Energy-Exergy Analysis of a Diesel Engine Running on Preheated SVO (Straight Vegetable Oil)

by

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1. Symbols

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

v	Specific volume, velocity
W	Work transfer
w	Work transfer per mole of fuel
y	Mole fraction
α	Power adjustment factor
β	Fuel consumption adjustment factor
ε	Second-law effectiveness or effectiveness
η_v	Volumetric efficiency
η_b	Brake thermal efficiency
η_{II} (η_{av})	Second-law efficiency (Availability efficiency)
μ	Chemical potential
v	Stoichiometric coefficient
ρ	Density
σ	Specific gravity
φ	Relative humidity
$\psi_{ch,f}$	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	Environmental dead state
wb	Wet bulb
x	Lab condition

3. Notations

Δ	Difference
$\bar{}$	Average or mean value

4. 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

Vegetable oils are emerging as promising fuel substitutes to the conventional petroleum fuels from the viewpoint of the energy crisis and emission problems. These seem to be attractive substitute of diesel fuels. Vegetable oils have the advantage of being geographically widely produced, in a variety of products and are renewable in nature and thereby not contributing to the net atmospheric concentrations of green house gas carbon dioxide.

However, higher viscosity and low volatility are identified as the main reasons for the unsuitability of straight vegetable oils as the substitute of diesel fuel. Researchers have reported that the problem of high fuel viscosity can be overcome by using esters, blending and heating. In this study, attempts have been made to evaluate the possibility of using vegetable oils as diesel fuel substitute by modifying their properties by means of preheating.

Engine performance evaluation has been carried out on the basis of both the first and the second laws of thermodynamics. The first law of thermodynamics has been employed to give a general macroscopic performance evaluation. In order to render a critical look into the performance of the engine using preheated soybean oil fuel, the second law of thermodynamics is also utilized. Although conventional performance analysis shows an improvement in engine performance (especially, brake thermal efficiency) due to preheating of soybean oil, the second-law analysis (known as 'availability analysis' or 'exergy analysis') reveals rather a different picture. It shows that the second-law efficiency (known as 'availability efficiency') of the engine running on preheated soybean oil fuel is somewhat less than the brake thermal efficiency of the same. Availability analysis also pinpoints the leakages of major energy losses in terms of 'availability destruction' or 'availability loss' that is a strong tool to ensure the best utilization of the energy input.

Chapter 1

Introduction

1.1 General

The second law of thermodynamics is a rich and powerful tool of related physical observations that has a wide range of implications with respect to operation of thermal systems. For example, the second law can be used as to determine the direction of processes, to establish the conditions at equilibrium, to specify the maximum possible performance of thermal system, and to identify those aspects of processes that are detrimental to overall performance.

Diesel or compression-ignition engines have a wide range of applications. These are characterized by their relatively high efficiency and their capability to meet current environmental and health standards. In general, in case of diesel engines about 50% of the input fuel energy is lost in the cooling water and exhaust gases. The wasted energy in the cooling water is usually considered useless due to low temperature level. However, much attention is focused upon the exhaust gas waste heat and several methods are suggested for recovering it. These include regeneration, turbo – charging, turbo compounding and Rankine engine compounding. For better energy utilization, it is important to take into account the quantity and quality of energy. This can be accomplished by employing both the first and second laws of thermodynamics. The methodology which utilizes both laws is called 'availability analysis' or 'exergy analysis'. Through exergy analysis, the locations where destruction or losses of useful energy occur within a device or process can be identified and rank ordered as to their significance.

The second-law analysis differentiates between high grade energy such as shaft work and low grade energy. Further, the first-law analysis might indicate no loss in energy for some processes e. g. a throttling process, but actually there is real loss in the value or quality of energy. Another point in the comparison between the first and second law analysis is datum (zero level) of energy and availability. While the datum for energy is arbitrarily chosen, the availability datum is chosen scientifically, at the level where there is no further useful work that can be produced. This means that for some processes the enthalpy drop and losses can be same whether it happens near the datum or away from it for the same processes. The availability drops and the loss will differ and depend on whether they are near or far from the datum.

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

The availability or exergy of a substance in a given state is a measure of the maximum obtainable work as the substance proceeds to the dead state while exchanging heat solely with the environment[1].

The availability changes and the irreversibility for chemical reaction of hydrocarbon fuels are studied from two viewpoints [Wark, K., Jr., 1995]:

i. Fuel and dry air composed of O_2 and N_2 react to form products of combustion in the *restricted dead state*. (When a system and its environment are in equilibrium with each other, the system is said to be in *dead state*. The restrictions of temperature, pressure, velocity and elevation characterize a *restricted dead state*. It is restricted in the sense that chemical equilibrium with the environment is not considered. That is, the control mass is not allowed to pass into or react chemically with the environment.)
ii. Fuel and dry air composed of O_2 and N_2 react to form products of combustion which end up in the *environmental dead state*. The recommended values of temperature and pressure for environmental dead state are 298.15K and 1.01325 bars.

The difference between (i) and (ii) is the *chemical availability* of the product gases as they proceed from restricted to unrestricted (environmental) dead state.

1 There may be a second (or more) heat sink instead of single environment. To avoid complexity in analysis heat transfer with only one environment will be considered in the present study.

To this end it is necessary first to determine the work potential of a system at a given state as it proceeds forward a state of equilibrium with the environment while exchanging heat solely with the environment once the system and environment are in equilibrium, no further change of state of the system can occur spontaneously, and hence no further work is performed. When a system and its environment are in equilibrium with each other, the system is said to be in its dead state. The methods for evaluating the availability of heat-transfer process are presented in several of the following section. The availability transfer associated with the transfer of work equals the value of the useful work itself.

An availability balance best defines the effect of all the losses in mechanisms on engine efficiency for the real engine cycle. The combustion and exhaust losses presenting the ideal cycle models are smaller. The loss in availability due to heat losses, flow losses, and mechanical friction are real engine effects.

1.2 Objectives of the Work

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

i. To conduct a detailed study based on both the first and the second laws of thermodynamics in order to examine the effects of speed and load on various thermodynamic processes of a diesel engine operation using preheated vegetable oil as substitute to diesel fuel.
ii. To determine an optimum preheat temperature for the vegetable oil fuel at which it produces performance which is comparable to that obtained with diesel fuel.

1.3 Scope of the Thesis

The first and second laws of thermodynamics are employed to analyze the quantity and quality of energy in a diesel engine running on a non-conventional fuel, i.e., soybean oil fuel. The second-law analysis is necessarily based on the empirical relations formulated by former researchers. Attempt to make an exact thermodynamic analysis has been avoided due to unavailability of some thermodynamic property data of the fuel used.

In this study, available literature regarding, especially, the second-law analysis of heat engine processes has been reviewed in Chapter 2. A precise thermodynamic analysis engine processes is given in Chapter 3, a major portion of which highlighted analysis using the second law of thermodynamics. In Chapter 4, a comprehensive treatment on the fuel chemical availability has been given separately in order to devise a roadmap to perform the so called 'availability analysis' of the test engine. Experimental procedure, description of the test set-up and engine performance parameters, which includes the conventional ones as well as the exergy parameters, is reported in Chapter 5. Finally, in Chapter 6, the outcomes of the experiments are presented graphically and attempt has been made to analyze those critically from the viewpoint of classical thermodynamics. Recommendation for further research works are also proposed in this chapter.

Chapter 2

Literature Survey of Works on Exergy Analysis of Heat Engines

Investigations that have used the second law of thermodynamics to study internal-combustion engines in detailed manner date back to the late 1950s. One of the earliest documented studies was a brief report presented by Traupel in 1957. He compared the availability destruction during combustion process of a naturally aspirated diesel engine and a turbocharged diesel engine and found these values as 22.5% and 21.9% of the fuel availability respectively.

Another pioneering work on this topic was reported by Patterson and van Wylen in 1964. They described an early version of a thermodynamic cycle simulation for spark-ignition engines in which they included determination of entropy values. With the entropy values they determined the availability for the compression and expansion strokes.

Clarke (1976) examined the Otto, Joule and Atkinson air-standard cycles from the perspective of availability and he associated availability destruction. He described the possibilities of achieving higher thermal efficiencies by recognizing the fundamental availability loss mechanisms in internal combustion engines. Clarke stated that to achieve minimum destruction of availability, the combustion process should be under conditions of near chemical equilibrium.

Edo and Foster (1984) reported an availability analysis for an engine which utilized dissociated methanol. The use of dissociated methanol was motivated by the potential to capture exhaust energy by dissociating liquid methanol into more readily used gaseous species such as carbon monoxide (CO) and hydrogen (H_2). They reported availability as a function of equivalence ratio, and showed the various transfer and destruction of the fuel availability.

In the mid-1980s researchers of Cummins Engine Company conducted a series of investigations using availability concept. In 1984, the first of these was reported by Flynn et al. They used a second-law analysis to study a turbocharged, intercooled diesel engine. They showed that of the original fuel availability about 46% was delivered as useful indicated work, 26% was destroyed, 10% was transferred as heat and 18% was exhausted. Primus et al. described another study which was a continuation of the early work (by Flynn et al.). In this study, they used the second law analysis to assess the benefits of turbocharging, charge air cooling, turbocompounding, the implementation of a bottoming cycle, and the use of insulating techniques.

Primus and Flynn (1985) reported on a continuation of their earlier works in which they described the use of the second law of thermodynamics to evaluate the performance of a diesel engine. They conducted a detailed parametric study which examined the effects of a number of engine parameters, viz., engine speed, load, peak cylinder pressure limit, compression ratio, intake air temperature, injection timing and apparent heat release rate shape on various thermodynamic processes of engine operation. They demonstrated the availability losses associated with combustion increase from 21.8 to 32.5% as the engine load is decreased. They attributed this increase in availability destruction with decrease in cylinder temperatures and pressures associated with the combustion process. They also reported that the most significant change in the availability as the engine speed is decreased is the increase in availability transfer to the cylinder walls (due to heat transfer).

Primus and Flynn, (1986) reported a further study which continued their previous work. They focused on itemizing the various loss mechanisms associated with a turbocharged and aftercooled direct-injected diesel engine by studying the in-cylinder and out-of-cylinder processes, viz, in-cylinder heat transfer, combustion, exhaust, friction, turbine, exhaust valve, compressor, aftercooler, intake valve and exhaust manifold heat transfer.

Alkidas (1988 – 89) reported a study which examined the application of second law analysis for a diesel engine. This work was different than many of the other investigations in two major ways. First, he defined the thermodynamic system as outside of the cylinder. Second, he used experimental measurements of the energy rejected to the coolant and lubricating oil, of the brake work, and of the air and fuel flow rates. He then calculated the availability values from the thermodynamic states based on the measured values.

Shapiro and van Gerpen (1989) also used a second law analysis with a standard cycle simulation for a diesel engine. In contrast to the previous investigations, this work included the chemical availability of fuels. They extended their earlier work to include a two-zone combustion model and applied this model to both a compression-ignition and a spark-ignition engine. As before, this study included chemical availability considerations. This work considered only the compression and expansion strokes, and included no consideration of intake or exhaust flows. They presented the time-resolved values of the availability for cases with different equivalence ratios, residual fractions and burn durations. They showed that the combustion irreversibility increases with increasing burn time.

Bozza et al. (1991) described a second law analysis of an indirect-injected, four-cylinder, turbocharged, diesel engine. They used experimental measurements to obtain information for the heat release and flow expressions in their simulation.

Al-Najem and Diab (1991) presented brief results for a turbocharged diesel engine. They reported that about 50% of the fuel availability is destroyed due to unaccounted factors such as combustion, 15% is removed via exhaust and cooling water and 1% is destroyed in the turbocharger.

Rakopoulos (1993) described a first and second law analysis of spark-ignition engine using a cycle simulation and experiment. The major parameters studied were the compression ratio, fuel-air ratio, and ignition advance.

Rakopoulos and Andritsakis (1993) presented results for the irreversibility rates of two four-stroke cycle diesel engines. The first engine was a high-speed, direct injection (DI), naturally aspirated, single-cylinder diesel engine and the second engine was a medium-speed, indirect injection (IDI), and turbocharged six-cylinder diesel engine. They used experimental information to determine the fuel burning rate, and then used the second law of thermodynamics to deduce the irreversibility rates for each engine. They showed that the accumulated irreversibility was proportional to the fuel burnt fraction for wide range of engine loads, speeds, and injection timings.

Rakopoulos and Giakoumis (1997) reported on the use of a computer analysis to assess the performance of a turbocharged, indirect-injected, multi-cylinder diesel engine operated over a range of engine speeds and loads. They provided a detailed parametric study on the effects of speed and load on the availability terms for a range of operations. This work showed that increase in load (with rated increase in equivalence ratio) caused increase in the availability of the cylinder and exhaust gases. As a percentage of fuel's availability, increase in load caused decrease in the combustion and total irreversibility and caused modest decrease in the availability transferred to the cylinder walls.

Caton (1999 and 2000) reported on the use of second law of thermodynamics to study a spark-ignition engine. A commercial V-8 spark-ignition was selected for this study. This work was based on the use of a comprehensive thermodynamic cycle simulation. In one portion of this study, he examined the effects of engine load and speed on a number of performance parameters in energy and availability terms.

Haq. (1995) studied the suitability of vegetable oils as diesel oil substitute. He reported that the calorific values of vegetable oils are slightly less than diesel values. Calorific values and energy densities of vegetable oils are very close to diesel values. But, viscosities of straight vegetable oils were found significantly higher than that of diesel. With adequate preheating, the viscosity of straight vegetable oil becomes comparable with that of the diesel fuel. Same inference was found for density. Here, a comparative study of the values of viscosity and density of diesel fuel and preheated soybean oil is mentioned from this study:

Fuel	**Density (kg/m^3)**	**Viscosity (cSt)**
Diesel at 30°C	818	5
Soybean oil at 30°C	910	60
Soybean oil at 50°C	900	35
Soybean oil at 75°C	882	17
Soybean oil at 100°C	863	11

In summary, most of the studies based on the second law analyses have used some type of engine simulation, although several based their results on measurements of the principal energy terms. The majority of the previous works have been completed for diesel engines. A few studies include some non-conventional characteristics like the use of alternative fuels such as butanol, ethanol and methanol. But, availability analysis of engine processes running with straight vegetable oil is still not that much focused area. Nwafor (2004) studied the emission characteristics of diesel engine running on vegetable oil with elevated fuel inlet temperature. He reported that the hydrocarbon emissions were significantly reduced when running on plant oils. The CO production with heated plant oil is a little higher than the diesel fuel at higher loads. The heated vegetable oil showed marginal increase in CO_2 emission compared to diesel fuel.

The vegetable oils have a high potentiality of as alternative fuel for heat engines. So, it claims more methodical and elaborate investigations to ensure its efficient use in heat engines.

Chapter 3

Availability Analysis of Heat Engines

3.1 Introduction

The first law of thermodynamics, when expressed as a conservation of energy principle, is concerned with the quantity of various forms of energy. From engineering viewpoint a quantity of energy also has quality. Quality is used in the sense of *degree of usefulness* to the society. Changes of system are brought about by work and heat interactions. Since work interactions have higher degree of usefulness, it may be said that such energy forms are of higher quality than heat interactions. Even the second law places a higher standard on work than on heat. Work is completely convertible to heat, but the conversion of heat into work by a cyclic device is highly restricted. Thus, not only is the work more useful than heat, but also more difficult to obtain in many instances [Wark, K., Jr., 1995].

One of the major goals of engineering design is the optimization of a process within given constraints. In the energy field, this implies the optimal use of energy during transfer or transformation. During the last two decades, it has become clear that first-law theory alone often fails to provide adequate insight into the engine operations [Moran, 1992 and Bejan, 1996]. In contrast, second-law analysis, with detailed study of what is happening during a process, has contributed a new way of thinking about and studying various thermodynamic engine processes. The measure of 'usefulness of energy' may be applied to forms of energy in a given state, as well as to energy transformations. Optimization of energy usage is based on the concept that energy has both quantity and quality. The capacity of a given quantity of energy to produce work is accepted as a meaningful measure of the *quality of energy*.

Of interest in engine performance analysis is the amount of useful work that can be extracted from the gases within the cylinder at each point in the operating cycle. The problem is that of determining the maximum possible work output (or minimum work input) when a system (the charge within the cylinder) is taken from one specified state to another in the presence of specified environment (the atmosphere). The first and second laws of thermodynamics together define this maximum or minimum work, which is best expressed in terms of 'availability', sometimes called 'exergy' or 'essergy' (essence of energy) [Caton, 2000].

