Sesame Oil as an Alternative Fuel for Diesel Engines in Bangladesh

by

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Table of Contents

List of Figures

List of Tables

List of Symbols, Subscripts and Abbreviations

1. Symbols

A	Non-flow availability
C	Specific heat
C_D	Coefficient of discharge
D	Diameter
G	Gibb's free energy
g	Gibb's free energy per mole of fuel
H	Enthalpy
h	Specific enthalpy
h_m	Manometric deflection
m	Mass
N	Crankshaft rotation per minute
P	Power
p	Pressure
Q	Heat transfer
R	Gas constant
S	Entropy
s	Specific entropy
T	Temperature
t	Time
U	Internal Energy
u	Specific internal energy
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
η_v	Volumetric efficiency
η_b	Brake thermal efficiency
η_{II}	Second law efficiency
μ	Chemical potential
σ	Specific gravity
$\psi_{ch,f}$	Fuel chemical availability

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	Environmetal dead state
wb	Wet bulb
x	Lab condition

3. Abbreviations

A/F	Air-fuel ratio
bmep	Brake mean effective pressure
bsfc	Brake specific fuel consumption
LHV	Lower heating value of fuel
WOT	Wide open throttle

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Abstract

The world now depends mostly on petroleum based fuels but recent concerns over the environment, increasingly higher price and depletion of the limited reserves of petroleum fuels have prompted the interest for development of alternative sources of energy. Vegetable oils seem to be a very promising alternative source of energy which can be employed in compression ignition (CI) engines. Different countries of the world are looking in different vegetable oils that grow enormously in that country. For the present study oil was extracted from sesame seed as it is the second largest oil producing crop in Bangladesh but not so popular as edible oil and several engine performance related properties are tested and compared with that of diesel fuel. The vegetable oil's volatility and heat content is found lower but density and viscosity is found higher. Viscosity is considered as the main obstacles of using them directly as CI engine fuel. Researchers have reported that this obstacle can be overcome by preheating, blending or using them as emulsion or esters. In the present study preheating technique is employed to the crude sesame oil and also to the 80% sesame blended with 20% diesel fuel oil and their effects on engine performance is investigated on a direct injection, 4 stroke, 3 cylinder diesel engine. Engine performance has been evaluated with the help of both conventional performance parameters and availability analysis. It is observed that the engine exhibits better results for higher preheating temperature and performance parameters become comparable with that of diesel when the preheating temperature reaches 80°C. The performance parameters of the blend preheated at 100°C become very close to that of diesel fuel. Availability efficiency shows also similar result but less than brake thermal efficiency. Major portion of the fuel chemical availability is wasted in uncounted factors and nearly 15% wasted with the exhaust gas which can be directly used for preheating. Technically diesel fuel can be replaced with preheated sesame oil but the major obstacle of using them is the high cost of sesame oil relative to diesel fuel.

Chapter 1

Introduction

1.1 Introduction

Since time immemorial securing the supply of energy has been one of mankind's basic needs. Energy is very much necessary to survive and to upgrade the standard of living. The present world mostly depends on petroleum based fuels and with time its consumption is increasing due to growth in population, economy, industrialization and urbanization. Limited petroleum fuel reserves are exhausting quickly and the reserves are highly concentrated in certain region of the world and so, those country does not having the reserves of petroleum has to pay a huge amount of foreign currency for importing petroleum fuel. So, now every country in the world is trying to introduce some alternative sources of energy which is sustainable and reliable and can be produced with the materials available within the country for their energy safety and sustainable development.

The compression ignition (CI) engines, also known as diesel engines, are generally high performance engines and play a great role particularly in the field of transportation (e.g., automobiles, ships), industrial sectors (e.g., power generation) and agricultural applications (irrigation pumps, tractors) and so on. The growth of internal combustion engines boosted from 1910 with the mass production of automobiles. With the increase of internal combustion engine, the demand for liquid petroleum fuel is also increased and resulted in increasing of atmospheric pollution. These concerns have highlighted the need for diversification and prompted increased research throughout the world into the potential fuel alternatives for internal combustion engines.

The use of vegetable oils, as alternative fuels for diesel engines dates back at around a century. Due to the rapid decline in crude oil reserves and increase of petroleum fuel price, the use of vegetable oils is prompted again in many countries. Depending upon climate and soil conditions, different countries are looking into different vegetable oils- for example, soybean oil in USA, rapeseed oil and sunflower in Europe, olive oil in Spain, palm oil in Southeast Asia, mainly in Malaysia and Indonesia and coconut oil in the Philippines are considered to substitute diesel fuels (Srivastava and Prasad, 2000). Different researcher's results show the vegetable oils are promising alternative fuels for CI engines. Because of high viscosity, low volatility and low cetane number of the vegetable oils, long term tests exposed some of the limitations with respect to injector choking, engine knocking, improper atomization to result in CO emission, gum formation, lubricant contamination and heavy particulate emissions (Altn *et al.*, 2001). But sulfur content of vegetable oil is negligible and hence the environmental damage caused by sulfuric acid is greatly reduced. However, the vegetable oil takes away more CO_2 from the atmosphere during their production than is added to it by their later combustion. Thereby, these oils alleviate the global warming problem (Srivastava and Prasad, 2000).

In the case of alternative fuels, the engines may need to be modified for which these were not originally designed and these may not generate optimum efficiency comparing to the conventional fuel. Researchers have indicated that higher viscosity of vegetable oil is the main impediment to possible problems encountered by engines. Therefore, methods of fuel property modification are evolved by them also. Economic and social impacts associated with the use of vegetable oil as fuel for IC engine need to be addressed further. If substantial amounts of vegetable oils are diverted from traditional markets for usage as fuel impacts on producers and consumers will be large and wide ranging.

1.2 Scopes of the Thesis

The present thesis reports the study of a diesel engine run by preheated raw sesame oil and preheated sesame oil blended with diesel. The oil was collected by crushing the sesame seeds and used in the crude form. Sesame oil is chosen as it is the second largest oil producing crop in Bangladesh but not so popular as edible oil. Experiments were carried out in a 3-cylinder, 4-stroke, air cooled diesel engine at three different speeds (e.g., 1750, 2000, 2250 rpm) for different loading conditions.

Initially the engine was run by diesel fuel under different loads at three different speeds and then the experiments were repeated replacing diesel by sesame oil under preheating temperatures of 60, 80 and 100°C and 80% sesame blended with 20% diesel under preheating temperature of 100°C. In doing so, various engine performance parameters are obtained, analyzed and reported.

In this thesis, available literature is reviewed in chapter 2. A brief introduction on the vegetable oil and a comprehensive comparison between the characteristics of diesel fuel and vegetable-oil fuels is compiled in chapter 3. Engine performance parameters for both the conventional practice and the analysis based on the second law of thermodynamics are discussed in chapter 4. Experimental procedure and Experimental setup is discussed in chapter 5. Outcomes of the experimental works, final conclusion of the thesis work and recommendations for further works are discussed in chapter 6.

Chapter 2

Literature Review

Fossil fuel accounts about 80% of the total commercial energy consumption of the world and oil share is about 40%. Oil consumption of the world is increased nearly 3.5% annually but total conventional oil reserves of the world has not significantly increased for the last 20 years. Again total oil reserves of the world is very unequally distributed, middle east accounts about 62% of the total reserves of the world where as Asia Pacific region accounts only 3.5% but in this region more than two third of the world population live (bp statistical review, 2005). Most of the countries of this region are developing. It becomes a great challenge for these countries to afford petroleum fuel for every sector. Increasing higher price of petroleum fuel makes the problem more challenging. At the same time securing the sources of energy is a fundamental need for every country and without sustainable sources sustainable development is impossible. So, every country is looking for alternative sources of energy, which is reliable, sustainable and can be produced with the materials available within the country. Vegetable oil is a very promising alternative source of energy.

The use vegetable oil as diesel fuel is not a new concept. Dr. Rudolph Diesel used peanut oil to fuel one of his early engines at the Paris Exposition of 1900. Vegetable oil fuels, however, were not accepted because of their high price compare to petroleum fuel. Now the interest for vegetable oil as a substitute to diesel fuel is enhanced due to the following reasons:

a) Petroleum as an exhaustible natural resource, its supply is expected to run out within this century, thereby making it necessary to find alternative fuels.
b) Petroleum fuel price is increasing with time.
c) Vegetable oil is renewable in nature and less harmful to environment.

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

a) The price of vegetable oil is dependent on the feedstock price.
b) The feedstock homogeneity, consistency and reliability are questionable. Homogeneity of the products depends on the supplier, feedstock and production methods.
c) Storage and handling of vegetable oil is difficult.
d) Compatibility of vegetable oil with diesel engine needs to be studied further.
e) Cold weather operation of the engine in not easy with vegetable oil.
f) Continuous availability of vegetable oil needs to be assured before embarking on the major use of it in IC engine.

The major technical areas (regarding the use of vegetable oils as fuels in IC engine), which need further attention, may be as follows (Ramadhas *et al.*, 2004)

a) Developing of less expensive quality tests.
b) Study on the effects of oxidized fuel on the engine performance and its durability.
c) Emission testing with a wide a range of feed stocks.
d) Studying on developing specific markets such as mining, municipal water supplies etc. that can specify bio-diesel as the fuel choice for environmentally sensitive areas.
e) Efforts to be focused on responding to fuel system performance, material compatibility, petroleum additive compatibility and low fuel stability under long term storage.
f) Environmental benefits over diesel fuel need to be polarized.
g) Attention is needed to reduce production cost to develop low feedstock and identify potential markets in order to balance the cost and availability.
h) Development of additives for improving cold flow properties, material compatibility and prevention of oxidation in storage.

In short term tests, vegetable oil's higher viscosity and lower volatility is the main reason of its poor performance. In long term test, rapid cocking of injector nozzle and other parts, e.g., piston heads, intake valves and exhaust valves, gumming up of the piston ring, higher lubricating oil contamination have been detected. (Bhattacharya and Reddy, 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. The 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 ways (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 collidal size.

3) Dilution: Dilution of vegetable oils can be accomplished with such materials as diesel fuel, as ethanol or kerosene.

4) Transesterification: It is the displacement of alcohol from an ester by another alcohol in a process similar to hydrolysis except then an alcohol is used instead of water. This process has been widely used to reduce the viscosity of vegetable oils.

Researcher in different countries are emphasizing on different vegetable that is hoped to be highly potential vegetable fuel for their respective country.

Rapeseed Oil

Peterson *et al.* (1987) carried out short term and long term test with high eurusic acid rapeseed (winter rape) and low eurusic 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 oil 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.

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, exhaust temperature. Engine related wear and lubricating oil contamination have been studied with processed sunflower oil and also with a 25/75% blend of crude sunflower oil and diesel oil. Lubricating oil contamination was found to be higher for the blend. Iron concentration in lubricating oil was the greatest when operating on de-gummed and de-waxed sunflower oil (Bhattacharyya and Reddy,1994).

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 loses 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 not 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.

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.

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 increase lubricating oil temperature and engine coolant temperature increase of soybean oil blend. While exhaust temperature increased over time, the engine compression pressure decreased with time.

Single cylinder engine screening with tracers, fluid modeling using high-speed photography and multi-cylinder durability tests were carried out to define a combustion system that could give a normal life even at higher engine ratings. Data on power output, thermal efficiency and lubricating oil from screening tests suggested a 25/75% blend of soybean oil and diesel. A higher injection pressure was used to enable proper atomization of the highly viscous vegetable oils in the experiments, with a pre-combustion chamber type diesel engine. The best injection timing and injection pressure for proper atomization were determined to 16^0C BTDC of piston position and injection pressure 20.6 MPa. The increase in thermal efficiency and lower black smoke concentration were attributed to the greater oxygen content in the vegetable oil tested (Bhattacharyya and Reddy, 1994).

Neem Oil

Bansal and Juneja, first reports 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^{0}C respectively. The authors suggested heating of fuel by recirlculating the exhaust to avoid cold starting problems during winter. The distillation curves reveal that thermal cracking of neem oil blends takes place earlier than diesel fuel and thus results in more smoke, particulates and fouling of lubricants if the blend contains more than 30% neem oil. Long term endurance tests, however, were not carried out (Bhattacharyya and Reddy, 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 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 as par with Malaysian gradeII petroleum diesel fuels. Except for the higher specific gravity, viscosity and narrowed boiling range, their properties closely matched those of diesel fuel (Bhattacharyya and Reddy, 1994).

Cottonseed 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 oil was 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 was attributed to low heat content and increased viscosity of blends compared to pure diesel oil. Further, fuel economy was higher with two stage combustion than with single-stage combustion system.

Jatropha Oil

K pramanik (2002) conducted a test using jatropha oil and diesel blends in different proportions. Blends containing up to 30% by volume jatropha oil have viscosity values close to that of diesel fuel. The viscosity of blends containing 70 and 60% vegetable oil became close to that of diesel in the temperature ranges of 70-75 and 60-65 ^{0}C, respectively. From his test result it has been established that up to 50% jatropha curcas oil can be substituted for diesel for use in CI engine without any major operational difficulties.

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 viscosity of this blend becomes comparable to that of diesel and engine performance with this blends also become close to that of diesel.

Chapter 3

Vegetable Oil as an Alternative to Diesel Fuel

3.1 Composition Vegetable Oils

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 polyunsaturated.

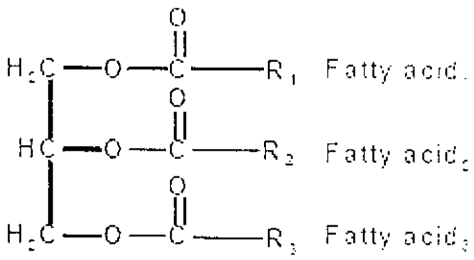

Fig. 3.1: General Structure of Triglycerides.

Table 3.1: lists of the most common fatty acids and their methyl esters (NERL,2004).

Fatty acids(trivial name/rational name)	*Structure*	*Common acronym*	*Class*
Palmitic acids/ Hexadeconaic acids	CH3-(CH2)14-COOH	C16:0	Saturated
Stearic acid/ octadeconaic acid	CH3-(CH2)16-COOH	C18:0	Saturated
Oleic acid/ 9(z) octadeconaic acid	CH3-(CH2)7-CH=CH-(CH2)—COOH	C18:1	Un-saturated
Linooleic acid/ 9(z),12(z) octadecadienoic acid	CH3-(CH2)4-CH=CH-CH2-CH=CH-(CH2)7-COOH	C18:2	Poly-unsaturated
Linolenic acid/9(z),12(z),15(z)-octadecotrionic acid	CH3-CH2-CH=CH-CH2—CH=CH-CH2-CH=CH-(CH2)7-COOH	C18:3	Polyunsaturated

Vegetable oil differs from each other due to difference in amounts of several fatty acids. Different fatty acids affect the stability and melting point of oil. As the amount of unsaturation increases so does the relative rate of oxidation. A vegetable oil that has a high degree of unsaturation would tend to polymerize sooner in the crankcase of an engine than a vegetable oil that is highly saturated. Also, as the amount of unsaturation increase, the melting point on the other hand, decreases. Thus, a vegetable oil with a high degree of saturation tends to be a solid at room temperature. A compromise between a low melting point and high oxidation rate must be made when selecting a suitable vegetable oil for use in diesel engines.

Table 3.2: Common fatty acids in some common vegetable oil (Demirbas, 1998)

Vegetable oil				*Fatty*	*acids*			
	16:0	*16:1*	*18:0*	*18:1*	*18:2*	*18:3*	*20:0*	*Others*
Cotton seed	28.7	0	0.9	13.0	57.4	0	0	0
Linseed	5.1	0.3	2.5	18.9	18.1	55.1	0	0
Peanut	11.4	0	2.4	48.3	32.0	0.9	1.3	3.7
Rapeseed	3.5	0	0.9	64.1	22.3	8.2	0	0
Safflower	7.3	0	1.9	13.6	77.2	0	0	0
Soybean	13.9	0.3	2.1	23.2	56.2	4.3	0	0
Sesame	13.1	0	3.9	52.8	30.2	0	0	0

3.2 Standard Fuel Properties of Vegetable Oils

The internal combustion engine was invented more than one hundred years ago and numerous improvements have been made since its invention. The development of fuels paralleled the development of the engine. Many standards concerning the required properties of engine fuels and tests for measuring those properties have been set. 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 of the many standards will be discussed here.

3.2.1 Specific Gravity of Fuels

Specific gravity is a measure of the density of liquid fuels. It is the ratio of density of fuel at 15.6^0C 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)

3.2.2 Heating Value of Fuels

The heating value is an important measure of finding out released energy of doing work. 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. Published heats of combustion are usually higher heating values and are therefore often used to calculate engine efficiencies. Standard diesel fuel heating value is around 42.5 MJ/kg and vegetable oil heating value is nearly 2 to 3 % less (Demirbas, 1983)

3.2.3 Fuel Volatility

Fuels must vaporize before they can burn. Volatility refers to the ability of fuels to vaporize; fuels that vaporize easily at lower temperatures are more volatile than fuels that require higher temperatures to vaporize. Volatility of diesel fuel is important, although it is not as critical as the volatility of gasoline. But if the fuel is too volatile, the fuel droplets will evaporate too quickly to permit adequate spray penetration in the chamber. As for example, a low T_{10} temperature will permit easier starting of diesel engine. A low T_{50} temperature helps to minimize diesel smoke and odor. A low T_{90} temperature helps to minimize crankcase dilution and improves fuel economy. Standard diesel fuel T_{90} temperature is in the range of 282^0C to 338^0C (Goering, 1986). Distillation of vegetable oil will not take place at this temperature at atmospheric pressure. Instead decomposition will gradually begin at temperatures of

300^0C, and distillate will consist of decomposition of products; the 90% point will never be reached (Haq, 1995).

3.2.4 Flash Point of Fuels

Flash point varies with fuel volatility. It is not related to engine performance. Rather, the flash point relates to safety precautions that must be taken when handling a fuel. The flash point is the lowest temperature to which a fuel must be heated to produce an ignitable vapor-air mixture above the liquid fuel when exposed to open flame. At temperature below the flash point, not enough fuel evaporates to form a combustible mixture. Insurance companies and governmental agencies classify fuels according to their flash points and use flash points in setting minimum standards for the handling and storage of fuels. Gasoline has flash points well below the freezing point of water and can readily ignite in the presence of spark or flame. Grade II Diesel fuel have the flash point 51.7^0C or above. The flash point for vegetable oil is considerably higher than diesel fuel (Haq, 1995).

3.2.5 Fuel Viscosity

Viscosity may be defined as the measure of liquid's resistance to flow. The viscosity of diesel fuel is important primarily for its effect of handling by pump and injector system. The direct ignition in the open combustion chamber through the nozzle and pattern of fuel spray decides the ease of combustion and thermal efficiency of the engine. The effect of viscosity is critical at low or light load conditions. Fuel viscosity exerts a strong influence on the shape of the fuel spray. High viscosities 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. In small engines, the fuel spray may impinge in the cylinder walls washing away the lubricating oil film and causing dilution of crankcase oil. Such a condition constitutes an excessive wear. Grade II diesel fuel viscosity at 40^0C lies in the range of 2 mm^2/s

to 4.3 mm^2/s. Vegetable oil is much more viscous nearly 10 to 12 times than diesel fuel (Altin, 2001).

