# THE CHARACTERISTICS OF COMBUSTION PROCESS OF DIESEL ENGINE USING VEGETABLE OIL METHYL ESTERS

by

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*Biodiesel is one of the promising renewable, alternative, and environmentally friendly biofuels that can be used in Diesel engine with little or no modification in the engine. The present paper investigates the combustion characteristics of single cylinder, naturally aspirated, air cooled, Diesel engine fueled with pure (100%) methyl ester of rapeseed oil, comparing to the corresponding characteristics when it was driven by diesel fuel. Combustion process analysis for both fuels was done at constant engine speed and at two load levels corresponding to the mean effective pressure of three and six bar. It was also concluded that the test engine can operate without problems, both with that fuel and with a lot of other biofuels and their mixtures that were used during long laboratory research.*

Key words: *Diesel engine, combustion process, vegetable oil methyl esters, heat release rate*

## Introduction

A great number of scientific papers that were dealing with the problem of vegetable oil methyl esters application for diesel driven engines has shown that the vegetable oil esters are renewable source of alternative and ecological bio fuels. Such fuels, usually called biodiesel can be used in Diesel engines with minimum, or even without any engine modifications [1-6]. Strict law regulations on the engine exhaust emission, exhaustion of fossil fuels and the constant political tensions about the oil sources in the world, have forced a lot of countries in the world to look for alternatives to fossil fuels. A lot of research, dealing with the vegetable oil esters (biodiesel) in Diesel engines has also shown the potential of these fuels for $CO_2$ emissions reduction [7, 8].

The performances of the engine with various bio fuel types of biodiesel and mixtures of pure vegetable oils with diesel fuel has also been a lot investigated. A lot of researches are based on a comparison of engine output performances such as engine power, specific fuel consumption and effective thermal efficiency when working with biodiesel or vegetable oils and their blends with performance of engines powered by diesel fuel. A number of researchers has found that the use of biodiesel as fuel leads to increased fuel consumption but also to something greater thermal efficiency of the engine compared to those obtained during the operation with standard diesel fuel [9-12]. On the other hand the use of various biodiesel

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in the engine results in a change of performance and exhaust emissions as a consequence of their different physical and chemical characteristics [13]. The research of influence of biodiesel on the characteristics of the fuel injection system, as well as the characteristics of the fuel spray is also carried out [14, 15].

In order to improve the output characteristics of the engine and exhaust emissions, and to keep the characteristics of diesel fuel operation, it is necessary to study in detail the process of combustion and heat release characteristics. Heat release rate (HRR) directly influences the history of pressure and temperature in the cylinder and consequently the performance and emissions of the engine. This means that the research of HRR and combustion process is very important regarding the possibilities of their optimization. The aim of this paper is to present the research results of combustion process characteristics in the case of direct injection Diesel engine running with pure rape oil methyl ester (RME) including the analysis of the dynamics of the combustion process. Also, the comparative study of combustion characteristics with standard diesel fuel and biodiesel has been carried out.

## Experimental set-up

### *Test engine and experimental procedure*

The study of combustion process in Diesel engine fueled with standard diesel fuel and RME are presented was carried out in the framework of doctoral dissertation [16]. All experiments were carried out in the laboratory of IC engine department, Faculty of Mechanical Engineering in Belgrade. In addition to standard engine testing equipment (dynamometer with engine torque and speed measuring system, intake air and fuel flow measuring systems, inlet air, exhaust gases, engine coolant and oil temperatures measuring system), the engine was equipped with the system for in-cylinder pressure recording. This equipment includes pressure transducer Kistler type 7031 mounted directly to combustion chamber, charge amplifier Kistler type 5001, angular encoder installed on engine crankshaft type COM1 and data acquisition system ADS 2000 developed in the IC engines department.

