Abstract: Renewable energy is getting more attention in recent times due to the rapid depletion of fossil fuel reserves. Production and consumption of biofuels derived from biomass has significantly increased. In the present work, *Spirulina* microalgae have been chosen as feedstock for biodiesel production. Diesel and biodiesel were mixed in different volumetric ratios to prepare fuel blends (SBF₀, SBF₂₀, SBF₄₀, SBF₆₀, SBF₈₀, and SBF₁₀₀). Energy and exergy analysis has been performed on a four-stroke, single-cylinder diesel engine. Experimentation was done under varying loads at 1500 RPM. The effect of multiple loads and blends was investigated for brake power (BP), cooling water losses (Qₜₑₜ), exhaust gas losses (Qₑₓₕ), and unaccounted losses (Qᵤₑₜ). Pure diesel SBF₁₀₀ has the highest and lowest exergy efficiencies, respectively equaling roughly 31.65% and 29.75%. It has been observed that BP and Qₜₑₜ increase with the increase in load whereas Qₑₓₕ and Qᵤₑₜ show a decreasing trend. It was also observed that with an increase in blending, Qₜₑₜ increases while Qₑₓₕ decreases.

In the exergy analysis, it was observed that the exergy destruction rate has a maximum fraction of input exergy values of 46.01% and 46.29% for Diesel and SBF₂₀ respectively. The system engine sustainability index was in the range of 1.27 to 1.46, which is directly related to exergy efficiencies.

Keywords: energy; exergy; biodiesel; diesel engine; analysis; sustainability

1. Introduction

As the world’s energy requirements are rising and conventional sources of energy are becoming depleted, the world is looking for alternative sources to meet its energy demands [1]. The World Energy Investment 2022 report [2] says that the amount of money spent on energy around the world will go up by 8% in 2022. Many developed and developing countries want to become net zero emitters by the end of 2050. India is expected to reach this goal by the end of 2070. Energy security is attracting the attention of researchers these days, as dwindling fossil fuel stocks and rapid increases in energy demands due to growing population [3] have led the researchers to shift their focus towards renewable energy, even though crude oil has significant contribution in the transportation sector [4]. There are growing concerns about the concentration of GHGs in the atmosphere, which is increasing with the increase in population and anthropogenic activities, which have a direct influence on human health. Biofuel is a type of fuel that can be produced directly from natural sources such as plants, animals, municipal wastes etc. [5]. It can be produced from environmentally friendly sources of materials and is additionally referred to as “renewable energy”. Crude oil and petroleum products will be in greater demand in a few years to match the demands of the growing population [6]. Due to their nonrenewable nature, crude oil reserves would be depleted as opposed to biodiesel, which can be obtained from trash, food, and non-food sources [7]. Medhat et al. [8] in their review article examine the significance of response surface methodology in the field of compression ignition engine performance and emission attributes using blends of diesel fuel, alternative fuels, and nanoparticle additives for obtaining the best outcome.
Biodiesels are made of long-chain alkyl esters [9]. Biodiesels are produced by the reaction of triglycerides present in oils with alcohols in the presence of catalysts. The finest biodiesel feedstock is microalgae, a fourth-generation biofuel that grows more quickly than other oil crops and can flourish in freshwater, sewage, and marine environments [10]. Algae can be classified as macro, micro, and cyanobacteria. Some of the microalgae that have been investigated for the generation of biodiesel are *Spirulina*, *Chlorella Vulgaris* and *Dunaliella* [11]. Biofuels are a promising renewable energy source which can serve as a replacement of traditional fuels and can reduce GHGs and other harmful emissions. Biofuels are facing the challenges of land, food and water consumption [12]. *Spirulina’s* transesterification is a particularly potent process in these microalgae. Palmitic, myristic, stearic, oleic, and linolenic acid are the main fatty acids found in spirulina oil [13]. Siraj et al. [14] examined how Nox modeling using ANN and dual blends of various biodiesels affects DICI engine characteristics. Using four distinct biodiesels, including Jatropha, Karanja, Mahua, and Neem, six sets of dual biodiesel/diesel mixes (10% and 90%) were created. The tests were run on a single-cylinder VCR engine with a constant compression ratio (CR) of 17.5:1 and full throttle. They concluded that the dual biodiesel blends under consideration might be utilized as alternate fuels in diesel engines; as a result, it is advised as an alternate fuel for DICI engines. Hoang, et al. calculated the viscosity, cetane number, calorific value and density of biodiesel and examined how its characteristics affect diesel engine performance [15]. Biodiesel blends made from *S. Marginatum* macroalgae of 20%, 50%, 75%, and 100% were investigated in the Kirloskar engine by Karthikeyan et al. [16]. Medhat et al. [17] used a mixture of sunflower and soyabean oil to produce biodiesel and used 29 pre-planned experiments to examine the significance of four reaction parameters including methanol-to-oil ratio, catalyst concentration, RPM and reaction time, and their combined impact on biodiesel yield. BBD, or box-Behnken design, is based on the response surface approach (RSM) The optimized values for BTE, UHC, and Nox values according to the RSM optimizer outputs were 13.7%, 121 ppm, and 234.9 ppm, respectively. Chandra et al. [18] used Pithecellobium dulce seed oil (PDSO) for transesterification to produce methyl esters. As PDSO contains 2.12% free fatty acids (FFA), an alkaline-based potassium hydroxide (KOH) catalyst was utilized to produce methyl ester and found to be effective. At a 1:6 molar ratio, 60 °C reaction temperature, 0.8 weight percent catalyst, and a reaction time of 90 min, an optimum PDSOME yield of 93.2% is attained. Because of higher oxygen content in the B20 blend compared to pure diesel, viscosity is reduced and engine efficiency is increased, resulting in greater HRR and CP when compared to pure diesel. At 23.0° b TDC and 220 bar injection pressure, Rajak et al. [19] investigated the effects of *Spirulina* Biofuel (SBF) blends with neat diesel (20, 40, 60, and 80%) on engine characteristics. The findings suggested, all blends have a higher ignition delay duration compared to pure diesel [20]. Thermodynamic studies, such as energy and exergy evaluations [21,22], are often performed in order to develop internal combustion engines since they were used to evaluate and define the efficiencies of thermal systems such as IC engines [23,24]. Siraj et al. [25], under various engine load levels, blended fuels created by blending two distinct biodiesels (10% by volume) with conventional diesel. To determine the appropriateness of blending two distinct biodiesels with diesel as a potential substitute for petroleum fuel, engine performance and emission characteristics were examined. As a result, several studies have been done on the energy and exergy evaluations of IC engines. The evaluation of quantitative and qualitative data for identifying energy and exergy efficiencies, losses, and exergy destruction was done in conjunction with an investigation of direct injection compression ignition (DICI) engines for biodiesel-diesel blended fuels. In the energy–exergy, study Siraj et al. [26] reveals that the B20 blend’s combustion and exergetic efficiency are highest at 2.5 kW and 3.3 kW BPs (+87.73% and (+52.04%), respectively. It has been found that three-fifths of the total exergy available has been destroyed. Cooling water removes half of the heat that is supplied, brake power absorbs a third of the heat, and the remaining heat is lost in exhaust gases and unreported losses. Javad et al. [27] investigated energy and exergy study of a combined
ORC and ERC system for waste heat recovery from biodiesel-fueled diesel engines. To test the engine’s exergy balance, López et al. [28] employed biodiesel made from oil produced from olive pomace to power a diesel engine. Two biodiesels made from petroleum diesel fuel and the methyl esters of soybean oil (SME) and yellow grease (YGME), and a 20% mix of each biodiesel with diesel fuel were used by Canakci et al. [29] for energy and exergy assessments of a diesel engine. The performance parameters of a multifuel diesel engine employing pure diesel and natural gas were studied theoretically [30] and empirically by Ramos et al. [31] in terms of both energy and exergy assessments. Using exergy and energy analysis, it is possible to distinguish between different types of combustion chambers such as hemispherical, trapezoidal and toroidal ones [32]. Because of its greater energy efficiency, biodiesel created with pumpkin seed oil was discovered to be a viable alternative to traditional diesel. Therefore, toroidal combustion chambers had superior engine characteristics than the other chambers. Chandra et al. [18] used *Pithecellobium dulce* seed oil (PDSO) for transesterification to produce methyl esters. As PDSO contains 2.12% free fatty acids (FFA), an alkaline-based potassium hydroxide (KOH) catalyst was utilized to produce methyl ester and found to be effective. At a 1:6 molar ratio, 60 °C reaction temperature, 0.8 weight percent catalyst, and a reaction time of 90 min, an optimum PDSOME yield of 93.2% is attained. Medhat et al. [33] investigated biodiesel that was trans-esterified from *Scenedesmus obliquus* algae, because pure biodiesel has drawbacks when used in engines. Compared to an exquisite biodiesel mix, it was discovered that CO₂ and Nox levels had increased while HC, CO, and O₂ levels had decreased. They concluded that adding pentane to a biodiesel blend considerably enhanced engine performance. The present work and literature comparison is represented in Table 1.

