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The Influence of N-Butanol Addition in Gasoline on the Combustion in the Spark Ignition Engine

Cristian Sandu [1], Constantin Pana [1], Niculae Negurescu [1], Gheorghe Lazaroiu [2,*], Alexandru Cernat [1], Rares Georgescu [1] and Cristian Nutu [1]

[1] Department of Thermotechnics, Engines, Thermal and Frigorific Equipment, University Politehnica of Bucharest, 060042 Bucharest, Romania; sandu_cristian2007@yahoo.com (C.S.); constantinpana1@gmail.com (C.P.); niculae_negurescu@yahoo.com (N.N.); cernatalex@yahoo.com (A.C.); raresgeorgescu@yahoo.com (R.G.); nikolaoscristian87@gmail.com (C.N.)

[2] Department of Energy Generation and Use, University Politehnica of Bucharest, 060042 Bucharest, Romania

* Correspondence: glazaroiu@yahoo.com

Abstract: Butanol has good combustion properties and it can be a viable alternative fuel for automotive spark ignition engines due to its ability to improve energy and pollution performance. This paper analyses the influence of butanol content in gasoline blends on engine operation with a focus on operation stability, thermal efficiency and emissions. A Cielo Nubira A15MF engine type with four cylinders and a 1.5 L displacement was turbocharged and it was fueled with butanol in a blend with gasoline in percent's of 10% vol. and 15% vol. An operation regime of 2500 1/min speed and 55% engine load was used, at different dosages, at which the engine power remained constant. Regarding the engine fueled with butanol in a blend with gasoline, the operation stability was improved, especially when lean dosages were used; the dosages at which the thermal efficiency was higher are comparative to classic fueling. Concerning the use of lean dosages, the combustion duration decreases and the energetic engine performance was improved when butanol was used comparative to gasoline. When butanol was used, polluting emissions and emission with a greenhouse effect were reduced. The sharp reduction in NO_x is highlighted in this paper.

Keywords: spark ignition engine; butanol–gasoline blend; combustion; n-butanol; emissions

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1. Introduction

1.1. Studies

The current climatic changes represent a challenge for the inhabitants of the "blue planet" (humans and animals) and sometimes their survival is threatened by pollution or weather conditions. These climatic changes include storms, heat waves, rising sea levels, melting glaciers, frequent and intense drought and so on. There are solutions to combat climate changes, the most important being the reduction in deforestation, the development of renewable and sustainable energy sources and engaging with activities to reduce carbon emission. The reduction in pollutants and greenhouse gas emissions produced by automotive engines appears to be an immediate necessity of civil society. In order to reduce the effect of pollution and mitigate climate change, the automotive industry has started to bring viable solutions such as alternative fuel use, hybrid powertrain use, the application of tighter emission standards (such as Euro 6 and possibly Euro 7 in the future), electrification use, etc.

Thus, in this context, the European Union has started to impose more severe rules against pollution and it has started to promote, more than ever, the use of alternative fuels instead of the classic ones. The European Commission's goal for 2030 is to reduce the dioxide carbon emission level by around 37% in terms of transportation-related emissions, and to promote the use of alternative fuels, like alcohols and alternative fuels that can be use in blends with classic fuels or as a single fuel, to fuel the internal combustion engines of automotives. The use of low carbon fuels like butanol, ethanol and methanol can reduce the emission levels of internal combustion engines and modern engines can be relatively easily converted to alcohol use. In this sense, a transition to a low-level carbon emission economy, based on the sustainable and efficient use of alternative energy resources, may be ensured. Regarding the production of butanol, there are modern solutions which can be used to produce butanol from bio-waste in a sustainable, efficient and simple manner [1]. Food waste is an important resource according to researchers in the field. Bio-resources, like agricultural waste materials, can be an efficient source from which bio-butanol can be produced in a sustainable way [2].

For many decades, of all alcohols, ethanol has been considered a "green solution" in terms of its ability to be used as an alternative fuel for gasoline-fueled engines and it has been used in different proportions in blends with gasoline to fuel automotives engines. Currently, because ethanol is used in blends with gasoline on a large scale, marketed under the name of gasohol, many studies have investigated its use in internal combustion engines. Today, due to its good combustion properties, butanol can be a viable alternative fuel for internal combustion engines, for diesel engines [3,4] and for spark ignition engines [5,6]. Butanol can be blended with diesel fuel or with biodiesel; the amelioration of the biodiesel's combustion properties at butanol use can assure the improvement of the diesel engine operation [3,4]. For example, Lapuerta et al. [5] used renewable normal butanol, n-butanol, to replace the ethanol in blends with gasoline in order to eliminate the disadvantages of ethanol use, like hydroscopic issues, in the gasohol fuel. As the percentage of the alcohol blended with gasoline increases, the butanol–gasoline blend keeps a more stable research octane number comparative to the ethanol blend [5] and, from this point of view, the modification of the engine tune and engine design are not necessary. The lower vapor pressure of butanol affects the easiness of an engine cold start compared with ethanol, but the use of a butanol–ethanol–gasoline blend will not influence the engine's ability to cold start [5]. Lapuerta et al. [5] show that a butanol percentage of $16(\%)_v$ can be used in blends with classic fuel with no further issues in terms of engine operation; this is due to its good combustion properties, like its higher heating value and a reduced hydroscopic tendency compared with ethanol–gasoline blends. Lapuerta et al. [5] recommend the substitution of ethanol with butanol to compensate for the engine cold starting issues that occur in ethanol–gasoline fueled engines, especially in the case of E70 fuels, which contain large volume percentages of ethanol. Shang et al. [6] used normal butanol as fuel in a spark ignition engine and they evaluated the combustion and emission characteristics using a regime of 1500 1/min speed and 10% engine load. Shang used different butanol injection ratios 20–100%, and different air–fuel ratios λ = 0.9–1.3. [6]. At a 20% butanol injection ratio, the maximum pressure and temperature during combustion starts to increase, in conjunction with the increase in the heat release rate. Small butanol quantities, until 20%, ensure a slight rise in the maximum pressure during combustion for all air–fuel ratios, but a 40% butanol ratio leads to a similar maximum pressure that is achieved when gasoline is used. An higher butanol ratio, 60–100%, leads to a reduction in combustion pressure for all used air–fuel ratios, and the reduction was more significant during lean mixture operations [6]. A similar tendency appears in terms of the variation between the maximum temperature and the butanol ratio at different air–fuel ratios. Small butanol amounts assure the acceleration of the combustion process, the maximum temperature being increased. Conversely, larger quantities of butanol reduce the combustion temperature because of the amplification of the local cooling effect of the in-cylinder mixture; this occurs due to the evaporation of butanol from the final mixture. The use of 100% butanol leads to an increase in combustion duration with almost a 10–20 crank angle degree comparative to classic fueling, especially when lean mixtures are used, λ = 1.2–1.3, depending on the quality of the mixture. For rich and stoichiometric mixtures, the influence of the butanol ratio on combustion is reduced [6]. The COV value of the IMEP starts to decrease for low butanol rates, around 20%, but a further increase in the butanol ratio to 100% leads to the increase in the IMEP cyclic variability with ~1% at λ = 1.2 and with ~11% at λ = 1.3, in correlation with the influence of the butanol ratio on the angle of the combustion end [6]. The CO emission level is slightly affected by the n-butanol ratio, but HC emission levels reach their lowest values for 40–60% butanol ratio for lean mixture use [6]. The use of large butanol quantities, around an 80–100% butanol ratio, leads to the increase in the HC emission level at rich and stoichiometric mixture conditions comparative to classic fueling. The use of large butanol ratios, 40–60%, at any air–fuel mixture, leads to the decrease in the NO_x emission level [6], but higher values of the NO_x emission level being obtained for lower butanol ratios, like 20%, comparative to classic fueling [6]. Shang [6] shows that butanol use in a 100% ratio leads to an increase in the particle emission level only for rich mixture operation. Lewandrowski et al. [7] concluded that the use of ethanol can reduce greenhouse emissions by up to 43% comparative to gasoline. However, a study published in 2022 by Tyler and Nathan [8] contradict the Lewandrowski's study, stating that ethanol failed to reach its gas emissions target and has also led to 30% increased corn prices. The researchers from University of Wisconsin concluded that ethanol's carbon emission level is at least 24% higher than gasoline [9] (supported by the U.S.A Renewable Fuel Standard [10]). From this point of view, butanol can be a viable solution to ameliorate ethanol use issues. Generally, alcohols, such as ethanol, butanol and methanol, can lower fuel consumption and their use may lead to the improvement of engine efficiency [11–13]. According to Wu et al.'s study [14], the use of butanol as fuel may reduce the fuel consumption by up to 56% and greenhouse gases may be decreased by 48% [14]. Additional cost savings may be achieved by the use of butanol due to a lower corrosion level of butanol compared to other alcohols [15]. The high density of butanol may lead to lower fuel consumption according to the Sarathy et al. study [16]. An improvement in coefficient of variability (COV) was observed at fueling with a blend of n-butanol–gasoline comparative to gasoline [17]. Blending 30% of butanol with 70% gasoline may have a negative impact on engine power according to Alasfour's study [18]. Because of the lower heat value (LHV) of butanol comparative to gasoline, a 7% power loss was observed during experimental investigation. Wallner et al. [19] observe a small difference in emission levels for 10% butanol–90% gasoline blend use comparative to gasoline. The effects of blending different alcohols with gasoline were studied by Rice et al. [20]; lower levels for CO and NO_x emissions in the case of alcohol–gasoline blend use comparative to gasoline were observed.