Angular encoder provides angular increment of 1 deg. (360 marks per revolution) which is by dint of software divided into five parts, so that the angular resolution of pressure sampling rate is 0.2 °CA. Pressure signal was recorded for 100 consecutive cycles and mean cycle was evaluated for subsequent analysis. A special attention was drawn to exact determination of absolute pressure level and pressure trace to crank angle synchronization, since these problems could significantly influence the results of heat release analysis. For this purpose the methodology reported in [17] and [18] was applied. More details about testing equipment and procedure can be found in [16].

Figure 1 shows the laboratory installations used in experimental investigation and in tab. 1 the basic test engine characteristics are listed. The engine is of the aggregate type and is

**Table 1. Main specifications of test engine**

| Engine type | Diesel 4 stroke, direct injection, air cooled, single cylinder for aggregate application |
|---|---|
| Bore/stroke | 85/80 mm |
| Compression ratio | 17.5 |
| Max. power output | 5 kW/2500 rpm |
| Fueling system | High pressure pump, injector with 4 jets, (4 × 0.28 mm) |

used to drive irrigation pumps and has a classic fuel injection system. Research of the characteristics of the combustion process was carried out on four load levels that correspond to the values of mean effective pressure of 1.5 bar, 3 bar, 4.5 bar, and 6 bar, with the engine speed range of 1600 rpm. The paper presents the results of the research into the characteristics of the combustion process for two representative operating regimes and the load level corresponding to the mean effective pressure of 3 bar and 6 bar.

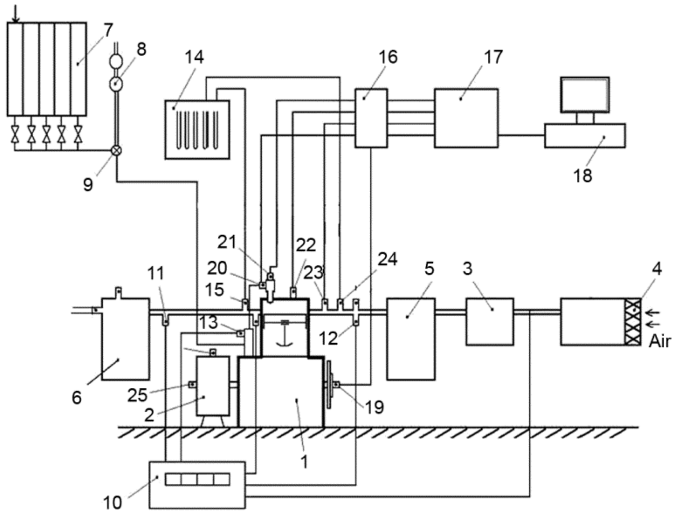


**Figure 1. Experimental installation**
*1 – Diesel engine, 2 – dynamometer 3 – air flow meter, 4 – air filter, 5 – compensating volume in intake, 6 – compensating volume in the exhaust, 7 – fuel tank, 8 – fuel consumption measuring, 9 – valve, 10 – display, 11 – exhaust gas temperature sensor, 12 – intake air temperature sensor, 13 – fuel temperature sensor, 14 – U-pipe system, 15 – in connection pipe, 16 – charge amplifier, 17– acquisition system, 18– computer, 19 – angular encoder, 20 – injection pressure sensor, 21 – injector needle lift sensor, 22– cylinder pressure sensor, 23 – intake pipe pressure sensor, 24– in connection pipe inlet, 25 – RPM sensor*

### *Fuel properties*

The fuel that is used in this research is a methyl ester of rapeseed oil obtained through the transesterification of vegetable oils and meets standard EN 14214: 2009, and the determination of the applied fuel was carried out in the laboratory for fuel, Faculty of Mechanical Engineering in Belgrade. Table 2 shows the basic characteristics of the fuel used in the research.

