

A SIMULATION STUDY ON PERFORMANCE CHARACTERISTICS OF SIMULATED BIOGAS ENGINE WITH HIGH COMPRESSION RATIO NGHIÊN CỨU ĐÁNH GIÁ ĐẶC TÍNH ĐỘNG CƠ BIOGAS TỈ SỐ NÉN CAO

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ABSTRACT

This paper presents a simulation study on engine performance characteristics of a high compression ratio biogas engine for high efficiency and low emission. A small diesel-type engine with a power of 14 horse power (HP) was modelled for simulating analysis with help of AVL Boost (a thermodynamic analyzing tool). Biogas includes CH₄ and CO₂ mainly. The ratio of CH₄-CO₂ was changed from 50%-50% (in volume) to 80%-20% with a step of 5%-5%. Original engine compression ratio is 18:1. Engine operating conditions were set up at: full-load, spark timing of 7 degrees before top dead center (bTDC), and lambda $\lambda=1$. In obtained results, in comparison of diesel operation, maximum engine output power was reduced but engine exhaust emission (CO, NO_x) was improved for operation with biogas that is significantly result in term of ecological protection. For example, in comparison of diesel operation, biogas operation could conduct a reduction of about 7.7% and 63% for ratio of 80%-20% and 50%-50%, respectively.

Keywords: High compression ratio SI engines; ratio of CH₄-CO₂; power; exhaust emission.

TÓM TẮT

Bài báo này trình bày một nghiên cứu mô phỏng đánh giá đặc tính động cơ một 01 xi-lanh tỉ số nén cao sử dụng nhiên liệu khí sinh học nhằm đạt hiệu suất cao và ô nhiễm thấp. Một động cơ diesel 01 xi-lanh với tỉ số nén 18:1 được mô hình hóa và mô phỏng dựa trên phần mềm phân tích nhiệt động AVL Boost. Khí sinh học bao gồm 2 thành phần chính là CH₄ và CO₂. Tỉ lệ CH₄-CO₂ được thay đổi về mặt thể tích từ 80%-20% đến tỉ lệ 50%-50% với bước thay đổi 5%-5%. Điều kiện mô phỏng bao gồm: tốc độ thay đổi (từ 1200 v/ph đến 2400 v/ph) trường hợp toàn tải toàn tải, đánh lửa tại 7° trước điểm chết trên và lamnda = 1.0. Kết quả mô phỏng cho thấy, khi so sánh với công suất cực đại động cơ nguyên thủy dùng diesel, khi vận hành với khí sinh học, công suất cực đại có giảm nhưng với nhiên liệu mới này, sản phẩm cháy có sự giảm các khí độc hại, góp phần bảo vệ môi trường. Cụ thể, công suất tối đa giảm khoảng 7.7% và 63% tương ứng với trường hợp sử dụng tỉ lệ 80%-20% và 50%-50%.

Từ khóa: động cơ đánh lửa cưỡng bức tỉ số nén cao; tỉ lệ CH₄-CO₂; công suất; khí thải.

1. INTRODUCTION

Biogas is the product of anaerobic digestion of waste, whether occurring spontaneously in landfills or under controlled conditions in digesters [1]. With the drastic increase in the need of renewable energy sources, it has recently gained great importance. Using biogas can improve the

environment issue and protect human health. It is also a great way to re-use waste-human, agricultural, industrial [2-10].

Effect of fuel properties: Biogas is used as individual fuel or combined with other fuels as dual type operation in engines. An investigation on utilization of biogas and Karanja oil biodiesel has conducted in India

[4]. Experiments were performed on a single cylinder DI diesel engine by using biogas as a primary fuel, Karanja oil biodiesel (KOB) and diesel oil as secondary fuels in dual fuel operation. Engine performance parameters such as: brake power (BP), brake thermal efficiency (BTE) and exhaust gas temperature gradually increases with increase in engine load for all test conditions using both pilot fuels diesel and KOB. 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. Researches on the effect of concentration of methane in biogas when using spark ignition engine always interesting by Porpatham [5, 13]. His report showed that HC and CO_2 could be reduced and the performance was improved. 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 12.65:1 and ignition advance of TDC was evaluated at 30° , 35° and 40° . The maximum brake power produced by the engine was found at ignition advance of 35°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.

