

## **Combustion characteristics of compression ignition engine operating on rapeseed oil-diesel fuel blends**

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**Abstract.** The effect of biofuels on the operational parameters of the combustion engines, such as performance parameters or emission production, are monitored often. These changes are, however, based on the effect of biofuels on the course of combustion pressure inside the combustion chamber. The contribution deals with the effect of rapeseed oil-diesel fuel blends on the combustion characteristics of turbocharged compression ignition engine. The course of cylinder pressure was monitored and analysed and heat release rate was calculated. The brake specific fuel consumption, indicated and brake thermal efficiency were calculated and evaluated, in-cylinder temperature and ignition delay were also evaluated. As a test fuels a 5% and 20% concentrations of rapeseed oil in diesel fuel were selected while 100% diesel fuel was used as a reference. Turbocharged CI engine Zetor 1204 located in the tractor Zetor Forterra 8642 was used for measurement. During measurement the rotation speed of the engine was kept constant at approx. 1,950 min<sup>-1</sup> and the load of the engine was selected at approx. 20, 60, 80 and 100%. The results showed decreased cylinder peak pressure, decreased intensity of heat release rate and earlier end of combustion in all tested loads for both tested fuel blends in comparison with diesel fuel, while the lowest peak cylinder pressure was reached using fuel with 5% rapeseed oil. Fuel with 5% rapeseed oil also showed highest indicated efficiency. Ignition delay was found shorter with both of the blended fuels in comparison with diesel fuel.

**Key words:** biofuels, cylinder pressure, heat release rate, engine efficiency.

### **INTRODUCTION**

The energy consumption, greenhouse gas emissions and fossil fuel consumption by agriculture sector is on the increase in present time (Garnier et al., 2019; Zhang et al., 2019). In agriculture sector the compression ignition engine is the most common source of the energy for machinery in the field conditions. For a diesel engine the liquid biofuels, based on vegetable oils are one of the most widely utilized alternatives to the fossil diesel fuel (Jindra et al., 2016; Babu et al., 2017; How et al., 2018; Mat et al., 2018a).

The vegetable oils have different properties in comparison with diesel fuel. The different origin of vegetable oils, edible or non-edible, means also different physical and

chemical properties. However, some properties are similar for vegetable oils in general in comparison with diesel fuel e.g. higher viscosity, density, surface tension, flash point and oxygen content and lower cetane number, calorific value and carbon content (Franco & Nguyen, 2011; Esteban et al., 2012; Sirviö et al., 2018). Preheat or modification of the fuel, such as hydrotreatment or transesterification, is necessary in order to utilize the neat vegetable oils as a fuel in diesel engines (Birzietis et al., 2017; Gad et al., 2018; Hsiao et al., 2018; Kim et al., 2018; Zahan & Kano, 2018). Alternatively, it is possible to blend the vegetable oils with diesel fuel or alcohols, such as butanol or methanol (Masjuki et al., 2001; Elango & Senthilkumar, 2011; Yilmaz & Morton, 2011; Pexa et al. 2014; Patel et al., 2016; Pexa et al. 2016; Gad et al., 2018; Mat et al., 2018a). The most of authors used vegetable oil-diesel fuel blends in concentration up to 20% of vegetable oil in the diesel fuel. According to Dabi & Saha (2019) blending vegetable oils in concentration up to 20% of vegetable oil makes a comparable performance with diesel fuel.

From the viewpoint of combustion characteristics vegetable oils and its blends in diesel fuel were found to decrease the peak cylinder pressure and maximum of heat release rate (HRR) (Nwafor et al. 2003; Devan & Mahalakshmi, 2008; Pradhan et al. 2014; Patel et al., 2016; Mat et al., 2018b; Sanli, 2018). Ignition delay (ID) was found shorter when using vegetable oil and its blends in comparison with diesel fuel (Devan & Mahalakshmi, 2008; Pradhan et al., 2014; Koder et al., 2018). However, some authors (Shah & Ganesh, 2016; Shah et al., 2018) state increased ignition delay and increased peak cylinder pressure when using neat karanj oil and sunflower oil in comparison with diesel fuel.

Vegetable oils and its blends in diesel fuel were found to decrease the brake thermal efficiency (BTE) of the engine while increasing the brake specific fuel consumption (BSFC) with increasing proportion of vegetable oil in the fuel blend (De Almeida et al., 2002; Devan & Mahalakshmi, 2008; Gad et al., 2018; Sanli, 2018; Dabi & Saha, 2019). However, other authors (Rakopoulos et al., 2006; Agarwal & Rajamanoharan, 2008; Bajpai et al., 2008; Rakopoulos et al., 2011) reported increased brake thermal efficiency in various of blending ratios from 10% to 75% of vegetable oil in the fuel blend. Rakopoulos et al. (2011) also found lower BSFC with 10% and 20% blends of sunflower and with 10% blend of corn oil in comparison with diesel fuel.

Higher exhaust gas temperature was found by many authors when using vegetable oil-diesel fuel blends or neat vegetable oils from different origin in comparison with diesel fuel (Pramanik, 2003; Hebbal et al., 2006; Agarwal & Kumar, 2007; Agarwal & Rajamanoharan, 2008; Devan & Mahalakshmi, 2008; Gad et al. 2018).

The aim of the paper is to experimentally verify the influence of rapeseed oil-diesel fuel blends, in concentrations of 5% and 20% of rapeseed oil, on the combustion characteristics, specific fuel consumption, thermal and indicated efficiency of the turbocharged compression ignition engine in comparison with diesel fuel.

## **MATERIALS AND METHODS**

For the measurement the turbocharged compression ignition engine Zetor 1204, mounted in the Tractor Zetor Forterra 8641 (Fig. 1), was used. The basic specification of the engine used for measurement are listed in the Table 1. The engine did not exceed 150 hours of operation time and it is unmodified. Start of injection (SOI) is given by

manufacturer and the injection pressure was checked before the measurement using manual testing device.



