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Comparative Analysis of the Combustion Stability of Diesel-Methanol and Diesel-Ethanol in a Dual Fuel Engine

Arkadiusz Jamrozik *, Wojciech Tutak, Renata Gnatowska and Łukasz Nowak

Faculty of Mechanical Engineering and Computer Science, Czestochowa University of Technology, 42-201 Czestochowa, Poland; tutak@imc.pcz.pl (W.T.); gnatowska@imc.pcz.pl (R.G.); luknowak@imc.pcz.pl (Ł.N.)

* Correspondence: jamrozik@imc.pcz.pl

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Abstract: The co-combustion of diesel with alcohol fuels in a compression ignition dual fuel engine is one of the ways of using alternative fuels to power combustion engines. Scientific explorations in this respect should not only concern the combustion process in one engine cycle, which is most often not representative for a longer engine life, but should also include an analysis of multiple cycles, which would allow for indicating reliable parameters of engine operation and its stability. This paper presents experimental examinations of a CI engine with a dual fuel system, in which co-combustion was performed for diesel and two alcohol fuels (methanol and ethanol) with energy contents of 20%, 30%, 40% and 50%. The research included the analysis of the combustion process and the analysis of cycle-by-cycle variation of the 200 subsequent engine operation cycles. It was shown that the presence and increase in the share of methanol and ethanol used for co-combustion with diesel fuel causes an increase in ignition delay and increases the heat release rate and maximum combustion pressure values. A larger ignition delay is observed for co-combustion with methanol. Based on changes in the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}) and the function of probability density of the indicated mean effective pressure (f(IMEP)), prepared for a series of engine operation cycles, it can be stated that the increase in the percentage of alcohol fuel used for co-combustion with diesel fuel does not impair combustion stability. For the highest percentage of alcohol fuel (50%), the co-combustion of diesel with methanol shows a better stability.

Keywords: co-combustion; dual fuel; combustion stability; coefficient of variation of IMEP; probability density of IMEP

1. Introduction

Compression ignition engines (CI) are commonly used in transport, industrial machines and in the agricultural economy due to their long life time durability. The CI engines are a subject of great criticism due to their emissions output. The biggest problem in these engines is the simultaneous reduction of NO_x and soot emissions [1,2]. Emissions of soot and NO are opposed to each other. In paper [3] authors presented results of an investigation of the emissions of dual fuel CI engine. Authors stated that in all analyzed cases, conditions were achieved in which soot emission starts to increase again with lower NO emissions. Emissions of NO_x and soot should be reduced because these are harmful to human health and environment as well [4]. Another motivation for these activities is the European Union Directive 2009/28/EC obliging the use of a 20% share of renewable biofuels in overall transport and diesel fuel consumption by 2020 [5]. One of the reasons for using biofuels is their smaller negative impact on the environment through lower greenhouse gas emissions, while another is in order to develop diversification, which can increase energy independence. Increased consumption of biofuels causes reduced consumer reliance on imported fossil fuels and the depletion of crude oil reserves [6,7]. Fuelling of compression ignition internal combustion engine with unconventional fuels seems to be an interesting alternative that allows for increasing the use of biofuels in the transport and energy sectors. Biofuels such as alcohols and biodiesel have been proposed as alternatives to fossil fuels for internal combustion engines [8]. Combined with diesel fuel or biodiesel, alcohol fuels may represent a valuable fuel for compression ignition engines. Alternative liquid fuels are used for powering compression ignition engines in two modes. The first one is powering engines by using a diesel-alternative fuel blend and second one is a co-combustion of fuels using dual fuel technology [9–12]. The main disadvantage of using a fuel mixture is the limited share of alcohol due to the phase separation [13,14]. In a dual-fuel engine, additional fuel is supplied to the intake system by low-pressure injection. Such produced air-alcohol mixture enters into cylinders during the inlet valve opening period. The engine in this mode of operation can burn alcohol to its 90% energy share [15,16]. Recently, there has been a growing interest in using organic oxygen compounds in engines of this type, especially alcohols [17]. Some disadvantages of alcohols compared to diesel fuel include a relatively high compression ignition temperature and a tendency for the decreasing of charge in-cylinder temperature due to the high value of heat of vaporization. Furthermore, the heating value of ethanol and methanol is lower compared to diesel. Thus, to provide the same energy of delivered fuel to the engine cycle, higher amounts of alcohol are needed [15,16]. The reduction of exhaust emissions of the compression ignition engine can be obtained by controlling the combustion process and by using alternative fuels [18]. Therefore, the substitution of diesel fuel with alternative fuels such as alcohol fuels represents a good approach to solve the environmental problems. With its properties, alcohol can also have a positive effect on NO_x and particulate matter emissions [19]. The diesel fuel consists of various hydrocarbons, the ratio of carbon to hydrogen (C/H) is high, and there is a tendency to form soot under fuel rich combustion conditions. The particulate matter is created in the internal combustion engine (mainly in the self-ignition engine), when combustion causes local or temporary lack of air, and more specifically oxygen. The additional oxygen delivered in the alcohol fuel promotes the burning of coal and reduces the formation of PM. Due to their organic nature, alcohols contain up to 50% oxygen in their structure, allowing for widespread oxidation of fuel, resulting in more complete combustion. One of the alcohols that can be used for co-combustion with diesel is ethanol. Ethanol can be produced from raw materials of plant origin and therefore can be considered a fully renewable fuel [17]. Methanol can be produced from such resources as natural gas, coal and various renewable biomasses. The main disadvantages of using ethanol in diesel engines include a low cetane number and problems with solubility in the petroleum diesel [20]. Among these alternative fuels, methanol also represents a promising alternative fuel [21]. Since methanol has a higher latent heat of vaporization than ethanol, methanol has the potential to reduce NO_x due to a cooling effect on the charge, leading to lower in-cylinder temperatures [20]. Another advantage of using alcohol is an oxygen content in methanol structure which reduces exhaust fume emissions. Ethanol or ethanol mixed with diesel leads to the occurrence of phase separation between these fuels. Phase separation occurs each time because diesel fuel and methanol or ethanol are immiscible [22]. Therefore, a better solution is to use alcohols in dual fuel mode. In the available literature, there are many publications about the co-combustion of ethanol or methanol with diesel fuel in internal combustion engine. The papers [14,23] presented the results of examinations of compression ignition engine powered by blends of diesel fuel with various alcohols. The authors demonstrated that diesel-ethanol blends could be used in the compression ignition engines with quite a large fraction. Using ethanol fuel in a blend with diesel fuel is supplied additional oxygen for the combustion, which can cause improvements in the overall combustion process. Another statement was that the combustion process of alcohol-diesel fuels blend in IC diesel engine takes place in a shorter time than in the diesel engine fuelled by pure diesel fuel. With an increase of the ethanol fuel fraction, the ignition delay increased. With the increase in the proportion of ethanol, the combustion duration decreases [14,20,24]. The paper [25] presented results of investigation of the combustion process in compression ignition engine powered by a diesel-methanol blend. They stated that up to a 30% methanol fraction, the combustion process occurred as it would in engines powered by pure diesel. A further increase in the alcohol content led to a significant decline in cylinder pressure and the abnormal combustion process. For mixtures up to 40% of methanol fraction, it did not exceed the value of COV_{IMEP} equal to 10% [25]. In another paper [24] results are presented which show that the use of dual fuel technology gave a satisfactory performance in a standard diesel engine without any modification in the engine. In addition, emissions of carbon monoxide, unburned hydrocarbon and smoke were lower, whereas nitrogen oxides emissions were almost equivalent with diesel. In paper [24], results are presented of varying the ethanol ratio on engine performance and emissions of the dual fuel combustion with various load conditions. The test engine was a single-cylinder diesel engine equipped with two direct injectors operated with constant speed. The authors stated, inter alia, that NO_x and PM emissions decreased with increasing ethanol fraction. Additionally, they stated that for low and high loads conditions, the ethanol fraction is limited due to insufficient ignition energy at a low load and at full load due to a high in-cylinder pressure rise. The engine can also burn so-called hydrated ethanol [26]. It seems that this is a good solution because the removal of water from fermented products of ethanol consumes lots of energy. In the paper [27], results are presented of hydrous ethanol combustion in a dual fuel engine, including impacts on the combustion and emissions. The thermal efficiency reduced significantly at the ethanol purity of 60%. The reduction of ethanol purity can reduce NO_x emissions but CO and THC emissions increase. In paper [4] results are presented of investigation of cylinder-to-cylinder pressure variation in a diesel engine fuelled with diesel/methanol using dual fuel technology. Research was carried out on the 4-cylinder turbocharged direct injection diesel engine. In the experiment, methanol was injected into the inlet duct by two low-pressure injectors, and directly mixed with intake air before entering the engine. The diesel fuel was directly injected into cylinder. The experimental results showed that the unevenness degree of engine increases with the increase of methanol percent under various engine loads. The uniqueness of pressure in subsequent cycles increases with the methanol share under a low engine load, and the COV_{pp} (coefficient of variation of peak pressure) curves vary very little under the low methanol percentage under 75% and 100% engine loads, but a significant rise appeared when the methanol substitution percent further increased [4]. Co-combustion of alternative fuels such as alcohols causes lowering in cylinder temperature which then affects the value of the ignition delay due to higher value of heat of evaporation of alcohols. In the paper [28] results are presented of investigation of in-cycle pressure fluctuation intensity under long ignition delay conditions. Authors stated that under low temperature charge conditions, the difference in peak pressure between the average cycle and cycles showed low fluctuation intensity. The average intensity of the pressure fluctuations showed an increase with increasing amounts of the premixed combustion phase. This resulted in higher cycle-to-cycle variations under these conditions [28]. Other works presented comparative analysis of the impact of fuel on the uniqueness of engine cycles. In the paper [29], results are presented of investigation of cycle-to-cycle variations of peak pressure in compression ignition engine powered by diesel and biodiesel fuel. The results show that at a lower load, in-cylinder pressure variations for biodiesel were lower compared to mineral diesel fuel. On the other hand, at medium and high loads, biodiesel dominated the peak n-cylinder pressure variations [29]. The cycle of unrepeatability influences emissions variation as well. In the paper [30], results are presented of assessment of cyclic variations on NO emissions of a direct injection diesel engine. Ono of statements was that the intensity of the pressure fluctuations increases with increasing ignition delay. Additionally, the authors stated that in cases with a significant diffusion combustion phase, cycles with pressure fluctuations showed higher-than average NO concentrations, with the intensity of fluctuation determining the increase in NO. Cycle-to-cycle variations in compression ignition engines are undesirable due to negative impacts on efficiency and higher emission. This undesirable phenomenon causes power output limitations.