Although numerous research has looked at the energy and pollutant characteristics of alternative fuels, particularly biodiesels, there are very few studies on the effects of microalgae biodiesel and diesel binary blends on the direct-ignition CI engine under different loading conditions. We examined the quantity and quality of energy and exergy in a direct ignition diesel engine that was running with the microalgae biodiesel and diesel blends. Additionally, each fuel blend’s sustainability index was determined.

Microalgae-based oil feedstock has an advantage over conventional biodiesel feedstock in that it does not add to the contentious “food vs. fuel” debate. The current investigation is to study the variation of loading conditions and blending ratios on the energetic and exergetic properties of microalgae biodiesel. The energy and exergy analysis for different blends and loading conditions of biodiesel will be worked on more by testing with a single cylinder diesel engine. The way biodiesel burns and what it puts out into the air was studied. The distribution of input fuel energy was observed and the analysis of sustainability was done to uncover the thermodynamics of irreversible processes. The outcomes of such a study could be very valuable to engineers, designers, and researchers in figuring out the best fuel mixtures and engine operating settings to produce more cost-effective and environmentally friendly operations.
Table 1. Comparison of present work with previous literature.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Speed (rpm)</th>
<th>Rated Power (kW)</th>
<th>Type of Fuel Blends</th>
<th>Engine Type</th>
<th>Observation Taken for</th>
<th>Exergy Efficiency (Max)</th>
<th>Thermal Efficiency (Max)</th>
<th>Sustainability Index (Max)</th>
</tr>
</thead>
<tbody>
<tr>
<td>The present study</td>
<td>1500</td>
<td>3.7</td>
<td>Diesel, microalgal-based biodiesel.</td>
<td>Kirloskar, 1/4, DI engine</td>
<td>25, 50, 75, and 100% (0.92, 1.96, 2.92 and 3.7 kW)</td>
<td>31.06%</td>
<td>32.7%</td>
<td>1.46</td>
</tr>
<tr>
<td>Romas et al. [31]</td>
<td>1800</td>
<td>188</td>
<td>Natural gas, Diesel, and dual fuel</td>
<td>commercial engine (Cummins 6CTA), with mechanical power of 188 kW @ 1800 rpm, coupled to an electric generator Onan Genset of 150 kW.</td>
<td>10 kW to 150 kW</td>
<td>32.6% diesel and 51.7% dual fuel</td>
<td>35% diesel and 53% dual fuel</td>
<td>-</td>
</tr>
<tr>
<td>Lopez et al. [28]</td>
<td>2250</td>
<td>34</td>
<td>Olive pomace oil and diesel blend</td>
<td>direct injection diesel engine Perkins AD 3–152 and 18.5:1 compression ratio</td>
<td>Full load with speed variation 1300–2250 RPM</td>
<td>24.27</td>
<td>NA</td>
<td>-</td>
</tr>
<tr>
<td>Bengi et al. [34]</td>
<td>2700</td>
<td>68 (91 hp)</td>
<td>Diesel, hazelnut biodiesel, and canola biodiesel fuels.</td>
<td>Mitsubishi Canter 4D31 Direct injection diesel with glow plug, 3296cc</td>
<td>At constant load with 1500, 1800, 2100 and 2400 RPM</td>
<td>36.45%</td>
<td>38.85%</td>
<td>1.43</td>
</tr>
<tr>
<td>Karthickeyan et al. [32]</td>
<td>1500</td>
<td>5.2</td>
<td>Methyl ester of Pumpkin seed oil (B1) and Moringa oleifera oil (B2)</td>
<td>Kirloskar/TV1 Direct injection, water cooled, vertical, diesel, naturally aspirated engine 01/04 5.2 kW 87.5/110 Compression Ignition (CI) 17.5:1 210 bar</td>
<td>combustion chamber bowl geometry modification 20, 40, 60, 80, 100% load</td>
<td>64.82%, 66.35% and 63.1% respectively for diesel B1 and B2</td>
<td>33.52%, 32.7% and 31.9% for diesel B1 and B2 respectively.</td>
<td>-</td>
</tr>
<tr>
<td>Abhishek et al. [35]</td>
<td>1500</td>
<td>3.6</td>
<td>blends of Diesel-ethanol and Pongamia methyl ester (PPME)</td>
<td>Kirloskar, Model TV-1, 4 stroke Water cooled, VCR Engine 87.5 mm and 110 mm</td>
<td>20, 40, 60, 80, 100 and 120%</td>
<td>30.09%</td>
<td>31.80%</td>
<td>-</td>
</tr>
</tbody>
</table>
2. Materials and Methods

