Thermodynamic Analysis of Synthetic Fuels in Compression Ignition Engines: Energy and Exergy Perspectives

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Abstract. Synthetic fuels, often considered carbon-neutral, offer a promising alternative to conventional fossil fuels by reducing environmental impacts. The production involves carbon capture and integration with other chemical components, aiming to mitigate emissions that typically contribute to air pollution, oil spills, and acid rain caused by sulphur in fossil fuels. This study investigates the energy and exergy performance of synthetic fuel blends in a compression ignition engine, comparing them with standard diesel fuel. Experimental tests were conducted at varying engine speeds to evaluate brake power, exhaust energy, fuel energy, exergy rates, heat loss, and exergy destruction. The results demonstrate that synthetic diesel and synthetic petrol exhibit energy and exergy distributions comparable to or superior to those of conventional diesel. These findings suggest that synthetic fuels can serve as environmentally friendly substitutes, offering viable alternatives for modern transportation energy systems.

1 Introduction

The increasing global energy demand accelerated research towards synthetic fuels as promising alternatives to conventional fossil fuels. The petroleum-based fuels and synthetic fuels are produced through processes such as Fischer–Tropsch synthesis and carbon dioxide capture, which enable the conversion of coal, biomass (or) natural gas into syngas and later into liquid fuels. These fuels are often considered carbon-neutral, as the CO_2 released during combustion can be balanced by the carbon captured during production. Equally, conventional fossil fuels, while dominant in transportation and power generation, contribute predominantly to greenhouse gas emissions, oil spills, and pollutants such as sulphur, which cause acid rain and result in damage to ecosystems. In contrast, synthetic fuels offer cleaner combustion and potential carbon neutrality. Synthetic fuels, such as syngas, consist of gaseous components [1], whereas solid fuels like coal or biomass are converted from natural gas. A mixture of carbon dioxide from the atmosphere and hydrogen, on the other hand, can be used to produce climate-neutral synthetic fuel. Compression ignition (CI) engines remain central to modern transportation, and evaluating the thermodynamic performance of synthetic

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fuels in these systems is critical. Energy and exergy analyses provide insights into efficiency, losses, and sustainability by examining brake power, exhaust energy, heat loss, and exergy destruction. Synthetic fuels are claimed to be more environmentally friendly than other fossil fuels since they are carbon neutral [2]. The manufacturing process entails collecting carbon and blending it with several different materials. This study investigates synthetic diesel and petrol fuels in CI engines, comparing them with standard diesel to assess their potential as environmentally friendly substitutes with comparable performance.

2 Material Composition and Design Parameters

Synthetic fuels are widely distributed globally, with an average production of approximately 380,000 barrels per day (52,000 m^3/d), alongside numerous additional projects currently under active development. Indirect conversion refers to the process in which coal, biomass, or natural gas is transformed into a mixture of hydrogen (H_2) and carbon monoxide (CO), commonly known as syngas, through gasification [2] (or) steam methane reforming, and the syngas is ground into fluid fuel transport utilising different transformation procedures. In the Fischer-Tropsch process, the syngas responds before the stimulus, changing them into fluid items and potential waxes [2, 11]. Depending on the underlying stock, the process for manufacturing synthetic fuels through an indirect change is referred to as gas-to-liquids (GTL), coal-to-liquids (CTL), and biomass-to-liquids (BTL). The impact of the feed on petroleum items delivered during the oligomerisation of the H-ZSM-5 Fischer-Tropsch feeders has been assessed. Under adequate response conditions to evaluate the carbon dioxide dispersion of the item, the isomer circulation was not estimated [2]. This makes the fuel more sensitive to the development of the feed. The pore-obliging calculation of ZSM-5 diminished fanning, which worked on the cetane number of distillates; however, it decreased the octane number of engine fuel. Distillate had a mean CH_3/CH_2 of 0.6 (thought about with> 2 of strong phosphoric corrosive catalysed oligomerisation). The thickness and consistency have not entirely settled by the refining scope of the refining and expanded by the last edge of boiling over of the item; the atomic construction assumes a small part. The carbonyl mixtures in the feed got from Fischer-Tropsch were, to some degree, switched over completely to carboxylic acids, which brought about items with higher sharpness than the server [2]. The four-cylinder compression ignition engine features a bore and stroke of 86 mm × 68 mm, with a displacement volume of 230 cc and total capacity of 1200 cc. Operating at a compression ratio of 18:1, it achieves 81% mechanical efficiency, delivering 3.8 kW power at 3800 rpm under 25-bar pressure. The driving rod is an important aspect of the motor's intricate mechanics [3]. The shaft parts that spin in the primary bearing make up the driving rod; huge closures of the interface bar are connected to the crankpins, the wrench arms or networks that connect the crankpins, and the shaft parts. Because the driving rod is subjected to several load cycles over its administration duration, its fatigue and strength must be considered in the plan interaction. The plan and advancement of the driving rod is a significant issue in the business since it is the primary power-transmitting component. Our point is to fabricate more affordable parts with the least weight and high strength [2, 3]. Improvement in the motor will give smaller motors with less weight and high-power yield. The driving rod of an IC motor ought to have the option to support high descending power with insignificant twisting. The dependability and life of the gas-powered motor are contingent upon the strength of the driving rod to a great extent. The loading control and engine torque were measured using a hydraulic dynamometer and a test. In addition, various parameters were measured using an instrumentation device. The consumption box was a viscous container that was used to measure the engine's air mass flow and was attached to the instrumentation unit. The mass flow rate of the fuel was also measured using a calibrated glass tube and a timer [3]. The exhaust gases, exit and input temperatures were all measured using Type K thermocouples.

