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Study of the effect of ignition crank angle and mixture composition on the performance of a spark-ignition engine fueled with ethanol

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The publication presents the results of the measurements of the operating parameters of a spark-ignition engine fueled with 95-octane unleaded gasoline (ES95) and ethyl alcohol, approx. 92%. The measurements were carried out at a constant load: an engine speed of 1500 rpm and a constant pressure in the intake system – MAP = 0.45 bar. For each type of fuel, the measurements were carried out in two series for two variables. The ignition crank angle was varied in the range of 0°÷40° and the mixture composition λ in the range of 0.85–1.25. The recorded engine performance parameters included torque, intake manifold pressure, intake air temperature, exhaust gas temperature and temporal fuel consumption; and exhaust gas composition was examined in terms of carbon monoxide, hydrocarbons and nitrogen oxides. The study showed that an ethanol-fueled engine has lower average efficiency compared to a gasoline one. The highest efficiency for ethanol was obtained for rich mixtures in the range λ = 0.85–1.0 and at high ignition advance angles. The use of alcohol fuel showed a very favorable effect on the composition of exhaust gas and a significantly lower content of harmful exhaust components was demonstrated. For the same operating points, carbon monoxide content was reduced by an average of 15%, and hydrocarbons and nitrogen oxides by an average of 80%.

Key words: *ethanol, alternative fuels, alcohol fuels, emissions, biofuels*

1. Introduction

Internal combustion engines still play an important role in the economy, providing propulsion for most vehicles and agricultural or industrial machinery. However, traditional fossil fuels used to power internal combustion engines have a negative impact on the environment due to emitted harmful substances such as carbon dioxide, nitrogen oxides, hydrocarbons and other chemical compounds during combustion [5]. It is clear that it is not possible to quickly replace internal combustion engines with electric ones [19], so work and research are being conducted so that internal combustion engines could remain in use but their consumption of fossil fuels was minimized [22]. In addition to zero-carbon fuels such as hydrogen [15, 21] or its mixtures with oxygen (HHO) [9, 11, 33], reduced-carbon fuels such as gases [10] or alcohols [12, 28] are being used. These can be the primary fuel or used as an additive to the original/factory fuel [1, 3]. Such fueled internal combustion engines can reduce the CO_2 emissions of the vehicle under study [20]. Often proposed as an alternative to fossil fuels are biofuels [8, 22] which have gained great popularity in recent years due to their environmental benefits and potential to reduce transportation dependence on traditional fuel sources [14]. Therefore, one of the most important areas of research is alternative fuels that would allow the continued use of traditional internal combustion engines [24, 32]. Alcohol fuels such as ethyl alcohol, which is produced by fermenting plant biomass, are an important type of fuel produced from plant components [31]. Many publications refer to the use of alcohol fuels in internal combustion engines and their impact on the operating parameters of such engines such as vibration and the way they are measured [6, 27], noise intensity [13], energy efficiency [2] or reduction in the number of harmful exhaust components [15, 23]. Ongoing research has shown that the use of alcohol additives and alcohol as a stand-alone fuel both in spark-ignition engines [16, 25] and diesel engines [17, 30] can significantly affect the engine performance. In the case of spark-ignition engines, ethyl and methyl alcohol fuels are among the most widely studied alternative fuels due to their relatively low viscosity and high vapor pressure. Many studies focus on comparing the combustion behavior of these fuels with the properties of gasoline which is currently the most widely used fuel in this type of engine. Their effect on the performance of such engines is also investigated. Studies have also been conducted on the physical and chemical properties of alcohol fuels and their blends [14], their effect on atmospheric emissions [7, 18] and their use to increase combustion efficiency and reduce harmful gas emissions [4].

2. Test stand and test methodology

To examine the effect of fuel type on the performance of an internal combustion engine, this paper compares two types of fuel: 95-octane gasoline: ES95 and ethyl alcohol of a concentration of about 92%. Two series of measurements were carried out in which the ignition crank angle and fuel mixture composition were changed, and engine torque, fuel consumption and exhaust gas composition were recorded in each series. The purpose of the study was to compare the effects of using ethyl alcohol on selected performance parameters of a spark-ignition engine.