1.2. Aim of Research

The objective of this paper is to study the effects of butanol use on an A15MF spark-ignition engine's performance. The current paper represents the continuation of the experimental investigations regarding the use of 10% n-butanol blended with gasoline in a spark ignition engine for the coefficient of air excess λ = 0.8–1.28 [21]. But, this is enhanced by increasing the percentage of butanol used, at 15% n-butanol, the use of the coefficient of air excess as λ = 0.9–1.3 and by expanding the study of the combustion process, especially from the point of view of analyses for the mass fraction burned (MFB-5%, MFB-50%, MFB-90%), cyclic dispersion and variability. The A15MF engine is fueled with blends of n-butanol–gasoline, butanol being used in different percentages (starting from gasoline G, followed by a blend of 10% vol. n-butanol and 90% vol. gasoline defined as GB10, and finally a blend of 15% n-butanol and 85% vol. gasoline defined as GB15). The experimental results bring new information which is presented comparative to other studies. For example, a study by Alasfour [22] shows that heating the inlet air can assure the cyclic temperature rise, which can lead to an increase in the NO_x emission level, misfire and knocking during engine operation. Alasfour [23] uses a blend of 30% butanol–70% gasoline which was tested in conditions of spark timing increase, but the increase in the spark timing can also have negative effects on the NO_x emission level [23]. Balaji and Venkatesan [24] concluded that the use of 10% ethanol and 5% iso-butanol offers the best experimental results and leads to the improvement of the engine performance and lower pollutant emissions levels. The addition of n-butanol to gasoline can reduce the coefficient of the cyclic variability, as COV, for the indicated mean effective pressure (IMEP), even at lean air–fuel mixture use [25], the vehicle drivability characteristics being improved according to Heywood [26], at rich and stoichiometric mixture conditions. Blending butanol with gasoline/diesel fuel may also lead to lower smoke emission levels [27]. Zheng et al. [28] observed that the use of butanol isomers in a diesel engine leads to lower soot emission levels and improved efficiency. An improvement in combustion can be achieved using n-butanol in an HCCI single-cylinder engine when misfiring occurs for very lean mixtures use [29]. Sandu et al. [21] use a 10% vol. butanol–90% vol. gasoline blend to fuel a spark ignition engine and observed lower engine performances, but with benefits for the engine strengths.

The combustion process and the engine stability in operation are influenced by the butanol properties. The main physical-chemical properties of butanol and gasoline are shown in Table 1.

Table 1. Physical-chemical properties of gasoline and butanol [24–30].

Fuel Property		Butanol	Gasoline
Chemical formula		C_4H_9OH	C_8H_{18}
Boiling temperature (at 101,325 Pa), [K]		390	303–463
Carbon content [% mass]		64.816	85.4
Hydrogen content [% mass]		13.59	14.2
Oxygen content [% mass]		21.6	0.4
Density at normal state (at 273.15 K, 101,325 Pa) [kg/m^3]		810	735–760
Kinematic viscosity (at 293.15 K, 0.1 MPa) [m^2/s]		3.6×10^{-6}	$(0.37–0.44) \times 10^{-7}$
Reid Vapor Pressure [kPa]		2.2	60–62
Flame temperature (in air at stoichiometric dosage) [K]		2195	2580
Laminar speed of flame (in air at stoichiometric dosage at 293.15 K, 101,325 Pa) [cm/s]		48	33–43
Lower heating value	[kJ/dm^3]	26,900–29,200	32,400
(gas at 273.15 K, 101,325 Pa)	[kJ/kg]	33,100–33,630	42,690
Molecular mass, [kg/kmol]		74.12	98.5
Octane number (research)		94-96	90–98
Cetane number		25	0–10
Temperature of autoignition [K]		616-618	257–327
Heat of vaporization (at 298.15 K) [kJ/kg]		585	351
Stoichiometric air fuel ratio		11.21	14.7

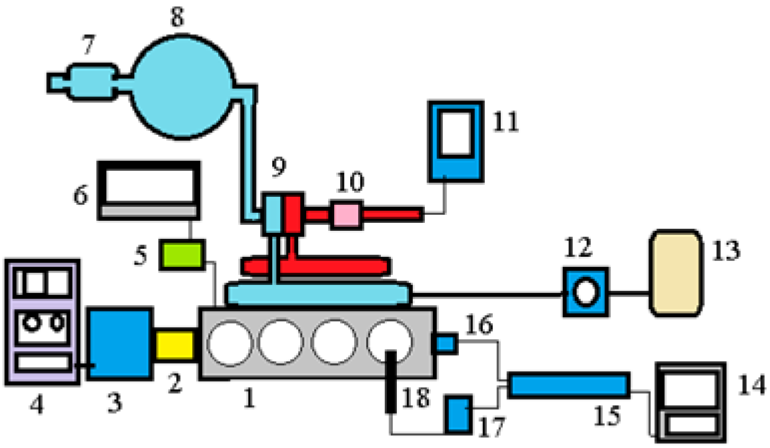
The higher octane number of butanol offers the possibility of operation without knocking phenomena, but the vapor pressure influences the ease of engine start, which becomes difficult for butanol use at low ambient temperatures. The higher lower heating value (LHV) of butanol comparative to other alcohols (33,630 kJ/kg versus 19,500 kJ/kg for methanol or 26,800 kJ/kg for ethanol) and, cumulative with its higher research octane number and higher combustion speed of air–butanol mixture, defines butanol as a viable alternative fuel for spark ignition engines. These combustion characteristics may improve the engine stability in operation for lean mixtures. The stability in engine operation can be evaluated with the calculated coefficient of variability (COV) determined for parameters like maximum pressure, indicated mean effective pressure, angle of maximum pressure, angles of mass fraction burned, etc., for many combustion cycles. In general, the operation stability of the engine is well evaluated based on the coefficient of variability of maximum pressure and indicated mean effective pressure. The engine response to the variability of the combustion process is evaluated based on the coefficient of variability of the indicated mean effective pressure, showing the unrepeatability and irreproducibility of the combustion phases from one cycle to another. The coefficient of variability for maximum pressure is more suitable for investigations of the operating regimes closer to the maximum torque brake speed regime when the spark timing tune is under optimization. For lean mixture operation, the engine stability is affected; the combustion total duration is increased because the initial phase and the final phase are also increased, as a common measure of correction is the tune of the spark timing. The use of butanol, which has a higher combustion speed (48 cm/s for butanol versus 33–43 cm/s for gasoline), in blends with gasoline can leads to an increase in the combustion speed, a reduction in the combustion duration and the amelioration of the lean mixture's influence on combustion duration and engine stability [26]. The instability of the combustion process, especially at lean and very lean mixture use, accentuates the cyclic dispersion through a higher irreproducibility of the combustion phases into consecutive combustion cycles, and such dispersion can be ameliorated by butanol use. For lean mixture use, because of the lower reaction speed, the combustion process is sensitively influenced by the fluctuation of the local dosages or by the concentrations of the residual gases into the mixture area closer to the spark plug electrodes. When the influence of the micro turbulence which dilutes the concentration of the active nucleus and accelerates the heat transfer is present, the initial phase of combustion increases, which leads to the prolongation of the combustion into the detent zone. The beginning and the end of combustion are largely influenced by the physical-chemical properties of the air–fuel mixture, the influence of the chemical nature of the fuel being major. The chemical composition of the fuel can influence the combustion process through the chemical reaction speed, with a sensible influence of the initial phase of combustion, MFB-5% (mass fraction burned) and the final phase of combustion, MFB-90%, through the intensity of the chemical transformations inside the reaction zone. These aspects are influenced by the presence of butanol in fuel [24–30]. Thus, at butanol use, if the spark ignition timing remains unchanged, a slight reduction in maximum pressure or in indicated mean effective pressure values may appear, especially at lean dosage use, in correlation with a lower value of the LHV of butanol versus gasoline. This aspect is compensated by the acceleration of the combustion processes and reduction in its duration at butanol use, especially for lean mixture use, comparative to classic fuel use.

Yusuf et al.'s [31] study shows the influence of butanol (B)–hydrogen (H) blend use for after the treatment of the exhaust gases on particle emission from city vehicles. Yusuf et al. [32] show that the increase in the n-butanol dose leads to the reduction in PN emissions. The use of butanol and hydrogen, as n-BH16, leads to the reduction in the levels of CO and NO_x by 84% and 31%, respectively, and a slightly increase of 12% for HC comparative with the reference BH00. The use of small quantities of butanol and hydrogen in gasoline leads to an increase in the in-cylinder temperature. BH196 use leads to an increase in the combustion maximum temperature comparative to BH00, a higher peak which appears sooner per cycle, but the ignition issues disappear at its use. Yusuf et al. [32] show that the use of butanol and hydrogen in a spark ignition engine leads to quicker ignition per cycle, a lower cyclic variability for the indicated mean effective pressure, under 5% for the COV of IMEP, a decrease in the brake-specific fuel consumption by 3.7% and a decrease in HC and NO_x emission levels, but a slight increase in CO emission levels by around 3%. The NO_x and HC emission levels can be further decreased by an ignition timing tune at 8 °CA. Generally, the use of n-butanol–hydrogen–gasoline blends leads to an increase in the heat release rate, in-cylinder pressure, maximum pressure rise rate and brake thermal efficiency, with the combustion being accelerated.

