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Experimental analysis of ethanol-diesel blends stabilised with jatropha methyl ester: Engine performance and emission characteristics

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Abstract

The growing environmental challenges and the rapid depletion of global fossil fuel reserves have driven the urgent need to explore alternative energy sources. In present work, an experimental investigation evaluated the stability of ethanol-diesel blends using jatropha methyl ester (JME) as a co-solvent, alongside engine performance and emissions at varying injection pressures and timings. Stability tests revealed that ethanol cannot blend with diesel without additives, requiring at least 4% JME for one-day stability, with E10B10D80 (10% ethanol, 10% JME, and 80% diesel by volume) blends remaining stable above 10°C. Optimal injection parameters were identified as 2.0×10^7 Pa pressure and 17° before top dead centre (BTDC) under different loads. JME proved effective as an additive, though its cost was higher than diesel, suggesting its long-term viability as fossil fuel resources diminish. Fuel consumption increased due to ethanol's lower calorific value, while thermal efficiency improved at low loads but decreased near full load. Emission analysis indicated that carbon monoxide (CO) emissions were lower at loads above half but higher at lower loads compared to pure diesel. Hydrocarbon (HC) emissions consistently rose with the blend, while a reduction in the nitrogen oxides (NO_x) emissions was observed at relatively lower load but increased near full load, showing no consistent trend. The study highlights the potential of JME as a biofuel additive, with its economic feasibility expected to improve as reliance on fossil fuels declines.

Keywords: Jatropha methyl ester; Diesel engine; Engine performance; Emission; Clean energy

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1. Introduction

Harnessing energy plays a pivotal role in the socio-economic development of the nation. However, the relentless exploitation of conventional energy resources has led to the quest for potential alternatives, prompting researchers to find potential alternatives [1]. The utilisation of conventional fuels has made various harmful impacts on the Earth, and one such impact is

Nomenclature

BTE – brake thermal efficiency,%
BSFC – brake specific fuel consumption, kg/kWh
BMEP – brake mean effective pressure, Pa
EGT –exhaust gas temperature, K

Abbreviations and Acronyms

BTDC– before top dead centre
CO – carbon monoxide
EBD – blended fuel, i.e. ethanol (E), biodiesel (B) and diesel (D)
JME – jatropha methyl ester
NOx – nitrogen oxides
GHGs– greenhouse gases
HC – unburned hydrocarbon

global warming [2]. This results in frequent shifts in climatic patterns, causing disasters. The emergence of biofuels can be viewed as a valuable alternative, serving as a bridge between economic viability and environmental sustainability [3]. Biofuels are usually derived from biomass resources, i.e. organic materials from plants and animals. Some examples of these organic materials are sugar, starch crops, oilseed crops, animal fats, algae, jatropha and many more [4,5]. These are processed and converted into by-products for obtaining bio-alcohols like ethanol, butanol, biodiesels and many more that offer a significant reduction in greenhouse gases, creating a more environmentally friendly substitution than conventional fuels [6].

In the past decades, ethanol-based engines were used in indirect combustion. But nowadays, ethanol has been used as a blending agent with advanced combustion methods that improve the overall system's performance and meet new regulations [7]. For instance, blending alcohol-based fuels (ethanol and methanol) with natural gas improves the pressure inside the chamber, resulting in proper combustion of fuels and reducing the NO_x emission as reported by Chen et al. [8]. Literature [9,10] proposed direct injection of ethanol and exhaust gas recirculation (EGR), which is being especially utilised in turbocharged engines, improving the overall engine efficacy and lowering harmful emissions, increasing overall utility. It was observed that direct injection enables charge stratification, which enables high performance and reduces fuel consumption, leading to a reduction in harmful pollutants [11]. Recently, blending of alcohols with primary fuels has gained popularity as it can improve the combustion chamber temperature and significantly reduce harmful gas emissions. Besides this, ethanol has also been utilised in pre-chamber ignition systems as a lean mixture [12,13].

