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Effective Parameters of a Compression-Ignition Engine Powered by a Mixture Consisting of Ethyl-Tetra-Butyl Ether and Diesel Fuel

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Abstract: The present article presents the results of analytical research on the possibility of using a mixture consisting of diesel oil and ethyl-tertiary-butyl ether in different percentages of EETB to power compression-ignition engines. The calculations were carried out for a four-stroke diesel engine intended for use as a power generator, among other things. In order to illustrate and verify the correctness of the calculations, a mathematical model was built that confirmed the correctness of the calculations. The calculations focused on a thorough analysis of the elemental composition (content of individual elements) of the fuel and, in particular, the carbon content in the fuel. A calculation algorithm was applied for mixing diesel fuel with ethyl-tertiary butyl ether in a share of EETB 5% + 95% ON, 10% EETB + 90% ON, 20% EETB ++80% ON, 30% EETB + 70% ON, 40% EETB + 60% ON. In this study, it determined the parameters of the working medium, the parameters of the environment and residual gases, and the processes (charge filling, compression, combustion, expansion) and effective parameters of the engine. The calculations used in this study led to heat balance, and a summary of the obtained results and their comparison with diesel oil are also described in this study. The results show the feasibility of using a mixture of ethyl-tertiary-butyl ether as a fuel in diesel engines. The results are very similar to those for 100% diesel. The results of our calculations confirming the possibility of using ether for fuel and thus maintaining similar engine operating parameters.

Keywords: biofuel; numerical simulation; diesel engine; ethyl-tertiary butyl ether

Citation: Mitan-Zalewska, P.; Klyus, O.; Kozak, M. Effective Parameters of a Compression-Ignition Engine Powered by a Mixture Consisting of Ethyl-Tetra-Butyl Ether and Diesel Fuel. *Energies* **2023**, *16*, 5163. https://doi.org/10.3390/en16135163

Academic Editors: Jiro Senda and Maria Cristina Cameretti

Received: 11 May 2023
Revised: 12 June 2023
Accepted: 28 June 2023
Published: 4 July 2023

1. Diesel Fuel with Ethyl Tertiary Butyl Ether—Biofuel for Compression Ignition Engines

The twentieth century ushered in significant economic developments around the world, resulting in an increase in energy demand. The main sources of energy have become oil, gas, and coal, which are characterized as cheap resources and are non-renewable raw materials with ending deposits. The increase in demand for transport services results in an increase in the production of mass transport and, as a result, an increase in the demand for fuel increases. The combustion of oil, gas, and coal negatively affects the Earth's natural environment, and nitrogen oxide, sulfur, and carbon emissions pollute the planet's atmosphere. In addition, excessive emissions of greenhouse gases in the environment, mainly derived from the combustion in internal combustion engines, promote global warming.

Increasing environmental awareness contributes to taking action to protect the environment. One of the ways to do this is to reduce the impact of emitted pollutants, particularly carbon dioxide. Standards regulating the content of toxins in exhaust gases are tightening, forcing engineers to look for new design solutions for engines that will burn fuels from renewable sources and whose exhaust emissions are less harmful to the environment.

Vehicles powered by electric motors further exemplify the desire of engineers to reduce exhaust emissions. The production of such cars is growing every year. This does not change the fact that the value of these cars are also seemingly increasing, and the cost of replacing an electric car battery is close to the value of the vehicle itself. In the future, electric motors will likely provide a popular means of powering light urban vehicles.

Compression-ignition and spark-ignition engines will be successfully used in transport and trade as the main source of propulsion for vehicles. In order to reduce the harmfulness of exhaust emissions, engineers are aiming to build engines that will successfully burn fuels from renewable sources. In addition, the use of new propulsion systems powered by electricity, as well as various fuels, e.g., hydrogen, alcohol (ethanol, methanol), ethers, natural gas, LPG, vegetable oils, and their esters, will also reduce environmental pollution. Research found in the literature indicates that the idea of using biofuels is correct and that the main impact of using biofuels will likely come in the form of reducing exhaust emissions [1–3]. However, the use of biofuels for engines involves the need to carry out technological processes and obtain biofuels with appropriate physicochemical properties. The use of biofuels must also properly affect the engine's operating processes, and the additives must be properly selected so that the process of consumption of fuel system components is the smallest [4,5]. The main purpose of the fuel used is to reduce fuel consumption, reduce exhaust toxicity, and generate high engine efficiency [6].

Biofuels that can be used for compression-ignition engines [7–9]:

Belonging to the group of unconventional liquid fuels, vegetable oils and their esters are mainly produced from plant seeds. The use of vegetable fuels, in particular rapeseed oil, has advantages and disadvantages. The advantages include the following:

- There is a large amount of oxygen bound in the fuel, which favorably affects the combustion process;
- The high kinematic viscosity seals the fuel injection system;
- The use of rapeseed oil has a beneficial effect on the environment;
- Low sulfur content (up to 0.05%);
- Strengthening the domestic biofuel market and the possibility of developing free land for rapeseed cultivation;
- The use of rapeseed oil reduces the NOx content in exhaust gases;

However, the disadvantages are as follows:

- High viscosity at low temperatures blocks the fuel filter in the system;
- Esters are oxidized, which increases their acidity and contributes to faster wear of the seal;
- At high temperatures, glycerin present in the composition turns into acroline, which, in the engine, forms carbon deposits (leading to engine failure);
- The use of pure vegetable oil necessitates the modification the engine designs and their supply systems (e.g., two-tank power system equipped with a heated oil tank, additional pumps, filters, etc.);
- The use of rapeseed oil reduces the operating parameters of the engine, i.e., power, torque, average pressure indicated, etc.;

Oils of vegetative origin include the following:

- corn oil,
- cotton oil,
- linseed oil,
- peanut oil,
- soybean oil,
- sunflower oil, etc.

Alcohols

Ethanol is the most well-known alcohol used for compression-ignition engines, but it has adverse physicochemical properties. Its mixing with diesel reduces lubricity, and under the influence of water contained in the fuel and low temperature, it tends to delaminate.

Additionally, a mixture of ethanol and ON reduces its viscosity, which may result in leaks in the fuel system [10].

Methyl alcohol, also known as methanol, has only one carbon atom and OH group in its structure. It is considered to be a very dangerous substance that poses a threat to human health and is characterized by a low calorific value in relation to diesel oil and high auto-ignition temperature.

Ethanol has two carbon atoms in its structure and can be mixed with diesel to a limited extent. The problems associated with this alcohol become visible with changes in temperature and include the presence of water in the fuel, which promotes the delamination of the mixture. Used as an oxygen additive by the Gdańsk Refinery for motor gasoline and thus increasing their octane number, 95% ethanol is biodegradable [11,12].

Propanol is a chemical compound from the group of alcohols found in two molecular structures: 1-propanol and isopropanol. The use of propanol to power compression-ignition engines is possible after mixing it with oils of vegetable or mineral origin. The increasing share of alcohols in the mixture of propanol and butanol isomers with rapeseed oil increases the emissions of unburned carbohydrates and reduces the cetane number and viscosity of the fuel.

