THE EFFECT OF TURBOCHARGER ON THE PERFORMANCE AND EXHAUST EMISSION OF DIESEL ENGINE RUN ON BIODIESEL FUEL

By

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DECLARATION

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

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DEDICATION

I want to specially dedicate this work to the almighty God for his guidance and protection during my period of study and also to my lovely family from whom I had a lot of moral support.

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In a special way, I recognise the support and encouragements of my family, in the execution of this research work. I would also like to register my profound gratitude to all those who, in one way or another, provided a helping hand in this work. I also like to extend my special thanks and appreciation to the following personalities, the late Mr. & Mrs. Samuel Danno Gaye, for the establishment of the educational foundation from which l benefited, Mr Charles B. Allen, Managing Director Liberia Water and Sewerage Corporation (LWSC), Mr Gabriel S. Flaboe Snr., Manager, ADB, Mr Chris Sherman, Institutional expert (LWSC). Finally, my classmates who were so supportive during my period of study, God bless you all.

ABSTRACT

Biodiesels have been used as renewable alternative fuels for the compression ignition (C.I.) engine. The use of biodiesel fuel in the CI engine leads to substantial reduction in hydrocarbon (HC) and carbon monoxide (CO) emissions. However, the C.I. engine run on petroleum diesel fuel has higher break power and thermal efficiency than when run on biodiesel fuel. A turbocharger is one of the accessories added to the C.I. engine to improve the engine's performance, efficiency and power. In this work, a palm kernel oil biodiesel fuel (B100) of Ghanaian origin is used to study the performance and exhaust emissions of a C.I. engine fitted with a turbocharger.

The experiments were conducted in the automotive engineering laboratory of the Department of Automotive Engineering of the Koforidua Polytechnic where a 4cylinder VW C.I. engine is dedicated to run on biodiesel and petroleum diesel fuels. The VW engine has a rated power of 55 kW, rated speed 4200 rpm and engine cylinder bore of 79.5 mm x 95.5 mm as well as compression ratio of 22.5 and turbocharger capacity of 0.7 bars.

The brake power of the C.I engine, when run on biodiesel fuel, naturally aspirator, recorded 40 kW while recording 50 kW with turbocharger connected respectively at 2200 rpm. In terms of torque, the engine, while running on biodiesel fuel at a speed of 1600 rpm, recorded maximum torque of 195 Nm with natural aspiration and 240 Nm

with the turbocharger. For brake specific fuel consumption (BSFC), the C.I engine, when run on biodiesel fuel, recorded 320 g/kWh with natural aspiration and 250 g/kWh with the turbocharger.

The C.I engine, while running at engine speed of 1600 rpm with the biodiesel fuel recorded 29% with natural aspiration and 36% with turbocharger. The engine, while running at a speed of 1200 rpm, with biofuel, naturally aspirator, recorded CO emissions of 2600 ppm while recording 1700 ppm with the biodiesel fuel and turbocharger (BF–TU) connected. The engine, when run on biodiesel fuel, naturally aspirator, at a speed of 1200 rpm, recorded NOX emissions of 1275 ppm while with the turbocharger connected recorded 1425 ppm.

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ACRONYMS AND ABBREVIATIONS

AC Allis Chalmers

ASTM	American Society of Testing and Materials
B	Biodiesel
BHP	Brake Horse Power
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTE	Brake Thermal Efficiency
CI	Combustion Ignition
CN	Cetane Number
CO_2	Carbon dioxide
DF	Diesel Fuel
DI	Direct Injection
EGR	Exhaust Gas Recirculation
HC	Hydro Carbon
IC	Internal Combustion
NA	Natural Aspiration
NOx	Nitrogen Oxide
PKOME	Palm Kernel Oil Methyl Ester
POME	Palm Oil Methyl Ester
SCR	Selective Catalytic Reduction
SVO	Straight Vegetable Oils
TU	Turbocharger
VW	Volkswagen
WC	Wood Chipper
NA	Naturally aspirator
NOx	Nitrogen Oxide
DF	Diesel Fuel

HC	Hydrocarbon
EGR	Exhaust Gas Recirculation
RPM	Revolutions per Cycle
BMEP	Brake Main Effective Pressure
TDC	Top Dead Centre
BTD	Bottom Dead Centre
FFA	Free Fatty Acids

CHAPTER ONE

1.1 INTRODUCTION

Biodiesel is a renewable source of fuel that is produced from vegetable oils. It is a cleaner burning fuel than petroleum diesel fuel. The physical characteristics of the biodiesel are similar to those of conventional Petroleum diesel making the former a perfect substitute for the later. Biodiesel is used in the compression ignition engine (CI) or the diesel engine. The biodiesel's viscosity is closer to diesel fuel than plants oil. The emissions of CO_2 are greater in biodiesel fuel than in petroleum diesel, indicating more complete combustion when biodiesel replaces petroleum diesel. The emissions of carbon mono-oxide (CO) and hydrocarbon (HC) decrease in the case of biodiesel mixture (Celktn et al. 2010). The temperature of exhaust gases and NOx emissions are similar to or more than those of petroleum diesel.

The biodiesel is an oxygenated fuel hence undergoes better combustion in the engine due to the presence of molecular oxygen which also leads to higher Nox emissions. The higher NOx emission can be effectively controlled by employing exhaust gas recirculation, (EGR). Recycled Exhaust gas lowers the oxygen concentration in the combustion chamber, and increases the specific heat of intake charge that results in lower flame temperature. Reduced oxygen and flame temperature will lead to lower Nox formation. However, EGR may increase if the O_2 in the re-circulated exhaust gas decreases below stoichiometric. Moreover, higher BSFC and particulate emissions would be observed at the exhaust of the engine when run on biofuel with EGR.

Rapidly depleting fossil fuels associated with serious environmental problems has forced the world to search for alternative renewable fuels like bio ethanol and biodiesel.

Straight vegetable oils from which biodiesel is prepared include: soybean, palm, rapeseed, and non-edible oils like jatropha, pongamia, mahua etc. Utilization of these oils directly in engines has received attention as a substitute for diesel but creates serious problems during engine operation, thereby, requiring frequent overhaul and maintenance of the engines. Although, the biodiesel is an oxygenated fuel and burns better, it has 5-12 % lower energy contents than petroleum diesel. Biodiesels can also be used as blends with petroleum diesel without any engine modification (Ma et al. 1999, Kulkarni 2006). And so far, the current practice in most countries is running the CI engine with blends of biodiesel and petroleum diesel fuels. Significant efforts are being made to use different types of fuels in present diesel engines in recent years. The use of straight vegetable oils (SVOs) is constrained by high oil viscosity, which causes delayed combustion, poor fuel atomization, incomplete combustion and carbon deposit in the injector and valve seats creating serious engine fouling (Sun et al. 2010). It is also known that when direct injection engines are fuelled by SVOs, the injectors are choked over few hours due to poor fuel atomization and inferior combustion. One possible solution to tackle such problems is either to blend SVOs with diesel or Trans-esterify SVOs to biodiesel in order to reduce the oil viscosity to the range of petroleum diesel (Peterson et al. 1992). Fuel injection timing of diesel engine is found as the main parameter affecting the combustion and exhaust emissions as evidenced by the fact that air/fuel ratio changes as the injection and the ignition time is varied. Therefore, the variation in injection timing has strong effect on Brake Specific Fuel Consumption (BSFC), Brake Thermal Efficiency (BTE) and NOx emissions due to changes in maximum pressure and temperature in engine cylinder (Borat et al. 2000, Heywood 1984). The fuel pressure in the combustion of diesel engine is a complex process and depends on proper mixing of fuel and air, and timing of injection. The cetane number

(CN) also affects the exhaust emissions, because any increase in CN reduces the ignition delay and increases the injection pressure, thereby, making the fuel particles finer giving lower smoke but higher CO emission (Heywood 1984, Yakup et al. 2003). This work reviews the problems encountered in the use of biodiesels from palm kernel vegetable oil from Ghanaian origin, factors affecting the engine performance and the possible suggested measures. The modification required in existing engines to achieve the targeted engine performance using biodiesel and its blends are also suggested.

1.2 MOTIVATION

Since biodiesel is a fuel which can be created from locally available resources, its production and use can provide a host of economic benefits for local communities. The community-based model of biodiesel production is particularly beneficial. In this model, locally available feedstock's are collected, converted to biodiesel, then distributed and used within the community. This model keeps energy dollars in the community instead of sending them to foreign oil producers and refineries outside the community. The peripheral benefits of this type of model are different for each case, but can include, increased tax base from biodiesel production operations, jobs created for feedstock farming and/ or collection, skilled jobs created for biodiesel production and distribution, income for local feedstock producers and farmers. Sustainable farming and value added agriculture biodiesel feedstock can come from a variety of agricultural crops.

When these crops are grown in a sustainable manner, using good stewardship practices, there are long term benefits to farmers, farming communities and the land.

Many crops which yield oils used for biodiesel production can be cultivated in rotation with other food crops including soybeans, corn, canola and wheat. Using crops in rotation can improve soil health and reduce erosion. The overall impacts of growing energy crops are complex with thousands of variables. However, the added value created for oilseed crops by the production of biodiesel is a tangible benefit for farming communities and when coupled with sustainable farming practices can provide benefits to farming communities and the environment.

As a renewable, sustainable and alternative fuel for compression ignition (CI) engines, interest in use of biodiesel in CI engine increased recently in order to study its effects on engine performance and emission. With commercialization of bioenergy, it has provided hope for solution to the fossil fuel depletion and its negative influence on the environment. Though many researchers pointed out that it might help to reduce greenhouse gas emissions, promote sustainable rural development, and improve income distribution, there still exists some resistance for using it. The primary cause is lack of new knowledge about the influence of biodiesel on diesel engine. One of such is the effect of turbocharger on biodiesel fuelled engine’s emission to which this work contributes.

Turbocharger has long been the standard technology used to boost the diesel engine power in passenger vehicles, on the highway trucks and stationery machines. Although majority of the gasoline engines are still Natural Aspirationtoday, a Natural Aspirationdiesel engine wastes a high proportion of heat energy released in the cylinder which exhausts to the ambient.

Some of the wasted heat energy in the exhaust gas can be recovered by a turbocharger and covered to useful work. In fact, turbocharger is widely employed in current diesel engines. The turbine of a turbocharger is driven by the energy available in the exhaust gas and its compressor, which is connected to turbine with a shaft, increases the pressure of the air supplied to the engine thus increase the air temperature. In this study, the performance parameters and exhaust emission of the four- cylinder, four- stroke diesel engine using biodiesel, from palm kernel oil methyl ester (PKOME) would investigated for the cases of both Natural Aspiration(NA) and turbocharger (TU) conditions. The test would be performed at various speeds and full load conditions, and the experimental results compared with each other.

1.3 PROBLEM STATEMENT

Biodiesel is an alternative fuel used in the C.I or diesel engines. However, when the diesel engine is run on biodiesel fuel, the torque and power developed are lower than using petroleum diesel fuel. The thermal efficiency of the C.I engine is also lower when run on biodiesel than when run on petroleum diesel. Optimization of the diesel engine to run on biodiesel includes the use of turbocharger connected to the C.I engine to improve the torque and power developed. This work focuses on experiments to determine the performance of a dedicated VW C.I engine run on a biodiesel fuel with a turbocharger attached.

1.4 MAIN OBJECTIVE

The aim of this study is to investigate the influence of turbocharger on the diesel engine which is run on palm kernel oil biofuel.

1.5 SPECIFIC OBJECTIVES

1. To determine the effects of turbocharger on fuel consumption of a diesel engine run on biodiesel fuel compared to a petroleum biodiesel fuel.

2. To determine the effects of turbocharger on the C I engine performance parameters: including engine torque, brake power, brake specific fuel consumption (BSFC), brake thermal efficiency (BTE) and thermal efficiency.
3. To determine the effects of turbocharger on CO and O_2 exhaust emissions.
4. To determine the thermal efficiency of the diesel engine with and without a turbocharger.

1.6 STRUCTURE OF THESIS

This thesis is organized into five chapters. The first chapter consists of an introduction, motivation, problem statement and objectives of the study. Chapter Two contains a literature review of diesel engine performance, the effect of biodiesel content on engine performance, theory of turbocharger and its effects on engine performance

Chapter Three presents the description of the experimental set-up and the procedures to achieve the specific objectives. The results and discussions of the experimental work to achieve the specific objectives are presented in Chapter Four. Chapter Five contains the conclusion and recommendation.

CHAPTER TWO

2 LITERATURE REVIEW

This chapter contains a literature review of diesel engine performance, the effect of biodiesel content on engine performance, theory of turbocharger and its effects on engine performance.

2.1 BIODIESEL FUELS

Biodiesel is the mono-alkylesters of fatty acids made after edible and non-edible oils or animal butters. The edible vegetable oils like sunflower, rapeseed, palm, soybean oil, etc., are constrained by high prices and competition with food. Alternately, the non-edible oils like jatropha, pongamiapinnata, linseed, mahua, neem, etc., are usually used for biodiesel production as these oils are not used as food.

2.2 ADVANTAGES AND DISADVANTAGES OF BIODIESEL FUEL

Biodiesel is miscible with petroleum diesel in any proportions with several advantages over absolute petroleum diesel. Biodiesel has better lubrication, lower toxicity, renewability and indigenous source of feedstock's, higher flash point, high biodegradability, negligible sulphur, and overall lower exhaust emissions. Its disadvantages include high feedstock cost, poor storage and oxidative stability, lower energy content, poor low-temperature operability and higher NOx productions (Oliveira et al. 2006, Knothe 2008). The low temperature operability and oxidation stability can be minimized by adding cold flow properties improvers and anti-oxidants to biodiesel and by blending with diesel. The methods to enhance the low-temperature engine operability are crystallization, fractionation and trans-esterification using long or branched-chain (Chiu et al. 2004, Soriano et al. 2006).

