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Effects of Actual Exhaust Gas Recirculation of the Direct Injection Spark Ignition Engine Fuelled by Compressed Natural Gas at Partial Injection Timing

Saheed Wasiu*, Rashid Abdul Aziz and Khairul Faiz.

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

Abstract

The exhaust gas that is released from vehicles contribute to pollution of the environment and create global warming, acid rain, haze, respiratory and other health problem. Due to this concerns, this research focus on exhaust gas recirculation (EGR) system for spark ignition (SI) engine fuelled by compress natural gas (CNG) at partial injection timing to reduce Nitrogen Oxides (NO_X) emissions which are very harmful to the environment and humanity health. Thus, the primary objectives of this study was to experimentally study the performance and emission characteristics of the SI engine fuelled by CNG under various actual EGR rates at partial injection timing. The experiment was performed using various engine operating conditions such as engine speed which varied from 2000-4000 rpm, part-throttle, stoichiometric mixture and various percentage of EGR. The results of the performance and the emissions level were compared for various EGR rates. The result revealed that as the EGR rates increase the brake torque and brake specific fuel consumption decrease respectively. However, Nitric Oxide emissions (NO) decrease as EGR rates increase. Unlike Unburned Hydrocarbon (UHC) and Carbon Monoxide (CO), the gas emissions decrease as the EGR rates decrease. Thus it can be concluded that by utilize various EGR rates in the DI SI engine, the performance of the engine decrease while the NO_X emission decrease which created a trade-off between performance and emission.

Keywords: Actual Exhaust Gas Recirculation (EGR), Compressed Natural Gas, Spark Ignition, Direct Injection and Partial Injection Timings.

NB: Partial Injection timing: It is the type of injection timing in which the delivery of fuel is neither early nor late. For the purpose of this research, it was taken as 180ºCA.

INTRODUCTION

Vehicle are a major source of air pollutants such as nitric oxides, carbon monoxide, and hydrocarbons as well as the greenhouse gas carbon dioxide and thus there is a growing shift in the transportation industry from the traditional petroleum-based fuels such as gasoline and diesel fuels to the cleaner burning alternative fuels friendlier to the environment such as natural gas, alcohols (ethanol and Methanol), liquefied petroleum gas (LPG) and hydrogen [1-6]. Natural gas is regarded as one the most promising alternatives fuels and probably one of the cleanest fuel in combustion. 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 [7-8]. However, natural gas as an engine fuel suffered from two major setbacks viz: Slow burning velocity and poor lean burn capability which often leads to incomplete combustion, high misfire ratio, and large cycle-by-cycle variation at lean mixture combustion which decreases the engine power output and increases the fuel consumption [7-9]. Traditionally to solve these problem, an increase in flow intensity in cylinder is introduced, unfortunately, this method always increase the combustion temperature and heat loss to the cylinder wall as well as high NO_X emission [10-11]. To solve these problem of high NO_X ; EGR is introduced into the intake system with the aid of regulating valve. EGR is one of the primary strategies implemented by researchers to reduce NO_X emission from the exhaust manifold as EGR addresses emissions within the combustion chamber by lowering the exhaust gas temperatures [12-15]. There were many published literatures regarding the implementation of EGR in internal combustion engine. According to Alain Maiboom, Xavier Tauzia, Jean-Francois, 2008 [16], they studied the effect of exhaust gas recirculation in direct injection compression ignition engine. The experimental results showed that the increase of inlet temperature at constant EGR rate has contrary effects on combustion and emissions, therefore typically giving opposite tendencies, as the reduction of NOx emissions with increased recess temperature. At low-load conditions, very low-NOx and particulate matter (PM) emissions can be obtained with high EGR rates at constant pressure, as a result of the delayed combustion due to the high dilution effect. Harshraj Dangar, Gaurav P. Rathod, 2013 [17] also perform the experimental investigation on the impact of EGR on the diesel engine performance and discovered that increasing inlet air pressure attachment and EGR system provided better result on engine performance. BSFC decreases and brake thermal efficiency increases by increasing inlet air pressure with EGR system. Gomaa M.A.A. 2011 [18] studied the effects of EGR rates on diesel engine and shows that the CO emissions increased with numerous EGR rates. The potential reason may be lower excess oxygen available for combustion. Lower excess oxygen results in rich air fuel mixture at different positions inside the combustion chamber. The NOx emission in combustion chamber is reduced due to higher pressure and higher temperatures during the combustion. Up till now, information is still lacking on the effects of actual EGR on the performance and emission characteristics of the direct injection spark ignition compressed natural gas engine at partial injection injection timing. Thus, the primary objective of this study will focus on the effect of EGR on the performance and emission characteristics of the direct injection spark ignition compressed natural gas engine at partial injection timing and clarify the behavior of the engine under these operating conditions.