This work aims to increase knowledge of influence of type of alternative fuel co-combusted with mineral diesel fuel on operation parameters and cycle-to-cycle variations of the CI engine. The results of experiments of co-combustion of diesel fuel with alcohols in the compression ignition engine working in the system of dual fuel are presented in these studies. This paper presents experimental examinations of a CI engine with dual fuel system, in which co-combustion was performed for diesel and two alcohol fuels (methanol and ethanol) with energy contents of 20%, 30%, 40% and 50%. The research concerned the analysis of the combustion process and the analysis of non-repeatability for 200 subsequent engine operation cycles. In the available literature there is not enough information about the influence of the share of alcohol on the stability of the combustion process. However, a lot of information is provided about the deterioration of the combustion by the share of alcohol. In this work, we showed the effect of alcohol on combustion process stability using indicators such as COV_{IMEP} and the probability density of IMEP.

2. Materials and Methods

2.1. Cycle-to-Cycle Variability Determination

During the observation of the course of the indicator pressures in successive engine operation cycles, it is easy to notice differences in the values of maximum pressure and indicated mean effective pressure in subsequent cycles. Cycle variability is an undesirable feature of combustion in internal combustion engines. Cyclic variations in diesel engines are undesirable since they are understood to lead to lower efficiency and higher emissions and to power output limitations [28]. They are caused by varied conditions in the cylinder from one combustion cycle to the other, under nominally identical operating conditions. Due to air pollution and increasingly stringent emission standards around the world, mitigation of the effects of cycle variability can reduce fuel consumption and minimize the emissions of harmful compounds in IC engines. Furthermore, cycle variability limitation allows for better engine optimization, i.e., it is possible to achieve efficiency close to optimal conditions. The main sources of the non-repeatability of engine cycles include:

- flow processes occurring in the engine cylinder,
- variability of air-fuel ratio in individual cycles,
- uneven composition of the combustible mixture in the cylinder due to incomplete mixing of air, fuel and residual gas,
- ignition non-repeatability.

The assessment of the of non-repeatability degree of the maximum pressure and indicated mean effective pressure of the thermal cycle occurring in the cylinder of the internal combustion engine can be conducted by analysing several dozen or several hundred consecutive engine cycles.

The obtained set of maximum pressures can be divided into classes k, from the interval:

$$k \in \langle p_{max\,A}, p_{max\,B} \rangle, \tag{1}$$

where $p_{max\,A}$ is the smallest value of maximum pressure in set of cycles, $p_{max\,B}$ is the maximum value of the maximum pressure in set of cycles.

The average value of maximum pressure pmax, determined on the set of pressure:

$$\overline{p}_{max} = \frac{1}{N}\sum_{i=1}^{N} p_{max\,i}, \tag{2}$$

where N is the cycle index, $p_{max\,I}$ is maximum pressure in individual cycles.

One of the most popular criteria for the evaluation of the correct operation of combustion engines is non-repeatability of the work cycles. The coefficient of non-repeatability of maximum pressure COV_{pp}, expressed in percentages and calculated as a ratio of standard deviation of the maximal pressure to its mean value from multiple engine operation cycles, was adopted as a measure of non-repeatability [4,31].

The coefficient of variation of peak pressure (COV_{pp}) is defined as:

$$COV_{pp} = \frac{\sigma_{pmax}}{\overline{p_{max}}} \cdot 100\%, \quad (3)$$

The standard deviation of peak pressure:

$$\sigma_{pmax} = \sqrt{\frac{1}{N}\sum_{i=1}^{N} (p_{max\ i} - \overline{p_{max}})^2}, \quad (4)$$

where $\overline{p_{max}}$ is the mean value of peak pressure of N cycle $p_{max\ i}$.

