

## Article

# Experimental Determination of Combustion Process Parameters of Biodiesel Fuel Made from Waste Grape Seeds

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## Abstract

Biodiesel fuel produced through transesterification is mainly used in blends with conventional diesel fuel (D100). The analysis of the combustion process parameters for each specific biodiesel fuel represents the basis for a rational approach to the utilization of available motor fuel quantities. In this study, the differential and cumulative heat release laws during the combustion of D100 and blends of biodiesel fuel made from waste grape seed oil and D100 were analyzed. In addition, the engine efficiency and economy for the cases of using the aforementioned fuels were analyzed. The tests were conducted on a single-cylinder, air-cooled diesel engine with direct fuel injection. The engine testing was conducted for two engine loads; that for which the brake was a mean effective pressure of 4.2 bar, and for the full load, that for the brake was a mean effective pressure of 5.6 bar at engine speeds of 1635 rpm, 1937 rpm, and 2239 rpm. All experimental work was conducted for conventional diesel fuel D100 and for biodiesel diesel blends B7 and B14. The combustion rates of D100, a blend containing 7% of biodiesel by volume (B7), and a blend containing 14% of biodiesel by volume (B14) were examined. However, the higher combustion rate of the B14 blend, particularly during the combustion of the first 50% of the fuel mass per cycle, could have a positive impact on the fuel economy of the working cycle and the brake thermal efficiency (BTE). The maximum heat release rates for D100, B7, and B14 at full load and an engine speed of 2239 rpm are 115.65 J/deg, 148.01 J/deg, and 152.99 J/deg, respectively. At full load and engine speeds of 1635 rpm and 2239 rpm, the brake thermal efficiencies (BTEs) for D100, B7, and B14 were 0.301, 0.285, and 0.296 and 0.281, 0.273, and 0.277, respectively. Under other tests, the highest BTE was observed for the B14 blend. Therefore, from the perspective of brake thermal efficiency (BTE), the most favorable blend for application is B14.

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**Keywords:** biodiesel fuel; waste grape seeds; combustion; heat release

## 1. Introduction

Biodiesel fuel is the most widely used biofuel in road transport [1]. Although biodiesel is a high-quality motor fuel, it can differ significantly from D100 in terms of its physico-chemical properties [2]. Differences in these properties affect combustion process parameters and, consequently, the efficiency and environmental performance of the

engine [3]. Furthermore, the differences between the physico-chemical characteristics of biodiesel fuels produced by different methods or even by the same method but from different feedstocks should not be overlooked.

Several biodiesel production processes have found commercial applications to date. The most common type of biodiesel on the market is produced by the transesterification process using alcohol [4].

Biodiesel fuel obtained through the transesterification process belongs to the traditional types of biofuel, that is, first or second generation biofuels. Chemically, it is a methyl ester of fatty acids and is known as FAME (Fatty Acid Methyl Ester) biodiesel or, less commonly, as FAEE (Fatty Acid Ethyl Ester) biodiesel. Biodiesel fuel is usually produced from oils derived from purpose-grown crops, but the production of biodiesel from waste generated during food and beverage production processes is becoming increasingly relevant [1].

Given the wide range of feedstocks that can be used to obtain oil for biodiesel production, the combustion process characteristics can vary significantly, which in turn affects the engine's energy and environmental performance indicators. Differences in combustion parameters, compared with those observed when using D100, are especially pronounced when neat (unblended) biodiesel fuels are used [5,6].

Today, biodiesel fuels are typically used in blends with D100. For many biodiesel-diesel blends, less-favorable engine energy performance indicators are obtained compared to the use of D100.

### *1.1. Objective and Scope of This Research*

For the rational use of blends, it is necessary to understand the characteristics of biodiesel fuels and to investigate the influence of the biodiesel content in the blend on the combustion process, the efficiency of the working cycle, and the BTE. The aim of this study is to examine the effect of the proportion of biodiesel produced from waste grape seed oil in a blend with D100 on combustion process parameters and engine efficiency. This paper will analyze heat release rates and engine efficiency under different operating conditions for D100 and B7 and B14 blends.

### *1.2. Review of Previous Research on Combustion Processes and Engine Efficiency When Using Blends*

Previous studies have mostly focused on first- and second-generation biodiesel fuels, particularly in blends with D100. In most cases, these blends have proven to be quality fuels from the standpoint of combustion processes and engine performance. However, engines often exhibit slightly less favorable performance compared to operation with D100.

Increasing the proportion of biodiesel derived from flaxseed oil in a blend with D100 reduces the BTE of the engine, and for all such blends, the BTE will be lower than when burning neat diesel fuel [7].

One way to influence the operating characteristics of an engine powered by neat biodiesel or a biodiesel-diesel blend is by adding a certain amount of water to the mixture. By adding 10% water to a blend of biodiesel produced from soybean oil and D100, it is possible to reduce the brake-specific fuel consumption (BSFC) [6].

Considering the physico-chemical properties of biodiesel fuel, it is interesting that under certain engine operating conditions, using an optimal proportion of biodiesel obtained from corn oil in a blend with D100 can yield more favorable engine performance, such as higher effective power output and lower BSFC, than in the case of using neat diesel fuel [8].

Biodiesel fuel derived from cultivated microalgae is a newer-generation biofuel belonging to the third generation [9]. Blends obtained by mixing biodiesel from cultivated microalgae and D100 burn at a higher rate in diesel engines than D100 [10].

Plant-based waste generated during various stages of plant processing, waste cooking oil, and animal fats and oils are increasingly being used as feedstocks for biodiesel production, mainly due to their low cost. Increasing the proportion of biodiesel derived from waste animal fats in blends with D100 results in lower peak cylinder pressures, while the combustion process begins earlier for these blends compared to neat diesel fuel [11].

Using a blend of D100 and biodiesel produced from waste fish oil (FB25) has yielded superior performance compared to a blend of D100 and biodiesel from waste cooking oil (CB25) [12]. However, when operating with both blends, the engine produced less power and had a higher BSFC than with D100 due to the lower heating value of the blends.

