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Authors:

Wojciech Tutak, Arkadiusz Jamrozik

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Comparative Analysis of Combustion Stability of Diesel/Ethanol Utilization by Blend and Dual Fuel

Wojciech Tutak and **Arkadiusz Jamrozik** *

Faculty of Mechanical Engineering and Computer Science, Czestochowa University of Technology, 42-201 Czestochowa, Poland; tutak@imc.pcz.pl

* Correspondence: jamrozik@imc.pcz.pl

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Keywords: diesel fuel; ethanol; dual fuel engine; rate of heat release; ignition delay; combustion duration; combustion stability

1. Introduction

Compression ignition engines are considered a significant source harmful emission that contributes to the increase in the greenhouse effect [1,2]. Eliminating them from use is now virtually impossible, because they are still used in various machines and cars due to relatively high efficiency and reliability [3]. However, wider use of alternative fuels to fossil fuels is possible. An additional difficulty is that the majority of fuels, apart from biodiesel, are not very friendly for self-powered diesel engines [4]. Research and development works have been carried out for many years indicating other possibilities of using alternative fuels. One method is to mix diesel fuel with another fuel and supply the engine with such a blend. Unfortunately, most alternative fuels have trouble creating a stable mixture. This disadvantage strongly limits the share of alternative fuel in the mixture. In such a power system, it is difficult to change the ratio of diesel/other fuel [5]. The second method is a dual fuel engine power system in which fuels are delivered to the engine separately. The latest solutions use direct injection of both fuels into the engine's combustion chamber. Currently, in most commonly used power systems, fossil fuel for compression ignition engine is delivered by a direct injection system and alternative fuel is injected into the intake port [6]. The ignition process is controlled by the injected dose of diesel fuel. This requires the addition of an injector, along with a separate fuel tank, lines and control system. Various alcohols are often used as alternative fuels, such as ethanol, methanol, butanol or propanol. Among alcohols, the most interesting seems to be ethanol. It can be produced from many plants that contain sugar or other components that can be converted into sugar, such as starch or cellulose in the fermentation, distillation and dehydration process. Ethanol is less toxic than gasoline and methanol, and is not carcinogenic. Bioethanol can be derived from a wide range of carbohydrates of general formula $(CH_2O)_n$. Recent years have seen intensifying explorations of opportunities to use alcohol-based fuels in compression ignition engines. In the works devoted to the combustion of fuels in the form of blends or dual fuel mode, reference is made to operating parameters of engine and emission standards fulfillment. Lee et al. presented results of investigation of a heavy-duty single-cylinder diesel engine with two direct injectors. Engine speed was fixed at 1000 rpm and the load condition varied for an indicated mean effective pressure ranging from 0.2 to 0.8 MPa. The ratio of ethanol to the total input energy was controlled from zero to nearly 50% of the input energy. The NO_x and PM emissions decreased with increasing ethanol substitution and the mean size of the PM emissions decreased. For the mid-load condition, the substitution was increased to 63%; however, for low and high loads, higher ethanol fractions could not be used because of insufficient ignition energy at low loads and high values of pressure rise under high loads [7]. Similar results were obtained by other authors; all of them showed that the share of ethanol has a positive effect on soot emissions, while there are different opinions regarding NO_x emissions [8–11]. On the one hand, ethanol reduces the temperature on the compressed load due to the high heat of vaporization. On the other hand, it increases the value of the maximum pressure and, consequently, the combustion temperature, contains additional oxygen in the structure, it promotes the formation of NO_x [12–14]. Ethanol greatly affects the combustion stage. This is confirmed by the result of basic research carried out using specialized combustion chambers [15–18]. The authors of research papers using piston engines came to similar conclusions with basic research [9,11,19]. In the paper of Paul et al., regarding the investigation of blend combustion, authors presented results of investigation of diesel-ethanol combustion process in diesel engine. Ethanol percentage was increased from 5% to 20% with intervals of 5%, thus reducing the diesel participation. Paul et al. stated that a blend with 15% ethanol showed best engine performance characteristics with 21.17% increase in brake thermal efficiency and 4.61% decrease in BSEC at full load. The combustion analysis also revealed increase in cylinder pressure and heat release rate indicating improvement in combustion condition for the above-mentioned blend. The blend also showed a substantial improvement in THC and CO emissions with a small increase in NO_x emission. The exergy analysis showed a 25.64% increase in exergetic efficiency [20]. Other researchers conducted similar studies [14,21,22]. They also confirmed that ethanol causes increase in engine efficiency.

