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LPG gaseous phase electronic port injection on performance, emission and combustion characteristics of Lean Burn SI Engine

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Abstract. Gaseous fuels have always been established as an assuring way to lessen emissions in Spark Ignition engines. In particular, LPG resolved to be an affirmative fuel for SI engines because of their efficient combustion properties, lower emissions and higher knock resistance. This paper investigates performance, emission and combustion characteristics of a microcontroller based electronic LPG gaseous phase port injection system. Experiments were carried out in a single cylinder diesel engine altered to behave as SI engine with LPG as fuel at a compression ratio of 10.5:1. The engine was regulated at 1500 rpm at a throttle position of 20% at diverse equivalence ratios. The test results were compared with that of the carburetion system. The results showed that there was an increase in brake power output and brake thermal efficiency with LPG gas phase injection. There was an appreciable extension in the lean limit of operation and maximum brake power output under lean conditions. LPG injection technique significantly reduces hydrocarbon and carbon monoxide emissions. Also, it extremely enhances the rate of combustion and helps in extending the lean limit of LPG. There was a minimal increase of NO_x emissions over the lean operating range due to higher temperature. On the whole it is concluded that port injection of LPG is best suitable in terms of performance and emission for LPG fuelled lean burn SI engine.

1. Introduction

The increasing energy demand globally compels the researchers to go for more and more alternative cleaner energy source. It also emphasizes the need to study the technological alternatives and the emerging situations to provide a fresh look towards energy supply and environmental protection. Gaseous fuels have proved to be a promising way to reduce emissions in spark ignition engines. In particular, Liquefied Petroleum Gas (LPG) established to be a recommendatory fuel for Spark Ignition (SI) engines because of their higher octane rating, efficient combustion characteristics and lower emissions [1]. Table 1 shows the properties of LPG and it is clear that the properties of LPG like hydrogen to carbon ratio and wide ignition limit makes them favourable for lean burn operation [2,3]. Nitrogen oxides (NO_x) emission are less in LPG compared to gasoline counterparts [4,5]. LPG could be considered as a powerful alternative and sustainable energy source, but the influence of the fuel remains oppressed by means of conventional conversion applications from gasoline. Dedicated fuel systems should be manufactured for LPG fuel systems which can obtain better combustion, better

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thermal efficiency at higher compression ratio. LPG in carburetted engines is undoubtedly unresolved and in such cases the control of fuel flow is not possible. This greatly affects the efficacy of carburetted engines where the emissions increase due to improper combustion of the mixture. One of the major problem faced by gaseous fuels is power loss acted upon by its volumetric efficiency [6]. Also Hydrocarbon (HC) emissions with LPG in a premixed carburetion SI engine employing gas carburetors have been significant. The suggested solution to overcome these problems is the precise electronically controlled LPG fuel injection method.

Table 1. Properties of LPG.

Property	LPG
Composition (% vol)	Propane (30%), Butane (70%)
Lower heating value (kJ/kg)	46000
Flame speed (cm/s)	38.25
Stoichiometric AFR	15.5
Flammability limits (% vol in air)	2.15
Octane number (Research)	103-105
Auto ignition temperature (° C)	405-450

Several researchers have reviewed the applicability of injection system in LPG SI engines but only a few discuss about standalone systems. On the whole LPG fuel was injected in the engine manifold either in liquid or gaseous phase [3,4,7–10]. Because of the density play, LPG liquid phase injection would eliminate the problem of volumetric efficiency as mentioned earlier but requires complex hardware. Hence, numerous tryouts have been carried out to foster arrangement for electronic control of LPG injection for higher power output and thermal efficiency [11–13]. Though liquid phase injection would eliminate the problem of power loss, it requires a very complex setup which has attracted many researches on computerized control of liquid phase injection to improve efficiency and power. Pradeep and Ramesh [14,15] worked on an innovative twin injector system for a two stroke SI engine using cylinder barrel injection technique. The novel direct gaseous injection technique contributed to the reduction of hydrocarbon emissions around 80% by reducing short-circuiting losses as compared to that of manifold injection. The results show that the system helps in improving thermal efficiency and reducing HC emissions. Enhanced combustion rates were also observed when related to customary port injection technique. Owing to charge stratification and rich in-cylinder charge, lower peak NO emissions and higher Carbon monoxide (CO) emissions were observed. Ceviz *et al.* [16] experimented with different temperature of intake LPG fuel. Based on the test results, the temperature of the intake LPG fuel before manifold injection had a positive impact on the performance and NO emission properties. Baris *et al.* [17] observed experiments on LPG SI engine by varying the excess air coefficient (1.0 and 1.3) and ignition timing. Tests were carried out to analyze the performance of lean burn at a speed of 4300 rpm and at wide open throttle condition. It may be noted that as the air excess coefficient was higher than 0.8, advanced spark timing showed positive influence on performance of the engine. At an excess air coefficient of 1.3, the emissions reported lower. On the whole, HC and NO_x emissions increased due to spark advance whereas the CO emissions were inconsequential due to the spark timing. Despite the fact that there are copious information about the study of LPG on engine performance and in reduction of emissions, there are very few analogous works on the consequences of carburetion and electronically controlled fuel injection control methods of LPG run SI engine and its operation under lean conditions.