2.1. Engine

The study used the Lublin University of Technology's dynamometer stand with a Holden C20LE engine (Fig. 1). It is a four-stroke gasoline engine to propels cars and vans. The engine is powered by electronically controlled multipoint indirect fuel injection. The engine is equipped with a DIS ignition system and an EGR exhaust gas recirculation system. A microprocessor-based ADAM 5510 system was responsible for regulating and maintaining the engine's thermal condition. The engine's technical data is shown in Table 1.

Fig. 1. Holden C20LE engine

Table 1. Technical data of Holden C20LE engine

Type of engine	C20LE
Layout and number of cylinders	4, in-line
Cylinder diameter	86 mm
Piston stroke	86 mm
Engine displacement	1 998 cm^3
Compression ratio	8.8:1
Max power	77 kW/5200 rpm
Max torque	164 Nm/2600 rpm
Number of valves	8

2.2. Measurement equipment

The proper construction of a test bench should be preceded by a CAD design and an analysis of the robustness of the structure with a view to its safe use in scientific research [26]. The test stand was equipped with the measurement systems necessary to measure the exhaust gas composition and energy efficiency of the engine and to read basic parameters of its operation. The measuring and regulating of the engine load was done with a brake, the SAK-670 N engine brake from VEB Elbtalwerk, connected to the engine via a 5-speed gearbox and a shaft with a flexible coupling. It is an electric brake whose braking torque is controlled by direct current. The accuracy of torque measurement by the brake measuring system was 0.5 Nm. The gearbox was locked in direct gear.

Energy demand was measured using a fuel scale. Engine speed (RPM), intake manifold pressure (MAP), composition and exhaust gas temperature (CHT) were also recorded. An AMX 200 CAN programmable logic controller (PLC) was used as the engine control system. The controller allowed arbitrary changes in the ignition crank angle and injector opening time. A proprietary strain gauge fuel scale was used to measure fuel consumption. The measurement time for fuel consumption at one operating point was 60 s.

Fig. 2. Test stand with electric brake

An HGA 400 exhaust gas analyzer was used to measure the composition of the exhaust gas during the tests. The measurement range of the various components of the exhaust gas is shown in Table 2.

Table 2. Technical data of the HGA 400 exhaust gas analyzer

Component	Measuring range	Measurement accuracy
CO	0–10%	±0.06%
CO_2	0–20%	±0.5%
HC	0–20,000 ppm	±11 ppm
O_2	0–22%	±5%
NO_x	0–5000 ppm	±10%

2.3. Research methodology

The study was aimed at making comparative measurements of the performance of a spark ignition engine fueled with the original fuel – ES95 and 92% ethanol. During the measurements, the same measuring apparatus and engine supply system were used. Table 3 shows the physicochemical data of the fuels used during the study.

Table 3. Physical and chemical parameters of the fuels used [29]

Parameter	ES95	Ethanol 92%
Calorific value	42.0 MJ/kg	26.2 MJ/kg
Density	750 kg/m^3	806 kg/m^3
RON	95	110
Enthalpy of vaporization	315–350 kJ/kg	879 kJ/kg
Specific heat	2.0 kJ/kg·K	2.2 kJ/kg·K
Freezing point	–40°C	–114°C
Viscosity (20°C)	0.37–0.44 mPa·s	1.19 mPa·s

It was decided to obtain two control characteristics of the engine: the characteristics of changing the ignition advance angle and the characteristics of the mixture composition. All measurements were carried out at an engine speed of 1500 rpm, controlled with the engine brake by adjusting the load. For each measurement point, the throttle opening angle was also selected to obtain an intake manifold pressure MAP = 0.45 bar. Such an operating point was chosen because an engine operating at low speed and low load is more sensitive to changes in feed conditions, changes in mixture composition, unevenness of operation, and deterioration of the uniformity of the air-fuel mixture in the cylinder. The use of such engine operating conditions was intended to highlight the change in the performance of the internal combustion engine caused by the change in fuel. Previous series of tests have shown that under these conditions the engine will operate at the limit of stability, so all changes in parameters caused by a change in fuel, will be easily noticeable.