Co-combustion of ethanol with diesel fuel, as shown in the cited articles, has many advantages. These are both advantages related to increased efficiency and emissions of most exhaust gas components. Unfortunately, there are no papers related to the stability of the combustion process in an engine co-firing ethanol with other petroleum fuels. The main idea of this work is the comparison of two fuel co-firing systems. Some conclusions in similar works are different. This is mainly due to the fact that the tests are carried out on different engines in different conditions. Fuels have different properties, for example ethanol with different water content. The authors compare both combustion systems using pure ethanol (99.9%) and diesel fuel and the tests were conducted on the same engine under the same conditions using the same fuels. In the presented work, we have attempted to assess the combustion stability of the co-combustion process of diesel fuel with ethanol in a dual fuel compression ignition engine. The research concerned the analysis of the combustion process and the analysis of non-repeatability for set of subsequent engine operation cycles. The analysis was made for heat release rate, combustion stages and combustion stability.

2. Materials and Methods

2.1. Experimental Test Stand

The research was carried out on the compression ignition engine operated with a constant rotational speed of 1500 rpm. It was one-cylinder air cooled, naturally aspired, direct injection compression ignition engine. This engine was modernized to work as a dual fuel engine. It was equipped with the independent port fuel injection system. The engine control system ensured that the engine speed was kept constant at various loads. Ethanol was injected in the intake manifold at 3 bar pressure and the value of the fuel dose was determined by the time of opening the injector. The injection system was equipped with an electronic control system connected with the signal of the crankshaft position.

Test engine operated with constant angle of beginning of diesel fuel injection equal to 17 degree before top dead center (TDC). The test bed is presented in Figure 1. The main engine parameters are presented in Table 1.

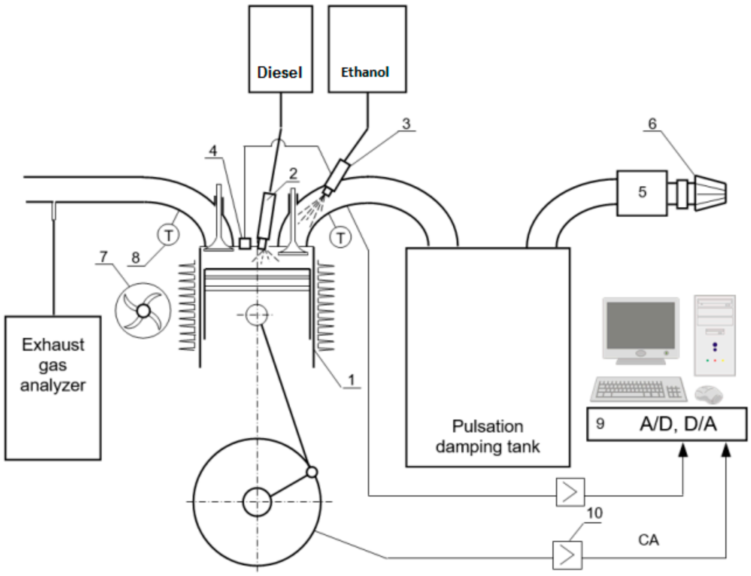


Figure 1. Diagram of the experimental setup. 1—engine, 2—diesel fuel injector, 3—ethanol fuel injector, 4—in cylinder pressure sensor, 5—intake air flowmeter, 6—air filter, 7—cooling fan, 8—exhaust gases temperature sensor, 9—pc with data acquisition system, 10—crank angle sensor.

