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Combustion Chamber Design for Lean Burn LPG Engines

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***Abstract*—Lean operation is an attractive operational method to increase thermal efficiency and to decrease exhaust emissions and fuel consumption. Gaseous fuels as clean, economical and abundant fuels can improve the lean operating limits. Liquefied petroleum gas (LPG) is one of the members of natural gases and declared as the cleaner fuel.**

Lean operation of homogeneous-charge spark-ignited engines reduces peak combustion temperatures, thereby reducing NOx emissions. Lean operation is normally restricted by the air-fuel ratio above which ignition is impossible, or combustion is incomplete. Operation under lean conditions also reduces the mixture burning rate, which can lead to increased spark advance and lower thermal efficiency.

In this paper, in order to increase the burning rate under lean air-fuel ratios combustion chamber shapes have been developed. The combustion chamber designs in such that they develop squish velocity inside combustion chamber, which generates increased levels of turbulence just before ignition and during the early phase of combustion. This increased burning rate gives the engine to operate with a smaller spark advance under lean conditions, thereby increasing the lean-limit of operation and increasing the thermal efficiency. The additional turbulence levels generated with the squish-jet type of combustion chamber is improves the completeness of combustion, thereby reducing unburned hydrocarbon emissions.

This paper presents three combustion chamber designs for lean operating LPG-SI engine, aimed at optimizing the squish velocity generation at optimized compression ratio as 11, in the mixture just before ignition and uses to increase the burning rate during lean operation, thereby increases the thermal efficiency and to decreases the exhaust emissions and fuel consumption.

Key Words— Lean burn, Spark Ignition, LPG, Thermal Efficiency, Economic Fuel Control, Emission Control.

I. INTRODUCTION

To meet future regulations for stringent emissions, LPG (Liquefied Petroleum Gas) fueled spark ignition engines are being used (1).A stoichiometric LPG fueled engine has limited applications due to high exhaust gas temperatures and a lower thermal efficiency. However, the lean burn technology implemented to overcome these difficulties.

Before 1970's a very few experimental works were made to study on the lean combustion technology in SI engines. This technology was studied first during 1908 to demonstrate the advantages of higher thermal efficiency (2).Later on the need for emission control and fuel economy improvement became evident and hence the lean combustion technology shows to offer the lower emissions, higher thermal efficiency and also improves the fuel economy (3).

The principal benefits of this operating technique are a reduction in greenhouse gas emissions and NOx emissions. Lean operation is normally restricted by the air-fuel ratio above which combustion is incomplete. A disadvantage of lean operation is that the burning rate can be significantly lower than with stoichiometric combustion. The reduction in burning rate increases the overall combustion duration and also leads to low flame velocities(4). For a successful implementation of the lean burn technology to decrease the exhaust emissions and increase the thermal efficiency, burn rate enhancement is necessary. A number of design variables such as spark plug location, intake port configuration and combustion chamber shape have been shown to influence the burn rate. Among these, "squish-jet" motion by using combustion chamber shape is having greater influence on the burn rate (5).

II. LITERATURE REVIEW

M.A.CEVIZ (6) investigated on the cyclic variations on liquefied petroleum gas (LPG) and gasoline lean burn spark ignition (SI) engine in order to reduce the cyclic variation in the SI engine; they use LPG as a fuel for the SI engine in terms of lean operation. Finally they concluded that higher laminar flame speed of LPG and good mixing of gaseous fuels in air causes the decreases in cyclic variation and LPG more suitable for lean operations in SI engines.

A.V. SITA RAMA RAJU (7) presented a paper which describe the effect of intensified swirl and squish on the performance of lean burn engine operated on liquefied petroleum gas (LPG).In order to produce the swirl and squish motion they masked the intake valve and provided swirl grooves on the piston crown. They found that combined swirl and squish configuration resulted in a small extension of the lean misfire limit and no significant change in the performance.