**Table 2. Test fuel properties**

| | DIESEL 100 | RME 100 |
|---|---|---|
| Density [kg $l^{-1}$] | 0.828 | 0.880 |
| Lower caloric value [kJ $kg^{-1}$] | 41494 | 37631 |
| Kinematic viscosity [$mm^2 s^{-1}$] | 3.16 | 4.59 |
| Stoichiometric air mass [kg $kg^{-1}$] | 15.08 | 12.65 |
| $O_2/C/H_2$ content [kg $kg^{-1}$] | 0.0/0.8496/0.1504 | 0.120/0.772/0.12 |

## Estimation of HRR from recorded cylinder pressure

The HRR is an important parameter to analyze the combustion phenomena in the engine cylinder. The important combustion phenomena parameters such as combustion intensity and duration can be easily estimated from the heat release rate diagram. The HRR diagram also provides key input parameters in the modelling of the $NO_x$ emission. The heat release rate is modelled by applying the first law of thermodynamics.

The HRR is modelled by applying the first law of thermodynamics. In reality, engine cylinder is an open thermodynamic system, even during high pressure part of the cycle (when intake and exhaust valves are closed) due to mass flow and loss through crevices between piston rings and cylinder liner. This is schematically illustrated in fig. 2.

The basic equation of the first law of thermodynamic applied to an open system is:

$$\mathrm{d}Q = \mathrm{d}U + p\mathrm{d}V \tag{1}$$

where $\mathrm{d}Q$ is the change of elementary energy entering/leaving the system (except kinetic energy which is omitted), $\mathrm{d}U$ – the elementary change of cylinder charge internal energy, and $p\mathrm{d}V$ – elementary mechanical work delivered to the piston. The derivative $\mathrm{d}Q$ consists of the energy released by fuel combustion $\mathrm{d}Q_\mathrm{f}$, the energy transferred to the walls $\mathrm{d}Q_\mathrm{w}$ and energies taken into/out the system by means of the mass flows through the boundaries $\Sigma h_i \mathrm{d}m_i$, where $h_i$ and $\mathrm{d}m_i$ are the enthalpies and elementary mass flows, respectively. During high pressure part of the cycle the gas exchange valves are closed and mass flow through crevices, provided that the proper cylinder sealing is encountered, is very small and can be neglected. Thus, it can be written:

$$\mathrm{d}Q = \mathrm{d}Q_\mathrm{f} + \mathrm{d}Q_\mathrm{w} \tag{2}$$

After simple evaluation:

$$\mathrm{d}Q_\mathrm{f} = m\mathrm{d}u + p\mathrm{d}V + \mathrm{d}Q_\mathrm{w} \tag{3}$$

where $m$ is the mass and $\mathrm{d}u$ – the elementary change of specific internal energy of cylinder content.

Pressure diagram recorded by dint of data acquisition system is the series of pressure values taken in discrete crank shaft positions. The series of pressure-crank angle data can be easily transformed into pressure-volume data. Since modern crank angle encoders enable very fine angular increment ($\Delta\alpha$), the changes in pressure and volume are small. Therefore, if we consider the changes between two recorded points as elementary, the error will be within reasonable limits.

During the gas state change between two consecutive points 1 to 2 (fig. 3), the amount of fuel chemical energy $\Delta Q_\mathrm{f}$ is released by fuel combustion. If we neglect the effect of crevice flow, this heat is partly transferred to the gas $\Delta Q_\mathrm{g}$ and partly transferred to the walls $\Delta Q_\mathrm{w}$. Then, we can write:

$$\Delta Q_\mathrm{f} = \Delta Q_\mathrm{g} + \mathrm{d}Q_\mathrm{w} \tag{4}$$

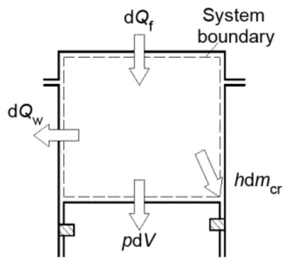


**Figure 2. Engine cylinder as an open thermodynamic system**

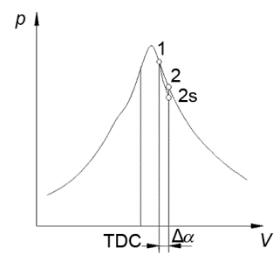


**Figure 3. Recorded *p-V* diagram; gas state change from point 1 to point 2**

The evaluation of $\Delta Q_\mathrm{g}$ and $\Delta Q_\mathrm{w}$ can be performed in a simplified way that is convenient for engine laboratory testing, and still sufficiently accurate. The real process (from point 1 to point 2) can be virtually divided into two steps (fig. 3). First step is isentropic expansion from point 1 to 2s, with no heat release. In the second step from point 2s to 2 the heat added to gas is being considered at constant volume ($V_2$ = const.). The required parameters for evaluation the isochoric heat $\Delta Q_\mathrm{g}$ can be obtained using the equation of state and the equation for isentropic state change. In this way heat release due to combustion that is transferred to the gas between points 1 and 2 can be calculated by the equation:

$$\Delta Q_\mathrm{g} = \frac{c_\mathrm{v}}{R} V_2 \left[ p_2 - p_1 \left( \frac{V_1}{V_2} \right)^\kappa \right] \tag{5}$$

For that, the thermodynamic properties for the gases in engine cylinder are required. The approximate expressions for gas constant $R$ and gas specific heat at constant volume $c_v$ and discussion about their accuracy can be found in [19]. For specific heat at constant volume the appropriate expression for lean mixture operation, used for Diesel engine heat release calculation is:

$$c_v = 692 + 0.15\,T + \frac{1}{\lambda}\left(17.5 + 0.1094\,T\right) \qquad [\mathrm{Jkg^{-1}K^{-1}}] \tag{6}$$

where $T$ is the temperature and $\lambda$ – the air excess ratio.

Gas constant $R$ can be used as for ideal gas with a very small error [20]. In that case a gas constant depends only on gas composition. For Diesel engines, before the start of combustion, pure air has been compressed and gas constant has the value 287 J/kgK. Gas constant of combustion products depends on fuel composition and mixture strength. In the case of usual diesel fuels considered the differences due to fuel composition are very small and can be neglected. With sufficient accuracy, gas constant of combustion products $R_{cp}$ can be calculated using the expression [20, 21]:

$$R_{cp} = 290.65 - 0.5\,\lambda \quad \mathrm{J/(kgK)} \tag{7}$$

The convective heat transfer rate to the combustion chamber walls can be calculated from the general relation:

$$\frac{\mathrm{d}Q_w}{\mathrm{d}t} = \alpha_w \sum_i A_{wi}(T - T_{wi}) \tag{8}$$

Heat transfer to the walls of the combustion chamber is usually considered separately for characteristic parts of combustion chamber surface area, since they have significantly different temperatures. Heat transfer coefficient is mainly taken as average for the whole combustion chamber due to lack of accurate data for different parts. Thus, in eq. (8) $\alpha_w$ is the heat transfer coefficient (averaged over the chamber surface area), $A_{wi}$ – the parts of combustion chamber surface area, $T_{wi}$ – the mean wall temperatures of appropriate combustion chamber surface area parts, and $T$ – the mean gas temperature. Usually, piston crown, cylinder head and cylinder liner are considered as the elements in eq. (8), the last having variable surface according to piston motion and variable temperature from the top to the bottom.

The amount of heat transferred to the walls between two consecutive points 1 and 2 can be calculated:

$$\Delta Q_w = \left[\alpha_w \sum_i A_{wi}(T - T_{wi})\right]\Delta t = \left[\alpha_w \sum_i A_{wi}(T - T_{wi})\right]\frac{\Delta\alpha}{6n} \tag{9}$$

where $\Delta\alpha$ is the angular increment and $n$ the engine speed in rpm.

For the heat transfer coefficient several models are widely used. One of the recent is Hohenberg's expression [22] which is relatively simple and convenient for use:

$$\alpha_w = 0.013\,V^{-0.06} p^{0.8} T^{-0.4} (c_m + 1.4)^{0.8} \qquad [\mathrm{Wm^{-2}K^{-1}}] \tag{10}$$

where $V$ is the instantaneous cylinder volume, $p$ and $T$ are the cylinder pressure and temperature, and $c_m$ is the mean piston speed. Alternatively, some other models, for example Woschni's or Annand's can be used.

