IOP Conference Series: Materials Science and Engineering

PAPER • OPEN ACCESS

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To cite this article: G Przybyła *et al* 2016 *IOP Conf. Ser.: Mater. Sci. Eng.* **148** 012082

IOP Conf. Series: Materials Science and Engineering **148** (2016) 012082 doi:10.1088/1757-899X/148/1/012082

The impact of air-fuel mixture composition on SI engine performance during natural gas and producer gas combustion

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Abstract. The paper summarizers results of experimental tests of SI engine fuelled with gaseous fuels such as, natural gas and three mixtures of producer gas substitute that simulated real producer gas composition. The engine was operated under full open throttle and charged with different air-fuel mixture composition (changed value of air excess ratio). The spark timing was adjusted to obtain maximum brake torque (MBT) for each fuel and air-fuel mixture. This paper reports engine indicated performance based on in-cylinder, cycle resolved pressure measurements. The engine performance utilizing producer gas in terms of indicated efficiency is increased by about 2 percentage points when compared to fuelling with natural gas. The engine power de-rating when producer gas is utilized instead the natural gas, varies from 24% to 28,6% under stoichiometric combustion conditions. For lean burn (λ=1.5) the difference are lower and varies from 22% to 24.5%.

1. Introduction

Running out of fossil fuels and increase in climate change are the main reason to seek alternative methods for more efficient use of fuel energy and increase in application of renewable resources. That is the resources which do not emit greenhouse gases and contribute to major reduction of emissions (considering fuel sustainability and carbon life cycle). Among all the types of renewable fuels, biomass is considered the most important with respect to the Polish climate and geographical conditions [1-3] Liquid fuels derived from biomass processing such as vegetable oils (i.e. fatty acid methyl esters - FAME) and alcohols are currently common in use as a blend component or additive to conventional fuels [4-6]. These fuels play an important role and can be considered as a possible alternative for the internal combustion engines. When considering biomass processing technologies, gasification process is one of the possible options for biogas production often called producer gas (PG) or syngas. The producer gas is characterized by its low calorific value and it can be used as a fuel in internal combustion engines and in process heat generation (boilers).

