COMBUSTION OF DIFFERENT REACTIVITY FUEL MIXTURE IN A DUAL FUEL ENGINE

by

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Original scientific paper
https://doi.org/10.2298/TSCI170606299T

The paper presents results of experimental investigation of co-combustion of fuels of various reactivity at dual fuel Diesel engine. The tested fuels were ethanol fuel (EF) and diesel. It was used one-cylinder direct injection Diesel engine. Researches were conducted at constant rotational speed and constant injection timing. With the increase in EF fraction the deterioration of combustion process was indicated. On the basis of heat release variation were determined combustion phases: ignition delay, combustion duration. With the increase in EF participation in combustion processes the IMEP increased as well. There was no noticeable effect of ethanol on unrepeatability of indicated mean effective pressure. The increase in EF fraction causes the specific NO_x emission decreases but specific total hydrocarbon emission increased.

Key words: *combustion, dual fuel, emission, engine, ethanol*

Introduction

Compression ignition engines are widely used to power engineering devices for its good durability and thermal efficiency. Unfortunately these engines are major contributors of air pollutant emission such as CO and NO_x. One of the more effective ways to reduce emissions is to control the combustion process. Currently, willingly is used the co-combustion of fossil fuels with renewable fuels which reduces the emission of toxic substances in the exhaust gases [1-3]. Biofuels are a renewable, less toxic, eco-friendly, and sustainable alternative fuel for compression ignition engines compared with fossil fuels.

The co-combustion process of fuels using dual fuel mode is one of the methods to utilize alternative fuels in internal combustion engines. In practice co-combustion of more than one fuel is possible by using dual fuel technology or blend mode. In dual fuel mode are used two independent power supply systems: the first is already existing direct injection system which delivers diesel fuel and the second injects alcohol fuel into the intake manifold of the internal combustion engine. Into the engine cylinder is delivered air-fuel mixture, nearly homogeneous. The ignition process is controlled by the injected dose of diesel fuel. The biggest disadvantage of this solution is the need to install an additional injector, along with a separate fuel tank, lines and controls. The greatest advantages of this solution are: replacement of diesel fuel with other fuel, the flexibility of power system and the engine can switch from dual fuel to diesel fuel operation, reduction of smoke emission.

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In case of providing fuels blend to engine combustion chamber by direct injection system, the mixture of fuels is already prepared. The greatest difficulties are that large percentages of alcohol do not mix with diesel fuel hence use of diesel-alcohol blends is not feasible. Also, the blends are not stable and separate in the presence of trace amounts of water. In such a power system can not change the ratio of diesel/alcohol [4-10]. The co-combustion of fuels blends is interesting as well. In papers [4-10] are presented results of the investigation of co-combustion of alcohol fuels with diesel or biodiesel fuels. It was stated that the combustion process in the dual fuel engine fueled with diesel and alcohol fuels has different pattern compared to combustion in the conventional compression ignition engine. The addition of alcohol leads to the increase in ignition delay and shortens the combustion duration. The increase in the phase of kinetic combustion is observed while limiting the phase of diffusion combustion, which leads to the emissions of toxic compounds in the exhaust. Co-combustion of alcohol fuels in the dual fuel engine may lead to the increase in NO_x for insignificant alcohol contents. The alcohol burned in the compression ignition engine has a substantial effect on reduction in soot emissions. The dual fuel engine is more sensitive to changes in the diesel fuel injection start angle compared to the engine fuelled only by diesel.

The process of biodiesel combustion with ethanol in the compression ignition engine was characterized by greater repeatability in consecutive work cycles compared to the combustion of diesel with ethanol. With larger contents of ethanol in the fuel blends, the process of combustion of the bio-ethanol (BE) blend was characterized by lower ignition delay time and elongation of combustion duration compared to combustion of diesel-ethanol blend (DE) burn. Combustion of BE blend can be the source of greater specific total hydrocarbon (THC) emissions at substantially lower specific NO_x emissions [4-5, 9]. Some authors [11] investigated the possibility of adding ethanol as an additive with animal fat biodiesel that is tested as an alternative fuel for diesel in a compression ignition engine. Biodiesel was obtained from waste pork lard by base-catalyzed transesterification with methanol when potassium hydroxide as catalyst. The experimental test results showed that the combustion and performance characteristics improved with the increase in the percentage of ethanol addition with biodiesel. When compared to neat biodiesel and standard diesel, an increase in brake thermal efficiency (BTE) of 5.8% and 4.1% is obtained for blend at full load of the engine. With the increase in the percentage of ethanol fraction in the blends, peak cylinder pressure and the corresponding heat release rate are increased. Biodiesel-ethanol blends exhibit longer ignition delay and shorter combustion duration when compared to neat biodiesel [11].

