Combustion Characteristics of Several Types of Biofuel in a Diesel Engine

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Alternative propulsion fuels for diesel engines were highlights for a variety of research activities within a large number of R&D labs in the world. Curiosity, as well as basic interest in alternative fuel has been immense because of its reproducibility, renewability, as well as good ecological characteristics. Another reason for this interest is a favorable impact on the economies of the countries engaged in their production, by processing that kind of bio fuel. This paper presents the results of testing and research of the three types of alternative fuel gained from bio oil. Analyses and comparison of the dynamics of combustion process of these alternative fuels were done in relation to the reference diesel fuel. Burning process of methyl esters rapeseed (RME100), soya oil (SME100), waste cooking palm oil (PME100), as well as of regular euro diesel fuel were investigated. It was found that these fuels were highly usable in diesel engines if they had been produced according to a proper standard procedure under standard EN 14214. The application of bio fuel is also very interesting for the propulsion of agricultural engines and machinery, because it enables the individual farming of bio fuel products necessary for the appropriate own machinery.

Keywords: *diesel engine, combustion process, biofuels, heat release rate*

1. INTRODUCTION

Diesel engines today represent one of the most important sources of mechanical energy for powering and propulsion of various machines and devices in the modern world. The first high-energy air self-propelled ignition engine was first introduced in patent application [1], by engine constructor Rudolph Diesel, who considered that in the future vegetables oils will prevail and will have same influence as diesel mineral fuel for which he had constructed and promoted patent activity for, i.e. diesel engine, [2]. After more than one hundred years later, in modern times, all over the world, a huge research in the large field of production and usage of bio fuel on existing diesel engines is done. When diesel engines are concerned, biodiesel fuel (BD), produced from various derivates from bio or animal origin, and from various wasted eatable cooking oils .can be used in the engines that work on the principle of diesel thermodynamical cycle [1]. This paperwork has presentation of universal testing process of fuel combustion of different biodiesel sorts due to following dynamic process of combustion in diesel engine which also shows that certain biodiesel fuels produced under standard EN14214 are examples of good quality fuels, and that their use has no side effects to the engine. The same was concluded in scientific articles [3,4,5,6]. On the other hand,Yuksek *et al.* in [7] showed that bio diesel fuel produced from rapeseed bio-oil decreases abrasion wear in comparison with conventional engine fueled on diesel, but often there is a slight degradation of engine oil viscosity which requires more frequent engine oil change.

When it comes to the quality and dynamics of the combustion process of soybean, rapeseed and palm oil methyl esters, compared to diesel fuel, it can be estimated that they are quite similar, which classifies these fuels as very high quality and suitable for diesel engine operation. The same is stated in the paper works [8,10,11,12], but in [9] had been noted slightly larger differences in the rate of heat release when working with pure rapeseed oil. Combustion process investigations were also conducted with the fuels produced from non-edible biodiesels, namely waste cooking oil and exotic macadamia oil [13] and these fuels have been shown to make little difference in engine performance and operational performances compared to diesel propulsion. Engine operation with used fish oil is also possible [14] in different oil ratios.

Research of this type allows a better view of changes in the engine workflow of biofuels. This allows further optimization of the workflow in terms of as close performance characteristics as diesel fuel and optimizing diesel engine exhaust emissions. The Heat release rate - HRR is of direct influence on pressure and temperature in cylinder of engine and thus on performance and therefore exhaust emission.

For this reason, HRR research is very important in terms of optimizing the above-mentioned parameters as well as the engine propelled on biodiesel. Goal of this paperwork is to present results of research process of combustion during testing various biodiesel fuels, and diesel engine with direct injection of biodiesel fuels.

The thing that, namely, could be a problem with wider biodiesel consumption for fuel production is the quantity of plants for such purposes: it should not be

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allowed for the plants used for engine fuels to be cultivated in the same fields as the edible ones. Also, wider production of bio fuels can lead to an increase in the price of food in the world market, a trend that has already been observed. On the other hand, there are plants that provide inedible oils and at the same time do not require high quality land, which makes it possible to use such land successfully. One example is the cultivation of algae to produce primary energy for engine propulsion [15]. On the other hand, the great benefit of using biofuels in internal combustion engines is a significantly lower global emission of CO_2 [16,17] in comparison to the conventional ones.

2. EXPERIMENTAL SETUP

The investigations of the combustion process characteristics presented in this paper were performed on a single-cylinder diesel engine with direct injection of domestic production LDA450. The engine is manufactured for the needs of agricultural machinery, small boats, irrigation pumps and power generators.

The engine has very robust construction so that it was suitable for reconstruction into a research engine for the purposes of concrete testing. Table 1 shows the technical characteristics of this engine and Figure 1 shows the scheme of the test installation as well as the engine image with the positions of the main sensors used in the measurements, figure 2.

Table 1. Technical specifications of test engine

Engine Type	DMB - LDA450
Engine specification	4-stroke, air-cooled, diesel
Aspiration	Supercharged ,ROOTS compressor
Bore x stroke (mm)	85 x 80
Number of cylinders	1
Cylinder volume (cm^3)	454
Compression ratio	17.5: 1
Speed range min-max (rpm)	1000 - 3000
Fuel injection pressure (bar)	185
Rated power DIN 70020 (kW)	7.3

Testing was done on The FA50 / 30 SL type electric dyno, Borghy & Saveri, Italy, for measurement engine performance. All studies were performed at a nominative load ratio 1600 rpm, with load levels varying from 25%, 50%, 75%, and 100% of described rpm, Table 2.

The specified engine speed was selected because of the extremely stable engine operation in this mode.

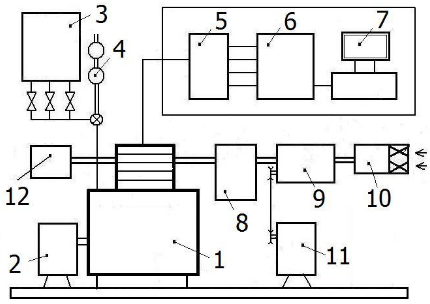
Table 2. Engine test conditions

Speed (rpm)	1600
Load (%)	25, 50, 75 and 100
Lubricating oil temperature (^{0}C)	85
Ambient temperature (^{0}C)	22
Stabilization time for nominative revolutions per minute (min)	15

All the measurements were repeated three times and the gain values for the three measurements were adopted, repeatedly.

