INTERNATIONAL JOURNAL OF
ENERGY AND ENVIRONMENT

Volume 2, Issue 4, 2011 pp.761-770
Journal homepage: www.IJEE.IEEFoundation.org

MPFI gasoline engine combustion, performance and emission characteristics with LPG injection

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Abstract
The present work is aimed at the study of a four cylinder multipoint port fuel injection gasoline engine combustion, performance and emission characteristics which is retrofitted to run with LPG injection. The findings of the experiments suggest that higher thermal efficiency and therefore improved fuel economy can be obtained from SI engines running on LPG as opposed to gasoline. The cycle-by-cycle variation of IMEP with LPG combustion can be reduced by advancing the idle ignition timing. The results of the study at wide open throttle opening conditions indicate that there is an increase in the brake thermal efficiency with LPG use in the engine at higher operating speeds when compared to gasoline at the factory set idle ignition timing of 5° bTDC. The exhaust emissions of CO and HC have reduced considerably. But the NO_X emission has increased considerably at elevated engine speeds with LPG fuel when compared to gasoline fuel operation. The results of LPG fuel operation at various idle ignition timings indicate that advancing the timing from 5° bTDC to 6° bTDC has resulted in increased brake thermal efficiency, and reduced emissions of CO and HC, compared to retarding the idle ignition timing to 4° bTDC and 3° bTDC. However advanced idle ignition timing has an adverse effect on NO_X emissions as it increases further.

Keywords: Gasoline engine; LPG injection; Combustion; Engine performances; Exhaust emissions; Idle ignition timing.

1. Introduction
Ever increasing cost of liquid fuels derived from crude oil and increasing concern about environmental pollution have increased attention in alternative internal combustion engine fuels. Various alternative fuels suited for spark ignition (SI) engines have been known almost since the automobile was invented. These fuels can be classified as synthetic gasoline, alcohols and gaseous fuels. Gaseous fuels are promising alternative fuels due to their economical costs, high octane numbers, higher heating values and lower polluting exhaust emissions. In the point of view of the reduction of exhaust emission, liquefied petroleum gas (LPG) is a useful alternative fuel because of sufficient supply infrastructure and higher heating value. LPG fuel also has merits in the operating characteristics under the high compression ratio because it has higher octane value and lower exhaust HC emission compared with the combustion of gasoline.

The use of Liquefied Petroleum Gas (LPG) as an alternative fuel for gasoline has been studied extensively in recent years. Many studies of the emissions from LPG powered vehicles have been reported in literature. The major attractions of these fuels, in comparison with conventional liquid fuels, lie in their relatively low carbon content, causing them to burn cleanly with lower emissions of CO, CO_2 and HC. Also a higher thermal efficiency and therefore improved fuel economy can be obtained from internal combustion engines running on LPG as opposed to gasoline [1]. The major reason for better performance of LPG is attributed to its high flame propagation speed compared to gasoline. The flame propagation speed of LPG is faster than that of gasoline at the range of lean to stoichiometric equivalence ratios, but at the rich mixtures range gasoline flame speed is superior to that of LPG. Hence LPG has better combustion characteristics in lean burn engines [2]. Compared to gasoline the energy content of LPG is slightly higher. The standard calorific value of 45.7 MJ/kgK for LPG and 44 MJ/kgK for gasoline [3] has been taken for the experimental purposes. LPG fuel also has merits in the operating characteristics under the high compression ratio because it has higher octane value compared with the combustion of gasoline.

The pressure–crank angle diagrams of consecutive cycles in the combustion chamber of an S.I engine shows that variations from one cycle to another exist. Since the pressure rate is uniquely related to the combustion, the pressure variations are caused by variations in the combustion process. This cyclic variation in the combustion process is generally accepted to be caused by variations in the mixture motion, in the amounts of air and fuel fed into the cylinder and their mixing, and in mixing with residual gases and exhaust gas recirculation (EGR), especially in the vicinity of the spark plug. Cyclic variations (CV) in internal combustion engines have long been recognized as a limiting factor in engine performance, fuel efficiency and emissions. CV occurs more frequently with lean fueling and EGR, resulting in large number of misfires and partial burns thus limit the potential benefits which can be derived from these operating modes. Therefore, minimization of cyclic variations is a key requirement for operating near to or extending the effective lean limit. A small amount of cyclic variability (slow burns) can produce undesirable engine vibrations. On the other hand, a larger amount of cyclic variability (incomplete burns) leads to an increase in hydrocarbon consumption and emissions [8]. The disappearance of cycle-by-cycle variations can lead to a fuel economy improvement up to 6%. One important measure of cyclic variability, derived from pressure data, is the coefficient of variation of indicated mean effective pressure (COV_{IMEP}) [9, 10]. It is the standard deviation in IMEP divided by the mean IMEP, and is usually expressed in percent given as

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\mu_{IMEP}} \times 100 \tag{1}$$

where σ and μ are Standard deviation and mean value of IMEP respectively over a number of combustion cycles. This percentage defines the variability in indicated work per cycle, and it has been found that vehicle drivability problems usually results when COV_{IMEP} exceeds about 10% [11].