Apart from ethanol, oil obtained from Jatropha has been emerging as a potential blending agent for biodiesel production. These plants are easy to cultivate, drought-tolerant, and have a high oil content (30–50%) compared to other biodiesel sources [14]. A notable example of large-scale Jatropha cultivation can be observed in Europe. To enhance oil yield, preheat treatment is applied to the seeds before oil extraction. Reports suggest that 1 kg of biodiesel can be produced from approximately 1.1 kg of raw jatropha oil [15]. Jatropha oil exhibits a calorific value comparable to that of conventional gasoline. According to Ong et al. [16], biodiesel and jatropha oil exhibit a calorific value of 38.96 MJ/kg and 40.42 MJ/kg, respectively. Despite this, jatropha-derived biodiesel has several advantageous properties, including lower viscosity, which enhances its volatility and makes it easier to transport.

The performance evaluation of jatropha biodiesel demonstrates its potential as a viable alternative fuel, showing satisfactory results with no significant technical challenges. Jatropha biodiesel exhibits high brake thermal efficiency (BTE) at a low fuel consumption rate. As per an investigation conducted by Sahoo et al. [17], the utilisation of a blend that is composed of 20% jatropha in biodiesel results in brake power values ranging from 0.09% to 2.64% higher compared to conventional diesel fuel. Interestingly, an increase in the value of brake-specific fuel consumption (BSFC) depends on higher blend ratios and decreases with increasing engine speed for all biodiesel blends. Similarly, brake-specific energy consumption (BSEC) is supposed to increase when the blend ratio is higher, but unfortunately, it decreases with an increase in the speed of the engine. Notably, the maximum increase in BSEC at 1200 rpm was recorded at 20.21% for B100, which exceeds that of conventional diesel.

Greenhouse gas (GHG) emissions have been observed to be significantly reduced with the utilisation of the jatropha biodiesel blends across various engine speeds. For instance, the B100 (pure biodiesel), B50 (50%biodiesel) and B20 (20% biodiesel) blends reduce harmful gas emissions by 64.28%, 40.9% and 28.57%, respectively, for an engine working at 2200 rpm. But if the engine speed goes down to 1200–1600 rpm, then B20 fuel showed a minimum GHG reduction of 1.29%. Upon increasing the engine speed to higher side, 1800–2200 rpm, the reduction in GHG emissions significantly increased and reached 15.84%. Notably, B100 demonstrated a maximum smoke reduction exceeding 60% at engine speeds between 1200 and 1600 rpm. However, increasing the biodiesel content in the blend generally affects the power output of the engine [18].

Huang et al. [19] investigated the performance and emissions of a diesel engine where a novel biodiesel was used, which was composed of jatropha and Chinese pistache oil. The output of the same was compared with the output of pure diesel. Their findings revealed that at higher engine speeds, if the weightage of biodiesel increases in the blend, then the BSFC can increase and reach a maximum of 6.8%. Improvements in BTE were marginal, ranging from 0.1% to 6.7%. At an engine speed of 1500 rpm, GHG emission reductions ranged from 8% to 35%, while at 2000 rpm, the reductions varied between 12% and 57%.

The aforementioned literature shows the potential of ethanol and jatropha as biofuels. However, their utility is not yet fully explored, and global petroleum prices have further intensified the urgency to identify cost-effective and environmentally friendly energy solutions. However, an appropriate blending of organic materials with diesel can be a durable solution at an affordable price. Based on the literature review, ethanol and jatropha oil have demonstrated their potential to develop a more sustainable fuel. But their blending is a tedious task that needs to be addressed properly. Therefore, in the present study combination of ethanol and jatropha blended biofuels at various proportions, along with their stability, is studied. The engine performance with the blended fuel at optimum injection pressure and timing is also studied. Besides this, engine exhaust emission was studied to evaluate the amount of emission of total carbon monoxide and unburned hydrocarbon. Lastly, a thorough comparison of the developed biodiesel was performed with diesel fuel alone to support the study.

2. Materials and methods

This section deals with the materials used and methods undertaken to achieve the objectives of the present investigation. It encompasses the utilisation of jatropha oil to prepare methyl ester, preparation of blends and examination of its stability, examination of prepared fuel properties and development of an experimental setup.

2.1. Preparation of methyl ester from jatropha oil and different blends

In the present work, the reactants used are jatropha oil, methanol and sodium hydroxide (NaOH). The glass flask was used to contain the mixture and execute the stirring magnetically. A digital weighing machine was used to measure the precise amount of catalyst to be utilised in the reaction. The flask used for mixing the methanol and NaOH was airtight. A separating funnel was used to collect the biofuel and by-product separately. In order to clamp and hold the magnetic stirrer, a dedicated stand and clamp were installed in the experimental setup. Heating was provided to the oil by using a heater, and to filter the oil, a special filter paper was used. The following procedure was opted for obtaining JME (jatropha methyl ester) from jatropha oil. A systematic approach was adopted to produce biodiesel from jatropha oil.

