A Study on Performance, Combustion & Emission Characteristics of Simulated Biogas Engine

HUYNH THANH CONG[1*], NGUYEN QUOC KHANH[2]
[1]VNU-HCM Key lab for Internal Combustion Engine, Ho Chi Minh City University of Technology, VNU-HCM
[2]Faculty of Transportation Engineering, Ho Chi Minh City University of Technology, VNU-HCM
Email: htcong@hcmut.edu.vn

Abstract: With the purpose to use biogas in an internal combustion engine with high compression ratio and in order to get a high output performance, this investigation used a stationary diesel engine model with maximum output power 10.3kW to simulating in AVL Boost Software. In this model, the engine was converted to spark ignition mode to use it with simulated biogas. The change of CH_4 between 50% and 80% with 5% step. The compression ratio was at 18:1. This simulation using Vibe 2-zone model combustion, in full load conditions the start of combustion at 7 degrees BTDC and the air ratio$\lambda = 1$. After simulation, the maximum output power of the engine decreased by 7.7% when using 80%CH_4 – 20%CO_2 and 63.1% when using 50%CH_4 – 50%CO_2. Emissions of nitrogen oxides, carbon monoxide decrease when using simulated biogas.

Keywords: *High compression ratio SI engines, simulated biogas, CNG, combustible mixture, Engine performance, Biogas enrichment with methane.*

Introduction:

Biogas is the product of anaerobic digestion of waste, whether occurring spontaneously in landfills or under controlled conditions in digesters. Due to the requirements for renewable energy production, reuse of materials and reduction of harmful emission, the using of biogas technology is increasing non-stop [1]. Biogas has probably never been as interesting as it is now than in any other time. With the drastic increase in the need of renewable energy sources, it has recently gained great importance. Using biogas can improve the environment and protect human health, it is a great way to reuse waste-human, animal, agricultural, industrial [2, 3].

Biogas not only using in engine but also combines biogas with another fuel in many researches. An investigation on utilization of biogas and Karanja oil biodiesel has conducted in India. Experiments were performed on a single cylinder DI diesel engine by using biogas as a primary fuel, Karanja oil biodiesel (KOBD) and diesel oil as secondary fuels in dual fuel operation. The engine performance parameters like Brake Power, Brake Thermal Efficiency and Exhaust Gas Temperature gradually increases with increase in engine load for all test conditions using both pilot fuels diesel and KOBD. CO_2, CO and NO_x emissions increase with increase in engine load for both single and dual fuel mode operation using both pilot fuels [4]. Researches on the effect of concentration of methane in biogas when using spark ignition engine always interesting. In experiments about reduction in the concentration of CO_2 in biogas by Porpatham [5], the CO_2 levels from 41% in biogas to 30% and 20%. Compression ratio 13:1 and constant speed 1500rpm. When reducing CO_2 level, the performance was improved, HC emission reduced with lean mixtures.

According to Semin [6], compressed natural gas (CNG) has long been used in stationary engines. For spark ignition engines there are two options: a bio-fuel conversion and use a dedicated to CNG engine. For compression ignition engines converted to run on natural gas, there are two main options discussed: there are dual-fuel engines and normal ignition can be initiated. The CNG engine research and development fueled using CNG are highlighted to keep the output power, torque and emissions of natural gas engines comparable to their gasoline or diesel counterparts. Biogas composed of various gases such as CH_4 (50 – 70%), CO_2 (30 – 50%), N_2 (<1%) and H_2S (10 – 2000 ppm) [2]. So investigating on Performance evaluation of a constant speed IC engine on CNG, methane enriched biogas and raw biogas are always attractive [7]. A diesel engine was converted into spark ignition mode, compression ratio 12.65 and ignition advance of TDC was evaluated at 30^0, 35^0 and 40^0. The maximum brake power produced by the engine was found at ignition advance of 35^0 TDC for all the tested fuels. In comparison to diesel as original fuel, the power deteriorations of the engine was observed to be 31.8%, 35.6% and 46.3% on compressed natural gas, methane enriched biogas and raw biogas.

