Experimental Investigation of Dual-Fuel Diesel Engine

By

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It is hereby declared that this thesis or any part of it has not been submitted elsewhere for the award of any degree or diploma.

Md. Habibur Rahman

To

My mother

who inspired me to become an engineer

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A	:	Air-Fuel ratio on mass basis
BDC	:	Bottom dead centre
BG	:	Bio gas
BG30	:	Bio gas containing 30% CO_2 and 70% CH_4 by volume
BG50	:	Bio gas containing 50% CO_2 and 50% CH_4 by volume
bmep	:	Brake mean effective pressure
bsfc	:	Brake specific fuel consumption
DI	:	Direct injection
N	:	Engine revolution per minute
NG	:	Natural gas
P_{max}	:	Maximum gas pressure
P_b	:	Brake power
Q	:	Heat input
S	:	Stroke length
TDC	:	Top dead centre
V_d	:	Displacement volume
$\overline{V}_p$	:	Mean piston speed
η_b	:	Brake efficiency
η_v	:	Volumetric efficiency
ϕ	:	Equivalence ratio
ρ_i	:	Inlet air density

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Abstract

Spreading concerns over problems related to energy, economy and environment have prompted a tremendous burst of research activities in the field of combustion of both conventional and alternative fuels. Among them, natural gas and biogas are the potential substitution to conventional fuels in our country.

Natural gas is cheap in Bangladesh and biogas is renewable in nature. Energy density of natural gas premixture is comparable to those of the diesel values; however energy density of biogas premixture is low as compared to diesel fuel. These gaseous fuels can be introduced in stationary diesel engine by fumigation method with minor physical modification of the engine.

In the present study, performance studies are carried out in a 4stroke 4-cylinder diesel engine using natural gas and biogas as the main source of energy. A fixed amount of diesel is used as a pilot fuel to ignite the mixture. Synthetic biogas is prepared by mixing CH_4 from main supply and CO_2 from the cylinder. Performance studies are carried out at four different engine speeds at different loading conditions. Various performance parameters are analyzed and reported. Engine performances are found to be not deviating from its diesel performance when these fuels are used.

Chapter 1

Introduction

Compression ignition (CI) engines, also known as Diesel engines, are widely used in power generation, transportation and irrigation purposes. These operate on the constant pressure diesel cycle or on dual cycle and generally use liquid fuels of low volatility varying from low-grade distillates to crude oils. In CI engines no ignition devices are used; load and speed are controlled by varying the amount of fuel injection. These have the advantages of low specific fuel consumption, ability to maintain economy and thermal efficiency at part loads, low fuel cost and practically no carbon-monoxide except at near full load or over-load conditions. However, the last three decades have witnessed growing concern over environmental impact and/or exhaustion of conventional fossil fuels. Gaseous alternative fuels, such as natural gas and biogas, can be used in diesel engine in dual fuel mode to substitute significantly the diesel consumption in CI engines.

Existing stationary diesel engines can be retrofitted fairly easily for operation with alternative gaseous fuels, such as natural gas and biogas. Natural gas is now being widely used to fuel combustion engines; however its reserve is limited in many areas of the world. Hence, biogas is a potential alternative fuel that is renewable in nature and thereby does not contribute to the net atmospheric concentration of the green house gas carbon dioxide. It is a colorless combustible gas produced by fermentation of cellulose material (woody and non-woody), manure or cow-dung. Methane, CH_4 and carbon di-oxide, CO_2, constitute about 95% of biogas and rests are trace organic / non-organic elements of variable compositions. Moreover, the amount of methane and carbon-dioxide in bio-gas varies with different sources of origin (Goodger 1980). The variation in its composition affects combustion process and the heat release rates (Haq and Mizanuzzaman 2002). Fortunately, methane is the simplest of the hydrocarbon fuel series and modeled combustion data is very consistent with the experimental results (Gu *et al.* 2000).

1.1 Scope of the Thesis

The present thesis reports the experimental study of diesel engines in dual fuel modes using natural gas and bio-gas as alternative fuels to diesel fuel. Experiments are carried out in a 4-cylinder, 4-stroke, water cooled Kubota Diesel engine at different speeds for various loading conditions. A fixed amount of diesel as a pilot fuel is used to ignite the mixture. Engine loads are varied by varying the amount of gaseous fuel varied engine loads. Natural gas and bio-gas of two compositions (70% methane and 30% CO_2, and 50% methane and 50% CO_2 on a volumetric basis) are introduced into the cylinder with the incoming air by fumigation method. Various engine performance parameters are obtained and reported.

A brief review of the CI engines, combustion in CI engines and alternative fuels of CI engines are presented in chapter 2. A brief introduction of natural gas and bio-gas as alternative CI engines are also presented in chapter 2. Experimental procedures, setup and performance parameters of CI engines are shown in chapter 3. Finally the results and discussions are exhibited in chapter 4. This is followed by conclusions of the study and the recommendations for further study.

Chapter 2
Literature review

An internal combustion engine (IC engine) is the most remarkable achievement of the last 100 years. The distinctive feature of IC engine is that combustion and conversion of heat energy into mechanical work occur inside a cylinder. Gaseous and liquid are commonly used and the oxygen required for combustion is taken from the atmospheric air. The force on the piston head is exerted by the products of combustion to produce mechanical work. These engines are noted for their high overall efficiency and low operating cost. Thereby the diesel engine has become the dominant type for heavy-duty applications for on-road, off-road, marine and industrial applications.

Diesel and gasoline engines have many similarities and there is little difference in their external appearance. There are differences in their construction, however, which are made necessary by the following three fundamental differences in operation. The SI engine burns an essentially homogeneous fuel-air mixture achieved generally by carburetion, and the CI engine burns a heterogeneous mixture, since liquid fuel is injected late on the compression stroke. Air-fuel mixtures are ignited by an electrical spark in gasoline engines but compression temperatures in diesel engines must be high enough to self ignite the air-fuel mixture. The speed control of a gasoline engine is achieved by throttling the air-fuel mixture. In diesel engine, only the fuel-flow is controlled, air is not throttled.

Gasoline engines are limited in efficiency by fuel octane number, the ability to propagate flames in lean mixtures, cycle-to-cycle variability and fuel trapped in crevices. These restrictions are partly overcome by direct injection of gasoline, but direct-injection stratified-charge spark-ignited engines have yet to be accepted as an alternative to homogenous spark ignited engines.

The importance of the CI engine arises because of its high thermal efficiency at either full or part load, and these are friendlier to the environment. The improved efficiency is caused by the relatively high compression ratios, low pumping losses due to lack of throttling, and the overall lean mixtures, all contributing to increase thermal efficiency. Unburned hydrocarbon emission problems are also significantly reduced because of the fact that the crevice volumes contain air or products rather than unburned fuel mixture. The efficiency of the diesel engine declines at part load, but the efficiency of the gasoline engine declines faster. Presented in Table 2.1 are the best thermal efficiency estimates of various types of internal combustion engines (Borman 1998).

Table 2.1. Best Thermal Efficiency Estimates for Various Internal Combustion Engines.

Combustion Engine Type	Efficiency (%)
Spark ignited, port injected stoichiometric	31.5
Direct injected, spark ignited, stoichiometric	33.0
Directed injected, spark ignited, lean injection	34.5
Indirect injected diesel	35.5
Direct injected, spark ignited, late injection	38
Gas turbine	38
High speed, direct injected diesel	43
Heavy duty, direct injected, diesel (HDDI)	46
Turbo compounded HDDI diesel	54

2.1 Combustion in CI Engines

In CI engine, air alone is introduced into the cylinder. The fuel is injected directly into the engine cylinder just before the combustion process. Load control is achieved by varying the amount of fuel injected in each cycle - the air flow at a given engine speed is essentially unchanged. There are a great variety of CI engine designs in use in a wide range of applications - automobile, truck, locomotive, marine, power generation. Naturally aspirated engines where atmospheric air is inducted, turbo-charged engines where the inlet air is compressed by an exhaust driven turbine compressor combination, and super-charged engines where the air is compressed by a mechanically driven pump or blower are common. Turbo-charging and super charging increase output by increasing the air mass flow per unit displaced volume, thereby allowing an increased fuel flow. These method are used, usually in larger engines to reduce engine size and weight for a given power output. Except in smaller engine sizes, the two-stroke cycle is competitive with the four-stroke cycle, in large part because, with the diesel cycle, only air is lost in the cylinder scavenging process.

The fuel is introduced into the combustion chamber of a diesel engine through one or more nozzles or orifices with a large pressure differential between the fuel supply line and the cylinder. Standard diesel injectors usually operate with fuel injection pressures between 20 and 170 MPa. At the time of injection, the air in the cylinder has a pressure of 5 to 10 MPa, temperature of about 1000 K, and a density between 15 and 25 Kg/m^3. Nozzle diameters cover a range of 0.2 to 1 mm diameter with length to diameter ratios from 2 to 8. Typical distillate diesel fuel properties are: relative specific gravity 0.8, viscosity between 3 and 10 Kg/m.s and surface tension about 3.10^{-2} N/m at 300 K (Heywood 1988).

The jet disintegrates into a core of fuel surrounded by a spray envelope of air and fuel particles. This latter zone is created both by atomization and vaporization of the fuel and the turbulence of the air in the combustion chamber passing across the jet and stripping the fuel particles from the core. The time between the beginning of injection and the attainment of chemical-reaction conditions is known physical delay. In the physical delay period, the fuel is atomized, vaporized, mixed with air and raised in temperature. In the next stage, called the chemical delay, reaction starts slowly and then accelerates until ignition takes place. At some locations or at many locations flame appears, but rather than an orderly propagation of flame along a definite flame front, entire areas may explode or burn because of the accumulation of fuel in the chamber during the delay period. Mixture of CI engine is heterogeneous, thereby flame will propagate if the region of mixture is continuous. Adjacent regions, however, on the verge of self-ignition may ignite from heat transferred from the burning region. In any event it would be difficult to distinguish between flame propagation and self-ignition, which is aided.

The term ignition delay is assigned to the time consumed by both the physical and chemical delays. For light fuel, the physical delay is small, while for heavy, viscous fuels, the physical delay may be the controlling factor. The physical delay is greatly reduced by using high injection pressures and high turbulence to facilitate breakup of the jet. In most CI engines, the ignition delay is shorter than the duration of injection. Then the combustion period can be considered to be divided into four stages (Obert 1973):

1. Ignition delay,
2. Rapid pressure rise (probable premixed flame),
3. Controlled pressure rise (probable diffusion flame),
4. Burning on the expansion stroke

Rapid pressure rise in the engine cylinder occurs because of myriad ignition points and the accumulation of fuel in the delay period. Following this stage, the final portions of the fuel are injected into flame and consequently combustion of this portion is somewhat regulated by the injection rate. Since the process is far from being homogeneous, combustion continues when the expansion stroke is well under way. Under normal diesel engine conditions, 70 to 90% of the injected fuel is in the vapor phase at the start of combustion. Evaporation is more than 90% complete after 1 ms. However only 10 to 35% of the vaporized fuel has mixed to within flammability limits in a typical medium speed DI diesel engine. Thus combustion is largely mixing limited, rather than evaporation limited (Heywood 1988)

2.2 Alternative Fuels for CI Engines

The last two decades have witnessed growing concern over the environmental impact and / or exhaustion of conventional fossil fuel energy sources in an environment that continuously increases its demand on liquid fuels. These concerns have highlighted the need for diversification and prompted increased research into potential alternative sources of fuel energy for IC engines. Combustion scientists and engineers are now faced with the problem of simultaneously achieving maximum energy efficiency and minimum emission of pollutants using alternative fuels and solution to these problems requires designs of combustion system.

Alternative fuels are of interest since they can be refined from renewable feed stocks and their emission levels can be much lower than those of gasoline and diesel fuels in engines. One feature of alternative source fuel energy is that they are well suited for decentralized production (e.g. bio-gas, vegetable oil, alcohols) to meet the needs for social and economic progress, especially in rural communities where the supply of conventional fossil fuels may be expensive. Another fact that has gained importance in recent years is the reduction of pollutant emissions by introducing alternative fuels. However, alternative fuels are not currently widely used in vehicular applications for economic and engineering reasons. The cost of alternative fuels per unit of energy delivered can be greater than gasoline or diesel fuels and the energy density of alternative fuels by volume is less than gasoline or diesel fuels. The smaller volumetric energy density requires larger fuel storage volumes to have the same driving range as gasoline fuel vehicles. This can be a drawback particularly with dual fuel vehicles where a significant portion of the trunk space is used by the alternative fuel storage tank. Alternative fuels also lack a wide scale distribution and fueling infrastructure comparable to that of conventional fuels.

Existing gasoline or diesel engines can be retrofitted fairly easily for operation with alternative fuels. However various operational considerations need to be taken into account. The different combustion characteristics of alternative fuels may require a change in the injection timing. Also many alternative fuels, especially those in gaseous form, have very low lubricity to cause increased wear of fuel components such as fuel injectors and valves.

The most important alternative fuels suitable for engine uses and covered by worldwide research and testing are:

- Alcohol's (Methanol CH_3OH and ethanol C_2H_5OH)
- Propane
- Vegetable oils and vegetable oil methyl Easters.
- Hydrogen
- Natural gas (Methane CH_4 and other gases)
- Bio-gas (Methane and other gases with carbon-dioxide contents in Bio-gas being upped 40%)

With the exception of vegetable oils all the above alternatives are suitable both for SI and CI engines. The exhaust emission characteristics of alternative fuels not only depend on engine consideration but to a decisive degree also on the chemical and physical properties of the fuels. Combustion of alternative fuels made up of hydrocarbons produces the known pollutants spectrum including carbon-dioxide. The higher the C content of the fuel the higher are the carbon-dioxide emissions under equal conditions.

