Article citation info:
Sander P, Longwic R, Tarkowski S. Analysis of the influence of the n-hexane content in the mixture with rapeseed oil on the auto-ignition delay angle of the fuel. Combustion Engines. 0000;XXX(X):xx-xx. https://doi.org/10.19206/CE-211729

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Analysis of the influence of the n-hexane content in the mixture with rapeseed oil on the auto-ignition delay angle of the fuel

ARTICLE INFO

Received: 20 June 2025
Revised: 11 September 2025
Accepted: 6 October 2025
Available online: 14 November 2025

Increasing emission-reduction requirements and EU renewable-energy targets have created a demand for biofuels that can be operated in diesel engines without modification. Current production routes – FAME and HVO – although markedly improving the physicochemical properties of rapeseed oil, are carried out in capital-intensive industrial plants and depend on continuous feedstock supply. To simplify the production chain, rapeseed oil was diluted with the inert hydrocarbon n-hexane; the resulting decrease in viscosity and flash point facilitated spray formation and ignition initiation. The effect of n-hexane content (v/v) in blends with rapeseed oil on the ignition-delay angle in a direct-injection diesel engine was analysed. Cylinder-pressure traces were used to determine the crank-angle interval between the start of injection and the start of combustion, and the results were compared with data for commercial diesel fuel and neat rapeseed oil. A progressive increase in the n-hexane fraction was found to shorten the ignition-delay angle relative to neat rapeseed oil, bringing it closer to the value observed for diesel fuel. The results confirm the suitability of the investigated blends as an alternative fuel and indicate that they can be used in currently operated diesel engines without costly modifications.

Key words: *diesel engine, rapeseed oil, combustion, auto-ignition delay, n-hexane*

1. Introduction

For nearly a century and a half, vehicles equipped with internal combustion engines powered by liquid fuels, most commonly hydrocarbons, have been a widespread means of transportation [25, 28]. Along with the growing environmental awareness of societies and the resulting tightening of exhaust emission standards, numerous research centers have been working on improving both fuel supply systems and the design of internal combustion engines themselves. Another important direction of research aimed at meeting strict emission regulations is the enhancement of fuel properties used in combustion engines [5, 11, 23, 30]. The physicochemical properties of fuels have a direct impact on the injection and combustion processes [3, 5, 16, 17, 26, 31, 34].

Achieving proper technical parameters in the operation of a compression ignition (CI) engine – namely, low fuel consumption, reduced emissions of toxic exhaust components, and desirable torque characteristics – depends primarily on the correct course of the combustion process. Despite decades of research, combustion in CI engines remains one of the least understood processes due to the complexity and interdependence of phenomena such as fuel injection and atomization, droplet evaporation, mixing with air, pre-flame reactions, and the initiation and development of combustion itself. These difficulties are largely due to the heterogeneous nature of the fuel–air mixture, which is characteristic of compression ignition [2, 15, 18, 20, 32, 33].

One of the key parameters describing the combustion process in a CI engine is the ignition delay – the period between the start of fuel injection and the onset of combustion, which causes a rapid increase in pressure and temperature within the cylinder. The ignition delay angle, expressed in degrees of crankshaft rotation (CA), significantly influences the entire combustion process: pressure rise rate, cold start capability, operating stability, and exhaust emissions. It can be defined as the difference between the start of injection angle (α_{pw}) and the start of combustion angle (α_{ps}) [6, 10, 13, 33].

The ignition delay consists of two parts: the physical delay (τ_f), related to the formation and evaporation of the fuel spray, and the chemical delay (τ_{ch}), associated with the chemical reactions leading to the formation of ignition kernels [4, 29, 33]:

$$\tau_s = \tau_f + \tau_{ch} \tag{1}$$

For hydrocarbon fuels, the chemical delay can be described as the sum of three phases [2, 33]:

$$\tau_{ch} = \tau_1 + \tau_2 + \tau_3 \tag{2}$$

where: τ_1 – cool flame delay, τ_2 – blue flame delay (chemiluminescence), τ_3 – hot flame delay (main combustion phase).

The length of the ignition delay is influenced by various physical and chemical parameters, including air temperature and pressure, the fuel's cetane number, piston speed, and engine design. A general mathematical expression describing this phenomenon is: [29, 33]:

$$\tau_s = A\rho^{-n}e^{E_A/RT} \tag{3}$$

where: τ_s – ignition delay period [ms], ρ – air pressure (fuel–air mixture) [bar], E_A– apparent activation energy [J/mol], R – universal gas constant [J/(mol·K)], T – air temperature (fuel–air mixture) [K], A – fuel-dependent constants, n – engine speed.

According to the formula proposed by Hardenberg and Hase, the ignition delay τ_s of a fuel in compression ignition engines can be determined using equation (4). This equation expresses the ignition delay as a function of compression temperature T, pressure p, and engine speed n [8, 9].

$$\tau_s = \frac{n}{6}\left(0.36 + 0.22 \cdot \bar{S}_p\right)$$
$$\exp\left[E_A\left(\frac{1}{RT} - \frac{1}{17190}\right) + \left(\frac{21.2}{p-12.4}\right)^{0.63}\right] \quad (4)$$

where: $\bar{S}_p$ – mean piston speed [m/s], R – gas constant, 8.3134 J/(mol·K), EA – apparent activation energy.

$$E_A = \frac{618.840}{LC+25} \quad [J/mol] \quad (5)$$

where: LC – cetane number.

$$\tau'_{id} = \frac{\tau_{id}(\alpha^o OWK)}{0.006 \cdot n} \quad [m/s] \quad (6)$$

The values of T and p can be estimated by assuming a polytropic compression process, using the following equations:

$$T_{TC} = T_i \cdot r_c^{(m-1)} \quad (7)$$

$$p_{TC} = p_i \cdot r_c^{m} \quad (8)$$

where: m – polytropic compression exponent, r_c – compression ratio, subscript "i" – denotes intake air parameters [8, 9].