Figure 1 illustrates a flowchart for production of microalgae biodiesel. Microalgae are extracted in the first step, followed by drying and dewatering in oven at 110 °C under atmospheric conditions (1 atm pressure, 27 °C). In the next step, oil is extracted by application of heat using a Soxhlet apparatus, keeping the reaction temperature about 65–70 °C using n-hexane as a solvent. The oil thus produced is utilized to produce biodiesel via a transesterification reaction of oil with methanol in the presence of catalyst. Biodiesel and glycerin are the outputs derived from transesterification reaction. A separating funnel was used to separate biodiesel and glycerin. The biodiesel produced was washed thoroughly with water to remove the remaining impurities. After water washing, the biodiesel was taken into a tank and utilized for research on diesel engines by blending it with pure diesel.

![Flow diagram of biodiesel production](image.png)

**Figure 1.** Flow diagram of biodiesel production.

2.1. Biodiesel Production

Biodiesel can be produced by various methods such as micro-emulsification, transesterification, pyrolysis, etc. It has been found out through extensive literature survey that the transesterification process (TP) is the most effective method for biodiesel production as it has high conversion rates, economical, mild reaction conditions and the biodiesel
obtained has properties closer to that of petroleum diesel, making it suitable for industrial production [2,5,7]. A wide variety of feedstocks could be utilized for the TP process for production of biodiesel as shown in Table 2, out of which the author has chosen *Spirulina* microalgae owing to the higher lipid contents present in them. Figure 1 shows the schematic process of oil extraction and biodiesel production.

Table 2. Comparison of sources of biodiesel feedstock for CI engine [36–39].

<table>
<thead>
<tr>
<th>S No</th>
<th>Crop</th>
<th>Oil Yield (Liters per Hectares)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Corn</td>
<td>172</td>
</tr>
<tr>
<td>2</td>
<td>Canola</td>
<td>1190</td>
</tr>
<tr>
<td>3</td>
<td>Coconut</td>
<td>2689</td>
</tr>
<tr>
<td>4</td>
<td>Cotton</td>
<td>325</td>
</tr>
<tr>
<td>5</td>
<td>Castor</td>
<td>1307</td>
</tr>
<tr>
<td>6</td>
<td>Camelina</td>
<td>915</td>
</tr>
<tr>
<td>7</td>
<td>Jatropha</td>
<td>1892</td>
</tr>
<tr>
<td>8</td>
<td>Hemp</td>
<td>363</td>
</tr>
<tr>
<td>9</td>
<td>Palm</td>
<td>5950</td>
</tr>
<tr>
<td>10</td>
<td>Microalgae (high oil content)</td>
<td>136,900</td>
</tr>
<tr>
<td>11</td>
<td>Microalgae (low oil content)</td>
<td>58,700</td>
</tr>
<tr>
<td>12</td>
<td>Microalgae (medium oil content)</td>
<td>97,800</td>
</tr>
<tr>
<td>13</td>
<td>Mustard</td>
<td>572</td>
</tr>
<tr>
<td>14</td>
<td>Rapeseed</td>
<td>974</td>
</tr>
<tr>
<td>15</td>
<td>Soybean</td>
<td>446</td>
</tr>
<tr>
<td>16</td>
<td>Sunflower</td>
<td>1190</td>
</tr>
</tbody>
</table>

2.2. *Oil Extraction from Dried Microalgae*

The oil was extracted from dried *Spirulina* microalgae feedstock using a Soxhlet apparatus as shown in Figure 2 [22]. The dried microalgae is presented to a Soxhlet unit, placed on top of the round bottom flask containing n-hexane. A condenser has been put on top of the Soxhlet unit to condense the vapors of n-hexane. The system is heated with a rate of 5 watts per minute, causing vaporization of n-hexane, which rises through the Soxhlet unit to the condenser, becomes condensed, and falls back into the feedstock in the Soxhlet unit. This process is repeated several times until no more extraction of oil from the feedstock is possible.