1.1. Producer gas utilization in SI engine

The number of studies on the use of synthesis gas from biomass gasification or pyrolysis in SI engines is quite limited considering the studies on eg. the use of biogas from fermentation processes. The authors of [7] presents the results of SI engine fuelled with natural gas and a gas having a composition corresponding to gas from the biomass gasification (CO = 17% CO_2 = 20% H_2 = 30%, CH_4 = 3%, N_2 = 30%). Tests were carried out in a range of variable composition of natural gas, which means it was diluted with CO_2, N_2 and air. For different values of the natural gas dilution with carbon dioxide, nitrogen and air parameters characterizing the combustion process were evaluated. The coefficient of variation of IMEP remained at favourable levels during the combustion of syngas in relation to diluted natural gas. Comparing the results for the same excess air ratio ($\lambda = 1.25$), the highest value COV of IMEP = 30% was observed in diluted gas by carbon dioxide, and the lowest value COV of IMEP ~ 0,7% was observed in the synthesis gas. In addition, indicated efficiency and utility range of the excess air ratio were higher during the combustion of the syngas. Both in terms of the stoichiometric and lean mixtures the combustion of the synthesis gas is accompanied with significantly higher content of CO in the exhaust gas in relation to the use of natural gas. On the other hand, the content of unburned hydrocarbons was several tens of times higher than during the combustion of natural gas. During the combustion of stoichiometric mixtures the content of NOx in the exhaust gas was significantly higher than during the combustion of natural gas. Realizing the combustion of lean mixtures the NOx content was at a similar level for both fuels. Although the work contains many valuable results and observations unambiguous comparison of emissions is impossible. Values given by the study's authors represent the molar proportions of individual components at the same excess air ratio. It should be stressed that natural gas and syngas have a different compositions, hence the content of ingredients such as CO_2 and N_2 in the engine exhaust is different for both fuels. One of the studies that is characterized by a wide range of research and unequivocal comparison of the parameters is the work [8]. The authors present SI engine research results fuelled with gas from the gasification of sewage sludge and a mixture of methane - the gas from the gasification of sewage sludge (CO = 16% CO_2 = 15% H_2 = 13%, CH_4 = 3%, N_2 = 53%). The study was conducted during the combustion of stoichiometric mixtures and for combinations of lean mixtures. The engine test were conducted at constant rotational speed, the load was a result of the intake pressure, which was set at constant value of 80 kPa. As shown in the presented report, the gas from the gasification of sewage sludge is troublesome fuel to power internal combustion engines. The combustion of fuel in the SI engine is unstable and often accompanied by the phenomenon of the misfire, consisting in the absence of combustion of a combustible mixture in some work cycles. Therefore, the authors determined the effect of the additive gas to the gas from the gasification of biomass on the correct operation of an internal combustion engine including the aspect of energy and environmental impacts. The optimum value of the volume fraction of methane in the mixture of the gas from the gasification of sewage sludge has been set at 40%. The addition of methane in to gas from gasification of sewage sludge significantly improves the stability of the internal combustion SI engine operating in the lean mixtures. Research in the combustion of lean methane-gas from the gasification of sewage sludge with the air shows no significant difference in the performance characteristics of the engine in relation to the energy supply of pure methane. Emissions of harmful substances (such as HC, NOx and CO) while using mixtures of synthesis gas - methane with variable ratio of excess air was at a similar level as it was when using pure methane. The test results of an installation consisting of a downdraft gasifier and SI engine shown in [9] indicate a higher efficiency of electricity generation during the combustion of the synthesis gas with respect to gasoline. In addition, presented in this paper results indicate that the molar ratio of CO and NOx in the exhaust gas of the engine fed with the synthesis gas was lower in relation to the petrol respectively 30-96% and 54-84% (depending on the engine load). The authors do not specify at what excess air ratio the tests were conducted. These values are ambiguous shares molar compared to the operating parameters of the engine during petrol and gas from the gasification, as well as energy efficiency of the system. The provided values of molar fractions and efficiency are ambiguous compared to the working parameters of the engine during combustion of fuel or syngas. Another example of research of gasification installation combined with SI engine is publication [10]. The tested object is based on commercial installations produced by the Canadian company Ankur code-named WBG-120. The internal combustion engine used in the system is a spark ignition engine type Cummins G-855-G driving the three-phase electricity generator type Stamford UC 27E (125 kVA). Average gas composition obtained by a gasification process of olive trees seeds was: CO = 10.7%, CO_2 = 4.6%, H_2 = 24.1%, CH_4 = 4.2%, N_2 = 55.1%, O_2 = 1.3%. During the research the engine worked at a constant speed (1500 rev/min) and full load. As a result the efficiency of electricity generation was 16.1%. The authors did not examine the effect of control parameters on engine efficiency and emission of harmful substances. One suspects that the low efficiency of power generation is a result of lack of control parameters optimization of the internal combustion engine.

The present work reports on the indicated performance of SI engine fuelled with a natural gas and producer gas substitute (clean and dry gas mixture obtained by mixing the gaseous compounds stored in high pressure cylinders). The stability limits in combustion process of lean air-fuel mixtures was investigated. As an stability indicator of combustion process the COV of IMEP was used.

2. The fuel properties and experimental test bench characteristic

Gaseous fuels are attractive for use in internal combustion engine (ICE) because of their wide ignition limits and ability to form homogeneous mixtures. Moreover, gaseous fuels usually have high hydrogen to carbon ratio, and thus relatively low CO_2 emissions are possible when they are used in SI engines. Natural gas is readily available from deep underground natural rock formations, or as a petroleum-based fuels, while producer gas can be obtained from renewable sources using biomass. When biomass is used sustainably to displace fossil fuels, the net impact is a lower CO_2 level in the atmosphere. The volumetric calorific value of producer gas is usually more than 5 times lower compared with natural gas. The LHV of producer gas varies with its composition that is heavily dependent on the gasification process. The main variable that determines the producer gas composition is the relative air flow direction. Gasifiers with a parallel flow of biomass feedstock and air impost higher technological requirements on the biomass feedstock compare to those with a counter- flow arrangement of biomass and air feeds. However, the producer gas from parallel flow gasifiers contains less undesired liquid residuals, such as tar and water. For application, where small and medium power outputs are needed, the most advanced technologies offer gasifiers with a parallel downdraft biomass and air delivery [11-13].

