# A STUDY OF ALTERNATIVE FUELS IN S.I. ENGINES FOR FUEL EFFICIENCY AND EXHUAST EMISSION CONTROL

A THESIS
SUBMITTED IN FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF
THE DEGREE OF

## DOCTOR OF PHILOSOPY

By

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# CERTIFICATE

Certified that the thesis entitled **"A STUDY OF ALTERNATIVE FUELS IN SI ENGINES FOR FUEL EFFICIENCY AND EXHAUST EMISSION CONTROL"** being submitted by Shri Sagar Maji, to the University of Delhi, Delhi, Faculty of Technology, in the Department of Mechanical Engineering at Delhi College of Engineering, in fulfillment of the requirements for the award of the degree of **DOCTOR OF PHILOSOPHY**,is a record of bonafide research work carried out by him. He has worked under our guidance and supervision and has fulfilled the requirements for the submission of this thesis which has attained the standard required for a Ph.D. degree of the University of Delhi. The results presented in this thesis have not been submitted elsewhere for the award of any degree of diploma.

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## *Nomenclature*

| | |
|---|---|
| A | area |
| Ap | area of piston |
| CL | connecting rod length |
| CO | carbon monoxide |
| D | diameter or bore |
| E | total internal energy |
| $\dot{E}$ | rate of change of internal energy |
| e | specific internal energy |
| F | fuel-air equivalence ratio |
| h | sensible or specific enthalpy |
| h' | heat transfer coefficient |
| $h_iM_i$ | energy convection in or out of the system at location i |
| k | ratio of specific heats |
| M | mass |
| $\dot{m}$ | mass flow rate |
| NO | nitric oxide |
| $NO_x$ | oxides of nitrogen |
| O | oxygen |
| P | pressure |
| $p\dot{V}$ | rate of change of mechanical work |
| Q | heat transfer |
| $Q_i$ | rate of change of heat transfer |
| R | specific gas constant or radial distance |
| Re | Reynolds number |
| RGF | residual gas fraction |
| S | length of stroke |
| Sp | mean piston speed |
| T | Temperature |
| V | cylinder volume |

| | |
|---|---|
| $V_{CL}$ | clearance volume |
| $Z_1$, $Z_2$, $Z_3$ | defined in section 5.3 |

### *Subscripts*

| | |
|---|---|
| ch | charge condition |
| e | equilibrium or exhaust gas |
| em | exhaust gas condition before mixing |
| g | gas side |
| i | any location or surface or reaction |
| mx | mixing |
| w | wall |

### *Greek Letters*

| | |
|---|---|
| $\phi$ | equivalence ratio |
| $\eta$ | thermal efficiency |
| $\theta$ $\theta$ | crank angle |

### *Abbreviations*

| | |
|---|---|
| BDC | bottom dead centre |
| bmep | break mean effective pressure |
| bsfc | break specific fuel consumption |
| CI | compression ignition |
| CNG | compressed natural gas |
| CN | cetane number |
| CO | carbon monoxide |
| $CO_2$ | carbon dioxide |

| | |
|---|---|
| CR | compression ratio |
| EFI | electronic fuel injection |
| EGR | exhaust gas re-circulation |
| ETBE | ethyl tertiary butyl ether |
| fmep | frictional mean effective pressure |
| HC | hydrocarbon |
| IC | internal combustion |
| imep | indicated mean effective pressure |
| isfc | indicated specific fuel consumption |
| LNG | liquefied natural gas |
| LPG | liquefied petroleum gas |
| MBT | minimum advance for best torque |
| MPFI | multipoint fuel injection |
| MTBE | methyl tertiary butyl ether |
| NG | natural gas |
| NGV | natural gas vehicles |
| NO | nitric oxide |
| $NO_2$ | nitrogen dioxide |
| $NO_x$ | oxides of nitrogen |
| OEM | original equipment manufacturer |
| ON | octane number |
| RVP | reid vapour pressure |
| SI | spark ignition |

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# ABSTRACT

The rapid depletion of petroleum fuels, their ever-increasing cost and increasing global concern for environmental pollution caused by the petroleum fuels have led to an intensive search for alternative fuels in internal combustion engines. This search has led to the recognition of several alternative fuels, which can be successfully used in spark ignition engines. However, the choice of an alternative fuel in a particular country will be governed by the availability of resources and economics apart from how efficiently a particular alternative fuel can be used in engines. Alternative fuels like ethanol and compressed natural gas can be attractive in India because of the country's wide agricultural base and large coal reserves. Ethanol can be produced from the fermentation of sugar cane, corn and grain stocks that are also renewable energy sources. Apart from natural gas reserves methane, which is one of its principal constituent, can also be produced from coal and biomass conversion thus making it much more available energy base.

Hence, in the present thesis, experimental and theoretical investigations that were undertaken to visualize the potential of ethanol and compressed natural gas as spark ignition engine alternative fuels and reducing the harmful emissions are reported. Experimental results of a single cylinder engine operated on gasoline, E-85 (85 percent ethanol and 15 percent gasoline by volume) and other ethanol-gasoline blends (E-20, E-10) at similar equivalence ratio, engine speed, bmep and compression ratio of 8.0 (nominal compression ratio of Indian 4-stroke gasoline operated vehicles) are reported. The results of these fuels are compared at similar equivalence ratios. WOT and constant speed performances were also compared. The performance of E-85 at different ignition timings and compression ratios are also presented. Other results presented include, the effect of intake air temperature, performance at idling, performance of catalytic converter and pressure crank angle history. Similar works were also carried out with the variation of equivalence ratio and engine speed using CNG and the results were compared with gasoline. Experimental work was also carried out to evaluate the performance of CNG using dual spark plugs.

The results showed that at a given equivalence ratio, ethanol fueling resulted in 2-3 percent greater thermal efficiency, up to 50% lower CO and 26 percent lower $NO_x$ emissions. However, bsfc and hydrocarbons emissions were higher with E-85 as compared to gasoline operation. It was observed that MBT timings for E-85 operation were 4–6 degrees less advanced than gasoline operation depending on equivalence ratios. At WOT, ethanol (E-85) fueling resulted in higher power and efficiency by 4-7% and 3-5% respectively as compared to gasoline. With gasoline an ignition timing of 20 degrees before top dead centre gave lowest fuel consumption while for E-85, an ignition timing of 16 degrees bTDC gave minimum fuel consumptions. An intake air temperature of 50 deg. centrigrade resulted in best performance with E-85 because of improved mixture formation. Increased compression ratio from 8 to 10 and 12 increased thermal efficiency by 3 and 8 percent respectively. It also increased exhaust emissions of $NO_x$ and HC by 15 and 40 percent respectively. Increased idling speed increased fuel consumption and decreased CO and HC emissions. Stable operation at idling was achieved at 900 rev./min. for both gasoline and E-85. The analysis of pressure-crank angle diagram at half throttle operation and WOT showed that with ethanol (E-85), pressure increased more rapidly. Peak pressures were also higher with E-85. A reasonably satisfactory correlation was found between theoretically predicted and experimentally obtained results on engine performance.

Single cylinder engine operation with CNG showed 3 to 5 percent higher thermal efficiency and 15 percent lower bsfc as compared to gasoline. Also CO emissions were lower by 30 – 80 percent in rich zone and $NO_x$ by about 12 percent at an equivalence of 1.0. At WOT CNG operation resulted in 10 to 12 percent lower power, however, thermal efficiency and bsfc was better with CNG as compared to gasoline. Dual spark plug operation increased power output by 3 to 5 percent.

# Chapter 1

## INTRODUCTION

Till the last half of 20th century Indian transport was mainly depended on railroad and the vehicles, which were pulled by animals. Railroad was one of the major source of transport which used conventional energy sources like coal, diesel and electricity. But today transport scenario in India has changed drastically. From traditionally railroad dependent economy, the country has moved on to become a road dependent economy in nineties, as depicted in Table 1.1 (1)*

**Table 1.1 Rising share of road traffic in India (1)**

| **Year** | **Road** *(Billion Passenger km)* | **Rail** *(Billion Passenger km)* | **Road** *(Billion Ton km)* | **Rail** *(Billion Ton km)* |
|---|---|---|---|---|
| 1951 | 40 | 67 | 12 | 44 |
| 1985 | 920 | 230 | 240 | 170 |
| 1995 | 3000 | 330 | 700 | 290 |
| 2000 | 5000 | 440 | 1500 | 360 |

As the share of the road transport has increased over the years the number of vehicles on road has also increased significantly. From about 2.5 million vehicles in1951 the country has today about 45 million vehicles and it is growing at a rate of 20 percent per annum for last three years (2).

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* Numbers in parentheses indicate references at the end of the thesis

Considering the energy scenario of the country, India can not afford to let the conventional automobile industry grow at this rate unless some solution to the energy requirement for automotive sector is found. This type of growth will result in a huge demand of petroleum products, most of which will have to be imported. Not only will it adversely affect India's energy security, but it will also mean a significant drainage of precious foreign exchange reserve. At the moment, the country consumes about 104 million tons/year of petroleum products as compared to 56 million tons/year of in 1990-91 (3). The percentage share of self sufficiency in petroleum products has also reduced to only 31 percent as compared to 56 percent in 1990-91, resulting in import of crude oil of about 15.37 million dollars in 2000-01 (3). With not much addition to the domestic reserve of crude oil expected in the foreseeable future, and with no switch over to alternative energy, India's dependence on imported oil may go up to 90 percent within a few years.

Another serious problem associated with increased number of vehicles is the growing contribution of vehicular pollutions to overall air pollution especially in urban areas. A study (4) on percentage contribution by various sources of air pollution in Delhi estimated that contribution of vehicular pollution would increase from 23 percent in 1970-71 to 72 percent in 2000-01, although most of the vehicles these days are equipped with modern emission control technology. It is concluded in that study that the primary reason for increased share of vehicular pollution is the increase of number of vehicles, which has increased by 10 folds in last three decades.

Interestingly, until early seventies, there was a preponderance of commercial vehicles and market was small and stagnant. By the end of seventies, personal transport vehicles increased nearly three times that of commercial vehicles. During the end of year 2000 personal transport vehicles constituted 12 times the commercial vehicles and the share of these vehicles is likely to increase further (4). Spark ignited gasoline engines power majority of these personal transport vehicles primarily due to its higher octane number and ability to form homogenous mixture with air. Because of these properties gasoline can be used in spark ignition engines to give smooth and less noisy combustion as compared to diesel fuel.

Since we have large population of personal transport vehicles powered by spark ignited gasoline engines, it is imperative to overcome the two basic problems i.e. large amount of gasoline consumption and exhaust emissions particularly so in the light of projected increase from the present population. In the next few paragraphs, the design and the operating variables, which effect fuel consumption and exhaust emissions, mechanism of formation of exhaust emissions, techniques for improving them and need for alternative fuels in spark ignition are discussed.

## 1.1 *SPARK IGNITION ENGINE--EFFECT OF OPERATING VARIABLES*

Spark ignition engines are widely used as a prime mover in the transport sector especially in passenger cars. The vehicles powered by this class of engines consume a large quantity of fuel and also contribute significantly to the air pollution especially in urban areas. Any improvement in their combustion quality and efficiency is likely to result not only in conserving fuel but also contribute significantly to the reduction of harmful emissions. In a spark ignition engine, fuel induction system outside the combustion chamber prepares the fuel-air mixture in accordance with the requirements of the engine. The fuel-air ratio is normally maintained closed to stoichiometric (chemically correct) for satisfactory ignition and flame propagation. The fuel-air equivalence ratio** or mixture strength affects the fuel consumption and exhaust emissions. For using alternative fuels in S.I. engines, temperature of intake air and time available for the preparation of mixture are important for better fuel economy and reduced emissions.

Four important factors, which significantly affect the process of mixture formation, are:

i) The time available for the preparation of mixture
ii) The temperature of the incoming air of the intake manifold
iii) Quality of the fuel supplied and
iv) The design of the induction system and combustion chamber.

For liquid alternate fuels atomization, mixing and vaporization are the processes, which require a finite time to occur. In high-speed engines the time available for mixture formation is very small. For example an engine running at 3000 rev/min, the induction process lasts less than 0.02 seconds. To complete these processes in such a small period of time require great ingenuity in designing the intake system. Because of the short time available, complete and efficient mixing, vaporization and distribution is difficult to achieve. Temperature is a factor, which controls the vaporization process of the fuel. A high temperature normally results in a high rate of vaporization. The temperature of the mixture can be increased by heating the intake manifold but this could reduce the power due to reduction in mass flow. The volatility of the fuel affects vaporization and distribution of the fuel.

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** *Actual fuel-air ratio divided by stoichiometric fuel-air ratio, normally denoted by* $\phi$

***So whenever any liquid alternative fuel is used in existing gasoline engine proper modifications in the intake system has to be done to compensate for different volatility characteristics of the new fuel.*** However, for gaseous alternative fuels a better mixture formation will result from the existing intake system as they mix with air quite easily to form a homogenous mixture. Apart from mixture formation other important parameters, which affect fuel consumption and exhaust emissions are the ignition timing and compression ratio.

### *1.1.1* ***Ignition Timing***

The combustion process in a spark ignition engine is initiated by an electrical discharge produced between two electrodes of a spark plug and the spark is triggered towards the end of combustion stroke. It is to be noted that spark is to be produced in a repeatable manner over full range of load and speed of the engine at the appropriate moment in an engine cycle.

With a homogeneous fuel air mixture in the cylinder, spark energy of the order of 1 MJ and duration of a few microseconds would suffice to initiate the combustion process (5). For a given engine design, the optimum spark timing varies with the type of fuel due to variation in flame velocities of different alternative fuels apart from engine speed, inlet manifold pressure and mixture composition. The ignition timing also has to be controlled accurately. If ignition is too late, there is a possibility that combustion will not be completed when the exhaust valve opens at the end of the expansion stroke and this may overheat the exhaust valve. Conversely, if ignition is too early, there will be too much pressure rise before the end of the compression stroke and power will be reduced. Thus, the increase in work during the compression stroke is greater than the increase in work done on the piston during the expansion stroke. Also, with early ignition the peak pressure and temperature may be sufficient to cause knock. ***So for using any alternative fuel in an engine, it will be essential to suitably optimize the ignition timing for improved performance and lower exhaust emissions.*** Ignition timing that is optimized for maximum power is called the MBT (Minimum Advance for Best Torque) ignition timing. The definition of MBT timing is somewhat arbitrary. It may correspond to the timing that gives a one percent fall in the peak torque. Alternatively, it might be when the operator of an engine first detects a fall in torque, as the ignition timing is being retarded. However for some engines operating at full throttle (particularly at low speed operation) knock will be encountered before the MBT ignition timing. In this case the ignition will be retarded from MBT to preserve a knock margin. This is to allow for manufacturing tolerances and engine aging, making some engines more susceptible to knock.

Fig. 1.1 shows the relation between spark timing (ST) and brake mean effective pressure (bmep) for number of spark ignited gasoline engines at different chassis dynamometer speeds (6). It can be noticed from the figure at all speeds the bmep curve is flat at the vicinity of maximum bmep. However, a large variation in nitrogen oxide emissions is found with very little variation in ignition timing (7). The variation of nitric oxide emissions with the variation of ignition timing is explained in section 1.6.2.

Clearly, if the ignition timing is slightly retarded say 5 degrees from that of maximum bmep then the engine power and in turn break specific fuel consumption (bsfc) will hardly suffer; yet under same operating conditions the oxide of nitrogen will be greatly reduced. Retarded timing also somewhat reduces the engine's octane requirement (8,9).

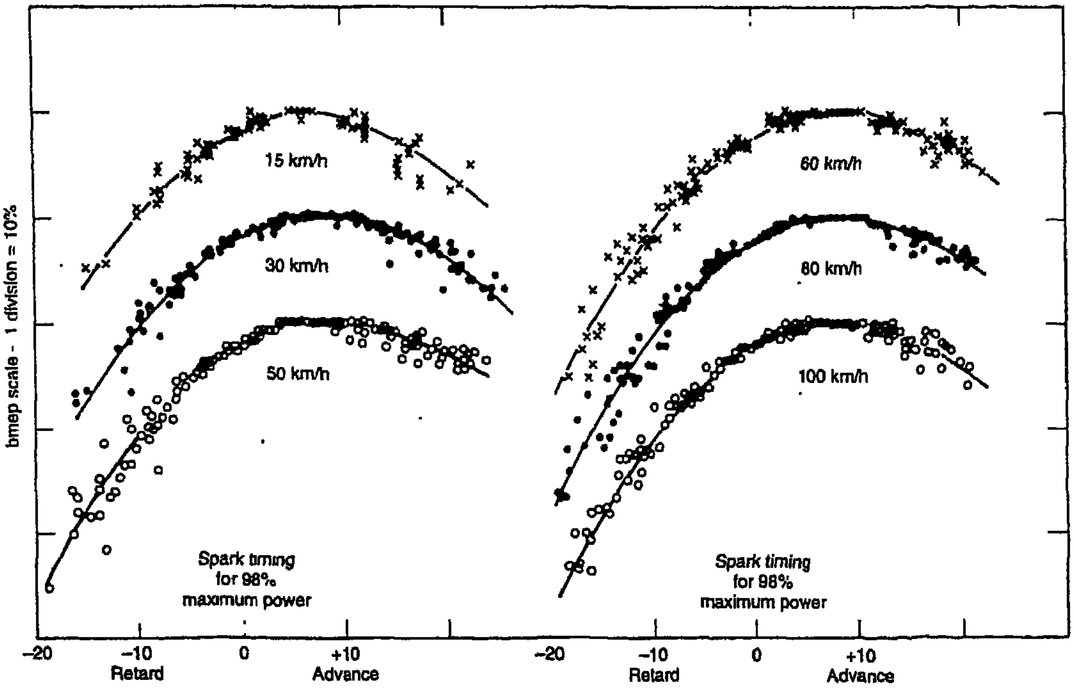

**Fig1.1-Relation between Spark Timing and Brake Mean Effective Pressure (6)**

### *1.1.2 Compression Ratio*

As shown in Fig.1.2 (10) increasing the compression ratio increases both the indicated and brake thermal efficiencies. In other words increasing compression ratio decreases both indicated and brake specific fuel consumption. The indicated specific fuel consumption (isfc) improves at a faster rate with increasing compression ratio than the brake specific fuel consumption (bsfc). This is due to the increased friction and heat losses at higher compression ratios. From the studies revealed by Heywood (11), over the compression ratio range that is accessible to spark ignition engines with available fuel and conversion efficiency, the engine power increases by about 3% per unit increase in compression ratio. The exhaust gas temperature decreases with the increase of compression ratio

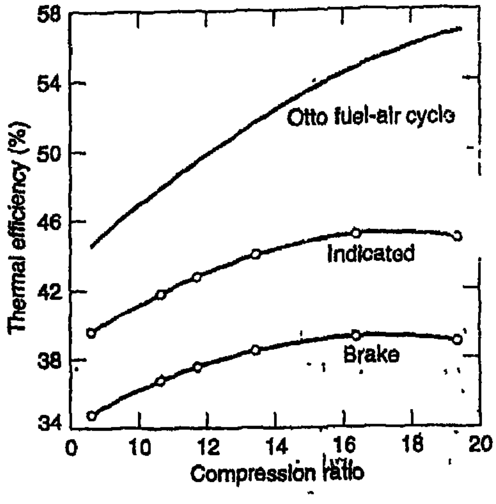

**Fig 1.2-Effect of Compression Ratio on Thermal Efficiency (10)**

and efficiency until the compression ratio corresponding to maximum efficiency is reached. It is also to be noted that as compression ratio increases exhaust emissions of hydrocarbons and oxides of nitrogen also increases (12,13). ***Since most alternative fuels have high octane numbers higher compression ratios can be used with these fuels.*** But due to increased compression ratio exhaust emissions of HC and $NO_x$ increases. This is due to the higher surface to volume ratio and elevated temperatures at higher compression ratios.

## *1.2 PART LOAD PERFORMANCE*

In spark ignition engines brake specific fuel consumption will be infinite at idling since engine is producing no useful work. But exhaust emissions of carbon monoxide at idling is very high because the engine uses richer mixture at idling. As the load increases, the brake specific fuel consumption drops, goes through a minimum and may or may not increase depending on how the load is increased at that point.

In the case of spark ignition engines, opening the throttle and increasing the delivery ratio increases the load. This has little effect on the indicated efficiency but it slightly increases the friction, and significantly reduces the pumping losses. Again, the dominant factor is the increase in mechanical efficiency. At constant fuel-air ratio, the brake specific fuel consumption drops with increasing load all the way to the point of maximum load so long as the indicated mean effective pressure (imp) increases faster than the frictional mean effective pressure (fmep). In engines running at a fuel-air ratio less than that corresponding to maximum power (about $\phi = 1.1$), the load can be increased further by increasing fuel-air ratio. This causes bsfc to begin increasing with load, once the engine is running at rich mixture.

The effect of the fuel-air ratio on brake specific fuel consumption of a spark ignition engine at part load is shown in Fig. 1.3 (6). The spark

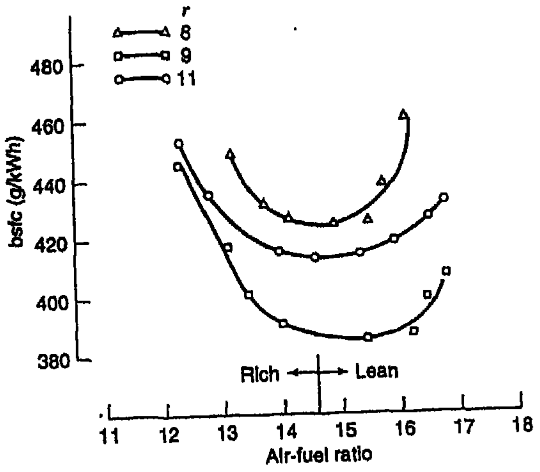

**Fig 1.3-Effect of A/F Ratio on BSFC (6)**

ignition engine is most efficient when running at stoichiometric or slightly lean mixture. At very lean fuel-air ratios, the engine wastes fuel Because of misfire, and at rich fuel-air ratios it wastes fuel since there is not enough oxygen present to liberate all of the fuel's energy. ***Alternative fuels like alcohols like CNG, LPG and hydrogen can operate at leaner mixtures than gasoline, which will result in better fuel economy and lower emissions levels.***

## *1.3 PERFORMANCE AT IDLING*

At idling, i.e. zero load, no useful work is done. The optimum fuel-air ratio must be such that the engine consumes the minimum fuel at this speed and yet the running is steady. This ratio is nearer to $\phi = 1$. But in actual practice, the normal idling mixtures are from $\phi = 1.2$ to 1.5. Under idling conditions, the pressure at the inlet manifold is much below atmospheric and there is always, some air leaking into the inlet system, through the inlet valve guides. Thus the combustion fuel-air ratio would be less than that delivered by carburetor. This leakage of air will depend upon valve guide clearance as well as pressure in the manifold and will increase with the wear of valve guides. Also, there is a dilution of the charge due to residual burnt gases. Because of uncertainty with regard to idling mixture requirements, carburetors are fitted with choke arrangements, i.e. manual arrangement to supply rich mixtures at starting, idling and very small part loads.

To understand this behaviour of fuel-air ratio being rich at idling, we may note that the presence of exhaust gases at the end of exhaust stroke does not vary much with changes in load, while, the temperature decreases with either load or a fall in speed because of heat losses. Since the volume of the combustion chamber is constant the mass of the exhaust gas remaining as residual tends to be constant, i.e. $m = pV/RT$.

On the other hand, mass of the fresh charge inducted on each stroke depends on manifold pressure and therefore, on throttle opening reaching a maximum, for each speed, at the wide open throttle (WOT). It follows that the percentage of the residual gas diluting the fresh charge increases as the throttle is closed (increases with manifold vacuum). The dilution is not serious at most loads and speeds but becomes a problem during idling and decelerating conditions. During idling, since the pressure in the inlet manifold is much less than that of residual gas, exhaust gas will flow from the combustion chamber into the inlet manifold when the inlet valve opens. Then as the piston descends on intake stroke, the exhaust gas is drawn back into the cylinder along with portion of fresh charge, and the over all mixture contains a high percentage of exhaust gas.

To offset the increasing dilution of fresh charge by residual gas as the throttle is closed, the carburetor must furnish an increasingly richer mixture or else the engine may misfire which increases fuel consumption and exhaust emission carbon monoxide and hydrocarbons.

## 1.4 ***CYCLIC VARIATIONS***

Cyclic variations are another important parameter in the combustion process, which are caused by variations in mixture motion within the cylinder at the time of spark. It is due to the cycle-by-cycle, variations in the amounts of air and fuel fed to the cylinder in each cycle, and variations in mixing of fresh mixture and residual gases within the cylinder of each cycle especially in the vicinity of the spark plug. Combustion variability for operation away from the lean limit is not strongly dependent on flow conditions at the spark plug and early flame behavior but more dependent on the cyclic repeatability of the overall cylinder flow pattern (14).

Because of variability in the combustion event, the spark must be retarded to prevent detonation in the highest burn-rate or most advanced cycles, placing the slow burn or retarded cycles further from optimum. This limits the peak output power, increases the exhaust gas temperature and reduces the peak load efficiency. Of greater importance is the influence of combustion variability accompanying lean or dilute operation. Poor combustion stability arising from the slower burning that accompanies increased dilution, limits the amount of dilution. This in turn limits the ultimate reduction in the level of nitrogen oxides.

## *1.5* ***EMISSIONS FROM SI ENGINES***

An ideal combustion process of the hydrocarbon fuel can be represented by the following chemical equation:

$C_mH_n$ + Air = $mCO_2$ + $n/2H_2O$ + Nitrogen

Thus, the products of ideal combustion are carbon dioxide, water vapour and nitrogen, all of which are harmless except that Carbon dioxide is a "greenhouse" gas. But by the nature of the combustion process itself and due to other physical factors spark ignition engines exhaust, emit oxides of nitrogen (nitric oxide and small amount of nitrogen dioxide – collectively known as NOx), carbon monoxide (CO) and organic compounds which are unburned or partially burned hydrocarbons (HC). The relative amount depends on engine design and operating conditions. Piston blow by gases and fuel evaporation, which are released by, vents in the fuel tank and carburetor after engine shut down are also sources of HCs. However, in most modern engines these non-exhaust sources are effectively controlled by returning the blow by gases from the crankcase to engine intake system and by venting the fuel tank and carburetor bowl through a vapour absorbing carbon canister, which is purged by some of engine intake air during normal engine operation. Whereas the gases which are coming out through exhaust or formed during combustion, required to be reduced further.

NO forms in both the flame front and the postflame gases. In engines, however, combustion occurs at high pressure so the flame reaction zone is extremely thin (~ 0.1 mm) and residence time within this zone is short. Also, the cylinder pressure rises during most of the combustion process, so burned gases produced early in the combustion process are compressed to a higher temperature than they reached immediately after combustion.

Chemical equilibrium considerations indicate that for burned gases at typical flame temperatures, $NO_2$ /NO ratios should be negligibly small. A plausible mechanism for the persistence of $NO_2$ is the following. NO formed in the flame zone can be rapidly converted to $NO_2$ via reactions such as

$$NO + HO_2 \rightarrow NO_2 + OH$$

Subsequently, conversion of this $NO_2$ to NO occurs via

$$NO_2 + O \rightarrow NO + O_2$$

unless the NO2 formed in the flame is quenched by mixing with cooler fluid.

### 1.5.1.2 *Formation of Carbon Monoxide*

Carbon monoxide is also formed during the combustion process. With rich fuel-air mixtures, there is insufficient oxygen to burn fully all the carbon in the fuel to $CO_2$ ; also, in the high temperature products, even with lean mixtures, dissociation ensures there are significant CO levels. Later, in the expansion process, the CO oxidation process also freezes as the burned gas temperature falls. Mostly, CO emissions depend on fuel-air ratio. As compared to gasoline some fuels give lower level of CO.

### *1.5.1 Formation of Exhaust Emissions*

#### 1.5.1.1 *Formation $NO_x$*

In conventional spark-ignition engines the fuel and air are mixed together in the engine intake system, and vigorous mixing with the residual gas within the cylinder occurs during the intake process. Thus the fuel air ratio and the amount of diluent (residual gas) is approximately uniform throughout the charge within the cylinder during combustion. Oxides of nitrogen are formed throughout the high temperature burned gases behind the flame through chemical reactions involving nitrogen and oxygen atoms and molecules, which do not attain chemical equilibrium. The higher the burned gas temperature and the residence time the higher the rate of formation of $NO_x$. As the burned gases cool during the expansion stroke the reactions involving $NO_x$ freeze, and leave $NO_x$ concentrations far in excess of levels corresponding to equilibrium at exhaust conditions.

While nitric oxide (NO) and nitrogen dioxide ($NO_2$) are usually grouped together as $NO_x$ emissions, nitric oxide is the predominant oxide of nitrogen produced insider the engine cylinder. The principal source of NO is the oxidation of atmospheric (molecular) nitrogen. The mechanism of NO formation from atmospheric nitrogen has been studied extensively (15,16). It is generally accepted that in combustion of near-stoichiometric fuel-air mixtures the principal reactions governing the formation of NO from molecular nitrogen (and its destruction are) :-

$$O + N_2 = NO + N$$

$$N + O_2 = NO + O$$

$$N + OH = NO + H$$

Carbon monoxide (CO) emission from spark ignition engines are controlled primarily by the fuel-air equivalence ratio. For fuel rich mixtures CO concentrations in the exhaust increase steadily with increasing equivalence ratio, as the amount of excess fuel increases. For fuel-lean mixtures, CO concentrations in the exhaust vary little with equivalence ratio and are of order $10^{-3}$ mole fraction.

Since spark-ignition engines often operate close to stoichiometric at part load and fuel rich at full load, CO emission are significant and must be controlled. The levels of CO observed in spark-ignition engine exhaust gases are lower than the maximum values measured within the combustion chamber, but are significantly higher than equilibrium values for the exhaust conditions. Thus the processes which govern CO exhaust levels are kinetically controlled. In premixed hydrocarbon-air flames, the CO concentration increases rapidly in the flame zone to a maximum value, which is larger than the equilibrium value for adiabatic combustion of the fuel-air mixture. It is generally assumed that in the postflame combustion products in a spark-ignition engine, at conditions close to peak cycle temperatures (2800 K) and pressures (15 to 40 atm), the carbon-oxygen-hydrogen system is equilibrated. Thus CO concentrations in the immediate postflame burned gases are close to equilibrium. However, as the burned gases cool during the expansion and exhaust strokes, depending on the temperature and cooling rate, the CO oxidation process may not remain locally equilibrated.

In multicylinder engines, because CO increases rapidly as the inlet mixture becomes richer than stoichiometric, cylinder to cylinder variations in equivalence ratio about the mean value are important. Because non-uniform distribution can significantly increase average CO emissions. Thus improved cylinder-to-cylinder fuel-air ratio distribution has become essential. Also, because it is necessary to enrich the fuel-air mixture when the engine is cold, CO emissions during engine warm-up are much higher than emissions in the fully warmed-up state. Further, in transient engine operation during acceleration and deceleration, control of fuel metering has had to be improved.

### 1.5.1.3 *Formation of Hydrocarbons*

Hydrocarbon emissions result from the presence of unburned fuel in the exhaust of an engine. Hydrocarbon fuels are composed of 10 to 20 major species and some 100 to 200 minor species. Most of these same species are found in the exhaust. However, some of the exhaust hydrocarbons are not found in the parent fuel, but are hydrocarbons derived from the fuel whose structure was altered within the cylinder by chemical reactions that did not go to completion. These are about 50% of the total hydrocarbons emitted. These partial reaction products include acetaldehyde, formaldehyde, 1,3 butadiene, and benzene, which are toxic emissions.

Hydrocarbon emissions are observed to be maximum during engine start due to decreased fuel vaporization and oxidation. Four principal mechanisms, which are responsible for exhaust hydrocarbons emissions, are:

i) Flow of hydrocarbons into crevices during compression and combustion (17)
ii) Absorption of fuel vapor in oil layers and engine deposits (18)
iii) Flame quenching at the combustion chamber wall (19)
iv) Incomplete combustion or misfiring in a fraction of engine's operating cycle.

The crevices in the combustion chamber walls are small volumes with narrow entrances – into which flame is unable to penetrate. The largest of these crevice regions is the volumes between piston, piston rings and cylinder wall. Other crevice volume in production engines are threads around the spark plug, the space around the spark plug center electrode, crevices around the intake and exhaust valve heads and the head gasket crevice.

During compression unburned mixture is forced into these crevice regions. Since these volumes are thin they have large surface to volume ratio; the gas flowing into the crevice cools by heat transfer to the wall. During the combustion process, when the pressure continues to rise, unburned fuel continues to flow into the crevice volumes. When flame arrives at each crevice it may or may not propagate. During expansion and exhaust processes they come out of the crevices because of reduction in cylinder pressure and goes out through exhaust.

Oil layers within an engine can also trap some of the fuel and later release it during expansion. Kaiser et. al. (20) added oil to the engine cylinder and found that the exhaust hydrocarbons increased in proportion to the amount of oil added when the engine was fueled on iso-octane. They verified that the increased emissions were unburned fuel and fuel oxidation species and not unburned oil and oil oxidation species. They also did experiments in which the engine was fueled with propane and found no increase in the exhaust hydrocarbons when oil was added. Since propane is not soluble in the oil, they concluded that the increase in exhaust hydrocarbons observed is caused by fuel having been absorbed into the oil layer during compression, later being released into the cooling burned gas during the expansion stroke. Thus, one can conclude that hydrocarbon emissions from engines will also depend on the amount of oil in the cylinder and the solubility of the fuel in the oil.

With continued use, carbon deposits build up on the valves, cylinder and piston heads of internal combustion engines. The deposits are porous, and the sizes of the pores in the deposits are smaller than the quenching distance, and as a result the flame cannot burn the fuel-air residual gas mixture compressed into the pores. This mixture comes out of the pores during expansion and blow down. Although some of it will burn up when mixed with the hotter gases within the cylinder, cylinder gas temperatures might have dropped to the level where the reactions fail to complete resulting in hydrocarbons being emitted from the engine.

Flame quenching along the surfaces is another mechanism that produces exhaust hydrocarbons. During flame propagation, the flame extinguishes at a small but finite distance away from the cylinder wall.

Flame extinction in the bulk gas before flame front reaches the wall and misfiring i.e. non-development of flame in some cycles is another source of HC emissions in spark ignition engines under idling and part load operating conditions. As cylinder pressure falls during the expansion stroke, the temperature of the unburned mixture ahead of the flame decreases. If the pressure and temperature fall too rapidly, the flame is likely to be extinguished.

## 1.6 *TECHNIQUES FOR IMPROVING ENGINE PERFORMANCE AND EXHAUST EMISSIONS*

The following techniques are available for improving the performance of spark ignition engines.

i) Fuel Intake System Design Modifications
ii) Engine Operating Variables Modifications
iii) Engine Design Modification and
iv) Fuel Modifications and use of an alternative fuel.

However, with all the above modifications, it has not been possible to meet the increasingly strict emission norms. Hence, exhaust gas (EGR recirculation) and exhaust gas after treatment are also used.

### 1.6.1 *Fuel Intake System Design Modifications*

The air-fuel ratio is one of the most important parameters, which significantly influences fuel economy and exhaust gas emissions. Fig. 1.4 (21) shows the variations of fuel consumption and exhaust emissions of CO, HC and $NO_x$ with change in air fuel ratio. The spark ignition engine has normally been operated close to stoichiometric, or slightly fuel rich to ensure smooth and reliable operation.

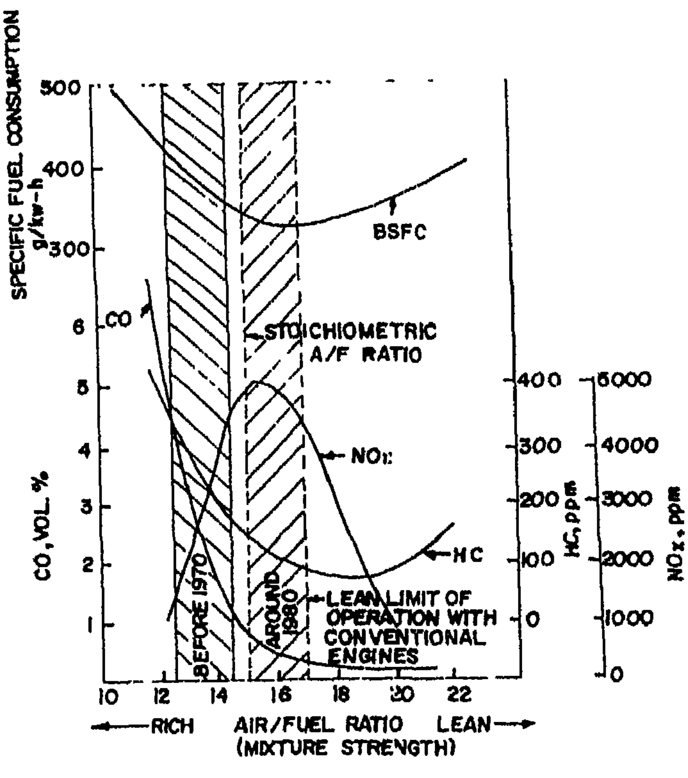

**Fig 1.4-Relation between A/F Ratio and Exhaust emissions (21)**

As it can be seen in Fig. 1.4 that leaner mixtures give lower emissions until the combustion quality becomes poor (and eventually misfire occurs) when HC emissions rise sharply and engine operation becomes erratic. The shape of these curves indicates the complexities of emission control. In cold engines, when. fuel vaporization is slow, the fuel flow is increased to provide an easily combustible fuel rich mixture in the cylinder. Thus until, the engine warms up and this enrichment is removed, CO and HC emissions are usually high. At part load conditions, lean mixtures could be used which would produce lower HC and CO emissions. Use of recycled gas during combustion lowers the $NO_x$ level but also deteriorates combustion quality.

In view of the influence of mixture strength on exhaust emissions, fuel intake system must be optimized for the whole range of engine operation. The introduction system must ensure a narrow tolerance band of air-fuel ratio over the whole range of engine operation. Particular attention is required in the following areas: -

a) Efficient atomization of fuel to ensure proper mixing with air.
b) Reduction of richness during deceleration and idling and avoidance of variation within a range of idling speed.
c) Proper mixture strength during transients such as progression and sudden acceleration.
d) Improved cold start system.

Initially, the carburetors were modified by compensating the main metering system to provide essentially constant lean or stoichiometric mixtures over to the 20 to 80 percent air flow range. Later the carburettors were improved to run on lean mixture and jet-manufacturing tolerances were reduced. Mixture distribution was improved by dual induction system and two/three barrel carburetors.

Recently, the carburettors are being replaced by multi-port fuel injectors (MPFI). The fuel is injected into the intake port of each engine cylinder. There are both mechanical and electronically controlled injection systems (EFI). However, electronically controlled systems are more commonly used due to their wider flexibility to control the fuel flow rate more precisely.

Due to very strict emission limits, the most recent trend is to use feedback controlled systems for which a three way catalytic converter system (described in a later section) is required. This in turn requires the engine to be operated quite close to the stoichiometric air-fuel ratio. The operating air-fuel ratio is maintained close to the stoichiometric through the use of a sensor in the exhaust system, which provides a voltage signal depending on the oxygen concentrations in the exhaust gas stream. This signal forms the input to a feed back system, which controls the fuel feed to the intake. The sensor called an "oxygen Sensor" or "Lambda Sensor" and it is used in the feed back controlled EFI engine.

### *1.6.2* ***Engine Operating Variable Modifications***

Apart from fuel intake system design modifications, other techniques like retarded spark timing, idle mixture adjustment, faster idle speed and leaner mixture operation can also be used to achieve lower exhaust emissions.

Spark timing significantly affects $NO_x$ emission levels. Retarding the timing decreases the peak cylinder pressure (because more of the fuel burns after top dead centre). Lower peak cylinder pressures result in lower peak burned gas temperatures, and hence lower $NO_x$ formation rates. Fig.1.5 (11) shows typical $NO_x$ emission data for a spark ignition engine as a function of spark timing.

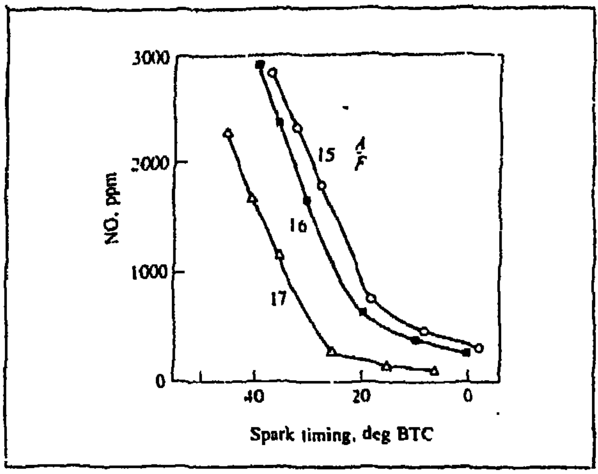

**Fig 1.5- NOx Emissions as a function of Spark Timing (11)**

$NO_x$ emission levels steadily decrease as spark timing is retarded from MBT timing and moved closer to top dead centre (TDC). Retarded timing also helps in reducing hydrocarbon emissions by increasing the fraction oxidized during expansion and exhaust due to the higher burned gas temperatures that result.

### *1.6.3 Engine Design Modifications*

Engine design modifications such as reduced compression ratio, improved combustion chamber design, lean burn engines and stratified charge engines help greatly in reducing exhaust emissions. A lower compression ratio reduces surface to volume ratio of the combustion chamber and hence the volume of gases in the quenched layer. Use of a lower compression ratio also results in a higher exhaust gas temperatures, which provides a better environment for the oxidation of unburned hydrocarbons in the exhaust system.

A combustion chamber design where the fuel burning process takes place faster, i.e. occupies a shorter crank angle interval at a given engine speed, produces a more robust and repeatable combustion pattern that provides emission control and efficiency gains simultaneously. The major combustion chamber design objectives, which relate to engine performance and emission, are (11)

- A fast combustion process, with low cycle-by-cycle variability, over the entire engine operating range.
- A high volumetric efficiency at wide-open throttle.
- Minimum heat loss to the combustion chamber walls.
- A low fuel octane requirement.

Many methods for producing a "Fast Burn" characteristics have been proposed. These include ways of making the combustion chamber shape more compact, moving the spark plug to a more central location within the chamber, using two plugs, and an increasing in-cylinder gas motion by creating swirl during the induction process or during the latter stages of compression.

The spark ignited engines conforming to a new concept known as "Stratified Charge Engine" have also been developed (22-25). In the cylinder of these engines, a rich mixture is provided in the vicinity of the spark plug and a very lean mixture in the remainder of the combustion chamber. Combustion of such a mixture provides very low levels of CO, HC and $NO_x$.

