EFFECT OF DIESEL-BIODIESEL-ETHANOL BLEND ON COMBUSTION, PERFORMANCE, AND EMISSIONS CHARACTERISTICS ON A DIRECT INJECTION DIESEL ENGINE

by

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The paper presents results of co-combustion of diesel-biodiesel-ethanol fuel blend in direct injection Diesel engine. Test was performed at constant rotational speed at three commonly used loads of this engine: 100%, 85%, and 70% of load. During the test hydrated ethanol was used at a concentration of 89% of alcohol. In this study, the ethanol fuel was added to diesel-biodiesel fuel blend with concentrations up to 50% with the increment of 5%. The biodiesel was used as an additive to prevent the stratification of ethanol and diesel blends. Thermodynamic parameters of engine were analyzed, and combustion process and exhaust emission were characterized. It turned out that with the increase in engine load is possible to utilize larger ethanol fraction in blend. With the increase of ethanol fuel in blend the increase in ignition delay (38.5% for full load) was observed, but burning duration decreased (49% for full load). The ethanol fuel share in blend generally causes the increase in NO_x *emission (42% for full load) due to higher oxygen content and higher in-cylinder temperatures. It turned out that, at full load the unrepeatability of indicated mean effective pressure was near the same up to 50% of ethanol fuel in blend (about 2%). In case of partial load at higher ethanol fuel fraction the increase in indicated mean effective pressure un-repeatability was observed.*

Key words: *co-combustion, diesel fuel, ethanol, blend*

Introduction

The increasing consumption of petroleum fuels and the growing requirements for purity of engine exhaust gas motivates to take up research into alternative fuels or new combustion systems in the engine. Biofuels such as alcohol and biodiesel, could partly replace petroleum fuel, reduce toxic emission, and more important restrain the life-cycle emission of CO_2 [1, 2]. Biodiesel has received wide attention as a replacement for diesel fuel because it is biodegradable, nontoxic, and can significantly reduce exhaust emissions and overall life cycle emission of carbon oxides, CO_2, from the engine. Biodiesel and ethanol can be produced from feedstock's that are generally considered to be renewable. Ethanol can be produced from agricultural products such as sugarcane, corn, sugar beet, molasses, cassava root, and barley by alcoholic fermentation process [3]. Biodiesel fuel is produced by chemically reacting a vegetable oil or animal fat with an alcohol such as methanol or ethanol through the chemical reac-

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tions transesterification [4, 5]. In the literature it can be found studied on alternative combustion systems such as stratified combustion system which provides the possibility of burning ultra-poor mixture [6-9]. In recent years great attention is paid to the use of alternative fuels [10-13].

Many investigations have studied the impact of blending various oxygenates additives with diesel or biodiesel fuel on engine performance and emission characteristics of Diesel engine [14]. The most investigated, in recent years, oxygenates are alcohols such as ethanol. It is known that diesel fuel and alcohol have difficulties in mixing and exhibit tendency to phase separation but biodiesel is known to be miscible with alcohols. Many researchers have proposed adding biodiesel as a solvent for ethanol in diesel fuel. Zhu *et al.* [15] investigated effect of co-combustion of diesel fuel, biodiesel, and ethanol-biodiesel (BE) blends in a four cylinder direct injection Diesel engine. The results indicated that when compared with biodiesel, the combustion characteristics of ethanol-biodiesel blends changed and the engine performance has improved slightly with 5% ethanol in biodiesel (BE5). In comparison with diesel fuel, the biodiesel and BE blends have higher brake thermal efficiency (BTE). On the whole, compared with diesel fuel, the BE blends could lead to reduction of both NO_x and particulate emissions of the Diesel engine. The effectiveness of NO_x and particulate reductions increases with increasing ethanol in the blends. With high percentage of ethanol in the BE blends, the HC, CO emissions could increase. But the use of BE5 could reduce the HC and CO emissions as well. Zhu *et al.* [1] in other paper presented the results of comparative studies of co-combustion ethanol or methanol with diesel and biodiesel fuel. They stated that the blended fuels could lead to reduction of both NO_x and particulate matter of a Diesel engine, with the biodiesel-methanol blends being more effective than the biodiesel-ethanol blends. The effectiveness of NO_x reduction is more effective with increase of alcohol in the blends. With high percentage of alcohol in the blends, the HC, CO emissions could increase and the BTE might be slightly reduced but the use of 5% blends could reduce the HC and CO emissions as well. Gnanamoorthi and Devaradjane [16] investigated the influence of compression ratio on the performance, combustion, and emission characteristics of a single cylinder 4-stroke direct injections naturally aspirated Diesel engine powered by diesel-ethanol blends. Compression ratios such as 17.5:1, 18.5:1, and 19.5:1 were considered and varied by altering the dimensions of piston bowl by keeping the standard stroke volume. Authors stated that the increase in compression ratio and ethanol blended fuel improves combustion and improves the performance of engine. Further there is a significant decrease in HC, CO, and smoke emissions however there is an increase in NO_x. The CO and total HC (THC) emission decreases with high compression ratio. This is achieved by faster mixing rate of ethanol and enhances the presence of oxygen at higher compression ratio. Increase in ethanol blend at high compression ratio causes higher steep heat release rate (HRR) which increases the adiabatic flame temperature. Pidol *et al.* [17] presented results of the study of the properties of ethanol blended fuels and evaluates their behavior in conventional diesel combustion and advanced combustion such as low temperature combustion (LTC).