JOSEPH SHANKAL (8) observed the flame propagation in an liquefied petroleum gas (LPG) lean burn spark ignition engine, to investigate the combustion characteristics of a heavy duty liquefied petroleum gas lean burn engine. They varied swirl ratio and piston cavity configuration to investigate their effects on combustion and engine performance. Finally they concluded that with decreases in

mixture strength flame speed and exhaust temperature also decreases and by increasing the squish area burn duration of flame also decreases.
LIGUANG LI (9) proposed a study on liquefied petroleum gas (LPG) lean burn in a motor cycle spark ignition engine. They compared the lean burn limits with the parameters such as engine speed; compression and advanced spark ignition etc…are tested. They concluded that the emission of liquefied petroleum gas engine is significantly reduced and lean burn limit can be improved by using the higher compression ratio and spark ignition.
SULAIMAN (10) presented a paper to analyse the performance of single cylinder spark ignition engine running with liquefied petroleum gas (LPG) as a fuel. They found that decreased on power output up to 4 % compared to petrol due to volumetric efficiency and specific fuel consumption is reduced.
O.BADR (11) reported a paper on parametric study on the lean misfiring and knocking limits of gas fueled spark ignition engines. They used three different criteria for defining the engine lean limit. Finally they concluded that as the compression ratio increased, the misfiring limit of liquefied petroleum gas and air mixture slightly decreased.
ERIC KASTANIS (12) presented a paper on The Squish-Jet Combustion Chamber for Ultra-Lean Burn Natural Gas Engines. In this paper they use the blow in piston concept to generate the squish jet motion. Finally they concluded that squish jet design operated with more advanced MBT ignition timing than the blow in piston chamber.
MIKIO FURUYAMA AND XU BO YAN (13) reported a paper on Mixing Flow Phenomena of Natural Gas and Air in the Mixer of a CNG Vehicle. In this paper they accomplished visualization by means of the Schlieren method in a two dimensional flow channel model of a CNG engine mixer.

III. PRESENT WORK

In this proposed work a four stoke, single cylinder, air-cooled, stationary diesel engine is modified to run as a spark ignited gas engine by replacing the injector and fuel pump by a spark plug, a gas carburettor and an ignition system. Specifications of the test engine are shown in Table 1.In this present work squish jet piston concept used. This concept uses “bowl in piston” to generate a squish motion and series of jets of it directed towards the centre of combustion chamber. The basic piston which is used in the desiel engine with compression ratio(C.R) 17.5. The volume of the bowl in the piston was simultaneously changed without altering its clearance volume in order to maintain the compression ratio 11. The volume of the bowl in piston based on the theoretical calculations (14)

Table1. Specifications of engine

parameters	values
Cylinder bore (cm)	87
Stroke length (cm)	110
Compression ratio	11
Volumetric efficiency (%)	85
Speed (rpm)	1500
Power (kw)	4.4

The basic piston which is used in the desiel engine with compression ratio(C.R) 17.5 is shown in figure 1, and for this compression ratio swept volume and clearence volume produced by piston are following,

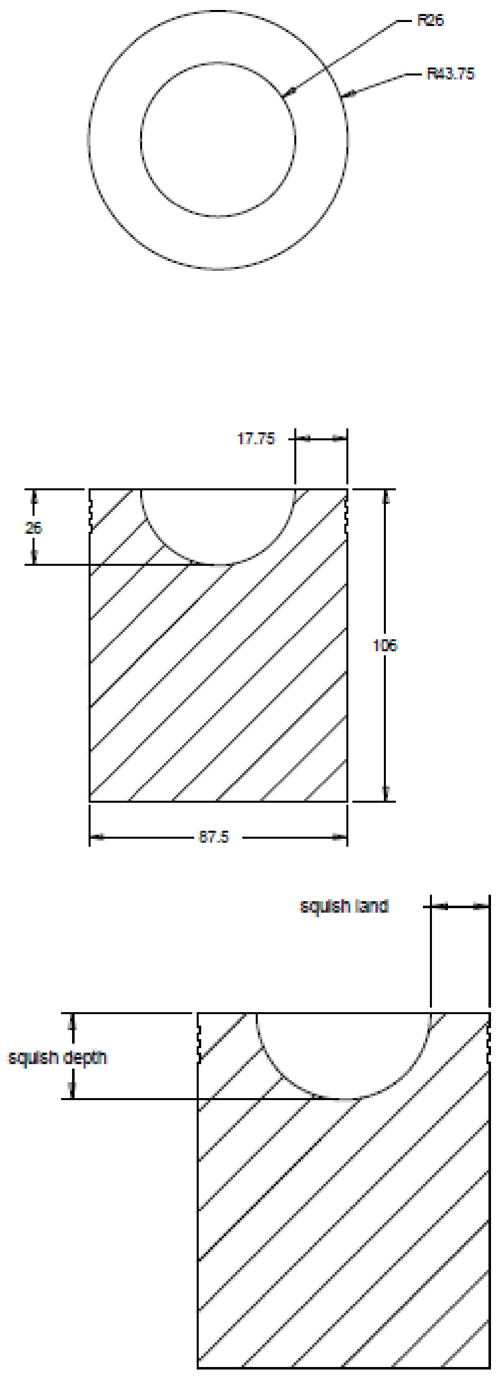

Fig.1 Basic pistion

$$r = \frac{\textbf{total volume}}{\textbf{clearence volume}} = \frac{\mathbf{v_s + v_c}}{\mathbf{v_c}}$$