The design modifications in intake manifold to improve mixture preparation, distribution among different engine cylinders and use of high turbulence combustion chamber has made it possible to run the engines much leaner than before (26-27). ***Lean mixture operation results in substantial reductions in CO and HC emissions (28). In addition, there is a definite reduction in the brake specific fuel consumption as the air-fuel ratio is made leaner.*** In order to accomplish significant reductions in the nitric oxide emissions, however, it is necessary to operate the engine on ultra lean air-fuel ratios (beyond 18:1). There is a limit beyond which further leaning causes increase in the hydrocarbon emissions and degradation of the driveability. This limit is normally called the lean misfire limit. However in cylinder gasoline fuel injection system based engines do operate at an overall air-fuel ratio of 40:1. In fact some of these engines do claim a better fuel economy than that of diesel engines (29-31).

### *1.6.4* ***Fuel Modifications and Use of Alternative Fuels***

Lean misfire limit may be extended by charge stratification, high-energy ignition, introduction of additional or alternative fuel. For fuel modifications in spark ignition engines two programmes currently being tried, namely oxygenated fuel programme and reformulated gasoline programme in United States, which resulted in mandated changes in the composition of gasoline. The oxygenated fuels programme is a winter programme used to reduce CO and HC emissions. Presence of oxygenates such as ethanol, methanol, MTBE and ETBE help in reducing CO emissions from older vehicles powered by spark ignited engines. They are also excellent octane improvers. Use of alternative fuels like Alcohols, CNG, LPG, hydrogen and electricity have been found capable of reducing exhaust emission considerably.

## 1.7 ***NEED FOR ALTERNATIVE FUELS***

The rapid depletion of petroleum fuels and their ever-increasing cost have led to an intensive search for alternative fuels in internal combustion (IC) engines. The search for such an alternative fuels have picked up considerable momentum during mid 70's due to the short supply of petroleum crude oil during that period. This search has led to recognize several alternative fuels, which can be successfully used in Internal Combustion engines. Several of these alternate fuels that look promising are inherently more suited to spark ignition (SI) engines rather than compression ignition (CI) engines due to their poor cetane numbers (CN) and good octane numbers (32). Further during this period of time, (i.e. from mid 70's) there has been a serious concern over growing air pollution especially in urban areas caused by vehicles powered by gasoline and diesel fuels. To reduce vehicular air pollution, exhaust emissions from these engines have been controlled by several approaches to meet the emission norms set from time to time (33). Out of several approaches, fuel modifications and clean alternative fuels are found to be excellent approaches to meet more stringent emission norms (34).

The ideal alternate fuel for SI engines should have the following properties: it should be liquid at ambient temperatures and pressures so that it can be stored in simple tanks. However, alternate fuels are being tried even if the same in question is a gas at atmospheric pressure and temperature. If it is a gas at normal ambient temperature, then it can be stored either as a compressed gas in high pressure or as a liquefied gas under high pressure. However, gaseous alternate fuel will have the advantage of better cold start and improved mixture formation with air. Liquid alternate fuels should have low vapour pressure to avoid vapour lock but some light components are also desirable to vaporize easily for better cold start performance. It should have good energy density i.e. good calorific value per unit mass and higher density. It should offer an ability to operate at low levels of harmful exhaust emissions. Last but not the least it should also have good octane numbers (ON) to enable high compression ratio (CR) to be used for achieving higher power and better fuel economy.

Important alternative fuels which are being used worldwide in SI engines are alcohols like methanol ($CH_3OH$) and ethanol ($C_2H_5OH$), gaseous fuels like compressed natural gas (CNG) and liquefied petroleum gas (LPG), hydrogen and ether. ***However, the choice of an alternative fuel in a particular country will be governed by the availability of resources and economics apart from how efficiently a particular alternative fuel can be used in engines.*** In other words, it can be said that success of an alternative fuel in a particular country will depend not only on the availability of raw material and economics but also will depend on the efficiency with which the alternative fuel can be used. The suitability of various alternate fuels, that could be tried as SI engine fuel, are discussed in the next chapter.

# Chapter 2

# ALTERNATIVE FUELS USED IN SI ENGINES

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## *2.1 INTRODUCTION*

Important alternative fuels which are being used worldwide in spark ignition engines are methane or natural gas commonly known as compressed natural gas (CNG), because the gas is kept under high pressure), propane or liquid petroleum gas, hydrogen gas, alcohols like methanol and ethanol, ethers like methyl tertiary butyl ether (MTBE) and ethyl tertiary butyl ether (ETBE). These alternative fuels are of interest since they can be refined from renewable feedstock, and their emissions levels can be much lower than those of gasoline fueled engines (Dhaliwal et. al. (35)). If there are availability problems with crude oil, due to worldwide geo-political problems, these alternative fuels can also be used as replacements. As of the year 2002, the most commonly used alternative fuel for spark ignition engines is propane, followed by natural gas, ethanol and methanol.

The properties of various alternate fuels, that could be tried as SI engine fuel, are shown in Table 2.1 (36-40). As it can be seen from the Table- 2.1, there is a range of energy densities (on fuel mass basis) for different alternative fuels but the energy densities on stoichiometric air mass basis are comparable. Gasoline has the greatest energy density by volume. Alternative fuels have higher-octane levels than gasoline. So engines fueled with alternative fuels can operate at higher compression ratios and thus likely to give higher thermal efficiency.

Table 2.1 Thermodynamic Properties of Gasoline and Alternative Fuels

| ***Properties/Fuel*** | ***Gasoline*** | ***Hydrogen*** | ***Propane*** | ***Natural Gas*** | ***$CH_3OH$*** (Methanol) | ***$C_2H_5OH$*** (Ethanol) |
|---|---|---|---|---|---|---|
| Molecular Weight, | 110 | 2.015 | 44.10 | 18.7 | 32.4 | 46.07 |
| Vapour Pressure, kPa @38 ° C | 62-90 | - | - | - | 32 | 17 |
| Boiling Point, °C, @ 1 bar | 30-225 | (-)253 | - | (-)160 | 65 | 78 |
| Lower Heating value (mass), MJ/Kg $_{fuel}$ | 44.5 | 120 | 46.4 | 50.0 | 19.9 | 26.8 |
| Lower Heating value (volume) MJ/l $_{fuel}$ | 32.9 | - | - | 8.1 | 15.7 | 21.1 |
| Lower Heating Value (Stoichiometric), MJ/Kg $_{air}$ | 2.96 | 3.52 | 2.98 | 2.92 | 3.09 | 3.00 |
| Octane Number (Research) | 90-98 | 106 | 112 | 120 | 112 | 111 |
| Octane Number (Motor) | 80-90 | 120 | 97 | 120 | 91 | 92 |
| Stoichiometric A/F ratio (mass) | 15.04 | 34.13 | 15.58 | 17.2 | 6.43 | 8.94 |
| Flammability limit in air, vol.% | 1.4-7.6 | 5-75 | 4.3-19 | 5.3-15 | 5.5-26 | 3.5-26 |
| Adiabatic Flame Temp; K (at stoich. Ratio) | 2266 | 2383 | 2268 | 2227 | 2151 | 2197 |
| Stiochiometric $CO_2$ emission, gm $CO_2$ / MJ fuel | 71.9 | 0.0 | 64.5 | 54.9 | 69 | 71.2 |
| Auto ignition temp., K | 743 | - | 813 | 853 | 790 | 780 |

### *2.1.1* ***Hydrogen***

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

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

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

The combustion characteristics of hydrogen are very different from gasoline combustion characteristics (39, 40) as the laminar flame speed of a hydrogen air mixture is about 3 m/s, which is about 10 times that of methane and gasoline, and the adiabatic flame temperature is about $100^\circ$ C higher than gasoline and methane. Since it has a wide flammability limit (5 to 75%), pre-ignition and backfiring can be a problem. It was found through a recent study (41) conducted on mechanism of backfire that the abnormal electrical discharge in the intake stroke is one of the causes of backfire. It was also found that the abnormal electrical discharge was caused by high energy retained in the ignition system due to low ion current flow resulting from lower ion concentration in hydrogen flame relative to gasoline flame.

### *2.1.2* ***Propane (LPG)***

Propane ($C_3H_8$) is a saturated paraffinic hydrocarbon. When blended with butane ($C_4H_{10}$) and ethane ($C_2H_6$) it is also designated as liquefied petroleum gas (LPG). LPG is obtained from the lighter hydrocarbon fractions produced during crude oil refining or from heavier components of wellhead natural gas.

LPG has been used a vehicular fuel since 1930s. In 1990 there were about 4 million LPG vehicles operating worldwide with majority of them in Netherlands, followed by Italy, the United States and Canada. There are number of original equipment manufacturers (OEM) that currently sell propane fuelled vehicles in Netherlands, Italy, US and Canada. Thus OEMs sell primarily light and medium duty fleet vehicles such as pick up trucks and vans. Conversion kits are also available to convert gasoline engine to a dual fuel engine.

In vehicle propane is stored as a compressed liquid typically from 0.9 - 1.4 MPa. Its evaporative emission is essentially zero, since it is used in a sealed system. A pressure regulator controls the supply of propane to the engine, and converts the liquid propane to a gas through throttling process. A typical LPG retrofit kit for SI engines is shown in Fig. 2.1. Propane has an octane number of 112 (RON), so vehicular application of propane will generally raise the compression ratio. As shown in Table 2.1 the $CO_2$ emissions on an equivalent energy basis are about 90% that of gasoline. The volumetric efficiency and the power of LPG vehicle will be reduced due to the displacement of about 5% to 10% fuel by air of the intake system as compared to gasoline. Propane requires about 5 degrees more spark advance at lower engine speeds due to its relatively low flame speed as compared to gasoline.

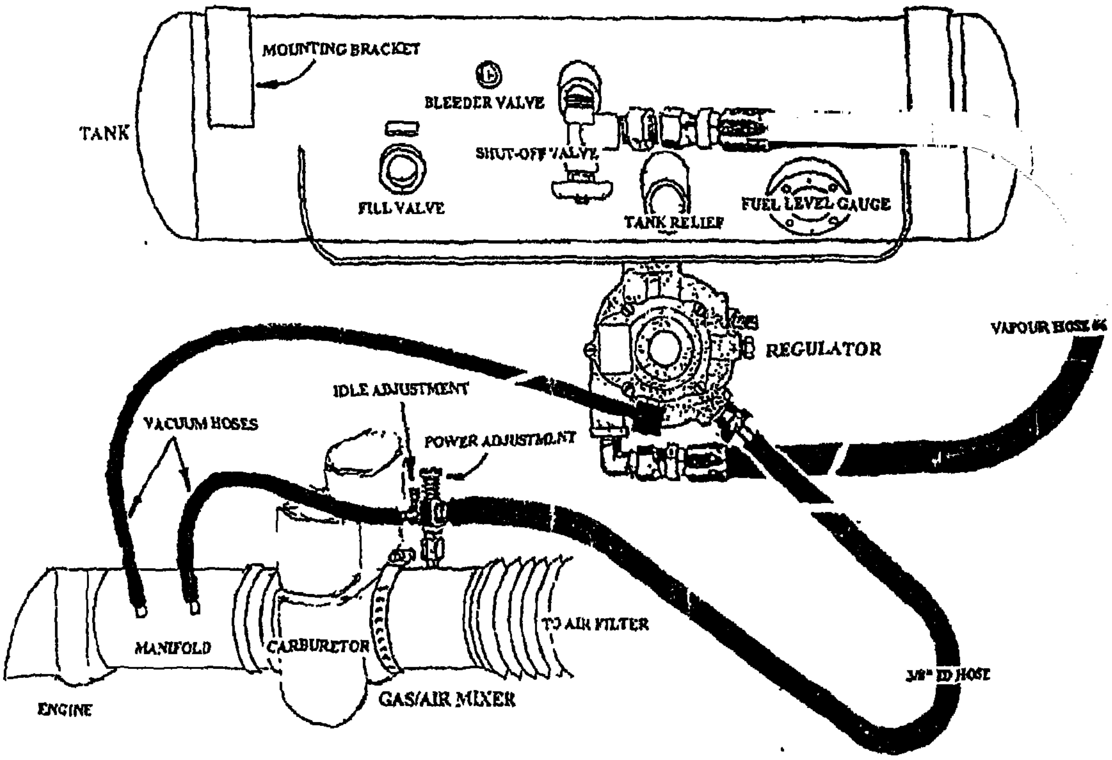

**Fig 2.1- A typical Propane retrofit kit**

### 2.1.3 *Natural Gas*

Natural gas has been used for many years in stationary engines for gas compression and power generation. Also natural gas fueled vehicles (NGV) have been in use since 1950 and conversion kits are available for existing vehicles powered by spark ignited gasoline engines. Recent research and development work has included development of dual fuel vehicles that can operate with both natural gas and gasoline. One advantage of a dual fuel operation is that the operating range of a vehicle is extended in comparison with a dedicated natural gas vehicle. Currently, original equipment manufacturers are selling production natural gas fueled vehicles, primarily to fleet owners. ***Natural gas vehicles were the first alternative fuel vehicles to be used in Delhi to meet the emission norms.***

Natural gas is stored in a compressed (CNG) state at room temperature and also in a liquid (LNG) form at – 160°C. Natural gas has an octane number (RON) of about 120, which permits the natural gas engines to operate at a compression ratio of 11:1. Natural gas is pressurized to 20 MPa in vehicular storage tanks, so that it has about one-third of the volumetric energy density of gasoline. The storage pressure is about 20 times that of propane. Like propane, natural gas is delivered to the engine through a pressure regulator, either through a mixing valve located in the intake manifold, port fuel injection at about 750 kPa, or direct injection into the cylinder. With intake manifold, mixing or port fuel injection, the engine's volumetric efficiency and power are reduced due to the displacement of about 10% of the intake air by the natural gas. Natural gas does not require mixture enrichment for cold starting thereby reducing the cold start HC and CO emissions (42,43).

The combustion of methane is different from that of liquid hydrocarbon combustion since only carbon hydrogen bonds are involved, and no carbon-carbon bonds, so the combustion process is more likely to be more complete, producing less non-methane hydrocarbons. Optimal thermal efficiency occurs at lean conditions at equivalence ratios of 1.3 to 1.5 (14). The total hydrocarbon emission levels can be higher than gasoline engines due to unburned methane. The combustion process of methane can produce more complex molecules, such as formaldehyde, a pollutant. The particulate emissions of natural gas are very low relative to any other liquid fuels.

### *2.1.4* ***Methanol***

Methanol ($CH_3OH$) is an alcohol fuel formed from natural gas, coal, or biomass feed stock (44-46). Methanol is also called wood alcohol. It is a liquid at ambient conditions. Its chemical structure is a hydrocarbon molecule with a single hydroxyl (OH) radical. The hydroxyl radical increases the polarity of the hydrocarbon, so that methanol is miscible in water, and has a relatively low vapor pressure, since oxygen is part of the chemical structure, less air is required for complete combustion. Methanol is toxic, and ingestion can cause blindness and death. Methanol has been used as a vehicular fuel since the early 1900s and is also used as a fuel for diesel engines and fuel cells.

Pure methanol is labeled as M-100 and a mixture of 85% methanol and 15% gasoline is labeled M-85. M-85 has an octane rating of 102. Adding gasoline to methanol provides more volatile components that can vaporize more easily at low temperatures. Methanol has been adopted as a racing vehicle fuel, both for performance and safety reasons. Since methanol mixes with water, a methanol fire can be extinguished with water, which is not the case of gasoline. The octane rating of methanol which is 112 RON allows the use of an increased compression ratio. The relatively high enthalpy of evaporation (1215 KJ/kg) of methanol relative to gasoline (310 KJ/kg) produces greater intake air-cooling leading to an increased volumetric efficiency. The energy density by volume of methanol is about half that of gasoline. However, because of its oxygen content, it has a higher stoichiometric energy density (3.09 MJ/kg air) relative to gasoline (2.96 KJ/kg air).

The low vapour pressure methanol causes cold starting problems. Satisfactory cold starting with M-85 requires a rich mixture so that enough volatiles are present to form a combustible mixture. Methanol is corrosive especially to rubber and plastic (47,48), so alcohol tolerant components, such as stainless steel and Teflon etc. are required for storage and transport.

### *2.1.5* ***Ethanol***

Ethanol ($C_2H_5OH$) is an alcohol fuel formed from the fermentation of sugar and grain stocks, (primarily sugar cane and corn) which are renewable energy sources. Its properties and combustion characteristics are very similar to those of methanol. Ethanol is also called "grain" alcohol. It is a liquid at ambient conditions, and nontoxic at low concentrations.

Two known blends of ethanol and gasoline which were used in spark ignition engines are gasohol (E-10) in which about 10% ethanol is mixed with 90 percent gasoline by volume and E-90, which is a blend of 90% ethanol, and 10% gasoline. In Brazil, about half of the vehicles use an ethanol based fuel "alcool", primarily E-93 is produced from sugar cane. The energy density by volume of ethanol is relatively high for an alternative fuel, about two-thirds that of gasoline. The octane rating of ethanol 111 (RON) allows the use of an increased compression ratio. The cetane number of ethanol is low, (about 8) but like methanol, it can be used in compression ignition engines with diesel fuel pilot ignition. As shown in Table 2.1, the $CO_2$ emissions on an equivalent energy basis are about 99% that of gasoline. From the above discussions, it is clear that there are number of alternative fuels which cad be used in spark ignition engines to tackle the availability problem of crude oil and to reduce harmful emissions. ***However, alternative fuels like ethanol and compressed natural gas can be attractive in India because of country's wide agricultural base and large coal reserves.*** Ethanol can be produced from the fermentation of sugar and grain stocks, primarily sugar cane and corn (49) that are also renewable energy sources. Apart from natural gas reserves methane, which is one of its principal constituent, can also be produced from coal and biomass conversion (50) thus making it much more available energy base. India occupies one of the top place in the world in respect of sugarcane production (51). About 35 lakh hectare of land is annually under cane cultivation. Also natural gas reserve of India is also quite encouraging. Table- 2.2 shows the growth of natural gas reserve in India (3).

***Table- 2.2 Growth of Natural Gas Reserves in India (3)***

| | **1975** | **1985** | **1995** | **2000** | **2001** |
|---|---|---|---|---|---|
| **ONGC** | 30.4 | 408.8 | 555.2 | 517.8 | 508.0 |
| **OIL** | 57.3 | 69.8 | 86.8 | 96.0 | 96.2 |
| **JVs&PVT** | 0.0 | 0.0 | 45.8 | 112.4 | 124.4 |
| **Total** | 87.7 | 478.6 | 687.8 | 726.2 | 728.6 |

- Figures in Billion Cubic Meters.
- Gas reserves of ONGC and OIL are as on Ist January up to 1985 and as on Ist April thereafter.
- Gas reserves of JVs/Pvt. Parties are as on Ist April.
- All above are balance of recoverable reserves.

From the above facts it is clear that both ethanol and natural gas both can be made available, so they can be more suitable alternative fuels for India. Also harmful emission levels from these fuels i.e. ethanol and natural gas in a well designed engine will be much lower than those of gasoline fueled engines especially in lean mixture zone (52-58). To reduce harmful emissions further, it will be essential to modify the fuel, engine design and operating systems apart from introducing exhaust gas recirculation (EGR) and exhaust gas after treatment concepts.

Hence, in the present work it is proposed to investigate fuel economy and emissions characteristics of above mentioned fuels. In the next few paragraphs fuel related properties of ethanol and natural gas and their performance and emission characteristics in spark ignition engines are discussed.

## 2.2 *FUELS RELATED PROPERTIES OF ETHANOL AND NATURAL GAS -- COMPARISON WITH GASOLINE*

The difference in emission characteristics for ethanol and natural gas relative to gasoline can be attributed to difference in physical and chemical properties and most importantly to difference in combustion behaviour arising from later. In view of this, it is instructive to note some of the ways in which ethanol and natural gas differ from gasoline.

### *2.2.1 Ethanol- Fuel Related Properties*

The most obvious difference is that, while gasoline is a complex and widely variable mixture of true hydrocarbons, i.e., molecules comprised of only carbon and hydrogen atoms, ethanol is a single specie that might be termed as, "partially oxidized hydrocarbon". Thus in a sense ethanol might be thought of as first a product of oxidation in a sequence of oxidation reaction steps in which a given hydrocarbon (RH) is first converted to ethanol (ROH) then sequentially aldehyde (RCHO), ketone, or carboxylic acid and then carbon dioxide via carbon monoxide (58). The presence of oxygen atom in molecules of ethanol coupled with the lower molecular weight (and more importantly lower carbon number) causes substantially different properties for ethanol as compared to gasoline. Some of the properties, which are important from spark ignition engine's performance and emissions point of view, are given in Table-2.3. As it can be seen from Table-2.3 ethanol is slightly denser

***Table-2.3 Comparison of Properties of Gasoline and Ethanol***

| ***Properties/Fuel*** | ***Gasoline*** | ***$C_2H_5OH$*** (Ethanol) |
|---|---|---|
| Density @ 15 $^0$C | .720 | 780 |
| Vapour Pressure, kPa @38 $^0$ C | 62-90 | 17 |
| Boiling Point, $^0$C, @ 1 bar | 30-225 | 78 |
| Enthalpy of Vapourization ($h_{fg}$), KJ/Kg, @ 298 K | 310 | 850 |
| Lower Heating value (mass), MJ/Kg $_{fuel}$ | 44.5 | 26.8 |
| Lower Heating value (volume) MJ/l $_{fuel}$ | 32.9 | 21.1 |
| Lower Heating Value (Stoichiometric), MJ/Kg $_{air}$ | 2.96 | 3.00 |
| Octane Number (Research) | 90-98 | 111 |
| Octane Number (Motor) | 80-90 | 92 |
| Stoichiometric A/F ratio (mass) | 15.04 | 8.94 |
| Flammability limit in air, vol.% | 0.6-8 | 3.5-26 |
| Adiabatic Flame Temp; K | 2266 | 2197 |
| Stiochiometric $CO_2$ emission, gm $CO_2$ / MJ fuel | 71.9 | 71.2 |
| Auto ignition temp., K | 743 | 780 |
| Burning velocity, cm/sec | 45 | 48 |

than gasoline and it has lower Reid vapour pressure (RVP). It has greater affinity for water and has much higher heat of vaporization.

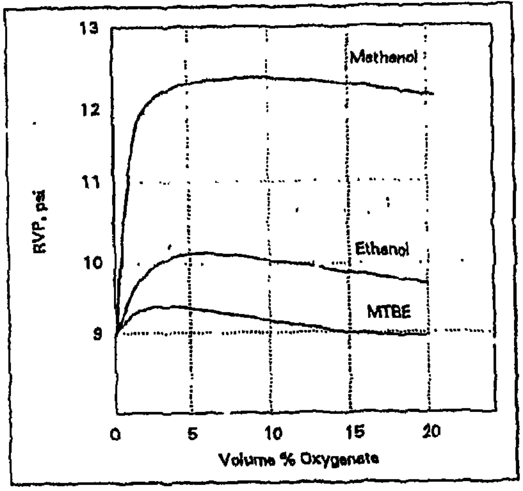

**Fig 2.2- Change in RVP as a result of adding alcohols (60)**

### *2.2.1.1 Vapour Pressure*

Most gasoline by comparison has wide mixture of molecular weight compounds and thus boiling points. The lower molecular weight compounds provide sufficiently high vapour pressures to permit ready ignition under ambient conditions. Ethanol's vapour pressure is considerably lower than these, which causes considerable difficulty in cold start. For better cold starting adding gasoline to ethanol is considered for using net ethanol. However, when ethanol is mixed with gasoline the blend vapour pressure rise as indicated in Fig.2.2 (60). As it can be seen in this figure, RVP increases when ethanol is added to gasoline. An increase in vapour pressure is important because it can cause elevated evaporative emissions (diurnal, hot soak and running loss).

### *2.2.1.2 Flammability Limit*

The wide flammability limits of ethanol present the possibility of a leaner combustion in engine but also create a potential safety problem in storage tanks.

### 2.2.1.3 *Enthalpy of Vaporization*

The hydroxyl group in ethanol makes it polar with strong hydrogen bonding between molecules. This gives ethanol an enthalpy of vaporization of 850 KJ/kg at 298K which 2.77 times that of gasoline (310 KJ/kg at 298 K). Hence more heat will be required to evaporate ethanol as compared to gasoline. Also stoichiometric air fuel ratio of ethanol is about 9:1 as compared to 15:1 for gasoline. Hence for the same output 1.4 times of fuel will be required when the engine runs on ethanol. Accordingly it will require 3.88 times as much heat to evaporate sufficient ethanol to deliver same heating value as gasoline in liquid fuel. Providing sufficient heat to vaporize ethanol has been one of the largest problems when ethanol is used in spark ignition engines, particularly if they are designed for use with hydrocarbon fuels. Lower volatility of ethanol raises the question about whether or not ethanol fuel can meet low emission standards. Low volatility may lead to difficulty in engine starting with high hydrocarbon emissions. Effective recovery of exhaust heat could possibly turn this to an advantage by improving overall thermal efficiency.

### 2.2.1.4 *Knock Resistance*

Reported value of RON of ethanol is about 111 as compared to gasoline's 90 to 98. These high octane ratings permit the use of higher compression ratios with ethanol than when using gasoline, thus permitting greater power and efficiency. Brinkman (61) studied ethanol in a single cylinder engine and found no objectionable knock up to a compression ratio of 9:1 when spark timing was set at MBT (Minimum Advance for Best Torque). The tendency of conventional fuels to knock has been reduced considerably in experiments in which the combustion chamber is altered to induce swirl into the fuel charges. These effects and the corresponding benefits of higher compression ratios may apply to ethanol also.

#### 2.2.1.5 *Energy Density*

Because of chemical structure ethanol, molecule contains fixed oxygen. This results in lower heating value of ethanol as compared to gasoline. To generate equal amount of energy 44.5÷26.8=1.66 times or 66% more mass of ethanol will be required as compared to gasoline. Also energy density on volume basis is lower in case of ethanol as compared to gasoline, so if ethanol is used in the existing gasoline vehicle, modifications to the fuel tank may be required to carry equal amount of energy. However, energy density of ethanol on a stoichiometric mass is slightly more which may result in slightly better thermal efficiency.

#### 2.2.1.6 *Corrosion and Swelling*

The reactive hydroxyl group in ethanol is noticeably different from petroleum hydrocarbons in its corrosion of different materials. This is compounded when water and salts are dissolved in the fuel. Higher corrosion rate with ethanol is noticed with zinc, lead and magnesium as compared to corrosion with hydrocarbon fuels. These materials need to be coated or lined with resistant coatings. Ethanol is also a strong solvent and noticeably swells or softens many of plastics or rubbers (62) that are often used as gaskets or floats in conventional gasoline engines. This is the reason that fuel filter problems were encountered when gasohol (10% ethanol + 90% gasoline) was first used in older vehicles. However ethanol is less corrosive than methanol. Table – 2.4 gives comparison of swell of various elastomers in ethanol and gasoline (62).

**Table 2.4 Elastomer Swell in Ethanol and Gasoline (62)**

| *Elastomer* | *Gasoline* | *Ethanol* | *E-10* |
|---|---|---|---|
| Fluorocarbon(FKM) | 0 | 2 | 3 |
| Polyester urethane | 11 | 19 | 37 |
| Fluorosillicone (FMQ) | 14 | 6 | 18 |
| Butadiene-acrylonitrile (NBR) | 43 | 8 | 51 |
| Polyacrylate (ACM) | 44 | 101 | 136 |
| Chlorosulfonated Polyethylene(CSM) | 49 | 1 | 56 |
| Ethylene-propylene-diene terpolymer (EPDM) | 137 | 13 | 124 |
| Natural Rubber (NR) | 169 | 2 | 176 |

### 2.2.2 *Effect on Distillation, Cold Starting and Vapour Lock of Ethanol-Gasoline Blends*

Adding ethanol to gasoline depresses the boiling temperature of individual hydrocarbons. Ethanol causes significant reduction in the front end distillation temperatures, thus affecting primarily the first 50 percent evaporated (64). Lower-molecular-weight alcohols have the greatest effect on boiling point depression. Higher molecular weight alcohols, such TBA, propanol, butanol and pentanol, exert smaller changes on the distillation characteristics.

### 2.2.3 *Review of Power, Fuel Consumptions and Emissions—Ethanol and Ethanol-Gasoline Blends*

One of the way of using ethanol in spark ignition engine is to blend them with gasoline and use them in cars which are either unmodified or have only small alteration made such as E-15, which signifies 85% gasoline and 15% ethanol. The most common commercially sold blend is gasohol, which is defined as a fuel containing 10% anhydrous (200 proof) ethyl alcohol and 90% unleaded gasoline. This fuel can be used in most spark ignition engines designed to burn lead-free fuel without any major engine modifications. However, operation of the engine may be slightly unstable, and thus the driveability of the vehicle may be poorer. This is due to a further leaning of the fuel air ratio in engines designed to be lean burning to meet emission standards (63). It is also possible to use ethanol in pure form but it cannot be used without incorporating major modifications to the intake system. This is because of the difference in the stoichiometric fuel-air ratios of gasoline and ethanol. Also to generate equal amount of energy 1.66 times or 66% more mass of ethanol will be required as compared to gasoline. So if pure ethanol is used in the existing gasoline vehicles the intake system has to be modified. In the following paragraphs the engine performance and the exhaust emissions of with ethanol and ethanol-gasoline blends are discussed.

Rich mixtures are always needed to start cold engines because enough fuel must vaporize to form a combustible air fuel mixture. Thus, these studies demonstrate only relative difficulties in starting characteristics and not whether an engine will start when fuelled with a blend. The relative cold starting difficulties largely depend on the fuel delivery system. The fuel delivery system must be able to provide a rich enough mixture to allow starting with an ethanol/gasoline blends.

#### *2.2.3.1* **Power and Fuel Consumptions**

As discussed earlier ethanol has a lower calorific value and therefore lower stoichiometric air-fuel is needed. Ethanol also has a much higher latent heat and these two factors together will have more cooling effect of the fuel on the intake than gasoline. Consequently, charge temperature at full load is reduced, resulting in increased volumetric efficiency and more torque and power from the engine. This has been observed experimentally by Overington and Thring (65). The effect on fuel consumption of ethanol is somewhat clouded by the fact the effects are different depending on the method used to measure fuel economy. As ethanol has lower calorific value than gasoline, it is necessary to supply about 1.6 times more ethanol for the same amount of stoichiometric air on mass basis. In practice, however, the brake thermal efficiency is slightly improved by about 3 percent at constant compression ratio and constant air-fuel ratio (61). Thermal efficiency can be improved further by increasing the compression ratio as octane rating of ethanol is higher than that of gasoline.

#### *2.2.3.2* **Evaporative Hydrocarbon Emissions**

The use of gasoline in vehicles with fuel systems incorporating evaporative emissions controls has been shown to affect the quantity of fuel vapors that escape from the fuel system. These vapors, defined as evaporative emissions, are generally higher when ethyl alcohol is mixed with gasoline compared to gasoline alone. Test results by Austin and Rubinstein (66) reported increases ranging from 42 to 65 percent depending on the type of system used. Others have also reported increased emissions. The actual change in evaporative emissions may also vary with the initial distillation characteristics of the gasoline.

Some gasohol fuels have been blended to match the volatility of normal gasoline to help minimize the increase in evaporative emissions. The adjusted volatility takes into consideration the fact that ethyl alcohol has a boiling point of 78.3°C. This temperature is near the 20% evaporation point of a typical gasoline. The evaporative emission test results on a 1978 passenger car showed that the levels were still higher with the adjusted blend than with gasoline alone (53). One theory offered by Furey and King (53) was that activated charcoal normally used in canister for trapping vapors absorbs alcohol vapor more strongly than hydrocarbons. This inhibits purging of the canister.

### *2.2.3.3* **CO Emissions**

Changes in exhaust emissions of a typical car are generally small when changing from gasoline to gasohol. The effect on carbon monoxide (CO) is partially dependent on the air-fuel equivalence ratio. Furey and King (53) reported a 27% reduction in engine (before catalyst) CO emissions on a 1978 passenger car. This is primarily the result of a lower equivalence ratio when using gasohol instead of gasoline.

The effect of ethanol on production of carbon monoxide was very clear in all of the articles reviewed. When ethanol was used in the existing gasoline engines CO was dramatically less (53,54, 67,68) as compared to pure gasoline fuel. This was due to leaning effect related to partially pre-oxidized nature of ethanol relative to gasoline. Apart from mixture ratio, CO emissions are also dependent of CR and fuel. Brinkman (61) had presented data for CO emissions with different equivalence ratios ($\phi$) at different CR for pure gasoline and pure ethanol. At an equivalence ratio of less than about 1.0, CO emissions were low essentially independent of fuel, CR and equivalence ratio. As equivalence ratio is increased above 1.0, CO emissions increased rapidly, but the rates of increase were different for different fuel- CR combinations.

#### *2.2.3.4* ***HC Emissions***

The effect of ethanol on production of exhaust HC emissions was not very clear. This due to complex nature of mechanism of formation of exhaust HCs, also nature of fuel, combustion temperature, exhaust manifold temperature, CR, equivalence ratio and quality of mixture will effect HC formation. The HC emission results reported by the University of Santa Clara (69) showed unburned hydrocarbon (UBF) emissions with ethanol to be twice than those with gasoline. The reasons for higher level of hydrocarbon emissions could be due to more flame quenching with ethanol as compared to gasoline. The boiling temperature of ethanol is higher, so ethanol may be evaporated till the end of combustion stroke. The result will be the cooler flame as compared to gasoline and the flame may get quenched much before it reaches the combustion chamber walls. Also, since these results were obtained at lower loads, there could be more misfired cycles with ethanol as compared to gasoline. Because of these reasons with ethanol they might have obtained twice the hydrocarbon emissions as compared to gasoline. Lowary and Devoto (70) found reduced UBF emissions with ethanol. But they used unheated HC analyzer, so much of the unburned fuel was probably lost in the water trap. Bechtod and Pullman (71) and Brinkman (61) found similar UBF with gasoline and ethanol. However, most of the work on ethanol gasoline blends show reduction in both CO and HC emissions.

#### *2.2.3.5* ***$NO_x$ Emissions***

As discussed earlier (Fig. 1.4) $NO_x$ emission is a function of temperature and availability of oxygen. Brinkman (61) found both CR, equivalence ratio and fuel affect $NO_x$ emissions. He found at leaner mixture NOx emissions with ethanol were substantially lower than that of gasoline. Maximum NOx emissions with ethanol at 7.5 CR were about 40% lower than corresponding maximum with gasoline. Others found reductions of 20 percent (69) and 25 percent (70) with ethanol as compared to gasoline. Astine and Rubinstein (66) reported that tests on a number of motor vehicles showed that HC emissions ranged from a 15% decrease to a 6% increase depending on type of emission control system used. In the same tests, CO emissions decreased from 7 to 28 percent and NO emissions increased from 4 to 13 percent.

#### *2.2.3.6* ***Aldehyde Emissions***

Exhaust emissions of aldehydes, though not currently regulated are of interest because they are irritants and contribute to photochemical smog reaction (71). The most prevalent aldehyde specie in ethanol combustion is acetaldehyde and in case of gasoline combustion it is formaldehyde. Brinkman (61) found at same compression ratio ethanol fuel emitted 100-360 percent more acetaldehydes as compared to formaldehyde emissions of gasoline, depending on air-fuel ratio. Acetaldehyde emissions were lowest at air-fuel mixtures near stoichiometric and increased as mixture was richened or leaned. An additional increase in acetaldehyde emissions occurred as compression ratio was increased. Others (69,71) also found increase in acetaldehyde emissions with ethanol as compared to gasoline's formaldehyde emissions.

Aldehydes are unstable at temperature existing in combustion flames (72), so exhaust aldehyde emissions are probably formed during partial oxidation of fuel remaining after flame extinction. Studies (72) of ethanol oxidation product at relatively low temperatures (270 –370$^0$ C) have shown that acetaldehyde is the first oxidation product that can be detected. Thus, formation of acetaldehyde appears to be a primary step in the mechanism of ethanol oxidation. This parallels current theory (73) on methanol oxidation via the formaldehyde intermediate. With hydrocarbon fuels, on the other hand, oxidation is believed (72,73) to occur through two paths, of which only one involves formation of aldehydes. Intuitively, these mechanisms suggest that aldehyde emissions should be greater with ethanol than with gasoline.

## 2.2.4 ***Natural Gas Composition and Properties***

### *2.2.4.1* **NG composition**

The exact composition of natural gas depends on whether the gas is sourced from an oil condensate field i.e. whether it is associated gas or it exists by itself, which is referred to as non-associated gas. Associated gas may contain significant amounts of heavier hydrocarbons (HCs) such as ethane, propane and butane together with lighter liquids such as pentane and hexane. In this category methane percentage can be as low as 50% (42). Non-associated gas contains a much higher percentage of methane. Additionally both these varieties contain varying amount of carbon dioxide, nitrogen and other contaminants. Table-2.5 shows the composition of pipeline quality gas in India (42).

### *2..2.4.2* **NG Characteristics**

Since natural gas is predominantly methane, the engine performance and exhaust emissions of a natural gas engine will primarily depend on properties of methane. The important properties of gasoline and methane are compared in Table-2.6. Some of these characteristics have specific importance with regard to use natural gas in spark ignition engines.

**Table 2.5 Typical Pipeline Quality Gas in India (42)**

| ***Constitute*** | ***Source I***<br>Ex HBJ (Delhi) | ***Source II***<br>Ex HBJ (IPCL) | ***Source III***<br>Ex Bombay |
|---|---|---|---|
| Methane | 84.50 | 88.42 | 82.55 |
| Ethane | 7.70 | 8.79 | 7.67 |
| Propane | 2.40 | 1.59 | 3.85 |
| I – Butane | 0.26 | 0.29 | 0.64 |
| N – Butane | 0.32 | 0.28 | 0.78 |
| I – Pentane | 0.18 | 0.05 | 0.13 |
| N – Pentane | 0.19 | 0.05 | 0.13 |
| Hexane | 0.17 | 0.04 | 0.09 |
| Nitrogen | 0.12 | 0.20 | 0.07 |
| Carbon – dioxide | 4.23 | 0.27 | 0.07 |

### 2.2.4.3 *Octane Number/Methane Number*

The compression ratio of a spark ignition engine is set in such a manner the air-fuel mixture will not auto ignite even when the flame starts propagating. The octane number of a fuel is a measure of the tendency to auto ignite. The higher the octane number, the higher the allowable compression ratio and greater the power and thermal efficiency. Natural gas has much higher octane number as compared to gasoline. For quantifying the knocking tendency of gaseous fuels, the parameter methane number is used. In this scale, pure methane is assigned 100, indicating extreme knock resistance and hydrogen is assigned a methane number of zero. Natural gas being predominantly methane, has a high methane number. On octane scale the value corresponds to approximately 120 RON, there by making natural gas as very highly knock-resistant fuel.

***Table-2.6 Comparison of Properties of Gasoline and Natural Gas***

| | ***Gasoline*** | ***Methane*** |
|---|---|---|
| Molecular Weight, | 110 | 18.7 |
| Vapour Pressure, kPa @38 $^0$ C | 62-90 | - |
| Boiling Point, $^0$C, @ 1 bar | 30-225 | -160 |
| Lower Heating value (mass), MJ/Kg $_{fuel}$ | 44.5 | 50.0 |
| Lower Heating value (volume) MJ/Kg $_{air}$ | 32.9 | 8.1 |
| Lowe Heating Value (Stoichiometric), MJ/Kg $_{air}$ | 2.96 | 2.92 |
| Octane Number (Research) | 90-98 | 120 |
| Octane Number (Motor) | 80-90 | 120 |
| Stoichiometric A/F ratio (mass) | 15.04 | 17.2 |
| Flammability limit in air, vol.% | 0.6-8 | 5.3-15 |
| Adiabatic Flame Temp; K | 2266 | 2227 |
| Stiochiometric $CO_2$ emission, gm $CO_2$ / MJ fuel | 71.9 | 54.9 |
| Auto ignition temp., K | 743 | 853 |
| Burning velocity, cm/sec | 45 | 37-45 |

### *2.2.4.4* **Density**

Lower density of gas means that for a given engine displacement, comparatively, lower amount of mixture can be inducted and burned, leading to power loss.

### 2..2.4.5 *Flame Speed*

The laminar flame speed of natural gas is lower than that of the liquid hydrocarbons (74), which results in slower combustion rate leading to a drop in efficiency of unmodified engines.

### 2.2.4.6 *Ignition Energy*

The methane molecule is stable and compact and hence has high activation energy. The minimum energy required for ignition is therefore higher compared to liquid fuel. This may require high-energy ignition source for combustion.

### 2.2.4.7 *Wobbe Number*

Another property, which is of interest, is the "Wobbe number". The relative presence of higher hydrocarbons and inert gases has a pronounced effect on gas density and the volumetric energy content. To account for this a parameter called 'Wobbe number' (w) is introduced, which is defined as : $W = H/(D)^{0.5}$

Where H = Higher heating value on volumetric basis, and

D = Specific gravity of gas.

Thus, if two gas mixtures have the same Wobbe number, the heat content of the air/fuel mixtures will be same in either case. Wobbe number is important because fuel composition and properties like heating value and molecular weight can affect the maximum power output of an engine (42).

The higher, RON, wider flammability limits, lower flame temperature, and gaseous state of natural gas, relative to gasoline, will work to increase thermal efficiency. (The gaseous state is an advantage because of a gas mixes thoroughly with air at any ambient temperature, even when starting a cold engine). The lower flame speed tends to reduce efficiency, but advancing the spark timing can compensate for this, at the expense of increasing $NO_x$ and HC emissions. It has been suggested that optimized natural gas vehicles should be about 20% more efficient than gasoline vehicles (75).

Natural gas can be used in spark ignition engines under two different modes, the dual fuel mode, in which the engine can run either with gasoline or CNG and dedicated mode, where the engine is modified to run on natural gas only. Each mode is discussed separately below:

### 2.2.5 **Duel Fuel Operation**

In this mode of conversion, natural gas is used in existing spark-ignited with practically no engine modification, except for a change in the intake system. The essential components of conversion, which relates only to the fuel intake system are:

(a) One or more storage cylinders depending upon the trunk space and the desired range. Sometimes the cylinders are mounted in the bed or alongside the chassis frame rails on heavy vehicle installations.

(b) Gas regulators to reduce the cylinder gas pressure down to delivery pressure for the gas carburetor. The reduction in pressure is either in two or in three stages. To avoid the charge chilling effect due to expansion of gas, the gas regulators are often heated by circulating the engine coolant.

(c) A gas/air mixer and metering system, to supply fuel charge to the engine in accordance with engine demand. The mixture units are generally venturi ring-type, specially matched to the engine displacement. Since the engine can be run on either fuel, the following additional components are also incorporated in a dual-fuel CNG engine.

(d) A fuel selector switch to select the fuel mode. It is generally a manual selector device of the rotary type. Electrically operated valves can achieve the changeover from one fuel to another also.

(e) Gasoline fuel lock, which is energized by the fuel selector switch, when the operator selects the gasoline mode. When the fuel lock is energized, gasoline is able to flow from the fuel pump to the carburetor.

