

THE EFFECT OF OXYGENATED DIESEL-N-BUTANOL FUEL BLENDS ON COMBUSTION, PERFORMANCE, AND EXHAUST EMISSIONS OF A TURBOCHARGED CRDI DIESEL ENGINE

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ABSTRACT

The article deals with the effects made by using various n-butanol-diesel fuel blends on the combustion history, engine performance and exhaust emissions of a turbocharged four-stroke, four-cylinder, CRDI 1154HP (85 kW) diesel engine. At first, load characteristics were taken when running an engine with normal diesel fuel (DF) to have 'baseline' parameters at the two ranges of speed of 1800 and 2500 rpm. Four fossil diesel (class 1) and normal butanol (n-butanol) fuel blends possessing 1 wt%, 2 wt%, 3 wt%, and 4 wt% (by mass) of n-butanol-bound oxygen fractions were prepared by pouring 4.65 wt% (BD1), 9.30 wt% (BD2), 13.95 wt% (BD3), and 18.65 wt% (BD4) n-butanol to diesel fuel. Then, load characteristics were taken when an engine with n-butanol-oxygenated fuel blends at the same speeds. Analysis of the changes occurred in the autoignition delay, combustion history, the cycle-to-cycle variation, engine efficiency, smoke, and exhaust emissions NO_x, CO, THC obtained with purposely designed fuel blends was performed on comparative bases with the corresponding values measured with 'baseline' diesel fuel to reveal the potential developing trends.

Keywords: Diesel engine; Diesel-n-butanol blends; Combustion phenomenon; Performance efficiency; Smoke; Pollutant emissions

INTRODUCTION

Growth of the economy and living standards of the population increase the need of people's and goods transportation by sea, railroads, the air, and highways. As a result, increases the need to use more ships, trains, airplanes, heavy-duty trucks, self-powered machines, city busses, and light-duty passenger cars. Unavoidably increases the demand of the fuel to be consumed, however the natural oil-resources are largely exhausted over hundreds of years. Moreover, the increased consumption of a fossil-origin fuel creates the urgent environment pollution problems and climate change. The automotive air-pollution problem emerged already in the 1940s in the Los Angeles basin [8]. The first

Clean Air Act enacted by Congress of the United States on July 14, 1955 identified the environmental air pollution related problems [17]. The Air Pollution Control Act documented the dangers to the public health and welfare, injury to agricultural crops and livestock, damage and deterioration of property, and hazards to the air, marine, and ground transport. This perception was the turning point at which the industrial, agricultural, and transportation infrastructure started to move towards green energy policy.

A legacy of the 20th century we still have today and the climate changes even more intensively over the last decades. Fulfilling historical agreement on curbing the climate change achieved in Paris (2015), the 195 Countries have made an important progress on the control of automotive emission and

set up increasingly strict emission regulations to reduce the human-made harm for the environment. Detailed analysis of the reasons leading to climate change, negotiations on reduction of carbon emissions and development of solutions for a gradual switch to renewable energy systems provides review [5]. Many of the research projects have been done to replace a fossil diesel fuel with renewable fuels such as neat oil derived from biomass of various origins, biodiesel, and ethanol. Using of renewable, oxygenated and sulphur-free fuels in internal combustion engines in a neat form or in blends with traditional diesel fuel saves fossil-fuel energy resources and makes less harm to the environment. 1-Butanol, also known as n-butanol, has a straight-chain structure with a hydroxyl group at the terminal carbon, and it has wide industrial applications. Butanol has attracted attention as an alternative biofuel because of its various advantages over other biofuels. The advances in butanol production were recently introduced to improve genetic modification of butanol-producing strains, and fermentation technology of butanol [27].

The influence of the addition of n-butanol on fuel properties, combustion, and emission characteristics for a diesel engine was studied and the engine performance and emissions were tested under nine loads at speed of 1200 rpm. The results showed that kinematic viscosity, net heating value, closed-cup flash point, density, and cetane number presented downward trends of varying degree with the increase in n-butanol content. Hydrocarbon (HC), carbon monoxide (CO), and particulate matter emissions decreased, while nitrogen oxides (NO_x) emissions and fuel consumption increased [25]. Lapuerta et al. [14] demonstrated that the increase of alcohol content in diesel or biodiesel, led to an increase in autoignition times. Such increase was not a linear but sharper for high alcohol contents, and was higher with ethanol than with butanol. For butanol blends, the increase in delay time was very similar when diesel or biodiesel were used. The maximum combustion pressure decreases as the alcohol content was increased, especially in the case of ethanol, as consequence of energy, chemical and dilution effects [14, 22]. The experiments of four-cylinder, four-stroke, turbocharged, CR diesel engine running with diesel and n-butanol blend with a volume ratio of 70:30 (D70B30) and a blend of diesel and gasoline with a volume ratio of 70:30 (D70G30) showed that the D70B30 blended fuel soot emission reduced more at the EGR ratio smaller than 25% and a constant speed of 1600 rpm. [10].

Normal butanol (CH₃CH₂CH₂CH₂OH – 74.0 g/mol) is an alcoholic-origin colourless liquid with a harsh fusel with banana odour, which possess 21.62 wt% of fuel-bound oxygen and differs as having low flash point of 28.9 °C and the boiling point of 117.7 °C at 760 mm Hg. On the one part, n-butanol added to commercial diesel fuel reduces density, viscosity, C/H atoms ratio and provides fuel-bound oxygen that along with good evaporative properties improves both the air and fuel vapours mixing rate and the combustion of fuel blend leading to more environment friendly exhaust. On the other part, the presence of a lighter and oxygenated

n-butanol reduces the cetane number, net heating value and thus mass of the fuel consumed per unit of energy developed by an engine. The sensitive interaction between advantages and disadvantages properties of n-butanol added to diesel fuel may lead to ambiguous development trends combustion, heat release rate and engine out emissions.

The purpose was to study the effects of oxygenated diesel-n-butanol fuel blends on the auto-ignition delay, combustion history, maximum heat release rate, burn angles MBF 50, MBF 90, brake specific fuel consumption, engine efficiency, smoke, and NO_x, CO, THC emissions of a turbocharged CRDI diesel engine running at various loads (bmep) and speeds of 1800 and 2500 rpm.

EXPERIMENTAL ENGINE TEST SET UP AND RESEARCH METHODOLOGY

A four-cylinder, turbocharged, CRDI diesel engine FIAT 1.9 JTD 8V 115 HP (85 kW) with a swept volume of 1.91 dm³ and compression ratio of 18:1 was used for the experimental tests. The uncooled air entered the capacity chamber and the cylinder at a controllable boost pressure of 0.160 MPa and the temperature of 85 °C. The EGR system was switched off to eliminate the potential side effects on the engine performance. Whereas the electronic control unit EDC-15C7 CR governed the timing and the duration of the fuel injection. The test setup contained a diesel engine, an engine test bed, the AVL indicating system, air and fuel mass consumption measuring equipment, a gas analyser, and a smoke meter as shown in Fig. 1. Specifications of an engine and fuel injection system are listed in Table 2.

Load characteristics with a fossil diesel fuel (DF) EN 590 (class 1) as a 'baseline' fuel and its 95.375/4.625 wt% (DB1), 90.749/9.251 wt% (DB2), 86.124/13.876 wt% (DB3) and 81.499/18.501 wt% (DB4) blends with n-butanol (B) components were taken at speeds of 1800 rpm and 2500 rpm. Changes in combustion, heat release characteristics, engine performance, smoke, and exhaust emissions revealed when using fuel blends DB1, DB2, DB3 and DB4 involving 1 wt%, 2 wt%, 3 wt% and 4 wt% of butanol-oxygen were compared with the respective values the reference diesel fuel develops for various brake mean effective pressures (bmep) at engine speeds of 1800 and 2500 rpm.