**Figure 1.** Tractor Zetor 8641, used for measurement (left), mobile dynamometer MAHA ZW 500 (right).

**Table 1.** Basic engine specification (\*according to Deutsche Landwirtschafts-Gesellschaft)

| Parameter                        | Specification                     |
|----------------------------------|-----------------------------------|
| Manufacturer and type            | Zetor 1204                        |
| No. and arrangement of cylinders | 4, in-line                        |
| Air fill                         | Turbocharged                      |
| Rated speed                      | 2,200min <sup>-1</sup>            |
| Rated power                      | 60 kW (53.4 kW on PTO*)           |
| Maximum torque                   | 351 Nm (312 Nm on PTO*)           |
| Engine displacement volume       | 4.156 l                           |
| Cylinder bore X stroke           | 105 X 120 mm                      |
| Compression ratio                | 17                                |
| Fuel supply                      | Mechanical in-line injection pump |
| Injection type                   | Direct injection                  |
| Start of injection (SOI)         | 12° before top dead center        |
| Injection pressure               | 22 MPa                            |
| Valve mechanism                  | OHV                               |
| Valves per cylinder              | 2                                 |

The engine was loaded via tractor PTO (Power Take Off) using mobile dynamometer MAHA ZW 500 (Fig. 1). Specification of the dynamometer can be seen in Table 2. A data acquisition unit, provided by manufacturer, was used to store the data from the dynamometer to the hard drive of the PC with the frequency of 10 Hz.

**Table 2.** Basic dynamometer specification

| Parameter             | Specification                  |
|-----------------------|--------------------------------|
| Manufacturer and type | MAHA ZW 500                    |
| Max. power            | 500 kW                         |
| Max. torque           | 6,600 Nm                       |
| Max. speed            | 2,500 min <sup>-1</sup>        |
| Torque inaccuracy     | < 1% over the full speed range |

The cylinder pressure was measured by means of pressure sensor Optrand C322-GPA (measuring range = 0–20.7 MPa, accuracy = 1%), mounted instead of the glow plug. The cylinder pressure was measured with a resolution of 1°CA (crankshaft angle), since the incremental sensor SICK DKS with 360 pulses per revolution was used as a trigger for cylinder pressure record. Therefore the frequency of cylinder pressure measurement is dependent on the engine speed.

The fuel consumption was measured by means of standard precision scale VIBRA AJ 6200 (range = 0–6200 g, accuracy = 0.1 g, readability = 0.01 g), where an external fuel tank was placed. Fuel consumption was measured with the frequency of 1 Hz.

The mass air flow through the engine was measured by means of mass air flow sensor Sierra FastFlo 620S (accuracy  $\pm 1\%$  of full scale, repeatability  $\pm 0.2\%$  of full scale). The data from the sensor were recorded to the hard drive of PC using A/D converter LabJack U6 with frequency of 10 Hz.

During the measurement the fuel temperature, exhaust gas temperature and ambient conditions, e.g. temperature, humidity and atmospheric pressure were monitored. Exhaust gas and fuel temperature were monitored by means thermocouples type K, the fuel temperature was monitored at the input of the injection pump and the exhaust gas temperature was measured in the muffler.

As a test fuels a mixtures of rapeseed oil and fossil diesel fuel were used. The rapeseed oil-diesel fuel blends were used in concentrations of 5% (R5D95) and 20% (R20D80) of rapeseed oil. As a reference the diesel fuel with no bio-components was used (D100). The basic fuel properties are listed in Table 3. Values of density and viscosity of the fuels were measured using Stabinger Viscometer SVM 3000 made by Anton Paar GmbH (measuring accuracy  $< 1\%$ , repeatability 0.1%). The values of calorific value of the fuels were reached by means of isoperibol calorimeter LECO AC600 (measuring range 23.1–57.5 MJ kg<sup>-1</sup> for a 0.35 g sample, accuracy 0.1% RSD) according to ČSN DIN 51900-1 and ČSN DIN 51900-2.

**Table 3.** Fuel parameters (\*EN 590, 2013; \*\*Cisek & Szlachta, 2001)

| Fuel            | Temperature<br>°C | Kinematic<br>Viscosity<br>mm <sup>2</sup> s <sup>-1</sup> | Dynamic<br>Viscosity<br>mPa s | Density<br>kg m <sup>-3</sup> | Calorific<br>value<br>MJ kg <sup>-1</sup> | Cetane<br>number |
|-----------------|-------------------|---|-------------------------------|-------------------------------|---|------------------|
| D100            | 15                | 2.843   | 2.329                         | 819.1                         | 43.15                                     | 50*              |
|                 | 40                | 1.801   | 1.444                         | 801.65                        |   |                  |
| R5D95           | 15                | 3.224   | 2.65                          | 821.9                         | 42.72                                     | -                |
|                 | 40                | 2.022   | 1.627                         | 804.4                         |   |                  |
| R20D80          | 15                | 5.042   | 4.216                         | 836.1                         | 41.74                                     | -                |
|                 | 40                | 2.984   | 2.443                         | 818.75                        |   |                  |
| Rapeseed<br>oil | 15                | 97.655  | 89.962                        | 921.2                         | 37.1                                      | 39.6–44**        |
|                 | 40                | 35.697  | 32.326                        | 905.33                        |   |                  |

The measurement was carried out in stabilized conditions at constant rotation speed of approx. 1,950 min<sup>-1</sup>. The engine rotation speed of 1,950 min<sup>-1</sup> was chosen because at this point the PTO shaft reaches rotation speed of 1,000 min<sup>-1</sup>, which is necessary for proper function of connected agricultural equipment, so it could be assumed that the engine spends most of its working time at this rotation speed. Load of the engine was selected 20, 60, 80 and 100%. The loads of the engine were calculated from maximum

torque at 1,950 rpm for each fuel. At each measurement point the monitored parameters were stabilized. After stabilization the monitored parameters were recorded for approx. 80 s. The mechanical losses in gearbox have no influence on comparative measurement and therefore they were not taken into account. The MS Excel was used for evaluation of the measured data.

Heat release rate (HRR) is one of the most effective ways to obtain information about the combustion process in internal combustion engines. HRR was calculated according to the first law of thermodynamics and Eq. (1). The calculation does not take into account the heat losses during the process. According to Ozsezen et al. (2008), temperature gradients, pressure waves, non-equilibrium conditions, fuel vaporization, and mixing can be ignored. In order to eliminate noise effects, the Savitzky-Golay smoothing filter was used on the recorded pressure data for HRR calculation.

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \quad (1)$$

where  $dQ/d\theta$  – heat release rate (J/°CA);  $V$  – instantaneous cylinder volume (m<sup>3</sup>);  $p$  – instantaneous cylinder pressure (Pa);  $\theta$  – crankshaft angle (°CA);  $\gamma$  – ratio of specific heats at constant pressure and volume (considered constant at 1.35 (Heywood, 1988; Imtenan et al., 2015)).

