

**Waste Vegetable Oil and Coconut Oil as an Alternative Fuel
For Diesel Engines**

by

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Nomenclature

1. Symbols

A	Non-flow availability
C	Specific heat
C_D	Coefficient of discharge
D	Diameter
G	Gibb's free energy, steady-flow availability
g	Gibb's free energy per mole of fuel
H	Enthalpy
h	Specific enthalpy
h_m	Manometric deflection
K	Constant
m	Mass
\dot{m}	Mass flow rate
N	Crankshaft rotation per minute
P	Power
p	Pressure
Q	Heat transfer
\dot{Q}	Heat transfer rate
\dot{q}	Heat transfer rate per unit mass of fuel
R	Gas Constant
S	Entropy
s	Specific entropy
T	Temperature
t	time
U	Internal energy
u	Specific internal energy
V	Cylinder volume
V_c	Clearance volume

V_d	Displacement volume
v	Specific volume, velocity
W	Work transfer
w	Work transfer per mole of fuel
y	Mole fraction
α	Power adjustment factor
β	Fuel consumption adjustment factor
ε	Second-law effectiveness or effectiveness
η_v	Volumetric efficiency
η_b	Brake thermal efficiency
$\eta_{II} (\eta_{av})$	Second-law efficiency (Availability efficiency)
μ	Chemical potential
ν	Stoichiometric coefficient
ρ	Density
γ	Specific gravity
ϕ	Relative humidity
E_i^{ch}	Fuel chemical exergy
Ex_{in}	Fuel chemical exergy input (approximate)
$\eta_{ex,shaft}$	Efficiency of chemical exergy output at shaft

2. Subscripts

a	Air
b	Brake parameter
db	Dry bulb
f	Fuel, friction
r	Reference condition as per BS 5514
i	Indicated, inlet, species i
in	Input
0	Reference value
00	Environmental dead state
wb	Wet bulb
x	Lab condition

3. Notations

Δ	Difference
-	Average or mean value

4. Abbreviations

<i>A/F</i>	Air-fuel ratio
<i>bme_p</i>	Brake mean effective pressure
<i>bsfc</i>	Brake specific fuel consumption
<i>LHV</i>	Lower heating value of fuel
<i>WOT</i>	Wide open throttle
<i>WVO</i>	Waste soybean oil

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Abstract

The subject of alternative fuels for diesel engines has become a popular one, particularly among environmentalists and political activists. In an effort to thwart human-induced environmental problems, such as global warming and climate change, increased efforts are taking place to find alternatives to the greatest producers of harmful greenhouse gases. Diesel fuel is one such producer. When burned, diesel emits carbon dioxide and other polluting gases that hurt the earth's atmosphere. Fortunately, some promising technologies, both existing and in development, can offer large-scale environmental solutions at reasonable costs. In this regard, vegetable oils are proved to be promising options. In this study waste soybean oil and raw coconut oil is used as alternative fuel. The lower heating value is found to be lower, on the other hand density and viscosity of waste soybean oil and coconut oil are found to be very close to that of diesel at higher temperature. In the present study, waste soybean oil and coconut oil is preheated and blended with 20% diesel fuel and their effects on engine performance is investigated on a direct injection, 4 stroke, single cylinder diesel engine. Engine performance has been evaluated with the help of both conventional performance parameters, viz; brake thermal efficiency and brake specific fuel consumption. In addition, the exergy analysis has been studied for the diesel as well as for alternative fuels. Major portion of the fuel chemical exergy is destroyed due to friction, cooling, combustion, etc. and due to exhaust gases.

CHAPTER 1

INTRODUCTION

1.1 Prologue

The search for renewable fuel sources is being made universally and this includes alternative fuels for internal combustion engines. Vegetable oil has been suggested and tested as one of the alternatives for conventional engine fuels; it would help to extend the life of fast depleting petroleum reserves. Alternative fuels promise to harmonize sustainable development, energy conservation, management, efficiency and environmental preservation.

Economic and political factors make supplies of petroleum uncertain and have given rise to tremendous price escalation. The effect on developing countries is much more severe. Thus, the search for alternative fuels for internal combustion engines, for both automobile applications and for stationary motive power, has become important.

Petroleum based fuels are obtained from limited reserves. These finite reserves are highly concentrated in certain regions of the world. Therefore those countries not having these resources are facing a foreign exchange crisis, mainly due to the import of crude oil. With the increase of the number of internal combustion engine, the demand for liquid petroleum fuel has also increased that resulted in increasing atmospheric pollution. These concerns have highlighted the need for diversification and prompted increased research throughout the world into the potential alternative fuels for internal combustion engines.

1.2 Use of Vegetable Oils as I.C Engine Fuels

For many developing countries, the concept of employing vegetable oil as sources for diesel fuels can be attractive. The energy crisis of 1974 brought pressure on various nations to search for non petroleum based alternative fuels. Since then many vegetable oils of commercial and non-commercial use have been tried and possible difficulties identified. The observed major differences between vegetable oil and diesel fuel, included, for the latter, the significantly higher viscosities and moderately higher densities, lower heating values, rise in the stoichiometric fuel /air ratio due to

the presence of molecular oxygen and the possibility of thermal cracking at the temperatures encountered by the fuel spray in the naturally aspirated diesel engines. These differences contributed to the poor atomization, coking tendencies, carbon deposits and wear that were generally experienced and which adversely affect the durability of the engine. The durability problems encountered in diesel engines fuelled with vegetable oils have been attributed to the following characteristics of the oil: (1) the high viscosities of these fuels with the degraded atomization as the fuel leaves the injector, this leads to improper of air and fuel leading to incomplete combustion and formation of deposits on engine parts (2) the chemical different structure that affects the chemistry of combustion and (3) the lubricating oil contamination due to incomplete combustion. For generating optimum efficiency by alternative fuels some engine modification is necessary. [Bhattacharyya 1994]

1.3 Analysis of The Second law Thermodynamics

The first law places no restriction on the direction of a process, but satisfying the first law does not ensure that a process will actually occur. This inadequacy of the first law to identify whether a process can take place is remedied by introducing another general principle, the second law of thermodynamics. A process will not occur unless it satisfies both the first and the second laws of thermodynamics. The second law of thermodynamics is a powerful statement of related physical observations that has a wide range of implications with respect to operation of thermal systems. For example, the second law can be used to determine the direction of processes, to establish the conditions at equilibrium, to specify the maximum possible performance of a thermal system, and to identify those aspects of processes that are detrimental to overall performance.

The first and second laws of thermodynamics can be employed to take into account the account the quantity and quality of energy. The methodology which utilizes both laws is called 'exergy analysis'(or 'availability analysis'). Through exergy analysis, the locations where destruction or losses of useful energy occur within a device or process can be identified and rank ordered as to their significance.

The second-law analysis deals with the quality of energy in particular. It is concerned with the degradation of energy during a process, the entropy generation and the lost opportunities to do work and it also offers opportunities for improvement. Furthermore, the first-law analysis might indicate no loss in energy for some processes e. g. a throttling process, but actually there is loss in the value or quality of energy. Another point in the comparison between the first and second law analysis is the datum (zero level) of energy and availability. While the datum for energy is arbitrarily chosen, the exergy datum is chosen scientifically, at the level where there is no further useful work that can be produced. This means that for some processes the enthalpy drop and losses can be the same whether it happens near the datum or away from it for the same processes. The exergy drop and loss will differ and depend on whether they are near or far from the datum.

The exergy (or availability) is the maximum useful work that could be obtained from a system at a given state in specified environment. It is also the useful work potential of a given amount of energy at some specified state. Work done during a process depends on the initial state, the final state and the process. The work output is maximized when the process between two specified state is executed in a reversible manner. The system must be in the dead state at the end of the process to maximize the work output. A system is said to be in the dead state when it is in thermodynamic equilibrium with the environment. Exergy is a property of a system environment combination.

An exergy balance best defines the effect of all the losses in mechanisms on engine efficiency for the real engine cycle. The loss in exergy due to heat losses, flow losses, and mechanical friction are real engine effects.

The purpose of the present analysis is to illustrate the capability of the exergy analysis to provide a systematic approach to focus on the waste and lost energy within diesel engines. This leads to a significant improvement in energy utilization.

1.4 Objectives of the Study

This research work comprises of both experimental and analytic investigations with the following specific aims:

- i. To estimate the physical properties of filtered waste soybean oil and coconut oil.
- ii. To conduct a detailed study based on both the first and the second laws of thermodynamics in order to examine the effects of speed and load on various thermodynamic processes of a diesel engine operation using preheated waste soybean oil, preheated raw coconut oil and a blend of waste soybean oil and coconut oil with raw diesel oil as substitutes to diesel fuel.
- iii. To compare performance parameters obtained from waste soybean oil, raw coconut oil and the blend oil with that of diesel oil.
- iv. Again to compare performance parameters obtained from raw coconut oil and the blend oil with that of diesel oil.

1.5 Scope of the Study

The main purpose of this study is not to convert edible oil into fuel but to carry out an experiment whether different grades of WVO (waste soybean oil) and coconut oil could be converted into economically viable fuel oil. According to WHO (World Health Organization) edible oil after being burnt several times lose quality for human consumption and coconut oil is chosen as it is not popular as edible oil in Bangladesh. Such type of oil could be converted into fuel to meet the fuel short fall. For the purpose of this study waste soybean oil, fried two times is collected from restaurants. It is then tested for its fuel related properties. Experiments are carried out in single cylinder, 4 stroke, water cooled diesel engine at three different speeds (at 2600, 2400 and 2200 rpm) and different loading conditions. Initially, the engine is run by diesel fuel under different loads at three different speeds and then the experiments are repeated consecutively replacing diesel by waste soybean oil and coconut oil under preheating temperatures of 80 and 100°C and 80% waste soybean oil and coconut oil blended with 20% diesel under preheating temperature of 100°C.

In this dissertation, in Chapter 2 the available literature is reviewed, also a brief description of use of vegetable oil as diesel fuel and its property is given in Chapter 2. The concept of exergy and exergy analysis of heat engines and its fuels are introduced in Chapter 3. Chapter 4 discusses with the experimental set up and procedure. In Chapter 5 the outcomes of the study are explained in detail. Conclusion of the work done and recommendations of the further related work are in Chapter 6

CHAPTER 2

LITERATURE REVIEW

In the literature review at first, there is a summary of the works on exergy analysis of heat engines have been arranged alphabetically. Then results on performances of diesel engines using vegetable oil fuels have been summarized later.

2.1 Literature Review of Works on Exergy Analysis of Heat Engines

Al-Najem and Diab [1992] made an energy-exergy analysis for a diesel engine. They reported that about 50% of the fuel availability was destroyed due to unaccounted factors such as combustion, 15% had been removed via exhaust and cooling water and 1% had been destroyed in the turbocharger.

Caton[1999] studied availability characteristics as a function of speed and load for a spark-ignition engine. For this study, a commercial V-8 spark-ignition engine was selected. This paper described the use of a comprehensive thermodynamic cycle simulation to examine the effect of engine load and speed on a number of performance energy and availability terms. In one portion of this study, he examined the effects of engine load and speed on a number of performance parameters in energy and availability terms.

Caton [2000] reviewed investigations that used the second law of thermodynamics in studying internal combustion engines in a detailed manner in the late 1950s. Over two dozen previous investigations which had used the second law of thermodynamics or availability analyses were identified. About two-thirds of these were identified for diesel engines and the rest for spark ignition engines.

Caton [2000] examined the destruction of availability during combustion processes for an adiabatic constant volume system. It was an analytical examination. The fraction of the fuel's availability that had been destroyed due to the irreversibility process was obtained as a function of temperature, pressure and equivalence ratio for

octane air mixtures. In general, the destruction of the fuels available energy due to the combustion process decreased for operation at higher temperatures. Specifically for the conditions of this study, the destroyed availability due to the combustion process ranged between 5-25% of the original availability. The implications of these results to combustion process in internal combustion engines were described.

Kumar and Minkowycz [1989] presented the methodology and some preliminary results in the application of exergy as a second law analysis parameter to a complete diesel engine cycle. Preliminary results showed that exergy was indeed a powerful tool in quantifying the losses and irreversibilities in internal combustion engines.

Rakopoulos and Giakoumis [2004] reported on the use of a computer analysis which was developed for studying the energy and exergy performance of a turbocharged diesel engine, operating under transient load conditions. In this report the effect of various thermodynamic and design parameters on the second law transient performance of the engine manifolds and turbocharger had been investigated. It was revealed from the analysis that the exhaust manifold process had been the second biggest irreversibility producer when load increased.

Rakopoulos and Giakoumis [2005] developed a simulation model for IDI turbo charged diesel engine with a prechamber. The study focused on the fundamental and irreversibilities in combustion analysis, and thereby, the possibilities of recovering work potential in heat losses. Availability equations have been used in both main and prechamber for each of the six cylinders. The model was then validated against experimental results for both steady state and transient flows. The authors studied, among other things, the effects of design and operating parameters such as, prechamber diameter, static injection timing and engine load. They concluded that combustion itself is responsible for the irreversibility produced in the engine. More specifically, main chamber contributed more to it than that in the prechamber. Finally, they opined for a detailed optimization of the engine systems using both the first and the second laws of thermodynamics.

Rakopoulos and Giakoumis [2006] surveyed the publications available concerning the application of second law of thermodynamics to internal combustion engines. The exergy balance equations of the engine cylinder and subsystems were reviewed in detail providing relations concerning the definition of state properties, chemical availability, flow and fuel availability and dead state. The review of the previous works was categorized according to compression ignition engines, spark ignition engines, engine subsystems, low heat rejection, alternative fuels, and transient operation.

Rakopoulos and Kyritsis [2001] used a method which was developed by them for both combustion irreversibility and working medium availability computations in a high speed, naturally aspirated four stroke internal combustion engine. In this report combustion irreversibility was found to decrease and exhaust gas availability to increase with increasing overall equivalence ratio, while both irreversibility and gas availability increased with engine rotational speed. The authors found that the use of alternative fuels could potentially lead to increase in second law efficiency.

Rakopoulos and Kyritsis [2001] compared the results of the second law analysis of engine operation with an n-dodecane with the results of a similar analysis for cases where a light, gaseous (CH_4) and an oxygenated (CH_3OH) fuel had been used. In this report combustion irreversibility was found to be the main source of irreversibility during engine operation and its differential variation was calculated as an analytic function of the differentials of the molar quantities of the constituents of the working medium.

Sezer and Bilgin [2008] was installed a two-zone quasi-dimensional cycle model to study the theoretical exergetic evaluation of spark ignition engine operation. The cycle simulation consisted of compression, combustion and expansion processes. The combustion phase was simulated as a turbulent flame propagation process. Intake and exhaust processes were also computed by a simple approximation method. The results of the model were compared with experimental data to demonstrate the validation of the model. Principles of the second law were applied to the model to

perform the exergy (or availability) analysis. In the exergy analysis, the effects of various operational parameters, i.e. fuel–air equivalence ratio, engine speed and spark timing on exergetic terms have been investigated. The results of exergy analysis show that variations of operational parameters examined have considerably affected the exergy transfers, irreversibilities and efficiencies. For instance, an increase in equivalence ratio caused an increase in irreversibilities, while it decreased the first and also the second law efficiencies. The irreversibilities had minimum values for the specified engine speed and optimum spark timing, while the first and second law efficiencies reached a maximum at the same engine speed and optimum spark timing.

2.2 Literature Review of Works on Using Vegetable Oils in Heat Engines

The last two decades has witnessed growing concern on the environment and the effect of greenhouse gases, a consequence of combustion of fossil fuel. This has revived interest in the use of vegetable oil as a fuel substitute, with re-evaluation of their performance in unmodified diesel engines. Literature reviews produce a number of recent studies that use vegetable oils as conventional diesel fuel substitutes.

Ali and Hanna (1994) studied the use of vegetable oils and animal fats as diesel fuel. One important conclusion that could be drawn from the work done on the utilization of vegetable oils as fuels was that they cannot be used directly in the engine. Vegetable oil processing costs could be reduced through increased plant capacities, it would become a more viable alternative to diesel fuel.

Altin et al [2001] reported that compared to diesel fuel a little amount of power loss happened with vegetable oil fuel operations. Particulate emissions of vegetable oil fuels were higher than that of diesel fuel, but NO₂ emissions were less. Vegetable oil methyl esters gave performance and emission characteristics closer to the diesel fuel. They concluded that raw vegetable oils could be used as fuel in diesel engine with some modifications.

Bhattacharya and Reddy [1994] worked on liquid and gaseous alternative fuels. This paper summarized the results of a compilation and review of the available work on vegetable oils as an alternative liquid fuel for stationary CI engines. They concluded that the most significant difference between diesel and vegetable oil was the latter's much higher viscosity; consequently problems such as poor atomization, coking tendencies, carbon deposits, and increased wear developed which adversely affected the durability of the engine.

Diaz [2000] had taken up attempts to prove that vegetable oil, especially; coconut oil was very clean fuel with excellent combustion properties.

Dorado et al. [2003] studied the exhaust emissions of a diesel direct injections Perkins engine fueled with waste olive oil methyl ester at several steady state operating conditions. Results revealed that the use of biodiesel resulted in lower emissions of CO (up to 58.9%), CO₂ (up to 8.6%), NO (up to 37.5%) and SO₂ (up to 57.7%), with increase in emissions of NO₂ (up to 81%). Biodiesel presented a slight increase in brake specific fuel consumption, but combustion efficiency remained the same.

Giannelos et. al. [2005] tested physical, chemical and fuel related properties of tomato seed oil for evaluating its direct use in diesel engines. It was found that the use of food industry by products such as tomato seeds, in non food applications, enhanced their value and importance for rural economy and reduced European dependence on petroleum.

Graboski and McCormick [1998] studied the status of fat and vegetable oil derived diesel fuels with respect to fuel properties, engine performance and emissions. The fuels considered were primarily methyl esters of fatty acid derived from a variety of vegetable oils and animal fats and referred to as biodiesel. Economics of biodiesel production were discussed, and it was concluded that the price of the feedstock fat or oil had been the major factor determining biodiesel price. It was found that the lubricity of these fuels was superior to conventional diesel, and this property ;;was

imparted to blends at level above 20 vol%. They found that the NO_x emission increase significantly for both neat and blended fuels in both two- and four-stroke engines.

Haq [1995] studied the suitability of vegetable oils as diesel fuel substitute. He reported that the calorific values of vegetable oils had been slightly less than that of diesel. Calorific values and energy densities of vegetable oils were very close to that of diesel. But, viscosities of straight vegetable oils were found to be significantly higher than that of diesel. With adequate preheating, the viscosity of straight vegetable oil became comparable with that of diesel fuel. Same inference was found for density.

Harrington [1986] recorded some chemical and physical characteristics of some compression ignition fuels derived from vegetable oils and established some relationships between them. The guidelines have proved successful in predicting the ignition quality of a range of vegetable oil fuels evaluated by means of a rapid ranking engine test method and have been further supported by cetane index calculations.

