

Article

Research of Parameters of a Compression Ignition Engine Using Various Fuel Mixtures of Hydrotreated Vegetable Oil (HVO) and Fatty Acid Esters (FAE)

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Abstract: The present study is aimed at studying the energy and environmental performance at various engine loads (*BMEP*) with identical start of injection (SOI) for all fuel types. The combustion parameters for the fuel mixtures were analyzed using the AVL BOOST software (BURN subroutine). Five different blends were tested, consisting completely of renewable raw materials based on hydrotreated vegetable oil (HVO) and fatty acid methyl ester (FE100), and the properties of diesel fuel (D) were compared with respect to these blends. The mixtures were mixed in the following proportions: FE25 (FE25HVO75), FE50 (FE50HVO50), FE75 (FE75HVO25). In this study, diesel exhaust was found to produce higher NO_x values compared to FE blends, with HVO being the lowest. Hydrocarbon and smoke emissions were also significantly lower for blends than for diesel. Possible explanations are the physical properties and fatty acid composition of fuel mixtures, affecting injection and further combustion. The results showed that blends containing more unsaturated fatty acids release more nitrogen oxides, thus having a lower thermal efficiency compared to HVO. No essential differences in CO emissions between D and HVO were observed. An increase in this indicator was observed at low loads for mixtures with ester. CO₂ was reduced in emissions for HVO compared to the aforementioned blends and diesel. The results of the combustion analysis show that with a high content of unsaturated fatty acids, mixtures have a longer combustion time than diesel fuel.

Keywords: diesel engine; fatty acid esters; combustion; performance; emissions



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1. Introduction

As a result of active industrialization, there has been a huge increase in the consumption of fossil fuels and, as a result, it threatens the ecological balance of the Earth [1,2]. The transport sector is the main source of more than one third of all environmental pollution [3,4]. In this regard, the European Environment Agency in its report “SOER-2020” [5] focuses on the transformation of key systems for the rational use of energy resources, fundamental changes in the systems of production and consumption of fossil resources, which are at the heart of environmental problems.

In modern conditions, road transport has taken an important place in human life, which has increased the dependence of humankind on energy [6,7]. Such a large number of diesel vehicles carry a heavy load on the environment [8]. The constantly growing global energy consumption and global climate change in consequence of greenhouse gases have led to the need to look to find solutions for renewable energy sources [9–12].

This led to the development of a new scientific direction based on the technological stages and peculiarities of processing various types of raw materials to obtain renewable and environmentally friendly fuels [8,13,14]. In many countries, conditions have been created for industrial research of renewable fuels [15–17].

Biodiesel is one of the most budding substitutes for diesel fuel [14,18–21]. Biodiesel can be made from vegetable oils, animal fats, waste oils and other raw materials by carrying out the transesterification reaction with alcohol and a catalyst [2,22–31].

Biodiesel is called an “environmentally friendly” source of energy, since its impact on the environment is much less compared to petroleum products [21,32]. Thus, the content of aromatic hydrocarbons and other substances with carcinogenic, mutagenic, teratogenic properties for living organisms in biodiesel is lower than in petroleum products if it gets into soil or water, as biodiesel is completely decomposed by microorganisms [33]. Biodiesel is essentially free sulfur, which results in an essential reduction in SO₂ emissions into the atmosphere in contrast to conventional diesel fuel [34]. Due to the higher percentage of oxygen, the combustion process of biodiesel is much more efficient than diesel fuel [35,36]. According to most studies [13,37–44], biodiesel significantly reduces the content of hydrocarbons, particulate matter, soot particles, carbon monoxide and aromatics in the combustion exhaust gases compared to traditional diesel fuel.

Currently, the technological development of society allows us to focus on a gradual transition from diesel fuel to more environmentally friendly biodiesel. The advantages of biodiesel are that it does not require engine modifications [45,46]. Additionally, the key aspects will be to reduce dependence on oil, which, firstly, will ensure national energy security, and secondly, will protect the environment.

When operating an engine on biodiesel, there are a number of important aspects to consider: physical and chemical properties, performance and exhaust emissions.

Higher viscosity biodiesel blends affect engine performance [47–49]. The increased viscosity of the fuel results the operation of the fuel pumps and the filtration system, as a result of which the fuel supply to the combustion chamber is disrupted, the combustion efficiency decreases, and the fuel consumption increases [50,51]. On the other hand, it is inversely proportional to the amount of acid double chains. This explains the high viscosity values of biodiesel produced from highly saturated feedstocks [52]. For this reason, the viscosity of biodiesel must be within the limits defined by international standards for biodiesel. As can be seen from Table 1, the studied samples are within the norm in accordance with the EN 14214 standard [26].

Table 1. Physicochemical properties of the analyzed fuels.

Sample	D100	HVO100	FE25	FE50	FE75	FE100
Density at 15 °C [kg/m ³]	823.00	776.00	800.30	824.60	848.90	873.20
Viscosity at 40 °C [mm ² /s]	3.5	2.9	3.2	3.4	3.9	4.3
Sulphur content [mg/kg]	7.25	4.16	4.12	4.27	4.37	5.41
Water content [mg/kg]	85	20	120	150	400	460
Flash point, °C	45	55	82	109	136	163
Hydrogen, %	0.1300	0.1520	0.1460	0.1400	0.1340	0.1280
Carbon, %	0.8700	0.8480	0.8233	0.7985	0.7738	0.7490
Oxygen, %	0.0000	0.0000	0.0308	0.0615	0.0923	0.1230
C/H	6,7%	5,7%	5,6%	5,7%	5,8%	5,8%
Lower Heating Value (LHV) [mg/kg]	42.70	43.70	42.30	41.05	39.53	38.00
Cetane number	45	76.89	69.95	64.52	59.41	58

One of main properties of the fuel is also density, which can be related to a number of other properties, for instance the cetane number, which is a parameter indicating the ignition delay time of the fuel. The sulfur content of biodiesel blends declines as the percentage of biodiesel increases [44].