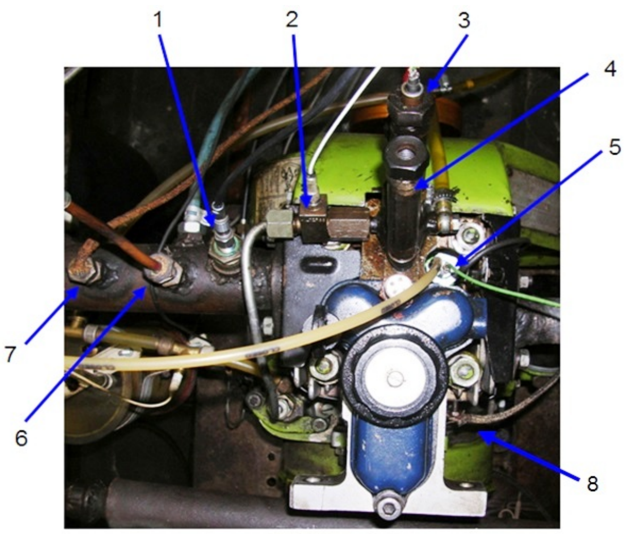
Further, the Kistler type 7031 of piezoelectric pressure transducer had usage for a 7507 water-cooled housing that was used for monitoring the pressure in the engine cylinder. This pressure transducer was integrated with a Kistler type 5001 charging amplifier of the range 1-100 kpC/V and with a linearity of +/- 0.05.

The crankshaft angle encoder was optical COM1 Austria type with an angular split of 360 slots on a rotating disc. During cylinder pressure measurement crank angle signal was divided by software,so that the acquisition was performend with the resolution of 0.2 crank angle degrees. Fuel injection pressure was measured with an AVL type 7ZP65 piezoelectric pressure transducer that was mounted at the end of the high-pressure pipe to the injector. Air flow was measured with a laminar flowmeter CUSSONS P7-201 of the range 0-100 l / s.



1- test engine, 2- electrical dynamometer, 3- fuel tank, 4- flowmeter, 5- amplifier of signal, 6- measuring-acquisition system, 7- computer, 8- volume compensator intake, 9-Roots compressor, 10-air flowmeter, 11-electric motor, 12-exhaust system

Figure 1. Schematic of test bench



1-Intake manifold pressure transducer, 2- Injection pressure transducer, 3- Injector needle stroke sensor, 4- Injector, 5- Cylinder pressure transducer, 6- U-tube connection, 7- Intake charge temperature encoder, 8- Temperature sensor engine heads

Figure 2. Sensor positions on the engine

Continuous measurement for several characteristic values (parameters) of the engine workflow was performed by using the ADS2000 metering system.

Fuels used in the survey were manufactured according to EN14214 and the reference diesel is according to EN590 Euro diese standard. Research results for pure 100% fuels are presented using the fuel labels: (D100) reference diesel, palm oil methyl ester (PME100), soybean oil methyl ester (SME100) and rapeseed oil methyl ester (RME100), Table 3. These labels will also be used in the survey results diagrams. as well as in the discussion of the results obtained.

Table 3 Specifications of testing fuel

	D100	PME100	RME100	SME100
Density(kg/l)	0.828	0.881	0.880	0.885
Net calorific value H_d (kJ/kg)	41494	35626.5	37631	37251
Kinematic viscosity (mm^2/s)	3,16	4,6	4,59	4,3
Air-fuel ratio Lo (kg/kg)	15.083	12.262	12.655	12.475
Temperatures of ignition (^oC)	55 [203]	183 [203]	80 [203]	178 [203]
O2 (kg/kg)	0	0.1210	0.1200	0.1145
C (kg/kg)	0.8496	0.7670	0.7720	0.7688
H2 (kg/kg)	0.1504	0.1120	0.1200	0.1167

Table 3 shows the main characteristics of the fuels used for testing the engine during the survey.

2.1 Heat Release Rate - HRR

The dynamics of the combustion process of the tested fuels had important role, because the key parameters of the engine workflow could be estimated on the basis of these dynamics. This primarily refers to the speed (intensity) of combustion and the duration of certain stages of the combustion process. The dynamics of the combustion process presented via HRR (Heat Release Rate) is also a key parameter in modeling engine exhaust emissions.

By general determination of heat release rate (HRR), Cylinder volume (proper volume $V=V_h+V_c$) was considered to be open thermodynamical system. The total elementary process energy dQ consisted of dQ_f energy developed by releasing internal chemical energy in combustion process, and energy (heat) lost by open boundaries of the thermodynamical system, i.e. from the cylinder walls, i.e.walls of work space dQw, and energy in enthalpy that is lost due to the flow of working gas through the cracks and gaps in the engine working space, primarily between the piston and the cylinder liner.

This can be written:

$$dQ = dQ_f - dQ_w + dm_{in}h_{in} - dm_{ex}h_{ex} - dm_{eel}h_{eel} \quad (1)$$

Here are next parameters:

dQ - an elementary change in the total energy input to the engine cylinder,

dQ_f - an elemental amount of energy released in the process of combusting a cycle amount of fuel,

$dm_{ex}h_{ex}$ - an elemental enthalpy of gas passing through the engine exhaust valve,

$dm_{in}h_{in}$ - an elemental enthalpy of gas passing through the engine intake valve,

$dm_{eel}h_{eel}$ - an elemental enthalpy of gas leaking through unsealed places in the engine cylinder (mainly through piston-cylinder gap),

dQ_w - an elemental amount of energy (heat) delivered to the walls of the workspace by convection and radiation.

Further analysis of this model would lead to complicated systems of differential equations, so that we considered somewhat simplified model of heat release rate calculation proposed in [19] and [20].

