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# Investigations on combustion characteristics of lean burn SI engine fuelled with Ethanol and LPG

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**Abstract.** This paper presents an effective experimental approach that can be applied to stationary single cylinder LPG fuelled lean burn SI engine. This approach has been applied to operate the engine under different equivalence ratio at wide open throttle condition. Experiments were conducted at a constant speed of 1500 rpm and at compression ratio of 10.5:1. In this study the effects of adding small amount of ethanol (5%, 10% and 20%) along with LPG were investigated. Ethanol on volume basis was injected along with carburetted LPG. A maximum brake thermal efficiency of 28.5% with 10% ethanol addition for LPG were observed at an equivalence ratio of 0.7. The ethanol addition with LPG extended the lean limit as compared to pure LPG. Also, it was observed that 10% Ethanol addition resulted in reduced HC and CO emissions due to oxygen present in the ethanol. The combustion parameter shows increased heat release rate and reduced cyclic variations with 10% ethanol addition. The optimal ethanol blend with LPG was found to be 10% for better performance and reduced emissions.

## 1. Introduction

The ongoing use of conventional energy resources towards various prime moving applications has dominated in terms of greater demands among the overall availability of the resources. The most prominent species of energy that is adaptable for the existing combustion technology are the alternative fuels. The alternative fuels have always been well known for its compatible properties that enhances the combustion efficiency which also improvises the performance as well as minimizes emissions [1].

The concept of dual fuel system tends to have positive effects on major parameters in SI engine. The dual fuel comprises of a baseline fuel along with an additive agent at specified ratios of mixture. The most well-known additive that enhances the fuel quality and is beneficial holding similar properties that is required to sustain engine operation is ethanol fuel. The ethanol is derived from biodegradable resources such as sugar, corn, barley and other forms that are all less toxic materials [2] [3]. The existence of ethanol in liquid form has greater compatibility with the existing engine technologies that require only minor modifications. The properties of the test fuels are shown in table 1. There are several ways in improvising engine efficiencies such as optimizing air-fuel ratio, spark timing, combustion chamber design and compression ratio. Amongst these the effect of dual fuel system tends to have significant stability and completeness of combustion that determines the maximum torque and reduced emission levels. This has more attraction towards flex fuel vehicle that



combines the use of more than one fuel that causes superior performances than compared to base line fuels [4].

In the concept of lean burn technique the combustion occurs by charging the mixture accompanied by excess amount of air causing the equivalence ratio to shift to lower regions. Hence in SI engine one such base fuel that operates at extended lean conditions under optimized higher compression is Liquefied Petroleum Gas (LPG). LPG is renowned for its properties that benefits in producing higher brake thermal efficiencies and suppresses engine knock under higher compression ratio at its corresponding spark timing [5]. However there are certain issues such as lesser control over air fuel ratio in the conventional LPG mixing as well as port injection system that results in incomplete combustion that leads to high level of emissions [6] [7]. Apart from this LPG displaces air that causes lower volumetric efficiency lowering partial pressure of air in the intake system, and also the lower vaporization effects results in loss of engine power output [8]. Hence all these phenomena depend internally upon a stable complete combustion process which is affected by optimum spark timing, flame motion pattern and uniformity of air fuel mixture.

**Table 1.** Properties of the fuels [9] [10] [11] [12].

| Properties                                | LPG  | Ethanol                    |
|---|--|----------------------------|
| Chemical Formula                          | $C_3H_8 + C_4H_{10}$                                 | $C_2H_5OH$                 |
| Composition (% vol)                       | Propane $C_3H_8 - 30\%$<br>Butane $C_4H_{10} - 70\%$ | C - 52<br>O - 13<br>H - 35 |
| Lower heating value (MJ/kg)               | 45.7   | 26.8                       |
| Density ( $kg/m^3$ )                      | 2.26   | 790                        |
| Flame velocity (cm/s)                     | 38.25  | 41                         |
| Stoichiometric Air Fuel Ratio(kg/kg)      | 15.5   | 9:1                        |
| Flammability limits (vol.% in air)        | 2.15   | 3.5 - 15                   |
| Research octane number                    | 103 - 105  | 106                        |
| Motor octane number                       | 90 - 97  | -                          |
| Auto ignition temperature ( $^{\circ}C$ ) | 405 - 450  | 420                        |
| Boiling Point ( $^{\circ}C$ )             | -  | -                          |
| Latent heat of vaporization (kJ/kg)       | 358.2  | 904                        |