In a broad sense, the quality of energy is the potential of that energy to produce useful work. The maximum possible useful work that can be obtained from a quantity of energy is called *work potential* in a given environment. Availability (or exergy) is a thermodynamic property that quantifies the quality of energy, but unlike energy it is not conserved. Irreversibility, like entropy production, is a measure of thermodynamic losses in a system and helps us locate and quantify wasteful uses of energy in engineering processes.

3.2 Concept of Availability or Exergy

Related to the analysis based on the second law of thermodynamics is the concept of 'availability' which is also known as 'exergy'. Availability, a thermodynamic property of a system and its surroundings, is a measure of 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. One key aspect of availability is the fact that a portion of a given amount of energy is 'available' to produce useful work, while the remaining portion of the original energy is 'unavailable' for producing useful work [Caton, 2000].

In general, the processes of interest are the thermal, mechanical and chemical processes. An example of the thermal aspect of availability is a case where the system temperature is above the environmental temperature. By utilizing an ideal heat engine (such as Carnot engine), the availability from the system could be converted to work until the system temperature equaled the environmental temperature (the remaining energy is, therefore, the unavailable portion of the energy). An example of mechanical aspect of availability is a system which is at a pressure above the environment. By utilizing an ideal expansion device (such as ideal turbine), the energy of the system could be converted to work until the system pressure equaled the environmental pressure [Caton, 2000].

3.3 Defining Availability or Exergy

A principle that can be revealed from the preceding section is that an opportunity exists for doing work whenever two systems at different states are brought into communication, for, in principle, work can be developed as the systems are allowed to come into equilibrium.

When one of the two systems is a suitably idealized system called an *exergy reference environment* or simply *environment* and the other is some system of interest, *exergy* is the maximum theoretical work obtainable as they interact to equilibrium. At this point, the concept of *dead state* is also important in completing our understanding of the property of exergy.

If the state of a fixed amount of matter in a closed system departs from that of the environment, an opportunity exists for developing work. However, as the system changes state toward that of the environment, the opportunity diminishes ceasing to exist when the two are in equilibrium with one another. This state of the system is called the *dead state*. At the dead state the fixed amount of matter is imagined to be sealed in an envelop impervious to mass flow, at rest relative to environment, and internally in equilibrium at the temperature T_o and pressure p_o. The numerical values of (T_o, p_o) recommended for dead state are those of the *standard atmosphere,* namely, 298.15 K and 1.0325 bars (1 atm). At the dead state, both the system and environment possess energy, but the value of exergy is zero because there is no possibility of a spontaneous change within the system or the environment, nor there can be an interaction between them [Wark, K., Jr., 1995].

On the basis of the above discussion we have the following definition [Wark, K., Jr., 1995]:

The ***availability*** (or ***exergy***) of a closed system in a given state is defined as the maximum useful work output that may be obtained from a system-environment interface as the system proceeds from a given equilibrium state to the *dead state* by a process where any heat transfer occurs only with the environment.

3.4 Exergy Aspects

The important aspects of the exergy concept are mentioned below [Moran and Shapiro, 2000]:

i. Exergy is a measure of the departure of the state of a system from that of the environment. It is therefore an attribute of the system and environment together. However, once the environment is specified, a value can be assigned to exergy in terms of property values for the system only, so exergy can be regarded as a property of a system.

ii. The value of exergy cannot be negative for heat engines (and also turbines). If a system were at any state other than the dead state, the system would be able to change its condition spontaneously toward the dead state; this tendency would cease when the dead state was reached. No work must be done to effect such spontaneous change. Accordingly, any change in a state can be accomplished with *at least zero* work being developed and thus the *maximum* work (exergy) cannot be negative.

iii. Exergy cannot be conserved but destroyed due to irreversibility. A limiting case is when exergy is completely destroyed, as would occur if a system were permitted to undergo a spontaneous change to the dead state with no provision to obtain work. The potential to develop work that existed originally would be completely wasted in such a spontaneous process.

iv. Exergy has been viewed thus far as the maximum theoretical work obtainable from the combined system plus environment as a system passes from a given state to the dead state while interacting with the environment only. Alternatively, exergy can be regarded as the magnitude of the minimum theoretical work input required to bring the system from dead state to the given state.

3.5 Second-Law Efficiency or Availability Efficiency

The first-law efficiency of any thermodynamic device is the ratio of selected energy quantities. But, the usefulness of energy is more appropriately described by its availability. Since availability has its origin in the second law, a performance parameter for a process based on availability concepts is known as *second-law or exergetic efficiency* η_{II}, or *availability efficiency* η_{av} or as *second-law effectiveness,* ε or simply as *effectiveness.* A first law efficiency gauges how well energy is used, whereas effectiveness indicates how well exergy or availability is used [Wark,K.Jr., 1995].

First-and second-law efficiency are different in one other important respect. The first-law is a conservation principle, and first-law efficiencies tend to fall into two general categories. Equipment first-law efficiencies compare actual energy changes to theoretical energy changes under specified conditions. Examples include turbines, compressors, nozzles and pumps. Cycle first-law efficiencies compare desired energy output to required energy input. Thermal efficiency and coefficient of performance (COP) are typical examples. On the other hand, entropy and availability from a second-law viewpoint are non-conserved properties. In the presence of irreversibility, entropy is produced and availability is destroyed. The former effect is measured by the entropy production, and the latter effect is measured by the irreversibility. Hence second-law efficiencies measure losses in availability during a process. A general definition of a second law efficiency is:

$$\eta_{II} = \frac{useful\ availability\ output}{availability\ input} = 1 - \frac{availability\ destrution\ and\ losses}{availability\ input}$$

where losses imply *nonuseful* transfers across the boundary. A second approach especially useful for steady-state devices is:

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

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

3.6 Availability Efficiency of Heat Engines

There are various definitions available in the literature for the efficiency based on second law analysis of processes. A general definition for the availability efficiency is given as follows [Cheng, Ikhumi, and Wen, 1980]:

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

where utility includes heat, electricity, etc.

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

$$\eta_{II} = \frac{A_{utility\ generated}}{A_{utility\ supplied}}$$

Assuming the work generated is converted into 100% electricity [Howell, Buckius, 1992]:

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

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

In summery, like mass, energy, and entropy, exergy is an extensive property that can be transferred across system boundaries. Exergy transfer accompanies heat transfer, work and mass flow. In contrast to entropy, exergy is not conserved. Exergy is destroyed within system whenever internal irreversibilities are present. Entropy production corresponds to exergy destruction and the reduction of the entropy generation is the desired goal of second law analysis.

Chapter 4

Engine Processes and Fuel Availability

4.1 Introduction

The operating cycle of an internal combustion engine can be broken down into a sequence of separate processes: intake, compression, combustion, expansion and exhaust. Thermodynamic analysis of these processes using the first law and the second law of thermodynamics are given in the following sections of this chapter. The aspect of chemical availability of fuels is also discussed in the later sections.

4.2 The First law of Thermodynamics and Combustion: Energy and Enthalpy Balances

In a combustion process, fuel and oxidizer react to produce products of different composition. The first law of thermodynamics can be used to relate the end states of mixtures undergoing a combustion process; its application does not require that the details of the process be known [Heywood, 1988].

The first law of thermodynamics relates changes in internal energy for closed system (or enthalpy for open system) to heat and work transfer interactions. Consider a closed system of mass m which changes its compositions from reactants to products by chemical reaction as indicated in Fig. 4.1.

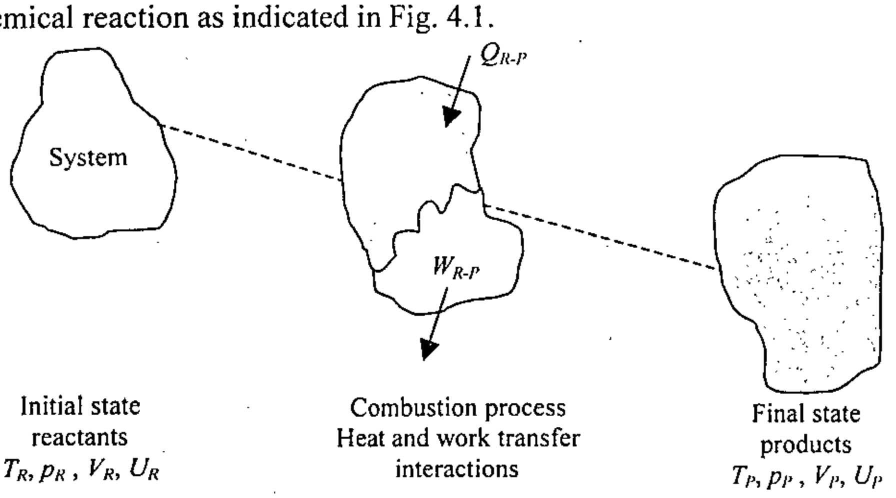

Fig. 4.1 System changing from reactants to products for first law analysis

Applying the first law to the closed system between its initial and final states gives

$$Q_{R-P} - W_{R-P} = U_P - U_R \tag{4-1}$$

Heat transfer Q_{R-P} and work transfer W_{R-P} due to normal force displacements may occur across the system boundary. The standard thermodynamic sign convention for each energy interaction – positive for heat transfer *to* the system and positive for work transfer *from* the system – is used.

We will consider a series of special processes; first, a *constant volume* process where the initial and final temperatures are the same, T'. Then Eq. (4-1) becomes

$$Q_{R-P} = U'_P - U'_R = (\Delta U)_{V,T'} \tag{4-2}$$

The internal energy of the system has been changed an amount $(\Delta U)_{V,T'}$ which can be measured or calculated. Combustion processes are exothermic [i.e., Q_{R-P} and $(\Delta U)_{V,T'}$
are negative]; therefore the system's internal energy decreases. If Eq.(4-2) is expressed per mole of fuel, then $(\Delta U)_{V,T'}$ is known as the increase in internal energy at constant volume and -$(\Delta U)_{V,T'}$ is known as the *heat of reaction at constant volume* at temperature T'.

Then, we consider a *constant pressure* process where the initial and final temperatures are the same, T'. For a constant pressure process

$$W_{R-P} = \int_R^P p\,dV = p(V_P - V_R) \tag{4-3}$$

So, Eq.(4-1) becomes

$$Q_{R-P} - p\left(V_P' - V_R'\right) = U'_P - U'_R$$

or
$$Q_{R-P} = (U'_P + pV_p') - (U'_R + pV_R') \tag{4-4}$$
$$= H_P' - H_R' = (\Delta H)_{p,T'}$$

The enthalpy of the system has changed by an amount $(\Delta H)_{p,T'}$, which can be measured or calculated. Again for combustion reactions, $(\Delta H)_{p,T'}$ is a negative quantity. If Eq.(4-4) is expressed per mole of fuel, then $(\Delta H)_{p,T'}$ is called the increase in enthalpy at constant pressure and $-(\Delta H)_{p,T'}$ is called the *heat of reaction at constant pressure* at temperature T'.

4.3 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 in 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 (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 Fig. 4.2. 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

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

Combining these two equations gives the total work transfer:

$$W_{t,1-2} = -[(U_2 - U_1) - T_0(S_1 - S_2)] \tag{4-5}$$

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:

$$W_{U,1-2} = -\left[(U_1 - U_2) + p_0(V_2 - V_1) - T_0(S_2 - S_1)\right] \tag{4-6}$$

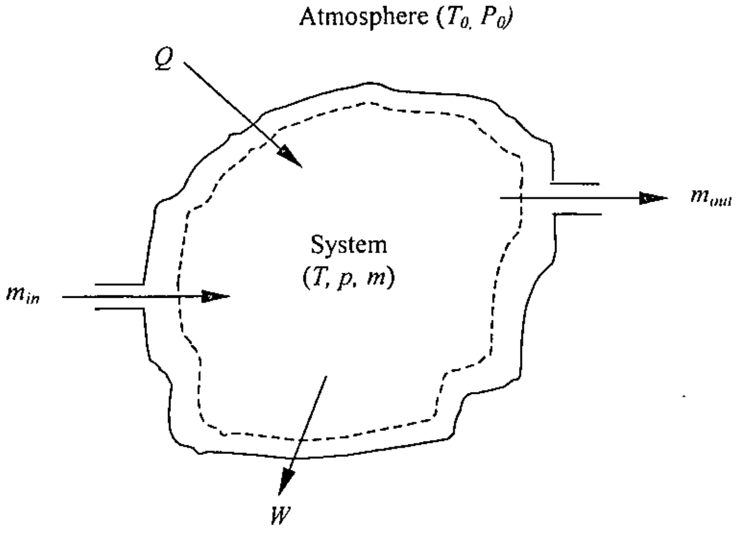

Fig. 4.2 System-atmosphere configuration for availability analysis engine processes

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 (known as 'availability function') which is in communication with the atmosphere

$$A = U + p_0 V - T_0 S \tag{4-7}$$

is thus 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:

$$W_{U,1-2} \leq -(A_1 - A_2) \tag{4-8}$$

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

$$G = H - T_0 S \tag{4-9}$$

which is the *Keenan function*[1] at the temperature T_0.

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 only with the atmosphere), the change in availability ΔA is given by

$$\Delta A = A_{in} - A_{out} - A_{destroyed} \tag{4-10}$$

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 work transfer is equal to the work transfer. The availability transfer dA_Q associated with a heat transfer δQ occurring when the system temperature is T is given by

$$dA_Q = \delta Q\left(1 - \frac{T_0}{T}\right) \tag{4-11}$$

since both an energy and entropy transfer occurs across the system boundary. The availability associated with mass transfer is Eq. (4-9).

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

$$A_{destroyed} = T_0 \Delta S_{irrev} \tag{4-12}$$

1 In general, $G = H - TS$ is known as *Gibb's function*. But, in case of atmospheric condition it is termed as *Keenan function*.

where ΔS_{irrev} is the entropy increase associated with the irreversibilities occurring within the system boundary.

4.4 Fuel Chemical Availability

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 an exergy reference environment. The *thermomechanical availability* is the value for work. The sum of the thermomechanical 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 change and the irreversibility for chemical reactions of hydrocarbon fuels can be studied 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 gases as they proceed from restricted to unrestricted dead state.

To extend these concepts to the most general situation, 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. 4.3 illustrates a control volume where pure fuel enters at the restricted dead state T_0, p_0 and oxidant (O_2) enters from the environment. In general, each product is at its state in the environment, namely, $\bar{h}_{i,00}, \bar{s}_{i,00}, \bar{g}_{i,00} = \mu_{i,00}$. A semipermeable membrane is used to introduce oxygen into the control volume. The products of combustion, primarily CO_2 and H_2O, leave the 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. 4.3 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 process shown in Fig: 4.3. In steady state the energy equation per mole of fuel is

$$q + w_{rev,f} = \Delta H = \sum_{P-R} v_i \bar{h}_i = \sum_{P} v_i \bar{h}_{i,00} - v_{o2} \bar{h}_{o2,00} - \bar{h}_{fuel,0} \qquad (4\text{-}13)$$

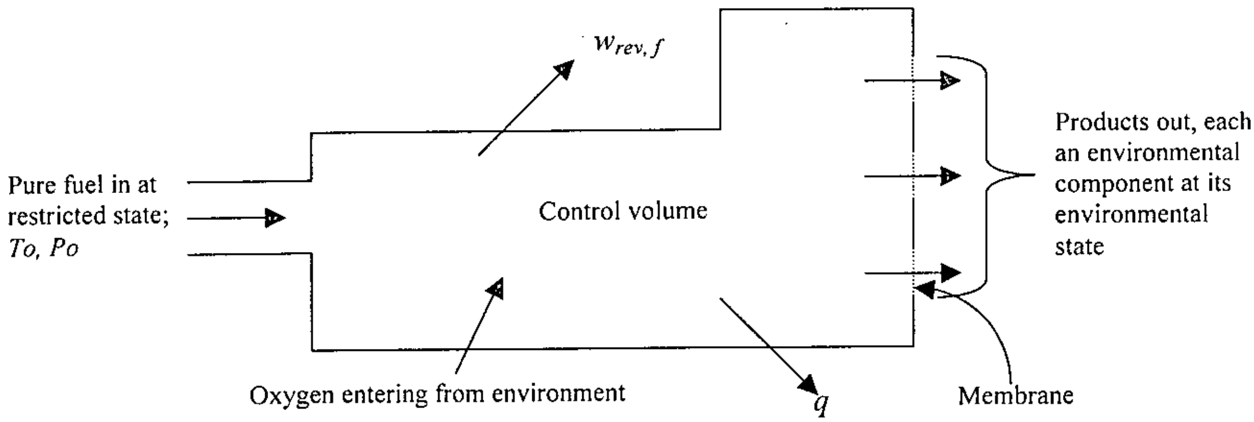

Fig. 4.3 Control volume for the development of fuel availability. Oxygen enter and products level through semipermeable membranes

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

$$q = T_0 \sum_{P-R} v_i \bar{s}_i = T_0 \Big(\sum_{P} v_i \bar{s}_{i,00} - v_{o_2} \bar{s}_{o_2,00} - \bar{s}_{fuel,0}\Big) \qquad (4\text{-}14)$$

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

$$w_{rev,f} = \sum_{P} v_i (\bar{h}_{i,00} - T_0 \bar{s}_{i,00}) - \left[\bar{h}_{00} - T_0 \overline{s_{00}}\right]_{o_2} - (\bar{h}_0 - T_0 \bar{s})_{fuel} \qquad (4\text{-}15)$$

where v_i is the stoichiometric coefficient for ith species.

