STUDY ON THE ENGINE CHARACTERISTICS (CI) OF USING PUMPKIN-MAIZE-BLENDED BIODIESEL MIXED WITH ADDITIVE (DIETHYL ETHER)

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ABSTRACT. This paper focused on the engine characteristics of a diesel engine fuelled using ternary blends. Initially, pumpkin and maize biodiesel were mixed in a volume ratio of 50:50. With a constant 0.5% diethyl ether (DEE) content, the binary combination of pumpkin and maize biodiesel was mixed with diesel at proportions of 10:90, 20:80, 30:70, 40:60, and 50:50 by volume. The prepared ternary mixtures were evaluated at varying engine loads to improve engine performance. Compared to diesel, the tested ternary blends had a reduced brake thermal efficiency (BTE). However, up to a 30% blending ratio, the BTE demonstrated by ternary blends was within the range of less than 0.5% concerning diesel fuel. The ternary blends' BSFC declined as the binary biodiesel mix increased. Diesel has a brake specific fuel consumption (BSFC) of 1.4%, 2.2%, and 3.4% lower than the ternary blends of 10%, 20%, and 30%. The decrease in the heat release rate of the ternary mixes meant that emitted less CO and NO_x than diesel. In contrast, ternary blends exhibited an increasing trend in smoke and HC emissions because of the rise in incomplete combustion that occurs as biodiesel content rises. Therefore, with appropriate engine modifications, the pumpkin and maize binary biodiesel blend can replace diesel by up to 30%.

KEY WORDS: Biodiesels, Binary blend, DEE, Ternary blend, Efficiency

INTRODUCTION

Diesel is widely used as fuel worldwide, intended to power the compression ignition engines used for carrying goods across the globe, generating electricity, farming, heavy earthworks, military vehicles, etc. Diesel fuel is used to drive the worldwide economy by playing a crucial part in bolstering the global economy and living standards [1]. The transport segment is mainly powered by diesel fuel, particularly in passenger vehicles and light- and heavy-duty trucks; these vehicles are widely used in urban mobility and freight transportation due to their high efficiency and load-carrying capacity. In order to meet the demands of the expanding population and industrial activities, there are more vehicles on the road. As a result, the amount of diesel fuel consumed each year rises. For example, diesel consumption in India in June 2022 was around 7.83 million tonnes, up 23.9% over last year [2]. Among these, around 70% is used by the transport sector. Also, rising fuel consumption might increase crude oil imports for countries that outsource a large portion of their total oil demand, particularly India. It currently outsources over three-fourths of its overall oil demand [3].

The biodiesel is a possible replacement for fossil fuels or diesel produced from petroleum that may also help reduce greenhouse gas emissions and carbon footprint. Biodiesel is a biodegradable and environmentally friendly product that may be made from sustainable and renewable feedstocks, such as plant-derived oils. Due to its ability to reduce direct and indirect greenhouse gas emissions, such as CO_2, CO, SO_2, and HC, biodiesel is considered a clean energy source [4]. Diesel engines may now run-on biodiesel, which is made from methyl esters of fatty acids derived from fats and seed oils [5]. Due to its clean combustion behavior, renewability, and biodegradability, biodiesel appears to be the most environmentally friendly biofuel [6].

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Additionally, excessive fuel usage has led to a rapid depletion of fossil fuel reserves. As they are non-renewable, this will lead to a fuel crisis in the future. On the other hand, carbon and other toxic emissions associated with dual-fuel vehicles are significant concerns as they cause adverse environmental effects; even excessive extraction of fossil fuel sources puts more pressure on the environment due to carbon emissions [7]. Therefore, environmentally favourable alternatives to diesel fuel are required to reduce the reliance on fossil fuels and their detrimental emissions.

Vegetable oil-based biodiesel, especially inedible oils, is viewed as a potential substitute for regular petroleum products, particularly diesel fuel. The major advantages of biodiesels are eco-friendly and renewable. Most importantly, their sources can act as a carbon sink, reducing the net CO_2 emissions and making them carbon-neutral fuels [8]. Several investigations have explored the viability of producing biodiesel from various crops and their feasibility as a replacement for diesel fuel [9]. Globally, over 350 crops have been identified as feedstock for biodiesel synthesis [10]. Availability, manufacturing procedures, and cost hugely influence the selection of oils for biodiesel production.