If we consider the high pressure part of the working cycle (intake and exhaut valves are closed), the simplified method is based on observing the increase in the amount of heat delivered to the gas (positive or negative) between two adjacent discrete points on the pressure line , which we will denote by ΔQ, Figure *3* If that amount of heat is added to the amount of heat handed over to the walls of the workspace; ΔQw sum represents the total amount of heat released by the combustion of fuel at a given interval, i.e. ΔQ_f. In doing so, the amount of energy lost through leakage through unsealed places was neglected, and it is negligible when properly sealed.

Therefore, equation stands for heat realization:

$$\Delta Q_f = \Delta Q + \Delta Q_w \text{ v} \quad (2)$$

Where:

ΔQ_f – is the part of the heat released by combustion of fuel between points 1 and 2,

ΔQ - the portion of heat that is delivered to the working gas between points 1 and 2,

ΔQ_w - the portion of heat that is transferred to the walls of the workspace between points 1 and 2.

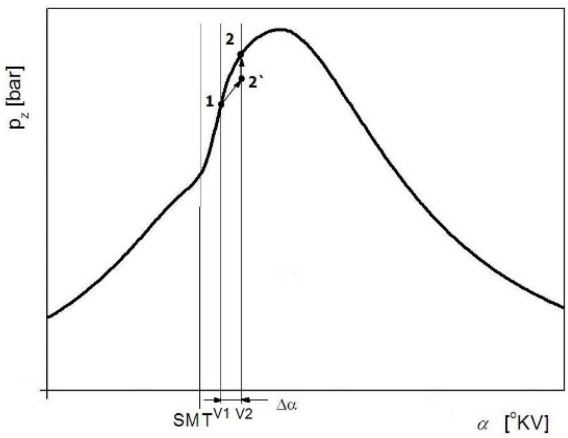


Figure 3. Cylinder pressure diagram and the change in the state of gas between points 1 and 2

The fraction of heat transferred to the working gas in the interval between the observed points 1 and 2, increasing its energy potential, ΔQ, it can be virtually considered as taking place in two steps, as shown in Figure 3. The first step is adiabatic, isentropic compression (or expansion) 1-2 'from volume V_1 to V_2 without heat transfer to gas. The second step is the isochoric heating of the working gas between points 2′-2

The required thermodynamic gas parameters during these two thermodynamic changes can be obtained from the ideal gas equation for the points 1 and 2:

$$p_z V_z = mRT_z \tag{3}$$

From the equation of ideal gas, we should have the following:

$$p_{2'} = p_1 (\frac{V_1}{V_2})^k \text{ ; } K = C_p/C_v \quad) \tag{4}$$

$$T_1 = \frac{p_1 V_1}{mR} \text{ ; } T_2 = \frac{p_2 V_2}{mR} \tag{5}$$

$$T_2^{'} = \frac{p_2^{'} V_2}{mR} = \frac{p_1 V_1^k V_2^{k-1}}{mR} \tag{6}$$

and with these values T_2 and $T_2^{'}$:

$$\Delta Q = \Delta Q\Big|_1^2 = mc_v (T_2 - T_2^{'}) = \frac{c_v}{R} V_2 (p_z - p_2^{'}) \tag{7}$$

The above equation shows the heat gain between points 2 and 2 'so that finally after the shift, the amount of heat released in the combustion process and transferred to gas from point 1 to point 2 is obtained as:

$$\Delta Q = \frac{c_v}{R} V_2 [p_2 - p_1 (\frac{V_1}{V_2})^k] \tag{8}$$

In [20] it is shown that the error due to the applied approximation (virtual heat transfer to gas in two steps) is very small (less than 0.15%) in the case of fine angular increment of cylinder pressure acquisition.

The application of the above equation requires knowledge of the thermodynamic parameters of the gas in the combustion chamber. In the paper [20] several approaches were presented that can be used to determine the thermodynamic parameters of a gas in a diesel engine cylinder as well as their influence on the accuracy of the determination of the heat release law.

2.2. Part of the heat transmitted to the walls of the workspace

The engine working space is surrounded by the piston head, the valve head with valve mushrooms and the cylinder liner wall. These elements have different temperatures and, in addition, the flow conditions in their surroundings are not the same, so the heat transfer coefficient on these elements may have different values. In practice, however, it is most often calculated with the same value of the heat transfer coefficient and different surface temperatures of these elements.

This can be written:

$$\frac{dQ_w}{d\alpha} = \alpha_w \sum_{i=1}^{3} A_{wi} (T_g - T_{w_i}) \tag{9}$$

The contact surface between the gas, on one hand, and the piston head and the cylinder head, on the other, do not change regardless of the perpendicular oscillatory motion of the piston in the cylinder, while the cylinder liner surface changes constantly depending on the position of the piston. In addition, the cylinder liner temperature is much higher at the top than at the bottom.

If, as in the case of heat transfer to the working gas, the amount of heat exchanged with the walls of the workspace between the observed discrete points 1 and 2 is observed, ΔQ_w wherein an interval of time Δt, it could be written:

$$\Delta Q_w = [\alpha_w \sum_{i=1}^{3} A_{wi} (T_g - T_{w_i})] \, \Delta t \tag{10}$$

that is, if the parameters are monitored as a function of the crankshaft angle $\Delta\alpha$, it follows:

$$\Delta Q_w = [\alpha_w \sum_{i=1}^{3} A_{wi} (T_g - T_{w_i})] \frac{\Delta\alpha}{6n} \tag{11}$$

where :

$\Delta\alpha = 6n\Delta t$ - angular increment from point 1 to point 2,

n - revolutions per minute (1/min)

It has been said earlier that the coefficient of heat transfer α_w a very sensitive parametre for defining it accurately, and that there are many ways of defining it, as a result of research by a large number of authors, but who do not yet have a universal character. The relation proposed by Hoenberg was used in this paper [21]:

$$\alpha_w = 0,013 \, V_z^{\ -0.06} p_z^{\ 0,8} T_z^{\ -0,4} (c_m + 1,4)^{0,8} \tag{12}$$

where:

V_z - is the current workspace volume

p_z - pressure and gas temperature in the engine cylinder,

$c_m = \frac{S\,n}{30} (m/s)$ - mean piston speed;

$S(m)$ - piston stroke; n (1/min) - rev. per minute.