Islam [2006] carried out the performance test of diesel engine run by preheated straight vegetable oil on the basis of both the first and second laws of thermodynamics. He has found that the brakes thermal efficiency of the engine apparently increases with increased preheat temperature of the soybean oil fuel oil and at 100°C it becomes very much compatible with the performance trends obtained using diesel fuel. Same implication is true in case of availability efficiency, but its value is less by 1-1.15%. The major part (almost 50%) of the available energy is wasted and lost due to uncounted factors. He also explained that the availability destroyed in exhaust gases is about 10% of the available energy.

Karaosmanoğlu et al. [2000] determined diesel fuel properties of sunflower oil and performed long term test of sunflower oil. Engine tests were conducted at 1600 rpm under part load condition for 50 hours. An overall evolution of results indicated that

the sunflower oil could be proposed as a possible candidate for substituting diesel fuel.

Kumar [2005] evaluated the effect of fuel inlet temperature on performance, emission and combustion characteristics of a diesel engine. It was found that preheated animal fat could be used in diesel engines with reduced smoke, hydrocarbon and CO emissions with no major deterioration in engine performance.

Lee [2001] studied the spray characteristics of alternative fuel in constant volume chamber. In this study DME, LPG and n-dodecane were injected into a high pressure and temperature chamber by means of an electronically controlled common rail injection system to reproduce their charge stratification process in the engine. It had been found that the mixture formation of DME and LPG sprays was faster than that of n-dodecane due to their flash boiling characteristics at high pressure and temperatures.

Morshed [2008] carried out performance study of a direct injection, 4 stroke, 3 cylinder diesel engine by blending sesame oil with diesel. Where 80% sesame oil was blended with 20% diesel. He found that Availability destroyed with exhaust gas increased with load and it was about 15% of the input availability at rated condition. That might be used directly for preheating purpose.

Nwafor and Rice [1996] carried out a study on the performance of rapeseed oil blends in a diesel engine. The test results showed increases in brake thermal efficiency as the amount of rapeseed oil in the blends had been increased. Reduction of power- output was also noted with increased amount of rapeseed oil in the blend.

Nwafor [2004] studied the emission characteristics of diesel engine running on vegetable oil with elevated fuel inlet temperature. The test results showed that there was no significant difference in BSFC between heated and unheated fuels at higher loading conditions. It revealed that the CO production with heated fuel was a little higher than the diesel fuel at higher loads. The CO concentrations in the exhaust gas were higher for unheated oil operation compared to other fuels. The heated oil

showed marginal increase in CO₂ emission compared to diesel fuel. The hydrocarbon emissions were significantly reduced when running on vegetable oils. The fuel consumption was a little worse when running on plant oil. The ignition delay was longer for unheated plant oil operation.

Ramadhas et al. [2004] investigated the effect of using various blends of rubber seed oil as fuel in CI engines. It was found that 50-80% of rubber seed oil blends gave the best performance.

Shay [1993]) reviewed some of the results obtained from using vegetable oils and their derivatives as fuel in CI engines and examined opportunities for their broader production and use

Uddin [2005] observed the behavior of SVO (soybean) with various performance parameters at different engine speeds and possibility of using as a substitute of diesel fuel. He concluded that straight soybean oil could be used to substitute diesel fuel in CI engines with very little power and efficiency loss. He also explained that increase of the preheat temperature of soybean oil reduces its viscosity that ensures better atomization and mixing of fuel with air and results in better combustion and enhanced engine performance

2.3 Use of Vegetable Oils as Diesel Engine Fuel

Rudolf Diesel, the inventor of the diesel engine, used peanut oil as fuel in one of his engines at the Paris exposition of 1900. But it is only in the last few decades that researchers around the world have been performing systematic studies on alternative fuels in diesel engines. Research is going on to find out suitable alternatives to diesel fuel, which do not require major engine modifications. For diesel engines, alternative renewable fuels are mainly vegetable oils referred to as bio-oils and transesterified bio-oil more popularly referred to as biodiesel. Biodiesel is a “diesel like” fuel which means it has properties very close to conventional diesel and can be used in land transport and stationary engines. Medium speed diesel engines can operate well on

the use of straight and unprocessed vegetable oil such as coconut oil, and soybean oil. Compared to of biodiesel, vegetable oils considerably reduce fuel cost in such engines. The growing global concern about environmental issues in the 90's (i.e. Clean Air Act) has increased the interest in alternative fuels paving the way to greater funding and effort for research studies. The increasing amount of greenhouse gases (GHG) such as CO₂ which is causing global warming and climate change, as well as the declining reserves of fossil fuels, and more importantly, the high fuel prices have strongly increased the interest in the use of alternative fuel for land transport and power generation.

Utilization of alternative fuels shows the following general advantages: (Diaz, 2000)

1. Lower dependence on crude oil,
2. Renewable fuel,
3. Favorable energy balance,
4. Reduction in greenhouse gas emission,
5. Lower harmful emission,
6. Biodegradable and non-toxic,
7. Option to the use of agricultural surplus

The major difficulties those are likely to be faced while using vegetable oil as IC engine fuels are listed below (Ramadhas *et al.*, 2004):

1. The price of vegetable oil is dependent on the feedstock price.
2. The feedstock homogeneity, consistency and reliability are questionable.
3. Homogeneity of the products depends on the supplier, feedstock and production methods.
4. Storage and handling of vegetable oil is difficult (particularly stability in long term storage)
5. Flash point in blends is unreliable.
6. Compatibility of vegetable oil with diesel engine needs to be studied further.
7. Cold weather operation of the engine in not easy with vegetable oil.
8. Acceptance by engine manufacturers is another major difficulty.

9. Continuous availability of vegetable oil needs to be assured before embarking on the major use of it in IC engine.

Vegetable oil as fuel cannot be used directly in the engine. In short term tests vegetable oils performed satisfactorily but their long term performance remains questionable due to heavy deposition problems and significantly greater engine wear. [Bhattacharya ,1994].

Since vegetable oils have different physical and chemical properties than those of diesel fuel, it is expected that a direct injection diesel engine designed for diesel operation will perform differently when fuelled by vegetable oils. So, if existing engines are to be used, then vegetable oil properties are to be modified. Considerable efforts have been made to develop vegetable oil derivatives that approximate the properties and performance of hydrocarbon based diesel fuels. Problems with substituting vegetable oils for diesel fuel are mostly associated with their high viscosities, low volatilities and polyunsaturated character. These can be changed in at least four methods [Ali and Hanna, 1994]:

1. **Pyrolysis:** It refers to a chemical change caused by the application of thermal energy in the presence of air and nitrogen spurge.
2. **Microemulsification:** It is a system consisting of a liquid dispersed, with or without an emulsifier, in an immiscible liquid usually in droplets larger than the colloidal size.
3. **Dilution:** Dilution of vegetable oils can be accomplished with such materials as diesel fuel, ethanol or kerosene.
4. **Transesterification:** It is the displacement of alcohol from an ester by another alcohol in a process similar to hydrolysis. This process has been widely used to reduce the viscosity of vegetable oils.

Bhattacharya and Reddy [1995] reviewed on each of the main vegetable oils tested for use as direct substitutes for diesel fuel either in their crude form or in the form of blends with diesel oil. Results are presented based on both short-term test durability of direct injection engines and in some cases of indirect injection (precombustion chamber) diesel engines. The major findings from these papers are listed as follows

Sunflower Oil

The performance of 25%-75% sunflower oil and diesel oil blend and a sunflower oil methyl ester with pure diesel oil was compared but no significant difference was noted with respect to characteristics such as thermal efficiency, ignition delay and exhaust temperature, CO and NO_x. Cylinder pressure rise and the rate of pressure rise increased with the blend. There was an increase in hydrocarbon emissions but a decrease in smoke was recorded. The methyl esters increased the cylinder pressure rise but reduced the smoke further compared with the blend

Hawkins et al. (1983) recommends that coking of the injector nozzle in direct injection diesel engines using pure sunflower oil could be reduced by esterification of these oils. Combustion studies, compatibility tests and engine durability tests for over 1500h have reportedly produced extremely good results with sunflower oil esters.

German et al. (1985) used two different blends of 25%-75% and 50%-50% alkali refined winterized sunflower oil in two different tractors with turbocharged diesel engines. No power losses were detected during the test period; however, one engine experienced cam shaft and valve train failure while in service. Heavier deposits occurred in most areas of the combustion chamber with 50%-50% sunflower oil and diesel oil blend hence these blends can't be recommended as long-term substitutes for diesel oil. However, the 25-75 blend was recommended for use under emergency conditions with the reduced engine life in mind. [Bhattacharyya 1994]

Soybean Oil

Pryor et al. (1982) tested the possibility of using soybean oils in a small diesel engine and concluded that crude soybean oil, crude de-gummed soybean oil and soybean

ethyl ester can be used as short term replacement for small direct injection engine. However, long-term use of 100% crude soybean oil was not recommended. Two hybrid fuels, an ionic and a non-ionic micro-emulsion of aqueous ethanol in soybean oil were tested for use in a naturally aspirated direct injection engine. The low cetane number of these hybrid fuels showed no adverse effect on short term engine performance. An ether addition was needed for cold starting

Ziemke et al. (1983) found that the closer Cetane rating and higher volumetric energy content of soybean oil as compared with diesel oil makes it preferable as a diesel engine fuel when compared with a biomass fuel such as ethanol. The 1000h performance recorded with soybean oil was the longest ever reported and reflects the potential of soybean oil for diesel engine use

Mazed et al. (1985) observed that a blend of 10% soybean oil and diesel oil gave higher brake thermal efficiencies than diesel oil.

Schlautman et al. (1986) employed a 75%-25% soybean oil and diesel blend in a naturally aspirated direct injection diesel engine. Tests revealed nearly the same power output for both the soybean oil blend and 100% pure diesel fuel but lubricating oil viscosity increased sharply and resulted in an increased lubricating oil temperature and engine coolant temperature incase of soybean oil blend. While exhaust temperature increased over time, the engine compression pressure decreased with time. [Bhattacharyya 1994].

Rapeseed Oil

Murayama et al. (1984) carried out on the use of rapeseed oil, methyl esters of rapeseed oil and rapeseed oil blended with diesel oil and ethanol in direct injection naturally aspirated diesel engine. No operational problems were encountered during short term tests. Problems such as carbon deposits and sticking piston ring during long term operationer by heating t were eliminated either by heating the fuel or by converting rapeseed oil into its methyl esters

Peterson et al. (1987) carried out short term and long term tests with high erusic acid rapeseed (winter rape) and low erusic acid rapeseed (spring rape). For winter rape the problems of ring gumming, lubricating oil thickening and injector gumming were absent. Some gumming was noticed on the upper piston land, but it was within acceptable limit. The study reported the successful use of a 70%-30% rapeseed oil and diesel blend to power a small single cylinder, swirl chamber diesel engine for 850 hours with no significant engine performance problems. For short term test, no operational problem was encountered but for long term tests, carbon deposit and sticking of piston rings were observed. [Bhattacharyya 1994]

Palm Oil

The palm oil research institute of Malaysia converted crude palm oil to methyl esters. These were referred to as 'palm oil diesel'. 'Palm oil diesel' and diesel blends of 25%-75%, 50%-50%, 75%-25% proportions were tested. The blend with the ratio of 1:1 by volume (50% and 50% diesel oil) produced the best performance for power, torque and specific fuel consumption. The performance of palm oil esters synthesized from crude palm oil, refined, bleached and deodorized olien (RBD), and crude palm stearin (CPS) showed a performance at par with Malaysian grad II petroleum diesel fuels. Except for the higher specific gravity, viscosity and narrowed boiling range, their properties closely matched that of diesel fuel (Bhattacharyya and Reddy, 1994).

Cottonseed Oil

Blumberg et al. (1982) compared the performance, gaseous emissions and limited cyclic durability (200h) of pure cottonseed oil and transesterified cottonsed oil (methyl esters) with diesel in a turbocharged, open chamber diesel engine. Performance and emission characteristics of cottonseed oil in a diesel engine were compared with diesel, sunflower, soybean and peanut oil.

Mazed et al. (1995) carried out short term tests burning blends of cottonseed oil in diesel engines, one with a direct fuel injection system and another with an indirect fuel injection system. Cottonseed oil in its pure form and also as 10% and 25% blends with diesel were tested. Results from the maximum power and fuel

consumption tests indicated decreasing power output as the percentage of cotton seed oil was increased in the blend. This resulted in low heat content and increased viscosity of blends compared to pure diesel. Further, fuel economy was higher with two stage combustion than with single-stage combustion system. [Bhattacharyya 1994]

Neem Oil

Bansal and Juneja (1993) first reported on the performance of typical Indian non-edible oil, neem oil. Short term tests were carried out using neem oil and diesel oil blends of 10, 20 and 30% in diesel engine. Viscosities of these blends were found to be nearly equal to the viscosity of diesel fuel at 25, 30, 44 and 60°C respectively. The authors suggested heating of fuel by recirculation the exhaust to avoid cold starting problems during winter. The distillation curves revealed that thermal cracking of neem oil blends took place earlier than diesel fuel and thus resulted in more smoke, particulates and fouling of lubricants if the blend has more than 30% neem oil. Long term endurance tests, however, were not carried out. [Bhattacharyya 1994]

Coconut Oil

Masjuki [2001] carried out dynamometer tests to evaluate performance, emission and wear characteristics of an indirect injection diesel engine when fueled by 10, 20, 30, 40 and 50% blends of coconut oil with diesel oil. The test was conducted for 100 h using each of the test fuels to monitor the effect of coconut oil blends on the wear and lubricating oil performance. Diesel oil was also used for comparison purposes.

The performance and emissions characteristics results showed that 10–30 per cent coconut oil blends produced slightly higher performance in terms of brake power than diesel oil. All the Coconut oil blends produced lower exhaust emissions including polycyclic aromatic hydrocarbons and particulate matter. The wear and lubrication oil characteristics results showed that Coconut oil blends up to 30 per cent produced similar results to diesel oil.

Sesame Oil

Haq [1995] carried out short term tests burning blends of sesame oil containing 50% kerosene and 50% sesame in a diesel engine. The engine performance with these blends was close to that of diesel.

Waste Vegetable Oil

Dorado et. al [2002] tried to determine the feasibility of running a 10% waste vegetable oil-90% diesel fuel blend during a 500-h period in a 3-cylinder direct-injection, 2500 cm³ Diter diesel engine. A waste vegetable oil-diesel fuel blend was recycled used frying oil, which was essentially a waste product. The long-term performance of this fuel was monitored by measuring the viscosity of the lubricating oil, abnormal functioning of the engine, power loss, and excessive smoke output compared to straight diesel fuel. The results revealed an approximately 12% power loss, slight fuel consumption increase, and normal smoke emissions. Combustion efficiency dropped slightly during the testing period. It can be concluded that the Diter diesel engine, without any modifications, ran successfully on a blend of 10% waste oil-90% diesel fuel without externally apparent damage to the engine parts. Nevertheless, it appears that the long-term use of waste oil blended with diesel fuel will need further testing before use as a viable energy solution.

2.4 Required Properties of Vegetable Oils

Fuel properties of vegetable oils for the suitability as an alternative fuel can be grouped conveniently into physical, chemical and thermal properties as follows [Demirbas, 1998]:

- 1. Physical properties:** include viscosity, density, cloud point, flash point, pour point, boiling range, freezing point and refractive index.
- 2. Chemical properties:** include chemical structure, acid value, saponification value, ash and sulfur contents, sulfur and copper corrosion, water and sediment residuum, oxidation and oxidation resistant and thermal degradation products.
- 3. Thermal properties:** are distillation temperature, thermal degradation point, carbon residue, specific heat content value, thermal conductivity, etc.

2.5 Composition of Vegetable Oils

Fats and oils are water-insoluble, hydrophobic substances in the plant and animal kingdom composed primarily of the fatty esters of glycerol, so called triglycerides. Structurally a triglyceride is the reaction product of one molecule of glycerol with three fatty acid molecules to yield three molecules of water and one molecule of triglyceride. The major components of vegetable oils are triglycerides. Triglycerides are esters of glycerol with long chain acids, called fatty acids. Fatty acids can be saturated or unsaturated depending upon the bonds between two adjacent carbon atoms. If there is any double bond between two carbon atoms then it is called unsaturated and if more than one double bond is present in the fatty acid then it is termed polyunsaturated.