The first measurements were made for a variable ignition advance angle. The engine was brought to a stable operating point. The ignition crank angle was varied from 0^o to 40^o, in 10^o increments. Once the ignition crank angle was set, the stoichiometric mixture composition (λ = 1) was established by changing the opening time of the injectors. Once the engine was stabilized, the measurement started. The recording time of a single measurement point lasted 60 seconds. During this time, the fuel consumption, the exhaust gas composition and the torque were measured. When the measurement was completed, the next value of the ignition crank angle was set, the mixture composition was determined and the measurement was repeated.

The mixture composition was plotted at a fixed value of the ignition crank angle of 20^o. Once a stable operating point was established, the injection time was changed to achieve the required mixture composition in the range of λ = 0.85–1.25, with a step of 0.1. The common measuring point for both characteristics was the operating point of λ = = 1, ICA = 20°. Figure 3 shows the distribution of the tested measurement points.

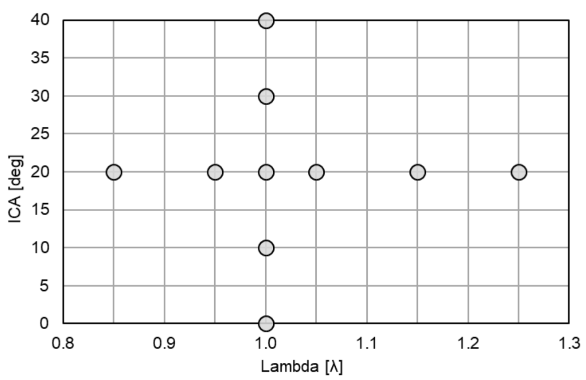

Fig. 3. Measurement points, MAP = 0.45 bar, n = 1500 rpm

3. Research results and analysis

3.1. Measurement results

From the torque measurement results, it can be concluded that when fed with lean mixtures and at a larger ignition crank angle (λ > 1; ICA > 20°), the gasoline-fueled engine had higher torque on average. In contrast, when fed with rich mixtures and at a smaller ignition crank angle (λ < 1; ICA < 20°), higher torque was generated when the engine was fed with ethanol.

For the measurement point with stoichiometric mixture and ICA = 20°, the torque values obtained were the same. The measured torque values are shown in Fig. 4.

For every measurement point except (ICA = 0°; λ = 1), the exhaust gas from ethanol combustion had a lower temperature than gasoline, by an average of 19.4°C. The highest exhaust gas temperature recorded in the tests was 655°C, while the lowest was 535°C. The measurement points are shown in Fig. 5. There is a clear tendency for the exhaust gas temperature to increase with decreasing ICA and with increasing excess air ratio λ.

