

**Effect of Intake Air Preheating on Performance and Emission
Characteristics of Compression Ignition Engine Fueled with Diesel-
Biodiesel Blends with Ethanol addition**



By

Temesgen Ashenafi Beyene

A Thesis Submitted to the Department of Mechanical Systems and Vehicle
Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirements for the award of the
Degree of Masters of Science in Automotive engineering

Office of Graduate Studies

Adama Science and Technology University

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Major Advisor: Dr. Menelik Walle

Co-Advisor: Mr. Mekonnen Liben

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Approval of Examiners

We, the undersigned, members of the Board of Examiners of the final open defense by Temesgen Ashenafi Beyene have read and evaluated his thesis entitled “**Effect of Intake Air Preheating on Performance and Emission Characteristics of Compression Ignition Engine Fueled with Diesel-Biodiesel Blends with Ethanol addition**” and examined the candidate. This is therefore to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of Master of Science in automotive engineering.

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DECLARATION

I hereby declare that this MSc thesis entitled “**Effect of Intake Air Preheating on Performance and Emission Characteristics of Compression Ignition Engine Fueled with Diesel-Biodiesel Blends with Ethanol addition**” in partial fulfillment of the requirements for the award of the Degree of Masters of Science in Automotive engineering is my original work carried out from February 2020 to January 2021 under the supervision of Dr. Menelik Walle, Vehicle Engineering Department, Defense Engineering college and Mr. Mekonnen Liben, Department of Mechanical System and Vehicle Engineering, Adama Science and Technology University, Ethiopia. It has not been presented for a degree in any other university, and all sources of material used for this thesis have been duly acknowledged.

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DEDICATION

I dedicate this research to my Love Mitin Alemayehu, my beloved mother Molalish Andualem and all my brothers and sisters. Also, I am pleased to memorialize my research in loving memory of my father, Ashenafi Beyene, who died in December 2005 E C.

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NOMENCLATURE

A	Heat transfer area
ASTM	American Standards for Testing
BSFC	Brake specific fuel consumption
BTE	Brake thermal efficiency
°C	Degree Centigrade
CI	Compression ignition
CN	Cetane number
CO	Carbon monoxide
CO ₂	Carbon dioxide
CR	Compression ratio
De	The equivalent diameter of shell
Di	Tube inner diameter
do	External diameter of tubes
EN	European Union
F _t	Friction factor
gm	milligram
H	Convective heat transfer coefficient
HC	Hydrocarbon
HV	Heating value of fuel
IP	Injection pressure

IT	Injection temperature
K	Thermal conductivity
kg/kW-h	kilogram per kilowatt hour
KOH	Potassium hydroxide
kW	kilo watt
L	Length of heat tube
ml	milliliter
Ms	Mass velocity of shell
N	Number of tubes
NO ₂	Nitrogen dioxide
NO _x	Oxides of Nitrogen
Nu	Nusselt number
P _b	Brake power
Pcs	Pieces
PM	particulate matter
ppm	Parts per million
Pr	Prandtl Number
Q	Heat transfer rate
Re	Reynolds number
rpm	Revolution per minute
T	Temperature

TDC	Top dead center
U	Overall heat transfer coefficient
Vol.	Volume
Vs.	Verses
wopa	Without preheated air
wpa	With preheated air
ρ	Density
μ	Viscosity
Δp	Pressure drop
C_p	Specific heat of fluid
$\dot{m}_c$	Mass flow rate
$(\Delta T_{LM})_{CC}$	Log mean temperature difference
%	Percentage

ABSTRACT

Lately, renewable biofuels could be considered leading candidates for softening society's liability to fossil derivative fuels and pollutants emission issues. In this study taking advantages of compression ignition engine better performance, diesel-cotton seed biodiesel blends with ethanol addition dawn as alternative commercial fuel in substitution to diesel fuel without the need of geometrical engine modifications. This work is aimed at an experimental investigation of the performance and emission characteristics of a diesel engine running on fuel concentrations of diesel (70%)-biodiesel (20%)-ethanol (10%), diesel (60%)-biodiesel (30%)-ethanol (10%), diesel (50%)- biodiesel (40%)- ethanol (10%), and diesel (100%) at two elevated intake air temperatures of 25°C and 51°C. The experiment was conducted on a single-cylinder, four-stroke, air-cooled and naturally aspirated direct injection diesel engine (TBMC3-02) with a maximum power output of 2.2kW by varying engine load at a constant speed of 1750rpm. The baseline test of a standard diesel fuel was used in the engine at the ambient air temperature. While running ethanol in a diesel engine, it is essential to preheat the intake air or the alcohols itself in order to sufficiently vaporize the fuel. In this regards, shell and tube counter flow heat exchanger is manufactured and used for preheating the intake air using engine exhaust gas as source, making the system more sustainable since no external energy is required. Performance and emissions of the diesel engine are compared based on diesel-biodiesel blends with ethanol addition, and standard diesel. An increase in the air intake temperature caused variations in the ignition delay period of the diesel-biodiesel blends with ethanol addition by improving the vaporization characteristic of ethanol, and provides a better combustion. The experimental result reveals that an increase in the air intake temperature results in an average increment in brake power, BTE, CO₂ and NO_x with B30E10 fuel of 2.78%, 4.1%, 22.14% and 8.68%. In addition, an average reduction in BSFC, CO and HC of 9.52%,36.84% and 13.23% was observed. Furthermore, an increase in the relative amount of biodiesel with diesel-biodiesel blends with ethanol addition showed that a slight decrease in brake power, medium increase in BTE and CO₂ however, there is a slight increment in BSFC, NO_x, CO and HC as well. The preheated intake air shows an effective result when compared to the ambient intake air.

Keywords: Biodiesel, Intake air preheating, Diesel engine, Performance, emission

CHAPTER ONE

INTRODUCTION

1.1 Background of the Study

Increasing population, the modernization in the world and increasing industrialization has significantly increased the total consumption of energy. Fossil fuels supply more than 80% energy for global energy consumption (Akhilendra et al., 2018). Globally, the transportation sector is the second largest energy consuming sector after the industrial sector and accounts for 30% of the world's total delivered energy, of which 80% is road transport. It is believed that this sector is currently responsible for nearly 60% of world oil demand and will be the strongest growing energy demand sector in the future (Nations, 2016). Currently, more than 99% of the transport sector is powered by diesel engines (Gautam, 2018). However, increased usage of diesel products results in a drop of oil reserves, global warming, increase in the world's average temperature and lessening of oil reserves. Additionally, emissions from diesel engine lead to an adverse effect on human health and the environment and are the primary cause of greenhouse gases (Doru et al., 2016).

But the applications of diesel engines are more as they have higher thermal efficiency and lower cost. Diesel engines are mostly employed in the field of transportation owing to higher fuel efficiency and greater power output compared to the other fuel engines (Sonsoles et al., 2017). Even though diesel engine exhaust has less carbon monoxide (CO) and hydrocarbon (HC) emissions than gasoline engine exhaust, it contains considerable amount of particulate matter (PM) and oxides of nitrogen (NOx) (Ibrahim et al., 2015). Additionally, the drawback of the CI engine is the high concentration of harmful components of its exhaust gases, being more expensive to eliminate, since the use of more advanced exhaust after treatment system (EATS) is a requirement to remove particulate matter (PM) and NOx. Moreover, the EATS used for CI engines tend to be more expensive than a regular three-way catalyst and have a limited life span (Addy and Hannu, 2019). Dwindling fossil fuels and continuously increasing demand for diesel fuel are the reason behind significant research being carried out in the direction of alternative fuel or new combustion systems in the combustion engine. The strict norms of fuel quality and emission also push researchers towards alternative fuel. Using alternative fuels, such as Hydrogen and Alcohol Fuels and derived from Biomass and Municipal Solid Waste (MSW) in internal combustion engines has the potential to reduce the dependency on petroleum fuels. Considerable attention has been paid in the development

of alternative fuel sources in various countries, with emphasis on biomass that possess the added advantage of being renewable (Jilin et al., 2011; Nadir et al., 2012 and Imran et al., 2013). Alternative fuel from biomass produces less emission and provides clean combustion (Ali and Mustafa, 2011 and Alan et al., 2005). Biomass is defined as any organic matter that is available on a renewable or recurring basis (Balat, 2009). It includes all plants and plant-derived materials, including agricultural crops and trees, wood and wood residues, grasses, aquatic plants, animal manure, municipal residues, and other residue materials (Singh and Gu, 2010). A variety of fuels can be produced from biomass resources including liquid fuels, such as biodiesel, ethanol, and methanol, and gaseous fuels, such as hydrogen and methane (Venghaus and Hoffmann, 2016). These biofuels could partly replace fossil-based fuels, reduce toxic emissions, and more importantly restrain the life-cycle emission of CO₂ (Zhu et al., 2010). Additionally, biofuels made from agricultural products (oxygenated by nature) reduce the world's dependence on oil imports, support local agricultural industries and enhance farming incomes and, moreover, offer benefits in terms of usually reduced exhaust emissions. Among those, vegetable oils, their derived biodiesels (methyl or ethyl esters) and bio-alcohols are considered as very promising fuels.

Biodiesel, chemically known as alkyl ester of fatty acids from different plant and animal sources. Biodiesel is a renewable and sustainable fuel that can be produced from any kind of edible and non-edible vegetable oils, waste vegetable oils, animal fats etc. Due to its organic nature, biodiesel has 10-11% oxygen in its structure that allows widespread oxidation of fuel, resulting in more complete combustion (Sureshkumar et al., 2008 and Abhishek et al., 2014). Due to its higher cetane number it has good self-ignition characteristics, which aids in its combustion and allows achieving similar values of thermal efficiency in comparison with engine powered by diesel fuel (Arkadiusz et al., 2017). Biodiesel also reduces the formation of CO, HC, and soot particles in engine (Murari et al. 2013; Cengiz and Sehmus, 2009 and Nurun Nabi et al., 2009,). Biodiesel also decreases the lifecycle emission of CO₂ (Mohammad et al., 2014). However, the NO_x emission from the engine with biodiesel is found to be higher (Kyunghyun, 2013 and Deepesh et al., 2014). This increased NO_x emission from biodiesel fueled engine can be countered by incorporating ethanol into the fuel blend.

Ethanol is the type of alcohol made from renewable resources such as biomass from locally grown crops and even waste products such as waste paper, grass, and tree trimmings. They are also an alternative transportation fuel since it has properties allowing its use in existing

diesel engines with minor hardware modification. Despite this, they are only used on a limited basis to fuel diesel engines due to their respective problems in toxicity, miscibility with water, or immiscibility with diesel. However, they have higher octane number enduring higher compression ratios before engine starts knocking, thus giving diesel engine an ability to deliver more power efficiently and economically and to produce lesser CO, HC, and NO_x/NO₂ with higher heat of vaporization, thereby reducing peak temperature inside combustion chamber (Cheung et al., 2008 and Agarwal., 2007). The use of ethanol on a limited basis in diesel-biodiesel blends can compensate the decreased octane caused by the presence of biodiesel in diesel fuel and produce lesser particulates and CO₂ due to the lower carbon-to-hydrogen ratio of these alcohols. On the one hand, the presence of biodiesel can act as stabilizer to stabilize the phase stability of diesel-biodiesel-ethanol in blend fuels.

Among alcohol fuels, ethanol and methanol have been widely reviewed and studied as additives to or blend fuels with diesel (Chang and Paul, 2019 and Sundar et al.2017). Some of the disadvantages of alcohol fuels are low lubricity, difficulty of vaporization and high auto-ignition temperature. It was also reported that a minimum in-cylinder initial temperature of 1100 K at TDC is require for the ignition time to be sufficiently short for methanol and ethanol (Gautam et al., 2006). Addition of 2% castor oil and preheat of intake air temperature could be used to improve lubricity and ignition characteristics of these fuels. It was shown that for bus applications where there is much stopping and starting, significant unburned fuel will pass through the engine and some of it will wash down into the crankcase oil which causes dilution of fuel and loss of lubricity. As a consequence of this, there is a chance of wear at metal to metal contacts (Arumugam and Sriram, 2012). Thus, heating of the intake air was suggested.

It is quite evident that the usage of ethanol in diesel–biodiesel blends has been a recent research area for partially replacing conventional fossil diesel fuel with biodiesel and ethanol. The oil-importing countries including Ethiopia aim to replace conventional fossil fuel with renewable sources of energy such as biodiesel and ethanol. And, a brief survey of the literature led to the finding that minimal work has been carried out using cotton seed biodiesel-diesel blends with addition of ethanol, as an alternate fuel, and preheated intake air, in a diesel engine. Thus, the objective and novelty of the current work is to perform experiments to investigate the effect of intake air preheating on performance and emission characteristics of diesel engine fueled with diesel-biodiesel blends with ethanol addition for a varying engine load.

1.2 Statement of the Problem

Energy plays an important role in social, economic and ecological advancement and creating superior living environment for human beings. The world energy usage, particularly demand and utilization of fossil fuel (diesel fuel) for diesel engines due to its higher efficiency and greater power output, has been increasing enormously with population and economic growth as well as improving lifestyles. However, fossil fuel resources are continuously depleting and these fixed reserves are very much concentrated in certain region of the world. As a result, countries like Ethiopia which have no these resources are facing a foreign exchange problem, particularly due to importing crude oil. The limited reserves and rising price of fossil fuels combined with the environmental harm created by them have led to a search, test and, implement to find alternative energy sources mainly emphasis on biofuels due to their eco-friendly nature. Biodiesel, an alternative diesel fuel, is made from renewable biological sources such as vegetable oils and animal fats. But, the direct use of biodiesel in a compression ignition engine is almost impossible due to poor fuel injection characteristics. Currently, adequate blending with diesel is widely used as one of the methods for improving the fuel properties (i.e., kinematic viscosity, density, and heating value) of biodiesel. However, to use in diesel engine percentage of biodiesel is limited to 20% due to higher viscosity and density of biodiesel affects engine performance and emission badly. Addition of ethanol, an alternative fuel, can further improve the fuel properties (i.e., kinematic viscosity and density) of diesel-biodiesel blends, but prolongs ignition delay and decrease heating value which can be improved by intake air preheating. Furthermore, addition of ethanol has an advantage of increasing biofuel percentage in substitution of petroleum based fossil fuel and prevent environmental pollution due to conventional fuel. Therefore, this research improves the utilization of biofuel for diesel engines by preheating intake air in addition to adding ethanol into diesel-biodiesel blends.

1.3 Objectives

1.3.1 General Objective

The general objective of this thesis is to investigate the effect of intake air preheating and percentage ratios of biodiesel in biodiesel-diesel blended fuels with addition of ethanol on diesel engine performance and emission parameters under varying engine loads.

1.3.2 Specific Objectives

The specific objectives of this thesis work are:

- To design and fabricate counter flow shell and tube heat exchanger for preheating intake air.
- To produce cotton seed oil biodiesel by transesterification method.
- To characterize the properties of biodiesel and its blends with diesel fuel.
- To examine the effect of preheated intake air and percentage variation of biodiesel on performance and emission of diesel engine.
- To evaluate and analyze the performance and emission characteristics of blended fuels with the baseline diesel.

1.4 Significance of the Study

The significances of the study are as follows:

- It helps in reducing the dependency on petroleum based fuels.
- Optimize the performance and reduce the emissions of diesel engine using preheated intake air and diesel-biodiesel blended fuels with addition of ethanol.
- Show how to use locally available energy to catalyze the all-rounded socioeconomic development process.
- Generally, the result is very important for the future that it will alleviate problems of energy supply while simultaneously helping mitigate greenhouse gases and other atmospheric pollution emissions.

1.5 Delimitation (Scope)

The scope of this thesis is experimentally investigating the effect of intake air preheating and percentage ratios of biodiesel on performance and emission characteristics of diesel engine fueled with diesel-biodiesel blends with addition of ethanol.

1.6 Limitations

While conducting the research various problems have faced like lack of copper tube with diameter of 31mm, and mild steel in the market. So, it was too difficult to wash the produced biodiesel until it gets clean. Finally, the testing engine was not well installed on the ground basement, thereby vibration occurred when it operated, and this affects the performance characteristics.

1.7 Organization of the Study

This thesis research work was organized into five chapters. **Chapter one:** introduces the background of the study, statement of the problem, objectives, significance, scope and limitations. **Chapter two:** discuss extensive literature reviews related to the application of biodiesel in a diesel engine; biodiesel feedstock's, biodiesel production processes and its properties, performance and emission characteristics, and also factors that affect the performance and emission characteristics of biodiesel-diesel blend fuel on diesel engine. **Chapter three:** refers to the experimentation (engine setup and its different measurement and instrumentation devices), experimental design methodology, procedure along with uncertainty analysis. It includes the design and manufacturing process of the heat exchanger and lastly, it includes experimental setup and test procedures. **Chapter four:** discusses about the efficiency of heat exchanger, biodiesel yield, characterization results of biodiesel and blended fuels properties, performance and emission characteristics of diesel engine under different biodiesel percentage and various load conditions at ambient and elevated intake air temperatures. Finally, concluding remarks with key findings of the present investigation with a recommendation and future research scopes have been summarized in **Chapter five**.

CHAPTER TWO

LITRATURE REVIEW

This chapter introduces about biodiesel, its feed stocks, production process, qualities and properties as an alternative fuel for internal combustion engines. It also includes detailed reviews of different journals related to design of heat exchanger device for preheating intake air, influence of intake air temperature, oxygenated additives and diesel- biodiesel blend ratio on performance, combustion and emission characteristic of diesel engine operated with diesel-biodiesel blended fuels.

2.1 Biodiesel as Alternative Fuel

The best way to use vegetable oil as fuel is to convert it to biodiesel. Biodiesel, also known as “mono-alkyl ester” based oxygenated fuel that is the product of the reaction of alcohol, such as methanol or ethanol with oil or fats (triglyceride) by transesterification reaction (Soo-Young, 2011). Biodiesel produced from raw vegetable oil has its properties almost similar to diesel, so that it can be used as an alternate fuel (Meher et al., 2006). Biodiesel can be used directly in engine in pure form or blend with diesel in numerous proportions to provide alternative solution of fuel in diesel engines. In compression ignition engine no engine mechanical part modification is needed while using biodiesel as fuel (Shahabuddin et al., 2013). Several reasons for biodiesel used as alternative fuel are concentrated on reduction of greenhouse gas (GHG) emissions, decrease effect on global climate, sustainable and renewable energy solution and to get more promising alternate fuel supply to encounter the current energy demand. With the aid of biodiesel emissions of CO, unburned HC, CO₂ and particulate matter can be reduced. The parameters of diesel engine performance like brake torque (BT), brake power (BP), brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) are also improved (Digambar et al., 2020). However, the main problem of biodiesel as compared to petro diesel are high cloud and pour point, high density, increased nitrogen oxides emission, high viscosity, less volatility, lesser energy content and poor spray characteristics (Digambar et al., 2019).

Biodiesel contains no petroleum products, but it is compatible with conventional diesel fuels and can be blended in any proportion levels with petro diesel to create a stable biodiesel blend. The proportion of blending with petroleum diesel is referred as “Bxx”, where xx indicates the amount of biodiesel in the blend. Slight concentration, such as B5, B10, B15 and B20 are suitably referred to as biodiesel blends have been accepted worldwide as

potential renewable alternate fuels to substitute petroleum diesel. This can be used in diesel engine with no major modifications in the engine mechanical parts and without using any fuel additive.

2.2 Biodiesel Feedstock's

Biodiesel can be found from different feed stocks like vegetable, algae, microbial oil and animal fats. Biodiesel obtained from different feed stocks have various purity and composition (Mohmmad et al., 2015). The main steps for the biodiesel production are availability of feed stocks, type of feed stocks, selection of feedstock and classification of feed stocks into edible, non-edible and waste-based origins. Selection of feedstock influence various factors, like purity of biodiesel, cost, composition and yield. Selection of feed stock for biodiesel production is also dependent on regions. Availability and economic aspect of country is mainly considered before selecting feedstock. Canola oil is used as feedstock in Canada and Soybean oil is used as feedstock in Brazil and USA. Coconut and Palm oils are used in Indonesia and Malaysia as biodiesel feedstock and Rapeseed oils used in Italy, Germany, Finland and UK as biodiesel feedstock. Karajan and Jatropha have considered as future feed stocks for biodiesel in India. Among them, Sunflower oil, Rapeseed oil, Soybean oil and Mustard oil have been used previously as biodiesel feed stocks. fBut due to unfavorable result on food plants slowed down their utilization as biodiesel feed stocks. The utilization of edible oils as biodiesel feed stocks have big issue because this directly affect food chain (Digambar,2020 and Digambar et al., 2020).

Choosing the proper feedstock is a vital subject to ensure low production cost of biodiesel. As much as possible the feedstock of biodiesel should satisfy two requirements for production of biodiesel which are low production costs and large-scale production. In general, biodiesel feed stocks can be categorized into four generations. These are:

2.2.1 First Generation Biodiesel

First generation biodiesels are produced from the edible feed stocks. These feed stocks are Coconut oil, Rapeseed oil, Olive oil, Corn oil, Palm oil, Rice oil, Soybean oil, Mustard oil etc (Mohmmad et al., 2015). At beginning of biodiesel era use of edible feed stocks for the production of biodiesel is quite popular. Availability of crops and relatively easy conversion procedure are the main benefits of the first-generation feed stocks. The main drawback in use of these feed stocks is limitation in food supply that increase the cost of food products (Digambar et al., 2019). High cost, limited area of cultivation and adaptability to

environmental conditions are also the difficulties for the production of biodiesel from edible feed stocks. These problems forced users to shift on the further alternate sources for the biodiesel production (Muhammad et al., 2012).

2.2.2 Second Generation Biodiesel

Second generation biodiesels are produced from the non-edible feed stocks. These feed stocks are Jatropha oil, Neem oil, cottonseed, Karajan oil, Calophyllum inophyllum oil, argemone Mexicana, Rubber seed oil, Mahua indica oil, linseed, Nagchampa oil, Rubber seed oil etc. Disadvantages of first-generation feedstock fascinate researchers to work on non-edible feed stocks. The main benefits of second-generation biodiesel are less production cost, eco-friendly, eliminates food inequality and less requirement of land for farming (Muhammad et al., 2012). Drawbacks of second-generation fuels is yields of plants, where yield falls for main non-edible plants like Jatropha oil, Jojoba oil and Karajan oil. These feed stocks can cultivate in unimportant lands. That's why it is forced to farm non-edible crops at farming lands; it directly influences economy of society and the food production. To solve the socioeconomic issues of nonedible oil, the researchers paying attention on new alternate solution, which are economically feasible and simply accessible at greater extent (Atabani et al., 2012)

2.2.3 Third Generation Biodiesel

Third generation biodiesel is the biodiesel produced from microalgae and waste oils (Atabani et al., 2012). The major benefits of third generation biodiesel are higher amount of oil percentage, reduced greenhouse effect, elevated growth rate and productivity, minor struggle towards farming land and smaller influence on food supply. The main problems of third generation biodiesel are required huge amount of investment, difficulties in oil extraction, matter of production at larger scale and requirement of sunlight. The main sources for third generation biodiesel are micro algae, animal fat, fish oil, waste cooking oil etc (Michael et al., 2014). All these possible resources of third generation biodiesel beat the issues faced by previous generation feed stocks that affect the food chain and economic feasibility. In case of waste fish oil, waste animal tallow oil, waste oils and used cooking oil are also sources for the third-generation biodiesel. It also reduces load of waste handling plant and reduction of water pollution. At current animal fats like goat, beef, poultry rising and pork as a possible and dependable source for biodiesel production (Teresa et al., 2010).

2.2.4 Fourth Generation Biodiesel

Photobiological solar fuels and electro-fuels are considered in fourth generation of biodiesels. Solar biofuels are produced by conversion of solar energy into biodiesel using raw material, this method of conversion is a new field of research. Raw materials are widely available, inexhaustible and cheap. Synthetic biology is an enabling technology for such a transformation (Ewen et al., 2014). For sustainable development, new-to-nature solutions must be found out, which make synthetic living firms and stylish microorganisms for efficient and direct change of solar energy to fuel. Similarly, a mixture of photovoltaic or inorganic water-splitting catalysts with metabolically engineered microbial fuel development is a rising methodology for efficient development and liquid fuel storage (Digambar et al., 2020).

2.3 Availability of Cotton Seed and Its Oil Composition

The main producer countries of Cotton crop are Europe, China and United States. In 2007, cotton was grown in 90 countries. In 2006/07, the four main producing countries were China, India, USA and Pakistan and accounted for approximately three quarters of world output. If we added Uzbekistan and Brazil, six countries would account for 83% of world cotton production. The leading cotton-growing countries in Africa are Egypt, Tanzania, Chad, Mali, Benin, Burkina Faso etc. Ethiopia's share is total cotton produced in Africa is about 5%. Ethiopia yearly produces around 120,000 tons of cotton (Central statistics Agency). Much of the cotton production in Ethiopia is from small-scale farmers, who cultivate about 39,600 hectares annually. The total area under cotton plantation by the private owned enterprises is 54,000 hectares. The country produces cotton both for the local and export markets. According to the Report on large and Medium Scale Manufacturing and Electricity Survey (2002) of the Central Statistical Authority (CSA). In 2007/08, the country's textile industries used 52,936 tons of locally produced cotton as raw material. In the same year, export of cotton from the Country was 14,360 tons valued at US\$ 19 million. In Ethiopia majority of the cotton cultivation takes place in the Awash Valley, with some cultivation also taking place in Gambela, Humera, and Metema. The cotton farming cycle varies from area to area. Cotton is planted earliest (by mid-April) in the case of Middle Awash, while in Lower Awash planting of cotton takes place in late June. Harvesting of cotton shows less time variation. There are four varieties of cotton that are usually cultivated in Ethiopia (Mid-Awash, Bazen, Hiwot and Semen Omo). In terms of production methods, almost all of the government-owned and private sector plantations adopt standard cultivation practices. Bulk of the

cultivation is in rain fed areas, with only a small part of total cultivated area under irrigation. Pesticides and fertilizers are typically applied whenever required. On the other hand, smallholder cotton farms are rain fed and cotton cultivation is typically organic. The cotton is hand-picked and transported to ginneries (Ethiopian Investment Agency, 2012).

The main species of Cotton plants are *Gossypium hirsutum* and *Gossypium herbaceum*, which are used for the production of Cotton seed oil. The height of cotton plants can reach up to 1.2 m. This oil has numerous kinds of non-glyceride things, like carbohydrates, gossypol, sterols, resins, phospholipids, and connected pigments (Fontaras et al., 2007). Cotton seed oil has density in the range of 917–933 kg/m³ at 15 °C. Heating value and viscosity of Cotton Seed oil are 39.5 MJ/kg and 34.79 mm²/s at 40 °C respectively. Cotton seed have oil in the range of 17–25%. The cotton seed oil has fatty acid, like oleic of 19.2–23.26%, palmitic of 11.67–20.1%, and linoleic acid of 55.2–55.5% (Digambar et al., 2020).

2.4 Biodiesel Production Process

Biodiesel production uses oils from vegetable feed stocks, animal fats, microalgae and waste oils. The yield of biodiesel from oil crops mostly depends on the crop species. The yield of biodiesel produced from oils of third generation feed stocks is less in comparison with oil from first and second-generation feed stocks (Aninidita et al., 2010). The production of biodiesel involves two major steps. The first one is extraction of oil from seeds and the second step is conversion of oil into biodiesel using different major techniques like transesterification, pyrolysis, dilution, micro-emulsification etc.