Dual fuel technology uses injects an alcohol fuel into the intake manifold of an internal combustion engine. Into the engine cylinder is delivered air-fuel mixture, nearly homogeneous. The ignition process is controlled by the injected dose of diesel fuel. This requires the addition of injector, along with a separate fuel tank, lines and controls [12]. Some advantages of dual fuel technology: some part of the fuel energy can be derived from non-fossil fuel, flexibility of power system – the engine can switch from dual-fuel to diesel fuel operation. Co-combustion of alcohols with diesel fuel can reduce smoke emission [12]. With increase in alcohol participation the peak combustion temperature decreases as well as the temperature of the mixture at the end of the compression stroke (preignition temperature). With the increase in the percentage of alcohol, the main combustion phase, expressed by crank angle (CA) 10-90°. On the other hand, the CA 0-10° lengthens itself due to lower preignition temperature by cooling effect by alcohol addition [12].

In the paper [13] authors investigated the effect of different high-octane fuels (such as eucalyptus oil, ethanol, and methanol) on engine's performance behavior of a biofuel based dual fuel engine. Results indicated the significant improvement in BTE simultaneous reduction in smoke and NO emissions in dual fuel operation with all the inducted fuels. At 100% load the BTE increased from 25.6% to a maximum of 32.3, 30.5, and 28.4%, respectively, with eucalyptus oil, ethanol, and methanol as primary fuels. Smoke was reduced drastically from 78% with neat mahua oil a minimum of 41, 48, and 53%, respectively, with eucalyptus oil, ethanol, and methanol at the maximum efficiency point. The optimal energy share for the best engine behavior was found to be 45, 27, and 23%, respectively, for eucalyptus oil, ethanol, and methanol at 100% load. Additionally, the eucalyptus oil showed the maximum BTE, minimum smoke and NO emissions and maximum energy replacement for the optimal operation of the engine [13].

In the paper [14] authors stated that increased ethanol energy fraction increases the engine efficiency until the operation is limited by misfiring associated with over-retarded combustion phasing. By energy fraction, up to 60% of diesel is replaced by ethanol, which achieves 10% efficiency gain compared with the diesel-only operation. They stated also that the decreased burn duration is the primary cause for the efficiency gain, *i. e.* the fast burning of ethanol improves the combustion. The burn duration increased with advancing the diesel injection timing at a fixed ethanol energy ratio. The combustion duration and combustion phasing should be considered to explain trends in the indicated mean effective pressure (IMEP) or efficiency of dual-fuel combustion engines. The THC, CO, and NO_x emissions increased with increasing ethanol fraction, which raises a question on the advantages of utilizing ethanol in a Diesel engine [14]. Results of the investigation of a fumigation system for introduction of ethanol in a small capacity Diesel engine and to determine its effects on emission are presented by Chauhan *et al.* [15]. Fumigation was achieved by using a constant volume carburetor. Different percentages of ethanol fumes with air were then introduced in the Diesel engine, under various load conditions. Results show that fumigated Diesel engine exhibited better engine performance with lower NO_x, CO, CO_2, and exhaust temperature. Ethanol fumigation has resulted in increase of unburned HC emission in the entire load range. Considering the parameters, the optimum percentage was found as 15% for ethanol fumigation [15].

From the literature review, it is clear that most of the research work on ethanol has been done on blend with diesel [4-9] and [16-18]. Although many publications on dual fuel engines can be found in the literature, it seems that the topic has not been exhausted. This paper presents results of experimental investigation of combustion of EF with diesel in dual fuel Diesel engine.

Experimental set-up

Test stand

In the study was used the one-cylinder direct injection natural aspired compression ignition engine, fig. 1. Detailed engine specifications are presented in tab. 1. Tests conducted at a constant angle of diesel fuel injection and constant rotational speed equal to 1500 rpm. The engine was air cooled. The engine was equipped with additional injection system which allows providing EF into the intake manifold. The injector was controlled by an external system that allowed the engine to synchronize with the

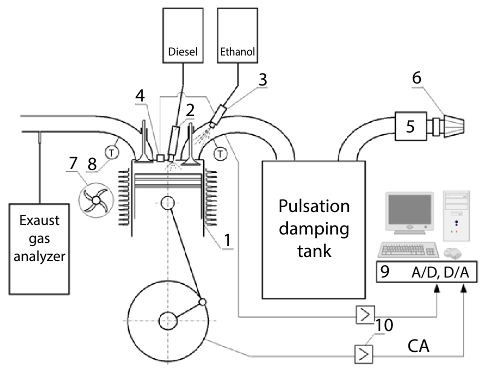

Figure 1. Diagram of the experimental set-up: *1 – engine, 2 – diesel fuel injector, 3 – EF injector, 4 – in-cylinder pressure sensor, 5 – intake air flowmeter, 6 – air filter, 7– cooling fan, 8 – exhaust gases temperature sensor, 9 – PC with data acquisition system, 10 – CA sensor*

Table 1. Main engine parameters

Parameter		Unit
Number of cylinders	1	–
Displacement volume	0.573	[dm^3]
Bore	90	[mm]
Stroke	90	[mm]
Compression ratio	17:1	–
Rated power	7	[kW]
Crankshaft rotational speed	1500	[rpm]
Injection pressure	21	[MPa]
Injection timing	17	[°bTDC]

engine operation and set an open time, which allowed the fuel to be properly dosed. At the beginning of the study made calibration of the injection system in order to determine the amount of fuel delivered by the injector at the various opening time settings. A Kistler piezoelectric pressure transducer and a CA encoder were used to measure the in-cylinder gas pressure and the corresponding CA. During the study, exhaust emission control was also carried out using an exhaust gas analyzer.