This paper presents some experimental results obtained via butanol use in a turbocharged spark ignition engine. The 10% and 15% of n-butanol is blended with gasoline, GB10 fuel and GB15 fuel, respectively. The fuel blends are used to fuel a spark ignition engine at the regime of 55% engine load and 2500 1/min speed in two mixture conditions: rich mixture and lean mixture. The influences of butanol addition quantity in blends with gasoline on combustion, maximum pressure, indicated mean effective pressure, cyclic variation (evaluated using the coefficients of cyclic variation COV_{pmax} and COV_{IMEP}), conventional mass fraction burned MFB, brake-specific energetic consumption, thermal efficiency and pollutant performance are presented and analyzed in this paper.

2. Experimental and Theoretical Investigation Design

The schema of the experimental engine test bench is represented in Figure 1.



1-A15MF engine, 2-dyno coupling, 3-AVL eddy curent dyno, 4-dyno cabinet, 5-engine control unit equipped with Dastek Unichip Q, 6-computer of engine control unit, 7-Krohne air flowmeter, 8-inlet air tank, 9-turbocharger, 10-catalytic converter, 11-AVL gas analizer, 12-Krohne fuel mass flowmeter, 13-fuel tank, 14-data acquisition PC, 15- AVL Indimodul, 16-AVL angle encoder, 17-AVL charge amplifier, 18-AVL piezoelectric transducer

Figure 1. Engine test bench schema.

The experimental investigations were carried out on a four-cylinder engine A15MF type, turbocharged, with a displacement of 1.5 L, designed for Cielo Nubira automotives. During the experimental investigations, the reference was set up for gasoline fueling at the engine load of 55%, engine speed of 2500 1/min and different coefficients of excess air for air–fuel mixtures. After setting the reference, the engine was fueled with an n-butanol–gasoline blend at different percentages (10%, 15% vol. butanol in blend with gasoline) and the engine power was maintained at a constant value. The fueling system is equipped with an open electronic control unit Dastek Unichip Q that provides the possibility to control the opening durations of the fuel injectors and to adjust the value of excess air-coefficient for the air–fuel mixture. For rich mixture use, the coefficient of excess air is set up at 0.9 and for the lean mixture use and the coefficient of excess air is set up at 1.3 to evaluate the influence of butanol percent in blends with gasoline on different air–fuel mixtures.

In the first place, the engine was fueled with gasoline in order to set up the reference for the experimental investigation. Secondly, two fuel blends were prepared, the first fuel GB10 which contained 10 $(\%)_{vol.}$ of n-butanol and 90 $(\%)_{vol.}$ of gasoline and the second fuel with 15 $(\%)_{vol.}$ of n-butanol and 85 $(\%)_{vol.}$ of gasoline. During the experimental investigations after reference set up, the fuel tank was emptied and immediately refiled first with the GB10 fuel and secondly with the GB15 fuel. In each fueling case, the engine operating regime and the air–fuel mixture condition, $\lambda = 0.9$ and $\lambda = 1.3$, were precisely restored. The experimental methodology assumed the acquisition of the in-cylinder pressure data for 250 consecutive combustion cycles, the measurement of the fuel consumption, the measurement of the level of polluting emissions CO, HC, NO_x and the CO_2 greenhouse gas in samples of three sets of data.

Comparative to the previous study [19], the novelty of the paper is assured by the following aspects: the percent of n-butanol used was increased; an electronic tune of the fuel cyclic dose was used in order to modify the coefficient of air excess at 15% n-butanol use; the optimum correlation was established between engine load, butanol quantity, injection duration, supercharging pressure–air/fuel ratio and exhaust gas temperature in order to control the combustion process and to obtain the best ecological and energetic performance of the spark ignition engine at butanol–gasoline blend fueling. This study is continued and completed with the analysis of a representative number of combustion cycles for maximum pressure, indicated mean effective pressure, the mass fraction burned (MFB-5%, MFB-50% and MFB-90%) and by establishing the degree of cyclic variability during combustion at the use of 10% and 15% n-butanol blended with gasoline.

3. Results and Discussion

The presented results were the subject of an uncertainty analysis in the conditions in which a set of three measurements registered for each operation point were recorded. Thus, the uncertainty analysis coefficient used to establish the error of measurement for the experimental results is established by the calculus formula:

$$uac = \frac{\sqrt{N^{-1} \cdot \sum_{i=1}^{N} \left(z_i - \frac{\sum_{i=1}^{N} z_i}{N} \right)^2}}{\frac{\sum_{i=1}^{N} z_i}{N} \cdot N^{-1}} \quad (1)$$

A set of three measurements is recorded for each engine load regime. The uncertainty analysis coefficient is used to evaluate the measurement error for the results. The values calculated for the uncertainty analysis coefficient determined for the experimental data like fuel consumption, thermal efficiency, levels of CO, CO_2, HC and NO_x did not exceed 1% due to the stability of the operating regimes. The measurement sample is uniform and the data spreading is narrow; the values of the uncertainty analysis coefficient are under the convergence limit for each one of the three sets of the measured data. For in-cylinder parameters, due to the AVL Indimodul data acquisition system measurement stability performance, the uncertainty is below $\pm$0.5%.

3.1. Operation Stability

3.1.1. Operation Stability for Rich Mixtures (Coefficient of Air Excess λ = 0.9)

The engine operation stability is evaluated based on the coefficient of cyclic variability, defined as COV, which in general is calculated for two parameters: maximum pressure and indicated mean effective pressure, as IMEP, considering that these two parameters reflect the degree of cyclic dispersion well enough. Thus, it becomes interesting to follow the variation of the maximum pressure and the IMEP along several consecutive operating cycles and to determine their COV values for operation stability evaluation. The COV of the IMEP, COV_{IMEP}, shows the engine's response to the cyclic variability of the combustion process and more concretely evaluates the degree of irreproducibility of the phases of the combustion process from one cycle to another. This is an important issue to evaluate when alternative fuels are used. The COV of the maximum pressure, COV_{pmax}, is usually used to evaluate the cyclic dispersion at operating regimes close to the regime of maximum engine torque, as maximum torque brake. The COV_{pmax} shows the dispersion of the maximum values of the pressure that depends on the duration of the combustion and its positioning in relation to the TDC area (top dead center), the duration of the initial phase of combustion and the heat release rate. These parameters, as well as the dispersion of the maximum pressure, depend on numerous factors; the most important factor is the mixture dosage, rich or lean, dosage state that will also influence the values for the IMEP and its dispersion. Thus, in Figure 2, the variations of maximum pressure are presented in blue for gasoline (G), in orange for GB10 fuel and in red for GB15 fuel. The GB10 fuel will be referred to as 90% gasoline–10% butanol and the GB15 fuel as 85% gasoline–15% butanol. For the GB10 fuel, the maximum pressure has values similar to those assigned to gasoline, but the combustion process is faster comparative to gasoline or GB15 fuel. The higher flame speed of butanol comparative to gasoline may ensure a faster combustion of air–fuel mixtures in which n-butanol is present. However, blending higher percentages of butanol with gasoline seems to have almost no impact on the maximum pressure, as the pressure diagram is almost identical to that of gasoline. Similar tendencies were observed by other researchers during their experiments [6,7,21,29].

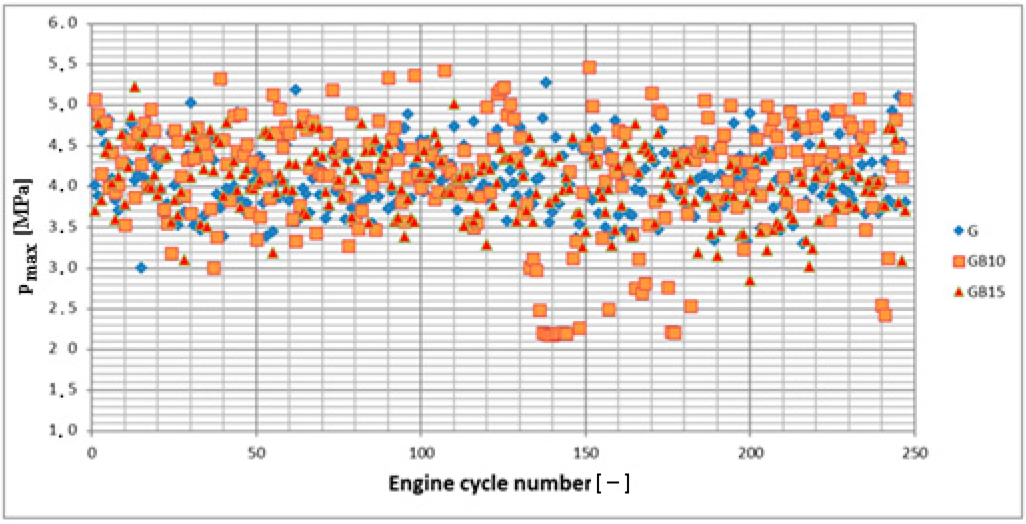


Figure 2. Peak pressure values for G, GB10 and GB15 fuels for rich mixtures.