Coffee husks also represent a potential feedstock for obtaining oil used in biodiesel production. By adding small amounts, e.g., 10%, of biodiesel produced from coffee husk oil to D100, blends can be obtained that can be used without major engine modifications. The engine performance with these blends is very similar to that achieved with neat diesel fuel, while the combustion process characteristics are even more favorable [13].

One of the promising feedstocks for biodiesel production that arises as a by-product of food and beverage manufacturing is waste grape seeds.

The five-year average wine production volume in the European Union (EU) for the period from 2019 to 2023 was 156 million hectoliters (mhl), while in 2023 and 2024, it amounted to 144 mhl and 139 mhl, respectively [14]. Global wine production in 2023 was around 236 mhl. According to the literature data, one kilogram of grapes yields between 0.6 and 0.75 L of wine. During wine production, a large quantity of waste is generated, posing an environmental burden. Grape seeds, skins, and stems represent solid waste in the wine industry. The mass fractions of grape seeds, skins, and stems in total grape mass are approximately 5%, 7%, and 5%, respectively [15,16]. Depending on the grape variety, the total mass of seeds, skins, and stems can reach as much as 20–30% of the processed grape mass [17].

Given the enormous quantities of grapes processed annually into wine in the EU and worldwide, it is evident that millions of tons of waste are generated every year during wine production, posing both economic and environmental challenges. For this reason, in recent years, such waste has been recognized as a suitable feedstock for biofuel production, specifically biodiesel from grape seeds and ethanol from grape skins and stems.

Combustion of neat biodiesel fuel obtained from waste grape seeds in a single-cylinder, direct-injection diesel engine results in lower BTE compared to D100. At full load and an engine speed of 1500 rpm, the maximum BTEs are 32.34% for D100 and 30.28% for pure biodiesel produced from waste grape seeds [18].

This difference in BTE can be considered negligible, given that the fuel used is neat biofuel derived from waste.

When using a blend of D100 and biodiesel from waste grape seeds (B5), favorable results are achieved in terms of combustion process parameters and energy efficiency, which differ only slightly from those obtained with D100. When running on D100, an engine achieved higher BTE than with the B5 blend, with the maximum difference being only 4.6% at full load [19]. Due to the lower heating value of the blend, engine power was also slightly reduced when using the blend compared to D100, with a maximum difference of 4.3% at full load.

In addition to improvements in fuels and engines, the development of engine lubricants and the application of nano-additives in lubricants are also of current interest [20]. The goal is to achieve a higher brake thermal efficiency through the combination of different measures.

### 1.3. Contributions

The applied biodiesel fuel is promising in terms of production cost, but in terms of heating value (33.965 kJ/kg) and kinematic viscosity (5.15 mm<sup>2</sup>/s), it significantly differs from D100 as well as from biodiesel produced from waste grape seeds used in previous studies. To keep the engine power close to that of D100, blends with low biodiesel content (B7 and B14) were used. The aim of this research is to determine:

- Whether an engine can operate stably with the applied blends at medium-high load (BMEP: 4.2 bar) and full load (BMEP: 5.6 bar);
- Whether biodiesel fuel (blends) significantly deviates from D100 in terms of combustion process parameters;
- Which fuel provides the highest engine efficiency and economy;
- Whether the blends are acceptable in terms of mechanical loads and engine noise, where combustion process parameters will be used as indicators for evaluation.

### 1.4. Organization of This Paper

This paper consists of four parts. The first part provides basic information on the possibility of using biodiesel fuel in internal combustion engines and a review of previous research on combustion process characteristics and the efficiency of engines powered by biodiesel fuels and blends of biodiesel and D100. The second part describes the research methodology. The third section presents the research results and a discussion of the obtained results. The fourth section provides concluding remarks and recommendations for future research directions.

## 2. Methodology

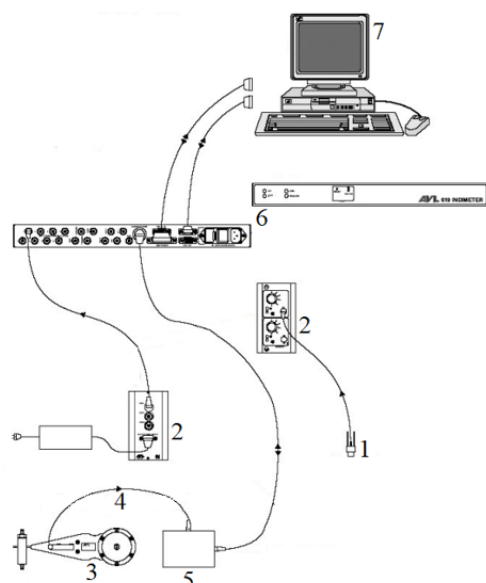
The combustion process of biodiesel–diesel fuel blends was investigated using an experimental single-cylinder, air-cooled, direct-injection diesel engine; the engine specs are given in Table 1. The engine was equipped with a mechanical fuel injection system and had a fixed injection timing. For the purpose of studying the combustion of biodiesel–diesel fuel blends, an injection timing of 21 deg before top dead center (BTDC) was selected, in that it was not changed, in order to investigate only the blend influence. The injection timing was selected based on achieving the most stable (smoothest) engine operation with the B7 blend, which represented a compromise solution for the three tested fuels and did not significantly deviate from the injection timing that is optimal for D100.

**Table 1.** Test engine characteristics.

Name	Description
Model	DMB
Power/engine speed, kW/rpm	7.3/2500
Torque/engine speed, Nm/rpm	28/1700
Engine cooling	Air cooled engine
Engine displacement, cm <sup>3</sup>	454
Number of cylinders	1
Piston diameter/piston stroke, mm	85/80
Compression ratio, -	17.5:1
Parameters of engine valve mechanism	Intake valve opens 16 deg BTDC
	Intake valve closes 40 deg ABDC
	Exhaust valve opens 40 deg BBDC
	Exhaust valve closes 16 deg ATDC
	Valve overlap 32 deg

### 2.1. Experimental Procedure

In order to determine the parameters of the combustion process, it is necessary to know the variation in the in-cylinder pressure throughout the engine cycle. The cylinder pressure is measured using a pressure indication measurement system, shown in Figure 1. The measurement system consists of both analog and digital components. The analog part includes a piezoelectric pressure sensor and a signal-conditioning amplifier. The digital part consists of a crankshaft angle optical marker and TDC position sensor, an optical transmitter, and a signal amplifier. The pressure sensor was calibrated before the experimental work, with so-called dead weight calibrated, where the calibrating machine uses special precise weights to simulate the specific pressure. That is, on the used calibrating machine, 1 kg is equal to 1 bar. The determined calibration coefficient amounted to 9.97 bar/V.