A higher cetane number corresponds to a lower activation energy, which translates into a shorter ignition delay. For practical purposes, equation (6) is also used to convert the delay angle into time [1, 29, 33].

In the context of alternative fuel research, a key factor is how the physicochemical properties of the fuel affect ignition delay. Rapeseed oil, which has a slightly lower cetane number than diesel fuel, exhibits a significantly longer ignition delay, as confirmed by numerous studies [14, 22, 23]. This is attributed to its higher heat of vaporization, greater viscosity, and lower chemical reactivity. In practice, this results in a longer time needed for the formation of initial ignition sites, which can hinder cold starts and negatively affect emissions of nitrogen oxides and particulate matter.

However, there are only a few studies [33] in which attempts have been made to explain the influence of chemical composition on the duration of the ignition delay period. Zablocki pointed out that the ignition delay varies depending on the properties of the fuel itself, which are determined by the chemical structure of its molecules. He identified a relationship between the ignition delay and the number of carbon atoms in the molecule for normal paraffinic, naphthenic, and olefinic hydrocarbons. The characteristics of the ignition delay process indicate that, as the number of carbon atoms in the fuel molecule increases, the ignition delay decreases. This leads to the conclusion that fuels containing large amounts of olefinic and paraffinic hydrocarbons are characterized by short ignition delay periods (τ_s), due to the ease of hydrocarbon chain decomposition at elevated temperatures.

The air temperature intake during the filling process also has a significant impact on the ignition delay time. The lower the air temperature, the longer the ignition delay. The combustion characteristics of a compression ignition engine cause the intake air temperature to be strongly correlated with the compression ratio [12, 33].

A few other factors, including the cetane number, temperature at the end of compression, intake air density, fuel injection timing, combustion chamber design, and engine speed, also influence ignition delay. These aspects make ignition delay a complex phenomenon.

The authors undertook a study aimed at modifying the physicochemical properties of rapeseed oil by adding a reactive solvent – n-hexane – in various volume proportions. The conducted analyses demonstrated that n-hexane had a positive effect on the operational properties of rapeseed oil, most notably by significantly reducing its viscosity [19, 36]. This reduction in viscosity can considerably enhance fuel atomization and evaporation in the combustion chamber, which in turn influences the ignition process and overall energy efficiency of the fuel in a diesel engine [7, 21, 23].

To mitigate this unfavorable effect, the present study introduced a 10% volume addition of n-hexane to rapeseed oil. n-Hexane, a hydrocarbon with a low boiling point, high volatility, and high reactivity, was intended to improve the ignition characteristics of the blend. Tests showed that this additive reduced the ignition delay compared to pure rapeseed oil, although the values still did not reach those typical of diesel fuel.

The experiments were conducted on a compression ignition engine using a chassis dynamometer, allowing the simulation of real-world operating conditions. Data analysis revealed that the rapeseed oil and 10% n-hexane blend exhibited a shorter ignition delay angle than pure rapeseed oil, indicating improved ignition behavior. However, the values remained higher than those for diesel fuel, confirming that while the modification had a positive impact, it did not eliminate the differences [19, 35, 36].

The aim of the study was to determine the influence of the physicochemical properties of the modified rapeseed oil on the ignition process, with particular emphasis on ignition delay as one of the main factors determining the efficiency, emissions, and reliability of a diesel engine.

2. Materials and methods

Rapeseed oil (Ro), called Kujawski (ZT "Kruszwica" S.A., Poland), n-hexane (ReagentPlus≥99%, Sigma-Aldrich) were used for measurements and the mixture preparation. The following fuels were studied:

- diesel oil – Df
- rapeseed oil – Ro
- mixture of rapeseed oil with 10% n-hexane – RoHex10.

Diesel fuel (Df) meeting the requirements of EN590 [27] commercial rapeseed oil (Ro), non-reactive solvent n-hexane were used for testing. n-Hexane (C_6H_{14}) is an organic chemical compound from the group of alkanos. n-Hexane isomers are very little reactive and often used as solvents in organic reactions because they are highly non-polar. On the basis of rapeseed oil (Ro) mixtures with n-hexane were made in proportions of 10% (RoHex10). The main physicochemical properties of the tested fuels are presented in Table 1.

Table 1. Basic physicochemical parameters of the tested fuels [19]

Fuel type	Density 20°C $[kg/m^3]$	Kinematic Viscosity 20°C $[mm^2/s]$	Surface tension 20°C [mN/m]	Calorific value [MJ/kg]
Ro	916.00	34.89	34.15	37.10
RoHex10	895.43	19.64	30.08	30.77
Df	840.00	2.70	29.15	43.84

The engine tests were carried out on a diesel engine equipped with a common rail injection system, installed in a Fiat Qubo vehicle that met Euro 5 emission standards. The vehicle was fitted with a five-speed manual gearbox. The test vehicle was additionally equipped with an external, independent fuel tank and an auxiliary fuel pump, which allowed for quick replacement of the tested fuels. The modification of the vehicle's fuel system was concerned only with the low-pressure side (approximately 0.3 MPa). When switching to the auxiliary (low-pressure) fuel system, the main fuel supply was automatically disconnected. The high-pressure fuel system was not modified, which ensured that the fuel pressure in this part of the system remained constant for all tests. The technical specifications of the engine and a view of the test vehicle are presented in Table 2.

