Sudan University of Science and Technology

College of Graduate Studies

Effect of Delay Period on Performance of Compression Engine Running on Jatropha Fuel

تأثير فترة تأخر الحرق على أداء محرك إشعال ضغطي يعمل بوقود الجاتروفا

Thesis Submitted to the College of Graduate Studies in partial fulfillment of the requirements for the Degree of Doctor of Philosophy in Mechanical Engineering

By:

Sulaiman Muhammed Dawood

Supervisor:

Dr. Tag-Elssir Hassan Hussan Ali

March, 2019

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بسم الله الرحمن الرحيم

جامعة السودان للعلوم والتكنولوجيا

كلية الدراسات العليا

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كلية الدراسات العليا

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بِسْمِ اللَّهِ الرَّحْمَٰنِ الرَّحِيمِ

ٱلَّذِى جَعَلَ لَكُم مِّنَ ٱلشَّجَرِ ٱلْأَخْضَرِ نَارًا فَإِذَآ أَنتُم مِّنْهُ تُوقِدُونَ ﴿٨٠﴾

صدق الله العظيم

سورة يـــــس آية (80)

Dedication

This work is dedicated to our families and lecturers, who gave me the desire and passion for learning as well as the vision to see and believe in the possibilities in life.

They gave me the courage to challenge conventional thinking and create pathways that were considered to be impossible. These timeless intangibles are, in essence, the critical reasons for my motivated and assiduous effort in pursuing this scholar work.

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My sincere thanks go out to all the academic and administrative staff of the college. I also would like to acknowledge the joint venture funding of my institutes for giving me this valuable chance of study.

My deepest gratitude goes forth to my strong and supportive friends, everywhere. Finally, I would like to say thanks once again to all our families and friends for their standing beside me patiently with no complain, whether knowingly or not, their undying support in finishing the project and to the staff of mechanical engineering and heat engines lab technicians of of Karary University who gave support on practical work.

Abstract

Jatropha Biodiesel was tested in a single cylinder direct-injection, water cooled diesel engine to investigate the operational parameters of a small capacity diesel engine under five engine loads, constant speed test. The jatropha oil is used as a non edible oil to produce the biodiesel. The investigated blends were B0, B25, B50, B75 and B100, where (B#) denotes bio-diesel fuel volume percentage in diesel oil. The jatropha biodiesel was prepared locally in Sudan, specifications of which are given with reference to Petroleum Laboratories, Research and Studies Centre, Sudan. Each blend was tested on a short term basis and the results linked to ignition delay effect, a good firing was attained with the engine at various blends of jatropha bio-diesel blends in diesel oil fuel up to Bio-100 (pure bio-diesel). Concerning ignition delay of the engine, which is a measure of ignition quality, it is less with low percentages of bio-fuel blends than that of high percentages. Finally it has been shown that blend (Bio75) was selected as a good blend since it has less mechanical friction loss, stability in exhaust heat loss, relatively good mechanical efficiency and, high stability in cooling engine heat loss.

مستخلص

تم اختبار أداء محرك احتراق داخلي من نوع الاشعال الضغطي (ديزل) أحادي الاسطوانة، ذو حقن مباشر للوقود وتبريد مائي لدراسة مؤشرات الاداء لمحرك صغير تحت التشغيل بسرعة ثابتة مع تغيير الاحمال. المحرك تحت الاختبار يزود بوقود حيوي من مستخلص زيت نبتة الجاتروفا في مزيج من وقود الديزل بنسب مختلفة رمز لها بـ (B0) لوقود الديزل الصافي و(B25) باضافة نسبة 25% وقود حيوي و (B50) نسبة 50% و (B75) بنسبة 75% واخيرا (B100) وقود حيوي كامل. الوقود الحيوي المستخدم محضر محليا في السودان وعولج كيميائيا ليتحول من زيت نباتي عادي إلى وقود حيوي مواصفاته كما في شهادة الاختبار من معامل ابحاث البترول. كل عينة من عينات مزيج الوقود اخضعت للاختبار حيث تم حساب مختلف معاملات الاداء وقرنت نتائج اداء المحرك بعامل مهم من معاملات الاداء هي ما يعرف بـ(زمن تأخرالحريق). تلاحظ من نتائج الدراسة احتراق جيد للوقود في مختلف العينات بما في ذلك وقود B100 (وقود حيوي كامل). بالنسبة لعامل تأخر الحريق كقياس لجودة الحريق في المحركات لوحظ ان المزيج المحتوي علي نسب أقل من الوقود الحيوي أعطى زمن تأخر أقل من تلك المحتوية على نسب أعلى. توصل البحث إلى اختيار المزيج B75 كمزيج جيد حيث تقل الفقودات الاحتكاكية مع استقرار الفقودات الحرارية في العادم وفقودات الحرارة إلى ماء التبريد بالاضافة إلى زيادة الكفاءة الميكانيكية مقارنة بنسب الخلط الاخرى.

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Symbols and Abbreviations (Nomenclatures)

Symbol / Abbreviation	Meaning	Units
A/F, FA	Air-fuel ratio	kg_f/kg_a
ASTM	American Society of Testing and Materials	
ASME	American Society of Mechanical Engineers	
BDC	Bottom-Dead-Centre	
CA	Crank angle	
CI	Compression Ignition	
CN	Cetane number	
CNG	Compressed Natural Gas	
C_p	Specific heat at constant pressure	kJ/kgK
DI	Direct injection	
GCV	Gross Calorific Value	
HC	Hydrocarbon	
HCV	Higher calorific value	kJ/kg
ID	Ignition delay	
JBD	Jatropha Bio-Diesel	
k, γ	Pressure ratio	
LCV	Lower calorific value	[kJ/kg
LDO	Light Diesel Oil	
LP	Liquefied Petroleum	
LPG	Liquefied Petroleum Gas	

LSHS	Low Sulpher Heavy Stock
NCV	Net Calorific Value
NO_x	Nitrogen oxides
ON	Octane number
ϕ	Equivalence ratio = $(A/F)_{th}/(A/F)_{act}$
P_{cyl}	Cylinder pressure bar
PM	Particulate matters
ppm	parts per million
S	Entropy
SI	Spark Ignition
SOI	Start of ignition
TDC	Top-Dead-Centre
TVO	Tractor Veporising Oil
u,v,w	Velocity vectors in (x,y,z) directions
UNC	Unburned hydrocarbon
β	Cut-off ratio
η	Thermal efficiency [%]
η	Efficiency
θ	Crank angle measured from TDC [0]
τ	Delay period ms

CHAPTER I

INTRPDUCTION

CHAPTER I

INTRPDUCTION

1.1 Introduction:

The importance of undertaking researches concerning bio-fuels; is for its importance as an alternative fuel for the known traditional fossil petroleum fuels.

Diesel engines as one of the most important prime movers are responsible of burning a lot of amount of the Diesel Oil consumed, and has a big share of the total pollutant gases generated. Diesel engines are of compression ignition (CI) type, where ignition of the fuel air mixture is taking place by the raising up of temperature of the compressed air when the fuel is injected into and are referred to as auto ignition engines.

Any research will have its importance if it help in controlling the pollution effect of using this kind of machines. The proposed research is to study combustion in a compression ignition engine when it runs on bio-fuel to enhance its efficiency and to improve its performance.

Delay period (DP) which is defined as the time taken (or crank angle) from the fuel injection starts to the onset of combustion (Saha). Efficiency of a CI engine running on blends of jatropha bio-diesel with diesel oil fuel is to be studied.

1.2. Research Objective

The research is to investigate experimentally the specified mixture fuel and to study delay period (DP) phenomena, to test a compression ignition (CI) engine performance under different blends of jatroph bio-diesel in diesel oil fuel and to link performance criteria with ignition delay characteristics under running conditions.

1.3. Scope and Limits of the research:

A lot of researches even here in Sudan were done towards the matter of Bio-Diesel (Ministry of Science, 2014), the important of using the bio-diesel make it necessary to go deep in the field have our own matter of research and invention towards its better performance.

1.4. Problem statement:

One of the important performance parameters of an engine burning bio-fuel, is the ignition delay, and its effect on engine performance as a part of research not well covered in the proposed way.

The type of bio-fuel on this proposed research is that of Jatropha oil extracted from the seeds of the plant, the blend of this oil in diesel forms a kind of bio-fuel to be used in ordinary diesel engine.

A four stroke diesel engine of direct injection type is tested for the research. Though Jatropha plant, has all ready been planted and oil from the seeds of the plant in now been extracted in good amounts for research purpose. Enough amount of a treated jatropha oil converted bio-diesel fuel is used for the taken tests as pure (B100) or in blends with natural diesel oil fuel.

1.5- Questions to be answered:

1.5.1 How does the blend percentage affect the delay period?

1.5.2 Is there any limit of blend percentage concerning the effective delay period?

1.5.3 How other parameters affect bio-diesel fuel?

1.5.4 What control methods can be used to improve Bio-fuel engine performance concerning delay period?

1.6. SIGNIFICANCE:

Why the research is important?

The importance of the work is the applications and usage of diesel engines in our life and; our seeking for better performance of engine on fixed or mobile machinery.

Since a lot of researcher locally and abroad have covered different areas of bio-diesel fuel, it will be better to extend their efforts for the benefits of our local community and the international one as well .

The knowledge of efficient use of bio-fuel, will help in good utilization of resources and energy saving.

1.7. Research Methodology

Samples were prepared in forms of (Bio-0.0, Bio-25, Bio-50, Bio-75 and Bio-100). Empirical formula using experimental data is used to calculate the delay period. Fuel blends fired in an experimental engine and readings taken for calculating all performance criteria. Results calculated of delay period and performance criteria calculated analyzed, discussed and presented in graphs form.

CHAPTER II

LITERATURE REVIEW

CHAPTER II

LITERATURE REVIEW

2.1 Introduction:

A lot of published work and research was reviewed concerning the research. The review was grouped into three categories namely; bio-fuels, engine fuelled with bio-fuel performance and ignition delay measuring and calculating.

2.2 Bio-fuel and jatropha:

According to Jatropha Handbook (Foundation, 2010) Jatropha curcas (Latin origin name) is often referred to as ‘jatropha’. It is a plant that produces seeds with high oil content. The seeds are toxic and in principle non-edible (not cooking oil). Jatropha grows under (sub) tropical conditions and can withstand conditions of severe drought and low soil fertility. This makes it a good desertification plant. Because jatropha is capable of growing in marginal soil, it can also help to reclaim problematic lands and restore eroded areas. As it is not a food or forage crop, it plays an important role in deterring cattle, and thereby protects other valuable food or cash crops.

The utilization of biodiesel is an old one, since it has been used as a substitute for mineral diesel since early 20th century, but in small amounts. The new is that from 2005 onwards biodiesel production and use has increased significantly, spearheaded by the EU (mostly in Germany and France), now responsible for about 80% of the world production. A part of this European dominance, biodiesel production is expected to stabilize in the coming years in the EU, with substantial growth expected in South America (Brazil, Argentina and Colombia) and Asia. (Frank Rosillo-Calle, 2009)

2.2.1 Jatropha fuel:

Fortunately, inedible vegetable oils, mostly produced by seed-bearing trees and shrubs can provide an alternative. With no competing food uses, this characteristic turns attention to Jatropha curcas, which grows in tropical and subtropical climates across the developing world.

2.2.2 Jatropha biodiesel:

What is bio-diesel? According to Akhihiero (2013), biodiesel is a non-polluting, locally available, accessible, sustainable and reliable fuel obtained from renewable sources such as vegetable oils or animal fats by transeserification process (Akhihiero E. T., 2013). An efficient method of producing jatropha bio-diesel from jatropha seeds oil is ultrasound assisted transeserification method used by Amish, India (Amish P. Vyas, 2011)

Hnny et al, (2008) treated crude jatropha curcas-seeds oil (CJCO) with a high content of Free fatty acids (FFA) and investigated its production of biodiesel fuel. In alkali base catalyzed and show that transesterification process, the presence of high concentration of FFA reduced the yield of methyl esters of fatty acids significantly (Hanny Johanes Berchmans and Hirata, 2008).

Abdulkareem A.S. et al (2012) the study report the production of biodiesel from non-edible oils (Jatropha Carcus and Ricinus Communis) as alternative to petrol diesel. Based on the results of experimental analysis, it can be concluded that oils from both feed stocks are suitable for the production of biodiesel.(Abdulkareem A.S., 2012).

Hanny J. B. et al (2008), searched on Bio-diesel production from jatropha seeds oil and the chemical treatment concerning the biodiesel production from jatropha oil with a high quality was possible. In alkali base catalyzed transe-sterification process, the presence of high concentration of FFA reduced the yield of methyl esters of fatty acids efficiently. In a two-stage transesterification process. The first stage was acid pretreatment process, and the second stage, alkali base cat-alyzed transesterification process. The produced bio-diesel was fully soluble in petroleum diesel oil fuel and gave no problems in injection in CI engine or in burning separately or in blends with diesel as observed on test. (Hanny Johanes Berchmans and Hirata, 2008).

2.3. Engine performance:

Benjamin et al (2014) were tested in a single cylinder direct-injection diesel engine fuelled with jatropha bio-diesel fuel blends in diesel fuel to investigate the operational parameters of a small capacity diesel engine under six engine loads. The investigated blends were 40/60%, 30/70%, 20/80% and 100% jatropha biodiesel at various loads. Each blend was tested on a short term basis and the result shows that the brake thermal efficiency increased for all tested blends at lower engine loads and decreases at higher engine loads.

The specific fuel consumption (S.F.C) increased for lower blends compared to jatropha oil while higher engine powers were obtained for lower blends compared to jatropha oil.

In all the investigated operational parameters, the diesel reference fuel had better performance to jatropha biodiesel blends except in the percentage heat loss to the exhaust where jatropha biodiesel blends had better performance (Benjamin Ternenge Abur. Abubakar Adamu Wara, 2014).

Elango et al, (2011) blends taken starts from B0.0 with increments of ten up to B50. The results were: The smoke opacity is found to be higher than diesel for all blends, but blends up to 20% substantially reduce CO_2 emissions with a marginal decrease. This research has focused mainly on engine emissions more than engine performance and its elements. A relatively high capacity single-cylinder water-cooled direct-injection diesel engine was evaluated using only two blends of biodiesel (B10 and B20) obtained from a mixture of two different oils (50:50) with high-speed diesel (HSD) in terms of brake specific fuel consumption, brake thermal efficiency, and exhaust gas temperature and emissions such as CO, HC, and NOx. It was shown that, based on performance and emissions, blend B10 was selected for long-term use. (Elango and Senthilkumar, 2011)

Hifjur Raheman, (2013) experiments were also conducted to assess emissions of soot deposits on engine components, such as cylinder head, piston crown, and fuel injector tip, and addition of wear metal in the lubricating oil of diesel engine when operated with the biodiesel blend (B10) for certainly long time. The amount of soot deposits on the engine components was found to be, on average, 21% lesser for B10-fueled engine as compared with HSD-fueled engine due to better combustion. The research also investigated engine wear of metals such as copper, zinc, iron, nickel, lead, magnesium, and aluminum, except for manganese, in the lubricating oil of B10-fueled engine after engine operation and was found to be 11% to 50% lesser than those of the HSD-fueled engine due to additional lubricity (Hifjur Raheman, 2013).

(Karthick.D, 2014)) conducted an investigation of CI engine fuelled with jatropha oil – referred to as (J) with addition of Di-etheyl Ether denoted as (DEE) blends in diesel oil is that of Karthick.D, where this time the works concentrates on varying compression ratio effect, he discloses that; brake thermal efficiency (BTE) of the blend increases with increase in applied load and maximum BTE at full load is 30.31% for J17+DEE 3% which is 7.3% higher than that of diesel. The NOx emission decreases when the compression ratio decreases but using of biodiesel increased NOx upto 10% because of higher cetane number (Karthick.D, 2014).

2.4 Ignition delay:

El-Kasaby et al, (2013) has an experimental investigations of ignition delay period and performance of a diesel engine operated with Jatropha oil biodiesel, their work has focused on the engine performance with DI measured using empirical formula without treating the delay itself. (El-Kasaby and Nemit-allah, 2013).

Ishag, (2006) conducted experimental Investigation of the effect of the protuberances in pre-combustion chamber on delay period the author has invented a highly sensitive technique for measuring DI of CI Engine by detecting start of injection and start of combustion events by using a photo detector for each of the two events and an oscilloscope to plot and measure delay time between the two (Ishag, 2006).

Md Nazeem Khan et al (2015) their studies were focused on a blend that behaves closer to ordinary diesel fuel. An experimental evaluation of C.I. engine performance using diesel blended with Jatropha biodiesel to study the effect of delay period on an engine running on bio-diesel fuel, the bio-diedel fuel testes here was that of coconut oil**.** Delay period investigations can also be studied in a rapid compression machine (Hao Liu et al., 2014), not a real running IC engine this method allow to study the phenomena at different pressure ratios with no link to specvific engine performance.

Kotzé, (2010) study the performance of biodiesel in a modern diesel engine taken by, the study has considered the general performance of in a turbo engine with no specific consideration effect of the delay period. The electronic engineering work is a review for designing high precise sensitive e-circuit for measuring time delay in different methods, one can use one suitable method to design his own for this specific application**.** Kotzé, J. (Kotzé, 2010)

Akbar, (2009) studied characteristic and Composition of Jatropha Curcas Oil, they concluded that the oil extracts of jatropha exhibited good physicochemical properties and could be useful as biodiesel feedstock and industrial application (Akbar, 2009)

Kuleshov A.S. et al (2010) published a report concerning ignition delay studies with computer simulation using different combustion criteria such as: (the piston bowl shape, the injector design and its location, the shape of the injection profile including multiple injection modes. The RK-mathematical model accounts, for the droplet sizes, interaction of free fuel sprays with the swirl, the spray/wall impingement, the evolution of the near-wall flow formed by the spray, fuel reaching the surfaces of the cylinder head and of the cylinder liner, mixing of the Near-Wall Flow (NWF) formed by adjacent sprays, the effect of the piston motion and the swirl intensity on the heat release rate, the exhaust gas remain (EGR), temperatures of the piston and the cylinder head in calculations of the heat release process. The application of multi-parametric optimization procedures in combination with the RK-model of engines provides the high efficiency of computational research which is an important in engine design process(Kuleshov, 2010).

Senthilkumar, et al (2011) studied the emissions of a diesel engine fuelled with blends of diesel bio-diesel fuel, the work reveals that; the smoke opacity is found to be higher than that of diesel for all blends, but blends up to 20% substantially reduce CO_2 emissions with a marginal decrease in brake thermal efficiency (Elango and Senthilkumar, 2011).

Hathout, J.P. et al (2000) investigate ignition time-delay and its effect induced on combustion instability. The combustion problems due to delay ignition can have short or long term solutions as: short-term by use of multiple injectors and low frequency pulsing. A long-term solution proposed, by the design of a high-speed, high-authority injector (Hathout, 2000).

Hillion M., et al present a research concerning combustion control of diesel Engines using injection timing, their final result was that, the method followed was a general and can be applied to a large scope of engine architectures. In particular, it is completely independent of the air-path architecture (Hillion, 2009).

