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Analysis of Potential Application of Biogas Fuel in Modern Compression-Ignition Engines

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Abstract: Limited fossil fuel supplies and the necessary reduction in toxic fumes emission to the atmosphere are the main motives in conducting a search for the new, effective energy supplies. The one with potential is biogas. It is the product of natural fermentation processes of municipal waste in landfills or is produced in biogas plants out of agricultural and green waste. Due to creation under different conditions, its chemical composition varies. This is enormous obstacle in its effective application. Biogas is easily applied to fuel spark-ignition engines however intensive attempts are made to employ it in much more effective compression-ignition engines. Application of biogas require the use of dual-fuel CI engine. The point of the research described in this paper is to show the influence of different methane-carbon dioxide composition ratio in biogas on dual-fuel CI engine effectiveness.

Keywords: biogas, compression-ignition engine, toxic fumes emission.

Conference topic: Environmental protection.

Introduction

The aim of reduction in toxic fumes and greenhouse gases emission to the atmosphere require to search for the new fuels which could drive modern combustion engines. The one of renewable fuels with great potential is biogas (Mikulski, Wierzbicki 2016; Wierzbicki 2012, 2016).

The main component of biogas is methane. It is the product of natural fermentation. Under conditions of high humidity and temperature the bacterial activity brakes complex organic compounds down to simpler organics and inorganic compounds such as methane and carbon dioxide in abundant amounts. The main sources of this fuel are agricultural biogas plants, sewage plants and municipal waste landfills. Besides these, it is the product of all natural decay processes (Wierzbicki 2012).

Biogas can be easily implemented to fuel spark-ignition engines because of its properties. However its use in powering much more effective compression-ignition engines is inhibited due to the high temperature of methane (main biogas component) ignition. In this case it is required do use dual-fuel fuelling system to feed it to the CI engine. The second ingredient is small dose of liquid fuel, which causes the ignition in combustion chamber (Wierzbicki 2012).

From the economic and toxic emission point of view it is advised to limit to the minimum the intake of gas oil dose. When fuelled with gas, the medium size CI engines' work is stable until the pilot dose is kept within 10–15 % of overall energy supplied with fuel (Hountalas, Papagiannakis 2001). The results shown by Albrecht (Albrecht 1995) indicate that possible reduction of an initiation dose to less than 5% is possible in stationary engines under strict load range. Smaller ignition doses lead to unsteady engine work and increased emission of THC, confirmed by (Alla *et al.* 2000; Basavarajappa, Banapurmath 2015).

Stable work of dual-fuel engine under set load, with the minimum usage of gas oil requires to control gas and liquid fuel ratio. Moreover is it wise to control the liquid fuel injection timing advance. Alternating this parameter in relation to rotation speed and engine load results in noticeable increase in engine performance (Wierzbicki 2016).

Recently, there have been some research undertaken about biogas application in dual-fuel engines (Barik, Murugan 2014; Barik, Murugan 2016; Bora, Saha 2016; Makareviciene *et al.* 2013; Sorathia, Yadav 2012; Wei, Geng 2016). Most of those papers set limits to biogas composition ratios, enrich biogas with methane or refine it in order to increase methane share.

The greater part of papers about biogas application in CI engines focus on influence of individual control parameters on engine effectiveness. Some conclusions can be drawn from those studies, however most of them employ different engines with distinct injection systems and combustion chambers' geometries as well as non-identical engine rotation speed. Those discrepancies preclude direct comparison of the results.

Bari (Bari 1996) draws the following conclusion from his research: increase of carbon dioxide in biogas composition over 40% leads to unstable engine work, significant power loss and toxic compounds emission rise. In summary he advises to remove the CO_2 from biogas in order to enhance engine performance. Similar results were presented by Van (Van *et al.* 2015). In addition, Van claims that high CO_2 share in biogas results in slower combustion inside the engine cylinder.

Bora and Saha (Bora, Saha 2016) conducted their research on small single cylinder engine with constant liquid injection timing advance and power of 3,5 kW. Other set condition were: rotation speed n = 1500 rpm, 60% CH_4 share in biogas. The engine construction allowed for a change in compression ratio from 17 to 18,5. During analysis a significant change is observed in combustion chamber pressure when fuelled with dual-fuel in comparison to gas oil feed only. First of all, a decrease in charge pressure within combustion chamber is noted. It is attributed to chemical composition change. The result is a delayed ignition and lower maximum pressure within the chamber than when fuelled by gas oil only in respect to TDC. The analysis of chemical composition of the fumes confirms a rise in carbon monoxide CO and THC and a decrease in nitrous oxides NOx when power by dual-fuel. However, an increase in compression ratio leads to decreased CO and THC emission and increased NOx emission.

