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## Operation of a Spark-Ignition Engine on Mixtures of Petrol and N-Butanol

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### Abstract

The Article analyses the use of n-butanol (biobutanol) in mixtures with petrol in spark-ignition (SI) internal combustion engines (ICE). Analysis of the scientific literature allowed determining that n-butanol content had a different effect on the same engine parameters in different tests, thus a decision was made to conduct additional research, where, having assembled the necessary equipment and conducted experimental planning, experimental studies of n-butanol (with its volumetric content being 0%, 20% and 40%) and petrol blends in a SI engine were carried out. Regression analysis of research results was conducted and regression dependences were formed allowing evaluating the impact of the key variables ( $\delta$  and  $\theta$ ) on engine parameters ( $M_e$ ,  $P_e$  and  $b_e$ ). The research results have shown that without changing the ignition advance angle  $\theta$ , petrol blend with 10% n-butanol has essentially no adverse effect on engine power  $P_e$ , engine torque  $M_e$  and specific fuel consumption  $b_e$ . Increasing n-butanol concentration  $\delta$ , the maximum engine torque  $M_e$  and effective power  $P_e$  decrease, while the specific fuel consumption  $b_e$  increases. Energy engine indicators decline as a result of a lower calorific capacity of n-butanol compared to that of petrol and lower combustion rate of n-butanol. Increasing n-butanol concentration  $\delta$  in fuel mixture, engine parameters can be improved by advancing the ignition angle  $\theta$ .

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**Keywords:** n-butanol, spark-ignition, fuel mixtures, power, torque, fuel consumption

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## 1. Introduction

### Nomenclature

SI	spark-ignition
ICE	internal combustion engines
$P_e$	engine power
$M_e$	engine torque
$b_e$	specific fuel consumption
$n$	crankshaft rotational frequency
$\delta$	n-butanol concentration in fuel mixture
$\theta$	ignition advance angle
RON	research octane number
MON	motor octane number
TDC	top dead center
$\lambda$	excess air factor

At present, the most topical transport problem worldwide is environmental pollution by exhaust gas and reckless waste of non-renewable energy resources, so abundant scientific research works on investigation of fuels of the new generation, modification of internal combustion engines of usual types or creation of new ones upon striving to reduce the environmental pollution caused by them, to use renewable energy resources and to win the economic benefit are carried out.

Recently, a particular attention is paid to reduction of greenhouse gas emissions. Statutory regulation of them began as early as in 1992, after signing the United Nations Framework Convention on Climate; in addition, Kyoto Protocol on reduction of greenhouse gas emissions was signed in 1997. The major sources of atmosphere pollution include transport and energy sectors that use fossil fuels, so EU encourages to replace resources of mineral fuels by renewable energy resources and extend use of the latter in various sectors of economy [1].

Road transport and the exhaust gas generated by it affect considerably the air quality and the human health. In EU Member States, regulations for road transport exhaust pollutants are being applied. The said regulations provide a serious challenge to manufacturers of vehicles and fuels. In the document of the European Commission, no specific technologies that would enable satisfying the requirements of “Euro 6” are pointed out; however, it is emphasized that this task may be accomplished by improving engines and their exhaust gas systems as well as by searching for opportunities of using alternative fuels.

Use of alternative fuels in internal combustion engines applied for operation on oil-based fuels causes a number of various problems. If only an alternative fuel is used, an expensive and complicated alteration of the structure of the engine and its control systems is required. Upon striving to pare down the said expenses and to simplify the vehicle use, it is confined only to a partial replacement of usual fuel with an alternative fuel; in the case under discussion, the alternative fuel is biobutanol.

Butanol (also called butyl alcohol) is a four-carbon alcohol with a formula of  $C_4H_9OH$ , which occurs in five structures, from a straight-chain primary alcohol to a branched-chain tertiary alcohol; all are a butyl or isobutyl group linked to a hydroxyl group (sometimes represented as BuOH, n-BuOH, and i-BuOH). These are n-butanol, 2 stereoisomers of 2-butanol, tert-butanol, and isobutanol. Butanol is primarily used as a solvent, as an intermediate in chemical synthesis, and as a fuel. It is sometimes also called biobutanol when produced biologically.

It is notable that until quite recently, butanol was not known to be an alternative fuel, because its production never was considered economically viable.

However, new high-efficiency butanol production technologies appear at present, so it, as a fuel for road transport, attracts an increasingly growing attention of professionals and it is not improbable that ethanol will loss a priority in future. At present, butanol is considered one of the advanced types of motor fuels.

