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Performance and Emission Characteristics of the Direct Injection Spark Ignition Engine Fuelled by Compressed Natural Gas

Saheed Wasiu*, Shaharin Sulaiman, Rashid Abdul Aziz and Ilyas Zarin.

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

This project aims to investigate the engine performance and exhaust emission of characteristics spark ignition engine fuelled by compressed natural gas. Conventional fuel used in internal combustion engines (ICE) are diesel and gasoline and the price of this fuel is not quite stable (increasing day by day) couple with environmental pollution (from emission) created or generated and the dispensation (non-renewable). Compressed Natural Gas (CNG) the most promising alternative fuel to be utilized in internal combustion engines (ICE) because of its lower price, cleaner burning characteristic and larger reservoir (renewable). Thus, the primary objective of this study is to experimentally investigate the performance and emission characteristics of the direct injection compressed natural gas engine at various air fuel ratios (excess air ratios). The experiment was conducted at various engine speed, wide open throttle (WOT) and various air fuel ratios such as (λ = 0.9, λ = 1.0, λ = 1.1) which represent rich, stoichiometric and slightly lean mixture respectively. The results showed that brake torque increases as the engine speed increases for all types of air fuel ratios due to turbulence that occurs within the engine cylinder while BSFC decreases at the early engine speed due to reduction in mixture concentration within the engine cylinder while for emission characteristics, the result revealed that increase in engine speed shows no significant effect on BSCO emissions for all the mixtures under consideration. BSNO emissions increases as the engine speed increases. BSUHC emissions decreases when the engine speed increases (lower speed) due to lower expansion and exhaust temperature. But at high engine speed, BSUHC increases due to slow combustion; partial burn and misfire. Thus, it can be concluded that, by utilizing various air fuel ratios in direct injection compressed natural gas engine is one of the favorable technique to improve the performance characteristics of the DI-CNG engine such as brake torque and BSFC most especially at a low engine speed (i.e. between 2000 rpm to 3000 rpm).

Keywords: Direct Injection, Compressed Natural Gas, Air-fuel ratios, Wide Open Throttle, Performance and Emission characteristics.

INTRODUCTION

Liquid fuel is one of the most energy sources for internal combustion engines. However, the usage of this energy sources has been rising with the amount of population growing in this world. In today's increasing and competitive sector of internal combustion engines, it has become necessity to upgrade or modify the engine functioning and to provide efficient and economical engines. Since the design of internal combustion engines, the regular fuel that have dominated are gasoline and diesel [1- 4]. The price of this fuel is not quite stable (increasing day by day) couple environmental pollution (from emission) created or generated and the dispensation (non-renewable), therefore investigating alternative fuel becomes imperative and important for the engine researchers. Compressed Natural Gas (CNG) is the most promising alternative fuel to be utilized in internal combustion engines (ICE) because of is lower price, cleaner burning characteristic and larger reservoir (renewable). Natural gas (NG) is created primarily of methane (CH_4) but frequently contains traceable amount of ethane, propane, nitrogen, helium, carbon dioxide, hydrogen sulfide, and water vapor. Methane is the standard component of natural gas. Usually more than 90 percent of natural gas is methane in natural gas composition [5-7].