2.1. Properties and production method of producer gas

The major components of producer gas are nitrogen (N_2), carbon dioxide (CO_2), hydrogen (H_2), carbon monoxide (CO), methane (CH_4), and water (H_2O). Occasionally minute amounts of higher hydrocarbons such as ethane (C_2H_6) can be traced. At the exit from a gasifier the producer gas is usually at elevated, 500 ºC to 800 ºC, than ambient temperature. Therefore, the producer gas energy content exceeds its lower caloric heating value, by the enthalpy change between these temperatures. It follows that efficiency of the gasification process can be asset based on hot or cooled gasification products. In both methods, the energy content of producer gas is referenced with respect to the biomass energy input. The process efficiency based on hot products can as high as 90%, while the cold products efficiencies are in 40 to 70% range. Mole fractions of combustible components in a dry producer gas vary and depend on the gasification process operational variables; 15 to 40% for CO, 10 to 35% for H_2, and 2 to 5% for CH_4. Similarly the producer gas LHV can usually range from 4 to 10 MJ/Nm^3 50 [14]. Literature reporting on producer gas properties and the gasification process efficiencies can be recognised as extensive [15-16].

There are various significant parameters that can be used to characterize the usefulness of a gaseous fuels to powering the internal combustion engine. Different fuels need different amount of air to create expected air-gas mixtures which can be burned in an engine. Thus, the calorific value ($e_{d,v}$) of an air-gas mixture, the flammability limits, and the flame speed, are the next important parameters to, consider [17].

The properties of fuels and air fuel-mixtures used during the experimental tests are presented in Table 1. The ed,v were computed using equation 1 and assuming the values of following data $T_1 = 298.15$ K, $p_1 = 101.325$ kPa, relative air humidity, $\varphi = 45\%$ to determine molar humidity ratio, the residual exhaust gases fraction $\delta_{ex} = 5\%$ as a molar fraction in fresh air, the air excess ratio $\lambda = 1$. The influence of the air excess ratio on the calorific value of air-gas mixture for fuel described in Table 1 is shown on Figure 2.

Table 1. Properties of tested fuels.

Fuel	Mixture description (used on Figures)	Composition (by volume)	LHVm *MJ/kg*	LHV *MJ/m³*	e_{dv} *MJ/m³*	$V_{a,min}$, *m³/m³*
Natural Gas	GZ50	CH_4 = 98,5% CO_2 = 0,1% N_2 = 1% other: C_2H_6, C_3H_8, C_4H_{10}	48.84	35.3	2.91	9.4
Producer gas	HC1	H_2 = 18,2% CO = 18,5% CH_4 = 3% CO_2 = 12% N_2 = 48,3%	4.85	5.38	2.19	1.16
Producer gas	HC0.4	H_2 = 10,5% CO = 27,5% CH_4 = 2% CO_2 = 8,5% N_2 = 51,5%	4.52	5.32	2.24	1.1
Producer gas	HC0.2	H_2 = 4,8% CO = 23,2% CH_4 = 2,8% CO_2 = 10,7% N_2 = 58,5%	3.55	4.45	2.03	0.93

The calorific value of air-gas mixture can be calculated using following equation:

$$e_{d,v} = \frac{p_1}{(MR)T_1} \frac{LHV}{\left[\frac{1}{M_f} + \lambda n'_{a,\min}(1 + X_{za} + \delta_{ex}) \right]} \tag{1}$$

where:
p_1, Pa; T_1, K – in-cylinder thermodynamic parameters after filling process, (MR) – universal gas constant; LHV, J/kg – Lower heating value; M_f, kg/kmol – fuel atomic weight, λ – air excess ratio; X_{za}, $kmol_{H2O}$/ $kmol_a$ – molar humidity ratio.

