STUDY OF THE EFFECT OF SPARK ADVANCE ON GAS RUN PETROL ENGINE

BY
MD.EHSAN

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Md. Ehsan

Submitted to the Department of Mechanical Engineering BUET, in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

Approved as to style and content :

1. Saifulla 7/9/93

Dr. Saiful Bari
Assistant Professor
Dept. of Mechanical Engineering
BUET, Dhaka

Chairman

2. 7/9/93

Dr. A. M ~~Aziz-ul Huq~~
Professor
Dept. of Mechanical Engineering
BUET, Dhaka

Member

3. Md. Wahhajuddin 7/9/93

Dr. Md. Wahhaj Uddin
Professor & Head
Dept. of Mechanical Engineering
BUET, Dhaka

Member (Ex-officio)

4. Musharraf Husain Khan 7/9/93

Prof. M.H. Khan
Member
University Grants Commission
Sher-e-Bangla Nagar, Dhaka.

Member (External)

CANDIDATE'S DECLARATION

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 or publication.

Signature of the Supervisors :

1. ______________________

Dr. Saiful Bari
Assistant Professor
Dept. of Mechanical Engineering
BUET, Dhaka

Supervisor

2. ______________________

Dr. ~~A. M. Aziz-ul Huq~~
Professor
Dept. of Mechanical Engineering
BUET, Dhaka

Co-Supervisor

Signature of the Candidate :

____

Md. Ehsan

Candidate

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ABSTRACT

In view of the energy crisis, alternative fuels are tried all over the world in IC engines. This research concerns the use of natural gas as alternative fuel in SI engines. Petrol engines can run on natural gas, when slightly modified. The combustion characteristics of natural gas is different from that of petrol, which eventually affects the engine performance. The engine performance with natural gas was tested at 2500 rpm and compared with that of petrol. Results showed some reduction in power and slight fall of efficiency. The exhaust temperature was found to be higher for natural gas. The air gas ratio for optimum performance was higher for gas than for petrol. These differences in performance were mainly due to the slower speed of flame propagation for natural gas. In this investigation the effect of spark advance on engine performance was also examined. The present spark advance mechanism consists of centrifugal and vacuum advance mechanisms. For the first set of experiments, for both fuels, the engine load was kept constant and by changing the spark advance positions best power spark advance (bpsa) was determined at various speeds. Under such conditions the centrifugal advance mechanism plays the vital part in spark advance control. For the second set of experiments, for both the fuels, the speed was kept constant and bpsa were determined at various loads. Under such conditions the vacuum advance mechanism plays the vital role in controlling the spark advance. In both the cases the bpsa for natural gas was found to have higher values than petrol. Analysis of the results showed that for best performance, using natural gas as the alternative fuel, both centrifugal and vacuum advance mechanisms of the petrol engine need modifications.

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ABBREVIATIONS

AF	Alternative Fuel
ASTM	American Society for Testing and Materials
BRLS	Barrels
BBS	Bangladesh Bureau of Statistics
BC	Bottom Centre
BDC	Bottom Dead Centre
bmep	Brake mean effective pressure
bsfc	Brake specific fuel consumption
BP	British Petroleum
bpsa	Best-power spark advance
BTDC	Before Top Dead Centre
C	Celsius
CI	Compression ignition
CN	Cetane Number
CNG	Compressed Natural Gas
CO	Carbon monoxide
ERL	Easter Refinery Limited
Eq.	Equvalent
F	Fahrenheit
IC	Internal Combustion
imep	Indicated mean effective pressure
klimep	Knock limited indicated mean effective pressure
LNG	Liquid Natural Gas
LPG	Liquified Petroleum Gases
MMCF	Million Cubic Feet
MON	Motor Octane Number
NG	Natural Gas
NO_x	Oxides of Nitrogen
PN	Performance Number
psi	Pound per square inch
Pr.	Pressure

rev/min	Revolutions per minute
rpm	Revolutions per minute
RH	Relative Humidity
RON	Research Octane Number
rpm	Revolution per minute
sa	Spark advance
SI	Spark Ignition
SIT	Spontaneous Ignition Temperature
SR	Bpsa Ratio
TC	Top Centre
TDC	Top Dead Centre
VR	Vacuum Ratio

TO MY PARENTS

CHAPTER – 1

INTRODUCTION

In view of the present energy crisis, different types of alternative fuels have been tried all over the world for the use in internal combustion engines. The importance of such studies and their application was first emphasized after the oil shock in 1973. Later the price of fossil fuels like petrol and diesel have reduced, but the consumption rate of the known reserves of conventional fossil fuels is growing fast. Table - 1.1 shows the estimated world energy reserves, consumption and life times. Also increasingly greater concern of the harmful effect on the environment, created by these fossil fuels have emphasized the need for study of alternative fuels. The exhaust from diesel and petrol engines are believed to be one of the major causes of air pollution and green house effect leading to the global warming.

Worldwide studies have established that there are several alternative fuels which appear to be prospective substitute for petrol. Some are gaseous and some are liquid. Some of them are also renewable in nature. Each of these alternative fuels has its advantages and limitations. The main alternative fuels for IC engines are Natural gas, Bio-gas, Hydrogen, Alcohol fuels, Vegetable oils etc. The use of gaseous fuels are intensively studied in the automotive sector, as gaseous fuels has the advantage of emitting less pollutant to the environment than liquid fuels.

Some countries have natural gas, but they import oil from abroad. Use of the natural gas in IC engines can reduce their dependency on imported oil and the atmospheric pollution level. In addition to the gaseous fuel found in wells it can also be produced from a wide variety of feedstock by chemical processing.

Bangladesh has a few proven mineral resources other than natural gas. Bangladesh thus depends on imported crude petroleum and related materials. A major portion of her foreign exchange spent in importing crude petroleum. Figure 1.1 shows a Pie-chart of the imported commodities of Bangladesh (1990).

On the other hand Bangladesh has a vast deposit of natural gas. Table 1.2 & 1.3 show the production quantity and chemical composition of natural gas at various gas fields of Bangladesh respectively. At present a little more than 1% of the gas reserves is being consumed annually. Running the large number of IC engines available in the country with natural gas may play a vital role in strengthening the economy of the country.

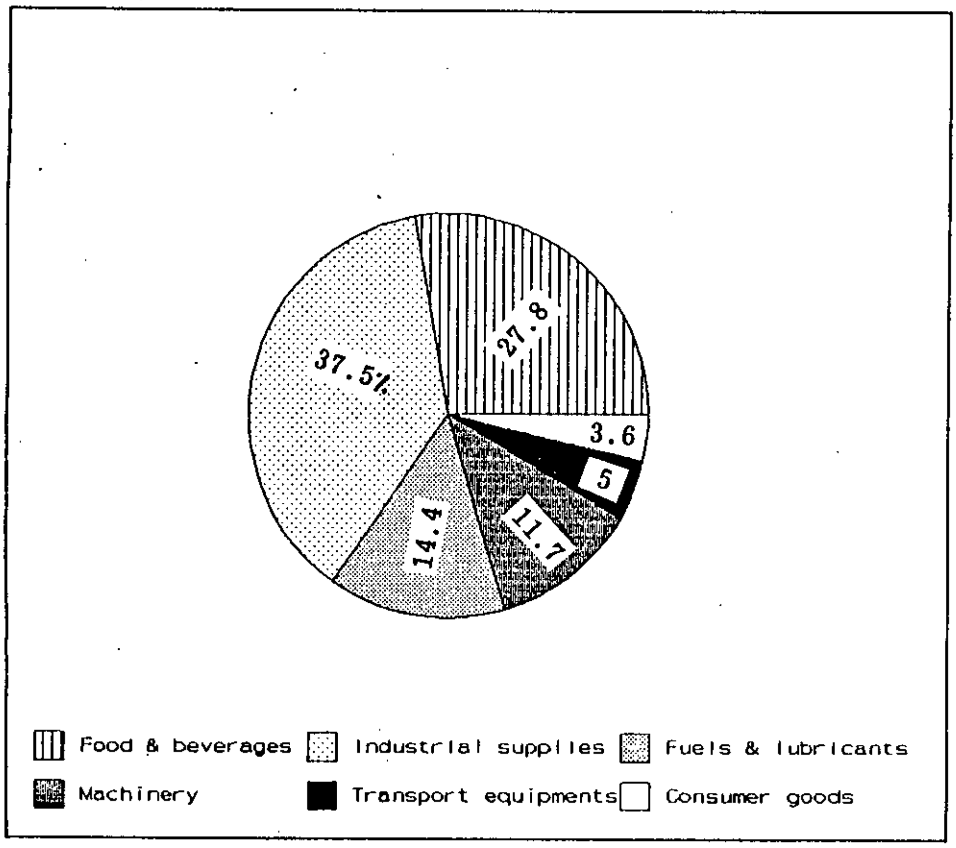


Figure 1.1 - Pie-chart of imported commodities in Bangladesh (1990).

Bio-gas contains Methane (60 - 65%) and by further Methanation process its percentage can be increased making it close to natural gas. Hence research on the use of natural gas in SI engines, which contains 95 - 99% Methane, is also relevant for bio-gas. Bio-gas can be produced from waste products and they are of renewable type. In developing countries, in remote and undeveloped areas bio-gas can play a very significant role in solving the energy crisis. In remote areas where supply of petrol and diesel are irregular and expensive, small scale power plants run on bio-gas may be an economic alternative.

The present work is related to the use of natural gas in spark ignition engine. The main concern of the study is to investigate the performance and the spark advance phenomena of a natural gas fuelled petrol engine.

Using Natural gas as the alternative fuel, the study of a particular petrol engine (1600 cc, Toyota Corona Car engine), has the following objectives:

(1) To generate the performance curves for the gas run engine at a constant speed.

(2) To examine the effect of spark advance on the performance of the engine.

(3) To study individually the centrifugal and vacuum advance requirements for optimum power of the engine.

CHAPTER – 2

LITERATURE REVIEW

(A) ALTERNATIVE FUELS FOR IC ENGINES

(B) SPARK IGNITION ENGINE

(C) USE OF NATURAL GAS IN SPARK IGNITION ENGINES

LITERATURE REVIEW

A. ALTERNATIVE FUELS FOR IC ENGINES

Gasoline and gas oil are the conventional fuels for spark-ignition (SI) and compression-ignition (CI) engines, respectively. Following the first oil crisis of 1973, various alternative fuels have been tried both in SI and CI engines. These fuels are gaseous, liquid or even solid. Fuels in gaseous or liquid form are commonly used in IC engines. The use of alternative fuels in IC engines depends on their physical properties, combustion characteristics, handling system, storage, availability, safety and above all, the cost. This chapter describes some of the alternative fuels which, from the author's point of view, have prospects to replace petroleum fuel in IC engines in the near future. Table - 2.A.1 gives some comparative properties of conventional and some alternative fuels.

A.1 Natural gas

Natural gas deposits exist either as free gas or in association with crude oil. The main constituent of natural gas is methane. Depending on the geological condition, the gas also contains a percentage of higher hydrocarbons, nitrogen, carbon dioxide, and hydrogen sulphide. The methane percentage can be as high as 99.5 percent (Ravenna, Italy). The presence of hydrogen sulphide tends to form acid after combustion, which is corrosive. Depending on the constituents present other than methane, natural gas needs some pre-treatment. The first treatment involves separation of higher hydrocarbons and water at the well head. Carbon dioxide and/or hydrogen sulphide can be removed by alkaline scrubbing or absorption. The nitrogen content usually remains in the gas untreated.

Some countries have deposits of natural gas. The use of natural gas in IC engine can reduce the dependency on imported petroleum oil. The gas can be derived directly from the well or can be derived from a wide variety of feedstock including the fossil range from petroleum gases to bituminous coal. Some of the gasification processes appear as follows:

A.1.1 Catalytic Steam Reforming of Light Hydrocarbons

At relatively low temperature, between 450 and 500°C, and in the presence of very active catalysts, light hydrocarbons may be partially converted by a process of hydrolysis into methane and carbon dioxide, together with hydrogen which can then help to convert the carbon dioxide to additional methane. At appropriate conditions, were the proportions of methane and carbon dioxide are equal, the reaction for a paraffinic feedstock is represented as follows

$$C_xH_{2x+2} + xH_2O \rightarrow \frac{1}{2}xCH_4 + \frac{1}{2}xCO_2 + (1 + x)H_2$$

Methanation of carbon oxides with hydrogen is shown as

$$CO_2 + 4H_2 \rightarrow CH_4 + 2H_2O$$

$$CO + 3H_2 \rightarrow CH_4 + H_2O$$

and these subsequent steps may be undertaken separately. The overall process is carried out at a relatively high pressure of about 20 bar, and careful control needs to be exercised over operating conditions and fuel contaminant concentrations. Suitable feedstock for this method include LPG, virgin naphtha and NG condensates which are generally free from catalyst poisoning sulphur, unreactive aromatic and the unsaturated hydrocarbons.

A.1.2. Hydrogasification of Middle Distillates and Light Residues

The direct bonding of hydrocarbons with hydrogen may be effected by hydrogenolysis at high pressure and low temperature, represented for the heavier C_xH_{2x} feedstock as follows

$$C_xH_{2x} + xH_2 \rightarrow xCH_4$$

A catalyst is not normally necessary, but some carbon is formed due to molecular cracking, particularly with heavier feedstock.

Suitable feedstock include gas oils, diesel fuels and light fuel oils which are generally free of sulphur, avoiding expensive removal of hydrogen sulphide, and of aromatics which lower the gasification efficiency.

A.1.3 Partial Oxidation of Residual Oils and coal

In this method, carbon is removed from heavy hydrocarbon feedstock by oxygenolysis at high temperature to give methane and carbon monoxide, the latter being converted subsequently by methanation. The overall reaction for the heavier C_xH_{2x} feedstock may be represented as follows

$$C_xH_{2x} + \frac{1}{2}O_2 \rightarrow \frac{1}{2}CH_4 + \frac{1}{2}CO_2$$

A catalyst is not normally necessary. Suitable feedstock include medium and heavy fuel oils, and stream coals, preferably low in ash, sulphur, nitrogen and moisture.

A.2 Bio-gas

Bio-matter comprises all natural materials associated with living organisms, including terrestrial and marine vegetable matter of all kinds from algae to trees, together with animal tissue and manure. As in the conversion of refuse, anaerobic digestion of biomatter to methane can be achieved through bacterial action, and is well suited to organic wastes of high moisture content (above 45 percent). This method is utilized in the treatment of sewerage sludge, which yields a biogas rich in methane (about 67 percent) together with carbon dioxide and traces of nitrogen, hydrogen and oxygen. The gas also contains hydrogen sulphide which provides the gas its bad odor, and when burned forms sulfuric acid which is highly corrosive. The slurry from the bio-gas plant can be used as a fertilizer.

A.3 Hydrogen

Among all the alternative fuels, hydrogen is the most clean burning fuel. Hydrogen when burned reacts with oxygen to produce energy with water as the only combustion product. Hydrogen is diatomic and it is the simplest molecular structure of a fuel. Hydrogen can be derived in many different ways which include:

i. Hydrogen can be derived from water. The application of electrical energy (direct current) ionizes water to hydrogen and hydroxyl ions, and produces particularly pure hydrogen gas at the cathode, and oxygen gas at the anode. But to make hydrogen by this method requires a cheap electricity supply. Solar cells, which can capture sunlight and convert it to electricity, can be used to split water into hydrogen and oxygen to make a portable fuel. But the cost of electricity generated by solar power, at present is around $4 per watt. But to generate electricity that can make hydrogen fuel compete with fossil fuels, the price will have to come down to about 25 cents per watt.(32)

ii. The major proportion of manufactured hydrogen gas is produced currently from sulphur cleaned natural gas by catalytic steam reforming in the presence of nickel oxide catalyst pellets.

$$CH_4 + H_2O \rightarrow CO + 3H_2$$

The hydrogen/natural gas volume ratio is 2.13 and the thermal efficiency of the process approaches 70 percent.

iii. Hydrogen can be produced by direct conversion of coal with oxygen and steam

$$3C + O_2 + H_2O \rightarrow 3CO + H_2$$

followed by the water-gas shift reaction as shown

$$CO(g) + H_2O(g) \rightarrow CO_2(g) + H_2(g)$$

and carbon dioxide removal.

A.4 Alcohol Fuels

The monohydric alcohol molecule consists of a hydrocarbon in which one atom of hydrogen has been substituted by a hydroxyl group, thus it is represented by the formula ROH, where R is the remaining hydrocarbon group. Of main interest as alternative fuels are the first four normal members of the alkyl alcohols: methyl (CH_3), ethyl (C_2H_5), propyl (C_3H_7) and butyl (C_4H_9).

Methanol (CH_3OH) was formerly derived from the destructive distillation of wood, but synthetic processes were introduced in 1923 and production now stems from feedstock from methane, carbon dioxide and water derived from naphtha, residual fuel oil, vacuum residue, or from coal. The ICI low-pressure process introduced in 1966, operating at about 50 bar and using a high active copper-based catalyst, led to large increases in plant capacity to the "jumbo" level of over 5000 tons/day. High pressure (300 bar) processes are also used, with zinc-chromium as catalysts.

The next member, ethanol (C_2H_5OH) is currently made from ethylene derived from petroleum, but can be produced in bulk using yeasts at controlled temperature, by fermentation of the carbohydrates from a variety of vegetable sources. These includes the starches from artichokes, potatoes, cassava and cereal grains, the sugars from sugar cane and beet, fruit juices and molasses, the cellulosic materials from wood, tropical grasses, straw and the waste sulphite liquor from the production of wood pulp for paper mills. Propanol (C_3H_7OH), butanol (C_4H_9OH) and other higher alcohols are derived either from fermentation of corn, maize or molasses, or from synthetic processes using lower alcohols, acetone, aldehydes, and so on.

A.5 Vegetable Oil

The vegetable oils comprise natural esters of the trihydric alcohol, glycerol ($HOCH_2CHOHCH_2OH$) with long straight-chain fatty acids (RCHOOH), where R hydrocarbon radial varies from about $C_{15}H_{31}$ to $C_{17}H_{35}$, the approximate concentrations being 12 percent paraffinic and 87 percent olefinic (including 68 percent diolefinic). These oils are generally miscible with gas oil, consequently solution-type hybrid fuels have been tested. Up to about 78 percent energy substitution has been reported with cottonseed oils in solution with gas oil (23) and successful road operation achieved by blending the gas oil with waste soyabean oil derived from deep-frying food preparation (8) (9) . Complete substitution is also reported with peanut oils, soyabean oil and sunflower oil (10) (14) (15). With vegetable oil the ignition delays tend to increase. Emissions of NO_x can be reduced, but smoke may be doubled (10).

Table - 2.A.2 gives comparative properties of some vegetable oil. Two important properties which should be met by vegetable oil are :

a. Ignition Quality: The term ignition quality is used to cover, loosely, the ignition temperature-versus-delay characteristics of a fuel when used in an engine. Good ignition quality means a short delay at a given speed, compression ratio, air inlet, and jacket temperature. The ignition quality of a fuel is usually indicated by its cetane number. Higher cetane number indicates case of starting from cold and smoothness of running (i.e. elimination of diesel knock).

For high speed diesel engine the cetane number required is about 50, for medium speed engines about 40, and for slow speed engines about 30. The cetane number of the vegetable oil should be close to that for diesel fuel. When the cetane number is lower than fuel, the engine will not start with vegetable oil.

b. Viscosity: Viscosity of the fuel is important as it affects the transfer of oil from the tank to the injector, the operation of the high pressure injection pump and the injector which in turn affects the atomization of fuel and mixing of fuel and air.

Too low viscosity can cause the following problems:

- Low lubricating value.
- Excessive wear and even seizing of the fuel pump plunger.
- Producing an excessive leakage past the plunger.

High viscosity can cause the following problems:

- Difficulties in transferring and delivering fuel by pump or gravity.
- Reduce atomization and increase penetration of the fuel on injection, which can cause incomplete combustion.

Some high viscosity vegetable oil may need pre-heating for their use as alternative fuel in the diesel engine.

B. SPARK IGNITION ENGINE

Spark ignition(SI) engines are the internal combustion engines which are generally supplied with air-fuel mixture, called the charge, which are mixed outside the combustion chamber. By the time the ignition occurs in the combustion chamber, the liquid fuel becomes essentially gaseous and well mixed with air. The combustion in the SI engine is initiated by a spark and characterized as combustion in homogeneous and gaseous mixture.

B.1 COMBUSTION IN SI ENGINE

In the SI engines, pre-mixed air-fuel mixture is ignited by an electric spark. At the commencing of ignition, a definite flame front travels from the ignition point and it spreads in a continuous manner outward from the ignition point. Because of turbulent motion of gases inside the cylinder, the shape of the flame front might be irregular. Though the whole combustion process in the engine occurs with great rapidity, there exists a finite time from the start of ignition to the end of the combustion process. The flame front travels through the gas-air mixture at a velocity equivalent to the flame speed.

Combustion starts before the end of the compression stroke, continues through the early part of the expansion stroke, and ends after the point in the cycle at which the peak cylinder pressure occurs. The burning rate of the gas-air mixture increases as the flame front passes through the mixture. This is caused due to larger flame radius and increase in temperature caused by rising pressure and heat transfer from the burned gases, as the rest of the charge burns. When the flame front touches the combustion chamber walls, the flame front breaks and hence, the burning rate decreases. The rate at which fuel-air mixture burns increases from a low value immediately following the spark discharge to a maximum about halfway through the burning process and then decreases to close to zero as the combustion process ends.

B.2 SPARK ADVANCE

In spark ignition engines, a high voltage electric spark at correct timing is needed to ignite the charge in the combustion chamber. The timing of the spark is measured by the crank angle. For best power and efficiency, spark occurs at a certain crank angle before the TDC (called, spark advance), so that the peak pressure occurs a few degrees of crank angle after TDC.

B.2.1 NECESSITY OF SPARK ADVANCE

In the SI engines, the pre-mixed air-fuel mixture is ignited by the spark created by the spark plug. As soon as the ignition commences, a definite flame front travels from the ignition point and it spreads in a continuous manner outward from the ignition point. Because of the turbulent motion of gas inside the cylinder, the shape of the flame front might be quite irregular. The whole combustion process in the engine occurs with great rapidity, but a finite time is needed from the start of the ignition to the end of the combustion process.

Generally the spark occurs at such time, so that the peak pressure rise is attained between 15 - 20 degrees after the piston passes the TDC in the power stroke (26). This ensures best power and efficiency of the engine. To get this , spark is done before the piston reaches the TDC in the compression stroke. The timing of the spark is measured by the crank-angle before TDC, and is called spark advance. The amount of spark advance needed depends on various factors like - engine speed, load, fuel characteristics, and temperature.

As any given SI engine is required to operate over a wide range of load and speed, a contineously changing spark advancing mechanism is needed to get best performance of the engine at different speeds and loads.

B.2.2 Factors Influencing Spark Advance :

The following factors govern the spark advance requirement of the engine : (For each case other conditions are considered to be constant)

Engine load : Lower load - As the throttle opening is less, lower is the intake pressure and lower combustion chamber temperature at the end of the compression stroke make combustion process slower, so more spark advance is needed.
Higher load - As the throttle opening is greater, higher is the intake pressure and higher combustion chamber temperature at the end of the compression stroke make the combustion process faster, so less spark advance is necessary.

Engine Speed : Lower speed - Greater time is available for combustion so less spark advance is necessary.
Higher speed - Less time is available for combustion so greater spark advance is needed.

Fuel Character : Low volatile fuel - Slower combustion, so greater spark advance is needed.
High volatile fuel - Faster combustion, so less spark advance is needed.

Air-Fuel Mixture: Below stoichiometric mixture, for leaner mixture greater spark advance is needed, as the combustion is slower.
Above stoichiometric mixture, for leaner mixture less spark advance is needed, as the combustion is faster.

Compression Ratio: Low compression ratio - Slower combustion rate and more spark advance is needed.

High compression ratio - Faster combustion rate and less spark advance is needed.

Engine Temperature: Cold Engine - Combustion is slower, so more spark advance is needed.
Hot Engine - Combustion is faster, so less spark advance is needed.

B.3 SPARK ADVANCE CONTROLLING MECHANISM

In modern SI engines a combined system of centrifugal and vacuum advance is used in controlling the spark-timing according to the load and speed of the engine automatically. Distributors are provided with a centrifugal-advance mechanism to produce advance based on engine speed. Fig - 2.B.1 demonstrates such a mechanism. It has a pair of pivoted advance weights with weight springs. At low speed, the spring holds the weighs in. As the speed increases the centrifugal force on the weights move them out against the spring tension. This movement causes the cam assembly to move ahead. Higher the engine speed, faster the shaft turns and further ahead the cam assembly is advanced. This causes the contact points to open and close earlier. Therefore the ignition coil produces its high-voltage surge earlier, hence the sparks appears earlier in the combustion chamber.