The addition of ethanol into diesel fuel affects some key properties such as the blend stability, the cetane number or the flash point, the fuel formulation was thus improved. The remaining weak ignitability of the blends requires an engine settings optimization, which associated to the high oxygen content allows a combine reduction of smoke levels and NO_x emissions, with a contained fuel consumption penalty. These oxygenated fuels also lead to an extended LTC operating range and improved maximum power output. The main advantage of these ethanol blended fuels is to generate low smoke levels thanks to the presence of oxygen (first order parameter), the decrease of soot precursors concentration and the higher volatility of ethanol blended fuel. Taghizadeh-Alisaraei *et al.* [18] investigated the impact of ethanol addition to diesel fuel on engine performance and engine vibrations. The results showed that the torque and power increase on average by 3.8% at fuel blend with concentration of 6% ethanol as compared with those of pure diesel fuel. For this fuel, the root mean square and kurtosis of vibration on the engine block enhance 4.75% and 7.75% as compared with pure diesel, respectively, that shows the more shocks is created. They stated that with increasing the ethanol concentration more than 8% in diesel fuel, ignition delay rises and engine operates irregularity [18]. Fang *et al.* [19] investigated the effect of ethanol on combustion and emissions in premixed LTC in a four cylinders heavy-duty Diesel engine. The biodiesel was used as an additive to prevent the stratification of ethanol and diesel blends. The premixed LTC is achieved by the medium level of exhaust gas re-circulation and the prolonged ignition delay. Compared with diesel fuel, ethanol-diesel-biodiesel blends have lower NO_x emissions due to lower combustion temperature, resulting from the higher latent heat of vaporization. Unfortunately, the lower combustion temperature also leads to the higher HC and CO emissions [20]. Qi *et al.* [21] investigated the performance, emission and combustion characteristics of a common rail direct injection compression ignition (CI) engine fueled with diesel tung oil-ethanol blended fuels with different ratio of volume fraction. The experimental results indicated that, compared with diesel fuel, the ignition delay showed a little longer, the peak in-cylinder pressure and HRR were higher, while the combustion duration was slightly shortened for the blended fuels. The brake specific fuel consumption was increased, while the BTE was increased with the growth of Tung oil and ethanol volume fraction. The CO emissions of the blended fuel were higher at low engine loads, and HC emissions were higher and increased with the tung oil and ethanol addition. The NO_x emissions of the blended fuel were lower at low engine loads and slightly higher at high engine loads. Smoke emissions were drastically reduced at high engine loads [21].

This paper presents the results of experimental researches of co-combustion of diesel-biodiesel-ethanol fuel in direct injection Diesel engine. The motivation of these researches was to investigate the limit of ethanol fraction in blend which is acceptable for Diesel engine. In previous investigations [22] it was found that diesel-ethanol fuel blend up to 30% of ethanol fuel (EF) is possible to stabile operation of the test engine (COV_{IMEP} < 10%). In this study as a base of blend with ethanol used mixture of diesel biodiesel (80% diesel fuel and 20% biodiesel in vol.).

Experimental set-up

In the experimental studies analyzed combustion characteristics and emission of exhaust gases of direct injection, naturally aspired Diesel engine. It is one-cylinder stationary engine which is commonly used to power various devices such as pumps and other equipment on farms. This is two-valve engine, with a vertical cylinder, equipped with air cooling system with an axial fan. The main engine parameters are presented in tab. 1.

Table 1. Engine main specifications

Number of cylinders	1
Displacement volume	573 cm^3
Engine rotational speed	1500 rpm
Bore × stroke	90 × 90 mm
Compression ratio	17:1
Injection pressure	21 MPa
Start of injection (SOI)	17 bTDC
Rated power	7 kW at 3000 rpm

Figure 1 presents the test stand. This engine is constructed to operate with constant rotational speed of 3000 rpm with injection timing equal to 28° bTDC.