(f) Fuel gauge to indicate the amount of CNG left in the cylinders. It essentially uses a pressure transducer, which converts the pressure signal to electrical signal for the display gauge.

(g) Master shut off valve, which cuts off the CNG supply in case of any emergency.

The major tuning problem to be tackled in a dual fuel CNG vehicle concerns the difference in the rate of burning of natural gas and gasoline. Natural gas having a slower laminar flame speed in air (76), burns slowly and needs optimization with regard to ignition timing. Since natural gas engines run leaner, it also becomes necessary to have a good ignition timing control, as spark timing is very sensitive to power characteristics under lean operation. Advancing the timing in natural gas mode results in improved power, and substantial improvements are observed by increasing the static advance (77). This, however, has two disadvantages. Firstly, it advances the timing over the complete cycle and therefore cannot take into account the variation in timing with speed. Secondly, the timing change is not easily reversible and can result in abnormal combustion when the engine is operating on gasoline mode. The electronic dual time principle overcomes the disadvantage of static advance. Commercial electronic dual curve ignition system is now available and form a part of the dual fuel conversion kit.

### *2.2.6* ***Dedicated CNG Engine***

The relative advantages of natural gas over gasoline, viz., its high antiknock property and ability to run lean, can be best exploited in dedicated CNG engines. This, however, require major engine modifications. The intake system requirements are the same as dual fuel engine. The only difference is that the system for gasoline carburetion is not there in this mode. The changes required in the power pack are discussed below.

(a) Compression ratio: To take advantage of high octane number of natural gas, the compression ratio of the engine has to be increased. This can be done by modifying the cylinder head or piston crown or by milling the mating surface of the cylinder head and block. However, there is a limit to which the compression ratio can be increased, because beyond a certain point there is a drop in the performance due to choking of the flame front (78).

(b) Valve timing: As natural gas burns slowly than gasoline, it is advantageous to open the exhaust valve as late as possible in the power stroke. A change in valve timing is therefore desirable (79).

(c) Ignition timing: The lower flame speed of natural gas requires advancing the ignition timing for optimized performance. Moreover CNG engines can be operated with very lean mixtures which result in longer burning time and the net result is the requirement of large amount of ignition advance. The amount of advance needed for best power, however, varies significantly from engine to engine.

(d) Ignition system: Ideally, natural gas will require more energy for ignition than gasoline, since it has got a compact structure and much higher auto ignition temperature. But it has been found in practice, that for similar engine operating conditions, the energy and voltage requirement for natural gas are comparable to that of gasoline (80). One probable reason may be that the ignition energy requirement is also a function of charge homogeneity, which is much better in the case of natural gas. For the same reason, smaller electrode gap can be used for natural gas compared to gasoline.

(e) Valve train: Valve seat wear may be a problem in natural gas engines. Dedicated CNG engines may therefore require alternate wear resistant material for valve train components.

### *2.2.7* ***Engine Performance***

Not all vehicles perform in the same manner when operated on natural gas. Further there is a difference in performance when operated with different conversion kits (81). The following general trend is however reported with regard to some vehicle performance parameters. In dual fuel operation, there is a power loss of the order of 10-15% under CNG operation (75,82-83). The power loss is observed even under optimized ignition timing. The reason for this is that the fuel, being in a gaseous state, displaces more intake air and hence accounts for more space being taken up by the fuel in a fuel air mixture. The power loss may be comparatively lesser at lower engine speed or load, where vaporization and hence combustion of gasoline tend to become less efficient (84).

Throttle response and hence acceleration is a bit slower with CNG operation, due to the lack of accelerator pump action common to liquid fuel carburetors. The higher octane number and gaseous state of natural gas can lead to an increase in thermal efficiency of the natural gas vehicles (NGV's) relative to gasoline ones. But these advantages can only be exploited in dedicated CNG vehicles, for which, an increase in thermal efficiency of the order of 20% is predicted (75). The efficiency of dual-fuel CNG vehicle is comparable to its gasoline counterpart. An improvement can, however, be expected at part-loads under cold running conditions.

Regardless of the ambient temperature, engine starting should be easier with natural gas operation, owing to its gaseous state and extremely low flash point (84). The use of CNG will also eliminate other driveability problems like vapour lock associated with gasoline operation.

Natural gas is a clean burning fuel compared to gasoline and hence engine deposits are reduced under CNG operation. Additionally since no antiknock compound is needed (specially TEL), the incidence of spark plug fouling is also reduced.

### *2.2.8* ***Environmental Impact***

The use of natural gas as an automotive fuel results in reduced concentration of major harmful species in the engine exhaust (85-87).

#### *2.2.8.1* **Carbon Monoxide (CO) Emissions**

Carbon monoxide is the result of incomplete combustion and is a function of overall mixture strength, the efficiency with which the fuel and air as mixed and the length of time available for combustion (88). The CO emission with natural gas is lower because it easily forms a more homogeneous mixture with air and can run leaner than gasoline vehicles (89,90). Since natural gas engines do not require cold enrichment, the contribution to reducing CO levels under cold conditions is substantial.

### 2.2.8.2 *Hydrocarbons (HC)*

Total hydrocarbon emissions from natural gas vehicles tend to be higher, since methane is slower to react than other HC's and in very lean mixtures, the flame speed may be too low for combustion to be completed in the power stroke (43,90). However, the non-methane hydrocarbons (NMHC) or reactive HC emissions, which are of real concern, are considerably lower. It is estimated than the reactive HC emissions are only 15-20% of the total HC emission from the NGV's (91).

### *2.2.8.3* ***Oxides of nitrogen ($NO_x$) Emissions***

The rate of formation of $NO_x$ is exponentially dependent on temperature. In S.I. engines, due to lean air-fuel ratio and lower flame temperatures of natural gas, lower levels of $NO_x$ emissions are encountered. However, in dedicated CNG vehicles, where the ignition timing and compression ratio are optimized, the $NO_x$ levels are expected to be higher (92).

### 2.2.9 *CNG Injection Systems*

One of the disadvantage of in-cylinder gasoline injection spark ignition engines is the time availability for the mixture formation. Since gasoline is in liquid stage at atmospheric temperature and pressure, it takes longer time for evaporation and mixing with air. However, in case of CNG injections the problem of time availability for mixture formation may be eliminated. Since CNG is in gaseous state at atmospheric temperature and pressure it can make homogenous mixture with air in a much less time as compared to gasoline. Conti (93) converted a 2.0 litre turbocharged SI engine with multi point injection to run on CNG. A better thermal efficiency by 16 percent was achieved with CNG injection as compared to gasoline operation.

From the above discussions it is clear that, most of the available alternative fuels that look promising can be used in spark ignition engines for reducing fuel consumptions and exhaust emissions. There appears to be widespread agreement that, relative to gasoline natural gas and ethanol have the positive features of generating less carbon monoxide when combusted. In reviewing the literature concern with emissions and fuel consumptions from spark ignition engines fueled with ethanol and compressed natural gas , several general conclusions can be made. The significant conclusions and the specific gaps, where there is a need to carry out work are explained in the next chapter.

# Chapter 3

## CONCLUSIONS FROM LITERATURE SURVEY AND STATEMENT OF THE PROBLEM

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### 3.1 *INTRODUCTION*

In this chapter the significant conclusions that have been derived from the literature survey carried out pertaining to the area of utilization of alternative fuels for spark ignition engines for better fuel economy and reduced exhaust emissions are briefly described. Based on these conclusions, the objective and need for present work are highlighted. Subsequently the statement of the problem is defined.

### 3.2 *CONCLUSIONS FROM LITERATURE SURVEY*

Based on the comprehensive literature survey carried out, the following conclusions have been derived.

1) Both CNG and ethanol are quite suitable alternative fuels for spark ignition engines. Since ethanol can be produced from sugar cane and India is one of the largest producer of sugar cane and has sufficiently large quantity of proven reserves of natural gas, they can be tried as potential substitutes for conventional gasoline.

2) Ethanol and CNG can be used in spark ignition engines to reduce vehicular pollutions and since they have higher octane ratings, higher compression ratios can be used to increase

power output and fuel economy without encountering engine knock.

3) Ethanol's high heat of vaporization requires special precautions to ensure adequate vaporization in the intake for better cold start and part load performance. As CNG is in a vapour form good starting and better mixture formation is achieved with this fuel.

4) Vapour lock may be a problem with ethanol gasoline blends due to increase in vapour pressure of the blends. Also phase separation of the blends have been reported at lower temperatures

5) Power losses of the order of 10-15% and slower throttle response have been reported under CNG operation in dual fuel mode. There is a need to improve the power and acceleration of CNG fueled engines. A better power output and energy efficiency have been reported with ethanol as compared to gasoline.

6) Higher flammability limits, higher octane number and gaseous state of CNG can be exploited to improve power and thermal efficiency. Up to 20% improvement in thermal efficiency have been predicted.

7) To generate the same amount of energy, 66% more mass of ethanol is required as compared to gasoline. This is because of lower heating value of ethanol per unit mass.

8) Corrosion rate of ethanol especially with zinc, lead and magnesium is higher that with gasoline. Ethanol also swells or softens many plastics and rubbers.

9) Ethanol has a boiling point of 78.3°C, which is about 20% evaporation point of a typical gasoline. For blending of ethanol and gasoline, hydrocarbons at this boiling temperature may be removed to reduce evaporative emissions.

10) Use of ethanol gasoline blends in the existing engines lean the charge, which result in reduced CO. HC may increase due to misfiring because of leaning effect. However, some researchers reporting lower HC emissions also.

11) Lower $NO_x$ emissions and higher aldehyde emissions are reported with ethanol as compared to gasoline by most authors. However, higher $NO_x$ have also been reported by some authors.

12) CO emissions are mainly a function of air fuel ratio. However, lower CO emissions as compared to gasoline have been reported with neat ethanol and CNG under rich operation zone.

13) There may be flame visibility problems with pure ethanol and ethanol-gasoline blends may give phase separation especially at lower temperatures.

14) Comparison of the performance and emission characteristics of a spark ignition engine operating on gasoline, ethanol and natural gas have been scantily reported in the literature.

### 3.3 *AREAS WHERE FURTHER WORK IS REQUIRED*

Until recently, most use of ethanol was in the form of ethanol-gasoline blends. These blends of ethanol in gasoline are commonly used in vehicles designed to operate in gasoline without modifications. As a result there appears widespread agreement that relative to gasoline, blends have positive features of generating less CO emissions, primarily due to "leaning effect". For fuel consumption and exhaust emissions of HC and $NO_x$, picture is somewhat ambiguous, with some researchers reporting lower HC and/or $NO_X$ emissions with blends while others reporting higher emission levels of these species. ***To obtain a true picture it is necessary to compare the engine performance and emissions of ethanol-gasoline blends with gasoline at similar fuel-air ratios. It is also necessary to find out whether or not the gasoline catalytic converters have similar conversion efficiency with ethanol-gasoline blends.***

Experience with pure ethanol engines, however, is quite limited. Some earlier studies (94,95) were carried out to investigate performance characteristics with ethanol but emission studies with ethanol were not carried out. Some more studies on fuel ethanol by Lowry and Devoto (70) measured emissions but no other parameters like compression ratio, ignition timing and effect of catalytic converters were studied. Brinkman's (61) study on pure ethanol was more complete but the effect of catalytic converter, performance at idling and wide-open throttle performance were not carried out. All the above studies were carried out on pure ethanol. But due to flame visibility and cold start problems with pure ethanol 10-15% gasoline may have to be added to ethanol. ***Keeping the above facts in mind this will be quite essential to carry out the performance and emissions characteristics of E-85 (a blend of 85 percent ethanol and 15 percent gasoline by volume) at different air-fuel ratios, compression ratio and ignition timing and to compare them with***

***gasoline. It is also necessary to carry out engine performance at idling, at part load and at WOT. Since harmful emissions like CO, HC, and NO$_x$ can be reduced by using a catalytic converter. It is also necessary to find out whether or not the gasoline catalytic converters have similar conversion efficiency with E-85.*** In conventional spark-ignition engines the fuel and air are mixed together in the engine intake system, and vigorous mixing with the residual gas within the cylinder occurs during the intake process. Thus the fuel-air ratio and the amount of diluent (residual gas) is approximately uniform throughout the charge within the cylinder during combustion. Ethanol's high heat of vaporization requires special precautions to ensure adequate vaporization of the fuel in the intake system to make uniform mixture throughout the charge within the cylinder during combustion. ***Hence, it may be necessary to find out the intake temperature at which homogenous mixture is formed to give best performance.***

Not all vehicles perform in the same manner when operated on natural gas. Further there is a difference in performance when operated with different conversion kits (81). Also percentage of methane vary from pipeline to pipeline. The following general trend is reported with regard to some vehicle performance parameters. In dual fuel operation, there is a power loss of the order of 10-15% under CNG operation. The power loss is observed even under optimized ignition timing. The use of natural gas as an automotive fuel results in reduced concentration of major harmful species in the engine exhaust. ***However, it is necessary to evaluate and compare the performance CNG under similar operating conditions. Also since very little or no work is done with regard to multi-point ignition to improve the performance of CNG engine, it is necessary to evaluate as performance with multi-point ignition.***

In view of the prevailing situation in the present work it is proposed to investigate the performance, fuel economy and emission characteristics of the above fuels by optimizing various operating parameters like compression ratio, equivalence ratio, ignition timing with E-85, ethanol gasoline blends and compressed natural gas.

## 3.4 ***OBJECTIVES***

To summarize in the proposed work, attempts are made to quantify, on a comparative basis the behaviour of ethanol, ethanol gasoline blends and CNG. In order to achieve this objectives the following were decided:-

- To investigate the performance and emission characteristics of spark ignition engines operating on gasoline, ethanol gasoline blends and CNG.
- To optimize various operating parameters like compression ratio, equivalence ratio and ignition timing to achieve optimum performance from the engine using above mentioned fuels.
- To incorporate intake heating system to ensure adequate varporization of ethanol for better part load performance.
- To assess potential improvements in engine performance using multipoint spark to ensure shorter combustion duration with CNG as methane-air flame takes longer time to travel.
- To develop computer simulation model for the combustion process with gasoline of E- 85.
- To conduct parametric studies of the model for obtaining optimum performance of the engine using the above mentioned fuels.
- To understand the combustion behaviour of these alternative fuels by analyzing cyclic variations, rate of pressure rise and maximum cylinder pressure.

## 3.5 *STATEMENT OF THE PROBLEM*

An attempt was made to attain the above envisaged objectives by defining the problems through the following phases:

1. Assessment of engine performance and emission characteristics with conventional gasoline fuel.

2. Assessment of engine performance and emission characteristics with CNG, E-85 , E-20 and E-10. E-85 is a blend of 85% ethanol, and 15% gasoline by volume. Gasoline was added to ethanol for the ease of cold starting and better flame visibility in case fire. Appendix -1 describes an investigation conducted to establish the flame visibility of ethanol. E-20 and E-10 are the blends of 20 % and 10% ethanol and 80% and 90% gasoline by volume respectively.

3. To compare the performance and emission characteristics of these alternative fuels with gasoline udder fuel-air ratios.

4. Design and development of an intake heating system for proper evaporation and mixing of ethanol.

5. Optimization of operating engine parameters like compression ratio, equivalence ratio and ignition timing.

6. Assessment of catalytic converter performance with alternate fuels.

7. To understand the combustion behaviour of alternative fuels by analyzing cyclic variations, rate of pressure rise and maximum pressure.

8. A comprehensive comparison of performance, combustion and emission characteristics of different alternative fuels and gasoline to assess the benefits and difficulties, if any.

To achieve these objectives a single cylinder variable compression ratio engine, which had inbuilt facilities to permit wide spectrum of controlling parameters and a multicylinder engine for carrying out tests at was chosen. Some tests at idling were also carried out using a multicylinder engine. The details of the test bed and the instrumentation is described in the next chapter.

# Chapter 4

## *EXPERIMENTAL SET-UP AND TEST PROCEDURE*

### *4.1* ***INTRODUCTION***

To appreciate the potential of ethanol and compressed natural gas as a spark ignition engine fuel adequate engine data is needed from an engine system. The chosen system should also facilitate the comparison of different fuels and also of mixture preparation methods, in which the operating conditions and engine parameters could be controlled independently. Although a multicylinder engine on a test bed would be more representative of an automobile engine, it would be rather difficult to obtain basic combustion data, which were aimed for this work, due to limited scope for the variation of important operating parameters, such as compression ratio and ignition timing. Furthermore, it would be difficult to vary the fuel flow at constant throttle, which would be needed to compare the operation of different alternative fuels over a wide range of mixture strengths. Hence, for the present experimental investigations, a single cylinder variable compression ratio engine was chosen, which had inbuilt facilities to permit wide spectrum of controlling the parameters and systematic relative evaluation of the performance of different fuels.

### *4.1.1* ***Test Engine***

The engine is of the single cylinder, poppet valve, four-stroke type having a bore of 76.2 mm and a stroke of 111.125 mm. The normal speed range of the engine is 1000-3000 rev/min. The compression ratio in the petrol version can be varied between 4.5 and 20 by raising or lowering the complete cylinder and cylinder head assembly by means of a captive nut operated by a handle. The movement of the cylinder assembly can be measured by means of a micrometer and compression ratio corresponding to a given micrometer reading can be read from a calibration curve. In the petrol version, the combustion chamber is cylindrical in shape, with the ends being formed by the flat surfaces of the cylinder head and the piston. This gives a very compact combustion chamber of good anti-knock qualities and also ensures that the chamber retains the same general form as the compression ratio is varied. The spark plug is situated at one side of the combustion chamber, between the valves. Thus maximum flame travel is not significantly affected very little by change of compression ratio (97). A similar hole on the other side of the combustion chamber enables the position of the spark plug to be changed, placing it near either the inlet or the exhaust valve. In addition a pressure-recording instrument may be fitted. The major specifications of the engine are given in Appendix –II and a photographic view of the engine is shown in Fig. 3.1

In a multicylinder automobile engine the temperature and the pressure inside the combustion chamber during idling is very low and variation of fuel-air mixture in different cylinders is very high (98). Because of this reason it was thought that using a multicylinder engine would be proper and a four stroke water cooled spark ignited gasoline engine was used to carry out the tests at idling. The major specifications of the engine are also given in Appendix –II and this engine will be termed as Engine-B in this thesis.

### *4.1.2* ***Engine Dynamometer***

The engine is coupled to a swinging field electric dynamometer. The dynamometer is of sufficient capacity to absorb the maximum power produced by the engine at all normal engine speeds. Simple switchgear

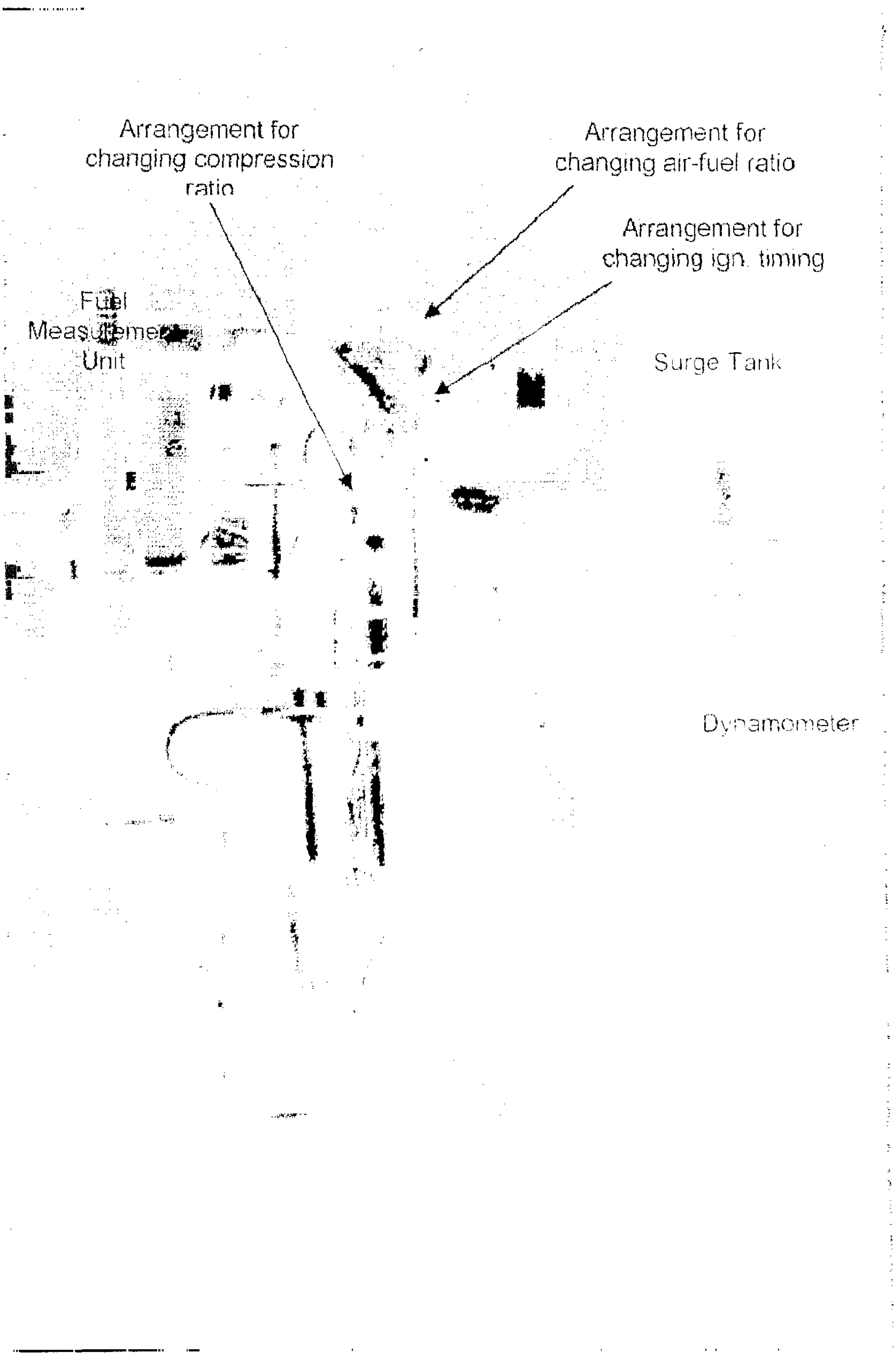

Plate 3.1 Photographic view of the Test Engine

enables the dynamometer to be operated as a motor to start the engine in order to motor it for the determination of friction and pumping losses. A bank of resistance grids having a negligible temperature coefficient of resistance absorbs the energy developed by the dynamometer. A series of knife switches enables the amount of resistance in the armature circuit to be varied, thus varying the load on the engine. Two rheostats, one coarse and one fine, enable the field current to be altered within fine limits, thus providing a further control of the load. A large open scale electrical tachometer driven from the dynamometer shaft and with the scale located in a position convenient to the observer is provided for speed measurement and control.

### *4.1.3* ***Test Fuel***

For carrying out experiments fuels used were

I) commercial gasoline for the purpose of understanding basic performance characteristics of the engine.

II) E-85, which is a mixture 85 percent anhydrous ethanol by volume and 15 percent commercial gasoline by volume. The purpose of using this fuel was to evaluate the performance characteristics of ethanol. However, 15 percent gasoline was added to ethanol for better cold start performance and fire safety point of view. The visibility of pure ethanol flame is poorer as compared to gasoline. (Appendix - I). Adding 15 percent ethanol to gasoline gives better flame visibility. ***Henceforth, this fuel will also be referred to as ethanol.*** E-10 and E-20, refer to the mixtures of 90 percent and 80 percent commercial gasoline and 10 percent and 20 percent anhydrous ethanol respectively.

III) For carrying out experiment with CNG, commercially available compressed natural gas was used.

## 4.2 *INSTRUMENTATION*

The instruments used for measuring the basic quantities are described here:

### 4.2.1 ***Thermocouples***

Different K type (chromel – alumel) thermocouples with digital type temperature indicators were used for measurement of temperatures at different points like intake manifold air temperature, cooling water inlet and outlet temperatures, engine oil temperatures and exhaust gas temperature.

### 4.2.2 ***Digital Tachometer***

A digital tachometer with a photo reflective sensor was used for measuring the engine speed. The photo reflective sensor was placed against a reflective strip mounted on the engine flywheel. Experiments were carried out in the speed range of 1000 rev/min to 2000 rev/min at different load conditions.

### 4.2.3 ***Exhaust Gas Analyzers***

A Horiba infrared gas analyzer was used for exhaust gas emission measurements, namely, carbon monoxide and unburned hydrocarbons. Carbon monoxide (CO) is measured as percent by volume and unburned hydrocarbon (HC) in parts per million (ppm) as n-hexane equivalent. The detailed specifications of the instrument are given in Appendix III.

A Thermo Electron's model 10A Rack mounted chemiluminescent NO-$NO_x$ Gas Analyzer was usedfor measuring oxides of nitrogen from the exhaust. The details of the chemiluminescent is given in Appendix-IV

### 4.2.4 ***Flash Gun (Timing Light)***

Ignition timing is measured by using a Xenon Flashing Gun, or timing light. The crank angle marking is provided on the circumference of the flywheel of the engine. Top Dead Centre is marked by (zero) degree on the flywheel. The timing torch also had a scale of 0 to 50 degrees marking corresponding to the engine crank angle, by which the changes in ignition timing could be read.

### 4.2.5 ***Pressure Transducer, Charge Amplifier and Oscilloscope***

A Kistler, 701 A, water cooled piezo electric, pressure transducer and a charge amplifier of type Kistler 5011, were used for measuring the cylinder gas pressure. The pressure transducer was mounted in the standard position of the spark plug. Piezo electric transducers have the advantage of good frequency response and linear operating range. The details of the instruments are given in Appendices V and VI. A continuous circulation of water was maintained for cooling the transducer by using a small water pump to maintain the required temperature. Distilled water was circulated through the transducer to avoid corrosion of the water passage. The charge amplifier and pressure transducer were calibrated by using a dead weight pressure gauge tester. The charge amplifier was used to amplify the output of pressure transducer into the desired voltage level so that the output of the charge amplifier could be used for recording or display on the oscilloscope screen. A four channel, 200 MHZ Kikusui make Digital Storage Oscilloscope was used for monitoring pressure signals. The specification of this instrument is given in Appendix VII.

## 4.3 ***TEST PROCEDURE***

The basic quantities that were measured during the experimental investigation were: fuel quantity, air quantity, engine speed, power output, exhaust gas temperature, intake air temperature, throttle position, ignition timing, compression ratio, CO, HC and $NO_x$ emission concentrations and pressure crank angle diagrams of cylinder for evaluation of cylinder peak pressure data and cycle – by – cycle variation. Ignition timing was controlled by a lever on the magneto marked off in degrees referred to the crankshaft. The magneto was directly coupled to the overhead camshaft assembly. Necessary readings were taken at MBT position obtained by ignition timing adjustment. Comprehensive experiments were carried out using gasoline, ethanol (E-85), E-20, E-10 and compressed natural gas over a wide range of operating parameters encompassing broad changes in engine speed, load, ignition timing, throttle position, compression ratio. The effects of any parameter on the engine performance are studied by keeping the remaining constant parameters at their base line conditions. The experimental results obtained from the present experimental investigations are given in chapter 6. Tests were conduced with commercial gasoline to find out the basic requirements of the engine, or base line data. The following procedures were followed for conducting the experiments:

1. Engine cooling water valve was opened for the proper cooling of the engine.
2. Carburetor's throttle valve was kept at wide-open throttle position when the tests were carried out at WOT
3. Carburetor's needle valve was kept closed initially and then it was gradually opened.
4. The engine was motored and gradually the needle valve of the carburetor was opened until stable running of the engine was achieved at the lowest power development.
5. Engine was run at idling speed for some time till the cooling water and engine oil temperatures reached the required values. Water flow rate was controlled manually to maintain the required temperature of the engine.

6. Needle valve was opened further for developing maximum power at a particular base line speed and throttle position.
7. Ignition timing was adjusted at a minimum advance for best torque (MBT) for the particular baseline speed and throttle position.
8. Engine loading was done through the loading hand wheel and weights were kept on the torque arm of the dynamometer for balancing and readings are recorded from the balancing dial after the torque arm was adjusted to horizontal position by rotating the balancing hand wheel provided for this purpose.
9. Compression ratio was kept at the base line value of 8.0
10. To study the effect of each parameter, other parameters were kept at base line position.

In the first phase, experiments were undertaken to compare the performance of ethanol (E-85) and ethanol gasoline blends (E-20 and E-10) with gasoline at similar air-fuel ratios. In these tests compression ratio, engine speed and engine power were kept constant at 8.0:1,1500 rev/min. and 2.2 kW respectively. The air-fuel ratio was changed using needle valve of the carburetor. Little variation in engine speed and power as a result of the changed position of the needle valve was adjusted using throttle valve. Fuel-air equivalence ratio was varied from 0.65 at the lean side to 1.2 at rich side. MBT timings at all equivalence ratios were obtained from the determination of start of fall of torque as ignition timing was retarded. For carrying out tests at WOT, the throttle valve was kept fully open and engine speed was changed from 1000 rev./min. to 2000 rev./min. For all WOT tests compression ratio was kept constant at 8.0:1 and ignition timings were kept at 18 degrees before top dead centre and 14 degrees before top dead centre for ethanol (E-85) and gasoline respectively. For ethanol-gasoline blends (E-20 and E-10) the ignition timing was kept at 18 degrees before top dead centre. Constant speed tests were carried out at 1500 rev/min. and 1800 rev/min. In these tests engine load was varied from a minimum load where stable operation was achieved under wide open throttle conditions at both the above mentioned speeds. The intake temperature of air was varied using an electrical heater and a rheostat connected to the surge tank in the intake system. Ignition timing was controlled by a lever on the magneto marked off in degrees referred to the crankshaft. The magneto was directly coupled to the overhead camshaft assembly. Necessary readings were taken at MBT position obtained by ignition timing adjustment. For comparing pressure-time history of various fuels, cycle pressures at various crank angles using these fuels were marked on a transparent paper which was fixed at the screen of the storage oscilloscope. The pressure points with respect to various crank angles were marked using a very fine permanent marker and then pressure crank angle diagrams of different fuels were plotted.

Although several parameters could be obtained from an engine test bed, certain basic data are difficult to measure. These include the continuous tracking of emission formation inside the engine cylinder and the gas temperature. Such studies could be easily carried out with computer simulation model. The next chapter describes one such model which can predict the above data.

# Chapter 5

# THEORETICAL INVESTIGATIONS AND COMPUTATIONAL TECHNIQUES

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## 5.1 *INTRODUCTION*

The combustion modelling and cycle simulation have proved to be quite valuable tool in the design of engines, narrowing down the range of experiments and predicting the performance and exhaust emissions of a combustion engine. These models can predict a wide range of performance and emission characteristics including time varying parameters which are difficult to measure under different operating conditions on an experimental test bed. Such model predictions can go long way in narrowing down the range of experiments on a test bed for a specific application.

During the past three decades, several mathematical models (99-105) catering to the specific areas of spark ignition engines have been developed. These models can ably predict the performance, fuel economy and exhaust emissions to a considerable level of accuracy of spark ignited gasoline engines. However, relatively very little work is available on modelling of spark ignition engines operating on ethanol (E-85). In view of this, an attempt was made to modify an existing combustion model (106) for a single cylinder SI operating on E-85. Several stages involved in the development of the model are illustrated below:-

## 5.2 *POWER CYCLE SIMULATION*

The period during which both the inlet and exhaust valves remain closed is the most significant part of the engine cycles, it being the power development period. The closed period, therefore, has been assumed to start from the point of inlet valve closing and extends until the exhaust valve opens. The assumptions that have been made in developing the incylinder model for the homogeneous charge spark ignition engine are mostly well accepted and are valid in cycle simulation work.

During the incylinder processes, the following assumptions have been made.

1. The pressure and temperature are uniform throughout the cylinder. During combustion, temperatures in the burnt and unburnt zones are those estimated by ideal gas laws.
2. The unburnt mixture at any instant consists of homogeneous fuel-air mixture and residual gases without any chemical reactions.
3. The heat transfer area consists of five different surfaces, namely
   (a) cylinder head without valves
   (b) the piston top
   (c) the instantaneous sleeve surface
   (d) the inlet valve and
   (e) the exhaust valve.
4. The metal temperature of each of the above surface is constant throughout the cycle.
5. The rate of heat transfer from gas to the wall is as calculated from the instantaneous heat transfer coefficient, concerned surface area and the difference in temperature between the gas and wall.

### 5.2.1 *BASIC APPROACH AND EQUATIONS*

The Basic equations involved in the general energy equation used for solving any thermodynamic system are:-

(1) Conservation of mass,
(2) Conservation of energy
(3) Conservation of momentum,
(4) Expression for internal energy of working fluid as a function of pressure, temperature and composition, and
(5) Equation of state.

## 5.3 *GENERAL ENERGY EQUATION*

The first law of thermodynamics put in the most general form as basic energy equation, helps in the estimation of temperature of any thermodynamic analytically. The general form of energy equation as applied to an open system may be written as

$$\dot{E} = \dot{\overline{Me}} = \Sigma\dot{Q}_i - p\dot{V} + \Sigma h_i\dot{M}_i \tag{5.1}$$

Where

$\dot{E}$ is the rate of change of internal energy of the system mass M,

$\dot{Q}_i$ is the rate of heat transfer through the boundary at location i,

$p\dot{V}$ is the rate of mechanical work done at the boundary,

$h_i\dot{M}_i$ is the energy convection in or out of the system at location i.

This energy conservation equation can be used to develop a differential equation for evaluating explicitly the change in temperature of thermodynamic system, by rearranging the equation in terms of dependent variables such as pressure P, temperature T, and equivalence ratio F. The equivalence ratio is defined as the ratio of actual fuel-air ratio to the stoichiometric fuel-air ratio.

The specific internal energy e, when expressed a function of dependent variables, is given as

$$e = e\ (T, P, F)$$

and by differentiating,

$$\dot{e} = \frac{\partial e}{\partial T}\dot{T} + \frac{\partial e}{\partial P}\dot{P} + \frac{\partial e}{\partial F}\dot{F} \tag{5.2}$$

Substituting equation (5.2) in equation (5.1),

$$\dot{E} = M\dot{e} + e\dot{M} = -\Sigma\dot{Q}_i - P\dot{V} + \Sigma h_i\dot{M}_i$$

or

$$M\left(\frac{\partial e}{\partial T}\dot{T} + \frac{\partial e}{\partial p}\dot{P} + \frac{\partial e}{\partial F}\dot{F}\right) = \Sigma\dot{Q}_i - P\dot{V} + hi\ \dot{M}_i - e\dot{M} \tag{5.3}$$

In order to obtain an equation independent of the cylinder pressure, the equation of state can be used. The equation of state for an ideal gas is given as

$$PV = MRT$$

Or, in a differential form,

$$P\dot{V} + V\dot{P} = MR\dot{T} + RT\dot{M} + MT\dot{R} \tag{5.4}$$

Expressing the specific gas constant R = R(T,P,F) in differential form,

the derivative $$\dot{R} = \frac{\partial R}{\partial T}\dot{T} + \frac{\partial R}{\partial P}\dot{P} + \frac{\partial R}{\partial F}\dot{F} \tag{5.5}$$

Substituting equation (5.5) in equation (5.4),

$$P\dot{V} + V\dot{P} = MR\dot{T} + RT\dot{M} + MT\left(\frac{\partial R}{\partial T}\dot{T} + \frac{\partial R}{\partial P}\dot{P} + \frac{\partial R}{\partial F}\dot{F}\right)$$

Rearranging,

$$\dot{P}(V - MT\frac{\partial R}{\partial P}) = M\dot{T}\,(R + T\frac{\partial R}{\partial T}) + RT\dot{M} - P\dot{V} + MT\dot{F}\frac{\partial R}{\partial F} \tag{5.6}$$

Substitution of equation (5.6) in equation 5.3) leads to

$$M\dot{T}\frac{\partial e}{\partial T} + M\frac{\partial e}{\partial P}\left[M\dot{T}(R + T\frac{\partial R}{\partial T}) + RT\dot{M} - P\dot{V} + MT\dot{F}\frac{\partial R}{\partial F}\right] /$$

$$(V - MT\frac{\partial R}{\partial P}) + M\dot{F}\frac{\partial e}{\partial F} = -\Sigma\dot{Q}_i - P\dot{V} + \Sigma h_i\dot{M}_l - e\dot{M}$$

Simplifying further and rearranging,

$$\dot{T}\left[M\frac{\partial e}{\partial T} + M\frac{\partial e}{\partial P}M(R + T\frac{\partial R}{\partial T}) / (V - MT\frac{\partial R}{\partial P})\right] = -P\dot{V} - \Sigma\dot{Q}_i +$$

$$h_iM_i - e\dot{M} - M\frac{\partial e}{\partial P}(RTM - P\dot{V} + MT\dot{F}\frac{\partial R}{\partial F}) / (V - MT\frac{\partial R}{\partial P}) - M\dot{F}\frac{\partial e}{\partial F}$$

$$\text{or} \quad \dot{T}\,[\frac{\partial e}{\partial T} + \frac{\partial e}{\partial P}\frac{P(1 - \frac{T}{R}\frac{\partial R}{\partial T})}{T(1 - \frac{P}{R}\frac{\partial R}{\partial P})}] = -\ \frac{RTV}{V} + \frac{1}{M}(-\Sigma\dot{Q}_i + \Sigma h_i\dot{M}_i - e\dot{M})$$

$$-\dot{F}\frac{\partial e}{\partial F} - \frac{\partial e}{\partial P}\left(\frac{P\dot{M}}{M} - \frac{P\dot{V}}{V} + \frac{P\dot{F}}{R}\frac{\partial R}{\partial F}\right) / (1 - \frac{P}{R}\frac{\partial R}{\partial P}) \tag{5.7}$$

Let

$$z_1 = -\ RTV/V + (-\Sigma\dot{Q}_i + \Sigma h_i\dot{M}_i - e\dot{M})/M \tag{5.8}$$

$$z_2 = 1 - \frac{P}{R}\frac{\partial R}{\partial P} \tag{5.9}$$

$$z_3 = 1 + \frac{T}{R}\frac{\partial R}{\partial T} \tag{5.10}$$

The equation (5.7) simplifies to

$$\dot{T} = \frac{z_1 - \dot{F}\frac{\partial e}{\partial F} - \frac{P}{z_2}\frac{\partial e}{\partial P}\left(\frac{\dot{M}}{M} - \frac{\dot{V}}{V} + \frac{\dot{F}}{R}\frac{\partial R}{\partial F}\right)}{\frac{\partial e}{\partial T} + \frac{P}{T}\frac{Z_3}{Z_2}\frac{\partial e}{\partial P}} \qquad (5.11)$$

Equation (5.11), giving the derivative of temperature, holds good for all incylinder processes except the combustion process wherein a different set of relations have been developed. This equation can be solved for the gas temperature, by suitable numerical integration, in conjunction with the equations for heat transfer, mass flow rates and volume change. It is also necessary, at all times, to keep track of the changing mixture composition in the cylinder to derive the property relations of gas constant and internal energy.

### 5.3.1 *Cylinder volume*

The cylinder volume at any instant of crank angle can be found out from the geometry of the engine and the slider-crank relationship. The instantaneous total volume of the gases in the cylinder is given by

$$V = V_{CL} + A_P[CL + \frac{S}{2}(1 - \text{Cos}\theta) - (CL^2 - (\frac{S}{2})^2\ \text{Sin}^2\theta)0.5] \qquad (5.12)$$

Also, differentiating equation (5.12), the rate of change of cylinder volume is obtained as

$$V = A_p\ [\frac{S}{2}\ \text{Sin}\theta + [\frac{S}{2}]^2\ \text{Sin}2\,\theta\ /\ (4CL^2 - 2(\frac{S}{2})^2(1 - \text{Cos}2\,\theta))\ )^{0.5}] \qquad (5.13)$$

Where θ is the crank angle measured from the top dead centre in radians.

### 5.3.2 *Heat Transfer*

The heat transfer rate for any surface at a particular instant can be determined by

$$\dot{Q}_i - h'_i A_i (T_g - T_{wi}) \qquad (5.14)$$

The heat transfer coefficient in the cylinder can be estimated by a number of empirical correlations. Gas to wall heat transfer in the cylinder of a spark ignition engine is controlled by a rapidly changing boundary layer, caused by gas motion due to piston movement, geometry of combustion chamber etc. The instantaneous heat transfer coefficient is calculated by using. Annand' equation (107).

$$h' = a' (KK/D) (Re)^{0.7} + b' (T_g^4 - T_w^4)/( T_g - T_w) \qquad (5.15)$$

where

KK = thermal conductivity

Re = Reynolds number

$a'$ = 0.27 to 0.93

$b'$ = $0.4292 \times 10^{-8}$

= 0.0 for compression process.

### 5.3.3 *Properties of reactants mixture*

Needless to stress, it is extremely important to have reliable physical data for the working species in order to evaluate the thermodynamic properties of the reactants mixture and products of combustion continuously during the entire cycle calculation. The ideal gas mixture assumptions have been used to estimate the gas constant, the internal energy and the enthalpy for the reactants mixture. For all the properties the datum has been fixed at absolute zero temperature.

In this model, the internal energy of air, the gas constant and internal energy of residual gases, sensible enthalpy and enthalpy of formation of fuel vapour and finally the internal energy of the fuel have been derived.

### 5.3.4 *Residual gas fraction*

The residual gas fraction is defined as the ratio of residual gas mass to the total mass of the reactants mixture. The variation of residual gas fraction with respect to equivalence ratio and intake temperature is assumed to be negligible. However, its variation in relation to changes in compression ratio and throttle opening is quite significant. Reasonable estimate of the residual gas fraction, to begin the calculations of closed period, can be obtained from the expression (108).

$$RGF = \frac{1}{1+\frac{T_{em}}{T_{ch}}[C_R(\frac{P_{mx}}{P_{em}})-\frac{P_{mx}}{P_{em}}^{(k-1)/k}]} \qquad (5.16)$$

Since the temperatures and pressures in the exhaust and intake have to be assumed to get the residual gas fraction from the above equation (5.16), it gives an approximate value, which anyway has been estimated and used during the first iteration only. In subsequent cycles, the actual quantity of residual gases, calculated from the open period analysis of proceeding iteration has been used.

With the evaluation of the above mentioned modeling functions, equation (5.11) can now be solved for various in-cylinder processes. During the compression process in spark ignition engines, the pre-flame reactions are usually neglected and hence equation (5.11) is considerably simplified. Furthermore, because of the closed valves, there is no mass transfer at the boundary.