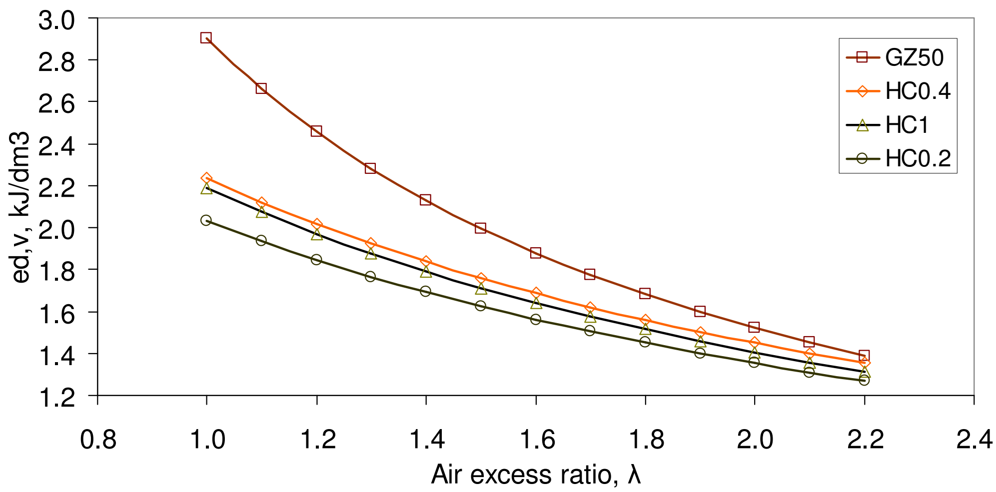


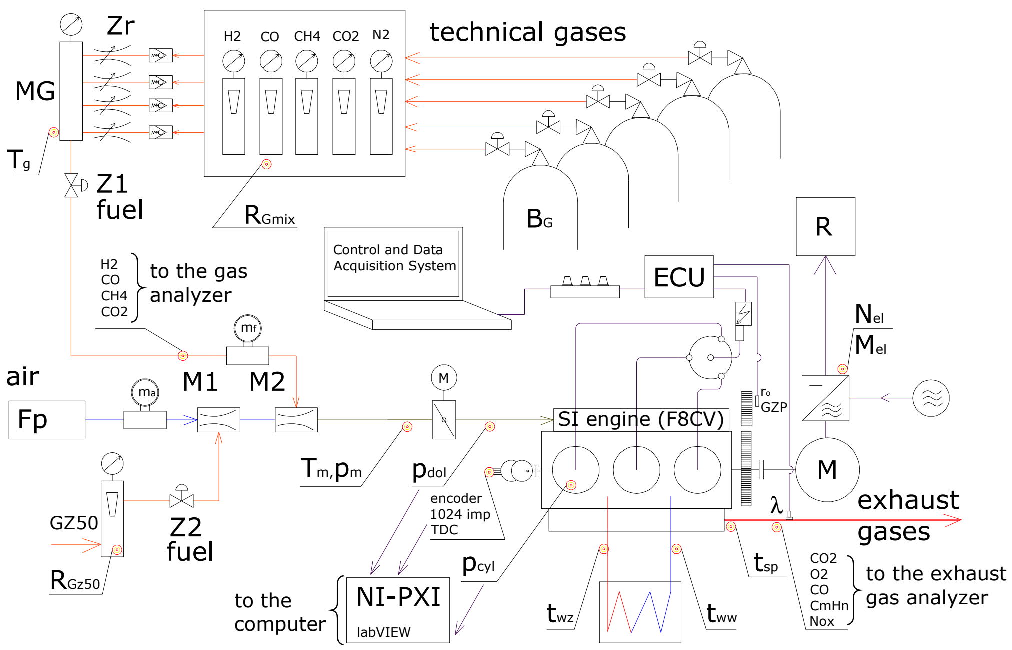
Figure 1. Calorific value of air – fuel mixture in a range of air excess ratio changes.

