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Combustion Characteristics of Late Injected CNG in a Spark Ignition Engine under Lean Operating Condition

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Abstract: Lean combustion strategy is used as a tool in the reduction of greenhouse gas emission in a natural gas fuelled engines. It also increases the thermal efficiency of the combustion. In this study, late injection and stratification effect of lean charge CNG combustion in a four stroke direct injection spark ignited engine was experimentally investigated. The fuel injection is kept at 90° Crank Angle (CA) Before Top Dead Center (BTDC) and the ignition timing at 28° CA BTDC at wide open throttle. Results from this study demonstrate that Indicated Mean Effective Pressure (IMEP), it's Coefficient Of Variation (COV), in cylinder pressure history, heat release rate and mass burn fraction all depend on the relative air fuel ratio (λ) and engine speed. With an increase in engine speed, the λ limit need to be increased so that the COV to remain well below 10%. However, the IMEP was found to decrease with an increase in λ. There existed an optimum λ for each specific engine speed where the peak cylinder pressure is maximum and the crank angle corresponding to peak pressure value is retarded.

Key words: Combustion, late injection, CNG, lean

INTRODUCTION

Greenhouse gas emission reduction, thus creating sustainable environment has become the main focus of global scientific world. Lion's share of emission of greenhouse gases is from the transportation sector. In US alone, more than half of all air pollution and 80% pollution in cities is caused by transportation sector. In the coming 25 years, energy demand for the transportation sector is projected to increase. This is due to an increase in ownership of motorized vehicles in both developing and developed countries and increased air travel in developing world as a result of urbanization (Zaretskaya, 2011). Reduction of emissions from transportation engines can be done by fuelling less pollutant fuels. Energy consumption for transport is the main sector which is heavily dependent on liquid and gaseous fuels, out of this gasoline takes the major share. The hydrogen to carbon ratio, 1.85 for gasoline, is very important factor in the greenhouse gas emission. An increase in the hydrogen to carbon ratio leads to less production of CO_2 and more production of H_2O in the combustion products (Basu, 2010). It is because of this reason substitution of gasoline with high hydrogen to carbon ratio (H:C) fuels like CNG and other biomass derived fuels. CNG on the other hand, has low flame speed, narrow combustible range and requiring high ignition energy. As a result, the combustion at low engine speeds is less complete with lower performance and higher CO and THC emissions. Moreover, application of CNG in automotive engines is coupled with engine performance reduction. This is mainly associated with the reduction of volumetric efficiency or displacement of air by CNG (Semin and Bakar, 2009). However, this fuel is regarded as the most promising and abundantly available fuel with high octane number (Cho and He, 2007; Zeng *et al*., 2006). As a result, CNG is applied at higher compression ratio (up to 15:1) engine to overcome the power reduction (Bhandari *et al*., 2005). Furthermore, with the emergence of direct injection application in spark ignited engines, lean combustion strategy has become a means for the reduction of greenhouse gas emissions and increasing thermal efficiency. This strategy is mainly accompanied with fuel stratification so that variable air-fuel ratio occurred around the combustion chamber. The stratification provides a relatively rich mixture near the igniter, compounded by ultra-lean uniformly mixed mixture all over the cylinder. Engine performance reduction can also be overcome by injecting the fuel very late after the inlet valve closes. However, this may lead to insufficient time for fuel-air mixing and slow combustion rate. The aim of this study is to investigate experimentally the effect of late injection

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and stratification on a lean operation condition of CNG combustion in a four stroke direct injection spark ignition engine.

MATERIALS AND METHODS

The experimental work was conducted in the Center for Automotive Research, Universiti Teknologi PETRONAS. The experimental test procedures follow the SAE standards of Engine Power Test Code (J1995-1995). The fuel injection was kept at 18 bar and the injector used in this experiment was Narrow Angle Injector (NAI) with a spray angle of 30°. NAI has better combustion performance over the Wide Angle Injection (WAI) at injection timings lower than 120°C BTDC due to its capability to create fuel stratification in the combustion chamber (Firmansyah and Aziz, 2011; Mohammed *et al*., 2011). The experimental matrix in this test is tabulated in Table 1. The parameters considered in the experiment are the load, Start of Injection (SOI), speed and the overall excess air ratio, sometimes also called relative air fuel ratio (λ).