Butyl alcohol—An alcohol with a higher molecular weight that can be used as a fuel is biobutanol. It is obtained through a two-stage biomass fermentation process using appropriate anaerobic bacteria. Butanol has physicochemical properties similar to that of gasoline, particularly in terms of calorific value and density. Unfortunately, butanol is more viscous than gasoline, which worsens the quality of fuel spraying. Ethers are also ecological fuels used for compression-ignition engines. Ethers are chemical substances with different physicochemical properties and complex molecular structures. These include dimethyl ether (DME), diethyl ether (DEE) [13], methyl-tetra-butyl ether (EMTB), and ethyl-tertiary-butyl ether (EETB), among others. Depending on the structure of the ether, they differ in terms of physicochemical properties. EETB is produced by reacting isobutylene with ethanol, which is obtained from renewable raw materials [14].

Table 1 shows the physicochemical properties of ethers.

Table 1. Physicochemical properties of selected ethers.

L.P.	Physicochemical Properties	Ether Name				
		DEE	DME	EETB	Diglyme	DBE
1.	Chemical formula	$C_4H_{10}O$	C_2H_6O	$C_6H_{14}O$	$C_6H_{14}O_3$	$C_8H_{18}O$
2.	Chemispider ID	3168	7956	11,996	7858	8569
3.	CAS number	60-29-7	115-10-6	637-92-3	111-96-6	142-96-1
4.	Molecular weight [g/mol]	74.1	46.07	102.18	134	130
5.	Carbon content [%, m/m]	64.9	52.2	70.53	54	74
6.	Hydrogen content [%, m/m]	13.5	13	13.81	10	14
7.	Oxygen content [%, m/m]	21.6	34.8	15.66	36	12
8.	Cetane number	120–140	55–78	8	100–210	91–100
9.	Flash point [°C]	−40	<−26	−19	51 (71)	25
10.	Freezing point [°C]	−116	−138.5	−94	−64	−95
11.	Auto-ignition temperature [°C]	165	235	310	190	185
12.	Boiling point [°C]	34.6	−24.9	72	160	142
13.	Density at 20 °C [g/cm^3]	0.71	0.668	0.745	0.944	0.77

The use of ethers for fuel purposes is very popular in Poland—PKN Orlen S.A. produces EETB. The use of 10% EETB for diesel fuel reduces particulate emissions, as well as cetane number, which extends the delay period of self-ignition of fuel. The effect of extending this period may come in the form of a greater increase in pressure in the combustion chamber, which in turn results in hard engine operation [15]. EETB is produced from ethanol and isobutylene. It can be obtained from bioethanol, making it a renewable fuel. Bioethanol also produces diethyl ether, which definitely exceeds EETB in terms of cetane number. The use of EETB and DEE to power compression-ignition engines accelerates the wear of friction nodes in the injection apparatus. The use of the above fuels requires an appropriate power supply system and the use of a dual-fuel engine power system. In the literature, data on the physicochemical properties of mixtures of diesel oil and ethyl-tertiary-butyl ether are available [9,15].

In order to check the operating parameters of the engine, based on the parameters of the fuel (a mixture of diesel oil with ethyl-tertiary butyl ether), thermal calculations were carried out for a compression-ignition engine powered by the said mixture. Their correctness was checked in matlab/Simulink, where a control system was built and processes were simulated. The results of these studies are presented in this article.

2. Thermal Calculations for a Mixture of EETB and ON

The calculations were carried out for a four-stroke compression-ignition engine intended for use as a power generator. The engine is an eight-cylinder engine with undivided combustion chambers, a volumetric method of formation of the fuel-air mixture, crankshaft speed up to 2600 rpm, and compression ratio of $\varepsilon = 17$. The engine is naturally aspirated with an effective power of $N_e = 170$ kW and supercharged with an air pressure of $p_k = 0.17$ MPa (centrifugal compressor with cooled body, vane diffuser, and radial turbine with constant pressure in front of the turbine). A calculation algorithm was applied to each mixture.

The calculations were carried out for a mixture of diesel fuel with ethyl-tertiary butyl ether in the share of EETB 5%, 10%, 20%, 30%, 40%. EETB from ON was chosen because it is commercially available domestically and is produced in Poland on the basis of it being a domestic raw material. EETB is obtained from bioethanol, which can be produced from food waste (a renewable energy source). The use EETB for diesel fuels reduces particulate emissions. For the purposes of thermal calculations, calculations were made, and Table 2 presents the obtained elementary composition of the mixture of chemical composition ON with the participation of 5%, 10%, 20%, 30%, 40% EETB.

Table 2. Elementary composition for selected fuel mixtures.

Fuel	Average Elementary Fuel Composition		
	Carbon C	Hydrogen H	Oxygen O
Diesel ON	0.870	0.126	0.004
5% EETB + 95% ON	0.862	0.127	0.012
10% EETB + 90% ON	0.854	0.127	0.019
20% EETB + 80% ON	0.837	0.128	0.035
30% EETB + 70% ON	0.821	0.130	0.050
40% EETB + 60% ON	0.804	0.131	0.065
100% EETB	0.705	0.138	0.157

For the above average elementary fuel composition for each mixture, the calorific value of H_u fuel is calculated according to the following formula (according to Mendelejewa) [16]:

$$H_u = 33.91\mathrm{C} + 125.60\mathrm{H} - 10.89(\mathrm{O} - \mathrm{S}) - 2.51(9\mathrm{H} + \mathrm{W})$$

Based on the above formula for the calorific value of fuel in Matlab, Simulink was used to build a model to check the correctness of the equations (Figure 1). The results obtained in the model agree with the results obtained via our calculations.

Already knowing the calorific value of the fuel, further calculations were carried out to determine the parameters of the working medium. One of these parameters is the theoretical air demand to burn 1 kg of Lo fuel (Figure 2).

$$L_0 = \frac{1}{0.208}\left(\frac{\mathrm{C}}{12} + \frac{\mathrm{H}}{4} - \frac{\mathrm{O}}{32}\right)$$