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swept volume,

$V_s = (\pi/4)\times 87.5^2\times 110 = 661452.5\ mm^3 = 661.5\ cm^3$

clearance volume,

$V_c = 66.15\ cm^3$

In this present work the deisel engine is modified in to spark ignition (S.I) engine and simultaniously redusing the compression ratio to 11. For this compresion ratio the required clearence volume is 66.15 cm^3. To achieve this clearence volume and different squish velocities the piston bowl is modified as shown below by varying the squish land, piston bowl depth and clearence hight as shown in Model 1, model 2, model 3 and the corresponding values are tabulated in the table 2.

The squish velocity is determined by using the following formula,

$$\frac{v_{sq}}{S_p} = \frac{D_B}{4z}\left[\left(\frac{B}{D_B}\right)^2 - 1\right]\frac{V_B}{A_C z + V_B}$$

V_{sq} = squish velocity (m/s)
D_B = Bowl diameter (m)
B = Cylinder Bore (m)
V_B = Volume of the piston bowl (m^3)
A_c = Cross sectional area of the cylinder= ($\pi B^2/4$) in m^2
z = c+Z

Where, c=clearance height
Z=l+a-s
l=connecting rod length=220mm
a=crank radius=55mm
s=distance between the crank axis and the piston pin axis (m)

$$s = a\cos\theta + (l^2 - a^2\sin^2\theta)^{1/2}$$

Θ=crank angle

S_p = Instantaneous piston speed

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2}\sin\theta\left[1 + \frac{\cos\theta}{(R^2 - \sin^2\theta)^{\frac{1}{2}}}\right]$$

S_p=mean piston speed=2LN/60 (m/s)
L =stroke length(110mm)
N=crank speed (1500rpm)
R= l/a =4

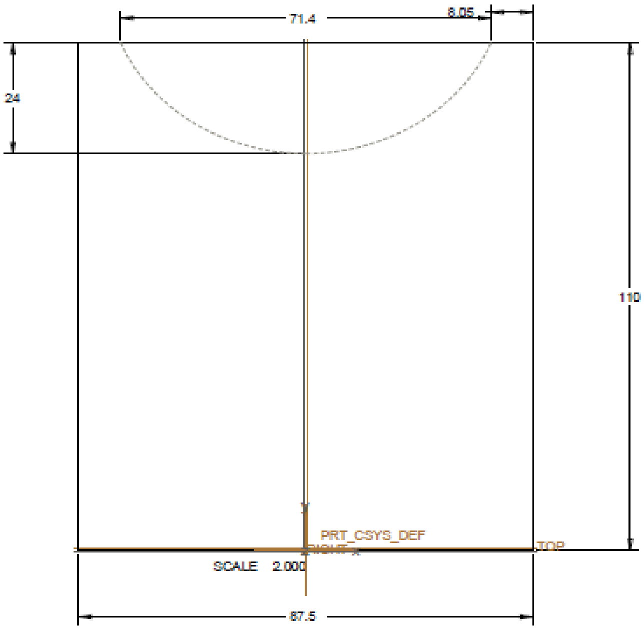

Fig.2 Mode 1(P1)

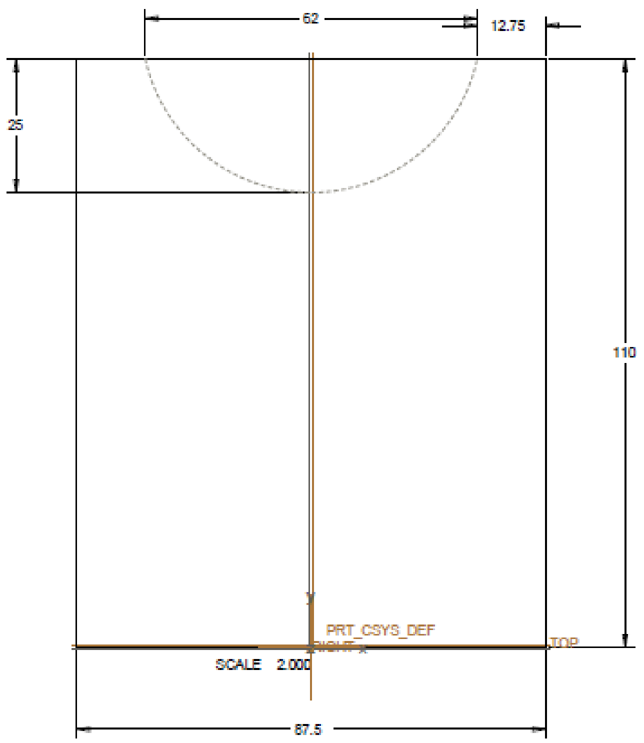

Fig.3 Model 2 (P2)