Initially, a known quantity of jatropha oil, based on the capacity of the flask, is filtered using filter paper and heated to 60^0C. This step removes moisture and reduces the oil's viscosity to facilitate better mixing with conventional fuel; care was taken to avoid overheating, which could break the molecular bonds. The amount of methanol and NaOH required was determined by the volume of jatropha oil used. Through experimentation, it was found that using 200 ml of methanol and 10 g of NaOH per 1000 ml of oil consistently resulted in successful transesterification. The measured methanol and NaOH were placed in an airtight glass flask, which was then vigorously shaken manually in a circular motion until the NaOH dissolved completely, forming sodium methoxide. This reaction is exothermic, causing the flask to warm slightly.

The prepared sodium methoxide was carefully poured into the reactor containing jatropha oil using a burette, ensuring no contact with the mixture or inhalation of fumes. A condenser was then attached to the reactor opening to prevent oxidation of the sodium methoxide, and the mixture was stirred continuously for 3 to 4 hours using a magnetic stirrer. After the stirring was completed, the mixture was transferred with the help of a separating funnel. The mix was then left to settle down, undisturbed, for a period of approx. 10 hours. During this settling-down period, the glycerin settled at the bottom as it was composed of free fatty acids. On the other hand, at the top, the biodiesel (jatropha methyl ester) floated. Glycerin was then carefully drained from the funnel, leaving only the biodiesel.

To purify the biodiesel, hot water was added to the separating funnel containing the jatropha methyl ester. The funnel was capped, shaken manually in circular motions, and inverted multiple times. This leads to the separation of water that contains impurities by its settling down at the bottom and followed by the draining. This washing was carried out up to 12 times until the obtained drained water had a pH value between 6 and 7. It is to be noted that the soap formation in the mix almost stopped when the pH value of the drained water reached 7.

In order to remove any moisture from the purified jatropha methyl ester, it was heated in an open environment up to 70°C. Biodiesel prepared in this way should be kept in an airtight container following the confirmation test.

Figure 1 depicts the visual comparison of ethanol-biodiesel-diesel, jatropha methyl ester (biodiesel) and high-speed diesel. After the preparation of jatropha methyl ester (biodiesel), a qualitative examination was performed for confirmation, including:

- **Visual confirmation:** The biodiesel should be light golden in colour; at least 80% of methyl ester must be produced from the raw jatropha oil after the completion of transesterification.
- **Smell:** The smell of jatropha oil is offensive, while the produced methyl esters smell pleasant and sweet.
- **Chemical examination:** The Iodoform test was used to confirm the formation of jatropha methyl ester (biodiesel) after transesterification. A few drops of Iodoform were mixed with methyl ester in a separate beaker. On the addition of Iodoform (NaOI), a sweet, fruity smell emerged, which confirms the presence of methyl ester.

Fig.1 Visual comparison of diesel, jatropha oil and jatropha methyl ester.

A simple mixing technique was adopted to prepare the blend in which ethanol, JME and high-speed diesel (HSD) were mixed together. The ratio of fuels was measured in volume before mixing them and characterised as E for ethanol, B for biodiesel, and D for diesel. Different types of blends prepared in the study are illustrated in Table 1 along with their mixing volume.

2.2. Experimental setup

For testing of the fuel, an AV1 engine of the company make Kirloskar was selected. It is a single cylinder, cooled by water and working on a 4-stroke mechanism. It is capable of producing 3.7 kW of power at an rpm of 1500. Several arrangements that are made on the engine are as follows:

1. An Eddy current dynamometer was installed in the engine to measure the torque.
2. A coolant water temperature measuring device and thermocouples for measuring the temperature of intake air and exhaust air were installed.
3. Spinner and pliers were incorporated for changing the injection pressure and timing.
4. A burette, as well as a stopwatch were installed to measure fuel consumption.
5. The flow of water was measured by a flow meter installed.

Necessary technical details of the Kirloskar engine AV1 are provided in Table 2.

