Comparative Analysis of Performance and Emission of a Homogenous Combustion Compressed Natural-Gas Direct Injection Engine

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Abstract-- **This study presents experimental results of a homogenous combustion compressed natural-gas direct injection (CNGDI) engine with 1596 cm^^{3} capacity, 4-cylinder spark-ignited with a compression ratio of 14. These results were compared with previous simulation results using Star CD, single cylinder CNGDI research experiment (CNGDI- SCRE (4x)) and port injection gasoline engine (CNG-BI, Gasoline-PI) experiment. The engine test was carried out at engine speed ranging from 1500 (RPM) to 4000 (RPM) with 500 (RPM) increments. The engine control unit (ECU) was using Motec 800. Modify the end of injection (EOI) to 120 BTDC. Performance and emission were recorded under wide-open throttle using an engine control system and a portable exhaust gas analyzer. Results show that power increases from 13.18 (KW) to 53 (KW). By comparing the power in CNGDI with the (Gasoline-PI, CNG-BI, simulation), it is increased (18%, 57%, 16%), respectively. The torque also increases from 83.9 (Nm) to 126.8 (Nm) in CNGDI, with high same percentage compared to the (Gasoline-PI, CNG-BI, simulation), respectively. For CNGDI, the brake specific fuel consumption (BSFC), the brake mean effective pressure (BMEP) is recorded. This study also shows the fuel conversion efficiency (η_f), volumetric efficiency (η_v) and lambda (λ). Furthermore, CO, CO_2 and O_2 are recorded. The study shows low CO emission in CNGDI in all speeds, but CO_2 shows low emission in low speed and high emission in high speed compared to (Gasoline-PI, CNG-BI).**

Index Term-- **Compressed natural gas, performance, emission, direct injection, homogenous piston.**

INTRODUCTION

In recent years, natural gas has been seen as an alternative clean fuel for spark ignition (SI) engines because of its relatively higher octane number. Lean burning of natural gas, which contains mostly methane, in SI engines has been potential to improve thermal efficiency and reduce emissions compared to that of gasoline. Due to its high research octane number (RON) which is higher than 120, natural gas allows combustion at the higher compression ratio without knocking. It also offers much lower CO_2 gas emissions compared with other hydrocarbon fuels as a result of its higher hydrogen to a carbon ratio. With increasing market demands in automotive industries, it is essential to design a new engine in shorter time. In some countries like Iran and Argentina, CNG is already the preferred fuel for the transportation sector, with more vehicles being converted to CNG from original manufacturers. In the 3rd world country, the conversion is mostly for an economic reason due to the increase in the price of gasoline and diesel fuels, while in the advanced world country; the conversion is principally motivated by the emission regulation and environmental concern (Pulkrabek 1997).

CNG is attractive for many reasons. It is the only fuel cheaper than gasoline or diesel. It has inherently minor air pollution emissions. It has lower greenhouse-gas emissions, and there are large quantities of the fuel available in the world (Semin and Bakar 2008).

CNG vehicles can be classified in three main groups: Bi-Fuel that means the CNG fuel is used in gasoline engine, Dual-Fuel which CNG fuel is used in diesel engine and engine innovation as CNG engine using (port injection or direct injection). Previous work investigating natural gas as a fuel in reciprocating piston engines, both spark-ignited (SI) and compression-ignited (CI) was reported (Korakianitis, T. Namasivayam et al. 2010). Gaseous fuels are more appropriate for higher compression engines since they resist knock more than conventional liquid fuels (due to high-octane value that permits a high compression ratio, leading to higher thermal efficiency at full-load condition) as well as produce less polluting to exhaust gases (Kirti Bhandari, Akhil Bansal et al. 2005).

Rising concern with exhaust emissions from internal-combustion engines has resulted in the implementation of strict emission regulations in many industrial areas such as the United States and Europe (Haeng Muk Cho and He 2007). In the meantime, as part of the Kyoto Protocol, many developed countries have agreed to legally binding limitations/reductions in their emissions of greenhouse gases in two commitments periods. The first commitment period applies to emissions between 2008-2012, and the second commitment period applies to emissions between 2013-2020. On the other side, the EURO 5 emission regulation that came into force in September 2009 requires both gasoline and diesel engines to reduce their emissions of nitrogen oxides (NOx) by about 30% (Commission. 2005). Furthermore, the EURO 6 emission regulation which will come into force in September 2014 focuses on reducing the emissions of NOx from diesel cars and vans in order to support efforts to achieve European air quality objectives (Commission. 2006).