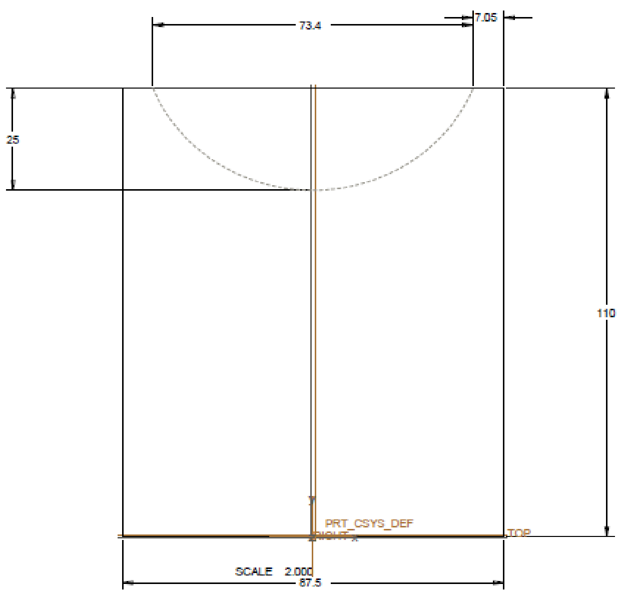

Fig4. Model 3 (P3)

Table 2:

Model	Clearance height (mm)	Squish land (mm)	Depth of bowl (mm)	Volume of bowl (cc)	Squish velocity (m/sec)
basic	0.846	17.75	25	35	17.79
P1	1.846	8.05	24	55.28	4.5
P2	2.52	12.75	25	45.91	5.5
P3	0.846	7.05	25	61.07	6.5

IV. EXPEREMENTAL SETUP AND EXEPEREMENTATION

The schematic diagram of the experimental setup is shown in figure. The modified engine was coupled to an eddy current dynamometer and LPG was supplied from a cylinder into the venturi of the gas carburetor through a pressure regulator, orifice meter, surge tank and a needle valve. The pressure drop across the orifice meter was measured with a micro-manometer to calculate the gas flow rate.

Arrangements were made to measure the temperature and pressure of LPG before it enters into the orifice meter.

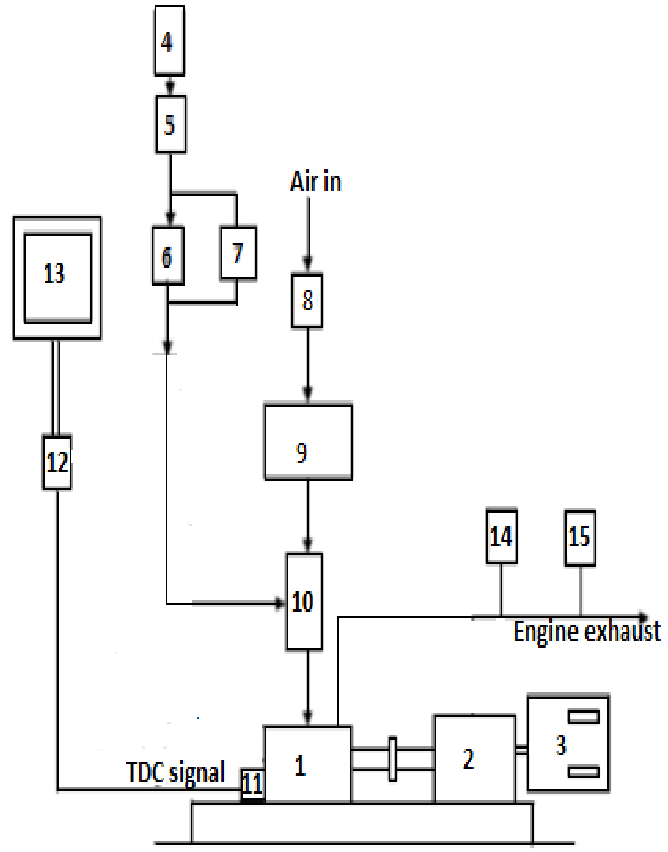

Fig.5 Experimental Setup

1. Engine 2 Dynamometer 3.Dynamometer controller 4.LPG cylinder 5.Pressure regulator 6. Vaporizer 7.Gas flow meter 8.Air flow meter 9.air drum 10.venturi 11.TDC encoder 12.Analog to Digital converter 13.Computer 14.HC/CO analyzer 15.Nox analyzer