Cumulative heat release was calculated as integral from HRR as can be seen in Eq. (2). Relative cumulative heat release (RCHR) was calculated from cumulative heat release and it is expressed in percentage.

$$Q_{CHR} = \int_{EOC}^{SOC} \frac{dQ}{d\theta} d\theta \quad (2)$$

where  $Q_{CHR}$  – cumulative heat release (J); SOC – start of combustion (°CA); EOC – end of combustion (°CA).

Indicated work was calculated according to (Heywood, 1988) as an area inside the p-V diagram according to Eq. (3). For the calculation of indicated work only compression and expansion strokes were used in order to exclude the gas exchange from the result, as described Gailis et al. (2017).

$$W_i = \int_{180}^{-180} p dV \quad (3)$$

From the measured and calculated data the indicated efficiency, brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) were calculated. Indicated efficiency is ratio between indicated work and chemical energy, delivered into cylinder in the fuel. Brake thermal efficiency is ratio between effective power and chemical energy delivered into the engine in the fuel.

For comparison of conditions in the cylinder before start of combustion (SOC) the temperature of charge at SOI was calculated according to the ideal gas law. The calculation was performed according to Eq. (4).

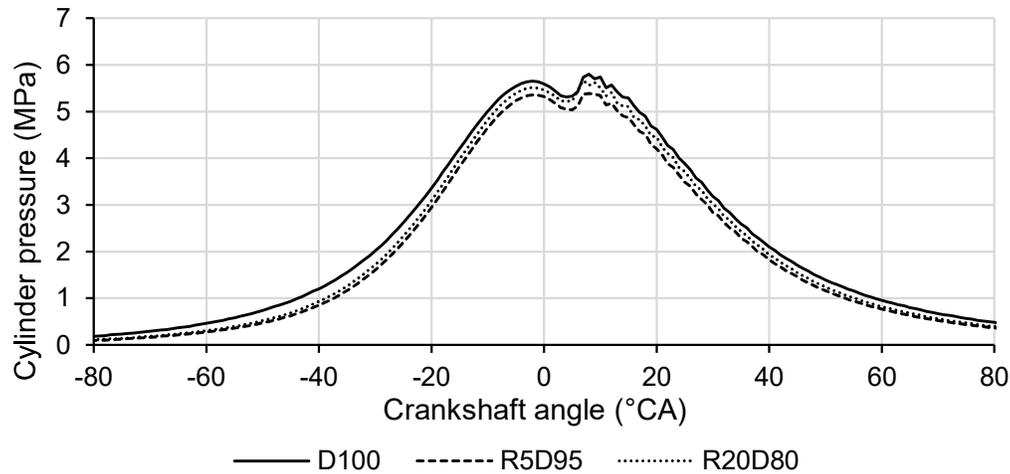
$$T_{SOI} = \frac{p_{SOI} V_{SOI}}{m R_i} \quad (4)$$

where  $T_{SOI}$  – temperature of charge at SOI (K);  $V_{SOI}$  – instantaneous cylinder volume ( $m^3$ );  $m$  – mass air flow per cycle (kg);  $R_i$  – specific gas constant of air ( $J\ kg^{-1}\ K^{-1}$ ).

The ignition delay (ID) was also determined from the data. ID is the period between SOI and SOC, stated in time or crankshaft angle. The SOI is kept constant at  $-12^\circ CA$  (crankshaft angle). SOC can be determined by various methods, including HRR profile (Aldhaidhawi et al., 2017). According to some sources (Heywood, 1988; Aldhaidhawi et al., 2017), change of slope in the HRR profile defines the SOC, other authors (Imtenan et al., 2015) state that SOC occurs once the HRR becomes positive. The moment when HRR becomes positive was considered as SOC. The moment when HRR becomes negative was considered as the end of combustion (EOC).

## RESULTS AND DISCUSSION

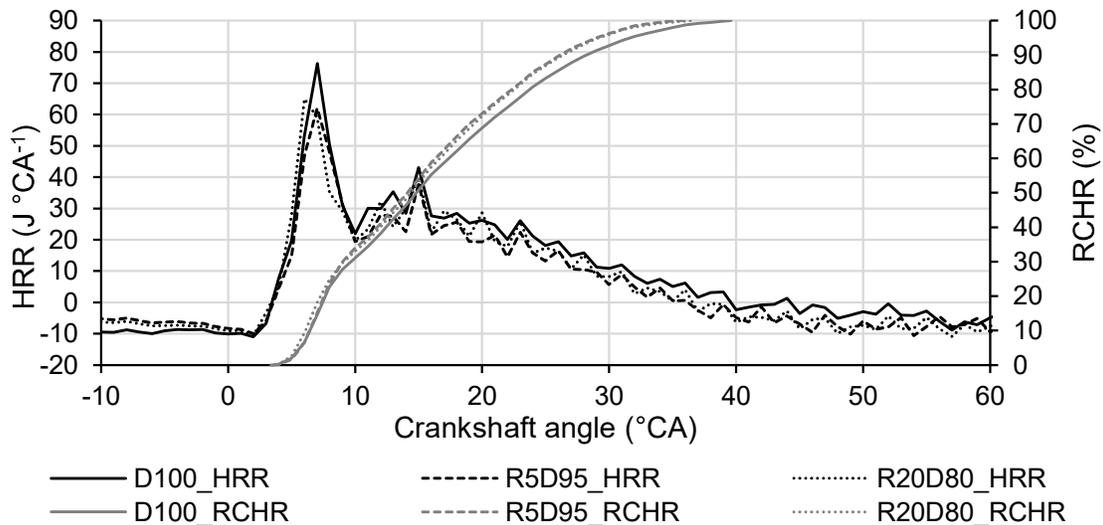
In Fig. 2 the cylinder pressure profile, measured at 20% engine load for all tested fuels can be seen. It is evident that at lower loads the SOC appears after the top dead center. Also, from the figure, it is evident that the cylinder pressure was lower for both of the tested fuel blends in comparison with D100. In the case of R5D95 the decrease of peak cylinder pressure was 7.13% and in the case of R20D80 approx. 2.25%. Also lower cylinder pressure during compression stroke can be seen when using both of the blended fuels in comparison with D100.