2.2.1 Methanol

Methanol (CH_3OH) is an alcohol fuel formed from natural gas, coal, or biomass feed stock. Methanol is also called 'wood alcohol'. It is liquid at ambient conditions, miscible in water, and has a relatively low vapor pressure. Since oxygen is part of the chemical structure, less air is required for complete combustion. Methanol is toxic, and ingestion can cause blindness and death. Methanol has been used as a vehicular fuel since the early 1900s, and also used as a fuel for diesel engines and fuel cells. Pure methanol is labeled M 100, and a mixture of 85% methanol and 15% gasoline is labeled M 85. M 85 has an octane rating of 102. Adding gasoline to methanol provides more volatile components that can vaporize more easily at low temperatures. Methanol has been adopted as a racing fuel, both for performance and safety reasons. Since methanol mixes with water, a methanol fire can be extinguished with water, which is not the case with gasoline. The octane rating of methanol of 111 (RON) allows use of an increased compression ratio. The relatively high enthalpy of evaporation (1215 KJ/Kg) of methanol relative to gasoline (310 KJ/Kg) produces greater intake air -cooling and a corresponding increase in volumetric efficiency relative to gasoline. The cetane number of methanol is low at about 5, but it can be used in compression ignition engines with diesel fuel pilot ignition. Methanol burns with a nearly invisible flame, and a relatively high flame speed. Formaldehyde is a significant decomposition product from methanol combustion and is expected to be higher from methanol than other fuels (Ferguson and Kirkpatrick 2001).

2.2.2 Ethanol

Ethanol (C_2H_5OH) is an alcohol fuel formed from the fermentation of sugar and grain stocks primarily sugars cane and corn, which are renewable energy sources. Its properties and combustion characteristics are very similar to those of methanol. Ethanol is also called 'grain alcohol'. It is liquid at ambient conditions and non toxic at low concentrations. Gasohol E10 is a gasoline ethanol blend with about 10% ethanol by volume. E85 is a blend of 85% ethanol and 15% gasoline. In Brazil, about half of the vehicles use ethanol based fuel 'alcool' primarily E93, produced from sugar cane. The energy density by volume of ethanol is relatively high for an alternative fuel, about 2/3 that of the gasoline. The octane rating of ethanol of 111(RON) allows an increased compression ratio. The cetane number of ethanol is low at about 8 but like methanol it can be used in CI engines with diesel fuel pilot ignitions.

2.2.3 Propane

Propane (C_3H_8) is a saturated paraffin hydrocarbon. When blended with butane (C_4H_{10}) or ethane (C_2H_6), it is also designated as liquefied petroleum gas (LPG). A common LPG blend is P92, which is 92% propane and 8% butane. In the United States, about one half of the LPG supply is obtained from the lighter hydrocarbon fractions produced during crude oil refining, and the other half from heavier components of wellhead natural gas.

Propane has been used as a vehicular fuel since the 1930s. In 1993, there were about 4 million LPG vehicles operating worldwide, with the majority in the Netherlands, followed by Italy, the United States, and Canada. There are a number of original equipment manufacturers that currently sell propane fuelled vehicles, primarily light and medium duty fleet vehicles, such as pick-up trucks and vans. Conversion kits are available to convert gasoline or diesel fueled engines to dedicated propane or duel fuel use.

In vehicles, propane is stored as a compressed liquid, typically from 0.9 to 1.4 MPa. Its evaporative emissions are essentially zero, since it is used in a sealed system. A pressure regulator controls the supply of propane to the engine, and converts the liquid propane to a gas through a throttling process. Propane gas can be injected into the intake manifold, into the ports, or directly into the cylinder. Propane has an octane number of 112 (RON), so vehicular applications of propane will generally raise the compression ratio.

2.2.4 Vegetable oils and their Easters

A wide variety of vegetable oils may be considered for potential use as fuels in diesel engines (Haq 1994). In addition to soybean oil, peanut oil, coconut oil, palm oil etc, rape seed oil is discussed as a particularly important raw material. Pure, untreated oils can cause malfunction in diesel engine. As their viscosity is fairly high, hence atomization is poor. These result incomplete combustion and carbon deposit formation in the injection nozzles. The polymerization tendency of these oils accelerates resin and soot formation. Suitable measures must therefore be introduced to prevent such processes from occurring when vegetable oils are used as fuels. Transesterification of the oil is one method of overcoming such problems. As a result glycerin-water and estar-alcohol mixtures are formed as soon as a certain temperature level is attained and when a catalytic converter is present after separating and distilling the excess alcohol this process yields the ester usable or combustion in engine. This process is suitable e.g for producing rape seed oil, rape seed methyl ester (RME) suitable as diesel fuel from raw rape seed oil. RME will be investigated here in greater detail as it is typical for a variety of vegetable oil derivatives (Schäfer and Van Basshuysen 1995).

Alternative vegetable oils have a higher cost, and lower volumetric energy density than diesel fuel, but do produce lower exhaust emissions. It is reported that RME had about 40% lower HC emissions, 35% lower CO, 35% lower PM, but about 15% greater NOx emissions (Ferguson and Kirkpatrick 2001).

2.2.4 Hydrogen

Hydrogen as a CI engine Fuel In recent years, Hydrogen as an energy source with a potential to solve energy shortage and environmental problems simultaneously is attracting attention. Hydrogen becomes water when it is burnt, and is a clean fuel because it does not produce any air pollution substances other than nitric oxides generated by combustion heat. However it is expensive and requires special safety measures for storage, it requires further research and development.

Hydrogen (H_2) can be produced from many different feed stocks, including natural gas, coal, biomass, and water. The production process include steam reforming of natural gas, presently the most economical method, electrolysis of water, and gasification of coal, which also produces CO. Hydrogen is colorless, odorless, and non-toxic, and hydrogen flames are invisible and smokeless. The global warming potential of hydrogen is insignificant in comparison to HC based fuels since combustion of hydrogen produces no carbon based compounds such as HC, CO, and CO_2.

At present the largest user of Hydrogen fuel is the aerospace community for rocket fuel. Hydrogen can also be used as a fuel in fuel cells. There have been a number of vehicular demonstration projects, but the relatively high cost of hydrogen fuel has hindered adoption as an alternative fuel. Dual fuel engines have been used with hydrogen, in which hydrogen is used at start up and low load, to reduce the cold start emissions levels (Ferguson and Kirkpatrick 2001).

One of the major obstacles related to the use of hydrogen fuel is the lack of any manufacturing, distribution, and storage infrastructure. The most economical would be to distribute hydrogen through pipelines, similar to natural gas distribution. The three methods used to store hydrogen fuel are:

1. In a liquid form at -253° C in cryogenic containers,
2. As a metal hydride or
3. In a pressurized gaseous form at 20 to 70 MPa.

Compressed hydrogen at 70 MPa has one third the energy density by volume of compressed natural gas, and liquid hydrogen has one fourth the energy density by volume of gasoline. If mixed with air in the intake manifold, the volume of hydrogen is about 30% of the intake mixture volume at stoichiometric, decreasing the volumetric efficiency. The octane rating of hydrogen of 106 (RON) allows use of an increased compression ratio.

2.3 Natural Gas and Biogas as Alternative CI Engine Fuel.

The mass production of automobiles, which began early in the 1910s, resulted in greater demand for petroleum fuels. The growth in internal combustion (IC) engines and the allied use of petroleum fuels resulted in increasing atmospheric pollution. Furthermore, as it has become strongly recognized that the petroleum resources will become scarce in the near future, there are urgent needs of establishing the counter measure, conserving petroleum products through systems design, developing alternative fuels and new energy sources. National interest in developing alternative fuels for IC engines continues to be strong due to environmental concerns and / or the given uncertainties involving the future availability of fossil fuel. Among the alternative fuels, natural gas and bio-gas are promising substitutes to the petroleum based fuel in Bangladesh's context.

2.3.1 Natural Gas

Natural gas is available in many parts of the world and in a high state of purity. CI engines are used as dual fuel engines and have the advantage of providing high compression ratio, which is possible with natural gas due to their high octane numbers. The dual fuel engine can be operated on either a mixture of a small quantity pilot fuel oil and gas, which constitutes the larger percentage of fuel, or fuel oil alone. The clean burning of natural gas does not create sludge and dilution of the lubricating oil, thus reducing the number of oil changes needed. It leaves spark plug and fuel injector's carbon free for better performance.

Means for the utilization of natural gas in SI engines are well established whilst developments efforts are still geared towards the use in CI engines. In gas fueled dual-fuel CI engines, it requires a pilot amount of diesel to be injected into the cylinder to cause ignition and the gas is mixed outside the cylinder before its induction into the cylinder (Bari 1986). The combustion process of dual-fuel engine lies between that of the SI engines and CI engines.

Natural gas is a naturally occurring fuel found in oil fields. It is primarily composed of about 90 to 95% methane (CH_4), with small amounts of additional compounds such as 0-4% nitrogen, 4% ethane, and 1 to 2% propane. Methane is a greenhouse gas, with a global warming potential approximately ten times that of carbon di-oxide. Methane has a lower carbon to hydrogen ratio relative to gasoline, so its CO_2 emissions are about 22 to 25% lower than gasoline.

Natural gas has been used for many years in stationary engines for gas compression and electric power generation. An extensive distribution network of natural gas pipe lines exists to meet the need for natural gas for industrial processes and heating applications. Natural gas fueled vehicles (NGV) have been in use since the 1950s, and conversion kits are essential for both spark and compression ignition engines. Recent research and development work has included of dual-fuel vehicles that can operate either with natural gas and gasoline or diesel fuel.

Natural gas is stored in a compressed (CNG) state at room temperature and also in a liquid form at -160° C. Natural gas has an octane number (RON) of about 127, so that natural gas engines can operate at a compression ratio of 11:1, greater than gasoline fueled engines. Natural gas is pressurized to 20 MPa in vehicular storage tanks, so that it has about one third of the volumetric energy density of gasoline. The storage pressure is about 20 times that of propane. Like propane, natural gas is delivered to the engine through a pressure regulator, either through a mixing valve located in the intake manifold, port fuel injection at about 750 KPa, or direct injection into the cylinder. With intake manifold mixing or port fuel injection, the engine's volumetric efficiency and power is reduced due to the displacement of about 10% of the intake air by the natural gas, and the loss of evaporative charge cooling. Natural gas does not require mixture enrichment for cold starting, reducing the cold start HC and CO emissions.

The combustion of methane is different from that of liquid hydrocarbon combustion since only carbon-hydrogen bonds are involved, and no carbon- carbon bonds, so the combustion process is more likely to be more complete, producing less non methane hydrocarbons. Optimal thermal efficiency occurs at lean conditions at equivalence ratios of 0.67 to 0.8. The total HC emission levels can be higher than gasoline engines due to unburned methane. The combustion process of methane can produce more complex molecules, such as formaldehyde, a pollutant. The particulate emissions of natural gas are very low relative to diesel fuel. Natural gas has a lower adiabatic flame temperature (approx. 2240 K) than gasoline (approx. 2310 K), due to its higher product water content. Operation under lean conditions will also lower the peak combustion temperature. The lower combustion temperatures lower the NO formation rate, and produce less engine output NOx.

Natural gas can replace diesel fuel in heavy duty engines with the addition of a spark ignition system. A number of heavy duty diesel engine manufacturers are also producing dedicated natural gas heavy duty engines. The natural gas fueled engines are operated lean with an equivalence ratio as low as 0.7. The resulting lower in cylinder temperatures reduce the NOx levels. Heavy duty natural gas engines are designed to meet LEV emissions standards without the use of an exhaust catalyst.

Natural gas can also be used in compression ignition engines if diesel fuel is used as a pilot fuel, since the auto ignition temperature of methane is 540° C, compared to 260° C for diesel fuel. This fueling strategy is attractive for heavy duty diesel applications, such as trucks, buses, locomotives, and ships compressors, and generators. These engines are also operated with a lean combustion mixture, so that the NOx emissions are decreased. However diesel engines are un-throttled, at low loads, the lean combustion conditions degrade the combustion process, increasing the HC and CO.

2.3.2 Bio-gas

Bio-gas is a very attractive alternative fuel candidate that is easily produced by fermentation of cellulose materials, manure and cow-dung. It contains 30-60% methane by volume, and together with CO_2 it constitutes 95% by the total volume and rest are the trace organic/non-organic elements of variable composition depending on the source of origin. Corrosive products of combustion are usually generated and these can result in corrosion of various components of the IC engines. Heat release from bio-gas is much lower than the hydrocarbon fuels because of its high CO_2 content and the reduction of power from a diesel engine is reported when diesel is partially substituted by bio-gas in a dual-fuel mode, and bio-gas containing more than 50% CO_2 is found to be practically non-feasible in internal combustion engines because of its very low energy density. The CO_2 can be removed from it which involves additional costs to make it economically non feasible.

Bio-gas has the advantage of being locally produced, cheap and renewable in nature, and thereby not contributing to the net atmospheric concentration of green house gas carbon dioxide (Goodger 1980). It is a colorless gas that is produced by fermentation of cellulose materials, manure and cow-dung. Methane and CO_2 constitute about 95% of bio-gas and rests of them are trace organic/non organic elements of variable composition that depends on the source of origin. Presented in Table 2.2 are the yield and methane contents of bio-gas originated from different common sources in Bangladesh. The production of bio- gas and its utilization is increasing day by day.

Table 2.2. Biogas production rates of some fermentation materials of Bangladesh and their main components.

Material	Yield of bio gas (m^3/Kg TS)	Methane Content (%)
Chicken excreta	0.58 – 0.60	60 – 65
Cowdung	0.26 – 0.28	68-75
Human excreta	0.3	52 – 76
Rice husk	-	52 - 76
Leaves	0.210 – 0.294	58
Wheat straw	0.432	59
Sludge	0.640	50
Green grass and weeds	-	53 – 65
Water hyacinth	0.35 – 0.42	-

Note: TS = Total Solid

Source: Project report on Biogas, LGED, 1994

In an experimental study, Bari *et al* (1994) reported the reduction of available power from a diesel engine when bio-gas was used to substitute the diesel fuel. In a recent analysis Mizanuzzaman (2001) supported this experimental result by quantifying the effects of carbon-di-oxide concentration bio-gas to reduce the energy density of the charge. However, Haq and Mizanuzzaman (2002) reported that the increase in carbon-di-oxide fraction in bio-gas also slows down the flame propagation in combustible charge. As a result some engine parameters (i.e. engine speed, load and injection timing) require to be optimized. Experimental results by quantifying the effects of carbon-di-oxide concentration bio-gas to reduce the energy density of the charge.

2.4 Energy Availability from NG and Bio-gas.

In literature, the maximum amount of heat available from stoichiometric combustion of unit mass of fuel, known as heating value, are reported. However, practical combustion processes are not essentially stoichiometric, and heat release depends on the equivalence ratio. Therefore, two useful definitions of heat release at standard condition of 0.1 MPa pressure and 25° C are:

1. Specific energy, SE: heat available from charge of unit mass [kJ/Kg-charge].
2. Energy density, ED: heat available from charge of unit volume [kJ/m^3-charge].

