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THE INFLUENCE OF SELECTED GASEOUS FUELS ON THE COMBUSTION PROCESS IN THE SI ENGINE

Summary. This paper presents the results of SI engine tests, carried out for different gaseous fuels. The analysis carried out made it possible to define the correlation between fuel composition and engine operating parameters. The tests covered various gaseous mixtures: methane with hydrogen from 5% to 50% by volume and LPG with DME from 5% to 26% by mass. The first group, considered as low-carbon-content fuels can be characterized by low CO_2 emissions. Flammability of hydrogen added in those mixtures realizes the function of the combustion process activator. Thus, hydrogen addition improves energy conversion by about 3%. The second group of fuels is constituted by LPG and DME mixtures. DME mixes perfectly with LPG, and differently than other hydrocarbon fuels, consisting of oxygen as well, which makes the stoichiometric mixture less oxygen demanding. In the case of this fuel an improvement in engine volumetric and overall engine efficiency has been noticed compared with LPG. For the 11% DME share in the mixture an improvement of 2% in the efficiency has been noticed. During the tests, standard CNG–LPG feeding systems have been used, which underlines the utility value of the research. The stand-test results have been followed by combustion process simulation including exhaust forming and charge exchange.

1. INTRODUCTION

The use of alternative fuels is one of the main solutions allowing the reduction of pollutant emissions nowadays. These fuels also allow the increase of energy conversion efficiency during the entire life cycle as well as during the conversion of chemical energy to mechanical or electrical energy. Among gaseous alternative fuels the most significant substitutes to conventional fuels are natural gas, LPG, and DME in their quite popular applications in both vehicles and stationary applications. Research carried out on these fuels proved that it is possible to replace conventional fuels with them, allowing them to differentiate energy sources as well. Gaseous fuels have been normalized by means of their chemical composition and physicochemical properties, whereas modern propulsion and engine control systems have become precise enough to achieve satisfactory ecological and economic results [1-3, 5-7]. The number of bi-fuel gas-powered vehicles is, therefore, increasing. Considering the use of already highly accurate and dedicated sequential injection systems, the effective use of gaseous fuel energy still becomes a challenge. The problems with effective combustion process control are mainly related to high ignition temperature and low flame-front speed, especially, in the case of natural gas [8-11].

One of the ways of solving this problem is increasing the flame-front propagation speed, mainly by adding more reactive fuels like hydrogen or DME. The influence of the addition of those fuels to natural gas and LPG on the combustion process has been presented in many publications.

Navarro et al. [10] show that the maximum cylinder pressure rises as the fraction of hydrogen in the blend increases and leads to a decrease in CO_2 emissions with no loss of performance. Zareei [12] carried out a numerical investigation on SI using a natural gas–hydrogen blend. By employing the same spark advance for CNG and HCNG fuels, the main result was the negligible variation in the engine brake efficiency despite a faster combustion rate. Instead, at MBT spark advance for two mixtures of 10% and 30% has shown relevant increments in engine efficiency. Hoekstra et al. [13] observed experimental results of natural gas-fueled internal combustion engines claiming that hydrogen as additive in NG can strongly improve the performance of such engines – especially, in terms of power, efficiency and emissions – allowing the engine to work with leaner mixtures. In addition, hydrogen does not affect the anti-knocking performance of NG fuel, and there is a strong reduction in NOx for a hydrogen percentage of up to 30%; an important point was the increment of the flame-propagation speed and a consequent reduction of the spark-advance angle to obtain the Maximum Brake Torque, as already indicated by Nagalingam et al. [14]. Fuel mixtures of NG and hydrogen have been widely studied in IC engines, and results show that engine performance can improve and the exhaust emissions can be reduced by adding a small amount of hydrogen. Some of studies carried out are presented below.

Dimopoulos et al. [15] carried out a well-to-wheel assessment for an NG–hydrogen mixture engine. They related engine test results with different hydrogen-producing methods and concluded that the application of hydrogen to IC engines is helpful in reducing greenhouse gas emissions. Apostolescu and Chiriac [16] studied the effect of hydrogen addition on the combustion process at mid to low loads; results show that the cyclic variation and 10–90% burn duration was greatly reduced when hydrogen mass fraction varied from 1.5% to 3%.

In the case of LPG–DME mixtures Nakazono et al. [17] showed that DME can be blended with LPG because characteristics of DME are similar to those of LPG, and may be used as gas fuel for a small gas engine. Moreover, the result of this test showed that the knocking limit using DME-mixture fuel could be improved by retarding ignition timing and A/F ratio under the limit of blend ratio without EGR being less than 30%. The limit of DME-blend ratio should be less than 40% by retarding ignition timing and controlling EGR ratio in the improvement of knocking limit. Chen et al. [18] investigated the effect of DME addition on engine performance. Experimental results showed that indicated thermal efficiency was increased by 25% and coefficient of cyclic variation in engine speed was decreased by 29.2% at a DME energy fraction of 85.2% in the total fuel. Seokhwan Lee [19] experimentally studied a spark ignition engine operated with DME-blended LPG fuel. They examined the effect of n-Butane and propane on performance, emission characteristics (including hydrocarbon, CO, and NOx), and the combustion stability of an SI engine fueled with DME-blended LPG fuel. The results showed that stable engine operation was possible for a wide range of engine loads with propane-containing LPG–DME-blended fuel rather than n-Butane containing LPG–DME-blended fuel as the octane number of propane is higher than that of n-butane. Also, engine power output, break-specific fuel consumption (BSFC), and combustion stability when operating with propane-containing DME-blended fuel were comparable to those values for pure LPG fuel operation.