Strategies to improve the exhaust emissions of biodiesel and its blends include Selective Catalytic Reduction (SCR) and Exhaust Gas Recirculation (EGR), diesel oxidation catalysts traps of NOx or particulate matters (Mc 2004, Knothe et al. 2006). Presently, the cost of the feedstock accounts for over 80% of biodiesel production cost, which is one of the serious obstacles in the economic capability of biodiesel production to which can be improved by using cheaper feedstock's like soap stocks, acid oils, waste cooking oils, waste micro algal oils, etc. Some feedstock may contain high free fatty acids (FFA) and water which can affect the biodiesel production process significantly (Dwivedi et al. 2013). The cost of biodiesel, however, depends on the economics of scale of industrial and the political result to stimulate biodiesel production, especially, in developing countries like Ghana. This factors like increased rural employment opportunities, indigenous energy sufficiency, and saving foreign exchange and environmental benefits, may are considered to improve the overall economic viability of biodiesel production (Openshaw, 2000).

2.3 TECHNIQUES USED TO IMPROVE THE PERFORMANCE OF A DIESEL ENGINE

Maximizing diesel engine performance is an important goal in diesel vehicles. The diesel engines need care and mechanical adjustments to improve diesel power.

There are a few ways to increase engine horsepower;

i. Larger engine displacement: This method of increasing engine performance is also efficient but it‟s costly and adds unwanted weight to the vehicle.

ii. Special fuel injectors: These fuel injectors are made with proprietary processes to create the right hole pattern necessary to get the most precise and optimal fuel spray. Its demerit is that one cannot quantify how much gain though it is noted to be able to get an increase of 50-100 diesel horsepower, and it is not easily accessible.

iii. Diesel fuels additives: Cetane improver is another way one can improve or increase diesel power and performance. Cetane can be found in engine oil products such as Super-Tane. Its high cetane rating improves drivability. The diesel engine runs smoother, less engine sound and improves combustion. It os relatively cost effective. Nitrous oxide in engine oil product on the other hand has a short duration for increasing engine performance.

iv. Turbocharger: A turbocharger is an apparatus that forces more air into the combustion chamber than it will normally thus more oxygen available and better temperature regulation. It is lightweight, relatively cheap, and provides a continuous supply of power.

2.4 PROBLEM WITH C.I ENGINE RUN ON BIODIESEL FUEL

Extensive literature survey reveals that most of the engine problems can be attributed to poor quality biodiesel. Some of the problems (primarily coldweather problems) are not due to poor fuel quality but are related to the biodiesel fuel properties. Most of these problems can be avoided or minimized. Table.1 reviews the possible engine problems while using

biodiesel and its blends.

Table 1.1 Details of Engine Problems and Suggested Remedial Measures When Biodiesel and Its Blends Were Used As C.I Engine Fuel

Engine trouble	Remedial Measures Suggested
Deposits on injectors affecting the fuel spray patterns	Injectors may be periodically cleaned. Using specialized cleaning equipment.

Cold-weather operation of engine using partially solidified or partially transformed biodiesel.	Use of Low-temperature properties improvers to improve the engine operation in cold conditions. To ensure complete conversion of oils to biodiesel free from contaminants.
Engine starting problems under cold weather conditions or run only a few seconds. Engine stops after operation for few seconds	Warm the fuel filter using 12-volt jacket heaters. Use additives to avoid gum/particles formation in biodiesel.
Fuel filter clogging due to poor biodiesel quality and also formation of resins or gels in the fuel supply system	The problem of algae build up can be removed by adding suitable algaecide. Use of moisture free fuel is recommended.

2.5 ENGINE PERFORMANCE USING BIODIESEL FUEL

Studies on biodiesel use in CI engines are still ongoing. Generally, quality biodiesel fuel performs well in the diesel engine. However, some well-known important points about biodiesel fuel use in diesel engine to note are as follows:

i. Engine power: Power and torque tend to be 3 to 5 percent lower when using biodiesel. Due to the fact that biodiesel has less energy per unit volume than traditional diesel fuel.

ii. Fuel efficiency: fuel efficiency tends to be slightly lower when using biodiesel due to the lower energy content of the fuel. Typically, the drop-off is in the same range as the reduction in peak engine power (3–5 percent).

iii. Engine wear: short-term engine wear when using biodiesel has been measured to be less than that of petroleum diesel. Engines are expected to experience less wear in the long run when using biodiesel.

iv. Deposits and clogging: deposits and clogging due to biodiesel have been widely reported but are generally traceable to biodiesel that is either of low quality or has become oxidized. If fuel quality is high, deposits in the engine should not normally be a problem.

v. Pollution from engine exhaust: biodiesels results in much less air pollution due to its higher oxygen content and lack of both "aromatic compounds" and sulphur. The one exception to this is nitrogen oxide (NO_x) emissions, which tend to be slightly higher when using biodiesel. However, proper tuning of the engine can minimize this problem.

Cold-weather performance: similar to petroleum diesel, engines tested in cold weather typically experience significant problems with operation caused primarily by clogging of the filters and/or coking of the injectors. The use of flow improving additives and "winter blends" of biodiesel and kerosene has proved effective at extending the range of operating temperatures for biodiesel fuel. Pure biodiesel tends to operate well at temperatures down to about 5°C (this varies noticeably depending on the type of oil used). Additives typically reduce that range by about 5 to 8 degrees, while winter blends have proved effective at temperatures as low as -20°C and below (*FreePubs, 2014*)

2.6 CRITERIA FOR A GOOD ENGINE FUEL

In IC engines, the thermal energy is released by burning the fuel in the engine cylinder. The combustion of fuel in IC engine is quite fast but the time needed to get a proper air/fuel mixture depends mainly on the nature of fuel and the method of its introduction into the combustion chamber. The fuel should therefore have the following characteristics or properties;

i. High energy density ii. Good combustion characteristics iii. High thermal stability iv. Low deposit forming tendencies

i. Compatibility with the engine hardware ii.

vi. Good fire safety vii. Low toxicity viii. Less pollution ix. Easy transferability and onboard vehicle storage

The combustion process in the cylinder should take as little time as possible with the release of maximum heat energy during the period of operation. Longer operation results in the formation of deposits which in combination with other combustion products may cause excessive wear and corrosion of cylinder, piston and piston rings. The combustion product should not be toxic when exhausted to the atmosphere. These requirements can be satisfied using a number of liquid and gaseous fuels. The biodiesel from non-edible sources like jatropha, pongamia, mahua, neem, etc., and other edible vegetable oils like palm kernel oil, soya, meet the above engine performance requirement and therefore can offer viable alternative to diesel fuel in Ghana and other countries.

2.7 THE DIESEL ENGINE

A diesel engine is an internal burning engine. The cylindrical stroke cycle of this engine is the same in a gasoline engine, supposing it is a four-stroke engine (ref. Figure 2.1).

Apart from the engines that use less fuel, the major difference between these two engines is the combustion itself. A gasoline engine also uses a spark plug to initiate burning. A diesel engine compresses the air then injects the fuel into the cylinder at the top of the stroke. The high temperature of the compressed air ignites the fuel. The hot gases expand, force the piston down, and create a torque on the crankshaft. The final stroke is the exhaust stroke, which releases the hot gases into the exhaust system.

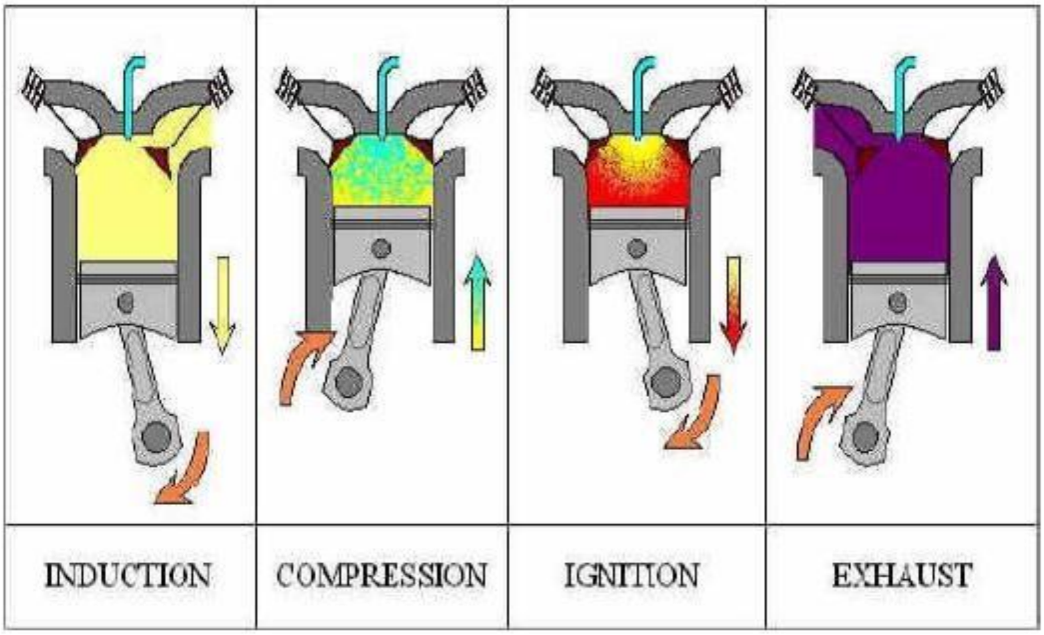


Fig 2.1 Four-Stroke Cycle Diesel Engine (*Diesel Engine, 2014*)

The torque created from the downward motion of piston acting on the crankshaft is transmitted from the crankshaft to flywheel and into the transmission. To increase power to the transmission, the power must be increased in the cylinder. There are several ways to increase the power of an engine. One of the more common ways of increasing engine power is to increase the airflow into the cylinder by increasing the density of the air entering the cylinder. A turbocharger uses waste energy from outside into the exhaust system to compress air entering the cylinder, thus increasing engine power.

2.8 TURBOCHARGER

Before you can truly appreciate what a turbo does for an engine, you need to understand the basics of internal combustion engines. Internal combustion engines are "breathing" engines. That is to say, they draw in air and fuel produce energy. This energy is realized as power when the air-fuel mixture is ignited. Afterward, the waste created by the combustion is expelled. All of this is typically accomplished in four strokes of the pistons.

What a turbocharger does is to make the air-fuel mixture more combustible by sucking more air into the engine's combustion chambers which, in turn, creates more power and torque when the piston is forced downward by the resulting explosion. It accomplishes this task by condensing, or compressing, the air molecules so that the air the engine draws in is denser. Now, how it does that is the following.

Turbocharger is an air pump that increases pressure in the engine by taking extra air from outside. . Is a tunnel like reaction engine in which air is drawn in, compressed by spinning blade attached to the turbine shaft, and mixed with atomized fuel, with the resultant mixture being ignited in combustion chamber to produce a power full jet that drive the engines turbine and provides thrust, directly to the turbine wheel side of the turbocharger to make it rotate. That turbine wheel is connected by a shaft to a compressor wheel. As the turbine wheel spins faster and faster, it causes the compressor wheel to also spin quickly. The rotation of the compressor wheel sucks, in ambient air and compresses it before pumping it into the engine's combustion chambers. (Fig 1.2 and 1.3)

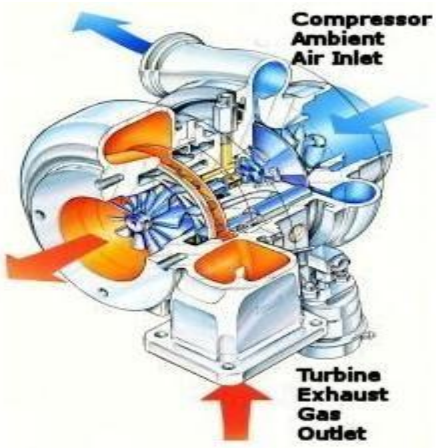


Fig.2.2.Turbocharger (Kadekticaret-turbocharger, 2014)

The compressed air leaving the compressor wheel housing is very hot as a result of both compression and friction. Thus the compressed air is passed through heat exchangers also called charge air cooler (Fig 1.4) to reduce its temperature as it enters the chambers. That is where a charge-air cooler (or "heat exchanger") comes in. It reduces the temperature of the compressed air so that it is denser when it enters the chamber. The air cooler also helps to keep the temperature down in the combustion chamber.