Thorough research in the subject of the application of biogas with varying methane share in fuel composition was presented in (Makareviciene *et al.* 2013). Turbocharged, four cylinder type 1Z (1.9TDI) engine with electronically controlled Bosch V37 injection pump system was employed. Biogas with methane ratio of 65 to 95% was acting as gas fuel. Constant dose of gas fuel was used with maximum 60% share in the whole dose. The outcomes confirm the increase in CO and THC emission as well as a decrease in engine efficiency when powered with dual-fuel. Authors highlight that the engine works noticeably better when gas oil pilot dose injection timing advance is increased.

Comparable results were obtained in (Bari 1996; Mustafi *et al.* 2013; Yoon, Lee 2011). It is emphasized that application of gas fuel in engine changes the combustion process and increases CO and THC emission. The decrease in NOx emission and less smoke in exhaust fumes is noted. The increase of gas fuel in the whole dose results in lower cylinder pressure and temperature by the end of compression stroke. It is caused by a change in specific heat of a charge. This leads to the delay of self-ignition of the liquid fuel, lowers pressure in combustion chamber and delays it reaching the maximum. It is suggested that this engine type performance could be improved by appropriate control of pilot dose injection.

The analysis of presented papers about biogas application in CI engines forced authors of this article to come to the conclusion that most studies conducted worldwide lack the influence of varying chemical composition of biogas on engines performance. The causes for this could be difficulties in developing a way to control changing chemical composition of biogas during tests. A test bench with such capabilities was built.

The aim of this paper is to assess the biogas chemical composition changes effect on dual-fuel compression-ignition engine performance.

Charge parameters in combustion chamber

The process of generation of fuel mixture and its combustion in dual-fuel CI engines is complex and dependent on gas fuel share in whole dose and the parameters of liquid fuel injection stream.

The most important parameters describing the charge in combustion chamber in dual-fuel engines are the following coefficients [18]:

λ – Total Air–fuel ratio (AFR): for the whole dose in the combustion chamber: gas and liquid fuel and air;

λ_o – AFR for the mixture of gas fuel and air

Those coefficients can be described as follows:

– Total Air–fuel ratio (AFR): for the whole dose in the combustion chamber: gas and liquid fuel and air:

$$\lambda = \frac{\dot{V}_p}{\dot{V}_g \cdot L_{tg} + \dot{m}_{df} \cdot L_{tdf}}, \tag{1}$$

where: $\dot{V}_p$ – air feeding stream [Nm^3/min], $\dot{V}_g$ – gas fuel feeding stream [Nm^3/min], $\dot{m}_{df}$ – liquid fuel feeding stream [kg/min], L_{tg} – theoretical gas fuel air need [Nm^3/Nm^3], L_{tdf} – theoretical liquid fuel air need [Nm^3/kg];

– AFR for the mixture of gas fuel and air:

$$\lambda_o = \frac{\dot{V}_p}{\dot{V}_g \cdot L_{tg}}. \tag{2}$$

The coefficients mentioned above allow to understand the complex conditions occurring during burning process, however they can be ambiguous.

Total AFR λ describes the capability to perfect combustion of both liquid and gas fuel. Its value should be close to the value of AFR λ of the traditional CI engine. This lowers the risk of engine heat overload [18].

If assumed that the whole gas fuel dose is injected and the air is fed, there should be homogeneous mixture of gas and air in the combustion chamber by the end of compression stroke. Its composition should therefore be close to λ_o coefficient.

In case of biogas application those coefficients are dependent on the ratio of gas and liquid fuel in the mixture and on gas fuel chemical composition.

Testing bench

The study was conducted on YANMAR L100N6CA1T1CAID compression-ignition engine. It was mounted on AUTOMEX AMX211 performance tester. Objects' basic data are presented in Table 1. Figure 1 shows testing bench. Genuine fuelling system was replaced with custom designed Common Rail type system. It allowed for precise control of liquid fuel pilot dose injection parameters (Śmieja *et al.* 2013; Wierzbicki *et al.* 2013).