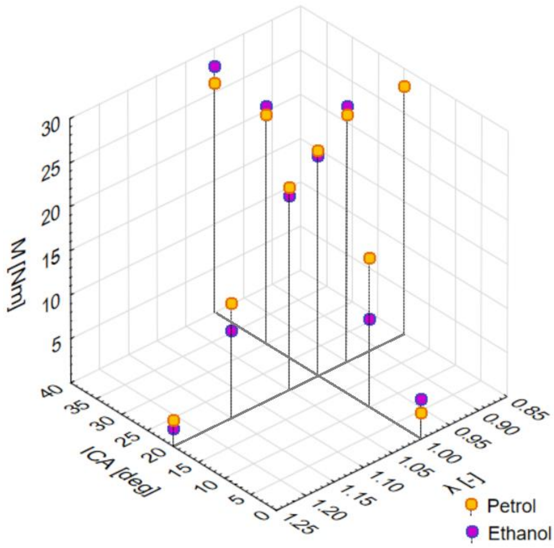

Fig. 4. Torque measurement results for gasoline and ethanol

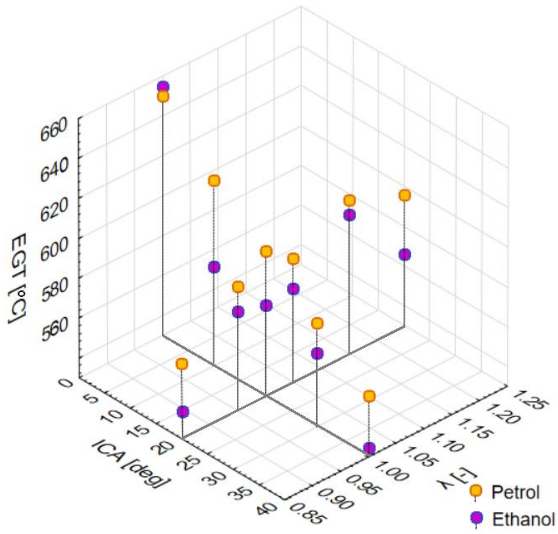

Fig. 5. Results of EGT measurements for gasoline and ethanol

The graph in Fig. 6 shows the carbon monoxide content of the exhaust gas for all measurement points. In the range of stoichiometric mixtures and lean mixtures, the carbon monoxide content of the exhaust gas oscillated between 0.12–0.53%, showing no significant differences between ethanol and gasoline. The maximum CO content was obtained for λ = 0.85 and reached 4.94% for gasoline and 3.94% for ethanol.

Figure 7 shows the hydrocarbon content of the exhaust gas during the testing of the engine fueled by ethanol and gasoline. The hydrocarbon content for ethanol was 22–78 ppm, with its average value of 41 ppm for all points, while for gasoline the range was 143–308 ppm with an average content of 209 ppm. There is a noticeable increase in the hydrocarbon content of the exhaust gas at λ = 1.25. Hydrocarbon emissions increased by 160% for ethanol (up to 78 ppm) and 76% for gasoline (up to 288 ppm). Taking into account the fact that the exhaust gas temperature decreased for this measurement point and the torque reached the lowest values, it can be concluded that this is the flammability limit of the mixture for the tested engine.