The engine speed was measured using a Movistrobe tachometer. The exhaust emissions were also analysed using a gas analyser. All of the CI engine's tests were conducted at speeds ranging from 500 to 3500 rpm, with 500 rpm increments. The tests were performed for long enough to use all of the fuel from the previous test, allowing the engine to run on a different type of fuel [12]. Furthermore, the test was carried out until the conditions were stabilised before the variables were used as a method for increasing the accuracy of fuel data testing. The same steps were repeated for each type of fuel [1-5]. The measurement, as mentioned above, was used to do an exergy and energy analysis of the compression ignition engine for each type of fuel under similar conditions.

2.1 Fuel Behaviour and Properties

In the present work, Synthetic Petrol Fuel (SPF) and Synthetic Diesel Fuel (SDF) blends, besides the reference conventional diesel, were utilised [4]. The research laboratories prepared the different variants of fuels. The properties of the Synthetic fuel and standard diesel are shown in Table 2 below. The colour test, distillation, viscosity, specific weight, flash point, Cetane number and lower heating value tests were taken as factors to calculate the parameters of various fuel kinds in the laboratory [4, 9]. The American Petroleum Institute (API) recommends using a manometer to determine the specific weight of the gasoline sample [4]. The colour has been measured with a Lovibond Tint metre. The cetane number for synthetic fuel mixes that were lower than pure diesel was tested, and the viscosity of each type of fuel was also determined. It was discovered that pure diesel fuel had a lower viscosity than synthetic fuel mixtures. The flashpoint, viscosity, and lower heating value tests were all completed. The designed model with different materials and components compared to those of previously used steel components, as motor burden builds, the temperature of the cylinder (Table 1) and chamber divider increments dramatically and have a positive relationship. [9] The cylinder temperature for each motor burden condition tried was assessed. This simulation is to check whether these materials will withstand the heat generated by internal combustion and shows significant results in the simulation, as we have considered the cylinder pressure to be 25 bar and the internal heat source temperature to be a maximum of 278℃. As a result, it was determined that these materials can withstand the internal heat and pressure generated by synthetic diesel fuel. Our simulation results confirm this, showing that the maximum temperature affecting the piston head surface is approximately 350 ℃, and the model is capable of enduring such extremes [2, 4].

Table 1. Comparative Properties of Synthetic and Conventional Fuels

Tested Properties	Synthetic Petrol Fuel (SPF)	Synthetic Diesel Fuel (SDF)	Standard Diesel
Specific gravity at 15.8 °C	0.8755	0.8615	0.8329
Colour	Slightly yellow	Pale Yellow	Clear yellow
Flash point (°C)	71	72	65
Viscosity @40°C (cSt)	3.87	3.75	2.95
Lower Heating Value (MJ/kg)	34.5	37.7	42.5
Cetane Number	50.67	51.55	56.35

2.2 Performance Modelling of CI Engine Thermodynamics

In spite of the fact that energy investigation was performed in light of the primary law of thermodynamics [3], exergy examination depended on the second law of thermodynamics. The engine balancing system was created to better understand the power balance of the test

engine. Before combining within the combustion chamber, air and gasoline enter the engine [3, 4]. To make the computations easier, several assumptions were made before beginning the energy and exergy study. (a) An ideal gas is a blend of gases from each type of fuel. (b) Exhaust gases and fuel mixes, considering kinetic and potential energy, were not considered. (c) The reference environment conditions were supposed to be the same P_0 = 1 atm and T_0 = 30°C, as in the reference. (d) The engine's energy loss and heat loss depart at a consistent rate from the control volume. (e) Complete combustion occurs, with exhaust gases exiting the P_p as pressure and T_p temperature are used to regulate volume. (f) The usable output of the engine is the braking power. (g) The engine was preserved as a steady-state open system for both the energy and exergy evaluations.

2.3 Combustion Heat Release Function

In contrast to more complex combustion models, the Wiebe function provides an effective and relatively simple method for estimating the mass fraction burned (MFB) during the combustion process. The Wiebe heat-release formulation is based on an exponential representation of the chemical reaction rate. This model assumes that all fuel has been injected before the end of the ignition delay period. The non-dimensional form of the Wiebe function is used to express the fractional heat-release profile, as shown in equation (1).

$$\frac{dQ_c}{d\theta} = \frac{a(m+1)}{\Delta\theta} Q_{av} \left(\frac{\theta - \theta_i}{\Delta\theta}\right)^m \exp\left[-a\left(\frac{\theta - \theta_i}{\Delta\theta}\right)^{m+1}\right] \quad (1)$$

Where, dQ_c = duration of combustion in °CA, θ = crank angle at any instance in the process, θ_i = At the commencement of the combustion process, it is the crank angle, Q_{av} = KJ of heat released per cycle, a = is the parameter which characterise the robustness of the combustion process, a = 6.908m (assumption as per Wiebe).

The mass balance for a compression ignition (CI) engine is expressed as:

$$\sum m_{in} = \sum m_{out} \quad (2)$$

where

m_{in} = mass flow rate entering the system,

m_{out} = mass flow rate leaving the system.

The energy balance of the engine is given by:

$$Q - W = \sum m_{out}\, h_{out} - \sum m_{in}\, h_{in} \quad (3)$$

where,

Q = heat input rate, W = net output power, h_{out} = rate of outlet specific enthalpy, h_{in} = rate of inlet specific enthalpy.

