International Journal of the Physical Sciences Vol. 6(8), pp. 1905-1914, 18 April, 2011
Available online at http://www.academicjournals.org/IJPS
DOI: 10.5897/IJPS11.288
ISSN 1992 - 1950 ©2011 Academic Journals

Full Length Research Paper

Performance and exhaust emission characteristics of a diesel engine running with LPG

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Accepted 31 March, 2011

This paper investigates the effects of LPG injection during air inlet period on emissions and performance characteristics. The engine has been modified to determine the best LPG composition for dual operation in order to improve the emissions quality while maintaining high thermal efficiency in comparison to a conventional diesel engine. An electronic controlled LPG injection system has been developed for this purpose. LPG injection rate were selected as 5, 10, 15 and 25% on a mass base. Minimum SFC and maximum brake efficiency obtained with 15% LPG between 1400 and 1800 rpm engine speeds. Optimum injection rates is found at 5% LPG in terms of exhausts emissions and performance. At this injection rate, SFC, NOx and smoke emissions decreased by 9, 27.6 and 20% at the test speed of 1600 rpm, respectively.

Key words: Diesel engine, dual fuel, LPG, emissions, performance.

INTRODUCTION

LPG is considered to be one of the most promising alternative fuels not only as a substitute for petroleum but also as a means of reducing NOx, soot and particulate matter. Therefore, it is more economical and of environmental advantage to use gaseous fuel in diesel engines adopted for the dual fuel concept. However, when LPG is used in the conventional diesel engine, due to its lower cetane number, self-ignition is always a problem. Thus, if LPG is to be used as an alternative to diesel, the cetane number needs to be improved with additives or other positive means of initiating combustion. Various studies on alternative fuels have been conducted for reducing consumption of Diesel fuel and important pollutant emissions such as nitrogen oxide (NOx) and particulate emissions (Snelgrove et al., 1996; Homeyer et al., 2002; Murillo et al., 2005).

As an alternative to Diesel fuel, LPG can be used in dual fuel compression ignition engines as a primary fuel (Mohamed et al., 2008; Miller et al., 2007; Li et al., 2008). In a dual fuel compressed ignition system (CI), the engine is operated with LPG as primary fuel providing with a pilot amount of diesel fuel is used as an ignition source. The LPG is inducted along with the intake air and is compressed as in a conventional diesel engine. Due to its high auto-ignition temperature, the mixture of air and LPG does not auto-ignite. However, a small amount of Diesel fuel needs to be injected near the end of the compression stroke to ignite the gaseous mixture. The pilot Diesel fuel injected for ignition can only contribute to a small fraction of the engine power output (Hountalas and Papagiannakis, 2000, 2003).

Karim and Zhiganag (1992), Karim and Zhaoda (1988) has reported that in dual fuel engines under low loads, when the LPG concentration is low, the ignition delay of the pilot fuel increases leading to some unburned LPG and poor emission quality. Due to dilute LPG-air mixture, poor combustion quality of LPG under low loads may often result in high CO and unburned HC emissions. Whereas, at high loads, with increased admission of LPG

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Nomenclature: Bx, Total systematic uncertainty; **BDC**, bottom dead centre; **CA**, crank angle (0); **CO,** carbon monoxide; **Cv,** specific heat for constant volume (kJ/kgK); **HC,** Hydrocarbon; **L,** moment arm (m); **LPG,** liquefied petroleum gas; **SFC,** specific fuel consumption (g/kWh). Standard random error of the mean: **N,** number of repetition of a measurement; **NOx,** nitrous oxides (ppm) **uR,** total uncertainty involving all the systematic and random uncertainties; **P**, pressure (bar) **Px,** total random uncertainty; **TDC,** top dead center; **T,** temperature (K) **t,** estimator; **rpm,** revolution per minute; **γ,** specific heats ratio.

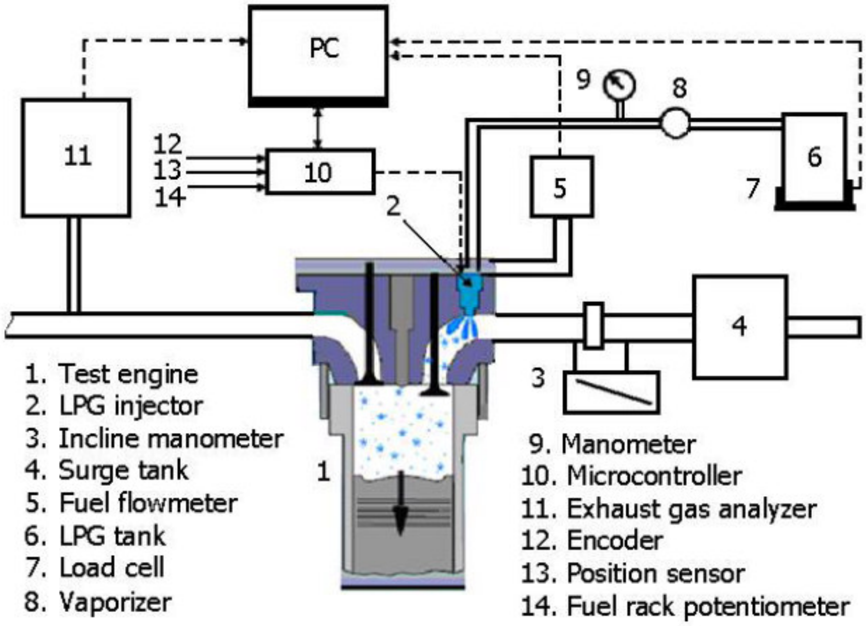


Figure 1. Block diagram of the experimental setup.

