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THE LEAN MIXTURE COMBUSTION OF SIMULATED PRODUCER GAS IN SI ENGINE

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Abstract

In this paper the experimental results of spark ignited (SI) engine fuelled with lean mixture of simulated producer gas compared with chosen gaseous fuels are presented. The SI engine test fuelled with simulated producer gas, natural gas and simulated biogas with variable value of excess air ratio has been done. The experiments were carried out on the petrol engine with a low engine displacement. Typical SI engine was selected in order to evaluate the potential application of gaseous fuel (i.e. producer gas, natural gas or biogas). These types of engines are available on a wide scale and commonly used in automotive sector because of the low purchase price and operating costs. It is expected that after minor modifications, the engine can easily operate in micro CHP system. The main goal of this work is to determine the performance of the engine and its impact on the environment during the combustion of the lean producer gas mixtures. The study shows the impact of both the excess air ratio and the type of fuel used for engine performance and emission index. Combustion of lean mixtures of producer gas leads to an increase of carbon monoxide in the exhaust. Increasing the value of excess air ratio affects the growth of indicated efficiency of the engine.

Keywords: *micro CHP, combustion engine, in-cylinder pressure*

1. Introduction

The global climate changes are one of the greatest menaces which the effects we observe increasingly at last years. Constantly growing the combustion gas emissions contributes significantly to the greenhouse effect. Therefore, considering the above, it has become crucial to reduce carbon dioxide emissions. One of the possible solutions to this problem is utilization of renewable energy sources, mainly biomass. One method of efficient biomass using (e.g. agricultural and forestry residues) is the gasification process in a downdraft fixed bed gasifier [1, 2]. This type of device can generate so called a producer gas (low calorific value gas - LCV) that can be used in spark ignition engine for heat and power generation purposes.

Since the late 90's to 2012 is observed significant increase in electricity consumption in Poland, with the exception of minor fluctuations in some years. The increasing demand for electricity in Poland will most likely be to increase rapidly in the coming years. The confirmation of this assumption may be in Annex 2 to the "Polish Energy Policy until 2030" that is part of Resolution No. 202/2009 of the Council of Ministers of 10 November 2009. This annex is expected to increase the demand for electricity in the amount of 217.4 TWh in 2030 relative to consumption of 157.9 TWh in 2011. With regard to such of data may be advantageous use of local low-power cogeneration systems. High performance micro-cogeneration (mCHP - called Micro Combined Heat and Power) can be realized on the basis of SI engine fuelled with natural gas, biogas and gas from biomass gasification. In the light of Directive 2004/8/EC, micro-cogeneration concerns combined production of electricity from the power of the maximum amounting to less than 50 kW. Implementation of mCHP systems can significantly contribute to the low-emission heat and electricity in our country.

2. Gaseous fuels for micro CHP systems

The cogeneration systems at small scale powered by internal combustion engines can be used in residential, countryside buildings, public buildings, hospitals, and especially in places where there is long-term preferably year-round demand for heat. The use of micro-cogeneration is justified wherever meets the needs of a single customer or a significant number of consumers are small consumers of electricity and heat, which connect to the central power plant is economically unjustified.

The spark ignition SI or compression ignition CI engine can be used to the powering of mCHP system. Wherein the CI engine can be operated in dual fuel system, what means that the main fuel is gaseous fuel and the combustion process is initiated on a pilot dose of liquid fuel. Currently most applicable fuel to power the CHP system is natural gas, mainly due to its good accessibility. In recent years more and more is also increased interest in biofuels to power this type of unit. Depending on the availability of a particular fuel and the type of engine (SI or CI) the mCHP systems are typically powered by biogas, LPG gas or liquid fuel. In the case of liquid fuel usage the most commonly, CI engines are used.