The study was conducted on the test bench, which included the following elements:

- modernized engine 1CA90 adapted for a multi-fuels powering,
- data acquisition system,
- exhaust gas analyzer: THC, CO, CO_2, O_2 – Bosch BEA 350:

THC:	range 0-9999 ppm vol.	accuracy: 12 ppm vol.
NO_x:	range 0-5000 ppm	accuracy: 10 ppm.
CO:	range 0-10 %vol.	accuracy: 0.06 %vol.
CO_2:	range 0-18 %vol.	accuracy: 0.4 %vol.
O_2:	range 0-22 %vol.	accuracy: 0.1 %vol.
λ:	range 0.5-9.999	accuracy: 0.01.

The digital measurement system for data acquisition:

- piezoelectric pressure transducer, Kistler 6061 SN 298131, sensitivity: ±0.5%,
- charge amplifier, Kistler 5011B, the linearity of FS < ±0,05%,
- data acquisition module, Measurement Computing USB-1608HS – 16 bits resolution, sampling frequency 20 kHz,
- computer PC,
- the CA encoder, resolution 360 pulses/rev,
- software for digital recording and analysis of the frequency signals [19].

This engine was equipped with independent port and direct injection (DI) fuel injection systems. For this study, intake manifold of the engine was modified to install the port fuel injection system, which enables the engine to operate in dual fuel mode. Port fuel injector was installed in the upstream of the intake manifold in such a way that fuel spray was directed delivered close to the intake valve. Ethanol was injected into the intake manifold at 3 bar injection pressure and diesel is directly injected into the cylinder at 200 bar injection pressure.

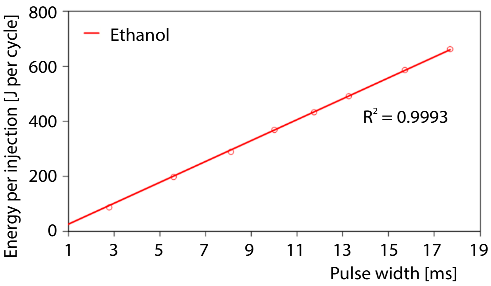

Figure 2. Characteristics of ethanol injector

The calibration result indicates that quantity of injected fuel by the injector in the intake manifold is a linear function of the pulse width was used to actuate the solenoid injector. Results of injection system calibration are visible in fig. 2.

A separate fuel injection controller was used to control the quantity of port injected fuel. Microcontroller acquired the signal from the cam sensor. The electrical pulse drives the solenoid fuel injector and duration of electrical pulse defines the fuel injection duration. Fuel injection duration was controlled by time of injector opened determined on the basis of the calibration characteristics.

During the tests recorded 100 consecutive engine cycles with resolution 1 °CA. It was recorded simultaneously: rotational speed of engine, air and fuel consumption, air temperature, fuel temperature, exhaust gas temperature, ambient temperature, and pressure. The excess air coefficient was constantly monitored. Using the exhaust gas analyzers recorded changes in the concentration of the components in the engine exhaust gases, such as NO_x, HC, CO, CO_2, and O_2.

Fuel characteristics

Alcohol is a form of renewable energy which can be produced from carbon based agriculture feedstocks, locally grown crops and even waste products including waste paper, tree trimmings and grass [20]. Sugarcane residue is other renewable energy source of alcohol production [20]. Ethanol is considered a potential substitute to diesel fuel. It is renewable fuel, has a high latent heat of vaporization, it is an oxygenated fuel, sulfur-free and has the high burning rate, with high potential to reduce NO_x and particulate matter (PM) emissions from Diesel engines [8-10]. One of the major limitations of using diesel fuel-ethanol blends in compression ignition engines is the low miscibility of ethanol in diesel fuel for a large temperature range, due to differences in the chemical structure of these components, which may cause phase separation [4, 5, 7]. Ethanol can be produced from any plants which contains sugar or other components which can be converted into sugar, such as starch or cellulose in the fermentation, distillation, and dehydration process [21]. This is especially important taking into account the global trend to reduce CO_2 emissions. The CO_2 is the main GHG and the CO_2 released by biofuel combustion can be fixed by growing plants and therefore makes no net contribution to global warming [21]. Concern about the environment is forcing reduction of GHG emission, in particular, significant cuts in CO_2 being released from the power industry [22-24]. In tab. 2 are presented main standards fuels parameters. Ethanol consists of carbon, hydrogen, and oxygen. Ethanol contains 2-carbon atoms having the molecular formula CH_3CH_2OH and isometric with di-methyl-ether [20].

