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Exergy Analysis of a CI Engine Operating on Ternary Biodiesel Blends

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Abstract: Exergy analysis is carried out on a single-cylinder CI engine fueled with biodiesel blends of palm, jatropha and cottonseed oils. This is to identify the blends with high exergy destruction. To this end, experimental and analytical methods were adopted. Three types of biodiesel blends incorporated in this study are primary, binary and ternary. The load was varied as an independent parameter, and mass flow rates of air and fuel, flue gas composition, etc., were measured during the study. Moreover, the chemical composition of the fuel blends and flue gas, as well as their flow rates, were used to determine the total exergy. The output parameters determined were 1st and 2nd law efficiency and fuel exergy destruction under all loading conditions. The inference obtained from the experiment suggests minutely higher 1st law efficiency for the biodiesel blends. Increasing the blending ratio led to an increase in efficiency indices.

Keywords: biodiesel; ternary blends; CI engine; energy analysis; exergy analysis

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1. Introduction

Alternative fuels will be at their culmination in the near future, replacing conventional fuels for thermal power generation. For automobile applications, conventional fuels cannot be completely replaced as per the current technological and economic status. However, blending conventional fuel with an alternative fuel appears to be an attractive option to diminish reliance on conventional fuels. Biodiesels are considered to be one of the main alternative fuels as they are easier to synthesize, transport and store. In order to evaluate the performance of engines using fuel blends, energy, exergy, emissions and life span of the engine must be studied. Energy and exergy analyses provide certain relevant data that can be used to compare the energy produced by diesel and biodiesel blends, thus determining the better one among different blends. Moreover, exergy analysis gives a quantitative estimate of the amount of useful energy recovered and, in the process, exposes the processes/components responsible for causing irreversibilities [1]. This information can be used in making a proper choice of the blend, improvement in the process, engine design, etc. Many researchers in the past have performed various investigations on energy and exergy on diesel engines operating on different fuel blends.

Khoobbakht et al. [2] experimented on a diesel engine by varying the load, speed and blend ratio and obtained maximum exergy efficiency of 36.92% for a blend containing 17% biodiesel (derived from waste cooking oil), 8% ethanol and the rest diesel, by volume. SayinKul and Kahraman [3] employed diesel-biodiesel blends with bioethanol acting as an additive, and the blends were tested at different engine speeds between 1000 and 3000 rpm. Biodiesel blends showed higher energy and exergy destruction rates at every engine speed, and fuel exergy rates were observed to rise with an increment in biodiesel concentration in the blend. Maximum exergy efficiency was found to be 29.38% for 92% diesel, 3% biodiesel and 5% bio-ethanol blend. Krishnamoorthi and Malayalamurthi [4] studied emissions and exergy in a CI engine at changing compression ratios and used blends comprising of diesel-Aegle marmelos-diethyl ether with the marmelos oil concentration varied from 0 to 40%, keeping the concentration of diethyl ether constant at 10%. The effect of bael oil in blends affected the combustion and caused high available energy destruction. The maximum exergy efficiency of 63.88% was observed for a blend containing 40% marmelos oil and 50% diesel at a compression ratio of 17.5. Paul et al. [5] used biodiesel blended with ethanol and observed an increment in brake thermal efficiency due to the addition of ethanol. They reasoned that exergy destruction decreased with adding ethanol because of the reduction in thermal gradient between the engine and the surroundings. Hoseinpour et al. [6] conducted research on the fumigation of gasoline in a direct injection diesel engine, and this proved to be advantageous as it led to an increment in exergy and energy efficiencies. Krishnamoorthi and Malayalamurthi [7] extended the discussion on blends containing bael oil, diethyl ether and diesel and observed minimum exergy destruction at a pressure of 250 bar at full load conditions. Moreover, first law efficiency increased with 60% diesel, 30% bael oil and 10% diethyl ether blend, and it also showed higher exergy efficiency of 62.17% at 100% loading. In their research, Odibi et al. [8] employed the use of oxygenated blends of cooking biodiesel and triacetin using different oxygen ratios (ratio of oxygen present in air + fuel to the Stoichiometric oxygen requirement). Oxygenation leads to higher thermal efficiency due to improved combustion. They also found that increased oxygen content is responsible for decreased exhaust exergy losses, and it also leads to a decline in fuel exergy, with triacetin at 10% and 90% biodiesel showing the least fuel exergy. Taghavifar et al. [9] conducted experiments involving a numerical study on the performance of an HCCI incorporating dimethyl-methanol blends. Higher thermo-mechanical energy is observed for diesel (50%), methanol (30%) and Di-methyl ether (20%) blend due to complete oxidation and higher release of chemical energy. Lower irreversibility is observed for diesel (60%), methanol (10%) and Di-methyl ether (30%) blend due to higher in-cylinder temperatures. Ma et al. (2020) used a dual fuel engine with diesel methanol blends and observed lower exergy losses because of high vaporization heat for methanol. Moreover, exhaust losses declined with subsequent increment in intake air temperature, and it also, in turn, led to better exergy efficiency. Das et al. [10] incorporated the use of waste plastic oil with diesel and observed an increase in fuel exergy up to a 20% blending ratio. Further, adding more waste plastic oil (WPO) led to excessive exergy destruction and marginal exergy efficiency improvements. Rangasamy et al. [11] compared two different modes of dual fuel reactivity and conventional diesel mode with oxygenated biofuels and observed lower loss of exergy in exhaust and coolant in dual fuel mode. This, in turn, led to an increase in the brake thermal as well as conversion efficiency. Finally, they concluded that about 9% of high-quality energy could be easily recovered from incomplete combustion. This could be achieved by setting the optimal operating parameters, which would result in an additional 26% recovery of low-quality energy by utilizing waste heat recovery methods. Sanli et al. [12] used micro-algae biodiesel in an exergy analysis at various ambient temperatures to deduce biodiesel blend to be a good substitute for diesel fuel as the difference between exergy efficiency was minute at 1.29%. Moreover, there was a rise in destroyed exergy with augmentation in the ambient temperature, and the maximum destroyed exergy was recorded at 35 °C. In addition, the decline in the exergy rate was observed with rising ambient temperatures.

The open literature shows that several studies were carried out to evaluate the suitability of different fuels and their blends in engines. The choice of the components of the blend was probably driven by their availability. The present work aimed to evaluate the diesel engine performance using palm oil, jatropha oil and cottonseed oil and their blends. To the best of our knowledge, blends of these oils have not been used earlier in diesel engines to carry out detailed energy as well as exergy analysis. The present work further aimed to investigate up to three levels of blending viz. primary, binary and ternary in order to arrive at the best blend based on exergy analysis.

2. Fuel Preparation

Direct use of biodiesel due to its relatively higher density and viscosity leads to injector choking and gum formation. Hence, in order to run the engine with biodiesel, lowering its viscosity was crucial. This was achieved by blending it with diesel which resulted in lower viscosity values. The blends were classified into three types viz. primary, binary and ternary, based on the number of biodiesel incorporated in them. One liter of the mixture was prepared for each blend with varying proportions of biodiesels from 10% to 30% increments of 10% by volume. Thus, a total of 22 blends, including pure diesel, were produced for experimentation. The blends were named after the biodiesel used. Palm oil is named P, jatropha oil is named J and cottonseed oil is CS. Regarding the composition, each blend name is followed by a number indicating the percentage composition of the biodiesel, the rest being diesel. For example, in the case of primary blends, P10 implies 10% palm oil and 90% diesel. In the case of binary blends, JCS20 implies 10% jatropha, 10% cottonseed and 80% diesel. In the case of ternary blends, PJCS30 implies 10% palm, 10% jatropha, 10% cottonseed and 70% diesel.

