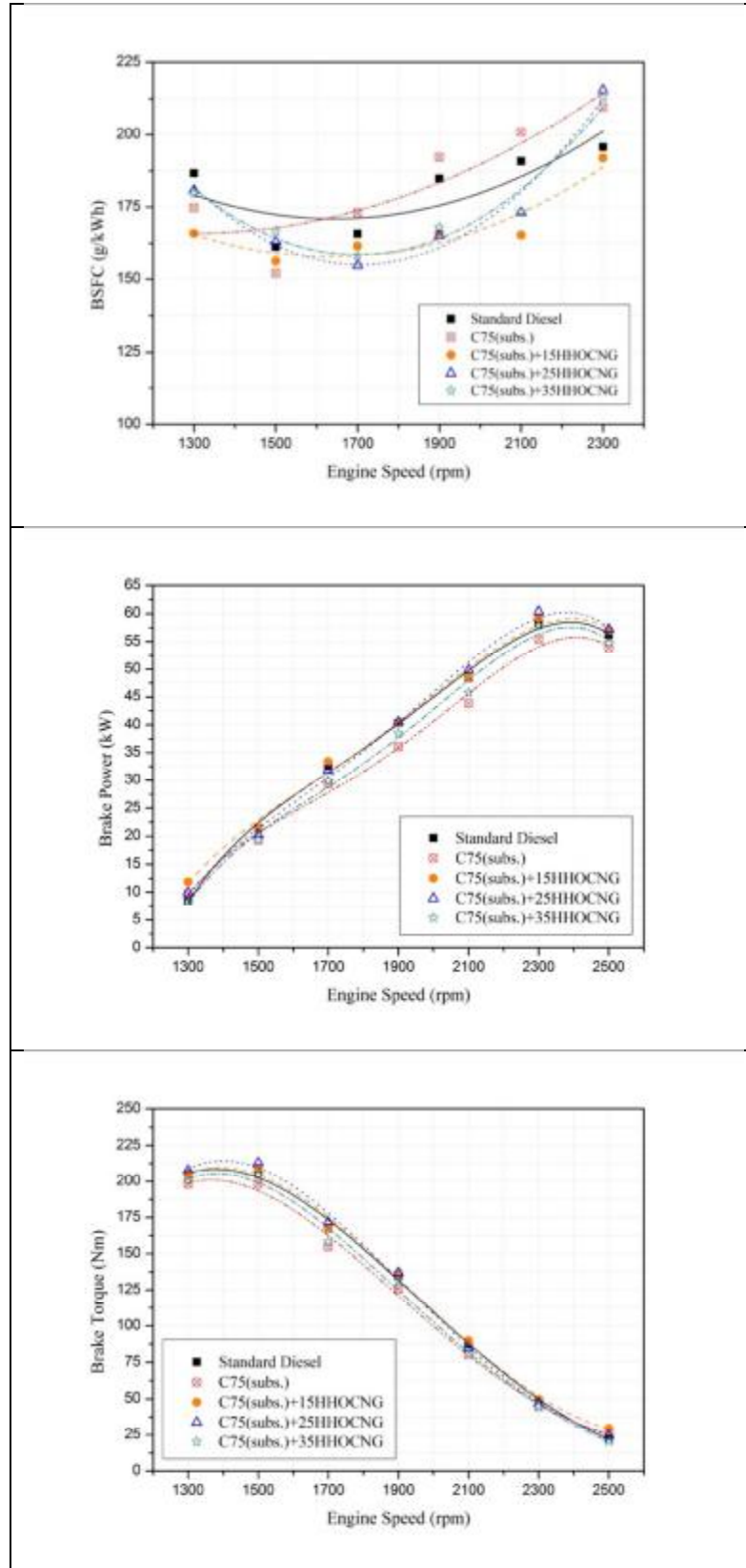


Figure 4.3. BT, BP and BSFC Compressions of C75 with 15-25-35 HCNG mixtures

Performance results of using C100 + 15-25-35 HHOCNG with semi pilot biodiesel injection are illustrated in figure 4.4. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HHOCNG fuel combinations on brake torque and brake are evidently shown in figure 4.4.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of Figure 4.4. It is clearly seen from the illustrated BSFC results that all mixtures have improved results according to diesel fuel. Single C50 usage decreased 1.37%, overall mean improvements of C100 + 15-25-35HHOCNG are 2.22%, 1.98% and 2.34%, respectively; which means best reduction for BSFC is obtained by using C100 + 35HHOCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.4. Except semi pilot injection of C100 + 35HHOCNG, all mixtures have slightly decreased brake power of the engine. Overall reductions are 2.97%, 0.62% 5.08% for C100, C100 + 15HHOCNG and C100 + 25HHOCNG, respectively. Increment of brake power is also slightly and at a level of 0.12% for C100 + 35HCNG fuel. The Brake Power data shows that C100 + 35HCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of C100 + (15-25-35) HCNG is placed at the bottom of Figure 4.4. Results are very similar to brake power. C100 and all fuel mixtures reduced brake power except C100 + 35HCNG. Overall reductions are 2.30%, 0.03% 3.18% for C100, C100 + 15HHOCNG and C100 + 25HHOCNG, respectively. Increment of brake power is also slightly and at a level of 0.42% for C100 + 35HCNG fuel. The Brake Power data shows that C100 + 35HCNG gives better improvement compared to other fuel mixtures and neat diesel.

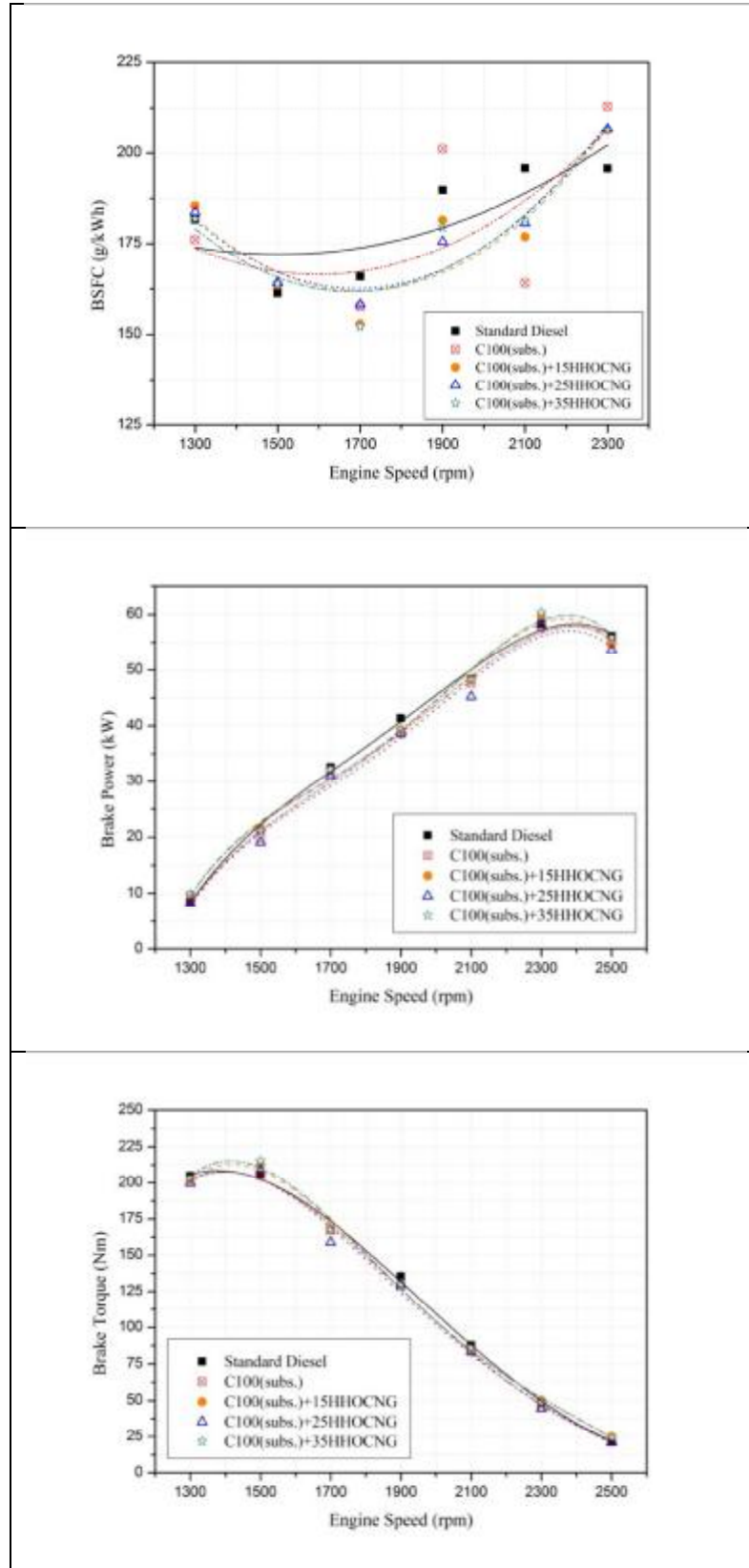


Figure 4.4. BT, BP and BSFC Compressions of C100 with 15-25-35 HCNG mixtures

4.1.2 Performance of palm biodiesel blends with HCNG and HHOCNG

Performance results of using P25 + 15-25-35 HCNG with semi pilot biodiesel injection are illustrated in figure 4.5. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake are evidently shown in figure 4.5.

BSFC - Engine Speed graph is shown at the top of Figure 4.5. It is clearly seen from the illustrated BSFC results that all mixtures have improved results according to diesel fuel. Overall mean improvements are 8.31%, 18.03%, 22.07% and 15,18%, respectively; which means best reduction for BSFC is obtained by using P25 + 25HCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.5. When compared all fuel mixtures with neat diesel, the brake power results as the similar with Brake Torque data. Except semi pilot injection of P25, all mixtures gave better results than diesel fuel. Very similar results are obtained between mixture P25 + 15HCNG, P25 + 25HCNG and P25 + 35HCNG versus neat diesel. Overall improvements are 12.27%, 13.01% and 10.5%, for mixtures P25 + 15HCNG, P25 + 25HCNG and P25 + 35HCNG, respectively. Unlike mixtures P25 + 15HCNG, P25 + 25HCNG and P25 + 35HCNG; semi pilot injection of P25 decreased brake power 0.24%. The Brake Power data shows that P25 + 15HCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of P25 + 15-25-35 HCNG is placed at the bottom of Figure 4.5. All fuel mixtures performed better performance results than neat diesel except semi pilot injection of P25. It is clear that P25 + 25HCNG has the best overall mean value about producing 10.1% more brake torque when compared with diesel fuel. Brake torque improvements of P25 + 15HCNG and P25 + 35HCNG are 8.99% and 7.92%, respectively.

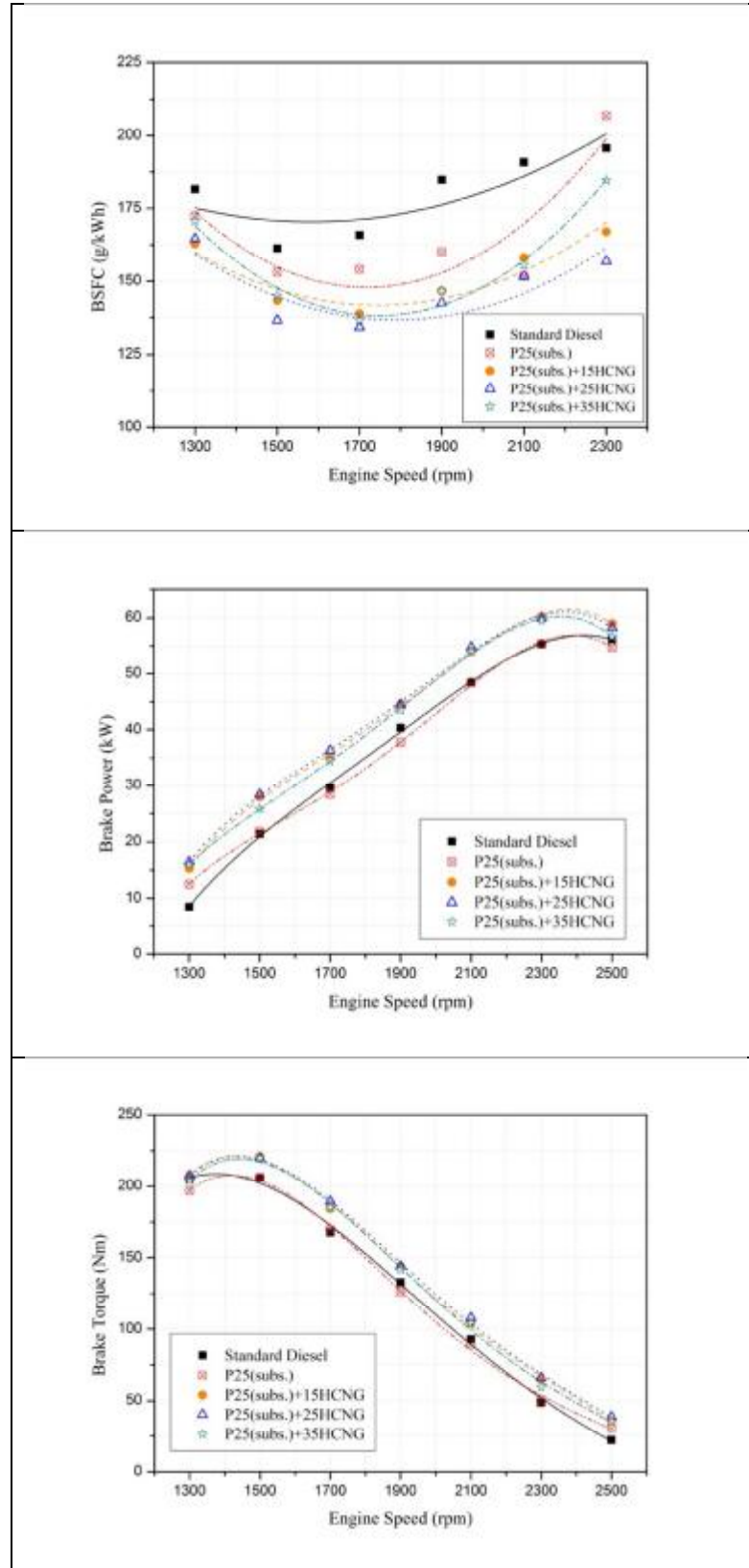


Figure 4.5. BT, BP and BSFC Compressions of P25 with 15-25-35 HCNG mixtures

Performance results of using P50 + 15-25-35 HCNG with semi pilot biodiesel injection are illustrated in figure 4.6. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake are evidently shown in figure 4.6.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of Figure 4.6. It is clearly seen from the illustrated BSFC results that all mixtures have improved results according to diesel fuel. Single P50 usage decreased 6.25%, overall mean improvements of P50 + 15-25-35HCNG are 21.32%, 19 and 16.53%, respectively; which means best reduction for BSFC is obtained by using P50 + 15HHOCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of figure 4.6. When compared all fuel mixtures with neat diesel, the brake power results as the similar with Brake Torque data. Except semi pilot injection of P50, all mixtures gave better results than diesel fuel. Overall improvements are 12.32%, 9.41% and 5.79% for mixtures P50 + 15HCNG, P50 + 25HCNG and P50 + 35HCNG, respectively. Unlike fuel mixtures, single P50 usage decreased brake power 0.91%. The Brake Power data shows that P50 + 15HCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of P50 + 15-25-35 HCNG is placed at the bottom of figure 4.6. All fuel mixtures performed better performance results than neat diesel except semi pilot injection of P50. It is clear that P50 + 15HCNG has the best overall mean value about producing 9.73% more brake torque when compared with diesel fuel. Brake torque improvements of P50 + 25HCNG and P50 + 35HCNG are 7.12% and 4.6%, respectively. Similar to brake power, P50 usage decreased brake torque 1.37%.

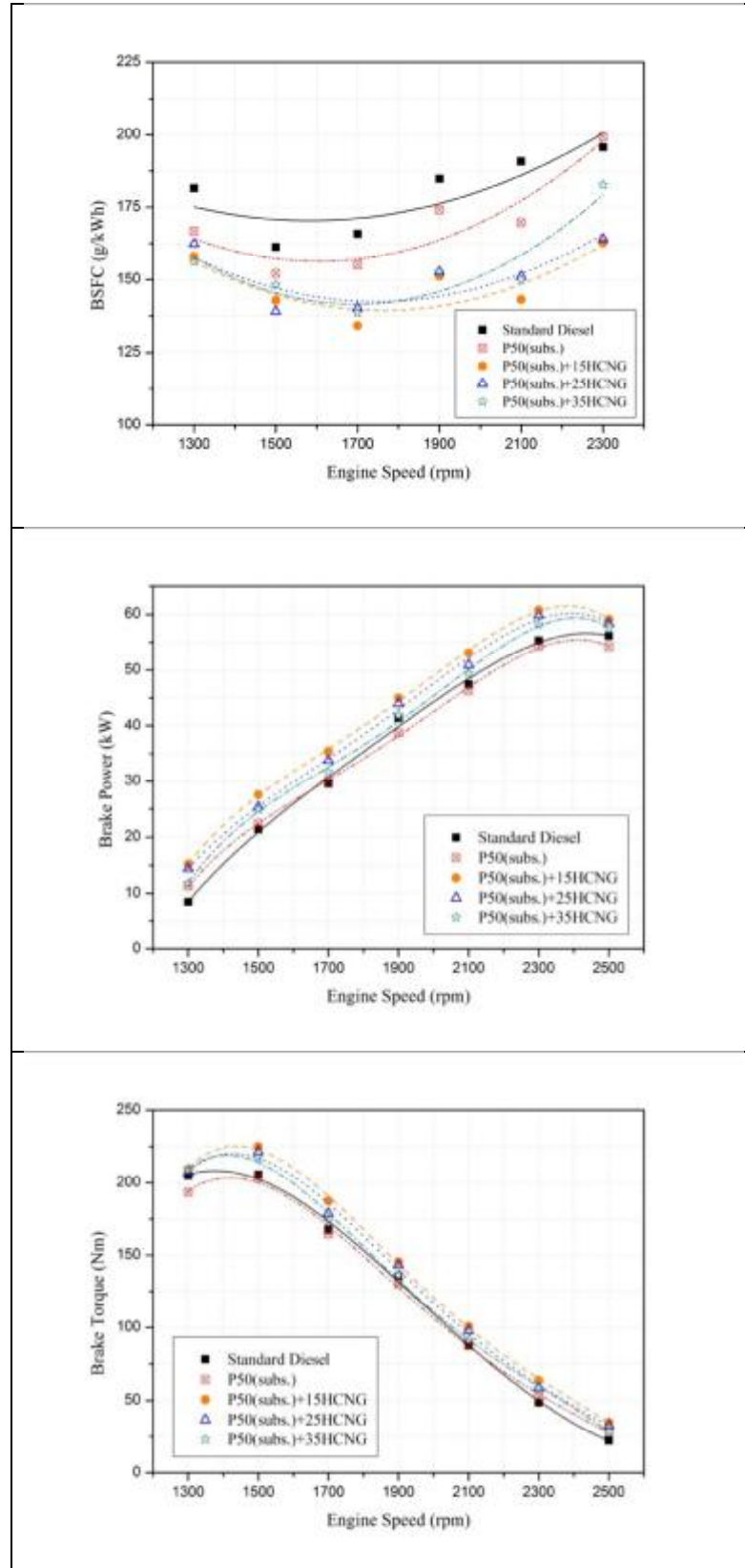


Figure 4.6. BT, BP and BSFC Compressions of P50 with 15-25-35 HCNG mixtures

Performance results of using P75 + 15-25-35 HHOCNG with semi pilot biodiesel injection are illustrated in figure 4.7. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake are evidently shown in figure 4.7.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of figure 4.7. It is clearly seen from the illustrated BSFC results that all mixtures have improved results according to diesel fuel. Single P75 usage decreased 4.45%, overall mean improvements of P75 + 15-25-35HHOCNG are 9.95% 7.94% and 11.6%, respectively; which means best reduction for BSFC is obtained by using P75 + 35HHOCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.7. When compared all fuel mixtures with neat diesel, the brake power results as the similar with Brake Torque data. Except semi pilot injection of P75, all mixtures gave better results than diesel fuel. Very similar brake power results are obtained versus neat diesel. Overall improvements are 1.47%, 1.11% and 3.63%, for mixtures P75 + 15HHOCNG, P75 + 25HHOCNG and P75 + 35HHOCNG, respectively. Unlike fuel mixtures, P75 decreased brake power 4.95%. The Brake Power data shows that P75 + 35HHOCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of P75 + 15-25-35 HHOCNG is placed at the bottom of figure 4.7. All fuel mixtures performed better performance results than neat diesel except semi pilot injection of P75. Semi pilot injection of P75 is decreased brake torque 4.03%. On the other hand, P75 + 35HHOCNG has the best overall mean value about producing 2.55% more brake torque when compared with diesel fuel. Brake torque improvements of P75 + 15HCNG and P75 + 25HCNG are 1.56% and 0.6%, respectively.

