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An Experimental Study of Different Effects of EGR Rates on the Performance and Exhaust Emissions of the Stratified Charge Piston Direct Injection Compressed Natural Gas Engine

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Abstract: Exhaust Gas Recirculation (EGR) is one of the principal techniques used to control spark ignition NO_X. A fraction of the exhaust gas is recycled through a control valve from the exhaust to the engine intake system. However, EGR has different effect on performance, combustion and emissions production that are difficult to distinguish such as increase of intake temperature, delay of Rate Of Heat Rrelease (ROHR), decrease of peak heat release, decrease in oxygen concentration etc. Therefore the impact of EGR on the aforementioned engine parameters (i.e., performance, combustion and exhaust emission) is not perfectly understood, especially under high EGR rates. An experimental study has been conducted to analyze various effects of EGR rates on the performance and emissions of the stratified charge piston direct injection compressed natural gas engine and to determine the stable operating limit of the engine at different excess air ratios (λ = 0.9, 1.0, 1.1 and 1.2) which represents rich, stoichiometric, slightly lean and moderately lean mixture respectively. The results showed that as the EGR is increased, the brake torque, brake specific fuel consumption decreased, while nitric oxide emissions (NO) reduced drastically at various fraction of EGR, just as Unburnt Hydro Carbon (UHC) increased. EGR has no significant effect on carbon monoxide (CO) emission. The addition of EGR also reduces cylinder's gas temperature and pressure. It can be concluded that in introducing EGR in DI-CNG engines, there is a tradeoff between the engine's performance and NO_X emission, while it is difficult to realize stable combustion at high temperature.

Key words: Compressed natural gas engine, exhaust gas recirculation, direct injection engine, performance and emissions

INTRODUCTION

Due to the skyrocketing of energy price; couple with the increasing concern about energy shortage and environmental protection, research on improving the engine fuel economy and reducing exhaust emissions has become imperative in combustion and engine development (Rousseau *et al*., 1999). The development of alternative fuel engine has been the major concern of the engine community and alternative fuels usually belongs to clean fuel compare to diesel and gasoline in engine combustion process. The introduction of the alternative fuel is expected to minimize the fuel shortage and reduces the exhaust emission from the engine (Ben *et al*., 1999).

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 less 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. However, natural gas as an engine fuel suffers two major setbacks viz: Slow burning velocity and poor lean burn capability which often leads to cycle-by-cycle variation (Rousseau *et al*., 1999; Ben *et al*., 1999). 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 (Akansu *et al*., 2004). To solve these problem of high NO_X , EGR is introduced into the intake system with the aid of regulating valve.

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Exhaust Gas Recirculation (EGR) is one of the common ways to control in-cylinder NO_X production and is used on most modern high-speed direct injection diesel engine. Because NO_X formation progresses much faster at high temperatures, EGR reduces the amount of NO_X the combustion generates. However, large EGR mass fraction may have an unfavorable effect on the engine performance, fuel economy and HC emission. Proper determination of EGR mass fraction needs to be determined by comprehensively evaluating engine performance, fuel economy and NO emission (Das and Watson, 1997; Heywood, 1988).

The most important engine variable that affects NOx emission are fuel/air equivalence ratio, spark timing and burnt gas fraction in charge. The burnt gas fraction in charge depends on the amount of diluents introduction such as EGR. NO_X emission can be greatly decreased via lean combustion, retarding the ignition timing and introducing EGR. Most of the previous work concentrated on the method of using lean combustion and retarding ignition timing, while few literature were found in introducing EGR with natural gas to reduce NO_X emission.

Amr and saiful investigated the use of EGR in a supercharged natural gas engine (SI) engine (Ibrahim and Bari, 2010; Zheng *et al*., 2004) studied the effects of EGR on the combustion and emission of the compression ignition engine. Peng *et al*. (2008) conducted on experimental study on the effect of Exhaust Gas Recirculation (EGR) on combustion and emission of the direct injection compression ignition engine. Their results indicated that the use of percentage EGR dilution in the inlet mixture decrease the oxygen concentration and consequently, it decreases the combustion rate significantly. Engine key performance parameter such as brake torque, brake power decreased with increased of EGR; while brake specific fuel consumption increased. The increased of EGR dilution in the inlet mixture decreases both the maximum cylinder temperature and oxygen concentration which leads to a significant reduction in NO emissions. Allenby *et al*. (2001) reported that methane-hydrogen gas engine (Hythane) engine could tolerates up to 25% EGR, while maintaining a coefficient of variability of indicated mean effective pressure below 5% (Allenby *et al*., 2001; Erjiang *et al*., 2009). To date, no literature has ever examined the impact of EGR on the performance and exhaust emissions of the stratified charge piston DI-CNG engine and explore the applicability limit of EGR most especially under lean combustion mode.

This study will seek to investigate the effects of different EGR rates on the performance and exhaust emissions of the stratified charge piston direct injection compressed natural gas engine and quantitatively analyze Brake torque, Brake power, Fuel conversion efficiency, Brake Specific Nitric Oxide (BSNO), Brake Specific Unburnt Hydro Carbon (BSUHC), Brake Specific Carbon monoxide (BSCO) emissions and Coefficient of variation of the indicated mean effective pressure (COV_{IMEP}). The study is expected to provide information on natural gas engine-EGR tolerance limit (driveability) and at this point, define the operating conditions for the engine application or operation.

MATERIALS AND METHODS

The specifications of the test engine are listed in Table 1 and Fig. 1 shows the schematic of the experimental set-up. A four-stroke single cylinder research engine was used to investigate the effect of EGR on the direct-injection compressed natural gas engine.

The supplying system of the engine consists of a natural gas, gas valve, gas flow meter, a pressure regulator. An electronic control unit was used to adjust the amount of CNG and the excess air ratio into the system, also controlled by the electronic are injection timing and spark timing. The composition of the natural gas was given in Table 2 below (Erjiang *et al*., 2009).

The recycle exhaust gas was taken from a hole located on the exhaust pipe with the aid of connecting pipe. Cooling of the hot exhaust gas was done by passing it through a water-cooled heat exchanger. A regulating valve was installed: in order to regulate the exhaust gas flow. Percentage increase of EGR in the inlet mixture was done by increasing the amount of the exhaust gas flowing back to the engine intake system. Figure 2 shows the schematic diagram of the EGR control system.