Investigation was controlled on a single-cylinder diesel engine by Jayaprabakar et al. [53] which used rice bran and algal methyl ester biodiesel mixtures with time-variable injection. Biodiesel blends have demonstrated poor fuel consumption for rice bran methyl ester mixtures to compare with algal methyl ester. However, BTE was increased for mixture with rice brain as opposed to algal methyl ester on account of higher calorific value. Mikulski et al. [54] used a four-cycle common rail diesel engine for research the swine lard methyl esters. It was noted that brake specific fuel consumption was increasing for the B25, B50, B75 blends by an average of 3.2%, 8.5%, 13.8%, respectively. Thus, addition of methyl ester increased the BSFC. It was accounted for lower heating values compared with diesel fuel and decreased ignition delay of the experimental blends. On the other hand, was showed the rise in BTE values on increasing biodiesel percent that can explain by the heating values of fuel mixtures. Another result for brake thermal efficiency was observed [55] to research of waste fried oil methyl ester in proportions of B50, B90, and B100. The higher the concentration of the mixture, the lower BTE was compared to mineral diesel. This conceivable explained by higher viscosity of blends. It should be noted that in another study [56], brake thermal efficiency for waste cooking oil was most lowest compared with diesel but for WCO emulsion was slightly higher. The cause can be poor combustion of the injected fuel due to high density and viscosity. Isik et al. [57] studied the n-butanol addition to waste cooking oil biodiesel with diesel fuel. For blend B20 with WCO/diesel, BSFC had highest values to compare with testing for n-butanol fuel with 10% and waste cooking oil also with 10% biodiesel/diesel. This could be attributed to the oxygen attend in the B20 that upgrades the combustion feature but have poor atomization. Chuah et al. [58] carried out experiments on a six cylinder internal combustion engine and found that BSFC of waste cooking oil methyl ester linearly increased due to the increased percentage fuel blends. However, BTE was slightly lower level for all mixtures to compare with diesel fuel at low and high engine loads.

The literature review based on the research of combustion characteristics involving such as cylinder pressure, rate of heat release and ignition delay, were studied at [54,58–61]. In this investigation, we found results indicating that the combustion of the biodiesel fuel began a little while later in compare of the diesel fuel. This can be explained due to longer injection delay of the biodiesel in contrast to the diesel fuel. Biodiesel, as known, has higher viscosity as opposed to diesel fuel and this important feature leads to higher rubbing around the needle injector [62]. This slow needle motion, and as a consequence, the longer-lasting injection delay, in spite of the fact that the cetane number of the biodiesel is slightly higher compared with the diesel fuel, leads to a lower volatility and higher viscosity of the biodiesel and poor atomization, and the mixture composed with air during the ignition delay time leads to a delay in starting the combustion of the biodiesel. In addition, the peak of the cylinder pressure and the peak of the heat release rate for the biodiesel are higher than the opposite of the diesel fuel. Additionally, the lower heating value of the biodiesel leads to a decrease of the heat release rate [30]. The result of peak pressure for waste cooking oil was slightly lower in comparison with diesel fuel when it was substituted for the fuel injection pressure and timing injection [63].

It was noted [39] that the use of various mixtures of biodiesel insignificantly reduces engine power and conducts to the increase of fuel consumption, but at the same time there is a significant decrease in most harmful substances. The only exception is NO_x [47,64]. Other studies [46] have been conducted in which BSFC was increased with an increasing percent of biodiesel blends owing to the lower heating value of the biodiesel. Another publication [65] observed that when using mixtures with diesel fuel, Brake Thermal Efficiency decreased in contrast with diesel due to the greater accumulation of energy during the ignition delay. Additionally, CO, CO_2 and smoke have decreased with biodiesel fuel mixture [59]. However, in [66,67], biodiesel blends had more particulate matter compared with D100.

After studying various literature reviews of alternative fuels for diesel engines [68–72], the authors of this research concluded that the literature does not discuss biodiesel based

on fuel mixtures from second generation biofuels, HVO, and fatty acid methyl ester, so it was motivation to research the combustion, energy and ecological parameters for these mixtures and discuss various aspects.

Given the above information, it is vital to investigate the effect of other mixtures in various percentages. The analysis of the physicochemical properties of these mixtures will aim at a comprehensive assessment of their potential for future operation in a diesel engine in accordance with the standards.

2. Materials and Methods

2.1. Fuel Preparation

In this research, the properties of mixtures for a diesel engine were considered in accordance with three standards in European Union standards for fuel. EN 590—this standard applies to diesel fuel intended for diesel engines and EN 14214—fuel for internal combustion engines. Fatty acid methyl esters (FAME) for internal combustion engines. Specification and Testing methods.

Additionally, Neste Renewable Diesel is separate recommendation for HVO. Considering HVO contains of paraffinic hydrocarbons, it cannot match the requirements of EN 14214 which was created and reasonable only for fatty acid methyl esters—FAME. Meanwhile, HVO matches EN 590 with the exception of the property for minimum density. Therefore, the article used the fuel standard of Neste Oil.

This research introduces the conclusions of a study of the 5 different mixtures, entirely consisting of renewable raw materials. During experimental part these blends were compared to diesel fuel (D). Mixing was carried out in the following proportions. Blends hydrotreated vegetable oil (HVO) and fatty acid methyl ester derived from duck fat (FE100) mixed in the ratio FE25, FE50, FE75,

Additionally, HVO and FE100 have been considered pure. The choice of biofuel based on animal fats justified by the fact that in a lot of countries is considered as a promising source for commercial production of biodiesel fuel. The qualities of the blends are given in Table 1.

Physicochemical properties of biodiesel, in particular, viscosity, density, heat of combustion, cetane number, etc., differ from those for diesel fuel. It can be noted that the fuel mixtures presented are within the normal range. The cetane number was calculated for FE25, FE50, FE75 according Kampf methodology.

2.2. Experimental Setup, Experimental Procedure and Processing Experimental Data

Experimental studies were carried out in the laboratory of the Transport Engineering Faculty of Vilnius Gediminas Technical University, Lithuania. Fuels were investigated in a four-cylinder internal combustion engine with direct injection of fuel into the combustion chamber in the piston (Table 2).

Table 2. Engine specifications.

General Informations	1.9 TDI Diesel Engine
Number of cylinders	4/In-Line
Bore	79.5 mm
Stroke	95.5 mm
Displacement	1896 cm ³
Compression ratio	19.5
Maximum torque	180 Nm/2000–2500 rpm
Maximum power	66 kW/4000 rpm
Combustion chamber	Direct injection
Nozzle opening pressure	190 bar
Fuel injection	Single
Cooling	Liquid

The engine test bench (Figure 1) used was composed of a 1.9 Turbocharged internal combustion engine with electronic fuel injection pump. The start of the injection (SOI) was managed by the engine electronic control unit (ECU) with the one injection procedure.