Effect of compression ratio: 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 (18:1, 17.5:1, 17:1, and 16:1) under dual fuel mode (biogas-diesel). At full load conditions, the BTE of the dual fuel mode are found to be 20.04%, 18.25%, 17.07% and 16.42% at compression ratios of 18:1, 17.5:1, 17:1 and 16:1. There is a reduction in carbon monoxide as well as hydrocarbon emission by 26.22% and 41.97% when compression ratio was increased from 16:1 to 18:1. In all the test cases, carbon monoxide and hydro-carbon 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 CR 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 NO_x 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 NO_x pollution.

The main objective of this work is to predict the engine performance characteristics (power, combustion, emission) with change of engine speed and ratio of CH₄-CO₂ at full load conditions. The obtained results could help to analyze the biogas fueled spark ignition engine with a proper compression ratio for high efficiency and low emission.

2. THEORETICAL STUDY

2.1 Simulation setup

A single cylinder make direct injection four stroke cycle diesel engine (Kubota RT-140) was used to model under AVL Boost ver.2013. The specification of engine shown in **Table 1**. The engine model is shown in **Fig. 1**.

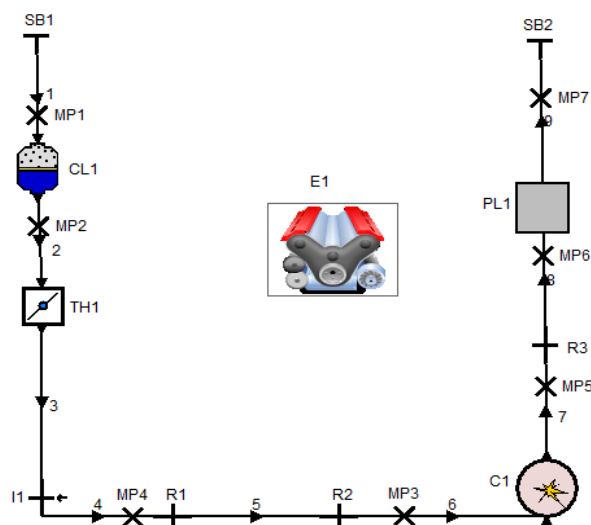


Figure 1. Simulated engine model

Table 1. Specification of 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

Engine simulation was modelled and shown in Fig.1. Original engine was 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 1200 rpm to 2400 rpm 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°bTDC and combustion duration of 70°[8]. The specifications about simulated biogas fuel are shown in Table.

Table 2. Characteristics of simulated biogas

CH ₄ -CO ₂ (%)	LHV (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

2.2 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. The state of first law of thermodynamic for this case is shown as follow:

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

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

where, d_{mu} : is the change of the internal energy in the cylinder,

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

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

$\frac{dQ_w}{d\alpha}$: 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.

$$\frac{dV_b}{d\alpha} + \frac{dV_u}{d\alpha} = \frac{dV}{d\alpha} \quad (3)$$

$$V_u + V_b = V \quad (4)$$

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:

$$M_e = \frac{N_e}{2\pi N} \quad (6)$$

$$g_e = \frac{\dot{m}_f}{N_e} \quad (7)$$

2.3 Emission model

CO Formation Model

CO formation following two reactions given in Table 3.

Table 3. CO formation reactions [13]

	Stoichiometry	Rate
R ₁	CO+OH=CO ₂ +H	$r_1 = 6,76 \cdot 10^{10} \cdot e^{\left(\frac{T}{1102,0}\right)} \cdot c_{CO} \cdot c_{OH}$
R ₂	CO+O ₂ =CO ₂ +O	$r_2 = 2,51 \cdot 10^{12} \cdot e^{\left(\frac{-24055,0}{T}\right)} \cdot c_{CO} \cdot c_{O_2}$

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

$$r_{CO} = C_{const} \cdot (r_1 + r_2) \cdot (1 - \alpha) \quad \left(\alpha = \frac{c_{CO,act}}{c_{CO,equ}} \right) \quad (8)$$

NO_x Formation model

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

Table 4. NO_x formation reactions [13]