Air, natural gas (from pipeline supply) and cooled exhaust gas were mixed in the engine intake. A pressure gauge was used to measure the inlet pressure. The research described in this paper was carried out at wide open throttled (WOT), with spark timing set at 31.5°CA, injection timing 300°CA (early injection timing) and four excess air ratio comprises of (λ = 0.9, 1.0, 1.1 and 1.2)

Table 1: Engine specification and test bed

Displacement volume	399.25 cm^3
Cylinder bore	76 mm
Cylinder stroke	88 mm
Compression ratio	14
Exhaust valve closed	ATDC 10°
Exhaust valve open	BBDC 45°
Inlet valve open	BTDC 12°
Inlet valve closed	ABDC 48°
Dynamometer	Eddy current with maximum reading of 50 Nm
E.C.U.	Orbital inc

Table 2: Composition of natural gas

Item	Volumetric fraction %
CH4	96.160
C2H6	1.096
C3H8	0.136
N2	0.001
CO2	2.540
Others	0.067

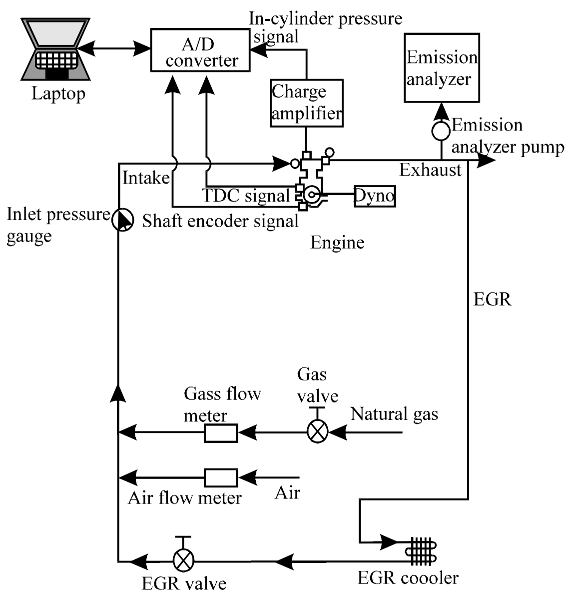

Fig. 1: Schematic diagram of the experimental setup

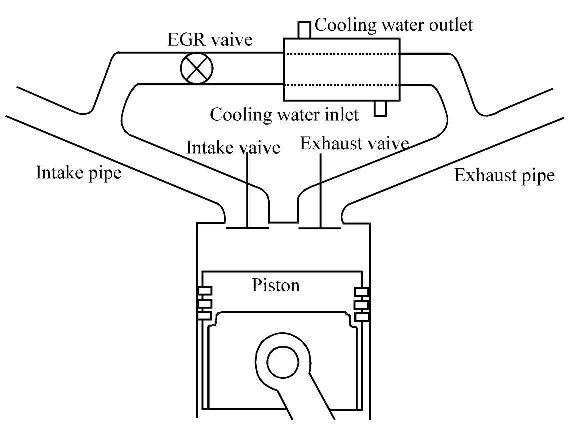

Fig. 2: Schematic diagram of EGR control system

which represent rich, stoichoimetric, slightly lean and lean mixture respectively and two engine speed were being utilized viz, 2000 and 3000 r min^{-1}. The excess air ratios are obtained by setting from the "Gasmet gas analyzer" which contains oxygen sensor.

The system for the acquisition of in-cylinder pressure is composed of:

- Piezo electric cylinder pressure sensor- AVL QH32D, gain 25.28pc bar^{-1} range 0-200bar
- Charge amplifier- AVL3066AO
- Shaft position encoder-AVL364C
- Piezos resistive pressure sensor fixed inside the inlet manifold

The exhaust gas is being measured with the aid of Gasmet gas analyzer which is capable of analyzing up to forty (40) exhaust species.

The reaction mechanisms of natural gas, EGR and air mixture can be represented as follows:

The overall combustion reaction for CH_4+EGR+air mixture can be written as:

$$(1-a)CH_4 + a\,EGR + \left(\frac{2-3a}{2+x}\right)(O_2+3.76N_2) \Leftrightarrow$$
$$(1-a)\,CO_2 + (2-a)\,H_2O + x\left(O_2+3.76N_2\right) +$$
$$3.76\left(\frac{2-3a}{2}\right)N_2$$

where, x is the excess air parameter, α is the EGR mole fraction knowing that the (methane + EGR) mole fraction is equal to 1. The equivalence ratio (ϕ) is defined by the fraction of O_2 needed to obtain complete combustion of the fuels. The EGR as it was being utilized comprises of CO_2 and H_2O only.

METHOD OF CALCULATION

A zero-dimensional thermodynamic model was used to calculate the heat release rate in the study. The model neglects the leakage through the piston rings and thus the energy conservation in the cylinder is written as follows:

From the first law:

$$dQ_{ch} = dU_s + dQ_{ht} + dW \tag{1}$$

$$\frac{dQ_{ch}}{d\theta} = \frac{dU_s}{d\theta} + \frac{dQ_{ht}}{d\theta} + p\frac{dV}{d\theta} \tag{2}$$

The gas-state equation is:

$$pV = mRT \tag{3}$$

The differential equation of the gas-state equation with crank angle θ is given as:

$$p\frac{dV}{d\theta} + V\frac{dp}{d\theta} = mR\frac{dT}{d\theta} \tag{4}$$

The heat release rate $dQ/d\theta$ can be derived from Eq. 2 and 3 as follows:

$$\frac{dQ_{ch}}{d\theta}=\left(\frac{c_v}{R}\right)V\frac{dp}{d\theta}+\left(\frac{c_v}{R}+1\right)p\frac{dv}{d\theta}+\frac{dQ_{ht}}{d\theta} \quad (5)$$

$$\gamma=\frac{c_p}{c_v} \quad (6)$$

Assuming ideal gas, specific gas constant $=R=C_p-C_v$ and therefore the value of specific heat at constant volume can be written as:

$$c_v=\frac{R}{\gamma-1}$$

Substituting Eq. 6 in 5; then equ. 5 becomes:

$$\frac{dQ_{ch}}{d\theta}=\left(\frac{1}{\gamma-1}\right)V\frac{dp}{d\theta}+\left(\frac{\gamma}{\gamma-1}\right)p\frac{dv}{d\theta}+\frac{dQ_{ch}}{d\theta} \quad (7)$$

More so, the convective heat-transfer rate to the combustion chamber wall can be calculated from the relation:

$$\frac{dQ_{ht}}{dt}=Ah_c(T-T_w) \quad (8)$$

The mass fraction burned can be calculated based on Rassweiler-Withrow method by using the average value of pressure data of 100 cycles.