The objective of the study reported in this paper is to study the combustion, performance and emission characteristics of gasoline and LPG fuelled four cylinder MPFI engine at lean burn operating conditions. Variation of cycle-by-cycle variation of IMEP, performances and emissions with the engine speeds are discussed in this paper. The effect of varying the idle ignition timing with LPG fuel is also discussed.

2. Experimental procedure

The experimental setup consists of a four-stroke, four cylinder, S. I engine of Maruti Zen make with multi point port fuel injection (MPFI) system, which is connected to an eddy current dynamometer for loading. The specifications of the test engine are given as Appendix I. It is provided with necessary instruments for combustion pressure and crank-angle measurements. A piezo-electric pressure transducer is used for recording the cylinder pressure for number of consecutive cycles for combustion variability studies. The set up has a stand-alone panel box consisting of air box, fuel tank, manometer, fuel measuring unit, differential pressure transmitters for air and fuel flow measurements, process indicator and engine indicator. The pressure and crank-angle signals are interfaced to a digital computer through engine indicator for P-θ and P-V diagrams. Also air flow, fuel flow, temperatures and load measurements are interfaced to the computer. An AVL Digas 444 five gas Exhaust gas analyzer is used to measure the NO_X (PPM), CO (%vol.), CO_2 (%vol.) and HC (PPM) emissions in the exhaust.

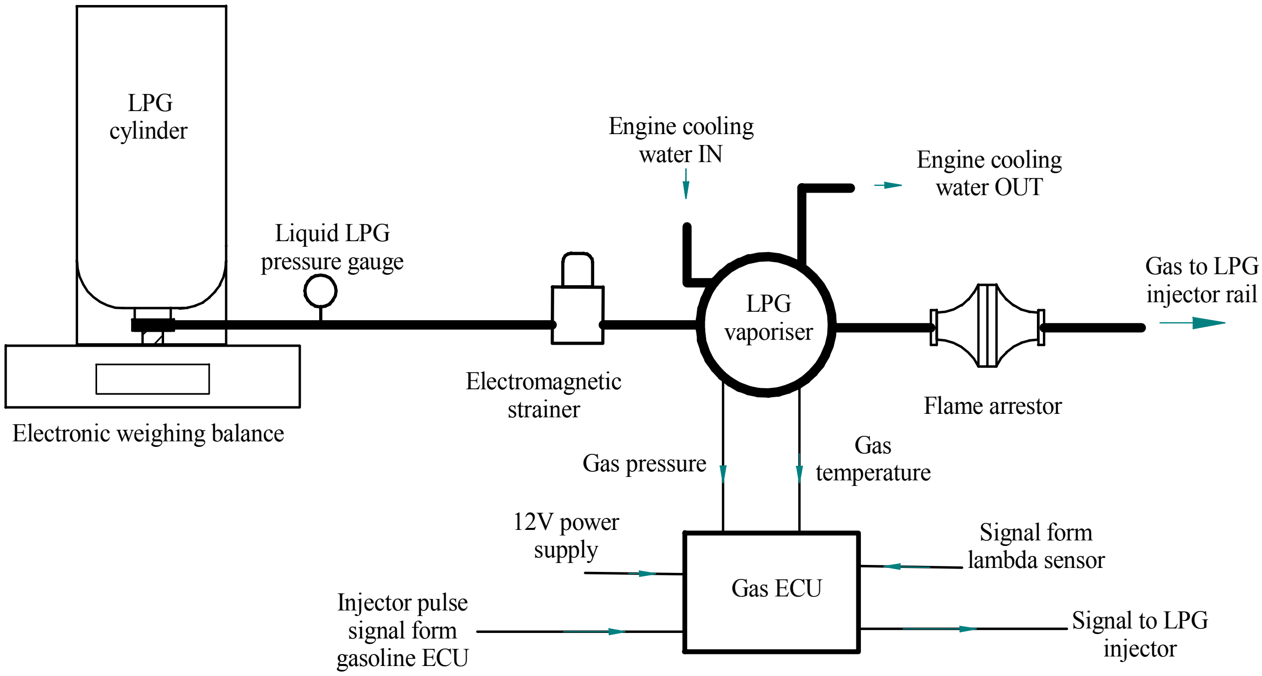

Figure 1. Block diagram of LPG injection system

The engine is operated with baseline fuel gasoline (petrol) by MPFI strategy. The engine has been retrofitted to operate with liquefied petroleum gas (LPG) fuel. Separate four gas injectors are attached to the inlet manifold near the inlet port of each cylinder for injecting LPG. The gas injectors are operated by solenoid valves driven by 12V DC power supply. A separate ECU has been used for driving the solenoid valves. The signals from the gas ECU controls the activation period of the gas injectors. The block diagram of the LPG injection system is shown in the Figure 1. Domestic LPG is stored in cylinders at a pressure of about 10 bar (Max. Vapour Pressure at 40° C is 1050 KPa gauge [4]) which weighs 14.2 kg. Unreduced pressure regulators have been fitted on to LPG cylinder which allows the gas to pass through it at high pressure. LPG consumption rate has been measured on the mass basis to minimize the error in measurement while operating the system under varying pressures. An electronic weighing balance along with a stopwatch has been used to measure the flow rate of gas. A copper pipe has been used to supply the LPG to the vaporiser. An electromagnetic strainer has been provided in the supply line to absorb the iron particles from the LPG cylinder which may travel along LPG. Power supply from the battery is given to activate the electromagnet. A LPG vaporiser which is provided in the supply line serves the purpose of vaporizing the fuel and supplies it at the required pressure. The function of the vaporiser is to transfer thermal energy into the LPG and to reduce the LPG (tank) pressure to the much lower system pressure such that the LPG evaporates to the superheated gas phase [5]. The thermal energy required is supplied by the engine cooling water which is made to pass through the vaporiser after the engine jacket circulation. To supply the required amount of gas at the required pressure for meeting the various load conditions a reference pressure from the engine inlet manifold is also connected to the evaporator. To avoid flash back, a flame arrestor has been connected in series in the fuel supply line between the injector lines and the vaporiser. From the flame arrestor LPG is passed to the gas injectors (four numbers) which are solenoid controlled valves, the functioning of which are controlled by the gas ECU signals.

