International Journal of Applied Engineering Research ISSN 0973-4562 Volume 13, Number 1 (2018) pp. 727-731


# Combustion Characteristics of Spark Ignition Engine Fuelled by Compressed Natural Gas in a Direct Injection Compressed Natural Gas Engine

**Saheed Wasiu, Rashid Abdul Aziz and Afiq Dahlan**

*Mechanical Engineering Section, University of Kuala Lumpur, Malaysia France Institute*
*Section 14, Jalan Teras Jernang, 43650, Bandar Baru Bangi, Selangor, Malaysia.*
**Corresponding author*
**Orcid id: 0000- 0002-1105-9214*

## Abstract

New alternative fuel is needed due to the increasing energy price of the conventional fuel (gasoline and diesel) couple with the environmental protection and fuel shortage which is a non-renewable fuel source. Research on improving the fuel economy and combustion characteristics of the internal combustion engine has become imperative. Compressed natural gas is the most promising alternative fuel to be utilized in the internal combustion engine. The reason being that the combustion characteristics of compressed natural gas generate a lower emission relative to conventional fuel. Thus, the purpose of this research was to study the combustion characteristics of a compressed natural gas in a direct injection compressed natural gas engine at stoichiometric and lean mixtures and quantitatively analyzed Cylinder pressure, Heat release rate and Mass fraction burned. The research was conducted experimentally under various engine conditions which were at stoichiometric and lean mixtures, wide open throttle and at various engine speeds so as to analyze the combustion characteristics of the engine. The result showed that, increasing the engine speed will increase the cylinder pressure due to the increase in burning velocity. While mass fraction burned also increase as the engine speed increase. This is due to the enhance turbulence within the engine combustion chamber. Heat release rate increase as the engine speed increases. This might largely be due to the increase in flame development duration within the combustion chamber. More so, the results also reveal that the maximum cylinder and heat release rate occurred at stoichiometric mixture. In addition, the maximum cylinder pressure for stoichiometric mixture peak at 61 kPa while for lean mixture was at 56 kPa which showed approximately 9% increment in cylinder pressure was achieved. While the heat release rate at 4000 rpm for both mixtures (stoichiometric and lean mixtures) under consideration respectively are 0.0219 kJ/ºCA and 0.0184 kJ/ºCA. This showed approximately 20% increment in heat release rate at that operating condition. More efficient and stable the combustion capable of enhancing the performance and drastically reduce the emission characteristics of the internal combustion engine were obtained.

**Keywords:** Direct Injection Compressed Natural Gas Engine (DI-CNG), Combustion characteristics, Stoichiometric and Lean mixtures; Wide Open throttled (WOT) and Engine Speeds.

## INTRODUCTION

Fuel plays a major impact in engine development. The earliest engine used for generating mechanical power burned gas. However, the conventional fuels which have dominated the ICE are basically petrol and diesel [1]. These fuels were preferred to be used because of their availability, competitive price and high energy density. The use of both petrol and diesel has resulted in a serious harmful impact on both environment and human health. In addition, these fuels are derived from petroleum oil which is non-renewable and it is expected to be totally consumed within the next few decades [1-3]. As the petroleum oil reserves declines, it cost is expected to increase dramatically in the future especially in the countries that have poor oil reserves and depends on imported oil for their energy supply. However, these factor put together enhance the investigation of alternative fuels in engine and seek the way to solve the fuel substitution and emission reduction. Natural gas is regarded as one of the most promising alternative fuels due to its lower cost, longer availability and probably one of the cleanest fuels in combustion [2-4]. The use of natural gas has been realized in both spark ignition and compression ignition engine. Natural gas comprises of mixture of different gases where methane is its major component. The combustion of natural gas produces lesser emission when compared to that of gasoline and diesel engine due to its simple chemical structure and absence of fuel evaporation. The engine possess high anti-knocking capability due to its high octane number and this allows it to operate at even high compression ratio; leading to further improvement of both power output and thermal efficiency [5-8]. Table 1 shows the combustion related properties for gasoline and CNG. However, the focus of this study research is to explore the combustion characteristics of the engine using compressed natural gas fuel.