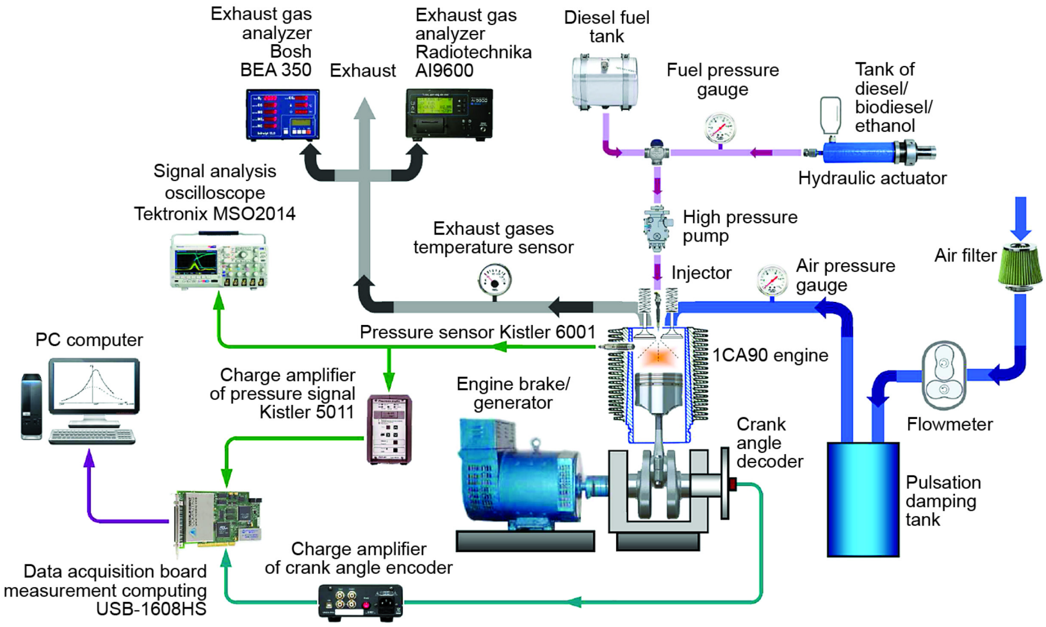

Figure 1. The test stand

After some mechanical modifications this engine can operate with lower rpm therefore the injection timing can be changed. It was due to changing in camshaft construction. During the tests the engine operated with 1500 rpm with injection timing 17° bTDC. For all analyzed blends, SOI was at the same set of the injection pump, *via* a suitable position of the pump shaft. In addition engine high pressure mechanical fuel pump was powered by gravity from the fuel tank placed above the fuel pump. In case of blend powering this supply system was ineffective. This was reflected by the observation of engine operating parameters, which were lower than it resulted from a decrease in energy consumption. In order to solve this unfavorable problem applied to the test bench pressure-powering system of fuel pump using hydraulic actuator [22].

The main measurement devices: piezoelectric pressure transducer, Kistler 6061 SN 298131, sensitivity ±0.5%, charge amplifier, Kistler 5011B, linearity of FS < ±0,05%, data acquisition module, Measurement Computing USB-1608HS – 16 bits resolution and sampling frequency 20 kHz, crank angle encoder, resolution 360 pulses/rev, software for digital recording, and analysis of the frequency signals [23]. Digital system for measuring the engine rotational speed based on the encoder position of the crankshaft with the resolution of 1 °CA. Exhaust gas analyzer: THC, CO, CO_2, O_2 – Bosch BEA 350 (CO, CO_2, and THC based on NDIR method, the measuring of the oxygen concentration based on an electrochemical cell O_2.) and exhaust gas analyzer: NO_x – Radiotechnika AI9600 (NO_x based on an electrochemical cell). The acquisition process covered 100 engine cycles. The combustion parameters calculated for every cycle and than averaged. The research conducted on the test engine powered by diesel-biodiesel-ethanol fuel blends. Table 2 presents fuel specifications of three fuels: diesel, biodiesel, and EF. Diesel fuel and biodiesel are available at petrol stations provided by a Polish distributor. The EF is available at retail in Poland. It was hydrated ethanol with water mass fraction 11%.

Alcohol, due to oxygen content in structure (CaHbOc), requires less air to maintain the same air-fuel equivalence ratio if compared to hydrocarbon based fuels [15]. Additionally, lower heating value (LHV) of alcohol is lower in comparison to diesel fuel hence to obtain the same engine performance the higher amounts of alcohol should be provided [15]. Analyzing the usefulness of alcohol to co-combustion in internal combustion (IC) engine it should be taken into account its properties such as: LHV and also stoichiometric air-fuel ratio and heat of evaporation. The EF belongs to the group of oxygenated fuels. Oxygen content of ethanol is equal to 34.8%. This high oxygen content has a big influence on the combustion phases of the Diesel engine. The oxygen participation if ethanol structure fuel causes the change in the ratio of combustion phases. With the increase in EF fraction it is observed the intensification of the premixed combustion phase and on the other hand it decreases diffusion phase. Ethanol has low stoichiometric air-fuel ratio, high oxygen content and high H/C ratio may be beneficial at improving the combustion process.

Table 2. Fuel specifications [15]

	Unit	Diesel	Biodiesel	Ethanol
Molecular formula		$C_{13}H_{23}$	$CH_3(CH_2)nCOOCH_3$	C_2H_5OH
Cetane number	–	51	56	~11
Molecular weight	g	205.2	~300	46
Liquid density	kgm^{-3}	840	832.5	789
Lower heating value	$MJkg^{-1}$	42.5	37.8	26.78
Heat of evaporation	$kJkg^{-1}$	260	300	840
Auto-ignition temperature	K	503	534	698
Stoichiometric air-fuel ratio	–	14.6	13.8	9.0
Viscosity at 25 °C	mPa·s	2.8	2-4.5	1.07
Carbon content	%	87	77.1	52.2
Hydrogen content	%	13	12.1	13.0
Oxygen content	%	0	10.8	34.8

Due to technical barriers of ethanol such as low value of cetane number, poor stability in diesel fuel and lower flash point it can not be used directly in Diesel engine [24]. Ethanol has lower stoichiometric air/fuel ratios than biodiesel and diesel fuel, thus blending ethanol into biodiesel leads to leaner combustion [15]. Table 3 presents the properties of diesel-biodiesel-ethanol fuel blends. Blends formed by adding EF to the base mixture which contained 80% diesel fuel and 20% biodiesel. The volume fraction of ethanol in the base mixture was varied from 5 to 50%. For example: DBE05 is a blend of diesel fuel and biodiesel in a ratio of 80:20 with ethanol, which share was equal 5%. As a limit of EF fraction in blend used unrepeatability of engine operating cycles described by COV_{IMEP}.