There are many published literatures regarding the implementation of CNG in internal combustion engine. There are a lot researches on CNG as an alternative fuel in the ICE, and this aspect plays the major role in this research work as it presents recent studies done on the CNG as vehicular fuel. M.I. Jahirul et al. (2010) [8] carried out a comparative engine performance and emission analysis of CNG and gasoline in a retrofitted car engine and came out with the following conclusions; the CNG produces lower brake power than the gasoline throughout the speed range. Retrofitted car engine runs on lower BSFC when utilizing CNG than on gasoline. The CNG has preferences of higher brake thermal efficiency on an average of 1.1% and 1.6% than that of gasoline. The engine exhaust gas temperature created by the CNG burning is always higher as compared with that of the gasoline. CNG fueled retrofitted car engine produced lower HC, CO, O_2 emission throughout the speed range than gasoline. Higher NOx emission is the main emission concern for CNG as car fuel. 41% and 38% higher NOx emissions have been recorded at 50% and 80% throttle position respectively contrasted to that of gasoline. Such a huge emission range should be a major environmental concern as CNG retrofitted automotives are currently mass created and utilized. Aljamali et- al, (2014) [9] studied the comparison of performance and emission of a gasoline engine fuelled by gasoline and CNG under different throttle positions from running 1596 cm^3, 4-cylinder spark-ignited gasoline port injection. Results showed that the gasoline produces more power, torque and BMEP than CNG yet BSFC of gasoline is not as much as CNG. Moreover it shows to a smaller extent CO_2 and CO in CNG as compared to gasoline. NOx emission for CNG is less than gasoline. E. Ramjee et al. (2011) [10], worked on experimental investigations of a single cylinder 4-stroke air cooled type Bajaj-Kawasaki petrol and CNG engine to compute performance and exhaust emissions of the test engine. All tests have been carried out under steady state conditions for both petrol and CNG fuels and the results have been compared. They found that for all range of speeds, the volumetric efficiency is reduced and varied between 10-14%; except thermal efficiency, the other performance parameters viz BMEP, Torque, Power and BSFC are decreased for CNG fuelled engine compared to petrol fuelled engine; in addition, with the exception of NOx, the other emission characteristics such as CO, CO_2, and HC are decreased. Munde Gopal et al. (2012) [11] carried out a review on compressed natural gas as an alternative fuel for spark ignition engine and came out with the following remarks, the engine thermal efficiency and exhaust gas temperature produced by the CNG burning is always higher as compared with that of the petrol. CNG produces less than 8-16% of effective brake torque, brake power and BMEP compared to gasoline fuel due to reduced volumetric efficiency and lower flame speed of CNG. On the average, the reduction of CO, CO_2 and HC emission are 20-98%, 8-20% and 40-87% respectively by CNG. Savanna V.S et-al. (2013) [12], carried out an experimental study on conversion of 4- stroke gasoline ICE into enriched CNG engine to achieve lower emissions and the following remarks were drawn; reduction in CO, CO_2 and HC emissions at 25% throttle opening with CNG as a fuel as against Gasoline were 81.81%, 34.96%, 39% respectively while NOx increasing 27.58%. Similar findings at 50% of throttle opening revealed 54.6%, 29.25% and 41.07%, while NOx increase was 76.97%. At 100% of throttle opening similar finding showed 76.2%, 28.85% and 77.37% respectively while NOx increase was 82.84%. Under the same engine operations and configurations, sequential port injection CNG operations showed 20% reduction in mechanical efficiency was observed at 25% throttle. Till date, data is still incomplete towards the implementation of CNG in a direct injection compressed natural gas engine (DI-CNG) engine at various air-fuel ratios such as (λ = 0.9, λ = 1.0, λ = 1.1) which represent rich, stoichiometric and slightly lean mixture respectively in engine. Therefore this study will investigate the performance and emission characteristics of the direct-injection compressed natural gas engine at various air-fuel ratios and clarify the engine behavior under this operation condition.

EXPERIMENTAL PROCEDURES

The specifications of the test engine are listed in Table 1 below. A four stroke single cylinder research engine was used to investigate the effect of performance and emission characteristics of the direct injection compressed natural gas engine at various air-fuel ratios. The schematic of the experimental set up is given in Figure 1 below.

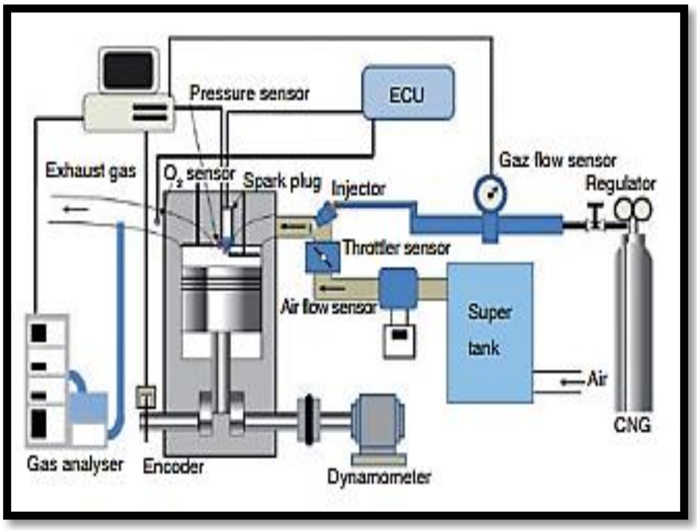

Figure 1: Schematic Diagram of Experimental set up.