Finally, recall that chemical potential for ith species is given by $\mu_i = \overline{g_i} = \bar{h}_i - T\bar{s}_i$.

As a result, the preceding equation in terms of chemical potential becomes

$$w_{rev,f} = \sum_P \nu_i \mu_{i,00} - \nu_{o_2} \mu_{O_2,00} - \mu_{fuel,0} \tag{4-16}$$

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 the fuel as it exists 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

$$\psi_{ch,f} \equiv g_{fuel,0} + \nu_{o_2} \mu_{o_2,00} - \sum_P \nu_i \mu_{i,00} \tag{4-17}$$

where $g_{fuel,0}$ replaces $\mu_{fuel,0}$ for a pure substance. Use for Eq. (4-17) 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. Chemical stream availability $\psi_{ch,f}$ is defined by

$$\psi_{ch} \equiv \sum_i y_i (\mu_{i,0} - \mu_{i,00})$$

where y_i is the mole fraction of the of the ith species.

For a pure fuel this becomes

$$\psi_{ch,f} = \mu_{fuel,0} - \mu_{fuel,00} = g_{fuel,0} - \mu_{fuel,00}$$

A comparison of this relation to Eq. (4-17) reveals that

$$\mu_{fuel,00} = \sum_P \nu_i \mu_{i,00} - \nu_{o_2} \mu_{o_2,00} \tag{4-18}$$

where the values come from the stoichiometric combustion relation.

For ideal-gas mixtures the chemical potential of the *i*th component takes on the format

$$\mu_{i,T,ideal} = g_{i,T}^{0} + RT \ln \frac{p_i}{p_0} \tag{4-19}$$

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

$$\mu_{i,00} = g_{i,0} + RT_0 \ln y_{i,00} \tag{4-20}$$

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 $\mu_{i,00}$ equations required for Eq. (4-20) are

$$\begin{aligned}
\mu_{O_2,00} &= g_{O_2,0} + RT_0 \ln y_{O_2,00} \\
\mu_{CO_2,00} &= g_{CO_2,0} + RT_0 \ln y_{CO_2,00} \\
\mu_{H_2O,00} &= g_{H_2O,0} + RT_0 \ln y_{H_2O,00}
\end{aligned}$$

The quantity $g_{i,0} = g_{i,T_0}^{0}$, 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 atmosphere also must be known, including water vapor. When the set of three equations for $\mu_{i,00}$ are substituted into Eq. (4-17), we find that

$$\psi_{ch,f} = -\Delta G_{R,0} + RT_0 \ln \frac{\left(y_{O_2,00}\right)^{\nu_{O_2}}}{\left(y_{CO_2,00}\right)^{\nu_{CO_2}} \left(y_{H_2O,00}\right)^{\nu_{H_2O}}} \tag{4-21}$$

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

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

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

In this format Eq. (4-21) becomes

$$\psi_{ch,f} = -\Delta G_{R,0} + RT_0 \ln \frac{\left(y_{O_2,00}\right)^{x+y/4}}{\left(y_{CO_2,00}\right)^{x}\left(y_{H_2O,00}\right)^{y/2}} \tag{4-22}$$

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

Equations (4-21) and (4-22) 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 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°C and P_0 = 1 atm. The values of Gibbs function of formation Δg_f^0 are found from JANAF Thermochemical Tables[1] and other[2] sources.

4.5 Approximation Methods to Find Fuel Availability of Higher Hydrocarbons

The steady-flow fuel availability of light hydrocarbon gases and their mixture 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 fuels 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

1 JANAF Thermochemical Tables, Dow Chemical Co., 1971
2 Chemical Thermodynamic Properties, NBS Technical Note 270-3, 1968 and *API* Research Project 44, Carnegie press, 1953

hydrocarbons the ratio of $\psi_{ch,f}$ to the lower heating value (LHV) is reasonably constant. Early work in this area was done Szargurt and Petela and subsequently revised by Rodriguez.

Brzustowski and Brena examined the ratio $\psi_{ch,f}/LHV$ homologous series of hydrocarbons. They concluded that the proportionality constant between fuel exergy and LHV is close to 1.065 based on $\psi_{ch,f}$ data at 56 percent relative humidity. 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 and Ferguson, 2004]. As we have adopted some empirical relations for measuring the fuel chemical availability so, from now on 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 $C_xH_yO_zS_w$:

$$A_{in} = Q_{in}\left[1.0374 + 0.0159\frac{y}{x} + 0.0567\frac{z}{x} + 0.5985\frac{w}{x}\left(1 - 0.1737\frac{y}{x}\right)\right] \tag{4-23}$$

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

Moran developed a correlation for liquid hydrocarbons having the simple formula of C_xH_y:

$$A_{in} = Q_{in}\left[1.04224 + 0.011925\frac{y}{x} - \frac{0.42}{x}\right] \tag{4-24}$$

A comparative study among the availability values of some liquid fuels as available in the texts and those calculated using the above Eqs. (4-23) and (4-24) is presented below:

Fuel	***LHV*** **(MJ/kg)**	A_{in} **(MJ/kg)** **[Ferguson & Kirkpatrick, 2004]**	A_{in} **(MJ/kg)** **[Eq. (4-23)]**	**% of Error**	A_{in} **(MJ/kg)** **[Eq. (4-24)]**	**% of Error**
Benzene (C_6H_6)	40.14	42.14	42.28	0.33	42.31	0.403
Octane (C_8H_{18})	44.43	47.67	47.68	0.021	47.5	0.356
Diesel ($C_{14.4}H_{24.9}$)	42.94	45.73	45.726	0.00875	45.64	0.197
Ethanol(C_2H_6O)	26.82	29.71	29.86	0.505	-	-

The above comparative study strongly recommends the use of Eq. (4-23) for determining the availability values of the conventional as well as non-conventional fuels. So, in this study this correlation is used in calculating availability of the test fuels. A point here to be noted that lower heating value of the fuels are used to calculate the exergy parameters instead of higher heating values because in combustion process of fuels, the water vapor is not totally condensed.

More detailed thermodynamic analysis of the engine processes including a number of other engine related parameters is available in the texts. Regarding the empirical formulae for second law analysis, there are many other complex ones. In this chapter, attempt has been made to present the thermodynamic analysis of engine processes in a bit simplified manner. This is well enough for drawing a general performance evaluation.

Chapter 5

Experimental Setup Test Procedure and Analysis of Data

5.1 Experimental Setup

The experimental setup consisted of engine test bed with diesel and soybean oil supply system, different metering and measuring devices along with test engine. Soybean oil feed systems consists with an electric heater placed in a heater box. Soybean oil is allowed to pass through the heater which is controlled by a temperature controller circuit which senses the inlet soybean oil before it enters the fuel injection pump. A thermocouple is placed at the soybean oil inlet line just before the fuel pump which is connected with the overload relay of the temperature controller circuit. Schematic diagram of the experimental set up is shown in Fig 5.1.

Experiments were carried out using a 33.75-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 105mm and a stroke length of 110mm. Its rated output is 33.5kW at a rated speed of 2250rpm. The engine was started by means of a self-starter motor run by a 12V battery. A water-brake dynamometer was used to apply desired loads on the engine to measure the torque. Then the brake power was calculated from this torque and the corresponding speed which was recorded by a built-in tachometer. The technical details of the engine and the dynamometer are given in Appendix A and B, respectively. In the present chapter, the experimental technique is presented in brief. Details of the required equations and sample calculation procedures are procedures are reported in Appendix C.

The dynamometer was connected with the engine by a universal joint and adequate care was taken so that no eccentricity might occur between them. Water was supplied to cool the dynamometer. A thermocouple was used to monitor the brake water temperature so that it can be kept within the limit.

Fuels (both diesel and soybean oil) were fed to the injector under gravity. A heating coil was dipped into the soybean oil container. In order to raise the temperature of the fuel up to a desired limit, the heater was controlled by a controller circuit, which cut off or connected on the electric current by sensing the temperature of the fuel before it enters the injector. For this purpose, a sensor was used at the fuel line just before the fuel 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 the fuel temperature over the set temperature its voltage difference changes and the relay breaks the circuit at that instant.

5.2 Test Procedure

Initially the test engine was run by diesel fuel at three different speeds of 1750, 2000 and 2250 rpm. Then the engine was run by 100% pure Soybean oil preheated by an electric heater at 50, 75 and $100^{o}C$ .At each pre-heating temperature, engine was operated at 1750, 2000 and 2250 rpm by the soybean oil under different loading condition. Engine speeds were maintained within ± 10 rpm of the desired speeds at different loads and the preheat temperature of the soybean oil was maintained within ± 2°C 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 in the experimental and measurement procedures. Any other additional guidelines required were taken from the procedures used by Plint and Böswirth (1986).

The values of the power and fuel consumption rate were properly derated following the derating procedure of BS 5514: Part I: 1982. A sample derating calculation is given in the Appendix D. In this connection, it is mention worthy here that in finding the derating factors the mechanical efficiency of the engine assumed to be 80%. Because, according to the clause 10, note 4 of 5514: Part I: 1982, "The value of mechanical efficiency shall be stated by the engine manufacturer. In absence of any such statement, the value of $\eta_m = 80\%$ will be assumed."

All measuring and metering instruments were calibrated according to the above standards prior to take any reading.

5.3 Measurement of Performance Parameters

5.3.1 Measurement of Engine Brake Power and Speed

A water-brake dynamometer of model no. TLJ-250L, manufactured by Tokyo Meter Co., 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 the rotational speed in rpm readily.

5.3.2 Measurement of Temperatures

At the wall of the silencer one hole was made for the provision of inserting the probe of a K-type thermocouple for measuring the exhaust temperature. Lubricant temperature temperatures were measured using a K-type thermocouple probe inserted into the lubricant oil sump. All temperatures were displayed by a LED digital display (OMEGA-K), connected to different K-types probes through a selector switch.

5.3.3 Measurement of Fuel Consumption Rate

The diesel and soybean oil supply were provisioned so that the either of the fuels could be supplied from a graduated burette instead of physically altering the fuel tank when needed. Fuel consumption was recorded observing the time by a digital stop watch for every 100cc of fuel.

5.3.4 Measurement of Air Flow Rate

Airflow rate was measured by drawing air through two circular nozzles of 30 mm diameter ($C_D = 0.98$) each into an air drum of standard size [in accordance with Plint and Böswirth] that is connected to the engine inlet. The pressure drops across the nozzles were measured by means of a U-tube manometer using water as manometric fluid. This pressure differential was converted into mass flow rate of air by using proper formulae.

5.4 Estimation of Engine Performance Parameters

5.4.1 Volumetric Efficiency, η_v

The power output of an internal-combustion engine is limited by its breathing capacity. There is never any difficulty in introducing any desired quantity of fuel into the cylinder but the maximum amount that can be burnt depends solely on the amount of air available for combustion [Lumley, 1999]. Volumetric efficiency η_v is introduced to account for the effectiveness of the induction and exhaust process. In terms of quantity applied to an actual engine, it is defined as the mass of charge inducted into the cylinder divided by the mass of the mixture that would fill the piston displacement volume at inlet air density in the intake manifold. For four stroke CI engines, volumetric efficiency is given by:

$$\eta_v = \left(\frac{1}{60}\right)\left(\frac{2\dot{m}_a}{NV_d\rho}\right) \tag{5-1}$$

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

N : engine speed in rpm

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

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

5.4.2 Brake Specific Fuel Consumption, *bsfc*

Brake specific fuel consumption is the rate of consumption of fuel per unit time and per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work. The brake specific fuel consumption *bsfc* can be written as

$$bsfc = \frac{\dot{m}_f}{P_b} \qquad \left[\frac{\mathrm{kg_f}}{\mathrm{kWh}}\right] \tag{5-2}$$

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

P_b : brake power in kW, measured at engine flywheel

Low values of *bsfc* are obviously desirable. Typical best values of *bsfc* are 270 for SI engines and 200 (even lower in large engines) for CI engines in g/kWh unit [Heywood, 1988].

5.4.3 Brake Thermal Efficiency, η_b

Brake thermal efficiency is a dimensionless parameter that relates the engine output to the necessary input (fuel flow). The ratio of the brake power output to the rate of fuel energy input is known as brake thermal efficiency η_b. It is given by

$$\eta_b = 3600 \times \frac{P_b}{\dot{m}_f \cdot LHV} = \frac{3600}{bsfc \cdot LHV} \tag{5-3}$$

where P_b : brake power measured at fly wheel in kW

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

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

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 brake thermal efficiency and lower heating value of fuel. If one or more of these parameters goes up then *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.

5.4.4 Brake Mean Effective Pressure, *bmep*

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

$$P_b = bmep.\frac{V_d N}{2} \tag{5-4}$$

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* available nowadays is between 0.9 and 1.1 MPa. However, CI engines have 25-30% lower value of bmep because of much leaner combustion.

5.5 Calculation of Availability Parameters

5.5.1 Fuel Chemical Availability or Availability Input, A_{in}

For both diesel and soybean oil fuel, the fuel exergy input is calculated by using the Eq. (4-23):

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

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

$$A_{in} = 1.06489\, Q_{in} \tag{5-5}$$

In our present study we have taken the formula of soybean oil fuel as $C_{56}H_{102}O_6$ [R. Altin et al., 2001]. So, putting x = 56 and y = 102, z = 6 and w = 0 the above correlation for soybean oil fuel becomes

$$A_{in} = 1.0724\, Q_{in} \tag{5-6}$$

5.5.2 Availability (or Exergy) of the Exhaust Gas

The availability of the exhaust gases is calculated from:

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

So,
$$A_{exst} = Q_{eg} + \dot{m}_{eg}T_0\left[C_{p,e}\ln\left(\frac{T_0}{T_{e,0}}\right)\right] \tag{5-7}$$

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

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

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

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

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

assuming that complete combustion of both of the diesel fuel and soybean oil fuel produces only carbon dioxide (CO_2) and water vapor (H_2O). The constant-pressure specific heat equations of the individual components of exhaust gas are adapted from the data in NASA SP-273. These are as follows:

$$C_{p,CO2} = (2.401 + 8.735\times10^{-3}T - 6.607\times10^{-6}T^2 + 2.002\times10^{-9}T^3)R_u$$

$$C_{p,H2O} = (4.0 - 1.108\times10^{-3}T + 4.152\times10^{-6}T^2 - 2.964\times10^{-9}T^3 + 0.807\times10^{-12}T^4)R_u$$

where R_u is the universal gas constant.

5.5.3 Second-Law Efficiency, η_{II}

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

$$\eta_{II} = \frac{\text{maximum useful work output}}{\text{fuel chemical availability}} = \frac{w_{\max}}{\psi_{ch,f}} \tag{5-8}$$

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

$$A_{shaft} = \frac{\text{brake power output}}{\text{availability input}} = \frac{P_b}{A_{in}} \tag{5-9}$$

There are several other parameters such as emissions, cylinder pressure, residual fraction, coolant temperature, oil temperature, and spark or fuel injection timing. Some measurements are rather straightforward and require little, if any, explanation. Some of the measurements require analysis to obtain the desired result. But, the performance parameters usually applied in practice are those as mentioned in the earlier sections of this chapter.