Table 1. Main engine parameters.

Parameter	Value	Unit
Number of cylinders	1	-
Displacement volume	0.573	dm^3
Bore	90	cm
Stroke	90	cm
Compression ratio	17:1	-
Crankshaft rotational speed	1500	rpm
Injection pressure	21	MPa
Injection timing	17	deg bTDC
Maximum rated power	7.4	kW

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,
- the CA encoder, resolution 360 pulses/rev, software for digital recording and analysis of the frequency signals [23].

In Table 2, the fuels properties can be seen. Diesel fuel ($C_{14}H_{30}$) used was commercial fuel provided by the Polish Refinery and commonly used to feed diesel engines in cars. The fuel is a mixture of liquid hydrocarbons obtained through crude oil distillation. In the case of the diesel fuel, one of the important parameters is the cetane number that denotes the auto-ignition capabilities of the fuel.

Ethanol (C_2H_5OH of 99.9% concentration) is an alcohol with two carbon atoms in its structure. This fuel is obtained through processing of biological matter. Therefore, it can be considered a renewable energy source. This alcohol is numbered among strongly oxygenated alcohols and is characterized by a lower value of the LHV compared to fossil fuels. LHV of ethanol is lower in 40% compared to LHV of diesel's fuel. Therefore, to keep the constant energy dose, comparable to that contained in diesel fuel, bigger ethanol dose, in mass, should be provided. The high heat of vaporization (840 kJ/kg) improves the filling coefficient but increases the ignition delay, which can cause the "hard" operation of the engine.

Table 2. Fuel properties [5,10,11].

Properties	Diesel Fuel	Ethanol Fuel
Molecular formula	$C_{14}H_{30}$	C_2H_5OH
Molecular weight	170–198	46
Surface tension (mN/m @ 15 °C)	26.9	21.78
Cetane number	51	8
Lower heating value, (MJ/kg)	41.7	26.9
Density at 20 °C, (kg/m^3)	856	789
Viscosity at 25 °C, (mPa s)	2.8	1.078
Heat of evaporation, (kJ/kg)	260	918
Stoichiometric air fuel ratio	14.7	9.06
Autoignition temperature, (°C)	300–340	698
Flash point, (°C)	78	16.6
Hydrogen content, wt%	13	13
Carbon content, wt%	87	52.2
Oxygen content, wt%	0	34.8

2.2. Calculation Methodology

The analysis of the combustion process was conducted on the basis of heat release. The heat release rate was calculated based on the data of in-cylinder pressure regarding crank angle. Analysis was based on the first law of thermodynamics and the equation of state. Due to omitting the 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 [8]. A detailed description of the research procedure and results development is included in our previous papers [10,11,14].

The unrepeatability of IMEP (COV_{IMEP}) was used as a parameter determined the cycle-by-cycle variations. The COV_{IMEP} is directly related to the investigated combustion stability. The COV_{IMEP} was calculated based on set of IMEP values from 200 following work cycles of the test engine:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP_{mean}}\,100\%, \tag{1}$$

where σ_{IMEP} is the standard deviation of IMEP and $IMEP_{mean}$ is the mean value of indicated mean effective pressure.