Table 1: 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

The supplying system of the engine consists of a natural gas, gas flow meter and pressure regulator. The effects of CNG with direct injection fuel on the engine performance and emission level are to be investigated. Initially the engine is started and no loads are applied for five minutes to allow the engine to reach a steady state operating condition. The experimental works was started by directly injecting CNG into the engine cylinder. An electronic control unit was used to adjust the amount of CNG gas and excess air ratios into the system and also controlled by electronic system is injection timing. The measurements for torque, volume flow rate and exhaust temperature of the engine are recorded when the value reach a steady state condition. The above procedure is repeated for engine performance between 2000 and 4000 rpm so that effective comparisons can be made.

RESULTS AND DISCUSSION

The results from the experiments performed on the single cylinder four-stroke engine for maximum load operating condition are shown below in graphical form and discussed. The several of the air fuel ratios used in this experimental are λ = 0.9 (Rich Combustion), λ = 1.0 (Stoichiometric Combustion) and λ = 1.1(Slightly Lean Combustion). The injection timing utilized for this experimental is 180° CA (Partial Injection Timing). For engine performance, the graphs of results which are engine brake torque, brake power and brake specific fuel consumption (BFSC) against varying engine speed from 2000 rpm to 4000 rpm are plotted. On the other hand, the graph of the emission of brake specific carbon monoxide (BSCO), brake specific unburned hydrocarbon (BSUHC) and brake specific nitrogen oxides (BSNO) against the same range of engine speed which is between 2000 rpm to 4000 rpm are equally shown.

Performance Characteristics

Brake Torque

Figure 2 shows the measured brake torque against various engine speed at different air fuel ratios which is at λ = 0.9 (Rich Combustion), λ = 1.0 (Stoichiometric Combustion) and λ = 1.1(Slightly Lean Combustion) operating condition. As the engine speed increases, the brake torque also increases for all type of air fuel ratio increases. This is because increasing of the engine speed created the turbulence within the engine cylinder; consequent upon, this the brake torque increases. Comparing the brake torque at various air fuel ratios, the air fuel ratio λ = 0.9 has the highest amount of brake torque. This is simply due to the fact that at λ = 0.9, which rich mixture that means the excess fuel is present at that mixture. This excess fuel will enhance the combustion process, which eventually leads to increase in brake torque. While the lowest brake torque is occurring at slightly lean mixture (λ = 1.1). This might be due to reduction in burning velocity within the engine cylinder. In addition, the brake torque increases for engine speeds ranging from 2000 rpm to 3000 rpm to a peak before dropping in magnitude for higher speeds up to 4000 rpm for all the mixtures under consideration (λ = 0.9, λ = 1.0 and λ = 1.1). Consider the brake torque for air fuel ratio λ = 0.9 at 2000 rpm and 3000 rpm engine speeds respectively. It is revealed that the brake torque for 2000 rpm is at 26.4 Nm while at 3000 rpm is 31.8 Nm. This is showing about 17 % increments in brake torque at this operating condition. While the brake torque for air fuel ratio at λ = 1.1 when the engine is running at 2000 rpm and 3000 rpm are 25 Nm and 28.7 Nm respectively which means there is an increment of about 13 % of brake torque at this operating condition. Thus, it is plausible to say that the higher brake torque is obtained at λ = 0.9 as compare to λ = 1.1. This might be so due to excess fuel which is present and this will enhance the combustion. Good agreement is achieved between this experiment result and [1].

