J. of Soil Sciences and Agricultural Engineering, Mansoura Univ., Vol. 15 (2):39 - 49, 2024

Journal of Soil Sciences and Agricultural Engineering

Journal homepage & Available online at: www.jssae.journals.ekb.eg

Analyzing the Performance and Emissions of Diesel Engines with Jatropha Biodiesel Using Both Numerical and Experimental Methods

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ABSTRACT

Diesel engines are the primary source of power in many types of transport applications, so many countries are currently working to reduce the vehicle exhaust emissions from diesel engines. Jatropha oil methyl ester was mixed in various amounts (20%, 40%, 60%, 80%, and 100%) during an experimental investigation to assess diesel engines' efficiency and emissions at different engine speeds, such as 500 rpm and steps increase with 250 till 1500 rpm. For various blending ratios at each speed, the engine was run at no load, quarter, half, three quarters, and full load. The brake thermal efficiency, the exhaust gas's temperature, and the brake specific fuel consumption (BSFC) are the performance characteristics that are assessed, whilst the exhaust emissions include the specific emissions of O_2, CO, CO_2, and NO_x. Combustion modeling for the fuel was done using ANSYS Fluent to predict the performance and emissions for diesel and biodiesel fuels. It was found from both the simulated and the experimental results that the use of biodiesel blends causes a 21% decrease in brake thermal efficiency and a 32% increase in brake-specific fuel consumption. The use of biodiesel fuel blends results in a rise in exhaust gas temperature. Increased biodiesel content in fuel blends results in a reduction in specific emission of CO_2. The results suggest that biodiesel made from oils that are not suitable for human consumption, such as Jatropha, may be a suitable replacement for diesel fuel in diesel engines.

Keywords*:* Ansys fluent; Biodiesel; performance; Emission; CFD simulation

INTRODUCTION

The development of alternative fuels from renewable sources that are more economical and environmentally friendly is greatly encouraged by the increasing environmental concerns and the decreasing fuel sources in recent years. Many researchers have attempted to produce derivatives based on vegetable oils that closely resemble the properties and function of diesel fuel made from petroleum (Che Mat *et al*., 2018, Jakub *et al*., 2018, Prasad *et al*., 2020, Uyumaz, 2020, Balasubramanian *et al*., 2021, Zhang *et al*., 2022, Mishra *et al*., 2023).

Vegetable oils' high viscosity is the main issue with using them directly as fuel in compression ignition (CI) engines since it results in issues like diminished fuel spray atomization and incorrect fuel injector operation (Mofijur *et al*., 2023). Four techniques, including dilution (blending), heating (thermal cracking), emulsification, and transesterification, can be used to solve the issue of higher viscosity (Xie *et al*., 2023, Basumatary *et al*., 2023). One of the most expense and effective ways to make biodiesel is through transesterification, which creates vegetable oil and fat mono-alkyl esters (Jaliliantabar *et al*., 2018).

The usage of vegetable oil increases fuel consumption, specifically brake specific fuel consumption (BSFC). Various studies found higher CO and HC emissions with vegetable oils and their blends, and lower NO_x and particulate emissions compared to mineral diesel (Agarwal *et al*., 2017, Devaraj *et al*., 2020, Seraç *et al*., 2020, Xiao *et al*., 2020, Surakasi *et al*., 2023, Gurusamy *et al*., 2023). Experiments on biodiesels with different blends as potential CI engine fuels for engine performance and emissions have been conducted. The results demonstrated that engine performance and emissions with biodiesel and its mixes were comparable to diesel engine fuel performance (Datta and Mandal, 2017, Jiaqiang *et al*., 2017, Damodharan *et al*., 2018, Ashour and Elwardany, 2020, Gad and Jayaraj, 2020, Altarazi *et al*., 2022).

Ruatpuia *et al*. (2023) studied a Jatropha plant's ability to produce biodiesel as a future source of energy. Krishana *et al*. (2023) used Ansys Fluent to analyse biodiesel mixes in CI engines. In comparison to pure diesel, their model predicted that the biodiesel blends will experience higher in-cylinder temperature and pressure during combustion. The performance of the diesel engine and its response to a B10 blended mixture were examined (Nema *et al*., 2023). Comparisons with regular diesel fuel revealed significant reductions in CO emissions, along with an increase in NOx levels, and demonstrated that the B10 fuel achieved maximum power. The biodiesel as an alternative fuel was studied (Doppalapudi *et al*., 2023). They place a strong emphasis on using biodiesel to reduce emissions. Additionally, they discussed how increased engine load and lower NO_x emissions in biodiesel combustion go in tandem, where exhaust gas recirculation can be used to reduce NOx. The heat generated during the combustion of fuels containing carbon, hydrogen, oxygen, nitrogen, and sulfur was estimated (Verevkin *et al*., 2022). Using Ansys Fluent, CFD analysis for biodiesel at high

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DOI: 10.21608/jssae.2024.261996.1214

ambient and pressure conditions as well as Jatropha burning characteristics in CI engines are described (Dixit *et al*., 2020). Many researchers worked hard to improve the Jatropha biodiesel fuel blend which ended in increasing NOx (Madiwale *et al*., 2018, Kavitha *et al*., 2019, Gautam *et al*., 2022, Khalaf *et al*., 2022, Yadav *et al*., 2023, D'Silva *et al*., 2023).

It is evident from the literature that Jatropha presents a viable option as an alternative fuel in compression ignition (CI) engines, resulting in reduced carbon monoxide (CO), hydrocarbon (HC) and smoke emissions, as well as increased nitrogen oxides (NOx). Nevertheless, it is imperative to explore its effects on engine performance, combustion characteristics, and pollutant emissions under varying operating conditions. A limited number of research studies have investigated the most optimal combination of load and speed to achieve enhanced engine efficiency and reduced emissions using Jatropha Oil Methyl Ester (JME). Furthermore, only a few investigations have developed a computational model that accurately represents the behavior of a diesel engine when using JME as fuel.

Therefore, the present study aims to provide valuable insights into the potential of JME as an alternative fuel for diesel engines. It highlights the impact of JME blend ratios on efficiency, combustion characteristics, and emissions under varying loads and speeds, and study utilizes Ansys Fluent software to model combustion and determine the optimal biofuel blend composition for improved performance and reduced emissions.

MATERIALS AND METHODS

Tested Fuels

From the seeds of the Jatropha plant, Jatropha oil methyl ester (JME) biodiesel was produced. First, the seeds of the Jatropha plant were used to make Jatropha oil. The glycerin is then removed from the Jatropha oil, and JME biodiesel is produced using a chemical process known as transesterification. A methyl ester of Jatropha oil and glycerin are the products of the procedure. The molecular structure of Jatropha oil is a triglyceride, which is common to other vegetable oils. Linoleic, Oleic, Palmitic, and Stearic acids can be found in Jatropha oil in the greatest concentrations. According to estimates, oleic acid makes up all of the fuel in biodiesel ($C_{18}H_{34}O_2$) (Li *et al*., 2019) . The molecular formula of Jatropha oil before transesterification is $C_{56}H_{101}O_6$. In this study, six fuel types (diesel, B20, B40, B60, B80, and B100 fuels) were investigated in order to fully understand the effect of biodiesel blend ratio on diesel engine performance and emissions. Table 1 provides an overview of the various Physical and chemical properties of diesel and biodiesel produced from Jatropha.

