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ANALYSIS OF COMBUSTION AND NO_X FORMATION IN A SPARK IGNITION (SI) ENGINE FUELED WITH HYDROGEN-HYDROGEN OXYGEN (HHO) ENRICHED BIOGAS

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Abstract

A numerical simulation was conducted to investigate the combustion characteristics and effects of Hydrogen-Hydrogen Oxygen (HHO) addition on performance and NO_x emission of a spark ignition (SI) biogas engine. At a given operating condition, indicative work cycle Wi increases with increase of HHO, CH_4 concentrations and load rate; but it decreases with engine speed increase and reaches peak value as variation of equivalence ratio and advanced ignition timing. Combustion temperature T and NO_x concentration in exhaust gas also increase with increase of HHO, CH_4 concentrations, load rate and advanced ignition timing; but it decreases with engine speed increase, and reaches peak value as variation of equivalence ratio. The increase rate of NO_x concentration with equivalence ratio is much higher than that of Wi and T. When HHO concentration in biogas is lower than 30%, the gain of Wi is advantageous before the increase of NO_x. The addition HHO is more interesting as biogas engine operates with lean mixture. At a given engine speed, optimal ignition timing reduces 6 crankshaft angle degrees as adding 30% HHO to biogas. A compromise between performance and NO_x emission can be obtained by appropriate adjustment of operating conditions of the engine fueled with HHO enriched biogas.

Key words: alternative fuels, biogas, biogas engines, HHO gas, renewable energy

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1. Introduction

The research on reduction of harmful emissions and energy consumption is becoming more and more important nowadays due to growing energy demands and environment pollution. Alternative renewable energy in place of fossil fuel is, therefore, becoming more realistic. The contribution of various renewable energies depends not only on available primary resources but also on the energy strategy of each country. In tropical area, solar power and biogas have proved to be highly potential renewable energy resources. These two types of energy can be used in a hybrid energy system (Reddy et al., 2016) where biogas is enriched with hydrogen or with Hydrogen-Hydrogen Oxygen (HHO) gas produced by solar energy. Biogas can be produced from poultry manure, food wastes, agriculture organic wastes (Bui et al., 2010). Raw biogas comprises of primary CH_4 that defines its heating value, and impurity CO_2, which reduces the flammability, and thus, the heat release rate. The presence of CO_2 in biogas worsens the combustion process, lowers brake power of engine. However, biogas engine can be designed with high compression ratio which allows an improvement of thermal efficiency thanks to the high-octane number of the fuel (Bui et al., 2013a, 2013b; Bui and Tran, 2015). In view of pollutant emission, Verma et al. (2017) highlighted that spark ignition (SI) biogas engine produces lower NO_x emission but higher

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unburned hydrocarbon and CO emissions as compared to gasoline engine. This is due to the presence of CO_2 in the fuel mixture reducing the flame temperature and this results in a reduction in NO_x formation reaction rate. In biogas diesel dual fuel engines, Bui and Bui (2017) pointed out that the soot formation rate is not quite different from diesel engine, but soot combustion rate is much higher than that of diesel engine resulting in a significant decrease of soot concentration in exhaust gas. Despite the positive effect pollutant emission reduction, CO_2 tends to increase ignition delay periods and reduce flame propagation speeds resulting in a decrease of engine thermal efficiency (Kim et al., 2016). The simulation research results (Bui et al., 2014, 2015a, 2015b) showed that the combustion of biogas engines was worsened, particularly in case of high-speed engines, as CO_2 concentration in biogas increases.

One possible solution of this biogas fuel problem, without having an adverse effect on emission levels, is to add hydrogen or HHO gas to the biogas mixture to improve the combustion properties. Ilbas et al. (2006) in his basic combustion research, showed that increasing the hydrogen concentration in the hydrogen-methane mixture resulted in an increase in burning velocity and a widening of the flammability limit. Porpatham et al. (2007) studied the influence of different equivalence ratios on the performance of spark ignition engine fueled with biogas mixed with 5%, 10% and 15% hydrogen. The study showed that with increasing hydrogen concentration, the burning rate was increased. The simulation research of Bui et al. (2018a) show that as fuels contain higher hydrogen concentration, optimum advance injection angle decreases, resulting in a significant decrease in the soot and NO_x emissions. For the SI engine fueled with gasoline, the addition hydrogen leads to a reduction of CO_2 emission and an improvement of thermal efficiency.

Generally, hydrogen can be considered as an additive to enhance the performance of engine. The employment of biogas-hydrogen mixture has tremendous prospects in view of energy saving and environment protection (Boretti, 2010; Bui et al., 2010). However, the challenge of wide use of hydrogen is associated with the storage of fuel, especially on board of vehicles. The main storage solution nowadays is to compress hydrogen up to 700 bar in cylinders, compared to only 200-bar cylinders employed for natural gases, to ensure the same cruising range. Otherwise the hydrogen generator also proves to be complicated and costly (White et al., 2006). Therefore, using hydrogen in the mixture with oxygen (namely HHO gas) produced onboard of vehicle by batteries or on site of stationary engine by renewable energy, such as solar panels, becomes more interesting. HHO gas is considered a promising alternative fuel and clean energy source (Musmar and Ammar, 2011; Subramanian and Ismail, 2018). It has recently been introduced to the auto industry (Ammar, 2010) and attracted the interest of scientists in around the world (Ammar and Musmar, 2018; Ma and Wang, 2008). The HHO gas is a mixture of H_2 and O_2 in a volume ratio of 2:1. It can be produced by water electrolysis process. HHO gas can be used in combination with traditional fuels in internal combustion engines to improve the overall efficiency and reduce emissions (Changwei and Wang, 2010; Rajasekaran et al., 2015). HHO is produced on demand of engines, not intended for storage. The HHO gas generator operates when engine starts and stops as engine is switched off. As the HHO gas generator is compact, it can be incorporated onboard of vehicles or on site of stationary engines. Many experiments on SI engines have been carried out using HHO gas as a fuel performance enhancer (Arjun et al., 2019). Babariya et al. (2015) had carried out an experimental research on a 4-stroke SI engine with HHO gas as a fuel additive. They concluded that the use of HHO in gasoline engines increased the power output of the engine around 5.7%, the thermal efficiency increased by around 5% leading to emission reduction of CO, HC. Musmar and Ammar (2011) has carried out the test of HHO additive to gasoline on Honda G200 engine and found out that NO_x was reduced to about 50%, CO was reduced to about 20% and a reduction in fuel consumption ranged between 20% and 30% was observed. Rimkus et al. (2018) observed that when HHO gas is added to conventional fuels, the indicated efficiency of the engine changes insignificantly, but the concentrations of CO and HC in the exhaust gas and smoke levels are reduced markedly.