Table 2. Properties of diesel fuel and ethanol

Properties	Diesel	Ethanol
Molecular formula	$C_{14}H_{30}$	C_2H_5OH
Molecular weight	198.4	46
Cetane number	51	~11
Lower heating value, [MJkg^{-1}]	41.7	26.9
Density at 20 °C, [kgm^{-3}]	856	789
Viscosity at 25 °C, [mPas]	2,8	0.983
Heat of evaporation, [kJkg^{-1}]	260	840
Stoichiometric air-fuel ratio	14.7	9.0
Autoignition temperature, [°C]	230	425
Flame speed, [m/s]	0.86	~3
Flame temperature, [°C]	2054	2120
Carbon content, [wt.%]	87	52.2
Oxygen content, [wt.%]	0	34.8

The EF is characterized by relatively low cetane number in comparison with diesel fuel what causes the increase in ignition delay. Additionally, EF has the high value of the heat of evaporation and lower value of lower heating value (LHV). The high value of the heat of evaporation causes a drop in temperature of charge before ignition what affects the ignition delay. Due to LHV of ethanol that changing the dose of energy in diesel fuel to the energy dose in ethanol, it should be used grater mass of EF. Air demand of EF is lower than for diesel fuel that the concentration of unused oxygen in the combustion chamber is higher. Ethanol is characterized by higher flame speed what decreased combustion duration of fuels in the dual fuel engine.

Test designations: D100 – diesel fuel with 0% ethanol energetic fraction, DE1 – diesel fuel with 11,3% ethanol energetic fraction, DE2 – diesel fuel with 18,5% ethanol energetic fraction, DE3 – diesel fuel with 24,8% ethanol energetic fraction, DE4 – diesel fuel with 33,0% ethanol energetic fraction, and DE5 – diesel fuel with 55,0% ethanol energetic fraction.

In fig. 3 are presented energy doses for one engine cycle for various used fuels: D100 (1155 J/cycle), DE1 (1149 J/cycle), DE2 (1153 J/cycle), DE3 (1121 J/cycle), DE4 (1118 J/cycle), and DE5 (1172 J/cycle). For all analyzed cases the energy dose is almost the same and varies within limits of 1120-1170 J/cycle.

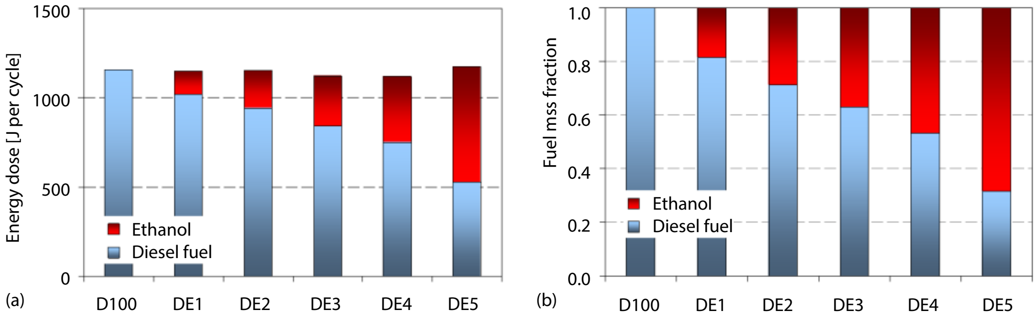

Figure 3. Energy doses of fuel (a) and fuels mass fraction (b)

Estimation of the experimental results

Analysis of combustion process in the internal combustion engine usually is carried out with the rate of heat release. Heat release rate ($\mathrm{d}Q/\mathrm{d}\varphi$) is calculated on the basis of the measured in cylinder pressure data and CA readings. The basis for determining the heat release rate is the First law of thermodynamics and the equation of state. After rearranging and simplifications, the heat release rate *vs.* CA is obtained in well-known form:

$$\frac{\mathrm{d}Q}{\mathrm{d}\varphi} = \frac{1}{\kappa - 1}\left[kp\frac{\mathrm{d}V}{\mathrm{d}\varphi} + V\frac{\mathrm{d}p}{\mathrm{d}\varphi}\right] \tag{1}$$

where κ is the ratio of specific heats, V – the cylinder volume, p – the cylinder pressure, φ – the crank angle. 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 heat release rate is termed as the net heat release rate. The cumulative net heat released is obtained by integrating eq. (1) over the CA φ [20].

The rate of pressure rise $\mathrm{d}p/\mathrm{d}\varphi$ is determined:

$$\frac{\Delta p}{\Delta \varphi} = \frac{p_i - p_{i-1}}{\varphi_i - \varphi_{i-1}} \tag{2}$$

where p is the in cylinder pressure and φ – the crank angle. A parameter indicating the performance of the engine is the IMEP which is determined on the basis of the instantaneous pressure in the cylinder. The IMEP is calculated:

$$IMEP = \frac{1}{V_d}\int_0^{720} p\frac{\mathrm{d}V}{\mathrm{d}} \tag{3}$$

where V_d [m^3] is the displacement volume, $V_d = 0.000573$ m^3. The cycle-by-cycle variation of combustion are usually investigated by experiments, and the measure of cyclic variability is represented by the coefficient of variation, $\mathrm{COV_{IMEP}}$. The $\mathrm{COV_{IMEP}}$ is calculated on the basis of IMEP data obtained from experimental results analysis. In this case, this factor is determined on the basis of 100 cycles. The COV_{IMEP} is a parameter that represents cyclic variability and it is defined as the standard deviation in IMEP divided by the mean IMEP:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{(IMEP)_{mean}}100 \tag{4}$$

where σ_{IMEP} is the standard deviation of IMEP. The mean IMEP was determined as:

$$IMEP_{mean} = \frac{\sum_{i=1}^{i=100} IMEP_i}{100} \tag{5}$$

where i is the number engine cycle. The COV_{IMEP} is directly related to the combustion stability.