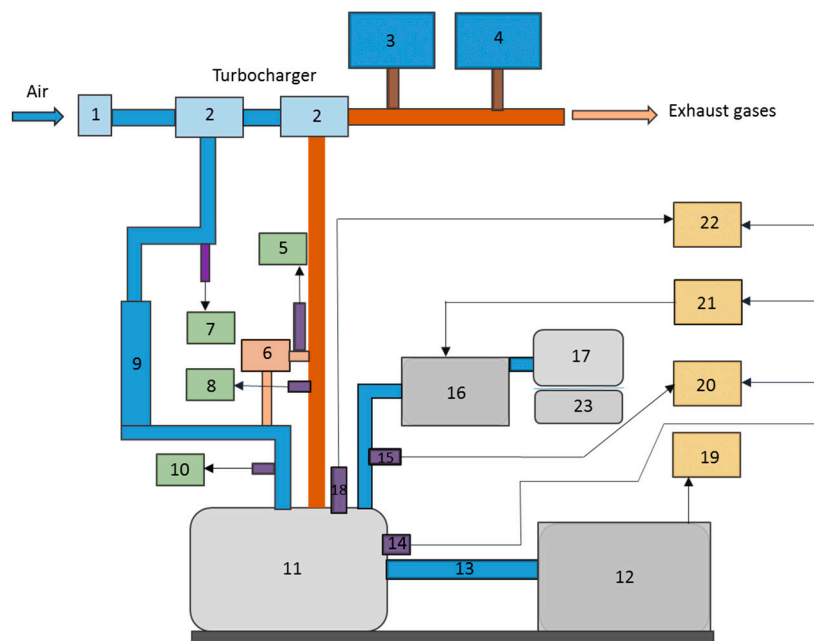


Figure 1. Schematic of engine testing equipment: 1—Air mass meter; 2—Turbocharger; 3—Gas analyser; 4—Smoke analyser; 5—Temperature sensor; 6—EGR valve; 7—Air pressure meter; 8—Exhaust gas temperature meter; 9—Air cooler; 10—Intake gas temperature meter; 11—1.9 TDI engine; 12—Engine load plate; 13—Connecting Shaft; 14—Crankshaft position sensor; 15—Fuel injection timing sensor; 16—Fuel pump; 17—Fuel tank; 18—Cylinder pressure sensor; 19—Engine torque and rotational speed recording equipment; 20—Fuel injection timing recording equipment; 21—Fuel injection timing control equipment; 22—Cylinder pressure recording equipment; 23—Fuel consumption calculation equipment.

The engine was loaded with a DC generator with an accuracy load measurement of 1.23 Nm. Measurement of load and speed, fuel and air consumption, temperatures in engine systems, composition and the smoke levels of the exhaust gases were produced simultaneously. Concentrations of carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x), in exhaust gases and smoke were determined by a five-component AVL DiCom 4000 gas analyzer (AVL—Anstalt für Verbrennungskraftmaschinen List, Graz, Austria) with a determination 0.01% for CO and 0.1% for smoke, 1 ppm for HC and NO_x. Fuel consumption was measured using a SK-5000 electronic scale (accuracy 1.0 g), air consumption was measured by air meter BOSCH HFM 5 (Bosch- Robert Bosch GmbH, Stuttgart, Germany) (accuracy 2%) Pressure of turbocharger was measured using pressure sensor Delta OHM HD 2304.0 (Delta—Delta OHM S.r.l., Padova, Italy) (accuracy 0.0002 MPa), temperature was measured using thermocouples (accuracy 1.5 C).

To determine the position of the piston at Top Dead Centre (TDC) an optical crankshaft position sensor A58M-F was used with a signal repeatability of 0.176 crank angle. Gas pressure in the cylinder was measured with an AVL quartz piezoelectric sensor GH13P, sensitivity 15.84 ± 0.09 pC/bar. An AVL DiTEST DPM 800 amplifying device was used to convert the signals of pressure and crankshaft position sensors. The pressure of gases in (100 cycles) was recorded by the high-speed software LabView Real engine indication system (LabView—LabView, National Instruments, Austin, TX, USA). The start of the fuel injection was registered by VAG-COM diagnostic equipment and fuel injection control is performed using a control signal modulator.

Engine testing data from the pressure in cylinder sensors fuel and air consumption, fuel properties, engine data and others were processed in the AVL BOOST/BURN software environment to determine start of combustion (SOC) and the ignition delay (ID) period, to calculate heat release rates, combustion duration, pressure and temperature rise, the average indicator pressure and other important features of combustion. The BOOST/BURN calculation methodology is based on the first law of thermodynamics, laws of mechanics and Vibe function [73].

The load characteristics of the engine were taken at 2000 rpm because the engine is studied mostly works in a similar rotation speed and maximum torque can be achieved at this speed. During the test, the engine was loaded with brake torque (M_B) of 30–120 Nm, (which corresponds to the Brake Mean Effective Pressure ($BMEP$) of 0.2–0.8 MPa). As the load increased, the start of fuel injection timing (SOI) was changed from 4 to 7 CAD before TDC. The engine test parameters are given in Table 3.

Table 3. Engine load and start of fuel injection.

Measuring Points	1	2	3	4
Engine speed n , rpm	2000	2000	2000	2000
Engine load M_B , Nm	30	60	90	120
$BMEP$, MPa	0.2	0.4	0.6	0.8
SOI, CAD	−4	−5	−6	−7

When the load is increased, the start of fuel injection timing must be earlier, as it takes longer to inject the fuel. This parameter is controlled during the test.

3. Results

3.1. Combustion Characteristics

Figure 2 shows the numerical analysis (using AVL BOOST/BURN software) results of combustion characteristics were presented in three parts: ignition delay, premixed combustion phase and mixing controlled combustion when the engine load ($BMEP$) is 0.4 MPa. Comparing the mixtures with each other, it can be noted that the start of combustion for HVO was earlier than the others, given the same moment of the beginning of the fuel supply process, and this, in turn, indicates the shortest ignition delay period for this fuel.

During this load ($BMEP = 0.4$ MPa), the test start of the fuel injection timing was constant at $SOI = -5$ CAD for all fuels. The start of combustion (SOC) and ignition delay (ID) according the fuel type are shown in Figure 2.

This experimental work has found that the ignition delay for HVO and mixtures is lower than for diesel, which is related with a higher cetane number for these fuels, which helps to reduce ignition delay [59]. At 4 CAD, the heat release rate for HVO is ~34% less than for D100, which indicates a reduction in the peak rate of heat release compared to diesel fuel. Diesel fuel also shows a longer ignition delay due to its higher viscosity, which results in a delayed evaporation and an atomization process that causes a longer ignition delay.

Additionally, it is worth noting that the ignition delay time also relies upon the number of carbon atoms in the molecules of blends. For our experimental mixture with FE100, we have less carbon to compare with the diesel fuel. Thus, the difference between 4 CAD rate of heat release for FE25 is ~29%, FE50 is ~23%, FE75 is ~15% and FE100 is ~14% to compare with mineral diesel. This is also confirmed by the lowest maximum combustion rate during fast combustion.