Hence the ideology of dual fuel system is an obvious method that forms LPG as base fuel mixed with the oxygenated additive ethanol. There is indication of tendencies that at higher cylinder volumes there is more affinity towards misfire occurrences. The remarkable properties of ethanol is known for its lower flammability limits making it preferable for lean combustion [13]. The ethanol additive sustains stable combustion by reducing knocking effects at higher compression ratio with increased peak pressure [14]. The pattern of flame developed and its propagation from the region of spark plug all the way down has influence upon the combustion variations. The higher flame velocity and the hydroxyl groups of ethanol results in superior combustion by with minimized heat losses to the cylinder maximizing thermal efficiencies [15]. The charged mixture containing alcohol such as ethanol additive results in higher volume of combustion products that results towards the rise in peak in-cylinder pressure causing improved work done on the piston [16]. The addition of ethanol forms homogenous air fuel mixing that causes complete combustion and moreover the presence of free radicle accelerate the combustion process owing to faster heat release rate [17]. The oxygenated properties of ethanol in the blend causes minimized coefficient of variation (COV) of indicated mean effective pressure (IMEP) [18], hence this makes it favorable with optimum concentration of ethanol present in the blend.

This experiential work was conducted to study the combustion effects upon injecting ethanol as an additive with carbureted LPG being the base fuel. The outcome was an improved effect in combustion characteristics being observed for 10% ethanol concentration present in the LPG ethanol mixture.

## 2. Methodology

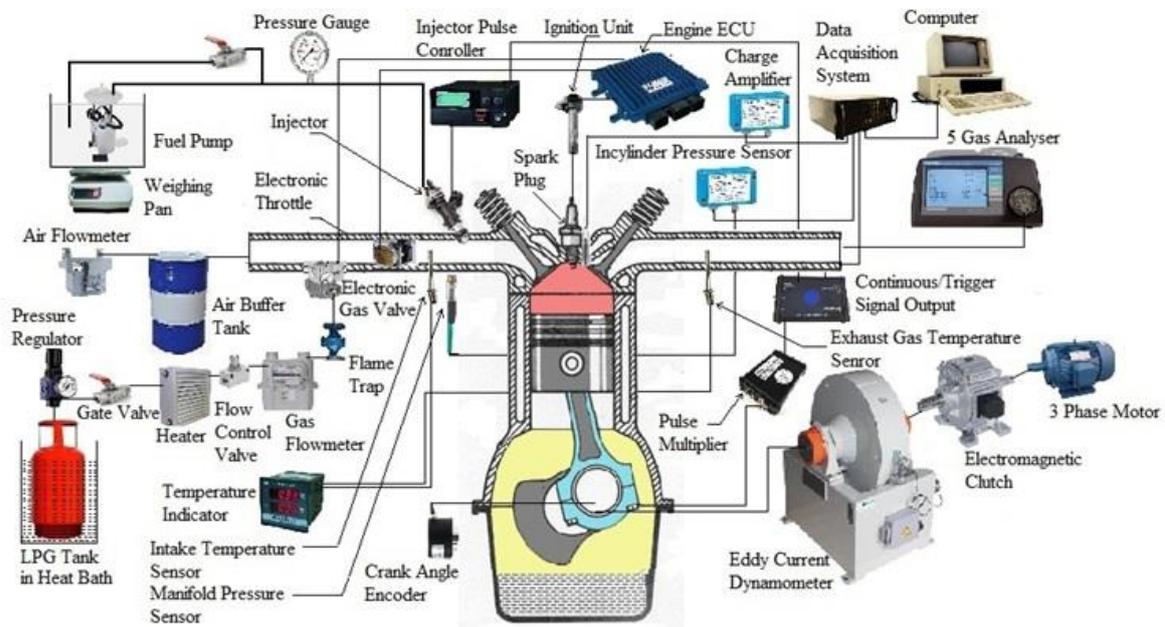
A diesel engine make Kirloskar TAF 1 model was modified to operate in spark ignition mode. The specification of the experimental engine is shown in table 2. This S.I engine is an inline single cylinder having a volume of 661.5 cc set at compression ratio of 10.5:1 having a constant speed of 1500 rpm and producing maximum power output of 4.4kW.