An experimental study was conducted in a four-stroke, single cylinder, direct injection SI research engine setup, schematic diagram shown Fig. 1 and its specification listed in Table 2. It was coupled with an eddy current dynamometer to measure the brake torque. Engine parameters such as injection timing, ignition timing and air-fuel ratio are controlled by Engine Control Unit (ECU) connected to a remote interface installed in a desktop computer measuring torque (0-50 Nm), speed (0-120 rev sec^{-1}), ignition advance, static injection timing, running hours and 8 channel digital temperature.

Table 1: Experimental Matrix followed in the study

Fuel	Load	SOI (°CA BTDC)	Speed	λ
CNG	Wide open throttle	90	1800	1.5-max lean
			2100	limit with 0.1
			2400	resolution
			2700	
			3000	

Table 2: Specification of research engine used in the study

Engine properties	Specification
Number of cylinders	1
Displacement volume	399.25 cm^3
Cylinder bore	76.0 mm
Cylinder stroke	88.0 mm
Compression ratio	14 (Geometric)
Number of valves	4
Inlet valve open (IVO)	BTDC 12°
Inlet valve closed (IVC)	ABDC 48°
Exhaust valve open (EVO)	BBDC 45°
Exhaust valve closed (EVC)	ATDC 10°

(Al-Khairi *et al*., 2011; Anbese *et al*., 2011)

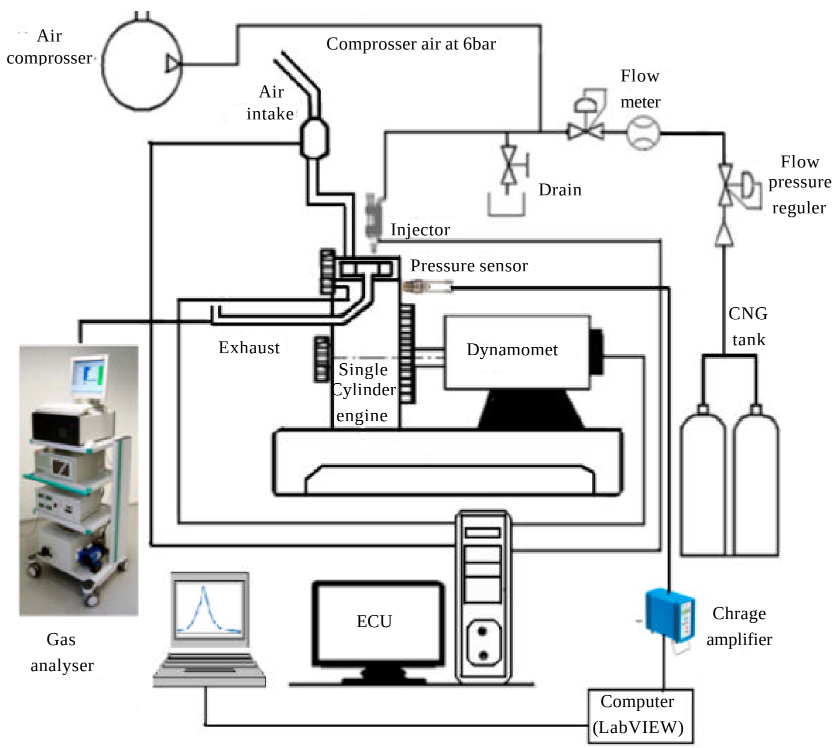

Fig. 1: Experimental setup arrangement

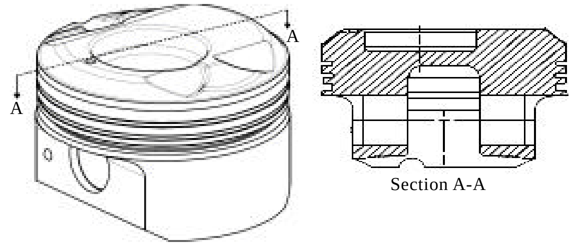

Fig. 2: Stratified charge piston head used in the study

Table 3: CNG Composition used in Malaysia

	Volumetric percentage	
Component	Leanest	Richest
Methane	96.42	89.04
Ethane	2.29	5.85
Propane	0.23	1.28
Iso-Butane	0.03	0.14
N-Butane	0.02	010
Iso-Pentane	NA	NA
N-Pentane	NA	NA
N-Hexane	NA	NA
Condensate	0.00	0.02
Nitrogen	0.44	0.47
CO_2	0.57	3.09

(Abdullah, 2009), NA: Not available

In this study the combustion analysis of the engine was performed with the help of pressure readings from the engine cylinder. A Kistler Piezoelectric pressure transducer was installed in the engine cylinder to record the in-cylinder pressure. The data capturing was synchronized with the crank angle encoder that determines the angular position for every pressure reading.