The LHVs of the simulated producer gases in this investigation varied from 4.45 MJ/m^3 to 5.38 MJ/m^3, compared against natural gas containing 35.3 MJ/m^3. The energy density per unit mixture volume of the producer gas is about 30% lower than that for a natural gas and air mixture at an air excess ratio λ = 1 despite the calorific value of the producer gas is 1/7.5 of natural gas. However, as the air excess ratio increase up to 2.2 the charge energy density differences become smaller and are equal 8.6% for HC0.2, 5.8% for HC1 and 2.2% for HC0.4 producer gas mixture. This shows that producer gas fuelling of the lean burn SI natural gas engine should not lead to significant power de-rating of the engine when operating with lean mixtures.

2.2. *Experimental test rig*

An overview of the engine test bench and measuring equipment is presented in Figure 2. The main components of the experimental set up include:

- Three cylinders SI engine with a capacity of 796 cm3 and compression ratio equal to 9.3. The engine is without turbocharging and was originally powered by petrol. For the purpose of experiment and possibility of gaseous fuel application, the control and power supply system of the engine have been modified.
- Electric motor with the power take-off system, capable of operating in two modes, the motor and generator. The main purpose of this system is to start the engine and then to apply load on the selected point of the operating cycle.
- High-pressure cylinders with gas mixtures and dual stage gas regulators.
- Measuring devices for flow rate, temperature and pressure evaluation including: rotameters, manometers and thermocouples.



where: Zr, Z1, Z2 – control valves, MG – gas mixer, M1, M2 – air-fuel mixers, m – mass flow meter, B_G – gas cylinders, R_{Gmix}, R_{Gz50} – rotameters, R – braking resistor, M – electric motor

Figure 2. The experimental test rig.

The engine test bench has a cooling system for engine lubrication and cooling liquids. Within the cooling system two plate heat exchangers connected to the valves have been employed. Controlling thermostat is located in the primary circuit of the engine cooling system. The settings of the control

valves on the secondary side of the heat exchangers allow to adjust the amount of the removed heat, which is then transmitted to the local central heating system. In this way, the stabilization of the engine temperature in the range of 80 ± 5°C can be assured.
During the experiment, the operation of the engine was controlled using Electronic Control Unit (ECU). This type of device driver has the capability to program and monitor engine operational parameters, i.e. pre-programmed ignition timing maps and mixture composition. The composition of the mixture was controlled using the signal from lambda sensor, which can operate in a closed loop mode with the controller. During the tests the natural gas (GZ50) was collected from local gas grid, while the producer gas substitute was stored at high pressure cylinders. The gas fuel was supplied to the engine through the gas mixer. Two mixers were selected to fuelling the engine by air-gas mixture, separately for natural gas and for producer gas. The volumetric flow of each gaseous component (CO, CO_2, CH_4, N_2) have been manually adjusted by special valve to obtain demanded producer gas mixture composition and value of air excess ratio during combustion process. Value of air excess ratio was controlled by wideband lambda sensor. The volumetric flows of the gases was measured by rotameters equipped in temperature and pressure sensor. Additionally the producer gas flow before air-fuel mixer was measured using Coriolis mass flow meter. The producer gas substitute composition was controlled using gas analyzer.
The pressure measurements in the first cylinder were performed using piezoelectric pressure transducer. This type of transducer through so called charge amplifier generates an analogue (voltage) pressure signal which is then sampled at a sufficient frequency by the data acquisition system. In addition, the absolute pressure within the intake manifold was recorded with piezoresistive absolute pressure transducer. At both measurement ducts the pressure signal was sampled at predefined crank angle using encoder. Measurements were carried out with a resolution of 1024 measurement points per revolution of the crankshaft. The encoder was also equipped with a position marker device for indicating the position of a piston in a cylinder. Each sets of measurement consisted of 100 consecutive engine cycles.