![Figure 2](image-url). Oil Extraction using Soxhlet apparatus.
2.3. Conversion of Oil to Biodiesel

Biodiesel can be made from edible and nonedible oils. Feedstock yield determines the feasibility of employing a certain feedstock for mass production, which can power the automobile sector. In the transesterification process, the fatty acid triglycerides in any type of oil or other fats are changed into fatty acid methyl esters (FAME) by reacting with ethanol in the presence of an NaOH catalyst. The transesterification process of microalgae oil to make biodiesel includes the following steps: first, before the production of biodiesel, the acid value was determined (found less than 2%) \([40]\) to avoid soap formation in the succeeding step. The microalgae oil was preheated to 60 °C in a 500 mL beaker. After that, oil and ethanol (molar ratio 1:6) and NaOH (0.6 wt% oil) were taken in a three-neck flat bottom flask. This flask was placed over hot plate magnetic stirrer at 60–64 °C and agitated at 500 rpm for 120 min. The processed oil was placed into a conical separating funnel. After a 24 h reaction period, two layers of the treated mixture were formed. Biodiesel, having a lower density, floats on top while higher-density glycerol sits at the bottom of the separating funnel. The two layers were separated through the tap provided in separating funnel. After that, the biodiesel was washed using 50 to 70% distilled water to get rid of the impurities, excess alcohol and unreacted catalyst. This washed sample was then placed again in separating funnel and the water was drained off through the tap. After that, the biodiesel was taken in a beaker and heated to over 100 °C for 25 to 30 min to remove the moisture from the biodiesel. The biodiesel formed was then blended with petro-diesel fuel in 20%, 40%, 60%, and 80% volume fractions respectively. SBF\(_{20}\), SBF\(_{40}\), SBF\(_{60}\) and SBF\(_{80}\) are the names of blended fuel biodiesels made from microalgae oil. The fatty acid composition of the produced biodiesel is shown in Table 3.

<table>
<thead>
<tr>
<th>Fatty Acid Name</th>
<th>Chemical Name</th>
<th>Composition (by wt%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lauric acid</td>
<td>C12:0</td>
<td>0.67</td>
</tr>
<tr>
<td>Mystic acid</td>
<td>(C14:0)</td>
<td>22.73</td>
</tr>
<tr>
<td>Plamitie acid</td>
<td>(C16:0)</td>
<td>49.28</td>
</tr>
<tr>
<td>Palmitoleic acid</td>
<td>(C16:1)</td>
<td>2.81</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>(C18:0)</td>
<td>5.32</td>
</tr>
<tr>
<td>Oleic acid</td>
<td>(C18:1)</td>
<td>2.44</td>
</tr>
<tr>
<td>Linoleic acid</td>
<td>(C18:2)</td>
<td>5.71</td>
</tr>
<tr>
<td>Linoleic acid</td>
<td>(C18:3)</td>
<td>7.89</td>
</tr>
<tr>
<td>Gondoic acid</td>
<td>(C20:1)</td>
<td>1.06</td>
</tr>
<tr>
<td>Eicosadienoic acid</td>
<td>(C20:2)</td>
<td>2.09</td>
</tr>
</tbody>
</table>

2.4. Test Fuel for Engine

As test fuels, diesel and Spirulina algae biodiesel mixes were used. The baseline fuel was pure diesel with no additives. Spirulina algae biodiesel was mixed with diesel in various volume fractions to form SBF\(_{20}\), SBF\(_{40}\), SBF\(_{60}\), SBF\(_{80}\), and SBF\(_{100}\) as test fuels. It is possible to use algae biodiesel as a replacement to lessen the added strain on diesel. Algal biodiesel’s main advantage is that it lowers greenhouse gas emissions [10–13]. Table 4 lists the essential chemical characteristics and other fuel characteristics of the used fuels. Spirulina algae biodiesel fuel blends were created in accordance with the American Society for Testing Materials (ASTM) standard based on volume to test the engine while preserving the essential characteristics within permissible limits [41]. Algae oil biodiesel was measured and combined with diesel fuel in varied ratios of 20, 40, 60, and 80.

2.5. Research Engine

Four-stroke internal combustion diesel engines were used in the experiment. The test engine is shown in Figure 3 of the schematic diagrams and Table 5 below gives a technical description of the engine. An experimental setup included dynamometers, fuel tanks, fuel control valves, data acquisition systems (DAQ), exhaust gas analyzers, and sensors. A stability test was done on the engine to make sure that data recording would be as accurate
as possible. The engine was started under no-load conditions, running at a steady speed of 1500 rpm, which continued for 25 min to get it stabilized. The performance parameters were calculated after stable operating conditions were achieved in engine on application of different loads. The engine was cooled by letting water flow through the cooling jacket.

**Table 4. Physical and chemical properties of test fuels.**

<table>
<thead>
<tr>
<th>Blending Ratio</th>
<th>Density at 15 °C (kg/m³)</th>
<th>Viscosity at 40 °C (mm²/s)</th>
<th>Cetane Number</th>
<th>Lower Heating Value (MJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pure Diesel</td>
<td>830</td>
<td>2.6</td>
<td>48.52</td>
<td>42.50</td>
</tr>
<tr>
<td>SBF₃₀</td>
<td>836.17</td>
<td>3.03</td>
<td>49.26</td>
<td>42.28</td>
</tr>
<tr>
<td>SBF₅₀</td>
<td>842.26</td>
<td>3.54</td>
<td>49.97</td>
<td>42.05</td>
</tr>
<tr>
<td>SBF₇₀</td>
<td>848.25</td>
<td>4.14</td>
<td>50.67</td>
<td>41.82</td>
</tr>
<tr>
<td>SBF₉₀</td>
<td>854.17</td>
<td>4.84</td>
<td>51.34</td>
<td>41.59</td>
</tr>
<tr>
<td>SBF₁₀₀</td>
<td>860</td>
<td>5.66</td>
<td>52.00</td>
<td>41.36</td>
</tr>
</tbody>
</table>

**Figure 3.** (a) Schematic diagram of experimental setup; (b) Test rig of diesel engine used for data collection and analysis.
2.6. Energy Calculations

Energy and exergy analysis is important for the design aspect of the engines as it can minimize the losses when using energy and exergy balance data [42]. This analysis deals with the first and second laws of thermodynamics on the IC Engine [43]. For the analysis, the engine was considered as a control volume under steady-state conditions as shown in Figure 4; due to limitations of the engine setup, some assumptions were made, which allow the author to simplify the thermodynamic calculations, and are as follows:

- The air–fuel mixture for combustion and exhaust gases were supposed as ideal gases.
- The total energy produced by the fuel–air mixture was considered to be the energy input to the control volume; however, some part of it leaves as exhaust and other losses.
- Kinetic and potential energies were neglected.