Today, organization of butanol production is one of the most perspective directions of development of biofuel based energy sector worldwide.

Physical and chemical properties as well as performance of butanol ensure its crucial advantage over already used spark-ignition engines operating on methanol and ethanol. In addition, performance of butanol is comparable to that of petrol and even surpasses it in certain aspects.

At present, the above-described causes an increasingly growing number of scientific research works on investigation of butanol as a motor fuel for engines [1–7, 9–11].

Let's examine how the physical and chemical properties as well as performance of butanol (Table 1) affect an opportunity of its use in engines, as compared to methanol and ethanol as well as the peculiarities of the engine operation process [8–12].

Table 1. Physical and chemical properties as well as performance of fuels.

Properties	Fuel				
	Methanol	Ethanol	Butanol	Petrol	Diesel fuel
Density, 20°C, kg/m <sup>3</sup>	791	810	810	730–784	820–840
Lower heating value, MJ/kg	19.6	26.9	33.1	43.5–44	42.5–43
Octane number:					
RON	136	130	96	91–99	–
MON	104	96	78	81–89	–
Cetane number	3	8	18	–	45–55
Volume of air required for complete combustion of 1 kg of fuel, kg	6.5	9.0	11.2	14.6	14.5
Heating value of stoichiometric mixture, MJ/kg	3.1	3.0	3.2	2.9	2.9
Heat of evaporation, kJ/kg	1200	920	570	300	230
Kinematic viscosity at 20°C, cSt	0.64	1.52	3.64	0.4–0.8	>3
Self-ignition temperature, °C	464	423	390	415	220

If butanol is used in a petrol engine, it would result an increase of fuel consumption by about 10%; however, testing of other alcohol fuels mixed with petrol shows the engine's economic performance to be not proportional to the changes of the lower heat of combustion.

Octane number of n-butanol almost equals to octane number of petrol; however, it is lower, as compared to ethanol and methanol. Use of n-butanol enables increasing the degree of compression that, in its turn, causes an improvement of fuel consumption economy and an increase of power and torque of the engine.

An alcohol fuel, including butanol and ethanol, is partially oxidized, so it enables working with fuel-enriched mixtures, as compared to petrol. In existing engines, butanol may be mixed with petrol in higher ratios, as compared to ethanol, without a necessity of alteration of the engine, because the values of the volume of air required for complete combustion of butanol and the lower heat of combustion are closer to the same of petrol.

An alcohol fuel contains less energy per unit mass or volume, as compared to petrol; however, it requires a fuel-enriched mixture. For making possible to compare amounts of net energy produced during a cycle, the so called heat of combustion of stoichiometric mixture is sometimes used. It is defined as a ratio of the generated heat to the volume of air required for complete combustion of it. In case of butanol, the net energy produced during a cycle is higher, as compared to ethanol or methanol; in addition, it is higher by about 10%, as compared to petrol.

In the engine, the fuel should be evaporated prior to beginning of the combustion process. Because the heat of evaporation for butanol is lower, as compared to ethanol or methanol, an engine operating on butanol should be easier started at a cold weather, as compared to those operating on ethanol or methanol.

Kinematic viscosity of butanol is several times higher, as compared to petrol and almost is the same, as of high quality diesel fuel. Use of fuels with viscosity that differs considerably from the one of traditional fuels requires a reconstruction of the fuel system of standard engines and a recalculation of their injection characteristics.

The aim of this study is to investigate the effect of n-butanol on the effective parameters of spark-ignition engine.

## 2. Literature review

Tornatore et al. [4] had investigated the combustion process in a spark-ignition internal combustion engine operating on a mixture of butanol (40%) and regular petrol (60%). The tests were carried out upon varying the ignition advance angle and phases of fuel injection. The intensity of combustion flame and the changes of combustion pressure were investigated for different fuels. Tornatore [4] found that in case of using an indirect injection system, zones of more enriched mixture appeared on the surface of the piston. So, both on using regular petrol and its mixture with butanol, the injected volume of fuel is higher, as compared to fuel injection in case of closed valves. The tests on combustion pressure in a cylinder and the rate of heat emission were carried out at ignition advance angle equal to  $14^\circ$  before the top dead center (TDC). Both in case of fuel injection at opened and fuel injection at closed intake valves, the maximum combustion pressure of butanol mixture was slightly higher, because the rate of combustion of butanol is higher, as compared to regular petrol.