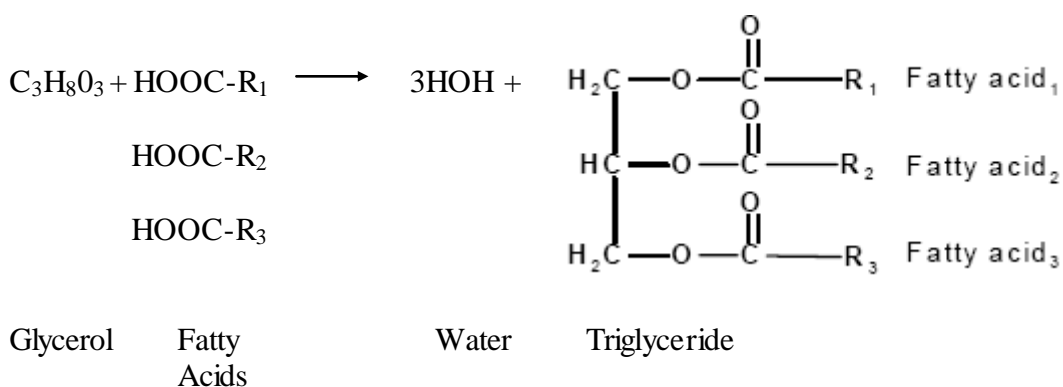


Fig. 2.1: Chemical structure of vegetable oil

Vegetable oil	16:0	18:0	20:0	22:0	24:0	18:1	22:1	18:2	18:3
Corn	11.67	1.85	0.24	0.00	0.00	25.16	0.00	60.60	0.48
Cottonseed	28.33	0.89	0.00	0.00	0.00	13.27	0.00	57.51	0.00
Crambe	2.07	0.70	2.09	0.80	1.12	18.86	58.51	9.00	6.85
Peanut	11.38	2.39	1.32	2.52	1.23	48.28	0.00	31.95	0.93
Rapeseed	3.49	0.85	0.00	0.00	0.00	64.40	0.00	22.30	8.23
Soybean	11.75	3.15	0.00	0.00	0.00	23.26	0.00	55.53	6.31
Sunflower	6.08	3.26	0.00	0.00	0.00	16.93	0.00	73.73	0.00
Coconut	13.1	3.9	0.00	0.00	0.00	52.8	0.00	30.2	0.00

Table 2.1: Common fatty acids in some common vegetable oil [Demirbas, 1998]

Chemical Names and Descriptions of some Common Fatty Acids				
Common Name	Carbon Atoms	Double Bonds	Scientific Name	Sources
Caprylic Acid	8	0	octanoic acid	coconut oil
Capric Acid	10	0	decanoic acid	coconut oil
Lauric Acid	12	0	dodecanoic acid	coconut oil
Myristic Acid	14	0	tetradecanoic acid	palm kernel oil
Palmitic Acid	16	0	hexadecanoic acid	palm oil
Oleic Acid	18	1	9-octadecenoic acid	olive oil
Linoleic Acid	18	2	9,12-octadecadienoic acid	grape seed oil
Alpha-Linolenic Acid (ALA)	18	3	9,12,15-octadecatrienoic acid	Flaxseed (linseed) oil
Gamma-Linolenic Acid (GLA)	18	3	6,9,12-octadecatrienoic acid	borage oil
Arachidic Acid	20	0	eicosanoic acid	peanut oil
Behenic acid	22	0	docosanoic acid	rapeseed oil
Erucic acid	22	1	13-docosenoic acid	rapeseed oil

Table 2.2: Lists of the most Common fatty acids and their methyl esters

The various vegetable oils are distinguished by their fatty acid composition. Petroleum based diesel fuels have different chemical structures from vegetable oils. The former contain only carbon and hydrogen atoms which are arranged in normal

(straight chain) or branched chain structure, as well as aromatic configurations. The normal structure preferred for better ignition quality. Diesel fuel can contain both saturated and unsaturated hydrocarbons, but the later are not present in large enough amounts to make fuel oxidation a problem. The aromatics present are generally oxidation resistant. On the other hand, in the case of vegetable oils, oxidation resistance is markedly affected by the fatty acid composition. The large size of vegetable oil molecules and the presence of oxygen in the molecules suggest that some fuel properties of vegetable oils would differ markedly from those of hydrocarbon fuels [Ali 1994]

2.6 Standard Fuel Properties of Vegetable Oils

While engineers and enthusiasts have been experimenting with using vegetable oils as fuel for a diesel engine since at least 1900, it is only recently that the necessary fuel properties and engine parameters for reliable operation have become apparent. Most of the standards were developed through the cooperative efforts of the American Society for Testing Materials (ASTM), the Society of Automotive Engineers (SAE), and American Petroleum Institute (API). Only the most important standards are given in Appendix E

CHAPTER 3

EXERGY ANALYSIS OF HEAT ENGINES AND ITS FUELS

3.1 Basic Concepts of Exergy

It has long been understood that traditional first-law analysis, which is needed for modeling the engine process, often fails to give the engineer the best insight into the engine's operation. In order to analyze engine performance- that is, to evaluate the inefficiencies associated with the various processes – second law analysis must be applied. For second law analysis, the key concept is 'exergy' (or availability). Unlike energy, exergy can be destroyed which is a result of such phenomena as combustion, friction, mixing and throttling. The exergy of a system in a given state can be defined as the maximum useful work that can be produced through interaction of the system with its surroundings, as it reaches thermal, mechanical and chemical equilibrium. Usually, the terms associated with thermomechanical and chemical equilibration are differentiated and calculated separately. [Rackopoulos, 2006]

Exergy is an extensive property with a value greater than or equal to zero. Exergy is a property, the value of which depends not only on the state of the system, but also on the ambient properties. There is no exergy in a system when thermal, mechanical and chemical equilibrium exists with the environment. Thermal equilibrium is achieved when the temperature of the system is equal to the temperature of the surrounding environment. In the same way mechanical equilibrium is achieved when there is no pressure difference between the working medium and the environment. [Rackopoulos, 2006]

3.2 Defining Exergy

When a new energy source, such as a geothermal well is discovered, the first thing the explorers do is to estimate the amount of energy contained in the source. This information alone, however, is of little value in deciding whether to build a power plant on that site or not. What one really needs to know is the work potential of the source- that is, the amount of energy one can extract as useful work. Thus, it would

be very desirable to have a property to enable the engineers to determine the useful work potential of a given amount of energy at some specified state. This property is exergy, which is also called the availability or available energy.

The work potential of the energy contained in a system at a specified state is simply the maximum useful work that can be obtained from the system. The work done during a process depends on the initial state, the final state and the process path. In an exergy analysis, the initial state is specified, and thus it is not a variable. The work output is maximized when the process between two specified states is executed in a reversible manner. Therefore, all the irreversibility are disregarded in determining the work potential. Finally, the system must be in dead state at the end of the process to maximize the work output.

In general, a system is considered to be at the so-called 'restricted' dead state when no work potential exists between the system and the environment due to the temperature difference or pressure differences. This is the dead state reached when calculating the thermomechanical availability. Some researchers in the field of second law application to internal combustion engines, including the present ones, define the restricted dead state (for any given state) to have the same chemical composition as the given state (thus no work potential exists due to compositional differences); whereas, some authors define the restricted dead state to be the chemical equilibrium state of the given state's components at p_o and T_o (but not in chemical equilibrium with environment components). On the other hand, if chemical equilibrium with the environment is of concern, then 'true' or 'unrestricted dead state' is referred, where the chemical potential of the system also equals to the environment. For engine applications the pressure and temperature conditions of the dead state are usually taken to be $p_o = 1.0325$ bars (1 atm) and $T_o = 298.15$ K [Rackopoulos, 2006]

3.3 Second-Law or Exergetic Efficiency of Heat Engines

Efficiency is defined in order to be able to compare different engine size applications or evaluate various improvement effects, either from the first or second law perspective. The consideration of energy quality leads to the concept of exergy. Energy efficiency does not include the usability of the converted energy. Exergy efficiency includes the quality level of the converted energy. Since exergy has its origin in the second law, a performance parameter for a process based on exergy concepts is known as second law or exergetic efficiency η_{II} .

Unlike first-law efficiencies, the second-law ones weigh the variable energy terms according to their capability for work production. Moreover, a second law efficiency includes, in addition to exergy losses (e.g. in exhaust gases) the exergy destructions (irreversibility) too. On the other hand, because energy is conserved, first law efficiencies reflect only energy losses. Moreover, energy losses are not representative of the usefulness of loss. And first law efficiencies do not explicitly penalize the system for internal irreversibility. The first-law is a conservation principle, and first-law efficiencies tend to fall into two general categories. Equipment first-law efficiencies compare actual energy changes to theoretical energy changes under specified conditions. Examples include turbines, compressors, nozzles and pumps. Cycle first-law efficiencies compare desired energy output to required energy input. Thermal efficiency and coefficient of performance (COP) are typical examples. On the other hand, entropy and availability from a second-law viewpoint are non-conserved properties. In the presence of irreversibility, entropy is produced and availability is destroyed. The former effect is measured by entropy production, and the latter effect is measured by the irreversibility. Hence second-law efficiencies measure losses in availability during a process. A second approach to define second law efficiency is especially useful for steady-state devices as:

$$\eta_{II} = \frac{\text{exergy output as work}}{\text{exergy input}}$$

The second law stresses the fact that two forms of the same quantity of energy may have quite different availabilities. The energy is ‘weighted’ according to its availability. Unlike a first-law efficiency, an effectiveness accounts for losses in work capability during a process.

There are various definitions available in the literature for the efficiency based on second law analysis of processes. A general definition for the availability efficiency is given as follows [Rakopoulos, 2006]:

$$\eta_{II} = \frac{\sum Av_{\text{utility generated}} + \sum \Delta Av_{\text{nonsource and nonpurge streams}}}{\sum Av_{\text{utility supplied}} - \sum \Delta Av_{\text{availability source streams}}}$$

where utility includes heat, electricity, etc.

The availability efficiency of a heat engine is written by using the following equation as:

$$\eta_{II} = \frac{AV_{\text{utility generated}}}{AV_{\text{utility supplied}}}$$

Assuming the work generated is converted into 100% electricity [Rakopoulos, 2006]:

$$\eta_{II} = \frac{W_{\text{generated}}}{Q_H \left(1 - \frac{T_0}{T_H}\right) - Q_C \left(1 - \frac{T_0}{T_C}\right)}$$

where the heat engine is assumed to consist of one hot temperature (T_H) reservoir and one cold temperature reservoir (T_C). Q_H and Q_C represent heat added and removed, respectively.

3.4 Chemical Exergy and Thermochemical Flow of Exergy

The flow exergy of a substance is the theoretically obtainable work when the substance is brought into total equilibrium with the local environment. It can be split into chemical exergy and thermomechanical flow exergy,

$$\text{Exergy} = \text{Chemical Exergy} + \text{Thermochemical flow Exergy}$$

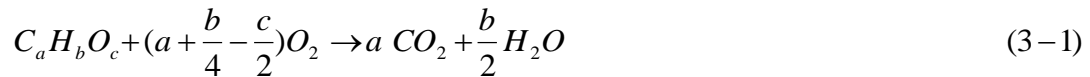
The thermomechanical exergy (also known as physical exergy) represents the deviation in temperature and pressure between the flowing matter and the ambient. It also includes the potential and kinetic energy, although these contributions are not treated in the present study. The chemical exergy represents the deviation in chemical composition between the flowing matter and the local environment. The molar thermomechanical flow exergy is expressed as

$$\text{Thermochemical Exergy} = h - h_0 - T_0 (s - s_0)$$

where h and s are the molar enthalpy and entropy, respectively, of the flowing matter, while $h_0 = h(T_0, p_0)$ and $s_0 = s(T_0, p_0)$

3.5 Determining Fuel Chemical Exergy

We limit our attention only to chemical exergy of fuel, because exergy supplied to a heat engine comes principally from the chemical exergy of fuel. The stoichiometric reaction between a fuel and oxygen can be written as



For hydrocarbons $c=0$, for hydrogen $a=c=0$ and $b=2$, while for carbon monoxide $b=0$ and $a=c=1$. The chemical exergy is determined from

$$E_i^{Ch} = -\Delta G(T_0, p_0) + \sum v_j E_j^{Ch} \quad (3-2)$$

where the chemical exergies of all products and co-reactants are evaluated at ambient temperature and pressure. Water H_2O is evaluated in gaseous state. The change of Gibbs function for the reaction is

$$\Delta G(T, p) = \sum_j v_j g_j(T, p) \quad (3-3)$$

when reactants are supplied and products released separately each at the relevant state (T, p) . Here $g_j(T, p)$ is the molar Gibbs function of each pure species and v_j is the stoichiometric coefficient of each species in the reaction, with negative values for the reactants and positive for products thus

$v_{C_a} v_{H_b} v_{O_c} = -1, v_{O_2} = -(a + b/4 - c/2), v_{CO_2} = a, \text{ and } v_{H_2O} = b/2$, Correspondingly the entropy and enthalpy changes are,

$$\Delta S(T, p) = \sum_j v_j s_j(T, p) \quad (3-4)$$

$$\Delta H(T, p) = \sum_j v_j h_j(T, p), \quad (3-5)$$

Where respectively $S_j(T, p)$ and $h_j(T, p)$ are the molar entropy and enthalpy of each pure species. For fuels the quantity of equation (3-5) equals the negative of the lower heating value (LHV), h_{LHV} , at the specified state. The relation between the three quantities is

$$\Delta G(T, p) = \Delta H(T, p) - T\Delta S(T, p) \quad (3-6)$$

Data for the molar Gibbs function molar entropy molar enthalpies are available for a certain reference state (T_{ref}, p_{ref}) , usually 25°C (298.15K) and 1 atm (101235 pa). similarly the molar enthalpies and entropies are available at this reference state. The values of Gibbs function of formation are found from JANAF Thermochemical Tables¹.

For deviating conditions the expressions of equations (3-3) and (3-5) assuming as ideal gases are elaborated as

$$-\Delta G(T_0, p_0) = -\Delta G^{ref} + (T_{ref} - T_0)(-\Delta S^{ref}) + W_1 + W_2 \quad (3-7)$$

$$-\Delta H(T_0) = h_{LHV}(T_0) = -\Delta H^{ref} \sum_j v_j \int_{T_{ref}}^{T_0} c_{p,j}(T) dt \quad (3-8)$$

Here the notation $\Delta G(T_{ref}, p_{ref}), \Delta S^{ref} = \Delta S(T_{ref}, p_{ref}),$ and $\Delta H^{ref} = \Delta H(T_{ref})$ is used for shortness

$$W_1 = \sum_j v_j \int_{T_{ref}}^{T_0} c_{p,j}(T) \left(1 - \frac{T_0}{T}\right) dt \quad (3-9)$$

$$W_2 = T_0 R \sum_j v_j \ln \frac{p_0}{p_{ref}} \quad (3-10)$$

Thus the chemical exergy at a certain state can be expressed as

$$E_i^{ch} = -\Delta G^{ref} + (T_{ref} - T_0)(-\Delta S^{ref}) + W_1 + W_2 + \sum_{j \neq i} v_j E_j^{ch} \quad (3-11)$$

Where the summation in the last term is made over products and co-reactants (here CO_2, H_2O and O_2) but not the substance itself. According to Szargut[1988], the terms here denoted W_1 and W_2 can be neglected. Using equation (3-6), alternative expressions of the two first right hand side terms of equation (3-7) are obtained

$$-\Delta G^{ref} + (T_{ref} - T_0)(-\Delta S^{ref}) = -\Delta G^{ref} \frac{T_0}{T_{ref}} + \frac{T_{ref} - T_0}{T_{ref}} (-\Delta H^{ref}), \quad (3-12)$$

$$-\Delta G^{ref} + (T_{ref} - T_0)(-\Delta S^{ref}) = -\Delta H^{ref} - T_0 (-\Delta S^{ref}) \quad (3-13)$$

The molar chemical exergy of an atmospheric gas is expressed as

$$E_i^{ch} = -RT_0 \ln X_i^e \quad (3-14)$$

Where R is the universal gas constant. This is equal to the work obtained from a reversible isothermal T_0 expansion from atmospheric pressure (p_0) to the atmospheric partial pressure (p_i^e).

By using equation (3-12) and (3-14) the chemical exergy from equation (3-11) can be re-expressed in terms of the chemical exergy at the reference condition-

$$E_i^{ch} = E_i^{ch,ref} \frac{T_0}{T_{ref}} + \frac{T_{ref} - T_0}{T_{ref}} (-\Delta H^{ref}) + W_1 + W_2 + T_0 R \sum_{j \neq i} v_j \ln \frac{X_j^{ref}}{X_j^e} \quad (3-15)$$

Here $E_i^{ch,ref}$ and X_j^{ref} are the values at the reference state (T_{ref} , p_{ref}). For a substance with $c < b/2$ in equation (3-1), the last term will be positive when X_j^e is less than X_j^{ref} . Hence, for hydrocarbon, this term will contribute to a larger chemical exergy for a lower ambient temperature.

3.6 Chemical Exergy of Higher Hydrocarbons

From equation (3-15) the chemical exergy of pure liquid fuels can be ascertained when vapor-pressure data are available. The steady-flow fuel exergy of light hydrocarbon gases and their mixture can be evaluated accurately when the composition is known. For determining chemical exergy of heavy hydrocarbon fuels which are pure or of undetermined composition latter substance data are usually lacking to calculate E_i^{ch} .

Szargurt and Petela studied that for large number of light hydrocarbons, the ratio of E_i^{ch} to the lower heating value (LHV) is reasonably constant. [Szargurt, 1988]. The ratio E_i^{ch}/LHV homologous series of hydrocarbons was also examined by Brzustowski and Brena. They determined that the proportionality constant between fuel exergy and LHV is close to 1.065 based on E_i^{ch} data at 56 percent relative humidity.[Brzustowski,1986]. These interpretations are only appropriate for heavy hydrocarbon of undetermined composition. [Al-Najem and Diab, 1991] used this approximation method and it was denoted by A_{in} . As some empirical relations for

measuring the fuel chemical exergy have been adopted in this study so, from now on, this Ex_{in} notation has been used instead of E_i^{ch}

The following relationship is used for computing the chemical exergy of liquid hydrocarbons having the general formula $C_xH_yO_zS_w$. [Szargurt,1988]

$$Ex_{in} = Q_{in} \left[1.0374 + 0.0159 \frac{y}{x} + 0.0567 \frac{z}{x} + 0.5985 \frac{w}{x} \left(1 - 0.1737 \frac{y}{x} \right) \right] \quad (3-16)$$

Where $Q_{in} = \dot{m}_f \cdot LHV$

A correlation for liquid hydrocarbons having the simple formula of C_xH_y was proposed by Moron [Moron,2000]

$$Ex_{in} = Q_{in} \left[1.04224 + 0.011925 \frac{y}{x} - \frac{0.42}{x} \right] \quad (3-17)$$

CHAPTER 4

METHODOLOGY OF THE STUDY

4.1 Fuel Property Test

The use of waste soybean oil and coconut oil as alternative to diesel depends on their fuel properties. The fuel property such as viscosity, heating value was evaluated by ASTM standards. The viscosity of waste soybean oil and coconut oil was found to be far greater than that of diesel and therefore, make it unsuitable for use as fuel in diesel engines. Because of this reason waste soybean oil and coconut oil was blended with diesel. The blend was prepared by mixing 80% waste vegetable oil and coconut oil with 20% diesel fuel by volume. Kinematic Viscosity of fuel was evaluated by ASTM D445 standard. On the other hand heating value of waste soybean oil and coconut oil was found less than the diesel. Heating Value of fuel was evaluated by ASTM D 240.