Significance Statement

This study discovers the possible effects of actual exhaust gas recirculation (EGR) on the performance and emission characteristics of the direct injection compressed natural gas (DI-CNG) engine at partial injection timings that can be beneficial towards obtaining the good fuel consumption and low engine out emissions. The study will help the engine researchers to understand the possible effects of actual EGR at partial injection timings on the direct injection compressed natural gas engine which has not been fully explored. Thus, an improved understanding on the effects of actual EGR at partial injection timing on the performance and emission characteristics of the DI-CNG engine may be obtained.

EXPERIMENTAL PROCEDURES

The engine used for this experiment is a single cylinder four stroke direct injection spark ignition engine. The engine crankshaft was coupled to an eddy current dynamometer to provide brake torque and it was fitted with the suitable instrumentations for control and measures the operating parameters. The schematic diagram of the experiment setup is shown in Figure 2.1 and its specifications in Table 2.1. The engine used Compressed Natural Gas (CNG) as a fuel during the experiments. The engine was run until it reached steady state, when the fuel temperature is at 60ºC and the coolant is 70ºC. The experimental works started by injecting the CNG into the system while simultaneously recycled the exhaust gas back into the system. The recycle exhaust gas was taken from hole located in the exhaust pipe with the aid of connecting pipe. A regulator was installed in order to regulate the flow of exhaust gas back into the system. An EGR rate in the inlet mixture was increased by increasing the amount of exhaust gas flow back into the engine intake. Two thermocouples placed in the exhaust pipe and in the air intake chamber were used to measure numerous exhaust emission temperatures and air intake temperatures. The engine exhaust manifold was connected to "Gasmet gas Analyzer" to measure the emission characteristics of the engine.