In an investigation, using natural-gas direct injection under various fuel injections starting (Zeng, Huang et al. 2006) found that fuel injection timing had a large influence over the

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engine performance, combustion and emissions and these influences became largely in the case of late injection. Over-late injection would supply insufficient time for the fuel–air mixing of the late part of the injected fuel, bringing poor quality of mixture formation and subsequently resulting in the slow combustion rate, the long combustion duration and high HC concentration. However, early injection gave a slight influence on both engine combustion and emissions. On another investigation, using natural gas blended with hydrogen with the effect of ignition timings (Huang, Wang et al. 2007); reported that the time intervals between the end of fuel injection and ignition timing are very sensitive to direct-injection gas engine combustion. Brake mean effective pressure and effective thermal efficiency increase with decreasing the time intervals from the ending of fuel injection to the ignition start while combustion durations decrease with decreasing the time intervals from the ending of fuel injection and ignition start.

For experiment using single cylinder different piston crown (homogenous, stratified), the results show the start of injection timing between 120° to 180° BTDC produce constant power and torque (Mohd.Khair Hassan, Ishak Aris et al. 2009).

For homogenous charge combustion, the injection timing is set at early in compression stroke to ensure better fuel/air mixing. For stratified charge combustion, natural gas is injected at the late of compression stroke with suitable injection timing to enable the engine to have stable operation at a very lean overall mixture (Yusoff Ali, Zailani Muhammad et al. 2005).

(Hassan, Kalam et al. 2009) tested the new design of CNGDI, and results show that slightly more power at 6000 RPM was achieved, less NO_x but high (HC, CO) compared with the origin engine fueled with gasoline. Therefore, a catalytic converter was manufactured to reduce the emission using titanium dioxide (TiO_2) and cobalt oxide (CoO) (Kalam, Masjuki et al. 2009).

On the other hand, the increasing in compression ratio influences high volumetric efficiency, high thermal efficiency and more power output, because more heat is released (Zheng, Wang et al. 2010).

In comparison study between CNG-BI, gasoline-PI and CNGDI, the results show CNGDI produces 23% more brake power than CNG-BI. NO_x reduces to 42% compared with the base engine, but HC and CO emissions are higher than the base engine (Kalam, M. A. Masjuki et al. 2011).

In this research, a purpose-built mono-fuel CNG-DI engine that was design and developed, based on a gasoline-built engine was used in the engine test. The compression ratio of the original engine was increased from 10 to 14 by modification of the cylinder head as well as development of a modern piston with the crown suitable for CNG operations. A high-pressure injector system (20 Bar) was developed and installed on a brand-new cylinder. A central injection system was selected in this design (Azhari Shamsudeen, Shahrir Abdullah et al. 2005), (Yusoff A. and I. 2005).

In our investigation, the results show that at the end of injection (EOI) of 120° BTDC the engine produces the highest power, torque as compared to other EOI settings. The intake valves close at 48° ABDC. Ignition timing changes from 24° to 32° BTDC for full load below 5000 RPM, but it changes from 35.5° to 36° BTDC for 5500 RPM, 6000 RPM, respectively.

The objective of this study is to investigate the performance and emissions of homogenous charge compressed natural-gas direct injection at a setting of end of injection (EOI) of 120 BTDC which is the optimized value. Increasing performance and decrease emission are the aims of the overall project. This investigation is earlier work as a baseline data for the subsequent investigation on stratified charge combustion.

MATERIALS AND METHODS

A 1.6 litre, 7.6 cm bore, 8.8 cm stroke, 4-cylinder spark ignition engine direct injection filled with compressed natural gas (CNG) were tested on an engine test bench. The engine specifications are given in Table 1. CNG was used as fuel. The substantial advantage that CNG has in antiknock quality is related to the higher auto ignition temperature and higher octane number compared to that of gasoline as shown in Table 2. Furthermore, CNG has a high air fuel ratio $(A/F)_s$ and heating value with 17.23 and 47.377 (MJ/kg) respectively. The composition of CNG used in Malaysia is as shown in Table III.

TABLE I
Engine specifications

Parameter	Value	Unit
Number of cylinders	4	-
Type	Inline	-
Capacity	1596	cm3
Bore	76	mm
Stroke	88	mm
Connecting rod length	131	mm
Crank radius	44	mm
Compression ratio	14	-
Intake valve opening	12	bTDC
Intake valve closing	48	aBDC
Exhaust valve opening	45	bBDC
Exhaust valve closing	10	aTDC
Maximum intake valve lift	8.1	mm
Maximum exhaust valve lift	7.5	mm

TABLE II
Combustion related properties of gasoline & CNG

Properties	Gasoline	CNG
Motor octane number	80–90	120
Molar mass (g/Mol)	110	16.04
Carbon weight fraction (mass %)	87	75
(A/F)s	14.7	17.23
Stoichiometric mixture density (kg/m3)	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

TABLE III
Typical composition (Vol. %) of CNG (source: PETRONAS Research & Scientific Services)

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+)	2.0

Figure 1 shows the experimental setup for this investigation. An engine control system and portable exhaust gas analyser were used for controlling engine operations and recording engine performance and emission's data. The KRONOS 4 software is the software used for the test bench. Results were recorded in steady-state condition so ambient pressure, ambient temperature and humidity were noted to estimate air inlet density. Portable exhaust gas analyser Kane-May which is an International Organization of Legal Metrology (OIML) class one certificate was calibrated for each test to get correct results. ECU setting is modified using Motec software.