The main function of the gas ECU is to open and close the gas injector at appropriate time to control the duration of injection. Concept of working of gas ECU for bi-fuel application is based on master slave theory [6]. The gasoline injector opening signal pulse from the preinstalled Gasoline ECU has been fed to gas ECU as an input. The gas ECU modifies the gasoline pulse width using a correction factor and sends it to gas injectors. This correction factor is calculated based on the fact that the densities of liquid gasoline and gaseous LPG are different. It also takes into account the signals from the other sensors such as exhaust lambda sensor (indicating oxygen content in the exhaust gases) and inlet manifold absolute pressure indicating the engine load. When the engine is running with LPG as fuel, the emulator system in the gas ECU cuts off gasoline injection signals and gives the emulated signal to the gasoline ECU so that it doesn’t give a fault signal.

Experiments have been conducted in the above described setup with baseline gasoline and LPG fuels with factory set idle ignition timing of 5° bTDC to evaluate the performance characteristics. A five gas exhaust gas analyzer for measuring engine emissions is used which can measure CO, CO_2, unburned HC and NO_X. The experiments have been conducted with different throttle positions & engine speed combinations. Later the experiments have been conducted to investigate the effect of variation of idle ignition timings on engine performance and emissions with LPG fuel. The idle ignition timings of 6° bTDC, 4° bTDC and 3° bTDC are used as testing points. The recorded pressure-crank angle data for 100 consecutive cycles were used for calculating the indicated mean effective pressures (IMEP) and COV_{IMEP} with the help of simple computer program written in visual C++. Analysis of the obtained data was performed and results were plotted.

3. Results and discussion

In this section the results of various experiments conducted are presented for the operating conditions at wide open throttle position. The gasoline ECU and the gas ECU have the fuel mapping programmed in such a way that it sends signals to injectors such that the engine develops constant IMEP at a particular throttle position. Hence the engine generates constant torque for a particular throttle valve opening position. The flame propagation speed of LPG is faster than that of gasoline in the range of stoichiometric equivalence ratios. However in the range of rich equivalence ratio conditions, the speed of flame propagation for the gasoline is superior to that of LPG [2]. The engine runs with too lean mixture at lower speeds for both LPG and gasoline at which both these fuels have same flame speed. At higher engine speeds the equivalence ratio of LPG-air mixture is nearly one (stoichiometric) and for gasoline, mixture is slightly rich. Hence at higher engine speeds, flames generated by LPG propagate faster than that of gasoline.

3.1 Combustion studies

3.1.1 Indicated Mean Effective Pressure(IMEP)

The indicated mean effective pressure developed by the engine remains constant for a particular throttle valve opening position. Figures 2 and 3 show the general trend of the variations of average IMEP at different engine speeds at WOT and half throttle opening operations respectively. The trends are for the standard idle ignition timing of 5° bTDC. It can be observed that LPG fuel combustion develops slightly lower IMEP at lower engine speeds and slightly higher IMEP at higher engine speeds compared to gasoline fuel combustion. But the variation is marginal. Hence the engine doesn't suffer much power loss.