	Stoichiometry	Rate	k_0 [cm ³ ,mol,s]	a [-]	T _A [K]
		$k_i = k_{0,i} \cdot T^a \cdot e^{\left(\frac{-E_{A,i}}{T}\right)}$			
R1	N ₂ + O = NO + N	$r_1 = k_1 \cdot c_{N_2} \cdot c_O$	4.93E13	0.0472	38048.01
R2	O ₂ + N = NO + O	$r_2 = k_2 \cdot c_{O_2} \cdot c_N$	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	N ₂ O + O = NO + NO	$r_4 = k_4 \cdot c_{N_2O} \cdot c_O$	4.58E13	0.0	12130.6
R5	O ₂ + N ₂ = N ₂ O + O	$r_5 = k_5 \cdot c_{O_2} \cdot c_{N_2}$	2.25E10	0.825	50569.7
R6	OH + N ₂ = N ₂ O + H	$r_6 = k_6 \cdot c_{OH} \cdot c_{N_2}$	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:

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

with:

$$\alpha = \frac{c_{NO,act}}{c_{NO,eq}} \cdot \frac{1}{C_{KineticMult}} ; \quad AK_2 = \frac{r_1}{r_2 + r_3} ;$$

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

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.

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

With: E=156 (J/mol), A=6,7.10²¹ (m³/mol), c_R is correction factor.

3. RESULTS AND DISCUSSION

The variation of brake power with engine speed is shown in Fig. 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 2400 rpm and 8.86 kW when using fuel 50%-50%. The maximum power when using diesel is 10.3 kW at 2400 rpm. So the power decreases 7.7% and 13.9% when using simulated biogas 80%-20% and 50%-50% due to the decrease of the lower heating value in mixture biogas. 2400 rpm 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 2400 rpm. In fact, may be the speed increases more than 2400 rpm and the power output may be increase higher.

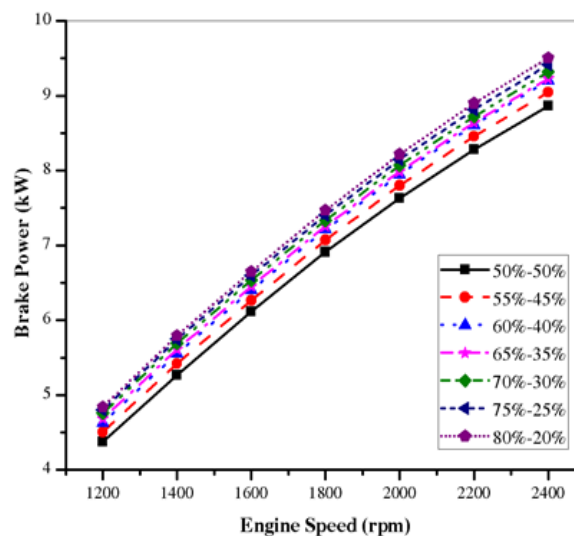


Figure 2. Brake power with engine speed

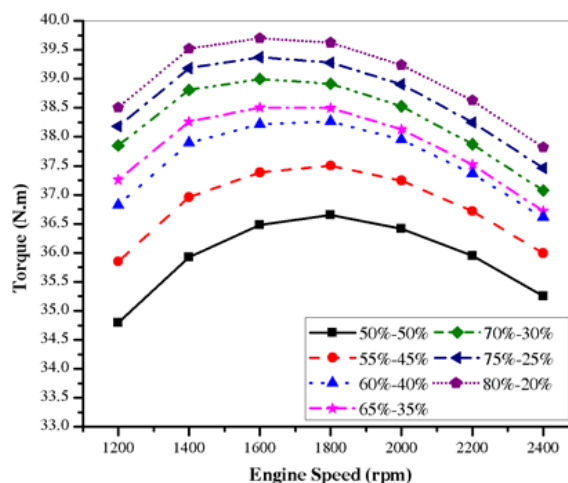


Figure 3. Brake torque with engine speed

Brake torque of the engine when using simulated biogas also presented in figure 3. The maximum of Torque got at 1600 rpm with every simulated fuel, the highest torque when using 80%-20% is 39.7 N.m and 50%-50% is 36.6 N.m. Maximum torque of the engine when using diesel is 48.9 N.m. The decrease of torque for comparing between diesel and biogas 50%-50% is 25%.