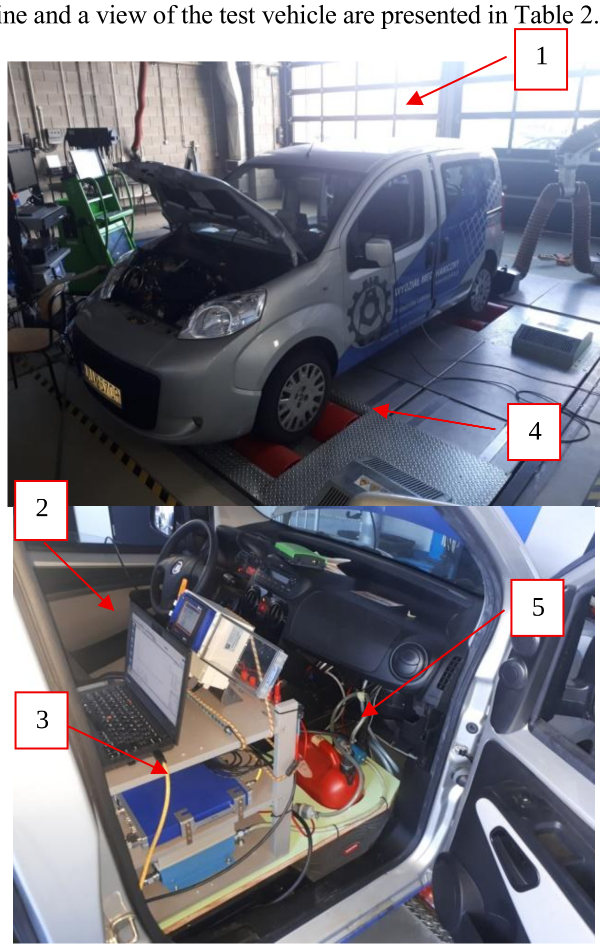


Fig. 1. Research station simulating vehicle motion under traction conditions: 1 – Fiat Qubo test car with 1.3 Multijet engine; 2 – computer with installed AVL IndiCom V2.7 software; 3 – Indimicro 602 engine indicator system; 4 – DF4FS-HLS chassis dynamometer; 5 – additional fuel system tank

An AVL pressure indication system – IndiMicro 602 – was used during the experiments, allowing for real-time acquisition of fast-changing engine parameters. Measurements of the diesel engine's performance under conditions simulating vehicle operation in traction mode were carried out using a DF4FS-HLS chassis dynamometer. The test bench used in the study is shown in Fig. 1.

Table 2. Technical specifications of the 1.3 Multijet engine in the Fiat Qubo test vehicle

Parameter	Characteristics
Name	Fiat Qubo
Production year	2015
Engine capacity	1248 cm^3
Cylinder number and arrangement	4, in line
Cylinder diameter	69.6 mm
Piston stroke	82 mm
Compresione ratio	16.8:1
Max power	55 kW CEE/75 KM CEE
Max tourqe	190 Nm CEE/kgm CEE
Idle speed	850 ±20 rpm
Engine speed at maximum torque	1750 rpm
Fuel injection/fuel supply system	Common Rail/diesel fuel
Exhaust after-treatment systems	EGR, DPF

The operation of an internal combustion engine in a traction vehicle occurs primarily under varying load and speed conditions, which significantly affect the combustion process. To evaluate the characteristics of this process – particularly its dynamics, efficiency, and in-cylinder pressure development – tests were conducted under conditions that reflected real-world vehicle operation. The experimental setup was described in detail in the study [18].

The WLTP (Worldwide Harmonised Light Vehicles Test Procedure) [21], a standardized driving cycle that reflects typical vehicle speed profiles, was used as the basis for the tests. The measurements were conducted on a direct-drive chassis dynamometer, where rolling resistance was simulated using the dynamometer's loading system. Based on the analysis of the WLTP cycle, two representative steady-state engine operating points were selected, corresponding to vehicle speeds of 50 km/h (approx. 1580 rpm) and 90 km/h (approx. 2860 rpm) in fourth gear (direct transmission ratio).

Dynamic conditions were reproduced by measuring the engine's maximum torque during the acceleration phase. This allowed for the analysis of key combustion parameters such as the in-cylinder pressure trace, pressure rise rate, start of injection (SOI), peak pressure location (Pmax), and the complete course of the combustion cycle, enabling the evaluation of combustion efficiency and stability under varying load conditions.

In addition to analyzing ignition delay, the start of fuel injection and the actual start of the combustion process were also evaluated. The start of injection was determined from the injector control signal, while the onset of combustion was identified primarily based on the heat release rate. The ignition initiation point was defined as the crank angle corresponding to approximately 5% of the total fuel mass burned, which represents a characteristic transition from ignition to the main combustion phase. This point coincided with the beginning of intensive heat release and a sudden increase in combustion pressure, corresponding to the actual initiation of combustion reactions of the air-fuel mixture in the cylinder combustion chamber.