RESULTS AND DISCUSSION

The experimental results in this section were obtained from single cylinder four-stroke direct-injection compressed natural gas engine having 76 mm bore and 88 mm stroke with a compression ratio of 14:1. The fuel comprises of compressed natural gas with different fraction of EGR (i.e., CNG-EGR) were being utilized. The experiment were carried out with EGR rate consists of 0, 10, 20, 30, 40, 50 and 60% at varying excessive air ratios (λ = 0.9, 1.0, 1.1 and 1.2) and engine speed of 2000 and 3000rpm at Wide Open Throttle (WOT). Ignition timing was set to 31.5°CA. A high pressure injector (18 bars) was used on the central direct injection system.

Effects of EGR Rates on performance parameters

Brake torque: Figure 3 shows the variation of brake torque with EGR rates for different speeds and injection timings at WOT. In the Fig. 3a it is obvious that the brake torque decreases as the EGR rates increases. This is largely due to the fact that as EGR rate is increased, the concentration of air and fuel in the cylinder decreased and consequently, it reduces the brake torque. The highest value of brake torque is presented at stoichiometric mixture (λ = 1.0) when compared to the other mixtures. The reason being that at stoichiometric mixture, there is efficient combustion, all the air provided is effectively utilized for combustion process. In addition, increasing the engine speed from 2000 to 3000 rpm at injection timing of 300°CA as shown in Fig. 3b increases the brake torque due to increase in friction mean effective pressure at higher speed which decreases the mechanical efficiency of the engine. Also the brake torque at rich mixture is higher than that at moderately lean mixture (λ = 1.2) due to the fact that at rich mixture (excess fuel), when the engine speed is increased, more fuel will be burnt to deliver more torque.

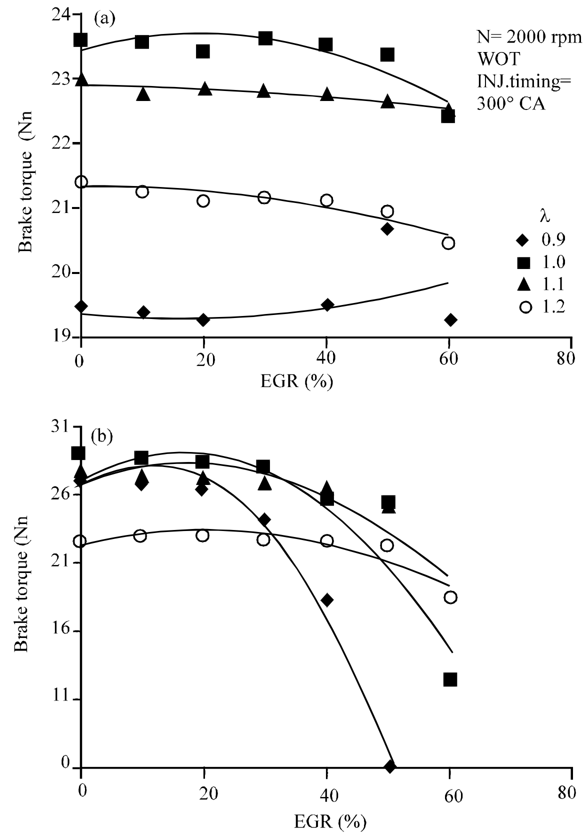

Fig. 3: Brake torque versus EGR rate

In furtherance, at 300°CA injection timing and engine speed of 2000 rpm i.e., Fig. 3a, it is obvious that at stoichiometric mixture the brake torque at both 0 and 60%

EGR rates are 23.5 and 22.7 Nm, respectively. This shows that the brake torque under Wide-Open-Throttle (WOT) and stoichiometric condition and at 60% EGR rate was about 4% lower than that under normal condition without EGR (i.e., 0% EGR rate). Also with the same operating conditions, but at slightly lean mixture (λ=1.1), the brake torque at both 0 and 60% EGR rates are 23 and 22.6 Nm. This is pointing to the fact that at 60% EGR rates brake torque is approximately 2% lower than that under normal condition without EGR (i.e., 0% EGR rate). While for the moderately lean mixture (λ=1.2) the brake torque respectively for both EGR rates under consideration are 21.5 and 20.5 Nm. This is an indication of about less than 5% reduction in brake torque.

Brake specific fuel consumption: Figure 4 shows the variation of Brake Specific Fuel Consumption (BSFC) at different EGR rates for different speeds and injection timings at WOT. It is clear from the Fig. 4a that BSFC decreases as EGR rates increases. This represent an improvement in fuel consumption and it is due to the following (1) reduction in pumping work as EGR is increased at constant brake load (fuel and air flows remain almost constant, hence intake pressure increases). (2) reduced heat loss to the walls because the burned gas temperature is decreased significantly and (3) a 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 TDC.

More so, BSFC decreases approximately as the mixture is richened (for maximum power) above stoichiometric due to the decreasing combustion efficiency associated with richening mixture. For mixture lean of stoichiometric, the theoretical BSFC decreases linearly as λ increases above stoichiometric. Combustion of mixtures leaner than stoichiometric produces product at lower temperature and with less dissociation of the triatomic molecules, CO_2 and H_2O. Thus the fraction of the chemical energy of the fuel which is released as sensible energy near TC is greater, hence a greater fraction of the fuel's energy is transferred as work to the piston during expansion and the fraction of the fuels available rejected to the exhaust decreases.

In addition, at injection timing of 300°CA, increasing the engine speed from 2000 to 3000 rpm, it shows that the rich mixture follows the anticipated pattern of significant BSFC reduction until at about 15% EGR before it start increase and this increment in BSFC is caused by larger cycle-by-cycle variations. Deterioration in combustion starts to occur almost immediately on the lean side of stoichiometric and the fuel consumption worsens for $\lambda \geq 1.1$.