Table 1. The fuel characteristics of biodiesel blends and diesel fuel. (Balasubramanian *et al*., 2021)

Properties	method of ASTM	Oil of Jatropha	Diesel	B20	B40	B60	B80	B100
Higher calorific value [MJ.kg^{-1}]	D-4809	39.5	42.5	42.1	41.3	40.1	39.9	39.5
Density [kg.m^{-3}]	D-4052	920	832	841	853	862	875	881
Kinematic viscosity @ 40 °C [mm^2.s^{-1}]	D-445	38	3.21	3.3	3.6	3.72	4.01	4.12
Flash point, °C	D-93	235	76	88	127	143	157	162

Experimental Study

An air-cooled, two-cylinder, four-stroke, direct injection, diesel engine is used in this study. This engine can produce a maximum output of 17 kW. It is a typical engine that is often used to drive water pumps in the agricultural sector. An alternative current (AC) alternator is connected to the engine. Table 2 shows the engine's specifications. The engine is loaded using the AC alternator via a resistive load bank. Twelve heating coils make up the load bank (1000 W of each one). The 12 kW engine is fully loaded with these twelve heating coils. There are four groups connected by the heating coil's on/off switch, and one group of every three coils is series-connected to correspond to one-quarter of the load of the engine. At ranges of engine speed rpms, such as 500, 750, 1000, 1250, and 1500 rpm, the engine's performance and emissions are measured. The engine was run at various loads for different diesel and biodiesel fuel blending ratios at each speed, including at no load, quarter, half, three quarters, and full load. The measurements were the amount of fuel used, the flow rate of the intake air, and the temperature of the exhaust. Using an IMR 2800P gas analyzer, the composition of exhaust gases was analysed. Table 3 provides the measuring range and accuracy of the instruments for measurement.

Table 2. Engine specifications

Parameter	Specification
Model	Deutz F2L11
Rated power	17 kW at 1800 rpm
Cooling system	Air-cooling
No. of cylinders	Two
Speed range	500 to 2500 rpm
Compression ratio	17:1
Aspiration	Natural
Cycle	4 strokes
Bore	100 mm
Stroke	120 mm

Table 3. Instruments for measurement

Instruments used	Parameter	Measuring Range	Accuracy	Errors
Air flow meter	flow of air	160 m3 hr−1	± 1	± 1
KISTLER pressure sensor	Pressure	0–200 bar	± 0.8	± 0.1
E50eddy current dynamometer	Torque	234 N.m	± 0.3	± 0.2
IMR 2800P gas analyzer	CO2	0 − CO2 max.	± 0.2	± 0.8
	O2	0 − 20.9 vol.%	± 0.2	± 0.9
	CO	0 − 2000 ppm	± 0.4	± 1.15
	NO2	0 − 100 ppm	± 0.5	± 1
	NO	0 − 2000 ppm	± 0.4	± 0.9
	NOx	0 − 2500 ppm	± 0.2	± 0.6
	SO2	0 − 4000 ppm	± 0.5	± 1.1

The experimental system's schematic layout shown in Fig.1. Diesel fuel must be used to start the engine, and it must then be allowed to run without any load for around 15 minutes to warm up. Similarly, 15 minutes before stopping, the engine is switched back to mineral diesel. By doing this, cold starting issues with the engine can be mostly avoided.

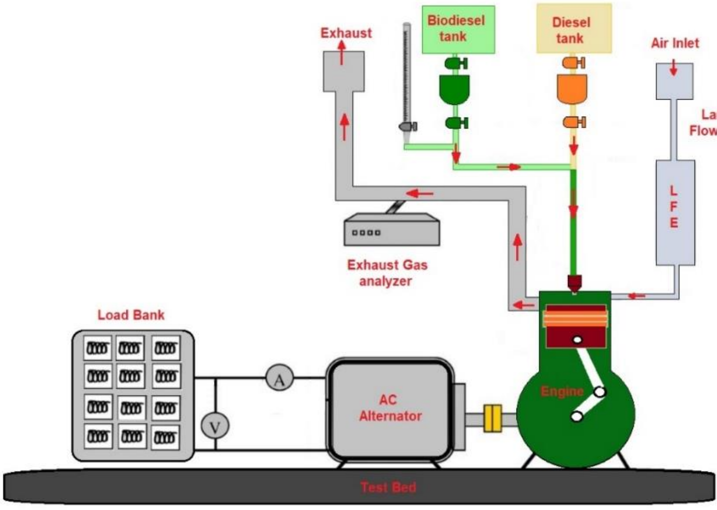


Fig. 1 Graphical view of the test system

Combustion Analysis Using ANSYS Fluent

The Modeling Challenge

The engine cylinder combustor considered is provided in Fig. 2. The geometry was designed with mesh using ANSYS Fluent and the boundaries were identified in accordance with the specifications for using boundary conditions. In this analysis, the term "flame" refers to a turbulent diffusion flame. With the atmosphere air moving at 0.5 m s^{-1} and biodiesel entering the cylinder at 50 m s^{-1} through a tiny nozzle at its center. There is approximately 28% extra air, or an equivalency ratio of 0.76. Since biodiesel injection speed is fast, the cylinder's outside wall presents less of a barrier, allowing it to expand quickly and mix with air entering the cylinder. The fundamental equations for combustion for various elements are:

Diesel: $C_{10}H_{22} + 15.5O_2 \Rightarrow 10CO_2 + 11H_2O + Heat$

B20: $C_{10}H_{22}$ (80%) + $C_{17}H_{34}O_2$ (20%) + $40O_2 \Rightarrow 27CO_2 + 28H_2O + Heat$

B100: $C_{17}H_{34}O_2 + 24.5O_2 \Rightarrow 17CO_2 + 17H_2O + Heat$

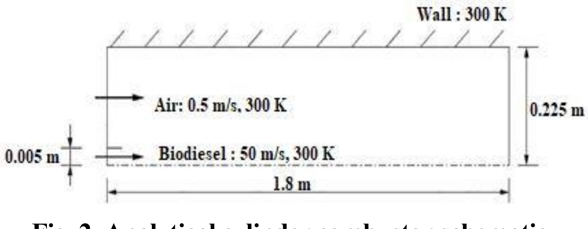


Fig. 2. Analytical cylinder combustor schematic

Ansys Fluent CFD Investigation of Biodiesel fuel Belnd

ANSYS Fluent is used to analyze the issue, and design is imported. With the help of fine mesh generation on the geometry, the entire domain is divided. Boundary conditions are used to apply the convection load. Any chemical reaction is analyzed using the ST-Model, and turbulent flow and fast reactions use the eddy dissipation model. The combustion reaction of diesel is the subject of the first analysis (B0). The same study is done for biodiesel after the outcomes (temperature, mass fraction and exhaust gases) have been taken. Different blends of diesel, such as B20, B40, B60, B80, and B100, have been studied.