For energy and exergy analysis, the chemical reaction represented in Equation (1) was used to determine the emission characteristics data at the experimental setup.

\[ a(C_xH_yO_z) + b(O_2 + 3.76N_2) \rightarrow cO_2 + dCO_2 + eCO + fNO_2 + gNO + hN_2 + jH_2O \]  

For energy and exergy analysis, the chemical reaction represented in Equation (1) was used to determine the emission characteristics data at the experimental setup.

\[ \sum m_{in} = \sum m_{out} \]  

### Table 5. Simulation test engine technical descriptions.

<table>
<thead>
<tr>
<th>S. No.</th>
<th>Parameters</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Engine</td>
<td>1/4, DI engine</td>
</tr>
<tr>
<td>2</td>
<td>Bore × stroke</td>
<td>80 × 110 mm × mm</td>
</tr>
<tr>
<td>3</td>
<td>Connecting rod length</td>
<td>235 mm</td>
</tr>
<tr>
<td>4</td>
<td>Compression ratio</td>
<td>17.5</td>
</tr>
<tr>
<td>5</td>
<td>Maximum power</td>
<td>3.7, kW</td>
</tr>
<tr>
<td>6</td>
<td>Fuel injection pressure</td>
<td>220 bar</td>
</tr>
<tr>
<td>7</td>
<td>Injection timing</td>
<td>23.5° b TDC</td>
</tr>
<tr>
<td>8</td>
<td>Dynamometer</td>
<td>Eddy current</td>
</tr>
</tbody>
</table>

---

Figure 4. Control volume for energy–exergy analysis.

For energy and exergy analysis, the chemical reaction represented in Equation (1) was used to determine the emission characteristics data at the experimental setup.

\[ a(C_xH_yO_z) + b(O_2 + 3.76N_2) \rightarrow cO_2 + dCO_2 + eCO + fNO_2 + gNO + hN_2 + jH_2O \]  

In the present study, the energy–exergy analysis includes finding major losses within the system, such as water cooling losses \( (Q_w) \), exhaust gas losses \( (Q_{exh}) \), along with other miscellaneous losses \( (Q_{im}) \). These analyses were done based on law of conservation of mass and energy as represented in Equations (2) and (3), respectively.

\[ \sum m_{in} = \sum m_{out} \]
\[ Q - W = \sum m_{\text{out}} h_{\text{out}} - \sum m_{\text{in}} h_{\text{in}} \]  

where \( m_{\text{in}} \) is the mass flow rate at the inlet and \( m_{\text{out}} \) is the mass flow rate at output. \( Q \) indicates the net heat input and \( h \) the specific enthalpy, respectively. Energy balance theory states that to calculate the heat lost from the engine to cooling water and heat carried by exhaust gases, the coolant inlet and outlet temperatures were needed. The flow rate of water was measured using a rotameter [44]. Net energy flow in \((Q_{\text{in}})\), the power developed by the engine (BP), water cooling losses \((Q_w)\), exhaust gas losses \((Q_{\text{exh}})\), along with other miscellaneous losses \((Q_{\text{un}})\) were calculated by using the following Equations (4)–(8)

\[ Q_{\text{in}} = m_f \times CV \]  

\[ BP = \omega \times T \]  

\[ Q_w = m_w \times C_{pw} \times (T_{w2} - T_{w1}) \]  

\[ Q_{\text{exh}} = (m_f + m_a) \times C_{pg} \times (T_{\text{exh}} - T_{\text{amb}}) \]  

\[ Q_{\text{un}} = Q_{\text{in}} - (BP + Q_w + Q_{\text{exh}}) \]  

The energy analysis is governed by the second law of thermodynamics, the principle of conservation of mass and the principle of conservation of energy. So, in the second part of the analysis, we started the exergy analysis of the IC engine by defining the exergy balance. The ratio of work done (BP) produced at the output shaft to the input fuel energy\((Q_{\text{in}})\) is termed as the piston thermal efficiency of the engine control volume, as described in Equation (9).

\[ \eta_{\text{th}} = \frac{\text{workdone} (BP)}{Q_{\text{in}}} \]  

2.7. Exergy Calculations

The maximum amount of useful work that can be obtained from any process dealing with two thermal reservoirs, the equation for exergy balance for the selected engine control volume, is represented in Equations (10) and (11).

\[ E_{\text{in}} = E_{\text{out}} + E_{\text{dest}} \]  

\[ E_{\text{air}} + E_{\text{fuel}} = E(m_{\text{out}} \times e_{\text{out}}) + E_{\text{heat}} + E_{\text{work}} + E_{\text{dest}} \]  

The terms \(E_{\text{air}}, E_{\text{fuel}}\) represent the exergy transfer rate for air intake and fuel, respectively, whereas \(E_{\text{heat}}, E_{\text{work}}\) and \(E_{\text{dest}}\) represent the exergy rate heat transfer from the source to the control volume to the environment, the exergy rate developed by the shaft power that is equivalent to the brake power calculated and the exergy rate of destruction, which is a measure of exergy destroyed in the control volume [45], and is the result of several factors such as friction and combustion. These exergy terms were calculated by Equations (12)–(19).

\[ E_{\text{fuel}} = m_f \times \varphi_{\text{fuel}} \times CV \]  

\[ E_{\text{air}} = m_{\text{air}} \times c_{p,\text{air}} \times \left[ (T_{\text{air},i} - T_{\text{amb}}) - T_{\text{amb}} \ln \left( \frac{T_{\text{air},i}}{T_{\text{amb}}} \right) \right] \]  

\[ \varphi_{\text{fuel}} = 1.0401 + 0.1728(\frac{H}{C}) + 0.0432(\frac{O}{C}) + 0.2169(\frac{S}{C}) \times \left[ 1 - 2.0628(\frac{H}{C}) \right] \]  

\[ E_{\text{heat}} = \sum \left( 1 - \frac{T_{\text{amb}}}{T_{cw}} \right) \times Q_{\text{loss}} \]  

\[ E_{\text{work}} = BP \]  

\[ E_{\text{out}} = \sum m_i (e_{tm} + e_{\text{chem}}) \]  

\[ e_{tm} = (h - h_0) - T_{\text{amb}}(s - s_0) \]
The engine rpm (1500 RPM) and compression ratio (17.5) were kept constant.