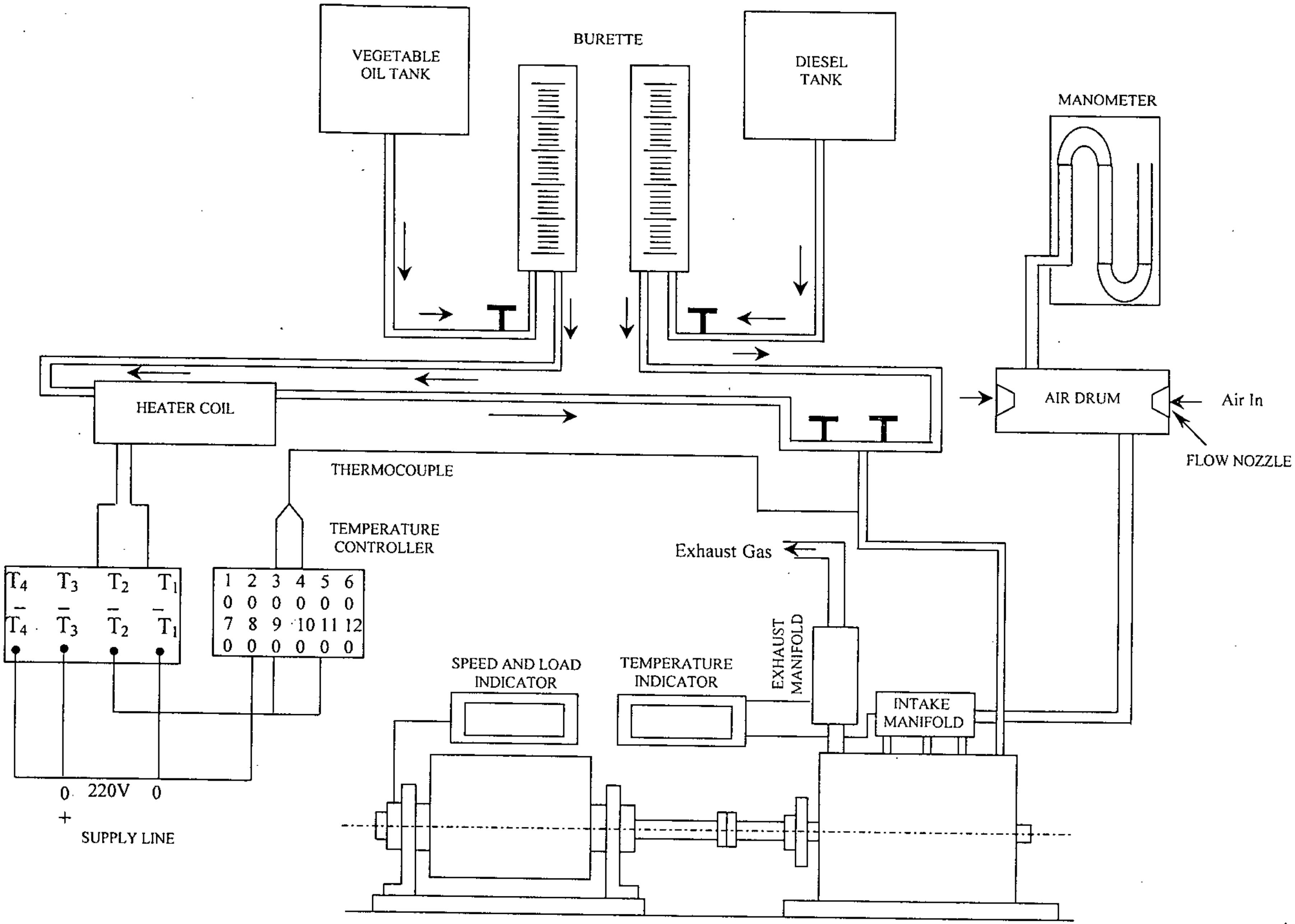

Fig. 5.1 Schematic diagram of the experimental setup showing the major instrumentation

Chapter 6

Results Discussions and Conclusion

6.1 Introduction

Energy and exergy analysis of the thermodynamic processes of a diesel engine are performed in this study. For this purpose first the experiments are carried out at three different speeds (1750, 2000 and 2250 rpm) under variable loading condition using diesel fuel. Then the experiments are repeated at the same speeds but this time the fuel was pure soybean oil preheated at three different temperatures (50, 75 and 100°C). So, a total of 12 sessions of detailed performance study of the engine are carried out using diesel fuel and soybean oil fuel. Thus, the data obtained are analyzed to produce performance parameters on both of the grounds of first law and the second law of thermodynamics. The findings are detailed in the subsequent sections.

At first performance of the engine at three different speeds using diesel fuel is evaluated against the conventional base parameter brake power output in order to test its credibility to employ as a test engine. Then the same parameters are plotted against brake mean effective pressure (*bmep*) to eliminate the effect of change in engine speed. This later technique of performance evaluation of heat engines which is a more recent practice will facilitate to observe the effect of preheating the soybean oil fuel on engine performance.

In the present study, along with the conventional performance analysis of heat engines, the availability (or exergy) analysis based on the second law of thermodynamics has been carried out. Although exact and elaborate second-law analysis of the heat engine processes has been presented by the physicist the thermochemical data like Gibb's function for a wide range of hydrocarbons especially the heavier ones, viz., soybean oil fuel are yet to be determined. So, the availability analysis using soybean oil as fuel is carried out depending, basically, on the empirical relations available in texts. In designating the availability (or exergy) parameters the letter '*A*' is used wherever these are derived from empirical formulas.

6.2 Results and Discussion

Shown in Fig. 6.1 are the variations of the measured air flow rate, m_a with brake power output, P_b of the test engine running with diesel fuel at three different speeds of 1750, 2000 and 2250 rpm. It is seen that for the same P_b, air flow rate increase with engine speed. This is due to the fact that the swept volume per unit time is higher for higher engine speeds. However, with the increase in P_b air flow rate slightly falls due to the increase in intake manifold temperature that reduces the intake air density which, in turn, reduces the air inhaling rate.

In the Fig. 6.2 the effectiveness of the engine's induction process is presented in terms of volumetric efficiency. In the present study, the inlet density is taken as atmosphere air density. So, in this case η_v measures the pumping performance of the entire inlet system. The volumetric efficiency drops with increasing brake power output, but the slope of the curve is not that sheer. During the exhaust stroke, not all of the exhaust gases get pushed out of the cylinder by the piston, a small residual being trapped in the clearance volume (the amount of this residual depends on the compression ratio, and somewhat on the location of the valves and valve overlap). In addition to displacing some incoming air, this exhaust gas residual interacts with the air in two other ways. With the increase in brake power output this exhaust gas residual becomes hotter and when the very hot gas mixes with the incoming air it heats the air, lower the air density, and decreases volumetric efficiency [Pulkrabek, 1997].

The other reasoning noticed from the same figure is that volumetric efficiency falls at higher speeds. There are three possible reasons behind this tendency. Firstly, short cycle time is available for the inhaling process of the engine. Secondly, at higher engine speeds, the flow into the cylinder during at least part of the intake process becomes choked. Once this occurs, further increase in engine speed does not increase the flow rate of air significantly. Thirdly, the viscous flow friction in the air filter, intake manifold and intake valve produces viscous drag which increases as the square of the engine speed. This causes viscous drag pressure drop that in turn reduces the volumetric efficiency of the engine intake system [Heywood, 1988].

Shown in Fig. 6.3 are the variations of fuel flow rate, m_f with brake power output, P_b of the test diesel engine running on diesel. In diesel engines, output power increase is achieved by increasing the fuel flow rate. Hence, the fuel flow rates increase with P_b. It is also observed that at part loads and at higher engine speeds, more fuel is consumed to generate the same power. But, near rated power, the engine consumes almost the same amount of fuel to generate the same P_b for all the three speeds.

The variation of air-fuel ratio, *A/F* with brake power output, P_b is shown in Fig. 6.4. It is readily evident from this figure that there is steep dwindle in the *A/F* up to about 45% of the rated power output; the falling trend continues over the full range of P_b, but gently. In CI engines, output power increase is achieved by increasing the fuel flow rate while, at the same time, air flow fall off very slightly with increasing P_b. So, requirement of fresh air for burning the same amount fuel gradually decrease. There is a noticeable 'cross-over' of the *A/F* curves at almost mid of P_b. That is, at higher loads, *A/F* slightly increases at elevated engine speeds. A CI engine operates with unthrottled intake air, controlling engine power by the amount of fuel injected. At heavy loading condition, more-than-normal amount of fuel is injected into the cylinder. So, at certain mean effective pressure inside the cylinder the engine will require increased amount of air to produce the same power output at higher speeds due to the decrease in the pumping capability (expressed in terms of η_v as shown in Fig. 6.2) of the engine intake system [Pulkrabek, 1997].

The efficiency of the test engine in using the fuel supplied in producing work is shown in Fig. 6.5. The variation of brake specific fuel consumption, *bsfc* as a function of brake power output, P_b demonstrate this fuel utilization efficiency. The early sharp decline of the curves manifest the poor fuel conversion efficiency of the engine at lower brake power output. The *bsfc* remains almost invariable in the average operating range, then again it increase after the rated power output (which is not shown in the figure as experiments were not carried out up to this zone). The other attention-grabbing point apparent from the same figure is that at relatively lower P_bs, higher *bsfc* is achieved with higher speeds. But, as the brake power output advances to the rated value the discrepancy among the *bsfc* curves tends to lessen and near rated P_b they coincide on each other. At low power, the effect of cylinder pressure is not significant. Frictional losses in this range are primarily dictated by the energy spent in moving shaft, valves, etc. Therefore, at lower engine speeds *bsfc* is low as frictional resistances in moving parts are less. With increasing P_b, the effect of cylinder pressure becomes more crucial. At relatively high P_bs the mean effective pressure for low engine speeds is comparatively higher. Hence, frictional resistance due to cylinder pressure at low speeds is somewhat greater than that at higher speeds. For this reason, with increase in brake power output the *bsfc* lines for different speeds get closer. Moreover, heat losses with exhaust gases at higher speeds are less than that at low speeds. All these factors together contribute to bring about the coincidence of *bsfc* lines at higher P_bs [Heywood, 1988 and Haq et. al., 2003].

The brake thermal efficiency, η_b (sometimes called enthalpy efficiency) of the test engine is plotted against brake power output in the Fig. 6.6. This is almost mirror of the previous figure (Fig. 6.5), i.e., η_b varies inversely as the *bsfc*. It is evident from this figure that at lower brake power outputs, η_b for lower speeds is higher compared to that for higher speeds. For a given mean effective pressure, the intake pressure increase with increasing thinning of air resulting from inhaling excess air at lower engine speeds. This reduces the pumping work of the engine intake system, thus improving η_b. Moreover, as the burnt gas temperature is lower due to dilute charge the heat losses to the cylinder walls are reduced. This also improves the η_b at lower speeds [Heywood, 1988].

Engine performance parameters presented as a function of brake power output gives a unique trend only for a particular engine at a particular speed. The performance curves for engines of different sizes or even the same engine at different speeds deviates far from each other. So, a dimensionless parameter is needed to characterize the engine processes in a precise manner. Unfortunately, there do not seem to be any appropriate dimensionless group that can serve the purpose duly. However, *bmep* is useful in this regard as it is roughly comparable even for very different engines. Because, different engines burn the same fuel necessarily under approximately the same condition, and hence produce almost identical mean effective pressure. This reasoning also holds for an engine running at different speeds [Lumley, 1999].

Both the figures Fig. 6.7 and Fig. 6.8 establish the above-mentioned fact. The *bsfc* and 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, from now on the performance and availability terms are evaluated as a function of *bmep*. This alternative base of performance evaluation is imperative in observing the effect of preheating the fuel when soybean oil was used instead of diesel. However, the trend of both *bsfc* and η_b curves are necessarily the same as those against brake power output, P_b.

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:

i. useful brake output availability
ii. availability transferred to cooling medium
iii. availability transferred to exhaust gases
iv. availability destroyed in engine accessories turbocharger, cooling fan, etc.
v. availability destroyed due to friction and radiation heat loss to surroundings

In the present analysis, due to scarcity of thermochemical data (especially, in case of soybean oil fuel) 'percent brake output availability', A_{shaft} is used in lieu of availability efficiency or second-law efficiency, η_{II}. The availability is destroyed or lost due to different irreversibilities 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 combinedly 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, Diab, 1991]. Availability destruction in exhaust gases, A_{eg}, is evaluated separately.

The Fig. 6.9 shows the portion of the input availability converted into brake output power which we denote by A_{shaft}. It can be noted from this figure that at higher *bmep*s availability output at shaft becomes quite constant with a slight declining trend after rated output. This trend is obvious as explained for the Figs. 6.6 (η_b vs P_b) and 6.8(η_b vs *bmep*); only the extent lower than in this case.

The Fig. 6.10 shows how the availability input is expensed in different processes of the engine. It can be noticed that availability transfer to the exhaust gases (denoted by A_{eg}) increase with increasing *bmep*, but is quite a small portion of the availability input, the maximum level reaching approximately 10% of A_{in}. In contrast, the availability destruction in friction, cooling, 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 facilitates to confirm an 'optimum operating point' from this graph. From this figure we notice that the optimum operating point is between 0.40 – 0.45 MPa *bmep*. Another feature that may be pointed out from this plot is that at optimum operating range the difference between A_{shaft} and $A_{uncounted}$ is of the order of about 20% or less and the difference between A_{shaft} and A_{eg} is of the order of about 35%.

In the Fig. 6.11, the comparison η_b with A_{shaft} is shown. To obtain a precise contrast of the two, the same are plotted for the rated speed 2250 rpm on the same plane in Fig. 6.12. Observation reveals that A_{shaft} is somewhat less than η_b throughout almost the entire range except at lower values (< 0.15 MPa) of *bmep*. Both of 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 (or higher) heating value of the fuel. And this available energy (or exergy) cannot be interpreted into shaft work due to inherent irreversibilities associated with pertaining engine processes.

The performance curves of the test engine using soybean oil as the fuel at three different preheat temperatures are presented in the Figs. 6.13 – 6.18.

Shown in Fig. 6.13 is the variation of brake thermal efficiencies as a function of *bmep* for three different preheat temperatures. It is evident from this figure that for a particular *bmep*, brake thermal efficiency η_b increases substantially with increasing preheat temperature. This is due to the fact that at higher preheat temperatures the viscous soybean oil gets leaner which improves the spray pattern of the fuel and helps penetrate deep into the charge air inside the cylinder As a result, a more homogeneous combustible air-fuel mixture is obtained by supplying preheated fuel to the combustion chamber. Fig. 6.14 shows the variation of percent availability output at shaft, A_{shaft} as a function of *bmep* for three different preheat temperatures. These curves comply with trend of those in Fig. 6.13 having the similar explanation behind.

In the Fig. 6.15 are shown the availability destructions from different sources and the impact of preheat temperature on their extent. As noted from the figure, the level of availability destruction in exhaust gases, A_{eg} is almost invariant with increasing preheat temperature. But, availability destruction in friction, cooling, etc., $A_{uncounted}$ decrease a bit with the increase in fuel preheat temperature. This is due to the fact that the increasing preheat temperature of the fuel increases the fuel conversion efficiency which results in increased temperature inside the cylinder. In this way the cylinder blocks and as well as the oil sump gets more heated that, in turn, heats up the lubricating oil. The lubricant gets leaner which reduces the viscous friction in moving parts of the engine.

Fig. 6.16 shows a comparison between the brake thermal efficiencies using soybean oil fuel at 100°C and diesel fuel at room temperature (30±2°C). It can be seen that brake thermal efficiency using soybean oil at 100°C has become comparable to that obtained using diesel fuel at room temperature. The heavier the fuel the less it is being atomized by the fuel injector. So, at lower temperatures the explosive mixture of air and fuel remains non-homogeneous, which affects the burning efficiency thus resulting in poor η_b. But, penetration rate of soybean oil spray is increased and cone angle is decreased as the viscosity is reduced by increasing the temperature of the oil. This contributes in improving the engine performance at elevated fuel inlet temperatures. Almost identical elucidation is associated with the Fig. 6.17, which shows a comparison between the percent availability output at shaft, A_{shaft} using soybean oil fuel at 100°C and diesel fuel at room temperature (30±2°C).

Shown in Fig. 6.18 a contrast between the availability destructions from different sources using soybean oil fuel at 100°C and diesel fuel at room temperature (30±2°C). Although the availability destruction in exhaust gases, A_{eg} does not vary that much the availability destruction in friction, cooling, etc., $A_{uncounted}$ using soybean oil drops slightly compared to that using diesel fuel. This is due to the fact that increased fuel inlet temperature helps to reduce some of the loss heads like combustion loss, fluid flow loss, etc., but the magnitude is not that significant ($< 3.5\%$).

Finally, Figs. 6.19 and 6.20 are presented to show the outcome of preheating of soybean oil fuel whether it improves the efficiency (both η_b and A_{shaft}) of the engine or not.