The base to creation of fuels mixture was blend of diesel fuel (D100) with biodiesel fuel (B100) with volumetric fraction 80/20. The B100 was used as a solvent to EF. This base mixture was blending with EF with volumetric fraction up to 50%. The mixture of D100 with

Table 3. Composition of diesel-biodiesel-ethanol blends

Properties	Unit	DB	DB E05	DB E10	DB E15	DB E20	DB E25	DB E30	DB E35	DB E40	DB E45	DB E50
Diesel--biodiesel blend	%	100	95	90	85	80	75	70	65	60	55	50
Ethanol fuel	%	0	5	10	15	20	25	30	35	40	45	50
Density	kgm^{-3}	838.5	836.0	833.6	831.1	828.6	826.1	823.7	821.2	818.7	816.2	813.8
Lower heating value	$MJkg^{-1}$	41.6	40.8	40.1	39.4	38.6	37.9	37.2	36.4	35.7	35.0	34.2
Heat of evaporation	$kJkg^{-1}$	268	296.6	325.2	353.8	382.4	411	439.6	468.2	496.8	525.4	554
Oxygen content	%	2.2	3.8	5.4	7.1	8.7	10.3	12.0	13.6	15.2	16.8	18.5

EF is characterized by phase separation and it limits the EF content in blend [12]. The use of B100 in blend has the impact not only on better miscible of fuels but has the impact on combustion process. The diesel fuel is consists of carbon and hydrogen atoms (C_nH_m) but biodiesel ($C_nH_mO_k$) consists of carbon, hydrogen, and oxygen atoms. With the increase in EF fraction in blend the oxygen content increased as well. It has the impact on combustion process phases. In the CI engine combustion process is divided in the premixed and diffusion phase. The increase in oxygen content in fuel causes shortens the combustion process. Consequently it has impact on exhaust emission of the engine. With the increase in oxygen content the combustion process occurs faster with higher temperature which can increase in NO formation. But on the other hand, the increase in EF fraction decreases C_nH_m content in blend, and the LHV of blend decreased as well. Blend DBE50 (DBE50 – with 50% of EF vol. fraction) the LHV is lower by over 20% comparison with DB blend (D80B20). With the increase in EF content in blend the LHV decreases and heat of evaporation increases. The heat of evaporation of EF has the significant impact on first stage of combustion process. The high value of this parameter makes it difficult to the process of fuel atomization and evaporation and in consequence causes the difficulties of auto-ignition. The increase in ignition delay has the impact mainly on the premixed combustion phase. It has the impact on exhaust emission as well. The small amount of ethanol could increase the oxygen content and reduce the viscosity and density of the blended fuel, leading to improved spray and atomization, better combustion and hence lower CO and HC emissions [15]. While for blends with higher ethanol fraction, the cooling effect of ethanol could reduce the in-cylinder gas temperature, leading to poorer oxidation reaction rate and hence increase in CO and HC emissions at low engine loads [15].

Results and discussion

As a result of engine indication process obtained the in-cylinder pressure traces *vs.* crank angle which give information on the in-cylinder processes as a mainly combustion process. The results of analysis of the indicator diagram described the combustion process and can contribute to performance evaluation of the test engine.

Performance characteristics

Figure 2 presents the courses of in-cylinder pressure for three commonly used loads and various EF fractions in blend. In the case of full load has been achieved the normal course of combustion for Diesel engine, even with a 50% share of alcohol (DBE50). For the 85% and 70% load the maximum fraction of EF, accepted by engine, was up to 45% (DBE45) and 40% (DBE40), respectively. After crossing these shares the combustion process was significantly deteriorated.

It is visible that with the increase in EF fraction in blend the higher value of peak pressure obtained. This trend is observed as long as the effect of the oxygen in the fuel for combustion process is greater than the impact of the increasing value of the heat of vaporization. For full load this limit obtained for EF fraction 45% (DBE45), for 85% load it was at 30% (DBE30), and for 70% load it was at 20% (DBE20). The obtained differences of peak pressure as compared to the combustion of diesel-biodiesel (DB) blends ware equal to 4, 3, and 1.5 bar, respectively, fig. 2.