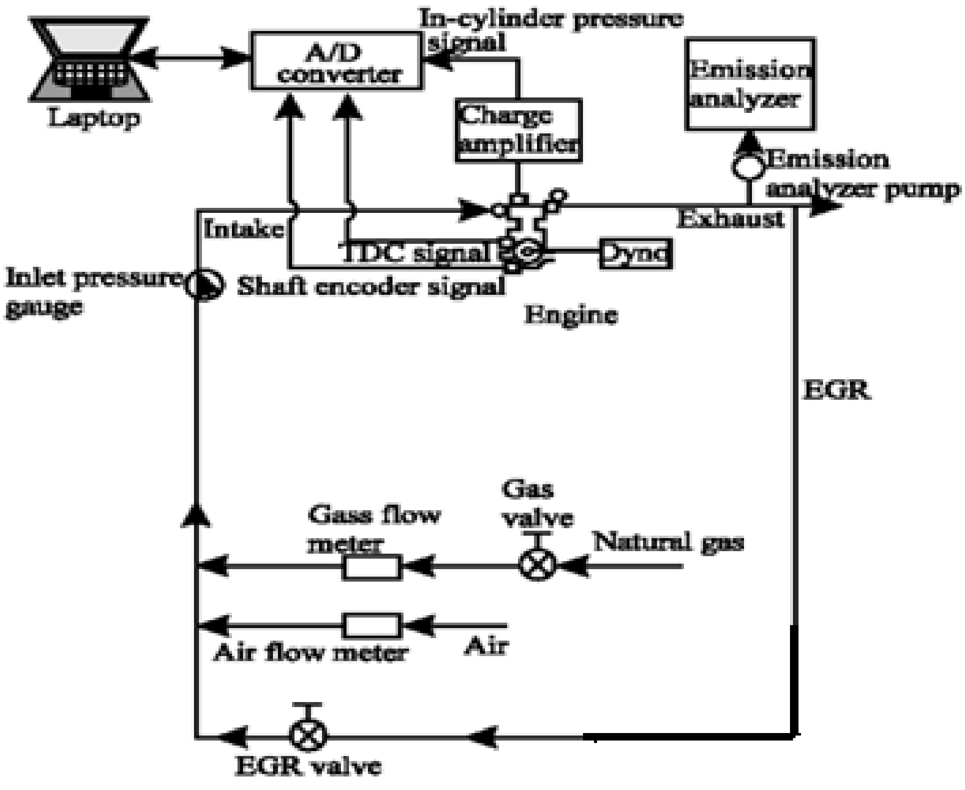

Figure 2.1: Schematic diagram of experimental setup

Table 2.1: Engine specification

Fuel	CNG
Type	One cylinder, 4 stroke, direct injection
Bore and stroke	76mm x 88mm
Compression ratio	19.5:1
Fuel injection timing	14° ±
Fuel injection pressure	19.6Mpa
Intake valve timing	Open at 12° BTDC
	Close at 48° ABDC
Exhaust valve timing	Open at 10° BBDC
	Close at 45° ATDC
Cooling system	Air system
Dynamometer	Eddy current with maximum reading 50 Nm

RESULTS AND DISCUSSION

Performance characteristics

The effects of variable EGR percentage on the CNG-SI engine performance are analyzed and shown graphically in Fig. 3.1 and Fig. 3.2. The engine performance characteristics are presented in terms of brake torque and brake specific fuel consumption to describe the engine output for various EGR conditions. The engine was run at different engine speeds from 2000, 2500, 3000, 3500 and 4000 rpm respectively with different EGR rates increases from 0% until 40% EGR. The research was carried out at part throttle, with injection timing set at 180°CA (represented partial injection timing) and relative air fuel ratio (λ=1.0) which represents stoichiometric mixture.

Brake torque

The Fig. 3.1 showed the relationship between brake torque against engine speed with various EGR rates. At early speed from 2000 rpm to 2500 rpm, the brake torque increase moderately due to turbulent within the cylinder, burning velocity and compensated by advancing spark timing. When the engine speed increase from 2500 until 3500 rpm, the brake torque drastically decrease due to reduction of combustion temperature. At 3500 rpm to 4000 rpm, the brake torque remains constant due to the variation in mixture composition caused by cycle by cycle combustion variation in the engine cylinder. Comparing the brake torque at various EGR rates, it revealed that the highest brake torque is obtained at 2500 rpm for 0% EGR. This was due to increase in turbulence occasioned by EGR rate at that operation condition. At 4000 rpm for 40% EGR rate achieve the lowest brake torque because EGR reduced the concentration of air/fuel mixture in the engine cylinder; consequence upon this, the brake torque reduces. Comparing the brake torque at 0% and 40% EGR and at an engine speed of 2000 rpm and 2500 rpm, it showed that for 0% EGR at 2000 rpm, brake torque is 26.1 Nm while at 2500 rpm brake torque was 30.05 Nm. This shows approximately 29% increment in brake torque. However, for 40% EGR at 2000 rpm. The brake torque respectively was 25.55 Nm and at 2500 rpm the brake torque achieved is 28.6 Nm. This showed about 27% increment. Hence, increase the exhaust gas recirculation proportion will decrease the brake torque of the engine compared to that used pure CNG with 0% EGR. This might largely be due to reduction in burning velocity of the occasioned by EGR. Good agreement was achieved between these experimental results and Hussain, J. [19].