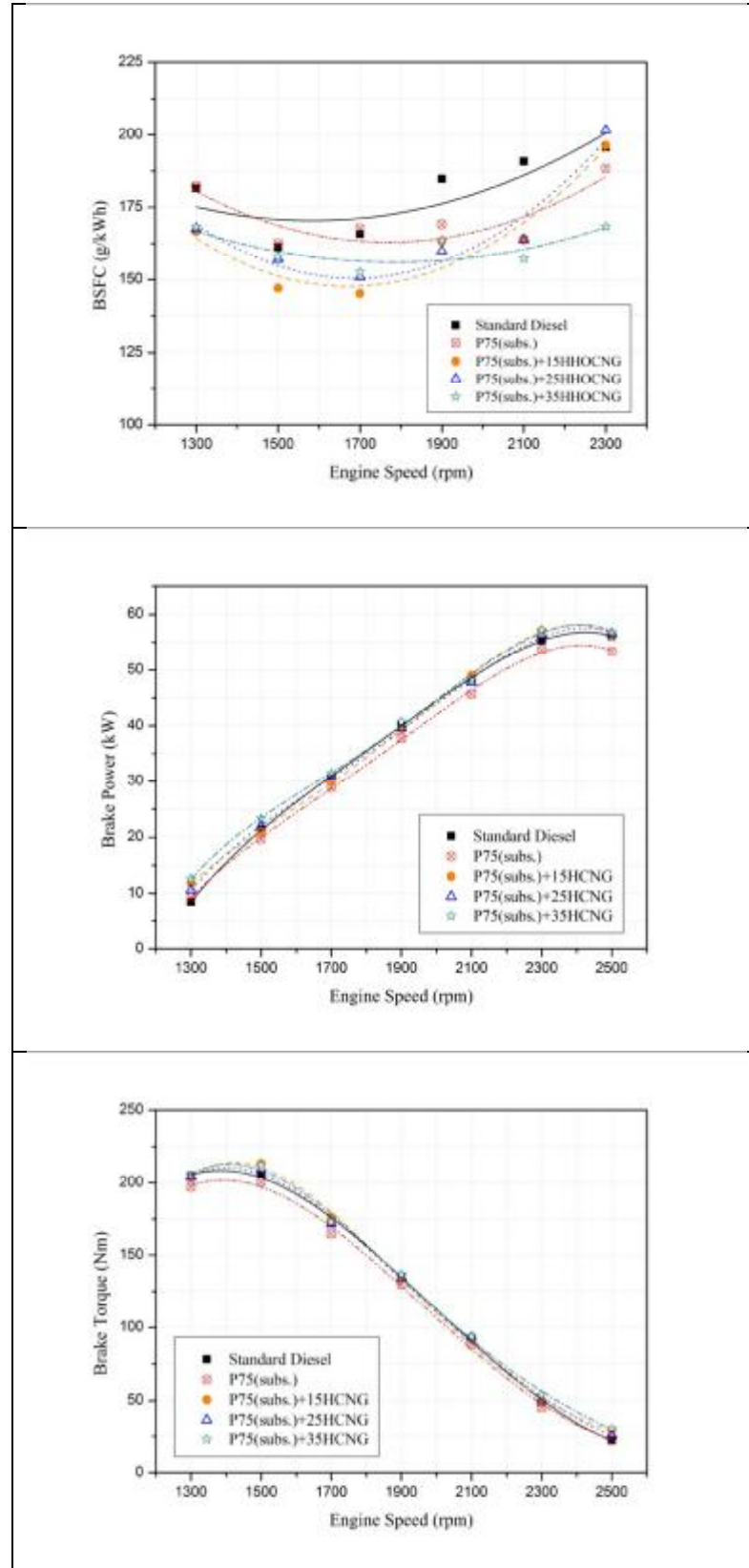


Figure 4.7. BT, BP and BSFC Compressions of P75 with 15-25-35 HCNG mixtures

Performance results of using P100 + 15-25-35 HHOCNG with semi pilot biodiesel injection are illustrated in figure 4.8. Using gas fuels in CI engines and their mixture ratios are very important in combustion phenomenon. This system ensured less fuel consumption without any structural modification. Besides, positive effects of using various HCNG fuel combinations on brake torque and brake are evidently shown in figure 4.8.

Brake specific fuel consumption (BSFC) - Engine Speed graph is shown at the top of figure 4.8. As seen from the figure, P100 + 15HHOCNG, P100 + 25HHOCNG reduced BSFC at level of 6.32% and 11.53%, respectively. Overall mean value change was negligible for P100 + 35HHOCNG fuel usage. Best reduction among P100 fuel enrichments is obtained by using P100 + 35HCNG. Substitution of diesel fuel by adjusting the mechanical fuel pump plunger should be considered about reduction of BSFC. Additionally, BSFC graph is representing only liquid test fuels consumption, not the gaseous test fuels.

The Brake Power - Engine Speed graph is shown at the middle of Figure 4.8. When compared all fuel mixtures with neat diesel, the brake power results as the similar with Brake Torque data. Except semi pilot injection of P100, all mixtures gave better results than diesel fuel. Overall improvements are 1.48%, 3.79% and 1.33% for mixtures P100 + 15HHOCNG, P100 + 25HHOCNG and P100 + 35HHOCNG, respectively. Unlike fuel mixtures, semi pilot injection of P100 decreased brake power 8.47%. The Brake Power data shows that P100 + 25HHOCNG gives better improvement compared to other fuel mixtures and neat diesel.

Brake Torque versus Engine Speed graph of P100 + 15-25-35 HHOCNG is placed at the bottom of Figure 4.8. All fuel mixtures performed better performance results than neat diesel except semi pilot injection of P100. Losses of brake torque was 6.53% with P100 single fuel usage. P100 + 25HHOCNG have the best overall mean value about producing 3.2% more brake torque when compared with diesel fuel. Brake torque improvements of P100 + 15HHOCNG and P100 + 35HHOCNG are 1.13% and 0.82%, respectively.

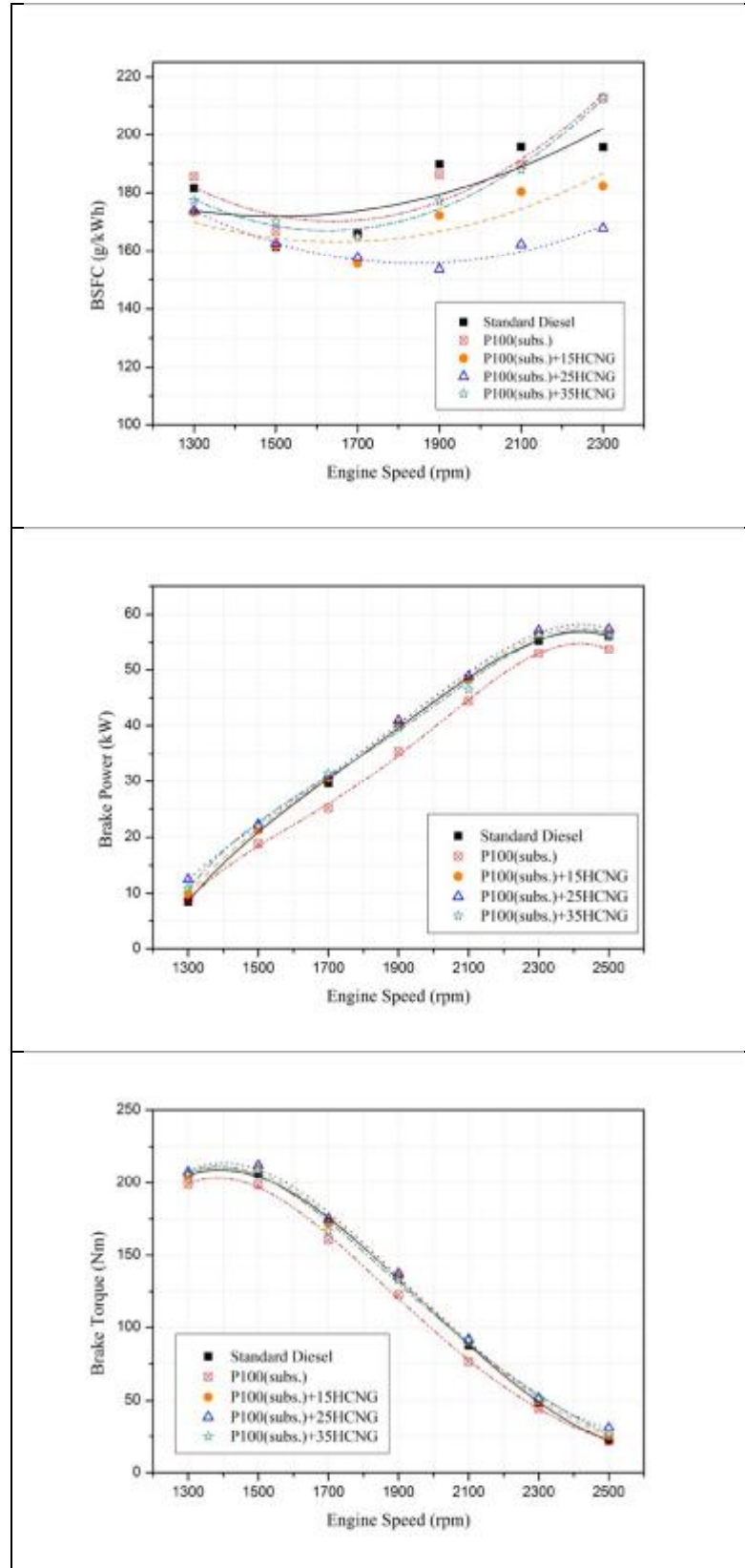


Figure 4.8. BT, BP and BSFC Compressions of P100 with 15-25-35 HCNG mixtures

The variation of Brake Thermal Efficiency (η_{bth}) & Volumetric Efficiency (η_v) is illustrated as a function of engine speed in figure 4.9. It is known that the brake thermal efficiency is the ratio of brake power to the energy released during the combustion process.

$$\eta_{bth} = \frac{\dot{W}_b}{\dot{m}_{d(B25,B50)} * LHV_{d(B25,B50)} + \dot{m}_{CNG} * LHV_{CNG} + \dot{m}_{H_2} * LHV_{H_2}} \quad (4.1)$$

$$\eta_{bth} = \frac{\dot{W}_b}{\dot{m}_{d(B75,B100)} * LHV_{d(B75,B100)} + \dot{m}_{CNG} * LHV_{CNG} + \dot{m}_{HHO} * LHV_{HHO}} \quad (4.2)$$

$$\eta_v = \frac{Z * \dot{m}_a}{\rho_a * V_d * N} \quad (4.3)$$

Equation 4.1, 4.2. are defined the brake thermal efficiency (η_{bth}). In eq. 4.1; hydrogen based test fuel mixtures (1 and 2) and in eq. 4.2; HHO based fuel mixtures (3 and 4) are defined. Before start to discuss η_{bth} of the test mixtures, it can be noticed that, except neat diesel operation, all test mixtures run under with semi-pilot diesel injection. The most important variants on calculation of η_{bth} are mass of fuel and lower heating values (LHV) of fuel. In this case, semi-pilot diesel injection (as substituted diesel amount) and LHV of biodiesels percentages, CNG, H_2 and HHO is playing an important role in this calculation. LHV of CNG, H_2 and COME fuel can be seen in tables 1 and 3. Hence the LHV of HHO needs clarification when used as a fuel and/or additive. For this issue; Al-Rababah (2014) and Chiriac (2006) mentioned that, LHV of HHO transformed to H-H ($2H_2$ and O_2) and use for the thermal efficiency with $0.66 * (\dot{m}_{H_2} * LHV_{dH_2})$ which contents of hydrogen ratio in HHO. In this study LHV of HHO for calculation of η_{bth} used with Al-Rababah (2014) and Chiriac (2006)'s approaches. Figure 4.9 showed that, when compared with neat diesel operation; all other mixture fuels decreased the η_{bth} slightly. Numerically; average decrement results can be listed from as 3.8%, 7.92%, 7.91% and 6.23% for C25 + 25HCNG, C50 + 25HCNG, C75 + 25HHOCNG, C100 +

25HHOCNG respectively. By the way, it should be noticed that, some critical points affected η_{bth} of this study. It can be listed like;

- mixture fuels operated with semi-pilot diesel injection;
- the additives of CNG and hydrogenated gases affected the η_{bth} (with higher LHV conditions);
- biodiesel percentage affected the results of η_{bth} .

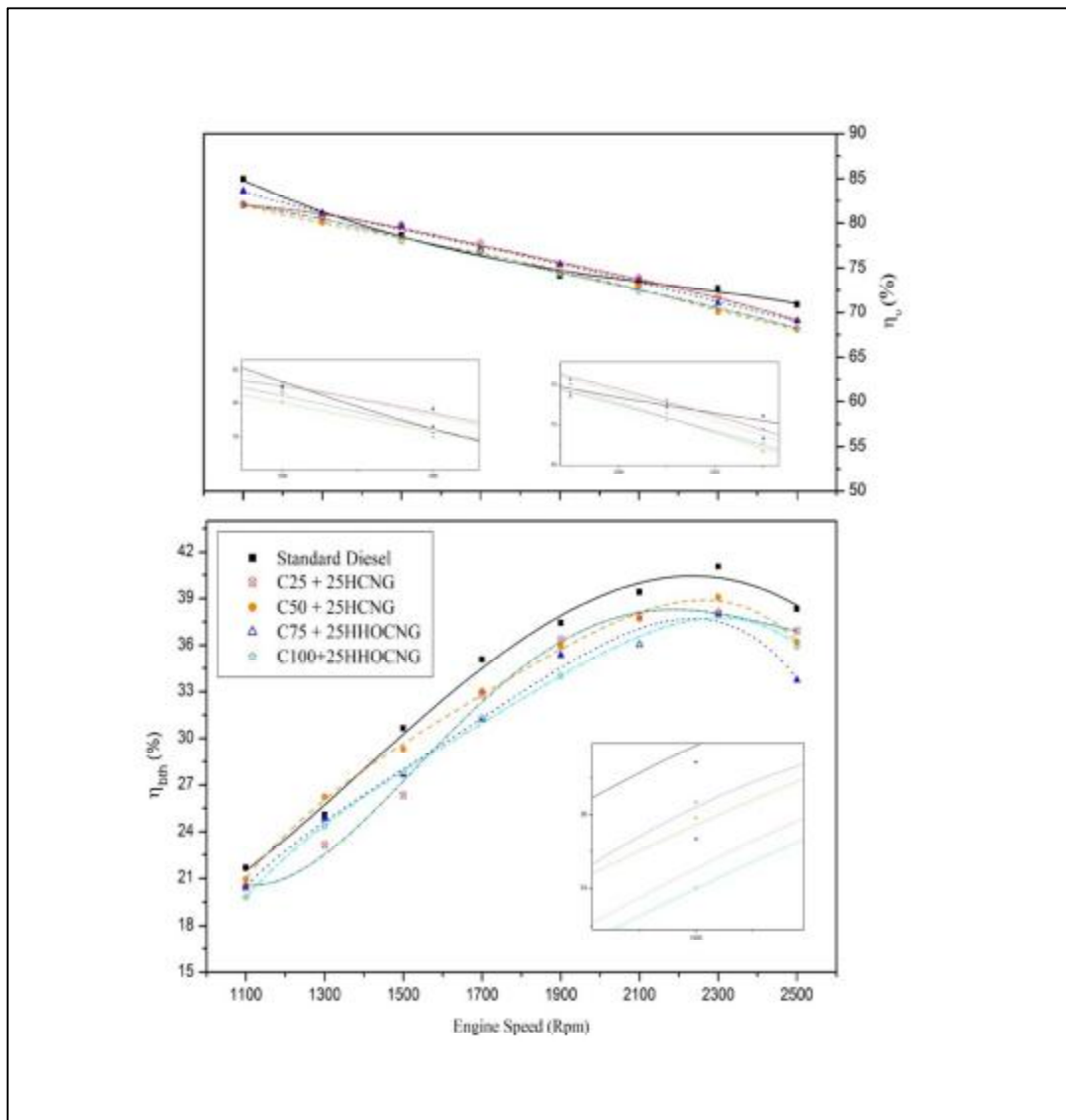


Figure 4.9. Brake Thermal Efficiency (η_{bth}) & Volumetric Efficiency (η_v)

In this case; some compromising studies mentioned that, the reason of this reduction *partly*; higher density and increased viscosity which leads to poor atomization and fuel vaporization and *mainly* related with the lower heating values of biodiesel (Gopala2014, Mehta 2015, Anbarasu 2014, Abedin 2014, Cecrle 2012). Although, CNG and hydrogenated gases (H_2 , HHO) have better LHV than neat diesel and biodiesel, the mass of these gaseous fuels slightly affected this situation. According to Kumar (2003) and Bose (2009), hydrogen improves the thermal efficiency which hydrogen has high flame speed and wide flammability limits helped the flame propagation through the hydrogen–air mixture leads to rapid heat release rates. Hence at higher loads and less pilot fuel quantity, the combustion becomes too rapid and affected thermal efficiency negatively, Abedin (2014), touched upon that, η_{bth} decreases with the increase of biodiesel percentage in the blends. The increasing biodiesel percentage leaded to lower calorific value. At full load condition, fuel consumption increases with the increasing engine speed; hence the η_{bth} decreases. Also, Abedin (2014) refereed that, the better combustion of oxygen rich biodiesels could be increased the thermal efficiency. η_{bth} results of this study showed that, the reduction value of η_{bth} minimized with CNG and hydrogenated gaseous fuel in spite of less diesel mass (pilot injection) and lower calorific values of biodiesels.

Volumetric efficiency, η_v , is defined as the ratio of the actual volume of air inducted to cylinders and the displacement volume. In upper side of figure 4.9, η_v vs engine speed was illustrated. η_v can be calculated with Equation 4.3. McCarthy(2011) mentioned the percentage of η_v for a normal aspiration engine was 80% and listed the various factors which affected the η_v suchas; mixture strength, compression ratio, specific enthalpy of vaporization of the fuel, heating of the induced charge, cylinder temperature, valve timing, induction and port design and the atmospheric conditions. In figure 4.9, neat diesel operation has slightly higher average results when compared all other mixtures results. This negligible differing is varying between 0.33% - 1.5%. According to Yüksek (2009), higher distillation temperature curve of biodiesel affected the η_v positively. Hence, hydrogen and CNG induction reduced the η_v in dual fuel mode because of the displacement of the intake air with gaseous fuels. Cecrle (2012) reported the results of their studies with the

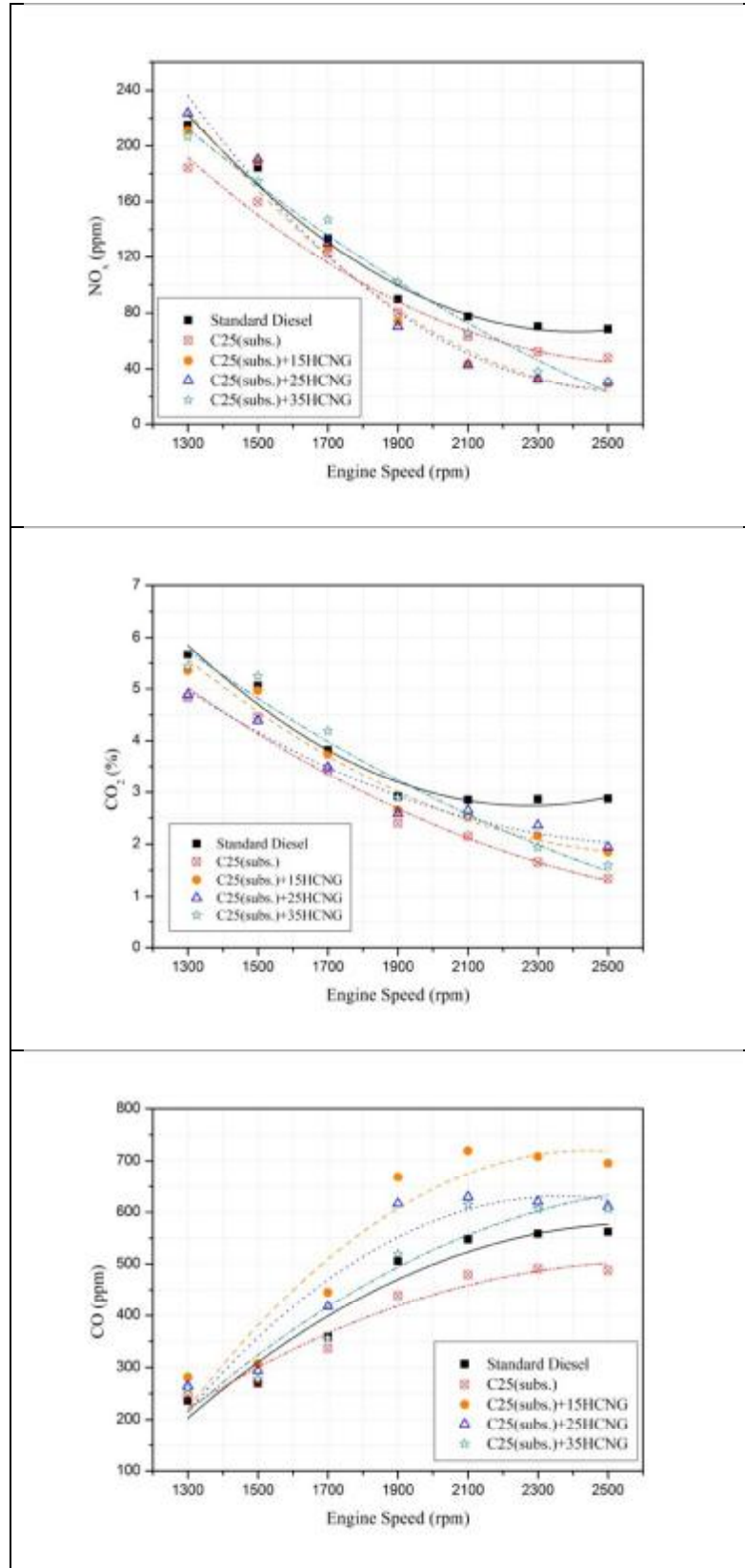
small degradation with 2.16%. η_v results of this study showed that, it was not observed a significant difference outputs between diesel operation and other mixture fuels.