Tab. 1. Engine specifications

Engine code	192A1000
Engine brand / model	FIAT 1.9JTD 8v 115 HP (85kW)
Engine type	Four-cylinder engine, in line, turbocharged, JTD
Turbine code	712766-1
Turbocharger	A variable geometry Garret GT1749V
Fuel injection system	Common rail, direct injection (DI)
Cylinder bore x piston stroke	82 mm x 90.4 mm
The length of connecting rod	144.5 mm
Total displacement volume	1910 cc
Compression ratio	18.0±0.45:1

Combustion chamber	The OMEGA-shaped in a piston head
Rated power	85 kW (115 HP) at 4000 rpm (bmep = 1.335 MPa)
Maximum torque	255 Nm (EEC) at speed of 2000 rpm
Idle speed	850±20 rpm
SOHC – Intake timing angles	Opens before TDC at 0°; closes after BDC at 32°
SOHC – Exhaust timing angles	Opens before BDC at 40°; closes after TDC at 2°
The gas exchange system	OHC with the four vertical valves per cylinder
Fuel injection system	Common rail, Bosch
Maximum injection pressure	1400 bar (140±0.5 MPa)
Codes injection pump / injectors	0445010007 / 0445110119 – with 6 injection holes

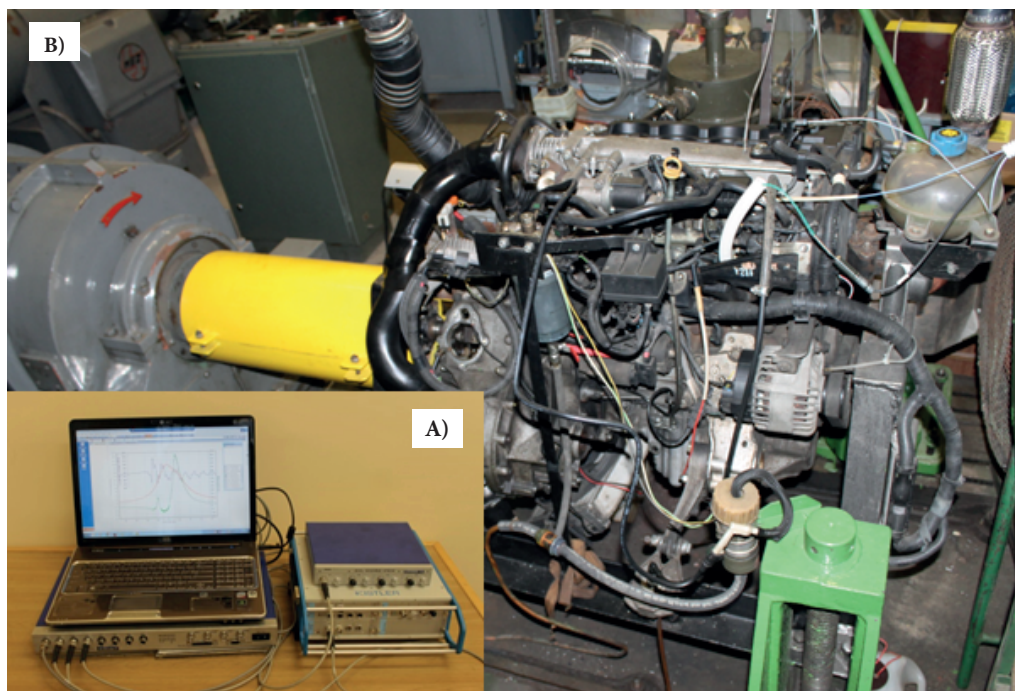


Fig. 1. Picture of the arrangement of the engine test stand: A) Measuring and recording apparatus; B) Research engine Fiat 1.9JTD 8v

A high-speed indicating system, which included the AVL angle encoder 365C and pressure transducer GU24D coupled to the AVL microIFEM piezoelectric amplifier and signal acquisition platform IndiModul 622, was introduced for the recording, acquisition, and processing of fast crank-angle gas pressure signals in the first cylinder. The single indicator diagrams, which reflected in-cylinder individual pressure signals over 100 cycles versus crank angle, were in series recorded for each fuel blend at every load-speed setting point of an engine. The data post-processing Software AVL CONCERTO™ advanced version 4.5 was used to increase productivity and improve measurement accuracy of the test results. The net heat release rate was calculated by using the AVL BOOST program. Summarized over the 100 engine-cycles in-cylinder pressure-data, instantaneous cylinder volume, and their first order derivative with respect

to crank angle have been used to calculate the heat release characteristics.

The engine torque was measured with an electric dynamometer KS-56-4 with a definition rate of ±1 Nm, and the speed with the AVL crank angle encoder 365C. A real-time air mass flow into the cylinders was measured with the AVL air mass flow meter and fuel mass consumption were recorded for every load-speed setting point with the AVL dynamic fuel balance 733S flex-fuel system. The fuel system was cleaned up after every test set and the engine operated for about 15 min intervals between test-series to consume the fuel remaining from the previous test and reach steady-state coolant temperature of 88–90 °C.

The start of injection (SOI) was recorded by using the Kistler piezoelectric pressure sensor ASMB 470004-1 connected on a high-pressure tube in front of the injector. The pressure sensor was coupled to the Kistler 2-channel charge amplifier-module 4665 mounted on the signals conditioning platform-compact 2854A to record high-pressure history at the injector with an accuracy of ±0.5% in the pressure range of 0–200 MPa.

The autoignition delay determined as a period in CADs between the start of injection (SOI) and the start of combustion (SOC) with an accuracy of ±0.1°. As the start of injection was taken crank angle, at which the fuel pressure in a high-pressure tube drops temporally down due to the opening of the nozzle-needle-valve of the injector. As the start of combustion taken crank angle, at which the total heat release-rate crosses the zero line and changes its value from the minus side to the plus side. The definition of the autoignition delay time is shown in graphs of Fig. 2.

Using the data post-processing software AVL CONCERTO™ advanced edition provided higher productivity and accuracy of the test results. The crank angles values of MBF 50 and MBF 90 for 50% and 90% of the mass burned fraction, net heat release versus crank angle, the total amount of heat released over an engine cycle, and heat losses to the cooling system were determined by using the AVL BOOST simulation software to improve accuracy of the calculated results. The accuracies of the measured experimental data of the combustion process and engine performance parameters and the uncertainties of the calculated test results are listed in Table 2.

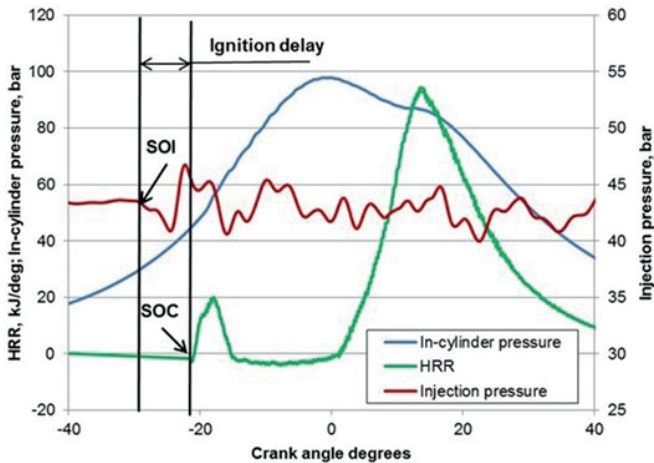


Fig. 2. Definition of the autoignition delay: the period expressed in crank angle degrees (CADs) between start of injection (SOI) and start of combustion (SOC).

Tab. 2. The accuracy of the measured engine performance and emission parameters and the uncertainty of the computed experimental results

Measurements	Accuracy
Engine torque	±1%
Engine speed	±0.1%
In-cylinder gas pressure	±0.1 MPa
Start of injection (SOI)	±0.1° (CADs)
Start of combustion (SOC)	±0.1° (CADs)
Calculated results	Total uncertainty
Engine power output	±1 %
Air mass flow rate	±1 %
Fuel mass flow rate	±0.1 %
Brake specific fuel consumption	±1.5 %
Brake thermal efficiency	±1.5 %

ANALYSIS OF MAIN PARAMETERS OF THE TESTED FUEL BLENDS

Conventional automotive fossil-origin diesel fuel (class 1) was produced at the oil refinery “Orlen Lietuva” and it satisfied the requirements of standard EN-590:2009+A1. Its composition consisted of C/H = 0.8608/0.1299 and the remaining 0.0093 was water, sulphur, and other impurities as determined at the refinery’s laboratory. The n-butanol (CH₃CH₂CH₂CH₂OH) was produced in Germany (Seelze) at Ltd. „Sigma-Aldrich” and satisfied the requirements of specification 1.00988.6025 1-Butanol EMPROVE® ESSENTIAL NF. At the respective producers’ laboratories, properties of the diesel fuel and n-butanol were examined by using the EU standards. These properties were approved with the certificates of quality analysis. Molecular weight of diesel fuel is about 180 [13] and that of 74 for n-butanol. Kinematic viscosity of a fossil diesel fuel (class 1) is 2.13 mm²/s at 40 °C whereas that of n-butanol compiles 2.97 mm²/s at 20 °C. Lubricity, HFRR VSD at 60 °C of n-butanol is lesser

(622 μm) than that (459 μm) of diesel fuel, but small amounts of n-butanol added to diesel fuel do not provide a risk to the reliability of the fuel system. More about properties of a fossil diesel, n-butanol and diesel-n-butanol fuel blends can be found in Table 1.