It was found that using butanol in the combustible mixture was accompanied by a reduction of the content of pollutants in it. It was caused by oxygen presented in the structure of butanol: oxygen improved the combustion process by reducing the content of solid particles in the exhaust gas. In addition, formation of hyperthermal compounds ( $\text{NO}_x$ ) was reduced as well, because heat of evaporation of butanol is higher and its calorific capacity is lower, as compared to petrol. If fuel is injected when the intake valves are opened, fuel consumption is reduced; however, CH and  $\text{NO}_x$  emissions increase, because the liquid fuel should be evaporated prior to blending with air. The data on the test show that in both cases of fuel injections, butanol reduced emissions of hydrocarbons, as compared to regular petrol. It was found that 40% share of butanol improved evaporation and combustion of the combustible mixture.

Szwaja et al. [7] carried out an investigation of a spark-ignition engine striving to examine the opportunities of using butanol in an internal combustion engine. For the tests, the following fuels were used: regular petrol; a mixture of petrol (80%) and butanol (20%); a mixture of petrol (40%) and butanol (60%) and pure butanol. The mixtures were tested by one-cylinder internal combustion engine cooperative fuel research engine (designed for operation on petrol) with controllable degree of compression. The engine was modified in a way that ensured a possibility of controlling the volume of air supplied to the intake manifold and the volume of the injected fuel. For monitoring the engine and controlling its operation, combustion monitoring equipment that enables observing the process in the combustion chamber at the present moment was installed. The tests were carried out when the engine operated on stoichiometric mixture, at fixed effective torque and fixed rotational speed ( $900 \text{ min}^{-1}$ ). The combustion degree was 8:1 and 10:1, the ignition advance angle varied from  $18^\circ$  to  $4^\circ$  before TDC.

It was found during the tests that when the engine operates on pure butanol upon the load of 330 kPa, the most effective ignition advance angle for such an engine is about  $10^\circ$  before TDC; at a higher value of the ignition advance (because the combustion rate of butanol is higher, as compared to regular petrol), operation of the engine becomes non-effective and signs of detonation appear. In addition, because of a higher combustion rate of butanol, as compared to regular petrol, the maximum pressure in the cylinder on combustion of butanol becomes in advance by about  $2^\circ$  of the crankshaft rotation angle, as compared to operation on regular petrol.

The investigation carried out by Elfasakhany [6] showed that if the content of biobutanol in its mixture with petrol is increased from 3% to 10%, at a middle rotational speed of the engine ( $3000 \text{ min}^{-1}$ ) and low throttle opening positions, the ignition delay time of the examined fuel became longer. The maximum combustion pressure may be achieved only at the optimal ignition advance moment that depends on the biobutanol concentration in petrol. In addition, it was found that even low biobutanol concentrations in petrol cause quite different maximum combustion pressures on combustion of the mixture, so for achieving the maximum combustion pressure in cylinder at the optimal moment, the ignition advance is required.

Galloni et al. [5] had compared the effects of three butanol-petrol mixtures with different concentrations (0, 20 and 40% of butanol) and found that on increasing the biobutanol concentration in petrol from 0% to 40%, the maximum power of the engine and the fuel mixture ignition rate decreased. During the investigation, the excess air factor ( $\lambda$ ) was controlled upon observing the oxygen concentration in the exhaust gas. The throttle opening position was 15%, the rotational speed of the engine was  $3000 \text{ min}^{-1}$  and the load of the engine was 30 Nm. Upon using lean combustible mixtures and increasing biobutanol concentration in petrol, Galloni found from the obtained results that

the examined fuel ignites later, as compared to regular petrol, and the engine develops lower engine power. The biobutanol additive causes a certain delay of ignition of the examined fuel, so an optimization of the ignition advance angle and the  $\lambda$  is required.

The above-described investigations show some contradicting results, so there was a reason for additional research that was accomplished in this work.

### 3. Material and methods

On the tests, a special stand has been designed and produced. The experimental tests were carried out on spark-ignition engine 4У 82/71 (BA3–2110). The engine was mounted on the stand equipped in accordance with ГOCT 14846-81 “Engines for vehicles. Methods of stand tests” [15].

The technical data for engine 4У 82/71 are provided in the Table 2. The scheme of the stand is provided in Fig. 1.

Table 2. The technical data for engine BA3–2110.