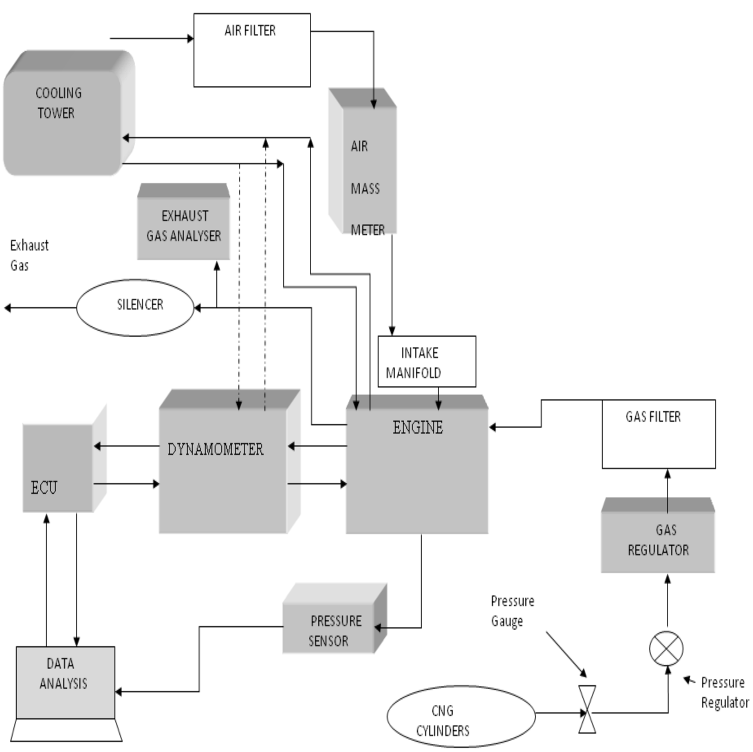

Fig. 1. Experiment setup

All sensors, including temperature's sensors, air mass flow sensor, and pressure cylinder sensor are calibrated with certain data. . The engine was running under full load wide-open throttle. Using water cooling, which consists of cooling tower, differential pressure switch, pipe strainer, filter and actuators (control moving water between dynamometer and engine). Fuel system has the pressure regulator to keep fuel pressure around 20 (Bar) which enter the combustion chamber. Installing the pressure sensor in cylinder one to record pressure cylinder with fast test. Installation of an air mass flow sensor before the throttle valve was carried out to record air mass flow (B.Heywood 1988). Using the dynamometer typed FR250 with maximum load 800 (N.m) and calibration the torque by control levers and weights. The homogenous piston crown used in this study is shown in Figure (2).

Fig. 2. Homogenous piston crown

RESULTS AND DISCUSSION

I. Performance

Performance includes brake power, brake torque, brake mean effective pressure, brake specific fuel consumption, fuel conversion efficiency, lambda and volumetric efficiency. Results were compared in brake power and torque but the others illustrated results for CNGDI empirical.

1. Brake Power

Figure (3) shows the brake power versus engine speed from 1500 (RPM) to 4000 (RPM). The results from the present study were compared with CNGDI experiment, single cylinder (SCRE) and simulation using STAR CD, gasoline port injection (Gasoline-PI) and CNG-BI. As demonstrated in the figure the CNGDI experimental is more 16% than simulation. From the result, the maximum power was 53.14 (KW) at 4000 (RPM) in the CNGDI experiment. The test was done until 4000 (RPM) for avoid engine problem. Comparison the power from CNGDI experimental is more (18%, 57%) than (Gasoline-PI, CNG-BI), respectively (Aljamali, Mahmood et al. 2014). The reasons with high power according to the brake torque are high volumetric efficiency and high thermal efficiency. Additionally, high heat released with the lean mixture produced high indicated power. In another study, the average brake power over the test cycle obtained was 47.39 (KW), 36.90 (KW) and 45.37 (KW) by the Gasoline-PI, CNG-BI and CNG-DI engines respectively (Kalam, M. A. Masjuki et al. 2011).