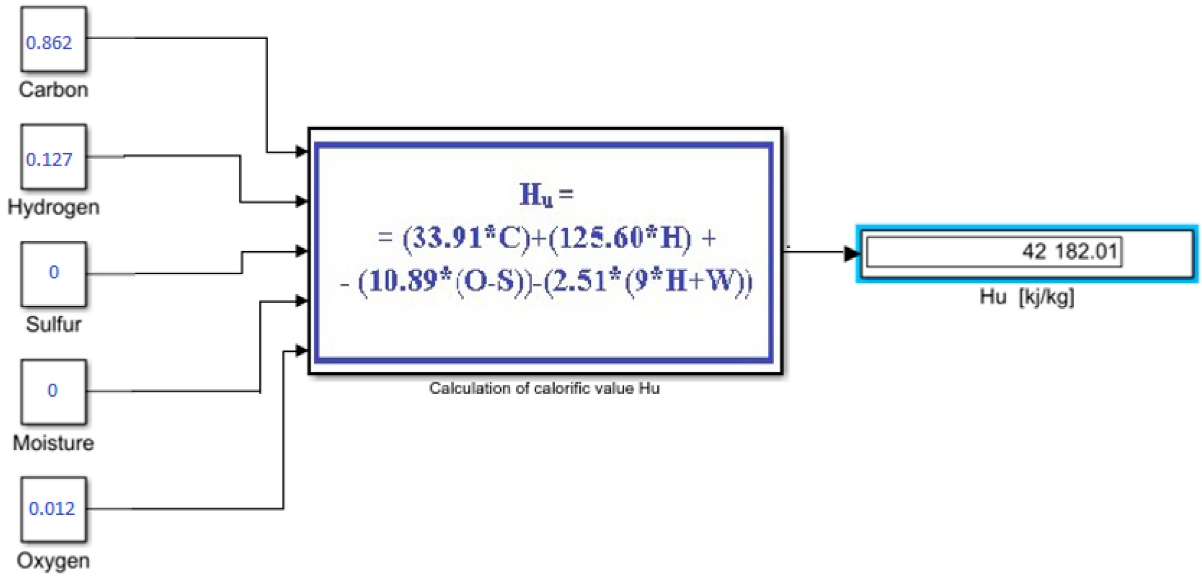

Figure 1. Diagram of the measuring system for determining the calorific value of the fuel.

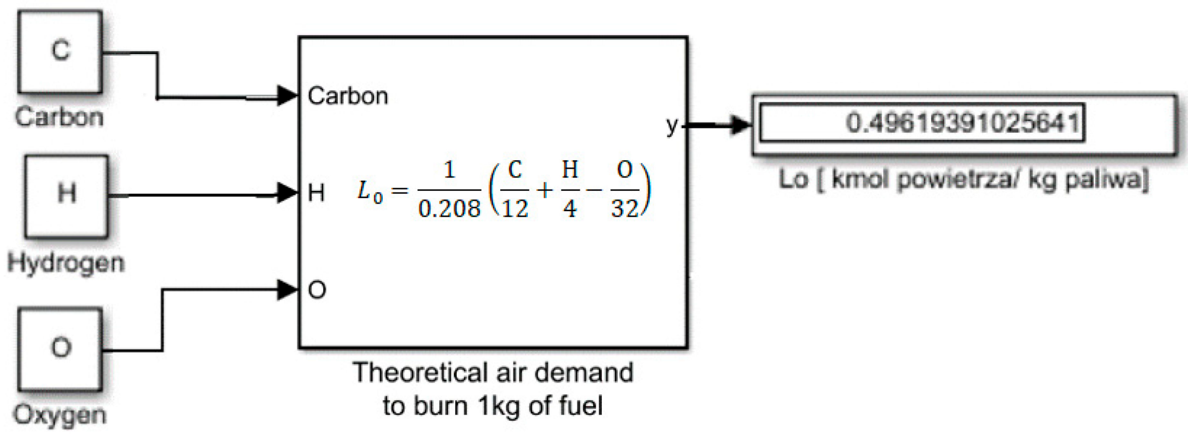

Figure 2. Diagram of the measuring system to determine the theoretical air demand to burn 1 kg of fuel.

The next stage of calculation is the calculation of the amount of fresh load M_t and the total amount of combustion products M_z. In order to calculate the amount of fresh cargo, the coefficient of excess air λ is necessary. It determines the ratio of the actual amount of air in which fuel is burned to the amount of air needed to completely burn the fuel. In order to achieve complete combustion, it is usually necessary to supply more air than the results from the chemical reaction equations indicate. Knowing the λ parameter makes it possible to regulate the combustion process to obtain the appropriate purity of the exhaust gases. The coefficient of excess air depends directly on the type of fuel used. If insufficient oxygen ($\lambda < 1$—rich mixture) is supplied for combustion, toxic gases such as carbon monoxide (CO), nitrogen oxide (NO), or sulfur oxide (SO) will appear in the exhaust gases. In order for all the fuel and exhaust gases to contain less harmful compounds, more oxygen must be supplied ($\lambda > 1$—lean mixture) [17]. For naturally aspirated engines $\lambda = 1.4 \div 1.5$, $\lambda = 1.4$ was used for the calculation. Knowing the previously calculated theoretical air demand, we used the following formula to calculate the amount of fresh cargo:

$$M_t = \lambda * L_0 \left[\frac{\text{kmol fresh cargo}}{\text{kg fuel}}\right]$$

Before proceeding to the calculation of the total amount of combustion products M_z, it is necessary to calculate the number of individual components of combustion products, $M_{CO_2}, M_{H_2O}, M_{O_2}, M_{N_2}$. For each mixture of ON and EETB, the following formulae were used to determine the individual combustion components. Knowing the number of combustion components, we calculated the total number of combustion products M_z, which can be facilitated by using the following formula:

$$M_{CO_2} = \frac{C}{12}\left[\frac{\text{kmol CO}_2}{\text{kg fuel}}\right];\ M_{H_2O} = \frac{H}{2}\left[\frac{\text{kmol H}_2\text{O}}{\text{kg fuel}}\right];$$

$$M_{O_2} = 0.208(\lambda - 1)L_0\left[\frac{\text{kmol O}_2}{\text{kg fuel}}\right]; M_{N_2} = 0.792\lambda L_0\left[\frac{\text{kmol N}_2}{\text{kg fuel}}\right]$$

$$M_z = M_z + M_{CO_2} + M_{H_2O} + M_{O_2} + M_{N_2} = 0.71058\left[\frac{\text{kmol exhaust gases}}{\text{kg fuel}}\right]$$

The results of the above calculations coincide with the results of the model (Figure 3).