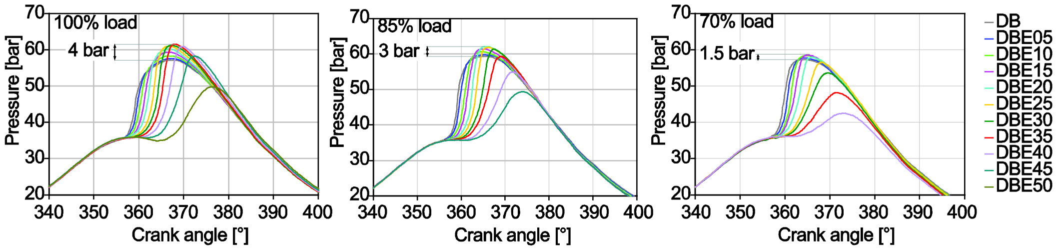

Figure 2. In-cylinder pressure for combustion of blends of diesel fuel, biodiesel with ethanol for three load of engine *(for color image see journal web site)*

On this basis, it can be stated that for high loads is possible of combustion a blends with higher ethanol content. An interesting dependence was observed for 80% load at which the higher peak in-cylinder pressure obtained than in case of full load. This can be explained by the fact that at full load the larger dose of fuel needs longer period of time to atomization and evaporation and combustion process occurs in a wider range of crank angle. In lower load (80%) the dose of fuels blend quickly reaches the parameters for burning and combustion process occurs in a narrower range of crank angle. Figure 3 presents the courses of HRR which are the basis for the evaluation of the combustion process in the IC engine. The HRR is calculated on the basis of the measured in-cylinder pressure and crank angle readings. The basis for determining the HRR is the first law of thermodynamics and the equation of state.

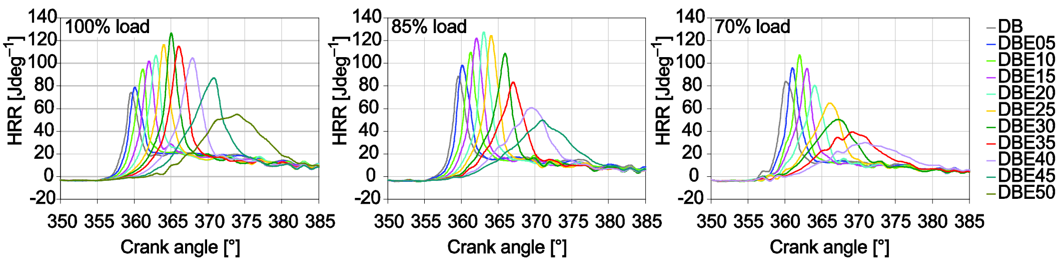

Figure 3. The HRR for combustion of DB blends with EF for three loads of engine
(for color image see journal web site)

Instantaneous cylinder volume, V, is precisely described by engine geometry. Due to omitting as follows: heat transfer to walls, crevice volume, blow-by, and the fuel injection effect, the resulted HRR is termed as the net HRR.

Similarly to in-cylinder pressure courses with the increase in EF fraction the peak HRR increased and that process occurs for some ethanol content determined for pressure courses. In case of high EF fraction in blend where HRR begins to outweigh the impact of the heat of vaporization the peak HRR decreased as well. These courses are used to determination of heat release which next are used to mass fraction burned (MFB) calculation. On the basis of MFB courses the combustion phases are determined. On the basis of MFB courses the ignition delay (ID) and burn duration (BD) is determined. The ignition delay is the time between the SOI and the start of combustion process. The ignition delay has a physical and a chemical delay phenomenon. The physical delay is the time required for fuel atomization, vaporization and mixing with the air. The chemical delay is the pre-combustion reaction of fuel with air [25]. Ignition delay in Diesel engines has a direct effect on engine efficiency, noise, and exhaust emissions. Experimentally, the start of ignition is mainly determined by the first appearance of visible flame on a high speed video recording, or sudden rise in-cylinder pressure or temperature caused by the combustion. In practice the start of combustion is defined as the beginning of heat release, and the end of combustion is defined as the crank angle where the summated heat release is 90% of the total heat release. The combustion duration is the time interval from the start of combustion to the end of combustion [1]. In these studies it is assumed that the start of combustion is time of 10% of the heat release (MFB). Figure 4 presents the combustion phases such as ignition delay and BD for all analyzed cases. In all analyzed cases the start of fuel injection was the same equal to 17° bTDC. It is visible that with the increase in EF fraction in blend the ignition delay increased as well. This relationship is maintained for all analyzed cases. A bit different relationship observed in case BD, at full engine load up to 25% of EF in blend the BD decreased slowly but after exceeding this value of EF fraction noticed significant drop in BD and decreasing of this parameter was continued. After exceeding 25% of EF in blend BD decreased due to relatively high value of ignition delay which increases the premixed combustion phase. For DBE50 the combustion duration was shorter compared to DB blend by almost 50%. In cases of partial loads with the increase in EF fraction the BD decreased simultaneously. Addition of EF makes the flame propagation faster and shortening the combustion process [16].