These observations induced the authors in a previous work [20, 21] to experience the simultaneous combustion of homogeneous mixtures of NG with hydrogen and LPG with DME in stoichiometric proportion with air (with different mixture proportions) on a series-production spark-ignition engine, so as to exploit the good qualities of both fuels to obtain cleaner and more efficient combustions. This study aims to discover the means to extend the use of NG–hydrogen and LPG–DME mixtures. The method introduced in this paper is as follows:

- mixing NG with hydrogen,
- dissolving DME into n-butane and propane (which was used as a representative LPG fuel), in order to directly inject the chosen proportion of blended fuels into the intake manifold. The effects of activators on the SI engine performance and combustion were examined.

2. FUEL PROPERTIES AND MEASUREMENT SET-UP

The selected chemical and physical properties of gaseous fuels have been presented in Fig. 1. A total of fourteen different fuels were used; these were gasoline, methane, LPG, and fuel blends consisting of the following:

- methane with hydrogen addition from 5% to 50%, by volume,
- LPG with DME addition from 0% to 28% DME by mass, respectively.

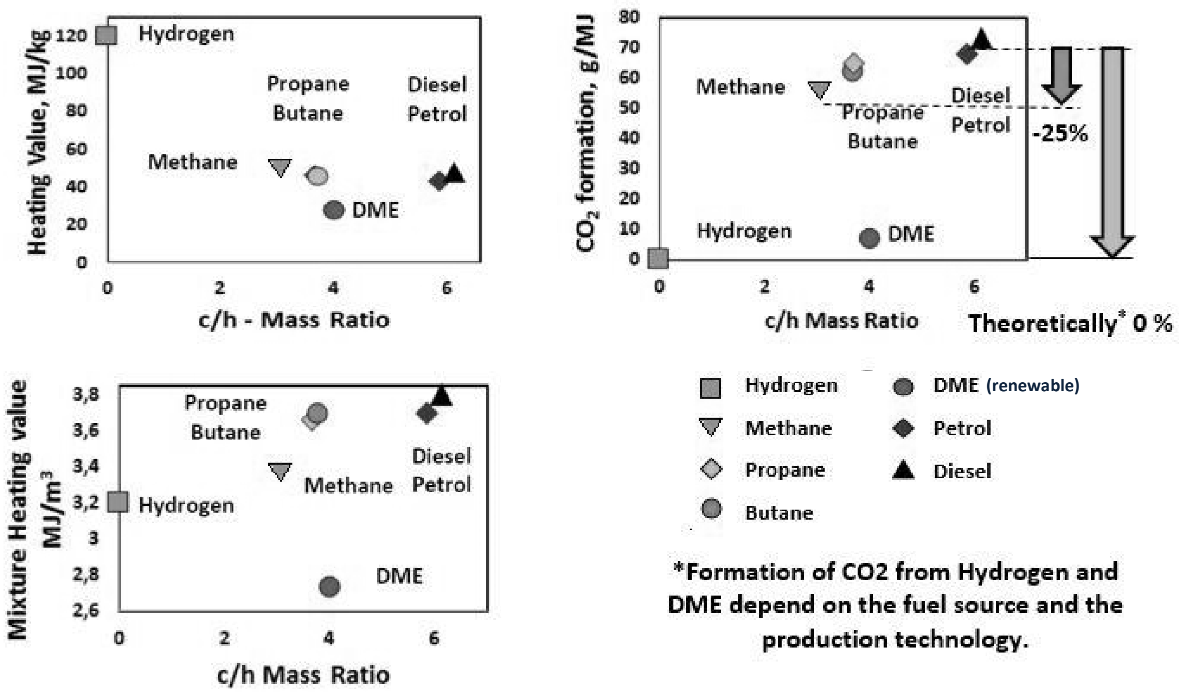


Fig. 1. Main properties of selected gaseous fuels as a function of c/h ratio (at p=100 kPa, T=283 K, λ=1)

Table 1

The composition of mixtures prepared for tests

Fuel	Symbol used in figures	Molecular mass of fuel [kg/kmol]	Stoichiometric air–fuel ratio A/F [kg/kg]
LPG 50/50	LPG	51	15.76
Methane	NG	16	17.48
95%CH_4 and 5%H_2 by vol.	95NG5H	15.3	17.596
90%CH_4 and 10%H_2 by vol.	90NG10H	14.6	17.72
85%CH_4 and 15%H_2 by vol.	85NG15H	13,9	17,86
80%CH_4 and 20%H_2 by vol.	80NG20H	13.2	18.01
70%CH_4 and 30%H_2 by vol.	70NG30H	11.8	18.37
60%CH_4 and 40%H_2 by vol.	60NG40H	10.4	18.826
50%CH_4 and 50%H_2 by vol.	50NG50H	9	19.424
95%LPG and 5%DME by mass	5DME	50.75	15.466
89%LPG and 11%DME by mass	11DME	50.45	15.101
83%LPG and 17%DME by mass	17DME	50.15	14.73
79%LPG and 21%DME by mass	21DME	49.95	14.482
74%LPG and 26%DME by mass	26DME	49.7	14.168