3.2.6 Cetane Number

The cetane number is a measure of the ignition delay period that is, the time delay between the beginning of injection into the combustion chamber and the beginning of combustion. Fuel accumulates during this delay period and then ignites explosively to produce the characteristic diesel knock. A short delay period, signified by a high cetane number, is desirable. Standard diesel fuels have cetane number 47. Vegetable oil's cetane number is inferior to diesel fuel but very close to minimum requirements of no. 2 diesel fuel. (Ramadhas *et al.*, 2004)

3.2.7 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)

3.3 Fuel Impurities

Crude oil contains many impurities that could be harmful in storage and use of petroleum fuels. Refining removes most of the impurities. Therefore, fuel specifications must include the permissible concentration of each impurity.

Sulfur

Sulfur compounds are present in crude oil, but most of the sulfur is removed during refining. Sulfur is undesirable in fuels. Sulfur oxides formed during combustion can convert to acids and cause corrosion of engine parts and accelerated wear of rings and cylinder liners. Typically, sulfur concentrations in gasoline are much smaller than in diesel fuel. In the USA, sulfur content of Gasoline averages less than 0.03% by weight and nearly always is less than 0.15%. The SAE maximum for sulfur content of diesel fuel is 0.5% by weight. Potential corrosively of sulfur is measured by immersing a copper strip in fuel for three hours at 50^0C and then comparing the resulting corrosion with that on standard corroded strip. Vegetable oil contains less sulfur than diesel fuel (Dorado, 2003).

Ash

Ash consists of small solid particles and oil or water –soluble metallic compounds found in fuels. Ash is measured by burning a small quantity of the fuel until all the combustible materials has been consumed. Ash is particularly harmful in diesel fuel. The abrasiveness of ash can contribute to wear of the close-fitting parts in the injection system and can contribute to plugging of the fuel filter and injection nozzles. Grade II diesel has the maximum allowable ash content by .01% (Goering, 1986) .Vegetable oil contain higher amount of ash than diesel fuel (Haq, 1995)

Water and Sediment

Water and sediment can enter fuel during handling and storage. Water vapor can move into storage tanks through vented openings and condense into liquid when air temperatures drop. Rain may also leak into some tanks and ground water may seep into leaky underground storage tanks. Sediment can consist of rust from metal tanks and/or small dirt particles that enter tanks with air or through careless handling. The presence of water in petroleum promotes the formation of slime and other organic sediment. Gasoline should be visually free of un-dissolved water and sediment; it should be clear and bright when observed at 21^0C. The SAE limits for water and sediment in diesel fuel are 0.05% by volume (Goering, 1986).

Chapter 4

Engine Performance Parameters

4.1 Conventional Engine Performance Parameters

4.1.1 Power Equations for CI Engines

A comprehensive CI engine analysis can be done by examining the energy input contained in the fuel and the available output power. However, output power available at the fly wheel is known as brake power, P_b. For multi operation, it is convenient to utilize brake thermal efficiency, η_b, which is defined as the ratio of brake power divided by the engine input contained in the fuel (Heywood, 1988).

Algebraically,

$$\eta_b = \frac{P_b}{m_f . Q} \tag{4.1}$$

Where

P_b : Brake power, measured at flywheel

η_b : Brake thermal efficiency

m_f : Mass flow rate of fuel

Q : Lower heating value of fuel.

If the fuel flow rate, m_f in Eqn. 4.1 is replaced by $F.m_a$, where, F is the fuel air ratio and m_a is the mass flow rate of air can be written as -

$$P_b = \eta_b . F . m_a . Q \tag{4.2}$$

Fuel-air ratio, F may be replaced by introducing equivalence ratio, ϕ, defined as the actual air-fuel ratio, divided by stoichiometric air fuel ratio, Fs (Heywood, 1988)

$$\phi = F/Fs \tag{4.3}$$

Equivalence ratio, ϕ, has the same meaning on the mole or mass basis.

$$P_b = \eta_b . \phi . Fs. m_a . Q \qquad (4.4)$$

The power output of an internal combustion engine is limited by its breathing capacity, there is never difficulty in introducing any desired quantity of fuel into the cylinder but the maximum amount that can be burned depends entirely on the amount of air available for combustion. If air were an incompressible, inviscid medium, then the displacement volume of the engine would be filled up into each two revolution for a four stroke engine. Air, however, is not an incompressible inviscid medium and the mass of air, which enters the cylinder, is somewhat less, depending on the engine speed and also on the residual gas present after exhaust process (Lumely, 1999).

Therefore, volumetric efficiency, η_v is introduced to account for the effectiveness of the induction and exhaust process and in terms of quantity applied to an actual engine, it is defined as the mass of fuel and air inducted into the cylinder, divided by the mass of the mixture that would fill the piston displacement volume at inlet density, ρ_i in the intake manifold.

For four stroke CI engines, volumetric efficiency, η_v can be expressed as -

$$\eta_V = \frac{2m_a}{NV_d \rho_i} \qquad (4.5)$$

Where

m_a : Mass of fresh air inducted per unit time

N : Number of engine revolution per unit time

V_d : Total displacement volume of the engine

ρ_i : Inlet air density.

The factor 2 arises from the fact that in the four stroke engine there is one cycle for every two crank revolution and displacement, V_d is given by-

$$V_d = n_c.A_p.S = n_c.(\Pi/4).b^2.S \tag{4.6}$$

Where

n_c : Number of engine cylinders

A_p : Engine piston area

S : Engine stroke length

b : Engine cylinder bore.

Combining equations, 4.5 and 4.6 results in the general power equation for engine. The appearance of chocking in a component prevents the piston from any more air into the cylinder and the volumetric efficiency drops. Its value can be improved by changes in valve timing, cam modifications, porting, intake and exhaust tuning and changes in the number of valves per cylinder (Pulkrabek, 2006).

4.1.2 Brake Mean Effective Pressure (*bmep*)

This is a constant pressure, which acting on the piston area through stroke would produce brake power at the flywheel. Therefore,

$$\begin{aligned} P_b &= bmep.V_d.(N/2) \\ &= bmep.A_p.S.n_c.(N/2) \end{aligned} \tag{4.7}$$

This mean pressure is fiction, but is useful as it is roughly comparable even in very different engines, as these different engines burn same fuel, necessarily under approximately the same conditions and hence similar pressures. In SI engines, typical WOT *bmep* at present is between 0.9 and 1.1 Mpa, however CI engines have 25-30% lower value of *bmep* because of much leaner combustion (Pulkrabek, 2006).

Comparing Eqns. 4.4, 4.6 and 4.7, *bmep* can be written as-

$$bmep = \eta_b.\eta_v\ Ø.F_s.\rho_i.Q \tag{4.8}$$

From Eqn. 4.8, it is clear that engines of completely different characteristics, but equal efficiency, will have equal value of *bmep*. Engines of different size or same engine running at different speeds can be compared with *bmep*. Table 4.1 illustrates the comparison of a model airplane running at 11400 rpm and a typical marine diesel engine running at 160 rpm.

Table 4.1: Comparison of two internal combustion engines (Ferguson and Kirkpatrick, 2000)

Characteristics	*Model Airplane*	*Marine*
B (m)	0.0126	0.737
S(m)	0.0131	1.016
V_d (m^3)	1.6×10^{-6}	0.433
P_b (kW)	0.1	529
bmep (MPa)	0.32	0.45

Relatively small differences in *bmep* between these two figures are grossly attributed to the compression ratio.

4.1.3 Brake Specific Fuel Consumption (*bsfc*)

Brake specific fuel consumption is the rate of consumption of fuel per unit time and per unit power output. The brake specific fuel consumption (*bsfc*) can be written as

$$bsfc = \frac{m_f}{P_b} = \frac{1}{m_f . Q} \tag{4.9}$$

For an engine operating on a given fuel, it is clear that the *bsfc* is a measure of the inverse of the product of the effectiveness. If one or more of the effectiveness goes up, the *bsfc* goes down and vice versa. However, two different engines can be compared on a *bsfc* basis provided they are operated on the same fuel.

4.2 Thermodynamic Analysis of CI Engine Processes

In order to render a critical look into the performance of the engine exergy analysis, which is a combination of both first and second law of thermodynamics is utilized. Exergy is a thermodynamic property of a system that provides a measure for the maximum useful work that a given system may attain as the system is allowed to reversibly transition to a thermodynamic state which is in equilibrium with its environment. Exergy is also termed as availability, which is not conserved like energy but destroyed due to irreversibility (Caton, 1999).

A performance parameter for a process based on availability concepts is known as exergetic efficiency, η_{II} or availability efficiency, η_{av} or second law effectiveness, ϵ or simply as effectiveness. A first law efficiency gauges how well energy is used where as effectiveness indicates how well exergy or availability is used. First and Second law efficiency also differ in another important respect. The first law is a conservation principle and first law cycle efficiency compare desired energy output to required energy input. On the other hand entropy and availability from a second law view point are non conserved property. In the presence of irreversibility entropy is produced and availability is destroyed. Hence second law efficiency measures losses in availability during a process (Wark, 1995). A general definition of a second law efficiency is:

$$\begin{aligned}\eta_{II} &= \frac{useful\ availability\ output}{availability\ input}\\ &= 1 - \frac{availability\ destruction\ and\ losses}{availability\ input}\end{aligned}$$

where, losses imply non-useful transfers across the boundary .

A second approach especially useful for steady state devices is:

$$\eta_{II} = \frac{rate\ of\ exergy\ output}{rate\ of\ exergy\ input}$$

4.2.1 Availability Analysis of Engine Processes

In the engine performance analysis, the focus is mainly in amount of useful work that can be extracted from the combustibles within the cylinder at every point of the operating cycle. The key problem is to determine the maximum possible work output (or minimum possible work input) when the charge within the cylinder is taken from one specified state to another in the presence of a specified environment and the atmosphere. The first and second laws of thermodynamics together define this maximum or minimum work, which is expressed in terms of availability or exergy (Heywood, 1988)

Consider the system-atmosphere combination shown in Figure 4.1

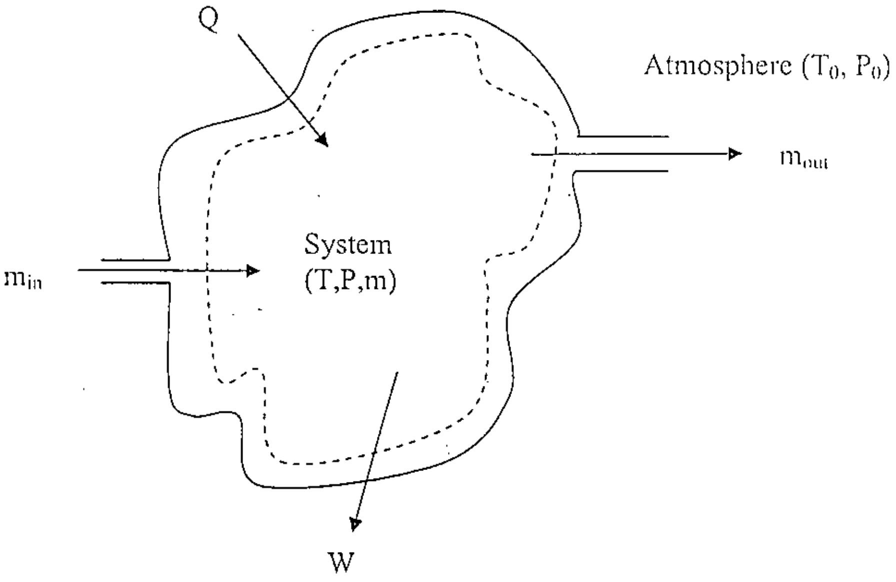

Fig. 4.1: System-atmosphere configuration for availability analysis of engine processes

In the absence of mass flow across the system boundary, as the system changes from state 1 to state 2, the first and second laws give

$$W_{1\text{-}2} = -(U_1 - U_2) + Q_{1\text{-}2}$$
$$Q_{1\text{-}2} \leq T_0 (S_2 - S_1)$$

Combining these two equations give the total work transfer:

$$W_{1\text{-}2} = -[(U_2 - U_1) - T_0 (S_2 - S_1)] \qquad (4.10)$$

The work done by the system against the atmosphere is not available for productive use. It must, therefore, be subtracted from the total work to obtain the useful work transfer:

$$W_{1-2} = -[(U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)] \qquad (4.11)$$

The maximum useful work will be obtained when the final state of the system is in thermal and mechanical equilibrium with the atmosphere. The availability or exergy of this system is:

$$A = U + P_0V - T_0S \qquad (4.12)$$

Availability is the property of the system-atmosphere combination which defines its capacity for useful work. The useful work such a system-atmosphere combination can provide, as the system changes from state 1 to state 2, is less than or equal to the change in availability.

$$W_{u,1-2} \leq -(A_1 - A_2) \qquad (4.13)$$

When mass flow across the system boundary occurs, the availability associated with this mass flow is

$$G = H - T_0S \qquad (4.14)$$

Which is the Kennan function at temperature T_0. (In general G= H-TS is known as Gibbs function, but in case of atmospheric condition it is termed as Keenan function)

With these relations, an availability balance for the gas working-fluid system around the engine cycle can be carried out. For any process between specified end states which this system undergoes (interacting with the atmosphere), the change in availability ΔA is given by-

$$\Delta A = A_{in} - A_{out} - A_{destroyed} \tag{4.15}$$

The availability transfers in and out occur as a result of work transfer, heat transfer and mass transfer across the system boundary. The availability transfer associated with heat transfer occurring with the system temperature T is given by:

$$dA_Q = \partial Q\left(1 - \frac{T_0}{T}\right) \tag{4.16}$$

Since both an energy and entropy transfer occurs across the system boundary. The availability associated with mass transfer is Eqn. 4.14.

Availability is destroyed by the irreversibilities that occur in any real process. The availability destroyed is given by:

$$A_{destroyed} = T_0 \, \Delta S_{irrev} \tag{4.17}$$

Where, ΔS_{irrev} is the entropy increase associated with the irreversiblities occurring within the system boundary.

4.2.2 Chemical Availability of Fuels

The chemical availability (or chemical exergy) is the maximum theoretical work that could be developed by a combined system consisting of a system of interest and exergy reference environment. The thermo-chemical availability is the value for work. The sum of the thermo-chemical and chemical exergies is the total exergy (total availability) associated with a given system at a specified state, relative to a specified exergy reference environment.

The availability changes and the irreversibility for chemical reactions of hydrocarbons fuels can be suited from two viewpoints; firstly, fuel and dry air composed of O_2 and N_2 react to form products of combustion in the restricted dead state and secondly, fuel and dry air composed of O_2 and N_2 react to form products of combustion which ends up in the environmental (unrestricted) dead state. The difference between these two is the chemical availability of the product gasses as they proceed from restricted to un-restricted dead state.

To extend these concepts to the most general situations, a steady-state control volume is considered where the fuel enters at the restricted dead state, the air (oxidant) is drawn from the environment, and the products are returned to the unrestricted dead state. Fig. 4.2 illustrates a control volume where pure fuel enters at the restricted dead state T_0 and P_0 and oxidant (O_2) enters from the environment. In general, each product is at its state in the environment, namely, $h_{i,00}$ $S_{i,00}$, $g_{i,00} = \mu_{i,00}$ (Wark, K. Jr., 1995)

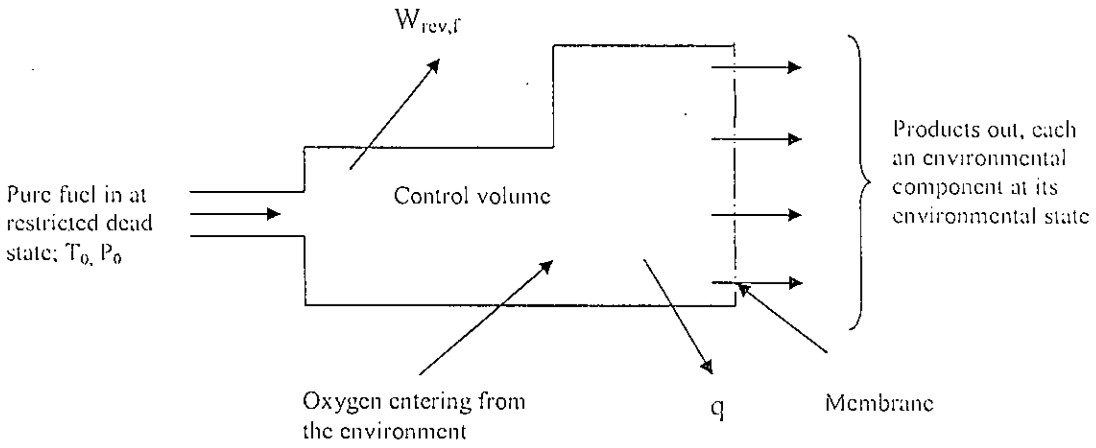

Fig. 4.2: Control volume for the development of fuel availability, oxygen enters and products leave through semi-permeable membrane.

A semi-permeable membrane is used to introduce oxygen into the control volume. The products of combustion, primarily CO_2 and H_2O leave control volume through semi-permeable membranes in order to adjust their state following combustion to their unrestricted dead state. The maximum work output per mole of fuel shown in Fig. 4.2 is $W_{rev,f}$. In addition, heat transfer, q with the environment will be necessary to assure that the products leave at T_0.

The maximum work $W_{rev,f}$ is found by combining the energy and entropy statements for the processes shown in Fig.4.2. In steady state the energy equation per mole of fuel is

$$q + W_{rev,f} = \Delta H = \sum_{P-R} \nu_i \bar{h}_i = \sum \nu_i \bar{h}_{i,00} - \nu_{O_2} \bar{h}_{O_2,00} - \bar{h}_{fuel,0} \tag{4.18}$$

Where, $\bar{h}_{i,00}$ is the partial molal enthalpy in the unrestricted dead state and $\bar{h}_{fuel,0}$ is the partial molal enthalpy of the fuel at the restricted dead state. For an isothermal reversible process Q=T.ΔS. In terms of fuel oxidation process this becomes

$$q = T_0 \sum_{P-R} \nu_i \bar{S}_i = T_0 (\sum_{P} \nu_i \bar{S}_{i,00} - \nu_{O_2} \bar{S}_{O_2,00} - \bar{S}_{fuel,0} \tag{4.19}$$

Elimination of q between these two equations yields, after rearrangement,

$$W_{rev,f} = (\sum_{P} \nu_i (\bar{h}_{i,00} - T_0 \bar{S}_{i,00}) - [h_{00} - T_0 S_{00}]_{O_2} - (\bar{h}_0 - T_0 \bar{S})_{fuel} \tag{4.20}$$

Where, ν_i is the stoichiometric coefficient for the ith species.