The indicated mean effective pressure is evaluated based on the recorded changes in the cylinder pressure and represents one of the indices that characterize operation of combustion engines in terms of the opportunities to ensure high and expected functional performance.

Indicated mean effective pressure for a single engine cycle:

$$IMEP_i = \frac{1}{V_d}\int_0^{720} pdV, \quad (5)$$

The average value of IMEP:

$$\overline{IMEP} = \frac{1}{N}\sum_{i=1}^{N} IMEP_i, \quad (6)$$

where $IMEP_i$ is indicated mean effective pressure in individual cycles.

The uniqueness of IMEP can be determined in the same way as the uniqueness of the maximum pressure value.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \cdot 100\%, \quad (7)$$

The standard deviation of IMEP:

$$\sigma_{IMEP} = \sqrt{\frac{1}{N}\sum_{i=1}^{N} (IMEP_i - \overline{IMEP})^2}, \quad (8)$$

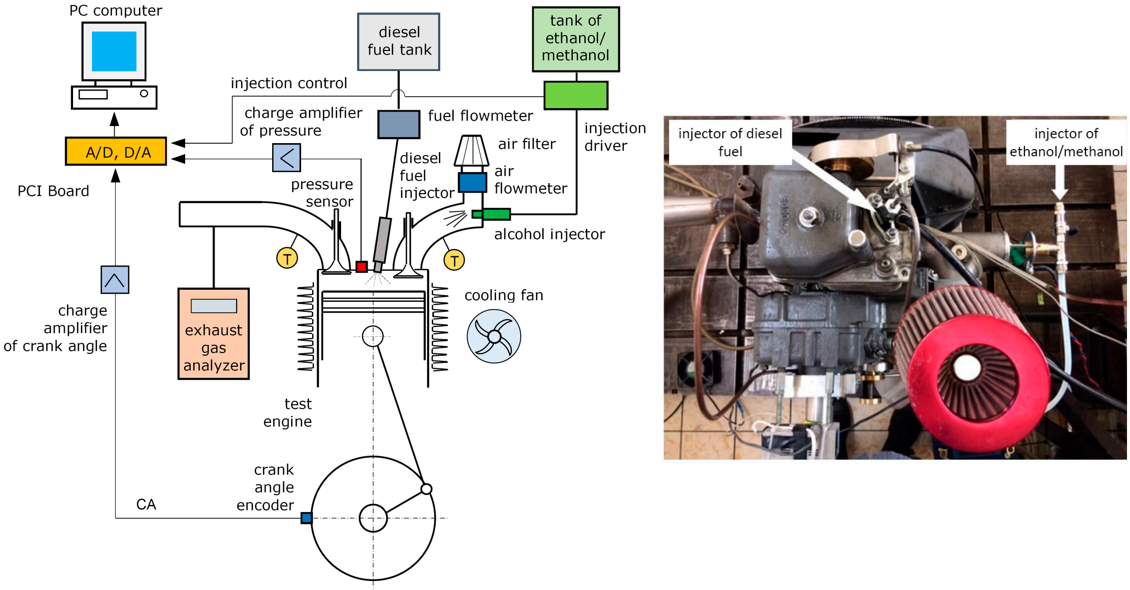
The probability density function allows for expressing the probability of obtaining or occurrence of a concrete value of the analyzed parameter. Distribution of probability density of the indicated mean effective pressure represents a repeatability index (index of probability of occurrence) of individual IMEP values obtained for multiple analyzed cycles of the test engine operation. This function can also be used as an indicator to assess the stability of operation of the internal combustion engine. Among other things, it indicates the frequency of occurrence of the most frequently repeated IMEP value, which is close to the mean value, indicating the repeatability of subsequent engine operation cycles. The IMEP probability density function has the profile of a normal distribution (Gaussian distribution), in which the density function is symmetrical in relation to the mean value of the distribution [32].

The probability density of the indicated the mean effective pressure:

$$f(IMEP) = \frac{1}{\sigma_{IMEP}\sqrt{2\pi}} \exp\left(\frac{-(IMEP_i - \overline{IMEP})^2}{2\sigma_{IMEP}^2}\right), \quad (9)$$

2.2. Test Stand

The experimental examinations of the process of co-combustion of diesel with alcohol fuels used the test engine based on a single-cylinder, four-stroke diesel engine 1CA90 manufactured by Andoria (Poland), with compression rate of 17 and engine displacement of 573 cm^3. This engine is a stationary two-valve unit with a vertical arrangement of cylinders and an air cooling system with an axial fan. The engine worked at constant rotational speed of 1500 rpm and constant injection angle of 17° crank angle before TDC. The engine was with the one single injection and was running at a steady-state full load. In order to change and ensure the optimal position of the initial injection angle, necessary changes were made in the design, which included changing the design of the camshaft. The engine was equipped in a high-pressure installation of fuel supply with direct injection of diesel fuel to the cylinder. The test engine was modernized and equipped with an additional installation of alcohol fuel supply to the intake manifold. The research stand, with its main component being the engine 1CA90, was equipped in the necessary control and measurement apparatus that allowed e.g., for indication and recording of the necessary parameters of the engine work. Engine indication included measurements and digital recording and analysis of fast-changing pressure in the combustion chamber of the research engine. The engine indication system included Kistler 6061B piezoelectric pressure sensor, charge amplifier Kistler 5011B, crank angle sensor and the module for data acquisition with the A/D converter (Measurement Computing USB-1608HS) (Figure 1). Table 1 presents the most basic technical data for this engine.



(**a**) (**b**)

Figure 1. Diagram of the experimental setup (**a**) and view from the top of the test engine (**b**).

Table 1. Main engine parameters.

Parameter	Value	Unit
Displacement volume	0.000573	m^3
Bore	90	mm
Stroke	90	mm
Compression ratio	17:1	-
Rated power	7	kW
Crankshaft rotational speed	1500	rpm
Diesel injection pressure	21	MPa
Alcohol injection pressure	0.3	MPa
Injection timing	17	deg bTDC

The test bench consisted of:

- compression ignition engine 1CA90,
- digital acquisition system for analysis of fast changing data consisted of: pressure transducer, Kistler 6061B of sensitivity: ±0.5%, charge amplifier, Kistler 5011B of linearity of FS < ±0.05%,

data acquisition module, crank angle encoder, resolution of 360 pulses/rev, Measurement Computing USB-1608HS–16 bits resolution of sampling frequency 20 kHz,

- exhaust gas analyzer Bosch BEA 350 for:

THC:	range 0–9999 ppm vol,	accuracy: 12 ppm vol,
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,
NO_x:	range: 0–5000 ppm,	According to OIML R99 Ed. 1998.

- Roots flowmeter Common CGR-01, range 0.2–650 m^3/h, accuracy < ±1%.
- temperature sensor TP-204K-1b-100-1.5, range +200 °C, resolution 0.1 °C.