2.4.1 Pyrolysis

Pyrolysis is one method which is used for biodiesel production in which thermal decomposition of material occur at high temperature in absence of air or in inert atmosphere. The characteristics of biodiesel gained from pyrolysis process have viscosity, flash point, cetane number and pour point less than diesel and calorific value similar to diesel fuel. Biodiesel produced by this method has adequate quantity of sulfur and water, but it has in adequate amount of ash content and residual carbon (George et al., 2005). High installation cost is the major problem of this process. Based on operating parameters pyrolysis method is classifying into three sub categories: (a) flash pyrolysis process: is a high heating rate (> 1000 °C/s) thermal cracking process, (b) conventional pyrolysis process: It takes very less vapour residence time to minimize cracking to provide high liquid yield and (c) fast pyrolysis process: It reduces the yield of bio-oil in comparison to conventional pyrolysis.

2.4.2 Micro-Emulsification

Micro-emulsification is the transparent, stable and isotropic mixture of oil, water and surfactant. It is colloidal dispersion that is thermodynamically stable and it is easy. Cetane improver, alcohol, co-solvent and surfactant are added during preparing micro-emulsion of oils. Production of biodiesel by micro-emulsion with butanol, hexanol and octanol is important when the maximum viscosity is required for fuel. Micro-emulsion method can eliminate the issue of viscosity of vegetable oil. some issues with micro emulsification are less stability, volatility and high viscosity (Veera Gnaneswar et al., 2013).

2.4.3 Dilution

Dilution is the method in which amount of solvent increased in solution by decreasing the amount solute. For dilution of oils ethanol and diesel fuel can be used as solvent. The results from this process are reduction in viscosity and density of oil. If 4% amount of solvent (ethanol to diesel fuel) is added to oil the brake thermal efficiency, brake torque and brake power increases but brake specific fuel consumption decreases. Dilution process is easy but there are some problems related with it like incomplete burning and carbon deposition in engine cylinder (Digambar et al., 2020).

2.4.4 Transesterification

Biodiesel produced from transesterification process have similar properties with petro diesel fuel and due to economic point of view transesterification process is favorable for commercial production. During transesterification process esters and glycerol are formed when triglyceride reacts with alcohol. Figure 2.1 shows the transesterification reaction for biodiesel production.

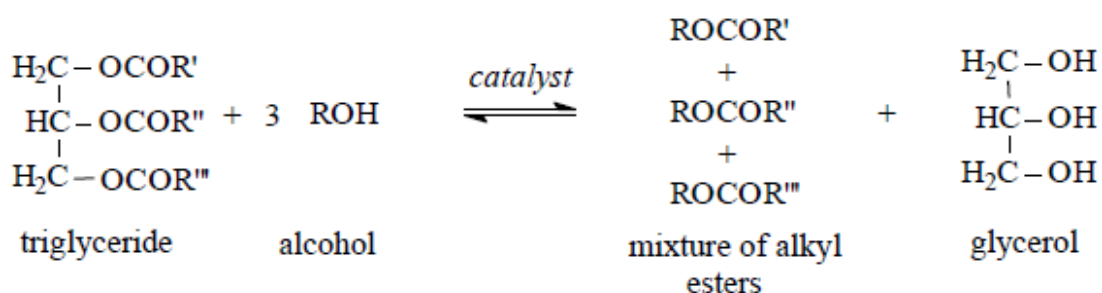


Figure 2.1: Transesterification of vegetable oils (Ulf et al., 1998).

Three fatty acids are connected to triglycerides (organic fats and oils) base and it has molecule of glycerin. Free fatty acids are formed by hydrolyzing triglycerides. After this,

these free fatty acids are reacting with alcohol and formed ester or biodiesel (methyl or ethyl fatty acid ester) and glycerol. Due to reaction of free fatty acid with alcohol transesterification is also known as alcoholysis. The final products of transesterification process are separated, biodiesel settles on the top and due to gravity glycerol settles down. The process of separation should be very fast to avoid reverse process (Fazal et al., 2011). Methanol and ethanol are usually used in transesterification process. If free fatty acids used methanol for reaction the transesterification process is known as methanolysis. In methanolysis process heat is applied to the mixture of methanol (10–20%) and oil (80–90%) and very less quantity of catalyst. Proper mixing of free fatty acids and methanol is essential because solubility of methanol in oil is less. Biodiesel produced from the methanolysis process is fatty acid methyl ester (FAME) (Puneet and Sharma, 2016). For transesterification methanol was preferred because it has higher reactivity and cheaper in price than other alcohols. Free fatty acids used ethanol for reaction the transesterification process is known as ethanolysis. Ethanol is slightly increasing heat content of fuel and cetane number and also ethanol has less toxic content as compared to other alcohol (Masoumeh et al., 2017). There are two main issues associated with ethanolysis, the first one is the reaction required high energy and the other one issue is in separation of ester and glycerol. That's why price of biodiesel produced through ethanolysis is increased (Digambar et al., 2020). The produced biodiesel from the ethanolysis process is fatty acid ethyl ester. Produced biodiesel from transesterification process have many advantages as compare to pure plant oil (PPO). Viscosity is one significant property of fuel is which directly effects engine performance. Pure plant oil has higher viscosity than biodiesel. High viscosity inversely influence the fuel injection time, atomization, and injection pressure for compression ignition engine. Characteristics of biodiesel are quite similar to petro diesel it can be used in diesel engines with small variations (Puneet and Sharma, 2016). Transesterification process is the most adaptive method for commercial biodiesel production because it is most economic process with high biodiesel yield (Digambar,2020).

For the production of biodiesel from transesterification two approaches are followed: (a) Catalytic Transesterification approach: Due to presence of catalyst glycerol and ester is formed when triglyceride reacts with alcohol, this approach is known as catalytic transesterification. The features of triglyceride or fat/oil defined by the nature fatty acids that means the behavior or feature of biodiesel directly influenced by nature of fatty acid (Veera Gnaneswar et al., 2013). Heat is applied to oil in a container to remove water content from

oil. After elimination of water, oil taken inside container and again heat is applied and continuously subjected to stirring. The oil reacts with alcohol in the presence of catalyst and form a mixture at 60 °C temperature of oil, and stirring process is sustained till glycerol separation initiates. After this mixture is taken into a separate vessel and kept it in for 8 hours. After the accomplishment of reaction glycerol settles down slowly due gravity and biodiesel remain top. The separation of glycerol and ester indicate effectiveness of transesterification process. glycerol may be purified for additional usage in cosmetic and pharmaceutical industries (Digambar et al., 2020) and (b) Supercritical Methanol Transesterification approach: In this approach instead of this two-phase mixture, single phase mixture is formed at supercritical state (pressure- 43 MPa, temperature- 340 °C). At these conditions dielectric constant of alcohol reduces. At supercritical state reaction time is very less and it is approximately 2 to 4 minutes. In this process purification of biodiesel is very easy because catalyst is not used. The main drawback of super critical transesterification process high cost of production (Masoumeh et al., 2017).

2.5 Use of Catalysts

Transesterification process is carried out in the presence of catalysts. The catalyst is used to accelerate the reaction rate and to get better quality biodiesel. The vegetable oils, used cooking oil or animal fats are reacted with alcohol in the presence of acid or base catalysts. Selection of both acid and base are made according to the characteristics of feedstock (Meher et al., 2006). The base catalysts are chosen over acid catalysts, due to their requirement of lower reaction temperature, higher conversion efficiency and their capability of completion of reaction at higher speed as compared to acid catalysts. Various types of base catalysts are used for the transesterification of vegetable oils. The most common among these are potassium hydroxide, potassium methoxide, sodium hydroxide and sodium methoxide. Particularly when the acid value of feedstock is high acid catalysts are used. The acid value of edible oils is normally low as compared to non-edible oils. However, the acid value of edible oils also increases when they are used for frying purpose for long time (Ejaz and Younis, 2011).

2.6 Most Important Variables That Influence the Transesterification Reaction

There are different factors that influence the transesterification process of biodiesel productions. The main factors affecting transesterification are reaction time, alcohol-oil

molar ratio, reaction temperature and pressure, catalyst, water content and free fatty acid content.

2.6.1 Reaction Temperature

The rate of reactions is strongly influenced by reaction temperature. However, the reaction is conducted close to the boiling point of methanol (60-70 °C) at atmosphere pressure for specified time. Though alkaline alcoholysis of vegetable oils is normally conducted near boiling point of the alcohol and that reaction can be carried out at room temperature. When the temperature exceeds the boiling point of alcohol, the alcohol will vaporize and form large bubbles which may hinder the reaction and decrease of biodiesel yield because higher reaction temperature accelerates saponification of triglycerides (Aninidita et al., 2010).

2.6.2 Ratio of Alcohol to Oil

Another important variable is the molar ratio of alcohol to vegetable oil that affecting the conversion efficiency, the yield of biodiesel as well as biodiesel production cost. Also, since the stoichiometric molar ratio of alcohol to oil for transesterification is 3:1 and the reaction is reversible, higher molar ratio are required to increase the miscibility and to enhance the contact between the alcohol molecule and triglyceride.

Transesterification is an equilibrium-controlled reaction in which excess of alcohol can be used to get complete conversion and alcohol is easily recoverable. Moreover, increase in the amount of alcohol to oil increase biodiesel yield and biodiesel purity. Therefore, higher alcohol to oil molar ratios gives rise to greater alkyl ester conversion in a shorter time. Molar ratio is associated with type of catalyst used such as; for alkali catalyzed transesterification reaction the alcohol to oil ratio is ranging 5:1 to 12:1 that are most acceptable for maximum conversion to ester. However, when using acid catalyst, the desirable maximum conversion is obtained with sulfuric acid with alcohol to oil molar ratio of 30:1. A molar ratio of 6:1 is normally used in industrial process to obtain methyl esters yields higher than 98%(w/w). The molar ratio of alcohol to oil has no effect on acid value, saponification values and iodine values of esters (Carraretto et al., 2004).

2.6.3 Catalyst Concentration

A catalyst is used to improve the transesterification reaction and amount yield produced. Alkali metal alkoxides are found to be effective transesterification catalyst compared to acidic catalysts. The most common and efficient catalysts for biodiesel production are KOH and NaOH. A concentration in the range of 0.5-1%(w/w) has found to yield 94 to 99%

conversion to vegetable oil esters and further increase in catalyst concentration does not affect the conversion but adds to extra cost, as the catalyst needs to be removed from the reaction mixture after completion of reaction. In consequence, with vegetable oils or fats containing low FFA and water, the base catalyst transesterification is much faster than the acid catalyst transesterification and is most commonly used commercially on the industrial scale (Aninidita et al., 2010).

2.6.4 Mixing Intensity

During the transesterification reaction, the reactants initially form a two-phase liquid system because oils and alcohols are not totally miscible, for this reason dynamic mixing is required to increase the area of contact between the two immiscible phases. Alcoholysis process can be enhanced by the agitation intensity of the reactor. Mass transfer of biodiesel from the oil phase towards the alcohol-oil interface might be a vigorous step that limits rate of alcoholysis reaction because the reaction is heterogeneous. The mixing effect has been found to play a significant role in the slow rate of the reaction. As phase separation ceases, mixing becomes insignificant. The effect of mixing on the kinetics of the transesterification process forms the basis for process scale up and design, therefore, fast mixing accelerates transesterification reactions that variation in mixing strength is expected to change the kinetics of the transesterification reactions (Carraretto et al., 2004).

2.6.5 Effect of Moisture and Free Fatty Acid

Sustainably, impurities in the oil affect the conversion level of crude oil to biodiesel. The free fatty acids and moisture content are the key parameters to determine the viability of vegetable oils to be used for transesterification process. The presence of moisture in the oil could increase the free fatty acids in the oil. To carry out the transesterification reaction, the free fatty acids in the oil must be less than 3%. Higher the acidity of the oil, the smaller is the conversion efficiency. As little as 3% water in the reaction mixture reduced glycerol yield consuming catalyst. High free fatty acid content oils are processed with an immiscible basic glycerol phase as to neutralize the free fatty acids and cause them to pass over the glycerol phase (Bouaid et al., 2016).

2.7 Properties and Qualities of Biodiesel

The properties and qualities of biodiesel must adhere with the international biodiesel standard specifications. These specifications include the American Standards for Testing Materials (ASTM6751-3) or the European Union (EN 14214) Standards for

biodiesel fuel (Digambar et al., 2020). The properties of biodiesel are characterized by physicochemical properties. Some of these properties include; density (kg/m^3), caloric value (MJ/kg), viscosity(mm^2/s), cetane number, pour and cloud points ($^{\circ}\text{C}$), flash point ($^{\circ}\text{C}$), ash content (%), copper corrosion, sulfur content, carbon residue, water content and sediment, distillation range, and oxidation stability. The biodiesel physical and chemical fuel properties is basically depending on the type of feedstock and their fatty acids composition (Atabani et al., 2012).

Sulfur: this method covers the determination of total sulfur in liquid hydrocarbons, boiling in the range approximately 25 to 400°C , with viscosities between approximately 0.2 and $20 \text{ mm}^2/\text{s}$ at room temperature. Even though, biodiesel feedstock typically having very little sulfur, this test is an indicator of contamination of protein material and carryover catalyst material from a production process.

Cloud Point: cloud point is a temperature at which a haze or cloud of wax crystals first appear in a fuel when it is cooled under test condition. The Cloud Point is the most conservative temperature at which components begin to precipitate from the fuel that may cause operability issues. Cloud Point results will vary significantly depending upon the feedstock from which the methyl ester was created and typically higher than the conventional diesel (Yasuaki et al., 2011). It can then be modified in the following ways to get the lower cloud point. These are: The use of additives that retard the formation of solid crystals in the B100 by various mechanisms. The blending feedstock that is relatively high in saturated fatty acids with feedstock that have lower saturated fatty acid content.

Cetane Number: cetane number (CN) is a measure of the ignition quality of diesel fuel during combustion ignition. It provides information about the ignition delay (ID) time of a diesel fuel upon injection into the combustion chamber. High CN implies short ignition delay. The value provides a measure of the ignition characteristics of the fuel in compression ignition engines. The lowest value for typical biodiesel is the same as a “premium petroleum diesel” at 47 (Yasuaki et al., 2011). The CN of biodiesel is generally higher than diesel fuels (Maeda et al., 2011). The CN of biodiesel specified by ASTM D613 is 47 min and EN ISO 5165 is 51 min (Leung et al., 2010).

Kinematic Viscosity: it is the resistance to flow of a fluid under gravity. It is a basic design specification for fuel injectors used in diesel engines. Maximum allowable viscosity, on the other hand, is limited by considerations involved in engine design and size, and the

characteristics of the injection system. The upper limit for the viscosity of biodiesel (6.0 mm²/s at 40°C) is higher than the maximum allowable viscosity in Specification D975 Grade 2-D and 2-D low sulfur (4.1 mm/s at 40°C). Blending biodiesel with diesel fuel close to its upper limit could result in a biodiesel blend with viscosity above the upper limits contained in Specification D975 (Manoj and Mausumi, 2016).

Acid Number: the acid number is a relative indicator of the acidic impurities, degradation and/or oxidation of the fuel. It is a direct measure of free fatty acids in the biodiesel (B100). It is a quantity of base that can be expressed as a milligram of potassium hydroxide per gram of sample, that required for titrate a sample to specified end point. Free fatty acids arise in part through the hydrolysis of the feedstock fats and oils. Improper processing or oxidation may also lead to higher values. These oxidative products are associated with fuel system deposits and corrosion (Maeda et al., 2011).

Free and Total Glycerin: the Free Glycerin value records the level of by-product glycerin that remains in B100. The Total Glycerin result measures the amounts of unconverted and partially converted fats and oils. Excessive levels of these components may lead to storage tank and fuel system filter plugging, along with engine fouling (Maeda et al., 2011).

Oxidative Stability: the oxidative stability of the fuel is related in part to the acid number results. As the biodiesel degrades, or oxidizes, organic acids or polymers are created as a by-product which may lead to corrosion or filter plugging. The Oxidation Stability result is an accelerated test used to predict the fuel's stability for longer-term storage, possibly up to six months (Maeda et al., 2011).

Water and Sediment: it is a test that determines the volume of free water and sediment in middle distillate fuels having viscosities at 40°C in the range 1.0 to 4.1 mm²/s and densities in the range of 700 to 900 kg/m³. For biodiesel, this test is determinant of presence of free water droplets and sediment particles.

Flash Point: flash point of a fuel is the temperature at which it will ignite when exposed to a flame or a spark. Flash point varies inversely with the fuel's volatility. The flash point is the lowest temperature at which fuel emits enough vapors to ignite. Biodiesel has a high flash point which is usually more than 150°C, while generally conventional diesel fuel has a flash point of 55–66°C. It is important to note that a typical result of 130°C for biodiesel is almost twice that of petroleum diesel (approximately 70°C) and therefore much safer to handle and transport. The Flash point is also used to ensure the residual methanol left in the

fuel after biodiesel processing will not negatively affect combustion and other fuel system components (Maeda et al., 2011).

Pour Point: pour point is the lowest temperature at which the diesel fuel will flow when cooled under test conditions. That is related to lower temperature at which diesel engine will be able to operate.

Iodine Value: indicates the level of saturation of the molecules according to ASTM, the iodine number helps to indicate the oxidation stability of the biodiesel. The higher iodine number represents the lower oxidation stability.

Sulfated Ash: it is the residue remaining after a fuel sample has been carbonized, and the residue subsequently treated with sulfuric acid and heated to a constant weight. This test monitors the mineral ash residual when fuel is burned. For biodiesel this test is an important indicator for the quantity of residual metals in the fuel that may be came from catalyst used in the transesterification process (Yordanov and Petkov, 2009).

Carbon Residue: in petroleum products, the part remaining after a sample has been subjected to thermal decomposition is the carbon residue. The carbon residue is a measure of how much residual carbon remains after combustion. The most common cause of excess carbon residues in biodiesel is an excessive level of total glycerin (Frantsina et al., 2019).

Biodiesel Qualities (Specifications): according to the American Society of Testing & Materials (ASTM), the Standard Specification for biodiesel (B100) is ASTM D6751-02 (Markley, 1961, p. 861). Also the European standard (EN 14214) is adopted by all 31 member states of the European Committee for standardization of biodiesel (B100) (Arild and Christian, 2005). The standard ASTM D6751 describes the quality requirements and the methods of analysis used for biodiesel blended with diesel oil, applying to methyl esters as well as for ethyl esters. The ASTM and EU biodiesel property specifications with the recommended test methods are given in the Table 2.1 below.

Table 2.1: Specifications of biodiesel according to ASTM and EN standards (Markley, 1961, p. 891)

Properties of biodiesel	Units	ASTM D-6751		EN 14214	
		Limits	Test methods	Limits	Test methods
Kinematic viscosity (40°C)	mm ² /s	1.9-6.0	D445	3.5-5.0	EN ISO 3104
Density(15°C)	Kg/m ³	N/S	N/S	860-900	EN ISO 3675
Ester content	mass%	N/S	N/S	96.5 min	EN 14103
Cetane number	-	47 min	D613	51 min	EN ISO 5165
Flash point	°C	130 min	D93	120 min	ISO/CD 3679
Water content	volume%	0.050 max	D2709	500 max	EN ISO 12937
Sulphated ash	mass%	0.020 max	D874	0.02 max	ISO 3987
Copper corrosion	grade	No.3 max	D130	No.1	EN ISO 2160
Acid number	mg KOH/g	0.80 max	D664	0.5 max	EN 14104
Iodine number	-	N/S	N/S	120 max	EN 14111
Oxidative ability (110°C)	H	N/S	N/S	0.8 max	EN 14112
Oxygen content	%	N/S	N/S	N/S	N/S
High calorific value	MJ/kg	N/S	N/S	N/S	N/S
Free glycerol	mass%	0.020 max	D6584	0.02 max	EN 14105
Total glycerol	mass%	0.240 max	D6584	0.25 max	EN 14105
Phosphorous content	mass%	0.001 max	D4951	10 max	EN 14107

2.8 Factors Affecting Performance and Emission of Diesel Engine Fueled with Diesel-Biodiesel Blend

The performance and emission characteristics of diesel-biodiesel blended fuel of diesel engine may be affected by the type of additives used, intake air temperature and diesel-biodiesel blend ratio etc., but also influenced by engine operating parameters like compression ratio, ignition timing, fuel-air equivalence ratio and etc. Some existing related previous works are reviewed and highlighted below.

2.8.1 Additives

Now a day, for vehicular fuels, there are numerous chemical additives used to improve the quality of biodiesel and diesel fuel to accumulate up the most wanted performance level. Different additives that used for diesel and biodiesel fuels in various forms to modify the properties of those fuels are being isolated in terms of their application and drawbacks.

Cetane Number Improver Additives: delay period is one of the most important role plays in diesel engine, in which the delay period varies with chemical and physical properties of the diesel or biodiesel fuels. To differ the delay period cetane number improvers are the main important parameter. To minimize the ignition delay during the combustion, cetane number improvers help to get better working in diesel engine ignition (Hong et al., 2001). During the combustion process, shorter ignition delay period leads to more complete combustion of the fuel there in the internal combustion engine, reduces the noise of the engine and also reduces the unwanted harmful pollutants in the environment. Alkyl nitrates have been proved the most important vital in commercially used cetane improvers. The 2-Ethylhexyl nitrate is the cetane improver which has been used very commercially for a number of years, but today it is major cetane number improving additives (Stein et al., 1999).

Metal Based Nano-Particle Additives: addition of some metal and metal oxide in the form of nano-powder to the base fuel could enhance the properties of the fuels. These additives consist of copper(II) chloride (CuCl_2), iron (III) chloride (FeCl_3), copper(II) oxide (CuO_2 -nano structured), cobalt (II) chloride (CoCl_2), cerium oxide (CeO_2), aluminum oxide (AlO_3) are added in fuel as a catalyst for diesel and biodiesel based on the necessity requirement. This is due to the stimulating properties of nano particles like higher specific surface area, thermal conductivity, catalytic activity and chemical properties as compared to their bulk form. Reduced size of the nano-particles increased specific surface, surface to volume ratio, and surface area improving catalytic reactivity and magnetic properties as compared to their bulk form. Hence metal and metal oxide nano-particles addition to biofuel will improve the performance as well as reduce the emissions from engine exhaust (Hariram et al., 2017).

Antioxidant Additives: the major purpose of using antioxidant to biodiesel fuels was to slow down the development of slush deposits and darkening of color. Petroleum products are very quick moving diesel fuels, but they are very rarely, are stored in for 1–3 years and are subjected to corrosion. Few names of antioxidant additives are butylated hydroxyl anisole (BHA), butylated hydroxyl toluene (BHT), pyrogallol (PL), diphenylamine (DPA), tert-butylhydroxyquinone (TBHQ) and propyl-gallate (PG). From the survey TBHQ, PL and

PG are mainly successful among all the antioxidants. These additives are mainly used in biodiesel mixing to get a better stabilizing potential. As the amount of free radicals and peroxide species increases, the oxidative chain reaction raises at a produced exponential rate. By decomposing the peroxides, the antioxidant works by disturbing the chain propagating steps (Vijay Kumar et al., 2018).

Oxygenated Additives: the most important additives for diesel engine are oxygenated additives. The fuels that are containing oxygen and blending components contain at least one oxygen atom by the molecules at the side of the hydrogen and carbon atoms. The oxygenated additives are very useful to develop the combustion process and octane rating.

By blending oxygenated additives in biodiesel and diesel sufficient cetane number should be there in oxygenated additives and allowed the blend to increase the cetane number. The oxygen helps to support for burning the fuel without emitting any high amount of inert material such as nitrogen into the air and it causes the harmful material such as NO_x emission at some operating load condition in CI diesel engine. The generally used oxygenated additives are alcohols, ether and ester. The few names of alcohols contain butanol, propanol, methanol and ethanol. By addition of oxygenated additives, the ignition temperature of biodiesel will be minimized and also reduction in smoke emission is observed in the diesel engine. According to the composition of diesel and biodiesel, the oxygenated additives will affect directly the properties such as cetane number, density, viscosity, volatility, flash point and calorific value (Stein et al., 1999). To ignite the fuel more efficiently oxygenated additives will support and as well as diminish environment pollution. The engine fuels will burn more completely due to the presence of oxygenated additives.

Mehta et al., (2012) carried out experimental investigation to determine the feasibility of utilizing renewable bioethanol as an oxygenated fuel additive with petro diesel in two phases:

- The first phase of the work deals with the study of stability of the blends with varying surfactant (biodiesel and span 80) concentration with the assistance of Turbiscan plots.

It was observed that the destabilization of the blends and percentage of backscattering reduced with increase in surfactant concentration, and the blends became most stable when 25% biodiesel (B5b) and 1.5% span 80 (B3s) were added.

- The second phase comprised determination of fuel properties and engine performance of most stable blends.

The result showed that both of the selected stable blends showed good resemblance in properties such as density, kinematic viscosity and copper strip corrosion with respect to diesel; whereas cetane index, and flash points showed marginal reduction due to presence of ethanol containing lower heating value, flash point, and cetane index. Properties were determined in accordance with ASTM standards.

And engine performance characteristics such as BSFC of B5b showed a drop of 2.7% and blend B3s showed almost same consumption as compared with diesel. The A/F ratio for the B5b blend was lower by 10.7%, whereas it increased by 10.4% for the B3s blend in comparison to diesel at medium loads. The EGT was high by 1.7% for B5b whereas B3s showed a reduction of 3.5%. Engines gave lesser BTE by 10.7% for B5b but an intuitive increment was observed for the B3s blend.

Channappagoudra et al., (2018.) explores the effect of dairy scum oil methyl ester (DSOME) blends and 3%, 6% and 9% ethanol additive on TV1 Kirloskar diesel engine performance, combustion and emission characteristics. The results showed that diesel engine operated with ethanol additive and 20% dairy scum biodiesel blended fuels shown the satisfactorily improved emission characteristics when compared to petroleum diesel fuel operation. Finally, they conclude addition of ethanol shown the slightly higher HC, CO emission and improved BTE, BSFC, NO_x and CO₂ than sole B20 biodiesel blend. Among all ethanol additive ratios, E6% (6%- ethanol with B20) ethanol additive exhibits slightly better BTE, BSFC, cylinder pressure and heat release rate hence 6% ethanol additive with B20 biodiesel blend would furnish beneficial effects in the diesel engine.

Jaremporn et al., (2020) studied the comparative phase stability of using diesel–palm fatty acid distillate ethyl ester (PFADE)– hydrous ethanol and diesel– palm fatty acid distillate (PFAD –hydrous ethanol in a direct injection diesel engine without any major modifications with varied blend proportions. The palm fatty acid distillate (PFAD) and PFADE were directly blended in ethanol and diesel. To evaluate the feasibilities of these fuel blends the fuel properties, emissions (CO, CO₂, NO_x, O₂, exhaust gas temperature), and fuel consumptions of diesel, PFADE, diesel–PFADE–hydrous ethanol were compared in a diesel engine at the engine speeds 1100, 1400, 1700, 2000, and 2300 rpm. At 2300 rpm, the maximum CO₂ emission with 10 % hydrous ethanol in the blend was approximately 2%. With regard to fuel consumption, clearly, 20 % diesohol gave higher consumption than 10

% ethanol at a maximum engine speed of 2300 rpm. The blend D50PE40E10 gave the lowest fuel consumption, while the maximum fuel consumption was with the D10PE70E20 blend. Therefore, both 10 and 20 % hydrous ethanol in the diesel fuel are alternatives usable in a diesel engine without modifications. But hydrous ethanol blending ratio of 10% was found to be the optimum blending ratio and thereby give a better performance and lower emission, than that of the other blending ratios.

