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Energy and Exergy Analyses of a Diesel Engine Fuelled with Biodiesel-Diesel Blends Containing 5% Bioethanol

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Academic Editor: Kevin H. Knuth

Received: 23 August 2016; Accepted: 24 October 2016; Published: 31 October 2016

Abstract: In this study, energy and exergy analysis were performed for a single cylinder, water-cooled diesel engine using biodiesel, diesel and bioethanol blends. Each experiment was performed at twelve different engine speeds between 1000 and 3000 rev/min at intervals of 200 rev/min for four different fuel blends. The fuel blends, prepared by mixing biodiesel and diesel in different proportions fuel with 5% bioethanol, are identified as D92B3E5 (92% diesel, 3% biodiesel and 5% bioethanol), D85B10E5 (85% diesel, 10% biodiesel and 5% bioethanol), D80B15E5(80% diesel, 15% biodiesel and 5% bioethanol) and D75B20E5 (75% diesel, 20% biodiesel and 5% bioethanol). The effect of blends on energy and exergy analysis was investigated for the different engine speeds and all the results were compared with effect of D100 reference fuel. The maximum thermal efficiencies obtained were 31.42% at 1500 rev/min for D100 and 31.42%, 28.68%, 28.1%, 28% and 27.18% at 1400 rev/min, respectively, for D92B3E5, D85B10E5, D80B15E5, D75B20E5. Maximum exergetic efficiencies were also obtained as 29.38%, 26.8%, 26.33%, 26.15% and 25.38%, respectively, for the abovementioned fuels. As a result of our analyses, it was determined that D100 fuel has a slightly higher thermal and exergetic efficiency than other fuel blends and all the results are quite close to each other.

Keywords: availability; biodiesel; bioethanol; diesel engine; energy analysis; exergy analysis; entropy; safflower; engine performance

1. Introduction

In internal combustion engines it is desirable to convert fuel energy into engine performance at the highest rate, but conversion of all the fuel energy into work thus a one hundred percent energy conversion is not possible [1,2]. Some of the losses that occur during energy conversion are caused by irreversibilities. The remaining energy, in other words the amount of the energy available is defined as exergy [3]. From the viewpoint of thermodynamics, exergy is defined as the maximum amount of work that can be obtained from the system at the specified state as the system is brought to equilibrium with the environment [1].

Exergy analysis is used to determine in detail the amounts of losses in a system and locations where they occur, and the processes that cause them [4,5], so the application of exergy analysis for internal combustion engines allows determining the sources of irreversibilities and obtaining more accurate information about engine efficiency.

In other respects, high energy costs combined with growing energy demand make saving energy increasingly important for the world. Fossil fuels, which have the largest share of the energy demand growth, are not sufficient to meet the energy needs. Additionally serious environmental problems such as air pollution, greenhouse gases, and the climate change caused by them are also globalized energy issues that have turned peoples' attention to eco-friendly renewable and alternative energy sources. In particular, considering air pollution the share of emissions caused by motor vehicles cannot be underestimated. Since the invention of the internal combustion engine many different fuel have been used such as gasoline and diesel, which are known under the name of conventional fuels, natural gas, liquefied petroleum gas (LPG), alcohols and even various alternative fuels such as biodiesel, biogas, hydrogen, and their blends in order to achieve lower fuel consumption and emissions and better engine performance [2,6]. While there are many options that can be fuels for internal combustion engines, the real issue is to determine how well a fuel serves the desired purpose [7]. Biodiesel is the first alternative fuel that comes to mind in terms of replacing diesel fuel and it has been subject to many studies due to the its attractive properties such as being non-toxic, sulphur-free, oxygenated, biodegradable, etc. [8]. Another one is bioethanol and when used with biodiesel, it overcomes the negative effects of the latter, such as high CO emissions, high viscosity, and high cold filter plugging point. Biodiesel conversely also improves the low lubricity and cetane number issues of bioethanol [9]. The reason why it is preferable to use these fuels together is to eliminate the negative effects of each fuel.

There are numerous studies in the literature on the effect of different diesel engine experimental conditions on exergy analysis. The history of the studies on this topic dates back to the mid-1900s [10]. Rakopoulos and Giakoumis [11] carried out a detailed energy and exergy performance computer analysis, verified experimentally, on a diesel engine under transient load conditions. In another study, Rakopoulos and Giakoumis [12] once again used a computer model to compare energy and exergy analysis terms of a transient diesel engine operation. Zheng and Caton [10] reported a study evaluating the energy and exergy distribution for eight operating conditions consisted of four different exhaust gas recirculation (EGR) levels and two different injection timings in a low temperature combustion diesel engine. Caton [13] presented an analytical study analyzing exergy destruction during the combustion process, based on the second law of thermodynamics. Tat [2] investigated the effect of cetane number and ignition delay on the energy and exergy analyses for a diesel engine run on two different biodiesels made from soybean oil methyl ester (SME) and the yellow grease methyl ester (YGME) and blends consisting of 0.75% and 1.5% of cetane improver by weight with SME. Giakoumis [14] studied the effect of cylinder wall insulation using silicon nitride and plasma spray zirconia as insulators for a turbocharged diesel engine operating under transient load conditions from both a first and second law perspective. Azoumah et al. [15] presented a study to examine the combined effect of exergy analysis and gas emissions analysis to optimize the performance of a diesel engine running on cotton and palm oils and their blends with diesel fuel under different engine loads. Caliskan et al. [16] conducted a study to investigate the effect of varying dead state temperatures on exergy efficiencies in a diesel engine fueled with high-oleic methyl ester (HOME). Özkan [17] applied energy and exergy analyses to a direct injection diesel engine running under different multiple injection modes. It was observed that energetic and exergetic efficiencies are inversely proportional to injection pressure.

Many other studies have investigated the effect of various fuels in diesel engines with similar characteristics based on a second law analysis and compared them with diesel fuel. Sahoo et al. [18] examined operation of a diesel engine which was dual fueled with syngas-diesel under varying load conditions from a second law viewpoint. Canakci and Hosoz [19] applied energy and exergy analyses to a diesel engine operated with two biodiesels made from soybean oil methyl ester (SME) and yellow grease methyl ester (YGME), petroleum diesel fuel and 20% blend of each biodiesel with petroleum diesel fuel. It was observed that the specific fuel consumption increased when using biodiesel. The most important term of exergy analysis is the destruction of exergy by irreversible processes; others following it are exergy losses due to the exhaust gas and heat transfer. Rakopoulos et al. [20] researched computationally the effect on the terms of the availability balance of mixing hydrogen, which has the potential for increased second-law efficiency, into natural gas for a direct injection engine. Sekmen and Yilbaşi [21] applied energy and exergy analysis to a direct injection diesel engine using petroleum diesel fuel and soybean oil methyl ester (SME). Rakopoulos and Kyritsis [22] presented a theoretical study to calculate the combustion irreversibilities for a four-stroke, naturally-aspirated diesel engine operated with methane, methanol, and dodecane as fuels. In their study it was observed that lighter fuels result in less entropy than heavy fuels. A review study applied energy and exergy analyses to a direct injection diesel engine dual fuelled with petroleum diesel oil and biogas was carried out by Sorathia and Yadav [3]. López et al. [5] studied exergy analysis applied to a direct injection diesel engine operated with olive pomace oil biodiesel/diesel fuel blends. Ramos da Costa et al. [4] investigated theoretically and experimentally the performance characteristics of a dual diesel engine using natural gas and diesel in terms of both energy and exergy analyses.

Considering all of these studies, it is anticipated that this study will contribute to the literature by applying exergy analysis to a diesel engine using biodiesel made from safflower oil, diesel and bioethanol blends. We aimed to comparatively investigate the effects of different fuel blends on the energy and exergy analysis using a single cylinder, four-stroke diesel engine. It was also our aim to determine in which way the exergy terms are affected by the engine speed. The blends prepared by mixing biodiesel and diesel in different proportions with 5% bioethanol were identified as D92B3E5 (92% diesel, 3% biodiesel and 5% bioethanol), D85B10E5 (85% diesel, 10% biodiesel and 5% bioethanol), D80B15E5 (80% diesel, 15% biodiesel and 5% bioethanol), D75B20E5 (75% diesel, 20% biodiesel and 5% bioethanol), respectively. Tests were carried out for each fuel blend for twelve different engine speeds between 1000–3000 rev/min. Energy analysis terms such as fuel energy, thermal efficiency, energy losses, brake power and exergy analysis terms such as fuel exergy, exergetic efficiencies, exergy losses, exergetic brake power, irreversibilities and exergy destructions of the combustion process for the diesel engine have been computed by determining the balances of energy and exergy rate.