A popular passenger car powered by a 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 on a BOSCH FLA 203 chassis dynamometer as presented in Fig. 2. Main features characterizing the engine installed on the tested vehicle have been listed in Table 2. Engine performance has been estimated on the basis of acquired dynamic characteristics, defining the power of the wheels as a function of vehicle speed. The test stand was equipped with various transducers and sensors allowing the identification of engine operating conditions. Basic measurements and control systems allowed continuous acquisition of engine operating conditions, through registrations of the following:

- in-cylinder pressures, crank angle, with the TDC identification,
- power on wheels, manifold pressure, inlet air temperature,
- exhaust gases’ temperature,
- fuel mass flow to the engine.

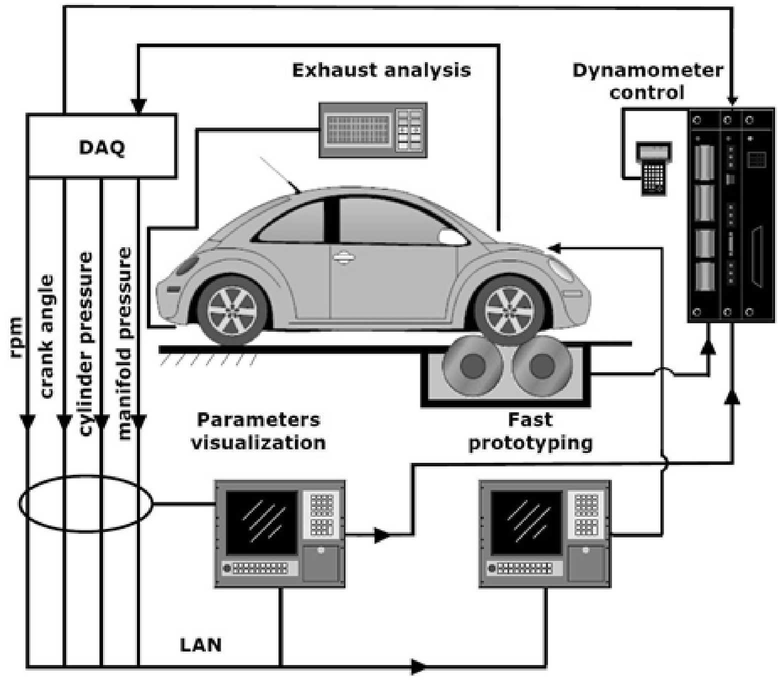


Fig. 2. Experimental setup

The in-cylinder pressure was measured using Kistler 6121 piezoelectric pressure transducers and a charge amplifier, Kistler 5011A. The signals were processed in a 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 CAD. The pressure measurements were recorded and stored on a computer, with recordings performed for 300 subsequent cycles in each test, and were further processed with the help of a script debugged in LabView 7.1 environment.

Table 2

Engine characteristics

Cylinder number and layout	4R
Maximum power	55 kW@5200 rpm
Maximum torque	128 Nm @ 2800 rpm
Displacement	1598 ccm
Bore stroke	79,0 mm x 81.5 mm
Compression ratio	9.6

3. RESEARCH METHODOLOGY

The research has been developed according to the predefined program which covered the following:
- estimation of power on wheels of the tested vehicle in the function of vehicle speed for the all the tested blends,

- estimation of in-cylinder pressure in the function of the crank angle,
- identification of specific fuel consumption.

The research covering engine indication and specific fuel consumption measurements were carried on the idle, and for WOT at speeds of 1500, 2000, 2500, and 3500 rpm, for each of the prepared blends. During the tests, no modifications in the engine control were done. Ignition timing was set up for the petrol operation, whereas stoichiometric air–fuel ratio was continuously controlled in a closed-loop mode by means of the ECU responsible for gas-fuel metering. The EGR valve remained closed.

Results recorded during stand tests were used as an input for the mathematical model in GT-Power. The so-called "reverse run combustion simulation" uses the recorded in-cylinder pressure traces as well as other required data including fuel mass and composition as well as engine volumetric efficiency to calculate main combustion parameters. The model calculations are based on the equations of energy balance in a closed combustion chamber. The model calculations are supplementary to conducted measurements.

4. RESULTS AND DISCUSSION

Because of the differences of selected properties of methane and hydrogen, an improvement in the conditions for the initiation of a flame and shortening the period of charge heating are expected. The minimum ignition energy of hydrogen is 0.02 mJ, and that of methane is 0.28 mJ. It can also be noticed that, speed of flame spread in case of hydrogen is 2.9 m/s, whereas in case of methane it equals 0.38 m/s. DME and LPG have similar properties if the flame speed for LPG equals 4.1–4.5 m/s and that for DME assumes a value in the range of 4.2–6.15 m/s [22].

