Combustion Characteristics of the Direct Injection Hydrogen Enriched Compressed Natural Gas Engine at Different Relative Air-Fuel Ratios

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Abstract:- The implementation of Compressed Natural Gas (CNG) in internal combustion engine (ICE) has many positive benefits such as to improve performance and reduce emission characteristics of the internal combustion engine. However, the engine combustion characteristics are badly affected (i.e. slow burning velocity and poor lean burn capability) as a result of CNG being used in ICE. To this end, research in finding a suitable additives to combine with the CNG becomes necessary. Hydrogen gas (H2) is the best additive to combine with CNG, as it possesses a faster burning and better combustion characteristics. Thus, the primary objective of this research is to conduct experimental investigation on the combustion characteristics of the ICE fuelled with hydrogen enriched compressed natural gas (DI-HCNG) engine at different relative air-fuel ratios λ=1.0, λ=1.2, and λ=1.4 which represent stoichiometric, lean, and very lean combustion respectively. This research was performed experimentally under the following technical operating parameters of the engine: engine speed (at 2000 rpm); percentage of H2 (0%, 28%, and 46% by volume); various air-fuel ratios λ=1.0, λ =1.2, and λ=1.4. The results showed that different percentages of hydrogen gas (H2) directly injected into the CNG engine significantly influenced the maximum in-cylinder pressure, the burning velocity (leading to faster mass fraction burned timings), and the maximum of the heat released rate under all mechanical operating conditions under consideration. Furthermore, the addition of 28% hydrogen gas (H2) to the CNG showed the most stable combustion characteristics relative to 0% and 46%. In light of the above, it can be concluded that direct injection of H2 into the CNG (DI-HCNG) engine would improve the combustion characteristics of the engine (such as faster burning velocity and better lean burn capability).

Keywords*:* *Hydrogen enriched compressed natural gas (HCNG), Mass Fraction Burned (MFB), In-cylinder pressure, Heat Released Rate (HRR), stoichiometric (λ=1.0), Lean (λ=1.2), Very Lean (λ=1.4) etc.*

1. Introduction

Today, the most dominant fuel for the Internal Combustion Engine (ICE) is fossil fuel and the reasons are because of its high energy density (resulting to a higher energy output per unit mass) and its availability. Two of the most well-known ICE – machinery are the Gasoline Engine and the Diesel Engine. These machines use pistons to increase the compression ratio within the engine before ignition takes place; thus, the higher compression ratio of ICE would result to a higher efficiency as shown in the Figure 1 [1, 2].

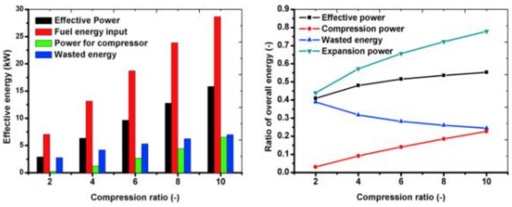


Fig 1: Comparing effective energy to compression ratio

However due to mass pollution, erratic fuel prices and the unsustainable nature of fossil fuel combustion, researchers are investigating for substitute fuel sources. Thus, the research on finding the suitable alternative fuel to be utilized in ICE has received increased attention. As a result, the development of alternative fuels such as biomass, ethanol, or natural gas may potentially provide a solution to the large consumption of fossil fuel.

Due to the abundance, clean burning and low cost of CNG, most analysts believe that it could be used as an alternative fuel. CNG is a gaseous hydrocarbon that is primarily made of methane with other smaller fractions of ethane, propane, and nitrogen, as shown in Table 1. The gas is harvested from underground wells, produced as a by-product from agricultural or human waste and decomposed garbage. According to Poulton, a NG powered engine has the capability of producing the same power, as well as at the same (or higher) level efficiency; and with lower discharge relative to the petrol-fuelled vehicle. On the contrary, this vehicle would have a shorter mileage unless the tanks are bigger, since CNG has a lower energy density than fossil fuel. Nonetheless, the CNG faces have two major disadvantages, which are its low burning velocity and poor lean burning capacity [3 - 5].