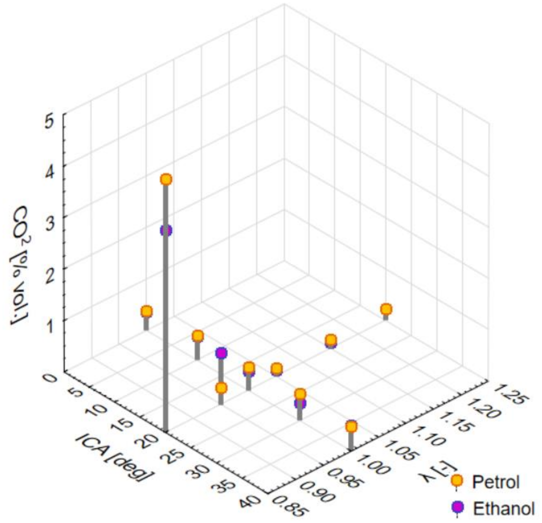

Fig. 6. Volumetric content of carbon monoxide in exhaust gases

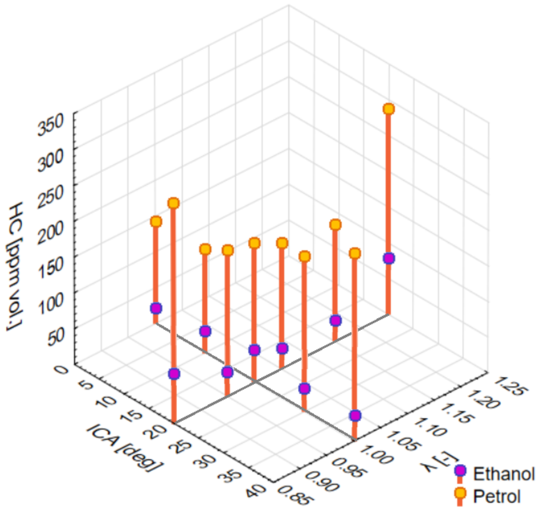

Fig. 7. Hydrocarbon content in exhaust gases

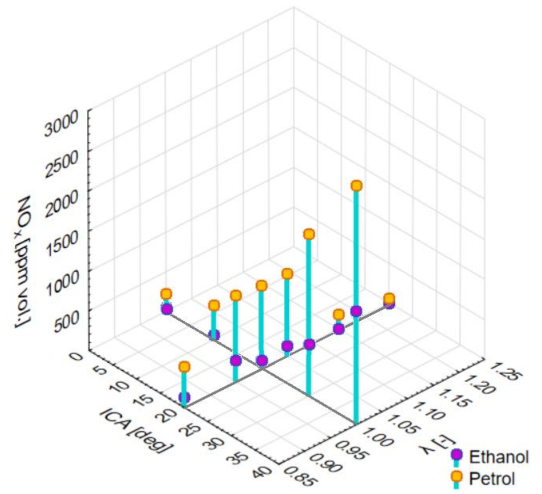

Fig. 8. Nitrogen oxides content in exhaust gases

Figure 8 shows the content of nitrogen oxides in the exhaust gas. The content of nitrogen oxides when the engine was fed with ethanol was 3–1407 ppm, with an average value of 268 ppm. For gasoline, the volumetric NO_x content was in the range of 65–2986 ppm, while the average content was 970 ppm. There was a clear increase in the exhaust gas NO_x content as the ignition advance angle increased.

The measured values of temporal fuel consumption as a function of the excess air ratio λ are shown in Fig. 9. The large difference in temporal fuel consumption for the same engine operating parameters is due to, among other things, the difference in the calorific value of the two fuels used.

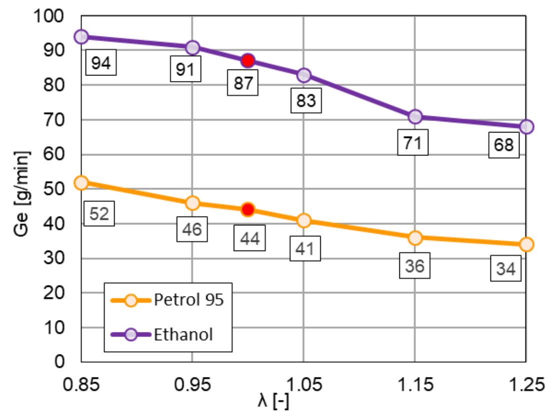

Fig. 9. Fuel consumption rate during operation with gasoline and ethanol

3.2. Results analysis

Based on the obtained data on temporal fuel consumption (Ge) and torque, the specific fuel consumption was calculated from the formula:

$$g_e = \frac{Ge}{\frac{M \cdot n}{9550}} \tag{1}$$

where: g_e – fuel consumption [g/kWh], Ge – temporal fuel consumption [g/min], M – torque [Nm], n – rotational speed – 1500 rpm.

Figure 10 and 11 shows the course of specific fuel consumption as a function of ignition crank angle and excess air ratio λ. It is noticeable that alcohol consumption is higher compared to gasoline at each recorded measurement point, which is due to, among other things, its significantly lower heating value.