2.3. Fuels

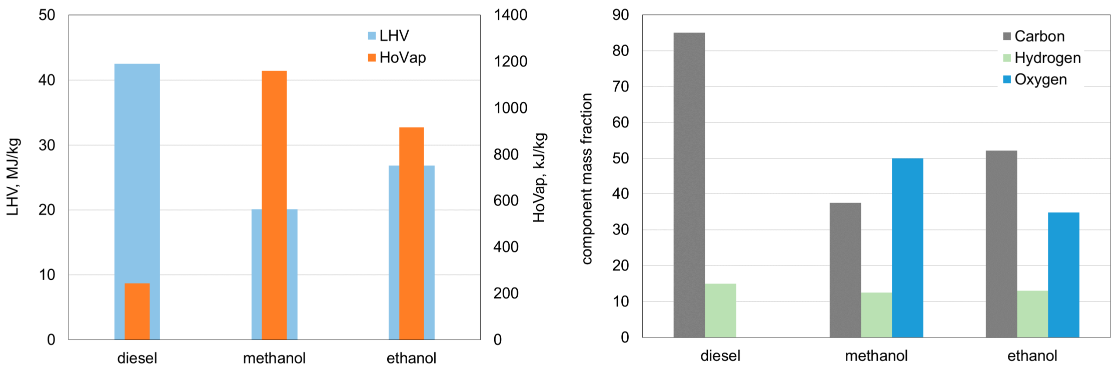
Diesel fuel and technical alcohols (methanol, ethanol), were used in the study. Due to the low value of the cetane number and difficulties with compression ignition, alcoholic fuels, i.e., methanol CH_3OH and ethanol C_2H_5OH, viewed as unconventional engine fuels, cannot be used as individual fuels in CI engines and, therefore, have been used only as fuels for SI engines. Nowadays, although methanol and ethanol belong to strongly oxygenated alcohols (with high content of oxygen in a fuel particle) and are characterized by a lower heating value compared to gasoline or diesel fuel, a continuous increase in interest in these alcohols as potential replacement fuels is being observed compared to conventional fuels, as well as in compression-ignition engines. Mass percentage of oxygen in structure of these alcohols is ca. 50% for methanol and ca. 35% for ethanol. High content of oxygen can have a positive effect on combustion in the engine's combustion chamber since this oxygen is characterized by higher activity compared to the molecular oxygen contained in air. It is conducive to intensification of combustion and contributes to the reduction in exhaust gas smokiness. Similar to ethanol, methanol is characterized by insignificant heating value compared to conventional fuels. The heating value of pure methyl alcohol (19.5 MJ/kg) is only 45%, whereas for ethyl alcohol (26.9 MJ/kg), this means 60% of the heating value of the diesel fuel or gasoline. Both alcohols are characterized by a high heat of vaporization, equal to 1100 kJ/kg for methanol and 840 kJ/kg for ethanol (Table 2). In compression ignition engines, this property reduces mixture temperature and improves the coefficient of cylinder filling, whereas in engines with compression ignition, it contributes to the ignition delay and impacts on the increased rate of pressure rise and can magnify the so-called hard operation of the engine. Hard operation is defined as engine work at a large pressure increase. For diesel engines, a limit of 1 MPa/deg is usually assumed. Alcohol fuels offer opportunities to reduce the use of fossil fuels in CI engines and to increase percentage of biofuels in the transport and energy sectors, where combustion engines are often used. Methyl and ethyl alcohols, due to relatively low costs and easy production, belong to the most popular fuels for combustion engines. Methanol can be manufactured on the industrial scale from broadly available resources, including both non-renewable (natural gas, coal) and renewable (gas generated from biomass or municipal waste) sources. Ethanol is typically obtained during alcohol fermentation of sugars, with sugar sources being mainly cereals (wheat, maize, barley) and potatoes.

Figure 2a shows the two most important features of the analyzed fuels from the standpoint of combustion engine operation. These are the calorific value, which determines engine efficiency, and heat of vaporization, which, in the case of compression ignition engines, impacts on the processes that initiate the ignition of the combustible mixture, causing a change in compression ignition delay and determining engine performance and levels of emissions. Figure 2b shows the molecular composition of diesel, methyl and ethyl alcohol. Hydrogen content in all the fuels is comparable. The main difference between them results from oxygen and carbon content. With its molecular structure, diesel fuel does not contain oxygen, whereas the most oxygenated alcohol (methanol) has a 50% oxygen content (Table 2).

The study examined the processes of co-combustion of diesel with methanol and diesel with ethanol. The percentages of alcohol fuels were 20%, 30%, 40% and 50%. The following symbols were used during the analysis: for pure diesel fuel: D100, for co-combustion of diesel fuel with methanol: DM20, DM30, DM40, DM50 and, for co-combustion of diesel with ethanol: DE20, DE30, DE40, DE50. The numbers in the symbols concern the percentage of alcohol energy dose in total fuel energy (alcohol and diesel fuel) supplied to the engine.

Table 2. Fuels properties [33–36].

Properties	Unit	Diesel	Methanol	Ethanol
Molecular formula	-	$C_{14}H_{30}$	CH_3OH	C_2H_5OH
Molecular weight	g	198.4	32.04	46.068
Cetane number	-	51	3	8
Research octane number	-	15–25	136	129
Boiling point	K	453–643	338	351
Liquid density	kg/m^3	840	796	789
Lower heating value (LHV)	MJ/kg	42.5	19.5	26.9
LHV of stoichiometric mixture	MJ/kg	2.85	2.68	2.69
Heat of evaporation	MJ/kg	243	1100	840
Auto-ignition temperature	K	503	736	698
Stoichiometric air-fuel ratio	-	14.6	6.45	9.06
Viscosity (at 40 °C)	cSt	4.59	0.65	1.52
Carbon content	%	85	37.5	52.2
Hydrogen content	%	15	12.5	13
Oxygen content	%	0	50	34.8



(**a**) (**b**)

Figure 2. Lower heating value and heat of evaporation of fuels (**a**) and fuels component mass fraction (**b**).

3. Results and Discussion

3.1. Combustion Characteristics

The research was based on the process of indicating the engine, which consists of measuring the pressure in the engine cylinder vs. crank angle. The examinations were started from indication of the engine fuelled with pure diesel as a reference. Figure 3 shows the pressure profiles in the cylinder for the engine fuelled by pure diesel and the engine for co-combustion of diesel with methanol and ethanol. The pressure profiles were used to determine the heat release rate (HRR), which characterizes the combustion process in the engine and allows for evaluation of the operation of the test engine.

The presence and the increase in the percentage of methanol and ethanol used for co-combustion with diesel lead to an increase in ignition delay of the charge caused by a higher value of latent heat of vaporization of the alcohol fuel compared to diesel [37]. Figure 3 shows that a greater ignition delay is observed for co-combustion with methanol. The increase in ignition delay leads to the rise in heat release rate and the maximum combustion pressure values. The paper [38] showed that with an increasing methanol substitution ratio in diesel engine with diesel/methanol compound combustion mode, the combustion starting point is delayed, both for the peak heat release rate and the peak cylinder pressure increase. The increase in ignition delay with methanol fraction results in more fuel burnt in the premixed combustion (kinetic combustion phase), which, coupled with the combustion of diesel fuel in a richer methanol–air mixture, leads to the increase of the maximum cylinder pressure and heat release rate. The objective of paper [39] was to investigate the thermal performance, exhaust emissions and combustion behaviour of small capacity compression ignition dual-fuel engine using fumigated ethanol. It was found that with the induction of ethanol in the CI engine, an increased rate of heat release and increased peak pressure occurs, thus enhancing the thermal efficiency. Cycle to cycle variations were very small with the diesel fuel alone. Distinct engine cycle variations as indicated by higher value of COV of IMEP were evident with various ethanol flow rates. Figure 3 was supplemented with the computed values of the indicated mean effective pressure, which show that co-combustion of diesel with alcohol fuel does not cause significant changes in the values of this parameter. Similar values of IMEP were obtained for both pure diesel oil and for all analyzed percentages of alcohol, at the level of ca. 0.66 MPa. Figure 3 presents the value of IMEP measurement error. The IMEP measurement error was ±3.1%.