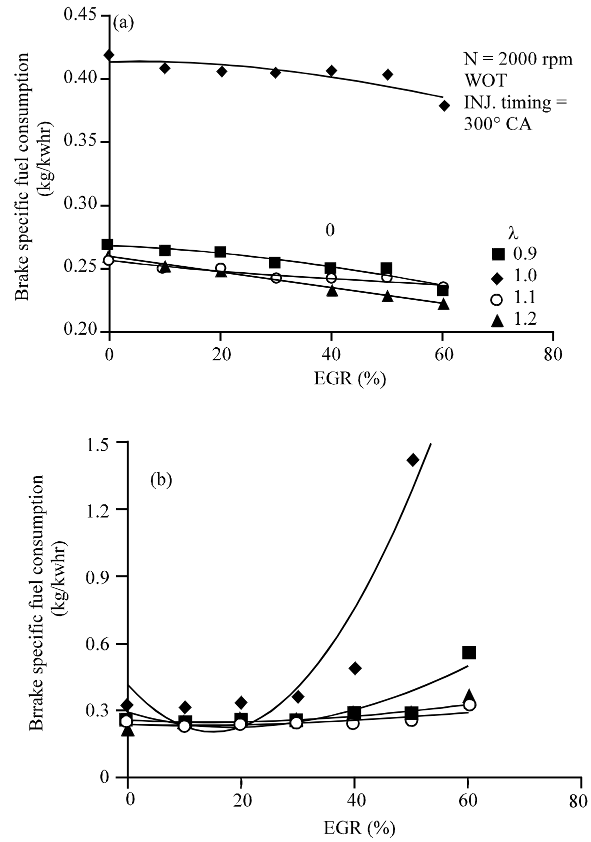

Fig. 4: Brake specific fuel consumption against EGR rates

Effects of EGR rates on emission parameters

Brake specific nitric oxide emissions (bsNO): Figure 5 shows the variation of brake specific nitric oxide emissions (bsNO) at different EGR rates for different speeds and injection timings at WOT. From Fig. 5 and 4a it is obvious that the Brake Specific Nitric Oxide (BSNO) decreases as EGR rate increases i.e., NO concentration is decreasing with increasing of EGR rates. This is because as EGR rate is increased, the burning velocity and combustion maximum temperature will reduce due to the dilution effect of EGR and also large specific heat capacity of CO_2 and H_2O will absorb more released heat and decrease the cylinder gas temperature. However, the decrease in cylinder gas temperature during combustion process reduces the formation of NO and eventually results in decreasing the NO concentration.

In addition, the highest NO concentration is presented at excess air ratio of 1.1 (λ = 1.1) which is slightly leaner than the stoichoimetric equivalence ratio. The oxygen concentration at slightly lean mixture results in higher NO concentration compared with that at stoichoimetric equivalence ratio. Further increase in excess air ratio will remarkably decrease the cylinder gas temperature and decrease NO concentration.

More so, lowest NO emissions consistent with good fuel consumption (avoiding the use of rich mixtures) are obtained with a stoichiometric mixture with as much as the engine will tolerate without excessive deterioration in combustion quality.

In addition at 300°CA injection timing and engine speed of 2000 rpm as shown in Fig. 5(a) it is obvious that at stoichiometric mixture (λ = 1.0) the concentration of NO emission released at both 0 and 60% EGR rates are 220 and 100 ppm/kW respectively. This shows that the concentration of NO emission under Wide-Open-Throttle (WOT) and stoichiometric condition was about 50% lower than that under normal condition without EGR (i.e., 0% EGR rate).While at slightly lean mixture (λ = 1.1) with the same operating conditions, the concentrations of NO emissions respectively for the EGR rates under consideration are 400 and 300 ppm kW^{-1}. This shows that at slightly lean mixture there is 25% reduction in concentration of NO. Also at moderately lean mixture (λ =1.2) the concentration of both EGR rates are 330 and 250 ppm kW^{-1} respectively. This reveals that there is about 24% reduction in the concentration of NO at moderately lean mixture. Thus it is reasonable to say that large fluctuation results at stoichiometric mixture as compare to the other mixtures i.e., NO is more sensitive to EGR and excess air ratio (λ).

More so, at injection timing of 300°CA, increasing the speed from 2000 to 3000 rpm moderately increases NO concentration. The reason being that the EGR rate decreases as the speed increases, the effects being greater at lower inlet manifold pressures (lighter loads). Also the relative importance of heat transfer per cycle is less as speed increases, which would be expected to increase NO concentration. More over, by comparing the graphs in Fig. 5a and b it is quite obvious that there is a large reduction in NO concentrations at engine speed of 3000 as compare to engine speed at 2000 rpm. The reason might be due to large cycle-to-cycle variation which is caused by variation in mixture composition or gas motion. However this cycle-to-cycle variation is measured by coefficient of variation in indicated mean effective pressure (COV_{IMEP}) and detail about this is given in Fig. 9 for better and more comprehensive understanding.

It is noteworthy to say that the use of rich mixture should be avoided while controlling the NO emissions in a direct injection compressed natural gas engine with

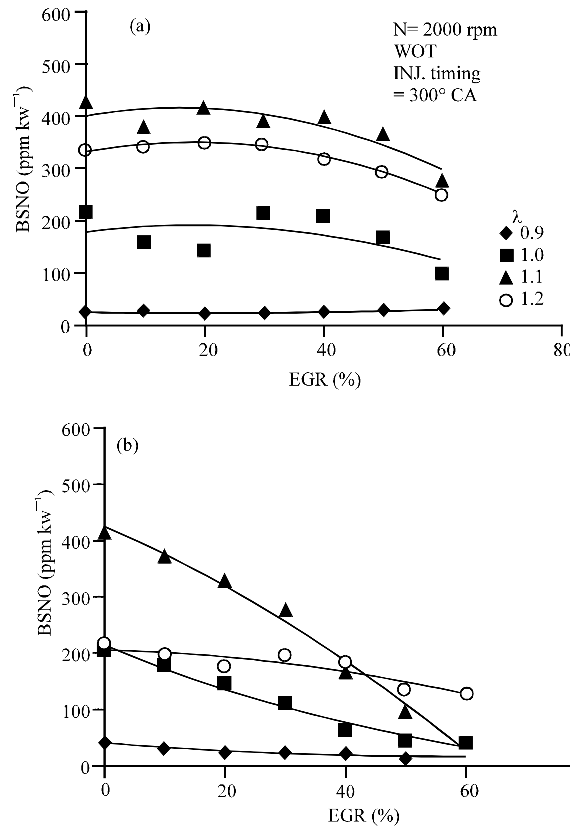

Fig. 5: Brake specific nitric oxide emissions (BSNO) versus EGR rates

EGR. Figure 5a and b supports this claim. Good agreement is achieved between these experimental results (Erjiang *et al*., 2009).