Heating Value of Diesel, WVO and Coconut oil are as follows

<i>Properties</i>	<i>Unit</i>	<i>Diesel</i>	<i>WVO</i>	<i>Coconut</i>
Calorific Value (LHV)	MJ/Kg	42.97	37.75	36.54

4.2 Engine Performance Test

4.2.1 Experimental setup

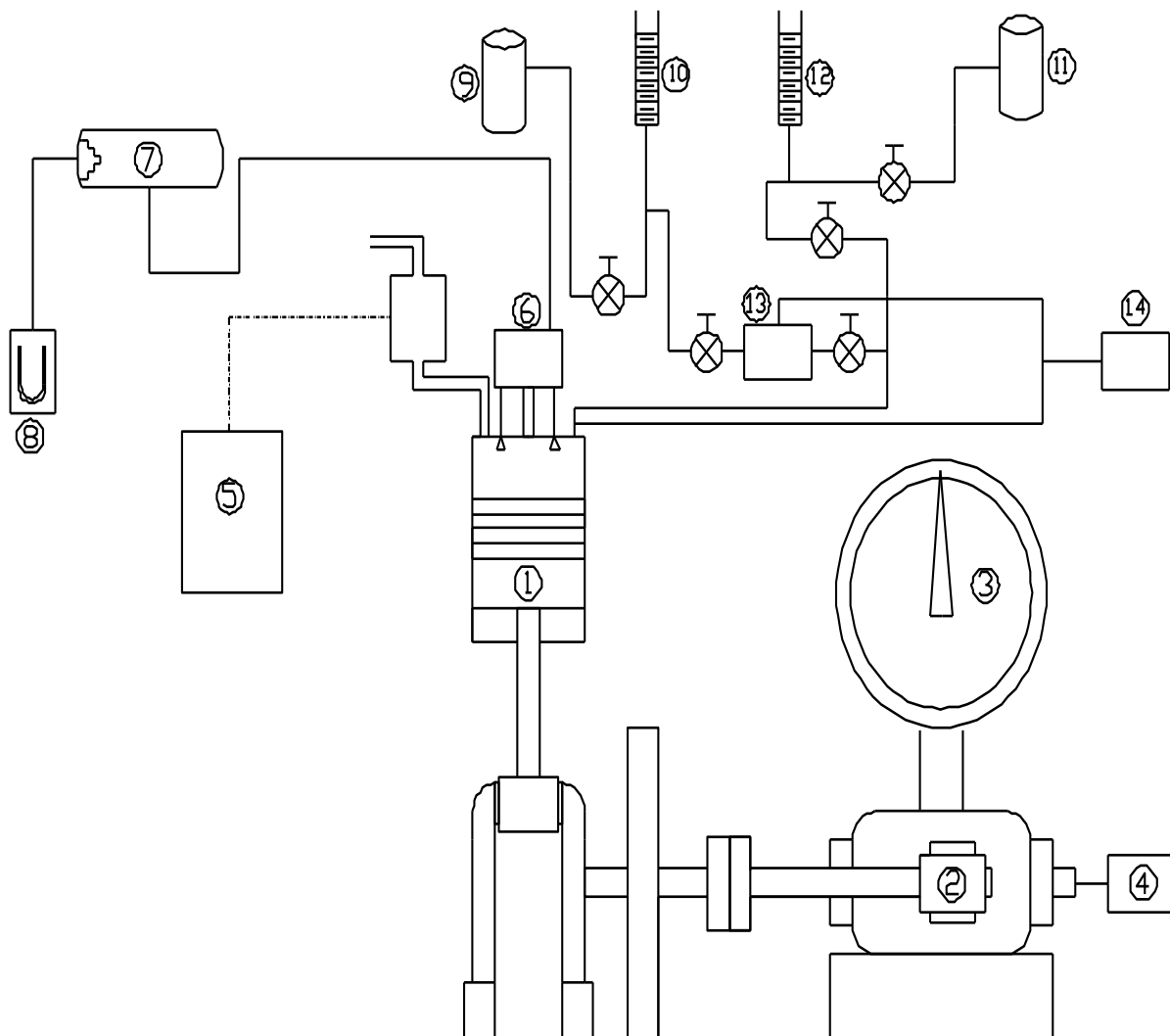
In this study, the experimental set up consists of a diesel engine, engine test bed with a water brake type dynamometer which was used to apply desired load on the engine, waste soybean oil, and coconut oil supply system, different along with the test engine. Waste soybean oil, and coconut oil feed system consists of an electric heater placed in a heater box. Oil is allowed to pass through the heater which is controlled by a temperature controller circuit which senses the inlet temperature of the oil before it enters the fuel injection pump. The schematic view of the experimental set up is shown in Fig. 5.1 Experiments are carried out on a single cylinder diesel engine

of model R175A having a rated output of 4 kW at a rated speed of 2600 rpm. In this study, BS standards for engine performance test BS 5514: Part I: 1982, equivalent to ISO 3046 and J 1349, ISO and SAE standards for the same respectively, has been followed. Any other additional guidelines required are taken from the procedures used by Plint and Böswirth [1986].

The gravitational force feeds the fuel to the injector. A coil is inserted into the waste soybean oil and coconut oil container for the purpose of heating the oil up to a desired temperature. A controller circuit arrangement is there to sense the temperature of the oil just before entering the injector. By sensing the temperature of oil entering the injector the controller the controller cuts off or connects the electric circuit to raise the temperature up to the desired limit. A sensor is used for this purpose at the fuel line just before the injector rail which is connected to the controller circuit through an overload relay. The overload relay is nothing but a transducer which is sensitive to thermal change in the form of voltage change. Whenever any rise in fuel temperature over the set temperature is sensed by the sensor, the voltage difference is changed in the thermal overload relay and at the instant the relay breaks the circuit.

4.2.2 Engine test procedure

Model R175A test engine is run by diesel fuel at three different speeds of 2600, 2400 and 2200 rpm and all the necessary parameters are recorded and different performance parameters are calculated to check the engine's reliability as a test engine. Then the performance tests are carried out using 100% waste soybean oil preheated to 80⁰C and 100⁰C and 80% waste vegetable oil blended with 20% diesel preheated to 100⁰C. Engine speed has been maintained within ± 5 rpm and the temperature has been maintained within $\pm 2^0$ C of the desired temperature.



- | | |
|--------------------------------------|---|
| 1. Engine | 8. Manometer |
| 2. Dynamometer | 9. Waste Soybean Oil Tank or Coconut Oil Tank |
| 3. Load Indicator | 10. Burette of Vegetable Oil Tank |
| 4. Speed Sensor | 11. Diesel Tank |
| 5. Exhaust Gas Temperature Indicator | 12. Burette of Diesel Tank |
| 6. Intake Manifold | 13. Heater Box |
| 7. Air Drum | 14. Temperature Controller |

Fig. 4.1: Schematic diagram of the experimental setup

4.3 Different Parameters Measuring Technique

4.3.1 Engine brake power and speed

To apply desired loads and to measure engine brake power a water brake type dynamometer, model no. AN3e has been used. Usually the brake is equipped with a pendulum balance. The torque of the brake housing is transmitted to the balance and indicated on a circle scale by a pointer. The units of measure can be force in Kg and torque in Kgm. An optical type tachometer is used to measure the engine speed.

4.3.2 Temperatures

To measure the exhaust gas temperature near the bottom of the silencer, a K-type thermocouple has been inserted. Lubricant temperature is measured using a K-type thermocouple probe inserted into the lubricant oil sump. All temperatures are measured using a digital meter (OMEGA-K), connected to the K-type probes via a selector switch.

4.3.3 Fuel consumption rate

Fuels were supplied from a graduated burette instead of fuel tank when needed. Fuel consumption rate has been recorded by a stop watch for 50cc of fuel.

4.3.4 Air flow rate

To measure air flow rate air was drawing through one circular nozzles of 13 mm diameter ($C_D = 0.92$) with an air drum of standard size (in accordance with Plint and Böswirth) that is connected to the engine air inlet and pressure drop is measured by means of a manometer, using water as the manometric fluid.



Fig. 4.2: Picture of the water brake type dynamometer, model no. AN3e



Fig. 4.3: Picture of the diesel engine model R175A

4.4 Estimation of Engine Performance Parameters

4.4.1 Volumetric efficiency, η_v

One of the most important processes that govern how much power and performance can be obtained from an engine is getting the maximum amount of air into the cylinder during each cycle. More air means more fuel can be burned and more energy can be converted to output power. [Pulkrabek, W. W, 2002] Volumetric efficiency η_v is introduced to account for the effectiveness of the induction and exhaust processes. In terms of quantity applied to an actual engine, it is defined as the mass of charge inducted into the cylinder divided by the mass of the mixture that would fill the piston displacement volume at inlet air density in the intake manifold. For four stroke CI engines, volumetric efficiency is given by:

$$\eta_v = \left(\frac{1}{60} \right) \left(\frac{2\dot{m}_a}{NV_d\rho} \right) \quad (4-1)$$

Where \dot{m}_a : mass of fresh air inducted per unit time in kg/h

N : engine speed in rpm

V_d : swept volume of all of the engine cylinders in m^3

ρ_i : density of inlet air at intake manifold in kg/m^3

4.4.2 Brake specific fuel consumption, $bsfc$

Brake specific fuel consumption is a measure of engine efficiency. In fact, $bsfc$ and engine efficiency are inversely related, so that the lower the $bsfc$ better the engine. It is the rate of consumption of fuel per unit time and per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work. The brake specific fuel consumption $bsfc$ can be written as

$$bsfc = \frac{\dot{m}_f}{P_b} \quad \left[\frac{kg_f}{kWh} \right] \quad (4-2)$$

where \dot{m}_f : mass of fuel consumed per unit time in kg/h

P_b : brake power in kW, measured at engine flywheel

4.4.3 Brake thermal efficiency, η_b

Brake thermal efficiency is a dimensionless performance measure of a device that uses the engine output to the necessary fuel input. The ratio of the brake power output to the rate of fuel energy input is known as brake thermal efficiency η_b . $bsfc$ and η_b are inversely related to each other. It is given by

$$\eta_b = 3600 \times \frac{P_b}{\dot{m}_f \cdot LHV} = \frac{3600}{bsfc \cdot LHV} \quad (4-3)$$

where P_b : brake power measured at fly wheel in kW

\dot{m}_f : mass flow rate of fuel in kg/h

LHV : the lower heating value of fuel in kJ/kg

4.4.4 Brake mean effective pressure, $b MEP$

Brake mean effective pressure is acting on the piston area through stroke would produce brake power at the flywheel. Therefore,

$$P_b = b MEP \cdot \frac{V_d N}{2} \quad (4-4)$$

Brake mean effective pressure $b MEP$ is a true indication of the relative performance of different engines.

4.4.5 Fuel chemical exergy or exergy input, Ex_{in}

For both diesel and soybean oil fuel, the fuel exergy input is calculated by using

Eq. (3-16):

$$Ex_{in} = Q_{in} \left[1.0374 + 0.0159 \frac{y}{x} + 0.0567 \frac{z}{x} + 0.5985 \frac{w}{x} \left(1 - 0.1737 \frac{y}{x} \right) \right]$$

In the present study, the formula of diesel is taken as $C_{14.4}H_{24.9}$ (Ferguson and Kirkpatrick, 2001). So, putting $x = 14.4$ and $y = 24.9$, the above correlation for diesel fuel becomes

$$Ex_{in} = 1.06489 Q_{in} \quad (4-5)$$

In the present study, the chemical formula of soybean oil fuel has been taken as $C_{56}H_{102}O_6$ and of coconut oil as $C_{12}H_{24}O_2$ (Altin et al., 2001). So, putting $x = 56$ and $y = 102$, $z = 6$ and $w = 0$ in the above correlation for soybean oil fuel and $x = 12$ and $y = 24$, $z = 2$ and $w = 0$ for coconut oil becomes

$$(Ex_{in})_{wvo} = 1.0724Q_{in} \quad (4-6)$$

$$(Ex_{in})_{coconut} = 1.0786Q_{in} \quad (4-7)$$

4.4.6 Exergy destruction in exhaust gas

Al-Najem and Diab [1992] calculated the exergy of the exhaust gases from

$$\begin{aligned} Ex_{exst} &= \dot{m}_{eg} [(h_{e,0} - h_0) + T_0 (S_{e,0} - S_0)] \\ &= Q_{eg} + \dot{m}_{eg} T_0 \left[C_{p,e} \ln \left(\frac{T_0}{T_{e,0}} \right) - R \ln \left(\frac{P_0}{P_{e,0}} \right) \right] \\ &= Q_{eg} + \dot{m}_{eg} T_0 \left[C_{p,e} \ln \left(\frac{T_0}{T_{e,0}} \right) \right] \quad \text{as } P_0 \cong P_{e,0} \end{aligned} \quad (4-8)$$

So,

$$Ex_{exst} = Q_{eg} + \dot{m}_{eg} T_0 \left[C_{p,e} \ln \left(\frac{T_0}{T_{e,0}} \right) \right]$$

where \dot{m}_{eg} : mass flow rate of exhaust gases = $\dot{m}_f(1 + A/F)$

Q_{eg} : heat energy of exhaust gases = $C_{p,e} \dot{m}_f(1 + A/F)(T_{exst} - T_{db})$

$C_{p,e}$: specific heat of the exhaust gas

The specific heat of exhaust is calculated using the formula of specific heat of mixture of gases as follows:

$$C_{p,e} = \frac{\sum_{i=1}^n N_{i,00} C_{p,i}}{\sum_{i=1}^n N_{i,00}} = \frac{N_{CO_2} C_{p,CO_2} + N_{H_2O} C_{p,H_2O}}{N_{CO_2} + N_{H_2O}}$$

assuming that complete combustion of both the diesel fuel and soybean oil fuel produce only carbon dioxide (CO_2) and water vapor (H_2O). The constant-pressure

specific heat equations of the individual components of exhaust gas are adapted from the data from Gordon and McBride, [NASA SP-273, March1976]. These are as follows:

$$C_{p,CO_2} = (2.401 + 8.735 \times 10^{-3}T - 6.607 \times 10^{-6}T^2 + 2.002 \times 10^{-9}T^3)R_u$$

$$C_{p,H_2O} = (4.0 - 1.108 \times 10^{-3}T + 4.152 \times 10^{-6}T^2 - 2.964 \times 10^{-9}T^3 + 0.807 \times 10^{-12}T^4)R_u$$

where R_u is the universal gas constant.

4.4.7 Second-Law Efficiency, η_{II}

The second-law (or exergy) efficiency is defined as the ratio of the maximum possible useful work output from the heat engine cycle to the fuel exergy input to the engine.

$$\eta_{II} = \frac{\text{maximum useful work output}}{\text{fuel chemical availability}} = \frac{w_{\max}}{E_i^{ch}} \quad (4-9)$$

But, due the complexity of calculation and lack of thermochemical data for soybean oil the brake power at engine flywheel is used instead of the maximum possible useful work output and the phrase ‘percent availability output at shaft’ ($\eta_{ex,shaft}$) is used instead of the term ‘second-law efficiency’. So, in the present study the second-law efficiency is calculated as:

$$\eta_{ex,shaft} = \frac{\text{brake power output}}{\text{exergy input}} = \frac{P_b}{Ex_{in}} \quad (4-10)$$

CHAPTER 5

RESULTS AND DISCUSSIONS

5.1 Introduction

Results are discussed in the following two sections

- 1) Results of fuel testing
- 2) Results of engine performance

A series of performance tests were carried out at three different speeds (2600, 2400, and 2200 rpm) under variable loading condition using diesel fuel. The experiments are repeated at the speed of 2600 rpm with waste soybean oil (termed 'WVO' from now on) and blend of WVO and raw diesel in 80:20 volumetric basis preheated at two different temperatures (80°C and 100°C). The experiments are repeated at the same speed with pure coconut oil and blend of pure coconut oil and raw diesel in 80:20 volumetric basis preheated at the same temperatures. The data obtained are analyzed to produce performance parameters from the viewpoints of first law and the second law of thermodynamics.

5.2 Results of Fuel Testing

The variation in viscosity of oils with temperature is shown in Fig. 5.1(a) and 5.1(b). The viscosity of WVO and coconut oil is nearly 10-12 times higher than that of diesel fuel at room temperature (27°C). But when WVO is heated up then its viscosity reduces significantly, viscosity of WVO and diesel blends decrease with increase in temperature and becomes close to that of diesel at temperature above 80°C. Coconut oil and its blend with diesel show similar characteristics.

The percentage of change in viscosity of diesel, WVO and WVO- diesel blend shows a gradually downward trend. In each of the three cases, percentage change in

viscosity falls in the range of 70-85 percent at 100°C with respect to the room temperature value. The highest change is observed in case of WVO-diesel blend and lowest for diesel. Coconut oil and its blend with diesel show similar characteristics.

In Fig. 5.2(a) and 5.2(b), variations of fuel density with temperature are shown. It is found that the densities of WVO and coconut oil are higher than that of diesel at room temperature. When WVO and coconut oil is blended with diesel and heated up to 100°C, its density becomes close to diesel fuel. However the percentage changes in density values are not as significant as that of viscosity. The highest change in this case is about 7%.

5.3 Results of Engine Performance

Variations of air flow rate, m_a with brake power, P_b of the test engine running with diesel fuel at three different speeds of 2600, 2400 and 2200 rpm at different loading conditions are shown in Fig.5.3. It is found that air flow rate increases with engine speed for the same output power because of swept volume per unit time is higher for higher engine speed. On the other hand, with the increase of engine power, mass flow rate of air decreases slightly due to increase of intake manifold temperature which reduces intake air density thereby reducing the induction air mass flow rate.

Volumetric efficiency is affected by the following factors relating to fuel, engine design and engine operating variables:

1. Fuel type, fuel/air ratio, fraction of fuel vaporized in the intake system, and fuel vaporization
2. Mixture temperature as influenced by heat transfer
3. Ratio of exhaust to inlet manifold pressures
4. Compression ratio
5. Engine speed
6. Intake and exhaust manifold port design
7. Intake and exhaust valve geometry, size, lift and timings(Heywood)

Fig. 5.4 shows the variations of volumetric efficiency, η_v with engine power. With increase in load, volumetric efficiency η_v decreases, since the temperature of combustion chamber walls are increased. In a diesel engine the cylinder and combustion chamber walls are hot, and confines the hot residual gas within the cylinder at the end of the exhaust stroke. When the residual flows into the inlet manifold, the inlet valve and port are heated. The entering fresh charge is also heated, not only by the walls, but also by the hot inlet valve and port. Thus η_v is decreased. At lower speed η_v is greater than at higher speed because of the fact that at higher engine speeds, the flow into the engine during the intake process a part of inlet becomes choked. Once this occurs, further increases in speed do not increase the flow rate significantly and so volumetric efficiency, η_v decreases. [Heywood, 1988]

In Fig. 5.5, fuel consumption rates of diesel fuel are presented at three different speeds at different loads. It is seen that fuel consumption rate increases with the increase in load. In diesel engine the absolute value of total friction work varies with load, and increases as speed increases. The friction work is the pumping work in the engine, to overcome the resistance to relative motion of all the moving parts of the engine, and to drive the engine accessories. To overcome these friction works for the same fuel and same power, the fuel consumption rate is higher for higher speeds. [Heywood, 1988]

In Fig. 5.6 variation of air fuel ratio is plotted against engine brake power. It is observed that the overall A/F ratio is in the range of 15.7-53 at rated speed. With change in load the quantity of fuel is changed, which changes the air fuel ratio. Whatever may be the overall A/F ratio in a diesel engine due to the injection of fuel, there is a heterogeneous mixture with A/F ratio varying widely in different areas within the chamber. Relative A/F ratio may be defined as

$$\lambda = \frac{(A/F \text{ ratio})_{\text{actual}}}{(A/F \text{ ratio})_{\text{stoichiometric}}}. \text{At low load } \lambda > 1 \text{ i.e fuel lean mixture and at high}$$

load $\lambda < 1$, i.e fuel rich mixture. In a diesel engine A/F ratio for stoichiometric mixture is 14.5 (Heywood). However, A/F ratio in a diesel engine cant approach the chemically correct mixture without the appearance of smoke. The engine should

never be operated at the point of maximum power, because extreme smoking and fouling of the engine would occur. Many engines will smoke even at A/F ratio of 30:1. In the diesel engine, injection must occur near the point where pressure rise is desirable and therefore little time is available for the fuel to find air. For this reason, the injection pump is equipped with a quantity stop that prevents injection of fuel beyond that of light smoke in diesel engine. Thus the points of maximum power and maximum economy in the diesel engine are shifted toward relatively high air fuel ratios than the chemically correct ratio [Obert, 1973]

Variations of brake specific fuel consumption, $bsfc$, with brake power, P_b at different speed is shown in Fig. 5.7. It is seen that $bsfc$ values all follow a hooked curve. At lower load, $bsfc$ is higher because A/F ratio is high, its value decrease with the increase of brake power until the rated output of the engine is reached. Minimum value of $bsfc$ is reached at each engine speed when all the fuel in the cylinder is most effectively consumed. After the rated output, $bsfc$ slightly increases again because air fuel mixture is richer due to scarcity of air. This parameter $bsfc$ is preferred, rather than thermal efficiency, because all quantities are measured in standard and accepted physical units: time, horsepower and mass. [Obert, 1973]

In Fig. 5.8 brake thermal efficiency, η_b is plotted against the same conditions mentioned above. $bsfc$ and η_b are inversely related to each other. So at lower brake power η_b is lower and it increases significantly up to the rated power and after this point it decreases slightly. For calculating η_b of a fuel higher heating value of the fuel should be used, because higher heating value at constant pressure represents the maximum amount of heat that can be transferred from steady flow machine. However, the heat that can be attained by condensing the water formed by combustion is (practically) not attainable, because exhaust gases invariably are discharged at high temperatures. For this reason, efficiency calculations are based upon the lower heating value of the fuel. [Obert, 1973]

Variations of exhaust gas temperature, T_{ex} with brake power is shown in Fig. 5.9. Exhaust temperatures are found to increase gradually with the increase of brake

power. With the increase of brake power more fuel is burned inside the cylinder and more temperature is generated and so, the exhaust temperature increases.