Finally, recall that chemical potential for the ith species is given by $\mu_i = \bar{g}_i = \bar{h}_i - T\bar{S}_i$. As a result, the preceding equation in terms of chemical potential becomes:

$$W_{rev,f} = \sum_{P} \nu_i \mu_{i,00} - \nu_{O_2} \mu_{O_2,00} - \mu_{fuel,0} \tag{4.21}$$

The first two terms on the right are measured at the environment state, whereas the last term is evaluated at the restricted dead state. $W_{rev,f}$ measures the chemical availability of fuel as it exits alone in the restricted dead state. The availability function ψ is the negative of the work output due to the adopted sign convention on work interaction. Therefore the fuel chemical availability $\psi_{ch,f}$ for the pure fuel in the restricted dead state is given by

$$\psi_{ch} = g_{fuel,0} + \nu_{O_2}\mu_{O_2,00} - \sum_{P} \nu_i \mu_{i,00} \tag{4.22}$$

Where, $g_{fuel,0}$ replaces $\mu_{fuel,0}$ for a pure substance. Use of Eqn. 4.22 requires selecting appropriate values for the reference chemical potentials. A chemical potential can be associated with a fuel in the environmental dead state, even though it is not a compound which makes up the standard environment.

For ideal-gas mixtures the chemical potential of the ith component takes on the format

$$\mu_{i,T,ideal} = g^0_{i,T} + RT\ln\frac{P_i}{P_0} \tag{4.23}$$

For the environmental state, where $P_i = y_{i,00} P_0$, this equation becomes

$$\mu_{i,00} = g_{i,00} + RT\ln y_{i,00} \tag{4.24}$$

For the complete combustion of a hydrocarbon fuel C_xH_y the only products of interest are carbon dioxide (CO_2) and water (H_2O) and the only environmental reactant is oxygen (O_2). Therefore the three equations required for equation (4-24) are:

$$\mu_{O_2,00} = g_{O_2,00} + RT\ln y_{O_2,00}$$
$$\mu_{CO_2,00} = g_{CO_2,00} + RT\ln y_{CO_2,00}$$
$$\mu_{H_2O,00} = g_{H_2O,00} + RT\ln y_{H_2O,00}$$

The quantity $g_{i,0} = g^0_{i,T_0}$ that is the standard state Gibbs function at T_0. This latter value is readily available for numerous substances. In these equations the $y_{i,00}$ values in the atmospheric also must be known, including water vapor. When the set of three equations for $\mu_{i,00}$ are substituted into equation 4.22, we find that

$$\psi_{ch,f} = -\Delta G_{R,0} + RT_0\ln\frac{(y_{O_2,00})^{\nu_{O_2}}}{(y_{CO_2,00})^{\nu_{CO_2}}(y_{H_2O,00})^{\nu_{H_2O}}} - \tag{4.25}$$

Where, $\Delta G_{R,0} = \nu_{H_2O} \cdot g_{H_2O,0} + \nu_{CO_2} \cdot g_{CO_2,0} - g_{fuel,0} - \nu_{O_2} g_{O_2,0}$

The quantity $\Delta G_{R,0}$ is the change in the Gibbs function per mole of fuel for the stoichiometric reaction at the state (T_0, P_0). For a hydrocarbon of general formula C_xH_y, the complete combustion with environmental O_2 to produce CO_2 and H_2O is:

$$C_xH_y + \left(x + \frac{y}{4}\right) O_2 \rightarrow xCO_2 + \frac{y}{2} H_2O(g)$$

In this format equation (4-22) becomes -

$$\psi_{ch,f} = -\Delta G_{R,0} + RT_0 \ln \frac{(y_{O_2,00})^{x+\frac{y}{4}}}{(y_{CO_2,00})^{x} (y_{H_2O,00})^{\frac{y}{2}}} \qquad (4.26)$$

where, $\Delta G_{R,0} = x.g_{H_2O,0} + \left(\frac{y}{2}\right).g_{CO_2,0} - g_{fuel,0} - \left(x + \frac{y}{4}\right) g_{O_2,0}$

Eqn. (4-25) and (4-26) enable one to evaluate the chemical exergy of a mole of fuel C_xH_y in the restricted dead state which is transformed into the products of CO_2 and H_2O in the unrestricted dead state or environmental state. All the Gibbs function data needed, $g_{i,0}$ and $g_{fuel,0}$ are for pure state at $T_0 = 25^0C$ and $P_0 = 1$ atmosphere. The values of Gibbs function of formation are found from JANAF thermo chemical tables and other sources.

4.2.3 Approximation Methods to Find Fuel Availability of Higher Hydrocarbons

The steady-flow fuel availability of light hydrocarbon gases and their mixtures can be evaluated accurately when the composition is known. As noted above the exergy of pure liquid fuels can also be ascertained when vapor-pressure data available. A question arises to the fuel availability of heavy hydrocarbon fuel which are pure or of undetermined composition. For this latter substance data are usually lacking to calculate $\psi_{ch,f}$. However, investigation indicates that for large number of light hydrocarbons the ratio of $\psi_{ch,f}$ to the lower heating value (LHV) is reasonably constant (Wark, 1995).

Brzustowski and Brena examined the ratio $\frac{\psi_{ch}}{LHV}$ homologous series of hydrocarbons. They concluded that the proportionality constant between fuel exergy and LHV is closed to 1.065 based on data at 65 percent relative humidity (Wark, 1995). The correlation improved as the molecular weight increased and was reasonably independent of the homologous series used. Another important analysis pointed out by the researchers that fuel exergy is not very sensitive to relative humidity. The authors concluded that these reasoning are applicable to heavy hydrocarbon of undetermined composition. It is to be noted, at this stage, that the fuel chemical availability $\psi_{ch,f}$ when calculated using approximation method is usually denoted by A_{in} (Al-Najem and Diab, 1991, Caton, 1999). As we have adopted some empirical relations for measuring the fuel chemical availability so, from now we will use this notation instead of $\psi_{ch,f}$.

Szargurt and Styrylska developed the following correlation for computing the chemical availability of liquid hydrocarbons having the general formula $C_xH_yO_zS_w$: (Rakopoulos, C. D. and Giakoumis, E. G., 2006)

$$A_{in} = Q_{in}\left[1.0374 + 0.0159\frac{y}{x} + 0.0567\frac{z}{x} + 0.5985\frac{w}{x}\left(1 - 0.1737\frac{y}{x}\right)\right] \qquad (4.28)$$

Moran developed the simple correlation for liquid hydrocarbons having the simple formula of C_xH_y: (Rakopoulos and Giakoumis, 2006)

$$A_{in} = Q_{in}\left[1.04224 + 0.011925\frac{y}{x} - \frac{0.42}{x}\right] \tag{4.29}$$

Equation 4.28 and 4.29 both can be used for determining chemical availability of both conventional and non-conventional fuels. But equation 4.28 reveals more accurate result than 4.29 (Islam, 2006) and so, in this study equation 4.28 is used for calculating availability of the test fuel.

CHAPTER 5

Experimental Setup and Test Procedure

Total experimental works of this study is divided mainly into two groups:

- Testing of the engine performance related properties of fuel
- Performance test of engine

5.1 Fuel Testing

Testing is required to evaluate the engine performance related fuel properties. Several properties of fuels are evaluated under this program. Before using all the fuels are filtered with cloth. The blend was prepared by mixing 80% sesame oil with 20% diesel fuel by volume. The mixture was left for 72 hrs and no separation was observed. Gum formation was observed after storing for 1 month. For evaluating all the properties ASTM standards are followed. The evaluated properties and corresponding ASTM standard no. is given in the following table.

Table 5.1: Fuel properties and determination method:

Fuel Properties	*ASTM No.*	*ASTM Description*
Kinematic Viscosity	ASTM D445	Kinematic Viscosity of Transparent and Opaque Liquids.
Heating Value	ASTM D 240	Heat of combustion of Liquid hydrocarbon fuels by bomb calorimeter
Flash point	ASTM D93	Flash Point by Pensky Martens Closed Cup Tester
Water and Sediments content	ASTM D96	Tests Methods for Water and Sediments in Crude Oil by Centrifuge Method
Carbon residue	ASTM D189	Conradson Carbon Residue of Petroleum Products
Measurement of Volatility	ASTM D86	Distillation of Petroleum Products at Atmospheric Pressure

5.2 Engine Performance Testing

5.2.1 Experimental Setup

The experimental setup consisted of engine test bed with diesel and sesame oil supply system, different metering and measuring devices along with the test engine. Sesame oil feed system consists with an electric heater placed in a heater box. The 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. Schematic diagram of the experimental setup is shown in the Fig. 5.1.

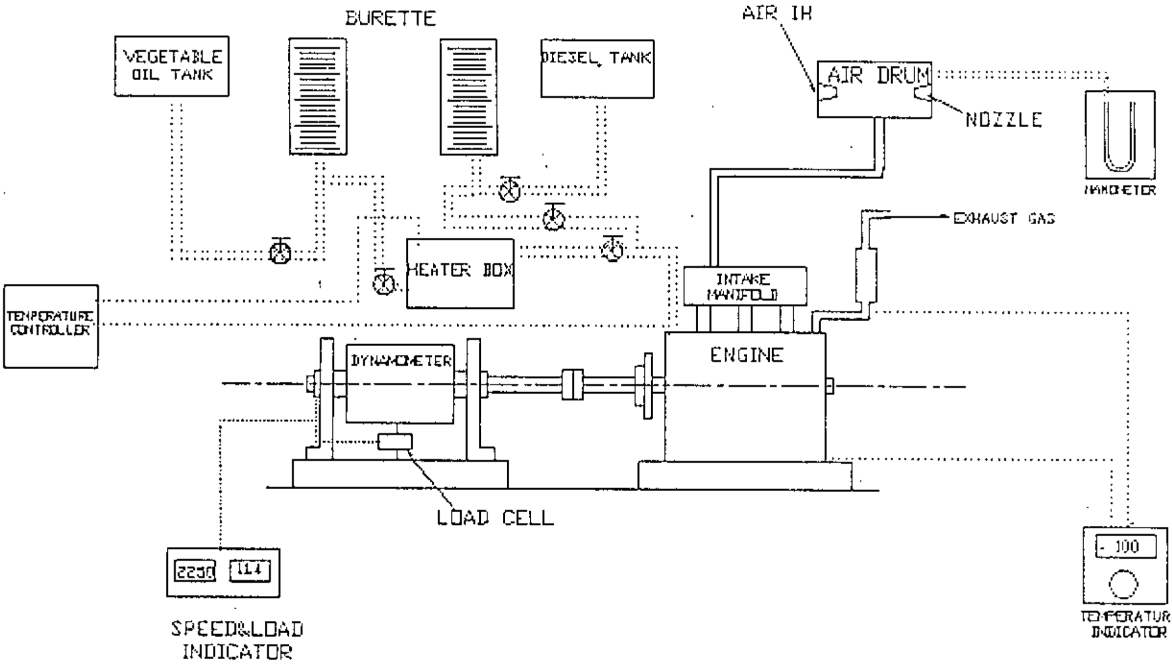

Fig. 5.1: Schematic Diagram of the experimental setup

Experiments were carried out using a 33.5 kW (45 hp) Motor Diesel (VM) Engine of model 1053 SU. It is air cooled high speed direct injection four stroke diesel engine. It has three cylinders each having a bore of 105 mm and a stroke length of 110 mm. Its rated output is 33.5 kW at a rated speed of 2250 rpm. The engine was started by a self starter-motor run by a 12 V battery. A water brake dynamometer was used to apply desired load on the engine. The dynamometer was connected with the engine by a universal joint. Technical details of the engine and the dynamometer are given in appendix A and B, respectively. In the present chapter, the experimental technique is given in brief. Details of the required equations and sample calculations are reported in appendix C.

Fuels were fed to the injector under gravity. A heating coil was dipped into the sesame oil container. In order to raise the temperature of the oil to a desired limit, the heater was controlled by a controller circuit, which cutoff or connects to the electric current by sensing the temperature of the oil before it enters the injector. For this purpose, the sensor was used at the fuel line just before the injector rail whose other end was connected to the controller circuit through a overload relay. The overload relay was very sensitive to voltage change whenever the sensor senses any rise in fuel temperature over the set temperature its voltage difference changes and the relay brake the circuit at the instant.

5.2.2 Engine Test Procedure

Initially the test engine was run by diesel fuel at three different speeds of 1750, 2000 and 2250 rpm and all the necessary parameters were recorded and different performance parameters were calculated to check the engine's credibility as a test engine. Then sequentially the performance tests were carried out using 100% pure sesame oil preheated at 60, 80 and 100^0C and 80% sesame oil blended with 20% diesel (by volume) preheated at 100^0C. Engine speed was maintained within ± 5 rpm and the temperature was maintained within ± 2^0C of the desired temperature.

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, was followed. Any other additional guidelines required were taken form the procedures used by Plint and Böswirth (1986).

Measurement of Engine Brake Power and Speed

A water brake type dynamometer, model no.- TFJ-250L, was used to apply desired loads and to measure engine brake power. It consisted of a load cell transducer to measure the reaction force acting on the dynamometer. A magnetic type tachometer was used to measure the engine speed. Two LED displays were provided to show the load in kg and speed in rpm

Measurement of Temperatures

Near the bottom of the silencer one hole was made for the provision of inserting the probe of a K-type thermocouple for measuring the exhaust gas temperature. Lubricant temperature was also measured using a K-type thermocouple probe inserted into the lubricant oil sump. All temperatures were measured using digital meter (OMEGA-K), connected to different K-types probes via a selector switch.

Measurement of Fuel Consumption Rate

Diesel and sesame oil supply systems were modified, so that fuels could be supplied from a graduated burette instead of the fuel tank when needed. Fuel consumption rate was recorded observing the time by a stop watch for every 100cc of fuel.

Measurement of Air Flow Rate

Air flow rate was measured by drawing air through two circular nozzles of 30 mm diameter (C_D = 0.98) with an air drum of standard size (in accordance with the plint and Böswirth) that is connected to the engine air inlet and pressure drop is measured by means of a manometer, using water as a manometric fluid. The mass flow rate of air is calculated by using the pressure drop in appropriate equation.

5.3 Uncertainty in Different Measurement

Considering instrument error, precision error and error associated with the resolution of the display unit, maximum error that might be associated with different measurements are estimated as below (calculations are shown in the Appendix F) :

Table 5.1: Lists of uncertainty in different measurement

Measurements	*:*	*Uncertainty in Measurement*
Temperature Measurement	:	± 2 ^{0}C
Load Measurement	:	± 0.2 Kg
Revolution Measurement	:	± 4 rpm
Fuel flow Measurement	:	± 1 ml
Time Measurement	:	± 1 sec

Uncertainties for the computed parameters are obtained from the measured parameters following the Kline and McClintock procedure (Holman, 1994). The uncertainties for the computed parameters are calculated with 95% probability as: brake power ± 1.96%, fuel flow ± 0.17%, bsfc 1.98% and bmep 2.02%.

Chapter 6

Results, Discussion and Conclusion

6.1 Introduction

In this chapter results obtained from fuel testing are presented which is followed by the engine performance results.

6.2 Results of Fuel Testing

In Fig. 6.1, variations of fuel density with temperature are shown. It is found that sesame oil density is 10% higher than diesel at 20°C. When sesame oil is blended with diesel and heated up to 100°C, its density become less than that of diesel fuel at 20°C.

In Fig. 6.2, viscosities of oils at different temperature are presented. It is seen that the viscosity of sesame oil is nearly 10 times higher than that of diesel fuel at 20°C but when sesame oil is heated up then its viscosity reduces significantly and at 100°C, viscosity of the both sesame oil and its blend become less than that of diesel fuel at 20°C.

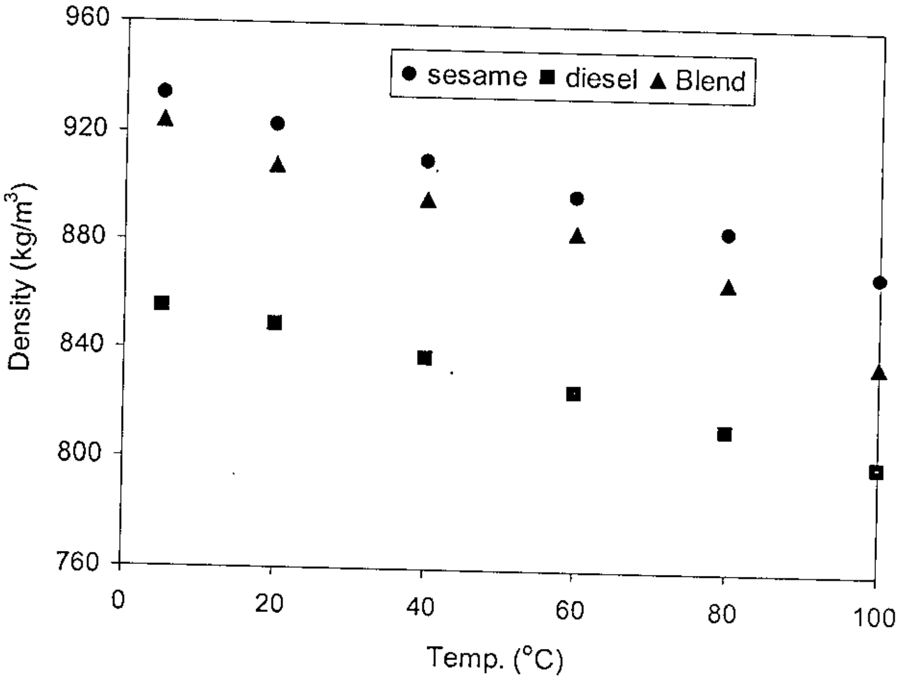

Fig. 6.1: Variations of densities of oil with temperature

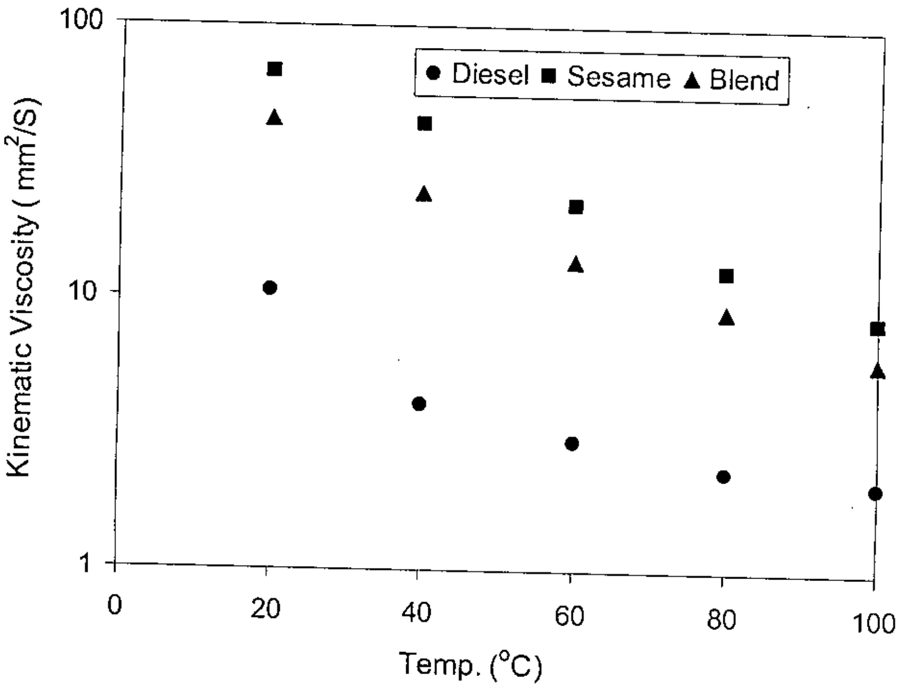

Fig. 6.2: Effect of temperatures on the viscosities of fuels

In Fig. 6.3, volatility of diesel and sesame oil is presented. It is seen that the volatility of sesame oil is much lower than that of diesel fuel but the initial boiling of sesame oil starts earlier than diesel fuel. T_{10}, T_{50} and T_{90} points on the curves are of special interest. Lower T_{10} temperature signify easy starting , A lower T_{50} point allow the engine to warm up and gain power quickly and T_{90} temperature associated with crankcase dilution and fuel economy, if it is too high, the larger fuel molecules will condense on the cylinder liners and passed down into the lubricating oil instead of burning. With sesame oil T_{90} point is not achievable as chemical decomposition of the fatty acids starts before reaching that point.