Almost the experiments often using biogas in a low compression ratio engine range of 12:1–15:1; to increase performance, the biogas has to use in higher compression ratio engine, up to 18:1 or 19:1. In a research of Montoya [8], a diesel engine (compression ratio 15.5:1) was converted to spark ignition mode to use three fuels: Simulated biogas, biogas enriched with 25% and 50% methane by volume. The optimum condition of experiment for the engine without knocking was using biogas enriched with 50% methane, with 12 degrees of spark timing advance and equivalence ratio of 0.95. In a research by Bora [9], Experiments have been conducted at various compression ratios (18, 17.5, 17 and 16) under dual fuel mode (biogas-diesel). At 100% load, the brake thermal efficiencies of the dual fuel mode are found to be 20.04%, 18.25%, 17.07% and 16.42% at compression ratios of 18, 17.5, 17 and 16. There is a reduction in carbon monoxide as well as hydrocarbon emission by 26.22% and 41.97% when compression ratio was increased from 16 to 18. In all the test cases, carbon monoxide and hydrocarbon emissions under dual fuel mode are found to be more than the diesel mode due to the reduction of volumetric efficiency of the former. In another research with a high compression ratio engine (19.5:1) to evaluate the impact of the carbon dioxide concentration in biogas on the operating characteristics and exhaust gas emissions of a diesel engine running on a mixture of biogas and mineral diesel fuel [10].

The tests were carried out in two stages. In the first stage, the impact of different biogas compositions and the exhaust gas recirculation system (EGR) on the engine parameters was determined. The NOx concentration decrease was directly proportional to the concentration of methane in the common fuel mixture. In the second stage, the gas with the highest methane content was used to determine the impact of the start of injection timing on the engine operating parameters. As the methane content in the common fuel mixture increased, the start of injection timing had to be progressively advanced to increase the thermal efficiency and to lower the fuel consumption, the CO and HC concentrations and the smokiness of the exhaust; however, advancing the start of injection timing increased NOx pollution.

Porpatham not only researches on effect of compression ratio between 13:1 and 15:1 on the performance and combustion but also working with the equivalence ratio between 1.08 and 0.95 [11]. The engine was operated at 1500 rpm at throttle opening of 25% and 100%. The spark timing was set to MBT. The peak pressure is higher with higher compression ratio. The MBT spark timing is retarded with in-crease in compression ratio. HC and CO emissions were low but the NOx values were high. Power and thermal efficiency reduced for leaner mixtures.

In this paper, a stationary diesel engine was converted into spark ignition using simulated biogas and simulating in AVL Boost software. Simulated biogas is formed by the mixing of methane gas (CH_4) and carbon dioxide (CO_2) under different mixing ratio between 50 – 50 and 80 – 20 in percentages. The experiment will evaluate performance, combustion and emission when using simulated biogas and diesel are compared.

Theoretical Study:

A. Simulation setup:

The simulation were conducted on single cylinder make direct injection four stoke cycle diesel engine Kubota RT-140. For basic technical characteristics of the engine, see Table 1.

Table 1. Parameters of converted engine

Parameter	Value
Displacement (cm^3)	709
Number of cylinders	1
Compression ratio	18:1
Power (kW/rpm)	10.3/2400
Torque (N.m/rpm)	49/1600
Bore (mm)	97
Stroke (mm)	96

AVL Boost simulation Model:

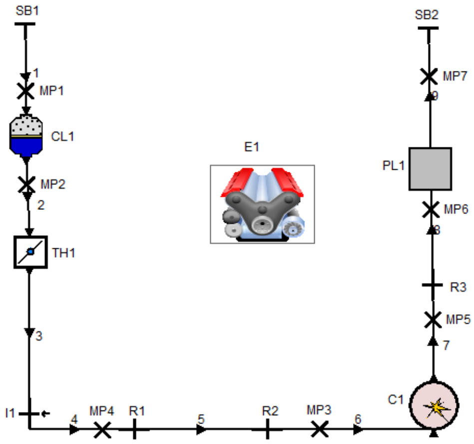

Figure 1: Layout of engine model

The engine simulation model is given in figure 1. The engine converted using a throttle valve to control the air and an injector to control simulated biogas. The model simulation kept the general specifications of the origin diesel engine. The simulation was conducted from 1200rpm to 2400rpm at 100% throttle valve opening to evaluated performance, combustion and emission characteristics. The base combustion model is Vibe 2-Zone with start of combustion at 8^0 BTDC and combustion duration 70^0 [8]. The specifications about simulated biogas fuel are shown in Table 2.

Table 2. Characteristics of simulated biogas

CH_4/CO_2 (%)	Lower Heating Value kJ/kg	A/F Stoi
80 – 20	29678.90	10.2
75 – 25	26143.42	9
70 – 30	23004.65	7.87
65 – 35	20205.70	6.9
60 – 40	17694.23	6.09
55 – 45	15428.11	5.3
50 – 50	13373.05	4.6

Combustion model:

For the current study Vibe two zone models was selected for the combustion analysis. This model divides the combustion chamber into unburned and burned gas regions [12]. However the assumption that burned and unburned charges have the same temperature is dropped. Instead, the first law of thermodynamics is applied to the burned charge and unburned charge, respectively.

$$\frac{dm_b u_b}{d\alpha} = -p_c \frac{dV_b}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_{Wb}}{d\alpha} + h_u \frac{dm_b}{d\alpha} - h_{BB,b} \frac{dm_{BB,b}}{d\alpha}. \quad (1)$$

$$\frac{dm_u u_u}{d\alpha} = -p_c \frac{dV_u}{d\alpha} - \sum \frac{dQ_{Wu}}{d\alpha} - h_u \frac{dm_B}{d\alpha} - h_{BB,u} \frac{dm_{BB,u}}{d\alpha}. \quad (2)$$

Where,

dm u: is the change of the internal energy in the cylinder,

$p_c \frac{dV}{d\alpha}$: is the piston work,

$\frac{dQ_F}{d\alpha}$: is the fuel heat input,

$\frac{dQ_W}{d\alpha}$: is wall heat loses,

$h_u \frac{dm_b}{d\alpha}$: is the enthalpy flow from the unburned to the burned zone due to the conversion of a fresh charge to combustion products. Heat flux between the two zones is neglected.

$h_{BB} \frac{dm_{BB}}{d_\alpha}$: is the enthalpy due to blow by,

u and b in the subscript are unburned and burned gas.

In addition the sum of the volume changes must be equal to the cylinder volume change and the sum of the zone volumes must be equal to the cylinder volume.

$$\frac{dV_b}{d\alpha} + \frac{dV_u}{d\alpha} = \frac{dV}{d\alpha}. \quad (3) \qquad V_u + V_b = V \quad (4)$$

Engine performance review:

Engine power (kW) is estimated via AVL BOOST as follows:

$$N_e = \frac{1}{2}\eta_f \cdot \eta_v \cdot N \cdot V_d \cdot LHV \cdot \rho_a \cdot \frac{\dot{m}_f}{\dot{m}_a} \quad (5)$$

Where,

η_f, η_v(%): fuel conversion efficiency and volumetric efficiency;

N: engine speed (rpm);

LHV: lower heating value (kJ/kg);

ρ_a (kg/m3): density of air;

$\dot{m}_f / \dot{m}_a$: Air-fuel ratio measured by engine test bed;

Brake torque (N.m)

Brake specific fuel consumption (g_e, g/kWh) of engine are calculated as:

$$M_e = \frac{N_e}{2\pi N} \quad (6) \quad ; \qquad g_e = \frac{\dot{m}_f}{N_e} \quad (7)$$

Emission model:

CO Formation Model:

The CO formation following two reactions given in Table 3.

