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Comparison of operational parameters and stability of performance of an automotive SI engine powered by methyl and ethyl alcohols

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The article presents research results performed using automotive spark ignition engine run on methyl and ethyl alcohols as well as gasoline as a reference fuel. The research was performed on automotive engine of Fiat 1100 MPI type. The aim of the conducted analysis was to compare the operational parameters of the engine. These parameters included engine power output, overall efficiency, emissions of toxic components in exhaust gases, and a comparison of the combustion process course. This comparison was conducted based on recorded and averaged indicator diagrams, as well as individual combustion cycles. The averaged diagrams were used in analysis of the pressure course during combustion, in analysis of pressure growth rate and heat release rate. Diagrams of individual combustion served for assessment of operational smoothness of the engine when fuelled with alcohols. As the reference, parameters measured in case of gasoline fuelling were used.

Key words: *methyl and ethyl alcohols, operational parameters, overall efficiency, indicated pressure, smoothness of engine operation*

1. Introduction

For nearly two decades, intensive research has been carried out worldwide on the gradual substitution of traditional petroleum-derived fuels with alternative fuels [2, 3, 5, 13, 14, 17, 19, 20, 21, 23–25]. Diversification of energy sources and ecological aspects are the main objectives of this research. Parallel to the research, in many countries regulations imposing an obligation to use biofuels as admixtures to gasoline and diesel oil, but also as homogenous fuels were introduced [4, 7, 10]. The basic biofuels include (used mainly in spark ignition engines) and esters of unsaturated fatty acids, used mainly to feed compression ignition engines. The biofuels are produced from biomass, an annually reproducible source of energy, obtainable in large quantities [1, 12–14, 16]. The biomass is created in the process of photosynthesis, using carbon dioxide from the atmosphere and solar energy. Under natural conditions, the biomass undergoes biological decomposition, which is accompanied by the emission of carbon dioxide and methane. As a result of the combustion of biofuels in combustion engines carbon dioxide is produced only, without emission of methane CH_4, having a considerably stronger greenhouse effect, compared with CO_2. Considering circulation of carbon dioxide in atmosphere, the process taking place in the engine is therefore much more advantageous than process of natural decomposition of the biomass. Some authors even believe that aggregated process of biomass formation, production, and combustion of biofuels, is close to zero emission in terms of carbon dioxide [1, 18]. Taking the above into consideration as well as unlimited resources of biomass, it can be assumed that importance of biofuels should increase in the future.

The obligation to use biofuels in Poland is due to the accomplishment of the National Indicator Targets (NCW in short), which assume a gradual increase in share of renewable fuels in the total amount of consumed engine fuels. Following the Regulation of the Polish Council of Ministers on the NCW of July 20th 2013, energetic share of biofuels in the year 2020 should reach the level of 10.0% [5, 22].

Legal regulations in the area of toxicity of exhaust gases mean that the share of compression ignition engines will decrease in the future. This will particularly apply to power systems of passenger cars driven mainly in the cities. It is anticipated that spark ignition engines will dominate in the future, while some of them will be powered with biofuels. From the point of view of the benefits of alcohol fuelling and the easiness of production, primary alcohols, like methylene and ethylene, will be of the greatest importance [8, 11].

Primary alcohols due to their perfect properties, mainly high octane number, high evaporation heat and high combustion rate, can be successfully used both in spark ignition engines (as homogenous fuels or as additives to traditional fuels), and in compression ignition engines as additives combusted simultaneously with diesel oil [9, 15, 17]. The advantages of alcohols cause that research on alcohol fuelling of engines is carried out presently in many research centres around the world.

The ethyl alcohol is produced in process of fermentation of plant materials (sugar cane, cassava, cereals, potatoes) and cellulose from timber waste. Production of ethanol from cellulose, which is intensively developed in Sweden and other Scandinavian countries, seems to be particularly attractive [20]. It allows for obtaining a significant amount of fuel without reduction of grocery products, which is an issue often raised by environmentalists. It seems that methyl alcohol produced from coal or natural gas may also play an important role in the future [1]. New technology for synthesis of methanol from carbon dioxide and hydrogen, which was developed in the recent years, also promises a certain future.

Both methyl and ethyl alcohols are fuels containing significant amount of oxygen in their molecule, and thus in combustion process in engine conditions, leading to reduction in emissions of toxic exhaust components, mainly nitrogen oxides and particulate matter.

Table 1. Properties of the fuels [1, 4, 11, 18]

Properties	Gasoline	Diesel oil	Methanol	Ethanol
Molecular formula	C_7H_{16}	$C_{14}H_{25}$	CH_3OH	C_2H_5OH
Octane/cetane number	95/~22	~15/51	115/~3	/~11
Lower heating value, MJ/kg	42–44	42.5	19.5	26.9
Density at 20°C, kg/m^3	720–760	820–845	792	789.5
Viscosity at 40°C, mPa s	1.07	3.11	0.57	1.1
Kinematic viscosity, mm^2/s	0.57	2.419	0.75	1.51
Heat of evaporation, kJ/kg	315–350	250	1101	841.5
Stoichiometric air fuel ratio	14.9	14.5	6.52	9.0
Explosion limits: lower, %vol upper, %vol			 5.5 44	 3.3 19
Flash point, °C	36	78/56	11.0	18.3
Auto-ignition temperature, °C	280/480–550	250/330–350	455	425
Flame speed, m/s		0.86		~3
Flame temperature, °C		2054	1890	2120
Carbon content, %wt	85.5	87	37.5	52.2
Hydrogen content, %wt	14.5	13	12.5	13
Oxygen content, %wt	0	0	50.0	34.8

Carbon dioxide and water are the products of the combustion of methyl and ethyl alcohol, while such a process takes place according to the following reaction:

$$\mathrm{CH_3OH} + \frac{3}{2}\mathrm{O_2} \rightarrow \mathrm{CO_2} + 2\mathrm{H_2O} + 29.7\,\frac{\mathrm{MJ}}{\mathrm{kg}} \tag{1}$$

$$\mathrm{C_2H_5OH} + 3\mathrm{O_2} \rightarrow 2\mathrm{CO_2} + 3\mathrm{H_2O} + 22.68\,\frac{\mathrm{MJ}}{\mathrm{kg}} \tag{2}$$

The mass fraction of carbon atoms in molecules of alcohol is lower compared to traditional fuels, and amounts to 0.375 for methyl alcohol, and 0.520 for ethyl alcohol, while for gasoline and diesel oil this ratio is approximately 0.845–0.850. However, taking into account differences in calorific values, the generation of the same unit of energy from alcohol results in only a slight, within 2%, reduction of CO_2 emissions in relation to gasoline.

The spark ignition engines are most commonly powered by mixtures of anhydrous alcohol, its esters, and gasoline. Due to the tendency for delamination of alcohol-gasoline mixtures at low temperatures and in the presence of water, the total content of alcohol and application time of the mixtures are limited. For this reason, it seems beneficial to feed the spark ignition engines with alcohols as homogenous fuels. Consequently, it is possible to use water-logged alcohols obtained directly from the distillery, which considerably reduces the costs of their production. Moreover, fuelling with homogenous alcohol makes it possible to take full advantage of the high anti-knocking resistance of alcohols through the increase in compression ratio. This results in an increase in overall efficiency and unit power output of the engine [2, 9]. The alcohol, as homogenous fuel, can be used in fleets of vehicles like taxis, post office cars, delivery vehicles or others. For example, a large number of cars powered by ethyl alcohol produced from cassava and sugar cane have been operated in Brazil for a long time [5], their number is around 2 million.

Currently, new possibilities to use alcohol additives to gasoline, or alternating usage of gasoline and alcohol are created by the common use of multipoint injection of light fuels in engine fuelling systems. With small modifications in the fuel supply system, engine start-up and its heating-up can be performed with gasoline alone, while further operation of the engine goes with alcohol fuelling [20]. In such engines, the compression ratio may be increased by 2.0–3.0 units, which should increase the power output and thermal efficiency of the engine. Research shows that also in the area of partial engine loads, the efficiency of the engine powered by alcohol only is higher, which is of the highest importance in the case of automotive spark ignition engines operated in cities.