There are many studies about the exhaust emissions, combustion and performance of internal combustion engines which use various fuels, including diesel, biodiesel and bioethanol, in terms of the first law of thermodynamics, but our assessments of these fuels, especially as a mixture in terms of the second law of thermodynamics, separate this study from the others.

2. Materials and Methods

A schema of the experimental setup which consists of a diesel engine connected with a hydraulic dynamometer, emissions measurement equipment, radiator, temperature measurement devices, flow meters and control panel is shown in Figure 1. A picture of the experimental setup is also given in Figure 2. A single cylinder, four-stroke, water cooled, direct-injection Antor 3LD510 diesel engine (Anadolu Motor Company, Kocaeli, Turkey) with maximum engine brake power of 9 kW at 3300 rev/min was used during the experiments.

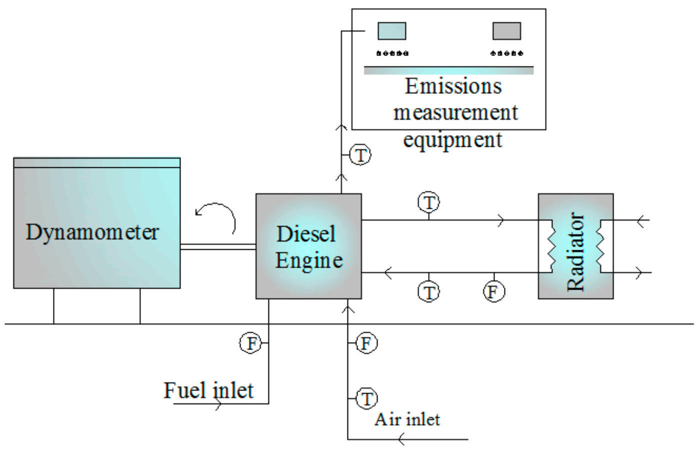


Figure 1. Schema of the experimental setup.

Figure 2. Picture of the experimental setup.

The bore of test engine was 85 mm, the stroke was 90 mm, and the compression ratio was 17.5:1. The speed range of the hydraulic dynamometer is 0–6500 rev/min and its torque range is 0–450 Nm. A flanged, sharp corner-type orifice plate was placed on the intake manifold line to measure air consumption. The engine speed was measured with a speed sensor and the flow rate of cooling water was measured with a flow meter. A fuel measurement unit consisting of a 2.5 liter tank and a load cell capable of measuring 0–3 kg were used for the determination of the fuel consumption. Air temperature, inlet and outlet temperatures of the cooling water and exhaust temperature were measured with K-type thermocouples and the values were read from the control panel. Carbon monoxide (CO), carbon dioxide (CO_2) were determined volumetrically (accuracy ± 0.01), and nitrogen oxides (NO) and unburned hydrocarbon (HC) in units of ppm (accuracy ± 1), were measured using an exhaust gas analyzer. Tests were initiated after the engine reached its operating temperature and tests were conducted at 1400 rev/min and 100% load using D100 (the pure diesel fuel), D92B3E5, D85B10E5, D80B15E5 and D75B20E5. The blends rates were selected by reference to previous studies in the literature.

The biodiesel for the present study was produced at the Biodiesel Laboratory of the Department of Agricultural Machinery, Selcuk University. It was produced by transesterification using as raw material safflower, an oilseed crop that contains 20%–45% oil, [23]. Sodium hydroxide was used as catalyst for the transesterification process. Bioethanol as monohydric alcohol was provided by the Sugar Refinery in Konya. Their chemical properties are presented in Table 1. These are obtained by fuels analysis tests carried out by the TUBITAK Marmara Research Center Energy Institute in Gebze.

Table 1. Chemical properties of the tested fuels.

Chemicals		Bioethanol (C_2H_5OH)	Diesel ($C_{12.226}H_{23.29}$)	Biodiesel ($C_{19}H_{35.64}O_2$)
By weight (%)	C	52	85.29	77.06
	H	13	13.64	12.13
	S	-	1.07	-
	O	35	-	10.81
Density (kg/m^3), at 15 °C		794	834.5	885.6
Kinematic viscosity (mm^2/s), at 40 °C		1.1	2.794	4.353
Content of fatty acid methyl ester (%, m/m)		-	-	97.24
Lower heating value (MJ/kg)		28.4	43.14	38.59
Cetane number		-	55.2	55.7
Water content (mg/kg)		170	70	400
Acid number (mg KOH/g)		-	<0.1	0.15
Iodine number (g iodine/100 g)		-	-	117
Flash point (°C)		-	68.5	156.5
Oxidation stability (h), at 110 °C		>1	2	0.31
Carbon residue (%, m/m)		0.007	<0.1	0.2
CFPP (°C)		-	−14	−8
Copper strip corrosion		1a	1a	1a
Cloud point (°C)		-	−12	−4
Oxygen content (%, m/m)		34.91	-	10.32
Purity (%)		99.8	-	-

2.1. Energy Analysis

Energy analysis provides a calculation of the internal energy variation as a function of energy transfers across the boundaries as heat or work and the enthalpy which is related to the mass flow passing these boundaries [7]. Before applying it to the test engine, the following assumptions must be made to simplify the calculations:

- The engine is in steady-state condition.
- It is assumed that the system is an open system and reference state is defined as T_0 = 293 K and P_0 = 1 atm.
- The combustion air and exhaust gases are ideal gas mixtures.
- The potential and kinetic energy effects of the incoming fluid streams and outgoing fluid streams are ignored [2].

Mass balance for the control volume in steady state condition can be written as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

where $\dot{m}_{in}$ represents the inlet flow rate of the mass that consists of air and fuel and $\dot{m}_{out}$ represents the of the outlet mass consisting of exhaust gases considering that fuel enters the engine, with mass flow rate $\dot{m}_{fuel}$ and is mixed with air $\dot{m}_a$ to form $\dot{m}_{in}$.

Energy balance for the control volume in steady state conditions—kinetic and potential energy being neglected—is given by Equation (2) in the general sense:

$$\dot{Q} + \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \tag{2}$$

where the subscripts *in* and *out* point out the incoming-outgoing fluid streams respectively, $\dot{m}$ is the mass flow rate; h is the specific enthalpy. In addition, $\dot{Q}$ indicates the net heat transfer rate and $\dot{W}$ indicates the brake power. When the energy balance is revised for the test engine by considering that the engine generates brake power and some heat produced by combustion is transferred to the environment, Equation (3) can be written, where $\dot{E}_{fuel}$ is the fuel energy rate, $\dot{Q}_{exh}$ is the exhaust energy rate, $\dot{Q}_{lost}$ is the lost energy rate which is transferred to the environment by other ways such as cooling, lubricating, heat transfer except for the exhaust:

$$\dot{E}_{fuel} = \dot{W} + \dot{Q}_{exh} + \dot{Q}_{lost} \tag{3}$$

To determine the fuel energy rate, 43144.9 kJ/kg, 42271.0 kJ/kg, 41952.2 kJ/kg, 41724.4 kJ/kg, 41496.7 kJ/kg are used as the lower heating values for D100, D92B3E5, D85B10E5, D80B15E5 and D75B20E5, respectively.

The brake power is denoted as:

$$\dot{W} = \omega\tau \tag{4}$$

where ω is the angular velocity; and τ is the engine torque.

Since the combustion air is in the same state as the standard reference state defined as T_0 = 293 K and P_0 = 1 atm, the energy of the combustion air can be neglected so the heat input rate to the control volume by mass is represented by only chemical energy of fuel. $\dot{E}_{fuel}$ can be calculated using the mass flow rate of the fuel $\dot{m}_{fuel}$ and lower heating value of the fuel H_u as expressed by Equation (5):

$$\dot{E}_{fuel} = \dot{m}_{fuel} H_u \tag{5}$$

Exhaust energy rate, $\dot{Q}_{exh}$ can be calculated as a function of the mass flow rate of each exhaust gas components $\dot{m}_{exh}$ which is obtained using combustion equations and the enthalpy change Δh represented by the difference between the enthalpy of the exhaust temperature and the reference temperature of each exhaust gas species as given in Equation (6). The mass flow rate and mole fractions of each exhaust component, and the chemical reactions of combustion were obtained by using molecular weight and emission measurement results which were converted into g/kWh units. Multipliers (35.91, 63.47) were used at the stage of converting from % unit to the g/kWh unit of CO and CO_2 emissions [24]:

$$\dot{Q}_{exh} = \dot{m}_{exh} \Delta h \tag{6}$$

Part of the energy obtained by burning the fuel passes into the cooling water and lubricating oil by heat transfer. The other part of it is discharged through the exhaust. The lost energy rate $\dot{Q}_{lost}$ consists of all that energy losses can be evaluated by substituting other parameters in Equation (3).