Table 3. CO formation reactions [13]

	Stoichiometry	Rate
R1	$CO+OH=CO_2+H$	$r_1 = 6,76.10^{10}.e^{\left(\frac{T}{1102,0}\right)}c_{CO}.c_{OH}$
R2	$CO+O_2=CO_2+O$	$r_2 = 2,51.10^{12}.e^{\left(\frac{-24055,0}{T}\right)}.c_{CO}.c_{O_2}$

The final rate of CO production/destruction in [mole/cm3s] is calculated as follow:

$$r_{CO} = C_{const}.(r_1 + r_2).(1-\alpha) \qquad \left(\alpha = \frac{c_{co.act}}{c_{co.equ}}\right)$$

(8)

NOx Formation model:

The following 6 reactions of the NOx formation model (based on the well-known Zeldovich mechanism) shown in Table 4:

Table 4. Nox formation reactions [13]

	Stoichiometry	Rate $k_i = k_{0,i} \cdot T^a \cdot e^{\left(\frac{-TA_i}{T}\right)}$	k_0 [cm³,mol,s]	a [-]	T_A [K]
R1	N2 + O = NO + N	$r_1 = k_1 \cdot c_{N2} \cdot c_O$	4.93E13	0.0472	38048.01
R2	O2 +N = NO + O	$r_2 = k_2 \cdot c_{O2} \cdot c_N$	1.48E08	1.5	2859.01
R3	N +OH = NO + H	$r_3 = k_3 \cdot c_{OH} \cdot c_N$	4.22E13	0.0	0.0
R4	N2O + O = NO + NO	$r_4 = k_4 \cdot c_{N2O} \cdot c_O$	4.58E13	0.0	12130.6
R5	O2 + N2 = N2O + O	$r_5 = k_5 \cdot c_{O2} \cdot c_{N2}$	2.25E10	0.825	50569.7
R6	OH + N2 = N2O + H	$r_6 = k_2 \cdot c_{OH} \cdot c_{N2}$	9.14E07	1.148	36190.66

All reaction rates r_i have units (mole/cm3s), the concentration c_i are molar concentrations under equilibrium conditions with units (mole/cm3s). The concentration of N2O is calculated according to:

$$c_{N_2O} = 1.1802 \cdot 10^{-6} \cdot T^{0.6125} \cdot e^{\left(\frac{9471.6}{T}\right)} \cdot c_{N_2} \cdot \sqrt{p_{O_2}} \quad (9)$$

The final rate of NO production/destruction in [mole/cm3s] is calculated as:

$$r_{NO} = C_{PostProcMult} \cdot C_{KineticMult} \cdot 2.0 \cdot \left(1-\alpha^2\right) \cdot \left(\frac{r_1}{1+\alpha \cdot AK_2} + \frac{r_4}{1+AK_4}\right) \quad (10)$$

With:

$$\alpha = \frac{c_{NO,act}}{c_{NO,equ}} \cdot \frac{1}{C_{KineticMult}}$$

$$AK_2 = \frac{r_1}{r_2 + r_3}$$

$$AK_4 = \frac{r_4}{r_5 + r_6}$$

HC Formation model:

The so-called mechanisms of HC development are in total very complex, and a quantitative calculation of HC emission in the SI engine is thus not yet feasible. For the estimation of the oxidation speed of the amount of HC originating in the cylinder, the following global relation is often recommended.

$$\frac{d[HC]}{dt} = -c_R A[HC]\left[O_2\right] e^{-\frac{E}{RT}} \quad (11)$$

With: E=156 (J/mol), A=6,7.10^{21} (m^3/mol), c_R is correction factor.

Results and Discussions:

The variation of brake power with engine speed is shown in figure 2 at full throttle opening. When simulating the operation of engine with simulated biogas from 80/20% to 50/50% at full load, the brake power output decreases significantly. When using fuel 80/20% the brake power get 9.5 kW at 2400rpm and 3.78kW when using fuel 50/50%. The maximum power when using diesel is 10.3kW at 2400rpm. So the power decreases 7.7% and 63.3% when using simulated biogas 80/20% and 50/50% due to the decrease of the lower heating value in mixture biogas. 2400rpm is the top speed of this engine when using diesel fuel. But in this simulation, this engine using simulated biogas and converted to SI engine, the speed of the engine in simulation fixed at 2400rpm. In fact, may be the speed increases more than 2400rpm and the power output may be increase higher.