Air flow rate was measured by using a flow meter. Pressure-crank angle data was acquired on a personal computer using a flush mounted piezoelectric pressure transducer. This data was processed by software to find combustion parameters. Arrangement was made for measuring the spark timing with the help of a stroboscope. A NDIR (non-dispersive infrared) gas analyzer was used for the measurement of HC & CO in the exhaust. A CLD (chemiluminiscence device) analyzer which works on the chemiluminiscence principle was used for measuring the NO concentration in the engine exhaust. Some relevant properties of LPG are given in Table 3.

Tests will conduct with LPG, using three different types of conventional "bowl-in piston" combustion chambers as shown in models (1, 2, 3). Initially, test conducted for finding out the MBT spark timing at different air fuel ratios and then performance test was conducted for three pistons at 25 % throttle and 100% throttle by using MBT spark timing.

Table 3.Properties of LPG

Parameter	Quantity	Units
Composition: Butane Propane	70 30	% by volume % by volume
Density	2.26	kg/m3
Calorific Value	47731	kJ/kg
Minimum Ignition Temperature	410	°C
Octane Number: Research Motor		 99 110
Flame Speed	0.37	m/s
Flammability Limits: Rich Lean	0.4019 1.9140	Excess air ratio

V.RESULTS AND DISCUSSIONS

Performance Parameters:

1. Brake Power:

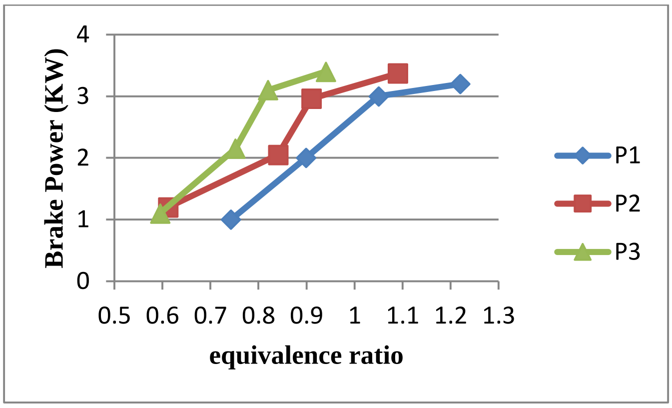

Fig.6 At 25% throttle

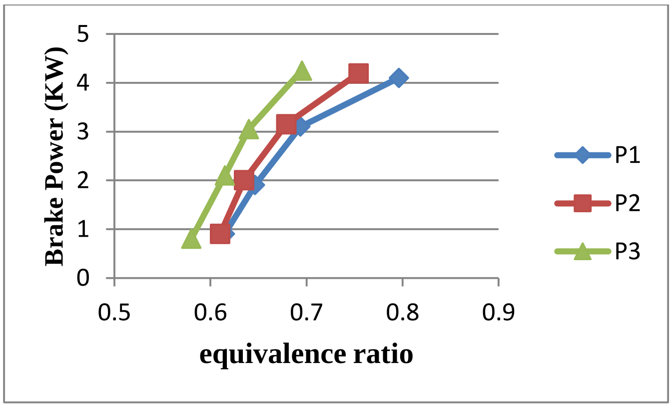

Fig.7 At 100% throttle

Figures 6,7 shows that variation of brake power with equivalence ratio at 25% and 100% throttle openings respectively. The model P3 gives maximum brake power 4.25 KW at equivalence ratio of 0.695 for 100% throttle opening.