Other well-known heat transfer coefficient models are incorporated into the heat release flow calculation program based on the exposed model can be used alternatively for example Woschni and Annand relations [22]. It must be said, however, that the calculated heat exchange between the gas and the walls of the workspace is still more an estimate than a reliable result in absolute terms. First, whatever model for the heat transfer coefficient is used, it is based on a limited amount of experimental data and can hardly be universal to all engine categories. Second, the exact temperatures of the walls of the workspace are unknown. Measuring them directly is a very demanding and complicated work, difficult to accomplish with conventional engine testing. These temperatures are usually estimated on the basis of literature data, but such estimates are quite unreliable on a case-by-case basis because the literature data generally refer to maximum values for certain engine categories, and wall temperatures change with the change of engine load.

In this paper, the values of workspace wall temperatures that had previously been estimated based on the procedure of identifying the parameters of a mathematical model of the workflow for the same engine were used [23]. For these reasons, the calculated heat transfer from the gas to the walls of the workspace is often corrected to obtain a logical result, e.g. a fine correlation good match between the calculated heat released at the end of the combustion process and the chemical energy of the fuel introduced into the engine by the cycle amount of fuel.

3. RESULTS OF THE RESEARCH

3.1 Investigation of the biodiesel combustion process in a direct injection diesel engine

The diesel combustion process is one of the basic processes that determine the quality of the entire duty cycle and the engine as a whole. Full knowledge of the physicochemical mechanism of this process is very complex because the combustion process is rather chaotic, very sensitive, even to relatively small variations of certain influencing factors. In addition, in the combustion chamber there is a diversity of physical state of the individual fuel particles, the degree of oxygen concentration as well as the temperature level in certain parts of the chamber. At the same time, the familiarity with the combustion process mechanism, gives the ability to control this process in the optimal direction.

The standard diesel fuel is Euro diesel (D100) fuel and the engines are tuned to optimally operate with this fuel. However, taking into account the fact that motor fuel can be obtained from several plants, it is of interest to study the characteristics and dynamics of the combustion process of these biofuels. In the EU, for the most practical reasons, rapeseed oil in the form of rapeseed methyl ester is used as biofuel for diesel engines-RME ($C_{15}H_{31}CO_2CH_3$), which must have the characteristics required by EN14214.

This methyl ester is a RME type biodiesel fuel and the results of combustion research of this fuel as the most prevalent in diesel engine will be discussed below, as well as the results of palm oil - PME and soybean - SME combustion research.

3.2. Gas pressure in cylinder

3.2.1 Gas pressure in cylinder for Pure Biofuels

For the sake of better clarification of the effect of selected biofuels on the combustion process , the pressure in the engine cylinder was indicated. The pressure in the cylinder – p_z was measured using a pressure transducer and a crankshaft angle encoder. As it is already said the p_z. signal was detected at 0.2 CA and an average value of 20 cycles was used to analyze and calculate the heat release rate (HRR) and calculate and analyze the combustion process parameters.

Figures 4, 5, 6 and 7 show the indicated gas pressure diagram in the engine cylinder for various fuels at 100%, 75%, 50% and 25% load, with the corresponding fuels. As noted, pure (100%) fatty acid methyl esters of rapeseed oil, soybean oil and waste edible palm oil were used as fuels. The standard euro diesel labeled D100 was used as the reference fuel.

From the attached diagrams the maximum pressure values in the cylinder are higher when working with biodiesel than the corresponding maximum pressure value achieved when working with diesel fuel. Also, as the load level increases, the maximum pressure in the cylinder rises, as expected. At all operating modes, the maximum pressure of the gas in the cylinder is always at RME100, with the maximum pressure at the highest load level being 101.6 Bar, which is 4.41% more than when operating with diesel fuel.

oiFigure 4. In-cylinder gas pressure of the engine when working with clean fuels at 100 % of engine load

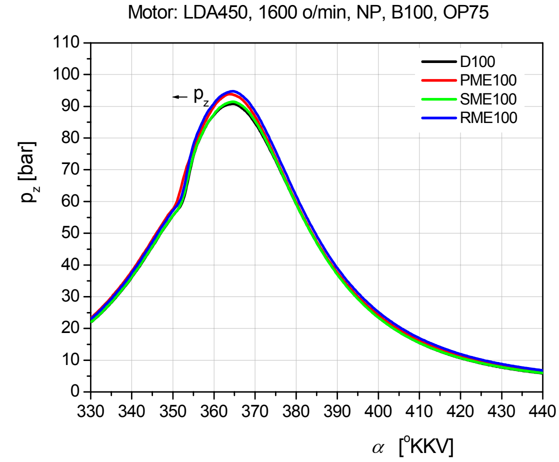


Figure 5. In-cylinder gas pressure of the engine when working with clean fuels at 75 % of engine load

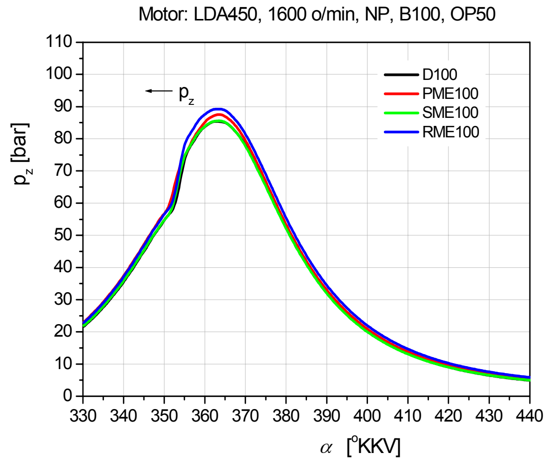


Figure 6. In-cylinder gas pressure of the engine when working with pure fuels at 50 % of engine load

During low engine load, the diversity of pressure values with propulsion on RME 100 had certain diversification and it is described in next sentence: at 75% load pressure value is 94.7 bar, on engine 50% load it was 89.2bar and at 25% engine load had value of 89.2bar. This is 4.42%, 5.06% and 4.35% selectively more effective in comparison with conventional diesel fuel, respectively.