One of the important performance factors is power measured on wheels P_w. The measurements were performed on a chassis dynamometer at the WOT, for different engine speeds. For the purpose of engine output power estimation, it is necessary to take into consideration transmission efficiency. For the investigated vehicle, a correction ratio of k=0.946 (according to EEC standard) was assumed.

4.1. Effects of Fuel Composition on Power

Tests on a chassis dynamometer determined the influence of different fuels on overall engine performance. Power and torque results have been presented in the chart.

Generally, torque and power curves have a similar shape, the only difference were regarding varied position of peak values for different fuels. Fig. 3 presents the power in the wheels for all tested blends. The power increases with rising hydrogen share. The exceptions are 5% and 10% hydrogen mixtures. Lower power output registered in these cases resulting from a 7% share of hydrogen in the mixture, which cause the flame front speed to be at its lowest value [23].

In the case of DME-enriched LPG, with a share not exceeding 17%, a small power increase was noticed. Increasing the DME share above 17% results in a drop in power compared with the propane–butane mixture.

4.2. In-Cylinder Pressure Traces and Heat Release

Combustion analysis has been undertaken using in-cylinder pressure traces. Fig. 4 shows the p–V diagrams and heat release, whereas the IMEP for the methane–hydrogen mixture at 2500 rpm and λ=1 has been presented in Fig. 5. To identify the cycle that most significantly represents the average burn characteristics of the evaluated point, the representative cycles have their IMEP, BMEP, peak pressure, and location of peak pressure closest to that of the mean value of each parameter, which, in turn, have been calculated on the bases of 300 subsequently recorded cycles.

The changes in the released energy have been presented in Fig. 4; the characteristics permit estimation of 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 a combustion activator.

Increasing the hydrogen share results in an IMEP drop for mixtures featuring 15% of hydrogen; higher IMEP values compared with methane can be noticed for shares of hydrogen over 15%. The reduction of IMEP values for a mixture of methane and hydrogen in the range of 5–15% results from the velocity of the stoichiometric mixture. The dependence of the combustion speed of the stoichiometric mixture on hydrogen content in the mixture is determined by the parabola for which the minimum value appears at 7% hydrogen.

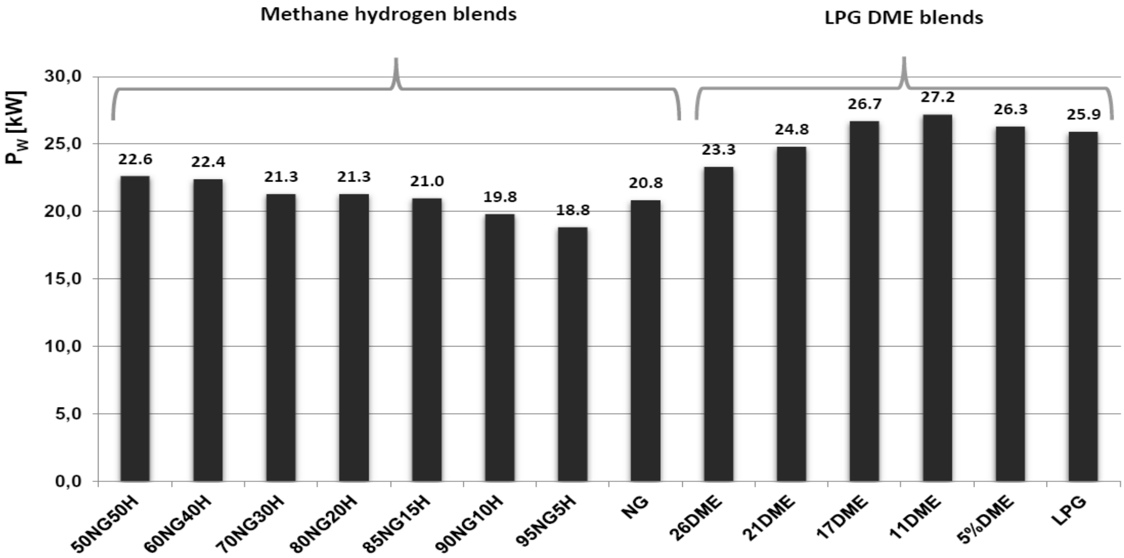


Fig. 3. The variation of power in the wheels, for rpm=2500, 100% load, λ=1.0

Increasing the hydrogen share in the mixture accelerates the combustion process; however, in the cases of hydrogen shares under 15% the amount of heat released in the combustion process is lower or equal to the one obtained for pure methane.

The DME–LPG blends insignificantly increased the peak and the mean pressure values despite the fact that the engine speed and ignition timing remained unchanged. The stability of the combustion process, described by COV_IMEP does not exceed 2.5% for the mixtures with the DME shares up to 17%, whereas for higher amounts of DME it reaches 4 %.

In the case of the LPG–DME-blend stoichiometric-mixture combustion, an early growing in-cylinder pressure and a higher value of the peak pressure have been observed but only in the case of mixtures that did not exceed 17% of the DME share: Fig. 6. With the increase of the DME fraction above 17% the cylinder pressure curve rise was retarded and the peak-pressure crankshaft angle was delayed leading to a decrease in the peak cylinder pressure along with the increasing DME fraction in the blends.