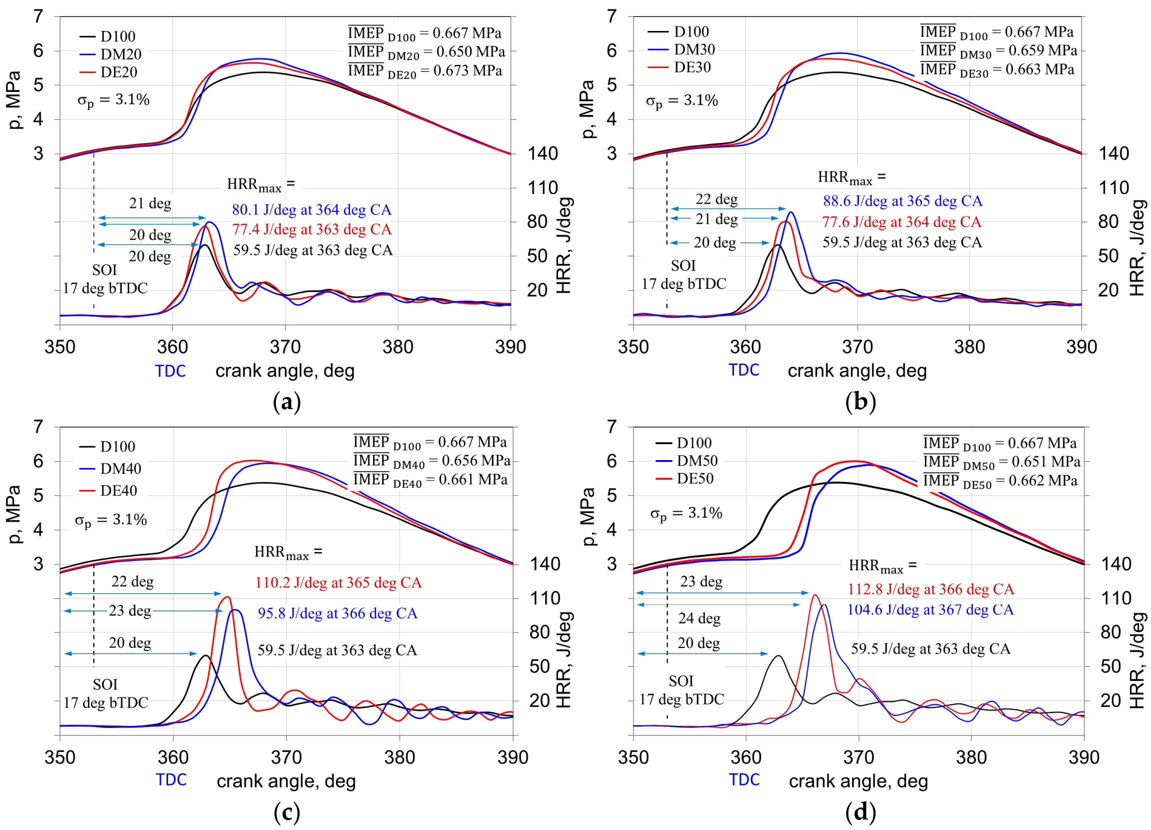


Figure 3. In cylinder pressure and heat release rate (HRR) traces of the engine for co-combustion of diesel fuel with methanol and diesel fuel with ethanol for 20% (**a**), 30% (**b**), 40% (**c**) and 50% (**d**) alcohol share.

In the conventional compression ignition engine, the combustion process is performed in two stages. The first, initial stage, is a period of kinetic combustion, which relates to combustion of the combustible mixture around the jet of injected fuel, whereas the second stage is diffusion combustion which characterizes the process of gradually vaporized fuel combustion. Depending on the conditions of work, the contribution of these stages changes, thus affecting engine performance and emissions. The stage of the kinetic combustion, which characterizes high values of temperature, is conducive to creation of the nitrogen oxides, whereas the stage of the diffusion combustion may be the source of non-burned hydrocarbons and soot particles. Co-combustion of fuels in the dual fuel engine differs from classical fuel combustion in the conventional engine. However, in most conditions, it is found that the combustion characteristics and heat release rate curves of dual-fuel engines can also be divided into two phases, and its shape is very similar to the typical curves [40]. In the dual fuel engine, the injection of additional fuel to the intake manifold, a homogeneous mixture is formed during the inlet stroke, which is then compressed in the cylinder during the compression stroke. With this prepared mixture, the injection of the dose of the diesel fuel occurs at the end of the compression stoke. This dose evaporates, ignites and leads to the ignition of the homogeneous mixture in the cylinder. Due to the reduced content of diesel fuel in total dose of the fuel supplied to the engine, its time of vaporization and combustion is shorter. This leads to shortening of the diffusion combustion stage and increases in the content of the stage of kinetic combustion. Co-combustion of fuels using the idea of dual fuel engine with two independent fuel supply systems, allows for control of individual stages of the combustion process through control of the moment of compression ignition through injection of the dose that initiates ignition and offers opportunities for smooth changes in the contents of individual fuels, adjusted to varied conditions of engine work.

Furthermore, Figure 4 shows the effect of alcohol energy percentage on the kinetic and diffusive combustion periods during co-combustion of diesel with methanol and ethanol. The division into two combustion phases was made arbitrarily on the basis of HRR analysis. The kinetic phase was defined as the period from the beginning of combustion to the local minimum on the HRR curve before the diffusion phase. The diffusion phase was defined as the period from the end of the kinetic phase to the end of combustion. 10% of the heat release was considered as the start of combustion and 90% of heat release was considered as the end of combustion. As the share of alcohol increased, the kinetic combustion period was extended due to the increased heat release rate and intensification of combustion, whereas the diffusion combustion period was reduced. Studies [40] showed that the increased in the percentage ratio of methanol energy to the total energy, from 0% to about 50%, enlarges the kinetic combustion phase, called the rapid burning duration. It is also shown that the proportion of rapid burning heat release in the whole combustion heat release increases dramatically with the increase of the alcohol energy in total fuel energy (alcohol and diesel fuel) supplied to the engine. In the case of DM50, the kinetic combustion stage accounted for 56% of the entire period of combustion of the load in the engine cylinder and was 31% longer than in the case of pure diesel. In the case of DE50, the kinetic combustion phase was 43% and was 18% longer than the kinetic phase accompanying the combustion of pure diesel.

For high energy contents, methanol, compared to ethyl alcohol, turned out to be a fuel that increased the rate of charge combustion, increasing the contribution of kinetic combustion and decreasing the contribution of diffusion combustion period. For 50% of alcohol, the use of methanol compared to ethanol increased the kinetic combustion period and led to a reduction in the diffusion combustion period by 13%.