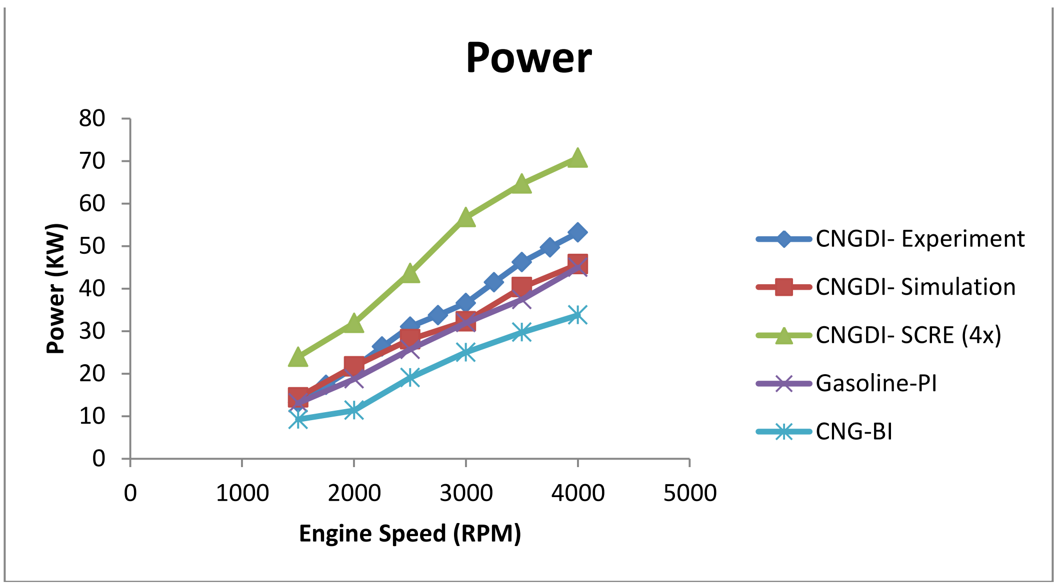

Fig. 3. Brake power versus engine speed

2. Brake Torque

Figure (4) shows the brake torque versus engine speed. The results show that brake torque increases with the engine speed. Maximum brake torque in the CNGDI experiment was recorded at 4000 (RPM) with 126.8 (N.m) which is (+16%, -6%) compared with simulation and SCRE, respectively. At 4000 (RPM), torque was (126.8 (N.m), 109 (N.m), 134 (N.m)) for present CNGDI experiment, simulation and SCRE, respectively. In comparison with previous study the torque from CNGDI is more 57% than CNG-BI (Aljamali, Mahmood et al. 2014). From different investigation, the average brake torque over the engine speed range for Gasoline-PI, CNG-BI and CNG-DI engines obtained are 120.54 (N.m), 92.36 (N.m) and 108.25 (Nm), respectively (Kalam, M. A. Masjuki et al. 2011). The reasons of high torque compared to CNG-BI is due to optimum injection timing which occurs with sufficient time for the fuel to mix and oxidize, resulting in good flame propagation. In (Korakianitis, T. Namasivayam et al. 2010) study, it is reported that when the fuel-injection timing starts early, the combustion chamber pressure and rate of energy release increases.

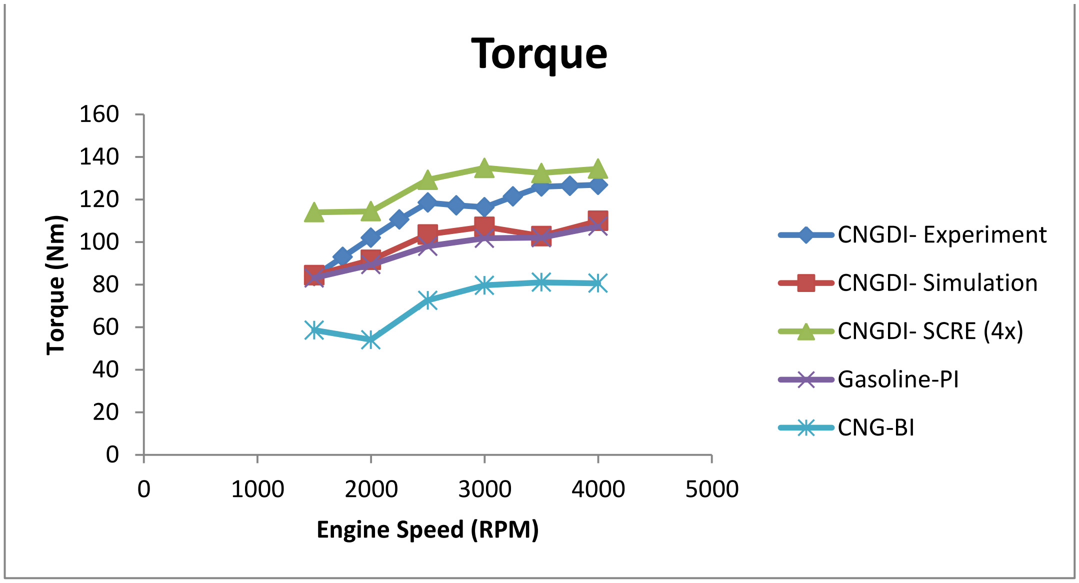

Fig. 4. Brake torque versus engine speed

3. BMEP

Figure (5) illustrates the brake mean effective pressure (BMEP) versus engine speed between 1500 (RPM) to 4000 (RPM). The results shows an increase from 6.6 (Bar) to 10 (Bar). The maximum value was at 4000 (RPM) with 10 (Bar) and this lead to high torque as we saw in the previous Figure (4). The reason of high BMEP is due to high lambda (≥ 1) which lead to lean air fuel ratio (A/F), high heating value 47.377 (MJ/kg) and high spontaneous ignition temperature 645 °C. Additionally, early start of injection (SIO) optimizes the mixture of fuel and air.