The fact is that, fuel is injected very late, a large piston bowl that creates fuel stratification in chamber is selected for combustion, it will reduce combustion instability and increase mixture distribution quality in cylinder to test stratified charge at lean air-fuel ratio (Fig. 2).

The ignition advance angle (θ_{ign}) was kept fixed at 28±1.5°CA BTDC.

The CNG used in this test was obtained from local NGV stations pressurized in a bottle with 200 bar. The gross heating value is in the range of 38.13-38.96 MJ kg^{-1}. Table 3 shows the properties and composition of CNG available and distributed in Malaysia.

RESULTS

The combustion study of late injected CNG under lean operation was conducted in a DI SI engine. The operation speed was varied from 1800-3000 rpm and air fuel ratio from stoichiometric to maximum lean under full load condition.

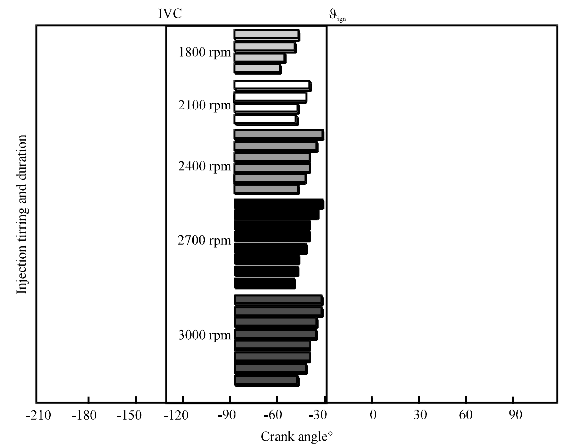

Fig. 3: Injection timing and duration for different engine speeds and their associated λ in reference to inlet valve closing and ignition advance crank angle

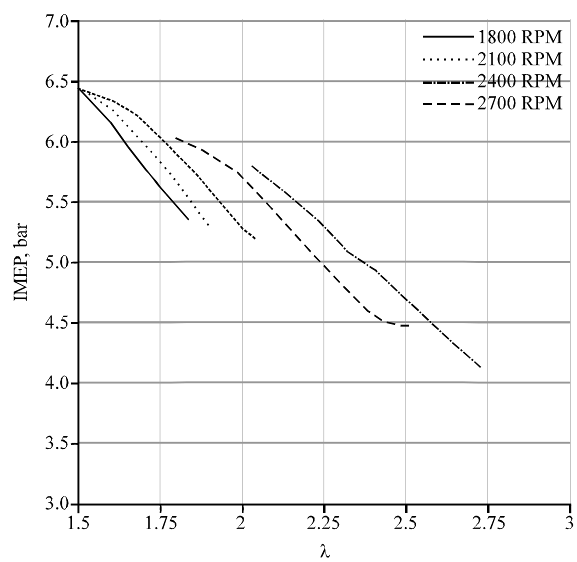

Fig. 4: Indicated mean effective pressure verses relative air-fuel ratio at different engine speeds

The relative air-fuel ratio of the study was varied from 1.5 to maximum lean with a resolution of 0.1. However, at engine speed of 2700 and 3000 rpm, combustion was unstable for relative air-fuel ratio less than 1.8 and 2.0, respectively. The start of injection and duration of injection for all operation speeds and their associated relative air-fuel ratio are depicted in Fig. 3.