Therefore,

$$\dot{F} = 0$$

$$\dot{M} = 0$$

## 5.4 *COMPRESSION PROCESS*

Compression process is modeled by a simplified form of equation (5.11), as it has been established beyond doubt that dissocation phenomena are insignificant for temperatures approximately below 1600 K.

$$\frac{\partial e}{\partial P} \approx 0 \text{ and } \frac{\partial R}{\partial P} \approx 0$$

Since the equivalence ratio and mass do not change with time during the compression process, equation (5.11) reduces to

$$\dot{T} = Z_1 / \frac{\partial e}{\partial T} = (-RT\dot{V}/V - \frac{1}{M}\Sigma Q_i) / \frac{\partial e}{\partial T} \tag{5.17}$$

The heat transfer term has been obtained by summing up the heat transfer through the five incylinder surfaces, for which the wall temperatures, initially, have been assumed, based on previous studies on similar engines (106). Compression process is assumed to begin at the inlet valve closing point. The temperature of the charge and volumetric efficiency have been provided as the model inputs, either assumed or derived from the experiment. With the estimated residual gas fraction available, the mass of the reactants has been found out by using a perfect mixing model for fresh charge and residuals. The gas constant and the rate of change of internal energy with respect of temperature ($\partial e/\partial T$) have been obtained from the property evaluation of the reactant mixture. From the known position of the piston at a given crank angle, the instantaneous volume and its derivative have been calculated.

Equation (5.17) has been integrated by fourth order Runge-Kutta method with an interval increment of $4^0$ crank angle. At the end of interval, pressure of the reactant gas has been obtained from the equation of state. Heat transferred across each particular surface is added up cumulatively. The compression process is assumed to be over when ignition crank angle is reached. Combustion process is assumed to start at the time of spark occurrence.

## 5.5 *COMBUSTION MODEL*

The success of any cycle simulation predominantly depends on the accuracy with which the combustion process can be predicted, and on the validity of assumptions and approximations made in the synthesis of other events that precede and follow the combustion phase. It is least surprising that this should be the situation, for the combustion is the process during which energy is released by the bruning of fuel. Both efficiency and emissions are influenced by the details of the combustion phenomena.

The resolution of the reacting flow both in space and in time is certainly a formidable task in spite of the fast computers, for the most general case would be a numerical solution of the entire set of Navier-Stroke equations along with the energy and species conservation equations. In order to simplify the problem to manageable proportion, several reasonable and restrictive assumptions are permissible, leading to different approaches in modeling techniques. Yet, in view of the ambiguity and complexity involved in the use of multidimensional combustion models, it has been felt that phenomenological or thermodynamic models are adequate enough to evaluate the relative performance, emission and combustion characteristics of methanol or gasoline operated spark ignition engine, provided that the basic physical processes are suitably represented.

In the combustion model detailed ahead, expressions to evaluate the burnt and un-burnt gas temperature, the rate of pressure rise and the rate of mass burnt have been derived. A methodology to determine the position of the flame, spreading out from an arbitrary location of the spark plug is presented. The effect of combustion induced turbulence and its decay as combustion progresses is taken care of. Using the equilibrium composition of the burnt gas, non-equilibrium rate controlled processes of nitric oxide and carbon monoxide formation and destruction in post combustion gases have been modeled. The solution technique to evaluate the above quantities has been described.

### 5.5.1 *Basic Assumptions*

Most of the assumptions stated below have been already employed in cycle simulation work and have been shown valid under normal operating conditions.

1. A two zone combustion model, consisting of an unburnt zone and a burnt zone, is used to describe the combustion process. The two zones are at different temperatures and are separated by a infinitesimal thin flame front.
2. Heat transfer does not occur between burnt and un-burnt zones. Heat transfer rate to the walls is estimated individually for each zone from the same empirical heat transfer coefficient relation.

3. Un-burnt zone consists of a non-reacting mixture of fuel vapour, air and residual products. Phenomena of dissociation is neglected in un-burnt zone.
4. Burnt zone consists of products of combustion which are in chemical equilibrium.
5. Pressure is uniform throughout the combustion space and is a function of time only. Medium in both the zones is idealized as perfect gas.
6. Instantaneous combustion occurs in the flame front, which propagates from an arbitrary location of spark plug by a specific geometric (spherical or cylindrical) mode. Initial temperature of ignition is equal to the constant pressure adiabatic flame temperature.
7. Nitric oxide emissions are calculated by using the extended Zeldovich kinetic mechanism, whereas carbon monoxide is estimated by the principal water gas shift reaction.
8. Blowby and valve leakage losses are neglected.

The combustion model that was employed has been described in detail (106). In the present analysis, pollutants (CO and $NO_x$ formation has been modeled (106) through kinetic mechanisms for nitric oxide and carbon monoxide

## 5.6 ***EXPANSION PROCESS***

The analysis of expansion process is similar to that of the compression process, except that dissociation of burnt products is no longer negligible and book keeping of pollutants formed throughout expansion process is to be adhered rigorously.

Equation (5.11), which is applicable to expansion period, can be written as

$$\dot{T} = (Z_1 + \frac{\partial e}{\partial P}\frac{P}{Z_2}\frac{\dot{V}}{V}) / (\frac{\partial e}{\partial T} + \frac{\partial e}{\partial P}\frac{P}{T}\frac{Z_3}{Z_2}) \qquad (5.18)$$

Interaction with the equilibrium routine has enabled the properties of burnt gases to be evaluated. The temperature and pressure have been evaluated by integrating equation (5.18) with a time step of $2^0$ crank angle. With the equilibrium composition as the input, nitric oxide and carbon monoxide kinetics have been followed throughout the expansion phase, which has been assumed to end with the exhaust valve opening.

The main programme consists of sequential segments for different phases of the cycle, i.e. compression, combustion, expansion, combustion, expansion exhaust and intake for .undertaking step by step calculations (full details are described in reference). The input to the computer programme are the engine geometry, operating variables such as speed, ignition timing, equivalence ratio and also various constants and empirical expressions and the assumed wall temperatures for different heat transfer surfaces. Fig. 5.1 shows the flow chart of the main programme and in the next chapter the results of the simulation model is compared with experimental results for validation and further analysis.

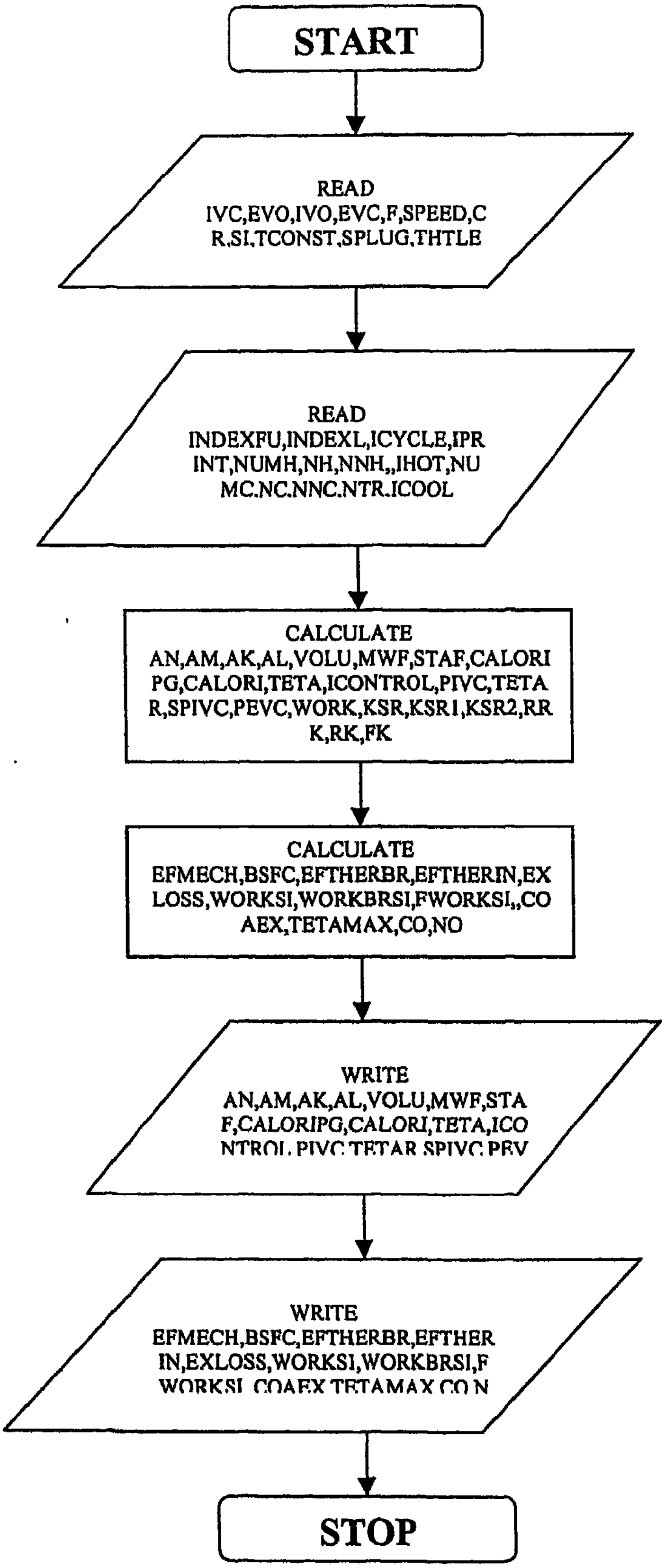

**Fig. 5.1 Flow Chart of the main programme.**

# Chapter 6

# RESULTS AND DISCUSSIONS

## *6.1* ***INTRODUCTION***

In this chapter, the results obtained from the experiments on test beds and simulation model are presented and discussed. The experiments were carried out using gasoline, ethanol (E-85), ethanol-gasoline blends (E-20 and E-10) and compressed natural gas. For the experimental study, the parameters that were varied include fuel-air ratio, ignition timing, engine speed, engine load, and intake air temperature and compression ratio. For carrying out tests at different equivalence ratio, fuel-air ratio was varied using tapered needle of the fuel metering system. Brake mean effective pressure was varied by increasing the throttle opening gradually and the corresponding bmep was calculated.

The results presented in this chapter highlight the effect of major parameters that influence engine operation with gasoline, ethanol (E-85) ethanol-gasoline blends (E-20and E-10) and compressed natural gas. Fuel consumption, power output, brake thermal efficiency, exhaust emissions (CO, HC & $NO_x$), exhaust gas temperature, peak pressure and variation of peak pressure were the parameters that were measured and analyzed. In the following paragraphs the experimental results obtained with ethanol (E-85) and ethanol-gasoline blends (E-20and E-10) are discussed first, followed by results obtained with CNG and then the results obtained from the simulation model.

## 6.2 *RESULTS OBTAINED WITH ETHANOL (85E) AND ETHANOL–GASOLINE BLENDS*

The results obtained from the experimental investigations with gasoline, E-85, E-20 and E-10 are presented and discussed in this section. For studying the performance of ethanol (E-85), and other ethanol-gasoline blends, first the performance of these fuels with the variations in equivalence ratio was studied. The parameters that were studied for the variations of equivalence ratio were thermal efficiency, fuel consumption and exhaust gas emissions of CO, HC and $NO_x$. The results were also compared with gasoline. Other results, which included in this section, are the effect of engine speed, engine load, intake air temperature and ignition timing. The engine performance at idling and the performance of catalytic converter with ethanol (E-85) are also discussed. Since trends of the performance with E-20 and E-10 were almost similar only the results obtained with E-10 presented and discussed except for WOT and constant speed performance where a set of results of E-20 are shown.

### 6.2.1 *Performance with the Variations of Equivalence Ratio*

To compare the performance of ethanol (E-85) and ethanol-gasoline blends with gasoline, fuel consumption and exhaust emission characteristics of these fuels were studied at several fuel-air ratios on a single cylinder engine. All the results are shown against the fuel-air equivalence ratio, which is defined as actual fuel-air ratio divided by the chemically correct fuel-air ratio and is denoted by $\phi$. If $\phi$ is more than one the mixture is said to be rich and if it is less than one it is designated as lean. Table 6.1 lists the engine operating conditions under which they were held constant

**Table 6.1 Constant Engine Test Conditions**

| | |
|---|---|
| Engine Speed, rev./min | 1500 |
| Coolant Temperature, deg. C | 80 |
| Oil Temperature, deg. C | 80 |
| Intake Air Temperature, deg. C | Atmospheric |
| Brake Power, kW | 2.2 |
| Compression Ratio | 8.0 |

#### 6.2.1.1 *Thermal efficiency and fuel consumption*

The fuel utilization by the engine is generally represented by both brake specific fuel consumption on mass basis and brake thermal efficiency. In the discussion of fuel consumptions of ethanol (E-85), ethanol-gasoline blends and gasoline, the thermal efficiency, which is independent of the heating value of these fuels, is discussed first. Fig. 6.1 shows the brake thermal efficiency data for gasoline, E-85 and E-10 as a function of equivalence ratio. As expected thermal efficiency was maximum at equivalence ratios between 0.95 to 1.0. With E-85 thermal efficiency was 2-3 percent greater than that with gasoline throughout the range of equivalence ratios. Greater thermal efficiency with E-85 as compared to gasoline may be due to faster burning and higher maximum pressure with the former. As it was seen from Table 2.3 then the burning velocity of ethanol was 48 cm/sec as compared to 45 cm/sec of gasoline. With E-10, thermal efficiency was slightly better when the mixture was leaner than stoichiometric ($\phi$ between 0.75 and 0.95) as compared to gasoline.

Although thermal efficiencies with ethanol (E-85) and ethanol gasoline blend (E-10) were better than with gasoline, fuel consumption, as shown in Fig. 6.2 was poorer with ethanol (E-85) as well as with 10-E

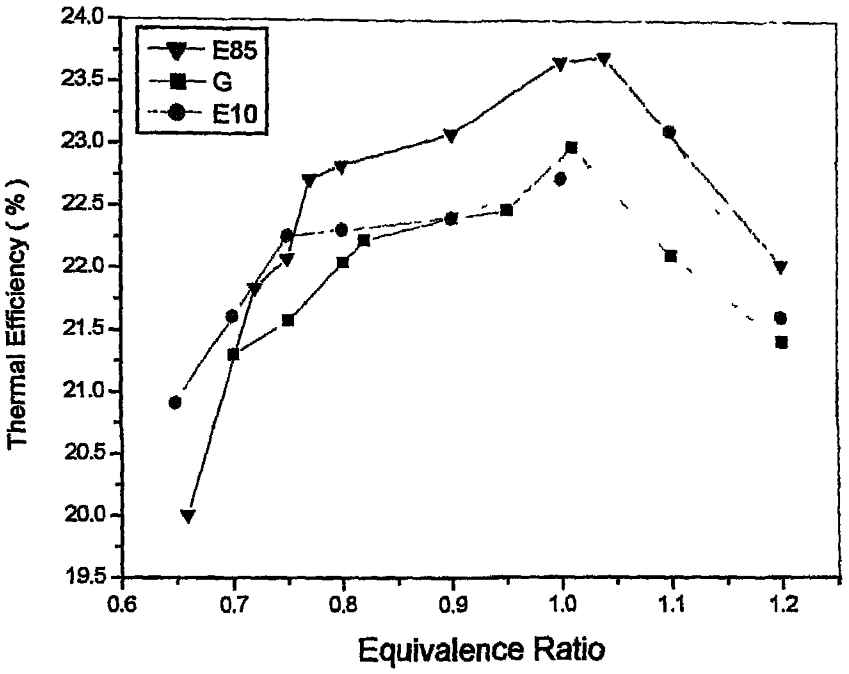

Fig. 6.1 Effect of A / F ratio and fuel on thermal efficency at 1500 rpm , power=2.2 kW, CR= 8.0

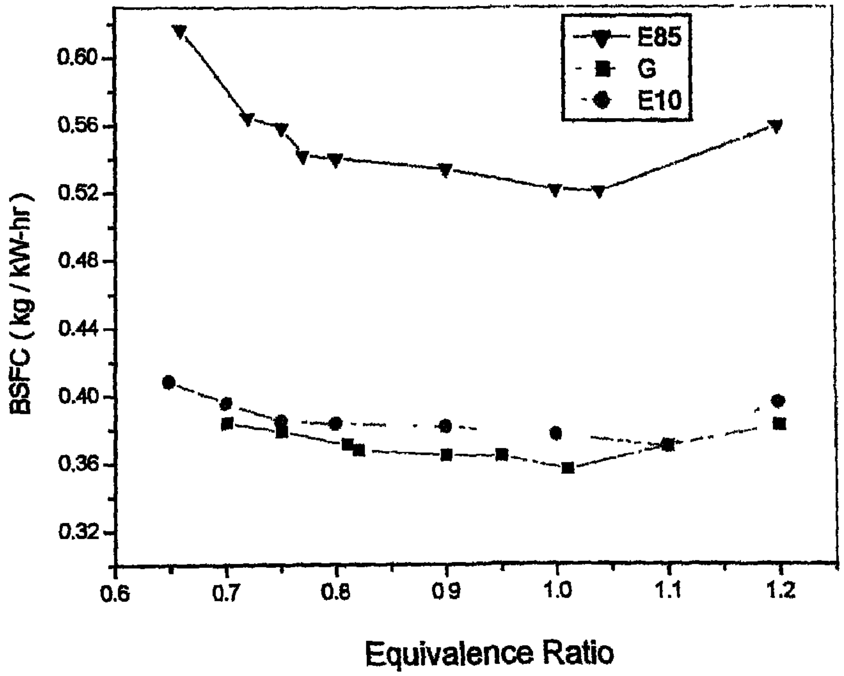

Fig. 6.2 Effect of A / F ratio and fuel on BSFC at 1500 rpm, power=2.2 kW, CR= 8.0

because of ethanol's lower heating value. At a given compression ratio, fuel consumption increased by 50 per cent for the case of E-85 and 2 per cent for E-10. Although 50 percent increase in the consumption of ethanol (E-85) was quite substantial, it was observed that the increase in energy input supplied was only 60 per cent.

### 6.2.1.2 *Hydrocarbon Emissions*

Based on the results of other studies (109,110) unburned hydrocarbon emissions assumed to be primarily ethanol vapour when engine was operated on ethanol and when engine was operated on gasoline, it was assumed to be gasoline vapour. However, the response of analyzer is not the same for ethanol and for gasoline. The response of the unburned hydrocarbon analyzer per carbon atom of ethanol was 80 percent of that for propane (61). Accordingly, the analyzer reading was divided by 0.8 when engine was fueled with ethanol (E-85).

Fig.6.3 shows the variation of unburned hydrocarbons with equivalence ratio for gasoline, ethanol (E-85) and E-10. The profiles of unburned hydrocarbon emissions versus equivalence ratio were conventional and minimum unburned hydrocarbon emission occurring between the equivalence ratios of 0.85 and 0.9. Unburned hydrocarbon emissions with ethanol (E-85) were found to be 4–10 percent higher as compared to gasoline especially in lean zone where equivalence ratio was varying between 0.65 to 0.95. In case of E-10 also unburned emissions were found to be 3 – 8 percent higher as compared to gasoline throughout the range of lean equivalence ratios ($\phi$ from 6.5 to 1.0). Since the entry of the fuel vapour to the crevices, flame quenching and absorption by oil layer would be more or less similar for both fuels, the higher unburned hydrocarbon emissions with ethanol (E-85) and E-10 were attributed to more misfired cycles under lean operating conditions.

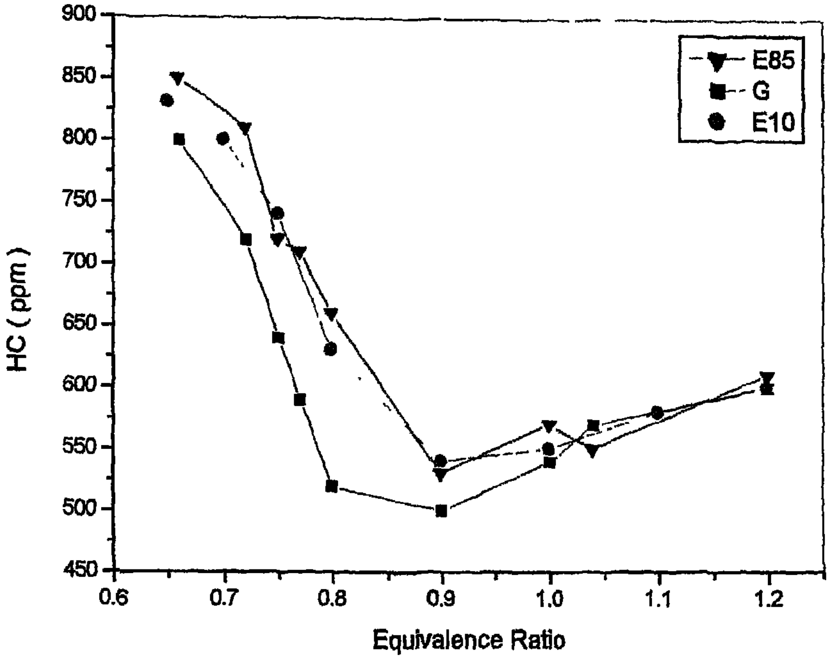

Fig. 6.3 Effect of A / F ratio and fuel on exhaust emissions of HC at 1500 rpm , power=2.2 kW, CR= 8.0

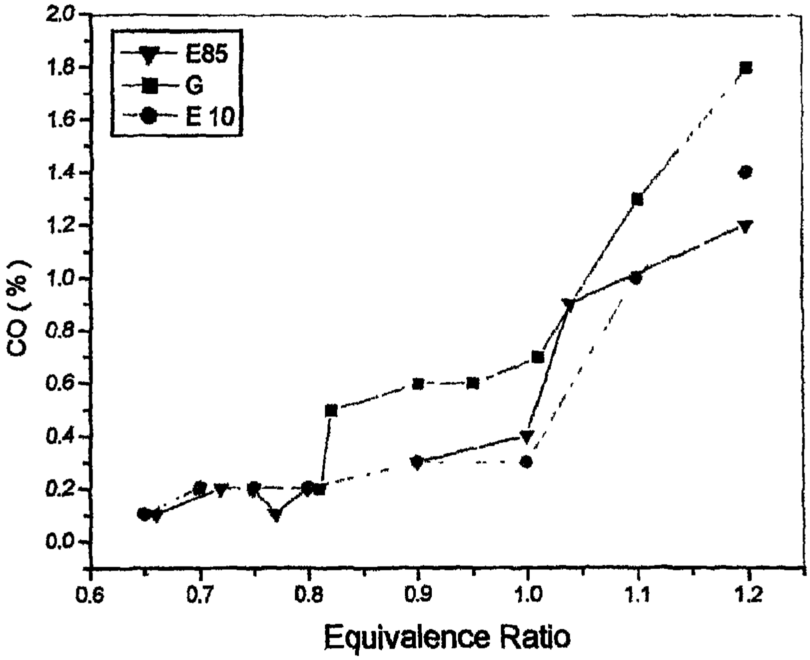

Fig. 6.4 Effect of A / F ratio and fuel on exhaust emissions of CO at 1500 rpm , power=2.2 kW, CR= 8.0

Similar trend in unburned hydrocarbon emissions were observed when operated at richer mixtures. Similar results were obtained by Bechtold and Pullman (71). However, these data disagree with those reported by Lowry and Deveto (70), which showed reduced unburned hydrocarbon emissions with ethanol. But they used unheated flame ionization detector (FID). Data reported by University of Santa Clara (69) showed that unburned hydrocarbon emissions with ethanol to be twice that of gasoline. Brinkman (61) found that with ethanol, unburned hydrocarbon emissions were quite similar to those with gasoline at a compression of 7.5, although some differences were observed on both lean and rich sides. However, Brinkman had reported the unburned hydrocarbon data as a mass of gasoline for gasoline fuel and as a mass of ethanol for ethanol fuel.

#### *6.2.1.3* ***Carbon Monoxide Emissions***

Exhaust emissions of carbon monoxide, as shown in Fig. 6.4 depended primarily on equivalence ratio. It was higher at richer mixtures and lower in case of leaner mixtures. Percentage of CO was low at equivalence ratios from 0.65 to 1.0. It varied from 0.1 percent to 0.3 percent for both ethanol (E-85) and E-10. However, it was slightly higher in case of gasoline as compared to ethanol (E-85) and E-10. As the value of equivalence ratio increased above 1.0, CO emissions increased rapidly, but the rate of increase was different for gasoline, ethanol (E-85) and E-10. The difference of CO emissions was quite prominent between these fuels at equivalence ratios richer than stoichiometric. At equivalence ratio of 1.2, CO emissions with gasoline were higher by 16 percent as compared to E-10 and was higher by 50 percent as compared to ethanol (E-85). Similar results were reported by Harrington and Pilot (111) with methanol fuel. Lower CO emissions with ethanol may be due to lower carbon to hydrogen ratio of ethanol as compared to gasoline.

#### 6.2.1.4 $NO_x$ Emissions

The effect of equivalence ratio on NOx emissions is shown in Fig. 6.5. For gasoline and ethanol (E-85) maximum $NO_x$ emissions were observed at an equivalence of 0.95 to 1.0 . However, maximum $NO_x$ with ethanol- gasoline blend (E-10) was observed at an equivalence of 1.0. $NO_x$ emissions were higher with gasoline throughout the range of fuel-air equivalence ratios except at very rich mixture ($\phi$ = 1.2). Maximum $NO_x$ emissions with ethanol (E-85) were about 26 percent lower than the corresponding maximum with gasoline. $NO_x$ emissions with E-10 fuel were also observed to be lower as compared to gasoline especially at equivalence ratios richer than stoichiometric. Lower $NO_x$ emissions with E-85 and E-10 as compared to gasoline were attributed to lower cylinder gas temperature with these fuels (Cylinder gas temperatures computed with gasoline and E-85 are discussed in section 6.5).

All of the above tests were conducted at MBT (Minimum Advance for Best Torque). For ethanol (E-85) and gasoline MBT timings under various equivalence ratios are presented in Fig. 6.6. As it can be seen from this figure that lesser spark advance (4 - 6 degrees) is required for ethanol (E-85) as compared to gasoline for the best torque. This reflects faster burning characteristics of ethanol as compared to gasoline. It was also observed that for E-85 and E-10, lowest spark advance was required at about 5 to 10 percent richer mixtures. This was attributed to faster burning of ethanol at these mixture ratios.

### 6.2.2 *Wide Open Throttle (WOT) Performance*

The effect of engine speed on thermal efficiency and brake specific fuel consumption with gasoline, E-85, E-20 and E-10 at WOT are shown in Figures 6.7 and 6.8 respectively. These tests were carried out at a

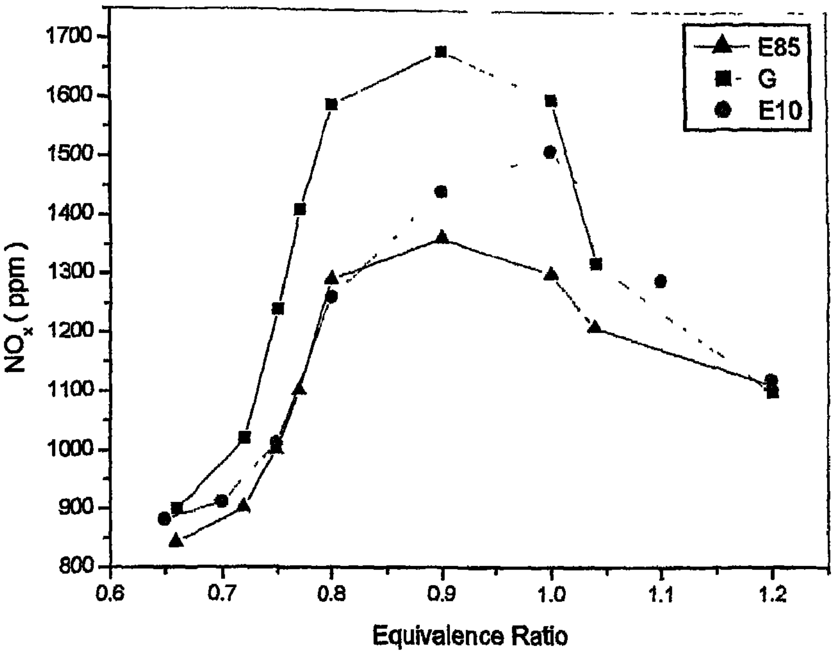

Fig. 6.5 Effect of A / F ratio and fuel on exhaust emissions of $NO_x$ at 1500 rpm , power=2.2 kW, CR= 8.0

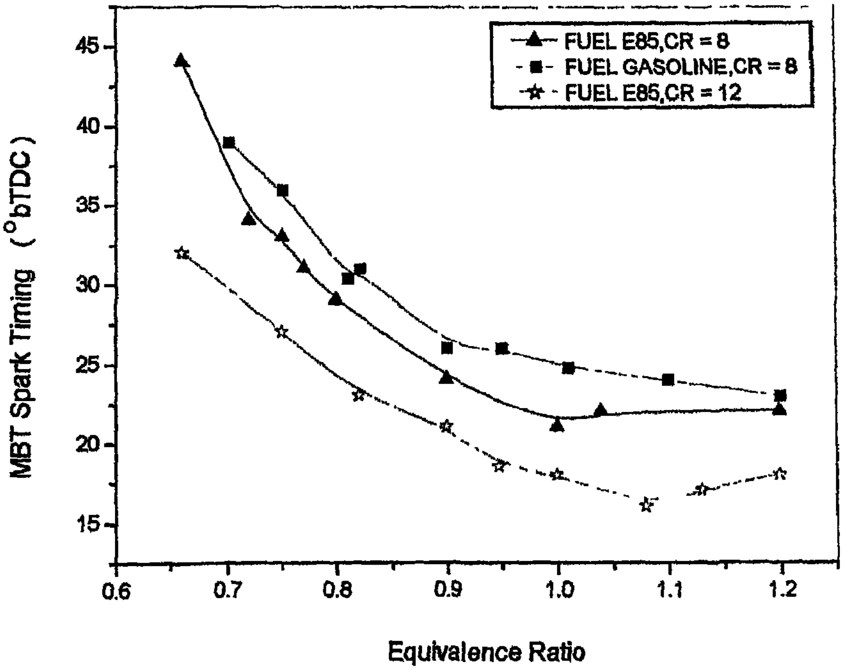

Fig. 6.6 Effect of A / F ratio and fuel on MBT spark timing at 1500 rpm, power=2.2 kW, CR= 8.0

constant compression ratio of 8.0 for all fuels. Ignition timing was kept at 18 degrees before top dead centre (bTDC) for gasoline, E-10 and E-20, while for ethanol (E-85) it was kept at 14 degrees bTDC. With E-85 thermal efficiency was observed to be better by about 3 to 5 percent throughout the range of speeds as compared to gasoline. While BSFC with E-85 was 45 to 55 percent higher as compared to gasoline. Better thermal efficiency with E-85 at WOT as well as at similar equivalence ratio was attributed to faster burning and higher peak pressure than those with gasoline (Peak pressures obtained with different fuels are discussed in the later sections of this chapter). Also since the latent heat of vaporization of ethanol is higher than that of gasoline, so adding ethanol to gasoline results in lower intake manifold temperature, which increases charge density to improve volumetric and thermal efficiencies.

Higher BSFC with E-85 is due to lower calorific value than gasoline. However, at WOT fuel consumption on mass basis was about 45 to 55% higher. Thermal efficiency with E-10 was slightly better as compared to gasoline. As compared to gasoline, BSFC was higher with both E-10 and E-20, which is due to lower energy densities of E-10 and E-20. Lowest BSFC was obtained at an engine speed of 1800 rpm for all the fuels. This is due to more heat rejection to coolant at lower speeds (lower than 1800 rev/min) for all fuels. Also at higher speeds (higher than 1800 rev/min) considerably more heat is carried away by the exhaust gases. Trends in engine power variation at WOT were similar until a speed of 1400 rpm (Fig. 6.9). However, oxygenated fuels i.e. E-10, E-20 and E-85 produced about 4-7% higher power as compared to gasoline at higher speed ranges.

Effects of engine speed on exhaust emissions of $NO_x$, CO and HC at WOT are shown in Figs. 6.10, 6.11 and 6.12 respectively. $NO_x$ emissions with E-85 at WOT was lower by about 2% as compared to gasoline.

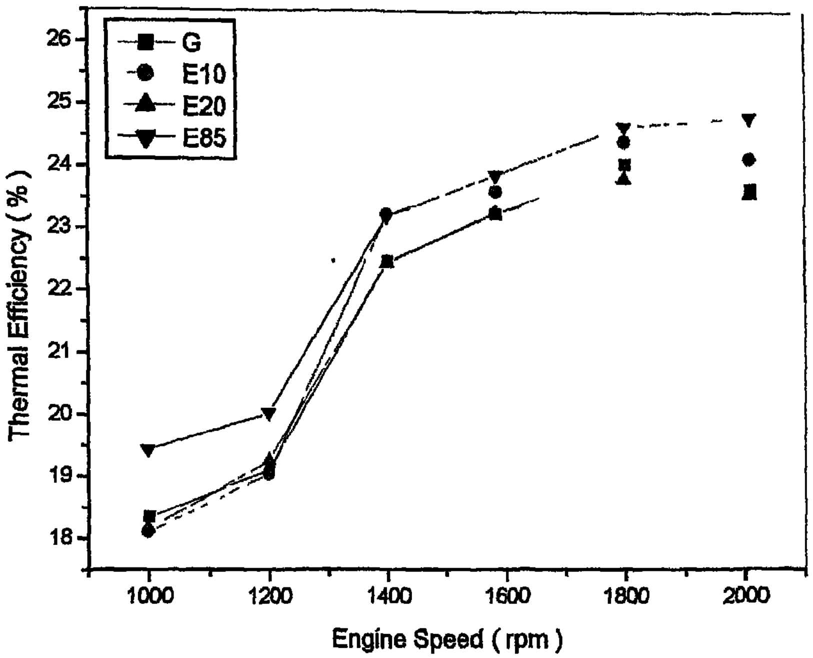

Fig. 6.7 Effect of engine speed on thermal efficency at WOT with gasoline,ethanol (E85) and ethanol - gasoline blends

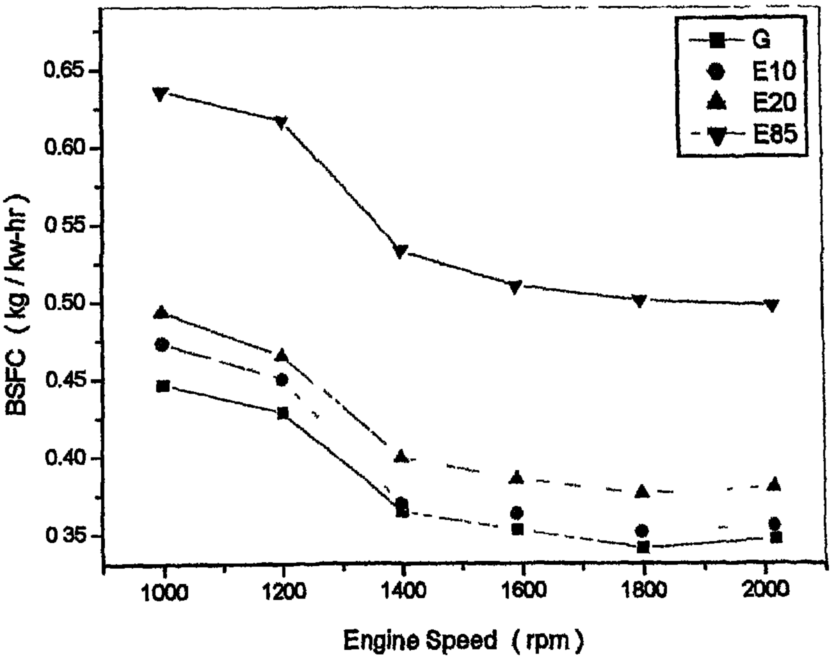

Fig. 6.8 Effect of engine speed on BSFC at WOT with gasoline, ethanol (E85) and ethanol - gasoline blends

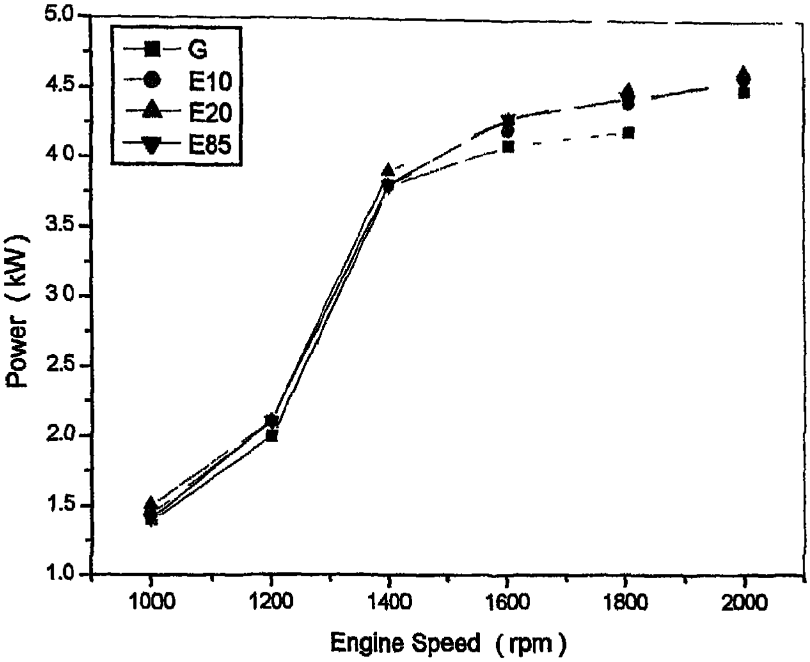

Fig. 6.9 Effect of engine speed on thermal efficency at WOT with gasoline,ethanol (E85) and ethanol - gasoline blends

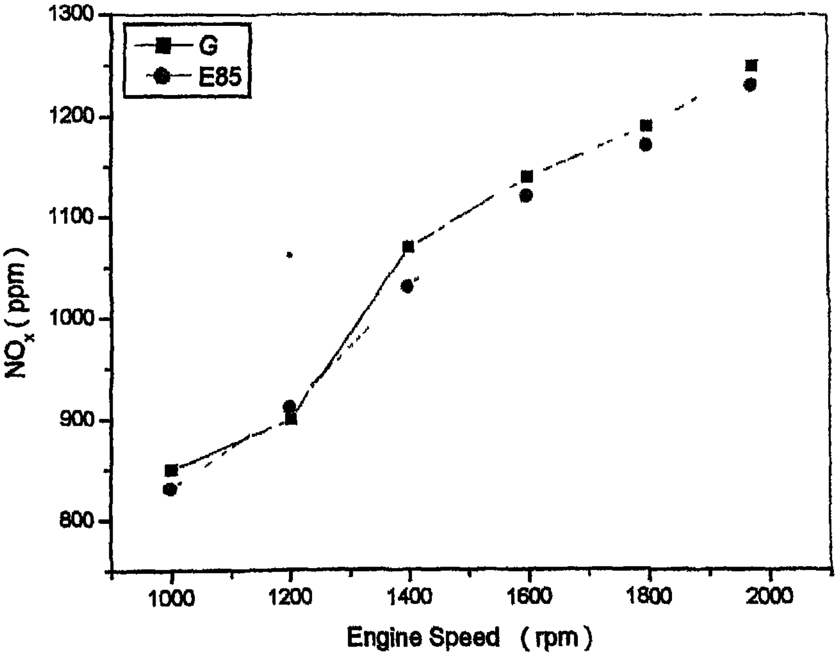

Fig. 6.10 Effect of engine speed on $NO_x$ at WOT with gasoline, ethanol (E85) and ethanol-gasoline blends

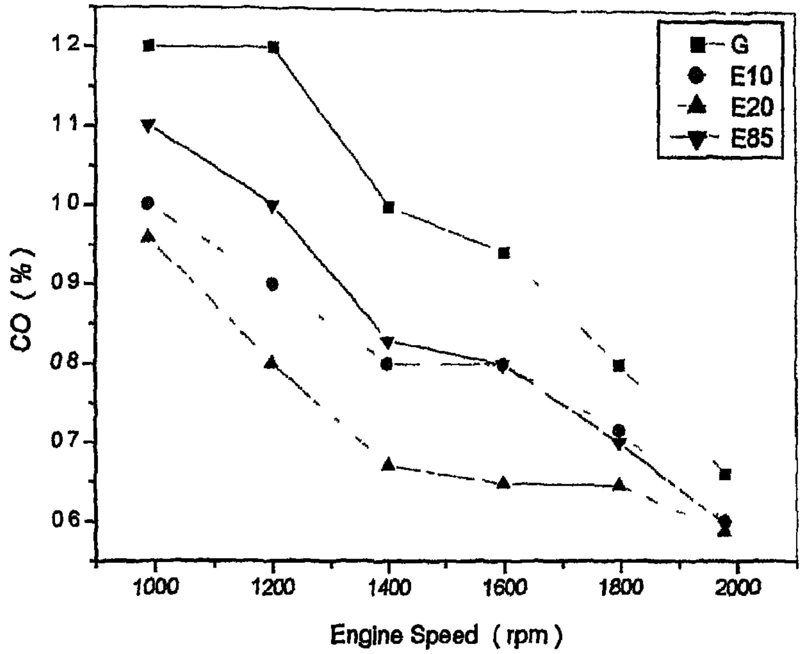

Fig. 6.11 Effect of engine speed on CO emissions at WOT with gasoline, ethanol (E85) and ethanol gasoline blends

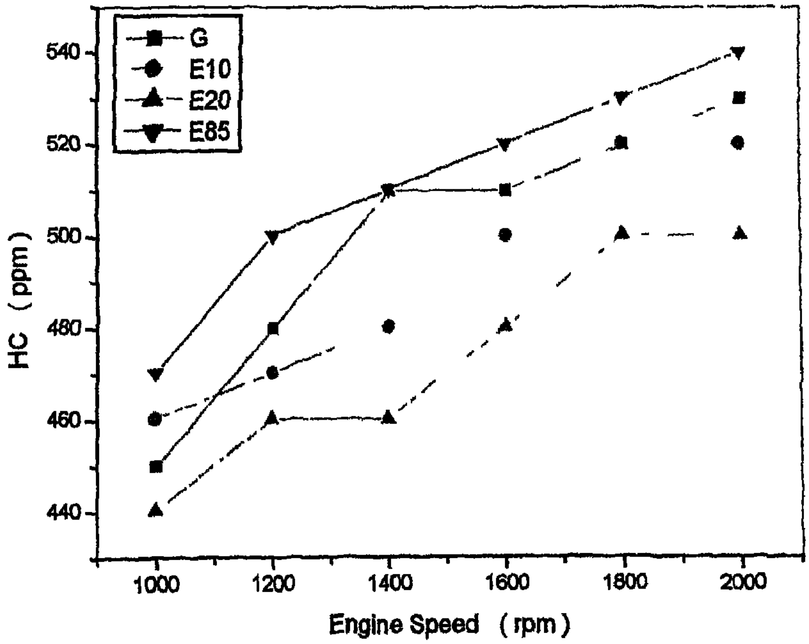

Fig. 6.12 Effect of engine speed on HC emissions at WOT with gasoline, ethanol (E85) and ethanol gasoline blends

At WOT, CO emissions were observed to be lower with ethanol (E-85) and other blends as compared to gasoline. This may be due to leaner mixture in case of E-85, E-20 and E-10. As mentioned earlier CO is strongly dependent of fuel-air ratio. However, when it was compared at same equivalence ratio, CO emissions of E-85 in rich zone were lower as compared to gasoline. Unburned hydrocarbon emissions were also lower with E-10 and E-20 but slightly higher with E-85. Mixture strength seems to be the reasons for this kind of unburned hydrocarbon emissions also. Lower hydrocarbon emissions with E-10 and E-20 may be due to lean mixtures used in these cases. When lean mixture was used less mass of fuel entered crevices and the mass of fuel absorbed by oil layer during compression was also less, which resulted in lower hydrocarbon emissions. In case of lean mixtures quenched layer thickness is also less, which will also result in lower hydrocarbon emissions.