By sustainable derivation with examinations on max pressure values it can be seen that on nominative engine loads the following pressure diversification of max pressure was conserved: the highest max is always for RME100, it is followed by PME100 and finally, by SME100. Under the exampled testing working conditions, for PME100 max value of pressure was higher for 1.44%,3.41% and 3.06% and 2.49% respectively from the highest engine load to the lowest engine load. Speaking about methyl ester form soya oil SME100 it can be noticed that max cylinder pressure peak with this fuel is around the maximum with diesel fuel but still with diversifications on the level of max pressure, except on the lowest load level of 25% where the difference was larger.

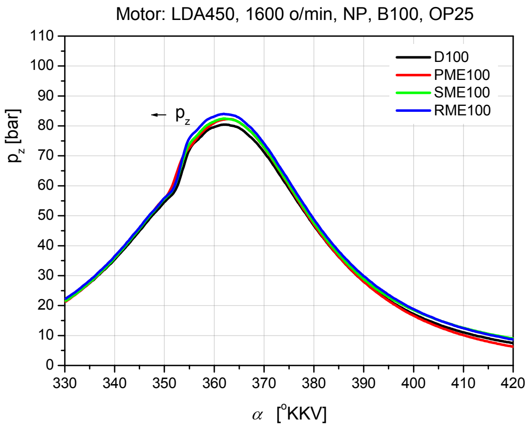


Figure 7. In-cylinder gas pressure of the engine when working with pure fuels at 25 % of engine load

The diagram of maximum cycle pressure values for all tested fuels and all operating modes is shown in Figure 8. At the highest load level, the maximum cycle pressure is about 100 bar for all the fuels used. This is a significantly higher-pressure value than the one commonly encountered with this type of engine and it is a consequence of the application of the supercharging procedure, as noted earlier. At lower load levels, the pressure values are lower and at the lowest load level of 25% these pressures are about 80 bar.

Regarding the position of the cylinder pressure maximum relative to the Top Dead Centre (TDC), it can be seen from the diagram that the maximum pressure is reached earlier when working with bio fuels than when working with diesel.

The reason for the earlier achievement of maximum cylinder pressure values when working with the methyl esters of these vegetable oils is twofold.

First, the latent combustion periods are shorter for methyl esters than for diesel, and secondly, the injection of these biofuels begins earlier than the injection of diesel, due to their increased density, kinematic viscosity, and higher modulus of elasticity of these fuels.

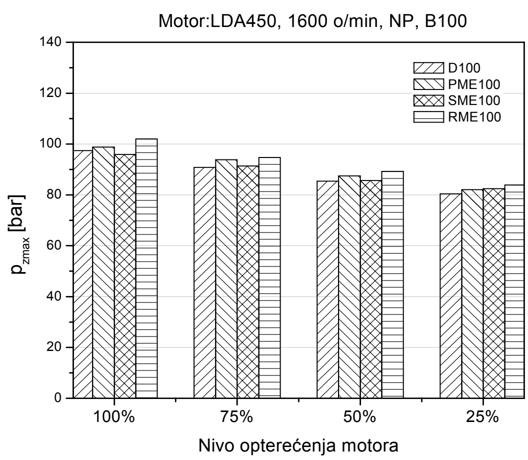


Figure 8. Histogram of maximum pressure value in engine cylinder when operating with pure fuels pure fuels

The change in pressure in the engine cylinder during combustion depends almost exclusively on the heat release rate rate Flow heat release rate during combustion, and the heat heat release rate, depends on many factors, the most influential being the physic chemical characteristics of the fuel and the flow of injection. In addition, the rate of heat release had been greatly influenced by the design of the combustion system, i.e. the geometry of the combustion chamber, as well as the characteristics of the current image in the engine cylinder just before the start of combustion and during the combustion itself.

3.3. Heat Release Rate - HRR

3.3.1 Differential law of heat release for pure biofuels

Figures 9. to 12. show the heat release laws for pure B100 biofuels and for all four operating modes. It can be observed that due to the evaporation of the fuel injected during the ignition delay period, there is initially a negative part of the function of the law of heat release. This negative part occurs due to the heat of evaporation of the fuel droplets from the jet during mixing with hot air, so that this process substantially lowers the pressure and temperature in the cylinder. On the other hand, the part of negative heat release is due to gas loss through crevices between piston and cylinder liner (sealing with piston rings). Therefore, by the size of this negative part of the diagram, the condition of the piston-cylinder assembly of the given engine can be quite accurately evaluated, given that this negative part of the diagram essentially gives information about the energy losses in the cylinder.