Brake Specific Fuel Consumption (BSFC) of the engine when using diesel and simulated biogas are shown in Fig.4. Because CH₄ decrease and CO₂ in-crease in simulated

biogas fuel (80% to 50% CH₄, 20% to 50% CO₂), so when CO₂ increase the BSFC also directly proportional increase. The BSFC when using simulated biogas always higher than using diesel. Maximum BSFC is 904 (g/kW.h) when using fuel 50%-50% because the lower heating value is lowest 13.37(MJ/kg) so the engine need more fuel to operate.

Operation at a ratio of 80%-20% has given the highest TE that is 31.61%. It is because of the lowest BSFC and the higher heating value 29.688(MJ/kg) (Fig.5). The Mass Fraction Burned Rate (MFBR) at 1800rpm is shown in figure 6. The MFBR of fuel 50%-50% is the best. In general, the fuel with the lowest level of methane begins combustion earlier. As CH₄ increase from 50% to 80% MFBR get worse.

Fig. 7 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 bar at 1800 rpm. 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% -20%. For pure biogas, the low heating value leading to a lower peak pressure.

Increase of CH₄ increases oxygen and decreases CO₂ in the mixture, the combustion process becomes better state that promote the flame speed faster. That is a reason why the temperature increases. The highest temperature got at using fuel 80-20%. Fig. 7 also indicated the profiles of the HRR, 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 CH₄ increase from 50% to 80% the Heat release pattern is steeper and close to TDC.