Table 1: Composition of CNG

Component	Symbol	Volumetric (%)
Methane	CH_4	94.42
Ethane	C_2H_6	2.29
Propane	C_3H_8	0.03
Butane	C_4H_{10}	0.25
Carbon Dioxide	CO_2	0.57
Nitrogen	N_2	0.44
Others	H_2O+	2.00

To overcome this problem, H_2 is injected into the CNG mixture. In addition, recent discoveries show that H_2 has attracted attention as a plausible source of energy for transport vehicles. It is known that H_2 possesses a wide flammability range in comparison to other fuel; this means that it works in the ICE with the CNG mixture. Simultaneously, it has a low ignition energy, which proves as an advantage, enabling it to run on a lean mixture while ensuring ignition. Often, when the ICE runs on a lean mixture, it produces a mileage and a more complete combustion. Since injecting hydrogen into the mixture only produces water vapour, it eliminates the harmful CO_2.

Injecting H_2 also allows high-speed engine operations allowing power increase and efficiency. Thus, it proves to be a possible solution to overcoming the drawbacks of the CNG (the low burning velocity and the poor lean capacity).

A lot of researches have been conducted to improve the combustion characteristics of ICE. Firstly, Elfasakhany A (2018) conducted an experimental investigation on “Exhaust emissions and performance of ternary iso-butanol–bio-methanol–gasoline and n-butanol–bio-ethanol–gasoline fuel blends in spark-ignition engines: Assessment and comparison”. The study concluded that using the two different fuel blends, which are iso-butanol–bio-methanol–gasoline (iBM) and n-butanol–bio-ethanol–gasoline (nBE) would improve the combustion characteristics in ICE [6]. Nieminen, J., & Dincer, I. (2013) investigated the “Comparative energy analyses of gasoline and hydrogen fuelled ICEs". The study concluded that utilizing the hydrogen enrichment strategy would decrease the flame development and propagation period, advances the central heat release and increases the heat release rate proportionally to the amount of hydrogen added to the air fuel mixture [7]. In furtherance Amrouche, F. at el. (2018) conducted "An experimental analysis of hydrogen enrichment on combustion characteristics of a gasoline (Wankel) engine at full load and lean burn regime". It was concluded that hydrogen enrichment strategy was able to decrease flame development, advances the central heat release and increases the heat release rate proportionally to the amount of hydrogen added to the air fuel mixture [8]. Furthermore, Chitragar, P. R., Shivaprasad, K. V. at el. (2016) conducted "An experimental study on combustion and emission analysis of four cylinder 4-stroke gasoline engine using pure hydrogen and LPG at idle condition". The investigation concluded by suggesting hydrogen and LPG as an alternative fuel for the SI engine operation. This is because, with hydrogen as an additive, the cylinder pressure would be increased by 18%, wherein for LPG, increase in cylinder pressure was 4.4%, improving the combustion characteristics of gasoline by having a higher burning rate [9]. Moreover, Kalsi, S. S., & Subramanian, K. A. (2017) conducted an "Experimental investigations of effects of hydrogen blended CNG on performance, combustion and emissions characteristics of a biodiesel fueled reactivity controlled compression ignition engine". The study concluded by stating that the in-cylinder pressure and heat-released rate with HCNG fuel increased when it is compared to the conventional CNG fueled engine. In furtherance, Song, J., & Park, S. (2017) conducted a study regarding "Combustion characteristics of a methane engine with Air- and N_2-assisted direct injection". The study concluded that by stating that IMEP (indicated mean effective pressure) is increased by three factors: strong turbulence and high O_2 concentration resulted to rapid combustion, presence of gas injection would result to a higher in-cylinder pressure, and the increase of combustion efficiency brought about by using the lean combustion strategy [10]. Furthermore Li B et al (2017) conducted an investigation regarding "Combustion and emission characteristics of diesel engine fueled with biodiesel/PODE blends", The research concluded that the biodiesel's combustion characteristics would improve by blending it with Polyoxymethylene Dimethyl Ethers (PODE), thus increasing combustion efficiency at high EGR (exhaust gas recirculation) conditions. At 40% EGR, combustion efficiency of BP15 (85% biodiesel and 15% PODE) is 5% higher than that of D100 and 3% higher than that of B100. Engine efficiency is also improved by blending PODE [11]. In addition, Lui et al (2018) conducted a "Comprehensive study of key operating parameters on combustion characteristics of butanol-gasoline blends in a high speed SI engine". The research concluded that the MFB is the key parameter in order to control the combustion process of engine fueled with butanol-gasoline blends. The PCP and rate of in cylinder pressure increase is obviously influenced by 50% MFB and 10–90% MFB [12]. Moreover, Najafi, G. (2018) conducted a research regarding "Diesel engine combustion characteristics using nano-particles in biodiesel-diesel blends". The study was concluded by stating that the reaction of the combustion is more efficient by mixing nanoparticle to the fuel in order to decrease the in-cylinder pressure, which in turn makes the combustion reaction more efficient as well as increasing the engine power with higher nano content blends [13]. Furthermore, Wang, X., & Ni, P. (2017) investigated on the "Combustion and emission characteristics of diesel engine fueled with diesel-like fuel from waste lubrication oil". The research concluded by stating that mixing diesel fuel with DLF (diesel-like fuel) displayed an increased heat release rate, in-cylinder pressure, and temperature of combustion at all measured points.