The dynamometer used in the study, sourced from Power Star, powered by electric current, was connected to the engine to measure the torque. An alternator was coupled with a dynamometer and, consequently, with bulbs to give a load to the engine. A burette of volume 50 cm^3 was used to measure the HSD, and its blend comprised ethanol and biodiesel. The mixture obtained in the burette was sent to the engine via the T-valve. During the operation, the fuel consumed by the engine is supplied only by the burette as the T-valve is closed, cutting the connectivity from the fuel tank. A stopwatch was utilised to measure the time elapsed in consuming a certain volume (20 cm^3) of fuel by the engine. Hence, the flow rate of volume was then calculated by dividing volume by the time recorded on the stopwatch.

Table 1. Stability of EBD blends with respect to time

No.	Jatropha methyl ester (%)	Ethanol (%)	Diesel (%)	Separation time	Stability
1	3	17	80	Unstable	Unstable
2	4	16	80	1	Unstable
3	5	15	80	2	Unstable
4	6	14	80	4	Unstable
5	7	13	80	12	Unstable
6	8	12	80	15	Unstable
7	9	11	80	20	Unstable
8	10	10	80	Till date	Stable
9	11	9	80	Till date	Stable
10	12	8	80	Till date	Stable
11	13	7	80	Till date	Stable
12	14	6	80	Till date	Stable
13	15	5	80	Till date	Stable

Table 2. Technical specification of the AV1 Kirloskar engine.

Number of cylinders	One
Bore × Stroke	80 mm × 110 mm
Cubic Capacity	0.553 L (553 cm^3)
Compression ratio	16.5 : 1
Rated output as per BS5514/ISO 3046/ IS 10001	3.7 kW (5.0 hp) at 1500 r/min
Injector opening pressure	2.0 × 10^7 Pa
Injection timing	21° BTDC

The temperature at different zones of the engine is measured with the help of thermocouples, which are installed in to six-channel selector switch. These thermocouples are also used to measure the temperature of exhaust gas, as well as cooling water. The injector opening pressure can be adjusted by modifying the spring load on the injector needle, using a screw located on top of the injector. Additionally, the injection timing can be altered by inserting shims beneath the pump top plate. Each shim, with a thickness of 0.16 mm, corresponds to a 2° retardation in timing. Initially, two shims were positioned at 21° before TDC (top dead centre). The number of shims varied from 2 to 5. A suitable gas analyser, AVLDi 4000, was utilised to measure various emissions (carbon monoxide (CO), nitrogen oxides (NO_x) and hydrocarbons) from the engine.

The concentrations of gases coming out of the exhaust valve are identified by the use of a filtered tube connected to the electrochemical sensors. The readings recorded on the panel of the electrochemical cell have a direct relation with the volume concentration of elements detected on it, and it is expressed in parts per million (ppm).

The engine was operated at a constant speed of 1500 rpm. The accuracies were: fuel measurement ±0.5 ml, torque ±0.1 Nm and emissions ±5 ppm.

The engine performance was determined by utilising two types of fuel, viz. diesel fuel and a blended fuel (ethanol, biodiesel and diesel). Important properties of these blends used in this study, such as viscosity, density, calorific value and cetane number of these test fuels were measured in the laboratory and are compared with diesel (Table 3). Several experiments were performed to examine the performance of the engine and the properties of exhaust gases. For this, the pressure at injection was varied, and also, the timing of injection was varied. The monitoring of the inlet and outlet points of cooling water was carried out during the experiment. Intake of air and exhaust air were also monitored for each change in the load. The engine was allowed to relax and reach a stable state after every change

Table 3. Fuel properties of diesel and EBD blend

Properties	Diesel	EBD
Calorific value (MJ/kg)	44.8	43.5
Specific gravity (kg/m^3)	0.815	0.8331
Viscosity at 27°C	4.3	–
Cetane number	47	46.85

in the loading condition.

The values of injector opening pressure that are considered for the analysis are as follows: 1.86×10^7, 1.96×10^7, 2.06×10^7, 2.16×10^7 and 2.26×10^7 Pa. The preliminary examination revealed that the best output was attained for the base fuel, i.e. diesel at a pressure of 2.0×10^7 Pa, while in the case of blended fuel, the best output was attained at 1.96×10^7 Pa. The variation in the timing of injection was set in the range between 21° to 17° before TDC. This was achieved by placing shims just below the top of the pump plate. Each shim plate has a 16 mm thickness, which is equal to 2° retardations of the timing of injection. At the full load, the AV1 engine of make by Kirloskar produces 3.7 kW of power. Therefore, for the analysis, the load was varied from minimum (0.5 kW or 72332.73 Pa BMEP) to maximum (3.5 kW or 506329.11 Pa BMEP). The loading on engine was performed by switching on the bulb of specific wattage. At first, the engine runs only on diesel so that baseline data could be formed. Thereafter, the prepared fuel (E10B10D80) was used to run the engine to check the performance and to make a comparison of the prepared fuel with diesel fuel.