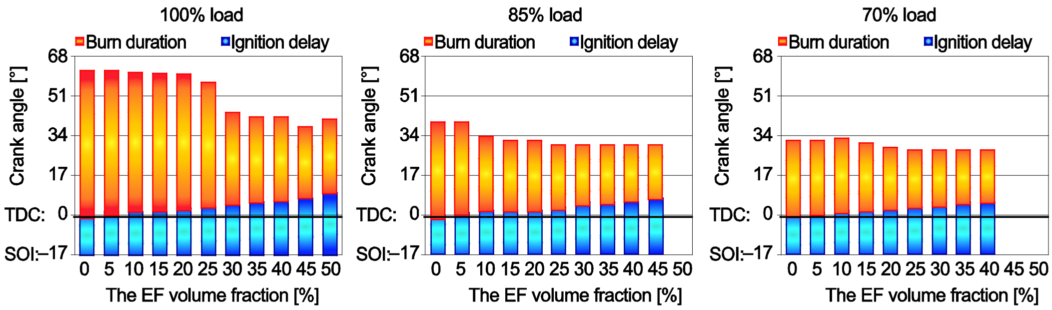

Figure 4. Ignition delay and burn duration for combustion of DB blends with ethanol for three loads of engine *(for color image see journal web site)*

On the basis of the pressure courses are calculated the values of indicated mean effective pressure (IMEP) and indicated thermal efficiency (ITE). The IMEP is one of the parameters which characterized engine operation in terms of obtaining a high, desired performance of engine. In fig. 5(a) is presented IMEP for all analyzed cases. For full load of engine IMEP was on near the same level (6.25-6.5 bar), the drop in value was lower than 4%.

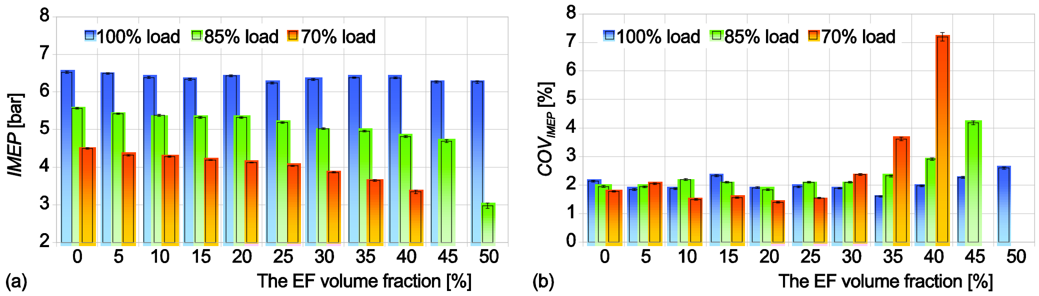

Figure 5. The IMEP (a) and coefficient of unrepeatability of IMEP (b) of test engine powered by diesel fuel and biodiesel with EF for three loads of engine *(for color image see journal web site)*

This shows that the test engine can operate on that fuels blend and reach high performance. In the case of partial loads with the increase in EF fraction IMEP started to decrease. Figure 5(a) shows also the uncertainty designation of the indicated mean effective pressure (ΔIMEP), which determines the dispersion (spread) around the average value calculation results of the IMEP in the individual cycles containing 100 recorded engine cycles. It was assumed that the uncertainty designation of the IMEP has a normal distribution and it is calculated from the equation [26].

Figure 5(b) presents the coefficient of IMEP variation (COV_{IMEP}) and uncertainty designation of COV_{IMEP} (ΔCOV_{IMEP}). It can be stated that up to 30% of EF fraction in blend the value of COV_{IMEP} was at the same near to constant level (COV_{IMEP} = 2%). For full load the unrepeatability of IMEP was acceptable for all EF fractions. For partial loads after exceeding 30% of EF in blend COV_{IMEP} started to increases. In case of 70% load at 40% of EF fraction began to strongly increased and due to this fact the engine operate with larger EF fraction was impossible. Figure 6 presents the specific fuel consumption (SFC) and ITE for analyzed cases. The SFC is an important parameter to compare the performance of various fuels used to power engine.

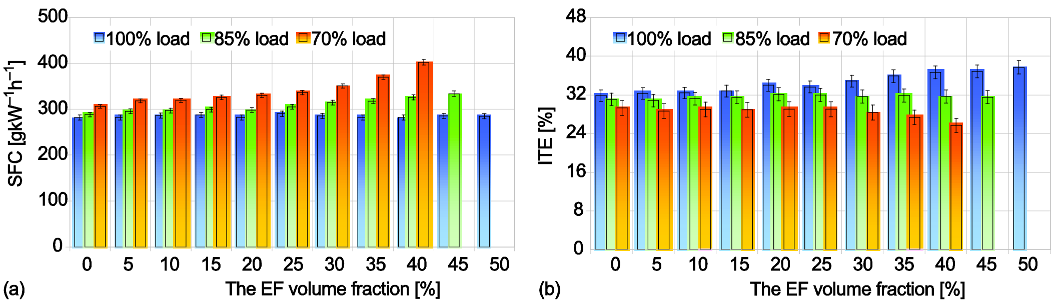

Figure 6. The SFC (a) and ITE (b) of the test engine powered by diesel fuel and biodiesel with EF for three loads of engine *(for color image see journal web site)*

As seen in fig. 6(a), SFC values increases with the decreasing the engine load for all analyzed cases. Based on the results stated that for full load SFC was kept on the same level equal to 280 g/kWh. It is interesting because with the increase in EF fraction LHV of blend decreases as given in tab. 3. This mean that more fuel amount is required for blend than for diesel or biodiesel fuel. This can be explained by the fact that with the increase in EF fraction ITE increased, fig. 6(b). For full load ITE increased and in case of 50% of EF fraction the gain of ITE was equal to 6%. It can be explained that with the increase in EF fraction in blend BD started to decrease and due to this fact the heat losses was lower. For partial load (85%) SFC started to increase (16%) with EF fraction in blend but ITE slightly increased (1%) up to 45% of EF fraction. In case of 70% load SFC increased (32%) with EF fraction in blend and at the same time ITE decreased (12.5%).