The input fuel energy (E) is calculated using the fuel mass fuel rate (m_{fuel}) and a lower heating value (H_u) as follows:

$$E = m_{fuel} \times H_u \quad (4)$$

The engine output power (W) is obtained using the torque (T) and the angular velocity (ω):

$$W = T \times \omega \quad (5)$$

Energy losses to the environment are evaluated using:

$Q_{loss} = E_{fuel} - (W + Q_{exh})$ (6)
where, E_{fuel} = rate of fuel energy.
The rate of energy transfer from exhaust gases to the atmosphere is defined as
$Q_{exh} = \sum m_{out} \Delta h_{out} = m_{CO} \Delta h_{CO} + m_{CO2} \Delta h_{CO2} + m_{NO} \Delta h_{NO} + m_{NO2} \Delta h_{NO2} + \ldots\ldots$ (7)
where, $\Delta h_{out} = h - h_0$, h_0 = specific enthalpy of reference gas, h = specific enthalpy of exhaust gas.

The thermal performance (η) is defined as the ratio of the power output rate to the intake (fuel) energy rate.
$\eta = (W / E_{fuel})$ (8)

To quantify irreversibilities in the system, an exergy balance is performed:

$$\sum \dot{m}_{in}\, \varepsilon_{in} = \sum \dot{m}_{out}\, \varepsilon_{out} + Ex_{heat} + Ex_{work} + Ex_{dest} \quad (9)$$

Where in and out are the input and output specific exergy, accordingly, and
Ex_{heat} = energy transfer via heat
Ex_{work} = energy associated with work output
Ex_{dest} = exergy destruction.

The chemical exergy of the fuel input was calculated, including the input exergy rate as:
$Ex_{in} = m_{fuel}\, \varepsilon_{fuel}$ (10)

The fuel's specific exergy is determined:
$\varepsilon_{fuel} = H_u\, \varphi$ (11)

The chemical entropy generation factor () is calculated using the following equation:
$\varphi = 1.0401 + 0.1728 \frac{h}{c} + 0.0432 \frac{o}{c} + 0.2169 \frac{a}{c} (1.216901 \frac{h}{c})$ (12)
where,
h = mass percentage of hydrogen
c = mass percentage of carbon
o = mass percentage of oxygen
a = mass percentage of oxygen and sulphur in the fuel.

At the same time, Ex_{work} is the rate of exergy work, which is derived by multiplying the rate of energetic work by the rate of exergy work and is calculated based on the rate of energetic work. The rate of transfer of exergy is the rate of heat exchange to the atmosphere, which may be approximated as follows:
$Ex_{work} = W$ (13)

The rate of exergy transfer is indeed the rate of heat exchange to the environment, which may be calculated as:

$$Ex_{heat} = \sum \left(1 - \frac{T_0}{T_{cw}}\right) Q_{loss} \quad (14)$$

where,
T_{cw} = average of cooling temperature of the profile
Tcwout = output temperature
Tcwin = input temperature

T_0 = dead state temperature

$Ex_{out} = \sum m_i (\varepsilon_{tm} + \varepsilon_{chem})$ (15)

where the thermo-mechanical exergy (ε_{tm}) of the exhaust is computed as follows:

$\varepsilon_{tm} = (h - h_0) - T_0(s - s_0)$ (16)

where,

s = specific entropy of the exhaust gas

h = specific enthalpy

T_0 = temperature of the reference atmosphere

Addendum zero identifies characteristics of the destroyed condition.

Here ε_{chem} is exergy of chemical of the exhaust gases, it may be calculated by using the following:

$$\varepsilon_{\text{chem},i} = \bar{R}\,T_0 \ln\left(\frac{y_i}{y_{i,0}}\right) \tag{17}$$

where,

R = the gas constant

y_i = mole fraction of exhaust gases

y_r = mole fraction of the provided object in the cited atmosphere as 0.02460 percent CO_2, 0.00040 percent CO, 4.02000 percent H_2O, 0.00006 percent H_2, 25.4500 percent O_2, 78.7600 percent N_2, and 0.91366 percent different gases.

The exergetic efficiency (e) is more precise than the thermal efficiency (t), which is derived by dividing the exergy work rate by the fuel exergy:

$$\eta_u = \frac{Ex_{work}}{Ex_{fuel}} \tag{18}$$

3 Results and Discussions

Estimates of fuel energy, loss of energy, and rate of gas emissions are estimated for each kind of fuel at varying engine speeds in energy analysis [8]. The rate of exergy destruction has also been calculated in the exergy analysis. At varying speed engines ranging from 500 to 3500 rpm, all findings were emphasized in phases [9] of standard diesel and the percentage volume of Synthetic Petrol Fuel (SPF) and Synthetic Diesel Fuel (SDF). A graph was plotted to furnish information on how the energy is distributed for each type of fuel in the CI engine. As a result, the energy balance for each kind of fuel at a static engine speed of 2500 rpm is represented in Fig. 1a to 1c [8]. The production of mean power for standard diesel fuel was determined to be approximately 2.824 kW, with an energy loss rate of roughly 2.643 kW due to exhaust gases. Furthermore, the loss of energy rate was 3.872 kW, although the normal fuel's input energy rate was roughly 9.334 kW. When expressed in percentages, it was discovered that approximately 30.23 percent of the rate of energy of input was transformed to output power, with losses of energy of around 41.53 percent of the energy input, resulting in a rate of energy transfer via exhaust gases of around 28.296 percent [9,10]. It was discovered that around 30.01 percent of the rate of energy of input was converted to power output, with energy losses accounting for the greatest proportion of exergy [11]. The energy distribution for SDF and SPF has been presented in a graph to demonstrate broad data about the impact of different types of fuel on the energy fraction. In the below Fig. 1b, 1c, it can be seen that, at the speed of engine at around 2600 rpm, the rate of input energy of SDF and SPF were found to be around 9.081 kW and 8.94 kW respectively. Clearly, it is noticeable that the rate of energy is inconsiderably lower than that of the standard diesel fuel by around 3.061 percent and 4.62 percent respectively. Also, it can be observed that the power output for SDF