Mbarawa M. (2003) studied ignition delay estimation, correlation for estimation of ignition delay of dual fuel combustion based on constant volume combustion, the research helps in ignition delay estimation techniques for dual fuel which is similar to that of jatropha bio-diesel blends in diesel oil, he concluded to that; a general correlation imperical equation was established to calculate ignition delay (τ_{id}) as a function of pressure, mass, temperature and equivalence ratio in the form (Mbarawa, 2003):

$$\tau_{id} = 19.3 P_0^{\,f(\phi_{gas})} g\left(\varphi_{gas} \cdot m_{inj}\right) Exp\left(\frac{500}{T_o}\right) \qquad \cdots\cdots \quad (2.1)$$

Stalin N. et al (2007) investigated the performance of IC engine fuelled with karanja biodiesel blending with diesel, they found that; it was possible to reach up to B100 with the type bio-fuel, for all fuel samples tested, torque, brake power and brake thermal efficiency with maximum values at 70% load. They recommended dual fuel combination of B40 for use in the diesel engines without making any engine modifications. Also the cost of dual fuel (B40) can be considerably reduced than pure diesel fuel (Stalin N., 2007).

Zhukov V.P. et al, the work estimate delay ignition time for a petroleum fuel of kerosene. The important is the technology used, as ignition of Jet-A/air mixtures was studied behind reflected shock waves. A heated shock tube at temperature of 150 ^{0}C was used to prepare a homogeneous fuel mixture. Ignition delay times were measured from OH emission at 309 nm ($A^2\Sigma - X^2\Pi$) and from absorption of He-Ne laser radiation at 3.3922 µm. The conditions behind shock waves were calculated by one-dimensional shock wave theory from initial conditions T_1, P_1, mixture composition and incident shock wave velocity. The ignition delay times were obtained at two fixed pressures 10, 20 atm for lean, stoichiometric and rich mixtures (ϕ = 0.5, 1, 2) at an overall temperature range of 1040-1380 K (Zhukov, 2006).

AlirezaValipour, (2014) has a valuable work on delay period experimentally measured for jatropha bio-diesel blends (JBD) in diesel oil fuel, the test was conducted on a fixed constant pressure cylinder using a photo method to detect injection and ignition events and an oscilloscope to draw and measure events duration and the delay between them. As the test was not on an actual running engine, but its result will be a useful and give good and clear image on what is happening for different blends combustion. The results reported that ignition delay of diesel fuel and JBD blended with diesel is highly depends on injection pressure and ambient air pressure with shorter ignition delay which is prolonged with increasing biodiesel content in the blends (Valipour, 2014)

Venkatraman, M., et al (2011) they studied ignition delay and CI engine performance through computer simulation. The suggested fuel is diesel bio-diesel blends seeking for optimum performance under different operation parameters. Bio-diesel fuel selected for the research was pungam methyl ester (PME) with blend percentages (10, 20 and 30). They concluded that; the developed simulation model seems to be a useful tool for analyzing the diesel engine combustion process accurately. Thus it can be used to predict the various performance and emission parameters of any vegetable oil esters with minimum inputs such as density, calorific value, chemical formula for a given engine specification. (Venkatraman and Devaradjane, 2011).

Rehman S., (2016), a new invented method of measuring ignition delay is the one used by the test was carried out in a fixed hot surface cylinder of the solid cone diesel fuel sprays impinging on the hot surface, where ignition delay of fuel sprays get reduced with the rise in injection pressure(Rehman, 2016).

Adam A. used the method on a real using optical signals running CI engine and, the previous studied the delay phenomena only while Adam has studied beside DI more other combustion characteristics like injection duration, rate of heat released, injection pressure and more other important combustion characteristics of diesel engine.

Nazeem Khan et al (2015), present an experimental work to investigate and measure ignition delay of blends of diesel oil and coconut oil-diesel fuel at various ambient pressures (cylinder pressures) shows that ignition delay is short at high cylinder pressure, photo-sensors were used for detecting the start of combustion for a single blend B50 under varying pressure conditions.(Nazeem Khan and Hussain, 2015)

Benjamin T., et al (2014) show that the parametric Study of Jatropha blended in diesel fuel was carried out for performance study with no link with delay period parameter (Benjamin Ternenge Abur. Abubakar Adamu Wara, 2014).

CHAPTER III

FUELS AND COMBUSTION

CHAPTER III

FUELS AND COMBUSTION

3.1.**Internal Combustion Engines (IC Engines)**:

Internal combustion engine is defined as an engine in which the chemical energy of the fuel is released inside the engine and used directly for mechanical work, as opposed to an external combustion engine in which a separate combustor is used to burn the fuel (Colin R. Fweguson, 2016). Internal combustion engines can be of rotary type like Wankle engine or the ordinary reciprocating piston engine.

3.1.1 Reciprocating IC engines:

There are basically two types of IC ignition engines. Spark ignition engines (SI) take a mixture of fuel and air, compress and ignite it using a spark plug. In a spark ignition engine a spark plug is required to transfer an electrical discharge to ignite the mixture. Compression ignition engines (CI engines) take atmospheric air compress it to high pressure and temperature, CI engines are auto-ignition engines, injected spray of fuel on the high temperature high pressure air ignites alone in the engine's combustion chamber.

The piston-cylinder engine is basically a crank-slider mechanism. The piston moves up and down by the rotary motion of the two arms or links. The crankshaft rotates which makes the two links rotate. The valves on top represent induction and exhaust valves necessary for the intake of an air-fuel mixture and exhaust of chamber residuals (Salazar, 1998).

3.1.2 Two and Four Stroke engines:

Reciprocating IC engines can be either two stroke engines or four stroke engines. In two stroke engines, the four basic events (induction, compression, power and exhaust) complete in two strokes of the piston movement from TDC to BDC and reverse back; four stroke engines need four stroke of the piston movement to complete its thermodynamic cycle.

3.1.3 Fuel ignition in IC engines:

Internal combustion engines are divided according to ignition principle type into spark ignition engines and compression ignition engines. Spark ignition engines use an air- fuel mixture that is compressed at high pressures. The mixture has to be near stoichiometric to be chemically inert and able to ignite and have an overall even amount.

In a compression ignition engine a high temperatures and pressures in the combustion chamber cause a flame to initiate at different sites of the combustion chamber. Combustion increases with increasing pressure and temperature. Compression ignition engines are divided into direct and indirect ignition engines. Diesel engines require fuel injection systems to inject fuel into the combustion chamber.

3.1.4 Direct and indirect fuel injection in CI engines:

Diesel engines require fuel injection systems to inject fuel into the combustion chamber. Fuel injection systems are either linear or rotary. Direct injection engines need high pressures rates to inject fuel into the combustion chamber, because heat addition process takes place at a compressed state, so in order for the fuel to inject well the pressure has to be greater than the one that the pressure accumulated through compression chamber. Indirect ignition engines have a pre-combustion chamber where the air-fuel mixture is stored. The purpose of the separate chamber is to speed up the combustion process in order to increase the engine output by increasing the engine speed. The two basic combustion systems are the swirl and pre-combustion chambers. Pre-combustion chambers depend on turbulence to increase the combustion speed and swirl chambers depend on the fluid motion to raise combustion speed. In divided chambers the pressure required is not as high as the pressure required for direct ignition engines. The Diesel engine has high thermal efficiencies, and therefore low fuel consumption. The disadvantage of diesel engines is their low power output relative to their weigh, as compared with spark ignition engines (Salazar, 1998).

3.1.5 Air intake and exhaust release in IC engines:

In internal combustion engines the induction and exhaust processes give importance to the performance and efficiency of the engine, in the four stroke engine the induction and exhaust processes are controlled through valves. The most commonly used valve is the poppet valve. The poppet valve has a straight cylinder rod and its end has the shape of a mushroom. A camshaft is used in the mechanism that operates the valves. Engines that use overhead poppet valves (OHV) use a camshaft that ‘is either mounted in the cylinder block or in the cylinder head’. Overhead camshafts (OHC) use chain or toothed belts to provide its drive. Valve timing is characterized by the camshaft and valve mechanism.

3.2. **Fuels General**:

Fuels were originally regarded as combustible substances used solely or mainly for the production of useful heat(Francis, 1965). Fuels could be divided into three natural classes: solids, liquids gases. Fuel definition in now widened to include uranium, plutonium and thorium, all of which can be include undergoing nuclear fission to produce useful heat for conversion into power(Mahir Husain, 2015).

3.2.1 Liquid Fuels:

Liquid fuels are those kinds of fuels found naturally or artificially in liquid form, natural liquid fuels include: petroleum liquids and bio-liquid fuels (vegetable oils and animal fats), artificial liquid fuels include: petroleum liquid products (gasoline, kerosene and residual oil) beside bio-liquid fuels like alcohols.

3.2.2 Bio-Fuels:

The material of plants and animals is called biomass. It is organic carbon based material that reacts with oxygen in combustion and natural metabolic processes to release heat. The initial material may be transformed by chemical and biological process to produce intermediate bio-fuels such as methane gas, ethanol liquid or charcoal solid.(Francis, 2000)

3.2.3 Bio-Diesel:

Biodiesel is the name of a clean burning alternative fuel, produced from domestic, renewable resources. Biodiesel contains no petroleum, but it can be blended at any level with petroleum diesel to create a biodiesel blend. It can be used in compression-ignition (diesel) engines with little or no modifications. Biodiesel is simple to use, biodegradable, nontoxic, and essentially free of sulfur and aromatics(Max J. A. Romero, 2014). A lot of benefits gained on using bio-diesel standard blends some of which are(Robert E. Reynolds, 2007):

1. Higher cetane
2. Improved lubricity
3. Low sulfur content
4. Reduces HC and CO emissions
5. Reduces PM emissions
6. Lowers visible smoke
7. Zero aromatics
8. Reduces net CO_2 emissions on life-cycle basis
9. Nontoxic and biodegradable

On the other hand using of bio-diesel blends should be under some important considerations listed below(Robert E. Reynolds, 2007):

1. Materials compatibility – especially with higher blend levels
2. Potential for increased water content and microbial contamination
3. Increase in NO_x emissions
4. Implications for emission control equipment
5. Impact on low-temperature operability if not properly additized
6. Reduced power and fuel economy (on B100) – lower energy content
7. Thermal and oxidative stability poorer than No. 2 diesel
8. Solvency effect may plug filters on initial use
9. Requires special care and handling
10. Spills could damage paint
11. Higher cost (present time)

3.2.4 Jatropha and Jatropha Bio-Diesel:

(a) Jatropha curcas:

Jatropha curcas as a biodiesel “miracle tree” can help alleviate the energy crisis and generate income in rural areas of developing countries. Jatropha is becoming a poster child among some proponents of renewable energy and appropriate technology, especially as an oil-bearing, “drought resistant” tree for marginal lands for small farmers.(Ministry of Science, 2014)

Oil from jatropha is extracted from the seeds of the plant in different methods. The more sophisticated and efficient method of extraction produces seed-cake with much lower oil content. (Benge, 2006)

(b) Jatropha Bio-Diesel:

Biodiesel is mono-alkyl esters of fatty acids derived from vegetable oils or animal fats, is known as a clean and renewable fuel. Biodiesel is usually produced by the transesterification of vegetable oils or animal fats with methanol or ethanol.(Akbar, 2009) Biodiesel has many advantages include the following: its renewable, safe for use in all conventional diesel engines, offers the same performance and engine durability as petroleum diesel fuel, non-flammable and nontoxic, reduces tailpipe emissions, visible smoke and noxious fumes and odors. The use of biodiesel has grown dramatically during the last few years. Feedstock costs account for a large percent of the direct biodiesel production costs, including capital cost and return. (Akbar, 2009)

3.3 **Combustion in internal combustion (IC) engine**:

In SI engine, uniform A/F mixture is supplied, but in CI engine A/F mixture is not. Homogeneous, and fuel remains in liquid particles, therefore quantity of air supplied is 50% to 70% more than stiochiometric mixture.(Jagadeesha T, 2007)

The combustion in SI engine starts at one point and generated flame at the point of ignition propagates through the mixture for burning of the mixture, where as in CI engine, the combustion takes place at number of points simultaneously and number of flames generated are also many. To burn the liquid fuel is more difficult as it is to be evaporated; it is to be elevated to ignition temperature and then burn.

3.3.1 Combustion Characteristics in compressive ignition (CI) engine:

Combustion occurs throughout the chamber over a range of equivalence ratios dictated by the fuel-air mixing before and during the combustion phase.

In general most of the combustion occurs under a rich conditions within the head of the jet, this produces a considerable amount of solid carbon (soot) in CI engine Types.i) **Direct-injection** engines – have a single open combustion chamber into which fuel is injected directly (Figure 1).

ii) **Indirect-injection** engines – chamber is divided into two regions and the fuel is injected into a "pre-chamber" which is connected to the main chamber via a nozzle, or one or more orifices (Figure 2).

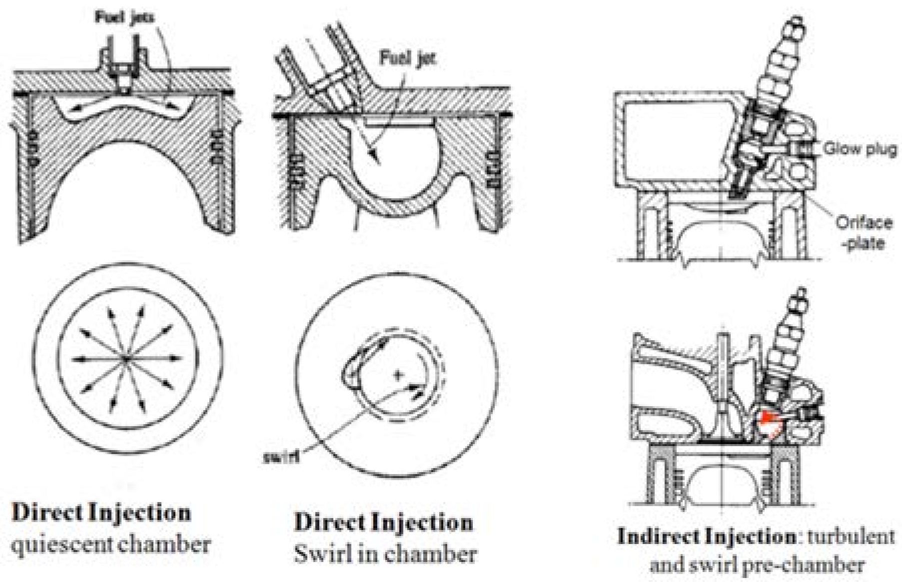

Figure (3.2) Direct Injection Engine

Figure (3.1) Indirect Injection Engine

3.3.2 Combustion in spark ignition (SI) engine:

- Combustion in a CI engine is quite different from that of an SI engine. While combustion in an SI engine is essentially a flame front moving through a homogeneous mixture, combustion in a CI engine is an unsteady process occurring simultaneously in many spots in a non-homogeneous mixture controlled by fuel injection.
- Fuel is injected into the cylinders in the compression stroke by one or more injectors located in each cylinder [Point (A) Figure 3]. Injection time is usually about 20^o of crankshaft rotation angle (15^o bTDC and 5^o aTDC).(Saha)

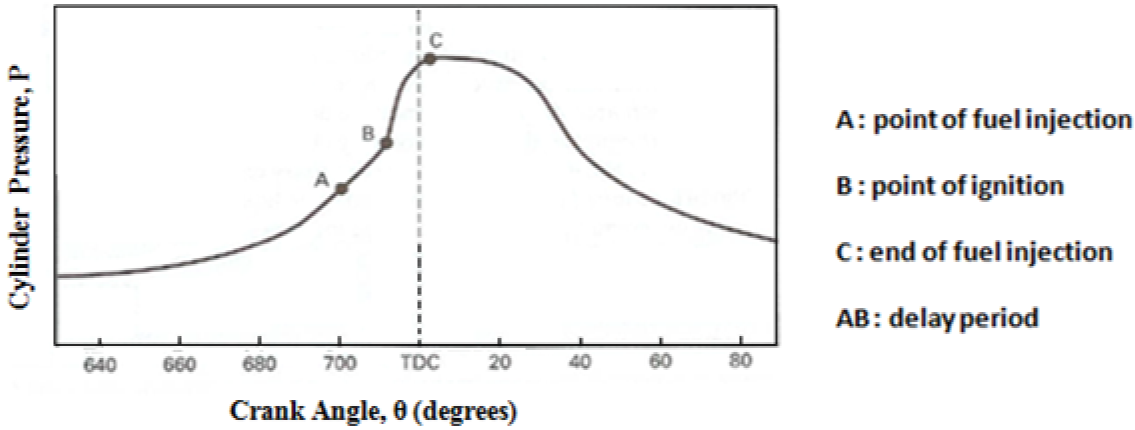

Fig. (3.3) Pressure vs. Crank Angle in CI engine Diagram

3.3.3 Delay Period (Delay Ignition) [DI]

Delay period can be defined as the time interval between the beginning of fuel injection and the beginning of fuel ignition (Figure3.5). The delay period has an important role in compression ignition engines (Diesel) which are widely used for power generation and other purposes.(Mahir Husain, 2015)

The ignition delay period should be kept as short as possible in order to limit the portion of fuel injected during the period, otherwise it will be difficult to start the engine since the rate of pressure rise after ignition may be excessive leading to noisy engine and highly stressed crankshaft.

3.3.4 Ignition Delay Measurements:

a. In Cylinder Measurements:

The measurement of Ignition Delay is done by the Scope Meter, and the values of parameters are fixed by Pressure gauges and Temperature controller. A Reciprocating axial flow Compressor is used for charging the Air in the combustion chamber.(Hao Liu et al., 2014)

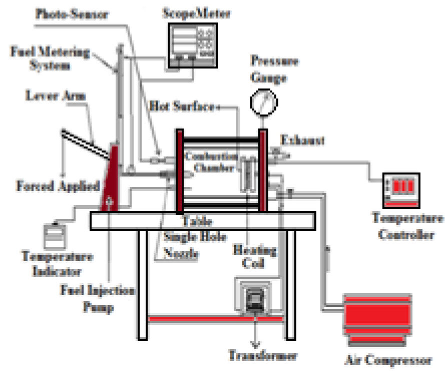

Figure (3.4) Delay Period Machine arrangement

b. Empirical Equation Method:

Test rig using special device, or to be calculated for different blends with diesel fuel as a function of cylinder pressure, cylinder temperature, and equivalence ratio using an experimental equation (El-Kasaby and Nemit-allah, 2013 as follows:

$$\tau_{id} = A \cdot P^{-n} \cdot \phi_1^{-m} EXP\left(E_a / \bar{R} \cdot T\right) \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (1)$$

From the ignition delay results using least square fitting technique, the coefficients A, n and m can be calculated and the general empirical relation of each blend, where (E_a) is a constant calculated from the relation:

$$E_a = 816{,}840 / \left(CN + 25\right) \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (2)$$

(El-Kasaby and Nemit-allah, 2013)

c. Computer simulation method:

Prediction of auto-ignition delay period for high temperature combustion using empirical equations (Computer simulation):

The auto-ignition delay period τ_i for each j-portion of fuel was calculated using different methods for various engines running under different operational conditions and the results were compared using special numerical software (DIESEL-RK software)

d. Cetane Number Test:

Cetne number can be used as a measuring tool for delay period calculation. The two types of these test are; the constant speed and load test, and the Throttling Test.