Although the flame-propagation speed rises with the increase in DME fraction in the blends, the retardation in the optimum ignition timing with the increase in DME fraction still delays the rise of the cylinder pressure. In addition, the constant heating value of the blend with an increase in DME fraction without the correction of the ignition angle lowers the peak value of the cylinder pressure. Lower ignition energy and temperature of DME is also the main reason for reaching maximum pressure values that are close to TDC. The DME mass share in the mixture, however, has a significant influence on the crankshaft angle at which the pressure reaches its maximum value. For the mixtures featuring 5 and 11% DME content the maximum pressure was obtained faster than in the case of the LPG fuel feed. The heat release of the blends with different DME fractions is shown in Fig. 6 and IMEP for all LPG–DME mixtures has been presented in Fig. 7. Similarly, as in case of the cylinder pressure for the stoichiometric mixture combustion, the fastest heat-release rate and the highest value of the maximum heat release have been noticed for the blend with the 11% DME share. Further increasing the DME fraction over 11% will decrease the maximum heat-release rate.

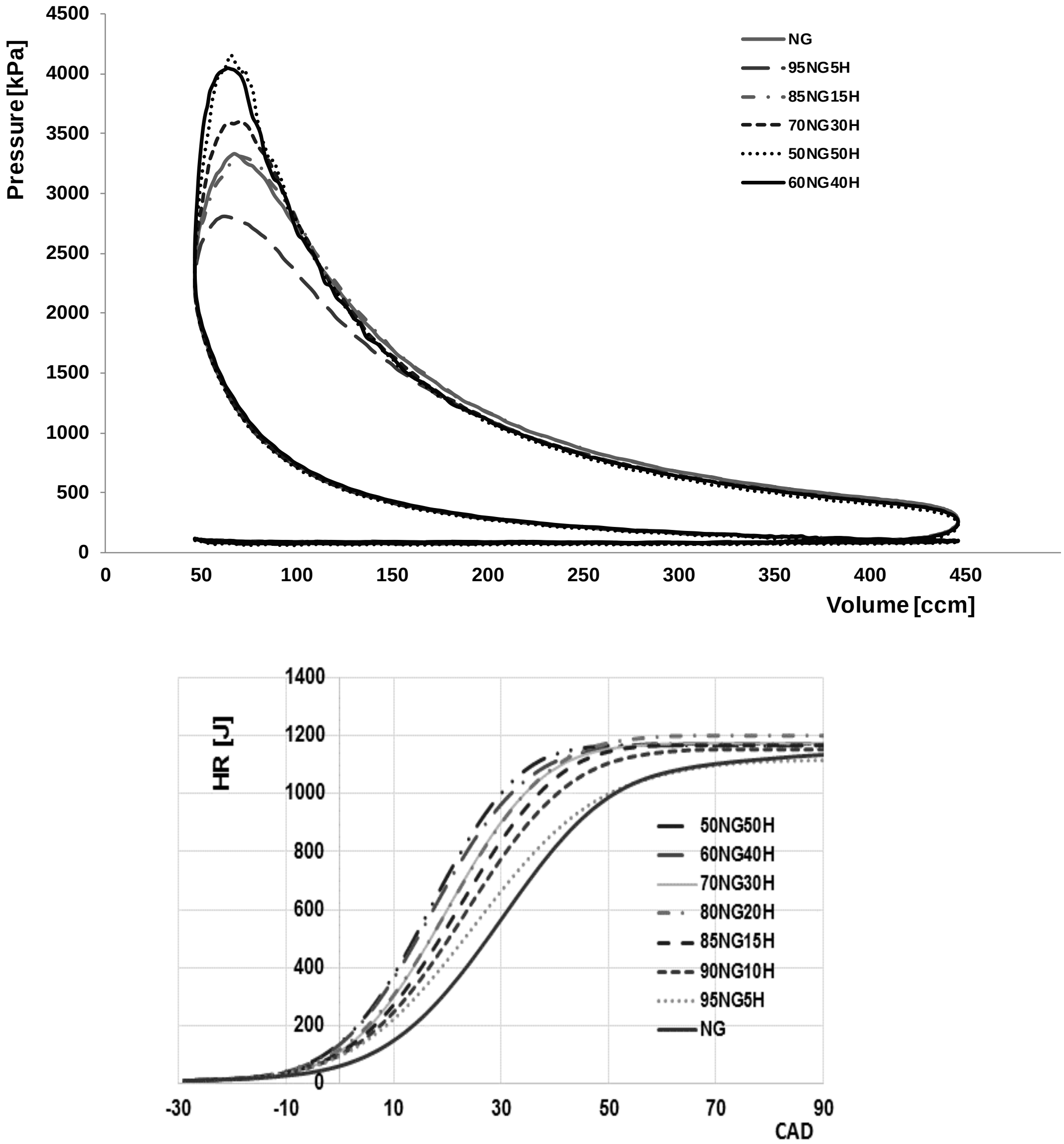


Fig. 4. p–V diagram and heat-release function for the engine fueled by methane, enriched with hydrogen, at 2500 rpm; WOT and $\lambda = 1.0$