and SPF was much lesser than the standard diesel by 6.64 percent and 10.22 percent respectively. The highest amount of output power was found in the standard diesel and was measured to be around 2.824 kW, whereas the lowest amount of output power was found in the SDF fuel and was observed to be 2.534 kW. The reason behind this observation can be explained as a result of the decreased heating value for the Synthetic Petrol and Diesel fuels when compared to standard diesel which further states that the rate of fuel exergy for the Synthetic Petrol and Diesel melds is much lower than pure diesel [10,11,14]. Furthermore, the energy losses for SDF and SPF were much larger than for normal diesel, at 4.181 percent and 8.082 percent, respectively. The loss of energy rate via exhaust gases for SDF and SPF was lesser than standard diesel. This finding may be explained by a number of important elements, including (i) the amount of oxygen in the fuel, (ii) the exhaust temperature, (iii) how viscous the fuel is, and (iv) the heating value of lower order. The different factors mentioned above lead to the drop of energy from the exhaust gases when it comes to SDF and SPF when compared to that of diesel [13]. Also, it can be further explained that the rates of both the rate of exhaust energy and the rate of energy losses produce identical outcomes for Synthetic Petrol and Diesel blends and standard diesel. The value of output power for standard diesel was reported to be around 26.64 percent of the net power, whereas 52.392 percent of the remaining energy was released to the environment and the other 29.99 percent of the energy was lost via the exhaust gases [12, 13].

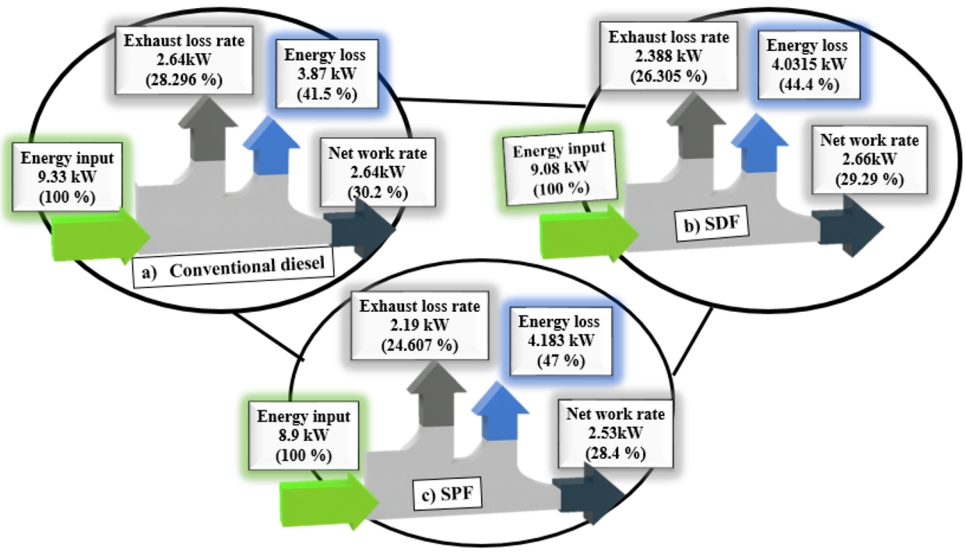

Fig. 1. Energy distribution of the engine at 2500 rpm: (a) standard diesel, (b) synthetic diesel fuel (SDF), and (c) synthetic petrol fuel (SPF).

The efficiency of the brake temperature is the output power of the input fuel. The thermal efficiency of the brakes on all types of fuel has been plotted in a graph along with the speeds of the engine, as shown in Figure 2. Clearly, it can be observed that initially the brake thermal efficiency is seen to be increasing with the increase of the speed of the engine, before it was noticed to be decreasing for each type of fuel. Also, when compared to Synthetic Petrol and Diesel, it is found that the BTE is observed to be decreasing as the blend contents in the fuel increase and as a result, the highest rate of power was found to be in standard diesel fuel at the varying speeds of the engine [14]. It was reported that the thermal efficiency of diesel is greater than SDF and SPF by around 0.912 percent and 1.81 percent for the engine static at 2500 rpm. The main cause for lower efficiency of exergy was that the brake power of almost all of the Synthetic Petrol and Diesel blends is lesser than standard diesel. Hence, this also

has an impact on the decrease of the cetane number because it has an effect on the degree of delay of ignition, braking power for Synthetic Petrol and Diesel blends [14, 15].