3. Results and discussion

In this study the first point of interest in evaluating the validity of the producer gas fuel as a suitable fuel for SI engine was to obtain stable combustion at a given intake air to fuel ratio. Once stable combustion was achieved the fuel flow was changed (and the ignition timing was adjusted) therefore varying the mixture equivalence ratio to determine an affective lean operating limit. The upper limit value of COV of IMEP on 5% was agreed as an indicator of combustion process stability. The tested engine fuelled with natural gas and producer gas HC0.2 mixture composition reached the stable combustion for an air excess ratio of $\lambda = 1.5$. When the producer gas mixture included more hydrogen the lean burn limit was higher. For the producer gas substitute HC0.4 mixture stable combustion was obtained for an air excess ratio of $\lambda = 1.65$, while for HC1 the limit value of air excess ratio was $\lambda = 1.95$. The MBT ignition timing for both fuels and for all tested mixtures in a range of air excess ratio are presented in Table 2.

Table 1. Optimal ignition timing for tested gaseous mixtures.

Fuel	Engine speed $\mathbf{r_o}$ = 1500 rpm, full open throttle					
	Air excess ratio λ					
	1	**1,25**	**1,5**	**1,65**	**1,85**	**1,95**
	MBT Ignition timing $\boldsymbol{\alpha_z}$, CA before TDC					
GZ50	**21**	**26**	**44**	P_{lim}	P_{lim}	P_{lim}
HC1	**21**	**30**	**34**	-	**59**	**64**
HC04	**24**	**31**	**42**	**51**	P_{lim}	P_{lim}
HC02	**33**	**41**	**49**	P_{lim}	P_{lim}	P_{lim}

where:- – outside the plan of experiment, P_{lim} – unstable combustion or misfire

Indicated efficiency of the engine is an significant parameter to report when discussing the indicated performance of an engine. In Figure 3 and 4 the indicated efficiency of the SI engine fuelled with natural gas and producer gas substitute (three mixtures) as a function of air excess ratio is presented.
The influence of fuel type and air excess ratio on the IMEP is presented on Fig. 5 and Fig. 6. IMEP values are always of prime interest in performance evaluation. The IMEPs for natural gas range between 6.6 bar (for λ=1.5) to 9.1 bar (for λ=1). When the producer gas is utilized the IMEP varied between 3.8 bar (for λ=1.95 and fuel composition HC1) to 6.9 bar (for λ=1 and fuel composition HC1) what is the maximum value obtained for producer gas. It is somewhat surprising because when the calorific value of air fuel mixture is compared at a range of air excess ratio (Fig. 1) the $e_{d,v}$ of producer gas composition HC1 is lower than for HC0.4. But for the air excess ratio λ=1 and λ=1.25 the utilization of the fuel composition HC1 brings higher engine indicated efficiency than fuel HC0.4 (see Fig. 4).

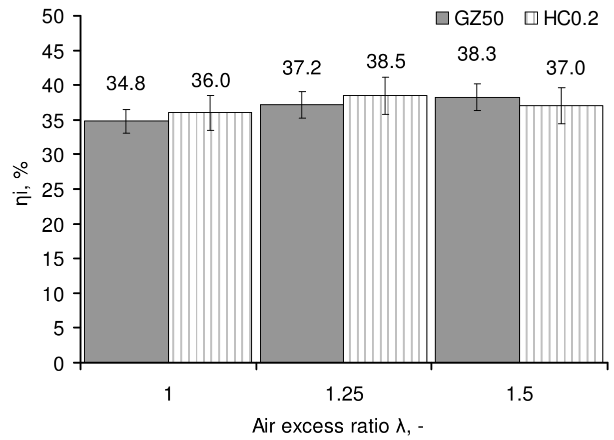


Figure 3. Indicated efficiency vs. air excess ratio utilizing natural gas and PG mixture HC0.2.

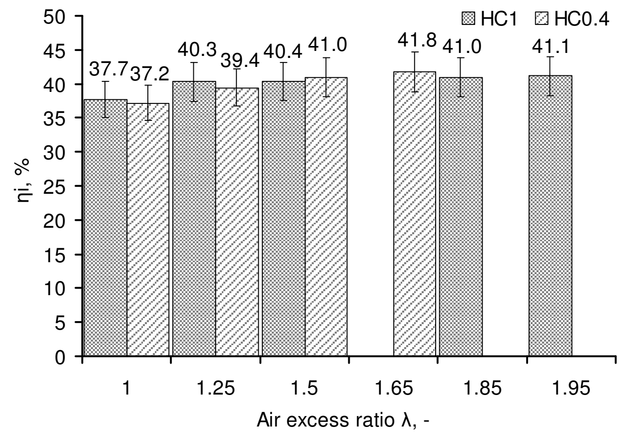


Figure 4. Indicated efficiency vs. air excess ratio utilizing PG mixture HC1 and HC0.4.