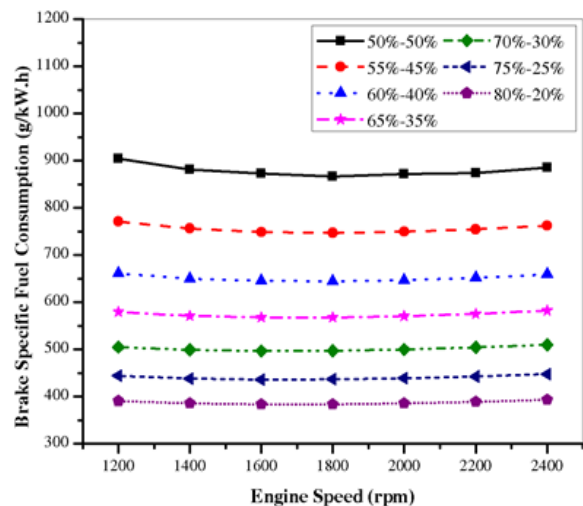


Figure 4. Brake specific fuel consumption

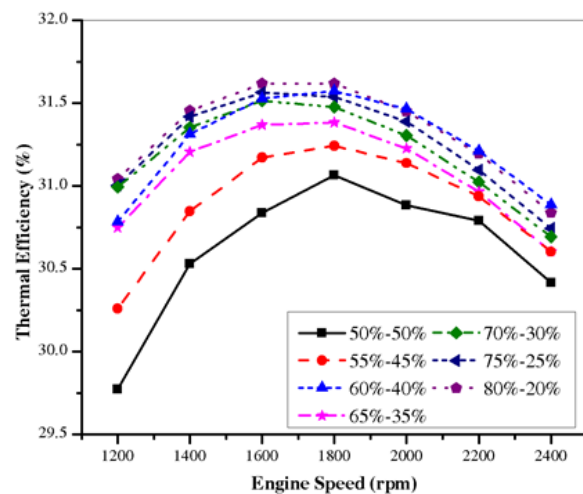


Figure 5. Thermal Efficiency

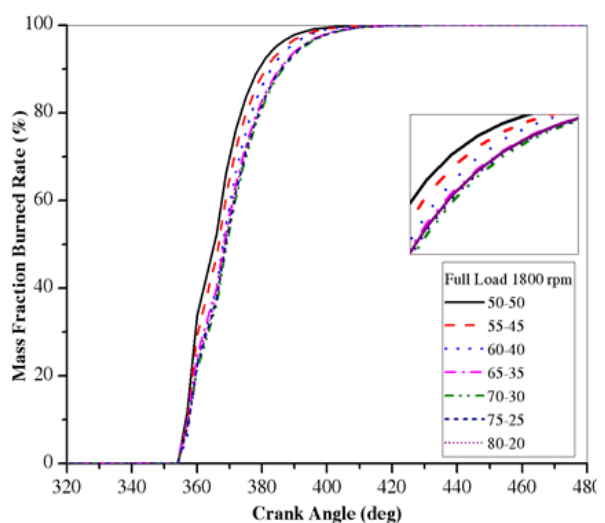


Figure 6. Mass fraction burned rate

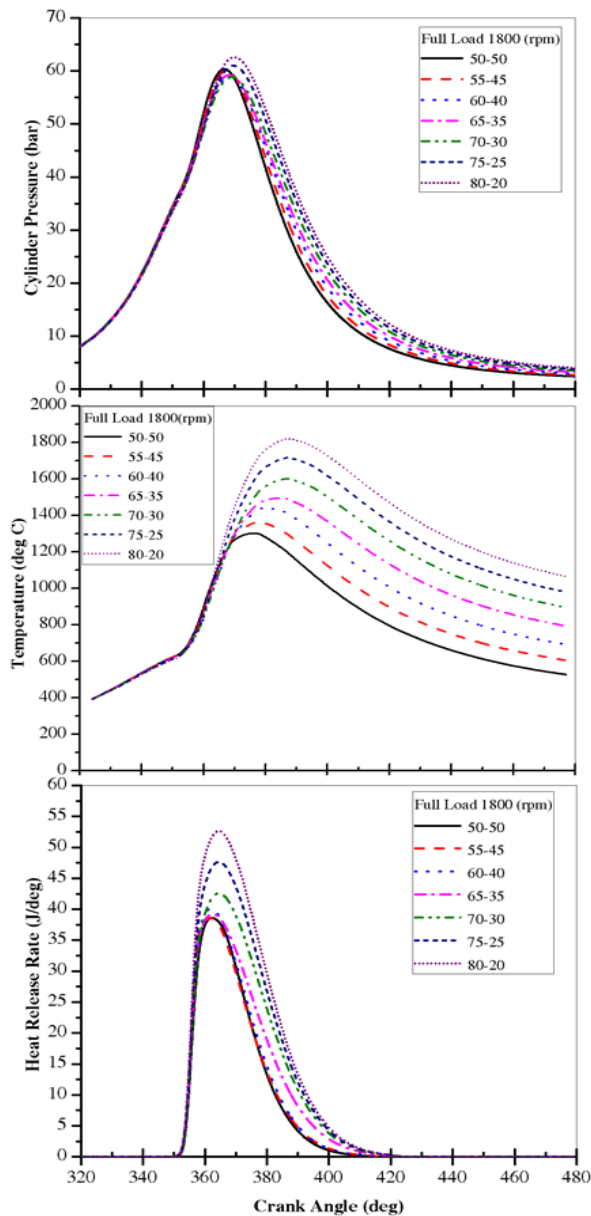


Figure 7. Cylinder pressure, temperature and heat release rate at 1800 rpm

Fig. 8 shows HC, NO_x and CO emissions. HC concentration in the exhaust rises with an increase in the CO₂ level in the fuel. The highest NO_x 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.

4. CONCLUSION

Engine performance characteristics were studied to predict the change of power,

cylinder pressure and temperature, exhaust emission with change of engine speed and ratio of CH₄-CO₂ by simulation.