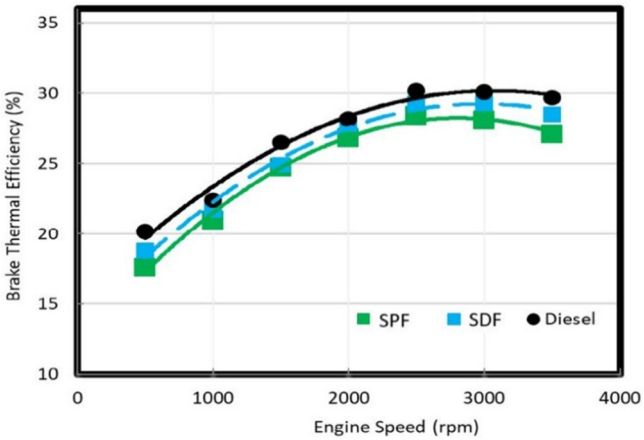

Fig. 2. Brake thermal efficiency of the engine.

A large part of the energy that is supplied to the compression ignition engine is destroyed on account of the irreversible back-up of the system [13-15]. Thus, exergy analysis was used to address this level of energy loss in the system. Therefore, the study of exergy was conducted by computing the exergies, which are linked to the power of output exergy, rate of exhaust exergy, input exergy, rate of exergy via loss of heat and irreversibility or destruction of a flowing or non-flowing system. Exergy distribution illustrations are used to represent the input exergy and output exergy rate [14,15]. Fig. 3a to 3c shows an illustration of the distribution of exergy of standard diesel, SDF and SPF, respectively, at a static engine speed of 2500 rpm. In the standard diesel, it was reported that the output power rate was found to be roughly 28.164 percent by the net exergy input, whereas the losses of exergy via the exhaust gases were found to be around just 15.293 percent by the exergy input [16, 17]. The rate of heat that is ejected is around 7.053 percent by exergy input. Also, 48.481 percent of the exergy input rate has been lost as destruction of exergy, as illustrated in Fig. 3a below. These studies have shown that around 30.01 percent of the energy input is changed into power output, whereas much of the energy input is exergy dissipation. Also, it is evident that, when it is compared to the energy rate of fuel, it is found that the fuel rate of exergy for each type of fuel is greater than the energy rate. The chemical exergy factor is bigger than unity, which explains this behaviour [18].

The input exergy rates for normal diesel, SDF, and SPF were 11.011 kW, 10.743 kW, and 9.542 kW, respectively, according to the exergy distribution diagram. This tendency may be explained by the fact that the standard diesel has a lower heating value than SDF and SPF, resulting in reduced fuel use. Furthermore, the fuel exergy trend for normal diesel is similar to the SDF and SPF. The output exergy for normal diesel, SDF, and SPF, respectively, was 3.823 kW, 2.664 kW, and 3.531 kW [20]. At 2500 rpm, the destruction of exergy of SDF and SPF was greater than that of standard diesel, according to these results. Exergy destruction was shown to account for the majority of fuel exergy losses, with 49.485 percent, 52.965 percent, and 56.021 percent for diesel, SDF, and SPF, respectively. Furthermore, the SPF fuel had the maximum exergy destruction. This can be defined by the existence of more oxygen atoms in Synthetic Petrol and Diesel melds, which causes the blends to lose more exergy. The exergy destruction for pure diesel was determined to be 51.60 percent, whereas various synthetic mixes were found to be between 54.41 and 55.12 percent.

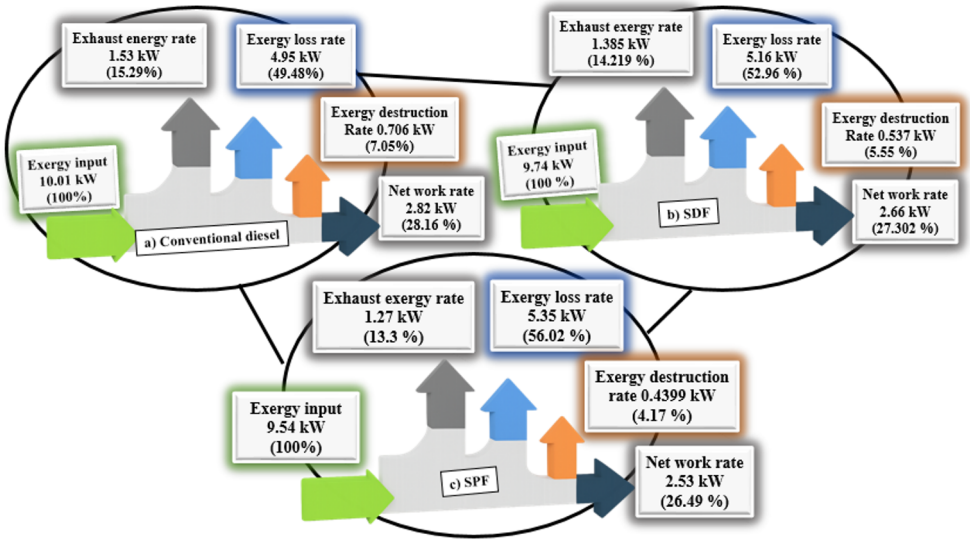

Fig. 3. Exergy distribution of the engine at 2500 rpm: (a) standard diesel fuel, (b) synthetic diesel fuel (SDF), and (c) synthetic petrol fuel (SPF).