2. Combustion Equations

The combustion equations used for the HCNG mixtures under are defined as follows:

λ: Refers to the relative air fuel ratio

λ=1.0: Stoichiometric combustion is shown in equation 1.1

$$(aCH_4+bH_2)+(2a+\frac{b}{2})(O_2+3.76N_2)\rightarrow aCO_2+(2a+b)H_2O+(2a+\frac{b}{2})3.76N_2 \quad (1.1)$$

λ=1.2: Lean combustion is shown in equation 1.2

$$(2aCH_4+bH_2)+1.2(a+b)(O_2+3.76N_2)\rightarrow aCO_2+2(2a+b)H_2O+(2a+b)3.76N_2 \quad (1.2)$$

λ=1.4: Very lean combustion is shown in equation 1.3

$$(4aCH_4 + bH_2) + 1.4(a+b)(O_2 + 3.76N_2) \rightarrow aCO_2 + 2(2a+b)H_2O + (4a+b)3.76N_2 \quad (1.3)$$

3. Experimental Set-up and Procedure

The combustion characteristics of the hydrogen enriched compressed natural gas (DI-HCNG) engine at different relative air-fuel ratios (λ=1.0, λ=1.2, and λ=1.4) was conducted using the single cylinder four-stroke engine located at Universiti Kebangsaan Malaysia (UKM). Figure 2 gives the schematic diagram of the experimental set-up and Table 2 gives the technical operative parameters of the engine used during the experimental operation.

Table 2: Technical Operations of the Engine

Serial Number	Parameters	Ranges
1	Speed	1000 to 5000 rpm
2	Throttle Position	Wide-Open Throttle
3	Percentage of H_2 by volume	0%, 20%, 28%, and 46%
4	Combustion Strategy	Rich, Stoichiometric, and lean Combustion (λ =0.8, λ =1.0, and λ =1.2)

The supply system of the experimental set-up consist of CNG-H_2 mixture, flow meter, pressure regulator, etc. The experiment was conducted by running the engine until the stability of the engine is achieved. A stable engine occur when the oil and coolant is at 60˚C to 70˚C respectively. The experiment was then started by directly injecting the different fractions of mixtures of H-CNG into the cylinder. The fractions of H_2 to be used are 0%, 28% and 46% by volume. The Electrical Control Unit (ECU) was used to control the technical operating parameters of the engine during the experiment. Technical operating parameters such as throttle position (wide-open-throttle), engine speed, and air-fuel ratios (λ=1.0, λ=1.2, and λ= 1.4). The experiment was repeated for the various percentages of H_2 in the CNG mixture (fractions of 0%, 28%, and 46% by volume). The combustion characteristics data (such as in-cylinder pressure, mass fraction burned, and heat release rate) was then obtained with the aid of in-cylinder pressure data acquisition system that comprises of the piezo electric pressure cylinder, charge amplifier, shaft piston encoder, and piezo resistance pressure sensor (transducer).