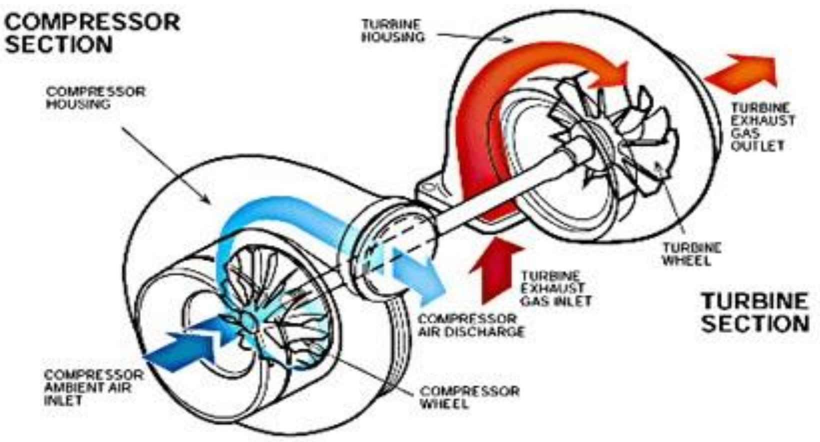


Fig.2.3 Section of Turbocharger (Kadekticaret-turbocharger, 2014)

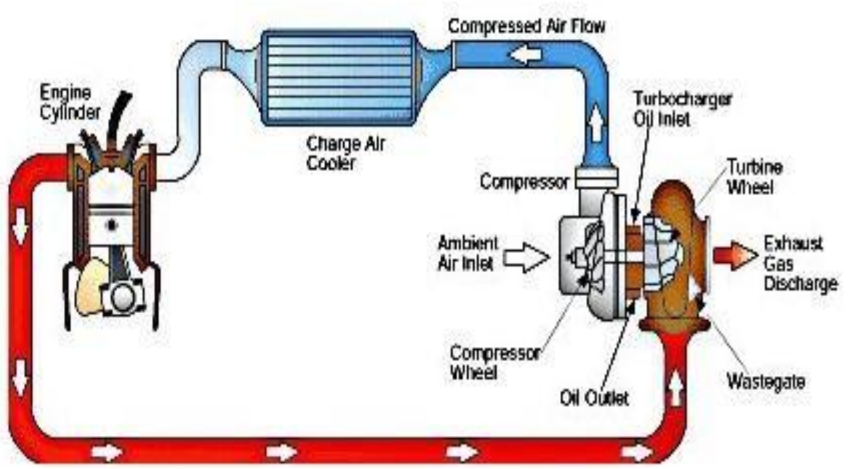


Fig.2.4 Turbocharger on Engine Assembly (Kadekticaret-turbocharger, 2014)

The basic principle behind turbocharging is fairly simple, but a turbocharger is a very complex piece of machinery (fig 1.4) Not only must the components within the turbocharger itself be precisely coordinated, but the turbocharger and the engine it services must also be exactly matched for efficient operation. Thus it is important to follow correct installation, operation and preventive maintenance procedures.

2.9 PERFORMANCE OF DIESEL ENGINE

Literature survey reveals that biodiesel perform satisfactorily during diesel engine operation and B_{20} blend provides the fuel economy almost similar to the diesel. Due to its high lubricity, biodiesel causes less wear and tear to engine parts. Studies have been reported on the performance and emission of CI engines, fuelled by B_{100} biodiesel as well as its blends with diesel. The biodiesel's oxygenated nature leads to more complete combustion, resulting in lower emission due to higher combustion temperature. The biodiesel blends with diesel give performance similar to diesel as the fuel properties of biodiesel and diesel are similar with the cetane number, flash point and lubricity of biodiesel been higher while the calorific value is lower. The following parameters are used to evaluate the performance of diesel engine using biodiesel and its blends:

2.9.1 Brake Specific Fuel Consumption (BSFC):

The BSFC is define as the fuel flow rate per unit of power output. It is a measure of the efficiency of the engine in using the fuel supplied to produce work. It is desirable to obtain a lower value of BSFC meaning that the engine used less fuel to produce the same amount of work. It can be calculated as;

BSFC (kg/kWh) $=W_f/P_b$

Where, W_f = fuel consumed (kg/h)

P_b = brake power (kW) which can be calculated by: $P_b = P_e / \eta_e$

Where, P_e = load (kW) on engine, η_e = efficiency of the engine

2.9.2 Brake Mean Effective Pressure (BMEP):

BSEC is defined as the amount of energy consumed per kilo-watt power developed in the engine in one hour. For the comparison of economy of two fuels, brake specific energy consumption is the better way of judgment as compared to brake specific fuel consumption because the heating value and density of the fuels exhibit slight different trends. BMEP is an important parameter for comparing the performance of different fuels and defined as the average pressure the engine can exert on the piston through one complete operating cycle. It is the average pressure of the gas in the fuel mixture inside the engine cylinder based on net power. BMEP is independent of the RPM and size of the engine. If N is the number of revolutions per second, and N_c the number of revolutions per cycle, the number of cycles per second is just their ratio

(W) which can be expressed by: (Borat et al., 2000 and Heyood, 1984)

$$W = \frac{PNc}{N}$$

2.9.3 Brake Horsepower (BHP):

It is the measure of an engine's horsepower before the loss in power caused by the gearbox, alternator, water pump, and other auxiliary components like power steering pump, muffled exhaust system, etc. Brake refers to a device used to load an engine and hold it at a desired RPM. During testing, the output torque and rotational speed can be measured to determine the brake horsepower which is the actual shaft horsepower and is measured by the dynamometer by;

$$BHP = IHP - FP$$

Where BHP is brake horse power and IHP is indicated horse power while FP is frictional power. The indicated power is produced from the fuel inside the engine while some power is lost due to friction and the remaining power available at the shaft of the engine is brake horse power.

2.9.4 Mechanical Efficiency:

Part of the indicated work per cycle is used to expel exhaust gases, induct fresh air, and also overcome the friction of the bearings, pistons, and other mechanical parts of the engine. The mechanical efficiency is the measure of the ability of the engine to overcome the frictional power loss and can be defined as:

$$\text{Mechanical Efficiency} = \frac{Work\ output}{Work\ Input}$$

The work output is also defined as brake horse power and input is indicated horse power and the ratio of BHP to IHP is defined as mechanical efficiency.

2.9.5 Brake Thermal Efficiency (BTE):

It is the ratio of the thermal energy in the fuel to the energy delivered by the engine at the crankshaft. It greatly depends on the manner in which the energy is converted as the efficiency is normalized with respect to the fuel heating value. It can be expressed by:

BTE (η_b) = $P_b / (m_f \times NCV)$

Where, P_b = brake power (kW)

m_f = fuel consumption (kg/s)

NCV = net calorific value (kJ/kg)

2.10 THEORY OF TURBOCHARGER

The most common turbocharger assembly used in the automotive industry consists of a radial compressor coupled to a radial turbine as shown in Figure 2.5.

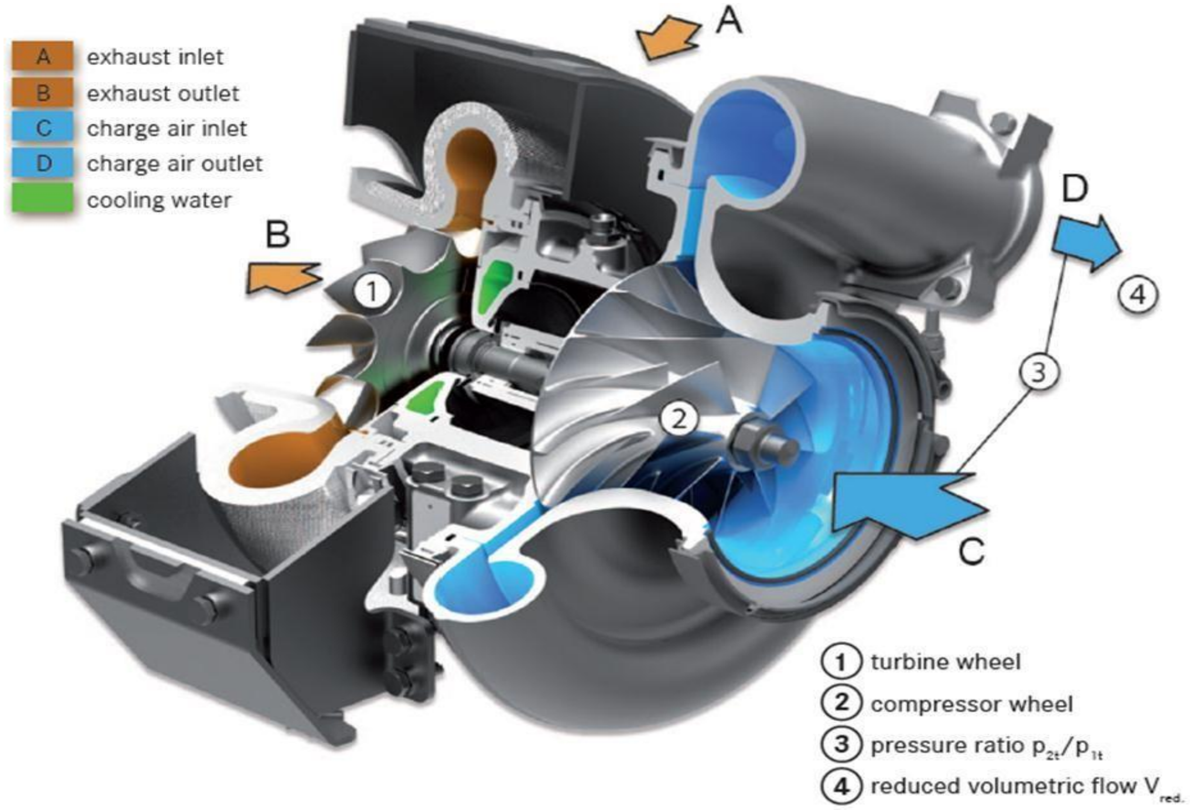


Fig. 2.5 Sectional View of a Turbocharger (Turbocharger, 2013)

The bearings are generally of the plain journal bearing type; however, for racing applications ceramic ball bearings are being used more frequently. On big engines such as those used for rail and marine applications, where the operating range is very narrow and operation is mostly steady state, an axial turbine coupled to a radial compressor is the most common configuration. Axial turbines are preferred for their superior efficiency to those of a radial turbine, but a radial turbine operating range is much wider. This makes radial turbines more suitable for automotive applications, where the operating range is very wide. Radial compressors also have a much wider operating range and are thus more widely used than axial compressors in turbocharger applications. Radial compressors are limited to a pressure ratio of about 3.5, because higher pressure ratios will cause supersonic flow and cause shockwaves to form at the compressor inlet. This will cause a rapid deterioration in the compressor efficiency.

2.11 TOTAL AND STATIC PRESSURE AND TEMPERATURE

The static pressure (P_1) of a fluid flowing in a duct is that measured at the surface of the wall. The total or stagnation pressure (P_{01}) is the pressure that will be measured in the stream if the fluid were brought to rest isentropically. Thus P_{01} can be related to P_1 as in Eq. 2.1

$$P_{01} = P_1 \left(\frac{T_{01}}{T_1}\right)^{\gamma/(\gamma-1)} \qquad (2.1)$$

Where gamma (γ) represents the polytropic coefficient (ratio of specific heats). Similarly the static temperature (T_1) is the free stream temperature and the total (or stagnation) temperature (T_{01}) is the temperature that will be measured if the gas were brought to rest. For a perfect gas it can be shown that Eq. 2.2 holds.

$$T_{01} = T_1 + \frac{1}{2}\frac{C_1{}^2}{c_p} \qquad (2.2)$$

Where C_1 is the velocity of the gas and Cp the specific heat at constant pressure.

2.12 THE RADIAL COMPRESSOR

In Figure 2.6 shown below, the three important parts of a radial compressor: impeller, diffuser ring and volute casing. In some applications there might be a diffuser ring included. The diffuser ring is optional and may or may not be present depending on size, use and cost of the compressor. The impeller is a solid rotating disc with curved blades standing out axially from the face of the disc. In most turbocharger applications the blade tips are left open and the casing of the compressor itself forms the solid outer wall of the blade passages. In some cases the blade tips may be covered with another flat disc to give shrouded blades. The advantage of the shrouded blade is that no leakage can take place from one passage to the next. The disadvantage of having shrouded blades is extra weight and a more complicated manufacturing process. In turbocharger applications where very high rotational speeds are required, the disadvantage of leakage is more than offset by the reduced weight of the impeller

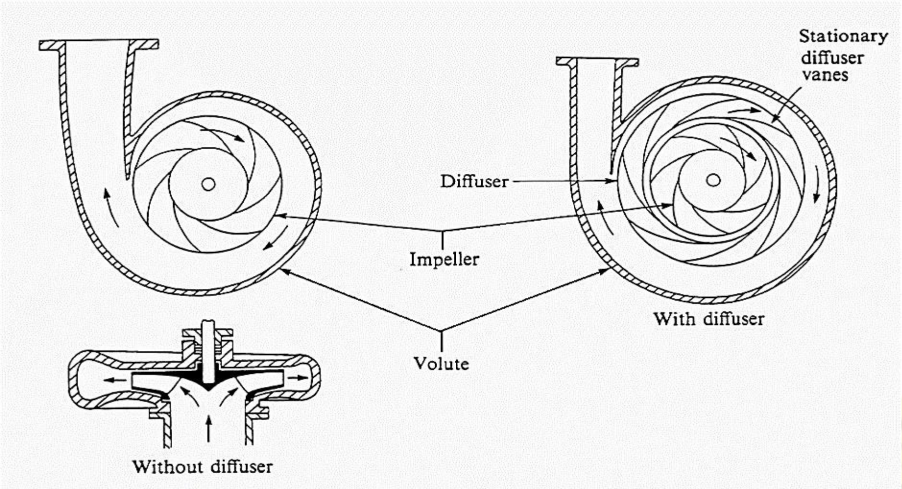


Fig. 2.6 Components of a Radial Compressor (Sayers, 1990)

As the impeller rotates, the fluid (air) that is drawn into the blade passages at the impeller inlet is accelerated as it is forced radially outwards. In this way, the static pressure at the outlet radius is much higher than at the inlet radius. The fluid has a very high velocity at the outer radius of the impeller and, to recover this kinetic energy by changing it to pressure energy, diffuser blades mounted on the diffuser ring may be used. The stationary blade passages so formed have an increasing cross- sectional area as the fluid moves through them, the kinetic energy of the fluid being reduced, while the pressure energy is further increased. Vane less diffuser passages may also be utilized.