Li *et al.* [18] established an electronic LPG gaseous phase injection technique for a 4-stroke water cooled SI engine. The study was carried out under injection pressure condition of 0.1 MPa and an

orifice of 1mm. The emissions obtained from the injection technique were compared with the mechanical mixer system with LPG. Using electronic LPG injection, better emission control was reported for HC, CO and NO_x. Smith *et al.* [19] observed experiments on a dual-fuel, four cylinder SI engine provided with a 3-way catalytic converter. Dual fuel involves single point LPG gas injection and multipoint gasoline injection. Over the studied speed range, the maximum Brake Mean Effective Pressure (BMEP) decreased with the use of LPG. On average the thermal efficiency under full load condition improved by 8%. The prime intention of the current research is to analyze the aftermath of injection and carburetion techniques on SI engine performance, emission and combustion characteristics. Moreover in the present study, the tests were executed using a dedicated LPG gaseous injection controller unit developed by the authors.

2. Experimental Setup

A four stroke, air cooled, single cylinder diesel engine with a displacement volume of 661 cc, with a compression ratio 17.5:1 and power of 4.4 kW at 1500 rpm was altered to run as SI engine. A diesel engine was altered because it can combat knock and higher compression ratio. The engine was operated in SI mode with gaseous fuels by replacing the fuel injection pump and the injector with carburettor system, a spark plug and an ignition system. The overall system was made favourable for LPG operation in SI mode.

Table 2. Engine specifications.

Type	Kirloskar TAF1, air cooled, single cylinder CI engine
Displacement	661 cc
Stroke	110 mm
Bore	87.5 mm
Connecting Rod	231 mm
Compression ratio	17.5:1 (CI version), 10.5:1 (SI version)
Rated Power	4.4 kW @ 1500 rpm
Inlet Valve Open	4.5 ° bTDC
Inlet Valve Close	35.5 ° aBDC
Exhaust Valve Open	35.5 ° bBDC
Exhaust Valve Close	4.5 ° aTDC

The spark timing could be varied precisely by 1° crank angle from 280° to 360° (0° to 80° CA before TDC) through a microcontroller based circuit with input from a proximity sensor. The compression ratio was modified by altering the clearance volume of a bowl-in type piston and was fixed at 10.5:1. Shim plates of standard thickness were placed between crank case and cylinder block to compensate the required clearance. The piston squish area was maintained at 30% of the total land area maintaining the a squish velocity of 4 m/s [20].

The engine used for experimentation was Kirloskar TAF1 engine as mentioned in Table 2. The intake manifold was especially designed to conduct experiments on carburetion and port-injection of gaseous phase LPG. The manifold was provided with throttle valve and a gas mixer unit for carburetion, while a 3-gas injector type rail was mounted for injection. The angle of injection was kept at 45° along the fuel line path. A dedicated LPG injection circuit using microcontroller ATMEGA 8A was developed and connected to the injectors. The pulse width and the delay of signals can be

controlled precisely by 1ms. The injector rail has provision for mounting 4 injectors on it but only 2 were used according to the flow requirement. The injector works on a pressure range of 0-3 bar and voltage of 10-14 V.

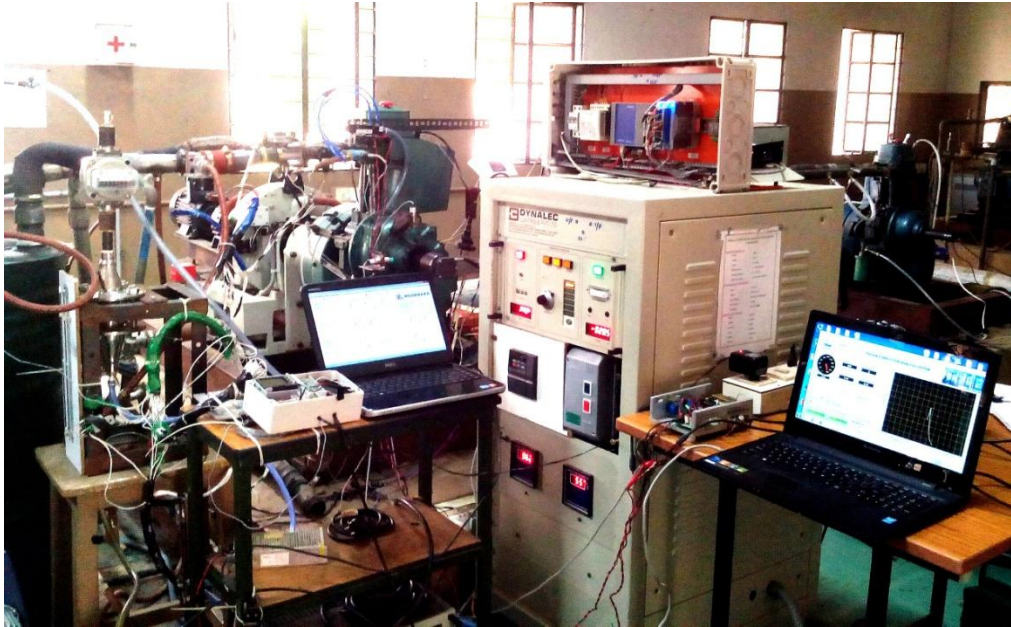
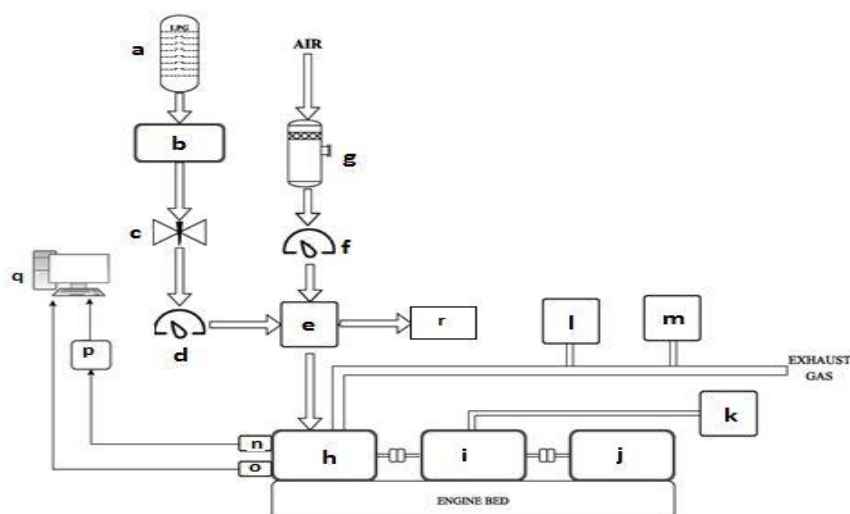