Generally, from the literature above it is observed that engine performance decreased with ethanol fraction increasing while the emissions of unburned hydrocarbon also increased. It can be concluded that oxides of nitrogen (NO_x) emission formation rate decreased drastically with ethanol based operations as compared to neat diesel operations of the test engine. This could be mainly due to the effect of ethanol addition which possesses high heat of vaporization. The brake power was very similar at all blend ratios and comparable with the brake power of diesel fuel at all engine speeds. Brake specific fuel consumption was increased in all alcohol blend. Introducing diesohol fuel can help increase biofuel consumption from the diesel segment. In addition, the fossil fuel can be suppressed and the greenhouse gas production can be mitigated.

2.8.2 Intake Air Temperature

Intake air temperature is one of the major factors that affect combustion efficiency, performance, and emissions characteristics of the diesel engine running with diesel-biodiesel-ethanol blends (Nadir, 2011).

The results of several studies showed that in-cylinder pressure and heat release rate increased with the increase of intake air temperature. The increase of intake air temperature caused combustion to advance and the combustion duration to decrease (Can et al., 2015). An increase in the air intake temperature also caused variations in the ignition delay period of the biodiesel–ethanol blend by improving the vaporization characteristic of ethanol, and provides a better combustion (Nantha et al., 2017).

As a consequence of better vaporization and in-cylinder combustion, intake air preheat could potentially help diesel-biodiesel–alcohol blends reduce CO and HC emissions but at the same time increase NO_x emission but the results would depend on the magnitude of the elevated temperature. Exhaust gas temperature increased by increasing intake air temperature as well. But, at full load, preheat may not make a difference perhaps because full engine load provides sufficient energy for diesel-biodiesel–alcohol blends to vaporize

and more completely burn (Nadir, 2012). And the BTE for preheated intake air ethanol blends was higher than that of the ambient temperature intake air (Nantha et al., 2017). For biodiesel–alcohol blended fuels evaluated in a diesel engine at ambient and preheated intake air temperatures, it was seen that intake air preheat helps biodiesel–alcohol blends reduce fuel consumption (Nadir, 2012).

Nadir, (2011) investigated the effects of intake air preheat on performance and emissions of a compression ignition engine running on fuel concentrations of biodiesel (85%)–methanol (15%), biodiesel (90%)–methanol (10%), biodiesel (95%)–methanol (5%), neat biodiesel (B100). Fuels were evaluated in the engine at two elevated intake air temperatures at 30°C and 85°C. Neat biodiesel and standard diesel fuels were used in the engine at the ambient air temperature. The resulting emissions indicate that preheating the intake air or lowering the methanol concentration in biodiesel–methanol blends tends to reduce the production of CO and HC while increasing NO_x emission.

Nadir, (2011) studied the effects of intake air preheat on performance and emissions of a compression ignition engine running on biodiesel (85%)–alcohol (15%) fuels and compared to baseline results of standard diesel and neat biodiesel. Emissions were compared at two elevated intake air temperature and results indicate that high heat of vaporization of alcohol fuels affect emissions significantly. Intake air preheat was proved to be one of the effective solutions to reduce CO and HC emissions. For biodiesel (85%)–alcohol (15%) blended fuels evaluated in a diesel engine at ambient and preheated intake air temperatures, it was seen that intake air preheat helps biodiesel–alcohol blends reduce fuel consumption. Exhaust gas temperature increased by increasing intake air temperature as well.

Senthil and Thundil, (2013) experimentally investigated the effect of fuel injection timing and intake air temperature using pure ethanol blended biodiesel fuel (B5E15), on combustion and emission characteristics of a single cylinder, four-stroke, air cooled direct injection diesel engine. The tests were carried out using B5E15 fuel under constant speed (1500 rpm) and load (2 kW) with different injection timing (12, 15, 18, 21 and 24° CA bTDC) and different intake air temperatures (40°C and 60°C) at 1.1 bar intake manifold pressure. They conclude that an increase in intake air temperature results higher in cylinder peak temperature, reduced ignition delay, increased NO_x emissions and decrease in smoke, CO and HC emissions.

Ramadhas and Hongming, (2017) conducted an experimental investigation to study the potential of air heating strategy in reducing the particulate emissions from the latest

generation of turbocharged direct-injection diesel engines at very cold ambient temperature conditions. New European Driving Cycle (NEDC) tests were conducted at 7°C ambient temperature conditions for the different intake air temperatures. It was observed that intake air heating reduced hydrocarbon emissions by 35 and 55% and nitrogen oxides emissions by 8 and 10% for intake air temperatures of 5 and 15°C, respectively, during the cold-start phase of the emission driving cycle. The cold-start phase of the NEDC cycle accounted for 1/4 of the total particulates, and the use of intake air heating reduced its proportion to 1/5. More than 50% of the particulates are less than 23 nm in diameter for all the intake air temperature conditions, and the intake air heating reduced the particulate number concentration significantly. The surface area of accumulation mode particles was higher than that of the nucleation mode, and the total particle surface area decreased with the intake air heating.

Nantha et al. (2017) studied the effects of preheating intake air on performance, emissions and combustion behavior for various compositions of cottonseed oil biodiesel–ethanol blends in a compression ignition engine. The characteristics were compared for intake air temperatures of 30°C and 80°C, respectively. It was obtained that an increase in the air intake temperature caused variations in the ignition delay period of the biodiesel–ethanol blend and provides a better combustion. It was found experimentally that the carbon monoxide (CO) as well as the unburned hydrocarbon (HC) emissions decreased with an increase in the preheat temperature, and were found to be slightly lower than those of biodiesel-fueled CI engines.

Generally, as noticed from the above literature it is clear that the fuel consumption reduces and brake thermal efficiency increases with increase in intake air temperature. The carbon dioxide increase and oxygen content decrease in the exhaust gas. Carbon monoxide content in the exhaust gas slightly reduces with increase in intake air temperature. Whereas NO_x content in the exhaust gas slightly increases with increase in intake air temperature. Hence, the test results indicated that the advantages gained are more with increase in intake air temperature though there are few disadvantages.

As observed from the literature above, preheating intake air is the major criteria for improving the performance and reduction in emission of the diesel-biodiesel-ethanol blended fuels of diesel engine. This could be achieved through employing an appropriate heat exchanger device as briefly discussed in the next section below.

2.8.2.1 Heat Exchanger Device for Preheating Intake Air

There are several different types of heat exchangers which include Double-pipe exchangers, Shell and tube exchangers, air coolers and many more. Where, 80% of the energy utilization systems uses heat exchangers (Chen et al., 2011).

Shell and tube heat exchangers are commonly used in the chemical industries and separations processes. At least 60% of heat exchanger types are the shell-and-tube type exchanger, indicating its prevalence in the process industry (Max et al., 2003), with applications in oil coolers and power condensers, as well as preheaters in power plants and steam generators in nuclear power plants, process applications, and the chemical industry (Kakac et al., 2020). They are made from bundles of tubes and may be composed of several types of tubes: plain, longitudinally finned, etc (Dilip et al., 2015). The ends of tubes are placed in sheets that divide the shell and tubes fluids (Sinnott, 2005). Baffles are also placed in the shells which aid in direction the shell-side fluid flow and create a more turbulent flow regime. There are several advantages with this heat exchanger type that include a large surface area for a small volume, easy cleaning, a good mechanical layout and well-known design procedures (Sinnott, 2005). It is necessary to know all the parameters and data of the requested design before starting calculations. Therefore, the design will be limited to certain specification requested by the company and these specifications will help to make the design applicable for the requested conditions.

Vijay et al., (2016) presents a concept of recovering waste heat energy of exhaust gas of diesel engine by placing a heat exchanger in the exhaust manifold so that energy from the exhaust gases can be used for preheating fuel. Initially a simple concentric tube type design is used to estimate the temperature of fuel by utilizing energy from exhaust gases. Based on the output obtained from this initial design, a shell and tube heat exchanger is fabricated for diesel engine to recover waste heat from exhaust. Maximum fuel temperature achieved for concentric tube heat exchanger is 133°C at 25% of full load at 1500 rpm. For shell and tube heat exchanger, 73°C and 90°C of fuel preheat temperatures are achieved for parallel and counter flow arrangements at 50% of full load at 1500 rpm. Effectiveness of heat exchanger is found to be 75% for parallel flow and 81% counter flow. Waste heat recovered at 50% full load condition is found to be 48.5% for parallel flow and 72% for counter flow arrangements respectively.

Jose et al., (2016) worked on experimental facility implemented with the aim of improving the performance of internal combustion engines working at low ambient temperatures. In

this work, an exhaust heat recovery system for a diesel engine has been proposed as a solution to cold operation negative effects. The energy obtained from the exhaust gases was used to increase the intake air temperature. Engine Intake manifold is replaced by a water/air heat exchanger, known as Water Charge Air Cooler (WCAC) that is joined to a short intake manifold. Intake air heating results with the heat recovery system shows a 10% in fuel use compared to standard air–air intercooler.

The conventional design method for segmentally baffled shell and tube heat exchangers applies an iterative procedure for several configurations until one is found that meets the system heat duty and both calculated pressure drops are below the maximum allowable values. Typically, the use of this approach results in oversized equipment and complex design procedures.

Polley et al., (1991) have reported different design algorithms that ensure that the pressure drops within the exchanger equal the specified values of the allowable pressure drops for the streams given as part of the design specifications. This full use of the allowable pressure drops provides the highest possible heat transfer coefficients and therefore smaller exchangers, thus saving capital costs.

Jegade and Polley, (1992) formulated simple equations for each stream based on the Kern method (Kern, 1950), which relates the fluid exchanger pressure drop with the film heat transfer coefficient and the heat exchanger area. The combination of the pressure drop relationships with the basic exchanger design equation gave rise to a simple design algorithm for shell and tube heat exchangers that avoids the iteration procedure for different geometries. However, the use of the Kern method may lead to significant errors in the calculations because of its simplified flow pattern model for the shell side, which does not take into account the effect of leakage and bypass streams.

The algorithm by (Polley et al., (1991) uses the Bell- Delaware method (Taborek, 1983) to describe the flow pattern of the shell-side fluid. The model accounts for leakage and bypass streams, according to the flow model proposed by (Tinker, 1951). Although the model by (Polley et al., 1991), provides better estimations than that by (Jegade and Polley, 1992), some shortcomings can be mentioned. In order to keep the accuracy of the Bell-Delaware method, (Polley et al., 1991), developed a rather complex relationship for pressure drop estimation on the shell side, which requires an iterative procedure that involves detailed estimations of exchanger geometries. A second limitation of the algorithm is its lack of flexibility for the shell side because it is restricted by the assumptions that the cross-flow areas are equal to

the window flow areas and that spacing for end baffles are equal to those for the central baffles. Although the first geometric restriction may promote a higher conversion from pressure drop into heat transfer, it implicitly determines the baffle cuts and commonly provides designs with a fractional number of baffles. The second geometric restriction ignores the fact that due to the large inlet and outlet nozzles, in some cases it is necessary to have higher inlet and outlet baffle spacing than central baffle spacing (Taborek, 1983). A third limitation of the algorithm by (Polley et al., 1991), is that it does not take into account the end pressure losses on the tube side due to contraction at the tube inlets, expansion at the exits, and flow reversal in the headers.

Generally, as noticed from the literature above a counter flow shell and tube heat exchanger is better from that of other types of heat exchangers for recovering waste heat energy of exhaust gas of diesel engine in the case of:

- Large surface area for a small volume
- Easy cleaning
- Good mechanical layout
- Well-known design procedures

And Kern method is better among the others as noticed since it provides design algorithms that ensure that the pressure drops within the exchanger equal the specified values of the allowable pressure drops for the streams given as part of the design specifications. This full use of the allowable pressure drops provides the highest possible heat transfer coefficients and therefore smaller exchangers, thus saving capital costs.

2.8.3 Diesel-Biodiesel Blend Ratio

Ilkilic and Bebcet, (2010) studied the emissions of biodiesel produced from cottonseed oil blended with diesel fuel, 20% biodiesel and 80% commercial diesel fuel, called B20, used as a fuel in a direct injection diesel engine. The study aimed to test the emissions of a diesel engine using this blend and diesel fuel, and characterize gaseous emissions that are harmful to humans and the environment. The result showed on the emissions of the engine that were about 19% reduction in NO_x, about 22% reduction in CO, and 13% reduction in CO₂. They conclude that the reduction in exhaust emissions has made biodiesel a suitable alternative for diesel engines and could contribute in controlling air pollution.

Karnwal et al., (2011) conducted an experimental study that involves an application of the Taguchi method and grey relational analysis to determine the optimum factor level to obtain

optimum multiple-performance characteristics of a diesel engine run with different low-percentage thumba biodiesel-diesel blends. Four factors, namely, low-percentage thumba biodiesel-diesel blend, compression ratio, nozzle opening pressure and injection timing were each considered at three levels. An L9 orthogonal array was used to collect data for various engine performance- and emission-related responses under different engine loads. The signal-to-noise (S/N) ratio and grey relational analysis were used for data analysis. The results of the study revealed that the combination of a blend consisting of 30% thumba biodiesel (B30), a compression ratio of 14, a nozzle opening pressure of 250 bar and an injection timing of 20° produces maximum multiple performance of a diesel engine with minimum multiple emissions from the engine.

Horng-Wen and Zhan-Yi, (2012) used a Taguchi method to determine the optimal combinations of concentrations for a diesel engine with diesel/biodiesel blend using H₂ and cooled exhaust gas recirculation (EGR) at the inlet port. They determined the optimal operating factors for achieving good combustion performance, low NO_x and smoke, at various engine loads and at 1500 rpm. Experimental results show that predictions by Taguchi's parameter design technique are in adequate agreement with the confirmation results, with a confidence interval of 95%, and this technique saves 67% of the time taken to perform the experiment in this research. The best brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC), NO_x and smoke at each load is achieved for a combination of B20 (A2), 30% hydrogen (B3) and 40% EGR ratio (C3). In addition, the best combination reduces the BSFC and inhibits both NO_x and smoke emissions. At a load of 60%, the reduction rate is 25.4% for BSFC, 74.1% for NO_x and 29.6% for smoke.

Ramesh et al., (2018) conducted experiment to study emission characteristics and performance parameters of biodiesel blend with and without ZNO additive on diesel engine in two phases:

- In the first part of the research there are four biodiesel blends studied and found that the B25 combination gives better result compare to the others. The CO and HC emission gets decreased.
- In the second part of the research the addition of 125 PPM of ZNO to the selected B25 blends gives better performance, the efficiency improvement is found to be 4.2% and the emission of NO_x is by 10.3% under full load condition.

Ali et al. (2018) carried out experiments on a single-cylinder engine in accordance to G-2 of ISO 8718 standard at wide ranges of blending ratios and operating conditions. The results

showed that (i) a maximum increase of 8% in the brake specific energy consumptions was received at a blending ratio of 30%, (ii) a slight increase in the brake thermal efficiency (around 1%) was obtained at a blending ratio of 20%, (iii) the best reduction in the carbon monoxide emission (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x) were 17%, 40%, and 0.05%, respectively at a blending ratio of 10%, (iv) the best reduction of 7.5% in carbon dioxide emissions was observed at 20% blending ratio, and (v) the minimum opacity level was observed at a blending ratio of 30%. They conclude that the 20% blending ratio is recommended to keep high engine efficiency without environmental deterioration.

Medhat et al. (2019) carried out an experimental study to investigate the influence of diesel/biodiesel blends on engine combustion, performance, and exhaust gas emissions at different engine loads and constant speed of 1400 rpm. Volumetric percentage of diesel/biodiesel blends: D70B30 (70% diesel–30% biodiesel), D50B50 (50% diesel–50% biodiesel) and D30B70 (30% diesel–70% biodiesel) were prepared to power a single cylinder diesel engine. The engine results show that a reduction in the rate of change of CO by 33.8% for D50B50, the slight decrease in maximum cylinder pressure for higher percentage of biodiesel blends, the reduction in HRR for biodiesel blends, HRR was about 31.7, 52.4 and 63.5 (J/deg) for 10%, 30% and 60% of maximum engine power. The highest reduction in HC emissions concerning diesel fuel was about 4.18% for D30B70. NO_x emissions of biodiesel blends were higher than diesel. Exhaust oxygen (EO) emissions for D30B70 was about 0.98% higher than diesel. Exhaust gas temperature (EGT) has observed for all biodiesel blends. Brake specific fuel consumption (BSFC) is increased until it reaches 11.43% for D30B70. A consequent reduction in brake thermal efficiency (BTE) and brake specific energy consumption (BSEC) is observed for all biodiesel blends.

Abed et al. (2019) carried out experimental work on a single cylinder diesel engine was run using different sources of biodiesel such as *Jatropha*, palm, algae and waste cooking oil biodiesel blends B10 and B20. Exhaust emissions were examined at different engine loads of 1, 2, 3 and 4 kW and a constant engine speed of 1500 rpm. Exhaust emissions such as CO, CO₂, NO_x, HC and smoke emissions were examined and compared with diesel fuel. They conclude the following:

- CO, HC and smoke emissions were lower for the tested four biodiesel types (*Jatropha*, algae and palm and waste cooking oil) mixtures B10 and B20 as compared to diesel fuel.

- CO₂ emissions from biodiesel blends B10 and B20 produced from waste cooking oil were higher compared to diesel fuel and the other biodiesel fuel.
- NO_x emissions; from biodiesel mixtures, B10 and B20 increased compared with diesel fuel for the examined biodiesel blends.

Generally, as seen from the literature reviewed above biodiesel has a great potential to reduce environmental pollution as an alternative fuel for diesel engines and conclude that B20 fuel is a suitable alternative fuel for diesel engines, and it can be easily produced and used without any modification of the diesel engine. As a result, B20 fuel is technically competitive with or offers technical advantages compared to conventional diesel fuel. Besides being a renewable and domestic resource, biodiesel blends reduce most emissions while engine performance and fuel economy are nearly identical compared to conventional fuels.

2.9 Summary of Literatures

Generally, from the reviewed literature above it is noticed that constantly converging emission norms have obliged the use of diesel engines, as it is very much culpable for high emissions of NO_x, particulate matters and other polycyclic aromatic hydrocarbons. One way to overcome this situation is incorporate oxygen releasing molecules in fuel. As seen from the reviewed literatures above rationalizing the usage of biofuels such as biodiesel, alcohols etc., rich in reactive oxygen molecules in their structure, as a sustainable, renewable and viable alternative for existing diesel power trains without the necessity to adopt radical changes in the existing diesel platforms becomes one part of an area understudied in the current research.

As seen from the literature reviewed above biodiesel is an alternative fuel that is cleaner than petro diesel and can be used directly as fuel for a diesel engine without having to modify the engine system. It has the major advantages of having high biodegradability, excellent lubricity and no sulfur content. As the biodiesel content in the fuel rises:

- There is an increment of BSFC and reduction in brake power and BTE.
- There is a reduction of CO, HC and an increment of NO_x and CO₂.

And as load varies from 20% to 80% for all tested fuels:

- There is a decrease in the BSFC and increase in the brake power and BTE of blends for a corresponding increase of engine load, following a similar trend to that for diesel.

The effect of biodiesel's fuel properties on the combustion characteristics, performance and exhaust emissions of diesel engines using biodiesels from edible and non-edible oils is other area understudied.

Blending biodiesel alone with diesel fuel is not enough work to reduce the harmful emissions from diesel engine. As noticed from the literature increased inert gases emissions such as NO_x from biodiesel fueled engines can be countered by incorporating oxygenated additives particularly ethanol into the fuel blend and preheating intake air. From the literature it apparent that diesel engine operated with ethanol additive and biodiesel blended fuels shown the satisfactorily improved emission characteristics when compared to petroleum diesel fuel operation and addition of ethanol shown the slightly higher HC, CO emission and improved BTE, BSFC, NO_x and CO₂ than sole biodiesel blend.

Intake air preheat was proved to be one of the effective solutions to reduce CO and HC emissions. Reduction of alcohol concentration in diesel-biodiesel-alcohol blends also showed similar effects to preheating intake air temperature.

2.10 Literature Gaps

The facts cited in relevant published articles have been analyzed critically and the following salient features are found not being addressed properly.

- The literature reviewed shows that the study on diesel engine operated with diesel-biodiesel blends with addition of ethanol and preheated intake air is very much limited. In most studies, the authors concentrate only on the effect of ethanol percentage on engine performance and emission parameters. The effect of preheating intake air with different percentage volume of biofuel (biodiesel and ethanol) on engine performance parameters and emission characteristics of a conventional diesel engine for different loading conditions is not studied exhaustively.
- The use of ethanol and biodiesel as a fuel of diesel engine is also limited. Many of the researches have used experimental analysis on biodiesel alone for diesel engine characteristics to substitute diesel fuel partially.

CHAPTER THREE

MATERIALS AND METHODS

This chapter comprises the computational and experimental analyses to study the effects of preheating intake air and percentage variation of biodiesel in diesel-biodiesel blends on diesel engine performance and emission parameters under certain operating conditions of diesel engine fueled with diesel-biodiesel blends with addition of ethanol. This also includes shell and tube counter flow heat exchanger that can be used to preheat the intake air. The numerical analysis for designing a heat exchanger was used for the computational methods that are described with the necessary illustrations. Whereas the experimental investigation was carried out using a single-cylinder, direct-injection four-stroke diesel engine equipped with an eddy current dynamometer for loading the engine. The detailed research methodologies and required materials are explained below in the next sections.

3.1 Materials

This section describes the equipment's being utilized in this research along with detail of its characteristics and measurement capabilities. It also includes the experimental setup and procedure, device calibration and equations and measurements that were used to obtain the performance and emission data. In any experimental study, instrumentation plays an important role, as it provides the required data for analysis. Before starting an experimental investigation, it is essential to acquire the basic knowledge about the instruments required for experimentation and details about their working principles, range, accuracy, etc. In this section, different instruments used for the experimental investigation are described with the necessary illustrations. In this thesis work, the main components were fabricated heat exchanger and produced biodiesel, and employed in the modified experimental setup for experimentation. In this regard, the materials and equipment used for manufacturing the heat exchanger, biodiesel production and experimental tests were mild steel, thermocouple, ethanol, drilling machine, cutting tool, stand funnel separators, Beakers, etc., are listed in Table 3.1, Table 3.2 and Table 3.3.

Table 3.1: Materials, tools, and machines used for heat exchanger manufacturing

No.	Item	Quantity
1	Mild steel	1 m x 0.5 m
2	Copper tube	30 mm diameter, 200 mm length
3	Electrodes	1 pack, 2.5 mm diameter
4	Drilling tool	30 mm diameter
5	cutting tool	2
6	Rolling machine	1
7	Welding machine	1
8	Drilling machine	1

Table 3.2: List of materials and equipment used for experimental test

No.	Item	Quantity
1	Diesel fuel	10 Liter
2	Ethanol	1 Liter
3	Biodiesel	3.6 Liter
4	Thermocouple(K type)	1
5	Flow control valve	1
7	Diesel engine	1
8	Exhaust gas analyzer	1
9	Dynamometer	1
10	Computer (desktop)	2

Table 3.3: Materials, tools, and equipment required for biodiesel production

No	Item	Unit	Quantity
1	Cotton seed oil	Liter	5
2	KOH	Gram	300
3	Methanol	Liter	1
4	Distilled water	Liter	100
5	Aluminum foil	Inch	4
6	Conical flasks	Pcs	6
7	Hot plate with magnetic stirrer	Pcs	3
8	Beakers (different sizes)	Pcs	6
9	Thermometer	Pcs	1
10	Digital Balance	Pcs	1
11	Stand funnel separators	Pcs	3

3.2 Methodology

To address the general and specific objectives of this work a well-designed methodology was being followed. There were combinations of procedures that were used to accomplish this study. These were design and then manufacturing of the heat exchanger, preparing fuels (producing biodiesel from cotton seed and purchasing ethanol and diesel fuel), preparing the experimental setup of the test engine for experiment and analysis of experimental results. This thesis work was conducted based on the following procedures summarized as shown in Figure 3.1 below.

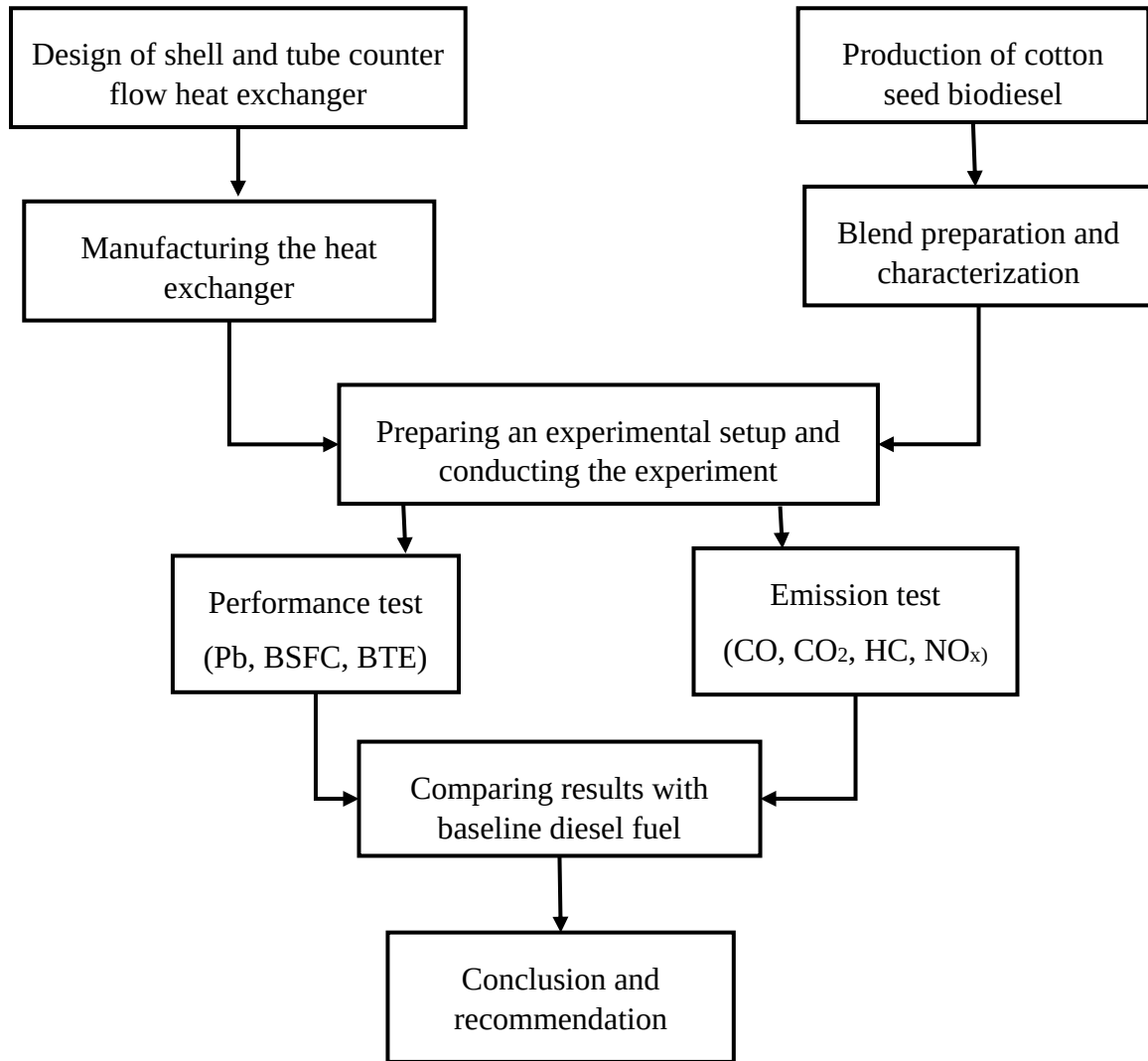


Figure 3.1: Block diagram of the procedure of this thesis work

3.3 Design of Heat Exchanger

Heat exchangers are devices that provide the transfer of thermal energy between two or more fluids at different temperatures. Heat exchangers are used in a wide variety of applications, such as power production, process, chemical and food industries, electronics, environmental engineering, waste heat recovery, manufacturing industry, air-conditioning, refrigeration, space applications, etc (Sadik et al., 2012, p. 1). In this study, intake air preheater is integrated with the test engine setup for heating the intake air prior to fuel-air mixing and subsequent entry to the combustion chamber. In the present study, a recuperator type of heat exchanger with counter flow arrangement as shown in Figure 3.2 is chosen for which the heat transfer occurs between two fluid streams separated by a heat transfer surface.