Shown in Fig. 2.1 are the heating values of bio-gas of various composition, methane, gasoline and diesel fuel. It is observed that the heating values of bio-gas are significantly lower than other hydrocarbon fuels and the deviation increases with increase in the CO_2 content of bio-gas. It explains the reduction of power in diesel engines (Bari *et al* 1994).

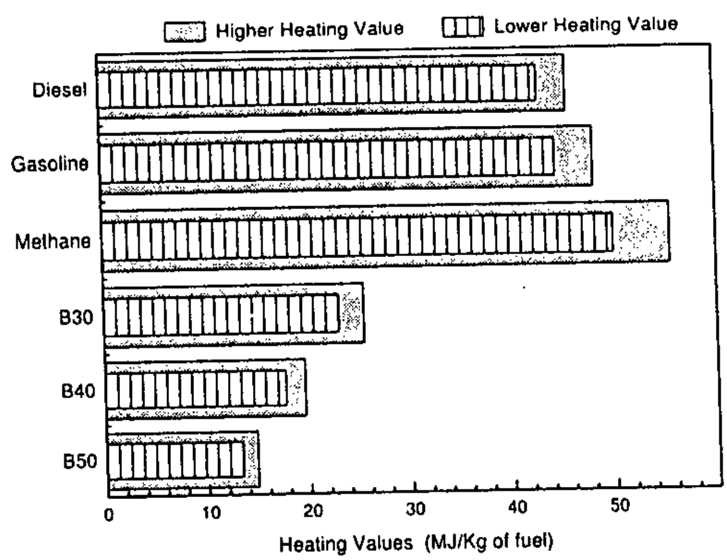

Fig. 2.1 Heating values of diesel, gasoline, methane and biogas of various composition. Hence, B30, B40 and B50 are biogas containing 30%, 40% and 50% CO_2, respectively.

Shown in Figs. 2.2a and 2.2b are the variations of specific energy and energy density, respectively as a function of ϕ for bio-gas, gasoline and diesel fuels, obtained from Haq and Mizanuzzaman (2002). It is observed that the heat release for a given amount of charge depends on both ϕ and the fuel composition itself. Heat releases decrease with increase in CO_2 content in bio-gas. However, for all the pre-mixtures, heat releases are maximum for stoichiometric mixtures, and their values decrease as ϕ deviates from stoichiometric conditions. Moreover, heat release from rich mixtures is higher than from the lean mixtures. This is due to the fact that, rich mixtures contain more fuel to burn and these are oxidized to release most of the heat leaving some species that are capable of slow oxidation and minor heat releases.

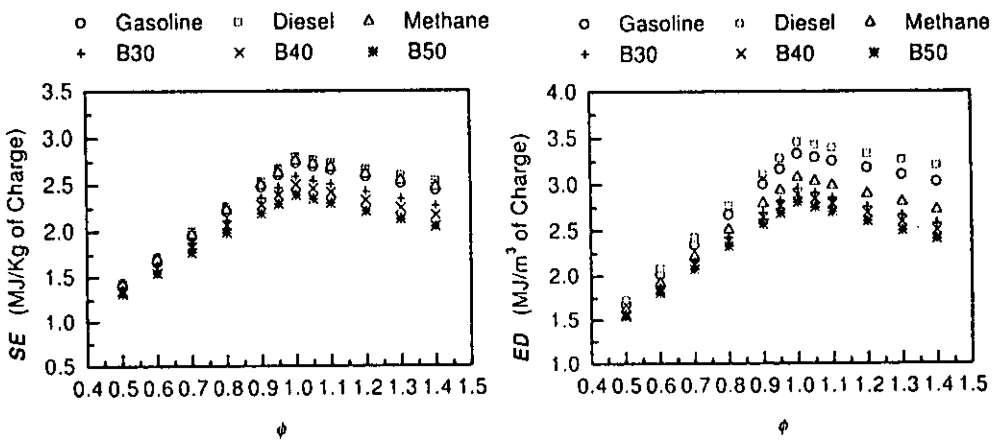

Fig. 2.2 (a) Specific energy (b) Energy density of fuel- air premixture.

2.5 Dual-Fuel Technology for CI Engines

Several technologies for CI engine conversion are being developed to convert standard diesel engines to duel- fuel operation (Cler 2000). Conversions can be relatively simple add-on processes or can require major engine modifications.

1. **Fumigation** is the least expensive and simplest way to convert diesel to dual fuel. It is commercially available, and virtually any manufacturer can use it. Conversion typically ranges from about $20 to $30 per kW of capacity. The conversion is accomplished by adding a natural gas carburetor in the diesel engines intake air system and pre-mixing natural gas with combustion air before the intake manifolds as well as adding or modifying appropriate controls.

   Fumigation dual-fuel engines are capable of displacing between 50 to 90 percent of the diesel fuel with natural gas. The reduction in NOx emission varies depending on the fraction of diesel fuel displaced, engine capacity, and loading and can range from less than 10% to 65% or more. Particulate and SO_2 emissions are also reduced roughly in proportion to the diesel fuel displaced by natural gas, but CO increases with increasing natural gas usage. Fumigation can be used with low-pressure natural gas, typically 0.007 to 0.035 MPa is all that is required.

2. **Multi port Injection** is the next level up from fumigation in conversion complexity, and potential benefit. Natural gas is sequentially injected to each cylinder individually just before the air intake valve. When valve opens the air and fuel mixture enters the cylinder. Diesel fuel is injected and compression ignition is initiated, which interns ignite the natural gas fuel. Multiport injection dual - fuel engines typically displace between 80% to 90% of the diesel with natural gas. Amricro-pilot version of this technologies being developed is expected to reduce requirements to 1% of the total energy requirements. This would result in NOx reduction of 50 to 70% compared with straight diesel operation. To inject the required quantity of natural gas within the available small time frame, natural gas pressure requirements for multi-port injection are typically on the order of 0.55 to 0.72 MPa. Conversion costs of $60 to $80 per kW for a 300 KW generator set are typical.

3. **High - Pressure Direct Injection** Converting a diesel engine to high-pressure direct injection dual - fuel operation may offer significantly reduced diesel requirements over other options. With this type of dual fuel engine, natural gas is directly injected into each cylinder when the piston is very near the top of its compression stroke, and the injected gas mixes with the already compressed combustion air. Pilot diesel fuel is injected after the natural gas, usually through a modified injector that supplies both fuels. Upon compression, the diesel fuel ignites and initiates combustion of the natural gas. With high pressure direct -injection dual fuel engines, natural gas can displace 98% to 99% of the diesel fuel, using just enough diesel fuel for proper ignition, however, in existing versions, 80% to 95% displacement is more common. As the amount of natural gas increases, additional controls must be to this system to ensure correct timing of the fuel injection. NOx reduction of about 40% has been achieved and according to the developer, greater reduction is expected. Converting a diesel engine to diesel fuel operation using this method requires natural gas pressures of 20 to 28 MPa.

4. **Major Engine Modifications** The diesel to duel fuel conversions just discussed are basically add-on conversions-that is natural gas-fueling equipment and control systems are added to an existing engine, or only relatively minor engine modifications are required. Modest natural gas pressure (0.1-0.5 MPa) is required and converted engines can achieve 60% or greater NOx reduction compared with diesel operation.

   Cylinder heads and piston must be replaced with this type of modified equipment to optimize performance. As with the other techniques, equipment for natural gas fueling and control is needed. The turbocharger after cooler will also require modifications to improve charge air cooling, as may the radiator, to maintain the full diesel power rating. Complete conversion costs range from about $80 to $180 per kW. Because this conversion requires major components replacement, they must be designed for specific engines and components. They are not transferable to other engines, as is frequently the case with other conversion technologies.

Table 2.3 Characteristic of various conversion technologies

Conversion type	NOx reduction % (relative to diesel)	Diesel substitute (%)	NG Pressure (MPa)	Conversion cost ($/KW)
1. Fumigation	10 – 65	50 – 90	0.007 – 0.035	20 – 30
2. Multi port injection	30 – 75	80 – 90	0.5 – 0.725	20 – 80
3. Direct injection	35 – 40	90 – 95	20 – 28	-
4. Major modification	70	92	.1 - .5	80 – 180

Duel fuel operating costs are generally lower than diesel operating costs as well, perhaps most importantly, at a capital cost as low as 20$ per kilowatt, converting existing diesel engines is far less expensive than buying new distributed generation equipment of any type. Efforts to broaden the duel-fuel approach face significant barriers often the some barriers that other distributed generation technologies face. These include permitting and inter connection requirements and in some emissions issues. The issue may divide the advocacy community for distributed generation between those who support dual-fuel technology for its economic benefits and those who oppose it because, although it is significantly cleaner than straight diesel operation, it offers little net environment gain over many traditional peaking plants. Views are conflicting as to whether dual fuel conversion might jump-start the broader distributed energy market or if it will have a negative influence (Cler 2000).

Chapter 3

Experimental Setup, Procedure and Post Processing of Data

Experiments were carried out in a diesel engine running in a dual-fuel mode using natural gas and bio-gas to substitute diesel consumption. A fixed amount of diesel (2 Kg/hr) was injected into the cylinder directly to use a pilot fuel to creating ignition and the engine loads were varied by regulating the gas (natural gas and bio-gas) flow rates. Hence, performance studies were carried out for natural gas and bio-gas of two composition (70% methane + 30% CO_2 and 50% methane + 50% CO_2 on volumetric basis) in addition to the diesel fuel at four different speeds of 2100, 2200, 2250 and 2300 rpm at different loading conditions. Synthetic bio-gas was utilized by producing it from mixing with CO_2 from cylinders and natural gas from main supply.

In the study, BS standard BS-5514 (1982), equivalent to ISO Standard-3046 and SAE standard J1349, was used in experimental and measurement procedures, and any other additional guidelines required were used from the procedures reported by Plint and Böswirth (1986). All measuring instruments were calibrated using relevant standard procedures prior to take any reading.

3.1 Experimental Setup

Present experimental set up consisted of engine test bed, diesel and gas supply system, different metering and measuring devices along with test engine. Schematic diagram of the experimental set up is shown in figure 3.1.

Experiments were carried out using a Kubota model V1502-B, 4 cylinder 1487cc diesel engine. It is water cooled, high speed direct injection four stroke diesel engine. Details of the engine specifications are given at Appendix A. The hydraulic dynamometer is used to apply desired load to the engine and for measuring the output power by measuring the resultant torque and the engine speed. Details of the dynamometer specifications are given at Appendix B. In the present chapter, only the brief overview of the experimental technique is presented, details of the required equations and sample calculation procedures are presented in Appendix C.

Dynamometer is connected with the engine by universal joint and care is taken not to cause eccentricity between them. Water is supplied to cool the dynamometer. Thermocouples were installed to monitor exhaust gas, lubricant, water, CO_2 gas and natural gas temperature. Diesel was fed to the injector pump under gravity. Natural gas-diesel fuel mixture and CO_2 gas, natural gas-diesel fuel mixture were controlled by the gas control valve with fueling taking place in the engine inlet manifold.

3.2 Experimental Procedure

Mixing of natural gas with CO_2 at different level of compositions was the main part of the experiment. However natural gas with different quantity mixed with air and allowed to enter into the cylinder during intake stroke. At the end of compression stroke a fixed amount of fuel was injected through the injector.

After checking the lubricant, water and fuel, engine was run by diesel fuel only for initial warming. Test was conducted using the conventional fuel (diesel) in order to establish base line for the candidate alternative fuels. Tests were conducted at constant speed at 2100, 2200, 2250, 2300 rpm and maintained within $\pm$ 10 rpm of the desired speed at different loads. For establishment of base line for diesel fuel, engine was run by 100% diesel fuel for constant rpm at any particular load. After 15 minutes, natural gas was supplied with the intake air and rpm was increased which was balanced by increasing the load and the diesel flow rate of 2 Kg/hr is ensured. Other parameter such as suction pressure of air was measured from the manometer.

Keeping the natural gas flow rate, load and rpm constant after 15 minutes, 30% CO_2 was supplied to the mixture. Hence rpm was lowered slightly. A heater also used at the delivery line of the CO_2 cylinder for the adequate flow of CO_2. To increase the rpm natural gas was increased slightly. Then time for the consumption of 50 cc fuel (diesel) same to those of the diesel and natural gas cases. Suction pressure for air, natural gas and CO_2 was measured from the penal board. Temperature for lubricant exhaust, CO_2, CH_4 was measured from the thermocouple. Load and rpm was measured from the digital meter. Similarly 50% CO2 was allowed to get mixture and the result had been noted.

3.2.1 Measurement of Air Flow Rate

Air flow rate was measured by drawing air through a parabolic nozzle into an air drum of standard size that is connected to the engine inlet and measured the pressure drop by means of a manometer, using water as a manometric fluid.

3.2.2 Measurement of Fuel Flow Rates

The diesel supply system was modified so that the diesel fuel could be supplied from a graduated burette instead of the fuel tank when needed. Fuel consumption is determined by observing the time from the stop watch for every 50 cc volume of fuel. Natural gas and bio-gas flow rates were measured using calibrated rotameters.