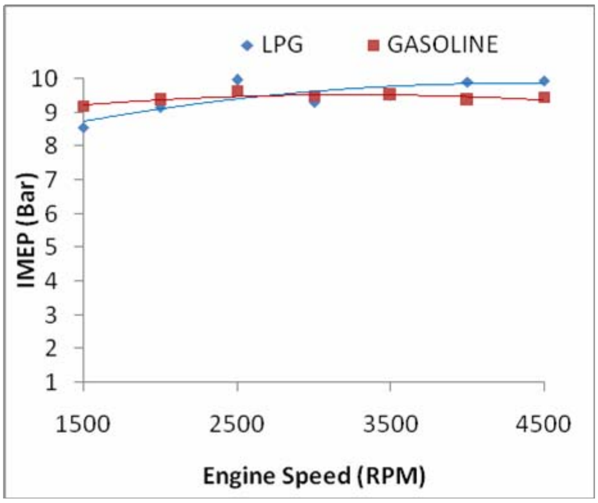

Figure 2. Variation of IMEP with engine speed at WOT opening

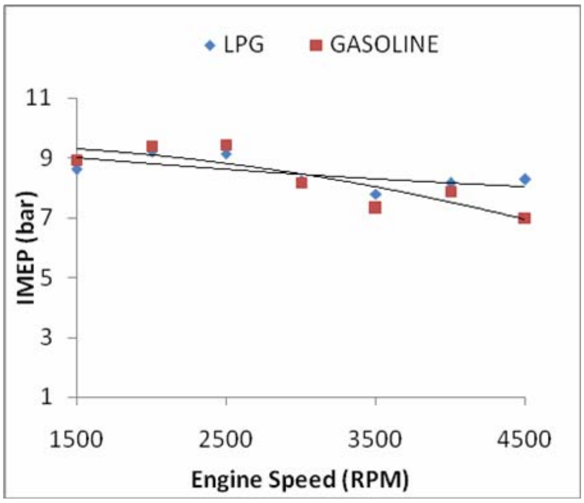

Figure 3. Variation of IMEP with engine speed at half throttle opening

3.1.2 COV of IMEP

The in-cylinder pressure data are being used for determining the coefficient of variation of IMEP. Figures 4 and 5 show the variation of COV_{IMEP} for LPG and gasoline fuel operation at WOT and half throttle conditions respectively. The WOT trends indicate that operation with LPG at idle ignition timing of 5° bTDC, which is the original set value, leads to a higher COV compared to other idle ignition timing operations and even the baseline gasoline fuel operation. Advancing the idle ignition timing to 6° bTDC has a positive impact on the COV as the higher laminar flame speed of LPG and the additional combustion duration available ensures more complete combustion. At part throttle operation, LPG operation at all idle ignition timings have higher COV than baseline gasoline value. Still competitively advanced idle timing has the closest trend to that of gasoline. Hence by advancing the idle ignition timing to 6° bTDC, better combustion and thus improved engine performance may be obtained from combustion variation point of view.

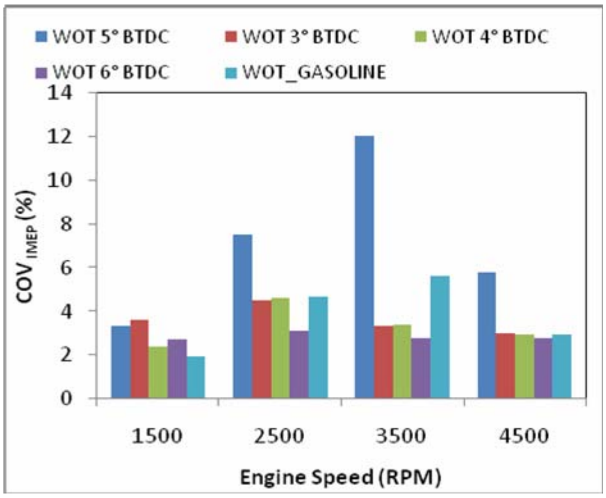

Figure 4. Variation of COV_{IMEP} with engine speed at WOT opening

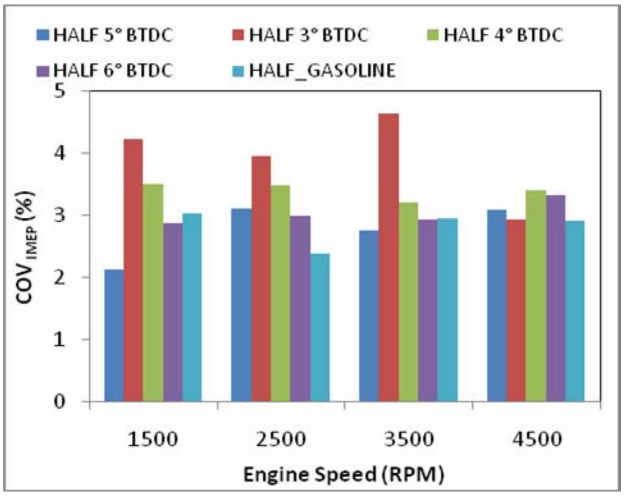

Figure 5. Variation of COV_{IMEP} with engine speed at half throttle opening

3.2 Performance and emissions at idle ignition timing of 5° bTDC

From the results of the experiments with LPG and gasoline at idle ignition timing of 5° bTDC, it is observed that the performance of LPG is less than gasoline at lower engine speeds even though a significant reduction is achieved in engine exhaust emissions and also better performance at higher engine speeds.

3.2.1 Brake thermal efficiency

The Variations of brake thermal efficiency for gasoline and LPG fuels at WOT operating condition is shown in Figure 6. The characteristics of these curves shows that the brake thermal efficiency is more for gasoline at lower speeds and in the speed range of 3000 RPM to 4000 RPM both fuels have nearly same efficiencies. LPG combustion exhibits higher thermal efficiency than gasoline after 3500 RPM. Since the ignition temperature of LPG is higher than that of gasoline, ignition delay and thus combustion duration is more for LPG [1]. This decreases the average burning rate. To accommodate this effect, engine consumes more fuel which in turn decreases its efficiency. Hence LPG has lower efficiency at lower engine speeds. At higher engine speeds, the higher flame propagation speed of LPG negates the effect of ignition temperature. Here the time duration for each cycle is very low which demands more rate of combustion to get the complete combustion of the fuel. The lower propagation speeds of gasoline flames cannot afford the requisite combustion rate; instead the engine takes more fuel to generate the required torque. The collective outcome of these factors lowers brake thermal efficiency of the engine for gasoline at higher engine speeds.