Leelakrishnan and Suriyan (2013) recently studied the effects HHO addition to gasoline on the performance of 4-stroke, 1-cylinder and 5.4-kW SI engines. The results showed that the effective power increased by 5%, the thermal efficiency increased by 7%, fuel consumption decreased by 6%, HC and CO emissions reduced by 90% at full load regime. The results of Ammar (2010) showed that the addition of HHO to the charge mixture reduced the level of fuel consumption, NO_x and CO emissions by 30%, 50% and 20%, respectively.

Mohamed et al. (2016) focused on the effects of addition of HHO gas to petrol on SI engine. They observed that engine thermal efficiency increased up to 10%, consequently reducing fuel consumption up to 34% when HHO gas was introduced into the air/fuel mixture. The concentrations of NO_x, CO and HC were reduced to almost 15%, 18% and 14% respectively on average when HHO was introduced into the system. Sharma et al. (2015) investigated the effect of HHO gas addition on performance characteristics of a 4-stroke multi-cylinder SI engine. They pointed out that the engine brake thermal efficiency was increased by almost 10.26% on average. The engine exhaust gas temperature was reduced by almost 4% on average leading to significant reduction in NO_x emission from the exhaust gas.

The above results are different from the observation of Patel et al. (2016) who noticed that there is an increase in emission level of NO_x as adding HHO to gasoline in a SI engine. The increase of NO_x

concentration in the diesel-HHO engine exhaust gas was also remarked by Kale and Dahake (2016) who studied the comprehensive review. The authors resumed that the brake thermal efficiency increased by 2.6%, brake specific fuel consumption reduced by 7.3%, 13.5% reduction in CO and HC emission but NO_x increased by 13% while using HHO in diesel engines.

The above literature research shows advantages of HHO gas adding to the others fuels on engine performance and emissions. Generally, adding HHO gas to traditional fuels enhances the combustion process, increases the brake power and reduces the harmful emissions of engines. This can be accounted by the presence of hydrogen in HHO gas which improves the combustion speed. There was a net decrease in specific engine fuel consumption due to the combined effect of hydrogen and oxygen which helps in achieving complete combustion. There was also a net decrease in emission of CO, HC, CO_2 present in the exhaust gas (Arjun et al., 2019). The results of bibliography research also show that there is a limit of hydrogen concentration in fuel where a compromise between performance and emission is obtained. The use of HHO gas as additive to traditional fuels exhibits a better improvement of engine performance as compared to hydrogen.

Besides the agreements of above results, there was a divergence about the NO_x emission in literature. In several works, an increase in NO_x emissions was observed but in some other publications, a decrease in NO_x emission was reported as adding HHO to traditional fuels. Although the effects of hydrogen additive to traditional fuels on performance and emission of engines have been demonstrated by the preceding studies, there have been few studies made on real biogas-HHO gas mixtures. Otherwise, most publications in literature on this topic seems to focus mainly on experimental researches. Very few fundamental studies on combustion characteristics inside the cylinder can be found in the literature to provide detail information for a fine analysis of the performance and the emission of engines. This study contributes to close the gap, attempting to provide insights into the combustion and emission characteristics, particularly NO_x emission, of the SI engines fueled with HHO enriched biogas. The objective of the study is to find out a compromise between engine performance and NO_x emission under different operating conditions of the engine.

2. Method of study

As it has been mentioned above, the available research results on biogas-HHO engine in the literature are mainly the output experimental data. They are not enough for a fine analysis of effects of different parameters on performance and emission of the engine. In fact, the measurement of instantaneous parameters of combustion phenomena inside the combustion chamber is always a challenge. Numerical simulation can be an effective way to overcome the difficulty. This method has been chosen for the research. The research results can be used to explain some available experimental results in the literature. They can also point out the appropriate operating conditions to improve the performance as well as to reduce the emission of a SI engine fueled with HHO enriched biogas.

In this research the commercial computational fluid dynamics (CFD) package ANSYS Fluent V15.0 is used for numerical simulation. The calculation of biogas-HHO combustion process and NO_x emission is performed on a four-cylinder, four-stroke, naturally aspirate, port injection, spark ignition DA465QE engine. The engine has a bore of 65.5 mm and a stroke of 72 mm with compression ratio of 8.8.

The fundamental governing equations of fluid dynamics (continuity, momentum, energy, species) closed by the turbulence model for unsteady flow were solved by a segregated pressure-based solution algorithm. The well-known Re-Normalized Group RNG k-ε model was used for modeling turbulence. It was analogous in form to the standard k-ε model but had the advantage of including the effect of swirl, which was important for mixture movement in combustion chamber of internal combustion engine.

NO_x formation is modeled by extended Zeldovich mechanism which is largely used in modeling pollutants emission of combustion process. As combustion temperature of HHO gas enriched biogas is normally higher than 1600K, NO_x formation rate via thermal mechanism described by Zeldovich is dominant.

Calculations are carried out in a closed system theoretically from intake valve closure at 0°CA to exhaust valve opening at 360°CA with different operating conditions. The grid independency study has been carried out for the present geometry of the cylinder space to identify the optimum number of cells that can be used in the simulation (Bui et al., 2018b). Initial conditions are taken from results of gas exchange calculation at the end of intake process (Bui et al., 2019b). The calculation procedure of effects of fuel composition and operating conditions on engine performance and pollutants emission has been presented in detail in (Bui et al., 2018a, 2019a). In the present study, the combustion process of HHO enriched biogas-air mixture is modeled via partially premixed model.