With a further increase in the concentration of ester in the mixture, there is also a noticeable tendency to decrease the maximum combustion rate during rapid ignition. Particularly, the oxygen content of the biodiesel mixtures improves the fuel–air mixing rate in the cylinder in comparison with the diesel fuel due to the extended combustion duration. In addition to this there is the vaporization of biodiesel, which is more slow than DF and

supplies a lower premixed phase of combustion, which correlates with the viscosity and density of the fuels. In addition, the cetane number effected the SOC timing [30].

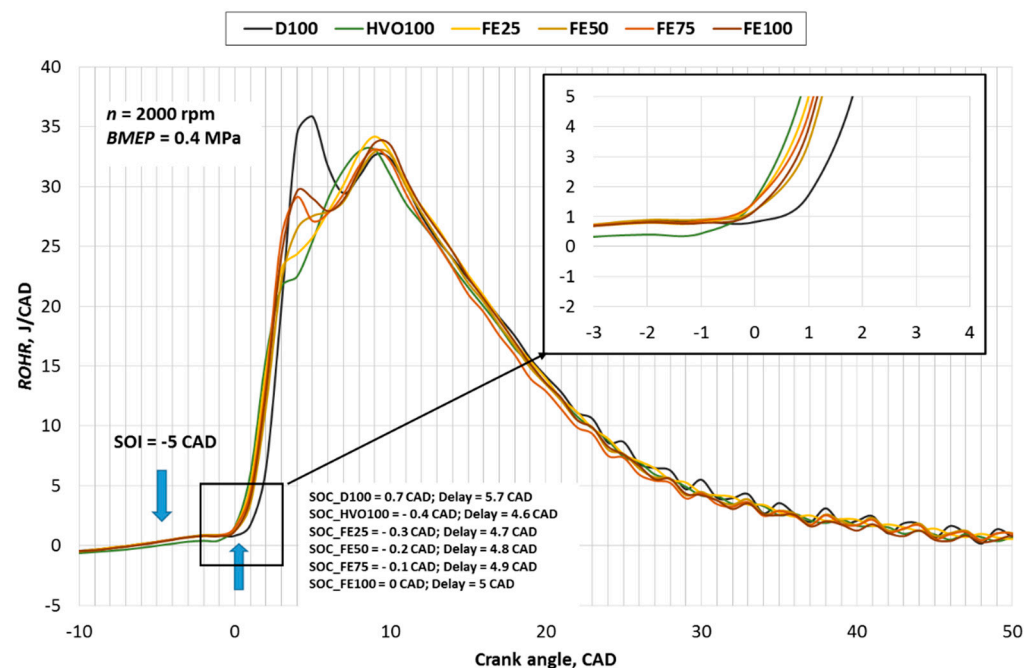


Figure 2. Heat release rate in the cylinder depending on the CAD.

The lower viscosity of HVO than diesel fuel contributes to improved the mixing characteristics in the premixed phase. This indicates that HVO evaporates faster and therefore mixes faster with the ambient air than diesel fuel. Moreover, the straight-chain paraffinic hydrocarbon HVO is more easily degraded than the diesel, and it can be noted that HVO is more easily sprayed onto the fuel, vaporized and mixed with the ambient air in the chamber [62]. For the blend with HVO, we observed the same tendency.

Analyzing the third combustion period (diffuse), it can be seen that HVO100 also reaches its peak earlier than other fuels, which is explained by the earlier onset of combustion. However, diesel fuel is characterized by the lowest maximum combustion rate in the mixing control combustion period, which is explained by a significantly larger proportion of fuel burned out during the first combustion period [74].

At 9 CAD, the heat release rate for HVO is ~1% higher than D100. Additionally, comparing mixtures with ester, we observed that for FE100 the heat release is ~3% higher equating to fossil fuel; for FE25, FE50 and FE75 the trend is ~4%, 1%, 1%, respectively.

Figure 3 presents that the temperature rise. It is very similar to the pressure rise presented in Figure 4. Additionally, the maximum for diesel that has the highest maximum on it is reached much later than for other fuels.

An absolute temperature rise for diesel fuel at 53 K/deg is observed at 4 CAD. The temperature rises for HVO at 40 K/deg is ~24% (8 CAD) less from diesel fuel, for mixtures with ester there is a similar trend in the range from ~15% for FE100 (4 CAD) at 45 K/deg to ~29% for FE50 (8 CAD) at 38 K/deg.

The reason is a later start of combustion, but we also see more intensive combustion of a pre-mixture phase, which in turn causes an increased rate of nitrogen oxide generation in the cylinder. Therefore, for all fuel mixtures, this is precisely why the rate of formation of nitrogen oxides in the phase is lower than for diesel fuel.

In the third combustion phase, diffuse heat release is very slightly different for all fuel mixtures.

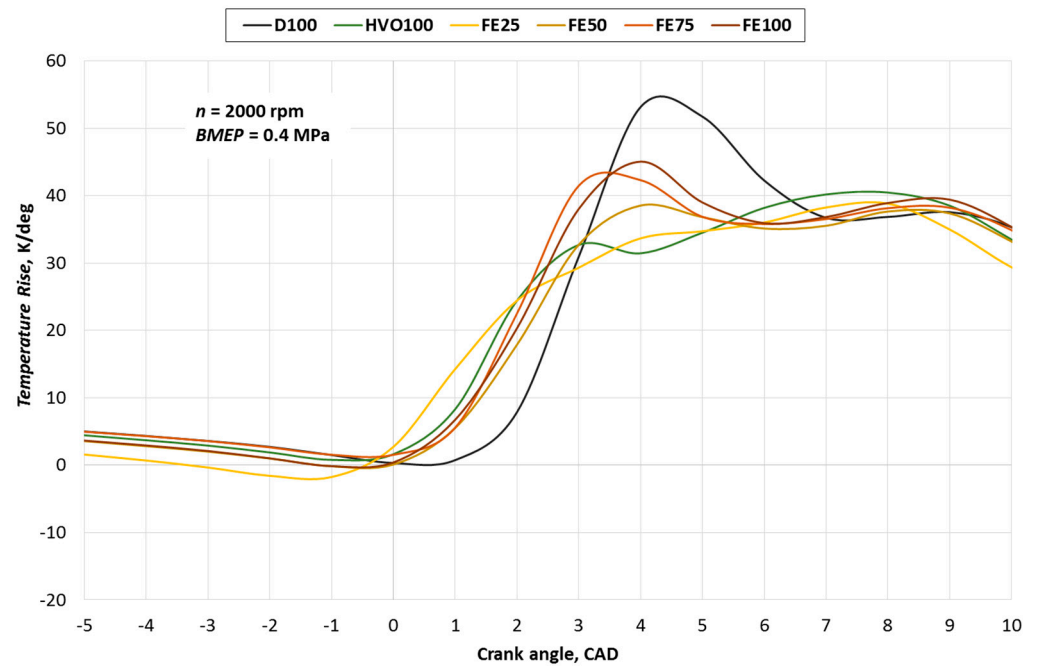


Figure 3. Temperature rise in the cylinder depending on the CAD.