In case of using the producer gas as a fuel to internal combustion engine, the very effective system removal of tars and particulates is needed. Tars removal is particularly necessary to avoid its condensation in pipes and other elements at inlet the appliance. The internal combustion gas engines (ICE) are more tolerant of contaminants than gas turbines. In particular, it is possible to have tar content up to 50-100 mg/Nm^3 for ICE and less than 5 mg/Nm^3 for gas turbines. Particulate levels must be reduced to below 50 mg/Nm^3 for ICE, whereas for turbines below 15 mg/Nm^3 [3, 4].

The small-scale cogeneration systems that use producer gas can be used in wood processing plants, such as sawmills and furniture factories. In those plants, it is possible year-round use of the heat from the mCHP system.

Out the world, there are several dozen of CHP plants based on gasification and ICE. Most of them are pilot plants, which are equipped in spark ignition engine. The most popular installations with SI engine in Europe can be listed: Harboore (Denmark), Güssing (Austria), Spiez (Switzerland), Kokemäki (Finland). Commercial solutions based on CI engine is offered by Mothermik Company (Germany). Due to a number of operational problems related with such kind of systems, very important is continuing the research in the field of use the producer gas as a fuel for internal combustion engines. Other important and current issues are optimisation of control system regarding to gas quality and variability. The effective organization of the combustion process, leads to the reduction the emission of harmful substances and increase energy efficiency.

Tab. 1. the main properties of tested fuels

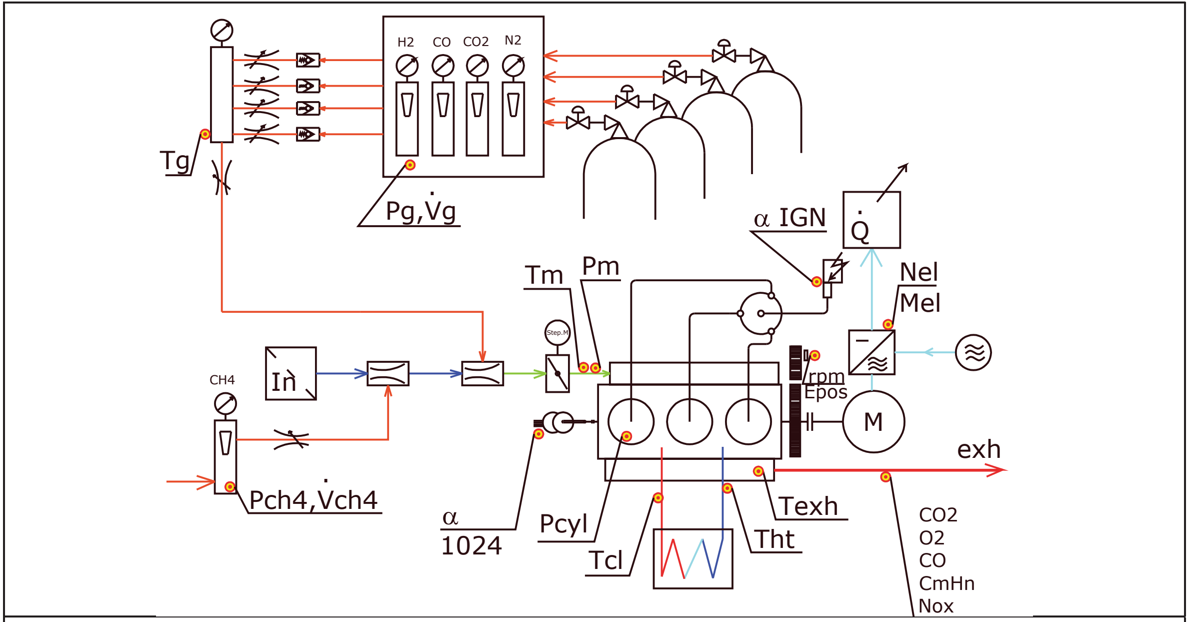
Fuel	Composition	LHV, MJ/Nm^3
Biogas	CH4 ≈ 60%, CO_2 ≈ 39%, other: C_2H_6, C_3H_8, C_4H_{10},	21.5
Natural gas GZ 50	CH_4 ≈ 98.5%, CO_2 ≈ 0.1%, N_2 ≈ 1%, other: C_2H_6, C_3H_8, C_4H_{10}	35.3
Producer gas	CO ≈ 27%, H_2 ≈ 16%, CO_2 ≈ 7%, CH_4 ≈ 4%, N_2 ≈ 46%	6.57