The addition of ethanol with the baseline fuel neat LPG were categorized based on quantity being added, such as 95% LPG with 5% ethanol (LE5) and 90% LPG with 10% ethanol (LE10), 80%LPG with 20% ethanol (LE20). The LPG fuel was stored in steel cylinder under 8 bar pressure and was regulated to 0.8 bar. Further this was sent through a secondary heater ensuring its vapor phase. The estimation of flow rate for LPG ethanol mixture was determined by energy basis. A flow control valve was used for adjusting the fuel flow and a TOSHNIWAL make turbine type flow meter was used to measure the fuel flow rate along with a flam trap arrangement. Figure 1 represents the layout of the experimental setup. The intake manifold consist of the air intake system, electronic gas valve, throttle body and an injector mounted near the port region. The intake air was measure by DRESSER make positive displacement type air flow meter. The throttle position was controlled by a BOSCH electronic throttle body ranging from 0 to 100. The injector was driven by separate microcontroller and has control over start and end of injection. The fuel injection system has a fuel pump connecting to the injector through a main line and a return line maintaining pressure of 3 bar. The return line consist of a gate valve which ensures that a constant injection pressure was maintained. A high resolution weighing pan was used for measuring the mass flow rate of the fuel being injected.

**Table 2.** Engine specifications.

|                       |  |
|-----------------------|--|
| Type                  | Four stroke, air cooled, single cylinder, overhead valve, compression ignition engine modified to run in the SI mode   |
| Make                  | Kirloskar TAF1   |
| Fuel                  | LPG, Gasoline, Ethanol   |
| Number of cylinders   | One  |
| Bore x stroke         | 87.5 x 110 mm  |
| Displacement volume   | 661.5 cc   |
| Compression ratio     | 10.5:1   |
| Connecting rod length | 231 mm   |
| Rated power           | 4.4 kW @ 1500 rpm  |
| Valve timing          | Inlet valve opening (IVO): 4.5° before Top Dead Centre<br>Inlet valve closing (IVC): 35.5° after Bottom Dead Centre<br>Exhaust valve opening (EVO): 35.5° before Bottom Dead Centre<br>Exhaust valve closing (EVC): 4.5° after Top Dead Centre |

As the engine load varies there is variation in flow rate of fuel from which the equivalence ratio was determined. In order to attain maximum brake torque (MBT) the ignition system was used to retard and advance of spark timing between 0 to 60 degrees.



**Figure 1.** Layout of the experimental setup.

The engine was loaded by means of DYNALEC make eddy current dynamometer that was coupled to the engine crank shaft through a propeller shaft, the speed was set to a constant mode of 1500 rpm and the load was varied.

The measurement of the pressure waves generated for each engine load was measured using the KISTLER make piezoresistive sensor that picks pressure signals from the intake manifold and similarly a KISTLER make piezoelectric sensor takes signals from the in-cylinder region. An AVL make optical crank angle encoder was used for obtaining the engine crank angle positions, and these signals were given to a data acquisition system which synchronizes these signals with the pressure signals and the obtained data were used later for combustion analysis.

The combustion analysis begins where the pressure sensors obtains data for 100 consecutive cycles which were then used for estimating average pressure rise, heat release rate and cumulative heat release rate, ignition delay, variations of peak pressure and COV of IMEP. A MATLAB algorithm was developed where the pressure data were given as input and the average pressure results were obtained denoting the peak in-cylinder pressure. The parameters that are necessary for giving as input are obtained from the data that are used for estimating the engine performance such as speed, torque, inlet temperature, exhaust temperature, spark timing, fuel flow rate for air and flow rate, calorific value, and the dimensional parameters such as connecting rod length, stroke length, bore diameter and compression ratio. This peak pressure gives the detail about work transfer occurring on the piston with simultaneous change in cylinder volume. The entire combustion process is accounted to take place in a closed system where work transfer occurs from the energy released during combustion process and in the later phase interaction of heat transfer occurs with the cylinder wall along with the movement of residual masses. It was assumed that there was no spatial variations and the model is considered to be zero dimensional. Hence based on the first law of thermodynamics the heat release leads to an approximation which is represented in the equation (1) shown below [19]. Where  $(dQ_n/dt)$  is the net heat release rate in to the system,  $\gamma$  is the ratio of specific heat ( $C_p/C_v$ ),  $p(dV/dt)$  denotes the work transfer taking place on the piston in the closed system. The residual gas constant was assumed to that of air, by using the standard expressions the value of  $C_p$  was determined from the composition of gases.