In this study the test results of the FIAT 1100 MPI engine, powered alternately with gasoline, methyl alcohol and ethyl alcohol, were presented. The subject of the research was to assess operational parameters of the engine, to compare some selected combustion parameters and assessment of stability of the engine operated on alcohol. Parameters of combustion process were calculated based on registered indicator diagrams, averaged for 50 successive cycles. Smoothness of engine operation was assessed based on an analysis of 50 successive cycles of individual combustion. Parameters from individual combustion diagrams were processed using mathematical statistical analysis. On this basis, stability of engine operation, so-called – *cycle by cycle* – was evaluated. The same parameters obtained in the case of gasoline fuelling were used as reference values.

Performed bench tests indicate stable operation of the engine powered by both methyl and ethyl alcohol, which enables usage of both alcohols in normal engine operation. Performance parameters, such as power output, overall efficiency, emissions of nitrogen oxides and hydrocarbons, as well as exhaust gas temperature, have clearly undergone improvement.

2. Test bench

The bench tests were performed on four cylinders, spark ignition, multipoint injection engine of Fiat 1100 MPI type. Technical data of the engine are specified in Table 2.

For the purpose of engine indication, a hole was drilled for an adapter of a non-cooled GH13 sensor in the second cylinder of the cylinder head. The INDIMETER 619 system from AVL was used to register the quick-changing pressures in the combustion chamber.

Table 2. Technical data of Fiat 1100 MPI engine

Engine type	Fiat 1100 MPI
Bore × stroke	70 × 72 mm
Swept capacity	1108 ccm
Compression ratio	9.6
Rated power/rotational speed	40 kW/5000 rpm
Max torque/rotational speed	88 Nm/3000 rpm

A system for automatic acquisition of measurement data to the Excel calculation sheet was installed on the test bench. For the particular requirements of the testing, a dual system of fuelling for alcohol and for gasoline was installed in the test bench. Each of the systems was equipped with an individual fuel pump and pressure stabilization system, and a system to measure the fuel consumption. The fuelling system enabled control of instantaneous consumption of the fuels, which considerably facilitated selection of engine adjustment and recording of time of consumption of a determined dose of fuel. A view of the test bench is presented in Fig. 1.

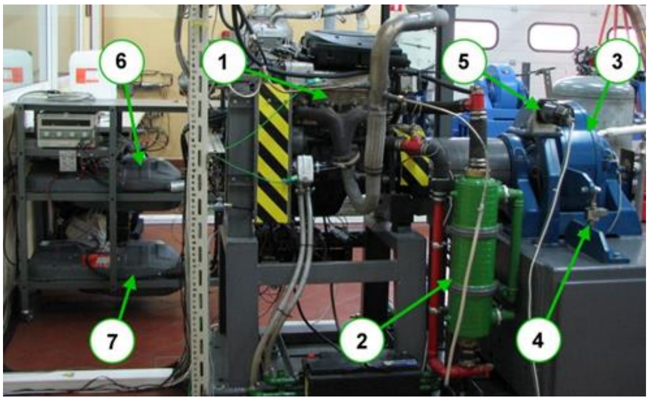

Fig. 2. Test bench – general view: 1 – research engine of Fiat 1100 MPI type, 2 – radiator, 3 – eddy-current engine brake made by Schenck, 4 – transducer of engine load, 5 – engine speed sensor, 6 – alcohol tank, 7 – gasoline tank

3. Analysis of the results of the research

3.1. Operational parameters of the engine running on methanol and ethanol

The use of alcohols has a positive effect on maximal power output, which is shown in Fig. 2. In the complete range of the investigated rotational speeds of the engine of 2000–3500 rpm the increase of the power output for alcohol feeding can be seen. This can be evaluated based on the relative increase of the power output, calculated from the following formula:

$$\mathrm{DNe_{Alc}} = \frac{\mathrm{Ne_{Alc}} - \mathrm{Ne_{G95}}}{\mathrm{Ne_{G95}}} \cdot 100\% \qquad (3)$$

where: $\mathrm{DNe_{Alc}}$ – relative increase of the power output when supplied with methyl or ethyl alcohol, $\mathrm{Ne_{Alc}}$ – power output when supplied with alcohol, $\mathrm{Ne_{G95}}$ – power output when supplied with G 95 gasoline.

Relative increase in the power output reached 1.7–3.3% when the engine was run on ethyl alcohol and 3.8–7.7% in case of methyl alcohol. A higher growth of the power output, when fuelled with methanol, could be caused by a higher combustion rate of combustible mixture and effect of a higher value of vaporization heat of methanol injected near the inlet valve, which resulted in better filling and smaller losses of the heat to cylinder walls during compression. It is worth underlying that these parameters were achieved for an unchanged compression ratio and without optimization of ignition advance angle. It can therefore be assumed that the engine performance when fuelled with alcohols may be even higher due to optimization of compression ratio and adjustment of ignition angle.

Conclusions about the beneficial effect of alcohols on the combustion of fuel charge can be confirmed by comparison of the overall efficiency of the engine shown in Fig. 3. This indicates that in the complete range of changes in the engine load, the efficiency when the engine was supplied with alcohols was higher than the efficiency when it was supplied with gasoline:

– with 1.2–2.5% for ethyl alcohol
– with 1.5–7.5% for methyl alcohol.

a) b)

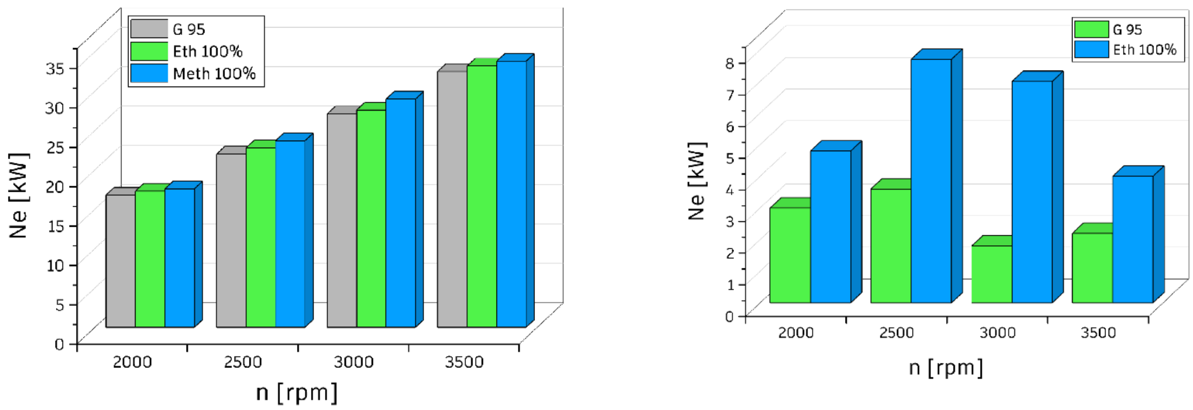

Fig. 2. Comparison of the power output and relative change of the power output in Fiat 1100 MPI engine run on G 95 gasoline and Eth ethyl alcohol and Meth methyl alcohol

A distinct increase in the efficiency when the engine was run on methanol was observed in the entire range of engine load changes, and the difference increased with increasing load. For ethyl alcohol, significant differences occurred in the range of medium and maximum engine loads, when an increase in the efficiency of 1.5–2.5% in absolute values was observed. For lower engine loads, gasoline and ethanol fuelling efficiency was comparable, especially for higher rotational speeds. Bigger changes in the efficiency as the engine load increases may be caused by the increase of temperature in the combustion chamber and of the temperature of fuel charge during compression, which promotes faster evaporation of alcohols, improves homogeneity of combustible mixture and promotes acceleration of combustion rate in the initial stages of the process.

It seems that both increase in maximum power output, as well as in the engine efficiency, when fuelled with methanol, is caused by a higher combustion rate of methanol in relation to ethanol and gasoline, which results in lower thermal losses in the cycle. Simultaneously, higher evaporation heat of methanol reduces the charge temperature during compression and at the beginning of combustions, which can lead to reduced mechanical losses during compression stroke. Both the above mentioned factors increase thermal efficiency of the engine.