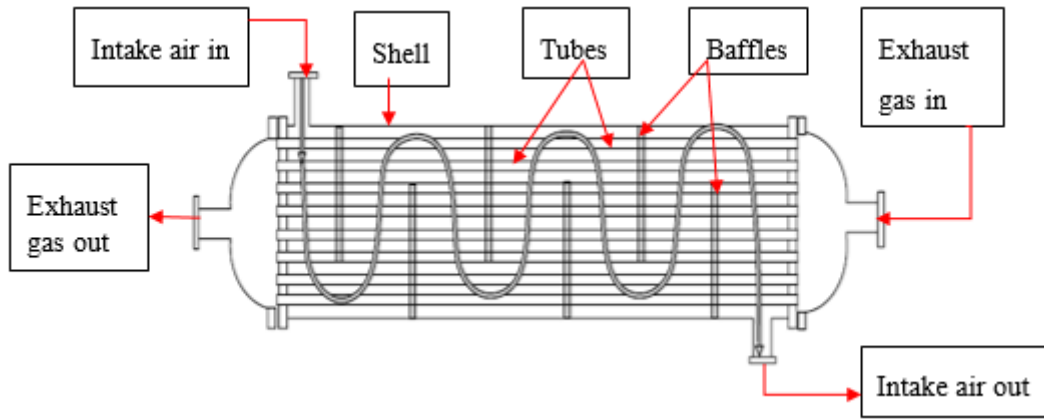


Figure 3.2: Counter flow shell and tube heat exchanger arrangement (Lei, et al., 2017).

3.3.1 Design Calculation of Heat Exchanger

The most common problem in heat exchanger design is thermal analysis (i.e., rating and sizing calculation). The “rating problem” is concerned with the determination of the heat transfer rate and the fluid temperatures based on the Kern method (Kuppan, 2013, p. 273), which involves “trial and error calculations” by using equation:

$$Q = \dot{m}_c c_{p,c} (T_{co} - T_{ci}) = \dot{m}_h c_{p,h} (T_{ho} - T_{hi}) \quad (\text{Eq. 3.1})$$

Where; Q = Heat transfer rate (W)

$\dot{m}_c$ = Mass flow rate of cold fluid (kg/s)

$\dot{m}_h$ = Mass flow rate of hot fluid ($\frac{\text{kg}}{\text{s}}$)

c_p = Specific heat of fluid ($\frac{\text{J}}{\text{kg.K}}$)

T_o = Exit temperature of fluid (K)

T_i = Inlet temperature of fluid (K)

And the subscripts h and c refers to hot and cold fluids respectively.

$$(\Delta T_{LM})_{cc} = \frac{[(T_{hi} - T_{co}) - (T_{ho} - T_{ci})]}{\ln \left[\frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})} \right]} \quad (\text{Eq. 3.2})$$

Where; $(\Delta T_{LM})_{cc}$ = Log mean temperature difference (K)

The “sizing calculation” focuses on the evaluation of the dimensions of the heat exchanger (Hesselgreaves, 2001, p. 64). For preheating intake air, in a “shell and tube heat exchanger” mild steel and copper tube materials are used for shell and tube part of the preheater, respectively. To extract the maximum amount of heat from a given waste heat exhaust gas

“counter flow, single pass, shell, and tube heat exchanger is preferred. The heat exchanger is indigenously designed with assumptions to meet the requirements with a specific intended purpose. The following parameters are assumed: tube diameter and length, fouling factor based on inside and outside tubes, fixed baffle cut of 25%, three known temperatures (ambient temperature and exit temperatures of air and inlet temperature of exhaust gas), overall heat transfer coefficient, etc.

Assumptions for Heat Transfer Analysis:

- The heat exchanger operates under steady-state conditions [i.e., constant flow rates and fluid temperatures (at the inlet and within the exchanger) independent of time].
- Heat losses to or from the surroundings are negligible (i.e. the heat exchanger outside walls are adiabatic).
- There are no thermal energy sources or sinks in the exchanger walls or fluids, such as electric heating, chemical reaction, or nuclear processes.
- The temperature of each fluid is uniform over every cross section (i.e., perfect transverse mixing and no temperature gradient normal to the flow direction).
- Wall thermal resistance is distributed uniformly in the entire exchanger.
- Longitudinal heat conduction in the fluids and in the wall is negligible.
- The individual and overall heat transfer coefficients are constant (independent of temperature, time, and position) throughout the exchanger.
- The specific heat of each fluid is constant throughout the exchanger, so that the heat capacity rate on each side is treated as constant.
- The heat transfer surface area A is distributed uniformly on each fluid side.
- The velocity and temperature at the entrance of the heat exchanger on each fluid side are uniform over the flow cross section. There is no gross flow maldistribution at the inlet.
- The fluid flow rate is uniformly distributed through the exchanger on each fluid side i.e., no passage-to-passage or viscosity-induced maldistribution occurs in the exchanger core. Also, no flow stratification, flow bypassing, or flow leakages occur in any stream. The flow condition is characterized by the bulk (or mean) velocity at any cross section.

Then considering the above assumptions and using measured as well as assumed parameters the required heat exchanger is designed as follows:

From Eq. 3.1 the heat transfer rate can be calculated as:

$$Q = \dot{m}_a c_{p,c} (T_{ao} - T_{ai}) \quad (\text{Eq. 3.3})$$

Heat lost by gas is equal to heat gained by air (Nageswara, 2015). Hence,

$$Q = \dot{m}_a c_{p,a} (T_{ao} - T_{ai}) = \dot{m}_g c_{p,g} (T_{go} - T_{gi}) \quad (\text{Eq. 3.4})$$

Where the subscripts a and g refers to ambient air and exhaust gas respectively.

With the four temperatures known, the $(\Delta T_{LM})_{cc}$ can be calculate as:

$$(\Delta T_{LM})_{cc} = \frac{[(T_{gi} - T_{ao}) - (T_{go} - T_{ai})]}{(T_{gi} - T_{ao}) / (T_{go} - T_{ai})} \quad (\text{Eq. 3.5})$$

Calculate R and S parameters:

$$R = \frac{(T_{gi} - T_{go})}{(T_{ao} - T_{ai})}$$

$$S = \frac{(T_{ao} - T_{ai})}{(T_{gi} - T_{ao})} \quad (\text{Eq. 3.6})$$

Where; R and S are unit less temperature difference ratios.

Then using these values $F_t = 0.96$, from the chart (Kothandraman and Subramanya, 2007, p. 510). Where F_t is the friction factor.

According to the value obtained for Q, the heat transfer surface area is calculated using (Abazar and Nader, 2016):

$$A = \frac{Q}{U \times LMTD \times F_t} \quad (\text{Eq. 3.7})$$

Where; A = Heat transfer area (m^2)

U = Overall heat transfer coefficient ($\text{W}/\text{m}^2 \text{ K}$)

LMTD = Log mean temperature difference (K)

Here, it is necessary to find a combination of number of tubes and tube length to satisfy the value of area calculated in Eq. 3.7, which is (Olaiya Kamorudeen et al., 2018):

$$A = N \times \pi \times d_o \times L \quad (\text{Eq. 3.8})$$

Where; N = Number of tubes

L = Length of heat tube (mm)

d_o = External diameter of tubes (mm^2)

In order to calculate the heat transfer coefficients, it will be necessary to calculate (μ/μ_w) for each fluid, where μ is the viscosity at the mean temperature, and μ_w is the viscosity at the tube wall temperature. The tube wall temperature is not known beforehand. Thus it must be considered that the tube wall temperature will have an intermediate value between T and t and will be closer to the temperature of the fluid with higher film coefficient h .

To do these obtain the physical properties of the fluids at the mean temperatures. These will be $(T_{go} + T_{gi})/2$ and $(T_{ao} + T_{ai})/2$ for the hot and cold fluids, and these mean temperatures will be called T and t respectively. Thermal and physical properties of air and exhaust gas at these mean temperatures are given in the Table 3.4 and Table 3.5 respectively.

Table 3.4: Thermal and physical properties of air at $(T_{ao} + T_{ai})/2 = 40^\circ\text{C}$
(Kothandraman and Subramanya, 2007, p. 41).

S/N	Property	Symbol	Unit	Values
1	Specific heat capacity	C_p	J/kg K	1005
2	Thermal conductivity	K	W/m K	0.02756
3	Prandtl No.	Pr		0.669
4	Density	ρ	kg/m ³	1.128
5	Viscosity	μ	Ns/m ²	19.12×10^{-6}

Table 3.5: Thermal and physical properties of exhaust gas at $(T_{go} + T_{gi})/2 = 104^\circ\text{C}$
(Kothandraman and Subramanya, 2007, p. 43).

S/N	Property	Symbol	Unit	Values
1	Specific heat capacity	C_p	J/kg K	1068
2	Thermal conductivity	K	W/m K	0.03128
3	Prandtl No.	Pr		0.69
4	Density	ρ	kg/m ³	0.95
5	Viscosity	μ	Ns/m ²	20.38×10^{-6}

The value of μ_w for air and for exhaust gas at 72°C is found to be 20.98×10^{-6} and 18.83×10^{-6} Ns/m² respectively (Kothandraman and Subramanya, 2007, p. 43).

In order to determine the tube side heat transfer coefficient, the Reynolds number must first be evaluated. It will also assume that the flow is fully developed; therefore, the entry region does not need to be considered (Abd and Naji, 2017).

$$Re_i = \frac{(Ms \times Di)}{\mu} \quad (\text{Eq. 3.9})$$

Where; Ms = Mass velocity of shell (kg/m² s)

$$Ms = \dot{m}_g / \text{tubetotalarea} \quad (\text{Eq. 3.10})$$

Di = Tube inner diameter (m)

μ = Absolut viscosity of exhaust gas at 104°C (Ns/m²)

Re = Reynolds number

The Nusselt number of the tube side is calculated using equation (Eq. 3.11) (Azad Abazar and Azad Nader, 2016). A turbulent flow is present and therefore the turbulent equations should be used. Eq. (3.11) will be used in determining the average Nusselt number and in the calculation, it will be assumed that the viscosity factor $(\mu/\mu_w)^{0.14}$ is one.

$$Nu_i = 0.023 \times Re_i^{0.8} \times Pr_i^{0.3} \times \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (\text{Eq. 3.11})$$

Where; Nu = Nusselt number

Pr = Prandtl number

Now, the convective heat transfer coefficient of the tube side is calculated using (Eq. 3.12) (Azad Abazar and Azad Nader, 2016).

$$h_i = Nu_i \times K_i / d \quad (\text{Eq. 3.12})$$

Where; h_i = Tube side convective heat transfer coefficient (W/m² K)

K = Thermal conductivity (W/m K)

In order to determine the shell-side heat transfer coefficient, the Reynolds number must first be evaluated. To complete the definition of the Reynolds number, Kern uses an equivalent diameter for the shell. The equivalent diameter for a square pattern is determined using Eq. 3.13 (Cao, 2010, p. 175).

$$De = \frac{4 \times (p_t^2 - \pi d_o^2 / 4)}{\pi d_o} \quad (\text{Eq. 3.13})$$

Where; De = The equivalent diameter of shell (mm)

p_t = Tube pitch (mm)

d_o = Outer diameter of tube (mm)

Then the shell Reynolds number is defined as (Cao, 2010, p. 175):

$$Re_o = \frac{D_e \times Ms}{\mu} \quad (\text{Eq. 3.14})$$

The Nusselt number and convection heat transfer coefficient of the shell side is calculated using Eq. 3.14a and 3.14b (Azad Abazar and Azad Nader, 2016).

$$Nu_o = 0.364 \times (Re_o^{0.55}) \times (Pr_o^{0.33}) \quad (\text{Eq. 3.15a})$$

$$h_o = Nu_o \times K_o / d_o \quad (\text{Eq. 3.15b})$$

It assumed that the thickness of the tubes, and fouling effects is negligible. so overall heat transfer coefficient calculated as follow:

$$U_t = A \times \left[\frac{1}{h_i} + \frac{1}{h_o} \right]^{-1} \quad (\text{Eq. 3.16})$$

In order to determine the pressure drop along the tubes, the friction factor must first be evaluated. For heat exchanger tubes, the friction factor in the turbulent region can be determined with the (Eq. 3.17) (Bichkar, et al., 2018).

$$f = 0.0014 + 0.125 \times Re_i^{-0.32} \quad (\text{Eq. 3.17})$$

Then the pressure drop along the tubes can be calculated with the Fanning equation:

$$\Delta p_i = 4f \times \frac{L}{d_i} \times \frac{Ms^2}{2\rho} \times \left[\frac{\mu}{\mu_w} \right]^{-0.14} \quad (\text{Eq. 3.18})$$

Where; Δp_i = Pressure drop in the tube fluid

f = Friction factor

According to Kern, the pressure drop in the shell fluid is proportional to the number of times the fluid crosses the tube bundle. If N_B is the number of baffles, the shell fluid crosses the bundle $N_B + 1$ times. The pressure drop is also proportional to the length of the path at every bundle cross, which can be represented by the shell diameter (Cao, 2010, p. 176). Thus

$$\Delta p_o = f \times \frac{(N_B + 1) \times D_i}{D_e} \times \frac{Ms^2}{2\rho} \times \left(\frac{\mu_w}{\mu} \right)^{0.14} \quad (\text{Eq. 3.19})$$

Where: Δp_o = Pressure drop in the shell fluid

$$f = 1.728 \times Re_o^{-0.188}$$

D_i = Internal diameter of shell (mm)

The heat exchanger is suitable for the required service since both the tube-side and shell-side calculated Δp values are lower than allowable values. Generally, the overall dimension of designed heat exchanger is given in Table 3.6 below.

Table 3.6: Technical specifications of intake air preheaters

Parameters	Dimension (mm)
Shell outer diameter, D_o	124 mm
Shell wall thickness, s_w	3.24 mm
Tube outer diameter, d_o	31 mm
Tube wall thickness, t_w	2.62 mm
Length of tube, L	400 mm
Number of tubes, N	5 no's
Number of Baffles, N_B	6 no's
Central Baffle space, L_B	55 mm
Baffle cut, B_c	31 mm
Pitch of the tubes, P_t	38.75 mm
No. of shell passes, n_s	01 no's
No. of tube passes, n_t	01 no's

3.4 Heat Exchanger Manufacturing Process

The designed intake air preheater heat exchanger device is manufactured from mild steel and copper tube materials by using rolling, welding, cutting, and drilling machines. During manufacturing the heat exchanger, the following operations were performed: **Cutting:** cutting was conducted to divide the mild steel into a number of circular and rectangular pieces which are used to produce baffles plates, tube sheets, and shell as per the dimensions of the design requirement. **Drilling:** the mild steel baffle plates and tube sheets are drilled by a diameter of 31mm at square pattern. **Rolling:** rolling operation is executed to roll different pieces of plate mild steel prepared by cutting, into round with 110mm diameter and conical shapes with a larger diameter of 110mm and 40mm smaller diameter. **Welding:** welding is performed to join baffle plates with tubes and shell, and shell with extender components of the exchanger. Finally, **the surface finishing** of the welded surface was performed by using glass paper. The overall heat exchanger manufacturing process is described in Figure 3.3 below.



Welding of baffles and tubes



Covering the central part by shell



Frontal views of heat exchanger



Final manufactured heat exchanger

Figure 3.3: Heat exchanger manufacturing process

3.5 Transesterification of Cotton Seed Oil

The Transesterification process of cottonseed oil was making triglycerides of cotton seed oil to react with methyl alcohol(methanol) by using Potassium hydroxide (KOH) as catalyst to produce biodiesel and glycerol. This process was performed at Chemistry department laboratory of Adama Science and Technology University.

The stepwise transesterification process of producing biodiesel from cotton seed oil is mentioned below:

1. Firstly, the required cotton seed oil was prepared, then 500ml cotton seed oil was measured and poured in beaker and heated to temperature of 60°C (Figure 3.4).



Figure 3.4: Preheating of the oil

2. 100 ml of methanol and 5 gm of Potassium hydroxide (KOH) were measured to prepare the solution of potassium methoxide. The molar ratio of methanol to catalyst is 20:1. Then, shake the mixture until the KOH was fully dissolved in methanol (Figure 3.5(b)).



Figure 3.5: (a) KOH and methanol and (b) measured KOH and methanol

After Potassium hydroxide (KOH) was fully dissolved in methanol, the potassium methoxide solution poured into heated oil and the dissolved mixture was stirred with 1000 rpm stirrer speed for 1 hour at fixed temperature of 60°C. During this process the cotton seed oil and the potassium methoxide solution was covered by Aluminum foil to prevent escaping of methanol during process because its boiling point closer to 60-65 °C (Figure 3.6).



Figure 3.6: Heating the mixtures of KOH and methanol with oil

3. When the reaction time reaches 40 minutes' glycerin starts to separate and settle at bottom and a clear solution appears at the top.
4. After the reaction is finished the products were poured into separating funnel and hanged for 24 hours. Then two phases layer formed, the lower layer consists of glycerol and soap due to gravity and the biodiesel formed at the top (Figure 3.7). Then removing the glycerol and collecting the biodiesel.



Figure 3.7: Separation of biodiesel and glycerol

5. The next step was washing of biodiesel with the hot distilled water that was heated at 55 °C and the water equivalent to half volume of biodiesel. biodiesel washing was used to remove any excess of glycerol, soap, alcohols and catalyst that may be left in the biodiesel. This was done until the water is mostly clear that would see the biodiesel in the separating funnel in Figure 3.8(final washing).



First washing



Second washing



Third washing



Fourth washing



Fifth washing



Final washing

Figure 3.8: Washing of biodiesel

6. Finally, the washed biodiesel was dried by placing on the hot plate for 1 hour about 140 °C to remove remaining alcohol and water in the biodiesel. Figure 3.9 shows biodiesel before drying and after drying.



a) Before drying

b) after drying

Figure 3.9: Drying of biodiesel

After washing, the final pure biodiesel was collected and yield of biodiesel was calculated. Generally following all the step by step listed above we got the pure biodiesel fuel shown in Figure 3.10 below.



Figure 3.10: The final collected pure biodiesel

The production process of cotton seed biodiesel is presented schematically in Figure 3.11.

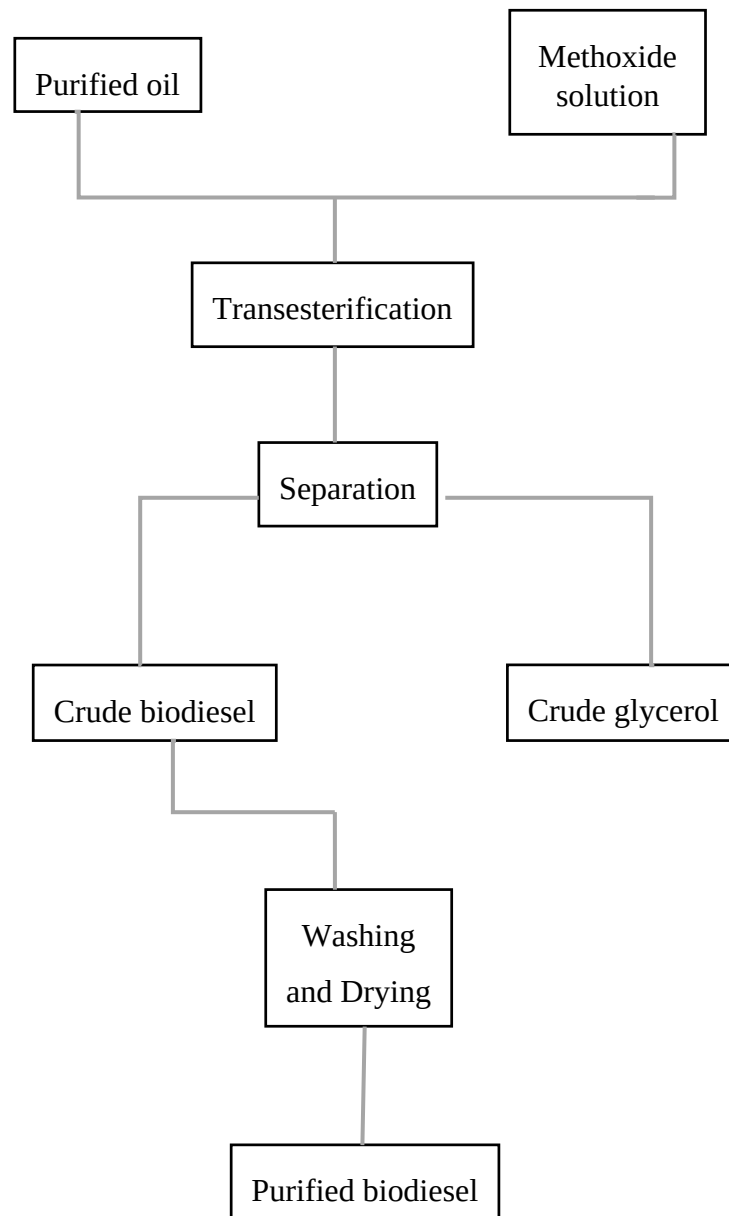


Figure 3.11: Conceptual flow chart of overall biodiesel production from cotton seed oil

3.6 Biodiesel Blend Preparation

As it is known a lot of research work has already been carried out to use vegetable oil both in a pure form and also blend forms. Studies show that the usage of vegetable oils in its pure form is possible but it preferable as the blended one. Because, biodiesel can be blended with any proportion B5(5% biodiesel and 95% petro diesel fuel) to B20(20% biodiesel and 80% petro diesel fuel) being the most common one without using any additives. Many researchers mentioned biodiesel blends at a range of B20 and lower can work on engine without modification of engine mechanical parts and without using any fuel additives. In this study the method used to blend the cotton seed oil biodiesel with petro diesel fuel and ethanol is splash mixing type rather than inline mixing type because, biodiesel is slightly heavier than

conventional diesel fuel and ethanol. The specific gravity of ethanol is less than fossil diesel and biodiesel. This allows splash blending by adding biodiesel on top of ethanol and diesel fuel for making blends in form of B20E10(20% biodiesel, 10% ethanol and 70% petro diesel fuel), B30E10(30% biodiesel, 10 ethanol and 60% diesel fuel) and B40E10(40% biodiesel, 10% ethanol and 50% petrol diesel fuel) shown in Figure 3.12. First ethanol was blended with diesel. Biodiesel should always be blended at top of diesel fuel blended with ethanol. If diesel fuel blended with ethanol is added at top of biodiesel, it will not mix.



Figure 3.12 (a) Measured diesel and biodiesel and (b) Diesel-biodiesel blend

3.7 Characterization

Characterization is the process of determining physiochemical properties of petroleum products and non-petroleum products like biodiesel and diesel-biodiesel blends. Before testing the engine performance and exhaust emissions it is very important to check physiochemical properties of biodiesel and diesel-biodiesel blends. The properties of fuel include density, kinematic viscosity, cloud point, calorific value and total acidity.

Fuel prepared for characterization is B100(pure biodiesel), B20(80% Petro diesel and 20% biodiesel), B20E10 (70% Petro diesel and 20% biodiesel with addition of 10% ethanol), B30E10 (60% Petro diesel and 30% biodiesel with addition of 10% ethanol) and B40E10 (50% Petro diesel and 40% biodiesel with addition of 10% ethanol) were shown Figure 3.13 below.



Figure 3.13: Biodiesel and diesel-biodiesel blend ready for characterization

The above-mentioned physiochemical properties of B100, B20 and B40 were performed in quality control laboratory of Ethiopian Petroleum Supply and Enterprise while physiochemical properties of B30 were performed in chemistry laboratory of Adama Science and Technology University. Furthermore, some of the physiochemical properties of B20E10, B30E10 and B40E10 were determined by using Eq. 3.20 and Eq. 3.21.

3.8 Experimental Set up and Procedure

The experimental test rig consists of a single-cylinder, four-stroke, naturally aspirated, direct injection, and computer-controlled test bench diesel engine TBM3-02. It is equipped with an eddy current asynchronous motor dynamometer (AM-1) for loading the engine. The setup also includes a fuel supply system, water cooling, and lubrication system and exhaust emissions. The engine produced a maximum power of 2.2 kW and a maximum torque of 7.4 N-m. The schematic and photographic representation of the experimental setup is illustrated in Figure 3.14 and Figure 3.15 respectively. The setup enables the evaluation of performance parameters and emission constituents of the diesel engine. The engine performance parameters include P_b , BSFC and BTE. The technical specification of a test engine is shown in Table 3.9 below. The engine exhaust emissions levels are measured using an SV-50 automotive emission analyzer. The CO, CO₂, emissions were measured in percentage volume (%Vol), and the HC and NO_x emissions were measured in parts per million (ppm) during each run of the engine.

The experimental investigation was done on three different biodiesel percentage ratios and two elevated intake air temperatures. The timing and duration of the injection of diesel were controlled by a factory engine electronic control unit (ECU) in which none of the programmed values was modified. Besides, all the sensors located on the engine were completely wired to the ECU by following the factory setting. In the engine, providing a compression ratio of 21:1, the nozzle opening pressure (190 bar) and the standard of the injection timing 15° bTDC.