Table 1. YANMAR engine technical specification.

Engine type	L100N6CA1T1CAID
Cylinders	1
Capacity	435 cm^3
Compression ratio	20
Piston diameter/stroke	86 / 75 mm
Max power	7,4 kW
Max torque	27 N·m
Max rotation speed	3600 rpm
Injection	Direct
Cooling system	Air

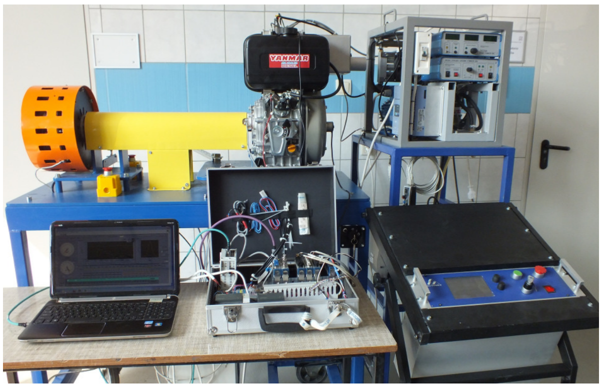

Fig. 1. Test bench view

A mixture of natural gas (CNG) with 98% methane content and 99.2% technical carbon dioxide (CO_2) was used as biogas gas fuel. In order to achieve close to biogas mixture a system of supply was developed. It allowed to easily manipulate the chemical composition by a means of a set of valves and mass flow controllers. Figure 2 shows the diagram for the supply system.

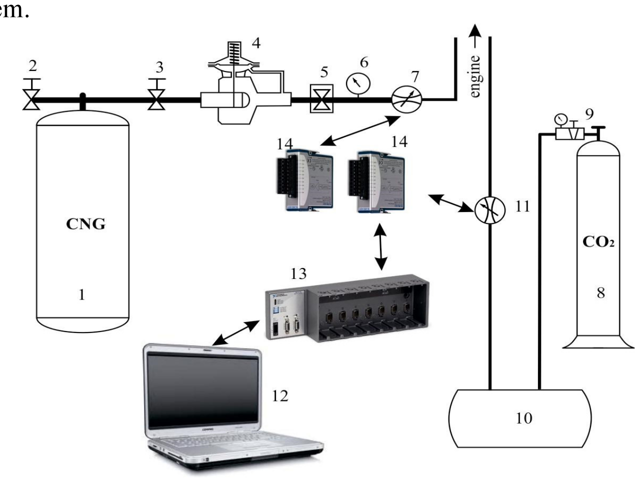

Fig. 2. Gas fuel mixture supply system. 1– CNG gas cylinder, 2 – fill up valve, 3 – cut off valve, 4 – two-stage reducer, 5 – solenoid valve, 6 – manometer, 7 – mass flow controllers MasStream, 8 – CO_2 gas cylinder, 9 – reducer, 10 – expansion tank, 11 – mass flow controller Bronks, 12 – PC, 13 – programmable controller CompactRio, 14 – input-output card adapters

Results

The research was conducted on the presented test bench. Its goal was to study compression-ignition dual-fuel engine performance powered by gas fuel with chemical composition close to that of biogas. Due to the fact that biogas chemical composition varies with raw material makeup, manufacturing technology and its refinement or enrichment the test was performed with the carbon dioxide changes in mixture between 10–50%.

Firstly, λ and λ_o coefficients for different compositions and shares of gas fuel were calculated. The liquid fuel dose was constant and equal to 10% of the fuel dose used during nominal load engine work. Figures 3 and 4 show those results.

The results shown in Fig. 3 indicate that mean total AFR coefficient λ in combustion chamber is proportional to the share of gas fuel in the whole dose and to the methane in gas fuel. The minimum value of λ coefficient is 1,3. It is close to the value of λ coefficient for the gas oil only CI engines.