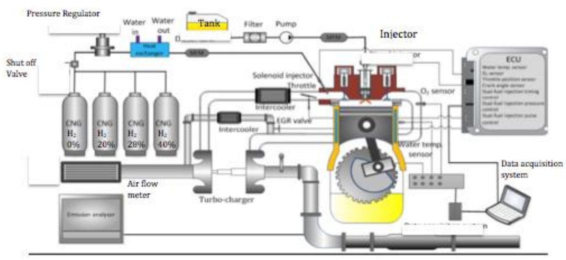


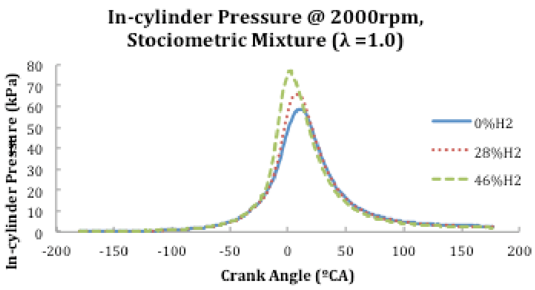
Fig: 2: Experimental Set-up

4. Research and Discussion

This section focused on the results of the experimental investigation of the combustion characteristics of the direct-injection hydrogen enriched compressed natural gas engine. The experiment was performed using different percentages of hydrogen gas (H_2) mixed with the compressed natural gas (CNG) at 0%, 28%, and 46% by volume under various air-fuel ratios which are λ= 1.0, 1.2, and 1.4, which represent stoichiometric, lean, and very lean combustion respectively. The results are presented in figure 4.1-4.3 and are fully explored in the following section, whilst the table of readings is provided in the appendix A-C.

4.1 In-Cylinder Pressure

Figure 3 (a-c) presents the relationship between in-cylinder pressures (kPa) against various crank angels (ºCA), under different hydrogen fractions at three different air-fuel ratios. It is apparent from the graphs, under all air-fuel ratios and at various hydrogen fractions that the graph is bell-shaped. This is due to the three stages of combustion (ignition lag, flame development, and after burning combustion). Under normal operating conditions, combustion is initiated towards the end of the compression stroke at the spark plug by an electric discharge. The ignition lag lasts for 0.00015 to 0.0002 seconds at low in-cylinder pressure. Following the ignition, the turbulent flame develops and propagates through the fuel and air, and the burned gas mixture until it reaches the combustion chamber walls; during this event the in-cylinder pressure is rapidly increasing. Finally, it peaks at maximum pressure due to the attainment of total combustion. Thus, explaining bell-shaped graph. In addition, comparing the in-cylinder pressure at different HCNG mixture (0%, 26%, and 48% H_2 by volume), it is evident that 46% H_2 gives the highest in-cylinder pressure under all air-fuel ratios under consideration. This is backed with theories suggesting that with higher percentage of H_2 in the HCNG it would increase the overall combustion temperature of the fuel; leading to higher in-cylinder pressure. Furthermore, when analyzing the in-cylinder pressure at different air-fuel ratios, stoichiometric combustion (λ = 1.0) gives the highest pressure at approximately 72kPa, followed by lean combustion (λ= 1.2) at approximately 62kPa, and then very lean combustion (λ = 1.4) at approximately 53kPa. The reason being that at λ= 1.0 the combustion process is complete, there is no excess air or fuel, enabling the flame to burn/propagate more efficiently; thus, explaining the decreasing trend of the in-cylinder pressure as the ratio deviates further from 1.0. Moreover, consider the data point of the maximum cylinder pressure of 0% H_2 and 46% H_2 under stoichiometric and very lean combustion respectively [i.e. Figure 4.1 (a) and (c)]. For stoichiometric combustion, the maximum cylinder pressures of the two hydrogen fractions under consideration are approximately 75kPa and 65kPa respectively. This has a peak-to-peak difference of approximately 26% under this mechanical operating condition of the engine. While for very lean combustion, the maximum pressure for the two hydrogen fractions under consideration are approximately 53kPa and 43kPa. This shows a peak-to-peak difference of approximately 10%. Thus, it can be concluded that both percentage of H_2 in the CNG mixture as well as the type of combustion strategy used heavily influenced the in-cylinder pressure of the engine. This is probably due to the chemical characteristics of the HCNG mixture occasioned by the air-fuel ratio. This is aligned with previous experimental results of R. Mehra et al (2017) and S. Wasiu et al (2018) [14,15].