2. Brake Thermal Efficiency:

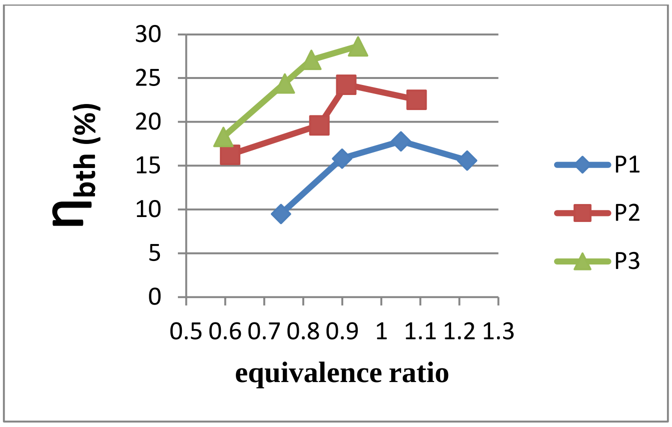

Fig.8 At 25% throttle

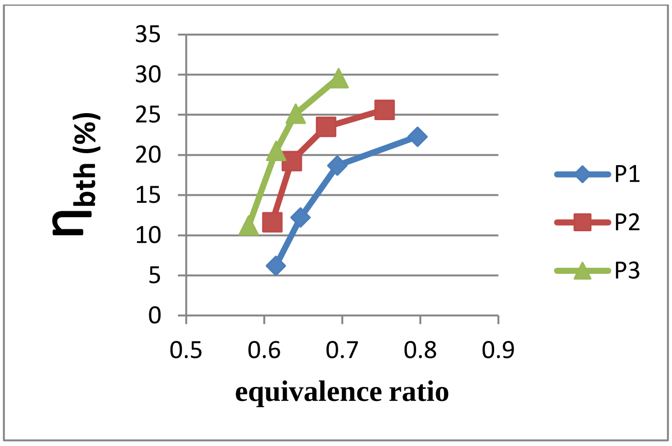

Fig.9 At 100% throttle

Brake thermal efficiency variation with equivalence ratio at 25% and 100% throttle are as shown in figures 8 and 9. The model P3 gives maximum brake thermal efficiency 29.56% at equivalence ratio of 0.695 for 100% throttle opening compared to other models.

3. Brake Specific Fuel Consumption:

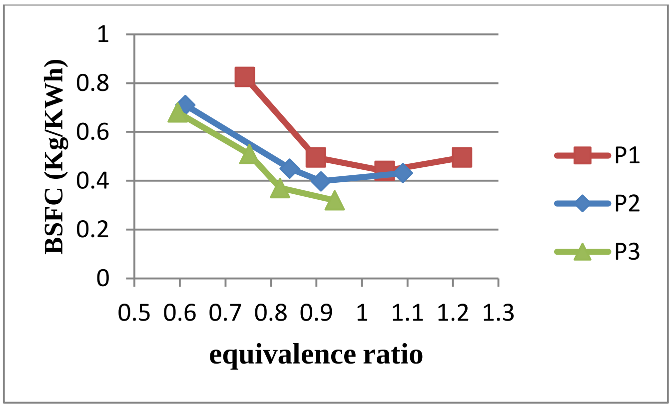

Fig.10 At 25% throttle

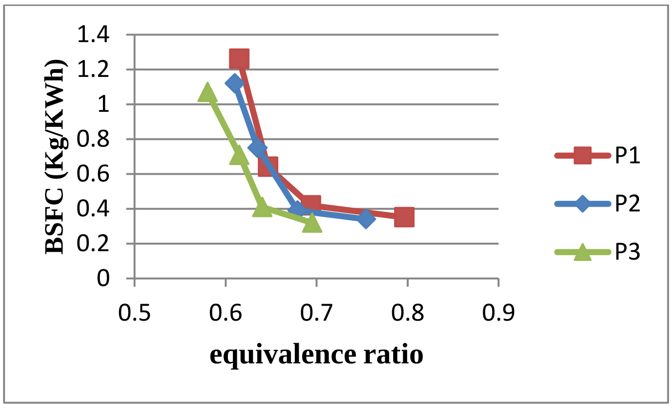

Fig.11 At 100% throttle

Figures 10,11 shows that variation of brake specific fuel consumption with equivalence ratio at 25% and 100% throttle openings respectively.