Many engine studies have already been carried out on these biodiesels individually as a substitute for diesel fuel [11]. Pumpkin biodiesel was produced from the raw oil extracted from its seed, blended with diesel in four different blending volume ratios (25:75, 50:50, 75:25, and 100:0), and tested with uniform speed. Adding 25% biodiesel to the mixture produced a BTE equivalent to diesel and slightly lower engine emissions. Wang *et al.* [12] explored the characteristics of a low-speed diesel engine using a 25% pumpkin biodiesel blend under different compression ratios (CR), injection pressure and timing (IP, IT). The blend yielded had a maximum performance with the least HC, CO, and smoke emissions at an operating CR, IP, and IT of 19.5, 250 bars, and 25°b TDC, respectively. Yuvaraja *et al.* [13], reported the effect of 10% and 20% pumpkin biodiesel blends at various compression ratios. The 20% blend exhibited maximum performance at 18 compared to diesel at standard CR (17.5). Similar kinds of engine studies were carried out for maize biodiesel, too. Manigandan *et al.* [14], carried out probe performance as well as emissions characteristics of two ternary blends formed by blending 25% corn biodiesel-diesel blend with pentanol (10% and 20%) and titanium oxide (5%) as oxygenated additives. The biodiesel blends with 20% pentanol improved the BSFC and brake power by 6.3% and 22%, and it reduced the NOx emission by 16% compared to diesel during a peak load operation. Sathyamurthy *et al.* [15], reported that fuelling diesel engines with a corn biodiesel-diesel blend resulted in a BTE that was lesser than diesel. Compared to diesel, the BSFC of 10%, 20%, and 30% blends increased by 2%, 4%, and 6%, respectively. Further, NO_x was higher, whereas CO and HC emissions were reduced with marginal changes in CO_2 emission

The test results show that pumpkin and maize biodiesels were tested individually as an alternative to diesel in a diesel engine. Both biodiesels exhibited analogous BTE as of diesel up to 20% blend ratio. However, they required modification of suitable engine parameters to obtain better performance than diesel. On the other hand, an engine study using these biodiesels combined as a binary blend has yet to be reported. From the physical properties of these biodiesels, it is noticed that one is found to be superior to the other in terms of specific properties. Therefore, combining these two biodiesels might result in a blend that performs better with higher blending ratios than observed when used separately. The engine performance of a diesel engine operating on ternary mixtures formed by blending 10-50% of a pumpkin-corn (50:50) binary biodiesel blend with diesel. In order to improve its performance further, DEE was added in 0.5% as an oxygenated additive to the prepared ternary biodiesel blends.

Fuel preparation

Diesel, diethyl ether, unprocessed pumpkin seed oil, and corn oil were bought commercially for this study. Initially, both crude oils were converted to biodiesel via transesterification. This procedure is important to decrease the viscosity of raw oil by converting its fatty acids into methyl esters. This process involves a chemical reaction between raw oil and methanol and base catalysts

(KOH/NaOH) when the temperature is around 60 °C for a certain period [16]. The specifics of the transesterification procedure used to convert unprocessed pumpkin seed oil and maize oil to corresponding biodiesel are given [15, 16]. The same procedure was followed in this study. Then, the extracted biodiesels from pumpkin seed oil and maize oil were blended in a ratio of 50:50 by volume. Then, this binary biodiesel blend was blended with pure diesel in five different volume ratios along with 0.5% diethyl ether (DEE) by volume, as shown in Table 1. The designations for the respective test blends are also tabulated in Table 1. The stability of the formed ternary mixtures was determined by storing them at ambient temperature for 14 days. There was no transition among the hydrocarbons in the blends, and all blends exhibited excellent blend stability.

Table 1. Fuel blend composition with its designation.