## Research results and discussion

### *Engine performance*

Figure 4 shows the output power, torque and specific effective fuel consumption during engine operation with both test fuels, with pure rapeseed methyl ester of the rapeseed – RME and with the mixtures of these fuels with the diesel fuel in the ratio of 50% and 7%, by volume.

When the mixture of fuel RME and diesel fuel are used, power and torque are lower than when we operate with diesel fuel, mainly due to lower heat value of RME and its mixtures in relation to diesel fuel. The largest reduction of engine power is when we work with pure rapeseed methyl ester and it is about 2.5% at rated speed. On the engine speed of 1600 rpm at which the characteristics of the combustion process were studied, that power reduction is about 8% for RME fuel comparing to diesel fuel. Somewhat bigger decrease in power at lower engine speeds, is probably the result of poorer formation of the mixture when working with RME and their mixtures, particularly due to lower volatility of the heavier fractions at lower engine speeds, more compact jet and larger diameter of the droplets in the stream. This leads to poor utilization of the of the fuel energy potential.

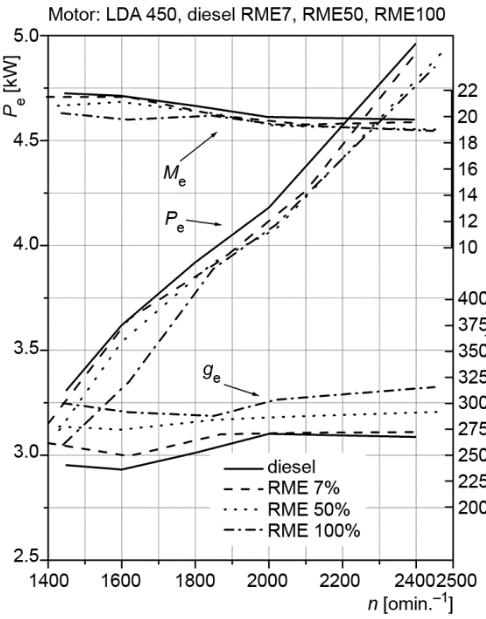


**Figure 4. Variation of engine power, torque and specific fuel consumption with respect to engine speed**

Specific fuel consumption is higher for RME comparing to diesel fuel, and it is increasing with the increase of RME share in the mixture. The difference in consumption between rapeseed methyl ester and its mixtures, and diesel fuel, is mainly due to its higher viscosity, lower heat value and higher density comparing to diesel fuel. The difference in consumption when using pure RME varies from 14% to 16% comparing to the diesel consumption, which is especially prominent at lower engine speeds.

### *Analysis of the combustion characteristics*

Figures 5 to 10 show the features of the combustion process of pure rapeseed methyl ester MER and their comparison to the same characteristics of the referent of diesel fuel combustion.

Figures 5 and 8 show the results of pressure diagram analysis using described method for one of recorded engine operating conditions using diesel fuel: the rate of heat release $dQ_f/d\alpha$, cumulative heat released $Q_f$, and the mean gas temperature $T$. The mean gas temperature is calculated using the equation of state for ideal gases. Figure 6 and 9 shows the same data in the case of engine operation with pure RME. In both cases engine speed and load were the same, 1600 rpm and mean effective pressure $p_e = 6$ bar and $p_e = 3$ bar, respectively.