4.2. Exhaust Emission Results

This subsection contains graphs of engine exhaust emission data that presented after detailed examination in between Figure 4.10. and 4.17. Exhaust emission results are separated in to section to discuss up to biodiesel usage in fuel mixtures. Nitrogen-oxides, carbon dioxide and carbon monoxide versus engine speed graphs are combined in single figures to avoid complexity and provide easier comparison.

4.2.1 Emissions of canola biodiesel blends with HCNG and HHOCNG

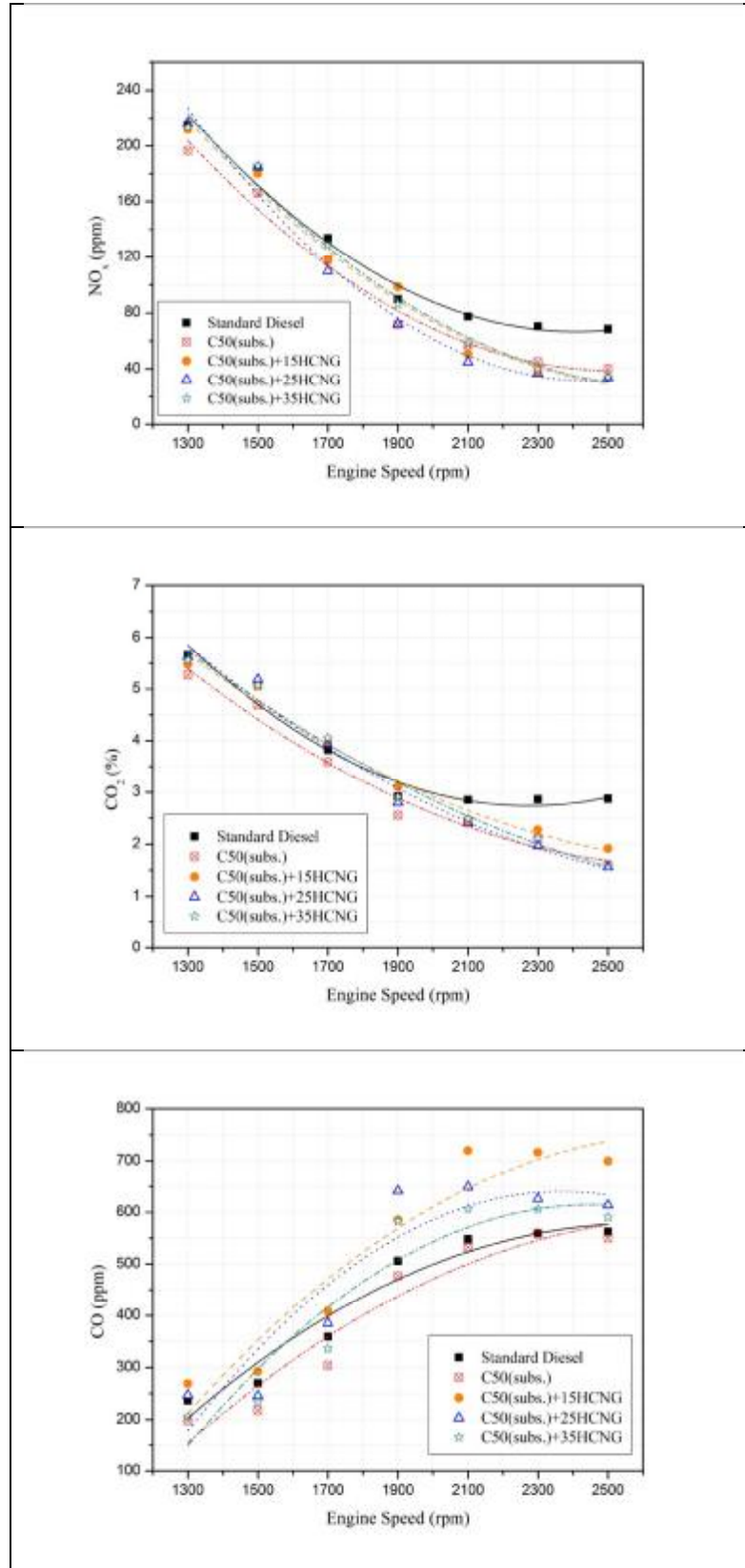
As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.10. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of C25 + 15HCNG, C25 + 25HCNG and C25 + 35HCNG are 18.77%, 16.55%, 9.72%, respectively. Single C25 usage with semi pilot injection decreased NO_x 15.1%. It is clearly seen from the figure; that C25 + 15HCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency. CO_2 versus Speed of C25 + 15-25-35 HCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.10. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These reductions for C25 + 15HCNG, C25 + 25HCNG, C25 + 35HCNG and C25 semi pilot injection (SPI) are 12%, 16.8%, 9.5% and 25%, respectively. C25 + 25HCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation. The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.10 for C25 + 15-25-35 HCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture C25 + 35HCNG with 6.1%. C25 + 15HCNG and C25 + 25HCNG fuel mixtures increased CO emissions 11% and 19%, respectively.

Figure 4.10. NO_x, CO₂ and CO Compressions of C25 with 15-25-35 HCNG mixtures

As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.11. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of C50 + 15HCNG, C50 + 25HCNG and C50 + 35HCNG are 14.86%, 20.18%, 12.31%, respectively. Single C50 usage with semi pilot injection decreased NO_x 14.2%. It is clearly seen from the figure; that C50 + 25HCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency.

CO_2 versus Engine Speed of C50 + 15-25-35 HCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.11. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These reductions for C50 + 15HCNG, C50 + 25HCNG, C50 + 35HCNG and C50 semi pilot injection (SPI) are 7.3%, 11.1%, 9.65% and 17%, respectively. C50 + 25HCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation.

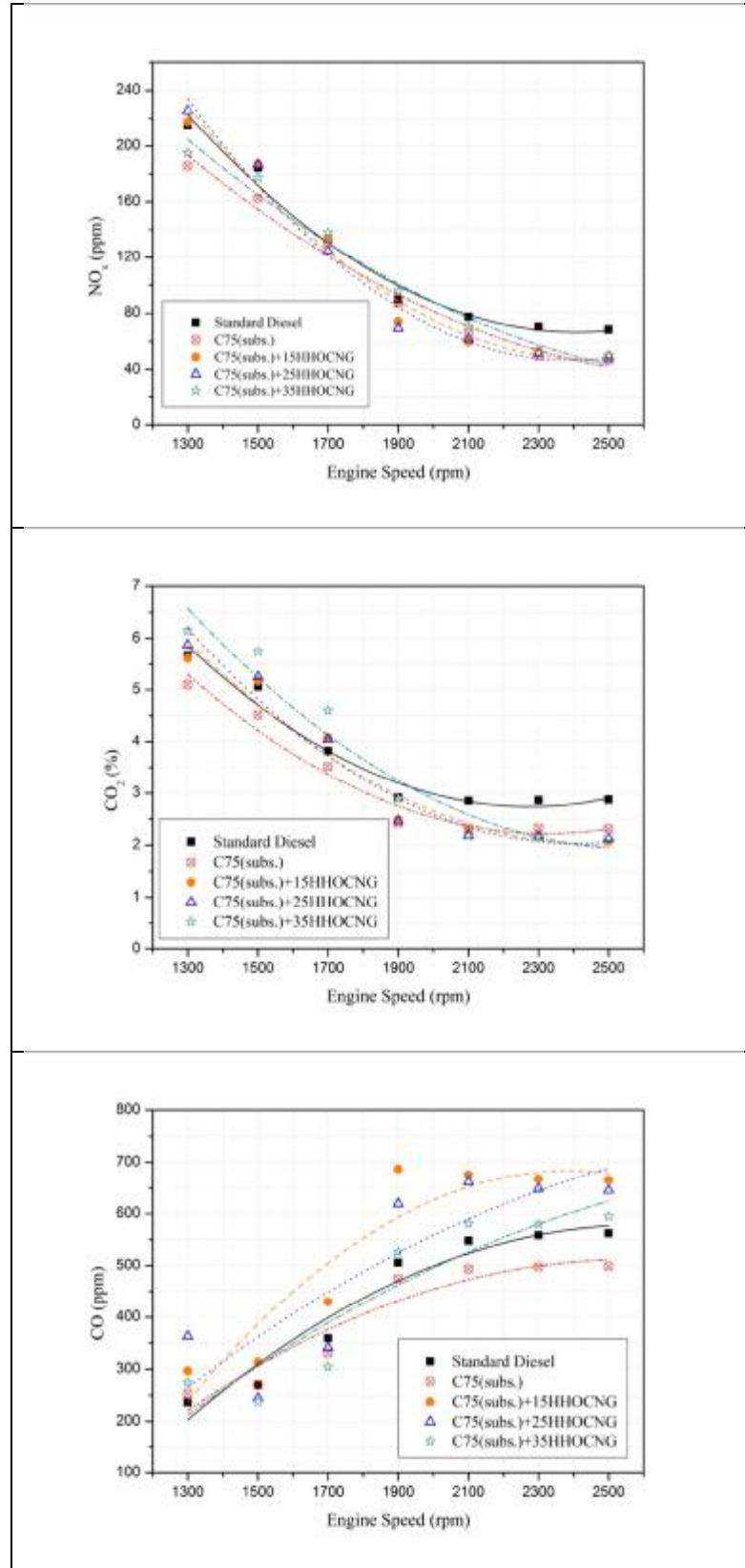
The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.11 for C50 + 15-25-35 HCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture C50 + 35HCNG with 3.8%. C50 + 15HCNG and C50 + 25HCNG fuel mixtures increased CO emissions 17.6% and 10.7%, respectively.

Figure 4.11. NO_x, CO₂ and CO Compressions of C50 with 15-25-35 HCNG mixtures

As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.12. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of C75 + 15HHOCNG, C75 + 25HHOCNG and C75 + 35HHOCNG are 8.23%, 9.35%, 7.7%, respectively. Single C75 usage with semi pilot injection decreased NO_x 13.5%. It is clearly seen from the figure; that C75 + 25HHOCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency.

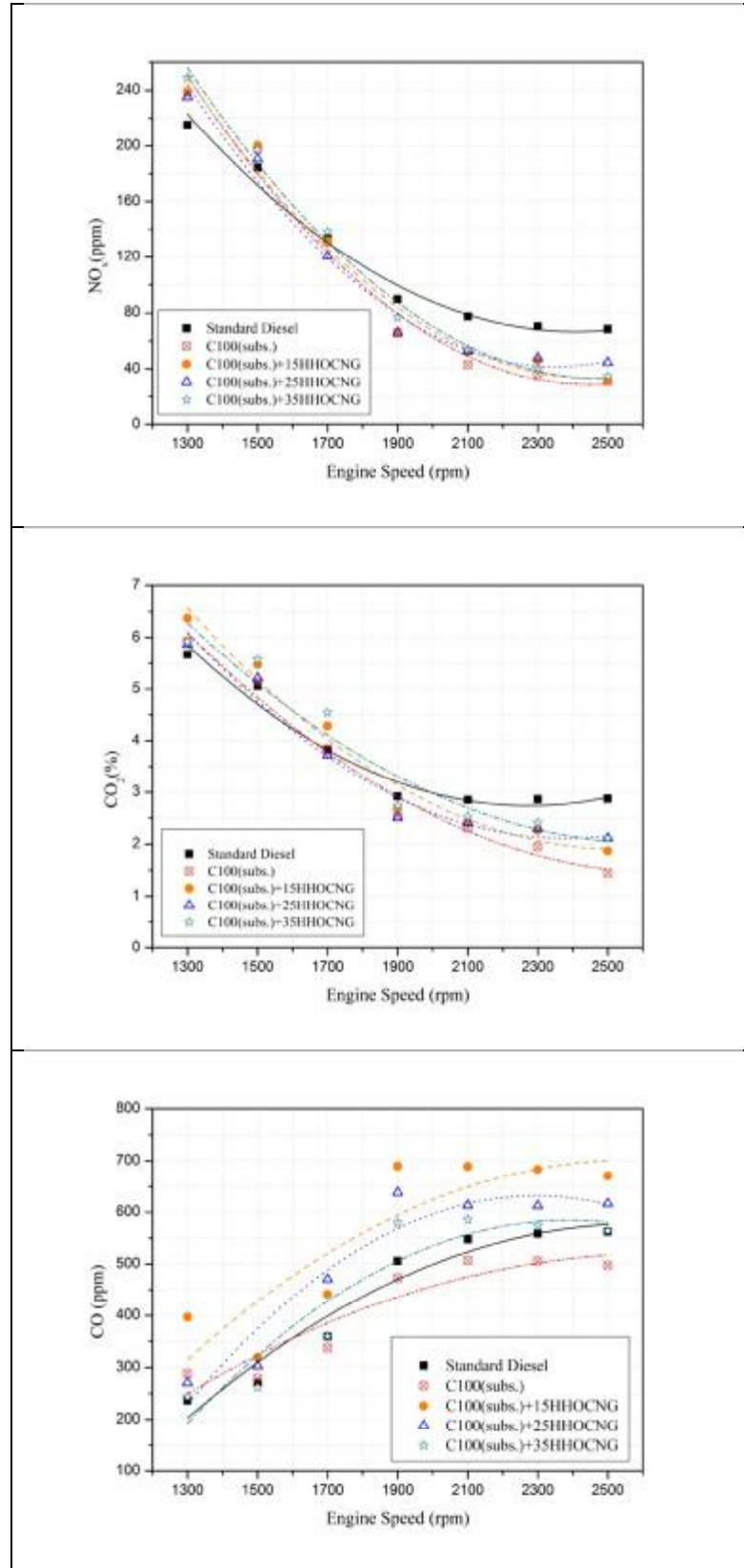
CO_2 versus Engine Speed of C75 + 15-25-35 HHOCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.12. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These overall reductions for C75 + 15HHOCNG, C75 + 25HHOCNG, C75 + 35HHOCNG and C75 semi pilot injection (SPI) are 9.15%, 8.08%, 1% and 13%, respectively. C75 + 25HHOCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation.

The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.12 for C75 + 15-25-35 HHOCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture C75 + 35HCNG with 2%. C75 + 15HCNG and C75 + 25HCNG fuel mixtures increased CO emissions 18% and 13.7%, respectively.

Figure 4.12. NO_x, CO₂ and CO Compressions of C75 with 15-25-35 HCNG mixtures

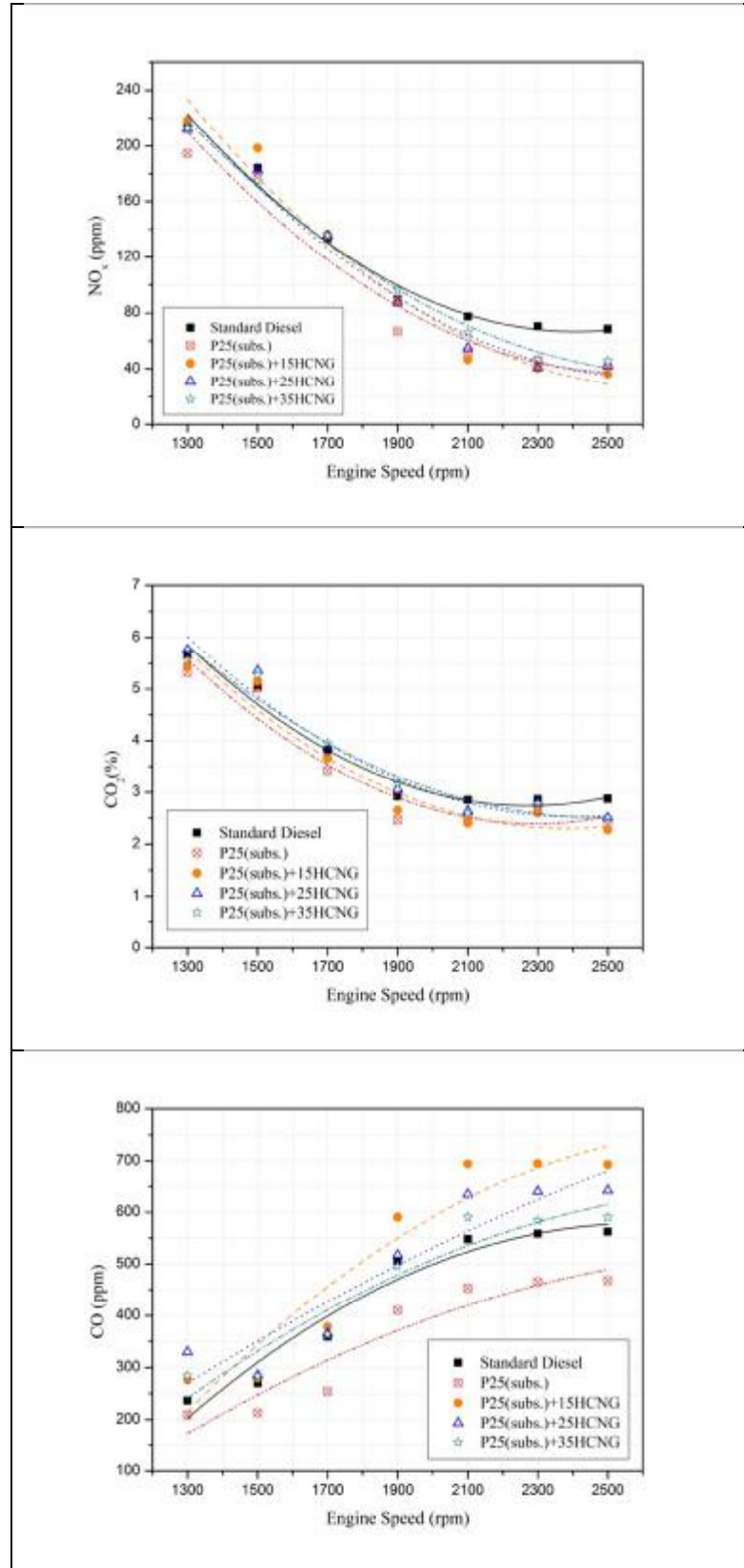
As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.13. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of C100 + 15HHOCNG, C100 + 25HHOCNG and C100 + 35HHOCNG are 9.42%, 10.8%, 6.2%, respectively. Single C100 usage with semi pilot injection decreased NO_x 12.2%. It is clearly seen from the figure; that C100 + 25HHOCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency.

CO_2 versus Engine Speed of C100 + 15-25-35 HHOCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.13. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These overall reductions for C100 + 15HHOCNG, C100 + 25HHOCNG, C100 + 35HHOCNG and C100 semi pilot injection (SPI) are 2.65%, 8%, 1% and 12.4%, respectively. C100 + 25HHOCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation. The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.13 for C100 + 15-25-35 HHOCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture C100 + 35HCNG with 4.1%. C100 + 15HCNG and C100 + 25HCNG fuel mixtures increased CO emissions 21.5% and 13.6%, respectively.

Figure 4.13.NO_x, CO₂and CO Compressions of C100 with 15-25-35 HHO CNG mix.