**Figure 1.** The measuring chain for the cylinder pressure: 1—pressure sensor AVL QC32D, 2—signal amplifier KISTLER 5007, 3—angle encoder, 4—optical transmitter, 5—signal multiplier AVL 365CC, 6—acquisition system AVL Indimeter 619, and 7—computer with acquisition software AVL Indi-Com 1.2.

The engine load was taken on the basis of the engine characteristics. In order to not take the engine speeds arbitrarily, for this research, the referent engine speeds were calculated according to the procedure defined by the European Stationary Cycle (ESC) [21]. The reason why the engine speeds were calculated according to the standard testing procedures is because in the future, this would allow easier results in comparison to other authors. However, considering the fact that the engine we used was an engine used to power small agricultural machines, where typical working regimes are the regimes at medium-high loads and high loads, only these regimes were taken to analyze the combustion process, that is, the experimental work, and the results will be presented only for the medium-high load (BMEP: 4.2 bar) regimes and full-load regimes (BMEP: 5.6 bar). The experimental engine was tested using blends of biodiesel produced from waste grape seed oil and D100 (B7 and B14), as well as with neat conventional diesel fuel (D100).

The measurement resolution in this research was 50 cycles per measurement, and each measurement was conducted three times, after which the results were averaged. As for how the tests were conducted according to the regimes defined by the ESC, the measurement procedure is quite simple. First, it is necessary to adjust the engine speed and engine load. As the parameter of successful measurement, the engine work stability is

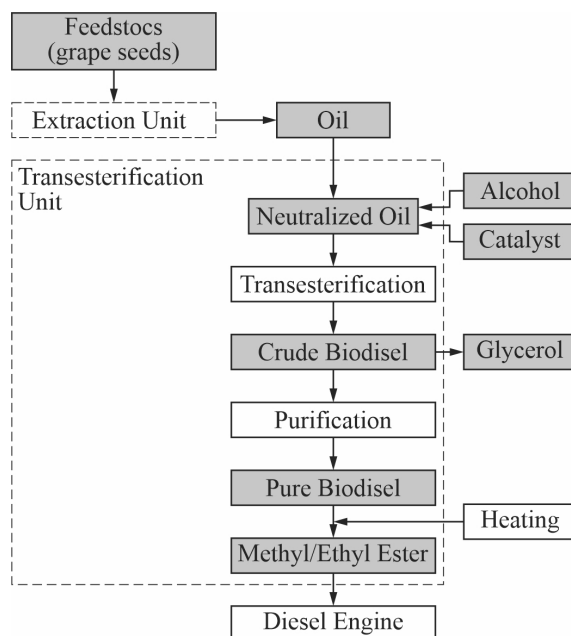
usually taken during the measurement and can be graded according to the engine speed fluctuations during the measurement. In Table 2 are given the minimal, mean and maximal rpm values, as well as the standard deviations, for the case of the 4.2 bar BMEP.

**Table 2.** Engine speed statistics.

	Minimal Engine Speed, rpm	Mean Engine Speed, rpm	Maximal Engine Speed, rpm	Standard Deviation, rpm
D100	1621	1630	1636	4.1
	1939	1949	1961	5
	2230	2236	2240	2.6
B7	1632	1641	1650	5.3
	1922	1938	1955	10.5
	2237	2245	2255	5.2
B14	1638	1642	1646	1.9
	1935	1952	1956	4.7
	2224	2236	2255	8.4

## 2.2. Biodiesel Production and Characteristics

The biodiesel fuel was obtained through the transesterification process of waste grape seed oil. Transesterification is based on the reaction of higher unsaturated fatty acids and alcohol (methanol— $\text{CH}_3\text{OH}$ ) in the presence of alkaline catalysts (potassium hydroxide— $\text{KOH}$ ). Each triglyceride molecule gradually releases three fatty acid molecules from its structure; they react with the alcohol to form three molecules of fatty acid alkyl esters, e.g., FAME, and one molecule of glycerol as a by-product. Figure 2 shows a simple processing flow diagram for biodiesel production. The characteristics of the biodiesel transesterification process used in this study are presented in Table 3.



**Figure 2.** Processing flow diagram for biodiesel production.

**Table 3.** Characteristics of the transesterification process.

Name	Value
Production method	Batch process
The molar ratio of oil:methanol	6:1

Catalyst	KOH
Catalyst amount	1%
Reaction temperature	63 °C
Reaction time	60 min
Pressure	Atmospheric
Post-treatment of FAME	Distillation of excess methanol, washing with distilled water, filtration.

The properties of the biodiesel fuel are presented in Table 4. The D100 used for preparing the blends and for combustion as neat diesel complies with the SRPS EN 590:2022 standard [22].

**Table 4.** Characteristics of the biodiesel fuel.

Property	Unit	Measured Value	EN-14214 Limit	Meth. of Testing
Density (at 15 °C)	kg/m <sup>3</sup>	885	860–900	SRPS EN ISO 3675 [23]
Kinematic viscosity (at 40 °C)	mm <sup>2</sup> /s	5.15	3.5–5	SRPS EN ISO 3104 [24]
Flash point	°C	>185	>101	SRPS EN ISO 2719/A1 [25] SRPS EN ISO 3679 [26]
Lower heating value	MJ/kg	33.965	-	ASTM D240 [27]
Ester content	%(m/m)	97.3	>96.5	SRPS EN 14103 [28]

### 3. Results and Discussion

The parameters of the combustion process, such as the combustion rates in individual phases of the combustion process and the amount of released heat, are most easily analyzed using diagrams of the differential and cumulative heat release laws. The differential heat release law is determined from Equation (1) [29]:

$$\frac{dQ}{d\alpha} = \frac{K}{n-1} (n \cdot p_i \cdot (V_{i+1} - V_{i-1}) + V_i (p_{i+1} - p_{i-1})), \quad (1)$$

where

$\frac{dQ}{d\alpha}$ , J/deg—heat release rate (HRR);

$\alpha$ , deg—crankshaft angle (CA);

$K$ —coefficient with a value of 100 when the volume is expressed in dm<sup>3</sup> and the pressure in bar units;

$n$ —polytropic expansion exponent;

$p_i$ , bar—instantaneous value of pressure;

$V_{i+1}-V_{i-1}$ , dm<sup>3</sup>—change in volume;

$V_i$ , dm<sup>3</sup>—instantaneous value of volume;

$p_{i+1}-p_{i-1}$ , bar—change in pressure.