3.2.3 Measurement of Exhaust and Lubricant Temperatures.

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

3.2.4 Measurement of Engine Brake Power and Speed

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

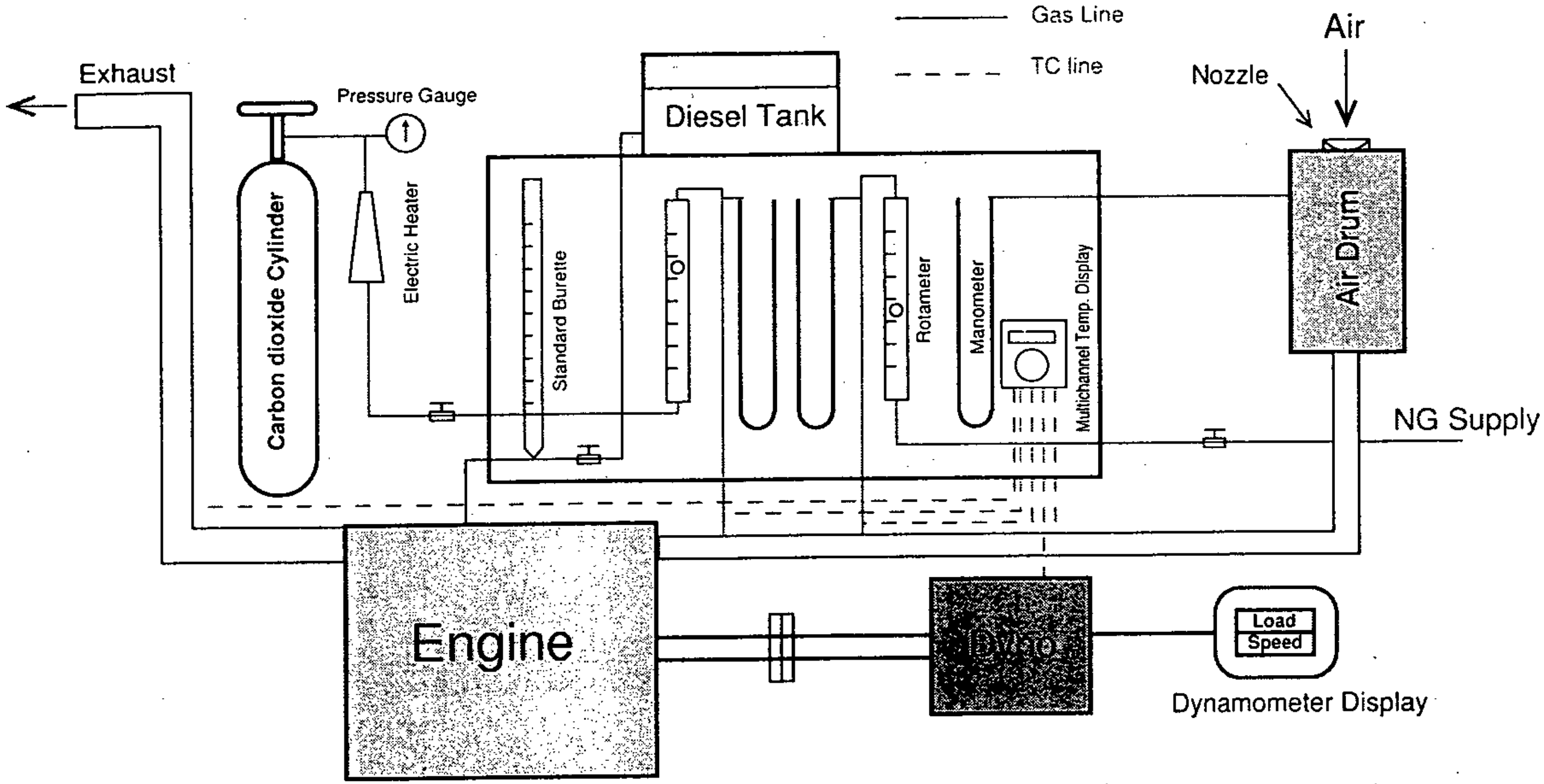

Fig. 3.1. Schematic diagram of the experimental setup showing the major instrumentation

3.3 Power Equations for CI Engines

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

$$\eta_b = \frac{P_b}{m_f Q} \tag{3.1}$$

Where,

P_b	=	Brake power, measured at fly wheel
η_b	=	Brake thermal efficiency,
m_f	=	Mass flow rate of fuel
Q	=	The lower heating value of fuel

If the fuel flow rate, m_f in equation 3.1 is replaced by Fm_a, where F is the fuel air ratio and m_a is the mass flow rate of air can be written as

$$P_b = \eta_b F m_a Q \tag{3.2}$$

Fuel-air ratio, F may be replaced by introducing equivalence ratio, ϕ, defined as the stoichiometric air-fuel ratio, A_s, divided by the actual air-fuel ratio, A, or equivalently, the actual fuel-air ratio, F, divided by the stoichiometric fuel-air ratio, F_S (Ferguson and Kirkpatric 2001):

$$\phi = A_S/A = F/F_S \tag{3.3}$$

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

$$\phi = \begin{cases} < 1 & \text{Lean mixture} \\ =1 & \text{Stoichiometric mixture} \\ > 1 & \text{Rich mixture} \end{cases}$$

Introduction of Eqs. 3.3 into 3.2 results in

$$P_b = \eta_b \, \phi \, F_S \, m_a \, Q \tag{3.4}$$

The power output of an internal combustion engine is limited by its breathing capacity, there is never difficulty in introducing any desired quantity of fuel into the cylinder, but the maximum amount that can be burned depends entirely on the amount of air available for combustion (Lumley 1999). If air were an incompressible, in viscid medium, then the displacement volume of the engine would be filled up into each two revolution for a four-stroke engine. Air, however, is not an incompressible in viscid medium, and the mass of air, which enters the engine cylinder, is somewhat less, depending on the engine speed and also on the residual gas present after exhaust process. Therefore volumetric efficiency, η_v is introduced to account for the effectiveness of the induction and exhaust process, and in term of quantity applied to an actual engine, it is defined as the mass of fuel and air inducted into the cylinder, divided by the mass of the mixture that would fill the piston displacement volume at inlet density, ρ_i in the intake manifold. For four stroke CI engines, volumetric efficiency, η_v , can be expressed algebraically as:

$$\eta_v = \frac{2\ m_a}{N V_d \rho_i} \tag{3.5}$$

Where,

m_a = Mass of fresh air inducted per unit time

N = Number of engine revolution per unit time

V_d = Total displacement volume of the engine

ρ_i = Inlet mixture density

The factor 2 in Eq. 3.5 arises from the fact that in the four stroke engine there is one cycle for every two crank revolution (Taylor 1985) and displacement volume, V_d is given by

$$V_d = n_c\, A_p\, S = n_c\, (\pi/4)\, b^2\, S \tag{3.6}$$

n_c = Number of engine cylinders,

A_p = Engine piston area,

S = Engine stroke length,

b = Engine cylinder bore.

Combination of Eqs. 3.5 to 3.6 results in the general power equation for engines as:

$$P_b = \eta_b \eta_v\, \phi\, F_S\, \rho_i\, Q\, N V_d/2 \tag{3.7}$$

Equation 3.7 is very useful as each term in it suggests some way in which the performance of the engine can be improved.

3.4 Influence of Engine Parameters

3.4.1 Combustion Efficiency and Equivalence Ratio

In practice, the exhaust of an internal combustion engine contains incomplete combustion-products. Under lean condition ($\phi < 1$), the amount of incomplete combustion products are small, however under fuel rich condition ($\phi > 1$), these quantities become more substantial since there is insufficient oxygen to complete combustion. Because, a fraction of the fuel's chemical energy is not fully released inside the engine during the combustion process, it is useful to employ combustion efficiency, η_C. Combustion-efficiency varies with equivalence ratio for internal combustion engines. For spark-ignition engines, for lean mixtures, the combustion efficiency is usually in the range of 95 to 98%. For mixtures richer than stoichiometric, lack of oxygen prevents complete combustion-of fuel, and the combustion efficiency steadily decreases as the mixture becomes richer. For diesel engines, which always operate lean, the combustion efficiency is normally high-about 98%. However, combustion efficiency of diesel engines can be reduced significantly for over-load conditions because of shortage of air to complete the combustion process.

3.4.2 Inlet Air Density ρ_i

The inlet density can be increased by super-charging, turbo-charging, inter-cooling, use of a fuel with a higher heat of vaporization, and water injection. It can also be increased if the engine is un-throttled, as in gasoline direct injection (GDI).

3.4.3 Volumetric Efficiency η_v

Separation, boundary layers and Mach numbers influence volumetric efficiency η_v. In internal combustion engines, air induction into the cylinder is limited by the appearance of chocking in a component (usually the inlet valve aperture), and this prevents the piston from any more air into the cylinder and the volumetric efficiency drops (Lumley 1999). Its value can be improved by changes in valve timing, cam modifications, porting, intake and exhaust tuning and changes in the number of valves per cylinder.

3.4.4 Average Piston Speed and Mechanical Efficiency

Engine speed, N, relates mean piston speed, $\overline{V}_p$ as

$$\overline{V}_P = 2NS \tag{3.8}$$

Hence, S is the stroke length and the term 2 arises from the fact that in one revolution of the crankshaft two stroke of the piston is completed. The mean piston speed an important parameter since stresses and other factors scale with piston speed rather than engine speed. It is fairly independent of engine size because engines of all sizes have piston rings made of roughly the same material, rubbing against cylinder wall of roughly same material, bearings of similar design using oils of similar viscosity, and so forth. Temperature rise of parts such as exhaust valves and piston crowns is proportionally related to the turbulent velocity, which is roughly proportional to the piston speed in a given engine. Moreover, there is a fundamental limitation on the maximum gas flow is the occurrence of sonic flow in the value aperture, and is related to the mean piston speed. In IC engines, mechanical efficiency is normally in the neighborhood of 0.85 for low engine speeds, drops drastically at high piston speeds, reaching 0.6 in the neighborhood of an average speed of 20 m/s. Average piston speed of 20m/s represents a practical maximum for most engines. Some racing car engines use a single piston ring to increase mechanical efficiency at a given speed. Therefore, piston speed is comparable among engines of otherwise very different characteristics. For example Table 3.1 illustrates, the model airplane engine has a mean piston speed of 5.0 m/s, while the enormous static marine diesel engine has a mean piston speed of 5.6 m/s.

3.4.5 Brake Mean Effective Pressure

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

$$P_b = \text{bmep}\,\frac{V_d N}{2} = \text{bmep}\,\frac{A_p S n_c N}{2} = \text{bmep}\,A_p\,n_c\,\frac{\overline{V_p}}{4} \qquad (3.9)$$

This mean pressure is fiction, but is useful as it is roughly comparable even in very different engines, as these different engines burn the same fuel, necessarily under approximately the same conditions and hence similar pressures. In SI engines, typical WOT bmep at present is between 0.9 and 1.1 MPa, however CI engines have 25-30% lower value of bmep because of much leaner combustion. As Table 3.1 illustrates, the bmep of model airplane of 1.6 x 10^{-6} m^3 displacement volume running at 11400 rpm is 0.32 MPa while the bmep of a typical Marine diesel engine of 0.433 m^3 displacement volume, running at 160 rpm is 0.45 MPa. The relatively small difference between these two figure are grossly attributed to the compression ratio (Lumley 1999). Comparing Eqs. 3.7 and 3.9, bmep can be written as

$$\text{bmep} = \eta_b\,\eta_v\,\phi\,F_s\,\rho_i\,Q \qquad (3.10)$$

In Eq. 3.10, the product, $\phi\,(F/A)_{stoic}\ \rho_i\ Q$ has the dimension of pressure and it reflects difference in design - engines of completely different characteristics (such as size, speed and so forth), but equal efficiency, will have equal value of bmep.

If it is evaluated for ρ_i, atmospheric and stoichiometric mixture, then it will represent a value of bmep, far above anything practically available, because it would correspond to efficiencies of unity. If ρ_i = 1.2 Kg/m^3 for gasoline, Q = 42.7 MJ/Kg, and Fs = 1/14.89, the bmep is equal to 3.44 MPa. However, if η_b = 0.30, η_v = 0.85 is assumed then a reasonable value of bmep, equal to 0.88 MPa, is obtained.

3.4.6 Specific Power

If bmep removes irrelevant differences among engines (such as physical size), and reflects only genuine design differences, and if the same can be said of $\overline{V}_p$, then the quantity $P_b/A_p\ n_c$, the power produced per unit piston area must also be such a quantity. It is called the specific power, P_s. Hence,

$$P_s = P_b/A_p\ n_c = \text{bmep}\ \overline{V}p/4. \tag{3.11}$$

In fact, the three quantities: bmep, P_b/A_pn_c, and $\overline{V}_p$ are not independent, any two being sufficient to determine the third thing. Eqn 3.11 for example as reported. Typical values of specific power of small, medium and large engines are presented in Table 3.1.

Table 3.1 Comparison of three internal combustion engines

Characteristics	Model Airplane	Automobile	Marine
b (m)	0.0126	0.089	0.737
S (m)	0.0131	0.080	1.016
V_d (m^3)	1.6 x 10^{-6}	4.98 x 10^{-3}	0.433
P_b (kW)	0.1	1.68	529
V_p (m/s)	5.0	6.6	5.6
Bmep (MPa)	0.32	0.80	0.45
P_s (kW/m^2)	400	1320	630

3.4.7 Brake Specific Fuel Consumption, bsfc

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

$$\text{bsfc} = \frac{mf}{Pb} = \frac{1}{\eta_b Q} \tag{3.12}$$

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

Chapter 4

Results, Discussions and Conclusions

In the present study, experiments were carried out in a CI engine using natural gas and biogas as the main source of energy input. The engine was operated in a dual-fuel mode at four different speeds of 2100, 2200, 2250 and 2300 rpm at different loading conditions. Data obtained are analyzed to have different performance parameters to study the effects of different operating conditions.

4.1 Results and Discussions

Shown in Fig. 4.1 are variation of the air flow rate with brake power of the test diesel engine running with pure diesel fuel at four different engine speeds of 2100, 2200, 2250 and 2300 rpm at different output loading. It is seen that the air flow rates increase with increase in engine speed and decrease slightly with increase in brake power. The increase in flow rate with speed is due to the fact that more volume is swept per unit time in case of higher engine speed. However, at higher engine output power, engine intake manifold temperature is high and therefore suction air density is reduced to cause reduced mass of air induction into the cylinder, as seen in fig. 4.1.

Shown in Fig. 4.2 are the variations of estimated volumetric efficiencies for the same operating conditions as presented in Fig. 4.1. It is seen that volumetric efficiencies decrease with increase in brake power, however at slower engine operations higher volumetric efficiencies are observed. It is apparently in contrast to the observations shown in Fig. 4.1. The lower volumetric efficiency at higher engine speeds is not surprising because of the rate of displacement volume increases proportionately to engine speed and air flow rate does not increase as proportional to the engine speed. The flow of air into the intake system has to overcome the frictional resistance of the various components of the intake system and this frictional resistance increases with engine speed and therefore air flow rate can not maintain its proportional increase with engine speed. In practice, air intake capacity of the system is limited by the occurrence of the sonic velocity of air at some position of the intake system and beyond that speed air flow does not increase any more.