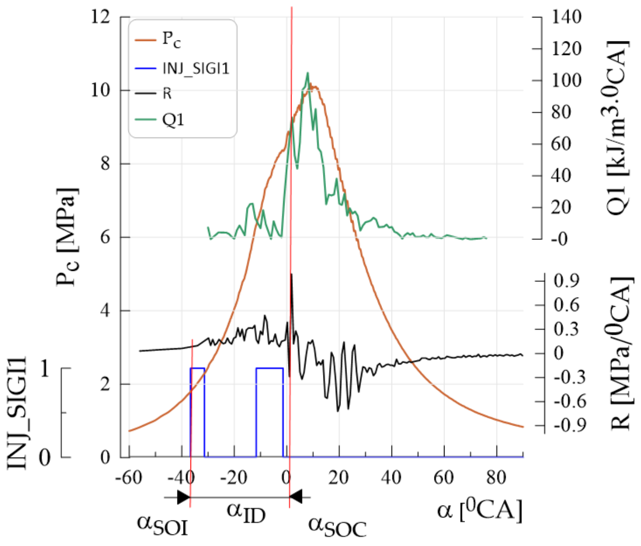


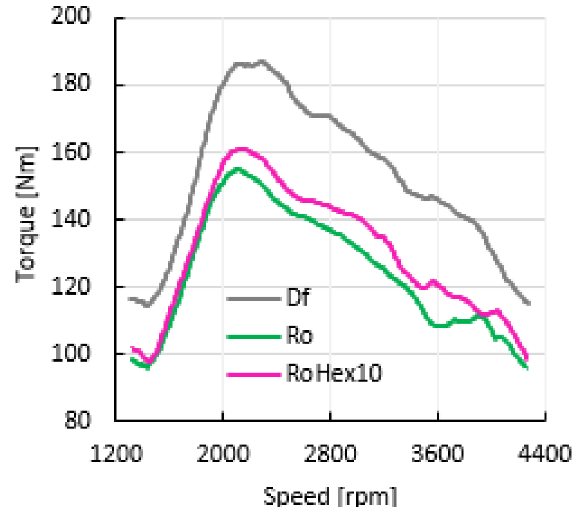
Fig. 3. Graphical interpretation of parameters necessary to calculate the self-ignition angle: α_{SOI} (Start of Injection), α_{SOC} (Start of Combustion) and α_{ID} (Ignition Delay). The course of the main injection and combustion parameters heat release rate (Q1), heat generation curves (I1), combustion pressure (Pc), rate of pressure rise (R) and fuel injector control signal (INJ_SIGI1) are presented

3. Results and discussion

In the experimental phase of the study, a fuel mixture Ro10hex – rapeseed oil with the addition of 10% n-hexane (fuel 3) – was used. The analysis of the physicochemical properties of this mixture showed that further increasing the share of n-hexane did not result in a significant improvement in viscosity; therefore, additional testing in this area was discontinued. The performance of the diesel engine powered by this mixture was compared with results obtained using conventional diesel fuel (Df, fuel 1) and refined rapeseed oil (Ro, fuel 2). The maximum torque and power values were determined based on engine tests (Fig. 4). For diesel fuel (Df), the torque reached 186.9 Nm at 2235 rpm, and the maximum power was 56.3 kW at 3849 rpm. For refined rapeseed oil (Ro), these values were 155.8 Nm at 2102 rpm and 45.7 kW at 3907 rpm, respectively, serving as a reference point for evaluating the performance of the alternative fuel blend. The use of the Ro10hex mixture allowed the engine to achieve a torque of 161.0 Nm at 2116 rpm and a power output of 47.7 kW at 4035 rpm, representing an increase of approximately 3.3% in torque and 4.4% in power compared to Ro. It should be noted, however, that both Ro and Ro10hex were characterized by extended ignition delay times, which is typical of biofuels with lower cetane numbers and lower reactivity. The longer ignition delay influences the timing of the combustion phase and may partly explain the lower torque and power values compared to diesel fuel.

The difference between rapeseed oil-based fuel (Ro) and the Ro10hex blend (rapeseed oil with the addition of 10% n-hexane) in terms of maximum torque and power output is relatively small. However, the course of the torque and power curves indicates that within the useful engine speed range, the difference between the Ro10hex and Ro curves becomes noticeably greater (Fig. 4). This suggests that the addition of n-hexane improves the engine's performance parameters compared to pure rapeseed oil. Such improvement may positively affect vehicle dynamics, especially during acceleration and operation under medium load conditions. These changes may also contribute to better environmental performance of rapeseed-based fuel, as confirmed in the authors' previous studies [19]. Indicated that while the addition of n-hexane slightly increases the heating value of rapeseed oil, it significantly improves key properties such as surface tension and viscosity. These changes can have a meaningful impact on combustion quality and ignition characteristics. Both Ro and Ro10hex produced lower torque and power than diesel fuel, primarily due to their lower chemical reactivity and different combustion profiles. Nevertheless, the analysis confirmed that n-hexane enhances the performance of rapeseed oil as an alternative fuel, improving engine output and vehicle dynamics across the useful speed range.

a)



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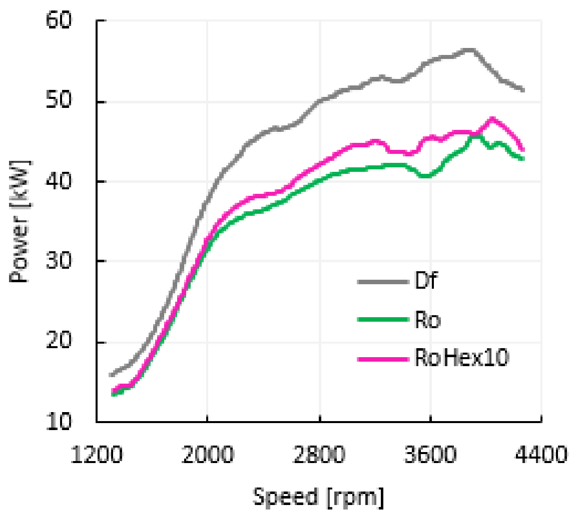


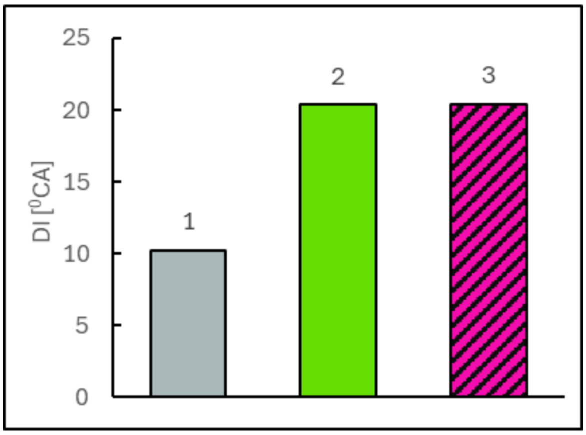
Fig. 4. The values of the Torque (a) and Power (b) values depending on the rotational speed, engine with Diesel supplied with the tested fuels, i.e. Df, Ro and Ro10hex