The brake power of an engine is dependent on its size and speed. Therefore, it is not possible to compare performance parameters of engine as a function of brake power. Brake mean effective pressure b_{mep} is a true indication of the relative performance of different engines. There is a good reason for this; all engines tend to be made from similar materials. Since material stress in an engine depends on a first approximation only on the b_{mep} and mean piston speed, it follows that for the same stress limit imposed by the material, all engines should have the same b_{mep} and mean piston speed. As a result, b_{mep} is used here, to compare performance parameters of engine as it is roughly comparable for different engines. [Ferguson 2000]. Fig. 5.10 and Fig. 5.11 show the $bsfc$ and η_b values for three different speeds as functions of b_{mep} . From now on the performance and exergy terms are evaluated as a function of b_{mep} .

The exergy analysis based on the second law of thermodynamics is utilized to identify the source of losses in useful energy within the components of diesel engine. About 50% of the chemical exergy of the fuel is destroyed due to unaccounted factors and about 15% is lost in the cooling water or exhaust gases. In general, diesel engines have efficiencies of about 35% and about 50% of the input fuel energy is lost in cooling water and exhaust gases. The wasted energy in the cooling water is usually considered useless due to its low temperature level. In internal combustion engine exergy is contained mainly in the form of chemical exergy. [Al-Najem, and Diab, 1992]

The second law efficiency or exergic efficiency is denoted by $\eta_{ex,shaft}$ in the present study. The exergy is destroyed or lost due to different irreversibilities which have been stated already. Exergy destruction due to all these sources together is expressed by $Ex_{uncounted}$. [Al-Najem, and Diab, 1992]

Exergy destruction in exhaust gases, Ex_{eg} is evaluated separately. It can be noted from figure 5.12 that at higher $bmep$ exergy output at shaft is almost constant with a slight declining trend.

The exergy input utilized in different processes of the engine is shown in Fig. 5.12. It can be noticed that exergy transfer to the exhaust gases (denoted by Ex_{eg}) increases with increasing $bmep$. In contrast, the availability destruction in friction, cooling, combustion, etc. (denoted by $Ex_{unaccounted}$) shows a declining trend. That is, with the increase in $bmep$ as Ex_{eg} goes up $Ex_{unaccounted}$ continues to go down. With increasing $bmep$, load increases that means A/F ratio shifted to rich mixture which increases the level of temperatures inside the cylinder, thus there is lesser degradation in the fuel chemical exergy as this now is transferred to ‘hotter’ exhaust gases. This decreased amount of exergy destruction is mainly reflected in increased amount of heat loss or exhaust gases exergy. [Rakopoulos, 2006]

The comparison of brake specific fuel consumption is plotted as a function of $bmep$ for two different preheat temperatures 80° and 100°C for pure WVO and 100°C preheat temperature for 80% WVO and 20% diesel blend and are compared with that of diesel fuel in Fig. 5.13. It was observed that $bsfc$ of all fuels tend to decrease with increasing load. The WVO diesel blends yielded a $bsfc$ closely matching that of diesel oil. Though the preheated WVO at 80° and 100°C temperatures maintained a similar trend to that of diesel, the value of $bsfc$ of heated WVO was higher than that of diesel in the entire load range of the engine. This is due to the combined effect of viscosity and lower heating value of WVO. High viscosity and poor volatility of the WVO results in poor atomization and mixture formation and increase fuel consumption to maintain the power. However preheated WVO shows improvement in energy consumption at higher load. In addition to that lower heating value of the WVO leads to more fuel delivery for the same load condition. Same trend is observed for coconut oil from Fig. 5.14.

In Fig. 5.15 and 5.16, a variation of brake thermal efficiency, η_b is shown for the above mentioned conditions. In Fig. 5.15 revealed that with increasing $bmep$, η_b of

WVO at 80°C and 100°C temperatures and blends of WVO and diesel was increased. In case of blend η_b closely matches with that of diesel oil. Same trend is observed for coconut oil from the Fig. 5.16. It is noticed that the use of oxygen-rich alternative fuels promote a better mixture formation and combustion, thus improving the thermal efficiency. The improvement in brake thermal efficiency can be attributed to the enhanced oxygen content which improves combustion especially during the phase of diffusion combustion, and to the higher lubricity of alternative fuels which reduces the friction loss.

Comparison of chemical exergetic output at shaft $\eta_{ex,shaft}$ as a function of $bmep$ for two different preheat temperatures of WVO and one preheat temperatures for WVO and diesel blend are shown in Fig. 5.17. The results $\eta_{ex,shaft}$, of diesel-WVO blend and preheated WVO shows almost similar result like diesel. Same trend is observed for coconut oil from Fig. 5.18.

Comparison between percent of exergy destroyed due to exhaust gas and percent of exergy destroyed in friction, cooling, combustion etc. and diesel fuel and other alternative fuels are shown in Fig. 5.19 and Fig. 5.20 respectively. Decrease in exergy destroyed for uncounted factor Ex_{un} was shown for the use of preheated WVO and blend of diesel WVO, Ex_{un} for alternative fuels is less than diesel. However the exergy destroyed due to exhaust gas as a fraction of the injected fuel exergy increases significantly for operation with preheated WVO and diesel WVO blend because of higher exhaust gas temperature for alternative fuels.

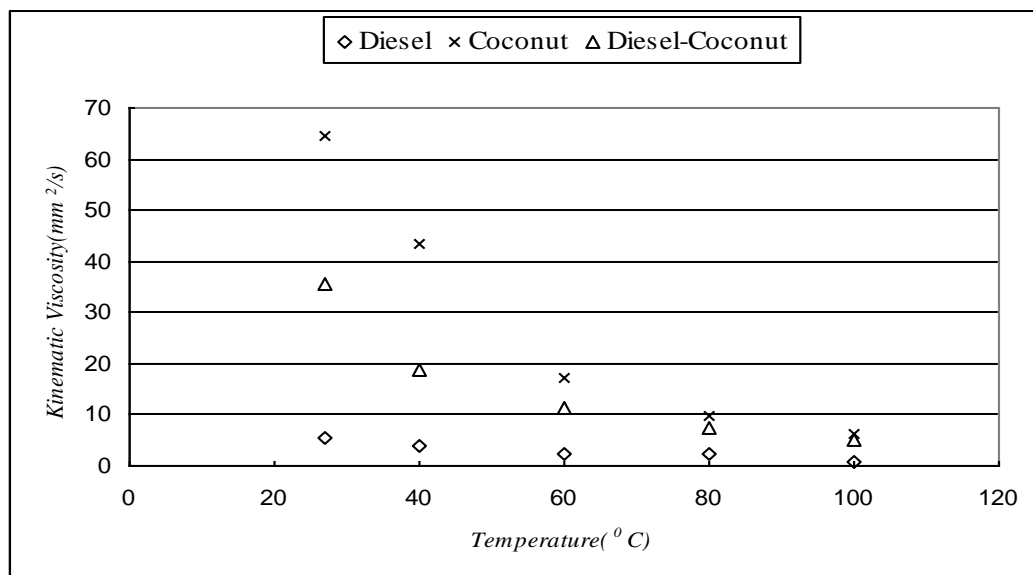
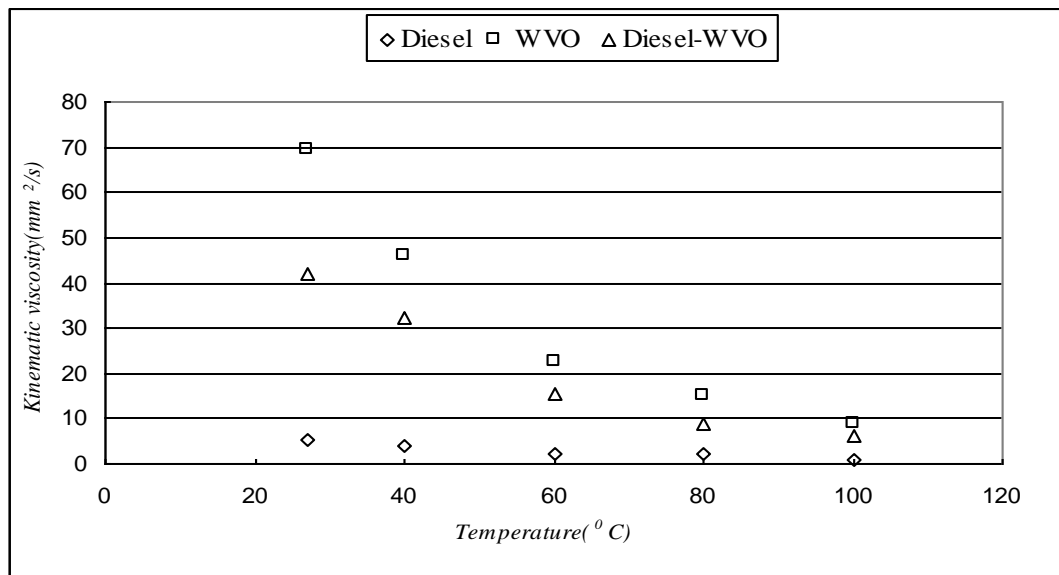


Fig 5.1 (a) Effect of temperature on the viscosities of Diesel, WVO and Diesel-WVO blend
 (b) Effect of temperature on the viscosities of Diesel, Coconut and Diesel-Coconut blend

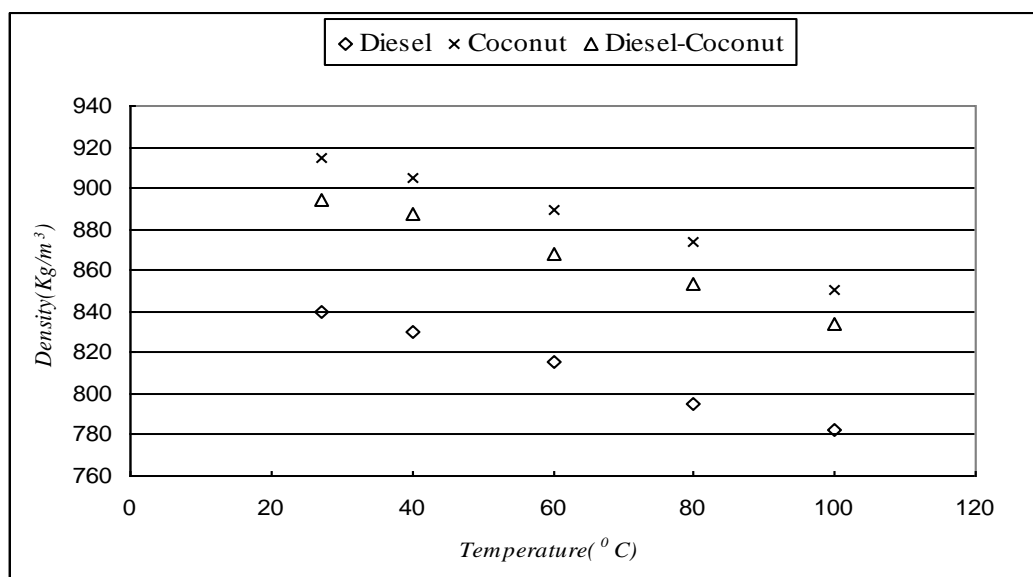
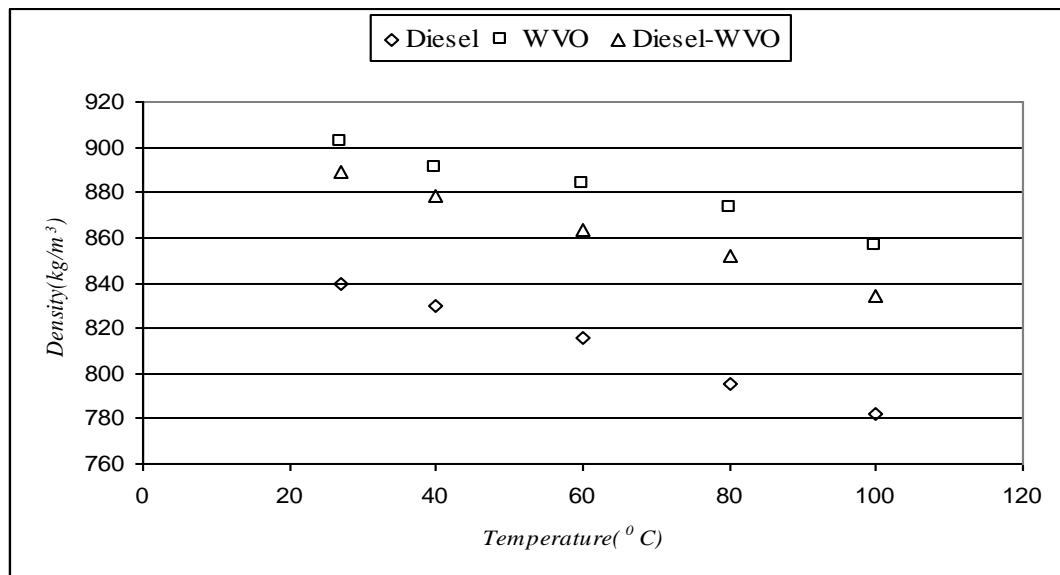


Fig 5.2 (a) Effect of temperature on the densities of Diesel, WVO and Diesel-WVO blend
 (b) Effect of temperature on the densities of Diesel, Coconut and Diesel-Coconut blend

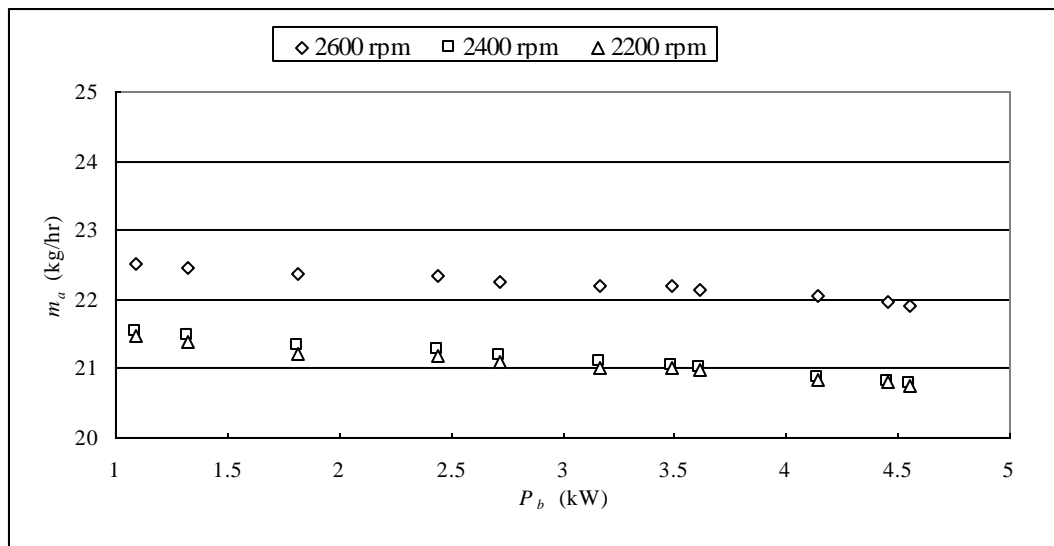


Fig. 5.3 Variation of mass flow rate of air as a function of brake power output at three different speeds running on diesel fuel

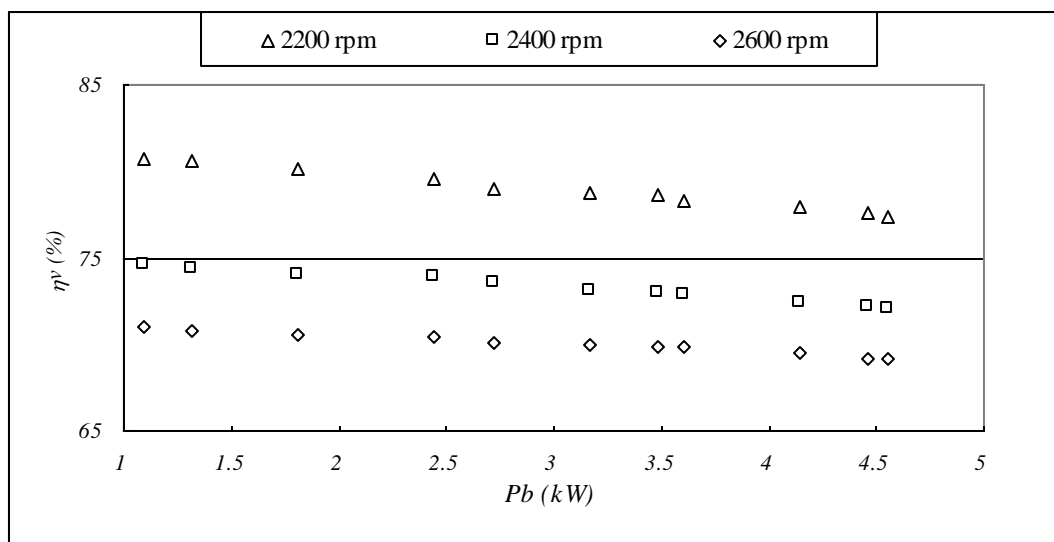


Fig. 5.4 Variation of volumetric efficiencies as a function of brake power output at three different speeds running on diesel fuel