Tab. 3. Basic properties of a fossil diesel fuel, n-butanol and diesel-n-butanol fuel blends

Property parameters	Diesel fuel (class 1) C13H24	N-butanol (CH ₃ CH ₂ CH ₂ CH ₂ OH)	Diesel-n-butanol DB1	Diesel-n-butanol DB2	Diesel-n-butanol DB3	Diesel-n-butanol DB4
Oxygen, max wt%	0.00	21.62	1.0	2.0	3.0	4.0
Percentage of butanol in blend, wt%	-	-	4.625	9.251	13.876	18.501
Density at 20 °C, kg/m ³	832.7	811.0	831.7	830.7	829.7	828.7
Cetane number	51.4	25.0	50.2	49.0	47.7	46.5
Carbon, max wt%	86.67	64.87	85.66	84.65	83.65	82.64
Hydrogen, wt%	13.33	13.51	13.34	13.35	13.35	13.36
Carbon-to-hydrogen ratio (C/H)	6.50	4.80	6.42	6.34	6.27	6.19
Stoichiometric air-fuel ratio, kg/kg	14.685	12.220	14.53	14.37	14.21	14.05
Net heating value, kJ/kg	43.00	33.10	42.54	42.08	41.63	41.17

The engine tests started with conventional diesel fuel (class 1) to determine the combustion, performance and emissions characteristics constituting the “baseline” level that was used for monitoring of developing trends of parameters measured when operating with diesel-n-butanol fuel blends involving 1.0, 2.0, 3.0 and 4.0 wt% of n-butanol-oxygen at the same test conditions. Stable fossil diesel and n-butanol oxygenated fuel blends DB1, DB2, dB3 and DB 4 were premixed in the pre-set proportions by mass without using of any co-solvents because n-butanol mixes properly with diesel fuel and does not show any signs of phase separation at the temperature of 20 °C.

Renewable n-butanol or n-butyl alcohol or normal butanol is a primary alcohol with a 4-carbon structure and the chemical formula C₄H₉OH. It is built up from straight chain hydrocarbons, contains 21.62 wt% of fuel-bound oxygen and practically free of aromatics, nitrogen, and sulphur and possess the cetane numbers value of 25.0 that is threefold higher than that of ethanol (Table 3). Initial/final boiling points of diesel fuel (177.8 / 345.0 °C) are higher than a single boiling point of 117.7 °C of n-butanol. This means that n-butanol added to diesel fuel contributes to faster evaporation and enhances mixing rate of the air and fuel vapours and thus homogeneity of combustible mixture. Butanol is miscible with hydrocarbons therefore it mixes well with the ‘baseline’ diesel fuel too [10]. After diesel-n-butanol

fuel blends were prepared, the canisters were splashed for a while to avoid phase separation.

The density and kinematic viscosity of fuel blends were measured by using the laboratory device Anton Paar density/viscosity meter SVM 3000 with an accuracy of $\pm 0.0002 \text{ g/cm}^3$ and 0.1%, respectively, at the temperature of $40 \pm 0.001 \text{ }^\circ\text{C}$. Whereas the cetane number of oxygenated diesel-n-butanol fuel blends was computed by using typical methodology developed in the U.S. at the National Renewable Energy Laboratory [26]. This methodology assumes that the cetane number of obtained blend is a linear combination of the cetane numbers of the original fuel-components in its composition. The added n-butanol portions to diesel fuel the cetane number of the tested fuel blends reduced by 2.3%, 4.7%, 7.2%, and 9.5%, respectively. This data will be used for comparative analysis and qualitative evaluation of the relative changes occurred in the combustion process, engine efficiency, and exhaust emissions.

RESULTS AND DISCUSSION

Fig. 3 shows how oxygenated diesel-n-butanol fuel blends affects the autoignition delay when running under various engine loads (bmep) and the two ranges of speed of 1800 and 2500 rpm. At first, it should be noted that the start of injection (SOI) of a pilot fuel portion was under computer control to be automatically advanced by 15.0° and 6.0° CADs with the engine load (bmep) increased within the tested ranges at the respective speeds. This means that the SOI occurred earlier in the cycle BTDC to ensure well-timed autoignition and complete combustion of more fuel injected per cycle needed to develop higher load. This is the answer to the question why the autoignition delay progressively increased with increasing engine load, - because the oxidation reactions of the fuel started at lower both pressure and temperature inside the cylinder. However, the research did not intend to examine the changes occurred in engine load or speed, since the investigation was focused mainly on the comparative analysis of the fuel properties-evoked and n-butanol-oxygen-made effects on the ignition delay and following combustion processes.

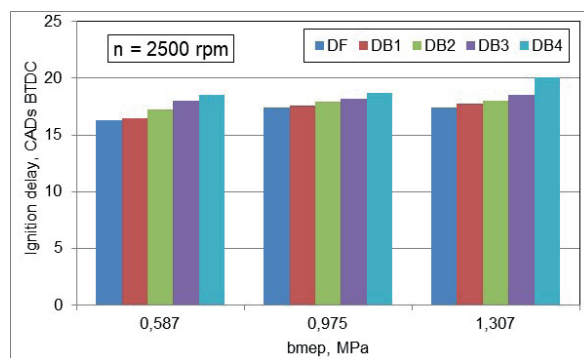
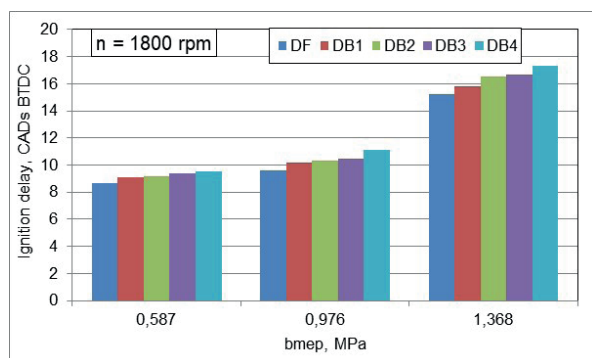


Fig. 3. The autoignition delay period in CADs as a function of engine load (bmep) at speeds of 1800 and 2500 rpm.

The autoignition delay time of diesel-n-butanol fuel blends DB1-DB4 increased by 4.6-9.2%, 7.4-16.8% and 3.9-13.8% against those values, 8.7° , 9.5° and 15.2° CADs, normal diesel fuel (DF) auto-ignites under respective 0.587, 0.978 and 1.368 MPa loads at the low speed of 1800 rpm. This pre-combustion period also increased by 1.2-13.5%, 1.1-7.5% and 2.3-15.5% when running with diesel-n-butanol fuel blends DB1-DB4 against, 16.3° , 17.4° and 17.4° CADs, diesel fuel auto-ignites under respective 0.587, 0.975 and 1.307 MPa loads at a higher speed of 2500 rpm. The obtained results show that the autoignition delay was always longer when running a turbocharged CRDI diesel engine with n-butanol-oxygenated fuel blends than with normal oxygen-free (class 1) diesel fuel. The longer auto-ignition delay for diesel-n-butanol fuel blends DB1, DB2, DB3 and DB4 can be directly attributed to the lower cetane number of the tested blends as compared to normal diesel fuel (Table 3). The next reason why the auto-ignition delay of diesel-n-butanol fuel blends was longer can be attributed to latent heat of vaporisation of n-butanol, 581.4 kJ/kg , being 2.15 times higher than, 270 kJ/kg , of diesel fuel. Finally, despite of a higher volatility, the auto-ignition temperature of alcoholic origin n-butanol is about $385 \text{ }^\circ\text{C}$, which is much higher than that, $250 \text{ }^\circ\text{C}$, of a fossil diesel fuel.