Table 1. Specification of the test engine.

Engine type	Super star water cooled
Bore (mm)	108
Stroke (mm)	100
Cylinder number	1
Stroke volume (l)	0.92
Power – 1500 rpm kW	14.7
Injection pressure (bar)	175
Injection advance (CA bTDC)	35
Maximum speed (rpm)	2500
Cooling type	Water
Injection type	DI

can result in uncontrolled reaction rates near the pilot fuel injection leading to knock.

Qi et al. (2007) conducted an experimental investigation on a single cylinder DI diesel engine modified to operate under LPG–Diesel dual fuel conditions. Using LPG–Diesel blends of various rates; 0, 10, 20, 30 and 40%, they compressed LPG by pressured N_2 gas to mix with the diesel fuel in a liquid form. They concluded that LPG-Diesel blended fuel combustion is a promising technique for controlling both NOx and smoke emissions even on existing DI diesel engines.

The purpose of the present study is to investigate the effects of LPG addition into the cylinder during inlet period on emissions and performance characteristics in a dual fuel engine running on diesel fuel. The optimum LPG rate for dual fuel operation is investigated in order to improve the emission quality while maintaining high thermal efficiency in comparison to a conventional diesel engine. The amount of LPG was determined on the mass basis with the percentages of 5, 10, 15 and 25 of standard engine fuel consumption for all over the engine speed tested for full load condition.

MATERIALS AND METHODS

Experimental setup and test conditions

The engine used in this study is a single cylinder, naturally aspirated, four-stroke, and water cooled, direct injection diesel engine with a bowl in piston combustion chamber. Figure 1 shows the experimental setup and the specifications of the engine are given in Table 1. For liquid fuel injection, a high-pressure fuel pump

Table 2. The errors in parameters and total uncertainties with 95% confidential level.

Parameters	Systematic errors, ±
Load (N)	0.1
Speed (rpm)	1.0
Time (s)	0.1
Temperature (℃)	1
Fuel consumption (g)	0.1
NOx (ppm)	5 ppm
CO (%)	0.06
HC (ppm)	12 ppm
Smoke number (%)	1
	Total uncertainity (%)
Specific fuel consumption	1.5
Brake torque (Nm)	1.0
Brake power	1.5
Volumetric efficiency	2.5

was used. The injector nozzle is located in the center of the combustion chamber which has an opening pressure of 175 bar. In order to measuring brake torque, the engine is coupled with a hydraulic dynamometer of 50 kW absorbing capacity. Using a load cell with the precision of 0.1 N. full load tests were conducted at the engine speeds of 1200, 1400, 1600, 1800, 2000 and 2200 rpm.

During tests, the inlet water pressure of dynamometer was kept constant at 3 bars. Outlet temperature of cooling water is also kept constant at 70℃. Fuel consumption was measured by an electronically controlled balance with the precision of ±0.1 g. The average and instant fuel consumption were directly transferred to a PC via RS-232 serial port of an electronic balance.

At each operating condition, after allowing sufficient time for the engine to stabilize, dynamometer load, engine speed, fuel and airflow rates were recorded. At stabilized condition, the engine was run for a period of 2 min to obtain 10 reading for each parameter concerned. Then the average values were obtained. After the load tests were conducted for the standard engine with injection timing of 35° crank angle (CA), the same procedure was followed to test engine with LPG injection system.

An electronically controlled LPG injection system was developed for dual fuel operation. The system composes of a rail, solenoid type injector, electronic control unit and an absolute encoder. The encoder gives a reference pulse signal (5 V) to determine the engine top dead center (TDC). Before testing, the TDC sign on the flywheel of the test engine was matched with the reference output of the encoder by using digital oscilloscope.

Fuel consumption values, were precisely measured during standard diesel engine test at full load condition and simultaneously transferred to the ECU unit of the system. The fuel consumption data was used as a reference in determining the LPG injection rate.

An injection test system was developed to control the time lag and injection time to determine the amount of injection fuel. The injection system consisted of an electrically driven fuel pump, an encoder with DC motor, GR-200 model balance with 0.0001 g resolution and a digital oscilloscope. Thus, the real injector opening time is equal to the calculated injection time which corresponding the amount of fuel, which will be injected into the manifold plus time lag of the injector coils to lift the needle. The time lag of the solenoid injector was 0.167 ms in average between 1200 and 2200 rpm. The amount of LPG was determined on the basis of mass base with the ratio of 5, 10, 15 and 25% of standard engine fuel consumption. A MATLAB code has been developed to electronically control the injection rates via PC.

NOx, CO and HC emissions were measured with MRU Spectra 1600 L gas analyzer. Smoke emissions were measured with Bilsa Mode 5000 opacimeter. Special emphasis has been given to possible measurement errors for pollutant emissions. To ensure the accuracy of the measured values, the gas analyzer was calibrated before each measurement using reference gases. The smoke meter was also set to zero point before each measurement.

