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SELECTED ASPECTS OF THE USE OF GASEOUS FUELS BLENDS TO IMPROVE EFFICIENCY AND EMISSION OF SI ENGINE

Summary. The paper presents results of SI engine tests, carried on for different gaseous fuels. The analysis carried out made it possible to define the correlation between fuel composition and engine operating parameters. Tests cover various gaseous mixtures: methane with hydrogen and LPG with DME. The first group, considered as low carbon content fuels, can be characterized by low CO_2 emissions. The flammability of hydrogen added in those mixtures realizes the function of combustion process activator and improves the energy conversion.

The second group of fuels is constituted by LPG and DME mixtures. DME mixes perfectly with LPG and differently than in the case of other hydrocarbon fuels also consisting of oxygen, which makes the stoichiometric mixture less oxygen demanding. In the case of this fuel, improvement in engine volumetric and overall engine efficiency has been noticed, compared to LPG. During the tests, standard CNG/LPG feeding systems have been used, which underlines the utility value of the research.

1. INTRODUCTION

Greenhouse gas emissions from transport constitute around 22% of the total CO_2 emissions of globally processed energy. However, this emission plays an important role primarily due to the rapid increase in traffic and almost total dependence of transport on fossil fuels. In addition, transport is the fastest growing sector of the economy among all emission sources. According to the latest ITF data, CO_2 emissions from passenger transport in the world will increase from 30% to 110% by 2050, depending on fuel prices and the development of urban transport [1]. Thus, the role of the transport sector in the implementation of activities related to climate change and sustainable development is fundamental. The adoption of the Paris Agreement allowed for setting a path to mitigate climate change and establishing a five-year cycle of reviewing national commitments on decarbonization from 2020. The most commonly undertaken commitments include the use of low carbon fuels, such as biofuels, LPG, or compressed natural gas (CNG), and the development of electric drives.

An attractive option in counteracting the effects of the emission of internal combustion engines is alternative gas fuels, including biogas, natural gas, dimethyl ether (DME), hydrogen (H2), and mixtures of them. This article focuses on mixtures that are easy to obtain: natural gas and hydrogen (HCNG) and petroleum gases with dimethyl ether (LPG and DME). Their impact on engine performance, combustion characteristics, and energy conversion efficiency has been determined. The use of these mixtures has been presented in many articles published in recent years. Verhelst and Wallner [2] published a review article based on hydrogen engines; this very important research is a milestone in both hydrogen and HCNG engines. Karim et al. [3] published the test results for an HCNG-powered engine, demonstrating that it is possible to increase the efficiency of energy conversion in internal combustion engines. Roopesh K. Mehra et al. presented the current state of

research on the use of HCNG for motor vehicles [4]. The results of research in this field are also presented in several other publications [5-7, 9].

The ease of mixing LPG and DME and properties of these fuels formed the basis for research conducted by Nakazono, T. et al. [10]. Economical and ecological benefits combined with very favorable physicochemical properties are the main causes of focusing on the research on these mixtures [11,12]. The most important aspect worth mentioning is the possibility of obtaining DME as a renewable fuel produced from biomass. As a result, it is possible to significantly reduce CO_2 emissions, compared to LPG, by 30–80% [11] and in the case of NOx by 5–15% [12, 13], in the life cycle assessment.

The results of testing LPG + DME mixtures, with a mass DME content in the range of 5–50%, allowed it to determine the optimal mass fraction of DME in the mixture - 7% for WOT operation, which has been presented [14].