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma-1} p \frac{dV}{dt} + \frac{1}{\gamma-1} V \frac{dp}{dt} \quad (1)$$

The heat transfer was estimated by implementing the Hohenberg's correlation [20], as shown in the following equation (2). This model consists of  $h$  which is heat transfer coefficient in  $\frac{W}{m^2 K}$ ,  $C_1$  and  $C_2$  are tuning constants given by 130 and 1.4. The cylinder volume is denoted by  $V$  in  $m^3$ , the in-cylinder pressure  $P$  in terms of bar, the gas temperature is  $T$  in Kelvin and the mean piston speed is given by  $\bar{V}_p$  unit as m/s.

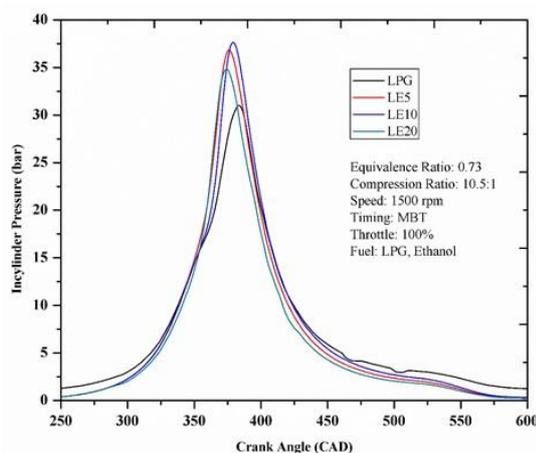
$$h = C_1 V^{-0.06} P^{0.8} T^{-0.4} (\bar{V}_p + C_2)^{0.8} \quad (2)$$

The ideal gas equation was used for determining the temperature of gases  $T$ ,  $t$  represents the time period,  $A_s$  is the area in  $m^2$  and  $T_w$  denotes the temperature of cylinder wall, and hence these values aid in estimating the heat transfer as well as the cumulative heat release rate. The exhaust emissions were determined using a HORIBA make five gas analyzer that measures the CO, HC, CO<sub>2</sub> and NO levels

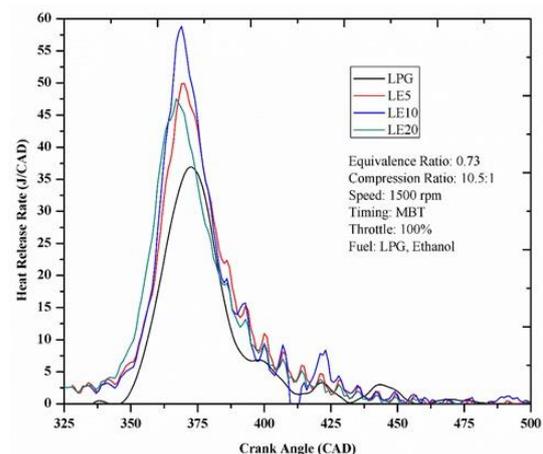
### 3. Results and discussion

The following section describes the combustion parameters such as in-cylinder pressure, heat release rate, cumulative heat release rate, ignition delay, COV of IMEP.

The in-cylinder pressure formation from the resultant combustion is described for all categories of fuel. The variation of in-cylinder pressure with respect to equivalence ratio can be seen from figure 2. It is observed that at the best thermal efficiency region of 0.73 equivalence ratio, among the LPG ethanol mixtures, LE10 indicated to have highest peak pressure of 37.65 bar at 379 crank angle degree (CAD), followed by LE5, LE20, and LPG having peak pressures of 33.60 bar at 379 CAD, 36.75 bar at 376 CAD, and 31.00 bar at 384 CAD. The oxygen content of ethanol tends to accelerate the flame propagation and combustion owing to the formation of higher cylinder peak pressure resulting in superior combustion. The addition of ethanol also provides wide spark timing which minimizes knocking and hence results in higher cylinder maximum pressure [21]. The presence of ethanol other than 10% in the mixture resulted in lower peak pressure which is observed in case of LE20 that results in excessive cooling effect due to latent heat of vaporization that lowers combustion temperature as well as lowering of heating values of the mixture which in turn reduces the overall peak pressure.