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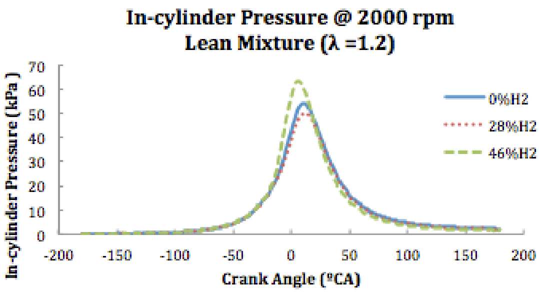

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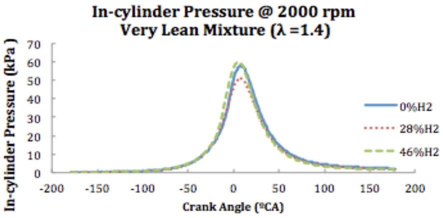
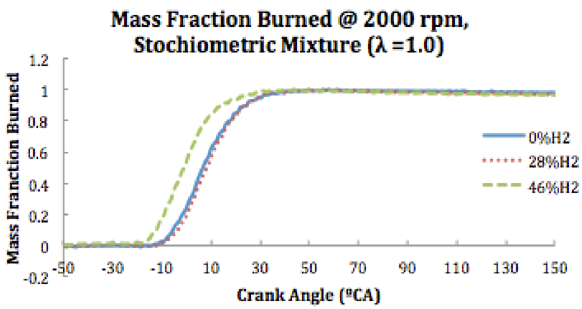


(c)

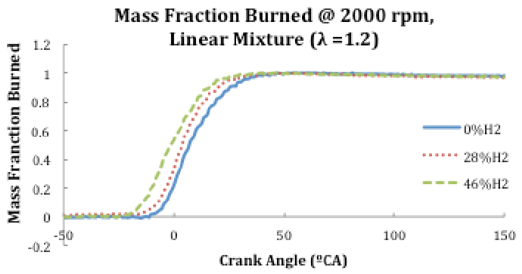
Figure 3. In-Cylinder Pressure (kPa) against Crank Angel (ºCA) at three different air-fuel ratios