A pressurized closed circuit cooling system was used to cool the engine. The outlet temperature of cooling water was kept at 70℃. The amount of air is measured by using rotameter-surge tank set. The fuel consumption was measured by computer controlled mass fuel flow meter.

LPG fuel composition used in the tests was 30% propane and 70% butane (% by volume). The specification of the LPG and Diesel fuel are given in Table 2.

Data acquisition systems

Cylinder pressure versus crank angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the combustion progress. The pressure time history was measured by Kistler 6061B type water cooled piezo-electric pressure transducer with the sensitivity of 25 pC/bar, PCB Piezotronic 422E15 type charge amplifier with the sensitivity of 0.5 mV/pC, National Instrument PCB ICP480C02 signal conditioner, National Instrument A1-16E-4 data acquisition card which collects data at a rate of 500 kHz per one channel and Koyo TRD J1000-RZ model encoder giving 1000 pulse per revolution. A MATLAB code has been developed to sample and store the pressure data fed into the channel of data acquisition card. The sampling program has been written to collect 400,000 pressure-time data points at a sampling rate of 80,000 data per second. An average of 100 consecutive engine cycles was taken into account for determining the cylinder pressure. The data were collected at 1600 rpm, at which maximum brake torque was obtained.

Uncertainty analysis

The measurements were repeated 10 times for each test and the

Table 3. LPG and diesel fuel properties.

Property	30% Propane/ 70% Butane	Diesel
Specific gravity, for gas at 15℃, 1 atm.	1.86	-
Density, for liquid (kg/m^3) at 15℃, 1 atm.	562	833
Cetane number	<3	45-55
Calorific value (kJ/kg)	45562	43162
Calorific value (kJ/Nm^3)	25960	35950
Theoretical air requirement (kg/kg)	15.5	14.5
Boiling point (℃)	-13	260
Sulphur content by weight (%)	0.02	14.5
Lower flammability limit (vol. %)	2	0.6
Upper flammability limit (vol. %)	8.7	5.6

average value has been taken into account. Data that lie outside the probability of normal variations will incorrectly offset the mean value, inflate the random error estimates. Therefore, a Chauvenet criterion was used to detect such data points known as outliers (Ayhan, 2009).

Performance parameters involve a number of independent variables. Thus, the uncertainty of each variable affects the result by propagating. Each independent variable x_i involves systematic uncertainty B_{xi}, and measurement standard random uncertainty P_{xi}. The true mean involving total uncertainty is

$$R = \overline{R} \pm u_R \quad (P\%) \tag{1}$$

(Ayhan, 2009).

Where $\overline{R}$ is the mean value of the result. The propagation of random and systematic uncertainties through the variables to the result are given, respectively by

$$P_x = \left(\sum_{i=1}^{N} [\theta_i P_{xi}]^2 \right)^{1/2} \tag{2}$$

(Ayhan, 2009).

$$B_x = \left(\sum_{i=1}^{N} [\theta_i B_{xi}]^2 \right)^{1/2} \tag{3}$$

(Ayhan, 2009).

Where θ_i is the sensitivity index. Combined uncertainty u_R can be as:

$$u_R = \pm \left[B_x^2 + (t_{v,99} P_x)^2 \right]^{1/2} \tag{4}$$

(Ayhan, 2009).

Where t is an estimator and can be obtained from tables called the Student's t distribution tables depending on the confidence levels. All the total uncertainties of performance characteristics are calculated as described previously. The accuracy and total uncertainty of characteristics calculated with respect to measured values are shown in Table 3.

RESULTS AND DISCUSSION

The main scope of the present work is to examine the effect of different rates of LPG injection into the engine manifold during inlet period on performance and the pollutant emissions. The results obtained from all test conditions are given subsequently.

The effects on performance parameters

Figures 2 compares brake torques variations of the engine working with pure diesel fuel and LPG added diesel, respectively. LPG injections into the engine through inlet manifold considerably increased the brake power and torque. As seen in the figure that as the injection rate of LPG increases, the power and the torque increase considerably. While maximum brake torque was obtained as 65.2 Nm at 1600 rpm with standard diesel operation, with the mixture of 25% LPG, the maximum torque was achieved at 1400 rpm. Compared to standard diesel fuel, increase amount of brake torques were measured as 7.2% at 2000 rpm, 9.6% at 1600 rpm, 11.9% at 1400 rpm and 13% at 1400 rpm with the LPG injection rates of 5, 10, 15 and 25%, respectively. The possible reason of this is that insertion of an additional energy input into the cylinder during dual fuel operation. Injecting of LPG in the form of gas might also be contributed towards improvement of combustion in the cylinder. Most probably, the combined effects of these two can lead to increase the power much more compare to that of obtained with pure diesel operation at full load conditions.

Figure 3 shows the variations in SFC with the LPG injection rates of 5, 10, 15 and 25% compared to that of standard engine at full load conditions. Fuel saving were improved with the injection of LPG for all over the engine speed tested. Maximum improvement in SFC is found with 15% LPG injection rate with the engine speed of 1400 rpm.