Brake Specific Fuel Consumption (BSFC)

The variations of brake specific fuel consumption (BFSC) with engine speed fuelled by different air fuel ratios are displayed in Figure 3. Based on the graph, the BFSC for air fuel ratio of λ = 0.9, λ = 1.0 decrease at the early engine speed 2000 rpm until 3000 rpm. This is due to the reduction in the concentration of this mixture in the engine cylinder at this operating condition and consequent upon this the brake specific fuel consumption (BSFC) decreases. Moreover, air fuel ratio of λ = 1.0 (Stoichiometric Combustion) recorded the lowest BSFC. This represents a good improvement in fuel consumption at this operating condition. This is might be due to efficient combustion occurring at stoichiometric mixture where all the air provided is effectively utilized for combustion process. In addition, the BFSC graph showed increasing trend at 3000 rpm until 4000 rpm simply because of the deterioration in combustion quality caused by cycle by cycle combustion variation in the engine. In addition, consider the data point at air fuel ratio λ = 0.9 for 2000 rpm and 3000 rpm respectively. It showed that BSFC for λ = 0.9 at 2000 rpm is 311 (g/kW.hr) while at 3000 rpm is 277.8 (g/kW.hr). This showed about 11 % of BSFC reduction. While BFSC for air fuel ratio λ = 1.0, at 2000 rpm and 3000 rpm are 261.4 (g/kW.hr) and 231.6 (g/kW.hr) respectively. This measured about 11 % decrement in BFSC. Thus the same percentage reduction in BSFC is achieved at both operating condition of the engine under consideration. The result obtained here is coherent with [9].

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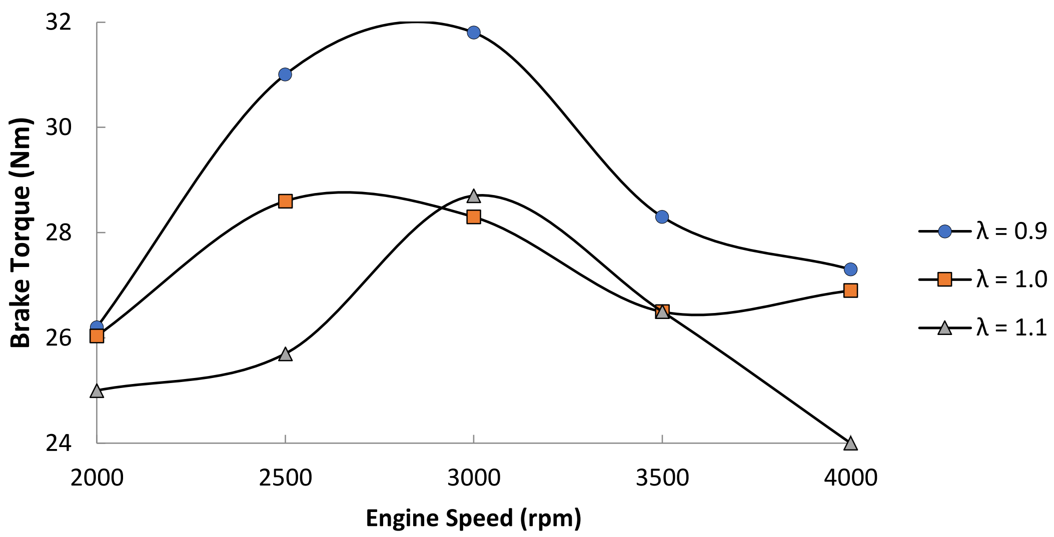