Shown in Fig. 6.19 is a comparison among the brake thermal efficiencies of the test engine run by diesel fuel at room temperature (30±2°C) and soybean oil fuel at three different preheat temperatures (50, 75 and 100°C), all at the rated speed of 2250 rpm. Engine with soybean oil at 50°C preheat temperature exhibit lower efficiency than that with diesel fuel. As the preheat temperature is increased (75 and 100°C) the performance of the engine using soybean oil fuel gets closer to that using diesel fuel. The performance curve of soybean oil fuel at 100°C preheat temperature becomes comparable with that of diesel fuel. This happens due to the fact that at 100°C, both the density and viscosity of soybean oil become very much closer to those values of diesel fuel [Haq, 1995]. So, at this temperature the spray pattern for soybean oil fuel bear a resemblance to that for diesel fuel. Also, the penetration of fuel injection becomes comparable. The consequence of the Fig. 6.20 is more or less alike to the previous figure. Obviously, the reasons behind are the same.

6.3 Conclusion

Experimental study of a diesel engine run by diesel fuel and preheated soybean oil fuel is carried out. In this present study, attempt has been made to perform a thermodynamic analysis of preheated SVO as the alternative of the conventional diesel fuel for CI engine. The test engine is run separately by diesel fuel and preheated soybean oil fuel. Engine performance parameters are obtained using both the conventional practice and the analysis based on second law of thermodynamics, popularly known as 'availability analysis'. The availability analysis is carried out using empirical correlations presented by other researchers. So, it is an approximate analysis rather than an exact one.

The major inferences that may be drawn from the above study are listed as follows:

- Preheated straight soybean oil may be a practical replacement of the conventional diesel fuel with a small power and efficiency drop.
- The brakes thermal efficiency of the engine apparently increases with increased preheat temperature of the soybean oil fuel and at 100°C it becomes very much comparable with the performance trends obtained using diesel fuel. Same implication is true in case of availability efficiency, but its value is less by 1 – 1.15%.
- The major part (almost 50%) of the available energy is wasted and lost due to uncounted factors.
- The availability destroyed in exhaust gases is about 10% of the available energy. This may be utilized directly for preheating purpose, thereby improving the actual efficiency of the engine.

6.4 Recommendations for Further Works

For further research works, the followings are recommended:

- To facilitate exact availability analysis of the of CI engine using preheated SVO as alternative fuel, extensive experiments should carried out in preparing a complete database of the thermochemical data of these heavy hydrocarbons.
- Mathematical model based on analytic treatment of the problem, using second law of thermodynamics, may serve to understand the interior

situation of the energy utilization and destruction in engine process that will help to reduce availability destructions.

- Experimental facility may be developed so that availability transferred to cooling air could be detached from $A_{uncounted}$.
- Emission characteristics of diesel engine using preheated SVO should be investigated.

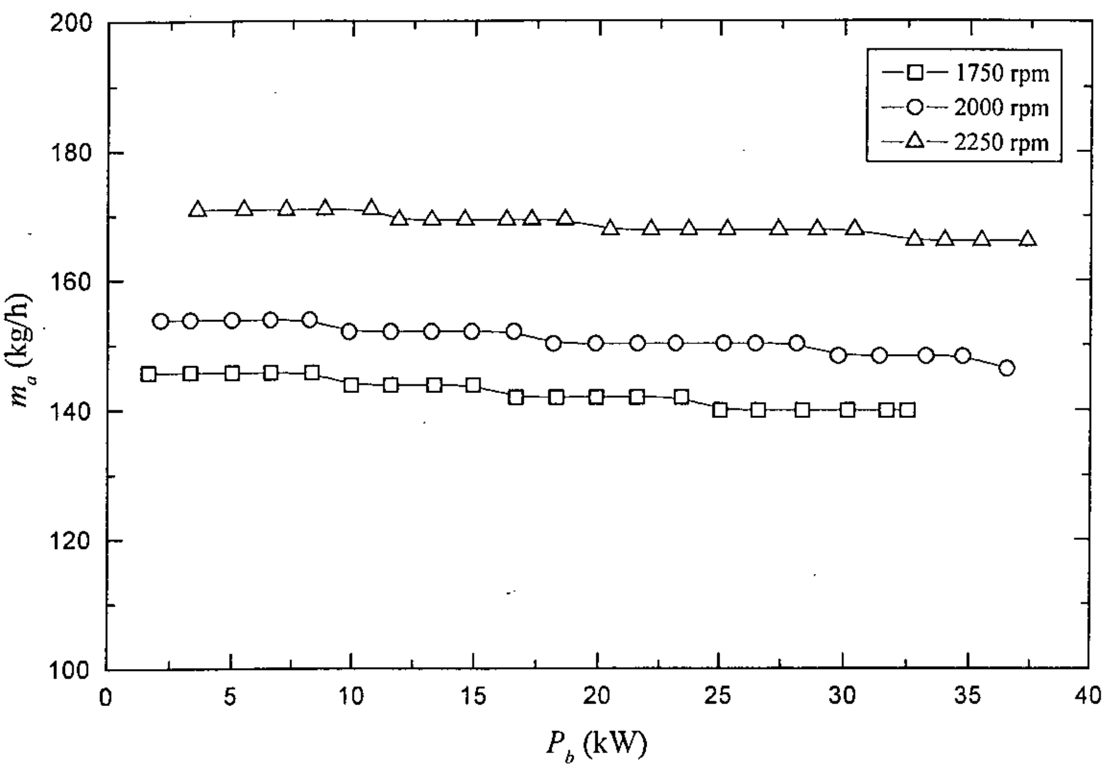

Fig. 6.1 Measured air flow rate as function of engine brake power output run by diesel fuel

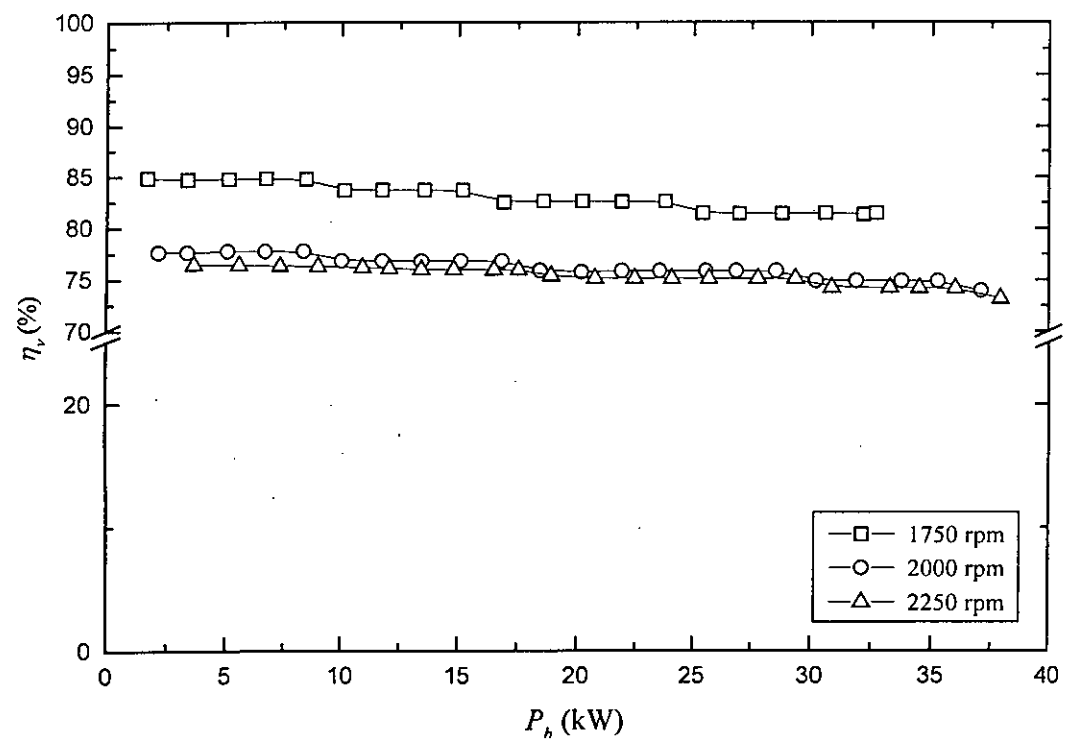

Fig. 6.2 Variation of volumetric efficiency with engine brake power output run by diesel fuel

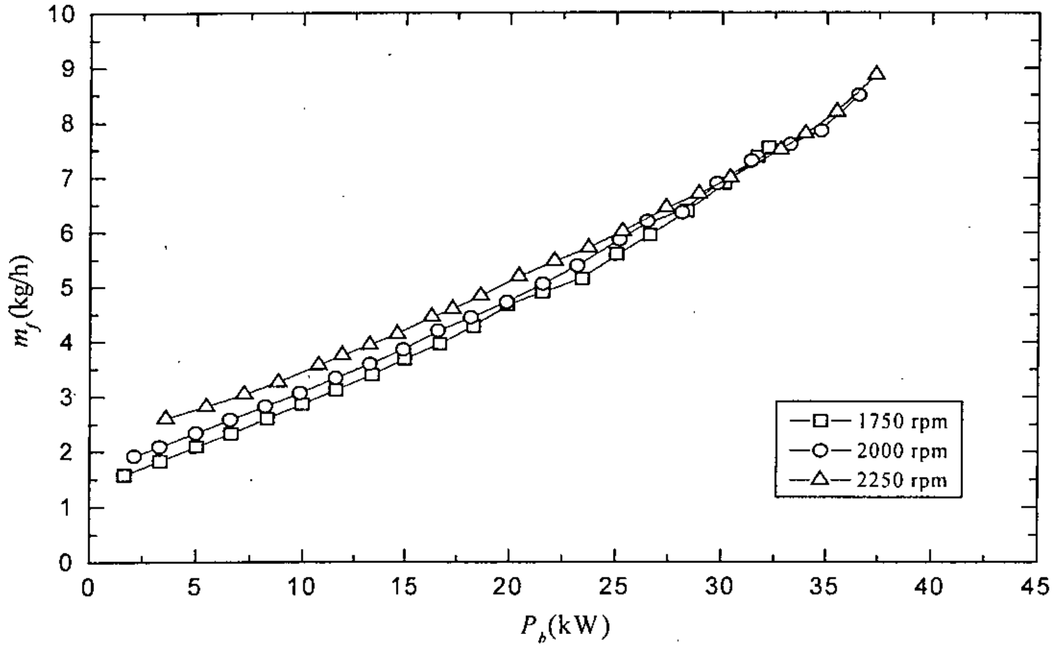

Fig. 6.3 Variation of mass flow rate of fuel with engine brake power output run by diesel fuel

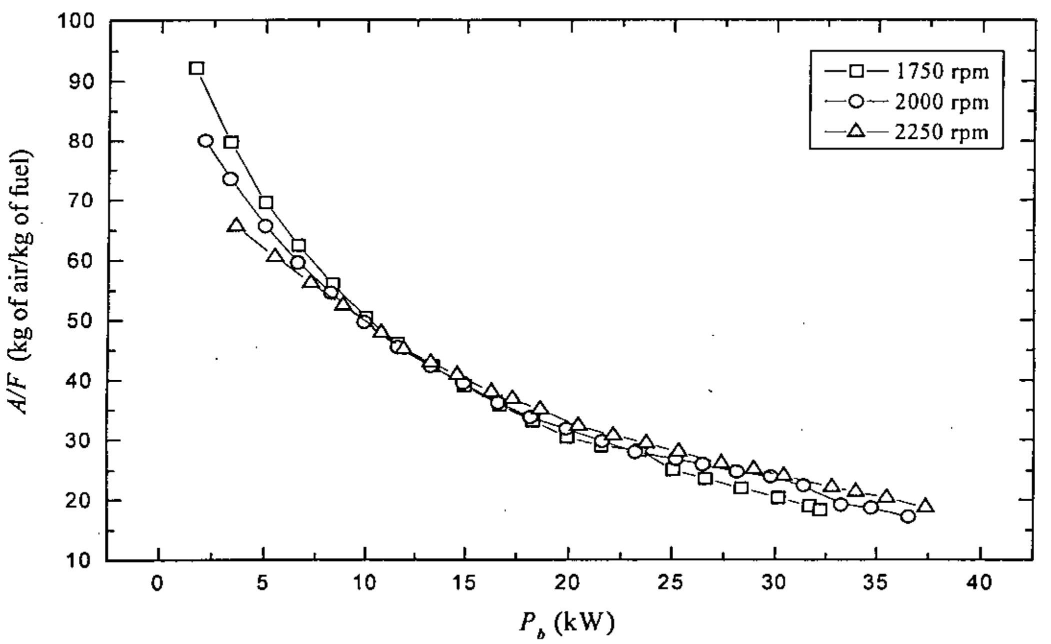

Fig. 6.4 Variation of air-fuel ratio with engine brake power output run by diesel fuel

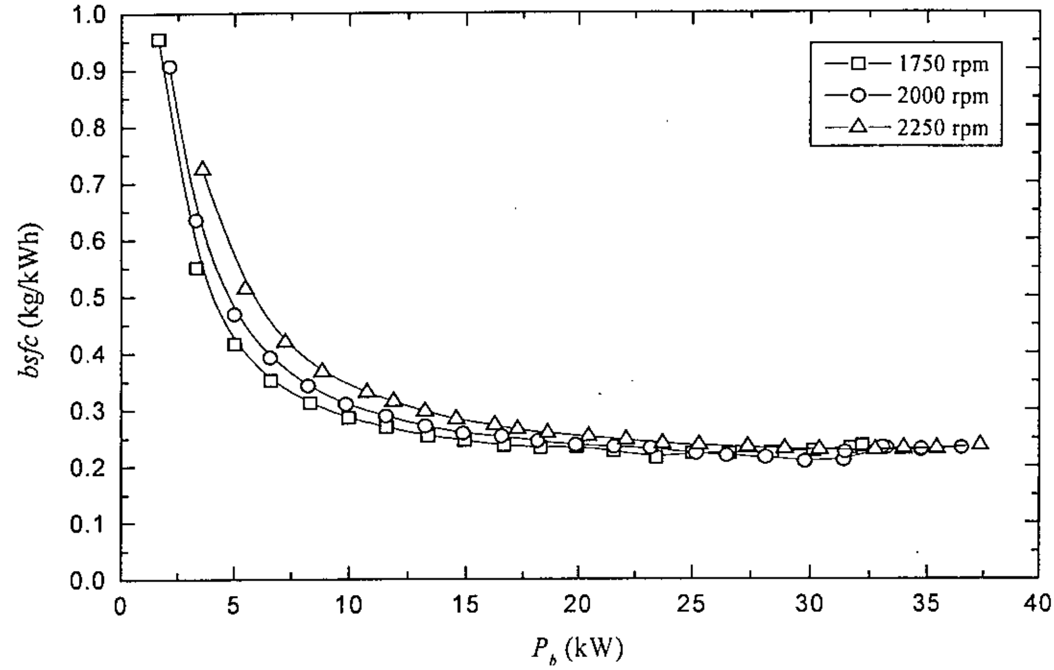

Fig. 6.5 Variation of brake specific fuel consumption with engine brake power output run by diesel fuel

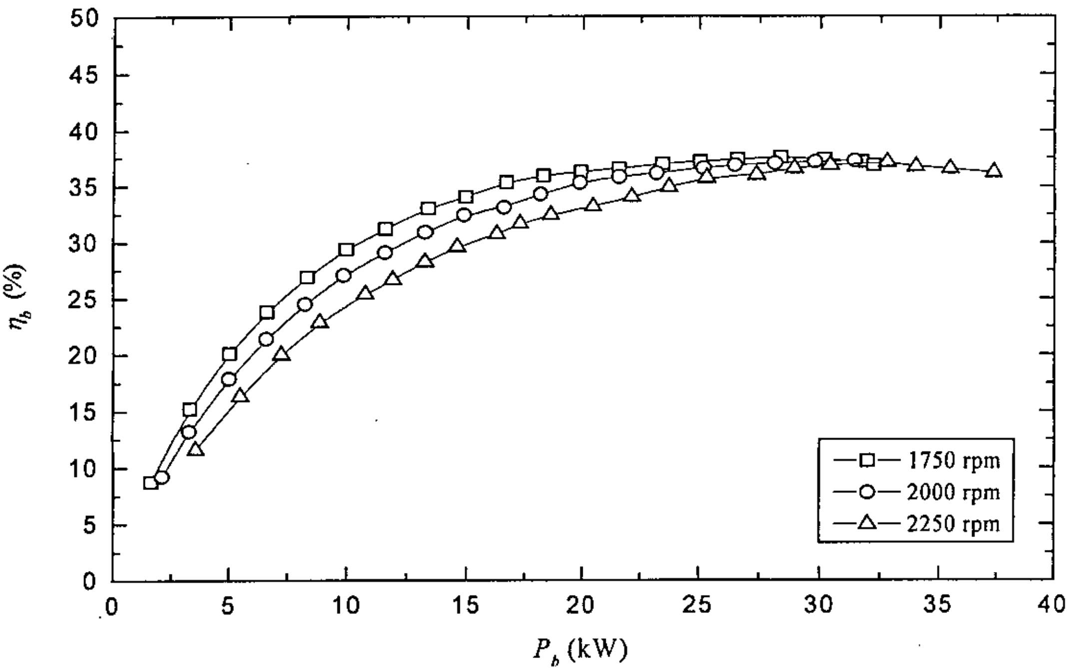

Fig. 6.6 Variation of brake thermal efficiency with engine brake power output run by diesel fuel