For the analysis steps for the programming of the Ansys fluid first, in Ansys fluent import the geometry with the diagram of the engine then, describe the overall symmetry into fine meshes and selected the number of meshes according to the accuracy that needed in this model, also, the energy was selected and determinate it on option from the program. Once finish this step, select the viscous model as a turbulence model as k-ε (2eqn). Transport the specs using species transport inside the program. Also, it should select the material and feed boundaries condition with the material selected in order to make the model. Solving the reaction for the temperature and exhaust outputs for the nitrogen dioxide gases it's have been selected again in the model for computing the nitrogen dioxide gas then then starting to solve these steps again for the other exhaust gases.

Regarding the boundary condition for the model in the Ansys fluent. The fuel is 50 m.s^{-1} from nozzles and at a temperature of 300 K. The airspeed it was 0.5 m.s^{-1} at 300 K, and 0.76 species mass fraction, and walls remained constant at 300 K.

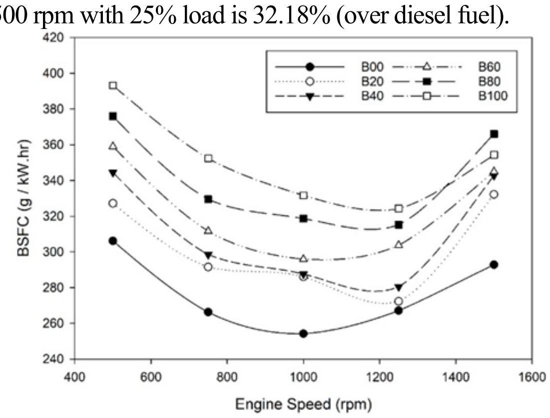
RESULTS AND DISCUSSION

Engine Performance Characteristics

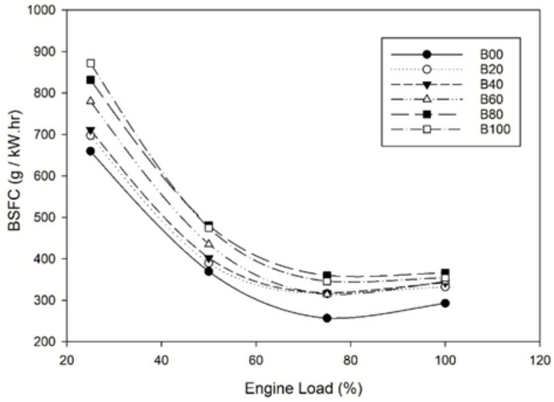
The engine performance parameters, brake specific fuel consumption, excess air factor, and brake thermal efficiency obtained versus engine loads at different speeds.

Brake Specific Fuel Consumption

As demonstrated in Fig.3.a Brake Specific Fuel Consumption (BSFC) reduced as engine speed increased, reached a minimum, and then increased at high speeds. The greater time between cycles allows for higher heat loss at low speeds, increasing BSFC. Due to higher friction losses at high speeds, BSFC rises. As may be seen in Fig.3.b the BSFC generally decreases as engine load increases. This decrease may have been caused by the higher proportion rise in braking force under load in relation to fuel consumption , which is one reason. However, there are a minor rise at the load when it is full because to poorer volumetric efficiency with increased biodiesel fuel content in the blends, BSFC rises. This may be explained by the decrease in heating value brought on by adding more biodiesel fuel to the blends (Lin *et al*., 2009). To sustain the created power, more fuel needs therefore be injected. The test results demonstrate that for speed 1500 rpm B100 under 25% load, the maximum value of BSFC is 872 (g kW^{-1} h^{-1}). The lowest value of BSFC for B100 under full load is 254 with 1000 rpm (g kW^{-1} h^{-1}). In comparison to the full test, B20 is the fuel mix that has the closest BSFC values to diesel fuel. The maximum gain in BSFC for the B100 at 1500 rpm with 25% load is 32.18% (over diesel fuel).



a) Brake Specific Fuel Consumption versus engine speed for diesel and biodiesel fuel at full load.



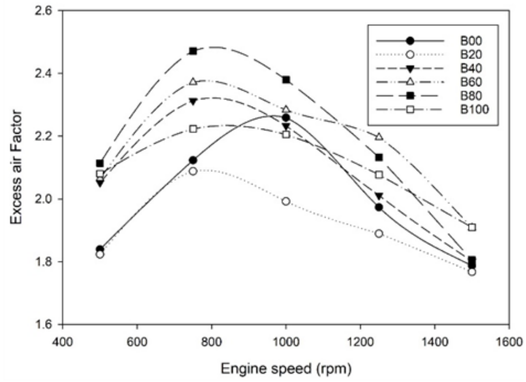
b) Brake Specific Fuel Consumption versus engine load for diesel and biodiesel fuel at 1500 rpm.

Fig. 3. Brake Specific Fuel Consumption for diesel and biodiesel fuel, a) Versus engine speed, b) Versus engine load

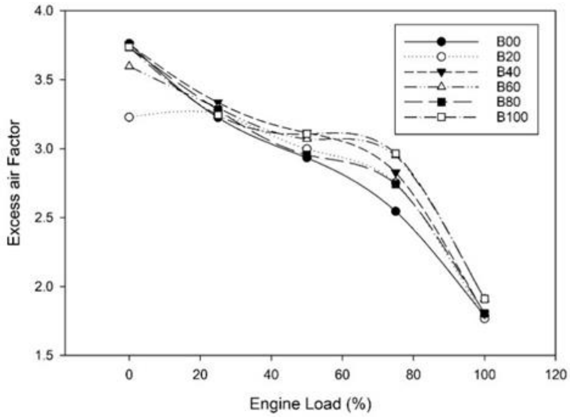
Excess air Factor

The excess air factor trends in Fig.4.a can be shown to increase with slow speeds, reach the highest, and then decline with great speed. This is a result of high specific fuel consumption caused by heat loss and friction power, respectively, at low and high speeds. Figure 4.b demonstrates how the extra air factor dropped as engine load increased as a result of higher fuel mass flow rates needed to raise the rate at which heat is released to overcome engine load. Additionally, when the proportion of biodiesel fuel in the blends grew, the extra air factor rose. This might be because lower air consumption was caused by biodiesel blends' higher oxygen content. Additionally, biodiesel fuel combines with less stoichiometric Air–fuel ratio (AFR) than diesel fuel. As a result, the surplus air factor is determined by the stoichiometric AFRs, which are less in biodiesel-containing fuel mixtures. As the proportion of blends with biodiesel fuel rises, this causes the excess air factor to rise.