The ignition delay period was also longer when running a naturally aspirated diesel engine D-243 with various ethanol-diesel-biodiesel blends under 'lambda' of 5.5, 3.0 and 1.5 at speeds of 1400, 1800 and 2200 rpm [13]. The negative impact on the ignition delay time of using oxygenated diesel-HRD fuel blends involving ethanol or biodiesel was also observed when running a turbocharged CRDI diesel engine FIAT 1.9JTD 8v 115 HP (85kW) under 'lambda' of 1.30, 1.25 and 1.20 at the respective speeds of 1500, 2000 and 2500 rpm [6]. To having longer auto-ignition delay of oxygenated fuel blends perhaps contributed n-butanol-bound oxygen the presence of which reduced net heating value of fuel blends. In result, the higher was engine load (bmep) and, hence, more fuel injected per cycle, the greater was evaporative cooling effect. This resulted in the difference in auto-ignition delay periods between the combustion of a 'baseline' diesel fuel and oxygenated diesel-n-butanol fuel blends being more perceptible when running with the most oxygenated fuel

blend DB4 (4 wt% oxygen) at both engine speeds. Besides, the relative increase in the ignition delay period was greater when running under both the lowest and the highest loads at high speed of 2500 rpm, because the effect of cetane number is always greater if time needed to preheat, evaporate of the fuel droplets, and auto-ignite of air-fuel vapours is extremely limited.

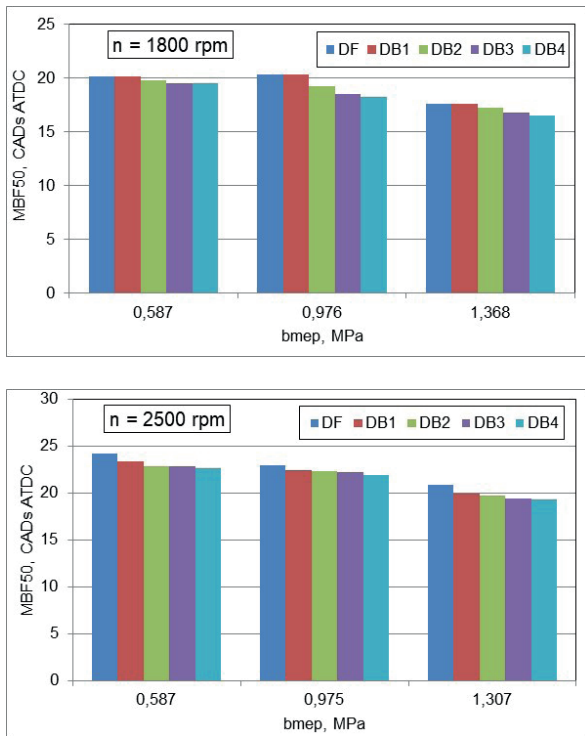


Fig. 4. Burn angle MBF 50 as a function of engine load (*bmep*) when running with oxygenated diesel-*n*-butanol fuel blends under various loads (*bmep*) and speeds of 1800 and 2500 rpm.

It is important to evaluate changes in burn angle MBF 50 caused by the combustion of various diesel-*n*-butanol fuel blends because this angle represents the centre of a gravity of heat release rate (HRR) characteristic and, thus, affects fuel-energy conversion efficiency of an engine. The shorter is the crank angle ATDC at which the 50% of energy releases in the cycle, the lower will be heat losses of the expansion stroke and thus higher engine efficiency can be attained. Nevertheless, angle MBF 50 should be neither too short, nor too long, but just as needed to harmonise sensitive interaction between the engine efficiency and the maximum torque (load) developed. Whereas the burn angle $AHRR_{max}$ should be approximately equal to 8° CADs ATDC as recommended by the AVL User's Guide instruction [2]. Burn angle MBF 50 decreased by 0.0–3.2%, 0.0–10.5%, 0.0–6.5% and 3.2–6.3%, 2.5–4.6%, 4.7–7.6% when running with oxygenated fuel blends DB1-DB4 against those values of 20.2° , 20.4° , 17.7° and 24.2° , 23.0° , 20.9° CADs the 50% of diesel fuel gets burned under the respective loads and engine speeds of 1800 and 2500 rpm (Fig. 4).

Again, the lover was the cetane number of fuel blends and, thus, the longer the ignition delay time available to improve the quality of air-fuel mixture, the higher was maximum heat release rate and the sooner in the cycle 50% of fuel-heat

releases from combustion of the most oxygenated fuel blend DB4 for the respective engine loads and speeds. To enhance the oxidation reactions, the *n*-butanol-based oxygen content should be progressively increased, 1, 2, 3 and 4 wt%, with increasing engine load to accelerate heat release rate against that the combustion of diesel fuel suggests for similar engine loads. Thus, burn angle MBF 50 can be shortened by 10.5% and 7.6% when running with the most oxygenated diesel-*n*-butanol fuel blend DB4 under medium load, 0.976 MPa, at speed of 1800 rpm and the high load, 1.307 MPa, at speed of 2500 rpm, respectively.

When running with a 'baseline' diesel fuel the end of combustion took place 4.2° (8.4%), 7.9° (15.8%) and 3.0° (5.6%), 8.1° (15.3%) CADs later in the cycle compared with that, 50.1° and 53.1° CADs, due to engine load increased from 0.587 MPa to 0.978, 1.368 MPa and from 0.587 MPa to 0.975, 1.307 MPa at the respective speeds of 1800 and 2500 rpm. The burn angle MBF 90 increased because burning of a bigger fuel portion delivered per cycle lasts always longer due to the reduced availability of air-born oxygen needed to complete combustion. However, a single-bound *n*-butanol-oxygen accelerated the combustion reactions to make a difference between oxygen-free diesel fuel and oxygenated diesel-*n*-butanol blends. Burn angle MBF 90 representing the end of combustion decreased by 1.4–3.4%, 3.7–30.2%, 2.1–26.7% and 0.0–1.5%, 2.0–2.1%, 1.0–6.5% when running with oxygenated fuel blends DB1-DB4 against those values of 50.1° , 54.3° , 58.0° and 53.1° , 56.1° , 61.2° CADs the combustion of diesel fuel ends up for the respective loads and speeds (Fig. 5).

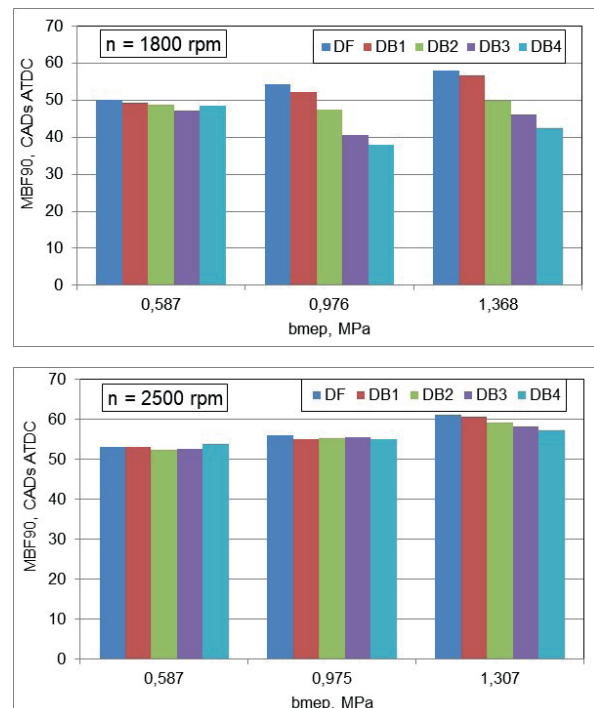


Fig. 5. Burn angle MBF 90 as a function of engine load (*bmep*) when running with oxygenated diesel-*n*-butanol fuel blends under various loads (*bmep*) and speeds of 1800 and 2500 rpm.

The end of combustion occurred by 2.7° – 12.2° and 0.2° – 7.6° CADs earlier in the cycle ATDC when running with oxygenated diesel-n-butanol fuel blends DB2-DB4 under medium (0.976 MPa) and high (1.368 MPa) loads than, 50.1° CADs, the combustion of diesel fuel proceeds under light (0.587 MPa) load and speed of 1800 rpm. This means that the combustion reactions of richer air-diesel-n-butanol fuel mixtures advanced with a higher average flame speed than normal air-diesel fuel mixture burns under easy loading conditions. Positive changes in the end of combustion of diesel-n-butanol fuel blends can be attributed to the longer auto-ignition delay (Fig. 3) that provides more time needed to improve the homogeneity of combustible mixture. In result, oxygenated air-diesel-n-butanol fuel mixtures burned with a higher flame speed than a straight diesel affords to cope with a problem at the respective loads and speeds. Next, n-butanol-oxygen contributed with an essential help accelerating the combustion reactions mainly at the low speed of 1800 rpm because of a lower turbulence intensity, swirl, and temperature inside the cylinder. Whereas the role of n-butanol-oxygen at the end of combustion seems to be less significant when running at the high speed of 2500 rpm.