Figure 1. Pictorial view of experimental setup.

Figure 1 shows the pictorial view of the experimental setup developed for the research work. The engine was connected to an eddy current dynamometer which was set to constant speed mode to carry out the experiments at 1500 rpm. The cylinder head surface was made to mount a KISTLER make piezoelectric pressure transducer in order to obtain the in-cylinder pressure. An AVL optical crank angle encoder was fixed to attain the corresponding pressure values. 100 cycle pressure data was recorded for each operating condition. The pressure values are corrected based on the intake manifold pressure obtained from the KISTLER make piezoresistive pressure sensor flush mounted in the engine intake manifold. The throttle position sensor was connected to a digital display in order to vary the exact throttle opening positions accordingly. A data acquisition system connected to a computer was used to acquire online in-cylinder pressure details from the flush mounted piezoelectric sensor and the intake manifold pressure from a piezoresistive sensor for every 1° resolution. The CO and CO₂ emissions were measured using NDIR analyzer and HC emissions were measured using FID analyzer and NO and NO_x were measured using CLD analyzer.

2.1. Experimental Conditions

Experiments were conducted at constant throttle opening condition of 20% based on the ECU controller module connected to a throttle position sensor fixed in the intake manifold. The LPG gas cylinder was immersed in a hot water tank which maintained a temperature of 50⁰C. A fuel surge tank was also made which was wrapped with tape heaters to ensure gaseous flow near the intake manifold as well. The fuel surge tank maintains a temperature of 50⁰C and also helps in maintaining constant supply and reduces pressure fluctuations during high load conditions. The flow of LPG gas was measured using thermal mass flow meter. Airflow rate was obtained from a positive-displacement type flow meter (Make: DRESSER) connected with a surge tank on the suction side.



^aLPG cylinder in hot water bath (50⁰C)

^bSurge Tank with electric heater

^cControl Valve

^dThermal mass Flowmeter

^eInjector unit

^fAir flowmeter ^g Air surge tank

^hEngine

ⁱEddy current dynamometer

^jMotoring unit

^kDynamometer controller

^lFID and CLD analyzer ^mNDIR analyzer

ⁿPressure pickup

^oCrank angle encoder

^pData Acquisition system

^qComputer

^rElectronic injection circuit

Figure 2. Schematic diagram of experimental setup

The mixture strength was varied from richest to the leanest possible by controlling the amount of fuel using a fine tune adjuster. Extreme care was taken in maintaining the gaseous phase addition of LPG fuel to the primary flow line without causing hindrance in the desired flow rate. The leanest mixture i.e. lowest equivalence ratio limit was limited by unstable operation of the engine. At each of the equivalence ratio, MBT (Minimum spark advance for best torque) was set. However, under knocking conditions the spark timing was adjusted for knock free operation rather than MBT adjustment. Such an adjustment was required only during rich operating conditions which are of no significance in this research work. Readings were always taken on after the engine attains stability of operation. Torque, MBT timing, intake pressure, cylinder pressure, air flow rate, LPG fuel flow rate, injection pulse width, injection delay, inlet and exhaust temperatures, exhaust emission levels of HC,CO,NO and NO_x were recorded for different equivalence ratios. The data acquisition system recorded the values of pressures at each crank angle degrees at regular time intervals for computing cycle pressure data. The pressure signals from the piezoelectric transducer were referenced based on the suction stroke cylinder pressure at BDC, which was equalled to the average manifold pressure obtained from the intake pressure piezoresistive transducer. Dedicated software was developed using LABVIEW to calculate the pressure average against crank angle for the obtained number of cycles.