**Figure 2.** Course of the cylinder pressure in dependence on crankshaft angle at 20% load.

Fig. 3 shows the results of HRR and RCHR at 20% engine load for all tested fuels. It can be seen that premixed combustion phase (the first peak on HRR profile after the start of combustion) takes the significant part of the heat, released during combustion (approx. 30–35%). Also, the premixed combustion phase is stronger for D100 fuel than for both of the tested fuel blends. Lower intensity of premixed combustion may be caused by worse atomization of the fuel blends with higher viscosity in combination with poor evaporating ability of rapeseed oil in the fuel blends. Diffusion and late combustion

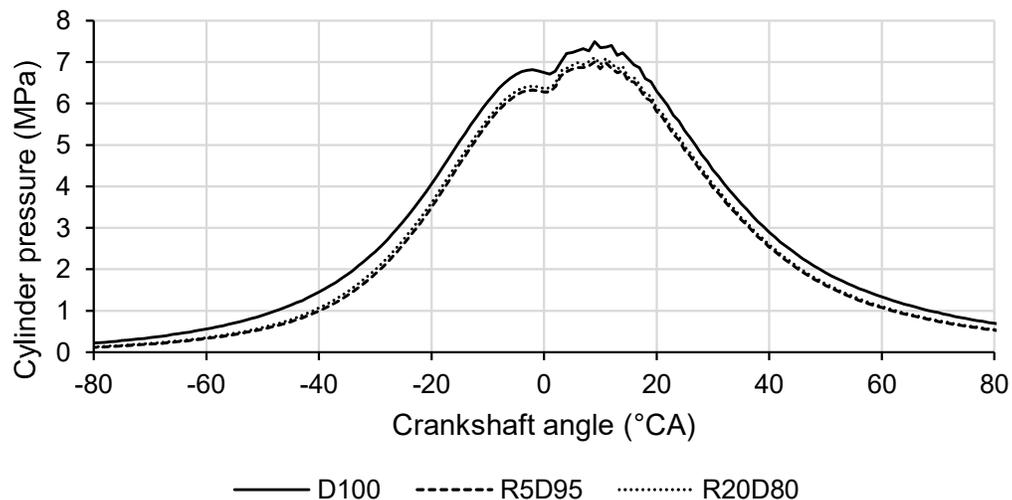
phases showed shorter duration and lower intensity in comparison with D100. Combustion duration was shorter when using R5D95 blend by 9.68% (3.5°CA) and R20D80 by 8.12% (2.93°CA).



**Figure 3.** Course of HRR and RCHR in dependence on crankshaft angle at 20% load.

ID for all tested fuels at all tested engine loads can be seen in Fig. 10. At 20% engine load the differences are under 1% (by 0.67% longer for R5D95 and by 0.51% shorter for R20D80).

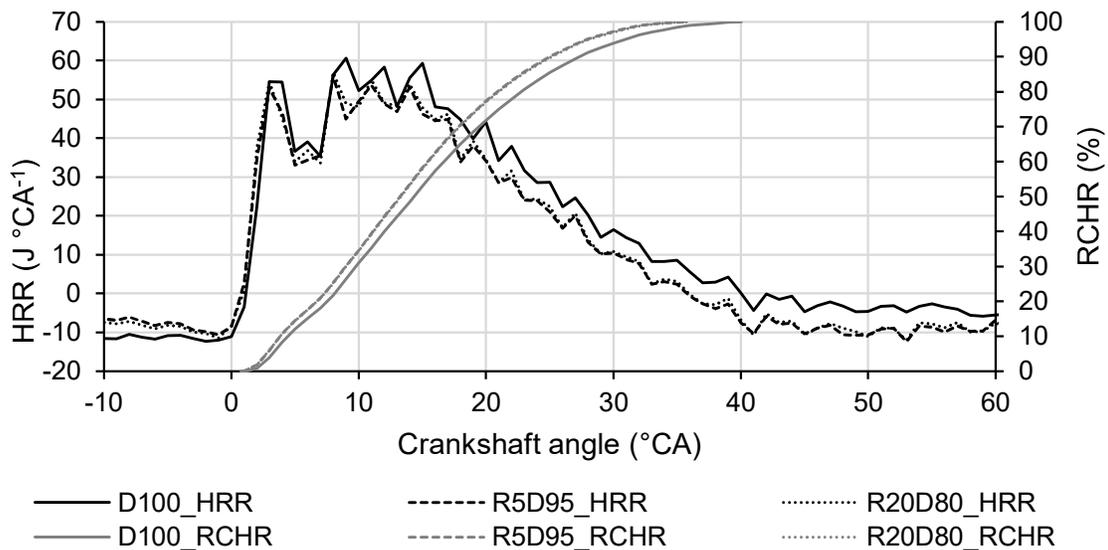
Fig. 4 shows the cylinder pressure profile for all tested fuel at 60% engine load. It can be seen that peak cylinder pressure is considerably lower for both of tested fuel blends in comparison with diesel fuel at this engine load (by 6.52% for R5D95 and by 5.05% for R20D80). Also, lower cylinder pressure can be seen during both compression and expansion strokes. This may be caused by lower calorific value of the blended fuels and by different energy of exhaust gas and thus different intake air pressure.



**Figure 4.** Course of the cylinder pressure in dependence on crankshaft angle at 60% load.

HRR and RCHR for 60% engine load are shown in Fig. 5. It is evident that with increasing engine load the higher proportion of fuel is burned during diffusion combustion phase. Both of the blended fuels reached higher RCHR during premixed combustion in comparison with D100 (R5D95 – 14.4%, R20D80 – 14.3%, D100 – 12%). In absolute values the fuel blends also released higher heat, released during the premixed combustion phase (by 2.97% using R5D95 and by 4.54% using R20D80). This may be caused by shorter ID and longer duration of the premixed combustion phase. Also higher oxygen content in the fuel contributes to premixed combustion. The diffusion and late combustion phases showed shorter duration in comparison with D100, since the EOC occurred earlier, the combustion duration was shorter by 9.97% (3.88°CA) for R5D95 and by 10.01% (3.9°CA) for R20D80. Also, the intensity of diffusion combustion phase was lower for both of blended fuels.