The positive effect of using n-butanol-oxygen on combustion reactions decreased significantly when the turbulence intensity, swirl, and temperature inside the cylinder increased at a higher engine speed of 2500 rpm. Nevertheless, the positive development trends in the end of combustion remained in value when running under high load and speed too because of extremely limited real time to accomplish the cycle (Fig. 5). Comparison of data in Figs. 4 and 5 shows that the end of combustion representing angle MBF 90 is more sensitive to changes in engine load, speed, and n-butanol-oxygen content than angle MBF 50, which represents a half (50%) fuel portion burned in the cylinder. This is because the oxidation reactions decrease for the end of combustion due to limited availability of air-borne oxygen and a lower mixing rate of the air and fuel blends at the late stages of the expansion stroke. Because the air-borne oxygen was about utilised for the end of combustion there emerged a chance for fuel-oxygen to show the own potentials in accelerating combustion.

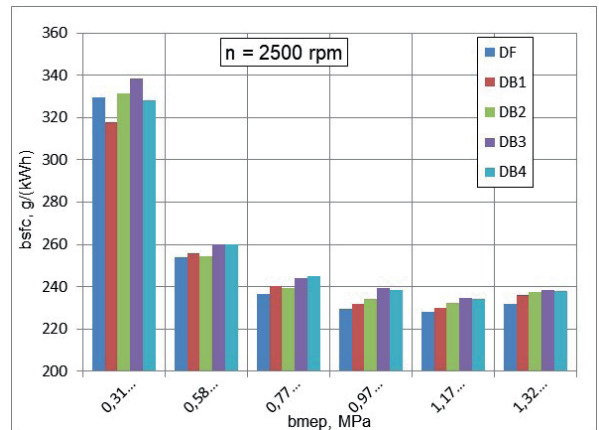
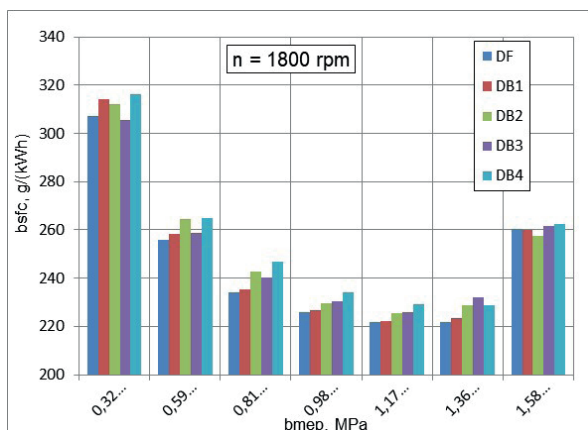


Fig. 6. The brake specific fuel consumption (bsfc) as a function of engine load (bmep) when using various diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm.

Columns of the brake specific fuel consumption (bsfc) in g/kWh as a function of engine load (bmep) obtained during operation with diesel fuel and diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm have been superimposed as shown in Fig. 6. As obvious from the figures, the bsfc decreased with the increasing load to the value that depends on the engine load, speed, and fuel blend used sustaining at the highest level for the most oxygenated and thus less calorific blend DB4 (4.0 wt% oxygen). The lowest brake specific fuel consumption of 221.6 g/kWh was obtained for diesel fuel and 223.3 g/kWh (0.8% higher) for slightly oxygenated (1.0 wt%) diesel-n-butanol blend DB1 when running under rational load of $bmep = 1.1362$ MPa at speed of 1800 rpm. The bsfc gradually increased to 228.8 (3.2%), 231.9 (4.6%) and 228.9 g/kWh (3.3%) against the respective value of diesel fuel due to lower heating value of oxygenated fuel blends DB2, DB3 and DB4. The amount of fuel blend DB2 (2.0 wt% oxygen) consumed per unit of energy developed increased with the lowest increment rate of 14.1% reaching 257.5 g/kWh against that of 260.5 g/kWh of diesel fuel needed to maintain the highest load of $bmep = 1.590$ MPa at the low speed of 1800 rpm. Further air-diesel-n-butanol mixture enrichment, and particularly air-diesel fuel mixture, was limited by critically high levels of smoke (Fig. 11).

The noted bsfc development trends with engine load remained in value when running with diesel fuel and oxygenated diesel-n-butanol fuel blends under various loads and speed of 2500 rpm. Again, the bsfc increased to 230.0 (0.9%), 232.3 (1.9%), 234.5 (2.9%), and 234.4 g/kWh (2.8%) for oxygenated fuel blends DB1, DB2, DB3, and DB4 against that, 228.0 g/kWh, a straight diesel consumes to develop reasonable load of $bmep = 1.174$ MPa at considered revolutions. However, the brake specific fuel consumption increased to 236.1 (1.8%), 237.7 (2.5%), 238.2 (2.7%), and 237.9 (2.5%) for the respective fuel blends DB1, DB2, DB3, and DB4 against that of 232.0 g/kWh a straight diesel consumes to maintain the highest load of $bmep = 1.320$ MPa at high speed of 2500 rpm. Specific fuel consumption increased mainly due to lower net heating value of oxygenated fuel blends (Table 3). Nevertheless, the revealed

tendency to reduce specific fuel consumption by using the most oxygenated (4.0 wt%) fuel blend DB4 was profitably utilised when running at the high engine load and speed. This was because the increased auto-ignition delay time (Fig. 3) provided advantages to improve quality in preparing air-fuel mixture, then the oxidation reactions proceed faster (Figs. 4) and the combustion ended up earlier in the cycle (Fig. 5) that enhanced engine efficiency (Fig. 7), reduced CO (Fig. 9), THC emissions (Fig. 10) and smoke of the exhaust (Fig. 11).

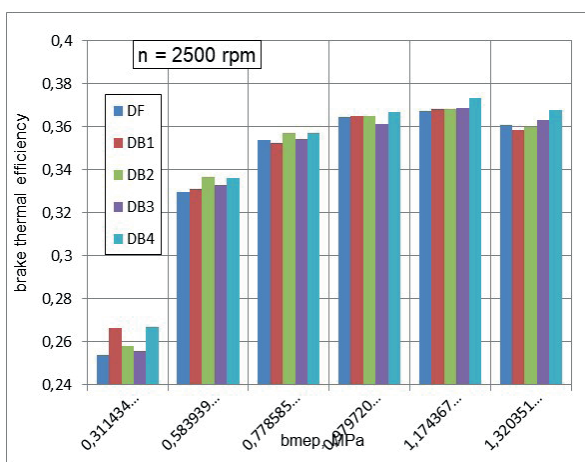
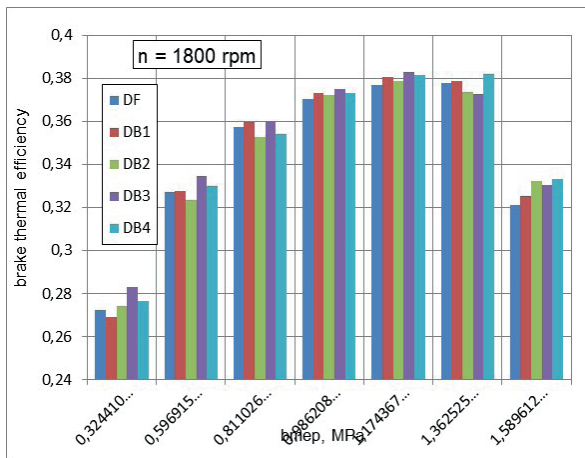


Fig. 7. The brake thermal efficiency (bte) as a function of engine load (bmep) when using various diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm.