4.2.2 Emissions of palmbiodiesel blends with HCNG and HHOCNG

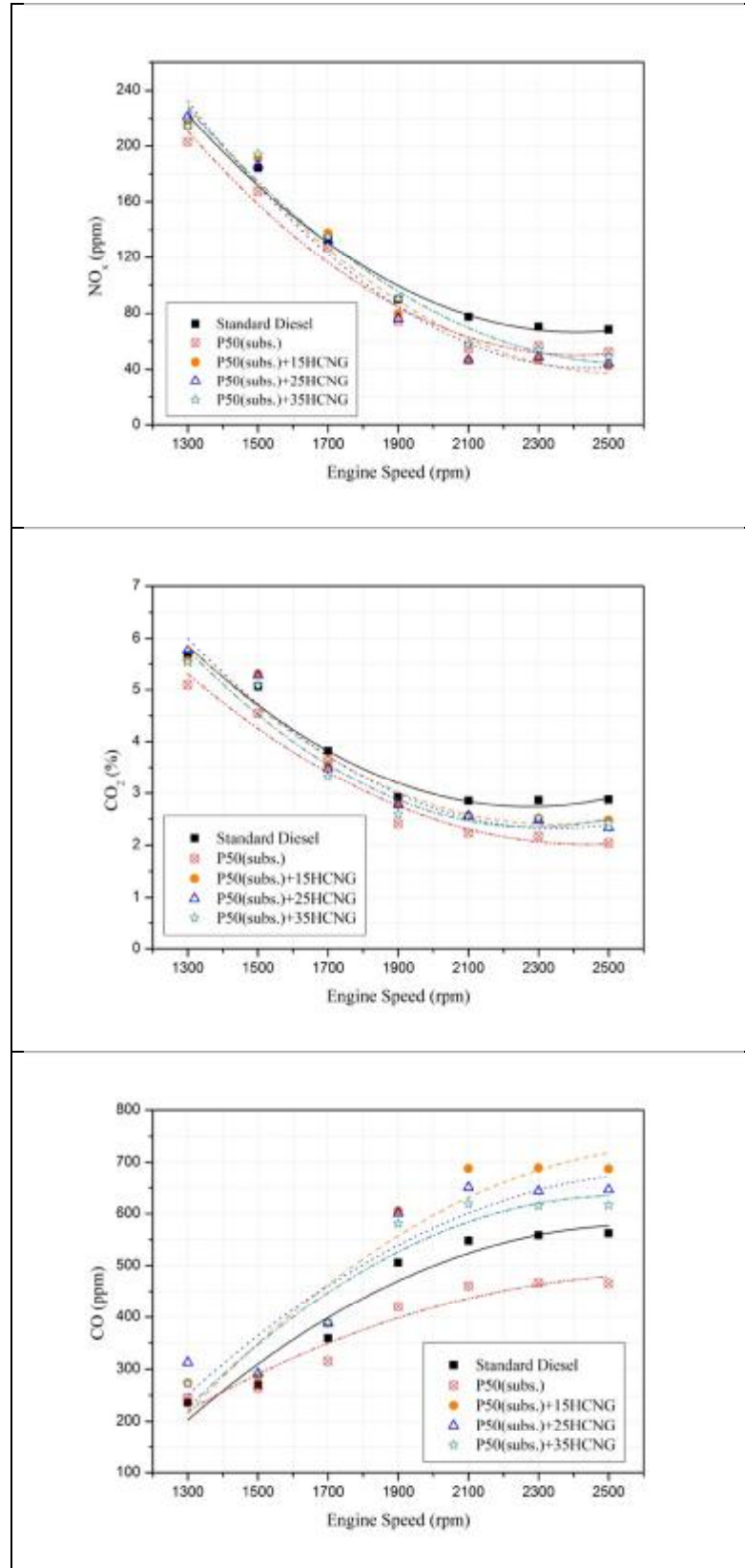
As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.14. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of P25 + 15HCNG, P25 + 25HCNG and P25 + 35HCNG are 9.7%, 11%, 7.7%, respectively. Single P25 usage with semi pilot injection decreased NO_x 16.1%. It is clearly seen from the figure; that P25 + 25HCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency. CO_2 versus Engine Speed of P25 + 15-25-35 HCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.14. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These reductions for P25 + 15HCNG, P25 + 25HCNG, P25 + 35HCNG and P25 semi pilot injection (SPI) are 7.7%, 0.3%, 0.7% and 9.11%, respectively. P25 + 15HCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation. The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.14 for P25 + 15-25-35 HCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture P25 + 35HCNG with 4.7%. P25 + 15HCNG and P25 + 25HCNG fuel mixtures increased CO emissions 15% and 10%, respectively.

Figure 4.14. NO_x, CO₂ and CO Compressions of P25 with 15-25-35 HCNG mixtures

As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.15. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of P50 + 15HCNG, P50 + 25HCNG and P50 + 35HCNG are 10.4%, 11.1%, 5.5%, respectively. Single P50 usage with semi pilot injection decreased NO_x 14%. It is clearly seen from the figure; that P50 + 25HCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency.

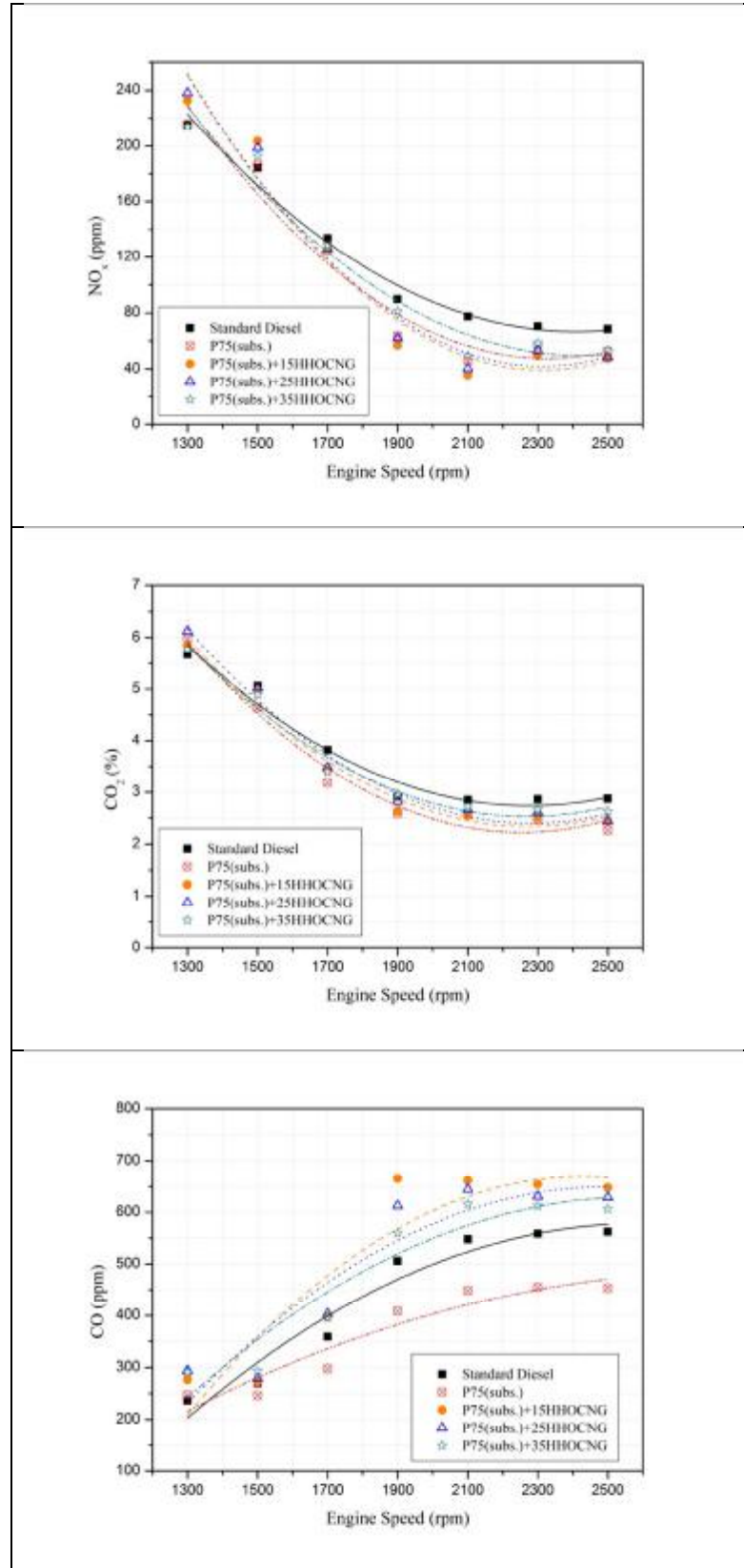
CO_2 versus Engine Speed of P50 + 15-25-35 HCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.15. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These reductions for P50 + 15HCNG, P50 + 25HCNG, P50 + 35HCNG and P50 semi pilot injection (SPI) are 5.2%, 5.6%, 8.4% and 17%, respectively. P50 + 25HCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation.

The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.15 for P50 + 15-25-35 HCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture P50 + 35HCNG with 10%. P50 + 15HCNG and P50 + 25HCNG fuel mixtures increased CO emissions 16% and 14%, respectively.

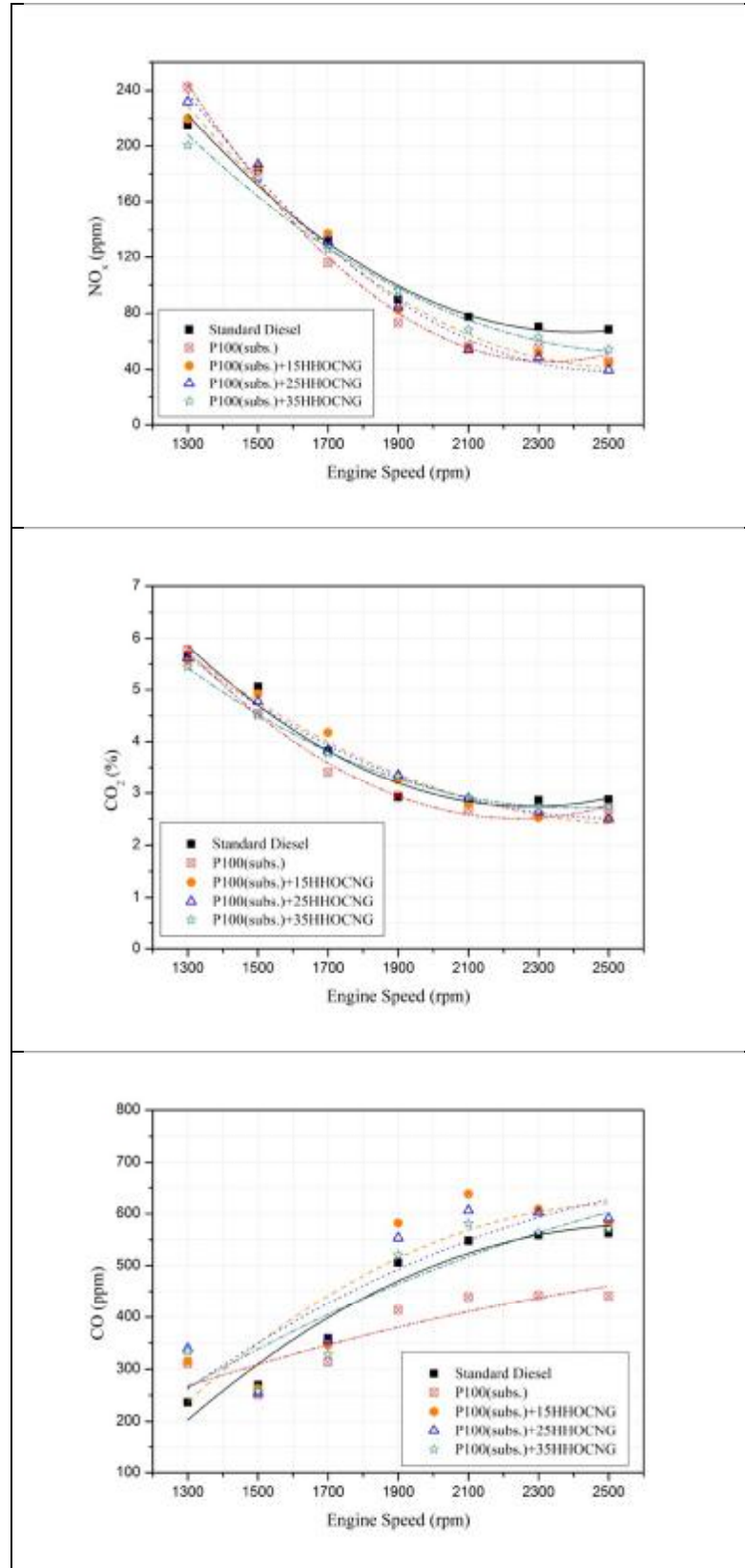
Figure 4.15. NO_x, CO₂ and CO Compressions of P50 with 15-25-35 HCNG mixtures

As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.16. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, successful reduction of NO_x emission is obtained. Numerically, mean reduction values of P75 + 15HHOCNG, P75 + 25HHOCNG and P75 + 35HHOCNG are 11.65%, 9.5%, 8%, respectively. Single P75 usage with semi pilot injection decreased NO_x 12.5%. It is clearly seen from the figure; that P75 + 15HHOCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency.

CO_2 versus Engine Speed of P75 + 15-25-35 HHOCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.16. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These reductions for P75 + 15HHOCNG, P75 + 25HHOCNG, P75 + 35HHOCNG and P75 semi pilot injection (SPI) are 6.6%, 3.5%, 4% and 10%, respectively. P75 + 15HHOCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation. The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.16 for P75 + 15-25-35 HHOCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture P75 + 35HHOCNG with 9.9%. P75 + 15HHOCNG and P75 + 25HHOCNG fuel mixtures increased CO emissions 14.9% and 12.9%, respectively.

Figure 4.16. NO_x, CO₂ and CO Compressions of P75 with 15-25-35 HCNG mixtures

As it is known, NO_x is very important emission parameter for diesel engines and especially biodiesel fuels. At the top of figure 4.17. NO_x output data of fuel mixtures are presented versus Engine Speed. Results show that, very similar reduction of NO_x emission is obtained. Numerically, mean reduction values of P100 + 15HHOCNG, P100 + 25HHOCNG and P100 + 35HHOCNG are 8.09%, 7.96%, 4.88%, respectively. Single P100 usage with semi pilot injection decreased NO_x 8.73%. It is clearly seen from the figure; that P100 + 15HHOCNG have resulted with better NO_x than all other fuel mixtures. Lean AFR could reduce the NO_x emissions and brings out an extra increase in engine efficiency. CO_2 versus Engine Speed of P100 + 15-25-35 HHOCNG fuel mixtures and diesel fuel are illustrated at the middle zone of Figure 4.17. It is clearly seen that; all fuel mixtures succeed about reducing CO_2 emission when compared with diesel fuel. These reductions for P100 + 15HHOCNG, P100 + 25HHOCNG, P100 + 35HHOCNG and P100 semi pilot injection (SPI) are 1.3%, 1.68%, 2.4% and 5.7%, respectively. P100 + 35HHOCNG is the most successful fuel mixture among all mixtures about reducing CO_2 emission. Significant reduction of CO_2 emissions mostly depends on the amount of hydrocarbon fuel inducted into the cylinders. Substitution of liquid fuel directly decreases the inducted amount of hydrocarbon fuel into the engine. Therefore, enrichment of semi-pilot injection process produces lower CO_2 levels than normal compressed ignition engine operation. The variations of CO emission levels versus engine speed are presented at the bottom of figure 4.17 for P100 + 15-25-35 HHOCNG fuels. There are obvious increments of CO emissions with dual-fuel compared to standard diesel results. Especially at higher engine speeds, the difference between fuel mixtures and diesel fuel is increasing. The reason of this could be explained with incomplete combustion of CNG due to insufficient ignition sources as well as higher fuel–air equivalence ratio. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation. Minimum average increment was obtained by using mixture P100 + 35HHOCNG with 3.65%. P100 + 15HHOCNG and P100 + 25HHOCNG fuel mixtures increased CO emissions 8.8% and 7.8%, respectively.

Figure 4.17.NO_x, CO₂and CO Compressions of P100 with 15-25-35 HHOCNG mix.

5. CONCLUSIONS

Developing countries spend lots of their money on imported petroleum diesel. Production and consumption of fossil fuel has been increasing every year around 5–6%. Requirement for more fossil fuel while aiming to reduce environmental pollution can be obtained through alternative fuels.

Aim of this experimental investigation is to reduce diesel fuel dependency by HHOCNG and HCNG fuel replacement with diesel. 15%, 25%, 35% (by volumes) hydrogenated compressed natural gas mixtures were used as experimental fuels to achieve the goals of this thesis. CNG fuel supplied from a bundle tank, hydrogen from single hydrogen tank while HHO gas supplied from HHO generator.

Intermediate results of experiments proofed that hydrogen addition to diesel or biodiesel fuel is more environmentally friendly solution. On the other hand, HHO addition provided higher engine performance which can help to solve one of the biggest handicap of high level biodiesel blends usage. Additionally, biodiesel is a naturally environmentally friendly and a good solution to decrease exhaust gas emissions. Unfortunately, high level of biodiesel fuel blends results with losses of performance for non-modified internal combustion engines. Under the guidance of these partial results, pure hydrogen used for 25% and 50% biodiesel fuel blends. Because 25% and 50% biodiesel fuel blends needed to support emission aspects than performance compared to 75% and 100% biodiesel fuel blends. So HHO became preferable addition to 75% and 100% biodiesel fuel blends which tolerated losses of the engines performances and provided better results than hydrogen addition.

Because of high auto-ignition temperatures of gas fuel mixtures, combustion of air and gaseous fuel does not take place in cylinders. A small amount of suitable liquid fuel needs to be injected near the end of the compression stroke to ignite the gaseous mixture. Basically, for economic and environmental reasons, the quantity of the liquid fuel pilot needs to be minimized relative to that of the cheaper fuel gases.

For this purpose, semi pilot biodiesel injection method is used in this thesis. Required amount of canola (COME) and palm (POME) biodiesels are produced from commercial canola and palm oils by transesterification method in fuel laboratory of automotive engineering. Palm and canola biodiesels are alternative fuels that can reduce environmental pollution and meet the demand for fossil fuel. Compared with other vegetable oils, palm and canola oil is a more sustainable and affordable fuel and also environment friendly.

To sum up the experimental procedure of this theses; HCNG, HHOCNG, canola and palm biodiesel fuel mixtures were used as a replacement fuel in a non-modified diesel engine. Gaseous fuel mixtures introduced to combustion chamber via intake manifold. Reduction of liquid fuels are obtained by dint of semi pilot injection and controlled by stepping motor devices. Determined fuel mixture combinations were tested in a 3.6 L, four cylinders, four stroke and water coolant diesel test engine between 1300-2500 rpms. Experimental results (engine performance and exhaust emissions parameters) were designed to compare and illustrated with graphics as brake specific fuel consumption, brake power and brake torque; NO_x , CO_2 and CO . The minimum and optimum quantities of pilot injection at specific engine operational conditions are determined, and the effectiveness of different measures to improve engine performance for various operational conditions are examined. Compression of fuel pump plunger effects are presented as 360° and 720° clockwise reset positions. Most promising results in engine performance and exhaust emissions are obtained by 720° clockwise reset position. Hence, experimental investigations of 720° clockwise pin rotation results are preferred and presented in this thesis. Main outputs of this thesis can be conclude as below;

- Successfully achieved to predicted method as dual-fuel operation with substitution of liquid fuel in a non-modified diesel engine. Thus, unnecessary modification costs and time consuming is prevented.
- Optimum fuel mixture ratios are determined to maintain and improve engine performance characteristics. At the same time very promising reduction of exhaust emission is provided.

- Hydrogenated gaseous fuel mixtures emission results were placed between single pilot biodiesel injection and standard diesel operations. CO₂ and NO_x reductions were obtained superior outputs compared to standard diesel fuel operation.
- Disadvantages of using biodiesel on performance results are compensated with HCNG and HHOCNG gaseous fuels by reducing the dependence on fossil fuels.
- Mixtures of hydrogen or HHO with compressed natural gas compensate each other's disadvantages and provide a promising gaseous fuel mixture with several advantages. Combination of hydrogen and compressed natural gas as fuel mixture for internal combustion engines provided improvement for engine performance results by hydrogen fuel properties; on the other hand, natural gas reduced the harmful exhaust emission and engine fuel operating cost.
- Reduction of η_{bth} is minimized with CNG and hydrogenated gaseous fuels in spite of less diesel mass (semi-pilot injection) and lower calorific values of biodiesels.
- Neat diesel operation had slightly higher η_v when compared with all other mixtures results.
- Economic, environmental and efficiency aspects are considered while determination of fuels.
- Under the guidance of experiment results, P25 + 25HCNG fuel mixture with semi-pilot injection operation was obtained the best combination to use as an alternative fuel for this test engine.
- Biodiesel blends at low ratios can be used in any normal internal combustion diesel engine without modifications. But higher blends of biodiesel (over 50%) may require minor modifications. Because the properties of biodiesel are close to diesel fuels but not the same.
- Under same conditions, palm biodiesel provided better engine performance results. However, canola biodiesel was better at the point of engine exhaust emission results.

- There was inverse correlation between hydrogen and HHO gas usage as fuel. Under same conditions, single hydrogen enrichment provided higher engine emission. On the other hand HHO usage was more preferable than hydrogen for engine performance.
- Substituted canola C25 and C50 blends performance results are more promising than C75 and C100. Similarly, palm biodiesel blends provided better results with low biodiesel blend.

As an overall result from the experimental study, effects of using HCNG and HHOCNG with palm and canola biodiesel fuel mixtures to engine performance and emissions in CI are added to literature. From the performance standpoint, enriching the intake air with 25HCNG and using P25 biodiesel fuel for pilot injection is more preferred than all other gaseous and liquid fuel mixtures. On the other hand, in terms of exhaust gas emissions, enriching the intake air with 25HCNG and using C25 biodiesel fuel for pilot injection provide better results compared to other fuel mixtures. In both case, 25HCNG is the best gaseous fuel mixture.

Finally, method of using HHOCNG and HCNG with semi pilot biodiesel injection as an alternative solution for diesel engines applications is performed successfully. Satisfactory results showed that methodology of this thesis is open area for more research and developments. It can be examined for future studies;

- Diesel engines with electronic control unit for more accurate pilot fuel injection,
- Other biodiesel fuels with various blend ratios, especially palm and canola can be mixed 50% ratios to optimize their performance and emission characteristic.
- Using simulation software and optimizing the combustion parameters,
- Changing fuel injection timing and pressure to obtain an optimization between performance and emissions.