Emission characteristic

The exhaust emissions measured during the tests were: NO_x, CO, THC, CO_2, and oxygen content. Emissions results were converted to the unit [$gkW^{-1}h^{-1}$] known as specific values. Figure 7 presents NO_x and total unburned HC emission. It can be stated that with the increase load the NO_x emission decreased as well. In case of full and partial loads with the increase in EF fraction in blend the NO_x emission increased up to DBE40 (full load), DBE30 (85% load), and DBE20 (70% load) and after exceeding this EF content its started to decreasing. This decrease in NO_x emissions is caused by deterioration of the combustion process by large share of EF. The relatively high value of heat of evaporation of blend causes the increase in ignition delay and due to this fact the peak of in-cylinder pressure and temperature is reached for larger °CA aTDC. In case of full load up to 40% of EF in blend NO_x increased (from 1.4 to 2.8 g/kWh) but after exceeding this content the NO_x emission started to decrease (to 2.4 g/kWh). With decreasing engine load the peak NO_x emission reached for lower EF fraction in blend. In case of partial load the maximal specific NO_x emission is reached for lower EF fraction in blend (5.8 g/kWh at DBE30 for 85% load and 6.3 g/kWh at DBE20 for 70% load).

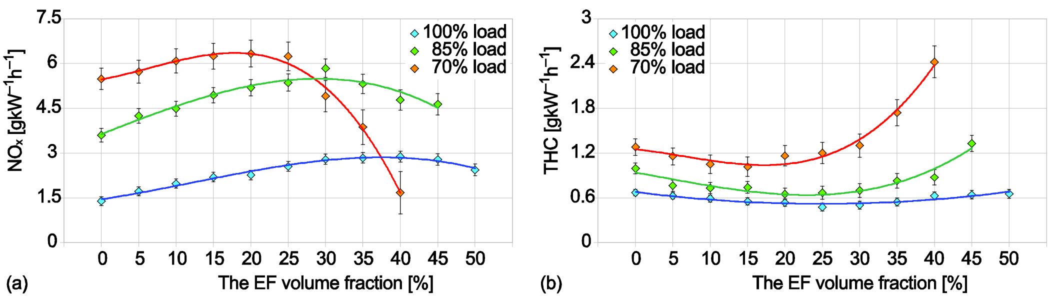

Figure 7. Emission of NO_x (a) and THC (b) of the test engine powered by DB blends with EF for three loads of engine *(for color image see journal web site)*

The thermal mechanism dominates the formation of NO_x in biodiesel combustion [15]. In case of biodiesel, the combustion temperature as well as the oxygen contents could be higher which leads to the higher NO_x emissions. On the other hand, the cooling effect of ethanol associated with its lower calorific value and higher latent heat of evaporation could reduce the combustion temperature and hence reduce the NO_x emissions. Formation of NO_x is strongly dependent on the in-cylinder temperature, the oxygen concentration and the residence time of the gases at high temperatures [27]. Ethanol fraction in blend with diesel fuel causes several effects that are opposite influence on combustion process in the IC engine. The low cetane number and the significant amount of oxygen in the ethanol blends causes high peak HRR. This can increase the in-cylinder temperature. On the other hand, high value of heat of evaporation of ethanol and low flame temperature tend to reduce the in-cylinder temperature.

The mechanism of the NO_x formation is dependent on the intensity of these effects during combustion [28]. However, ethanol could reduce the cetane number of the blended fuel, which means longer ignition delay period and a larger amount of fuel burned in the premixed phase and causes the increase in NO_x emission. The higher oxygen contents of ethanol could enhance NO_x emission as well. These mentioned factors act against each other. For the BE fuels, the cooling effect of ethanol seems to be the dominating effect leading to the overall reduction of NO_x emission [15]. Analyzing the emissions of THC it can be stated that for partial loads of engine up to 30% of EF fraction emission is kept near on the same level, respectively, 1.2 g/kWh and 0.7 g/kWh. After exceeding 30% EF emission of THC started to increase. In case of the full load emission of THC was near the same level equal to 0.6 g/kWh. The THC emission is increased under lower engine load, fig. 7(b). The THC emissions depend on the combustion quality, engine load and physical properties of the fuel. There are different mechanisms that affect THC emission in CI engine. The HC emission generally increases compared to neat diesel fuel due to high heat of evaporation of ethanol leaving unburned ethanol in the exhaust [16]. The results showed that this effect was more significant at low and medium loads. Lower emissions of THC can be a consequence of higher fuel burn efficiency. When ethanol is added to the fuel, more oxygen is available in the combustion which improving the burning quality [29].

Alptekin *et al.* [30] stated that ethanol participation in blend causes decrease in THC emission. In analyzed cases this fact was confirmed but it was truth up to 25-30% of EF fraction in blend. Figure 8(a) presents CO emission. For full load it was observed that with the increase in EF fraction CO emission decreases (from 0.13 g/kWh at DB to 0.04 g/kWh at DBE50). For 50% of EF in blend obtained three times lower emission. In case of partial loads up to 25-30% of EF fraction CO emission decreased (to 0.02 g/kWh at DBE25 for 85% load, and 0.03 g/kWh at DBE15 for 70% load) but after exceeding this EF content in blend emission started to grow (to 0.06 g/kWh at DBE45 for 85% load and 0.1 g/kWh at DBE40 for 70% load). The CO formation in engines is strongly connected with combustion quality, and tends to increases with insufficient oxygen and incomplete combustion.