In this paper, the main attention has been focused on the combustion process of lean mixtures and comparing the results obtained in the stoichiometric conditions. The study was conducted for the three gaseous fuels, such as; gas from biomass gasification, natural gas and biogas. Both gas from biomass gasification processes (producer gas) and biogas were cleaned and simulated by mixing the proper ingredients stored in high-pressure cylinders. The main properties of the investigated fuels is reported in Tab. 1.

3. Engine tests and results

An overview of the engine test bench and measuring equipment is presented in Fig. 1. The main components of the experimental set up include:

- three cylinders SI engine with a capacity of 796 cm^3 and compression ratio equal to 9.3. The engine is naturally aspired 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:
T_g – gas temperature, T_m – air-gas mixture temperature, T_{cl} – engine coolant input temperature, T_{ht} – engine coolant output temperature, T_{exh} – exhaust temperature, P – CO_2 pressure, Pch4 – CH_4 pressure, P_{cyl} – in cylinder pressure, N_{el} – electric power, exh – exhaust gas composition, α1024ppr – encoder resolution, αIGN – ignition advance angle control system.

Fig. 1. The scheme of engine test rig with measurement equipment

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. The controller also has an adjustable spark plug discharge energy by changing the loading time of high voltage coils.

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 adjusting the amount of the removed heat, which is then transmitted to the local central heating system.

The pressure measurements in the first cylinder were performed using piezoelectric pressure transducer KISTLER type. 6117BF17. 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. Additional parameters measured during experiment include:

- flow rate and composition of the gas,
- pressure and temperature of the gas powered the engine,
- temperature of the oil and engine cooling liquid,
- ambient parameters (temperature, pressure, relative humidity),
- exhaust gas temperature,
- exhaust gas composition (dry products).

It should be noted that the engine was not equipped with the exhaust aftertreatment system. Therefore, the composition of the exhaust gas was measured directly at the outlet of the exhaust system.

In the first stage of the experimental data, handling the indicated mean effective pressure (IMEP) has been calculated using in cylinder pressure data. Unlike the stoichiometric mixtures, the combustion of lean mixtures leads to reduction in IMEP values. It is mainly due to a decrease in the heating value of the air-fuel mixture as the excess air ratio is increased.