### *6.2.3* ***Constant Speed Performance***

A comparison of engine performance at a constant speed of 1800 rev./min. is shown in Figs 6.13 to 6.18. Variations of thermal efficiency, brake specific fuel consumption, CO, HC, $NO_x$ and exhaust gas temperatures are shown against brake mean effective pressure (bmep). These tests were conducted at a constant compression ratio of 8.0 and the ignition timing was kept at 18 degrees before top dead centre for Gasoline, E-10 and E-20. For E-85 the ignition timing was kept at 14 degrees before top dead centre. The relation between brake mean effective pressure and thermal efficiency at 1800 rev./min. is shown in Fig.6.13. Higher bmep was obtained with ethanol (E-85) and also with ethanol-gasoline blends as compared to gasoline. With ethanol (E-85) and ethanol gasoline blends, brake mean effective pressures were higher by about 0.025 to 0.03 Mpa. It can be mentioned here that power output was also higher at WOT with blends especially at higher engine speeds. Highest thermal efficiency and minimum BSFC were observed at brake mean effective pressures ranging from 0.5 to 0.6 Mpa for all fuels in 1800 rev/min. The relation between brake mean effective pressure and brake specific fuel consumption at 1800 rev./min. is shown in Fig.6.14. Lowest BSFC was when engine was operated on gasoline. BSFC increased as the percentage of ethanol increased due to lower calorific value of ethanol. Figs 6.14, 6.16, 6.17 and 6.18 show the relationship between brake mean effective pressure and HC emissions, CO emissions, $NO_x$ emissions and exhaust gas temperature respectively at 1800 rev/min. Exhaust emissions of hydrocarbons and carbon monoxide were observed to decrease with increasing bmep. However, $NO_x$ emissions increased as the bmep increased because of increased exhaust gas temperature. Similar results at 1500 rev./min. are shown in Figs. 6.19 to 6.23.

### 6.2.4 ***Effect of Ignition Timing***

In order to obtain maximum power and better fuel economy the maximum pressure must be developed at a time when the piston is about to commence its downward stroke. Since there is a time lag between the occurrence of spark and the burning of the mixture, the spark must take place before the piston reaches TDC on its compression stroke. If the spark occurs too soon combustion will commence before compression stroke is completed and the pressure so developed would oppose the piston motion and thereby, reduce the engine power. If spark occurs too late, the piston will have already completed a certain part of the expansion stroke before the pressure rise occurs and corresponding amount of engine power will be lost. The correct instant for the introduction of spark is mainly determined by ignition lag and flame velocity. Some of the important factors affecting ignition lag and flame velocity are compression ratio, equivalence ratio, throttle opening, combustion chamber design, engine speed and the type of fuel.

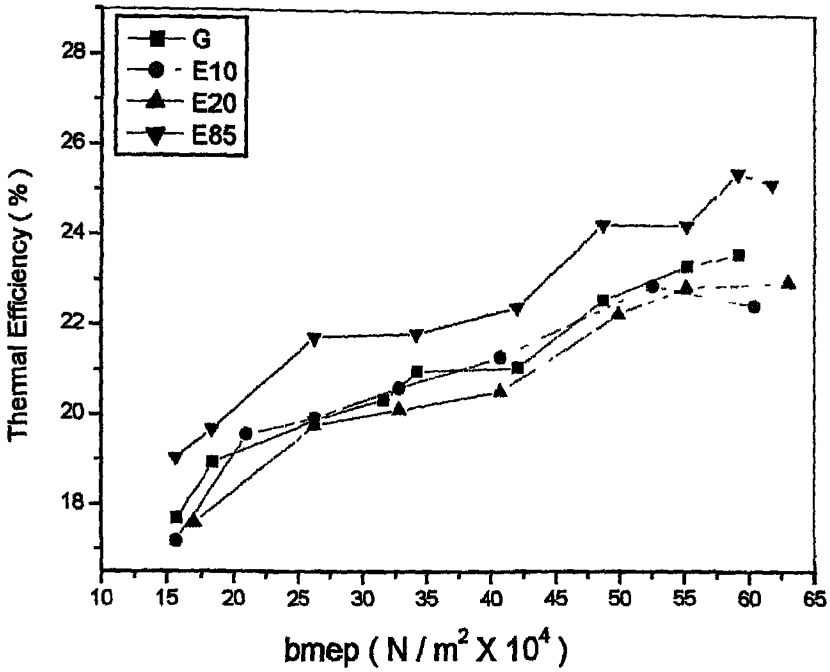

Fig 6.13 Effect of brake mean effective pressure and fuel on thermal efficiency at constant speed of 1800 rpm

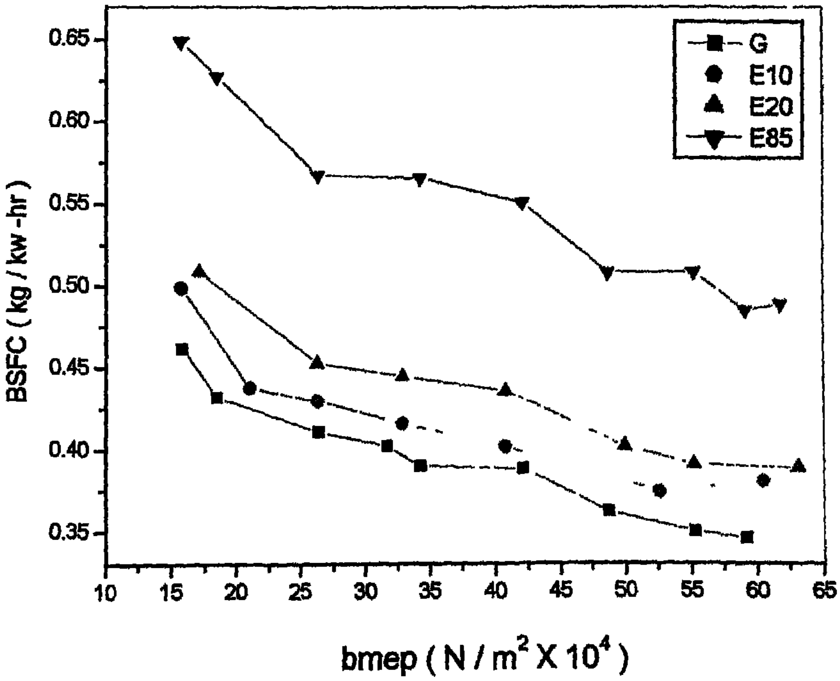

Fig. 6.14 Effect of brake mean effective pressure and fuel on BSFC at constant speed of 1800 rpm

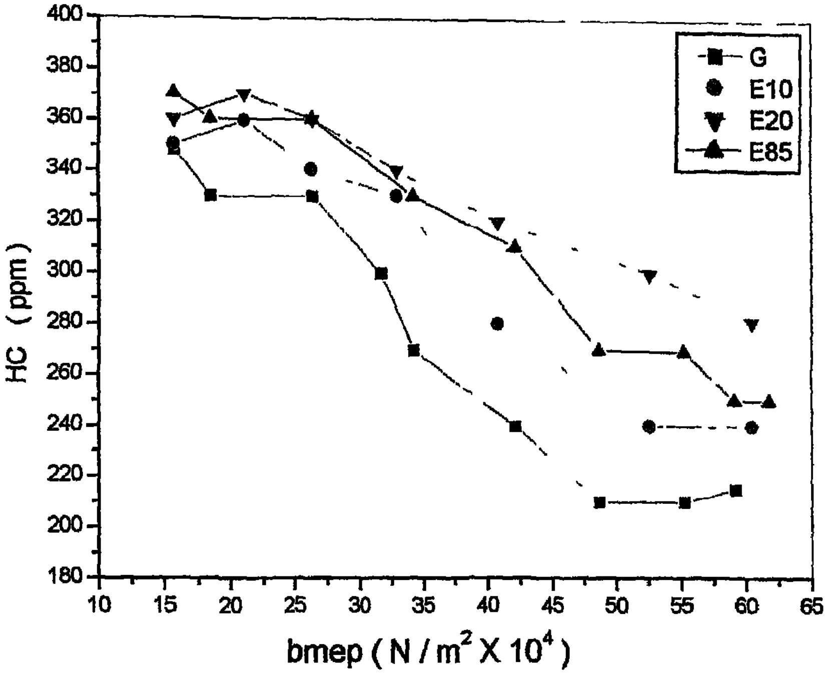

Fig. 6.15 Effect of brake mean effective pressure and fuel on HC emissions at constant speed of 1800 rpm

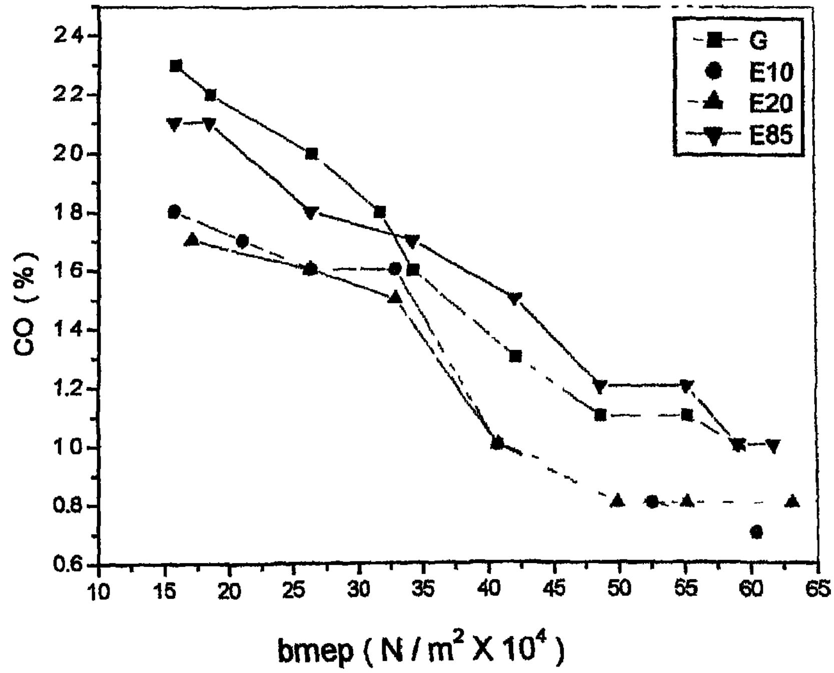

Fig.6.16 Effect of brake mean effective pressure and fuel on CO emissions at constant speed of 1800 rpm

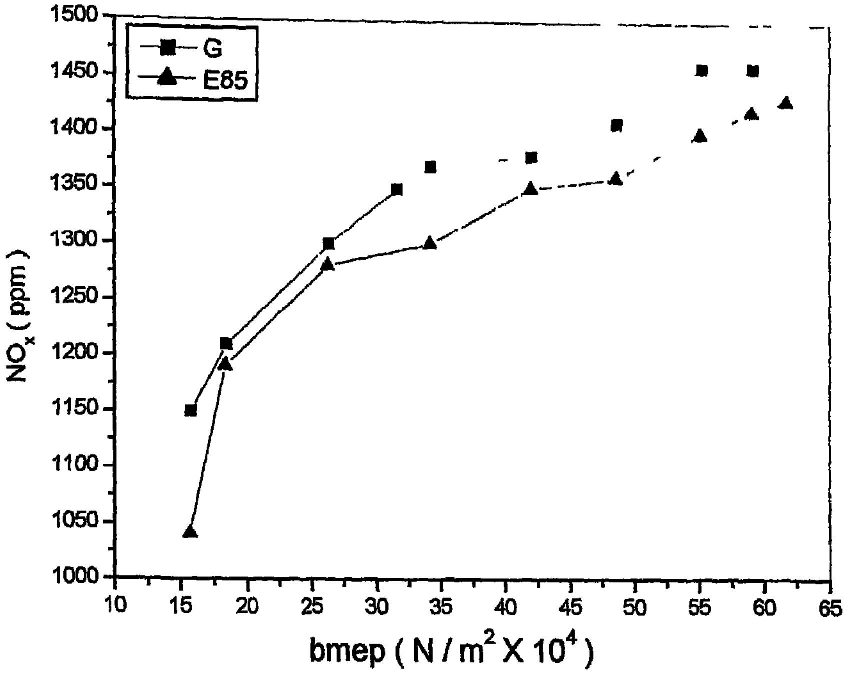

Fig. 6.17 Effect of brake mean effective pressure and fuel on NO$_x$ emissions at constant speed of 1800 rpm

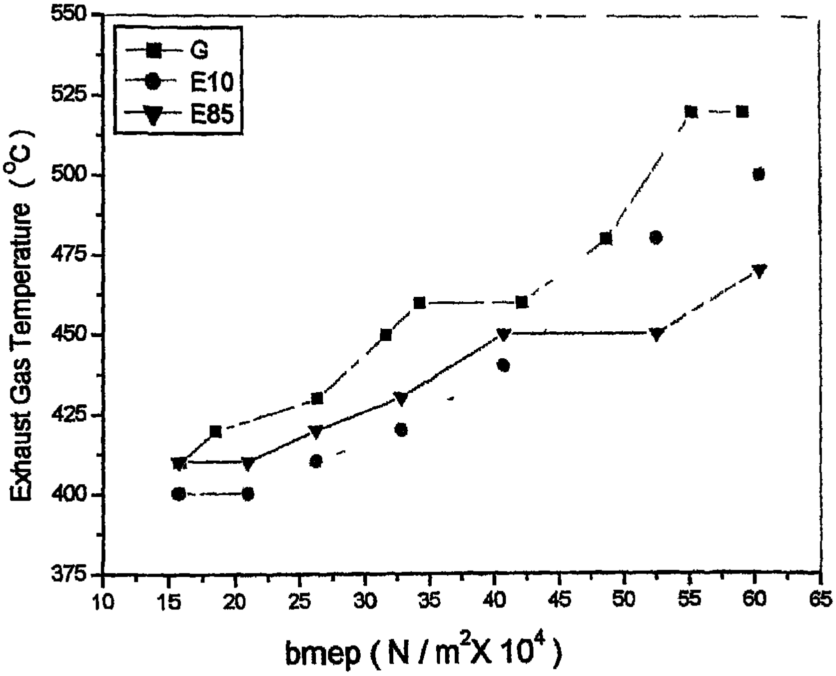

Fig. 6.18 Effect of brake mean effective pressure and fuel on exhaust temperature at constant speed of 1800 rpm

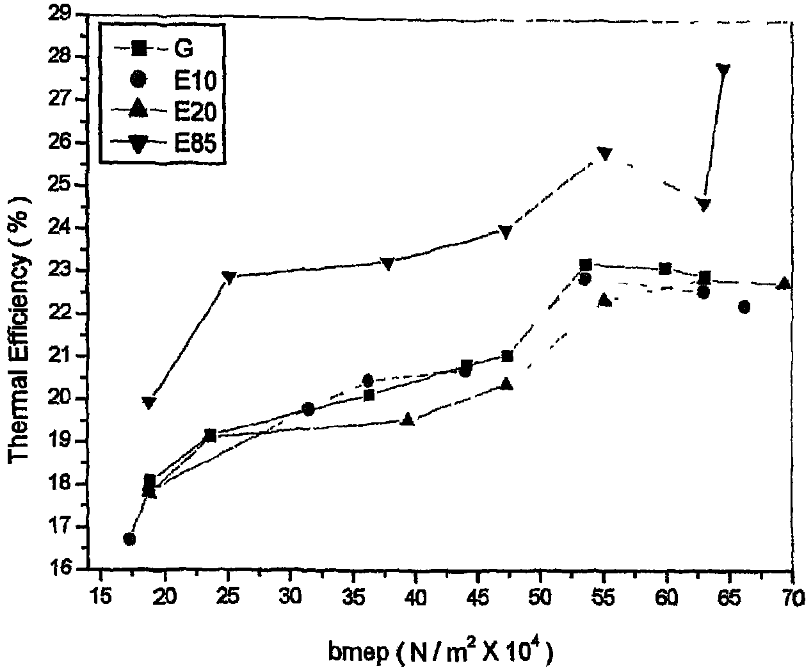

Fig. 6.19 Effect of brake mean effective pressure and fuel on thermal efficiency at constant speed of 1500 rpm

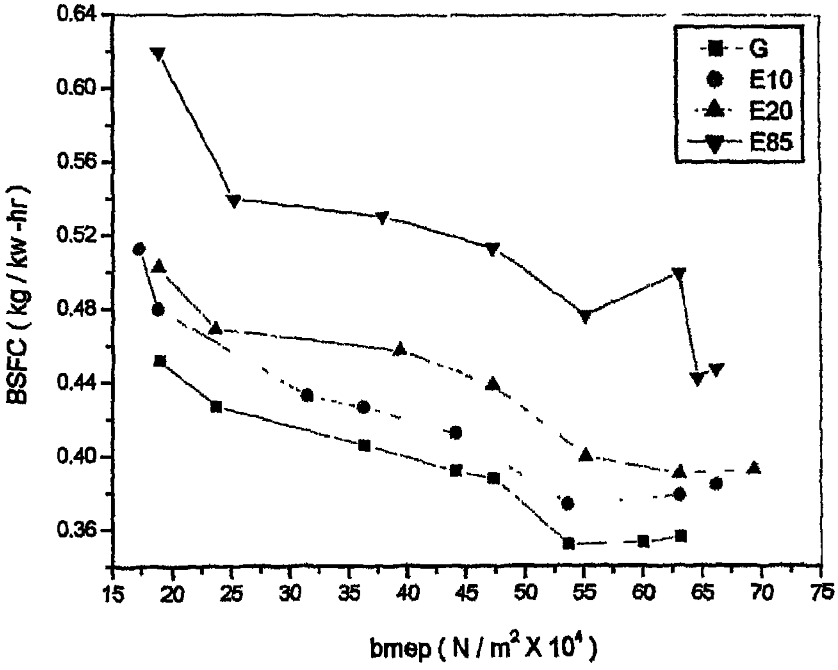

Fig. 6.20 Effect of brake mean effective pressure and fuel on BSFC at constant speed of 1500 rpm

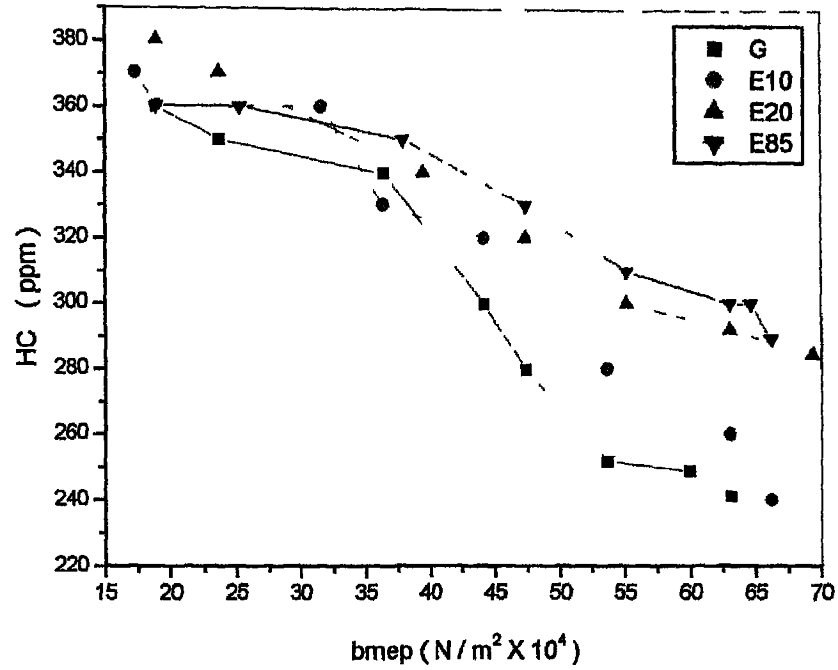

Fig. 6.21 Effect of brake mean effective pressure and fuel on HC emissions at constant speed of 1500 rpm

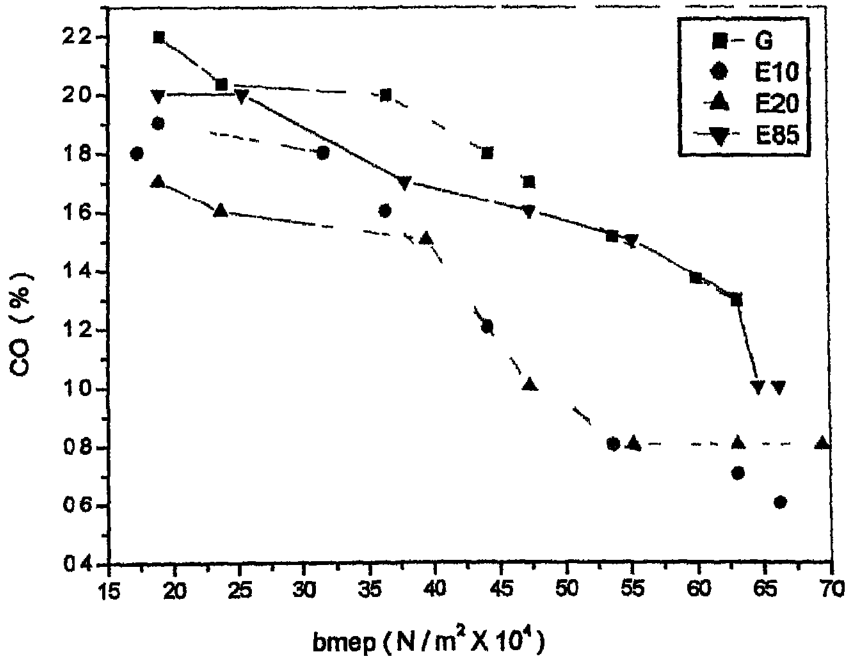

Fig. 6.22 Effect of brake mean effective pressure and fuel on CO emissions at constant speed of 1500 rpm

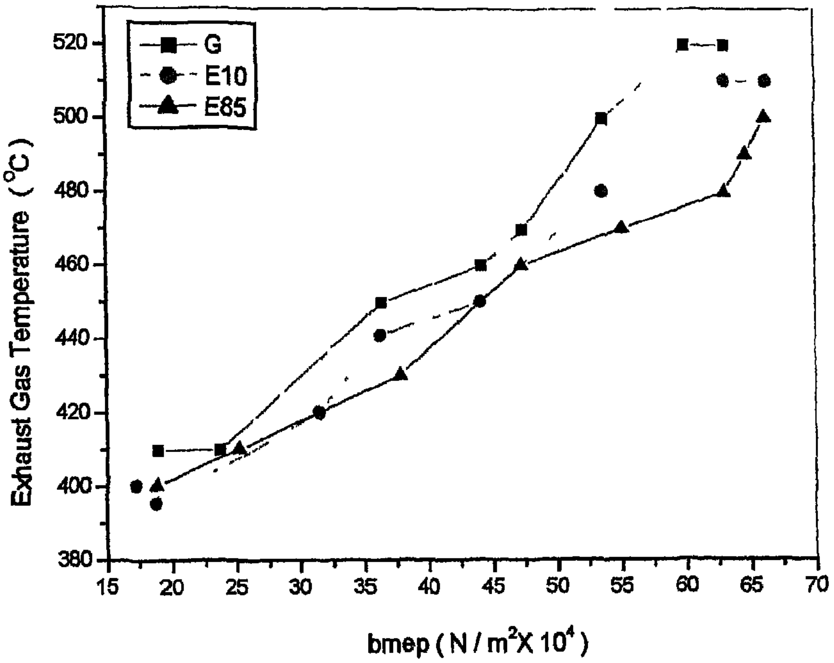

Fig. 6.23 Effect of brake mean effective pressure and fuel on exaust temperature at constant speed of 1500 rpm

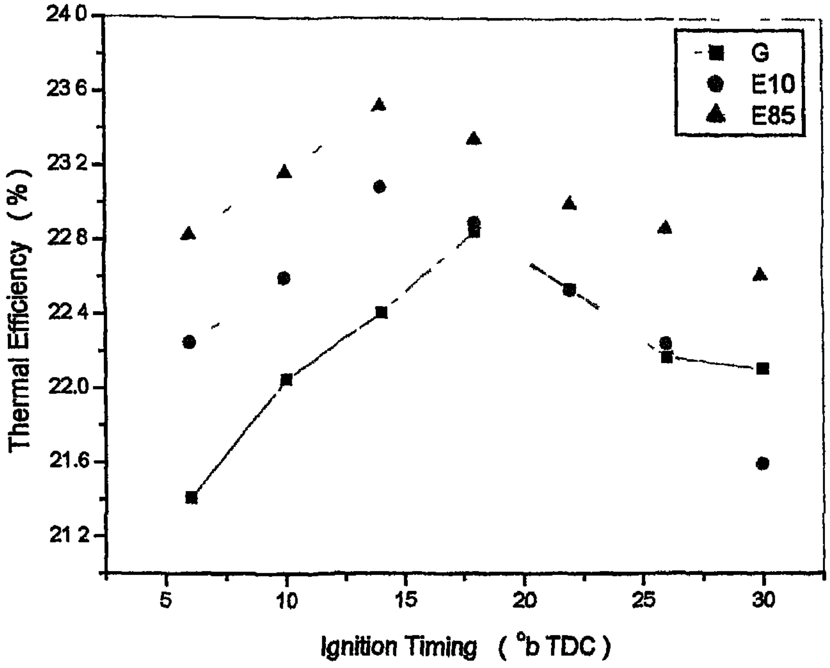

Fig. 6.24 Effect of ignition timing on thermal efficiency at 1500 rpm,1/2 throttle, CR = 8

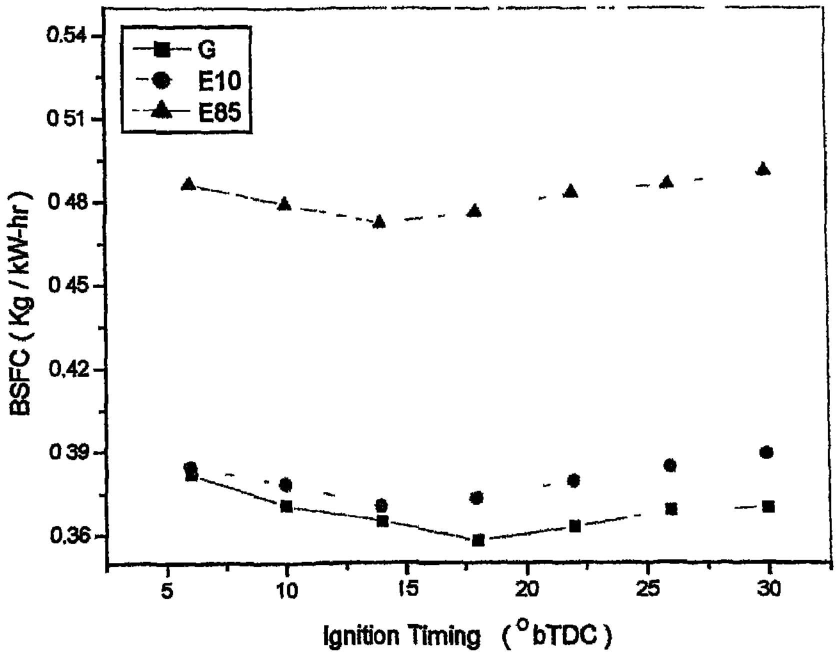

Fig. 6.25 Effect of ignition timing on BSFC at 1500 rpm, 1/2 throttle, CR = 8

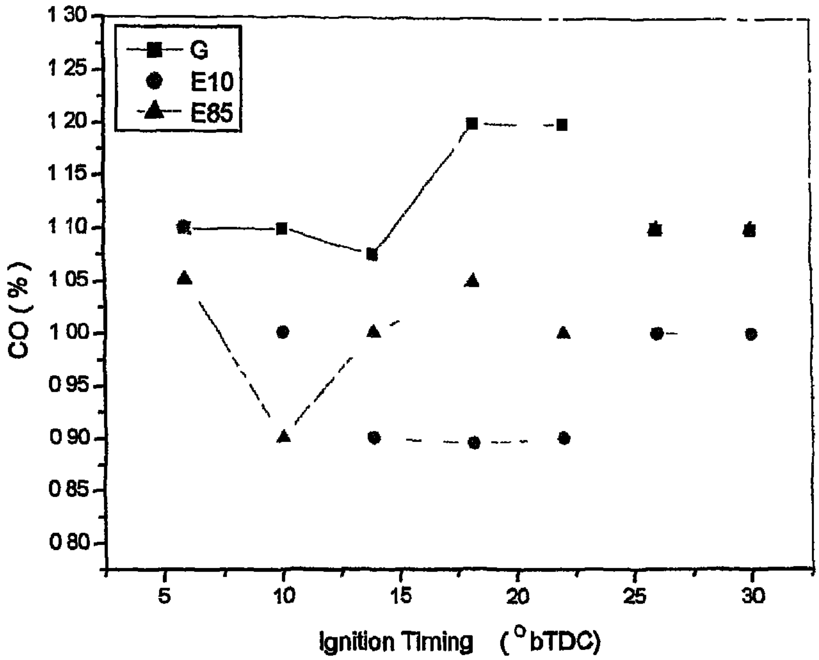

Fig. 6.26 Effect of ignition timing and fuel on CO at 1500 rpm , 1/2 throttle, CR= 8

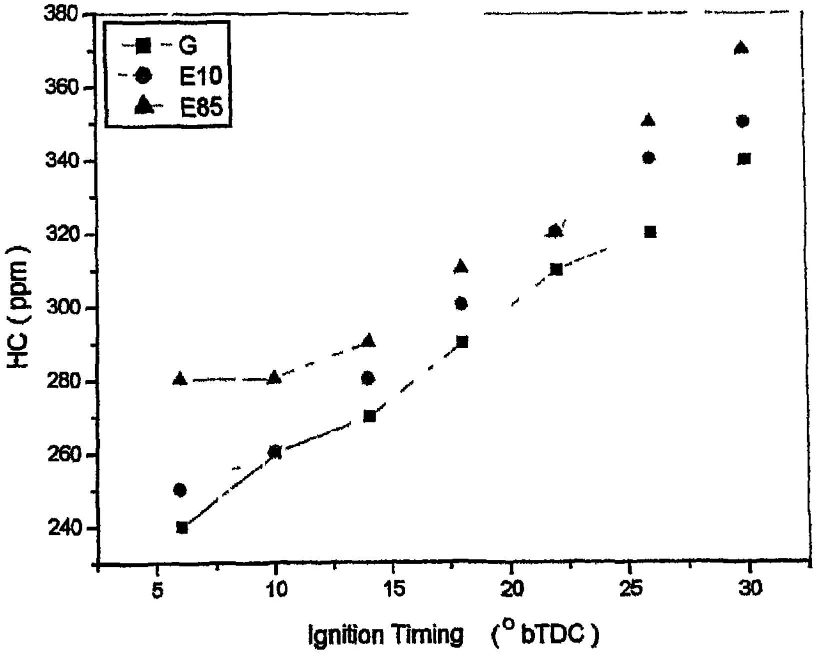

Fig. 6.27 Effect of ignition timing and fuel on HC at 1500 rpm , 1/2 throttle, CR= 8

Figs 6.24 to 6.27 show the effect of ignition timing and the type of fuel on engine power, fuel consumption and exhaust emission. Three fuels namely gasoline, E-85 and E10 were used for these experiments. Other important factors, which affect ignition lag and flame velocity were kept constant. Engine speed and compression ratio were maximum at a value of 1500 rev/min and 8.0:1 respectively, while the throttle opening was kept at 50 percent of WOT. It was found that maximum thermal efficiency for gasoline could be obtained at 18 degrees before top dead centre. While for while for E-10 and E-85 best fuel economy was obtained at 14 degrees bTDC. Exhaust emission of CO and HC are shown in Figs 6.26 and 6.27. It was found that unburned hydrocarbon emissions increased as the ignition timing was advanced. The reason for increasing HC emissions at advanced timings was attributed to lower exhaust gas temperatures at advanced timings.

### *6.2.5* ***Effect of Intake Air Temperature***

For complete vaporization of ethanol more heat is to be transferred to ethanol air mixture, as compared to gasoline air mixture. It was therefore, thought intake system design for gasoline operation might not be suitable for ethanol operation. Heating of intake air was considered to improve the mixture preparation, which was then expected to give improved engine performance. Accordingly arrangement was made to heat the intake air up to $90^0$ C and engine performance was taken at different intake air temperatures. Effect of intake air temperature on thermal efficiency and brake specific fuel consumption using E-85 at 1500 rev./min and 1800 rpm are shown in Figs 6.28 and 6.29 respectively. For both engine speeds best thermal efficiency and minimum BSFC was obtained at an intake temperature of 50 deg. C. Maximum power was obtained at the same temperature (Fig.6.30). Figs. 6.31 to 6.33 shown the variation of exhaust gas emissions of $NO_x$, CO and HC with the variation of intake air temperature. $NO_x$ increased as the intake air temperature is increased. CO was minimum at about 60 degree C intake air temperature which hydrocarbon emissions were lowest at 50 to 60 deg. C intake air temperature.

### *6.2.6* ***Effect of Compression Ratio***

A higher compression ratio increases the pressure and temperature of the working mixture and decreases the concentration of residual gases. These favourable conditions reduce ignition lag of combustion. High pressures and temperatures of the compressed mixture also speed up the second phase of combustion. Hence with higher compression ratio total combustion time is reduced. Maximum pressure and mean effective pressure are increased. Also higher compression ratio increases surface to volume ratio of the combustion chamber, thereby increasing the part of the mixture which after burns in the third phase of the combustion. To exploit the benefits of ethanol's higher octane number some tests were carried out at higher compression ratios with E-85.

Figs. 6.34 and 6.35 show the effect of compression ratio on thermal efficiency and brake specific fuel consumption using E-85. These tests were conducted when the throttle opening was full (WOT), while the compression ratio and the engine speed were kept constant at 8.0:1 and 1500 rev./min. As it can be seen in Fig. 6.34 that when compression ratio was increased from 8.0 to 10.0 and 12.0, the thermal efficiency increased by about 3 percent and 8 percent respectively, whereas calculations for an ideal thermodynamic cycle indicate that efficiency improvements of 6.6 and 11.6 percent could be expected. The observed efficiency improvements were lower than the theoretical values. The reason for this could be higher rate of heat transfer to the cylinder walls at higher compression ratios. This effect has been observed also with gasoline engines (112). Increased compression ratio decreased fuel

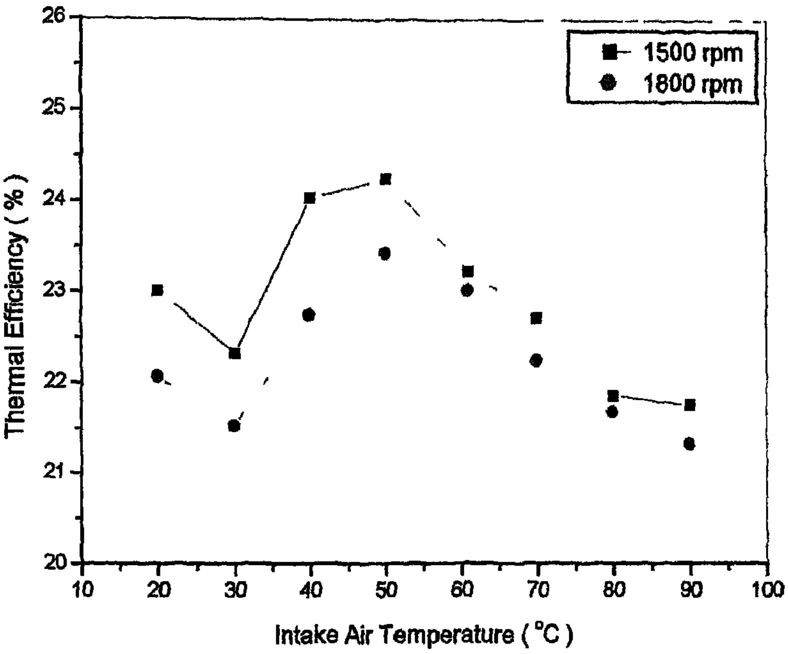

Fig. 6.28 Effect of intake air temperature on thermal efficiency using E85 with CR=8, ign. timing = 14°bTDC, throttle opening = 50%

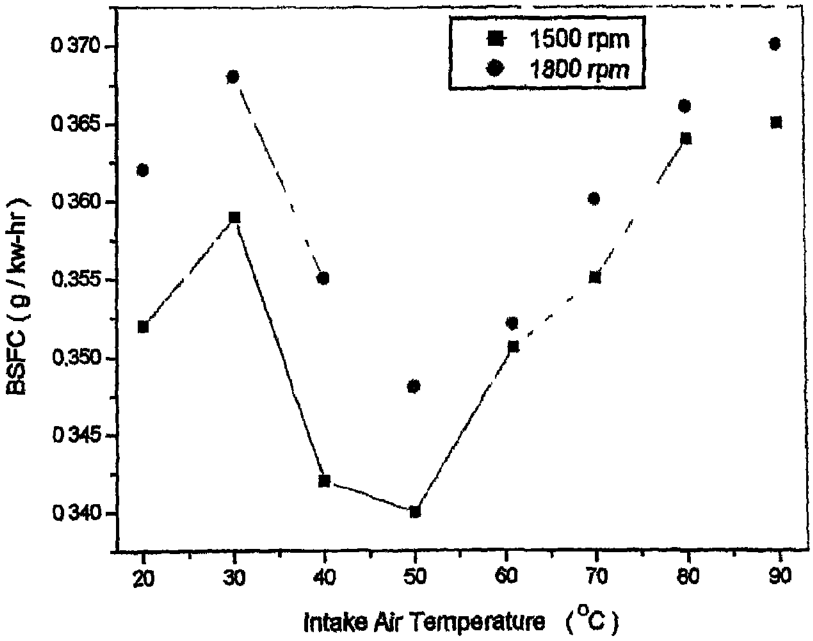

Fig. 6.29 Effect of intake air temperature on BSFC using E85 with CR=8, ign. timing = 14°bTDC, throttle opening = 50%

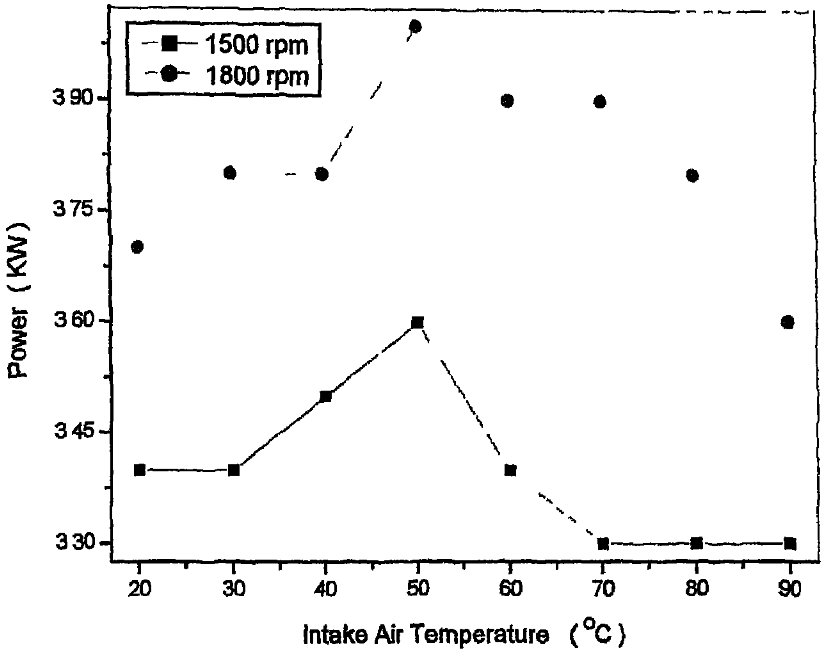

Fig. 6.30 Effect of intake air temperature on power output using E85 with CR=8, ign. timing = 14°bTDC, throttle opening = 50%

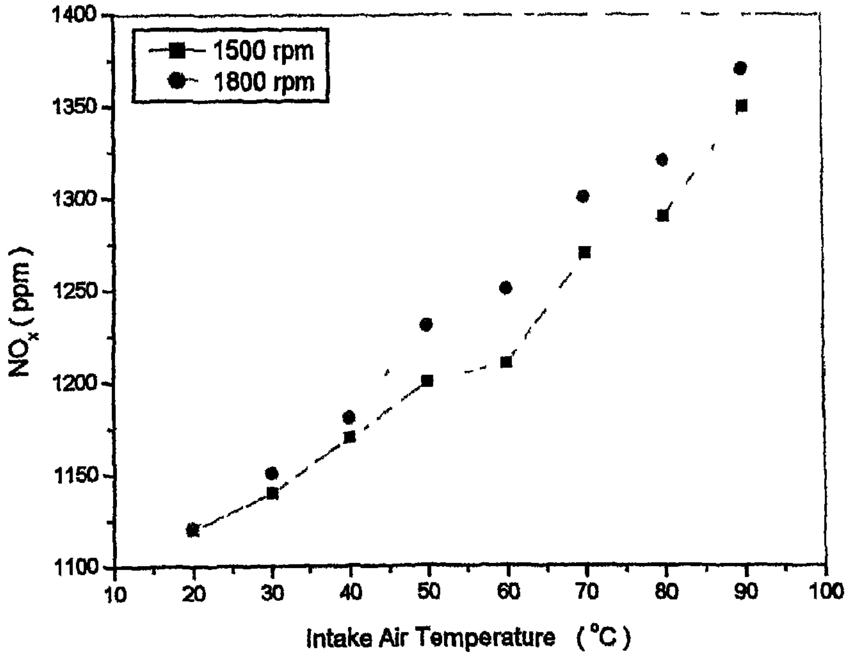

Fig. 6.31 Effect of intake air temperature on $NO_x$ emissions using E85 with CR=8, ign. timing = 14°bTDC, throttle opening = 50%