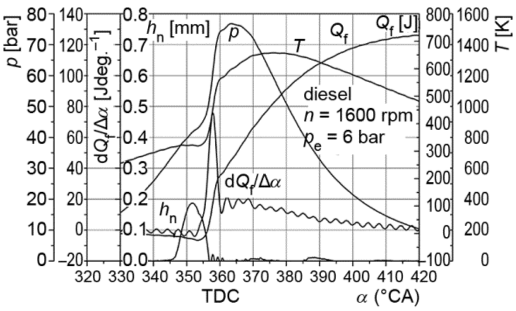

**Figure 5. Recorded pressure, *p*, injector needle lift, $h_n$, rate of heat release, d*Q*/d$\alpha$, cumulative heat released, *Q*, and mean gas temperature, *T*; fuel standard diesel**

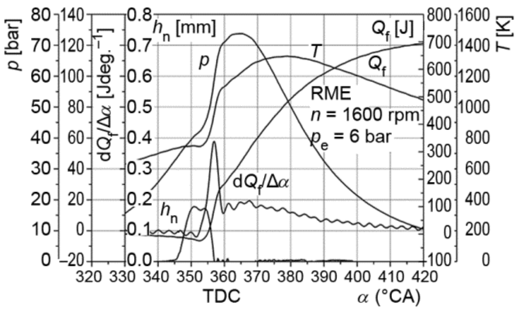

**Figure 6. Recorded pressure, *p*, injector needle lift, $h_n$, rate of heat release, d*Q*/d$\alpha$, cumulative heat released, *Q*, and mean gas temperature, *T*; fuel pure RME**

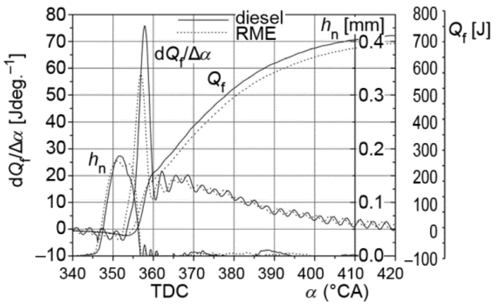

**Figure 7. The comparison of combustion dynamics of standard diesel (full line) and pure RME (dashed lines); $h_n$ – the needle lift, d*Q*/d$\alpha$ – the rate of heat release, *Q* – the cumulative heat released, $p_e$ = 6 bar, *n* = 1600 rpm**

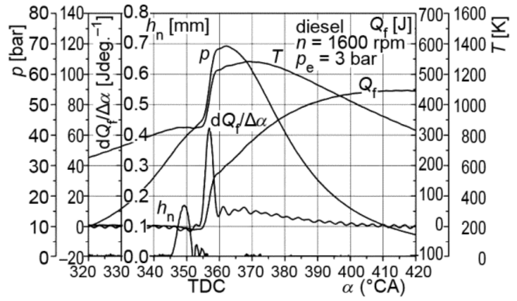

**Figure 8. Recorded pressure, *p*, injector needle lift, $h_n$, rate of heat release, d*Q*/d$\alpha$, cumulative heat released, *Q*, and mean gas temperature, *T*; fuel standard diesel**

The heat release laws have a similar shape with the both fuels. Also, it can be noticed that for both fuels the HRR has the form that is typical for Diesel engines with direct injection. High peak in the phase of premixed combustion, which reaches app. 75 J per degree at higher load and app. 60 J per degree at lower load, is pursued by relatively low intensity and lengthy combustion in diffusion phase. The duration of premixed phase is app. 5-6 °CA while total combustion duration is about 60-70 °CA.

It is interesting to compare combustion dynamics of standard diesel fuel and pure RME. This is shown in figs. 7 and 10. At higher engine load dynamic injection timing of RME is app. 0.5 °CA earlier (injector needle lift starts 0.5 °CA earlier), while the start of combustion is app. 1.5 °CA earlier, compared with diesel fuel. Ignition delay periods for RME and diesel fuel are app. 6.5 °CA and 7.5 °CA, respectively, *i. e.* RME has app. 1 °CA shorter ignition delay period.

At lower engine load the difference is even higher. The curves for needle lift $h_n$ are nearly coincident for both fuels, *i. e.* dynamic injection timing is almost the same, while combustion of RME commences app. 1.5 °CA earlier, *i. e.* ignition delay period is 1.5 °CA shorter. Ignition delay periods for RME and diesel fuel are app. 7.5 °CA and 9 °CA, respectively. Taking into account that engine speed is 1600 rpm, the duration of ignition delay is 0.78 ms for RME, and app. 0.93 ms for diesel fuel.