A comparison of concentrations of nitrogen oxides NO_x and Total Hydrocarbons Content THC is presented in Fig. 4. From the point of view of ecology, it is especially important to reduce concentration of NO_x in the exhaust gases when fuelling with both alcohols, occurring in the entire range of engine load changes, as shown in Fig. 4a and 4b. Differences in concentrations of NO_x were respectively 15–0% for methyl alcohol and 35–75% for ethyl alcohol. Compared to gasoline, differences in concentrations of NO_x for both alcohols were growing as the engine load increased, and the biggest differences were observed in the area of medium and maximal engine loads. A greater reduction in the concentration of NO_x, observed for ethanol is particularly important, because this alcohol is used in some regions of the world as admixture to gasoline or as homogenous fuel.

The fact of NO_x emissions reduction when fuelling with alcohols was also signalled in some previous reports found in the literature [1, 9, 18], while the analysis of the obtained results of the research indicates a considerable scale of such phenomenon. Due to huge difficulties in the reduction of nitrogen oxides by catalytic converters, the advantageous feature of the combustion of alcohols, especially ethanol, described here requires special emphasis.

It seems that shorter combustion time of methanol-air and ethanol-air mixtures, and lower temperatures of the flame of alcohol vapours are the reasons for the reduction of NO_x quantities generated in the combustion process of alcohols. Temperatures of the working medium behind the front of the flame, which are lower in the case of alcohol vapours, have an especially important effect [8, 11]. This is due to the fact that the effect of time on quantities of generated NO, the main constituent of nitrogen oxides in exhaust gases of the spark ignition engine, is linear. In contrast, the effect of temperature is exponential, which considerably influences quantities of generated NO.

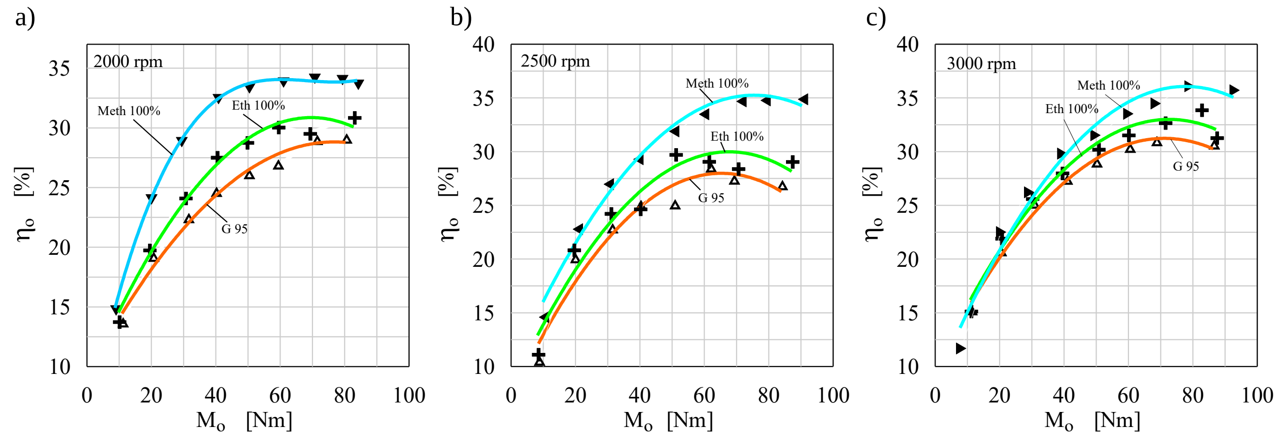

Fig. 3. Comparison of overall efficiency of Fiat 1100 MPI engine fuelled with G 95 gasoline and Eth 100% ethyl alcohol and Meth 100% methyl alcohol

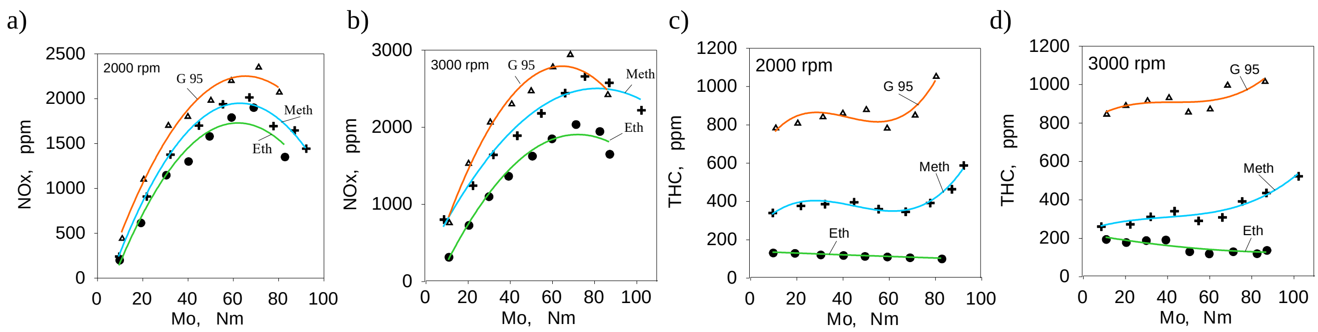

Fig. 4. Comparison of concentration of toxic components of exhaust gases from Fiat 1100 MPI engine run on G 95 gasoline and ethyl alcohol Eth and methyl alcohol Meth

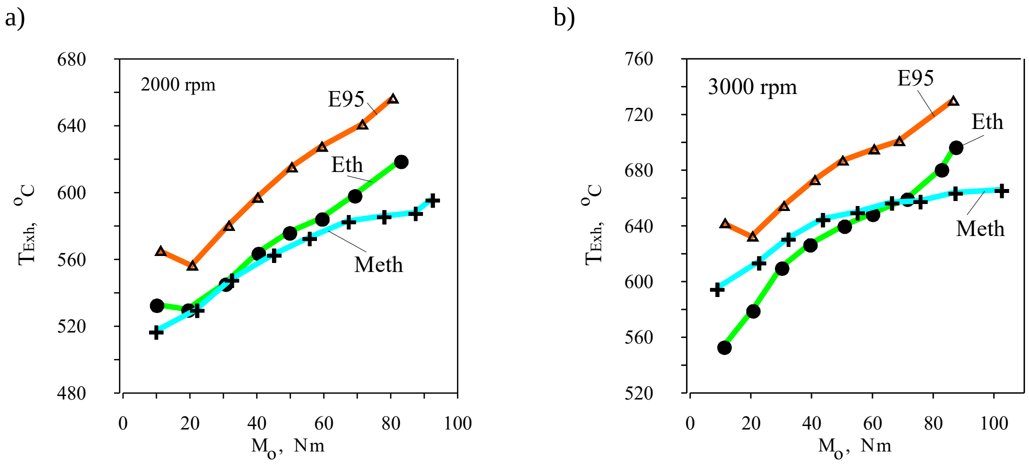

Fig. 5. Comparison of the exhaust gases temperature in Fiat 110 MPI engine run on gasoline and methyl and ethyl alcohols

The combustion of alcohols has advantageous effects on emissions of THC, as shown in Fig. 4c and 4d. At methanol fuelling, concentrations of THC in a complete range of change of engine loads were nearly 2–3 times lower than gasoline fuelling. Even greater differences in concentration of THC were observed at ethanol fuelling, where the reduction was 4.5–8.2 times.

Such significant changes in THC concentration can be explained by the different chemical composition of methanol, CH_3OH, and ethanol, C_2H_5OH, compared to gasoline that is a mixture of different hydrocarbons with complex chemical composition. Simple chemical structure and the presence of oxygen in molecule of alcohols contribute to rapid and complete oxidation of these fuels. Oxygen contained in the molecules of alcohols is released in the combustion process directly in zones of the reaction, which considerably accelerates oxidation of carbon and hydrogen atoms. The additional factors contributing to the rapid oxidation of alcohols are the higher combustion rates of alcohol vapours and thus, higher temperatures of the entire load during the initial stages of combustion compared to the ones occurring in the case of fuelling with gasoline. In the case of gasoline combustion, the oxidation time of higher hydrocarbons increases as the number of carbon and hydrogen atoms increases, which in high-speed spark ignition engines may affect concentrations of total hydrocarbons in the exhaust gases.