The maximum decrease in SFC was measured as 1.4% at 1200 rpm, 5.7% at 1400 rpm, 8.7% at 1400 rpm and 8.2% at 1400 rpm for the LPG injection rates of 5, 10, 15 and 25%, respectively.

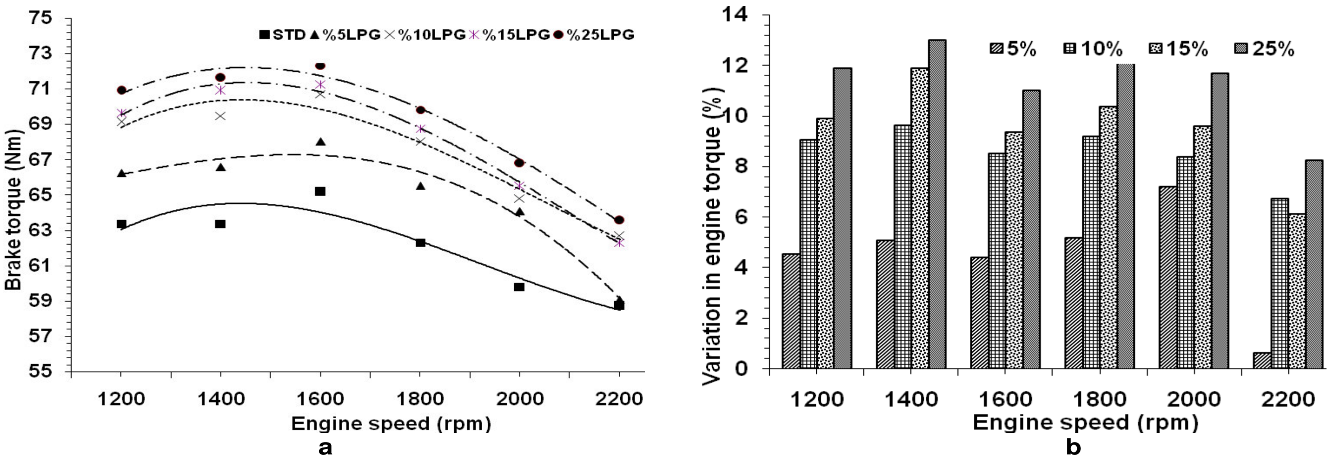


Figure 2a. The effects of LPG injection on brake torque compare to diesel fuel operation. **b.** Variations in brake torque depending on various LPG injection rates.

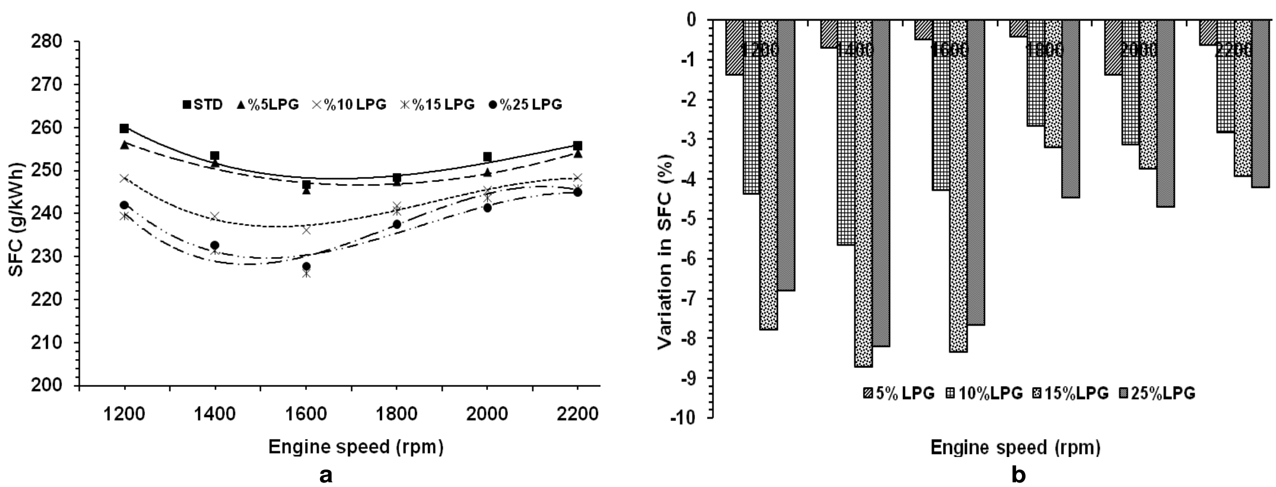


Figure 3a. The Effects of LPG injection on SFC. **3b.** Variations of SFC depending on various LPG injection rates compare to diesel fuel operation.

Volumetric efficiency

Figure 4 shows the variation of volumetric efficiency at various LPG injection rates. Volumetric efficiency decreases with increasing LPG injection rate. Maximum increase in volumetric efficiency was observed with the 25% which LPG varied from 7% (1200 rpm) to 6.3% (2000 rpm). On the other hand minimum decrease is observed at 5% LPG injection rate. At this rate, the decrease in the efficiency was found as 2.1% in average. Although volumetric efficiency slightly decreases, the reasons for improvement in performance and SFC at all the engine speeds can be explained from a view of number of aspects. One of these is that LPG is directly injected just before the inlet valve during suction period. This enables the air and LPG mixture to mix homogeneously and to improve combustion efficiency. Injection of LPG just before the inlet valve instead of mixing it with air in the manifold may provide further decrease in volumetric efficiency. However, no attempt was made to control the amount of diesel fuel injected during testing.