3.2.2 Carbon monoxide (CO) emission

The variation of carbon monoxide (CO) emission with speed at WOT condition is shown in Figure 7. From the figure it can be inferred that CO emissions with LPG is far less than that with gasoline. The CO emissions are reduced from an average value of 5% to 1.5% with the use of LPG compared to that with gasoline. For LPG, at all throttle positions the CO emissions is found to be less than 2% which is well within the limits of EURO V pollution norms. The higher flame propagation speed and proper mixing of gaseous LPG with the air enhances the combustion and thus reduces the CO emissions. At wide throttle valve opening positions, the total quantity of air and fuel which is inducted into the engine is more than that of half or quarter throttle positions. The lower combustion duration at higher engine speeds is not sufficient for the complete combustion of fuel. As a result of this, CO emissions increase at higher engine speeds.

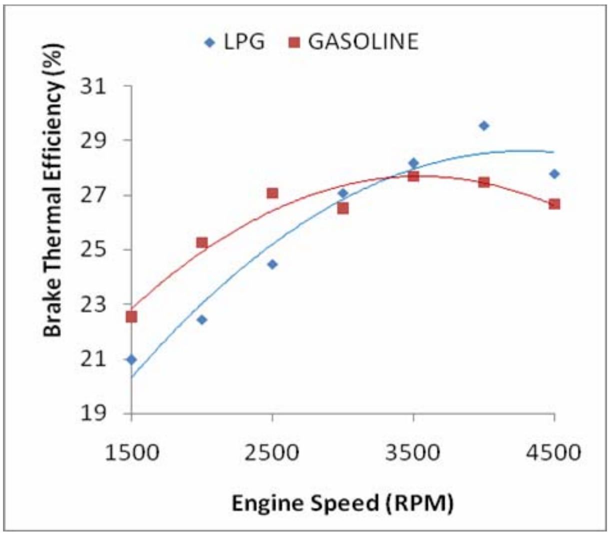

Figure 6. Variation of brake thermal efficiency with engine speed at WOT

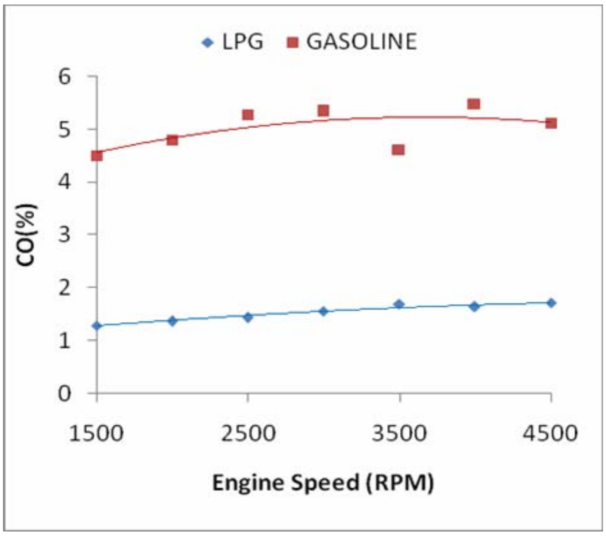

Figure 7. Variation of CO with engine speed at WOT

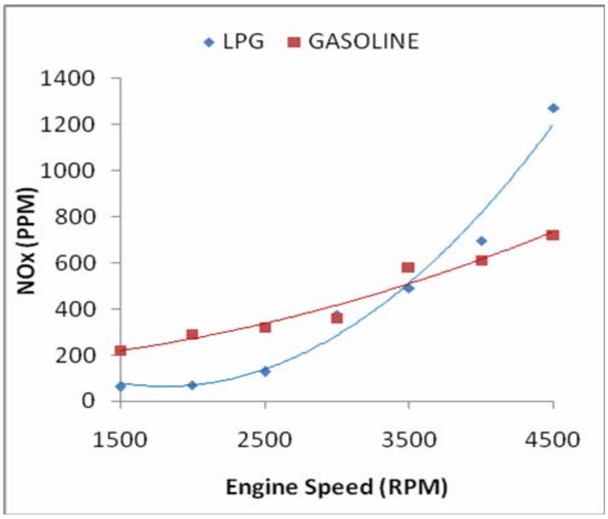

Figure 8. Variation of NO_X with engine speed at WOT

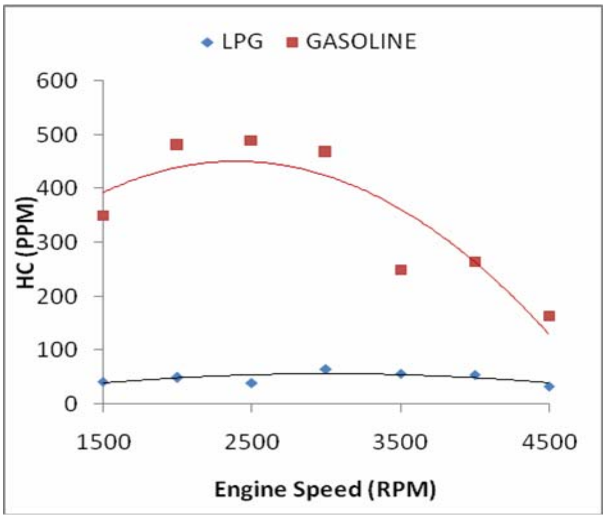

Figure 9. Variation of HC with engine speed at WOT

3.2.3 Oxides of Nitrogen (NO_X) emission

The variation of NO_X at different engine speeds at wide open throttle opening is shown in Figure 8. Generally as the engine speed increases the NO_X emissions also show an increasing trend. But LPG combustion results in more elevated NO_X emissions compared to gasoline at higher engine speeds after 3500 RPM. Higher flame propagation speed of LPG and proper mixing of gaseous fuels with air causes an increase in the burning rate of the fuel and thus results in the complete combustion of the fuel. Hence the cylinder pressures and combustion temperatures of LPG are higher than those obtained for gasoline [7]. As a final outcome of this, more NO_X emissions occur in LPG combustion at higher speeds. The value of NO_X emission of LPG is almost double the emission of gasoline at all throttle positions at a speed of 4500 RPM. The reason for lower NO_X emissions for gasoline and LPG at lower engine speeds may be attributed to lower peak combustion temperature due to too lean mixtures as it might result in quenching effect.

3.2.4 Hydrocarbons (HC) emission

Figure 9 depicts the variation of un-burnt hydrocarbon emissions with engine speed at WOT condition. Contrary to gasoline which shows a drastic reduction trend of HC as the engine speed increases, LPG values are almost constant. LPG combustion results in drastic reduction in HC emissions when compared to gasoline values at all the operating speeds. Oxidation of unburned HC within the cylinder and in exhaust pipe line may be enhanced at higher speeds, since expansion and exhaust stroke gas temperatures increases significantly due to reduced heat transfer [4]. This may be the reason for the gasoline HC emission trend. The fact that LPG combustion being more complete than that of gasoline has resulted in substantial reduction in HC emission at all throttle open conditions.

3.3 Performance and emissions at various idle ignition timings