The exponent of the polytropic expansion is determined from Equation (2):

$$n = \frac{\log\left(\frac{p_1}{p_2}\right)}{\log\left(\frac{V_1}{V_2}\right)}. \quad (2)$$

Based on Equation (2) and the variation in the in-cylinder pressure obtained from the experimental research, the values of the polytropic expansion exponent were calculated for D100 and the blends:

- $n_D = 1.31$ —for D100;
- $n_{B7} = 1.27$ —for the B7 blend;
- $n_{B14} = 1.28$ —for the B14 blend.

### 3.1. Results for Medium-High Load Regimes

Table 5 summarizes the parameters of the differential and cumulative heat release laws, including the maximum combustion rates for the blends and D100; the crank angle positions of the maximum combustion rates relative to TDC; and the combustion progress at 5%, 10%, 50%, and 90% of the total fuel injected per cycle. The differential and cumulative heat release laws for the tests conducted at a 4.2 bar BMEP are graphically illustrated in Figure 3.

**Table 5.** Differential and cumulative heat release parameters for the 4.2 bar BMEP.

Parameter	Value/Description		
	D100	B7	B14
Engine speed, rpm		1635	
Maximum heat release rate, J/deg	96.78	132.41	145.31
Position of the maximum heat release rate relative to TDC, deg	0	0	1
Crank angle for 5% fuel burned, deg	-1.81	-1.98	-1.12
Crank angle for 10% fuel burned, deg	-1.14	-1.24	-0.38
Crank angle for 50% fuel burned, deg	7.07	4.93	5.11
Crank angle for 90% fuel burned, deg	22.76	19.98	20.55
Engine speed, rpm		1937	
Maximum heat release rate, J/deg	106.56	119.25	125.37
Position of the maximum heat release rate relative to TDC, deg	2	2	2
Crank angle for 5% fuel burned, deg	0.23	-0.55	0
Crank angle for 10% fuel burned, deg	1.08	0.42	0.86
Crank angle for 50% fuel burned, deg	8.75	7.18	7.25
Crank angle for 90% fuel burned, deg	25.27	21.76	22.47
Engine speed, rpm		2239	
Maximum heat release rate, J/deg	114.41	149.22	142.88
Position of the maximum heat release rate relative to TDC, deg	5	6	6
Crank angle for 5% fuel burned, deg	2.35	2.89	2.50
Crank angle for 10% fuel burned, deg	3.24	3.77	3.54
Crank angle for 50% fuel burned, deg	9.76	8.86	8.89
Crank angle for 90% fuel burned, deg	27.99	25.62	26.25

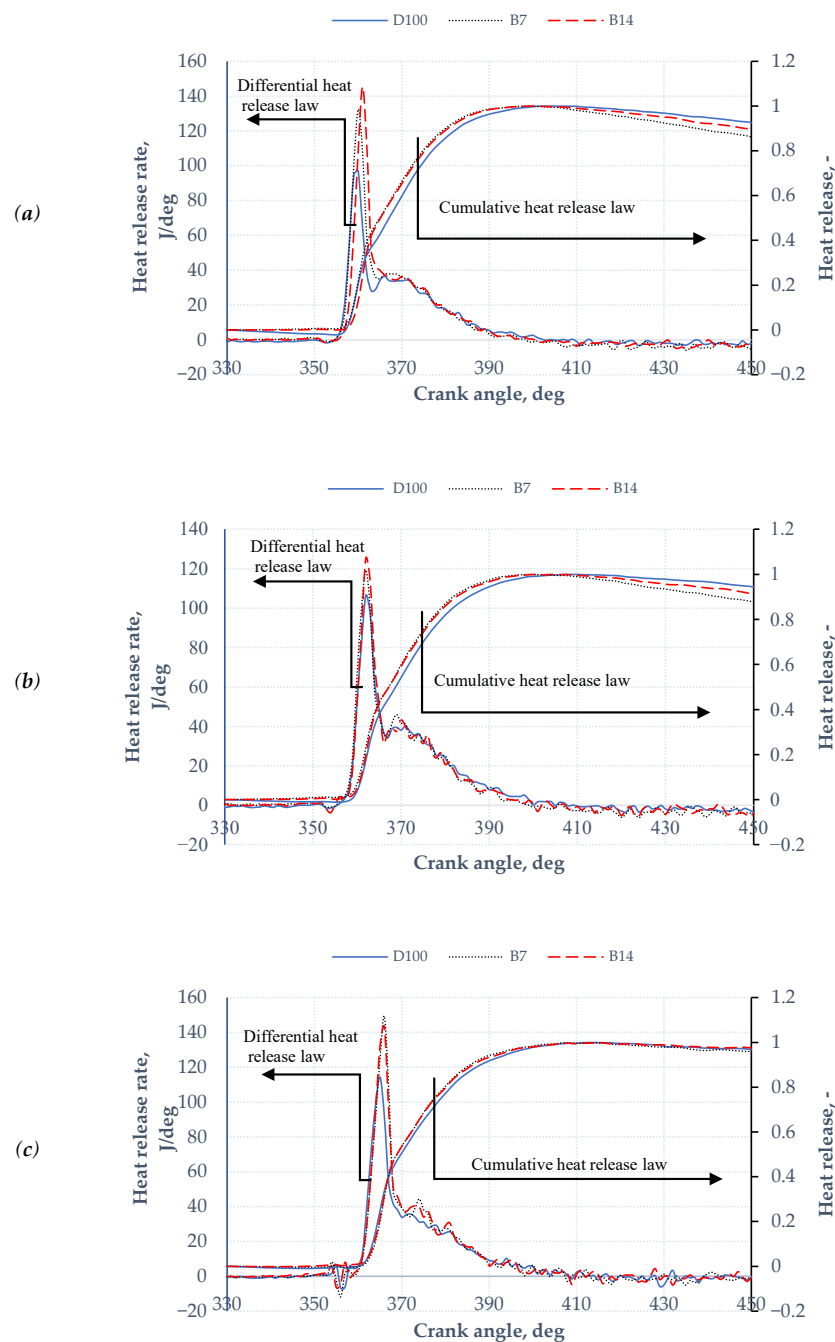
The heat release characteristics during the combustion process influence not only the engine's energy efficiency but also the mechanical loading of its components and the noise level. Rapid heat release and high peak heat release rates lead to steep pressure rises and elevated peak pressures within the engine cylinder, resulting in increased mechanical stresses on engine parts and a progressive increase in noise levels. The timing of the maximum heat release rate is also a critical parameter affecting both mechanical loading and engine noise.