According to the experiment results, the maximum Excess Air Factor is 3.7 for B100 at 1500 rpm under no-load conditions. At 1500 rpm and full load, the lowest Excess air factor value is 1.7 for B20. The greatest excess air factor increase for B100 at 1500 rpm at 75% load is 16.5% (over diesel fuel).



a) Excess air factor versus engine speed for diesel and biodiesel fuel at full load.

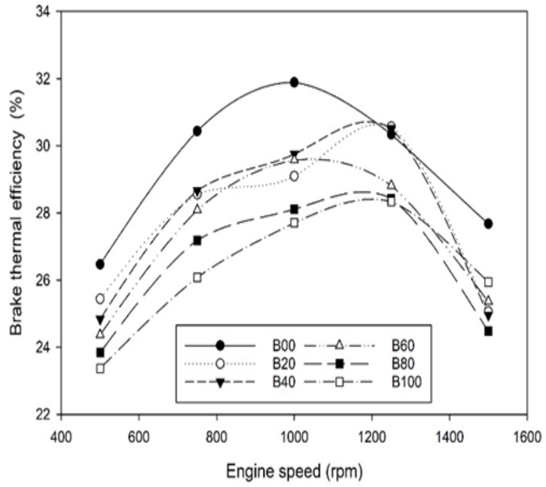


b) Excess air factor versus engine load for diesel and biodiesel fuel at 1500 rpm

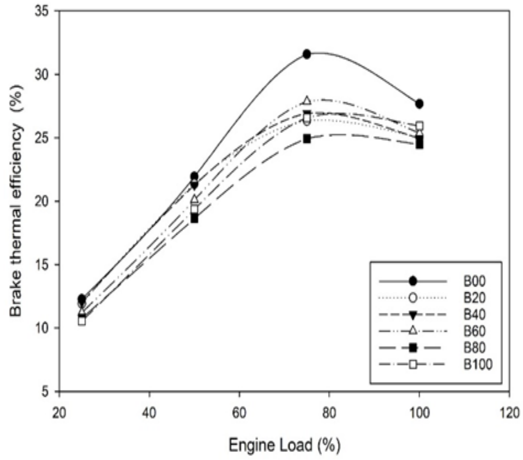
Fig. 4. Excess air factor for diesel and biodiesel fuel, a) Versus engine speed, b) Versus engine load

Brake Thermal Efficiency

According to Fig.5.a brake thermal efficiency grew as engine speeds increased, reached a maximum, and then began to decline at high speeds. This is due to high specific fuel consumption caused by heat losses and friction power, respectively, at low and high speeds. Figure 5.b demonstrates that as engine load grew, the brake thermal efficiency also increased as a result of the additional power that was produced. At full load, there is a very little decrease, but this decrease may be caused by reduced volumetric efficiency. As the ratio of biodiesel fuel in the blend increases, the brake thermal efficiency declined. Due to the mixes' poor combustion properties, which are caused by their comparatively high viscosity and low volatility, thermal efficiency has decreased. Brake specific fuel consumption (BSFC) increases for biodiesel blends over diesel fuel are another factor. The test results indicated that, at full load and 1000 rpm, brake thermal efficiency achieves its maximum value of 31.8% for B0. At 1500 rpm and 25% load, the minimum brake thermal efficiency is 10.5% for B100. The greatest reduction in brake thermal efficiency is 21% (less than diesel fuel) for B80 at 1500 rpm and 75% engine load.



a) Brake thermal efficiency versus engine speed for diesel and biodiesel at full load

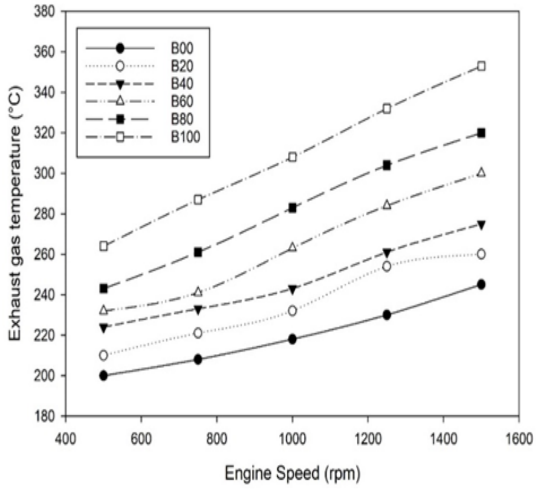


b) Brake thermal efficiency versus engine load for diesel and biodiesel at 1500 rpm

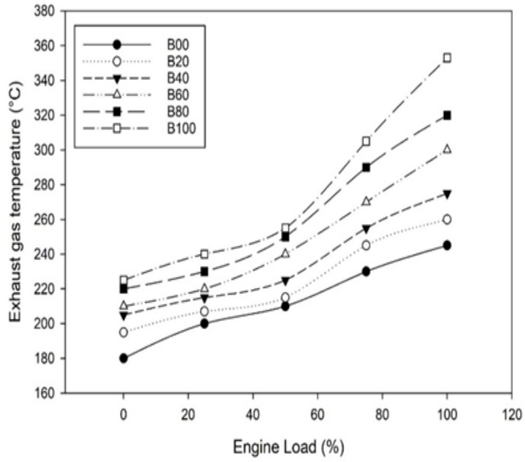
Fig. 5. Brake thermal efficiency for diesel and biodiesel fuel, a) Versus engine speed, b) Versus engine load

Exhaust gas Temperature

The temperature of the exhaust gases reveals how much waste heat is carried by the exhaust gases. As presented in Fig. 6.a the temperature of the exhaust gas rises as engine speed rises. This is caused by an increase in the volume of fuel and air entering the cylinder as the engine speed rises. This implies that more fuel is used at high speed, which raises the temperature of the exhaust gas at high speed. The temperature of the exhaust gases rises as engine load increases, as shown in Figure 6.b. This is because the engine needs more fuel to produce the additional power needed to handle the increased loading. As the proportion of biodiesel fuel in the mixes increased, so did the temperature of the exhaust gases. The reasons for this are the shorter ignition delay, longer premixed combustion, and regulated combustion stage mixing. When using biodiesel mixtures, the maximum heat release rate occurs earlier, resulting in a greater increase in in-cylinder gas temperature at the end of the compression stroke. As a result, the exhaust gas temperature rises further during the exhaust stroke due to the equal relationship between the in-cylinder gas temperature and the exhaust gas temperature. Another factor is the increased fuel pumping of biodiesel and its blends because of the lesser heating value. Test results show that the highest brake thermal efficiency value of the B100 at 1500 rpm under full load reaches 353°C. The lowest brake thermal efficiency for the B100 is 180°C at 1500 rpm no load. The B100 experiences a 44% increase in exhaust gas temperature at full load and 1500 rpm.



a) Exhaust gas temperature versus engine speed for diesel and biodiesel at full load.



b) Exhaust gas temperature versus engine load for diesel and biodiesel at 1500 rpm.