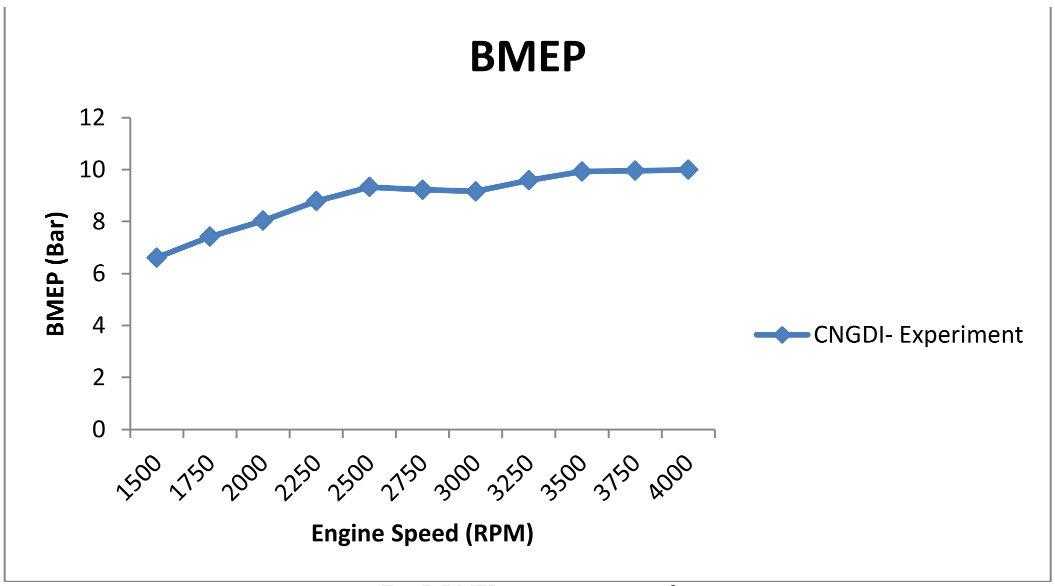

Fig. 5. BMEP versus engine speed

4. BSFC

Figure (6) shows the brake specific fuel consumption (BSFC) versus engine speed. The results show a decrease from 194.8 (g/KW.h) to 88 (g/KW.h). These results are lower than gasoline-PI and CNG-BI due to high A/F ratio, which is more than 17.23 (Kalam, M. A. Masjuki et al. 2011) . Furthermore, the lambda did not exceeds 1.4, and exhaust temperature not more than 550 °C.

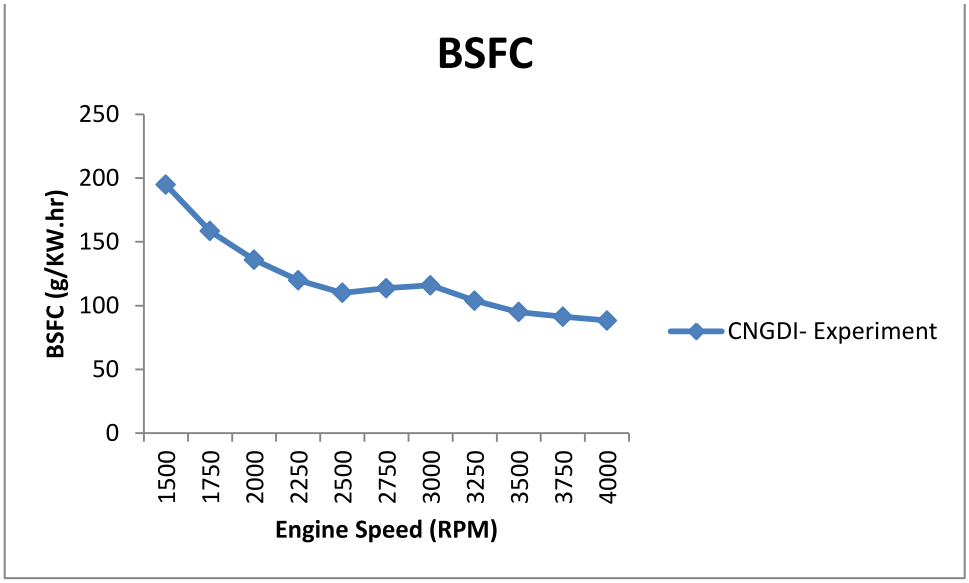

Fig. 6. BSFC versus engine speed

5. Fuel Conversion Efficiency

Figure (7) shows the fuel conversion efficiency (η_f) versus engine speed. The results show an increase from 0.39 to 0.69 as engine speed increases from 1500 (RPM) to 2500 (RPM). The maximum value was recorded at 4000 (RPM) with the value of 0.86.