In diesel engines, power increase is achieved by increasing the fuel flow rates. Shown in Fig. 4.3 are the variation of the fuel flow rate with brake power of the test diesel engine running with pure diesel fuel at four different engine speeds. Hence, fuel flow rate increase with brake power for all the engine speeds considered. The effect of the engine speed is not clearly demonstrated in Fig. 4.3, as the engine efficiencies were not significantly different in the small speed range considered in the present study.

Shown in Fig. 4.4 are the variations of the air fuel ratio with brake power of the test diesel engine running with pure diesel fuel at different test conditions. It is seen that the air fuel ratio gradually decreases with increase in brake power. However, air fuel ratio was found slightly high at high engine speed, in line with the observation of the increased air flow rate with high engine operations, as shown in Fig. 4.1. Shown in Fig. 4.5 are the corresponding equivalence ratio plotted as a function of brake power where a trend similar to air fuel ratio variation with brake power is obtained. It is found that, for part load conditions, combustion is very lean and its value approaches to stoichiometric value only at over load condition. Lean combustion is essential in CI engines as mixture is heterogeneous and excess air is required to ensure complete combustion.

Shown in Fig. 4.6 and 4.7 are the variation of lubricant and exhaust temperatures, respectively, with brake power at different test conditions. Both of these temperatures were found to increase gradually with increase in brake power, however the effect of engine speed is not evident. Theoretically, higher engine speeds are associated with higher lubricant and exhaust temperatures, especially in gasoline engines. In the present study, higher lubricant temperature is not observed because of the lubricant cooling mechanism of the test engine, and the higher exhaust temperature is not evident because of the presence of excess air in the combustion process. It may also be noted that both of these temperature are significantly lower than those of a typical SI engine operation.

Shown in Fig. 4.8 are the variations of brake specific fuel consumption with brake power at different test conditions. It is seen that bsfc is high at light loading and its value decreases with increase in the brake power until the brake power reaches the rated value where the bsfc is minimum and any further increase in brake power results in higher bsfc. Although the bsfc's at different speeds at same load did not vary significantly, bsfc of the slowest engine operation showed lower values of bsfc at part load condition, and the situation was reversed at rated and overload condition. This behavior of bsfc vs brake power can be explained by considering the different components involved in frictional losses, fractions of which are directly related to the engine speed and fraction of which is directly related to the peak engine cylinder pressure. At lower load conditions, the effect of engine cylinder pressure is not significant, rather the frictional losses are dictated by the energy expanded in moving the shafts, valves and pumps, and these losses increase with speed. Therefore, at low load conditions, bsfc is low for low speed operations. However, at higher load, the effect of peak cylinder pressure becomes more significant and peak engine pressures for slower engine operations are higher for same power generation and hence higher frictional losses. Moreover, at higher engine speed less heat is lost through the cooling system. Therefore, at higher speed operations, bsfc are lower for rated power condition. However, beyond the rated power, supply of air is not sufficient to ensure complete burning although the supplied air is more than the stoichiometric condition. CI requires more air than the stoichiometric amount because of the very nature of its combustion system where fuel is directly injected into the cylinder and heterogeneous mixture is formed and in case of overall stoichiometric mixture some portion of the charge will starve from oxygen to complete the combustion. Therefore, beyond rated load condition, combustion is not completed and some of the energy input is lost in the form of incomplete combustion products and results in higher values of bsfc's.

The bsfc is a measure of engine overall efficiency. In fact, bsfc and engine efficiencies are inversely related as seen in Eq. 3.12, so that the lower the bsfc, the better the engine. Shown fig. 4.9 are the variations of brake thermal efficiency plotted as a function of brake power at different test condition. However, for different fuels with different heating values, the values of bsfc's are misleading and hence the brake thermal efficiency is employed in the analyses of engine performance when the engines are fueled by natural gas and biogas of different composition.

Values of brake thermal efficiencies are plotted as a function of specific power, P_S, and values of brake specific fuel consumption's are plotted as a function of brake mean effective pressure, bmep, in Figs. 4.10 and 4.11, respectively. The bmep removes the effect of the engine size and specific power relates the piston speeds and the bmep, as discussed in § 3.4.4 and 3.4.5. It is observed that, maximum bmep is lower than the typical SI engine value of 1.0 MPa and the maximum specific power is lower than corresponding value of 2400 kW/m^2 (Lumley 1999).

Values of brake thermal efficiency and volumetric efficiencies for different test conditions using diesel fuel is plotted in Fig. 4.12 and 4.13.

Shown in Fig. 4.14 and 4.15 are the variations of brake thermal efficiency plotted as a function of brake power and bmep, respectively, at different test condition using different fuels considered in the present study. Hence, results for pure diesel shown in Fig. 4.9 and 4.10 are reproduced in Fig. 4.14 (a) and 4.15 (a), respectively, for comparison purposes. In case of tests using natural gas and bio gas of two compositions, a pilot amount of diesel equal to 2.00 kg/hr is injected directly into the cylinder and the rest of the input energy is supplied by supplying the gases into the intake manifold by fumigation method. Overall engine performance, as shown in Fig. 4.14 (b), (c) and (d), is similar to that of the diesel fueling as shown in Fig. 4.14(a), when diesel is substituted partially by natural gas, bio gas containing 70% methane and 30% carbon dioxide by volume and bio gas containing 50% methane and 50% carbon di-oxide by volume respectively. Incase of pure diesel operation as shown in Fig. 5.14(a), brake thermal efficiency reaches it maximum value and the corresponding brake power is the rated power of the engine.

Beyond the rated power, efficiency is reduced because of the decrease of combustion efficiency resulting from the shortage of oxygen required to burn the fuel completely. In case of natural gas operation, as shown in Fig. 4.14(b), the efficiency continues to increase even beyond the rated power for the diesel operation. It is because, of the increased combustion efficiency resulting from the combustion of premixed natural gas and diesel, as natural gas was mixed homogeneously with air and thus demand lesser amount of excess air. Engine efficiency is slightly reduced in case of biogas fueling. The output power greater than the rated power (17 kW) for diesel operation suggests a possible decrease in efficiency at higher engine load. The decrease in efficiency was not measured in case of natural gas operation as experiments were not carried out at further higher loads because of the possible engine damage resulting from inadequate engine cooling, and in case of bio gas containing 50% CO_2 by volume, the efficiencies seemed to start falling, as shown in Fig. 4.14 and 4.15. The slight reduction of efficiency in case of biogas is due to the presence of CO_2 which did not supply any energy but absorbed some heat when exhausted in the form of raised temperature. The results obtained in the present study also suggest that the slow speed operate at 2100 rpm results in slightly higher efficiency. Equivalence ratio, volumetric efficiencies and exhaust temperatures, for the test condition are reported in Fig. 4.16, 4.17 and 4.18 respectively. No significant effect of speed is evident in the test results. Lower volumetric efficiencies are due to the substitution of some intake air by gaseous fuels and the substitution is more in case of biogas containing 50% CO_2 by volume and the effect is clearly reflected in volumetric efficiency, as shown in Fig. 4.17(d). Exhaust temperature of natural gas operation is lower than the diesel values because of lower adiabatic flame temperature of 2227^0C for NG stoich mixture lower than the corresponding diesel value of 2310^0 C (Lumley 1999).

Shown in Fig. 4.19 is the brake thermal efficiency plotted as a function of *bmep* for straight diesel operation. Brake thermal efficiencies, as shown in Figs. 4.20-4.22, have trend similar to the diesel fueling as shown in Fig. 4.19, when the diesel is substituted partially by natural gas, BG30 and BG50. Both the experimental and modeled results are plotted in these figures. It is seen that, in case of pure diesel operation as shown in Fig. 4.19, the experimental values of the brake thermal efficiency increases with increase in *bmep* until it reaches its maximum value at the rated brake power of the engine corresponding to that speed. Beyond the rated power, brake thermal efficiency is reduced because of the decrease of combustion efficiency resulting from the shortage of oxygen required to burn the fuel completely. Although the modeled results are within 2-3% of the experimental results up to the rated power (Bhutto 2003), it predicts higher thermal efficiency in all cases considered. In case of natural gas fueling, as shown in Fig. 4.20, the efficiency continues to increase even beyond the rated power for the diesel operation. It is because of the increased combustion efficiency resulted from the combustion of premixed natural gas which is inducted into the intake manifold by fumigation method. However, for gaseous fuel operations shown in Fig. 4.20-4.22, the reduction in thermal efficiencies beyond the rated power can be anticipated. Hence, the experimental results are within the reasonable agreement with the modeled results.

4.2 Conclusions

Major conclusions of the present study are:

- In CI engines, diesel fuel can be partially substituted by natural gas and bio gas by fumigation method. The diesel engine conversion to dual fuel mode operation demands very simple modifications of the engine intake system only. For that, pilot amount of diesel is required to be injected into engine cylinder to ignite the charge.

- Effect of moderated speed variation is found non in significant in engine performances.

- Slightly more power and better efficiency is obtained in case of natural gas operation, however these gains reduce with increase in CO_2 in biogas.

- Engine exhaust temperature and volumetric efficiency decreases with natural and biogas, especially with biogas containing more CO_2.

The following observations are made with regard to the future works:

- The effect of injection timing is important in the performance parameter as natural gas and biogas have different combustion characteristic to that of the diesel combustion characteristic.

- The effect of engine speed, specially with much slower than considered in the present study, might provide optimum combustion because of the slower flame propagation in biogas air premixture.

- Engine endurance test with natural gas and biogas will focus the overall effects on the engine life expectancy.

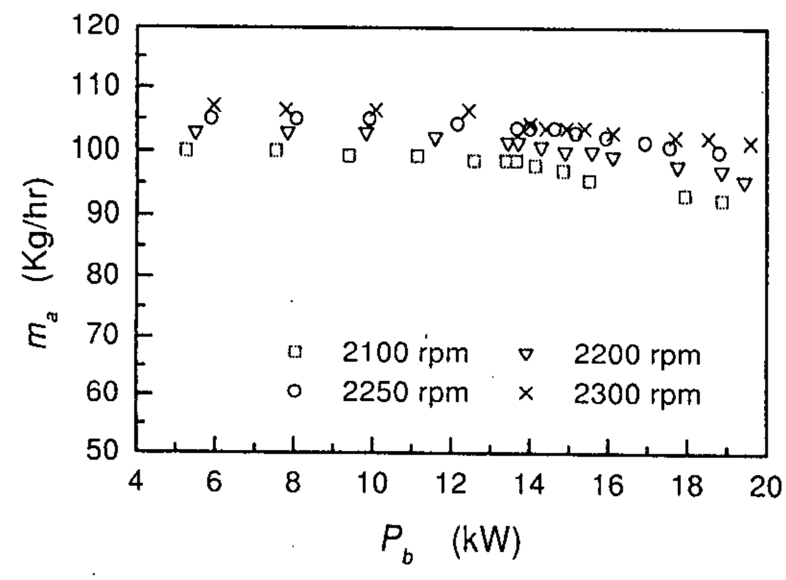

Fig. 4.1. Measured air flow rate as a function of engine brake power.

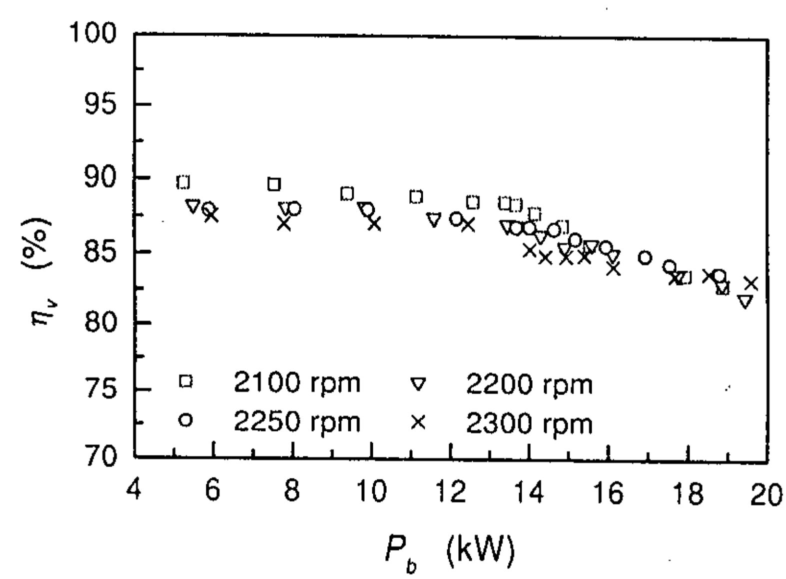

Fig. 4.2. Estimated volumetric efficiency a function of engine brake power.

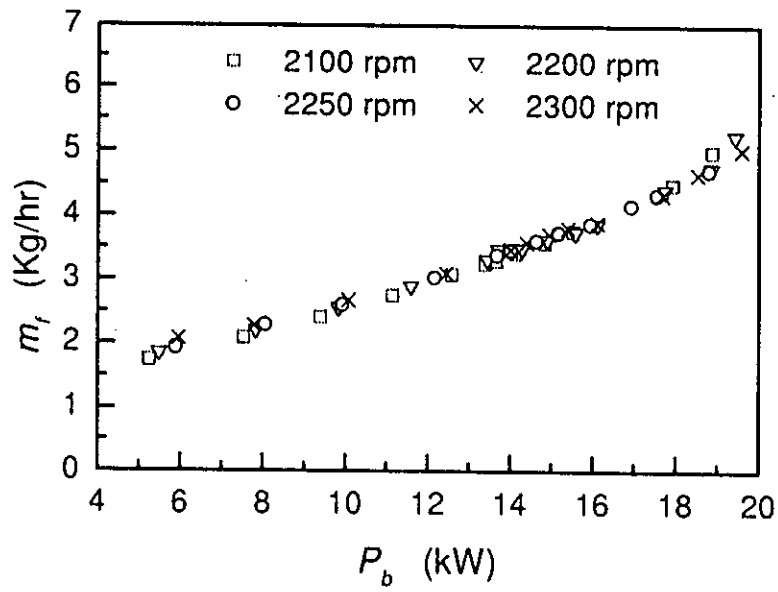

Fig. 4.3. Measured fuel flow rate as a function of engine brake power.