3. Fuel Properties

The properties of the fuel blends were measured in the laboratory as per standard methods. All the biodiesel that was utilized possessed higher values of a flash point than pure diesel. In addition, as the percentage of biodiesel in the blend increased, the values of flash and fire point correspondingly increased. Cottonseed had the highest value of flash and fire point, followed by P100 and J100 biodiesel. When all other mixed combinations were considered, CS100 depicted the highest flash point, and PCS30 and JCS30 blends were the successors in that order. Regarding the fire point, CS100 again depicted the highest value, followed by PCS30 and PCS20. Moreover, pure diesel possessed the lowest flash point and fire point among all the blends. The density values depict that biodiesels had a higher density compared to diesel. CS100 had the highest density, with PCS30 and P100 following it. On the contrary, PJ10 depicted the lowest density, followed by J10 and pure diesel. Giakoumis and Sarakatsanis [13] experimentally concluded that there was a link between the density and the degree of unsaturation of the biodiesel as there was a surge in density value with an increase in the double bonds. Similarly, in the experiment conducted by Ramírez-Verduzco et al. [14], it was found that with a decrease in the molecular mass and increasing degree of unsaturation, the density of biodiesel was observed to increase. Biodiesels are having hiher viscosity than diesel [15]. As depicted in Table 1, it can be seen that biodiesels have a higher viscosity than diesel. PJ10 had the highest calorific value, and the next higher values were depicted by J10 and P10 blends, while CS30 possessed the lowest calorific value, trailed by P30.

Table 1. Properties of biodiesel blends.

Blends/Properties	Flash Point, °C	Fire Point, °C	Density, kg/m^3	Viscosity, mm^2/s @40 °C	Calorific Value, MJ/kg
Diesel-only(D)	62	68	834	4.2	40.91
Palm 10% + diesel 90% (P10)	68	74	834.8	4.26	40.61
Palm 20% + diesel 80% (P20)	79	86	835.6	4.32	38.32
Palm 30% + diesel 70% (P30)	87	94	836.4	4.39	36.54
Jatropha 10% + diesel 90% (J10)	65	72	833.8	4.3	40.64
Jatropha 20% + diesel 80% (J20)	73	82	838.6	4.41	39.38

Table 1. *Cont.*

Blends/Properties	Flash Point, °C	Fire Point, °C	Density, kg/m^3	Viscosity, mm^2/s @40 °C	Calorific Value, MJ/kg
Jatropha 30% + diesel 70% (J30)	78	86	842.4	4.51	37.11
Cottonseed 10% + diesel 90% (CS10)	75	84	842.6	4.26	40.61
Cottonseed 20% + diesel 80% (CS20)	80	91	845.4	4.32	39.31
Cottonseed 30% + diesel 70% (CS30)	89	96	848.4	4.38	36.01
Palm 5% + jatropha 5% + diesel 90% (PJ10)	76	82	833.6	4.37	40.8
Palm 10% + jatropha 10% + diesel 80% (PJ20)	78	85	839.2	4.53	40.2
Palm 15% + jatropha 15% + diesel 70% (PJ30)	81	87	842.8	4.7	39.22
Jatropha 5% + cottonseed 5% + diesel 90% (JCS10)	84	91	836.2	4.36	40.35
Jatropha 10% + cottonseed 10% + diesel 80% (JCS20)	89	95	842.4	4.53	39.78
Jatropha 15% + cottonseed 15% + diesel 70% (JCS30)	92	97	847.2	4.69	38.22
Palm 5% + cottonseed 5% + diesel 90% (PCS10)	87	95	843.3	4.32	40.32
Palm 10% + cottonseed 10% + diesel 80% (PCS20)	91	98	847.4	4.45	39.7
Palm 15% + cottonseed 15% + diesel 70% (PCS30)	97	99	849.6	4.57	38.13
Palm 3.33% + jatropha 3.33% + cottonseed 3.33% + diesel 90% (PJCS10)	72	77	842.8	4.426	39.05
Palm 6.66% + jatropha 6.66% + cottonseed 6.66% + diesel 80% (PJCS20)	81	88	846.6	4.65	38.19
Palm 10% + jatropha 10% + cottonseed 10% + diesel 70 (PJCS30)	88	94	848.4	4.87	37.83
Jatropha 100% (J100)	76	84	846	5.23	38.25
Cottonseed 100% (CS100)	131	142	859	4.81	37.92
Palm 100% (P100)	84	91	849	4.82	37.95

4. Engine Setup

The engine setup utilized is depicted in Figure 1. The engine employed for testing was the Kirloskar AV1 single-cylinder, 4-stroke, water-cooled diesel engine with direct ignition (DI). Engine specifications are provided in Table 2. An eddy current dynamometer was used for loading the engine, which was coupled to the engine shaft. A pressure sensor and crank angle encoder were installed onto the engine for measuring the peak pressure values and the required p-theta curves through the DAQ. Thermocouples were installed at required points to measure the temperature of exhaust gases and other temperature values. The AVL 5 Gas analyzer was used to record the emission values viz. HC, CO and NOx. The smoke values were documented using the smoke meter. The following assumptions were made before computing and interpreting the results:

i. The engine operated at steady state conditions. Enough time, about 15 min, was given in order to attain a steady state in each test case;
ii. The entire engine, including the fuel, air, coolant and exhaust lines, was treated as a control volume;
iii. Air and flue gas were treated to be ideal in nature;
iv. Changes in kinetic and potential energies of flow streams were ignored.

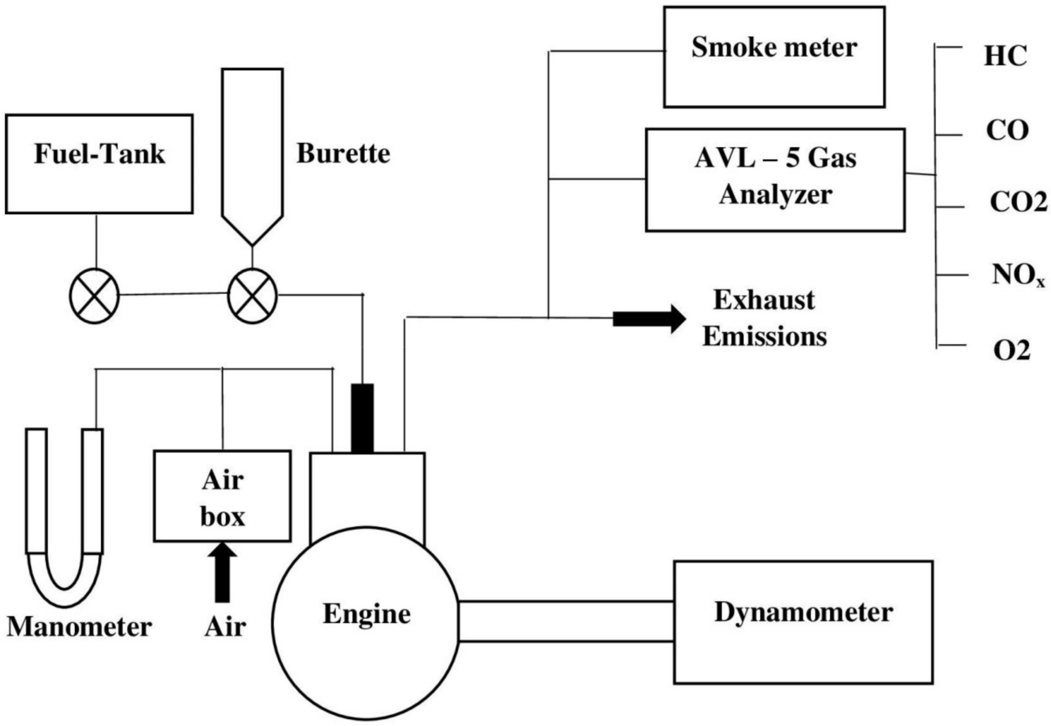

Figure 1. Engine Setup.