Many researches have been conducted and published on the characteristics of internal combustion engine fueled by compressed natural gas [10-16]. However, no such research or study on the comparative combustion characteristics of the direct injection compressed natural gas engine fuelled by compressed natural gas at stoichiometric and lean mixtures has been conducted. Therefore, the research is to investigate the combustion characteristics of compressed natural gas in a direct injection compressed natural gas engine at stoichiometric and lean mixtures and clarify the combustion behavior of the direct injection compressed natural gas (DI-CNG) engine fuelled by compressed natural gas.

**Table 1**: Combustion Related Properties of Gasoline & CNG [9]

| **Properties** | **Gasoline** | **CNG** |
|---|---|---|
| Motor octane number | 80–90 | 130 |
| Molar mass (kg/mol) | 110 | 16.04 |
| Carbon weight fraction (mass %) | 87 | 75 |
| $(A/F)_s$ | 14.6 | 16.79 |
| Stoichiometric mixture density ($kg/m^3$) | 1.38 | 1.24 |
| Lower heating value (MJ/kg) | 43.6 | 47.377 |
| Lower heating value of stoic. mixture (MJ/kg) | 2.83 | 2.72 |
| Flammability limits (vol% in air) | 1.3–7.1 | 5–15 |
| Spontaneous ignition temperature (°C) | 480–550 | 645 |

## EXPERIMENTAL PROCEDURES

The specifications of the test engine are listed in Table 2 below. A four stroke single cylinder research engine was used to investigate the combustion characteristics of the direct injection compressed natural gas engine fuelled by compressed natural gas at stoichiometric and lean mixtures respectively. The schematic of the experimental set up is given in Figure 1 below.

**Table 2:** Engine Specification

| **Engine Properties** | |
|---|---|
| Displacement Volume | 399.25 $cm^3$ |
| Cylinder Bore | 76 mm |
| Cylinder Stroke | 88 mm |
| Compression Ratio | 14 |
| Exhaust Valve Open | BBDC 45$^o$ |
| Exhaust Valve Closed | ATDC 10$^o$ |
| Inlet Valve Open | BTDC 12$^o$ |
| Inlet Valve Closed | ABDC 48$^o$ |
| Dynamometer | Eddy current with maximum reading of 50 Nm |
| ECU | Orbital Inc |

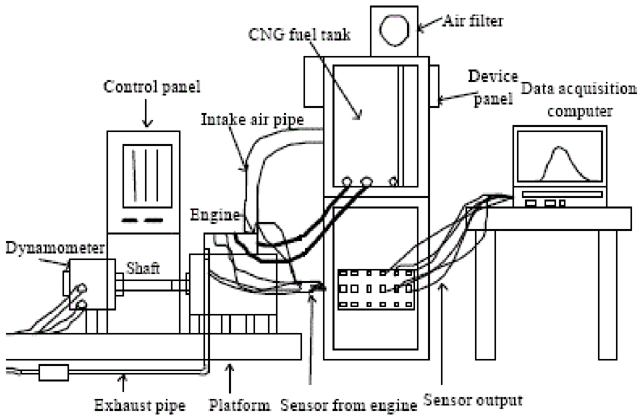



**Figure 1:** A schematic Diagram of The Experiment Set-up.

The supply system of the experiment set-up consists of fuel line, pressure gauge and flowmeter. The experiment was conducted after running the engine until the engine become stable where the temperature for oil and coolant are at 60$^o$C and 70$^o$C respectively. The experimental works was started by injecting compressed natural gas fuel into the engine cylinder. The compressed natural gas fuel was adjusted with the aid of regulating flowmeter to get the right mass flowrate through valve into the engine. Electronic control unit (ECU) was used in this experiment to control all the engine parameters as mentioned before which are wide open throttle, stoichiometric and lean mixtures and various engine speed. Combustion characteristics data such as cylinder pressure, mass fraction burn, and heat release rate was obtained with the aid of the combustion analyzer which comprises of

i. Piezoelectric Pressure Sensor
ii. Charge Amplifier
iii. Measurement Cable
iv. Shaft Encoder.