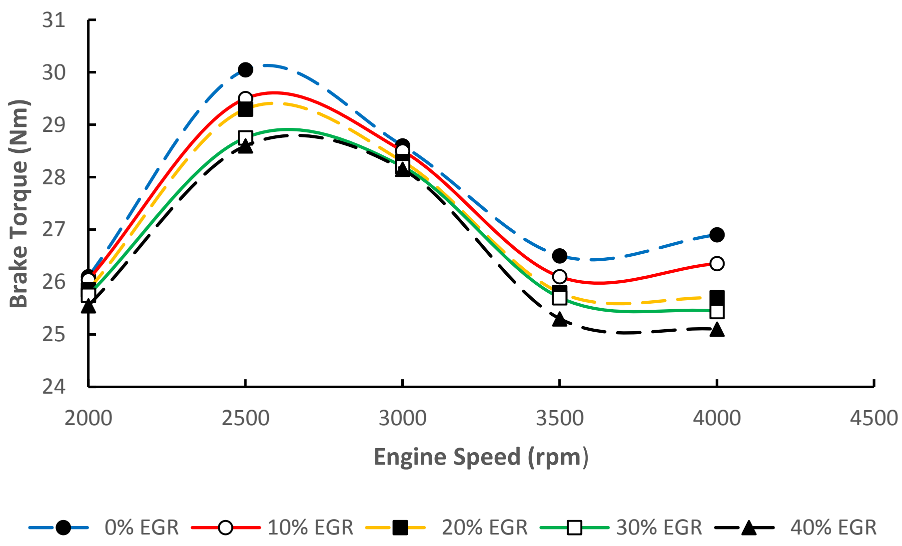

Figure 3.1: Brake torque against engine speed with various rate of EGR.

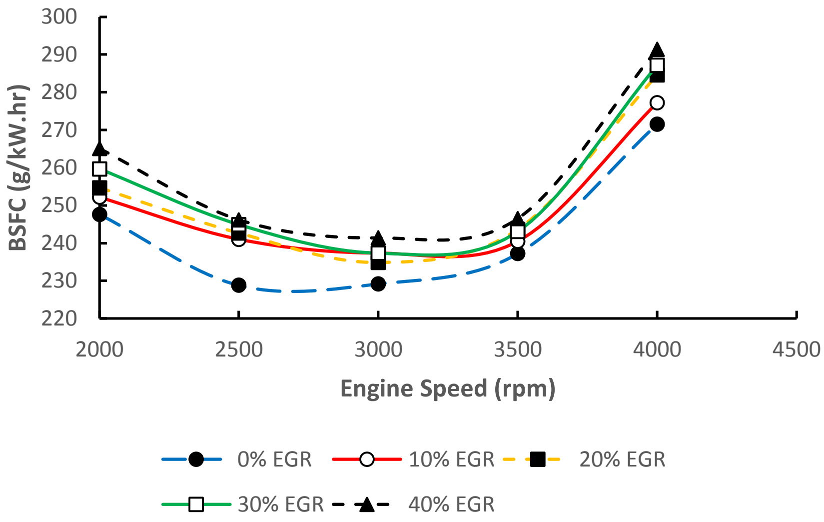

Figure 3.2: BSFC against engine speed with various EGR condition.