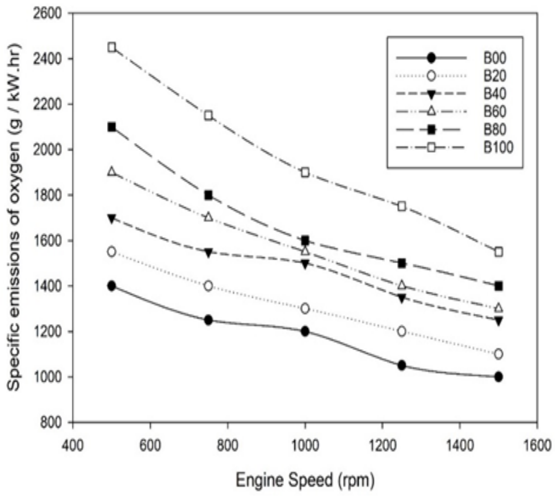
Fig. 6. Exhaust gas temperature for diesel and biodiesel fuel, a Versus engine speed, b Versus engine load

Emission Characteristics

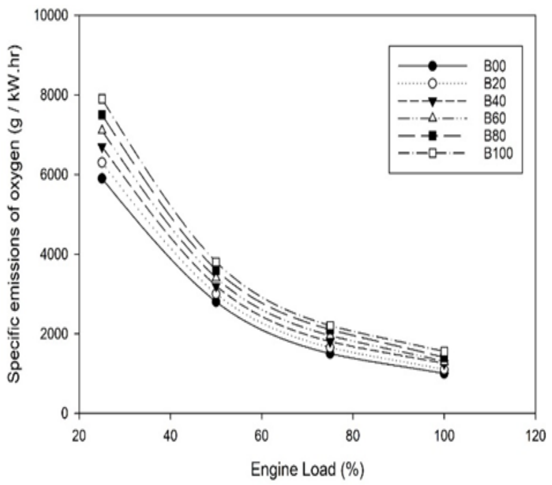
The following subsections illustrated the emissions of nitrogen oxides, carbon dioxide, carbon monoxide, and oxygen while using diesel/Jatropha biodiesel fuel blends at various engine speeds and loads.

Emissions of Oxygen

According to Fig. 7.a the specific oxygen emissions decrease as engine speed rises. This occurs as a result of an increase in engine speed, which raises the temperature of the gas inside the cylinder and causes a greater dissociation of O_2 atoms into CO_2. Figure 7.b illustrates how specific oxygen emissions decrease as engine load increases. Higher engine loads required a higher fuel consumption rate. Under conditions of constant inlet-air consumption rate and constant speed, this results in a higher fuel-to-air ratio. Therefore, less excess oxygen can be created when the engine was under more loads. With an increase in the amount of biodiesel fuel in the blends, specific emissions of Oxygen rise. This may be the result of poor fuel atomization brought on by the high viscosity of biodiesel mixes, which led to incomplete combustion. Another cause for this increase could be the presence of oxygen in biodiesel fuel mix (approximately 11% by volume).



a) Specific Emissions of Oxygen versus engine speed at full load

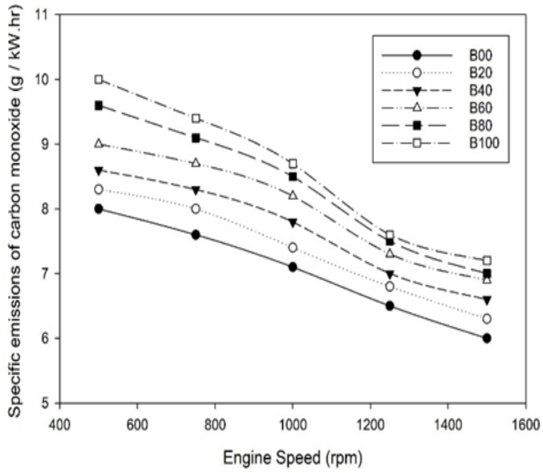


b) Specific Emissions of Oxygen versus engine load at 1500 rpm

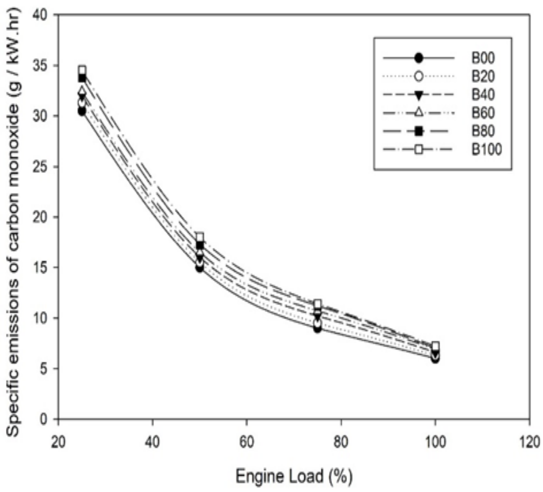
Fig. 7. Specific Emissions of Oxygen for diesel and biodiesel fuel, a) Versus engine speed, b) Versus engine load

Emissions of Carbon Monoxide

It was found that when engine speed and load rise, the specific emissions of carbon monoxide decrease, as illustrated in Fig. 8. The main cause of this is the CO_2 oxidation conversion takes longer brought on by the lower gas temperature inside the cylinder at lower engine speed and load. Increases in engine load or speed raise the temperature of the reacting gases inside the engine cylinder, which causes a decline in CO specific emissions and an increase in CO_2 specific emissions. As the quantity of biodiesel fuel in the blends increases, specific carbon monoxide emissions increase. This is feasible because biodiesel fuel mixes have a high viscosity, which makes the process of atomizing them more difficult. As a result, the engine cylinder produced locally rich mixtures. Due to the absence of oxygen, this resulted in more carbon monoxide being produced during burning. This trend is also brought on by fuel excess brought on by poor volumetric efficiency and a lack of air for complete combustion.



a) Specific Emissions of Carbon Monoxide versus engine speed at full load.