Brake specific Unburnt Hydro Carbon (bsUHC): The variations of brake specific Unburnt Hydrocarbon (UHC) emissions with EGR rates are shown in Fig. 6. From Fig. 6 (a) and (b) it is obvious that at first the increase in HC is modest and is due primarily to decrease HC burn up due to lower expansion and exhaust stroke temperatures. The HC increases become more rapid as slow combustion, partial burning and even misfire in turn occur with increasing frequency. Exception to these, is the rich mixtures (λ =0.9) at engine speed of 2000 rpm and injection timing of 300°CA i.e., Fig. 6a. It shows an decreasing trend up to about 25% EGR and then starts to increases. More so, rich mixture shows that emissions are high. This is primarily due to lack of oxygen for after burning of any unburnt hydrocarbon that escape the primary combustion process within the cylinder and exhaust system. HC emission decrease as the stoichoimetric point is approached; increasing oxygen concentration and

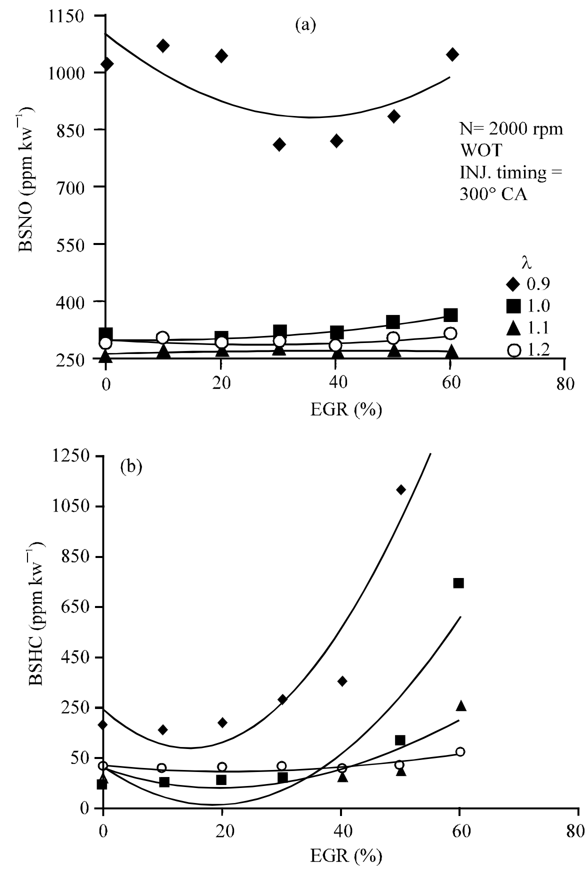

Fig. 6: Brake specific Unburnt Hydrocarbon (bsUHC) versus EGR rates

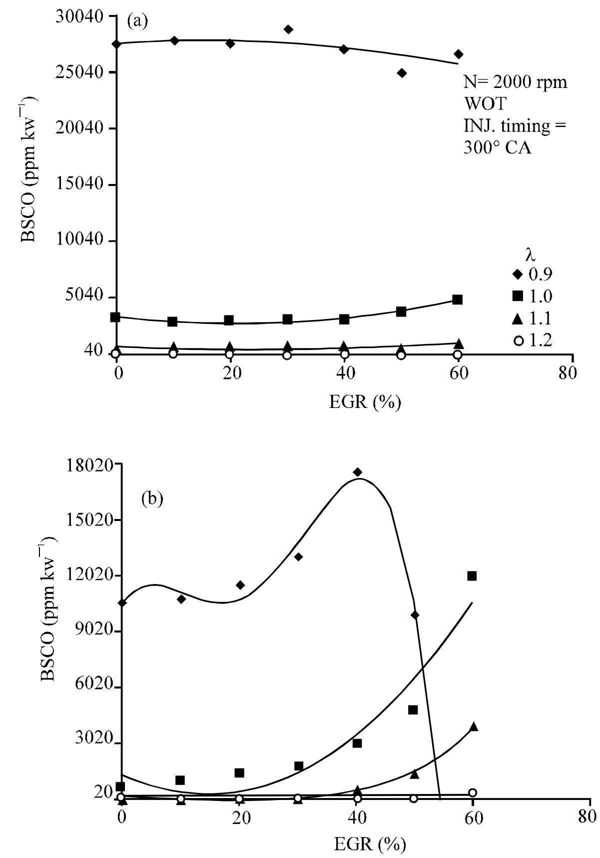

Fig. 7: Brake specific carbon monoxide emission versus EGR rates

increasing expansion and exhaust stroke temperature results in increasing HC burn up.

For moderately lean mixture, HC emission level varies little with excess air ratio. Decreasing fuel concentration and increasing oxygen concentration essentially offset the effect of decreasing bulk gas temperatures. As the lean operating limit of the engine is approached, the combustion quality deteriorates significantly and HC emissions start to rise again due to occurrence of occasional partial burning cycles. For still leaner mixture, HC emission rise more rapidly due to the increasing frequency of partial-burning cycles and even the occurrence of completely misfiring cycle.

Comparing the concentration of HC emission at both engine speeds of 2000 and 3000 rpm respectively and at injection timing of 300°CA shows that higher concentration of HC occurs at 3000 rpm at rich mixture which is about 1250 ppm kW^{-1} i.e., Fig. 6 (b) as compared to the engine speed at 2000 rpm (i.e., 1100 ppm kW^{-1}) Fig. 6 (a). This shows that when engine speed is increased from 2000 to 3000 rpm there is about 12% increment in the concentration of HC emissions. This suggests that if oxygen is available, oxidation of unburned hydrocarbons both within the cylinder and in the exhaust system will be significantly enhanced by increases in speed since the expansion stroke and exhaust process gas temperature increase substantially, due to the reduced significance of heat transfer per cycle with increasing speed. This more than offsets the reduced residence time in the cylinder and in the exhaust.

Brake specific carbon monoxide (bsCO): Figure 7 shows the graphs of brake specific CO emission versus EGR rate for different engine speeds and injection timings at WOT. From the Fig. 7a and b it shows that EGR has no significant effect on engine CO emissions. Highest level of CO emission is presented at rich mixture when compared to stoichoimetric and lean mixture respectively, because complete oxidation of the fuel carbon to CO_2 is not possible due to insufficient oxygen (incomplete combustion). For lean mixtures, CO levels are approximately constant at low level of about 0.5% or less.

Effects of EGR rates on combustion parameters

Maximum cylinder pressure: Figure 8 gives the maximum cylinder pressure at different EGR rates. The maximum cylinder pressure decreased as EGR rates increases, this is largely due to the fact that with increase of EGR mass fraction, the burning velocity and combustion maximum temperature will reduced due to the dilution effect of EGR and consequent upon this cylinder maximum pressure will decrease. Also the large specific heat capacity of CO_2 and H_2O will absorb more released heat and decrease the cylinder gas temperature and this will give rise to reduction in cylinder maximum pressure. The highest value of maximum pressure is presented at stoichoimetric in comparison to lean and rich mixture, this is because there is efficient utilization of oxygen concentration during combustion process at stoichoimetric as compared to lean mixture.