3.3.5 Factors affecting Ignition Delay

1. **Compression Ratio** (CR): DP decreases with increase of CR.
2. **Engine Speed**: DP decreases with increase of engine speed.
3. **Power Output**: DP decrease with increase of power output.
4. **Fuel atomization**: DP decrease with fineness of atomization
5. **Fuel Quality**: Decreases with higher cetane number.
6. **Intake Temperature & Pressure**: DP decreases with increase of temperature and pressure.

3.3.6 Combustion Stages in CI Engine:

Figure (3.5), show the fuel injection flow rate, net heat release rate (Q) and cylinder pressure (P) for a direct injection CI engine. Relative locations of various points on the graph have an indication on the pattern of combustion taking place. The figure (3.5) is for good and smooth form of combustion.

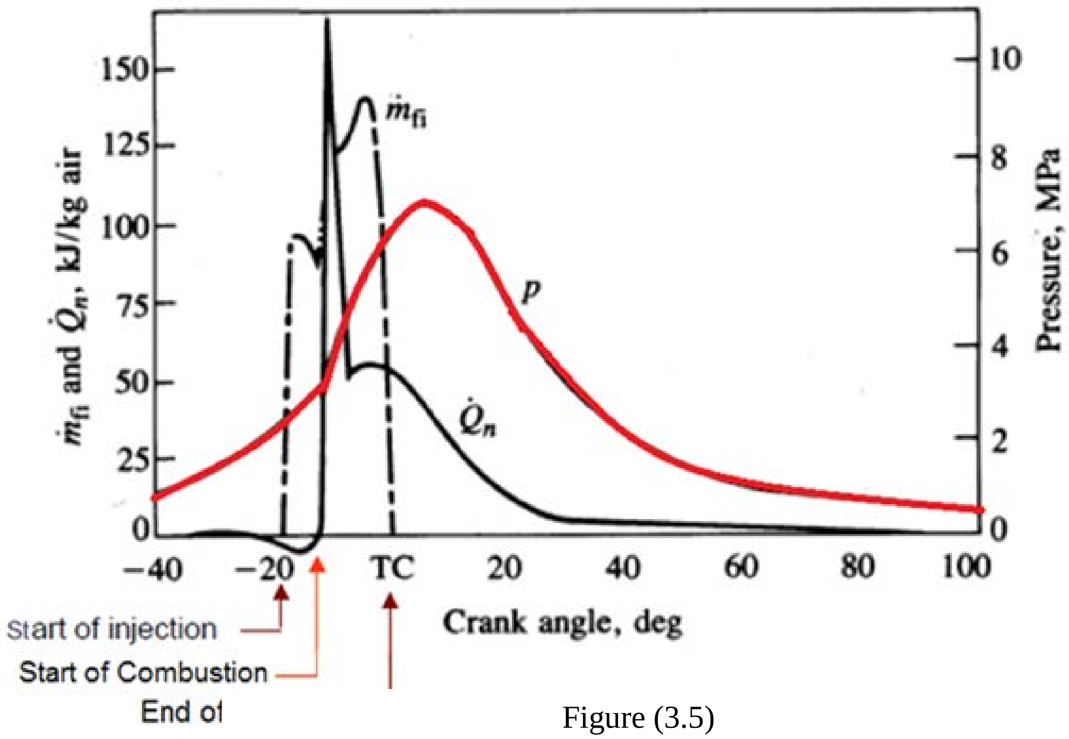

Figure (3.5)
[fuel quantity (Q) vs Crank Angle ic CI engine Diagrams

Figure (3.6) below, shows the stages of combustion in CI engine and the events in each of the stages are described for the analysis of the combustion.(Saha)

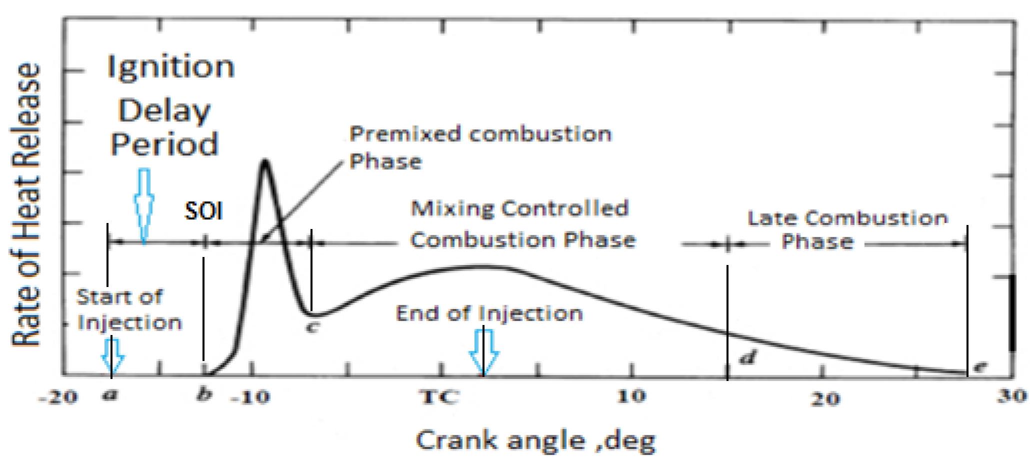

Figure (3.6)
Stages of Combustion Diagrams

(1) Ignition delay (a-b) - fuel is injected directly into the cylinder towards the end of the compression stroke. The liquid fuel atomizes into small drops and penetrates into combustion chamber. The fuel then vaporizes and mixes with the high-temperature high-pressure.

(2) Premixed combustion phase (b-c) combustion of the fuel which has mixed with the air to the flammability limits during the ignition delay period occurs rapidly in a few crank angles.

(3) Mixing controlled combustion phase (c-d) after premixed gas consumed, the burning rate is controlled by the rate at which mixture becomes available for burning. The rate burning is controlled in this phase primarily by the fuel-air mixing process.

(4) Late combustion phase (d-e) heat release may proceed at lower rate well into the expansion stroke (no additional fuel injected during this phase). Combustion of any unburned liquid fuel and soot is responsible for the stage. Two processes take place before significant fraction of the chemical energy of the injected liquid fuel is released; namely, physical and chemical processes.

   A. **Physical processes** are fuel spray atomization, evaporation and mixing of the vapor with cylinder air.

   B. **Chemical processes** are similar to those of auto-ignition phenomenon in premixed fuel-air, only more complex since heterogeneous reactions (reactions occurring on the liquid fuel drop surface) also occur.

3.3.7 Fuel Ignition Quality

Generally, two characteristic fuel numbers affect ignition quality in Internal combustion engines (Octane Number for SI engine and Cetane Number for CI engine)(Francis, 1965)

(a) **Octane Number**

The octane number of a gasoline is a measure of its tendency to nock, or detonate, when burned in a spark-ignition engine. Detonation is a rapid and uncontrolled burning of fuel and air mixture in a cylinder, which results in an abnormally rapid pressure rise (Salazar, 1998). The vibrations of the gases the cylinder walls and other metallic surfaces giving a distinct knock or noise.

Combustion phenomena may be affected by the tendency for any fuel to varies with factors as; type of engine, engine speed, air fuel ratio, temperature of fuel and air entering the cylinder, compression ratio and many others.

It has been proved that detonation tendencies are related to the auto-ignition temperature of a fuel and the chain reaction mechanism by which the fuel burns. A liquid such as iso-octane exhibits smooth burning characteristics. In contrast with his longer chain compound, normal heptane, displays a a strong tendency towards detonation.

(b) **Cetane Number (CN)**:

Cetane Numbers are used to indicate the quality of a fuel oil for compression-ignition engines as follows (Saha).

1. The straight-chain hydrocarbon cetane ($C_{16}H_{34}$) is perhaps the best high-speed diesel fuel known, and is given a rating of 100.
2. Aromatic hydrocarbons are poor diesel fuels, and the aromatic hydrocarbon methyl-naphthalene is given a rating of 0.
3. The cetane number of diesel oil is the percentage by volume of cetane in a cetane/methyl naphthalene mixture that has the same performance in a standard compression ignition engine as that of the fuel.

- By definition, isocetane (**heptomethylnonane**, **HMN**) has a cetane 15 and cetane (n-hexadecane, $C_{16}H_{34}$) has a value of 100.

- The higher the CN the better the ignition quality, i.e. shorter ignition delay time.

These following criteria are used to indicate the quality of a fuel oil for compression-ignition engines.

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3. The cetane number of a diesel oil is the percentage by volume of cetane in a cetane/methyl naphthalene mixture that has the same performance in a standard compression ignition engine as that of the fuel.

- For low cetane fuels, the ignition delay is long and most of the fuel is injected before auto-ignition and rapidly burns, under extreme cases this produces an audible knocking sound referred to as "diesel knock".(Hathout, 2000)

- For high cetane fuels the ignition delay is short and very little fuel is injected before auto-ignition, the heat release rate is controlled by the rate of fuel injection and fuel-air mixing – smother engine operation.

(c) **Cetane Number Measurement**

- The method developed to measure CN uses a standardized single-cylinder engine with variable compression ratio run at the following operation condition

The operating condition is according to table below:

Table (3.1) Standard operation condition for CN defining

Inlet temperature (^oC)	65.6
Speed (rpm)	900
Spark advance (^{o}BTDC)	13
Coolant Temperature (^oC)	100
Injection Pressure (MPa)	10.3

- With the engine running at the shown conditions on the test fuel, the compression ratio is varied until combustion starts at TC, ignition delay period of 13^o.

- The above procedure is repeated using blends of cetane and HMN. The blend that gives a 13^o ignition delay with the same compression ratio is used to calculate the test fuel cetane number.

- **Method of Determining the "Cetane Number" in Test Engine:**

  Because of the high cost of pure cetane and methyl-naphthalene, a number of standard reference fuel oils are available with a range of cetane numbers. Two methods of test are specified, both of which may be carried out on any compression-ignition engine:

  A. Constant speed and Constant Load Test:

  This test is carried out at constant speed and load. The delay time is measured for the oil under test with an electronic delay meter, and compared with standard reference fuels having delay periods shorter and longer than that of the sample fuel. The cetane number is obtained by interpolation.

  B. Throttling Test:

  Test engine is run at the lowest load, which gives steady conditions. A surge chamber and throttle device is attached to the engine intake port. This device reduces the surge chamber pressure and increases the delay period until a misfire occurs, which is indicated by a puff of white smoke. The air pressure at this point is related to the delay period and is a function of the cetane number. By bracketing the pressure for misfire on the sample fuel with reference fuels of higher and lower quality, the cetane value can be calculated.

(e) **Cetane V.S Octane Number**:

The octane Number and cetane number of a fuel are, inversely correlated.

Gasoline is a poor diesel fuel and vice versa. (Figure (6) Below).(T)

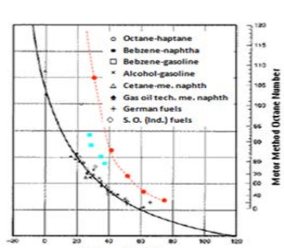

Figure (3.7)
Cetane Number vs Octane Number Diagrams

3.3.8 **Knock in CI Engines**:

- Knock in SI and CI engines are fundamentally similar. In SI engines, it occurs near the end of combustion; whereas in CI engines, it occurs near the beginning of combustion.

- Knock in CI engines is related to delay period. When DP is longer, there will be more and more accumulation of fuel droplets in combustion chamber. This leads to a too rapid pressure rise due to ignition, resulting in jumping of forces against the piston and rough engine operation. When the DP is too long, the rate of pressure rise is almost instantaneous with more accumulation of fuel.

- It is evident that straight alkane hydrocarbons ignite more readily than aromatics and so are more suitable diesel fuels. In fact, all types of light fuel oils with low-octane numbers are satisfactory high-speed diesel fuels. High-octane fuel oils are not satisfactory high-speed diesel fuels. In other words, good spark ignition fuels make poor compression-ignition fuels, and vice versa.

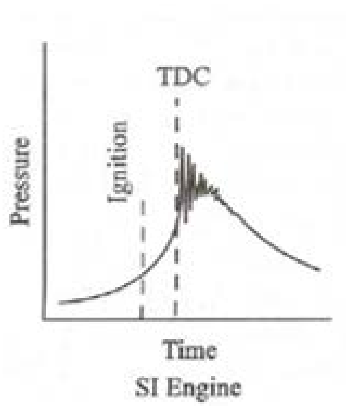

Figure (3.8) Detonation Wave in SI engine Diagrams

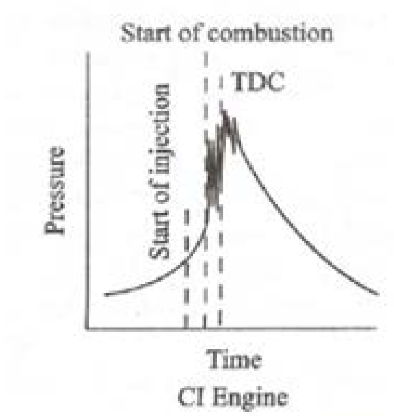

Figure (3.9) Pressure Nock Wave in CI engine Diagrams

3.4 **Bio-fuels/ Bio-Diesel Blend**:

3.4.1 **Bio-fuels:**:

Biodiesel as an alternative fuel derived from vegetable (green) oil or animal fats are oxygenated, biode-gradable, non-toxic and environmentally safe. It consists of al- kyl monester of fatty acids from tri-acyglycerols. Biodiesels are classified into two categories; namely edible and non-edible oils. Edible oils are such that sunflower, corn, rapeseed, palm, soybean and waste vegetable oils. The non-edible oils are such that Jatropha, Jojoba, Karanja, Polanga oils and likes. All can be used separate or been mixed in some percentage with ordinary diesel oil(Antony Raja, 2011).

3.4.2. Bio-Diesel Blend:

Physical and chemical properties of jatropha crude, jatropha bio-diesel and diesel oils fuel are given in table below(PLRS, 2004):

Table (3.2) fuels properties

Properties	Jatropha crude	Jatropha bio-diesel	Diesel	Test standard
Density, g/ml	0.92	0.865	0.841	--------
Viscosity, C. stoke @ 40 ^{o}C	32.5	5.2	4.5	ASTM D445
Heating value MJ/kg	35.2	34.5	42	ASTM D240
Flash point ^{o}C	240	175	50	ASTM D93
Cloud point ^{o}C	16	13	9	ASTM D2500
Cetane Number	38	51	49	ASTM D93

CHAPTER IV

EXPERIMENTAL WORK

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4.1. Test Rig

4.1.1 **Base frame**: Base frame is made of M.S. C-channel and powder coated for durability. The Engine is mounted on separate frame provided with 'caster wheels' to allow shifting of engines on the frame, while generator is mounted on stationary frame provided with stationary frame provided with 'anti vibration pads' to damp the engine vibrations. A magnetic coupling is used to transmit the engine power to dynamometer. The engine can be coupled to the shaft of the magnetic coupling by means of removable teflon toothed sleeve also the magnetic coupling is joined to the dynamometer in the same way. The engine is brought close adjacent to the magnetic coupling and bolted to the dynamometer frame by means of four high tension bolts.

4.1.2 **DC Generator cum motor**: A DC generator (capacity 8 kW) which can be also used as Motor acts as dynamometer to measure power of the engine. It can be run as a motor to conduct motoring test and start the engine. The control panel for generator is fitted on load bank. It is a swinging field generator supported on bearings at both ends and a load cell measures the load acting on the generator due to torque generated.

4.1.3 **Control Panel**: Control panel is fixed over the dynamometer frame and houses following accessories and instruments:-

a- Fuel tank and measurement unit:

b- Measuring instruments / indicators / switches

Load cell	Temperature indicator
Load indicator	Air box
Inductive RPM sensor	Differential pressure gauge
RPM indictor	Timer
K' type thermocouples	Torque reversal switch

a) **Fuel tank and fuel measurement unit**:

Separate fuel tanks are fitted for pure petrol, oil mixed petrol (for 2-stroke engine) and diesel. Each fuel tank is provided with fuel measurement unit comprising of float operated level switch, solenoid valve and digital timer. The level sensor is fitted in a cylindrical container. When the fuel level in the container starts decreasing, the timer starts and when float reaches lower limit, the timer stops so that time required for consumption of measured quantity of fuel can be measured. When float is at lower limit, supply signal is send to solenoid valve which opens and fuel starts flowing into the container. When the container is filled with fuel, float is at upper limit, supply for solenoid is cutoff as soon as fuel starts decreasing, timer stats again.

b) **Measuring instruments / Indicators / Switches**:

Load cell: The load cell is connected to the swinging field generator. It measures the load acting on generator due to torque generated. It is basically a strain-stress gauge.

Load indicator: The digital load indicator indicates the load acting on generator in kilograms (kg) measured by load cell connected to the dynamometer.

Inductive RPM sensor: The engine speed is measured by means of an inductive RPM sensor. The sensor sends a pulse when a metallic part passes in front of it. These pulses are sent to the indicator which in turns displays the engine speed.

RPM indicator: The RPM indicator displays engine speed in RPM measured by inductive sensor.

K-type thermocouples: ‘K’ type thermocouples suitable for high temperature measurements are used to measure the temperatures at different points. The temperatures measured by thermocouples are indicated on temperature indicator.

Temperature indicator: A multi channel temperature indicator shows the temperature recorded by the thermocouples at different points. Temperature reading at any specific point can be taken by turning selector switch to corresponding channel (1, 2, 3, … etc).

The sequence for temperatures on the indicator is as follows:-

T1	Exhaust gas inlet temperature
T2	Exhaust gas outlet temperature
T3	Calorimeter water inlet temperature
T4	Calorimeter outlet temperature
T5	Engine coolant inlet temperature
T6	Engine coolant outlet temperature
T7	Ambient temperature

Air Box: The air box is used to measure the amount of air intake by the engine. The air box is fitted with an orifice inlet. The pressure difference between atmospheric air and air inside the box is measured on 'differential pressure gauge' which can be used to determine air consumption of the engine. The air box acts as a damping reservoir to damp the air pulses.

Differential Pressure Gauge: the differential pressure gauge measures the pressure difference (inches of water column) between atmospheric air and inside air box.

Timer: Digital timer indicates the time required for consumption of measured quantity of fuel. Three separate timers are provided for each fuel tank according to the engine under test.

Torque reversal switch: The torque reversal switch is used while testing different type of engines with different direction of rotation since the direction of load applied and hence torque developed changes with change in direction of rotation. The switch should be in the position '1' for engines running in clockwise direction and should be in position '2' for those in anticlockwise direction.

4.1.4 **Load bank**: The load bank (capacity 8 kW) consists of resistive load (air heater). The load bank is provided with 16 air heaters of capacity 500 watts each. Engine can be loaded gradually by switching on the resisters separately one by one. Exhaust fans are provided for forced air circulation over heaters' surfaces.

4.1.5 **DC Generator control panel**: The control panel for generator has the following instruments:

a) Voltmeter: it is used to measure the supply voltage developed by the generator.

b) Ammeter: This is used to measure the current output of generator.

c) Excitation system: It is used to vary excitation voltage supplied to the generator. The generator output (in generator mode) and supply voltage for motor (in motor mode) can be changed by changing excitation voltage.

d) Mode selector switch: Mode selector switch is used to select mode of operation i.e. Motor or Generator mode.

e) Reference Potentiometer: The reference potentiometer is used to regulate motor speed.