It can also be observed that the blends exhibit higher peak heat release rates compared to D100 (Figure 3). For the engine speeds of 1635 rpm and 1937 rpm and the 4.2 bar BMEP, blend B14 shows the highest peak heat release rate, while at an engine speed of 2239 rpm and the 4.2 bar BMEP, blend B7 reaches the maximum. The higher peak heat release rates of the blends compared to the D100 are a consequence of the longer ignition delay period, caused by the higher density and viscosity of the blends. The difference in peak heat release rates between the blends decreases with increasing engine speed, reaching only 4.2% at the maximum engine speed.

The B7 blend, at 2239 rpm and 4.2 bar BMEP, achieves a higher maximum heat release rate compared to B14 due to the dynamics of the injection system. The injection system of

the tested engine is mechanical, and therefore the injection process directly depends on the engine speed.

Taking into consideration the timing of the peak heat release rate, it can be observed that the blends generally reached their maximum heat release rates later than D100 at 1635 rpm and the 4.2 bar BMEP, or, in some cases, simultaneously, which was the case for the B7 blend at engine speeds of 1635 rpm and 1937 rpm and the 4.2 bar BMEP. For all tested fuels, the timing of the peak heat release rate shifted further from TDC as the engine speed increased, meaning that the maximum combustion rate occurred later.



**Figure 3.** Differential and cumulative heat release laws: (a) 1635 rpm and 4.2 bar BMEP, (b) 1937 rpm and 4.2 bar BMEP, and (c) 2239 rpm and 4.2 bar BMEP.

The cumulative heat release law is a combustion process parameter used to assess the impact of combustion on engine cycle efficiency and exhaust emissions. From an

efficiency perspective, the crank angle at which 50% of the cycle's heat is released is considered particularly important.

At a 4.2 bar BMEP, the highest heat release rates during the combustion of the first 5% and 10% of the cycle fuel occurred with blend B7 at engine speeds of 1635 rpm and 1937 rpm and with D100 at an engine speed of 2239 rpm. For the first 50% of the fuel cycle, combustion was slowest for D100 and fastest for B7 across all three tests. The same trend was observed for the combustion of 90% of the cycle fuel. Thus, combustion lasted the longest for D100 and the shortest for B7.

### 3.2. Results for the Full-Load Regimes

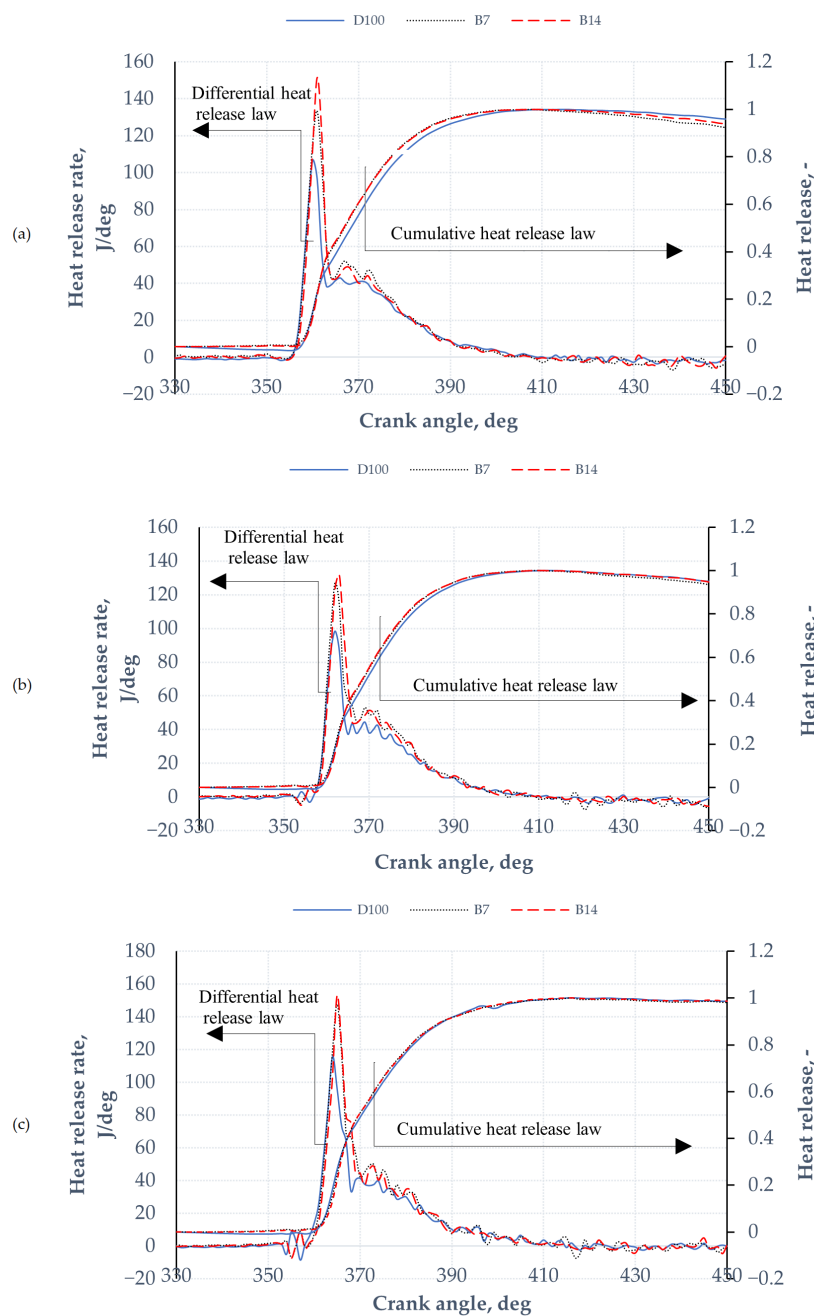
The combustion process parameters, i.e., the heat release laws for full-load regimes that were conducted for a 5.6 bar BMEP, are given in Table 6, while a graphical representation of the differential and cumulative heat release laws is shown in Figure 4.