The Torque of the engine when using simulated biogas also presented in figure 3. The maximum of Torque got at 1600rpm with every simulated fuel, the highest torque when using 80/20% is 39.7N.m and 50/50% is 17N.m. And the maximum torque of the engine when using diesel is 48.9N.m. The decrease of Torque compare between diesel and biogas 50/50% is 65.2%.

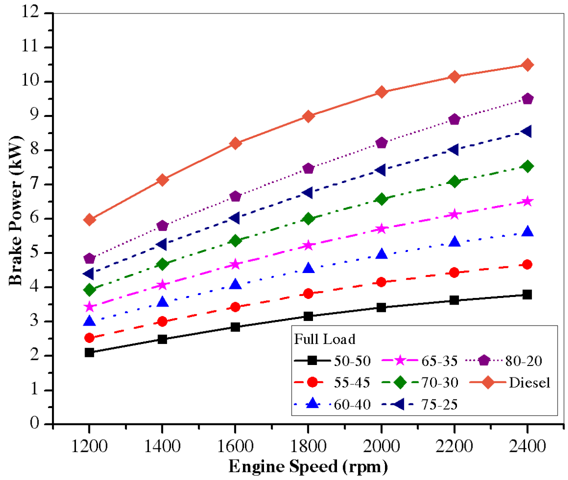

Figure 2: Variation of brake power with engine speed at full throttle.

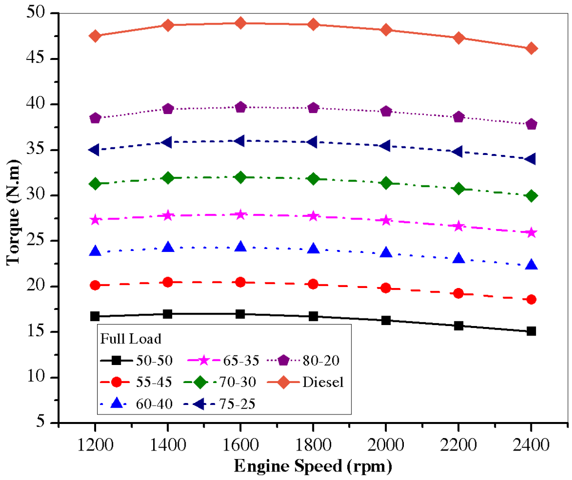

Figure 3: Variation of Torque with engine speed at full throttle.

The Brake Specific Fuel Consumption (BSFC) of the engine when using diesel and simulated biogas are shown in figure 4. Because CH_4 decrease and CO_2 increase in simulated biogas fuel (80% to 50% CH_4, 20% to 50% CO_2), so when CO_2 increase the BSFC also directly proportional increase. The BSFC when using simulated biogas always higher than using diesel. The maximum of BSFC is 1015 (g/kW.h) when using fuel 50/50% because the lower heating value is lowest 13373.05(kJ/kg) so the engine need more fuel to operate.

The Mass Fraction Burned Rate (MFBR) at 1800rpm is shown in figure 5. The MFBR of fuel 50/50% is the best. In general, the fuel with the lowest level of methane begins combustion earlier. When the CH_4 increase from 50% to 80% the MFBR also get worse.

Figure 6 shows the Pressure and Temperature inside the engine cylinder as function of the crankshaft angle, 100% throttle. The highest peak pressure when using diesel is 67.5 bars at 1800rpm. And the peak pressure in cylinder when using simulated biogas focussed around 60 bars. The figure shows that peak pressure increases significantly with increase in methane and the highest peak pressure occurs for 80%CH4-20%CO_2. For pure biogas, the low heating value leading to a lower peak pressure.

