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EFFECT OF PILOT CHARGE SIZE AND BIOGAS COMPOSITION ON THE OPERATING EFFICIENCY OF A DUAL-FUEL COMPRESSION-IGNITION ENGINE

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Abstract

Reduction of greenhouse gases emissions into the atmosphere, as well as increasing the share of renewables in the overall energy balance, forces the search for new, alternative energy sources. One of the fuels, which presents high potential for combustion engines are biomethane or biogas, with methane as the main flammable component.

Biogas can be obtained from different products and using a variety of technologies which results in its wide availability and relatively easy manufacture.

The largest sources of biogas can be animal farms or sewage treatment plants and waste dumps in which significant quantities of biogas are obtained as a result of naturally occurring processes. Biogas can also be obtained from processing of energy crops or waste processing in agricultural, food and meat processing plants.

In this article, the possibility of using biogas as a fuel for CI engines has been examined. In such engine, combustion of biogas (methane) requires the use of dual fuel supply system, in which in addition to methane, liquid fuel is injected into the combustion chamber, in order to initiate the self-ignition of gaseous fuel. The paper presents exemplary results of the impact of the proportion of different fuels and biogas composition on the efficiency of the engine work cycle.

***Keywords**: CI engine, biogas, dual-fuel engine, pilot dose, efficiency of the engine*

1. Introduction

Reducing greenhouse gas and toxic compound air emissions is currently one of the fundamental challenges to humankind. A number of rigorous standards and laws aimed at not only curbing toxic compound emissions, but also promoting new eco-friendly solutions in the power and automotive industries have been introduced in recent years [1, 4, 6, 9].

The search for new alternative and eco-friendly fuels is favoured not only by the rise in ecological awareness and concern for the environment, but also by shrinking oil reserves and a substantial increase in prices in the world markets in recent years. Additionally, interest in the use of alternative fuels is favoured by substantial subsidies launched by state governments and international organizations for companies investing in power engineering based on alternative, renewable energy sources. Current legislation on the use of renewable fuels, e.g. Directive 2001/77/EC of the European Parliament and of the Council of 27 September 2001 on “the promotion of electricity produced from renewable energy sources in the internal electricity market,” requires a 20% share of energy to be produced from renewable energy sources in the overall energy balance by 2020 [2, 3, 6, 9].

One of the sources which can be used for energy production is biogas, which not only can be obtained by intentional processing of biomass and waste, e.g. from animal production, but substantial amounts of this gas form spontaneously as a result of phenomena occurring in sewage treatment plants or dumping sites. Depending on the production technology and the used raw materials, the forming biogas may have different chemical composition, different combustible component contents and therefore also variable calorific value. The basic combustible biogas component is methane, whose percentage ranges from 40-75%. Besides methane, biogas contains substantial amounts of carbon dioxide and small several percent amounts of oxygen and nitrogen. Therefore, depending on methane content, the calorific value of crude biogas ranges from 15-27 MJ/m^3. Another important parameter for biogas as engine fuel is the methane number, which is the equivalent of the octane number for liquid fuels. This number is 100 for methane; however, because of a substantial content of non-combustible compounds in biogas such as nitrogen and carbon dioxide, which increase the methane number, this number is typically around 130 for biogas [4-6, 9].

Biogas purification technologies are also currently able to obtain gas with over 90% methane content; purified gas (called biomethane) can be pumped into the gas grid and transported over long distances.

Because of its properties, biogas is used, above all, as fuel for spark-ignition engines. This way of using biogas is currently most often applied in sewage treatment plants and dumping sites. These engines, as a rule, drive power generators or cogeneration units [8-10].

2. Feeding compression-ignition engines with gaseous fuels

Compression-ignition engines are characterized by much higher efficiency and lower sensitivity to fuel quality compared to spark-ignition engines. Unfortunately, despite these characteristics, the use of gaseous fuels in such engines poses many problems.

The direct reason for the impossibility of using both biogas and methane in compression-ignition engines is the relatively high methane ignition temperature (over 900 K). One of the methods for using gaseous fuels containing methane in compression-ignition engines is the application of a dual-fuel system. In this solution, a mixture of air and gaseous fuel is sucked into the combustion chamber and a small liquid fuel charge, which serves as the charge igniting the air/gas mixture, is injected into the combustion chamber towards the end of the compression stroke [1, 5, 7].