3. Materials and methods

Prior to the start of the experiment, the aim was to find the optimum injector opening pressure and timing for the prepared fuel, i.e. EBD. Among the selected injection pressures, as specified in section 2.2, 1.96×10^7 Pa was learned to be the optimum injector opening pressure and shim number of 4 was noted as optimum for the injection timing, at which minimum BSFC was recorded and maximum BTE was obtained. Therefore, the results that are going to be discussed afterwards are based on the injection pressure of 1.96×10^7 Pa and at 17° before TDC.

The parameter that suggests how efficient an engine is in converting fuel into output work is BSFC. To examine BSFC of diesel and prepared fuel, Fig. 2 has been plotted at different loading conditions. Upon giving a close observation to the results depicted in the figure, it was learned that as the load on the engine increases, BSFC decreases. However, once a particular load point is reached, BSFC begins to increase. In Fig. 2, the load at which BSFC has its minimum is referred to as the best economical mixture, which in this case was obtained at 80–85 % of loading for both diesel and EBD blend. For all cases of loading, BSFC was noted to be higher for E10B10D80 as compared to the BSFC for diesel. It is also apparent that at 0.5 kW or 72332.73 Pa BMEP of loading, BSFC was recorded slightly higher than BSFC for diesel. As the load on the engine increases, the BSFC for EBD starts decreasing until the load reaches 2.5 kW or 361663.65 Pa BMEP. But, as the load increases beyond 2.5 kW or 361663.65 Pa BMEP, BSFC shows an increment. The higher value of BSFC for the case of EBD was attributed to the lower heating as compared to diesel. This suggests that to obtain a particular amount of output, more EBD fuel has to burn out when compared to pure diesel.

The examination of BTE was carried out to calculate the ability of engine in terms of net power output. A graph showcasing the variation of BTE against the applied load for both diesel and EBD is presented in Fig. 3. It has been found that the engine runs on both diesel and EBD, showing an increase in BTE for every increase in the load on the engine. Notably, the highest BTE was obtained at 2.5 kW or 361663.65 Pa BMEP loading of the engine. The reason behind the higher BTE in the case of EBD fuel is attributed to the greater consumption of fuel and low content of energy for the same power output.

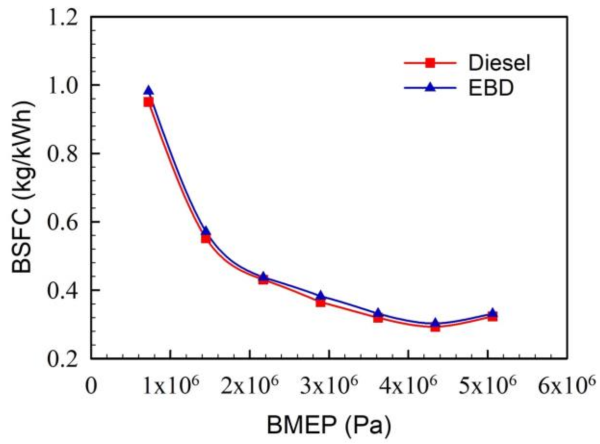

Fig. 2. Effect of load on the brake specific fuel consumption.

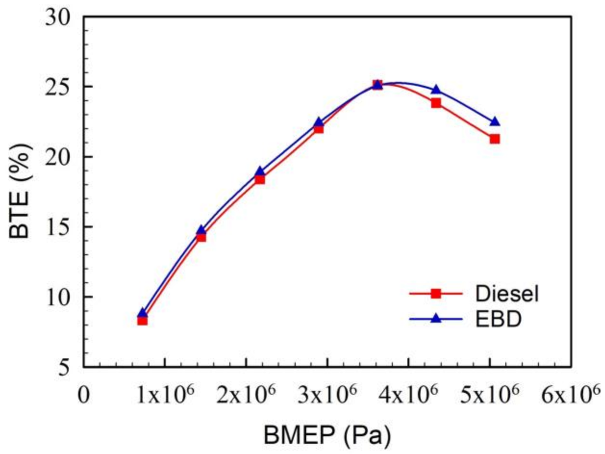

Fig. 3. Effect of load on brake thermal efficiency.