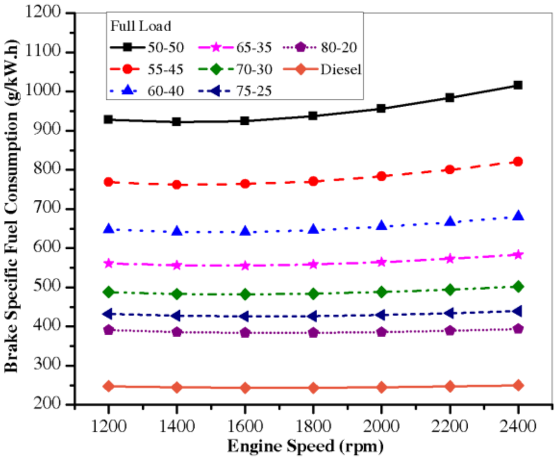

Figure 4: The Brake Specific Fuel Consumption

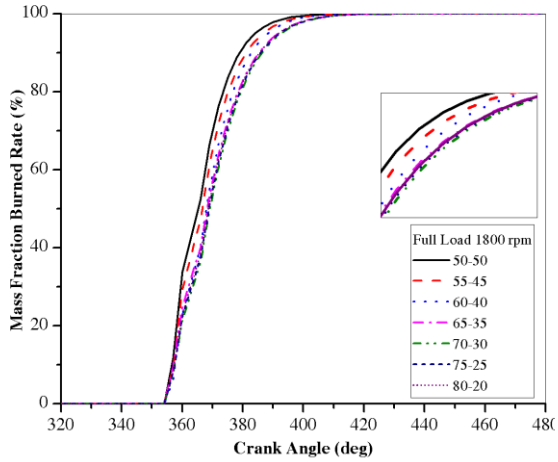

Figure 5: Mass Fraction Burned Rate (%) at 1800rpm

When the CH_4 increase in the mixture, it also increase oxy and decrease CO_2 in the mixture, the combustion better and flame speed faster, that the reason why the temperature increase. The highest temperature got at using fuel 80-20%.

Figure 6 also indicated the profiles of the Heat Release Rate, which take in account the total heat release and the heat lost to the walls for the simulated biogas fuels. The faster and larger net heat release occurs because the mixture is more reactive with a larger heating value and flame speed, leading to higher temperature and pressure profiles. In case of the CH_4 increase from 50% to 80% the Heat release pattern is steeper and close to TDC.