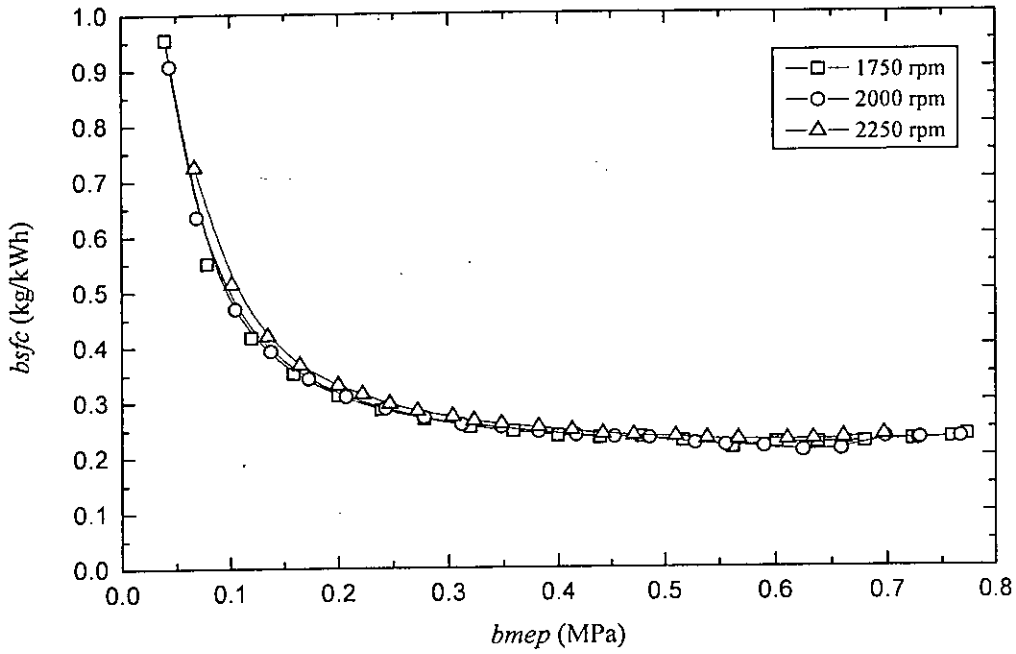

Fig. 6.7 Variation of brake specific fuel consumption with engine brake mean effective pressure run by diesel fuel

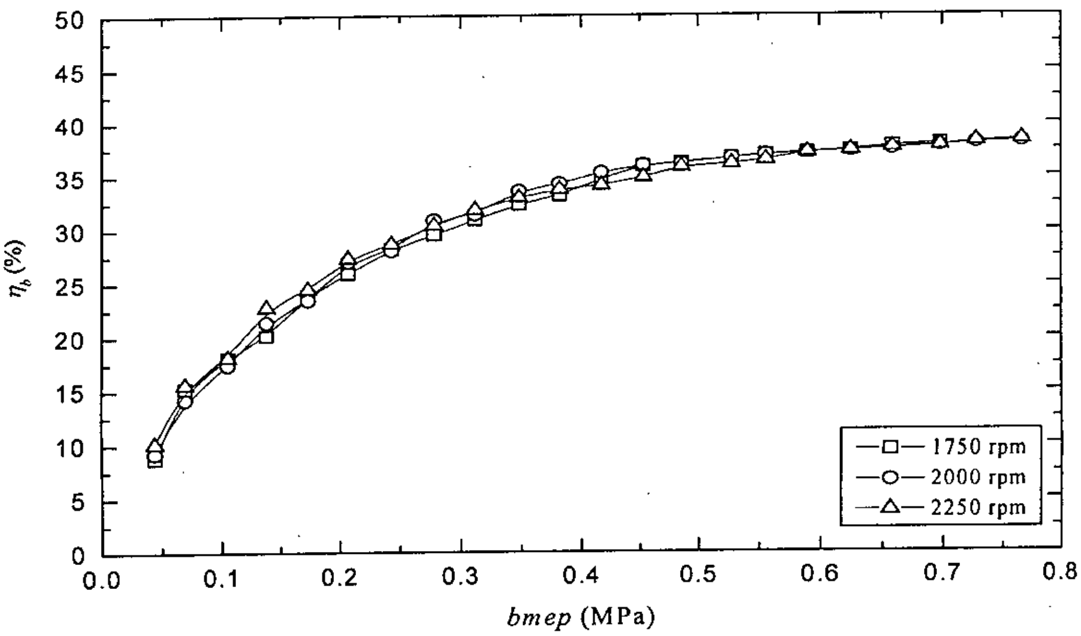

Fig. 6.8 Variation of brake thermal efficiency with engine brake mean effective pressure runs by diesel fuel

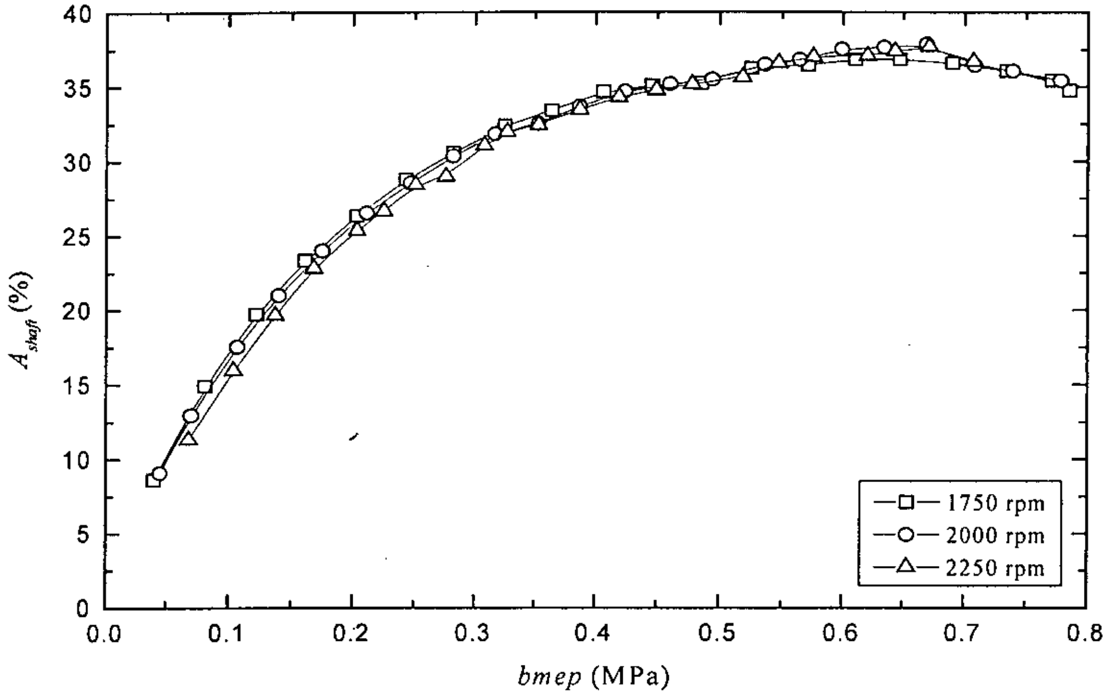

Fig. 6.9 Variation of percent availability output at shaft with engine brake mean effective pressure run by diesel fuel

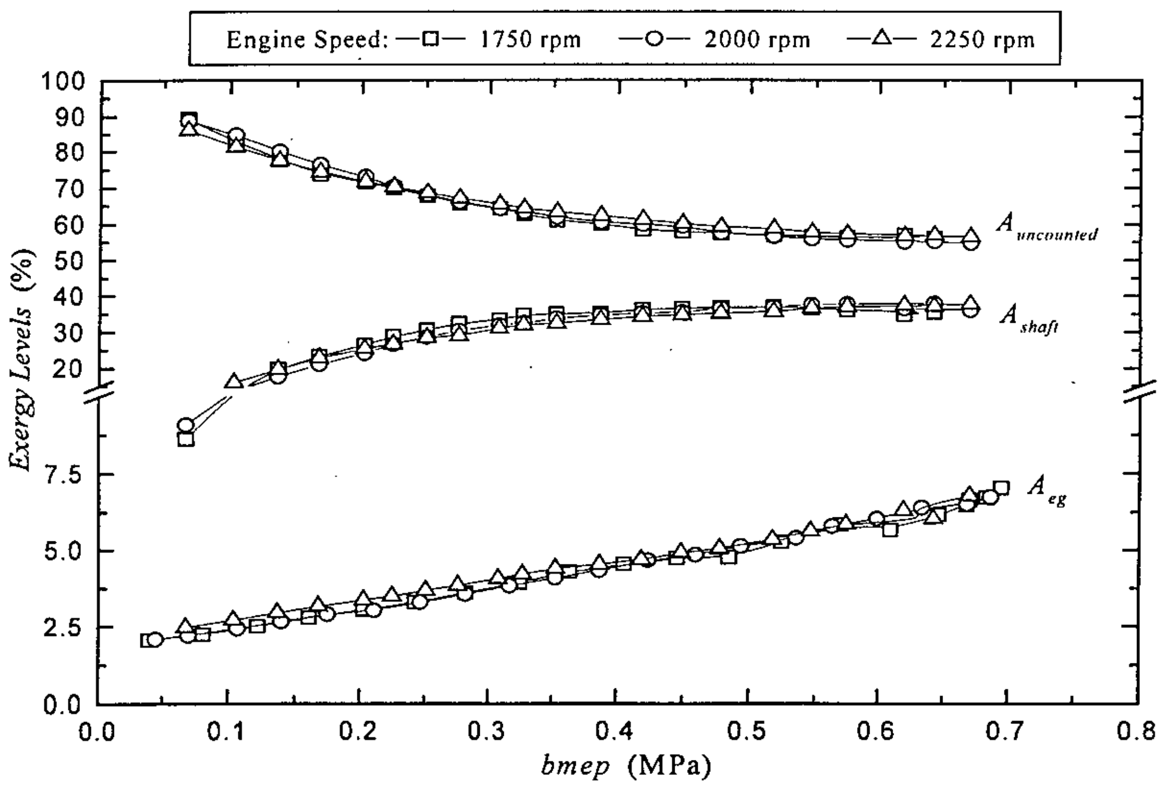

Fig. 6.10 Level of different types of availabilities associated with engine operation as a function of brake mean effective pressure run by diesel fuel

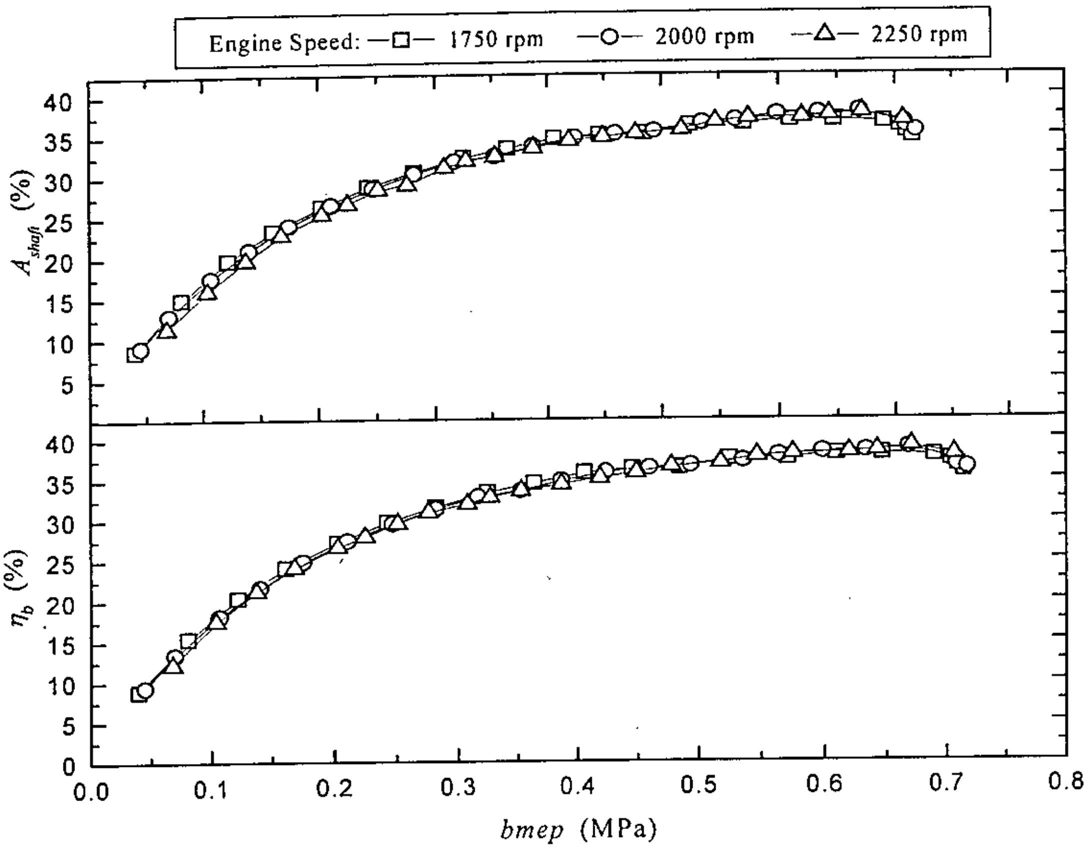

Fig. 6.11 Comparison of brake thermal efficiency with percent availability output at shaft as a function of brake mean effective pressure run by diesel fuel

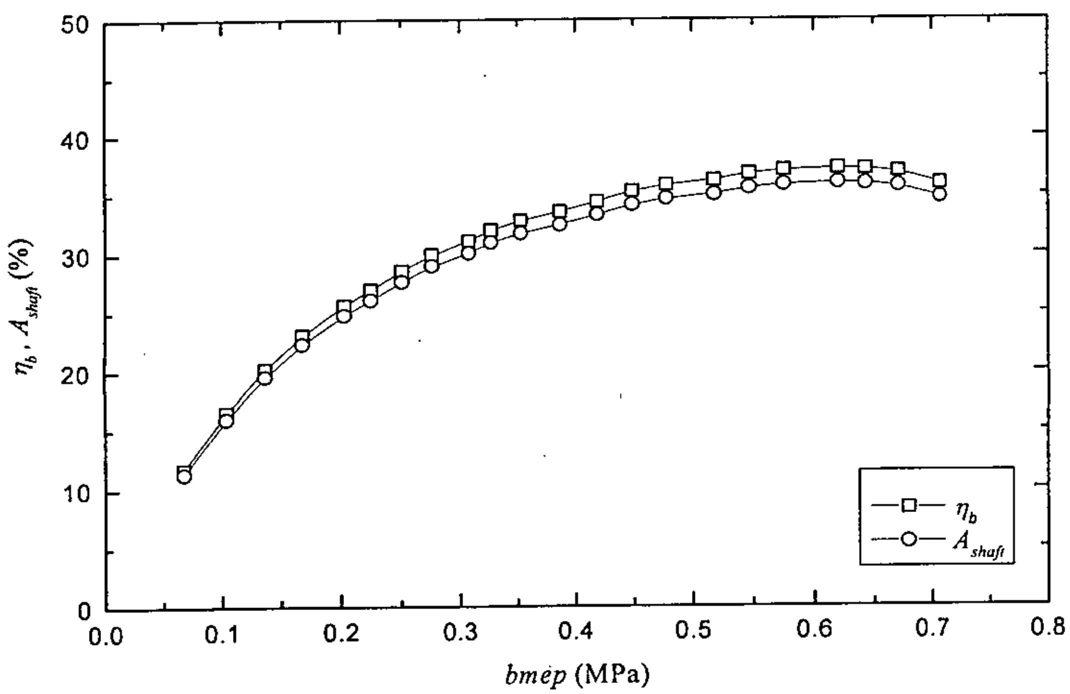

Fig. 6.12 Comparison of percent availability output at shaft with corresponding brake thermal efficiency as a function of brake mean effective pressure run by diesel fuel at 2250 rpm