Brake specific fuel consumption

The Fig. 5.2 showed the relationship between BSFC against engine speed with various EGR rates. Based on the graph below, it showed that the BSFC graph decrease at early speed from 2000 rpm to 2500 rpm because of EGR lengthen the flame development and propagation process. However, brake specific fuel consumption remain constant while it cruising from 2500 rpm to 3500 rpm due to increase in magnitude of friction (increasing pumping work); increase relative important to heat transfer. The BSFC graph increases at 3500 rpm to 4000- rpm simply because of the more fuel needed to be burnt to overcome dilution effects of incoming charge and increase specific heat of exhaust gases. Comparing the BSFC at various EGR rate showed that the highest BSFC was presented at 40% EGR because of deterioration in combustion quality caused by cycle by cycle combustion variation while the lower BSFC is presented at 0% simply because of reduction in the degree of dissociation in the high-temperature burned gases which allows more of the fuel's chemical energy to be converted to sensible energy near TC. Also, considered the data point at 0% EGR rate for 2000 rpm and 2500 rpm respectively. It was obvious that BSFC for 2000 rpm is 247.62 g/kW.hr while BSFC for 2500 rpm is 228.80 g/kW.hr. This showed about 8% reduction in BSFC at that operating condition. While for 40 % EGR rate at the same operating condition, the BSFC are 265.06- g/kW.hr and 246.17 g/kW.hr respectively. This shows approximately 7% reduction in BSFC (i.e. almost the same reduction as the previous.) Thus, it was plausible to say that, decreasing the EGR rates will decrease the BSFC of the engine. Good agreement achieved between this experiment result and Shehata, M.S & Abdel Razek, S.M [20].

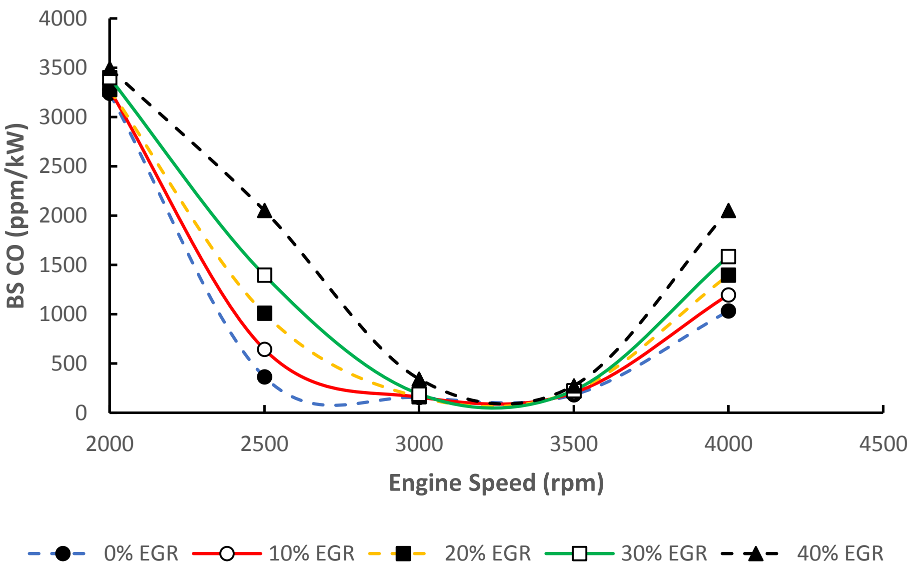

Figure 3.3: BSCO against engine speed with various EGR rates.

Emission Characteristics

The Fig. 3.3, 3.4 and 3.5 represented the BSCO, $BSNO_X$, and BSUHC emissions of CNG fuel with an engine speed increases from 2000 rpm until 4000 rpm for the different EGR rates from 0% to 40% EGR. The experiment was carried out at part throttle, with 180° CA (Partial injection timing) and stoichiometry as air fuel ratio.

