THE EFFECT OF DIESEL FUEL MIXTURE AND CAMELINA OIL ESTER ON SELECTED PARAMETERS OF COMBUSTION PROCESS

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Abstract

This paper presents the results of comparative research into basic physical and chemical properties of pure diesel fuel and two types of mixtures: Camelina oil ester (Camelina oil ester content in the mixture was 10% and 15%) and diesel fuel. Camelina oil ester was used to show that it can also be used as an additive plant to diesel fuel (according to the plans of the European Union). Similarly to on a large scale used ester of rapeseed oil. The base fuel in the research was clean diesel fuel (without the addition of ester as in the case of diesel fuel available at petrol stations).The use of fuels with different physical and chemical properties to supply the diesel engine and the use of measuring equipment used for determining the parameters of fast-changing operation resulted in the determination of the operating medium average pressure during the combustion process. On the basis of the experimental pressure characteristics of heat release were identified. Analysis of these characteristics determined the proportion of total combustion phases of kinetics and diffusion, depending on the type of fuel and operating conditions.

Finally, the paper shows that the fuel with different physicochemical properties has a significant effect on the combustion process. A comparative assessment of the suitability of tested mixtures for operational use was made.

Keywords: Camelina oil ester, combustion, diesel engine

1. Introduction

In the European Union countries including also Poland rapeseed oil is the basicoily raw material for biodiesel production, however, its potential resources are limited as cultivation of rapeseed requires specific climate and soil conditions. Thus other sources of oily raw materials are sought after and it seems that the oil obtained from a plant known as camelina can be quite important [5].

Camelina (Latin: Camelina sativa) belongs to the oldest oilseeds ever known. In the past this plant was cultivated all over Europe but in the last few decades it was not very popular due to the low crops [4, 5, 11]. Recently, thanks to agrotechnical progress the camelina crops have improved. What is more, in comparison to rapeseed this plant can be cultivated on poorer soils and the expenses related to fertilizers and crop protection chemicals are much lower [6]. Another important aspect is the results of analyses which indicate lower emissions of greenhouse gases not only in comparison to fossil fuels but also with reference to other liquid biofuels [8, 10].

Methyl esters obtained from camelina oil in their pure form cannot be marketed as biodiesel due to their high iodine value and cold filter plugging point (CFPP) [6, 7]. For this reason obtaining a standard-conformable product is possible only by making an appropriate mixture of diesel fuel with little amount of the camelina derived ester [6, 7].

2. The aim of the research

The aim of the research was to assess the effect of mixtures of diesel fuel and camelina-derived ester on selected parameters of the combustion process in the self-ignition Perkins engine, model 1104C–E44T. During experimental tests the engine worked at maximum volumetric fuel charge at the same speeds. The tests were carried out at two characteristic rotational speeds of the engine crankshaft

corresponding to: the engine's maximum power (2000 rpm) and maximum torque (1400 rpm). In order to determine the quantities describing the combustion process, a thermodynamic (zero-dimensional) model was used based on momentary pressures of the operating medium in the combustion chamber and the volume of the combustion chamber corresponding to them [8, 10].

3. Engine test stand and physicochemical properties of examined fuels

The test stand was based on a four-cylinder self-ignition engine - Perkins 1104C–E44T (EuroStage II) model manufactured in 2010. This model was equipped with a turbo-compressor unit and rarely encountered system of two controllers. One was responsible for the motor operation supervision and the other one was located on a radial piston distributor injection pump, Bosch VP-30 type. The stand was also equipped with a measuring system enabling determination of the engine's performance parameters and fast-changing pressures of the operating medium.

The fuels tested were chosen in view of their availability on the market and their non-uniform physicochemical properties which usually exert a significant effect on the course of the combustion process [1-3, 8, 11, 13, 14].