4.1.6 Trolley mounted Exhaust gas calorimeter: To measure the heat carried away by exhaust gases, exhaust gas calorimeter is used. It is basically a shell and tube heat exchanger in which copper tubes are forged inside a MS cylindrical shell. The gas flows through tubes while water flows surrounding them in opposite direction, hence it is a counter flow type heat exchanger. Four k-type thermocouples measure temperatures of exhaust gas and water inlet and outlet. These thermocouples are connected to the temperature indicator for display of recorded temperatures. A water flow meter (Rotameter) is used to measure the flow-rate of water passing through calorimeter.

4.1.7 **Auxiliary cooling unit**: For water cooled engines. Coolant is supplied from an auxiliary cooling unit. It consists of a coolant storage tank, centrifugal pump to circulate the coolant through the engine and flow-meter (Rota-meter) to measure the coolant flow-rate. Two thermocouples measure the temperature of coolant entering and leaving the engine. The coolant is re-circulated from auxiliary cooling unit. There is a temperature controller attached in the circuit relay which is connected to a hooter. When the temperature of the water of the water in the tank equals the set temperature controller, the hooter is set of, which symbolizes that the water in the tank should be changed with water with at room temperature.

4.2 Engine Specifications:

Single cylinder, four stroke diesel engine (compression ignition) mounted on a moving frame is used for the conducted test. It is a variable speed, water cooled, direct injection engine and natural air breathed. The general specifications of the engine (Engineers, 2006) are:

Displacement = 499 cc

Bore (d) = 85 mm

Stroke = 87.8 mm

Make = Force Motors, India

Tests were taken at constant speed of 1940 RPM (recommended speed from manufacturer) under varying loads conditions with a fuel of normal diesel oil fuel and blends (Bio-0.0, Bio-25, Bio-50, Bio-75 and Bio-100) of jatropha bio-diesel in diesel oil fuel. The blends taken were in volume ratio, noted as Bio-0.0 for pure diesel, Bio-25 for 25% blend and so on up to Bio-100 for 100% bio-fuel. For each type of fuel blend, seven readings set were taken. The first and second of which were not considered in analysis for allowing the engine to attained stable condition.

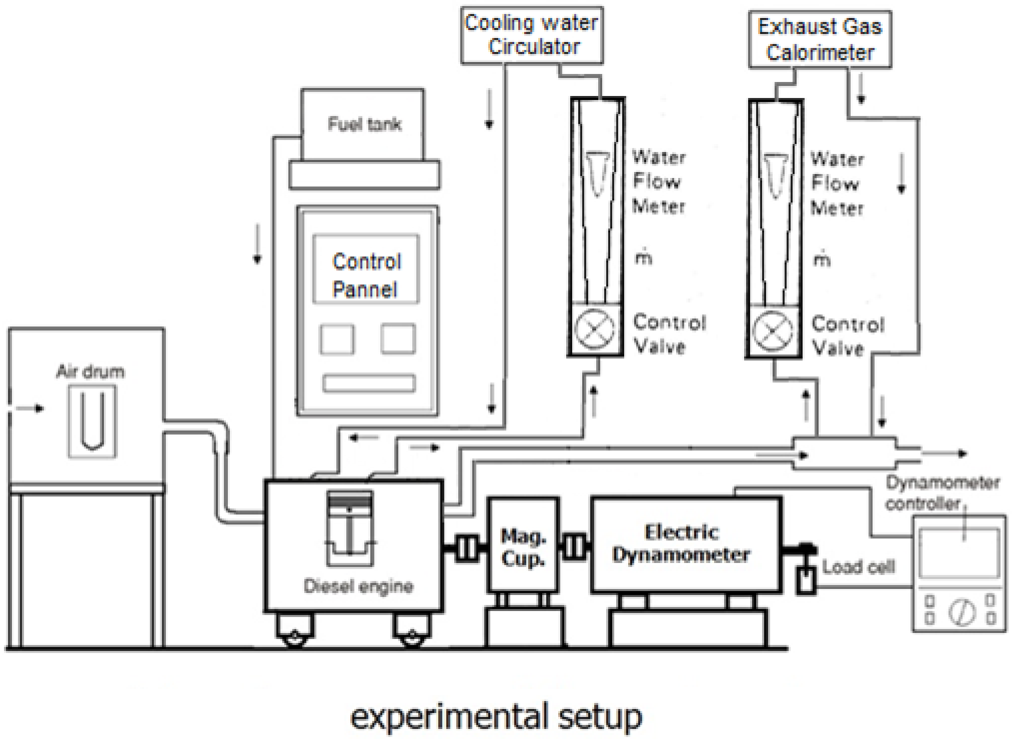


Figure (4.1) Test Rig layout

Tables of sample of readings taken and calculated results obtained are given in the following pages.

4.3 Performance Equations (Engineers, 2006):

Engine Brake Power (B.P):

$$\text{B.P.} = \frac{2\pi NT}{60\times1000}\ \text{kW} \qquad \text{---------------------------} \ (1)$$

B.P. = Brake Power (kW), N = Engine speed (RPM),

T = Shaft Torque (Nm)

$$\text{Or, B.P.} = \frac{V\times I\times10^{-3}}{\eta_{Gen.}},\ \eta_{Gen.} = 0.85 \ \text{---------------------------} \ (2)$$

Brake Mean Effective Pressure (B.M.E.P):

$$\text{B.M.E.P.} = \frac{B.P.}{100\times L\times A\times(N/120)}\ (\text{bar}) \ \text{-----------------------} \ (3)$$

B.P. = Brake Power (kW)

L = Stroke length (m) = 0.878 m

A = Area of engine bore (m^2) = 0.00567 m^2

N = Engine speed (RPM) = 2000 RPM

Frictional Power (kW):

$$\frac{Voltage(V) \times Current(I)}{1000} \times \eta_{Mot.}, \quad \text{-------------------------------------} \ (4)$$

$\eta_{Mot.}$ = 0.9, V = Voltage (Volts), I = Current (Amps),

η_{Mot} = Mechanical Efficiency of the dynamometer when runs as motor

Fuel Consumption Rate [Q_f], (kg/sec):

$$\dot{Q}_f = \frac{Volume\ (ml) \times \rho_{fuel}\,(kg\ /\ m^3)}{1000 \times 1000 \times t\ (\text{sec})} \quad kg\ /\ \text{sec} \text{--------------------------} \ (5)$$

ml = mille liter, ρ_{fuel} = fuel density

Brake Specific Fuel Consumption (B.S.F.C.), kg/(sec-kW),

$$\text{B.S.F.C.} = \frac{\dot{Q}_{fuel}\,(kg\ /\ \text{sec})}{B.P.(kW)} \quad kg\ /\ \text{sec} - kW \ \text{-----------------------------} \ (6)$$

Q_{fuel} = fuel consumption rate

Indicated Power (I.P.) = (B.P.) + (F.P), kW

Brake Thermal Efficiency (η_{th}):

$$\eta_{th} = \frac{B.P.}{\dot{Q}_{fuel} \times (C.V.)_{fuel}} \times 100 \ \text{--} \ (7)$$

$(CV)_{fuel}$ = Calorific Value of fuel

Mechanical Efficiency ($\eta_{mech.}$):

$$\eta_{mech.} = \frac{B.P.}{I.P.} \times 100 \ \text{---} \ (8)$$

Exhaust Calorimeter water flow rate,

$$m_{H2O,1} = \frac{\dot{V}_{Cal.}\ (LPH)}{3600 \times 1000} \times \rho_{H_2O} \quad kg\ /\ \text{sec} \text{------------------------} \ (9)$$

$m_{H2O,1}$ = water flow rate in flow-meter one,

Cooling water mass flow rate,

$$m_{H2O,2} = \frac{\dot{V}_{flow\ met.}\ (LPH)}{3600 \times 1000} \times \rho_{H_2O} \quad kg\ /\ \text{sec} \text{-------------------} \ (10)$$

$m_{H2O,2}$ = water flow rate in flow-meter two, Air intake Rate, m_{air} (kg/sec):

$$\dot{m}_{air} = \dot{V}_{air} \times \rho_{air} \text{ kg/sec},$$

$$\dot{V} = C_d \times A \times \sqrt{\frac{20 \times g \times h \times 2.54}{\rho_{air}}} \quad m^3/\sec, \text{ --------------- (11)}$$

C_d = Coefficient of discharge of the orifice, = 0.65,

A = Aria of the orifice, = $\frac{\pi}{4} d_o^{\,2} = 0.000314\, m^2$

A/F ratio = $\frac{\dot{m}_{air} \ kg/\sec}{\dot{m}_{fuel} \ kg/\sec}$ --- (12)

m_{air} measured from air drum flow metre device,

m_{fuel} measured from fuel consumed at given time,

Equivalence Ratio (ϕ):

$$\phi = \frac{(A/F)_s}{(A/F)_{act}} \text{ -------------------------------------- (13)}$$

Stochiometric Air/Fuel Ratio $(A/F)_s$:

For any hydro-carbon fuel of general chemical formula (C_xH_y)(Francis, 1965), where x= number of carbon atoms in the compound, y = number of hydrogen atoms in the compound,

$$(A/F)_s = \frac{137.9(x + \frac{y}{4})}{12x + y} \text{ --------------------------- (14)}$$

For diesel fuel, ($C_{12}H_{26}$),

$$(A/F)_s = \frac{137.9(12 + \frac{36}{4})}{(12 \times 12) + 26} = 15.007 \quad (kg_{Air}/kg_{Fuel}) \text{---- (15)}$$

For any hydro-carbon compound fuel containing oxygen, ($C_xH_yO_z$),

$$(A/F)_s = \frac{137.9}{(12x) + y} \cdot \frac{(4x + y - 2z)}{4} \quad (kg_{Air}/kg_{Fuel})$$

For Jatropha Bio-Diesel fuel, compound, ($C_{19}H_{36}O_2$)(Francis, 1965),

$$\left(A/F\right)_s = \frac{137.9}{\left(12\times19\right)+36}\cdot\frac{\left(4\times19\right)+36-\left(2\times2\right)}{4} = 14.103 \quad \left(kg_{Air}/kg_{Fuel}\right)$$

For any hydro-carbon liquid fuel blend,

$$\left(A/F\right)_s = 137.9\times\frac{\sum m_s `X`\times molar\ \ fraction\ 'X' - molar\ \ fraction\ O_2}{\sum\left(RMM\right) Fuel\ Component\times molar\ \ fraction\ Fuel\ Component} \quad (16)$$

m_s = stochiometric moles of oxygen per mole of fuel,

'X' = molar fraction of oxygen consuming component element,

RMM = Relative Molar Mass.

For diesel/jatropha bio-diesel blend, $(A/F)_s$;

$$(A/F)_{s,blend} = (A/F)_{s,diesel} \times (1\text{-}Bio) + (A/F)_{s,Bio} \times (Bio)$$

$$\left(A/F\right)_{s,blend} = \left\{\left(\left(A/F\right)_{s,diesel}\times\left(1-Bio\right)\right)+\left(\left(A/F\right)_{s,Biol}\times Bio\right)\right\} \text{--------------} \ (17)$$

For any blend (B_i),

$$(A/F)_{s,i} = [(15.007 \times (1\text{-} B_i)] + [(14.103 \times (B_i)] \text{(Francis, 1965)}$$

Volumetric Efficiency, ($\eta_{vol.}$);

$$\eta_{vol.} = \frac{\dot{V}_{act.}}{\dot{V}_{th.}} = \frac{\dot{V}\ \ (m^3/\sec)}{(L.A.N)/120\ (m^3/\sec)} \text{--------------------------} \ (18)$$

Heat Balance Equations:

i. Fuel intake power (kW):

Fuel Power = $\dot{m}_{fuel}\times(C.V.)_{fuel}$

ii. % heat utilized in (B.P.) = $\dfrac{B.P.(kW)}{Fuel.P.(kW)}\times100$ ------------------------- (19)

iii. Heat Carried away by Exhaust Gas =

$$Q_{Exhau} = \frac{(T_1-T_7)}{(T_1-T_2)}\times\dot{m}_{H_2O}\times C_{p_{H_2O}}\times(T_4-T_3)\ \ \text{(kW)} \text{----------} \ (20)$$

T_1 = Temperature of Exhaust Gas in (^oC),

T_2 = Temperature of Exhaust Gas out (^oC),

T_3 = Temperature of Calorimeter Water in (^oC),

T_4 = Temperature of Calorimeter Water out (^oC),

T_7 = Ambient Temperature (^oC),

iv. % Heat Carried by exhaust gas = $\frac{Q_{Exhaust} \quad (kW)}{Fuel.P \quad (kW)} \times 100$ -------------- (21)

v. Heat taken by cooling water, $Q_{cool} = \dot{m}_{H_2O_{Eng.}} \times C_{H_2O} \times (T_6 - T_5)$ -------(22)

T_5 = Temperature of Engine Cooling -Water, out (^oC),

T_6 = Temperature of Engine Cooling-Water, in (^oC),

vi. % of Heat taken by cooling water, $\frac{Q_{Cool} \quad (kW)}{Fuel.P \quad (kW)} \times 100$ ------------ (23)

vii. Heat unaccounted Loss =

Q_{other} = Fuel Power – [(B.P.) + (F.P.) +$Q_{Exhau.}$ + $Q_{cool.}$] ----- (24)

viii. % Unaccounted loss, = $\frac{Q_{other} \quad (kW)}{Fuel.P. \quad (kW)} \times 100$ ------------------------- (25)

Ignition Delay (τ) [ms]

$$\tau_{id} = \left\{ \left(0.0039 \times [\%B] + 1.7031\right) \times \phi^{\left[0.0072 \times (\%B) - 1.3916\right]} \right\} \text{------- (26)}$$

(El-Kasaby and Nemit-allah, 2013)

4.4 Fuel Blends preparing:

Jatropha bio-diesel is used as the bio-fuel, its blends in ordinary diesel oil were prepared, percentages were in volume bases. The investigated blends were B0, B25, B50, B75 and B100, where (B#) denotes bio-diesel fuel volume percentage in diesel oil. The jatropha biodiesel was prepared locally in Sudan from the local grown and harvested jatropha plant, specifications of which is given with reference to Petroleum Laboratories, Research and Studies, Sudan (PLRS, 2004).

4.5 Data Collection and Calculation Equations of Calculations

For data collection (observation tables 4.4 to 4.8), and using equations (1) to (24), criteria of performance and heat balance percentages were calculated.

For each of the blends, fuel specifications and properties are given in table (4.1) below.

Table (4.1) Blends specifications

Blend Designation	Blend Expressions				Density ρ kg/m^3	Calorific Value MJ/kg
	JBD %vol	JBD %wt	kg_Jat/kg Blend	kgDies/kg Blend		
Bio-0.0	0	0	0.000	1.000	804.00	43.70
Bio-25	25	28.6	0.286	0.714	837.15	43.52
Bio-50	50	57.2	0.572	0.428	870.29	43.35
Bio-75	75	85.8	0.858	0.142	903.44	43.17
Bio-100	100	100	1.000	0.000	919.90	43.09

Table (4.2) Summary of observation tables:

Observation Table No.	Blend Designation	JBD% In the blend
4.4	Bio-0.0	0
4.5	Bio-25	25
4.6	Bio-50	50
4.7	Bio-75	75
4.8	Bio-100	100

Table (4.3) observation table Bio0.0

Constant Speed Test [For Max. Torque], 1940 RPM, Direction of rotation …………CW, ……….ACW				
Observation Table No. (…[1]..), Date: 09/11/2017			**Bio-Fuel Blend ([B00])**	
Motoring Test,				
	Voltage (V) Volts	157	**F.P. (kW)**	$\rho_{BioZero}$ = 804 kg/m3 $Cv_{BioZero}$ = 43.7 MJ/kg
	Current (I) Amps	22.29	**3.15**	

S/No.	**Speed (N) RPM**	**Load (W) kg**	**Voltage (V) Volts**	**Current (I) Amps**	**Fuel Volume (Vf) cc**	**Time (t) Sec.**	**Pr. Diff (h) in H_2O**	**T_1**	**T_2**	**T_3**	**T_4**	**T_5**	**T_6**	**T_7**	**Cal_H_2O_Flow1 (Lit_P_Hr)**	**Cal_H_2O_Flow2 (Lit_P_Hr)**
1	1932	1.4	140	0.1	10	47.93	2.3	60	42	37	35	38	36	31	1460	200
2	1945	1.9	138	0.1	10	47.37	2.0	62	48	41	37	40	38	31	1410	200
3	1955	2.1	143	0.9	10	40.2	2.4	62	38	35	30	33	31	28	1410	200
4	1923	2.6	161	3.9	10	40.7	2.4	80	44	39	34	37	35	29	1410	200
5	1928	3.1	139	5.1	10	40.8	2.3	66	40	34	32	38	36	29	1410	200
6	1921	3.7	138	6.2	10	40.7	2.3	73	41	36	33	40	38	29	1410	200
7	1902	4.1	135	7.3	10	40.5	2.3	83	39	39	34	41	39	29	1410	200

* Flow1 = Engine Cooling Water, Flow2 = Exhaust Gas Calorimeter

Table (4.4) observation table Bio25

Constant Speed Test [For Max. Torque], 1940 RPM, Direction of rotation …………CW, ……….ACW

Observation Table No. ([2]), Date:12/11/2017 **Bio-Fuel Blend ([B 25%])**

Motoring Test,

Voltage (V) Volts	157	**F.P. (kW)**	
Current (I) Amps	22.29	**3.15**	

ρ_{Bio25} = 837.15 kJ/kg Cv_{Bio25} = 43.52 MJ/kg

S/No.	Speed (N) RPM	Load (W) kg	Voltage (V) Volts	Current (I) Amps	Fuel Volume (Vf) cc	Time (t) Sec.	Pr. Diff (h) in H_2O	T_1	T_2	T_3	T_4	T_5	T_6	T_7	Cal_H_2O_Flow1 (Lit_P_Hr)	Cal_H_2O_Flow2 (Lit_P_Hr)
1	1900	1	143	0.1	10	47	2.3	74	37	38	34	44	45	29	1460	130
2	1963	1.6	142	1.3	10	45.1	2.3	86	40	38	35	46	45	30	1410	130
3	1947	2.1	139	2.6	10	42.4	2.3	88	39	38	34	47	45	29	1410	120
4	1911	2.6	138	3.8	10	41.2	2.2	93	40	39	34	48	46	29	1410	120
5	1921	3.2	137	5	10	39.4	2.2	100	40	39	34	39	47	30	1410	120
6	1950	3.6	140	6.3	10	36.9	2.2	99	41	40	34	50	46	30	1410	120
7	1925	4.1	139	7.5	10	35.7	2.2	102	42	40	34	52	50	30	1410	120