## RESULTS AND DISCUSSION

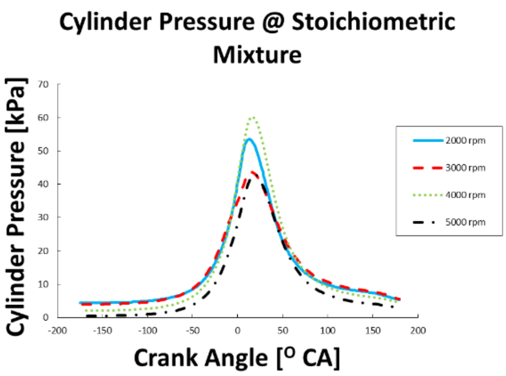
### Combustion Characteristics

The combustion characteristics of the direct injection compressed natural gas engine fuelled by compressed natural gas at stoichiometric and lean mixtures were analysed and shown graphically in figure 2, figure 3 and figure 4 respectively. The engine combustion characteristics is presented in terms of cylinder pressure, mass fraction burned and heat released rate to describe the engine output conditions. Different engine speeds were used to run the engine at 2000, 3000, 4000 and 5000 rpm respectively for the stoichiometric and lean mixtures. The research was carried out at wide open throttle, with injection timing at 300$^o$ CA (represented early injection timing) and relative air-fuel ratios set at $\lambda$=1.0 and $\lambda$=1.2 which represented stoichiometric and lean mixtures. Appendix A, B and C shown further details on the analysis of raw data for cylinder pressure, mass fraction burned and heat release.

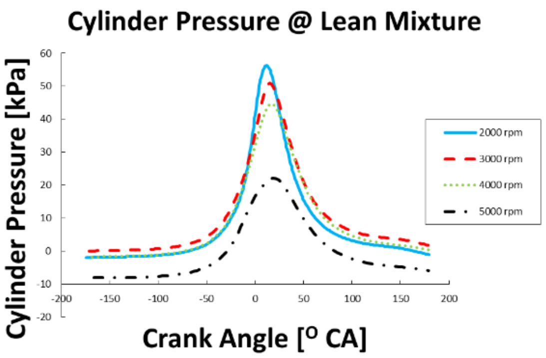
### *Cylinder Pressure*

Figure 2 shows the relationship of the cylinder pressure at various engine speeds for stoichiometric and lean mixtures by using direct injection compressed natural gas engine. There is a period which the fire created discharge to increase the pressure in the cylinder for the ignition to occur. As the flame keeps on developing and spread over the cylinder chamber, the cylinder pressure then increases more than the value it would have without burning. The cylinder charge is burned before achieving the maximum cylinder pressure which occur after top dead center (ATDC), before it start to diminish as the chamber volume keep on increasing amid the expansion stroke. For stoichiometric mixture, cylinder pressure change as a result of change in burning velocity under various engine speeds. Increasing the burning velocity will cause the cylinder pressure to increase variedly. Maximum cylinder pressure peak at stoichiometric mixture and at engine speed of 4000 rpm as compared to 2000 rpm, 3000 rpm and 5000 rpm respectively. This might largely be due to increase in engine speed increase the turbulence within the combustion chamber and consequent upon this, in-cylinder pressure will increase. While the lowest cylinder pressure occurs at 5000 rpm. This might be due to the cycle-by-cycle combustion variation occasioned by variation in the mixture composition.

Comparing the cylinder pressure between stoichiometric and lean mixtures, it is quite obvious that the higher cylinder pressure occurs at stoichiometric mixture. This is so because stoichiometric mixture will enhance the efficient combustion at that operating condition. While, the lower one occurs at lean mixture due to inefficient combustion (not all the oxygen available effectively utilized for the combustion process). The maximum cylinder pressure for stoichiometric mixture peak at 61 kPa while for lean mixture was at 56 kPa. This shows that almost 9% increment in cylinder pressure was achieved with stoichiometric mixture as compared to lean mixture. Thus, it is reasonable to say that for direct injection compressed natural gas (DI-CNG) engine at early injection timing will attained its highest combustion efficiency at stochiometric mixture with the maximum pressure cylinder occuring at 61 kPa and at engine speed of 4000 rpm engine speed as compared to lean mixture with maximum cylinder pressure occur 56 kPa at engine speed of 2000 rpm. A good agreement was achieved between these experimental results and [15].