In addition, when engine speed is increased from 2000 to 3000 rpm, there is an increase in cylinder maximum pressure at excess air (λ = 1.1) above stoichoimetric and this due to the fact that oxygen concentration will be higher at excess air or slightly lean condition.

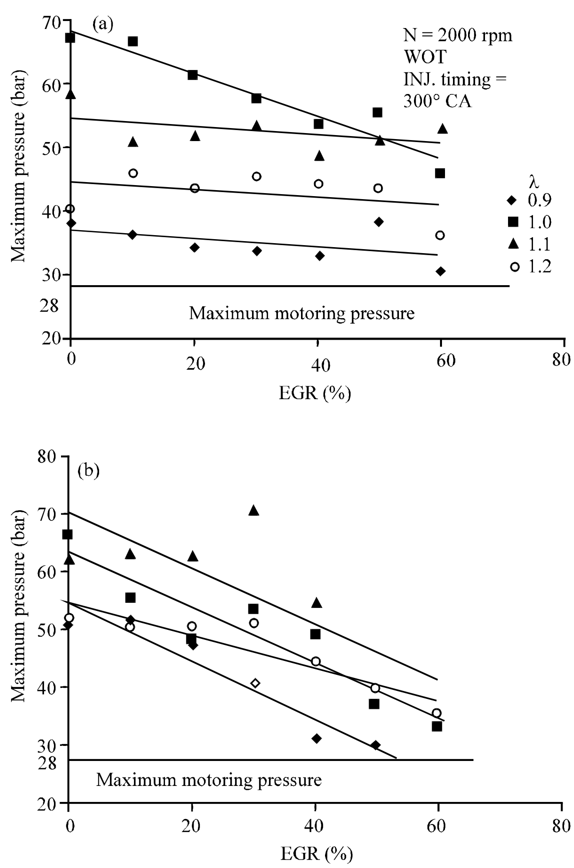

Fig. 8: Maximum cylinder pressure versus EGR rates

Consequently, the burning velocity and combustion maximum temperature will rise and this will automatically leads to increase in cylinder maximum pressure.

The results equally shows that at the engine speed of 2000 rpm, when EGR exceed 40%, the maximum cylinder gas pressure is almost equal to the maximum motoring pressure (28bar) due to occurrence of misfire and/or the partial burning cycles. Therefore at engine of 2000 rpm and EGR rate of over 40% will not take into account in combustion analysis. When engine speed is increased to 3000 rpm, the increase in turbulence intensity increases the flame propagation speed and the engine can still maintain the normal combustion at EGR rate of 50% or more depending on the combustion strategy (i.e., rich or stoichoimetric or slightly lean or lean) and this is pointing to the fact that increasing the engine speed can extend the tolerated EGR limit. However, the maximum pressure variation depends on both changes in phasing and burning rate.

The magnitude of this variation depends on whether the combustion chamber is faster or slower burning, on average. It is also depend on the cyclic cylinder fuel and air charging variations.

Heat released rate: Figure 9 shows the effects of varying EGR dilution in the inlet mixture on the net heat release rate. As seen in this figure, there are cycle-by-cycle variations in the early stage of flame development (from 0 to few percent heat released rate) and in the major portion of the combustion process-the rapid burning phase-indicated by variation in a maximum burning rate. The maximum heat release decreases as the EGR rates increases. This is largely due to the fact that with the increase of EGR rates, the concentration of air and fuel in the cylinder decreases and this decrease the cylinder gas temperature and consequent upon this; the heat release rate decreases.

Also, considering Fig. 9a-d, starting from (a) it shows that there is about -0.005 kJ/°CA heat released rate in the beginning. The negative value implies that the energy is absorbed by the fuel from the ignition source prior to combustion. However, the energy absorption gives rise to increase in ignition delay and consequent to this, the combustion duration increases, that explains the reason why the combustion duration at stoichiometric (150°CA) is longer as compared to that at moderately lean mixture (50°CA). In other words, it shows that moderately lean mixture has a faster burning i.e. shorter combustion duration (due to increase in turbulence intensity and also