Experimental study

Thermodynamic performance parameters

The analysis of the thermal cycle of the test engine was performed on the basis of an analysis of the indicator chart. During the tests ware analyzed: pressure and pressure rise variation, temperature, and heat release. The engine performance indicators: IMEP, indicated thermal efficiency (ITE), brake specific fuel consumption were analyzed as well. The diesel fuel was taken as a reference for other cases.

In fig. 4 are presented pressure variation for all analyzed cases. The maximal value of pressure in the combustion chamber of the test engine was equal to 70 bar for DE4 (with 33% of EF energetic fraction). With the increase in EF fraction, the deterioration of combustion process was indicated. It is clearly visible that injected EF into intake manifold causes pressure drop in the compression stroke. This phenomenon is due to the cooling effect of evaporated ethanol during the intake stroke. For next analyzed EF fraction (55%) it was noticed significant decrease in peak pressure and clearly higher ignition delay period.

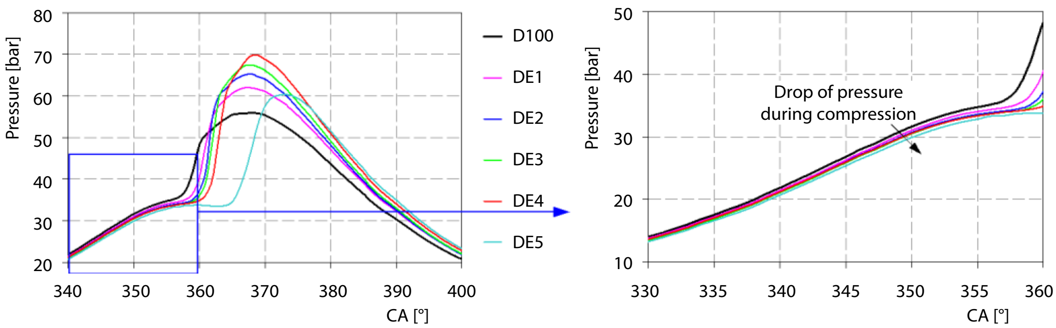

Figure 4. Pressure variation

In fig. 5 it is visible negative loop which indicted that for that engine setting it cause drop in engine performance parameters. The tests were performed at constant start of diesel fuel injection. The easiest way to get rid of this unfavorable effect is to advance injection of diesel fuel.

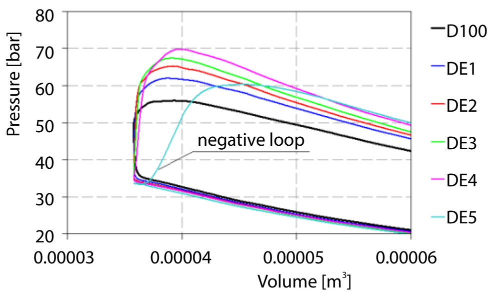

Figure 5. Closed indicated chart

In fig. 6 are presented temperature variation for all analyzed cases. The cooling effect is visible as in case of pressure variation in fig. 4.

This cooling effect has in consequence impact on ignition delay. The drop in temperature in the time of injection beginning of diesel fuel was near to 60 K for DE5 fuel. It is visible that despite the temperature drop during intake and compression stroke the peaks temperature were higher in comparison with combustion of reference fuel.

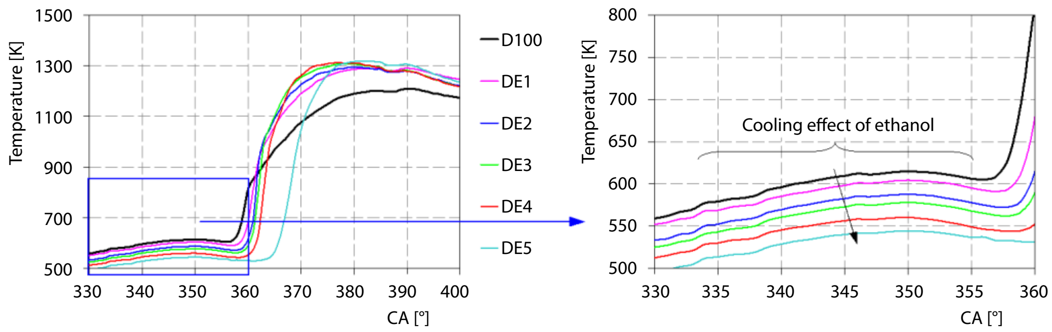

Figure 6. Temperature variation

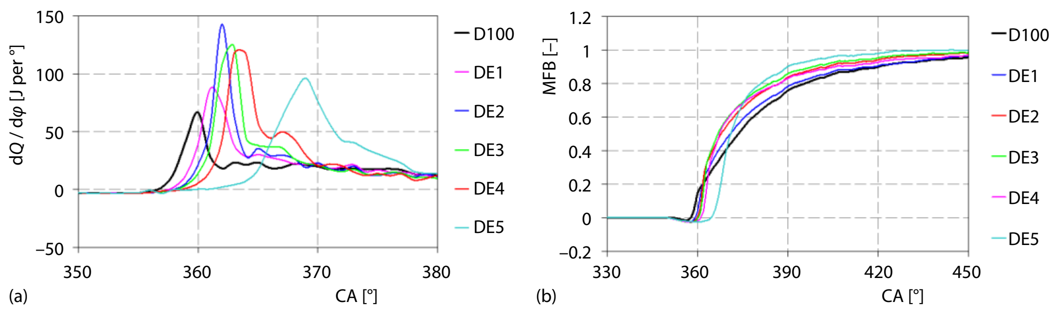

Figure 7. Heat release rate (a) and MFB (b)