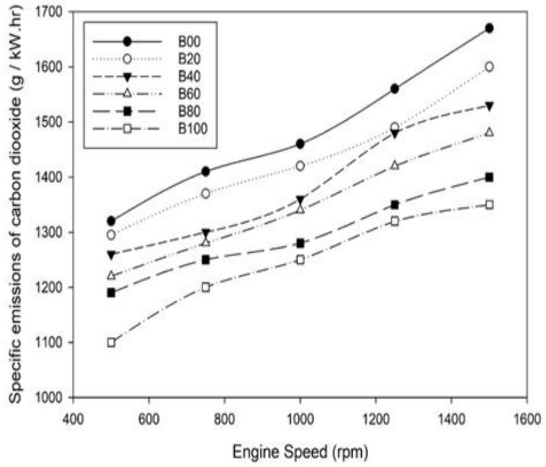


b) Specific Emissions of Carbon Monoxide versus engine load at 1500 rpm

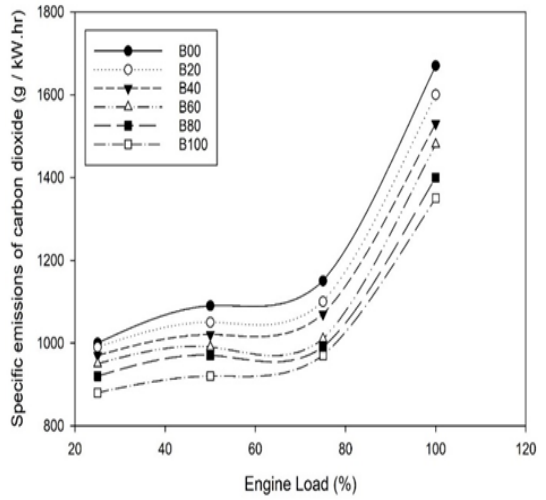
Fig. 8. Specific Emissions of Carbon Monoxide for diesel and biodiesel fuel, a versus engine speed; b versus engine load

Emissions of Carbon Dioxide

According to Fig. 9.a the specific emissions of carbon dioxide rise as engine speed rises. This is because more CO is converted to CO_2 as a result of higher engine speed, which raises the temperature of the gas in the engine cylinder. According to Fig. 9.b when engine load increases, the CO_2 specific emissions also rise. The reacting gas temperature within the engine cylinder increased as a result of the need for a higher fuel consumption rate for higher engine loads. Thus, there are higher CO_2 specific emissions created. As the amount of biodiesel fuel in the blends increased, the specific emissions of carbon dioxide gradually decreased. This is due to the low carbon content of biodiesel fuel. Moreover, it has a lower ratio of hydrogen to pure carbon than diesel fuel. Lower CO_2 emissions than diesel fuel as a result of biodiesel being burned with air.



a) Specific Emissions of Carbon dioxide versus engine speed at full load



b) Specific Emissions of Carbon dioxide versus engine load at 1500 rpm

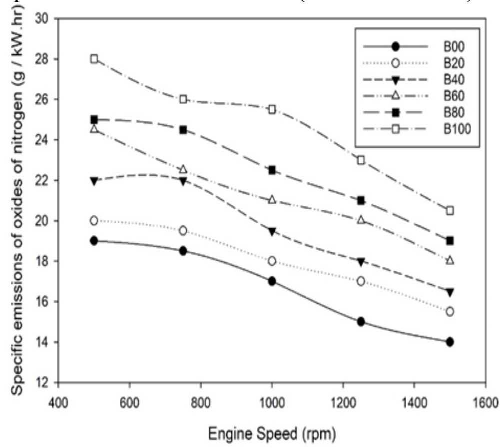
Fig. 9. Specific Emissions of Carbon dioxide for diesel and biodiesel fuel, a) versus engine speed, b) versus engine load

Emissions of Oxides Nitrogen

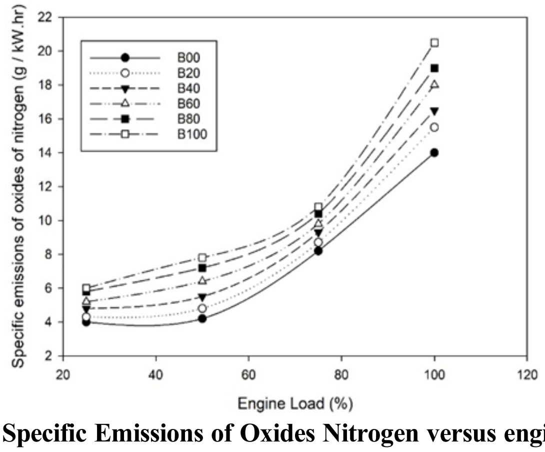
The reductions in NO_x are caused by the reductions in N_2 and O_2 atoms that occur at high speeds as a result of the excess air factor decreasing at high speeds, as illustrated in Fig.4.a. As engine speeds rise, there is a corresponding rise in internal turbulence, which causes the fuel and air mixture to mix more quickly. As a result, each engine cycle's reaction time was subsequently decreased, reducing the amount of time high gas temperatures were allowed to remain inside the cylinder and causing NO_x levels to drop. On the other hand, Fig.6.a illustrates how exhaust gas temperatures increased in response to increased engine speed (as a result of an increase in internal gas temperature), which in turn led to a rise in NO_x generation.

It is clear from Fig. 10.a that as engine speed increases, particular NO_x emissions decrease. This is probable because as engine speed increases, oxygen and nitrogen atoms and residence time decrease. An approximately constant residence duration results from the constant engine speed. A higher fuel consumption rate is required due to the increased engine load. As a result, the temperature of the gas rises, increasing the rate at which NO_x is produced. On the other hand, Fig.4.b illustrates the decreases in the excess air factor during high loads also lead to reductions in the quantities of N_2 and O_2 atoms, which results in reductions in NO_x. As the engine load increases, it is shown in Fig. 10.b that the particular NOx emissions also increase. This is most likely a result of the increased gas temperature with increased engine load. According to Fig. 10.a a certain increase in nitrogen oxide emissions occurs as the proportion of biodiesel fuel in the blends rises. The oxygen content of biodiesel may be responsible for this rise in particular nitrogen oxide emissions. This oxygen tends to react with nitrogen and produce NO_x at higher flame temperatures.

According to the experiment results, the highest value of specific nitrogen oxide emissions for the B100 at 500 rpm under full load is 0.028 kg kW^{-1} h^{-1}. For no blending B100 at 1500 rpm and 25% load, the lowest value of specific nitrogen oxide emissions is 0.004 kg kW^{-1} h^{-1}. For B100 at 500 rpm at full load, the biggest increase in NO_x-specific emissions is 47.3%. (Over diesel fuel).



a) Specific Emissions of Oxides Nitrogen versus engine speed at full load

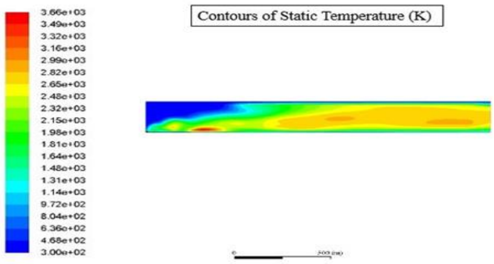


b) Specific Emissions of Oxides Nitrogen versus engine load at 1500 rpm

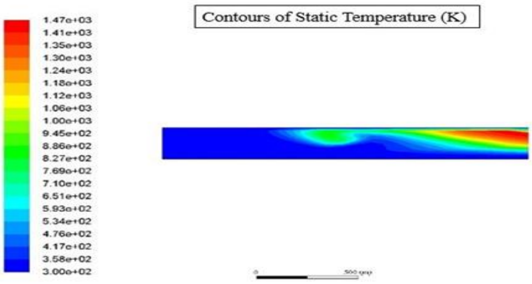
Fig. 10. Specific Emissions of Oxides Nitrogen for diesel and biodiesel fuel, a versus engine speed, b versus engine load

Emission Characteristics

Biodiesel performs better in terms of less knocks, fewer emissions, quicker ignition, and full combustion. Figure 11 depicts the temperature distribution during the combustion of diesel and biodiesel. Compared to biodiesel, diesel combustion has higher temperature characteristics. Because biodiesel does not produce as much heat as diesel, it can be blended with diesel to increase efficiency.