Finally, the fluid moves from the diffuser blades into the volute casing, which collects it and conveys it to the compressor outlet. As the fluid moves along the volute casing, further pressure recovery occurs. Sometimes only the volute casing exists without the diffuser. This process can be plotted on an enthalpy versus entropy diagram as shown in Figure 2.7, so that any departures from isentropic compression can be shown. In figure 2.7, station 01 represents ambient pressure of the air. Acceleration of the fluid in the inlet causes a pressure drop from P_{01} to P_1 or P_{00} to P_1 when considering losses in the inlet, the change in enthalpy being equivalent to the increase in kinetic energy ($C_1^2/2$). Isentropic compression to the delivery stagnation pressure P_{4s} is shown by the vertical line 01-4s. Energy transfer to the fluid takes place in the impeller and the line 1-2 indicates this process. The corresponding isentropic process is shown by 1-2s. If the total kinetic energy of the fluid leaving the impeller ($C_2^2/2$) were converted to pressure, isentropically, the delivery pressure would be P_2 (point 2). This describes the basic working of a radial compressor (Sayers 1990, Watson and Janota 1984).

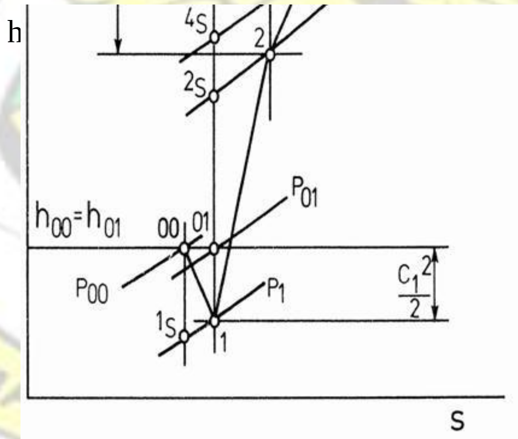


Fig.2.7 h-s Diagram for a Radial Compressor (Watson & Janota, 1984)

2.13 COMPRESSOR EFFICIENCY

The efficiency of the radial compressor can be defined as the work required for ideal adiabatic compression divided by the actual work required to achieve the same pressure ratio. From the second law of thermodynamics it is clear that this definition is equivalent to Eq 2.3.

$$\eta_C = \frac{isentropic\ work}{actual\ work} \qquad (2.3)$$

From the first law of thermodynamics, assuming that the heat transfer rate to and from the compressor can be neglected as well as the change in potential energy, Eqn. 2.3 can be rewritten in the following form;

$$\eta_{cTT} = \frac{h_{02s}-h_{01}}{h_{02}-h_{01}} \qquad (2.4)$$

Assuming that air is a perfect gas, thus c_p is constant

$$\eta_{cTT} = \frac{T_{02s}-T_{01}}{T_{02}-T_{01}} \qquad (2.5)$$

The expressions are for total-to-total isentropic efficiency.

An evaluation based on Eqn. 2.5 assumes that all the kinetic energy at the compressor outlet can be used. This is true in the case of a gas turbine, since the velocity at the compressor delivery is maintained at the combustion chamber. However, the compressor of a turbocharger must supply air via a relatively large inlet manifold to the cylinders. Hence the engine will only 'feel 'the static pressure at the compressor delivery and is unlikely to benefit from the kinetic energy at the compressor outlet. Thus a turbocharger compressor should be designed for high kinetic to potential energy conversion before the outlet duct. Since the engine benefits little from the kinetic energy of the air leaving the compressor, a more realistic definition of the compressor efficiency is based on static delivery temperature as in Eqn. 2.6, where T_{2S} denotes total-to-static.

$$\eta_{cTT} = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} \quad (2.6)$$

It is common practice for manufacturers to quote total-to-total efficiencies for turbocharger compressors, and quite often those are quoted without declaring the basis on which the efficiency values are calculated.

2.14 THE RADIAL TURBINE

The radial flow turbine is main used in turbo charger. It consists of a scroll or inlet casing, a set of inlet nozzles (sometimes omitted) followed by a short vane less gap and the turbine wheel itself (Figure 2.8). Most small turbocharger turbines use a vane less casing; the nozzle is then in the form of a slot running all the way between the scroll and turbine wheel. A vane less casing can be used to improve flow range at some penalty in peak performance, while also reducing cost. However, considering the more conventional type with nozzles, the function of the inlet casing is purely to deliver a uniform flow of inlet gas to the nozzle entries. The nozzles accelerate the flow, reducing pressure and increasing the kinetic energy. A short vane less space prevents the rotor and nozzle blades from touching and allows wakes coming off the trailing edge of the nozzle blades to mix out. Energy transfer occurs solely in the impeller, which should be designed for minimum kinetic energy at the exit.

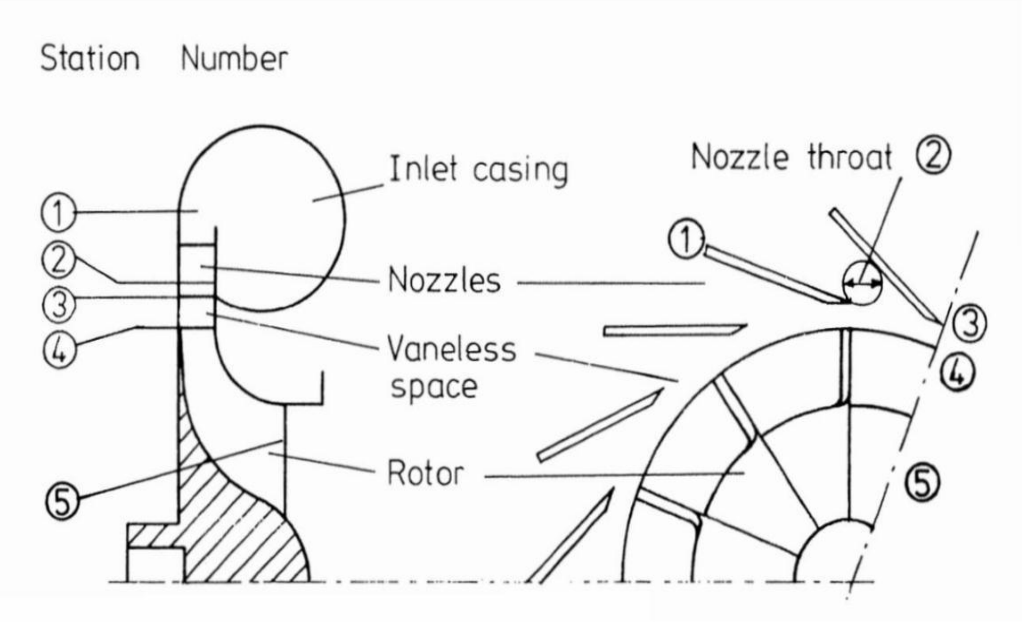


Fig.2.8. Components of a Radial Turbine (Watson & Janota, 1984)

The thermodynamic states of the flow process through the turbine may be plotted on an enthalpy versus entropy diagram as shown in Figure 2.9. Station 01 refers to stagnation conditions at the entry to the casing. The gas will already have a significant velocity (C_1), hence the stagnation pressure is P_{01}. The inlet nozzles accelerate the flow from station 1 to 2. If this process were isentropic, the end point would be 2s. Energy transfer occurs in the rotor, between station 4 and 5 (4 and 5s if isentropic) down to the exit pressure P_5. The stagnation P_{05} will be higher than P_5 since the exit velocity will remain significant. Station 3 is the nozzle exit or the nozzle throat, denoted as station 2.

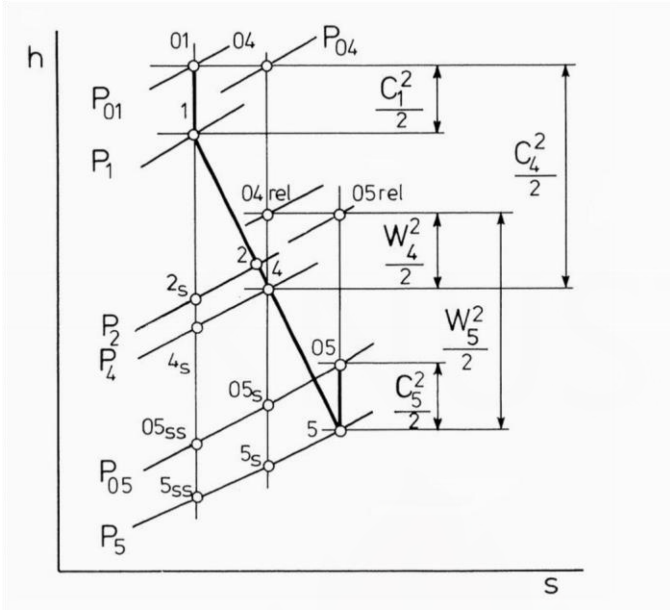


Fig. 2.9 h-s diagram for a radial turbine (Watson & Janota, 1984)

2.15 TURBINE EFFICIENCY

The isentropic efficiency of a turbine may be defined as the actual work output divided by that obtained from reversible adiabatic (isentropic) expansion between the same two pressures.

$$\eta_t = \frac{actual\ work}{isentropic\ work} \qquad (2.7)$$

Assuming a perfect gas (cp = constant) and following the same reasoning as with compressors, it can be shown that Eqn. 2.7 can be expressed in terms of temperatures as in Eqn. 2.8.

$$\eta_{nT} = \frac{T_{03}-T_{04}}{T_{03}-T_{04s}} \qquad (2.8)$$

The total-to-total efficiency given in Eqn 2.8 assumes that the kinetic energy leaving the turbine exit can be harnessed. In most applications this is not possible. The energy leaving the turbine exit goes to waste through the exhaust pipe. Thus a more relevant isentropic efficiency could be based on the static exit temperature. The total-to-static isentropic efficiency would be defined as the actual work output divided by isentropic expansion between the stagnation inlet and static outlet pressures.

$$\eta_{nT} = \frac{T_{03}-T_{04}}{T_{03}-T_{4s}} \qquad (2.9)$$

2.16 ENERGY AVAILABLE IN THE EXHAUST GAS

In Figure 2.10, shows the ideal limited pressure engine cycle in terms of a pressure/volume diagram for a Natural Aspirationengine. Superimposed is a line representing isentropic expansion from point 5, at which the exhaust valve opens, down to the ambient pressure (Pa), which could be obtained by further expansion if the piston were allowed to move to point 6. The shaded area 1-5-6 represents the maximum theoretical energy that could be extracted from the exhaust system; this is called the blow-down energy.

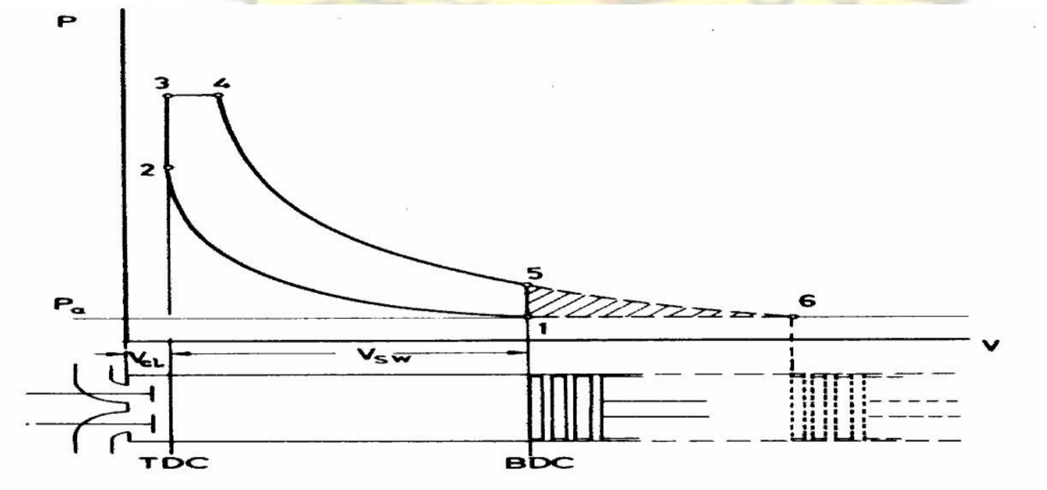


Fig.2.10. Natural AspirationIdeal Limited Pressure Cycle (Watson & Janota, 1984)

Consider now the turbocharged engine; the ideal four-stroke pressure/volume diagram would appear as shown in Figure 2.11, where P_1 is the turbocharging or boost pressure and P_7 is the exhaust manifold pressure. Process 12-1 is the induction stroke, during which fresh air at the compressor delivery pressure enters the cylinder. Process 5-1-13-11 represents the exhaust process. When the exhaust valve first opens (point 5) some of the gas in the cylinder escapes to the exhaust manifold expanding along 5-7, if the expansion is isentropic. Thus the remaining gas in the cylinder is at P_7, when the piston moves toward top dead centre (TDC), displacing the cylinder contents through the exhaust valve against the backpressure P_7. At the end of the exhaust stroke the cylinder retains a volume (V_{cl}, clearance volume) of residual combustion products, which for simplicity can be assumed to remain there. The area 7-8-10-11 will represent the maximum possible energy that could be extracted during the expulsion stroke, where 7-8 represents isentropic expansion down to the ambient pressure.