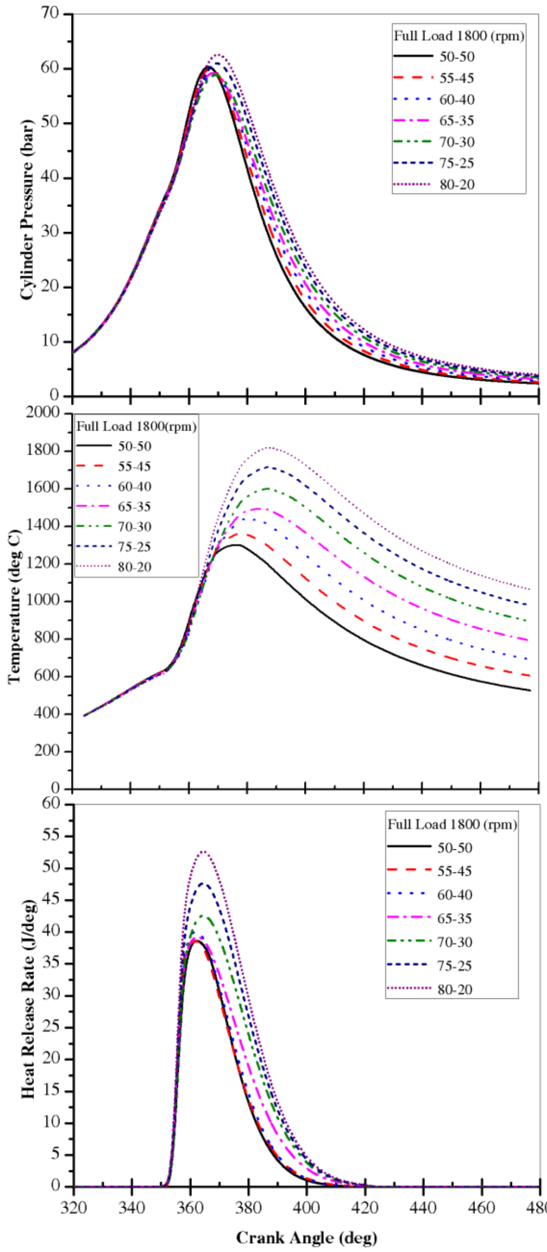

Figure 6: Cylinder Pressure, Temperature and Heat Release Rate at 1800rpm

Figure 7 shows HC, NOx and CO emissions. HC concentration in the exhaust rises with an increase in the CO2 level in the fuel. The highest NOx emission at 80/20% simulated biogas. A decrease in the CO2 fraction leads to increased methane and oxygen concentrations, and thus faster combustion and higher temperatures. It is also the reason why CO level emission increase.

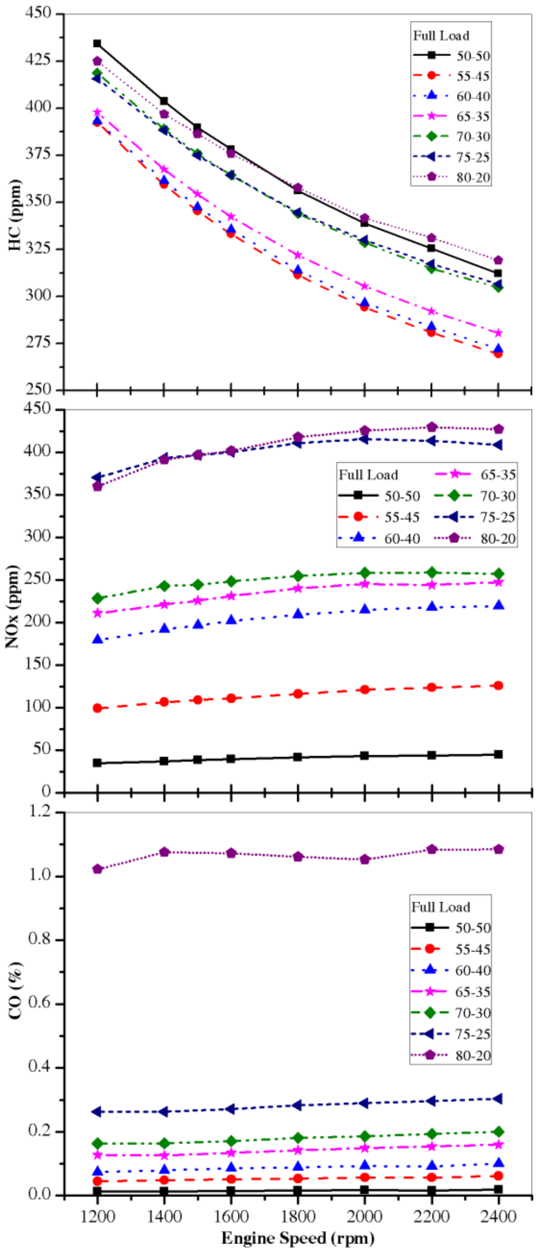

Figure 7: Hydro carbon Emission, Nitrogen Oxide and Carbon Monoxide emissions at full load

Conclusion:

The present demonstrates the influences of simulated biogas on SI engine (converted from CI engine) performance, combustion and emissions characteristics. Simulated biogas fuel show lower brake power and brake torque and higher BSFC than diesel.

When methane percentage increases, the combustion of engine also got better, the peak pressure and heat release rate got closer to TDC. The result shows the significant increases in NO_x, HC and CO with the increase of percentage of methane.

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