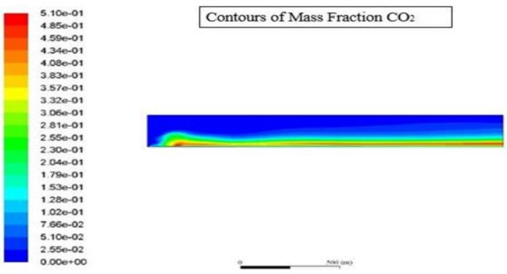
a) Temperature Profile - Diesel



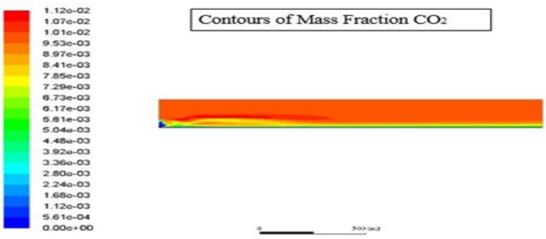
b) Temperature Profile - Biodiesel

Fig. 11. combustion's static temperature profile a Diesel, b Biodiesel

In the combustion of Diesel, CO_2 produced up to 0.51 biodiesel produces CO_2 up to 0.0112 mass fractions, which is less than diesel fuel. Fig. 12 in the combustion study of diesel and biodiesel, represents the CO_2 mass fraction. Which demonstrates that biodiesel is more environmentally beneficial than diesel fuel.



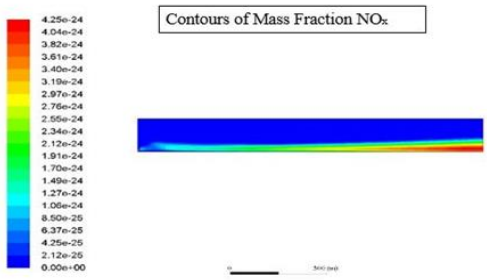
a) Mass Fraction of CO2 - Diesel



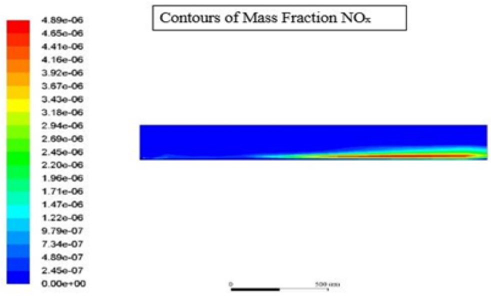
b) Mass Fraction of CO2 - Biodiesel

Fig. 12. The mass percentage of CO_2 in contour profile, a Diesel, b Biodiesel

Diesel engine combustion of NO_x can yield up to 4.25×10^{-24} mass fraction while biodiesel produces up to 4.89×10^{-6} mass fraction. When biodiesel is burned, more NO_x is created than when diesel is burned. This is one disadvantage of utilizing biodiesel as a fuel. The regulation of NOx up to a predetermined level by employing the exhaust system's Exhaust Gas Recirculation (EGR). Figure 13 shows the NOx exhaust gas created when burning diesel and biodiesel fuel.



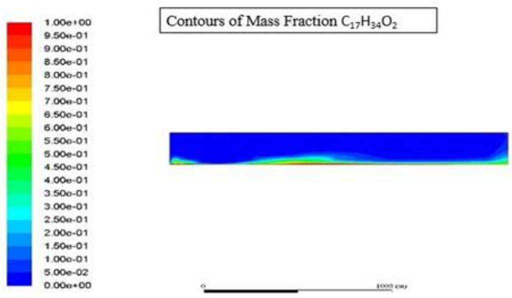
a) Mass Fraction of NOx - Diesel



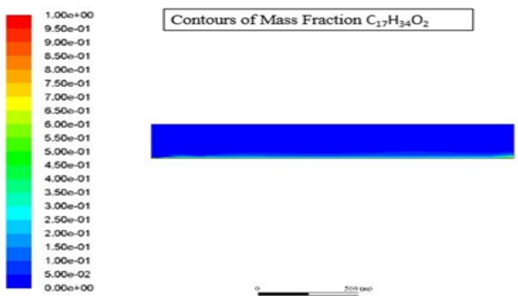
b) Mass Fraction of NOx - Biodiesel

Fig. 13. The mass percentage of NO_x in contour profile, a Diesel, b Biodiesel

Figure 14 represents the mass fraction of $C_{18}H_{34}O_2$ in diesel and biodiesel in order to forecast the lubricating property of the fuel. Biodiesel fuel is less viscous than conventional diesel fuel. In contrast to diesel, biodiesel has a lubricating quality that makes it flow smoothly into the combustion chamber. As a result, engine life extends, benefiting biodiesel applications.



a) Mass Fraction of $C_{18}H_{34}O_2$ – Diesel



b) Mass Fraction of C18H34O2 - Biodiesel

Fig. 14. The mass percentage of $C_{18}H_{34}O_2$ in contour profile, a Diesel, b Biodiesel

As previously described, biodiesel is combined with diesel while various quantities are being studied. The more biodiesel is blended into the fuel, the more environmentally friendly it is.

Biodiesel fuel burns completely since it contains oxygen by nature. NO_x emissions can be controlled depending on the engine type and operational circumstances. B20 is widely used and resembles diesel fuel in several ways. Blended fuels can reduce emissions because they enable cleaner combustion and produce less pollution in the exhaust.

Figure 15 depicts the combustion temperature and contour profile for the blends B0, B20, B40, B60, B80, and B100.