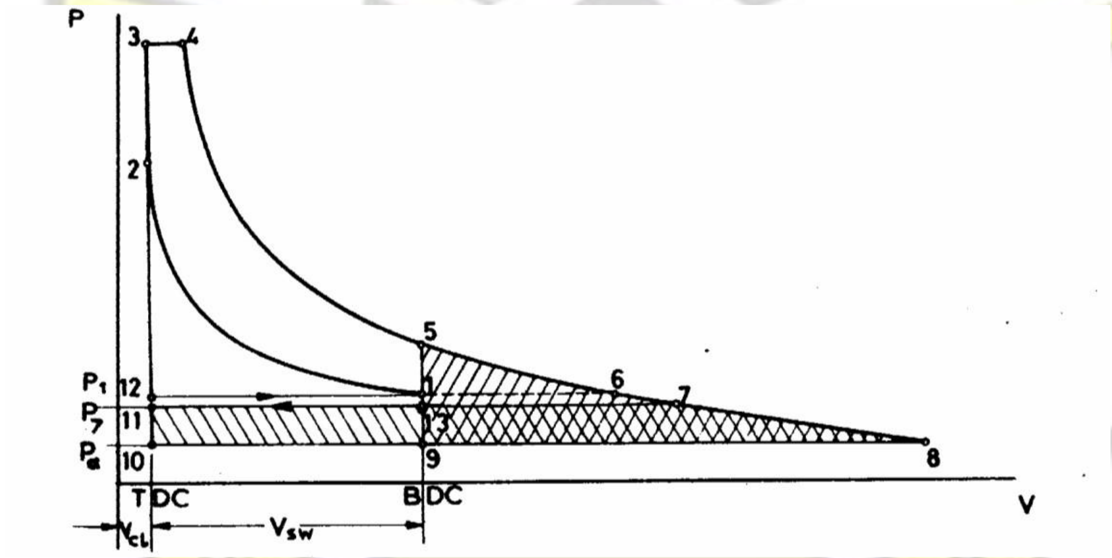


Fig. 2.11 Turbocharged Ideal Pressure Limited Cycle (Watson & Janota, 1984)

There are two distinct areas in Figure 2.11 representing energy available from the exhaust gas, the blow-down energy (area 5-8-9) and the work done by the piston (area 13-9-10-11). The maximum possible energy available to drive the turbocharger turbine will clearly be the sum of these two areas. Although the energy associated with one area is easier to harness than the other, it is difficult to devise a system that will harness all the energy. To harness all the energy, the turbine inlet pressure must rise instantaneously to P_5 when the exhaust valve opens, followed by isentropic expansion of the exhaust gas through P_7 to the ambient pressure (P_8=P_a). During the displacement part of the exhaust process (expulsion stroke) the turbine inlet pressure must be held at P_7. Such a series of processes is impractical.

Consider the simpler process in which a large chamber is fitted between the engine and the turbine inlet, in order to damp out the pulsating exhaust gas flow. By forming a restriction to flow, the turbine may maintain its inlet pressure at P_7 for the whole cycle. The available work at the turbine will then be given by area 7-8-10-11. This is the ideal constant pressure turbocharging system. Next, consider an alternative system, in which a turbine wheel is placed directly downstream of the engine close to the exhaust valve. If there were no losses in the port, the gas would expand directly out through the turbine alone line 5-6-7-8, assuming isentropic expansion. If the turbine area were sufficiently large, both cylinder and turbine inlet pressures would drop to P_9 before the piston has moved significantly up the bore. Hence the available energy at the turbine would be given by area 5-8-9. This can be considered the ideal pulse turbocharging system. The systems commonly referred to as “constant pressure turbocharging” and “pulse turbocharging” are based on the above principles, but in practice they differ from the ideal theoretical cycles.

2.17 SUMMARY OF LITERATURE

In light of an extensive literature, the performance of a diesel engine with petroleum diesel fuel and biodiesel was reviewed and biodiesel was considered to be a better option with its competitive advantage in reduction in exhaust emission gases. Engines with turbocharger installed, records relative improvement in fuel economy and engine performance. The next chapter, methodology explains experimental procedures undertaken to achieve my specific objectives.

CHAPTER THREE

3 METHODOLOGY

This chapter presents the description of the experimental set-up. The experimental procedures to achieve the specific objectives are also cited in this chapter.

3.1 EXPERIMENTAL SET UP AND TEST PROCEDURE

The experiments were conducted in the automotive engineering laboratory of the Department of Automotive Engineering of the Koforidua Polytechnic where a 4cylinder VW C.I. engine is dedicated to run on biodiesel and petroleum diesel fuels.

The following devices were also installed and used for the experiments,

i. A diesel engine stand was fabricated to accommodate a VW diesel engine ii. A VW diesel engine with the specifications in Table 3.1 below was mounted on the stand iii. Exhaust gas analyzer machine was used to measure CO and NOx emissions iv. A turbocharger with boost power of 0.7 bar was mounted on the diesel engine.

v. Four-gas exhaust analyzer was used to measure CO and NOx emissions vi. Fuel consumption meter was used to measure rate of fuel consumption vii. The biodiesel fuel used was acquired from the renewable energy department of Koforidua Polytechnic – Ghana

3.2 THE VW DIESEL ENGINE

The experimental set up consists of a direct injection diesel engine, fuel consumption metre and Exhaust gas analyzer. The specifications of the four-stroke, direct injection (DI), Turbocharging diesel test engine are given in Table 3.1. Two fuel tanks, one for diesel fuel and another one for biodiesel fuel was used to supply the fuels to the test engine. The fuel delivery was controlled by the fuel injection pump installed on the engine. Fuel consumption metre and a stopwatch was utilized to measure the fuel consumption at different operating conditions of the test engine. The exhaust gas emissions were measured at different engine speeds using the exhaust gas analyzer. The lubrication oil for the turbocharger was supplied through a pipe attached to the main oil channel of the engine. The specifications of the VW engine used for the experiment are shown in the table 3.1.

Table 3.1 Specifications of the VW engine

Engine specification	Details
Engine make	VW Golf 3 water-cooled
Bore x Stroke	79.5 x 95.5 mm
Aspiration	Turbo
Rated power	55 KW
Rated speed	4200 rpm
Compression ratio	22.5
Injection timing	336^0 CAD
Injection pressure	150 bar
Fuel type/system	Diesel/Bosch
Engine size/cylinders	1.896 cm^3/ 4 cylinders
Engine dynamometer	Alternator with water heaters
Cylinder number	4
Type	water cooled, four stroke diesel
Fuel type	Diesel
Charging	Turbocharging

3.3 OPERATING THE EXHAUST GAS ANALYSER

The instrument used to monitor the engine exhaust is called TECALEMIT GARAGE TG400 EXHAUST GAS ANALYSER (Fig.3.1) .

Fig.3.1. Exhaust gas analyser

The exhaust gas analyzer is an instrument used to measure the products of combustion of vehicles. There are various types of exhaust gas analyzer which can be used for various purposes. The TG400 analyzer is a 4Gas analyzer which is designed to measure carbon monoxide (CO), carbon dioxide (CO_2), unburned hydrocarbons (HC), Lambda and NOx simultaneously in the exhaust gases of vehicles. The 4gas analyzer has been used by most developed countries for similar purpose and has been found to be very accurate (Thomas, 2006). The analyzer is able to printout the results of the test for analysis on engine.

The analyser was operated in the following steps:

i. The analyzer was switched on, then it went through the preheating and calibration stage for some minutes. After that a leak test was performed, a fail leak test would not allow the test to continue. After the 'pass' leak test, a hydrocarbon residual test was done.

ii. The VW engine dip stick was removed and inserted into a probe into the engine; this allowed the analyzer to read the temperature of the engine.

iii. The researcher finally inserted another probe into the exhaust tail pipe as the engine run on idling speed

iv. The analyzer then displayed results of the constituents of the exhaust gas on the screen

v. Measurements were performed when the temperature was above 80 degrees, which is the engine's working temperature

vi. The results were printed for analysis.

The measurement range of accuracy of the analyzer are shown in table 3.2 below,

Table 3.2 Measurement Range of Exhaust Analyser

Exhaust Gas Analyser		
Exhaust gas	Measurement range	Accuracy
CO	0-10 % vol.	<.06 vol.%:±0.03 vol.% P0.6 vol.%:±5% of ind. val.
HC	0-20,000 ppm vol.	<200 ppm vol.:±10 ppm vol.
NO	0-5000 ppm vol.	P500 ppm vol.:±10% of ind. val.

3.4 PROPERTIES OF FUELS MEASURED

The properties of the biodiesel palm kernel oil methyl ester are shown in Table 3.3 compared with petrol diesel and other standards.

Table 3.3 Fuel Properties of PKOME and Petroleum Diesel Used (Milton, 2013)

Properties	PKOME	Petroleum Diesel	ASTM D6751	EN 14214

Kinematic Viscosity @ 40 ^{0}C (mm^2/s)	3.7	2.6	1.9-6	3.5-5
Cetane number	50	49	47min	51min
Pour point (^{0}C)	1	1	-15 to 6	-
Cloud point (^{0}C)	6	2	-	-
Flash point (^{0}C)	170	90	93min	120min
High calorific value (MJ/kg)	44	46	-	35
Acid value (mg KOH/g)	3.4	0.17	0.8 max	0.5 max
Density (kg/m^3)	894	839	880	860-900

3.5 MEASUREMENT OF ENGINE PERFORMANCE WHEN RUN ON PETROLEUM DIESEL FUEL

i. The VW diesel engine (Naturally aspirator) was run on petroleum diesel fuel for about 30 minutes. This was to allow the engine to reach its operating temperature.

ii. Measurements of engine parameters as well as CO and NO_X were then taken with the exhaust gas analyser at engine speeds of 1200, 1400, 1600, 1800, 2000, 2200 and 2400 rpm respectively.

iii. The engine was then fitted with a turbocharger with boost power of 0.7 bars iv. Steps 1 and 2 were repeated with the turbocharger installed. The CO and NOx measurements with turbocharger installed were then noted. The engine performance parameters were also recorded including BSFC, BTE, thermal efficiency.

3.6 MEASUREMENT OF ENGINE PERFORMANCE WHEN RUN ON BIODIESEL FUEL

1. The VW diesel engine (Naturally aspirator) was run on palm kennel oil methyl ester (PKOME) for about 30 minutes. This was to allow the engine to reach its operating temperature.
2. Measurement of engine parameters as well CO and NOx were then taken with the exhaust gas analyser at engine speeds of 1200, 1400, 1600, 1800, 2000, 2200 and 2400 rpm respectively.
3. The engine was then fitted with a turbocharger with boost power of 0.7 bars
4. Steps 1 and 2 were repeated with the turbocharger installed. Measurement of engine parameters as well CO and NOx were then taken with the exhaust gas analyser at engine speeds of 1200, 1400, 1600, 1800, 2000, 2200 and 2400 rpm respectively.

CHAPTER FOUR

4 EXPERIMENTAL RESULTS AND DISCUSSION

This chapter presents findings, results and discussions of the experimental work to achieve the specific objectives in Chapter one. In particular, the results of the measurements of the engine torque, brake power, brake specific fuel consumption (BSFC), brake thermal efficiency (BTE) and thermal efficiency as well as the CO and O_2 exhaust emissions were tabulated, graphs plotted and discussed.

4.1 PROPERTIES OF DIESEL FUEL AND BIODIESEL

The physical characteristics of the palm kennel oil biodiesel fuel can be compared with the characteristics of the experiment on jatropha biodiesel and petroleum diesel fuel conducted by Mohammed and Nemit-allah (2013) as shown in Table.4.1.

Table 4.1 Properties of Diesel Fuel and Biodiesel Mohammed and Nemit-allah (2013)

Property	Diesel fuel	Biodiesel (RME)
Chemical formula	$C_{14.09}$ $H_{24.78}$	C_{19} $H_{35..2}O_2$
Kinematic viscosity (mm^2/s) at 40 C	2.82	4.7
Specific gravity	0.85	0.88
Net heating value (MJ/kg)	42.64	37.23
Cetane number	42.6	51
Oxygen content (%)	–	10.9

4.2 RELATIONSHIP BETWEEN BRAKE POWER AND ENGINE SPEED FOR PETROLEUM AND BIODIESEL FUELED ENGINE

Table 4.2 shows the engine brake power at various engine speeds. In Figure 4.1, the variation in the brake power of the test engine operated with Natural Aspiration (NA) and turbocharger (TU) conditions are shown as a function of the engine speed for diesel fuel (DF) and biodiesel fuel (B). The brake power reached its peak value at the speed of about 2200 rpm for all fuels and engine operations. The brake power of the engine with diesel fuel is higher than that with biodiesel for both NA and TU operations. While operating Biodiesel with Natural Aspiration (B-NA), the mean reduction in the brake power is 4.8% compared to petroleum diesel fuel with Natural Aspiration (DF-NA) operation. Due to the fact that the lower heating value of biodiesel is about 12% lower than that of petroleum diesel fuel, both torque and brake power reduce. However, differences are very small in most cases. Fig. 4.1 also shows that the difference in the break power between petroleum diesel fuel and biodiesel fuel reduces with the turbo charger (TU) operation. The brake power produced with Biodiesel and turbocharger (B-TU) operation is on an average 3.25% lower than that of Diesel fuel with Turbocharger (DF-TU) operation due to better combustion resulting from increased air supply.