In fig. 7 are presented variation of heat release rate (dQ/dφ) for all analyzed cases. It can be stated that with EF in burned fuels the heat release rate increases. In case of DE2 fuel (18.5% of EF) reached the highest value equal to 143 J/deg and it was two times higher in comparison with combustion of diesel fuel (67 J/deg). On the basis of heat release variation were determined combustion phases: ignition delay, combustion duration. The combustion delay period depends on chemical and physical processes. Before the energy of fuel injected into cylinder is released the physical and chemical processes must take place. The physical processes are atomization of the liquid fuel, vaporization of droplets and mixing with air. The chemical processes are the precombustion reactions which causes autoignition of air-fuel mixture. On the basis of (mass fraction burn) MFB variation it can be stated that with the increase in EF participation during combustion increase the ignition delay but the main combustion phase is shortened. This cause larges values of pressure rise and in consequence engine operation becomes hard.

In a Diesel engine, there are two combustion phases, the premixed combustion phase and diffusion combustion phase. The oxygenated fuel causes the change in the ratio of these phases. The share of the combustion kinetics and decreased the share of combustion diffusion. Intensification is the first phase of combustion. On the other hand, ethanol is characterized by higher carbon content and it is more energy fuel than for example methanol. Ethanol has low stoichiometric air-fuel (A/F) ratio, high oxygen content and high H/C ratio may be beneficial at improving the combustion process [7]. Based on the literatures review, it is clear that alcohol fumigation in a Diesel engine affects the BTE in two ways. Alcohol fumigation decreases the BTE at lower engine load condition and increases the BTE at medium and higher engine load condition [20]. Alcohol has lower cetane number which increases the ignition delay hence energy is released within a very short time, resulting the reduction in the heat loss from the engine as there is no sufficient time for transferring heat through the cylinder wall to the coolant.

In fig. 8 are presented IMEP and ITE for all analyzed cases. It is visible that with the increase in EF participation in combustion processes the IMEP increases as well. The highest value of ITE obtained for 33% of EF and it was equal 38%.

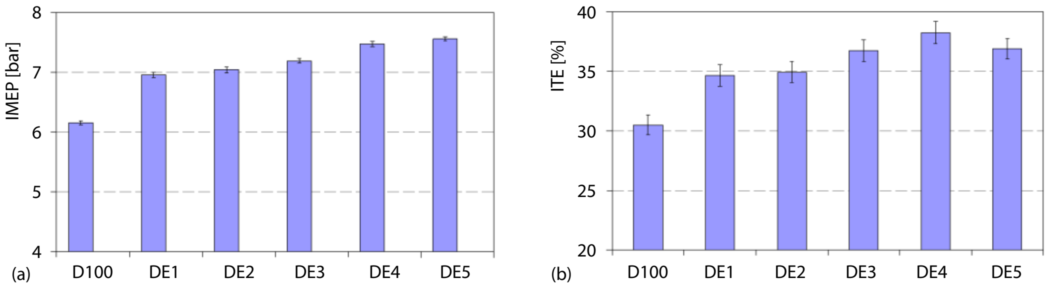

Figure 8. (a) IMEP and (b) ITE

In fig. 9 are presented specific fuel consumtion (SFC) and COV_{IMEP} for all analyzed cases. Up to 33% of EF fraction, the SFC was near at the same level, at 55% of EF fraction in fuels the SFC increased. This increase was due to deterioration of combustion process. The dominant influence on the combustion process has the high value of the heat of evaporation of ethanol. There was no noticeable effect of ethanol on unrepeatability of IMEP which was near at the same level of 3%. Diesel engine operated in dual fuel mode, consumes more fuel to maintain same thermal efficiency compared to diesel fuel. Alcohol has the higher heat of evaporation compared to diesel fuel. Thus, less amount of heat is extracted during combustion process that must be compensated with higher fuel consumption [20].