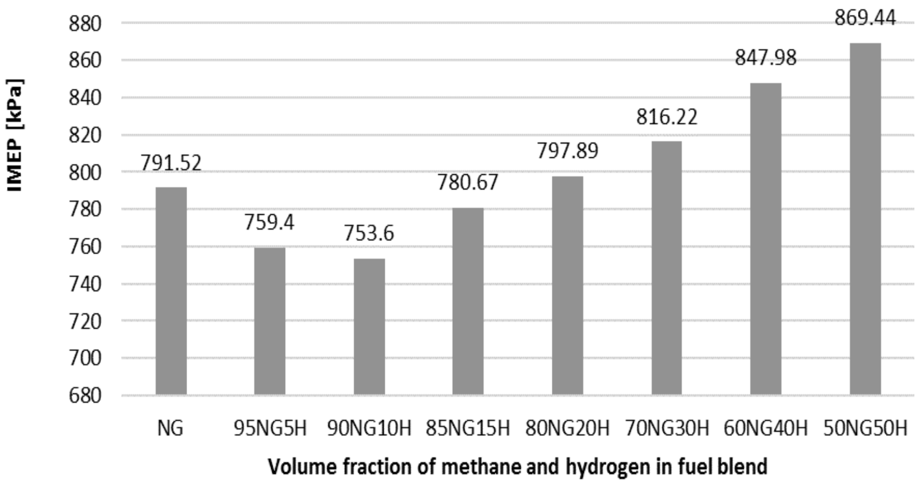


Fig. 5. IMEP for the engine fueled by methane enriched with hydrogen, at 2500 rpm; WOT and $\lambda = 1.0$

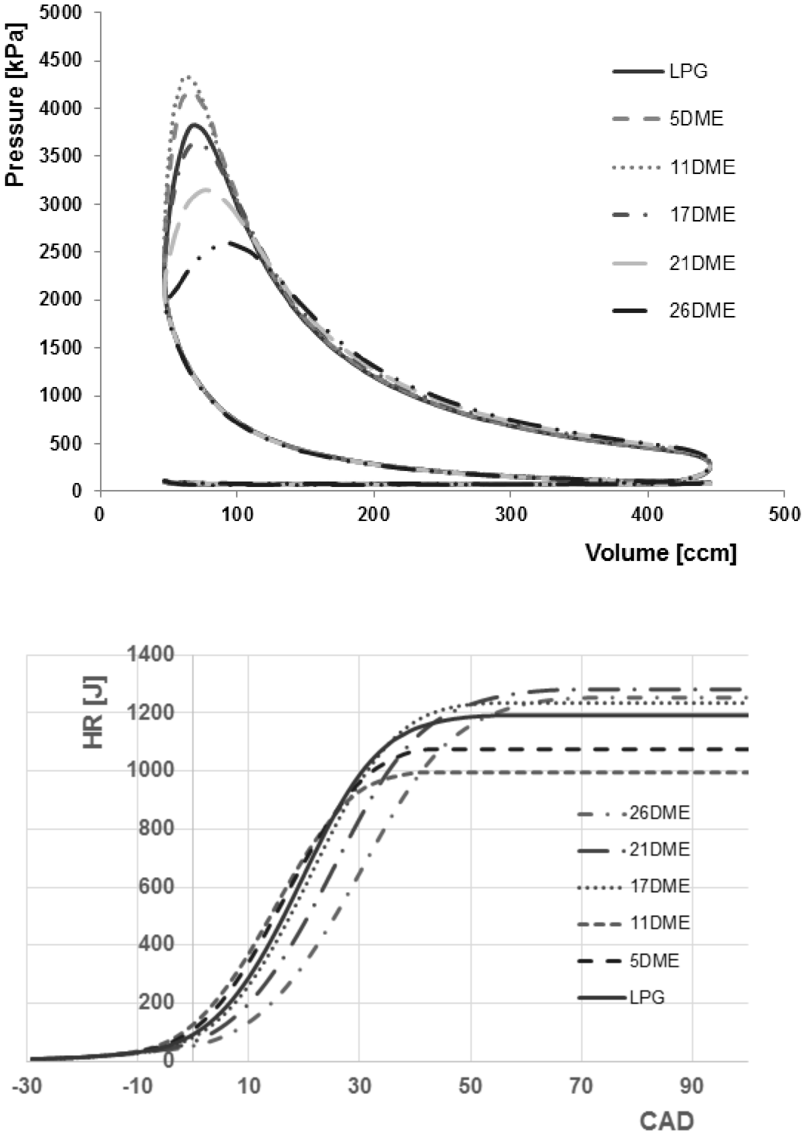


Fig. 6. p–V diagram and heat-release function for the engine fueled by LPG–DME mixtures, at 2500 rpm; WOT and λ = 1.0