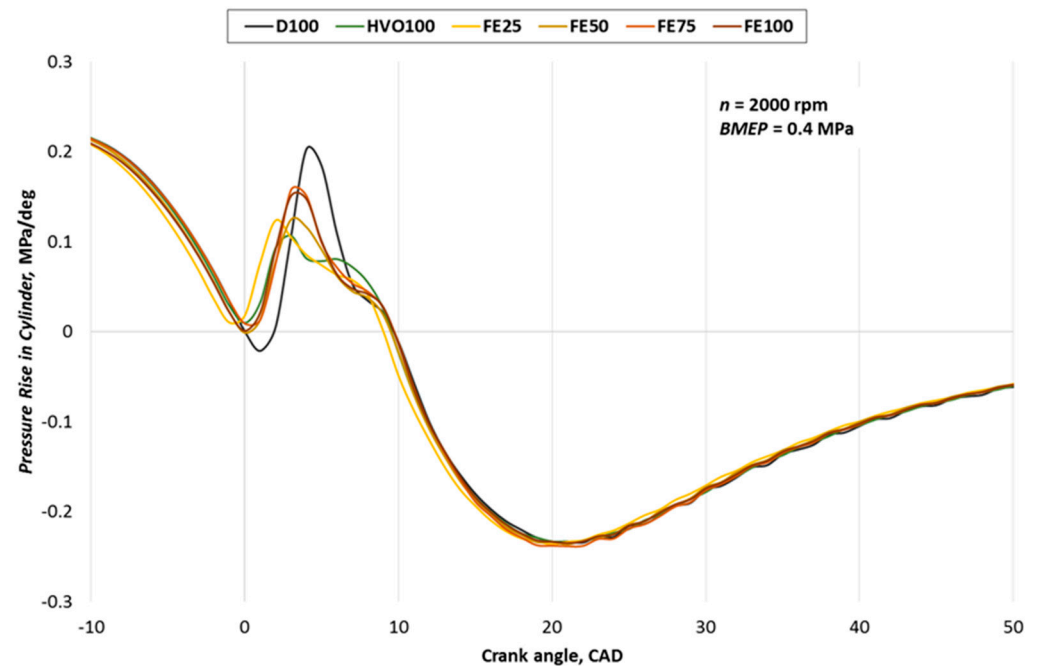


Figure 4. Pressure rise in the cylinder depending on the CAD.

The pressure rise graph shows exactly the same thing as described earlier. The moment of the beginning of combustion is clearly visible, as is the moment of a sharp increase in the rate of pressure change.

When using a pure HVO, the pressure rise is $\sim 47\%$ (3 CAD) less than of diesel fuel (4 CAD), and for the FE100 it is $\sim 25\%$ (3 CAD) lower than of the fossil fuel. For other mixtures with ester FE25 (2 CAD) and FE50 (3 CAD), it is also $\sim 38\%$ lower in comparison with diesel fuel; for FE75 it is $\sim 22\%$ (3 CAD) lower in opposite of diesel.

This graph shows all the other fuels, but shows that the maximum visible is for the diesel engine, and it corresponds to the premixed combustion phase. This is described by

the longest ignition delay period, due to which the largest amount of fuel is accumulated in the cylinder [56].

From a practical point of view, the largest peak of the pressure rise rate means the highest level of shock loads on the parts of the cylinder–piston group and the highest noise level on this type of fuel (diesel).

Figure 5 shows that with increasing pressure between the mixtures, a difference is observed from 0.02% to 1.4%. For HVO, it decreased $\sim 0.3\%$ (10 CAD) compared to fossil fuel (9 CAD), for FE25, FE50, FE75 and FE100 $\sim 0.7\%$ (9 CAD), 1% (10 CAD), 0.02% (10 CAD) and 1.4% (10 CAD), respectively.

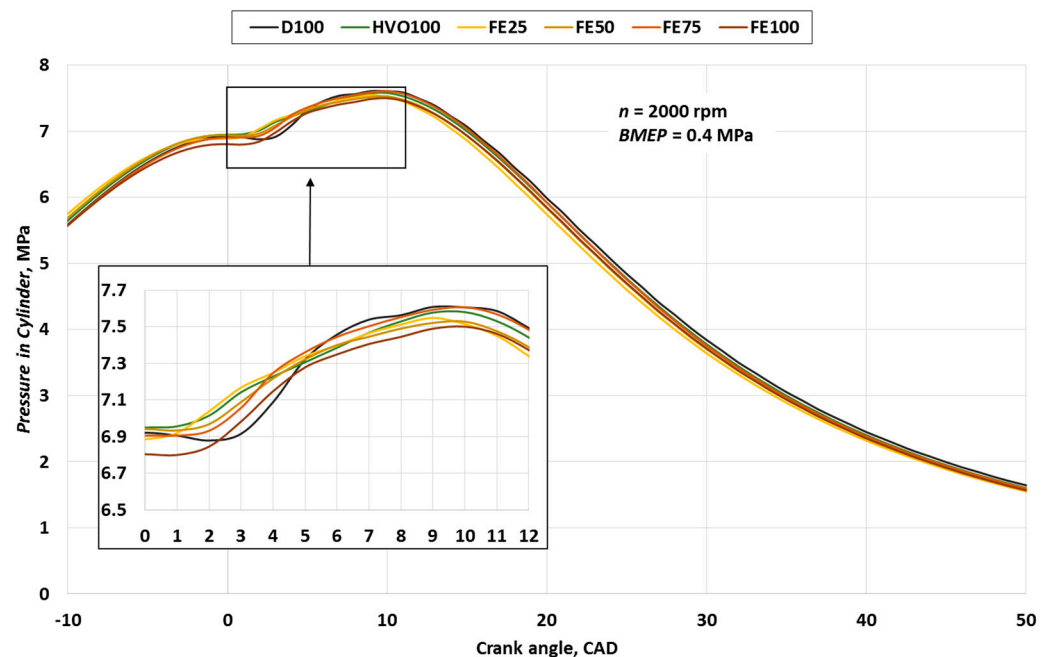


Figure 5. Pressure in the cylinder depending on the CAD.

During the experiment, it was found that compression stroke, temperature and pressure are very similar for all fuels, but after the start of combustion they differentiate from -0.4 to 0.7 CAD. However, it is from the pressure graph that all the rest are calculated, including the most important one—the rate of diffuse heat release.