Additionally, there are presented evaluation results of the probability density function of indicated mean effective pressure f(IMEP). This parameter can also be used as an indicator to assess the stability of operation of the internal combustion engine. The probability density of the indicated mean effective pressure:

$$f(IMEP) = \frac{1}{\sigma_{IMEP}\sqrt{2\pi}} \exp\left(\frac{-(IMEP - IMEP_{mean})^2}{2\,\sigma^2_{IMEP}}\right), \tag{2}$$

3. Results Analysis

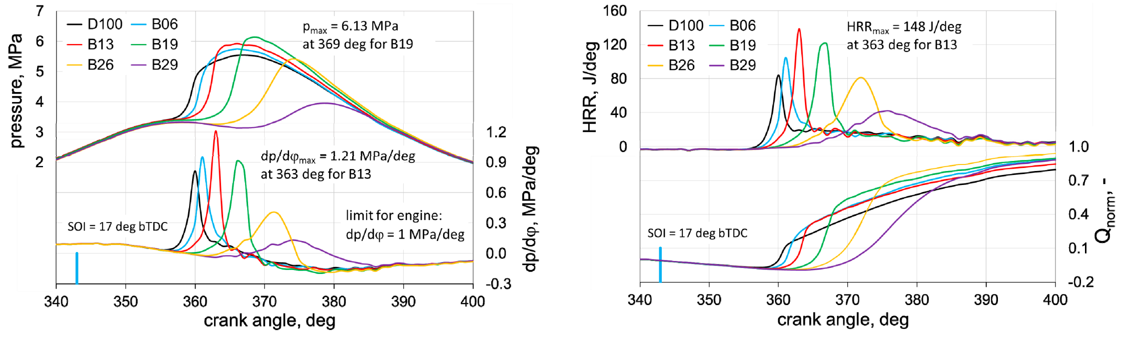
There are presented results of assessment of combustion stability of the co-combustion process of diesel fuel and ethanol using two combustion modes. The first is co-combustion as a fuel blend (B). The main disadvantage of this solution is problem with stability and separation of blends and lack of flexibility on the ratio of diesel/alcohol. The second one is dual-fuel mode (DF) in which additional fuel is supplied by injection to intake port of IC engine. Into the engine cylinder is delivered air–fuel mixture, nearly homogeneous. In that combustion system, the ignition of nearly homogeneous mixture of air and fuel is realized by injection of diesel fuel 17 degree of crank angle before top dead center (TDC). This construction system requires the additional power supply by injector or injectors, tank with alternative fuel and other necessary equipment [10,11,22].

At each test point, the engine was fully warm up and its parameters were stabilized. The engine was run until the engine reached a constant temperature of the exhaust gases and invariable emission. The measurement system allowed for recording of the in-cylinder pressure with resolution of 1 deg CA of 200 engine operating cycles. It was recorded simultaneously: rotational speed of engine, air and fuel consumption, air temperature, fuel temperature, exhaust gas temperature, ambient temperature and pressure. The examinations started from indication of the engine fueled with pure diesel as a reference. The tests were carried out at a constant angle of diesel injection start. The variations of energy doses were in range of 3%. Preliminary tests were performed using mathematical modeling using the Fire program. The results of simulation tests allowed to determine the maximum share of ethanol in dual fuel mode, which was 60%; after that, the combustion process deteriorated.

The in-cylinder pressure, heat release rate (HRR) and normalized heat release (Q_{norm}) for various ethanol fraction in blend or in dual fuel mode are used for analysis. On the basis of Q_{norm} traces, the combustion process phases were determined. For the combustion process in the IC engine, two phases are the most important: first is the ignition delay (ID), second the combustion duration (CD). The ignition delay is defined as the time between the start of diesel fuel injection and the crank angle of 10% heat release (CA 0–10% Q_{norm}). This delay period consists of physical delay and chemical delay which occur simultaneously. In the physical delay takes place atomization, vaporization and mixing of air fuel, and in the chemical delay attributed pre-combustion reactions. Burn duration is also calculated by reading the time between CA 10–90%, the crank of heat release (CA 10–90% Q_{norm}).