The brake specific fuel consumption and net heating value of each fuel blend were evaluated to calculate brake thermal efficiency of an engine. As columns in Fig. 7 show, the engine efficiency progressively increased with increasing load reaching the highest brake thermal efficiency of 0.383 for diesel-n-butanol fuel blend DB3 (3.0 wt% oxygen) and 0.377 for a straight diesel running at $bmep = 1.174$ MPa and speed of 1800 rpm (Fig. 7). As engine load further increased to maximum of $bmep = 1.363$ and 1.590 MPa, the leader's position took over the most oxygenated blend DB4 (4.0 wt%) suggesting the best engine efficiency of 0.382 (1.0%) and 0.333 (3.7% higher) than the respective values, 0.378 and 0.321, a straight diesel produces from combustion

of critically enriched air-fuel mixture. It is worth to note that no decrease in engine efficiency was observed and fuel consumption increased proportionally to its lower heating value when diesel-n-butanol fuel blends up to 20% (by volume) were used in a Euro 6 engine following the New European Driving Cycle [15].

It is obvious from figures that the brake thermal efficiency depends on engine load, speed and the fuel used. At a higher speed of 2500 rpm, the need for fuel-bound oxygen increased with load, therefore the highest efficiency values produced the combustion of fuel blends DB1 (0.266), DB2 (0.337, 0.357), and the most oxygenated DB4 (0.367, 0.373, 0.368). The highest efficiency values from combustion of diesel-n-butanol fuel blend DB4 (4.0 wt% oxygen) were 0.8%, 1.6% and 1.9% higher than, 0.364, 0.367 and 0.361, a straight diesel develops for the respective loads of $bmep = 0.980$, 1.174 and 1.320 MPa. The obtained results proved that there does not exist an exceptional blend using of which could ensure the best possible engine efficiency within wide ranges of loads and speeds. The matter is that the amount of fuel-bound oxygen needed for complete combustion depends on load and thus availability of air-born oxygen as well as engine speed, turbulence intensity, swirl, and temperature inside the cylinder. The bigger is a lack of air-born oxygen inside the cylinder, the greater the need for the fuel-bound oxygen to burn the fuel completely and ensure engine efficiency.

The NO_x production depends on the ignition delay time and the amount of the fuel premixed for rapid combustion, maximum heat release rate, pressure inside the cylinder, and adiabatic flame temperature [4]. Analysis of biodiesel and second-generation biofuels showed that the NO_x emissions have trends to increase with increasing biodiesel blend level, but the magnitude of this effect differs for different feedstocks, engines, and cycles [7]. A review of the effects of biodiesel on NO_x emissions shows that the NO_x increased with increasing unsaturation, but they decrease with increasing chain length [9]. Heterogeneous air-fuel mixture contributes to uneven temperature distribution in the cylinder and thus stimulates the formation of NO , which always dominates in NO_x production [8]. In this case, the changes in NO_x emissions are largely determined by sensitive interaction between the longer auto-ignition delays (Fig. 3) and the shorter combustion durations specified by burn angle MBF 90 (Fig. 5). For similar reasons, the NO_x emissions (in g/km) remained unchanged with n-butanol addition to diesel fuel up to 20% (by volume) when running a Euro 6 diesel engine in the New European Driving Cycle [15].

As columns in Fig. 8 show, the temperature related NO_x production increased with engine load and was higher when running with diesel fuel and all fuel blends tested at speed of 1800 rpm. The combustion of oxygenated diesel-n-butanol fuel blend DB2 (2.0 wt%) generated NO_x emissions 5.2%, 3.4%, 4.8%, 6.3%, 7.9%, 5.1% and 3.7% more than a straight diesel produces, 225, 415, 583, 828, 1207, 1438 and 970 ppm, for respective loads at the low speed of 1800 rpm. Despite of a higher brake thermal efficiency (Fig. 7), the combustion of oxygenated blends DB3 and DB4 produced almost always

NO_x emissions less than blend DB2 does because of a shorter burn angle MBF 90 (Fig. 5) and thus reduced residence time left for the combustion reactions [23]. The NO_x emissions reached the lowest level of 955 ppm when running under the highest load of bmep = 1.590 MPa with the most oxygenated blend DB 4 (4.0 wt%) at the low speed of 1800 rpm.

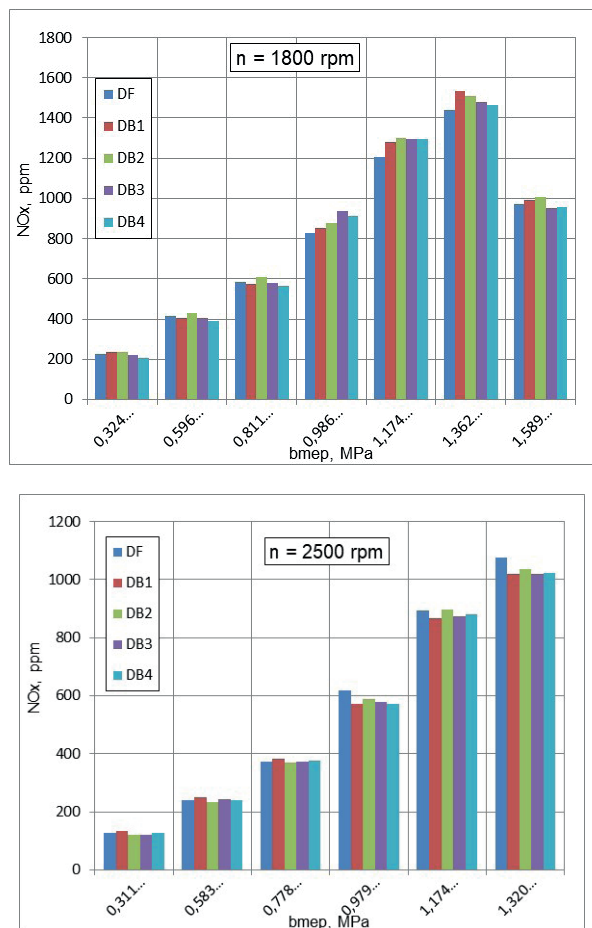
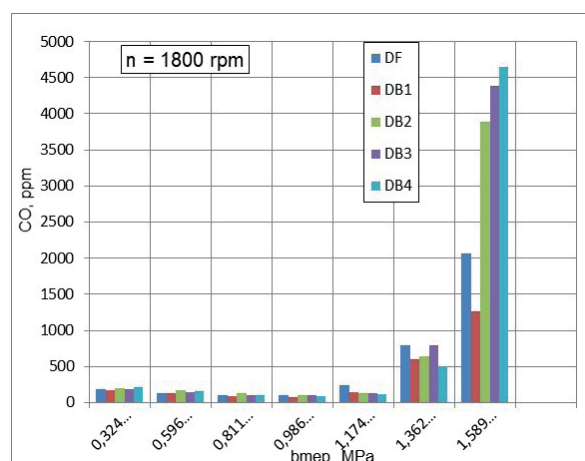


Fig. 8. The nitrogen oxide emissions (NO_x) as a function of engine load (bmep) when using various diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm.

The NO_x emissions decreased when running at speed of 2500 rpm because of the volumetric efficiency reduced and thus in-cylinder pressure was lower since boost pressure remained at the same level of 0.16 bar to eliminate the potential effects of the intake temperature on auto-ignition delay, combustion, and NO_x production. The positive effect of using oxygenated fuel blends on the NO_x production was less discernible due to shorter combustion duration in units of time at a higher speed. Therefore, the NO_x emissions produced from combustion of fuel blend DB1 (1.0 wt%) increased by 3.9%, 3.8% and 2.1% only when running under light loads of bmep = 0.311, 0.583 and 0.778 MPa against, 128, 240 and 374 ppm, a straight diesel produces at speed of 2500 rpm. Whereas the biggest NO_x emissions of 618, 892 and 1075 ppm emerged namely from combustion of oxygen-free diesel fuel when running under higher loads of bmep = 0.979, 1.174 and 1.320 MPa at the latter speed.

The combustion of fuel blend DB2 generated NO_x emissions similar as a straight diesel does, but slightly more, 588, 897 and 1036 ppm, than the remaining diesel-n-butanol blends DB1, DB3 and DB4 produce for considered test conditions. Thus, the NO_x emissions changing behaviour shows that the increased fuel-bound oxygen mass content is important, but evenly important is air-to fuel equivalence ratio 'lambda' (load) and the temperature inside the cylinder on which the NO_x production mainly depends [8].