The engine characteristics are essentially expressed by thermal efficiency η and brake specific fuel consumption *bsfc*. While thermal efficiency is determined as the ratio of brake power to the fuel energy rate of the control volume, *bsfc* is defined as a measure of how much fuel is consumed in one hour to obtain one kilowatt brake power and they are denoted as in Equations (7) and (8):

$$\eta = \frac{\dot{W}}{\dot{E}_{fuel}} \tag{7}$$

$$bsfc = \frac{\dot{m}_{fuel}}{\dot{W}} \tag{8}$$

2.2. Exergy Analysis

Exergy has been described by scientists in many different forms. It was first defined by Baehr as the part of the energy converted into other forms of energy. Amore quantitative and detailed description of exergy corresponds to Bosnjakovic, who expressed it as the maximum amount of work that can be obtained by a reversible process, if the system goes to equilibrium with the surroundings [25]. Unlike energy, which is conserved during any process, exergy is destroyed because of irreversibilities such as combustion, heat transfer through a finite temperature difference, air-fuel mixing, throttling, friction, etc. [1,4,10].

According to Kotas [26], even though exergy analysis is similar to energy analysis in characteristics, there are some fundamental differences between them. While the energy analysis base on the law of energy conservation, exergy analysis is based on the law of degradation of energy. Degradation of energy means the loss of exergy stems from the irreversibilities mentioned in the preceding paragraph [26]. Using only energy analysis is not adequate to analyses energy utilization processes and additionally it leads to poor decisions about them [4,21]. Exergy analysis, more complicated than energy analysis, determines the portion of energy which cannot be used and it states where the losses occur in a system [1,4]. By applying it to a thermal system in addition to the energy analysis, irreversibilities could be detected and so the best ways to minimize energy destruction, losses and increase efficiency can be understood.

The assumptions made for energy analysis is also applicable for exergy analysis. Based on these assumptions, the exergy balance for the control volume can be given in the general sense by Equation (9):

$$\dot{E}x_Q + \dot{E}x_W + \sum \dot{m}_{in}\varepsilon_{in} - \sum \dot{m}_{out}\varepsilon_{out} - \dot{E}x_{dest} = 0 \tag{9}$$

where $\dot{E}x_Q$ is the exergy transfer rate related to heat transfer between the control volume and the environment; $\dot{E}x_W$ is the exergy transfer rate associated with work transfer; $\dot{E}x_{dest}$ is the exergy destruction rate of the control volume. Furthermore, $\dot{m}_{in}\varepsilon_{in}$ and $\dot{m}_{out}\varepsilon_{out}$ are the exergy transfer through the intake and exhaust process where ε_{in}, ε_{out} are the specific exergies of the fuel and exhaust gas, and similarly $\dot{m}_{in}$, $\dot{m}_{out}$ are the mass flow rates of the fuel and exhaust gas.

Exergy input rate with mass transfer to the engine consists of fuel exergy and combustion air exergy. However the effect of combustion air exergy can be neglected by assuming that combustion air enters the engine at ambient conditions. The specific chemical exergy of fuel is computed by multiplying the lower heating value and chemical exergy factor as in the following equation:

$$\varepsilon_{fuel}^{ch} = H_u \varphi \tag{10}$$

The chemical exergy factor for liquid fuels can be calculated by Equation (11), where h, c, o and α are the mass fractions of H, C, O and S, respectively [26]:

$$\varphi = 1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{\alpha}{c}(1 - 2.0628\frac{h}{c}) \tag{11}$$

The exergy transfer rate associated with work is equal to the net work for the engine:

$$\dot{E}x_W = \dot{W} \tag{12}$$

Exergy output rate with mass transfer from the engine consists of exhaust exergy which can be expressed as in Equation (13):

$$\dot{E}x_{exh} = \dot{m}_{exh}\varepsilon \tag{13}$$

Exhaust exergy per unit mass can be expressed as the sum of the specific thermo mechanical and chemical exergies of the exhaust gases as seen in Equation (14). Using the assumption that the exhaust gas is an ideal gas mixture, the specific thermo mechanical and chemical exergies of the exhaust gas containing n components can be calculated by Equations (15) and (16), respectively. The term T_a in these equations represents the ambient temperature and it has been obtained approximately as 19 °C, 23 °C, 20 °C, 21 °C, 21 °C for D100, D92B3E5, D85B10E5, D80B15E5, respectively.

$$\varepsilon = \varepsilon^{tm} + \varepsilon^{ch} \tag{14}$$

$$\varepsilon^{tm} = \sum_{i=1}^{n} a_i \left\{ (\overline{h_i} - \overline{h_{0i}}) - T_a(\overline{s_i} - \overline{s_{0i}}) \right\} \tag{15}$$

$$\varepsilon^{ch} = \overline{R}T_a \sum_{i=1}^{n} a_i \ln \frac{y_i}{y_i^e} \tag{16}$$

where s is the specific entropy and h is the specific enthalpy of the exhaust gas and $\overline{R}$ is the universal gas constant, y_i is the molar fraction of the component in the exhaust gas and y_i^e is the molar fraction of the component in the exhaust gas for the reference environment. Subscript "0" represents the reference state values. It is accepted that the reference state, defined as T_0 = 25 °C and P_0 = 1 atm, consists of N_2, O_2, CO_2, H_2O and other components, the molar fractions of which are 75.67%, 20.35%, 0.03%, 3.12%, and 0.83%, respectively [27].

Exergy transfer rate related to heat transfer can be given by Equation (17) by using Q_{lost} which is a parameter of the energy analysis:

$$\dot{E}x_Q = \sum (1 - \frac{T_0}{T_{cw}})\dot{Q}_{lost} \tag{17}$$

where, T_{cw} is the cooling water temperature considered to be the same as the system boundary temperature. Exergy destruction rate of the control volume $\dot{E}x_{dest}$ can be evaluated by replacing other parameters in Equation (9).

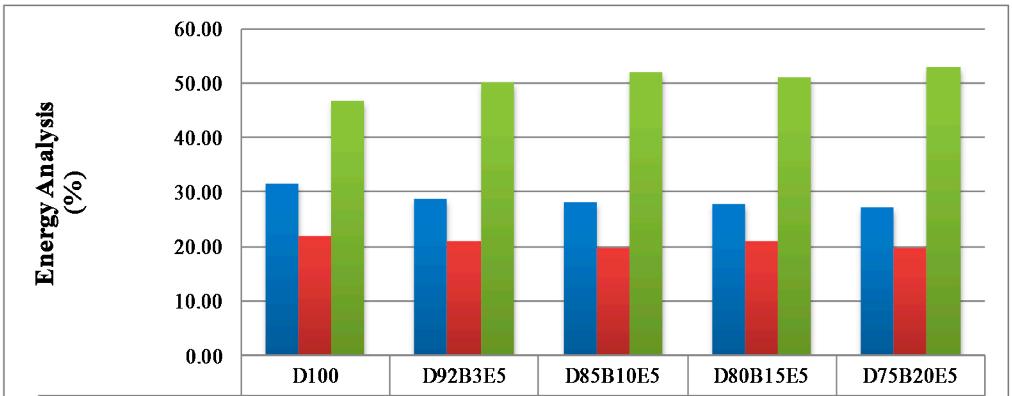
The exergetic efficiency is a more accurate measurement of the system performance compared to the first law efficiency and it answers the question "how much of the fuel exergy is converted into power". The exergetic efficiency, expressed as the ratio of brake power exergy to the fuel exergy of control the volume, can be calculated as follows:

$$\eta_{II} = \frac{\dot{E}x_W}{\dot{E}x_{fuel}} \tag{18}$$

3. Results and Discussion

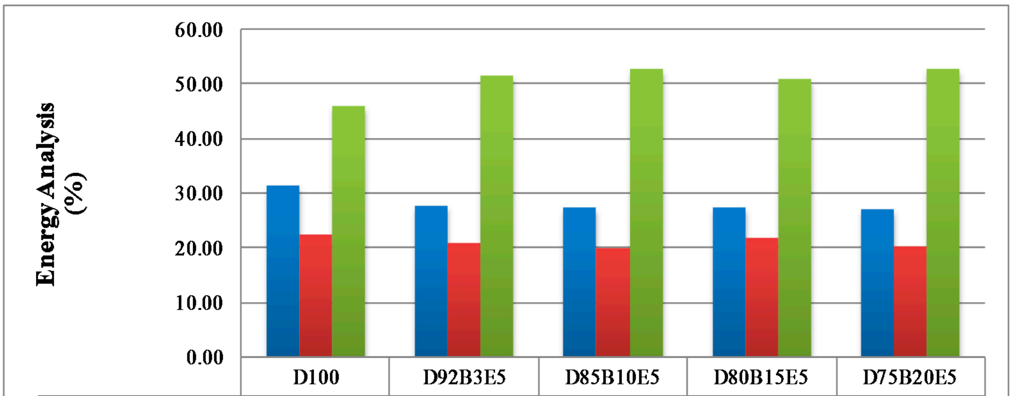
3.1. Energy Analysis

The energy analysis results for the D100 reference fuel and the D92B3E5, D85B10E5, D80B15E5, D75B20E5 fuel blends at 1400 rev/min, at 1500 rev/min and at 2800 rev/min maximum power speed are shown in Figures 3–5 (1500 rev/min maximum torque speed for D100 and 1400 rev/min maximum torque speed for the others). Fuel energy rate, brake power, exhaust energy rate, lost energy rate all form part of the energy distribution. These figures are given to provide insight into how the energy distribution for the test engine is realized in a general sense without indicating the energy distribution of all engine speeds. At 1400 rev/min 31.42% of the fuel energy is converted to brake power, 46.75% lost by the engine to the environment, 21.83% is expelled by the exhaust gases for D100. Considering other fuels at the same speed, it is seen that the conversion rate of fuel energy to brake power has a gradually decreasing trend. This ratio is the expression of thermal efficiency, and has been calculated as 28.68%, 28.1%, 28%, and 27.18% for D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively. Similarly, at 1500 rev/min 31.50%, 27.71%, 27.42%, 27.37%, 25.11% and at 2800 rev/min 27.62%, 26.15%, 25.45%, 25.02%, 24.13% of the fuel energy is converted to brake power. These results show that operation with biodiesel blends leads to less brake power for the same fuel energy rate since biodiesel has a lower heating value than diesel.



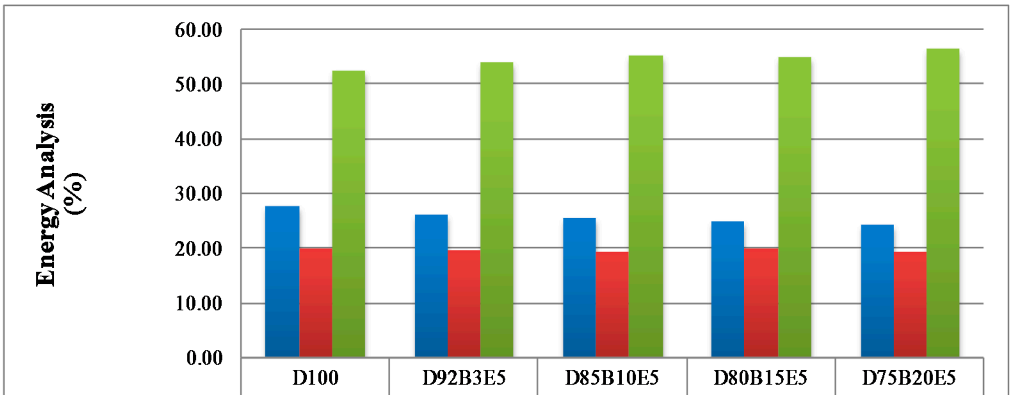
	D100	D92B3E5	D85B10E5	D80B15E5	D75B20E5
Brake Power	31.42	28.68	28.10	28.00	27.18
Exhaust Energy Rate	21.83	21.15	19.92	20.99	19.93
Lost Energy Rate	46.75	50.17	51.99	51.01	52.89

Figure 3. Energy distribution of tested fuels at 1400 rev/min.



	D100	D92B3E5	D85B10E5	D80B15E5	D75B20E5
Brake Power	31.52	27.72	27.40	27.35	26.97
Exhaust Energy Rate	22.47	20.84	19.92	21.72	20.28
Lost Energy Rate	46.01	51.44	52.68	50.92	52.76

Figure 4. Energy distribution of tested fuels at 1500 rev/min.



	D100	D92B3E5	D85B10E5	D80B15E5	D75B20E5
Brake Power	27.62	26.15	25.45	25.02	24.13
Exhaust Energy Rate	19.99	19.71	19.45	20.08	19.39
Lost Energy Rate	52.39	54.13	55.10	54.90	56.49

Figure 5. Energy distribution of tested fuels at 2800 rev/min.

Fuel energy rate which is one of the energy distribution parameters is related to fuel density (molecular mass of fuel), the amount of displaced air due to the volume of the fuel vapour, lower heating value, and thermal efficiency [6]. As shown in Figure 6, the highest fuel energy rate belongs to D100 at 1600 rev/min and at subsequent speeds. When evaluated in terms of speeds, fuel energy rate increases as long as the engine speed increases for all fuel. The average values of the fuel energy rate are 22.81 kW, 22.21 kW, 22.17 kW, 21.93 kW, 22 kW for D100, D92B3E5, D85B10E5, D80B15E5, D75B20E5, respectively.

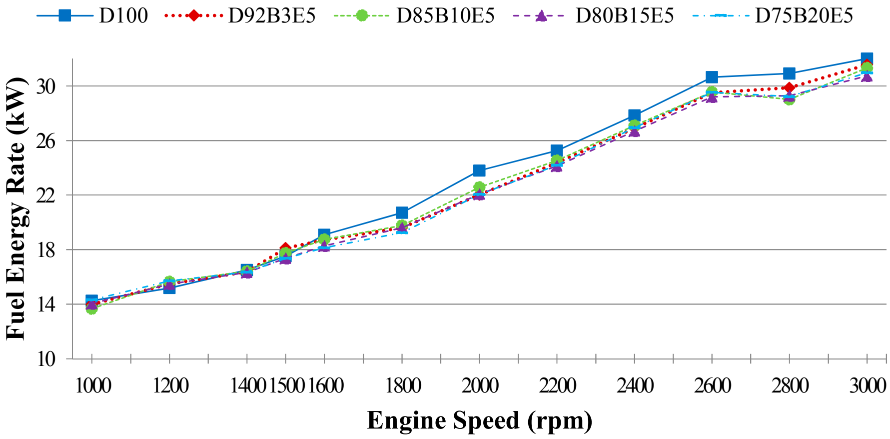


Figure 6. Variation of fuel energy rate with engine speed.

The highest brake power has been obtained when operating with diesel fuel at all speeds as shown in Figure 7. This is explained by the fact that engine torque is higher for diesel fuel than other fuels. We can say that the lower viscosity of diesel causes higher engine torque values. The average torque values are 32.46 Nm, 29.27 Nm, 28.41 Nm, 27.84 Nm, 27.02 Nm for D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively. When comparing fuels other than diesel, it has been determined that brake power decreases as the biodiesel content in fuel increases so the highest power has been obtained for D92B3E5.

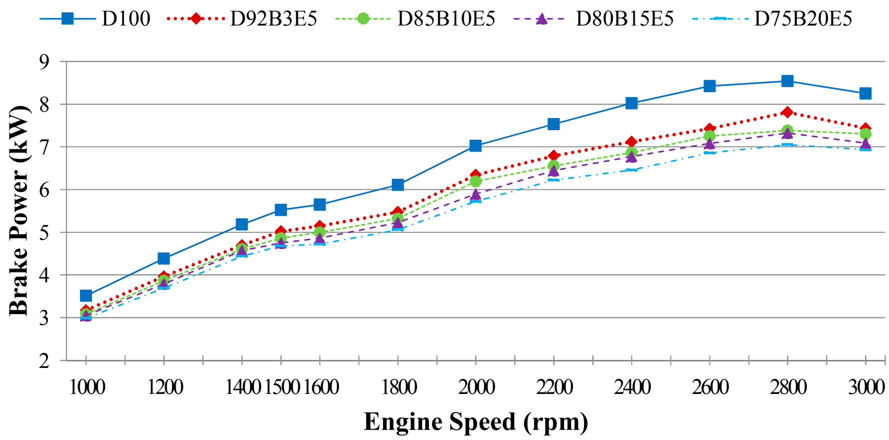


Figure 7. Variation of brake power with engine speed.