Brake specific Carbon Monoxide (BSCO) emissions

The Fig. 3.3 represented brake specific carbon monoxide which show the relationship between BSCO against engine speed with different EGR rates. The trend of the graph showed that BSCO decrease while it cruising from 2000 rpm until 3000 rpm due to increasing amount of inert gases in the mixture, which reduces the adiabatic flame temperature. But it keep constant from 3000 rpm to 3500 rpm because of deteriorates of combustion quality. Nevertheless, the BSCO rise when the engine speed increase to 4000 rpm due to decrease of O_2 concentration in the fresh charge which decreases rate of different reactions. 0% EGR showed the lowest concentration of BSCO emission due to reduction of adiabatic temperature in the cylinder. While 40% EGR recorded the highest concentration of BSCO emission due to occurrence of unstable combustion at that operating condition. Comparing the data points for 0% EGR at 3500 rpm and 4000 rpm correspondingly. It showed that at 3500 rpm, BSCO emissions is 181.72 g/kW.hr while at 4000- rpm is 1030.72 g/kW.hr which revealed increment of 82%. Data point for 40% EGR rate showed that at 3500 rpm is 273.11 g/kW.hr while for 4000 rpm the recorded BSCO emission is 2049.77- g/kW.hr. Therefore it represented 87% of increment. Hence, BSCO emissions get higher with EGR operation rates and did give significant effect by increasing the concentration of BSCO at that operating operation. The result obtain here was coherent with Shehata, M.S & Abdel Razek, S.M [20].

Brake specific Nitrogen Oxide (BSNO) emissions

The Fig. 3.4 illustrated Brake Specific Nitric Oxide emission graph which represent BSNO- emissions against engine speed with various EGR rates. The graph showed that at the low engine speed of 2000 rpm, the emission of BSNO is at the lowest point before it increases moderately when engine speed is increased from 2000 rpm to 3000 because sufficient amount of oxygen may lead to formation of NO. However the emissions decrease from 3000- rpm to 4000 rpm. This might be so because large specific heat capacity of CO_2 and H_2O will absorb more released heat and decrease the cylinder gas temperature and consequent upon this the concentration of NO emission in the engine cylinder will decreases. The highest concentration of BSNO emissions occurred at 0% EGR due to increase of oxygen concentration which leads to formation of more BSNO and for 40% EGR has the lowest BSNO emissions concentrations because EGR decrease the amount of oxygen concentrations in the cylinder which reduce BSNO emission. Compare the BSNO emission concentrations at 0% EGR and 40% EGR respectively from 3500 rpm to 4000 rpm. The data recorded showed that BSNO emission concentration for 0% EGR at 3500 rpm is 225.31 g/kW.hr while at 4000- rpm is 112.57 g/kW.hr. Decrement of about 50% in BSNO emission concentrations is achieved. While, for 40% EGR with the same operating condition. The BSNO emission concentrations were 149.26 g/kW.hr and 66.85 g/kW.hr respectively. This showed decrement of approximately 55%. Hence increase of the EGR rates will leads to decrease in concentration of BSNO emission gas. The result showed a good relationship between these experimental results and Hussain, J. [19].

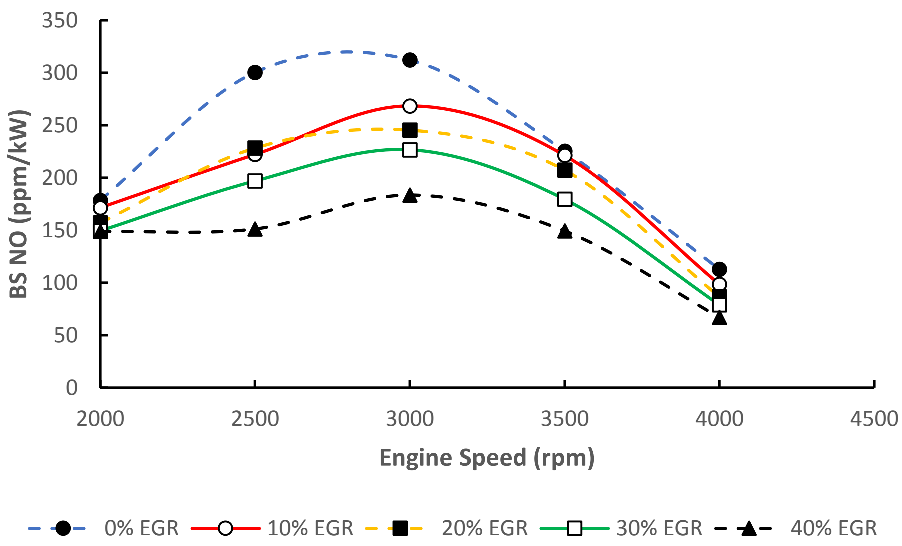

Figure 3.4: BSNO against engine speed with various EGR rates.