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CURRICULUM VITAE

Mustafa Kaan BALTACIOĞLU was born on July 17, 1984 in İskenderun. After completing his education in İstiklal Makzume Anatolian High School, he enrolled in Çukurova University Mechanical Engineering Department in 2003. He had graduated from the Mechanical Engineering Department of Çukurova University with a Bachelor of Science degree in 2007. After graduated from university he performed his military service and moved to United States of America. He finished his Master of Science degree in 2012 in Mustafa Kemal University. He had started to Doctor of Philosophy education in Çukurova University with the same department. He has been working as a research assistant in İskenderun Technical University since 2010. He has published 17 reaseach papers in national&international journals. His research areas are; internal combustion engines, biofuels, alternative gas fuels, fluid mechanics, energy sustainability and environmental progress, energy and mechanical engineering subdivisions.

APPENDIX

APPENDIX 1. Papers

The following publications are prepared from this thesis.

PAPER I-EXPERIMENTAL COMPARISON OF PURE HYDROGEN AND HHO (HYDROXY) ENRICHED BIODIESEL (B10) FUEL IN A COMMERCIAL DIESEL ENGINE

This paper was presented at the 6th International Conference on Hydrogen Production, Oshawa, Canada (2015) and was accepted for publishing in “International journal of Hydrogen Energy”

PAPER II-EXPERIMENTAL INVESTIGATION OF USING 30HCNG FUEL MIXTURE ON A NON-MODIFIED DIESEL ENGINE OPERATED WITH VARIOUS DIESEL REPLACEMENT RATES

This paper was published in International journal of Hydrogen Energy, 2016, doi:10.1016/j.ijhydene.2015.12.112

PAPER III-OPTIMIZING THE QUANTITY OF DIESEL FUEL INJECTION BY USING 25HHOCNG GAS FUEL MIXTURE

This paper was published in Advanced Engineering Forum Journal, Vol. 14 (2016) pp 36-45

PAPER I

----- Forwarded message -----

From: Ibrahim Dincer <Ibrahim.Dincer@uoit.ca>

Date: 2015-11-30 0:29 GMT+02:00

Subject: Your Submission - HE-D-15-01637R1

To: kaanbaltacioglu@gmail.com

Ms. Ref. No.: HE-D-15-01637R1

Title: EXPERIMENTAL COMPARISON OF PURE HYDROGEN AND HHO (HYDROXY) ENRICHED BIODIESEL (B10) FUEL IN A COMMERCIAL DIESEL ENGINE
International Journal of Hydrogen Energy

Dear Mr. BALTACIOGLU,

I am pleased to inform you that your paper "EXPERIMENTAL COMPARISON OF PURE HYDROGEN AND HHO (HYDROXY) ENRICHED BIODIESEL (B10) FUEL IN A COMMERCIAL DIESEL ENGINE" **has been accepted for publication in the International Journal of Hydrogen Energy.**

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For further assistance, please visit our customer support site at <http://help.elsevier.com/app/answers/list/p/7923> Here you can search for solutions on a range of topics, find answers to frequently asked questions and learn more about EES via interactive tutorials. You will also find our 24/7 support contact details should you need any further assistance from one of our customer support representatives.

Yours sincerely,

Rami S. El-Emam, Ph.D.

Special Issue Guest Editor

EXPERIMENTAL COMPARISON OF PURE HYDROGEN AND HHO (HYDROXY) ENRICHED BIODIESEL (B10) FUEL IN A COMMERCIAL DIESEL ENGINE

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ABSTRACT

Main objective of this study is to compare performance and emission characteristics of a pilot injection diesel engine with the additions of alternative fuels like pure hydrogen, HHO and biodiesel. In order to achieve this goal, helianthus annuus (sunflower) biodiesel was produced and blended with volumetric ratio of 10% with diesel fuel. Additionally, intake air was enriched with pure hydrogen or HHO via intake manifold without any structural changes except reduction of injected diesel fuel on the 3.6 L, four cylinders, four stroke diesel engine. Amount of Hydrogen fuel supplied to the engine was adjusted to 10 L/min during the experiments. The effects of pure hydrogen and HHO usage with the addition of biodiesel to the engine performance values (Brake Torque, Brake Power and Brake Specific Fuel Consumption) and exhaust emission values (NO_x, CO₂, CO) were investigated in between 1200-2600 rpm engine speeds. Engine performance values were increased with the enriching the intake air with HHO than pure hydrogen compared to the standard diesel fuel operating condition. On the other hand, in terms of exhaust gas emissions, pure hydrogen provided better results than HHO. In both cases, changes on the engine performance results were minimal however improvements on exhaust gas emissions were very promising.

Keywords: Pure hydrogen, HHO (Hydroxy), biodiesel, engine performance, exhaust gas emissions.

INTRODUCTION

It is very well known that environmental and economic criteria push scientists to find out alternative solutions to energy demand. Hydrogen is one of the alternative fuels to create clean and environment friendly future. It is an energy-efficient, low-polluting fuel. When hydrogen is used in a fuel cell to generate electricity or is combusted with air, the only products are water and a small amount of NO_x [1]. In last decades researches have been carried out lots of studies on hydrogen usage in diesel engines. Diesel engine has wide range of using area at transportation. Some of the researchers used pure hydrogen [2-6] and others tried with hydrogen-rich gases (HRG) and hydrogen/oxygen mixture produced by water electrolysis [7-11]. Birtas, A. et al. (2011), mentioned that using pure hydrogen or hydrogen containing gas produced through water electrolysis, are notably different [10]. However, previous studies have a lack of comparison between the using pure hydrogen and hydroxy gas (HHO) in diesel engine. The present experimental study presents a comparison between effects of pure hydrogen and HHO to performance and emissions of a diesel engine. Required pure hydrogen is obtained from hydrogen tank and HHO with processes such as water electrolysis. Prigent, M. (1997), used similar method to produce small amount of hydrogen, or when high-purity hydrogen is required [12]. Water electrolysis is one of the most important industrial processes for hydrogen production today, and is expected to become even more important in the future [13]. Also it is a practical solution for onboard

production of hydrogen, which avoids the storing of heavy pressurized hydrogen tanks [7]. Another way to obtain hydrogen is heavy pressurized (170 bar) commercial tanks which contain hydrogen at 99.99% purity.

Additionally, sunflower biodiesel is used with commercial diesel fuel at a level of 10% volumetric ratio. Lot of different vegetable oils, like canola, palm, cottonseed, soybean, sunflower etc. have been used to produce biodiesel. Vegetable oils offer almost the same power output with slightly lower thermal efficiency when used in diesel engines [14]. Among these, sunflower oil has very good physical properties. Calorific value and Cetane number are higher compared to many others and slightly less than diesel fuel. The sunflower plant can grow all over the Turkey. Biodiesel fuel can effectively reduce engine-out emissions of unburned hydrocarbons, particulate matter, and carbon monoxide in compression ignition (CI) engines. However, a slight increase in emissions of nitrogen oxides has been observed in the use of oxygenated fuels in general [15-18]. High viscosity, a low ignition point, and unsaturated hydrocarbons molecules are the disadvantages of biodiesel fuels.

Biodiesel fuels are non-toxic and do not contain aromatic and can be resolved in nature better than fuels used in the industry and transportation. The most important point is that biodiesel fuel does not contain sulphur [19, 20].

METHODOLOGY

All tests were carried out in the Internal Combustion Engine (ICE) Laboratory of Automotive Engineering Department of Çukurova University. A schematic of experimental set up is shown in figure 1 below. Also technical specifications of the test engine and hydraulic dynamometer are given in table 1 and table 2 respectively. Performance tests were conducted on four cylinders, four-stroke, naturally aspirated, water-cooled direct injection CI engine which connected to a Netfren mark hydraulic dynamometer for loading the test engine. Exhaust emissions were measured by MRU Delta 1600 V gas analyzer. A data logger was used to collect test engine, dynamometer, volumetric flow meters and gas analyzer data's. Acquired data was then transferred to host computer from data logger.

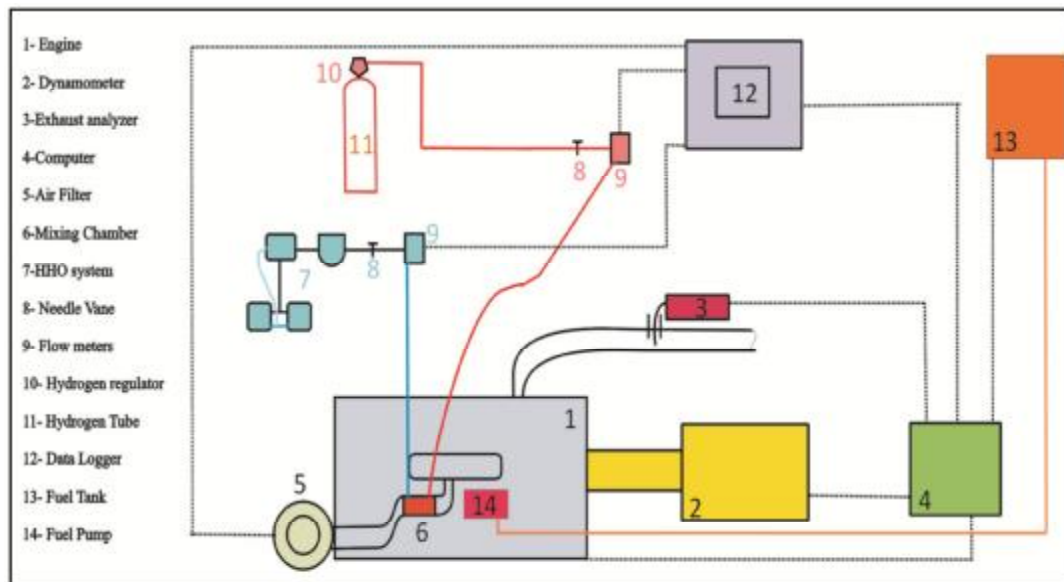


Fig. 1: Schematic view of experimental set up.

Four sets of experiments were performed. First set of experiments was performed with neat diesel fuel, second set of experiments with B10, third set of experiments with pure hydrogen enriched B10 and fourth set of experiments was with hydroxy gas (HHO) enriched B10 fuel. The engine was operated to reach steady state conditions for about 10 minutes at the

beginning of each test. All experiments were repeated three times to increase accuracy of results and decrease experimental errors. The engine was subjected constant speed, full load conditions during tests [21]. The engine performance values of brake power, brake torque, and brake specific fuel consumption, and exhaust emission values of carbon monoxide, carbon dioxide and nitrogen oxide were obtained and results were evaluated.

Table 1: Test engine specifications.

Brand	Mitsubishi Canter
Model	4D34 - In line 4
Type	D.I. diesel with glow plug
Displacement	3567cc
Bore	104mm
Stroke	105mm
Power	89kW @ 3200rpm
Torque	295Nm @ 1800rpm

Table 2: Hydraulic dynamometer specifications.

Brand	Netfren
Torque range	0-1700 Nm
Speed range	0-7500 rpm
Body diameter	250mm
Torque arm length	250mm
Torque	295Nm @ 1800rpm

Constant rate of 10 L/min pure hydrogen (purity: 99,999%) and HHO mixed with intake air respectively by using mixture chamber which is shown in figure 2. Mixture chamber is placed before the intake manifold entrance as shown below. "A, B and C" in figure 2, point the pure hydrogen entrance from commercial hydrogen tank with 99,999% purity, hydrogen/oxygen mixture (hydroxyl gas) entrance produced by water electrolysis and intake manifold of the test engine in sequence.

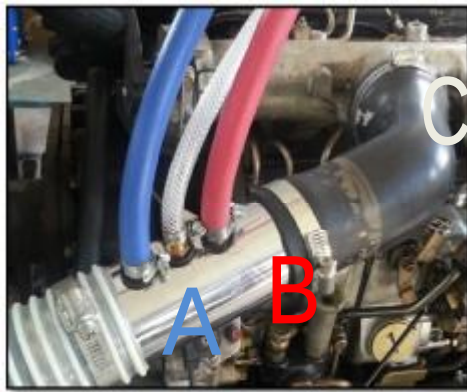


Fig. 2: Mixture Chamber.

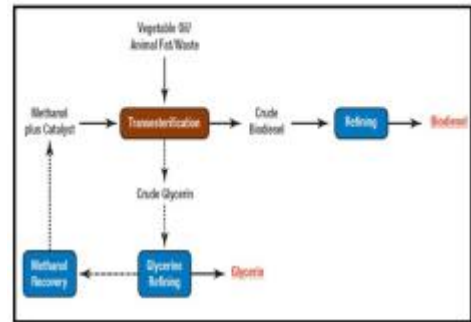


Fig. 3: Schematic view of biodiesel production path [22].

Basically, HHO system is electrolysis of distilled water with electrolyte to any substance containing free ions that make it electrically conductive. In this experimental study, KOH (potassium hydroxide) is chosen as an electrolyte. System contains double parallel connected dry cells with 14 plates total, water reservoir, bubbler, solenoid relay switch, constant current Pulse Width Monitor (PWM) with liquid crystal display, fittings and electrical wires. HHO generator runs with engine battery (24 V). Installed HHO generator needs 30 amps to achieve 10 LPM of oxy-hydrogen gas. Properties of HHO and hydrogen are very similar to each other therefore HHO has been adopted instead of hydrogen while representing the fuel properties [23]. Some of the important specifications of hydrogen and diesel fuel compression with sunflower biodiesel fuel are shown in table 3 and table 4 respectively. Required amount of sunflower biodiesel is produced by transesterification method in fuel laboratory of automotive engineering. The most common way to produce

biodiesel is the transesterification method, which refers to a catalyzed chemical reaction involving vegetable oil and alcohol to yield fatty acid alkyl esters (i.e., biodiesel) and glycerol [24]. A simple biodiesel production path is shown in figure 3 above. B10 fuel is prepared by mixing commercial diesel fuel and produced biodiesel with a volumetric ratio of 90% diesel fuel +10% biodiesel respectively.

Table 3: Fuel Properties of diesel and hydrogen [23, 25].

Property	Diesel fuel	Hydrogen
Density (kg/m^3)	840 (l)	0.082(g)
Limits of flammability in air	-	4-75 vol. %
Auto-ignition Temperature $^{\circ}\text{C}$	254-285	585
Laminar Flame Speed (cm/s)	128	265-325
Lower Heating Value (MJ/kg)	42.61	120.21

Table 4: Fuels Properties of neat diesel fuel and sunflower biodiesel [26].

Property	Diesel fuel	Sunflower biodiesel	Standards
Kinematic viscosity [mm^2/s]	3.06	4.6	EN ISO 3104
Cetane number	50	49	EN ISO5165
Pour point [$^{\circ}\text{C}$]	-	1	ISO 3016
Cold filter plugging point [$^{\circ}\text{C}$]	-16	-2	EN 116
Flash point [$^{\circ}\text{C}$]	76	183	EN ISO3679

RESULTS AND DISCUSSION

All experiments were conducted at full load condition in the range of 1200-2600 rpm and increased by 200 rpm. Experimental results are presented under engine performance and exhaust gas emissions subsections shown below.

Engine Performance

Three figures illustrated in this subsection that contains brake power, brake torque and brake specific fuel consumption. Brake power variations versus engine speed are shown in figure 4. Results show that 10 L/min HHO enrichment of B10 fuel provides an average of 8.31% improvement according to neat diesel fuel. Similar results for pure hydrogen enrichment B10 fuel also are obtained with an average of 4.33% improvement. Maximum improvement was obtained at medium engine speed range around 1750-2000 rpm. Brake power of HHO+B10 enrichment fuel was an average of 13% higher than neat diesel fuel at 1800 rpm.

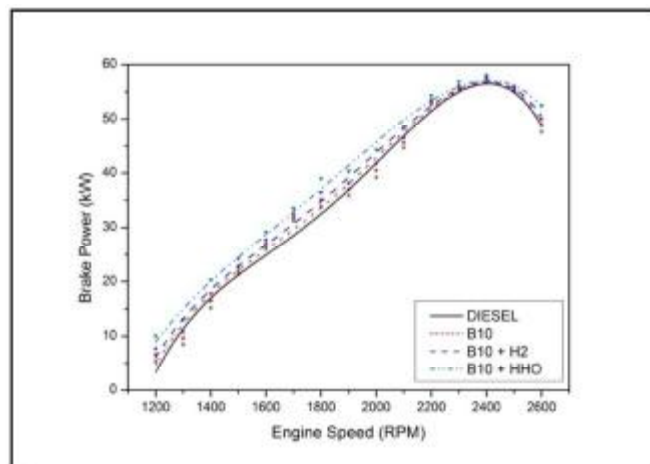


Fig. 4: Brake power versus engine speed.

Using pure hydrogen or HHO in combustion chamber can lead better thermodynamic efficiency. Hydrogen has higher heating value and flame speed compared to conventional

liquid fuels. Hence, both of the hydrogen enriched fuels in this study provide better brake torque outputs than neat diesel fuel and B10 as shown in figure 5. Peak points of brake torque values measured at 1400 rpm for all test fuels. Similar to the brake power outputs, maximum brake torque point of 209.57 Nm at 1400 rpm is obtained by HHO+B10 enrichment fuel.

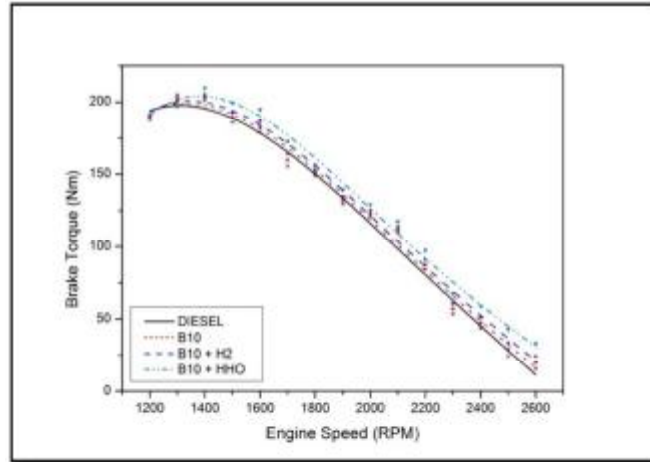


Fig. 5: Brake torque versus engine speed.

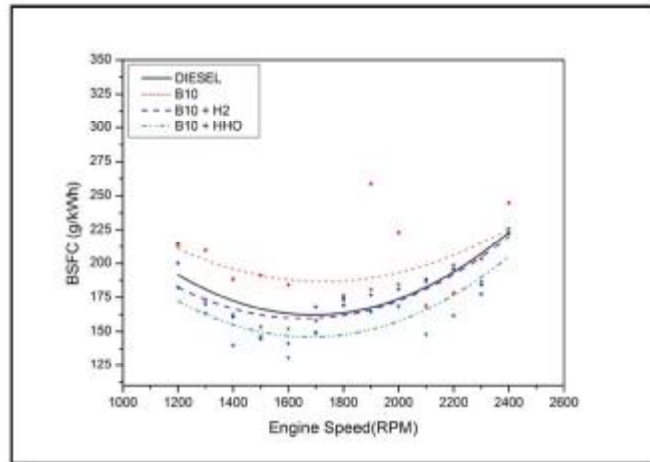


Fig.6. Brake specific fuel consumption versus engine speed.

Brake specific fuel consumption (BSFC) is a measure of the fuel efficiency of any prime mover that burns fuel and produces rotational, or shaft power. It is typically used for comparing the efficiency of ICE with a shaft output. It is the rate of fuel consumption divided by the power produced [27]. BSFC of hydrogen enriched fuels were less than neat diesel fuel and B10 for all engine speeds due to high energy content of hydrogen. BSFC results are demonstrated in the figure 6 above for all the test fuels. H_2 +B10 fuel decreased BSFC an average of 2% while decrement for HHO+B10 was 10%. B10 fuel increased the BSFC due to less heating value than neat diesel fuel.

Exhaust Gas Emissions

The biggest advantage of alternative fuels such as biodiesel and hydrogen over diesel fuel is their environmental friendliness. Hydrogen enrichment to the fuels reduces exhaust emission values. Carbon dioxide, carbon monoxide, and nitrogen oxide results of all experiments represented in figures 7-9 in this subsection.

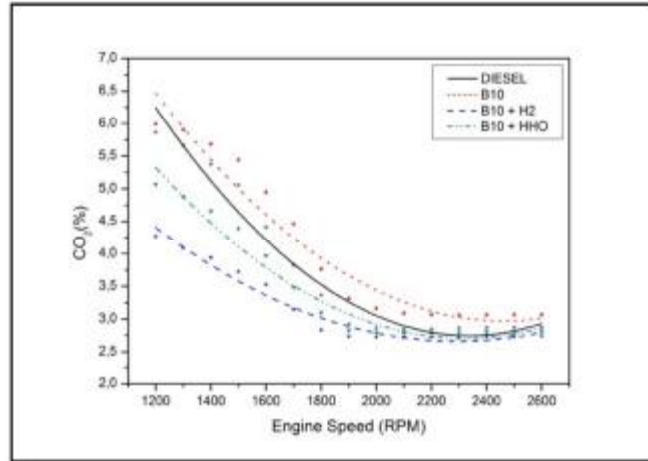


Fig. 7: CO₂ versus engine speed.