The coefficient of variability (COV) is calculated with the following formulas [26]:

$$COV_p = \frac{\sqrt{\frac{1}{N_c-1} \cdot \sum_{k=1}^{N_c} (p_k - p_m)^2}}{p_m} \tag{2}$$

$$p_m = \frac{\sum_{k=1}^{N} p_k}{N_c} \tag{3}$$

where N_c is number of the combustion cycles, p_k is the current value of the parameter, p_m is the average value of the parameter, $p_{\max}$ is the maximum pressure of the parameter and the indicated mean effective pressure is IMEP. If the value of the COV does not exceed 10% for the maximum pressure and IMEP, the normal and stable operation of the engine is assured [26].

The peak pressures values for gasoline and both butanol-based fuels, GB10 and GB15, can be similar in some individual combustion cycles. It was observed that in 250 consecutive cycles the dispersion between the values is greater for the GB10 fuel, with a coefficient COV_{pmax} = 18%, comparative to 10% for gasoline. A 57% increase in variation is noticed at fueling with the GB10 fuel, while for GB15 fuel use the COV_{pmax} is 10.23%, only 0.23% higher comparative to gasoline. Butanol's good miscibility properties, especially at higher butanol percentages, seems to have no impact on the engine operation for rich mixtures, where pressure values are similar to pressure values recorded at gasoline fueling. The values measured for the indicated mean effective pressure (IMEP) are presented in Figure 3. Fueling with the GB10 fuel leads to lower averaged IMEP values comparative to gasoline and the GB15 fuel. The dispersion between IMEP values is also more noticeable for the GB10 fuel, with a COV_{IMEP} value of 9% versus COV_{IMEP} = 2.76% for gasoline and COV_{IMEP} = 3.19% for the GB15 fuel. The increase in variation is around 0.43% for the GB15 fuel. For both butanol-based fuels, the variation of the IMEP is lower than 10%, and the engine's normal operation stability is assured. Similar results were obtained by other researchers [17,21,25,30–32].

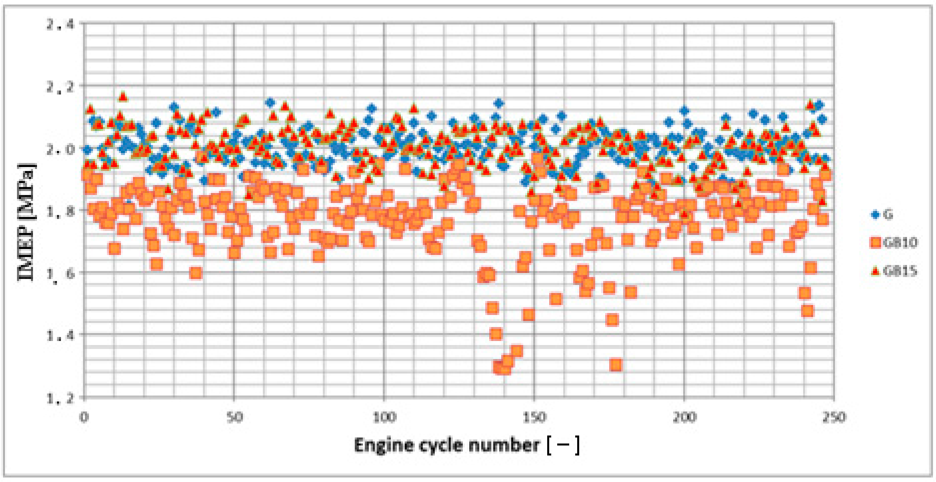


Figure 3. The IMEP for G, GB10 and GB15 fuels for rich mixtures.

The heat release rate is calculated using the following formula [21,26]:

$$\frac{dQ}{d\alpha} = \frac{V}{k-1} \cdot \frac{dp}{d\alpha} + \frac{k}{k-1} \cdot p \cdot \frac{dV}{d\alpha} \quad (4)$$

where:

- $\frac{dQ}{d\alpha}$—heat release rate;
- $\frac{p \cdot dV}{d\alpha}$—mechanical work variation;
- $\frac{dp}{d\alpha}$—pressure variation;
- k—adiabatic exponent;
- V—volume;
- p—pressure;
- α—crankshaft angular position.

The angles of mass fraction burned for 5%, 50% and 90%, as the angles at which the conventional fractions of combustion heat are released, are presented for 250 consecutive combustion cycles in Figures 4–6 for all tested fuels. The angles of mass fraction burned for 5%, 50% and 90% correspond to the mass fraction burned MFB-5%, MFB-50% and MFB-90%. The start of combustion appears sooner for butanol use, at a 16.5 crank angle degree (°CA) for the GB10 fuel and at a 12.2 °CA for the GB15 fuel comparative to the 18.2 °CA registered at gasoline fueling, taking into consideration the averaged values. For 5% heat released, the use of gasoline leads to the most stable operation, with a $COV_{5\%}$ = 9% compared to $COV_{5\%}$ = 14% for the GB10 and GB15 fuels. The main phase of combustion tends to appears later per cycle for butanol use, at 32.3 °CA for the GB10 fuel and at 28.2 °CA for the GB15 fuel comparative to 15 °CA for gasoline, as averaged values; this aspect may influence the increase in the COV_{pmax} for butanol use at rich dosages in conditions of unchanged spark ignition timing. The influence on the COV_{IMEP} value is not significant. However, at 50% heat released, the gasoline fueling operation is the least stable, with a $COV_{50\%}$ of 16.52% compared to a $COV_{50\%}$ = 10.5% for the butanol-based fuels. A possible explication is that the addition of oxygen stabilizes the main combustion phase. The end of combustion appears later in the cycle for butanol use, with an average value of 7.5 °CA for the GB10 fuel and 0.5 °CA for the GB15 fuel comparative to gasoline. Other researchers [6] obtained similar results.

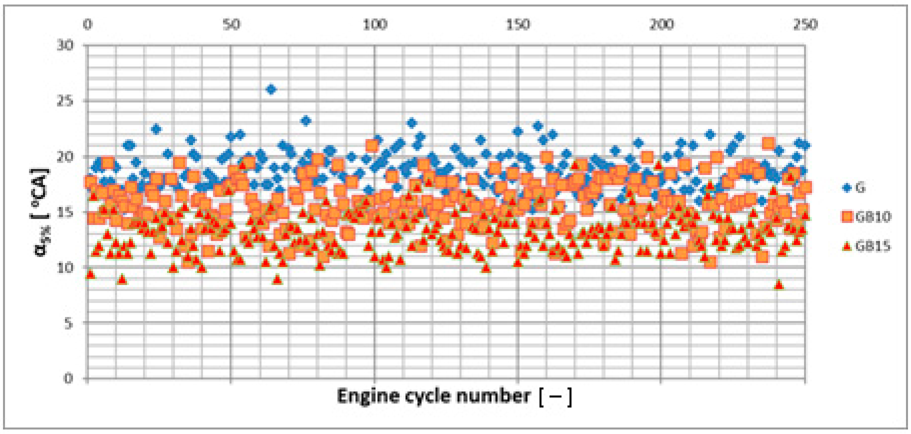


Figure 4. Heat released (5%) angles for G, GB10 and GB15 fuels for rich mixtures.

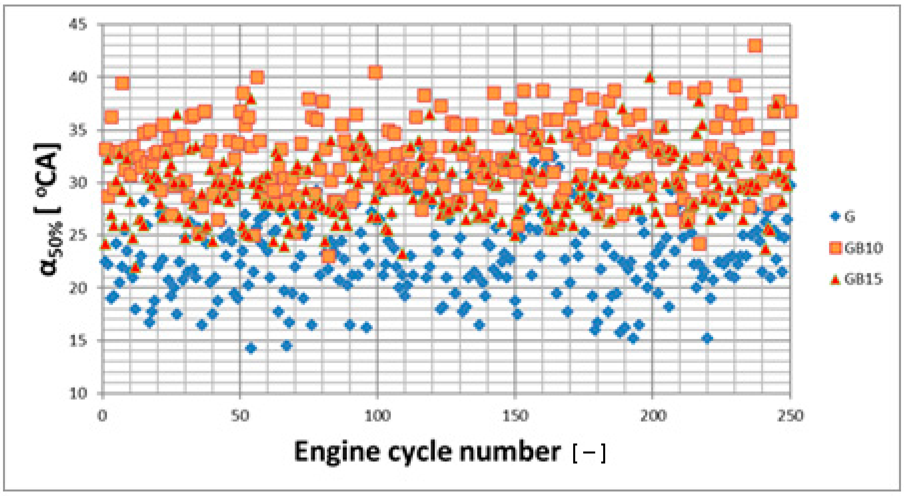


Figure 5. Heat released (50%) angles for G, GB10 and GB15 fuels for rich mixtures.