Brake specific Unburned Hydrocarbon (BSUHC) emissions

The Fig. 3.5 showed the relationship between BSUHC emissions against engine speed with variables condition of EGR. The trend of the graph showed that it decrease moderately as the speed changes from 2000 rpm until 4000 rpm due to lack of oxidation within the cylinder and the intake manifold. As the exhaust gas flow increases, the residence time in critical sections of the exhaust system decreases and reduction in exhaust port BSUHC occurs. The highest concentrations for BSUHC occurred at 40% EGR due to reduction of oxygen concentration in the fresh charge and decrease flame temperature. While 0% EGR showed the lowest point of BSUHC emission concentrations because of fuel composition can have a significant influence on the concentrations of the emission at that operating condition. Compare the concentration of BSUHC emissions for 0% EGR at 2000 rpm and 2500 rpm respectively. The result showed that at 2000 rpm is 231.50 g/kW.hr while for 2500 rpm is 140.56 g/kW.hr. This revealed decrement of 40% for 0% EGR rate. While for 40% EGR rate showed that at 2000 rpm the BSUHC is 247.64 g/kW.hr while for 2500 rpm recorded BSUHC is 164.03 g/kW.hr. This represented 34% decrement in BSUHC emission concentrations for 40% EGR rate. Hence, BSUHC emissions get higher with EGR operation rates increase. Results obtained here was coherent with Heywood, J [2].

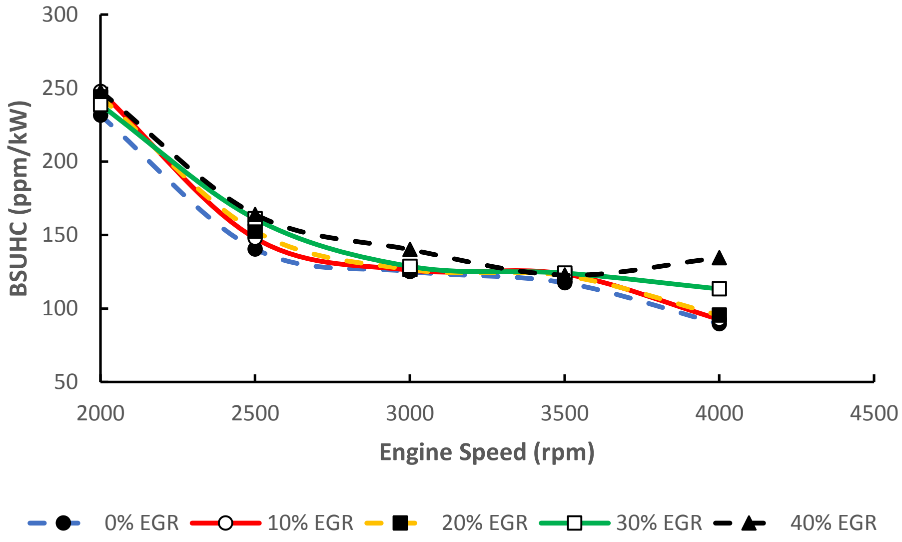

Figure 3.5: BSUHC against engine speed with various EGR rates.

