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*International*
*Journal*
*Of Modern Engineering Research (IJMER)*

# Potential Use of LPG in A Medium Capacity Stationary HCCI Engine

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***ABSTRACT:*** *Internal combustion engines are extensively used in every field of life in today's world. Diesel engines being more efficient are preferred in the industrial and transportation sector in comparison to spark ignition engines for their higher efficiency, versatility and ruggedness. The major emissions of diesel engines are oxides of nitrogen (NOx), particulate matter (PM), carbon dioxide (CO2), carbon monoxide (CO). Among these emissions, oxides of nitrogen (NOx) and the particulate matter are the reasons of serious concern. For reduction of oxides of nitrogen (NOx) and particulate matter simultaneously, the use of Homogeneous Charge Compression Ignition (HCCI) have provided a sustainable solution in the present scenario.*

*This paper presents the findings of an experimental investigation into the operation of a compression ignition (CI) engine in homogeneous charge compression ignition (HCCI) mode using LPG as the main fuel and diesel as pilot injection. Factors that were investigated include engine performance and emission characteristics and based on the results, LPG was found to be a possible fuel for operation of a CI engine in HCCI mode.*

*A methodology using a small pilot quantity of diesel fuel injected during the compression stroke to improve the power density and operation control is carried out for an HCCI engine based on a stationary, constant RPM, water cooled diesel engine. The objective of this study is to investigate the performance and emission characteristics of HCCI engine fuelled with LPG and help understand the viability of LPG as an alternative fuel in diesel engines for use in the automotive industry.*

***Keywords:*** *LPG , Homogeneous Charge Compression Ignition (HCCI). Performance Characteristics, NOx Emissions,*

## I. INTRODUCTION

Oil accounts for 33.1% of global energy consumption. Global oil consumption grew by a below-average 0.6 million barrels per day (b/d), or 0.7%, to reach 88 million b/d [1]. In the present scenario of energy crisis and global warming, energy demand is exponentially increasing due to ever increasing number of vehicles employing internal combustion engines [2]. There is a growing need to decrease this consumption; Indian transport sector majorly consists of Diesel engines which carries many disadvantages such as more fuel consumption, lower power with higher level of harmful emissions. The excessive dependency on these Internal Combustion engines is forcing the automotive sector to utilize the technically advanced combustion technique to compensate with current demand.

Ever-increasing stringent legislations imposed by emission regulatory organizations on NOx emissions from engines make achieving near zero NOx emission combustion strategies more challenging. Hence, in-cylinder NOx reduction methods such as low temperature combustion are being widely studied. Homogeneous Charge Compression Ignition (HCCI) combustion is a reliable method that has been found to produce ultra-low NOx levels and near zero soot emissions and to provide equal or greater fuel conversion efficiencies compared to that of conventional Direct Injection (DI) diesel combustion [3, 4].

There is no direct control over the combustion timing in an HCCI engine, unlike the Diesel and Otto engines which use injection timing and spark timing respectively to control the start of combustion. Recently a lot of researches have been performed to investigate the potential control methods such as the inlet air heating [5, 6], variable compression ratio (VCR) [7, 8], variable valve actuation (VVA) [9, 10] and EGR rates [11, 12]. Moreover many studies also focused on the effects of different fuel physical and chemical properties, for instance the octane number and the cetane number, using the primary reference fuels and fuel additives [13, 14, 15, 16]. Within these attempts, attracting progresses have been reported, to some extent, about gaining control of the HCCI combustion process. In a HCCI engine, the fuel is injected into the (preheated) air in the intake manifold, to create a homogeneous charge. The charge is then further heated during the compression stroke to get auto-ignition, close to Top Dead Centre (TDC). With HCCI, there is no direct control over the ignition timing. The ignition timing can only be controlled indirect. By adjusting the operating parameters correctly, ignition will occur near TDC [17]. The auto-ignition timing in an HCCI engine is dependent on many parameters including equivalence ratio, pilot injection timing, fuel composition and properties, compression ratio, intake pressure and temperature, and external and internal EGR. The overall efficiency of the engine depends strongly on the combustion timing and duration. The homogeneous mixing of the fuel and air before the combustion, opposite to that in diesel engines, reduces the level of soot particles in the exhaust leading to cleaner combustion. HCCI provides up to a 30-percent fuel savings, while meeting current emissions standards. In regards to gasoline engines, the emission of throttle losses improves HCCI efficiency [18]. Recent research has shown that the use of two fuels with different reactivity's can help solve some of the difficulties of controlling HCCI ignition and burn rates. RCCI or Reactivity Controlled Compression Ignition has been demonstrated to provide highly efficient, low emissions operation over wide load and speed ranges [19].