ID at 60% engine load, as can be seen in Fig. 10, was shorter for both blended fuels in comparison with D100. Fuel blend R5D95 showed by 2.85% (0.37°CA) and R20D80 by 1.56

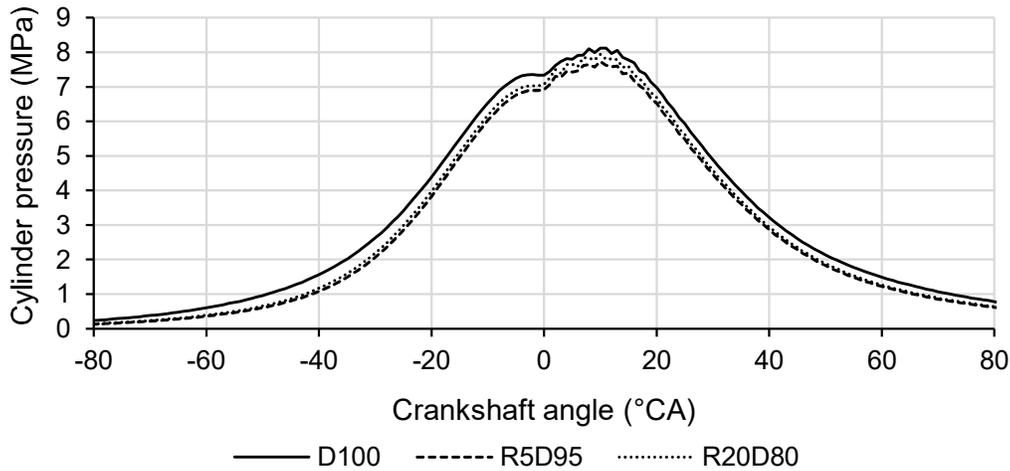


**Figure 5.** Course of HRR and RCHR in dependence on crankshaft angle at 60% load.

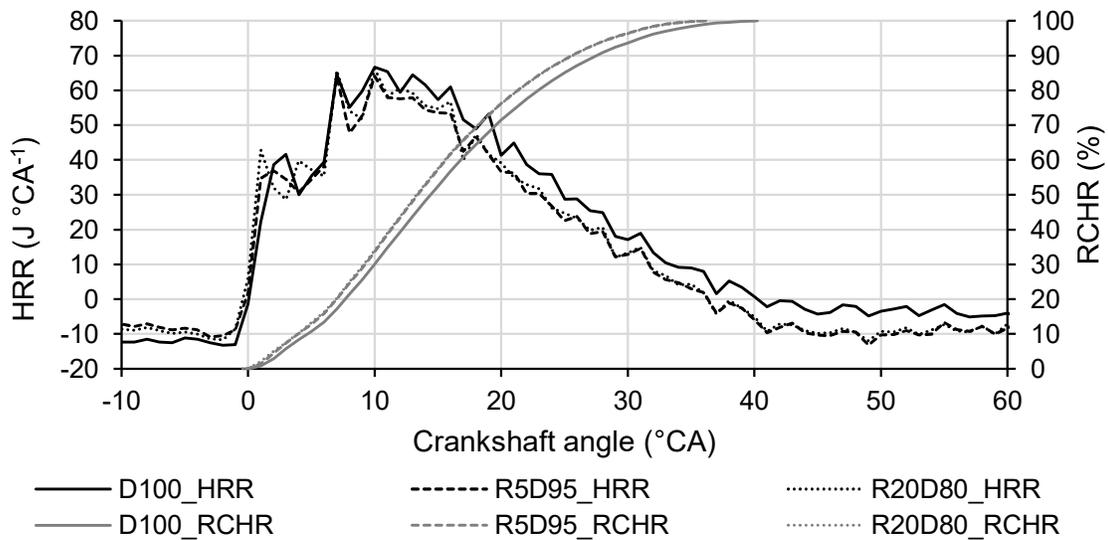
Fig. 6 shows the cylinder pressure profile for all tested fuels at 80% engine load. In comparison with D100 the peak cylinder pressure was lower by 4.75% using R5D95 and by 2.18% using R20D80 fuel blend. Similarly to 60% engine load the lower cylinder pressure can be seen during compression and expansion strokes for both of the tested fuels.

In Fig. 7 HRR and RCHR at 80% engine load for all tested fuels are shown. From the figure it can be seen that at this engine load the peak of the premixed combustion appears earlier for both of the blended fuels in comparison with D100. This is connected with the ignition delay which was shorter for both of the blended fuels in comparison with D100 and with oxygen content in the blended fuels. However, the duration and RCHR of the premixed combustion phase in the case of R20D80 fuel blend was lower than in case of D100 (RCHR for D100 – 8.36%, RCHR for R20D80 – 7.48%, premixed combustion shorter by 1°CA). It is evident that at higher engine load the 20% proportion

of rapeseed oil in the fuel significantly affects the ability of the fuel to evaporate and lowers the intensity of premixed combustion. When using R5D95 fuel blend the RCHR during premixed combustion was higher by 10.11% than in the case of D100. This may be the result of shorter ID and thus longer premixed combustion phase while the peak value remains lower than in case of D100. The EOC also occurred earlier for both of the blended fuels, combustion duration was shorter by 9.17% ( $3.67^{\circ}\text{CA}$ ) when using R5D95 and by 8.65% ( $3.48^{\circ}\text{CA}$ ) when using R20D80.



**Figure 6.** Course of the cylinder pressure in dependance on crankshaft angle at 80% load.

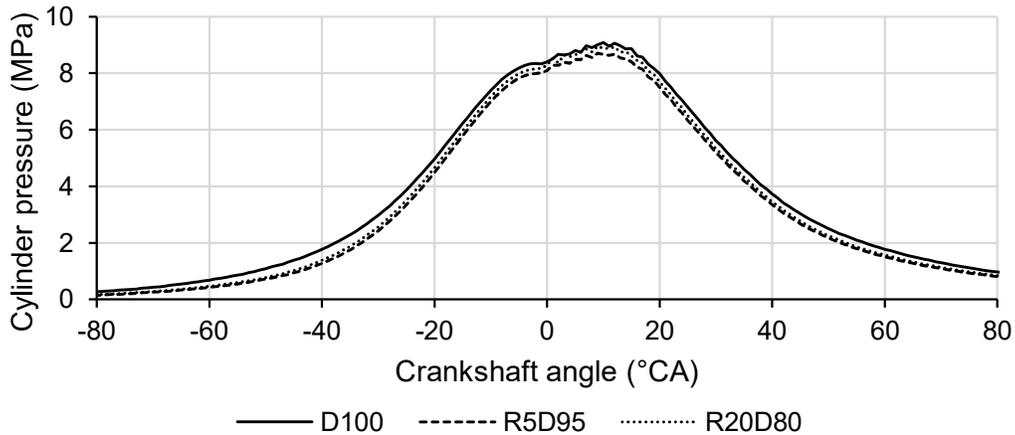


**Figure 7.** Course of HRR and RCHR in dependance on crankshaft angle at 80% load.

ID at 80% load was shorter for both blended fuels in comparison with D100. When the engine was running on the fuel blend R5D95 ID decreased by 1.93% ( $0.23^{\circ}\text{CA}$ ), during operation on the fuel blend R20D80 ID decreased by 3.95% ( $0.48^{\circ}\text{CA}$ ).