**Table 6.** Differential and cumulative heat release parameters for the 5.6 bar BMEP.

Parameter	Value/Description		
	D100	B7	B14
Engine speed, rpm		1635	
Maximum heat release rate, J/deg	106.8	133.26	151.54
Position of the maximum heat release rate relative to TDC, deg	0	1	1
Crank angle for 5% fuel burned, deg	-1.38	-1.62	-1.19
Crank angle for 10% fuel burned, deg	-0.59	-0.66	-0.34
Crank angle for 50% fuel burned, deg	8.33	6.91	6.75
Crank angle for 90% fuel burned, deg	25.52	22.97	23.29
Engine speed, rpm		1937	
Maximum heat release rate, J/deg	98.32	127.32	131.6
Position of the maximum heat release rate relative to TDC, deg	2	2	3
Crank angle for 5% fuel burned, deg	0.08	-0.29	0.20
Crank angle for 10% fuel burned, deg	0.92	0.73	1.17
Crank angle for 50% fuel burned, deg	9.46	8.62	8.46
Crank angle for 90% fuel burned, deg	26.51	25.08	24.93
Engine speed, rpm		2239	
Maximum heat release rate, J/deg	115.65	148.01	152.99
Position of the maximum heat release rate relative to TDC, deg	4	5	5
Crank angle for 5% fuel burned, deg	2.02	2.07	2.39
Crank angle for 10% fuel burned, deg	2.94	3.16	3.47
Crank angle for 50% fuel burned, deg	10.34	9.75	9.75
Crank angle for 90% fuel burned, deg	28.26	27.99	27.93

For all three tests, in the cases of using fuel blends, higher maximum heat release rates were achieved compared to D100 (Figure 3), which is logical. The highest maximum heat release rate was obtained at the highest engine speed test, for the blend with the higher biodiesel content (B14), and it amounted 152.99 kJ·deg<sup>-1</sup>.

Furthermore, for each test, the blends reached the maximum heat release rate later than D100, except for the B7 blend at an engine speed of 1937 rpm and a 5.6 bar BMEP, where B7 and D100 simultaneously reached their maximum values at 2 deg.



**Figure 4.** Differential and cumulative heat release laws: (a) 1635 rpm and 5.6 bar BMEP, (b) 1937 rpm and 5.6 bar BMEP, and (c) 2239 rpm and 5.6 bar BMEP.

When observing the integral combustion law, i.e., combustion by phases, it is noticeable that the first 5% and 10% of the cycle fuel quantity burned the fastest in the case of the B7 blend, except in the case of the 2239 rpm engine speed and 5.6 bar BMEP, where D100 had the fastest combustion.

During these phases of the combustion process, fuel injection and mixture formation are still intense. Since the mixture formation process depends on many factors, the behavior of the blends in terms of the combustion rates of the first 5% and 10% of the burned fuel cannot be generalized. For all three tests, 50% of the cycle fuel quantity burned faster when using the blends than when using D100. A higher biodiesel content in the blend corresponds to a faster combustion of 50% of the cycle fuel quantity, or, in some cases, this combustion phase has lasted the same for both blends at an engine speed of 2239 rpm.

The combustion of 90% of the cycle fuel quantity, for all three tests, lasted the longest for D100. It can also be observed that the differences in the combustion duration of 90% of the cycle fuel quantity between B7 and B14 are small and depend on the engine speed.

Finally, it can also be observed that with an increase in the biodiesel share in the blend, the maximum heat release rate increases, which is unfavorable from the standpoint of mechanical loads and engine noise. However, it can be seen that under certain operating conditions, with an increasing biodiesel share, the maximum heat release rate occurs later (during the expansion stroke), which is favorable.

It is also important to note that the center of combustion (50% of the cycle fuel mass) is located closer to TDC in the case of blend combustion than in the case of D100 combustion. The position of the combustion center is one of the influential parameters on the efficiency of the working cycle, i.e., the overall engine efficiency, and it can be concluded that the blends have a more favorable combustion center position than D100.

The BTE and BSFC values for the blends and D100 are shown in Table 7 and Figures 5–8.

From the standpoint of BTE, the most favorable for application is the B14 blend, followed by B7 and then D100. Therefore, the research has shown that under most operating regimes, a higher share of biodiesel fuel in the blend results in greater engine economy. In addition to the oxygen contained in the blends, a parameter that contributes to this trend is the selected fuel injection advance angle (21 deg BTDC). Injection timing influences the ignition delay period as well as the heat release rate. It is typical for biodiesel fuels to exhibit a longer ignition delay period than D100. It has already been stated that a fixed injection timing was applied in this study and that the injection timing was selected to ensure the smoothest engine operation with the B7 blend. The chosen injection timing did not significantly deviate from the optimal injection timing for D100 operation, thereby reducing potential bias in the comparison of the obtained HRR and BTE results for D100 and the blends. The influence of the injection timing selection, as well as the determination of the optimal injection timing for B7 and B14, are planned for future research.