2. RESEARCH OBJECT AND MEASUREMENT SET-UP

The popular passenger car powered by 1.6-liter engine, naturally aspirated with a compression ratio of 9:6, port fuel injection, two valves per cylinder, flat pistons, and without external EGR was used in the experiments. The experiments were performed using a BOSCH FLA 203 chassis dynamometer as presented in Fig. 1. Main features characterizing the engine installed on the tested vehicle have been listed in Table 1. Engine performance was estimated on the basis of acquired dynamic characteristics, defining the power on wheels in the function of vehicle speed. The test stand was equipped with various transducers and sensors allowing the identification of engine operating conditions.
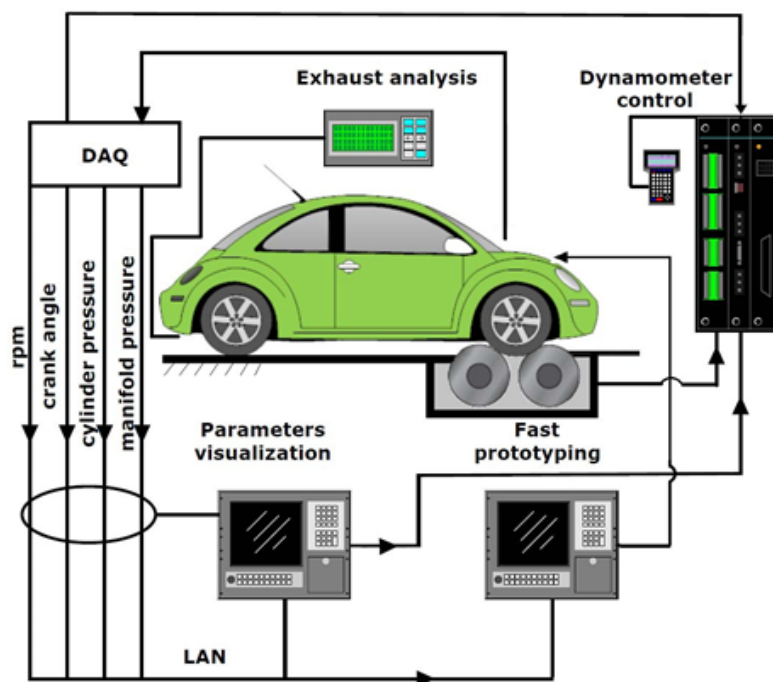

Fig. 1. Diagram of experimental stand [5]

Basic measurements and control systems allowed the continuous acquisition of engine operating conditions, through registrations of:
- in-cylinder pressures, crank angle, with the TDC identification,
- power on wheels,
- manifold pressure,

− inlet air temperature,
− exhaust gases temperature, and
− fuel mass flow to the engine.

The in-cylinder pressure was measured by Kistler 6121 piezoelectric pressure transducers and a charge amplifier, Kistler 5011A. The signals were processed in the type NI PCI-6143 board in a computer for online pressure measurements. The pressure recording system was also connected to the Kistler 2613B crank angle encoder giving the temporal resolution of the pressure recordings of 0.5 CA. The pressure measurements were recorded and stored in a computer, with recordings performed for 200 subsequent cycles in each test, and were further processed with the help of a script debugged in LabView 7.1 environment.

Table 1

Engine specifications

Engine code	X16SZR
Cylinder number and layout	4 R
Maximum power	55 kW@ 5200 rpm
Maximum torque	128 N·m @ 2800 rpm
Displacement	1598 ccm
Bore × stroke	79.0 × 81.5 mm
Compression ratio	9.6

3. RESULTS AND DISCUSSION

The first group of fuels selected for tests consisted of eight methane/hydrogen blends, with various hydrogen shares (by volume): 0% (pure methane), 5%, 10%, 15%, 20%, 30%, 40%, and 50%. On the basis of hydrogen properties (Table 2), it can be noticed that hydrogen burns at the air–fuel ratio of 9, whereas methane and gasoline are capable of burning at the air–fuel ratios no lower than 1.9 and 1.4, respectively. Hydrogen density of 0.0838 kg/m^3 (at normal temperature and pressure) volumetric low heating value equals 10.046 kJ/m^3, and it is lower than methane 32.573 kJ/m^3 and petrol 195.8 kJ/m^3 [10]. As a consequence, hydrogen occupies a greater proportion of volume with respect to air. An approximate seven-fold increase in the burning speed of hydrogen flame (265–325 cm/s) over methane or petrol results in shorter burn time. This shorter burn time results from lower heat transfer from hydrogen flame, compared to either methane or petrol flame.

Table 2

Main properties of hydrogen, methane, and petrol [5]

	Hydrogen	Methane	Petrol
Air–fuel ratio lower limit	~9	~1.9	~1.4
Range of burning [% gas volume in air]	4–75	5.3–15.0	1.2–6.0
Minimal ignition energy [mJ]	~0.02	~0.28	~0.25
Burning speed [m/s]	~2.90	~0.38	~0.37–0.43
Adiabatic flame temperature [K]	~2318	~2190	~2470
Self-ignition temperature [K]	~858	~813	~500–750
Density [kg/m^3] at 293,15 K and 101,3 kPa	0.082	0.717	4.4
Stoichiometric air to fuel ratio [kg/kg]	~34	~17.2	~14.8
Volumetric lower heating value [MJ/kg]	~3.37	~2.56	~2.79
Mass lower heating value [MJ/kg]	~120	~50	~44.5