When comparing the different fuel types with regard to the rate of exhaust exergy, it was discovered that the exhaust exergy rate declines as the mass amount in the fuel increases [22]. Therefore, standard diesel has the greatest exhaust exergy rate, as illustrated in Figures 4b and 4c. Many factors contribute to these trends, including a reduction in viscosity and a reduction in heating value of lower order of SPF and SDF melds. At the same 2500 rpm engine speed, the exergy loss rate via heat exchange of SDF and SPF was substantially lower than that of standard diesel for losses to the environment excluding exhaust [22]. The least value of losses of exergies was recorded for exergy lost by heat exchange, which were around 7.053 percent, 5.551 percent, and 4.172 percent, respectively, for standard diesel, SDF, and SPF. It can be explained that this difference between the diesel and Synthetic Petrol and Diesel melds was caused by the heating values of lower order of Synthetic Petrol and Diesel melds, which were much lower than standard diesel. Hence, resulting in the standard diesel having a higher fuel exergy rate than the Synthetic Petrol and Diesel mixes [22]. The exergy input rate is influenced by elements such as chemical exergy, as well as the rate of mass flow and heating value of the fuel. As a result, the rates of fuel energy and exergy have both followed the same path. Because the chemical exergy factor exceeds unity, the fuel exergy rate exceeds the fuel energy rate under similar circumstances. For all fuel types, the exergy rate of fuel increases as the speed of the engine increases. This may be explained by the fact that as the engine speed rises, the engine uses more fuel. SDF and SPF have lower fuel exergy rates than standard diesel, at 2.691 percent and 4.692 percent, respectively, at 2500 rpm, as compared to pure diesel. Furthermore, the compression ignition engine with the base diesel fuel at varying speeds produced the maximum rate of input exergy, whereas the SPF produced the lowest input exergy rate as shown in 4a to 4c. This tendency may be described by a decrease in the calorific value of lower-order Synthetic Petrol and Diesel blends, indicating that the energy input of Synthetic Petrol and Diesel blends is much less than standard diesel [22, 25].

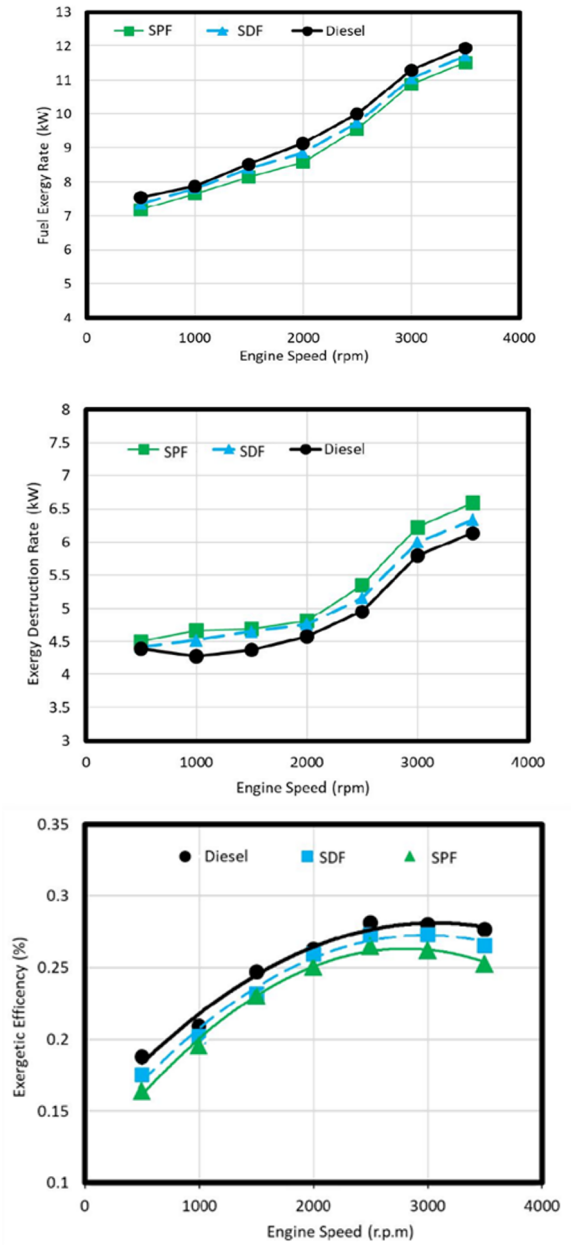

Fig. 4. (a). Thermal efficiency for each type of fuel, (b) Rate of exergy destruction for each type of fuel, (c) Exergetic efficiency for each type of fuel.

Many irreversible processes, such as the participation of turbulence within the chamber of combustion, air-fuel mixing, decrease of pressure, expansion of gases, and so on, destroy a portion of the input exergy. For all fuel types, the rate of exergy degradation rises as engine speeds increase, as illustrated in Figure 4 a to c. This phenomenon can be described by the combustion process's strong irreversibility, which results in high exergy destruction levels. Furthermore, when engine speed rises, the temperature of the cylinders rises, as does the rate of fuel consumption [27, 28]. The rate of exergy destruction of the SDF and SPF was likewise found to be somewhat greater than the rate of pure traditional diesel, exergy destruction is