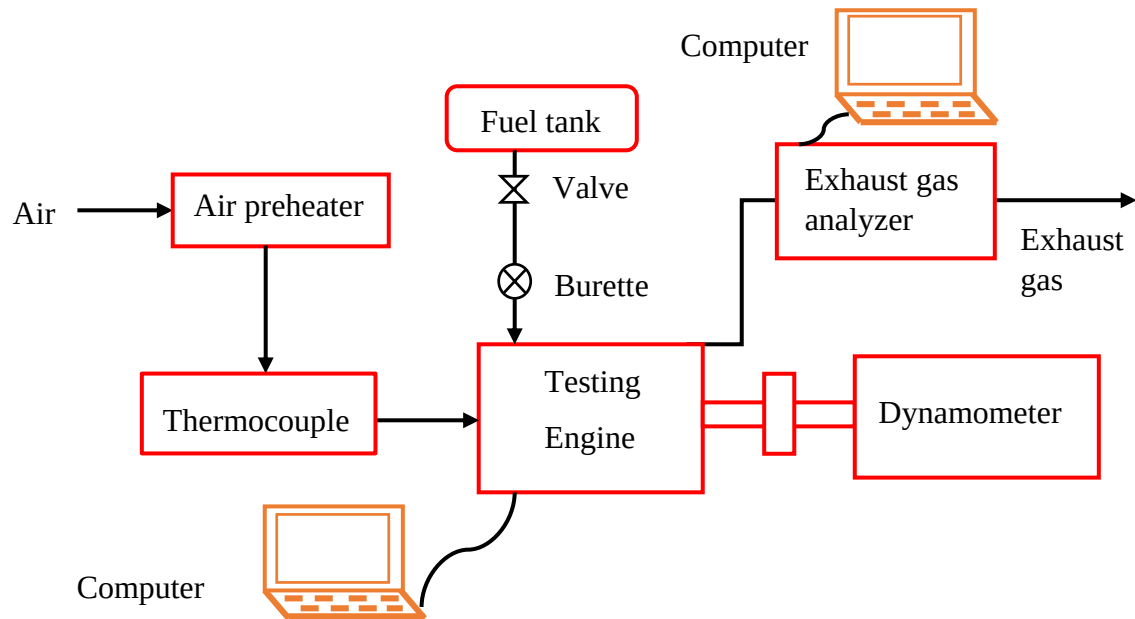


Figure 3.14: Schematic experimental setup of the test engine

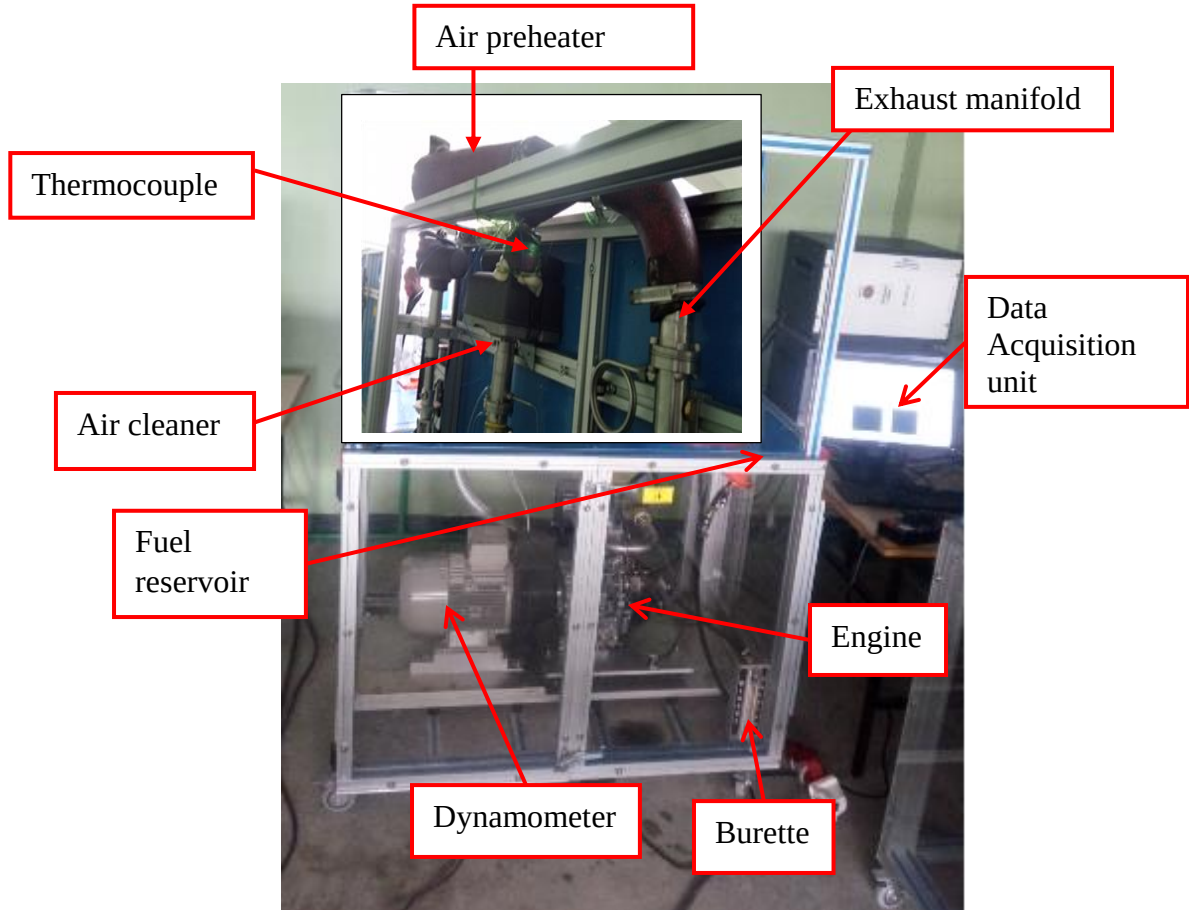
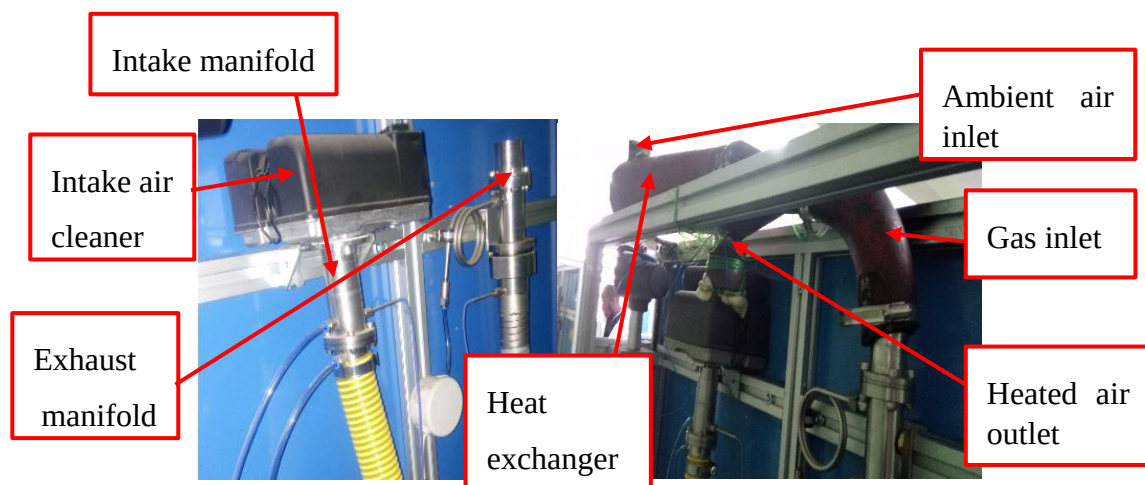


Figure 3.15: Photographic experimental setup of the test engine.

The engine was first operated on diesel fuel at ambient intake air temperature with no load for few minutes at rated speed of 1750 rpm, which is set fixed on the computer. All the tests were taken after the engine reached its operating temperature. The diesel engine is tested

initially with diesel fuel for baseline data for each load condition according to the baseline test matrix shown in Table 3.10. The baseline parameters were obtained at the rated speed by varying 0 to 80% of load on the engine with an increment of 20% and as the load varies consequently the brake power varied and displayed on the data acquisition computer. The diesel fuel was replaced with the diesel-biodiesel blends with ethanol addition and test was conducted by varying 0 to 80% of load on the engine with an increment of 20%. The volumetric proportions of diesel, biodiesel and ethanol in blend fuel mode was demonstrated with B20E10, B30E10, and B40E10 as the blending test matrix is shown below in Table 3.11. Then the performance and emissions of the engine were tested and the values were recorded with diesel fuel, B20E10, B30E10, and B40E10 for each load conditions at a constant engine speed of 1750 rpm operating at ambient intake air temperature. In each experiment, the various emissions concentration such as CO, HC, CO₂ and NO_x emissions were measured using an exhaust gas analyzer, and the fuel flow rate was also recorded using the fuel burette as shown in Figure 3.17 below. The performance parameters of BSFC and BTE were calculated by using equation (3.20 and 3.21).



a) Before installing heat exchanger b) After installing heat exchanger

Figure 3.16: Installation of heat exchanger at the air intake and exhaust manifolds.

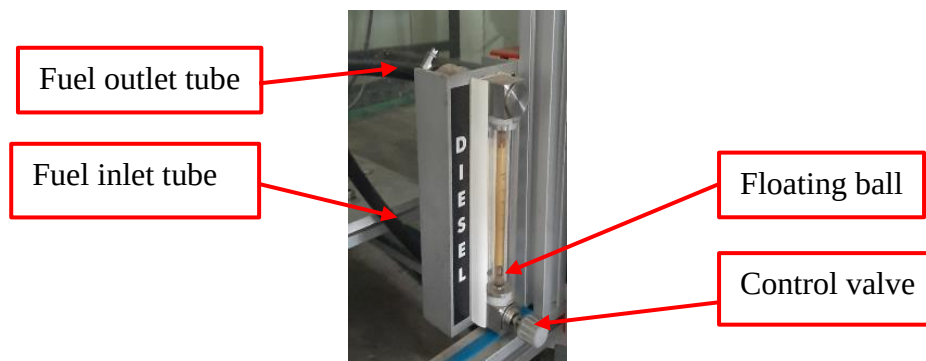


Figure 3.17: Flow meter(Burette).

After various experiments have carried out for baseline data at ambient intake air temperature, the counter flow shell and tube heat exchanger is installed at the intake manifold before air cleaner and exhaust manifold as shown above in Figure 3.16b for preheating intake air. After installing the heat exchanger, the engine restarts with diesel-biodiesel blends with addition of ethanol and runs for few minutes until intake air temperature becomes constant. As the engine operates with diesel-biodiesel blends with addition of ethanol and installed heat exchanger, the intake air extracts heat from exhaust gas in such a way that the exhaust gas flows through the tubes of the heat exchanger and the intake air flows in the shell in opposite direction of exhaust gas as shown in Figure 3.2 which enables them to exchange heat. Preheated air was approximately 51°C with a deviation of about 2°C due to thermocouple precision, while ambient air was approximately 25°C. Then various experiments were also executed again once the counter flow shell and tube heat exchanger is installed by running the engine with blend fuel mod as shown below in blending test matrix in Table 3.11 following the same procedure of baseline data recording, performance analysis, load variation and etc as explained above. Finally, the engine performance and emission parameters were plotted against varying engine loads.

The ethanol (97% pure) used in these tests was bought from FIRMAMENT TRADING PLC chemical selling and suppliers' company in Ethiopia. Fuel properties that are important from engine performance and emission point of view such as density, viscosity, net heating value, flash point and cetane index of diesel and ethanol is shown in the Table 3.7.

Table 3.7: Properties of the base fuel (Gopal et al., 2017).

S. no.	Fuel property	Diesel	Ethanol
1	Cetane index	39	14
2	Viscosity (mm ² /s) at 40°C	1.3	1.1
3	Calorific value (kJ/kg)	42,500	28,180
4	Flash point °C	53	12
5	Density (kg/m ³) at 15°C	824	799.4

Density and Calorific values of B20E10, B30E10 and B40E10 are determined using equation 3.20 and equation 3.21. Table 3.8 presents composition and calculated property values of blended fuels.

Table 3.8: The composition and fuel properties of tested fuels.

Fuels	Unit	B20E10	B30E10	B40E10
Diesel	%(v/v)	70	60	50
Cotton seed biodiesel	%(v/v)	20	30	40
Ethanol	%(v/v)	10	10	10
Density at 20°C	kg/m ³	820.5	824.5	828.5
Calorific value	MJ/kg	39.16	38.9	38.6

The equations used to predict the density of ethanol-diesel-biodiesel blends were based on a Kay's mixing rule, in which the density of the blends was determined using the individual characteristics of the components of the blends (Razzaq et al., 2020).

$$\rho_{blend} = \rho_D \times D + \rho_{BD} \times BD + \rho_{EtOH} \times EtOH \quad (\text{Eq. 3.20})$$

where ρ_{blend} is the predictable density of the blend, ρ_D , ρ_{BD} , ρ_{EtOH} are the density of the diesel, biodiesel, and ethanol, respectively, and D, BD, EtOH are the mass fraction or volume fraction of diesel, biodiesel, and ethanol, respectively.

The impact of biodiesel and ethanol additions on the heating value (HV) of diesel was predicted for different ethanol-diesel-biodiesel blends using the following formula (Al-Esawi et al., 2019):

$$HV_{blend} = (\nu_D \times HV_D \times \rho_D + \nu_B \times HV_B \times \rho_B + \nu_E \times HV_E \times \rho_E) / \rho_{blend} \quad (\text{Eq. 3.21})$$

Where HV_D , HV_B and HV_E refer to the heating values (MJ/Kg) of diesel, biodiesel and ethanol respectively: and ν_D , ν_B and ν_E refer to the volume fractions of diesel, biodiesel and ethanol respectively. ρ_D , ρ_B , ρ_E and ρ_{blend} refer to the densities of diesel, biodiesel, ethanol, and their blend, respectively.

Table 3.9: Test engine TBMC3-02 technical specification.

Engine model	TM3-02
Engine type	Compression-Ignition
Fuel type	Diesel oil
Number of cylinders	1
Type of air intake	Naturally aspirated
Cooling type	Air
Compression ratio, CR	21:1
Bore diameter	69 mm
Stroke	60 mm
No. of stroke	4 stroke
Torque	7.4 N m
Engine power	2.2 kW
Internal dia. of the air inlet manifold (D)	36 mm

3.8.1 Experimental Test Matrix

An experimental test matrix is the number of tests that were being executed and guidance for experimental work. Baseline and Blending mode experimental test matrix are listed in Table 3.10 below.

Table 3.10: Experimental test matrix for blending and Baseline data.

Types work	Engine operating parameters				Test fuel	Evaluating Engine Parameters
	Compression ratio (CR)	Injection timing (IT)	Load	Air preheating temperature		
Baseline test	Default setting (CR=21:1)	Default setting (IT=15° bTDC)	0%-80% in a step of 20%	Without preheating (ambient temperature, 25° C)	Pure diesel fuel	(a) Performance parameters: i. Brake power (P_b) ii. BSFC iii. BTE (b) Emission parameters: i. CO Emission ii. CO ₂ Emission iii. HC Emission iv. NOx Emission
				Without preheating (ambient temperature, 25° C)	B20E10, B30E10, B40E10	(a) Performance parameters: i. Brake power (P_b) ii. BSFC iii. BTE (b) Emission parameters: i. CO Emission ii. CO ₂ Emission iii. HC Emission iv. NOx Emission
Major experiments	Default setting (CR=21:1)	Default setting (IT=15° bTDC)	0%-80% in a step of 20%	With preheating at Elevated temperature (51°C)	B20E10, B30E10, B40E10	(a) Performance parameters: i. Brake power (P_b) ii. BSFC iii. BTE (b) Emission parameters: i. CO Emission ii. CO ₂ Emission iii. HC Emission iv. NOx Emission

3.9 Performance Parameter Analysis

The performance of the diesel engine such as brake power, brake specific fuel consumption and brake thermal efficiency was evaluated by fueled the engine; diesel fuel, B20E10, B30E10 and B40E10 at ambient and elevated air preheated temperatures with different engine load conditions.

i) Brake Power (P_b)

One of the most important metrics for engine performance is output power. Brake power is the actual engine power output at the output shaft of the engine in kW. This power is related to the torque applied to the crankshaft. Similar to brake torque, the brake power is also displayed on the computer during the engine runs and its value changes as the engine loads changes.

ii) Brake Thermal Efficiency (BTE)

Brake thermal efficiency is the ratio of output brake power to the energy supplied by the fuel and it is calculated as (Teoh et al., 2019):

$$BTE(\%) = \frac{3600 \times P_b}{\dot{m}_d \times HV_f} \times 100 \quad (Eq. 3.22)$$

Where; BTE = brake thermal efficiency(%)

$\dot{m}_d$ = mass flow rate of fuel in kg/h

HV_f = Heating value of fuel (MJ/kg)

i) Brake Specific Fuel Consumption (BSFC)

Specific fuel consumption is the measure of the engine's ability to produce power by utilizing the energy content of the fuel. The specific fuel consumption of an engine should be minimized for the desired power output to optimize engine performance (Sandalcı, et al. , 2019). Brake specific fuel consumption is the ratio of the mass flow rate of fuel consumed by the engine to the brake power in which the engine is produced. The BSFC is calculated as (Ambarita, 2017):

$$BSFC = \frac{\dot{m}_d}{P_b} \quad (Eq. 3.23)$$

Where; BSFC = brake specific fuel consumption in kg/kWh

3.10 Emission Parameters and Measuring Procedure

The emission parameters of diesel engine such as CO, CO₂, HC, and NO_x were measured by using an exhaust gas analyzer when the engine operates in diesel only and blended fuel mode at each load conditions. During the experiments, exhaust gases were measured using Automobile Exhaust Gas Analyzer SV-50 (which is found in Federal TVET Institute, Ethiopia) as shown below in Figure.3.18 and its measuring range is listed in Table 3.11 below.



Figure 3.18 Automobile exhaust gas analyzer SV-50

Table 3.11: Measurement range of SV-50 automobile exhaust gas analyzer

species	Measurement range	Resolution	Allowed error	Relative error
HC	0 - 1000 ppm	1 ppm Vol.	± 12 ppm Vol.	$\pm 5\%$
CO	0 - 10.0 in % Vol.	0.01% Vol.	$\pm 0.06\%$ Vol.	$\pm 5\%$
CO ₂	0 - 20.0 in % Vol.	0.1% Vol.	$\pm 0.5\%$	$\pm 5\%$
NO _x	0 – 800 ppm	1 ppm	± 10 ppm	± 5

Measuring Procedure

Initially connect (plugin) the exhaust gas analyzer from the power source and turn it ON. Since the probe and the sampling tube were connected to the gas analyzer. Then the probe tip was closed by plastic for at least 10 minutes before starting measurement. After 10 minutes remove the enclosed plastic and insert its probe tip into the engine exhaust pipe. Some portion of exhaust gases was taken by the gas analyzer through the sampling tube and sensing it and then measurement was started by pressing the record icon on the control unit of the analyzer. The content of the exhaust gas was started to display on the gas analyzer control unit display. The readings of HC (in ppm), NO_x (in ppm), CO₂ (in % Vol.), and CO (in % Vol.) for each test were allowed to stabilize from 2 to 5 minutes and the results were recorded.

3.11 Experimental Uncertainties

Uncertainty is a measure of the ‘goodness’ of a result. Without such a measure, it is impossible to judge the fitness value. There are a number of physical and operational parameters that causes some degree of uncertainty during working with an instrument.

Hence, an uncertainty or error analysis is of prime importance for establishing the bounds on the accuracy of the estimated parameters. To this end, a comprehensive error estimation was conducted by means of universally known theory of sequential perturbation technique (Paul et al., 2017). The method is described as follows. If N is a dependent measuring parameter which is a function of the independent variables $x_1, x_2, x_3, x_4, \dots, x_n$. Therefore,

$$N = N(x_1, x_2, \dots, x_n) \quad (\text{Eq. 3.24})$$

If ΔN is the uncertainty created due to the individual uncertainties of the independent parameters termed as $\Delta N_1, \Delta N_2, \Delta N_3, \Delta N_4, \dots, \Delta N_n$. Then the uncertainty of the dependent variable can be written as:

$$\Delta N = \left[\left(\frac{\partial N}{\partial x_1} \Delta N_1 \right)^2 + \left(\frac{\partial N}{\partial x_2} \Delta N_2 \right)^2 + \dots + \left(\frac{\partial N}{\partial x_n} \Delta N_n \right)^2 \right]^{1/2} \quad (\text{Eq. 3.25})$$

The overall measurement uncertainties of emission constituents are tabulated in Table 3.12.

Uncertainty in Emission Constituents

Uncertainty in emission constituents are obtained by computing its Resolution/Range.

$$\begin{aligned} \Delta \text{CO} / \text{CO} &= (\text{Resolution/range})_{\text{CO}} \times 100 \\ \Delta \text{CO}_2 / \text{CO}_2 &= (\text{Resolution/range})_{\text{CO}_2} \times 100 \\ \Delta \text{HC} / \text{HC} &= (\text{Resolution/range})_{\text{HC}} \times 100 \\ \Delta \text{NO}_x / \text{NO}_x &= (\text{Resolution/range})_{\text{NO}_x} \times 100 \end{aligned} \quad (\text{Eq. 3.26})$$

Table 3.12: Overall measurement uncertainty for emission parameters.

Emission parameters	Diesel mode uncertainty (%)	Blend mode at 25°C uncertainty (%)	Blend mode at 51°C uncertainty (%)
CO emission	±0.1	±0.1	±0.1
CO ₂ emission	±0.5	±0.5	±0.5
HC emission	±0.05	±0.05	±0.05
NO _x emission	±0.02	±0.02	±0.02

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Introduction

In this chapter the efficiency of shell and tube heat exchanger and biodiesel yields are presented. This chapter also discusses about the experimental work analysis on performance and emission characteristics of diesel engine running with diesel-biodiesel blends with addition of ethanol. The effect of preheating intake air, variation of biodiesel percentage and load in diesel engine performance and emission characteristics is also explained in this chapter.

4.2 Heat exchanger Efficiency

The heat exchanger was designed to give an output temperature of 55°C. However, practically the output temperature is found to be 51°C which in turn leads the heat exchanger to have overall efficiency of 93.09%. This is due to many factors such as manufacturing processes, the materials used to manufacture may not be as recommended in the design, and others. Among all factors the heat loss from the exhaust gas before it reaches to the heat exchanger is the main one.

4.3 Biodiesel Yields

The biodiesel production was done by transesterification process using methanol and KOH. During the transesterification reaction the oil to methanol molar ratio and methanol to KOH was 5:1 and 20:1 respectively. After washing and drying the final pure biodiesel was collected and yield of biodiesel obtained was 85%. The amount of biodiesel obtained and byproducts of biodiesel production in one liter of cotton seed oil was listed in the Table 4.1 below.

Table 4.1: Biodiesel yield and its byproducts

Oil	Quantity used	Methanol used	KOH used	Biodiesel obtained	Byproducts obtained	Biodiesel yield (%)
Cotton seed	1000 ml	200 ml	10 gm	850 ml	350 ml	85%

4.4 Characterization Results

Characterization was done to know the properties of the biodiesel produced whether it satisfies the ASTM standards for biodiesel to test its performance in the existing diesel engines without doing modification. The characterization results of the biodiesel produced from cotton seed oil and its blends with diesel were certified from Ethiopian petroleum supply enterprises shown in the Table 4.2. From Table 4.2, it is observed that the density of B100 was increased while B40, and B20 were decreased. These density results were within the range given by Europe standard (EN 14214). The viscosity of B100 was within the range given by ASTM standards (D6751) and the viscosity of B20, and B40 were nearly same with diesel fuel which also satisfies ASTM standards (D6751).

Table 4.2: Characterization results of biodiesel and its blends

Sn	Properties	Test method ASTM	BD limits	Results		
				B100	B20	B40
1	Density at 15°C(g/mL)	D4052	Report	0.8886	0.8459	0.8514
	Density at 20°C(g/mL)	D4052	Report	0.8851	0.8424	0.8479
2	Calorific value (BTU/LB)	Calculated	-	19,172.6	19,556.9	19,507.4
3	Cloud point(°C)	D2500	-	+10	+1	+8
4	Kinematic viscosity at 40°C(c St)	D445	-	5.2829	3.0578	3.1871
5	Total acidity (mg KOH/g)	D974	-	0.890	0.3667	0.400

In general, it can be concluded that these properties of biodiesel produced and its blends with diesel found to be within the range of ASTM standards (D6751) and Europe standard (EN 14214).

4.4.1 Fuel Properties

Fuel properties are the sole important reason for the variance in combustion characteristics of the test fuels. Key properties that has vast influence on combustion are density, kinematic viscosity, heating value, Cetane number, Flash point, Oxygen content etc. Transesterified

Cotton seed biodiesel was compared with ASTM biodiesel standards which showed closer values proving no significant change needed for the prevailing engine.

4.4.1.1 Kinematic Viscosity

The kinematic viscosity (ν) of cotton seed oil biodiesel, diesel, diesel-biodiesel blends and diesel-biodiesel blends with ethanol addition is shown in Figure 4.1. The test was carried at a temperature of 40°C. Slight increase in kinematic viscosity was found with diesel-biodiesel blend fuels with increase in biodiesel volume in comparison to diesel fuel. The increase in KV was due to the higher kinematic viscosity of biodiesel. A decreased kinematic viscosity was found with diesel biodiesel blend fuels with ethanol addition due to the lower KV of ethanol. However, the KV of diesel-biodiesel blend and diesel-biodiesel blend fuels with ethanol addition were lower in comparison with cotton seed oil biodiesel (5.28 cSt).

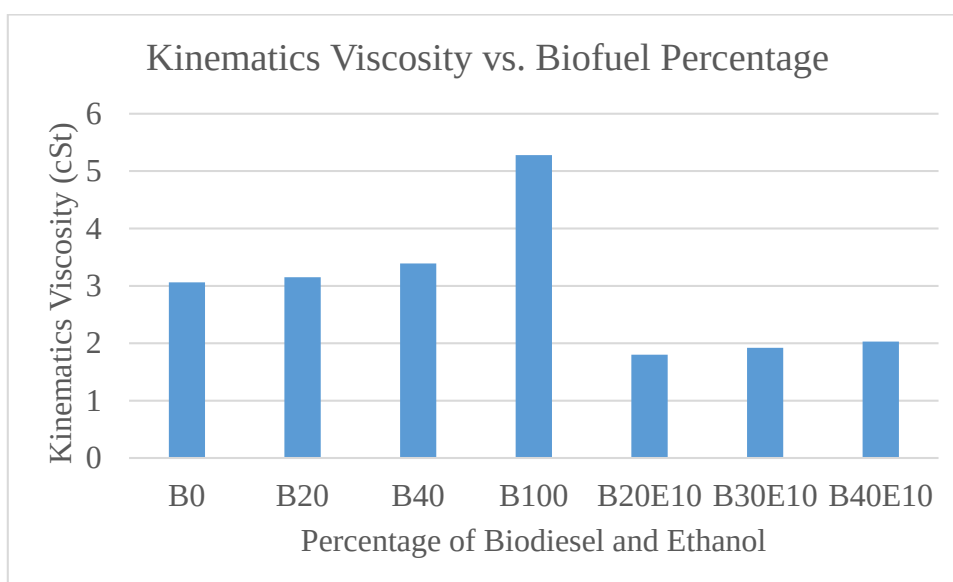


Figure 4.1: Kinematic viscosity of test samples.

4.4.1.2 Density

The density of diesel, cotton seed oil methyl ester, blends of diesel-cotton seed oil biodiesel and blends of diesel-cotton seed oil biodiesel with addition of ethanol is as shown in Figure 4.2. The test was carried out at 15°C. From Figure 4.2 it can be seen that the density of the fuel blends was increased with increase in biodiesel volume compared to fossil diesel. This is due to the higher density of biodiesel. However, there is a slight decrease in density of fuel blends with the addition of ethanol in the fuel blends due to the lower density of ethanol (0.7996 g/m³).