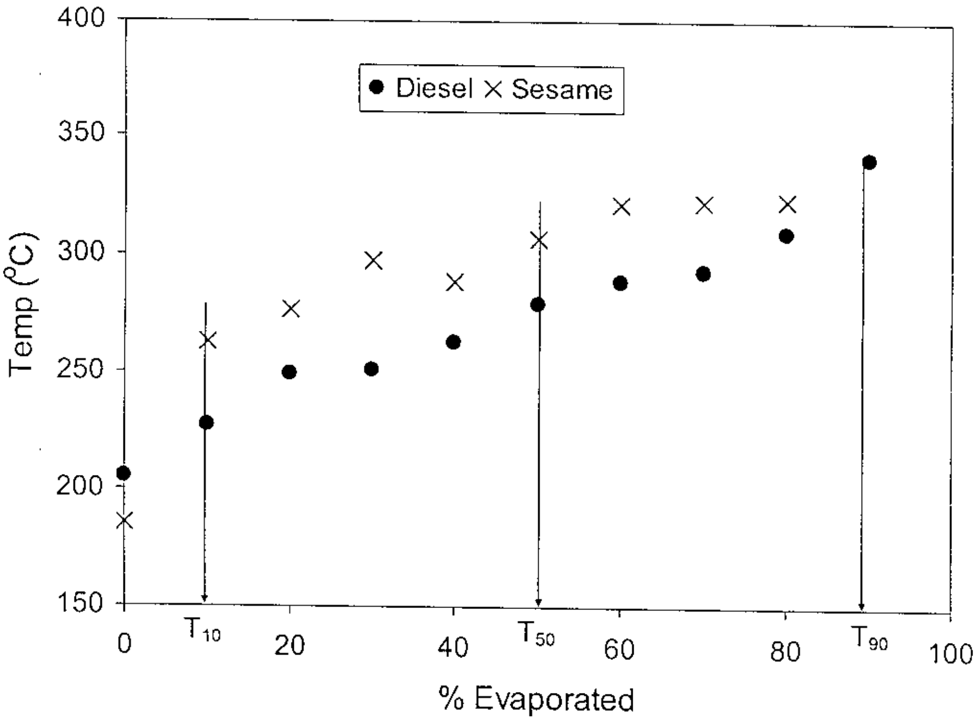

Fig. 6.3: Distillation curves

Some other important properties of fuel are presented in Table 6.1. It is seen that the sesame oil flash point is much higher than that of diesel fuel, so from safety point of view sesame oil is better than diesel. Calorific value of sesame oil is 10% less than diesel fuel. When same amount of vegetable oil is used in a diesel engine some energy loss is observed. Carbon residue of sesame oil is much higher than that of diesel fuel and so, when vegetable oil is used in an engine more deposits will be formed inside the engine and hence it needs cleaning more frequently. Again, higher carbon residue signifies higher ash content which is abrasive in nature results in more wear.

Table 6.1: Physical and chemical properties of diesel and sesame oil

Properties	*Unit*	*Diesel*	*Sesame*
Flash Point	°C	58	230
Calorific Value (LHV)	MJ/Kg	42.94	37.43
Carbon Residue	%	0.02%	0.35%
Water and Sediments	%	Trace	

6.3 Engine Performance Results

Potentiality of using sesame oil as an alternative to diesel fuel is judged in the present study. For this purpose, first the experiments are carried out at three different speeds (1750, 2000, and 2250) under variable loading condition using diesel fuel to evaluate the engine's credibility to employ as a test engine and also to raise base performance parameters for later comparisons.

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 1750, 2000 and 2250 rpm at different loading conditions is shown in the Fig.6.4. It is seen that for the same output power air flow rate increases with engine speed due to the fact that swept volume per unit time is higher for higher engine speed. However, with the increase of engine power mass flow rate of air decrease slightly due to increase of intake manifold temperature which reduces intake air density to reduce the induction air mass flow rate.

Air induction of engine depends on the ratio of intake and exhaust pressure, residual gas volume, intake air velocity, engine speed, size and shape of the passages. Residual gas fraction effect is less in CI engine due to higher compression pressure. As the gas velocity increase pressure drop also increase and breathing reduces. Again, at higher engine speed intake system or a part of the intake system is chocked and once this occurs further increase in speed does not increase the flow rate significantly and so, volumetric efficiency, η_v decrease. Again at higher speed, air flow rate per cycle decreases due to short cycle time. Echo of the above discussion is shown in Fig. 6.5 where variations of volumetric efficiency, η_v for the conditions mentioned in Fig. 6.4

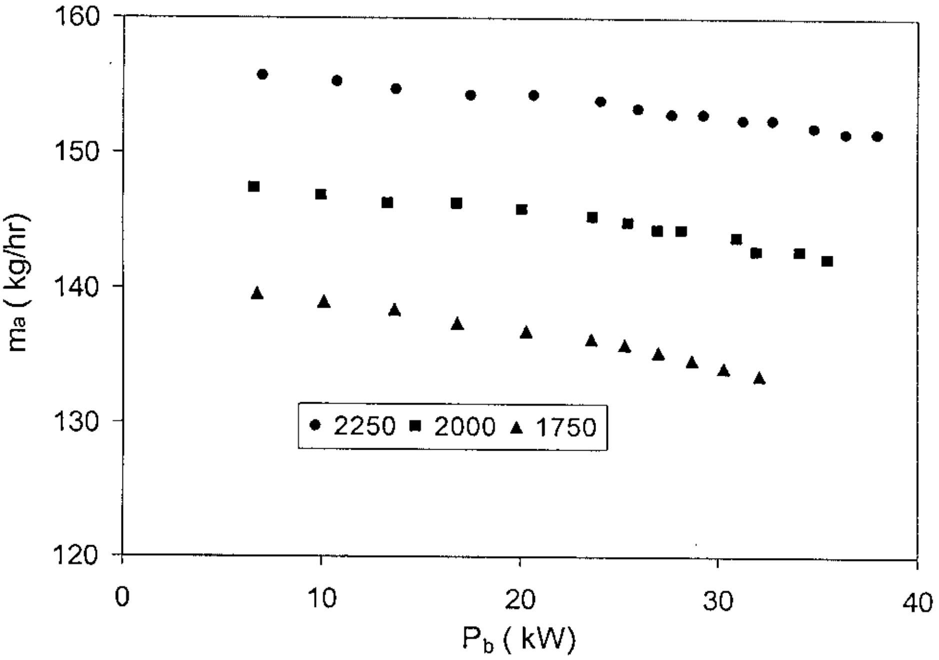

Fig. 6.4: Air flow rate as a function of engine brake power run by diesel fuel

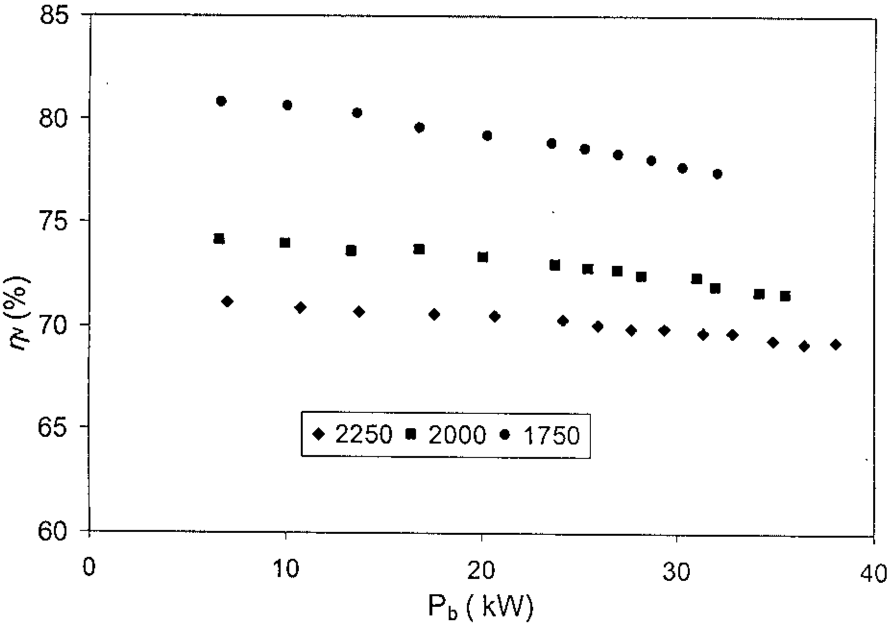

Fig. 6.5: Variations of volumetric efficiency with engine brake power run by diesel fuel

Fuel consumption rates are presented in Fig. 6.6 at the above mentioned conditions. In diesel engine higher power is achieved by increasing the fuel flow rate and it is also observed from the graph. Again fuel consumption rate increases with speed. With increase of speed the frictional loss increases and to overcome that more power required inside the cylinder and so, fuel flow rate increases with speed. At part load more fuel is consumed for higher engine speed but near the rated power, the engine consumes almost same amount of fuel to generate the same amount of power for all three speeds. This behavior is directly related to the friction. Engine friction is either related to the engine speed or related to peak engine cylinder pressure. At low loading condition the effect of engine cylinder pressure is insignificant and frictional losses is dictated by the energy consumed by the shafts, valves and pumps and these losses increases with the speed of the engine. So, at low loading conditions fuel consumption is higher at higher speed. However at higher load, the effect of peak cylinder pressure becomes more significant. For the same power output the slower speed generates more pressure and frictional loss increases and so, at higher loading fuel consumption rates become all most same for all the three speeds.

Variation of air fuel ratio is plotted against engine brake power for the above mentioned conditions in Fig. 6.7. It is observed that only lean combustion is employed in diesel engine because of the very nature of the diesel engine combustion system where fuel is directly injected into the cylinder and heterogeneous mixture is formed and in case of stoichiometric mixture some portion of the charge will starve from O_2 to complete the combustion. Stoichiometric combustion of diesel fuel requires 14.30 A/F ratio (Ferguson, 2000) but from the graph it is seen that the air fuel ratio at low load is nearly 55. With the increase of load A/F ratio reduces and up to 45% of the rated power it is steeply reduces. The falling trend continues over the entire range but gently. In CI engine, output power is increased by increasing fuel flow rate and at the same time the air flow rate decreases with the increase of load which is already mentioned. A 'cross over' point is observed almost at the middle of P_b.

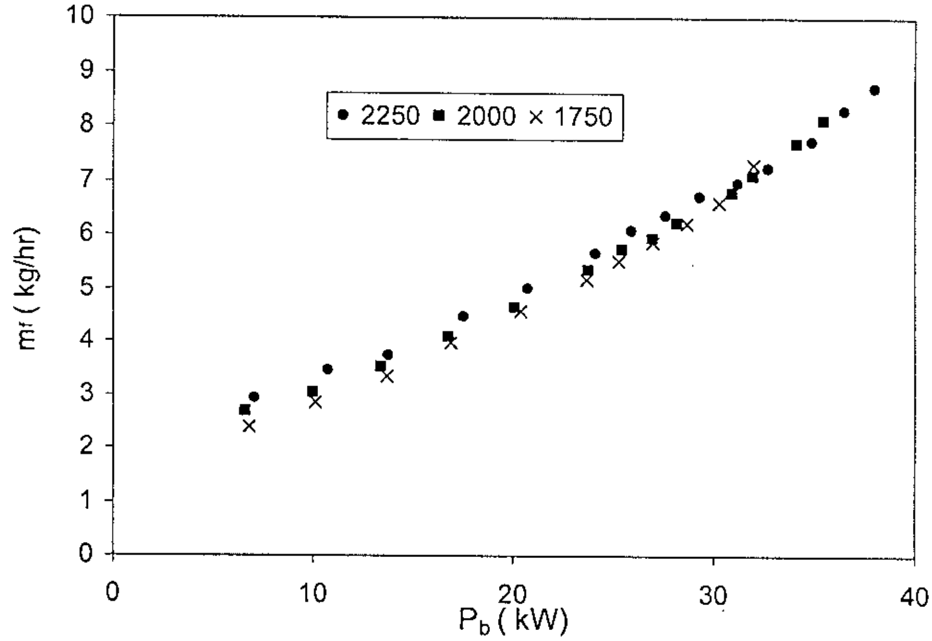

Fig. 6.6: Fuel flow rate as a function of engine brake power run by diesel fuel

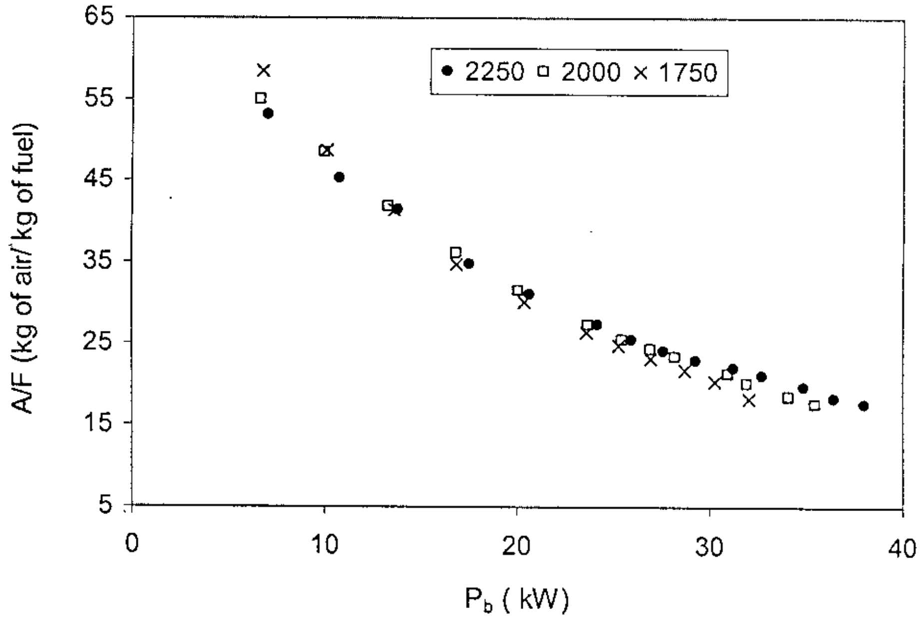

Fig. 6.7: Variations of Air Fuel ratio as a function of engine brake power run by diesel fuel.

Variations of brake specific fuel consumption, *bsfc* with brake power, P_b at different speed is shown in Fig. 6.8. Different engines can be compared with *bsfc*. It is seen that *bsfc* is high at low loads and its value decrease with the increase of brake power until the rated output of the engine is reached beyond that value again it increases. At lower load A/F ratio is much leaner, at that time residual gas temperature, wall temperature is lower and ignition delay period is higher and so, fuel utilization efficiency is less. So, *bsfc* value is higher at lower load again at higher load beyond the rated point all the fuel is not burned properly due to scarcity of air. Energy input in the fuel is lost in the form of incomplete combustion product and results in higher values of *bsfc*. It is also observed that, for the same output power, at part load condition *bsfc* is higher for the higher rpm but at rated or higher than rated rpm *bsfc* is independent of speed. The main reason here is the friction and it is already mentioned earlier.

The *bsfc* is a measure of overall engine efficiency, η_b and these quantities are inversely related. So, that lower the values of *bsfc* higher the overall efficiency of the engine. However for different fuels with different heating values, the values of *bsfc*'s are misleading and hence brake thermal efficiency is employed when the engines are fueled with different types of fuels (Haq , 1995). In Fig. 6.9 brake thermal efficiency, η_b is plotted against the same above mentioned conditions.

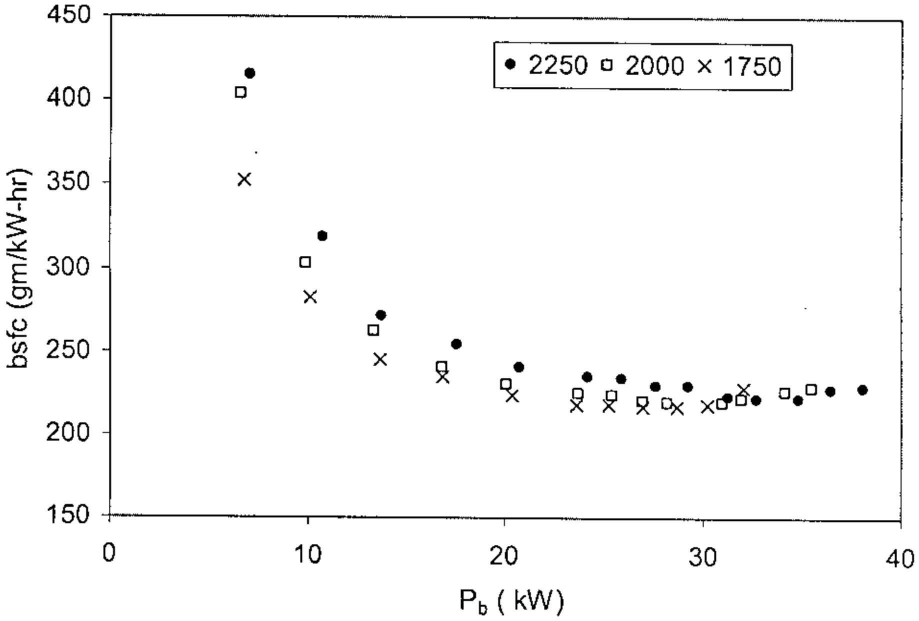

Fig. 6.8: Variations of brake specific fuel consumption with brake power run by diesel fuel

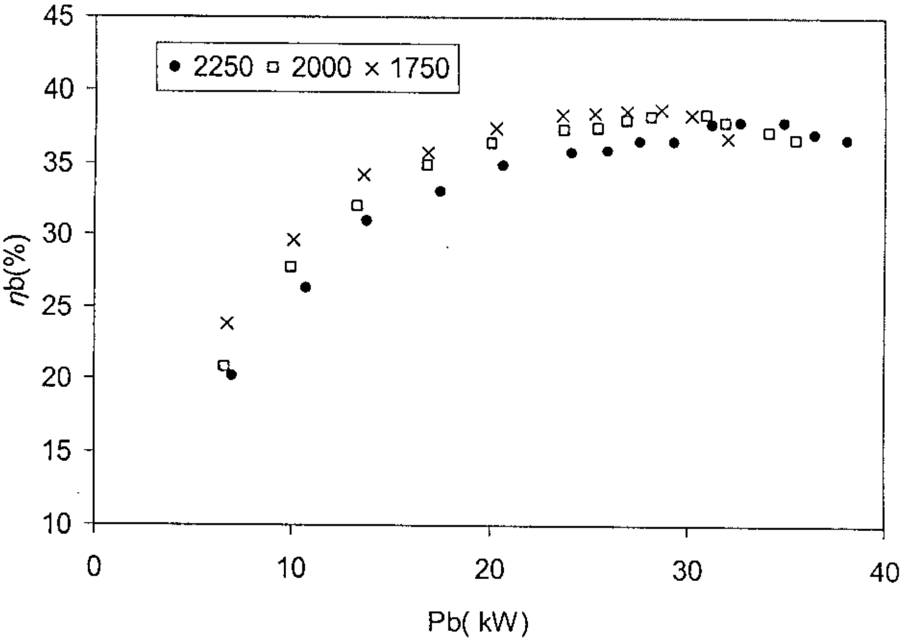

Fig. 6.9: Variations of brake thermal efficiency with engine brake power run by diesel fuel

Variations of exhaust gas temperature, T_{exht} is shown in Fig. 6.10 for the above mentioned conditions. The exhaust temperatures are found to increase gradually with increase of brake power. With the increase of brake power more fuel is burned inside the cylinder and more heat is generated and so, exhaust temperature increase. It is also observed that, at part load for the same output power, at lower speed exhaust temperature is lower and reverse trend is observed when the load advances to rated or over load.