When the engine is operated at partly open throttle, there is a vacuum in the intake manifold. The partly closed throttle valve prevents the maximum amount of air-fuel mixture from entering the intake manifold. So less air-fuel mixture enters the engine cylinders on the intake stroke. With less air-fuel mixture in the cylinders, the mixture is not compressed as much during the compression stroke as there is less mixture to compress and also the temperature inside the cylinder is less. These results the mixture to burn at a lower rate when ignited. Unless there is some additional spark advance when operating at part throttle, best power and efficiency from the mixture will not be realized. The piston will be moving down after TDC before the

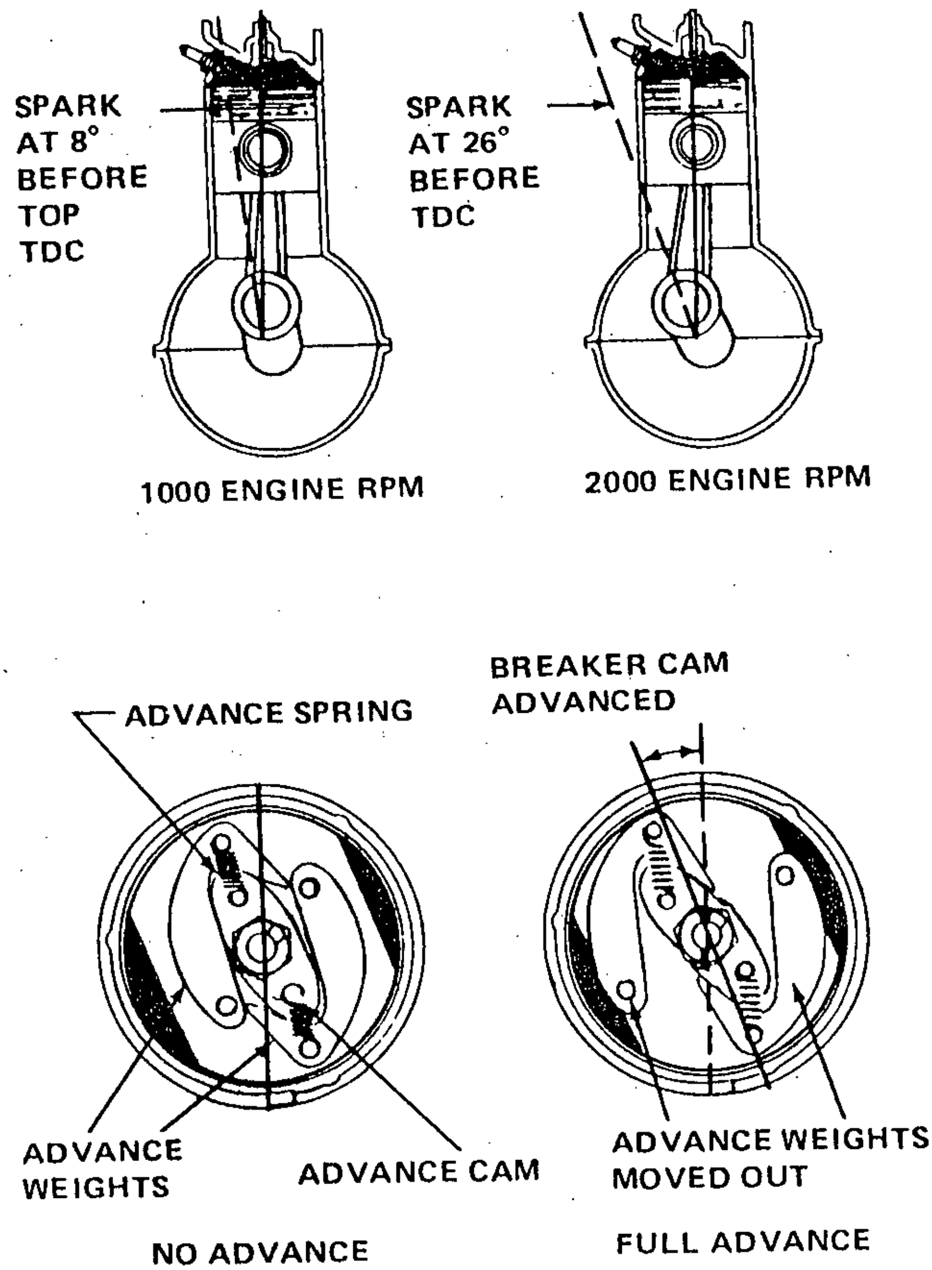


Fig -2.B.1 Centrifugal-advance mechanism in no advance and full-advance position. In the typical example shown, the ignition is timed at 8 degrees before TDC on idle. There is no additional centrifugal advance at 1000 rev/min. There is 26 degrees total advance (18 degrees centrifugal plus 8 degrees due to original timing) at 2000 rev/min. (6)

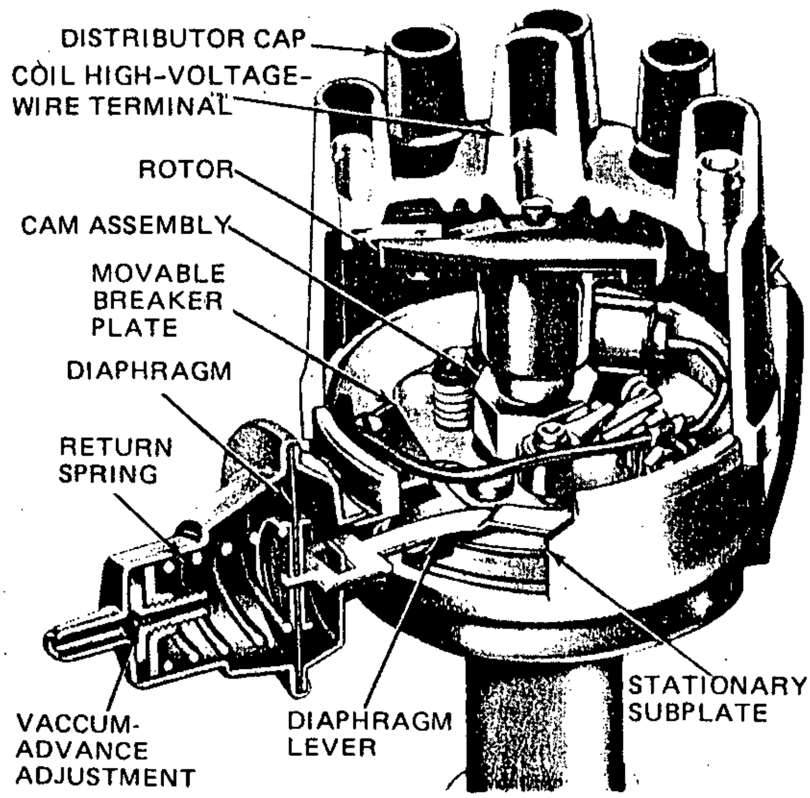


Fig – 2.B.2 A cutway vacuum-advance mechanism. The diaphragm is linked to the movable breaker plate.

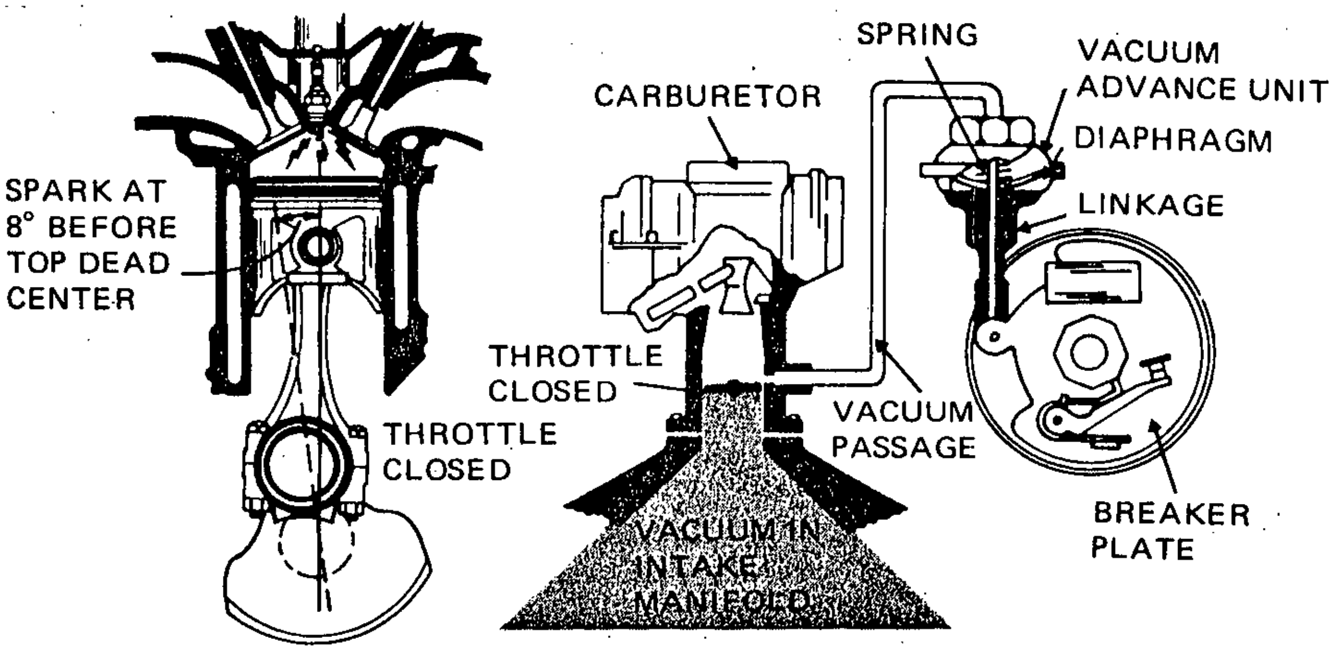


Fig - 2.B.3 (a) When the throttle valve is closed, there is no vacuum advance.

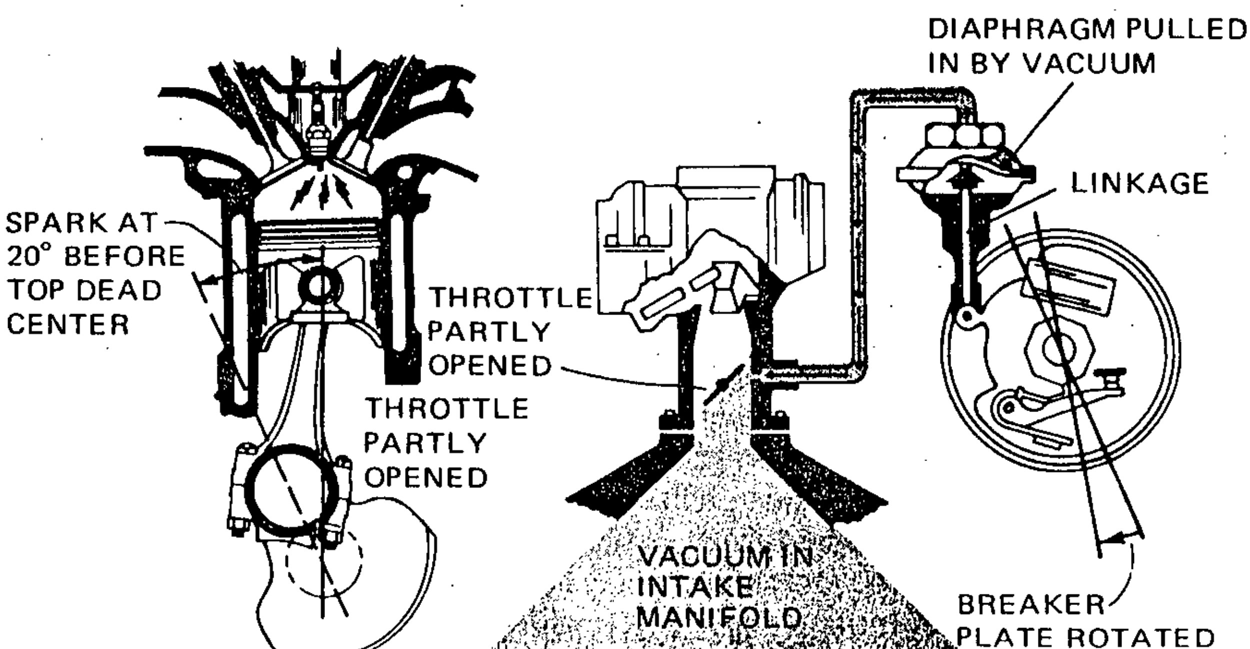


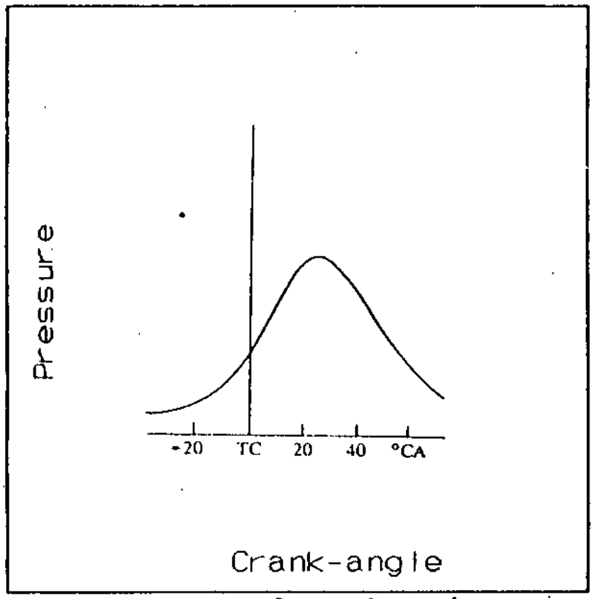
Fig - 2.B.3(b) When the throttle valve swings past the port, a vacuum is admitted to the vacuum - advance mechanism on the distributor. The breaker plate with the points, is rotated to advance the spark.

combustion pressure reaches the maximum. As a result the piston would keep ahead of the pressure rise and the power stroke would be weak. Fig - 2.B.2 and 2.B.3 shows a typical mechanism of vacuum advance based on part-throttle operation, using the intake manifold vacuum. It has a spring loaded flexible diaphragm linked to the movable breaker plate in the distributor. A vacuum passage connects the diaphragm chamber to the carburetor. When the throttle valve is closed, there is no vacuum applied to the vacuum passage as it is above the closed throttle valve. When the throttle valve is partly opened, it moves past the opening of the vacuum passage. As the intake manifold vacuum is applied through the passage it pulls the diaphragm outwards against the spring force. This rotates the breaker plate, so the contact breaking points open and close earlier resulting in advanced sparks. When the throttle valve is wide open, intake manifold vacuum is almost zero, therefore the vacuum advance mechanism produces no advance.

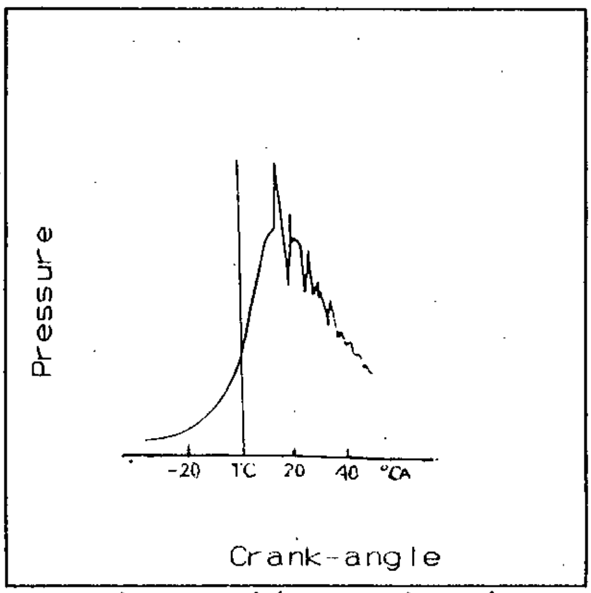
In case of engines with electronic ignition system the spark timing is varied electronically according to load and speed requirements.

B.4 KNOCKING IN THE SI ENGINE

In a SI engine, air and fuel are mixed outside the combustion chamber, then the mixture is drawn into the engine cylinder during the suction stroke of the engine. The mixture is compressed by the piston in the compression stroke. The combustion of the gaseous air-fuel mixture is ignited by spark created by the spark plug. The combustion is then characterized by the more or less rapid development of a flame that starts from the ignition point and spreads in a continuous manner outward from the ignition point. When this spread contines to the end of the chamber without abrupt change in its speed or shape, combustion is called normal. When the mixture appears to ignite and burn ahead of the flame, the phenomenon is called auto-ignition. When there is a sudden increase in the reaction rate, accompanied by measurable pressure waves, the phenomenon is called knocking. Fig - 2.B.4 shows the pressure versus crank-angle diagram for a non-knocking and for a knocking fuel. In the expansion stroke, the normal cycle shows a smooth change in pressure,



(a) Normal combustion
spark 28°BTDC



(b) Knocking combustion
spark 32°BTDC

Figure 2.B.4 : Pressure versus crank-angle diagrams. (a) pressure vs. crank-angle without knocking; (b) pressure vs. crank-angle with knocking. 4000rev/min, wide-open throttle, $3.81 \times 10^{-4} m^3$ displacement single-cylinder engine (Heywood).

while the knocking cycle shows severe pressure fluctuations, indicating a vibrating motion of the gases.

Spark knocking in the SI engine is due to auto-ignition of the end gas, which is that part of the charge that has not yet been consumed in the normal flame-front reaction. When the knocking of the end gas occurs, it is because of the compression of the end gas by the expansion of the burned gas parts, which raised its temperature and pressure to the point where the end gas auto-ignites. When the reaction of the auto-ignition is sufficiently rapid and a sufficient amount of end gas is involved, knocking can be observed. Such a reaction will result in a high local pressure, and the portion of the gas in question will subsequently expand rapidly, sending a pressure wave across the chamber. This pressure wave will be reflected from the walls, and the frequency of this wave will decrease as the piston moves outward in the expansion stroke. Such a process results in significant noise generation within the structure of the engine.

Octane Requirement

The tendency of the engine to knock depends on the quality of the fuel to be used. One of the accepted scale for evaluating the resistance to knocking of fuel is the Octane Number. The octane number (ON) scale is based on two hydrocarbons which defines the end of the scale. Normal heptane (n-C_7H_{16}) which knocks easily, has an octane number of zero, and iso-octane (C_8H_{18}:2,2,4-tri methyl penate) which has a great resistance to knock, has an octane number of 100. Blends of these two hydrocarbons define the knock resistance of intermediate octane numbers: e.g., a blend of 10 percent n-heptane and 90 percent iso-octane has an octane number of 90. A fuel's octane number is determined by measuring what blend of these two hydrocarbons matches the fuel's knock resistance.

Performance Number

Another way of expressing the knocking tendency of the fuel is by Performance Number (PN). The number is defined as the ratio of klimep with the fuel in question to klimep with iso-octane when inlet pressure is used as the dependent variable. In this method, with a given fuel the inlet pressure of the engine is varied until knocking is observed. The indicated mean effective pressure measured at this point is abbreviated as klimep. Higher PN means greater resistance to knock.

C. USE OF NATURAL GAS IN SPARK IGNITION ENGINES.

Born and Durbin (7) showed that, for gasoline engines ran on natural gas, at the equivalence ratio of 1.0 (i.e. the stoichiometric mixture), the spark advance needed for best power is 32° before TDC (btdc). While gasoline needs 20 degree spark advance. Natural gas needed more spark advance because of its slower flame speed than that of gasoline. The amount of air intake into the engine was less for running with natural gas, because natural gas replaced more air than the liquid gasoline when they are mixed with air and also evaporation of the liquid gasoline fuel in the air increased the engine's air mass breathing capacity. Due to less flame speed and less air intake into the engine, the maximum power of the engine was less for running it with natural gas than running it with gasoline (petrol). Their results also showed that pollutants for the natural gas-fuelled engine were less than those of gasoline.

G.A. Karim and I.A. Ali (17) tested a single cylinder spark ignited research engine fuelled with natural gas. Their results showed that the exhaust NO_x always decreased as the spark timing was retarded. The emission of CO increased with decreasing mixture temperature and also with increasing compression ratio.

Ets Duvant of Valenciennes, France (13) has developed and commercialized complete assembly lines for the conversion of bio-mass energy into electricity. Duvant's generator set has power extending from 100 up to 1000 KVA.

R.H. Twring (27) studied the prospect of using Alcohol, Methane based gases, Hydrogen, Produced gas and Liquified Petroleum gas as alternative fuel in SI engines. As an alternative fuel methane offers several advantages as a fuel for the SI: high octane number (RON = 120, MON = 120), good cold weather starting, and low exhaust emission. But its disadvantage is that, as a gaseous fuel, it displaces much more intake air than gasoline which enters mostly as droplets and because of this, the engine's volumetric efficiency and maximum power output is reduced.

TNO, a scientific research institute have been doing research for the application of LPG, Natural gas, Bio-gas, Producer gas, Alcohol fuels in the engines for vehicles as well as for power generation. According to their results, using natural gas as on engine fuel could reduce usual exhaust emission (CO, HC, NO_x, Particles). The emission of CO_2 could be reduced by some 26 percent (assuming the same efficiency, compared to diesel oil and petrol). This is caused by the high hydrogen content of methane. TNO also did some research on the safety of gaseous fuels (28). In one of their tests, a passenger car with a speed of 70 km/h was crashed straight into the LPG tanks which were mounted without any special protection under the near end of a bus. From this test programme it was learned that LPG system in collisions are no less safe or even safer than existing gasoline and diesel oil systems.

Balasubramanian, Sarma, Sridhara, and Ramchardra (4) investigated the performance of compressed natural gas (CNG) as an alternative fuel for vehicular diesel engine. They also performed field tests and investigated the technical feasibility of using CNG in automotive diesel engines. The study showed that natural gas (CNG) is a very attractive fuel for Internal combustion Engine (Petrol and Diesel) applications in India.

Nagosh and Samaga (21) optimized a biogas fueled SI engine. In their test result, it is reported that engine performance in general is more satisfactory at advanced ignition of 35° BTDC and at larger spark-plug gap (0.45mm). The tested engine showed a minimum requirement of bio-gas at the rate of 0.7 m^3/bhp-hr. The biogas contained 59% methane and 41% other combustible and CO_2.

Cambell (5) conducted experiments on the use of CNG in SI engine and in dual fuel diesel engine. They converted a single cylinder diesel engine to a natural gas fuelled SI engine. The compression ratio was reduced from 17:1 to 12:1. Test results showed that exhaust temperature decreased significantly with ignition timing advance. An increase in the ignition advanced from 2° BTDC to 15° BTDC resulted in decrease of exhaust temperature from 702°C to 620°C at 2000 rev/min at rated load. It was also mentioned that exhaust temperature in natural gas fuelled engine was overall high compared to diesel fuel operation. Exhaust emission was found to follow the typical spark engine trend.

Bari (2) conducted an experiment on the study of the effect of dual fuel on some performance parameters of a single cylinder small diesel engine. Three types of mixers (Direct, L-shaped and cross-shape) were used in the experiment and among those the cross-shape mixer was found to be more efficient.

Sayed Ali Mollah (20) studied the performance of a single cylinder SI engine (converted from a diesel engine), using CNG as the alternative fuel. He also studied the effect of varying spark advance at 6,7.5,9,12 and 15 degrees before the TDC. Results showed that at greater spark advance the performance was better. Exhaust gas temperature was found to be higher for natural gas. The exhaust analysis showed that amount of CO, CO_2, HC were less but formation of NO_x was higher.

CHAPTER – 3

DESCRIPTION OF EXPERIMENTAL SETUP AND PROCEDURE

DESCRIPTION OF THE EXPERIMENTAL SETUP

Experimental set-up consists of engine test-bed, petrol and natural gas supply system, different metering and measuring systems of air, petrol and gas flow, with additional facilities of spark advance and temperature measurements. Fig : 3.1 - 3.8 show the photographs and schematic diagrams of the experimental setup, which is described in brief below :

A 4-stroke, 4-cylinder, inline, water cooled petrol engine (1600 cc) of a Toyota Corona car was used for the test purpose. The engine was provided with an electrical spark-ignition system. Photograph of the engine is shown in Fig - 3.1.

A model no. TFJ-250L water-brake type dynamometer was used to measure the engine performance. The dynamometer unit contained an electronic load cell transducer to measure the force acting on the dynamometer and a magnetic type tachometer to measure the speed. Two LED displays were provided to show load and rotational speed readily. The engine was connected with the dynamometer assembly through a spline shaft and universal joints. The connecting structure was covered with a metallic guard fence as safety measure. Fig: 3.2 shows the photograph of the engine and the dynamometer connection. To reduce noise a silencer box was connected to the exhaust pipe of the engine. Fig - 3.9 shows the block diagram of the various components of the engine test bed assembly.

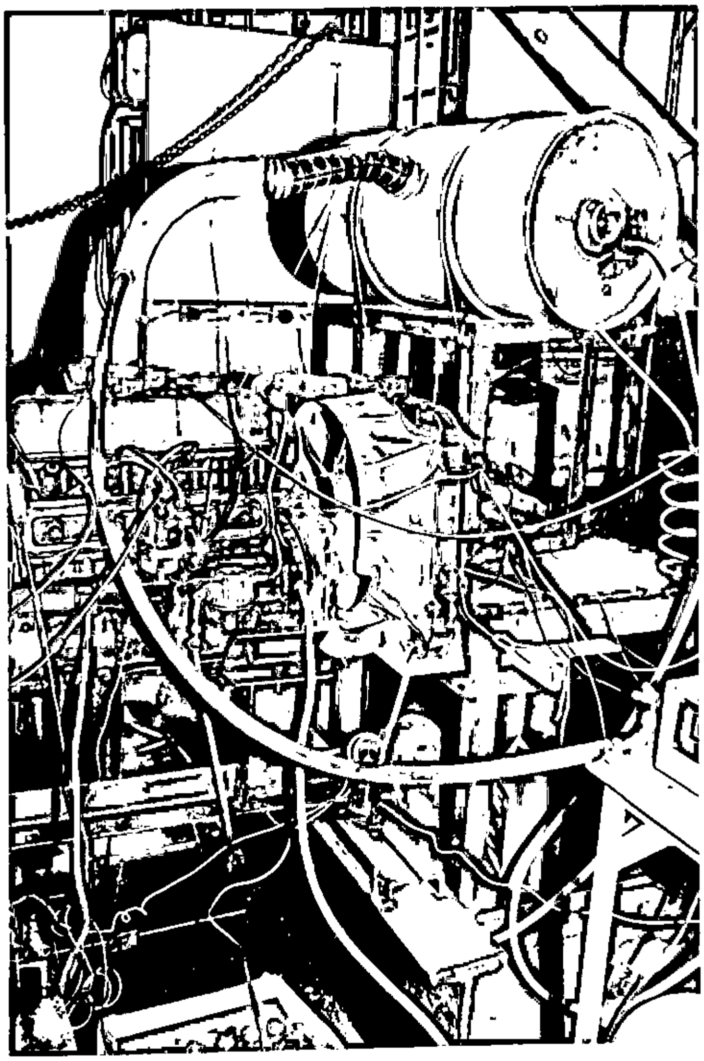


Fig : 3.1 Photograph of the test engine.