Emission Parameters:

1. HC Emissions:

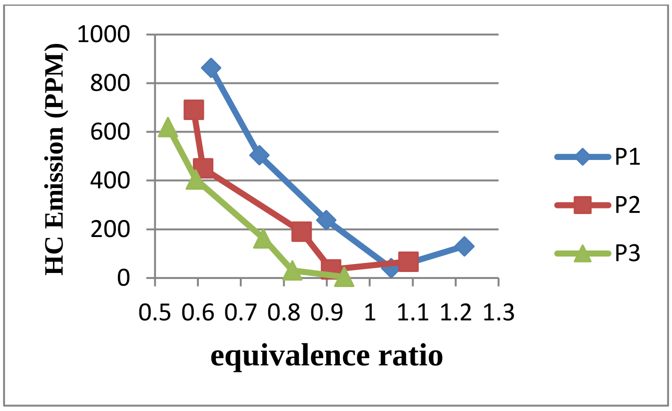

Fig.12 At 25% throttle

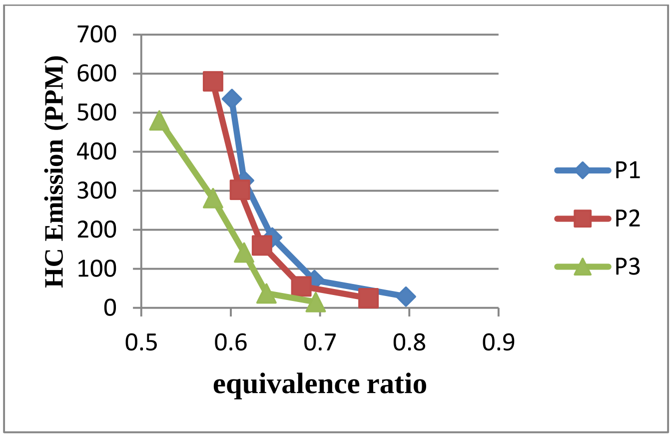

Fig.13 At 100% throttle

Hydrocarbon emission variation with equivalence ratio at 25% and 100% throttle are as shown in figures 12 and 13. The model P3 gives less HC emissions at 25% and 100% throttle openings compared to other models. We can see that as air fuel mixture becomes rich HC emissions increases.

2. CO Emissions:

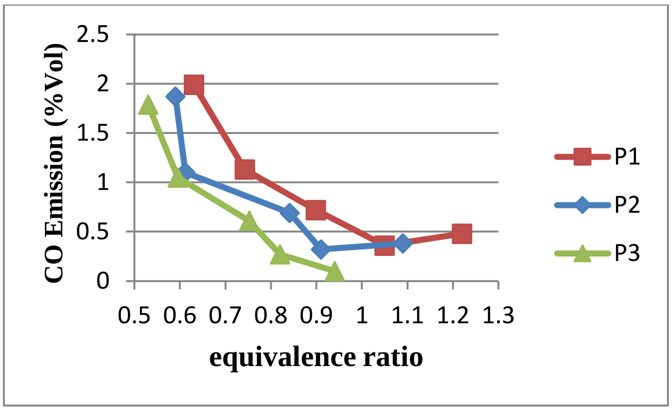

Fig14. At 25% throttle

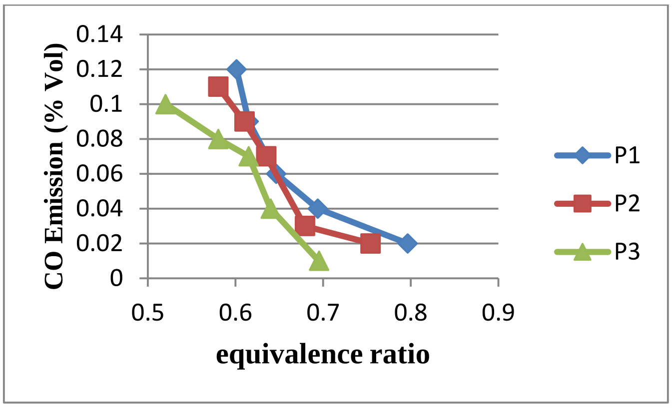

Fig15. At 100% throttle

Corbonmonoxide emissions variation with equivalence ratio at 25% and 100% throttle are as shown in figures 14 and 15. The model P3 gives less CO emissions at 25% and 100% throttle openings compared to other models. We can see that as air fuel mixture becomes rich CO emissions increases.