CONCLUSION

An experimental study has been performed on the DI SI engine fueled by CNG so as to study the effect of EGR rates on the aforementioned engine at part throttle and stoichiometric mixture. The main results contributed to enhance knowledge for performance and emissions characteristics of the engine are summarized below:

➢ The brake torque decrease as EGR rate increases. This is due to reduction of burning velocity occasioned by increase in EGR rates while BSFC decrease as EGR rate increase (improvement in fuel consumption). BSNO emissions decrease as EGR rate increase. This is due to capability of EGR to reduce oxygen concentration. While BSCO decrease as the EGR rates increase due to increasing amount of inert gases in the mixture where it reduces the adiabatic flame temperature. BSUHC decrease as the EGR increase because lack of oxidation within the cylinder and intake manifold.

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REFERENCES

[1] Brady, R. N., 2013. Internal Combustion (Gasoline and Diesel) Engines.

[2] Heywood, J., 1988. *Internal Combustion Engine Fundamentals.* New York: McGraw-Hill.

[3] Hollembeak, B., 2005. *Automotive Fuels and Emissions.* United States of America: Thomson Delmar Learning.

[4] Fairbanks, W., 1997. *Diesel Engine & Spark Ignition Gasoline Engines- Which Is Green?* U.SDepartment Energy.

[5] Bellis, M., 2016. *Inventors* . Retrieved from The History of the Automobile : http://inventors.about.com/library/weekly/aacarsgasa.html.

[6] Ashwin, S., 2009. *Seminar4u.blogspot*. Retrieved from Diesel Engine Fundamentals: http://seminar4u.blogspot.my/2009/02/diesel-engine-fundamentals.html.

[7] Yunus, A.C. and A. Michael, 2011. Thermodynamics: An engineering approach. United State:McGraw Hill.

[8] Cho, H. M. and B.Q. He, 2014. Spark ignition natural gas engines. *Energy Conversion and Management*, 608-618.

[9] Guwahati, T. 2013. Internal Combustion Engines. Alternative Fuels Data Centre, Alternate Fuels Technologies Inc.

[10] Jiang Deming, Chen Changyou, Yang Jialin, 2006. Advance Internal Combustion Engine Fundamentals. Xi'an: Xi'an Jiaotong University Press. 103-107.

[11] Darade, P. M. 2012. Performance and Emissions of Internal Combustion Engine Fuelled With CNG - A Review, 1(5), 473–477.

[12] Ajinkya B, Amritkar, Nilesh Badge. 2016. EFFECT OF EXHAUST GAS RECIRCULATION (EGR) IN INTERNAL. 1181.

[13] Münz, S., Römuss, C., Schmidt, P., Brune, K.-H., Schiffer. 2008. Diesel engines with low-pressure exhaust-gas recirculation. Germany: Vieweg Verlag.

[14] Zandaryaa, S. and A. Buekens, 2014. Control of Nitrogen Oxides. *Pollution Control Technologies-II*.

[15] Reifarth, S. H.-E, Ångström, 2009. Transient EGR in a Long-Route and Short-Route EGRSystem. *Efficiency and Mixing Analysis of EGR Systems for Diesel Engines*.

[16] Alain Maiboom, Xavier Tauzia., Jean-Franc-ois . 2008. Experimental study of various effects of EGR on combustion and emission of an automotive direct injection diesel engine. . *Energy*, 22-34.

[17] Harshraj Dangar, P. Gaurav, Rathod. 2013. Combine Effect of Exhaust Gas Recirculation (EGR) and Varying Inlet Air Pressure on Performance and Emission of Diesel Engine. *IOSR Journal of Mechanical and Civil Engineering*, 26-33.

[18] Gomaa, M.A.A. 2011. The effect of EGR rates on NO_X and smoke emissions of an IDI diesel engine fuelled with Jatropha biodiesel blends. *INTERNATIONAL JOURNAL OF ENERGY AND ENVIRONMENT*, 477-490.

[19] Hussain, J. 2012. Effect of Exhaust Gas Recirculation (EGR) on Performance and Emission characteristics of a Three Cylinder Direct Injection Compression Ignition Engine, 245-246.

[20] Shehata M.S and S.M. Abdel Razek, 2008. Engine performance parametres and emissions reduction methods for spark ignition engine, 38-48.