L10 and L15 mixtures used in the tests were volumetric mixtures consisting of 10% of camelina derived ester or 15% camelina-derived ester and low-sulphur diesel fuel (ON). The block diagram of the test stand is depicted in Fig. 1. Tab. 1 compares selected (main) physicochemical properties of fuels used in our research. Due to the fact that the mixtures contain more than 7% volumetrically of the methyl ester they should be treated as biofuels and therefore their physicochemical properties should be referred to the standard for biodiesel fuel (EN 14214).However, in accordance with the European Union plans concerning the biofuel market, in future such mixtures can be regarded as diesel fuel. This resembles the current situation of the B5 fuel (5% of methyl ester from rapeseed plus diesel) or B7 fuel (7% of methyl ester from rapeseed plus diesel fuel). This is the reason why a comparison of physicochemical properties of these mixtures with diesel properties specified in the EN 590 standard is justified.
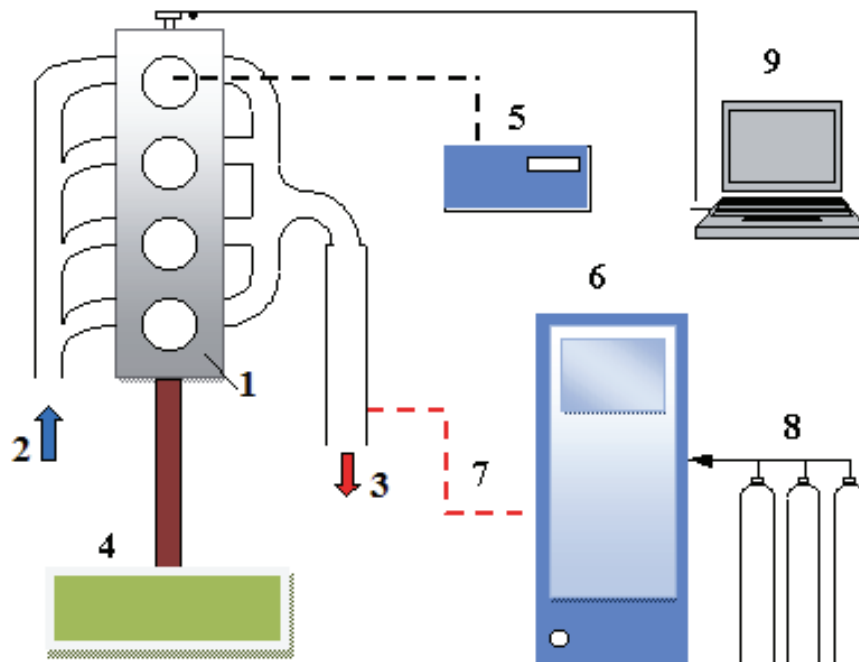

Fig. 1. Block diagram of engine test stand [7]: 1 – PERKINS engine, Model 1104C-E44T; 2 – air intake; 3 – exhaust gas outlet; 4 – SCHENCK II eddy current brake; 5 – AVL IndiSmart system of fast-changing parameters; 6 – AVL CEB 2 exhaust gas analyser; 7 – heating path; 8 – set of reference gases; 9 – control station – notebook

Tab.1. Selected results of physicochemical tests for mixtures of camelina-derived ester and diesel fuel

Parameter	Diesel fuel- ON (in parentheses – values according to EN 590 standard)	L10	L15
Cetanenumber	52.4 (min. 51)	51.6	51.4
N.c.v. [MJ/kg]	43.2	42.5	42.2
Density at 15°C [kg/m^3],	835.4 (820-845)	846	848
Kinematic viscosity [mm^2/s] (~40°C)	2.64(2.00 - 4.50)	3.26	3.40
Ignition temperature [°C]	63 (min. 55)	75	80
Cold filter plugging point (CFPP) [°C]	-15 (max. 0)	-12	-11
Sulphur content S [mg/kg]	9 (max. 10)	8.9	8.9
Water content [mg/kg]	43.8 (max. 200)	58	63