Parameter	Value
Bore, mm	82
Stroke, mm	71
Engine displacement, l	1.5
Number of cylinders	4
Compression ratio	9.8
Number of valves per cylinder	2
Nominal power according to ГOCT 14846–89 (net), kW (HP), no less than	54 (73.4)
Crankshaft rotation frequency at the nominal power, $\text{min}^{-1}$	5600
Maximum torque according to ГOCT 14846–81 (net), N·m, no less than	103.9
Crankshaft rotation frequency at the maximum torque, $\text{min}^{-1}$	3400–3600
Minimum crankshaft rotation frequency under no-load conditions, $\text{min}^{-1}$	750–800
Feeding system	Distributed fuel injection

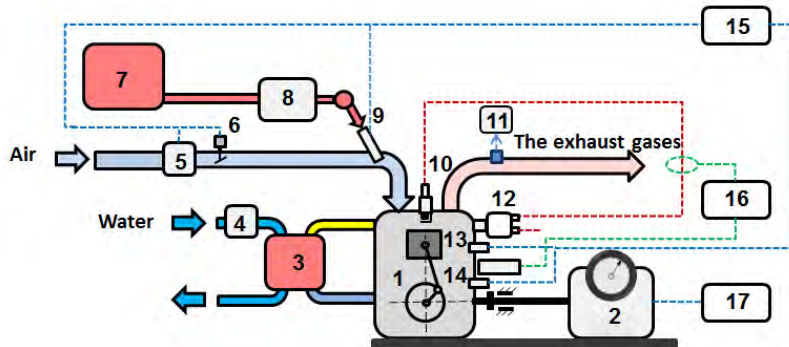


Fig. 1. Laboratory equipment scheme: 1 – SI engine BA3 –2110; 2 – engine load stand DS 736-4/V; 3 – engine cooling heat exchanger; 4 – electromagnetic valve; 5 – air mass meter; 6 – throttle; 7 – fuel tank; 8 – fuel consumption measurement equipment; 9 – fuel injector; 10 – spark plug; 11 – exhaust gas temperature sensor; 12 – ignition distributor; 13 – engine coolant temperature sensor; 14 – engine crankshaft sensor; 15 – engine control unit; 16 – motortester M3-2; 17 – engine load stand control unit. Source: authors.

The power of the engine is consumed by the braking system. The stand is equipped with braking system DS 736-4/V. The maximum power in the generation mode is 86 kW and the maximum speed is  $6400 \text{ min}^{-1}$ . The braking system is equipped with a sensor and a register for measuring the rotor’s rotational speed that is the rotational speed

of the crankshaft of the engine as well. The engine torque may be established not only by weighing equipment, but also in digital form on a display using special equipment.

In the described tests, the mass measuring method is applied for determination of fuel consumption. For this purpose, weighing machine BMI-2 with inductive sensors and a powered stopwatch that registers the time of consumption of the preset fuel dose by the engine is used.

During the tests on the engine, the temperatures of the air in the box, the exhaust gas, the liquid coolant and the oil were measured. The cooling system of the engine was of a closed type, with forced circulation of the coolant by water-circulating pump. The temperature of the liquid coolant was measured at the output of the cylinder head. The temperature of oil was measured in the crankcase of the engine. Temperatures of the liquid coolant and the oil were measured by a resistance thermometer. Operation of such a thermometer is based on an ability of some current conductors to change their resistance on heating. The temperature of exhaust gas was measured in the distance of no more than 0.1 m from the output flange of the exhaust manifold of the engine. For measuring the temperature of exhaust gas, a thermocouple thermometer (pyrometer) was used. In this case, chromel and alumel were used as current conductors, i.e. a signal from a chromel-alumel thermocouple was sent to an automated potentiometer of KPI type that registered the temperature of exhaust gas.

The devices used for the engine tests passed metrological verification. Their list is provided in the Table 3.

Table 3. The devices used for the engine tests.

Description	Units of measurement	Range of expected values	Permissible error	Description, type	Measurement range	Normed metrological characteristics, accuracy class	Conditions of measurement
Torque of the engine	N·m	0...110	±1%	Force-measuring equipment for a balancing machine	0...120	±1%	20 ± 10°C 93...105 kPa 30...80%
Ambient temperature	°C	10...25	±1	Psychrometric hygrometer ВИТ-1	0...25	±0.2	0...25°C
Ambient humidity	%	30...80	±1	Psychrometric hygrometer ВИТ-1	20...90	±0.2	5...25°C
Ambient pressure	kPa	90...106	±0.2	Aneroid barometer БАММ-1	80...106	±0.2	20 ± 5°C to 80%
Time of fuel consumption	s	–	±0.1	Electronic stopwatch “Integral C-01”	0...3600	±(9.6×10 <sup>-6</sup> ×t+0.01) ±1 s/day	–10...+50°C
Ignition advance angle	Degree of crankshaft turning angle	0...60	±1	Motor tester M3-2	–15...60	±1	5...40°C 84...107 kPa
Liquid coolant temperature	°C	85...95	±2	Regulator tester MT2	–50...200	±0.5%	5...50°C 84...107 kPa to 80%
Fuel consumption	g	–	±1%	Weigh scales, ET-6000-H	0...6000	According to ГOCT 24104 High (II)	20 ± 5°C to 80%