3. NOx Emissions:

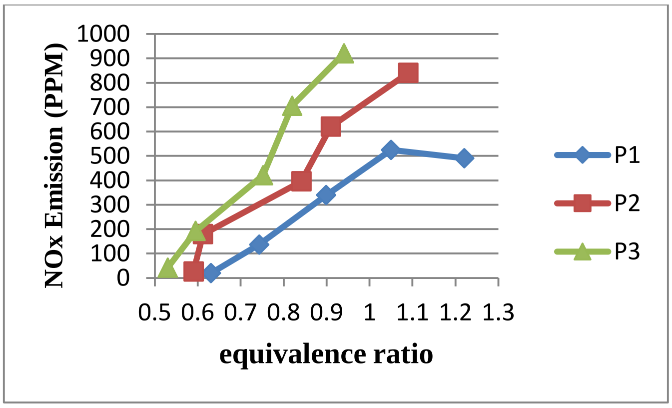

Fig.16 At 25% throttle

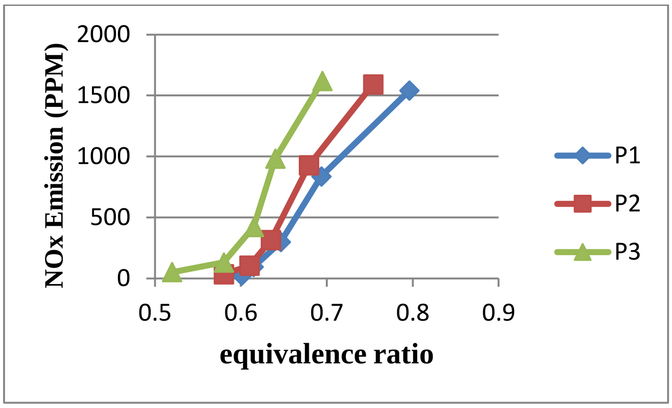

Fig.17 At 100% throttle

Nitric oxide emissions variation with equivalence ratio at 25% and 100% throttle are as shown in figures 16 and 17. The model P1 gives less NO_x emissions at 25% and 100% throttle openings compared to other models. We can see that as air fuel mixture becomes rich NO_x emissions decreases.

Combustion Parameters:

1. Spark Timing:

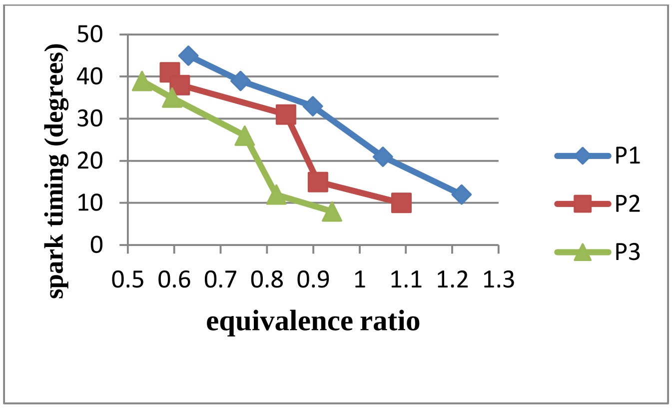

Fig.18 At 25% throttle

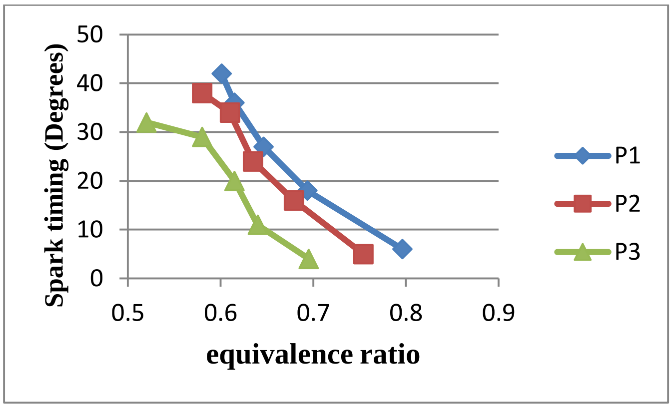

Fig.19 At 100% throttle

Spark timing (MBT) variation with equivalence ratio at 25% and 100% throttle are as shown in figures 18 and 19. With increase in squish velocity the combustion duration and ignition delay decreases.

VI. CONCLUSIONS

This experimental work was aimed at investigating the optimum squish velocity for an optimum compression ratio 11 based on performance and emissions of a lean burn operated on liquefied petroleum gas (LPG). Based on results obtained the following conclusions are drawn.

1. No significant improvement in power output and thermal efficiency, the model P3 gave best thermal efficiency and emission characteristics at 25%, 100% throttle compared to the model P2 and model P1.So the maximum possible squish velocity for compression ratio 11 is (6.56 m/sec) the optimum squish velocity corresponding to model P3.
2. With increase in squish velocity the lean burn limit increases at 25%, and 100% throttle openings. For model P3 lean burn limits are 0.53 and 0.52 at 25% and 100% throttle openings respectively.
3. With increase in squish velocity the combustion duration and ignition delay decreases.

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