A similar trend, in terms of heat release and engine efficiency, was observed for the blends used in this study, as well as for blends of biodiesel derived from waste fish oil and D100 (B25, B50, B75, B100), which were tested in a single-cylinder diesel engine.

With an increasing proportion of waste fish oil biodiesel in the blends, at loads of 50% and 100%, the maximum heat release rate increased, while the peaks occurred almost simultaneously or slightly later [30]. At full load, the B25, B50, B75, and B100 blends achieved higher brake thermal efficiency compared to D100, by 0.74%, 1.77%, 2.75%, and 3.74%, respectively. These authors attribute this to more complete combustion due to the oxygen content present in biodiesel fuel.

In study [19], in which blends of biodiesel derived from waste grape seeds (B5 and B10) were tested in a four-cylinder engine under full load, it was found across all operating conditions that the maximum heat release rates were identical for B10 and D100, while B5 exhibited a slightly lower maximum heat release rate compared to D100 and B10. In comparison to D100, the engine showed lower BTE, i.e., higher BSFC. The authors concluded that a higher proportion of biodiesel in a blend leads to a reduction in engine efficiency.

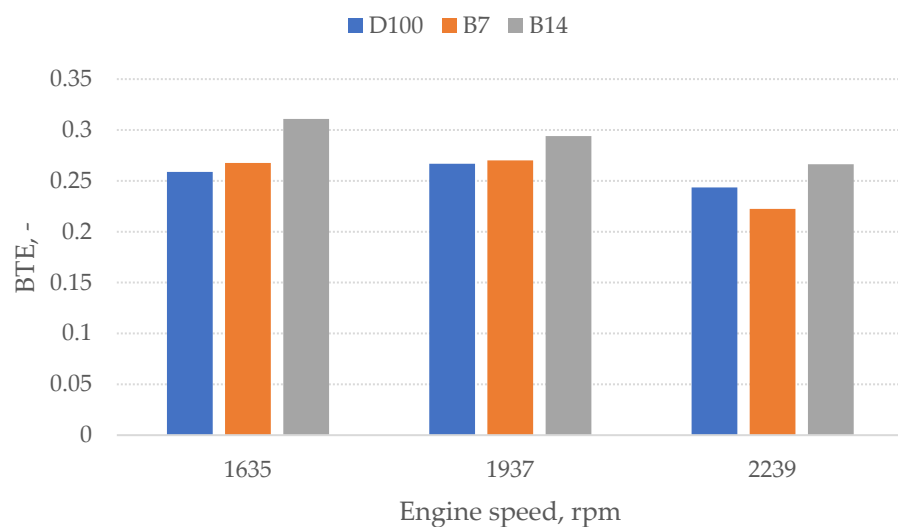
For the case of testing a four-cylinder engine with biodiesel blends from waste grape seeds (B0, B30, B50, B70, B100), the authors of [31] found that at full load, the highest efficiency is achieved with B70, while the lowest is with B100. In that case, there was no clear trend in efficiency variation with changing biodiesel share in the blend.

However, it should be emphasized that in both studies (19 and 24), the engines differed from the engine used in this work (number of cylinders, injection timing, compression ratio, combustion chamber shape...).

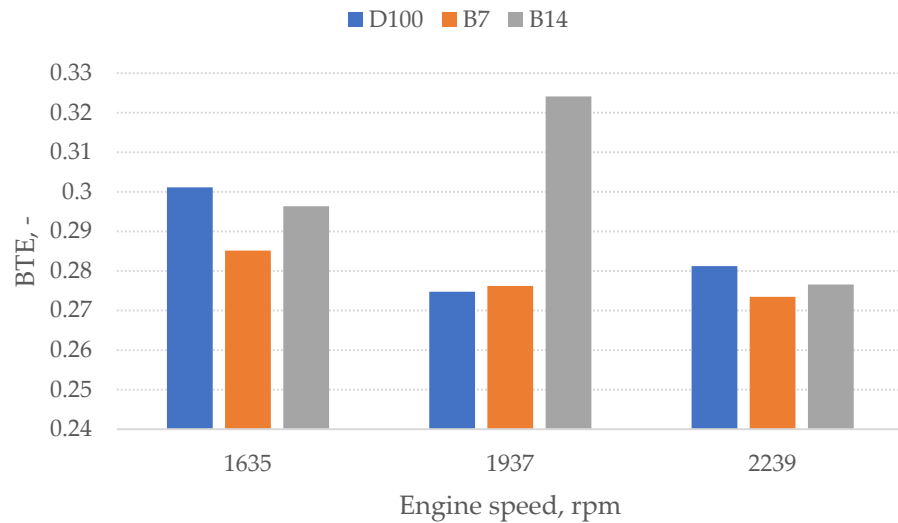
The most widely used biodiesel fuel derived from waste materials is biodiesel produced from waste cooking oil. Testing of this biodiesel in a single-cylinder engine showed that the B5 and B10 blends achieved nearly the same maximum heat release rate, which was slightly lower compared to D100 [32]. It was also observed that the peak heat release rate for these blends occurred slightly earlier than for D100. The authors found that the addition of 5% and 10% biodiesel leads to a slight increase in BSFC (up to 4%) and a reduction in BTE (up to 2.8%).