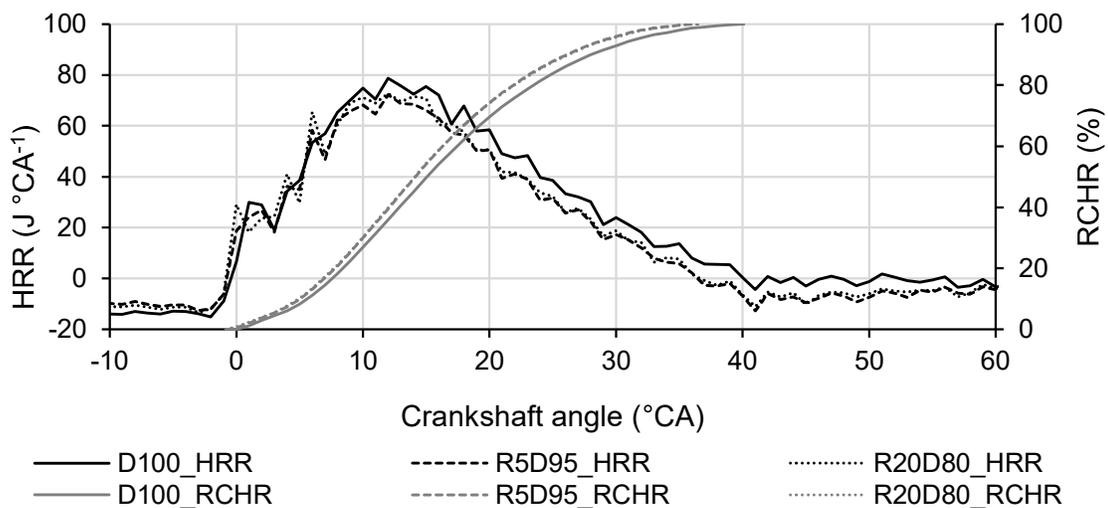
In Fig. 8 the pressure profile in dependence on crankshaft position for all tested fuels at full engine load is shown. It is evident that peak cylinder pressure is also lower

for both of the tested fuel blends in comparison with D100, but the difference is lower in than in previous measured engine loads. Especially when using R20D80 where the difference in peak cylinder pressure was close to the sensor accuracy (1.44%), when using R5D95 fuel blend the peak cylinder pressure at this engine load was lower by 4.17%. However, lower cylinder pressure during compression stroke can be seen, similarly to previous measured engine loads.



**Figure 8.** Course of the cylinder pressure in dependance on crankshaft angle at 100% load.

HRR and RCHR for 100% engine load are shown in Fig. 9. SOC and premixed combustion phase, similarly to 80% engine load, appears earlier with blended fuels in comparison with D100. When using R20D80 the duration of the premixed combustion is also considerably shorter (by 2°CA). RCHR at the end of premixed combustion phase was 5.42% for R5D95, 2.06% for R20D80 and 4.49% for D100. The combustion duration was shorter when using R5D95 fuel blend by 8.2% (3.32°CA) and when using fuel blend R20D80 by 7.53% (3.05°CA) in comparison with D100.



**Figure 9.** Course of HRR and RCHR in dependance on crankshaft angle at 100% load.

The shorter ID at full engine load was found when using both of the blended fuels in comparison with D100. When the engine was running on the fuel blend R5D95 ID decreased by 2.75% (0.32°C<sub>A</sub>), during operation on the fuel blend R20D80 ID decreased by 3.34% (0.37°C<sub>A</sub>).

At all of the measured engine loads the statistically significant difference in peak cylinder pressure was found between all fuels. In Table 4. the analysis of variance (ANOVA), complemented with Tukey HSD post-hoc test for peak cylinder pressure at full engine load can be seen.

**Table 4.** ANOVA with Tukey HSD post-hoc test for peak cylinder pressure at full engine load

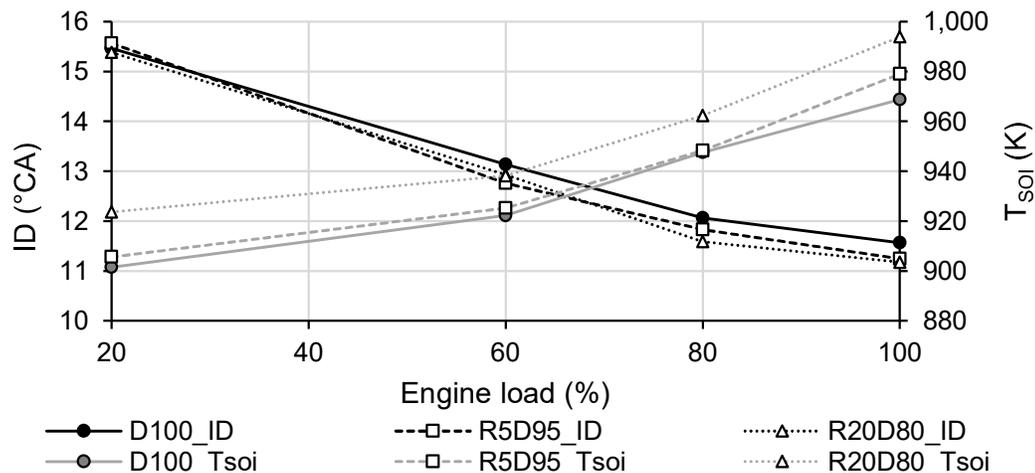
| ANOVA           |                |                    |          |          |
|-----------------|----------------|--------------------|----------|----------|
| $\alpha = 0.05$ | Sum of squares | Degrees of freedom | Variance | F        |
| Between groups  | 73.5574        | 2                  | 36.7787  | 7,743.04 |
| Within groups   | 16.9429        | 3,567              | 0.0047   |          |
| Total           | 90.5003        | 3,569              |          |          |

| Tukey HSD Post-hoc Test  |  |
|--|--|
| D100 vs R5D95: Diff = -0.3788, 95%CI = -0.3862 to -0.3714, $p = 0.0000$  |  |
| D100 vs R20D80: Diff = -0.1311, 95%CI = -0.1378 to -0.1243, $p = 0.0000$ |  |
| R5D95 vs R20D80: Diff = 0.2477, 95%CI = 0.2413 to 0.2541, $p = 0.0000$   |  |

The lower RCHR and duration of premixed combustion phase when using R20D80 fuel blend at higher engine loads (80% and 100%) may be a result of poor evaporating ability of the rapeseed oil in the fuel mixture. Also, shorter ID causes lesser amount of fuel is being injected into the combustion chamber before the combustion begins.