Table 2. Kirloskar AV1 Engine Specifications.

Engine Specifications	
Brand	Kirloskar
No of Cylinders and Stroke	1 and 4
Bore × Stroke Length	80 × 110 mm
Compression Ratio	16.5:1
Fuel Tank Capacity	6.5 litre
Rated Speed	1500 RPM
Torque at Full Load	2.387 kg-m
Cubic Capacity	0.553 L

5. Energy and Exergy Analysis:

Energy and exergy analysis was conducted in accordance with the methodology and equations used by Sayin and Kahraman [3] and Kotas [16]. All the relevant data related to temperatures were recorded using a thermocouple attached. All flue gas data were calculated using the mass flow rate of air and fuel. Emissions were monitored using AVL 5 Gas Analyser, and smoke was measured using a smoke meter attached in parallel to the exhaust line. The flue gas temperature was assumed to be 250 °**C**. The dead state conditions were taken as 1 atmosphere and 25 °**C,** and the air is assumed to contain 75.65% N_2, 20.3% O_2, 3.12 H_2O, 0.03% CO_2 and 0.9% Ar.

The following equations are employed to conduct energy and exergy analysis:

Mass balance:

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

Energy balance:

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

Fuel energy rate:

$$\dot{E}_{fuel} = \dot{m}_{fuel} LHV \tag{3}$$

Shaft power:

$$\dot{W}_{shaft} = \frac{2\pi N\tau}{60} \tag{4}$$

Thermal efficiency:

$$\eta_I = \frac{\dot{W}_{shaft}}{\dot{E}_{fuel}} \tag{5}$$

Exergy balance:

$$\dot{E}x_Q + \dot{E}x_W + \sum \dot{m}_{in}\varepsilon_{in} - \sum \dot{m}_{out}\varepsilon_{out} - \dot{E}x_{dest} = 0 \tag{6}$$

Chemical exergy of the fuel:

$$\varepsilon^{ch}_{fuel} = LHV \times \varphi \tag{7}$$

Chemical exergy factor for liquid fuel:

$$\varphi = 1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{s}{c}\left(1 - 2.0628\frac{h}{c}\right) \tag{8}$$

Exergy associated with shaft work:

$$\dot{E}x_w = \dot{W} \tag{9}$$

Exergy of the exhaust:

$$\dot{E}x_{exh} = \dot{m}_{exh}\varepsilon \tag{10}$$

Specific exergy of the exhaust:

$$\varepsilon = \varepsilon^{tm} + \varepsilon^{ch} \tag{11}$$

Thermo-Mechanical exergy of exhaust gas:

$$\varepsilon^{tm} = \sum_{i=1}^{n} a_i \left\{ \left(\overline{h}_i - \overline{h}_{0i}\right) - T_a(\overline{s}_i - \overline{s}_{0i}) \right\} \tag{12}$$

Chemical exergy of exhaust gas:

$$\varepsilon^{ch} = \sum y_i \varepsilon^{ch}_i + \overline{R}T_a \sum_{i=1}^{n} y_i \ln y_i \tag{13}$$

Exergy due to heat transfer:

$$\dot{E}x_Q = \dot{m}_{cw} c_{p,water} \left[(T_{out} - T_{in}) - T_0 \ln \frac{T_{out}}{T_{in}} \right] \tag{14}$$

Exergy efficiency:

$$\eta_{II} = \frac{\dot{E}x_{work}}{\dot{E}x_{fuel}} = \frac{BP}{\dot{E}x_{fuel}} \tag{15}$$

The symbols h, c, o and s indicate the mass fractions of hydrogen, carbon, oxygen and sulfur, respectively, present in the liquid fuel. A flow-sheeting software Cycle-Tempo was used to evaluate the exergy of the fuel blends as well as the exhaust. The software implemented the above-mentioned Equations (1)–(3) and (6)–(14) and offered the ease of a graphical user interface for the purpose. The composition of the fuel blend and exhaust gas was input, and the exergy was computed.

6. Uncertainty Analysis

Uncertainties in the output parameters are determined using the approach given by Moffat [17]. Less than 3% uncertainty was found for all output parameters.

7. Results and Discussions

7.1. First Law Efficiency

First law efficiency is also commonly known as brake thermal efficiency (BTE). The BTE trends of primary, binary and ternary blends versus load are depicted in Figures 2–4. Primary biodiesel blends, the 30% blends, show higher brake thermal efficiency with augmenting biodiesel concentration due to the increased amount of oxygen presence dominating over the viscous attribute, thus producing higher heat energy. Binary and ternary blends of biodiesel show efficiency values closer to diesel but lower than primary biodiesel blends due to reduced energy content released on combustion and relatively higher brake-specific fuel consumption leading to a higher accumulation of fuel to compensate for the power output produced. At a load of 20 N-m, the highest values of BTE were observed for J30, P30 and CS30 at 36.91%, 36.56% and 36.83%, respectively.