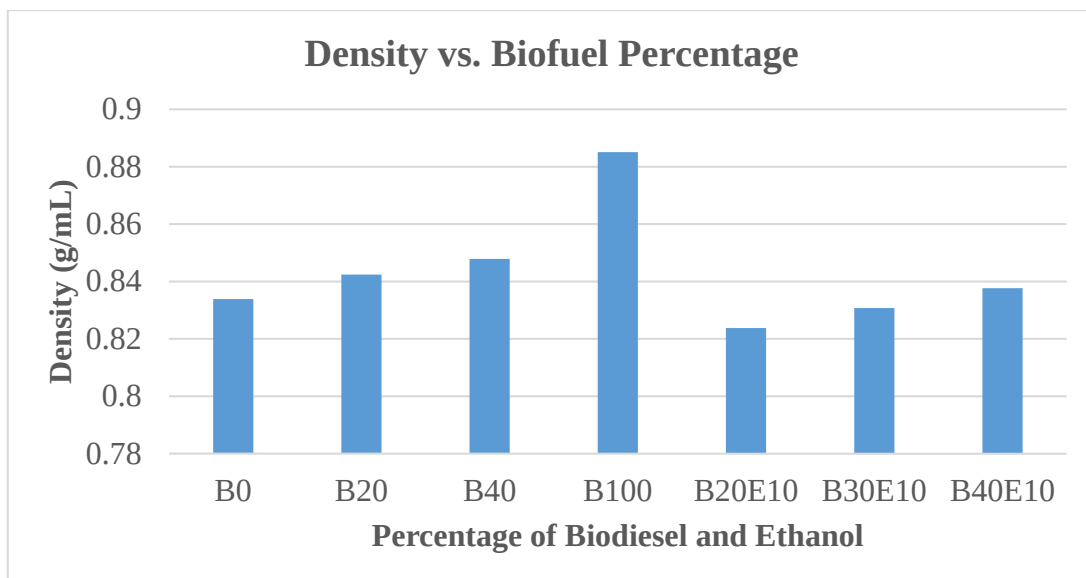


Figure 4.2: Density of test samples.

4.4.1.3 Calorific Value

The calorific value (CV) of diesel, cotton seed biodiesel, blends of diesel-cotton seed oil biodiesel and blends of diesel-cotton seed oil biodiesel with addition of ethanol is as shown in Figure 4.3. The CV of fuel blends were found to be decreased with increase in biodiesel percentage and addition of ethanol in the fuel blends due to the lower calorific values of the cotton seed oil biodiesel (40646.58 kJ/kg) and ethanol (29700 kJ/kg) in comparison with CV of diesel fuel (43225.6 kJ/kg).

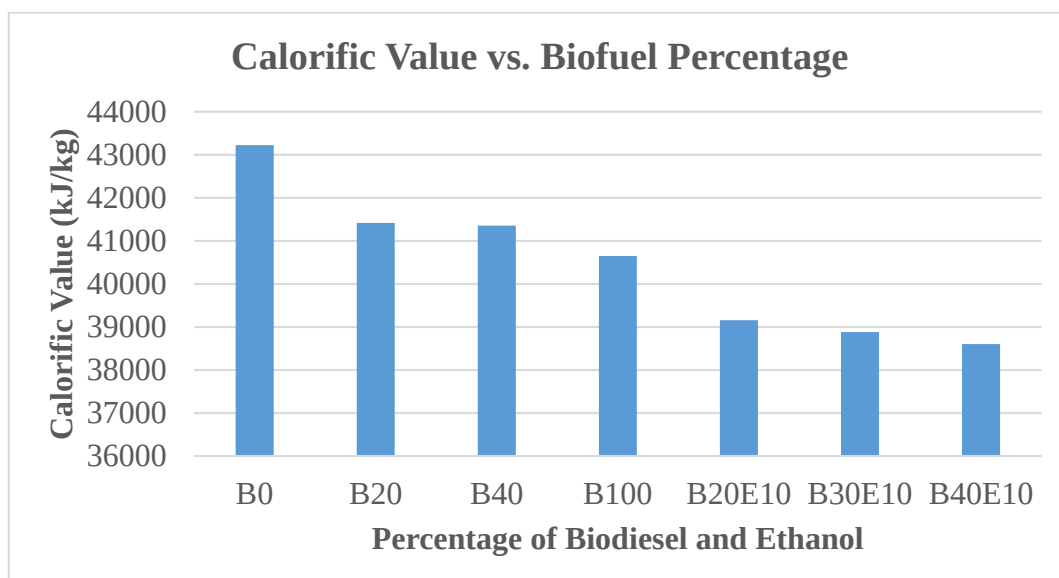


Figure 4.3: Calorific value of test samples.

4.5 Engine Performance and Emission Characteristics

The results pertaining to performance and emission characteristics of the engine are demonstrated with the help of graphs. The variation of performance characteristics such as brake thermal efficiency, brake specific fuel consumption and brake power and emissions parameters such as CO, CO₂, NO_x and HC with respect to various loading conditions for diesel fuel and diesel-biodiesel blends with addition of ethanol at ambient and elevated intake air temperatures are discussed in the following section.

4.5.1 Engine Performance

4.5.1.1 Brake Power (P_b)

The variation of brake power versus engine load for diesel and biodiesel-diesel blended fuels with ethanol addition at different temperatures (ambient temperature, i.e. 25°C and preheated air at 51°C) is shown in Figure 4.4 and Table 4.3 show its numerical values.

Table 4.3: Brake power for all tests.

Load (%)	Brake Power (kW)						
	Diesel	B20E10 (wopa)	B30E10 (wopa)	B40E10 (wopa)	B20E10(wpa=51°C)	B30E10(wpa=51°C)	B40E10(wpa=51°C)
20	0.85	0.8	0.85	0.81	0.85	0.85	0.85
40	1.3	1.32	1.25	1.22	1.25	1.3	1.3
60	1.82	1.75	1.75	1.75	1.75	1.78	1.8
80	2.1	2	1.95	2.01	2.01	2.02	2.03

The brake power is defined as the rate at which work is done by the engine. The average percentage in engine brake power values change with respect to load for all tests as recorded in table 4.3. Properties of blended fuels, especially heating value, viscosity and lubricity, have a significant effect on brake power of the engine. It is observed that variation of brake power with load is approximately linear irrespective of fuel used and no significant variation in brake power is evident due to the use of different fuels injected. From Figure 4.4 it is clearly observed that the trend of the graph is almost similar for all fuel samples. In general, as engine load increases, brake power is also increasing up for all tested fuels. The brake power for diesel and B20E10 is almost equal at ambient intake air temperature. However,

B30E10 and B40E10 shows lower output brake power. This lower brake power is due to the lower calorific value of B30E10 and B40E10 compared to diesel and B20E10. This is due to the heat of combustion (MJ per kg or liter) is a measure of the amount of energy available to produce work. In general, in almost all loads, power resulted from pure diesel fuel is greater than the power of biodiesel-diesel blended with addition of ethanol fuels, though the B20E10 fuels has relatively equal power when the intake air is not preheated. Generally, the reason for the lower brake power of higher percentage of biodiesel at ambient in take air in diesel-biodiesel blended with addition of ethanol fuels compared to diesel can be attributed to their lower heating value, higher viscosity and density causing poor atomization of the fuel.

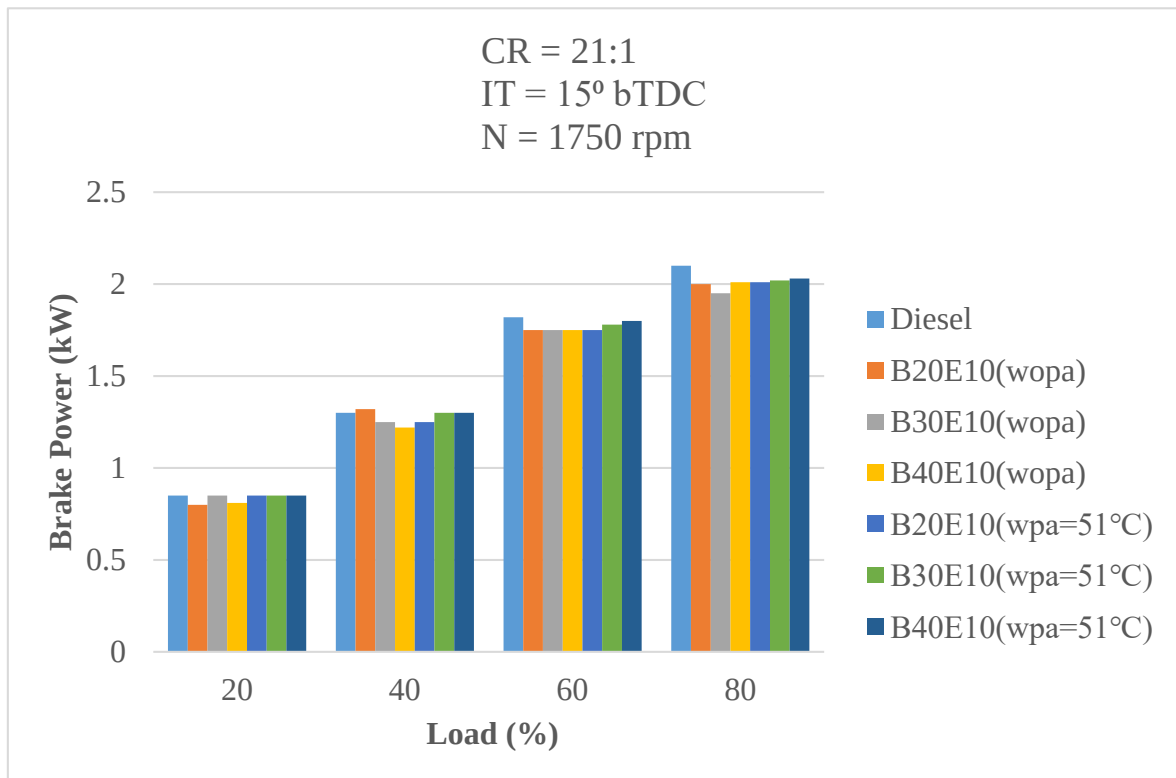


Figure 4.4: Variation of brake power with respect to engine load.

As observed in the test at preheated intake air, the brake power of B40E10 is higher than that of B30E10 and B20E10. The reason for increase in brake power for B40E10 is due to preheating intake air leads in decreases the ignition delay which in turn leads the available oxygen in the combustion chamber to be employed for combustion that results complete combustion of the fuel, in which higher energy conversion is found. From Figure 4.4, it is observed that preheating intake air has a substantial effect on fuel atomization which leads to have complete combustion and thus have higher brake power when compared to tests

without preheating intake air. Therefore, the average brake power reduction for B20E10, B30E10 and B40E10 in compared to diesel fuel at ambient intake air temperature is 1.97%, 4.61% and 4.76% respectively and the average percentage increased brake power of B20E10, B30E10, and B40E10 at preheated intake air of 51°C are 1.34%, 2.78%, and 3.25% respectively compared to the operation of the engine without intake air preheating (25°C).

4.5.1.2 Brake Thermal Efficiency (BTE)

BTE indicates the ability of the combustion system to accept the experimental fuel, and provides comparable means of assessing how efficient the energy in the fuel was converted to mechanical output (Dattatray et al., 2011). The effects of engine load on brake thermal efficiency (BTE) for various test fuels at different temperatures (ambient temperature, i.e. 25°C and preheated air at 51°C) are depicted in Figure 4.5 where BTE increases with an increase in load.

Table 4.4: Brake Thermal Efficiency for all tests.

Load (%)	Brake Thermal Efficiency(%)						
	Diesel	B20E10(wopa)	B30E10(wopa)	B40E10(wopa)	B20E10(wp a=51°C)	B30E10(wp a=51°C)	B40E10(wp a=51°C)
20	18.83	18.51	17.72	16.9	20	18	16
40	22.3	22.5	21.23	21.5	23.4	23	21
60	26.6	26.12	25.2	25	26.57	25	24
80	27.72	27.2	26.91	26.1	28.01	26	25

Almost similar BTEs have been observed for all blends and base diesel. Presence of the excess oxygen in cotton seed biodiesel and ethanol helps in better combustion which is the main reason for improved BTE for all blends. It also could be inferred due to the lower boiling point of ethanol which improves the spray quality of the blend. Due to spray quality of ethanol blend, complete combustion occurs at fuel-rich zones in the combustion chamber. Flame temperature of ethanol is lower than the neat diesel so that heat losses during the combustion were decreased and hence combustion efficiency enhanced. However, fractional decrease in BTE has been observed with an increase in biodiesel percentage. The reason is that the high density and viscosity of cotton seed biodiesel results poor atomization which further deteriorates overall combustion efficiency and due to lower calorific value of biodiesel. It is also seen that BTE for B20E10, B30E10, and B40E10 inducted fuels is increased with preheating intake air. This can be related to the increase in the temperature of

the intake air, which leads to improved combustion behavior of biofuel blends as compared to the ambient intake air temperature.

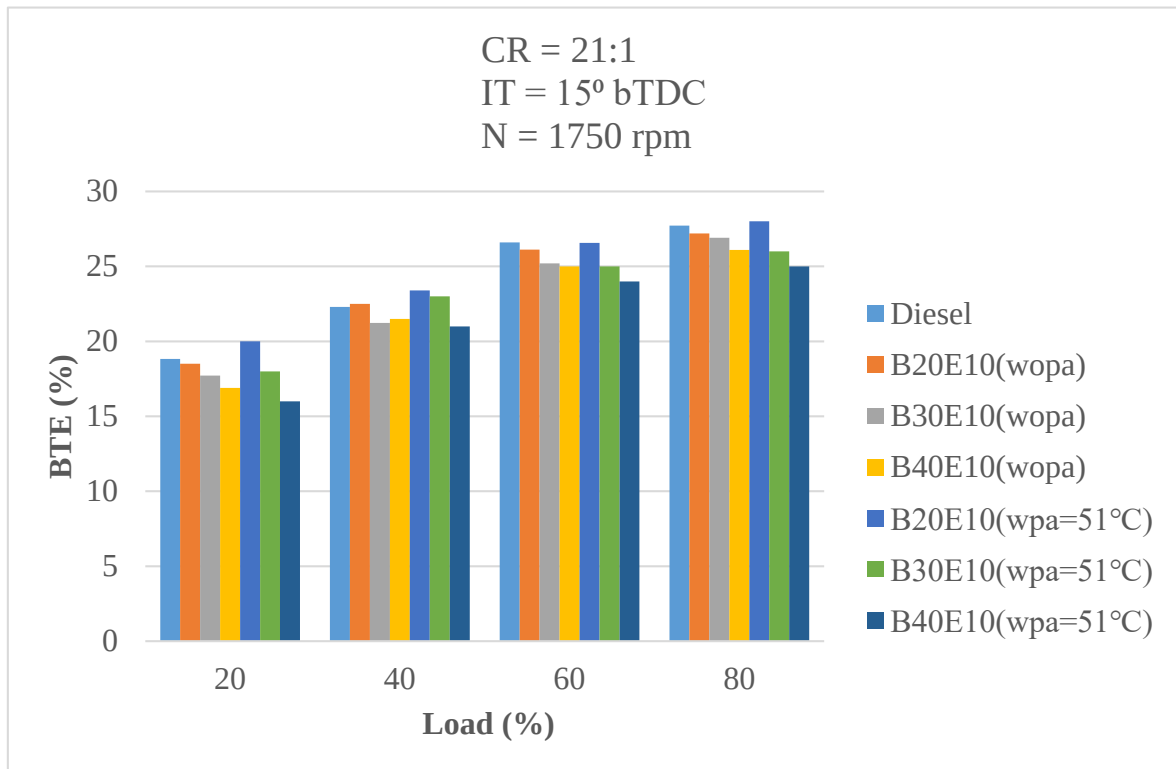


Figure 4.5: Variation of brake thermal efficiency versus engine load.

BTE attains maximum for all the test fuels (B20E10, B30E10, and B40E10) at 80% load operating condition at preheated intake air temperature of 51°C, i.e., 28.01%, 26%, and 25%, respectively as can be seen from Table 4.4. Therefore, the average BTE reduction for B20E10, B30E10 and B40E10 at ambient intake air temperature in compared to diesel fuel is 1.17%, 4.57% and 6.20% respectively and the average percentage of increase in BTE for all the test fuels at intake air preheating temperature of 51°C is seen to be 3.9%, 4.1%, and 4.28% for B20E10, B30E10, and B40E10 respectively as compared to the conventional engine without preheating intake air at 25°C.

4.5.1.3 Brake Specific Fuel Consumption (BSFC)

Figure 4.6 represents the deviation of BSFC with load for petrol diesel and various blends of cotton seed biodiesel with addition of ethanol at both ambient and preheated intake air temperatures. BSFC acts as the metering scale on determining the engine efficiency in converting the heat energy of the fuel into required brake power (Devarajana et al., 2020). From the graph, it is noticed that the BSFC increases with the increase in the proportion of

biodiesel in the blend. This could be due to the inferior calorific values and higher specific gravity of biodiesel. At ambient intake air temperature, it observed that the BSFC for diesel is lower when compared to all other blends. Among all blends at ambient intake air temperature, B20E10 gives slightly lower in BSFC compare to B30E10 and B40E10 blends, the reason may be higher heating value, lower specific gravity and lesser kinematic viscosity. The trend could be due to enhanced air–fuel mixing rate with increased oxygen presence in the biodiesel and ethanol and fuel vaporization results in improved combustion. When the engine runs with ambient intake air temperature the higher BSFC is noticed for B30E10 and B40E10, it could be due to higher content of biodiesel cause improper air-fuel mixing which in turn cause the incomplete combustion.

Table 4.5: Brake specific fuel consumption for all tests.

Load (%)	Brake Specific Fuel Consumption (kg/kW h)						
	Diesel	B20E10 (wopa)	B30E10(wopa)	B40E10(wopa)	B20E10(wp a=51°C)	B30E10(wp a=51°C)	B40E10(wp a=51°C)
20	0.42	0.47	0.58	0.59	0.44	0.49	0.54
40	0.36	0.37	0.4	0.43	0.37	0.39	0.38
60	0.3	0.34	0.36	0.4	0.32	0.33	0.35
80	0.28	0.32	0.35	0.39	0.3	0.32	0.34

Results also indicate that at preheated intake air temperature of 51°C BSFC of B20E10 is the lowest which is perhaps due to the higher calorific value of B20E10 fuel as compared the other blended fuels. Overall, preheated intake air show some improvement in the performance of the engine due to better vaporization of fuel mixtures. Further, an increase in intake air temperature results in decreased BSFC due to a lower reduction in latent heat vaporization which causes a reduction in the ignition delay period that provides improved combustion efficiency.

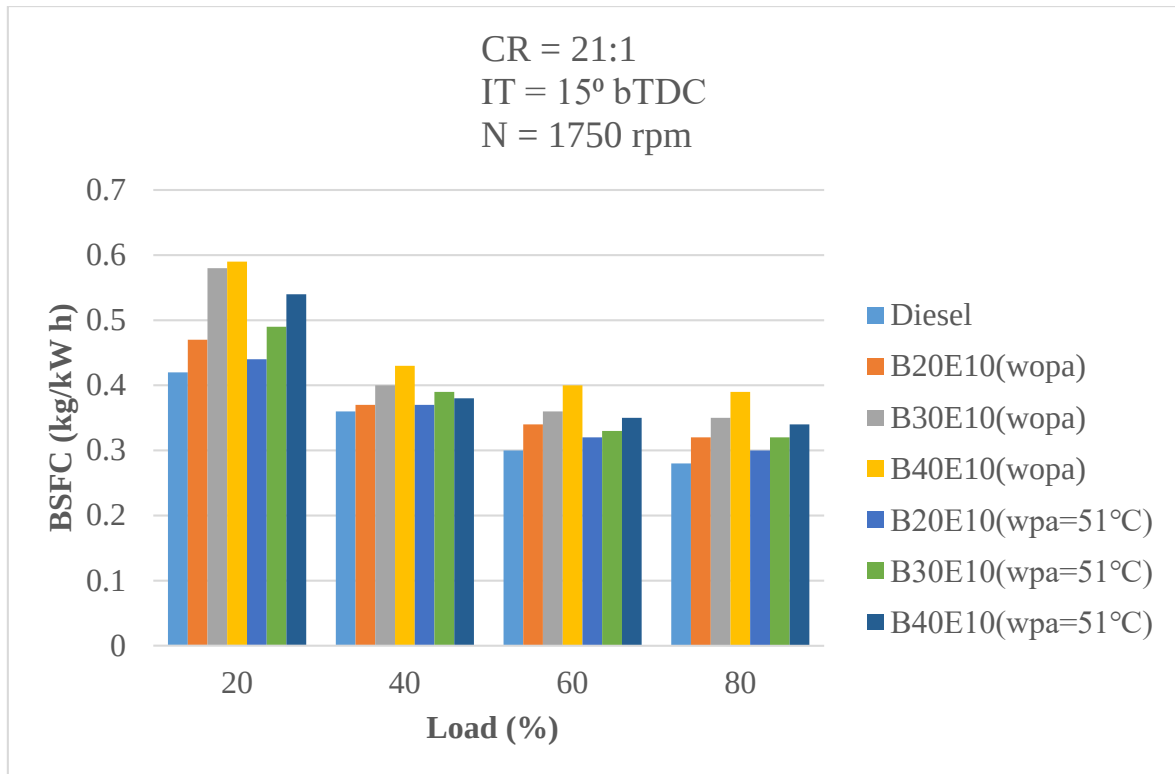


Figure 4.6: Variation of brake specific fuel consumption with respect to engine load.

Generally, the average brake specific fuel consumption increment of B20E10, B30E10 and B40E10 with respect to diesel fuel at ambient intake air temperature was obtained 11.76%, 23.53% and 32.35% respectively and the average percentage of drop in BSFC for all the test fuels at intake air preheated temperature of 51°C for B20E10, B30E10, and B40E10 is seen to be 5.2%, 9.52%, and 11.11% respectively when compared to the conventional engine without preheated intake air at 25°C. Table 4.5 above shows the numerical value of brake specific fuel consumption for all tested fuels both at ambient and preheated intake air temperatures.

4.5.2 Exhaust Gas Emissions

Emission standards are requirements that set specific limits to the amount of pollutant which can be released into the environment. Many emission standards focused on regulating pollutant released by automobile and other powered vehicles but they can also regulate emissions from industry, power plants and diesel generators. In this study the emissions output of fuel samples are measured, they are for CO, CO₂, HC and NO_x emissions with reference to diesel fuel emissions.

4.5.2.1 Exhaust Emissions of Carbone Monoxide

The comparison of carbon monoxide (CO) emissions with load at constant engine speed both at the elevated intake air temperature of 51°C and ambient intake air temperature of 25°C for diesel and diesel-biodiesel blends with addition of ethanol are represented in Figure 4.7 and its numerical values are tabulated in Table 4.6. It can be observed from Figure 4.7 that the CO emissions for higher percentage of biodiesel in diesel-biodiesel blends with ethanol addition are generally lower than for lower percentage of biodiesel across the entire load. This can be explained by the enrichment of oxygen owing to the biodiesel, in which an increase in the proportion of oxygen will promote the further oxidation of CO during the engine exhaust process. From the graph, it clears that the CO emission level is higher at the beginning as cooling effect plays a major role at the beginning and reduces at medium and later increased sharply up to maximum load. This may be due to the poor burning of the fuel with a very little time available for burning. The study noticed that the CO emissions for the engine operation with higher biodiesel blends are lower among different blends because of better air availability and proper mixing of air and fuel for better combustion.

Table 4.6: Carbon monoxide exhausts emission for all tests.

Load (%)	Carbone Monoxide Emissions (CO) in % Vol.						
	Diesel	B20E10 (wopa)	B30E10 (wopa)	B40E10 (wopa)	B20E10(w pa=51°C)	B30E10(w pa=51°C)	B40E10(w pa=51°C)
20	0.043	0.042	0.04	0.035	0.025	0.02	0.023
40	0.038	0.035	0.03	0.03	0.02	0.018	0.015
60	0.04	0.035	0.035	0.029	0.025	0.023	0.019
80	0.051	0.05	0.045	0.04	0.04	0.035	0.034

As observed, the CO emission is greatly influenced by engine load, biofuel proportion and intake air temperature. Additionally, intake air preheated to 51°C helps biodiesel-diesel blends with addition of ethanol reduce CO emissions as compared to ambient air (25°C). Overall, preheating air indicates lower CO emission compared to ambient conditions. The main reason is that more complete combustion is achieved due to better vaporization at higher intake temperature. It can be also concluded that emissions are strongly influenced by intake air temperature and biofuel concentration in the blends. The reduction is more

effective for the higher biofuel concentration in the blended fuel mixture, since the oxygen content is greater in biodiesel and in ethanol, which enhances the oxidation reaction, that reduces the CO emission. At preheated intake air temperature, the cylinder gas temperature increased, leading to improved chemical reactions; CO was oxidized to CO₂, hence decreasing overall CO emissions which is similar with the result obtained (Gopal et al., 2017).

Another observation is that diesel-biodiesel blend with addition of ethanol shows lower CO emission at full load regardless of preheat. In fact, preheat and ambient conditions show no difference in CO emissions at full load. This implies that there is no need to preheat the intake air if the engine is to be operated at full load because it provides enough energy to allow the majority of ethanol to make the phase change, resulting in complete combustion.

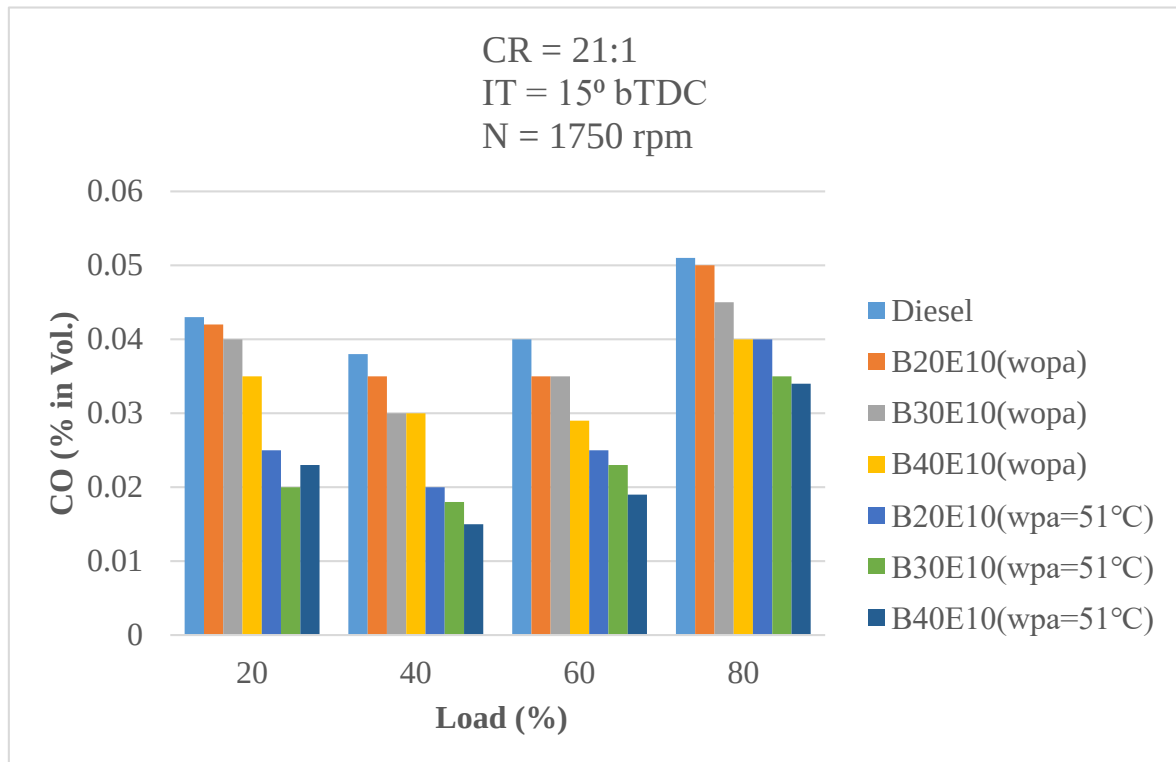


Figure 4.7: Variation of CO emissions with respect to load.