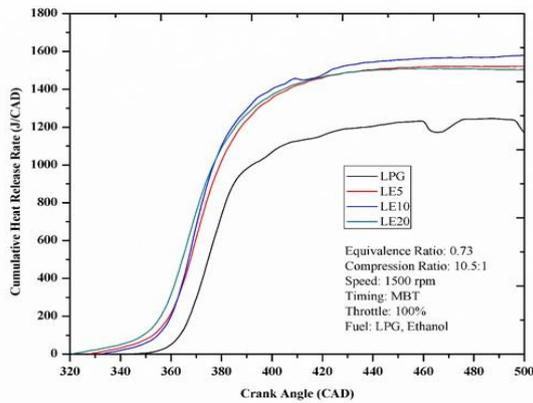
**Figure 2.** Variation of in-cylinder pressure with crank angle degree at 0.73 equivalence ratio.



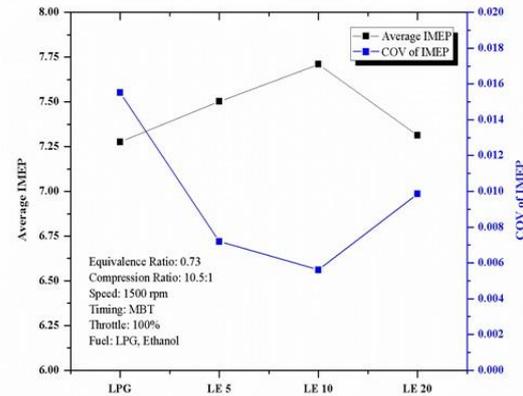
**Figure 3.** Variation of heat release rate with crank angle degree at 0.73 equivalence ratio.

The variation of heat release rate with respect to crank angle is shown in figure 3. The peak heat release rate was highest for LE10 of 64.88 J/CAD at 371 CAD followed by LE5, LE20 and LPG. Therefore the concentration of 10% ethanol present in the mixture causes a dense reaction zone of the charge mixture that results in higher flame velocity, hence the optimum levels of ethanol results towards faster burn rates due to their higher flame propagation reducing heat loss to the wall.

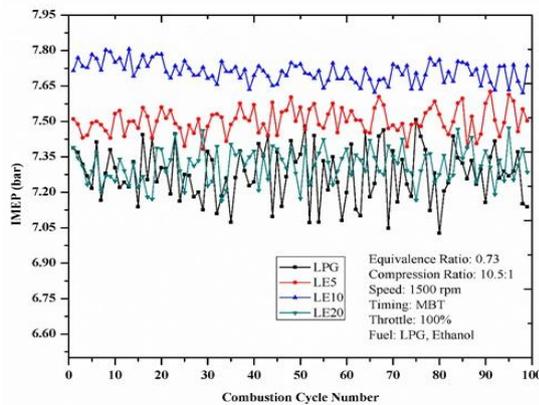
Therefore, the energy transfer occurs at maximum in doing the work on the piston as a result of higher combustion temperature and pressure which in a whole result in higher heat release rate. The excess content of ethanol raises the value of latent heat of vaporization in the mixture which results in lower combustion temperature causing lower heat release rate. The cumulative heat release rate with respect to crank angle is shown in figure 4 which indicates increase in heat release rate on the account of ethanol addition to LPG as well as gasoline. LE10 on a whole has the highest heat release rate of 1734 J/CAD at 505 CAD.



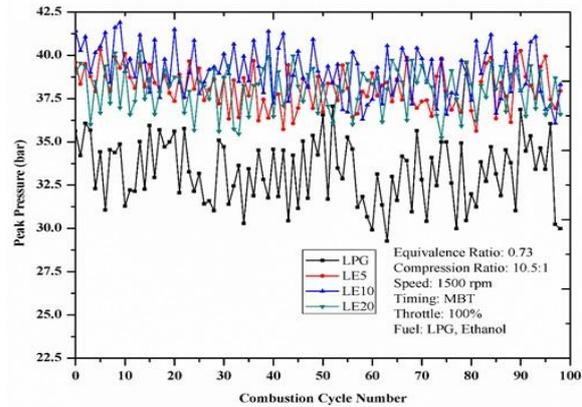
**Figure 4.** Variation of cumulative heat release rate with crank angle degree at 0.73 equivalence ratio.