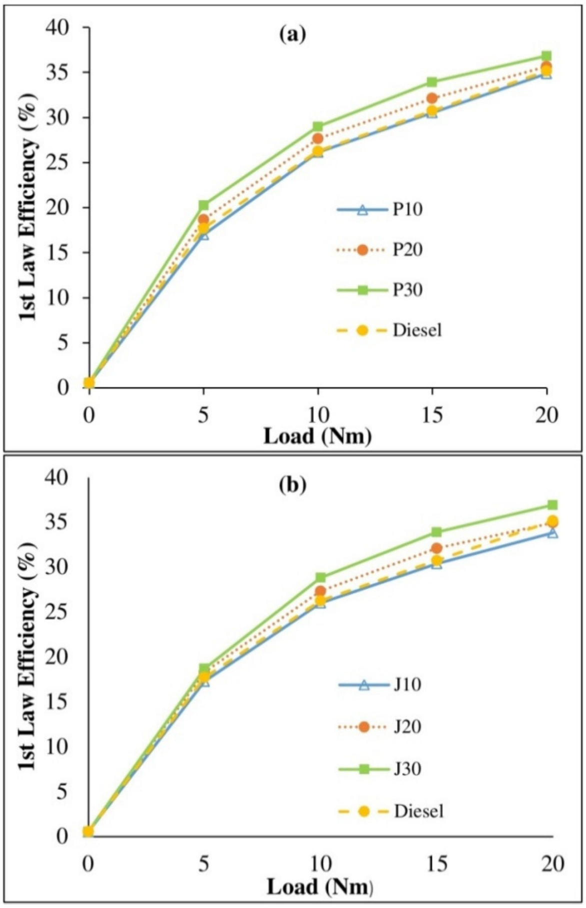

Figure 2. *Cont.*

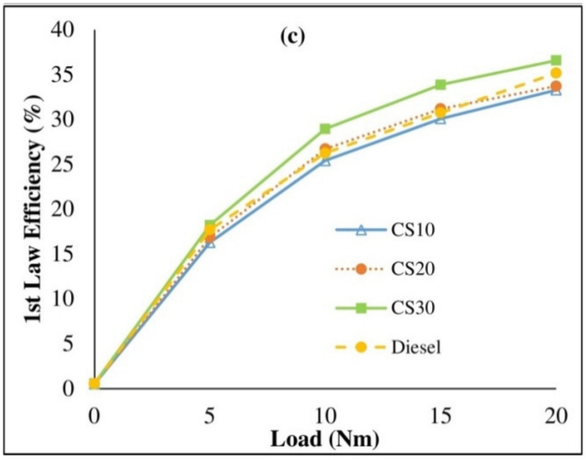

Figure 2. 1st law efficiency of primary biodiesel blends in comparison with diesel for (**a**) palm, (**b**) jatropha and (**c**) cottonseed blends.

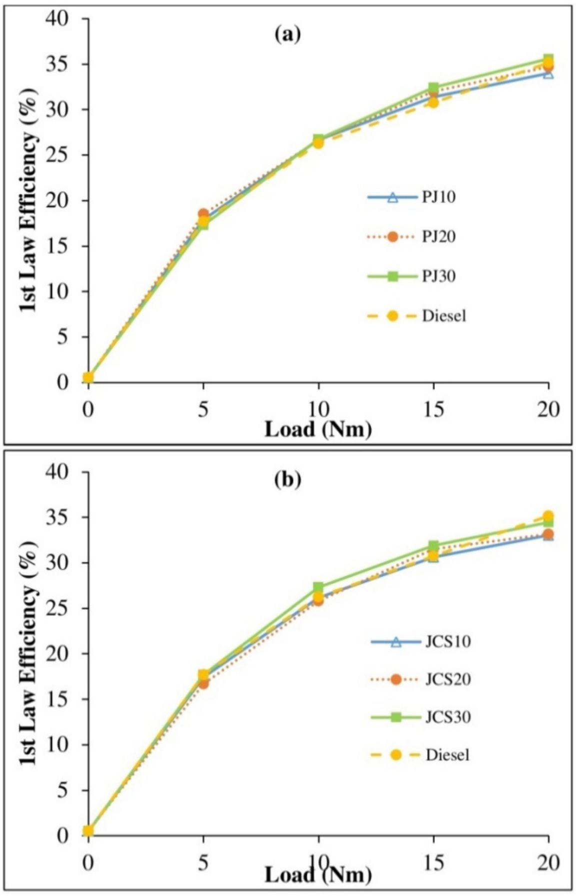

Figure 3. *Cont.*

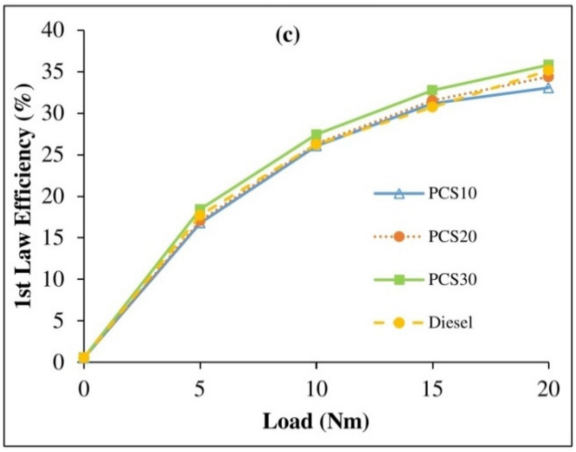

Figure 3. **1st law efficiency** of binary biodiesel blends in comparison with diesel for (**a**) palm-jatropha, (**b**) jatropha-cottonseed and (**c**) palm-cottonseed blends.

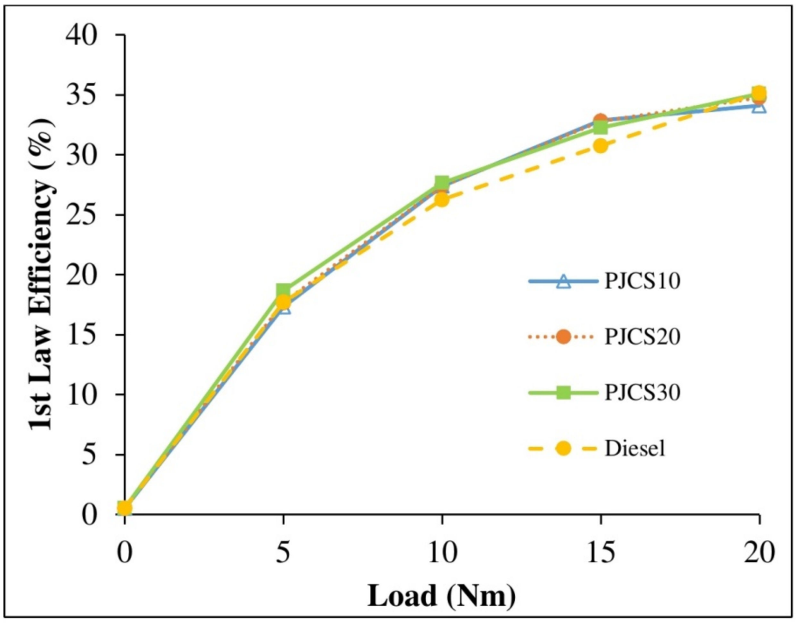

Figure 4. **1st law efficiency** of ternary biodiesel blends in comparison with diesel.

The 1st law efficiencies obtained in the present study are close to those reported in most of the CI engine literature. Khoobbhakt et al. [2] reported a maximum efficiency of 36.6% while using biodiesel and ethanol as blends in a diesel engine. Canakci and Hosoz [18] reported a maximum efficiency of 37.3% while using soya beans and methyl ester as blends. However, Caliskan et al. [19] and Odibi et al. [8] reported a maximum efficiency of over 40%, which is on the higher end of the reported values. All three fuels tested by Caliskan et al. [19] gave almost similar results, indicating that the high efficiency is due to the procedure adopted. Odibi et al. [8] used a turbo-charged engine which could be a possible reason for the higher efficiency. Moreover, these two engines are of high power ratings (Caliskan 66.5 kW and Odibi 162 kW). The advantage of conducting experiments in a larger engine is that they are better equipped with energy conservation provisions. Sarikic et al. [20] used a 9 kW diesel engine to test biodiesel and butanol and reported a maximum efficiency of 32.5%.

7.2. Second Law Efficiency

Second law efficiency is defined as the ratio of brake power and the total fuel exergy. Second law analysis points to those areas where useful energy can be conserved or extracted.