(a)



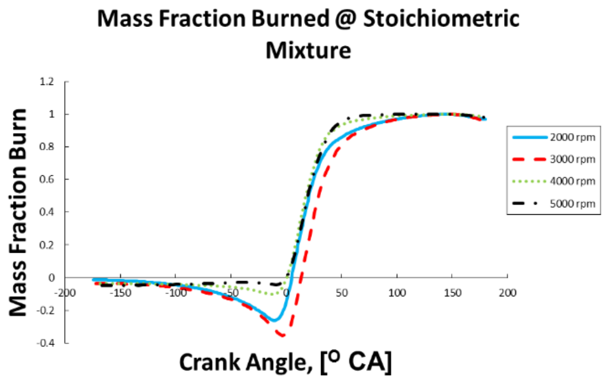

(b)

**Figure 2:** Cylinder pressure at various speeds for stoichiometric and lean mixtures.

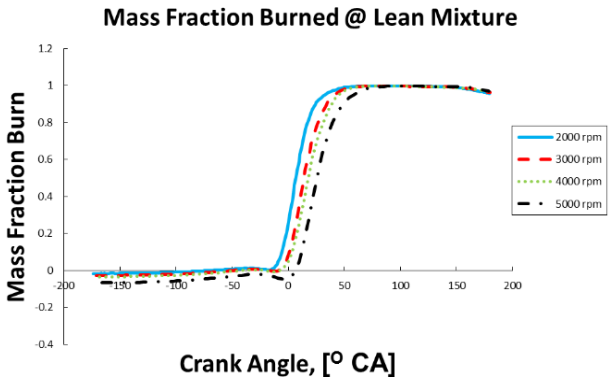
### Mass Fraction Burned

Figure 3 shows the relationship between mass fraction burned and crank angle at various engine speeds for stoichiometric and lean mixtures. At the start of the combustion process in figure 3, there is a flame development duration where the mass fraction burned increase with the crank angle until it reaches the maximum value approximately one (≈1) and become constant. Before that, at the beginning of the flame development the ignition delay might occurs which is influenced by the mixture (stoichiometric or lean mixture). At the end of mass fraction burned the value is approximately equal to one (≈1). If not, the combustion will be a wasted combustion which is either lean with excess air or rich combustion with excess fuel.

For stoichiometric mixture, the mass fraction burned shows a quick growth when increasing the engine speed. This is because the flame propagation speed varies with the increasing the engine speed. For evidence, at 5000 rpm the mass fraction burned reach maximum at one (1) at 136$^o$ crank angle (CA) after top dead center (ATDC) which is the fastest compared to 2000 rpm and 4000 rpm respectively while at 3000 rpm was the slowest, occurring at 142 $^o$ crank angle (CA) after top dead center (ATDC) due to the slower burning velocity. Figure 3 (a) supports this discussion. By referring to the figure 3 (a), The total combustion duration which is the sum of flame development duration (0%-10% mass fraction burnt) and rapid burn duration (10%-90% mass fraction burnt) at stoichiometric mixture for all the engine speeds under consideration occurs approximately at 65.5$^o$, 79$^o$, 39$^o$ and 35.5$^o$ crank angle (CA) after top dead center (ATDC) respectively. While for lean mixture, the optimum mass fraction burned occurs at 2000 rpm and the lowest occur at 5000 rpm. By referring to figure 3 (b), the total combustion duration at lean mixture for all the engine speeds under consideration occurs nearly at 14$^o$, 30$^o$, 33$^o$ and 46$^o$ crank angle (CA) after top dead center (ATDC) respectively. From the above analysis, it is obvious that; the shortest combustion duration occurs at 2000- rpm while the longest combustion duration occurs at 5000 rpm. Comparing the total combustion duration for the two mixtures under consideration (lean and stoichiometric mixtures), It reveals that the shortest combustion duration occurs at lean mixture due to inefficient combustion (not all the oxygen available are effectively utilized in the combustion process) while the stoichiometric mixture had the longest combustion duration due to efficient combustion. The results obtained here is coherent with what achieved in [16].