To unearth the possibilities of enhancing the performance of LPG at lower engine speeds, experiments have been conducted with different idle ignition timings. The different idle ignition timings set for LPG are 3° bTDC, 4° bTDC and 6° bTDC. Performances were found to be reducing drastically at 3° bTDC and tendency of knocking was observed when ignition timing was set at values above 6° bTDC. Hence experimental studies on performances and emission characteristics of LPG at different idle ignition timings are limited to the range of 3 to 6° bTDC.

3.3.1 Brake thermal efficiency

Figure 10 shows the variations of the brake thermal efficiency with engine speed for various ignition timings. Advancing the ignition timing increases the combustion duration which helps to overcome the effect of ignition delay. This enhances the cylinder pressure and power output at lower engine speeds. Hence efficiency is highest for 6° bTDC at lower engine speeds. The equivalence ratio is nearly stoichiometric at higher engine speeds at all ignition timings. At higher engine speeds, the high propagation speeds of LPG flames allow the fuels to burn with high rate of combustion. Hence at higher engine speeds, the efficiency is more or less same irrespective of the ignition timing. The observations from the graph show that the combustion of fuel with 6° bTDC provides the maximum average efficiency over the given range of engine speeds.

3.3.2 Brake specific energy consumption

The variations of BSEC with engine speeds at various ignition timings is depicted in Figure 11 at wide open throttle opening condition. The incomplete combustion of fuels results in the higher specific energy consumptions for 3° bTDC and 4° bTDC. The optimal energy consumption can be achieved by running the engine at 3500 RPM with ignition timing being set at 6° bTDC.

3.3.3 Carbon monoxide emission

The comparison of CO emissions of LPG and gasoline at various engine speeds are shown in the Figure 12 with various idle ignition timings at WOT condition. The combustion of fuel primarily depends on two factors. Air-fuel ratio being the first and flame speed is the other one. Higher engine power output requires lower values of equivalence ratios whereas a value above stoichiometric ratio produces large amount of CO emissions. A fine trade-off between these two factors gives better results. At lower engine speeds, equivalence ratio is too lean at all throttle positions which results in complete combustion of the fuel and hence CO emissions are less. As speed increases air-fuel ratio increases and hence the CO emissions. At higher engine speeds, equivalence ratio is stoichiometric and hence flame propagation speed is higher. The high flame propagation speed results in complete combustion of fuel and thus engine produces very low values of CO emissions. This trend can be observed at higher engine speeds at half throttles. At wider throttle valve opening positions, the total quantity of air and fuel which is inducted into the cylinder is more than that of half or quarter throttle positions. The lower combustion duration at higher engine speeds is not sufficient for the complete combustion of fuel. As a result of this, CO emissions increase at higher engine speeds. Retarding the ignition timings enhances the chances of incomplete combustion which increases the CO emissions. Hence higher CO emissions are observed at 3° bTDC and 4° bTDC. Here again the effective combustion may the reason for reduced CO emissions when the idle ignition timing is advanced to 6° bTDC.