The enrichment of methane with hydrogen changes the value of p_{max} (Fig. 2). The higher content of hydrogen accelerates the burning speed of the mixture, increasing the rate of growth pressure. It is also worth noting that a change of fuel composition does not cause considerable displacement p_{max} in the engine operating cycle. For the presented series of results, this point occurs between $\varphi = 21 \div 24$ °CA after TDC. The changes in the released energy have been presented in Figure 3; the characteristics permit it to estimate the influence of hydrogen share in the first stage of flame development. This phase has been reduced with the increasing amount of hydrogen, confirming that hydrogen acts as combustion activator. Power measurements on wheels (Fig. 4) confirmed the trends of observed changes. The next graph (Fig. 5) shows the effect of hydrogen addition on the overall efficiency of the tested engine.
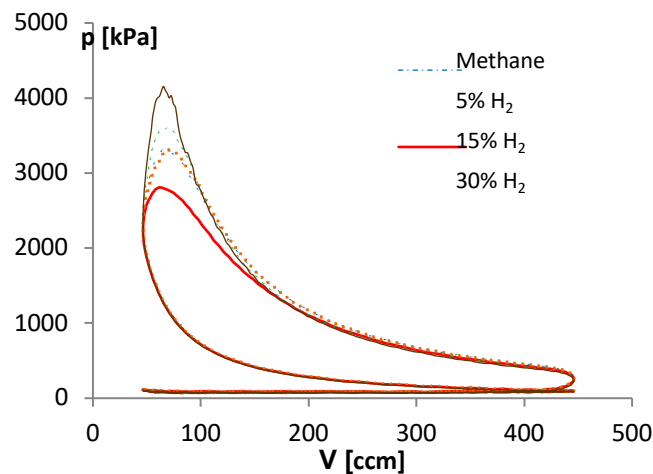
Fig. 2. Pressure changes in the p–V system, an engine Fueled with methane/hydrogen blends, $n = 2500$ rpm, WOT=100%, and $\lambda = 1.0$
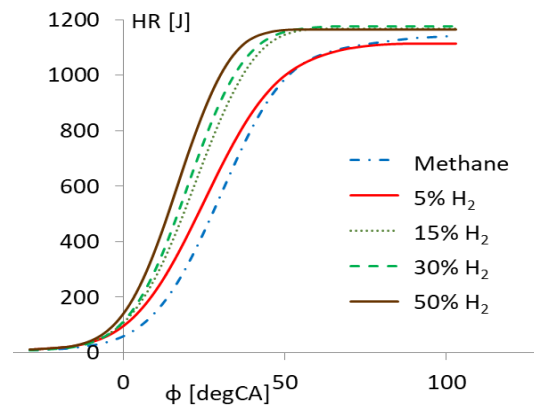
Fig. 3. Heat release function, an engine fueled with methane/hydrogen blends; $n = 2500$ rpm; WOT=100%; $\lambda = 1.0$
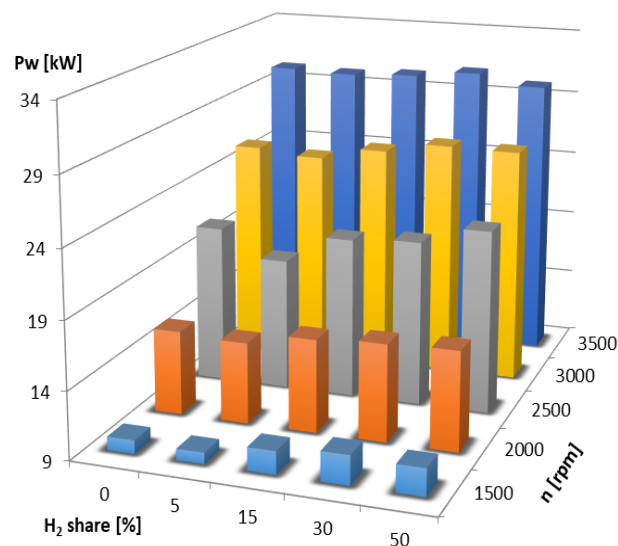
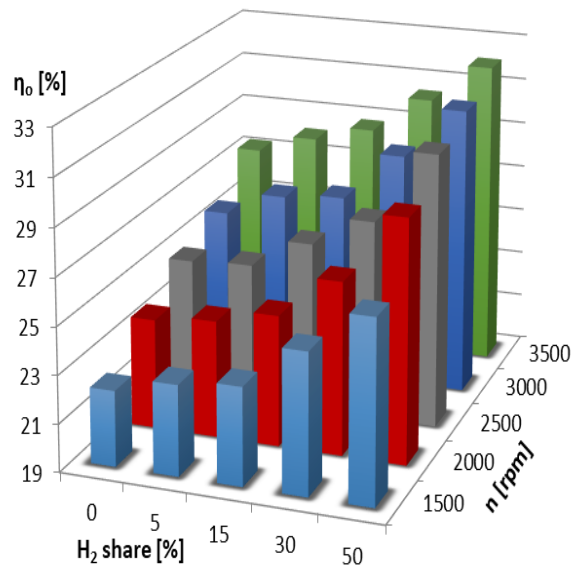Fig. 4. Measured values power on wheel, WOT=100% and $\lambda = 1.0$