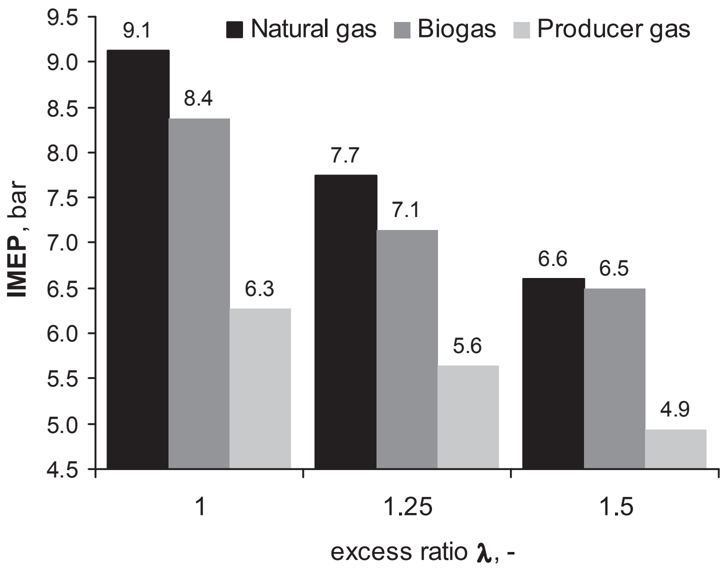


Fig. 2. Influence of air excess ratio and kinds of fuel on IMEP

In Fig. 2 the influence of air excess ratio and kinds of used fuel on indicated efficiency of the tested engine is presented. A significant increase of indicated efficiency during combustion of lean mixtures (λ = 1.5) is observed. It should be considered as a typical phenomenon, caused mainly due to lower heat losses, what accompanies during combustion of lean mixtures. Interesting is the fact that over the range of used gaseous fuels, the highest internal efficiency of tested engine is obtained for producer gas.

Higher values of internal efficiency engine powered by producer gas, can be caused by a better matching of the ignition advance angle. During the experiment, the ignition advance angle has been adjusted with steps of 5 deg for all tested fuels.

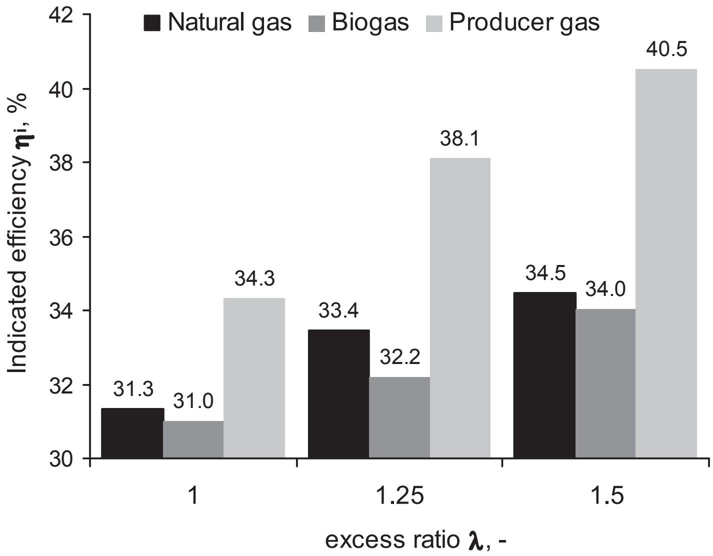


Fig. 3. Influence of air excess ratio and kinds of fuel on indicated efficiency

The emission index of harmful exhaust species, such as carbon dioxide (CO), hydrocarbons (HC) and nitrogen oxide as sum with nitrogen dioxide (NOx) is presented in Fig. 3-5.

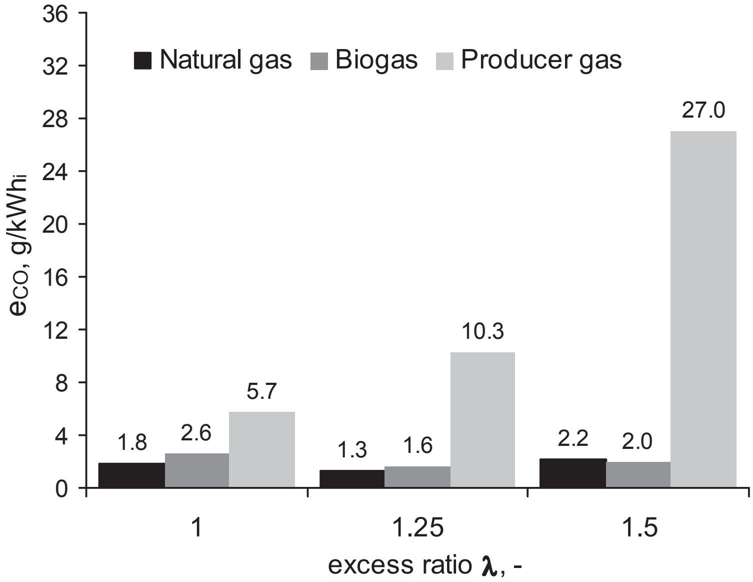


Fig. 4. Influence of air excess ratio and *kinds of fuel on the specific emission of carbon monoxide*

The values of emission index refer to internal power of the engine. Taking in to account the mechanical efficiency of the internal combustion engine and electromechanical efficiency of the electric generator, these values can be converted to the final result, what in the mCHP system is electricity. In this paper the attention is paid on the analysis of indicators characterized processes inside the cylinder of the ICE.