required at all speeds of the engine. When compared to standard diesel, the rate of exergy destruction for the SDF and SPF increases by 4.23 percent and 8.07 percent at 2500-rpm, respectively [29]. It was also noticed that the behavior of the exergy destruction rate followed a similar pattern. This is due to incomplete combustion, which happened with Synthetic Petrol and Diesel melds, resulting in a reduction of combustion efficiency [29]. It is known that the exergetic efficiency is much highly precise than the efficiency of thermal factor, and it's been applied to assess the compression engine's accuracy. The exergetic efficiency, which improves with increasing engine speed until it reaches its maximum value before decreasing for each type of fuel. It was discovered that the exergy efficiency of standard diesel fuel rose till 3000 rpm and then fell beyond that. Furthermore, at all engine speeds, the exergetic efficiency is lesser than the thermal efficiency. The difference in efficiency between thermal and exergy was roughly 1.82 percent. Furthermore, when the exergetic efficiency of different fuel types was compared, standard diesel had the greatest exergetic efficiency for varying speeds of the engine. It can also be shown that standard diesel has a greater exergetic efficiency than SDF and SPF, with 0.862 percent and 1.671 percent, respectively. This behavior can be explained by a decrease in the calorific value of Synthetic Petrol and Diesel mixtures. The poor exergy efficiency was due to the fact that the braking power of most Synthetic Petrol and Diesel blends is much lesser than standard diesel [31]. As per the findings of the engine's energy and exergy study, as the speed of the engine accelerates, so does amount of fuel provided to the CI engine. As per the findings, the majority of fuel exergy is lost as a consequence of exergy degradation. As a result, when high efficiency is desired, it is essential to minimize destruction causes. Exergy losses occur as a result of exhaust are the second source of exergy losses [32]. As a result, enhancing exergetic efficiency requires recycling exhaust gases. Furthermore, heat transfer-related exergy losses lead to a drop in output exergy power. It is reasonable to infer that engine optimization based on exergy is achievable [33, 34] based on the exergy data.

4 Conclusion

This study examined the energy and exergy performance of synthetic fuel blends in a compression ignition engine and compared them with standard diesel fuel. Experimental tests conducted at varying engine speeds revealed that fuel exergy and exergy destruction both increase with speed for all fuels. However, synthetic diesel (SDF) and synthetic petrol fuel (SPF) demonstrated lower energy and exergy losses through exhaust gases and heat transfer compared to conventional diesel. Despite this, standard diesel achieved the highest thermal and exergetic efficiencies, while synthetic fuels showed competitive values. Exergy destruction was identified as the dominant source of exergy loss, followed by exhaust gases and heat transfer, underscoring the importance of minimizing these losses to improve efficiency. The lower brake power observed for synthetic fuels was attributed to their reduced heating values. Overall, the findings suggest that SDF and SPF can serve as viable, environmentally friendly alternatives to standard diesel, offering comparable energy and exergy distribution characteristics. Optimization of CI engines based on combined energy and exergy analysis can therefore play a vital role in advancing sustainable fuel applications.

References

1. Z. Hu, J. Fu, X. Gao, P. Lin, Y. Zhang, P. Tan, D. Lou, Waste cooking oil biodiesel and petroleum diesel soot from diesel bus: A comparison of morphology, nanostructure, functional group composition and oxidation reactivity, Fuel, **321,** 124019 (2022).

2. C. Sejkora, L. Kühberger, F. Radner, A. Trattner, T. Kienberger, Exergy as criteria for efficient energy systems – Maximising energy efficiency from resource to energy service, an Austrian case study. Energy, **239**, 122173 (2022).
3. A.G.M.B. Mustayen, M.G. Rasul, X. Wang, M. Negnevitsky, J.M. Hamilton, Remote areas and islands power generation: A review on diesel engine performance and emission improvement techniques, Energy Conversion and Management, **260,** 115614, (2022).
4. A. Tozlu, Techno-economic assessment of a synthetic fuel production facility by hydrogenation of CO_2 captured from biogas. International Journal of Hydrogen Energy, **47(5)**, 3306–3315 (2022).
5. M. Mikulski, J. Hunicz, K. Duda, P. Kazimierski, T. Suchocki, A. Rybak, Tyre pyrolytic oil fuel blends in a modern compression ignition engine: A comprehensive combustion and emissions analysis. Fuel, **320**, 123869 (2022).
6. Y. Chen, C. Shi, T. Jia, Q. Cai, L. Pan, J. Xie, L. Wang, X. Zhang, J.J. Zou, Catalytic synthesis of high-energy–density jet-fuel-range polycyclic fuel by dimerisation reaction. Fuel, **308**, 122077 (2022).
7. J.D. Morris, S.S. Daood, W. Nimmo, Machine learning prediction and analysis of commercial wood fuel blends used in a typical biomass power station. Fuel, **316**, 123364 (2022).
8. W. Meng, D. Wang, G. Li, H. Li, Z. Liu, S. Yang, A novel coal chemical looping gasification scheme for synthetic natural gas with low energy consumption for CO_2 capture: Modelling, parameters optimisation, and performance analysis. Energy, **225**, 120249 (2021).
9. T.N. Verma, P. Shrivastava, U. Rajak, G. Dwivedi, S. Jain, A. Zare, A.K. Shukla, P. Verma, A comprehensive review of the influence of physicochemical properties of biodiesel on combustion characteristics, engine performance and emissions, Journal of Traffic and Transportation Engineering, **8,** 510-533 (2021).
10. S. Katekaew, C. Suiuay, K. Senawong, V. Seithtanabutara, K. Intravised, K. Laloon, Optimisation of performance and exhaust emissions of single-cylinder diesel engines fueled by blending diesel-like fuel from Yang-hard resin with waste cooking oil biodiesel via response surface methodology. Fuel, **304**, 121434 (2021).
11. F. Andrade Torres, O. Doustdar, J.M. Herreros, R. Li, R. Poku, A. Tsolakis, J. Martins, S.A.B. Vieira de Melo, A comparative study of biofuels and Fischer–Tropsch diesel blends on the engine combustion performance for reducing exhaust gaseous and particulate emissions. Energies, **14**, 1538 (2021).
12. J. Chen, H.L. Lam, Y. Qian, S. Yang, Combined energy consumption and CO_2 capture management: Improved acid gas removal process integrated with CO_2 liquefaction. Energy, **215**, 119032 (2021).
13. Z. Lee, S. Park, Particulate and gaseous emissions from a direct-injection spark ignition engine fueled with bioethanol and gasoline blends at ultra-high injection pressure Renew. Energy, **149,** 80-90 (2020).
14. M.K. Yesilyurt, The examination of a compression-ignition engine powered by peanut oil biodiesel and diesel fuel in terms of energetic and exergetic performance parameters. Fuel, **278,** 118319 (2020).
15. M.A. Fayad, et al., Influence of alternative fuels on combustion and characteristics of particulate matter morphology in a compression ignition diesel engine. Renewable Energy, **149,** 962–969 (2020).