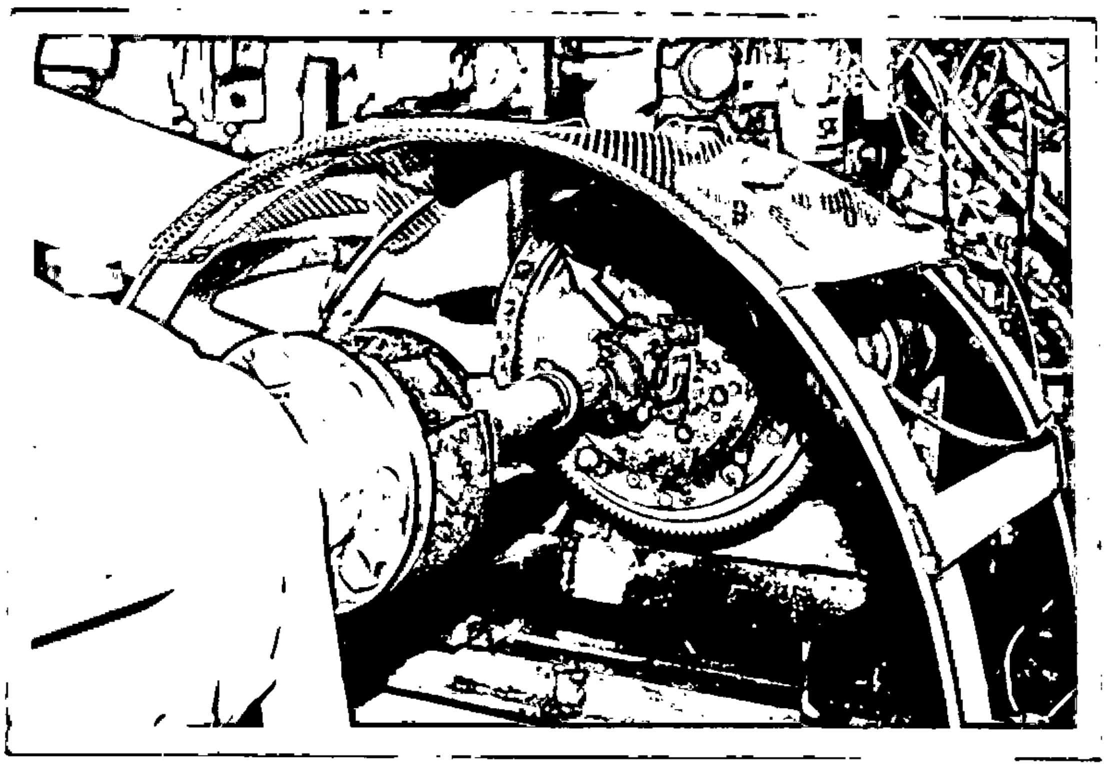


Fig : 3.2 Photograph of the engine and Dynamometer coupling.

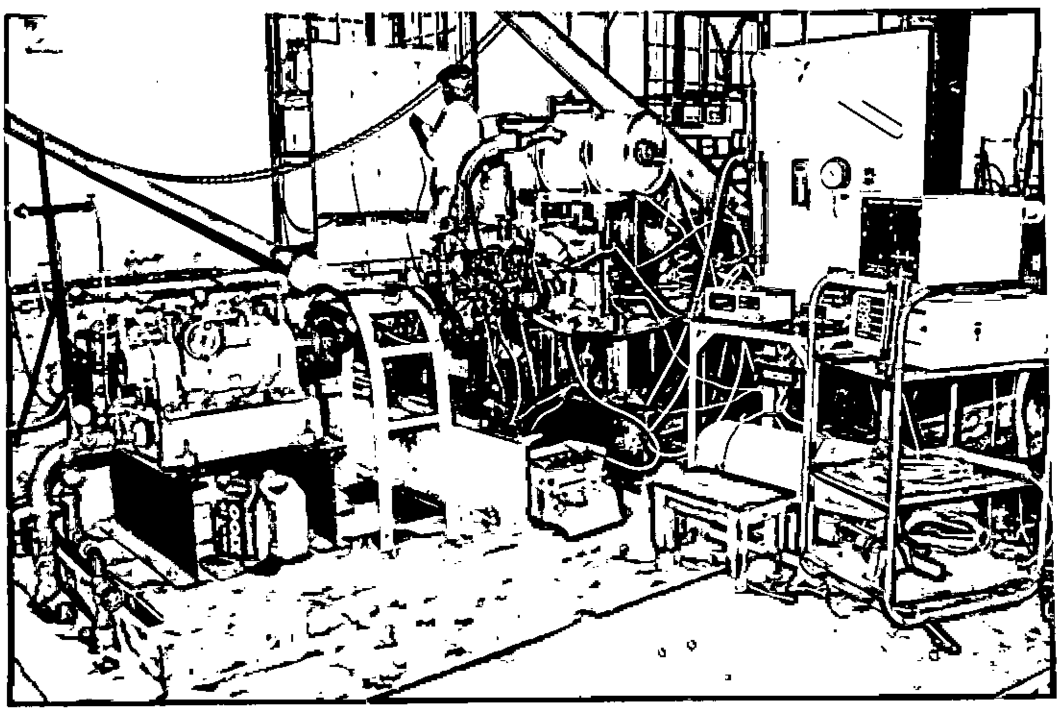


Fig : 3.3 Photograph of the Engine Test Bed.

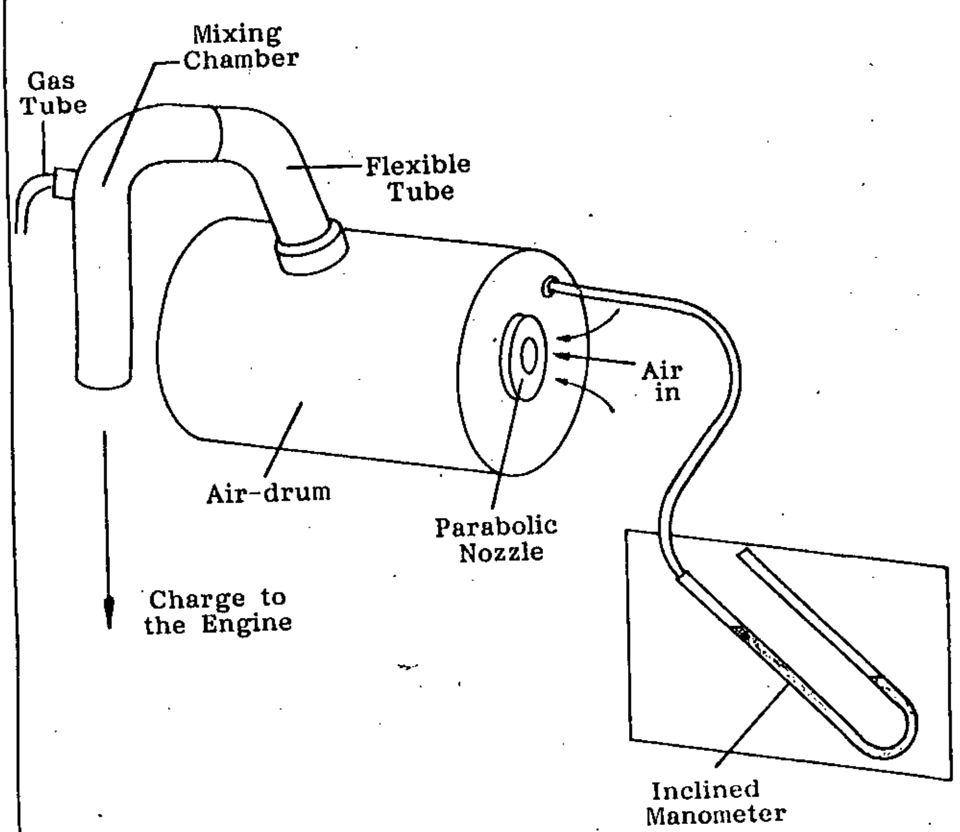


Fig : 3.4 Schematic diagram of the Air - flow line.

Three separate metering systems were used to measure the flow rates of air, petrol and natural gas. The air line consists of a drum, which contains a parabolic nozzle at one end and the other end was connected to the carburetor through a flexible pipe connection. The gas supply tube was connected to the air intake tube through a mixing chamber. The parabolic nozzle was replaceable and nozzles of various sizes (0.5" & 0.75" dia) could be used according to the engine requirement. A piezometric tube, placed on the air-drum was connected to an inclined manometer. The inclined manometer was placed at an angle of 30° with the horizontal and mercury was used as the manometric fluid. Figure 3.4 shows the schematic diagram of the air-flow line.

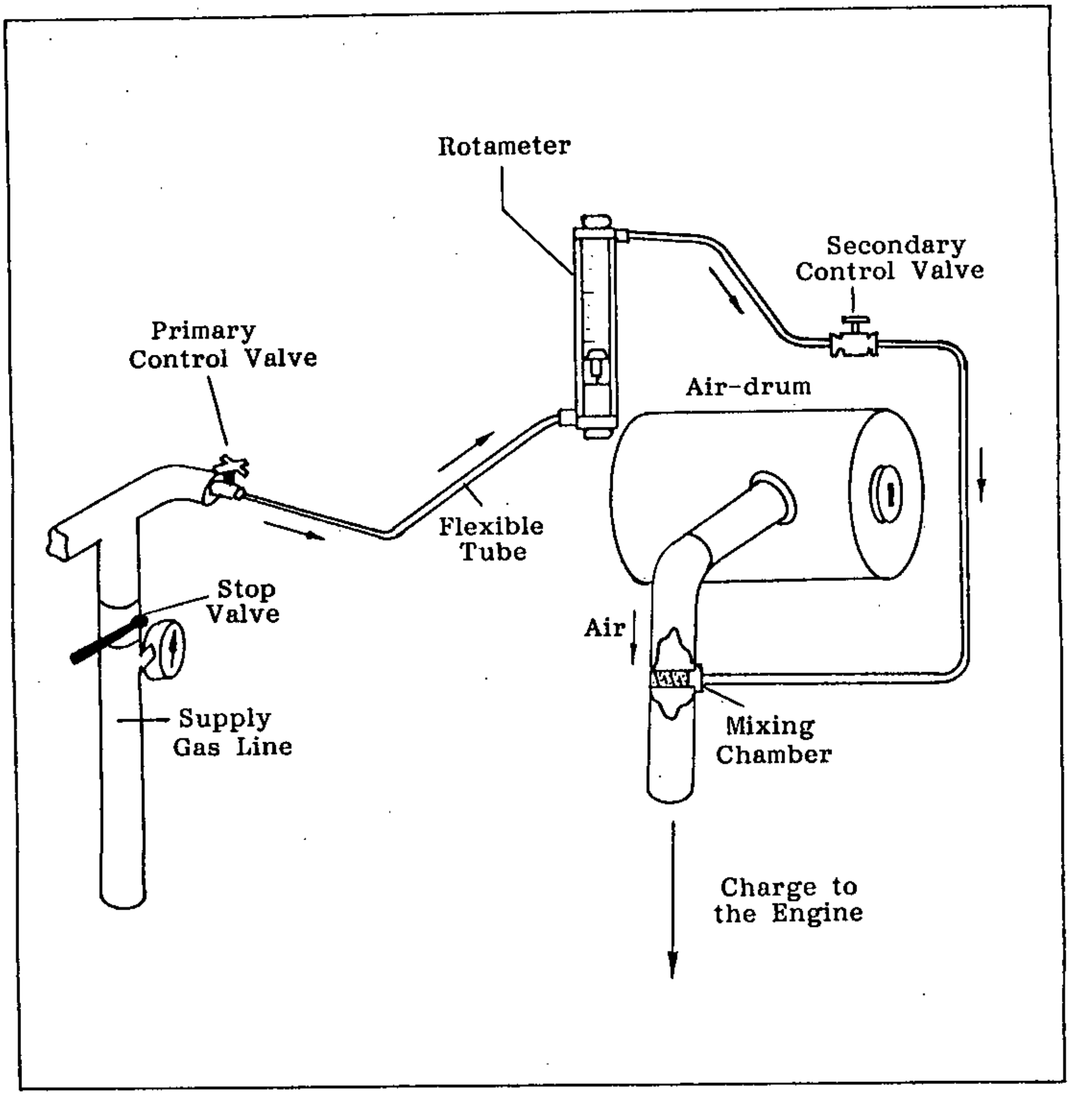


Fig - 3.5 Schematic diagram of the Natural gas flow line.

The source of natural gas used for the engine was a 2 inch commercial Titas gas line available at the laboratory. The gas was passed to the mixing chamber through a stop valve, a primary control valve, a rotameter (variable area flow meter) and a secondary control valve, using a flexible rubber pipe. The rotameter was used to measure the volume flow rate of natural gas. The rotameter was calibrated for natural gas prior to the experiment. A schematic diagram of the gas flow line is shown in figure - 3.5. The mixing chamber was cross flow type, where natural gas was passed through a perforated copper tube (having forty-two, 1/16" dia holes around the surface) placed at right angles to the air flow.

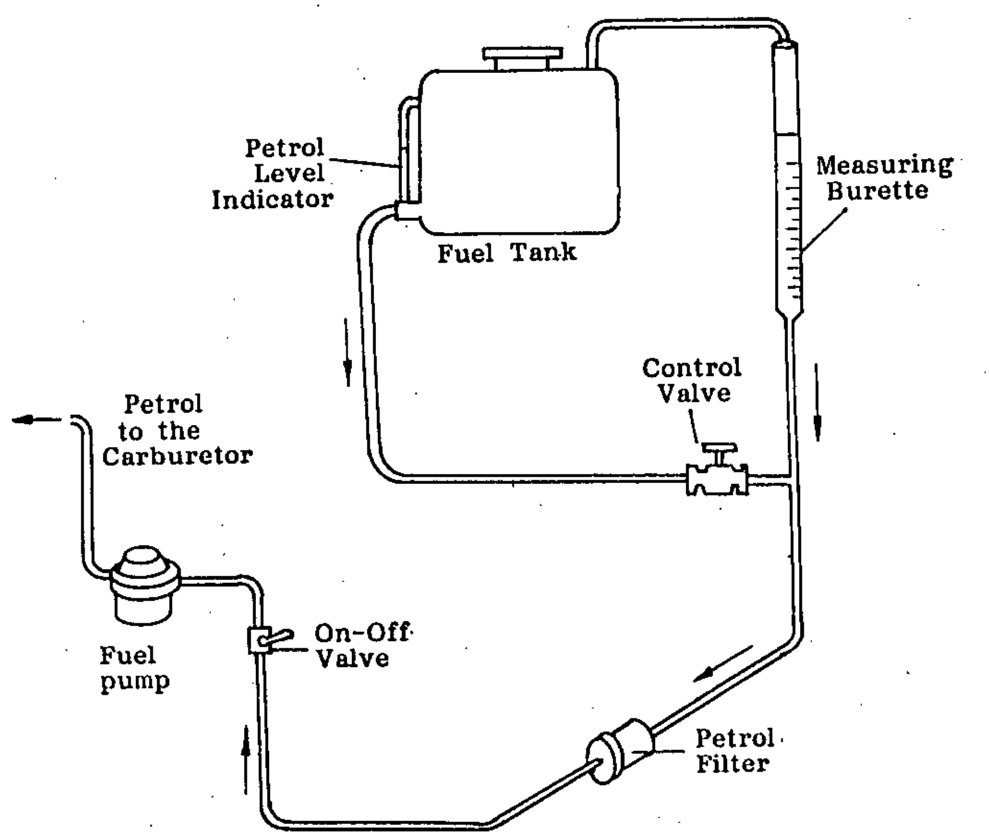


Fig : 3.6 Schematic diagram of the petrol flow line.

The petrol line consisted of a air-tight petrol tank, measuring burette, fuel filter and flexible tubing connected to the fuel pump of the petrol engine. An on - off valve was connected prior to the fuel pump to switch off the petrol flow when the engine ran on natural gas. The acceleration paddle control mechanism of the engine was replaced by linear motion of a spring-locked screw adjuster. The petrol flow rate was determined by closing the valve and recording the time required for a certain volume of fuel flown from the burette. Fig - 3.6 shows the schematic diagram of the petrol line.

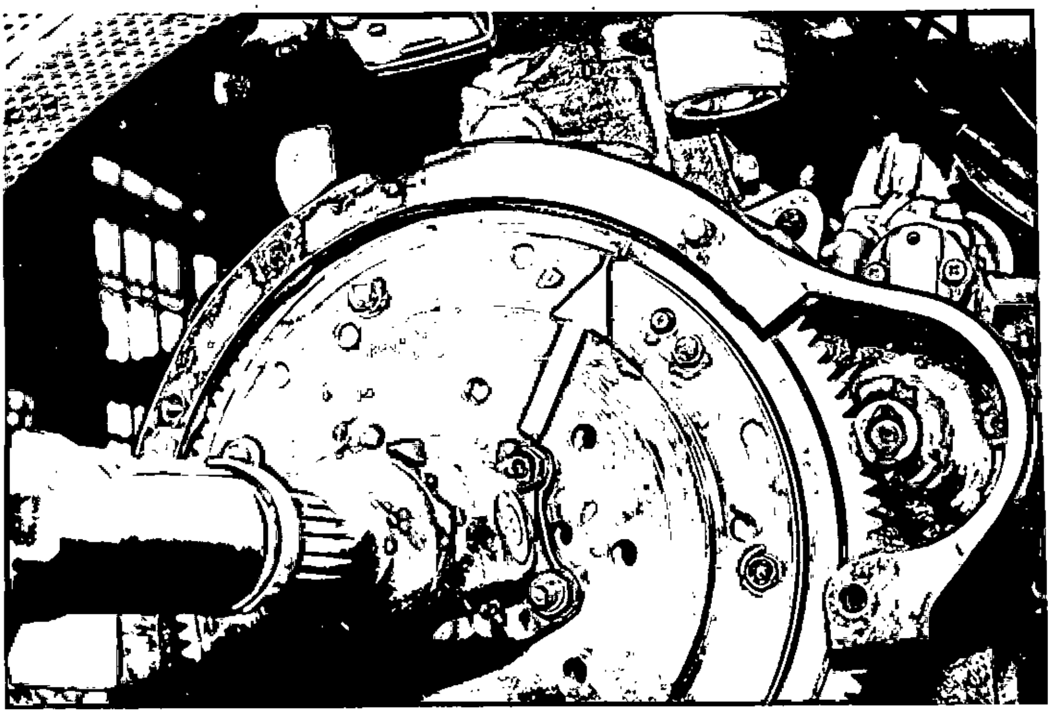


Fig : 3.7 Photograph of the Spark Advance measuring mechanism.

The spark advance measuring mechanism consists a high voltage electric bulb, cord connections and a light reflecting marker. The high surge carrying electric cord from the distributer, which is aimed for no. 1 cylinder of the engine, was connected to the input cord of the special electric bulb. The output cord of the electric bulb was connected to the spark plug of the first cylinder (firing order 1-3-4-2). Each time a high voltage surge from the ignition coil passes through this connecting system, the bulb blinks. A light reflecting marker was placed on the flywheel plate indicating the TDC of the first cylinder. A stationary scale was placed on the engine body (as in Fig - 3.8), which showed the angle in degrees before top dead center (btdc). Each time the a high voltage surge passes through the circuit, the lamp blinks and the reflector becomes visible. The relative position of the reflector marker with respect to the stationary scale indicates the angle at which the spark occurs before the piston reaches the TDC. The mechanism used is similar to stroboscope in operation. Fig : 3.7 and Fig : 3.8 present the system.

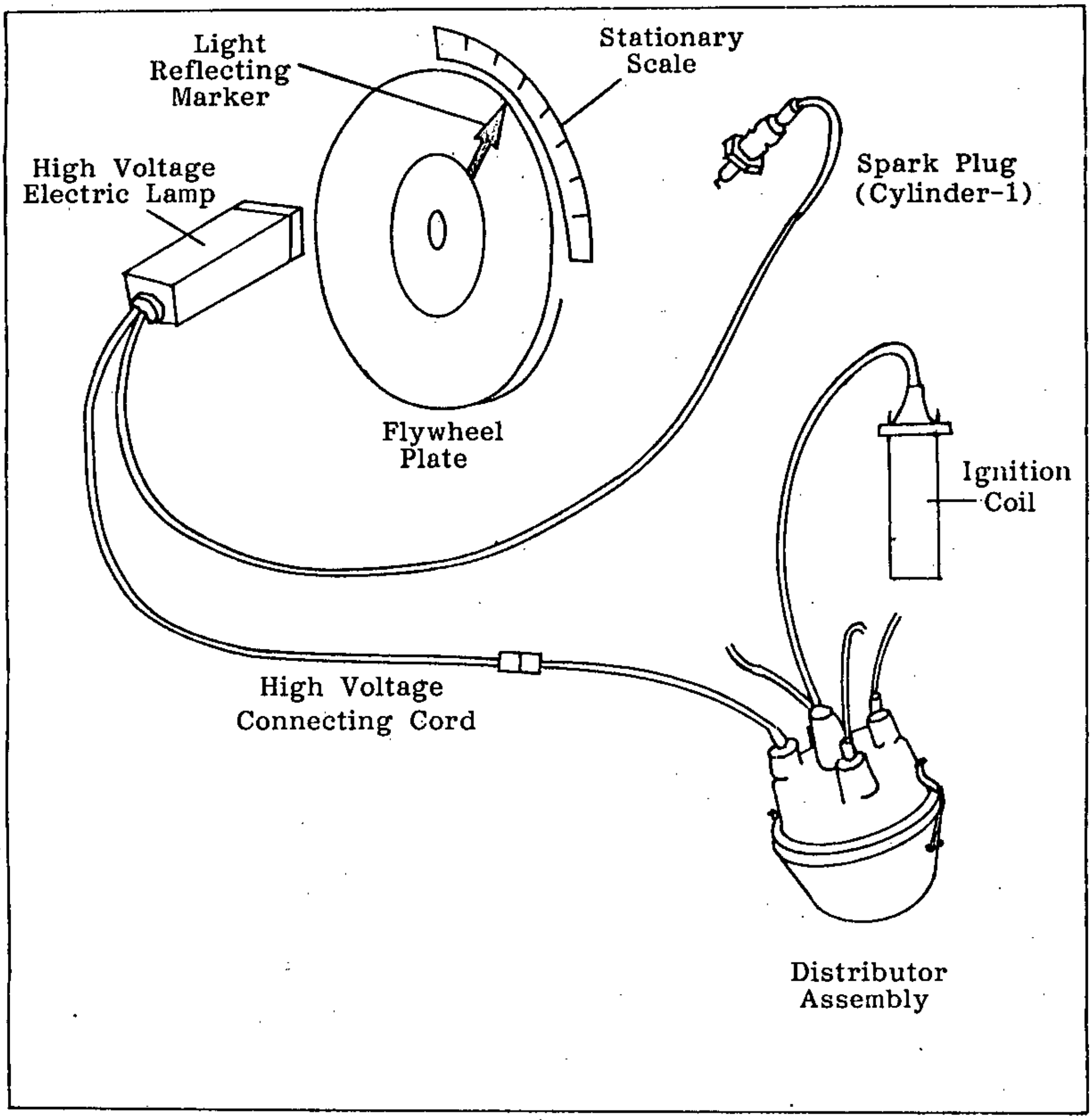


Fig : 3.8 Schematic diagram of the Spark Advance measuring system.

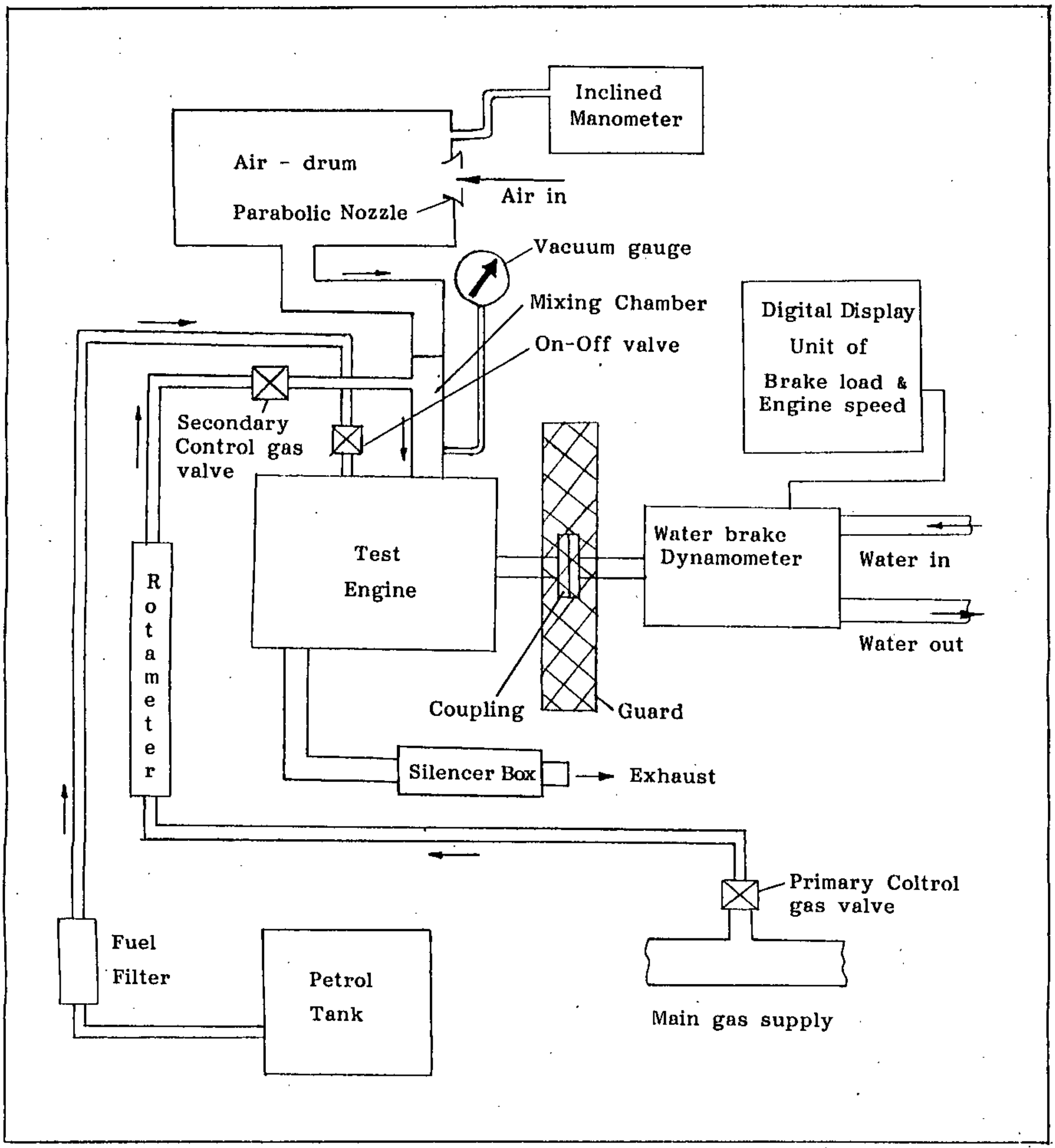


Fig : 3.9 Block Diagram of the Engine test bed assembly.