Motor tester M3-2 was used for measuring the ignition advance angle. The motor tester is a microprocessor-based device with a set of sensors and connectors usable for measuring parameters and examining the technical condition of 2–8-cylinder four-stroke internal combustion engines.

The engine was tested while operating on petrol and on mixtures of petrol with n-butanol where the volumetric content of n-butanol was 0, 20, and 40%. In the first phase, no settings of the engine were changed.

The extent and the procedure of tests of engines are provided in ГOCT 14846-81. “Engines for vehicles. Methods of stand tests” [13, 15]. During the tests, the following indicators were measured: the torque of the engine; the

crankshaft rotational frequency; fuel consumption; temperature of the liquid coolant; temperature of the exhaust gas; oil pressure; air temperature, humidity and barometric pressure.

In the first phase of the tests, the following parameters of the engine were established:

- the speed characteristics of the engine showing the dependence of its performance on the crank rotation frequency;
- the load characteristics showing the dependence of its performance on its load at constant crank rotation frequency;
- the ignition mapping in respect of ignition advance angle showing the dependence of engine performance on the moment of spark occurrence in a cylinder of the engine on its operation in the preset speed mode and a constant position of the fuel feeding control equipment;
- the control characteristic in respect of hourly fuel consumption showing the dependence of engine performance on the hourly fuel consumption in the preset speed mode.

In the second phase, upon striving to explore the impact of petrol–n-butanol mixture on the engine performance, an experiment of achievement of the values close to the reference D-optimal plan to the maximum possible extent for three variable parameters was outlined (Table 4) [14].

Table 4. The plan of achievement of the values close to D-optimal plan.

$j$	$x_1^*$	$\delta, \%$	$x_2^*$	$\theta, \text{ }^\circ\text{bTDC}$	$x_3^*$	$n, \text{ min}^{-1}$
1	+1	40	-1	30	+1	3500
2	+1	40	+1	50	-1	2500
3	+1	40	-1	30	-1	2500
4	+1	40	+1	50	+1	3500
5	-1	0	+1	50	0	3000
6	-1	0	0	40	+1	3500
7	-1	0	0	40	-1	2500
8	-1	0	-1	30	0	3000
9	0	20	-1	30	+1	3500
10	0	20	0	40	0	3000

The plan includes:  $j$  – the number of experiments;  $x_i$  – the reduced value of the  $i$ -th variable parameter on the natural measurement scale ( $i = 1, 2, 3$ ).

During the tests, the position of the throttle remained the same.

The variable parameters  $x_1, x_2, x_3$  predetermine the modes of engine operation: the content of n-butanol in its mixture with petrol  $\delta$ , the ignition advance angle  $\theta$  and the crankshaft rotation frequency  $n$ .

The parameters to be established include: the engine torque  $M_e$ , engine power  $P_e$ , and specific effective fuel consumption  $b_e$ .

The second-order regression equation for the saturated plan close to D-optimal plan in case of three variable parameters will be expressed as follows:

$$y = a_0 + a_1 \cdot x_1^* + a_2 \cdot x_2^* + a_3 \cdot x_3^* + a_{11} \cdot x_1^{*2} + a_{22} \cdot x_2^{*2} + a_{33} \cdot x_3^{*2} + a_{12} \cdot x_1^* \cdot x_2^* + a_{13} \cdot x_1^* \cdot x_3^* + a_{23} \cdot x_2^* \cdot x_3^*, \quad (1)$$

here  $a_0, a_1, a_2, a_3, a_{11}, a_{22}, a_{33}, a_{12}, a_{13}, a_{23}$  – the regression coefficients that will be established after procession of the results of the experiment.

$$x_i = 0.5x_i^* \cdot (x_{i \max} - x_{i \min}) + 0.5 \cdot (x_{i \max} + x_{i \min}), \quad (2)$$

here  $x_{i \min}$  and  $x_{i \max}$  – the limit values of the range of the  $i$ -th variable parameter.