Exhaust energy rate, which is the measure of the heat energy of the exhaust gases, increases as the exhaust gas temperature rises. The average exhaust gas temperatures of D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5 are 465.11 °C, 460.45 °C, 447.35 °C, 456.25 °C, and 453.64 °C. When low engine speeds are examined, it is not possible to determine the fuel which has the lowest exhaust energy rate but at 1800 rev/min and at following engine speeds, the minimum exhaust energy rate has been obtained for D85B10E5, as shown in Figure 8. In order to increase the energy efficiency of the system, the exhaust energy can be recovered.

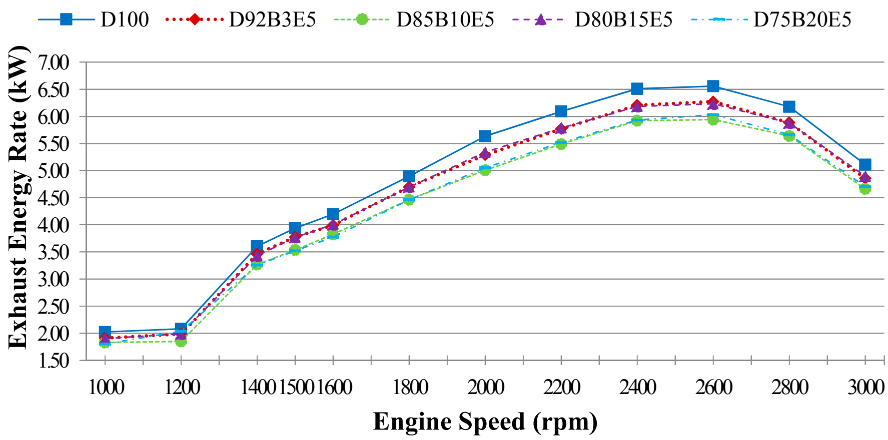


Figure 8. Variation of exhaust energy rate with engine speed.

Figure 9 shows the lost energy rate by the engine to the environment excluding exhaust. We can see that the lost energy rate increases with the engine speed. Fuels containing biodiesel give higher lost energy rates from the engine. When the engine is fueled with biodiesel blends, it rejects on average 2.31% for D92B3E5, 5.22% for D85B10E5, 2.47% for D80B15E5, 5.95% for D75B20E5 higher energy lost to the atmosphere compared to the D100.

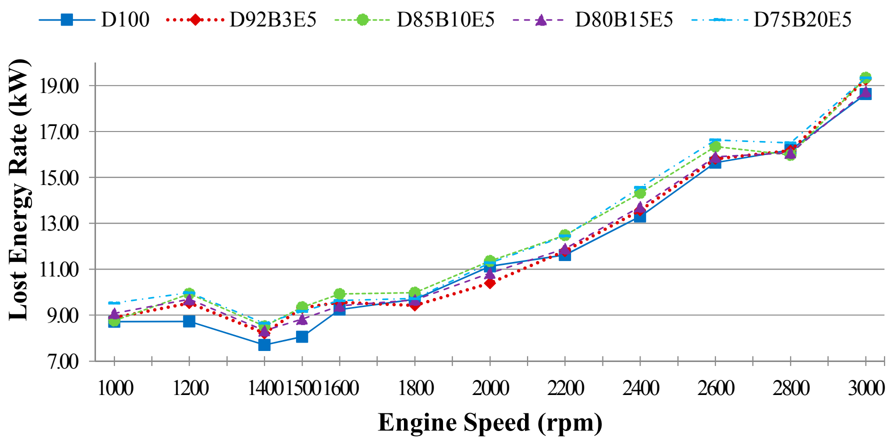


Figure 9. Variation of lost energy rate with engine speed.

Variation of thermal efficiency as a function of fuel composition and engine speed is shown in Figure 10. If the engine runs with D100, we can see that thermal efficiency initially increases to a certain point which is maximum torque speed and then it decreases significantly. This decrease of thermal efficiency despite the increasing engine speed value can be explained by mechanical friction in the engine, the inertial forces which must be met for rotating parts. Reduction of the combustion time as engine speed increases is also a key factor of this case. Thermal efficiency has the highest value for D100 at all speeds. When compared other fuels, it has been observed that the thermal efficiency has a downward trend with the increase in the biodiesel content of the fuels. Among them the thermal efficiency shows the highest value for D92B3E5, and the lowest value for D75B20E5. The engine operation with biodiesel blends yields approximately 7.68%, 10.32%, 11.22% and 14.08% lower thermal efficiency compared to D100. This decrease in thermal efficiency as biodiesel content increases in fuel is due to the lower calorific value of biodiesel.

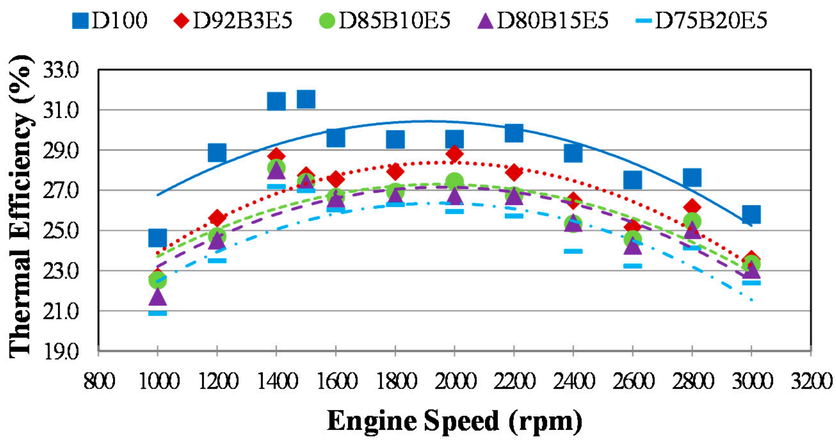


Figure 10. Variation of thermal efficiency with engine speed.

Brake specific fuel consumption is defined as the ratio of the mass flow rate of fuel to the engine brake power. The variation dependence on engine speed and fuels is given in Figure 11. Specific fuel consumption is inversely proportional to the lower heating value of a fuel, so it has the highest value for D100 at all speeds. It is also associated with the lower density of diesel fuel then biodiesel. The amount of fuel required to obtain the same amount of brake power from the engine increases when using biodiesel blends because a rise in the biodiesel content decreases the lower heating value. It is thus an expected result that D75B20E5 has the highest specific fuel consumption than the others blends. As it is seen there has been a minimum of *bsfc* at 1400 rev/min for all fuels. At this engine speed, compared to D100, specific fuel consumption is approximately 12% more for D92B3E5, 15% for D85B10E5, 16% for D80B15E5 and 20% for D75B20E5.

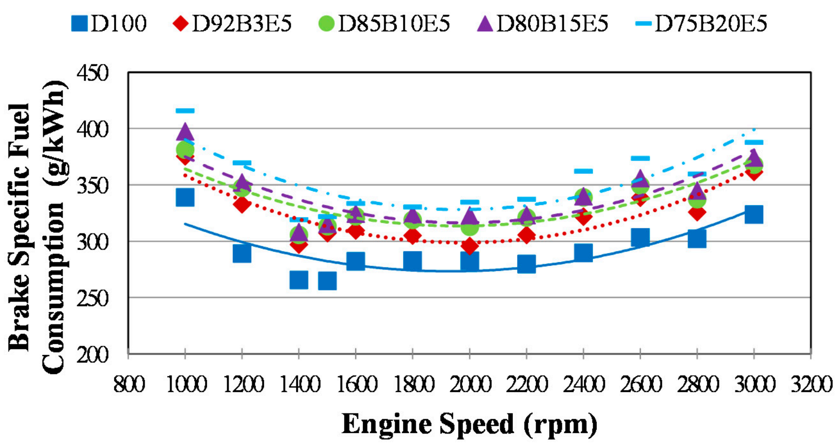
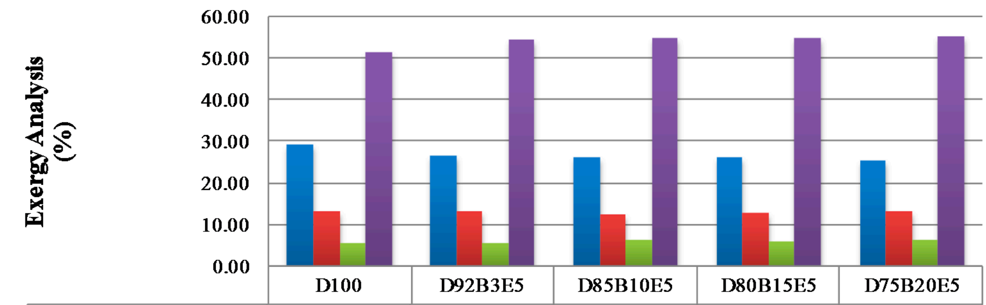


Figure 11. Variation of brake specific fuel consumption with engine speed.