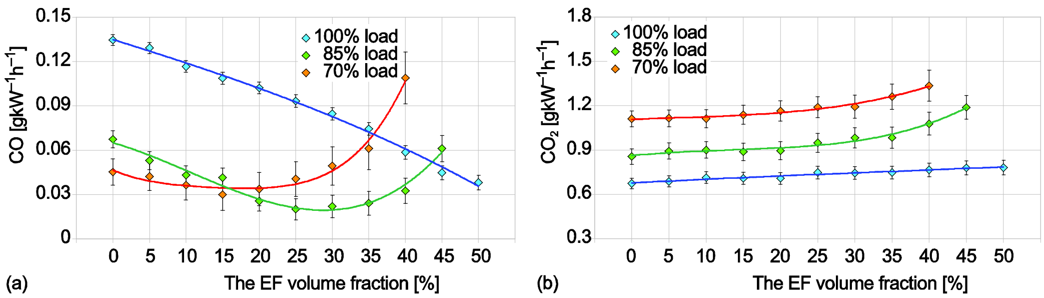

Figure 8. Emission of CO (a) and CO_2 (b) of the test engine powered by DB blends with EF for three loads of engine *(for color image see journal web site)*

As well known from the literature [28], the CO emissions increase especially for fuel-rich mixtures. The addition of ethanol to the fuel reduces the exhaust gas temperature, due to the high value of heat of evaporation of ethanol. This results in lower CO oxidation, increasing the emission rate of this component [29]. On the other hand, the higher oxygen concentration in fuel-rich zones of the combustion chamber with the addition of ethanol may lead to more complete combustion and reduce CO formation [18]. The obtained results showed that CO concentration is depended on engine load and ethanol fraction in blend. For full load it was observed the reduction in specific CO emission which can be explained by improvement of the combustion process by ethanol participation. In case of lower loads the dominant is cooling effect of ethanol by the high value of heat of evaporation. Similar results obtained by Rakopoulos [31] and Alptekin [30] which stated that CO emission increases at low loads while it decreased at high load with the addition of ethanol to fuel blend. High load of engine causes high in-cylinder pressure and temperature which provide better combustion process.

Analyzing the CO_2 emission stated that with the increase engine load the specific CO_2 emission decreased, fig. 8(b). It can be stated that for low and medium EF fraction the specific CO_2 emission is near to constant. The CO_2 is main components of exhaust emissions built up by burning HC fuels. The reason of decrease in CO_2 emissions is the low C/H ratio of bioethanol.

Conclusions

The paper presents results of experimental investigation on the combustion process and emission characteristics of direct injection Diesel engine fueled with diesel-biodiesel-ethanol blend. It turned out that it is possible to power Diesel engine with relatively high ethanol fraction: up to 50% at full load, 45% at 85% of load, and 40% at 70% of load. It can be stated that up to 30% of EF fraction in blend the value of COV_{IMEP} was at the same near to constant level (COV_{IMEP} = 2%). In case of 70% load at 40% of EF fraction began to strongly increased and due to this fact the engine operate with larger EF fraction was impossible. It turned out that with the increase in EF fraction in blend the higher value of peak pressure obtained and the ignition delay increased as well but BD decreased. For DBE50 the combustion duration was shorter compared to DB blend by almost 50%. At full load the IMEP was on near the same level for all EF fractions, at partial load the IMEP decreased with the increase in EF fraction. With the increase in EF fraction the SFC increased due to lover value of LHV of ethanol, the ITE increased in case of full load, at partial load 85% was kept on the same level and at lowest load decreased. With the load increase the specific NO_x emission decreased and in all analyzed cases for low and medium EF fraction noticed increase in NO_x emission and then was decreasing due to deterioration of combustion process. In case of full load up to 40% of EF in blend NO_x increased (from 1.4 to 2.8 g/kWh) but after exceeding this content the NO_x emission started to decrease (to 2.4 g/kWh). With decreasing engine load the peak NO_x emission reached for lower EF fraction in blend. In case of partial load the maximal specific NO_x emission is reached for lower EF fraction in blend (5.8 g/kWh at DBE30 for 85% load and 6.3 g/kWh at DBE20 for 70% load). The THC emission decreased, with full load THC emission was on the same level, for partial loads with higher EF fraction THC emission started to increase. In case of the full load emission of THC was near the same level equal to 0.6 g/kWh. With the load increase the specific CO emission increased, at full load with the EF fraction increase the CO emission decreased, at partial load this trend was maintained for low and medium EF fraction and for higher EF fraction the specific CO emission started to increase. For full load with the increase in EF fraction CO emission decreased (from 0.13 g/kWh at DB to 0.04 g/kWh at DBE50). Presented study showed that it is possible to co-combustion EF with relatively high water content in CI engine without decreasing its performance and to the benefit of the environment.

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