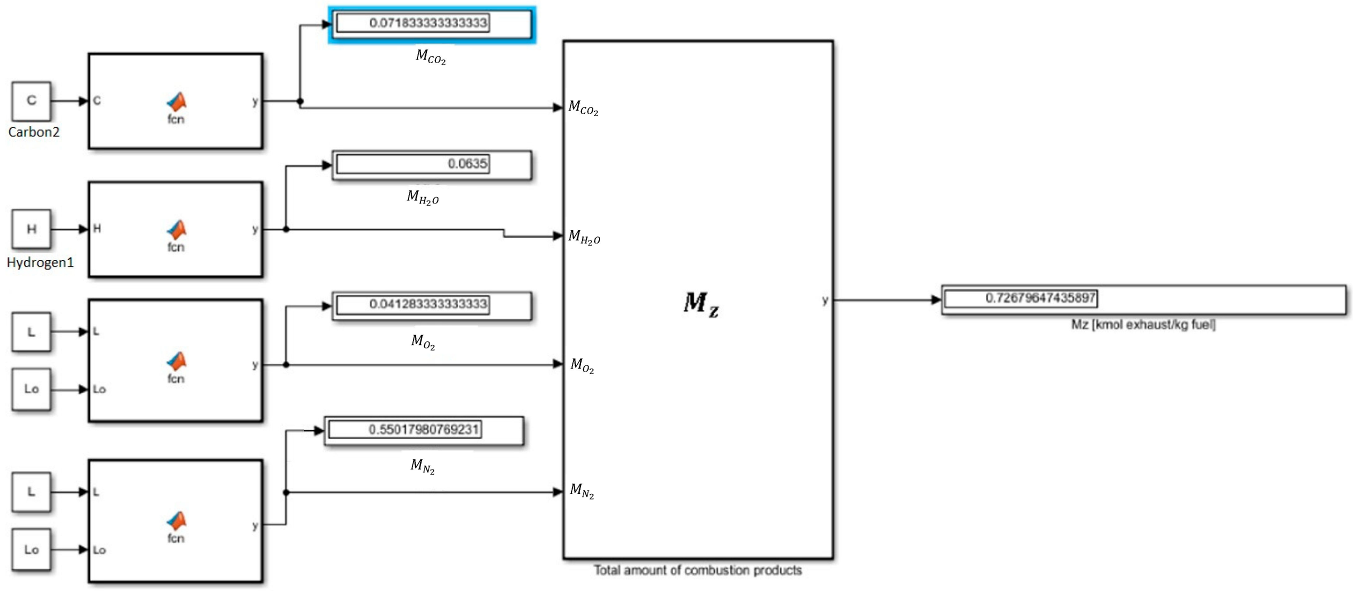

Figure 3. Diagram of the measuring system for determining total amount of combustion products.

The table (Table 3) below summarizes the results of the working medium parameters depending on the share of EETB in the fuel.

Table 3. List of working medium parameters depending on the EETB share in the fuel.

	Mixture 5%EETB + 95%ON	Mixture 10%EETB + 90%ON	Mixture 20%EETB + 80%ON	Mixture 30%EETB + 70%ON	Mixture 40%EETB + 60%ON
H_u [kJ/kg]	42,182.01	41,834.5	41,186.8	40,686.9	40,050.1
$L_0\left[\frac{\text{kmol air}}{\text{kg fuel}}\right]$	0.49619	0.49194	0.48392	0.47766	0.46980
$M_t\left[\frac{\text{kmol fresh cargo}}{\text{kg fuel}}\right]$	0.69467	0.68872	0.67749	0.66872	0.65772
$M_z\left[\frac{\text{kmol exhaust gases}}{\text{kg fuel}}\right]$	0.72679	0.72106	0.71058	0.70279	0.6925

2.1. Filling Process

In the next stage, calculations will be made. The algorithm is the same for each type of EETB share in the fuel, while the results presented are for the compositions of 95% ON and 5% EETB. The calculations take into account the ambient parameters and residual gases, the filling process, the compression process, the combustion process, and the expansion process. The ambient conditions were assumed to be as follows:

Ambient pressure for motor:$pk = po = 0.1$ [MPa]

Ambient temperature for engine $Tk = To = 293$ [K]

Both the temperature and pressure of residual gases depend on the compression value, rotational speed, fuel quality, temperature and intake pressure, and fuel temperature and pressure, among other things. Fairly high compression ratio values (e.g., values of approx. ε= 17) lower the temperature of residual gases, while a high frequency of rotational speed increases the temperature and pressure of residual gases. Accordingly, the pressure and temperature of the residual gases are as follows:

$$Pr = Pk * 1.05 = 0.105 \text{ [MPa]}$$

$$T_r = 750 \text{ [K]},$$

By adopting the above-listed parameters, we can proceed to the calculations regarding the filling process. The first parameter is the temperature of the fresh cargo. In the calculations, it was assumed that the engine does not have a special device for heating fresh cargo. The actual heating of the load in naturally aspirated engines can reach ~15 ÷ 20 K. For the calculation, the following values were taken: ΔT = 20 [K]. Other parameters that should be taken into account include the following:

- charge density when filling:

$$\rho_k = \frac{\rho_k * 10^6}{R_B T_k} = 1.18919 \left[\frac{\text{kg}}{\text{m}^3}\right]$$

- pressure trats at filling:

$$\Delta p_a = \left(\beta^2 + \xi\right)\omega^2 * \rho_k * \frac{10^{-6}}{2} = 0.00787 \text{ [MPa]}$$

where $\left(\beta^2 + \xi\right) = 2.7$; $\omega = 70$.

These parameters were selected in accordance with the speed regime of the engine, taking into account the low hydraulic resistance in the engine intake system.

$$p_a = p_k - \Delta p_a = 0.09213 \text{ [MPa]}$$

- in the residual gas coefficient:

$$\gamma_r = \frac{T_k + \Delta T}{T_r} * \frac{p_r}{\varepsilon p_a - p_r} = 0.030$$

- fill end temperature:

$$T_a = \frac{T_K + \Delta T + \gamma_r T_r}{1 + \gamma_r} = 326 \text{ [K]}$$

- into the filling coefficient:

$$\eta_V = T_K \frac{\varepsilon * p_a - p_r}{(T_K + \Delta T) * (\varepsilon - 1) p_k} = 0.85490$$

For the above-mentioned and performed calculations, a model was built (Figure 4).

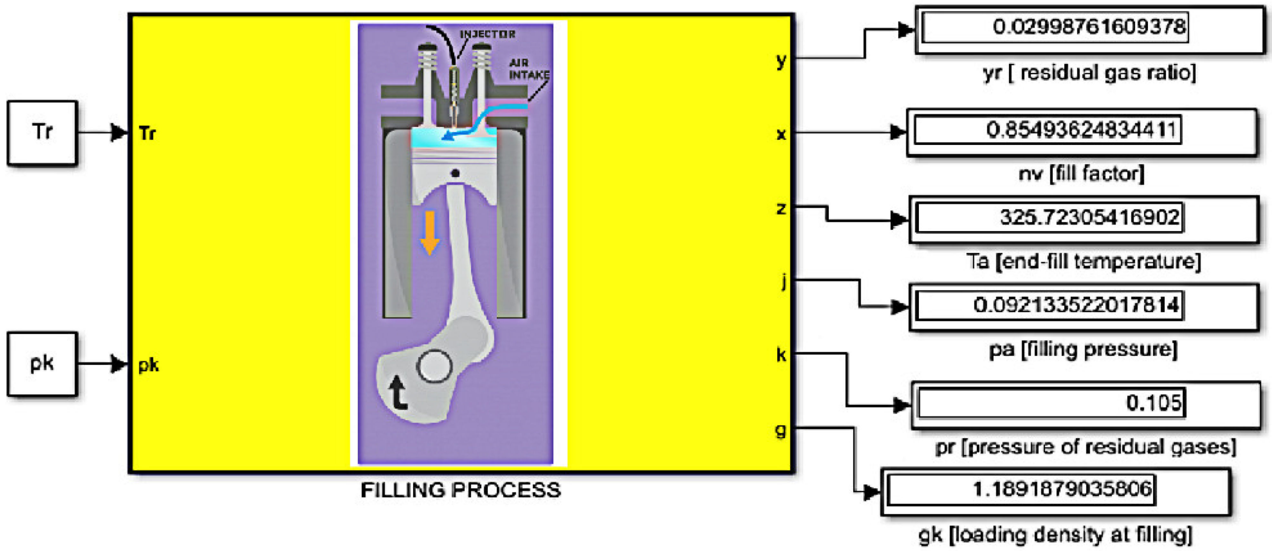

Figure 4. Diagram of the measuring system of the filling process.