Faster combustion of alcohol vapours causes that temperatures of the exhaust gases downstream of the exhaust valve are lower than the ones observed in the case of gasoline fuelling, as in Fig. 5. The temperature differences are present in the entire range of load variations and increase with the increase of the engine load. Depending on engine load and rotational speed, the observed temperature differences amounted to 45–90°C for methanol and 30–60°C for ethanol. It can be assumed, that lower temperatures of exhaust gases have an advantageous effect on the durability of the engine fuelled with alcohols.

3.2. Comparison of combustion parameters for methanol and ethanol fuelling

The pressure trace in the second cylinder was recorded using a piezoelectric sensor type GH13 made by AVL mounted in the engine's cylinder head. Average indicator diagrams for 50 successive cycles of engine operation and pressure runs of 50 individual and successive cycles of engine operation were the subject of the analysis.

Figure 6 presents a comparison of average cylinder pressure during combustion for two selected rotational speeds and different loads of the engine running on methanol and gasoline. From this comparison, it can be concluded that considerably higher maximal pressures are present during the combustion of methanol. The heat release process for methanol is greater at the beginning of combustion, which increases rate of pressure rise compared to gasoline. Such a tendency is noticeable for all engine loads and for all rotational speeds. It was also ascertained that for medium and maximum engine loads, when methanol is combusted, the maximal pressures are reached faster than gasoline feeding, thus approaching TDC. This is one of the reasons for increasing engine power output and efficiency [9, 20]. A different tendency was observed for the lowest engine loads (green colour), where in the case of methanol, the maximum pressures were reached slightly later than for gasoline.

In the case of ethanol supply (Fig. 7), similar tendencies to those discussed earlier for methanol occur for lower rotational speeds of 2000 rpm, where for the engine load of 30–80 Nm, both maximal pressures and pressure growths at the beginning of combustion are higher comparing to gasoline. For higher rotational speeds of 2500–3000 rpm, the values of the average pressures in the course of combustion for ethanol and gasoline are similar. It is worth emphasizing, however, that also for rotational speeds discussed here, in the range of medium engine loads, a tendency to slightly higher pressures occurring when feeding with ethanol can be noticed.

In a spark-ignition engine, there is a significant uniqueness of work from cycle to cycle. This is caused by changes in the quality of the fuel charge in subsequent suction cycles but mainly by changes in the delay time of combustion and the formation of a stable flame around the spark plug. To check courses of the pressure in individual cycles for the combustion of ethanol and gasoline, the indicator diagrams of individual combustion for rotational speed of 3000 rpm and for two engine loads, minimal 10 Nm and maximal 86

Nm, were compared in Fig. 8. In the Figures, the courses with the maximum pressure close to the average value, calculated from 50 consecutive cycles of engine operation are depicted in black, and two courses with the maximum pressures lower are depicted with coloured dashed lines while the ones with the maximal pressure higher than the average are presented with continuous lines.

The comparison shows that lower maximal combustion pressures are accompanied by later angles of termination of the kinetic combustion phase (reaching the maximal pressure), depicted with dashed lines. This phenomenon is more clearly visible for low engine loads, such as in Fig. 8c and 8d. Protracted combustion is visible for the minimal engine load of Mo = 10 Nm in cycles with low maximal pressures, as illustrated in Fig. 7a and 7b. This is due to a longer ignition delay, which causes active combustion to shift towards later angles after TDC, accompanied by higher pressures in the latter stages of combustion and expansion.

The comparison outlined in Fig. 8 shows that for both tested fuels, ethanol and gasoline, both the combustion courses at high pressures p_{max} and at low pressures are similar. The heat release rate dQ/dα can be a measure of the rate of fuel combustion. Its value was calculated based on the recorded indicator diagrams. To perform the calculations, an author's proprietary computer program designed to analyse indicator diagrams was used, which was developed by the Combustion Engines and Automobiles Department of Bielsko-Biala University, described in detail in the publications [20]. The averaged diagrams from 50 consecutive cycles were used as representative cycles for engine operation in a selected points defined by n-Mo (rotational speed and torque). As a result of the calculations, the courses of combustion parameters were obtained, such as: pressure increase rate, heat release rate, the temperature of the working medium, composition of the charge as a function of crank angle, as well as aggregated and maximal values of selected parameters. An analysis of only some of the calculated parameters is presented in this paper.

Comparison of values of maximal heat release rates $(dQ/d\alpha)_{max}$ for the investigated fuels confirms greater dynamics of the heat release during the combustion of alcohols, and especially of methanol. In the complete range of changes of the engine load, for both analysed rotational speeds, values of $(dQ/d\alpha)_{max}$ are higher for alcohol than the ones calculated for gasoline. It is worth emphasizing at this point that the differences in the heat release rate for

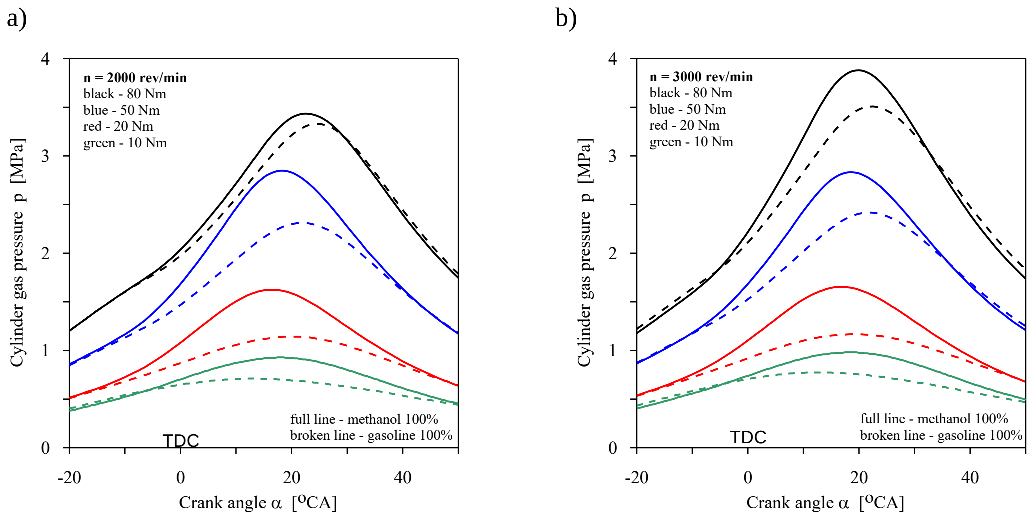

Fig. 6. Comparison of cylinder pressure in function of crank angle in Fiat 1100 MPI engine run on gasoline and methanol for a various engine loads

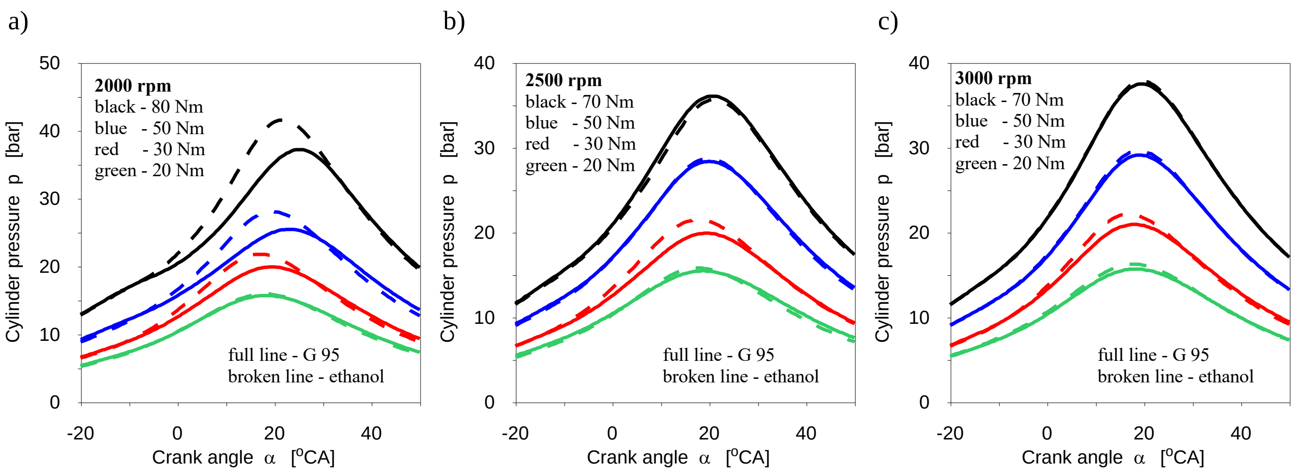

Fig. 7. Comparison of cylinder pressure in function of crank angle in Fiat 1100 MPI engine run on gasoline and ethanol for a various engine loads