In the experimental studies that were conducted, clear differences were observed in ignition delay (ID) values depending on the type of fuel used. Diesel fuel (Df), due to its favorable physicochemical properties (high cetane number, high reactivity), exhibited the shortest ignition delay times among all analyzed fuels – 10.25°CA at 50 km/h and 25.97°CA at 90 km/h, respectively. For refined rapeseed oil (Ro), ID values were significantly higher – 20.4°CA and 36.88°CA – confirming the lower chemical reactivity of plant-based biofuels and their slower ignition initiation. The use of the Ro10hex blend (rapeseed oil with 10% n-hexane) resulted in a noticeable, although limited, improvement in ignition properties. At 50 km/h, the ignition delay was 20.42°CA, practically unchanged from Ro. Only at 90 km/h was a reduction observed, down to 35.77°CA, which represents a decrease of approximately 3.0% compared to the base fuel (Fig. 5).

Figure 7 shows the heat release profiles as a function of crank angle for the tested fuels: diesel fuel (Df), refined rapeseed oil (Ro), and the mixture of rapeseed oil with 10% n-hexane (Ro10hex), at two vehicle speeds (50 km/h – part a, and 90 km/h – part b). In part a, corresponding to lower engine load (50 km/h), a distinct kinetic combustion phase is observed for Df, represented by a sharp peak just before top dead center (TDC). This indicates that combustion begins almost immediately after injection. In contrast, Ro and Ro10hex show almost no kinetic combustion phase – their curves rise more slowly and reach maximum values much later, confirming the predominance of diffusion combustion in these biofuels. It is also evident that the Ro10hex curve precedes the Ro curve slightly, suggesting improved fuel volatility due to the addition of n-hexane. In part b (90 km/h), similar trends are observed – the kinetic combustion phase for Df appears clearly before TDC and is the most dynamic. Both Ro and Ro10hex demonstrate delayed and more prolonged heat release, with peak values occurring after TDC. The Ro10hex curve lies between those of Df and Ro, indicating a partial improvement in combustion characteristics compared to pure rapeseed oil. At both engine loads, the addition of n-hexane leads to a slightly earlier onset and greater intensity of heat release, confirming improved ignition and combustion behavior. This has a positive impact on thermal efficiency and contributes to lower peak mechanical loads in the engine. Despite these limitations, it was found that under stationary conditions, Ro and Ro10hex released a greater total amount of heat compared to Df, and the addition of n-hexane increased both the heat released and the degree of fuel combustion. This effect partially compensated for the adverse effects of delayed ignition, resulting in improved engine performance when running on Ro10hex compared to pure rapeseed oil, as confirmed by the analysis of power and torque curves.

The experimental findings are consistent with literature data, which indicate that ignition delay is directly influenced by cetane number, ignition temperature, pressure–temperature conditions in the combustion chamber, and injection dynamics. In light of established models (e.g., Hardenberg and Hase), it is confirmed that fuels with lower chemical reactivity and weaker autoignition properties, such as vegetable-based biofuels, tend to exhibit longer ignition delays. The addition of components that improve volatility and ignition behaviour – such as n-hexane – can enhance their operating properties. However, fully matching the performance of conventional diesel fuel remains a technical challenge. The results of the auto-ignition delay are shown in Fig. 5. The experimental studies revealed clear differences in the values of the maximum rate of pressure rise (Rmax) depending on the type of fuel used. Diesel fuel (Df), due to its favorable physicochemical properties, showed the lowest Rmax values at both tested speeds – 0.308 MPa/°CA at 50 km/h and 0.399 MPa/°CA at 90 km/h – indicating a more controlled and smoother combustion process. This was also influenced by combustion taking place mainly in the kinetic phase (Fig. 7).

a)



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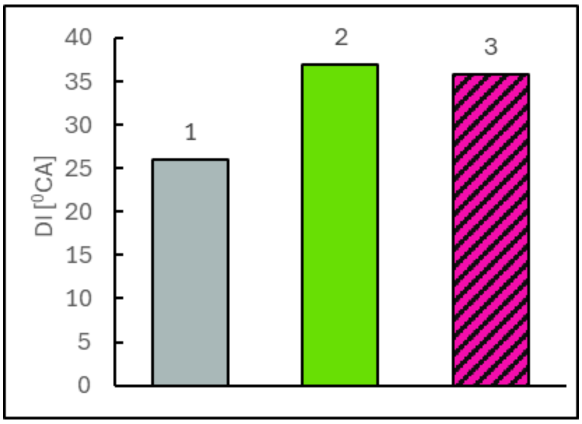
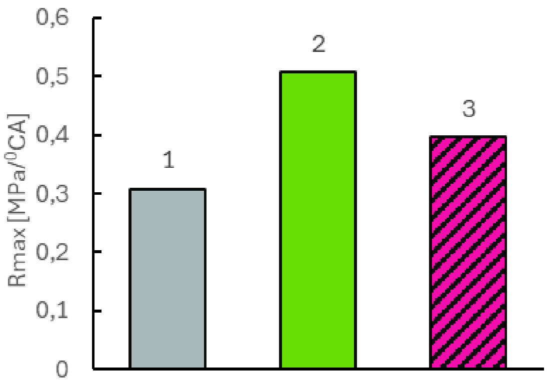