Knowledge of the pressure variation in the combustion chamber will enable calculation of energy changes taking place in an engine by thermodynamic analysis. Also important aspects like peak pressure, indicated mean effective pressure, ignition delay, combustion duration and cyclic variations in pressure can be studied with the aid of cylinder pressure crank angle data. To obtain an in-depth view of combustion process, it is mandatory to correlate the above mentioned factors with in-cylinder pressure. This can be termed as Heat Release Analysis. This data provides a clear picture of what happens inside the cylinder during combustion process. First law of thermodynamics is used to interpret the pressure data to calculate the mass fraction of the fuel burnt and the rate of heat release [21, 22]. The analysis is done considering a closed cycle where the intake and exhaust valves are closed. Hence, a simple approach of single zone model is considered where the properties and the thermodynamic state are considered uniform throughout the cylinder and represented by the average values. The model is considered to be zero dimensional since no spatial variations are taken into account. For a closed cycle period, the first law can be written with suitable assumptions as follows,

$$\frac{dQ_{hr}}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP}{d\theta} + hA_s(T - T_w) \frac{dt}{d\theta}$$

where Q_{hr} is the rate of heat release and θ is crank angle in degrees, γ is the ratio of specific heats of the air, fuel and residual gas mixtures. The total mass of the mixture per cycle is computed as the addition of air mass, residual gas mass present in the clearance volume and the mass of fuel. Ideal gas equation is used to calculate the mass of residual gas trapped where the temperature and pressure are assumed equal to the exhaust manifold properties respectively [23]. The residual gas constant was assumed as that of the air. Based on the above information and assumptions, the molecular weight of the charge, mole fraction of fuel, residual gas and air and the charge gas constant are found. Using the expression of polynomials, C_p value of each of the LPG gas constituent was determined knowing the composition of the gas. Cylinder gas temperature T was obtained using ideal gas equation at every crank angle degree. Based on the engine speed, t is the time period and T_w being the temperature of combustion chamber wall. The coefficient of heat transfer was obtained using the Hohenberg's correlation as shown below [24]

$$h = C_1 V^{-0.06} P^{0.8} T^{-0.4} (\overline{Vp} + C_2)^{0.8}$$

where h is the coefficient of heat transfer in $W/m^2 K$, C_1 and C_2 the tuning constants (130 and 1.4), V volume of cylinder in m^3 , P the in-cylinder pressure in bar, T the cylinder gas temperature in Kelvin

and \overline{Vp} the piston mean speed in m/s. A_s is the area in m^2 where the heat transfer between the gas and combustion chamber wall occurs. This comprises top squish area, piston bowl surface area, cylinder wall and cylinder head. All the areas are considered constant excluding the cylinder wall area for a desired form. Along with the heat release, cumulative heat release and mass fraction burned at every crank angle degree was calculated.

3. Results and Discussions

All test results were obtained at a constant speed of 1500 rpm and throttle opening position of 20%. The load was varied based on the fuel inlet from rich to leanest possible limit. The performance, emissions and combustion characteristics were compared between LPG gas phase port injection and LPG carburetion system.

3.1. Performance parameters

Figure 3 shows the variation of brake power with equivalence ratio for carburetion and injection mode. The maximum brake power for carburetion was observed to be 2.5 kW at an equivalence ratio of 0.8 while for injection it was 3.1 kW. The values higher than equivalence ratio of 1 are not of interest in this research study hence maximum power output was considered for comparison at an equivalence ratio of 0.8. It may be noted that an increase of 24% higher power output is observed by injection technique. This clearly states that electronic control of fuel injection precisely meters the quantity of fuel and provides an enhanced breathing capacity when compared to that of carburetion which improves the brake power output and also brake thermal efficiency as shown in figure 4. The peak brake thermal efficiency for injection was found to be 34.1% at an equivalence ratio of 0.88 and was about 27% respectively for LPG carburetion. This shows an improvement of 22% higher brake thermal efficiency using LPG electronic injection technique as compared to that of carburetion. It can also be observed that the LPG electronic injection method helps in extending the lean limit from an equivalence ratio of 0.69 by carburetion to 0.63 by injection method. An increase of 8.6% of lean limit certainly reduces the fuel consumption of LPG using injection technique. On the whole, the maximum efficiency was observed at a much leaner mixture for injection on comparison and hence it can be stated that gas phase manifold electronic injection is much suitable for leaner operations for improving higher thermal efficiencies and brake power output.

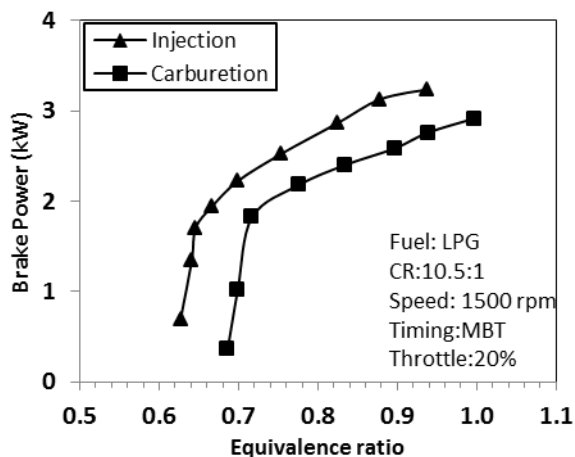


Figure 3. Variation of brake power.