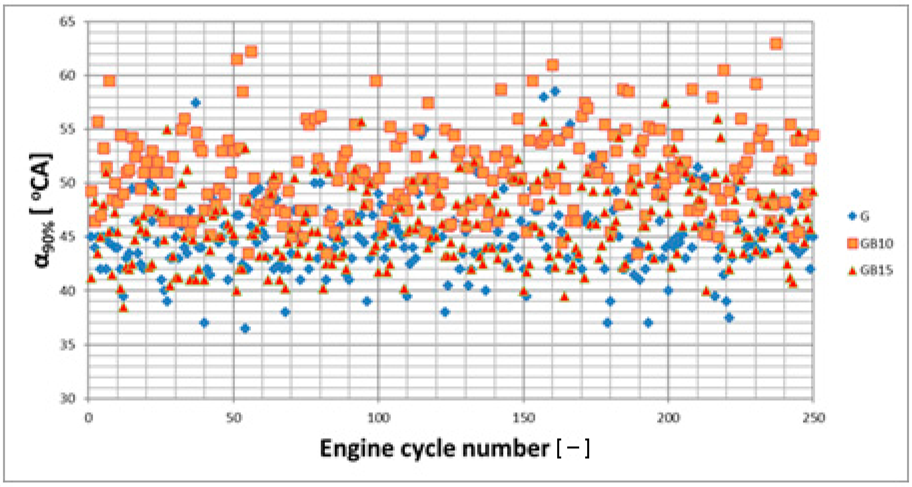


Figure 6. Heat released (90%) angles for G, GB10 and GB15 fuels for rich mixtures.

At 90% heat released, there is a small difference in $COV_{90\%}$ values between the three fuels and the dispersion is greatly reduced to less than 1%. For rich mixtures, the use of the GB15 fuel favors the reduction in the combustion duration in a more significant way comparative to the GB10 fuel. Other researchers obtained similar results [6].

3.1.2. Operation Stability for Lean Mixtures (λ = 1.3)

The peak pressure variation for both butanol-based fuels is shown in Figure 7; values registered in 250 combustion consecutive cycles are presented. At fueling with butanol-based fuels, the maximum pressures are lower compared to gasoline. A drop of 8.7% in peak pressure is observed at fueling with the GB15 fuel and a decrease of 12% for the GB10 fuel. It is also noticed that, overall, the GB10 fuel achieves lower pressures during the combustion process for lean mixtures. The measured values of maximum pressure, Figure 7, have a relatively higher dispersion rate for lean mixture use comparative to rich mixture use. GB10 fuel use leads to a much more stable operation; the value of the COV_{pmax} is 11% for the GB10 fuel comparative to the COV_{pmax} = 15.34% for gasoline (showing a 33% reduction) and the COV_{pmax} = 16% for the GB15 fuel. The use of 10% butanol may improve the operation stability for lean mixtures, but for 15% butanol use, the spark timing tune must be considered in further investigation. Similar results were obtained by other researchers [5,6,17,21,25,30–32].

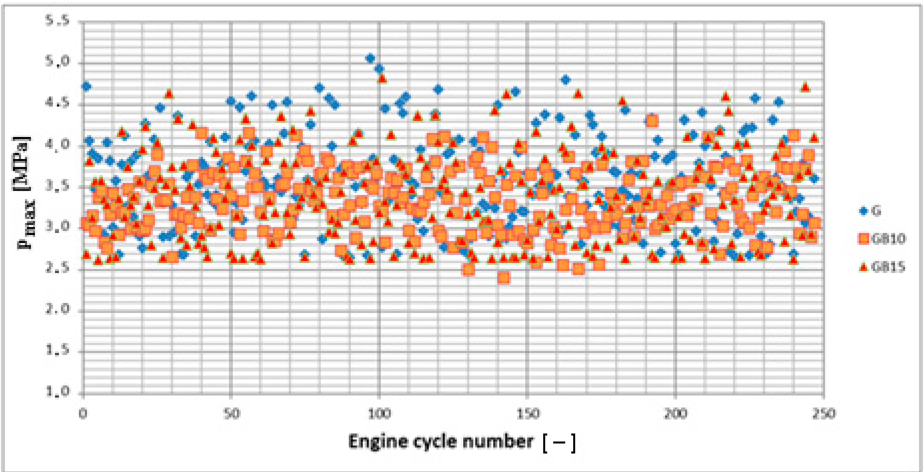


Figure 7. Peak pressure values for G, GB10 and GB15 fuels for lean mixtures.

The indicated mean effective pressure values for lean mixture operation are presented in Figure 8. Lower average IMEP values are achieved for GB10 fuel use, but with a lower dispersion in consecutive values, with a COV_{IMEP} of 3.25% versus a COV_{IMEP} = 4.9% for gasoline and a COV_{IMEP} = 5.3% for the GB15 fuel. In all cases, the dispersion values are below 10% so there should be no negative influence on the engine operation stability or on the vehicle's drivability.

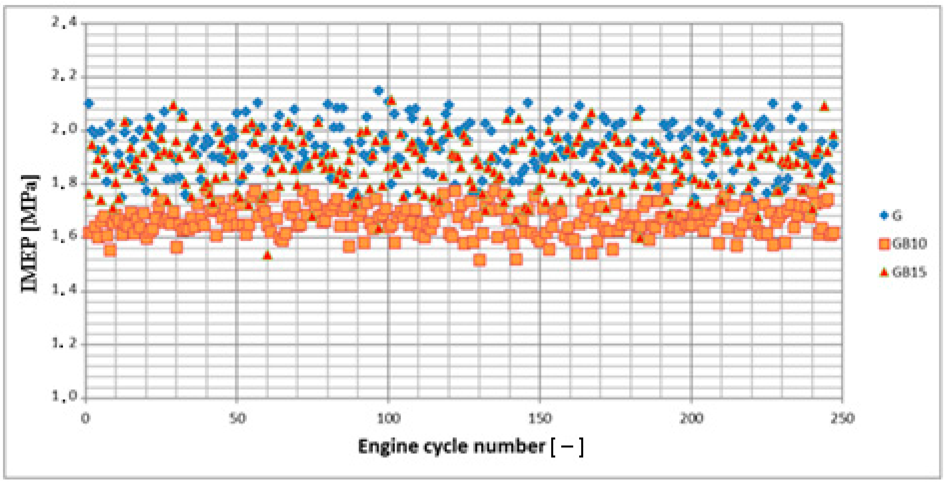


Figure 8. The IMEP for G, GB10 and GB15 fuels for lean mixtures.

For lean mixture use, the conventional mass fraction burned (MFB) is presented in Figures 9–11. GB10 fuel use leads to a faster achievement per cycle of the combustion initial phase, with an average value of 12.3 °CA for MFB-5% comparative to 13.5 °CA for gasoline fueling for lean mixtures. The main phase of combustion appears at 31.2 °CA for the GB10 fuel with a 3.4 °CA closer to the top dead center comparative to gasoline use (for gasoline, the MFB-50% appears at a 34.6 °CA) and at a 36.6 °CA for the GB15 fuel, 2 °CA later versus gasoline use. GB10 fuel use offers more stable combustion in terms of 5% and 50% of heat released. Comparative to gasoline and GB15 fuel use, GB10 fuel use leads to a 65% improvement in the $OV_{5\%}$ ($COV_{5\%}$ = 12.81% for the GB10 fuel versus $COV_{5\%}$ = 25.23% for gasoline) and a 44% improvement in the $COV_{50\%}$ ($COV_{50\%}$ = 16.66% for GB10 versus $COV_{50\%}$ = 10.58% for gasoline). GB15 fuel use leads to similar COV values to the ones registered for gasoline use, but slightly increased. It seems that the GB10 fuel offers a more stable engine operation for lean mixtures comparative to gasoline, but with loss of performance, lower averaged values for peak pressure and IMEP being registered. The GB15 fuel assures a slight loss of performance but shows no improvement in engine stability in a significant way. The end of combustion appears 8.1 °CA sooner per cycle for the GB10 fuel and 4 °CA later per cycle for the GB15 fuel comparative to gasoline. Further improvements may be possible with the optimization of the engine tune.

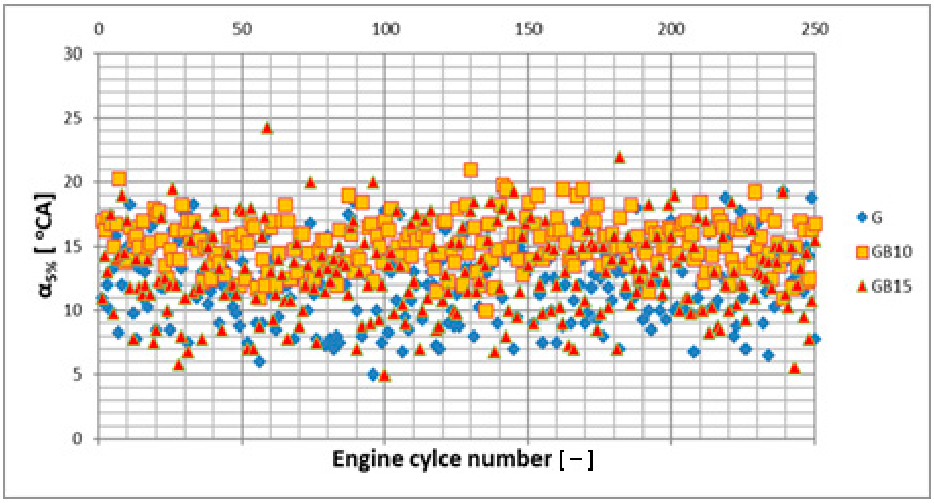


Figure 9. Heat released (5%) angles for G, GB10 and GB15 fuels for lean mixtures.