Both the Indicated Mean Effective Pressure (IMEP) and its Coefficient of Variation (COV) as a function of λ are shown in Fig. 4 and 5, respectively for SOI of 90°. The IMEP of the engine shows a reducing trend with speed. As can be seen from Fig. 5, λ is also increasing with an

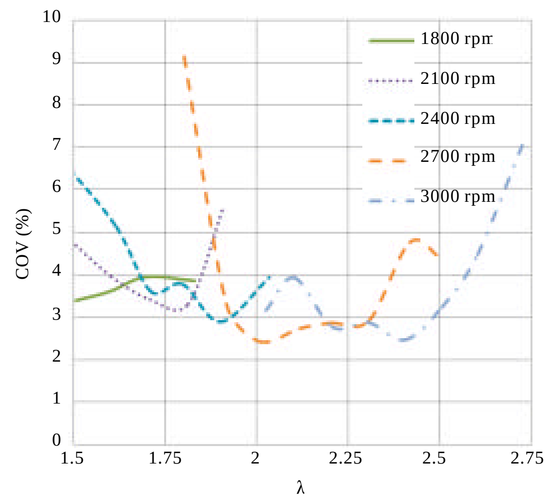

Fig. 5: Coefficient of variation verses relative air-fuel ratio at different engine speeds

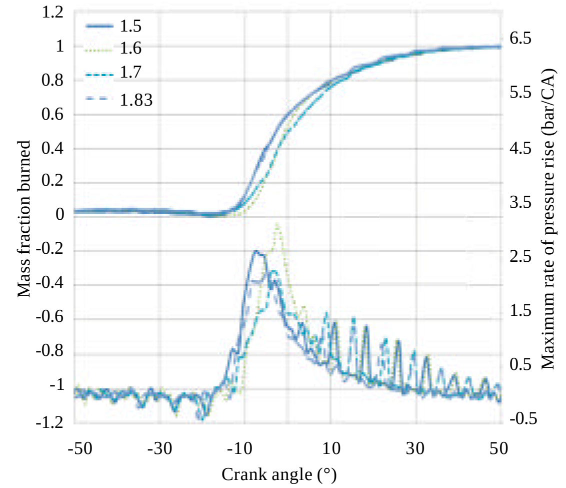

Fig. 7: Mass fraction burned and maximum rate of pressure rise at 1800 rpm

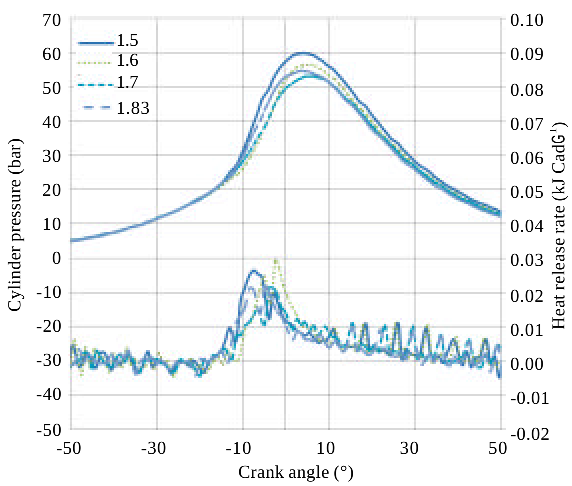

Fig. 6: Pressure history and heat release rate at 1800 rpm

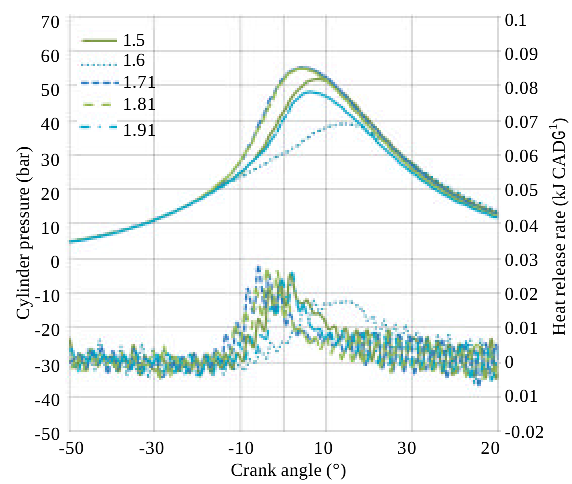

Fig. 8: Pressure history and heat release rate at 2100 rpm

increase in engine speed. The decrease in IMEP is therefore as a result of an increase in λ and due to the need for significant ignition retard at higher speeds. As can be seen from Fig. 4, COV was well below 10% and this put forward that combustion was steadfast.