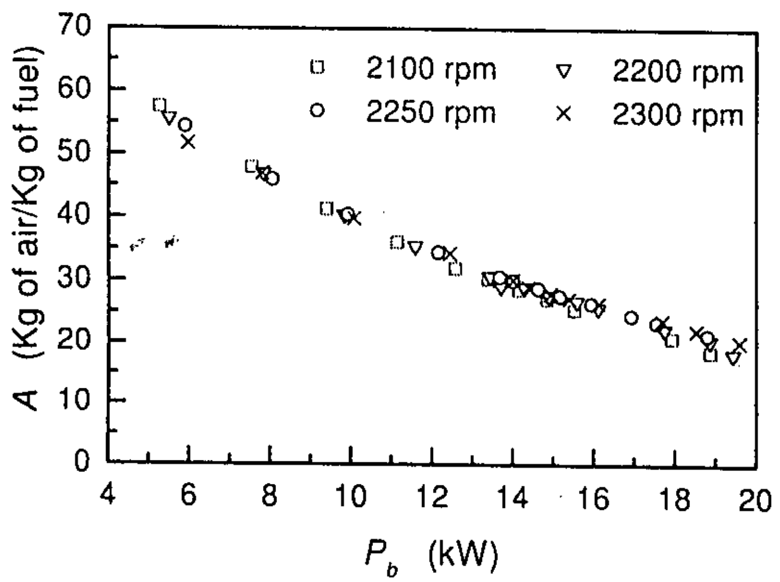

Fig. 4.4. Estimated air-fuel as a function of engine brake power.

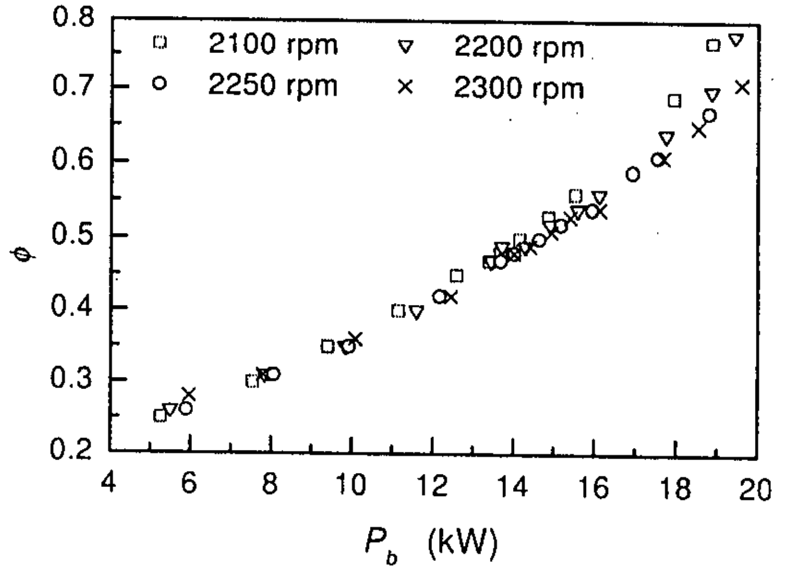

Fig. 4.5. Estimated equivalence ratio as a function of engine brake power.

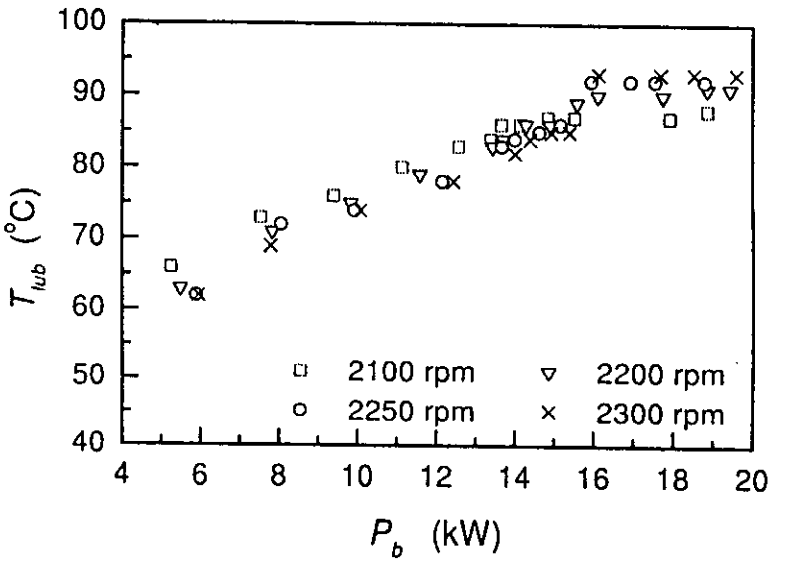

Fig. 4.6. Measured lubricant temperatures as a function of engine brake power.

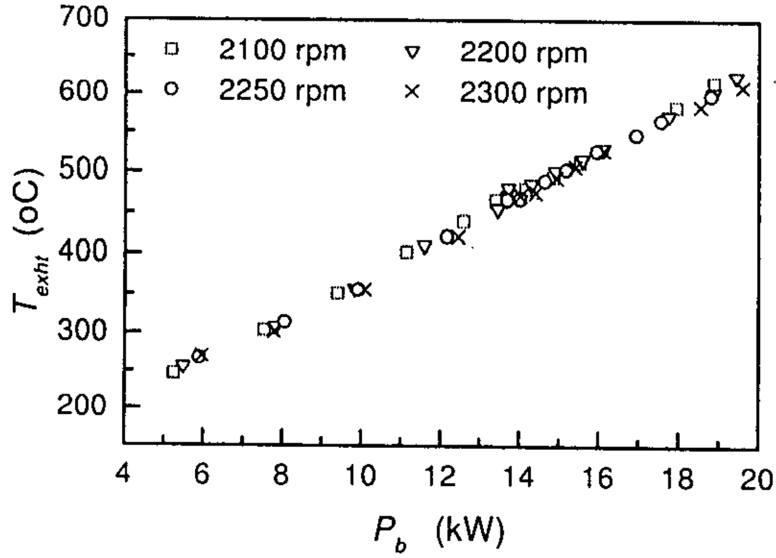

Fig. 4.7. Measured exhaust temperature as a function of engine brake power.

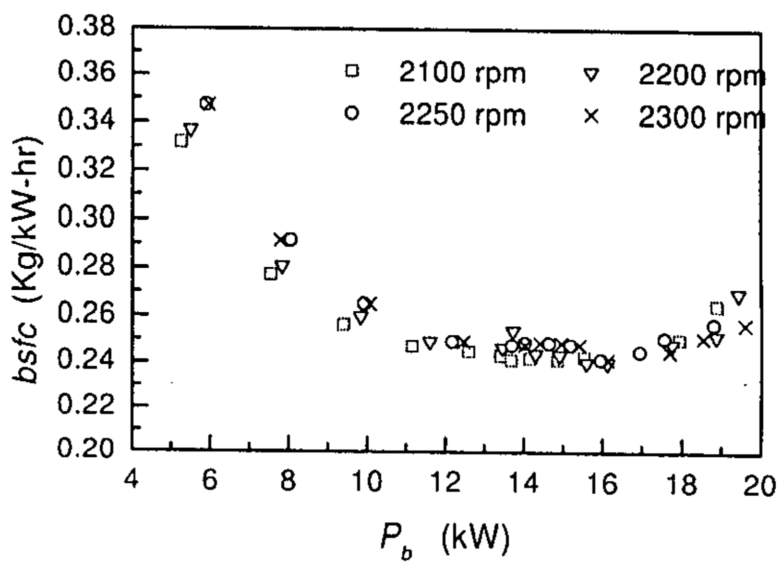

Fig. 4.8. Measured brake specific fuel consumption as a function of engine brake power.

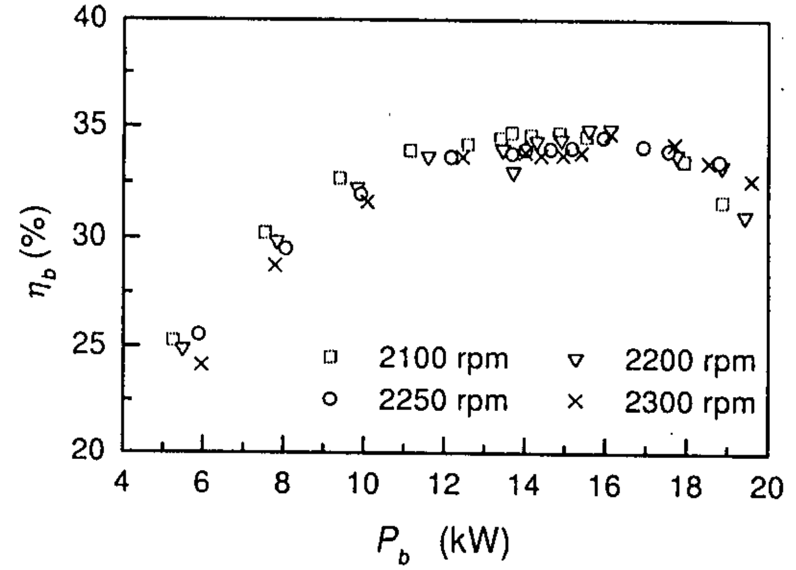

Fig. 4.9. Estimated brake thermal efficiency as a function of engine brake power.

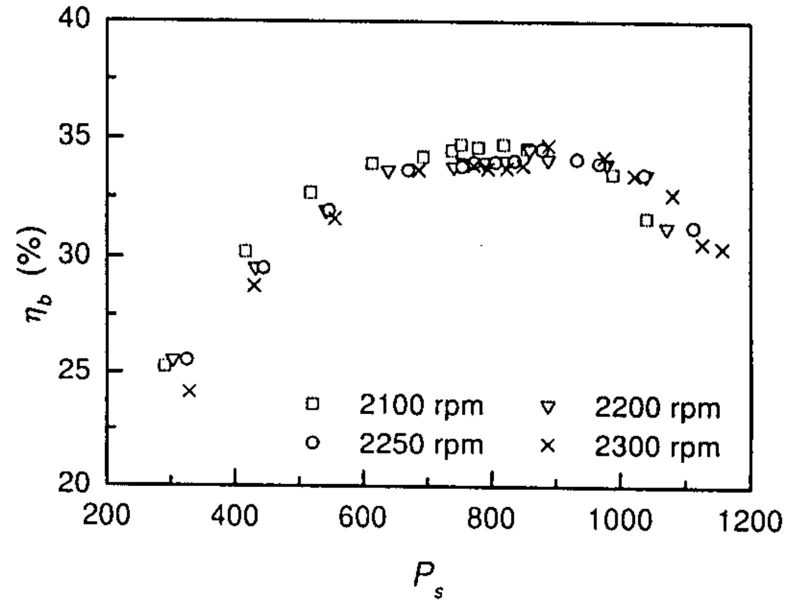

Fig. 4.10. Estimated brake thermal efficiency as a function of engine specific power.

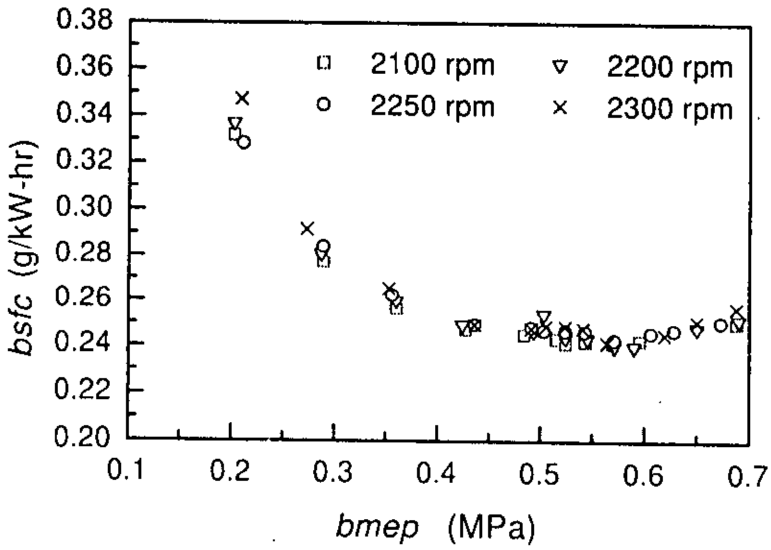

Fig. 4.11. Estimated brake specific fuel consumption as a function of brake mean effective pressure.

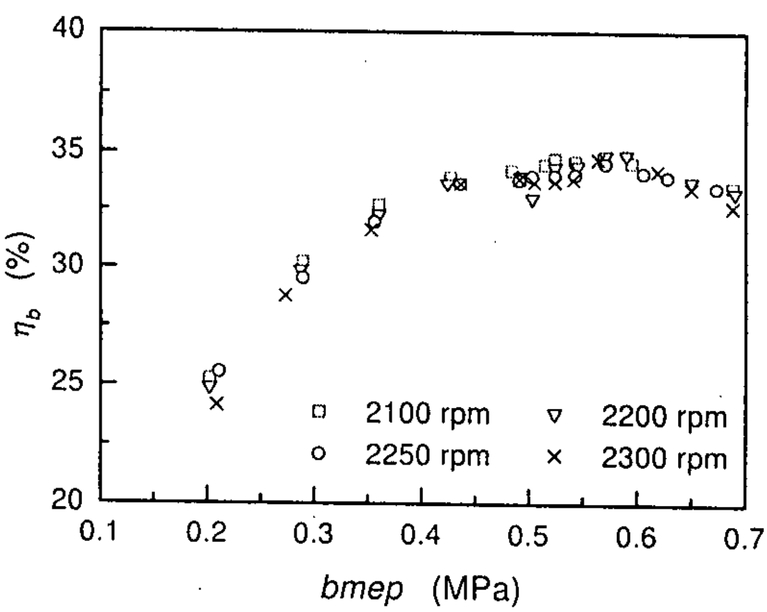

Fig. 4.12. Estimated brake thermal efficiency as a function of brake mean effective pressure.