4. Description of the method of determining basic parameters of the combustion process

The research object was three fuels. The diesel fuel without any additives of plant-derived esters (like commercial oils available at petrol stations) or improvers was adopted as reference fuel. From the point of view of quality this product meets specifications of the EN 590 standard. The remaining two fuels were two mixtures of the methyl ester derived from the camelina oil whose contents in the mixtures were 10% and 15%, respectively. The said mixtures did not contain any additives, either. Experimental tests on mixtures of the camelina oil ester and pure diesel were conducted in the engine test bed of the Institute of Vehicles at the Faculty of Automotive and Construction Machinery Engineering of Warsaw University of Technology. Research works were carried out on the test engine 1104C–E44T (manufactured in 2010) with the use of the measuring equipment made by Schenck company and an AVL system of fast-changing parameters (dedicated to this engine). Thanks to the appropriately selected measuring equipment momentary pressures of the operating medium in the combustion chamber were determined as well as corresponding to them volumes of the combustion chamber. Knowing these quantities it was possible to calculate the rate of heat release. To this end the equation of the first law of thermodynamics was applied [1, 3, 10]:

$$dQ_x = dU + pdV + dQ_{sc},$$ (1)

Following transformations and substitution of finite differences for differentials, the quantity dU + pdV takes the form [10]:

$$dU + pdV = \frac{1}{k-1}\left[k\,\frac{p_i + p_{i-1}}{2}(V_i - V_{i-1}) + \frac{V_i + V_{i-1}}{2}(p_i - p_{i-1}) \right],$$ (2)

The heat transferred to the combustion chamber walls is [10]:

$$dQ_{sc} = \alpha_g \cdot F \cdot \Delta T \cdot dt.$$ (3)

Quantity α_g was determined according to the well-known and quoted in specialist literature [10] empirical correlations which allow for calculation of heat transfer coefficient. F is a heat transferring area, where dt value can be replaced with a finite quantity $dt = \frac{\alpha_i - \alpha_{i-1}}{6 \cdot n}$, where α is the crankshaft rotation angle expressed in degrees of the engine crankshaft rotation.

Taking into account the heat transferred between the operating medium and the cylinder walls, by using the above relationships we obtain [10]:

$$x(\alpha)=\left\{\frac{1}{k\text{-}1}\left[k\frac{p_i(\alpha)+p_{i\text{-}1}(\alpha)}{2}\left(V_i(\alpha)-V_{i\text{-}1}(\alpha)\right)+\frac{V_i+V_{i\text{-}1}}{2}\left(p_i(\alpha)-p_{i\text{-}1}(\alpha)\right)\right]\right\}$$

$$+\alpha_g(\alpha)F(\alpha)\frac{\alpha_i\text{-}\alpha_{i\text{-}1}}{6n}\Delta T, \tag{4}$$

where:

$\Delta T = T - T_{sc}$ – is the difference in temperatures between the operating medium and the cylinder walls,

k – heat capacity ratio.

The final value of the rate of heat release $\dot{x}$(in literature it is usually denoted by Q) was determined from the formula [10]:

$$Q(\alpha) = \dot{x}(\alpha) = \frac{x_i(\alpha)-x_{i\text{-}1}(\alpha)}{\alpha_i-\alpha_{i\text{-}1}}. \tag{5}$$

To determine the heat transfer coefficient$\alpha_g$in the combustion engine theory the correlations put forward by G. Woschni are widely used [10]. They reflect the quality aspect of the phenomenon of the heat transfer between the operating medium and the cylinder walls and quantitative correlations of the heat transfer process parameters. In this paper we used the method of determining the heat transfer coefficient put forward by Woschni as the one most suitable for this type of research [10].

Woschni proposed the following correlation:

$$\alpha_g=127.93\cdot D^{-0,2}\cdot p_i^{0,8}\cdot T_{SC}^{-0,53}\cdot w^{0,8}, \tag{6}$$

at:

$$w=A_1 c_m+A_2\frac{V_S T_1}{p_1 V_1}\left(p_i-p_0\right), \tag{7}$$

where:

$A_1 = 2.28 + 0.308\, c_u/c_m\ [-]$,

$A_2 = 3.24\ 10^{-3}\ [\text{m/s}\cdot\text{K}]$ (for the engine with fuel directly injected to the combustion chamber),

D – cylinder diameter [m],

$c_m = S\, n/30$ –mean piston speed [m/s],

$c_u = D\,\pi\, n/60$–circumferential air speed in the cylinder [m/s],

V_s – piston-swept volume [m³],

p_0 – pressure at start of combustion [Pa],

p_1 – pressure in the cylinder at the moment the inlet valve is closed [Pa],

T_1 – temperature in the engine cylinder at the moment the inlet valve is closed [K],

V_1 – engine cylinder volume at the moment the inlet valve is closed [m³].