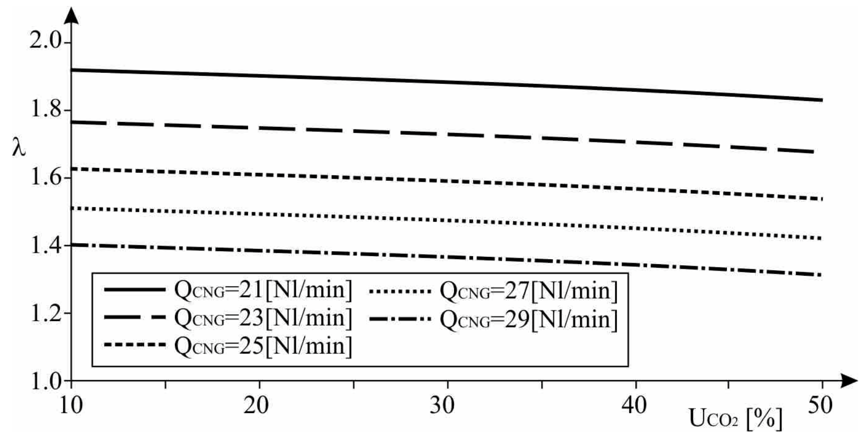

Fig. 3. Total AFR (λ) changes in relation to different CO_2 share in gas fuel. Different lines indicate different methane dose. Liquid fuel dose is 10% of the nominal value

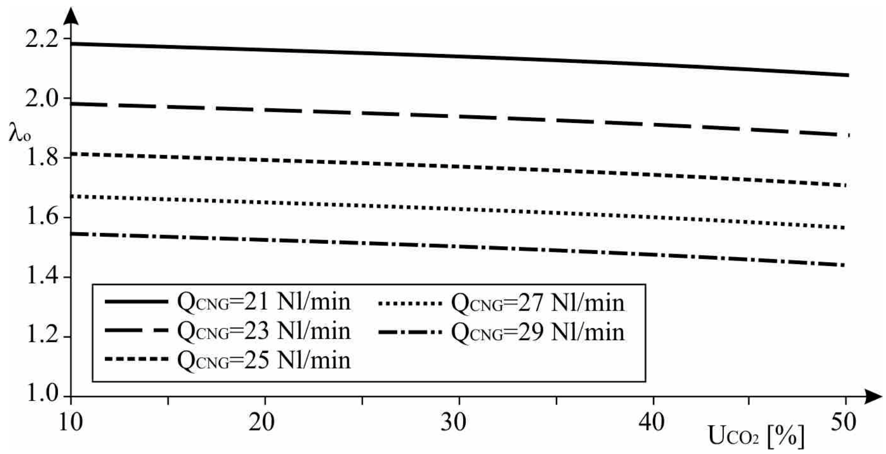

Fig. 4. AFR changes in relation to different CO_2 share in gas fuel. Different lines indicate different methane dose

Figure 5 shows the gas fuel CO_2 share influence on engine torque and efficiency at constant liquid fuel dose. It can be concluded that a rise in CO_2 share leads to lowering the engine efficiency. It is explained that carbon dioxide presence causes the decrease of temperature in cylinder during the compression stroke. This usually ends with the delayed ignition of liquid fuel. In addition CO_2 seizes the heat during the combustion and decreases oxygen content in cylinder which disturbs the flame distribution.

Figure 6 presents changes of toxic compounds contents in exhaust fumes in relation to gas fuel CO_2 share. A rise in CO emission is linked to CO_2 share increase, due to impoverishment of gas-air mixture. Percentage CO content in exhaust fumes rises with the decrease of injection timing advance. The concentration of NO_x in emitted fumes lowers with the increasing of CO_2 in fuel. It is the effect of lower temperature in the combustion chamber.

Figure 7 shows the relation of the size of natural gas dose and gas oil injection timing advance on torqe and engine efficiency. The rotation speed is n = 3000 rpm, 10% gas oil pilot dose, injection pressure of 50 MPa and 66% of CNG content in gas fuel.

The findings clearly indicate that a rise of gas fuel content in the whole dose leads to engine efficiency improvement. It is the effect of richer gas-air mixture and thus better combustion. Pilot dose injection timing advance influences engine efficiency. The highest engine efficiency was recorded for the greatest gas fuel content and pilot dose injection timing advance equal to 22° before TDC. Anything more and the engine work becomes unstable.