In this engine fuelling solution, combustion proceeds differently than in modern compression-ignition engines. Currently used compression-ignition engines have complex control algorithms and enable fuel injection in several charges during one work cycle. This solution controls the course of fuel combustion by graduated injection, which improves engine performance. For dual-fuel engine operation, the control of the course of fuel combustion is limited. As soon as, the first fuel charge is injected and ignited, the air/gas mixture in the combustion chamber starts to burn. The further course of this mixture's combustion depends only on the conditions in the combustion chamber. The main parameter determining the course of this mixture's combustion is the excess air coefficient. Methane forms a combustible mixture with air in the range of $\lambda = 0.6\text{-}2.0$. Because gaseous fuel is fed to the suction manifold, it forms a homogeneous mixture with air in the combustion chamber. For CI engines operating at partial loads, a lean mixture is therefore in the combustion chamber, which may not burn fully. In this case, the mixture within the liquid fuel stream is most often combusted. It is therefore extremely important to properly select the parameters of the liquid fuel stream [1, 5-7, 11].

It should also be remembered that when the engine is fed with gaseous fuel, a part of the combustion chamber is filled with gaseous fuel, which has a much higher volume compared to liquid fuel. Therefore, the volume of air sucked into the combustion chamber decreases, which reduces the burnable fuel charge and this, in turn, diminishes the maximum engine power. While it is true that real CI engines operate at high global excess air coefficients, it should be remembered that, in this case, an air/gas mixture with a high λ coefficient is formed in the dual-fuel engine, which considerably hinders its combustion. Fig. 1 shows the change in the theoretical combustion chamber air filling coefficient (at $\eta_v = 1$) when the engine is fed with gaseous fuel with variable methane content.

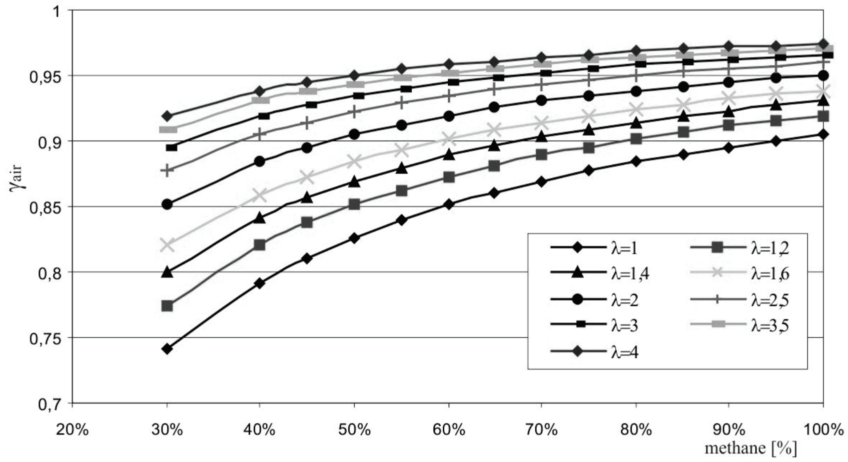

Fig. 1. Changes in the combustion chamber air filling coefficient depending on the biogas methane content with different λ coefficients

3. Test stand

To determine the effect of individual liquid fuel pilot charge parameters on the operating efficiency of a dual-fuel compression-ignition engine, a test stand was built allowing free regulation of the size and composition of the gaseous fuel charge as well as the injection of a pilot charge with the adopted parameters.

A Yanmar L100N6 single-cylinder compression-ignition engine was used for the tests and the basic data of this engine is compiled in Tab. 1.

Tab. 1. Basic technical parameters of the Yanmar engine

Engine type	L100N6CA1T1CAID
Displacement	534 cm^3
Compression ratio	20
Piston stroke/diameter	86/75 mm
Max power	7.4 kW
Max torque	27 Nm
Max speed	3600 min^{-1}
Injection type	direct
Cooling system	air cooling

The original mechanical fuel system of the tested engine was replaced with a laboratory common rail fuel system controlled from the level of a PC application [8, 10]. The used system allowed free regulation of the following system operation parameters:

- common rail fuel pressure value,
- fuel charge size, adjusted by the injector opening time with an accuracy of 1 μs,
- fuel injection advance angle, adjusted every 0.35°CA,
- the possibility of free regulation of the voltage and current changes at the injector coil.