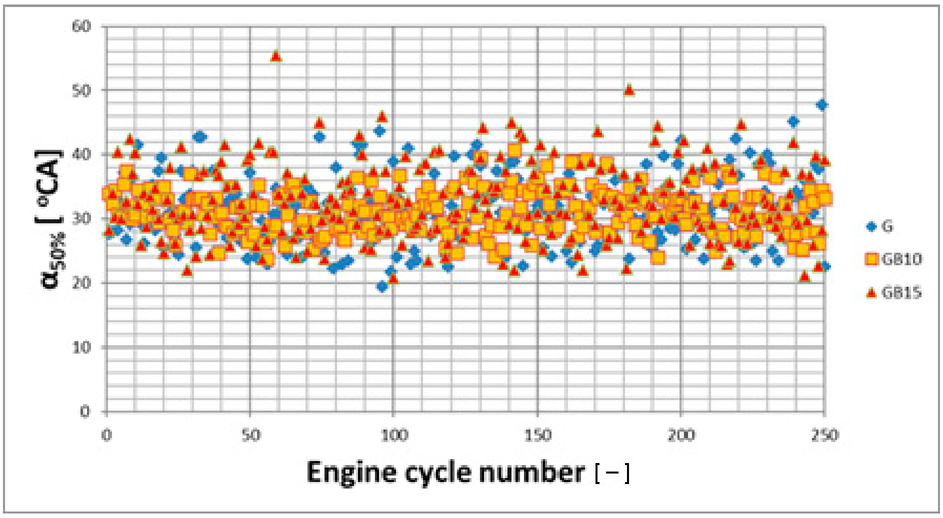


Figure 10. Heat released (50%) angles for G, GB10 and GB15 fuels for lean mixtures.

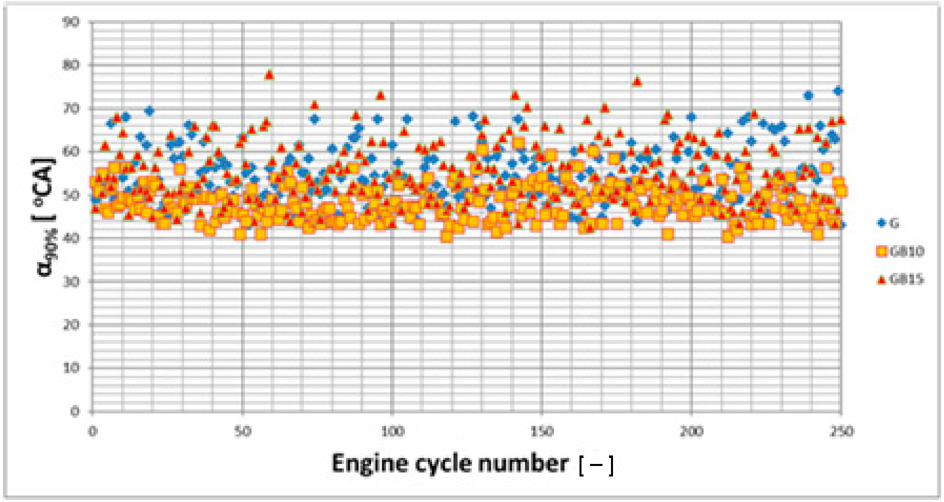


Figure 11. Heat released (90%) angles for G, GB10 and GB15 fuels for lean mixtures.

For lean mixtures, the use of the GB10 fuel favors the reduction in the combustion duration Δ_{5-90} until 10 °CA, while the GB10 fuel ensures a slight increase in combustion duration of around Δ_{5-90} = 5 °CA. Similar results were achieved by other researchers [5,6,17,21,25,30,31].

3.2. Brake-Specific Energy Consumption and Emissions

The brake-specific energy consumption (BSEC) variation versus the coefficient of air excess λ is presented in Figure 12 for gasoline G and butanol-based fuels and the GB10 and GB15 fuels. In the area of λ = 0.9–1.1, the lowest BSEC is achieved for gasoline use. A rich mixture and gasoline use leads to a lower BSEC, 5.7% lower comparative to the GB10 fuel and 37% lower comparative to the GB15 fuel. GB15 fuel use leads to an improvement of the BSEC for lean mixtures, λ = 1.3. Comparative to gasoline use, the BSEC is 3.97% lower comparative to gasoline or the GB10 fuel. Similar results were obtained by other researchers [14,16,21,28,31,32].

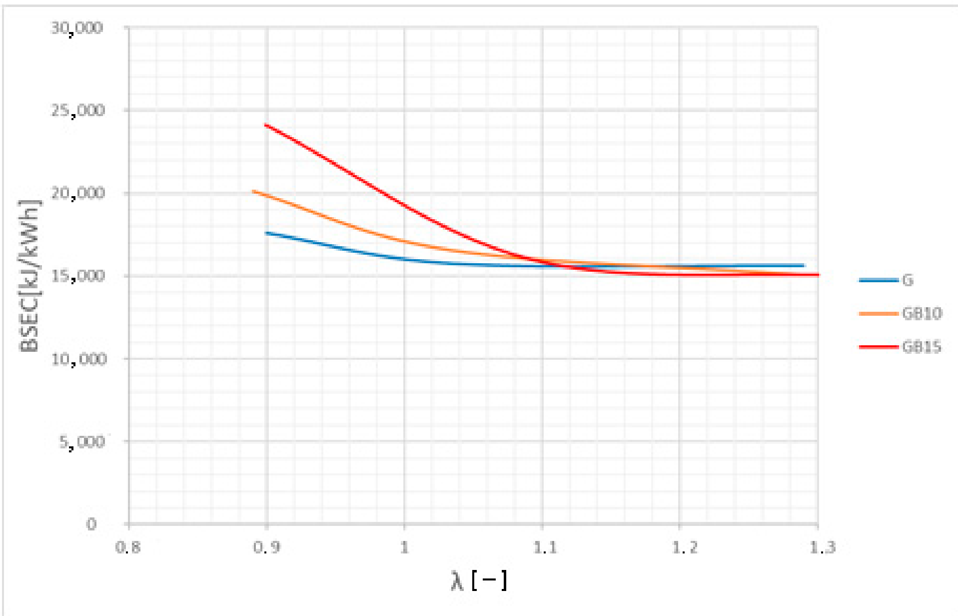


Figure 12. The BSEC for G, GB10 and GB15 fuels.

The high value of the BSEC recorded at λ = 0.9 for GB15 fuel use can be explained by a low thermal efficiency, as Figure 13 shows. For rich mixture and gasoline use, the engine thermal efficiency is 12.5% higher comparative to butanol-based fuels, with a low difference of 0.01 between the GB10 and GB15 fuels. For lean mixture use, λ = 1.15–1.3, the use of gasoline leads to a reduction in the engine thermal efficiency, and so it is advantageous to use the GB15 fuel, especially for λ = 1.3, due to a higher thermal efficiency achievement, Figure 13, and a lower brake-specific energy consumption, Figure 12. GB10 fuel use leads to similar performance in the domain of lean mixtures. Similar results were obtained by other researchers [14,16,21,28,32].

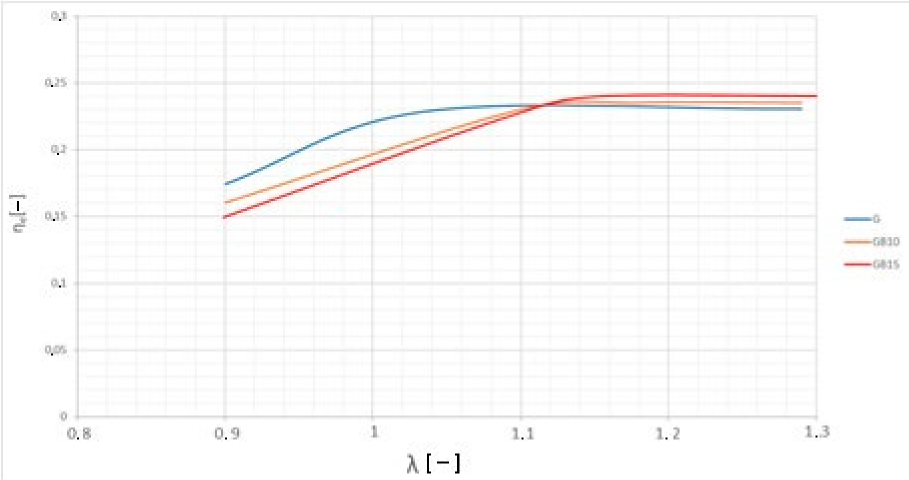


Figure 13. The engine thermal efficiency for G, GB10 and GB15 fuels.