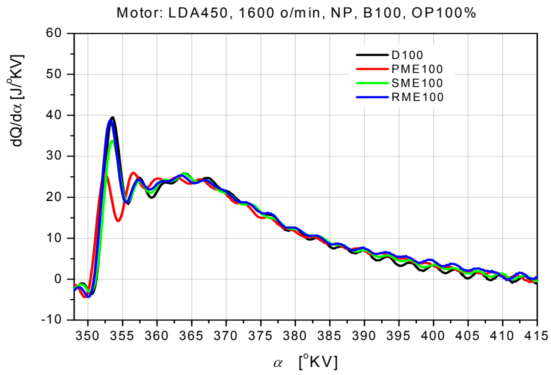


Figure 9. Heat Release Rate for biofuels and diesel fuel for 100 % load level

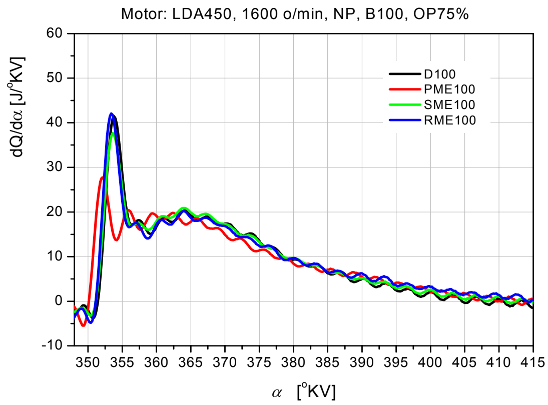


Figure 10. Heat Release Rate for biofuels and diesel fuel for 75 % load level

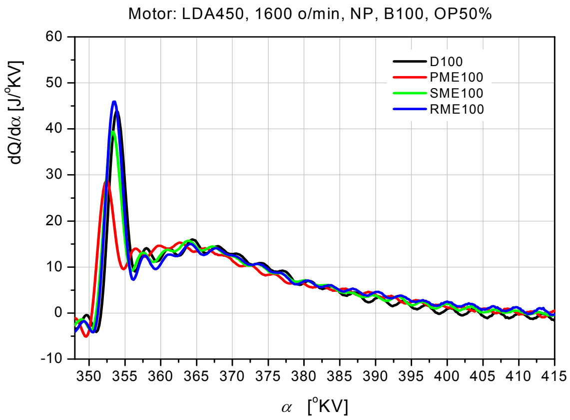


Figure 11. Heat Release Rate for biofuels and diesel fuel for 50 % load level

If there is no large gas leakage through the crevices between piston and cylinder liner. If which means that the piston-cylinder assembly is in good condition, this negative part of the diagram is small and applies only to the evaporation of the fuel from the jet. In this case, it can be noted that the piston-cylinder assembly of the engine used in the research was in good condition.

Upon initiation of the combustion process, which was previously defined as the intersection point of the dQ/dα diagram and the abscissa, a very turbulent combustion occurs with a large gradient of increase in the rate of heat release. The diagram shows that both diesel and biodiesel burn very quickly at this stage. After this phase i.e. the period of unregulated combustion (or premixed combustion phase), the process continues with a purely diffuse character. This second part of the combustion process is much more soothing and it is called the period of regulated combustion.

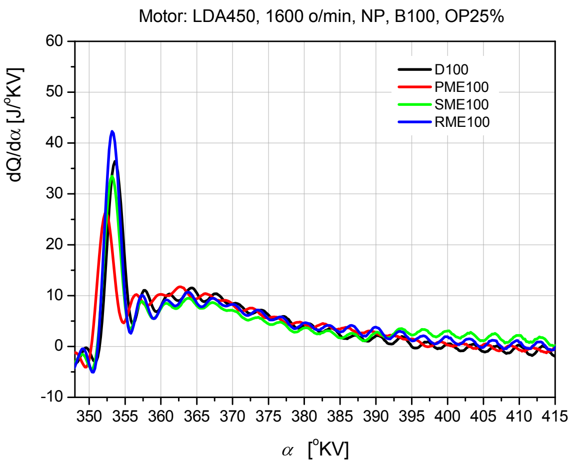


Figure 12. Heat Release Rate for biofuels and diesel fuel for 25 % load level

The speed of this part of the process is mainly defined by the rate of formation of the mixture and the main feature of this diffuse combustion is the gradual release of heat till the completion of the entire combustion process.

If you look at Figures 8 to 11, it can be observed that the combustion process always starts earlier when working with RME100, PME100 and SME100 biodiesel than with diesel. When it comes to the value of the maximum heat release stream, it is observed that for all the fuels used, these maximums are increasing as the load level decreases.

In addition, the maximum heat release rates for biofuels are moving beyond Top Dead Center (TDC), i.e. they are being reached earlier comparing to the rates for diesel fuel. When it comes to the order of the position of the maximum of this law, the maximum for PME100 is always reached at the earliest, followed by the maximum for SME100 and finally for RME100. Of course, the maximum heat release rate is last reached when operating with the D100 reference diesel. At the highest load level, all dQ/dα maxima are lower for biodiesel than for diesel. At a load level of 75%, the maximum heat release rates for RME100 and D100 are practically the same. If the load level goes down, then the maximum level with RME100 exceeds the one achieved when working with the D100. This difference is expressed to the most at the lowest load level of 25%. For the remaining two fuels, the maximum dQ/dα is always lower than the maximum with diesel.

The fact that is particularly interesting as well as evident from the diagram is that PME100 biodiesel burns extremely diffusely with a large proportion of the third and fourth combustion phases, that is, the diffuse combustion phases, here treated as unique and referred to as pure diffuse combustion.

3.3.2 Normalized cumulative heat release for pure biofuels

Figures 13 to 15 show the cumulative or integral heat release for all B100 fuels compared to diesel when operating at different engine loads. These diagrams reaffirm the earlier onset of combustion for biodiesel in comparison to diesel. It can be noted that the overall combustion characteristics for these biofuels are like those obtained with diesel fuel. Using the cumulative or integral heat release, it is possible to estimate the way the combustion fuel content changes during the combustion process, that is, the combustion for different fuels from the beginning to the end of the combustion cycle. The negative part at the beginning of the heat release stream shows that most of the evaporative energy is consumed at RME100, after that at PME100, only then at SME100, and at the least at D100.

Although combustion starts earlier for bio fuels than for diesel, the overall duration of combustion is longer for bio fuels (biodiesel), that is, biodiesel burns more slowly – in a more elongated way, looking at the whole combustion process from the beginning to the end. The difference in combustion dynamics between the diesel and the biofuels under investigation is greater at lower loads, whereas with the increase of the load this difference is gradually reducing to become minimal at full load.

In the beginning of combustion, biodiesel burns faster up to about 354 CA and the diesel burns slower; however, later diesel burns faster and this goes up to about 415 CA when combustion is completed.

The D100 burns fastest because it forms the mixture with the air fastest for it has the lowest density and viscosity. Furthermore, this fact is the reason why the average *Saunter* droplets diameter is the smallest when injecting diesel and the diesel is the fastest to find oxygen from the combustion air available. We also have to bear in mind that the final outcome of all the parameters related to higher kinematic viscosity of biofuels significantly affect the fuel spray, droplet size distribution, droplet evaporation rate and spray atomization process.