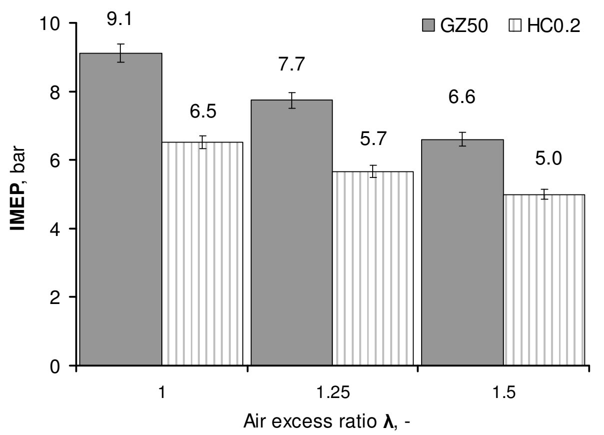


Figure 5. IMEP vs. air excess ratio utilizing natural gas and PG mixture HC0.2.

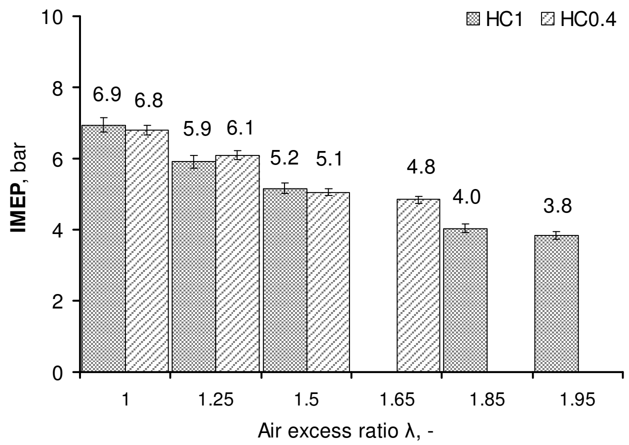


Figure 6. IMEP vs. air excess ratio utilizing natural gas and PG mixture HC1 and HC0.4.

The similar impact of air excess ratio change on the indicated efficiency value for all tested fuels was observed. For most tested fuels as the air excess ratio is increased and the indicated efficiency increased too. Exception is producer gas mixture HC0.2 (with the highest content of carbon monoxide from all used mixtures), for this fuel the maximum of indicated efficiency was achieved when the engine was operated with air excess ratio λ=1.25. However, for all tested fuels combusted with this

value of air excess ratio (λ=1.25) the highest increase of indicated efficiency was observed comparing the values obtained for stoichiometric air fuel mixture. In this range of air excess ratio (λ=1 to λ=1.25) the indicated efficiency increases averagely of about 2 percentage points. The increase of indicated efficiency for lean burn condition is mainly associated with lower level of in cylinder temperature what is directly related with lower heat losses during combustion period. The tested engine reached highest level of indicated efficiency when producer gas mixture HC0.4 has been burned with air excess ratio λ=1.65. It should be noted here that as the air excess ratio is increased the indicated mean effective pressure of the engine is decreased and it is the effect of calorific value change of air-fuel mixture (Fig. 1). Because the friction work of the engine internal parts is the same in all tested conditions covered by the experiment, the engine efficiency (based on the crankshaft power output) can be constant or can decrease with increase of air excess ratio.
In Fig. 7 and Fig. 8 the Coefficient of Variation (COV) of the IMEP values are depicted. In the constant air excess ratio one can see that the variation in combustion stability between the tested fuels is relatively small as the span of variations is 0.6 percentage point for λ=1.