The formation of carbon monoxide is generated by incomplete combustion because of the lack of oxygen in the in-cylinder. Engine operation for rich mixtures favors an increase in the carbon monoxide emission levels and also, for the combustion of lean mixtures, the dissociation reactions leads to the formation of low amounts of carbon monoxide [26].

An increased level of CO emission, caused by an incomplete combustion of the rich air–fuel mixture, is shown in Figure 14. For rich mixtures, GB10 fuel use generates the lowest level of the CO emission, almost 22% lower comparative to the GB15 fuel and 11% lower comparative to gasoline. For very lean mixtures, there are no CO emission level improvements at fueling with the GB10 or GB15 fuels. The variation tendency of the carbon monoxide emission level for butanol use was studied by other researchers and similar aspects are present [6,20,21,24,31].

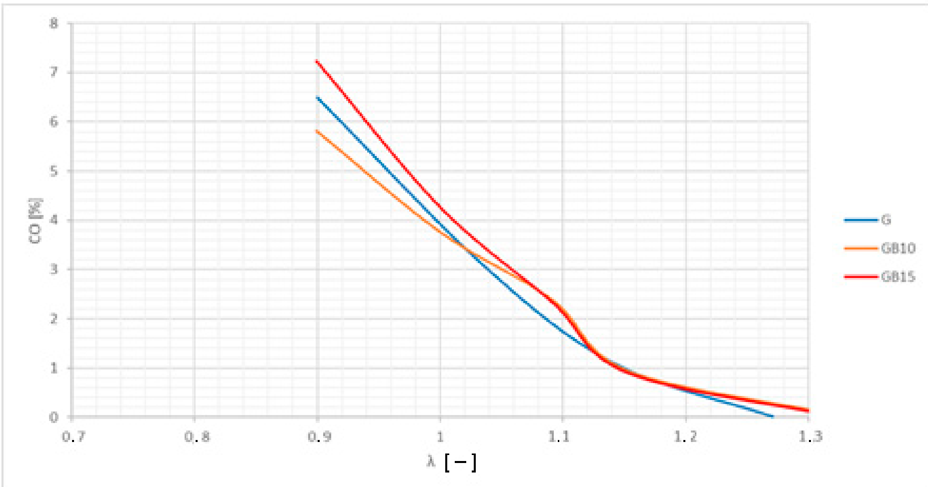


Figure 14. The CO emission levels for G, GB10 and GB15 fuels.

The CO_2 emission level, Figure 15, can be an indicator regarding the complete combustion of the air–fuel mixture. For rich mixtures, GB10 fuel use leads to an increase in the CO_2 emission level of 23% comparative to the GB15 fuel and of 11% comparative to gasoline. For lean and very lean mixtures, the difference between CO_2 emission levels registered for all fuels are negligible. Similar results of CO_2 emission level reduction were obtained by other researchers [6,7,21,31].

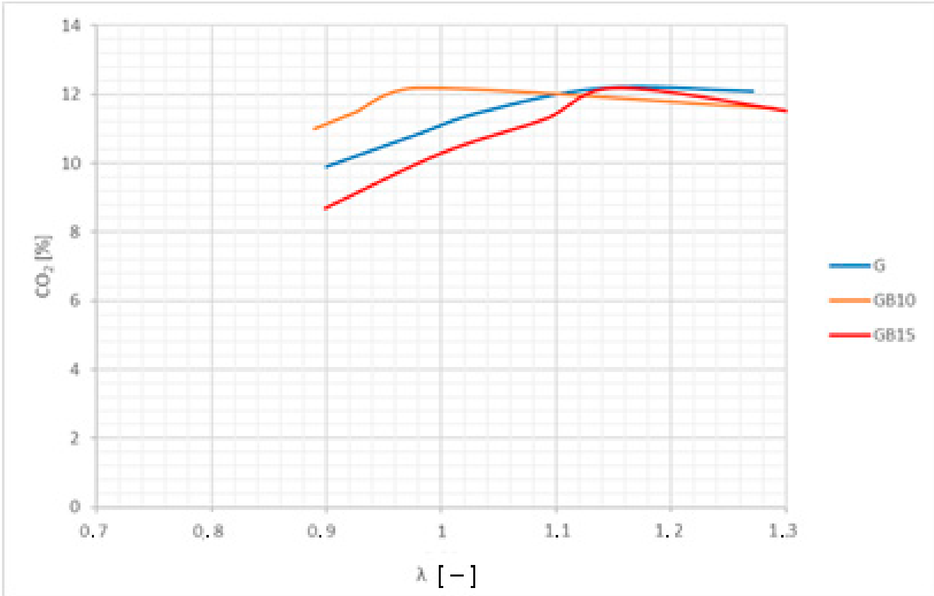


Figure 15. The CO_2 emission levels for G, GB10 and GB15 fuels.

The emission of unburnt hydrocarbons, HCs, is generated during an incomplete combustion process which may appear because of the flame extinguishing on the in-cylinder walls or because of the flame extinguishing in the mass of in-cylinder gases. The low temperature of the walls can assure the interruption of chemical reactions and the formation of unburned HCs in the boundary layer at the wall. The flame extinguishing in the in-cylinder gas may appear because of important concentrations of residual burnt gases. The flame extinguishing in the combustion chamber crevices can be a phenomenon that leads to HC formation. Others sources of unburnt hydrocarbons can be misfire cycles, but a part of the unburned HCs evacuated from the cylinder can continue to combust in the exhaust system in the presence of oxygen in the exhaust gases at local temperatures of over 600 °C [26]. This process of HC degradation can be initiated during the exhaust process and continues as the gas flows inside the exhaust system [26]. The use of lean mixture can be a solution for reducing the HC concentration inside the boundary layer, reducing the thickness of the boundary layer and the surface where it forms, by using fuels with lower carbon content, like butanol. The unburned hydrocarbon emission (HC) level variation is presented in Figure 16. The use of GB10 fuel leads to a 10% lower level of HC emission comparative to gasoline. For lean mixtures, both butanol-based fuels, GB10 and GB15, assure a 26% lower HC emission level comparative to gasoline. For very lean mixtures, the results are similar. A reduction in the HC emission level for butanol use was obtained by other researchers [6,21,24,31,32].

The emission level of nitrogen oxides depends on the combustion temperature, the amount of oxygen available in air–fuel mixture and the time allocated for chemical reactions. During the combustion process, the NO_x formation processes develop with finite speed and follow a reactions chain, defined by the Zeldovici mechanism [26]:

$$O + N_2 = NO + N \quad (5)$$

$$NO + N = O + N_2 \quad (6)$$

$$O_2 + N = NO + O \quad (7)$$

$$NO + O = N + O_2 \quad (8)$$

Nitrogen oxides are formed during the first and the third reaction. The first reaction (5) triggers the formation of nitrogen oxides because the atomic oxygen concentration is higher comparative to atomic nitrogen [26]. Nitrogen oxide decomposition is made at low speed and temperature; it conforms to reactions (6) and (8).

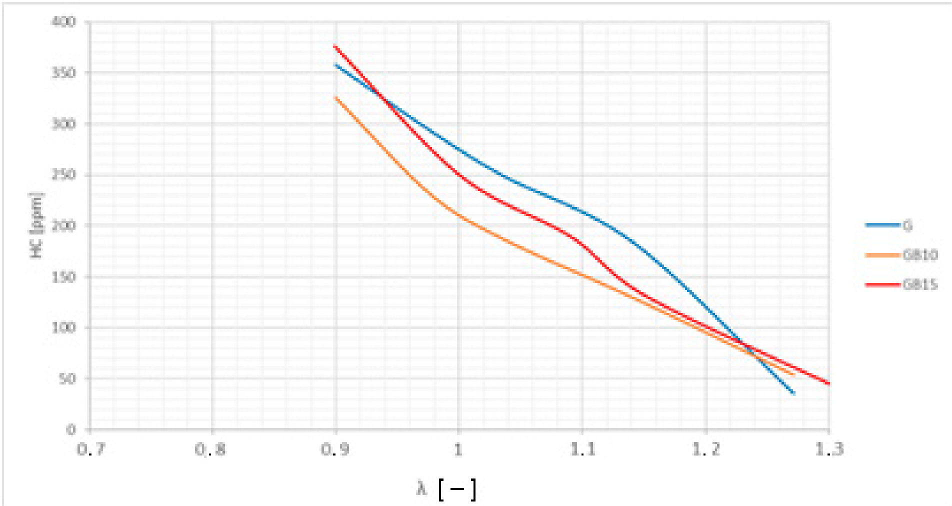


Figure 16. The HC emission levels for G, GB10 and GB15 fuels.