The test bed was also provided with temperature measuring facilities of exhaust manifold, cooling water radiator and dynamometer water outlet.

A vacuum gauge, ranging 0 - 1 bar vacuum, was connected to the intake manifold of the engine. This allowed the measurement of intake manifold vacuum during operation of the engine.

Initially the exhaust gas analysis was also planned, but finally it could not be carried out due to instrumental problems.

PROCEDURE

Before starting the engine necessary precautions such as - checking mechanical joints and connections, lubricating oil level, cooling water level, calibration of the dynamometer, dynamometer water supply etc. were done. When the engine needs to run on natural gas, two methods can be followed. While running the engine on petrol, the engine could be switched to natural gas. In this method the engine was started on petrol fuel only and load was gradually applied to the engine and correspondingly the fuel throttle was shifted to maintain the speed of the engine to the desired level, the gas main valve and the gas control valve were kept at closed position. Then the on-off valve of the petrol line was closed and the engine was allowed to run with the petrol present in the carburetor. As soon as the petrol in the carburetor finishes the speed of the engine tends to drop and at this condition the gas flow valve is opened carefully and the engine is switched over from petrol to natural gas. In the other method the engine could be started with natural gas only. The throttle valve position was adjusted to a wider open condition and by controlling the opening of the natural gas control valve and with normal cranking, the engine could be started. The petrol fuel line was kept closed for this method. Among these two methods the second one needs more skill or practice.

In taking readings two approaches were adopted. In the first approach the engine was tested at varying speeds (1025, 1200, 1500, 1700, 2000, 2200, 2500, and 2700 rev/min) keeping the load constant (15 kg). For petrol at each speed the variation of power was measured through the dynamometer and tachometer readings, at various spark advance positions. The variation of speed and load was controlled by the throttle valve control mechanism. The spark advance positions were changed by manual adjustment of the distributor unit (in this process the position of the contact breaker plate is changed relative to the distributor casing.). The spark advance position at which best power is attained, is known as the best power spark advance (bpsa). At each bpsa condition the fuel flow rate, exhaust temperature, cooling water temperature, manometer reading for air flow rate and intake manifold vacuum were recorded. A similar procedure was followed for the engine when running with Natural gas at various speeds (1025, 1200, 1500, 2000, 2500 rev/min, at load of 15kg). In this case the load and speed were controlled by the simultaneous adjustment of the gas flow control valve and the throttle position. For natural gas, a rotameter was used to measure the gas flow rate. The rotameter was calibrated prior to the experiment.

In the second approach the speed was kept constant at 2500 rev/min. For petrol the readings similar to the first approach were taken, but this time at variable loads (at 18,19,20,21,22,23,24 and 25 kg) keeping the speed constant (2500 rev/min). Also for natural gas, readings similar to the first approach were taken at the best power spark advance positions for variable loads(at 15,17,18,19,20,21,22,23 and 24 kg loads at 2500 rev/min). Similar to the first approach natural gas needed simultaneous adjustment of gas flow control valve and throttle control mechanism.

After all the readings were taken in case of operation with petrol fuel the engine speed was gradually decreased by the throttle adjustment, and the engine was finally stopped by turning off the ignition switch. In case of operation with natural gas, at the end of the experiment the speed was decreased by simultaneous adjustment of the gas flow control valve and the throttle valve position. Finally the engine was stopped by completely closing the gas flow valve. This process allowed removal of the unburnt natural gas remaining in the carburetor, the mixing chamber and the engine cylinders. Then the ignition switch was turned off.

CHAPTER – 4

RESULTS AND DISCUSSION

This chapter presents the results obtained from observations and experimental data along with the discussion based on results. Table: 4.1 - 4.7 shows the results obtained by experiments and Figure : 4.1 - 4.14 presents the graphs obtained from these results. A sample calculation has also been shown in the Appendix - A.

This is an experimental investigation and the discussions are mainly regarding the experimental configuration, spark advance, air-fuel ratio, load, rotational speed, specific fuel consumption and exhaust temperature. Some explanations have been made to explain the observations in the sight of known physical phenomena.

The 1600 cc, 4-cylinder, Toyota engine which normally runs on petrol as the fuel, was run with 100 per cent natural gas, without making much modification to the engine. Available commercial Titas gas line was used as the supply for running the engine with natural gas. Gas was admitted into the air intake manifold of the engine through a mixer. Generally three types of mixer eg, the Direct mixer, the L-shape mixer, and the Cross-shape mixer are used. The performance of the engine depends on the effeciency of the mixing process. The Cross-shape mixer showed better performance than the Direct or L-shape mixer (2). Therefore, a cross-flow type mixer was used as the mixing chamber, where gas was flown through a perforated copper tube placed perpendicular to the direction of intake air flow. Fig - 4.0 showes a cross flow type mixer.

In this research the effect of the spark advance on the performance of the SI engine was studied. The study had two main objectives, one was to study the effect of change of spark advance on performance parameters like - Air flow rate, Brake-Specific Fuel Consumption, Air-fuel ratio, Exhaust temperature. The other one concern was to study whether the spark advance

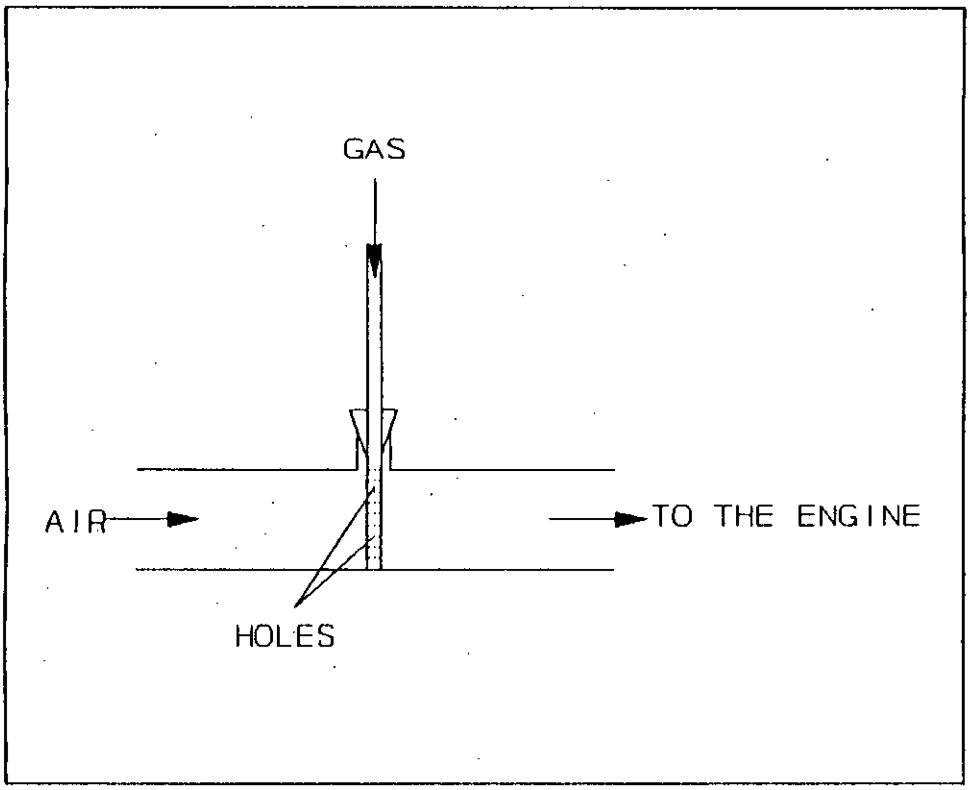


Figure - 4.0 Cross-shape Mixer

mechanism used in petrol engine, which consists of centrifugal and vacuum advance mechanism, was suitable for the same engine when running with gas as the alternative fuel.

A. PERFORMANCE OF THE SI ENGINE USING NATURAL GAS AS THE ALTERNATIVE FUEL.

The SI engine tested was designed for using petrol as the fuel. In case of running with natural gas, the properties of natural gas will effect the combustion characteristics of the engine. This will eventually effect the performance of the engine. In this research the alternative fuel was introduced into the air before the mixture enters the combustion chamber. A cross flow type mixing chamber was used for better mixing of natural gas with air. The combustion process of the gas-air mixture can be assumed to be similar to those for other homogeneous mixtures.

The flame speed of natural gas, used as the alternative fuel in this research, is less than that of petrol(gasoline) fuel ( ). Without changing the spark-timing, this could lead the combustion process to continue well into the expansion stroke. As a result there will be loss of power and efficiency. Generally for all fuels for better performance, the cylinder pressure should continue to rise after TDC, with peak pressure occuring a few degrees after the TDC.

Performance tests were done at a speed of 2500 rev/min. Some engine parameters, as measured in this research with natural gas are discussed as follows :

A.1 Air Fllow rate

Natural gas is mixed with air before the mixture enters the combustion chamber. At constant speed, as the load increases the fuel flow rate increases which demands the air flow rate to increase. Fig - 4.1 shows the variation of air flow rate at various loads at 2500 rev/min, for both petrol and natural gas. At higher load the throttle opening is wider, more fuel enters the engine and more ar is needed for its combustion. The curve for natural gas has higher values than that of petrol. This indicates that greater amount of air is needed for combustion of natural gas, than petrol fuel. So for natural gas the throttle position has to be opened more, even for the same load. At loads higher than 22 kg, for natural gas the air flow was almost constant, which indicates that the extreme limit of the throttle position for the petrol engine has been reached. Fig 4.2 shows the variation of brake power developed with air flow rate, for both the fuels.

A.2 AIR-FUEL RATIO

Here three ratios are considered. The A/F ratio is the ratio of air flow rate to petrol flow rate, in mass basis. The A/G ratio is the ratio of air flow rate to gas flow rate, in mass basis. The Equivalent A/F ratio is the ratio of air flow rate to the same petrol fuel flow rate as equivalent to the natural gas flow. The A/G ratio, was found to be higher than A/F ratio. This is caused as greater amount of air is needed for combustion of same mass of natural gas, relative to petrol. In the operational range the A/F or A/G ratio changes little, but at high loads the ratios decrease as the throttle approaches its end limit. Fig - 4.3 shows the variation of A/F ratio, A/G ratio and equivalent A/F ratio for natural gas.

A.3 Brake-Specific Fuel Consumption (bsfc)

The brake specific fuel consumption is defined as the fuel flow rate per unit brake power. For comparison between natural gas and petrol, equivalent bsfc, was considered. It is defind as the equivalent petrol flow rate of natural gas per unit of brake power. Fig - 4.4 shows the load vs.bsfc graph for petrol fuel and the load vs. equivalent bsfc graph for natural gas. The power developed by the engine is related to the air intake. As can be seen from Fig.4.3 after about 22.5 kg load, the air-gas ratios decreases when the engine runs on natural gas, because the throttle reaches its maximum position. Because of limitation of air intake the maximimum power of the engine when running with natural gas is less than when running with petrol fuel. Also the rated load is less for natural gas. The minimum equivalent bsfc for natural gas was found to be slightly higher (1.7 %) than the minimum bsfc for petrol. This is mainly caused by the slower flame propagation speed of natural gas. The slower burning rate allows exit of a greater amount of unburnt natural gas through the exhaust. In case of petrol, the fuel enters the caburetor in the form of fine liquid along with air. By the time the charge enters the cylinder the liquid particles become completely vapourised this increases the inlet pressure of the cycle. In case of natural gas, the fuel is supplied in gaseous form, hence such increase in inlet pressure is not present. These two factors might cause the slight rise of the bsfc (346.35 g/kW-h for petrol and 353 g/kW-h for natural gas, at rated load) and fall of brake thermal effeciency (23.62% for petrol and 23.14% for natural gas).

A.4 Exhaust Temperature:

Fig - 4.5 shows the variation of exhaust temperatures with load for both petrol and natural gas. At all loads the exhaust temperature is higher for natural gas than petrol fuel. The flame speed of natural gas is less than that of petrol. Therefore, burning of the air gas mixture continues well in the expansion stroke, which causes the exhaust temperature to rise.

B. SPARK ADVANCE MECHANISM FOR NATURAL GAS FUELLED SI ENGINE

The first intention was to get the data required to establish a graphical presentation of the available spark advance mechanism of the petrol engine under test - which consists of both centrifugal and vacuum advance mechanisms. These two are governed by the engine speed and throttle position, where as throttle position depends on the load. These results were to be compared with the same for natural gas. Two approaches were used for both the fuels. The first approach was to study the spark advance phenomena at various speeds, keeping the load constant. The second one involved the variation of load on the engine, keeping the speed constant.

In the first approach using petrol as the fuel, spark advance was measured at a speed of 2500 rev/min and loads of 15 kg, using the stroboscopic spark advance mechanism. Keeping the speed at 2500 rev/min, the spark advance was changed manually from 10 to 40 degree btdc, to find the variation of power. Figure - 4.6 shows the variation of power for various spark advances at 2500 rev/min. From the peak of the curve the best power spark advance (bpsa) could be determined. Keeping the load same, the engine speed was changed and the corresponding bpsa was recorded. These results are given in Table - 4.2 and Figure - 4.7 shows the graphical presentation. The graph showed a linear relation of bpsa with speed.

Similar experiments were carried out using natural gas as the fuel. At the speed of 1000, 1250, 1500, 2000 and 2500 rev/min the spark advances were recorded along with the load. At each speed, spark advance and air-gas ratio were manually changed to get best possible power. The data recorded at various speeds, with different spark advances are given in Table - 4.3. From the graphs shown in Figure - 4.8 the best power spark advances (bpsa) were determined. Figure - 4.9(c) shows the variation of bpsa with speed for both natural gas and petrol. For the engine, spark advance needed for best power at any speed is higher for natural than petrol fuel. Moreover these two graphs have different slopes.

Results and graphs show that with increased speed at a constant load, spark advance needs to be increased as less time is available for combustion. The two curves of fig - 4.9(c) show that the spark advance needed is different for petrol and natural gas. This is mainly caused by the lower flame speed of natural gas.

In the second approach the spark advance phenomena was studied for both petrol and natural gas at various loads, keeping the speed constant. For petrol at first the engine was set at the speed of 2500 rev/min, load 15 kg and at the best power spark advance position (which was about 30 btdc). Keeping the speed constant the bpsa were recorded at different loads up to overload condition. Results are recorded in Table - 4.5 and Fig - 4.10. The same procedure was repeated for natural gas, here the air-gas ratio was also varied. For each load the best throttle position was adjusted and then the spark advance was changed manually for the best power. Fig - 4.11 shows the variation of power at various spark advance positions. The variation of bpsa with load is shown in Fig - 4.12. Fig - 4.12(c) shows variation of bpsa with load for natural gas and petrol fuel at constant speed.

Since the speed is constant (2500 rev/min), for both the fuels, the effect of the centrifugal advance mechanism is the same . Therefore the required variation of spark advance for best power has to be done with the vacuum advance mechanism. For both the fuels the spark advance required decreases at higher loads due to increased rate of combustion. The advance needed for natural gas is much higher than that of petrol, while the vacuum effect created in the intake manifold is less for natural gas (Fig - 4.13). So there exists scope for modification in the vacuum advance mechanism. At the same speed the variation of spark advance for both fuels should be caused by the vacuum advance mechanism. For analysis two terms "Bpsa Ratio" and "Vaccum Ratio" were introduced. Otherwise direct comparison between the spark advance and vacuum needed for two different fuels is difficult. These terms are defind as:

$$Bpsa\ Ratio = \frac{Spark\ advn.\ at\ load}{Spark\ advn.\ at\ rated\ load}$$

Both at the same speed.

$$Vaccum\ Ratio = \frac{Vacuum\ at\ load}{Vacuum\ at\ rated\ load}$$

Both at the same speed.

Fig - 4.14(a) and 4.14(b) gives the variation of "Vacuum Ratio" and "Bpsa Ratio" with load at a constant speed of 2500 rev/min for both the fuels. The two dimensionless curves for petrol were found to be almost identical, which signifies that the designed vacuum created at the intake manifold and the corresponding vacuum advance are well matched. The two dimensionless curves for natural gas was found to be different. The rate of decrease of vacuum created in the manifold is much higher than the rate of decrese of vacuum advance needed by the spark advance mechanism.

CHAPTER – 5

CONCLUSION AND RECOMMENDATION

The need for alternative fuels are increasing everyday. This is caused by the increasing rate of consumption and limited reserves of the conventional fuels. A vast number of internal combustion engines are used in various fields in the present world. These internal combustion engines mainly use petrol or diesel as fuel, which are conventional fuels of limited reserves. Operating these engines with alternative fuels could help in encountering the energy crisis in near future. Hence research and use of various alternative fuels in internal combustion engines, should be enhanced before the energy crisis begins. Various alternative fuels include - Natural gas, Bio-gas, Alcohol fuels, Hydrogen and Vegetable oil etc.

Bangladesh has a vast deposit of natural gas. Hence the use of natural gas in IC engines may be very helpful for strengthening the economy of Bangladesh and reduce dependency on other countries. So research on the use of natural gas in IC engines is very relevant to our country.

The engine performance is governed by various parameters like air flow rate, load, speed, spark advance etc. Also combustion will depend on the combustion characteristics of the fuel used. Some performance parameters when running a SI engine using natural gas as the alternative fuel was the main concern of this study. The test results obtained in this research can be summarized as follows :

5.1 PERFORMANCE OF THE ENGINE

Among the performance parameters the flow rate, air-fuel ratio, brake specific fuel consumption and exhaust temperature were considered. The performance tests were performed at a speed of 2500 rev/min for petrol fuel and natural gas respectively. For comparing the results obtained from two fuels, petrol flow rate equivalent to the natural gas flow rate was calculated. As the engine was ran on natural gas equivalent air-fuel ratio, equivalent brake specific fuel consumption were also calculated.

For same speed the air flow rate needed for the gas was higher than that for petrol. The throttle had to be set at a wider opening position for natural gas than petrol fuel, at the same speed and load. The A/F, A/G and equivalent A/F ratio was found to vary little in the operating range of the engine. The values are about - 17 for A/G ratio, 16.1 for equivalent A/F ratio for natural gas and 11.8 for A/F ratio for petrol.

For both fuels the brake specific fuel consumption curves have the same nature. The rated load at 2500 rev/min was found to be less for natural gas than for petrol fuel (23 kg for gas and 25 kg for petrol). The power developed at these conditions were 18.96 kW for petrol and 17.45 kW for natural gas, which shows 7.9 % reduction of rated power at 2500 rev/min. This is mainly caused by the displacement of the air in the charge by natural gas at a proportion greater than petrol.

The exhaust temperature was found to be higher for natural gas. This is mainly caused due to the slower flame speed of natural gas relative to petrol. As a result some combustion may continue well into the exhaust stroke. Though there is some loss of power and higher exhaust temperature is attained, the use of natural gas in SI engine is prospective as the engine needs little modification.

5.2 EFFECTS OF SPARK ADVANCE :

In this study effect of spark advance on the engine performance was tested using petrol fuel and natural gas individually. For this two approaches were used for both the fuels. In the first approach the load was kept constant at 15 kg and by measuring the power the best power spark advances (bpsa) were measured at various speeds. The results showed that the bpsa needed at the same speed is higher for natural gas than petrol fuel. This is mainly caused due to the lower flame propagation speed of natural gas. The difference of spark advance needed for petrol and natural gas also varies with speed, which is presented by the non-parallel nature of the bpsa curves of the two fuels. It was also found that as the speed increased, the rate of increase of centrifugal advance needed for natural gas was greater than petrol fuel.

In the second approach the speed was kept constant at 2500 rev/min and the load was varied. At constant speed the change of spark advance is caused by the vacuum advance mechanism. This procedure was applied for both the fuels. For both fuels the spark advance needed decreases with higher load. The bpsa needed for natural gas was higher, due to its lower flame speed. At various loads, the difference of spark advance needed for petrol and gas is different, though the variation of difference is much less then that of the first approach.

For analysis the Bpsa ratio and Vacuum Ratio were introduced. For petrol the curves presenting the variation of Bpsa Ratio and Vacuum ratio were found to be almost identical, indicating that the vacuum created in the intake manifold and vacuum advance mechanism are in match for petrol. But these two curves for natural gas showed different nature.

5.3 RECOMMENDATIONS:

For future work in this line the authors recommendations are as follows:

1. While studying the spark advance mechanisms, two approaches were used. The first one keeping the load constant at 15 kg and varying the speed. This approach can be repeated for other loads of higher and lower values and varying the engine speed.

2. In the second approach the speed was kept constant at 2500 rpm and the load was varied. This approach can be adopted for other constant speeds by varying the loads.

3. The same study can be done for other engines of different - size, capacity and geometry.

4. The performance and the effect of spark advance on SI engine can be studied for other alternative fuels like - Biogas (which contains about 65% Methane).

5. Exhaust gas analysis should also be included in the performance analysis.

REFERENCES

REFERENCES

1. Bari,S.,"Alternative Fuels in Diesel Engine", University of Reading, UK, 1991.

2. Bari,S.," Study of the Effect of Dual Fuel on Some Performance Parameters of A Single Cylinder Small Diesel Engine", M.Sc.Engineering Thesis, BUET, Dhaka, Bangladesh,1986.

3. Bari,S.,"Natural Gas in Diesel Engine", M.Sc. in Alternative Energy for Developing Countries Thesis, Reading University, U.K., June, 1989.

4. Baluswamy,N. and Subramanyam,S., " Investigation on the Performance of an Automotive Diesel Engine Using LPG as a Substitute Fuel", Proceedings of IX National Conference on IC Engines and Combustion. Vol:1 IC Engines, Dehra Dun,India, November, 19-22, 1985.

5. Cambell,A, "Engine testing Data and Results, for a Single Cylinder Duetz Diesel Engine converted to SI Engine run by CNG Fuel",COMSEC Sponsored CNG Conversion Programme carried out at BUET by LFMG Limited, April, 1989.

6. Crouse, Anglin, "Automotive Mechanics" Taya McGraw-hill company ltd,1986.

7. Born,G.J., and Durbin,E.J.,"The Natural Gas Fueled Engine", Published in "Methane for the Future" by Patrick McGeer and Enoch Durbin, Plenum Press, New York, USA, 1982.

8. Eangleman, H.W., Guinther,D.A. and Silvis, T.W., "Vegetable Oil as a Diesel Fuel", ASME 78-DGP-19, pp-7, Houston, Texas, November, 1978.

9. Fishinger, M.K.C., et al., "Service Trals of West Vegetable Oil as a Desel Fuel", SAE 811215, pp-10.

10. Forgiel, R., and Vard, K.S., "Experimental Investigation of Vegetable Oils Utlization in a Direct Injection Diiesel Engine", SAE 811214, pp-8.

11. Goering,C.E., Schwab,A.W., Daugherty,M.J., Pryde,E.H., and Heaking,A.J., "Fuel Properties of Eleven Vegetable Oils", ASAE Winter Meeting, Paper no. 81-3579, USA, 1981.

12. Goering,C.E., and Daugherty,M.J., "Energy Accounting for Eleven Vegetable Oil Fuels", ASAE Paper no.81-3586, Transactions of the ASAE, pp.1209-1215, USA, 1982.

13. Goodger,E.M., "Alternative Fuel", Macmillan Press Ltd.,U.K.,1980.

14. Goodger,E.M.,"Alternative Fuels for Locomotive Diesel Engines", Journal Institution of Mechanical Engineers, UK., 1982, C85/82.

15. Hall, D.O., "Put a Sunflower in Your Tank", New Scientiist, pp. 524-6, 26 February, 1981.

16. Heywood,J.B.,"Internal Combustion Engine Fundamentals", McGraw-Hill Publishing Company, USA, 1988.

17. Karim,G.A. and Ali,A.I., "Combustion, Knock and Emission Characteristics of a Natural Gas Fuelled Spark Ignition Engine with Particular Reference to Low Intake Temperature Conditions", Proc. Instn. Mech. Engrs. Vol:189, 24/75, 1975.

18. Karim,G.A.,"Methane and Diesel Engines", published in "The Natural Gas Fueled Engine" by Patric McGeer and Enoch Durbin, Plenum Press, New York, 1982.

19. Land,K.L.,"Performance and Operation of Spark Ignited Gas Engines", Institute of Mechanical Engineering, Vol. 181, Part-I, 1967.

20. Mollah, S.A.M., "Study of Performance Parameters and Exhaust Gas Pollution of Natural Gas Fuel Spark Ignition Engine", M.Sc. Thesis, Bangladesh University of Engineering and Technology, Dhaka, Bangladesh, 1990.

21. Nagosh, N.S.and Sagma,B.S.,"Optimisation of Operating Parameters For Bio-mass Fuelled SI Engine",Proceedings of IX th National Conference on IC Engines and Combustion.1985.

22. Picken,D.J., Seare,K.D.R., Fox,M.P., and Bush,J.P.,"An Investigation into Piston Ring Blowby and a Discussion of its Impact on Biogas Engines", Proceedings of the Conference on the Internal Combustion Engine Research in Universities, Polytechnic and Colleges, 30-31, January, 1991, Institution of Mechanical Engineers, U.K. C433/001.

23. Ryan,T.W.and Lestz,S.S.,"The Laminar Burning Velocity of Iso-octane, N-Heptane, Methanol, Methane and Propane at Elevated Temperatures and Pressures in the Presence of a Diluent", SAE 800103. p 652-644, 1980.