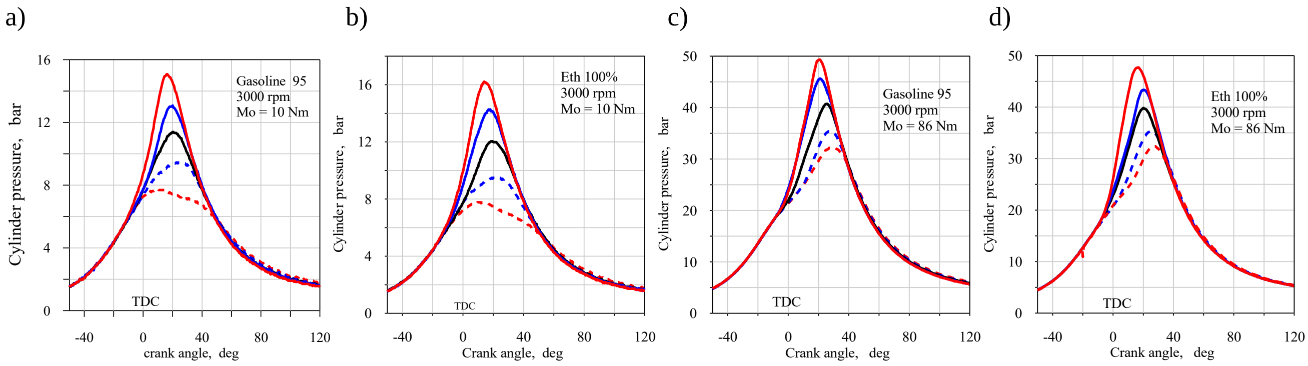

Fig. 8. Comparison of cylinder pressure in function of crank angle in Fiat 1100 MPI engine run on gasoline and methanol for a various engine loads

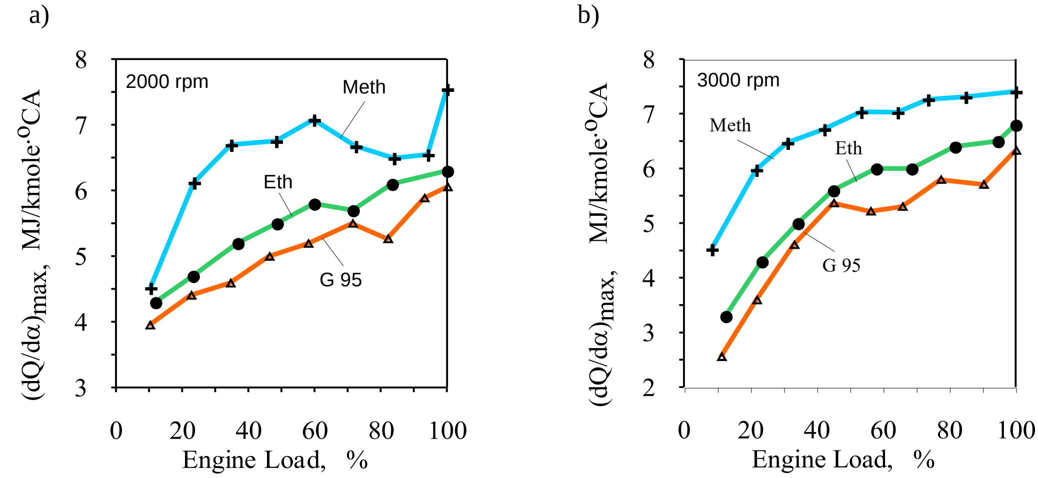

Fig. 9. Maximal heat release rate in Fiat 1100 MPI engine fuelled with gasoline, ethanol and methanol, for various rotational speeds and engine loads

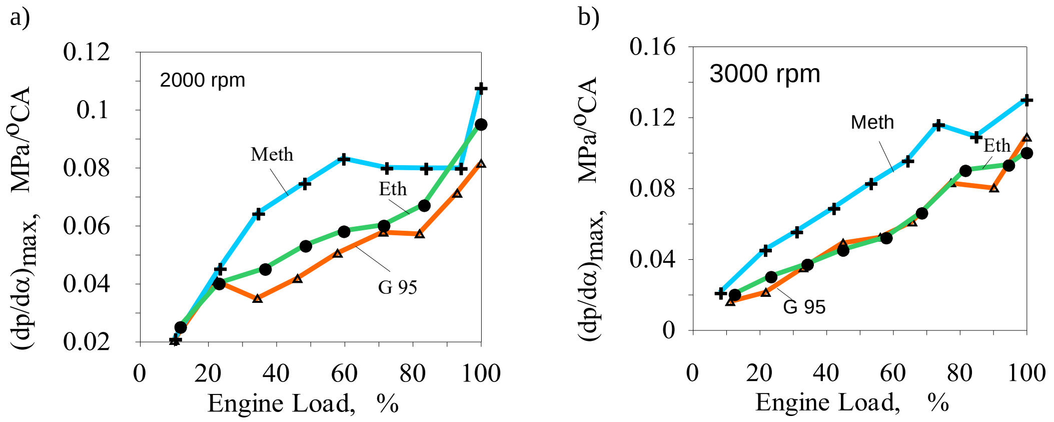

Fig. 10. Maximal rates of pressure growth in Fiat 1100 MPI engine run on gasoline and methanol for various rotational speeds and engine loads

ethanol are only slightly greater within the range of 10–18%, while for methanol are clearly greater by 50–80% compared to those recorded when fuelling with gasoline (Fig. 9).

As a measure of engine operational noise, the value of the maximal pressure increase rate $(dp/d\alpha)_{max}$ may be used, in the interval from the beginning of the combustion to the moment when the maximal pressure is reached. The analysis of this parameter, presented in Fig. 10, shows that operation of the engine fuelled with methanol is characterized by rougher run of the engine, which is evidenced by higher values of $(dp/d\alpha)_{max}$. Value of $(dp/d\alpha)_{max}$ for maximal engine loads exceeds 0.1 MPa/°CA and is higher with about 0.025 MPa/°CA in relation to gasoline fuelling. Also, in the range of medium engine loads, the values of $(dp/d\alpha)_{max}$ for methanol are distinctly higher compared to gasoline. However, due to the fact that they are lower than 0.9 MPa/°CA, this should not have any significant effect on the operational roughness of the engine run on methanol.

Higher values of $(dp/d\alpha)_{max}$ for methanol are connected with higher combustion rate of methanol-air mixture in the first stage of kinetic combustion, which results in a faster growth of pressure and heat release rate $(dQ/d\alpha)_{max}$ as presented in Fig. 7 and Fig. 9.