Fuel blend composition	Designation
100% Diesel	Diesel
50% Pumpkin + 50% Maize biodiesel	PB50MB50
89.5% Diesel + 10% PMB50 + 0.5% DEE	PMB10
79.5% Diesel + 20% PMB50 + 0.5% DEE	PMB20
69.5% Diesel + 30% PMB50 + 0.5% DEE	PMB30
59.5% Diesel + 40% PMB50 + 0.5% DEE	PMB40
49.5% Diesel + 50% PMB50 + 0.5% DEE	PMB50

The fuel features of the diesel, pumpkin, and maize biodiesel and blends are measured as per ASTM standards [17]. Adding Diethyl Ether (DEE) to a ternary blend can enhance ignition characteristics; improve combustion efficiency, and lower emissions such as carbon monoxide and unburned hydrocarbons. The oxygen content of DEE contributes to cleaner combustion, potentially reducing soot formation and promoting a faster and more controlled flame speed. However, challenges such as low energy density and storage issues need consideration.

Engine setup and its operating procedure

For the test, a Kirloskar TV1-diesel engine with one cylinder was used. The engine specifications are cylinder volume 661-CC, compression ratio 17:5:1, rated power 5.2 kW @1500 rpm and dyno arm length 185 mm. Fuel was injected using a mechanical injector at the standard IT and IP of 21° b TDC and 220 bars, respectively. An eddy current dynamometer, coupled to the engine via a crankshaft, was used to load the engine. The dynamometer was loaded electrically using a load control mechanism. Then, a load cell (S-type) connected to the dynamometer assessed the pressure exerted on the engine. An electronic converter takes the load cell's electrical signal into a numerical value. Installing a burette apparatus to determine the engine's fuel consumption in each operating state. A pressure sensor and an encoder were attached to the engine head and crankshaft to measure the cylinder pressure and crank angle. The data acquisition system comprised of an amplifier and dynamometer was provided to acquire the electrical signals from the encoder and transducer; then, the signal was converted to a digital value and stored on a desktop installed with AVL Indicom software. The pressure data was accessed and analysed using this software for further study. Nitrogen oxides, hydrocarbons, and carbon monoxide were measured using an AVL-444 gas analyser. The smoke's visibility was measured with an AVL 437C opacimeter.

The engine's performance fuelled by the various test fuels was evaluated at five distinct engine loads ranging from 20% to 100% by 20% increments. Regardless of load, the engine speed was maintained at 1500 rpm. The 100% load represents that the engine was operated at its maximum capacity; the brake power corresponding to this load is 5.2 kW. The engine was run idle for 15 min before the measurement to warm it up. Fuel usage, in-CP, and exhaust gas components were measured at each operating condition. The fuel consumed was calculated by timing how long it

took to fill the burette with ten cc. The cylinder pressure was recorded over 200 successive engine cycles. The pressure measurements over 200 cycles were averaged, and then, using the first law of thermodynamics, the program automatically estimated the rate of heat emission based on the mean pressure data [18]. Finally, the exhaust products HC (ppm), CO (% vol), NO (ppm), and smoke opacity (%) were measured for all the test fuels at each operating condition. Before each emission measurement, the analyzer instrument of the gas analyzer was entirely purged to remove any residual gases from the previous measurement. In order to minimize experimental error, each measurement was made three times, and the average was calculated. The uncertainty caused by those errors in the measured and computed parameters was determined and given as error bars in the results graphs from Figures 1-9. The formula for computing uncertainty was taken from [19].

RESULTS AND DISCUSSION

Diesel and ternary biodiesel mixes were tested at five different loads, and their performance, combustion properties, and emissions were analysed.

Performance characteristics

This section compares the various biodiesel blends and pure diesel regarding the variance in brake thermal efficiency and brake-specific fuel consumption that occurs as a function of engine load.

Brake thermal efficiency (BTE)

Based on engine load, Figure 1 depicts the disparity in BTE between diesel and ternary biodiesel mixtures. The BTE of the engine increases as the engine load increases for all test fuels. This happened due to increased fuel injection quantity and enhanced in-cylinder operating conditions, which resulted in enhanced combustion and thus improved the BTE. Across the evaluated engine loads, biodiesel blends had lower BTE values than diesel, indicating they were less efficient. Diesel, PMB10, PMB20, PMB30, PMB40, and PMB50 all had respective BTEs of 28.87%, 28.76%, 28.6%, 28.44%, 27.65%, and 27.14% at full load. Based on the findings, the blends with higher amounts of biodiesel had relatively lesser BTE. The main factor behind this trend is the blends' calorific value decreasing due to the addition of biodiesel, which has a calorific value that is generally lower than diesel. This finding is consistent with that of Ashok *et al.* [20], reported that the decrease in calorific value is the main reason for the decrease in BTE when powered by biodiesel blended diesel.