The effects on pollutant emissions

NOx emissions

At the full load conditions the variations of NOx emissions from the engine exhaust when using both diesel fuel and various LPG-Diesel fuel mixtures are given in Figures 5 (a) and (b). Minimum NOx was observed with the 5%

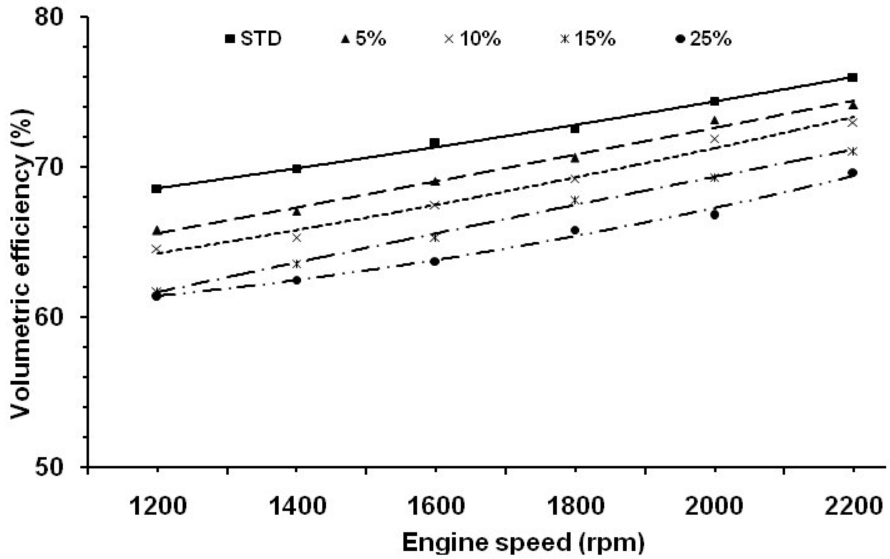
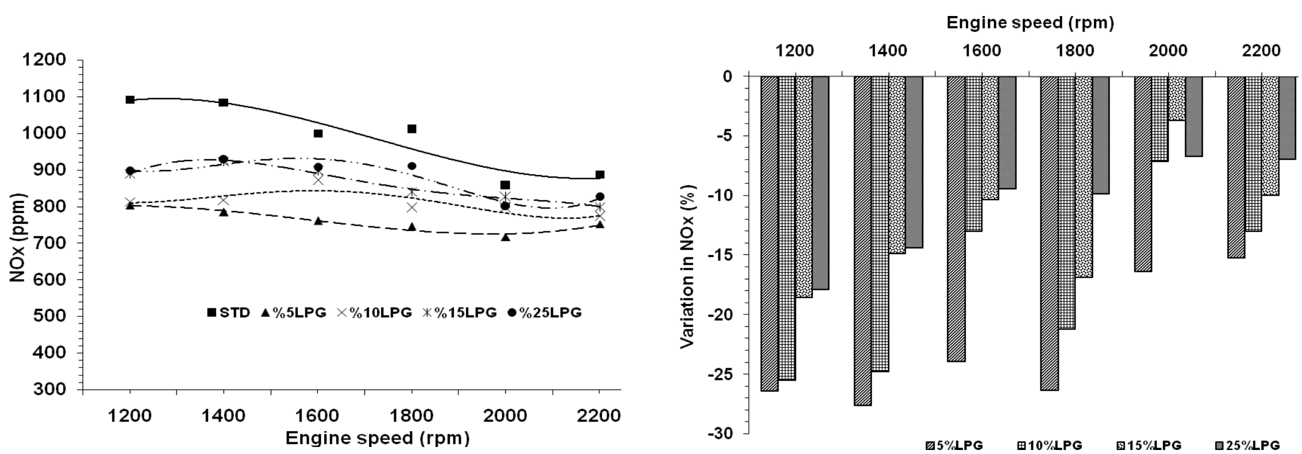


Figure 4. Comparison of volumetric efficiencies for various rate of LPG injection.



a b

Figure 5a. The effects of LPG injection on NOx emissions. **5b.** Variation of NOx emission for various rate of LPG injection compare to diesel fuel operation.

LPG injection rate at the speed of 2000 rpm. Maximum decreases of NOx in the modes are obtained as 27.6 with 5% LPG, 24.7% with 10% LPG, 14.8% with 15% and 14.3% with 25% LPG at 1600 rpm.

The main reason for this decrease in NOx at this speed is that overall air-fuel ratio was very close to the rich mixture. As is widely known, NOx formation rate strongly depends on gas temperature and air/fuel ratio in the post flame gases. Moreover, NOx emission strongly depends on the fuel distribution and on how that distribution changes with time due to mixing. NOx forms in the high temperature burned gas regions. Since temperature and air/fuel distributions within the burned gases are non-uniform, formation rates are non-uniform and highest in the close-to-stochiometric regions. The critical time period for NOx emission is when the burned gas temperature is at a maximum, that is, between the start of combustion and shortly after the occurrence of peak cylinder pressure.