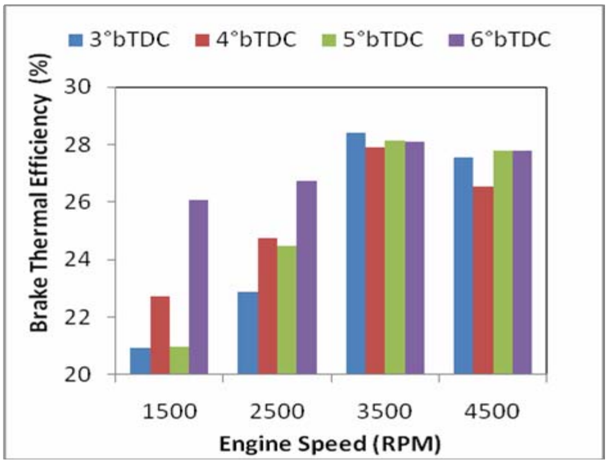

Figure 10. Variation of brake thermal efficiency with engine speed at WOT for different ignition timings

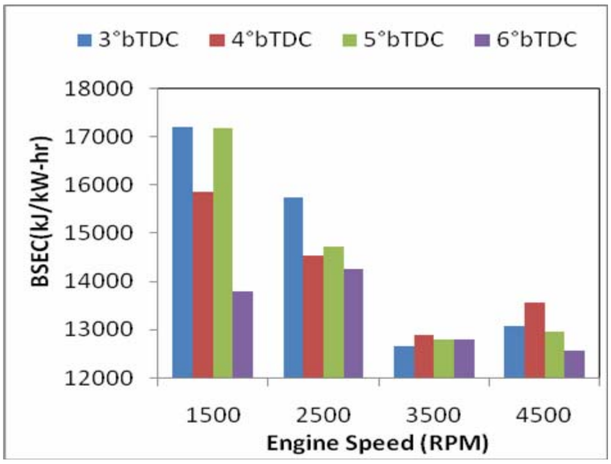

Figure 11. Variation of brake specific energy consumption with engine speed at WOT for different ignition timings

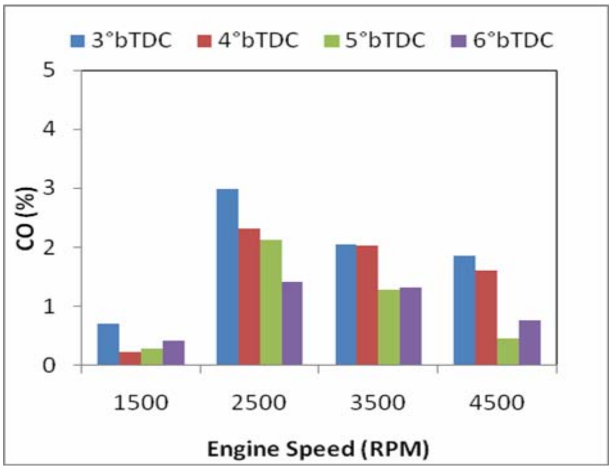

Figure 12. Variation of CO with engine speed at WOT for different ignition timings

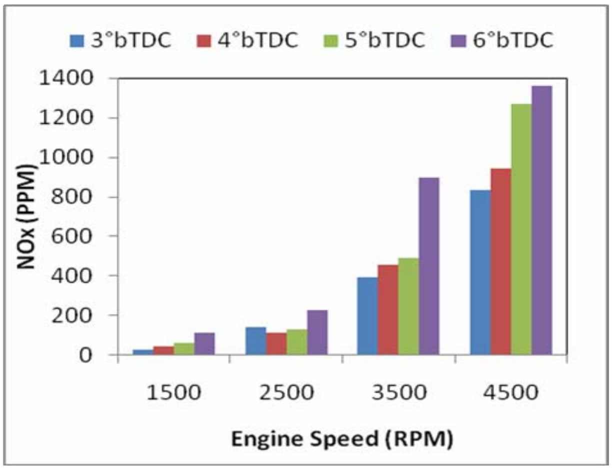

Figure 13. Variation of NO_X with engine speed at WOT for different ignition timings

3.3.4 Oxides of Nitrogen emission

Variation of NO_X with engine speeds at various idle ignition timing at WOT are shown in Figure 13. At lower engine speeds mixture is lean for all throttle positions. Hence at lower RPM, combustion of fuel remains same irrespective of the ignition timings and so is the case of combustion temperature. The combustion temperature at lower engine speeds is low at all throttle valve positions. As the engine speed increases, the air-fuel ratio of the mixture gradually increases and eventually becomes stoichiometric. LPG has higher flame speed at stoichiometric ratio than that at lean mixture. Since NO_X is a function of temperature, its value increases with engine speed and is highest at 4500 RPM for all operating conditions. As the ignition point retards afterburning increases at higher RPM which lower the value of combustion temperature. The lower value of NO_X emission at 3° bTDC attributes to its lower value of combustion temperature due to incomplete combustion. It can be observed that at wide open throttle, the value of NO_X emission for 4500 RPM at 6° bTDC is around 1400 PPM where as it is nearly 500 PPM at 3° bTDC.

3.3.5 Unburnt hydrocarbon emission

Figure 14 depicts the variation in hydrocarbon emissions at various idle ignition timing at WOT. When engine runs with 6° bTDC, combustion duration is more. This increases the chances of complete combustion and thus reduces the HC emissions. Hence HC emissions are found to be less when idle ignition timing is set at 6° bTDC. The incomplete combustion of fuel results in the higher emissions of hydrocarbons at 3° bTDC and 4° bTDC idle ignition timings. The highest HC emissions are observed when ignition timing is set at 3° bTDC. As the speed increases, both equivalence ratio and flame propagation speeds also increases. At lower engine speeds, the effect of flame propagation speed is less which results in incomplete combustion. Hence HC emission increases at lower engine speeds. The high flame propagation speeds at higher engine speeds drive the engine to complete combustion and thus reduces the HC emissions.