2.2. Compression Process

At this stage of the calculation, it was assumed that the average exponent of adiabate and compression polytropics can be determined by the respective nomograms. The approximate value of the polytropic exponent, which is equal to the exponent of the adiabate, was read from the nomogram.

$$\varepsilon = 17;\ T_a = 326\ [\mathrm{K}]; \qquad n_1 \approx k_1 = 1.370$$

Knowing the compression ratio, adiabate exponents, and polytropics, the pressure and temperature at the end of compression were determined to be as follows:

$$p_c = p_a \varepsilon^{n_1}\ [\mathrm{MPa}]; T_c = T_a \varepsilon^{n_1 - 1}\ [\mathrm{K}];$$

$$p_c = 0.09213 \times 17^{1.37} = 4.46803\ [\mathrm{MPa}];\ T_c = 326 \times 17^{1.37-1} = 930\ [\mathrm{K}]$$

Next, the average heat and proper at the end of the compression was calculated:

$$\text{Air} \qquad (mc_v)_{t_o}^{t_c} = 20.6 + 2.638 * 10^{-3} t_c = 22.333 \left[\frac{\mathrm{kJ}}{\mathrm{kmol} * \mathrm{deg}}\right];$$

where:

$$t_c = T_c - 273 = 657\ [^\circ\mathrm{C}]$$

Residual gases (interpolation method)

$$\alpha = 1.4;\ t_c = 657\ [^\circ\mathrm{C}]$$

$$(mc_v'')_{t_o}^{t_c} = 24.168 \left[\frac{\mathrm{kJ}}{\mathrm{kmol} * \mathrm{deg}}\right]$$

Working mixture:

$$(mc_v')_{t_o}^{t_c} = \left[\frac{1}{1 + \gamma_r}\right] \left[(mc_v)_{t_o}^{t_c} + \gamma_r \left(mc_v''\right)_{t_o}^{t_c}\right] = 22.386 \left[\frac{\mathrm{kJ}}{\mathrm{kmol} * \mathrm{deg}}\right]$$

The pre-compression process model is shown below (Figure 5).

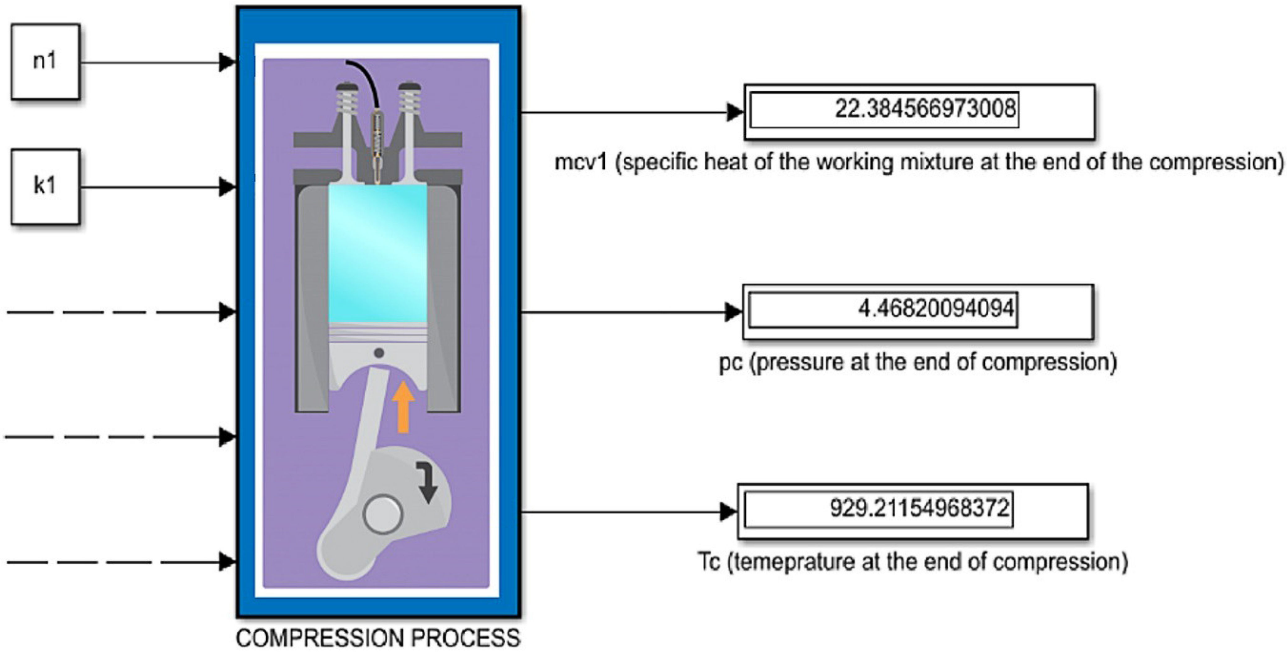

Figure 5. Diagram of the measuring system of the compression process.

2.3. Combustion Process

The combustion process is a process by which it is possible to analyze the selection of parameters for the operation of the engine, the impact of which significantly affects the effective parameters of the engine. For this purpose, it is necessary for us to perform calculations and analyze the achieved parameters. The first stage of the combustion process is to determine the molecular value of the diesel fuel–air mixture μ_o and the molecular value of the mixture change in diesel μ:

$$\mu = M_z / M_t = 1.04624$$

$$\mu = (\ \mu the+) / (\ 1 + \gamma_r \gamma_r\ ) = 1.04489$$

Then, thanks to the previously calculated calorific value of the fuel and the amount of fresh load, the heat of combustion of the working mixture in Hmiesz rob diesels and the average specific heat of the combustion products are calculated $(mc_p'')_{t_o}^{t_z}$ using the following formula:

$$Hmiesz\ rob. = \frac{H_u}{[\ M_t\ (\ 1 + \gamma_r\ )]} = 58953.759 \left[\frac{\text{kJ}}{\text{kmol}}\right]$$