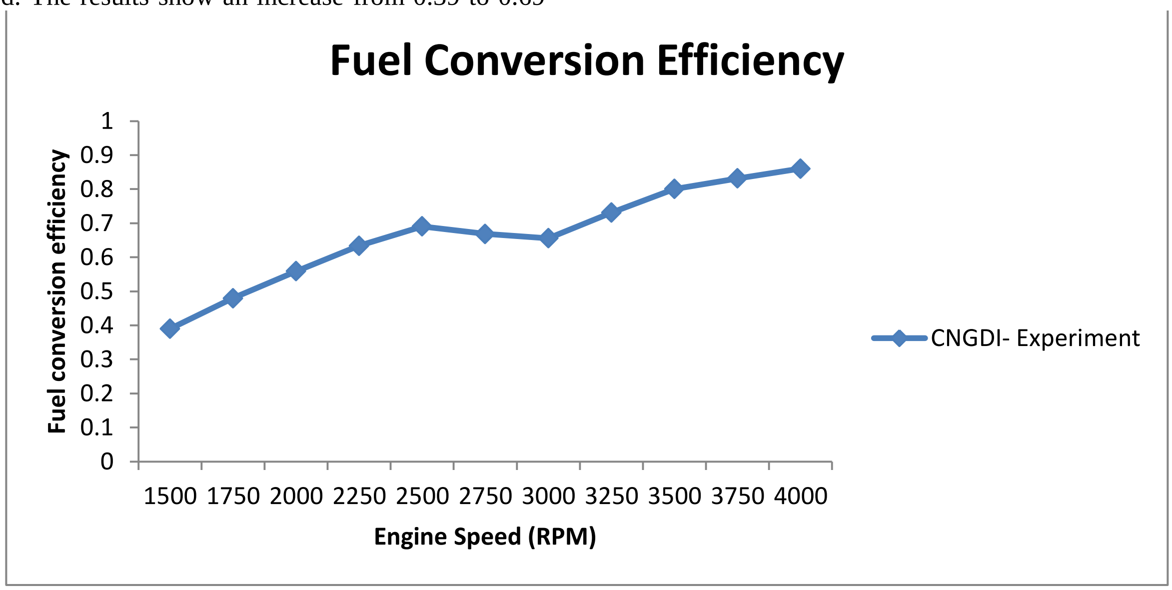

Fig. 7. Fuel conversion efficiency versus engine speed

6. Volumetric Efficiency

Figure (8) shows the volumetric efficiency against engine speed. The results show various irregular trend from 0.35 at 1500 RPM to 0.42 at 4000 RPM with a peak and trough. The volumetric efficiency depends on the quantity of the air inlet to the intake manifold. To increase the volumetric efficiency, it is useful to add a turbocharger.

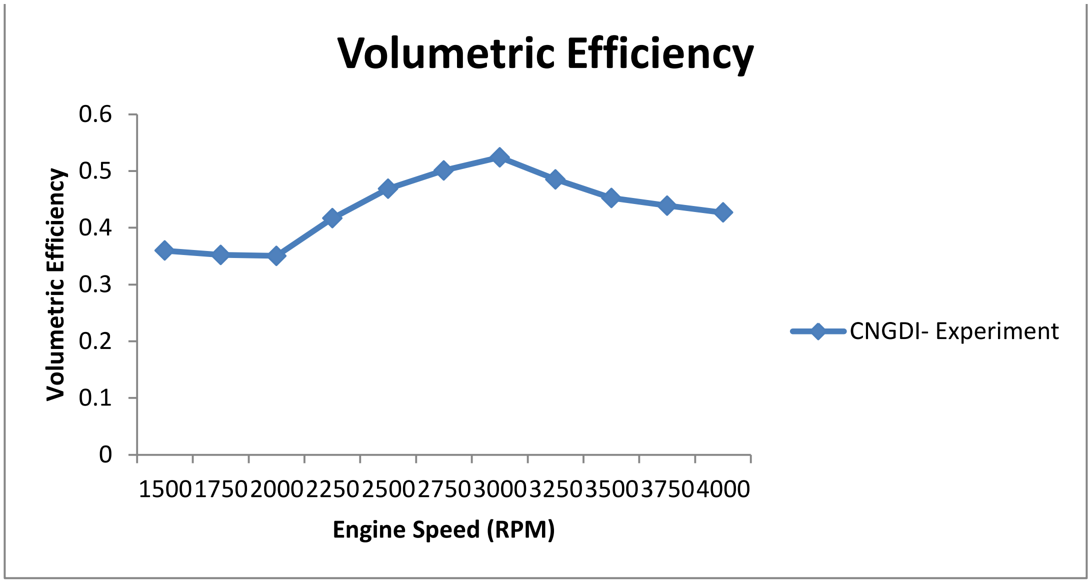

Fig. 8. Volumetric efficiency versus engine speed

7. Lambda

Figure (9) shows the lambda versus engine speed. The results show that the lambda was almost continuously bigger than unity (≥ 1) for the entire speed range which produces lean mixture. However, the lambda settings in the ECU were set between (0.98 to 1.0) for different loads.