To ensure that a methane/carbon dioxide mixture with the adopted proportions is fed to the engine, a preparation system was developed for this mixture. Both methane and carbon dioxide were fed from cylinders containing these gases. After the pressure was reduced by pressure regulators, the gases were fed to mass flow controllers, which were responsible for adjusting the flow rate of individual gases. The gases were then fed to a special mixing chamber, where they were mixed with air fed to the engine. An application operated from the PC level was used to control the flow rate of the gases.

The tested engine was mounted on an Automex AMX211 engine test bed. The used test bed allows, among others, stabilization of engine revolutions, which considerably facilitates tests on engines under defined conditions. The test bed is controlled by an individual operator panel. However, because of test bed controller communication with the operator panel by a CAN network (Controller Area Network), it is possible to control brake operation from the level of a PC application [8, 10].

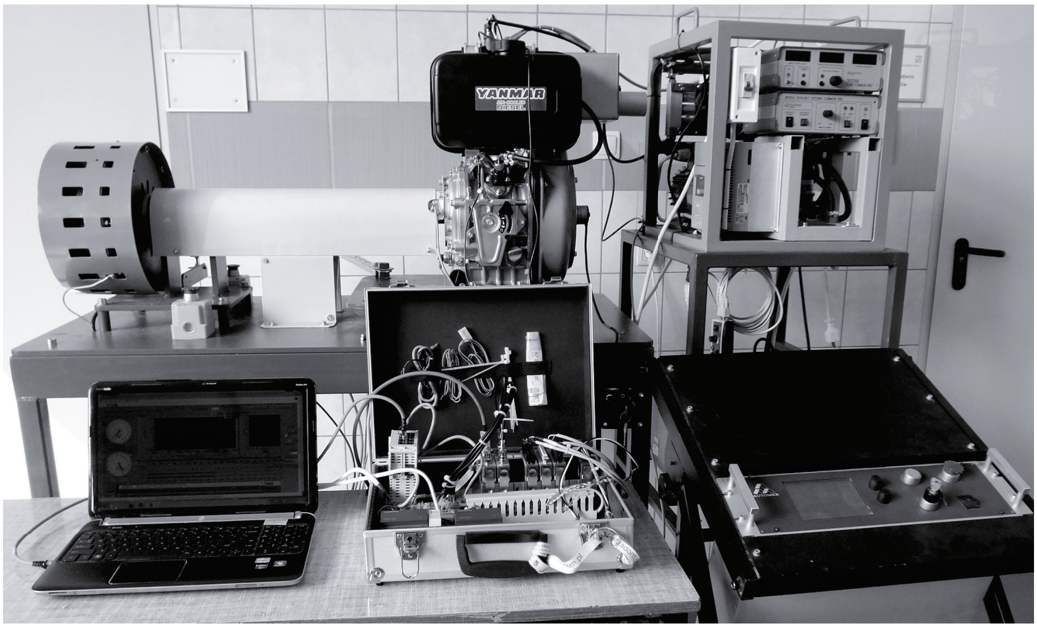

Fig. 2. Test stand

A special application was developed in LabVIEW to integrate the control of individual engine fuel systems and test bed operation. A cRIO controller allowing real-time operation was used for control. A detailed description of the test stand is presented in [8, 10].

4. Conducted tests

A series of tests was conducted at the above-described stand to determine the effect of pilot charge injection parameters and gaseous fuel charge composition. A mixture of CNG, the methane source and CO_2 was used as the gaseous fuel during the performed tests. The work gases from cylinders, after pressure reduction by regulators, were fed to the mixing chamber before the air filter, where they mixed with air fed to the engine. The flow rate of individual gases was adjusted by mass flow controllers, which ensured the adopted gaseous fuel composition.

The test results presented below were obtained at the following constant parameters:

- engine speed – 3000 ± 20 rpm,
- liquid fuel temperature – 45 ± 5°C,
- pilot charge injection pressure – 50 MPa.

The following were changed during the tests:

- size and injection advance angle of the diesel fuel pilot charge,
- size and chemical composition of the gaseous fuel (mixture of CH_4 and CO_2).

Fig. 3 shows changes in the torque of the tested engine at a pilot charge of 3 mm^3 per injection, at different methane charges (a mixture of 66.6% CNG and 33.3% CO_2 was used as the gaseous fuel in this case), for different pilot charge injection advance angles. Fig. 4 shows the recorded torque changes at a constant liquid fuel charge of 3 mm^3 per injection and a methane charge of 29 Nl fed in a mixture with CO_2 in different proportions. Torque changes at a pilot charge of 2 mm^3 and similar gaseous fuel parameters are shown in Fig. 5.