Average specific heat of combustion products (Figure 6):

$$(mc_v'')_{t_o}^{t_z} = \left(\frac{1}{M_z}\right)[\ M_{CO_2}(mc_v''\ CO_2\ )_{t_o}^{t_z} + M_{H_2O}(mc_v'' H_2O)_{t_o}^{t_z} + M_{O_2}(mc_v'' O_2)_{t_o}^{t_z} + M_{N_2}\left(mc_v'' N_2\right)_{t_o}^{t_z}\ ]$$
$$= 24.161 + 0.00191 t_z \left[\frac{\text{kJ}}{\text{kmol}*\text{deg}}\right]$$

$$(mc_p'')_{t_o}^{t_z} = (mc_v'')_{t_o}^{t_z} + 8.315 = 32.476 + 0.00191 t_z \left[\frac{\text{kJ}}{\text{kmol} * \text{deg}}\right]$$

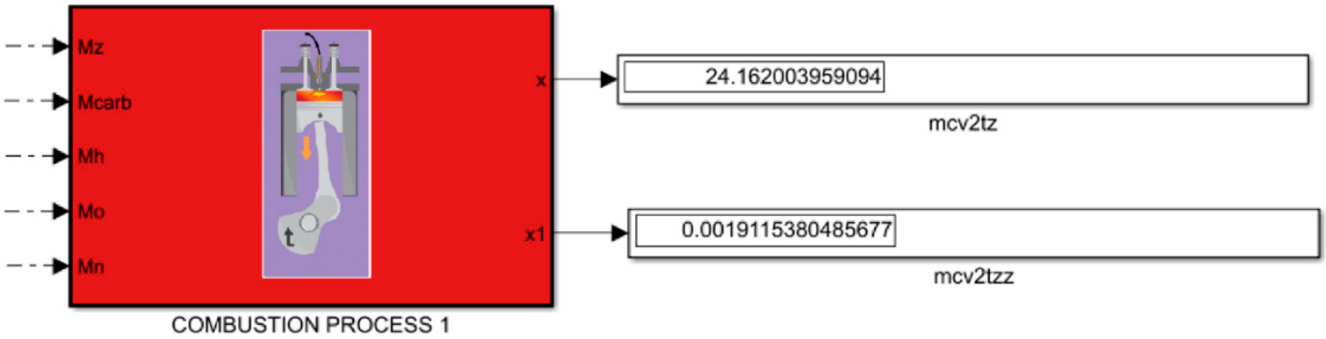

Figure 6. Diagram of the measuring system of the combustion process.

Knowing the above value of the heat of combustion of the mixture in diesel and the average specific heat of the combustion products, we can calculate the temperature at the end of the combustion process T_z . In order to reduce the bearing load, the engine should have a combustion pressure of no more than 11–12 MPa. Therefore, it is necessary to calculate for diesel without supercharging λ = 2.0.

$$\xi_z H_{miesz.rob.} + \left[\left(mc'_v\right)_{t_o}^{t_c} + 8.315\lambda\right] t_c + 2270(\lambda - \mu) = \mu\left(\left(mc''_p\right)_{t_o}^{t_z} t_z;\right.$$

$$t_z = \frac{-33.93385 + \sqrt{33.93385^2 + 4 * 0.00199 * 76143.694}}{2 * 0.00199} = 2007\ [°\mathrm{C}]$$

$$T_z = t_z + 273 = 2280\ [\mathrm{K}]$$

Maximum pressure for diesel: $pz = \lambda * p * c = 8.93606\ [\mathrm{MPa}]$.

Degree of expansion preparation for diesel (Figure 7):

$$\rho = \mu * \frac{T_z}{(\lambda * T_c)} = 1.28083$$

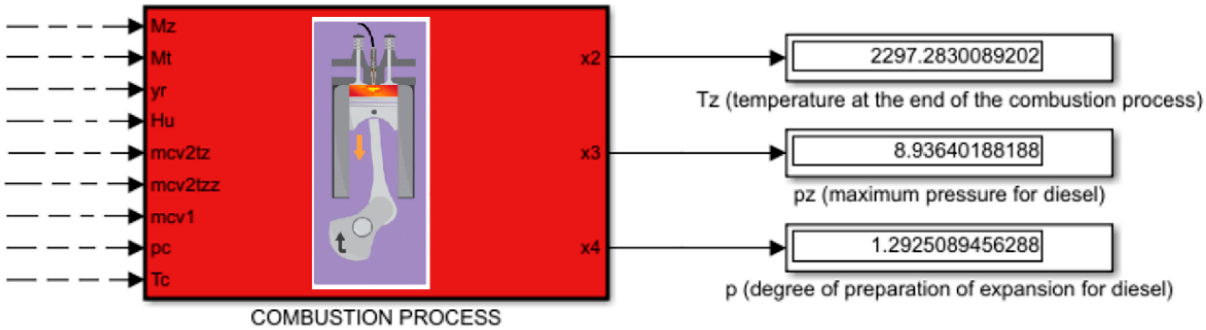

Figure 7. Diagram of the measuring system of the combustion process.

2.4. The Process of Expansion

The last process for which the calculations were made is the process of expansion. As was the case before, on the basis of our previous calculations, the degree of volume increase for diesel δ was calculated, and on the basis of the aforementioned tables, adiabate and expansion polytropic for diesel were selected.

$$d = \frac{e}{p} = 13.273$$

The average values of adiabate and expansion polytropic for diesel are selected as follows: On the nominal (small) extension you have to take adiabate, and on large cylinders, we take polytope. To determine the average values, we used arrays.

$$\delta = 13.273,\ T_z\ = 2280[\mathrm{K}],\ \alpha = 1.4,\ k_2 = 1.273,\ n_2 = 1.260,$$

Pressure and temperature finally expand for diesels.

$$p_b = \frac{p_z}{\delta^{n2}} = 0.34372\ [\mathrm{MPa}];\ T_b\ = \frac{T_z}{\delta^{n2-1}} = 1164\ [\mathrm{K}];$$

Checking the previously accepted temperature of the final (exhaust) gases (Figure 8).