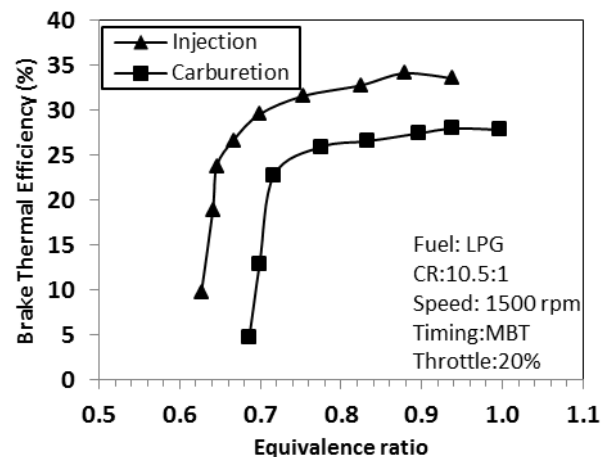


Figure 4. Variation of brake thermal efficiency.

3.2. Combustion Parameters

Figure 5 shows the rate of in-cylinder pressure rise, maximum cylinder pressure and occurrence of maximum pressure rise which was around 29.5 bar at 374 deg.CA at an equivalence ratio of 0.88 was favourable for gaseous injection method than carburetion which was around 24 bar at 376 deg.CA. This may also be noted in the rate of heat release curves as shown in figure 6. The HRR values are highest with about 42.3 J/deg.CA at 364 deg.CA for injection and were also better by 16.5% than carburetion which was around 36.3 J/deg.CA. The cumulative heat release which gives the total energy from combustion was around 850 Joules for carburetion which was also lower when compared to 960 Joules observed from electronic injection technique. This indicates that injection paves way for enhanced combustion. Since the charge is injected at a precise timing, the fuel is possibly rich at the vicinity of the spark plug which is comparably better than the homogenous charge inducted during carburetion. The charge stratification by injection and the turbulence created by the bowl type piston helps in flame propagation which results in enhanced combustion of LPG. LPG injection has the least ignition delay period i.e. 5% mass fraction burnt (5%MFB) of 52 deg.CA when compared to that of carburetion which can be clearly seen from figure 6. Figure 7 shows the variation of ignition delay and it can be seen that spark timing has been advanced earlier in LPG gaseous phase injection over

carburetion from figure 8. The total burn duration as well as the segment wise combustion duration for the LPG port injection technique are less with minimum spark timing and combustion duration. This indicates enhanced combustion rate involving flame travel and turbulent kinetic energy and minimum spark advance is possible in lean burn condition by gaseous phase LPG injection method.

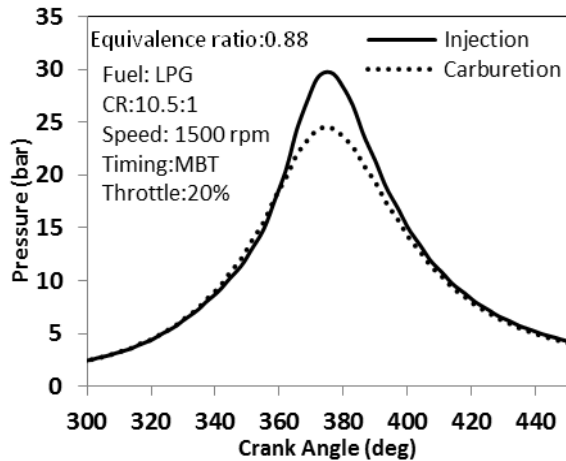


Figure 5. Variation of In-cylinder pressure.

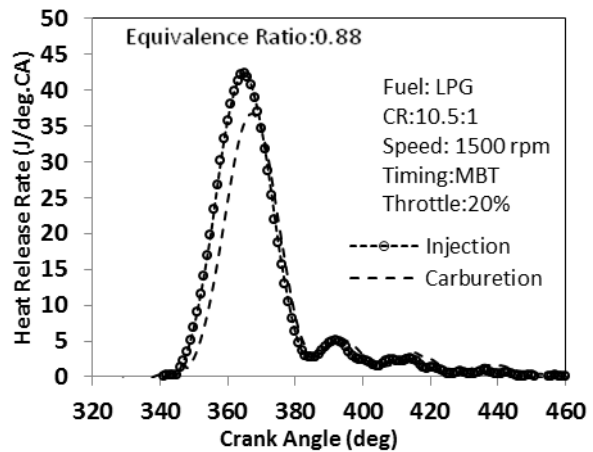


Figure 6. Variation of Heat Release Rate (HRR).