Biogas fuel is denoted by MxCy, in which 10x is the volume percentage of CH_4 and 10y is the volume percentage of CO_2 in biogas. HHO gas is a mixture containing two-thirds of the H_2 volume and one-third of the O_2 volume. The HHO content is calculated by the ratio of HHO volume in the total volume of HHO and biogas mixture. For example, the mole fractions of the mixture biogas M6C4 enriched by 20% HHO are: CH_4 (48%), CO_2 (32%), H_2 (13.33%) and O_2 (6.67%).

The low heat value by volume of hydrogen is 10.8 MJ/m^3, much more inferior to that of methane (35.8 MJ/m^3). When the HHO is added to biogas, oxygen-containing HHO is enough for a complete combustion of supplied hydrogen. The supplied air is needed for combustion of CH_4 in biogas only. Thus, the quantity of inert gas N_2 introduced to the engine fueled with HHO enriched biogas is less than that of the engine fueled with neat biogas.

3. Results and discussion

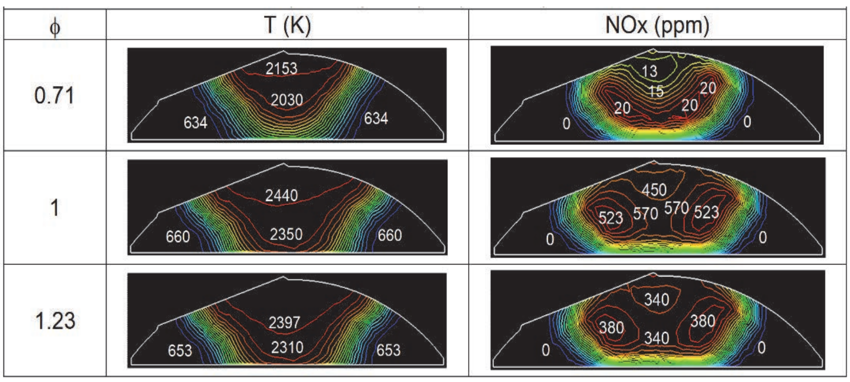
3.1. Effects of the equivalence ratio

Fig. 1 illustrates effects of the equivalence ratio on the contours of temperature and NO_x concentration at TDC on the longitudinal section xy. The engine operates at 4000 rpm, fueled with biogas M6C4 enriched by 30% HHO at a fixed advanced ignition timing φ_s =25^{0}CA. The maximum temperature is found behind the flame front while the maximum NO_x concentration is found in the reaction region in early stage of combustion. As the flame front propagates toward cylinder wall, the zone with maximum NO_x concentration move along with it. But in the late stage of combustion, the maximum NO_x concentration is found in the region where it is formed initially (spark plug region). In fact the concentration of NO_x is controlled by kinetic reaction, thus it depends on the existence time of the combustion products in high temperature medium.

Comparing with stoichiometric mixture, the volume of burned zone of poor mixture ϕ=0.71 is smaller due to lower burning velocity. The flame front propagates faster with stoichiometric mixture (ϕ=1). The larger volume of burned zone in this case results in an increase of heat release rate, thus an increase of temperature. Rich mixture (ϕ=1.23) lowers the combustion velocity, thus, reduces temperature and NO_x concentration as well.

Figs. 2a-b present the variation in the CH_4 and H_2 concentrations during combustion process of biogas enriched by 20% HHO. It can be observed that the rate of consumption of both CH_4 and H_2 decreases with poor or rich mixture. When the rich mixture is supplied, the gas-water reaction CO_2+H_2=CO+H_2O in the combustion process takes place intensely, producing hydrogen in the expansion process. Under normal combustion, the combustion of CH_4 starts just after ignition (at 160°CA) and ends at about 240°CA. Meanwhile, the complete combustion of H_2 depends on conditions of the chemical thermodynamic equilibrium including temperature, pressure and mixture's composition. As ϕ<1, the complete combustion of CH_4 and H_2 achieved almost at the same time. However, as ϕ>1, the ratio of H_2/CH_4 in the combustion mixture rises in function of ϕ (Figs. 2a-b). Fig. 2c shows the effect of equivalence ratio ϕ on the variation of pressure in the cylinder respect to crankshaft angle. It can be seen that the lean mixture results in a reduction of maximum pressure. Simulation study shows that when poor biogas M6C4 is employed, the engine can not operate with an equivalence mixture less than 0.8. But when 30% of HHO is added to the biogas, the engine can run with an equivalence ratio around 0.6. The addition of HHO gas positively influences combustion due to the high laminar burning velocity and the wide flammability limits of hydrogen. This represents a possible means to use leaner fuel-air mixtures, with favorable effects on exhaust emissions and thermodynamic efficiency, particularly at low load regime.

Figs. 3a-c compare the indicative work cycle, the maximum temperature in the combustion chamber and the NO_x concentration as the engine operates with biogas M6C4 only mode and with biogas enriched by 10% and 30% HHO mode. It can be observed that at a given equivalence ratio, the indicative work cycle, the temperature and the NO_x concentration increased as HHO concentration in the fuel mixture increased. At a given HHO concentration, the curves Wi, T, NO_x concentration show peak values at $\phi\approx$1.1. The increase rate of NO_x concentration with ϕ is much higher than that of Wi or T. The maximum Wi increases by 4% and 12% as adding 10% and 30% HHO to biogas M6C4, respectively. This result agrees with experimental research results of Arjun et al. (2019), Babariya et al. (2015), Leelakrishnan and Suriyan (2013) who observed an improvement of brake power of SI engines as adding HHO to gasoline. However, NO_x concentration as adding 10% HHO and 30% HHO to biogas M6C4 is 1.5 times and 2.8 times, respectively, as compared to NO_x concentration of biogas only operation mode.