HCCI engines do present some technical challenges that have so far kept them from widespread commercialization. The main hurdles are combustion timing control, low specific power output, high emissions of hydrocarbon (HC) and carbon monoxide (CO), and difficulty to start when cold [20]. These are formidable technical challenges for transportation applications, due to the fast transients required to meet the road load and due to the size restrictions inside a vehicle. However, for stationary applications, these issues are not nearly as challenging, because a stationary engine runs predominantly at a constant speed and the load changes relatively slowly. Under these conditions, combustion control becomes much more tractable. External components (e.g. a burner and heat exchanger for starting the engine can easily be installed in stationary engines, since size restrictions are typically not as strict as for transportation applications [21].

Liquefied Petroleum Gas (LPG) accounts for 16.8% of global energy consumption by 2011 end with production increased to 35.9% in less than 20 years [1]. As LPG has a low auto ignition temperature it is more likely that it will cause HCCI mode of ignition and is quite prone to knocking.

HCCI engines can operate on gasoline, diesel fuel, and most alternative fuels [22]. Further, on using LPG as a fuel in HCCI engine results in even lower levels of NOx and other particulate emissions are observed. LPG is made by liquefying petroleum gas (which is mainly composed of butane and propane), to less than 1% of the volume it occupies at standard atmospheric pressure. It is stored and distributed in hard containers in the liquid form with a pressure of 200–248 bar (2,900–3,600 psi), usually in cylindrical or spherical shapes. Being a gaseous fuel, LPG mixes easily and evenly in air, thus producing a homogeneous mixture essential for working of HCCI engine along-with increasing the life of lubricating oils, as LPG does not contaminate and dilute the crankcase oil. As LPG is less likely to ignite on hot surfaces, since it has a high auto-ignition temperature around 450 °C and a narrow range (5–15%) of flammability, therefore a methodology using a small pilot quantity of diesel fuel injected during the compression stroke to improve the power density and operation control is implemented, thus helping the homogeneous mixture of LPG and air to combust without altering the compression ratio of the engine or installing a spark plug inside the combustion chamber.

In present study the Compression Ignition engine is used which is modified to LPG-Diesel HCCI engine in which Diesel works on pilot-injection (volumetric ratio of 1/10) and LPG as the main fuel. Combustion control is carried out by LPG and its quantity varies at different loads. The objective is to investigate the performance and emission characteristics. This technique reduces NOx by 40.53% at 40% load and increases at 80% and 100% load. Kerosene when used as an alternative for diesel in HCCI engines at high compression ratio, higher BTE was observed with low exhaust emissions.

It is proposed that a small (<20% of fuel energy) pilot injection of a fuel (diesel) with high volatility and low auto-ignition temperature will act to initiate combustion of a premixed liquefied petroleum gas charge in an HCCI engine. The pilot fuel is injected shortly before TDC to burn as a stratified charge or a diffusion jet, and not as a homogeneous mixture. As such, it will cause a relatively slow rise in the cylinder pressure and temperature and cause the homogeneous natural gas charge to auto-ignite at an appropriate timing. The pilot injection is used to lower the variation in IMEP at the lean limit, as well as prevent misfiring of the engine at marginal conditions. It can also be used to control the combustion timing in an HCCI engine and increase the efficiency of fuels like liquefied petroleum gas that have very high reaction rates. With HCCI, cycle-to-cycle variations of combustion are very small since combustion initiation takes place at many points at the same time. HCCI has no flame propagation; instead the whole mixture burns close to homogeneous at the same time.

Till now the study in this field resulted into the engines used for power generation, but here the engine optimized for this use is an engine used in rural area as a transport unit so if we can control the combustion of this technology with emission control it could result into its commercialization.

## II. LITERATURE REVIEW

Diaz, p.m. et al. [23] did an experimental study on emission characteristics on a single cylinder engine, he blended reformer gas (RG) with different mass% with LPG, formed a lean preheated mixture which worked under the limit of misfiring and knocking which is 10-20 bar/CAD. Reformer gas mainly increased the maximum cylinder pressure and thus NOx increased when mass% RG is more than 60% and is reduced before it. Junjun Ma et al. [24] did experimental study on n-heptane diesel HCCI and concluded that NOx emissions decreases at low premixed ratios and increases at high premixed ratios, this is due to the diffusive combustion and increased peak pressure.