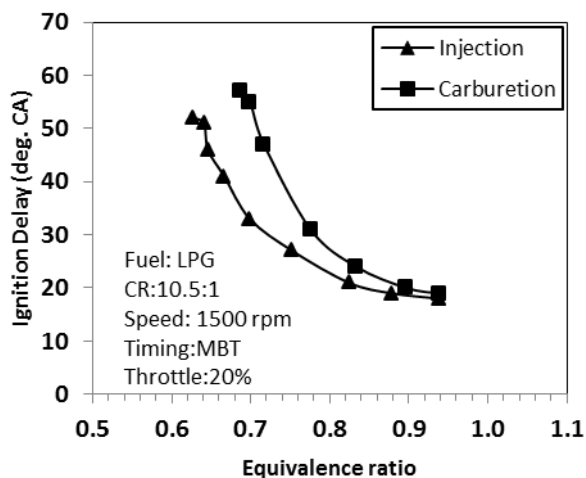


Figure 7. Variation of Ignition delay.

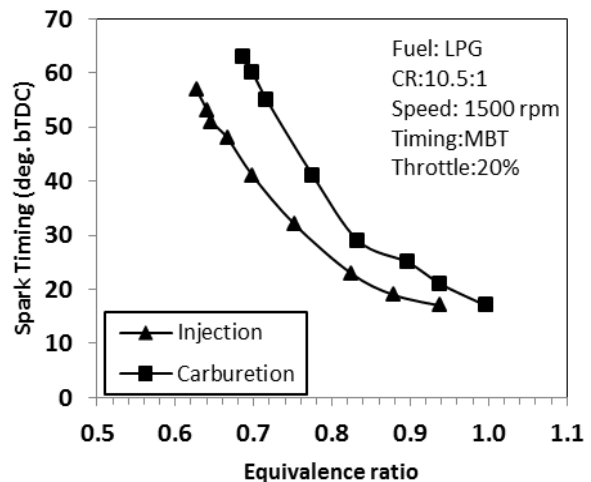


Figure 8. Variation of Spark timing.

3.3. Emissions

The variations of HC and CO emissions could be clearly seen from the lean equivalence ratios between 0.6 and 0.8 for carburetion and injection. Figure 9 shows that within the stable limit of combustion on the leaner side of 0.70 equivalence ratio HC emissions of carburetion was around 3720 ppm whereas for injection it reported as low as 300 ppm respectively. As the equivalence ratio reaches a particular lower limit there is a sudden rise in HC level due to misfiring. The decreasing trend of HC towards rich limit is primarily due to the influence of increase in the in-cylinder temperature. This can be attributed towards the enhanced combustion by injection and the combined effect of turbulence and laminar flame speed of the bowl-in-piston type. Figure 10 shows that at an equivalence ratio of 0.75 the CO emissions were 0.08% and 0.05% respectively for carburetion and injection. This can also be observed from Fig. 11 which shows the trend of CO₂ emissions of LPG injection which was higher because of complete combustion. The NO and NO_x emission comparison from figure 12 and 13 clearly indicate that there was an increase of nearly 58% at an equivalence ratio of 0.82. The NO_x

emission were certainly higher for LPG injection than LPG carburetion because of higher combustion temperature and heat release rate, this can also be inferred from higher exhaust gas temperature obtained from injection technique compared with carburetion as shown in Figure 14. Hence, it can be stated that LPG electronic control manifold injection provides a better suitable way to reduce HC and CO emissions.

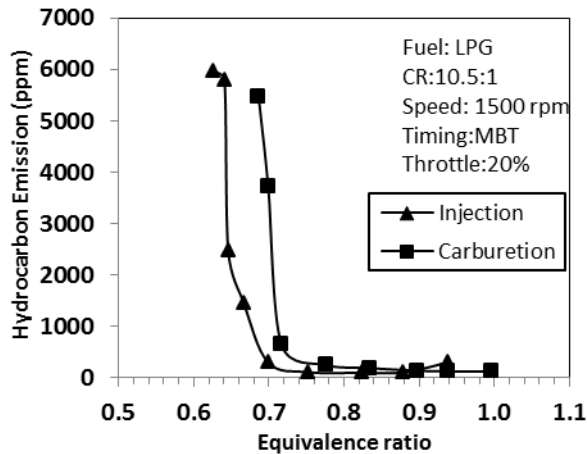


Figure 9. Variation of HC.

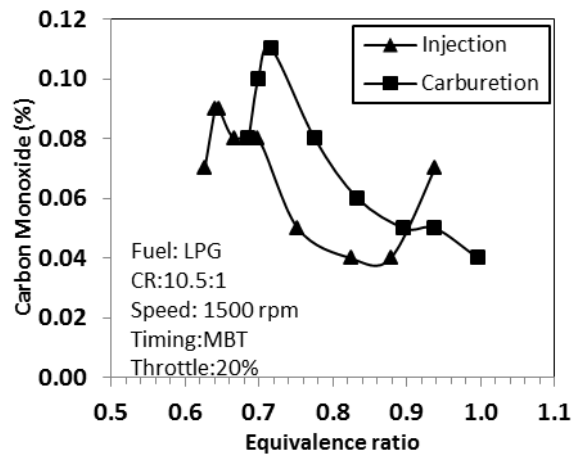


Figure 10. Variation of CO.