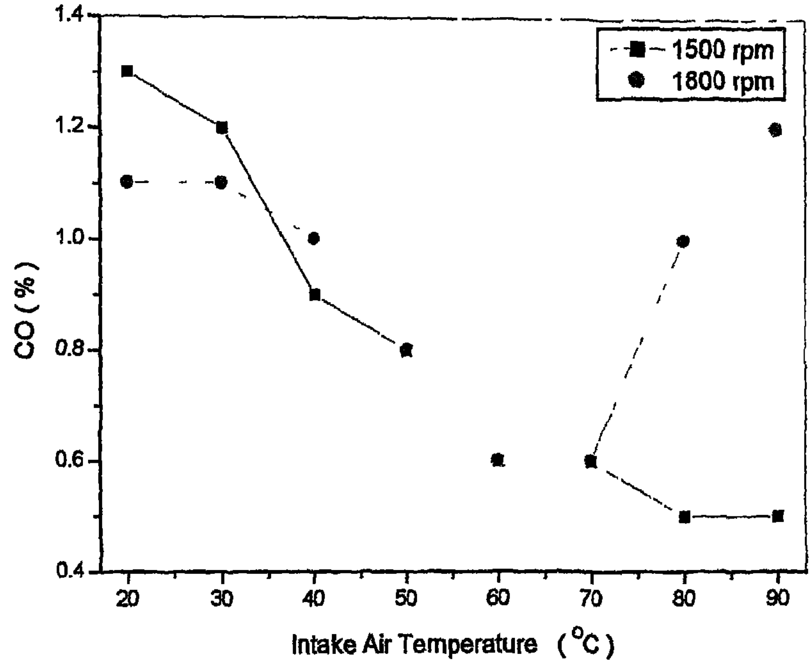

Fig. 6.32 Effect of intake air temperature on CO emissions using E85 with CR=8, ign. timing = 14°bTDC, throttle opening = 50%

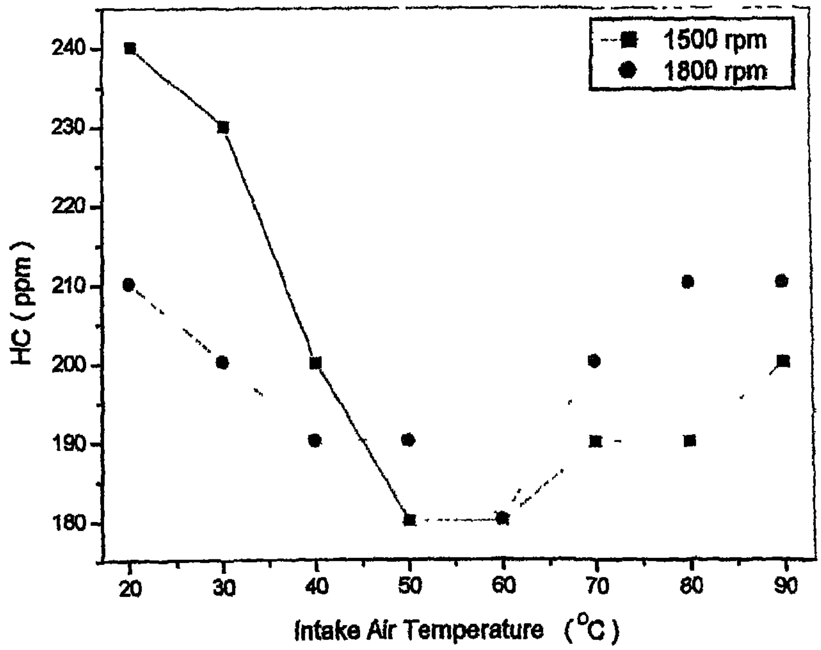

Fig. 6.33 Effect of intake air temperature on HC using E85 with CR=8, ign. timing = 14°bTDC, throttle opening = 50%

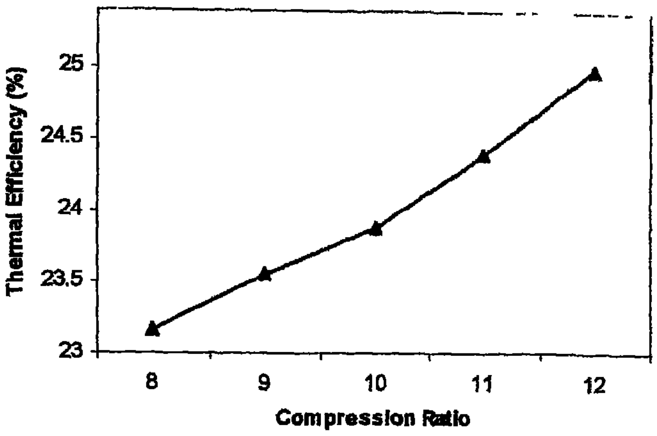

Fig. 6.34 Effect of compression ratio on thermal efficiency *using ethanol (E 85)*

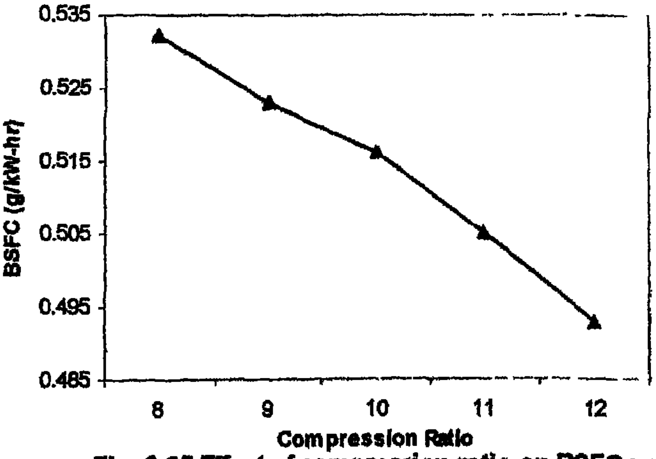

**Fig. 6.35 Effect of compression ratio on BSFC using ethanol (E 85)**

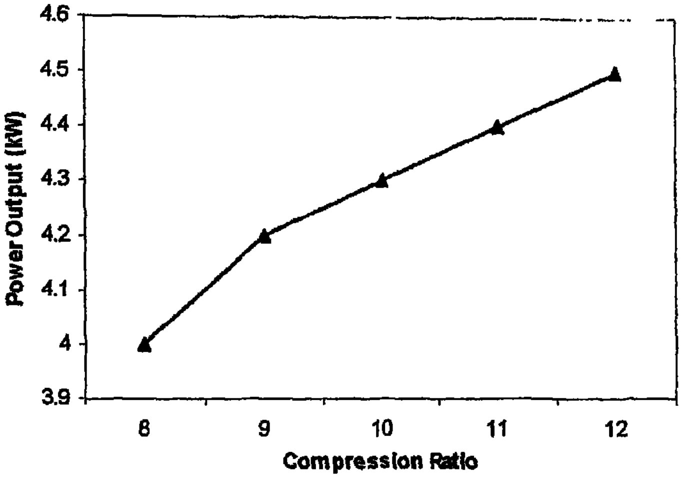

Fig. 6.36 Effect of compression ratio on power output using ethanol (E 85)

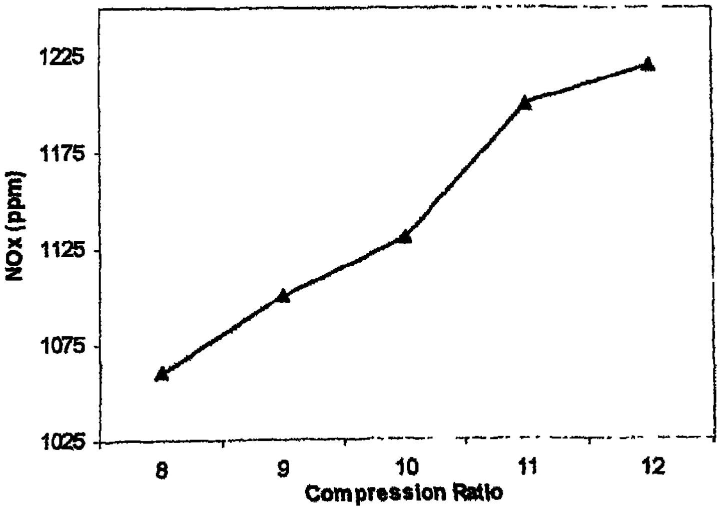

Fig. 6.37 Effect of compression ratio on exhaust emissions of $NO_X$ using ethanol (E 85)

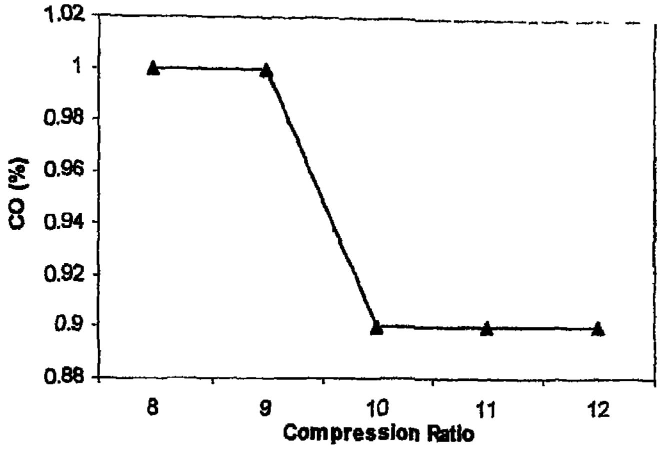

Fig. 6.38 Effect of compression ratio on exhaust emissions of CO using ethanol (E 85)

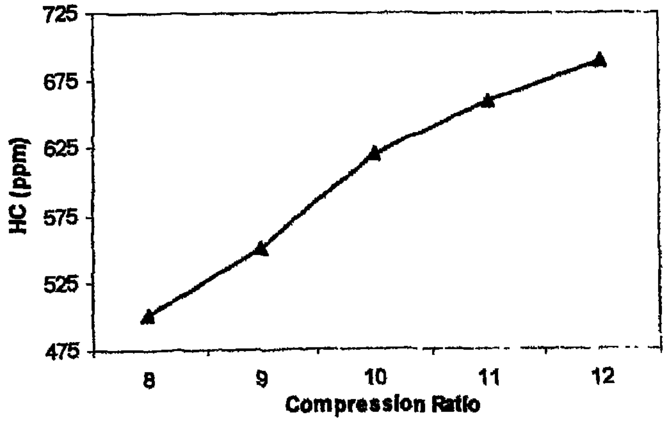

Fig. 6.39 Effect of Compression Ratio on exhaust emissions of HC using ethanol (E 85)

consumption. However, even with a compression ratio of 12:1; the fuel consumption of E-85 was higher than that of gasoline (at compression ratio of 8.0:1) by about 38 percent (Fig.6.35).

Fig. 6.36 shows the variation of engine power with variation of compression ratio. It was found that an improvement of engine power by about 12.5 percent could be achieved when compression ratio was increased from 8.0 to 12.0. However, with increased compression ratio, $NO_x$ increased due to higher temperature in the combustion chamber (Fig.6.37). Maximum $NO_x$ emission at the compression ratio of 12 was 15 percent higher than that with compression ratio of 8.0. Bechtold and Pullman (71) also found increase in $NO_x$ emissions as compression ratio increased with ethanol fuel.

Fig. 6.38 shows the variation of CO with variation of compression ratio. It was found that a reduction in CO emissions by about 10% could be achieved when compression ratio was increased from 8.0 to 12.0. Variation unburned hydrocarbon emissions with variation of compression ratio is shown in Fig. 6.39. Unburned hydrocarbon emissions increased substantially when compression ratio was increased. When compression ratio was increased from 8.0 to 12.0 UBF emissions increased by about 40%.

### 6.2.7 *Fuel Consumptions and Emissions at Idling*

During idling air supply is restricted by nearly closed throttle, and the suction pressure and exhaust temperature are very low. Also backflow during the valve overlap period occurs since exhaust pressure is higher than intake pressure. This increases amount of residual gases. During suction process the residual gases expand, thereby, reducing the fresh mixture inhaled. Increased dilution cause the combustion to be erratic or even impossible.

To get rid of erratic running, rich mixture is used during idling. The enriching of mixture increases the probability of contact between fuel and air particles which could improve the combustion process. However, enrichening the mixture strength increases fuel consumption and also CO and HC emissions. Fuel consumption and exhaust emissions may be reduced by reducing engine speed and using as lean mixture as possible. Although decreasing idling speed would improve fuel economy, too low idling speed is not possible because of unstable engine operation (113). Apart from the above factors, other factors that effect idling operations are induction system, combustion system and the type of fuel (114-116).

To assess the performance of E-85 fuel and gasoline at idling experiments were carried out on a 4-cylinder, 4-stroke, and water-cooled spark ignited engine. For carrying out these experiments engine speed was varied from 700 rpm to 1200 rev/min and ignition timing was varied from 6 degrees before top dead centre to 34 degrees before top dead centre. Fig. 6.40 shows the variations of energy consumptions with the variation of engine speed. It was observed that energy consumption increased as speed increased for both gasoline and E-85. Although energy consumption was lower at lower engine speeds, but stable engine operation was obtained at about 900 rev/min with both gasoline and E-85. Engine idling speeds below 900 rev/min, speed fluctuation was very high (of the order of 50 rev/min) with both E-85 and gasoline. It was found that at similar engine idling speeds energy consumption with gasoline was lower as compared to E-85. Better energy consumption with gasoline as compared to E-85 was observed to be 4 percent at 700 rpm and 2 percent at 1200 rpm. As idling speed was reduced improvement in energy consumption with gasoline was more as compared to E-85. This could be due to better mixture formation by gasoline-air mixture than ethanol-air mixture. The boiling point of ethanol is 78 deg. C, which is very high as compared to gasoline's initial boiling point of 30 deg. C (Table 2.1). Since the intake at idling is low, so an inferior mixture is expected with ethanol as compared to gasoline.

The variation of fuel consumption with ignition timing at idling is shown in Fig. 6.41. Lowest fuel consumption with gasoline was achieved at about 14 degrees before top dead centre. Fuel consumption increased as ignition timing was advanced beyond 14 degrees before top dead centre. This could be due to the fact that as ignition timing was advanced the temperature and the pressure of mixture was less at the time of spark which led to burning of less homogeneous mixture. Minimum fuel consumption with ethanol (E-85) was achieved at 10 degrees before top dead centre. Energy consumption with ethanol was higher by 2.5 to 3 percent throughout the range of ignition timing. Although the energy consumption with ethanol (E-85) was higher as compared to gasoline at idling but single cylinder results for thermal efficiency at constant equivalence ratio (Fig-6.1) showed 2-3 percent better thermal efficiency with E-85 as compared to gasoline. In multi-cylinder engine higher energy consumption may be due to formation of less homogeneous mixture with ethanol (E-85) as compared to pure gasoline.

The variations of carbon monoxide and hydrocarbon emissions at different engine speeds are shown in Figs. 6.42 and 6.44 respectively. It was observed that both CO and HC emissions increased as idling speed was reduced and also CO and HCs were higher with ethanol (E-85) as compared to gasoline. Higher CO and HC emissions at lower idling speeds could be due higher back pressures and higher residual gas fractions at lower speeds, which resulted in higher CO and HC emissions. Due to inferior mixture formation with ethanol (E-85) at idling, CO and HC emissions were also higher with E-85 as compared to gasoline. The variations of CO and HC emissions with variations of ignition timing at idling are shown in Fig. 6.43 and 6.45 respectively. With E-85 exhaust emissions of CO and HC were observed to be lowest at 10 degrees before top dead centre. CO and HC emissions increased as ignition timing was advanced beyond 10 degrees before top dead centre. This could be due to the fact that when ignition timing was advanced the temperature and the pressure of the air-fuel mixture was low, which led to lower flame velocity.

### 6.2.8 ***Performance of Catalytic Converter with E-85 and E-10***

Currently, the most important after treatment device is a there-way catalyst (TWC) first installed on the exhaust systems in passenger cars in 1975 (117). It derives its name from the fact that it works on all the three gaseous pollutants of concern: Carbon monoxide, hydrocarbons and oxides of nitrogen. The operation of the TWC is severely affected by the presence of lead and sulphur in conventional gasoline and normally uses three noble metals, platinum (pt), palladium (pd) and rhodium (Rh). Rhodium is the principal metal used for reduction of $NO_x$ to nitrogen ($N_2$) and oxygen ($O_2$). While platinum is the principal metal used to oxidize CO and HC to $CO_2$ and water vapour ($H_2O$). In gasoline engines highest conversion efficiency for all the three gases are achieved if the exhaust gas composition corresponds to nearly (± 1%) stoichiometric combustion. To evaluate the performance of the three way catalytic converter with ethanol, exhaust emissions of CO, HC and $NO_x$ were measured using a three-way catalyst on the exhaust system of a single cylinder engine. This catalytic converter is designed for an 800cc spark ignited gasoline engine. The fuel used for these experiments were E-85 and E-10. These

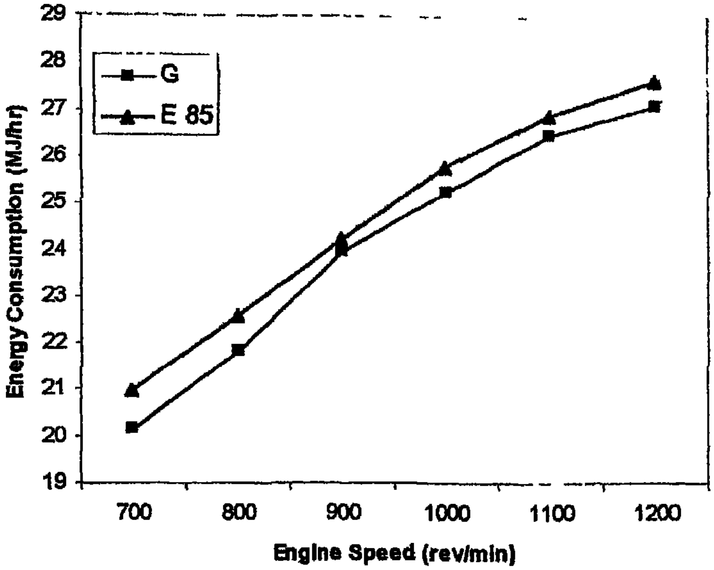

**Fig. 6.40 Effect of engine speed on energy consumption at idling : Engine B**

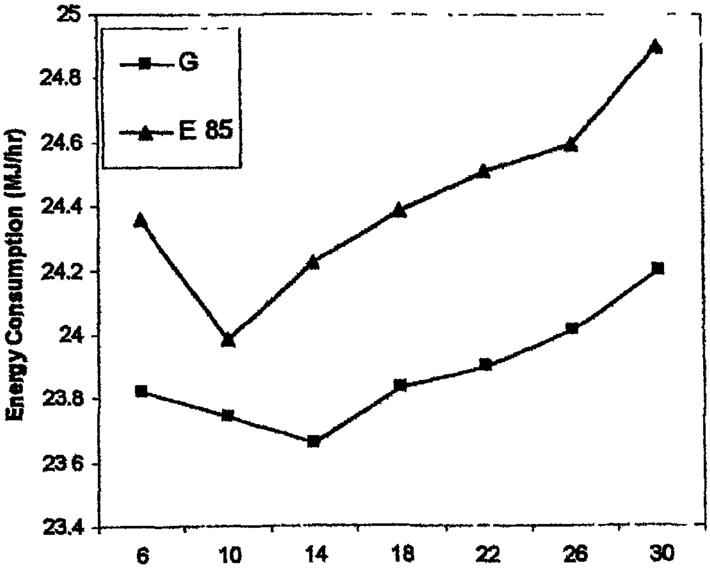

**Fig. 6.41 Effect of ignition timing on energy consumption at idling : Engine B**

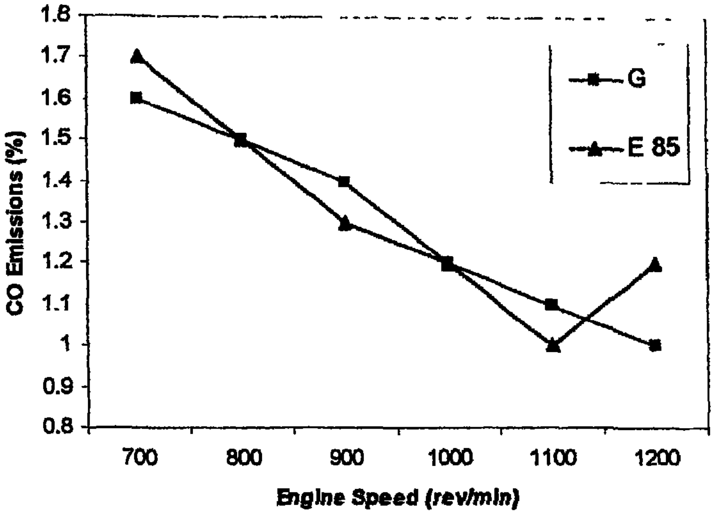

Fig. 6.42 Effect of engine speed on CO emissions at idling : Engine B

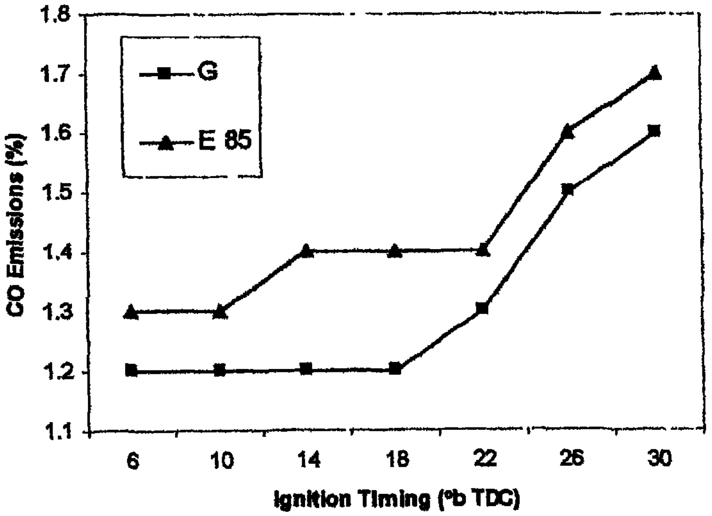

Fig. 6.43 Effect of ignition timing on CO emissions at idling : Engine B

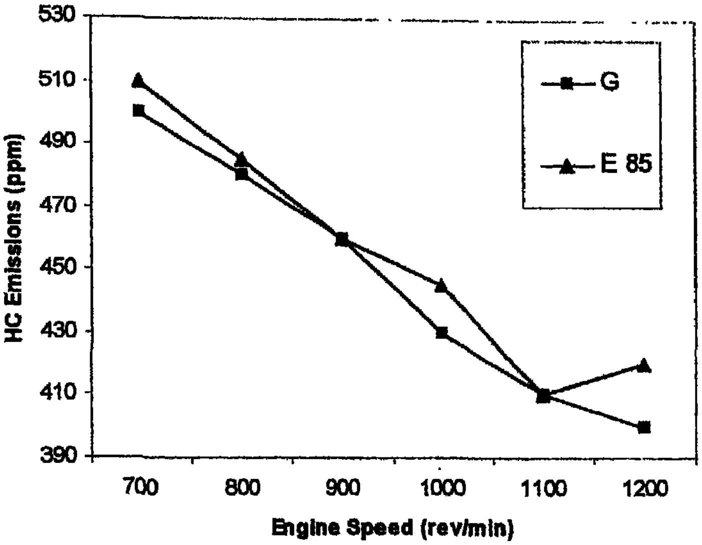

Fig. 6.44 Effect of engine speed on HC emissions at idling : Engine B

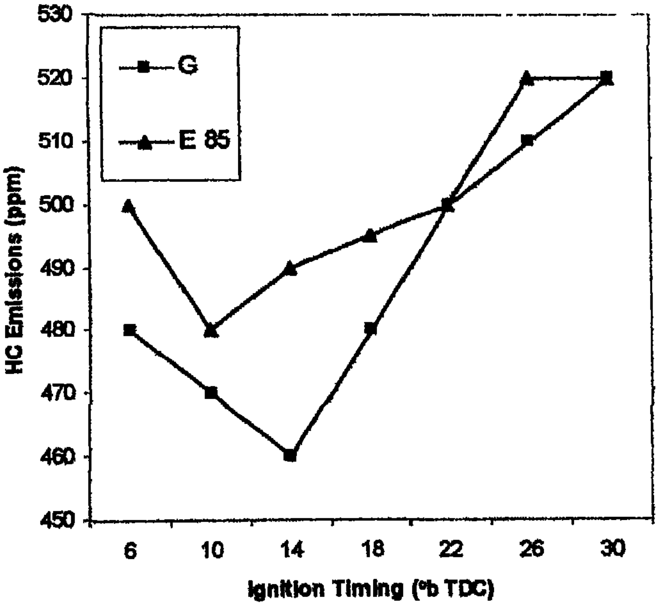

Fig. 6.45 Effect of ignition timing on HC emissions at idling : Engine B

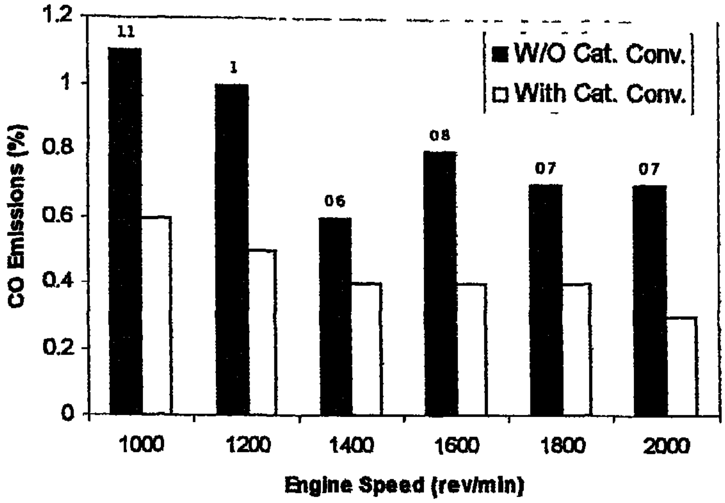

Fig. 6.46 Comparison of CO emissions with and without catalytic converter, Fuel: Ethanol (E 85)

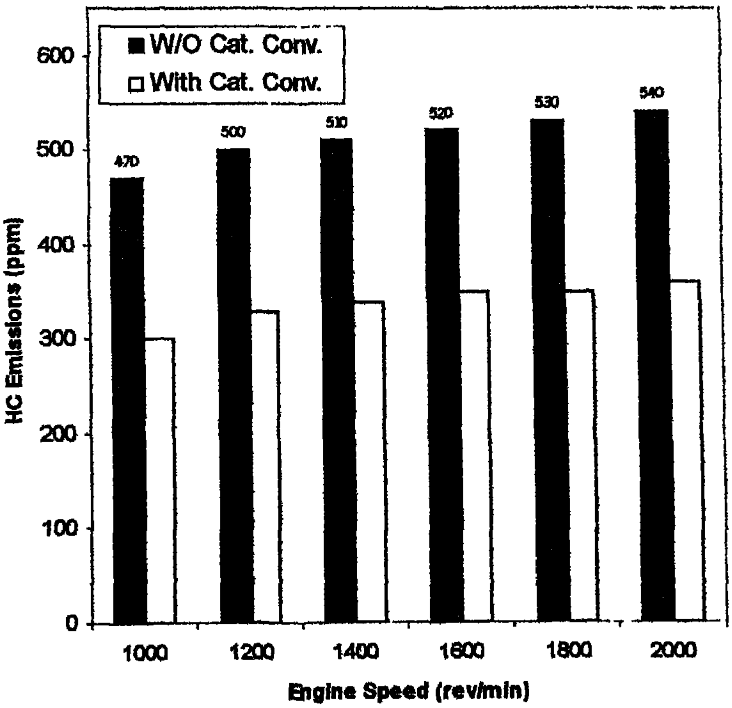

Fig. 6.47 Comparison of HC emissions with and without catalytic converter, Fuel: Ethanol (E 85)

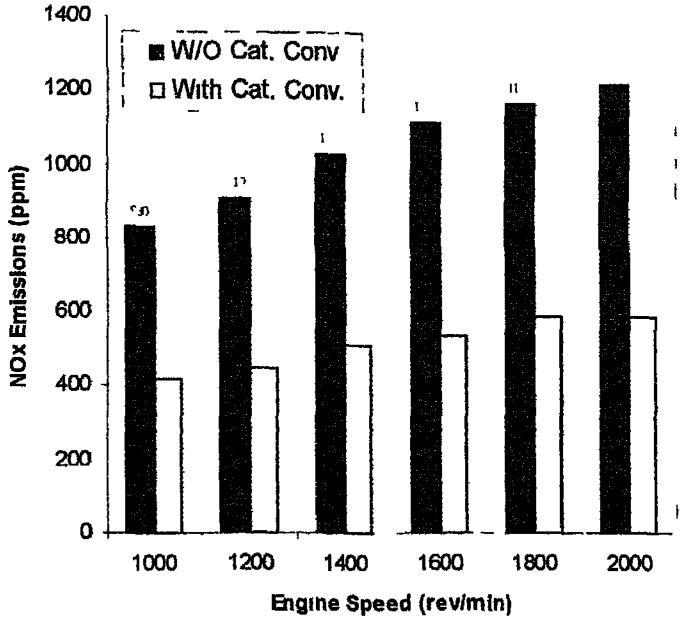

Fig. 6.48 Comparison of NOx emissions with and without catalytic converter, Fuel: Ethanol (E 85)

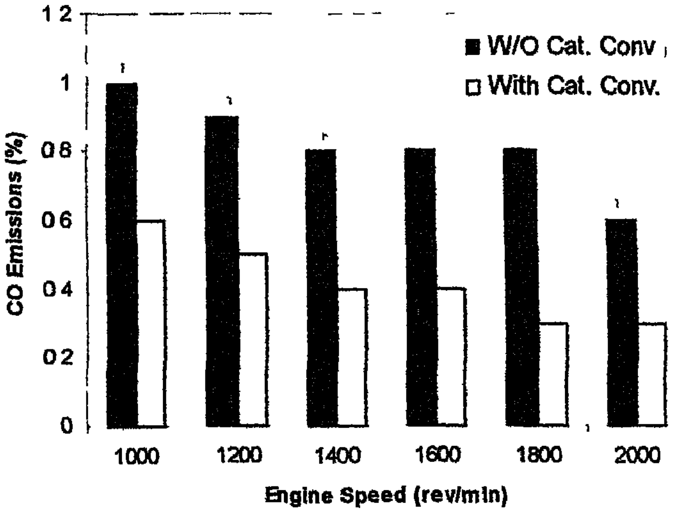

Fig. 6.49 Comparison of CO emissions with and without catalytic converter, Fuel: Ethanol (E10)

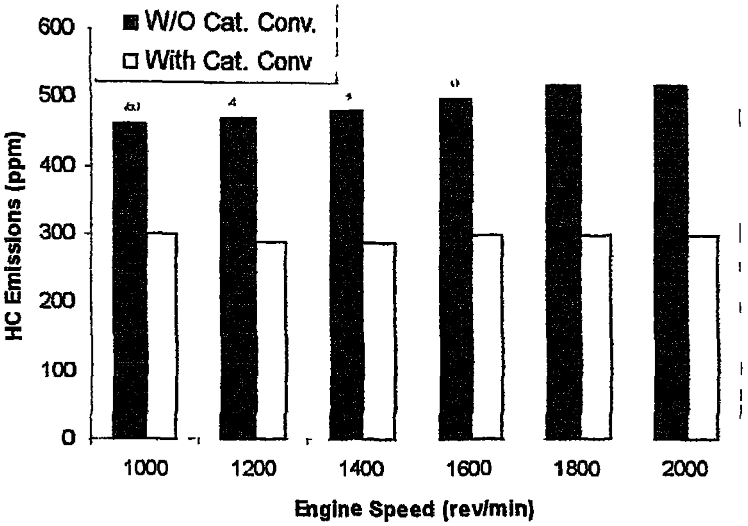

Fig. 6.50 Comparison of HC emissions with and without catalytic converter, Fuel: Ethanol (E 10)

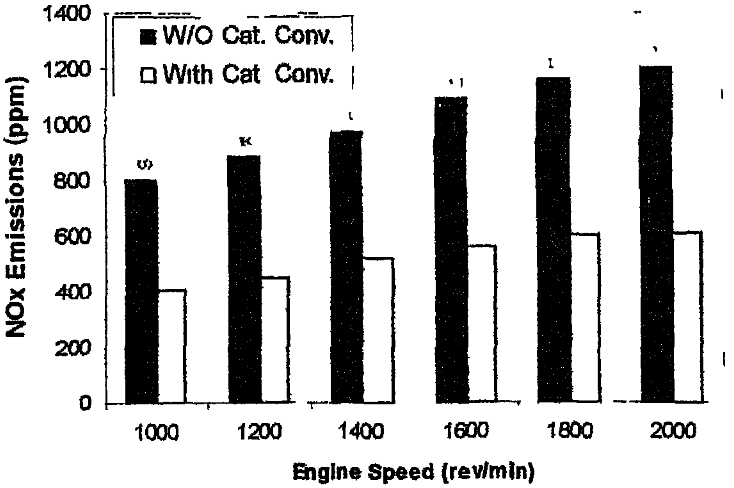

Fig. 6.51 Comparison of NOx emissions with and without catalytic converter, Fuel: Ethanol (E 10)

tests were conducted when the throttle opening was full (WOT), and the compression ratio and the equivalence ratio were kept constant at 8.0:1 and 1.0.

A comparison of exhaust emissions of CO, HC and $NO_x$ using E-85 with and without TWC is shown in Figs 6.46, 6.47 and 6.48 respectively. These tests were carried out at WOT and the compression ratio was kept at 8.0. It was observed that with the use of this three-way catalyst, CO, HC and $NO_x$ could be reduced by 46 percent, 34 percent and 50 percent respectively. The highest CO reduction of 57 percent was achieved at an engine speed of 2000 rpm. With the same catalytic converter slightly higher conversion efficiency resulted using E-10. In case of E-10, the reduction of CO, HC and $NO_x$ were 49 percent, 39 percent and 48 percent respectively. This catalytic converter also gave similar conversion efficiency with gasoline fuel. From these results it can be concluded that the existing 3 way catalytic converters being used in gasoline engines could also be to reduce CO, HC and $NO_x$ when ethanol is added to gasoline.

## 6.3 ***PERFORMANCE OF CNG – COMPARISON WITH GASOLINE***

In this section, the results obtained with compressed natural gas for the variations of equivalence ratio at part load and with the variations of engine speed at WOT on a single cylinder engine are discussed. The results obtained with compressed natural gas are also compared with gasoline. The flame velocity of methane is lower than that of gasoline. This may require longer time for combustion to complete when compressed natural gas is used, which may result in lower power and thermal efficiency. Use of multipoint spark plug may improve the combustion rate to enhance the combustion process. Hence it was also decided to evaluate the performance of CNG using two spark plugs.

### 6.3.1 *Thermal Efficiency and Fuel Consumption*

As discussed earlier the fuel utilization by engine can be represented by both brake specific fuel consumption on mass basis and brake thermal efficiency. Fig. 6.52 shows the variations of brake thermal efficiency with of equivalence ratio for gasoline and CNG operations. Thermal efficiency was maximum at the equivalence ratio of 1.0 for both CNG and gasoline. With CNG thermal efficiency was 3-5 percent greater than that of gasoline throughout the range of air-fuel ratios. Fuel consumption on mass basis was also lower with CNG by about 3 percent (Fig.6.53). Lower fuel consumption on mass basis with CNG as compared to gasoline was attributed to greater heating value of methane as compared to gasoline (Lower heating value of methane is about 12 percent more than gasoline). Similar results were also obtained by Evens and Blaszeyzyk (56). 3 to 5 percent greater thermal efficiency with CNG as compared to gasoline was attributed to better mixing of methane-air mixture as compared to gasoline-air mixture. Heenan (82) observed about 7.5 to 10 percent improvement in thermal efficiency in a multi cylinder carbureted engine.

### 6.3.2 *Exhaust Emissions*

Exhaust emissions of total hydrocarbon, carbon monoxide and oxides of nitrogen with CNG and gasoline at similar mixture strengths (similar equivalence ratios) are compared in Figs. 6.54, 6.55 and 6.56 respectively. The trends in the variations of total hydrocarbon emissions versus equivalence ratio were conventional and minimum hydrocarbon emissions occurring between equivalence ratios of 0.9 to 1.0. Hydrocarbon emissions with CNG were found to be 14 to 20 percent lower in lean zone (at equivalence ratios 0.65 and 0.85) as compared to gasoline. While in the richer side ($\phi$>1.0) hydrocarbon emissions were almost similar for gasoline and CNG. Lower level of hydrocarbon

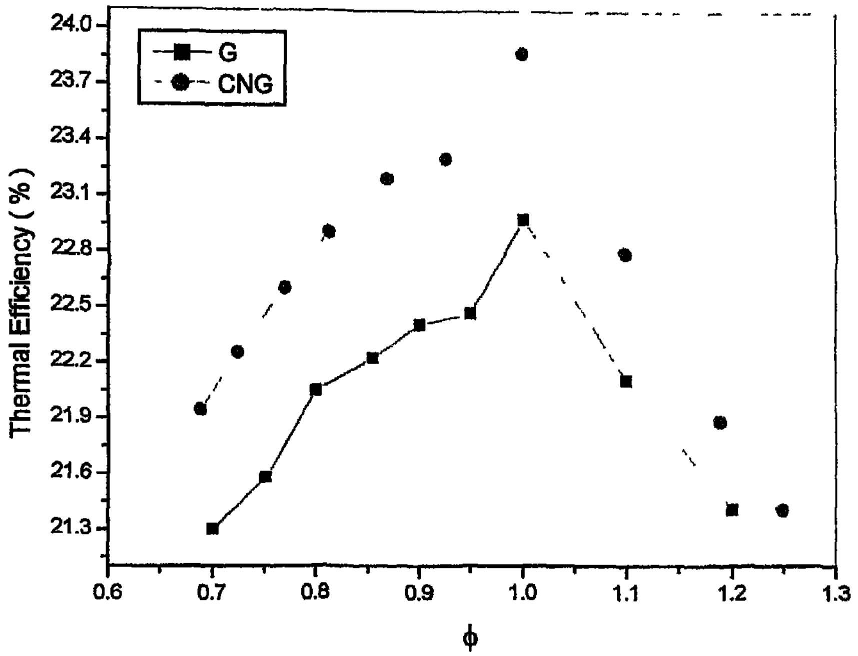

Fig. 6.52 Effect of equivalence ratio and fuel on thermal efficiency at CR=8,1500 rpm, 1/2 throttle

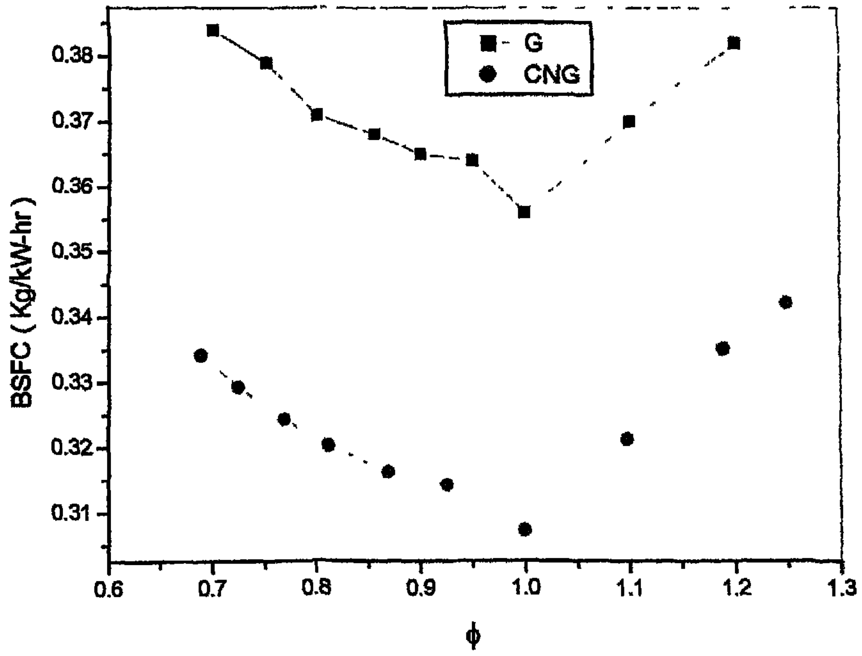

Fig. 6.53 Effect of equivalence ratio and fuel on BSFC efficiency at CR=8,1500 rpm, 1/2 throttle

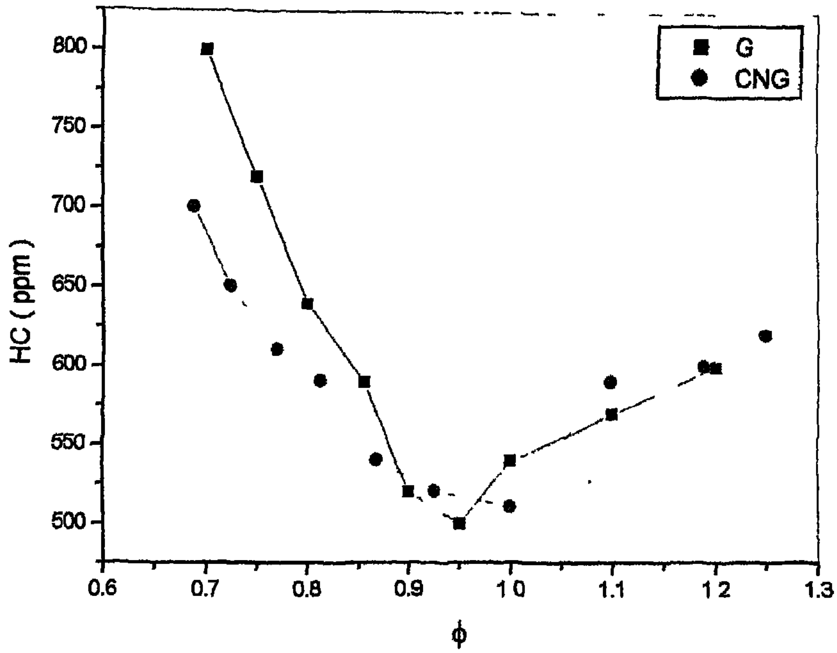

Fig. 6.54 Effect of ϕ and fuel on HC emissions at CR=8, 1500 rpm, 1/2 throttle

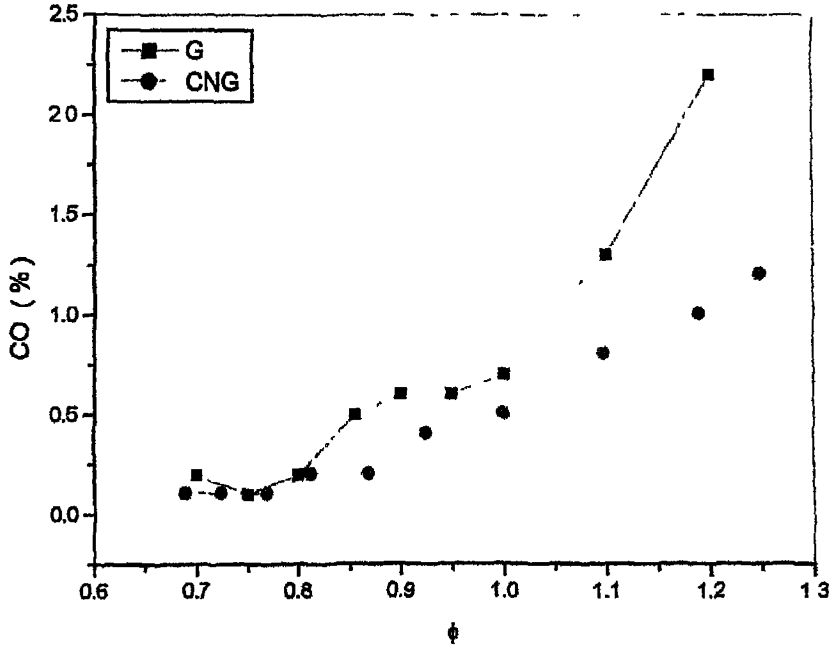

Fig. 6.55 Effect of ϕ and fuel on CO emissions at CR=8, 1500 rpm, 1/2 throttle

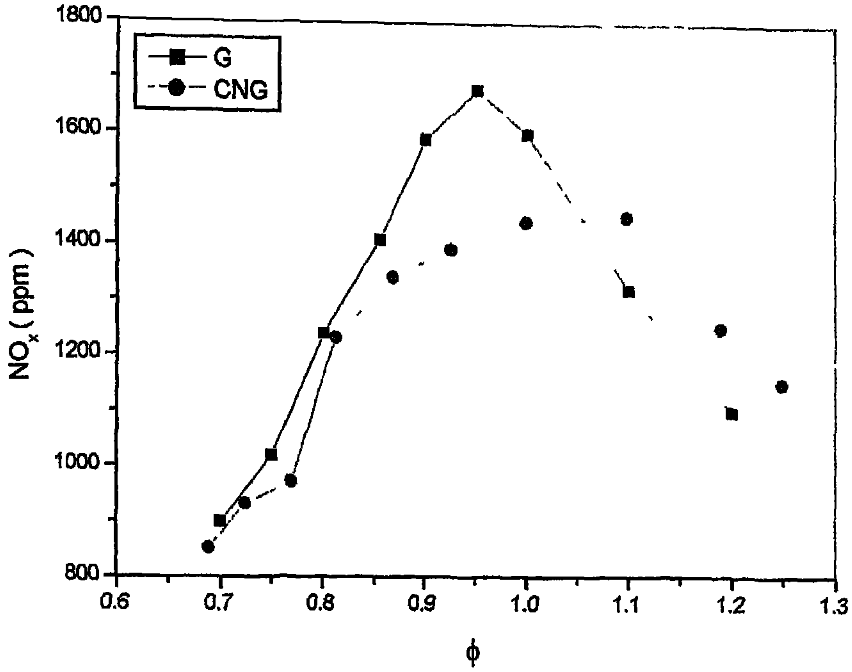

Fig. 6.56 Effect of ϕ and fuel on NO$_x$ emissions at CR=8,1500 rpm, 1/2 throttle

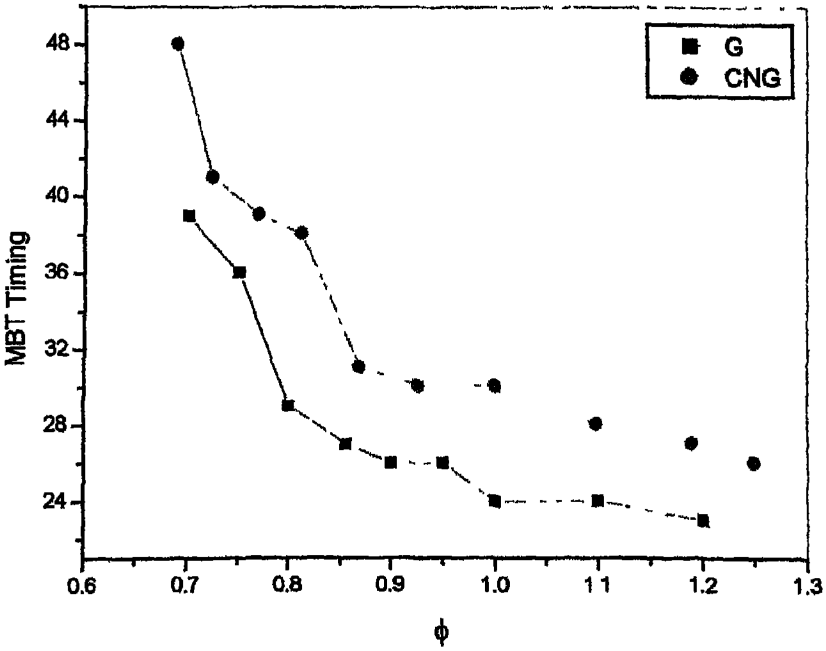

Fig. 6.57 Effect of ϕ and fuel on MBT Timing at CR=8,1500 rpm, 1/2 throttle

emissions with CNG as compared to gasoline at leaner equivalence ratios were attributed to better mixing and lesser-misfired cycles. Previous work (118-120) reported by researchers had shown that for gasoline most of the hydrocarbon emissions were non-methane compounds, while for natural gas 70 percent to 85 percent of total hydrocarbons were methane hydrocarbons. So far CNG operation non-methane hydrocarbon emissions could be quite less.