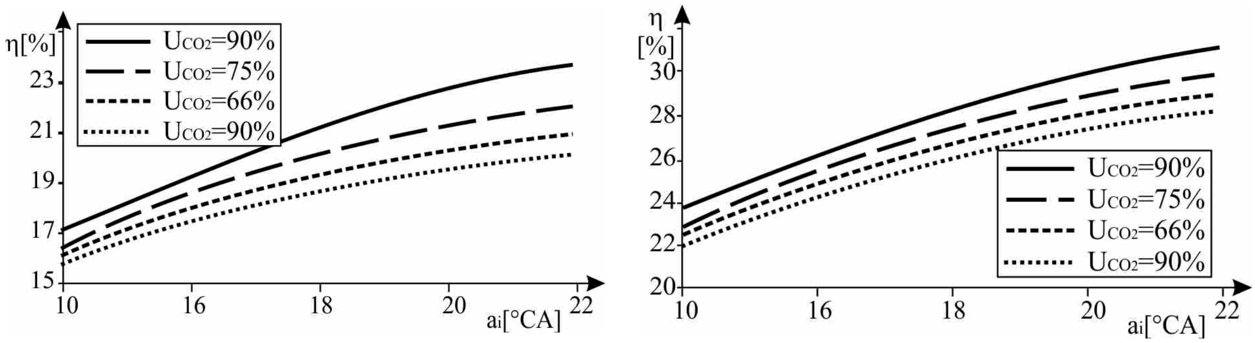

Fig. 5. The influence of gas fuel CO_2 content on the torque.
Different lines indicate different liquid fuel injection timing

Data shown in 8 clearly indicate that rise of gas fuel in the injected dose share and increase in gas oil pilot dose injection timing advance leads to decreased content of CO and THC in exhaust fumes. It is caused by mixture enrichment and improvement in combustion conditions. On the other hand improved combustion conditions and sooner pilot dose injection prompt the rise of nitrous oxides. Higher combustion temperature inside the chamber can be blamed.

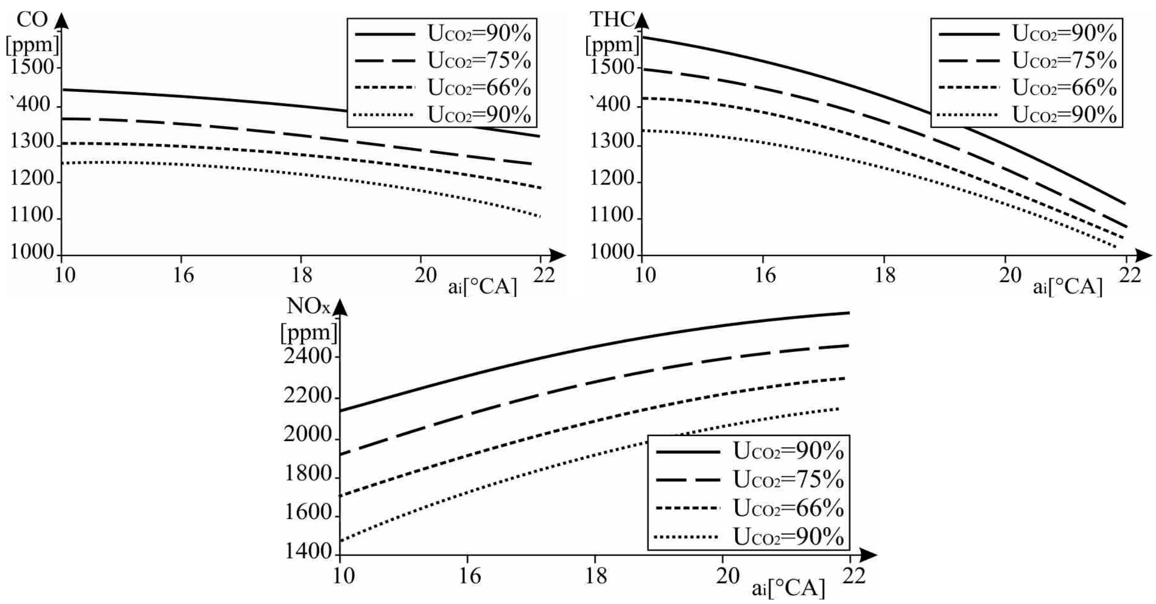

Fig. 6. The influence of gas fuel CO_2 share on the content of toxic compounds in exhaust fumes.
Different lines indicate different liquid fuel injection timing.