3.1. Energy Analysis

Figure 5 shows the variation in the energy distribution of different biodiesel blends with pure diesel. Power output (BP), heat lost to the exhaust gases (Qexh), heat carried away by the cooling water (Qw) and other unaccounted losses (Qun) were the major contributors in the energy analysis.

The percentage variation of BP developed compared with the pure diesel was lower by 0.76%, 1.35%, 1.94%, 2.67% and 3.64% for SBF20, SBF40, SBF60, SBF80, and SBF100, respectively, which is mainly because the higher biodiesel density and lower biodiesel LHV relative to diesel fuel values are connected to the use of biodiesel and its blends. In other words, when the engine is operating at the same speed, the mass flow is enhanced as compared to the same volumetric flow of fuel provided by the injection pump, which leads to higher fuel consumption and lowers the power developed [47,48]. The water-cooling loss was increased by 1.28%, 2.67%, 4.21%, 6.59%, and 8.69% for SBF20, SBF40, SBF60, SBF80, and SBF100.

$$
\epsilon_{\text{chem}} = RT_{\text{amb}} \ln \left( \frac{y_i}{y_r} \right) \tag{19}
$$

$$
\eta_{\text{ex}} = \frac{E_{\text{work}}}{E_{\text{in}}} \tag{20}
$$

$\eta_{\text{ex}}$ is called exergetic efficiency and is characterized as the ratio of exergetic work output to the total exergy input to the system [46] as mentioned in Equation (20), which is based on the second law of thermodynamics and gives a more accurate calculation for the performance of the system.

3. Result and Discussion

In the present study, the biodiesel produced from microalgae biomass was utilized as fuel in a research CI engine to measure the energy–exergy characteristics of the engine. Six different fuel blends were operated at four loading levels (25, 50, 75, and 100% as 0.92 kW, 1.96 kW, 2.91 kW and 3.7 kW). The engine rpm (1500 RPM) and compression ratio (17.5) were kept constant.
and SBF100, respectively. This is because biodiesel fuel has a large amount of oxygen, which encourages full combustion and raises the temperature within the cylinder [49], causing loss of power increases by cooling water [50]. The exhaust losses were decreased [27] by 0.35%, 0.80%, 1.35%, 2.34%, and 3.39% for SBF20, SBF40, SBF60, SBF80, and SBF100, respectively, due to the higher oxygen content present, which promotes the proper combustion and hence reduction the exhaust gases temperature [51]. As a result, the unaccounted losses were found to increase with the blends with high microalgae concentrations, and were observed as 2.11%, 4.01%, 5.98%, 8.59%, and 11.31% for SBF20, SBF40, SBF60, SBF80, and SBF100, respectively, when compared to pure diesel [34].

Figures 6–11 represent the variation of fuel energy contribution for diesel and blends of biodiesel and diesel blend. It was observed that the percentage share of BP and Qw increases with the increase in load whereas Qexh and Qun decrease with the increase in load for diesel and all biodiesel-diesel fuel blends. It was observed that the total input energy converted to useful output was 33.12%, heat loss by cooling water was 24.02%, heat loss by exhaust gases was 19.57%, and unaccounted losses were 23.27% for pure diesel at full loading conditions of 3.7 kw [52]. It was observed in Figures 6 and 7 that the percentage increases the amount of heat lost via water cooling by 2.57%, 2.03%, 1.85%, and 1.28% for SBF20 as compared to diesel for 0.92 kW, 1.96 kW, 2.92 kW and 3.7 kW loading conditions, respectively; the reason behind this is that biodiesel contains more oxygen so there is complete combustion, which increases the cylinder temperature and leads to higher losses [53,54]. Similarly, the exhaust gas losses were decreased by 1.73%, 0.929%, 0.8%, and 0.35%, respectively, for SBF40 as compared to diesel for 0.92 kW, 1.96 kW, 2.92 kW and 3.7 kW loading conditions [55]. The unaccounted heat loss was decreased by 2.29%, 2.24%, 1.89 and 1.71% for SBF20 as compared to diesel for 0.92 kW, 1.96 kW, 2.92 kW and 3.7 kW loading conditions, respectively.

![Image](image_url)  
**Figure 6.** Energy distribution of Diesel fuel under different loading conditions.
Figure 7. Energy distribution of SBF20 fuel under different loading conditions.

Figure 8. Energy distribution of SBF40 fuel under different loading conditions.
Figure 9. Energy distribution of SBF₆₀ fuel under different loading conditions.

Figure 10. Energy distribution of SBF₈₀ fuel under different loading conditions.
Similar results were observed for SBF_{20}, SBF_{40}, SBF_{60}, SBF_{80}, and SBF_{100}. From Figures 8–11, SBF_{40} shows percentage increases in cooling water losses of 5.71%, 4.13%, 3.49%, and 2.67%, whereas the percentage decreases in exhaust gases loss were 4.66%, 2.01%, 1.3% and 0.8%, respectively, for 0.92 kW, 1.96 kW, 2.92 kW and 3.7 kW loading conditions compared to pure diesel, as described in Figure 8. From Figure 9, it was observed that, with an increase of biodiesel proportion in the blend, for SBF_{60}, 8.4%, 6.3%, 5.96 and 4.22% water-cooling losses and 7.13%, 3.22%, 2.73%, and 1.36% exhaust losses were observed for an incremental loading condition of 25%. The losses were found to be higher for a higher blend ratio [55].