The rate of heat release and mass fraction burned are estimated from the instantaneous cylinder pressure history. A MACRO code was developed in Microsoft Excel to analyze the data and estimate both the rate of heat release and mass fraction burned based on a single zone thermodynamic model developed by Rassweiler-Withrow (Abdullah, 2009). Analysis of both IMEP and COV is also included in the MACRO. Figure 6 though Fig. 15 shows the pressure history, heat release rate, mass fraction burned and maximum rate of pressure rise as a function of engine crank angle for different speeds ranging from 1800-3000 rpm.

In Fig. 6 and 7, the relative air-fuel ratios considered are 1.5, 1.6, 1.7 and 1.83. Combustion was unstable out of this range. Maximum pressure recorded with λ = 1.5. In all of the cases maximum pressure rise was seen around 5° CA ATDC. Moreover, fastest heat release with a very short rate was also recorded with λ = 1.5.

In Fig. 8 and 9, the relative air-fuel ratios considered are 1.5, 1.6, 1.71, 1.81 and 1.91. Combustion was unstable out of this range. Maximum pressure get increased with

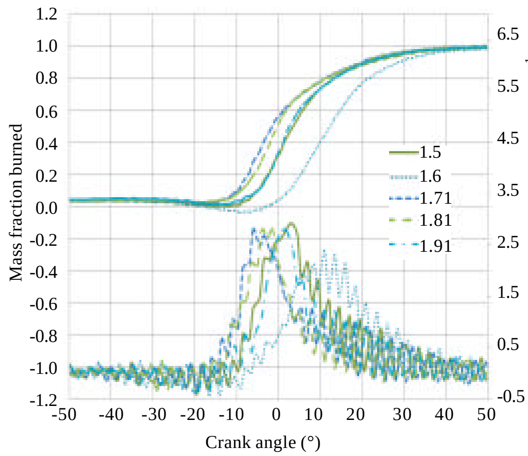

Fig. 9: Mass fraction burned and maximum rate of pressure rise at 2100 rpm

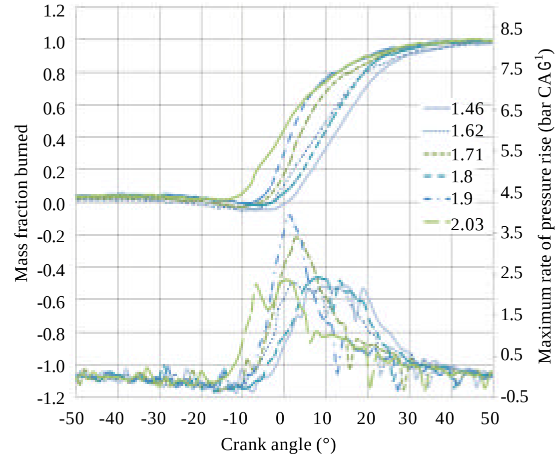

Fig. 11: Mass fraction burned and maximum rate of pressure rise at 2400 rpm

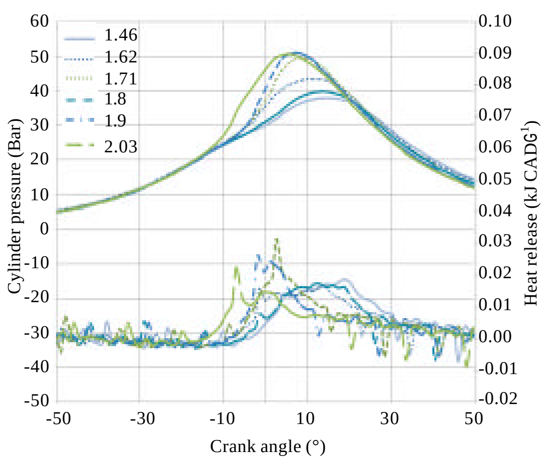

Fig. 10: Pressure history and heat release rate at 2400 rpm

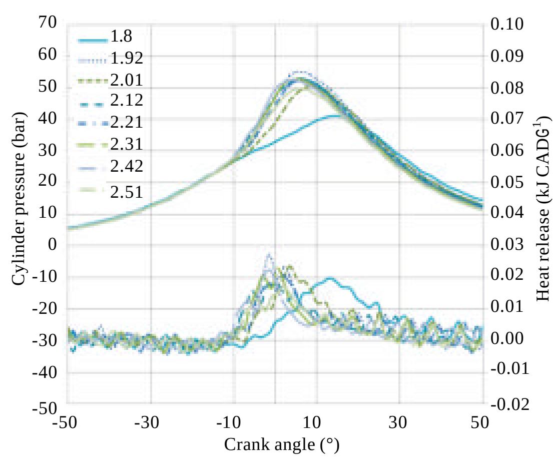

Fig. 12: Pressure history and heat release rate at 2700 rpm