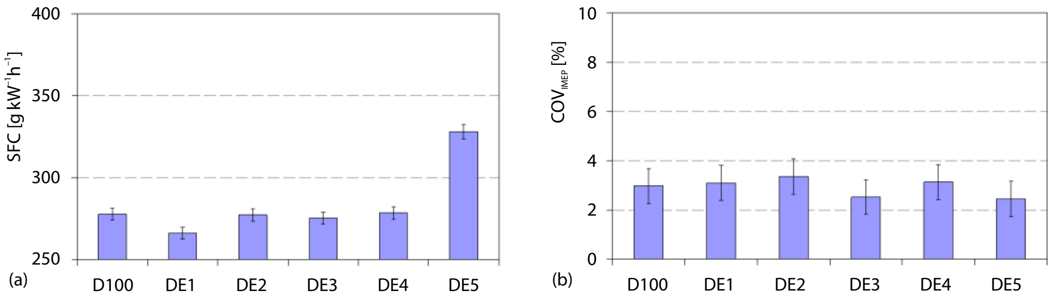

Figure 9. (a) SFC and (b) unrepeatability of IMEP (COV_{IMEP})

Emission characteristics

The Diesel engines are characterized by NO_x and soot emission. The NO_x in the exhaust emissions contains NO and NO_2 [20]. The NO_x formation rate has a strong relationship to temperature. The high temperature in the combustion chamber of the engine during the relatively short phase of premixed combustion is characterized by high NO_x formation rate. On the other hand, the decreasing temperature in the combustion chamber causes the lower thermal efficiency of the engine and cause the increase of THC emission. In the Diesel engines operation which decreased NO_x emission simultaneously increased soot emission. Emission of NO_x is combined with the premixed phase of combustion process but the soot emission is combined with diffusion phase. A proper control of these two phases of combustion can leads to the low emission of NO_x and soot. By the temperature over 2200 K, the production rate of NO_x is doubled for every 90 K. In dual fuel engines the value of peak temperature can be reduced by using fuels with the high value of the heat of evaporation. This property of the fuel causes a decrease of the temperature of in-cylinder charge before the start of the combustion and as a consequence decreases maximum temperature during the burning process [25-27]. Three mechanisms are involved in the formation of NO_x: thermal, prompt and nitrous oxide. According to the thermal mechanism, the reaction between nitrogen and oxygen occurs at high temperatures inside combustion chamber according to the Zeldovich mechanism. The NO_x formation occurs at temperatures above 1500 °C, and the rate of the formation increases rapidly with increasing temperature [20, 26]. According to the prompt mechanism, fuel bound nitrogen is one of the significant parameters for the formation of prompt NO_x [28].

In fig. 10 are presented the results of NO_x and THC emission obtained for test engine. It was stated that with small EF ratio NO_x emission increases slightly what is confirmed in others researchers [4-5, 7]. From 20% of EF the specific NO_x emission started to decreases. The highest value of NO_x emission was reached at DE1 and it was equal 1.83 g/kWh and the lowest value at DE5 equal 0.97 g/kWh. In case of THC emission, it was noticed that EF cases slight increase in specific emission. For powering by pure diesel fuel THC emission was equal 0.57 g/kWh but at DE5 it increased up to 0.73 g/kWh. The THC from Diesel engines comes primarily from the fuel trapped in the injector at the end of the injection that later diffuses out, the fuel mixed into air surrounding the burning spray so lean that it can not burn, and the fuel trapped along the walls in crevices, deposits, or oil due to impingement by the spray. The increase in HC with the addition of EF is due to the higher heat of evaporation of the diesel in air-ethanol lean mixture, which causes slower evaporation and thus slower and poorer fuel-air mixing. Also, the increase in the ignition delay by ethanol increases the HC emissions. Consequently, high HC emissions are exhausted in the overall engine operating conditions [29, 30].

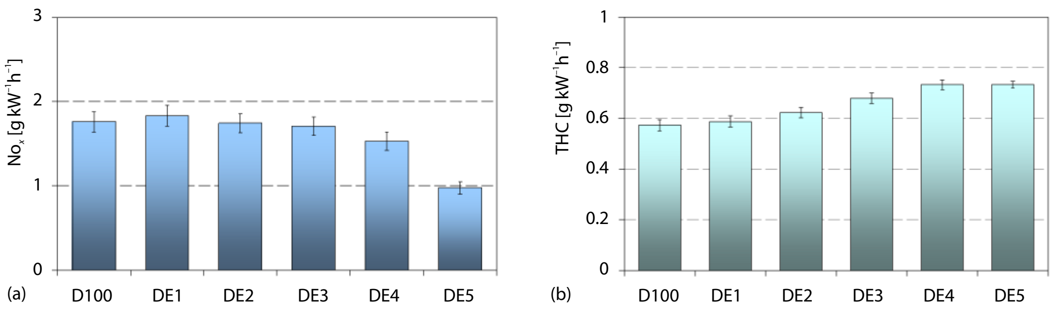

Figure 10. Emissions of (a) NO_x and (b) THC

The other harmful gaseous emission from Diesel engine is CO. Formation of CO is the result of in complete combustion. The CO emissions are an intermediate product in the combustion of HC fuels, resulting from incomplete combustion. Therefore, CO emissions are largely dependent on the air-fuel ratio relative to the stoichiometric mixture. The fuel-rich combustion necessarily produces CO emissions, and it increases nearly linearly with deviation from stoichiometry [20, 31]. The wall-wetted fuels can not take part in the combustion reaction. Therefore, the combustion temperature is relatively low, and the oxidation of CO is not active. This result is related to the decrease of the NO_x emissions. The oxygenate components in a fuel reduce the formation of soot precursors and decrease the carbon concentration, which reduces the carbon-carbon bonds (the cause of the soot formation) [32, 33].