16. S.A. Mirbagheri, et al., Modeling of the engine performance and exhaust emissions characteristics of a single-cylinder diesel using nano-biochar added into ethanol-biodiesel-diesel blends. Fuel, **278,** 118238 (2020).
17. B.G. Şanli, E. Uludamar, Energy and exergy analysis of a diesel engine fuelled with diesel and biodiesel fuels at various engine speeds. Energy Sources, Part A: Recovery, Utilisation, and Environmental Effects, **42,** 1299–1313 (2019).
18. B.D. Patterson, F. Mo, A. Borgschulte, M. Hillestad, F. Joos, T. Kristiansen, S. Sunde, J.A. Van Bokhoven, Renewable CO_2 recycling and synthetic fuel production in a marine environment. Proceedings of the National Academy of Sciences of the USA (2019).
19. B. Gozmen Şanli, et al., Evaluation of energetic-exergetic and sustainability parameters of biodiesel fuels produced from palm oil and opium poppy oil as alternative fuels in diesel engines. Fuel, **258,** 116116 (2019).
20. M.N. Nabi, M.G. Rasul, M. Anwar, B.J. Mullins, Energy, exergy, performance, emission and combustion characteristics of diesel engine using new series of non-edible biodiesels. Renewable Energy, **140,** 647–657 (2019).
21. S.H. Teo, A. Islam, E.S. Chan, S.Y. Thomas Choong, N.H. Alharthi, Y.H. Taufiq-Yap, M.R. Awual, Efficient biodiesel production from Jatropha curcus using $CaSO_4/Fe_2O_3–SiO_2$ core–shell magnetic nanoparticles. Journal of Cleaner Production, **208,** 816–826 (2019).
22. C. Odibi, M. Babaie, A. Zare, M.N. Nabi, T.A. Bodisco, R.J. Brown, Exergy analysis of a diesel engine with waste cooking biodiesel and triacetin. Energy Conversion and Management, **198,** (2019).
23. K. Vengatesvaran, N. Prithiviraj, N. Periyasamy, Thermal analysis and material optimisation of piston in IC engine. International Journal of Advance Research and Innovative Ideas in Education, 4(3), (2018).
24. A. Jamrozik, W. Tutak, M. Pyrc, M. Gruca, M. Kočiško, Study on co-combustion of diesel fuel with oxygenated alcohols in a compression ignition dual-fuel engine. Fuel, **221,** 329–345 (2018).
25. N.A. Ansari, A. Sharma, Y. Singh, Performance and emission analysis of a diesel engine implementing polanga biodiesel and optimisation using Taguchi method. Process Safety and Environmental Protection, **120,** 146–154 (2018).
26. M. Dalla Nora, T.D.M. Lanzanova, H. Zhao Investigation of performance and combustion characteristics of a four-valve supercharged two-stroke DI engine fueled with gasoline and ethanol fuel, Fuel, **227,** 401–411 (2018).
27. S.K. Som, A. Datta, Thermodynamic irreversibilities and exergy balance in combustion processes, Progress in Energy and Combustion Science, **34,** 351-376 (2008).
28. N. Panigrahi, Energy and exergy analysis of a CI engine fuelled with Polanga oil methyl ester. Energy & Environment, **29,** 1155–1173 (2018).
29. A. Karthikeyan, J. Jayaprabakar, Energy and exergy analysis of compression ignition engine fuelled with rice bran biodiesel blends. International Journal of Ambient Energy, **40(2),** 1–11 (2017).
30. G. Karavalakis, N. Gysel, D.A. Schmitz, A.K. Cho, C. Sioutas, J.J. Schauer, D.R. Cocker, T.D. Durbin, Impact of biodiesel on regulated and unregulated emissions, and redox and proinflammatory properties of PM emitted from heavy-duty vehicles Sci. Total Environ., **584-585**, 1230-1238 (2017).

31. M. Hoseinpour, H. Sadrnia, M. Tabasizadeh, B. Ghobadian, Energy and exergy analyses of a diesel engine fueled with diesel, biodiesel–diesel blend and gasoline fumigation. Energy, **141**, 2408–2420 (2017).
32. B.S. Kul, A. Kahraman, Energy and exergy analyses of a diesel engine fuelled with biodiesel-diesel blends containing 5% bioethanol. Entropy, **18(11),** 387, (2016).
33. A. Yaşar, A.A. Ali, Investigation of effects of diesel and biodiesel fuels on energy and exergy analysis in diesel engines. Çukurova University Journal of the Faculty of Engineering and Architecture, **31(1),** 159-174, (2016).
34. N. Panigrahi, M.K. Mohanty, S.R. Mishra, R.C. Mohanty, Performance, emission, energy, and exergy analysis of a C.I. engine using Mahua biodiesel blends with diesel. International Scholarly Research Notices, **29,** 207465 (2014).