Figure 2: Engine Brake Torque against Engine Speed for Various Air-Fuel Ratios

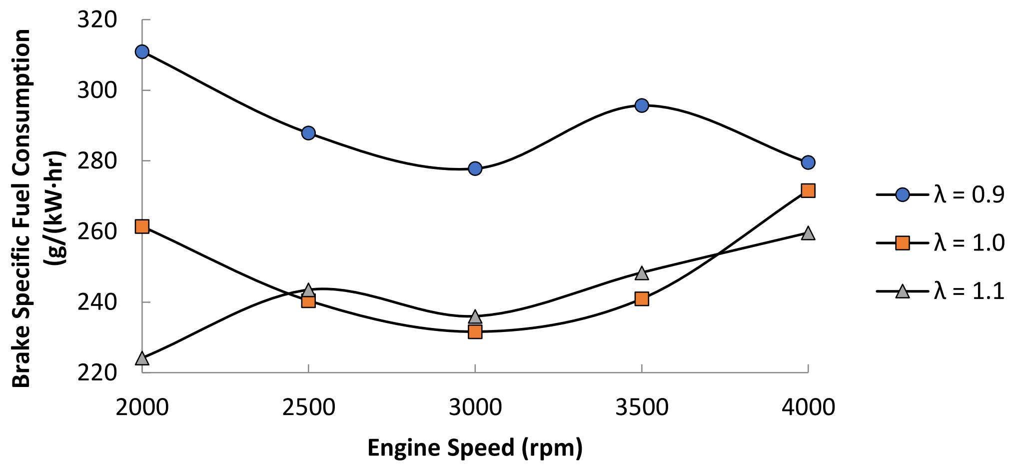

Figure 3: Brake Specific Fuel Consumption against Engine Speed for Various Air-Fuel Ratios

Emission Characteristics

Brake Specific Nitric of Oxide (BSNO)

Figure 4 shows the variation of Brake Specific Nitric Oxide emission (BSNO) at various air fuel ratios and different engine speeds. It is obvious from the graph, when engine speed increases, at early engine speed (from 2000 rpm to 3000 rpm), the nitric oxide emission also increases for all type of air fuel ratios under consideration. The nitric oxide emission is at lowest point before it increases moderately when engine speed is increased from 2000 rpm until 3000 rpm. This is so because, increase in engine speed, increase the turbulence within the engine cylinder and this also increases combustion temperatures. This formation of BSNO emission within the engine cylinder strongly depends on the combustion temperature; hence, the reason for increase in BSNO emission. In addition, the emission reduces from 3000 rpm until 4000 rpm because of reduction in oxygen concentration inside the combustion engine cylinder and this decreases the combustion temperature and thus BSNO reduces. Moreover, highest nitric of oxide emission trend occurs at excess air fuel ratio of λ = 1.1(Slightly Lean). This is largely due to the fact that oxygen concentration at slightly lean mixtures results in higher BSNO emission concentration as compare with the other mixtures. Further, increases in excess air ratio will remarkably decreases the cylinder gas temperature and this decrease the BSNO emission concentration. The lowest BSNO emission, occurs at λ = 0.9 due to excessive deterioration of the combustion quality engine. The result obtain here is coherent with [5].