**(a)**

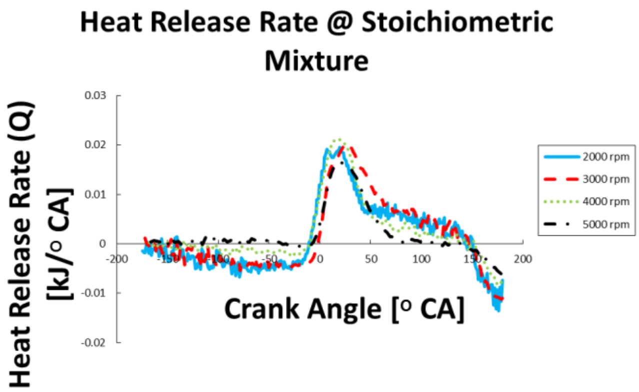



**(b)**

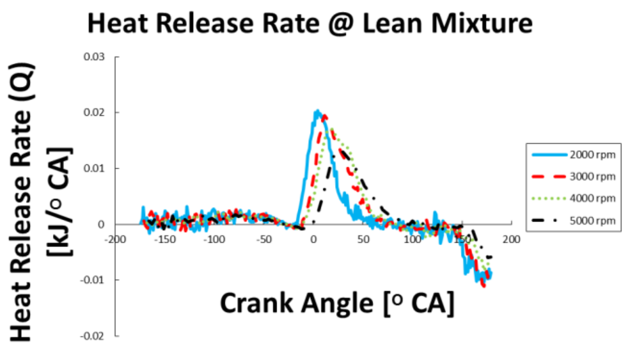
**Figure 3:** Mass fraction burned at various speeds for stoichiometric and lean mixtures.

### *Heat Release Rate*

Figure 4 shows the heat release rate at various engine speeds against crank angle for stoichiometric and lean mixtures respectively. From this figure, it is reasonable to say that cycle-by-cycle combustion variation had occurs in the early stage of flame development process. Increasing the engine speed caused the heat released rate to increase due to the increase in turbulence within the engine combustion chamber. For stoichiometric mixture, the highest released rate occurs at 4000 rpm, 0.0219 kJ/°CA due to the high turbulence in the combustion chamber which varies with the engine speed as compared to 2000 rpm, 3000 rpm and 5000 rpm (0.0218 kJ/°CA, 0.0206 kJ/°CA and 0.0174 kJ/°CA). More so, 5000 rpm had the lowest heat release rate with 0.0174 kJ/°CA. This might be largely due to the deterioration in the combustion quality which caused the heat release rate to drop drastically. While for lean mixture, the highest release rate was obtained at 2000 rpm which is 0.0248 kJ/°CA and 5000 rpm showed the lowest heat released rate 0.0135 kJ/°CA as compared to 3000 rpm and 4000 rpm (0.0207 kJ/°CA and 0.0184 kJ/°CA). It must however be noted that due to inefficient combustion (lack of proper mixing of fuel and air) of lean mixture that affected the turbulence in the combustion chamber, the rate of heat release was inversed with the effect of increasing the engine speed. Thus, the heat release rate will decrease as the engine speed increase. By comparing the heat release rate at 4000 rpm for both mixtures under consideration, stoichiometric mixture has the higher rate of heat release which occurs at 0.0219 kJ/°CA as compared to the lean mixture which is 0.0184 kJ/°CA. This is so because of an efficient combustion occurring at that operating condition. In addition, approximately 20% increment of heat release rate is achieved between the heat release rate for the operating condition under consideration (i.e. Heat release rate at stoichiometric mixture is nearly **20%** higher than at lean mixture for the operating condition under consideration). This is because of complete combustion occurring at stoichiometric mixtures which enhance the turbulence within the combustion chamber. A good agreement was achieved between these experimental results and [17].




**(a)**




**(b)**

**Figure 4:** Heat release at various speeds for stoichiometric and lean mixtures.

## CONCLUSIONS

An experimental study had been performed to study the combustion characteristics of the direct injection compressed natural gas (DI-CNG) engine fuelled by compressed natural gas. The main results are summarized below:

- Cylinder pressure change as a result of change in burning velocity under various engine speeds. Increasing the burning velocity will cause the cylinder pressure to increase variedly. For stoichiometric mixture, it has the maximum peak cylinder pressure while lean mixture had the minimum cylinder pressure.
- The mass fraction burned shows a rapid growth when increasing the engine speed. This is because the flame propagation speed varies with the increasing engine speed. The shortest combustion duration occurs at lean mixture due to inefficient combustion (not all the oxygen available is effectively utilized in the combustion process) while the stoichiometric mixture has the longest combustion duration due to efficient combustion.
- Heat release rate was affected by the turbulence condition in the combustion chamber. For stoichiometric mixtures, higher turbulence in the combustion chamber varied with the engine speed which produces more heat release rate while reverse is the case for the lean mixture (i.e Heat released rate vary inversely with the engine speed) due to inefficient combustion.