Fig. 5. Total efficiency of engine fueled with methane/hydrogen blends, WOT=100% and $\lambda = 1.0$

The composition of the exhaust gas is tightly related to the composition of the fuel. The most conspicuous is the reduction in the emission of CO_2, which is a result of a reduced share of carbon in the charge prepared for combustion (Fig. 6). With the combustion process properly realized (temperature, ignition conditions, mixture composition, etc.), we may also expect a lower value of CO. An increase in the share of H_2O in the exhaust is an effect of an increased share of H_2 in the fuel.
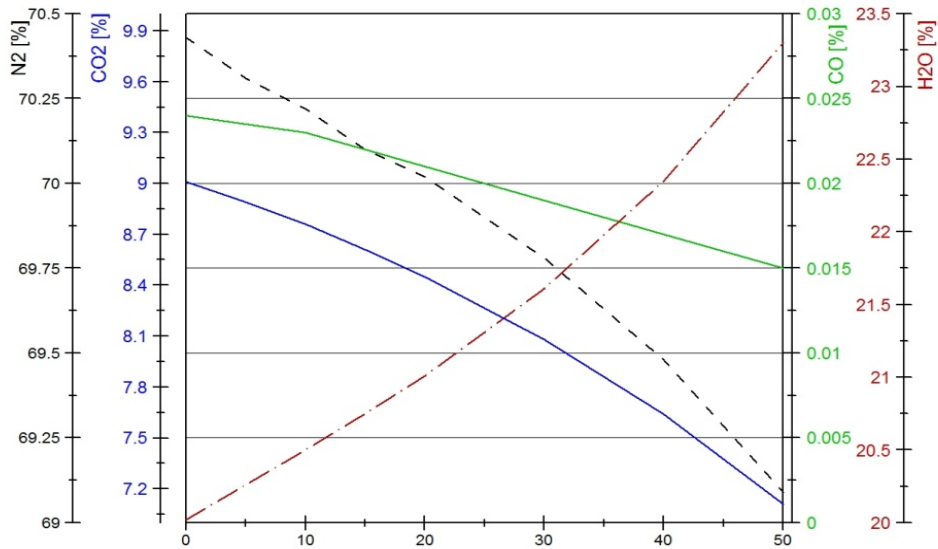

Fig. 6. Changes in the concentrations of main exhaust components depending on the share of the addition of H_2 to CH_4 (combustion of stoichiometric mixtures)

The density of DME as well as of crude oil-based gases is similar both in the gaseous and liquid phases although its low pressure allows storing it in the liquid phase in ambient temperatures. A theoretical air demand for DME is the lowest among all the gaseous fuels, which results from the oxygen content in the fuel. As a result, the calorific value of the air–fuel mixture, in the case of DME–LPG blend remains constant even with the increasing mass share of DME (Fig. 7).
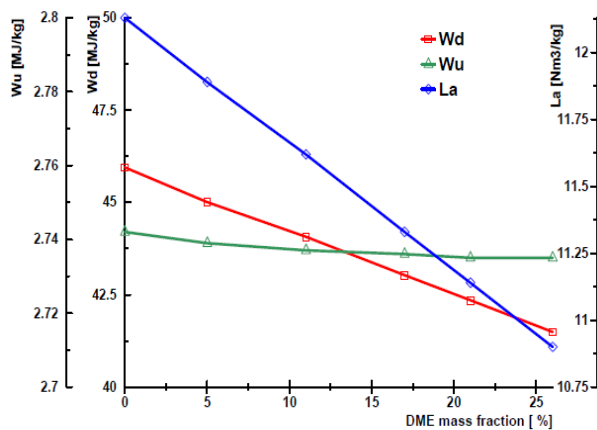