The production of CO emissions depends on engine load, speed and the availability of air-borne oxygen in the cylinder. When running with diesel-n-butanol blends, an extra fuel-bound oxygen comes with an essential help to burn the fuel completely and reduce CO emissions when in the combustion chamber is a lack of air-borne oxygen and time needed for oxidation reactions is limited at a high speed. The effect of n-butanol-oxygen was minor because of the presence of air-borne oxygen in the cylinder when running a turbocharged CRDI diesel engine under light loads and the low speed of 1800 rpm. Emissions of CO decreased from 180 to 100 ppm for a straight diesel and from 210 to 90 ppm for the most oxygenated fuel blend DB4 due to bmep increased from 0.324 to 0.986 MPa and higher temperature inside the cylinder. The need for fuel-bound oxygen increased with engine load and, thus, CO emissions produced from combustion of diesel-n-butanol fuel blends were 54.2% (DB4), 38.0% (DB4) and 38.8% (DB1) lower than the respective values of 240, 790 and 2060 ppm a straight diesel produces for bmep = 1.174, 1.363 and 1.590 MPa at the latter speed. Whereas CO emissions progressively increased to 3890 (93.2%), 4390 (2.1 times) and 4650 ppm (2.3 times) when running with more oxygenated fuel blends DB2, DB3 and DB4 under the highest load, bmep = 1.590 MPa, and the low speed of 1800 rpm.



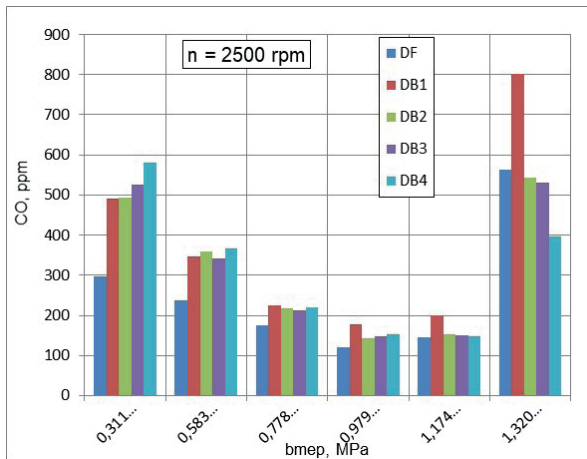


Fig. 9. The carbon monoxide emissions (CO) as a function of engine load (bmeP) when using various diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm.

Emissions of CO from combustion of diesel fuel decreased from 300 ppm to 235, 175, 120 ppm with engine load changed from bmeP = 0.311 MPa to 0.584, 0.779, 0.980 MPa to increase again to 145 and 560 ppm for higher bmeP = 1.174 and 1.320 MPa when running at high speed of 2500 rpm. These species are always higher when running at light loads because of too low temperature inside the cylinder to burn the fuel completely whereas the following CO emissions increase with engine load caused the lack of air-borne oxygen needed to convert all carbon in the fuel to CO₂ and all oxygen to H₂O. For these reasons, the CO production increased by 63.3% (DB1), 65.0% (DB2), 75.0% (DB3) and 93.3% (DB4) with increased fuel-oxygen content because n-butanol added to diesel fuel reduced net heating value of the fuel and thus temperature inside the cylinder for low load of bmeP = 0.311 MPa. Whereas the CO production progressively decreased from the highest value of 800 ppm (DB1) to 545 (DB2), 530 (DB3) and 395 ppm (DB4) with adding of n-butanol to diesel fuel when running under the highest load of bmeP = 1.320 MPa at speed of 2500 rpm. Thus, using of oxygenated fuel blends DB2, DB3 and DB4 CO emissions reduced by 2.7%, 5.4% and 29.5% against, 560 ppm, a straight diesel produces at high bmeP = 1.320 MPa and speed of 2500 rpm. Naturally, the need for fuel-bound oxygen increased with increasing speed of an engine because time available to burn the fuel completely was extremely limited.

Small 2.0 wt% content of n-butanol added to diesel fuel the THC production increased 7.6, 2.9, 2.3 times and by 8.8% and 39.1% against the respective values a straight diesel produces running under bmeP = 0.324-1.174 MPa and the low speed of 1800 rpm. However, THC emissions sustained at the lowest levels of about 80 ppm and were from 2.0 to 4.3 times lower when using the most oxygenated fuel blend DB4 (4.0 wt% oxygen) at the respective loads. Moreover, THC emissions decreased by 35.1% (DB1), 27.3% (DB2), 26.0% (DB3) and 43.4% (DB4) when using oxygenated fuel blends against that value of 3960 ppm a straight diesel produces for the highest

load of bmeP = 1.590 MPa at the low speed of 1800 rpm (Fig. 10).

Especially big THC emissions increase 5.3, 4.1, 3.9, 4.5, 4.0 and 2.1 times against that the normal diesel produces for the respective loads of bmeP = 0.311-1.320 emerged when running with slightly (1.0 wt%) oxygenated fuel blend DB1 at the higher speed of 2500 rpm. Moreover, the increased THC emissions were accompanied by a bigger exhaust smoke emerged from incomplete combustion of fuel blend DB1 under considered test conditions (Fig. 11). However, using more oxygenated fuel blends DB2, DB3 and DB4 the production of THC emissions reduced 2.4, 2.0 and 3.7 times, respectively, against that value of 260 ppm a fully loaded, bmeP = 1.320 MPa, straight diesel produces at a higher speed of 2500 rpm.

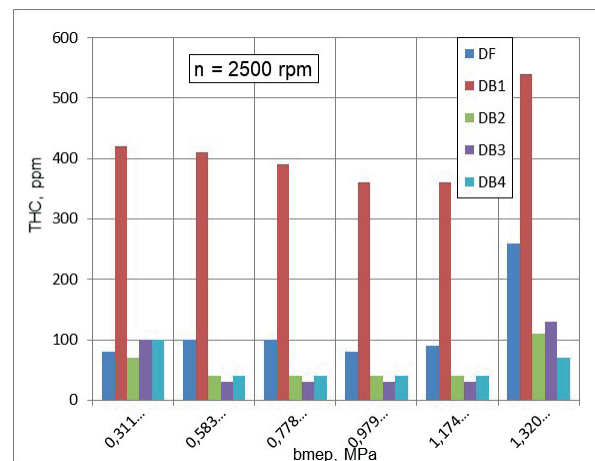
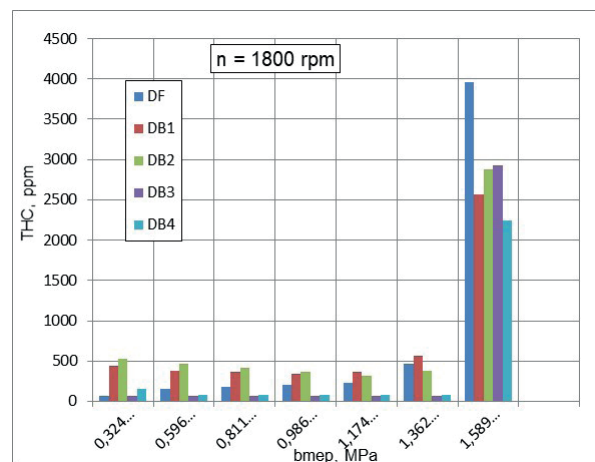


Fig. 10. The total unburned hydrocarbons (THC) as a function of engine load (bmeP) when using various diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm.

Smoke of the exhaust was at negligible and varied between 3.2-10.5% when running with normal diesel fuel under light and medium loads of bmeP = 0.324-1.362 MPa at low 1800 rpm speed (Fig. 11). Transparency of the exhaust further improved 1.5-2.8 times using of the most oxygenated fuel blend DB4 (4 wt% oxygen) against that a straight diesel produces for considered test conditions. As could be expected, the biggest positive effect of using diesel-n-butanol fuel blends

was achieved when running a turbocharged CRDI diesel engine under the highest load of $b_{mep} = 1.590$ MPa. Using oxygenated fuel blends DB1, DB2, DB3 and DB4 smoke of the exhaust reduced by 6.2%, 26.0%, 20.1% and 24.8%, respectively, against that value of 77.5% the combustion of diesel fuel generates for the highest load at the low speed of 1800 rpm. Apart from the help of fuel-bound oxygen in the combustion reactions, to having lower smoke contributed all of them, - lower density, C/H atoms ratio and stoichiometric air-fuel ratio of the tested fuel blends that reduced the need for air-born oxygen on which transparency of the exhaust mainly depends.