CO₂ emission values of H₂+B10 and HHO+B10 decreased at lower engine speed between 1200-1800 rpm compared to the medium or higher engine speeds. Meanwhile B10 fuel increased CO₂ gas emissions at all engine speeds. Improvements by hydrogen enriched fuels were noticeable as shown in figure 7. Maximum decrement of 38% was achieved using 10 L/min pure hydrogen with B10 fuel at 1300 rpm. Average reductions of CO₂ emissions for H₂+B10 were 22% while 12% for HHO+B10.

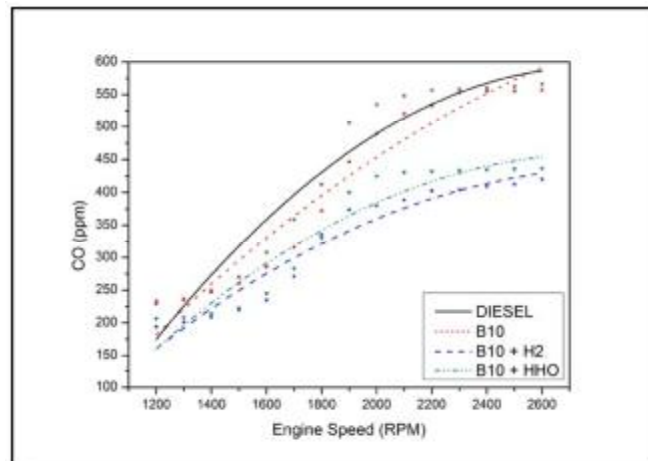


Fig. 8: CO versus engine speed.

CO emission is toxic and an intermediate product in the combustion of a hydro carbon fuel, so it results from incomplete combustion. Emission of CO is greatly dependent on the air-fuel ratio relative to the stoichiometric proportions [15]. As shown in figure 8, observation indicates that neat diesel fuel has higher CO gas emissions than pure H₂+B10 and HHO+B10 fuels. Higher oxygen content of B10 and small quenching distance of hydrogen improves combustion in the cylinder. As a result CO gas emissions are dropped down for all engine speeds. It is clear in figure 8 that maximum decrement occurred at high engine speeds while there is fast combustion without enough time. H₂+B10 and HHO+B10 fuels performed the similar characteristics however CO gas emissions H₂+B10 fuel reduced 5.8%

compared to the HHO+B10 fuel. Maximum reduction of CO emissions of 38% using H₂+B10 fuel was obtained compared to the neat diesel fuel at 2300 rpm.

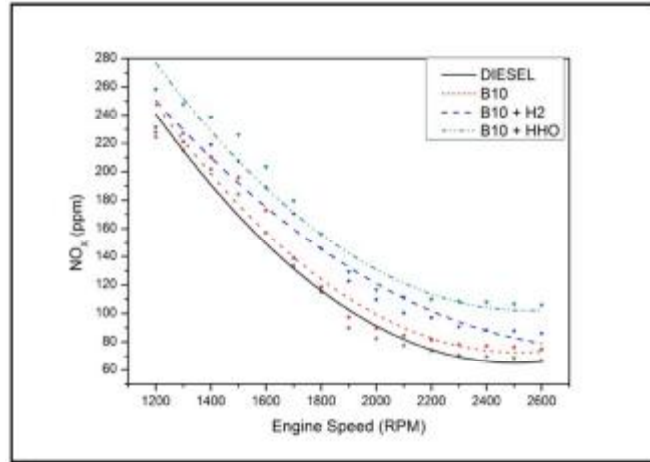


Fig. 9: NO_x versus engine speed.

NO_x is formed during the combustion because of three factors, high temperature, oxygen concentration, and residence time. If these three factors present in a combustion chamber, the NO_x formation is more [7]. When hydrogen is inducted, the enhanced combustion rate increases the temperature and thus the NO emission. This is the main environmental problem with hydrogen induction [28]. Beside, higher oxygen content of B10 fuel increases the combustion efficiency and the NO_x emissions. As an expected result, higher in-cylinder temperature and oxygen content of B10 are increased NO_x emissions that presented in figure 9. Average increment of NO_x emissions were 20% and 16% for HHO+B10 and H₂+B10 fuels respectively.

CONCLUSIONS

Main point of this experimental study was to investigate the effect of pure hydrogen, hydroxy (HHO) gas and biodiesel addition to the diesel fuel in diesel engines. Experimental results are summarized and simply presented in table 5. Intake air is enriched with 10 L/min of H₂ and HHO before intake manifold without modification on the test engine. Effects of HHO+B10 and H₂+B10 fuels on engine performance and exhaust emissions compared with neat diesel fuel results are presented in table 5.

Table 5: Comparison of experimental results HHO+B10 versus H₂+B10.

	Output	Preference (HHO+B10/H ₂ +B10)	Effect (%)
Emissions	CO (ppm)	H ₂ +B10	5.80 less
	CO ₂ (%)	H ₂ +B10	8.72 less
	NO _x (ppm)	H ₂ +B10	9.70 less
Performance	Brake Power (kW)	HHO+B10	4.33 more
	Brake Torque (Nm)	HHO+B10	2.15 more
	BSFC (g/kWh)	HHO+B10	7.60 less

On the basis of the experimental investigation on diesel engine, conclusions can be drawn below;

- Ø Enrichment of intake air with pure hydrogen or hydroxy gas gives promising results compared to neat diesel fuel according to the experiments.
- Ø All other performance and exhaust emissions have improved except NO_x exhaust emissions.
- Ø According to neat diesel fuel, reductions of CO emission were 29% and 22% via H₂+B10 and HHO+B10 usage respectively.
- Ø Similar to the CO results, both of the hydroxy gas and hydrogen enriched fuels have provided better CO₂ exhaust emissions than neat diesel fuel. Lesser CO₂ exhaust emissions resultshavebeen measured with H₂+B10 fuel usage compared to the HHO+B10 and neat diesel fuels (8.72% and 22.3% respectively).
- Ø On the other hand, HHO+B10 fuel has performed higher engine performance results compared to the H₂+B10 and neat diesel. While brake power and torque outputs have improved and BSFC values have decreased which has inverse ratio with fuel efficiency.
- Ø As an overall result, HHO+B10 usage in diesel engines is preferable because of easy and safety assembles to commercial transportation vehicles. Additionally low operation and consumables cost according to H₂+B10.

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NOMENCLATURE

HHO Hydroxy, Oxy-Hydrogen
B10 10% biodiesel included diesel fuel
ICE Internal Combustion Engine
KOH Potassium hydroxide
Consumption
PWM Pulse Width Modulation

DI Direct injection
LPM Liter per Minute
RPM Revolution per minute
BSFC Brake Specific Fuel
H₂ Pure hydrogen

Subscripts

l Liquid

g Gas

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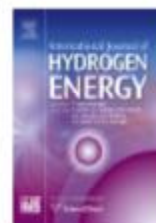
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Experimental investigation of using 30HCNG fuel mixture on a non-modified diesel engine operated with various diesel replacement rates

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ABSTRACT

Pilot diesel injection phenomenon consisted by gaseous fuels such as hydrogen and natural gas; ensure the way of exert fuels without any structural modification in compression ignition engines. This study investigates of using 30HCNG (Hydrogenated Compressed Natural Gas) fuel mixture on a non-modified diesel engine operated with diesel fuel replacement rate experimentally. HCNG was used as an additional fuel with air fumigation in a non-modified diesel engine via intake manifold and reduction of diesel fuel by dint of pilot injection. Reduction of diesel fuel was prepared as two different substitutions like 25 and 50% diesel replacement rates controlled by stepping motor devices. 30% Hydrogen and 70% CNG (Compressed Natural Gas) (by volumes) fuel mixture were tested in a 3.6 L, four cylinders, four stroke and water coolant diesel test engine between 1200 and 2600 rpm. Effects of these two different diesel reductions on engine performance and exhaust emissions parameters were compared and illustrated with graphics as brake power, brake torque, and brake specific fuel consumption; CO, CO₂ and NO_x. Brake thermal efficiency and a minor cost analyses were also performed. It was observed that, 30HCNG fuel mixture with 50% diesel replacement rate affect the engine performance as brake torque, brake power and BSFC values were decreased 4.3%, 8%, 26.8% respectively. Exhaust emission values of NO_x and CO₂ were decreased 47.2%, 16.7% respectively; however CO emission values were increased 8%.

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1- Introduction

Fuel research and developments are always possessing importance as much as engine technologies at automotive industry. Common objective is to create more economical and environmental friendly future for both. Some of the important alternative fuels are gaseous fuels, such as hydrogen and natural gas which have rising momentousness of utilization. Especially degradation of exhaust emissions and fuel consumptions can be obtained by using these gaseous fuels in internal combustion engines. Gaseous fuels can be used as a fuel for SI (spark ignition), CI (compression ignition) and NG (natural gas) engines. For SI and NG engines these fuels combustions are easier than CI engines. In CI engine combustion state occurs with penetration of diesel fuel to the pressurized air in combustion conditions. Natural gas and hydrogen is favorable couple for using as a fuel in ICE (Internal Combustion Engine)'s. Natural gas has environmental advantages and reserve capacity. Hydrogen has also environmental advantages and significant power inside. Their engineering and combustion properties are very suitable when they become a mixture in certain percentages. Researchers have focused on this point and they mentioned; the optimum usage of mixture percentages as a fuel between 18% - 35% by volumes of these gases [1-5, 15, 16]. Basically, a singular usage effect of NG is reducing emission parameters while causing losses in torque and power. However H_2 is considerable improving power while increasing NO_x exhaust emissions. When they mixed up with each other their disadvantages becomes their advantages. Previous studies showed that it can be more selectable for choosing these mixtures rather than singular usage. Pichayapat K. et al, mentioned that using combinations of hydrogen and diesel mixing together or natural gas and diesel mixing together and even HCNG (Hydrogenated Compressed Natural Gas) blended together in the internal combustion engine will reduce emission levels and provide better overall performances [3].

In CI engines, dual-fuel mode operation gives promotional utilize when use gaseous fuels. In dual-fuel operations mostly two options have been preferred to induct gases into the combustion chamber; generally called fumigation. First one is directly mixed with air and straight the mixture (air and fuel) via intake manifold. The second one is mixed in two fuels in an injector that used gas and diesel [6]. At the first option any kind of structural modification is not required on the engine expect set up to pilot injection system for combustion. Minimum diesel fuel injected towards of air and gas fuel mixtures in combustion chamber. On the other hand; original injectors of engine has to exchange with the new ones for second option. Additionally; in transportation sector, diesel engines recompose like a gases engine with some structural engine modifications. These modifications of diesel engines to gas engines, causes necessary changes of parts with higher costs. Beside, this transformation in CI engines, allowed the system only mono-fuel. Obviating these problems, diesel engines combust with pilot diesel injection by using gaseous fuels without any structural modification.

Cheolwoong Park et al., touched upon and explain the importance of HCNG fuel that, “An HCNG engine exhibits a leaner combustion than a CNG (Compressed Natural Gas) engine, which is possible because of the stable lean combustion characteristics of hydrogen. Consequently, nitrogen oxides (NO_x) and CO₂ emissions can be significantly reduced.” [7, 8]. From this point of view, some important previous studies of HCNG fuel experiments can be listed below. Arat H.T. et al, prepared a critical review about HCNG fuel usage in diesel engines. They focused on the usage of HCNG fuel, details on the different mixture formation strategies and fuel properties, analyze their engine performance and their emissions characteristics, and look through the benefits and challenges of HCNG for diesel engines [1]. Papagiannakis R.G., et al., studied NG usage on diesel engines. They examined their tests on a single cylinder, direct injection CI engine with dual fuel, NG and diesel, which operated pilot diesel injection. Test results were worthwhile for exhaust emissions, especially for NO_x results [9, 10]. Cheolwoong P. et al., studied on 11 L, 6 cylinder CNG engine with addition of hydrogen as a supplementary fuel. They mentioned the volumetric ratio of hydrogen to natural gas was maintained at 3:7, which was considered the most appropriate ratio based on their researches. Experiments results showed that proposed operating strategy for satisfying the limit of the EURO-VI standard is effectively in specific spark timing condition [7, 8]. Another study was reached by Fanhu M. et al, studied on performance and emission parameters on a hydrogen-enriched compressed natural gas engine with various hydrogen quantities. They mentioned emission parameters reduce with optimum usages however increasing the hydrogen quantity occur NO_x increase [11]. From the point of view, using hydrogen and/or natural gas as an additive to enhance the conventional diesel engine performance and by the way reducing the exhaust emissions have been investigated by several researchers and the outcomes are very promising. Some of the remarkable studies can be refereed with for HCNG and CNG usage in CI [2, 11-20]. All together touched upon; the important factors of H₂NG vehicles, improving the engine performance (torque, power, brake thermal efficiency, total engine efficiency improvement, lean combustion parameters like higher equivalence ratio and excess air, degradation the engine acoustic, etc.) decreasing the exhaust emission outputs (THC, CO, CO₂ decreases with both and also CNG balanced the unwanted hydrogen’s NO_x) and reducing the specific fuel consumption and fuel economy. In this experimental study, 30HCNG fuel mixtures were used in a non-modified diesel engine and operated with 25 and 50% diesel replacement rates. HCNG was used as a replacement fuel in a non-modified diesel engine via intake manifold and reduction of diesel fuel was obtained by dint of pilot injection. Reduction of diesel fuel was prepared as two different substitutions like 25 and 50% diesel replacement rates and controlled by stepping motor devices. Engine performance and exhaust emissions were plotted and explained in details. Also brake thermal efficiency and minor cost analysis of fuel consumption were performed.

2- Experimental Procedures

2.1 Experimental Setup

All tests are carried out on a 3.6 L, non-modified, commercial, four cylinders, four-stroke, naturally aspirated, water-cooled direct injection compression ignition engine. Test engine has 17.5:1 compression ratio and mechanical direct injection fuel pump with injection starts at 23 deg BTDC (Before Top Dead Center). It should be noted here, the usage of gaseous fuels in a fumigated diesel engine tented to operation under lean limits [18]. In this experiment, combustion operation is under lean limits which calculated results for relative AFR is $\lambda=1.35$; equivalence ratio is $\phi=0.74$ and air-fuel ratio is, AFR=21.04. By the way, the stoichiometric AFR of 30HCNG is 16.4 [24]. A Netfren mark hydraulic dynamometer (specifications are presented by [16]) was used for loading the test engine and MRU Delta 1600V gas analyzer was used to measure exhaust gas emissions. Diesel fuel consumption has measured with liquid fuel flow meter which addressed the data to main computer. Additional gas fuels flow parameters controlled and measured with needle vanes and ALICAT mark flow meters respectively. Test engine specifications and accuracies of measurement devices were exposed in Table 1.

Hydrogen and CNG fuels supplied from pressured tanks which has gas capacity of 12.5 m³ and fixed on 200 bar pressure for CNG and 170 bar for hydrogen. A back fire eliminator and CNG check valve used behind the checkout for hinders the reverse flow through the tanks. Regulators were used for reducing the pressures of hydrogen and CNG. Hydrogen regulator that adapted after back fire eliminator regulated the high pressure to atmospheric pressure. Lovato brand CNG regulator kit has 2- stage regulation which throttled in first stage 200 bar to 12-14 bar and the second stage decreased the 12-14 bar to 1-0 bar. Exit from regulators, Hydrogen and CNG pass through the needle valve and flow meters. After controlled and measured the flow of gaseous fuels with needle valves and flow meters, measurement data were collected through the data logger. Air quantity was also gauged with conversation of the air velocity meter which was adapted on air filter. In front of the intake manifold, a mixture chamber was used for the mixing hydrogen, CNG and air. After combustion MRU Delta 1600 V gas analyzer was used to measure exhaust gas emissions. Experimental set-up is illustrated in Figure 1.

2.2. Fuel properties and Determination of Mixture

In this study; diesel fuel, hydrogen and CNG were used as test fuels. CNG used in this study contains 97.372% Methane, 2.327% Nitrogen, 0.230% Ethane and 0.068% Propane. Methane is the main constituent of the CNG used that; its C/H ratio is also low, resulting in a significant reduction of the specific CO₂ emissions [10]. Hydrogen purity is 99.999%. Diesel fuel is a typical commercial fuel. Fuel properties and contents of mixture proportions under the present experimental investigation are given in Table 2.

Using gas fuels in CI engine and their mixture ratios are very important items in combustion of pilot injection diesel fuel. As we mentioned before, between 18%-35% (vol/vol) ratios of hydrogen to CNG are used for several experiments. In this experiment study, 30HCNG fuel mixtures were used to analyze the engine performance and emission parameters with diesel replacement rates like pilot

injection. The volumetric hydrogen content is calculated according to equation 1 [24]. In this experimental study, gas fuel mixtures, used in CI engine with substitution of diesel fuel, are termed by HCNG₁ and HCNG₂ and one test average mass flow rates also given for H₂ and CNG. 30% Hydrogen and 70% CNG gas fuel ratio was not changed in the fuel mixture except diesel fuel ratio. Termed fuels and contents of mixture proportions, and the fuel definitions and quantities were illustrated in Table 2 and Table 3, respectively.

$$H_{2(vol\ \%)} = \frac{V_{H_2}}{V_{H_2} + V_{CNG}} \quad \text{Eq.1}$$

2.3. Substitution of Diesel Fuel

One of the research experimental goals was to use hydrogen and CNG gas fuel mixtures in an original diesel engine without any modifications. For this reason, substitution of diesel fuel was adjusted with the help of stepping motor and devices. A bipolar, 1.8 deg/step, 6.2 Nm holding torque stepping motor, equipped with Toshiba TA 8435 stepping motor driver was used for adjustment of the mechanical fuel pump plunger pin. Stepping motor driver controlled with software in main computer and obtained more sensitive measurement results. Several optimization experiments were tried for substitution of diesel fuel. As a result, reducing the diesel fuel injection was handled via twisting the plunger pin one tour (360°) and two tours (720°) of fuel pump plunger pin clockwise. In the normal diesel operation conditions, experiments showed that one tour twisted fuel pump plunger pin procured approximately 20-25% reduction of fuel injection. And two tours twisted procured approximately 30-45% reduction in fuel consumption [15]. Engine drivability problem has been observed with more than three and higher twisted rotations. The test engine has a mechanical fuel pump and injection system, therefore more choking of the diesel fuel caused pressure drops in the injection system. Hence fuel droplet size increased with a fuel pressure drop. The self-ignition of the fuel in the combustion delayed and this caused unstable engine running.

Pilot diesel injection phenomenon summarized by Papagiannakis R.G. et al “In dual fuel CI engines operating with gaseous fuels as primary fuel and a ‘pilot’ amount of liquid diesel fuel as an ignition source, the gaseous fuel is inducted along with the intake air and is compressed like in a conventional diesel engine. The mixture of air and gaseous fuel does not auto ignite due to its high autoignition temperature. A small amount of liquid diesel fuel is injected near the end of the compression stroke to ignite the gaseous mixture” [10]. Also some researchers mentioned this small quantity of pilot diesel fuel is between 2-10% by mass [12]. Similar approach with these literatures, 25 and 50% diesel replacement rates were used for substitution of diesel fuel.

3- Experimental Results

The engine was operated for enough time with diesel fuel to reach the operation temperature before the tests. All tests were implemented with the engine speed between 1200 to 2600 rpm with an interval of 100 rpm at WOT and full load

condition. Separately three sets of experiments were performed and average test data were used for the results for optimization of the test results. The engine was operated with normal diesel fuel, HCNG₁ and HCNG₂. Engine performance and exhaust emission parameters were measured for all tests and compared each other.

3.1. Performance Characteristics

Performance of 30HCNG fuel mixtures on a non-modified diesel engine with substitution of diesel fuel like pilot diesel injection is discussed with respect to brake torque, brake power and brake specific fuel consumption parameters. These parameters are illustrated in following figures.

Figure 2 illustrates the brake torque of the engine as a function of engine speed of diesel fuel, HCNG₁ and HCNG₂ fuel mixtures. Gaseous fuel mixtures were played a relatively negative role and reduced the torque and power. It is highly probable due to the volumetric efficiency, gas inlet temperature, gas mixture distribution, AFR, net heat release, cylinder pressure and lack of chemical energy conversion to mechanical energy. These criteria occurs incomplete combustion which redirect the researches minimizing this incident. In this study, when brake torque values of fuels were compared, HCNG₁ and HCNG₂ fuels reduced the torque parameters about 6.2% and 4.3% respectively. Although replacement rate of diesel fuel for HCNG₁ is higher than HCNG₂, the hydrogen quantity of HCNG₂ mixture occur more powerful combustion.

Figure 3 demonstrated the break power versus engine speed of diesel fuel, HCNG₁ and HCNG₂. Similarly engine torque values, pilot diesel fuel injection with gas fuel mixtures decreased the brake power of engine using HCNG₁ and HCNG₂ with the ratio of 9% and 8%. Additionally in lean burning period of combustion, HCNG₂ power values were better than HCNG₁ values caused of more hydrogen concentration inducted.