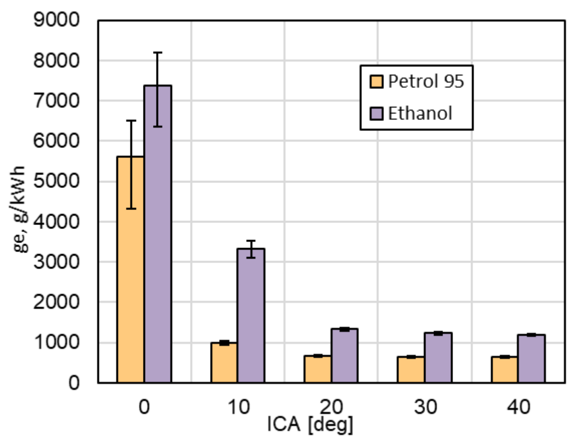

Fig. 10. The specific fuel consumption for gasoline 95 and ethanol at λ = 1.0

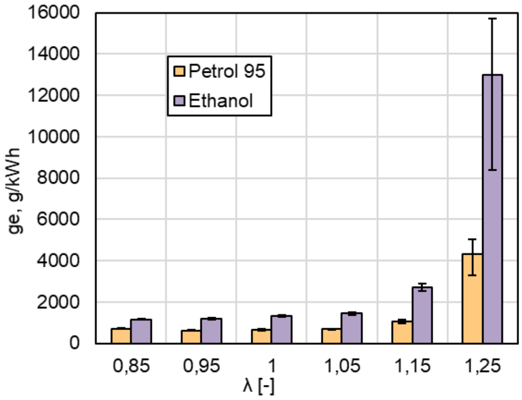

Fig. 11. Specific fuel consumption for ethanol and gasoline 95 at ICA = 20°

By determining the calorific value of gasoline in the range of 40.1–43 MJ/kg and that of ethanol – 24.03–28.31 MJ/kg, the total efficiency of the engine was determined. The highest efficiency for both fuels was obtained for ICA ranging 20–40° BTDP and for a mixture composition in the range λ = 0.85–1.05. When feeding the engine with ethanol, higher efficiency is recorded for richer mixtures. The course of engine efficiency for ethanol and gasoline fueling is shown in Fig. 12 and 13. The overall efficiency at the most favorable measurement points oscillates within 10–14%, which is a relatively small value, but this is due to the low engine load (MAP = 0.45 bar). Such an unfavorable operating condition in terms of efficiency, however, makes the differences between the two fuels become more apparent.

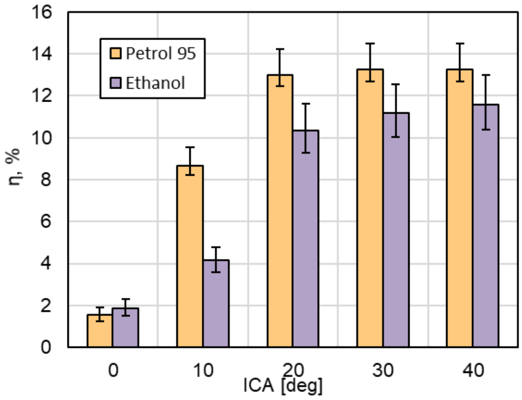

Fig. 12. Engine efficiency during fueled with gasoline 95 and ethanol at λ = 1.0

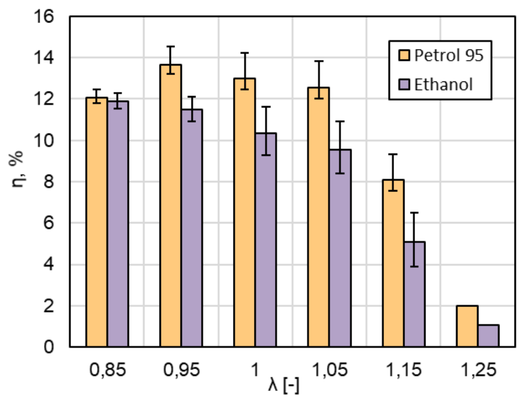

Fig. 13. Engine efficiency when fueled with gasoline 95 and ethanol at ICA = 20°

Using the neural network method prediction module in Statistica, a complete map of internal combustion engine efficiency was determined by using the neural network-based prediction method. Input data for the neural network were the efficiency values at the measurement points shown in Fig. 14 and 15, whereas output data for the algorithm were the specified missing points, identically distributed as in the experiments, for the area bounded by the parameters λ ∈ (0.85–1.25) and ICA ∈ (0–40°).