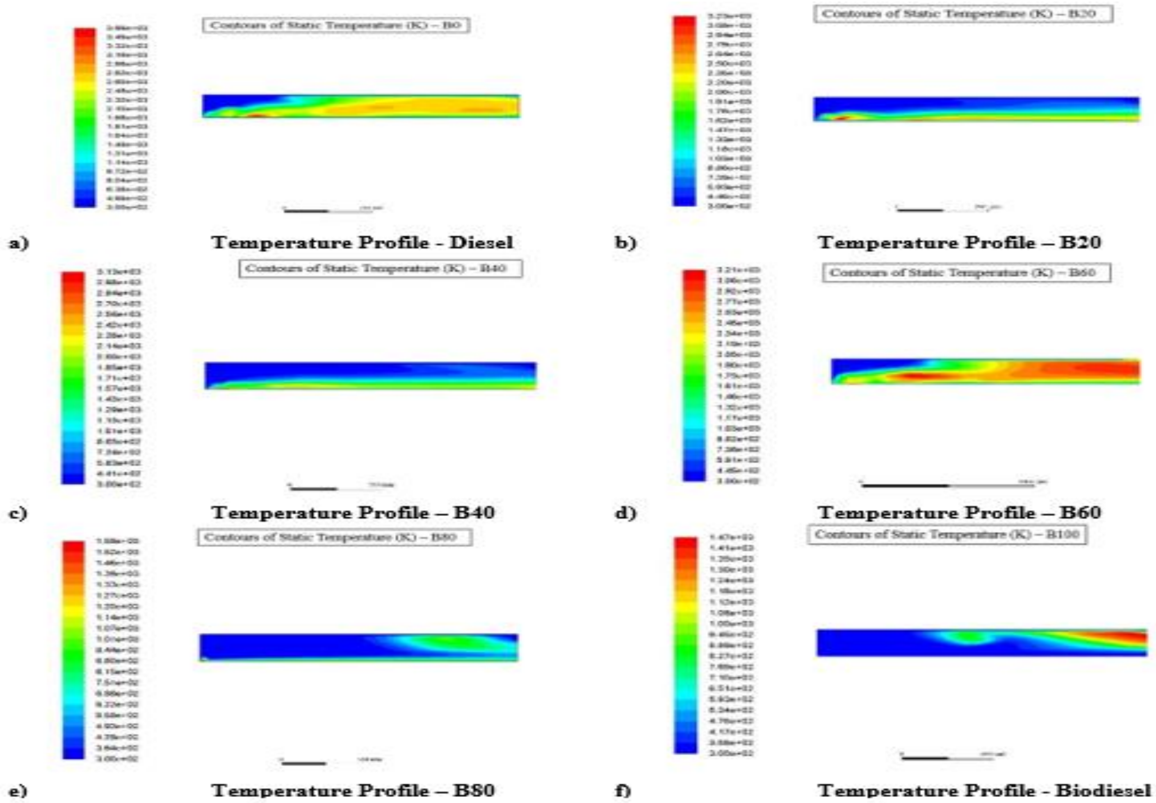


Fig. 15. The static temperature profile, a B0, b B20, c B40, d B60, e B80, f B100

Numerical and Experimental Comparison

The simulation was run using the Ansys Fluent programme, and the results for the mass fraction of NO_x and CO_2 emissions were predicted. When compared to diesel fuel, the estimated NO_x emission results for B100 indicated an increase of 15.4%. The largest increase in NO_x emission for B100 in compared to diesel fuel was 14.6%, according to experimental results. In comparison to the experimental results, the percentage variation in the simulated findings was greater between the numerical and experimental comparison for NO_x emission, there was an error of around 5.1%. the error that occurred was within the permitted limits. When compared to diesel fuel, the estimated CO_2 emission results for B100 showed a drop of 11.5%. The experimental CO_2 emission results for B100 showed a decrease of 12.3% when compared to diesel fuel. Between the numerical and experimental comparisons for CO_2 emission, there was an error of around 6.5%. Therefore, the error that occurred was within the permitted limits.

CONCLUSION

The current study was conducted on a diesel engine that had not been modified but had been made to operate in two modes. Evaluation of the engine's performance, combustion, and exhaust characteristics was the main goal of this study. Using either pure Jatropha oil methyl ester or a combination of diesel fuel, the engine was run successfully. The outcomes show that Jatropha and its mixes' biodiesel engines' performance and emissions were on par with those of conventional diesel engines.

Blends of biodiesel produce a 21% decrease in brake thermal efficiency and a 32.18% increase in brake specific fuel consumption. For B100 at 1500 rpm and full load, the biodiesel fuel blends result in a 44% increase in exhaust gas temperature.

With an increase in the amount of biodiesel in fuel blends, specific oxygen emission increases. The greatest increase in specific oxygen emissions for the B100 at 1500 rpm and 25% load is 33.8%. The specific CO emission for B100 at 1500 rpm at 25% load rises with a 13.1% increase in the biodiesel content of fuel mixtures (over diesel fuel). When the quantity of biodiesel in fuel blends is increased, the specific emission of CO2 for B100 at 1500 rpm under full load decreases by 19.16%. (Less than diesel fuel). The particular emission of NOx increases for B100 at 500 rpm at full load as fuel blends contain 47.3% more biodiesel (over diesel fuel).

To forecast the temperature profile and the mass proportion of NOx and CO_2, fuel combustion modelling for diesel, B20, B40, B60, B80, and B100 fuels was also performed using ANSYS Fluent software. The predicted results and experimental results match very well.

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تحليل أداء وانبعاثات محركات الديزل باستخدام وقود الديزل الحيوي الجاتروفا باستخدام الطرق العددية والتجريبية

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الملخص

يهدف هذا البحث الى تعظيم الكفاءة وخصائص الاحتراق والانبعاثات لمحركات الديزل التى تعمل بوقود الديزل الحيوى الجاتروفا. تم خلط إستر الميثيل لزيت الجاتروفا بنسب مختلفة (20٪، 40٪، 60٪، 80٪، و 100٪) لتقييم كفاءة محركات الديزل وانبعاثاتها عند سرعات محرك مختلفة وأحمال مختلفة.. تم دراسة أداء المحرك عن طريق دراسة الكفاءة الحرارية ، درجة حرارة غازات العادم ، واستهلاك الوقود (BSFC)، كذلك تم دراسة الانبعاثات النوعية للأوكسجين (O_2) وأول أكسيد الكربون (CO) وثاني أكسيد الكربون (CO_2)وأول أكسيد النيتروجين(NO_x) .تم إجراء محاكاة للاحتراق باستخدام برنامج Ansys Fluent لتوقع الأداء والانبعاثات لوقودي الديزل والبيوديزل. وجد من النتائج المحاكية والتجريبية على حد سواء أنة بزيادة نسب وقود البيوديزل فان ذلك يؤدي الي انخفاضًا في الكفاءة الحرارية وزيادة في استهلاك الوقود وكذلك ارتفاع في درجة حرارة غازات العادم كما أنها تعمل على تقليل انبعاثات غاز ثاني أكسيد الكربون . كما.تشير النتائج إلى أن البيوديزل المصنوع من الزيوت التي لا تصلح للاستهلاك البشري، مثل جاتروفا، قد يكون بديلاً مناسبًا لوقود الديزل في محركات الديزل.