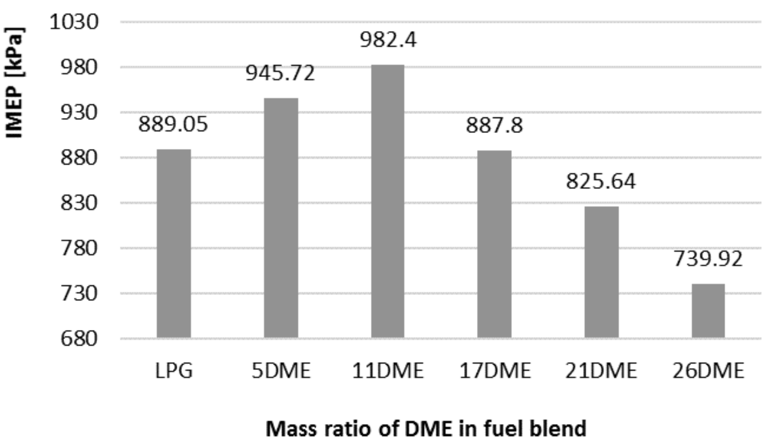


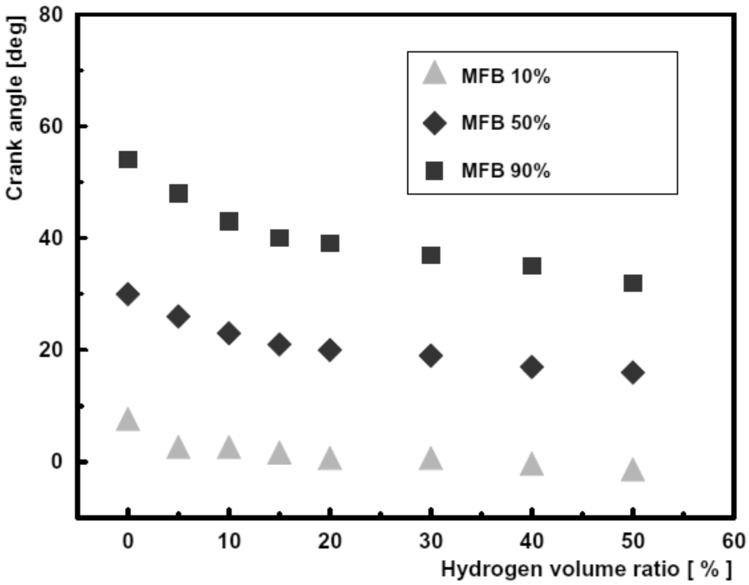
Fig. 7. IMEP for the engine fueled by LPG-DME mixtures, at 2500 rpm; WOT and λ = 1.0

4.3. Burn Rates

Fig. 8 shows typical mass fraction burned (MFB) curves obtained from the GT-Power reverse-run simulations.

For a given fuel, at constant λ and spark advance, as well as with reduction of the final MFB inside the cylinder, the burn rate also falls as the air–fuel mixture is progressively diluted. When comparing results between the different fuels, it is evident that natural gas has the lowest burn rate and is considerably more susceptible to dilution than the hydrogen-bearing fuels. This is consistent with its lower laminar flame speed and narrower flammability limits.

On the basis of MFB, presented in Fig. 8, it can be noticed that mixtures featuring DME shares of 5% and 11% not only initiate the combustion faster but also the dynamics of their combustion is higher than that for LPG. For these mixtures, the combustion duration is also shorter, whereas the DME share rising over 11% retards the combustion initiation and prolongs the process of combustion. This tendency is characteristic for the entire range of engine speeds.



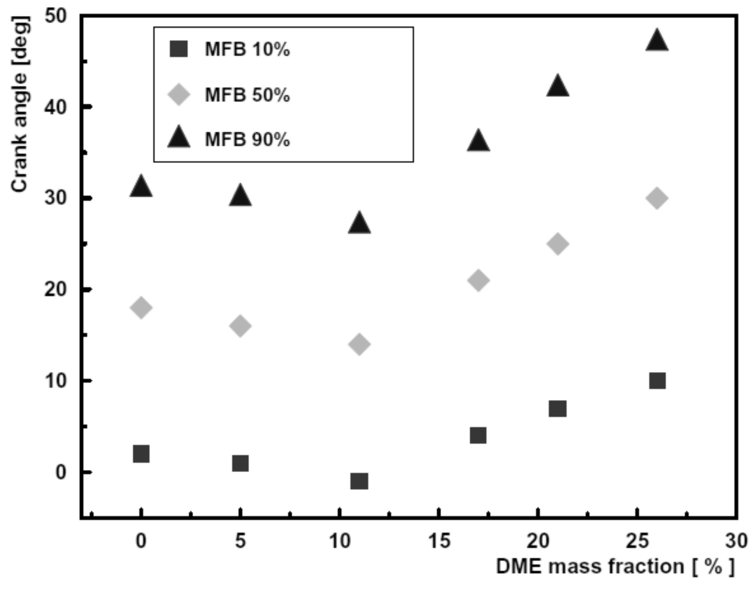


Fig. 8. Variations of CA corresponding to 10, 50, and 90% MFB for various cases of hydrogen share and DME mass fraction; 2500 rpm, load 100%, spark advance 30°, $\lambda = 1.0$

5. EFFICIENCY OF ENERGY CONVERSION

One of the most important factors describing the efficiency of energy conversion in the IC engine is definitively brake-specific fuel consumption [g/kWh]. The value of this indicator in the case of the described research has been calculated on the basis of the fuel consumption and engine power measurements acquired during completed stand tests. The individual fuel consumption of an engine powered by methane with hydrogen addition decreases with increasing volume of hydrogen in the mixture: Fig. 9.