## ACKNOWLEDGEMENT

This research has been financially supported by Short Term Research Grant (STRG) provided by University of Kuala Lumpur (UniKL) (Grant No. UnikL / CoRI / strl 7086).

## REFERENCES

[1] Heywood, J. B. (1988). Internal combustion engine fundamentals. New York: McGraw-Hill.

[2] IANGV, IANGV reports. 2007.

[3] Mistry, C.S., Comparative Assessment on Performance of Multi cylinder Engine Using CNG, LPG and Petrol as a Fuel. SAE International, 2005(2005-01-1056).

[4] Kato, K., et al., Development of Engine for Natural Gas Vehicle. SAE, 1999(1999-01-0574).

[5] Rao, G. L. N., & Ramadhas, A. S. (2010). Compressed Natural Gas. Alternative Fuels for Transportation, 227.

[6] Ben,L., Dacros, NR., Truquet, R & Charnay,G. Influence of air/fuel ratio on cyclic variation and exhaust emission in natural gas (SI) engine,SAE ,1999 paper no.992901.

[7] Rousseau,S., Lemoult, B & Tazerout,M. Combustion characteristics of natural gas in a lean burn spark-ignition engine.Proc Inst Mech Eng Part D: j Automobile Engineering, 1999. 213(D5), pp 481-9.

[8] Cho, H. M., & He, B. Q. (2007). Spark ignition natural gas engines—A review. Energy Conversion and Management, 48(2), 608-618.

[9] Huang, Z., et al., Basic Characteristics of direct injection combustion fuelled with compressed natural gas and gasoline using a rapid compression machine. Proc. Instn Mech. Engrs. Part D: j. Automobile Engineering, 2003. 217.

[10] Hatakeyama, S., Murayama, T., Sekiya, Y., Nakai, S., Sako, T., and Tsunemoto, H., Proc. of the 16th International Combustion Engine Symposium, Tokyo, (2000– 9), 277, (in Japanese).

[11] Fitwi Y., et al ., Combustion characteristics of late Injected CNG in a Spark Ignition Engine under Lean Operating Condition. Journal Applied Science 12 (23): 2368-2375, 2012.

[12] M.A. Kalam, H.H. Masjuki and M.A. Maleque, Gasoline engine operated on compressed natural gas. Proc. Advances in Malaysian Energy Research (AMER), Malaysia (2001), pp. 307–316. M.U.

[13] Aslam, M. U., Masjuki, H. H., Kalam, M. A., Abdesselam, H., Mahlia, T. M. I., & Amalina, M. A. (2006). An experimental investigation of CNG as an alternative fuel for a retrofitted gasoline vehicle. Fuel, 85(5), 717-724.

[14] Uger Kesgin, Study on prediction of the effects of design and operating parameters on NOx emissions from a lean burn natural gas engine. Energy Conversion and Management 44 (2003), pp. 907–921.

[15] Rałiu, S., Nicolae, G., & Vasile, P. (2009). Monitoring of the Pressure Inside the Cylinder for an Internal-Combustion Engine, 8(1), 105–114.

[16] Cooney, C., Worm, J., Michalek, D., & Naber, J. (2008). Wiebe Function Parameter Determination For Mass Fraction Burn Calculation In An Ethanol-Gasoline Fuelled Si Engine, 15(3), 567–574.

[17] Ma, F., Zhao, C., Zhang, F., Zhao, Z., Zhang, Z., Xie, Z., & Wang, H. (2015). An Experimental Investigation on the Combustion and Heat Release Characteristics of an Opposed-Piston Folded-Cranktrain Diesel Engine, 6365–6381. https://doi.org/10.3390/en8076365