P. Saisirirat et al. [25] concluded that in HCCI combustion the usage of alcohol resulted into cooler flame and lover OH radicals, constant volume was kept hand in hand with pressure which is maintained at 20 and 40 bar to attain these conditions. S. Swami Nathan et al. [26] ran single cylinder diesel engine working on biogas-diesel HCCI and tested its performance and emissions where NOx emissions were decreased when the lean air-fuel mixture is given intake charge of 135 C thus less is the work done by compression to ignite fuel thus fuel burned evenly at multiple points and combustion control is done by adjusting amount of diesel injected with a constant supply of biogas, BTE was maximum at 135 C temperature as the expansion ratio of biogas is greater than diesel. Kimura S et al. [27] are continuously interested in developing a better understanding 0f the mechanism of HCCI combustion over a wide range of operating conditions. This study is motivated by the need to improve combustion efficiency and performance of current used engines and to reduce harmful pollutants emissions generated in the combustion. As a result many studies took place. Magnus Christensen et al. [28] used a Volvo TD100 Diesel engine was converted for HCCI operation with the capability of variable compression ratios, intake air preheating, air/fuel equivalence ratio and intake boost pressure. It was found that the level of intake charge preheating needed to accomplish ultra-lean HCCI combustion could be drastically reduced almost to the point of elimination with increasing inlet pressure (boost). The idea of charge dilution from excess air inducted from boost (creation of extremely lean charge) aids in the problematic nature of the speed of the combustion caused by the chemical reaction rates.

Rakesh Kumar Maurya et al. [29] did an experimental study where an HCCI engine was operated with a fixed compression ratio of 16.5 and fixed engine speed of 1500 RPM fuelled with ethanol. The results are reported with respect to $\lambda$ (the excess air ratio $\{1/\lambda = \Phi\}$) instead of equivalence ratio. In this study it was determined that at lower intake temperatures a richer mixture ( $\Phi = 0.5$ or $\lambda = 2$) could be ignited without causing engine knock, as the intake temperature was increased at the same rich equivalence ratio knock was encountered. However it was found that the lean operational range (up to $\Phi = 0.18$ or $\lambda = 5.5$) could be extended with increases in intake temperature. John E. Dec et al. [30] resulted maximum IMEP to 16.34 bar with no knock and intake pressure ringing at 3.25 bar. Noguchi et al. conducted a spectroscopic analysis on HCCI combustion in an opposed piston, two-stroke engine. They measured high levels of CHO, HO2, and O radicals within the cylinder prior to auto ignition, which demonstrated that pre-ignition chemical reactions had occurred and these reactions contributed to the auto ignition. After auto ignition took place, H, CH, and OH radicals were detected, which were indicative of high-temperature chemical reactions. In a traditional SI engine, these radical species are only associated with end-gas auto ignition, namely knock, which confirmed the similarities between the reactions of HCCI and knock in an SI engine.

## III. FUEL PROPERTIES

Table 1: Properties of LPG

| S.No. | Constituents | Value |
|---|---|---|
| **1.** | Molecular weight | 4409 |
| **2.** | Carbon content (wt %) | 8172 |
| **3.** | Hydrogen content (wt %) | 1828 |
| **4.** | Carbon: hydrogen ratio by weight | 447 |
| **5.** | Density of liquid at 15ºC (kg/l) | 510 |
| **6.** | Boiling point of liquid at atm. pres. (ºC) | -421 |
| **7.** | Density of gas at 15ºC & atm. pres. (kg/m3) | 186 |
| **8.** | Volume ratio of gas: liquid at STP* | 274:1 |
| **9.** | Vapour pressure at 20ºC (kPa abs.) | 710 |
| **10.** | Net calorific value at 25ºC (MJ/kg) | 460 |
| **11.** | Wobbe number (kcal/Nm3) | 19000 |
| **12.** | Limits of flammability in air (vol% gas) | 22 – 10 |
| **13.** | Limits of flammability in oxygen (vol% gas) | 2 – 50 |
| **14.** | Max. flame temperature in air (ºC) | 1930 |

| | | |
|---|---|---|
| **15.** | Max. flame temperature in oxygen (ºC) | 2740 |
| **16.** | Max. flame speed in 25 mm tube (cm/sec) | 82 |
| **17.** | Air required for combustion at STP (m3/kg LPG) | 1210 |
| **18.** | Air: gas vol. ratio for combustion at STP | 225 |
| **19.** | O2 vol. for combustion at STP (m3/kg fuel) | 256 |