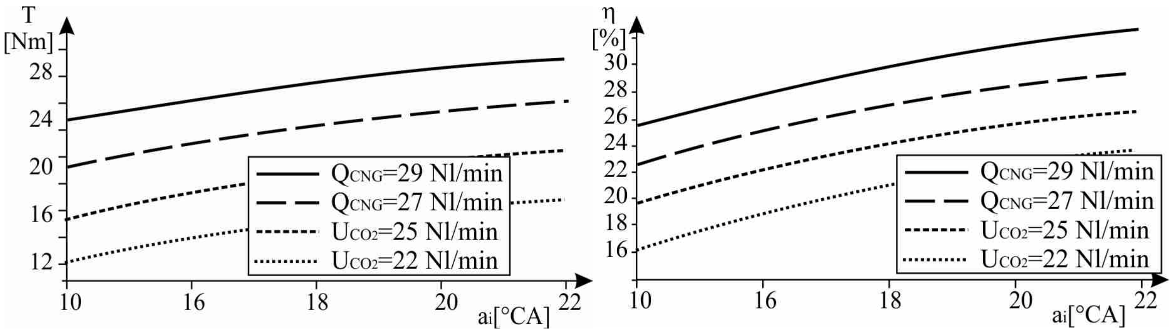

Fig. 7. The Influence of injection timing on the torque and efficiency. RPM = 3000. Liquid fuel share = 10%, methane share = 66%, CO_2 share = 34%. Different lines indicate different gas fuel dose

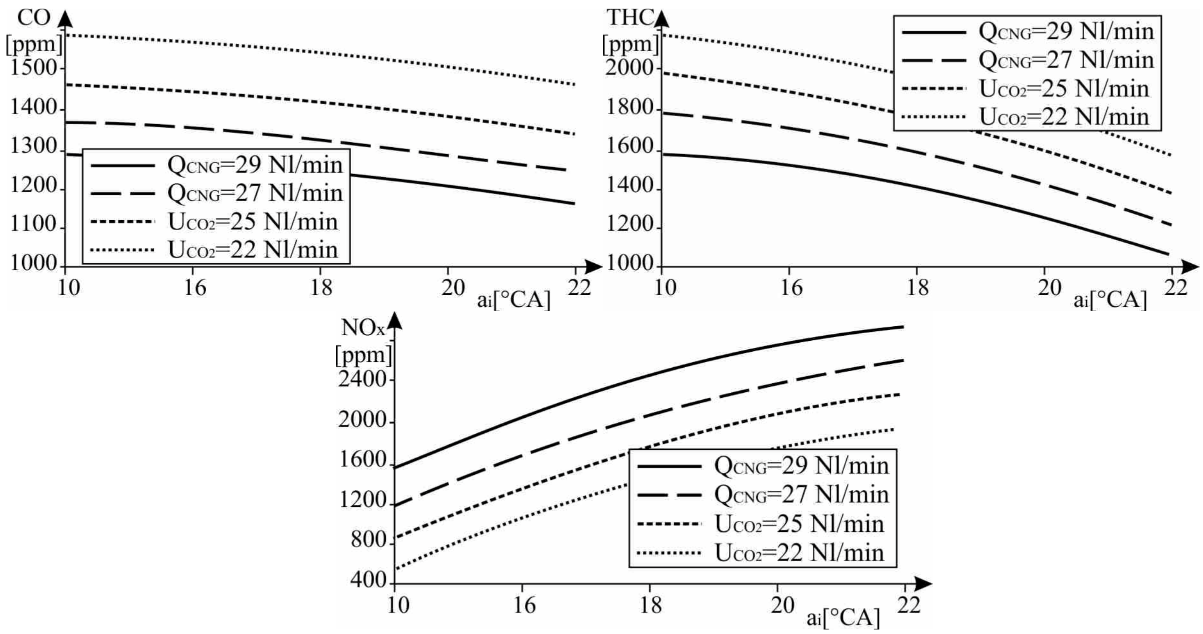

Fig. 8. The Influence of injection timing on the content of toxic compounds in exhaust gas. RPM = 3000. Liquid fuel share = 10%, methane share = 66%, CO_2 share = 34%. Different lines indicate different gas fuel dose

Conclusions

The results of the research presented in this paper allow to conclude that it is possible to effectively employ biogas as main power source in dual-fuel CI engines. The highest efficiency rates were recorded when the engine worked under heavy loads. They are even higher than when this engine works under those same loads in gas oil only configuration. The engine performance is influenced by CO_2 content in gas fuel. The increase of CO_2 share in biogas leads to lowering the overall engine efficiency. High dual-fuel CI engine efficiency under heavy loads when powered by biogas qualify it to be employed in cogeneration or hybrid systems where it can be limited to work under certain loads.

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