The ranges of variation for:  
 $\delta - (0...40) \%$ ,  
 $\theta - (30...50)$  degrees of crankshaft position angle,  
 $n - (2500...3500) \text{ min}^{-1}$ .

#### 4. Results and discussion

In accordance with the plan of the experiment, we establish the values of the parameters and carry out the tests. Results of experiment are presented in the Table 5.

Table 5. Results of experiment.

<i>j</i>	Variable parameters			Defined parameters		
	$\delta, \%$	$\theta, \text{ }^\circ\text{bTDC}$	$n, \text{ min}^{-1}$	$M_e, \text{ Nm}$	$P_e, \text{ kW}$	$b_e, \text{ g/(kW}\cdot\text{h)}$
1	40	30	3500	20.5	7.51	592.4
2	40	50	2500	49.0	12.83	318.1
3	40	30	2500	41.0	10.73	373.7
4	40	50	3500	29.5	10.81	407.5
5	0	50	3000	46.0	14.45	305.1
6	0	40	3500	36.0	13.19	347.8
7	0	40	2500	55.0	14.40	292.8
8	0	30	3000	41.0	12.88	346.0
9	20	30	3500	24.5	8.98	503.0
10	20	40	3000	39.0	12.25	347.6

For establishing  $M_e$ ,  $P_e$  and  $b_e$  at different combinations of  $\delta$ ,  $\theta$ , and  $n$ , we form four regression dependences. For the said purpose, we calculate the regression coefficients on the base of the above-described method and the obtained experimental results and include the calculated values in the formula (1).

On the base of the provided characteristics, it should be noted that engine operation on petrol and engine operation on petrol blend with 10% n-butanol, no significant differences in power (Fig. 2b) and specific effective fuel consumption (Fig. 2c) were observed.

The ignition mapping in respect of ignition advance angle enables to single out the range of variation of the ignition advance angle when the specific effective fuel consumption is minimum (Fig. 2c) and the value of the engine power is maximum (Fig. 2b). It is notable that engine operation on petrol and engine operation on petrol blend with 10% n-butanol, no differences in the minimum value of the specific effective fuel consumption and the maximum value of the effective power were observed.

After establishing the characteristics, calculation of the engine performance parameters was performed and its results were fixed in the record.

The values of fuel consumption, power and engine torque depend on the atmospheric conditions during the tests. So, for comparison of the results of engine tests, obtained upon different atmospheric conditions, the values provided in the results for certain modes are reduced for the standard atmospheric conditions.

The regression dependences for establishing  $M_e$ ,  $P_e$  and  $b_e$  are expressed as follows:

$$M_e = 39 - 4.47 \cdot \delta^* + 3.38 \cdot \theta^* - 9.75 \cdot n^* + 1.59 \cdot \delta^{*2} - 1.56 \cdot \theta^{*2} + 0.44 \cdot n^{*2} + 0.88 \cdot \delta^* \cdot \theta^* - 0.25 \cdot \delta^* \cdot n^* + 0.25 \cdot \theta^* \cdot n^*, \quad (3)$$

$$P_e = 12.25 - 1.39 \cdot \delta^* + 1.07 \cdot \theta^* - 0.96 \cdot n^* + 0.56 \cdot \delta^{*2} - 0.54 \cdot \theta^{*2} - 0.41 \cdot n^{*2} + 0.28 \cdot \delta^* \cdot \theta^* - 0.35 \cdot \delta^* \cdot n^* + 0.3 \cdot \theta^* \cdot n^*, \quad (4)$$



$$b_e = 347,6 + 42,4 \cdot \delta^* - 40,3 \cdot \theta^* + 52,4 \cdot n^* + 2,4 \cdot \delta^{*2} + 17,9 \cdot \theta^{*2} + 12,6 \cdot n^{*2} - 19,8 \cdot \delta^* \cdot \theta^* + 24,8 \cdot \delta^* \cdot n^* - 32,3 \cdot \theta^* \cdot n^*, \quad (5)$$

here  $\delta^*$ ,  $\theta^*$ ,  $n^*$  – the reduced values of percentage of n-butanol content in its blend with petrol, the ignition advance angle and the crankshaft rotation frequency on the natural measurement scale.

The values of  $\delta^*$ ,  $\theta^*$ ,  $n^*$  are established from the following dependence:

$$x_i^* = \frac{2 \cdot x_i - (x_{i \max} + x_{i \min})}{(x_{i \max} - x_{i \min})}. \quad (6)$$

Let's examine the impact of  $\delta$ ,  $\theta$  and  $n$  on  $M_e$ ,  $P_e$  and  $b_e$  on the base of the obtained regress dependences.