(a) (b)

Fig. 1. Contours of instantaneous temperature (a) and NO_x concentration (b) at TDC on xy section as the engine fueled with biogas M7C3 enriched by 30% HHO (n=4500 rpm, φ_s =25^{0}CA, full load)

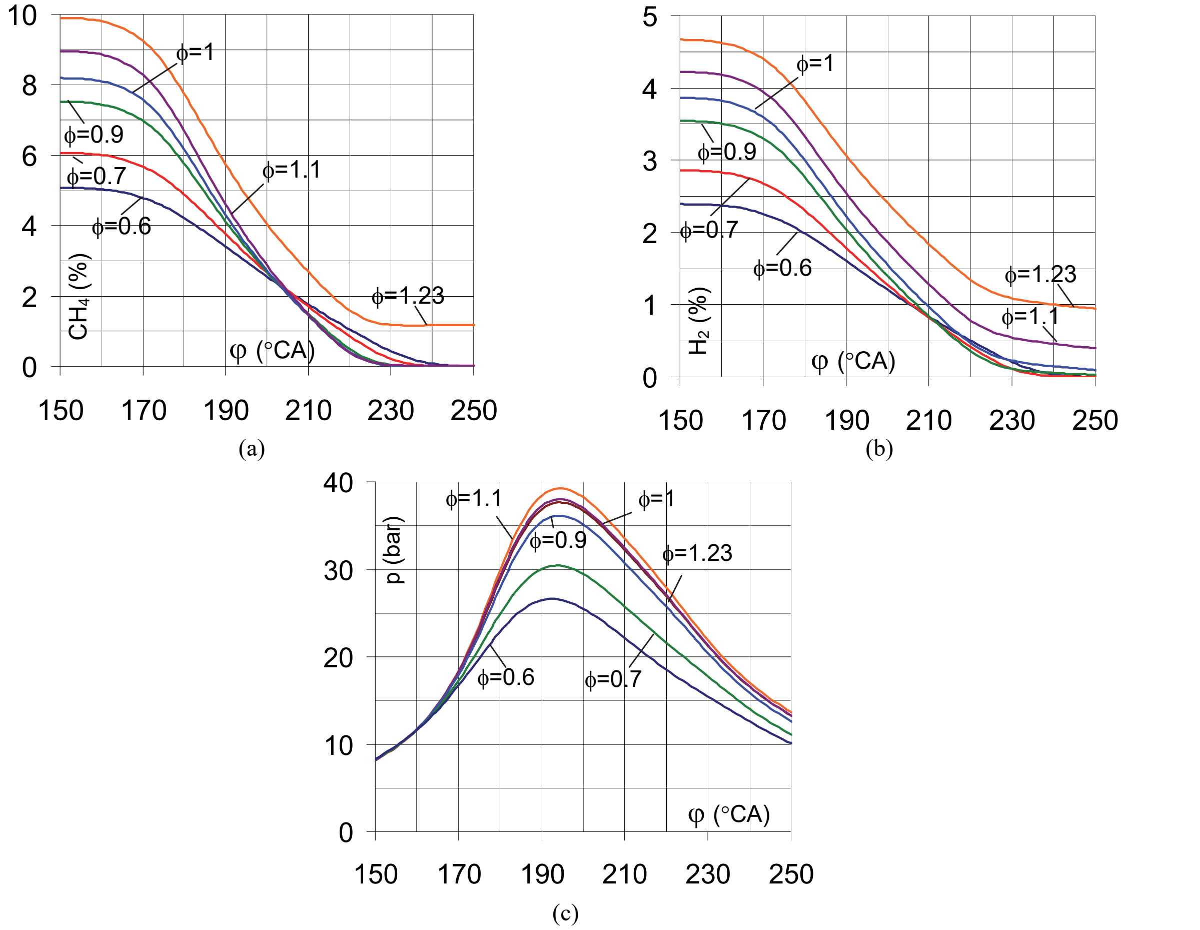


Fig. 2. Effects of the equivalence ratio ϕ on variation of CH_4 concentration (a), H_2 concentration (b) and pressure (c) in function of crankshaft angle as the engine is fueled with biogas M6C4 enriched by 20% HHO (n=4000 rpm, φ_s =25^{0}CA, full load)

Inversely, near the lean limit of combustion (ϕ<0.75), the NO_x emission of biogas only operation mode and HHO enriched biogas operation mode is not significantly different, but the indicative work cycle increases by 10% and 20% as adding 10% HHO and 30% HHO, respectively, to the biogas. Thus, the addition HHO to biogas is more interesting when the engine is fueled with lean mixture.

3.2. Effects of advance ignition timing

Due to the presence of CO_2 in biogas, the combustion rate of biogas-air mixture is normally lower than that of purified fuels. Thus, increasing advanced ignition timing is crucial to improve the combustion efficiency of the biogas engine. As mentioned above, the combustion rate is particularly improved as adding HHO to biogas, thus, the advanced ignition timing can be reduced as compared to neat biogas operation mode.

If the advanced ignition timing is fixed, since the available time for combustion process decreases as increasing engine speed, the combustion will continue during the expansion stroke. The peak of pressure shifts away TDC, results in a reduction of maximum pressure and temperature, consequently, a reduction of NO_x concentration. As shown in Fig. 4a the maximum pressure decreases from 31 bar to 21.5 bar and NO_x concentration decreases from 2800 ppm to 1050 ppm as engine speed increases from 2500 rpm to 4500 rpm with fixed φ_s at 25°CA.

To maintain the indicative work cycle, the advanced ignition timing must be increased as the engine speed increases. Fig. 4b shows the variation of indicative work cycle Wi and NO_x concentration with engine speed when the advanced ignition timing is fixed and when it is altered. As the engine speed increases from 2500 rpm to 5000 rpm, Wi decreases from 175 J/cyc to 161 J/cyc in case of fixed advanced ignition timing at 25°CA. In case of advanced ignition timing changed from 22°CA at n=2500 rpm to 37°CA at 5000 rpm, Wi decreases slightly from 175 J/cyc to 170 J/cyc. In the same conditions of engine speed increase, NO_x concentration decreases from 2800 ppm to 800 ppm in the first case and decreases from 2400 ppm to 1900 ppm in the second case.

Fig. 5a synthesizes the correlation between the indicative work cycle and NO_x concentration when the engine runs at 5000 rpm with different biogas enriched by 30% HHO. The advanced ignition timing varies from 20°CA to 40°CA. In all cases, when the advance ignition angle is lower than the optimal value, the rate of increasing Wi is higher than that of NO_x concentration. But after this value, rate of increasing of NO_x concentration is much higher than that of Wi. Therefore, to obtain harmonized values of Wi and NO_x

concentration, advanced ignition timing should be chosen on the contour of Wi-NO_x curves as shown in Fig. 5a. Concretely, the advanced ignition timing in range of 28°CA-35°CA is appropriate to the engine operating at speed of 5000 rpm and fueled with any biogas enriched by 30% HHO.