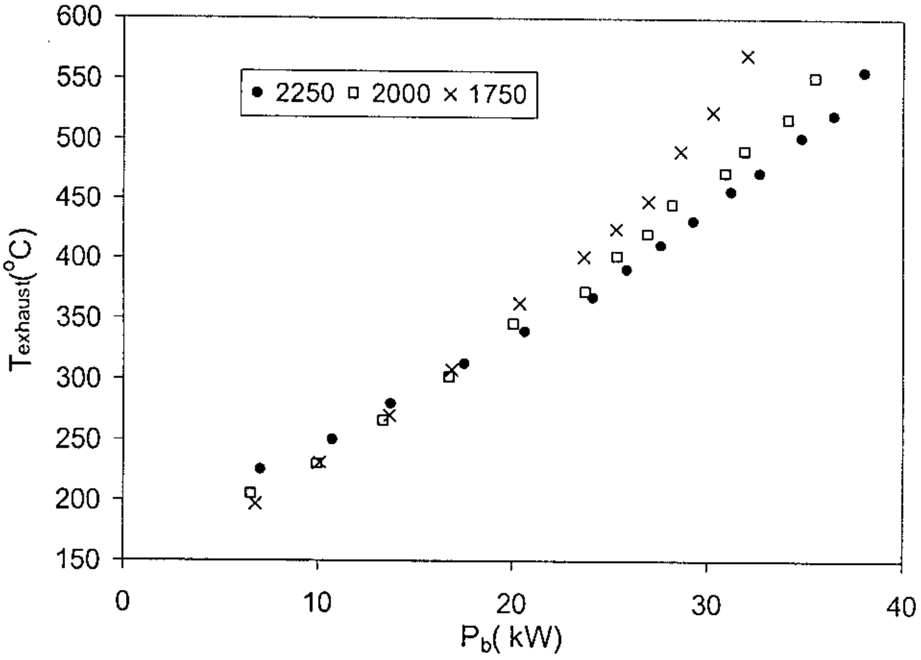

Fig. 6.10: Exhaust gas temperature as a function of brake output power

The brake thermal efficiency, η_b of the test engine is plotted against brake mean effective pressure (*bmep*) in Fig. 6.11. Engine performance parameters presented as a function of engine brake power is unique trend only for a particular engine at a particular speed. The performance curves for engines of different sizes or for same engine at different speeds derivates far from each other. So, a dimensionless parameter is required to characterize the engine processes in a precise manner. Unfortunately, there do not seem to be any appropriate dimensional group that can serve the purpose duly. However, *bmep* is useful in this regard as it is roughly comparable even for different engines. The trend is same as it was in Fig. 6.11. The brake thermal efficiency, η_b lines for three different speeds have almost coincided on each other, which ascertain the acceptability of *bmep* as equivalent to non-dimensional parameter. So, all the next performance and availability terms are evaluated as a function of *bmep* .

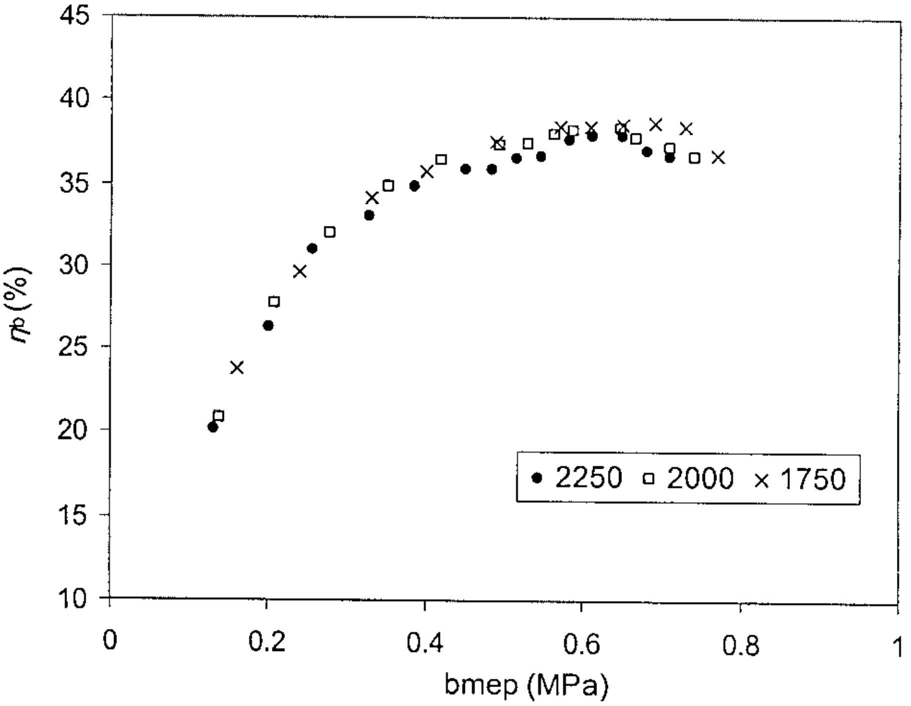

Fig. 6.11: Variation of engine brake thermal efficiency with brake mean effective pressure

The availability (or exergy) input to an internal combustion engine is contained in its fuel chemical availability. In CI engines, the input availability contained in fuel is converted into:

a) Useful brake output availability
b) Availability transferred to cooling medium
c) Availability transferred to exhaust gases
d) Availability destroyed in engines accessories turbocharger, cooling fan, etc.
e) Availability destroyed due to friction and radiation heat loss to surroundings.

In the present analysis, due to scarcity of thermo chemical data (specially for sesame oil) 'percent brake output availability', A_{shaft} is used in lieu of second law efficiency, η_{II} . The availability is destroyed or lost due to different irreversibility include combustion losses, friction losses, heat loss to lubricating oil, power consumed by auxiliary equipment (axial blower in the present test engine), radiation losses, fluid flow losses, etc. Availability transfer to cooling medium (air in this case) has been included into the above category because of the lack of facilities to compute the heat loss in the air-cooling system by axial blower. Availability destruction due to all these sources are combined expressed by $A_{uncounted}$. This is justified by the fact that availability transfer to cooling medium in an air-cooled engine is only a fraction (less than 2.5%) of the availability input to the engine (Al-Najem and Diab, 1991). Availability destroyed in the exhaust gas is evaluated separately.

In Fig. 6.12 the portion of the input availability converted into brake output power which is denoted by A_{shaft}.is shown. It can be noted from this figure that at higher *bmep* availability output at shaft becomes quite constant with a slight declining trend after rated output, same trend as is observed in Fig. 6.11 and the reason is also same.

The availability input is expensed in different processes of the engine is shown in Fig. 6.13. It can be noticed that availability transfer to the exhaust gases (denoted by A_{eg}) increases with increasing *bmep*, but is quite a small portion of the availability input, the maximum level reaching approximately 12% of A_{in}. In contrast, the availability destruction in friction, cooling, combustion etc. (denoted by $A_{uncounted}$) shows a declining trend. That is, with increase in *bmep* as A_{eg} goes up $A_{uncounted}$ continues to go down although its extent is far higher. However, this opposite trend of these lines facilities to confirm an 'optimum operating point' from this graph. From this figure it is noticed that the optimum operating point is around 0.45 MPa *bmep*.

The comparison of η_b with A_{shaft} is shown in Fig. 6.14 to obtain a precise contrast of the two, the curves are plotted at the same plane for the rated speed 2250 rpm. Observation reveals that A_{shaft} is somewhat less than η_b throughout the entire range. Both the graphs urge the fact that the capability of an engine to utilize the available energy successfully is rather less than that articulated in brake thermal efficiencies. The major reason behind this is that the fuel chemical availability, A_{in} is about 3.35 to 7.25% as higher (depending on the chemical formula of the fuel) than the heat input Q_{in} calculated from the lower heating value and this available energy can not be interpreted into shaft work due to inherent irreversiblities

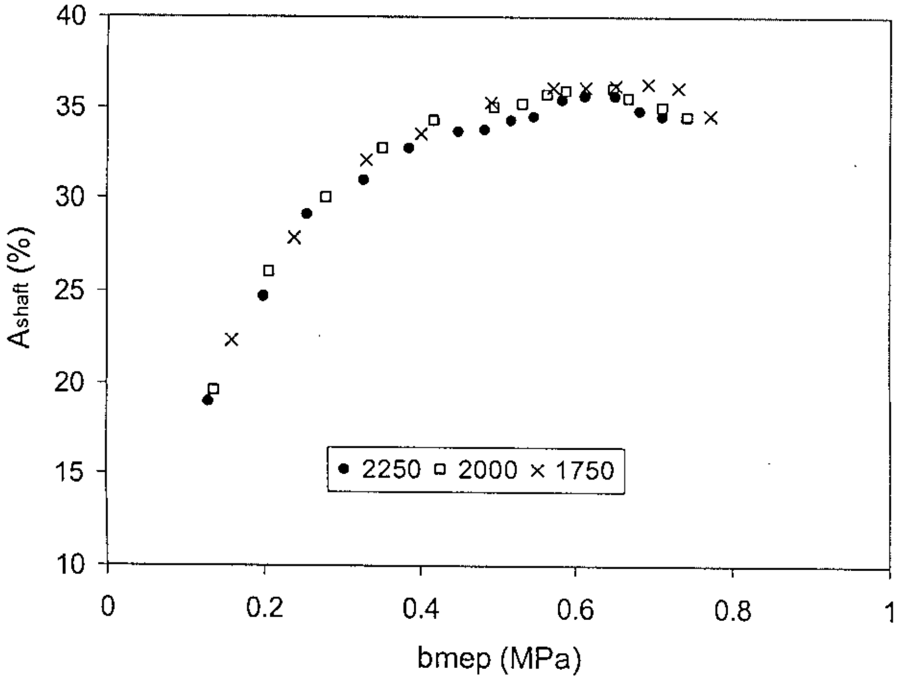

Fig. 6.12: Variations of percent availability output at shaft with engine brake mean effective pressure run by diesel fuel

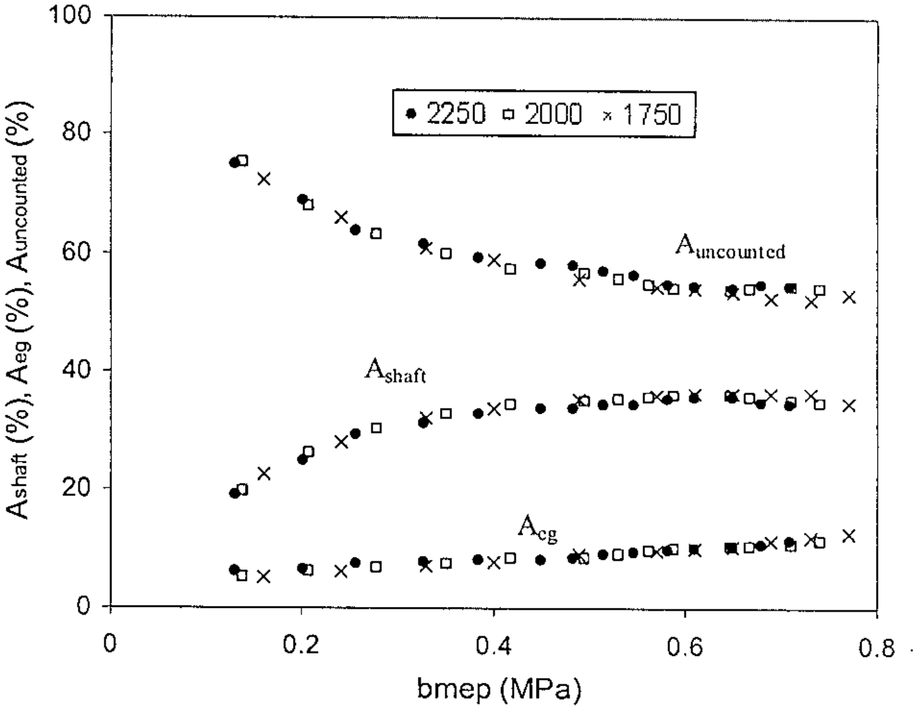

Fig. 6.13: Level of different types of availabilities associated with engine operation as a function of brake mean effective pressure run by diesel fuel.

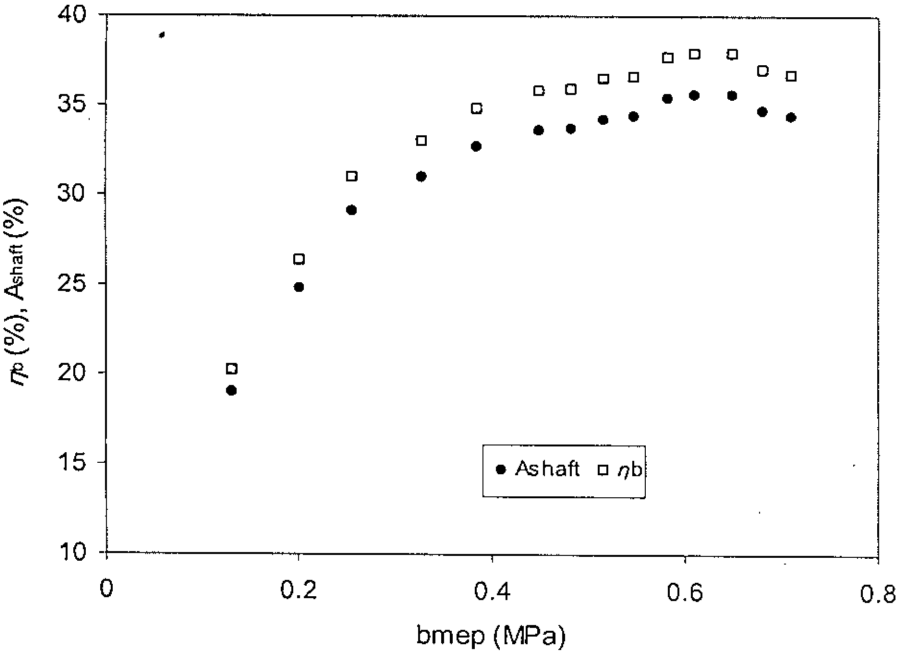

Fig. 6.14: Comparison of percent availability output at shaft with corresponding brake thermal efficiency as a function of brake mean effective pressure run by diesel fuel at 2250 rpm.

In Fig. 6.15 variations of brake specific fuel consumption is plotted as a function of *bmep* for three different preheat temperature 60, 80 and 100 °C of pure sesame oil and 100 °C preheat temperature for 80% sesame and 20% diesel blend is compared with that of diesel fuel. From this figure, it is evident that, specific fuel consumption is higher for all the vegetable oils. Vegetable oil's heating value is much lower than that of diesel fuel. Again high viscosity and poor volatility of the vegetable oil results in poor atomization and mixture formation and increases the fuel consumption to maintain the power. Again with the increase of preheat temperature viscosity of oil greatly reduced to improve fuel injection, atomization and mixing quality of the charge and so, fuel consumption rate decrease. But at high load temperature effect become less significant. At higher load, fuel flow rate and velocity is dominated by the combustion temperature and hence fuels arrive at the point of injection of any preheated temperature with about same temperature, providing similar injection conditions. Again blend of 20% diesel and 80% sesame preheated at 100°C does not increase the efficiency over 100°C preheated straight sesame oil. It may be due to the fact at that temperature extra fine spray formation occurs which reduce penetration and mixing with air that offsets the decrease of consumption of fuel.

In Fig. 6.16 variations of brake thermal efficiency is shown for the above mentioned conditions. Here it is seen that the brake thermal efficiency of diesel fuel is lower than sesame oil and blend preheated at 100°C. High viscosity lubricating oil act as a sealant between the piston rings and cylinder wall. High viscosity sesame oil forms coarse atomization, which sometime wash away the lubricating film and again vegetable oil creates more heating problem which reduce the lubricating oil viscosity to reduce the frictional loss. Vegetable oil's ignition delay period is longer than diesel and so, pressure rise inside the cylinder is lower and so, frictional loss due to pressure is also less. Brake thermal efficiency as a function of engine brake power is plotted in Fig. 6.17. The trend of the curve is same but from the figure it is clear that some power losses is experienced for the fuels other than diesel.