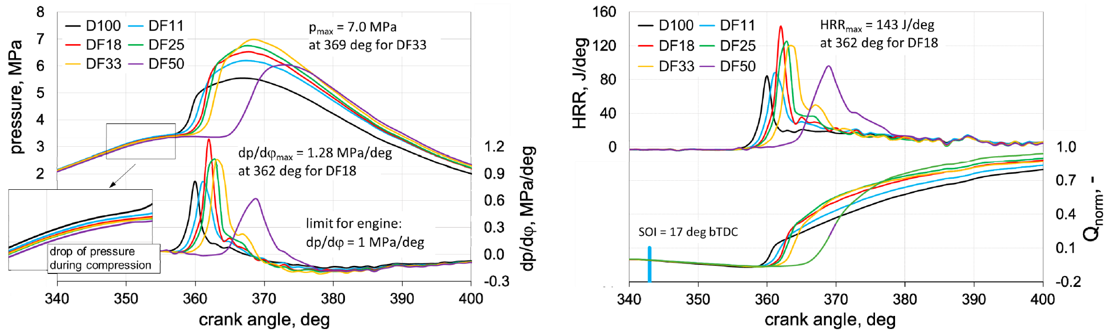
Figure 2 shows the results of the investigation of the compression ignition engine powered by diesel/ethanol blend. It can be stated that with the increase in ethanol energetic fraction in blend up to near 20% the peak pressure increases. The highest pressure was obtained for 19% of ethanol fraction and it was equal to 6.12 MPa and was reached 8 deg after TDC. A further increase in the proportion of ethanol worsened flame propagation in the combustion chamber as evidenced by a significant decrease in combustion pressure. This was due to the poor auto-ignition properties of ethanol and relatively high value of heat of evaporation. In case of the rate of pressure increase ($dp/d\varphi$) the highest value obtained for 13% of ethanol fraction and it was equal to 1.2 MPa/deg. The so-called hard engine running was noticed at this operating point. This is reflected in the heat release in the combustion chamber of engine. The highest value of HRR was obtained for 13% of ethanol fraction and it was equal to 138 J/deg. Figure 2b also shows Q_{norm} waveforms. Based on the analysis of these waveforms, combustion stages can be described. For such a fuel mixture, increased ignition delay and combustion process occurred very rapidly.

Figure 3 shows the results of investigation of dual fuel engine. In comparison with the blend combustion case, it can be stated that up to 33% of ethanol energetic fraction the combustion process was correct for the internal combustion engine. The problem can only be the high rate of the pressure rise, which was slightly higher than 1 MPa. As in the case of combustion of the fuel mixture, in this case for 18% ethanol fraction, the combustion showed the highest value of rate of pressure increase. Analyzing the course of heat release and Q_{norm}, it can be stated that it occurs in a relatively narrow crank angle range. The rapid combustion process is usually accompanied by an increase in the uniqueness of subsequent engine cycles. As a consequence, this affects the uniformity of engine operation and the repeatability of its operating parameters such as IMEP, but also has an impact on exhaust emissions. The choice of combustion model has a great impact on the repeatability of engine cycles.



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

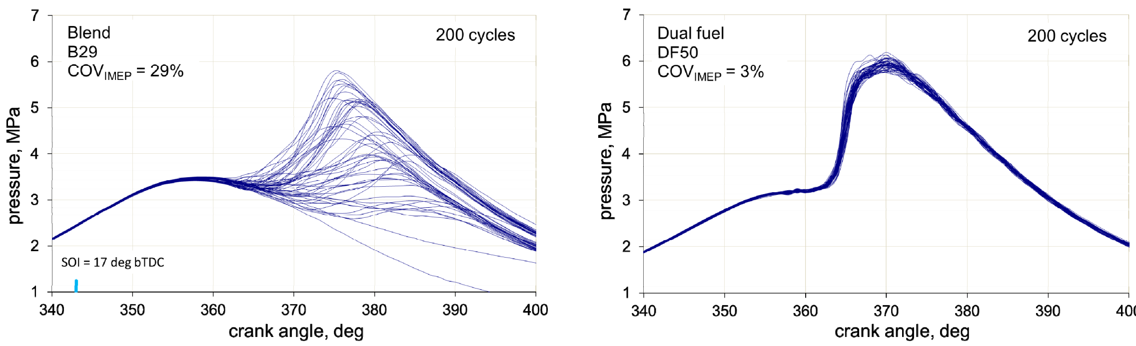
Figure 2. Pressure (**a**) and heat release (**b**) courses for blend case.