The temperature of gas exiting the cylinder, termed as exhaust gas temperature (EGT), signifies the condition of combustion and the state of ignition. For the present study, the EGT was measured for all the conditions assumed in the experiment. The variation of EGT with the change in the loading condition for both ED and diesel fuel is represented in Fig. 4. Observations revealed that with an increase in the load on the engine, the EGT increases for both the fuels, whether it is diesel or EBD. For a fixed period of time, the heat generation increases due to an increase in the load that leads to an increase in the EGT. Apparently, the EGT for the case of EBD was much lower as compared to EGT for the case of diesel in all loading conditions. The probable cause for this is an improved atomisation of the fuel during the combustion due to the comparatively low viscosity of EBD fuel.

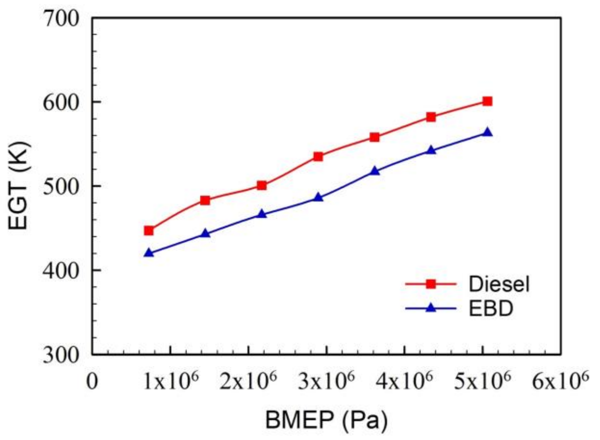

Fig. 4. Effect of load on the exhaust gas temperature.

In Figs. 5–7, the emissions of various exhaust gases at different loading conditions are depicted. The data collected on nitrogen oxides (NO_x) emissions is plotted in Fig. 5. It can be observed from Fig. 5 that the level of emission of NO_x was lower for EBD as compared to diesel. However, at full load conditions, NO_x for EBD was found to be higher than diesel. The emissions of NO_x were observed to rise progressively as the engine load increased across all tested fuel blends. This trend can be attributed to the higher quantity of fuel injected and burned within the engine cylinder under the elevated loads. The increased combustion activity led to a significant rise in cylinder gas temperatures, which, in turn, accelerated the formation of NO_x within the engine. Consequently, the higher temperatures resulted in the elevated NO_x emissions being released from the engine. These findings underscore the critical role of combustion temperature as a key factor influencing NO_x emissions. As the combustion process intensifies with greater fuel input and load, the elevated thermal conditions create an optimal environment for NO_x formation, highlighting the direct correlation between engine load, combustion temperature and NO_x output.

Figure 6 illustrates the carbon monoxide (CO) emissions produced by the engine under various load conditions. At lower engine loads, the CO emissions were notably higher when the engine was operated with ethanol-diesel blends compared to pure diesel fuel. This increase in CO emissions was more pronounced with higher ethanol content in the blends. However, as the engine load increased beyond half of its capacity, a reversal in this trend was observed. Under these higher load conditions, the CO emissions from the ethanol-diesel blends were consistently lower than those from pure diesel. The most significant reductions in CO emissions occurred at the maximum engine load tested, highlighting the potential benefits of ethanol-diesel blends in reducing CO emissions under high-load operating conditions. This behaviour can be attributed to improved combustion efficiency at elevated loads, which offsets the incomplete combustion characteristics observed at lower loads.

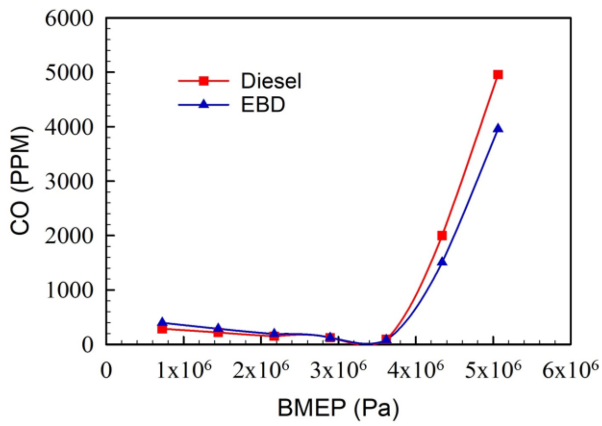

Fig. 6. Variation of CO emissions with load.