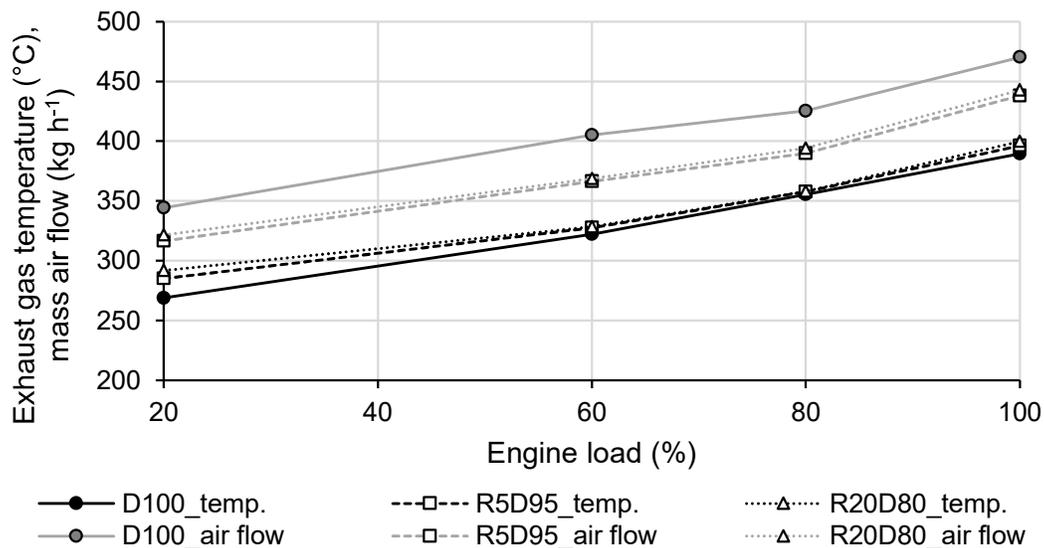
ID and temperature of charge at SOI for all tested fuels at all measurement points are shown in Fig. 10. It is evident that differences in ID between individual tested fuels are increasing with the engine load. It can be seen that in 3 out of 4 measurement points the fuel blend R20D80 showed the shortest ID. Similar results of ID was reached by Devan & Mahalakshmi (2008) with 20% blend of poon oil who also found shorter ID in comparison with mineral diesel fuel. Koder et al. (2018) with neat jatropa and soybean oils also reported shorter ID when using vegetable oils in all measured engine modes.



**Figure 10.** Ignition delay and temperature of charge at SOI for all tested fuels at all measured points.

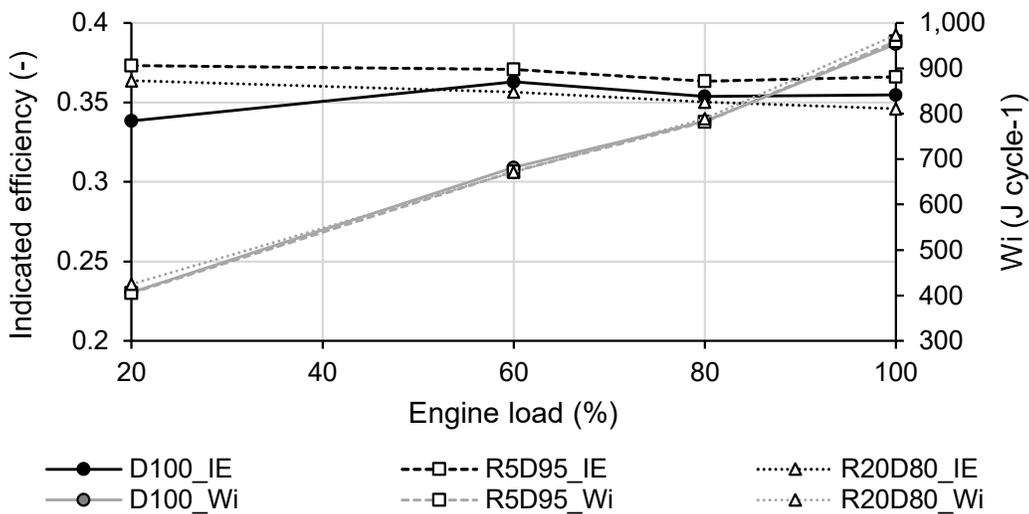
Shorter ID may be result of the temperature of charge at SOI which was highest at all measurement points when using fuel blend R20D80. Also, increased value of compressibility of vegetable oil may cause a slightly earlier injection of the fuel blends into the cylinder (Varde, 1984; Rakopoulos et al., 2005). From the results it can be stated that when using the tested fuel blends in comparison with D100, the temperature of charge at SOI in combination with physical and chemical properties of rapeseed oil shortened the ID despite the cetane number, which was lower for both of the tested fuel blends. However, the differences in temperature of charge between individual fuels are on the border of measurement accuracy (especially the sensor of mass air flow) and the calculation does not take into account the heat losses during the compression.

The temperature of charge is to a large extent affected by speed of turbocharger and its temperature. The speed of the turbocharger depends on kinetic energy of exhaust gas. The kinetic energy of exhaust gas could be affected by the earlier EOC that occurs when using both of the tested fuel blends in comparison with D100. Temperature of the exhaust gas (Fig. 11), that affects the temperature of the turbocharger, was higher when using both of the blended fuels at all measured engine loads in comparison with D100. This can be explained by higher oxygen content for both of blended fuels, especially for R20D80. Gad et al. (2018) also found higher exhaust gas temperature when using 20% blend of palm oil in diesel fuel caused by poor combustion characteristics of the blend. Lower speed and higher temperature of the turbocharger means less mass of air in the cylinder with higher temperature. Also, temperature of the intake air and other ambient conditions affects to some extent the temperature of charge in the cylinder. The ambient temperature variations between individual measurement points were very small, the ambient temperature for D100 was approx. 24.03 °C, for R5D95 approx. 25.58 °C and for R20D80 approx. 26.64 °C.