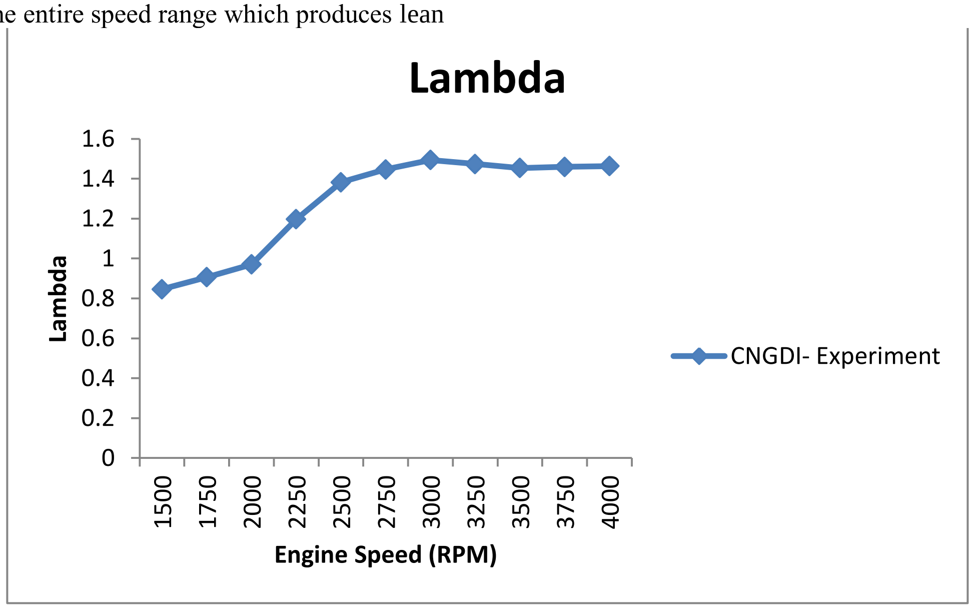

Fig. 9. Lambda versus engine speed

II. EMISSIONS

Figures (10), (11) and (12) show the emissions (CO, CO_2, O_2) versus engine speed. Figure (10) shows the comparison between CNGDI experiment, simulation, CNG-BI and Gasoline-PI. The results shows 0.34 (%) volume to 0.38 (%) volume in the simulation, but in the CNGDI experiment in 4000 (RPM) with value 1.84 (%) volume. From results, CO emission the CNGDI experiment was high than the simulation at high engine speed. Also CNGDI experiment is less than both (CNG-BI, Gaoline-PI) in the engine speed range.

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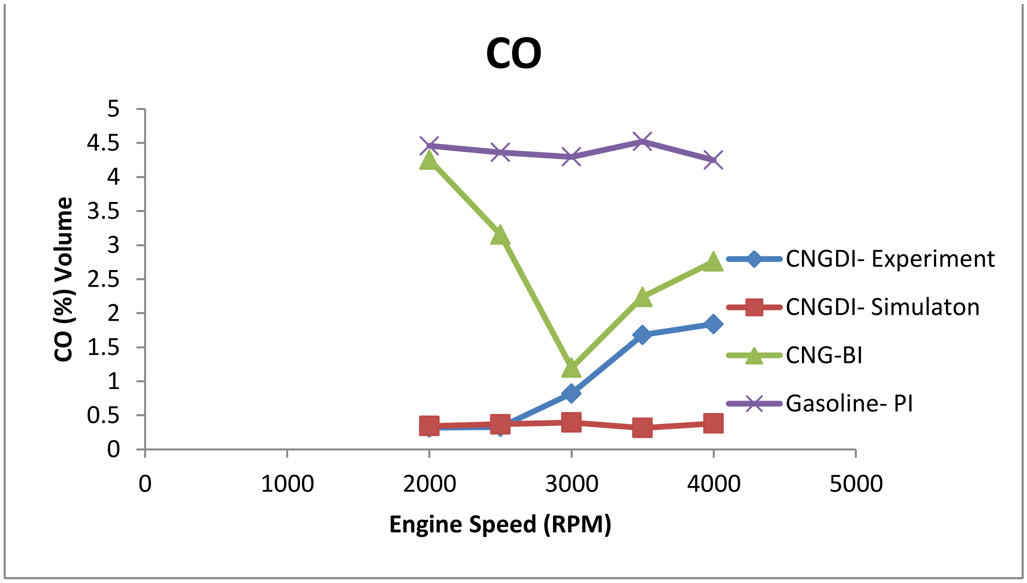

Fig. 10. CO versus engine speed

In Figure (11), the results shows that CO_2 emission is higher 64% in the CNGDI experiment compared to the simulation and higher 21% than Gasoline- PI at high engine speed. Figure (12) shows O_2 versus engine speed. The results show that a high value of O_2 due to high lambda. In another study, the results showed that natural gas composition did have an impact on some emissions components and the performance (Karavalakis, G. Durbin et al. 2012).