After the rate of heat release has been determined in accordance with the above outlined methodology, we obtain the course of this quantity depicted in Fig. 6 and Fig. 7.

5. Research results for selected combustion process parameters

Figure 2 and 3 depict pressure increments in the combustion chamber of the tested engine, whereas the remaining ones (Fig. 4-7) show the quantity and the rate of heat released in the combustion process. All the mixtures were tested in identical engine operating conditions (speed and load). Tests were conducted at two characteristic rotational speeds of the engine's crankshaft: the speed corresponding to the engine's maximum power (2000 rpm) and rotational speed of the maximum torque generated by the engine (1400 rpm). For the sake of our research no adjustments were made to the engine. It worked according to the original manufacturer's settings.
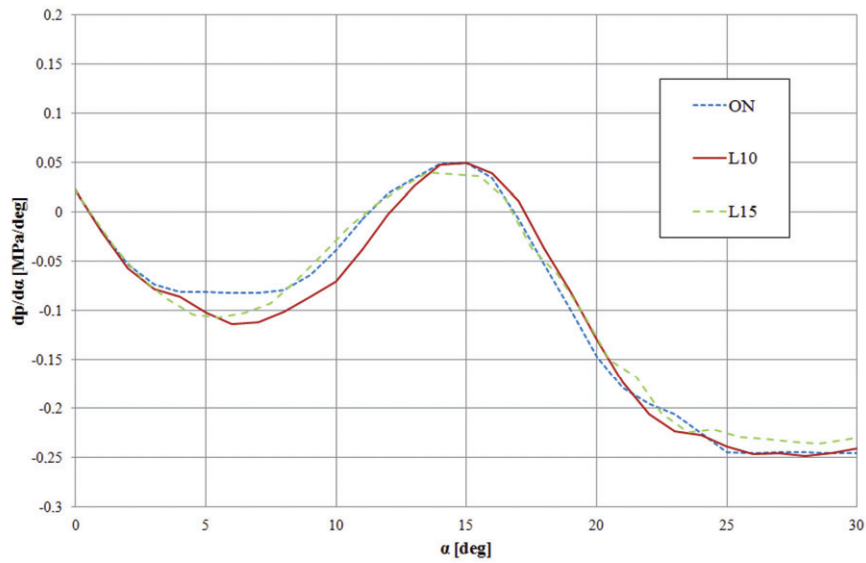
Fig. 2.   Averaged pressure increments in the combustion chamber noted at the engine's crankshaft rotational speed corresponding to the maximum torque (1400 rpm) and at maximum volumetric fuel charge
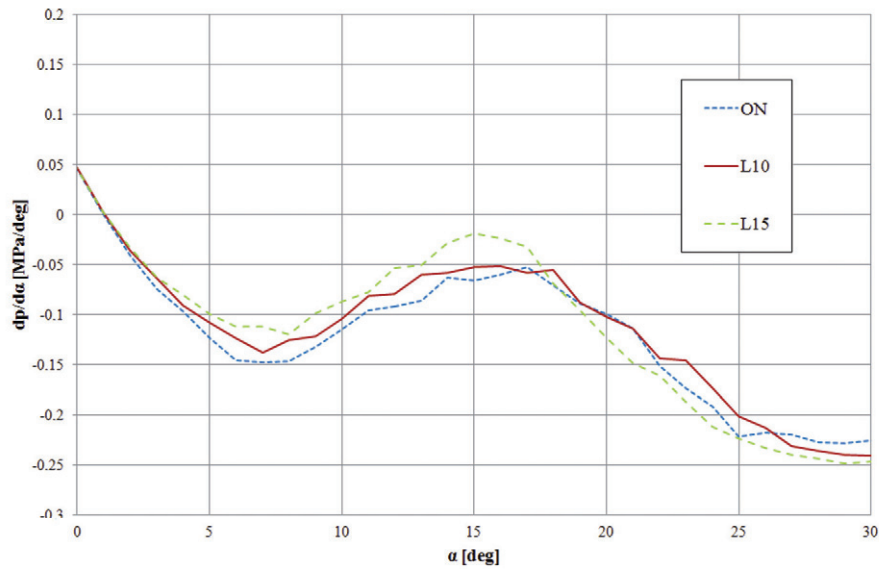