3.2. Energy Indicators

The brake specific fuel consumption (BSFC_m, g/kWh) for the D100 at medium load was lower than for all mixtures, but at the same time it was higher by $\sim 3\%$ compared to HVO. We see how with a growth in the rate of ester in the fuel mixtures, as shown in Figure 6, the fuel consumption increases for FE100 $\sim 13\%$ compared to fossil fuels. If we compare D100 and other blends, we observe an increase in fuel consumption for FE25 $\sim 1\%$, FE50 $\sim 5\%$, and FE75 $\sim 9\%$ in comparison with diesel fuel, which in turn affects the combustion process.

During engine refueling with ester blends, the BSFC for these test fuels was high in contrast with diesel and HVO under all load conditions. The trend towards higher BSFC values has been reported to be associated with a lower calorific value of the test fuel, which results in more fuel being consumed to sustain the power output [75].

Diesel fuel with 0% oxygen showed the lowest BSFC to compare with biofuel mixtures. Less heating of the tested fuels (Table 1) was reported as the cause for the increase in BSFC values [71]. However, it should be noted that HVO has the highest fuel consumption rate due to the higher hydrogen content. Therefore, the calorific value per mass HVO is higher (Figure 7).

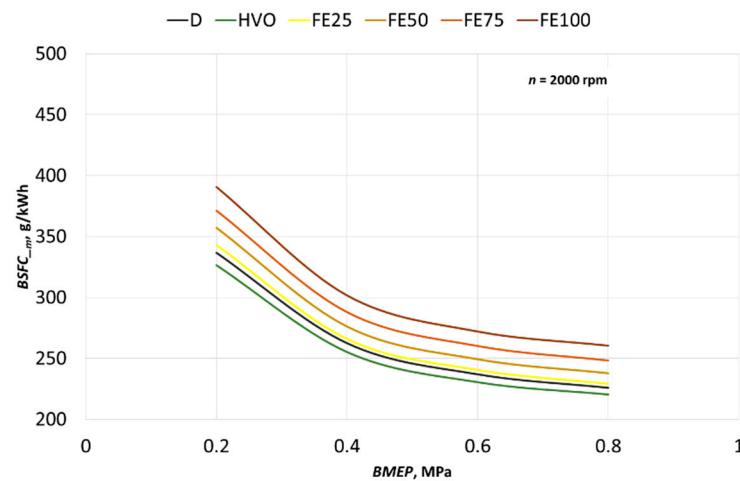


Figure 6. Dependence of BSFC_m, g/kWh on the load.

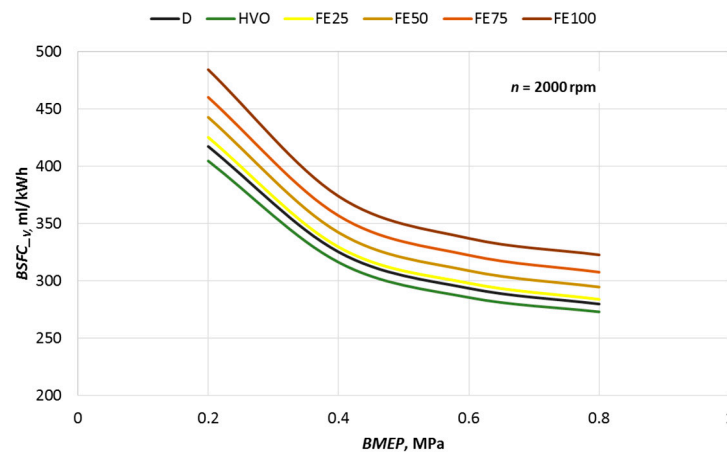


Figure 7. Dependence of BSFC_V, mL/kWh on the load.

Such a correlation can be explained by the ratio of intake pressure on the brake mean effective pressure (*BMEP*). Engine performance during low loads is relative on the altitude, considering that pressure of inlet manifold can straightly influence to the quantity of air mass induced in the cylinder [76]. Therefore, by growing elevation, the lower amount of air would enter the cylinder and *BMEP* decreases which involves additional fuel consumption with identical output power.

Figure 8 shows that BTE increases for all fuels with increasing load, which is associated with increased power. Additionally, the higher the specific heat of combustion (the amount of heat released for complete combustion) of the fuel, the lower its consumption [53]. Lower BTE values are observed for all mixtures, with the highest for HVO followed by diesel fuel. The difference between these type of fuels is ~0.5%. This is due to the specific heat of combustion.

At medium load (*BMEP* = 0.4 MPa) for mixtures with ester, we can observe a reduction of BTE at an average of ~0.3% in comparison with diesel fuel.

The resulting low BTE for mixtures can be associated with a lower heating value and increased fuel consumption in comparison with diesel. It could also be on account of their low heat consumption for higher power outputs at a given load. As the proportion of biodiesel in the mixtures increases, the BTE decreases due to poor atomization of the mixtures by reason of their high viscosity. Fuels with a higher proportion of ester gave less torque due to less energy released due to their lower heating value [77,78]. A lower BTE can also be attributed to a higher BSFC. In addition, the oxygen content grows with the

increase of percent for ester blends (Table 1), which decreases the heating value and, in turn, decreases the BTE.

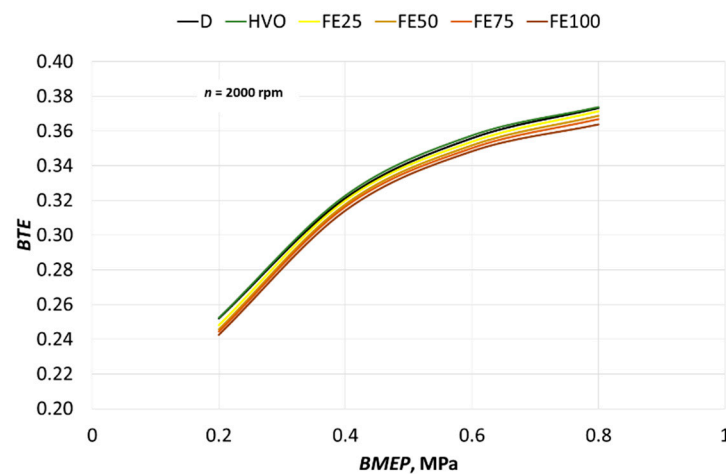


Figure 8. Dependence of BTE on the load.

3.3. Ecological Indicators

Carbon dioxide (CO_2) emissions decreased for all types of fuels with a growth in the load as shown in Figure 9. At medium load ($\text{BMEP} = 0.4 \text{ MPa}$), the CO_2 emissions of mixes including FE75 and FE100 averaged $\sim 2\%$ and $\sim 3\%$, respectively, which is higher in comparison to diesel. The higher CO_2 emissions in the mixtures are related with the higher oxygen and carbon content of the tested fuels, in opposition to pure diesel fuel (Table 1).