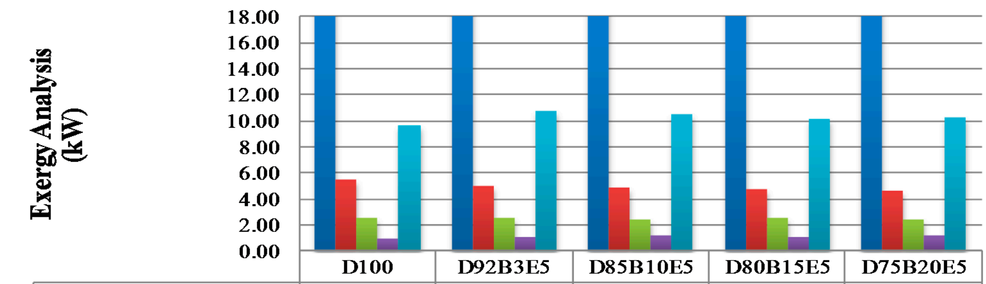
3.2. Exergy Analysis

The result of exergy analysis for D100, and D92B3E5, D85B10E5, D80B15E5, and D75B20E5 fuel blends at 1400, 1500, 2800 rev/min is shown in Figures 12–14. Fuel exergy rate, brake power exergy, exhaust exergy rate, the exergy rate through heat transfer and exergy destruction rate constituting the exergy distribution for the fuels used in the study are shown in them. For D100 at 1400 rev/min 29.34% of the fuel exergy is converted to brake power, 5.55% is lost through heat transfer, 13.42% is expelled by the exhaust gases, and 51.55% is lost by exergy destruction.



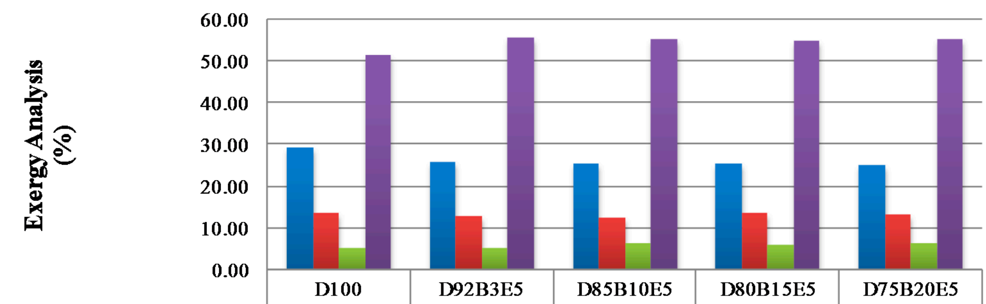
	D100	D92B3E5	D85B10E5	D80B15E5	D75B20E5
Power Exergy	29.38	26.80	26.33	26.15	25.38
Exhaust Exergy Rate	13.45	13.14	12.48	12.97	13.11
Exergy Rate Through Heat Transfer	5.58	5.59	6.30	6.03	6.38
Exergy Destruction Rate	51.60	54.46	54.90	54.85	55.13

Figure 12. Exergy distribution of the tested fuels at 1400 rev/min.



	D100	D92B3E5	D85B10E5	D80B15E5	D75B20E5
Fuel Exergy Rate	18.74	19.38	18.94	18.58	18.60
Power Exergy	5.52	5.02	4.86	4.75	4.68
Exhaust Exergy Rate,	2.58	2.53	2.37	2.50	2.49
Exergy Rate Through Heat Transfer	1.00	1.04	1.24	1.14	1.15
Exergy Destruction Rate	9.64	10.80	10.47	10.19	10.27

Figure 13. Exergy distribution of the tested fuels at 1500 rev/min.



	D100	D92B3E5	D85B10E5	D80B15E5	D75B20E5
Power Exergy	29.47	25.91	25.67	25.55	25.18
Exhaust Exergy Rate	13.79	13.04	12.54	13.46	13.37
Exergy Rate Through Heat Transfer	5.32	5.35	6.54	6.14	6.21
Exergy Destruction Rate	51.43	55.70	55.25	54.85	55.24

Figure 14. Exergy distribution of tested fuels at 2800 rev/min.

Considering the other fuels at the same speed, it is seen that the conversion rate of fuel exergy to the brake power has a gradually decreasing trend. This ratio, the expression of exergetic efficiency, has been calculated as 26.82%, 26.34%, 26.12%, and 25.37% for D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively. Similarly, at 1500 rev/min 29.45%, 25.90%, 25.65%, 25.56%, 25.16% and at 2800 rev/min 25.82%, 24.44%, 23.83%, 23.35%, 22.53% of the fuel exergy is converted to brake power. These results show that operation with biodiesel blends lead to less power for the same fuel exergy rate since biodiesel has a lower heating value than diesel.

Fuel exergy rate is a parameter which has a similar trend as the fuel energy rate because both are functions of the mass flow rate and lower heating value of fuel. Chemical exergy factor, which is another dependent variable of fuel exergy rate, is greater than one. This explains why the fuel exergy rate has a greater value than the fuel energy rate. Fuel exergy rates are about 6.95%, 7.01%, 6.73%, 7.07%, and 7.1% times higher than the fuel energy rate for D100, D92B3E5, D85B10E5, D80B15E5, D75B20E5, respectively. As shown in Figure 15, the highest fuel exergy rate corresponds to D100 at 1600 rev/min and at subsequent speeds as in the fuel energy rate. When evaluated in terms of speeds, fuel exergy rate increases as long as engine speed increases for all fuels. The average values of the fuel exergy rate are 24.39 kW, 23.76 kW, 23.66 kW, 23.48 kW, and 23.56 kW for D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively.

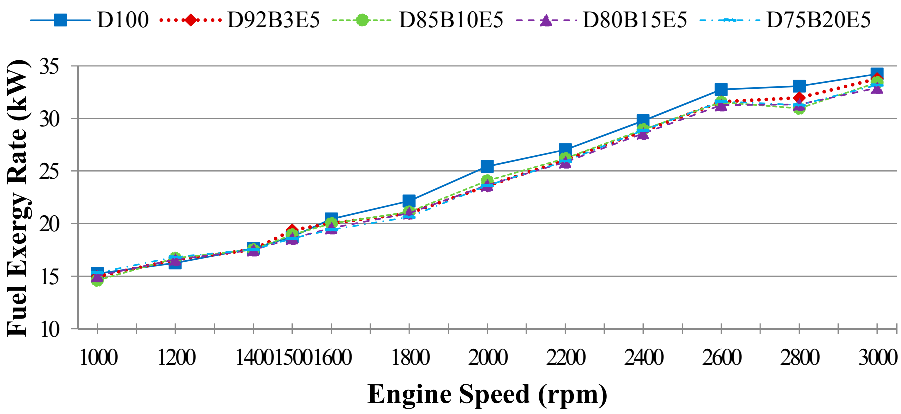


Figure 15. Variation of fuel exergy rate with engine speed.

Exhaust exergy rate, which is the expression of the useful part of heat energy carried by the exhaust gas, reaches the highest value for D100 and the lowest value for D85B10E5, as shown in Figure 16. It was detected that exhaust exergy rate is affected by the exhaust gas temperature. Exergy rate through heat transfer has a similar trend with lost energy rate. It is also related to the cooling water temperature which has instabilities. The average values of exhaust energy rate are 3.19 kW, 3.07 kW, 2.93 kW, 3.03 kW, and 3.1 kW for D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively.