Fig. 5. The values of the ignition delay (ID) for vehicle speeds of 50 km/h (a) and 90 km/h (b) fuelled with the test fuels: Df – diesel fuel (bar 1); Ro – the rapeseed oil (bar 2), Rohex10 – the mixture of rapeseed oil with 10% n-hexane (bar 3)

The torque characteristics obtained during tests on a Dynorace chassis dynamometer (Fig. 4a) show a noticeably faster decrease in engine torque after reaching its maximum value compared to the reference characteristics provided by the manufacturer (Fiat Powertrain Technologies). According to catalogue data and engine-bench measurements (engine dynamometer), the torque of the 1.3 Multijet 55.15 kW engine is approximately 190 Nm in the range of 1500–3000 rpm, whereas in tests on a chassis dynamometer a decrease in engine torque was observed already at approximately 2000 rpm. This difference is typical and has been reported in scientific studies – it results from driveline losses occurring during chassis-dynamometer operation, different engine-cooling conditions, and consequently a different course of the boost-control process.

For refined rapeseed oil (Ro), the rate of pressure rise was significantly higher – 0.507 MPa/°CA at 50 km/h and 0.540 MPa/°CA at 90 km/h – confirming that the combustion of plant-based biofuel is characterized by a more rapid pressure increase and, consequently, greater loading of the piston-crank system. This was due to the delayed start of combustion compared to Df, leading to the dominance of diffusion combustion. The use of the mixture of rapeseed oil with 10% n-hexane (Ro10hex) resulted in a noticeable reduction in the intensity of the pressure rise compared to pure rapeseed oil. At 50 km/h, the R_{max} value was 0.396 MPa/°CA, and at 90 km/h 0.417 MPa/°CA, representing a decrease of approximately 22% and 23%, respectively, compared to pure Ro. This result confirms the positive effect of n-hexane in improving the combustion process of the fuel blend. The ignition delay of these fuels had a decisive influence on these results, as the delay determined the accumulation of the fuel-air mixture prior to the start of combustion and, consequently, the magnitude of the pressure rise. The results of the maximum rate of pressure rise are shown in Fig. 6.

a)



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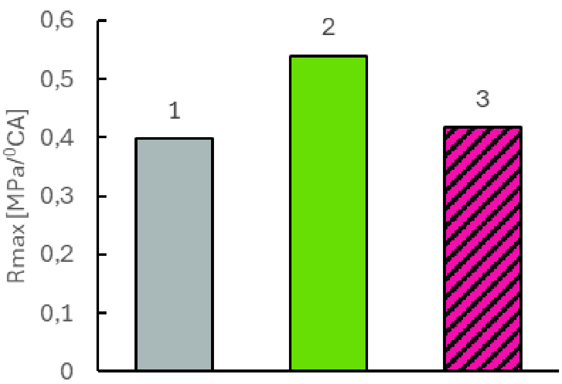
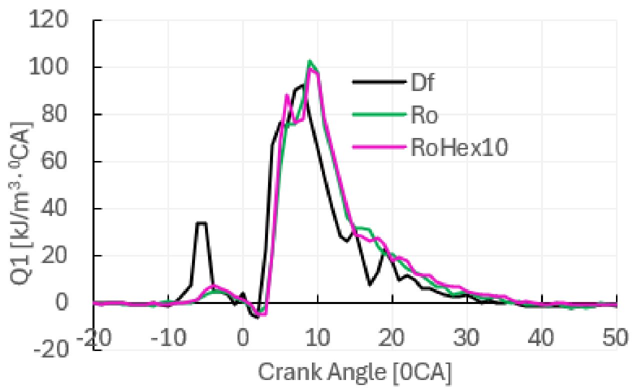


Fig. 6. The values of maximum rate of pressure rise (R_{max}) for vehicle speeds of 50 km/h (a) and 90 km/h (b) fuelled with the test fuels: Df – diesel fuel (bar 1); Ro – the rapeseed oil (bar 2), Rohex10 – the mixture of rapeseed oil with 10% n-hexane (bar 3)

The experimental studies were carried out on a diesel engine whose ECU operated on factory-calibrated control maps designed for diesel fuel supply. The use of alternative fuels (Ro, RoHex10), characterized by lower energy content, required corrections of the control strategy by modifying injection parameters, mainly by extending the injection duration and increasing the fuel dose. This resulted in higher specific fuel consumption while maintaining comparable torque values. In this way, the ECU adapted the combustion process to the so-called “driver wish”, i.e., the driver’s demand, ensuring the required torque on the crankshaft. The physicochemical properties of rapeseed fuels (Ro, RoHex10), different from those of diesel fuel (Df), caused the observed variations in ignition delay (Fig. 5) and significantly influenced the course of the combustion process (Figs. 6 and 7). Consequently, combustion was characterized by a higher share of the diffusion phase and a less stable course compared to diesel fuel. The effect of these differences was reflected in the torque and power curves (Fig. 4). It can therefore be concluded that the EDC system played an essential compensatory role – by automatically adjusting injection parameters to the properties of alternative fuels, it enabled the maintenance of the required operating parameters of the engine despite differences in calorific value and chemical reactivity of the fuels.

a)



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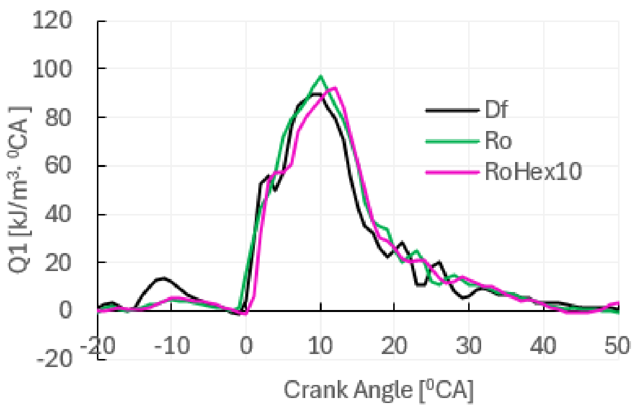