**Figure 5.** Variation of average IMEP and COV of IMEP at equivalence ratio 0.73.

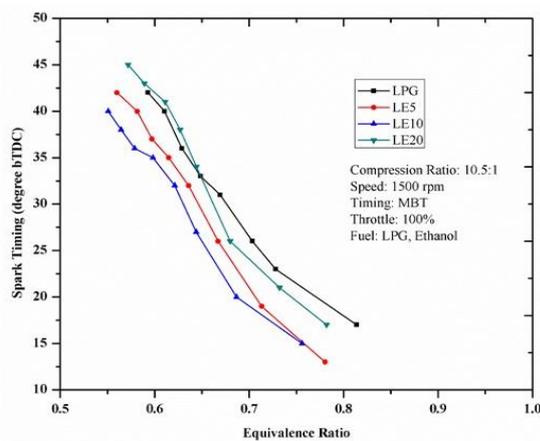


**Figure 6.** Variations of IMEP at equivalence ratio 0.73.

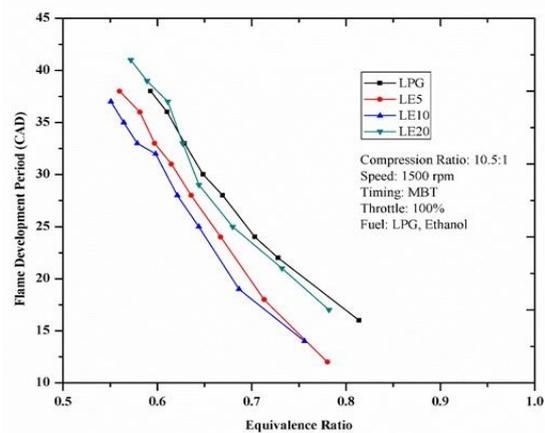


**Figure 7.** Variations of peak pressure at equivalence ratio 0.73.

The COV of IMEP and the cyclic variations are plotted in figure 5 and figure 6. At the maximum brake thermal efficiency region of 0.73, unlike stoichiometric combustion, lean burn combustion tends to misfire due to quenching at outer flame zones resulting in HC emission, reduced flame propagation, lower octane rated fuel that causes knocking at higher compression ratio and lesser range of spark timing. However, with the addition of ethanol, there is a major advantage in minimizing all these factors and sustains stable combustion. It can be seen that for LE10 mixture there is a positive effect upon minimizing variations in IMEP when compared with neat LPG. Beyond this, the addition of ethanol results in reduced combustion performance due to lower combustion temperature. The variation of peak pressure shown in figure 7 was observed to have minimum variations of peak pressure with the addition of ethanol owing to stable combustion.



**Figure 8.** Variation of spark timing with equivalence ratio.



**Figure 9.** Variation of flame development period with equivalence ratio.

The precise control over the spark timing determines the maximum brake torque. The variation of spark timing with equivalence ratio are shown in Figure 8. The higher flame speed characteristics of ethanol causes the combustion performance greater improvement at lower equivalence ratio since faster flame travels along the longitudinal axis of the flame. Therefore, faster burn rate of LE10 requires retarding of spark timing compared to flame zone which is disoriented near the vicinity of spark plug due to slower burn rate. The flame development period represents time period from start of ignition source until combustion takes place. Figure 9 shows the flame development period with equivalence ratio. The presence of ethanol results in increased flame velocities which is visible for LE10 that require lesser flame development period. This indicates that the presence of ethanol in the mixture with the specified ignition timing results in higher combustion performance.

#### 4. Conclusion

The experimentation was carried out by manifold injection of ethanol in carburetted LPG air mixture for the compression ratio 10.5:1 under wide open throttle, constant speed of 1500 rpm and the following conclusions were drawn. The combustion parameters were studied based on the interaction of ethanol with LPG fuel. The highest in-cylinder peak pressure and heat release pattern were observed in case of LE10. The stable combustion was also noted with least variations of IMEP. Overall, the use of 10% ethanol for the LPG ethanol mixture exhibited superior combustion characteristics. Hence, this method can be applied in lean burn engines and other dual fuel technologies where stable combustion performance is required.

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