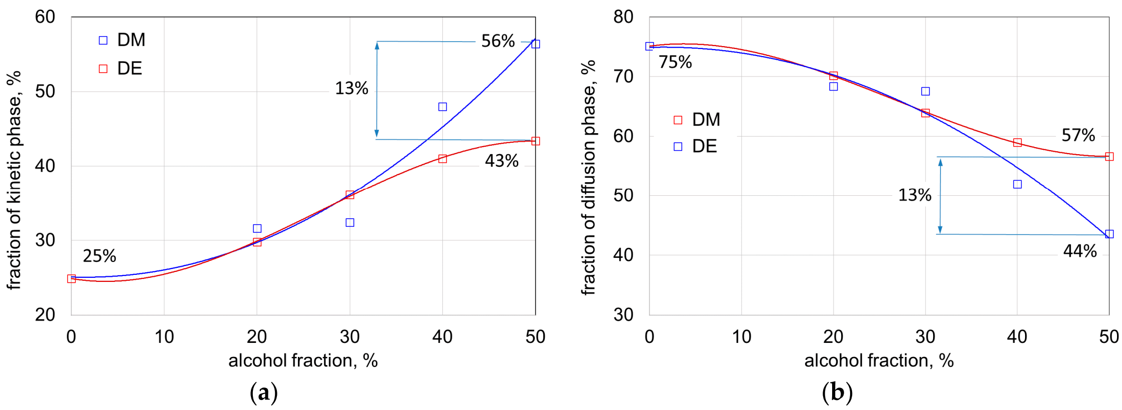


Figure 4. Kinetic (**a**) and diffusion (**b**) phase of the co-combustion process of diesel oil with methanol and ethanol for the analyzed energy shares of an alcohol fuel.

3.2. Cycle-to-Cycle Variation of the Engine

Combustion engine operation is characterized by the non-repeatability of subsequent operation cycles caused by the continuous changes in the conditions present in the engine cylinder, both during filling and combustion processes. Instantaneous conditions in the combustion chamber depend, among other things, on the wave phenomena in the intake channels during the suction stroke, instability of the ignition system and the ignition misfiring (SI engines) and the change in ignition delay (CI engines) associated with the heterogeneity of the combustible mixture. Furthermore, the increased non-repeatability of engine operation is affected by the instability of the rotational speed and engine temperature. The profiles of pressure changes in the cylinder obtained during engine indication for several hundred cycles differ from each other and always deviate to a greater or lesser extent from a single profile usually adopted as a representative profile. Figure 5 illustrates in cylinder pressure profiles and heat release rate during combustion of D100 and DM50 for 200 individual test engine operation cycles.

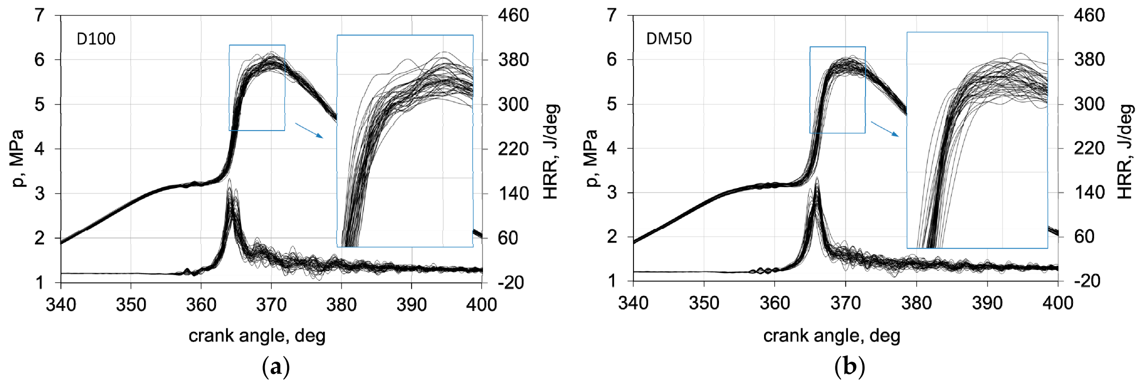
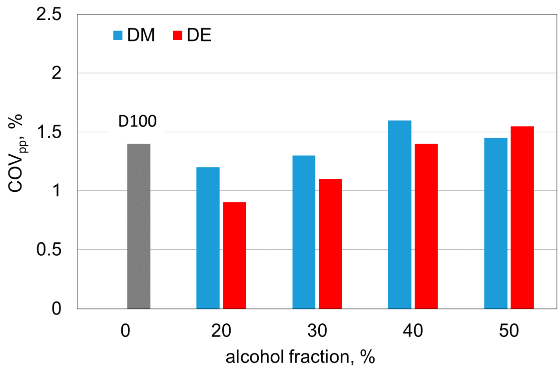


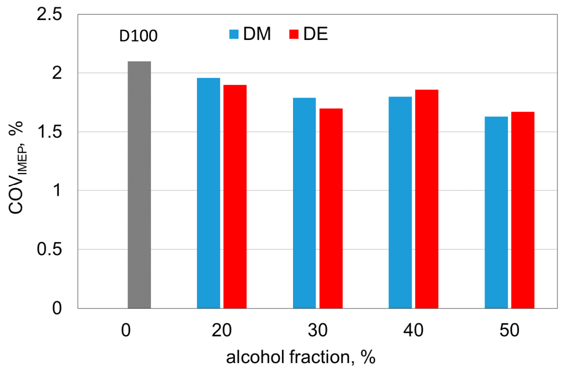
Figure 5. In cylinder pressure and heat release rate during combustion of D100 (**a**) and DM50 (**b**) for 200 individual cycles of a test engine.

The indicator diagrams can be used to determine characteristic values that are representative for many individual engine operation cycles, such as maximum combustion pressure or indicated mean effective pressure. Changes in these values can be computed using statistical analysis methods and presented in the form of the coefficient of variation of peak pressure (COV_{pp}) or of the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}). This parameter can be used to evaluate stability of the internal combustion engine related to the non-repeatability of subsequent engine operation cycles and ignition misfiring. In paper [41], the COV of IMEP and p_{max} were both used to evaluate cyclic variations in multi-cylinder diesel engine fueled with diesel methanol dual fuel. Three different methanol injection positions were chosen to investigate their effect on cylinder-to-cylinder variation. According to literature, the threshold of correct operation of combustion engine, expressed by the maximal value of the coefficient of variation of the indicated mean effective pressure is 2–5% [35,42].