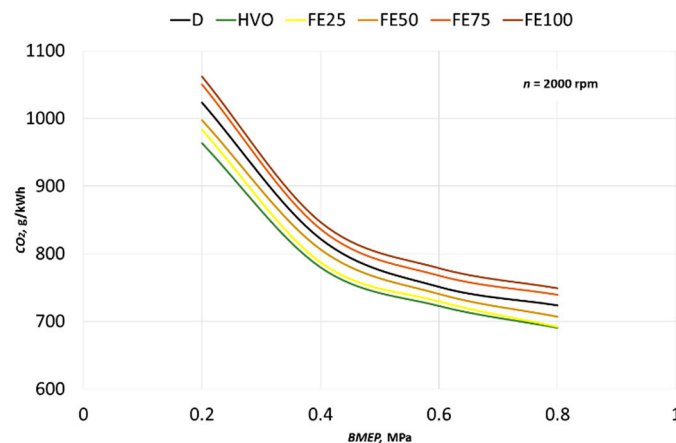


Figure 9. Dependence of carbon dioxide emissions on the load.

For HVO100 carbon dioxide emissions was $\sim 5\%$ lower than diesel fuel. Additionally, we have observed the decrease CO_2 for FE25 and FE50 $\sim 4\%$ and 2% accordingly.

Additionally, in this experiment it was found that mixtures that have a higher hydrogen/carbon (H/C) ratio contribute the most to CO_2 reduction, since their combustion is more complete (Table 1).

HVO, in turn, has a higher hydrogen/carbon (0.1520%/0.8480%) ratio, which contributes to the most CO_2 reduction for this fuel type compared to the aforementioned mixtures and diesel fuel.

During combustion, the fuel is split into CO and then oxidized to carbon dioxide. Consequently, the CO concentration can rapidly decrease with increasing temperature. The presence of hydrogen-containing substances such as water can help speed up the process.

Additionally, ester blends are oxygenated fuels, and the additional oxygen molecule influences help to achieve better combustion, resulting in lower CO emissions [79–81]. Adequate oxygen content and high cylinder biodiesel temperatures help reduce CO emissions.

Figure 10 presents the CO emissions for HVO100 at medium load, which were ~5% less than for D100, and ~15% higher than for FE100. As reported by the figure, CO emissions are lower for HVO due to the lower ignition delay, and combustion time is increased, thus contributing to the oxidation process of CO emissions. Diesel fuel has lower emissions than mixtures, which is explained by the high C/H ratio in diesel fuel.

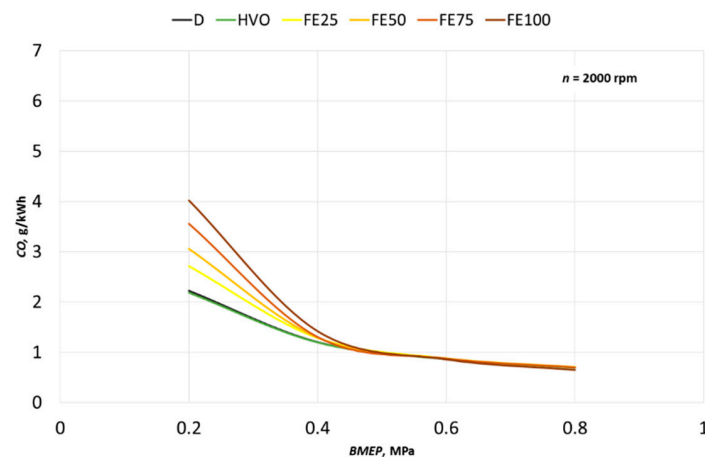


Figure 10. Dependence of carbon monoxide emissions on the load.

For blends with ester, this tendency was also higher compared to fossil fuel. For FE25, FE50 and FE75, carbon monoxide emissions were ~6%, 7%, 8%.

At $BMEP = 0.4$ MPa, there was a marked reduction in HC emissions. It is also seen that the curves of this dependence for mixtures with an intermediate concentration of ester in the fuel occupy an intermediate position between HVO100 and diesel fuel.

This can be explained by an increase in the combustion temperature of the fuel and the increase of the quality of fuel atomization over the volume of the combustion chamber with increasing load.

Additionally, low rates for mixtures can be associated with the content of fatty acids, which promote complete combustion, due to the fact that oxygen molecules are also present in the droplets of this mixture, which contributes to more complete combustion.

In part, unburned hydrocarbons can result from poor air and fuel homogeneity due to incomplete mixing before or during combustion.

Satputalei et al. [79] noticed that a higher cetane number on methyl ester decreases HC emissions in comparison with diesel under all load conditions.

All mixtures have lower values than diesel as we can see from Figure 11. For example, FE100 has hydrocarbon emissions ~14% less than D100, and HVO ~37%. On average for other mixtures was found the same tendency. FE25 was ~33%, FE50 ~30%, and FE75 ~25% lower compared with diesel fuel, while oxygen-free diesel showed the highest hydrocarbon emissions.

According to the results of experimental studies, we found that the lower the carbon content in the fuel, the less smoke will be [82].

With a decrease in the ignition delay, the combustion process starts earlier and the content of harmful substances, such as smoke, decreases (associated with a higher cetane number).

Smoke emissions are increased for D100 and HVO100 compared to blends with ester, as is illustrated in Figure 12, because oxygenated fuel contributes to the oxidation of soot. Behcet et al. [80] found that the smoke level of diesel fuel was high, while for biodiesel it decreased. Additionally, earlier soot formation for diesel fuel correlates with the expected high soot tendency of diesel fuels containing aromatics and cycloalkanes, which increase the formation of soot precursors.

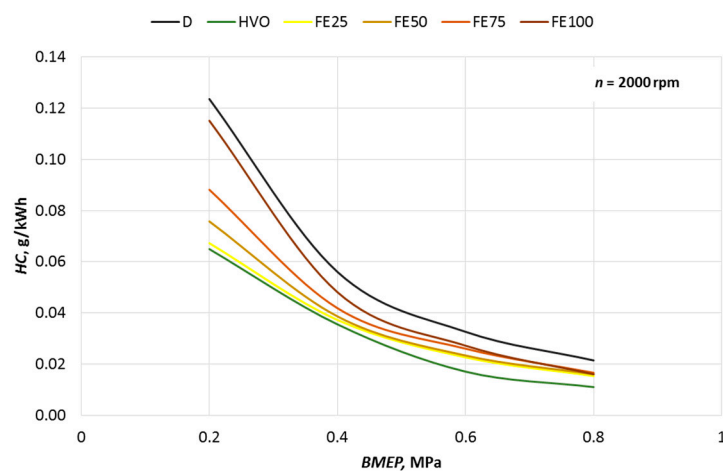


Figure 11. Dependence of hydrocarbon emissions on the load.