Figure 4 shows the total brake specific fuel consumption (BSFC) variation with engine speed. The BSFC is directly calculated from the ratio between the sum of diesel fuel mass flow rates and the engine brake power output. It should be noticed that; in this study, BSFC performed from liquid diesel engine fuel replacement rates and does not include the gaseous fuel. It can be observed that, 30HCNG fuel variances with substitution of diesel fuel like pilot diesel injection, decreased the BSFC and forms more lucrative fuel consumption. Improvement of fuel consumption in this experimental test engine is observed as 18% and 26.8% for HCNG₁ and HCNG₂ fuel mixtures, respectively. One of the major intentions of this study was reducing BSFC while minimum torque and power loses. Additionally gaseous fuels consumption and effects of total BSFC have been mentioned in 4th section in cost analyses part.

Due to the high laminar velocity of combustion and the wide flammability limits of hydrogen, its addition is utilized as a method to shift stable engine operation to leaner mixtures, with positive effects on emissions and thermal efficiency. The use of

hydrogen as an additive was also considered as a solution to enhance the combustion rates of some fuels like methane [15]. The high RON and low lean-flammability limit of hydrogen and natural gas provide the necessary elements to attain high thermal efficiencies in an ICE [25]. η_{bth} thermal efficiency is an important performance indicator in engine performance. In formula form of η_{bth} can be written as equation 2;

$$\eta_{bth} = \frac{\dot{W}_b}{\dot{m}_d * LHV_d + \dot{m}_{CNG} * LHV_{CNG} + \dot{m}_H * LHV_H}$$

Eq. 2

where η_{bth} ; is the brake thermal efficiency; $\dot{W}_b$ is brake power; $\dot{m}_d, \dot{m}_{CNG}, \dot{m}_H$ is the mass flow rates of substituted diesel, CNG and HHO respectively; and LHV_d, LHV_{CNG}, LHV_H is the lower heating values of fuels.

Figure 5 shows the variations in the brake thermal efficiency (η_{bth}) as a function of engine speed for the cases of using diesel fuel, HCNG₁ and HCNG₂ fuel mixtures. In results, the average of η_{bth} reduces with 3% and 3.6%, respectively for HCNG₂ and HCNG₁ when compared with neat diesel operation. This can be explained by the fact that the high diffusion and flame speeds of hydrogen help improve the charge homogeneity and tended to increase the efficiency, hence the amount of replacement rate of liquid fuel and quantities of CNG affected the outputs slightly negative. As an additional note; from the characteristics nature of hydrogen, it can be clearly said that; the high flame speed of hydrogen can greatly compensate the low flame speed of CNG. For this reason, open-literature referred that; the η_{bth} for HCNG operation is substantially higher with respect to neat CNG operation [24].

3.2. Emission Characteristics

In ICE's, hydrogen and natural gas components have significant environmentally improve when it comes to exhaust emission parameters. For single usage of hydrogen in a diesel engine, due to high flammability limits and increasing the cycle by cycle variation, nitrogen oxides (NO_x) values were increasing. Therewithal only CNG or NG usage in diesel engines, due to lower flammability limits of methane and high C/H ratio, CO parameters were increasing undesirable values. Eliminating of this downbeat situation, HCNG was preferred as a favorable fuel mixture for decreasing the emissions. NO_x emissions were thrilled by some factors like; oxygen concentration, combustion temperature and reacting time. CNG balanced the hydrogen's faster combustion and make more stable performance [25] and emission outputs especially in lower and middle engine speeds.

NO_x emissions of diesel fuel, HCNG₁ and HCNG₂ fuel mixtures values are plotted versus engine speed in Figure 6. 30HCNG fuel mixtures were decreased the NO_x emissions worthwhile. It was observed that HCNG₁ and HCNG₂ fuel mixtures improved the NO_x emissions with 28% and 47.2%, respectively. Surely, pilot diesel injection assisted this reductions. Also these results are similar and better condition of previous studies [1, 2, 3, 5, 10].

Carbon dioxide (CO₂) concentrations in the atmosphere have been increasing significantly over the past century, compared to the rather steady level of the pre-industrial era [14]. Although it is a product of complete combustion, CO₂ is a major source of air pollution. It can be clearly seen in Figure 7, in dual fuel mode with replacement diesel injection, HCNG₁ and HCNG₂ fuel mixtures decreased the CO₂ values with 11.4% and 16.7%, respectively when compared with the diesel fuel in this experimental study.

Figure 8 shows the plot of carbon monoxide (CO) emissions versus engine speed for diesel fuel, HCNG₁ and HCNG₂ fuel mixtures. It is well known that the rate of CO formation is a function of the available amount of unburned gaseous fuel as well as the mixture temperature, both of which control the rate of fuel decomposition and oxidation [15]. In this experiment, it is obvious that CO emissions under gaseous dual fuel operation are significantly higher. HCNG₁ was increased the CO emission 12% and HCNG₂ was increased the CO emission 8%. The main reason for the CO emissions increment could be result of incomplete combustion of CNG (especially less hydrogen mixture quantity which seen in HCNG₁) and amount of pilot diesel fuel ignition on HCNG mixture. One of the solutions for solving this problem is the amount of diesel fuel for pilot ignition can be minimized for the reduction of CO emissions.

4- Cost Analyses

As an engineering approach, being only environmentally friendly is not enough. Economical point of this study is briefly explained in this subsection. Equation 1 is presented below by considering current fuel prices for calculation. Instead of converting diesel engine to natural gas engine, which is expensive and needs time consumption, limited modifications on diesel engine is easier and cheaper. Additionally, converted engines are not suitable to work with diesel fuel anymore. Comparison of these two options is illustrated in Table 4.

National current fuel prices (*in date: 20.09.2015*) are \$1.5 for 1 liter of diesel fuel, \$1 and \$2 for 1m³ of CNG and H₂, respectively. Volumetric flow rates of gaseous fuels are measured from flow meters multiplied by duration of experiment (in unit of minute), meanwhile mass flow rate of liquid diesel fuel is measured and transferred to the computer software simultaneously. Substitution of diesel fuel provided %45 reduction on liquid fuel consumption.

$$\left[\frac{V_{diesel} \times \$1.5}{\left\{ (V_{subs.diesel} \times \$1.5) + \left(\left(\dot{V}_{CNG} \times \frac{\$1}{1000} \right) + \left(\dot{V}_{H_2} \times \frac{\$2}{1000} \right) \right) \times ETP \left(\frac{1}{60} \right) \right\}} - 1 \right] \times 100 = A (\%)$$

(1)

A: Percentage of saving in cash

V_{diesel} : Total amount of diesel fuel consumption during the experiment with original set up of fuel pump (m/r).

$V_{subs.diesel}$: Total amount of substituted diesel fuel consumption during the experiment after fuel pump set up (m/r).

$\dot{V}_{CNG}$: Volumetric flow rate of CNG fuel measured by flow meter during experiment (L/min).

$\dot{V}_{H_2}$: Volumetric flow rate of H_2 fuel measured by flow meter during experiment (L/min).

ETP : Experimental Time Period (second).

An explanatory example;

To get a consistent idea, Equation 1 is applied on experimental engine output data which also includes measured gaseous fuel during the test.

$\dot{V}_{diesel}$: 0.2 L, $\dot{V}_{subs.diesel}$: 0.165 L, $\dot{V}_{CNG}$: 19 L/min, $\dot{V}_{H_2}$: 8 L/min, ETP : 100 sec.

$$\left[\frac{0.2 \times \$1.5}{\left\{ (0.11 \times \$1.5) + \left(\left(19 \times \frac{\$1}{1000} \right) + \left(8 \times \frac{\$2}{1000} \right) \right) \times 100 \left(\frac{1}{60} \right) \right\}} - 1 \right] \times 100 = 34\%$$

If a commercial truck is considered which has 30 L/100 km fuel consumption and travels around minimum of 120,000 km/year, fuel cost becomes around \$54,000 (36,000 (L) x 1.5 (\$/L)). Additionally, even investment cost and constant operating costs included, this experimental usage is extremely advised to protect environment and micro-macro economies. It must be noticed two important assumptions that; firstly, this is a micro financial analysis which does not includes necessary gas stations or infrastructure preparations. Secondly, calculations of vehicle fuel consumption such as this approachment, needed to evaluate with transient tests (i.e. world harmonized transient cycle) which is achieved more accomplished results than the steady test data.

5- Conclusion

In this present work, an experimental investigation has been conducted to examine the effect of the performance and pollutant emissions of a non-modified diesel engine operated with liquid diesel replacement rates by using 30HCNG fuel mixtures. Mechanical diesel fuel pump pin was mounted with a stepping motor for substitution of diesel fuel and two different situations were tested. 25% and 50% replacement rates of diesel were feeding with 30HCNG mixture. The cost analyses, methodology, experimental set-up and measurement devices were also given in details. The major conclusions can be drawn on the basis of experimental results;

- ü HCNG₂ fuel has significantly preferable fuel mixture rather than HCNG₁ due to engine performance outputs. Although pilot diesel fuel amount was lower

than the other, higher hydrogen contents of mixture has improved the results when compared between HCNG₂ and HCNG₁.

- ü Comparison with diesel operation; both HCNG₂ and HCNG₁ mixtures decreased the brake torque outputs 4.3% and 6.2 % while brake power outputs decreased 8% and 9% respectively.
- ü Brake thermal efficiency follows the similar effect with brake power and decreased 3 and 3.6% for HCNG₂ and HCNG₁ fuel mixtures, respectively when compared the normal diesel operation.
- ü BSFC results provided 18% and 26.8% improvement for HCNG₁ and HCNG₂ fuel mixtures, respectively.
- ü Cost analysis and economical approach of this experimental study is calculated and supported with an example. Results showed that in case of using non-modified engine with diesel replacement rates, supplied with 30HCNG is approximately 30% profitable than base diesel engine. Meanwhile, it should be remembered that, further vehicle fuel consumption analyses evaluate with additional approaches like WHTC.
- ü Very promising improvements obtained for NO_x and CO₂ emissions. Both of the fuel mixtures decreased the NO_x amounts 28% for HCNG₁ and 47.2% for HCNG₂ respectively. While reduction of CO₂ emissions were 11.4% and 16.7% respectively.
- ü CO emissions under gaseous dual fuel operation are significantly higher than diesel fuel. Both of the fuel mixtures increased the CO emissions 12% and 8% respectively.

Overall, it can be concluded that, 30HCNG fuel mixtures usage on a non-modified commercial diesel engine has promising improvements for BSFC, CO₂ and NO_x outputs. Especially for this reason, HCNG blends are candidate to be piquant alternative fuel for internal combustion engines.

Acknowledgement

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Table Captions:

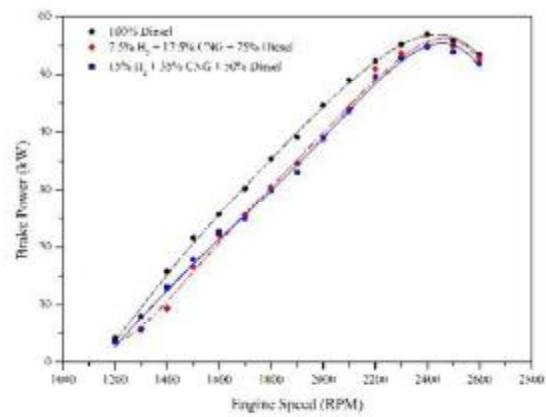
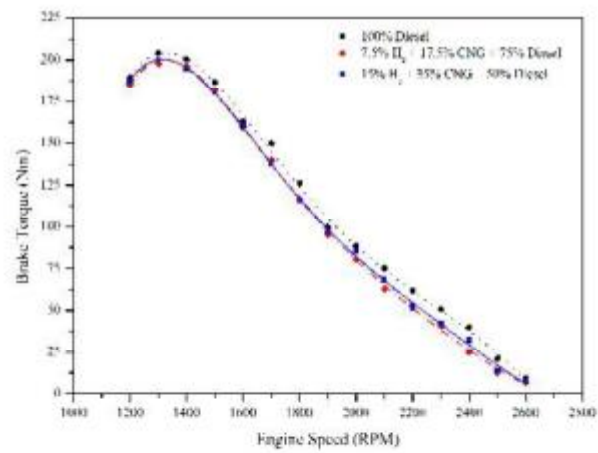
- Table 1: Test Engine Specifications and Accuracies of Measurement Devices
- Table 2: Fuel Properties and contents of percentages mixtures
- Table 3: The fuel definitions and quantities
- Table 4: Approximate investment cost of converting modified diesel engine and present work

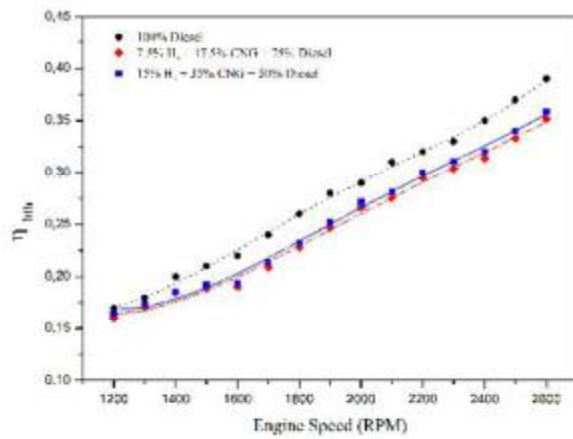
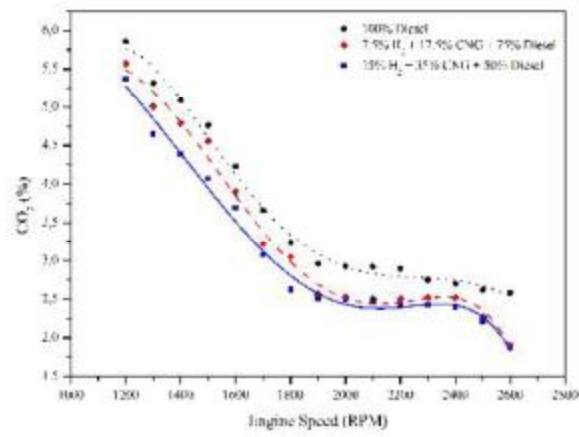
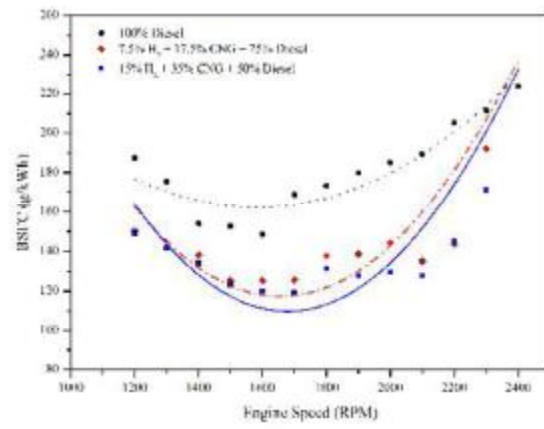
Figure Captions:Figure 1- Experimental Setup

- Figure 2-Brake torque vs engine speed of HCNG₁ and HCNG₂
- Figure 3- Brake power vs engine speed of HCNG₁ and HCNG₂
- Figure 4- BSFC vs engine speed of HCNG₁ and HCNG₂
- Figure 5- BTE vs engine speed of HCNG₁ and HCNG₂
- Figure 6- NO_x vs engine speed of HCNG₁ and HCNG₂
- Figure 7- CO₂ vs engine speed of HCNG₁ and HCNG₂
- Figure 8- CO vs engine speed of HCNG₁ and HCNG₂



1- Engine 2- Dynamometer 3- Exhaust gas analyzer 4- Data logger 5- CNG line 6- Hydrogen tank 7- Stepping motor on fuel pump plunger 8- Stepping motor driver 9- CNG regulator 10- Flow meters 11- Needle valves 12- Gas fuels lines 13- Mixture chamber 14- Air filter 15- MAF 16- Fuel Flowmeter 17- Dynamometer Control Unit 18- Main Computer





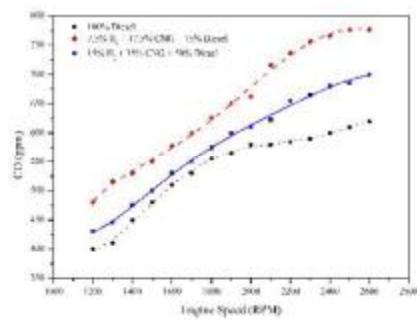
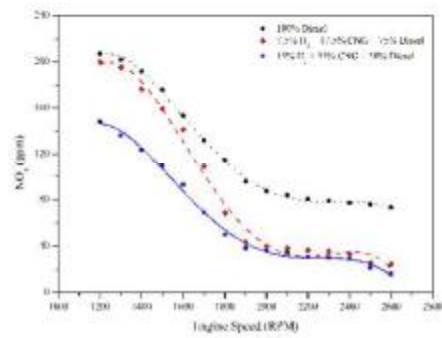


Table 1. Test Engine Specifications and Accuracies of Measurement Devices

Brand	Mitsubishi Canter	Performance	Alicat Flow Meters	MRU Delta 1600 v
Model	4D32 - In line 4	Accuracy at calibration conditions	± 0.8%	-
Type	D.I. diesel with glow plug	High Accuracy at calibration conditions	± 0.6%	-
Displacement	3567cc	Repeatability	± 0.2%	-
Bore	104mm	Typical Response Time	10 ms	-
Stroke	105mm	Measuring range-CO ₂ - 0-22%	-	OIML-class 1
Power	89kW @ 3200rpm	Measuring range-NO _x -(0-4000 ppm)	-	OIML-class 1
Torque	230Nm @ 1350rpm	Measuring range-CO - 0-10%	-	OIML-class 1
Hyd. Dynam. Torque/Speed Range	0-1700 Nm / 0-7500 rpm	Lambda	-	0-9.99

Table 2. Fuel Properties and contents of percentages mixture

Property	Diesel	Methane	Hydrogen	
Molar Mass (kg/kmol)	~200	16.04	2.02	1 Diesel
Density (kg/m ³)	840(l)	0.72(g)	0.082(g)	
Stoichiometric air/fuel weight	14.7	17.2	34.3	2 0.75 Diesel 75/245 25/105
Auto-ignition Temperature °C	254-285	595	585	
Laminar Flame Speed (cm/s)	128	42	230	3 0.5 Diesel 50/250 50/105 50/105
Lower Heating Value (MJ/kg)	42.61	47.14	120.21	
Higher Heating Value (MJ/kg)	45.58	52.23	142.18	

Table 3. The fuel definitions and quantities

Fuel	Definition	Liquid Fuel Flow rate*	Gas Fuel Flow rates*	Mass %**	Energy %**	LHV**
Diesel	Non substituted, standard diesel oper.	2.17 g/sec	-	100	100	42.6 MJ/kg
HCNG ₁	7.5% H ₂ +17.5% CNG+75% Diesel	1.53 g/sec	8.2 l/min, 19.57 l/min	4.9	11.2	52.2 MJ/kg
HCNG ₂	15% H ₂ +35% CNG+50% Diesel	1.11 g/sec	8.2 l/min, 19.57 l/min	95.1	88.8	52.2 MJ/kg

* General one set test average (90 sec.) ** Adapted from Munshi S. R. [23]

Table 3. Approximate investment cost of converting modified diesel engine and present work

Options	Changes/Additions	Approximate investment cost
Converting diesel engine to NG	- Modification on cylinder head and piston - Reduce compression ratio - Incorporate a spark plug - Gas fuel tanks, piping systems, ECU and calibrations and necessary equipment.	\$12.000
Limited modifications on engine	- Mixing chamber before intake manifold - Set up the fuel pump - Gas fuel tanks, piping systems, ECU and calibrations and necessary equipment.	\$4000

PAPER III

Optimizing the quantity of diesel fuel injection by using 25HHOCNG gas fuel mixture

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Keywords:Hydroxy (HHO), CNG, Fuel injection optimization, performance, emissions.

Abstract.Injection behaviors of internal combustion engines are very substantial fact that provides developments to future strategies about optimizing the engine and fuel parameters. During the combustion process, pilot diesel injection technique is more preferable option while using alternative gas fuels in a diesel engine. In this experimental study, a 3.6 L commercial, four stroke, four cylinders and mechanical fuel pump non-modified diesel test engine operated with hydroxy (HHO) and compressed natural gas (CNG) fuel mixtures under 25% and 75% (vol/vol), respectively. Diesel fuel injection quantities were reduced with the help of steeping motor devices which mounted on mechanical fuel pump plunger pin. Sensitive removes of steeping motor, plunger pin twisted clockwise 360°, 720° and 1080°, respectively. Comparisons of engine performance and exhaust emissions were explained briefly and illustrated via graphs. As a result, 720° clockwise twisted pin is the optimum point for experimental fuel pump plunger while using 25HHOCNG fuel mixtures.