Another factor contributing to decrease in NOx could be the fact that the heat release occurred later phase with LPG addition compared to that of standard diesel engine as concluded by Qi et al. (2007). During the tests running with LPG mixtures, the effect of post flame gases, which dominates the NOx formation may decrease since the piston moved further towards bottom dead center

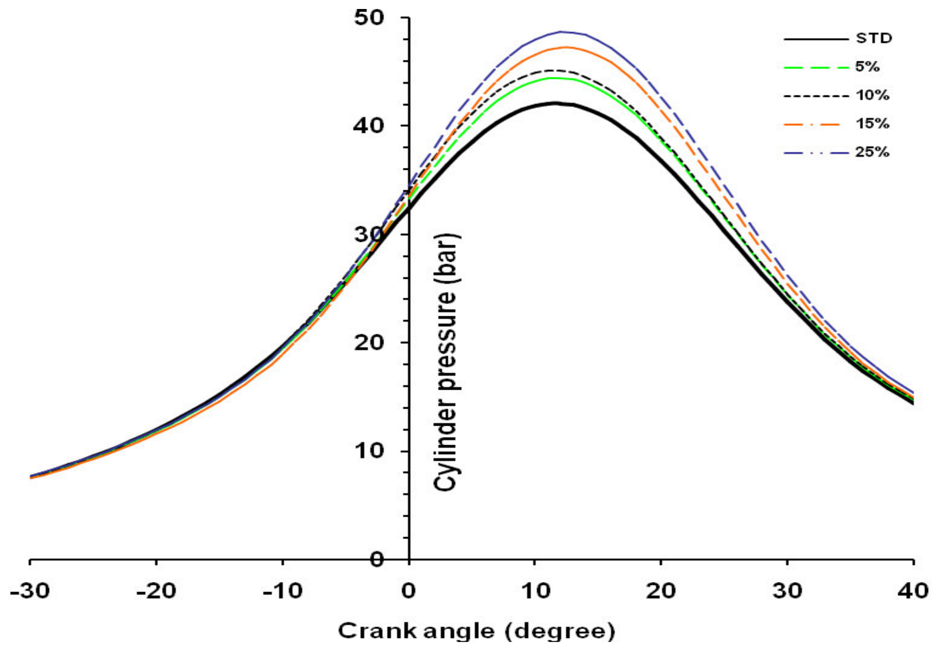


Figure 6. Comparison of cylinder pressure for various rates of LPG injections.

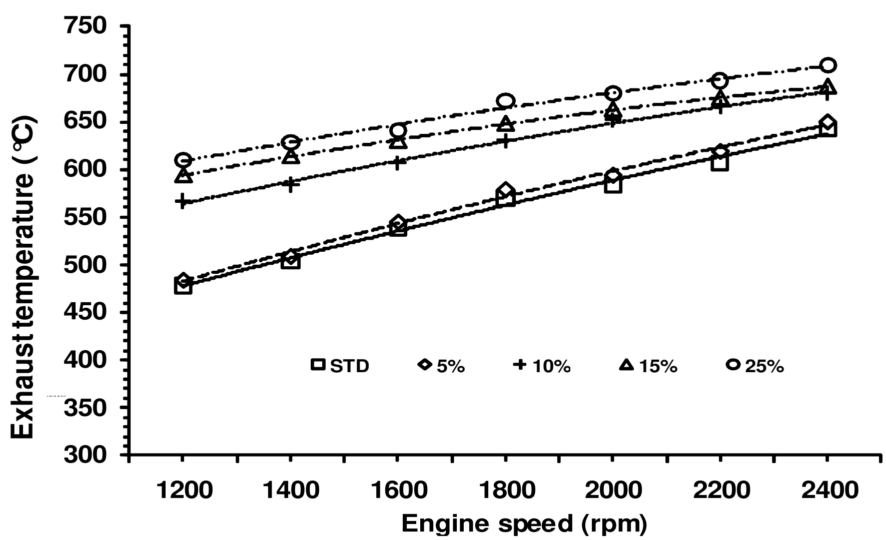


Figure 7. Variations of exhaust temperatures for various rates of LPG injection.

(BDC) during expansion period leading to decrease in gas temperature and residence time for NOx formation. Figure 6 shows the variation of cylinder pressures for various LPG injection rates. As can be seen from the figure, similar trend was observed. Increasing LPG rate caused to increase pressure after TDC and shift the point where maximum pressure occurred to further towards BDC. The reason for higher pressure obtained with LPG mixture can be explained by an additional energy input with LPG injection. As indicated previously, shifting the maximum pressure occurring point further towards BDC may cause to decrease NOx emission considerably.

Exhaust temperature

Figure 7 shows variations of exhaust temperatures for various LPG injection rates. Exhaust temperature increased with increasing LPG amount in the mixture for all the engine speeds tested. The increase can be explained by postponing the peak heat release rate to later stages of expansion period as reported by Qi et al.