Fig. 7. The values of Heat relase (Q1) for vehicle speeds of 50 km/h (a) and 90 km/h (b) fuelled with the test fuels: Df – diesel fuel; Ro – the rapeseed oil, Ro10hex – the mixture of rapeseed oil with 10% n-hexane

At partial loads, the ECU compensates for the fuel's lower energy content by extending the injection time and increasing the dose, which, within the limits of adaptation, allows comparable torque to be maintained. However, under maximum power demand (full utilization of the "driver wish" function), the ECU already provides the maximum injection dose specified in the control maps, taking into account boost pressure, exhaust gas temperature, and smoke limits. When using fuel with a lower calorific value, the maximum dose corresponds to a lower amount of chemical energy, which makes it impossible to reach the nominal engine power. This results in reduced maximum speed, slower acceleration, and potentially increased exhaust smoke due to less efficient combustion of the larger fuel dose. Ultimately, at partial loads, the differences are masked by ECU adaptation, while under full load, the lack of correction margin causes a drop in maximum power proportional to the energy deficit of the fuel

4. Conclusions

Based on the experimental investigations conducted, it was found that the use of the Ro10hex blend – consisting of rapeseed oil with a 10% addition of n-hexane – leads to a relative improvement in the operational parameters of a diesel engine compared to pure rapeseed oil. An increase in maximum power by approximately 4.4% and torque by 3.3% was observed, indicating the positive effect of n-hexane on the combustion process. Additionally, at higher engine speeds, a slight reduction in ignition delay (ID) was recorded, suggesting an improvement in the autoignition properties of the fuel.

However, it should be emphasized that despite these beneficial effects, both pure rapeseed oil (Ro) and its mixture with n-hexane (Ro10hex) still showed significantly inferior performance parameters compared to conventional diesel fuel. This was particularly evident in the ignition delay time, which for Ro and Ro10hex was considerably longer. The analysis indicates that this is due to a different combustion mechanism: in the case of diesel fuel, combustion begins with a clearly defined kinetic phase that occurs before top dead center (TDC), whereas for Ro and Ro10hex the combustion process is predominantly diffusive and begins only after the fuel has been fully atomized and mixed with air. As a result, the peak combustion pressure occurs significantly after TDC, which negatively affects the thermodynamic efficiency of the cycle. This phenomenon was also accompanied by a more abrupt rise in combustion pressure, particularly noticeable for pure rapeseed oil, which led to increased mechanical loading of the piston-crank system.

From a physicochemical perspective, n-hexane clearly improves the practical usability of rapeseed oil as a fuel – particularly by reducing its viscosity – bringing its physical properties closer to those of diesel fuel (although not reaching its nominal values). However, the relatively high ignition temperature of rapeseed oil means that the onset of combustion for the Ro10hex mixture remains significantly delayed compared to diesel fuel. Therefore, it is justified to continue research aimed at further modifying the composition of the blend by incorporating additives that improve combustion properties, especially those that lower the ignition temperature and reduce ignition delay time, ultimately aligning the combustion characteristics more closely with those of conventional diesel fuels.

Nomenclature

CI	compression ignition
DI	direct injection
Hex	n-heksan
ID	ignition delay
Ro	Rapeseed oil
TDC	top dead center

Bibliography

[1] Al-Shahrani AS. Theoretical analysis of ignition in die sel engines. Journal of Power and Energy Engineering. 2017; 5(6). https://doi.org/10.4236/jpee.2017.56001

[2] Baczewski K, Kałdoński T. Paliwa do silników o zapłonie samoczynnym. WKŁ. Warszawa 2008.

[3] Bayindirli C. Celik M. Investigation of combustion and emission characteristics of n-hexane and n-hexadecane additives in diesel fuel. J Mech Sci Tech. 2019;33:1937-1946. https://doi.org/10.1007/s12206-019-0344-8

[4] Cheng C, Faurskov Cordtz R, Dyhr Pedersen T, Winther K, Langballe Førby N, Schramm J. Investiga-tion of combustion characteristics, physical and chemical ignition delay of methanol fuel in a heavy-duty turbo-charged compression ignition engine. Fuel. 2023;348:128536. https://doi.org/10.1016/j.fuel.2023.128536.

[5] Cisek J, Borowski A, Całkowska J, Wichary Ł. Effect of nitrON® cetane-detergent additive to B7 fuel on energy parameters and exhaust gas composition of a 6Dg locomotive with a Caterpillar C27 engine. Combustion Engines. 2021; 186(3):51-58. https://doi.org/10.19206/CE-140113

[6] Crua C, Kennaird DA, Sazhin SS, Heikai MR, Gold MR. Diesel autoignition at elevated in-cylinder pressures. Int J Engine Res. 2004;5(4):365-374. https://doi.org/10.1243/146808704323224259

[7] Górski K, Sander P, Longwic R. The assessment of ecological parameters of diesel engine supplied with mixtures of canola oil with n-hexane. IOP Conf Ser Mater Sci Eng. 2018;421(4): 042025. https://doi.org/10.1088/1757-899X/421/4/042025

[8] Hardenberg HO, Hase FW. An empirical formula for computing the pressure rise delay of a fuel from its cetane number and from the relevant parameters of direct-injection diesel engines. SAE Technical Paper. 790493. 1979. https://doi.org/10.4271/790493

[9] Hardenberg HO, Hase FW. Empirical formula for computing the pressure rise delay of a fuel from its cetane number and from the relevant parameters of direct-injection diesel engines. SAE Preprint 1979.