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

Figure 3. Pressure (**a**) and heat release (**b**) courses for dual fuel case.

Figure 4 shows the pressure traces for co-combustion diesel/ethanol fuels in blend mode and dual fuel engine. The courses for the highest analyzed ethanol fractions for both combustion systems were selected. It is clear that for dual fuel technology, the cycles are in a compact group. In the case of blend combustion with 29% of energetic ethanol fraction, the cycle spread is very large, from the cycles perfectly correct for the engine to the cycles without ignition of the fuel. The assessment of this spread will be parameterized. The uniqueness of the subsequent combustion cycles is also affected by the uniqueness of the fresh charge speed field [24].



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

Figure 4. Variation of pressure traces for blend (**a**) and dual fuel (**b**) powering.

Figure 5 shows the combustion stages as the ignition delay and combustion duration. The ignition delay period consist of physical and chemical delay phase, which occur simultaneously. The physical delay is the time required for fuel atomization, vaporization and mixing with the air. The second ignition phenomenon is the chemical delay, which consists of the pre-combustion reaction of fuel with air. Ignition delay in compression ignition engines has a direct effect on engine efficiency, noise and exhaust emissions [25]. It was observed that in case of blend combustion, the ignition delay increased more than in case of dual fuel mode. The mixture of diesel/ethanol was characterized lower capabilities for ignition. In dual fuel mode, the ignition is initiated by a dose of diesel only, and the ignition delay is affected by a lowering of the temperature in the combustion chamber by the evaporating ethanol. Increasing the ignition delay time usually results in very rapid combustion, i.e., the duration of combustion decreases. In case of dual fuel mode, the maximal increase in ignition delay was 7 deg of CA (for 50% of ethanol fraction), but in case of blend combustion, this increase was 11 deg of CA (for 29 ethanol fraction). The period of 10 to 90% heat release occurs more quickly in case of blend combustion. Combustion process of blend was characterized by high increase in COV of IMEP. Already for 19% ethanol, energetic fraction COV_{IMEP} exceeded 5%, which is a limit value for industrial engines [10]. In case of dual fuel technology, the COV of IMEP was below 5% and its maximal value was reached for 50% of ethanol fraction, equal to 3%.