The reaction speed depends on the temperature, thus in the combustion chamber areas appear with different concentrations of nitrogen oxides behind the ignition front. The temperature of the combustion gases changes and the variation of the rate of formation of nitrogen oxides appears [26]. Thus, the variation of the in-cylinder temperature field during the combustion process influences nitrogen oxide formation. Usually, the highest concentrations of nitrogen oxides are found in the spark plug area because of the presence of high temperatures for a longer period. The NO_x emission levels are presented in Figure 17. Butanol use assures a reduction in the NO_x emission level. For rich mixtures, the use of GB10 and GB15 fuels leads to a reduction in the NO_x emission level of 45% comparative to gasoline. For lean mixtures, the use of GB15 fuel leads to a reduction in the NO_x emission level of 32% comparative to gasoline. The use of GB10 fuel leads to a reduction in the NO_x emission level of 66% comparative to gasoline for lean mixtures. Knowing that in general, spark ignition engines operate with coefficients of air excess between 0.97 and 1.03, the advantage of NO_x emission level reduction for lean mixture operation and butanol use appears to be important. Comparing the NO_x emission levels for stoichiometric mixtures for all fueling cases, GB10 fuel use assures a reduction in the NO_x emission level of 72% versus classic fuel use, since the use of the GB15 fuel assures a reduction of 42% comparative to gasoline. The vaporization of the butanol can create a cooling effect inside the in-cylinder air–fuel mixture, which collaborates with a lower flame temperature of butanol and may lead to a reduction in the NO_x emission level. Similar results were obtained by other researchers [6,20,21,23,24,30–32].

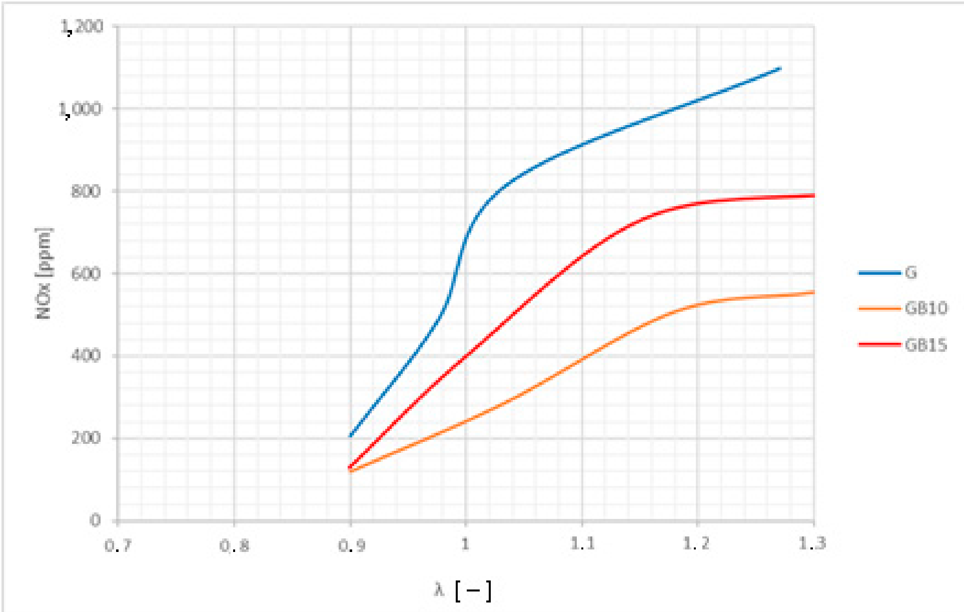


Figure 17. The NO_x emission levels for G, GB10 and GB15 fuels.

4. Conclusions

The blending of n-butanol in 10% vol. and 15% vol. with gasoline provides stable miscible fuel blends named GB10 and GB15, respectively, and their use as fuel for a turbo-supercharged spark ignition engine leads to mixed results, bringing improvements in some areas and some regression in others.

From an engine performance point of view, blending gasoline with 10% vol. n-butanol for rich mixtures, indicates no loss in performance; however, the maximum pressure is achieved sooner during combustion comparative to gasoline and GB15 fuel use, probably due to butanol's higher combustion speed. GB15 fuel use leads to a similar combustion pressure evolution to gasoline use. A 57% increase in maximum pressure variation is noticed at fueling with the GB10 fuel comparative to gasoline for rich mixtures, indicating that gasoline use may lead to a much more stable operation. A similar observation can be made it for GB15 fuel use as the degradation in the COV value was very small at 2.27% comparative to the value calculated for gasoline fueling. For lean mixtures, in some individual combustion cycles, the maximum pressure decreases with the use of the GB10 and GB15 fuels comparative to gasoline use. The use of the GB10 fuel leads to worse performance for lean mixtures, with a decrease of 12% in peak pressure comparative to gasoline; the use of the GB15 fuel leads to a 9% decrease in maximum pressure in some individual cycles. The reduction in the engine performance for the use of the GB10 fuel is compensated by the overall lower COV values comparative to gasoline and the GB15 fuel. Further improvements may be achieved for butanol use if several engine parameters are adjusted, such as spark timing, supercharging pressure, exhaust gas recirculation flow, etc.

For rich mixtures, the combustion process is improved, the isochoric combustion phase appears sooner per cycle for butanol use comparative to gasoline and the mass fraction burned MFB-90% tends to get closer to the TDC. In terms of MFB-90% cyclic variability, for gasoline use the dispersion coefficient is 16.52% while for GB10 and GB15 fuel use the dispersion coefficient is 10.5%. For rich mixtures, the MFB-5% is more stable for gasoline use, with a COV of 9% versus a COV of 14% for the GB10 and GB15 fuels. For lean mixtures, GB10 fuel use leads to an improvement in the cyclic variability of the MFB-5% of 65% and of the MFB-50% of 44%, while for GB15 fuel use the COV values are slightly above the values recorded for gasoline use.

From the point of view of cyclic variability, lean mixtures better support low percentages of butanol (10% butanol is well tolerated for lean mixtures, $\lambda = 1.3$). Rich mixtures more easily allow for the use of higher percentages of butanol (15% butanol is well tolerated for rich mixtures, $\lambda = 0.9$). From the point of view of the positioning of the combustion phases per cycle versus the TDC position, lean mixtures easily accept higher percentages of butanol. The main phase of combustion is accelerated, while the beginning of combustion favors a higher percent of butanol for rich mixtures than lean mixtures. Higher percentages of butanol, such as 15%, stabilize the combustion process comparative to low butanol percentages, such as the GB10 fuel. GB10 fuel use favors the reduction in the combustion duration Δ_{5-90} by up to 10 °CA for lean mixtures.

Regarding the exhaust emission levels, the use of butanol leads to a reduction in the CO and HC emission levels for rich mixtures comparative to gasoline, but similar levels are registered comparative to classic fueling for lean and very lean mixtures. GB10 fuel use leads to a reduction in the CO emission level of 11% and in the HC emission level of 10% comparative to gasoline for rich mixtures. For lean mixtures, the use of GB10 and GB15 fuels leads to a reduction in the HC emission level of 26% comparative to gasoline. The biggest improvement was noticed in the sharp reduction in the NO_x emission level; the use of both butanol-based fuels, GB10 and GB15, leads to a reduction in the NO_x emission level for rich and lean mixtures. For stoichiometric mixtures, the use of the GB10 fuel leads to a reduction in the NO_x emission level of 72% and the use of the GB15 fuel assures a reduction in the NO_x emission level by 42% comparative to gasoline. Lower NO_x emission levels could be explained by lower in-cylinder temperatures during combustion because of n-butanol's reduced LHV and lower combustion flame temperature, which may produce a cooling effect during the vaporization of the air–fuel mixture inside the cylinder.

Regarding the stability of fuel blends, butanol has good miscibility properties, which make it attractive to use blended with gasoline at higher percentages than could ever be possible with ethanol, as its properties are more similar with gasoline.

For leaner mixtures, the use of butanol leads to a reduction in the combustion duration in some individual cycles; the end of combustion appears sooner per cycle and the reproducibility of the main combustion phase is improved. The response of the engine to the cyclic variability of the combustion process is also improved, and the IMEP value dispersion is reduced. Similar trends can be found for rich mixtures, but to a lesser extent.

Butanol can be a viable alternative fuel for spark ignition engines and its use can assure a reduction in CO, HC and NO_x pollutant emission levels and in the CO_2 greenhouse gas emission level comparative to gasoline fueling. An advantage of using butanol is given by the reduction in the NO_x emission level of 27% for GB15 fuel use and 49% for GB10 fuel use, especially for lean mixture operation, in the dosage area $\lambda = 1.1$–1.2 where the NO_x emission level is significant at gasoline fueling. A reduction in the NO_x emission level is achieved in terms of a 3.9% improvement in the efficiency for lean mixture use.

For butanol use, the engine stability is improved at rich and lean dosages, with the butanol blending percent depending on mixture state: rich or lean.

Butanol can be a viable alternative fuel for spark ignition engines and its use does not imply major changes to the engine construction.

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Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

AVL	Anstalt für Verbrennungskraftmaschinen List GmbH- automotive research institute which fabricates the test bed equipment
BSEC	brake-specific energetic consumption
°C	Celsius degree
°CA	crank angle degree
CO	carbon monoxide
CO_2	carbon dioxide
COV	coefficient of variation
NO_x	nitrogen oxides
G	gasoline
GB	gasoline–butanol blend
ECU	electronic control unit
EU	European Union
GHG	greenhouse gas
HC	unburned hydrocarbon
HCCI	homogeneous charge compression ignition
ICE	internal combustion engine
LHV	lower heating value
MFB	mass fraction burned
rpm	revolutions per minute
TDC	top dead center
α	crankshaft angular position
$\Delta_{5\text{-}90}$	combustion duration
λ	coefficient of air excess

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