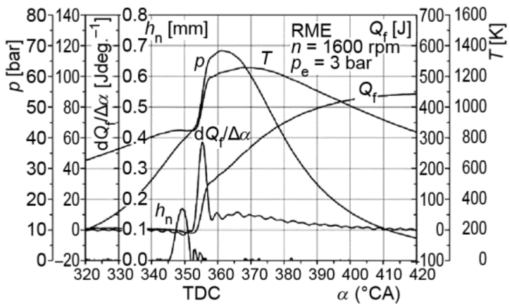

**Figure 9. Recorded pressure, *p*, injector needle lift, $h_n$, rate of heat release, d*Q*/d$\alpha$, cumulative heat released, *Q*, and mean gas temperature, *T*; fuel pure RME**

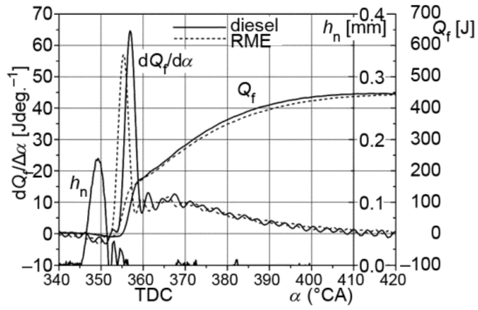

**Figure 10. The comparison of combustion dynamics of standard diesel (full line) and pure RME (dashed lines); $h_n$ – the needle lift, d*Q*/d$\alpha$ – – the rate of heat release, *Q* – the cumulative heat released, $p_e$ = 3 bar, *n* = 1600 rpm**

As can be seen, ignition delay period for both fuels is shorter at higher engine load. This is because of the fact that the temperature of elements that form the engine working area increases with the increase of engine load. This leads to higher temperatures of the inlet charge, and therefore higher temperatures at the end of compression *i. e.* at the time of injection. Also, at both considered engine loads RME has shorter ignition delay period, although the difference decreases with engine load increasing.

Shorter ignition delay means higher cetane number of RME with a positive effect on premixed combustion yielding lower peak of the HRR in premixed phase of combustion thereafter. However, after premixed phase, in diffusion phase of combustion RME burns slower and combustion is finished a few degrees later than in the case of diesel fuel. This is despite of the fact that RME contains a considerable amount of oxygen, which in principle should accelerate combustion. The reason is probably slower process of mixture formation due to poorer atomisation of the fuel spray caused by higher density and viscosity of RME.

## Conclusions

In this paper, the results of the experimental research of the combustion process with direct-injected Diesel engines were presented. The comparison of the pressure trace and the mean temperature of the process was made, as well as the HRR during the operation with pure methyl ester of rapeseed oil RME and diesel fuel. Maximum pressure values and the mean temperature of the process are slightly lower while operating with RME in comparison to the operation with diesel fuel. Combustion is finished a few degrees later than in the case of diesel fuel.

The comparison of combustion for standard mineral diesel fuel and RME provides very useful information of combustion dynamics for both fuels. It clearly shows that RME has shorter ignition delay period with a positive effect on premixed combustion yielding lower peak of the rate of heat release. However, after premixed phase, in diffusion phase of combustion RME burns slower and combustion is finished a few degrees later than in the case of diesel fuel.

## References

[1] Hammond, G., *et al.*, Development of Biofuels for the UK Automotive Market, *Applied Energy*, *85* (2008), 6, pp. 506-515