Exhaust emissions of carbon monoxide was dependent on equivalence ratio. It was lower at lean mixture and increased as mixture became richer. However, at richer mixture CO emission with natural gas was much lower as compared to gasoline. With gasoline CO emissions was higher by 83% as compared to natural gas at an equivalence ratio of 1.2. This was attributed to lower carbon to hydrogen ratio of methane as compared to gasoline. Evens and Blaszeyzyk (56) in a single cylinder engine study at WOT also found 50-75% lower CO emission with natural gas as compared to gasoline at similar equivalence ratios.

The relationship between oxides of nitrogen ($NO_x$) and equivalence ratio for both fuels is shown in Fig. 6.56. The $NO_x$ emissions from the engine when operating on both natural gas and gasoline followed classical relationship with maximum $NO_x$ occurring at slightly leaner than stoichiometric air-fuel ratio for gasoline. Peak $NO_x$ emission with natural gas was lower by about 18 percent as compared to gasoline. With decreasing equivalence ratio (or, increasing air-fuel ratio) $NO_x$ decreased sharply for both fuels and reached similar levels at about $\phi = 0.7$. However, with wider flammability limits, natural gas can provide greater possibilities for $NO_x$ reduction at very lean air-fuel ratios.

The relationship between MBT Timings, and equivalence ratio, which provides an indication of the combustion rate for both fuels, is shown in Fig. 6.57. At these operating conditions that is, at 1500 rev/min and at half throttle opening, MBT timings for natural gas was between 4 degrees to 6 degrees more advanced than for gasoline throughout the range of equivalence ratios. These results were attributed to lower flame velocity of methane-air mixture as compared to gasoline-air mixture.

### 6.3.3 ***WOT Performance***

The effect of engine speed on performance and exhaust emissions for natural gas and gasoline is shown in Figs 6.58 to 6.64. These tests were carried out at a constant Compression Ratio of 8.0:1 for both fuels and ignition timing was kept at 20 degrees before top dead centre for gasoline while for CNG ignition timing was kept at 24 degrees before top dead centre.

As it can be seen from Fig. 6.58 power output with natural gas operation was 2-12 percent lower as compared to gasoline depending on the engine speed. Lower power output with CNG was obtained even though ignition timing in case of natural gas operation was kept at 24 degrees before top dead centre as compared to 18 degrees before top dead centre for gasoline operation. The reason for lower power output with CNG was that natural gas being in a gaseous state displaced more intake air and hence, more space in the intake system was occupied by fuel. However, the difference in power output between natural gas and gasoline was lesser at lower engine speeds. At lower engine speeds vapourization and hence combustion of gasoline air mixture was less efficient.

The relationship between engine speed and thermal efficiency of both fuels at WOT are shown in Fig. 6.59. Thermal efficiency with natural gas was higher by about 3 percent as compared to gasoline through out the range of engine speeds. This was attributed to better mixing of methane-air mixture as compared to gasoline-air mixture. Brake specific fuel consumption with natural was gas also lower by about 7 percent (Fig. 6.60). Lower fuel consumption on mass basis with CNG as compared to gasoline was attributed to greater heating value of methane in CNG as compared to gasoline. Similar results at WOT and at 65% of full load were obtained by Evens and Blaszezyk (56).

Exhaust emissions of hydrocarbon, carbon monoxide and oxides of nitrogen at WOT are depicted in Figs 6.61, 6.62 and 6.63 respectively. The results in these figures substantiate the fact that for all regulated harmful emissions (i.e. CO, HC and $NO_x$), CNG fuelling gives rise to lower emission levels than those observed with gasoline. For CNG, the average reduction in volume of CO emissions with respect to gasoline at WOT thought out the speed range was 40 percent, whereas average reduction in volume of $NO_x$ for CNG with respect to gasoline at WOT was about 10 to 12 percent. Reduction in CO emissions with CNG as compared to gasoline is attributed to lower carbon to hydrogen ratio as it was seen earlier (Fig. 6.55). Hence, at the same equivalence ratios CNG combustion produced lower CO emissions. Lower $NO_x$ emissions at WOT was attributed to lower flame temperature of methane-air mixture as compared to gasoline-air mixture. Measured exhaust gas temperature at the same location of the tail pipe also shows lower exhaust gas temperature with CNG as compared to gasoline (Fig. 6.64)

### 6.3.4 *Performance Using Twin Spark Plugs*

The laminar flame speed of natural gas is lower than that of the liquid hydrocarbons (74), which results in slower combustion rate

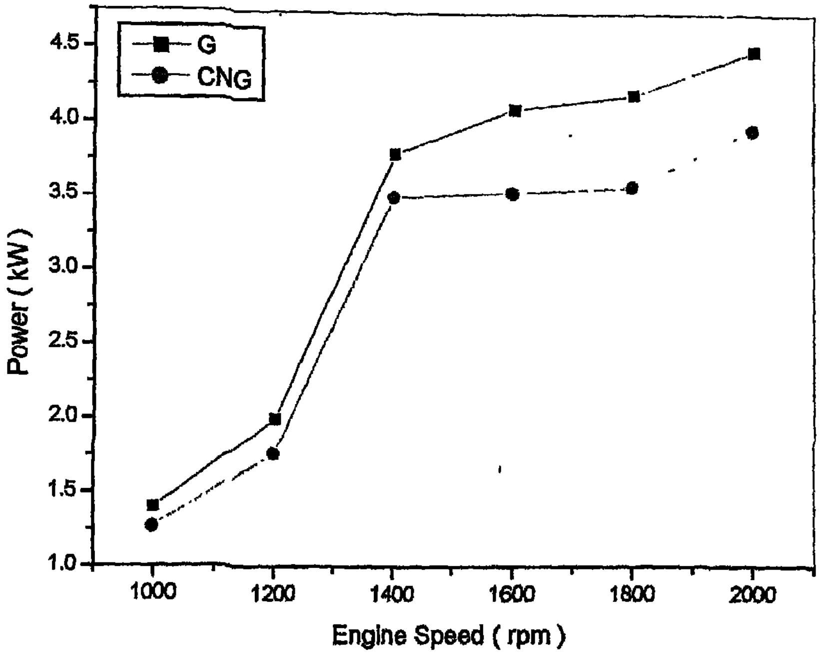

Fig. 6.58 Effect of engine speed and fuel on power at W.O.T

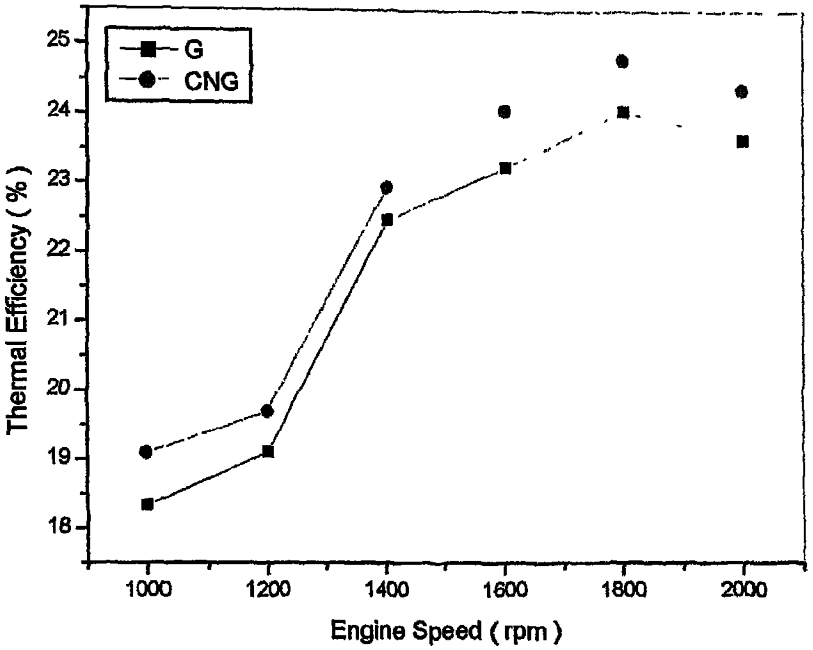

Fig. 6.59 Effect of engine speed and fuel on thermal efficiency at W.O.T

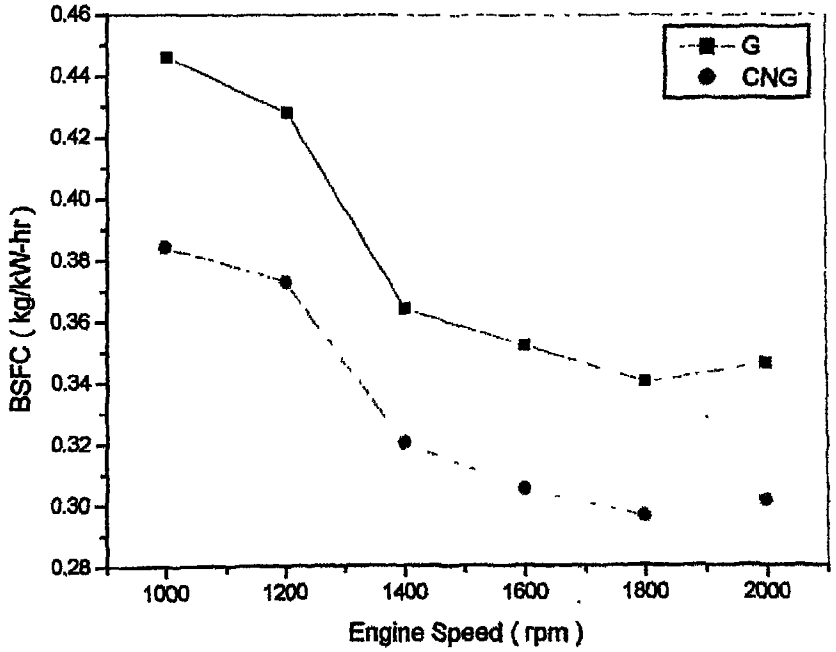

Fig. 6.60 Effect of engine speed and fuel on BSFC at W.O.T

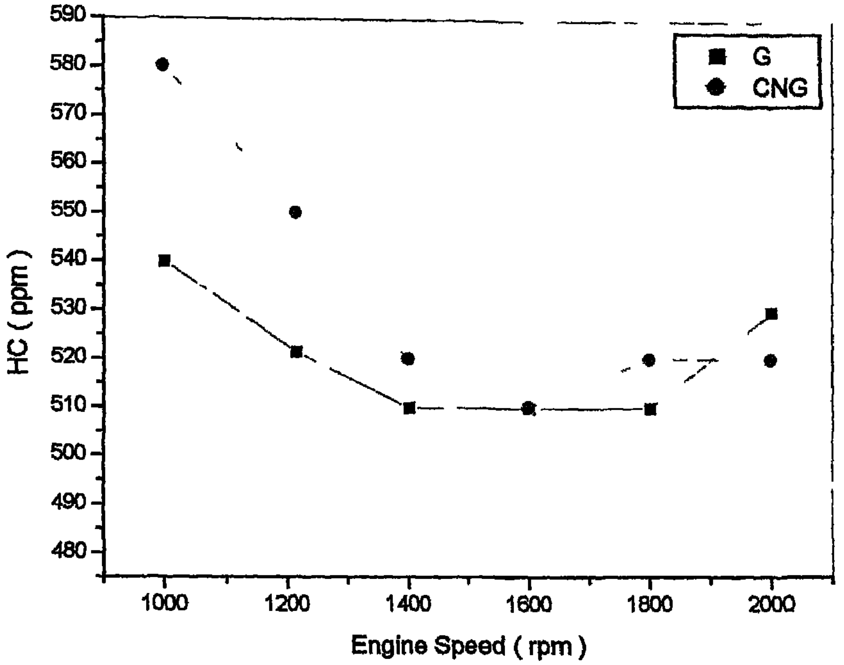

Fig. 6.61 Effect of engine speed and fuel on HC emissions at W.O.T

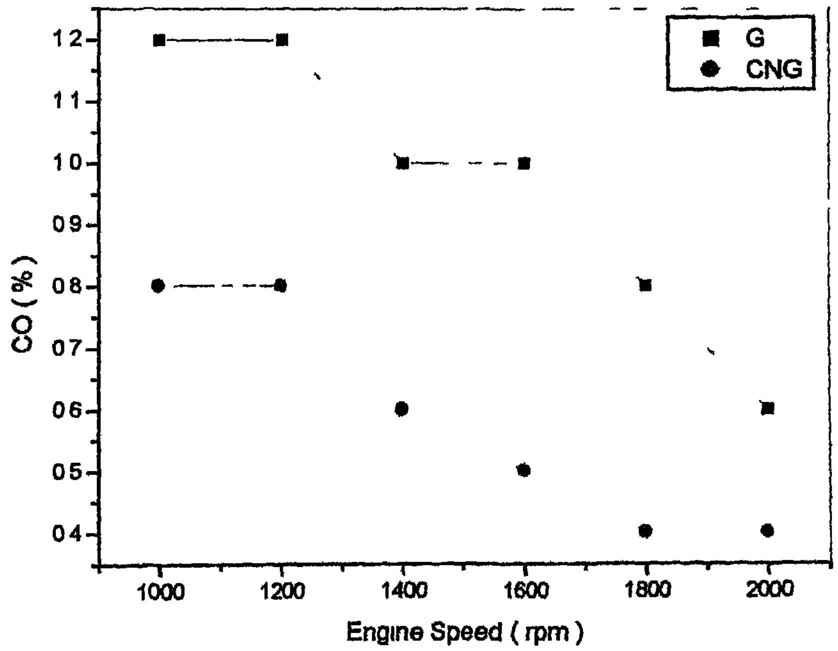

Fig. 6.62 Effect of engine speed and fuel on CO emissions at W.O.T

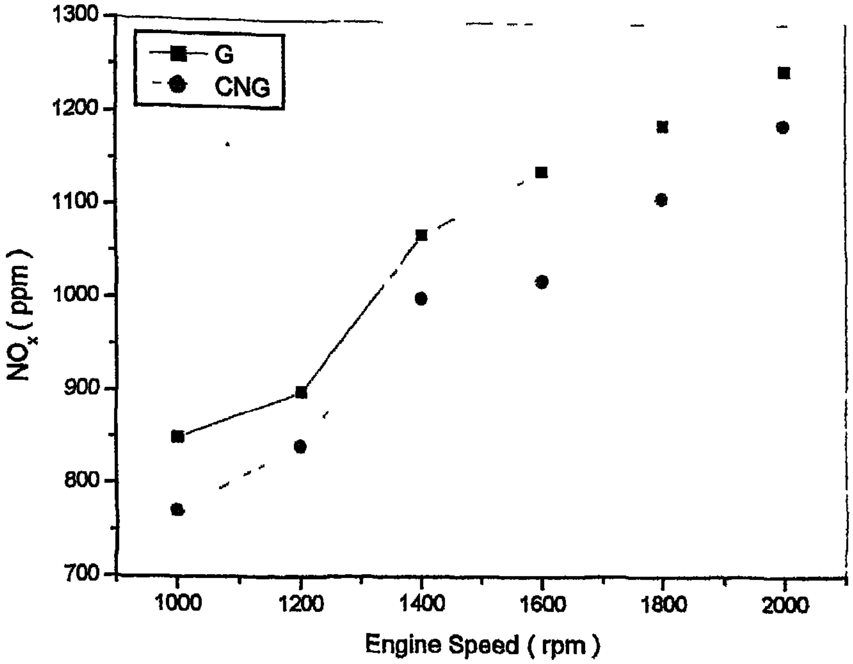

Fig. 6.63 Effect of engine speed and fuel on NO$_x$ emissions at W.O.T

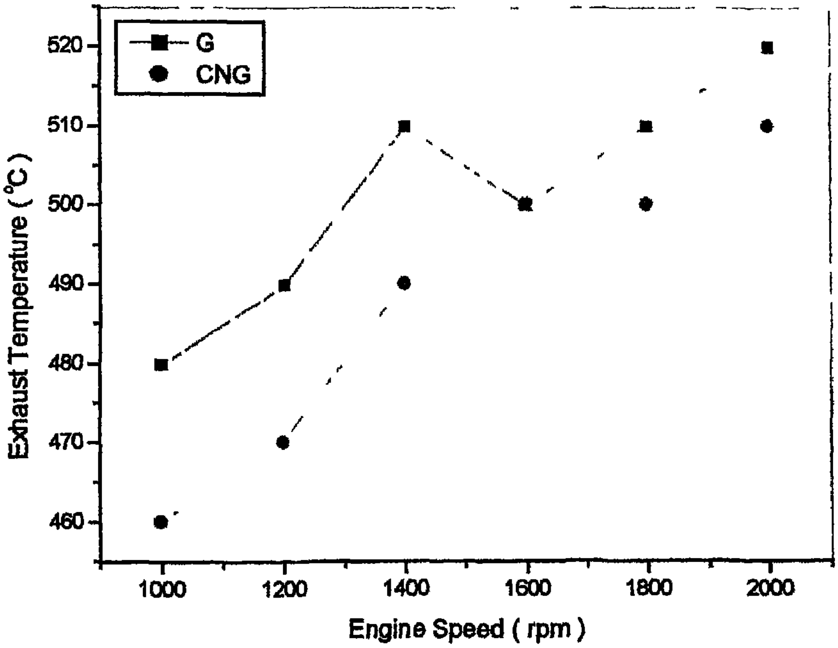

Fig. 6.64 Effect of engine speed and fuel on exhaust temperature at W.O.T

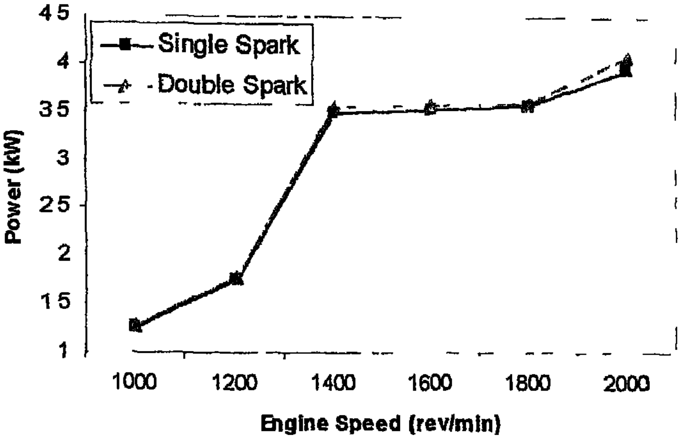

Fig. 6.65 Comparision of power output using single and double spark plug at WOT, Fuel CNG, CR=8

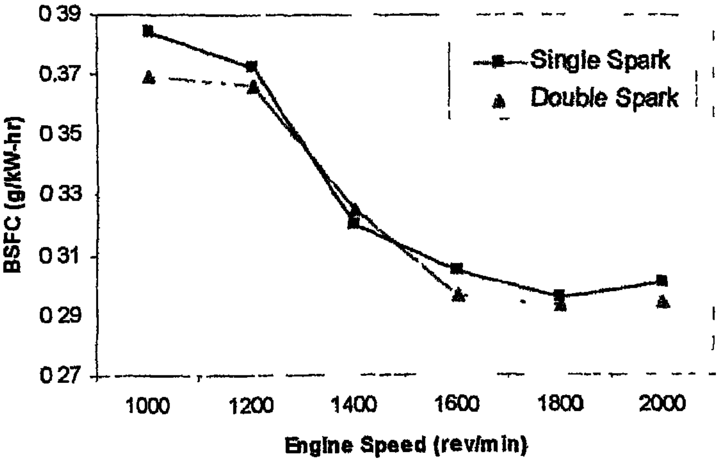

Fig. 6.66 Comparision of BSFC using single and double spark plug at WOT, Fuel CNG, CR=8

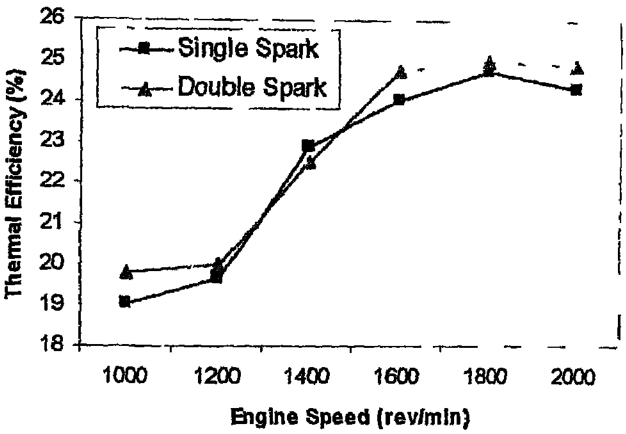

**Fig. 6.67** Comparision of thermal efficiency using single and double spark plug at WOT, Fuel CNG, CR=8

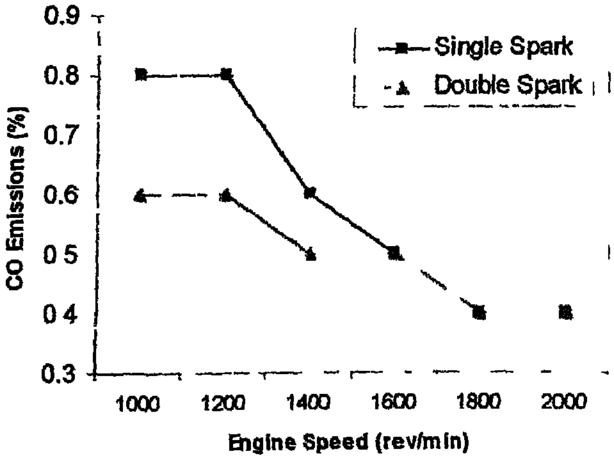

**Fig. 6.68** Comparision of CO emissions using single and double spark plug at WOT, Fuel CNG, CR=8

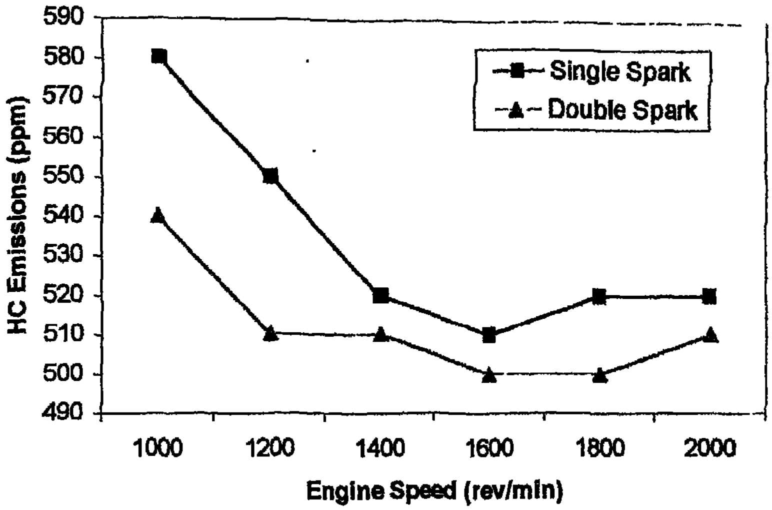

**Fig. 6.69 Comparision of HC emissions using single and double spark plug at WOT, Fuel CNG, CR=8**

leading to a drop in efficiency of unmodified engines. It was therefore, thought that using single spark plug might not be suitable for CNG operation as the flame would take longer time to reach the other end of the combustion chamber. Using two spark plugs was considered to improve the rate of combustion, which was then expected to give enhance the engine performance. In the experimental engine the spark plug was situated at the side of the combustion chamber, between the valves and a similar hole on the other side of the combustion chamber enabled two spark plugs to be used simultaneously, placing them near either side of the inlet and the exhaust valve.

Fig. 6.65 shows a comparison of power output using twin spark plug and single spark plug. It was observed that a marginal improvement in power output was possible by using double spark plugs as compared to a single spark plug. Marginal improvement in power output was attributed to peak pressure more closer to TDC by using double spark plug than that of the single spark plug. However, for CNG operation even with double spark plugs and advanced ignitions timing power output was less as compared to gasoline. Using double spark plugs fuel consumption and thermal efficiency were improved by about 3 - 5 per cent as compared to single spark plug (Figs. 6.66 and 6.67). Comparisons of exhaust gas emissions of carbon monoxide and hydrocarbons are shown in Figs. 6.68 and 6.69. Exhaust emissions of CO and HCs were lower with double spark plugs as compared to single spark plug. Lower CO and HC were attributed to improved combustion using double spark plugs.

## 6.4 *COMBUSTION CYCLE ANALYSIS*

In the attempt to explain the improved efficiency with ethanol (E-85) and ethanol-gasoline blends cylinder pressure data were recorded and analyzed, so that combustion characteristics of E- 85 and E-10 can be compared with gasoline. The method of analysis and results are described below.

Pressure data were collected with ethanol (E-85), ethanol-gasoline blend (E-10) and gasoline at WOT or half throttle conditions. For carrying out all these tests, engine compression ratio was maintained at 8.0:1 and the engine speed was kept at 1500 rev/min. Engine with gasoline and E-10 fuels was operated at the spark timing of 18 degrees before top dead centre, while for ethanol (E-85) spark was triggered at 14 degrees before top dead centre. For all these tests carbon monoxide emissions were adjusted at 0.5%, which corresponds to air equivalence ratio of 1.0. Cylinder gas pressure was measured using quartz piezoelectric transducer, charge amplifier and digital oscilloscope.

Figs. 6.70 and 6.71 show comparisons of pressure-time diagrams of E-85 and gasoline at 50% throttle opening and WOT respectively. Comparing the results of E-85 and gasoline at same compression ratio, engine speed and throttle opening showed that pressure increased more rapidly with E-85 even though spark was triggred later in the cycle. Peak pressures were higher with E-85, as were pressures throughout the expansion stroke. A critical examination of the pressure traces showed that with E-85 cycle pressure started increasing rapidly from 10-15 degrees after top dead centre, whereas, in the expansion stroke higher pressures were maintained throughout. So the work done by the piston (positive work) with E-85 was more as compared to gasoline. This resulted in higher-thermal efficiency with E-85 as compared to gasoline. Figs. 6.72 and 6.73 show comparison of pressure curves of E-10 and gasoline at 50 percent throttle opening and WOT respectively. Higher cylinder gas pressure throughout the expansion stroke and peak points were obtained with E-10 also as compared to gasoline.

Figs 6.74 and 6.75 shows variations of maximum pressure at different engine speeds at half throttle and WOT respectively using E-85, E-10 and gasoline. It was observed that at all engine speed maximum pressures were higher *for ethanol operation. Coefficient of variation of* cylinder peak pressures were observed to be marginally less with gasoline as compared to E-85 (Fig. 6.76)

## 6.5 ***RESULTS FROM COMPUTER SIMULATION MODEL***

In order to understand the process of formation of exhaust gases, the behaviour of gas temperature and pressure inside the combustion chamber, computer simulation analysis with gasoline and E-85 were carried out. Already available computer simulation model (106) was modified for the present engine to obtain gas temperature and cylinder pressures at different crank angle and exhaust emissions of carbon monoxide and hydrocarbons at different fuel-air equivalence ratios. Figures 6.77 shows a computed P-$\theta$ diagram for E-85 with experimental one during the power cycle. Reasonable agreement could be noticed between the two except during the expansion process. Figs. 6.78 and 6.79 show the comparison of predicted $NO_x$ and CO concentrations at the end of the expansion process with the measured values of exhaust $NO_x$ and CO when engine was operated on E-85. Similar to the case of gasoline, the characteristic variation of $NO_x$ and CO with respect to fuel air equivalence ratio is evident. The validity of the model for application to E-85 operation is substantiated by the good agreement of the predicted emissions of both these species with the measured values.

Cylinder gas temperatures at different crank angle with gasoline and E-85 are shown in Fig. 6.80. While table 6.2 shows the different engine component temperatures (computed) with gasoline and E-85. These results indicated slightly lower cylinder gas temperature with E–85 as compared to gasoline. It may be due to higher latent heat of vaporization and lower flame temperature of E-85 as compared to gasoline.

**Table 6.2, computed component temperatures**

| Fuel/Engine Components | Cylinder Head | Piston | Sleeve | Inlet valve | Exhaust valve | Exhaust Port |
|---|---|---|---|---|---|---|
| Gasoline | 401.70 | 554.84 | 377.14 | 770.25 | 1025.50 | 518.23 |
| E-85 | 396.75 | 550.17 | 371.25 | 764.35 | 1011.15 | 515.34 |

The significant conclusions that have been derived based on the above experimental and theoretical investigations are summarized in the next chapter.

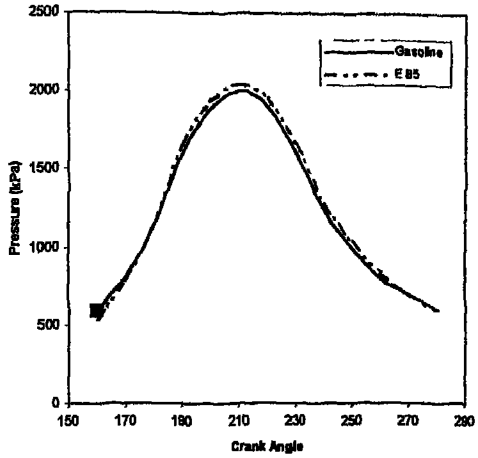

Fig. 6.70 Pressure-crank angle diagram for Gasoline and E 85 at half throttle, CR = 8.0

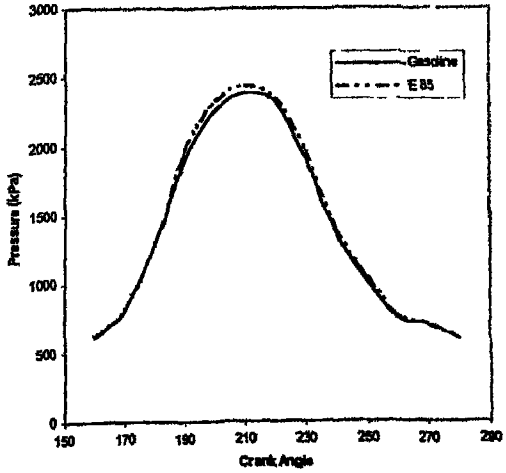

Fig. 6.71 Pressure-crank angle diagram of gasoline and E 85 at WOT, CR = 8.0

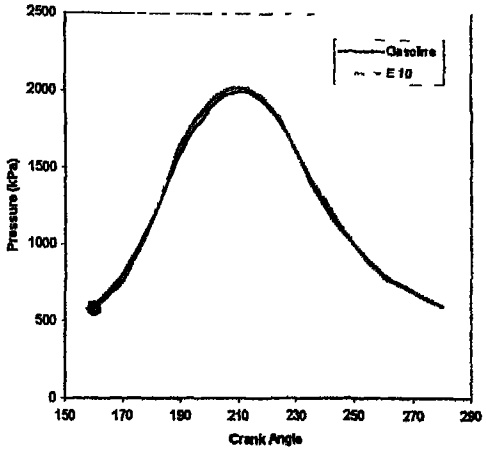

Fig. 6.72 Pressure-crank angle diagram for gasoline and E 10 at half throttle, CR = 8.0

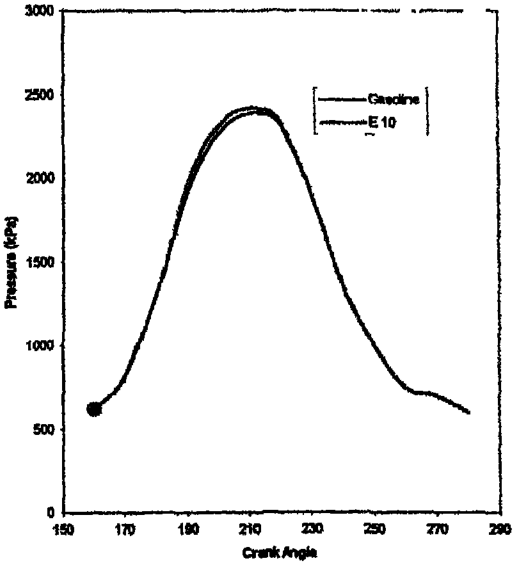

Fig. 6.73 Pressure-crank angle diagram of gasoline and E 10 at WOT, CR = 8.0

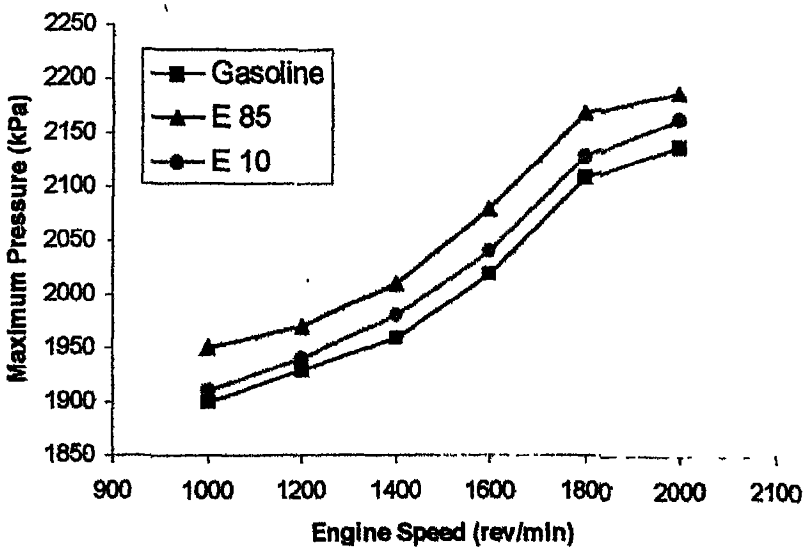

Fig. 6.74 Effect of engine speed on maximum cylinder pressure at half throttle conditions, MBT Timings, CR = 8.0

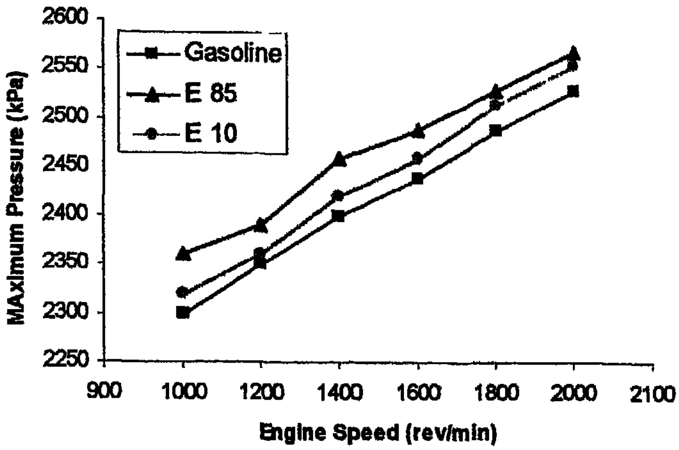

Fig. 6.75 Effect of engine speed on maximum cylinder pressure at WOT, MBT Timings, CR = 8.0

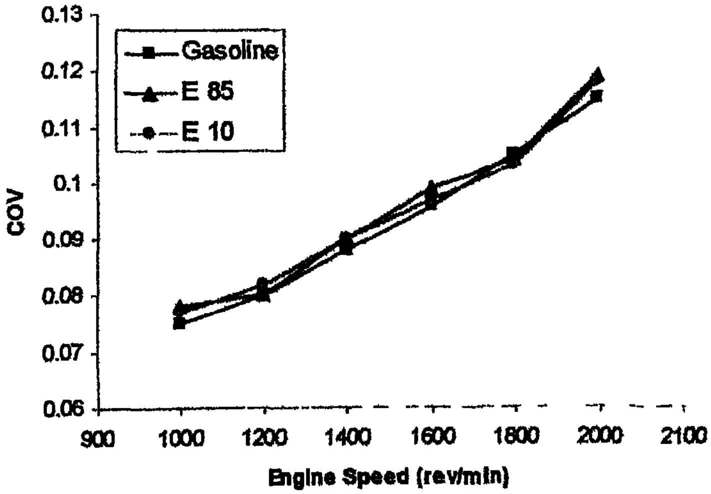

Fig. 6.76 Effect of engine speed on cyclic variations at WOT, MBT Timings, CR = 8.0

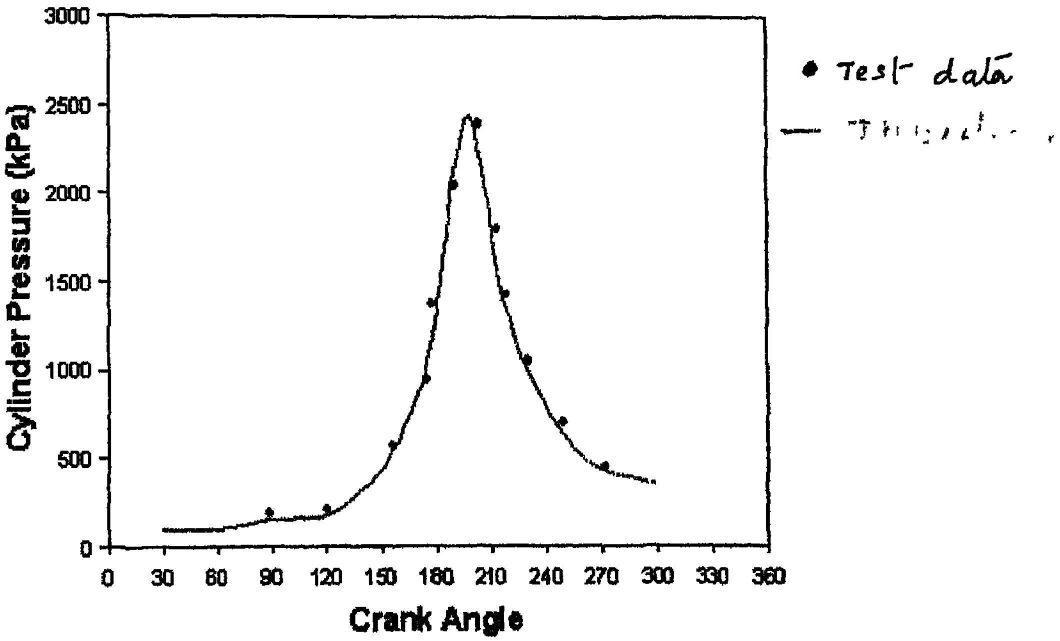

Fig 6.77 Comparison of computed and test results of cylinder pressure at WOT, CR=8, equivalence ratio=1.0, engine speed=1500 rpm

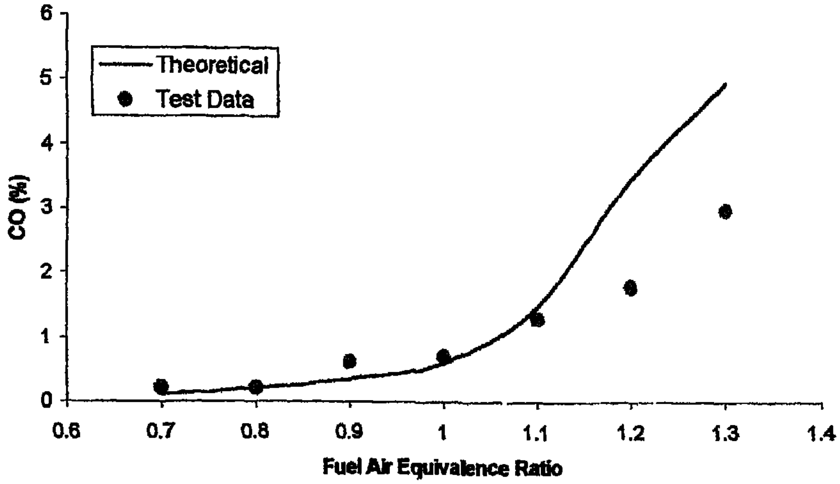

Fig. 6.78 Comparision of computed and test results of carbon monoxide levels

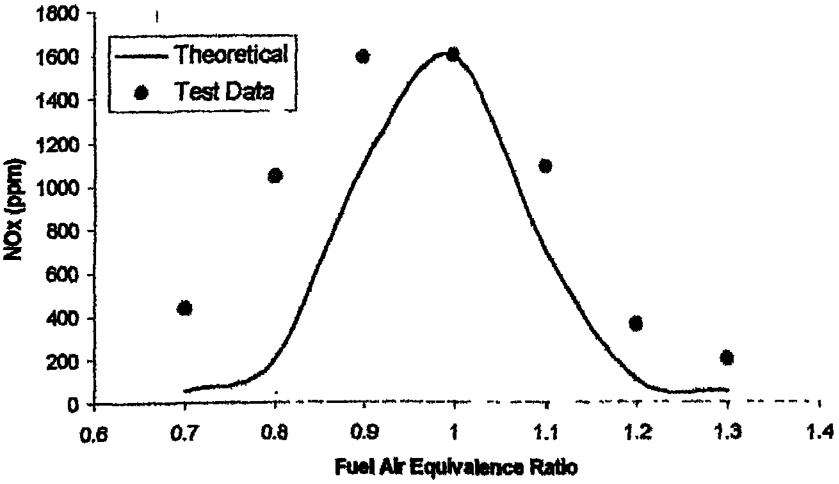

Fig. 6.79 Comparision of computed and test reults of NOx levels

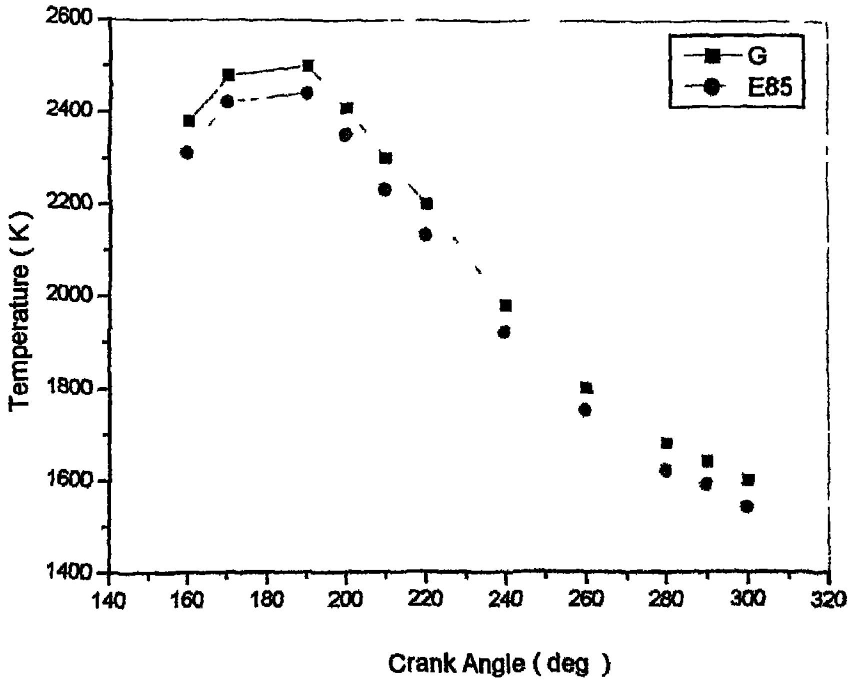

Fig. 6.80 Comparison of cylinder gas temperature of gasoline and E-85

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# APPENDIX- I

## *FLAME VISIBILITY ETHANOL AND PHASE SEPARATION OF ETHANOL-GASOLINE BLENDS*

The important factors to be considered for any alternative programme includes a careful examination of the following factors.

i. Whether the resource base for the alternative fuel is large enough to provide supply for a significant period of time.
ii. Technical feasibility of its use
iii. Environmental implications with its use
iv. Economic considerations and
v. Safety considerations

Available work on safety aspects of compressed natural gas is quite significant (121-124). However, no work or very little work has been done on safety aspects of ethanol especially from the point of view of flame visibility. Since poor visibility of the flame can cause fire hazards, so it was decided to carry out some simple tests on flame visibility of ethanol. For these tests flames of pure ethanol and E-85 (85 percent ethanol and 15 percent gasoline by volume) were observed on bright sun. Figs. A-1 and A-2 show the flames of pure ethanol snd E-85 respectively from a transparent container, while Figs. A-3 and A-4 show the flames of pure ethanol E-85 respectively from a non-transparent container. As it can be seen from these figures that the visibility of pure ethanol flame is little poorer than normal flame visibility. When the pure ethanol flame was seen on a bright sunlight the flame was hardly visible. To avoid accident due to fire a better flame visibility is required. Adding 15% gasoline to pure ethanol resulted in much better flame visibility.