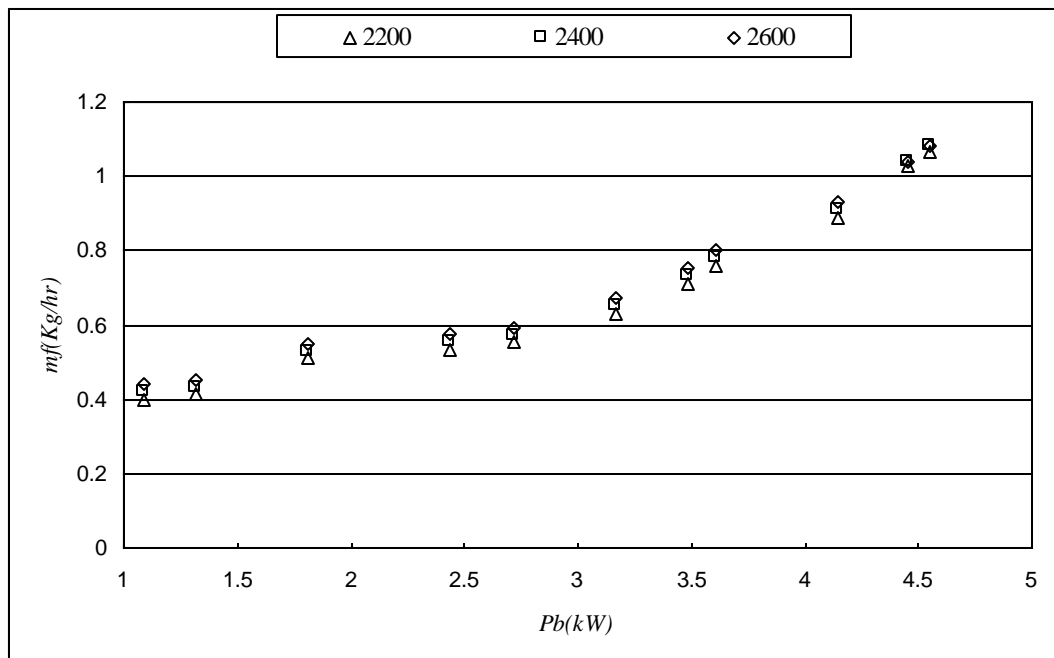


Fig. 5.5 Variation of mass flow rate of fuel as a function of brake power output at three different speeds running on diesel fuel

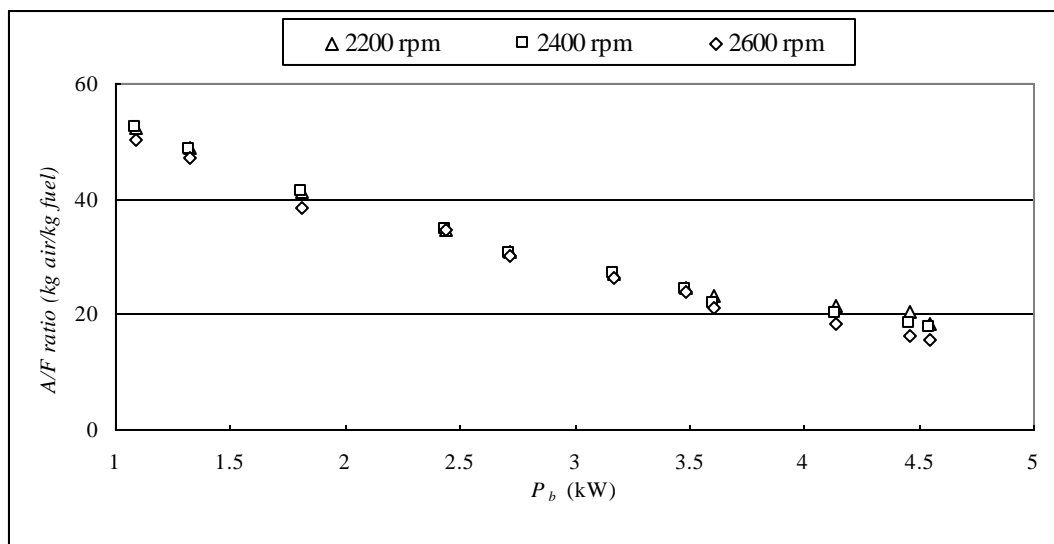


Fig. 5.6 Variation of air-fuel ratio as a function of brake power output at three different speeds running on diesel fuel

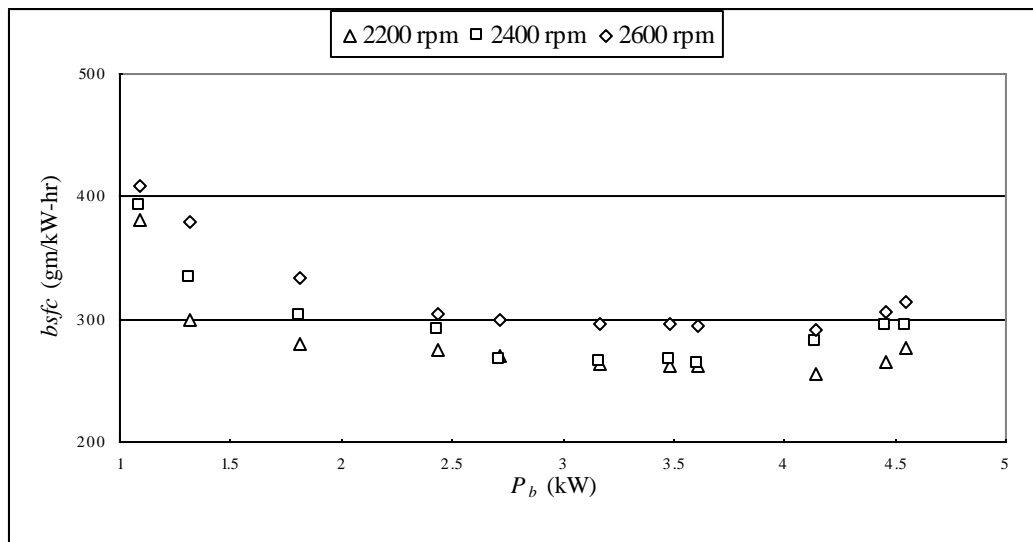


Fig. 5.7 Variation of brake specific fuel consumption as a function of brake power output at three different speeds running on diesel fuel

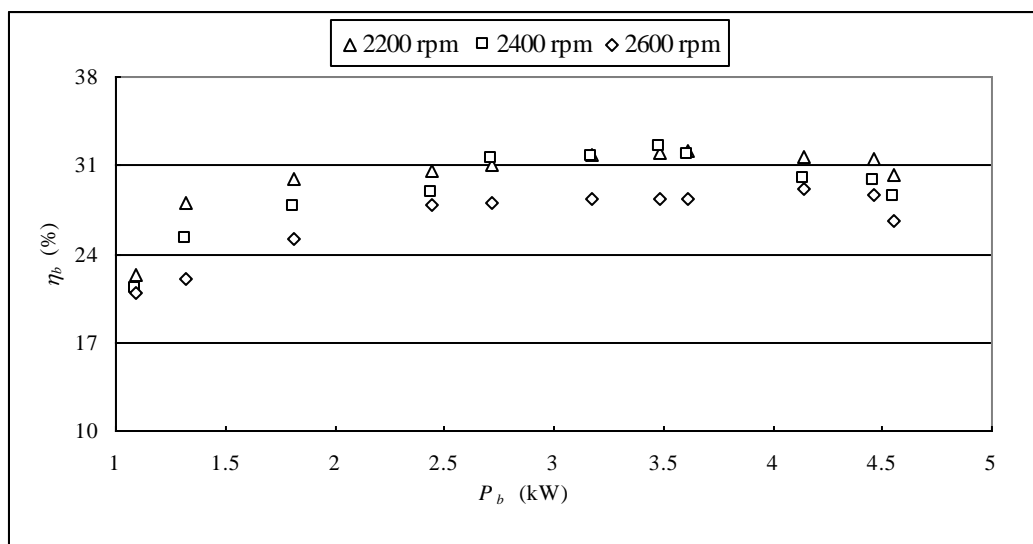


Fig. 5.8 Variation of brake thermal efficiency as a function of brake power output at three different speeds running on diesel fuel

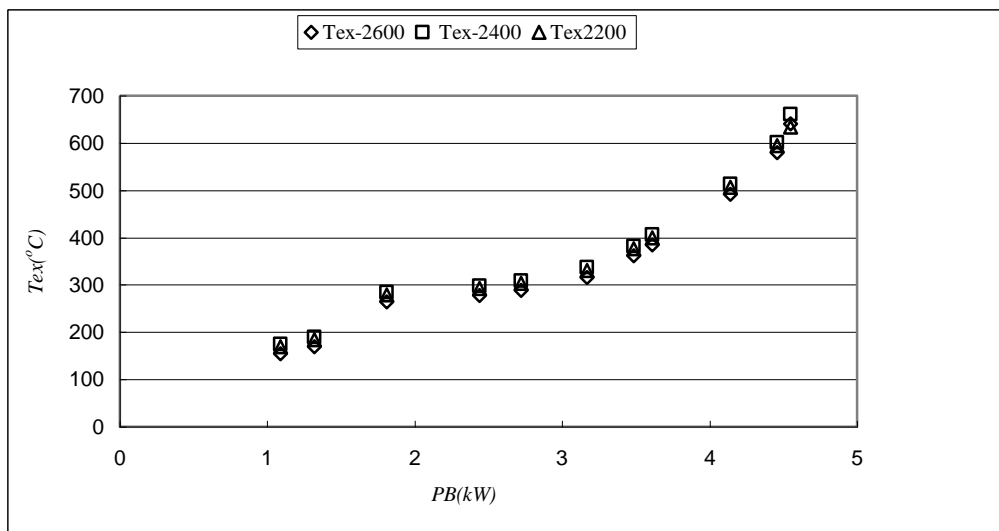


Fig. 5.9 Variation of exhaust gas temperature (T_{ex}) as a function of brake power

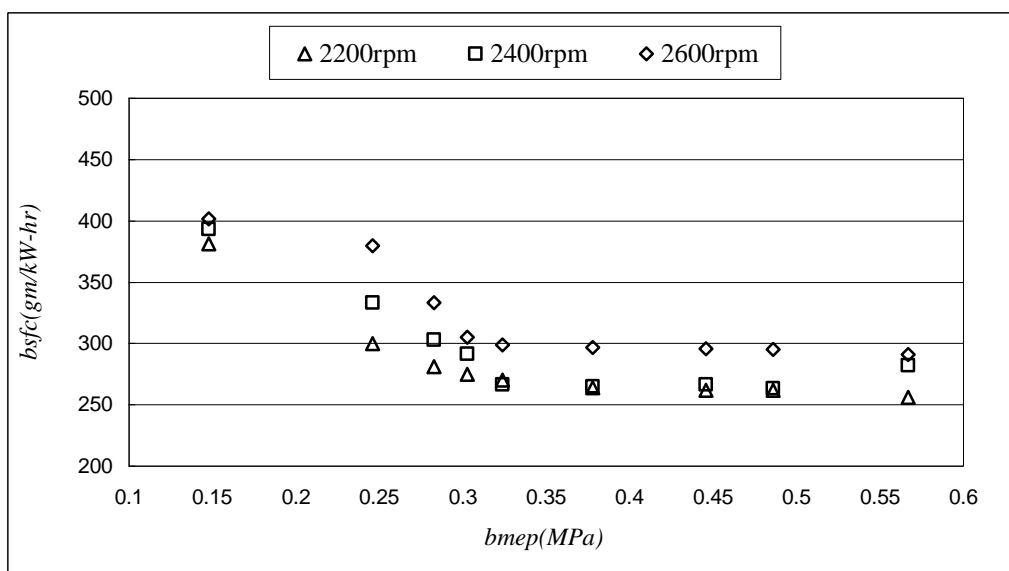


Fig. 5.10 Variation of brake specific fuel consumption as a function of brake mean effective pressure at three different speeds run by diesel fuel

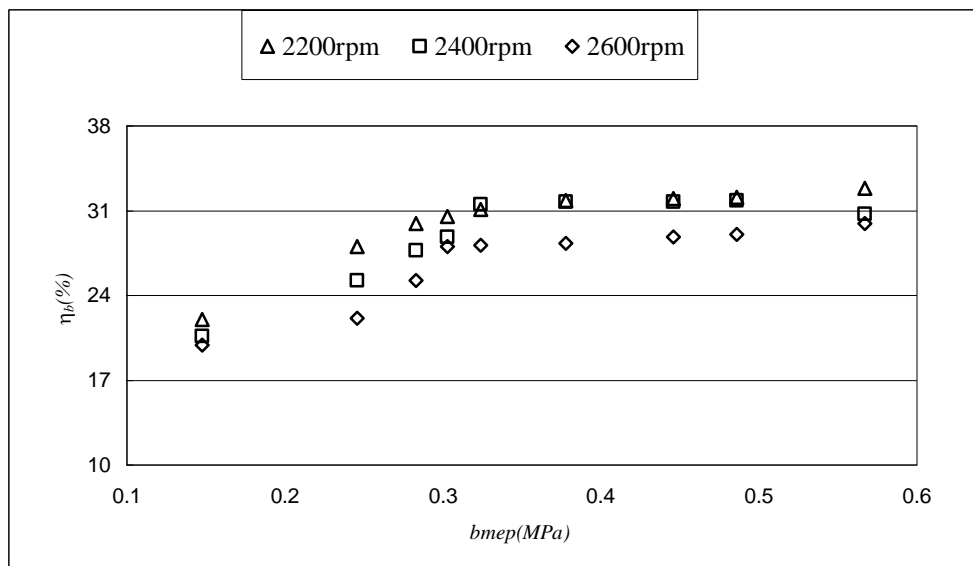


Fig.5.11 Variation of brake thermal efficiency as a function of brake mean effective pressure at three different speeds run by diesel fuel

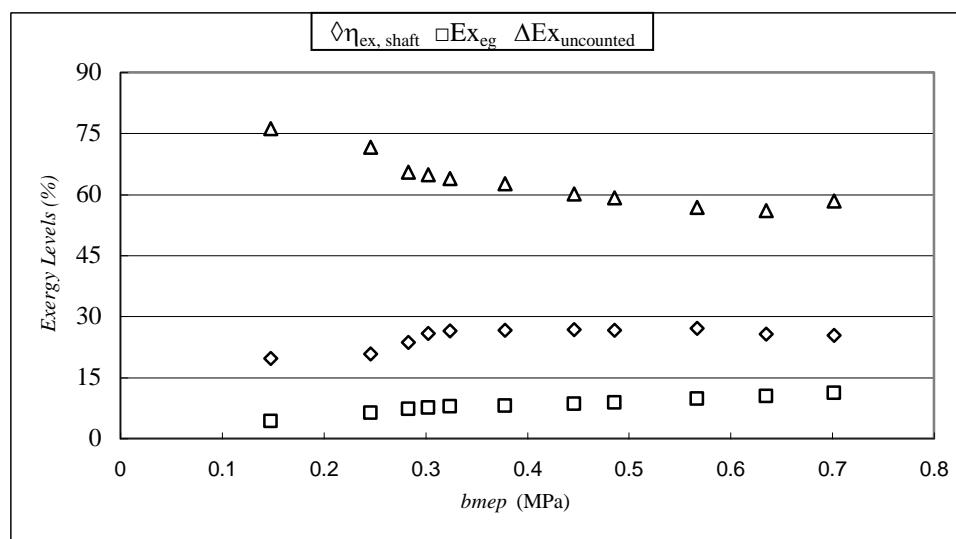


Fig. 5.12 Comparison among the exergy levels, viz., exergetic efficiency, exergy destroyed in exhaust gas and exergy destroyed from unaccounted sources at 2600rpm run by diesel fuel

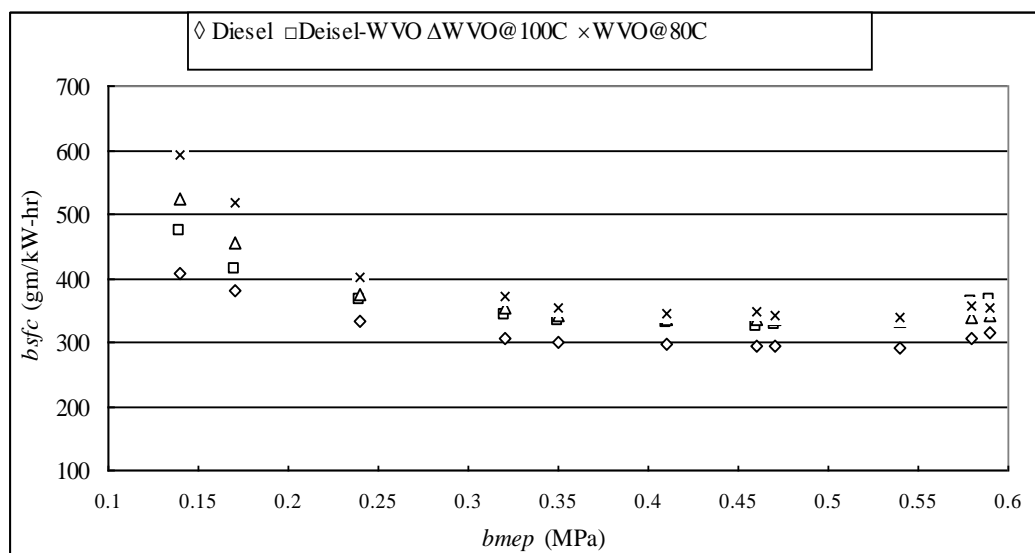


Fig. 5.13 Comparison of brake specific fuel consumption levels using raw diesel, WVO at 80° and 100° C preheat temperature and blend of diesel fuel with WVO as a function of brake mean effective pressure at 2600rpm