As described in Figures 10 and 11, for SBF_{80} the cooling water losses observed were 10.99%, 8.72%, 7.57% and 6.54%, while for SBF_{100} cooling water losses were recorded as 13.76%, 11.89%, 10.68% and 8.69% respectively at 0.92 kW, 1.96 kW, 2.92 kW and 3.7 kW loading conditions. This occurs due to the high oxygen content of biodiesel, which encourages complete combustion and raises internal cylinder temperature, increasing loss via cooling water [56,57]. The exhaust gas losses for SBF_{80} were observed as 10.25%, 4.64%,3.13% and 2.34% while for SBF_{100} it was recorded as 13.46%, 7.22%, 5.21%, and 3.39%, respectively.

### 3.2. Exergy Analysis

Exergy analysis may help design more effective energy systems by reducing the system’s irreversibilities. For use with diesel fuel, the exergies related to input energy, generated power, heat loss from exhaust gases, other heat loss and system irreversibilities or destruction of input energy were assessed during the exergy study [24,27,50]. The differences in thermal and exergy efficiency for test fuels with 1500 RPM speed, 17.5 CR and full load are shown in Figure 12.
The experimental engine’s exergy efficiency and thermal efficiency both followed the same pattern for all test fuel blends. Exergy efficiency has a lower value under identical circumstances than thermal efficiency. Between these two there was a 3–6% difference. Low energy effectiveness was brought on by the conversion of just a small quantity of work exergy and the destruction of most of the provided fuel exergy, which is also called inlet exergy rate [30,58], by the fuel. Diesel fuel has a higher value of both efficiencies as compared to other test fuels because of the higher calorific content of diesel [59]. When the amount of biodiesel in the mixture is raised, the exergy flow is (slightly) decreased as a result of increased fuel consumption. Due to the fuel’s chemical makeup and lower calorific content, this has occurred.

The exergy analysis depicted in Figure 13 shows that for pure diesel, brake power exergy was 31.06%, exergy exhaust rate was 18.35%, exergy destruction was 46.14% and exergy rate by heat transfer was 4.43%. For other fuels, the brake power was found to have a declining trend and decreased by 30.79%, 30.57%, 30.35%, 30.09%, and 29.75% for SBF20, SBF40, SBF60, SBF80, and SBF100 respectively. This result shows that biodiesel blends lead to a reduction in power output due to their lower calorific value [60]. Exhaust exergy and exergy transfer by heat were following an increasing trend. Exhaust gas exergy flow rate increased as a result of the fuel mixture, presumably as a result of incomplete combustion brought on by the addition of more long-chain hydrocarbons to the fuel, and was observed to increase by 18.35%, 18.4%, 18.46%, 18.54%, 18.7%, and 18.86% for exergy exhaust rate and by 4.43%, 4.5%, 4.56%, 4.62%, 4.69% and 4.77% for the exergy rate transfer, for SBF20, SBF40, SBF60, SBF80, and SBF100, respectively. Most input energy is lost during combustion, and only a tiny portion of it can be transformed into exergetic work during an engine cycle. The major portion of input exergy lost is in the form of exergy destruction [23,59], which was observed as an increasing trend with the increase of blend ratio, was found to be 46.14%, 46.29%, 46.39%, 46.47%, 46.51%, and 46.61% for SBF20, SBF40, SBF60, SBF80, and SBF100 respectively. The exergy analysis shows that engine operation was more sustainable for diesel fuel than other biodiesel blends [24,61]. SBF20 shows the closest result to diesel fuel. SBF20 biodiesel blends have good potential for working in a diesel engine.

![Figure 12. Thermal and Energetic Efficiency of different biodiesel blends.](image-url)
rate and by 4.43%, 4.5%, 4.56%, 4.62%, 4.69%, and 4.77% for the exergy rate transfer, for SBF20, SBF40, SBF60, SBF80, and SBF100, respectively. Most input energy is lost during combustion, and only a tiny portion of it can be transformed into exergetic work during an engine cycle. The major portion of input exergy lost is in the form of exergy destruction, which was observed as an increasing trend with the increase of blend ratio, was found to be 46.14%, 46.29%, 46.39%, 46.47%, 46.51%, and 46.61% for SBF20, SBF40, SBF60, SBF80, and SBF100, respectively. The exergy analysis shows that engine operation was more sustainable for diesel fuel than other biodiesel blends. SBF20 shows the closest result to diesel fuel. SBF20 biodiesel blends have good potential for working in a diesel engine.

Figure 13. Exergy balance of Diesel and biodiesel blends at 1500 RPM and full loading conditions.

3.3. Sustainability Index

Figure 14 depicts the sustainability index (SI) parameter for the diesel system used with different fuel blends under four different loads. The engine’s energy efficiency has a direct impact on the sustainability score. As a result, the operational parameters' impacts under the same operating circumstances were comparable to their effects on the exergy efficiency. Since the environmental effect and the sustainability index of a process are mutually exclusive, and were examined for all fuel sources, the diesel engine sustainability index increased at 1500 revolutions per minute. Then, during the engine tests taken into consideration, the diesel engine’s sustainability index varied from 1.27 to 1.46. When the results were calculated based on the fuel type, it became clear that the engine running on diesel fuel was more environmentally friendly than the engine running on biodiesel blends.

3.4. Statistical Analysis

According to the ANOVA results as represented in Tables 6 and 7, the Model F-value of 41,217.33 implies the model is significant. There is only a 0.01% chance that an F-value this large could occur due to noise. p-values less than 0.0500 indicate model terms are significant. In this case A, B, AB, A^2, B^2, A^2B, AB^2, and A^3 (A stands for load and B stands for blend ratio) are significant model terms, where the load has the biggest impact on BP, followed by the blend ratio. The p-value reflects this, and the inference made is within sizable error bounds. The surface plot of BP (percentage of energy) versus blend and load (percentage) is shown in Figure. It demonstrates that the maximum BP is observed at the highest load; the surface plot can be used to observe the relative variation of BP, and load. The lack-of-fit F-value of 0.04 implies that the lack-of-fit is not significant relative to the pure error. There is a 99.98% chance that a lack-of-fit F-value this large could occur due to noise. A non-significant lack of fit is good. Adeq. Precision measures the signal to noise ratio. A ratio greater than four is desirable. Our ratio of 562.754 indicates an adequate signal. This model can be used to navigate the design space.
Figure 14. The variation of the sustainability index.