24. Sichuan Institute of Agricultural Machinery, China, "Application of Bio-Gas on Farm Internal Combustion Engine".

25. Taylor,F.C.,"The Internal Combustion Engine in Theory and Practice" Volume.1, The MIT Press, USA, 1989.

26. Taylor,F.C.,"The Internal Combustion Engine in Theory and Practice", Volume.2, The MIT Press, USA, 1985.

27. Thring,R.H., "Another look at Alternative Fuels",Automotive Engineering, January, 1984, p.60-64, volume 92, Number 1.

28. TNO, Research Institute for Road Vehicle, Netherlands,"Substitution of Diesel Fuel by Gaseous Fuel".

29. Weide,J.V.D., "Uses of Methane in Diesel and spark Ignition Engines", World Conference on Methane as a Motor Vehicle Fuel, Vancouver,B.C., 8 29 September, 1981.

30. Weide,J.V.D., Seppen,J.J., Ling,J.A.N.V., and Dekker,H.J.,"Natural Gas in Heavy Duty Engines", TNO Road-Vehicles Research Insitute, The Netherlands, 1988.

31. Whiston,P.J., Goodwin,M.J., and Girgis,N.S.,"Analysis of the Combustion of Methane in a Spark Ignition Internal Combustion Engine", Proceedings to the Conference on the Internal Combustion Engine Research in Universities, Polytechnics and Colleges, 30-31 January, 1991, Institution of Mechanical Engineers, UK, C433/001.

32. Williams,Nigel,"Water is 'Fuel of the Future'",The Guardian, 10th April, 1991.

33. Wong,J.K.S.,"Study of Mixtures of Methane and Carbon Dioxide as Fuels in a Single Cylinder Engine (CLR)", National Research Council, Canada, 1976, also SAE 770796.

GRAPHICAL PRESENTATION OF RESULTS

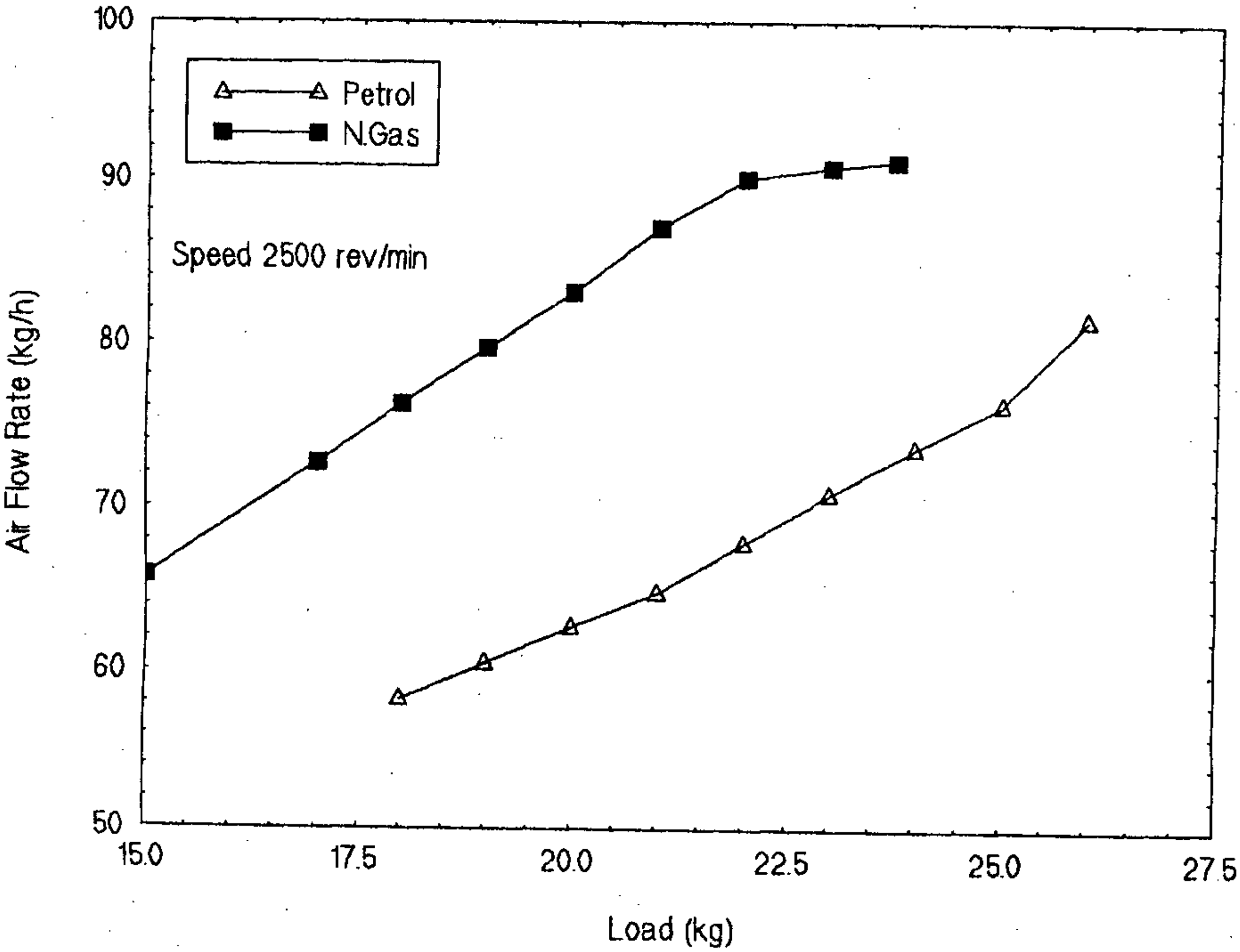


Figure - 4.1 : Variation of Air flow rate with Load.

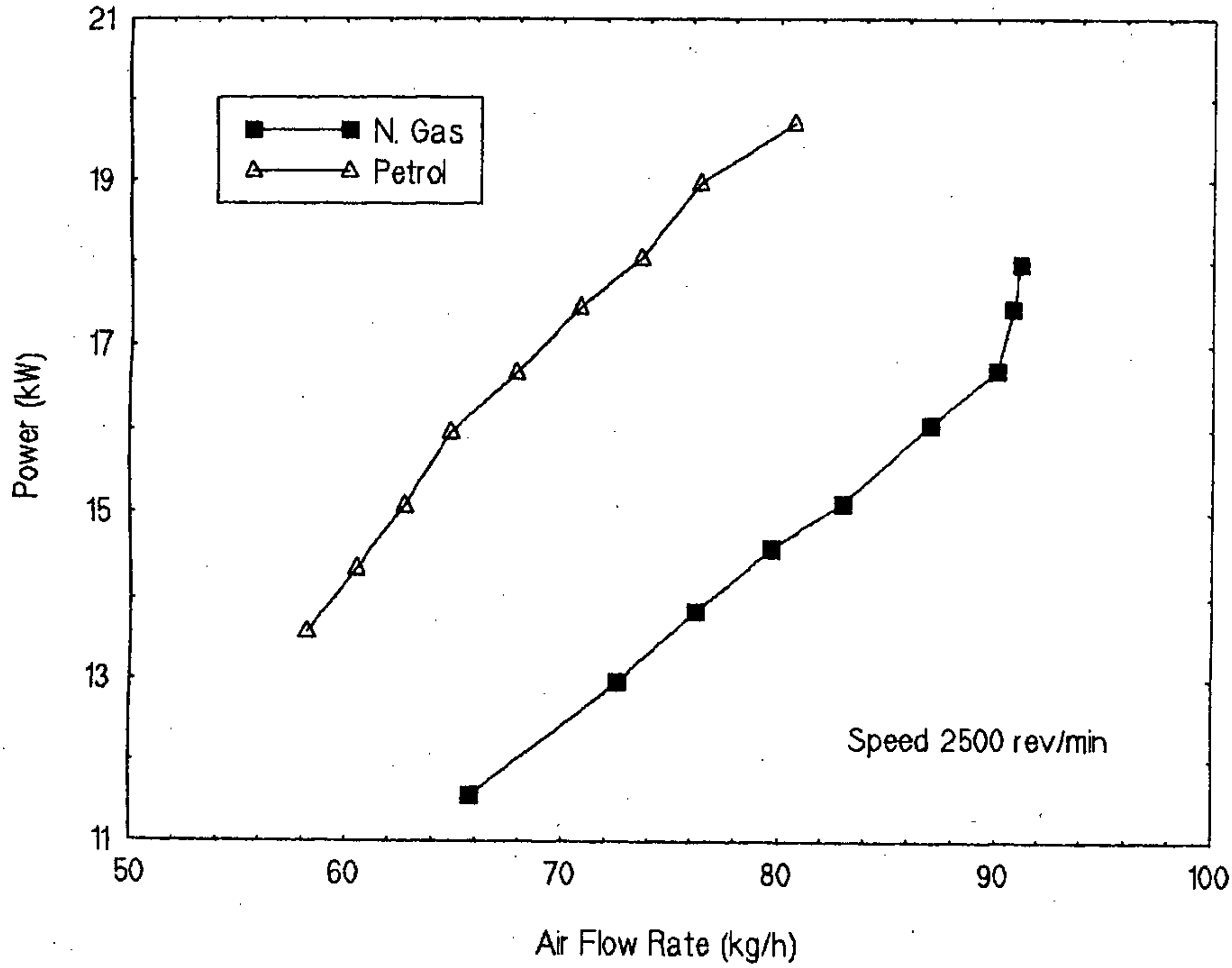


Figure – 4.2 : Power Vs. Air flow rate Graph.

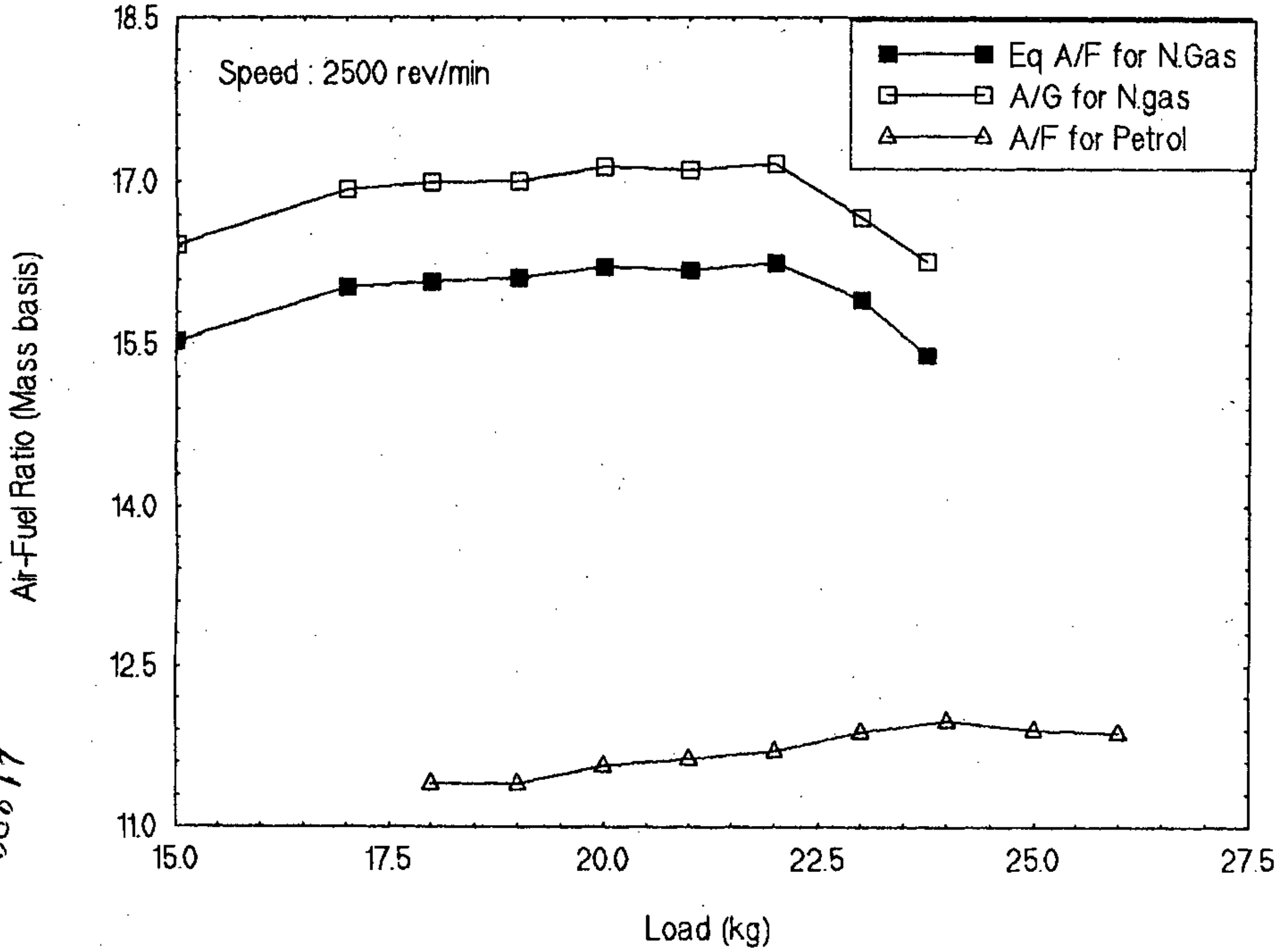


Figure - 4.3 : Variation of Air-Fuel Ratio with Load.

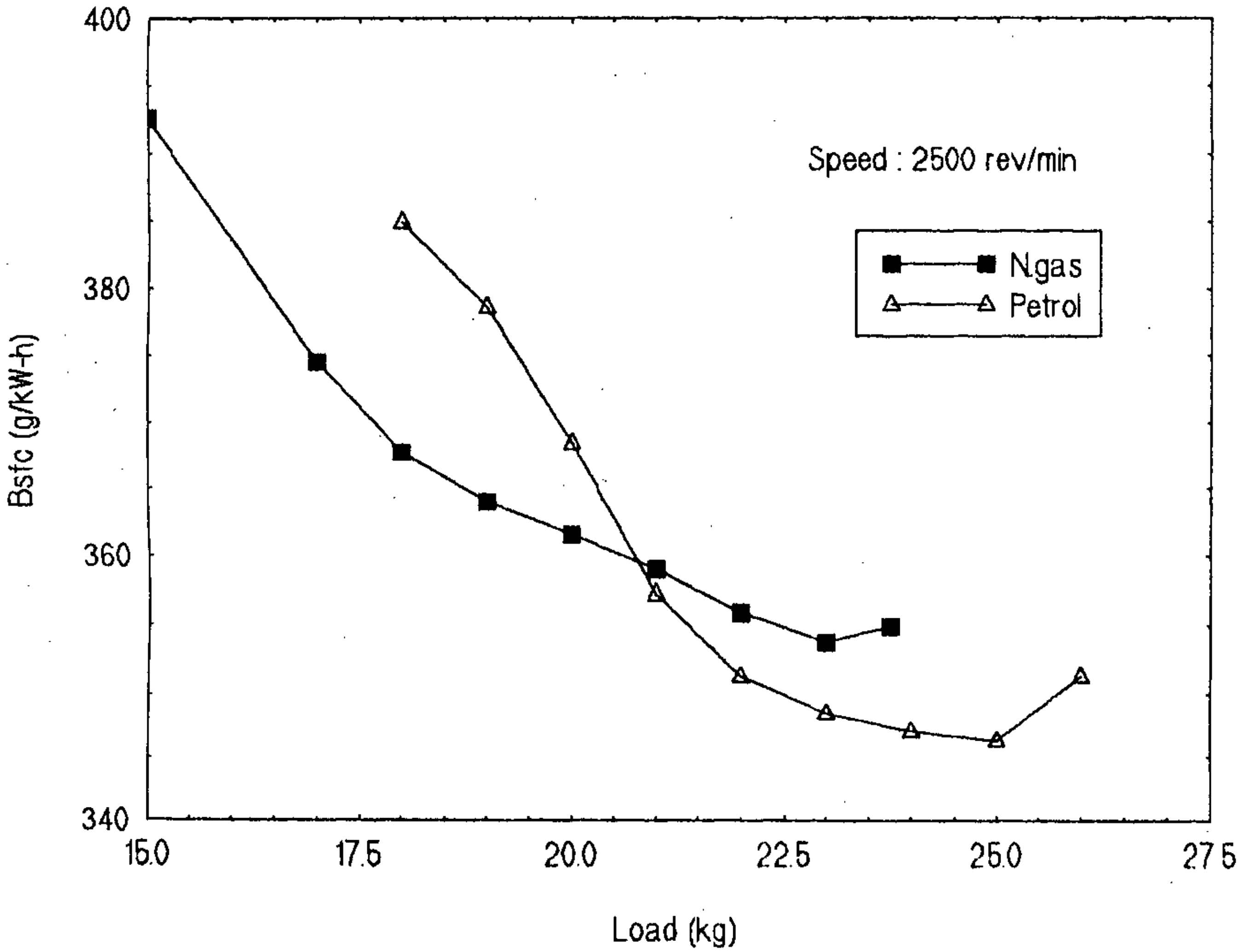


Figure – 4.4 : Variation of Bsfc with Load.

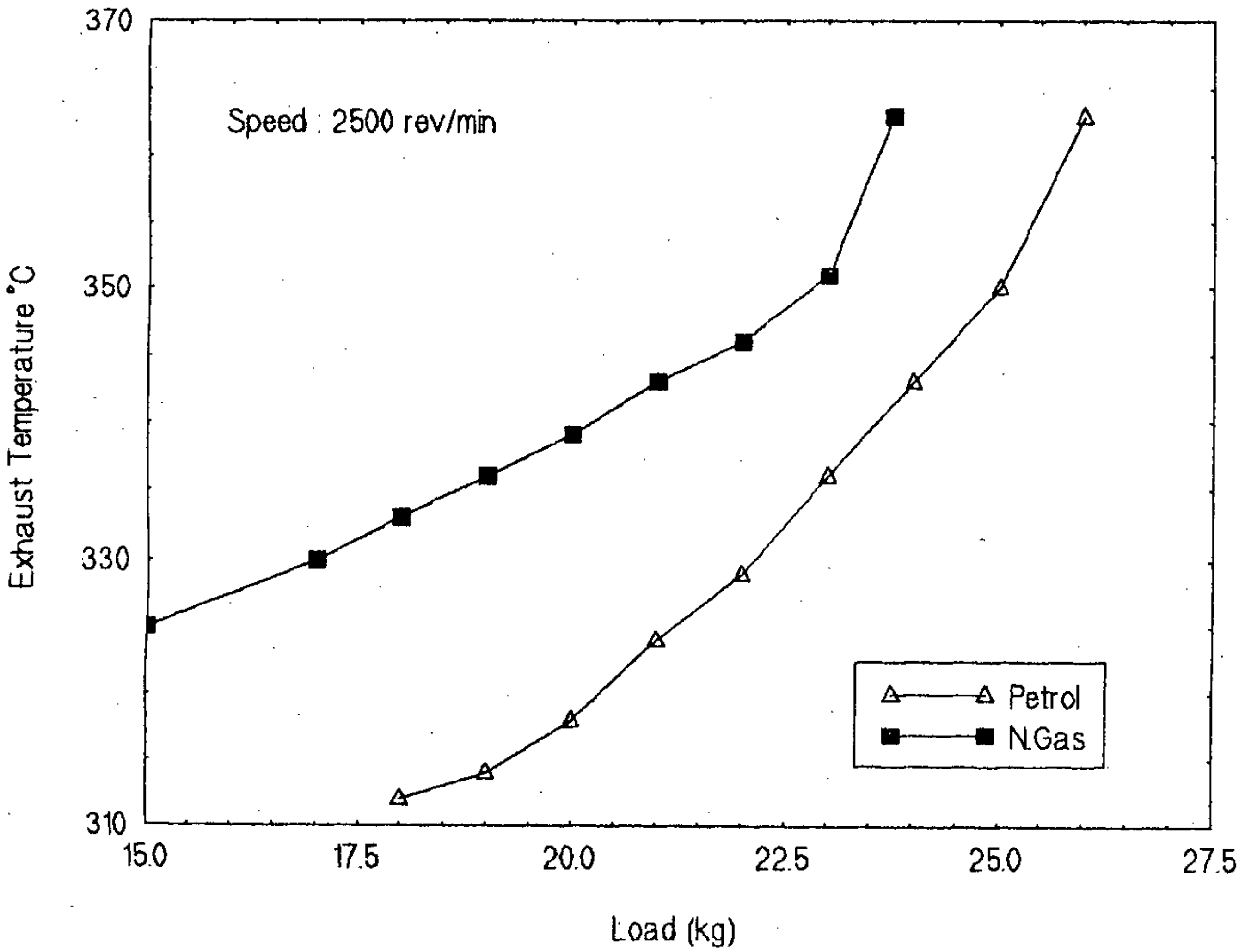


Figure – 4.5 : Variation of Exhaust Temperature with Load.

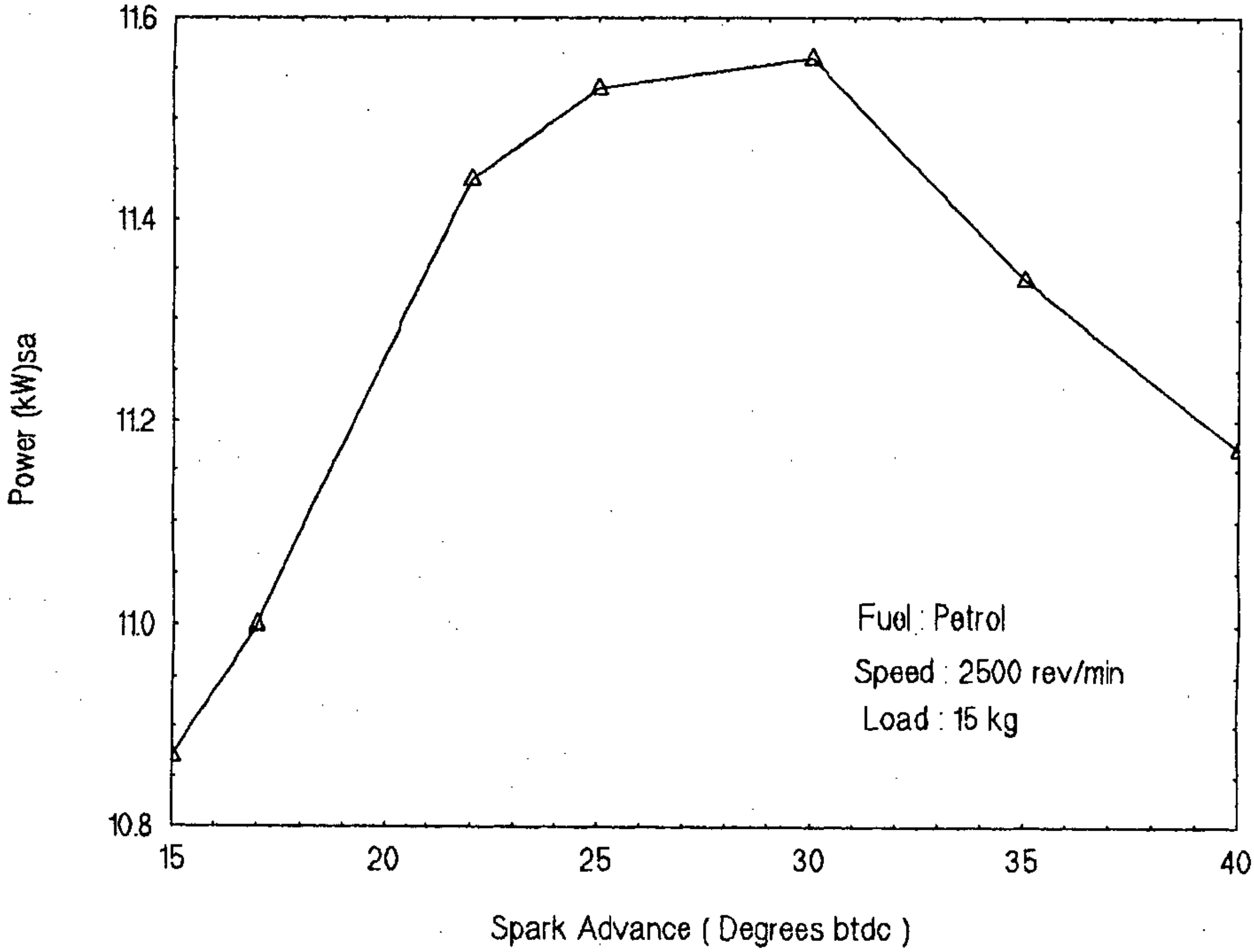


Figure – 4.6 : Variation of Power with Spark Advance.