Table 4.2 Engine Speed and Brake Power

Engine speed (rpm)	Brake power (kw)			
	DF-NA	B-NA	DF-TU	B-TU
1000	25	24	30	28.7
1200	29.8	28	37	35
1400	34	33	42.5	40
1600	37.5	34	45	42.2
1800	40	37.5	47.5	45
2000	42.7	40	50	48

2400	46	42	53.2	52

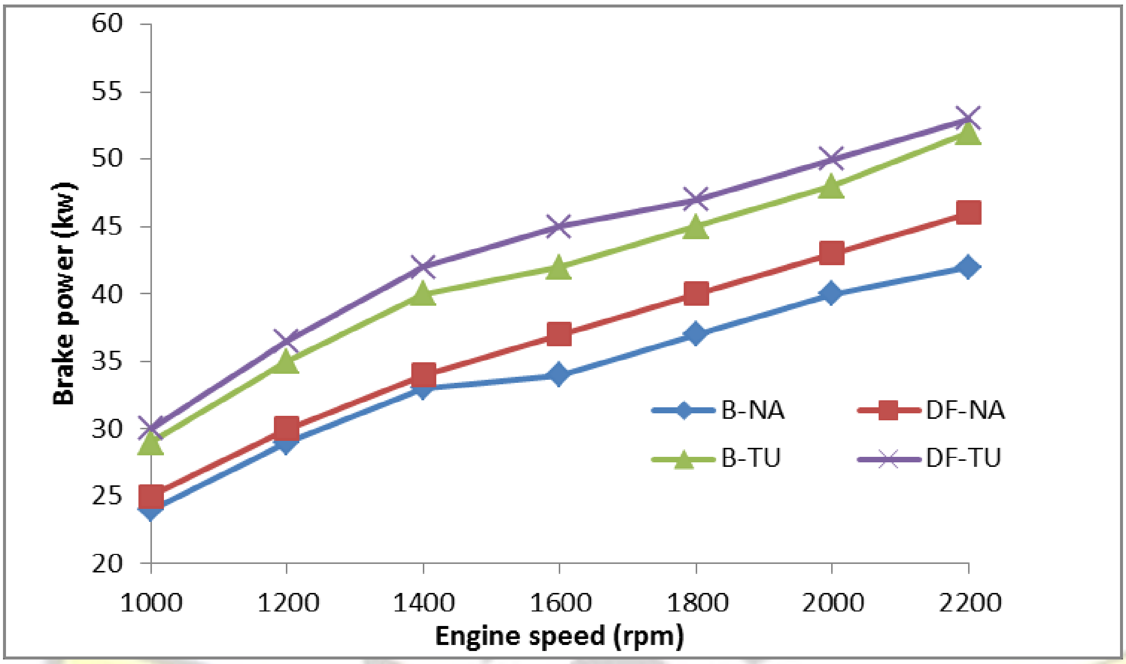


Fig.4.1 Brake Power versus Engine Speed for Fuels Tested At Full Load

4.3 RELATIONSHIP BETWEEN ENGINE SPEED AND TORQUE FOR PETROLEUM AND BIODIESEL FUELED ENGINE

Table 4.3 shows the engine speed and torque. Figure 4.2, shows the variation in the torque of the engine fuelled with petroleum diesel fuel and biodiesel fuel versus the engine speed. It is observed that the engine yields the maximum torque at about 1600 rpm.

Table 4.3 Engine Speed and Torque

Engine speed (rpm)	Torque (Nm)			
	B-NA	B-TU	DF-NA	DF-TU

1000	190	235.6	200	239.4
1200	195	240.7	210.4	247.3
1400	195	242.5	211.7	250.3
1600	190	237.5	212.2	245
1800	186.3	225.4	200	241
2000	180.1	221	190	232.4
2200	175	211.8	186.5	226

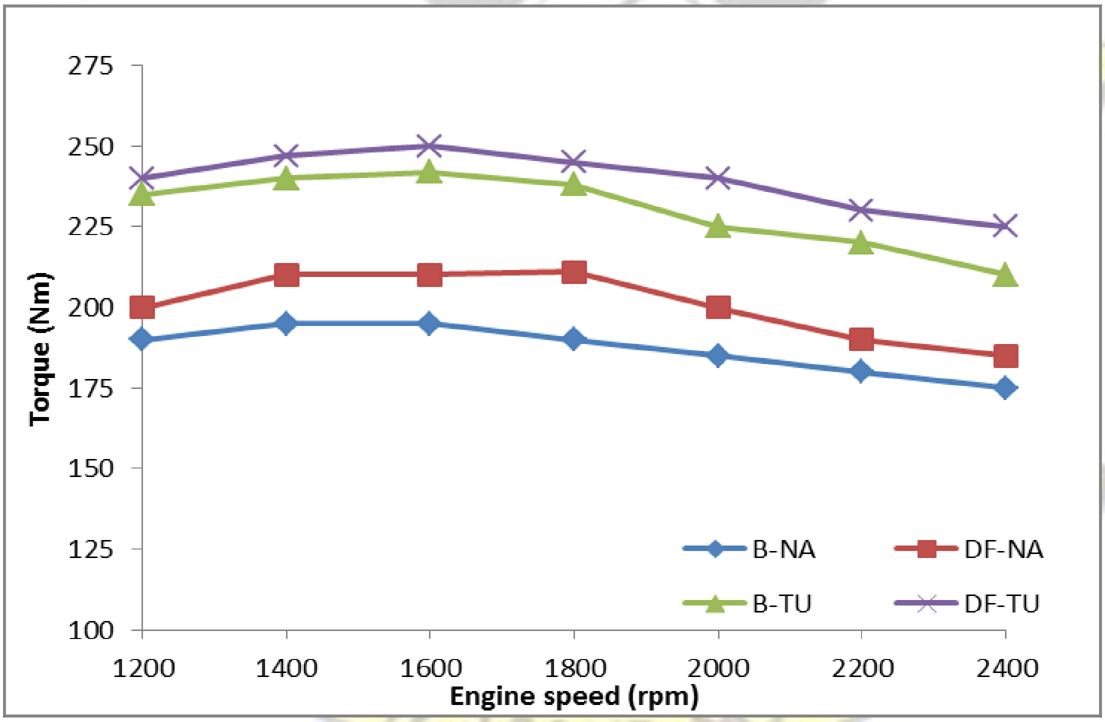


Fig.4.2 Torque versus Engine Speed For Fuels Tested At Full Load.

The reason for the reduction of torque with biodiesel can also be attributed to the lower heating value of the fuel. The mean increases in the torque with biodiesel and diesel fuel in TU operation are determined as 18.7% and 16.8%, respectively.

4.4 RELATIONSHIP BETWEEN THE BRAKE SPECIFIC FUEL CONSUMPTION (BSFC) AND ENGINE SPEED FOR PETROLEUM AND BIODIESEL FUELED ENGINE

Table 4.4 shows the engine BSFC at various engine speeds. In Figure 4.3, the variations in the BSFC of both petroleum diesel fuel and biodiesel fuel with respect to the engine speed are shown. The BSFC is the ratio of the fuel consumption to the brake power of the engine. The BSFC for the B-NA operation is on an average 11.5% higher than that for DF-NA operation. BSFC also increases with the use of biodiesel fuel. This increase may be attributed to the collective outcomes of the higher fuel density, higher fuel consumption and lower break power due to lower heating value of the biodiesel fuel.

Table 4.4 Engine Speed and BSFC

Engine speed (rpm)	BSFC (g/kWh)			
	B-NA	B-TU	DF-NA	DF-TU
1200	338	265	300	238
1400	324	262	295	230
1600	225	239	282	219
1800	213	243	260	225
2000	330	262	283	237
2200	332	270	291	251
2400	337	271	296	262

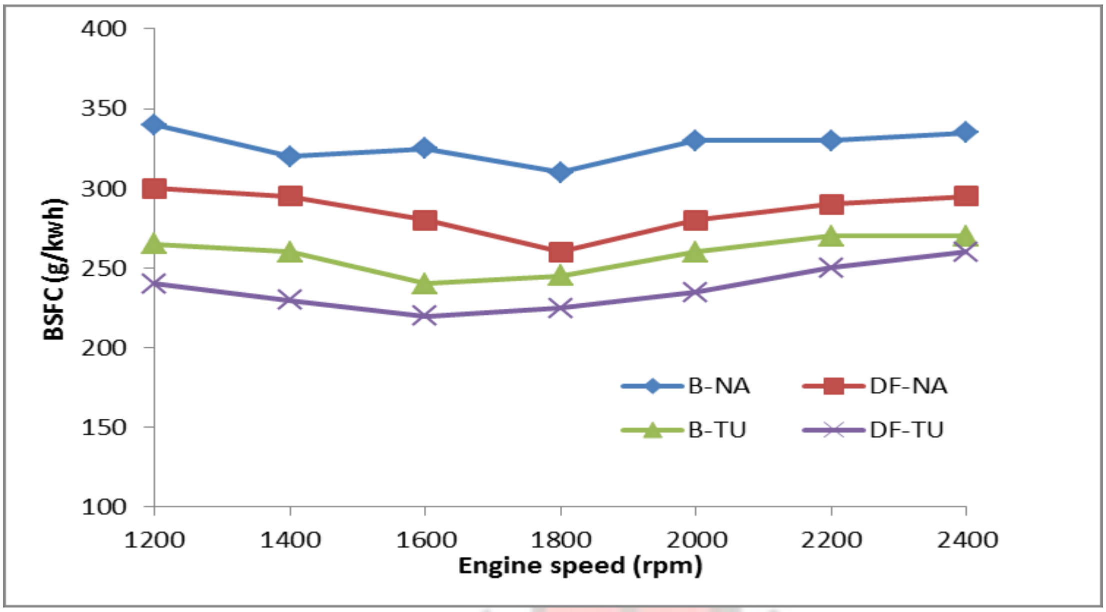


Fig.4.3 Brake Specific Fuel Consumption versus Engine Speed for Fuels Tested At Full Load

It is also seen from Fig.4.3 that the BSFC decreases for both diesel fuel and biodiesel fuel in the TU operation compared to NA operation. Compared to NA, the BSFC for the TU operation is averagely 15.7 and 17.7% lower for diesel fuel and biodiesel, respectively. The BSFC for the B-TU operation is averagely 8.9% higher than that for the DF-TU operation. This reduction is mainly caused by the improvement in fuel atomization, air–fuel mixing and combustion characteristics of the fuel due to the high air temperature and increased air charge in the cylinder of the diesel engine in TU operation. The collective factors of lowering fuel consumption and increasing brake power cause an improvement in BSFC with the application of turbocharger. It was also reported that there is a reduction of about 15% in the BSFC of a diesel engine fuelled with untreated cotton seed oil which is a similar biodiesel fuel with a supercharger at 0.4 bar compared to natural aspiration (NA) condition.

4.5 RELATIONSHIP BETWEEN THE BRAKE THERMAL EFFICIENCY (BTEs) AND ENGINE SPEED FOR PETROLEUM AND BIODIESEL FUELED ENGINE

Table 4.5 shows the brake thermal efficiency and various engine speeds. BTEs for diesel fuel and biodiesel as a function of engine speed are shown in Fig.4.4. The maximum BTE values are observed in the ranges of 1600–2000 rpm for the NA and TU operations. It is seen that petroleum diesel fuel has higher BTEs than biodiesel fuel for all cases of turbocharging and natural aspiration. However, the mean difference in the BTEs between B-TU and DF-TU operations is about 2.6%.

Table 4.5 Engine Speed and Thermal Efficiency

Engine speed (rpm)	Thermal efficiency (%)			
	B-NA	B-TU	DF-NA	DF-TU
1200	0.28	0.337	0.29	0.361
1400	0.275	0.349	0.298	0.362
1600	0.28	0.366	0.292	0.379
1800	0.288	0.36	0.3	0.37
2000	0.27	0.344	0.28	0.361
2200	0.275	0.338	0.275	0.351
2400	0.26	0.332	0.265	0.342

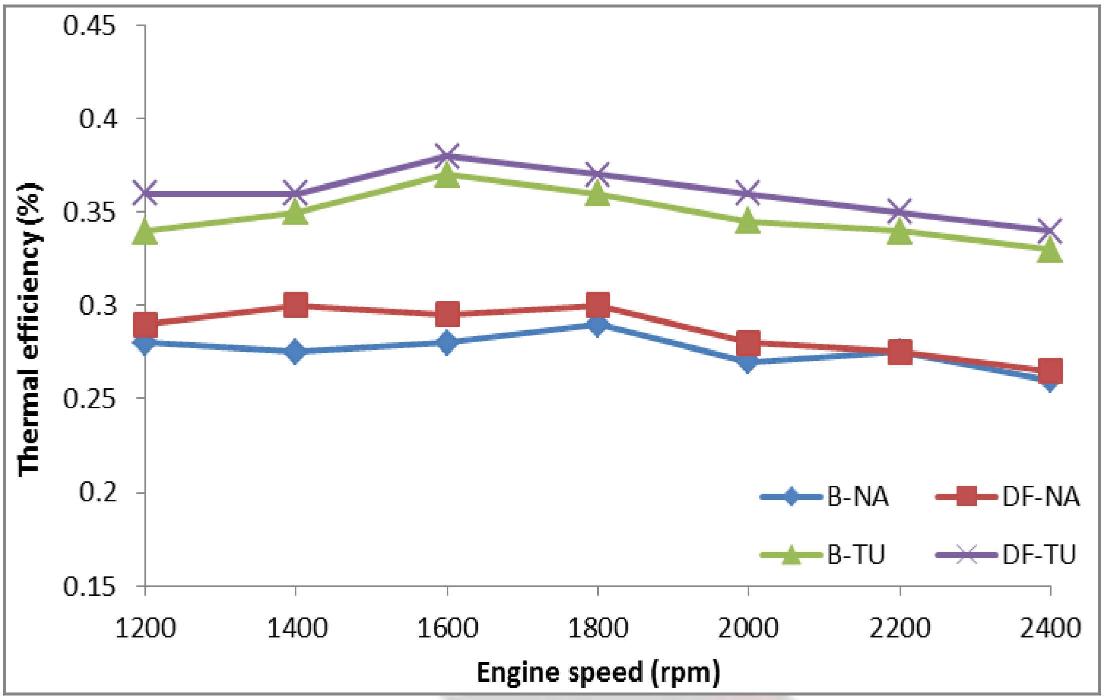


Fig.4.4 Thermal Efficiency versus Engine Speed For Fuels Tested At Full Load.