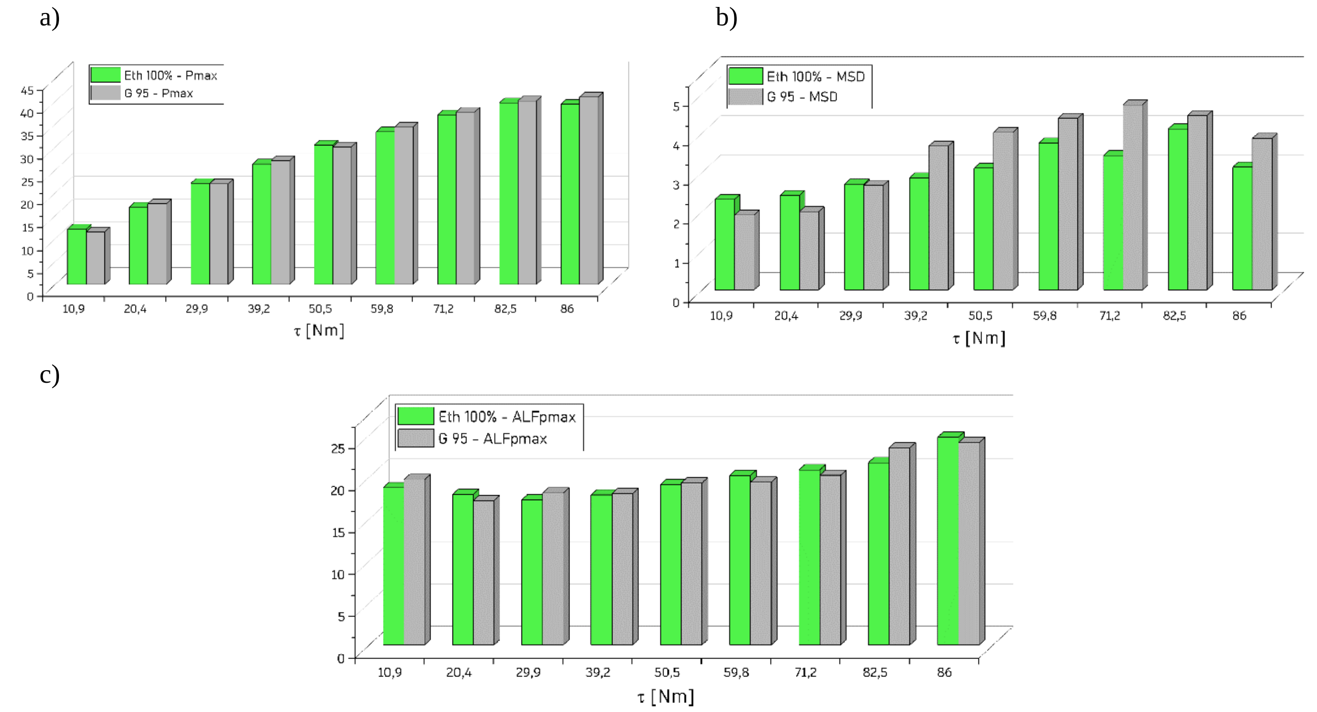

Fig. 11. Comparison of parameters calculated from diagrams of individual combustion for the engine fuelled with ethanol and gasoline: a) mean value of maximal combustion pressure, b) mean standard deviation σ_{pmax} of the maximal combustion pressures, c) maximal pressure variation coefficient s_p

The values of $(dp/d\alpha)_{max}$ for supply with gasoline and ethyl alcohol are similar, especially in the case of the higher rotational speed of 3000 rpm, as presented in Fig. 9. Therefore, it can be assumed that supply of the engine with ethyl alcohol will not affect the noise levels of its operation.

3.3. Smoothness of the operation of the engine powered by alcohol

The spark ignition engines are characterized by greater operational smoothness cycle by cycle in relation to the compression ignition engines. This is due to cyclical changes in mass of the intake air, the course of the ignition spark, changes in the ignition delay time, the time of a stable front flame formation, and changes in the combustion rate depending on the composition of the homogeneous combustible mixture. The majority of fuels for the spark ignition engines is characterized by narrow flammability limits and a large change in combustion rate, dependent on excess air coefficient λ. Due to ecological reasons, modern spark ignition engines operate on stoichiometric mixtures and the excess air coefficient λ = 1.0 is controlled precisely by the oxygen sensor (λ sensor). With stochastic changes in amount of the air sucked in, the oxygen sensors can act with some delay, causing that for a number of cycles, changes in the excess air coefficient can occur around the value of λ = 1.0. It is also worth mentioning here that the lambda sensors available on the market were developed for petroleum-derived fuels. When using alcohols, i.e. fuels with molecules significantly different than gasoline hydrocarbons, there is a need to investigate the impact of fuel on the repeatability of cycles of engine operation. It is also important to check if there is no misfiring during the combustion of alcohols. This was the basis to undertake research in this area.

The assessment of smoothness of engine operation for gasoline and alcohol fuelling was performed based on recorded diagrams of individual combustion for 50 successive cycles of engine operation. The following parameters were used for the assessment:

$$\bar{p}_{max} = \frac{\Sigma p_{max-i}}{n} \tag{4}$$

$$\sigma_{p_{max}} = \sqrt{\frac{\Sigma(p_{max-i} - \bar{p}_{max})^2}{n-1}} \tag{5}$$

$$s_p = \frac{\sigma_{p_{max}}}{\bar{p}_{max}} \cdot 100\% \tag{6}$$

where: $\bar{p}_{max}$ – mean value of maximal pressure for n successive cycles, p_{max-i} – maximal pressure of the next cycle, n – number of analysed cycles, σ_{pmax} – mean standard deviation, s_p – PVC – pressure variation coefficient.

Figure 11a presents a comparison of the average maximal pressures calculated on the basis of indicator diagrams of individual combustion for 50 successive cycles of engine operation. The engine was operated at rotational speed of 3000 rpm and variable engine loads from the minimal of 10.9 Nm to the maximal of 86.0 Nm. This comparison shows that over the entire range of engine loads, the values of the mean pressures for gasoline and alcohol were similar. Also, in the case of other investigated rotational speeds of 2500 rpm and 3500 rpm, maximal pressures for the both fuels were similar.

The comparison of the values of the standard deviation σ_{pmax} and maximal combustion pressures shown in Fig. 11b indicates that for engine loads higher than 30 Nm, the engine operated on ethanol is characterized with bigger operational repeatability, cycle by cycle. This is evidenced by significantly lower values of σ_{pmax}. Only in the area of lower engine loads, higher values of σ_{pmax} are present in the case of ethanol fuelling, which proves that there are greater fluctuations of maximal pressures. Greater fluctuations of the pressures at low engine loads can be caused by a reduction in the charge temperature associated with higher vaporization heat of alcohol, and prolonged stabilization time of the flame after ignition.

The pressure variation coefficient s_p shown in Fig. 11c can prove the effect of pressure fluctuations on the stable operation of the engine. This coefficient decreases with increasing engine load, suggesting pressure fluctuations' diminishing effect on stable engine operation. A comparison of the s_p coefficient for both investigated fuels shows that the engine fuelled with ethanol in the range of mean and maximum loads is characterized by greater repeatability cycle by cycle. It demonstrates that this type of fuel supply brings about smoother and more stable operation comparing to standard operation on gasoline.

Fluctuations of the maximal pressures for individual combustion of gasoline and ethanol are presented in Fig. 12. From the comparison, it can be found that at the engine load close to the maximal M_o = 86 Nm, in successive cycles of engine operation, there are significant absolute differences in the pressures for both fuels. At the same time, slightly higher fluctuations of the maximum pressures for gasoline can be observed, Fig. 12a.

For the quantitative assessment of the fluctuations of combustion pressure, the absolute differences and relative differences of the maximal pressures related to the medium value of the maximal pressure calculated for 50 successive cycles of engine operation are presented in Fig. 13.

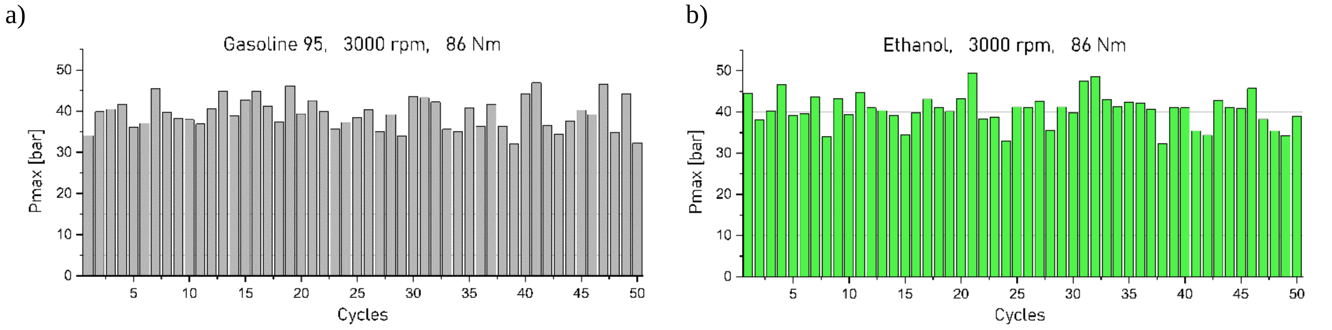

Fig. 12. Fluctuations of the maximum combustion pressures in successive operational cycles of Fiat 1100 MPI engine fuelled with gasoline and ethyl alcohol