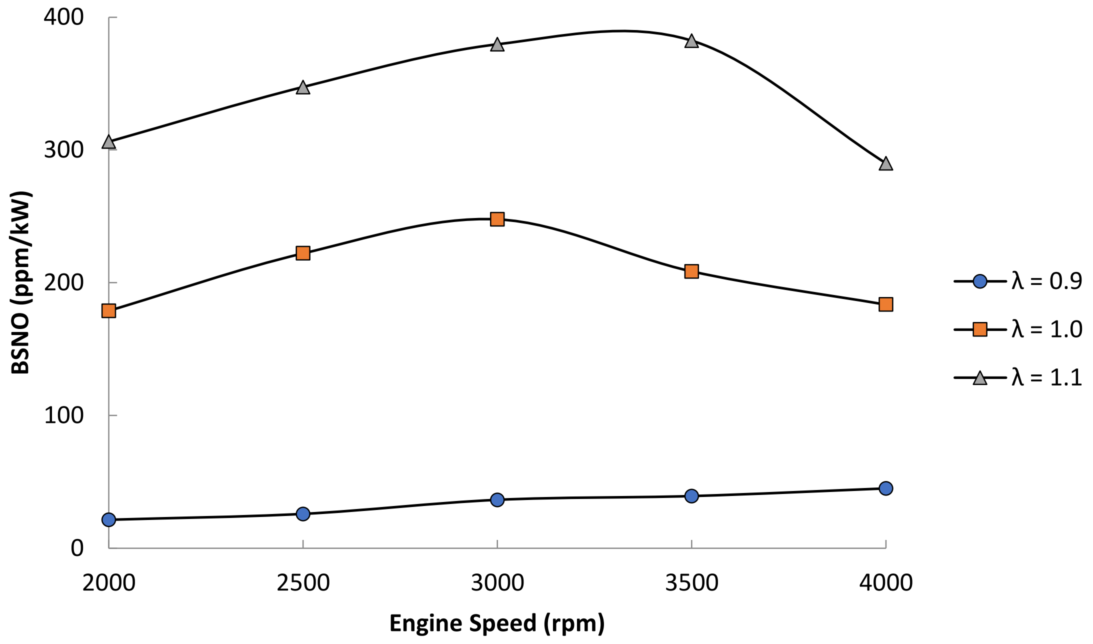

Figure 4: BSNO against Engine Speed at Various Air-Fuel Ratios.

Brake Specific Carbon Monoxide (BSCO)

Figure 5 shows the relationship between BSCO emissions against engine speed at various air fuel ratios. The trend of the graph shows the approximately linearly constant graph. Based on the trend of the graph obtained, there is no significant effect on BSCO emission for all the mixtures under consideration λ = 0.9 (Rich Combustion), λ = 1.0 (Stoichiometric Combustion) and λ = 1.1(Slightly Lean Combustion). The graph on BSCO emission, shows that the highest BSCO occurs at rich mixtures (λ = 0.9) compared with stoichiometric mixtures (λ = 1.0). The reason being that, complete oxidation of the fuel carbon to carbon dioxide is not possible due to insufficient oxygen (incomplete combustion). For lean mixtures, CO levels are approximately constant at a low level of about 0.5 percent or less. Good agreement achieved with this experiment result and [1].

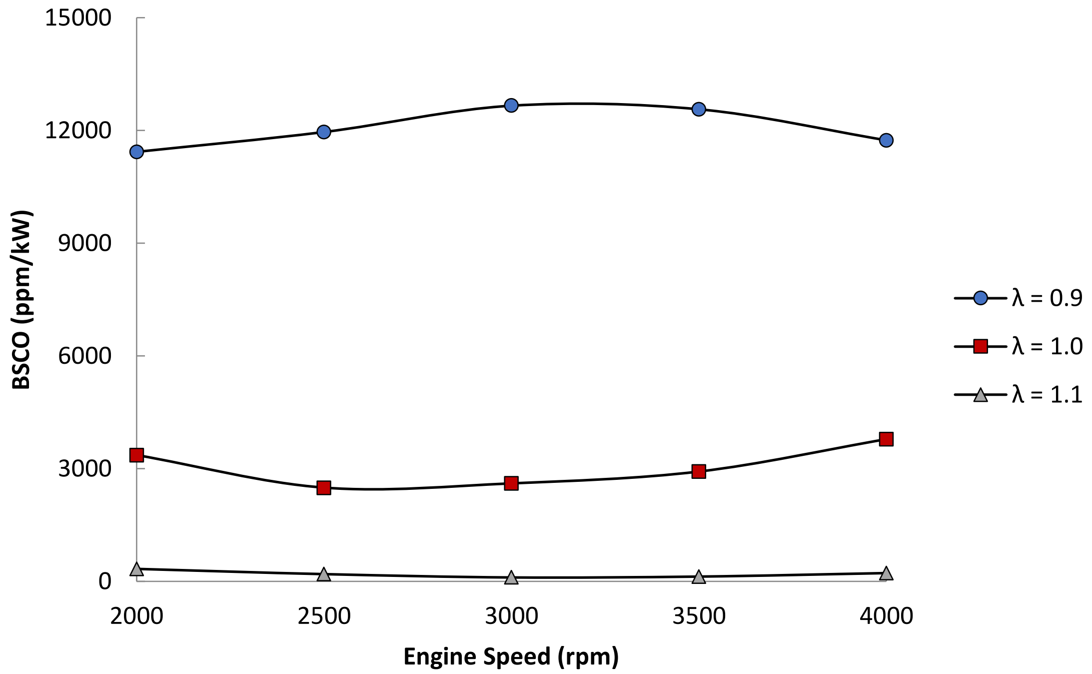

Figure 5: BSCO against Engine Speed at Various Air Fuel Ratios.