The correlation between Wi and NO_x concentration as the engine is fueled with biogas M7C3 enriched by different compositions of HHO at speed of 4000 rpm is presented in Fig. 5b. It can be seen clearly the derivation of the curve Wi-NO_x from the contour after the optimal value of φ_s. The results show that if the engine operates around 4000rpm with biogas M7C3 enriched by different HHO compositions, the advanced ignition timing in range of 25°CA-30°CA will be coherence between performance and NO_x emission. Fig. 6a present variation of Wi and NO_x concentration with respect to φ_s as the engine operates at different speeds. It can be seen that each curve Wi(φ_s) exhibits a peak at 20, 25, 29 and 33°CA corresponding to engine speed of 2000, 3000, 4000 and 5000 rpm. These values can be considered as optimal advanced ignition timing for corresponding engine speeds. The indicative work cycle is 291, 289, 285 and 283 J/cycle corresponding to engine speed of 2000, 3000, 4000 and 5000 rpm. The indicative work cycle decreases by 2.5% as engine speed increases from 2000 rpm to 5000 rpm. As the engine operates with optimal advanced ignition timing, at speed of 2000, 3000, 4000 and 5000 rpm, NO_x concentration in exhaust gas will be 3100, 2600, 2100 and 2000 ppm, respectively. The NO_x emission thus, decreases by 30% as engine speed increases from 2000 rpm to 5000 rpm.

Without addition of HHO, the optimal advanced ignition timing is larger as shown in Fig. 6b. It can be seen that the optimal value of φ_s is 24, 30, 36 and 42°CA corresponding to engine speed of 2000, 3000, 4000 and 5000 rpm. The maximum Wi corresponding to these speeds are 278, 269, 258 and 249 J/cycle. The NO_x concentration is 1590, 950, 800 and 700 ppm, respectively. The indicative work cycle decreases by 10% and NO_x decreases by 56% as engine speed increases from 2000 rpm to 5000 rpm. As compared with the previous case, the optimal advanced ignition timing reduces by 6°CA and Wi, NO_x concentration are less sensitive to variation of engine speed as adding 30% HHO to biogas M7C3.

The results in Figs. 6a-b show that the optimal advance ignition timing and the corresponding indicative engine cycle work can be expressed by linear relations while NO_x concentration can be represented by a parabolic curve with respect to the engine speed. As the engine operates with biogas fueling mode and 30% HHO enriched biogas fueling mode, the rate of increase of optimal advance ignition timing ($d\varphi_{so}/dn$) is 0.006 and 0.0043 respectively; the rate of decrease of indicative engine cycle work (dWi/dn) is -0.0097 and -0.0028 respectively and the rate of decrease of NO_x concentration (dNO_x/dn) can be expressed by (0.0002n-1.227) and (0.0002n-1.08) respectively.