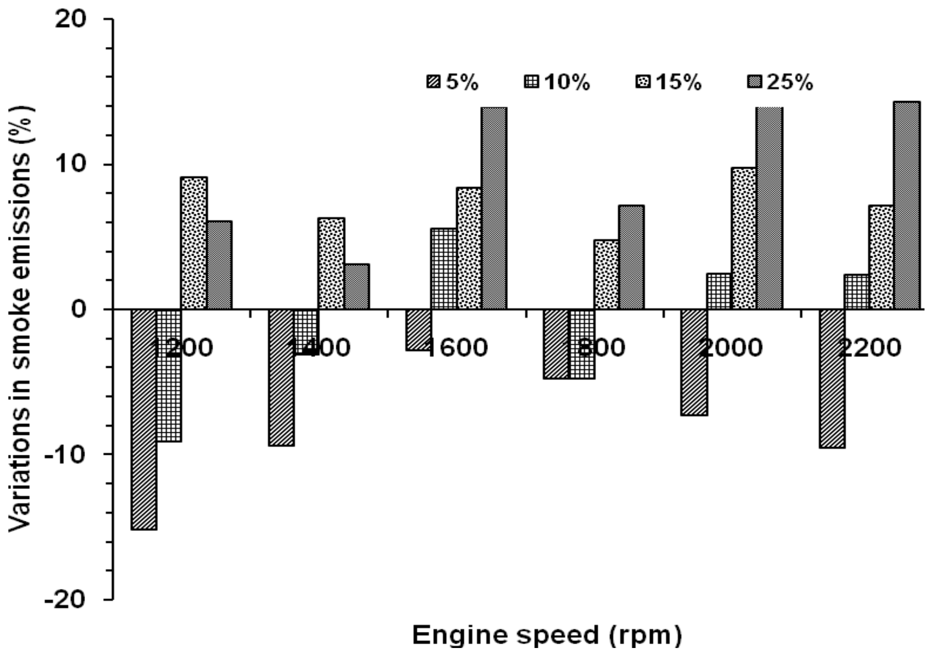


Figure 8. Variations of smoke emissions depending for various LPG injection rates.

(2007). Shifting heat release to a later phase may result in the increase in exhaust temperature.

Smoke emissions

Figure 8 shows the smoke emission of the engine operated with LPG compared to the standard diesel operation. It can be observed that the engine operated with 5% LPG exhibits a significant reduction in smoke emissions at all loads. As can be seen from the figure, smoke emission decreases at all engine speeds when using 5% LPG injection, whereas, some reduction was observed at only low and moderate engine speeds for 10% LPG injection. Smoke emissions were measured higher than those of standard diesel engine with increase of the LPG mass fraction after 10% for all the engine speeds. The main mechanism, which affects the formation of smoke emission, is gas temperature in the cylinder just after the combustion. Smoke emissions decreases with the increase of gas temperature in contrary to NOx formation. The conditions reducing the NOx emissions adversely affect the smoke formation.

Carbon monoxide emissions

The results for CO emissions of the engine, which was operated under both Diesel fuel and Diesel fuel with different LPG compositions, are shown in Figure 9 for all the speeds tested under full load condition. It is seen from the figure that CO emissions significantly increased between 1200 and 1800 rpm with 25% LPG rate. While CO emissions were measured as 0.6% with the diesel fuel operation at this speed, the emissions with the other LPG compositions at the same speed are 0.65% with 5% LPG, 0.8% with 10% LPG, 0.85% with 15% LPG and 1.2% with 25% LPG. This can be attributed to the volumetric efficiencies of the LPG mixtures which are lower than that of standard diesel engine. As it is well known that the rate of CO formation is a function of unburned fuel and mixture temperature during combustion, since both factors control the rate of fuel decomposition and oxidation. Decreases in volumetric efficiency and richer mixture seem to be significantly affecting the rate of fuel decomposition and oxidation adversely. The increase in CO emissions with 5% LPG composition remains tolerable when the improvements in the performance and reductions in NOx and smoke emissions are considered.

Unburned hydrocarbon emissions

The variation of unburned HC emissions of the engine, which was operated under both Diesel fuel and with different diesel fuel- LPG compositions are shown in Figure 10. The HC emissions increased with increase of the LPG mass fraction at all the engine speeds tested. It is seen from the figure that HC emissions were higher for all the LPG ratios compared to the pure diesel fuel operation. The reason of increase in HC emissions at low speeds can be explained by lower volumetric efficiency. Richer mixture and lower volumetric efficiency lead to