[2] Lapuerta, M., *et al.*, Diesel Emissions from Biofuels Derived from Spanish Potential Vegetable Oils, *Fuel*, *84* (2005), 6, pp. 773-780
[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*, *34* (2000), 3, pp. 349-355
[4] Ramadhas, A., *et al.*, Performance and Emission Evaluation of a Diesel Engine Fueled with Methyl Esters of Rubber Seed Oil, *Renewable Energy*, 30 (2005), 12, pp. 1789-1800
[5] Tomić, M., *et al.,* Effects of Fossil Diesel and Biodiesel Blends on the Performances and Emissions of Agricurtular Tractor Engines, *Thermal Science*, *17* (2013), 1, pp. 263-278
[6] Sivalakshmi, S., Balusamy, T., Experimental Investigation on a Diesel Engine Fuelled with Neem Oil and its Methyl Ester, *Thermal Science, 15* (2011), 4, pp. 1193-1204
[7] Dorado, M. P., *et al.*, Exhaust Emissions from a Diesel Engine Fueled with Transesterified Waste Olive Oil, *Fuel*, *82* (2003), 11, pp. 1311-1315
[8] Utlu, Z., Kozak, M. S., The Effect of Biodiesel Fuel Obtained from Waste Frying Oil on Direct Injection Diesel Engine Performance and Exhaust Emissions, *Renewable Energy*, *33* (2008), 8, pp. 1936-1941
[9] Lapuerta, M., *et al.*, Diesel Emissions from Biofuels Derived from Spanish Potential Vegetable Oils, *Fuel*, *84* (2005), 6, pp. 773-780
[10] Lapuerta, M., *et al.*, Effect of Biodiesel Fuels on Diesel Engine Emissions, *Progress in Energy and Combustion Science*, *34* (2008), 2, pp. 198-223
[11] Altparmak, D., *et al.*, Alternative Fuel Properties of Tall Oil Fatty Acid Methyl Ester-Diesel Fuel Blends, *Bioresource Technology*, *98* (2007), 2, pp. 241-246
[12] Monyem, A., Van Gerpen, J. H., The Effect of Biodiesel Oxidation on Engine Performance and Emissions, *Biomass and Bioenergy*, *20* (2001), 4, pp. 317-325
[13] Canakci, M., Combustion Characteristics of a Turbocharged DI Compression Ignition Engine Fueled with Petroleum Diesel Fuels and Biodiesel, *Bioresource Technology*, *98* (2007), 6, pp. 1167-1175
[14] Gao, Y. *et al.*, Experimental Study of the Spray Characteristics of Biodiesel Based on Inedible Oil, *Biotechnology Advances*, *27* (2008), 5, pp. 616-624
[15] Tesfa, B., *et al*., Prediction Models for Density and Viscosity of Biodiesel and Their Effects on Fuel Supply System in CI Engines, *Renewable Energy*, *35* (2010), 12, pp. 2752-2760
[16] Knežević, D., The Research of Combustion Process of Diesel Engine using Biofuels and Exhaust Gas Recirculation (in Serbian), Ph. D. thesis, Faculty of Mechanical Engineering, University of Belgrade, Belgrade, 2015
[17] Hohenberg, G., Killman, I., Basic Findings from Measurement of the Combustion Process, *Proceedings,* XIX Fisita Congress, paper 82126, Melbourne, Australia, 1982
[18] Tomić, M., The Identification Technique for Determination of the Indicator Diagram Pressure Level and TDC Position Errors, *Proceedings,* Motorsympo '88, VI International Symposium, 1988, Strbske Pleso, Brnik, Czechoslovakia
[19] Tomić, M. V., *et al.*, A Quick, Simplified Approach to the Evaluation of Combustion Rate from an IC Engine Indicator Diagram, *Thermal Science*, *12* (2008), 1, pp. 85-102
[20] Jankov, R., *Mathematical Modelling of Flow and Thermodynamic Processes and Operating Characteristics of CI Engines, I part, Fundamentals* (in Serbian), Naucna knjiga, Beograd, 1984
[21] Petrović, S., *et al*., *Combustion Modelling in SI engines* (in Serbian), Faculty of Mechanical Engineering, University of Belgrade, Belgrade, 1995
[22] Hohenberg, G., Calculation of the Gas Heat Transfer in Diesel Engines (in German), *Motortechnische Zeitschrift – MTZ, 41* (1980), 7/8, pp. 321-326

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