$$T_r\ = T_b / \sqrt[3]{\frac{p_b}{p_r}} = 784[\mathrm{K}]$$

$$\Delta = 100\ (784 - 750)/784 = 4.3\ \%$$

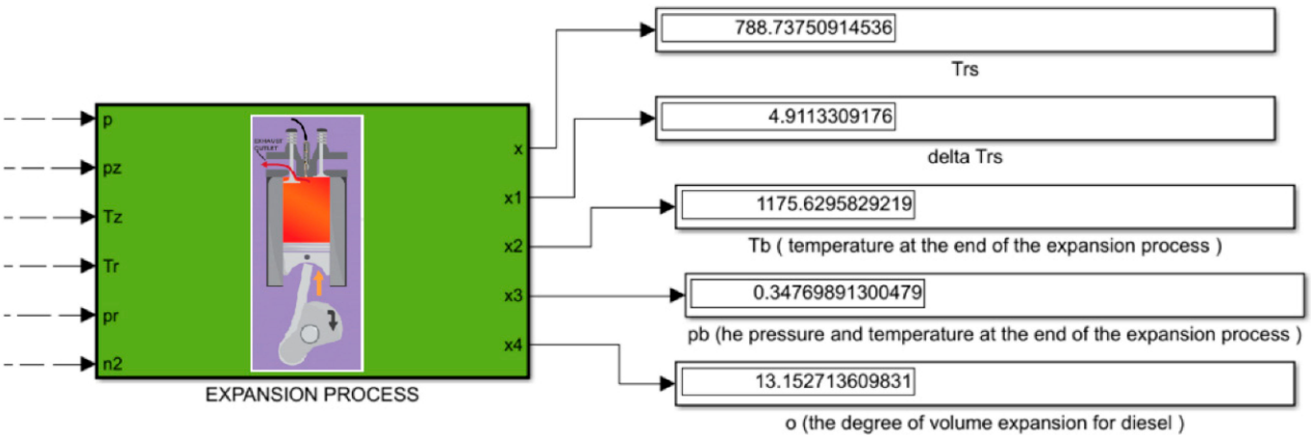

Figure 8. Diagram of the measuring system of the expansion process.

3. Engine Efficiency Parameters

Analyzing all the processes of engine operation, you can proceed to the calculation of the effective parameters of the engine. First of all, p'_i= was determined. Mean pressure is indicated by the formula below:

$$p'_i = \frac{p_c}{\varepsilon - 1}\left[\lambda(\rho - 1) + \frac{\lambda\rho}{n_2 - 1}\left(1 - \frac{1}{\delta^{n_2 - 1}}\right) - \frac{1}{n_1 - 1}\left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right)\right] = 1.01632[\mathrm{MPa}]$$

The following parameters are determined in turn:

indicated pressure p_i, $\quad p_i = \varphi_n p'_i\ = 0.9655\ [\mathrm{MPa}]$

where $\varphi_n = 0.95$

Indicated efficiency η_i $\quad \eta_i = \dfrac{p_i l_0 \alpha}{(H_u \rho_K \eta_v)} = 0.45261$

Indicated specific fuel consumption g_i $\quad g_i = \dfrac{3600}{(H_u \eta_i)} = 188.561\ \left[\dfrac{\mathrm{g}}{\mathrm{kWh}}\right]$

Medium mechanical pressure p_n $\quad p_n = 0.089 + 0.0118 \nu_{ncp} = 0.20936\ [\mathrm{MPa}]$

where ν_{ncp}= 10.2 [m/c].

The effective and mechanical parameters of the engine operation were also calculated:

Average effective pressure p_e : $p_e = p_i - p_n = 0.75614$ [MPa]

Mechanical efficiency η_M $\eta_M = \frac{p_e}{p_i} = 0.78316$

Effective efficiency η_e $\eta_e = \eta_i \eta_M = 0.35447$

Efficient specific fuel consumption g_e $g_e = \frac{3600}{(H_u \eta_e)} = 240.76633 \left[\frac{\text{g}}{\text{kWh}}\right]$

The main indicators of the engine are as follows: effective power, effective torque, fuel consumption, and power. Below are the dependencies according to which these parameters were calculated.

Effective power N_e $N_e = \frac{p_e V_\pi n}{30\tau} = 173.365$ [kW]

Effective torque M_e $M_e = 3 * 10^4 * \frac{N_e}{(\pi n)} = 637.059$ [Nm]

Fuel consumption G_T $G_T = N_e * (g_e/1000) = 41.741 \left[\frac{\text{g}}{\text{h}}\right]$

Power N_π $N_\pi = \frac{N_e}{V_\pi} = 16.383 \left[\frac{\text{kW}}{\text{dm}^3}\right]$

The above calculation algorithm has been performed for all EETB and ON mixtures, and the results are summarized in Table 4.

Table 4. Summary of parameters for each type of mixture EETB with ON.

Fuel and Engine Parameters	**Mixture EETB with ON**					
	100%ON	**5%EETB+ 95%ON**	**10%EETB+ 90%ON**	**20%EETB +80%ON**	**30%EETB +70%ON**	**40%EETB +60%ON**
H_u [kJ/kg]	42,440	42,182.01	41,834.5	41,186.8	40,686.9	40,050.1
p_i [MPa]	0.9655	0.96550	0.96548	0.96378	0.96302	0.9614
η_i	0.4528	0.45261	0.45251	0.45128	0.45056	0.44943
g_i [g/kWh]	187.340	188.561	190.169	193.686	196.379	200.003
p_e [MPa]	0.75617	0.75614	0.756116	0.75442	0.75366	0.75224
η_e	0.35462	0.35447	0.35438	0.35325	0.35261	0.35158
η_m	0.78318	0.78316	0.78315	0.78277	0.78260	0.78228
g_e $\left[\frac{\text{g}}{\text{kWh}}\right]$	239	241	243	247	251	256
N_e [kW]	177.796	173.365	173.360	172.971	172.797	172.471
M_e [Nm]	653.341	637.059	637.041	635.611	634.972	633.774
G_T $\left[\frac{\text{g}}{\text{h}}\right]$	42.529	41.741	42.097	42.799	43.360	44.095

By analyzing the table above, it can be seen that, with the increase in the share of EETB, the calorific value decreases, which is caused by the decreasing share of coal in the fuel. The effective performance of an engine powered by a mixture of ON and EETB slightly differs from the parameters of a diesel engine. The pressure and efficiency indicated decrease as the proportion of ether in the fuel increases. The highest of these values are characterized by a mixture of ON and 5% EETB, which are $p_i = 0.9655$ [MPa] and $\eta_i = 45.26$ [%], respectively. It is also worth noting that, with the decrease in the above parameters, the indicated fuel consumption increases by 6.3% compared to ON.

The mechanical efficiency for mixtures of ON and EETB is almost at one level (on average, varying by 0.51% compared to the efficiency for diesel). The increasing amount of EETB added to diesel increases hourly fuel consumption. It is worth noting that, for a mixture of 5%, EETB+95%ON is the lowest fuel consumption, amounting to 41.741 g/h, 1.9% less than 100% diesel. No mixture of ON and EETB achieved such high effective power and effective torque as diesel. The obtained results coincide with the results reported by other authors in the literature [8,15].

4. Conclusions and Further Work

The main purpose of the calculations was to check what effective parameters will be achieved by a compression-ignition engine powered with biofuel supplementation (a mixture of diesel oil with ethyl-tertiary butyl ether). After the initial assessment of the physicochemical properties of ecological fuels, EETB is characterized by good miscibility with diesel fuel, with no changes in temperature and low hygroscopicity [2]. The disadvantages of ether include a small cetane number and a decrease in the lubricity of the mixture. The basis for the thermal calculations was the elemental composition of the fuel, which was different for each mixture depending on the percentage of ethyl-tertiary-butyl ether in the ON. On its basis, the calorific value of the fuel and the amount of individual combustion products were calculated. Subsequently, the parameters of the engine environment were appropriately selected and then used in calculations for filling, compression, combustion, and expansion processes. The calculations obtained in this way were used to calculate the efficiency parameters and engine indicators.