Figure 6 shows non-repeatability of the value of indicated mean effective pressure and maximum pressure, for a conventional diesel engine and a dual fuel engine for co-combustion of diesel with methanol and ethanol. It can be observed that in the case of both alcohols, the increase in their energy content leads to a reduction in the COV_{IMEP} coefficient, which suggests an improvement in engine operation stability. This may result from an increase in the intensification and speed of the combustion process that improved process repeatability due to the presence of active oxygen supplied in the alcohol fuel. The highest oxygen content is observed in methanol, with the lowest COV_{IMEP} = 1.63% values obtained for DM50. In paper [43] COV_{IMEP} coefficient was analyzed for a dual-fuel engine supplied with diesel fuel and methanol, at 60% load. The COV_{IMEP} of the dual-fuel engine remained smaller than 2% in all test conditions. This means the combustion of dual-fuel is very stable without large cycle-by-cycle variations. In paper [44] results are presented of investigation of the diesel/methanol dual-fuel combustion mode as a new combustion mode for NO_x and PM reduction from the diesel engine. The aim of this research was to investigate the effect of diesel fuel injection pressure on the performance and emissions characteristics in a 6-cylinder common-rail diesel engine. It showed that the COV_{IMEP} of the dual-fuel combustion mode remains under 2.1% among all the tests. Experimental investigations [31] of the engine with the ethanol-diesel dual-fuel combustion mode showed that, in the dual-fuel mode, more premixed combustion yielded higher levels of COV_{IMEP} than the conventional diesel combustion mode case for higher engine loads. The increase in COV_{IMEP} was proportional to the ethanol energy fraction. In addition, the dual-fuel operation resulted in higher COV_{pp} than the conventional diesel combustion mode at all engine loads except a lower load. Nevertheless, the COV_{IMEP} and COV_{pp} could be controlled between 1.0% and 3.0%. In the case of the analysis of non-repeatability of the maximal combustion pressure, no unambiguous changes in the COV_{pp} values were obtained. A low percentage of alcohol, both in the case of methanol and ethanol, led to a decrease of the COV_{pp} coefficient, which indicated the improved engine stability. After exceeding the level of 20% alcohol, an insignificant increase was observed in non-repeatability of engine operation, eventually reaching the value comparable to D100 for DM50 and DE50. It should be emphasized that both the engine powered by pure diesel and the engine for co-combustion of diesel with alcohol fuels were characterized by stable and even operation, for which the coefficient of non-repeatability was much lower than the acceptable limits for the reciprocating combustion engine (2–5%).



(**a**)



(**b**)

Figure 6. Coefficient of variation of peak pressure (**a**) and coefficient of variation of the indicated mean effective pressure (**b**) for a conventional engine and a dual fuel engine co-combustion diesel with ethanol and methanol.

One of the indicators that allow for the assessment of the operation stability of an internal combustion engine is probability density for the occurrence of selected parameters of engine operation. Figure 7 presents the probability density for the occurrence of IMEP in a conventional engine and a dual fuel engine for co-combustion of diesel with 20%, 30%, 40% and 50% of ethanol and methanol. The distribution of probability density of the indicated mean effective pressure, based on normal distribution, represents a repeatability index for individual IMEP values obtained for 200 analyzed cycles of the test engine operation. The mean values of IMEP presented in the drawings show insignificant differences in performance between combustion of pure diesel and co-combustion of diesel with methanol and ethanol. The largest differences were found for the lowest percentage of alcohol fuel, equal to 20% (Figure 7a).

The probability density curves of the indicated mean effective pressure show, among other things, the repeatability of the mean IMEP value, which may be considered a determinant of the stability of operation of the test engine. The probability density for IMEP only during combustion with the lowest percentage of alcohol fuel (20%) was lower compared to combustion of pure diesel (Figure 7a). In the case of higher percentage of both methanol and ethanol, higher values of f(IMEP) were recorded compared to D100. For the highest percentage of alcoholic fuel (50%) (Figure 7d), this value was 37.5% for methanol and 36.2% for ethanol, respectively.

Analysis of the profile of the probability density function in the case of DM50 for 200 engine operation cycles reveals that the individual IMEP values for the largest number of cycles were close to the mean value of IMEP. This suggests the best repeatability of subsequent cycles of the test engine operation for co-combustion of diesel with 50% energy from methyl alcohol. These findings are consistent with previous results obtained for the analysis of the stability of a dual fuel engine operation based on the changes in the coefficient of variation of the indicated mean effective pressure.

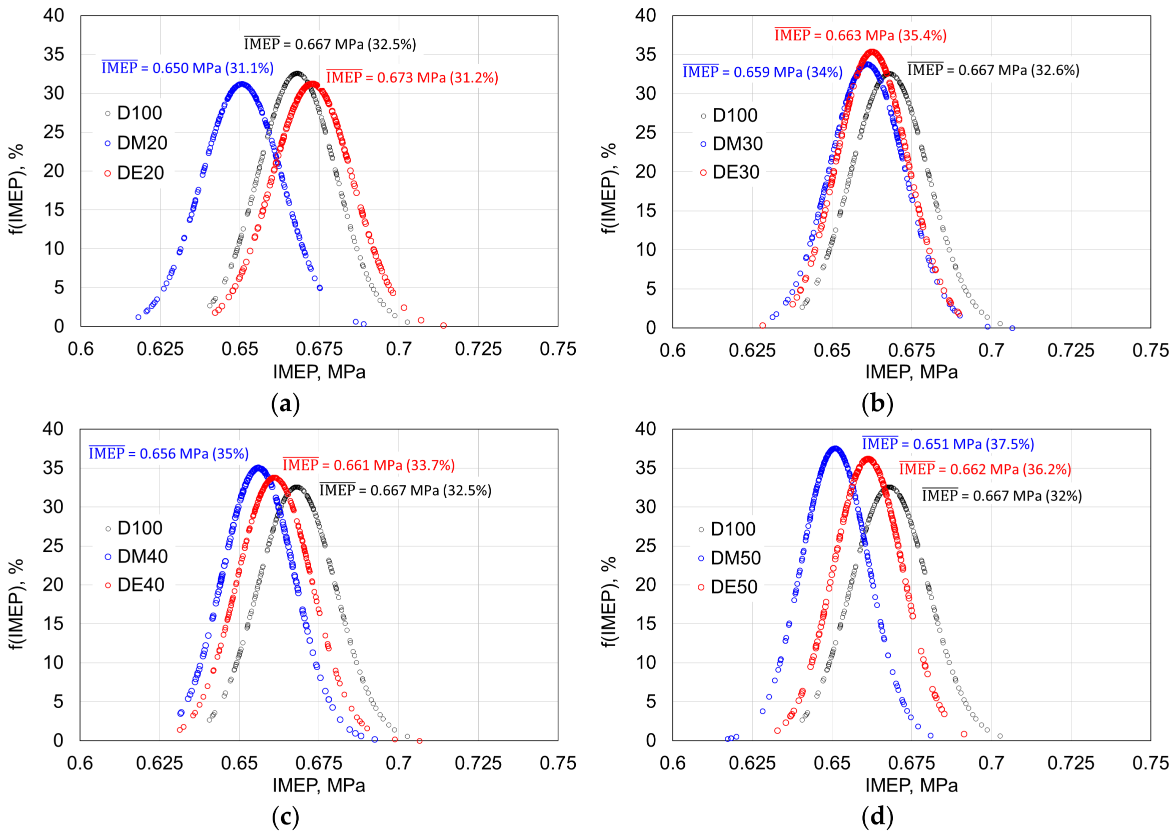


Figure 7. The probability density of the indicated mean effective pressure for co-combustion of diesel fuel with methanol and diesel fuel with ethanol for 20% (**a**), 30% (**b**), 40% (**c**) and 50% (**d**) alcohol share.