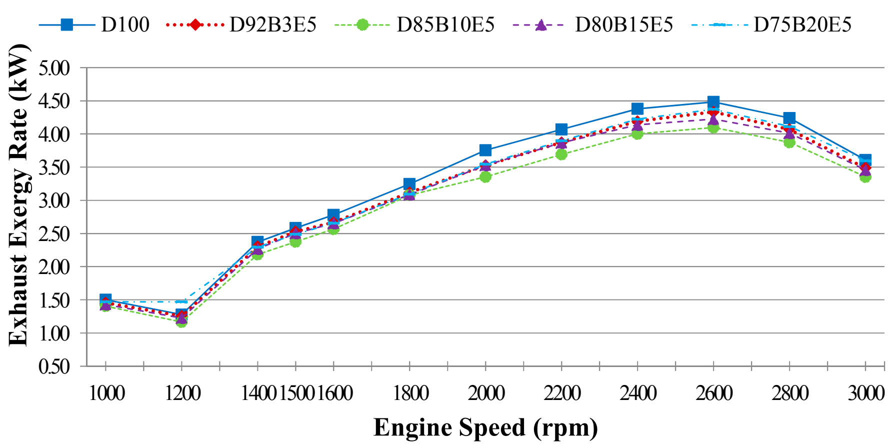


Figure 16. Variation of exhaust energy rate with engine speed.

Figure 17 shows the exergy rate through heat transfer at different engine speeds for D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5. It is expected that the exergy rate through heat transfer will have similar trends as the lost energy rate. When the lost energy rate examined, it increases depending on the engine speed and it takes its lowest value for D100 fuel. Exergy rate through heat transfer also takes its lowest value for D100 at low engine speeds. At high engine speeds, it is not possible to determine which fuel has the lowest value of this factor.

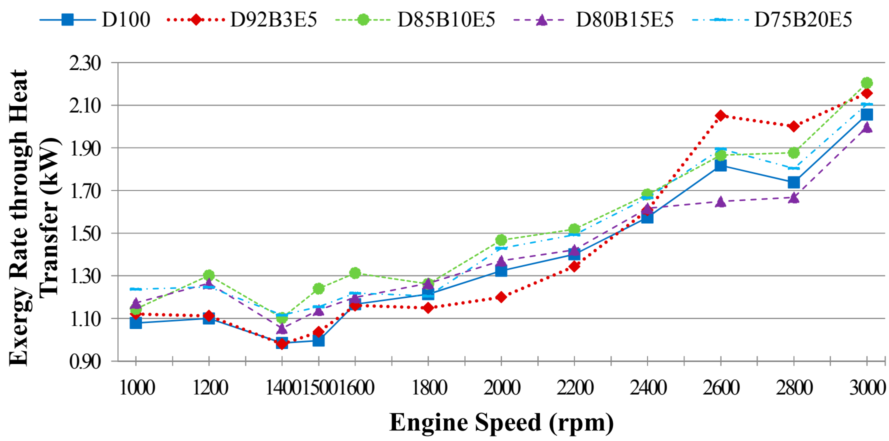


Figure 17. Variation of exergy rate through heat transfer with engine speed.

Exergy destruction rate is the result of a number of irreversible processes such as mixing of the different reactants during the chemical reactions, heat transfer between molecules at different temperatures, friction and combustion processes, expansion of gases, turbulent flow inside the cylinder, etc. [10]. It depends on many factors such as mixing the gases at different temperatures, turbulence, flow losses, and mixing of the fuel-air mixture in the intake valve with residual gas in the internal combustion engine [28]. Exergy destruction rate is a term that is quite complicated as it is affected by many factors, and it increases as the engine speed increases as shown in Figure 18. The average values are 13.32 kW, 13.42 kW, 13.55 kW, 13.48 kW, 13.6 kW for D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively.

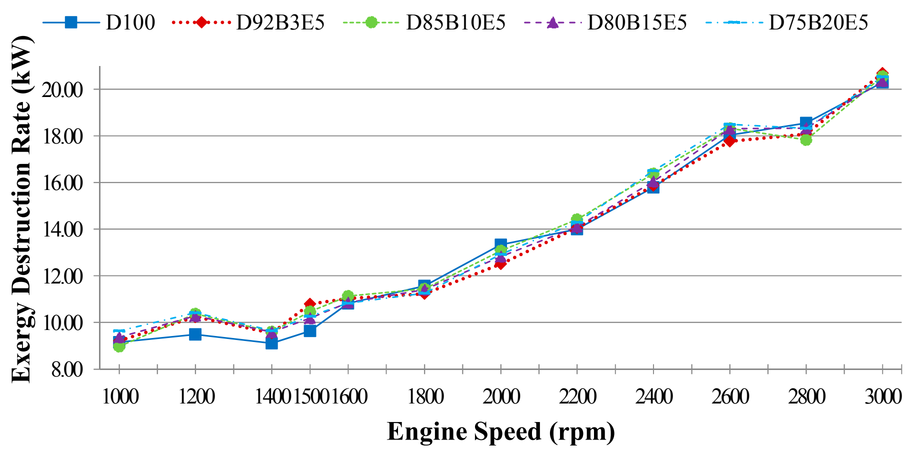


Figure 18. Variation of exergy destruction with engine speed.

Uncertainty of measuring instruments used for the experiments and some parameters of energy and exergy analyses are presented in Table 2. These parameters are brake power, thermal efficiency and exergetic efficiency. Uncertainty values calculated for these three parameters has been obtained in the case of an engine operating at 1400 rev/min with D100.

Table 2. Uncertainty of measuring instruments and some parameters of analyses.

	Parameter	**Unit**	**Measuring Instruments**	**Uncertainty**
Uncertainty of measuring instruments	Temperature	°C	Thermometer	1 °C
	Mass flow	g/s	Load cell	±0.01 g/s
	Torque	Nm	Dynamometer	±0.0001
	Engine speed	rev/min	Speed sensor	±1
	Parameter	**Unit**	**Value**	**Uncertainty**
Uncertainty of some parameters of analyses	Brake power, $\dot{W}$	kW	5.1841	±0.0037
	Thermal efficiency, η	-	0.314	±0.008
	Exergetic efficiency, η_{II}	-	0.294	±0.007

Second law efficiency, or in other words, exergetic efficiency is a more accurate measure of the performance of the system from the point of view of thermodynamics and it is more expressive, objective and useful compared to the first law efficiency [27]. By comparing the curves of both with each other, it can be seen that the exergetic efficiency curves present a similar trend as the thermal efficiency curves, but exergetic efficiency takes lower values under the same conditions. The exergetic efficiency is approximately 6.5%, 6.55%, 6.31%, 6.61%, 6.63% lower than thermal efficiency for D100, D92B3E5, D85B10E5, D80B15E5, and D75B20E5, respectively. When evaluated in terms of fuel, D100 gets the maximum value of exergetic efficiency for all speeds as seen in Figure 19, so if exergy is considered as a measure of quality, D100 is a better quality fuel than others.

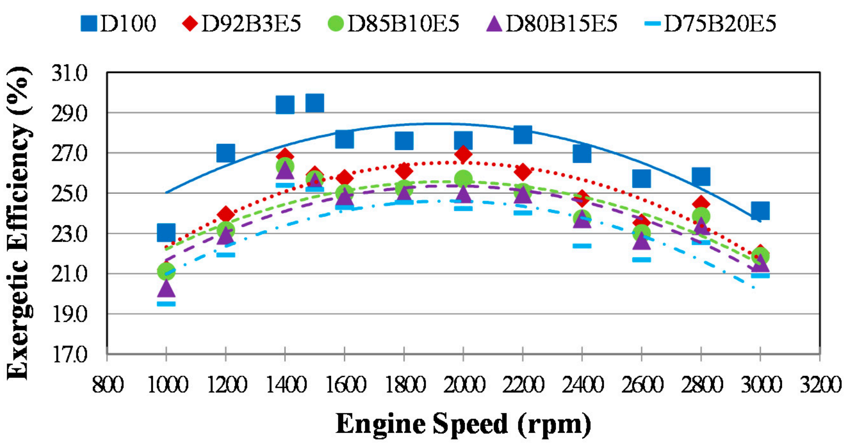


Figure 19. Variation of exergetic efficiency with engine speed.