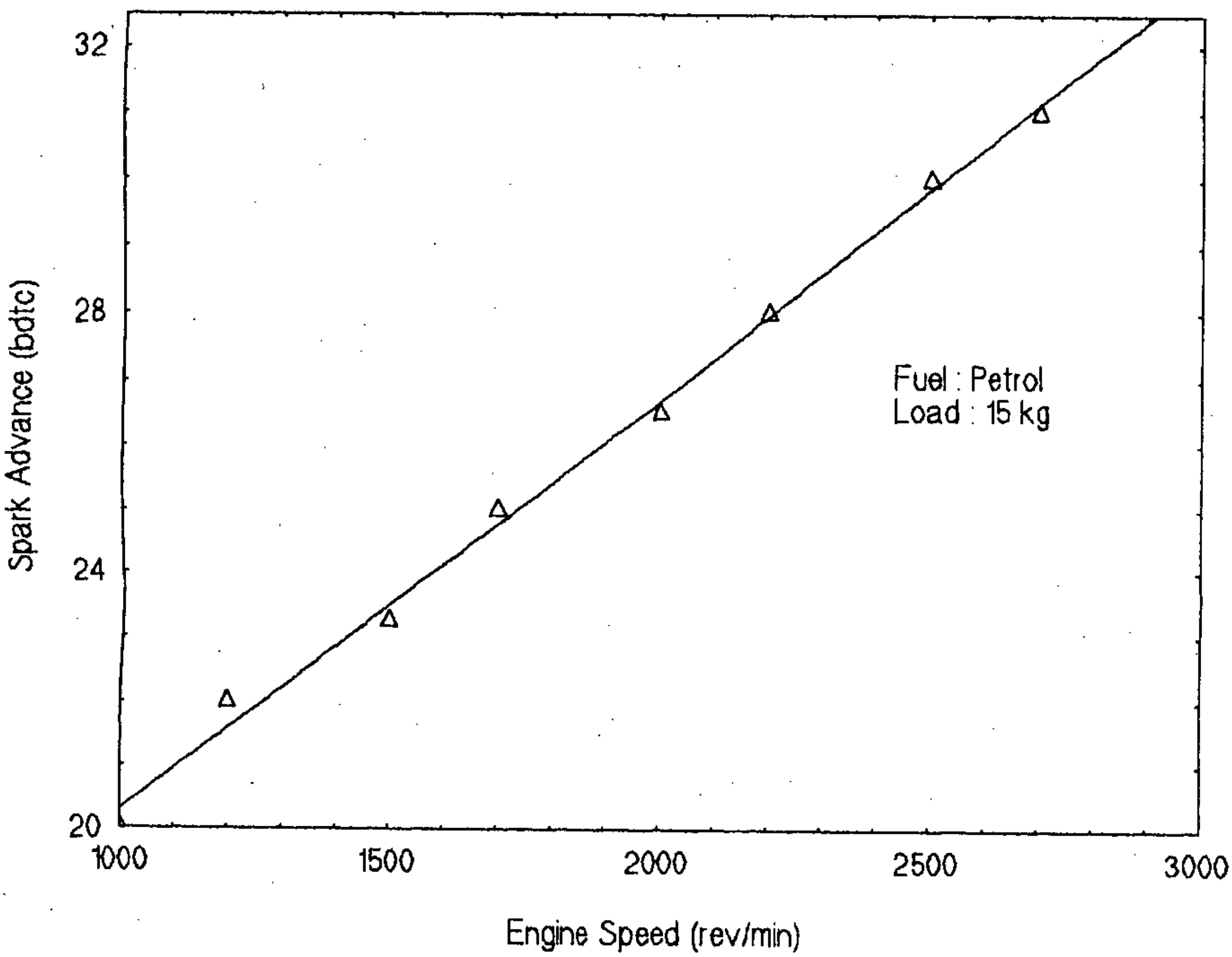


Figure - 4.7 : Speed Vs. Best Power Spark Advance Graph.

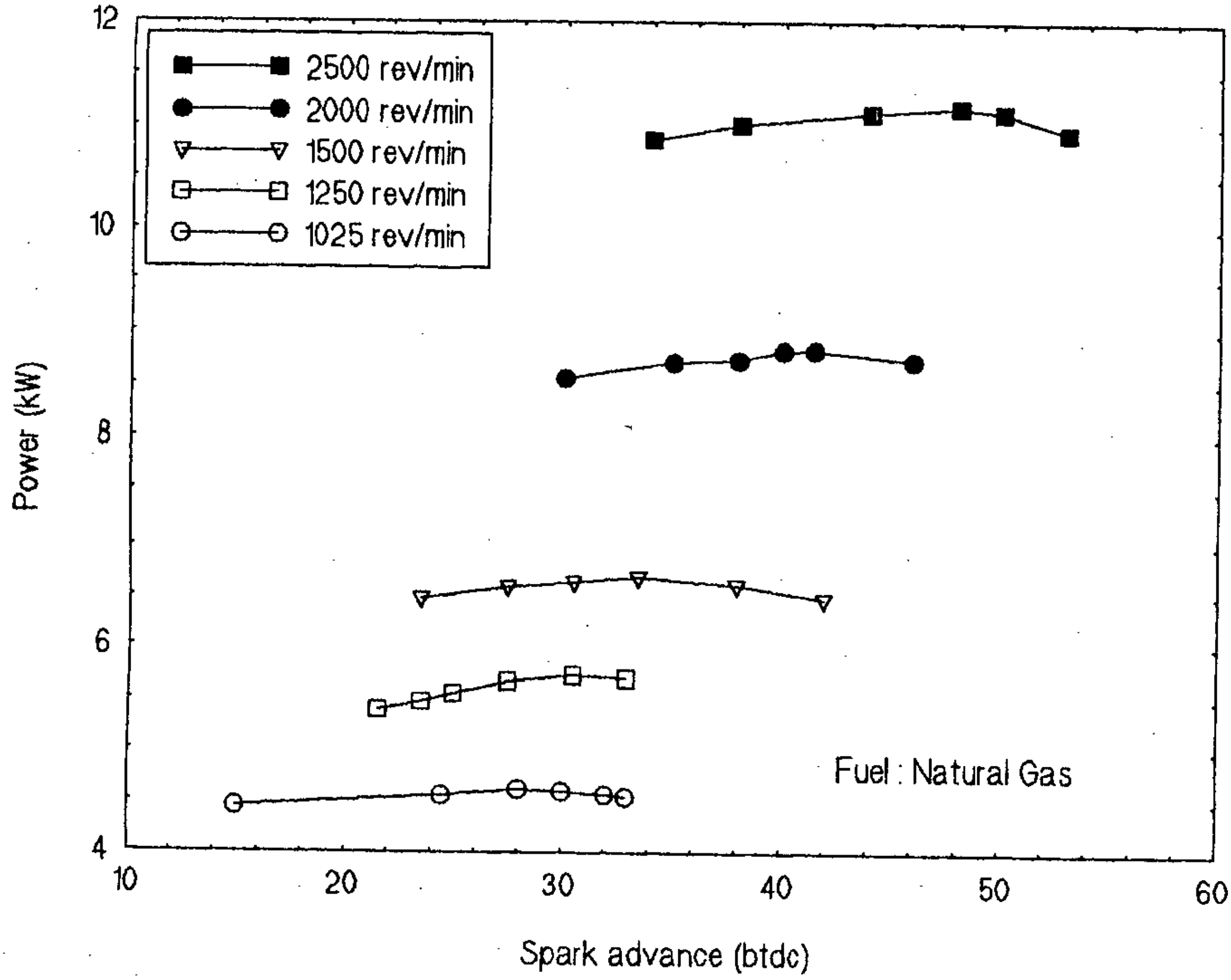


Figure – 4.8 : Variation of Power with Spark Advance at various speeds.

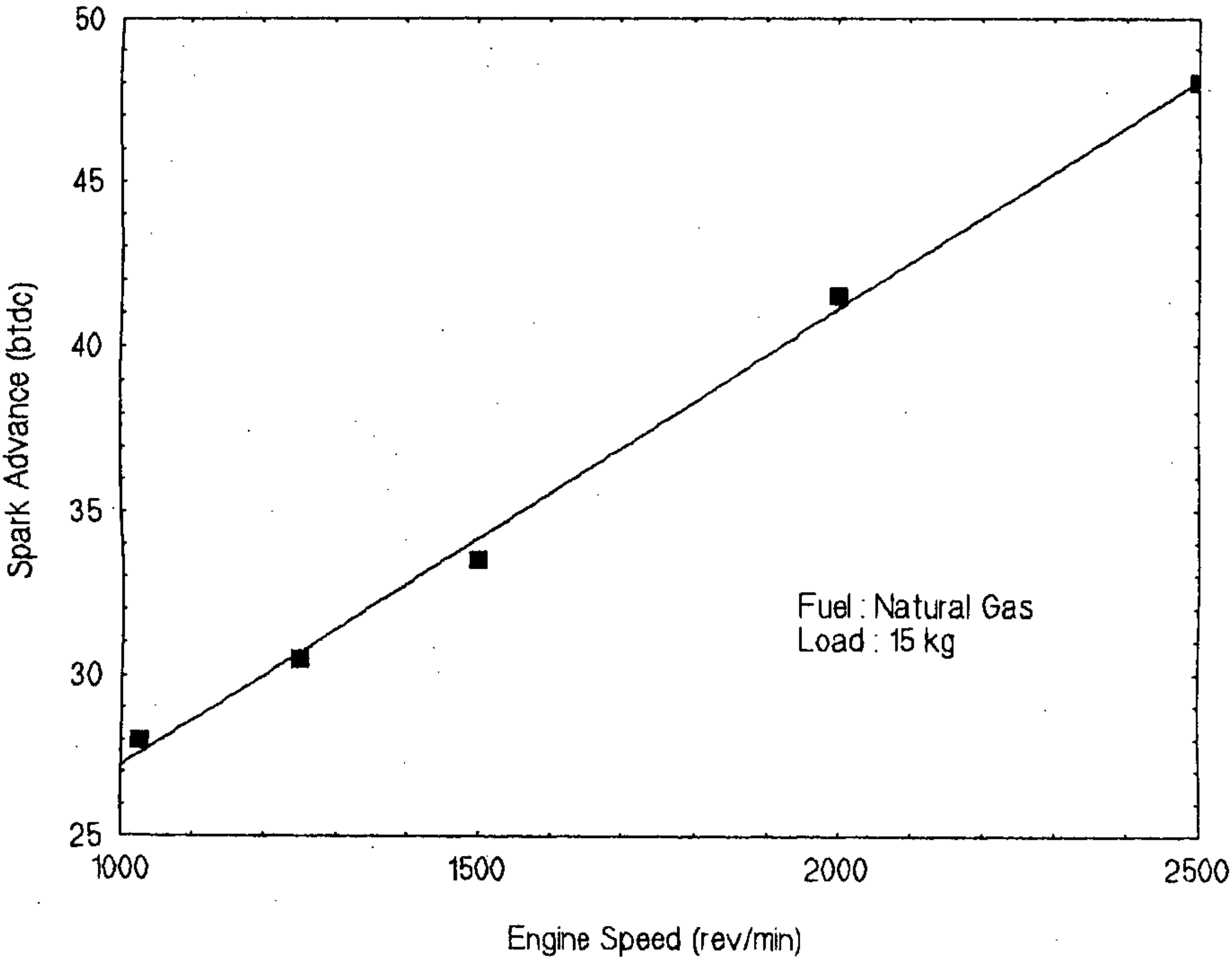


Figure – 4.9 : Speed Vs. Best Power Spark Advance

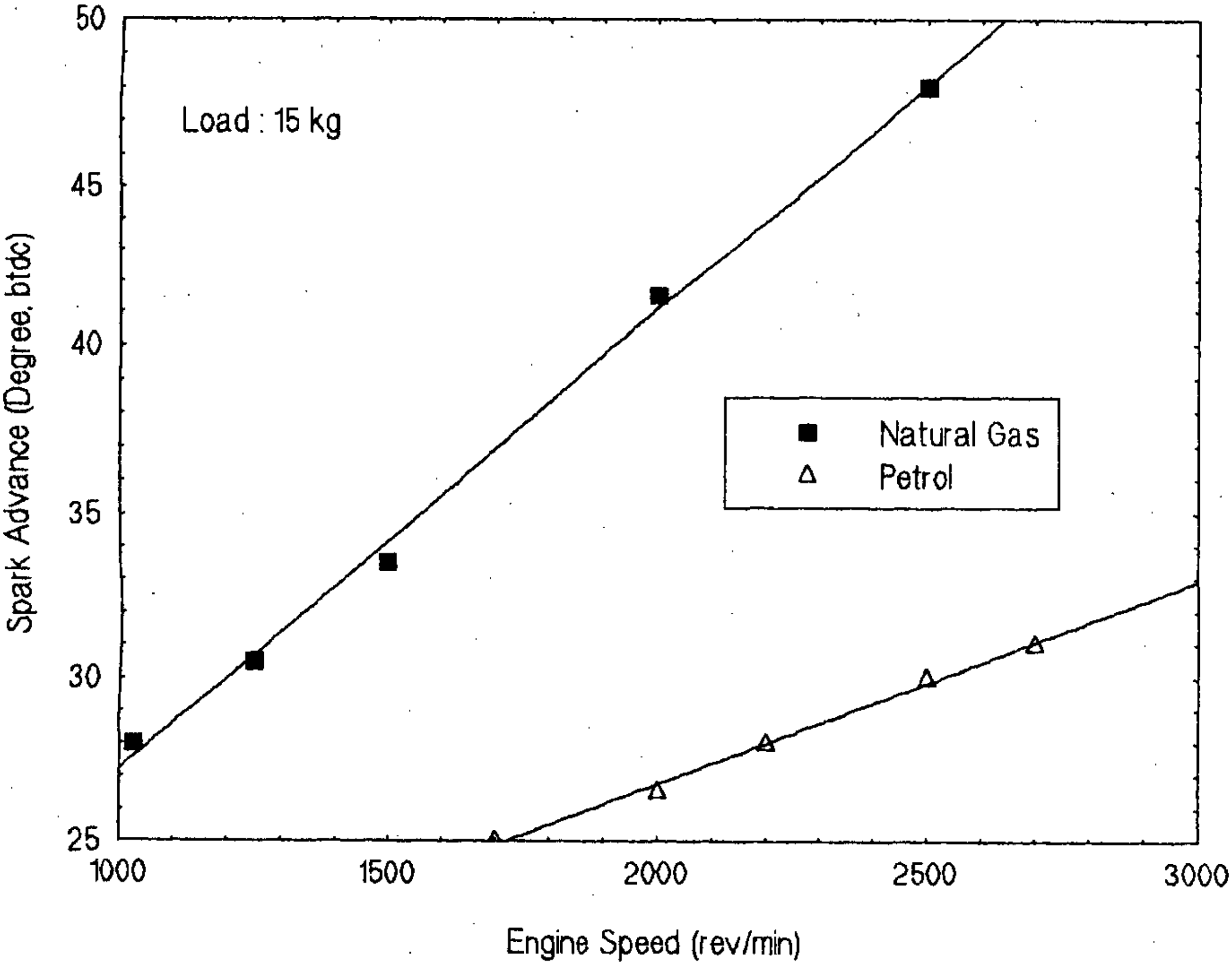


Figure - 4.9(c) : Speed Vs. Best Power Spark Advance for Petrol & Natural Gas.

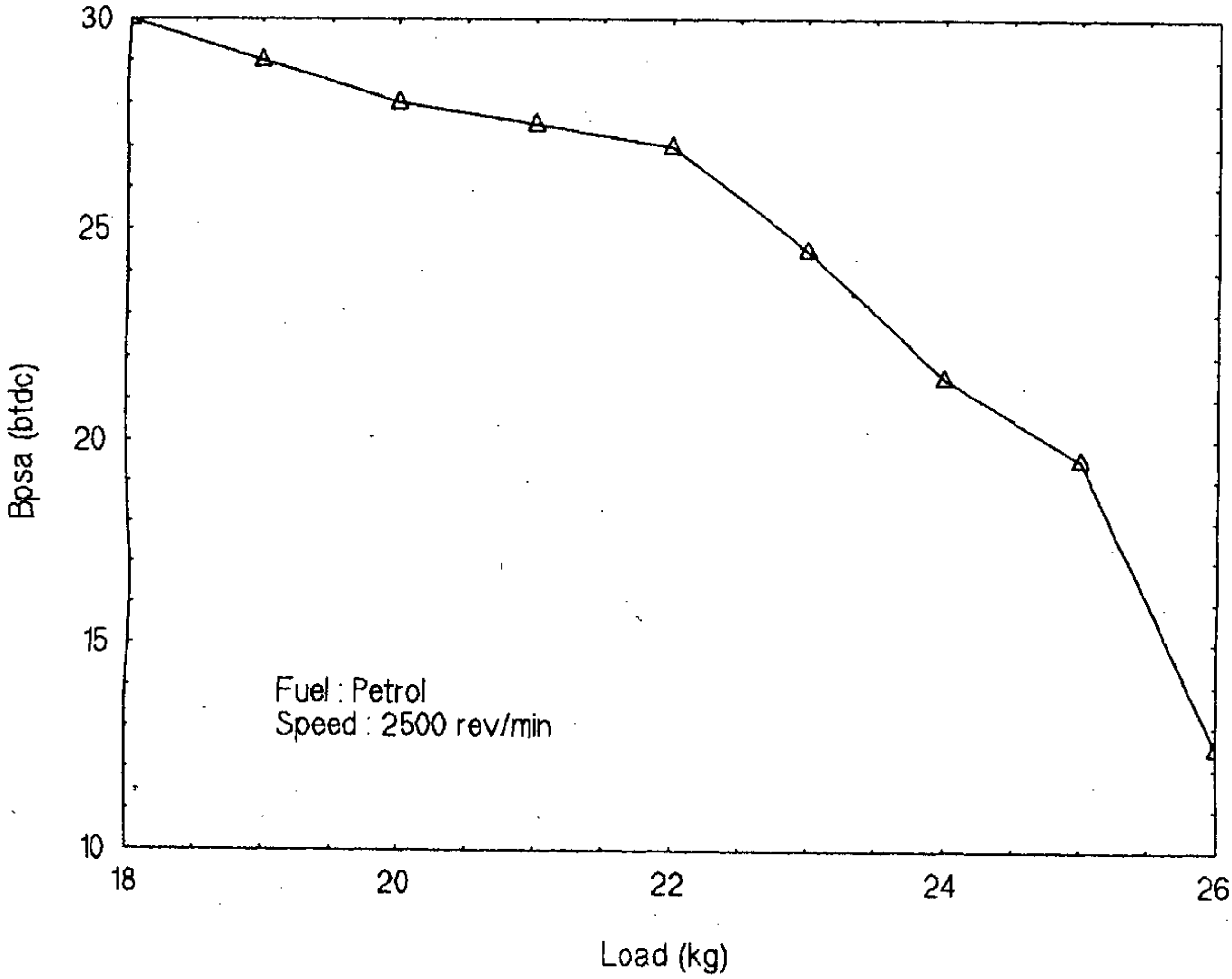


Figure - 4.10 : Variation of Best Power Spark Advance with Load.

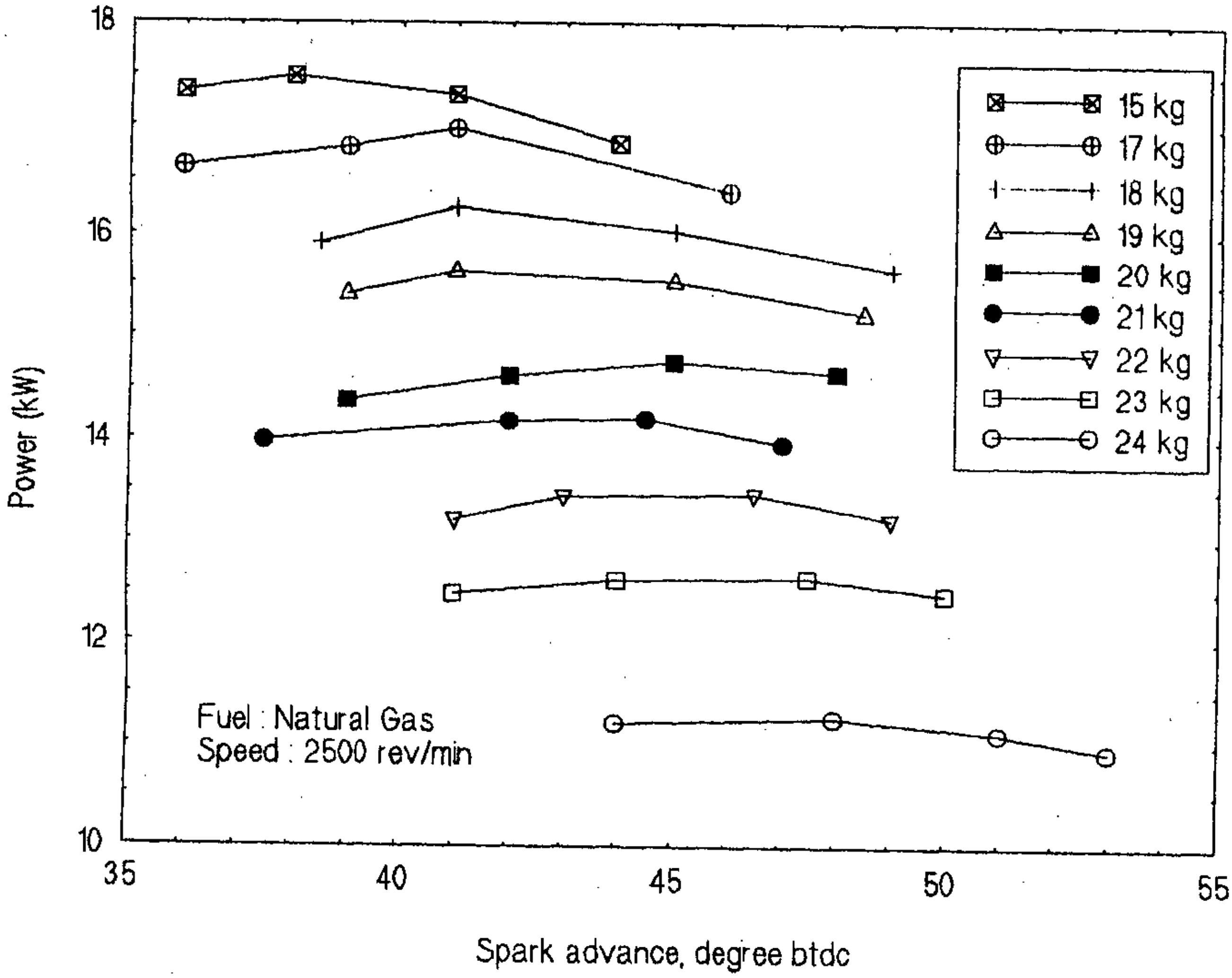


Figure - 4.11 : Variation of Power with Spark Advance.

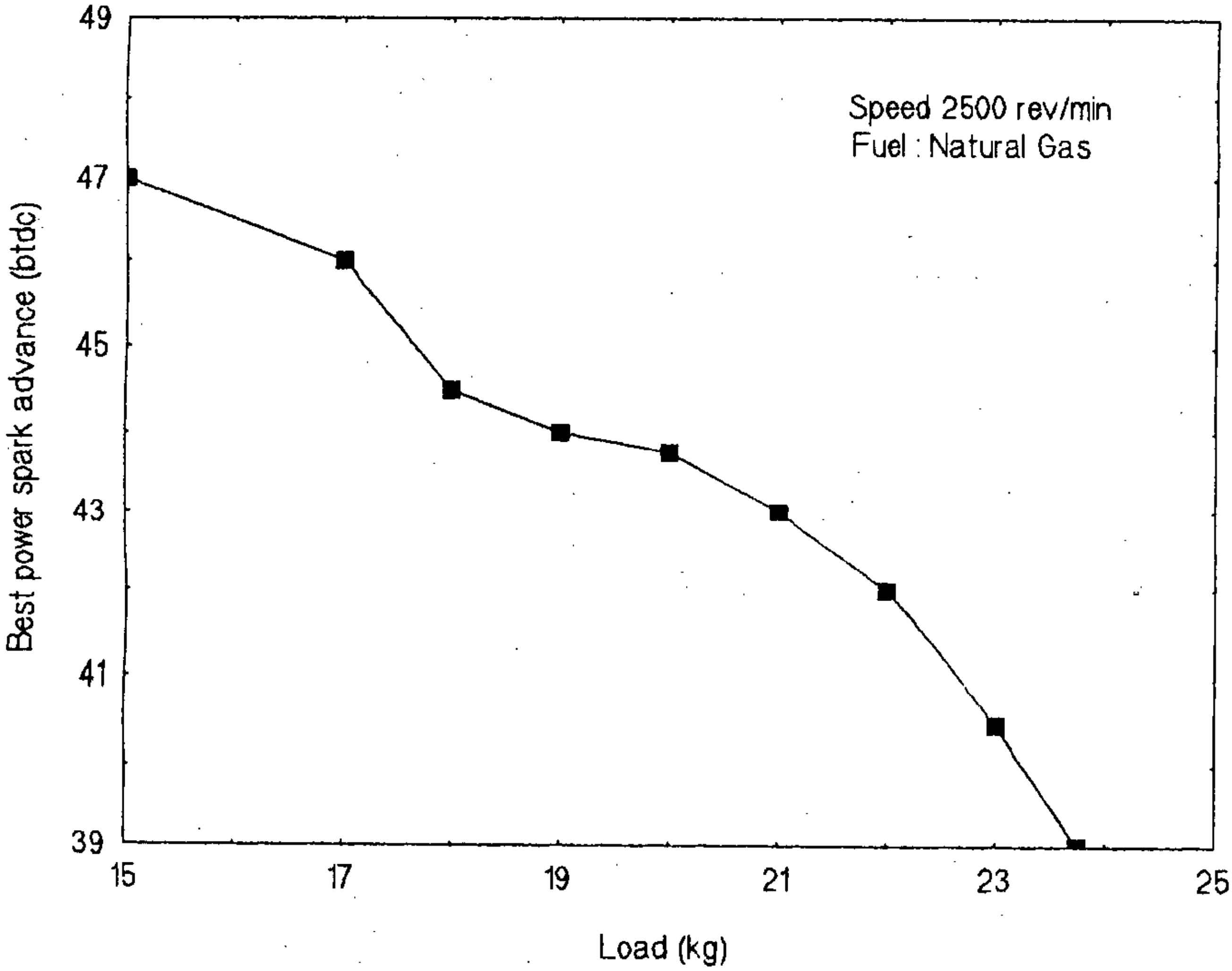


Figure – 4.12 : Variation of Best Power Spark Advance with Load.