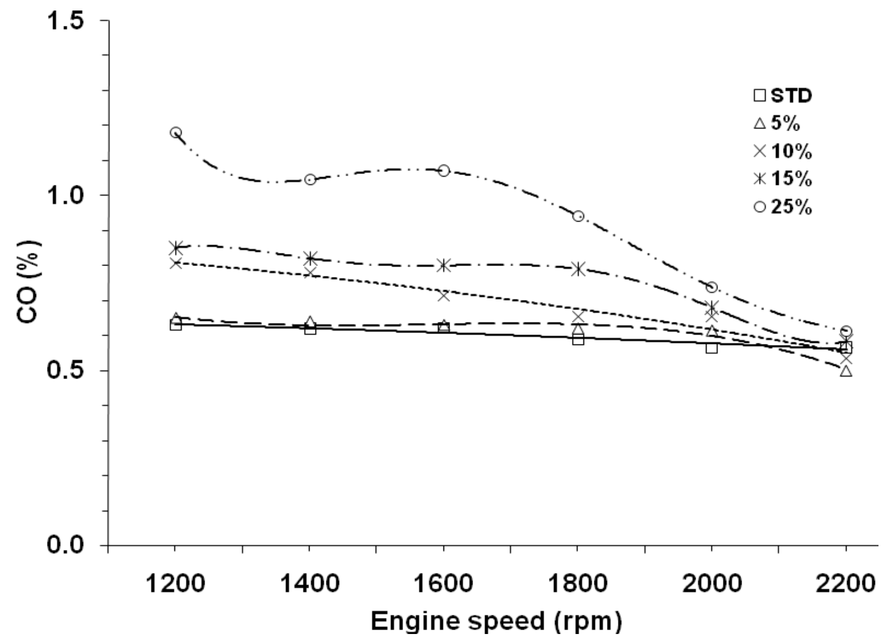


Figure 9. Variation of CO emission for various rate of LPG injection.

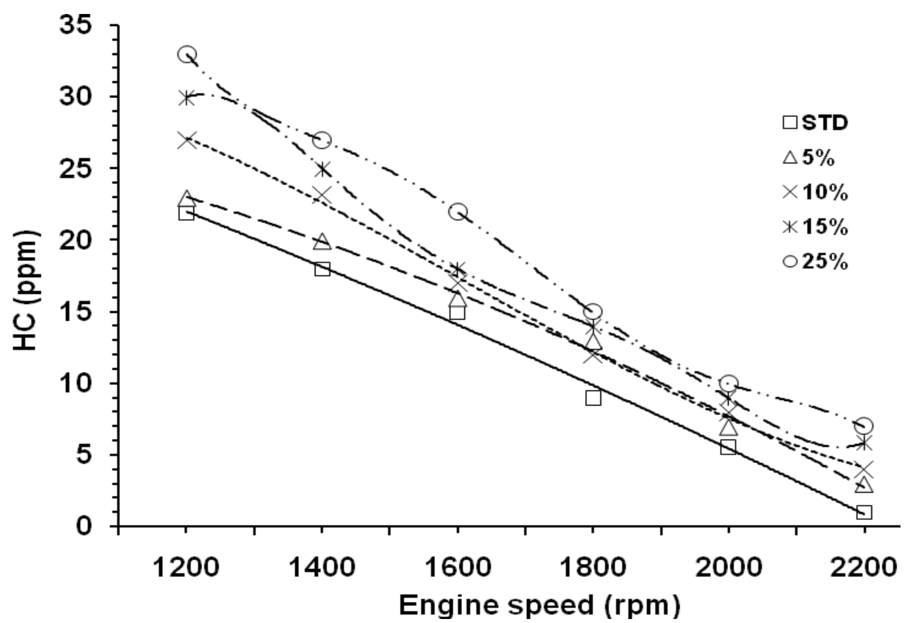


Figure 10. Variation of HC emission for various rate of LPG injection.

decrease combustion quality under lower cylinder temperature and hence, leads to increase the unburned HC emissions. However, HC emissions emitted from exhaust are in low levels for both diesel fuel and LPG mixtures. When the accuracy of HC (±12 ppm) is considered, most of the HC emissions measured with LPG are within the uncertainty limits.

Conclusions

In the present work, an experimental investigation has been conducted to examine the effects of injection of LPG into the engine manifold (just adjacent to inlet valve) during inlet. From the analysis of the experimental data, it has been shown that injection of LPG during inlet period of the engine considerably improved the brake power, specific fuel consumption, brake efficiency and brake torque. This has been attributed to the additional energy input into the cylinder with LPG injection.

NOx emissions also decreased considerably with LPG injection. This can be explained with the two main reasons. One reason may be that injecting LPG into the cylinder in the form of gas caused air and-LPG mixed homogeneously lean. Secondly, as ignition temperature of LPG is higher than that of diesel fuel, ignited diesel fuel starts combustion of LPG leading maximum pressure to further towards BDC compare to that of Diesel fuel operation. Thus, as the LPG composition in mixture increases, net heat release rate postpones to a later phase by decreasing the temperature of post flame gases. When exhausts emissions and performance trade-off considered, optimum injection rate is found at 5% LPG injection rate. The maximum net heat release rate with the 5% LPG took place 4° CA later compared to diesel fuel operation. As the postpone in peak heat release is small compare to the other LPG modes. The time for oxidation of particulate at this period might be sufficient for reducing smoke emissions. However, for all the other LPG mixtures, richer mixture along with the reduced gas temperature might lead to increase smoke emissions.

It seems that LPG-diesel fuel mixture combustion is a promising technique for controlling performance, NOx and smoke emissions even on existing DI. Since brake specific fuel consumption also considerably decreased by the optimized LPG injection rates depending on engine speeds, it will give additional advantages adapting an electronic controlled LPG injection system to a diesel engine in such a way that injection will be performed according to optimal rates.

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