4.2 Mass Fraction Burned

Figure 4 (a-c) shows the relationship between mass fractions burned against different crank angels (ºCA) using three different hydrogen fractions at three different air-fuel ratios. The graph is shaped as a ‘S-curve’ and it consists of 3 different phases (a constant phase, followed by a linear increment, ending with a constant phase). This is due to the 3 stages of combustion (flame development, flame propagation, and total combustion duration). Mass fraction burned (MFB) in each individual engine cycle has a standard quantity with a scale of 0 to 1, which describes the process of chemical energy release in respect to the crank angle. The determination of MFB is commonly based on burn rate analysis. In the beginning there is little mass (fraction) burned when the flame initially starts, as shown at the start of the graph. However after the flame is self-sustainable it propagates through the fuel and air; thus, increasing the mass (fraction) burned rate, represented with the linear increase phase of the graph. After the entire mass fraction is burned, the graph “flattens” out at approximately 1.0 (100% mass fraction burned); thus, explaining the S-shape curve). Also when analyzing the MFB at all air-fuel ratios, the fastest mass fraction burned was the HCNG with 46% H_2. This is contributed to the fact that hydrogen gas has a fast burning velocity; as a result, a more hydrogen-enriched CNG would lead to a faster mass fraction burned. Furthermore, examining the MFB at various air-fuel ratios shows that the total combustion duration (the sum of the flame development and the flame propagation duration) for stoichiometric, lean, and very lean combustion (λ=1.0, 1.2, and 1.4), for each is approximately 20ºCA, 10CA, and 8ºCA respectively. This shows that very lean combustion has the fastest total combustion duration while stoichiometric has the slowest. It can be explained by the air-fuel ratio present; as very lean combustion contain more excess air as compared to the others. Combustion will occur faster as the latent heat and ignition temperature of air is lower than fuel; thus allowing for more rapid combustion duration. Moreover, comparing the data-points between 0% and 46% H_2 content for each total combustion duration under stoichiometric and very lean combustion (λ=1.0 and 1.4) [i.e. figure 4.2 (a) and (c)]; under stoichiometric combustion the total combustion duration took approximately 34ºCA and 20ºCA respectively. There is a reduction of approximately 40% in the total combustion duration under this technical operating condition of the engine. While under very lean operation the total combustion duration took approximately 16ºCA and 8ºCA respectively. This indicates approximately 50% total combustion duration reduction between the two respective points in consideration. This shows that both the air-fuel ratio as well as the hydrogen fraction greatly affects the MFB. The results obtained here are coherent with Chang, Y., Sterniak, J., Mendrea, B., & Bohac, S. V. (2016) and S. Wasiu et al. (2018) [15].



(a)



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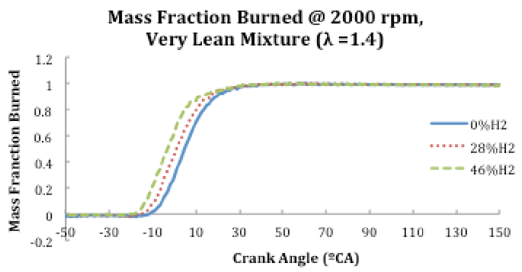
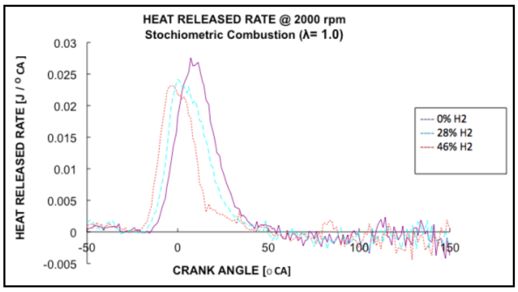


(c)

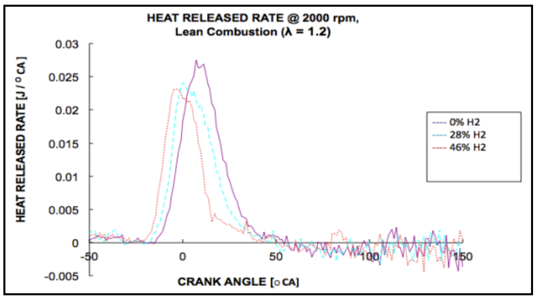
Figure 4.2 Mass Fraction Burned against Crank Angel (ºCA)