The total fuel exergy is a sum of thermo-mechanical and chemical exergy. Since the fuel is at ambient conditions at the feed point, the thermo-mechanical exergy is negligible, and the total exergy is equal to the chemical exergy of the fuel. According to Kotas [16], the chemical exergy is slightly greater than the lower heating value of the fuel. The chemical exergy factor that relates to the chemical exergy and a lower heating value is given in Equations (7) and (8). Due to the manner in which it is defined, the 2nd law efficiency would be slightly lower than the 1st law efficiency. Trends for biodiesel and diesel are illustrated in Figures 5–7. It was deduced from the results that diesel shows higher second law efficiency values due to the high amount of fuel exergy being sufficiently utilized during combustion. This could be because the engine is designed for diesel and was not modified to take care of the different properties of other blends. In comparison to binary blend trends, diesel shows almost the same values at 10 N-m and 15 N-m loads. Moreover, increasing biodiesel volume percentage for all the blends led to an increment in the 2nd law efficiency, and the trend is seen as analogous to that observed in the 1st law efficiency.

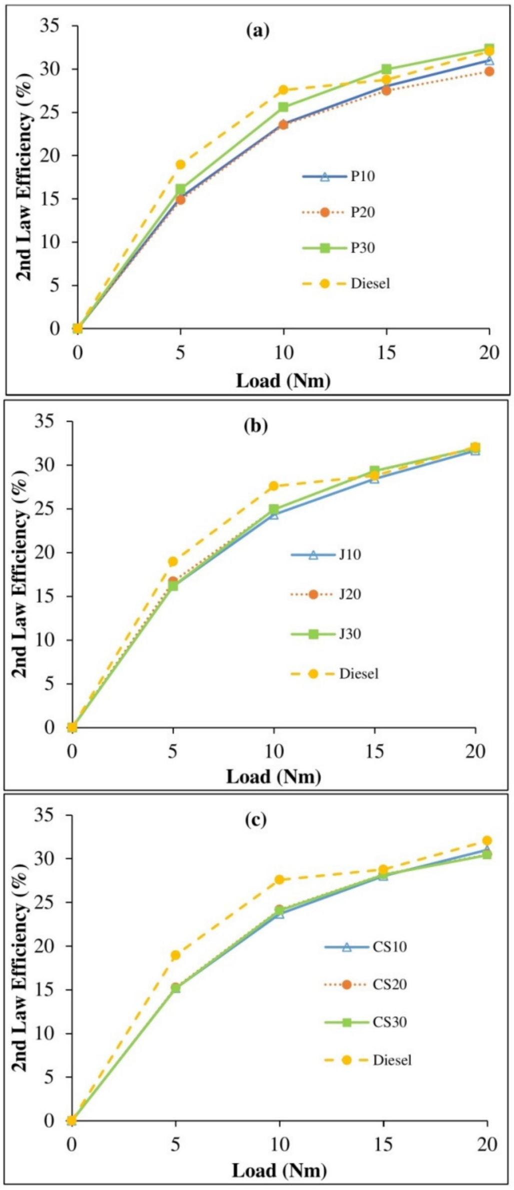

Figure 5. 2nd law efficiency of primary biodiesel blends in comparison with diesel for (**a**) palm, (**b**) jatropha and (**c**) cottonseed blends.

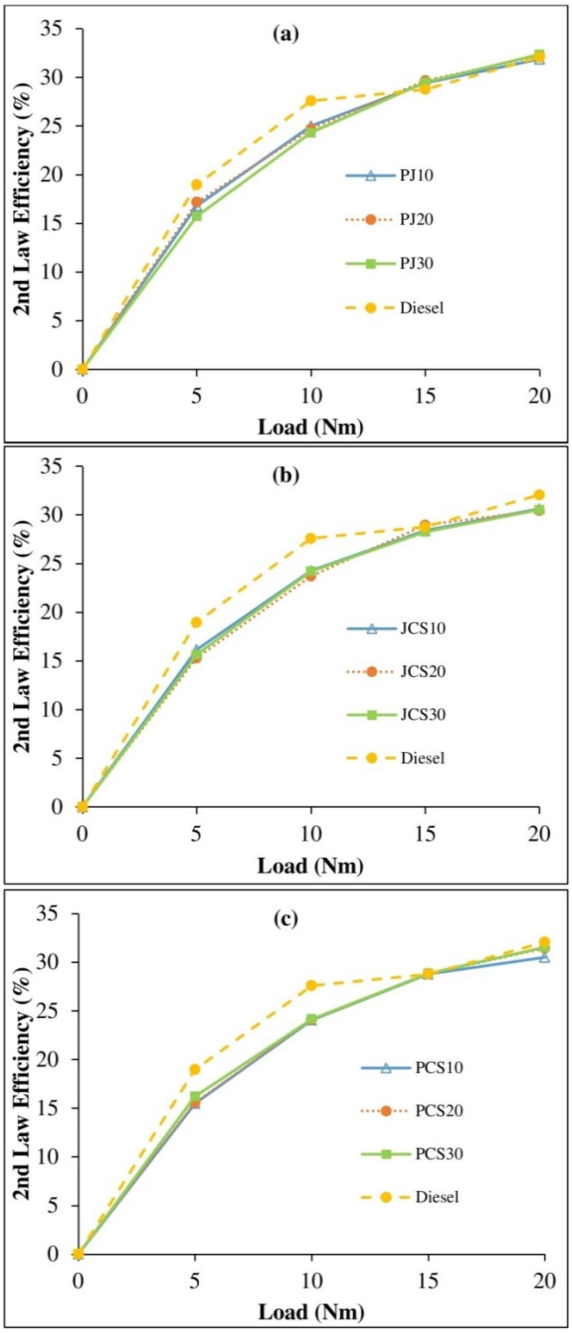

Figure 6. 2nd law efficiency of binary biodiesel blends in comparison with diesel for (**a**) palm-jatropha, (**b**) jatropha-cottonseed and (**c**) palm-cottonseed blends.

The higher second law efficiency while using diesel could be due to the irreversibilities associated with blending, which involves mixing different fuels with different properties. Firstly, the constituents of the blend have different physical properties and hence may not form a homogenous mixture. Secondly, the fuel injection system and the combustion chamber design are all meant for diesel. Hence, the primary, binary and ternary blends undergo mixing and combustion under unnatural and unfavorable conditions resulting in lower exergy efficiency.

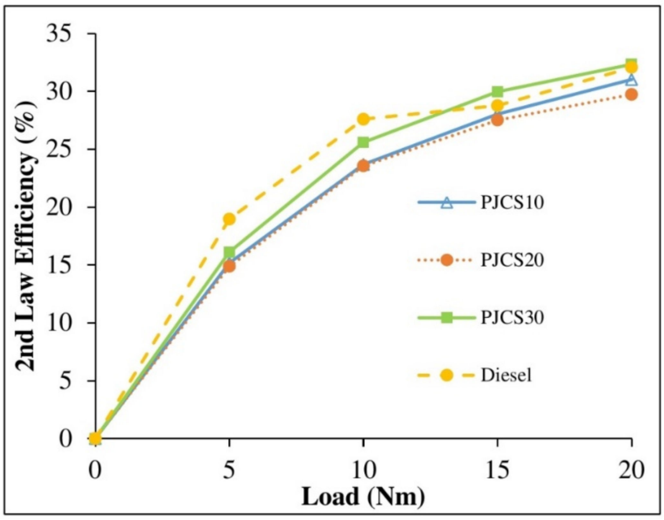

Figure 7. 2nd Law Efficiency of ternary biodiesel blends in comparison with diesel.