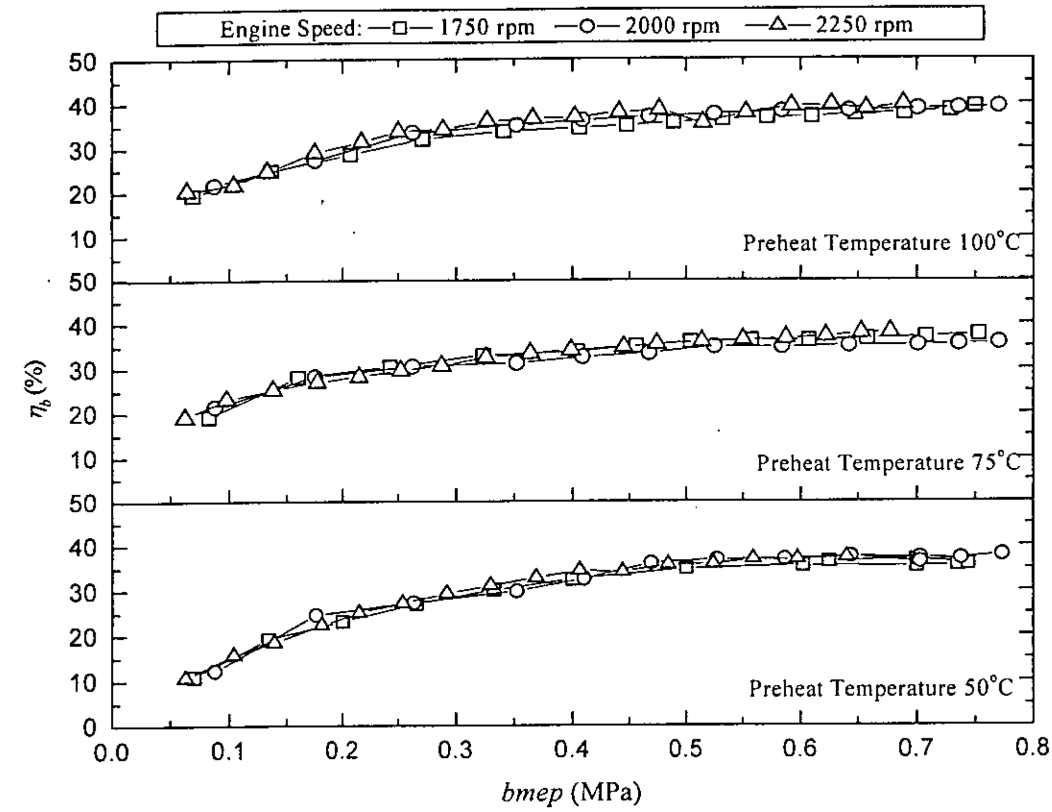

Fig. 6.13 Estimated brake thermal efficiency with brake mean effective pressure run by soybean oil fuel at different preheat temperatures

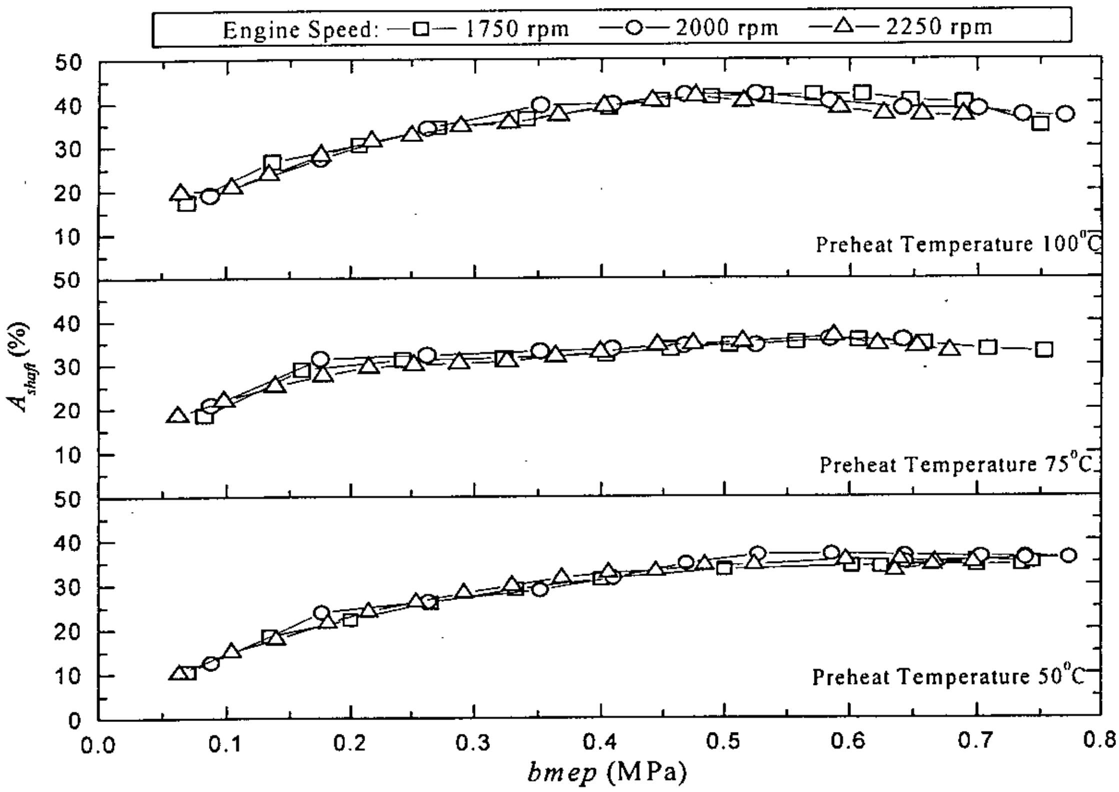

Fig. 6.14 Estimated percent availability output at shaft with brake mean effective pressure run by soybean oil fuel at different preheat temperatures

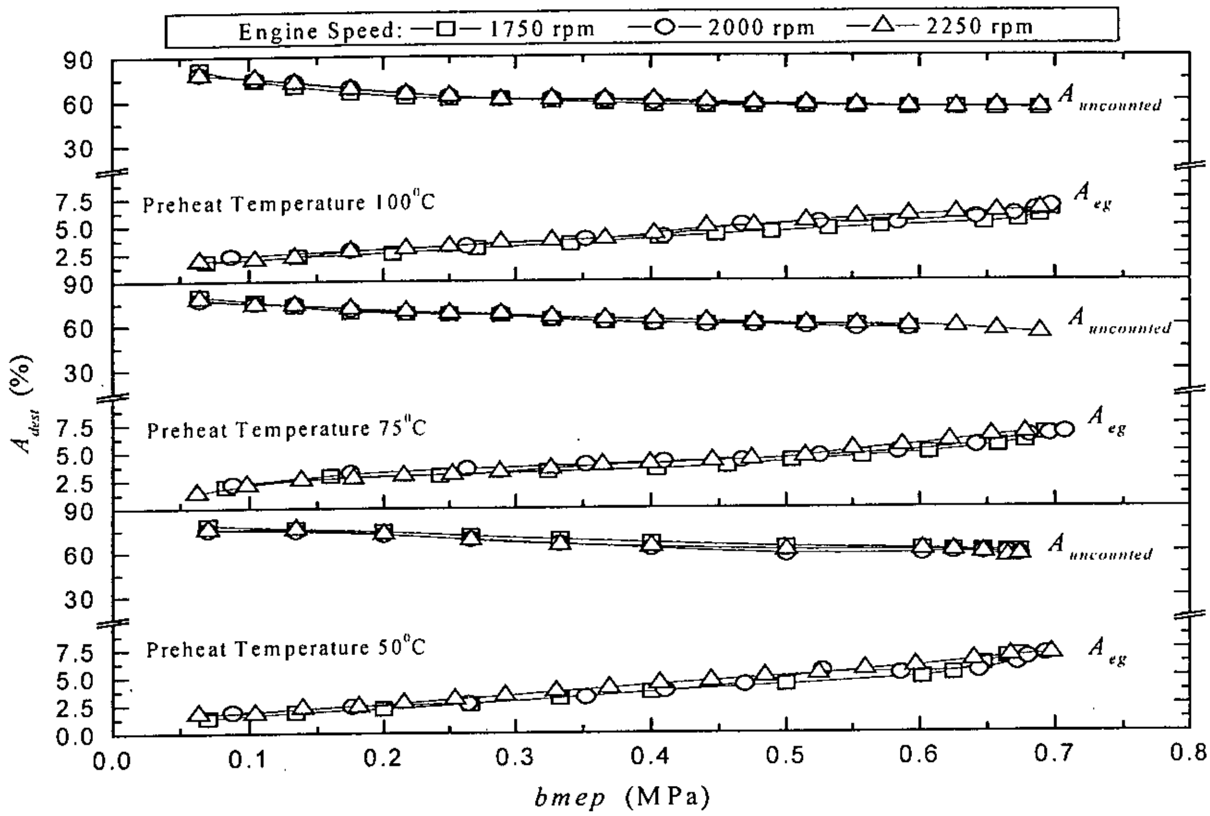

Fig. 6.15 Availability destruction as a function of brake mean effective pressure at different preheat temperatures run by soybean oil fuel

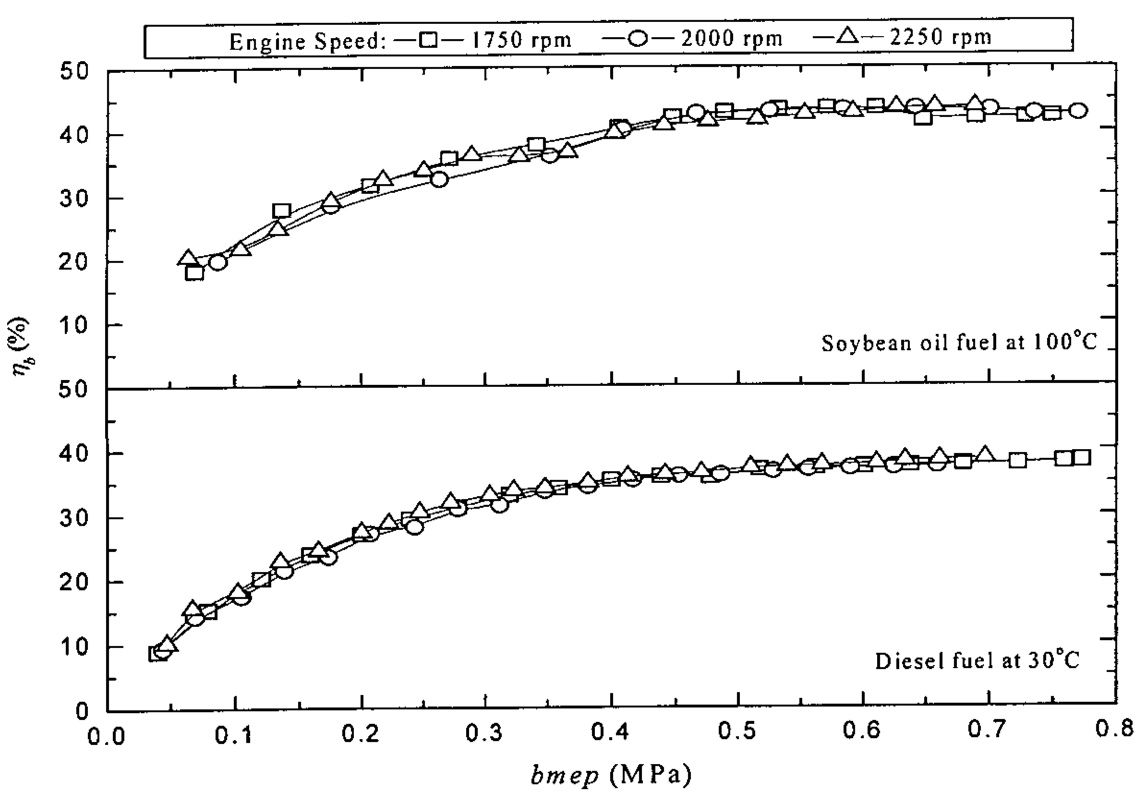

Fig. 6.16 Comparison of brake thermal efficiency as a function of brake mean effective pressure run by soybean oil fuel at 100°C and diesel fuel at 30±2°C

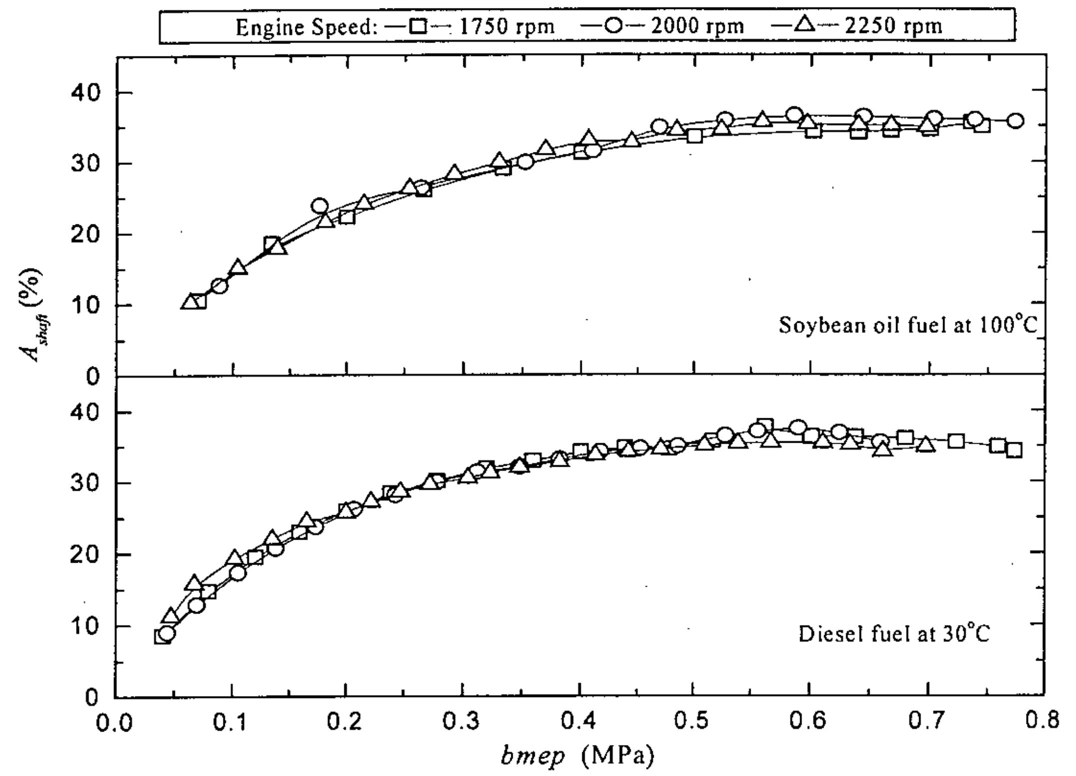

Fig. 6.17 Comparison of percent availability output at shaft as a function of brake mean effective pressure run by soybean oil fuel at 100°C and diesel fuel at 30±2°C

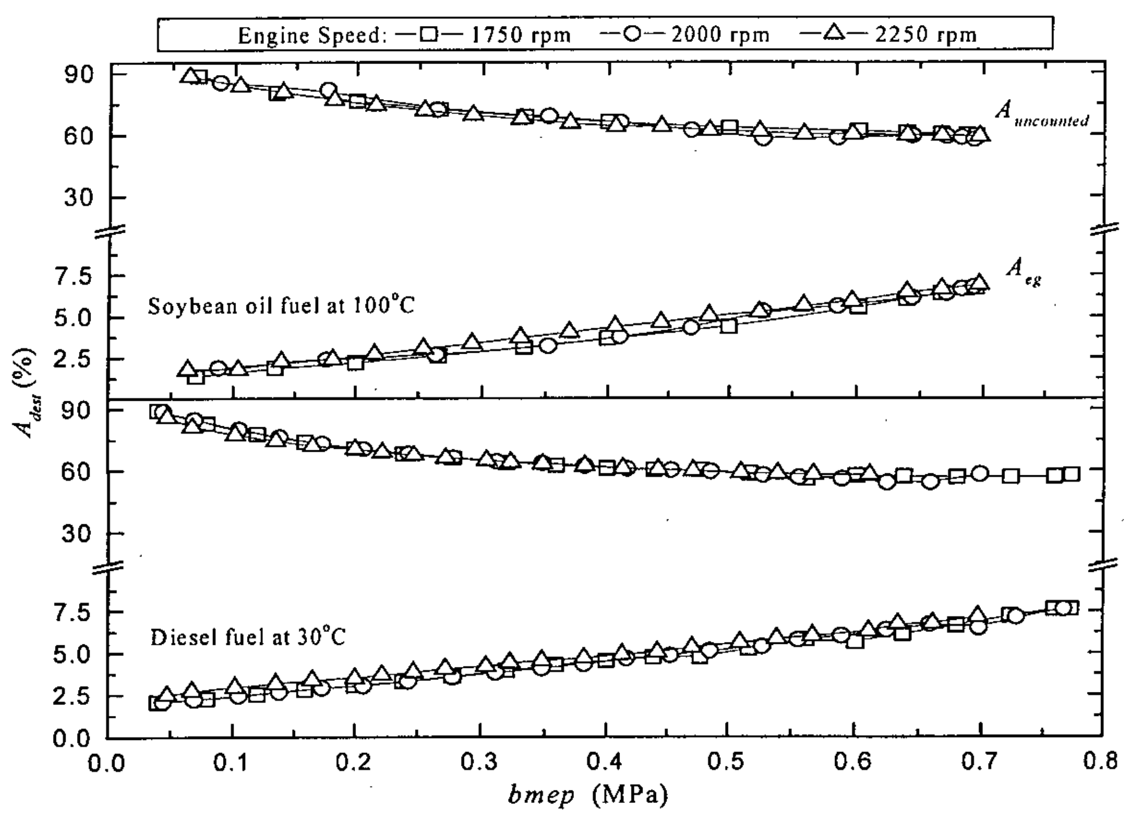

Fig. 6.18 Comparison of availability destruction as a function of brake mean effective pressure run by soybean oil fuel at 100°C and diesel fuel at 30±2°C