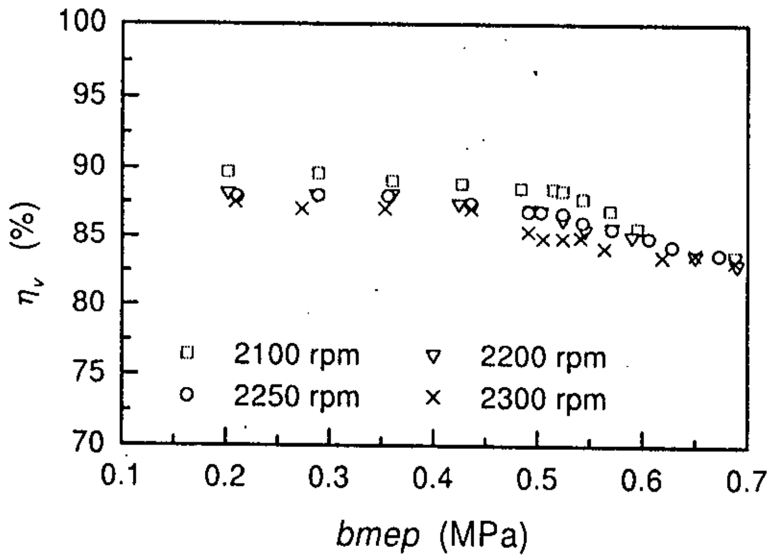

Fig. 4.13. Estimated volumetric efficiency as a function of brake mean effective pressure.

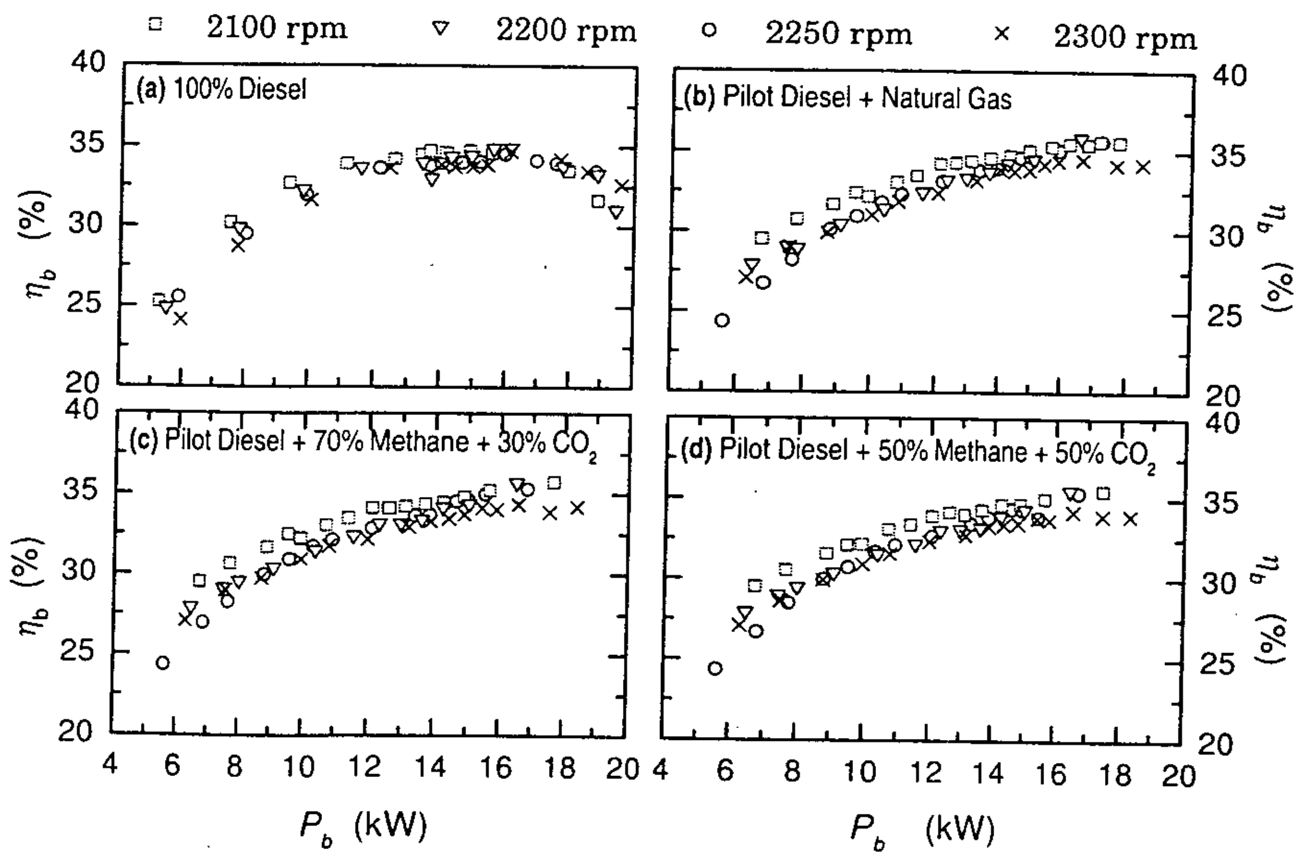

Fig. 4.14. Estimated brake thermal efficiency as a function of engine brake power.

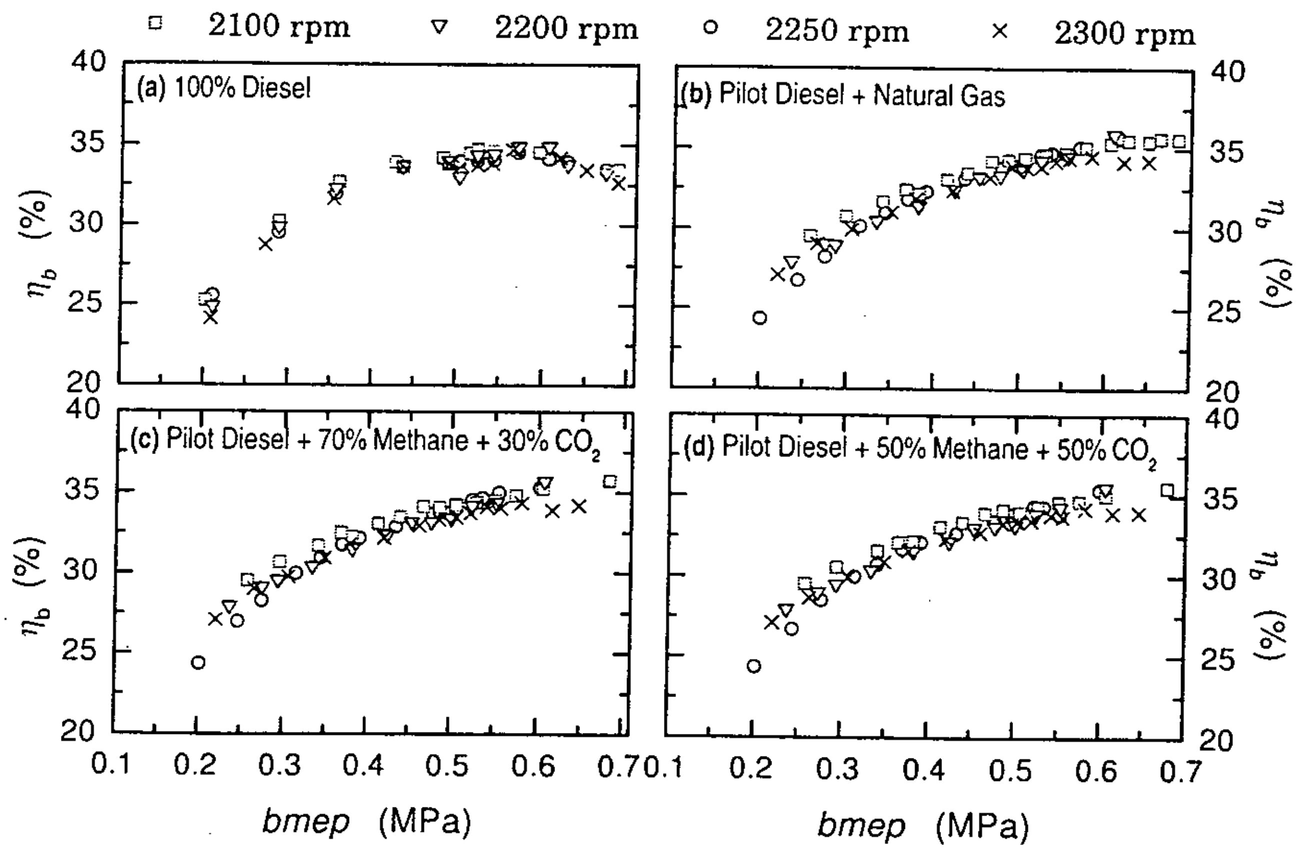

Fig. 4.15. Estimated brake thermal efficiency as a function of engine brake mean effective pressure.

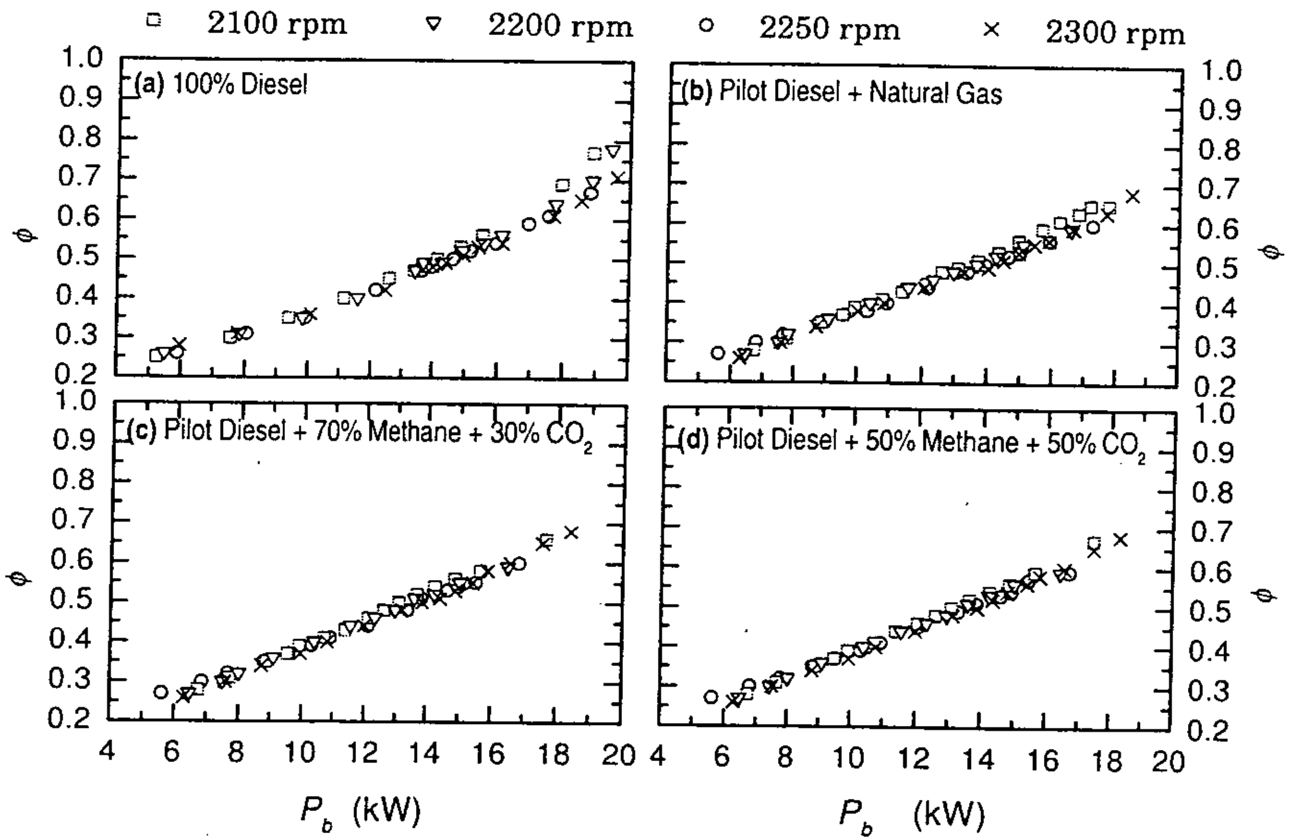

Fig. 4.16. Estimated equivalence as a function of engine brake power.

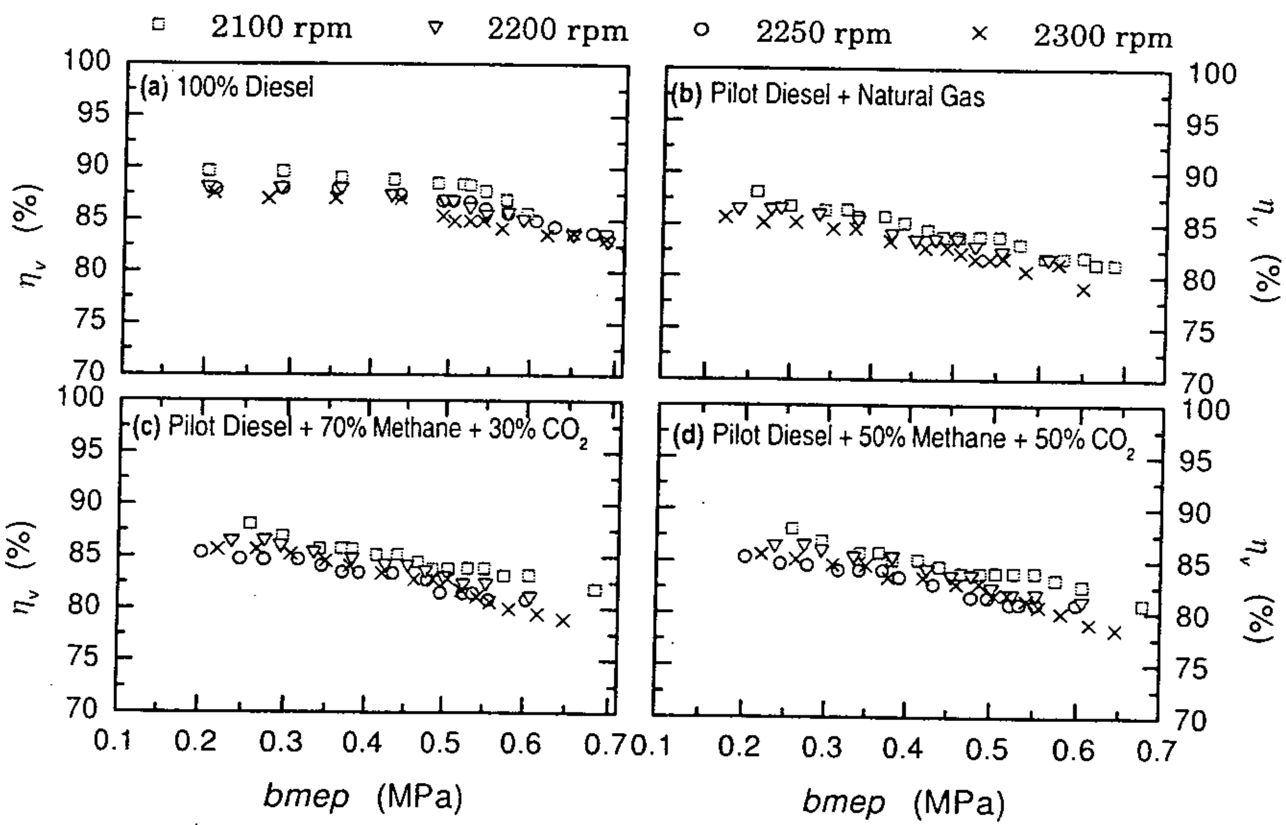

Fig. 4.17. Estimated volumetric efficiency as a function of engine brake mean effective pressure.