**Table 7.** Brake thermal efficiency and brake-specific fuel consumption.

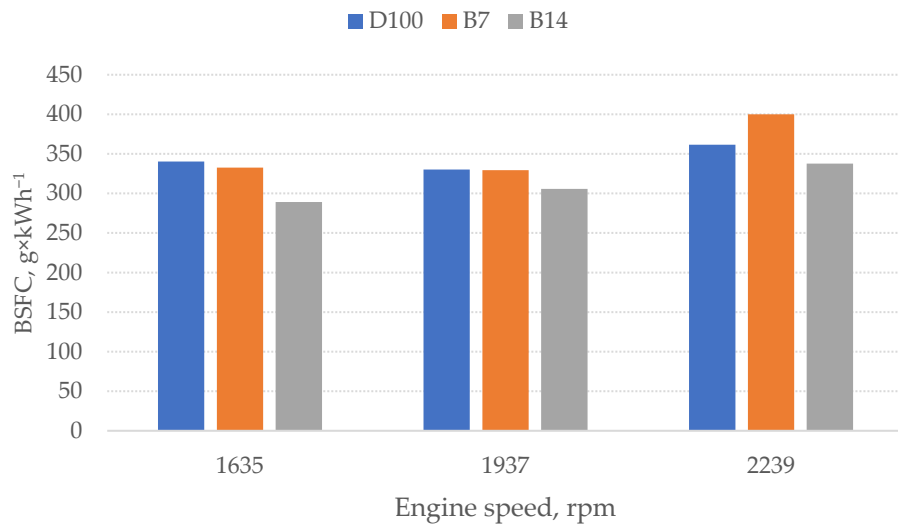
Parameter	Value/Description		
	D100	B7	B14
BMEP, bar		4.2	
Engine speed, rpm		1635	
Brake thermal efficiency, -	0.259	0.267	0.311
Brake-specific fuel consumption, g·kWh <sup>-1</sup>	340.33	332.56	289.08
Engine speed, rpm		1937	
Brake thermal efficiency, -	0.267	0.270	0.294
Brake-specific fuel consumption, g·kWh <sup>-1</sup>	330.13	329.35	305.66
Engine speed, rpm		2239	
Brake thermal efficiency, -	0.244	0.222	0.266
Brake-specific fuel consumption, g·kWh <sup>-1</sup>	361.61	400.08	337.62
BMEP, bar		5.6	
Engine speed, rpm		1635	
Brake thermal efficiency, -	0.301	0.285	0.296
Brake-specific fuel consumption, g·kWh <sup>-1</sup>	292.49	312.03	303.22
Engine speed, rpm		1937	
Brake thermal efficiency, -	0.275	0.276	0.324
Brake-specific fuel consumption, g·kWh <sup>-1</sup>	320.58	322.12	277.29
Engine speed, rpm		2239	
Brake thermal efficiency, -	0.281	0.273	0.277
Brake-specific fuel consumption, g·kWh <sup>-1</sup>	313.20	325.36	324.93



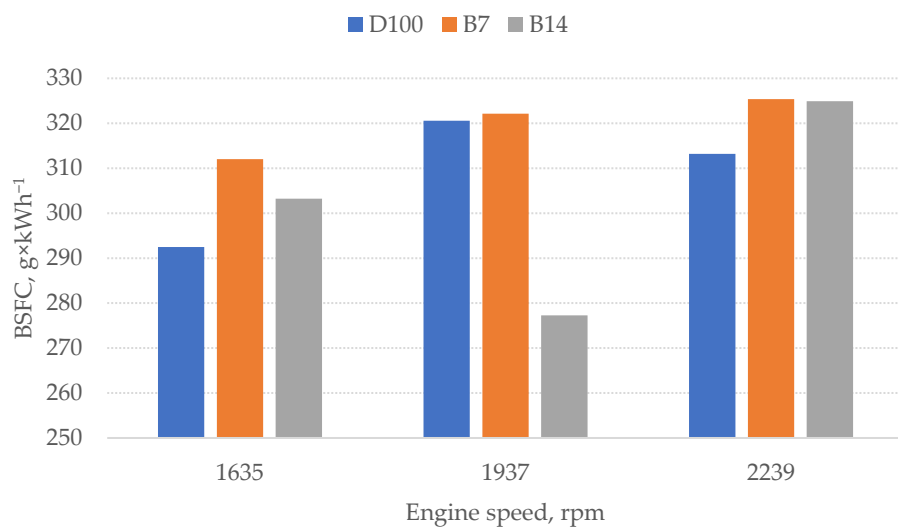
**Figure 5.** BTE vs. engine speed for a 4.2 bar BMEP.



**Figure 6.** BTE vs. engine speed for a 5.6 bar BMEP.



**Figure 7.** BSFC vs. engine speed for a 4.2 bar BMEP.



**Figure 8.** BSFC vs. engine speed for a 5.6 bar BMEP.

#### 4. Conclusions

Biodiesel fuel is a promising biofuel that is increasingly being considered for use in road transport. It is particularly interesting when produced from waste generated as a by-product in various stages of food and beverage production. Biodiesel fuel obtained from waste grape seed oil can be successfully combusted in a diesel engine, without any modifications, in the forms of blends B7 and B14. Both blends have proven to be high-quality motor fuels because:

- Biodiesel fuel from waste grape seeds is produced from waste materials and is cost-effective from a production standpoint;
- Based on the combustion of 50% of the cyclic fuel quantity, it can be concluded that in the case of using B7 and B14 blends, the engine's working cycle is economical, and that these two blends are comparable to diesel fuel in terms of efficiency. Such energy performance can be attributed to the oxygen content present in biodiesel and its blends, as well as to the injection timing;
- Under most operating conditions, the blends have higher BTE than D100, especially B14.

A drawback of using the B7 and B14 blends is the higher maximum rates of heat release. However, in modern engines, this issue is overcome through multiple fuel injections, which optimize the combustion process.

Therefore, it can be concluded that both blends are acceptable as engine fuels, with B14 having an advantage since its use replaces a larger portion of D100 with biofuel.

In future research, in addition to the parameters analyzed in this paper, it is necessary to investigate the influence of other parameters on engine efficiency. Of particular importance would be to determine the influence of the injection timing and to identify the optimal injection timing for the applied blends in order to achieve the maximum possible engine efficiency. It would also be interesting to determine how engine efficiency changes with variations in the compression ratio and at which compression ratio the engine achieves maximum efficiency for the applied blend.

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#### Abbreviations

The following abbreviations are used in this manuscript:

BMEP	Brake Mean Effective Pressure
FAME	Fatty Acid Methyl Ester
FAEE	Fatty Acid Ethyl Ester
BTE	Brake thermal efficiency

BSFC	Brake-specific fuel consumption
FB25	Fuel blend with fish oil
CB25	Fuel blend with cooking oil
BTDC	Before top dead center
B7	Fuel blend with 7% of biodiesel made from grape seeds
B14	Fuel blend with 7% of biodiesel made from grape seeds
D100	Conventional diesel fuel
TDC	Top dead center
ESC	European Stationary Cycle
HRR	Heat release rate
CA	Crankshaft angle

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