Fig. 7. Mass heating value as a function of DME mass ratio: W_u – the mass heating value of a stoichiometric mixture, W_d – the mass heating value of fuel, and L_a – the stoichiometric air to fuel ratio
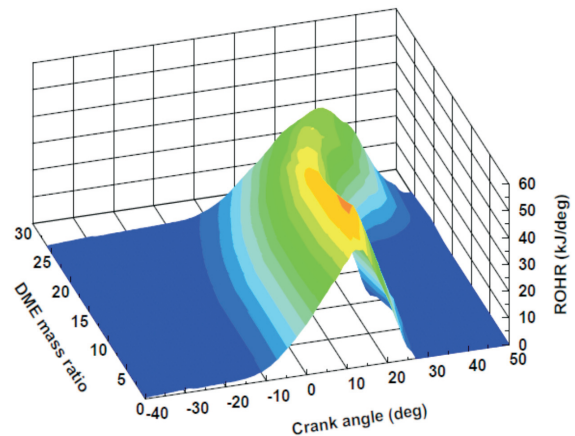

Fig. 8. ROHR as a function of DME ratio, at 2500 rpm, WOT and $\lambda = 1.0$

In the previous studies, carried out at the Faculty of Transport of Silesian University of Technology [14], the changes that occur in the combustion process, depending on the amount of DME's participation, were determined. These tests were conducted at idle and at full engine load for selected speeds. In those preliminary studies, the range of changes in the participation of DME in the blend with LPG, which ensures the proper combustion process, has been determined. The results showed that a limited DME additive, in the range of approximately 7–15% by weight, increased engine performance. The interpretation of these changes is the result of the analysis of the function of the heat release rate during the combustion process (Fig. 8). A too large amount of DME in the fuel causes a reduction in combustion rate and prolongs the process. The engine management during the test was performed according to the settings developed by the manufacturer.
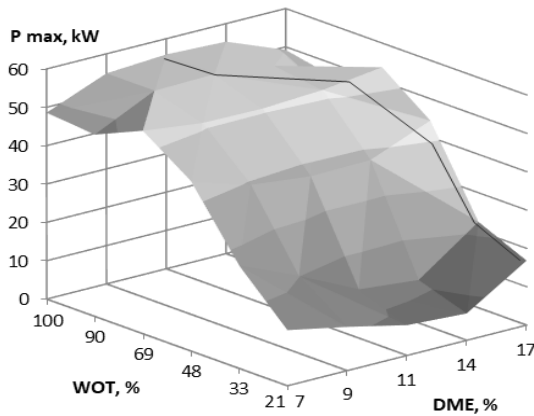

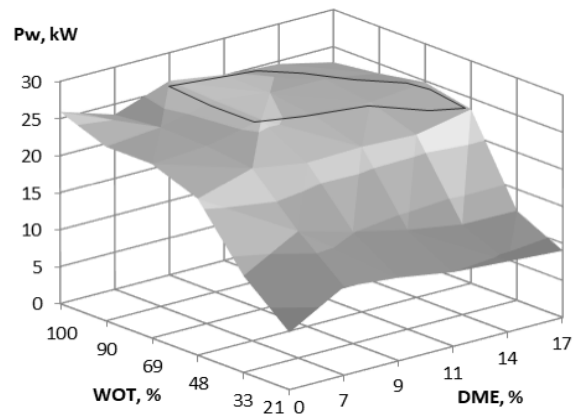Fig. 9. Values of engine maximum power depending on DME share and engine load, λ=1.0

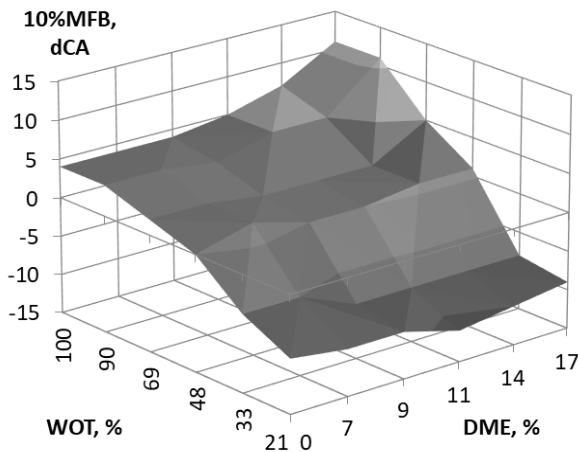Fig. 10. The results of power on wheel measured during the experimental test, n = 2500 rpm