4. Conclusions

This study examines the energy and exergy analysis of a diesel engine using D92B3E5 (92% diesel, 3% biodiesel and 5% bioethanol), D85B10E5 (85% diesel, 10% biodiesel and 5% bioethanol), D80B15E5 (80% diesel, 15% biodiesel and 5% bioethanol), D75B20E5 (75% diesel, 20% biodiesel and 5% bioethanol) and D100 reference fuel. The fuel energy rate, exhaust energy rate, lost energy rate, brake power with first law analysis, fuel exergy rate, brake power exergy, exhaust exergy rate, the exergy rate through heat transfer and exergy destruction rate with exergy analysis were calculated for each fuel using results of tests performed at different speeds (from 1000 rev/min to 3000 rev/min in intervals of 200 rev/min) and then compared with each other. Additionally thermal efficiency and exergetic efficiency were determined. The results of this study can be summarized as follows:

- Highest brake power, highest thermal efficiency and highest exergetic efficiency for all engine speeds have been obtained for D100 and as biodiesel content in the fuel increases, each tends to decrease.
- The nearest thermal and exergetic efficiency to D100 is obtained for D92B3E5 which has 3% biodiesel.
- The thermal and exergetic efficiency for D92B3E5 is closer than others to that of D100, so D92B3E5 can be used instead of D100 because many features of the fuels used in this study are quite close to each other.
- The most important contributor to the system inefficiency is the destruction of exergy by irreversible processes, mainly caused by the combustion. Exergy losses due to the exhaust gas and heat transfer are other contributors in decreasing order. By concentrating further studies on these items we may be able to determine how the engine can utilize the fuel energy better.
- Application of exergy analysis in addition to the energy analysis led us to obtain more realistic and more accurate results.
- Engine brake power has only net potential to produce useful work.

Acknowledgments: This work was supported by the Coordination Office for Research Projects (BAP) of Necmettin Erbakan University.

Author Contributions: Ali Kahraman put forward the original ideas and performed the research. Bahar Sayin Kul designed and prepared the text and graphics of the manuscript. Ali Kahraman reviewed the paper and provided useful comments. All authors have read and approved the final manuscript.

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

References

1. Cengel, Y.A.; Boles, M.A. *Thermodynamics: An Engineering Approach*; McGraw-Hill College: Boston, MA, USA, 2006.
2. Tat, M.E. Cetane number effect on the energetic and exergetic efficiency of a diesel engine fuelled with biodiesel. *Fuel Process. Technol.* **2011**, *92*, 1311–1321. [CrossRef]
3. Sorathia, H.S.; Yadav, H.J. Energy analyses to a CI-engine using diesel and bio-gas dual fuel—A review study. *Int. J. Adv. Eng. Res. Stud.* **2012**, *1*, 212–217.
4. Da Costa, Y.J.R.; de Lima, A.G.B.; Filho, C.R.B.; de Araujo Lima, L. Energetic and exergetic analyses of a dual-fuel diesel engine. *Renew. Sustain. Energy Rev.* **2012**, *16*, 4651–4660. [CrossRef]
5. López, I.; Quintana, C.E.; Ruiz, J.J.; Cruz-Peragón, F.; Dorado, M.P. Effect of the use of olive–pomace oil biodiesel/diesel fuel blends in a compression ignition engine: Preliminary exergy analysis. *Energy Convers. Manag.* **2014**, *85*, 227–233. [CrossRef]
6. Caton, J.A. Implications of fuel selection for an SI engine: Results from the first and second laws of thermodynamics. *Fuel* **2010**, *89*, 3157–3166. [CrossRef]
7. Abedin, M.J.; Masjuki, H.H.; Kalam, M.A.; Sanjid, A.; Rahman, S.M.A.; Masum, B.M. Energy balance of internal combustion engines using alternative fuels. *Renew. Sustain. Energy Rev.* **2013**, *26*, 20–33. [CrossRef]
8. Murugesan, A.; Umarani, C.; Subramanian, R.; Nedunchezhian, N. Bio-diesel as an alternative fuel for diesel engines—A review. *Renew. Sustain. Energy Rev.* **2009**, *13*, 653–662. [CrossRef]
9. Alptekin, E.; Canakci, M.; Ozsezen, A.N.; Turkcan, A.; Sanli, H. Using waste animal fat based biodiesels–bioethanol–diesel fuel blends in a DI diesel engine. *Fuel* **2015**, *157*, 245–254. [CrossRef]
10. Zheng, J.; Caton, J.A. Second law analysis of a low temperature combustion diesel engine: Effect of injection timing and exhaust gas recirculation. *Energy* **2012**, *38*, 78–84. [CrossRef]
11. Rakopoulos, C.D.; Giakoumis, E.G. Availability analysis of a turbocharged diesel engine operating under transient load conditions. *Energy* **2004**, *29*, 1085–1104. [CrossRef]
12. Rakopoulos, C.D.; Giakoumis, E.G. Comparative first- and second-law parametric study of transient diesel engine operation. *Energy* **2006**, *31*, 1927–1942. [CrossRef]
13. Caton, J.A. On the destruction of availability (exergy) due to combustion processes—With specific application to internal-combustion engines. *Energy* **2000**, *25*, 1097–1117. [CrossRef]

14. Giakoumis, E.G. Cylinder wall insulation effects on the first- and second-law balances of a turbocharged diesel engine operating under transient load conditions. *Energy Convers. Manag.* **2007**, *48*, 2925–2933. [CrossRef]
15. Azoumah, Y.; Blin, J.; Daho, T. Exergy efficiency applied for the performance optimization of a direct injection compression ignition (CI) engine using biofuels. *Renew. Energy* **2009**, *34*, 1494–1500. [CrossRef]
16. Caliskan, H.; Tat, M.E.; Hepbasli, A. Performance assessment of an internal combustion engine at varying dead (reference) state temperatures. *Appl. Therm. Eng.* **2009**, *29*, 3431–3436. [CrossRef]
17. Özkan, M. A Comparative study on energy and exergy analyses of a CI engine performed with different multiple injection strategies at part load: Effect of injection pressure. *Entropy* **2015**, *17*, 244–263. [CrossRef]
18. Sahoo, B.B.; Saha, U.K.; Sahoo, N. Theoretical performance limits of a syngas–diesel fueled compression ignition engine from second law analysis. *Energy* **2011**, *36*, 760–769. [CrossRef]
19. Canakci, M.; Hosoz, M. Energy and exergy analyses of a diesel engine fuelled with various biodiesels. *Energy Sources Part B Econ. Plan. Policy* **2006**, *1*, 379–394. [CrossRef]
20. Rakopoulos, C.D.; Scott, M.A.; Kyritsis, D.C.; Giakoumis, E.G. Availability analysis of hydrogen/natural gas blends combustion in internal combustion engines. *Energy* **2008**, *33*, 248–255. [CrossRef]
21. Sekmen, P.; Yilbasi, Z. Application of energy and exergy analyses to a CI engine using biodiesel fuel. *Math. Comput. Appl.* **2011**, *16*, 797–808. [CrossRef]
22. Rakopoulos, C.D.; Kyritsis, D.C. Comparative second-law analysis of internal combustion engine operation for methane, methanol, and dodecane fuels. *Energy* **2001**, *26*, 705–722. [CrossRef]
23. Karaosmanoglu, F.; Isler, A. Assessment of Safflower Oil Methyl Ester and Its Life Cycle. Ph.D. Thesis, Istanbul Technical University, Istanbul, Turkey, 3 April 2012. (In Turkish)
24. Pilusa, T.; Mollagee, M.M.; Muzenda, E. Reduction of vehicle exhaust emissions from diesel engines using the whale concept filter. *Aerosol Air Qual. Res.* **2012**, *12*, 994–1006. [CrossRef]
25. Ucgul, I.; Koyun, T. Evaluation of First and Second Law Efficiency of Bifacial Cylindrical Reflection and Flat Surface Collectors. *Eng. Mach. (TMMOB-MMO)* **2004**, *530*, 18–24. (In Turkish)
26. Kotas, T.J. *The Exergy Method of Thermal Plant Analysis*; Elsevier: Amsterdam, The Netherlands, 2013.
27. Moran, M.J.; Shapiro, H.N. *Fundamentals of Engineering Thermodynamics*; Wiley: Hoboken, NJ, USA, 2000.
28. Rakopoulos, C.D.; Giakoumis, E.G. *Diesel Engine Transient Operation: Principles of Operation and Simulation Analysis*; Springer: Berlin/Heidelberg, Germany, 2009.