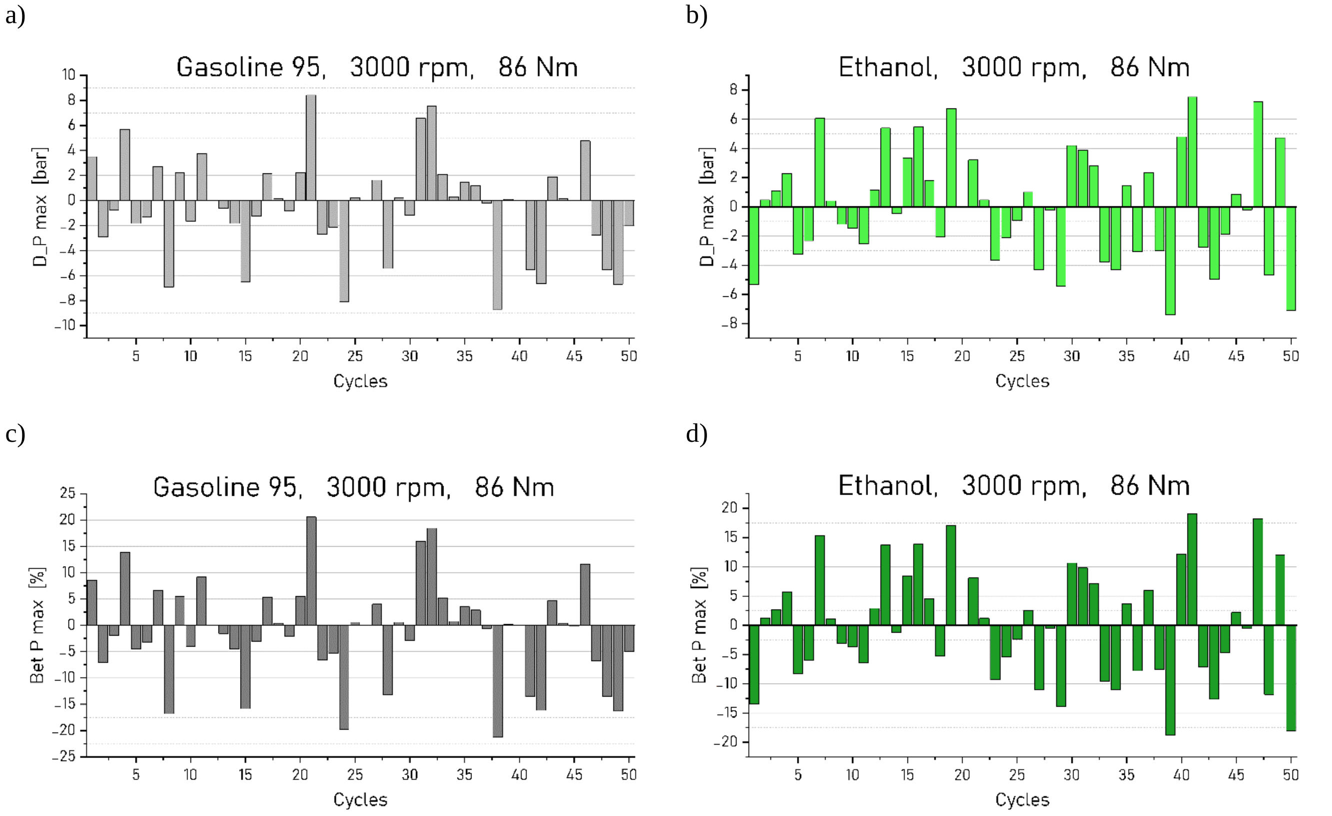

Fig. 13. Absolute and relative differences of maximal combustion pressures in successive operational cycles of Fiat 1100 MPI engine run on gasoline and ethyl alcohol

The differences were calculated from the following equations:

$$AD_{p_{max}} = p_{max-i} - \bar{p}_{max} \tag{7}$$

$$RD_{p_{max}} = \frac{Ap_{max}}{\bar{p}_{max}} \cdot 100\% \tag{8}$$

where: $AD_{p_{max}}$ – absolute difference of maximal pressure, $RD_{p_{max}}$ – relative difference of maximal pressure, $\bar{p}_{max}$ – medium value of maximal pressure calculated from 50 successive cycles of individual combustion.

The absolute differences of the maximal pressures for gasoline are high. The maximal value of $AD_{p_{max}}$ for the highest pressures exceeds +8 bar, and the minimal ones more than –8 bar, which results in maximal fluctuations of the pressures of about 17 bar, as depicted in Fig. 13a. For ethanol feeding, the values of $AD_{p_{max}}$ are slightly lower and fluctuate within the range of –7.2 to 7.4 bar, and hence, maximal fluctuations of the pressures are equal to 14.6 bar, as illustrated in Fig. 13b.

The relative values of pressure changes for gasoline amount to $RD_{p_{max}}$ = –21.5–20.5% (Fig. 13c), while for ethanol fuelling to $RD_{p_{max}}$ = –18.5–19.0% (Fig. 13d). Such changes should be evaluated as quite high. However, it seems that they should not have any adverse effect on loading of the crank system and the durability of the engine.

Both indicators, $AD_{p_{max}}$ and $RD_{p_{max}}$, are a little lower for ethanol, which proves the beneficial effect of ethanol fuelling on the smoothness of engine operation.

Regarding the conversion of chemical energy into mechanical work, not only the value of maximum combustion pressure is important, but so is the angle at which the p_{max} which respect to TDC is reached.

Both changes in the p_{max} as well as the value of the angle at which the p_{max} is reached in successive cycles of the operation cause fluctuations of the mean indicated pressure and the torque, having adverse effect on the cooperation of the engine with the receiver. High fluctuations in the cycles result in accelerated wear of bearings, clutches and transmissions. For this reason, smoothness of engine operation is of utmost significance and should be carefully evaluated when fuelling systems or fuel type is to be changed.

In Figure 14 a comparison of average values of the p_{max} angle for gasoline and ethanol fuelling is presented. The comparison was performed for the rotational speed of 3000 rpm and variable engine load. From the analysis of Fig. 14a, it can be perceived that the maximum p_{max} angles initially decrease, and then increase, as the engine load increases. This is connected with changes in the combustion course and fluctuations in successive cycles of the engine operation. At minimal loads of the spark ignition engine, a small amount of the charge is burnt in the combustion chamber at lower temperatures of the walls of the chamber and the working medium. It results in the prolongation of the first stage of kinetic combustion. At the same time, due to the lower energetic density of the charge, fluctuations in the course of combustion are more pronounced. As a result, at minimal engine loads, the time to reach the p_{max} is extended. With increasing engine load, amount of the sucked charge increases, and so does its energetic density, which increases the combustion speed and shortens the time to reach the p_{max}. However, a further increase in the engine load is related to an increase in the angle to reach the p_{max}, which is induced by the longer time of kinetic combustion of the growing quantity of the mixture.

From the comparison presented in Fig. 14a it may be inferred, that for the majority of the cycles, the maximal combustion pressures for both fuels were reached for the angles of 18–24°CR after TDC. Average values of these angles for alcohols and gasoline are similar, and the difference can be neglected.

Figure 14b presents a comparison of the variation coefficient of occurrence of the maximal combustion pressure angle calculated from the following formula:

$$s_{\alpha p_{max}} = \frac{\sigma_{\alpha p_{max}}}{\bar{\alpha}_{p_{max}}} \cdot 100\% \tag{9}$$

where: $s_{\alpha p_{max}}$ – coefficient of variation of the maximal combustion pressure angle, $\sigma_{\alpha p_{max}}$ – mean standard deviation of the p_{max} angle, $\bar{\alpha}_{p_{max}}$ – average value of the p_{max} angle calculated from 50 successive cycles of engine operation.