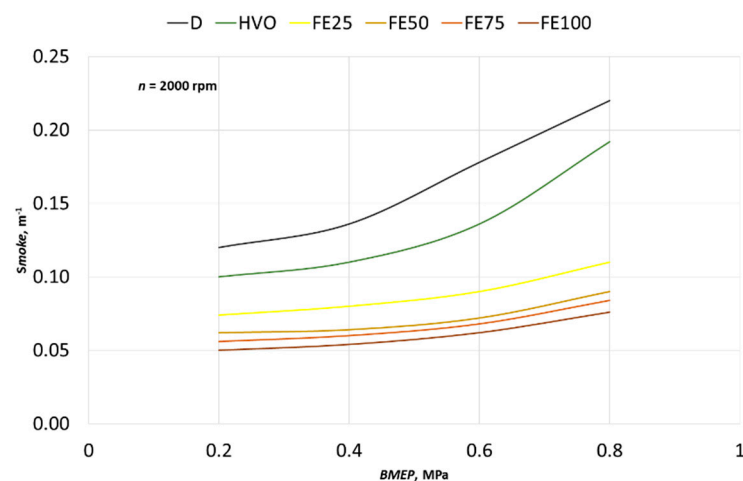


Figure 12. Dependence of smoke emissions on the load. Should be replaced by smoke.

For HVO, we can observe reduce of smoke of ~18% on average in comparison with diesel fuel. HVO belongs to the paraffinic fuel and contains a higher H/C ratio (Table 1). This type of fuel does not have aromatic hydrocarbons, sulfur and another mineral contaminations, which affect the formation of soot [83]. The mixtures also have indicators which are lower than of diesel. For FE25, they are ~47%, for FE50 ~55%, for FE75 ~58%, and for FE100 ~62%. Reduced smoke emissions at all engine loads are on account of the high mass oxygen content and lower C/H ratio (Table 1).

NO_x emissions were increased with a growing load in all samples examined, with diesel fuel being the highest indicator. The difference between D100 and HVO100 was ~19% on average between D100 and FE100 ~10% in Figure 13.

At medium load ($BMEP = 0.4$ MPa), the nitrogen oxide emissions reduce for FE25~12%, FE50 ~10%, FE75 ~10%, FE100 ~10%, and HVO ~20% in comparison with D100.

Consequently, a higher combustion chamber temperature results in higher NO_x values.

Numerous literature review have explained the influence of biodiesel on NO_x emissions due to the fact that biodiesel contains the oxygen. This improves fuel oxidation in the process of combustion, which has the effect of higher temperature.

The influence of HVO on NO_x emissions appears to be positive compared to the increase in NO_x emissions with ester/diesel mixtures. The ambiguous impact on NO_x emissions may depend on the combined effects of ignition delay, fuel injection quantity, and the distribution of the injection quantity between the pilot and main injection [84].

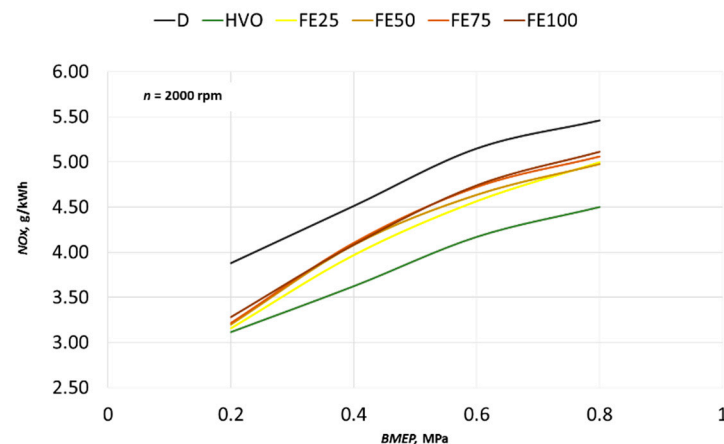


Figure 13. Dependence of nitrogen oxide emissions on the load.

Additionally, the second reason can be that a lower iodine number and oxygen content in mixtures with ester reduces the NO_x value by 20% lower than that of a diesel engine.

4. Conclusions

1. The use of fat esters results in a deterioration of their physical and chemical properties with increasing fat concentration. However, it is emphasized that the cetane number of pure fats is higher than compared with mineral diesel.
2. One of the important advantages of testing mixtures is that it lowers the levels of harmful pollutants in the exhaust of diesel engines. One exception to this is oxides of nitrogen (NO_x), which are implicated in ozone and smog formation.
3. It was determined the D100 has a higher exhaust gas temperature in comparison with other mixtures due to the longer combustion process of the fossil fuel.
4. We investigated whether when using blends, the combustion started earlier, in opposition to when using diesel fuel. The testing blends could give a positive impact on efficiencies (NO_x -Soot trade-offs) and GHG, especially for a heavy-duty engine application. In addition, FE25, FE50, FE75 and FE100 blends reduced CO from approximately 7 to 13%. CO_2 reduction was observed for all mixtures except FE75 and FE100. Additionally, the concentration of hydrocarbon emissions decreased for all mixtures. Lower values have HVO to compare with D100. Smoke emissions were also significantly lower for blends than for diesel. On average, HVO was reduced by ~18%, FE25 by ~47%, FE50 by ~55%, FE75 by ~58%, and FE100 by ~62%. We have shown increased NO_x emissions for all blends and fossil fuel with an increasing load.
5. The brake thermal efficiency (BTE) indicators for the mixtures were lower compared to D100 for all loads, but for HVO, they were slightly higher than diesel (~0.5%). Considering that the heating values of the mixtures are lower in comparison with diesel fuel, it was found that the brake specific fuel consumption (BSFC) for all types of the mixtures was higher in opposition to fossil fuel, besides HVO, which has a higher LHV.

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Abbreviations

BTE	brake thermal efficiency
BSFC	brake specific fuel consumption
BMEP	Brake Mean Effective Pressure
CAD	crank angle degree
CO	Carbon monoxide
CO ₂	Carbon dioxide
D	diesel fuel
ECU	electronic control unit
FE100	fatty acid methyl ester from duck fat
FE25	25% fatty acid methyl ester derived from duck fat and 75% HVO
FE50	50% fatty acid methyl ester derived from duck fat and and 50% HVO
FE75	75% fatty acid methyl ester derived from duck fat and and 25% HVO
FAME	Fatty Acid Methyl Ester
HC	Unburned Hydrocarbons
HVO	Hydrotreated Vegetable Oils
M_B	brake torque
CN	cetane number
ID	ignition delay
NO _x	Nitrogen oxides
O ₂	Oxygen
SOI	start of injection
SO ₂	sulfur dioxide
LCV	lower calorific value
TDC	top dead center
WCO	waste cooking oil

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