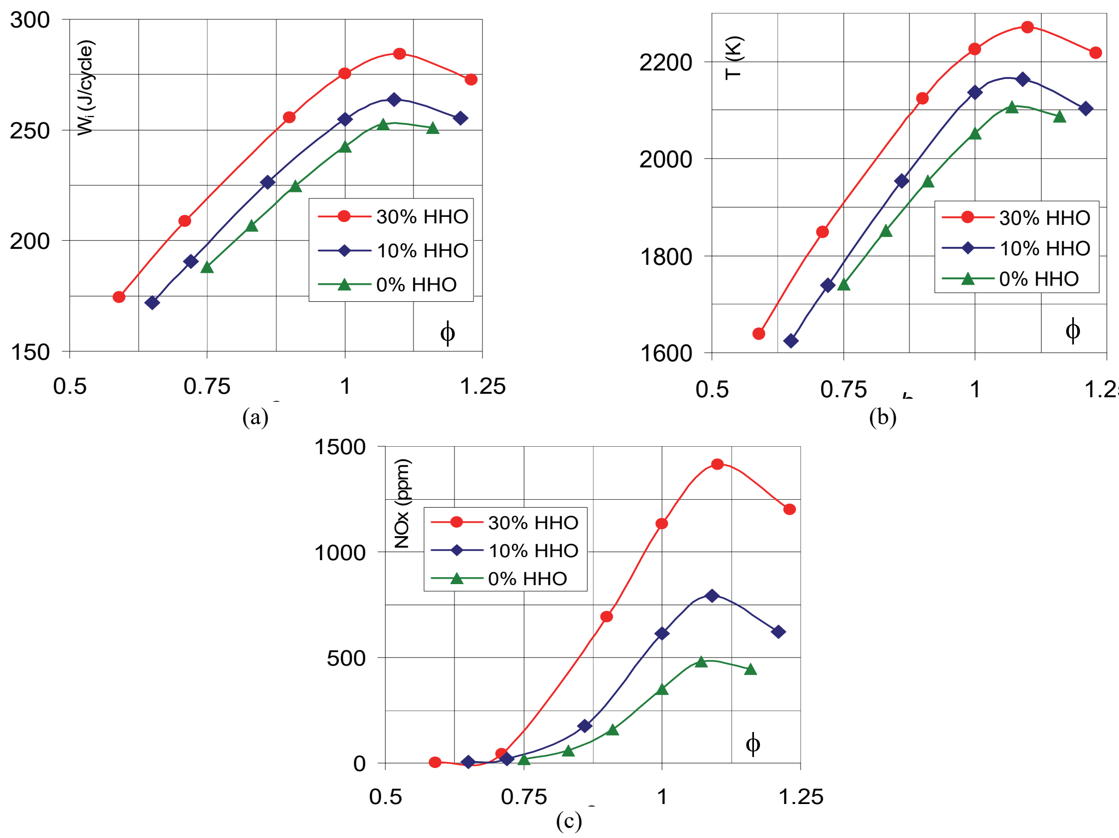
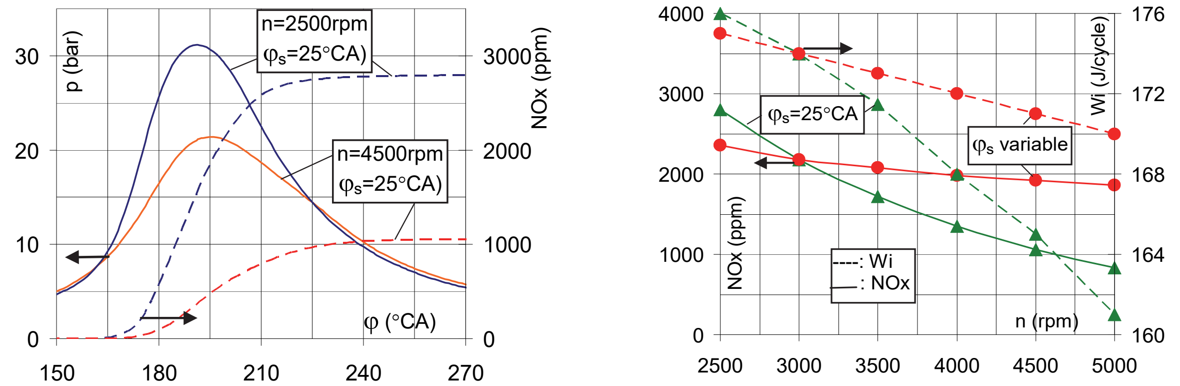
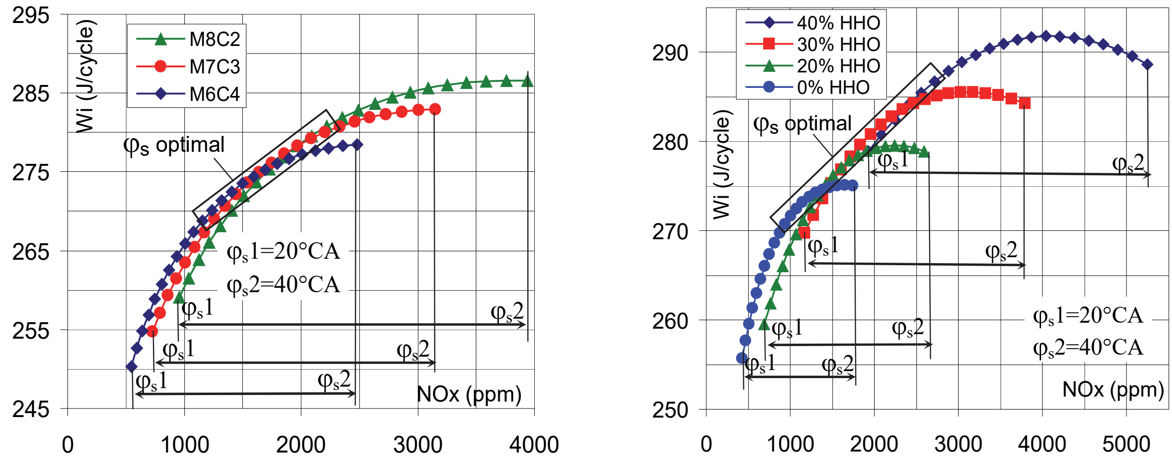


Fig. 3. Effects of HHO concentration on variation of Wi (a), T (b) and NO_x concentration (c) in function of equivalence ratio ϕ (Biogas M6C4, n=4000 rpm, φ_s =25^{0}CA, full load)



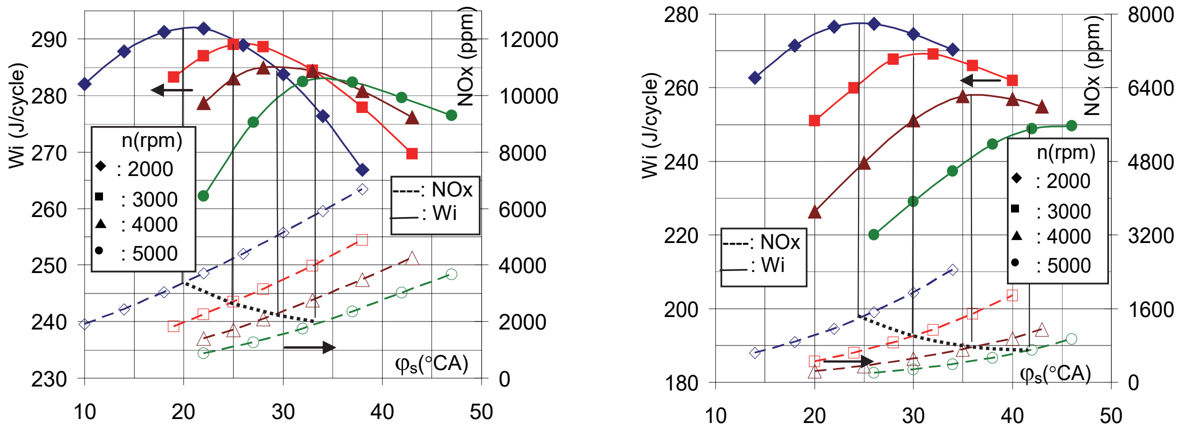
(a) (b)

Fig. 4. Effect of engine speed on combustion and NO_x emission of engine fueled with biogas M7C3 enriched by 30% HHO (ϕ=1, 60% load rate): (a) Variation of p and NO_x concentration in function of crankshaft angle in case of φ_s constant; (b) Variation of Wi and NO_x concentration in function of engine speed in case of φ_s constant and φ_s variable



(a) (b)

Fig. 5. Effect of biogas composition and HHO concentration on optimal φ_s: (a) n=5000 rpm, full load, biogas enriched by 30% HHO; (b) n=4000 rpm, full load, biogas M7C3



(a) (b)

Fig. 6. Effects of engine speed on variation of Wi and NO_x concentration in function of φ_s: (a) Biogas M7C3 enriched by 30% HHO (ϕ=1, full load); (b) Biogas M7C3 only (ϕ =1, full load)