As can be seen from the results shown in Fig. 4, the CO emission increase with increasing air excess ratio, but only for the producer gas and methane. The air excess ratio have no significant influence on the CO emissions during powering the engine with biogas.

The HC emission increases with increasing the air excess ratio for all tested fuels. The highest values are obtained for methane, and the lowest for producer gas. Low values of HC emissions for procucer gas are directly associated with low methane content in its composition.

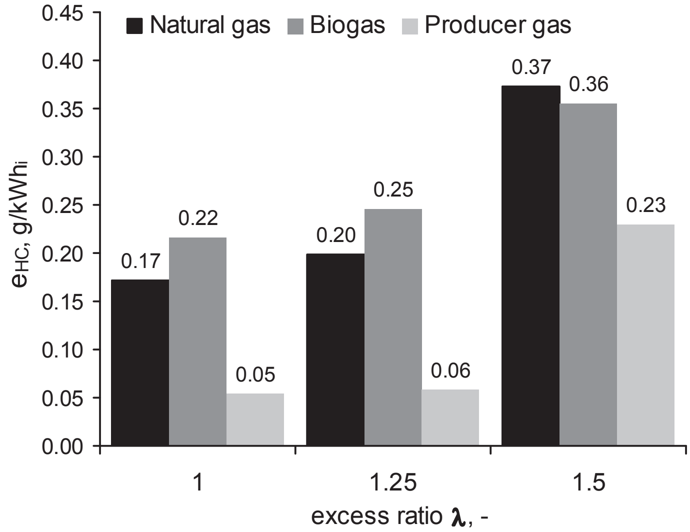


Fig. 5. Influence of air excess ratio and kinds of fuel on the specific emission of hydrocarbons

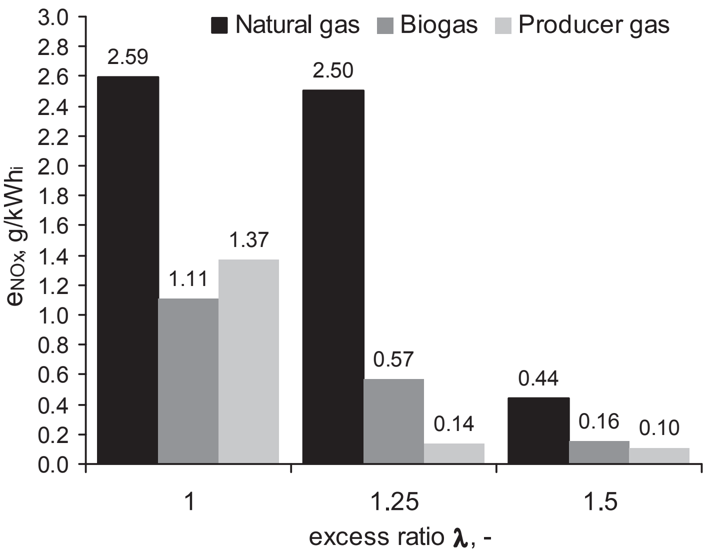


Fig. 6. Influence of air excess ratio and kinds of fuel on the specific emission of NOx

For all tested fuels, the NOx emission decreases with increasing the air excess ratio. It is mainly caused by decreasing the in cylinder temperature when the engine is operated on the lean mixture. The evident disproportion in the values of NOx emissions for the analysed fuel may be caused by different content of OH radicals and the values of local temperature peaks for a individual fuel.