Decreasing the brake-specific consumption affects the efficiency of the engine; a change of efficiency is shown in Fig. 9. The increasing efficiency of the engine with rising hydrogen amount in the blend is, however, limited, because it is connected with the risk for knocking combustion occurrence. The presence of this phenomenon can be noticed in the acquired in-cylinder pressure signal, and was audible in the engine during the test.

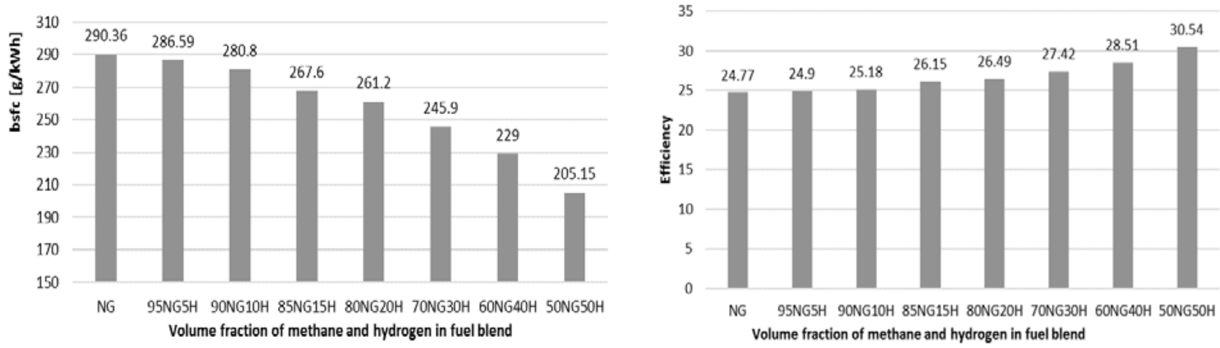


Fig. 9. Brake-specific fuel consumption and engine efficiency for all methane and hydrogen mixtures

In the case of LPG and DME mixture the bsfc and overall efficiency evolution have been presented in Fig. 10. Overall efficiency values have been averaged for all recorded points, obtained for defined DME shares in the mixture, and for different engine speeds at WOT. Higher values of overall efficiency have been obtained for the mixtures featuring DME shares from 5 to 11%. In this range, the DME share allowed obtaining efficiency comparable to that of LPG propulsion. Generally, an increase of DME content in the blends caused a slight overall efficiency drop.

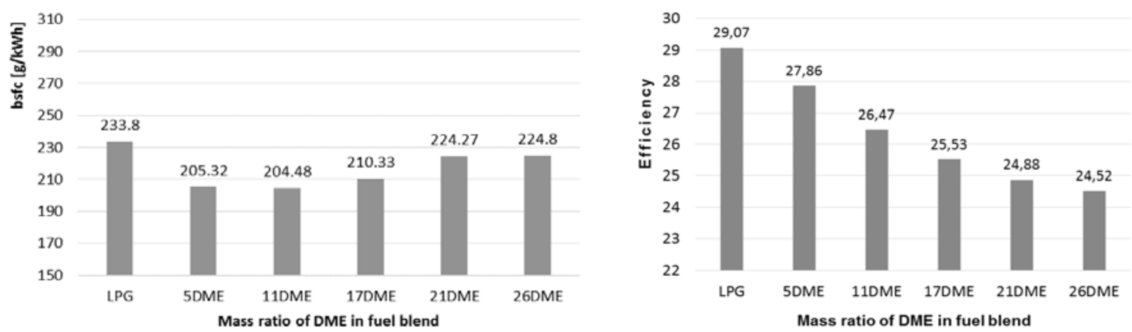


Fig. 10. Brake specific fuel consumption and Engine efficiency for all LPG DME blend

6. CONCLUSIONS

Research covered stand tests of the vehicle equipped with the SI engine, fueled with methane–hydrogen and DME–LPG blends. It was possible to identify the influence of hydrogen volume rate and DME mass share in the mixtures on the overall engine performance, its efficiency, and on the combustion process. Moreover, carried out tests give a possibility to understand fundamental combustion properties of HNG and LPG+DME blends, which is important for developing advanced NG- and LPG-based combustion engines with necessary operating strategies.

The chemical energy of the charge enclosed in a combustion chamber is released during the combustion process and the speed and energy profiles depend on the gaseous fuel composition.

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 methane, hydrogen, DME, and LPG an improvement in the conditions for the flame initiation and shortening of the charge-heating period are expected.

The following has been shown:

- stoichiometric combustion of HNG and LPG–DME mixtures reduced ignition delay and created conditions for faster burn.
- HNG reacts faster than pure methane, allowing shortening of combustion duration, especially, in the first phase of flame development.
- LPG and DME blend reacts faster than pure LPG but only for mixtures that consist of no more than 11% of DME, by mass ratio.
- heat release during combustion of HNG compares to that for pure methane, increasing for mixtures containing more than 15% of hydrogen. Increasing HR does not exceed 4.5% for the tested engine.
- the share of DME in the mixture of LPG–DME significantly influences changes in HR for the tested engine.

Future experimental development would foresee the optimization of emissions of both pollutants and CO_2 along with the reduction of fuel consumption during 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,
- EGR.

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