3.3. Effects of load regime

When the engine operates at low load regime, the addition of HHO to biogas significantly improves the maximum pressure. Fig. 7 shows that the increase of maximum pressure from 8.5 bar to 13.5 bar results in an increase of indicative work cycle from 74.2 J/cycle to 92.3 J/cycle as the engine operates at 30% load regime, fueled with biogas M7C3 only and with biogas M7C3 enriched by 50% HHO. The addition of

HHO also raises the level of NO_x concentration in exhaust gas. The NO_x concentration in exhaust gas reaches 192 ppm as the engine is fueled with biogas M7C3 only; but it increases to 396 ppm, 598 ppm, 996 ppm, 1650 ppm and 2870 ppm when 10%, 20%, 30%, 40% and 50% of HHO are added to biogas respectively (Fig. 7a). The correlation between NO_x concentration and HHO content in the mixture with biogas can be thus expressed by a parabolic equation with the slope $dNO_x/dC_{HHO}=2.47C_{HHO}-11.65$. HHO composition thus, exhibits important effects on NO_x emission as the biogas engine operates at any load regimes. Besides, in case of partial load operation mode and with a given HHO composition in biogas, peaks of pressure and NO_x concentration increase with CH_4 concentration in biogas increase. It can be observed in Fig. 7b that when the engine operates at 4000 rpm and 60% of load regime fueled with a fixed 30% HHO concentration in biogas, the maximum pressure increases from 21.5 bar to 27 bar and NO_x concentration increases from 800 ppm to 2500 ppm as shifting fueling mode from biogas M6C4 to biogas M9C1.

Fig. 8a presents the correlation between indicative work cycle and NO_x concentration when the engine load varies from 30% to 100% and HHO content in biogas M7C3 increases from 0% to 50%. It can be seen from the figure that as HHO content in biogas is lower than 30%, the rate of increasing Wi is higher than that of NO_x concentration. However, when HHO content exceeds 30%, the rate of increasing Wi is significantly lower than that of NO_x. Thus, for the best compromise of Wi/NO_x concentration, HHO content in biogas M7C3 should be lower than 30% under any engine load regimes. In case of hydrogen in mixture with methane, Wang et al. (2007) pointed out that the optimal concentration of H_2 is 20%. Thus, the optimal concentration of HHO in biogas is higher than that of H_2 in methane. Fig. 8b shows the effects of biogas composition in mixture with 30% HHO on Wi and NO_x concentration at different load regimes. The indicative work cycle and NO_x concentration increase as CH_4 content in biogas or/and load regime of the engine increase.

The slope of Wi-CH_4 concentration curves is slightly affected by engine's load regime. The rate of increasing of indicative work cycle is practically the same, about 2.46%, as load regime of the engine increases from 40% to 100% at CH_4 content in biogas of 60% or 90%. At the same conditions, the rates of increasing of NO_x concentration are 1.57 and 1.45 for biogas M6C4 and M9C1 respectively when load regime of the engine increases from 40% to 100% (Fig. 8b).

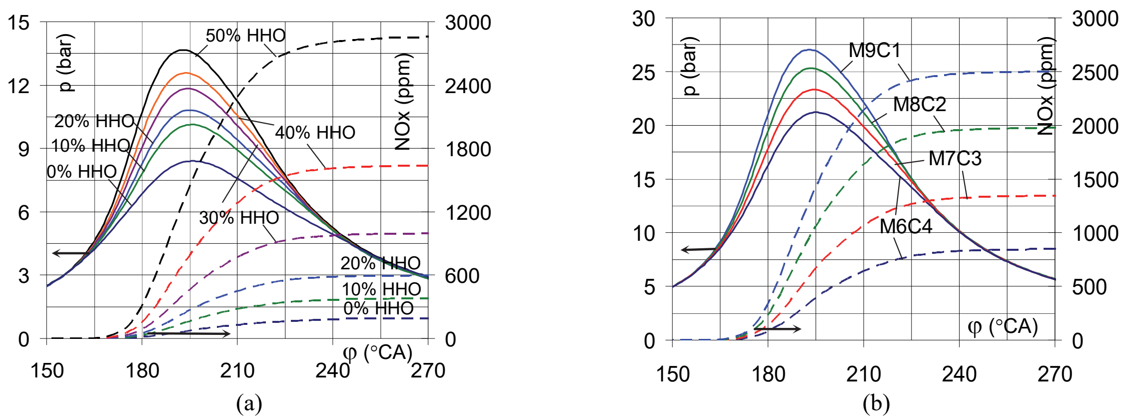


Fig. 7. Effect of HHO content and biogas composition on in-cylinder pressure and NO_x emission (n=4000 rpm, , φ_s=25^{0}CA, ϕ=1): (a) Engine operates at 30% load regime fueled with biogas M7C3 enriched by different HHO content; (b) Engine operates at 60% load regime with different biogas enriched by 30% HHO

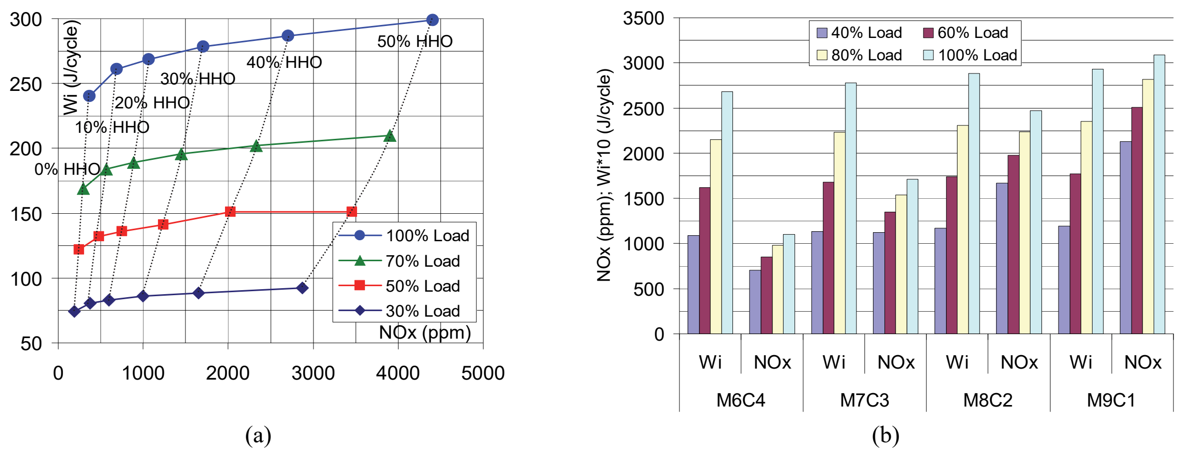


Fig. 8. Correlation between Wi and NO_x concentration (n=4000 rpm, , φ_s =25^{o}CA, ϕ=1): (a) Under effects of different HHO contents enriched biogas M7C3 and load regime; (b) Under effects of different biogas enriched by 30% HHO and load regime