Introduction

In Internal Combustion Engines (ICEs), injection behaviors play an influential and important role that affects the combustion, engine performance and exhaust emissions. Alternative fuels have been getting more attention as concerns escalate over exhaust pollutant emissions produced by ICEs, higher fuel costs, and the depletion of crude oil. Among ICE's applications, one of the most promising options is the diesel derivative dual fuel engine with gaseous fuels as the supplement fuel [1]. Additionally for optimizing the engine performance and exhaust emission parameters; with the ability to reduce oxides of nitrogen (NOx) emissions, carbon dioxide (CO₂) emissions, and particulate matter (PM) emissions, dual-fuel operation is environmentally viable [2]. An effective method for controlling the start of diesel combustion is through pilot fuel injection that named other words, dual fuel combustion. Dual fuel engines are conventional diesel engines converted to operate

on a gaseous fuel while using liquid diesel fuel injection to provide mainly ignition. These engines which tend to have common features with both spark-ignition and diesel engines. There are three different kinds of dual fuel engines according to the means of the induction of the gaseous fuels: 1) Common type of dual fuel engines with pilot injection into homogeneous charge gaseous fuel-air mixture; 2) Dual fuel engines with the timed port injection; 3) Dual fuel engines with both pilot and gaseous fuels injection into high pressure air [3]. The minimum pilot quantities may be different for different combustion chamber structures and operating conditions. Conventional diesel fuel injection systems limit the minimum pilot fuel quantity to not less than 5% of the full load fuel quantity when operating on diesel alone [4]. However, in practical dual fuel engines, the diesel flow is usually not reduced below 10 or 15 percent of the total energy required - in part to maintain the fuel injection system for diesel engine operation and to ensure adequate injection nozzle cooling [3.] Miyake-et al [5] suggested that, "natural gas and pilot liquid diesel fuel were mixed prior to injection and found that 40% pilot diesel fuel was needed for stable ignition". It would be ideal to simply replace diesel fuel with CNG in Compression Ignition (CI) engines, removing the need for significant investment in new trucks and rail freight; however, due to the auto-ignition characteristics of CNG, it cannot be directly substituted without heating the intake because of its relatively high auto-ignition temperatures [6]. Thus, in order to use CNG as a diesel substitute in existing CI engines, dual fuel operation must be employed where the CNG is drawn into the engine through the intake port or directly injected into the combustion chamber and a small diesel pilot, which is injected at the correct timing, is used to initiate the combustion process. This can drastically decrease the overall amount of diesel used, while improving fuel economy through a more homogeneous (and constant-volume like) combustion process [6]. Alternative gas fuels have occurred the same situation either CNG in using CI as a supplementary fuel. Alternative fuels inducted the CI combustion has different situational phenomena. The gas-air mixture is introduced into the cylinder during the intake stroke. Diesel fuel is directly injected into the cylinder during the compression stroke and compression-ignites which in turn ignites the surrounding gas-air mixture. After the period between diesel pilot injection and ignition known as ignition delay, the diesel pilot releases energy and initiates the combustion process. Then, the surrounding gas-air mixture ignites. Finally, the combustion process continues by flame propagation through the remaining lean gas-air mixture [7]. Hydrogen and HHO (oxy-hydrogen, hydroxy) gases are more attractable alternative fuels in ICEs. Alike CNG, these fuels can be combustible in CI engine with dual-fuel mode cause of CI combustion nature. Some researchers, like [1,2,3,4,5,6,7,8,9] related to the use of environmentally friendly fuels like CNG and HHO in diesel engines is seen in intensive way with exertion.

In this experimental study, three different quantities of pilot injection at specific engine operational conditions are determined, and the effectiveness of different measures to improve engine performance and exhaust emissions for various operational conditions are examined. Gaseous fuel mixture affected some effective results for non-modified diesel engine at dual fuel combustion with pilot injection into diesel combustion both of engine performance and exhaust emissions. 25HHOCNG gas fuel mixture operated in a non-modified diesel engine by

substitution of diesel fuel with helped of steeping motor and pilot diesel injection quantities effects are examined. All results and experimental setup is illustrated and explained briefly.

Experimental set-up

Test engine and fuels properties

3.6 L, non-modified, commercial, four cylinders, four-stroke, naturally aspirated, water-cooled direct injection compression ignition engine is used in all test which is placed in Cukurova University Automotive Engineering Laboratories in TURKEY. A Netfren brand hydraulic dynamometer was used for loading the test engine and MRU Delta 1600V gas analyzer was used to measure exhaust gas emissions. Data outputs of dynamometer and gas analyzer were collected via Elimko PR-100 data logger and were transferred to main computer. Table 1 presented the technical specifications of test engine.

Table 1. Test Engine Specifications

Brand	Mitsubishi Canter
Model	4D32 - In line 4
Type	D.I. diesel with glow plug
Displacement	3567cc
Bore x Stroke	104mm x105 mm
Power	89kW @ 3200rpm
Torque	295Nm @ 1800rpm

In this experimental study; CNG and HHO are used as gaseous fuels and commercial diesel is used as a liquid fuel. Fuel properties can be reachable author's previous studies [8]. In this study, CNG is supplied by cryogenic tank under 200 bar pressure which contain 97.372% Methane, 2.327% Nitrogen, 0.230 Ethane and 0.068 Propane. In tests, behind the CNG bottles exist, a check-valve is used for prevent back-flow. After the check-valve, CNG is transferred to regulator in an ex-proof pipeline and that regulator has 2-stage regulation system. 200 bar CNG pressure firstly reduced 12-15 bar in a first stage and secondly regulated to the 0-1 bar for attendance the air-gas mixture chamber. CNG heating-cooling systems are rebuilt using the engine cooling system thought radiator input and output. The quantity of CNG gas flow is controlled by a needle vane after the regulator. CNG gas is entered to ALICAT flow meter for the measurement of the gas flow after needle vane.

HHO system is electrolysis of distilled water with using suitable catalyst. In this experimental study, KOH (potassium hydroxide) is chosen as a catalyser. System contains double parallel connected dry cells, water reservoir, bubbler, solenoid relay, constant current PWM (Pulse Width Modulation), fittings and electrical wires. HHO generator runs with engine battery (24 V), 15-20 amperage and 500 hertz frequency. When hydroxy gas is generated, gas pours itself into the needle vane and flow meter. For this experiment; (for a single test average) measurement from flow meters, the total average of hydroxy gas was adjusted to 3.04 LPM (Liter Per Minute) and the total average of CNG gas was adjusted to 9.104 LPM during the experiments, and

these values indicates give approximate percentages of 25% Hydroxy gas and 75% CNG gas mixture by volume. Additionally this value has a similarity to literature for optimum usage (10-30% HCNG (Hydrogenated Compressed Natural Gas)) of these gas fuels.

Determination of substituted diesel fuel as pilot diesel injection

Optimizing the substituted diesel fuel quantities is the major goal of this study. For this reason substitution of diesel fuel was obtained with the help of stepping motor and devices which mounted on mechanical diesel fuel pump plunger. A bipolar, 1.8 deg/step, 6.2 Nm holding torque stepping motor adjusted to plunger and controlled with Toshiba TA 8435 stepping motor driver. It operated with software in main computer and obtained more sensitive measurement results. Optimizing the quantities of diesel fuel, three clockwise rotations were selected.

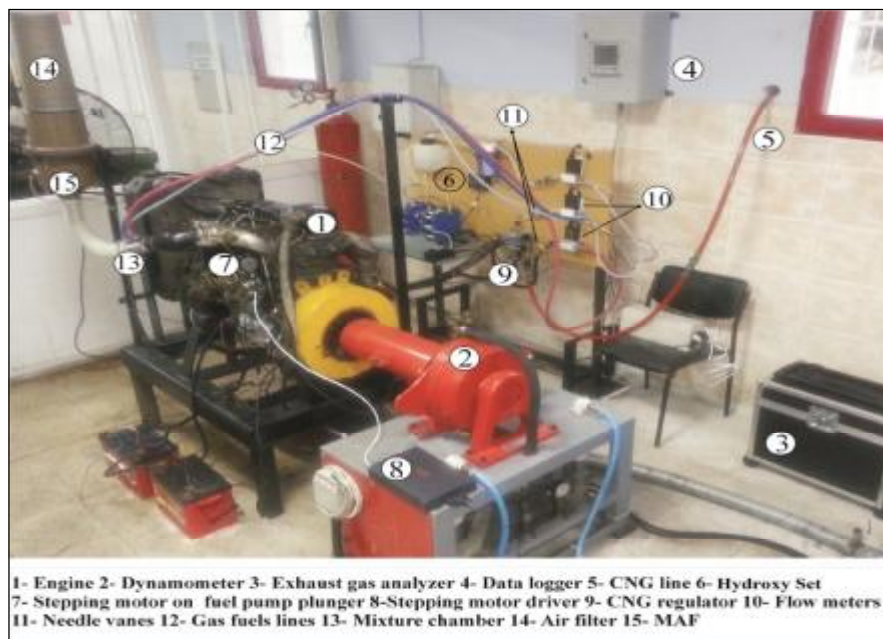


Figure1.Experimental set-up

Pilot diesel injections occurred with twisting the plunger pin one tour (360°), two tours (720°) and three tours (1080°) of fuel pump plunger pin. Additionally middle degrees of rotations were examined with 540°, 900° and 1440° but half rotation of plunger, 180°, did not show a noticeable decrease in this mechanical fuel pump for using pilot diesel injection quantity. Because of this it was not found necessary for presented of these data. As we mentioned previously, minimum pilot quantities changeable in diesel engines for different combustion chamber structures and operating conditions. All test apparatus and measurement devices of experimental setup illustrated in figure 1.

Before the tests, the engine was operated for enough time with diesel fuel to reach the operation temperature. Test fuels were tested from 1300 to 2600 rpm with an

interval of 100 rpm at full load condition. For optimization of the test results, separately three sets of experiments were performed and average test data were used for the results. The engine was operated with normal diesel fuel and 25HHOCNG gas fuel mixture with various substituted diesel fuel quantities as pilot diesel injection systematic.

Result and Discussion

Engine Performance

One of the main handicaps of gaseous fuels usage in diesel engines, created negative effects in performance parameters. Especially for the CNG fuel used in diesel engines, exhaust gas emissions decreased and affected positively, but in performance up to 15% reduction, CNG was required using in dual-fuel system with hydrogen-enrichment. HCNG improves the diesel performance nearly 10% better than CNG with diesel. In this study HHO is the major player for amelioration the performance of diesel test engine. HHO contains an extra oxygen atom compared to hydrogen that provides the engine more powerful by mixture of air-gas fuels and oxygen before intake manifold. As a general one set average (60 sec), diesel substitution observed with instantaneous fuel consumption for diesel, 360°, 720° and 1080° is 1.75 g/sec, 1.34 g/sec, 1.118 g/sec and 1.59 g/sec respectively. In other word substituted diesel fuel affected like pilot diesel injection reductions handled with 360°, 720° and 1080° is 21%, 30% and 18% respectively.

Fig. 2 presented the Brake Torque as a function of Engine speed. In figure it can be clearly seen that, effect of addition HHO to CNG improves the torque values relative to diesel operation. 25HHOCNG mixture improved the torque values versus diesel engine can be reformed for 360°, 720° and 1080° is 10%, 15% and 13% respectively. Major reason for these improvements is HHO addition into the cylinders. Molar concentration of oxygen in air-gas mixtures occurred more flexible combustion and hydrogen atoms prevented the combustion phases faster. With this situation, CNG has playing critical role for the combustion and due to this faster combustion, CNG combust more willingly. Especially middle and high engine speeds, this factor can be seen expressly.

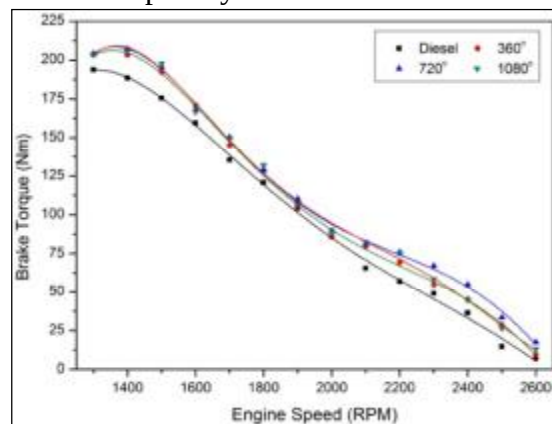


Fig. 2. Brake torque vs engine speed of diesel and various rotations of pilot diesel injection substitution

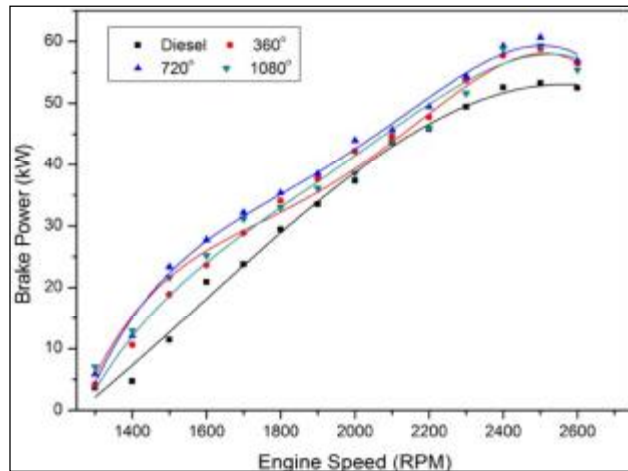


Fig. 3. Brake power vs engine speed of diesel and various rotations of pilot diesel injection substitution

Brake Power as a function of Engine speed graphic is shown in Fig. 3. In figure, similarly brake torque values, 13%, 19% and 15% increases presented for 360°, 720° and 1080° power values compared with neat diesel. Once again, gas fuels combustion properties affected the power development. Brake Specific Fuel Consumption (BSFC) versus Engine speed curve illustrated in Fig. 4. One of the major goals of this study, while improving performance values, decreasing the BSFC values with pilot diesel injection parameters helped of substituting diesel fuel amount. In figure it can be clearly seen that, 25HHOCNG gas mixture with substituted diesel fuel, affected positively in BSFC and forms more lucrative fuel consumption. Decreasing values can be listed for 360°, 720° and 1080° compared to neat diesel is 21%, 29% and 15% respectively.

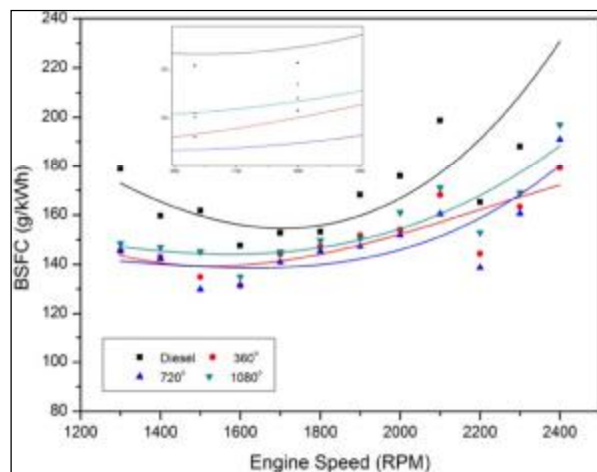


Fig. 4. BSFC vs engine speed of diesel and various rotations of pilot diesel injection substitution

Exhaust Emissions

A precise control of the fuel delivery is needed for increasing stringency of the emission standards for CI engine. Not only to meet the standards for NO_x , PM and other air pollutions, but also to reduce CO_2 emissions with a minimum penalty, in fuel economy and cost [10].

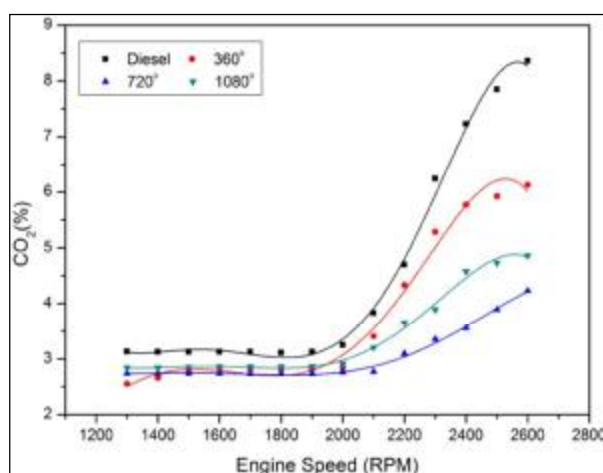


Fig.5. CO_2 vs engine speed of diesel and various rotations of pilot diesel injection substitution

When using alternative gas fuels in a non-modified diesel engine, it is the most important unforgettable thing that; while these gas fuels improve the engine performance, they have to balanced the harmful exhaust emissions values. CO_2 emission curves as a function of engine speed graphic illustrated in Figure 5. It is clearly seen that, 720° clockwise twisted pin position has placed the best CO_2 values. 25HHOCNG mixture with pilot diesel injection has decreased the CO_2 values significantly either neat diesel operation. When we compared the diesel and substituted pilot quantities operated with gas fuels; 360° , 720° and 1080° generated the CO_2 values with; 26%, 32% and 24% respectively. If HHO used singularly in diesel engine as a additive fuel, it increased the NO_x emissions because of higher flammability limits and rapid combustion willing. It is also known that, oxygen concentration, reacting time and in-cylinder gas temperature affected occurring NO_x formation. When increasing the hydrogen content, NO_x emission values are also increased. For all that, CNG plays an important role for reducing NO_x emission values. The amount of hydroxy gas and CNG ratio in the mixture is also affected by this role. Fig. 6 shows NO_x emissions between engine speed of diesel and 360° , 720° and 1080° pilot diesel quantities run under 25HHOCNG mixture. NO_x emission values of mixture fuel are approximately decreased 20%, 21% and 1.5% when compared to normal diesel fuel with 360° , 720° and 1080° respectively.

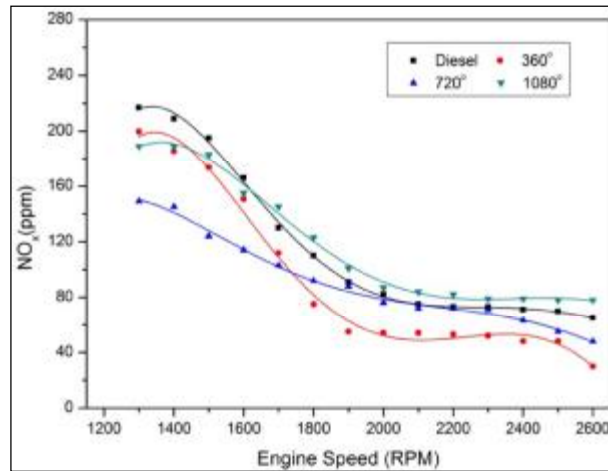


Fig. 6. NO_x vs engine speed of diesel and various rotations of pilot diesel injection substitution

Fig. 7. illustrates CO emissions as a function of engine speed for normal diesel operation and 25HHOCNG mixture with various amounts of pilot diesel injection. CO emission values were decreased soupcon when compared with normal diesel fuel operation. With numbers; CO emission values were decreased 1.05%, 1.17% and 0.55% for 360°, 720° and 1080°, relative to neat diesel operation. The reason for this very little reducing of CO emissions could be result of gas fuel mixtures nature, combustion discontinuity, faster combustion and oxygen concentrations in the cylinders. One of the solutions for these problems is the amount of diesel fuel for pilot ignition, which can be more minimised for the reduction of CO emissions.

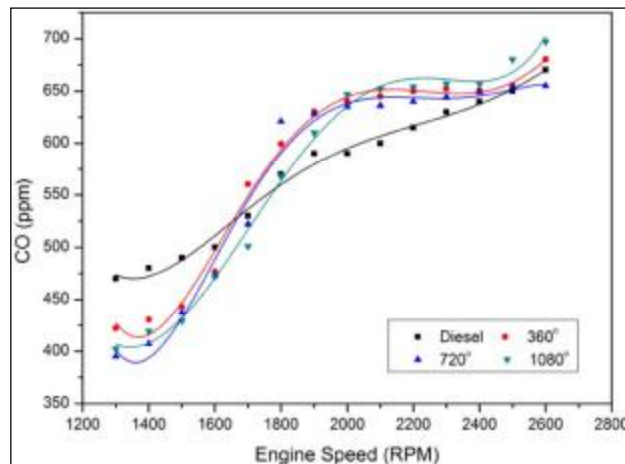


Fig. 7. CO vs Engine speed of diesel and various rotations of pilot diesel injection substitution

Conclusion

In this experimental study, a 3.6 L commercial, four stroke, four cylinders and mechanical fuel pump non-modified diesel test engine operated with hydroxy (HHO) and compressed natural gas (CNG) fuel mixtures under 25% and 75% (vol/vol), respectively. Diesel fuel injection quantities were reduced with the help of steeping

motor devices which mounted on mechanical fuel pump plunger pin. Sensitive removes of steeping motor, plunger pin twisted clockwise 360°, 720° and 1080°, respectively. Experimental procedure and measurement devices were also explained briefly. The major goal of this study, while improving combustion values (not only performance also exhaust emissions), decreasing the BSFC values with pilot diesel injection parameters helped of optimizing and substituting diesel fuel amount. As a general result, for this test engine and these studies equipments, 720° clockwise twisted pin is the optimum point for experimental fuel pump plunger while using 25HHOCNG fuel mixtures. 720° point affected the performance improvingly as brake torque and power with 15%, 19% respectively and in BSFC, this point of pilot injection decreased the results up to 29%. On the other hand, 720° point generated and developed the exhaust emissions positively with run under with 25HHOCNG fuel mixtures. 32%, 21% and 1.17% reducing values presented for CO₂, NO_x and CO respectively.

Acknowledgements

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