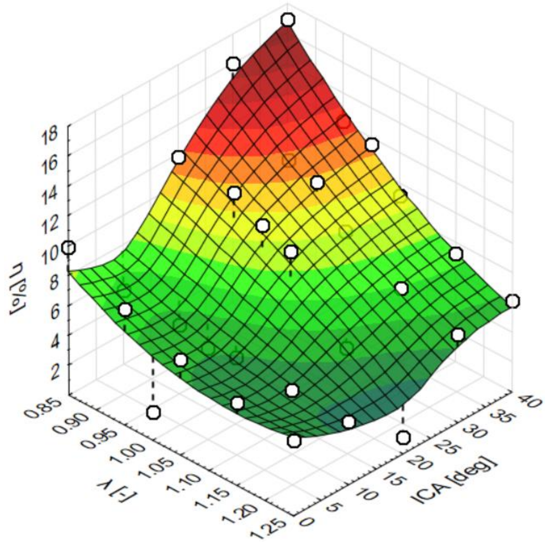

Fig. 14. Map of engine efficiency during operation fueled with ethanol, generated by predicting operating points within the range of λ ∈ (0.85–1.25) and ICA ∈ (0–40°)

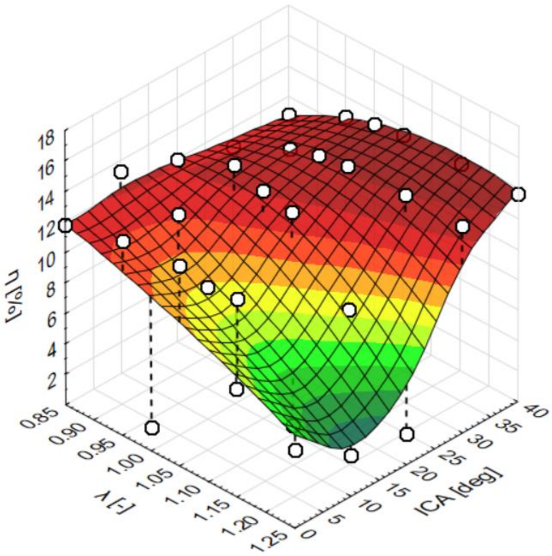

Fig. 15. Map of engine efficiency during operation fueled with gasoline 95, generated by predicting operating points within the range of λ ∈ (0.85–1.25) and ICA ∈ (0–40°)

The values determined by prediction indicate that the gasoline-fueled engine maintains its efficiency above 10% over a much wider range than the ethanol-fueled engine. The most unfavorable operating range of the engine in terms of efficiency is the operation when fed with lean mixtures (λ > 1) and at low values of ICA < 30°.

Figures 16 and 17 shows the degree of reduction in the content of harmful exhaust components after replacing gasoline with ethanol.

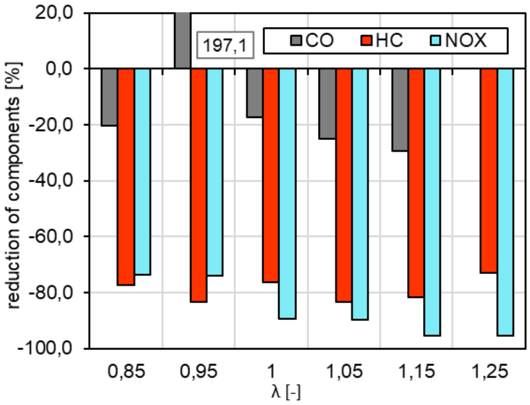

Fig. 16. Reduction of toxic emissions content after replacing gasoline with ethanol at ICA = 20°