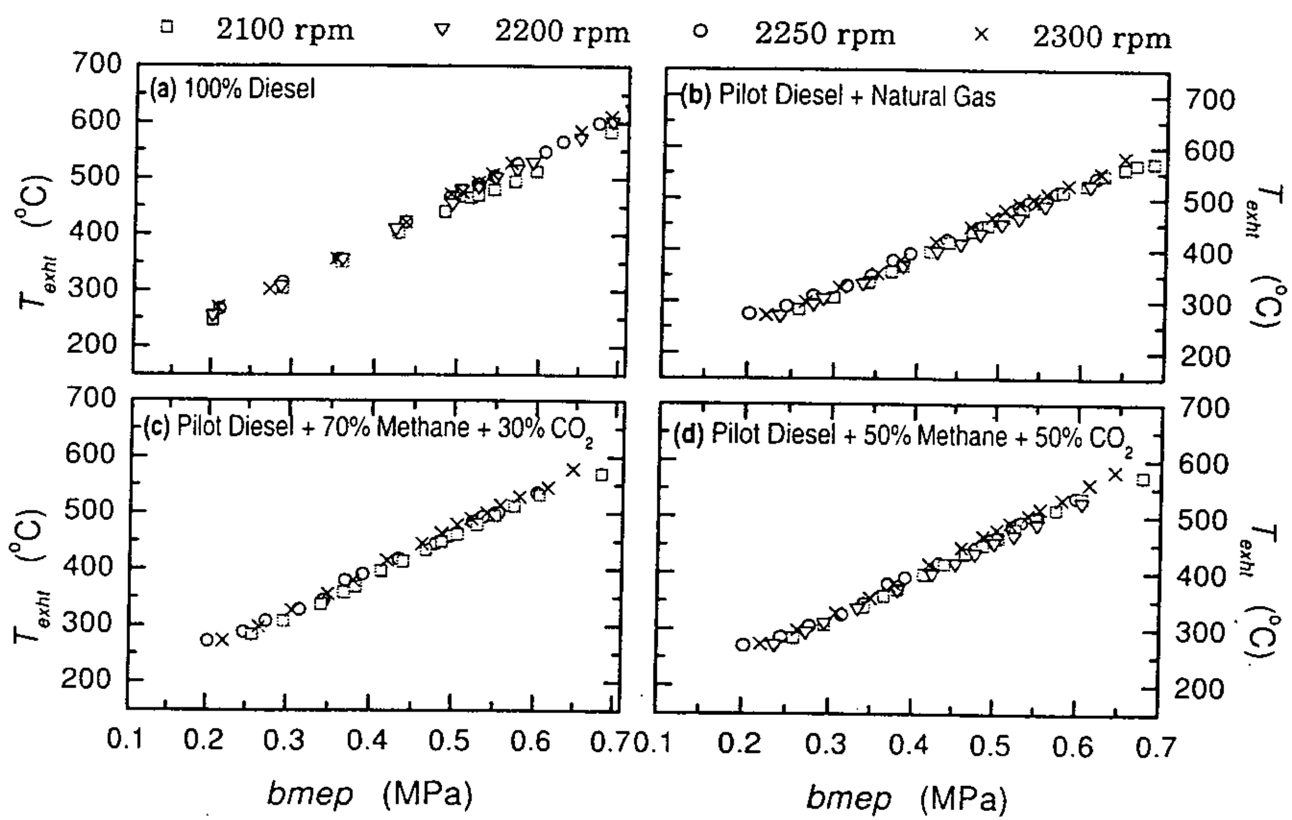

Fig. 4.18. Estimated exhaust temperature as a function of engine brake mean effective pressure.

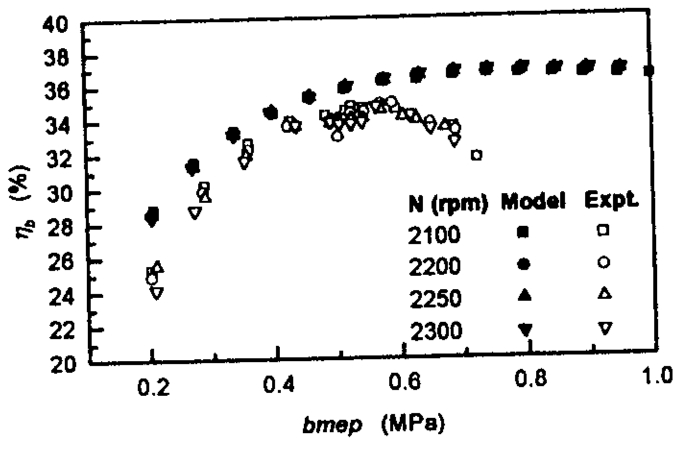

Fig. 4.19. Brake thermal efficiencies for different engine speeds when the engine is fueled by straight diesel.

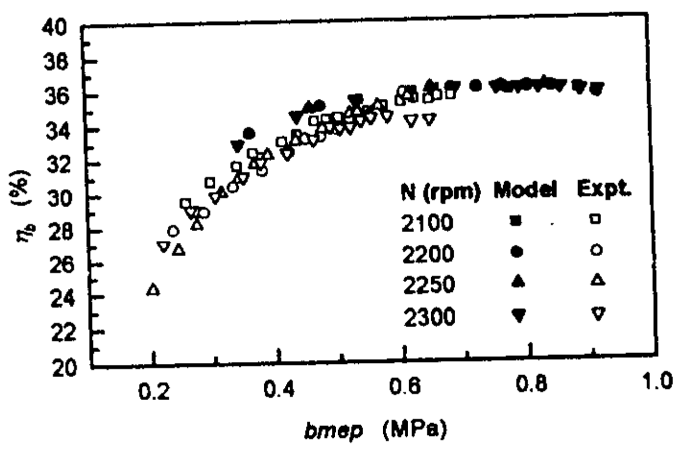

Fig. 4.20. Brake thermal efficiencies for different engine speeds when the engine is fueled by natural gas and pilot diesel fuel.

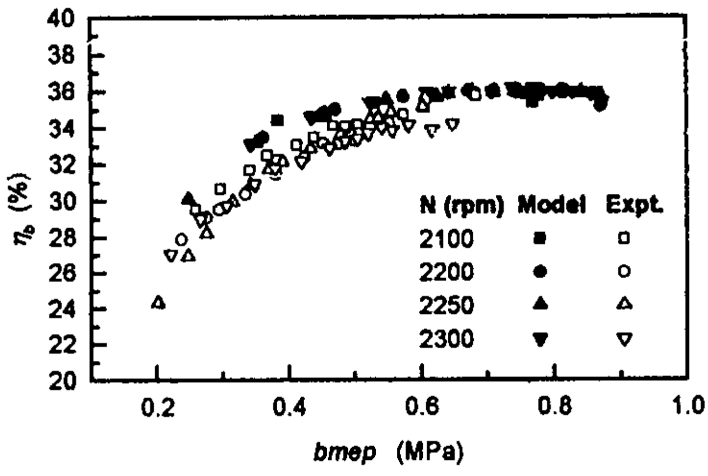

Fig. 4.21. Brake thermal efficiencies for different engine speeds when the engine is fueled by biogas BG30 and pilot diesel fuel.

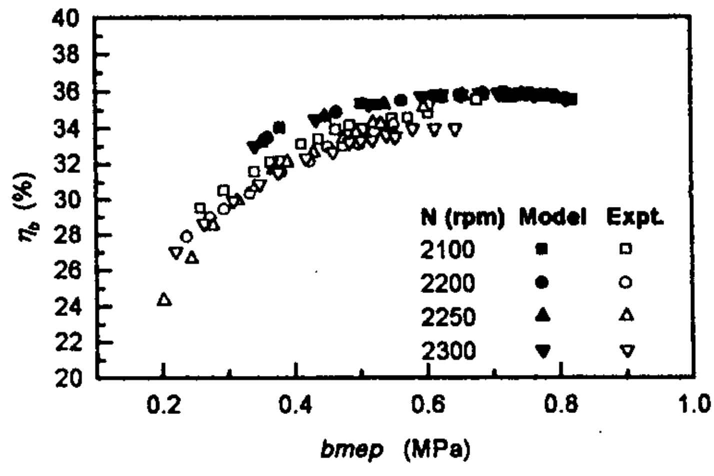

Fig. 4.22. Brake thermal efficiencies for different engine speeds when the engine is fueled by biogas BG50 and pilot diesel fuel.

Appendix A

Engine Specifications

Model and type	: V1502B, Vertical, Water-cooled 4-cylinder diesel engine
Bore (mm)	: 76 mm
Stroke (mm)	: 82 mm
kW Rating At 2250 rpm	: 17
Number of cylinder	: 4
Method of cooling	: Pressurized Radiator Forced Circulation with water pump.
Compression ratio	: 21
Fuel injection timing	: 25 BTDC
Volume	: 1487 cc

Appendix B

Dynamometer Specifications

Type	: Water Brake Dynamometer
Model	: TFJ-250L
Max. braking horsepower (PS)	: 250
Revolutions at max. braking horsepower point (rpm)	: 2500 to 5000
Max. braking torque (kg.m)	:71.6
Max. braking water quantity (lit/min)	: 75
Weight (kg)	: 575
Max. revolutions (rpm)	:5500

Appendix C

Sample Calculation

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

$$P = \frac{WN}{3400} \tag{C.1}$$

$$\dot{m}_f = 0.036\frac{V\rho_f}{t} \tag{C.2}$$

Where,

P = Engine brake power

W = load shown in the dynamometer

N = rpm of the shaft connected to the dynamometer

$\dot{m}_f$ = fuel consumption rate in (kg/hr)

V = calibrated volume in ml

t = time in second to consume V amount of fuel

ρ_f = density of fuel in kg/m^3

Engine brake power was standardized using BS:5514:1982

$$P_b = \frac{P}{\alpha} \tag{C.3}$$

Where,

P_b = Brake power after standardization

α = Power correction factor

Brake thermal efficiency was estimated using Eq. 4.14

Air flow rate was estimated by measuring the pressure drop created when it was passed through a 30 mm flow nozzle and its value is estimated using

$$\dot{m}_a = C_D A_n \sqrt{2g \frac{h_m \rho_w}{\rho_a} 3600} \tag{C.4}$$

Where,

C_D = Coefficient of discharge of the nozzle

h_m = Manometer deflection

ρ_w = Density of water kg/m^3

ρ_a = Density of air kg/m^3

g = Gravitational acceleration m/s^2

A_n = Nozzle area.

Experimental Data and Results :

W	=	8.5
N	=	2248 rpm
V	=	50 ml
t	=	78.6 sec
h_m	=	72 mm
C_D	=	0.98
A_n	=	4.536×10^{-3} m^2
ρ_w	=	995.7 Kg/m^3
ρ_a	=	1.199 Kg/m^3
P	=	5.62 kW
$\dot{m}_f$	=	1.94 Kg/hr
P_b	=	5.62/0.9735 = 5.77 kW
bsfc	=	0.336 Kg/kW-hr
$\dot{m}_a$	=	102.61 Kg/hr
A	=	102.61

Appendix D

Engine Derating

Engine brake power, p_b and brake specific fuel consumption bsfc is standardized using BSS: 5514 Part 1: 1982.

Rated (BS) condition,

P_r = 100 KPa

T_r = 300 K = 27°C

ϕ_r = 0.6

η_m = 0.85

Where,

P_r = Rated pressure

T_r = Rated temperature

ϕ_r = Rated humidity

η_m = Mechanical efficiency

Lab condition,

P_x = 100.3 KPa

T_x = 293 K = 20°C

ϕ_x = 0.82

Where,

P_x = Relative pressure at lab

T_x = Relative temperature

ϕ_x = Relative humidity

From Annex F, BSS 5514:

Temperature ^{o}C	Relative humidity ϕ_x	ϕ_x P_x (KPa)
20	0.8	1.9
20	0.6	1.4

For relative humidity $\phi_x = 0.82$, from interpolation we get,

$$\phi_x P_x = 1.95 \text{ KPa} \tag{D.1}$$

From Annex E, BSS 5514:

For ϕ_x P_x = 1.95 KPa and P_x = 100.3 KPa, from interpolation we get,

$$E = \frac{P_x - a\ \phi_x P_x}{P_r - a\ \phi_r P_r} = 1.005 \tag{D.2}$$

Where,

E = the dry air pressure ratio

A = 1

From Annex D, BSS 5514:

$$\left(\frac{T_r}{T_x}\right)^n = \left(\frac{300}{293}\right)^n = (1.0238)^{0.75} = 1.0177 \tag{D.3}$$

Where,

n = 0.75 for CI engine

n = 0.50 for SI engine

Multiplying eqn. (D.3) with (D.2) we get the ratio of indicated power K,

$$K = E \times 1.0177 = 1.005 \times 1.0177 = 1.02 \tag{D.4}$$

From Annex C, BSS 5514:

Using value K = 1.02 and

η_m = 0.85 we get

β = 0.998

From Annex B, BSS 5514:

Using $\eta_m = 0.85$ and K = 1.02 we get

α = 1.023

Therefore the bhp and bsfc in BS condition

$$bhp = \frac{\text{bhp in lab condition}}{\alpha} \text{ hp}$$

$$bsfc = \frac{\text{bsfc in lab condition}}{\beta} \text{ gm / bhp − hr.}$$

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