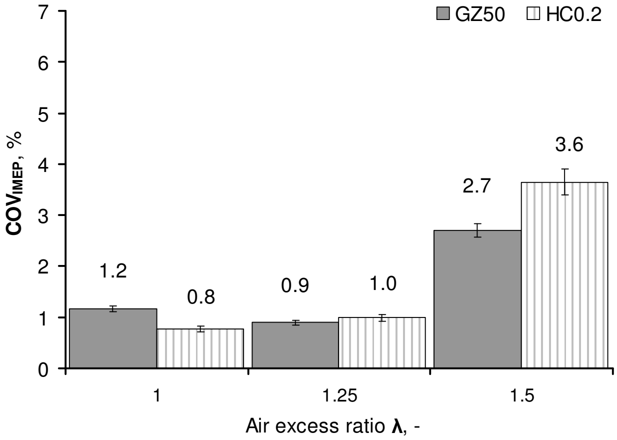


Figure 7. COV of IMEP vs. air excess ratio utilizing natural gas and PG mixture HC0.2.

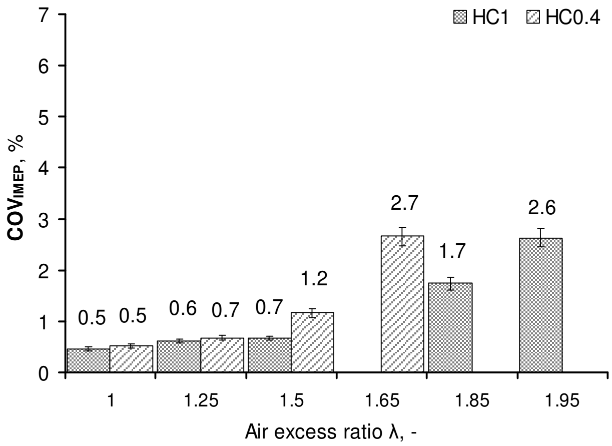


Figure 8. Indicated COV of IMEP vs. air excess ratio utilizing PG mixture HC1 and HC0.4.

The observed values of the COV for both fuels are acceptable and compare well to those reported in literature [7-10] For natural gas and producer gas composition HC0.2 (see Fig. 7) a COV of IMEP range of 1.2% (for λ=1) to 2.7% (for λ=1.5) for GZ50 and 0.8% (for λ=1) to 3.6% (for λ=1.5) for HC0.2 fuel.
For fuel composition HC1 a COV of IMEP range of 0.46 % (for λ=1) to 2.6% (for λ=1.95) what was the lowest value of COV obtained during all engine tests. Fuel HC1 includes most hydrogen in the composition comparing to the other producer gas mixtures. In this case the role of hydrogen in producer gas mixture is visible. The hydrogen can extend the flammability limits of the fuel and stabilising the combustion process of producer gas.

4. Conclusions

In this work the influence of air-fuel mixture change and a simulated biomass gas composition change that resembles in its composition so called producer gas for SI engine performance has been investigated. The engine performance was evaluated based on indicated performance parameters derived from in-cylinder cycle-resolved pressure traces.
It is shown that the simulated biomass gas can be stable burned in SI engine. The useful range of air excess ratio change is wider for producer gas mixtures with higher H_2/CO ratio. In particular the results indicate the following:

- The engine performance in terms of indicated efficiency is increased by about 2 percentage points when compared to fuelling with natural gas.
- The engine power de-rating when producer gas is utilized instead the natural gas, varies from 24% for producer gas HC1 to 28,6% for producer gas HC0.2 under stoichiometric combustion conditions. For lean burn (λ=1.5) the difference are lower and varies from 22% for producer gas HC1 to 24.5% for producer gas HC0.2.
- To obtain maximum value of IMEP the ignition timing should be advanced for the engine fuelled with producer gas if compare with the values for natural gas. The ignition timing must be more advancing for producer gas with low H_2/CO ratio and for lean mixture combustion.
- For most of engine working conditions the COV of IMEP has lower values when the engine is fuelled with simulated producer gas. The lowest values is obtained for stoichiometric combustion.

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Acknowledgments
This work is supported by the Silesian University of Technology from the statutory research source of Institute of Thermal Technology.