The analysis of the calculations showed that a mixture consisting of diesel and ethyl-tertiary butyl ether can be used in a compression-ignition engine. Despite the different physicochemical properties of the mixture of ON and EETB, the obtained parameters and engine-operating indicators are satisfactory and do not differ much from the parameters of the diesel engine [15]. The heat loss of the mixtures with ether was up to 50% lower than for diesel, and the exhaust emissions are less harmful. It should be emphasized that our obtained results align with the results presented in the works of Prof. W. Lotko [7,15,18]. In addition, other studies have confirmed that it is possible to use biodiesel to power diesel engines [8,9,19,20]. The calculation model presented in this study can also be used for thermal calculations of the supercharged engine and in the analysis of other biofuels.

Author Contributions: Conceptualization, P.M.-Z., M.K., and O.K.; methodology, P.M.-Z. and O.K.; software, P.M.-Z.; validation, P.M.-Z., M.K., and O.K.; formal analysis, M.K.; investigation, P.M.-Z. and O.K., resources, P.M.-Z., M.K., and O.K.; writing—original draft preparation, P.M.-Z. and O.K.; writing—review and editing M.K.; funding acquisition M.K. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by Maritime University of Szczecin grant number S/1/S/RD/WMiE/23.

Data Availability Statement: The data presented in this study are publicly available in the Library of the Maritime University of Technology in Szczecin at https://katalog.pm.szczecin.pl/3653200683202/mitan-zalewska-paulina/obliczenia-cieplne-silnika-o-zaplonie-samoczynnym-zasilanego-paliwami-ekologicznymi?bibFilter=365, accessed on 1 March 2021. reference number 005 18040.

Conflicts of Interest: The authors declare no conflict of interest.

References

1. Sezer, I. Review study on the use of diethyl ether in compression-ignition engines: Effects on NOx emissions. *Int. J. Automot. Eng. Technol.* **2018**, *7*, 164–183. [CrossRef]
2. Krishnamoorthi, M.; Malayalamurthi, R. Review of the effect of additive diethyl ether on combustion, performance and emission characteristics of diesel and biodiesel/vegetable oil fuelled engines. *Adv. Nat. Appl. Sci.* **2016**, *10*, 9–17.
3. Smigins, R.; Zakis, A.A. Effect of Diethylether/Rapeseed Oil Blends on Performance and Emissions of a Light Diesel Vehicle. *Energies* **2020**, *13*, 3788. [CrossRef]
4. Pehan, S.; Jerman, M.S.; Kegl, M.; Kegl, B. Effect of biodiesel on the tribological characteristics of a diesel engine. *Fuel* **2009**, *88*, 970–979. [CrossRef]

5. Sudan, R.D.M.; Nanthagopal, K. Tribological aspects of biofuels—Review. *Fuels* **2019**, *258*, 116066. [CrossRef]
6. Górski, K.; Smigins, R.; Longwic, R. Studies on physicochemical properties of mixtures of diethyl ether/linseed oil for use as fuel in diesel engines. *Energies* **2020**, *13*, 6564. [CrossRef]
7. Górski, K.; Olszewski, W.; Lotko, W. *Alcohols and Ethers as Fuels for Compression-Ignition Engines*; Technical Journal 7-M/2008; Cracow University of Technology Publishing House: Cracow, Poland, 2008.
8. Hoekman, S.K.; Broch, A.; Robbins, C.; Ceniceros, E.; Natarajan, M. Review of the composition, properties and specifications of biodiesel. *Refurbish. Sustain. Energy Rev.* **2012**, *16*, 143–169. [CrossRef]
9. Corsini, A.; Marchegiani, A.; Rispoli, F.; Sciulli, F.; Venturini, P. Vegetable oils as fuels in diesel engines. Engine performance and emissions. *Energy Procedia* **2015**, *82*, 141–148. [CrossRef]
10. Shamun, S.; Belgiorno, G.; Blasio, G.D. *Evaluation of Engine Parameters for the Use of Alcoholic Fuels in Compression-Ignition Engines*; Springer: Singapore, 2020; Volume 8, pp. 125–139.
11. Lotko, W.; Górski, K. Biofuels for Compression Ignition Engines. Logistics 6/2011. Available online: https://repozytorium.biblos.pk.edu.pl/redo/resources/34119/file/suwFiles/LotkoW_OcenaWybranych.pdf (accessed on 25 January 2020).
12. Caligiuri, C.; Renzi, M.; Bietresato, M.; Baratieri, M. Experimental studies on the effect of the addition of bioethanol in diesel and biodiesel mixtures on the emissions and performance of a micro-cogeneration system. *Energy Drains Manag.* **2019**, *185*, 55–65. [CrossRef]
13. Ibrahim, A. Investigation of the effect of the use of diethyl ether as a fuel additive on the performance and combustion of a diesel engine. *Appl. Therm. Eng.* **2016**, *107*, 853–862. [CrossRef]
14. Szczepański-Sala, W.; Strzępek, P. *Lubricating Properties of Selected Biofuel Compositions for Compression-Ignition Engines*; Technical Journal; Cracow University of Technology Publishing House 8-M/2008: Cracow, Poland, 2008; pp. 143–150.
15. Lotko, W.; Górski, K. *Diesel Engine Power Supply with ON and EETB Mixtures*; Wydawnictwo Naukowo-Techniczne: Warsaw, Poland, 2011.
16. Kolczyn, A.I.; Dmitriov, W.P. *Calculations of Car and Tractor Engines*; Waldorf School: Moscow, Russia, 1980.
17. Muślewski, Ł.; Markiewicz, M.; Pająk, M.; Kałaczyński, T.; Kolar, D. Analysis of the use of fatty acid methyl esters as an additive to diesel fuel for internal combustion engines. *Energies* **2021**, *14*, 7057. [CrossRef]
18. Lotko, W.; Górski, P.; Przedlacki, P. Parameters of Lubricity and Resistance to Scuffing of Mixtures of Diesel oil with Ethyl-Tetra-Butyl Ether. Automotive Archive No. 4/2009. Available online: http://yadda.icm.edu.pl/baztech/element/bwmeta1.element.baztech-article-BGPK-2716-0475/c/httpwww_bg_utp_edu_plartarchiwum20motoryzacji2009lotko2.pdf (accessed on 12 May 2022).
19. Madura Electronics. Combustion Process Parameters, Information Brochure, 03/2007. Available online: https://www.madur.com/ (accessed on 10 March 2022).
20. Markiewicz, M.; Aleksandrowicz, P.; Muślewski, Ł.; Pająk, M. Examination and analysis of selected operating parameters of a vehicle powered by fuel with the addition of biocomponents. *Energies* **2023**, *16*, 3159. [CrossRef]