Generally, the average reduction of exhaust emission of carbon monoxide for B20E10, B30E10 and B40E10 at ambient intake air temperature in compared to diesel fuel is 4.65%, 11.63% and 20.93% respectively and the average percentage of reduction in CO emissions for all the test fuels at intake air preheating temperature (51°C) for B20E10, B30E10, and B40E10 is seen to be 31.71%, 36.84%, and 32.35% respectively when compared to the

conventional engine without preheating intake air at 25°C. Thus, fuel blend ratio and operating conditions play important roles in the change of CO emissions.

4.5.2.2 Exhaust Emissions of Carbone Dioxide

The variation of CO₂ emissions with load for diesel and three other diesel-biodiesel blends with addition of ethanol is shown in the Figure 4.8. The CO₂ emissions increased with load for all the fuel modes. Compared with diesel, the CO₂ emissions of biodiesel-diesel blends with addition of ethanol are higher. This is due to the cause of more oxygen content in the biodiesel and ethanol leads to complete oxidation of CO into CO₂. Also the CO₂ emissions slightly increased with the percentage increase of biodiesel in the biodiesel blends because of better air availability and proper mixing of air and fuel for better combustion. And is due to fact that biodiesel has lower elemental carbon to hydrogen ratio than diesel fuel. For preheated intake air CO₂ emission further increased compared to ambient intake air temperature. This may be due to the improved combustion in the engine cylinder. The numerical values of Carbon dioxide exhausts emissions for all tests fuels is listed in Table 4.7 below.

Table 4.7: Carbon dioxide exhausts emissions for all tests.

Load (%)	Carbon Dioxide Emissions (CO ₂) in % Vol.						
	Diesel	B20E10 (wopa)	B30E10 (wopa)	B40E10 (wopa)	B20E10(wpa=51°C)	B30E10(wpa=51°C)	B40E10(wpa=51°C)
20	2.31	2.5	2.4	2.3	3.4	3.2	2.9
40	3.22	3.52	3.41	3.3	4.3	4.2	4
60	4.21	4.51	4.4	4.32	5.5	5.2	5.4
80	5.2	5.4	5.5	5.5	6.6	6.6	6.5

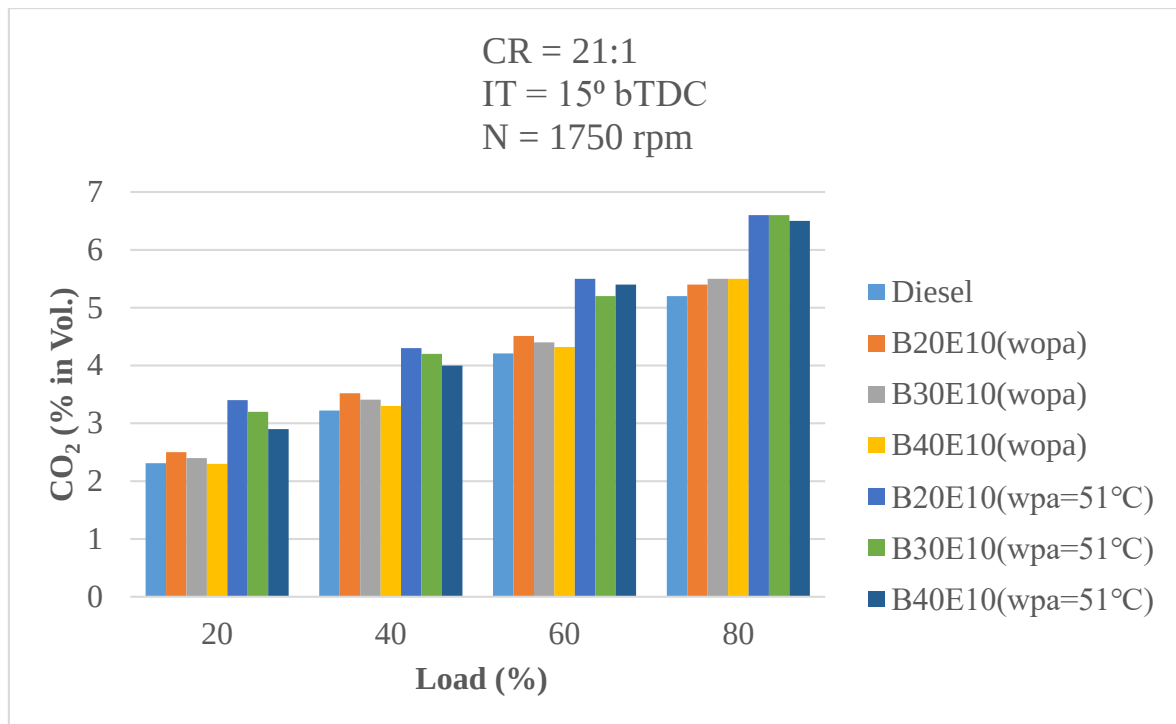


Figure 4.8: Variation of CO₂ emissions with respect to load.

Generally, the average exhaust emission of carbon dioxide increment for B20E10, B30E10 and B40E10 with respect to diesel fuel at ambient intake air temperature was obtained 7.83%, 5.34% and 2.85% respectively and the average CO₂ emissions increment for all the test fuels at intake air preheated temperature (51°C) for B20E10, B30E10, and B40E10 is seen to be 24.3%, 22.14%, and 21.24% respectively when compared to the operation of engine without preheated intake air.

4.5.2.3 Exhaust Emissions of Hydrocarbons (HC)

Changes in HC emission for diesel and all blended fuel at different loads and different intake air temperatures have been shown in Figure 4.9 and its numerical value is listed in Table 4.8. Partial combustion of fuel inside the combustion chamber is the main reason for the HC emission. There is an increasing trend in HC emissions for all test fuel as load increases. Due to the incomplete combustion, higher HC emission has been observed for the blended fuels compared to diesel at ambient intake air temperature. According to the results, blend with 40% biodiesel exhibits minimum HC emissions compared to B20E10 and B30E10. The reason may be the excess oxygen in B40E10 blend results better combustion that helps in lowering the HC emission compared to other blends.

Table 4.8: Hydrocarbon exhaust emissions for all tests.

Load (%)	Hydrocarbons Emissions (HC) in ppm Vol.						
	Diesel	B20E10 (wopa)	B30E10 (wopa)	B40E10 (wopa)	B20E10(wpa=51°C)	B30E10(wpa=51°C)	B40E10(wpa=51°C)
20	25.1	30	27.2	26.4	26	23.2	24
40	30.89	34	34	31.6	31	30.5	27.6
60	40.12	49	46	43	43.5	41	37
80	65.2	75.22	67	65	66	58	57

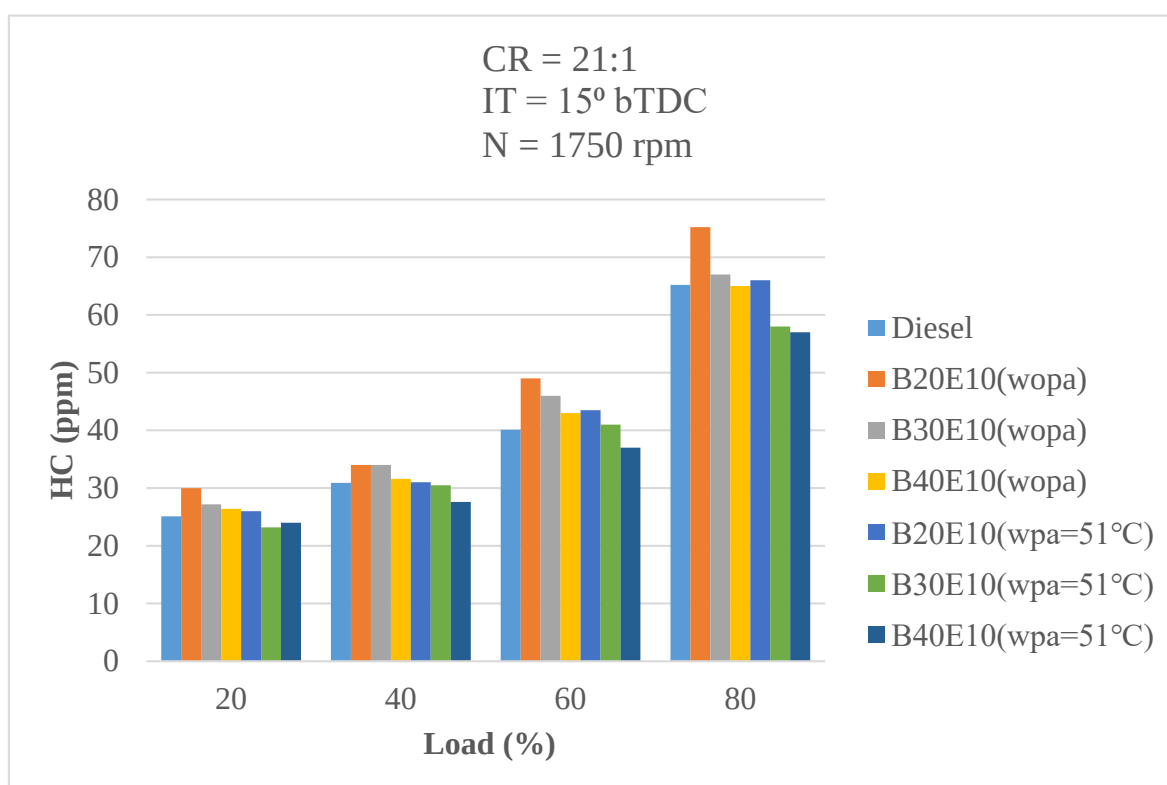


Figure 4.9: Variation of HC emissions with respect to load.

The intake air temperature played an important role in the HC emissions. At preheated intake air also the HC emission for B40E10 fuel has lower values compared to other tested fuels over the entire load range. This may be a shorter ignition delay and early combustion attributed because of the higher Cetane number of biodiesel. This results sufficient time for the combustion of fuel and decrease HC emission. The presence of more oxygen content in

the biodiesel also contribute to have a better combustion, and this results in lowering HC emissions. Additionally, due to preheated intake air temperature the combustion temperature increased, results in lowering in the levels of HC emission which is in line with the reported data (Papagiannakis, 2013). The maximum of HC emission for B20E10, B30E10, and B40E10 fuels at intake air temperature of 25°C is found as to be 75.22, 67, and 65 ppm at 80% loading conditions and the minimum values at intake air preheating temperature of 51°C are 26, 23.2, and 24 ppm at 20% loading conditions, respectively

Generally, the average exhaust emissions of hydrocarbons (HC) increment of B20E10, B30E10 and B40E10 with respect to diesel fuel at ambient intake air temperature of 25°C was obtained 16.79%, 8% and 2.9% respectively and the percentage reduction of HC emissions for B20E10, B30E10, and B40E10 fuel is 11.49%, 13.23%, and 12.29%, respectively, at 51°C intake air operation compared to without preheating intake air (25°C).

4.5.2.4 Nitrogen Oxide Exhaust Emissions

Nitrogen oxide is one of the most relevant air pollutants composed of nitrogen and oxygen, among which nitric oxide (NO) and nitrogen dioxide (NO₂) are most toxicologically significant compounds. The formation of NO_x is highly influenced by combustion, heat release rate and oxygen concentration. In this study, the variations of NO_x at both ambient and preheated intake air temperatures with different loads have been shown in Figure 3.10 for different blends and base diesel. Cylinder temperature and oxygen presence in the fuel will result the NO_x formation in a diesel engine. As load increased nitrogen oxides also increased. It has been seen in Figure 4.10 that NO_x is observed to be greater for B40E10 fuel when compared to other blends for all loads. This is because higher heat release rate (HRR) during premixed burning phase observed with higher proportion of biodiesel fuels led greater cylinder temperature and improved combustion. This might be attributed to proper air-fuel mixing during compression and complete combustion process and during power stroke with greater oxygen presence in the cylinder. Because of the low heat generation, there is no such change in NO_x emission at low load. At medium to high load, the high oxygen content in biodiesel and ethanol enhances the burning and causes a rise in combustion temperature which further increases NO_x emission. The reason for higher NO_x emission may also be due to the generation of high premixed combustion temperature because of the higher premixed combustion duration.

Table 4.9: Nitrogen oxides exhaust emissions for all tests.

Load (%)	Oxides of Nitrogen Exhaust Emission (NOx) in ppm						
	Diesel	B20E10(wopa)	B30E10(wopa)	B40E10(wopa)	B20E10(wpa=51°C)	B30E10(wpa=51°C)	B40E10(wpa=51°C)
20	221.7	177.35	166.29	210.12	224	210	260
40	286.3	236.3	229.89	280.3	290	286	330
60	433.9	347.45	382.9	421.12	440	400	468
80	762.3	753.43	777.8	798.54	774	796	820

Another observation is that the NOx emission of biodiesel-diesel blend with addition of ethanol fuels was found to be higher at intake air preheated temperature of 51°C when compared to the operation of the engine without preheated intake air (25°C). The increase in the intake air preheated temperature improves the combustion characteristics, which causes an immediate increase in engine gas temperature that leads to an increase in NOx emission which is comparable well with the reported data (Menelik and Niranjana, 2018). In general, higher NOx emission formation is mainly due to elevated cylinder temperature and better fuel burning followed by improved air-fuel mixing rate with higher oxygen content in cotton seed biodiesel blends.

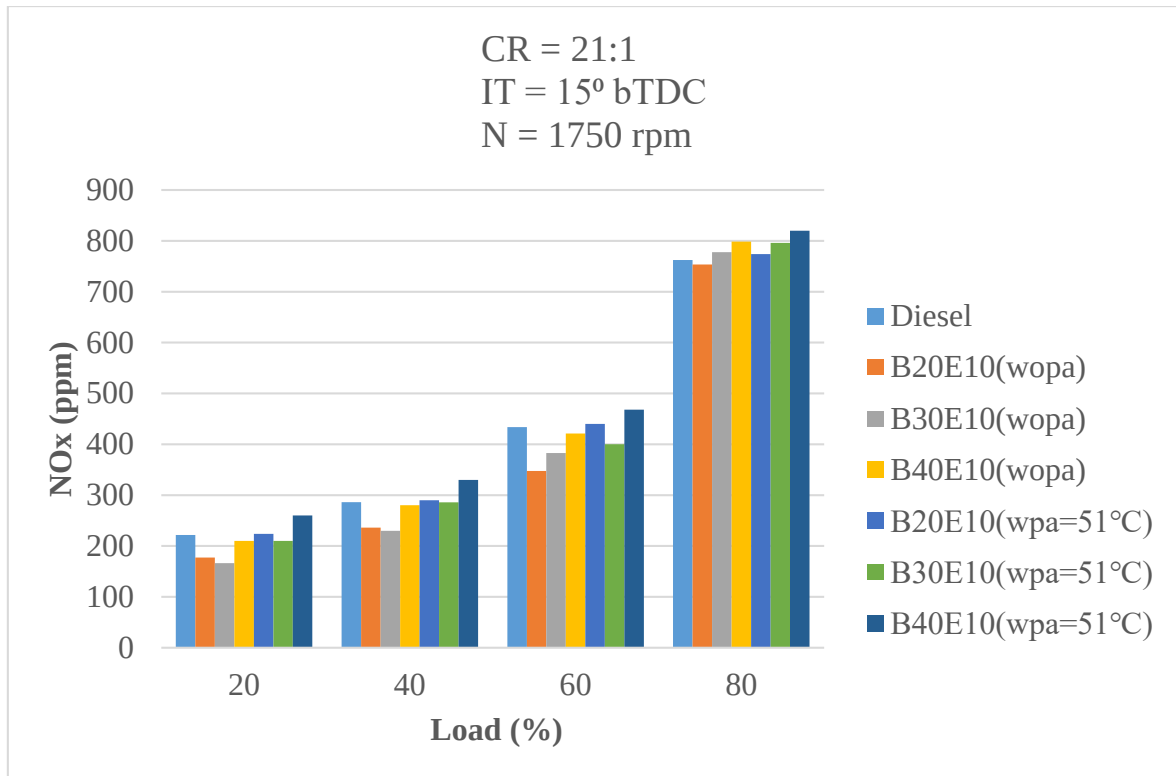


Figure 4.10: Variation of NOx emissions with respect to load.

Generally, the average reduction of exhaust emission of Nitrogen oxide for B20E10, B30E10 and B40E10 in compared to diesel fuel at ambient intake air temperature is 11.13%, 8.65% and 0.19% respectively and the average percentage increased NOx emissions of B20E10, B30E10, and B40E10 at preheated intake air temperature of 51°C are 14.13%, 8.68%, and 9.82%, compared to the operation of the engine without intake air preheating (25°C).

CHAPTER FIVE

CONCLUSION AND RECOMMENDATIONS

5.1 Summary

The main objective of this work was to investigate the effect of preheating intake air and proportion of biofuels on performance and emission characteristics of diesel engine operating by diesel-biodiesel blends with addition of ethanol fuel at elevated preheating intake air temperature by varying the engine load and percentage by volume of biodiesel at a constant engine speed of 1750 rpm by using counter flow shell and tube heat exchanger device. Using counter flow shell and tube heat exchange device for the blended fuels has advantages for preheating the intake air which shortened ignition delay and thereby due to complete combustion, improves performance and reduction of emission characteristics as compared without preheating the intake air.

Experiments were carried out on a diesel engine fueled with firstly diesel and blended fuels at ambient intake air temperature for baseline data and secondly blended fuels at elevated intake air temperature for examining the performance and emission parameters as compared to ambient intake air temperature at constant engine speed for all operations. The effect of preheating the intake air and percentage variation of biodiesel on variation of the performance and emission characteristics of a diesel engine at various engine loads was also examined.

Based on the experimental work, the following main findings are obtained:

- Almost for all loads: brake power and brake thermal efficiency of blended fuels shown slight reduction and slight increment of brake-specific fuel consumption was noticed as compared to baseline diesel fuel when the engine is operated at ambient intake air temperature. Whereas emissions; CO₂ and HC showed that little increment and slight reduction of CO and NO_x emissions as compared to baseline data.
- As percentage of biodiesel increases for all load conditions at ambient intake air temperature: no significant change in brake power, little reduction in brake thermal efficiency, and slight increment of brake specific fuel consumption was noticed. Whereas emissions; it was noticed little increment in CO₂ and NO_x and little reduction in CO and HC emissions.
- For all loads and blended fuels when the engine was operated at elevated preheated intake air: brake power and brake thermal efficiency of blended fuels shown slight

increment and little drop of brake-specific fuel consumption was noticed as compared to the operation of the engine without intake air preheating. Whereas emissions; CO₂ and NO_x showed that little increment, slight reduction of HC and medium drop of CO emissions as compared to the operation of the engine without intake air preheating.

- As percentage of biodiesel increases for all load conditions with preheated intake air: no significant change in brake power, little reduction in brake thermal efficiency, and slight increment of brake specific fuel consumption was noticed. Whereas emissions; it was noticed little increment in CO₂ and NO_x and little reduction in CO and HC emissions.

5.2 Conclusions

From the study, cotton seed biodiesel diesel blend with addition of ethanol can be considered a highly promising alternate fuel for CI engine application. Using the aforementioned fuel blends, a number of tests were carried out at an engine working speed of 1750 rpm. It was investigated the performance and emission parameters for biodiesel diesel blends with addition of ethanol at different intake air temperatures. From the experimental results the following conclusions are drawn:

In the case of biodiesel-diesel blends with ethanol addition as compared to that of diesel at ambient intake air temperature as biodiesel percentage varies from 20% to 40%:

- There is an increment of BSFC from 11.76% to 32.35% and reduction in brake power from 1.97% to 4.76%, and BTE from 1.17% to 6.2%.
- There is a reduction of CO and NO_x from 4.65% to 20.93% and from 0.9% to 11.13% respectively and an increment of CO₂ and HC from 2.85% to 7.83% and from 2.9% to 16.79% respectively.

And as load varies from 20% to 80%:

- There is a decrease in the BSFC and increase in the brake power and BTE of blends, following a similar trend to that for diesel.

In the case of preheated intake air as compared to that of ambient intake air temperatures of the engine running with biodiesel-diesel blends with ethanol addition as load varies from 20% to 80%:

- There is an increment of brake power and BTE from 1.34% to 3.26% and from 3.9% to 4.28% respectively and drop of BSFC from 5.2% to 11.11%.
- There is reduction of CO and HC from 31.71% to 36.81% and from 11.49% to 13.23%, a respectively and an increment of CO₂ and NO_x from 21.24% to 24.3% and from 8.68% to 14.13% respectively.

Among the diesel-biodiesel blended with addition of ethanol test fuels B20E10 shows significant improved of performance and emission characteristics of the engine when the engine is operated at ambient intake air and when the engine operates with preheated intake air B30E10 shows significant improvement of performance and emission characteristics of the engine.

Generally, the performance and emission characteristics of a direct injection diesel engine without any engine modification are improved by using diesel-biodiesel blends with addition of ethanol as a fuel and preheating the intake air. The preheated intake air shows an effective result when compared to the ambient intake air.

5.3 Recommendations

- In the present study the characteristics were only carried out for some properties of the cotton seed biodiesel and blended fuels due to financial constraint. To have a better comparison the characteristics for all properties of biodiesel and blends should have to be carried out.
- In this study due to the limitation of analyzer used there was only focused on the emissions of CO, CO₂, HC and NO_x but to have a better investigation on the effects of preheating intake air and biofuel proportion on soot emissions and particulate matter emission test should have to be carried out.
- When the engine is operating at ambient intake air temperature B20E10 fuel relatively has the best performance and emission characteristics, so it is recommended to substitute diesel fuel with it.
- It is recommended to use counter flow shell and tube heat exchanger in the diesel engine fueled with diesel biodiesel blends with ethanol addition to have better combustion.
- When the engine is operating with preheated intake air B30E10 fuel relatively has the best performance and emission characteristics, so it is recommended to substitute diesel fuel with it.

- Finally, I recommended the Department of Mechanical Systems and Vehicle Engineering of Adama Science and Technology University to have set-up of biodiesel laboratory and an alternative fuel performance and emission testing laboratory which is very important to make a research work on renewable energy sources.

5.4 Future Work

From different aspects as has been mentioned in chapter one under the significance of the study, using renewable alternative fuel such as biodiesel and ethanol for the IC engine has several advantages. From this point of view, several things can be made and improved in the future in which this study couldn't address. So the following areas need further improvement for future studies.

i. Percentage of ethanol variation: since higher viscosity and density of biodiesel affects engine performance and emission badly, the use of biodiesel in partial replacement of fossil diesel in a diesel engine is limited to minimum. So to improve the properties of biodiesel (viscosity and density) and thereby increase biofuel percentage and prevent environmental pollution due to conventional fuel, addition of ethanol should be practiced in substitution of petroleum based fossil fuel. This work was conducted only on addition of 10% ethanol with three diesel-biodiesel blends. But for having better understanding the effect of ethanol addition and further evaluation and comparison, studies should be conducted at varies ethanol percentage addition for the sake of finding the optimum blending ratio to get better performance and emission.

ii. Intake air temperature variation: preheating intake air is the major criteria for improving the performance and reduction in emission of the diesel-biodiesel-ethanol blended fuels of diesel engine. And this study was carried out only at preheated intake air temperature of 51°C. But for having better comparison and further evaluation, there should be study conducting at varies preheated intake air temperatures for observing optimum intake air temperature to get better performance and emission along with maximum biofuel proportion. Therefore, conducting studies at various intake air temperature to find the optimum intake air temperature that helps to use maximum biofuel proportion is the future research work.