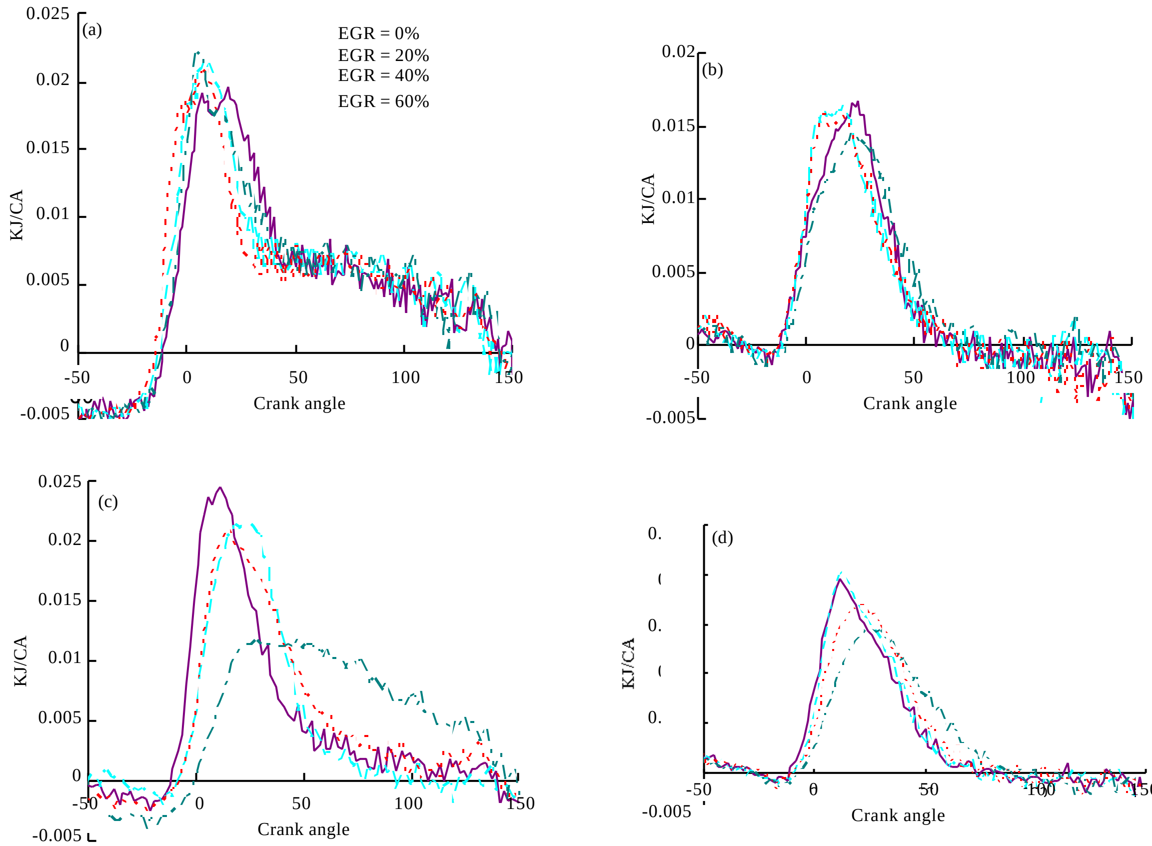

Fig. 9: Heat released rates characteristics for different EGR rates at 2000, 3000 and 5000 rpm and at different excess air ratios. (a) Heat released rate @ 2000 rpm @ stoichiometric mixture (b) Heat released rate @ 2000 epm @ moderately lean mixture (c) Heat released rate @ 3000 rpm @ stoichiometric mixture

on the reaction rate which is dependent on the mixture composition) as compared to stoichiometric mixture at the same engine speed. In addition, at engine speed of 3000 rpm, the combustion duration respectively for both mixtures (i.e. stoichiometric and moderately lean mixtures are 100 and 75°CA. These results obviously show that the moderately lean mixture has a shorter combustion (due to faster burning) as compared to stoichiometric mixture. However, the maximum heat released rate peaks at stoichiometric as compared to that of lean burn. The reason being that at stoichiometric, all the air provided is effectively utilized for combustion process (efficient combustion).

At engine speed of 2000 rpm, the maximum heat release rate occurs at stoichiometric mixture which is almost equal to 0.023 kJ/°CA at 20°CA ATDC as compared to moderately lean mixture which is 0.017 kJ/°CA at 18°CA ATDC, while at engine speed of 3000 rpm, the maximum heat released equally occurs at stoichiometric mixture which is approximately equal to 0.025 kJ/°CA at 20°CA as compared to moderately lean mixture which peaks at 0.020 kJ/°CA at about 15°CA ATDC.

In addition, as the mixture become leaner with excess air or more dilute with a higher burned gas fraction from residual gas or EGR, the magnitude of cycle-by-cycle combustion variation increases. Eventually some cycles become sufficiently slow burning that combustion is not completed by the time the exhaust opens, a regime where partial burning occurs in a fraction of the cycle is encountered. For even leaner or more dilute mixtures, the misfire limit is reached. At this point, the mixture in a fraction of the cycle fails to ignite.

Furthermore, heat released rate increases as the speed increases, i.e., increasing the engine speed from 2000 to 3000 rpm, the maximum heat release rate varies from 0.023 to 0.025 (stoichiometric mixture) and 0.017 to 0.020 kJ/CA (moderately lean mixture). This is so, because increase in speed will increase the turbulence intensity within the cylinder and since the rate of heat-release

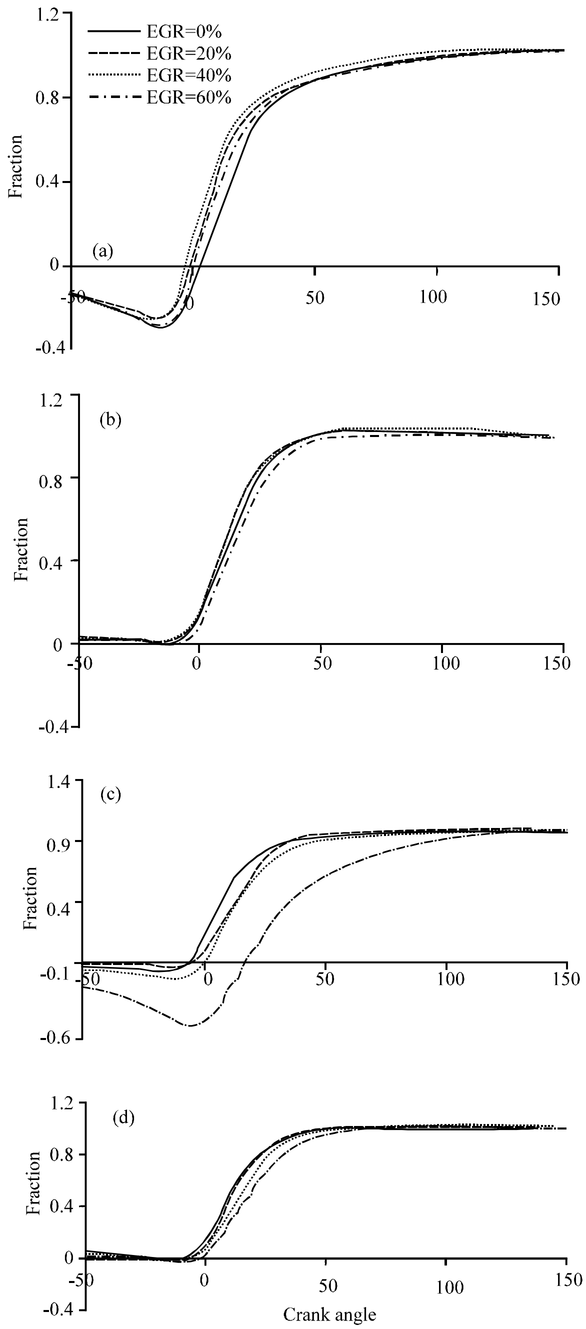

Fig. 10: Mass fraction burned profile for different EGR rates at 2000, 3000 and 5000 rpm and at different excess air ratios. (a) Mass fraction burned @ 2000 v rpm @ (stoichoimetric), (b) Mass fraction burned @ 2000 rpm (moderately lean), (c)Mass fraction burned @ 3000 rpm (stoichoimetric), (d) Mass fraction burned @ 3000 rpm (moderately lean)

depends largely on turbulence intensity and the reaction rate which is dependent on the mixture composition, hence the heat-release rate will be increased. Figure 9a-d supports this claim. While the relative rates are essentially independent of speed, indicating that combustion rate which depends on fuel-air mixing scale approximately with engine speed.

Mass fraction burned: Figure 10 shows the mass fractioned burned as a function of a crank angle at different EGR rates. It is quite obvious that, increasing the EGR rates, the accumulated mass fraction burned grows up slowly, due to the decrease in burning (flame propagation) velocity. In furtherance, parts of the heat released will be absorbed as a result of introduction of large specific heat capacity gas like CO_2 and H_2O vapour from the exhaust and decrease the combustion temperature and consequent upon this, flame propagation speed decreased.