Fig. 3.   Averaged pressure increments in the combustion chamber noted at the engine's crankshaft rotational speed corresponding to the maximum engine power (2000 rpm) and at maximum volumetric fuel charge
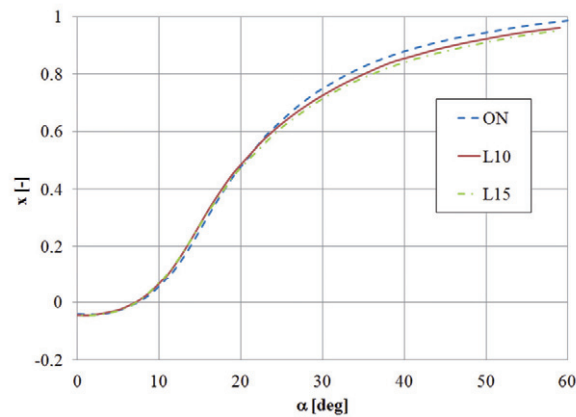

Fig. 4.   Averaged values of the mass fraction burnt x determined at the engine's crankshaft rotational speed corresponding to the maximum torque (1400 rpm) and at maximum volumetric fuel charge
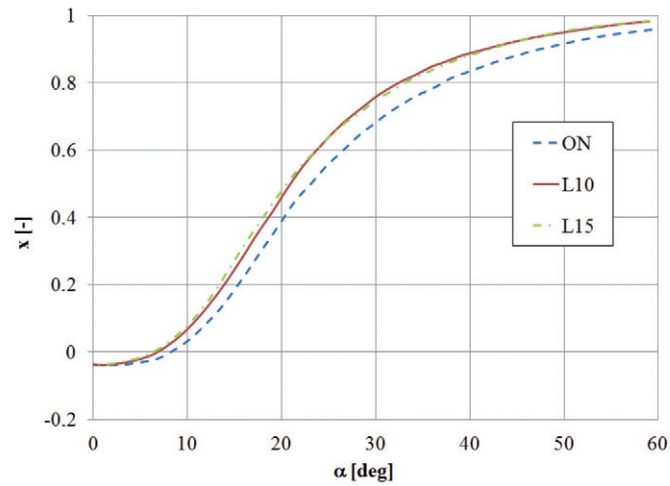
Fig. 5. Averaged values of the mass fraction burnt x determined at the crankshaft rotational speed corresponding to the maximum engine power (2000 rpm) and at maximum volumetric fuel charge
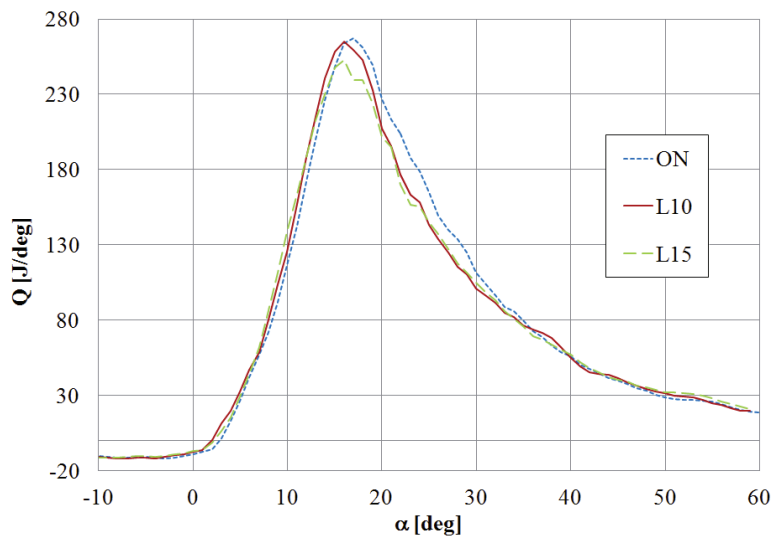