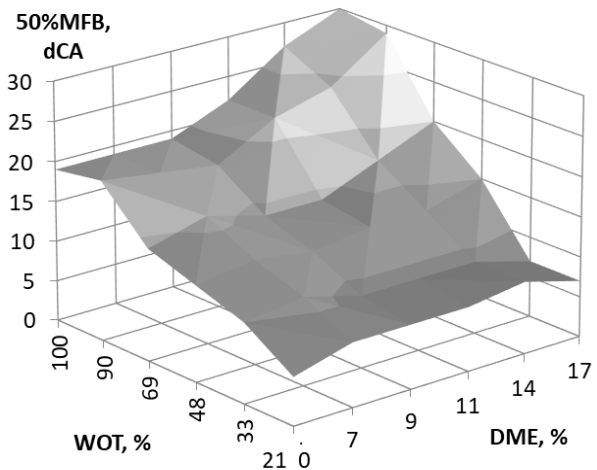Fig. 11. A point of 10% MFB during the combustion process, n = 2500 rpm

Fig. 12. A point of 50% MFB during the combustion process, n=2500rpm

The chart showing the maximum values of the measured engine power (Fig. 9) indicates that with the engine load drop, the fuel composition changes to give the maximum power (red line in the graph). The change in fuel composition consists of increasing DME's share. The power measured on the wheels of the vehicle (Fig. 10) also shows that, with less engine load, an increase of the DME share has a beneficial effect on the performance of the vehicle. This area is marked on the chart.
The following graphs (Figs. 11 and 12) show the angle of the crankshaft position corresponding to the characteristic points of the combustion process, 10 and 50% of the mass fraction burned MFB. In both cases, there is a strong relationship between the share of DME in the fuel and the rate of heat release.

In the conditions that prevail in the combustion chamber at a load of 90–100%, only a small addition DME allows us to keep appropriate combustion velocity.

Increasing DME share over 11% slows down the process under these conditions. However, decreasing engine load moves this limit, making it possible to gradually increase the addition of DME. The load change is also associated with a change in the ignition advance angle that was determined by the original engine driver.

Because of the total efficiency of the engine (Fig. 13), the range of partial loads is most advantageous because the highest values (LPG-like) are obtained there. This is the most common operating range of the engine.

Another aspect of the analysis is the composition and emission of combustion products. The changes in the concentration of selected exhaust components are shown in the following graphs. Emissions of carbon-containing compounds, CO_2 (Fig. 14) and CO (Fig. 15), are not subject to major changes. The influence of the type of fuel is not noticeable because in the group of the studied fuels the total share of the C-component in the stoichiometric mixture is similar, from 2.83 to 2.86%. CO emissions are maintained at a constant level of $0.045 \div 0.035\%$, indicating the correct course of the combustion process. The NOx concentration decreases with the decrease in engine load.
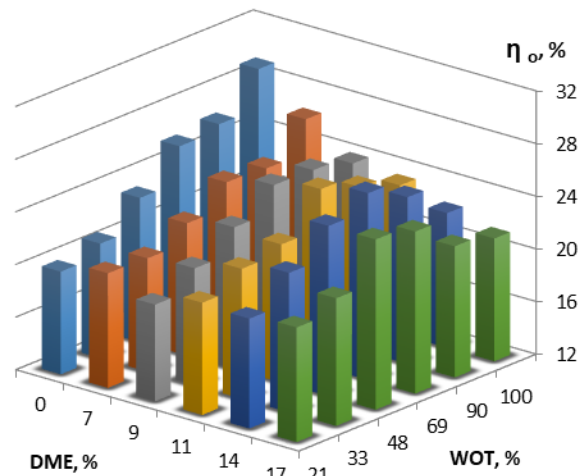
	100	90	69	48	33	21
0	29,07	26,1	25,65	22,95	20,7	19,8
7	26,28	23,76	23,94	22,05	20,61	20,7
9	23,94	24,66	24,75	22,77	20,88	19,35
11	23,31	24,39	25,47	22,5	21,87	20,52
14	22,23	24,66	26,19	24,93	22,59	20,34
17	21,33	21,96	24,3	24,93	21,69	20,7