The obtained regression dependences were used for assessing the impact of the crankshaft rotation frequency, the concentration of n-butanol in the combustible mixture and the ignition advance angle on the engine performance parameters.

The results for the analysis were obtained in the following way: the values of the engine torque, the engine power and the specific effective fuel consumption were calculated for a case when one of the variable parameters was provided a value from the range and two other parameters were varied.

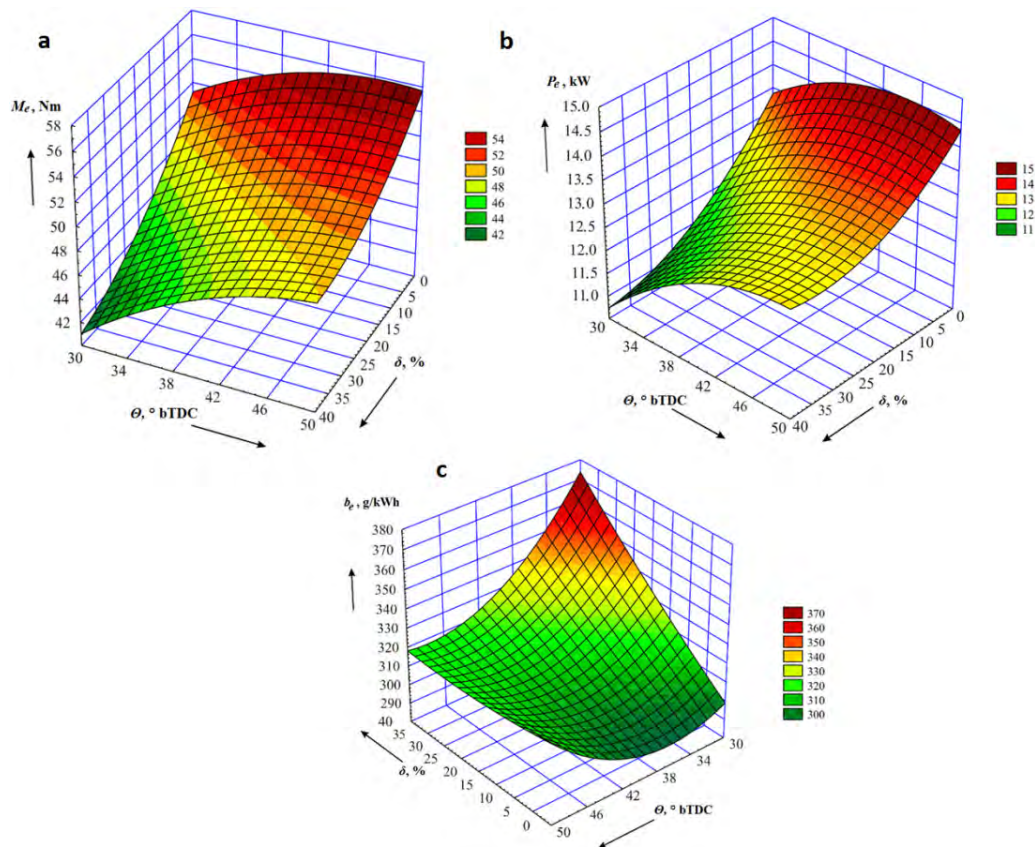


Fig. 2. The ignition advance angle and n-butanol concentration in the fuel dependence of the engine torque  $M_e$  (a); the engine power  $P_e$  (b); the specific effective fuel consumption  $b_e$  (c). Source: authors.

For example, the engine performance dependence on the ignition advance angle and the concentration of n-butanol in the combustible mixture were calculated for crankshaft frequency of  $2500 \text{ min}^{-1}$ . The surfaces shown in Fig. 2 were formed by the combination of the ignition mappings in respect of ignition advance angle obtained for different compositions of the combustible mixture.

The results show that increasing the n-butanol concentration in the blend causes changes of the engine performance. The power and the torque decrease by 1.6 kW (12.7%) and 6.8 N·m (13.8%), respectively; the specific effective fuel consumption increases by 6.3 g/(kW·h) (2%) at increase of the n-butanol concentration to 40%. In fact, no reduction of power and torque and increase of the specific effective fuel consumption take place when the engine operates on petrol blend with 10% n-butanol. Achievement of the best performance for engine operation on petrol and engine operation on petrol blend with 10% n-butanol is ensured at the same values of the ignition advance angle.