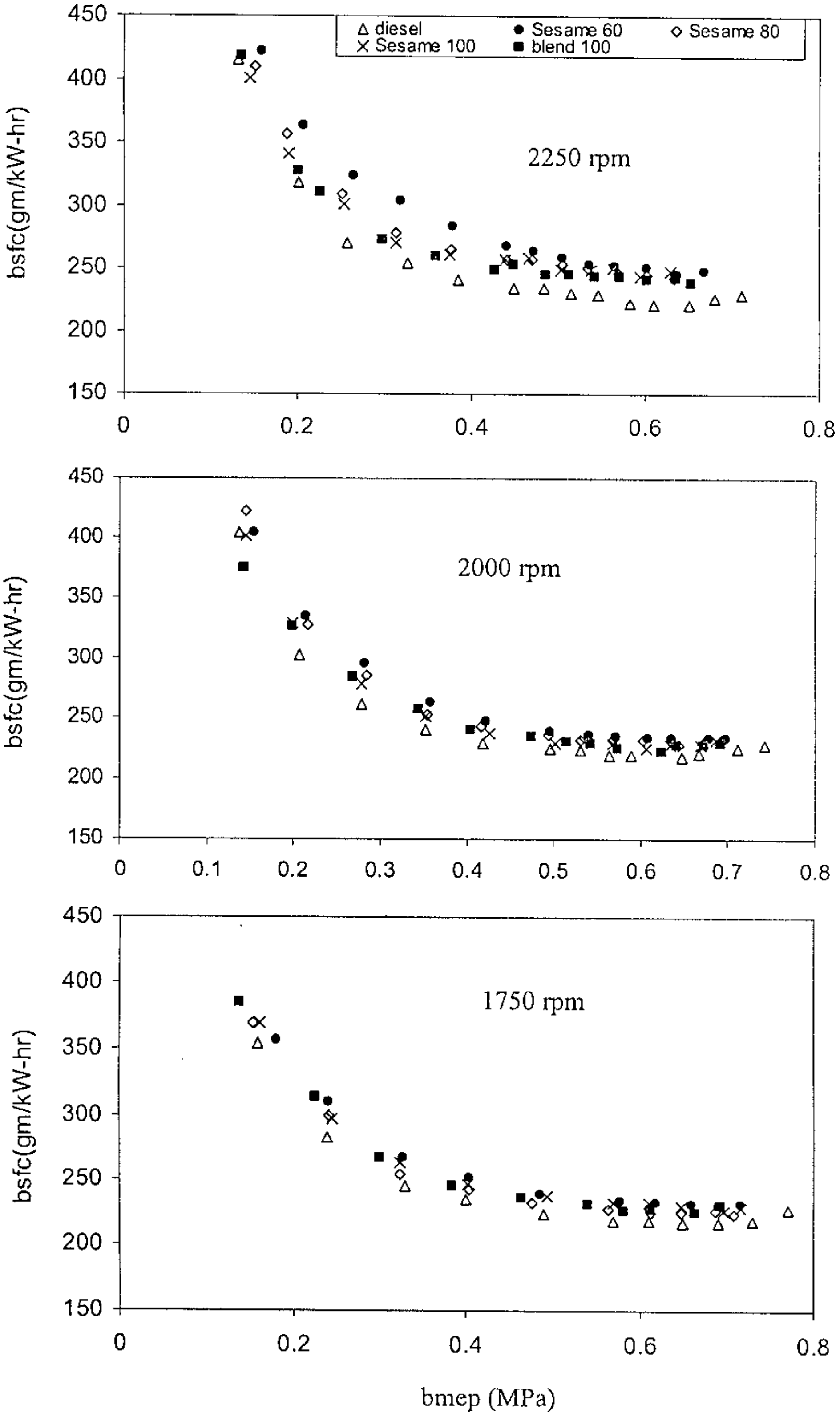

Fig. 6.15: Comparison of bsfc of different fuels at particular speed as a function of brake mean effective pressure

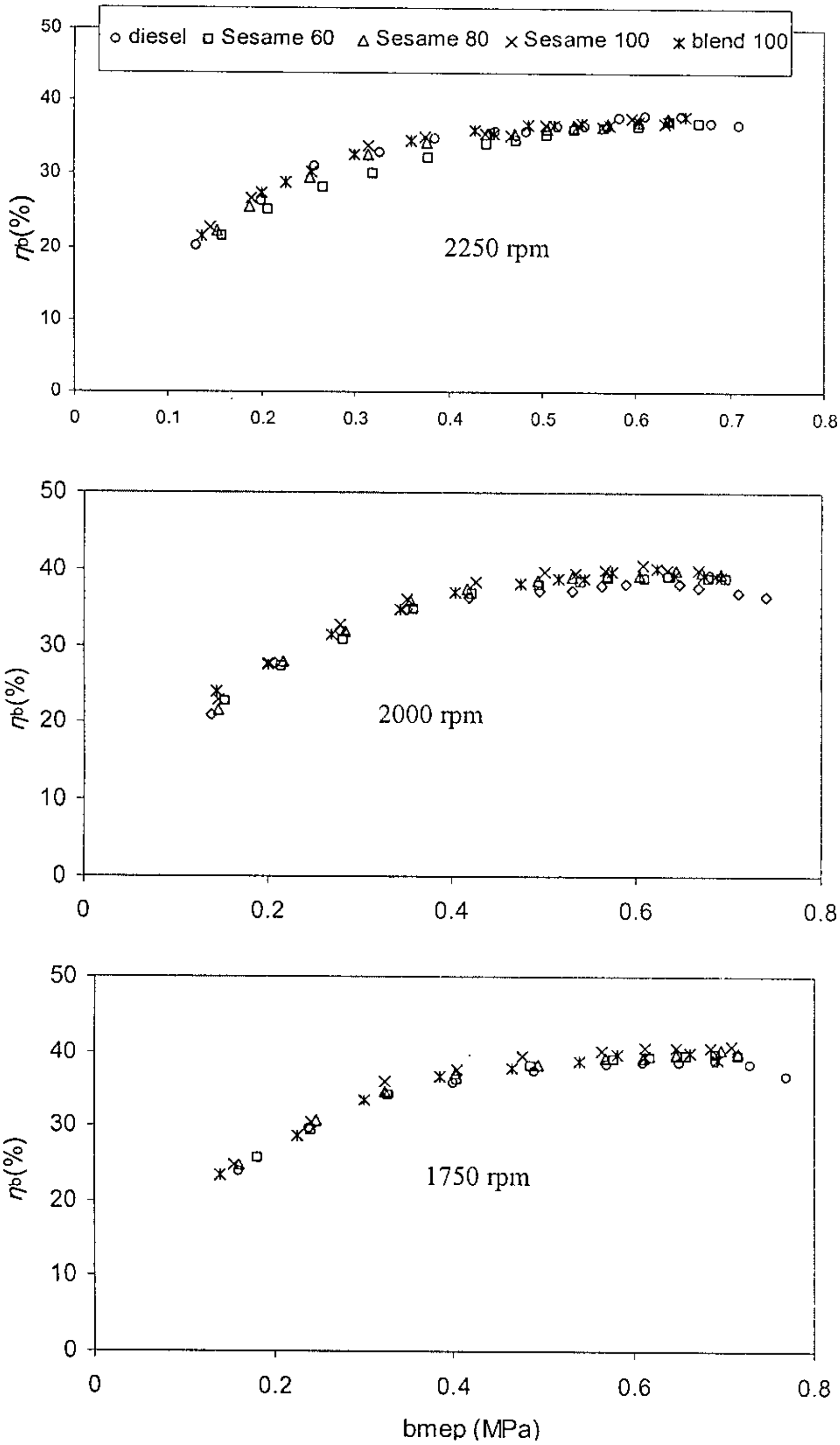

Fig. 6.16: Comparison of the brake thermal efficiency of different fuels at particular speed as a function of brake mean effective pressure

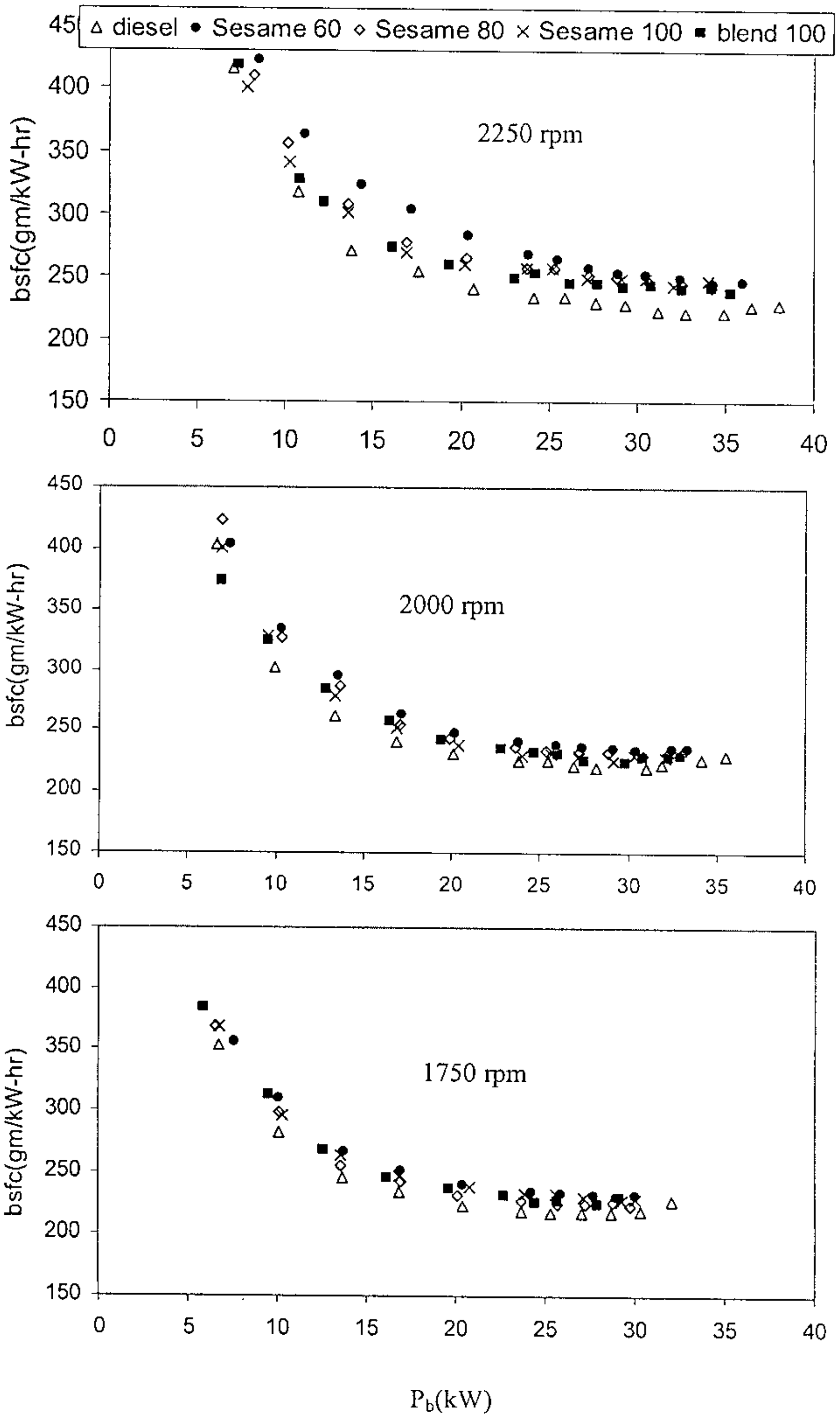

Fig. 6.17: Comparison of the bsfc of different fuels at particular speed as a function of brake power.

Variations of brake thermal efficiency for different engine speed for different fuels are shown in Fig. 6.18. For every case, at lower rpm highest efficiency is observed. Ignition delay period is much higher for vegetable oil and so, at higher speed fuel gets less time for heat realising or heat realsing occurs late of the expansion stroke and so, effective use of heat becomes less. Variations of percent availability output at shaft, A_{shaft} as a function of *bmep* for three different preheat temperatures of pure sesame oil and one preheat temperatures for the sesame and diesel blend is shown in Fig. 6.19 to Fig. 6.22. The trend is same as in figure 6.17 having the similar explanations behind.

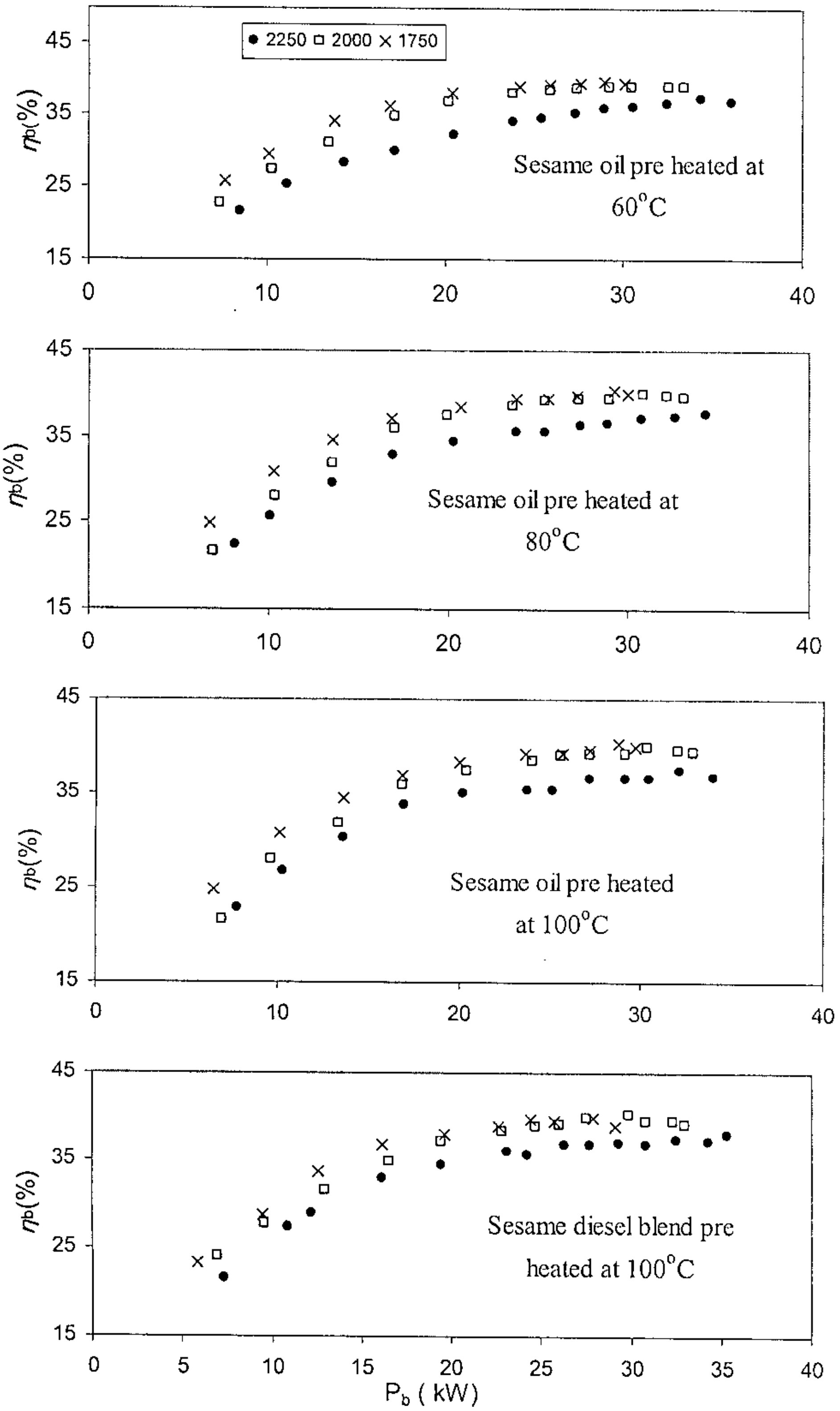

Fig. 6.18: Variations of brake thermal efficiency for different speeds at varying loads for a particular fuel.

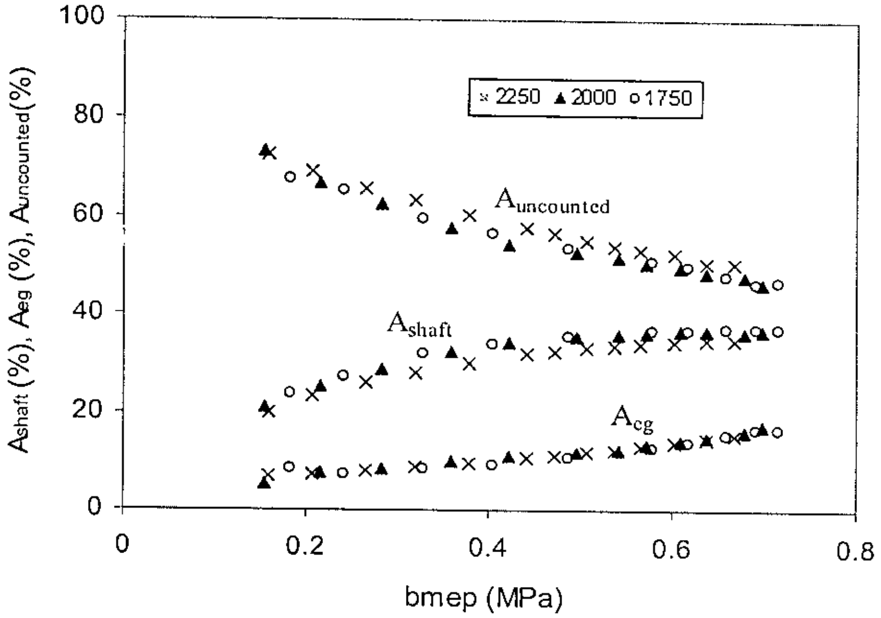

Fig. 6.19: Variations of percent available at shaft and destruction of availability as a function of brake mean effective pressure running by 60^0C preheated sesame oil.

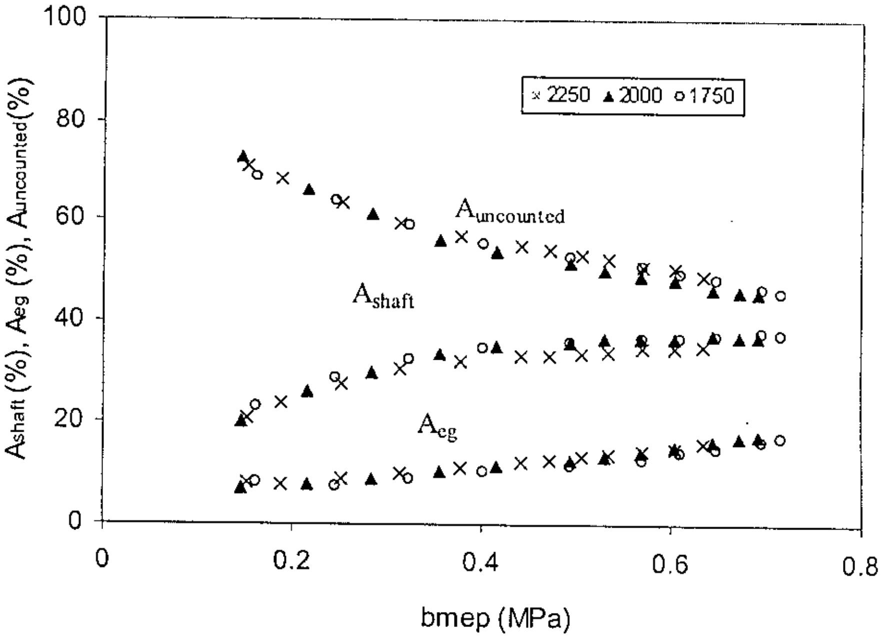

Fig. 6.20: Variations of percent available at shaft and destruction of availability as a function of brake mean effective pressure running by 80^0C preheated sesame oil.

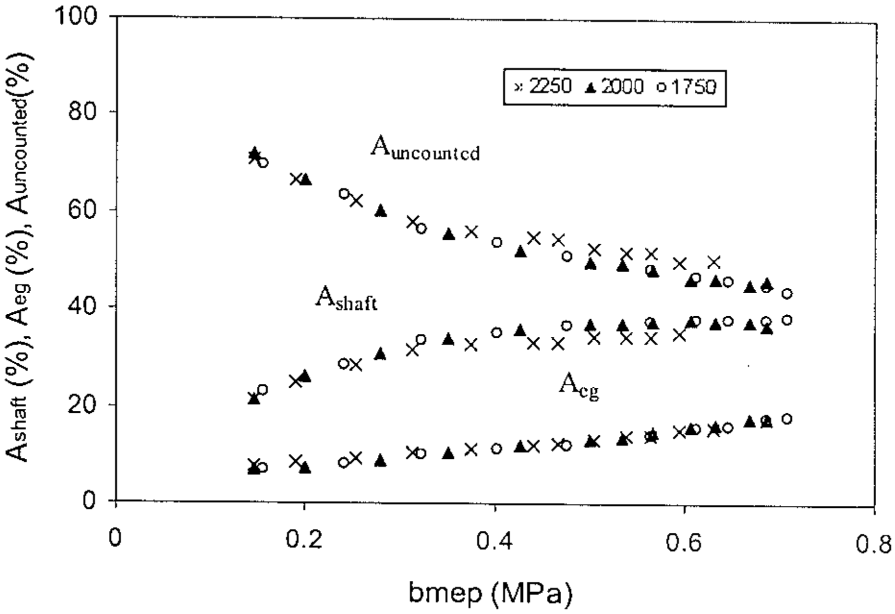

Fig. 6.21: Variations of percent available at shaft and destruction of availability as a function of brake mean effective pressure running by 100^0C preheated sesame oil.