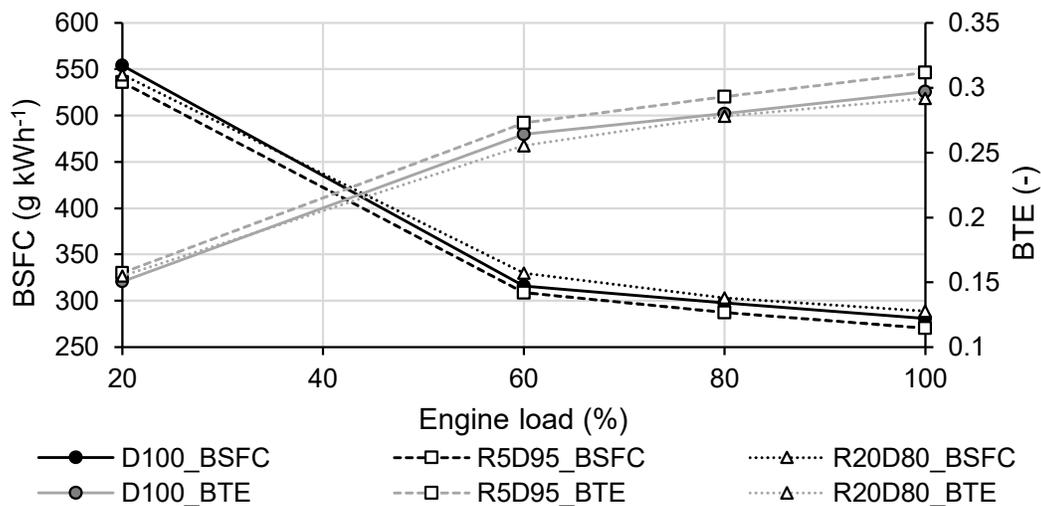


**Figure 11.** Ignition delay and temperature of charge at SOI for all tested fuels at all measured points.

In Figs 2, 4, 6 and 8 the lower pressure during compression stroke (approx. from  $-80^{\circ}\text{CA}$  to  $-20^{\circ}\text{CA}$ ) can be seen. This may be caused by different kinetic energy of exhaust gas. The lower kinetic energy of exhaust gas causes the lower speed of turbocharger and lower mass of charge. This is confirmed by the mass air flow (Fig. 11), which was lower when using both the blended fuels in comparison with D100 at all measured points. Also, from the Figs 3, 5, 7 and 9 it is evident, that during the combustion the R5D95 and R20D80 released lower amount heat than D100 at all measurement points. However, lower amount of intake air takes lower energy during the compression stroke, so that during the whole cycle, the engine produced similar amount of indicated work, as can be seen in Fig. 12. In the case of R5D95 the maximum difference of indicated work in comparison with D100 was 1.27%, in the case of R20D80 the maximum difference was 4.46%.



**Figure 12.** Indicated efficiency and indicated work per cycle for all tested fuels at all measured points.



**Figure 13.** Brake specific fuel consumption and brake thermal efficiency for all tested fuels at all measured points.

Indicated efficiency of the engine at all measurement points for all tested fuels is shown in Fig. 12, BTE and BSFC are shown in Fig. 13. It is evident that fuel blend R5D95 showed the highest indicated efficiency and BTE and the lowest BSFC. R5D95 fuel blend reached the best results in terms of utilization of chemical energy, given in the fuel. This may be caused by increased oxygen content in the fuel in comparison with D100 in combination with fuel properties closer to diesel fuel than in case of R20D80. Rakopoulos et al. (2011) also found increased BTE and even decreased BSFC when using sunflower oil-diesel fuel and corn oil-diesel fuel blends in comparison with diesel fuel.

## CONCLUSIONS

The article is focused on comparison of combustion characteristics and engine efficiency of CI engine, operated on rapeseed oil-diesel fuel blends in comparison with neat diesel fuel. From the results of the measurement following results were made:

- Lower cylinder pressure during compression and expansion stroke was found for both of tested fuel blends in comparison with diesel fuel. Statistically significant decrease of peak cylinder pressure was found at all loads for both of the fuel blends. This may be caused by lower calorific value in combination with lower mass air flow.
- Earlier EOC was found and combustion duration was decreased for both of tested fuel blends in comparison with diesel fuel. As a result, lower heat was released during combustion.
- Despite the lower heat, released during combustion, indicated work was comparable in all measurement points for all fuels. This is a result of lower mass air flow, which may be caused by different kinetic energy and temperature of exhaust gas, which affects the turbocharger. Lower mass of air in the cylinder then consume less energy during compression stroke.
- Ignition delay was shorter using the blended fuels in all measured loads in comparison with the diesel fuel. The fuel, containing 20% rapeseed oil reached the shortest ID in 3 out of 4 measured loads. The cause may be higher temperature of charge when using biofuels in combination with physical and chemical properties of rapeseed oil.
- In comparison with diesel fuel the indicated efficiency and BTE was considerably increased while BSFC was slightly decreased using fuel blend with 5% of rapeseed oil. On the contrary, 20% blend of rapeseed oil caused increase of BSFC and slight decrease of BTE and indicated efficiency.

From the obtained results it is evident that from the viewpoint of combustion characteristics and engine efficiency, the fuel with 5% rapeseed oil showed the best results from the tested fuels since the fuel blend and its combustion are closer to the diesel fuel than blend with 20% rapeseed oil, while higher amount of oxygen contributes to higher engine efficiency in comparison with diesel.

Research in the field of biofuels is still very important. In present time the large development of electric mobility is reflected only in some areas. In many areas the utilization of current electric drive technologies is absolutely impossible. Agriculture is one of these important areas. In agriculture it is very interesting to explore the

possibilities of using various products of plant and livestock production as a fuel to power machinery and equipment. Even a few percent of biofuel in the diesel fuel could significantly contribute to cleaner environment and fulfilment of internationally binding agreements.

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