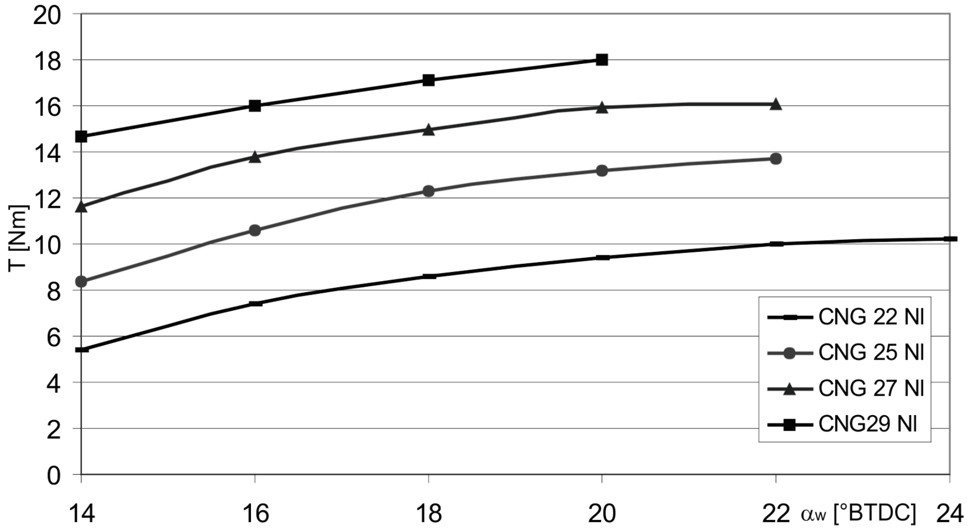

Fig. 3. Engine torque change at a pilot charge of 3 mm^3 and different methane charges for different pilot charge injection advance angles, (NL – normal litters gas volume under normal conditions: pressure 1013.25 hPa and gas temperature 273.16°K)

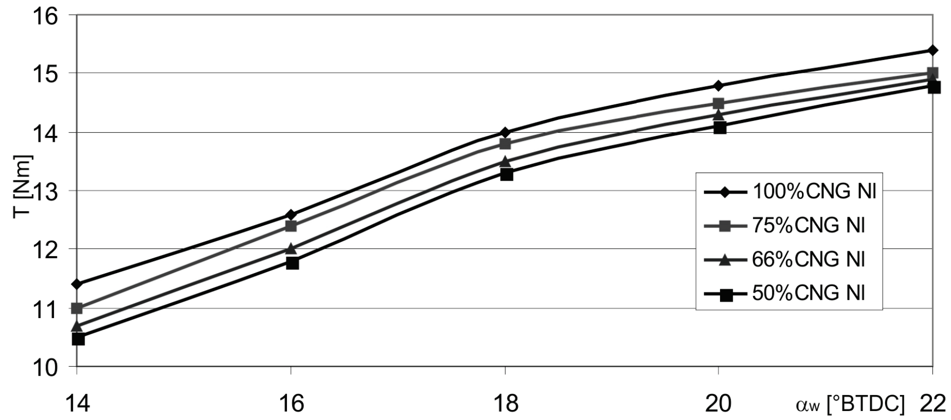

Fig. 4. Engine torque change at a pilot charge of 3 mm^3 and a constant methane charge fed in a mixture with a different CO_2 percentage