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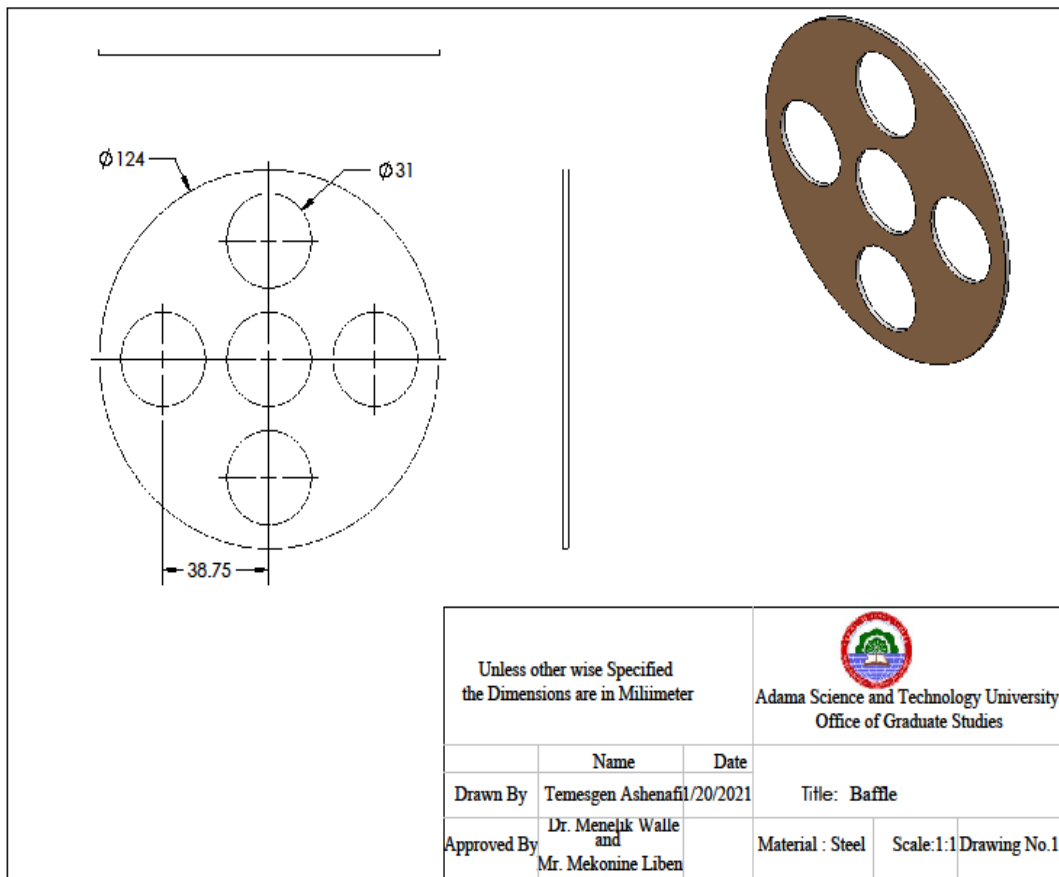
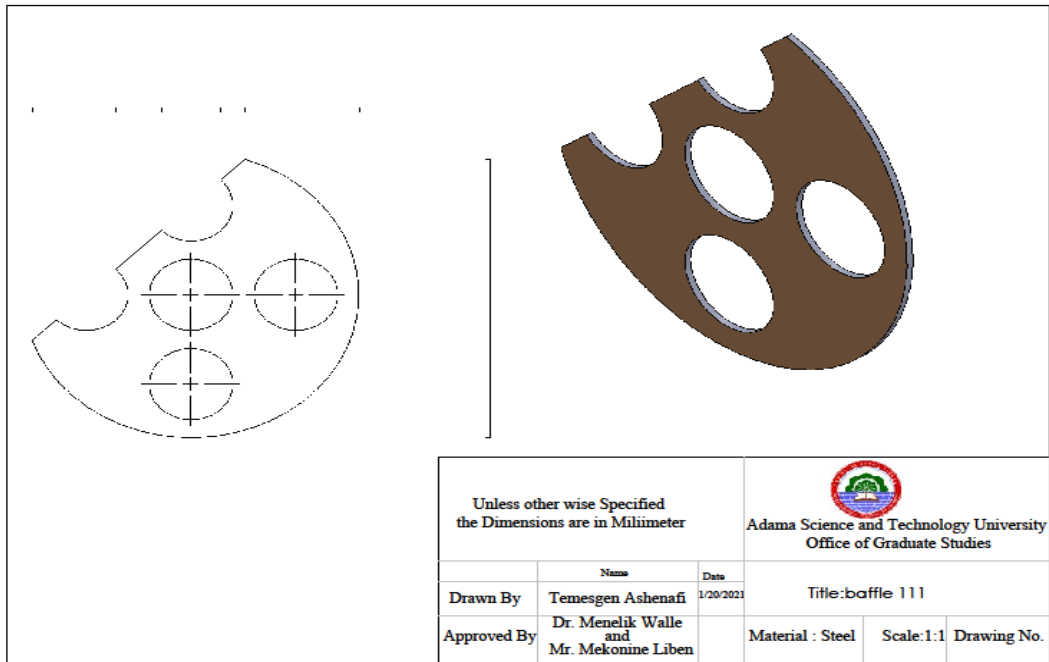
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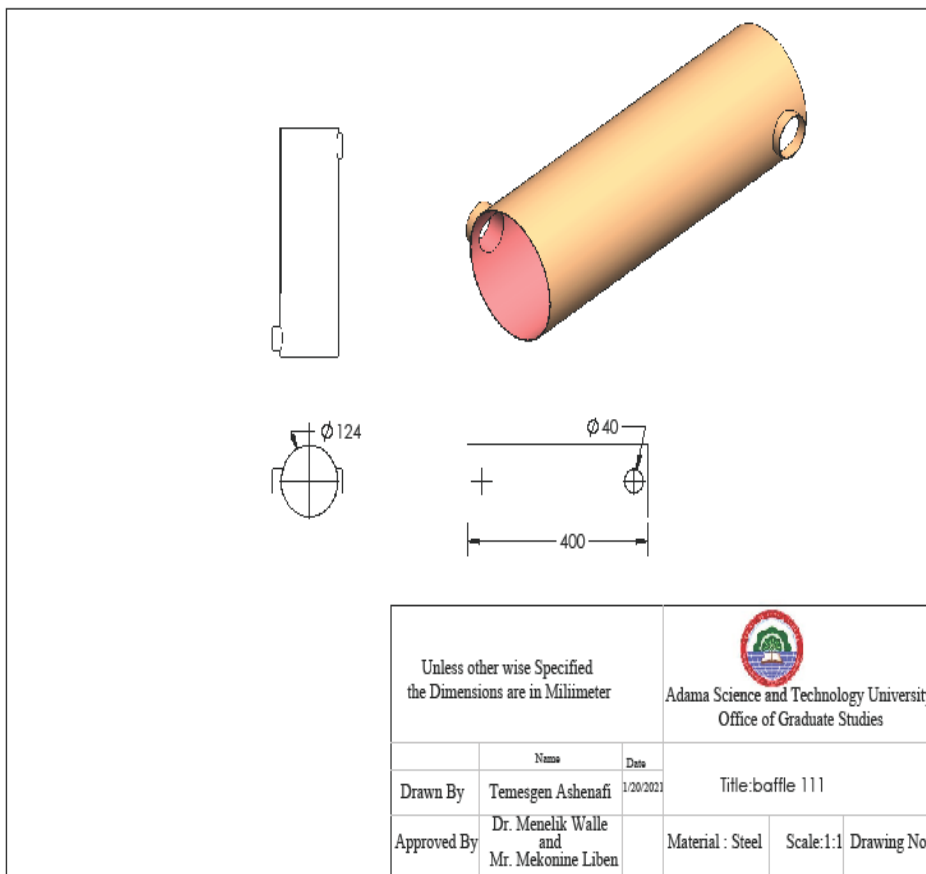
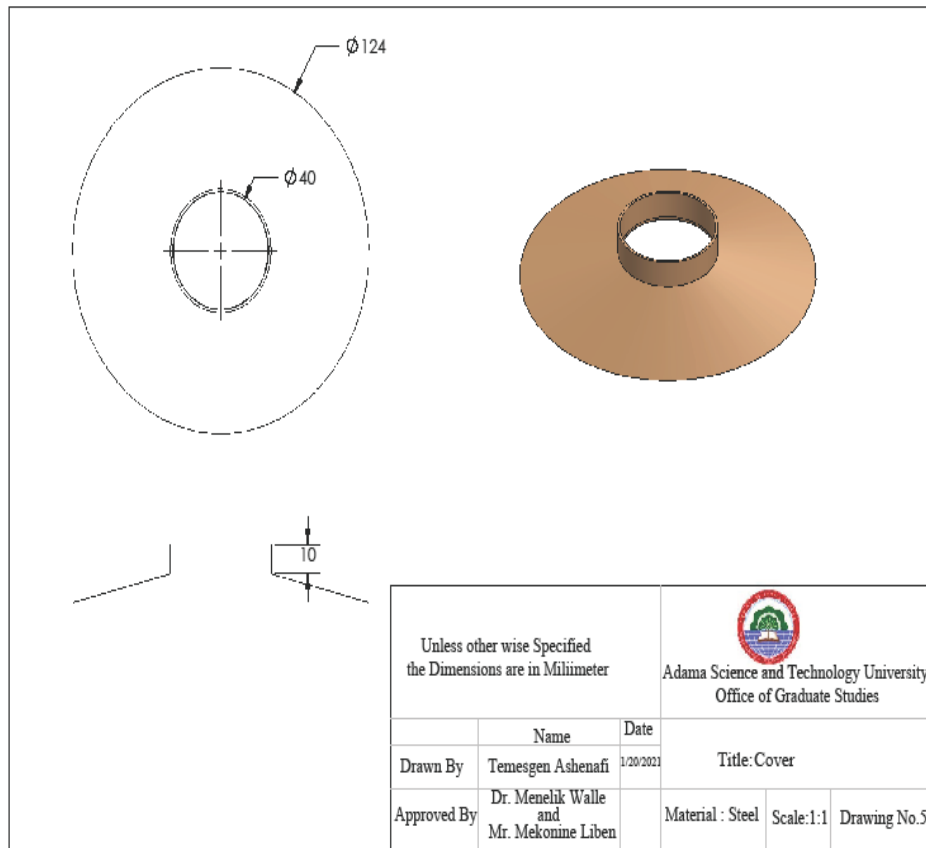
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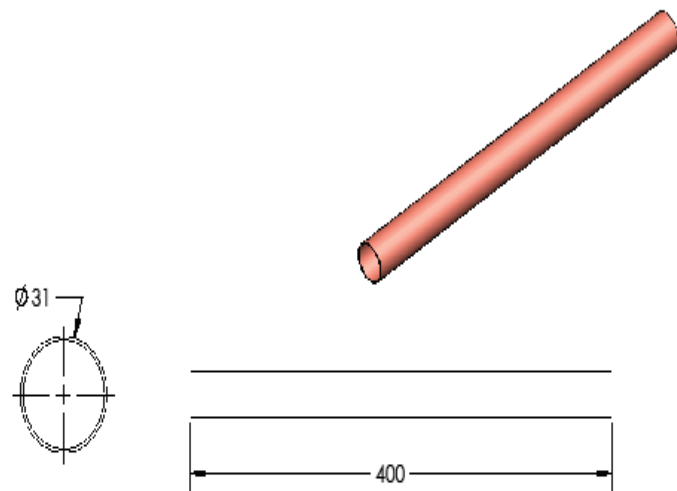
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APPENDICES

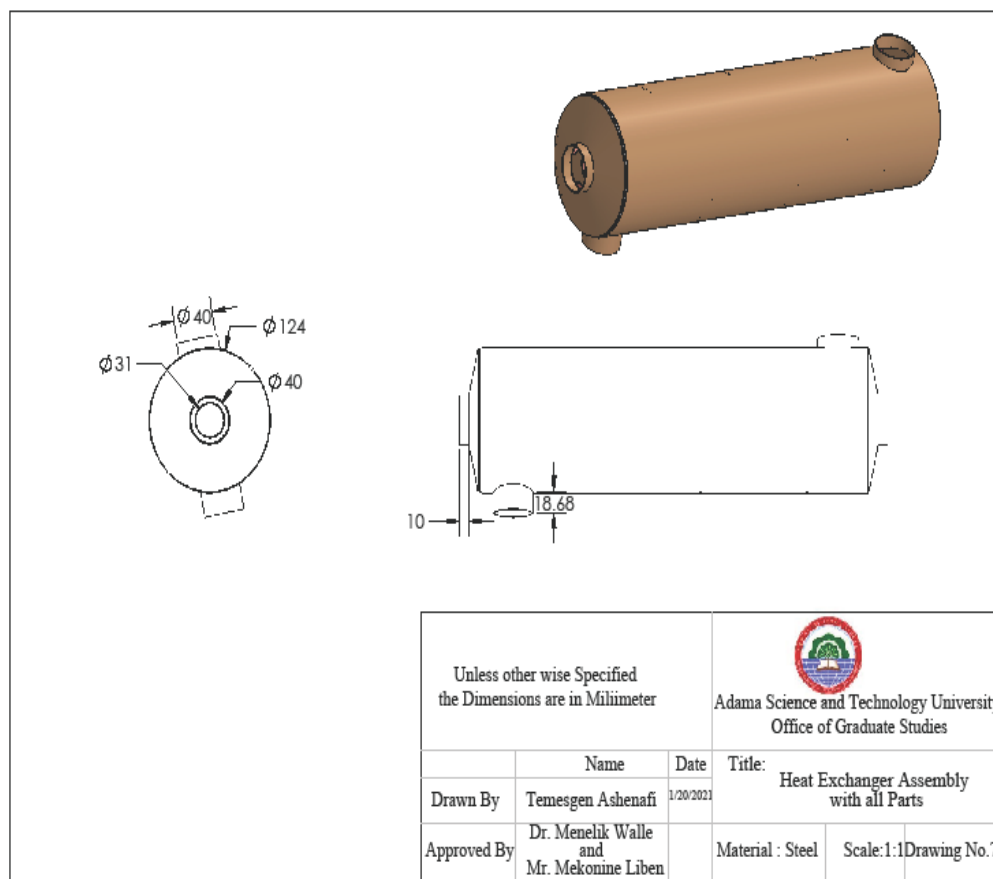
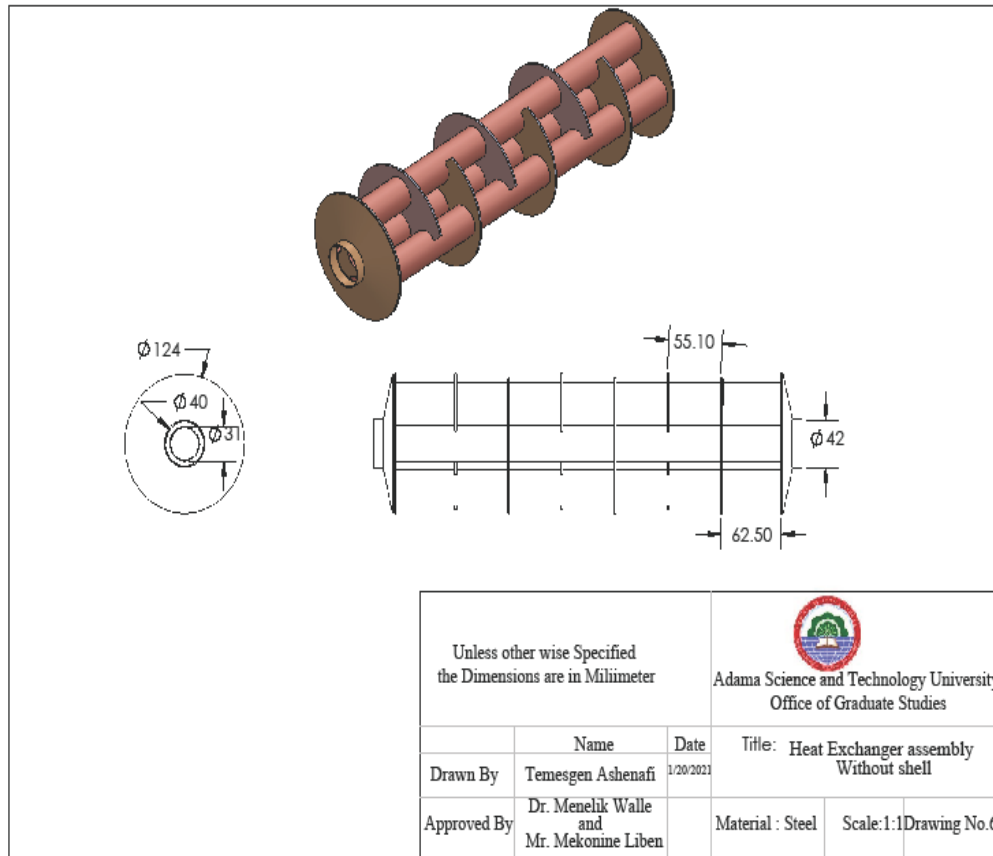
Appendix A: Drawings







Unless other wise Specified the Dimensions are in Milimeter			 Adama Science and Technology University Office of Graduate Studies		
	Name	Date	Title: tube		
Drawn By	Temesgen Ashenafi	1/21/2021			
Approved By	Dr. Menelik Walle and Mr. Mekonine Liben		Material : Steel	Scale: 1:1	Drawing No. 4



Appendix B: Material Specification

i. Automobile Exhaust Gas Analyzer SV-50

- Environment conditions: Temperature range from -5~40°C, Humidity $\leq 95\%$ and atmosphere pressure from 670~10⁶kpa
- Power supply: AC220V $\pm 10\%$; 50Hz ± 1 Hz
- Sampling mode: tail gas directly sampling with a sampling tube length of 5m and length of sampling head is 900mm.
- Oil temperature: inserting the head of a thermo scope into the staff gauge of engine lubricant, the length of which is the same as that of oil staff gauge, and the length of lead is 5m.
- Preheating time: 10 minutes and Oil temperature measuring range: 0~120°C.

Table B.1: Automobile Exhaust Gas Analyzer SV-50 measuring range.

species	Measurement range	Resolution	Allowed error	Relative error
HC	0 - 1000 ppm	1 ppm	± 12 ppm	$\pm 5\%$
CO	0 - 10.0 in % Vol.	0.01%	$\pm 0.06\%$	$\pm 5\%$
CO ₂	0 - 20.0 in % Vol.	0.1%	$\pm 0.5\%$	$\pm 5\%$
NO _x	0 – 800 ppm	1 ppm	± 10 ppm	± 5

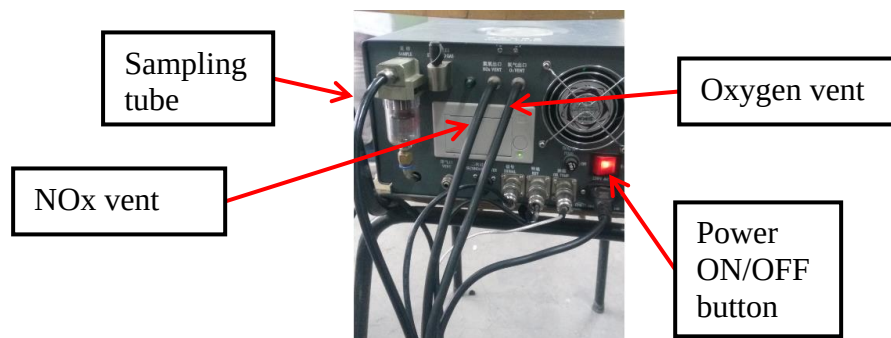
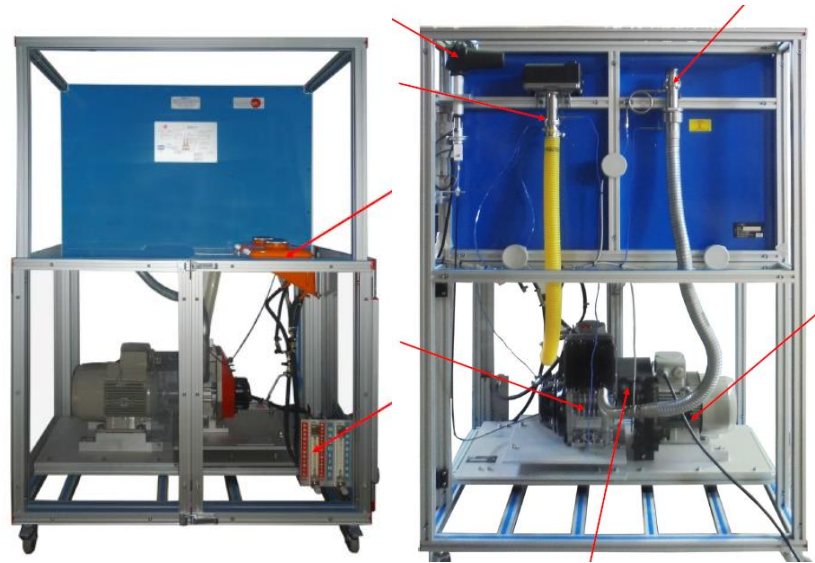


Figure B.1: Rear view of automobile exhaust gas analyzer SV-50.

ii. TBMC3-02 Test Bench Diesel Engine



a. Front view

b. Rear view

Figure B.2: TBMC3-02 Test Bench Diesel Engine.

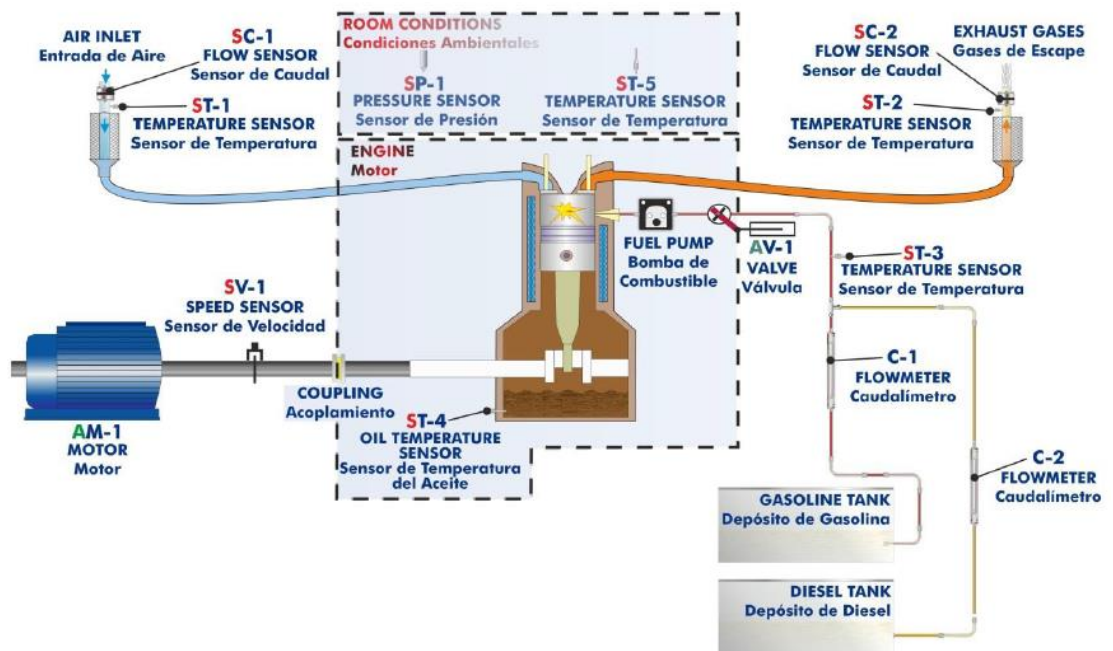


Figure B.3: Description of the process.

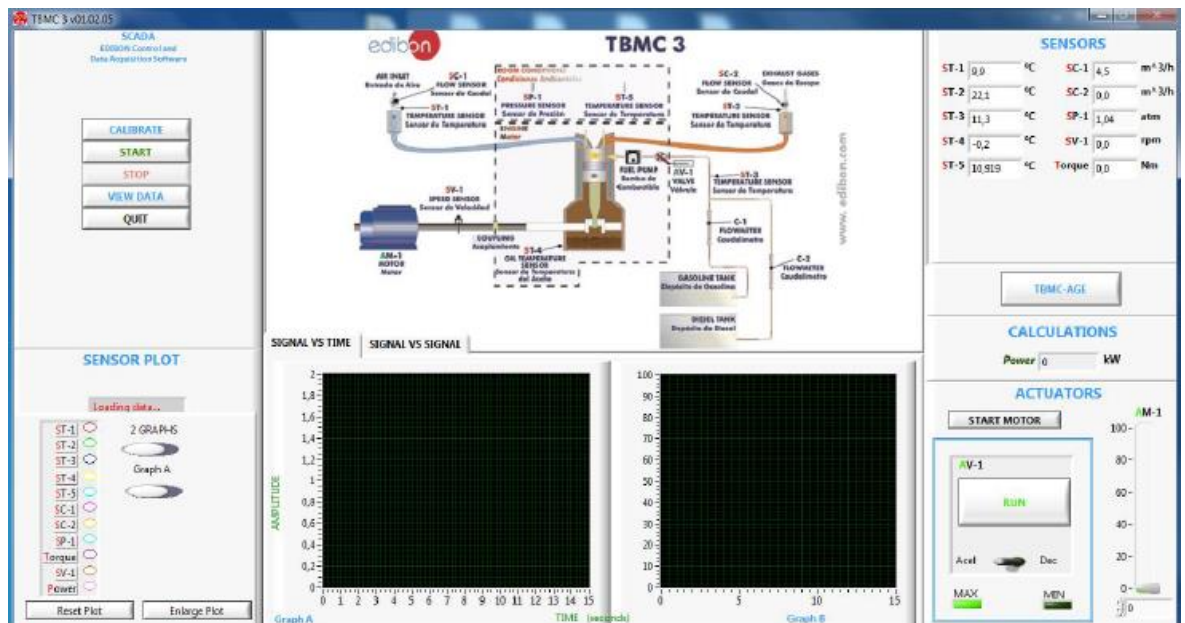


Figure B.4: TBM-3 Software unit.

iii. Thermocouple

A thermocouple is an electrical device consisting of two dissimilar electrical conductors forming an electrical junction. A thermocouple produces a temperature-dependent voltage as a result of thermoelectric effect, and this voltage can be interpreted to measure temperature.



Rear view



Front view

Figure B.5: Thermocouple.

Appendix C: Letters

i. Characterization results

	Company Name: የኢትዮጵያ ንግድ አቅርቦት ድርጅት ETHIOPIAN PETROLEUM SUPPLY ENTERPRISE	Document number: OF/EPSE/QAS/110
		Page --- of ---
TITLE: CERTIFICATE OF TEST REPORT		
Issue No.: 2	Date of issue: July 17, 2019	
Prepared by: QM	Approved by: Tadesse H/Mariam	

Sample Originator: ADAMA UNIVERSITY (ASTU) Request No: 7/2.1.3/Hm80/8898/12
 Sample Type: BIO DIESEL Analytical Method: ASTM /IP/ISO.
 Sample Number: 03 Analytical Result: Interpreted with Stds.
 Lab. Number: 2019-ADO - EXT-7 Report No: QA/0009/2020
 Sample Submitted By: CUSTOMER Date Submitted: 13/03/2020
 Test date: 18/03/2020 Issued Date: 20/03/2020
 Customer Address: - ADAMA Telephone: +251 913 81 31 89

	Property	Test method ASTM	BD Limits	Test results		
				B20	B40	B100
1.	Density@15°C, g/mL	D4052	Report	0.8459	0.8514	0.8886
	Density@20°C, g/mL	D4052	Report	0.8424	0.8479	0.8851
2.	Calorific value, calc, BTU/LB	Calculated		19,556.9	19,507.4	19,172.6
3.	Cloud point, °C	D 2500		+1	+8	+10
4.	Kinematic viscosity @ 40 °c	D 445		3.1871	3.0578	5.2829
5.	Total acidity, mg KOH/g	D974		0.3667	0.400	0.890

Liability: Ethiopian Petroleum supply Enterprise, its Laboratory, its Servants or agents shall not be held liable for any damages, loss, claims or expenses direct or indirect howsoever caused, arising in connection with the laboratory test carried out on **BIODIESEL** sample submitted to it for analysis whether in tort or in contract, due to any act, omission or error of whatever nature, whether or not negligence and howsoever caused. Furthermore all expenses and implied warranties are specifically disclaimed. This test report shall not be reproduced or given to a third party without written approval of the Petroleum Quality Assurance Service.

Lab. expert Senior expert Quality Control Head
 Name: Mekdes Abera Name: Abelaw Ayalew Name: Abelaw Ayalew
 Signature: [Signature] Signature: [Signature] Signature: [Signature]



PQA Lab. Address Addis Ababa Telephone: - +251 115 52 66 63 OR +251 115 51 39 10 Fax +251 115 51 29 38
 PLEASE MAKE SURE THAT THIS IS THE CORRECT ISSUE BEFORE USE

ii. Experimental work recommendation letter

Letter of recommendation as shown below, which explains experimental work performed in the automotive laboratory, mechanical engineering department from Addis Ababa Science and Technology University.

አዲስ አበባ ሳይንስና ቴክኖሎጂ ዩኒቨርሲቲ
ኤሌክትሪካል እና ሜካኒካል ምህንድስና ኮሌጅ
መካኒካል ምህንድስና ትምህርት ክፍል



Addis Ababa Science and Technology University
College of Electrical and Mechanical Engineering
Department of Mechanical Engineering

ቁጥር: 11/01/2013

ቀን Mech/170/13

ለአዳማ ሳይንስና ቴክኖሎጂ ዩኒቨርሲቲ

አዳማ

ጉዳይ: የላብራቶሪ አገልግሎት ያገኙ መሆኑን ሰለማሳወቅ

ተማሪ **ተመስገን አሸናፊ** ለጥናት እና ምርምር የሚሆን ዲግሪ እና ባዮዲግሪ ናሙና ይዘው የቤተሙከራ አገልግሎት እንዲያገኙ ትብብር እንዲደረግላቸው በጠየቃችሁ መሰረት ተማሪው በመካኒካል ምህንድስና ትምህርት ክፍል ሰር በሚገኝ አውቶሞቲቭ ላብራቶሪ Engine dynamometer performance test እና exhaust gas analysis test አገልግሎት ያገኙ መሆኑን እናሳውቃለን።



ከሰላምታ ጋር

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