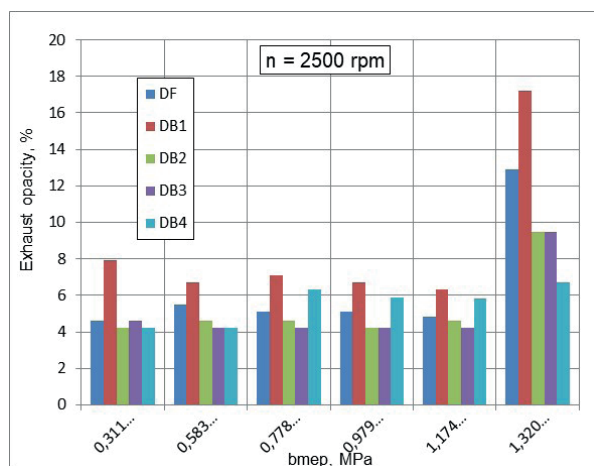
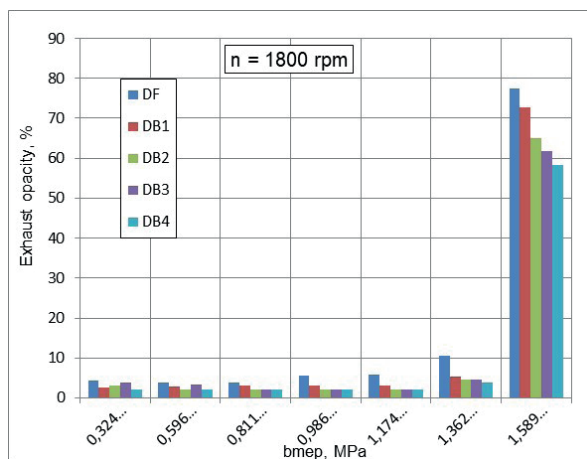


Fig. 11. The nitrogen oxide emissions (NO_x) as a function of engine load (b_{mep}) when using various diesel-n-butanol fuel blends at speeds of 1800 and 2500 rpm.

As swirl, squish and turbulence intensity of in-cylinder compressed air increased, smoke of the exhaust from a straight diesel did not change greatly and varied between 4.6-5.5% reaching the biggest value of 12.9% for the highest load, $b_{mep} = 1.320$ MPa, at high speed of 2500 rpm (Fig. 11). On the contrary, smoke of the exhaust increased by 71.7%, 21.8%, 39.2%, 34.0%, 28.6% and 33.3% when using slightly oxygenated fuel blend DB1 against the respective values the combustion of oxygen-free diesel fuel produces for the tested engine loads. Moreover, the higher smoke opacity was accompanied by

spontaneous eruption of unburned hydrocarbons as can be seen in Fig. 10. Apart of fuel-bound oxygen, the production of smoke depends on many variables, some of which are not fully understood. Therefore, it is difficult to reveal a real reason of this phenomenon. The increase in smoke opacity and, with some exceptions in THC, was also observed when running with slightly (0.91 wt%) ethanol-oxygenated diesel-HRD fuel blends a turbocharged CRDI diesel engine under close to stoichiometric conditions, $\lambda = 1.30, 1.25$ and 1.20 , at speeds of 1500, 2000 and 2500 rpm [12].

A half fuel portion burned earlier (Fig. 4) and the end of combustion took place sooner in the cycle (Fig. 5) when running with more oxygenated fuel blends DB2, DB3 and DB4 under the highest load, $b_{mep} = 1.320$ MPa, at speed of 2500 rpm. The positive changes in combustion resulted in brake thermal efficiency higher (Fig. 7) owing to lower CO (Fig. 9), THC (Fig. 10) emissions and reduced by 25.6%, 26.4% and 48.1% lower smoke opacity against that, 12.9%, a straight diesel produces for considered test conditions. The matter is that diesel fuel contained allowable by standard EN 590 amounts of polycyclic aromatics and sulphur whereas an alcoholic origin n-butanol did not. Therefore, the effects of oxygenated diesel-n-butanol fuel blends on the auto-ignition delay, combustion reactions, engine efficiency, smoke and exhaust emissions depend on the composition of the tested fuel blends, their chemical and physical properties as well as on engine load, speed, availability of air-born oxygen and temperature conditions inside the cylinder.

CONCLUSIONS

The auto-ignition delay period of a pilot diesel-n-butanol fuel blends DB1-DB4 portions increased by 3.9-13.8% and 2.3-15.5% against, 15.2° and 17.4° CADs, the normal diesel fuel auto-ignites when running a turbocharged CRDI diesel engine under full loads of $b_{mep} = 1.368$ and 1.307 MPa at the respective speeds of 1800 and 2500 rpm.

Burn angle MBF 50 occurred by 0.0-6.5% and 4.7-7.6% earlier in the cycle for oxygenated fuel blends DB1-DB4 than 17.7° and 20.9° CADs the 50% mass-portion of diesel fuel gets burned when running a fully loaded, $b_{mep} = 1.368$ and 1.307 MPa, turbocharged CRDI diesel engine at speeds of 1800 and 2500 rpm.

The combustion of air-fuel mixtures prepared over the longer ignition delay ended by 2.7° - 12.2° and 0.2° - 7.6° CADs earlier in the cycle ATDC when using fuel blends DB2-DB4 at medium ($b_{mep} = 0.976$ MPa) and high ($b_{mep} = 1.368$ MPa) loads than, 50.1° CADs, diesel fuel gets burned at light ($b_{mep} = 0.587$ MPa) load and speed of 1800 rpm. Whereas the effect of oxygenated fuel blends on the end of combustion was minor for all loads at a higher speed of 2500 rpm.

Brake specific fuel consumption decreased with engine load to 221.6 g/kWh for diesel fuel and 223.3 (0.8%), 228.8 (3.2%), 231.9 (4.6%) and 228.9 g/kWh (3.3%) for fuel blends DB1, DB2, DB3 and DB4 when running at $b_{mep} = 1.1362$ MPa and speed of 1800 rpm. The bsfc further increased to 230.0

(0.9%), 232.3 (1.9%), 234.5 (2.9%), and 234.4 g/kWh (2.8%), respectively, against, 228.0 g/kWh, a straight diesel consumes at bmep = 1.174 MPa and speed of 2500 rpm.

Brake thermal efficiency increased with engine load to reach maximum of 0.383 for diesel-n-butanol fuel blend DB3 (3.0 wt% oxygen) and 0.377 for a straight diesel running at bmep = 1.174 MPa and speed of 1800 rpm. Whereas the highest efficiency of 0.367 (0.8%), 0.373 (1.6%) and 0.368 (1.9%) suggested the most oxygenated blend DB4 (4.0 wt%) against, 0.364, 0.367 and 0.361, a straight diesel develops for bmep = 0.980, 1.174 and 1.320 MPa at speed of 2500 rpm.

The combustion of fuel blend DB2 (2.0 wt% oxygen) generated NO_x emissions 5.1% more than a straight diesel produces, 1438 ppm, for bmep = 1.362 MPa at the low speed of 1800 rpm. The NO_x emissions relatively decreased to the lowest levels of 955 (1.5%) and 1020 (5.1%) ppm when running a fully loaded engine, bmep = 1.590 and 1.320 MPa, with the most oxygenated fuel blend DB 4 (4.0 wt%) at the respective speeds of 1800 and 2500 rpm.

CO emissions increased to maximum of 3890 (93.2%), 4390 (2.1 times) and 4650 ppm (2.3 times) when running with oxygenated fuel blends DB2, DB3 and DB4 under the highest load, bmep = 1.590 MPa, and the low speed of 1800 rpm. Whereas CO emissions decreased by 2.7%, 5.4% and 29.5% against, 560 ppm, a straight diesel produces when running a fully loaded engine, bmep = 1.320 MPa, with diesel-n-butanol blends DB2, DB3 and DB4 at speed of 2500 rpm.

THC emissions decreased by 35.1% (DB1), 27.3% (DB2), 26.0% (DB3) and 43.4% (DB4) when using oxygenated fuel blends against, 3960 ppm, a straight diesel produces for the highest load of bmep = 1.590 MPa at the low speed of 1800 rpm. Using oxygenated fuel blends DB2, DB3 and DB 4 the THC production reduced 2.4, 2.0 and 3.7 times, respectively, against, 260 ppm, a fully loaded, bmep = 1.320 MPa, straight diesel produces at a higher speed of 2500 rpm.

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