both λ = 1.71 and 1.81. These two relative air-fuel ratios experienced similar in cylinder pressure history, heat release rate and the mass fraction of the fuel burned. It was also noted that the maximum pressure rise for these two relative air-fuel ratios was near to TDC. λ = 1.6 experienced decrease in the rate of pressure rise and in heat release rate. There was also retarding of heat release process. Rapid burning of the fuel was observed for both λ = 1.71 and 1.81 while reduction in rate of burning with λ = 1.6.

Figure 10 and 11 show the combustion characteristics at a speed of 2400 rpm. The relative air-fuel ratios considered in the specific speed are 1.46, 1.62, 1.72, 1.8, 1.9 and 2.03. An increase in peak pressure and heat release rates observed with an increase in λ. However, λ = 1.8 experienced different trend from both λ = 1.9 and 2.03. Major part of heat release occurred slightly before TDC for both λ = 1.9 and 2.03. Rapid burning of the fuel was observed for both λ = 1.9 and 2.03.

Figure 12 and 13 show the combustion characteristics at a speed of 2700 rpm. λ considered here ranges from 1.8-2.5. The pressure history, heat release rate, mass fraction burned and maximum rate of pressure rise experienced similar trend for all except for λ = 1.8. In this specific relative air-fuel ratio, there was decrease in the rate of pressure rise and in heat release rate. There was also retarding of heat release process. The effect of this λ can be further seen from Fig. 5 where the COV of the IMEP is seen maximum. Maximum pressure recorded with λ = 1.92 near TDC. Major part of heat release occurred around TDC for all λ except λ = 1.8. Rapid burning of the fuel was observed for both λ=1.92 and 2.42.

Figure 14 and 15 show the combustion characteristics at a speed of 3000 rpm. λ considered here ranges from