In the Fig. 7 the impact of air excess ratio on the coefficient of variation ($COVl_i$) of indicated mean effective pressure (IMEP) for all investigated fuels is presented. This coefficient is very important for the smooth running of the electric generator due to production of electrical energy on expected parameters (frequency, amplitude voltage). The calculations were done using 100 working cycles of the engine. A clear influence of air excess ratio regarding to values of COVli was observed. For all tested fuels the COV values increase as the air excess ratio is increased. Higher variations of engine working cycles fuelled with leaner mixture is are mainly caused by lower presence of combustible components probability near spark plug electrodes. The lowest values of COV were obtained for producer gas, what is achieved probably by presence of hydrogen in the fuel, and enhanced by it probability of proper ignition of the air gas mixture.

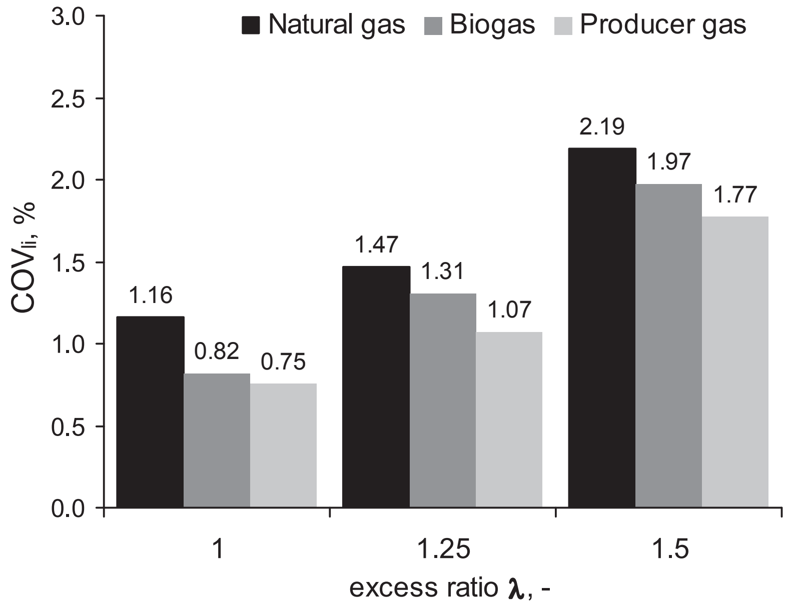


Fig. 7. Influence of air excess ratio and kinds of fuel on coefficient of variation of indicated work

It is a bit surprising fact that the powering by natural gas leads to a higher value of COVli, relative to biogas. The one reason can be the higher concentration of methane around the spark plug when running on biogas, which can occur due to the higher concentration the components with lower density in biogas-air mixture, relative to methane-air mixture.

4. Conclusions

The study shows the impact of both, the air excess ratio and type of used fuel on the internal combustion engine performance. Combustion of lean air-producer gas mixtures leads to increasing of carbon monoxide emission in the exhaust. Increase the air excess ratio has an impact on increase of the engine indicated efficiency. It is mainly caused by lower heat loss to the environment during the combustion process.

Significantly higher content of CO in the exhaust gas during fuelling the engine with producer gas compared to other fuels can be related with a lower content of OH radicals what can be associated with the lowest presence of NOx at the same time in the range of lean mixtures.

Of course, the use of fuel with a low calorific value is associated with a decrease in engine power at a specific crank shaft speed. The difference in the reduction of engine performance fueled by biogas and producer gas, as regards to methane (as a high caloric value fuel), decreases in the direction of lean mixtures. It is directly caused by the decrease of the difference in the calorific value of air gas mixture of these fuels.

The unrepeatability of the engine working cycles, fuelled by tested fuels is at an acceptable level. For the internal combustion engines powered the electric generator, the COV_{li} should be less than 5%.

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