4.6 RELATIONSHIP BETWEEN THE CARBON MONOXIDE EMISSIONS AND ENGINE SPEED FOR PETROLEUM AND BIODIESEL FUELED ENGINE

Table 4.6 shows the values of carbon monoxide emissions at various engine speeds. The variations in the CO emission with the diesel fuel and biodiesel in NA and TU operations as a function of the engine speed are shown in Fig.4.5. CO emissions of B-NA operation are averagely 17% lower than those of DF-NA operation. CO emission from diesel engine is related to the fuel properties as well as combustion characteristics. It is well known that better fuel combustion usually results in lower CO emission. Biodiesel has higher cetane number compared to diesel fuel, which causes lower ignition delay period and auto ignition capability.

Table 4.6 Carbon monoxide emissions and engine speeds

Engine speed	CO (ppm)			
(rpm)	B-NA	DF-NA	B-TU	DF-TU
1200	2400	2700	1500	1800
1400	1700	1900	1000	1100
1600	1200	1500	950	900

1800	1000	1200	400	600
2000	500	700	300	400
2200	450	600	200	350
2400	400	500	300	350

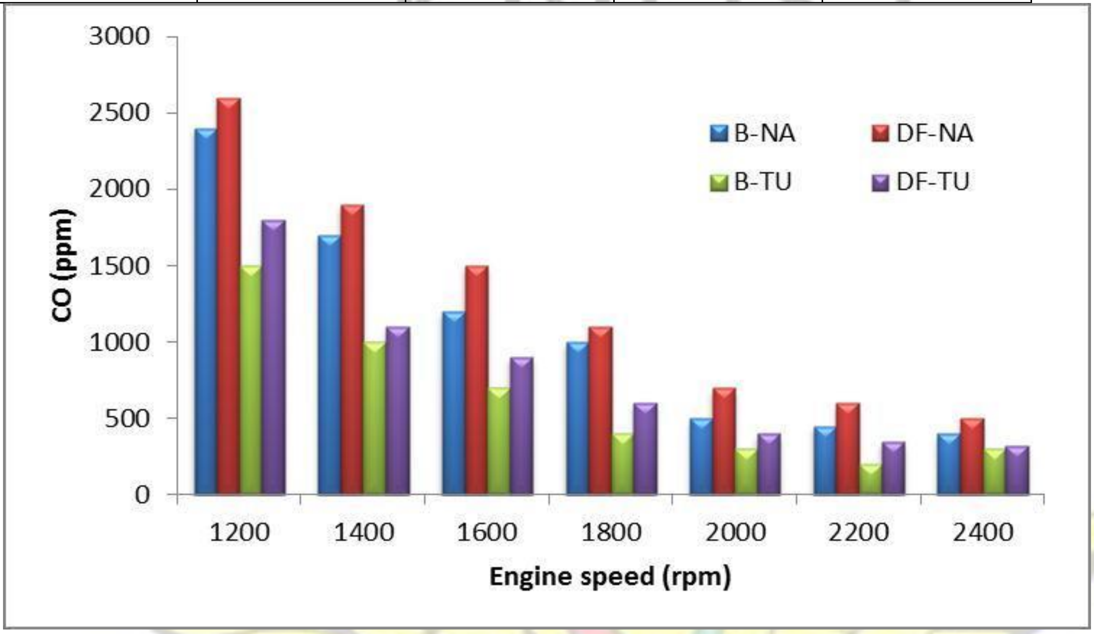


Fig.4.5 Carbon Monoxide Emissions versus Engine Speed for Fuels Tested At Full Load

High oxygen content of biodiesel associated with lower ignition delay period provides an important reduction in the CO emission by improving combustion. Moreover, the carbon/hydrogen (C/H) ratio of biodiesel is slightly lower than that of diesel fuel, which yields diminished CO emissions with the use of biodiesel. It was experimentally determined that the TU operation causes a noticeable reduction in CO emission. Compared to NA operation, in the TU operation the CO emissions for diesel fuel and biodiesel are on an average 47% and 52% lower, respectively. CO emissions in the B-TU operation are averagely 26% lower than those in the DF-TU operation. The application of turbocharger provides increased air to the diesel engine and enables mixing of fuel-air easily in the combustion chamber, thereby causing better combustion and lower CO emission values.

4.7 RELATIONSHIP BETWEEN THE NITROGEN OXIDES EMISSIONS AND ENGINE SPEED FOR PETROLEUM AND BIODIESEL FUELED ENGINE

Table 4.7 shows the NO_X emissions at various engine speeds. In Figure 4.6, variations in the NOx emissions plotted against engine speed for diesel fuel and biodiesel fuel. It is known that formation of NOx emissions are strongly dependent upon the equivalence ratio, oxygen concentration and burned gas temperature. The oxygen content of biodiesel is the main reason for higher NOx emissions. The oxygen in the biodiesel can react easily with nitrogen during the of combustion process, thus causing higher emissions of NOx. The NOx emission with biodiesel is higher than that with diesel fuel in both NA and TU operations. In B-NA operation, an average of 10% increase in the NOx emission was measured compared to DF-NA operation. A noticeable increase in the NOx emissions was observed with the use of diesel fuel and biodiesel in TU operation. Application of turbocharger provides more air to the engine and causes a higher combustion temperature which yields an increase in the formation of NOx emission. It was determined that in the TU operation, the NOx emissions with diesel fuel and biodiesel are higher on an average of 27% and 21%, respectively, compared to NA operation.

Table 4.7 NO_X and Engine Speed

Engine speed (rpm)	NO_X (ppm)			
	B-NA	DF-NA	B-TU	DF-TU

1200	1300	1150	1450	1300
1400	1200	1100	1400	1350
1600	1200	1000	1350	1300
1800	1100	1050	1300	1250
2000	1000	1000	1250	1250
2200	1000	1150	1250	1200
2400	950	1100	1250	1150

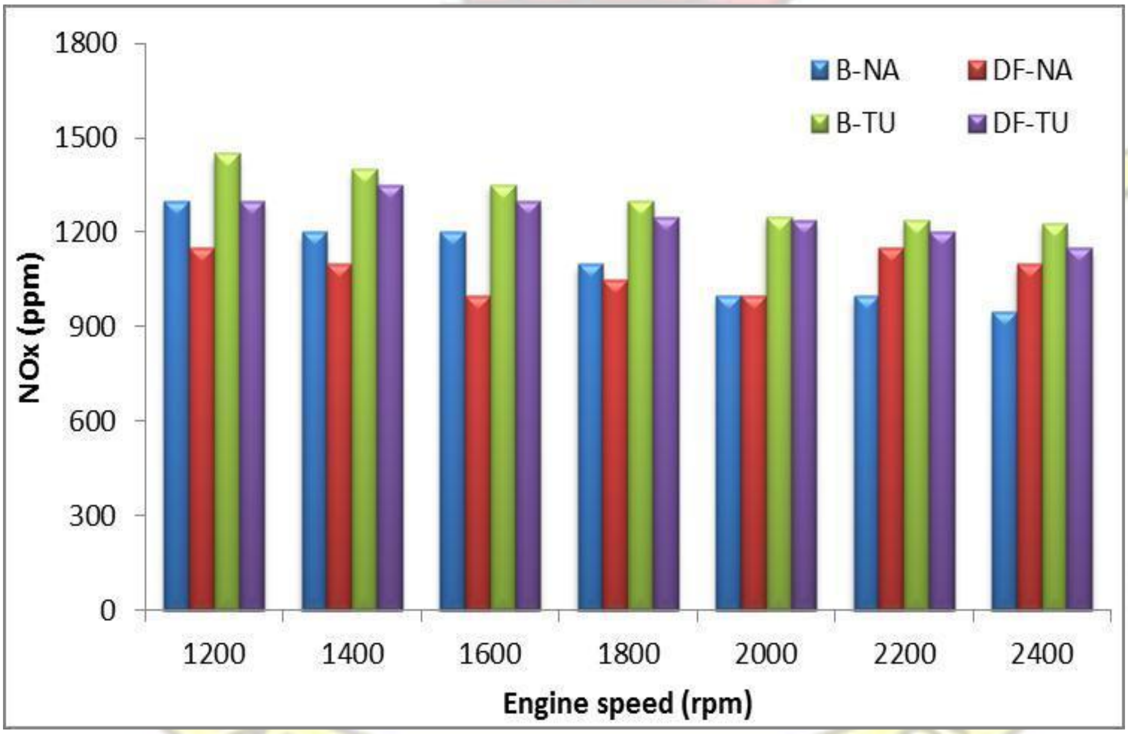


Fig.4.6. Nitrogen Oxides Emissions versus Engine Speed For Fuels Tested At Full Load.

4.8 EFFECT OF TURBO WITH ENGINE PERFORMANCE

The engine performance was increased when run on biofuel with turbocharger connected.

Table 4.8 Percentage increase in Engine torque for B-TU and B-NA

Engine speed	Torque (Nm)			% INCREASE FOR BIOFUEL	% INCREASE FOR B-TU AND DF-NA
(rpm)	B-TU	B-NA	DF-NA		
1200	235.6	190	200	0.2400	0.178
1400	240.7	195	210.4	0.2344	0.1440
1600	242.5	195	211.7	0.2436	0.1455
1800	237.5	190	212.2	0.2500	0.1192
2000	225.4	186.3	200	0.2099	0.1270
2200	221	180.1	190	0.2271	0.1632
2400	211.8	175	186.5	0.2103	0.1357

Table 4.9 Percentage increase in Brake Power for B-TU and B-NA

Engine Speed	Brake Power (Nm)			% INCREASE FOR BIOFUEL	% INCREASE FOR B-TU AND DF-NA
(rpm)	B-TU	B-NA	DF-NA		
1000	28.7	24	25	0.1958	0.1480
1200	35	28	29.8	0.2500	0.1745
1400	40	33	34	0.2121	0.1765
1600	42.2	34	37.5	0.2412	0.1253
1800	45	37.5	40	0.2000	0.1250
2000	48	40	42.7	0.2000	0.1241

2200	52	42	46	0.2381	0.1304

Table 4.10 Percentage increase in Thermal Efficiency for B-TU and B-NA

Engine speed	Thermal Efficiency (%)			% INCREASE FOR BIOFUEL	% INCREASE FOR B-TU AND DF-NA
(rpm)	B-TU	B-NA	DF-NA		
1200	0.337	0.28	0.29	0.2036	0.1621
1400	0.349	0.275	0.298	0.2691	0.1711
1600	0.366	0.28	0.292	0.3071	0.2534
1800	0.36	0.288	0.3	0.2500	0.2000
2000	0.344	0.27	0.28	0.2741	0.2286
2200	0.338	0.275	0.275	0.2291	0.2291
2400	0.332	0.26	0.265	0.2769	0.2528

Table 4.8, 4.9 and 4.10 shows the improvement of engine torque, brake power and thermal efficiency of the biofuel engine run with turbocharger. The values of DFNA was also compared to B-TU. From the results, there is an increase in engine performance indicators (i.e. torque, brake power, thermal efficiency) when the engine was run on B-TU compared to DF-NA.

Table 4.11 Percentage reduction in CO emissions for B-TU and B-NA

Engine Speed	Carbon monoxide emissions (CO)	% Reduction for Biofuel	% Reduction for B-TU and DF-NA

(rpm)	DF-NA	BF-NA	BF-TU		
1200	2700	2400	1500	0.3750	0.4444
1400	1900	1700	1000	0.4118	0.4737
1600	1500	1200	900	0.2500	0.4000
1800	1200	1000	400	0.6000	0.6667
2000	700	500	300	0.4000	0.5714
2200	600	450	200	0.5556	0.6667
2400	500	400	300	0.2500	0.4000

From table 4.11, CO emissions of the engine running on biofuel reduced when connected to turbocharger compared to natural aspiration. For B-TU and DF-NA emissions, there was a percentage decrease in CO emissions when engine was run on biofuel with turbocharger.

Table 4.12 Percentage reduction in NO emissions for B-TU and B-NA

Engine speed	Nitrogen Oxide emissions (NOx)			% Increase for Biofuel	% Increase for B-TU and DF-NA
(rpm)	BF-TU	BF-NA	DF-NA		
1200	1450	1300	1150	0.1154	0.2609
1400	1400	1200	1100	0.1667	0.2727
1600	1350	1200	1000	0.1250	0.3500
1800	1300	1100	1050	0.1818	0.2381
2000	1250	1000	1000	0.2500	0.2500
2200	1250	1000	1150	0.2500	0.0870

2400	1250	950	1100	0.3158	0.1364

From table 4.12, NOx emissions of the engine running on biofuel increased when connected to turbocharger compared to natural aspiration.

CHAPTER FIVE

5 CONCLUSION AND RECOMMENDATIONS

The conclusion and recommendations from this study and experiments are discussed in this chapter.

5.1 CONCLUSION

From the results and analysis, the following conclusions were made:

1. The brake power increases with increasing engine speed for Biofuel Turbocharger (B-TU), Biofuel Natural Aspiration(B-NA), Diesel Fuel Turbocharger (DF-TU), Diesel Fuel Natural Aspiration(DF-NA). Diesel Fuel Turbocharger (DF-TU) recorded highest brake power at 52.3 kW, compared to others. The percentage increase in brake power for the engine running on biofuel and diesel fuel with turbocharger compared with Natural Aspiration is 17.98% and 16.67% respectively.