Furthermore, it is identified that the drop in BTE of the mixtures with a biodiesel ratio of up to 30% from that of diesel is within the 0.5% limit. This marginal decrease is that the variation in lower calorific value between the diesel and blends is small. Additionally, the inherent oxygen quantity in the biodiesel and the DEE existing in the mix assist in improving combustion, blend contributes to combustion enhancement and thus resulting in a BTE comparable to that of diesel. However, if the biodiesel mix percentage goes up to 40% and 50%, there is a significant decrease in the BTE. This is because blends that contain higher biodiesel percentages have a greater decrease in calorific value. In addition to this, the spray combustion stage is also heavily influenced by physical qualities like viscosity and density. Due to the fuel's high density and viscosity, which interfere with the spray pattern and evaporation process, heterogeneous mixtures occur, which causes incomplete combustion.

Because biodiesel mixes have a higher density and viscosity than diesel, the above effect comes into play and reduces the engine BTE when powered by the biodiesel blends. The above effect starts to dominate more when the blend proportion is increased beyond a certain percentage. PMB40 and PMB50 mixes have substantially lower BTE than other blends. This BTE finding is consistent with previous findings [3], who claimed that an increase in density and viscosity of the

fuel negatively affects the fuel injection as well as evaporation, resulting in an improper mixture formation and subsequently poor combustion which in turn reduced the BTE.

A combination of factors influences the brake thermal efficiency of ternary fuel blends in internal combustion engines. These include the energy content, octane or cetane numbers, and chemical composition of the fuel components, as well as the ratio of their proportions. According to the results, the blends with more significant percentages of biodiesel had comparatively lower BTE. The primary cause of this tendency is the inclusion of biodiesel, which often has a lower calorific value than diesel, which causes the blends' calorific values to decrease.

Hegde *et al.* [21], examined the transesterified CI and the biodiesel's physicochemical characteristics. An addition called SC5D was added to the blends. The blends that were developed and used for engine studies were B10-B40. Using additives resulted in a decrease in fuel usage and an increase in BTE. Because of the mixture's more extensive oxygen content and improved heating value, biodiesel burns more quickly following the premixed combustion phase, improving efficiency [22]. The rise in peak pressure and combustion temperature is also accountable for the BTE increase (Chen *et al.* [23]. BTE somewhat declines with increasing blend % [24]. Combining diesel and pumpkin -maize and adding DEE additive decreased the blend's density and cetane number value.

Additives are essential for enhancing the characteristics and functionality of biodiesel. Cold flow additives improve low-temperature operability; antioxidants prolong fuel shelf life by improving oxidation stability. Lubricity additives reduce friction and wear in fuel system components. Stabilizers ensure fuel stability during storage and transportation. Additives make biodiesel a more reliable and efficient alternative fuel by addressing these challenges.
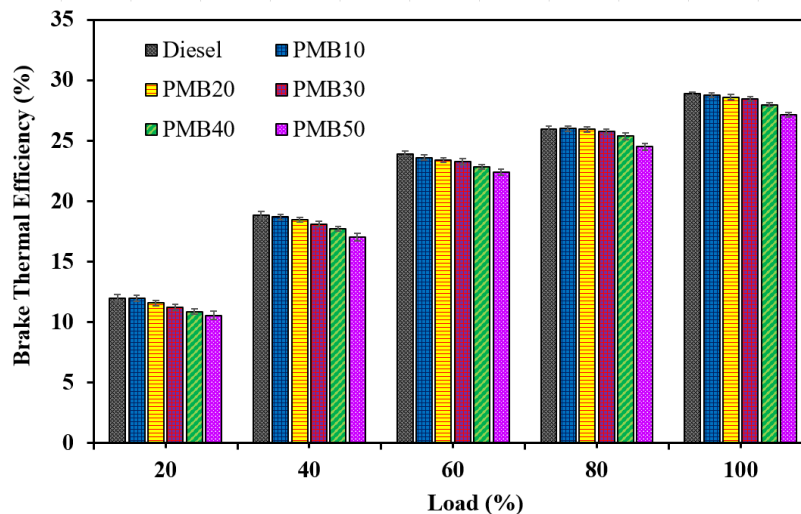

Figure 1. Engine BTE of ternary biodiesel blends and diesel at various engine loads.