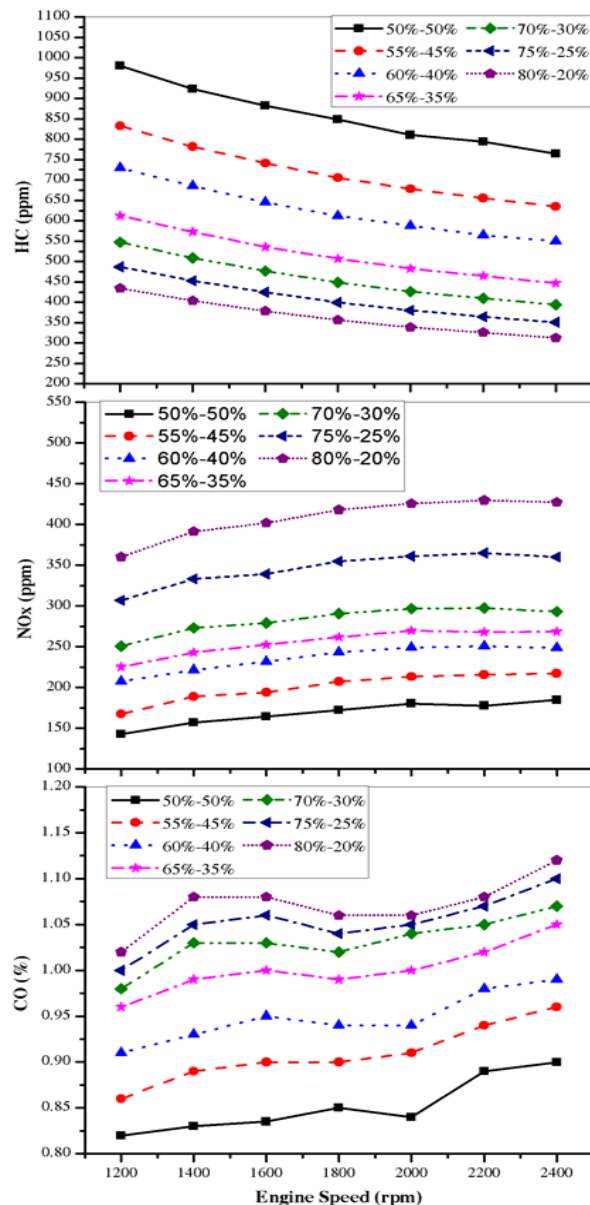


Figure 8. Typical exhaust emission

The obtained results could help to analyse the trend of biogas fuelled high CR engine with high efficiency and low emission. Power was reduced with decrease of CH₄, but exhaust emissions (such as CO, NO_x) were improved.

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REFERENCES

- [1] Sari L., Argo N., Mats E., (2011) “Overview of Biogas Technology” Baltic Forum for Innovative Technologies for Sustainable Manure Management, December 2011
- [2] Ertugrul D. O., (2013) “Biogas Production from Household Wastes: A Quantitative Feasibility Study for Student Apartments in Albano”, MJ153x Bachelor’s Thesis for Energy and Environment, Stockholm 2013
- [3] NSCA, (2006) “Biogas as a road transport fuel”, The National Society for Clean Air and Environmental Protection, ISBN 0-903-47461-1, England, June 2006
- [4] Bhabani P. P., (2013) “Investigation on utilization of biogas & Karanja oil biodiesel in dual fuel mode in a single cylinder DI diesel engine”, International Journal of Energy and Environment Volume 4, 2013 pp.279-290
- [5] Porpatham E., (2008) “Investigation on the effect of concentration of methane in biogas when used as a fuel for a spark ignition engine”, Fuel, Fuel 87 (2008) 1651 - 1659
- [6] Rosli A B., (2008) “A Technical Review of Compressed Natural Gas as an Alternative Fuel for Internal Combustion Engines”, American J. of Engineering and Applied Sciences 302 -311, ISSN 1941 – 7020, 2008
- [7] Chandra R., (2011) “Performance evaluation of a constant speed IC engine on CNG, methane enriched biogas and biogas”, Applied energy 88 (2011) 3969 – 3977
- [8] Juan P. G. M., (2015) “Experimental study of spark ignition engine performance and emissions in a high compression ratio engine using biogas and methane mixtures without knock occurrence, University of Antioquia, Calle 67 No. 53-108. Medellin–Colombia, 2015
- [9] Ujjwal K. S., (2014) “Effect of compression ratio on performance, combustion and emission characteristics of a dual fuel diesel engine run on raw biogas”, Energy Conversion and Management 87 (2014) 1000–1009
- [10] Violeta M., (2013) “Performance and emission characteristics of biogas used in diesel engine operation”, Energy Conversion and Management 75 (2013) 224–233
- [11] Porpatham E., (2012) “Effect of compression ratio on the performance and combustion of a biogas fuelled spark ignition engine”, Fuel 95 (2012) 247 – 256
- [12] AVL, Boost Theory, AVL Boost Users Guide ver2013.2, AVL Boost.
- [13] Merker P., Christian S., “Simulating Combustion”, Berlin, 2006.

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