In the present work, the 2nd law efficiencies are around 30% for a maximum load of 20 N-m. Sayin and Kahraman [3], as well as Chintala and Subramanian [21], reported similar values (~29%) in their work, while Caliskan et al. [18] reported 37.8% of 2nd law efficiency. Once again, one can note that Savin and Kahraman and Chintala and Subramanian [21] used small engines (9 and 7 kW, respectively), which were probably less equipped with energy conservation provisions, while Caliskan et al. [19] used a bigger engine. Another probable reason for lower 2nd law efficiency noted in the present study could be the unnatural blending. Primary blends showed efficiencies that were slightly greater than the secondary blends, which were further greater than ternary blends. Hence blends made by mixing too many components could be detrimental to engine performance. In the literature, Lopez et al. [22] reported the lowest 2nd law efficiency of 25% while using blends of oil-pomace oil biodiesel and diesel blends. It can be seen that even with the differences in the experimental conditions, the 2nd law efficiency of this work and those reported in the literature are in the same ballpark.

7.3. Fuel Exergy Destruction

Fuel exergy is an important parameter to calculate the maximum available amount of energy that can be extracted, and fuel exergy destruction represents the loss of work potential that cannot be recovered for use in the system due to the presence of irreversibilities. It is well known that certain processes such as combustion and heat transfer have a high amount of irreversibilities and hence would contribute to a larger share in exergy destruction. Figures 8–10 represent the trends of fuel exergy destruction for all biodiesel blends with respect to diesel. Fuel exergy destruction is the ratio of the amount of exergy destroyed to the fuel exergy (both in kW). The exergy destroyed considers the exergy carried away by cooling water and the flue gases (both thermo-mechanical as well as chemical exergy). Diesel is shown to have lower fuel exergy destruction in comparison with the biodiesel blends at lower loads due to its higher second law efficiency and also higher amounts of actual work produced during engine operation. It was consistently observed that the higher the amount of diesel, the lower the exergy destroyed. This could be due to the fact that the biodiesel blends have been artificially made, while diesel is a naturally occurring compound in crude oil. Unnatural blends could result in greater amounts of irreversibilities in the combustion process. At higher loads of 15 and 20 N-m, fuel exergy destruction is nearly identical for all the blends P, J, PJ and PJCS. Moreover, 30% of blends in all cases showed low exergy destruction. Moreover, in Figures 8–10, it can be seen that for diesel, the exergy destruction is the same for loads of 10 and 15 N-m. A similar observation can be made for the second law efficiency from Figures 5–7. This needs to be further studied.

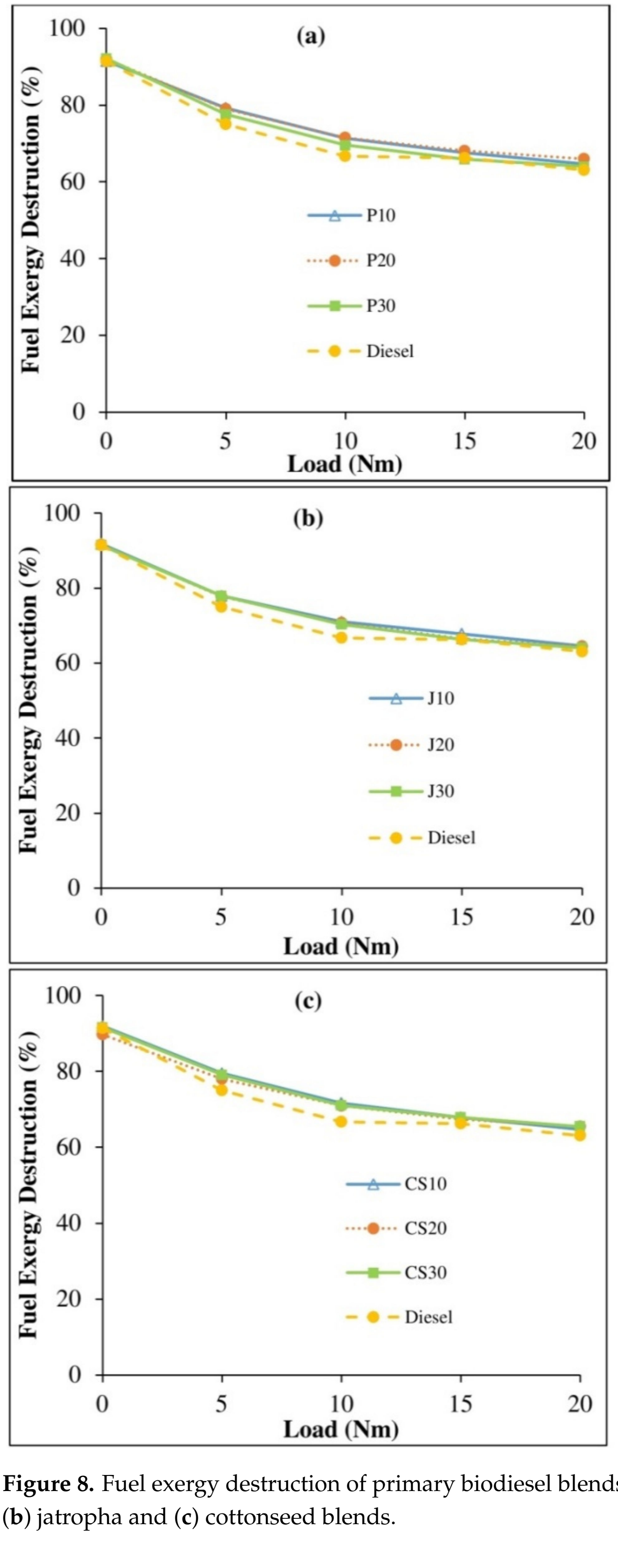

Figure 8. Fuel exergy destruction of primary biodiesel blends in comparison with diesel for (**a**) palm, (**b**) jatropha and (**c**) cottonseed blends.