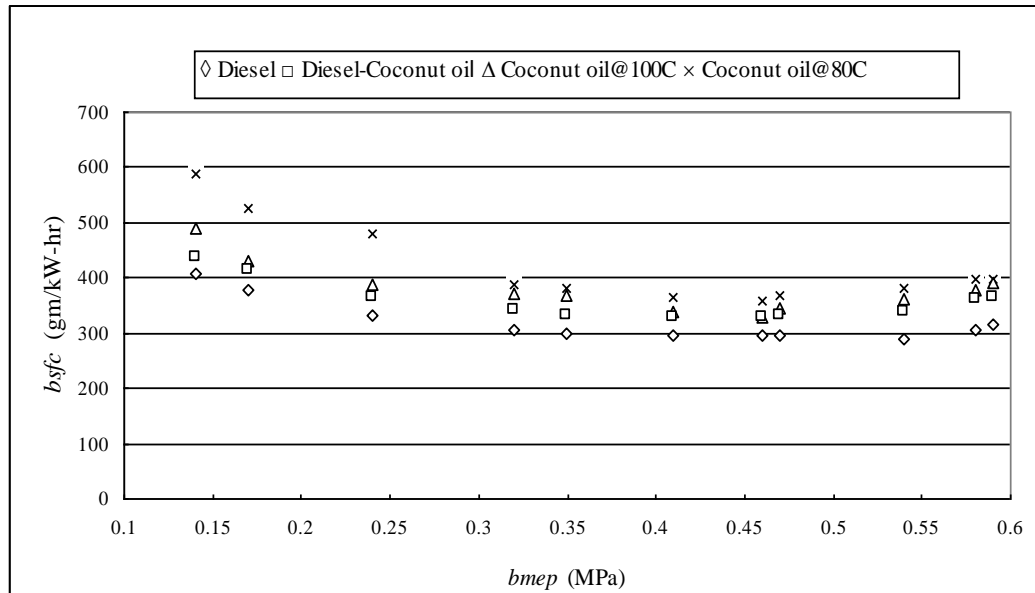


Fig. 5.14 Comparison of brake specific fuel consumption levels using raw diesel, coconut at 80° and 100° C preheat temperature and blend of diesel fuel with coconut as a function of brake mean effective pressure at 2600 rpm

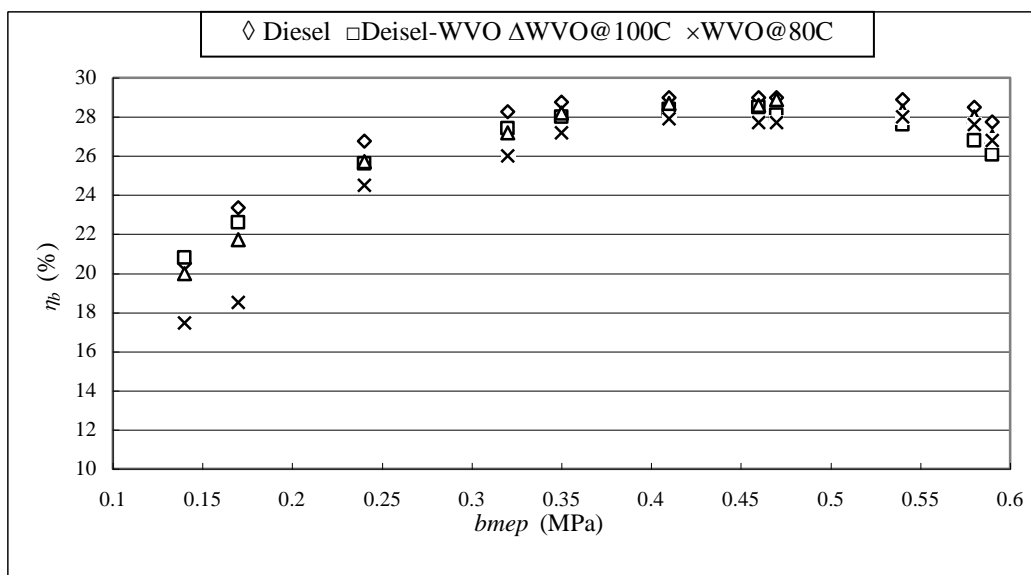


Fig. 5.15 Comparison of brake thermal efficiencies using raw diesel, WVO at 80° and 100° C preheat temperature and blend of diesel fuel with WVO as a function of brake mean effective pressure at 2600 rpm

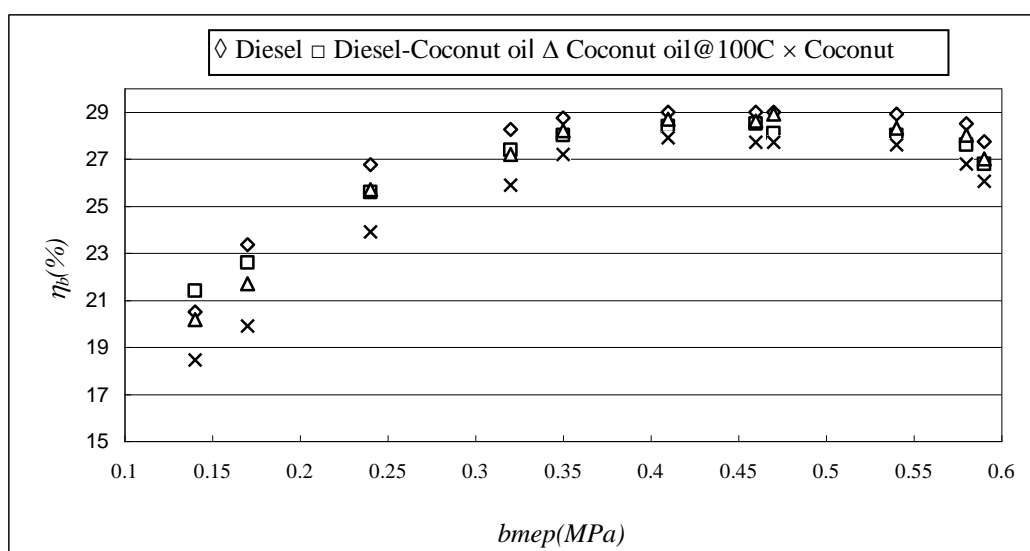


Fig. 5.16 Comparison of brake thermal efficiencies using raw diesel, coconut at 80° and 100° C preheat temperature and blend of diesel fuel with coconut as a function of brake mean effective pressure at 2600 rpm

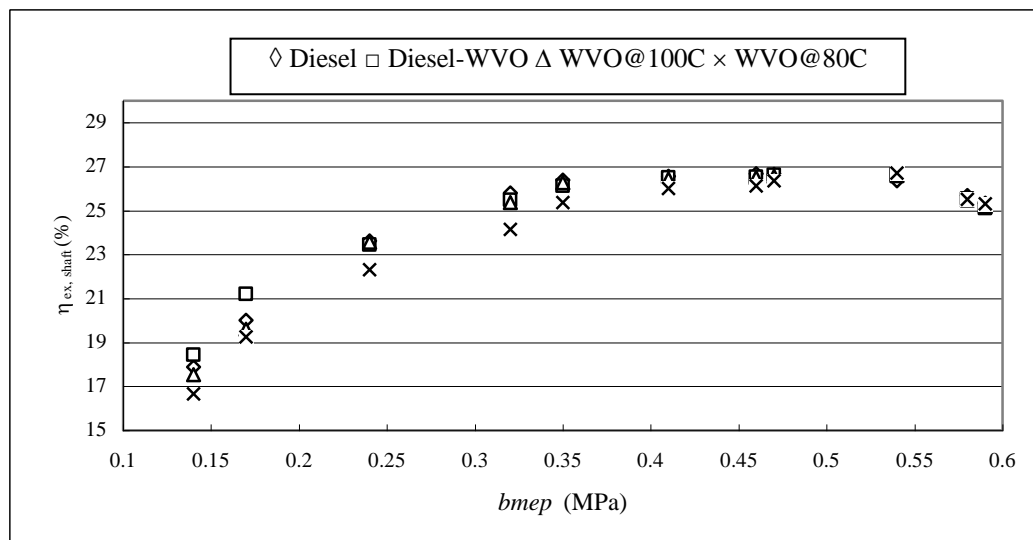


Fig. 5.17 Comparison of chemical exergetic output at shaft using raw diesel, WVO at 80° and 100° C preheat temperature and blend of diesel fuel with WVO as a function of brake mean effective pressure at 2600 rpm

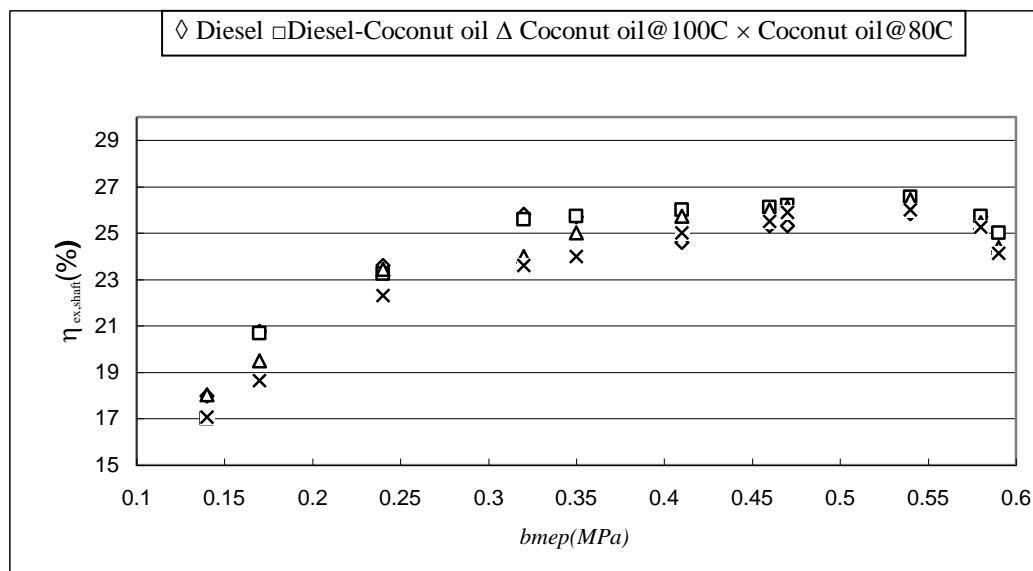


Fig. 5.18 Comparison of chemical exergetic output at shaft using raw diesel, Coconut oil at 80° and 100° C preheat temperature and blend of diesel fuel with Coconut oil as a function of brake mean effective pressure at 2600 rpm

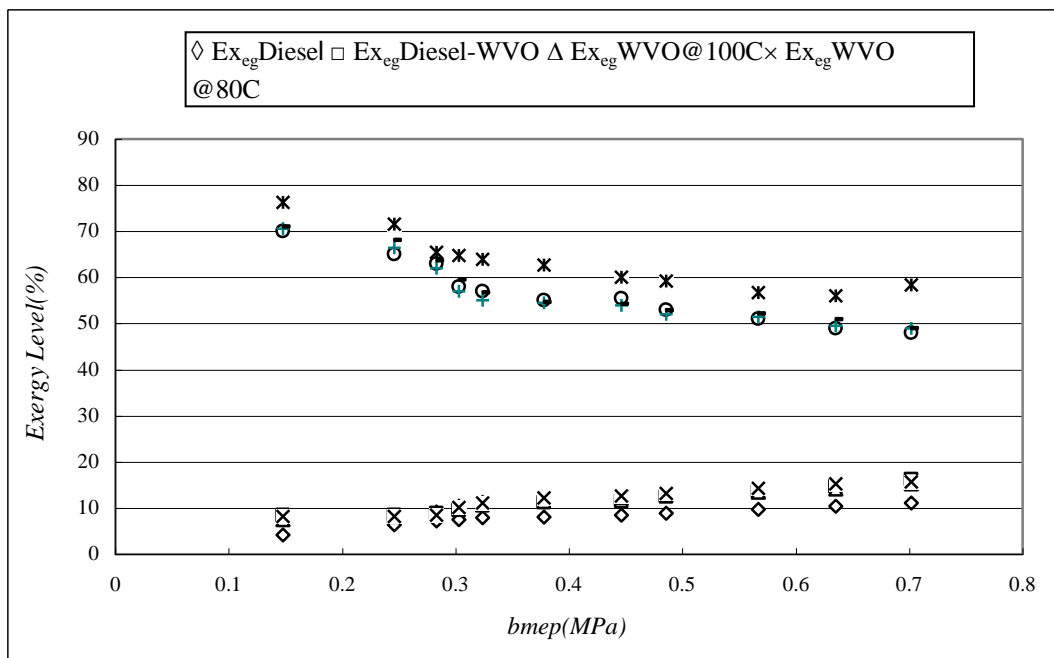


Fig. 5.19 Comparison among the exergy levels, viz., exergy destroyed in exhaust gas and exergy destroyed from unaccounted sources at 2600 rpm run by diesel fuel and WVO

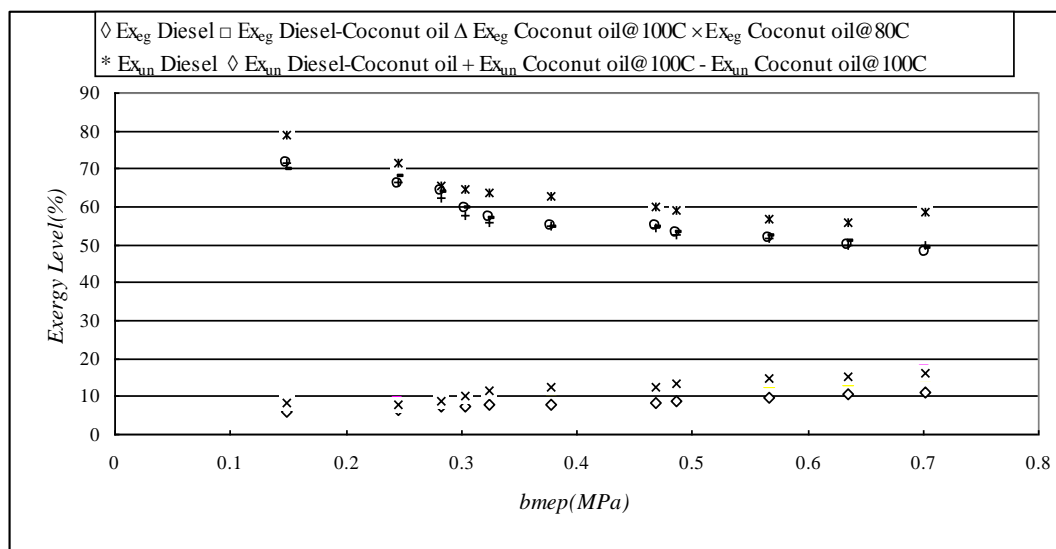


Fig. 5.20 Comparison among the exergy levels, viz., exergy destroyed in exhaust gas and exergy destroyed from unaccounted sources at 2600 rpm run by diesel fuel and Coconut oil.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The major inferences that can be drawn from the present study are summarized below:

1. The most significant difference between diesel oil, WVO and coconut oil is the latter's much higher viscosity and higher density. When WVO and coconut oil are heated up to 80-100°C, then viscosity and density of WVO and coconut oil become close to that of diesel.
2. From engine performance test it is found that air flow rate increases with engine speed with respect to brake power. On the other hand, with the increase of engine power, mass flow rate of air decreases slightly. With increase in load, volumetric efficiency η_v decreases. It is seen that fuel consumption rate increases with the increase in load.
3. Though the preheated WVO and coconut oil at 80° and 100°C temperatures maintain a similar trend of *bsfc* to that of diesel, the value of *bsfc* of heated WVO is higher than that of diesel in the entire load range of the engine. Because high viscosity and poor volatility of the WVO results in poor atomization and mixture formation thereby increasing the fuel consumption to maintain power.
4. It can be noticed that exergy transfer to the exhaust gases (denoted by Ex_{eg}) increases with increasing *bme_p*. The availability destruction in friction, cooling, combustion, etc. (denoted by $Ex_{unaccounted}$) shows a declining trend.

5. Decrease in exergy destroyed for uncounted factor Ex_{un} for alternative fuels is less than that of diesel. However, the exergy destroyed due to exhaust gas as a fraction of the injected fuel exergy increases significantly for operation with alternative fuels. It means that there are possibilities of improvement in case of alternative fuels.
6. According to the existing study using exergy as a measure of quality, diesel fuel is a better quality fuel than WVO and coconut oil. Because, the net lower heating value of diesel is greater than that of WVO and coconut oil. It means that to run the engine for a certain period at the same running condition, greater amount of alternative fuel is needed.

6.2 Recommendations for Further Works

The following aspects demand attention and are recommended for further extensive studies:

1. Exergy destroyed (also called '*anergy*') with exhaust gas may be used directly for preheating purposes. The cost thus saved to preheat the fuel in this course can be calculated to have a clear idea on whether it is economic to employ vegetable oil or not.
2. Major portion of the fuel chemical exergy is destroyed due to friction, cooling, combustion etc. Therefore, more research should be devoted to minimize such losses. Further extensive studies should be carried out to find the levels of exergy destruction from these sources.
3. Long term tests with alternative fuels may be carried out in order to find out the type of problems such as (material compatibility, component wear, component malfunction) that may occur with the engine.