[10] Internal combustion engine fundamentals. Choice Reviews Online. 1988;26. https://doi.org/10.5860/choice.26-0943

[11] Jaroń A, Borucka A, Sobecki G. Assessment of the possibility of using nanomaterials as fuel additives in combustion engines. Combustion Engines. 2022;189. https://doi.org/10.19206/CE-143824

[12] Jo S, Park S, Kim HJ, Lee JT. Combustion improvement and emission reduction through control of ethanol ratio and intake air temperature in reactivity controlled compression ignition combustion engine. Appl Energy. 2019;250:1418-1431. https://doi.org/10.1016/j.apenergy.2019.05.012

[13] Kowalewicz A. Application of biofuels to compression ignition engines. Mech Mech Eng. 2005;9:69-90.

[14] Kowalewicz A, Lotko W. Alternative fuels for automotive I.C. engines and some results of application to S.I. and C.I. engine. Proceedings of the Third Asia-Pacific International Symposium on Combustion and Energy Utilization n.d.. 2005; 1. https://doi.org/10.1243/095440705X6399

[15] Kryshtopa S, Górski K, Longwic R, Smigins R, Kryshtopa L. Increasing parameters of diesel engines by their transformation for methanol conversion products. Energies. 2021; 14(6):1710. https://doi.org/10.3390/en14061710

[16] Krzemiński A, Jaworski A, Kuszewski H, Woś P. A comparative study on selected physical properties of diesel–ethanol–dodecanol blends. Combustion Engines. 2024;196(1):99-105. https://doi.org/10.19206/CE-170267

[17] Longwic R. Diesel engine operations with alternative fuels. Boca Raton: CRC Press. 2024. https://doi.org/10.1201/9781003466291

[18] Longwic R, Sander P. The characteristics of the combustion process occurring under real operating conditions of traction. IOP Conf Ser Mater Sci Eng. 2016;148:012071. https://doi.org/10.1088/1757-899X/148/1/012071

[19] Longwic R, Sander P, Jańczuk B, Zdziennicka A, Szymczyk K. Modification of canola oil physicochemical properties by hexane and ethanol with regards of its application in diesel engine. Energies. 2021;14(15):4469. https://doi.org/10.3390/en14154469

[20] Longwic R, Sander P, Lotko W, Gorski K, Janczuk B, Zdziennicka A et al. Self-ignition of rapeseed and n-hexane mixtures in diesel engine. Przem Chem. 2020;99:206-210. https://doi.org/doi.org/10.15199/62.2020.2.3

[21] Longwic R, Sander P, Zdziennicka A, Szymczyk K, Jańczuk B. Combustion process of canola oil and n-hexane mixtures in dynamic diesel engine operating conditions. Appl Sci. 2020; 421(4):042025. https://doi.org/10.3390/app10010080

[22] Lotko W. Studium zastosowań paliw alternatywnych do silników o zapłonie samoczynnym. Radom 1999.

[23] Lotko W. Self-ignition delay and control parameters of diesel engines for different vehicle feeding systems and different fuels. Adv Sci Technol Res J. 2021;15(1):245-254. https://doi.org/10.12913/22998624/132474

[24] Martins J, Brito FP. Alternative fuels for internal combustion engines. Energies. 2020;13. https://doi.org/10.3390/en13164086

[25] Mohammed O, Suleiman E. The history of the internal combustion engine. Annuls of the Faculty of Engineeeing Hunedoara. 2017;3.

[26] Pirouzfar V, Zarringhalam Moghaddam A, Mirza B. Physicochemical properties and combustion performance of gas oil-fuel additives. J Energy Resour Technol. 2012;134(4):041101. https://doi.org/10.1115/1.4007483

[27] Polska norma PN-EN 590:2013. n.d.

[28] Rațiu S. The history of the internal combustion engine. Annals of the Faculty of Engineering Hunedoara. 2003;3.

[29] Rosseel E, Sierens R. The physical and the chemical part of the ignition delay in diesel engines. SAE Technical Paper. 961123. 1996. https://doi.org/10.4271/961123

[30] Singh AP, Mustafi NN, Sharma YC, Agarwal AK. Introduction to alternative fuels and their utilization strategies in internal combustion engines. Energy, Environment, and Sustainability. 2020. https://doi.org/10.1007/978-981-15-0418-1_1

[31] Sun B, Zhao S, Zhai Y, Liu Q, Wu G, Wu H. Effect of fuel physicochemical properties on spray and particulate emissions. ACS Omega. 2022;7(48):44251-44265. https://doi.org/10.1021/acsomega.2c05747

[32] Wajand JA, Wajand JT. Tłokowe silniki spalinowe średnio i szybkoobrotowe. Warszawa: WNT 2005.

[33] Zabłocki M. Wtrysk i spalanie w silnikach wysokoprężnych. Wydawnictwo Komunikacji i Łączności. Warszawa 1976.

[34] Zaharin MSM, Abdullah NR, Najafi G, Sharudin H, Yusaf T. Effects of physicochemical properties of biodiesel fuel blends with alcohol on diesel engine performance and exhaust emissions: a review. Renew Sustain Energy Rev. 2017;79:475-493. https://doi.org/10.1016/j.rser.2017.05.035.

[35] Zdziennicka A, Szymczyk K, Jańczuk B, Longwic R, Sander P. Adhesion of canola and diesel oils to some parts of diesel engine in the light of surface tension components and parameters of these substrates. Int J Adhes Adhes. 2015;60:23-30. https://doi.org/10.1016/j.ijadhadh.2015.03.001

[36] Zdziennicka A, Szymczyk K, Jańczuk B, Longwic R, Sander P. Surface, volumetric, and wetting properties of oleic, linoleic, and linolenic acids with regards to application of canola oil in diesel engines. Appl Sci. 2019;9(17):3445. https://doi.org/10.3390/app9173445

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