Fig. 13. Total efficiency of the tested engine depending on DME share and engine load, n = 2500 rpm
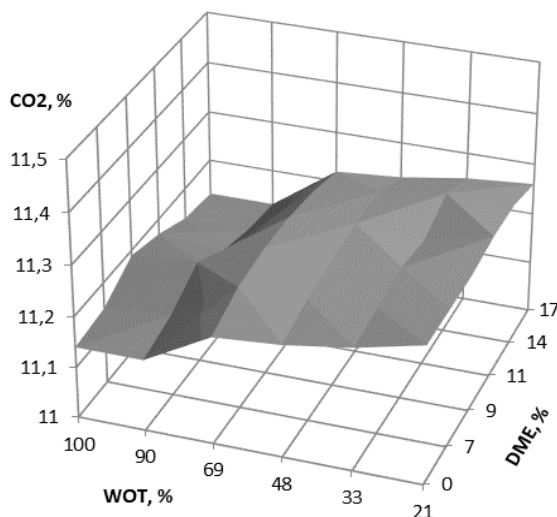

Fig. 14. An influence of DME share and engine load on emission CO_2, $\lambda = 1.0$ and n = 2500 rpm
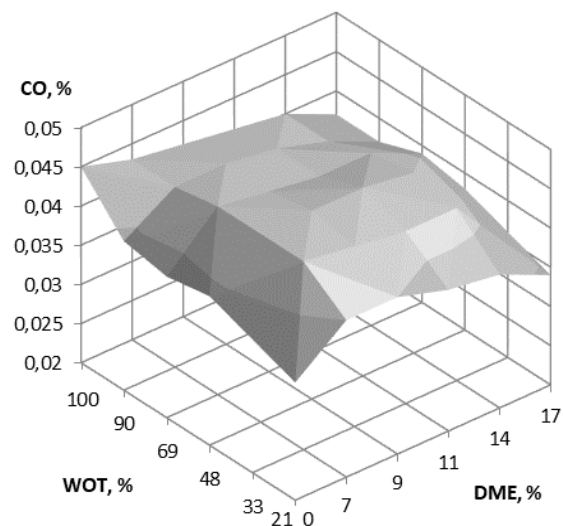

Fig. 15. An influence of DME share and engine load on emission CO, $\lambda = 1.0$ and n = 2500 rpm

4. CONCLUSIONS

1. The carried out tests give a possibility of understanding the fundamental combustion properties of methane/hydrogen and LPG/DME blends, which is important for developing advanced combustion engines with necessary operating strategies. Increasing the content of hydrogen in a gas fuel (e.g. methane and natural gas) significantly affects the improvement in the efficiency of energy conversion in the SI engine, without interfering in the original feeding and ignition systems algorithms.

2. Basic changes in the course of the heat release function are ensued the shorten period of burning initiation thanks to good flammability of hydrogen. As a result of these changes, higher engine power and total efficiency η_o can be achieved. An effective increase in power on wheels has been measured for fuel blends containing: 30% of hydrogen, which ranged from 12% (1500 rpm) to 3% (3500 rpm), whereas η_o increased by c.a. 12% in this range of speed.

3. The general aim of adding hydrogen to methane and DME to LPG is to minimize negative parts in their combustion. Because of the differences in selected properties of these fuels, an improvement in the conditions for the flame initiation and shortening of the charge heating period are expected.

4. At full engine load, DME share should be limited to approx. 10%. In the range of partial loads, 50–70% its share can be increased to 17%.

5. Future experimental development would foresee the optimization of the emissions of both pollutant and CO_2 along the reduction in fuel consumption for the vehicle driving cycle. It is necessary to investigate widely the following set of engine variables:
   - Spark advance,
   - Compression ratio,
   - A wide spectrum of λ values, and
   - EGR.

Nomenclature

CA	crank angle	LNG	liquefied Natural Gas
CI	compression ignition	LPG	liquefied Petroleum Gas
CN	Cetane number	MFB	mass fraction burned
CNG	compressed Natural Gas	SI	spark ignition
DME	dimethyl ether	TDC	top dead centre
EGR	exhaust gas recirculation	WOT	wide open throttle

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