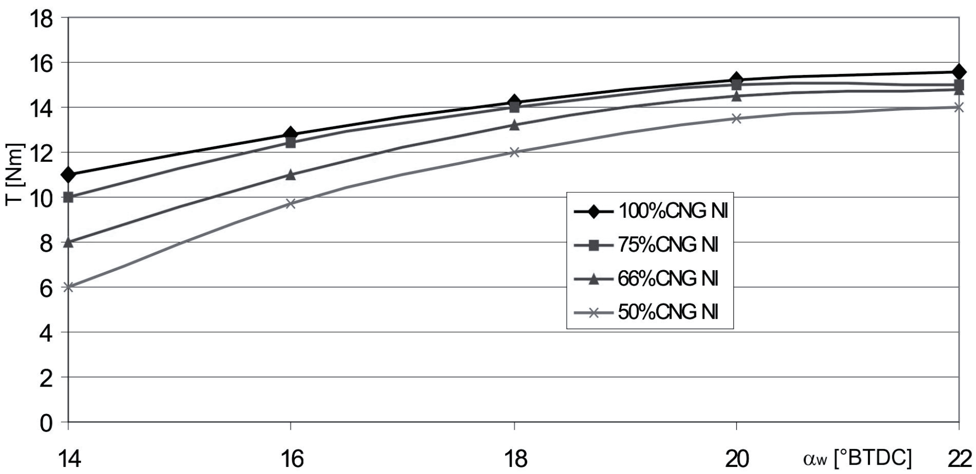

Fig. 5. Engine torque change at a pilot charge of 2 mm^3 and a constant methane charge fed in a mixture with a different CO_2 percentage

5. Summary

The above-presented example results of tests on a dual-fuel compression-ignition engine with the gaseous fuel charge fed in the form of a mixture of methane and carbon dioxide (the main biogas components) confirm the possibility of using low calorific gases as fuel for compression-ignition engines. When these engines are fed with gaseous fuels, it is necessary to properly select the pilot charge injection advance angle, which has a decisive effect on the course of the combustion process and engine performance, but it should be remembered that too early pilot charge injection can cause too rough engine operation, which results in high loads on the piston-crank system. The value of the liquid fuel injection advance angle for fuels with a lower combustible component concentration should be higher because of a lower flame rate in leaner mixtures.

The combustible component concentration in gas has a significant effect on engine efficiency, which is presumably related to the decreasing oxygen concentration in the mixture for fuels containing substantial percentages of non-combustible components.

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References

[1] Azimov, U., Tomita, E., Kawahara, N., Dol, S. S., *Combustion Characteristics of Syngas and Natural Gas in Micro-pilot Ignited Dual-fuel Engine*, World Academy of Science, Engineering and Technology, Vol. 72, pp. 1618-1625, 2012.

[2] Chmielewski, A., Lubikowski, K., Radkowski, S., Szczurowski, K., *Research and simulation work of TEG in cogeneration task of the exhaust system*, Journal of KONES, Vol. 20, No. 2, pp. 41-49, 2013.

[3] Korakianitis, T., Namasivayam, A.M., Crookes, R.J., *Diesel and rapeseed methyl ester (RME) pilot fuels for hydrogen and natural gas dual-fuel combustion in compression–ignition engines*, Fuel, Vol. 90, pp. 2384-2395, 2011.

[4] Kowalczyk-Juśko, A., Mazanek, A., *Agricultural biogas – characteristics, substrates and its use*, Silniki spalinowe, 1, (148), 2013.

[5] Makareviciene, V., Sendzikiene, E., Pukalskas, S., Rimkus, A., Vegneris, R., *Performance and emission characteristics of biogas used in diesel engine operation,* Energy Conversion and Management, Vol. 75, pp. 224-233, 2013.

[6] Ray, N. H. S., Mohanty, M. K., Mohanty, R. C., *Biogas as alternate fuel in diesel engines: A literature review*, Journal of Mechanical and Civil Engineering, Vol. 9(1), pp. 23-28, 2013.

[7] Stelmasiak, Z., *Studium procesu spalania gazu w dwupaliwowym silniku o zapłonie samoczynnym zasilanym gazem ziemnym i olejem napędowym*, Wyd. ATH, Bielsko-Biała 2003.

[8] Śmieja, M., Wierzbicki, S., *The concept of an integrated laboratory control system for a dual-fuel diesel engine*, Journal of KONES, Vol. 19, No. 3, pp. 451-458, 2012.

[9] Wierzbicki, S., *Biogas as a fuel for diesel engines,* Journal of KONES, Vol. 19, No. 3, pp. 477-482, 2012.

[10] Wierzbicki, S., Śmieja, M., Grzeszczyk, R., *Zintegrowane sterowanie stanowiskiem badawczym silników o ZS w środowisku fast prototyping,* Combustion engines, 3(54), 2013.

[11] Wierzbicki, S., *Analysis of the effect of the chemical composition of low calorific gaseous fuels on workload concentration in an engine's combustion chamber*, Journal of Polish CIMAC, Vol. 8, pp. 89-96, 2013.