The correlation between the maximum combustion pressure and the indicated mean effective pressure for a conventional engine and a dual fuel engine for co-combustion of diesel with ethanol and methanol is presented in Figure 8. The figure illustrates the scattering of the pmax and IMEP values obtained for 200 engine cycles and the mean values of these parameters adopted as representative. The analysis of the scattering in subsequent cycles reveals that the increase in the indicated mean effective pressure was associated with the increase in the maximum combustion pressure for each case. Due to the fact that the tests were performed at constant engine load, insignificant changes in IMEP were obtained for each case, which did not differ much from the value of 0.66 MPa. Larger differences are observed in p_{max} values. Analysis of this parameter shows that for the lowest alcohol percentage (20%), the differences between co-combustion of methanol and ethanol are the greatest. The increase in the alcohol content to over 20% yielded p_{max} values similar for DM and DE while the differences were increased compared to D100. It is possible to obtain the expected IMEP in an engine for co-combustion of diesel with alcohol, even though the engine has a wider range of p_{max} changes compared to the combustion of pure diesel.

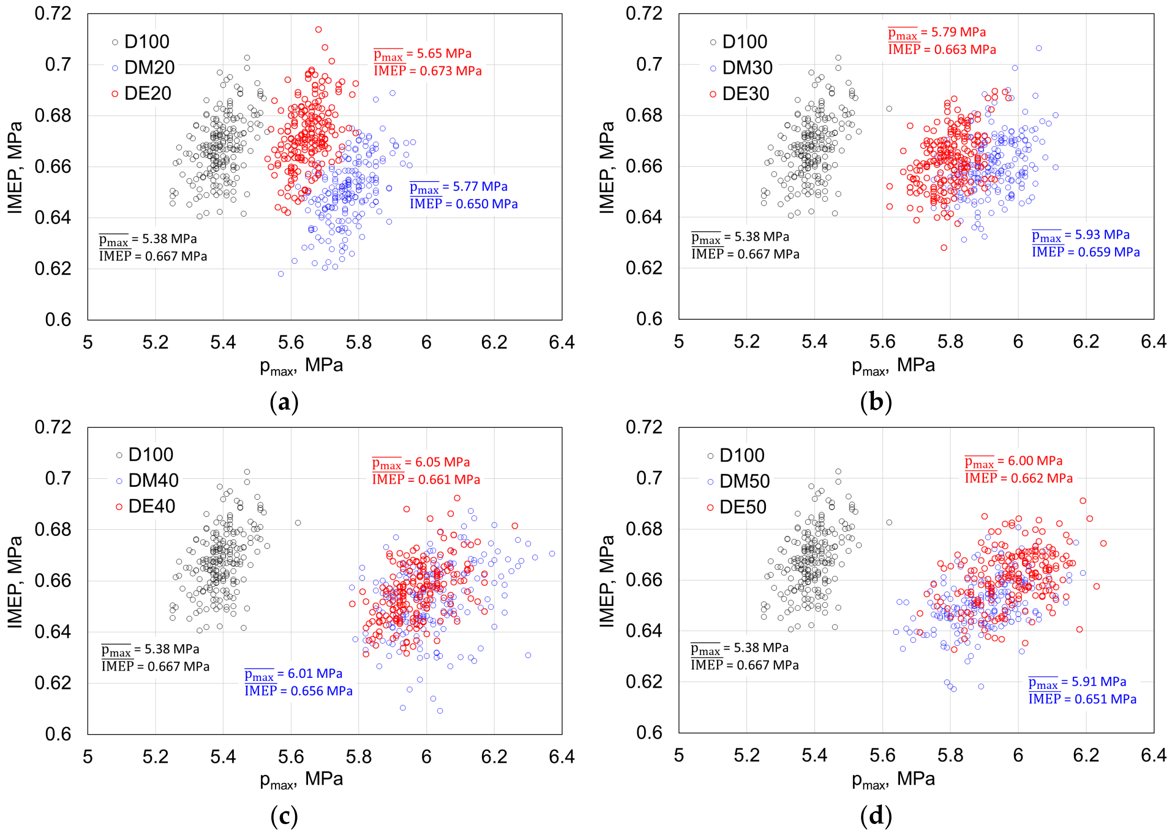


Figure 8. The correlation between the maximum combustion pressure and the indicated mean effective pressure for co-combustion of diesel fuel with methanol and diesel fuel with ethanol for 20% (**a**), 30% (**b**), 40% (**c**) and 50% (**d**) alcohol share.

4. Conclusions

Alcohol fuels offer opportunities to reduce the use of fossil fuels in CI engines and to increase the percentage of biofuels in the transport and energy sectors, where combustion engines are often used. This study presents experimental examinations of a CI engine with dual-fuel system, in which co-combustion of diesel with two alcohol fuels (methanol and ethanol) with energy content of 20%, 30%, 40% and 50% was performed. The research included the analysis of the combustion process and the analysis of the non-repeatability of the 200 subsequent engine operation cycles. Analysis of the results obtained in the study leads to the following conclusions:

- presence and increase in the share of methanol and ethanol used for co-combustion with diesel causes an increase in compression ignition delay and increases the heat release rate and maximum combustion pressure values,
- for high energy percentages of over 30%, methanol, compared to ethyl alcohol, turned out to be a fuel that is accompanied by a higher rate of charge combustion, contributing to intensification of the process and the increase in the contribution of the kinetic combustion period while decreasing the contribution of the diffusion combustion stage,
- the analysis of the operation of a dual fuel engine based on the value of the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}) for 200 operation cycles revealed an improvement in the stability of its operation with an increase in the percentage of alcohol. With the largest share of alcohol (50%), methanol proved to be a better fuel than ethanol. For DM50, the lowest COV_{IMEP} value of 1.63% was obtained.
- based on the function of probability density of the indicated mean effective pressure (f(IMEP)) prepared for 200 engine operation cycles, it can be concluded that the increase in the percentage

of alcohol fuel used for co-combustion with diesel is conducive to stability of operation of the dual-fuel engine. For a 50% share of alcoholic fuel, the higher value of f(IMEP), which demonstrates better engine stability, was characterized by methanol and was equal to 37.5%.

- assessment of engine operation stability based on the probability density function of selected engine performance parameters such as indicated mean effective pressure, leads to results similar to those obtained from the assessment based on the analysis of changes in the coefficient of variation of the indicated mean effective pressure,
- with the increase in the percentage of methanol and ethanol used for co-combustion with diesel, the dependence of the indicated mean effective pressure on the maximum combustion pressure becomes lower. Although combustion in a dual fuel engine was characterized by greater differences in maximum pressure compared to combustion in a conventional engine, the values of the indicated mean effective pressure (that reflect engine performance) were similar for both engines.

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Abbreviations

CI	compression ignition engine
IMEP	indicated mean effective pressure, MPa
p_{max}	maximum pressure, MPa
COV_{IMEP}	coefficient of variation of the indicated mean effective pressure, %
COV_{pp}	coefficient of variation of peak pressure, %
σ_{IMEP}	standard deviation of the IMEP, MPa
σ_{pmax}	standard deviation of peak pressure, MPa
f(IMEP)	probability density of the indicated mean effective pressure, %
HRR	heat release rate, J/deg
V_d	displaced cylinder volume, m^3
p	pressure, bar
n	engine speed, rpm
TDC	top dead centre
λ	excess air ratio
SOI	start of injection
NO_x	nitrogen oxides
THC	total hydrocarbons
CO	carbon monoxide
CO_2	carbon dioxide
O_2	oxygen
PM	particulate matter

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