* Flow1 = Engine Cooling Water, Flow2 = Exhaust Gas Calorimeter

Table (4.5) observation table Bio50

Constant Speed Test [For Max. Torque], 1940 RPM, Direction of rotation ....CW, ...ACW				
Observation Table No. ([3]), Date:12/11/2017			Bio-Fuel Blend ([B 50%])	
Motoring Test,				
Voltage (V) Volts	157	F.P. (kW)	ρ_{Bio50} = 870.29 kJ/kg	Cv_{Bio50} = 43.35 MJ/kg
Current (I) Amps	22.36	3.16		

S/No.	Speed (N) RPM	Load (W) kg	Voltage (V) Volts	Current (I) Amps	Fuel Volume (Vf) cc	Time (t) Sec.	Pr. Diff (h) in H_2O	T_1	T_2	T_3	T_4	T_5	T_6	T_7	Cal_H_2O_Flow1 (Lit_P_Hr)	Cal_H_2O_Flow2 (Lit_P_Hr)
1	1945	1.1	138	0.1	10	50.1	2.2	70	37	38	34	50	48	30	1410	110
2	1950	1.6	139	1.3	10	47.1	2.2	79	39	38	34	50	48	30	1410	110
3	1920	2.2	139	2.6	10	43.5	2.2	87	41	39	37	50	49	30	1410	110
4	1922	2.7	139	3.8	10	42.8	2.2	89	39	39	34	51	49	30	1410	110
5	1955	3.1	138	5.1	10	39.4	2.2	100	45	40	34	52	50	30	1410	110
6	1959	3.8	140	6.3	10	32.5	2.2	110	39	36	34	54	53	31	1410	120
7	1959	4.2	139	7.5	10	32.2	2.2	102	41	40	34	58	53	31	1410	120

* Flow1 = Engine Cooling Water, Flow2 = Exhaust Gas Calorimeter

Table (4.6) observation table Bio75

Constant Speed Test [For Max. Torque], 1940 RPM, Direction of rotation ....CW, ...ACW

Observation Table No. ([4]), Date:15/11/2017 **Bio-Fuel Blend ([B 75%])**

Motoring Test,

Voltage (V) Volts	157	**F.P. (kW)**	
Current (I) Amps	22.36	**3.16**	

ρ_{Bio75}= 903.44 kJ/kg CV_{Bio75} = 43.17 MJ/kg

S/No.	Speed (N) RPM	Load (W) kg	Voltage (V) Volts	Current (I) Amps	Fuel Volume (Vf) cc	Time (t) Sec.	Pr. Diff (h) in H_2O	T_1	T_2	T_3	T_4	T_5	T_6	T_7	Cal_H_2O_Flow1 (Lit_P_Hr)	Cal_H_2O_Flow2 (Lit_P_Hr)
1	1963	1.2	150	0.09	10	43.7	2.3	73	35	34	30	32	30	29	1410	110
2	1970	1.9	141	1.3	10	39.3	2.3	75	43	47	36	36	35	30	1410	110
3	1944	2.4	139	2.6	10	39.3	2.3	81	44	52	37	37	35	30	1410	110
4	1924	2.9	139	3.8	10	38.4	2.3	90	47	58	38	38	36	30	1410	110
5	1941	3.5	138	5.8	10	37.5	2.3	96	58	68	40	39	37	31	1410	110
6	1971	4.1	142	6.4	10	34.9	2.3	101	57	57	45	40	38	31	1410	110
7	1934	4.4	140	7.6	10	43.4	2.3	122	66	60	46	41	39	31	1410	110

* Flow1 = Engine Cooling Water, Flow2 = Exhaust Gas Calorimeter

Table (4.7) observation table Bio100

Constant Speed Test [For Max. Torque], 1940 RPM, Direction of rotation …………CW, ……….ACW

Observation Table No. (…[5]..), Date: 15/11/2017 **Bio-Fuel Blend ([B 100%])**

Motoring Test,

Voltage (V) Volts	157	**F.P. (kW)**	
Current (I) Amps	22.36	**3.16**	

ρ_{Bio100} =919.9kJ/kg CV_{Bio100}= 43.088 MJ/kg

S/No.	Speed (N) RPM	Load (W) kg	Voltage (V) Volts	Current (I) Amps	Fuel Volume (Vf) cc	Time (t) Sec.	Pr. Diff (h) in H_2O	T_1	T_2	T_3	T_4	T_5	T_6	T_7	Cal_H_2O_Flow1 (Lit_P_Hr)	Cal_H_2O_Flow2 (Lit_P_Hr)
1	1909	1.3	142	0.1	10	45.6	2.3	71	57	57	49	43	41	31	1460	110
2	1961	2.0	140	1.3	10	44.8	2.3	81	60	56	46	46	42	31	1410	110
3	1954	2.5	140	2.6	10	41.5	2.3	86	62	56	45	44	42	31	1410	110
4	1963	3.0	142	3.9	10	38.9	2.3	93	64	73	44	45	43	32	1410	110
5	1944	3.5	139	5.1	10	37.8	2.3	97	68	74	46	46	44	32	1410	110
6	1940	4.0	139	6.3	10	36.4	2.3	102	68	78	50	47	47	32	1410	110
7	1928	4.4	137	7.3	10	35.3	2.3	104	71	85	54	48	46	32	1410	110

* Flow1 = Engine Cooling Water, Flow2 = Exhaust Gas Calorimeter

CHAPTER V

RESULTS AND DISCUSSION

CHAPTER V

RESULTS AND DISCUSSION

5.1 Results

Using equations (1) to (26), the observations taken (tables 4.4 to 4.8) were analyzed. Results obtained in tables 5.2 to 5.11 analyzed and presented in plotted diagrams. Performance results give numerical calculated values.

Brake power (BP), values given in column (2) in all performance tables. BP was calculated using equation (1). Brake mean effective pressure (bmep), values given in column (5) in all performance tables, (bmep) was calculated using equation (3). Indicated power (IP), values given in column (6) in all performance tables, (IP) was calculated with reference to equations (1) and (4). Volumetric efficiency, values given in column (8) in all performance tables, (η_{vol}), this was calculated using equation (18). Thermal efficiency (η_{th}) values given in column (9) in all performance tables, calculated using equation (7). Air fuel ratio (A/F), values given in column (10) in all performance tables, A/F is calculated using equation (12) with reference to equations (5) and (11). Frictional power (FP), values given in column (11) in all performance tables, (FP) calculated from equation (4). Equivalence ratio (ϕ), values given in column (12) in all performance tables, (ϕ) is calculated using equation (13) with reference to equations (12) and (17) and a final performance criteria calculated is delay period (τ), values given in column (13) in all performance tables, it is calculated using equation (26).

Heat balance analysis has also worked out for each of the blends, where fuel power, percentage of heat utilized in (BP), values given in column (4) in all heat balance tables, (BP%) is calculated using equation (19). Percentage of heat carried away by exhaust gases, values given in column (5) in all heat balance tables, it is calculated using equation (21). Percentage of heat lost as mechanical waste, values given in column (6) in all heat balance tables, it is calculated using equation (20). Percentage of heat lost in engine coolant, values given in column (7) in all heat balance tables, it is calculated using equation (25) and other unaccounted losses such as; (radiation lost, sound, vibration …etc) are also given values given in column (8) in all heat balance tables, it is calculated using equation (25).

Table (5.1) Summary of performance results table**:**

S. No.	Blend Designation	Performance Table Number	Heat Balance Table Number
1	Bio-0.0	5.2	5.3
2	Bio-25	5.4	5.5
3	Bio-50	5.6	5.7
4	Bio-75	5.8	5.9
5	Bio-100	5.10	5.11

Table (5.2) Performance table Bio0.0

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
S/No.	**Torque (T), (N-m)**	**BP (kW)**	**SFC (Brake) kg/kW_hr**	**bmep (bar)**	**I.P (kW)**	η_{Mec} **%**	η_{Vol} **%**	η_{Th} **%**	**Air-Fuel Ratio**	**F. P. (kW)**	**ϕ**	τ (ms)
1	5.54	1.05	0.68	1.32	4.22	3.59	80.19	1.73	32.52	3.16	0.46	5.00
2	6.77	1.28	0.55	1.64	4.43	16.67	81.52	8.56	32.92	3.15	0.46	5.09
3	8.04	1.54	0.46	1.96	4.69	17.80	79.60	9.68	32.31	3.15	0.46	4.95
4	9.42	1.83	0.39	2.33	4.97	20.25	79.89	11.66	32.23	3.15	0.47	4.94
5	10.40	2.00	0.36	2.59	5.14	22.56	80.69	13.36	32.07	3.14	0.47	4.90

Performance analysis of B0.0 calculated from observation table 4.4

Table (5.3) Heat Balance Bio0.0p

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
S/No.	**Torque T, (Nm)**	**Fuel intake power (kW)**	**%Heat utilized for BP**	**%Heat carried away by Exhaust Gas**	**% Heat in Mechanical waste**	**%Heat Carried away by Cooling water**	**% Heat Unaccounted**
1	5.54	8.74	12.07	18.85	36.21	5.32	27.55
2	6.77	8.63	14.88	19.09	36.45	5.38	24.20
3	8.04	8.61	17.83	7.69	36.58	5.42	32.49
4	9.42	8.63	21.15	11.11	36.44	5.22	26.07
5	10.40	8.68	23.10	16.45	36.14	4.64	19.67

Heat balance analysis for B0.0 calculated from observation table 4.4

Table (5.4) Performance table Bio25

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
S/No.	**Torque (T), (N-m)**	**BP (kW)**	**SFC (Brake) kg/kW_hr**	**bmep (bar)**	**I.P (kW)**	**η_{Mec} %**	**η_{Vol} %**	**η_{Th} %**	**Air-Fuel Ratio**	**F. P. (kW)**	**ϕ**	τ (ms)
1	5.54	1.05	0.68	1.32	4.21	24.95	78.82	12.73	32.25	3.16	0.46	5.06
2	6.77	1.28	0.57	1.64	4.42	28.90	78.54	15.03	30.64	3.14	0.48	4.71
3	8.04	1.58	0.48	2.02	4.73	33.43	78.13	17.78	29.30	3.15	0.50	4.43
4	9.42	1.80	0.45	2.27	4.97	36.32	76.97	19.02	27.45	3.16	0.54	4.04
5	10.40	2.03	0.42	2.59	5.18	39.18	77.97	20.69	26.55	3.15	0.56	3.86

Performance analysis of B25 calculated from observation table 4.5

Table (5.5) Heat Balance Bio25

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
S/No.	**Torque T, (Nm)**	**Fuel intake power (kW)**	**%Heat utilized for BP**	**%Heat carried away by Exhaust Gas**	**% Heat in Mechanical waste**	**%Heat Carried away by Cooling water**	**% Heat Unaccounted**
1	5.54	8.25	12.73	13.58	38.30	4.79	30.61
2	6.77	8.49	15.03	16.54	36.97	4.66	26.79
3	8.04	8.88	17.78	15.28	35.42	17.23	48.75
4	9.42	9.48	19.02	17.51	33.35	8.23	21.89
5	10.40	9.80	20.69	17.09	32.12	4.02	26.09

Heat balance analysis for B25 calculated from observation table 4.5

Table (5.6) Performance table Bio50

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
S/No.	**Torque (T), (N-m)**	**BP (kW)**	**SFC (Brake) kg/kW_hr**	**bmep (bar)**	**I.P (kW)**	**η_{Mec} %**	**η_{Vol} %**	**η_{Th} %**	**Air-Fuel Ratio**	**F. P. (kW)**	**ϕ**	τ (ms)
1	5.54	1.09	0.66	1.39	4.23	78.17	78.17	13.61	32.35	3.15	0.45	5.20
2	6.77	1.33	0.55	1.70	4.48	78.09	78.09	16.45	31.83	3.15	0.46	5.08
3	8.04	1.56	0.51	1.96	4.72	76.77	76.77	17.69	29.30	3.16	0.49	4.53
4	9.42	1.91	0.50	2.40	5.08	76.62	76.62	17.92	24.17	3.17	0.60	3.47
5	10.40	2.11	0.46	2.65	5.28	76.62	76.62	19.63	23.95	3.17	0.60	3.43

Performance analysis of B50 calculated from observation table 4.6

Table (5.7) Heat Balance Bio50

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
S/No.	**Torque T, (Nm)**	**Fuel intake power (kW)**	**%Heat utilized for BP**	**%Heat carried away by Exhaust Gas**	**% Heat in Mechanical waste**	**%Heat Carried away by Cooling water**	**% Heat Unaccounted**
1	5.54	9.04	12.01	6.38	34.80	2.25	44.57
2	6.77	9.19	14.51	14.94	34.25	4.21	32.08
3	8.04	9.98	15.60	17.80	31.71	4.18	30.70
4	9.42	12.10	15.81	4.28	26.18	1.51	52.22
5	10.40	12.21	17.31	13.30	25.94	7.82	35.63

Heat balance analysis for B50 calculated from observation table 4.6

Table (5.8) Performance table Bio75

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
S/No.	**Torque (T), (N-m)**	**BP (kW)**	**SFC (Brake) kg/kW_hr**	**bmep (bar)**	**I.P (kW)**	**η_{Mec} %**	**η_{Vol} %**	**η_{Th} %**	**Air-Fuel Ratio**	**F. P. (kW)**	**ϕ**	**τ (ms)**
1	5.54	1.20	0.69	1.51	4.36	78.94	78.94	13.57	29.89	3.16	0.48	4.77
2	6.77	1.43	0.59	1.83	4.58	79.76	79.76	15.86	29.20	3.15	0.49	4.62
3	8.04	1.75	0.50	2.21	4.90	79.06	79.06	18.86	28.52	3.16	0.50	4.47
4	9.42	2.08	0.45	2.59	5.25	77.86	77.86	20.88	26.54	3.17	0.54	4.05
5	10.40	2.19	0.34	2.78	5.34	79.35	79.35	27.34	33.01	3.15	0.43	5.48

Performance analysis of B75 calculated from observation table 4.7

Table (5.9) Heat Balance Bio75

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
S/No.	**Torque T, (Nm)**	**Fuel intake power (kW)**	**%Heat utilized for BP**	**%Heat carried away by Exhaust Gas**	**% Heat in Mechanical waste**	**%Heat Carried away by Cooling water**	**% Heat Unaccounted**
1	5.54	10.00	11.98	48.07	31.57	4.52	51.93
2	6.77	10.24	14.00	63.40	30.74	4.47	50.79
3	8.04	10.48	16.65	65.00	30.11	5.35	47.89
4	9.42	11.27	18.43	39.42	28.17	4.63	48.77
5	10.40	9.06	24.13	58.41	34.80	5.88	35.18

Heat balance analysis for B75 calculated from observation table 4.7

Table (5.10) Performance table Bio100

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
S/No.	**Torque (T), (N-m)**	**BP (kW)**	**SFC (Brake) kg/kW_hr**	**bmep (bar)**	**I.P (kW)**	**η_{Mec} %**	**η_{Vol} %**	**η_{Th} %**	**Air-Fuel Ratio**	**F. P. (kW)**	**ϕ**	τ (ms)
1	5.54	1.26	0.64	1.58	4.42	78.54	78.54	15.76	31.56	3.16	0.45	5.21
2	6.77	1.51	0.56	1.89	4.68	78.18	78.18	18.69	29.58	3.17	0.48	4.76
3	8.04	1.75	0.50	2.21	4.91	78.94	78.94	19.88	28.75	3.16	0.49	4.57
4	9.42	1.99	0.46	2.52	5.15	79.10	79.10	18.70	27.68	3.16	0.51	4.34
5	10.40	2.18	0.43	2.78	5.33	79.60	79.60	20.26	26.85	3.15	0.53	4.16

Performance analysis of B100 calculated from observation table 4.8

Table (5.11) Heat Balance Bio100

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
S/No.	**Torque T, (Nm)**	**Fuel intake power (kW)**	**%Heat utilized for BP**	**%Heat carried away by Exhaust Gas**	**% Heat in Mechanical waste**	**%Heat Carried away by Cooling water**	**% Heat Unaccounted**
1	5.54	9.38	13.38	62.51	33.73	8.01	44.88
2	6.77	10.01	15.12	52.00	31.67	6.89	46.32
3	8.04	10.30	16.97	50.00	30.67	7.14	45.22
4	9.42	10.69	18.64	55.00	29.51	30.00	51.84
5	10.40	11.03	19.76	53.00	28.56	6.49	45.19

Heat balance analysis for B100 calculated from observation table 4.8

5.2- **Discussion:**

From the tables of calculated results and the graphs obtained, the following general observations were found.

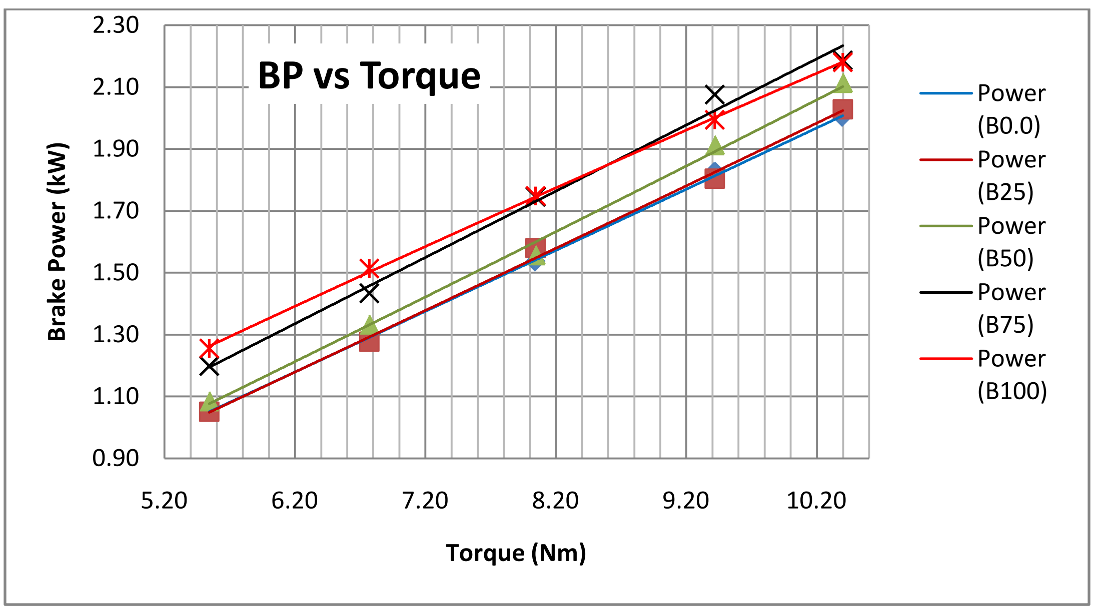
5.2.1. **Brake power**:

Brake power of the engine is generally increasing with the applied torque for all the blends, with higher values at low torque condition for Bio-0.0 and higher close to maximum torque for Bio-75. (Figure 5.1).