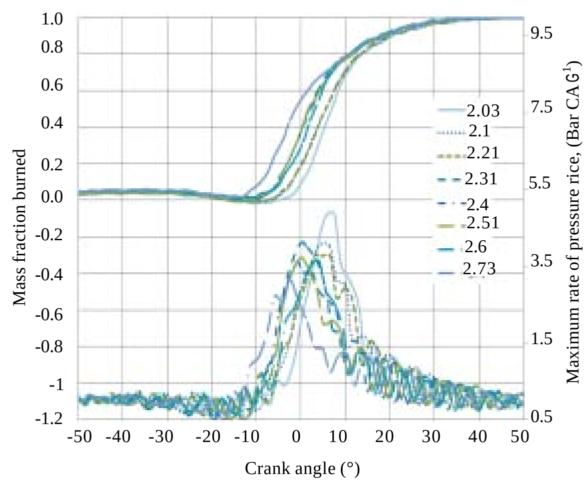

Fig. 13: Mass fraction burned and maximum rate of pressure rise at 2700 rpm

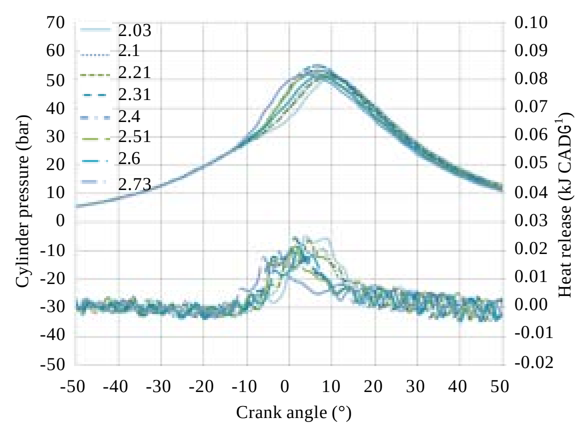

Fig. 14: Pressure history and heat release rate at 3000 rpm

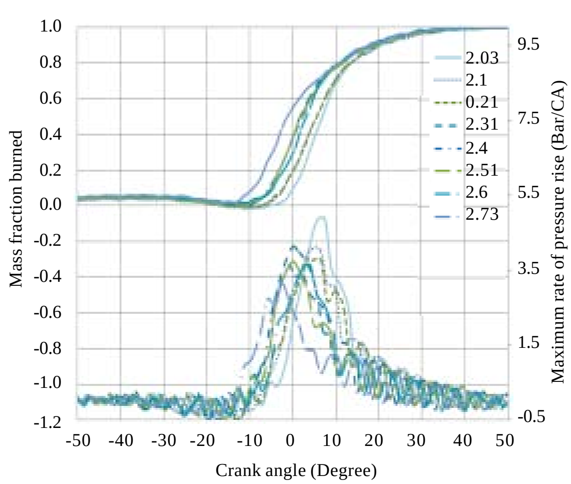

Fig. 15: Mass fraction burned and maximum rate of pressure rise at 3000 rpm

2.03-2.73. The pressure history, heat release rate, mass fraction burned and maximum rate of pressure rise experienced similar trend for all the relative air-fuel ratios except for $\lambda = 2.73$. In this specific relative air-fuel ratio, major part of heat release occurred slightly before TDC. Rapid burning of the fuel was also observed with this relative air-fuel ratio.

DISCUSSIONS

As can be seen from Fig. 3, duration of injection of fuel goes until the ignition advance angle. This reduces the time available for the fuel-air mixing, resulting in poor combustion stability and misfiring. Similar effect of shortening the duration of injection timing and ignition timing on the overall combustion is reported elsewhere (Huang *et al*., 2003; Zeng *et al*., 2006). As both the injection and ignition timing are fixed in this experiment. Smooth combustion was attained by increasing the relative air-fuel ratio, most importantly at higher engine speed. This arrangement has helped the fuel injection process to finish before the ignition starting.

It is observed from Figure 4 that IMEP decreased with an increase in relative air-fuel ratio. This is because the air-fuel mixture energy density is decreasing with an increase in relative air-fuel ratio. Insufficient time for fuel-air mixing and slow combustion has also partly influenced the IMEP. As can be seen from Fig. 5, only COV less than 10% are considered in this combustion data and the combustion instability near the lower relative air-fuel ratio limit is mainly associated to insufficient time for fuel-air mixing.

The engine peak pressure was recorded maximum at engine speed of 1800 rpm and minimum at engine speed of 2400 rpm. This is mainly associated with the energy density inside the engine. The relative air-fuel ratio operation range for the speed 1800 rpm is from 1.5-1.83 compared to 1.46-2.03 of that of 2400 rpm.

λ is observed to be increasing with an increase in engine speed, $\lambda = 2.73$ for a speed of 3000 rpm. The reason behind this is due to charge stratification in the combustion chamber. A 10% increase in lean limit with the application of partial stratified charge (Evans, 2009). Furthermore, there is high turbulence of charge in the combustion chamber due to the large piston bowl shape which also facilitates the combustion process.

CONCLUSIONS

Combustion characteristics of a late injected and stratified CNG was experimentally investigated at lean strategy in a four stroke direct injection spark ignition engine with a constant ignition advance angle and recapitulated as follows are the main conclusions:

- The authors have made an effort to improve the lean limit of CNG combustion by late injection under stratified charge. As a result, a relative air-fuel ratio of 2.73 was attained at a speed of 3000 rpm with a smooth combustion
- Besides, based on the pressure rise and the crank angle corresponding to the peak pressure value, the optimal relative air-fuel ratios for the specific engine speeds are 1.5, 1.71, 2.03, 2.41 and 2.73 for speeds 1800, 2100, 2400, 2700 and 3000 rpm, respectively. Maximum cylinder pressure and higher rate of pressure rise are registered with these relative air-fuel ratios. The crank angle corresponding to the peak pressure value is also seen to retard. Fastest heat release rate and shortest heat release duration was also witnessed with these relative air-fuel ratios
- In this study only the combustion characteristics was dealt and further investigation of the performance and emission is required in order to assert the above remarks, more importantly the effect of lean limit improvement on the brake specific fuel consumption, brake power, brake torque, CO and THC emissions

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