2. Diesel Fuel Turbocharger (DF-TU) recorded highest torque at all engine speeds compared to the others. The highest value, 250.3 Nm is recorded at optimum speed of 1600 rpm. The percentage increase in torque for the engine running on biofuel and diesel fuel with turbocharger compared with Natural Aspiration 18.63% and 16.08% respectively.

3. Diesel Fuel Turbocharger (DF-TU) recorded higher values of thermal efficiency at all the engine speeds compared to Biofuel Turbocharger (BF-TU). Diesel Fuel Turbocharger (DF-TU) recorded highest value of 38% compared to 36% of

Biofuel Turbocharger (BF-TU) at optimum speed of 1600 rpm. The percentage increase in thermal efficiency for the engine running on biofuel and diesel fuel with turbocharger compared with Natural Aspiration 20.49% and 20.83% respectively.

4. Biofuel Turbocharger (B-TU) recorded the highest emission of NOx (Nitrogen Oxide) at all engine speeds.

5. Biofuel Turbocharger (B-TU) compared to Biofuel Natural Aspiration (B-NA), Diesel Fuel Turbocharger (DF-TU) and Diesel Fuel Natural Aspiration (DF-NA) recorded least value of carbon Monoxide emission at all engine speeds. CO emission reduces at increasing engine speeds.

5.2 RECOMMENDATIONS;

1) Nitrogen Oxide (NOx) reduction strategies should be considered, since biodiesel usage increases Nitrogen Oxide emission.
2) Turbocharger production should be encouraged as the country adopts policy on biofuel use in diesel engine.

REFERENCES

Alcantara, R, Amores, J, Canoira. L, Fidalgo, E, Franco, MJ and Navarro, A 2000, „Catalytic production of biodiesel from soybean oil used frying oil and tallow‟, *Biomass Bioenergy*, 18, pp. 515–27.

Al-Widyan, MI, Tashtoush, G and Abu-Qudais, M 2002, „Utilization of ethyl ester of waste vegetable oils as fuel in diesel engines‟, *Fuel Process Technology,* 76: pp. 91–103.

Aydin, H, Bayindir, H 2010, „Performance and emission analysis of cottonseed oil methyl esterin a diesel engine‟, *Renewable Energy*, 35(3): pp.588-592.

Borat, O, Balci, M and Surmen, A 2000, *internal combustion engines*, Gazi University Publishing, Turkish.

Buyukkaya, E 2010, „Effects of biodiesel on a DI diesel engine performance, emission and combustion characteristics‟, *Fuel 2010*; 89: pp. 3099–105.

Carraretto, C, Macor, A, Mirandola, A, Stoppato, A and Tonon, S 2001, „Biodiesel as alternative fuel: experimental analysis and energetic evaluations‟, *Energy 2004* ;29: pp. 2195–211

Chiu, CW, Schumacher, LG and Suppes, GJ 2004, „Impact of cold flow improvers on soybean biodiesel blend‟, *Biomass Bioenergy,* 27, pp. 485-91.

Cvengros, J , Povazanec, F 1996, „Production and treatment of rapeseed oil methyl esters as alternative fuels for diesel engines‟, *Bio resource Technology*, 55, pp. 145–52.

Dwivedi, G, Sharma, MP and Jain, S 2013, „Diesel engine performance and emission analysis using biodiesel from various oil sources – Review‟, *JMES,* 4 (4), pp. 434-447

Ghobadian, B, Rahimi, H, Nikbakht, AM, Najafi, G, Yusaf, TF 2009, „Diesel engine performance and exhaust emission analysis using waste cooking biodiesel fuel with an artificial neural network‟, *Renewable Energy* ;34: pp. 976–82.

Godiganur, S, Murthy, CH and Reddy, RP 2010, „Performance and emission characteristics of a Kirloskar HA394 diesel engine operated on fish oil methyl esters‟, *Renewable Energy* ;35: pp. 355–9.

Heywood, JB 1984, *internal combustion engines,* Mc-Graw Hill, USA. http://www.kadekticaret.com/whatisaturbocharger.htm [Accessed, 15/1/2014])

Jain, S, Sharma, MP 2009, „Prospects of Biodiesel from Jatropha in India: A Review‟, *Renewable Sustainable Energy Reviews*, 14(2), pp. 763-771.

Knothe, G 2008, „Optimizing fatty ester composition to improve fuel properties‟, *Energy Fuel*, 22, pp. 1358–1364.

Knothe, G, Sharp, CA and Ryan, TW 2006, „Exhaust emissions of biodiesel, petro diesel, neat methyl esters, and alkanes in a new technology engine‟, *Energy Fuel*, 20, pp. 403–408.

Kulkarniand, MG, Dalai, AK 2006, „Waste cooking oil an economical source for biodiesel: a review‟, *Ind. Eng. Chem. Res*, 45, pp. 2901–13.

Lin, BF, Huang, JH, Huang, DY 2009, „Experimental study of the effects of vegetable oil methyl ester on DI diesel engine performance characteristics and pollutant emissions‟, *Fuel 2009*;88: pp. 1779–85.

Luján, JM, Bermúdez, V, Tormos, B, Pla, B 2009, „Comparative analysis of a DI diesel engine fuelled with biodiesel blends during the European MVEG-A cycle: Performance and emissions (II)‟, *Biomass Bioenergy*; 33: pp. 948–56.

Ma, F, Hanna, MA 1999, „Biodiesel production: a review‟, *Bio resource Technology,* l70, pp. 1–15.

Mahanta, P, Mishra, SC and Kushwah, YS 2006, „An experimental study of Pongamia pinnata L. oil as a diesel substitute‟, *P I Mech Eng A-J Pw*; 220: pp. 803–8.

Mc, G 2004, „Diesel engines have a future and that future is clean‟, *SAE TechPaper*, 2004,01-1956

Nwafor, OM, Rice, G and Ogbonna, AI 2000, „Effect of advanced injection timing on the performance of rapeseed oil in diesel engines‟, *Renewable Energy,* 21, pp. 433–44.

Oliveira, ED, Quirino, RL, Suarez PA and Prado AG 2006, „Heats of combustion of biofuels obtained by pyrolysis and by trans esterification and of biofuel/diesel blends‟, *Thermochim Acta*, 450, pp. 87–90.

Openshaw, K 2000, „A review of Jatrophacurcas: an oil plant of unfulfilled promise‟, *Biomass and Bioenergy,* 19, pp. 1–15.

Peterson, CL, Reece, DL, Hammond, R, and Thompson, JA 2002, „Comparison of ethyl and methyl esters of vegetable oils as diesel fuel substitute‟, *Proceeding Alternate Energy Conference* ASAE, 1992, pp. 99-110. pubs.cas.psu.edu/ FreePubs/pdfs/uc204 diesel engine.pdf, [accessed 15/1/2014]

Qi, DH, Geng, LM, Chen, H, Bian, YZ, Liu, J and Ren, XC 2009, „Combustion and performance evaluation of a diesel engine fueled with biodiesel produced from soybean crude oil", *Renewable Energy* ;34: pp. 2706–13.

Raheman, H, Phadatare, AG 20004, „Diesel engine emissions and performance from blends of karanja methyl ester and diesel", *Biomass Bioenergy*; 27: 393–7.

Ramadhas, AS, Muraleed, C, Jayaraj, S 2005, „Performance and emission evaluation of a diesel engine fueled with methyl esters of rubber seed oil", *Renewable Energy*; 30: pp. 1789–800.

Reyes, JF, Sepúlveda, MA 2006, „PM-10 emissions and power of a diesel engine fuelled with crude and refined biodiesel from salmon oil", *Fuel 2006*; 85: pp. 1714–9.

Sayers, T. 1990 As the impeller rotates, the fluid (air) that is drawn into the blade passages at the impeller inlet is accelerated as it is forced radially outwards.

Sahoo, PK, Das, LM, Babua, MK, Arora, P, Singh, VP, Kumar, NR 2009„„Comparative evaluation of performance and emission characteristics of jatropha, karanja and polanga based biodiesel as fuel in a tractor engine", *citeseerx.ist.psu.edu* :88; pp. 1698–707.

Song, JT, Zhang, CH 2008, „An experimental study on the performance and exhaust emissions of a diesel engine fuelled with soybean oil methyl ester", *P I Mech Eng D-J Aut*; 222: pp. 2487–96.

Sun, J.F., Caton, J.A. and Jacobs, T.J. 2010, „Oxides of nitrogen emissions from biodiesel-fuelled diesel engines", *Prog Energy Combust Sci*, 36, pp. 677–

95

Usta, N, Öztürk, E, Can, Ö, Conkur, ES, Nas, SC 2005, „Combustion of biodiesel fuel produced from hazelnut soap stock/waste sunflower oil mixture in a diesel engine", *Energy Conversion Management 2005;* 46: pp. 741–55.

Utlu, Z, Koc¸ MS 2008, „The effect of biodiesel fuel obtained from waste frying oil on direct injection diesel engine performance and exhaust emissions", *Renewable Energy 2008;* 33: pp. 1936–41

Verma, a, Pal, a, Kachhwaha, SS and Maji, S 2010, „Biodiesel production through hydrodynamic cavitation and performance testing", *Renewable Energy*; 35: pp. 619–24.

Zhang, Y, Dube, MA, McLean, DD and Kates, M 2003, „Biodiesel production from waste cooking oil: Process design and technological assessment", *Bio resource Technology*, 89, pp. 1–16.

APPENDIX A

The processes involved in the lab preparation of the biodiesel from vegetable are shown below (Appendix: Figure A1-A10)

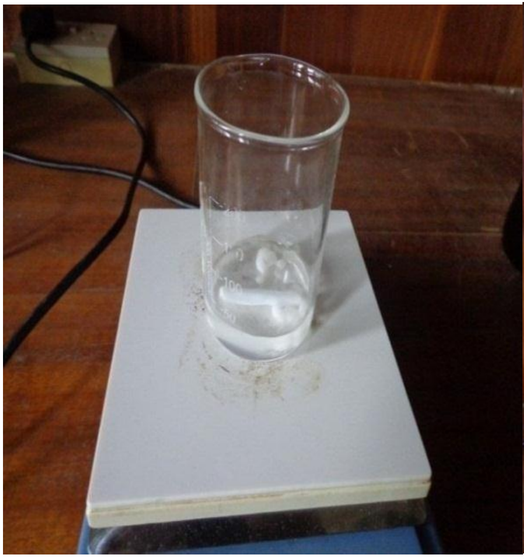
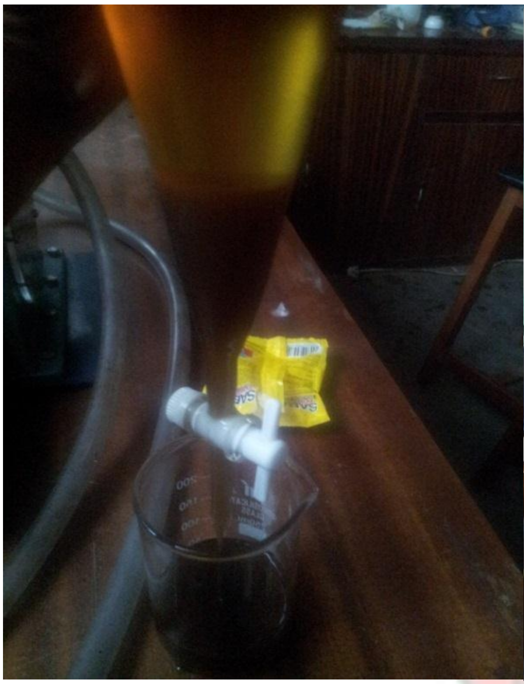


Figure A1 Figure A2 (a) Stirring of Methanol and sodium hydroxide (b) Adding the solution to the oil

Figure A3 Figure A4

(a) Solution might cake after an hour (b) Sulphuric acid is added at 35°C.



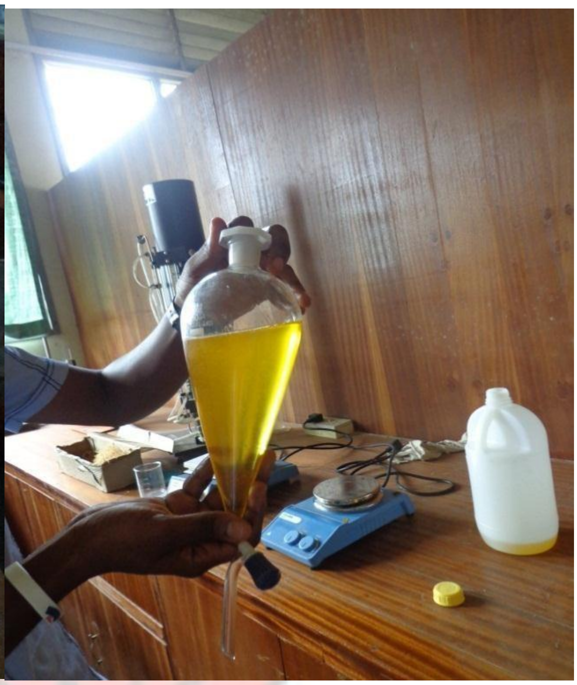


Figure A5 Figure A6 (a) Separation stage (b). Crude biodiesel before washing

Figure A7 Figure A8
(a) Washing biodiesel with warm water. (b) Separating water from biodiese

Figure A9 Figure A10 (a) Heating to remove traces of moisture (b) Refined biodiesel