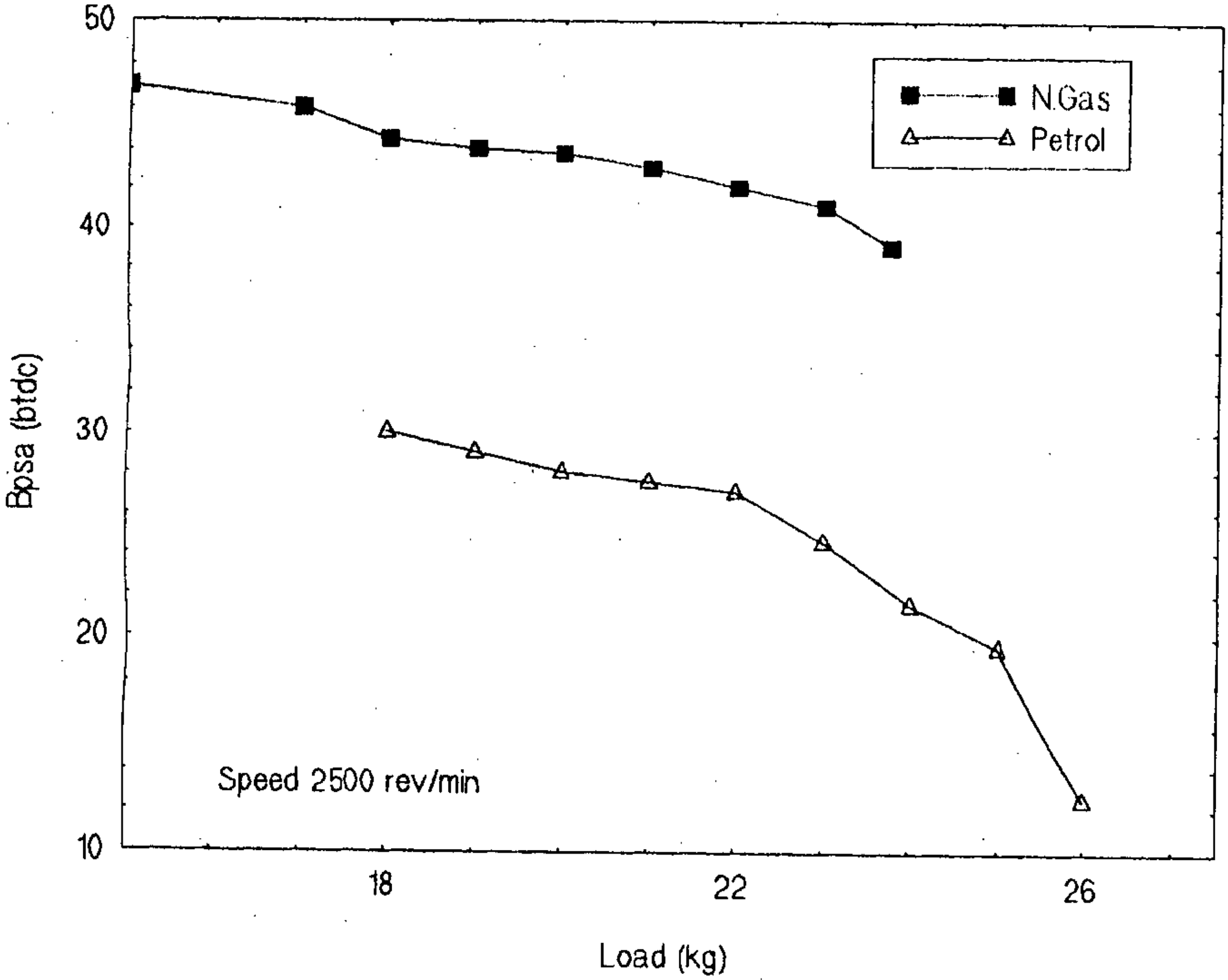


Figure - 4.12(c) : Variation of Best power Spark Advance with Load.

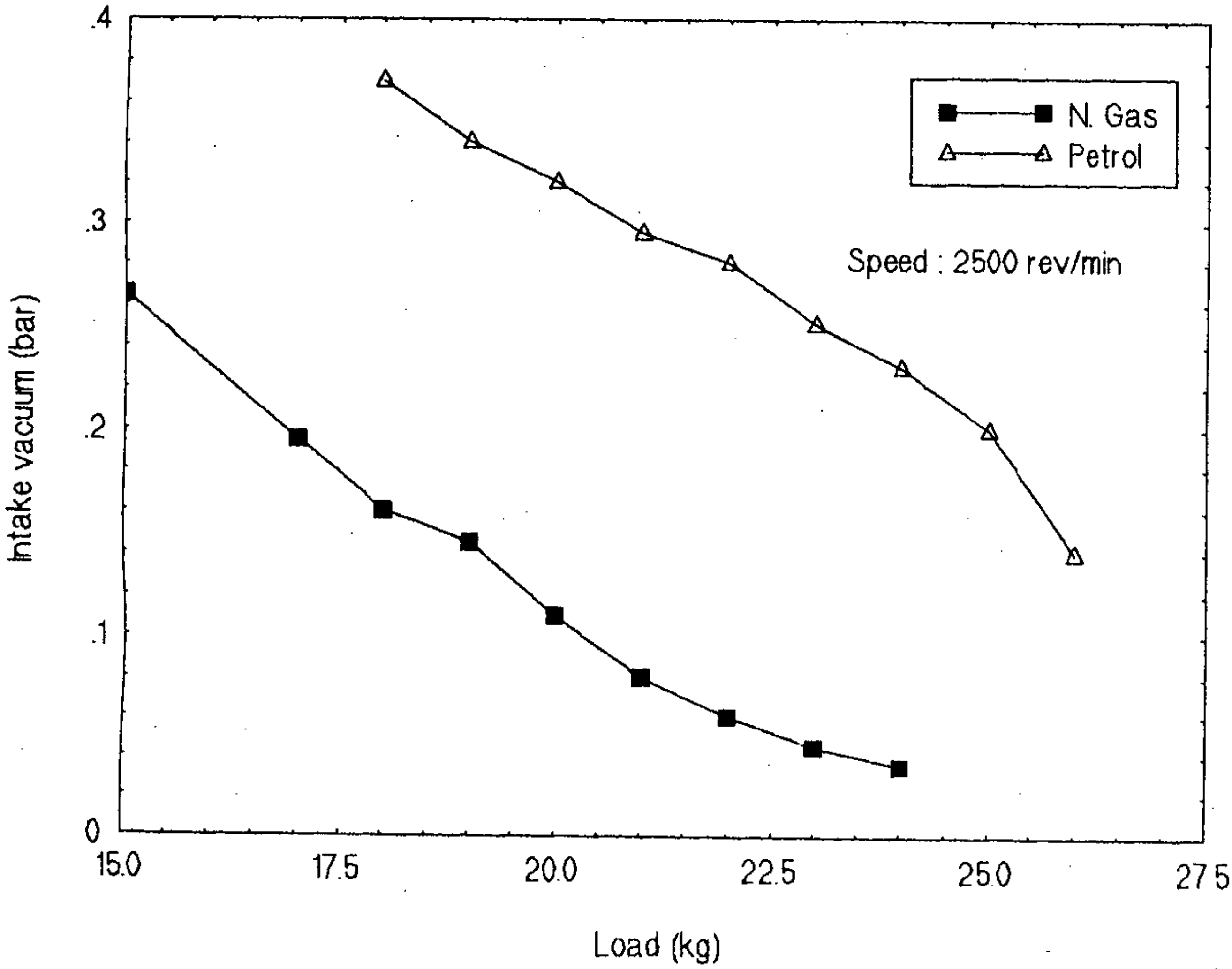


Figure - 4.13 : Variation of Intake Vacuum with Load.

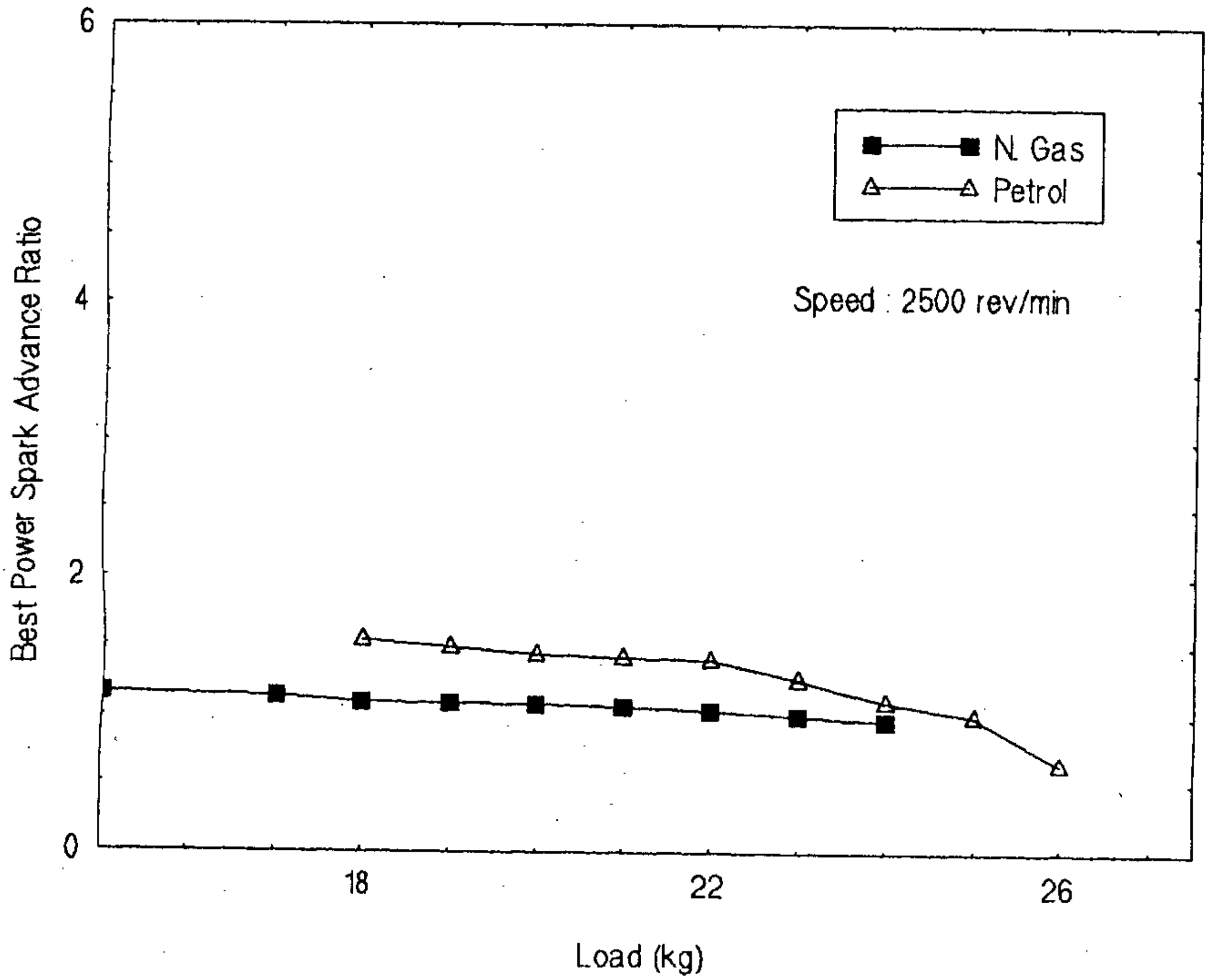


Figure - 4.14(a) : Load Vs. Bpsa Ratio

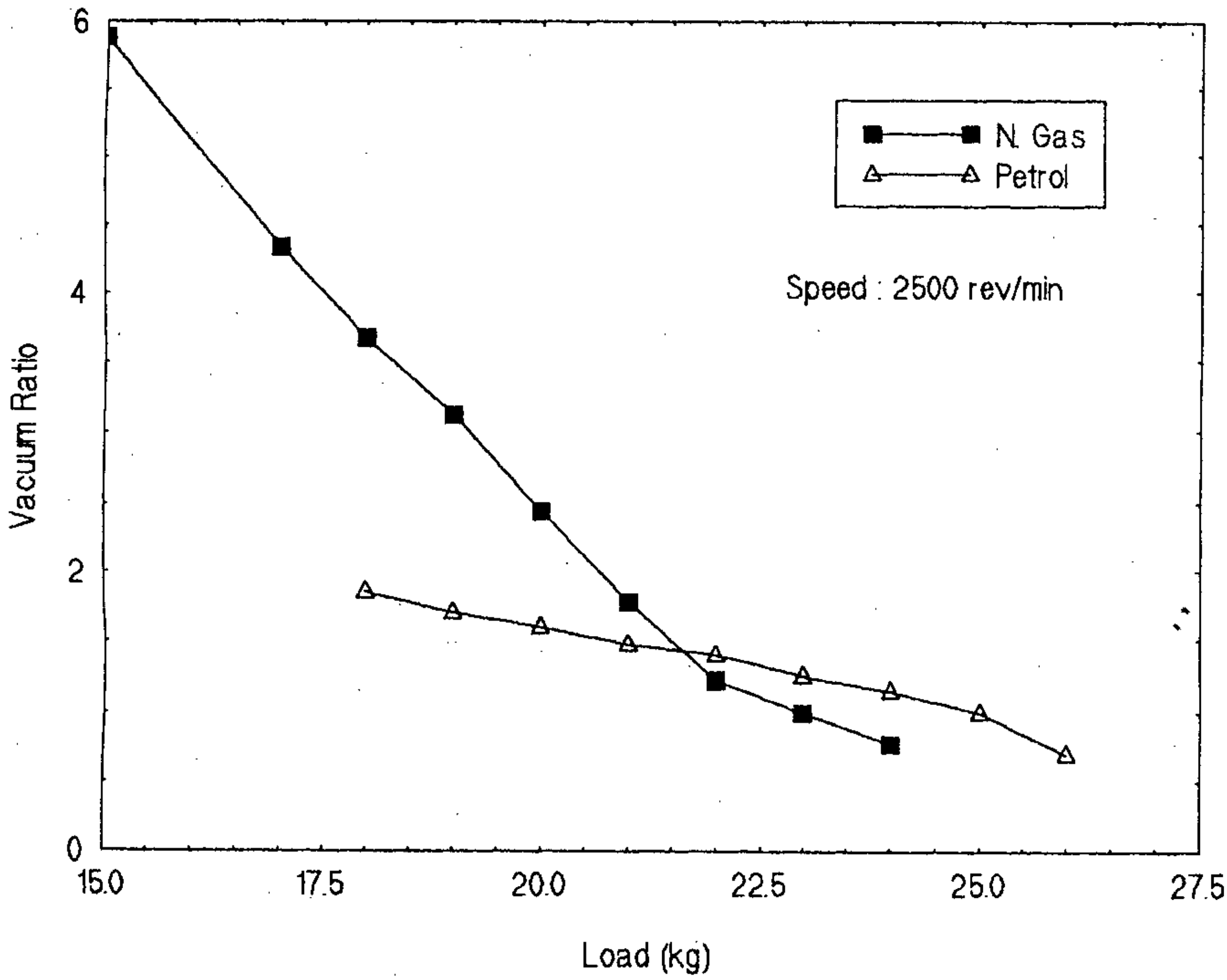


Figure - 4.14(b) : Load Vs. Vacuum Ratio

APPENDIX - A

SAMPLE CALCULATIONS

SAMPLE CALCULATION

This chapter shows the different methods of calculating different parameters from the readings taken while the engine was tested. Table - A1 shows a typical set of readings while the engine was tested with petrol and natural gas, respectively.

TABLE - A1 : FUEL : PETROL LOAD: VARIABLE SPEED : 2500rev/min

DB temp = 36 °C ϕ = 63 % α = 0.97424
WB temp = 30 C ρ_{air} = 1.1451 kg/m^3 β = 1.00448

Obs	Av. Speed (rev/min)	Load (Kg)	Stand Power (kW)	Spark Advan (btdc)	Exhaust Temp. °C	Cooling Water Temp. °C	Manometer reading (cm of Hg)
8	2508	25.00	18.96	19.5	350	92.5	4.3

Fuel Collected (cc)	Time (sec)	Fuel Flow Rate (g/h)	Air Flow Rate (kg/h)	A/F Ratio (Mass)	Stand. Bsfc (g/kW-h)	Intake Manifold Vacuum (bar)
40	15.55	6418	76.35	11.90	346.35	0.2

FUEL: N.GAS. LOAD : VARIABLE SPEED : 2500 rev/min

DB temp = 37°C ϕ = 58.5 % α = 0.97242
WB temp = 30°C ρ_{Air} = 1.142 kg/m^3 β = 1.00479

Obs	Av. Speed (rev/min)	Av. Load (kg)	Stand. Power (kW)	Spark Advn. (btdc)	Rota meter reading	Gas Flow Rate m^3/h	Eq. Oil Flow Rate (g/h)	Mano. reading (cm of Hg)
7	2513	22.00	16.71	41	31	6.64	5812	6.0

Air Flow Rate (kg/h)	Eqvt. A/F Ratio (Mass)	A/G Ratio (Mass)	Stand.Eqv Bsfc. (g/kW-h)	Vacuum Gauge (bar)	Exhaust Temp. °C	Cooling Water Temp. °C
90.06	15.47	17.16	355.91	0.06	346	89.5

SPEED:

The speed was measured with a magnetic pick-up and digital display type tachometer which was provided with the dynamometer set-up.

LOAD:(kg)

Load was measured with an electronic load-cell transducer attached with the dynamometer assembly. The system was calibrated with dead weights, prior to experiments.

POWER AT SITE:(kW)

For the dynamometer used (Model no. TFJ-250L), power in kW was calculated using the relationship:

$$Power\ (kW) = \frac{W \times N \times 0.7355}{2500}$$

Where,

W = Load on dynamometer (kg)

N = Speed in rev/min

2500 = Dynamometer constant

1 PS = 0.7355 kW.

From the data :

N = 2508 rev/min

W = 25.0 kg

P = (25 x 2508 x 0.7355) / 2500

= 18.44 kW

STANDARD POWER:(kW)

Standard power is calculated according to the ISO Standard 3046 (British Standard 5514). Standard reference conditions are -

Barometric pressure:	P_r =	100 kpa
Air temperature:	T_r =	300°k (27°C)
Relative humidity:	ϕ_r =	60%

Site Conditions :

Barometric pressure:	P_x =	101.32 kpa
Air temperature:	T_x =	309°k (36°C)
Relative humidity:	ϕ_x =	63%

From British Standard 5514 :

Power Adjustment Factor, α = 0.97242

Fuel Consumption Factor, β = 1.00448

Standard power (kW) = Power at site / α

Standard Power (kW) = 18.44 / 0.97242

= 18.96 kW

FUEL FLOW RATE:(g/h)

The time taken for the consumption of 40 cc petrol fuel was 15.55 sec. The specific gravity of the fuel used was 0.693. Hence the density of the fuel was 0.693 gm/cc.

The fuel flow rate = (40 $\times$ 0.693 / 15.55) $\times$ 3600

= 6418 g/h.

BRAKE SPECIFIC FUEL CONSUMPTION (bsfc) : (g/kW-h)

bsfc = Fuel flow rate (g/h) / Brake power at site (kW)

= 6418/18.44

= 347.9 g/kW-h

Standard bsfc is calculated as, Standard bsfc = bsfc/β

Where, β = Fuel consumption adjustment factor determined by using ISO 3046 (BS 5514) for site conditions.

Using BS 5514, for Petrol (Liquid,SI) β = 1.00448

So, Standard Bsfc = 347.9/1.00448 = 346.34 g/kW-h

GAS FLOW RATE:(m^3/hr)

The gas supplied to the engine was passed through a rotameter(ie, an area flowmeter). From the rotameter float reading the gas flow rate was directly calculated using the following relationship, which was determined by calibration of the rotameter.

$$Y = 0.359 + 0.203\ (X) \quad m^3/h$$

$$\text{or,} \quad Y = (0.359 + 0.203\ X) \times (1000/60) \quad l/min$$

Where, Y = Gas flow rate.

X = Rotameter reading (Height of the float).

From data :

X = 31

So, Y = 0.359 + 0.203 (31)

= 6.64 m^3/h

Density of Natural gas = 0.79 kg/m^3

Hence, Mass flow rate of gas = 0.79 x 6.64

= 5.246 kg/h

EQUIVALENT OIL FLOW RATE : (g/h)

Equivalent oil flow rate corresponding to the gas flow rate was calculated using the relation,

Eq.oil flow rate = (Y $\times$ Calorific Value of N.gas)/(Calorific value of petrol) g/h

Where Y is the gas flow rate in m^3/h.

Calorific value of Natural gas used (Titas) = 1036 BTU/ft^3 = 38.5 MJ/m^3

Calorific value of Petrol used = 44 MJ/kg

Hence, Eq. Oil flow rate = (6.64 x 38.5 x 1000) / (44)

= 5812 g/h

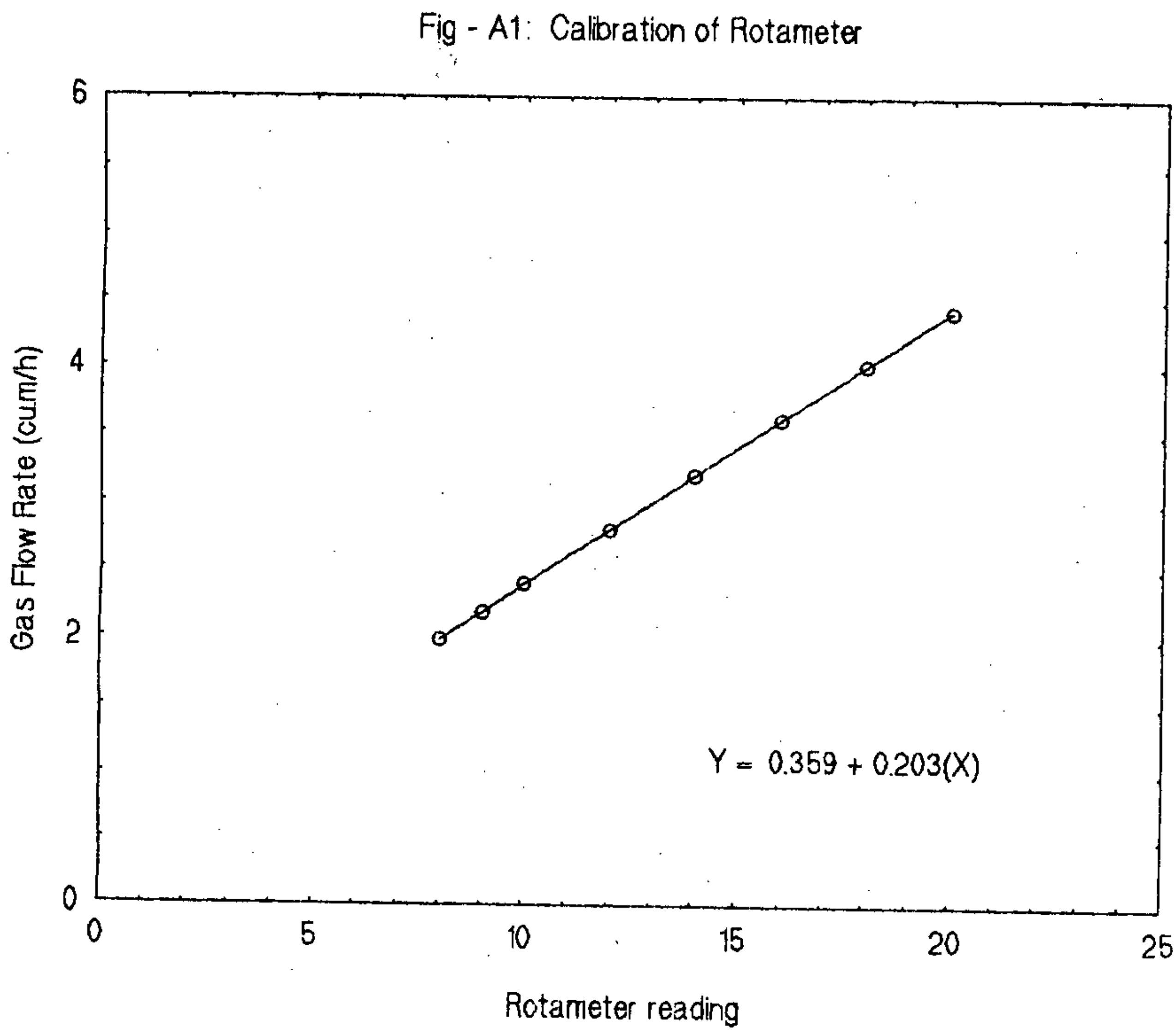


Fig - A1 : Calibration of the Rotameter.

AIR FLOW RATE: (kg/h)

Air flow rate was measured using an airdrum,a parabolic nozzle and an inclined draft gauge. Mercury(Hg) was used as manometric fluid in the manometer which was inclined at an angle of 30 degree with the horizontal.

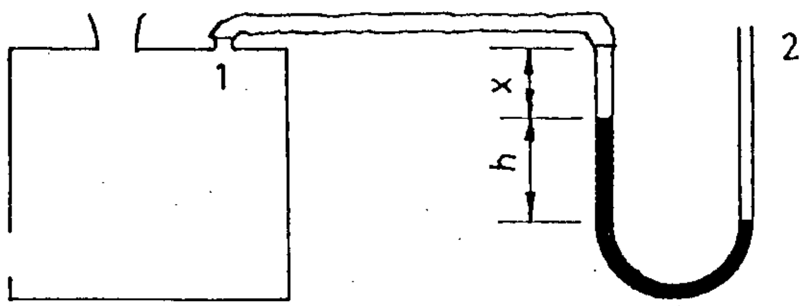
Density of Hg, γ_{Hg} = 13543 kg/m^3

Density of Air, γ_{Water} = 1.1451 kg/m^3

Co-efficient of discharge of the nozzle C_d = 0.92

Nozzle dia used = 0.01905m (0.75inch)

Vertical Manomiertic deflection = Manometer reading / 2



For manometer,

$$P_1 + x.\gamma_{Air} + h.\gamma_{Hg} = P_2 + (x+h).\gamma_{Air}$$

$$\therefore \quad P_2 - P_1 = h(\gamma_{Hg} - \gamma_{Air})$$

Applying Bernoulli's Equation at positions inside(1) and outside(2) the air-drum,

$$\frac{P_1}{\gamma_{Air}} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma_{Air}} + \frac{V_2^2}{2g} + Z_2$$

Since, $Z_1 = Z_2$ and $V_2 = 0$.