Table (5.12) Compiled results of brake power

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Brake Power (kW)				
1	5.54	1.05	1.05	1.09	1.20	1.26
2	6.77	1.28	1.28	1.33	1.43	1.51
3	8.04	1.54	1.58	1.56	1.75	1.75
4	9.42	1.83	1.80	1.91	2.08	1.99
5	10.40	2.00	2.03	2.11	2.19	2.18

Compiled from column (3) in all performance result tables



(Figure 5.1): Comparisons of brake power a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

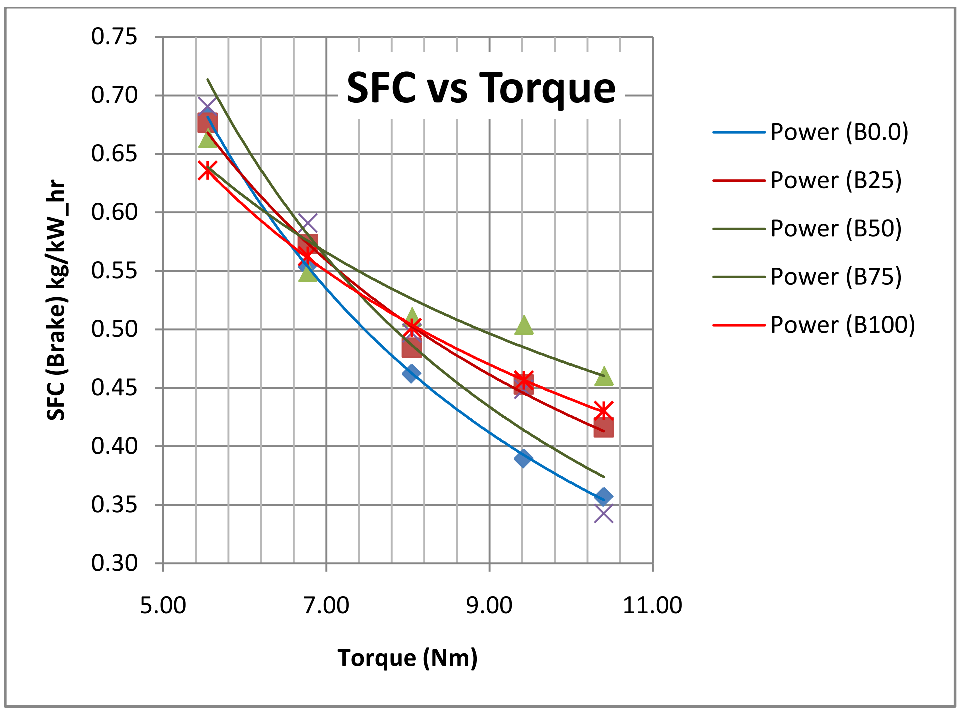
5.2.2. **Specific fuel consumption:**

It is high of all blends at low torque condition down towards the maximum torque with no significant variations at low torque, but with higher value for Bio-75 towards maximum torque. (Figure 5.2).

Table (5.13) Compiled results of SFC

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Specific Fuel Consumption (kg/kW hr)				
1	5.54	0.68	0.68	0.66	0.69	0.64
2	6.77	0.55	0.57	0.55	0.59	0.56
3	8.04	0.46	0.48	0.51	0.50	0.50
4	9.42	0.39	0.45	0.50	0.45	0.46
5	10.40	0.36	0.42	0.46	0.34	0.43

Compiled from column (4) in all performance result tables



(Figure 5.2): SFC as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

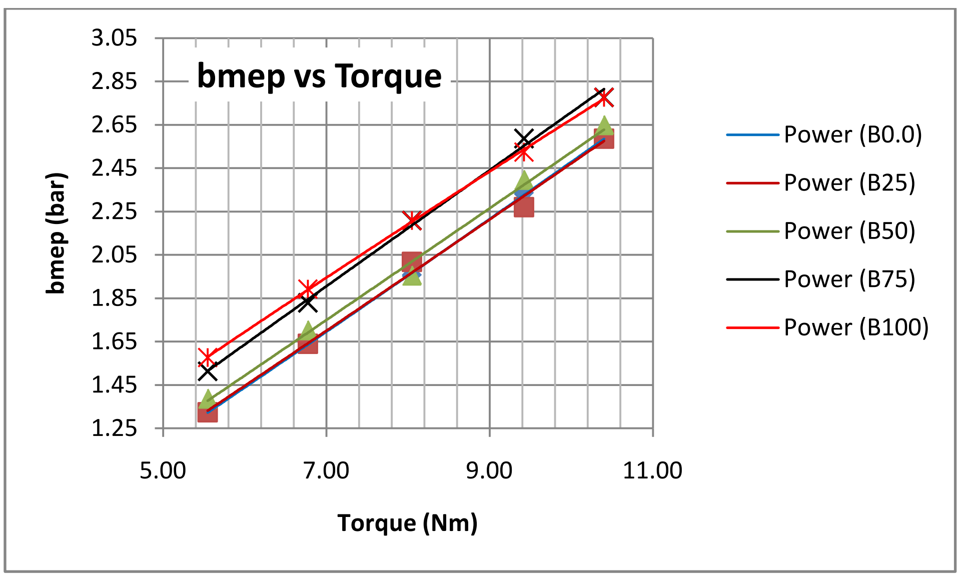
5.2.3. **Brake mean effective pressure (bmep):**

Brake mean effective pressure curves are looking similar to those of brake power curves, which in a normal situation. A maximum value of about 2.7 bar was attained with pure diesel fuel as well as with Bio-75 blend under applied torque of 10.4 Nm for the running test speed. (Figure 5.3).

Table (5.14) Compiled results of bmep

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Brake Mean Effective Pressure (bar)				
1	5.54	1.32	1.32	1.39	1.51	1.58
2	6.77	1.64	1.64	1.70	1.83	1.89
3	8.04	1.96	2.02	1.96	2.21	2.21
4	9.42	2.33	2.27	2.40	2.59	2.52
5	10.40	2.59	2.59	2.65	2.78	2.78

Compiled from column (5) in all performance result tables



(Figure 5.3): bmep as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

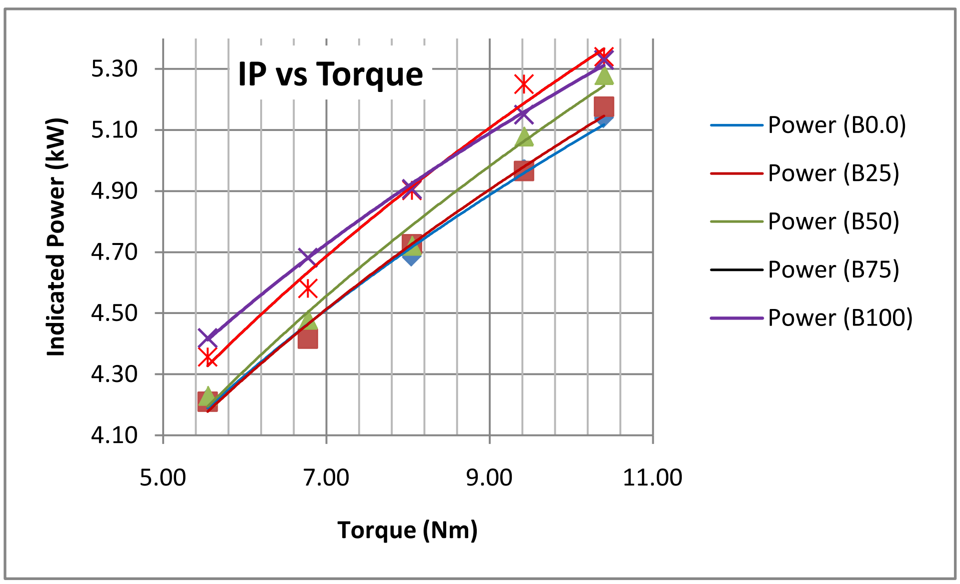
5.2.3. **Indicated power:**

Indicated power of the engine is greater than that of pure diesel for all bio-blends, with Bio-100 above all, this may be for that frictional loss is higher for jatropha bio-diesel than that of pure diesel for its higher viscosity. (Figure 5.4).

Table (5.15) Compiled results of indicated power

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Indicated Power (kW)				
1	5.54	4.22	4.21	4.23	4.36	4.42
2	6.77	4.43	4.42	4.48	4.58	4.68
3	8.04	4.69	4.73	4.72	4.90	4.91
4	9.42	4.97	4.97	5.08	5.25	5.15
5	10.40	5.14	5.18	5.28	5.34	5.33

Compiled from column (6) in all performance result tables



(Figure 5.4): Indicated Power as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

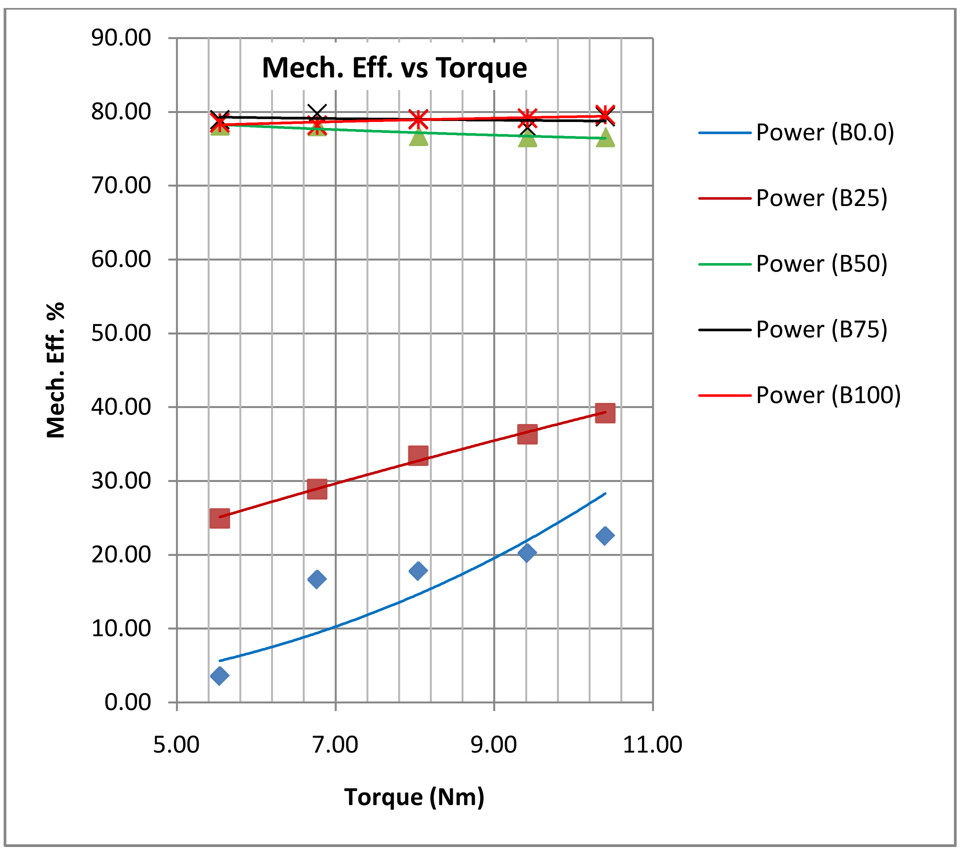
5.2.4. **Mechanical efficiency:**

Mechanical efficiency of the engine is low generally, for higher blends (Bio-100) and for all other blends close to that of pure diesel oil, with a value around 80% (Figure 5.5).

Table (5.16) Compiled results of mechanical efficiency

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Mechanical Efficiency				
1	5.54	3.59	24.95	78.17	78.94	78.54
2	6.77	16.67	28.90	78.09	79.76	78.18
3	8.04	17.80	33.43	76.77	79.06	78.94
4	9.42	20.25	36.32	76.62	77.86	79.10
5	10.40	22.56	39.18	76.62	79.35	79.60

Compiled from column (7) in all performance result tables



(Figure 5.5): Mechanical efficiency as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

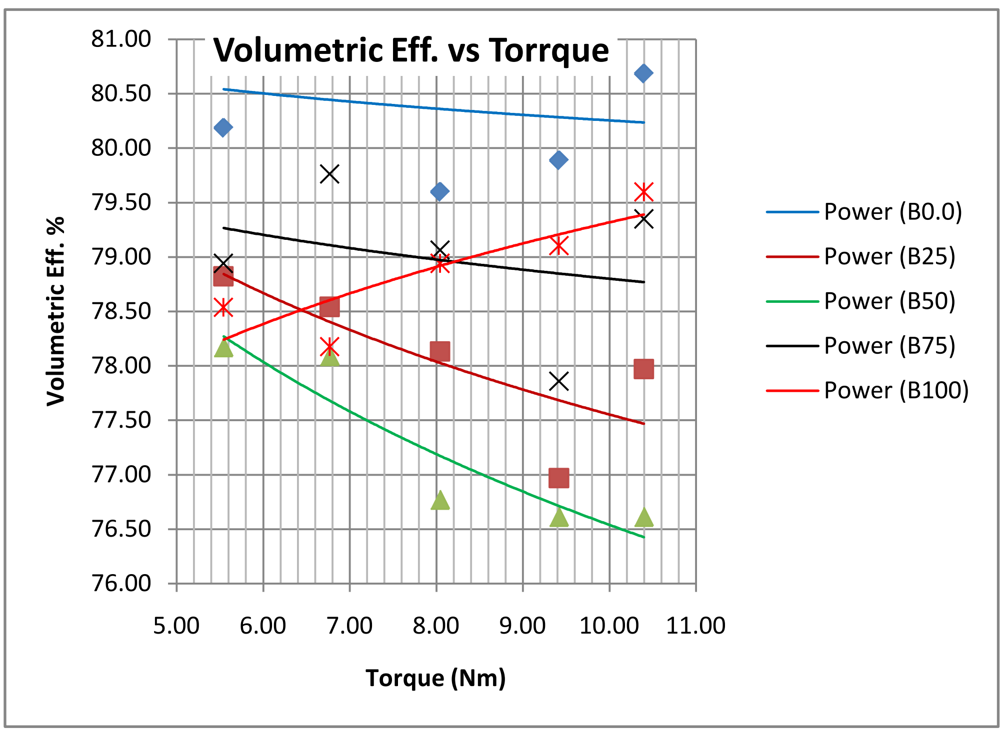
5.2.5. V**olumetric efficiency:**

The **volumetric efficiency** of the engine under test conditions (with natural air charge) gives values ranging from (76% - 81%) with no significant variations for the different blends under the various loading conditions but, for pure diesel oil fuel, it is slightly higher. (Figure 5.6).

Table (5.17) Compiled results of volumetric efficiency

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Volumetric Efficiency				
1	5.54	80.19	78.82	78.17	78.94	78.54
2	6.77	81.52	78.54	78.09	79.76	78.18
3	8.04	79.60	78.13	76.77	79.06	78.94
4	9.42	79.89	76.97	76.62	77.86	79.10
5	10.40	80.69	77.97	76.62	79.35	79.60

Compiled from column (7) in all performance result tables



(Figure 5.6): Volumetric efficiency as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

5.2.6. **Equivalence ratio (ϕ):**

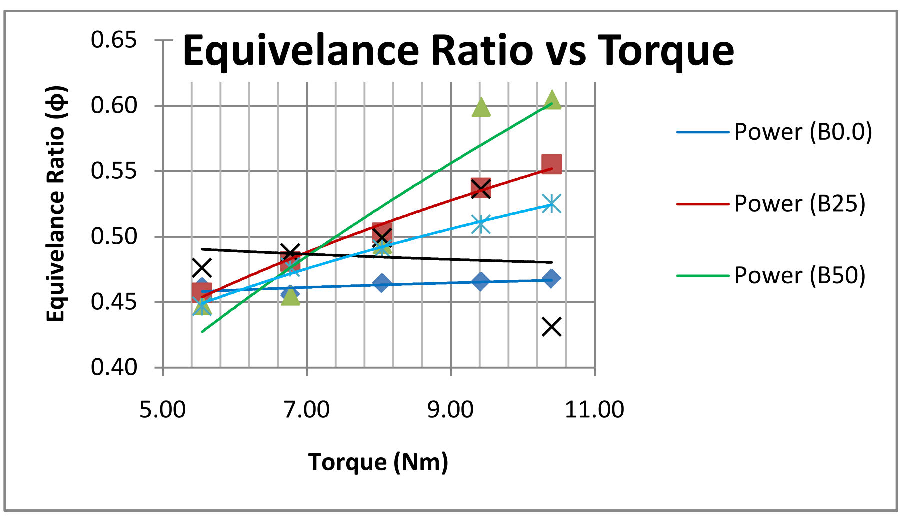
Equivalence ratio (ϕ) of the engine under test, which is the theoretical air fuel ratio (A/F)s to actual air fuel ratio (A/F) of a given condition, the recorded values for (ϕ) are all less than unity (1.0) which mean that, the air fuel mixtures are all lean. The best of them is for (ϕ) equals 0.6 for Bio-50 with maximum torque.

For other blends under all loading conditions, (ϕ) varies between 0.45 to 0.55. The test device is not equipped with means for (A/F) control, though the air taken is natural breathing method, still (ϕ) is less than one. It was observed that equivalence ratio is generally increasing with applied torque, since for doing this under constant speed, much fuel should be injected. (ϕ) values are better for Bio-75 at low loads and for Bio-50 close to maximum torque applied. (Figure 5.9).

Table (5.18) Compiled results of equivalence ratio

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Equivalence Ratio				
1	5.54	0.46	0.46	0.45	0.48	0.45
2	6.77	0.46	0.48	0.46	0.49	0.48
3	8.04	0.46	0.50	0.49	0.50	0.49
4	9.42	0.47	0.54	0.60	0.54	0.51
5	10.40	0.47	0.56	0.60	0.43	0.53

Compiled from column (12) in all performance result tables



(Figure 5.7): Equivalence ratio (ϕ) as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

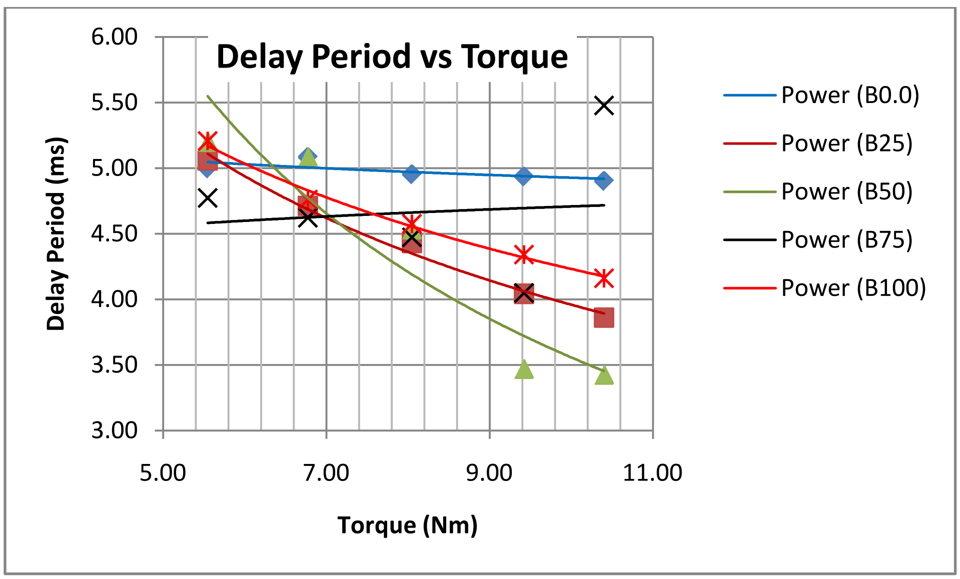
5.2.7. **Ignition delay:**

Ignition delay period was calculated using the empirical equation (16), the equation that links (ϕ) with blend percentage (%Bio). This makes it possible to trace the effect of loading conditions effect for each of the blends on delay of ignition. Delay time in milli-seconds (ms) found to be in range 3.5 ms as a minimum value with Bio-50 close to maximum applied torque to a value of 5.2 ms for pure jatropha bio-diesel oil fuel i.e. Bio-100 at low torque condition. Generally, delay time is less with high applied loads than that with low loads, with no significant variations between the blends. (Figure 5.8).