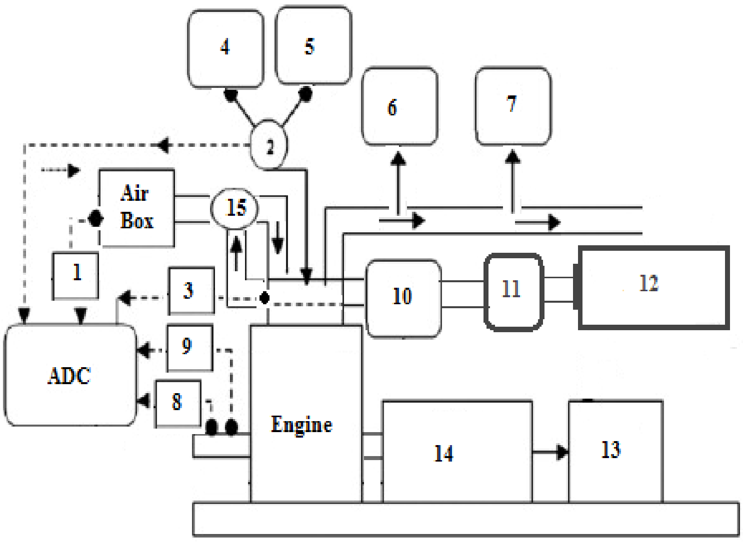
These properties of LPG is quite useful in controlling the combustion as this gas has flammability entirely different from the other fuels which brings changes in the type of combustion and temperature program which is followed with low peak in pressure and gives low harmful emissions with similar power output.

## IV. EXPERIMENTAL SETUP

The engine used for the study is Kirloskar Make, single cylinder, 4-stroke, constant speed, vertical, water cooled, direct injection, 5.2 kW diesel engine, Table 2. The engine was coupled to a swinging field separating exciting type DC generator and loaded by electrical resistance. Exhaust gas temperature was measured by an iron–constantan thermocouple. Fuel consumption was measured by U-tube manometer. The engine was started with standard diesel fuel and warmed up. The warm up period ends when the cooling water temperature is stabilized. Then the fuel consumption and different exhaust emission like $NO_x$, HC, CO, $CO_2$ and smoke were measured.

Table 2: Engine Specifications

| | |
|---|---|
| Make | Kirloskar |
| Ignition Type | Compression Ignition |
| Type of Cooling | Water Cooled |
| Injection Type | Direct Injection |
| Number of cylinders | 1 |
| Bore (mm) | 87.5 |
| Stroke (mm) | 110 |
| Cycle | 4-stroke |
| Compression Ratio | 17.5 |
| Rated Power (HP) | 5.2kW @1500 rpm |
| Swept Volume (cc) | 661 |




**1. Air flow sensor** | **2. Fuel flow sensor** | **3. Pressure sensor**
**4. Diesel tank** | **5. Biodiesel tank** | **6. AVL Digas Analyser**
**7. AVL Smoke meter** | **8. RPM sensor** | **9. Crank angle encoder**
**10. LPG Vaporiser** | **11. Multivalve** | **12. LPG cylinder**
**13. Load Cell** | **14. Eddy Current Dynamometer** | **15. Venturie**

Figure 1: Experimental Setup

The LPG is induced into the combustion chamber by installing a LPG gas kit (including a LPG cylinder, a multivalve, a vaporiser along-with a battery, lambda controller) along-with a venturi in the intake manifold just after the air filter. It assures proper mixing of the air and LPG and hence forming a homogenous and a lean air/fuel mixture before combustion. The quantity of the diesel entering the combustion chamber is controlled through the help of the diesel governor ensuring on pilot quantity of diesel entering the chamber enough for starting the combustion process of the lean mixture.