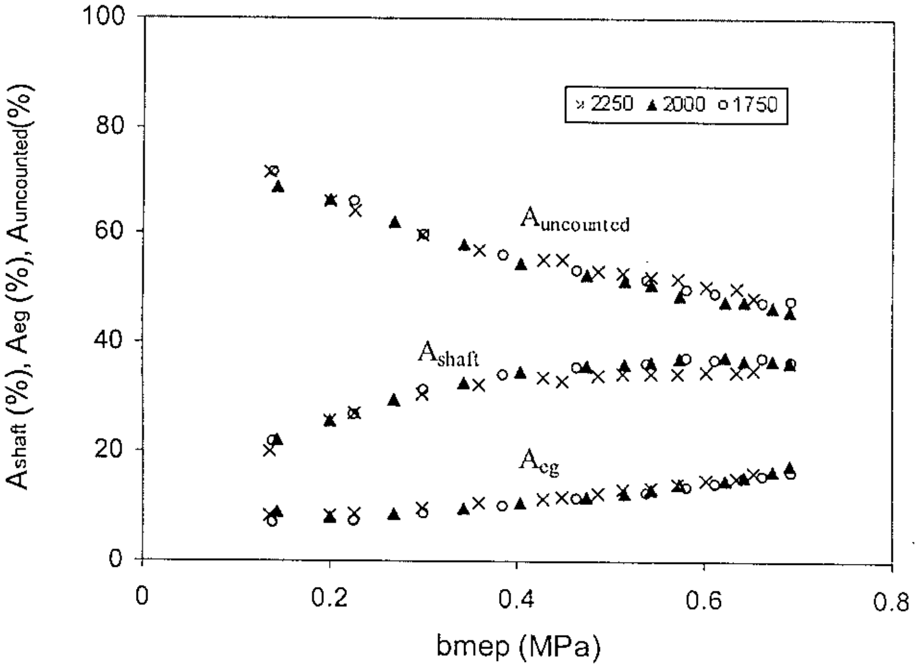

Fig. 6.22: Variations of percent available at shaft and destruction of availability as a function of brake mean effective pressure running by 100^0C preheated sesame oil and diesel blend.

Comparison of the percent of input availability available at the output shaft, percent of availability destroyed due to exhaust gas and percent of availability destroyed in friction, cooling, etc. between diesel fuel and other alternative fuels are shown in Fig. 6.23 to Fig. 6.25. With the increase of preheat temperature A_{shaft} increases.

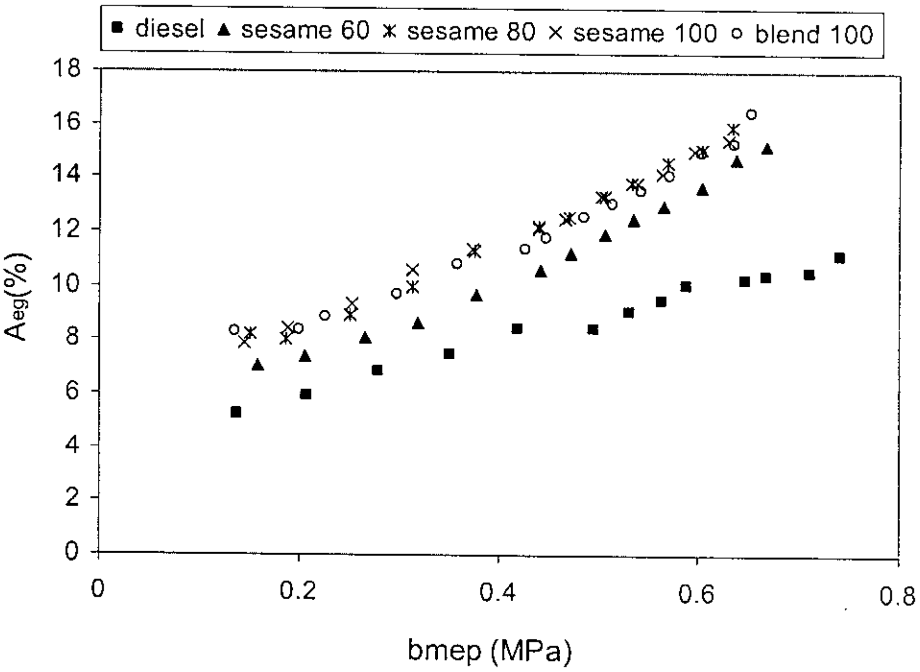

Fig. 6.23: Comparison of percent availability destroyed in uncounted factor as a function of brake mean effective pressure for 2250 rpm

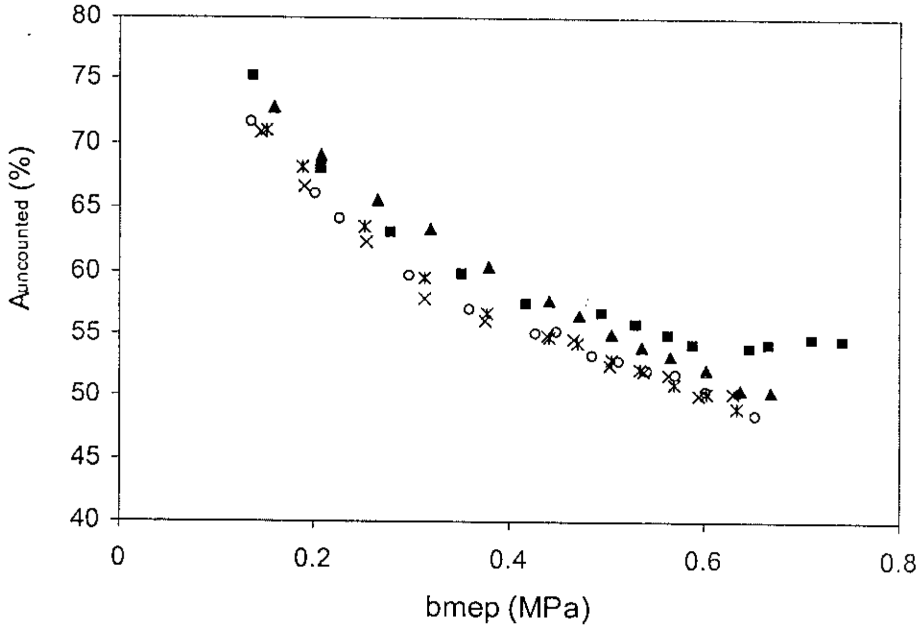

Fig. 6.24: Comparison of percent availability destroyed with exhaust gas as a function of brake mean effective pressure for 2250 rpm

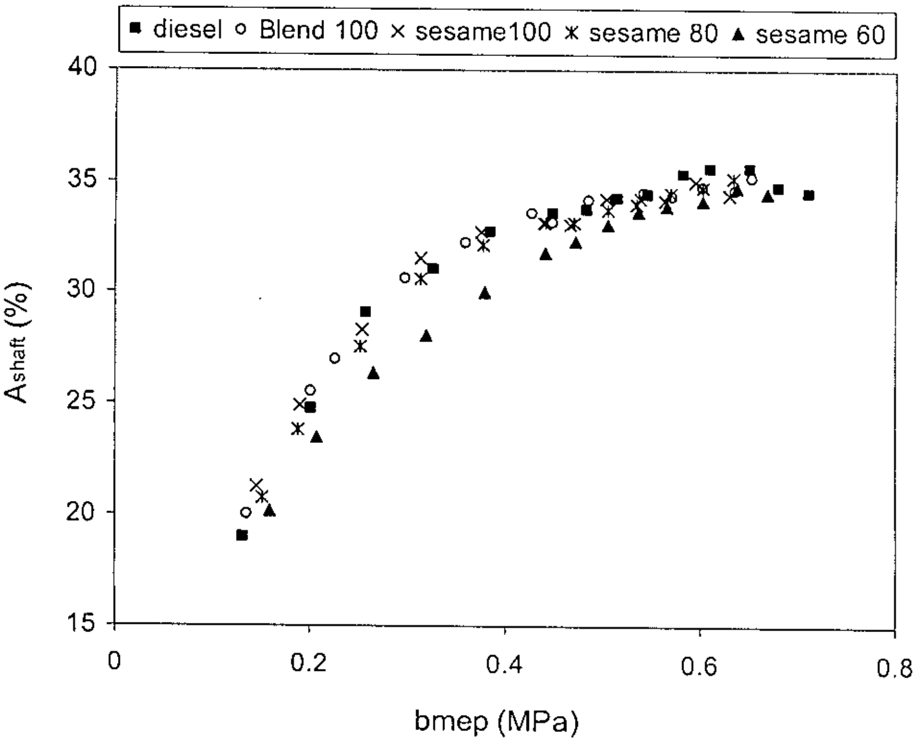

Fig.6.25: Comparison of percent availability output at shaft as a function of brake mean effective pressure for 2250 rpm.

6.4 Cost Analysis

For economic analysis of any system involves new equipment costs, fuel costs, operating and maintenance costs.

For introducing preheated pure sesame oil or preheated sesame diesel blend, does not require much of initial investment. Only need new fuel tank and new piping system. Heating energy can be used from the exhaust gas and for that heat recovery arrangement will be required but this cost will not be a very significant one, specially for a stationary engine.

Fuel cost is the main deciding factor for using sesame oil. Sesame oil costs more than twice of diesel fuel and refined oil costs more. Oil is sold on volume basis. Sesame oil is much denser than diesel, so, the better way to compare diesel and sesame is on the basis of their energy content per liter.

Using the equation 4.2, fuel chemical availability costs

Diesel	1.36 MJ/Taka
Sesame	0.58 MJ/Taka

Using this engine at the best operating condition for 1 kW-hr energy production costs

Diesel	10.5 Taka/ kW-hr
Sesame	19.7 Taka/ kW-hr

Again, vegetable oil increases engine wear, injector coking and contamination of lubricating oil. So, maintenance costs will increase for sesame oil.

So, from the economic view point sesame oil will not be a good or accepted alternative fuel to diesel at the present fuel costs. But if some other point is considered then the scenario can be different. Government is giving a huge amount of subsidy on diesel fuel and huge amount of foreign currency is spent for importing diesel. Transportation to the remote areas also costs much. Whereas sesame will not costs any foreign currency, it can be used, where it is produced, much people can be involved in production and processing work.

The main obstacles of widespread use of sesame oil lies in their cost relative to diesel. The cost of sesame oil can not be reduced by more cultivation as they are edible in nature and if its price becomes lower than other edible oil then its share will increase in the food market and consequently price will rise again.

6.5 Conclusions

The major conclusions that can be drawn from the present study are listed below:

- Without preheating crude sesame oil can not be used directly.
- Compared to diesel fuel operation a little amount of power losses happened with sesame oil operation.
- Fuel consumption becomes similar for heated and unheated oil operations at higher load but higher than diesel fuel.
- At lower rpm operation sesame oil exhibits better results for any pre heat temperature.
- The heating energy of sesame oil is almost same effectively as diesel is converted into mechanical energy when preheating temperature is 80°C or more than that.
- Preheating the blend of sesame oil with 20% diesel fuel exhibits almost same results as is for same temperature preheated net sesame.
- Availability destroyed with exhaust gas increases with load and it is about 15% of the input availability at rated condition. This may be used directly for preheating purpose.
- Major portion of the available energy is wasted due to friction, cooling, combustion etc., which is counted here as uncounted factors. So, for engine performance improvement much focus requires here.
- At present, Sesame oil will not be a good alternative to diesel fuel due to its higher price.

6.6 Recommendations for Further Works

- Long term tests are required to observe the oil's effect on engine.
- Emission characteristics need to investigate as vegetable oil's net contribution to the global warming gases is supposed to be less.
- Some non edible type of oil that can be planted in the waste land and can be sold at lower price (like jatropha curcas) need to investigate.
- Potentiality of the seeds having less commercial value like seeds of cotton, seeds of tomato need to investigate in respect of Bangladesh.
- Some correlations between fuel properties and engine performance parameters need to establish to reduce the experimental cost and time.

Appendix A

Engine Specification

Model No. :	:	1053 SU
Manufacturer and Country of Origin :	:	VM Motor Diesel Engine, Italy.
Cycle:	:	4 Stroke Direct Injection Diesel Engine
Number of Cylinders :	:	3
Bore :	:	105 mm
Stroke :	:	110 mm
Total Swept volume :	:	2856 L
Rated Speed :	:	2250 rpm
Speed Control :	:	Lever Control
Power rating at Rated rpm :	:	45 hp
Cooling :	:	Air Cooling by Axial Blower
Lubrication :	:	Forced feed with Oil cooling
Oil sump Capacity :	:	6.9 Kg
Oil filter :	:	with cartridge and its container
Air filter :	:	Oil bath filter
Injection Pressure (Injectors setting) :	:	230 kg/cm^2
Injection Order :	:	1-3-2
Engine weight with Electric start :	:	306 kg
Rotation of the crankshaft :	:	counter clockwise (from flywheel side)

Appendix B

Dynamometer Specifications

Type	:	Water brake Dynamometer
Model	:	TFJ 250L
Manufacturer	:	Tokyo Meter Corporation
Country of origin	:	Japan
Maximum brake power	:	250 hp
Power supply	:	AC 200/230V, 50Hz, 1-ϕ
Revolutions at Maximum brake power point	:	2500 to 5000 rpm
Maximum revolution	:	5500 rpm
Maximum braking torque	:	71.6 kg-m
Maximum braking water quantity	:	75 L
Weight	:	575 kg

Appendix C

Dynamometer Calibration

Before taking any reading the dynamometer was calibrated to make sure the validity of the instruments with dead weight following its manual procedure.

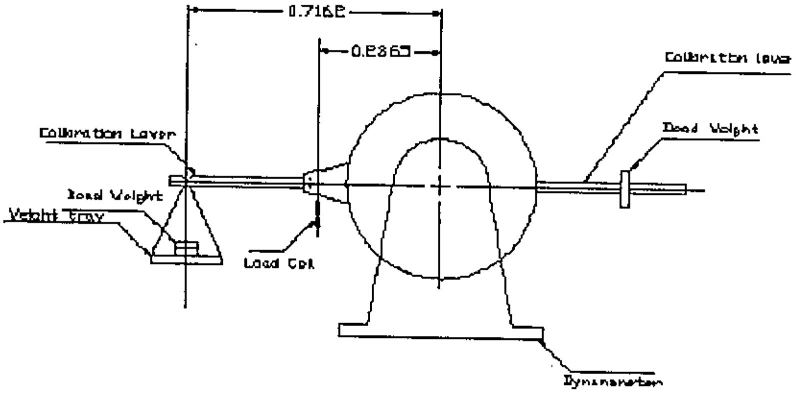

Fig. C.1: Setup for calibration of hydraulic brake dynamometer.

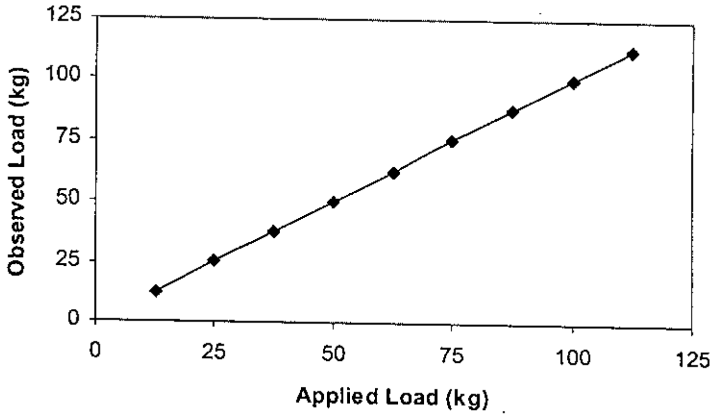

Fig. C.2: Observed load as a function of applied load for dynamometer

Clearly, the dynamometer was displaying consistently accurate result and so, dynamometer displayed load is acceptable without any correction factor.

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 Standard Reference Conditions

The following reference conditions are set by the BS 5514.

- Barometric Pressure, $P_r = 100$ kPa
- Air Temperature $T_r = 300$K ($27\,^0$C)
- Relative Humidity $\phi_r = 60\%$
- Charge in coolant temperature $T_{cr} = 300$K ($27\,^0$C)
- Mechanical efficiency of engine $\eta_m = 0.80$
  (according to the clause 10, note 4 of BS 5514)

D.2 Derating Factor Calculations

Here is given an example to show the derating factors calculation

The environmental condition at lab,

- Barometric Pressure $P_r = 98.9$ kPa
- Air Temperature $T_r = 305$K ($32\,^0$C)
- Relative Humidity $\phi_r = 80\%$

From Annex F, BS 5514:

By interpolation,

- Water vapor pressure $\phi_x P_{sx} = 3.8$

From Annex E: BS 5514 :

- The dry air pressure ratio, $\dfrac{P_x - a\phi_x P_{sx}}{P_r - a\phi_r P_{sr}} = 0.972$

From Annex D: BS 5514 :

The ratio of indicated power, $K = (R_1)^{y1} . (R_2)^{y2} . (R_3)^{y3}$

$$R_1 = \frac{P_x - a\phi_x P_{sx}}{P_r - a\phi_r P_{sr}} = 0.972$$

$$R_2 = \frac{T_r}{T_x}$$

$$R_3 = \frac{T_{cr}}{T_{cx}}$$

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

From table 1, m=1, n= 0.75, q=0.

$$(R_2)^{y2} = \left(\frac{T_r}{T_x}\right)^n = \left(\frac{300}{305}\right)^{0.75};$$

$$\therefore \left(\frac{T_r}{T_x}\right)^n = 0.988$$

- $K = 0.972 \times 0.988 = 0.9603$

From Annex – C

The fuel consumption adjustment factor, β=**1.0024**

From Annex – B

Power consumption adjustment factor, α =**0.9824**

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

$$P_B = \frac{p_B\,(Lab)}{\alpha}$$

$$bsfc = \frac{bsfc\,(lab)}{\beta}$$

Appendix E

Sample Calculation of Engine Performance

A set of calculation with actual data is presented here

E.1 Data Table

Lab Condition:

Date of Experiment : June 20, 2007
Fuel Used : Diesel
Atmospheric Pressure, P_x : 741.5 mm of Hg (98.93 KPa)
Dry bulb temperature, T_{db} : 32°C(305K)
Wet bulb temperature, T_{wb} : 29°C (302 K)
Relative Hmidity, ϕ_x : 80%

Derating Factors:

Power adjustment factor,α : 0.9824
Fuel consumption adjustment factor,β : 1.0024

Experimental Observations and Data:

Diameter of the flow nozzle, D_n : 30mm
Cross-sectional area of the flow nozzle, A_n : 7.0685×10^{-4} m^2
Co-efficient of discharge of the flow nozzle, C_D : 0.98
Specific gravity of diesel, σ : 0.846
Lower heating value of diesel, LHV : 42.94 MJ/Kg
Density of manometric fluid(water at 32 ^{0}C),ρ_w : 995.2 kg/m^3
Density of air at 32 ^{0}C, ρ_a : 1.113 Kg/m^3
:

Load on the Dynamometer, W 50.5 kg
Speed of the Dynamometer, N : 2255
Volume of Diesel collected at time t, V : 100 ml
Time of collection V volume of diesel ,t : 39 S
Manometric deflection, h_m : 1.56 inch(0.0396 m) of water
Lubricating oil temperature : 573 ^{0}C
Exhaust gas temperature : 114 ^{0}C

E. 2 Brake Power out put

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

$$\mathrm{P(hp)} = \frac{W(kg).N(rpm)}{2535}$$

$$\mathrm{P(kW)} = \mathrm{P(hp)} \times 0.746$$

Where

P(hp) : brake power output in hp

W : Load of the dynamometer in Kg

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

$$P(hp) = \frac{50.5 \times 2255}{2535}$$
$$= 44.92$$

$$\mathrm{P(kW)} = 44.92 \times 0.746$$
$$= 33.51 \text{ kW}$$

E.3 Fuel Consumption Rate

The fuel consumption rate of the engine is calculated using the following equation:

$$m_f = \frac{\sigma.\rho_w.V}{t} \times 3600 \times 10^{-6}$$

Where

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

V : Volume of fuel flown in time t in ml.