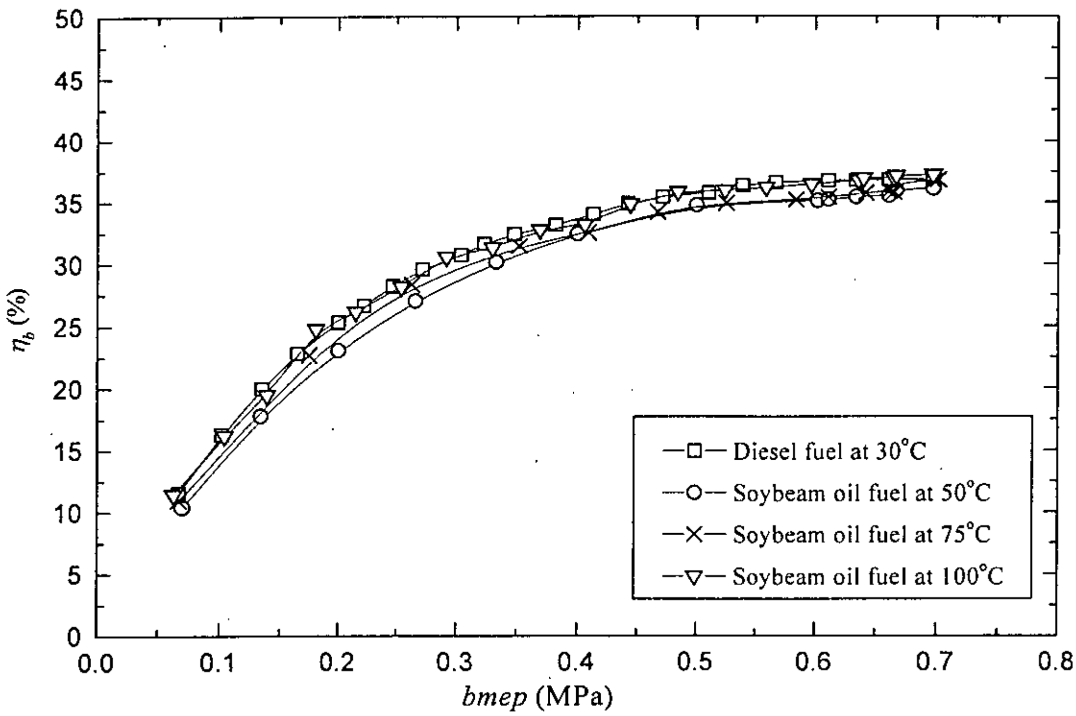

Fig. 6.19 Comparison of thermal efficiency as a function of brake mean effective pressure run by preheated soybean oil fuel and diesel fuel both at 2250 rpm

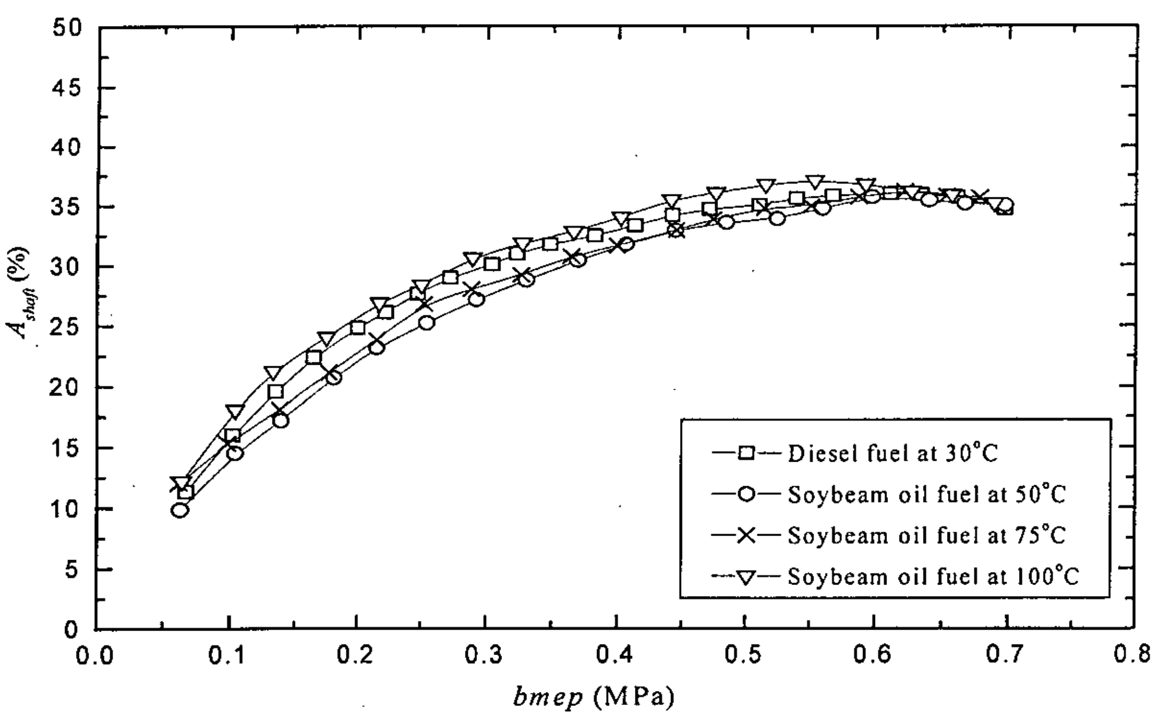

Fig. 6.20 Comparison of percent availability output at shaft as a function of brake mean effective pressure run by preheated soybean oil fuel and diesel fuel both at 2250 rpm

Appendix A

Engine Specifications

Model No.	:	1053 SU
Manufacturer and country of origin	:	VM Motor Diesel Engine, Italy
Cycle	:	Four stroke DI 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
Maximum power "F" (DIN 70020)	:	60 (44.1) hp(kW)
Continuous power "B" (DIN 6270)	:	56.25 (41.3) hp(kW)
Continuous power "A" (DIN 6270)	:	52.5 (38.6) hp(kW)
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
Engine weight with hand start	:	281 kg
Rotation of the crank shaft	:	counter clockwise (from flywheel side)

Appendix B

Dynamometer Specifications

Type	:	Water brake dynamometer
Model	:	TFJ 250L
Manufacturer	:	Tokyo Meter Corp.
Country of origin	:	Japan
Maximum brake power	:	250 hp
Power supply	:	AC 220/230V, 50Hz, 1-Ø
Revolutions at maximum braking hp point	:	2500 to 5000 rpm
Maximum revolutions	:	5500 rpm
Maximum braking torque	:	71.6 kg.m
Maximum braking water quantity	:	75 L/min
Weight	:	575 kg

Appendix C

Sample Calculation

Governing equations are presented first which is followed by a complete set of sample calculation with actual data.

C.1 Brake power output

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

$$P(\mathrm{hp}) = \frac{W \cdot N}{2500}$$
$$P(\mathrm{kW}) = 0.746 \cdot P(\mathrm{hp})$$

where

$P(\mathrm{hp})$	:	brake power output in hp
$P(\mathrm{kW})$	:	brake power output in kW
W	:	Load on the dynamometer in kg
N	:	speed of the shaft connected to the dynamometer in rpm

C.2 Fuel Consumption Rate

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

$$\dot{m}'_f = \frac{(\sigma \cdot \rho_w \cdot V)}{t} \times 60 \times 10^{-6}$$

where

$\dot{m}'_f$	:	fuel consumption rate in kg/h
V	:	volume of fuel flown in time t in ml
ρ_w	:	density water in kg/m^3
σ	:	specific gravity of the fuel
t	:	time to measure V ml fuel flow in minutes

C.3 Standardized Brake Power and Fuel Consumption Rate

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

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

where

P_b : adjusted brake power output in kW

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

α : power adjustment factor

and

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

where

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

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

β : fuel consumption adjustment factor

C.4 Air Flow Rate and A/F Ratio

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

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

where

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

C_d : co-efficient of discharge of the flow nozzle

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

ρ_{air} : density of air in kg/m^3

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

h_m : manometric deflection in m of H_2O

and

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

where

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

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

C.5 Volumetric Efficiency

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

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

where

η_v	:	volumetric efficiency in %
$\dot{m}_a$	:	mass of fresh air inducted per unit time in kg/h
N	:	engine speed in rpm
V_d	:	swept volume of all of the engine cylinders in m^3
ρ_i	:	density of inlet air at intake manifold in kg/m^3

C.6 Brake Thermal Efficiency

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

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

where

η_b	:	brake thermal efficiency in %
$\dot{m}_f$	:	adjusted fuel consumption rate in kg/h
P_b	:	adjusted brake power output in kW
LHV	:	lower heating value of the fuel in MJ/kg

C.7 Brake Specific Fuel Consumption

The brake specific fuel consumption is computed as follows:

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

where

$bsfc$	:	brake specific fuel consumption in kg/kWh
$\dot{m}_f$	:	adjusted fuel consumption rate in kg/h
P_b	:	adjusted brake power output in kW

C.8 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 soybean oil fuel,

$A_{in} = 1.0724\, Q_{in}$

where

A_{in} : availability input in kW

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

C.9 Percent Availability Output at Shaft

The fuel chemical availability or availability input 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

C.10 Availability Destructions

The sources of availability destruction 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

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

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

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

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

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

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

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

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

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

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

C.11 Sample Calculation

A set of real time calculation procedure of the above parameters are given as below:

Lab Condition:

Date of experiment		:	May 12, 2005
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}	:	28.5°C (301.5K)
Relative Humidity,	φ_x	:	75%

Derating Factors†:

Power adjustment factor,	α	:	0.9891
Fuel consumption adjustment factor,	β	:	1.00326

† From appendix D

Experimental Observations and Data:

Experimental constants:

Diameter of the flow nozzle,	D_n	:	30mm
Cross-sectional area of the flow nozzle,	A_n	:	7.065×10^{-4} m^2
Coefficient of discharge of the flow nozzle,	C_D	:	0.98
Specific gravity of diesel,	σ	:	0.845
Lower heating value of diesel,	LHV	:	42.94 MJ/kg
Density of manometric fluid (water) at 32°C,	ρ_w	:	996 kg/m^3
Density of air at 32°C,	ρ_w	:	1.129 kg/m^3

Experimental Data:

Load on dynamometer,	W	:	45.0 kg
Speed of the dynamometer,	N	:	2252 rpm
Volume of diesel collected at time t,	V	:	100 ml
Time of collection of V volume diesel,	t	:	43.56 s (0.726 min.)
Manometric deflection,	h_m	:	51 mm of H_2O

Calculation:

Engine brake power output in hp,

$$P(\text{hp})=\frac{45.0\times 2252}{2500}=40.54\,\text{hp}$$

Engine brake power output in kW,

$$P(\text{kW})=0.746\times 40.54=30.243\,\text{kW}$$

Fuel consumption rate,

$$\dot{m}'_f=\frac{(0.845\times 1000)\cdot(100)}{0.726}\times 60\times 10^{-6}=6.9835\,\text{kg/h}$$

Adjusted brake power output,

$$P_b=\frac{30.243}{0.9891}=30.58\,\text{kW}$$

Adjusted fuel consumption rate,

$$\dot{m}_f=\frac{6.9835}{1.00326}=6.96\,\text{kg/h}$$

Brake specific fuel consumption,

$$bsfc = \frac{6.96}{30.58} = 0.23\,\text{kg/kWh}$$

Brake thermal efficiency,

$$\eta_b = \left(\frac{3.6\times 30.58}{6.96\times 42.94}\right)\times 100 = 36.84\%$$

Air flow rate,

$$\dot{m}_a = 2\times 3600\times\frac{0.98\times 7.065\times 10^{-4}}{1.129}\sqrt{2\times 9.81\times 0.051\times 996}$$
$$= 167.62\,\text{kg/h}$$

Air-fuel ratio,

$$A/F = \frac{167.62}{6.96} = 24$$

Volumetric efficiency,

$$\eta_v = \left(\frac{2\times 167.62}{2252\times 2.875\times 10^{-3}\times 1.165}\right)\left(\frac{1}{60}\right) = 76.92\%$$

Availability input, (or Fuel chemical availability)

$$A_{in} = 1.06489\cdot\left(\frac{6.96\times 42.94\times 10}{36}\right) = 88.40\,\text{kW}$$

Percent availability output at shaft,

$$A_{shaft} = \frac{30.58}{88.40}\times 100 = 34.59\%$$

Availability destroyed in exhaust gases,

$$A_{exst} = \begin{bmatrix} 1.266246\times 6.96\cdot(1+24)\cdot(728-305)+ \\ 6.96\cdot(1+24)\{1.266246\ln(305/728)\} \end{bmatrix} = 9.682334$$

where

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

Percent availability destroyed in exhaust gases,

$$A_{eg} = \frac{9.682334}{85.815}\times 100 = 11.283\%$$

Percent availability destroyed in friction, cooling, etc,

$$A_{uncounted} = [100-(34.59+11.283)] = 54.127\%$$

Appendix D

Engine Derating

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

D.1 The standard Reference Conditions:

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

Total barometric pressure, $P_r = 100$ KPa

Air temperature, $T_r = 300$K (27°C)

Relative Humidity, $\varphi_r = 60$ %

Charge air coolant temperature, $T_{cr} = 300$K (27°C)

Mechanical efficiency of engine $\eta_m = 0.80$ (according to the clause 10, note 4 of the BS 5514 : Part 1: 1982)

D.2 Derating Calculations

Here is given an example to show the derating of the engine running at 2250 rpm with diesel fuel.

The data was taken on the date of May 12, 2006 and at the following lab condition:

Barometric pressure at lab, $P_x = 98.93$ kPa (741.5 mm of Hg)

Dry bulb temperature, $T_{db} = 32$°C (305K)

Wet bulb temperature, $T_{wb} = 28.5$°C (301.5K)

Relative humidity, $\varphi_x = 75\%$

From Annex F, BS 5514:

For relative humidity, φ_x = 75% and air temperature T_{db} = 32°C (305 K), by interpolation, the water vapor pressure $\varphi_x p_{sx} = 3.55$

From Annex E, BS 5514:

For water vapor pressure of $\varphi_x p_{sx} = 3.48$ and barometric pressure in lab $p_x = 98.93$ kPa by interpolation we get the dry air pressure ratio = 0.97464

From Annex D, BS 5514:

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

where R_1 = dry air pressure ratio
$R_2 = T_r/T_x$
$R_3 = T_{cr}/T_{cx}$

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

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

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

$R_1 = P_x/P_r = 98.93/100 = 0.9893$ and $m = 1$. So, $(R_1)^m = (0.9893)^1 = 0.9893$
$R_2 = T_r/T_x = 300/305 = 0.984$ and $n = 0.75$. So, $(R_2)^n = (0.984)^{0.75} = 0.988$

Therefore, $k = 0.988 \times 0.9893 \times 1 = 0.9774$

From Annex C, BS 5514:

The fuel consumption adjustment factor (β) for k= 0.9774 and $\eta_m = 0.85$ is found by interpolation as:

$\underline{\beta = 1.00326}$

From Annex B, BS 5514:

The power adjustment factor (α) for k = 0.9774 and η_m = 0.85 is found by interpolation as:

$\underline{\alpha = 0.9891}$

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

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

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

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