The result shows that at a given HHO composition, to achieve the required indicative work cycle, the engine can operate at full load regime with poor biogas or at partial load regime with rich biogas. Concretely, the engine generates an equivalent indicative work cycle as it operates at full load regime and fueled with poor biogas M6C4 enriched by 30% HHO, or as it operates at 90% load regime and fueled with rich biogas M9C1 enriched by 30% of HHO. However, the NO_x emission in the first case is only 1000 ppm while in the second case is 3000 ppm. Therefore, with a given HHO composition, using poor biogas enriched HHO at full load regime is more beneficial than rich biogas at partial load regime for environment protection aspect.

The overall above results show that there are several parameters affecting engine performance (Wi) and NO_x emission. The main parameters are equivalence ratio, HHO content, CH_4 content, advanced ignition timing, engine speed and load regime. Generally, Wi increases with the increase of HHO content, CH_4 concentration, load regime; but it decreases with the increase of engine speed and it reaches peak value in function of ϕ and φ_s. NO_x concentration increases with the increase of HHO content, CH_4 concentration, load regime, φ_s; but it deceases with the increase of engine speed, and reaches peak value with ϕ. When adding HHO to traditional fuels, depending on the relative burning velocity between HHO and the fuel, optimal value of advanced ignition timing can be more or less reduced. Therefore, NO_x emission of the engine can be higher or lower than that of the original fuel operation mode. This is the reason for the divergent conclusions on NO_x emission when adding HHO to gasoline and diesel that has been mentioned in the literature (Hüseyin et al., 2016; Mohamed et al., 2016; Patel et al., 2016; Sharma et al., 2015). In case of biogas enriched by HHO, a compromise between performance and harmful emission can be obtained by appropriate adjustment of operating conditions. In general, Wi and NO_x emission increases as adding HHO to biogas comparing to neat biogas operation mode.

4. Conclusions

Based on the analysis of simulation results, the following conclusions can be drawn:

- At a given HHO concentration, the curves Wi, T, NO_x concentration present peak value at $\phi\approx1.1$. The rate of increasing NO_x concentration with ϕ is much higher than that of Wi or T. As adding 10% and 30% HHO to biogas M6C4, Wi increases by 4% and 12%, respectively; however, NO_x concentration is 1.5 times and 2.8 times higher, respectively, as compared to that of neat biogas operation mode.
- As HHO content in biogas is lower than 30%, the rate of increasing Wi is higher than that of NO_x concentration. However, when HHO content exceeds 30%, the rate of increasing Wi is significantly lower than that of NO_x. Thus, for the best compromise of Wi/NO_x concentration, HHO content in biogas M7C3 should be lower than 30% under any engine load regime.
- The addition HHO to biogas is more interesting as the engine operates with lean mixture. With $\phi<0.75$, the NO_x emission is slightly different as comparing neat biogas operation mode and biogas enriched HHO operation mode, but the indicative work cycle increases by 10% and 20% as adding 10% HHO and 30% HHO, respectively, to the biogas M6C4.
- As φ_s increases, Wi reaches a peak, while NO_x concentration continuously increases. The appropriate φ_s which harmonizes values of Wi and NOx concentration should be chosen on the contour of Wi-NO_x curves. The advanced ignition timing in range of 28°CA-35°CA is appropriate to the engine operating at speed of 5000 rpm and fueled with any biogas enriched by 30% HHO. Moreover, the advanced ignition timing in range of 25°CA-30°CA is appropriate as the engine operates around 4000 rpm with biogas M7C3 enriched by any HHO composition.
- The optimal advanced ignition timing increases with engine speed. With biogas M7C3 enriched by 30% HHO, as the engine speed increases from 2000 rpm to 5000 rpm, appropriate φ_s increases from 20°CA to 33°CA, the indicative work cycle and NO_x concentration decrease by 2.5% and 30%, respectively. Optimal φ_s as the engine operates with neat biogas M7C3 increases by 6°CA compared to that of biogas M7C3 enriched by 30% HHO; Wi and NO_x concentration in this case decrease by 10% and 20%, respectively, as engine speed increases from 2000 rpm to 5000 rpm.
- To obtain a desired Wi, with a given HHO composition, using poor biogas enriched HHO at full load regime is more beneficial than rich biogas at partial load regime for environment protection aspect.
- In case of biogas enriched by HHO, at a given operating condition, Wi increases with the increase of HHO content, CH_4 concentration, load regime but it decreases with the increase of engine speed and it reaches peak value in function of ϕ and φ_s. NO_x concentration increases with the increase of HHO content, CH_4 concentration, load regime, φ_s, but it deceases with the increase of engine speed, and reaches peak value in function of ϕ. Therefore, a compromise between engine performance and NO_x emission can be obtained by appropriate adjustment of fuel compositions or/and operating conditions of the engine.

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Nomenclature

CA Degree crankshaft angle
C_{HHO} Percent content of HHO in the mixture with biogas by volume
HHO Mixture of 2/3 H_2 and 1/3 O_2 by volume
MxCy Biogas constituted by 10x% CH_4 and 10y% CO_2 by volume
n Engine speed (rpm)
p Pressure (bar)
SI Spark ignition
T Temperature (K)
TDC Top dead center
V In-cylinder volume (liter)
Wi Indicative work cycle per cylinder (J/cycle)
ϕ Equivalence ratio
φ Crankshaft angle
φ_s Advanced ignition timing (°CA)
φ_{so} Optimal advance ignition timing (°CA)

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