3.3.6 Carbon dioxide emission

Figure 15 shows the trends of carbon dioxide emissions for LPG at various idle ignition timings. It can be observed that the value of CO_2 emission increases with engine speed. This can be seen as an outcome of the larger fuel air ratio at higher engine speeds. There is no considerable variation in CO_2 emissions at different ignition timings.

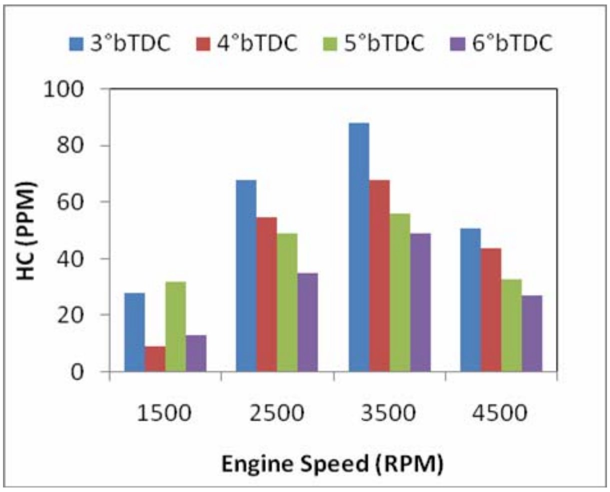

Figure 14. Variation of HC with engine speed at WOT for different ignition timings

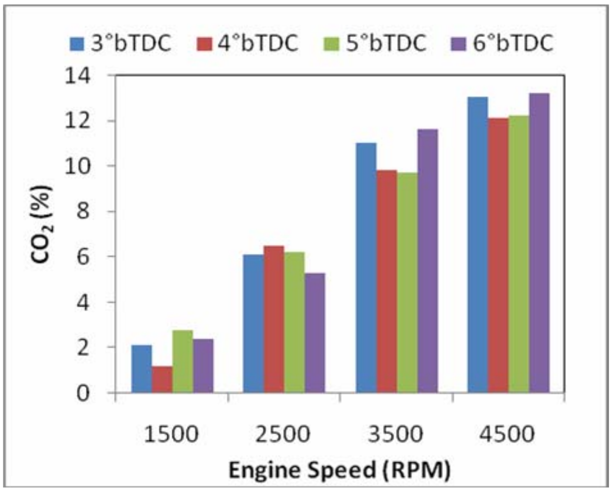

Figure 15. Variation of CO_2 with engine speed at WOT for different ignition timings

4. Conclusions

From the extensive study conducted on the MPFI 4 cylinder SI engine with gasoline and LPG as fuels at different ignition timings the following conclusions can be drawn:

- The COV of IMEP can be reduced with advancing the idle ignition timing from 5° bTDC to 6° bTDC.
- At factory set idle ignition timing of 5° bTDC at WOT condition, brake thermal efficiency is more for gasoline at lower engine speeds and it is more for LPG at higher engine speeds. LPG has a positive effect on the CO and HC emissions when compared to gasoline. The CO emission has reduced from an average value of 5 % to around 1 % and corresponding change in HC is noticed was from 350 PPM to 50 PPM when LPG is used instead of gasoline. As a result of higher combustion temperature, comparatively more NO_X emissions occur with LPG combustion than with gasoline combustion. The NO_X emission with LPG is almost double when compared to that with gasoline at higher engine speeds.
- Advancing the idle ignition timings gives better thermal efficiency at lower engine speeds and thermal efficiency is nearly same for all ignition timings at higher engine speeds. When engine runs with LPG, better performance has been observed when ignition timing is set at 6° bTDC. Advancing the idle ignition timing has also resulted in reduced CO and HC emissions. But the advanced ignition timing shows an increase in NO_X emissions. Retarding the ignition timing achieves lesser NO_X emissions at higher engine speeds. At wide open throttle opening condition, the value of NO_X emission for 6° bTDC is around 1400 PPM where as it is nearly 500 PPM for 3° bTDC at 4500 RPM speed.

Nomenclature

BSEC	Brake specific energy consump. (kJ/kW-hr)	bTDC	Before TDC
CO	Carbon monoxide (% vol.)	ECU	Electronic control unit
HC	Hydro carbon (PPM)	IMEP	Indicated mean effective pressure (bar)
LPG	Liquefied petroleum gas	MPFI	Multipoint port fuel injection
NO_X	Oxides of nitrogen (PPM)		

Appendix I: Specifications of the experimental setup

Engine Make	: Maruti Zen MPFI
Cylinders	: 4
No. of strokes	: 4
Fuel	: Petrol (MPFI)
Power	: 44.5kW @ 6000 RPM
Torque	: 59Nm @ 2500rpm
Stroke	: 61mm
Bore	: 72mm
Capacity	: 993 cc
Compression ratio	: 9.4:1

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