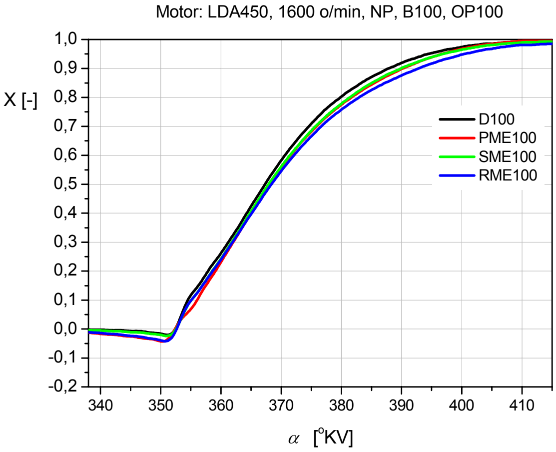


Figure 13. Normalized cumulative heat release at 100% load

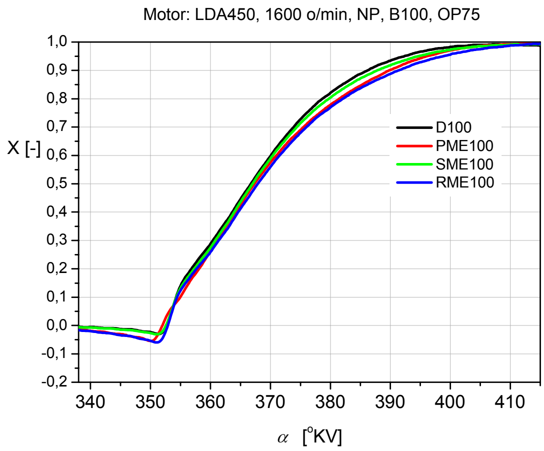


Figure 14. Normalized cumulative heat release at 75% load

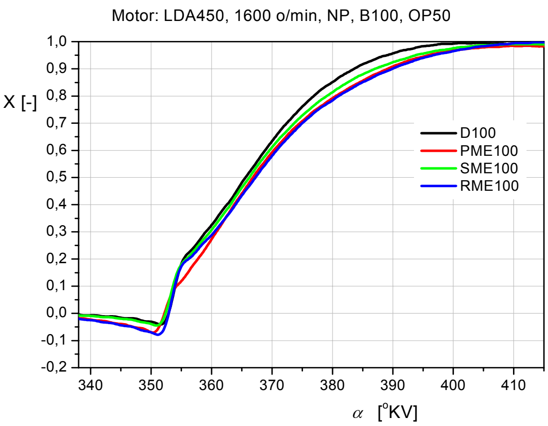


Figure 15. Normalized cumulative heat release at 50% load

It can also be concluded that the differences in heat release dynamics at a later stage of the process go according to the order of D100, SME100, PME100 and RME100, respectively. From this it can be concluded that it burns the RME100 fuel burns longest and PME-100 is close to it.

4. CONCLUSION

Studies of the combustion process conducted on the given diesel engine and with the above mentioned biofuels presented in this paper show the following:

- The diesel engine can run smoothly, powered by pure methyl esters of the appropriate vegetable oils.
- There are no problems with the engine operation or the formation of deposits on vital parts of the engine, which was confirmed by multiple dismantling of the engine during the experimental study. All the facts stated above are true provided that the fuel is manufactured in accordance with EN14214 standard.
- The dynamics of the combustion process expressed through the heat release rate is similar to the one the engine has when it is driven by diesel fuel.
- Differences observed in the relevant diagrams are common because of the physico-chemical differences of the fuels themselves.
- An important conclusion can be drawn from the research that biofuels produced from vegetable oils are excellent fuels for diesel engines.

The use of vegetable oils as fuel for diesel engine propulsion can be very interesting regarding the fact that they are renewable raw materials. It should be noted that alternative fuels are not fully CO_2 neutral but can have considerable potential in developing the country's economy. Should also be noted that this type of fuel can be very interesting for farmers who can produce fuel for their own mechanization in considerable quantities.

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REFERENCES

[1] Diesel, R. K.: *Internal Combustion Engine;* Patent No. 608,845. Patented Aug. 9, 1898, United States Patent Office

[2] Diesel, R.: The Diesel oil-engine. *Engineering*, **93**, 395-406 (1912). *Chem. Abstr*., 6, 1984 (1912).

[3] Durbin, T. et al., Effects of Biodiesel, Biodiesel Blends, and a Synthetic Diesel on Emissions from Light Heavy-Duty Diesel Vehicles, Environmental Science & Technology, vol. 34, no. 3, pp. 349-355, 2000.

[4] Ramadhas, A., Muraleedharan, C., Jayaraj, S., Performance and emission evaluation of a diesel engine fueled with methyl esters of rubber seed oil, Renewable Energy, vol. 30, no. 12, pp. 1789-1800, 2005.

[5] Tomić, M., Savin, L., Mićić, R., Simikić, M., Furman, T.: Effects of Fossil Diesel and Biodiesel Blends on the Performances and Emissions of Agricultural Tractor Engines, Thermal Science, vol. 17, No. 1, pp. 263-278, 2013

[6] Sivalakshmi S., Balusamy T., Experimental Investigation on a Diesel Engine Fuelled with Neem Oil and its Methyl Ester, Thermal Science, vol. 15, No. 4, pp. 1193-1204, 2011

[7] Yuksek, L., Kaleli, H., Ozener, O., Ozoguz, B.: The Effect and Comparison of Biodiesel-Diesel Fuel on Crankcase Oil Diesel Engine Performance and Emissions, FME Transactions 37, pp 91-97, 2009

[8] Ozsezen, A.N., Canakci, M., Turkcan, A., Sayin, C.: Performance and combustion characteristics of a DI diesel engine fueled with waste palm oil and canola oil methyl esters. Fuel, vol. 88, pp 629 - 636, 2009

[9] Buyukkaya, E.: Effects of biodiesel on a DI diesel engine performance, emission and combustion characteristics. Fuel, vol. 89, pp 3099–3105, 2010

[10] Ghobadian, B., Rahimi, H., Nikbakht, A.M., Najafi, G., Yusaf, T.F.: Diesel engine performance and exhaust emission analysis using waste cooking biodiesel fuel with an artificial neural network. Renew Energy, vol. 34, pp 976–982, 2009

[11] Qi, D.H., Geng, L.M., Chen, H., Bian, Y.Z.H., Liu, J., Ren, X.C.H.: Combustion and performance evaluation of a diesel engine fueled with biodiesel produced from soybean crude oil. Renew Energy, vol. 34, pp 2706–2713, 2009

[12] Serac, M.R. et al.: Evaluation of comparative combustion, performance, and emission of soybean-based alternative biodiesel fuel blends in a CI engine, Renewable Energy, Article in press, https://doi.org/10.1016/j.renene. 2019.10.090.