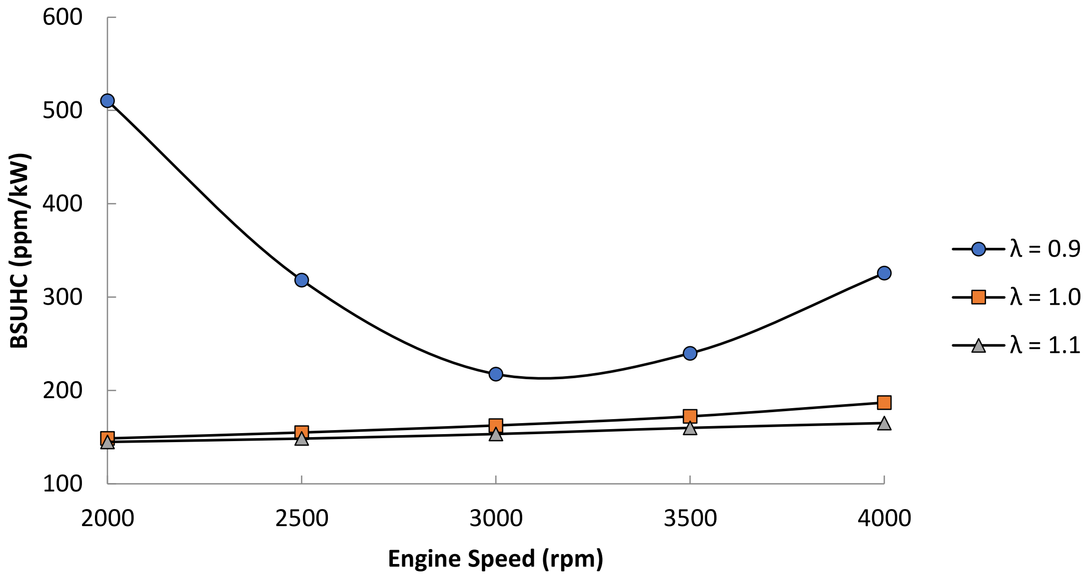

Figure 6: BSUHC against Engine Speed at Various Air Fuel Ratios

Brake Specific Unburned Hydrocarbon (BSUHC)

Figure 6 shows the relationship between brake specific unburned hydrocarbons against engine speed with various air fuel ratios. For a rich mixture λ = 0.9, the trend of the graph showed decrement in the BSUHC emission when the engine speed is increases from 2000 rpm to 3000 rpm. This is largely due to the lower expansion and exhaust stokes temperatures. As the speed increases from 3000 rpm upward, the BSUHC increases. This might be so because of slow combustion, partial burning and even misfire in turn occurs with increasing frequency. In addition, the highest BSUHC occurs at rich mixture. This is because of lack of oxygen for after burning of any unburnt hydrocarbon that escape the primary combustion process within the engine cylinder and exhaust system. For the other mixture (i.e. Stoichiometric and slightly lean mixture), BSUHC emission varies little with excess air ratio due to lower combustion efficiency. The results obtained here is coherent with [8].

CONCLUSIONS

An experimental study has been performed on the direct injection spark ignition engine fuelled by compressed natural gas under various operating condition. The main results contributed to enhance knowledge for performance and emissions characteristic of the engine are summarized below:

- The brake torque increases as the engine speed increases for all types of air-fuel ratios due to turbulence that occurs within the engine cylinder. Exceptions to this occur due to reduction in burning velocity. While BSFC decreases at the early engine speed due to the reduction in mixture concentration within the engine cylinder, while at high engine speed, BSFC increases due to the cycle by cycle combustion variation occasioned by deterioration in combustion quality.
- For BSCO emissions; increase in engine speed shows no significant effect on BSCO emissions for all the mixtures under consideration. While BSNO emissions increases as the engine speed increases. This is so because the formation of BSNO emission depends on the combustion temperature which is enhanced by turbulence occasioned by increased in engine speed. While BSUHC emissions decrease when the engine speed increases (lower speed) due to lower expansion and exhaust temperature. But at high engine speed, BSUHC increases due to occurrence of slow combustion, partial burn and misfire with increasing frequency.

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