ρ_w : Density of water in kg/m^3

Σ : Specific gravity of the fuel

T : Time to measure the V volume of fuel in Seconds

$$m_f = \frac{0.846 \times 996 \times .153}{60} \times 3600 \times 10^{-6}$$
$$= 7.809\ kg/hr$$

E.4 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:

$$P_B = \frac{p\ (kW)}{\alpha}$$

$$m_f = \frac{m_f}{\beta}$$

Where

P_b : adjusted brake power output in kW

P(KW) : Brake power output in kW

α : Power adjustment factor

m_f : Adjusted fuel consumption rate in kg/h

m_f : Fuel consumption rate in Kg/h

β : Fuel consumption adjustment factor

$$P_B = \frac{33.981}{0.9824}$$
$$= 34.88\,kW$$

$$m_f = \frac{7.74}{1.0024}$$
$$= 7.756\ kg/hr$$

E.5 Air Flow Rate

The mass flow rate of fresh air and the air fuel ratio is determined as follows:

$$m_a = 2.[3600 \times C_d \times A_n \times \sqrt{\frac{2g \times h_m \times \rho_m}{\rho_a}}] \times \rho_a$$

where

m_a : Mass of fresh air inducted per unit time in kg/h

C_d : Co-efficient of discharge of the flow nozzle

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

ρ_a : Density of air in kg/m^3

ρ_w : Density of water in kg/m^3

h_m : Manometric deflection in m of water.

$$m_a = 2.[3600 \times 0.98 \times 7.065 \times 10^{-4} \times \sqrt{\frac{2 \times 9.81 \times 0.0396 \times 995.2}{1.133}}] \times 1.129$$
$$= 150.4\, kg/hr$$

E.6 A/F Ratio

$$A/F = \frac{m_a}{m_f}$$

Where

m_a : Mass of fresh air inducted per unit time in kg/h

m_f : Fuel flow rate in kg/h

$$A/F = \frac{150.4}{7.756}$$
$$= 19.39$$

E.7 Volumetric Efficiency

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

$$\eta_v = \left(\frac{2.m_a}{N V_d \rho_i}\right)\left(\frac{1}{60}\right)$$

Where

η_v : Volumetric efficiency in %

m_a : Mass of fresh air inducted per unit time in kg/h

V_d : Swept volume of all the engine cylinder in kg/m^3

ρ_i : Density of intake air in intake manifold in kg/m^3

N : Engine speed in rpm

$$\eta_v = \left(\frac{2. \times 150.4}{2255 \times 2.875 \times 10^{-3} \times 1.133} \right) \left(\frac{1}{60} \right)$$

$$= 0.0368$$

$$= 68.7\%$$

E.8 Brake Specific Fuel Consumption

The brake specific fuel consumption is computed as follows:

$$bsfc = \frac{m_f}{P_b}$$

Where

bsfc : Brake Specific fuel consumption in gm/kWh

m_f : Adjusted fuel consumption rate in gm/h

P_b : Adjusted brake power in kW

$$bsfc = \frac{7.756}{34.88} \times 1000$$

$$= 222.4\, gm / kW.hr$$

E.9 Brake Thermal Efficiency

The overall brake thermal efficiency of the engine is calculated using the following formula:

$$\eta_b = \frac{p_B}{m_f\, Q_{LHV}}$$
$$= \frac{1}{bsfc.Q_{LHV}}$$
$$= \frac{3600}{bsfc(gm/kW.hr)\,Q_{LHV}\,(Mj/kg)}$$

Where

m_f : Mass of fresh air inducted per unit time in kg/h

P_b : Adjusted brake power in kW

Q_{LHV} : Lower heating value in Mj/kg

bsfc : Brake specific fuel consumption in gm/.kW-hr

$$\eta_b = \frac{3600}{222.4 \times .42.9}$$
$$= 0.379$$
$$= 37.9\%$$

E.10 Fuel Chemical Availability or Availability Input

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

For diesel fuel, $A_{in} = 1.06489\ Q_{in}$

For Sesame oil fuel, $A_{in} = 1.0724\ Q_{in}$

Where

A_{in} : Availability input in kW

Q_{in} : Heat energy iput by fuel in kW

Q_{LHV} : Lower heating value in Mj/kg

bsfc : Brake specific fuel consumption in gm/.kW-hr

$$A_{in} = 1.06489 \times \left(\frac{7.756 \times 42.94 \times 1000}{3600} \right)$$
$$= 98.51\,kW$$

E.11 Percent Availability output at shaft:

The percent availability output at shaft is calculated as follows:

$$A_{Shaft} = \frac{P_b}{A_{in}}$$

Where

A_{shaft} : Percent availability output at shaft in %

A_{in} : Availability input in kW

P_b : Adjusted brake power output in kW

$$\begin{aligned} A_{Shaft} &= \frac{34.88}{98.51} \\ &= 35.4\% \end{aligned}$$

E. 12 Availability Destructions

The sources of availability destructions are:

a) Availability destruction in exhaust gases, A_{eg}

b) Availability destruction in uncounted sources as friction, cooling, etc., $A_{uncounted}$

The availability destruction in the exhaust gases is calculated as follows:

$$A_{ext} = Q_{eg} + m_{eg} T_o \left[C_{p,e} \ln\left(\frac{T_o}{T_{e,o}}\right)\right]$$

Where

m_{eg} : Mass flow rate of exhaust gases = $m_a + m_f$

Q_{eg} : Heat energy of exhaust gases = $C_{p,e}.m_f(1+A/F)(T_{ext} - T_{db})$

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

The specific heat of exhaust gases is calculated as follows:

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

$$\frac{N_{co_2} C_{p,co_2} + N_{H_2O} C_{p,H_2o_2}}{N_{co_2} + N_{H_2O}}$$

$$C_{p,e} = \frac{14.4\times 1.429324 + 12.45\times 1.077626}{26.85}$$
$$= 1.266246$$

$$A_{ext} = 1.266246\times 8.07\times(1+24)\times(728-305) + 8.07\times(1+24)\left\{1.266246\ln\left(\frac{305}{728}\right)\right\}$$
$$= 9.682334$$

The percent availability destruction in exhaust gases is calculated as follows:

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

$$A_{eg} = \frac{9.682334}{98.5}\times 100$$
$$= 9.83\%$$

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

$$A_{uncounted} = [100 - (A_{shaft} + A_{eg})]$$

$$A_{uncounted} = [100 - (35.4 + 9.83)]$$
$$= 54.77\%$$

Appendix F

Uncertainty Analysis

The uncertainty was evaluated based on Kline and Mclintock method (Holman,1994)

$$W_R = \left[\left(\frac{\partial R}{\partial x_1} \omega_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} \omega_2 \right)^2 + \text{----------} + \left(\frac{\partial R}{\partial x_n} \omega_n \right)^2 \right]^{\frac{1}{2}}$$

where, R= R(X_1, X_2, X_3,------------------X_n)

ω_1, ω_2, ω_3,................ω_n, are uncertainties in the independent variables.

A set of calculation with actual data is presented here:

F.1 Data Table

Experimental Data

Applied Load, W	:	10.2 kg
Shaft revolution, N	:	2254 rpm
Volume collected, V=	:	100
Time of collection, t	:	103
Specific gravity of diesel, σ 0.	:	0.846

F.2 Uncertainty in Load Measurement

Error associated with the instrument:

Information provided by the manufacturer

Repeatability : ± 0.03% R. O.

Lineraty : ± 0.05% R. O

$B_1 = .05\% \times 10.2 = \pm 0.0051$ kg.

$P_1 = .03\% \times 10.2 = \pm 0.00306$ kg

Error associated with the resolution

$B_2 = \pm\ 0.05$ kg. (95%) (Figolia, Beasley, 2005)

Precision error associated with the measurement

For measuring the precision error associated with the measurement 30 samples are taken and mean value was computed .

$$\bar{L} = \frac{1}{30}\sum_{m=1}^{30} L_m$$
$$= 10.11\,kg$$

Standard deviation was calculated

$$S_L = \sqrt{\frac{\sum_{m=1}^{30}(L_m - \bar{L})^2}{29}}$$
$$= 0.079\,kg$$

Uncertainty associated for 95% probability is found from Student-t distribution

$P_2 = t_{29,95\%}\, S_L$

$= 2.026 \times 0.079$

$= 0.160$ kg

Total Precisión error $P = |0.160| + |0.00306| = 0.163$ kg

Total Bias Error, $B = |0.0051| + |.05| = 0.0551$ kg

$$Total\ error, U_T = \sqrt{P^2 + B^2}$$
$$= 0.172\,kg$$
$$\approx 0.2\,kg$$

F.3 Uncertainty in Temperature Measurement

Bias error associated with the instrument, $B_1 = \pm 1^{\circ}C$.

Bias error associated with the resolution, $B_2 = \pm 0.5\ ^{\circ}C$

Precision error associated with the measurement is evaluated following the above mentioned procedures, $P_1 = 1.27\ ^{\circ}C$.

Total Bias error, $B = 1.5\ ^{\circ}C$

Total precision error, $P = 1.27\ ^{\circ}C$

Total uncertainty, $U_T = \sqrt{1.5^2 + 1.27^2}$

$$= 1.96\ ^{\circ}C$$

$$\approx 2\ ^{\circ}C$$

F.4 Uncertainty in rpm Measurement

Following the above mentioned conditions uncertainty in rpm measurement was estimated as:

$$U_T = \sqrt{(1+0.5)^2 + 2.9^2}$$

$$= 3.26\,rpm$$

$$\approx 4\,rpm$$

F.5 Uncertainty in Fuel Volume Measurement

Uncertainty in fuel volume measurement was estimated as:

$$U_T = \sqrt{(.8)^2 + .5^2}$$

$$= 0.94$$

$$\approx 1\,ml$$

F.6 Uncertainty in Time Measurement

It is estimated as $U_T = U_o = 0.5$ sec ≈ 1 sec..

F.7 Error in Power Measurement

$$P(kW)=\frac{WN}{2535}\times 0.746=\frac{10.2\times 2254}{2535}\times 0.746=6.77\,kW$$

$$\frac{\partial P}{\partial W}=\frac{0.746\times N}{2535} \qquad \frac{\partial P}{\partial N}=\frac{0.746\times W}{2535}$$

$$=0.663 \qquad =2.99\times 10^{-3}$$

$$\omega_P=0.2\,kg \quad \omega_N=4$$

$$W_P=\sqrt{(2.99\times 10^{-3}\times 4)^2+(0.663\times 0.2)^2}$$

$$=0.133$$

$$\frac{W_p}{P}=1.96\%$$

F.8 Error in Fuel Flow rate

$$m_f=\frac{V\,(ml)}{t}\times\frac{60\times 60}{10^6}\times\sigma$$

$$=\frac{100}{103}\times\frac{60\times 60}{10^3}\times 0.846$$

$$=2.944\ \frac{kg}{hr}$$

$$\frac{\partial m_f}{\partial V}=\frac{60\times 60}{1000\times 103}\times 0.846$$

$$=0.5076$$

$$\frac{\partial m_f}{\partial t}=-\frac{V}{t^2}\times\frac{60\times 60}{1000}\times 0.846$$

$$=0.049068$$

$$\frac{\partial m_f}{\partial\sigma}=\frac{V}{t}\times\frac{60\times 60}{1000}\times 0.846$$

$$=\frac{58}{60}\times\frac{60\times 60}{1000}=3.48$$

$$W_{m_f} = \left[(3.48\times 0.0005)^2 + (0.049068\times 1)^2 + (1\times 0.05076)^2\right]^{\frac{1}{2}}$$
$$= 0.0706$$
$$\frac{W_{m_f}}{m_f} = \frac{0.0706}{2.944} = 0.169\%$$

F.9 Error in bsfc

$$bsfc = \frac{m_f(\frac{kg}{hr})\times 1000}{P_b(kW)}(gm/kW-hr) = 431.9\,gm/kW-hr$$

$$\frac{\partial(bsfc)}{\partial m_f} = \frac{1000}{6.77} = 147.93$$
$$\frac{\partial(bsfc)}{\partial P_b} = -\frac{1000}{(2.924)^2} = -64.423$$

$$W_{bsfc} = \left[(147.93\times 2.944\times 0.169\%)^2 + (-64.423\times 6.77\times 1.96\%)^2\right]^{\frac{1}{2}}$$
$$= 8.579$$
$$\frac{W_{bsfc}}{bsfc} = 1.98\%$$

References

Ali, Y., and Hanna, M. A., "Alternative Diesel Fuels From Vegetable Oils", Bioresearch Technology, vol. 50, pp. 153-163, 1994.

Altin, R., Ceninkya, S. and Serdar, H., "The Potential Of Using Vegetable Oil Fuels As Fuels For Diesel Engine", Energy Conversion and Management, vol. 42, pp. 529-538, 2001.

Al-Najem, N. M. and Diab, J. M., "Energy-Exergy Analysis of A Diesel Engine", Heat Recovery Sytems & CHP, vol. 12, No. 6, pp. 525-529, 1992.

Bejan, A., Tsatsaroins, G. and Moran, M. J., "*Thermal Design and Optimization*", John Wiley and Sons, Newyork, 1996.

Bhattacharyya, S. and Reddy, C. S., "Vegetable Oil as Fuel for IC Engines- A Review", Journal of Agricultural Engineering Research, vol 57, pp. 157-166, 1994.

Bp statistical review 2005.

Canton, J. A., "On the destruction of availability (exergy) due to combustion processes — with specific application to internal-combustion engines", Energy, vol. 25, pp. 1097-1117, 1999.

Demirbas, A., "Fuel Properties and Calculation of Higher Heating Values of Vegetable Oils", Fuel, vol. 77, pp 1117-1120, 1998.

Dorado, M.P. et. al., "Exhaust Emission From A Diesel Engine Fueled With Transesterified Waste Oilve Oil", Fuel, vol 82, pp. 1311-1315, 2003.

Ferguson, C. R. and Kirkpatrick, A. T., "*Internal Combustion Engines*", John Wiley & Sons, Inc., 2000.

Figiola,R. S. and Beasley, D. E., "*Theory and Design for Mechanical Measurements*", John Willey and Sons, Inc., 2005.

German, T. J., "Field Endurance Test of Diesel Engines Fueled with Sunflower Oil Diesel Fuel Blends", SAE paper no. 850239, 1985.

Gerpen, V. J., *et al.*, "Biodiesel Production Technology", National Renewable Energy Laboratory, SR-510-36244, 2004.

Goering, E.C., "*Engine and Tractor Power*", Breton Publishers, 1986.

Haq, M. Z., "*Study Of The Properties Of Vegetable Oil As An Alternative To Diesel Fuel*", M. Sc. Thesis, Department of Mechanical Engineering, BUET, Dhaka, 1995.

Heywood, J. B., "Internal Combustion Engine Fundamentals", McGraw-Hill Book Company,1988.

Holman, J. P., "Experimental Methods For Engineers", McGraw-Hill, Inc., 1994.

Islam, M. M., "*Energy- Exergy Analysis of a Diesel Engine Running on Preheated SVO* ", Department of Mechanical Engineering, BUET, Dhaka, 2006.

Lumely, J. L., "*Engines: An Intorduction*", Cambridge Univeristy Press, 1999.

Mazed, M. A., Summers, J. D. and Batchman, D. G., "Peanut, Soybean and Cottonseed Oils as Diesel Fuels", Transaction of ASAE, vol. 28, pp. 1375-1377, 1985.

Peterson, C. L., Wanger, G. L. and Auld, D. l. , "Vegetable oils substitute for diesel fuel", Transaction of SAE, vol. 26, pp. 322- 327, 1983.

Peterson, C. L., Kornus, R. A, Mora, P. G. and Madson, P. G. , "Fumigation with propane and transesterification effects on injector coking with vegetable oil fuels", Transaction of SAE, vol. 30, pp. 28- 35, 1987.

Plint, M. A., and Böswirth, L., "*Mechanical Engineering Thermodynamics: A laboratory course*", Charles Giffin & Company Ltd., London, 1986.

Pryor, R. W., Hanna, M. A., Shinstock, J. L. and Bashford, L. L., "Soybean Oil Fuel in a Small Diesel Engine ", Transaction of ASAE, vol. 26, pp. 333-342, 1982.

Pulkrabek, W. W,"*Engineering Fundamental of the Internal Combustion Engine*". Prentice Hall of India, 2006.

Rakopoulos, C. D. and Giakoumis, E. G., "Second-law Analyses Applied to Internal Combustion Engines Operation", Progress in Energy and Combustion Science, vol. 32, pp. 2-47, 2006.

Rmadhas, A. S. *et al.*, "Use Of Vegetable Oils As IC Engine Fuels- A Review", Renewable Energy, vol 29, pp. 727- 742, 2004.

Ramadhas, A. S. *et al.*, "Use of Vegetable Oils as I.C. engine fuels-A Review", Renewable Energy, vol. 29, pp. 727-742, 2004.

Schlautman, N. J., Schinstock, J. L. and Hanna, M. A., "Unrefined Expelled Soybean Oil Performance in a Diesel Engine", Transaction of ASAE, vol. 29, pp. 70-80, 1986.

Srivasatava, A. and Prasad, R., "Triglyceride Based Diesel Fuels.", Renewable and Sustainable Energy Reviews volume 4 pp. 111-133, 2000.

Wark, K. Jr., "*Advanced Thermodynamics for Engineers*", McGraw-Hill Inc., 1995