## V. EXPERIMENTAL PROCEDURE

In order to achieve HCCI the engine was initially hand started same likely the normal diesel engine with a steady and constant rpm program with no load condition or the Brake Mean Effective Pressure (BMEP) was maintained constant to 1 bar, then steadily the diesel consumption is minimised from the governor followed with the flow of LPG in the intake manifold with a constant ratio with diesel to maintain the steady rpm at 1500 and controlled combustion state. The intake was given a constant charge of 300 F. The limitations found in this operational working are misfire (at high LPG flow rates followed by low diesel flow rate) and Knocking (at low LPG flow rate followed by high diesel flow rate). Thus the misfire and knocking is controlled by injecting a controlled and appropriate amount of fuel and feasible combustion state is obtained followed with the stabilization.

Data points were recorded at steady state operating conditions. Engine speed for the tests was held constant at 1500 RPM. Data sets were taken at different loads of 0%, 20%, 40%, 60%, 80%, 100%(full load i.e. 5.2kW).

Each data set contained numerous points as:
-Amount of diesel consumption in 60 sec.
-NOx emissions
-CO emissions
-HC emissions
-smoke opacity
- λ value
-exhaust temperature
-voltage and current
The data sets were first taken without LPG Induction i.e. engine operating fully on diesel fuel and then repeated with LPG fuel along with pilot diesel injection as HCCI.

## VI. RESULTS AND DISCUSSIONS

### *6.1 Brake Thermal Efficiency (BTE):*

Fig .2 shows relationship between brake thermal efficiency (BTE) and brake mean effective pressure (BMEP). In all the three cases BTE increases with increase in BMEP. It is found that brake thermal efficiency of the engine when run on LPG-HCCI mode (with diesel pilot injection) is more than that when run on diesel alone at all loads. Power available at the crankshaft increases with increase in the load due to increase richness of the air fuel mixture but as with increase in the load mass flow rate also increases, so on the whole the outcome being increase in thermal efficiency. Also, as the loss of energy increases at higher loads the Brake thermal efficiency decreases at higher loads. Except for the engine loads that are very close to the maximum load (in the range 90–100%), this analysis combined with the analysis of the BSFC (Fig.2) may suggest that the higher the engine load, the better is its performance in terms of fuel consumption and thermal efficiency, as shown by some investigations [31].

### *6.2 Brake Specific Fuel Consumption:*

Fig .3 shows relationship between brake specific fuel consumption and brake mean effective pressure. Brake specific fuel consumption (BSFC) is a measure of the fuel efficiency of an engine. It is the rate of fuel consumption divided by the power produced. It may also be thought of as power-specific fuel consumption. BSFC decreases with increase in brake mean effective pressure in all the three cases as seen in the Fig.3, due to the increment in the value of BTE for higher loads. BSFC, being inversely proportional to Brake Thermal Efficiency, is highest for neat diesel mode, and lowest for normal pilot diesel assisted LPG-HCCI mode. To give the same power output i.e. to run on the same load range for different modes the engine starts to consume larger volume of fuel for diesel mode and LPG-HCCI water injection mode than normal pilot diesel assisted LPG-HCCI mode. With increase in the load, Brake specific fuel consumption will decrease. All these results agree with the literature [32, 33, 34].