If the in cylinder temperature during combustion process is not sufficient to support the complete combustion then transformation of CO to CO_2 has not occurred. It can be caused by evaporated alcohol which decreasing peak temperature in-cylinder, fig. 11. In case of CO and CO_2 emission, the lover emissions obtained with the participation of EF. On the basis of these components, it can be concluded of combustion process efficiency. With the increase in EF participation the CO emission started to increase. In case of CO_2 emission for all analyzed range of EF participation, obtained the slight decrease in this component. Nowadays it attaches great importance to the reduction of CO_2 emissions that dual fuel technology using EF fits into this trend [22-24].

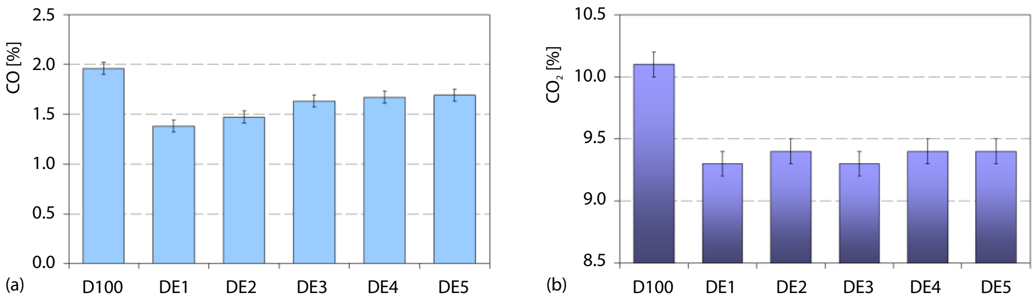

Figure 11. Emissions of (a) CO and (b) CO_2

During the combustion process, air/alcohol mixture gets trapped in crevices, deposits and quench layer in the engine. Alcohol also tends to lower the in-cylinder gas temperature which might be not able to ignite the trapped alcohol during expansion stroke [20].

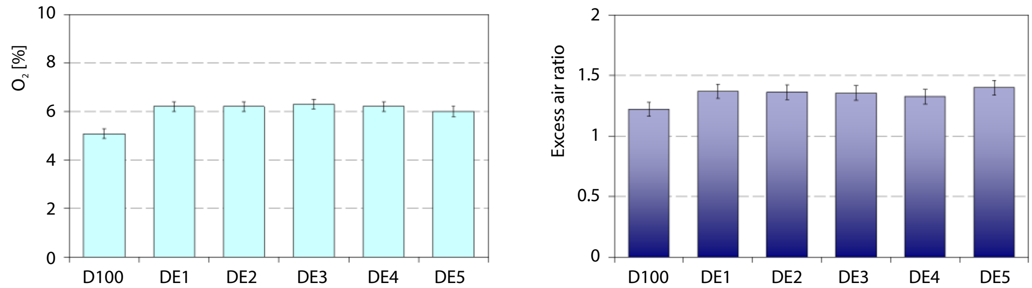

Figure 12. Oxygen fraction in exhaust gases and excess air ratio

In fig. 12 are presented oxygen content in exhaust gasses and mean excess air ratio. With the increase of EF participation in the energetic dose of fuels the excess air ratio increased. It is due to oxygen content in EF. Ethanol is included in the group of oxygenated fuels (C_2H_5OH) and ethanol in mass contains near to 35% of oxygen. This is reflected in stoichiometric air-fuel ratio of ethanol which is more than 60% less than for diesel fuel. The unused oxygen in the combustion process is reflected in the exhaust as O_2 or excess air ratio.

Conclusions

On the basis of investigations can be stated that EF could be utilized in the compression ignited engines with quite large energetic share using dual fuel technology.

Conclusions on the evaluation of engine performance parameters are as follows.

- With the increase of EF up to 33% in energetic dose delivered to the engine the peak pressure increased, after exciding this fraction was noticed decrease in this value.
- The injection of EF into intake manifold cause decrease in temperature of intake charge.
- In case of 55% of EF fraction on the p-V chart, the negative loop was visible.
- In case of 18% of EF fraction obtained the pressure rate over 1 MPa/deg – the work of engine started to be hard.
- With the increase in EF fraction, the ignition delay increased but the combustion duration is shortened.
- The highest value of ITE was obtained for 33% of EF fraction.
- The EF does not affect the coefficient of IMEP unrepeatability.

Conclusions on exhaust emissions are as follows.

- Generally the increase in EF fraction causes the specific NO_x emission decreases.
- With the increase in EF fraction in energetic dose, the specific THC emission slightly increased.
- The EF participation in combustion process causes decrease in CO and CO_2 emission.

Dual fuel technology represents the most efficient way of using alcohol in the Diesel engine. There is a possibility to utilize EF with the increase in BTE at high engine loads.

Acknowledgment

Research was financed by the Ministry of Science and Higher Education of Poland from the funds dedicated to scientific research No. BS/PB 1-103-3030/2017/P.

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Paper submitted: June 6, 2017
Paper revised: December 24, 2017
Paper accepted: December 24, 2017