The peculiarity of the obtained dependences is a shift of the minimum effective fuel consumption and the maximum values of the torque and the power towards an increase of the ignition advance angle on increasing the content of n-butanol in the blend.

## 5. Conclusions

The engine performance for cases of its operation on petrol and on n-butanol–petrol mixtures with different concentrations of n-butanol (0, 20, 40%) has been established according to the load, speed and control characteristics that enable assessing the impact of the combustible blend on the engine operation.

The obtained regression dependences of the effective engine performance on concentration of n-butanol in its blend with petrol, the ignition advance angle and the crankshaft rotation frequency enable calculating the torque, the engine power, and the specific effective fuel consumption on various combinations.

It was found that in cases of engine operation on petrol and engine operation on petrol blend with 10% n-butanol, the engine performance remains practically the same and no alteration of the set ignition advance angle and hourly fuel consumption is required.

Increasing the concentration of n-butanol maximum torque and effective motor power decreases and the comparative fuel costs are rising. Energy characteristics of the engine is exacerbated by the lower n-butanol in the net calorific value compared to gasoline and lower n-butanol burning rate. Increasing the concentration of n-butanol fuel mixture, engine performance can be improved setting ignition advance angle.

## References

- [1] Directive 2009/28/EC of the European Parliament and of the Council of 23 April 2009 on the promotion of the use of energy from renewable sources and amending and subsequently repealing Directives 2001/77/EC and 2003/30/EC.
- [2] A. Rimkus, M. Melaika, J. Matijošius, S. Mikaliunas, S. Pukalskas, Investigation of combustion, performance and emission characteristics of spark ignition engine fuelled with butanol – gasoline mixture and a hydrogen enriched air, *Advances in Science and Technology Research Journal* 10 (2016) 102–108.
- [3] P. G. Aleiferis, M. K. Behringer, Flame front analysis of ethanol, butanol, iso-octane and gasoline in a spark-ignition engine using laser tomography and integral length scale measurements, *Combustion and Flame* 162 (2015) 4533–4552.
- [4] C. Tornatore, L. Marchitto, G. Valentino, F. E. Corcione, S. S. Merola, Optical diagnostics of the combustion process in a PFI SI boosted engine fueled with butanole – gasoline blend, *Energy* 45 (2012) 277–287.
- [5] E. Galloni, G. Fontana, S. Staccone, F. Scala, Performance analyses of a spark-ignition engine firing with gasoline–butanol blends at partial load operation, *Energy Conversion and Management* 110 (2016) 319–326.
- [6] A. Elfasakhany, Experimental study of dual n-butanol and iso-butanol additives on spark-ignition engine performance and emissions, *Fuel* 163 (2016) 166–174.
- [7] S. Szwaja, J. D. Naber, Combustion of n-butanol in a spark-ignition IC engine, *Fuel* 89 (2010) 1573–1582.
- [8] American Chemistry Council's Oxo Process Panel, in United States Environmental Protection Agency. In. SIDS Initial Assessment Report for SIAM 13. Bern, 6–9 November, 2001.
- [9] Ю. М. Лужков, Альтернативные источники энергии – будущее Москвы, Автогазозаправочный комплекс + Альтернативное топливо. 3 (2005) 3–5.
- [10] П. И. Степаненко, Российский бутанол тулунского разлива, *The Chemical Journal* (2008) 34–37.

- [11] А. С. Хачиян, Применение различных топлив и энергетических установок в автомобилях будущего, Двигателестроение 1 (2004) 28–32.
- [12] А. М. Данилов, Альтернативные топлива: достоинства и недостатки. Проблемы применения, Российский химический журнал 6 (2003) 4–11.
- [13] В. Н. Луканин, К. А. Морозов, А. С. Хачиян, И. В. Алексеев, Л. Н. Голубков, Б. Я. Черняк, В. И. Трусов, Г. М. Камфер, В. З. Махов, С. А. Пришвин, В. В. Синявский, Л. М. Матюхин, Н. И. Назаров, М. Г. Шатров, Двигатели внутреннего сгорания. Теория рабочих процессов: учебник для вузов, 2-е изд., перераб. и доп., Высшая школа, Москва. 2005.
- [14] К. Хартман, Э. Лецкий, В. Шефер, Планирование эксперимента в исследовании технологических процессов, Мир, Москва. 1977.
- [15] ГОСТ 14846–81. Двигатели автомобильные. Методы стендовых испытаний, Изд-во стандартов, Москва. 2003.