The EBD blend shows a decrease in hydrocarbon (HC) emissions compared to mineral diesel (Fig. 7). This reduction is likely due to the improved combustion of biodiesel blends, attributed to the presence of oxygen. In fact, HC emissions further decreased as the engine load increased.

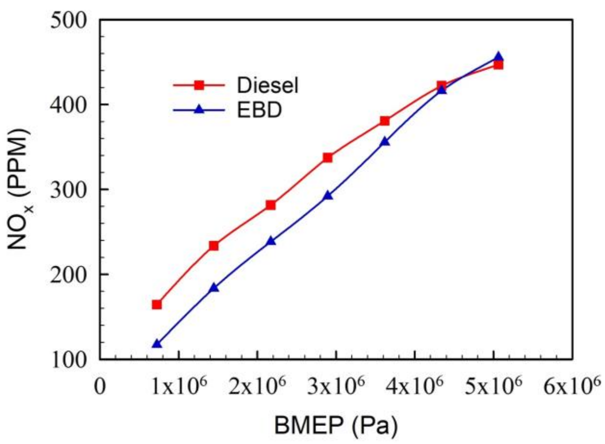

Fig. 5. Variation of NO_x emissions with load.

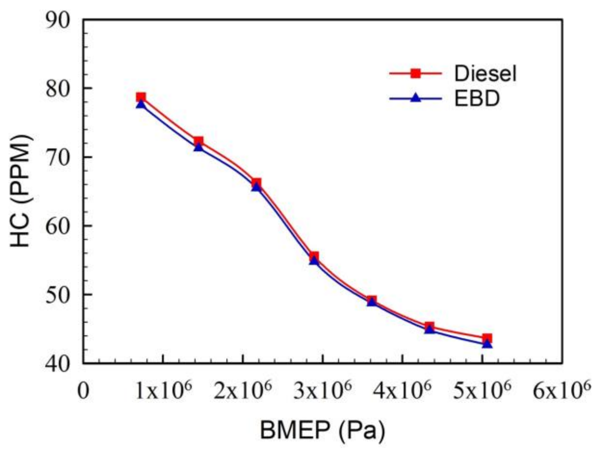

Fig. 7. Variation of unburnt hydrocarbon emissions with load.

4. Conclusions

An experimental study was conducted to examine the stability of ethanol-diesel blends and assess the engine performance parameters and emissions characteristics of these blends when combined with jatropha methyl ester (JME) as a co-solvent. The tested fuel mixture consisted of 10% ethanol, 10% JME, and 80% diesel fuel by volume. The engine was operated using this blend under varying injection pressures, ranging from 190 to 220 MPa, and different injection timings, at various load conditions. Key findings from the investigation are as follows:

1. Ethanol could not form a stable blend with diesel without the inclusion of an additive such as JME. A minimum of 4% JME (by volume) in an 80% diesel fuel mixture was necessary to achieve stability, but the blend remained stable for only one day. When the additive percentage was below 4%, blending was unsuccessful. The E10B10D80 blend remained stable only when the temperature did not fall below 10°C.
2. JME proved to be an effective additive for facilitating ethanol-diesel blending, despite being more expensive than diesel during the study period. Over the long term, as fossil fuel reserves diminish, biofuels like JME are expected to become economically viable alternatives.
3. Engines fuelled by the blend showed higher fuel consumption compared to pure diesel due to the lower calorific value of ethanol.
4. Thermal efficiency improved at lower loads when using the blend and diesel, but decreased near full load.
5. Carbon monoxide (CO) emissions varied with engine load. Above half load, CO emissions from the blend were lower than those from pure diesel, while below half load, CO emissions were higher.
6. Unburned hydrocarbon (HC) emissions were consistently higher when using the EBD blend as compared to diesel.
7. Nitrogen oxide (NO_x) emissions decreased at low engine loads, but increased near full load. However, NO_x emissions did not follow a consistent trend across different operating conditions.

This study underscores the potential of JME as a co-solvent for ethanol-diesel blends, while highlighting the trade-offs in performance and emissions associated with its use.

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