Table (5.19) Compiled results of delay period

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Delay Period (ms)				
1	5.54	5.00	5.06	5.20	4.77	5.21
2	6.77	5.09	4.71	5.08	4.62	4.76
3	8.04	4.95	4.43	4.53	4.47	4.57
4	9.42	4.94	4.04	3.47	4.05	4.34
5	10.40	4.90	3.86	3.43	5.48	4.16

Compiled from column (13) in all performance result tables



(Figure 5.8): Ignition delay as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

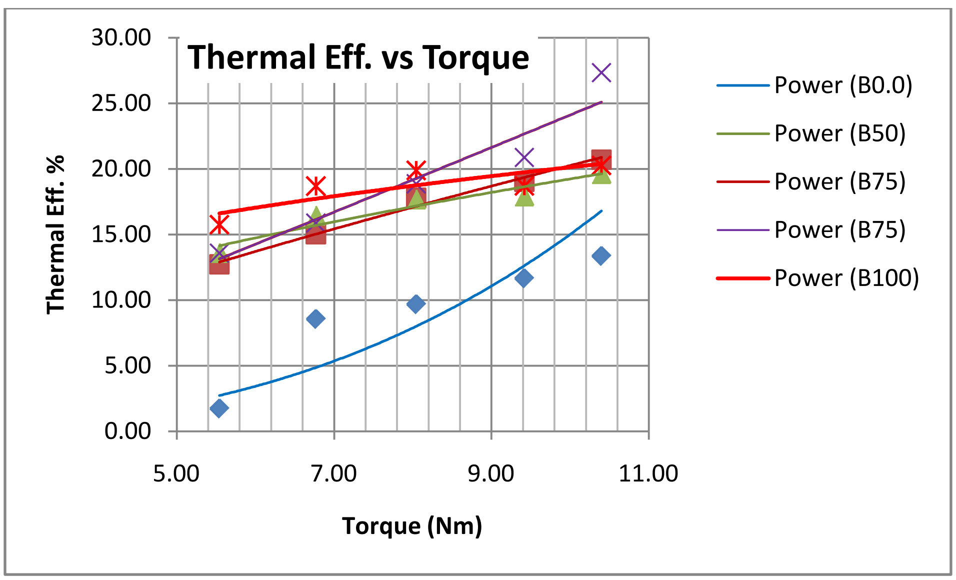
5.2.8. **Thermal efficiency:**

Thermal efficiency of the engine, tends to increase for all blends with increasing loads, all bio blends have less thermal efficiency than that of pure diesel oil fuel, except for that of Bio-75 at maximum applied load. (Figure 5.9)

Table (5.20) Compiled results of thermal efficiency

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Thermal Efficiency				
1	5.54	1.73	12.73	13.61	13.57	15.76
2	6.77	8.56	15.03	16.45	15.86	18.69
3	8.04	9.68	17.78	17.69	18.86	19.88
4	9.42	11.66	19.02	17.92	20.88	18.70
5	10.40	13.36	20.69	19.63	27.34	20.26

Compiled from column (9) in all performance result tables



(Figure 5.9): Thermal efficiency as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

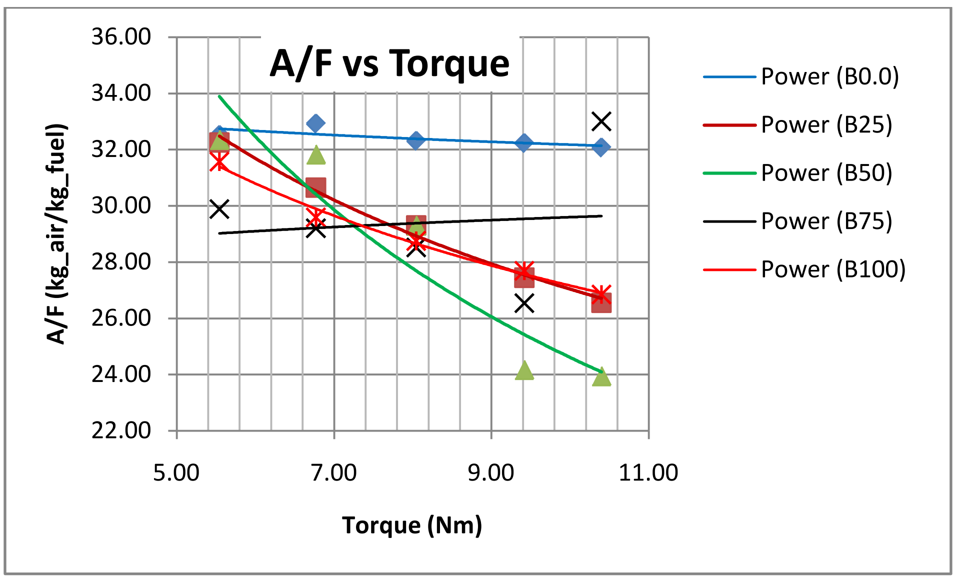
5.2.9. **Air/fuel ratio:**

Air/fuel ratio for the engine under test is all higher than that required for theoretical complete combustion according to the chemical composition of the fuel with its different blends; the values recorded were in range 24 kg_{air}/kg_{fuel} up to thirty three kg_{air}/kg_{fuel} compared to a value of 15 kg_{air}/kg_{fuel} for pure diesel oil fuel and 14 kg_{air}/kg_{fuel} for pure jatropha bio-diesel. This shows clearly that equivalence ratio is less than one and that the charge is lean (with excess amount of air). Bio-0.0 (pure diesel oil fuel) is the best for this aspect at all loading conditions. All other blends have (A/F) values close to that of Bio-0.0 at low loads the fall down with greater loads for better value of (ϕ). (Figure 5.10).

Table (5.21) Compiled results of A/F ratio

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Air-Fuel Ratio				
1	5.54	32.52	32.25	32.35	29.89	31.56
2	6.77	32.92	30.64	31.83	29.20	29.58
3	8.04	32.31	29.30	29.30	28.52	28.75
4	9.42	32.23	27.45	24.17	26.54	27.68
5	10.40	32.07	26.55	23.95	33.01	26.85

Compiled from column (10) in all performance result tables

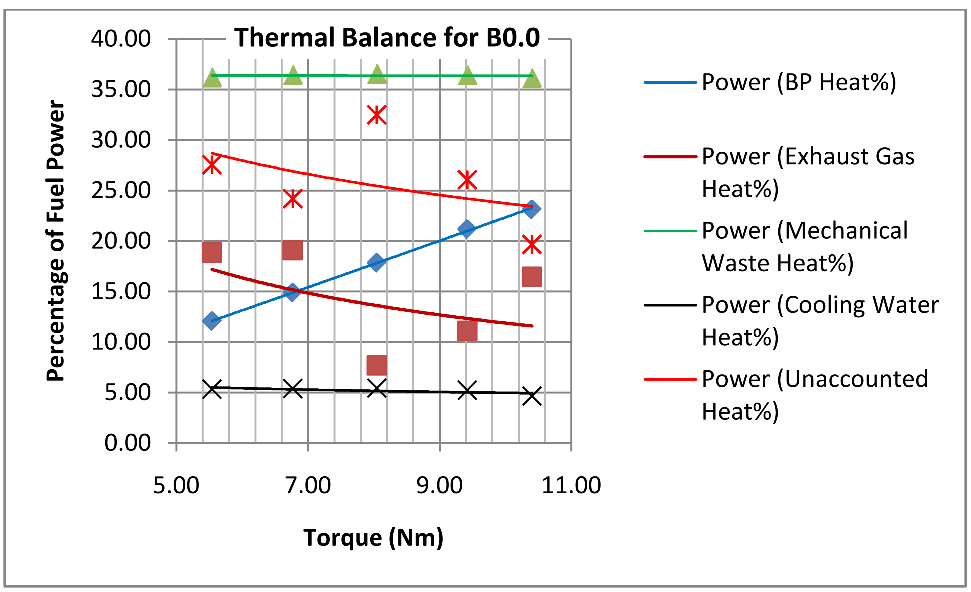


(Figure 5.10): Air-Fuel ratio (A/F) as a function of applied torque at a constant speed test condition for blends of Jatropha-bio diesel in diesel fuel.

5.2.10. **Heat balance analysis:**

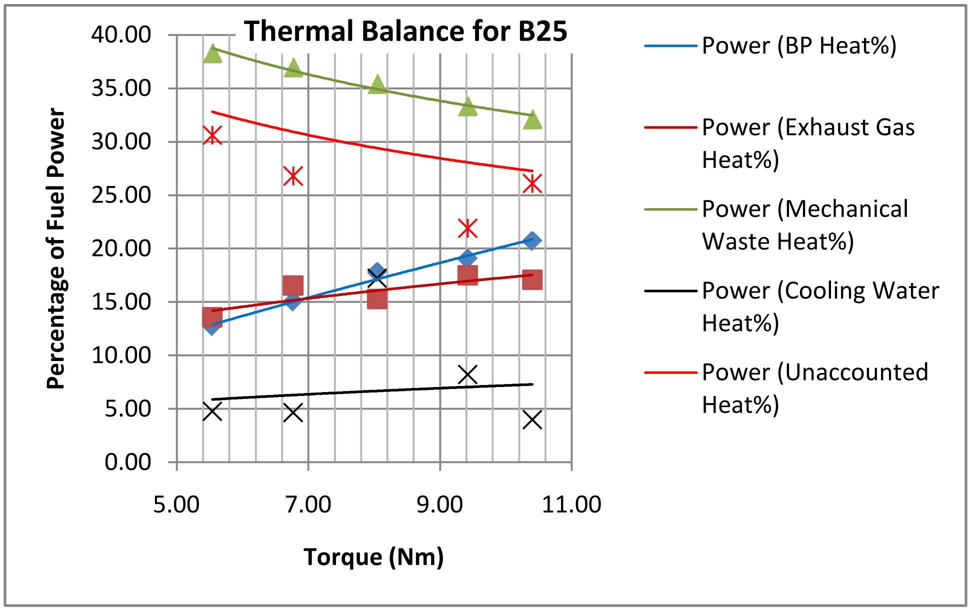
Heat balance was considered for the engine and charts were plotted first for each blend separately with group of elements of heat distribution percentage arranged in sequence of (brake power, exhaust gas, mechanical waste, cooling water and unaccounted heat loss). Figure (5.11) for Bio-0.0, Figure (5.12) for Bio-25, Figure (5.13) for Bio-50, Figure (5.14) for Bio-75 and Figure (5.15) for Bio-100. The last two charts are without columns for exhaust gas power loss for odd readings from the gas calorimeter. Any set of reading are giver in the bar chart for the given applied torque for any specific blend. It can be observed easily that mechanical loss for any blend chart is of a constant value under all loading conditions since the test is a constant speed test.

With reference to table (5.3)



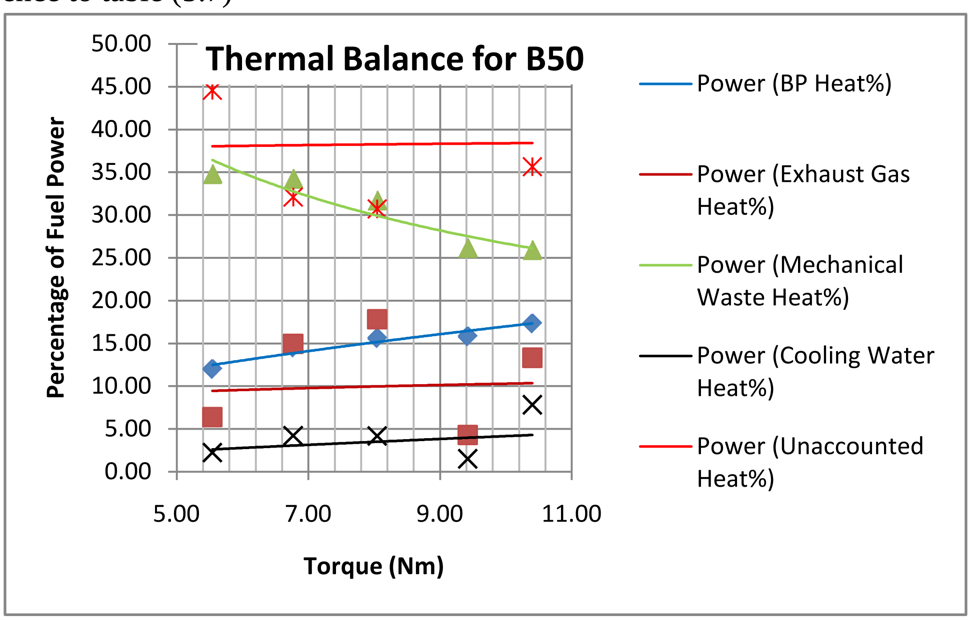
(Figure 5.11): Heat balance diagram against applied torque at a constant speed test condition for Bio-0.0 fuel.

With reference to table (5.5)



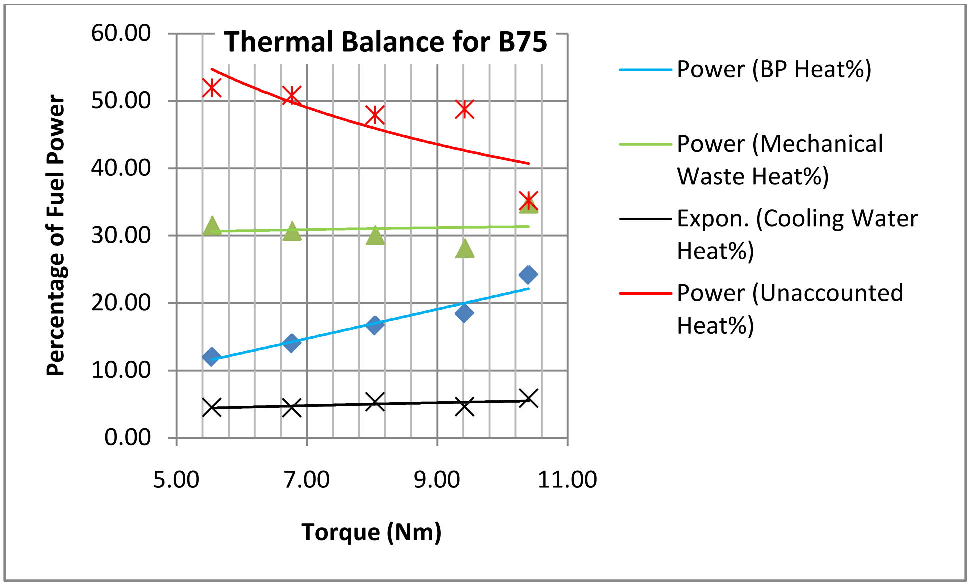
(Figure 5.12): Heat balance diagram against applied torque at a constant speed test condition for Bio-25 fuel.

With reference to table (5.7)



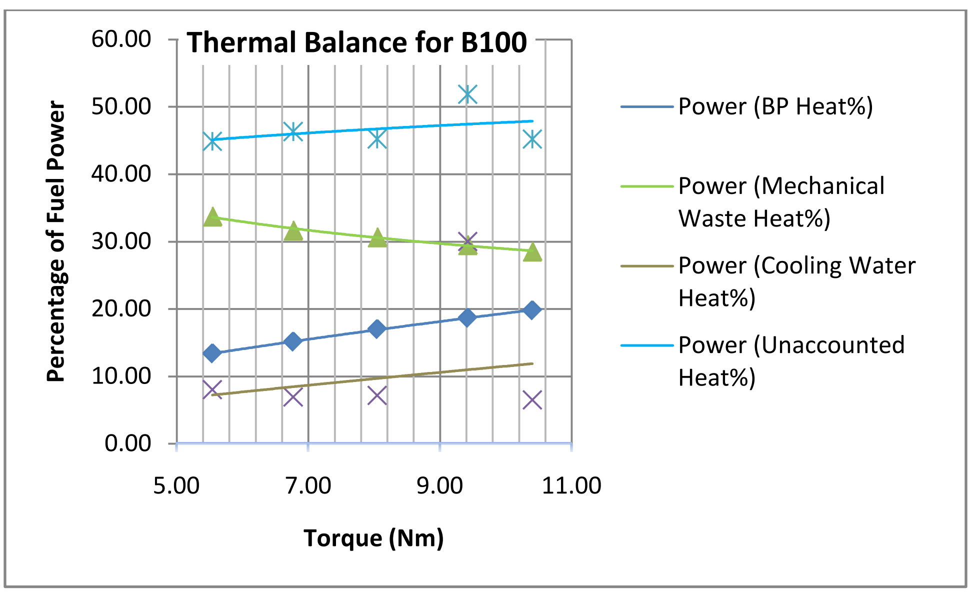
(Figure 5.13): Heat balance diagram against applied torque at a constant speed test condition for Bio-50 fuel.

With reference to table (5.9)



(Figure 5.14): Heat balance diagram against applied torque at a constant speed test condition for Bio-75 fuel.

With reference to table (5.11)



(Figure 5.15): Heat balance diagram against applied torque at a constant speed test condition for Bio-100 fuel.

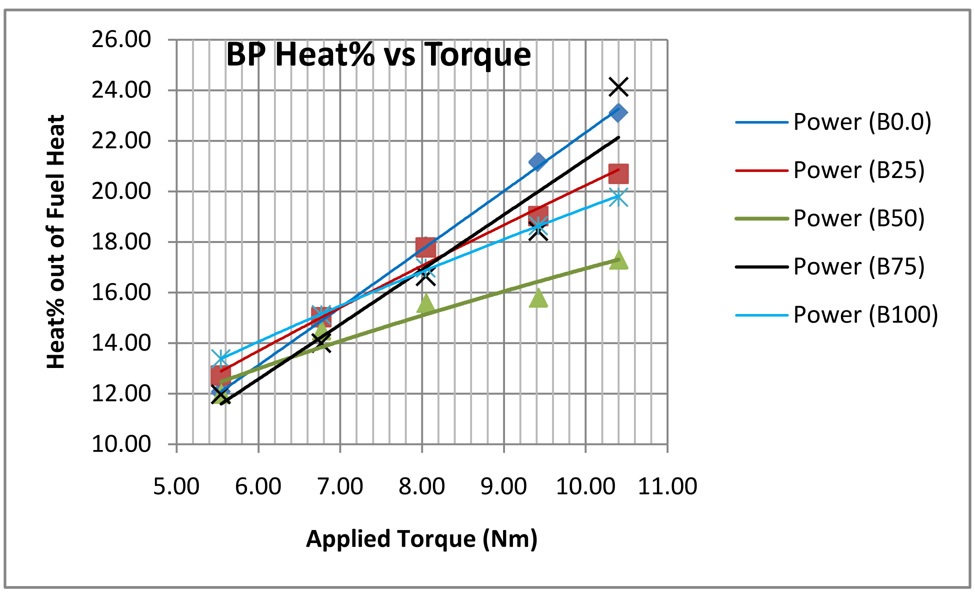
Other set of graphs for heat balance analysis are given in figures 5.16, 5.17, 5.18 and 5.19 to simplify the comparison of beat balance element for all blends at a time.

a- **Brake power** heat percentage, In Figure 5.16, the brake power heat percentage is shown. Values of brake power percentage out of that of fuel power are increasing with load from low values of minimum (12%) to highest value of 24% Bio-75 at maximum applied torque, with minor variations between the blends.