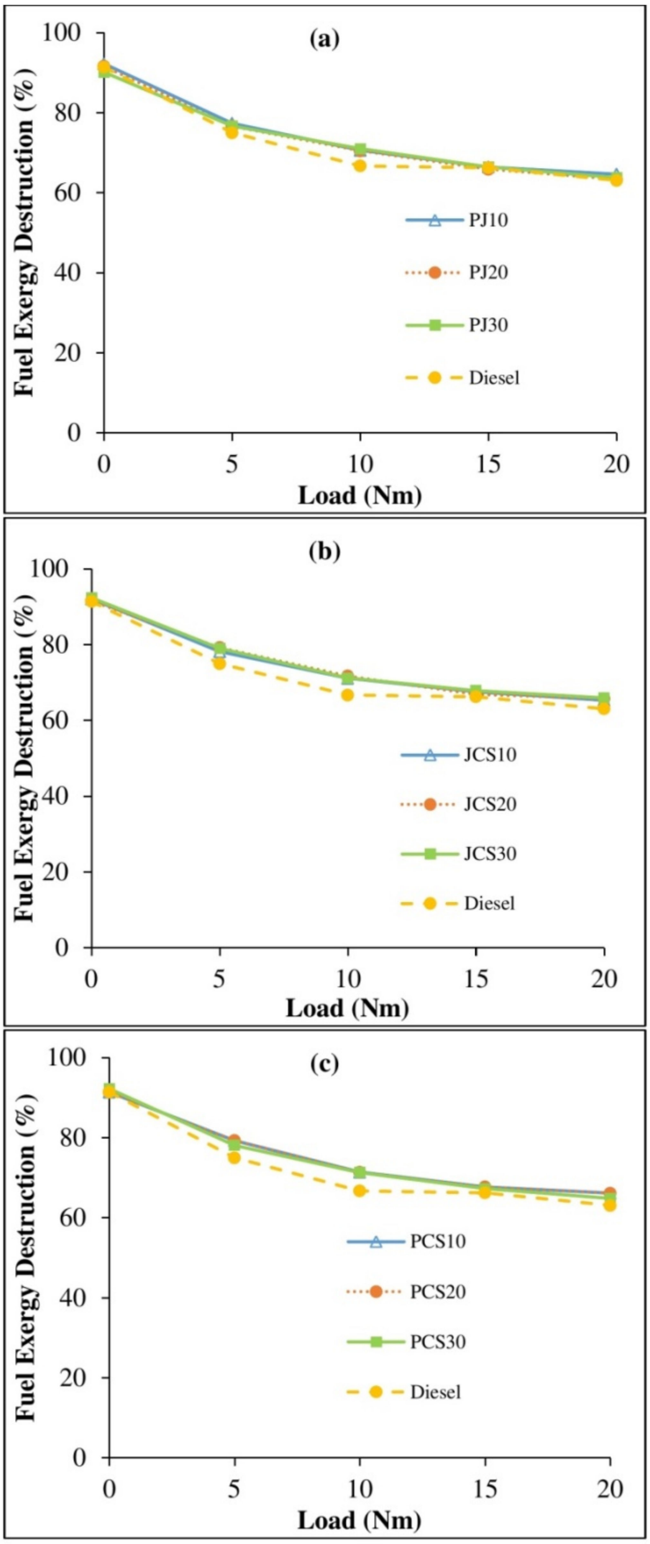

Figure 9. Fuel exergy destruction of binary biodiesel blends in comparison with diesel for (**a**) palm-jatropha, (**b**) jatropha-cottonseed and (**c**) palm-cottonseed blends.

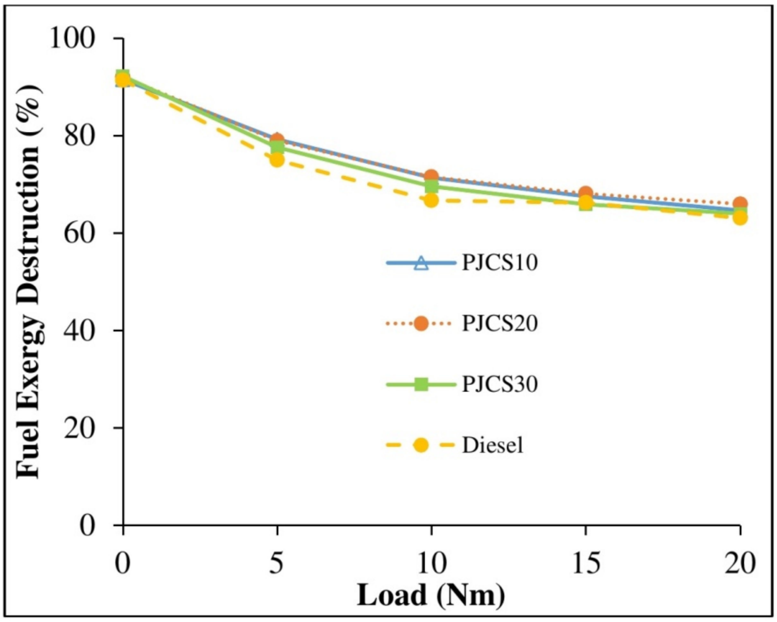

Figure 10. Fuel exergy destruction of ternary biodiesel blends in comparison with diesel.

From the open literature, one can find that the exergy destroyed is typically in the range of 40–65%. Chintala and Subramanian [21] reported a similar amount of exergy destruction in their entire range of experiments. The least amount of exergy destruction (~45%) was consistently reported by Karthickeyan et al. [23] in their entire range of experiments. They used pomegranate oil-methyl ester in their studies. They attribute the better performance to the engine modifications carried out in their study. The exergy destruction is attributed to irreversibilities due to combustion, heat transfer and friction. This emphasizes that suitable engine modifications need to be carried out depending on the fuel blends used.

7.4. Total Exergy Destruction

Total exergy destruction is the exergy destroyed expressed in kW. This quantity gives an understanding of the quantity of useful energy lost and not as a fraction. As observed in Figures 11–13, increasing load led to more exergy destruction due to the widening of cylinder temperature (2017). Ma et al. [24] obtained similar results in their study involving methanol in dual-fuel engines. In all cases, the exergy destructed falls in the range of 3 to 6.5 kW. Diesel shows a minimal amount of total exergy destruction when compared to all types of biodiesel blends; the reason justified is due to diesel showing lower fuel consumption. Increasing biodiesel quantity in the blends led to a decline in exergy destruction as more oxygen quantity of biodiesel led to cleaner combustion and more energy extraction. Moreover, the exergy carried out by cooling water and flue gases is considerable in quantity. This can be utilized, and the energy can be tapped in some applications downstream, such as process heat. Moreover, one can contemplate the operation of a vapor-absorption refrigeration system to use the heat contained in the water. Moreover, the flue gases can be used in a turbo-charger. Such retrofitting can decrease the exergy destruction and improve the second law efficiency.