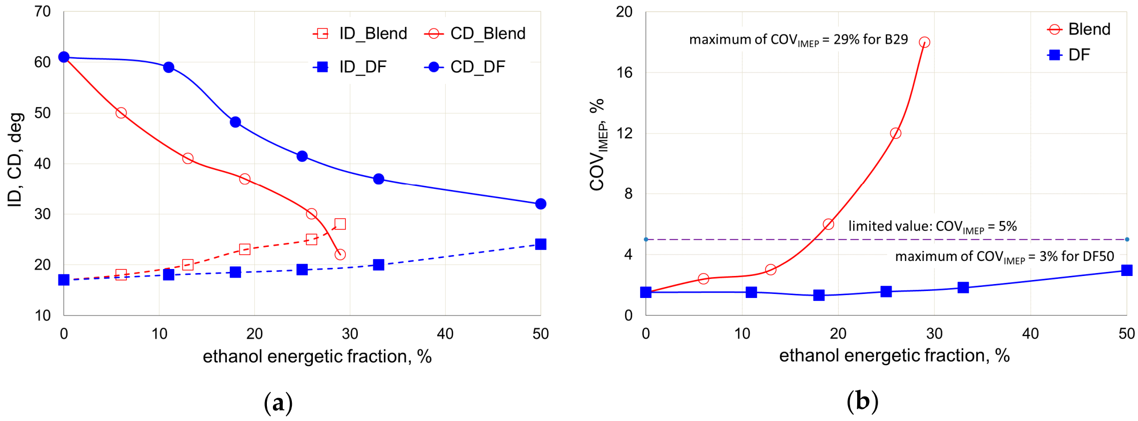
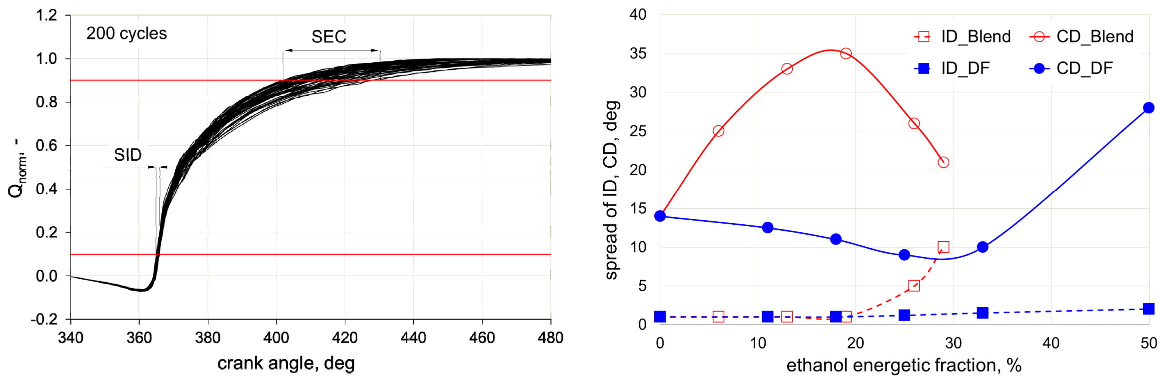


Figure 5. Combustion stages (**a**) and variation of IMEP (**b**).

Combustion stages are determined on the basis of the normalized average heat release traces. These curves are obtained on the basis of set of individual engine cycles. In the IC engine, a cycle-by-cycle variation phenomena was observed. Consequently, such determined combustion stages are also valid to some extent in ambiguity.

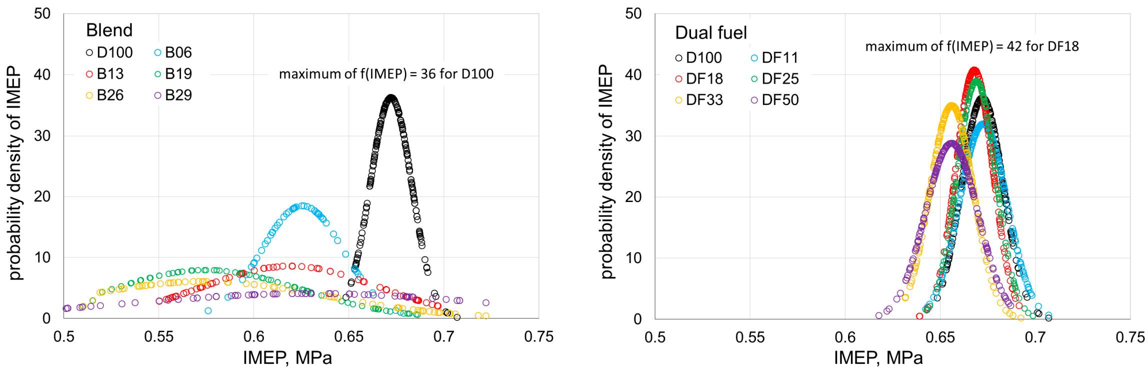
Figure 6 presents the results of the assessment of the unrepeatability of the combustion stages for both analyzed cases. The first area of ambiguity is named SID (spread of ignition delay), and the second one SEC (spread of end of combustion) (Figure 6a). Figure 6b shows the results of assessment of spread of ignition delay and end of combustion period. A larger area of ambiguity occurs when determining the end of combustion because it is determined on the already flattened part of the heat release course. It was noticed that for dual fuel technology, SID is in a very narrow range (2 deg of CA), but for blend combustion, its start increases after exceeding 19% of ethanol fraction. For blend combustion with 19% of ethanol fraction SID was equal 10 deg of CA. Analyzing end of combustion it can be stated that dual fuel technology is characterized by greater repeatability of the end of combustion up to 33% of ethanol fraction; after exceeding this fraction, it starts to increase due to deterioration of combustion conditions. In case of blend combustion with the increase of ethanol fraction (up to 19%), the spread of end of combustion was increased, from 10 to 35 deg of CA. Thus, this ethanol fraction of SEC decreased.