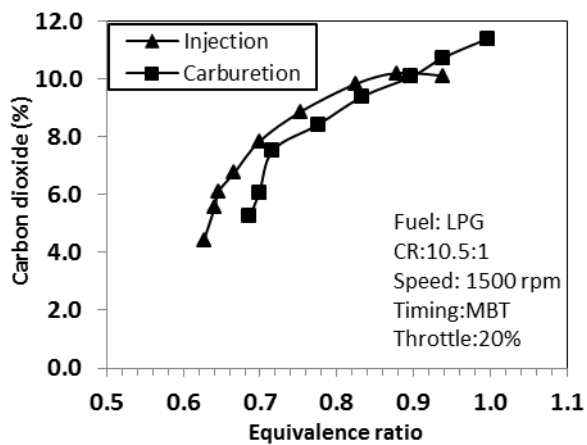


Figure 11. Variation of CO₂.

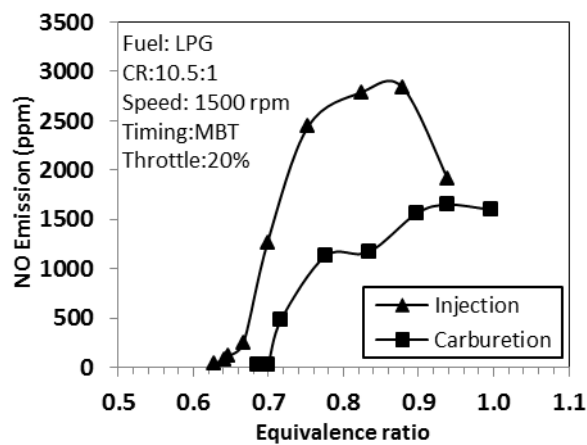


Figure 12. Variation of NO.

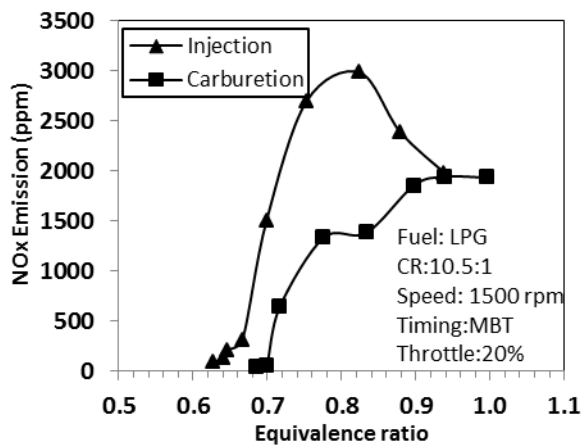


Figure 13. Variation of NO_x.

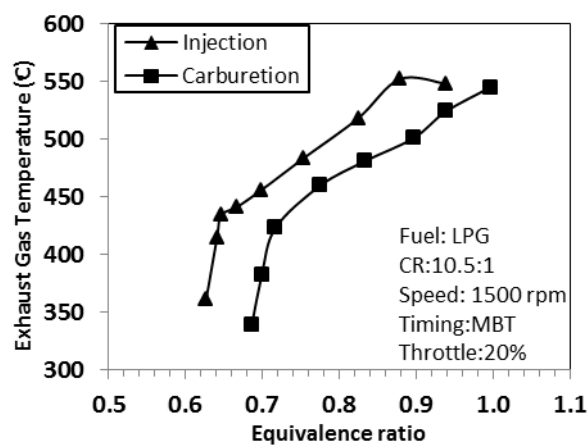


Figure 14. Variation of EGT.

3.4 Aspects of LPG electronic injection

From the above research, it is clear that LPG gaseous phase electronic injection technique paves way for improvements in terms of brake power, brake thermal efficiency, lean limit and also helps in enhanced combustion of LPG fuelled lean burn SI engine. Even on the emissions front, notable reductions in HC and CO were observed. Although, there have been a rise in NO and NO_x emissions, this could well be reduced by further implications like Exhaust gas recirculation, hydrogen supplementation or by other simple techniques. If the heat release rate was analysed in the real condition the results would have been more realistic compared to the assumption of pressure and temperature being uniform. On the whole, LPG in SI engine can be established as an alternate fuel for ecological reasons and can be extended over agricultural sector for irrigation purposes as well as to other remote places in India. This definitely will promote a newer way of utilizing alternate energy source for sustainable growth and development.

4. Conclusion

An electronically controlled port-injection system was successfully developed and tested; comparison of its performance, combustion and emission characteristics with carburetion was also done. Based on the experimentation the following outcomes can be arrived.

- There was a significant improvement in port injection technique as compared to the carburetion in both power output and brake thermal efficiency. There was an improvement of 22% increase in brake thermal efficiency and 24% higher brake power output for injection system. The maximum brake thermal efficiency under lean operating range reduces the fuel consumption as well.
- Lean limit extension was possible from an equivalence ratio of 0.69 using carburetion to an equivalence ratio of 0.63, about 8% increase using gaseous port injection technique.
- In general, HC and CO emissions were less for port-injection as compared to carburetion but witnessed an increase of NO and NO_x emission of about 58% due to higher in-cylinder temperature and enhanced combustion.
- The LPG electronic control port-injection has clear advantage over carburetion in terms of performance, combustion and emission characteristics of LPG lean burn SI engine.