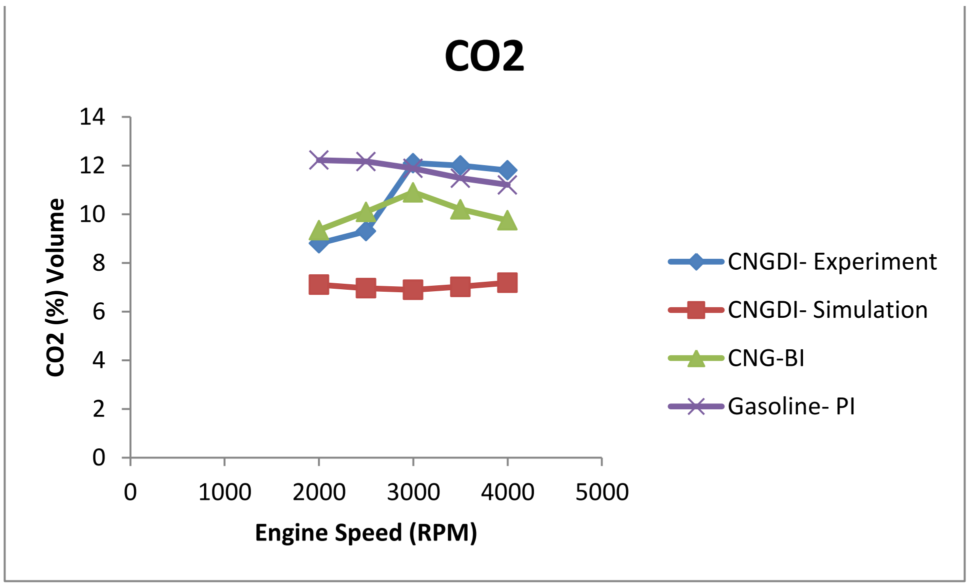

Fig. 11. CO_2 versus engine speed

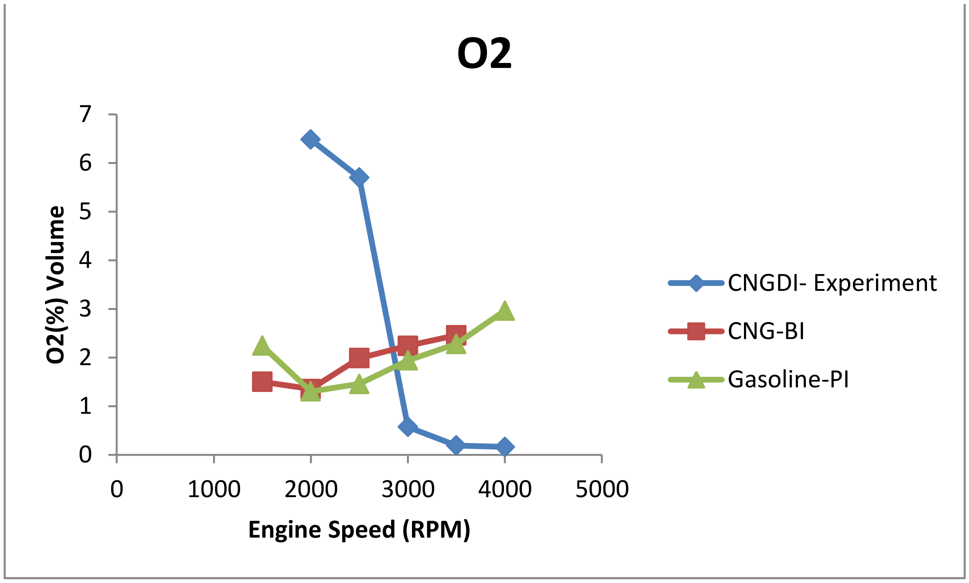

Fig. 12. O_2 versus engine speed

CONCLUSION

CNGDI with homogenous charge is attractive for many reasons. The main reasons are lower fuel economy, lower greenhouse-gas emissions, high-power output and high torque. The following remarks can be drawn from the present investigation:

- The power produce from the CNGDI-Experiment is higher to the CNGDI- Simulation, but it is lower than CNGDI- SCRE (4x).
- The maximum brake torque at 4000 (RPM) for CNGDI experiment, which is (+16%, -6%) compared with CNGDI- Simulation and CNGDI-SCRE (4x).
- In general, BSFC in CNGDI is lower than Gasoline-PI and CNG-BI.
- On average over the speed range, CNGDI experiment produces more power and torque than (Gasoline-PI, CNG-BI).
- CNGDI produces lower CO emission than Gasoline-PI and CNG-BI.
- The CNGDI experiment produces higher CO_2 emission than the (simulation, Gasoline-PI, CNG-BI) at high engine speed.
- Natural-gas homogeneous charge engines require high compression ratios because of the high auto-ignition temperature of natural gas.

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