Fig. 6. Averaged values of the rate of heat release Q determined at the crankshaft rotational speed corresponding to the maximum torque (1400 rpm) and at the maximum volumetric fuel charge
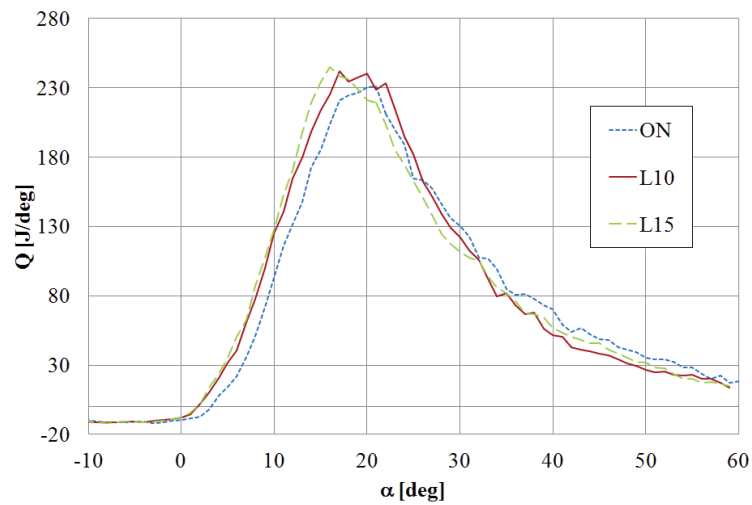

Fig. 7. Averaged values of the rate of heat release Q determined at the crankshaft rotational speed corresponding to the maximum engine power (2000 rpm) and at the maximum volumetric fuel charge

6. Summary

On the basis of conducted laboratory tests it was found out that the examined mixtures:
- differ from one another in respect of their physicochemical properties (Tab. 1),
- meet the requirements of the EN 590 standard (Tab. 1).

An analysis of the results obtained from the engine test bed leads to the following conclusions:
- in averaged diagrams of the operating medium pressure increments in the combustion chamber, in comparison to the L10 and L15 mixtures, self-ignition for diesel fuel is initiated later by ca. 1-3 CA deg.(Fig. 2 and Fig. 3) at two characteristic rotational speeds of the engine's crankshaft, which results mainly from different physicochemical properties of the examined mixtures,
- the highest pressure increments in the combustion chamber were noted for the L10 and L15 mixtures,
- when the engine was supplied with the mixtures of diesel fuel and camelina oil ester it was found out that the total amount of the released heat is higher for the mixtures of diesel fuel and camelina oil ester (max. by 6%),
- application of boost pressure in the engine causes that more heat is released in the diffusion phase of the combustion process (in comparison to undercharged engines – see: articles[1, 3, 6, 7]). For this reason the values of this indicator are comparable to the ones noted in the premixed combustion phase. Therefore it cannot be unambiguously decided at which moment (and at what crankshaft rotation angle) the premixed combustion phase is over and the diffusion phase starts.

The research results show that camelina methyl esters can be used as a component of diesel fuel for self-ignition engines. Physicochemical properties of the examined mixtures meet the main parameter requirements specified by the EN 590 standard for diesel fuels. Their combustion process is not very different from that determined for the diesel fuel. However, it must be borne in mind that examining of the remaining parameters set in the EN 590 standard and not covered by this paper would be appropriate.

Acknowledgement

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