Table 6. Statistical analysis (ANOVA) for Brake power.

<table>
<thead>
<tr>
<th>Source</th>
<th>Sum of Squares</th>
<th>df</th>
<th>Mean Square</th>
<th>F-Value</th>
<th>p-Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>25.17</td>
<td>9</td>
<td>2.8</td>
<td>41,217.33</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>A-load</td>
<td>1.6</td>
<td>1</td>
<td>1.6</td>
<td>23,536.72</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>B-Blend</td>
<td>0.0056</td>
<td>1</td>
<td>0.0056</td>
<td>82.34</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>AB</td>
<td>0.0049</td>
<td>1</td>
<td>0.0049</td>
<td>72.39</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>A²</td>
<td>0.037</td>
<td>1</td>
<td>0.037</td>
<td>545.32</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>B²</td>
<td>0.0013</td>
<td>1</td>
<td>0.0013</td>
<td>19.1</td>
<td>0.0006</td>
</tr>
<tr>
<td>A²B</td>
<td>0.0033</td>
<td>1</td>
<td>0.0033</td>
<td>48.13</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>AB²</td>
<td>0.0007</td>
<td>1</td>
<td>0.0007</td>
<td>10.44</td>
<td>0.006</td>
</tr>
<tr>
<td>A³</td>
<td>0.0011</td>
<td>1</td>
<td>0.0011</td>
<td>16.94</td>
<td>0.001</td>
</tr>
<tr>
<td>B³</td>
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<td>0.0003</td>
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</tr>
<tr>
<td>Residual</td>
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<td>14</td>
<td>0.0001</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lack of Fit</td>
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<td>13</td>
<td>0</td>
<td>0.0397</td>
<td>0.9998</td>
</tr>
</tbody>
</table>

Table 7. Regression analysis (R²) table for Brake power.

<table>
<thead>
<tr>
<th>Std. Dev.</th>
<th>0.0082</th>
<th>R²</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean</td>
<td>3.23</td>
<td>Adjusted R²</td>
<td>0.9998</td>
</tr>
<tr>
<td>C.V.%</td>
<td>0.2547</td>
<td>Predicted R²</td>
<td>0.9997</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Adeq Precision</td>
<td>562.7536</td>
</tr>
</tbody>
</table>

4. Discussion

According to the findings of the property tests, biodiesels have lower calorific values than diesel and lower kinematic viscosity, density, and cetane number. The energy–exergy study revealed that when the volume fraction of biodiesel fuel increases relative to diesel, the heat loss via cooling water increases. This is because biodiesel fuel has a higher oxygen
concentration, which causes better combustion, a rise in cylinder temperature and an increase in heat loss via cooling water.

The primary reason for the system’s inefficiency is the exergy being destroyed by irreversible processes, mostly burning. Other factors which affect the exergy destruction include energy losses from heat transfer and exhaust gas energy losses. By devoting further research, we may be able to determine how the fuel energy can be utilized by the engine more efficiently based on these variables.

5. Conclusions and Future Scope

The research presents an energy and exergy analysis of diesel and biodiesel blends. Transesterification was used to create the biodiesel in consideration of its compatibility. The engine was operated at 1500 RPM and 17.5 CR for diesel and different blends under four loading conditions.

- The energy–exergy analysis and SI value of a direct-injected diesel engine have been significantly impacted by the characteristics of biodiesel.
- With a minor exception, the results for all blends were in good agreement with the pure diesel.
- The deviation of exergy efficiency and thermal efficiency was observed as 5.09 and 5.71% for Diesel and SBF20, respectively.
- Maximum destruction exergy was observed at the 100% loading condition with SBF100 fuel blend, at 46.60%, whereas maximum power output was for diesel and SBF20, with 31.06% and 30.79% of total fuel energy input.
- The sustainability index was found in the range of 1.27 to 1.45.
- The increase in engine load increases the sustainability index and exergy efficiency in all fuel blends.
- With the use of exergy analysis along with energy analysis, we were able to obtain findings that were more accurate and realistic.

It was shown that the evaluated biodiesels provide competitive energetic performance with diesel. The best alternative in terms of using renewable resources and reducing exhaust is biodiesel. Therefore, biodiesel may be seen as a future fuel that will become widely used when petroleum is overexploited to the point of no return. Assessment of energy and exergy distribution is important when evaluating the performance of a thermal engine operated with different types of fuels or under different operational systems, because it can give the researcher a clear picture of the energy conversion processes and recommend the best way to reduce emissions, energy losses, or performance optimization. The appropriate alternative fuel can be chosen for future studies by looking at the impact of energy losses on the sustainability index.

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Nomenclature and Symbols

BP  Brake power,
Q_{exh}  Exhaust gas losses
YGME  Yellow Grese methyl ester
FAME  Fatty ester methyl easter
HRR  Heat Release rate
ANN  Artificial neural network
DICI  Direct ignition combustion engine
PDSO  Pithecellobium dulce seed oil
CO_2  Carbon dioxide
NOx  Nitrogen oxide
ASTM  American Society for Testing Materials
RPM  Revolution per minute
DAQ  Data acquisition systems
SBF_{20}  20% of spirulina microalgae biodiesel and 80% diesel
SBF_{40}  40% of spirulina microalgae biodiesel and 60% diesel
SBF_{60}  60% of spirulina microalgae biodiesel and 40% diesel
SBF_{80}  80% of spirulina microalgae biodiesel and 20% diesel
SBF_{100}  100% of spirulina microalgae biodiesel
Q_w  Cooling water losses
Q_un  Unaccounted losses
TP  Transesterification process
GHGs  Greenhouse gases
CP  Cylinder pressure
RSM  Response surface method
SME  Soyabean methyl ester
FFA  Free fatty acid
HC  hydrocarbon
CR  Compression ratio
LHV  Lower heating value
ANOVA  Analysis of variance

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