[13] Nabi, M.N., Rasul, M.G., Anwar M., Mullins, B.J., Energy, exergy, performance, emission and combustion characteristics of diesel engine using new series of non-edible biodiesels, Renewable Energy, vol. 140, pp. 647-657, 2019

[14] Gharehghani, A., Mirsalim M., Hosseini, R., Effects of waste fish oil biodiesel on diesel engine combustion characteristics and emission, Renewable Energy, vol. 101, pp. 930-936, 2017

[15] Kuberan, J., Alagumurthi, N.: Performance and Emission Characteristics of Algae Bio-fuelled Diesel engine. Int.J.Chem.Sci 14 (4), pp 2973-2980, 2016

[16] Dorado M.P., Ballesteros E., Arnal J.M., Gomez J., Lopez F.J., Exhaust emissions from a Diesel engine fueled with transesterified waste olive oil, Fuel, vol. 82, no. 11, pp. 1311-1315, Jul. 2003

[17] Utlu, Z., et al.: The Effect of Biodiesel Fuel Obtained from Waste Frying Oil on Direct Injection Diesel Engine Performance and Exhaust Emissions, Renewable Energy, vol.33, no. 8, pp. 1936-1941, 2008

[18] Knežević, D.M., The Research of Combustion Process of Diesel Engine Using Biofuels and Exhaust Gas Recirculation, Ph.D. thesis (in Serbian), University of Belgrade, Belgrade, RS, 2014

[19] Hohenberg, G., Killman, I., Basic Findings from Measurement of the Combustion Process, XIX Fisita Congress, Melbourne, Australia, 1982, paper 82126

[20] Tomić M.: A Quick Simplified Approach to the Evaluation of Combustion Rate from an Internal Combustion Engine Indicator Diagram, Thermal Science, Vol. 12 (2008), No.1, pp 85-102

[21] Hohenberg, G., Heat Transfer Calculation in CI Engine (in German), *Motortechnische Zeitschrift*, (1980), 7/8

[22] Jankov, R.: Mathematical Modeling of Current-Thermodynamic Processes and Driving Performance of Diesel Engines, Part One - Fundamentals, Scientific Book, Belgrade, 1984

[23] Tomić, M.: Identification of parameters of mathematical model of current-thermodynamic processes of diesel engines, Doctoral dissertation, Belgrade, 1987.

[24] Internal Reports, IMR Institute d.o.o., Belgrade, Rakovica

NOMENCLATURE

TDC	Top Dead Center
HRR	Heat Release Rate
D100	Euro Diesel
SMO100	Soybean Methyl Ester
PMO100	Palm Oil Waste Methyl Ester
RME 100	Rapeseed Oil Methyl Ester
EN14214	Standard for production bio diesel oil fuel
dQ	an elementary change in the total energy input to the engine cylinder,
dQ_f	an elemental amount of energy released in the process of combusting a cycle amount of fuel,
$dm_{ex}h_{ex}$	an elemental enthalpy of gas passing through the engine exhaust valve,
$dm_{in}h_{in}$	an elemental enthalpy of gas passing through the engine intake valve,
$dm_{eel}h_{eel}$	an elemental enthalpy of gas leaking through unsealed places
dQ_w	an elemental amount of energy (heat) delivered to the walls of the workspace by convection and radiation,
ΔQ_f	is the part of the heat released by combustion of fuel between points 1 and 2,
ΔQ	the portion of heat that is delivered to the working gas between points 1 and 2,
ΔQ_w	the portion of heat that is transferred to the walls of the workspace between points1and 2,
V_z	is the current workspace volume,
p_z	pressure and gas temperature in the engine cylinder,
$c_m = \frac{S\,n}{30}(m/s)$	mean piston speed;
$S(m)$	piston stroke; n (1/min) - rev. per minute.

КАРАКТЕРИСТИКЕ ПРОЦЕСА САГОРЕВАЊА НЕКИХ ВРСТА БИОГОРИВА У ДИЗЕЛ МОТОРУ

Д. Кнежевић, В. Петровић, Г. Воротовић, В. Пајковић, П. Браловић

Интересовање за алтернативна горива која би се користила за погон мотора сус је веома велико у читавом свету. Зато се истраживања везана за ову проблематику врше у многим водећим светским

лабораторијама за моторе. Разлог овог интересовања су доступност оваквих горива, повољне еколошке карактеристике и позитиван утицај на економије земаља које оваква горива могу да производе. У предметном раду се даје део резулта веома комплексних и дуготрајних истраживања динамике процеса сагоревања код дизел мотора погоњеног метилестрима биљних уља и то отпадног палминог јестивог уља, сојиног уља и уља уљане репице. Поређење промена у радном процесу мотора, пре свега процесу сагоревања, је вршено при синхронизованом раду мотора на уобичајеним стандардним дизел горивом минералног порекла. Анализом закона (тока) сагоревања сваког појединачног горива, утврђено је да постоје разлике које нису великог карактера, из чега се може закључити да су горива произведена из биљних уља, а по важећем стандарду, веома квалитетна горива за погон дизел мотора. Ово даје могућност нарочито пољопривредним газдинствима да самостално производе део потребног горива са сопствених земљишних парцела уз услов да поседују одговарајућу опрему за производњу биодизела.