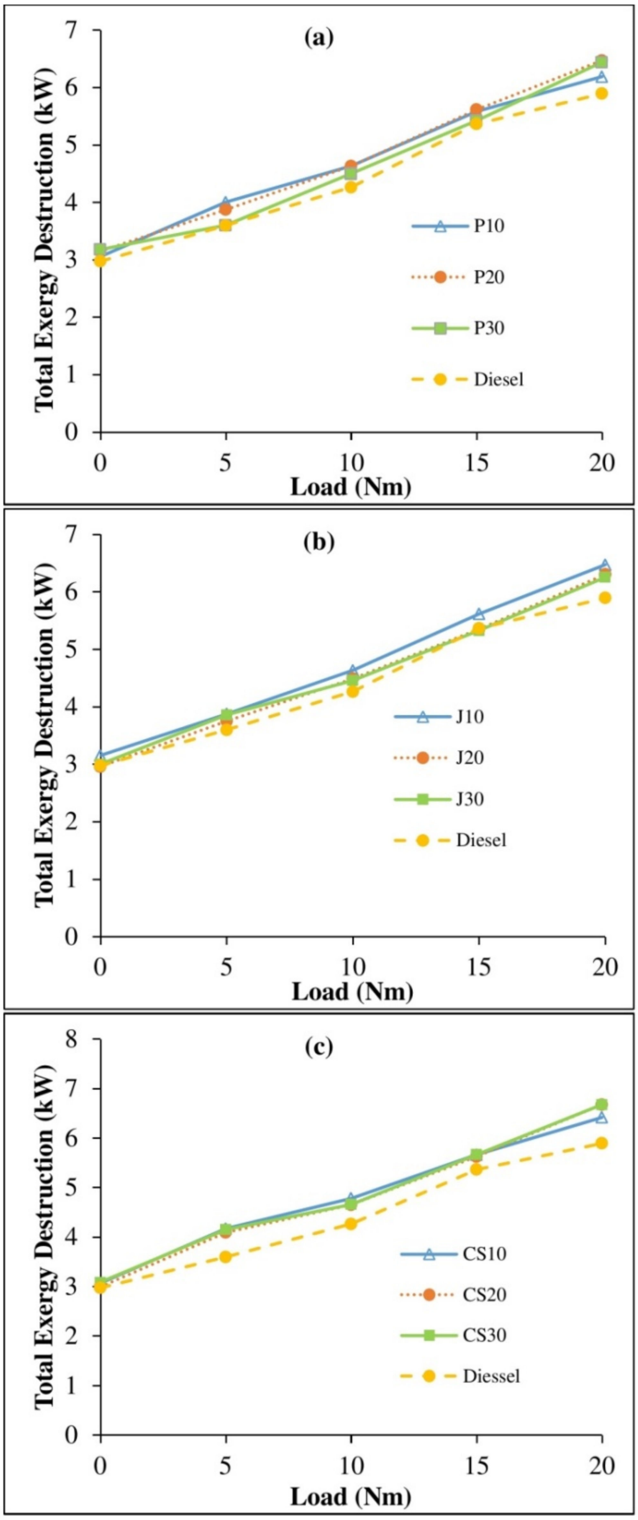

Figure 11. Total exergy destruction of primary biodiesel blends in comparison with diesel for (**a**) palm, (**b**) jatropha and (**c**) cottonseed blends.

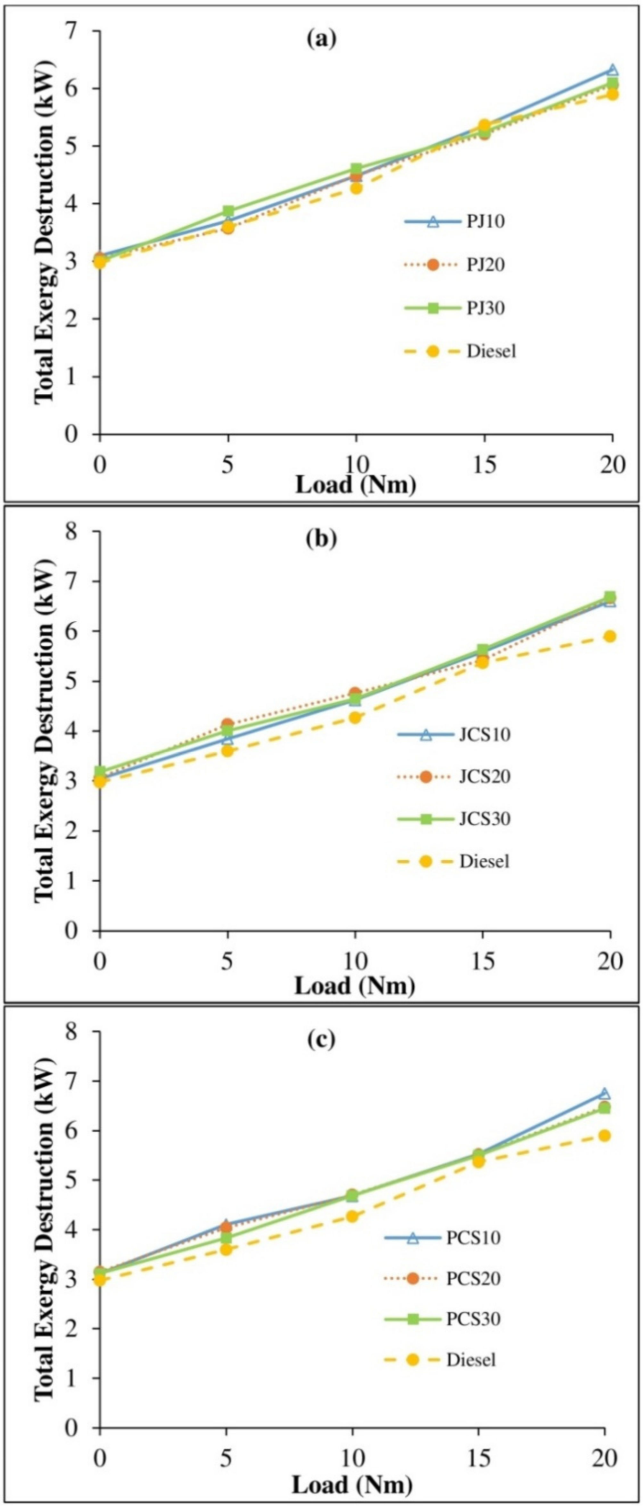

Figure 12. Total exergy destruction of binary biodiesel blends in comparison with diesel for (**a**) palm-jatropha, (**b**) jatropha-cottonseed and (**c**) palm-cottonseed blends.

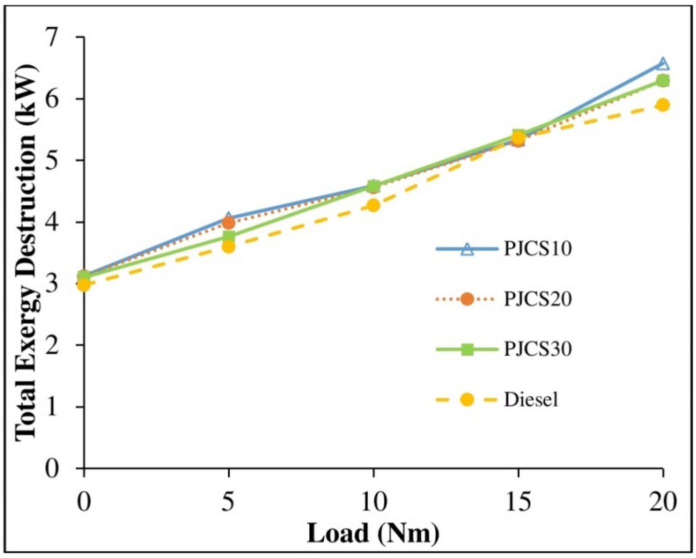

Figure 13. Total exergy destruction of ternary biodiesel blends in comparison with diesel.

8. Conclusions

The exergy analysis performed on the CI engine running on biodiesel provides various inferences about the available amount of heat being destroyed inside the engine while in operation:

- Primary biodiesel blends (J30) showed higher 1st Law efficiency due to reduced density and an additional amount of oxygen present;
- Higher 2nd law efficiency is observed nearly at all loading conditions while using diesel;
- 1st law and 2nd law efficiencies rise with augmentation in the amount of biodiesel. This encourages the use of biodiesel blended with diesel;
- Exergy destruction for diesel is lower at low loads;
- At higher loads of 15–20 N-m, fuel exergy destruction is nearly identical for diesel and biodiesel blends. This and the above point imply that an intelligent blending control system is needed when one opts for biodiesel fuels. The control system should inject only diesel at lower loads while at higher loads, more blending can be performed;
- Increased biodiesel quantity in the blends led to very minute variation in exergy destruction;
- Primary, binary and ternary biodiesel blends nearly show the same exergy destruction ranging from 63 to 65% at 20 N-m loads;
- Increasing biodiesel percentage in P and CS primary blends led to an increment in exergy destruction;
- In the case of binary and ternary biodiesel blends, augmenting biodiesel quantity in the blend caused little fluctuations in total exergy destruction;
- The lowest exergy destruction at 20 N-m loading condition was shown by PJ20 and Diesel at 6 and 5.9 kW, respectively;
- The 1st and 2nd law efficiencies obtained in the present study are in the same ballpark as those reported in the open literature;
- The manufacture of biodiesel and its blends must be determined by the crop availability at a specific geographical location.

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