Appendix A

Engine Specifications

Model No.	:	R 175A
Manufacturer and country of origin	:	Changhai Co., Ltd., China
Cycle	:	Four-stroke DI diesel engine
Number of cylinders	:	1 (one)
Bore	:	75 mm
Stroke	:	80 mm
Total swept volume	:	0.000353 m ³
Rated speed	:	2600 rpm
Starting method	:	Hand cranking
Maximum power	:	6 hp
Cooling	:	Water evaporative
Lubrication	:	Combined pressure and splashing
Oil sump capacity	:	4 L
Injection pressure (injectors setting)	:	13.72 MPa
Valve clearance	:	Intake valve : 0.15-0.25 mm Exhaust valve : 0.25-0.35 mm
Type of injector pump	:	Helical groove, individual rack
Fuel injection timing	:	22° ± 2°
Rotation of the flywheel	:	counter clockwise (from flywheel side)

Appendix B

Dynamometer Specifications

Type	: Hydraulic dynamometer
Model	: AN3e
Manufacturer	: Gebr. Hofmann KG Werk iii Elze
Country of origin	: German
Length of torque lever	: 0.7162 m
Torque formula	: $Md = 716.2 \cdot (N/n)$ $Md = \text{torque(kgm)}$ $N = \text{power(hp)}$ $n = \text{speed(rpm)}$

Appendix C

Sample Calculation

A complete set of sample calculations with actual data is followed by the governing equations respectively are presented in details as follows

Lab Condition:

Date of experiment	:	April 8, 2009
Fuel used	:	Diesel
Atmospheric pressure, p_x	:	746.98 mm of Hg (99.59 KPa)
Dry bulb temperature, T_{db}	:	28.5°C(301.5K)
Wet bulb temperature, T_{wb}	:	26°C (299 K)
Relative Humidity, ϕ_x	:	81.6%

Derating Factors:

Power adjustment factor,	α	:	0.978
Fuel consumption adjustment factor,	β	:	1.004

From appendix D

Experimental Observations and Data:

Experimental constants:

Diameter of the flow nozzle,	D_n	:	13mm
Cross-sectional area of the flow nozzle,	A_n	:	$2.85 \times 10^{-4} \text{ m}^2$
Coefficient of discharge of the flow nozzle,	C_D	:	0.92
Specific gravity of diesel,	σ	:	0.85
Lower heating value of diesel,	LHV	:	42.94 MJ/Kg
Density of manometric fluid (water) at 28.5°C,	ρ_w	:	996.1 kg/m ³
Density of air at 28.5°C,	ρ_w	:	1.151Kg/m ³

Experimental Data:

Load on dynamometer,	W	:	1.123kg
Speed of the dynamometer,	N	:	2604
Volume of diesel collected at time t ,	V	:	50 ml

Time of collection of V volume diesel,	t	:	341 S
Manometric deflection	h_m	:	0.996 inch of water 0.0252 m of water
Lubricating oil temperature		:	68 °C
Exhaust gas temperature		:	197 °C

C.1 Brake power output

The brake power of the test engine is calculated using the following equations:

$$P(\text{hp}) = \frac{W \cdot N}{2000}$$

$$P(\text{kW}) = 0.7355P(\text{hp})$$

Where

$P(\text{hp})$: brake power output in hp

$P(\text{kW})$: brake power output in kW

W : Load on the dynamometer in kg

N : speed of the shaft connected to the dynamometer in rpm

Engine brake power output in hp,

$$P(\text{hp}) = \frac{1.123 \times 2604}{2000}$$

$$= 1.46 \text{ hp}$$

Engine brake power output in kW,

$$P(\text{kW}) = 1.46 \times 0.7355$$

$$= 1.07 \text{ kW}$$

C.2 Fuel Consumption Rate

The fuel consumption rate of the engine is calculated as below:

$$\dot{m}_f = \frac{(\gamma \cdot \rho_w \cdot V)}{t} \times 60 \times 10^{-6}$$

where

\dot{m}_f : fuel consumption rate in kg/hr

V : volume of fuel flown in time t in ml

ρ_w : density water in kg/m^3

γ : specific gravity of the fuel

t : time to measure V ml fuel flow in minutes

Fuel consumption rate,

$$\dot{m}_f = \frac{0.85 \times 996.1 \times 50}{341} \times 3600 \times 10^{-6}$$

$$= 0.449 \text{ kg/hr}$$

C.3 Standardized Brake Power and Fuel Consumption Rate

The engine brake power and fuel consumption rate are standardized or derated according to the BS 5514: Part 1 1982 as follows:

$$P_b = \frac{P(\text{kW})}{\alpha}$$

where

P_b : adjusted brake power output in kW

$P(\text{kW})$: brake power output in kW

A : power adjustment factor

and

$$\dot{m}_f = \frac{\dot{m}'_f}{\beta}$$

where

\dot{m}_f : adjusted fuel consumption rate in kg/hr

\dot{m}'_f : fuel consumption rate in kg/hr

β : fuel consumption adjustment factor

Adjusted brake power output,

$$P_B = \frac{1.07}{0.978} \\ = 1.094 \text{ kW}$$

Adjusted fuel consumption rate,

$$\dot{m}_f = \frac{0.449}{1.004} \\ = 0.447 \text{ kg/hr}$$

C.4 Air Flow Rate and A/F Ratio

The mass flow rate of intake fresh air and the air –fuel ratio are determined as below:

$$\dot{m}_a = \left[3600 \times \frac{C_d \times A_n}{\rho_{air}} \sqrt{2g \times h_m \times \rho_m} \right]$$

where

\dot{m}_a : mass of fresh air inducted per unit time in kg/hr

C_d : Co-efficient of discharge of the flow nozzle

A_n : cross-sectional area of the flow nozzle in m^2

ρ_{air} : density of air in kg/m^3

ρ_m : density of manometric fluid in kg/m^3

h_m : manometric deflection in m of H₂O

and

$$A/F = \frac{\dot{m}_a}{\dot{m}_f}$$

where

\dot{m}_a : mass of fresh air inducted per unit time in kg/hr

\dot{m}_f : adjusted fuel consumption rate in kg/hr

Air flow rate,

$$m_a = [3600 \times 0.92 \times 2.855 \times 10^{-4} \times \sqrt{\frac{2 \times 9.81 \times 0.0252 \times 996.1}{1.151}}] \times 1.151$$

$$= 22.52 \text{ kg / hr}$$

Air-fuel ratio,

$$A/F = \frac{22.52}{0.447} = 50.38$$

C.5 Volumetric Efficiency

The overall volumetric efficiency of the engine breathing system is calculated using the following formulae:

$$\eta_v = \left(\frac{2 \cdot \dot{m}_a}{NV_d \rho_i} \right) \left(\frac{1}{60} \right)$$

where

η_v : volumetric efficiency in %

\dot{m}_a : mass of fresh air inducted per unit time in kg/hr

N : Engine speed in rpm

V_d : swept volume of all of the engine cylinders in m³

ρ_i : Density of inlet air at intake manifold in kg/m³

Volumetric efficiency,

$$\eta_v = \left(\frac{2 \times 22.52}{2605 \times 0.000353 \times 1.151} \right) \left(\frac{1}{60} \right)$$

$$= 70.97\%$$

C.6 Brake Thermal Efficiency

The brake thermal efficiency of the test engine is calculated as follows:

$$\eta_b = \left\{ 3.6 \cdot \left(\frac{P_b}{\dot{m}_f \times LHV} \right) \right\} \times 100$$

where

η_b : brake thermal efficiency in %

\dot{m}_f : adjusted fuel consumption rate in kg/hr

P_b : adjusted brake power output in kW

LHV : lower heating value of the fuel in MJ/kg

Brake thermal efficiency,

$$\begin{aligned}\eta_b &= \frac{3600}{408.6 \times 42.9} \\ &= 0.205 \\ &= 20.5\%\end{aligned}$$

C.7 Brake Specific Fuel Consumption

The brake specific fuel consumption is computed as follows:

$$bsfc = \frac{\dot{m}_f}{P_b}$$

where

$bsfc$: brake specific fuel consumption in kg/kW-hr

\dot{m}_f : adjusted fuel consumption rate in kg/hr

P_b : adjusted brake power output in kW

Brake specific fuel consumption,

$$\begin{aligned}bsfc &= \frac{0.447}{1.094} \times 1000 \\ &= 408.6 \text{ gm/kW.hr}\end{aligned}$$

C.8 Fuel Chemical Exergy or Exergy Input

The fuel chemical exergy or exergy input is calculated as follows:

For diesel fuel,

$$Ex_{in} = 1.06489Q_{in}$$

where

η_{ex} : Exergy input in kW

Q_{in} : heat energy input by fuel in kW

For soybean oil fuel,

$$Ex_{in} = 1.0724Q_{in}$$

For coconut oil fuel

$$Ex_{in} = 1.0786Q_{in}$$

Exergy input,
(or Fuel chemical Exergy)

$$Ex_{in} = 1.06489 \cdot \left(\frac{0.447 \times 42.94 \times 10}{36} \right) = 5.68 \text{ kW}$$

C.9 Percent of Fuel Exergy Output at Shaft

The percent of fuel chemical exergy output at shaft is calculated as follows:

$$\eta_{ex,shaft} = \frac{P_b}{Ex_{in}}$$

where

$\eta_{ex,shaft}$: Percent of fuel chemical exergy output at shaft in %

Ex_{in} : Fuel exergy input in kW

P_b : adjusted brake power output in kW

Percent exergy output at shaft,

$$\eta_{ex,shaft} = 1.094/5.68 = 19.26\%$$

C.10 Exergy Destructions

The sources of availability destruction are:

(a) Exergy destruction in exhaust gases, Ex_{eg}

(b) Exergy destruction from uncounted sources as friction, cooling, etc., $Ex_{uacncounted}$

The exergy destruction in the exhaust gases is calculated as

$$Ex_{exst} = Q_{eg} + \dot{m}_{eg} T_0 \left[C_{p,e} \ln \left(\frac{T_0}{T_{e,0}} \right) \right]$$

where \dot{m}_{eg} : mass flow rate of exhaust gases = $\dot{m}_f (1 + A/F)$

Q_{eg} : heat energy of exhaust gases = $C_{p,e} \dot{m}_f (1 + A/F) (T_{exst} - T_{db})$

$C_{p,e}$: specific heat of the exhaust gas

The specific heat of exhaust is calculated using the formula of specific heat of mixture of gases as follows:

$$C_{p,e} = \frac{\sum_{i=1}^n N_{i,00} C_{p,i}}{\sum_{i=1}^n N_{i,00}} = \frac{N_{CO_2} C_{p,CO_2} + N_{H_2O} C_{p,H_2O}}{N_{CO_2} + N_{H_2O}}$$

The percent exergy destruction in exhaust gases is calculated as below:

$$Ex_{eg} = \frac{Ex_{exst}}{Ex_{in}}$$

The percent exergy destruction from other sources like friction, cooling, combustion etc. is calculated as below:

$$Ex_{unaccounted} = [100 - (\eta_{ex,shaft} + Ex_{eg})]$$

exergy destroyed
in exhaust gases,

$$Ex_{ex} = 1.4307 \times .447 \times (1 + 53.2) \times (470 - 301.5) + .447 \times (1 + 53.2) \left\{ 1.4307 \ln \left(\frac{301.5}{470} \right) \right\} \times 301.5$$

$$= 0.237561$$

where

$$C_{p,e} = \frac{14.4 \times .993 + 12.45 \times 1.937}{26.85} = 1.4307$$

Percent exergy destroyed in
exhaust gases,

$$Ex_{eg} = \frac{0.237561}{5.68} \times 100 = 4.18\%$$

Percent exergy destroyed in friction,
cooling, combustion etc,

$$Ex_{unaccounted} = [100 - (19.61 + 4.18)] = 76.21\%$$

Appendix D

Engine Derating

The engine brake power, P_b and brake specific fuel consumption, $bsfc$ are standardized following the BS-5514 : Part 1: 1982 which identical with 1981 revision of ISO 3046 /1 'Reciprocating internal combustion engines- Performance Part 1'.

D.1 The Standard Reference Conditions:

For the purpose of determining the power and fuel consumptions of engines, following reference conditions are set by the BS 5514.

1. Total barometric pressure, $P_r = 100 \text{ KPa}$
2. Air temperature, $T_r = 300\text{K} (27^\circ\text{C})$
3. Relative Humidity, $\phi_r = 60 \%$
4. Charge air coolant temperature, $T_{cr} = 300\text{K} (27^\circ\text{C})$
5. Mechanical efficiency of engine $\eta_m = 0.80$ (according to the clause 10, note 4 o the BS 5514 : Part 1: 1982)

D.2 Derating Calculations

Here is given an example to show the derating of the engine running at 2604 rpm with diesel fuel.

The data was taken on the date of April 8, 2009 and at the following lab condition:

1. Barometric pressure at lab, $P_x = 99.59 \text{ kPa} (741.5 \text{ mm of Hg})$
2. Dry bulb temperature, $T_{db} = 28.5^\circ\text{C} (301.5\text{K})$
3. Wet bulb temperature, $T_{wb} = 26^\circ\text{C} (299\text{K})$
4. Relative humidity, $\phi_x = 81.6\%$

- **From Annex F, BS 5514:**

For relative humidity, $\phi_x = 81.6\%$ and air temperature $T_{db} = 28.5^\circ\text{C} (301.5 \text{ K})$, by interpolation, the water vapor pressure $\phi_x p_{sx} = 3.180872$

- **From Annex E, BS 5514:**

For water vapor pressure of $\phi_x p_{sx} = 3.180872$ and barometric pressure in lab $p_x = 99.59$ kPa by interpolation we get the dry air pressure ratio = 0.984775

- **From Annex D, BS 5514:**

The ratio of indicated power (k) is given by $k = (R_1)^{y_1}(R_2)^{y_2}(R_3)^{y_3}$

where R_1 = dry air pressure ratio

$$R_2 = T_r/T_x$$

$$R_3 = T_{cr}/T_{cx}$$

and $y_1 = m, y_2 = n, y_3 = q$

The values of the power adjustment exponents obtain from Table- 1, BS 5514 are:

$$m=1, n=0.75, q=0$$

$$R_1 = P_x/P_r = 99.59/100 = 0.9959 \text{ and } m = 1. \text{ So, } (R_1)^m = (0.9959)^1 = 0.9959$$

$$R_2 = T_r/T_x = 300/301.5 = 0.995 \text{ and } n = 0.75. \text{ So, } (R_2)^n = (0.995)^{0.75} = 0.9962$$

Therefore, $k = 0.9959 \times 0.9962 \times 1 = 0.992$

- **From Annex C, BS 5514:**

The fuel consumption adjustment factor (β) for $k = 0.992$ and $\eta_m = 0.85$ is found by interpolation as:

$$\underline{\beta = 1.004}$$

- **From Annex B, BS 5514:**

The power adjustment factor (α) for $k = 0.992$ and $\eta_m = 0.85$ is found by interpolation as:

$$\underline{\alpha = 0.978}$$

Therefore the brake power output, P_b and brake specific fuel consumption, $bsfc$ in BS condition is adjusted as:

$$P_b = \frac{(\text{Brake power output at lab condition})}{\alpha}$$

$$bsfc = \frac{(\text{bsfc in lab condition})}{\beta}$$

Appendix E

Standard Fuel Properties

E.1 Fuel Properties

E.1.1 Heating value of fuels

The heating value or energy value of a fuel, is the amount of heat released during the combustion of a specified amount of it. It is measured in units of energy per unit of the fuel, usually mass, such as: kJ/kg, kJ/mol, kcal/kg, Btu/m³. It can be measured by bomb calorimeter. The heating value measured by the bomb is therefore called the lower or net heating value. The energy needed to vaporize the water can be calculated and it can be recaptured by letting the steam condense to water. The sum of the water vaporization energy and lower heating is called the higher or gross heating value of the fuel. When engine efficiencies are calculated, it is important to state whether higher or lower heating value of the fuel is used in calculation. Available heats of combustion are usually higher heating values and are therefore often used to calculate engine efficiencies. Standard heating value of diesel fuel is around 42.5 MJ/kg and vegetable oil and heating value is nearly 2 to 3 % less [Demirbas, 1998]

E.1.2 Specific gravity of fuels

Specific gravity is a measure of the density of liquid fuels. It is the ratio of density of fuel at 15.6⁰C to the density of water at that temperature. Specific gravity of fuel is measured with the help of a hydrometer. Specific gravity of fuel lies in the range of 0.82 to 0.87 and vegetable oil's specific gravity is about 10% higher than diesel fuel [Altin , 2001]

E.1.3 Flash point of fuels

The flash point of a volatile fuel is the lowest temperature at which it can vaporize to form an ignitable mixture in air. Measuring a fuel's flash point requires an ignition source. At the flash point, the vapor may cease to burn when the source of ignition is removed. It is not related to engine performance. Rather, the flash point relates to safety precautions that must be taken when handling a fuel. At temperature below the

flash point, not enough fuel evaporates to form a combustible mixture. Gasoline has flash points below the freezing point of water and can ignite in the presence of spark or flame. [Haq, 1995] found that Grade II Diesel fuel have the flash point 51.7⁰C or above. The flash point for vegetable oil is considerably higher than diesel fuel.

E.1.4 Fuel volatility

Fuels function by releasing combustible gases (vapours). Boiling Point is an indicator of volatility, the higher the boiling point, the less volatile the fuel. Vapour pressure is an indicator of volatility, the higher the vapour pressure, the more volatile the fuel. Vapour pressure increases with temperature, so the volatility of a fuel can be increased by raising the temperature. Standard diesel fuel T₉₀ temperature is in the range of 282⁰C to 338⁰C [Goering, 1986]. [Haq, 1995] found that distillation of vegetable oil will not take place at this temperature at atmospheric pressure. Instead decomposition will gradually begin at temperatures of 300⁰C, and distillate will consist of decomposition of products; the 90% point will never be reached

E.1.5 Cetane number

Cetane number or CN is actually a measure of a fuel's ignition delay; the time period between the start of injection and the first identifiable pressure increase during combustion of the fuel. In a particular diesel engine, higher cetane fuels will have shorter ignition delay periods than lower cetane fuels which is desirable. Cetane numbers are only used for the relatively light distillate diesel oils. Fuel accumulates during this delay period and then ignites explosively to produce the characteristic diesel knock. Standard diesel fuels have cetane number 47. [Ramadhas *et al.*, 2004] has studied that Vegetable oil's cetane number is lower to diesel fuel but very close to minimum requirements of no. 2 diesel fuel.

E.1.6 Pour Point and Cloud Point

The ability of a diesel engine to operate in cold weather depends on the pour and cloud points of fuel. The pour point is the lowest temperature at which the fluid movement can be detected. The pour point is important since below that temperature the fuel will not flow to the outlet pipe in the fuel tank. On the other hand, at the

cloud point of petroleum oil is not flow to the outlet pipe in the fuel tank. The cloud point of petroleum oil is the temperature at which paraffin wax or other solid substances begin to crystallize or separate from solution when oil is chilled. The cloud point temperature is usually several degrees higher than the pour point. If the ambient temperature is below the cloud point, then the wax crystals will block the filter. Both these measurements are highly significant when operating at low temperatures. Many Vegetable oils have higher cloud point than diesel fuel [Ali,1994]

E.1.7 Fuel viscosity

Viscosity describes a fluid's internal resistance to flow and may be thought of as a measure of fluid friction. High viscosities of fuel can cause poor atomization, large droplets and high spray jet penetration. With high viscosities, the jet tends to be a solid stream instead of a spray of small droplets. As a result, the fuel is not distributed or mixed with, the air required for burning. This result in poor combustion is accompanied by loss of power and economy. Too low viscosity can lead to excessive internal pump leakage whereas the injector pressure reaches an unacceptable level and will affect injection during the spray atomization. [Altin, 2001] has studied that grade II diesel fuel viscosity at 40⁰C lies in the range of 2 mm²/s to 4.3 mm²/s. Vegetable oil is much more viscous nearly 10 to 12 times than diesel fuel

E.2 Fuel Impurities

The sulfur content, carbon residue and ash are responsible for corrosion and forming a residue on the engine parts which will affect the engine life. These values should be as small as possible. Practical values are 0.5% sulfur, 0.27% carbon residue and 0.01% ash. [Ramadhas, 2004]

Since the 1990's, fuel quality has been increasingly more regulated by the US EPA (Environmental Protection Agency) under the authority of the Clean Air Act. In the context of the increasingly more stringent diesel emission standards, the most important fuel property regulated by the EPA became the sulfur content. Historically,

the sulfur content in diesel fuels for highway and non road vehicles was limited to 0.5% (wt.) by ASTM specifications. Sulfur is not desirable in fuel. Forms acidic by-products that are harmful to the environment, and eat away the engine (SO,SO₂, SO₃).

Diesel Fuel contains small amount of ash forming material, such as suspended solids and soluble organometallic compounds. These ash particles can cause damage to close tolerances of fuel injection system, and abrasive wear on engine components (piston rings).

Diesel fuel should be clear in appearance and free of water and sediment. The presence of these materials generally indicates poor fuel handling practices. Water and sediment can cause shortened filter life or plugged fuel filters which can in turn lead to fuel starvation in the engine. In addition, water can have negative impact on fuel corrosion and on microbial growth. It is for that reason we recommend separate analysis and maximum levels.

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