4.3 Heat Release Rate

Figure 4.3 (a-c) shows the relationship between the heat-released rates against different crank angels (ºCA) using three different hydrogen percentages (at 0%, 28%, and 46% H_2 by volume) at three different air-fuel ratios (λ=1.0, 1.2, and 1.4). The graph represents the combustion cycles present in the three aforementioned stages of combustion. As the graph starts to increase (around -10ºCA) it shows the ignition lag stage, where the flame starts to develop. Once the flame is sustainable, it starts to propagate through fuel and air increasing the heat released. The graph peaks at after burning stage; however, the dip and rise after the peak indicates that the engine had experienced an abnormal combustion, which could be due to surface ignition or spark knocking. The hear-release rate in figure 4.3 (a) – (c) shows that there are cycle-by-cycle variations in the early stages during the flame development stage of combustion (from zero percent to a few percent of the total heat release). In addition, the major portion of the combustion process (during the rapid-burning phase) which is indicated by the variations in the maximum burning rate. As the mixture becomes leaner with (excess) air, the magnitude of the cycle-by-cycle combustion variations increases. Then, some cycles become sufficiently slow burning that combustion is not completed by the time the exhaust opens. Also when analyzing the different hydrogen fractions under the aforementioned conditions, 0% hydrogen has the highest heat release rate relative to the others. This is due to the cycle-by-cycle variations present during the operation of different composition mixture. Furthermore, under the different air-fuel ratios, stoichiometric combustion presents the highest heat released rate at approximately 0.0275 J/ºCA, followed by lean combustion and very lean combustion with heat released rates of 0.026 J/ºCA and 0.0162 J/ºCA respectively. The reason being that stoichiometric combustion is more efficient relative to the others. Moreover consider the heat release rate data points at 0% and 46% H_2 under stoichiometric and very lean combustion [figure 4.3 (a) and (c)]. At λ= 1.0, the peak heat released rates are approximately 0.0275 and 0.0256 J/ºCA respectively. This indicates a reduction of 7% between the peak-to-peak measurements. While under λ = 1.4, for the mechanical operating condition under consideration the maximum heat release rates are approximately 0.0157 and 0.0146 J/ºCA. This shows a reduction of approximately 7%. This shows that the type of mixture used as well as the combustion operation affects the heat released rate; however, the two factors together does not impact the difference in reduction of the peak-to-peak difference. This experimental result is in line to the previous work of Z. Huang et al (2007) and S. Wasiu (2018) [15-17].



(a)



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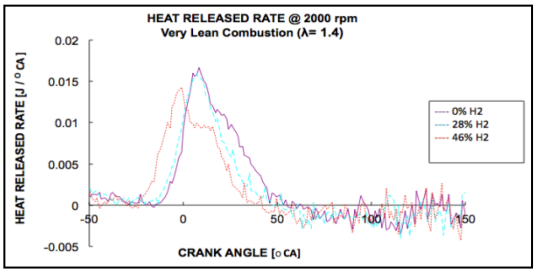

(c)

Figure 4.3 Heat Release Rate against Crank Angel (ºCA)

5. Conclusion

An experimental study has been performed to study the combustion characteristics of the direct injection compression ignition engine under the influence CNG-H_2 Mixtures at full load (WOT) load condition: The main results are summarized below:

(a)

- Under the three different Hydogen fractions (0% H_2, 28% H_2 and 46% H_2), the cylinder pressure gives a bell-shaped. This is largely due to the three (3) stages of combustion such as ignition lag, flame development and after burning combustion. Maximum cylinder pressure peaks at approximately 80 kPa due to the attainment of total combustion duration.
- For the Hydogen fractions under consideration, the mass fraction burned graph gives S-shaped with three different stages such as flame development (0 -10% burned), flame propagation (10 – 90 % burned) and total combustion duration (100% burned). The total mass fraction burned peaks at 1 (100%) due to the attainment of total combustion duration (100% burned).
- For different fractions of Hydrogen under consideration, Heat release rate was affected by the turbulence condition in the combustion chamber. For stoichiometric mixtures, highest turbulence in the combustion chamber varied with the hydrogen with highest fractions (46% H_2) which produces more heat release rate than the other fractions (0% H_2 and 28 % H_2)

(b) From the graphs, it is obvious that 46% Hygrogen gas gives the best combustion characteristics because the Maximum Cylinder Pressure, Mass fraction burned and Heat Released Rate peak at that point in comparison to other hydrogen fractions.

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