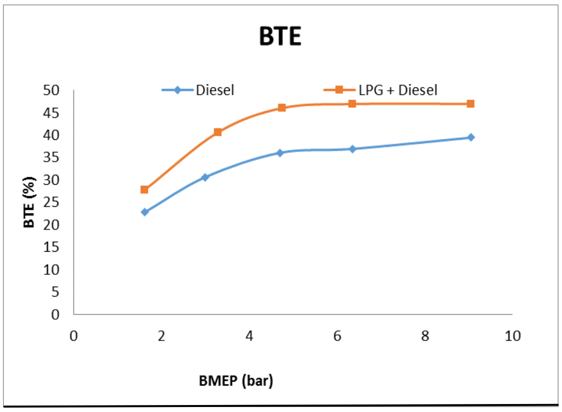



Figure 2: BTE vs BMEP

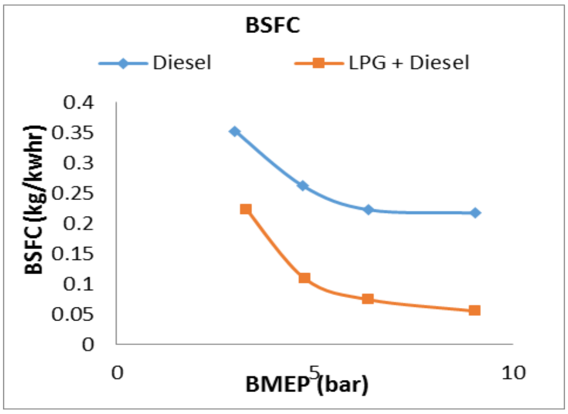



Figure 3: BSFC vs BMEP

### *6.3 Nitrogen Oxides (NOx):*

Fig.4 depicts the NOx versus the BMEP values for various load ranges for neat diesel mode, LPG-HCCI mode. NOx formation is highly sensitive to temperature in the combustion chamber. The NOx formation rate is governed by the Zeldovich Mechanism [35]. Homogeneous charge in combination with lean mixtures of LPG gives low maximum temperature as LPG storage is in high pressured liquid form and when mixed with normal air the temperature is maintained to <273 K which when enters the combustion chamber lower its temperature to a great extent and much lower NOx are observed.

HCCI combustion is well known for its ultra-low NOx and soot emission. Thus it is considered in this study that the NOx and soot emissions are mainly due to the diffusive combustion of direct injection. The formation of Nitrogen Monoxide and Nitrogen Dioxide can be divided into thermal route, prompt route, $N_2O$ route and fuel-bound nitrogen route [36]. The major NOx formation route in IC engine combustion is the thermal route [37]. The thermal NO route is the major constituent to the NOx emission and can be described with the following three elementary reactions called as extended Zeldovich mechanism

Eq. 1-3:

O+$N_2$®NO+N (1)

N + $O_2$®NO+O (2)

N+OH®NO+H (3)

It is evident from the Fig.4 that NOx is reduced dramatically with the pilot diesel aided HCCI mode engine running on LPG along-with exhibiting an increasing trend in NOx levels at higher loads, just as in a neat diesel engine. This reason being the trade-off relationship between the reduction of NOx during HCCI combustion and the increase of NOx during diffusive combustion. On one hand, the LPG undergoing HCCI combustion produces nearly no NOx so that increasing premixed ratio would reduce the overall NOx. On the other hand, the heat release prior to the diffusive combustion raised the in-cylinder temperature that offset the NOx decrease. Therefore at higher loads (above 85% load) the NOx emissions are greater

### *6.4 Hydrocarbons (HC):*

Fig.5 shows relationship between hydrocarbon formation and brake mean effective pressure for different loads (0%,20%, 40%, 60%, 80% and full load (5.2KW)). Hydrocarbon formation results from unburned carbon presence inside engine. The HC emissions mainly originate from the HCCI combustion since the prototype engine run is a diesel engine and without varying its compression ratio the complete combustion of LPG cannot take place and hence unburned HCs are emitted. The low combustion temperature prevents NOx formation, but the combustion temperature becomes too low to fully oxidize the fuel completely. This low combustion temperature results in high emissions of unburned hydrocarbons. During no load condition, unburned carbon is present inside the engine which go on to produce hydrocarbons when engine is run on diesel pilot injection assisted LPG, in great amount as compared to when run on diesel alone. The increase in HC can also partly be explained by incomplete vaporization of the fuel when the humidity of the inlet air becomes very high. Further as load increases, hydrocarbon formation decreases rapidly when engine is run on LPG (with diesel pilot injection) for both modes with and without water injection as higher temperature is achieved.