The electronic LPG gas injection system in SI engine will promote a new way to develop cleaner sustainable power engines with reduced emission levels. The developed system provides a fresh look on lean burn combustion of LPG and helps in reduced fuel consumption.

Acknowledgement

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References

- [1] Morganti K J, Foong T M, Brear M J, Da Silva G, Yang Y and Dryer F L 2013 The research and motor octane numbers of Liquefied Petroleum Gas (LPG) *Fuel* **108** 797–811
- [2] Li L, Wang Z and Wang H 2014 A study of lpg lean burn for a small SI engine *SAE Technical Paper* 2002-01-2844; 2002
- [3] Bayraktar H and Durgun O 2005 Investigating the effects of LPG on spark ignition engine combustion and performance *Energy Conversion and Management* **46** 2317–33
- [4] Ceviz M A and Yüksel F 2006 Cyclic variations on LPG and gasoline-fuelled lean burn SI engine *Renewable Energy* **31** 1950–60
- [5] Akif Ceviz M, Sen A K, Küleri A K and Volkan Öner İ 2012 Engine performance, exhaust emissions, and cyclic variations in a lean-burn SI engine fueled by gasoline–hydrogen blends *Appl. Therm. Eng.* **36** 314–24
- [6] Sunwoo M, Sim H and Lee K 1999 Design and development of an ECU and its air-fuel ratio control scheme for an LPG engine with a Bypass injector, Vehicle Electronics Conference,

- (IVEC '99) Proceedings of the IEEE International page. 508 - 513 Suppl. IVEC.1999.830740; 1999
- [7] Pourkhesalian A M, Shamekhi A H and Salimi F 2010 Alternative fuel and gasoline in an SI engine: A comparative study of performance and emissions characteristics *Fuel* **89** 1056–63
- [8] Li L, Wang Z, Deng B, Han Y and Wang H 2002 Combustion and emissions characteristics of a small spark-ignited lpg engine *SAE Technical* **01** 1738
- [9] Mistry, C., Comparative assessment on performance of multi cylinder engine using CNG , LPG and petrol as a fuel, *SAE Technical Paper* 2005-01-105; 2005
- [10] Campbell M, Wyszy Ł P and Stone R 2004 Combustion of LPG in a spark-ignition engine *SAE Technical Paper* 2004-01-0974; 2004
- [11] Watson H C and Phuong P X 2007 Why liquid phase LPG port injection has superior power and efficiency to gas phase port injection *SAE Technical Paper* **22** 48-52
- [12] Wendeker M, Jakli P, Czarnigowski J, Boulet P and Breaban F 2007 Operational parameters of LPG Fueled SI Engine – Comparison of Simultaneous and Sequential Port Injection *SAE Technical Paper* 2007-01-2051;2007
- [13] Sierens R 1992 *An Experimental and theoretical study of liquid LPG injection SAE Tech. SAE International* 922363; 1992
- [14] Pradeep V and Ramesh A 2015 A novel twin injector strategy for direct cylinder barrel injection of LPG in a 2S-SI engine *Fuel* **162** 128–37
- [15] Pradeep V, Bakshi S and Ramesh A 2015 Direct injection of gaseous LPG in a two-stroke SI engine for improved performance *Appl. Therm. Eng.* **89** 738–47
- [16] Ceviz M A, Kaleli A and Güner E 2015 Controlling LPG temperature for SI engine applications *Appl. Therm. Eng.* **82** 298–305
- [17] Baris Erkus, Ihsan M and Sürmen A 2015 Enhancing the heavy load performance of a gasoline engine converted for LPG use by modifying the ignition timings *Appl. Therm. Eng.* **85** 188-94
- [18] Li L G, Liu Z M, Wang H P, Deng B Q, Wang Z S, Xiao Z C, Su Y and Jiang B 2003 Development of a gas-phase LPG injection system for a small SI engine *SAE Technical Paper* 2003-01-3260; 2003
- [19] Smith W J, Timoney D J, Lynch D P 1997 Emissions and efficiency comparison of gasoline and LPG fuels in a 1.4 litre passenger car engine *SAE Technical, SAE International* 972970; 1997
- [20] Krishnaiah R, Ekambaram P and Jayapaul P B Investigations on the effect of piston squish area on performance and emission characteristics of LPG fuelled lean burn SI engine *SAE Technical Paper* 2016-28-0123; 2016
- [21] Gatowski J A, Balles E N, Chun K M, Nelson F E, Ekchian J A and Heywood J B 1984 Heat release analysis of engine pressure data *SAE Technical Paper* 841359; 1984
- [22] Hayes T K, Savage L D and Sorenson S C 1986 Cylinder pressure data acquisition and heat release analysis on a personal computer *SAE Technical Paper* 860029; 1986
- [23] Pedersen P S and Qvale B 1974 A Model for the physical part of the ignition delay in a diesel engine *SAE Technical Paper* 740716; 1974
- [24] Hohenberg G F 1979 Advanced approaches for heat transfer calculations *SAE Technical Paper* 790825 ;1979