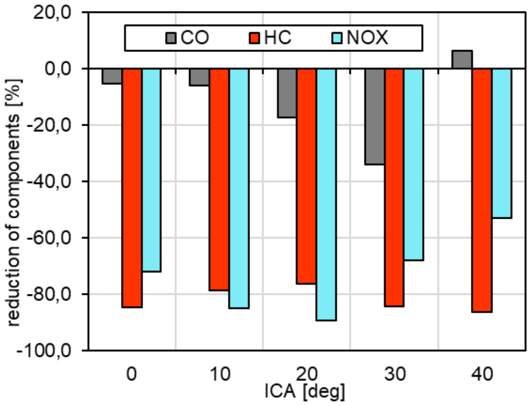

Fig. 17. Reduction of toxic emissions content after replacing gasoline with ethanol at λ = 1.0

The use of ethanol is very beneficial in terms of emission reduction. In most cases, the hydrocarbon content was reduced by approx. 75–80%. When feeding the engine with a lean mixture and at ICA = 20°, the content of nitrogen oxides was reduced by more than 90%. With increasing ICA, the degree of NO_x reduction decreases, but in the least favorable case it is: –53%. The only component of the exhaust gas for which an increase in emissions was registered at individual measurement points is CO. At the measurement point of ICA = 20° and λ = 0.95, the carbon monoxide emissions almost doubled, but with an overall content of 1.01% vol. this can be considered a measurement error. At the measurement point of ICA = 40° and λ = 1.00, ethanol showed a 6.3% increase in CO emissions relative to that of gasoline. The percentage of carbon monoxide at this point for gasoline is 0.48% vol., and for ethanol is 0.51% vol.

Knowing the parameters of the exhaust gas analyzer, it can be claimed that this difference is within the margin of a measurement error. This makes it clear that for all analyzed engine operating conditions, the use of ethanol as an alternative to gasoline allows a significant reduction in emissions.

4. Conclusions

The study showed that under the operating conditions analyzed, the SI internal combustion engine can be fueled with ethanol via the original injection system without any design changes. However, it is necessary to select the correct injection timing for the alcohol fuel. In addition, by changing the fuel from gasoline to ethanol, emissions of harmful exhaust components were reduced at all operating points. The content of carbon monoxide (CO) in the exhaust gas decreased by an average of 14.8% and the content of hydrocarbons (HC) and nitrogen oxides (NO_x) by an average of 80%. Because the heating value of gasoline is about 61% higher than that of gasoline, fuel consumption increased significantly. Temporal fuel consumption increased by 90% on average, while specific fuel consumption increased by 115%. Therefore, ethanol-fueled vehicles could have a much shorter range with the same tank capacity. At the tested operating point, the ethanol-fueled engine showed an efficiency decrease by 1.9 percentage points on average, reaching the highest values for large ICA values (20–40°) and rich mixtures λ = (0.85–0.95). The limit of operation of the tested engine, both with gasoline and alcohol fueling, is the mixture composition λ = 1.25 at which torque reached values close to zero. The study showed that the use of ethanol to power a spark-ignition internal combustion engine can provide significant emissions benefits without significantly degrading engine performance. The only modifications required to start and operate an ethanol-fueled engine are to increase the volumetric fuel flow, which can be achieved by installing higher capacity injectors or making changes to the ECU. This does not mean, however, that making such modifications will be sufficient for long-term operation of the engine on alcohol fuel. Design changes would also be needed to account for the higher temperature amplitude of the engine, the higher water content of the exhaust, or problems associated with alcohol dilution and evaporation.

Nomenclature

CO	carbon monoxide content of the exhaust gas
CO_2	carbon dioxide content of the exhaust gas
EGT	exhaust gas temperature
g_e	specific fuel consumption
Ge	timing fuel consumption
HC	hydrocarbons
ICA	ignition crank angle
M	torque
MAP	manifold absolute pressure
n	rotational speed
NO_x	content of nitrogen oxides in the exhaust gas
RON	research octane number
η	total efficiency of the internal combustion engine
λ	excess air factor in the flue gas

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