Another problem with alcohol fuel flame is phase separation when blended with gasoline in small percentages specially at low temperatures. Phase separation problems is more severe with methanol than with ethanol. Some tests were carried out to examine the phase separation problem of ethanol-gasoline blends. Fig. A- 5 shows the blends of E–5 (5 percent ethanol and 95 percent gasoline by volume), E-10 (10 percent ethanol and 90 percent gasoline by volume) and E–20 (20 percent ethanol and 80 percent gasoline by volume) at normal temperature. No phase separation was observed when these blends after 48 hours at normal atmospheric temperature. Fig. A-6 shows similar blends at 5 degree centigrade. No phase separation of ethanol-gasoline blends were observed even at 5 degree centigrade after 48 hours.

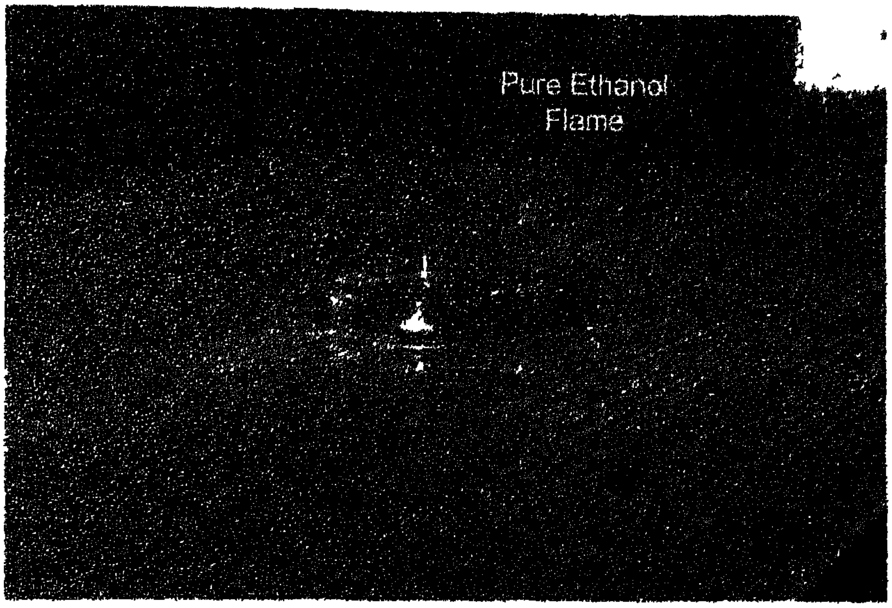

Plate A-1: Flame visibility of Ethanol, the fuel kept in a transparent container.

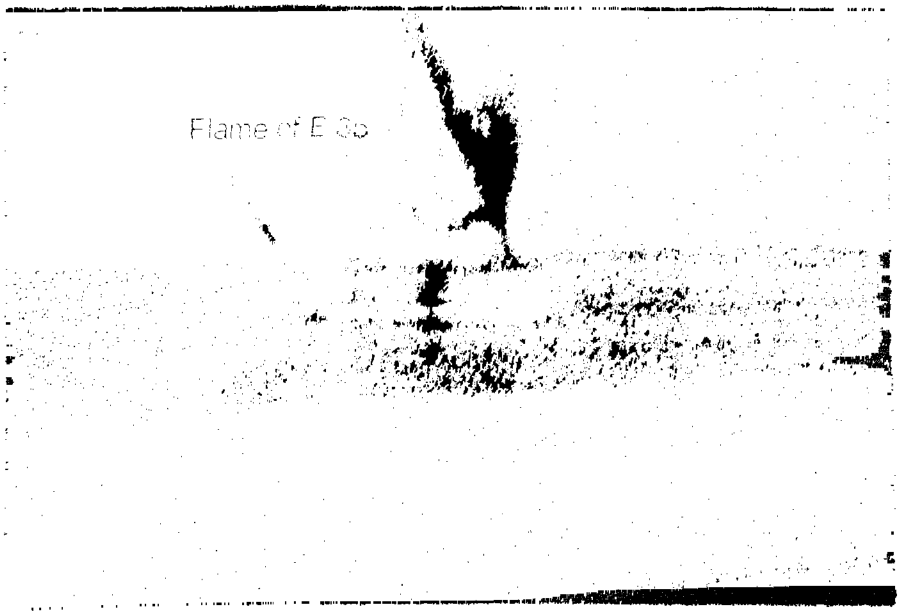

Plate A-2: Flame Visibility of E 85, the fuel kept in a transparent container.

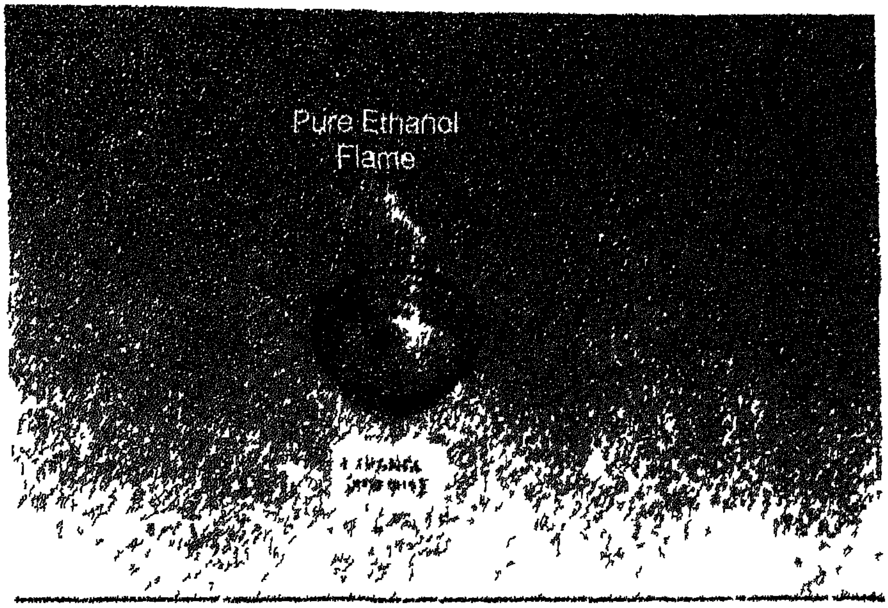

Plate A-3: Flame visibility of Ethanol, the fuel kept in a non-transparent container

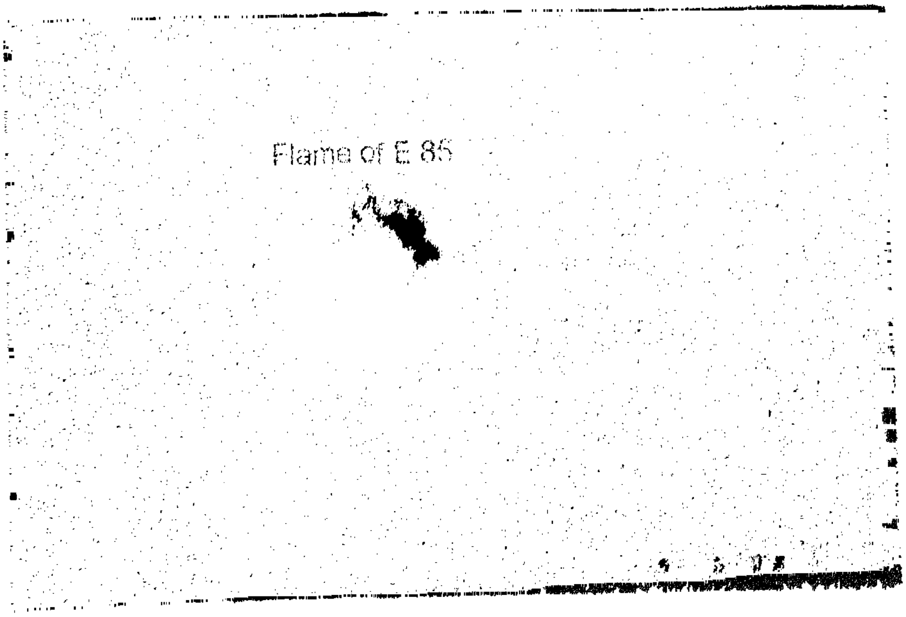

Plate A-4: Flame visibility of E 85, the fuel kept in a non-transparent container.

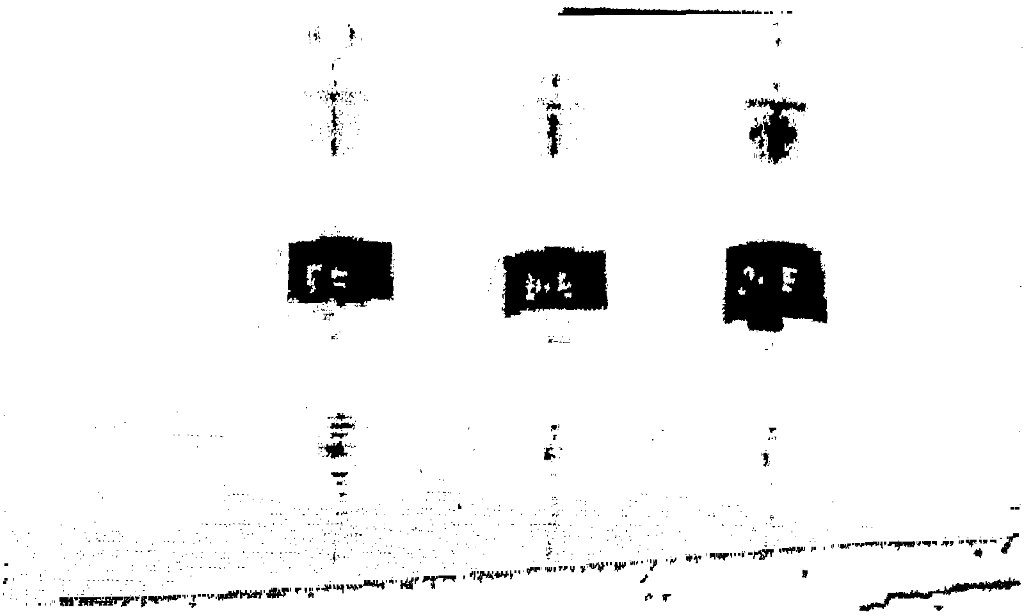

Plate A-5· Phase separation of Ethanol-gasoline blends at normal temperature

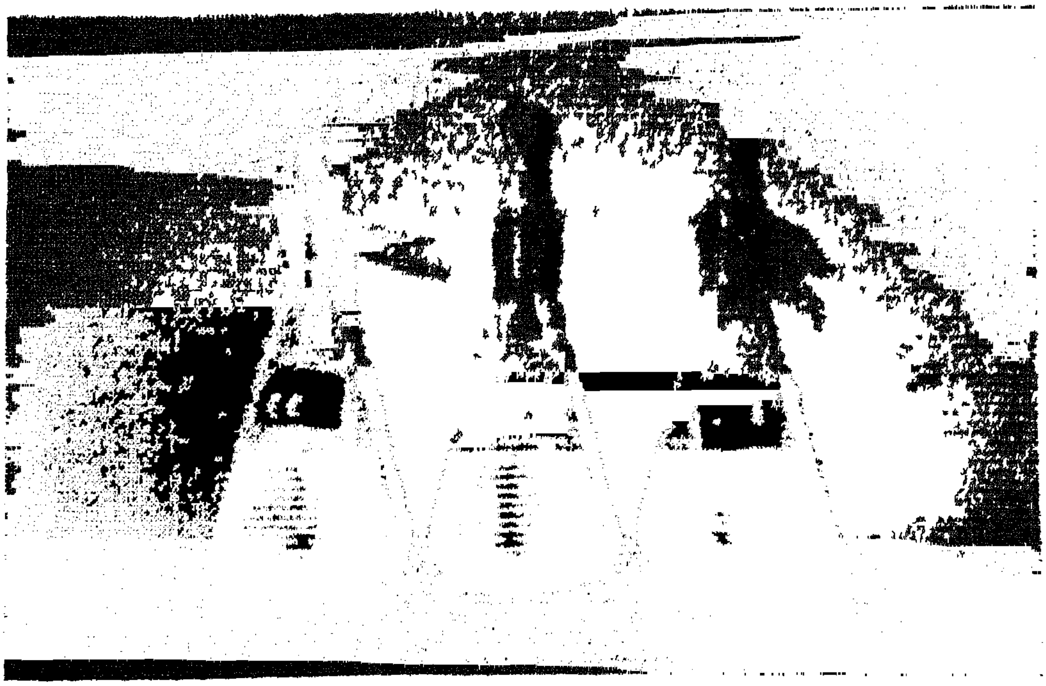

Plate A-6· Phase separation of Ethanol-gasoline blends at $5^{O}C$.

# APPENDIX- II

## ENGINE SPECIFICATIONS

Type : Ricardo E6/T. Variable Compression Ratio

No. of Cylinders : One

Cycle Operation : Four Stroke

Ignition : Spark Ignition

Cooling System : Water Cooled

Speed Range : 1000-2500 rev/min

Ignition Setting : 20-60 deg. TDC

I.V.O. : 9 Degrees bTDC

I.V.C. : 36 Degrees bTDC

E.V.O. : 41 Degrees bBDC

E.V.C. : 8 Degree aTDC

Bore : 76.2 mm

Stroke : 111.125 mm

Max. power : Approximately 5.5 kW

### DYNAMOMETER

Type : Lawrence Scott "NS", Swinging Field A.C. Dynamometer

## *SPECIFICATIONS OF ENGINE-B*

| | | |
|---|---|---|
| Bore | : | 69.0 mm |
| Stroke | : | 72.0 mm |
| Capacity | : | 1089 cc |
| No. of Cylinders | : | Four |
| Cycle Operation | : | Four Stroke |
| Ignition | : | Spark Ignition |
| Cooling System | : | Water Cooled |
| Maximum Speed | : | 4500 rev./min. |
| Idling speed | : | 900 rev./min. |
| Ignition Timing | : | 20 deg. bTDC |

# *APPENDIX – III*

## *Instruments for the Measurements of Carbon Monoxide and Hydrocarbon Emissions*

Horiba MEXA – 324 GB NDIR (Non Dispersive Infrar-red) Automotive exhaust gas analyzer was used to measure the concentrations of carbon monoxide and hydrocarbons in the engine exhaust. In the NDIR analyzer the exhaust gas species being measured is used to detect itself. This is done by selective absorption. The infrared energy of a particular wavelength or frequency is peculiar to a certain gas in that the gas will absorb the infrared energy of this wavelength and transmit the infrared energy of other wavelengths. For example, the absorption band for carbon monoxide is between 4.5 and 5 microns. So the energy absorbed at this wavelength is an indication of concentration of CO in the exhaust gas.

The NDIR analyzer as shown in Fig.A-7 consists of two infrared sources, interrupted simultaneously by an optical chopper. Radiation from these sources pass in parallel paths through a reference cell and a sample cell to opposite side of a common detector. The sample cell contains the compound to be analyzed, whereas this compound is not present in the reference cell. The latter usually filled with an inert gas, usually nitrogen, which does not absorb the infrared energy for the wavelength corresponding to the compound being measured. A closed container filled with only the compound to be measured works as a detector.

The detector is divided into two equal volumes by a thin metallic diaphragm. When the chopper blocks the radiation, the pressure in both parts of the detector is same and the diaphragm remains in the neutral position. As the chopper blocks and unblocks the radiation, the radiant energy from one source passes through the reference cell unchanged whereas the sample cell absorbs the infrared energy at the wavelength of the compound in the cell. The absorption is proportional to the concentration of the compound to be measured in the sample cell. Thus unequal amounts of energy are transmitted to the two volumes of the detector and the pressure differential so generated causes movement of the diaphragm of the detector. This changes the capacitance between the diaphragm and a fixed probe, thereby generating an a.c. signal which is amplified and, after rectification to d.c., displayed on a meter. The signal is a function of the concentration of the compound to be measured.

The NDIR can accurately measure CO, $CO_2$ and those hydrocarbons which have clear infrared absorption peaks. However, usually the exhaust sample to be analysed contains other species which also absorb infrared energy at the same frequency. For example, an NDIR analyzer sensitized to n-hexame for detection of HC responds equally well to other paraffinic HC but not to olefins, acetylens or aromatics. Therefore, the reading given by such analyzer is multiplied by 1.8 to correct it to the total UBHC as measured by an FID analyzer in the same exhaust stream.

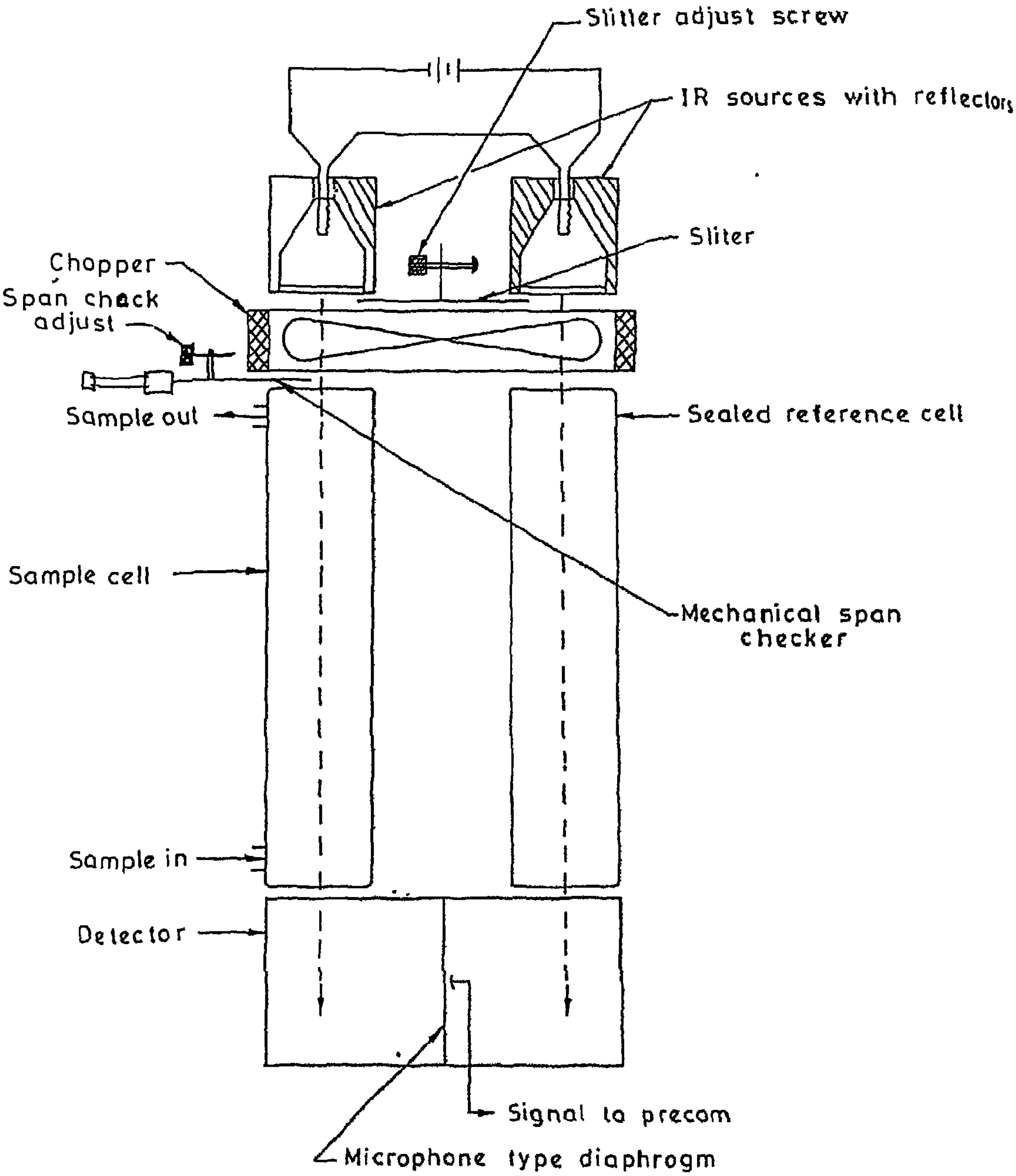

Fig. A-7, Schematic diagram of NDIR analyzer

## *APPENDIX- IV*

### *Instruments for the Measurements of of Exhaust $NO_x$*

For accurate measurement of exhaust $NO_x$ concentration Thermo Electron's model 10A Rack mounted chemiluminescent NO-$NO_x$ Gas Analyzer was used.

Chemiluminsesçent is the emission of light energy resulting from a chemical reaction. It was found in the late 1960's that the reaction of NO and ozone ($O_3$) produces $NO_2$ in activated state which emits light in due course as it converts to normal state.

$$NO + O_3 \rightarrow NO_2^* + O_2$$

$$NO_2 \rightarrow NO_2 + \text{light emissions}$$

The light emitted is proportional to the concentration of NO in the sample stream.

To measure NO concentration the gas sample to be analyzed is send to a reaction chamber, where it combines with ozone in a flow reactor as shown in A.8 and A.9. The resulting chemiluminscent is monitored through an optical filter by a high sensitivity photomultiplier positioned at one end of the reactor. The filter photo multiplier combination responds to light in a narrow wavelength band unique to the above reaction. The output from the photo multiplier is linearly proportional to the NO concentration.

To measure $NO_x$ concentration ( i.e. NO plus $NO_2$) the sample gas flow is diverted through an $NO_2$-to–NO converter. The chemiluminescent response in the flow reactor to the converter effluent is linearly proportional to the $NO_x$ concentrations entering the converter. The photomulitplier output is amplified and given to an indicator for measuring NOx concentration directly. A model 800 heat $NO_x$ sampler is a module, which contains the particulate filter and sample plumbling for the chemiluminescent analyzer in a heater chamber. The sampler utilizes the temperature pressure characteristics of water to eliminate the formation of condensate in the sample. Within the module the sample is heater to a temperature, which is above the gas dew point. As the sample flows through the plumbing, the pressure is reduced to approximately 15 Torr and heated to prevent condensation at this pressure is necessary. Thus the analysis of $NO/NO_x$ in an emission gas is conveniently made without sample loss or alteration.

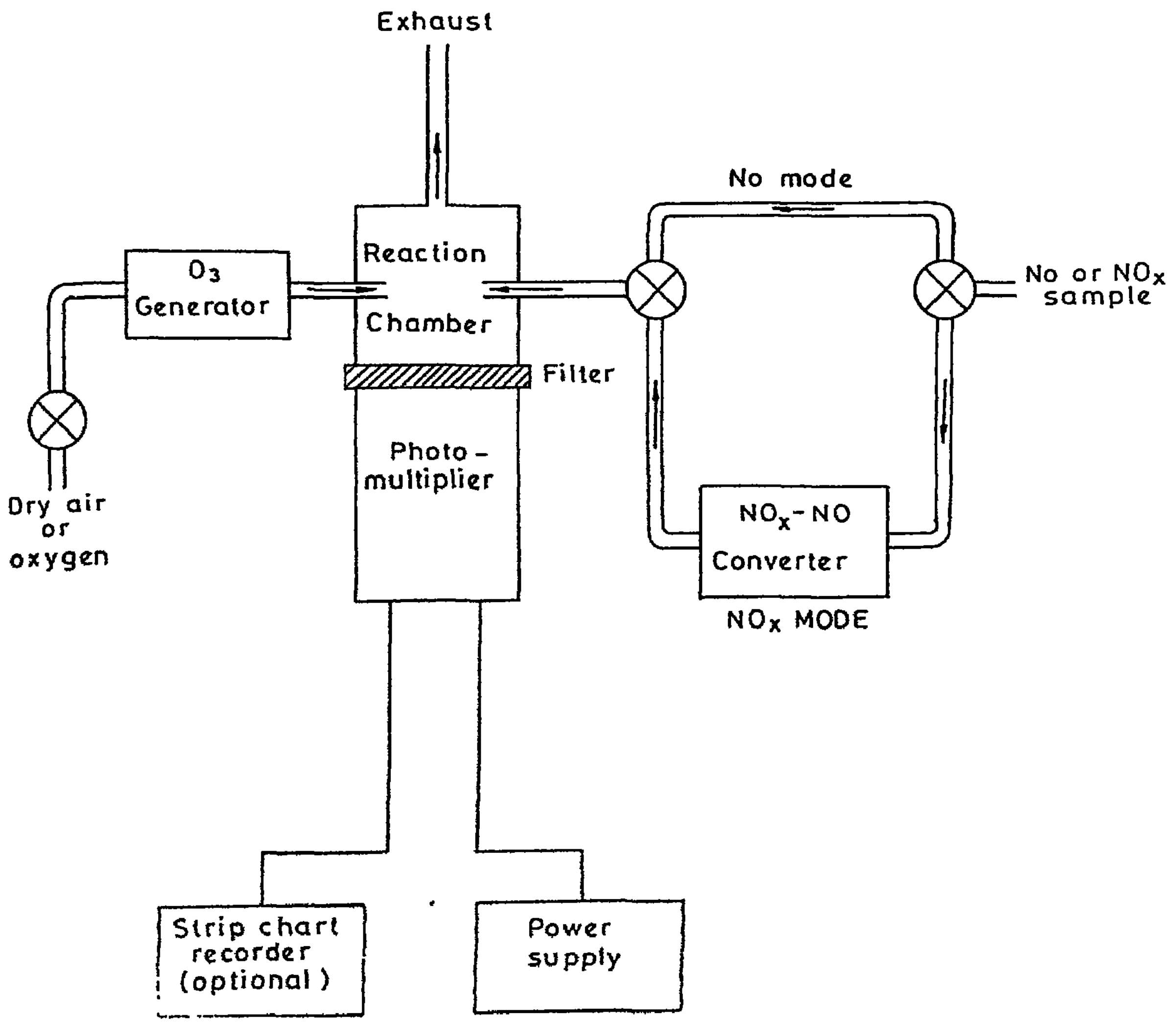

Fig. A-8, Schematic diagram of Chemiluminescent Analyzer

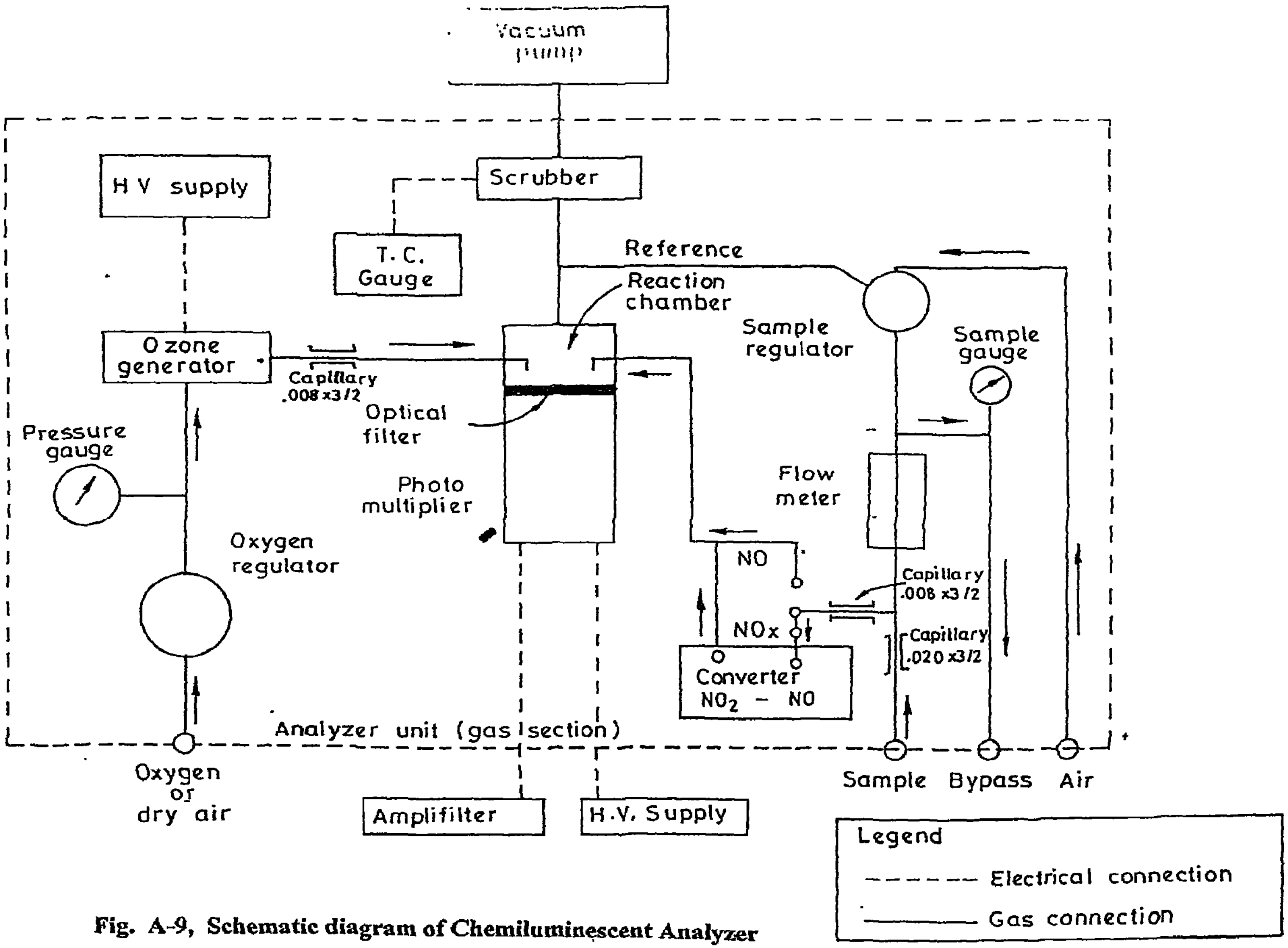

Fig. A-9, Schematic diagram of Chemiluminescent Analyzer

## *APPENDIX- V*

### *SPECIFICATIONS OF PRESSURE TRANSDUCER*

Type : Pies-Electric, Kittler, 701A

Calibrated range : 0...250 bar

Sensitivity : -82,1 pick/bar

Linearity : ≤ ± 0.1 % FSO

Operating Temperature : - 150 ...$200^{0}$C

## APPENDIX- VI

## SPECIFICATIONS OF CHARGE AMPLIFIER

| | | |
|---|---|---|
| Type | Kittler 5011 | |
| Measuring range, *Continuous setting* | PC | ± 10....999'000 |
| Transducer sensitivity (T) | PC/M.U. | 0,01....9'990 |
| a) | | |
| Scale (S) | M.U./V | |
| 0,001....9'990'000 | | |
| Output Voltage | V | ± 10 |
| Current (short-circuit Protected) | am | ± 0...5 |
| Impedance | Ohm | 10 |
| Linearity | % FS | ≤ ± 0,05 |
| Accuracy to + 99,9 pick FS | % | ≤ ± 3 |
| b) | | |
| from + 100 pick FS | % | ≤ ± 1 |
| Frequency range -3 dB (Filter off) | KHz | ≈ 0 - 200 |
| Frequency response error (Filter off) | | |
| to 50 KHz | | - 1...+ 2 |
| to 200 KHz | % | - 30...+3 |
| Time constant resistant Rag (long) | Ohm | > $10^{14}$ |

| | | | |
|---|---|---|---|
| (Medium) | : | Ohm | ca. $10^{11}$ |
| (Short) | : | Ohm | ca. $10^{9}$ |
| Time constant (TC) (Long) | : | | c) |
| ($T + R_{ag} C_g$) (Medium) | : | S | 1...10'000 |
| (short) | : | S | 0,01...100 |
| Zero point transition, (reset-Operate) switching | : | PC | < ± 0,2 |
| Drift (25° C) MOSFET Leakage current | : | pC/s | < ± 0,03 |
| Input voltage, max. value For pulse lengths < 0,3s | : | V | ±125 |
| Insulation resistance at Input | : | Ohm | > $10^{14}$ |
| Zero offset, for (Reset) (max/typ.) | : | mV | < 2/0, 4 |
| (automatic zero point correction) | | | |
| Temperature coefficient of Zero point for (Reset) | : | u V/K | ca. 10 |
| Adjusting range of zero Point setting at input | : | mv | ca. ± 250 |

a) M.U. = Mechanical Unit, eg. Bar, N, g.
b) FS = Full Scale
c) The time constant (Long cannot be considered as a real time constant. The signal behaviour vs. Time is mainly influenced by drift effects. (Long) is used for quasistatic measurements.

## *APPENDIX – VII*

## *SPECIFICATIONS OF OCILLOSCOPE*

Type : KIKUSUI COM7201 – Storage

No. of Channels : Four

Sensitivity : DC – 200 MHz:1.5 Div

DC – 10 MHz : 0.4 DIV

Band Width : DC – 4 MHz : Within – 3db

Rise Time : 1.75 ns

Sweep Speed : 10 ns/DIV – 0.5 s/DIV

Setting Display : CH1, CH2, CH3, CH4
With scale factors and coupling modes

Input Impedance : 1 M Ohm ± 1%, 18 puff ±

Maximum Input : 1 M Ohm circuit: 400 v (DC + AC Peak)

# A STUDY OF ALTERNATIVE FUELS IN S.I. ENGINES FOR FUEL EFFICIENCY AND EXHUAST EMISSION CONTROL

A THESIS
SUBMITTED IN FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF
THE DEGREE OF

**DOCTOR OF PHILOSOPY**

By

**SAGAR MAJI**

Department of Mechanical Engineering,
Delhi College of Engineering
Faculty of Technology, University of Delhi
July, 2002

# 7

# CONCLUSIONS

A single cylinder engine was operated on gasoline, E-85, E-20, E-10 and compressed natural gas at the same equivalence ratios, engine speed, power and compression ratio of 8.0, which is nominal compression ratio of Indian 4-stroke gasoline operated vehicles. Theoretical studies of the performance characteristics were also carried out.

The results showed that at similar equivalence ratios engine thermal efficiency with E-85 was 2-3 percent greater than that of gasoline at the same engine operating conditions due to faster burning of ethanol. However, mass specific fuel consumptions were greater with the E-85 case than gasoline. The increase in mass fuel consumption was up to 50% with E-85, while for the case of E-10 it was 2 percent more.

At similar equivalence ratios, hydrocarbon emissions with E-85 were observed to be 3-8 percent higher as compared to gasoline throughout the range of lean equivalence ratio (ϕ varying from .65 to 1.0) due to a higher level of misfiring at leaner mixtures. Minimum HC emissions were observed to be occurring at an equivalence ratio of 0.8 to 0.9 for E-85 and also for E-10.

CO emissions depended primarily on equivalence ratio (ϕ) but an effect of fuel was apparent. On rich side CO emissions with E-85 and E-10 were lower. At equivalence ratio of 1.2, CO was lower by 16 percent with E-10 and was lower by 50 percent with E-85 as compared to gasoline due to ethanol's higher hydrogen to carbon ratio.

$NO_x$ emissions were nearly same at $\phi=1.2$ for E-10, E-85 and gasoline operations. At an equivalence ratio of less than 1.2, $NO_x$ emission were lowered by 10 percent and 26 percent respectively for E-10 and E-85 cases, as compared to neat gasoline operation due to lower burned gas temperature.

For the optimum performance, 4-6 degrees less advanced timing is required for E-85 as compared to gasoline. Single cylinder engine tests at WOT and at a constant compression ratio of 8.0 using gasoline, E-85, E-20 and E-10 showed that, with E-85, efficiency and power were greater by 3 – 5 percent and 4-7 percent respectively and also CO and $NO_x$ emissions were lowered marginally as compared to gasoline. However, HC emissions were higher with E-85. E-10 and E-20 case showed similar trends of $NO_x$ emissions.

With gasoline an ignition timing of 18 degrees before top dead centre gave lowest fuel consumption while for E-85 and E-10 case 14 degrees advance from top dead centre gave lowest fuel consumptions.

An intake air temperature of 50 degree C resulted in best performance with E-85 because of improved mixture formation. $NO_x$ emissions increased as intake air temperature increased. Minimum CO was obtained at 50 degree C while HC emissions were lowest at in the range of 50 to 60 degree C.

Increasing the compression ratio from 8 to 10 and from 8 to 12 improves the thermal efficiency by 3 percent and 8 percent respectively, but it also increases exhaust emissions of $NO_x$ and HC by 15 percent and 40 percent respectively.

Increasing idling speed increases fuel consumption and decreases CO and HC emissions. Stable operation at idling was achieved at 900 rev./min. for both gasoline and E-85.

Single cylinder engine operation with CNG showed 3 to 5 percent higher thermal efficiency and 15 percent lower BSFC. Also CO emissions were lower by 30 – 80 percent in rich zone and $NO_x$ by about 12 percent at $\phi = 1.0$.

At WOT CNG operation resulted in 10 to 12 percent lower power, However, thermal efficiency and BSFC were better with CNG as compare to gasoline. Twin spark plug operation resulted in the increase in thermal efficiency by 3 to 5 percent.

E-85 and E-10 produced higher cycle pressures throughout the expansion stroke which resulted in better thermal efficiency. Higher peak pressures with E-85 and E10 were observed throughout the speed range.

A reasonably good agreement was observed between Computer simulation results and experimental results for cylinder pressure and exhaust emission of CO and $NO_x$. Computer simulation results indicated lower cylinder gas temperature with E-85 as compared to gasoline. Further E-85 operation results in lower engine component temperatures.