Therefore,

$$\frac{V_1^2}{2g} = \frac{P_2 - P_1}{\gamma_{Air}}$$

Hence from the last two equations :

$$\frac{V_1^2}{2g} = \frac{h(\gamma_{Hg} - \gamma_{Air})}{\gamma_{Air}} \qquad \therefore V_1 = \sqrt{\frac{2gh(\gamma_{Hg} - \gamma_{Air})}{\gamma_{Air}}}$$

So, the actual air flow rate is ,

$$= C_d\left(\frac{\pi d^2}{4}\right) V_1 . 3600 \quad m^3/hr$$

$$= C_d\left(\frac{\pi d^2}{4}\right) V_1 . \gamma_{Air} . 3600 \quad kg/hr$$

$$= C_d\left(\frac{\pi d^2}{4}\right) \sqrt{\frac{2gh(\gamma_{Hg} - \gamma_{Air})}{\gamma_{Air}}} . \gamma_{Air} . 3600 \quad kg/hr$$

Using the data from Table - A1 , Air flow rate -

$$= \frac{(0.92)\,(3.1416)\,(0.01905)^2}{(4)} \sqrt{\frac{2x9.81x(4.3/2)x(13543-1.1451)}{2x1.1451}}$$

$$x\ 1.1451\ x\ 3600$$

$$= 76.34 \quad kg/hr$$

AIR-FUEL RATIO (A/F) :

This term applies for petrol, A/F Ratio in mass basis is defind as,

A/F Ratio = Mass flow rate of Air intake / Mass flow rate of Petrol used

= (76.34 x 1000) / (6818)

= 11.89

AIR-GAS RATIO (A/G) :

This term is applied for Natural gas, A/G Ratio in mass basis is defind as,

A/G Ratio = Mass flow rate of Air intake / Mass flow rate of gas used

= (90.06) / (5.246)

= 17.16

EQUIVALENT AIR-FUEL RATIO :

This term is also used for Natural gas, Eq. A/F Ratio in mass basis is defind as,

Eq. A/F Ratio = Mass flow rate of air intake / Equivalent oil flow rate

= (90.06) / (5.812)

= 15.49

APPENDIX -B

RELEVANT TABLES

TABLE 1.1 **Estimated world energy reserves, consumption and "lifetimes" (13).**

Source of Energy	Reserves Joule x 10^{-20}		Yearly Consumption rate Joule x 10^{-20}		Life of Reserves
	Proven	Ultimate	1979(%)	2000(%)	(Years)
Natural gas	30	80	0.50(16)	0.80(13)	60
Petroleum	40	200	1.40(43)	1.90(31)	30; 60
Oil shale	20	230	-		
Tar sands	30	100	-		
Coal	200	3000	0.70(22)	1.50(25)	280
Uranium	110	3×10^6	0.1(3)	1(16)	1100
Breeder Reactor	$>10^{10}$	$>10^{15}$			
Renewables	-	-	0.50(16)	0.90(15)	-
World Total	--	--	3.2(100)	6.1(100)	-

* Assumed demand growth to year 2000 = 2.9% per annum

TABLE 1.2 : Fieldwise Production of Natural Gas.

Gas field	Year of discovery	Proven reserves x 10^{10} cu.m	Cumulative production x 10^{10} cu.m	Total discounted reserves x 10^{10}cu.m
Titas	1962	8.430	1.657	6.773
Habiganj	1962	6.003	0.600	5.403
Sylhet	1655	1.926	0.399	1.527
Chhatak	1959	0.566	0.074	0.492
Kailashtilla	1962	1.699	0.068	1.631
Bakhrabad	1968	7.872	0.241	7.631
Rashidpur	1960	3.002	--	3.002
Semutang	1969	0.654	--	0.654
Kutubdia	1977	2.195	--	2.195
Begumganj	1980	0.371	--	0.371
Feni	1981	1.028	--	1.028
Beani Bazar	1982	0.688	--	0.688
Kamta(Tongi)	1982	0.365	0.048	0.317
Fenchyganj	1988	0.991	--	0.991
Total	--	35.790	3.087	32.703

Source: Petro-Bangla.

TABLE 1.3 : Field wise chemical composition (mol.%) of naturl gas.

Field	Molecular contents in percentage					
	Methane	Ethane	Propane	Butane and Higher	Nitrogen	carbon dioxide
Titas	96.85	1.81	0.39	0.30	0.31	0.34
Habiganj	97.63	1.32	0.27	0.12	0.57	0.09
Sylhet	95.15	1.51	0.34	0.35	2.14	0.51
Chhatak	97.72	0.24	Trace	Nil	2.00	0.04
Kailashtila	95.99	2.59	0.81	0.37	Nil	0.24
Bakhrabad	94.31	3.54	0.75	0.45	0.39	0.56
Rashidpur	99.60	0.12	0.04	0.02	0.17	0.05
Seumtang	96.40	1.70	0.14	0.01	0.35	0.86
Kutubdia	95.72	2.87	0.67	0.30	0.37	0.07
Begumganj	94.98	3.64	0.75	0.24	0.29	0.10
Beani Bazar	92.69	4.74	1.42	0.76	0.28	0.11
Feni	95.70	3.29	0.68	0.18	--	0.15
Kamta	94.66	3.57	0.47	0.07	0.81	0.42
Fenchuganj	96.67	2.33	0.58	0.26	0.11	0.05

Source : BBS

Notes: (--) Not yet determined.

TABLE 1.3: Continued.

Field	Total Sulphur grains/100 SCF	Condensate BRLS/100 MMCF	Calorific value (BTU/Cft)
Titas	Nil	1.50	1036
Habiganj	Nil	0.3-3.03	1020
Sylhet	0.38	3.70	1052
Chhatak	Nil	0.004	1007
Kailashtila	Nil	10.13	1050
Bakhrabad	Nil	2.00	1022
Rashidpur	--	0.30	1014
Semutang	--	--	n.a.
Kutubdia	--	Trace	1043
Begumganj	--	0.29	1064
Beani Bazar	--	18.20	n.a.
Feni	--	3.60	n.a.
Kamta	--	0.16	--
Fenchuganj	--	--	--

Source: BBS

TABLE 2.A.1: Properties of gasoline, gas oil and some alternative fuels (13).

Fuel	Formula	Density @ 20/4°C kg/l	b.p. or range °C	m.p. pour point °C	Viscosity @ 40°C Ost
Gasoline	$C_nH_{1.87n}$	0.78	30-200	-60	3.5
Gas oil	$C_nH_{1.8n}$	0.84	180-360	-20	10.0
Hydrogen	H_2	0.071*	-252.7	-259.1	0.18*
Methane	CH_4	0.424*	-161.5	-182.3	-
Methanol	CH_3OH	0.796	64.7	-97.8	0.55
Ethanol	C_2H_5OH	0.794	78.5	-114.9	1.10
Ammonia	NH_3	0.615	-33.4	-77.7	0.20
Hydrazine	N_2H_4	1.013	113.5	1.4	0.70
Nitromethane	CH_3NO_2	1.123	101.1	-28.5	0.47
Sunflower oil	♦	0.924	90 @ 360	-15	30.0
Carbon black	C	1.860	4827**	3367**	-

* At boiling point

** Sublimes

♦ C 77.5%, H 10.9%, (O + N) 11.6% mass

TABLE 2.A.1: Continued.

Fuel	Vapn.enthalpy @ 37.8°C kJ/kg°K	Sp. heat capacity @ mid.b. range kJ/kg°K	Sp. energy Gross MJ/kg	Sp. energy Net MJ/kg	Flash point °C
Gasoline	350	2.4	47.3	44.0	−45
Gas oil	230	1.9	45.7	42.9	68
Hydrogen	450	8.7*	142.4	120.2	–
Methane	512	3.9	55.5	50.0	–
Methanol	1080	2.5	22.7	19.9	15
Ethanol	845	2.4	30.2	27.2	14
Ammonia	1370	4.7	22.4	18.6	−62
Hydrazine	1254	3.1	19.4	16.7	52
Nitromethane	564	1.8	12.0	10.9	–
Sunflower oil	–	2.6	39.5	36.9	180
Carbon black	–	–	31.4	31.4	–

TABLE 2.A.1: Continued.

Fuel	(A/F) mass		Minimum ignition energy mJ	S.I.T. °C
	Stoi.	Flamm. range		
Gasoline	14.6	25-4	-	312
Gas oil	14.5	21-3	0.3	247
Hydrogen	34.2	345-5	0.019	574
Methane	17.2	34.3-10.2	0.28	540
Methanol	6.5	12.6-1.6	0.19	385
Ethanol	9.0	18.5-2.7	0.65	365
Ammonia	6.1	8.9-4.6	7.5	651
Hydrazine	4.3	18.3-0	-	270
Nitromethane	1.7		-	419
Sunflower oil	12.3		-	360
Carbon black	11.5		-	-

TABLE 2.A.2: Fuel Properties of Vegetable Oils (12).

Vegetable oil	Visc.* mm^2/s	Cetane** no.	Hg kJ/kg	Cloud point °C	Flash point °C	Denisty kg/l	Pour point °C
Castor	297.0	?	37274	None	260	0.9537	-31.7
Corn	34.9	37.6	39500	-1.1	277	0.9095	-40.0
Cottonseed	33.5	41.8	39468	1.7	234	0.9148	-15.0
Crambe	53.6	44.6	40482	10.0	274	0.9044	-12.2
Linseed	27.2	34.6	39307	1.7	241	0.9236	-15.0
Peanut	39.6	41.8	39782	12.8	271	0.9026	-6.7
Rapeseed	37.0	37.6	39709	-3.9	246	0.9115	-31.7
Safflower	31.3	41.3	39519	18.3	260	0.9144	-6.7
H.O. Safflower	41.2	49.1	39516	-12.2	293	0.9021	-20.6
Sesame	35.5	40.2	39346	-3.9	260	0.9133	-9.4
Soyabean	32.6	37.9	39623	-3.9	254	0.9138	-12.2
Sunflower	33.9	37.1	39575	7.2	274	0.9161	-15.0
Typical No. 2 diesel	2.7	47.0	45343	-15.0	52	0.8400	-33.0

TABLE 2.A.2: Continued.

Vegetable Oil	Water & sed. % v	Carbon residue % w	Ash % w	Sulphur % w
Castor	trace	0.22	< 0.01	0.01
Corn	trace	0.24	0.01	0.01
Cottonseed	0.04	0.24	0.01	0.01
Crambe	0.20	0.23	0.05	0.01
Linseed	trace	0.22	< 0.01	0.01
Peanut	trace	0.24	0.005	0.01
Rapeseed	trace	0.30	0.054	0.01
Safflower	trace	0.25	0.006	0.01
H.O. Safflower	trace	0.24	< 0.001	0.02
Sesame	trace	0.25	< 0.01	0.01
Soyabean	trace	0.27	< 0.01	0.01
Sunflower	trace	0.23	< 0.01	0.01
Typical No. 2 diesel	< 0.05	< 0.35	< 0.01	< 0.01

* Measured at 38°C

** Measured using a modified form of ASTM D613 in which ignition were observed visually.

APPENDIX - C

EXPERIMENTAL DATA

'ABLE-4.1: FUEL : PETROL LOAD: 15 Kg SPEED : 2500 rev/min.

DB = 36°C ϕ = 63 %
WB = 30°C ρ_{air} = 1.1451 kg/m^3

Obsr.	Speed rev/min	Load Kg	Power kW	Spark Advn. btdc	Exhaust Temp. °C	Cooling Water Temp.	Manometer Reading cm of Hg	Intake Manifold Vacuum (bar)	Fuel Collected CC
1	2465	14.6	10.59	15	291	86.5	2.1	0.41	40
2	2480	14.7	10.72	17	289	86.5	2.1	0.41	40
3	2510	15.1	11.15	22	288	86	2.1	0.41	40
4	2512	15.15	11.23	25	287	87	2.1	0.41	40
5	2518	15.2	11.26	30	281	86	2.1	0.41	40
6	2503	15	11.05	35	279	87	2.1	0.41	40
7	2482	14.9	10.88	40	275	87	2.1	0.41	40

'able-4.1 Continued.

Time of Collection sec.	Bsfc	α	β	Stand Power kW	Stand. Bsfc g/kW-h	Air Flow Rate g/h	Fuel Flow Rate g/h	A/F Ratio (Mass)
21.44	439.6	0.97424	1.00448	10.87	437.64	53.35	4654.47	11.46
21.44	434.2	0.97424	1.00448	11.00	432.26	53.35	4654.47	11.46
21.44	417.44	0.97424	1.00448	11.44	415.58	53.35	4654.47	11.46
21.44	414.47	0.97424	1.00448	11.53	412.62	53.35	4654.47	11.46
21.44	413.36	0.97424	1.00448	11.56	411.58	53.35	4654.47	11.46
21.44	421.22	0.97424	1.00448	11.34	419.34	53.35	4654.47	11.46
21.44	427.8	0.97424	1.00448	11.17	425.90	53.35	4654.47	11.46

TABLE-4.2 : FUEL-PETROL LOAD: 15 Kg SPEED : VARIABLE

DB temp= 36°C ϕ = 63 % α = 0.97424
WB temp= 30°C ρ_{air} = 1.1451 kg/m3 β = 1.00448

Obsr.	Av. Speed rev/min.	Load Kg	Power kW	Spark Advan. (btdc)	Exhaust Temp. °C	Cooling Water Temp. °C	Manometer Read. cm of Hg
1	2711	15.1	12.04	31	297	89	2.3
2	2522	15.0	11.13	30	281	86.5	2.1
3	2192	14.75	9.51	28	253	86	1.5
4	2001	15.0	8.83	26.5	241	85	1.25
5	1704	15.0	7.52	25	229	82	0.9
6	1508	14.9	6.61	23.5	224	82	0.7
7	1203	15.3	5.415	22	221	72	0.5
8	1020	15.2	4.56	20	220	69	0.3

Table-4.2 Continued.

Fuel Collected CC	Time sec.	Fuel Flow Rate g/h	Air Flow Rate g/h	A/F Ratio (Mass)	Stand. Bsfc g/kW-h	Intake Manifold Vacuum (bar)
40	20.14	4954.9	55.83	12	411.5	0.41
40	21.44	4654.5	53.354	11.463	418	0.41
40	24.01	4156.3	45.09	10.85	437	0.405
40	25.76	3874	41.163	10.625	438.7	0.4
40	30.07	3318.6	34.928	10.52	441.3	0.4
40	33.97	2937.7	30.804	10.48	444.5	0.395
40	40.11	2488	26.034	10.46	459.5	0.365
40	47.51	2100.5	20.166	9.6	460.62	0.325

'ABLE-4.3: FUEL : NATURAL GAS LOAD: 15 Kg SPEED : VARIABLE

DB = 35.5°C ϕ = 65.8 % α = 0.97425
WB = 30°C ρ_{air} = 1.147 kg/m^3 β = 1.00447

Obsr.	Av. Speed rev/min	Load kg	Standard Power kW	Spark Advance (btdc)	Exhaust Temp. °C	Cooling Water Temp. °C	Manometer reading cm of Hg
1	1009	15.00	4.56	20	235	70	0.4
	1020	15.20	4.68	24.5	234	72	0.4
	1024	15.35	4.74	28	233	70.5	0.4
	1019	15.35	4.72	30	235	71	0.4
	1016	15.30	4.69	32	236	71	0.4
	1015	15.35	4.67	33	236	73	0.4
2	1210	15.15	5.53	21.5	237	68.5	0.7
	1220	15.25	5.61	23.5	236	71	0.7
	1230	15.35	5.69	25	236	70.5	0.7
	1240	15.55	5.82	27.5	237	70.5	0.7
	1249	15.60	5.88	30.5	238	71.5	0.7
	1245	15.55	5.85	33	238	71	0.7
	1243	15.50	5.83	35	239	72	0.7
	1236	15.40	5.75	39	238	73	0.7
	1200	14.95	5.15	44	241	79	0.7
3	1480	14.85	6.64	23.5	249	69	0.9
	1485	15.05	6.75	27.5	248	66	0.9
	1487	15.15	6.81	30.5	248	68	0.9
	1494	15.20	6.85	33.5	247	67	0.9
	1480	15.15	6.77	38	247	70.5	0.9
	1465	15.00	6.64	42	249	69.5	0.9
	1438	14.50	6.29	42	250	72	0.9
4	1987	14.65	8.78	30	261	82	2.15
	2002	14.80	8.95	35	262	82	2.15
	2005	14.80	8.96	38	262	84	2.15
	2005	14.95	9.05	40	265	84	2.15
	2015	14.95	9.07	41.5	266	83	2.15
	2009	14.80	8.98	46	266	81	2.15
	1994	14.70	8.85	48.5	267	82	2.15
	1982	14.55	8.70	52	269	84	2.15
	1945	14.15	8.31	55	270	85	2.15
5	2480	14.90	11.15	34	300	82	3.1
	2495	15.00	11.29	38	301	82	3.1
	2505	15.10	11.42	44	300	83.5	3.1
	2510	15.15	11.48	48	301	85	3.1
	2500	15.15	11.43	50	302	86	3.1
	2480	15.00	11.23	53	304	86.5	3.1
	2460	14.80	10.99	56	307	87	3.1

able-4.3 Continued.

Rotameter reading	N.Gas Flow-rate m^3/h	Equivalent Oil Flow-rate g/h	Air Flow-rate kg/h	A/G Ratio (Mass)	Equvt. A/F Ratio (Mass)	Stand. Eqvt.Bsfc g/kW-h	Intake Vacuum (bar)
10	2.38	2087.3	23.30	12.31	11.16	467.0	0.26
10						455.8	
10						449.5	
10						451.8	
10						454.7	
10						456.8	
12.5	2.89	2531.2	30.83	13.49	12.18	467.5	0.27
12.5						460.6	
12.5						454.5	
12.5						444.4	
12.5						439.7	
12.5						442.0	
12.5						443.6	
12.5						445.0	
12.5						477.1	
14	3.19	2797.18	34.88	13.81	12.47	430.4	0.265
14						423.2	
14						420.1	
14						417.0	
14						421.9	
14						430.3	
14						454.2	
19	4.21	3683.9	53.91	16.21	14.63	428.4	0.26
19						420.6	
19						420.1	
19						415.8	
19						414.9	
19						419.1	
19						425.5	
19						432.5	
19						452.8	
23.25	5.07	4437.6	64.74	16.16	14.59	406.4	0.265
23.25						401.2	
23.25						396.9	
23.25						394.9	
23.25						396.5	
23.25						403.8	
23.25						412.5	

-LE-4.4: FUEL: N. GAS LOAD: 15 Kg SPEED : VARIABLE

DB temp = 35.5°C ϕ = 65.8 %

WB temp = 30°C ρ_{air} = 1.147 kg/m^3

-peed rev/min.	1025	1250	1500	2000	2500
BPSA (btdc)	28	30.5	33.5	41.5	48
Stand. Power (kW)	4.74	5.86	6.85	9.07	11.98

TABLE-4.5 : FUEL : PETROL LOAD: VARIABLE SPEED : 2500rev/min

DB temp = 36°C ϕ = 63 % α = 0.97424

WB temp = 30°C ρ_{air} = 1.1451 kg/m^3 β = 1.00448

Obs	Av. Speed rev/min	Load Kg	Stand Power kW	Spark Advan (btdc)	Exhaust Temp. °C	Cooling Water Temp. °C	Manometer reading cm of Hg
1	2503	17.9	13.58	30	312	84	2.5
2	2497	19	14.34	29	314	87	2.7
3	2497	19.94	15.07	28	318	88.5	2.9
4	2507	21.06	15.96	27.5	324	90	3.1
5	2508	21.96	16.67	27	329	92	3.4
6	2500	23.06	17.45	24.5	336	91	3.7
7	2493	24.96	18.04	21.5	343	93	4.0
8	2508	25.01	18.96	19.5	350	92.5	4.3
9	2497	26.1	19.71	12.5	363	94	4.8

Table-4.5 Continued.

Fuel Collected CC	Time sec.	Fuel Flow Rate g/h	Air Flow Rate kg/h	A/F Ratio (Mass)	Stand. Bsfc g/kW-h	Intake Manifold Vacuum (bar)
40	19.54	5107	58.21	11.39	384.88	0.36
40	18.80	5308	60.50	11.40	378.88	0.34
40	18.40	5423	62.70	11.56	368.44	0.32
40	17.91	5572	64.83	11.63	357.24	0.295
40	17.21	5802	67.88	11.69	351.42	0.28
40	16.81	5936	70.82	11.92	348.53	0.25
40	16.31	6118	73.64	12.03	347.07	0.23
40	15.55	6418	76.35	11.90	346.35	0.2
40	14.73	6775	80.66	11.89	351.43	0.14

TABLE 4.6: FUEL: N.GAS. LOAD : VARIABLE SPEED : 2500 rev/min

DB temp = 37°C ϕ = 58.5 % α = 0.97242
WB temp = 30°C ρ_{Air} = 1.142 kg/m^3 β = 1.00479

Obsr	Av. Speed rev/min	Av. Load kg	Stand. Power (kW)	Spark Advn. (btdc)	Rota meter reading	Gas Flow Rate m^{3}/h	Eq. Oil Flow Rate g/h	Mano. reading cm of Hg
1	2518	15.12	11.52	44	23.25	5.07	4437.6	3.2
	2522	15.16	11.56	48	23.25			3.2
	2521	14.98	11.42	51	23.25			3.2
	2494	14.98	11.30	41	23.25			3.2
	2487	14.94	11.29	53	23.25			3.2
2	2505	16.92	12.82	41	25	5.43	4747.9	3.9
	2514	17.04	12.96	44	25			3.9
	2517	17.04	12.97	47.5	25			3.9
	2505	16.92	12.82	50	25			3.9
3	2498	17.96	13.57	41	26.25	5.68	4969.7	4.3
	2517	18.14	13.81	43	26.25			4.3
	2517	18.16	13.83	46.5	26.25			4.3
	2494	18.00	13.58	49	26.25			4.3
	2478	14.88	13.41	51	26.25			4.3
4	2482	19.14	14.37	37.5	27.5	5.93	5191.3	4.7
	2501	19.26	14.57	42	27.5			4.7
	2513	19.20	14.59	44.5	27.5			4.7
	2490	19.04	14.34	47	27.5			4.7
	2459	18.76	13.96	50	27.5			4.7
5	2485	19.66	14.78	39	28.5	6.13	5368.7	5.1
	2498	19.90	15.04	42	28.5			5.1
	2516	19.98	15.11	45	28.5			5.1
	2503	19.90	15.07	48	28.5			5.1
	2457	19.30	14.35	51	28.5			5.1
6	2502	20.92	15.83	39	30	6.44	5634.7	5.6
	2514	21.12	16.06	41	30			5.6
	2508	21.04	15.97	45	30			5.6
	2481	20.84	15.64	48.5	30			5.6
7	2493	21.70	16.36	38.5	31	6.64	5812	6.0
	2513	21.98	16.71	41	31			6.0
	2501	21.78	16.49	45	31			6.0
	2472	21.58	16.08	49	31			6.0
8	2497	22.62	17.08	36	32.25	6.89	6033.1	6.1
	2510	22.76	17.27	39	32.25			6.1
	2517	22.92	17.45	41	32.25			6.1
	2485	22.44	16.87	46	32.25			6.1
9	2494	23.64	17.84	36	33.25	7.01	6211	6.15
	2503	23.66	17.91	38	33.25			6.15
	2492	23.60	17.79	41	33.25			6.15
	2461	23.26	17.32	44	33.25			6.15

Table 4.6 Continued.

Air Flow Rate	Eqvt. A/F Ratio (Mass)	A/G Ratio (Mass)	Stand.Eqv Bsfc. g/kW-h	Vacuum Gauge bar	Exhaust Temp. °C	Cooling Water Temp. °C
65.77	14.82	16.42	394.31	0.265	328	85
65.77			392.62	0.265	325	86
65.77			397.60	0.265	321	80.5
65.77			401.78	0.265	317	86.5
65.77			401.87	0.265	324	86.5
72.61	15.29	16.93	379.18	0.195	335	86
72.61			374.87	0.195	334	87
72.61			374.48	0.195	330	87.5
72.61			378.98	0.195	329	88
76.24	15.34	16.99	374.70	0.16	340	87
76.24			368.83	0.16	339	87
76.24			367.73	0.16	333	88
76.24			374.50	0.16	330	89
76.24			379.38	0.16	326	89
79.71	15.35	17	369.63	0.145	345	90
79.71			364.38	0.145	346	90
79.71			363.99	0.145	336	90
79.71			370.32	0.145	337	90
79.71			380.70	0.145	334	90
83.03	15.46	17.13	371.66	0.11	339	89
83.03			365.34	0.11	340	88
83.03			361.50	0.11	339	89
83.03			364.56	0.11	334	89.5
83.03			382.95	0.11	331	89
87.00	15.44	17.1	364.22	0.08	349	87.5
87.00			358.98	0.08	343	77
87.00			361.17	0.08	340	87
87.00			368.63	0.08	339	87
90.06	15.47	17.16	363.48	0.06	348	89
90.06			355.91	0.06	346	89.5
90.06			360.77	0.06	339	89.5
90.06			369.82	0.06	338	90
90.81	15.05	16.67	361.38	0.045	357	92
90.81			357.49	0.045	353	91
90.81			353.85	0.045	351	94
90.81			366.12	0.045	348	92
91.18	14.68	16.26	356.41	0.035	365	92
91.18			354.95	0.035	363	93
91.18			357.23	0.035	362	92.5
91.18			367.04	0.035	360	93

TABLE 4.7 :

FUEL : Natural Gas   Speed = 2500 rev/min.

Load	15	17	18	19	20	21	22	23	24
Sp. Adv	47	46	44.5	44	43.75	43	42	40.5	39
Vacuum	0.265	0.195	0.16	0.145	0.11	0.08	0.06	0.045	0.035
SR	1.16	1.136	1.098	1.086	1.08	1.062	1.037	1.0	0.963
VR	5.89	4.33	3.55	3.22	2.44	1.77	1.33	1.0	0.77

FUEL : Petrol   Speed = 2500 rev/min.

Load	18	19	20	21	22	23	24	25	26
Sp. Adv	30	29	28	27.5	27	24.5	21.5	19.5	12.5
Vacuum	0.37	0.34	0.32	0.295	0.28	0.25	0.23	0.2	0.14
SR	1.54	1.48	1.43	1.41	1.38	1.25	1.1	1.0	0.64
VR	1.85	1.7	1.5	1.475	1.4	1.25	1.15	1.0	0.7