Table (5.22) Compiled results heat balance analysis for BP

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Brake Power Heat Percentage				
1	5.54	12.07	12.73	12.01	11.98	13.38
2	6.77	14.88	15.03	14.51	14.00	15.12
3	8.04	17.83	17.78	15.60	16.65	16.97
4	9.42	21.15	19.02	15.81	18.43	18.64
5	10.40	23.10	20.69	17.31	24.13	19.76

Compiled from column (4) in all thermal balance result tables



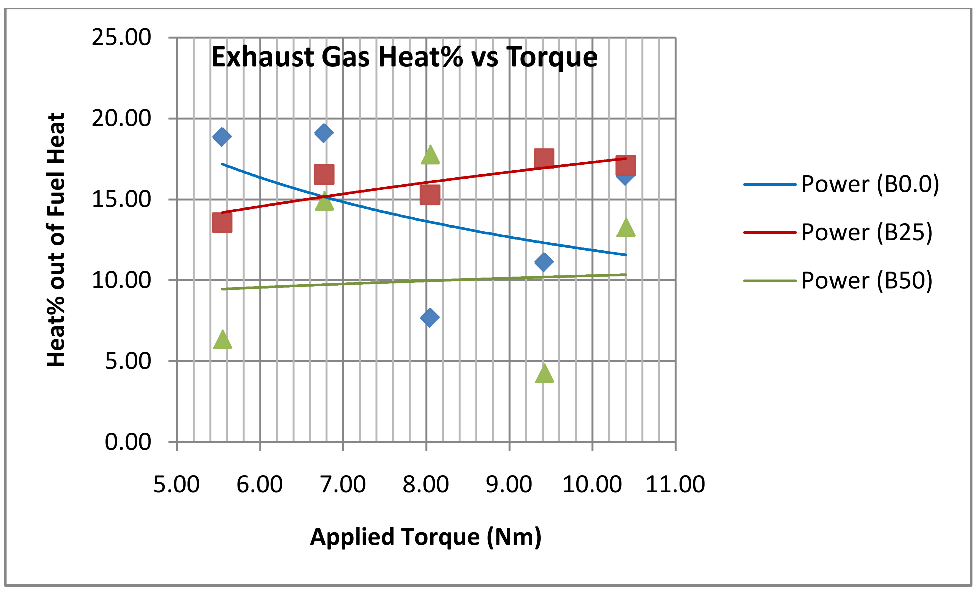
(Figure 5.16): Heat balance diagram for percentage of brake power heat against applied torque at a constant speed test condition for the different blends fuel.

b- **Exhaust gas heat loss**, was measured by a calorimeter attached to the engine for the three blends Bio-0.0, Bio-25 and Bio-50, heat loss in exhaust gas was of a minimum of 5.0% with Bio-50 at low load to a highest value of about 20% for pure diesel oil fuel also at low load, for Bio-25, exhaust gas loss percentage was fluctuating around 15% under all loads.

Table (5.23) Compiled results heat balance analysis for exhaust heat

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Heat Loss in Exhaust Gas Percentage				
1	5.54	18.85	13.58	6.38	48.07	62.51
2	6.77	19.09	16.54	14.94	63.40	52.00
3	8.04	7.69	15.28	17.80	65.00	50.00
4	9.42	11.11	17.51	4.28	39.42	55.00
5	10.40	16.45	17.09	13.30	58.41	53.00

Compiled from column (5) in all thermal balance result tables



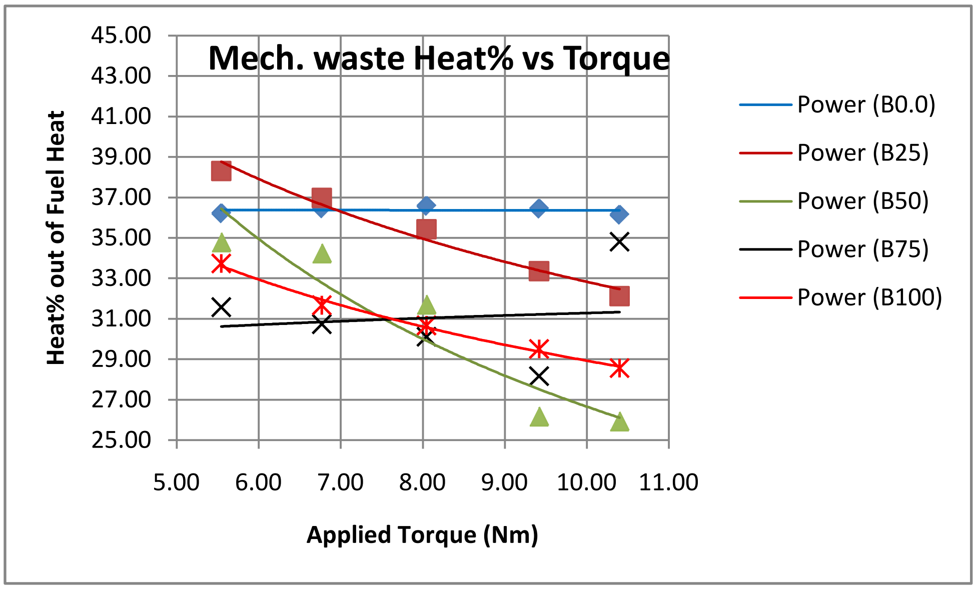
(Figure 5.17): Heat balance diagram for percentage of Exhaust gas heat against applied torque at a constant speed test condition for the different blends fuel.

c- **Mechanical waste** in heat balance analysis, should give stable close figures for any specific blend since the test is a constant speed test. Pure diesel oil fuel fives a good example for this with values around 36%, other blends vary in mechanical loss percentage in heat balance analysis with small variations through the loading torques. The blend Bio-75 can be considered good for mechanical waste percentage heat. (Figure 5.18).

Table (5.24) Compiled results heat balance analysis for friction loss

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Mechanical Heat Loss Percentage				
1	5.54	36.21	38.30	34.80	31.57	33.73
2	6.77	36.45	36.97	34.25	30.74	31.67
3	8.04	36.58	35.42	31.71	30.11	30.67
4	9.42	36.44	33.35	26.18	28.17	29.51
5	10.40	36.14	32.12	25.94	34.80	28.56

Compiled from column (6) in all thermal balance result tables



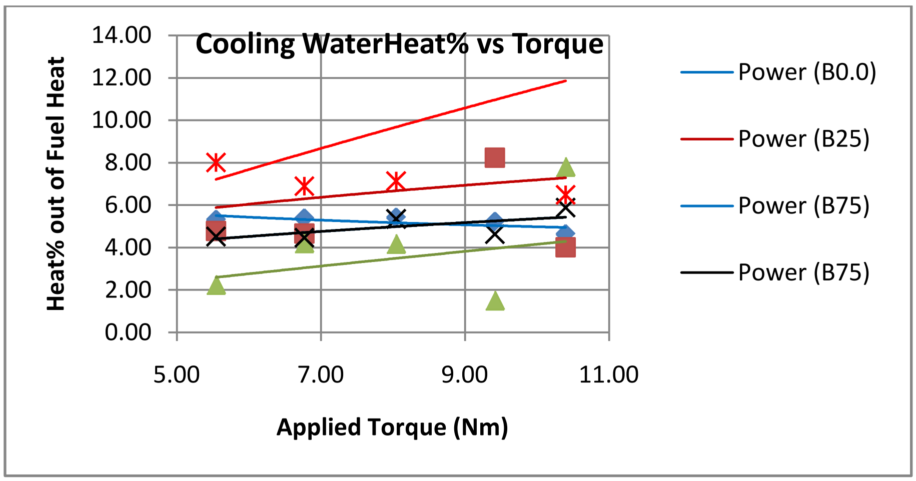
(Figure 5.18): Heat balance diagram for percentage of Mechanical waste heat against applied torque at a constant speed test condition for the different blends fuel.

d- **Cooling water heat loss** in heat balance test has recorded values between a small value of 3% for Bio-15 up to 15% with Bio-25 an odd figure of 30% was observed with Bio-100 under 9.42 Nm load. It can be summarized that cooling water waste percentage bio-fuel blends is close to that of pure diesel oil fuel in heat balance test. (Figure 5.19).

Table (5.25) Compiled results heat balance analysis for cooling loss

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Heat Loss Percentage in Engine Coolant				
1	5.54	5.32	4.79	2.25	4.52	8.01
2	6.77	5.38	4.66	4.21	4.47	6.89
3	8.04	5.42	17.23	4.18	5.35	7.14
4	9.42	5.22	8.23	1.51	4.63	30.00
5	10.40	4.64	4.02	7.82	5.88	6.49

Compiled from column (7) in all thermal balance result tables



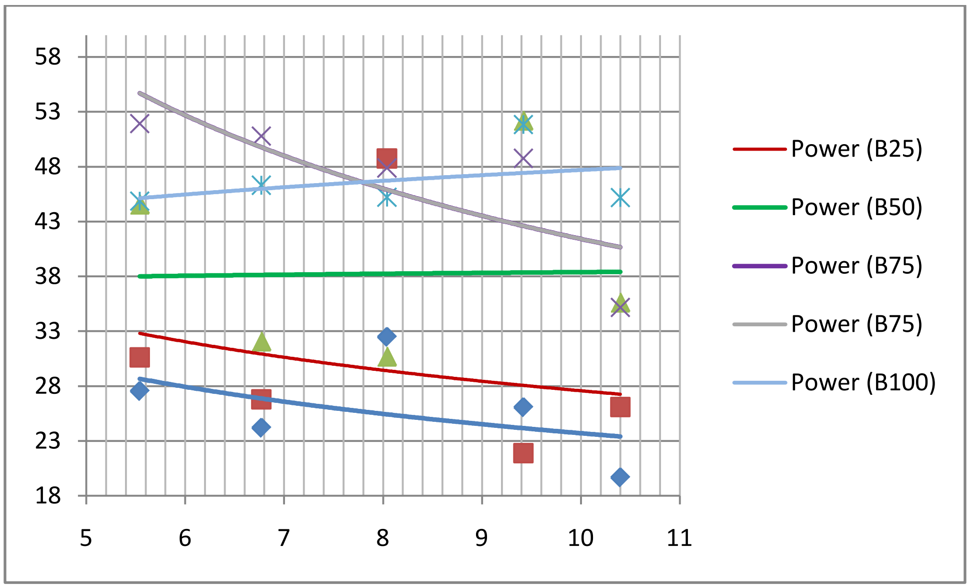
(Figure 5.19): Heat balance diagram for percentage of cooling water heat loss against applied torque at a constant speed test condition for the different blends fuel.

e- **Unaccounted heat loss** percentage, it's the other form of energy loss in form of sound, vibration, radiated heat … etc. In Figure 5.20, the unaccounted form of heat loss is percentage is shown. Values of power percentage out of that of fuel power are decreasing with load from low values of minimum of about (20%) to highest value of 55% Bio-75 at minimum applied torque, with considerable variations between the blends. For all the blends this kind of loss is decreasing with greater applied torque.

Table (5.26) Compiled results for unaccounted heat loss percentages

S/No.	Torque T, (Nm)	B0.0	B25	B50	B75	B100
		Unaccounted Heat Loss Percentage				
1	5.54	27.55	30.61	44.57	51.93	44.88
2	6.77	24.20	26.79	32.08	50.79	46.32
3	8.04	32.49	48.75	30.70	47.89	45.22
4	9.42	26.07	21.89	52.22	48.77	51.84
5	10.40	19.67	26.09	35.63	35.18	45.19

Compiled from column (7) in all thermal balance result tables



(Figure 5.20): Heat balance diagram for percentage of unaccounted heat loss against applied torque the different blends fuel

CHAPTER VI

CONCLUSION AND RECOMMENDATIONS

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

6.1. **Conclusion**:

A concerns a constant speed test for a direct injection, water-cooled engine fuelled with diesel oil and blends of jatropha bio-diesel in diesel oil fuel. Blend percentages were on volume base of zero percent, twenty five percent, fifty percent, seventy five percent and hundred percent. Tests were conducted on constant speed with seven gradually increasing loads, first two of which were not used for analysis to allow engine stabilizing. Performance criteria investigated were; brake power, (BP), brake mean effective pressure (bmep), brake specific fuel consumption (BSFC), brake thermal efficiency (η_{thb}), mechanical efficiency (η_{mech}), volumetric efficiency (η_{vol}), air fuel ratio (A/F), equivalence air fuel ratio (ϕ) and ignition delay (τ). Thermal balance analysis were done for each of the blends to heat percentage elements of; heat percentage consumed in developing brake power, heat percentage lost for mechanical losses, heat percentage lost in engine cooling water, heat percentage lost with exhaust gas and unaccounted heat los percentage. From general observation and result the author has come out with these following points:

1- Jatropha bio-diesel has a high promising chance in the future as an alternative fuel.
2- Jatropha plant can be grown locally for bio-diesel fuel production beside other ecological and economical benefits.
3- Good firing was attained with the engine at various blends of jatropha bio-diesel blends in diesel oil fuel up to Bio-100 (pure bio-diesel).
4- Bio-75 was good blend for;
    i. Less mechanical friction loss,
    ii. Stability in exhaust heat loss,
    iii. Relatively g mechanical efficiency,
    iv. High stability in cooling engine heat loss.
5- Concerning ignition delay of the engine, which is a measure of ignition quality, it is less with low percentages of bio-fuel blends than that of high percentages.

6.2. **Recommendations**:

i. More investigations are needed to adopt jatropha bio-diesel fuel as an alternative fuel for C.I. engines.
ii. Research for low percentage blends of jatropha bio-diesel blend in CI engines for uncertainty of high percentages of bio-diesel fuel.
iii. Mechanical investigation solution of pro-turbulence for more shortening ignition delay time.

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APENDEXES

Appendix (A1), Jatropha Oil Properties

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TEST REPORT

Sample Type جاتروفا
Sample Code SP_6
Sample ID 0015571
Customer Name Student
Customer Ref. _
Date /Time Received 4-Jun-2014

Report Number 0015571
Date /Time Report 9-Jun-2014

Test Name	Test Method	Unit	Result
Kinmatic viscosity @40^{0}C	ASTM D445	cSt	4.631

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Date : 9-11. 2014

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Appendix (A2), Jatropha Bio-Diesel Properties (General)

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فاكس: +(249) -1-83429641 ص.ب: 2986

TEST REPORT

Sample Type	Biodiesel
Sample Code	sp-109
Sample ID	14412
Customer Name	Student
Customer Ref.	-
Date /Time Received	17-Nov-2013

Report Number
14412
Date /Time Report
24-Nov-2013

Test Name	Test Method	Unit	Result
Sulfur content	ASTM D4294	Wt%	0.0077
Density @15^{0}C	ASTM D4052	g/cm3	0.9199
TAN	ASTM D664	mgKOH/g	3.59
Calorific value gross	Calc	MJ/Kg	43.76
Viscosity @40 °C	ASTM D445	cSt	36.39

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TEST REPORT

Sample Type	Biodiesel
Sample Code	sp-110
Sample ID	0014458
Customer Name	Student
Customer Ref.	-
Date /Time Received	24-Nov-2013

Report Number
0014458
Date /Time Report
26-Nov-2013

Test Name	Test Method	Unit	Result
Kin Viscosity @40°C	ASTM D 445	cSt	6.625
Cloud point	ASTM D2500	°C	11

MINISTRY OF PETROLEUM
SUDANESE PETROLEUM CORPORATION
CENTRAL PETROLEUM LABORATORIES
(C.P.L)
Date : 26-11-2013

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Appendix (B), ASTM Requirements for Bio-Diesel (B100)

ASTM Requirements for Biodiesel (B100)			
Property	**Units**	**Grade S15 Limits**	**Importance**
Calcium and magnesium, combined	ppm (µg/g)	5 max	To protect against wear of injectors, pumps, pistons & ring and also engine deposits & premature failure of particulate filters
Flash point	°C	93 min	Safety during fuel handling & storage
Alcohol control - One of following must be met: 1. Methanol content 2. Flash point	 % mass °C	 0.2 max 130 min	To ensure alcohol from manufacturing process is properly removed
Water and sediment	% volume	0.050 max	Filter plugging, injector wear, increased corrosion
Kinematic viscosity, 40°C	mm2/s	1.9-6.0	Injector wear & spray pattern, pump wear, filter damage
Sulfated ash	% mass	0.020 max	Limits unremoved catalyst from fuel to protect against wear in injector pumps, pistons, rings & reduce engine deposit
Sulfur	% mass (ppm)	0.0015 max (15)	To protect emissions control equipment
Copper strip corrosion		No. 3 max	Protect copper, brass, bronze fuel system parts
Cetane number		47 min	Measure of ignitability (ignition quality)
Cloud point	°C	ReportD	Low-temperature operability
Carbon residue	% mass	0.050 max	To reduce deposits in fuel system and engine
Acid number	mg KOH/g	0.50 max	Protect against fuel system deposits & corrosion
Free glycerin	% mass	0.020	Injector deposits & fuel system clogging
Total glycerin	% mass	0.240	Injector deposits, filter plugging & low-temperature operability
Phosphorus content	% mass	0.001 max	Protect catalysts in exhaust after-treatment devices
Distillation temperature, 90% recovered	°C	360 max	Affects fuel economy and power under varying loads/speeds
Sodium and potassium, combined	ppm (µg/g)	5 max	Limits unremoved catalyst from fuel to protect against wear in injector pumps, pistons, rings & reduce engine deposit
Oxidation stability	hours	3 min	Storage stability, prevent degradation of fuel

Note that the above specifications are identical for biodiesel blended into ULSD (grade S15) and low-sulfur diesel (grade S500), except that grade S500 is permitted to have up to 500 ppm sulfur while grade S15 is limited to 15 ppm sulfur.

Appendix (c), **Benefits and Concerns –Biodiesel and Biodiesel Blends** standard by ASTM

Benefits and Concerns – Biodiesel and Biodiesel Blends

Benefits:

1. Higher cetane
2. Improved lubricity
3. Low sulfur content
4. Reduces HC and CO emissions
5. Reduces PM emissions
6. Lowers visible smoke
7. Zero aromatics
8. Reduces net CO_2 emissions on life-cycle basis
9. Nontoxic and biodegradable

Concerns:

1. Materials compatibility – especially with higher blend levels
2. Potential for increased water content and microbial contamination
3. Increase in NO_x emissions
4. Implications for emission control equipment
5. Impact on low-temperature operability if not properly additized
6. Reduced power and fuel economy (on B100) – lower energy content
7. Thermal and oxidative stability poorer than No. 2 diesel
8. Solvency effect may plug filters on initial use
9. Requires special care and handling
10. Spills could damage paint
11. Higher cost