It is shown in Fig. 10a, that mass fraction burnt starts with almost -0.15 at the time of the spark and then starts to increase from approximately -30° to -15°CA after the spark timing. This crank angle interval between the spark timing and start of combustion is called flame development duration or sometimes ignition delay. However, flame development duration reflects the flame development at early stage and it is related to the ignition delay which is dependent on the mixture concentration and temperature.

More so, considering Fig. 10a and c i.e., increasing the engine speed from 2000 to 3000 rpm at stoichiometric mixture clearly shows that increasing the engine speed increases the ignition delay simply because residual gases in the cylinder will increases as a result of increase in speed.

Following, the flame development duration, the mass fraction burned noticeably increases with crank angle until it reaches its maximum value where essentially almost all the fuel chemical energy has been released at basically the end of combustion process. Figures 10a-d indicates that the maximum mass fraction burnt, which essentially identifies the end of combustion occurs later as the percentage of EGR dilution increases.

Determination of EGR-stable operating limit: COV_{IMEP} is one important measure of cyclic variability, derived from pressure data. It is the standard deviation in imep divided by the mean imep and it is usually in percent (%). Mathematically, it is given as:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP_{avg}} \quad (9)$$

It is a measure of the cyclic variability in indicated work per cycle. Empirically, it has been found that vehicle driveability problem usually results when COV_{IMEP} exceeds

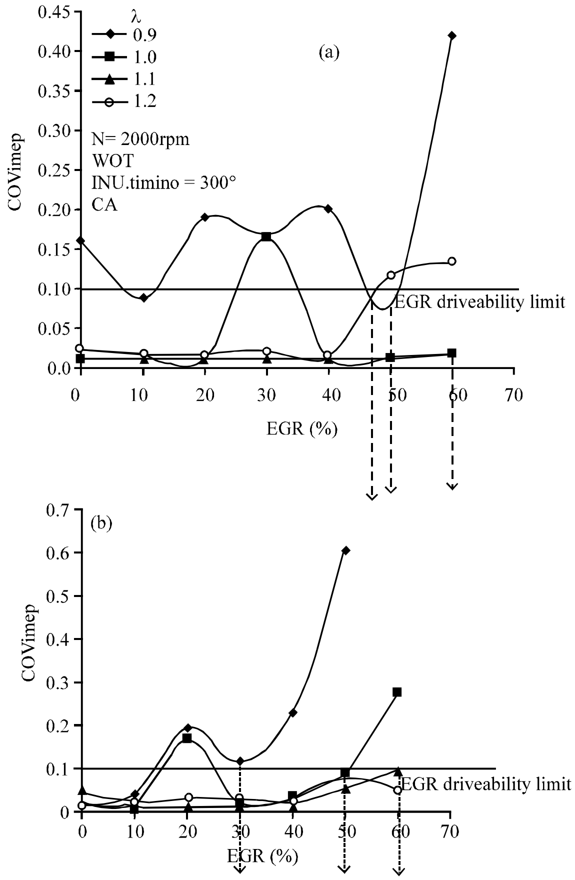

Fig. 11: Coefficient of variation in indicated mean effective pressure

10%. It is also an important parameter to determine the EGR tolerance limit by the engine. Based on these, the direct injection compressed natural gas engine-EGR tolerance limit could therefore be determined by using COV_{IMEP} not exceeding 10% and this would be applied to Fig. 11 (a) and (b) to determine the engine-EGR tolerance limit and also define the engine operating condition at this point.

Figure 11 the result showed that when an engine speed of 2000 rpm, ignition timing of 31.5° CA, injection timing of 300° CA (early injection timing), 18 bar and 315 k of pressure and temperature respectively is applied, the EGR tolerance limit by the engine is indicated by the dotted arrows pointing down for each of the mixture in Fig. 11a and b, respectively and it is given as: (a) Rich mixture (λ = 0.9) is 50%, Stoichoimetric (λ =1.0) is 60% or more, slightly lean (λ =1.1) is 60% or more and Leaner mixture (λ =1.2) is 45%.

When the engine speed is increased to 3000 rpm, while other operating condition remains the same. (b) Rich mixture (λ =0.9) is 30%, Stoichoimetric (λ =1.0) is 50%, Slightly lean (λ =1.1) is 60% or more and Leaner mixture (λ =1.2) is 60% or more.

From the foregoing, it is cleared that the engine could still tolerate more EGR at leaner mixture when engine speed is increased. this is so because an increase in engine speed will increase the turbulence intensity and this in turn increase the flame propagation speed and engine will maintain a normal combustion when egr rates is increased. The experimental result is consistent with (Erjiang *et al*., 2009). Also at rich and stoichoimetric mixture; the engine will tolerate lesser EGR when the engine speed is increased.

CONCLUSIONS

An experimental study of the different effects of EGR rate on the performance and exhaust emissions of the stratified charge piston DI-CNG engine has been conducted and the following are the major contribution to the body of knowledge:

- Engine could still tolerate more EGR, especially when operating at moderately lean condition (λ = 1.2) provided the engine speed is increased and this is largely due to increase in turbulence intensity which give rise to increase in flame propagation speed and engine will maintain normal combustion when the EGR rate is increased
- Degradation must be accepted in engine performance and efficiency when using EGR to reduce NO_X emissions
- By carefully optimizing the choice of operating parameters, EGR can improve the fuel economy in DI-CNG. Figure 4 (a) supports this claim

NOMENCLATURE

$IMEP_{avg}$	=	Average indicated mean effective pressure calculated for number of a cycle (kPa)
P	=	Cylinder gas pressure (kPa)
V	=	Cylinder volume (m^3)
m	=	Mass of the cylinder gases (kg)
σ_{imep}	=	Standard deviation in indicated mean effective pressure (kPa)
COV_{imep}	=	Coefficient of variation in indicated mean effective pressure
dQ_{ch}	=	Chemical energy released by combustion (J)
dQ_{ht}	=	Heat transfer to chamber wall (J)
dU_s	=	Change in sensible energy (J)

dW	=	Piston work (J)
T	=	Mean gas temperature (K)
T_w	=	Mean wall temperature (K)
C_v	=	Specific heat at constant volume (J kg k^{-1})
C_p	=	Specific heat at constant pressure (J kg k^{-1})
γ	=	Specific heat ratio
R	=	Specific gas constant (J kg k^{-1})
$d\theta$	=	Change in crank angle (CA deg)

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