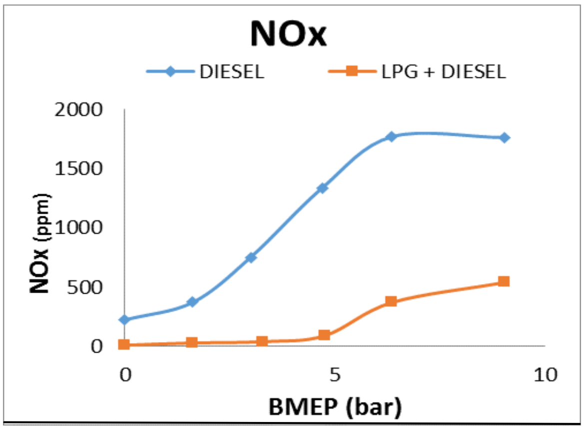



Figure 4: NOx vs BMEP

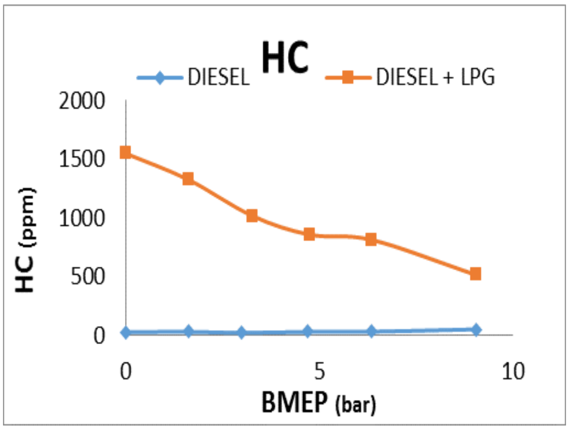



Figure 5: Hydrocarbons vs BMEP

### *6.5 Carbon Dioxide ($CO_2$):*

Fig 6 shows relationship between carbon dioxide and brake mean effective pressure. In this present study the $CO_2$ is found to be increase gradually with the increase in the load for diesel fuel but this pattern was not found to be uniform throughout the investigation. At the same time, at maximum load the $CO_2$ measured is lower in HCCI-LPG mode as compared to the diesel because of the complete combustion of the butane.

### *6.6 Smoke Opacity:*

Fig.7 shows relationship between smoke opacity (%) and brake mean effective pressure for different loads (0%, 20%, 40%, 60%, 80% and full load (5.2KW)). Smoke opacity increases with increase in the amount of unburned carbon. Unburned carbon will go on to form soot which will result in smoke formation. It is found when brake mean effective pressure approaches from 0 to 3 bar, smoke opacity increases exponentially in case when engine is run on diesel whereas it increases first then decreases in case when run on LPG (with diesel pilot injection). Amount of unburned carbon formed by LPG is more than that formed by diesel when brake mean effective pressure increases from 0 to 3.

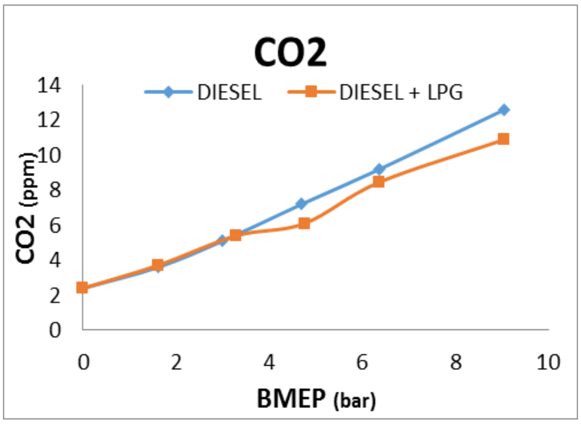



Figure 6: CO2 vs BMEP

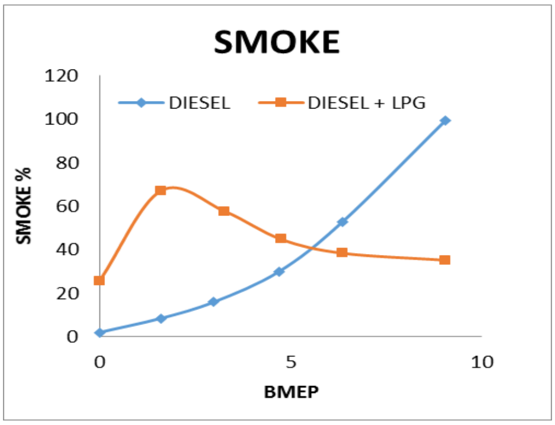



Figure 7: Smoke Opacity vs BMEP

## VII. Conclusions

NOx emissions decreased at low premixed ratios with the pilot diesel aided HCCI engine running on LPG. Generally the NOx formation in HCCI could be dramatically reduced in comparison with the prototype diesel engine. Note that the specific NOx levels continue to be high at high loads and further combustion enhancements is needed to make the LPG fuelled HCCI engine practical. Smoke opacity in HCCI mode decreases with increasing load. Overall smoke opacity in HCCI mode is quite low as compared with diesel mode. Extreme low levels of smoke opacity are attained at brake mean effective pressure being above 3 bar, when the engine is operated on HCCI mode. The change of carbon monoxide with premixed ratio mainly depends upon the premixed equivalence ratio which exceeds the critical value when the unburned hydrocarbon increases linearly with the premixed ratio due to the incomplete oxidation at the boundary layer and in the crevices. Higher Brake Thermal Efficiency was obtained at all loads and offers 20-25% overloads in HCCI mode in comparison to 10% in conventional diesel operation.

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