Figure 7 shows the assessment results of the IMEP spread expressed in a probability density. It can be seen that in the case of dual fuel technology, the peak values of this function are in range from 29 to 42 and the spread of IMEP is in the rather narrow range of variation. It looks completely different in the case of combustion of fuels in the form of a mixture. Even the smallest fraction of ethanol causes a large spread of IMEP values. This may be due to the difficulty in maintaining a stable mixture of diesel and ethanol.



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

Figure 6. Assessment of spread of combustion stages (**a**) set of Q_{norm} for 200 cycles (**b**) spread of ID and CD.



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

Figure 7. Probability density of IMEP for blend (**a**) and dual fuel (**b**) combustion.

4. Conclusions

This paper presented the results of assessment of combustion stability of the IC compression ignition engine powered by diesel/ethanol blend or using dual fuel technology. In case of blend combustion, the maximum ethanol fraction was 29%, but in case of dual fuel technology, the energetic fraction was 50%. On the basis of results, the following conclusions can be formulated:

(i) With the increase in the ethanol energetic fraction in the blend, up to nearly 20%, the peak pressure increases. The highest pressure was obtained for an ethanol fraction of 19%, equal to 6.12 MPa at 8 deg after TDC. Regarding DF, all values of the peak pressures were higher than those for the diesel engine.

(ii) In case of dual fuel mode, the maximum increase in ignition delay was 7 deg of CA (for 50% ethanol fraction), but in case of blend combustion, this increase was 11 deg of CA (for 29% ethanol fraction).

(iii) Crossing 19% ethanol energetic fraction in the blend, the COV_{IMEP} exceeded 5%; in case of dual fuel technology, the COV of IMEP was below 3% for all ethanol fractions.

(iv) Up to 19% ethanol fraction, the spread of ID was in the range of 2 deg of CA; for the blend combustion, SID started to increase after exciding 19% ethanol fraction.

(v) Ethanol fraction in dual fuel technology increases the repeatability of the end of the combustion, but in case of blend combustion, the increase in ethanol causes an increase in the SEC parameter.

Author Contributions: Conceptualization, W.T. and A.J.; Data curation, W.T. and A.J.; Formal analysis, W.T.; Investigation, W.T. and A.J.; Methodology, W.T. and A.J.; Writing—original draft, W.T.; Writing—review and editing, W.T. and A.J.

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

Abbreviations

IMEP	indicated mean effective pressure, MPa
HRR	heat release rate, J/degree
COV_{IMEP}	coefficient of variation of indicated mean effective pressure, %
Q_{norm}	normalized heat release
σ_{IMEP}	standard deviation of indicated mean effective pressure, MPa
LHV	lower heating value, MJ/kg
BSEC	brake specific energy consumption, MJ/kWh
DF	dual-fuel mode
ID	ignition delay, degrees
CD	combustion duration, degrees
SID	spread of ignition delay
SEC	spread of end of combustion
f(IMEP)	probability density of indicated mean effective pressure
CI	compression ignition engine
TDC	top dead center
CA	crank angle
SOI	start of injection
NO_x	nitrogen oxides
THC	total hydrocarbons
CO	carbon monoxide
PM	particulate matter
φ	crank angle, degrees

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