From the analysis of Fig. 13b, it may be concluded, that the effect of the changes in the $\alpha_{p_{max}}$ angle of individual cycles on the value of the coefficient of variation $s_{\alpha p_{max}}$ decreases with the increase of engine load. At the same time, in the range of average engine loads, the values of the $s_{\alpha p_{max}}$ for ethanol are lower than for gasoline. Only at the lowest and the maximal engine loads, the values of the $s_{\alpha p_{max}}$ for ethanol are higher than values for gasoline. The reason for this phenomenon in the case of the lowest engine load may be greater fluctuations in the combustion rate of ethanol caused by an excessive decrease in temperature of the charge due to the higher evaporation heat of alcohol, as mentioned before. On the other hand, in the case of maximal engine loads, different air excess coefficients of the

a) b)

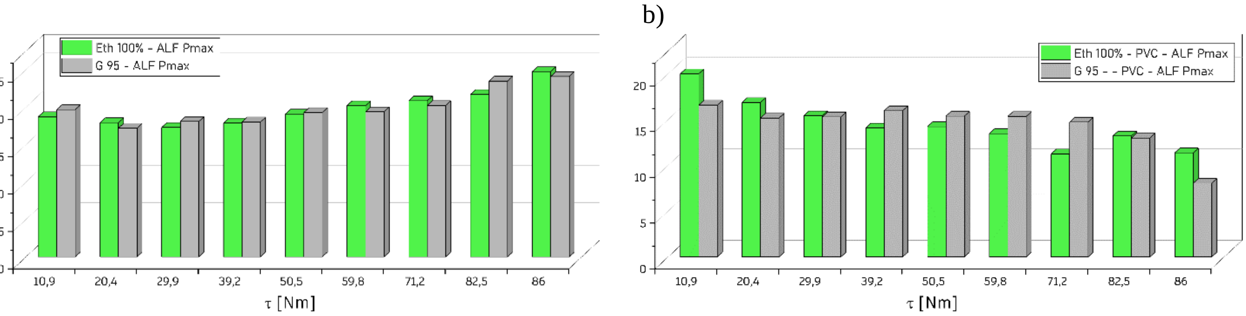

Fig. 14. Comparison of the angles of occurrence of the maximal pressures and the coefficient of angle variations $\sigma_{\alpha p_{max}}$ in course of combustion in Fiat 1100 MPI engine run on gasoline and ethyl alcohol

sucked mixture were found. A lower air excess coefficient for the gasoline-air mixture could cause a faster combustion of the mixture and smaller fluctuations of the angle.

Changes in the course of the combustion process during successive cycles of engine operation, manifested by fluctuations in maximum combustion pressure p_{max} and its angle α_{pmax} affect the changes in the crankshaft torque and the values of the mean indicated pressure pi for individual cycles of engine operation.

However, as demonstrated by the previous research of the authors, described in [23], the fluctuations of pi in subsequent cycles of engine operation are small at constant engine load, and they are similar for both fuels. The analysis of the relative coefficient of variability of the mean indicated pressure s_{pi} (calculated similarly as for the maximal pressure angle according to the formula 9) indicates that in the range of mean and maximal engine loads, the fluctuations of the p_i in individual cycles for both fuels are similar, while their values are lower than 2%, which should be considered irrelevant for the spark ignition engine, see Fig. 15.

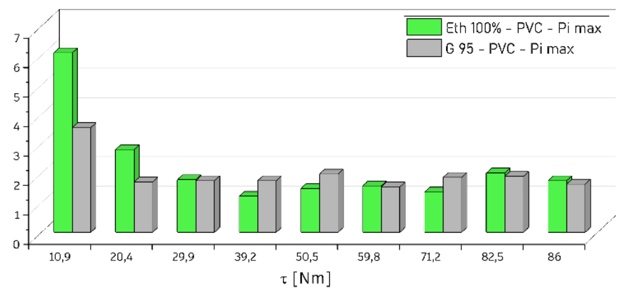

Fig. 15. Comparison of variability coefficient s_{pi} of the mean indicated pressure in Fiat 1100 MPI engine fuelled with gasoline and ethyl alcohol

4. Summary

In this paper, the results of the research performed on the engine fuelled with two kinds of primary alcohols. i.e. ethanol and methanol, as homogenous fuels, are presented. Gasoline was the reference fuel, and all the analysed parameters obtained from alcohol fuelling were compared to gasoline. Using alcohol as a homogenous fuel allows the application of water-logged ethanol obtained directly from the distillery, considerably reducing production costs.

The research performed in this paper has shown that using methyl and ethyl alcohols in a spark ignition engine improves its performance and ecological parameters with simultaneous improvement of smoothness of engine operation.

In relation to gasoline fuelling, the following parameters were found to be improved in the tested car engine of the passenger model of Fiat 1100 MPI:

- increase of maximal output power of the engine within the range of rotational speeds of 2000–3500 rpm, by 3.8–7.5% for methanol fuelling and by 1.7–3.3% for ethanol fuelling
- increase the overall efficiency of the engine by 1.5–7.5% for methanol and by 1.2–2.5% for ethanol
- reduction in the concentration of toxic components of exhaust gases: nitrogen oxides NO_x by 15–60% for methanol and by 35–75% for ethanol, total hydrocarbons content THC by 2–4 times for methanol and 4.5–8.2 times for ethanol.

The increase in efficiency was observed in the entire range of changes in engine load and its rotational speed, with higher values of the changes occurring as the engine load increased.

The increase in the performance parameters was obtained with the unchanged compression ratio and the unchanged ignition advance angle. Both these parameters were optimized in the tested engine for gasoline fuelling. It seems that a further increase in the power output and efficiency can be achieved by increasing the compression ratio, which will allow better utilization of high anti-knock resistance of alcohols.

Feeding the engine with alcohol reduces the exhaust gas temperature behind the exhaust valve in the range of 45–90°C for methanol and 30–60°C for ethanol. This should lead to an increase in the durability of the engine.

The combustion of methanol vapours proceeds faster than gasoline, which is accompanied by higher maximum pressures in the cylinder p_{max}, higher heat release rate $(dQ/d\alpha)_{max}$ and higher maximal rate of pressure growth $(dp/d\alpha)_{max}$. As a result, the kinetic combustion phase of methanol is shorter, and the maximum pressure of the process is achieved faster than for gasoline. This explains the significant increase in the maximum power output and overall efficiency of the methanol-supplied engine. The increase in the maximum pressures is not high, and it seems that it should not change the engine’s durability.

The analysed parameters, like pressure p, heat release rate $(dQ/d\alpha)_{max}$ and $(dp/d\alpha)_{max}$ during the course of combustion of ethyl alcohol, are similar to gasoline. As a result, the engine's operational noise and its durability should not change when the engine is run on ethanol as homogenous fuel.

The analysis of 50 successive cycles of individual combustion of alcohol and gasoline has shown that changes in the maximum pressure p_{max} in subsequent cycles of engine operation are stochastic and typical for the spark ignition engine. The character of changes of the p_{max} is similar for ethanol and gasoline.

The quantitative assessment of fluctuations of the maximal pressure defined by the mean value of the maximal pressure, mean standard deviation σ_{pmax} and coefficient of variability of pressures s_p indicates that in the range of engine loads higher than 20 Nm, fluctuations of the maximal pressure for ethanol are lower comparing to gasoline. It proves the improvement of smoothness of the engine powered by ethyl alcohol. Only for the lowest engine loads below 20 Nm, when the engine is fed with ethyl alcohol, an increase in the pressure fluctuations p_{max} and deterioration in smoothness of engine operation from cycle to cycle were observed. This can cause start-up difficulties and rough engine operation in warm-up phase during winter time.

This is probably due to excessive reduction in temperature of the fuel charge triggered by the greater heat of evaporation of ethanol in relation to gasoline, which affect the time of stabilization of the flame after ignition of the combustible mixture.

The research conducted for this paper fully confirms the possibility of using alcohols as homogenous fuels to feed the engines. It is especially recommended to use ethyl alcohol as a less aggressive fuel to the materials used in construction of the engines and less toxic to the environment.

Nomenclature

CI compression ignition
CNG compressed natural gas
DI direct injection
LPG liquefied petroleum gas
Ne power output
NO_x nitrogen oxides
SI spark ignition
THC total hydrocarbons
η_o overall efficiency

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