Brake specific fuel consumption (BSFC)

Figure 2 compares the variation in BSFC for ternary blends of various biodiesel ratios at various engine loads to that of single diesel. All test fuels used in the experiments showed a similar BSFC trend concerning engine load; as the load increases, the BSFC decreases. This is because, as engine load rises, cylinder temperature increases, and heat loss decreases. This leads to combustion enhancement and a more significant proportion of fuel power converted into a

productive mechanical effort. At all engine loads, the ternary blends' BSFC increases as the biodiesel concentration in the blend increases. This increasing BSFC trend is due to biodiesel and DEE in ternary blends, which have a lower calorific value than pure diesel [25].

The BSFC given by diesel, PMB10, PMB20, PMB30, PMB40, and PMB50 at full load is 0.296, 0.3, 0.303, 0.306, 0.313, and 0.325 kg/kWh, respectively. Figure 2 further shows that the increase in BSFC for larger loads for ternary mixes is minor, up to 30%. This is due to the lower calorific value for ternary blends, which are comparatively lesser, up to a 30% blend ratio. Furthermore, intrinsic oxygen in the biodiesel and DEE also significantly affects the BSFC. The inherent oxygen enhances combustion efficiency by inducing a locally leaner air-fuel mixture ratio [26]. Consequently, combustion is improved.
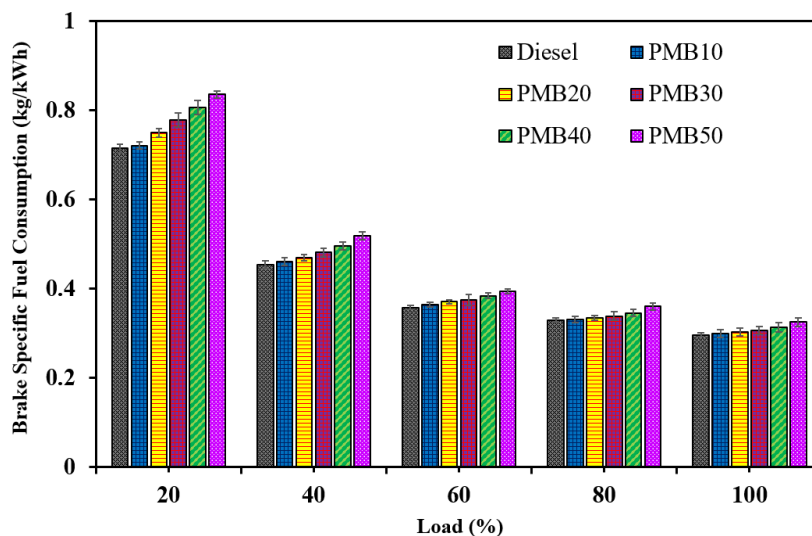

Figure 2. BSFC for diesel and ternary blends.

Emission characteristics

Diesel fuel and ternary biodiesel blends exhaust emission loads, including carbon monoxide (CO), hydrocarbons (HC), and nitrogen oxides (NO_x), are discussed in this section.

Carbon monoxide emission. Figure 3 depicts the CO emission change for ternary biodiesel mixes across various engine loads. The CO emission given by diesel fuel is compared with them. Suppressed oxidation of CO generated during combustion leads to increased CO in the emissions, primarily due to low temperatures and incomplete combustion resulting from a rich air-fuel ratio combination [27]. Observations indicate that CO emissions grew progressively with increasing engine load, reaching their peak at maximum load circumstances. This is because the production of regionally fuel-rich zones slows the transformation of CO into CO_2 as the load increases due to the shorter time available for effective mixture formation. The blending of biodiesel and DEE dropped the CO emission drastically. The CO emission reduces as the amount of biodiesel in the mixture increases. Ternary blends have lower CO emissions than diesel. The CO emission of PMB10, PMB20, PMB30, PMB40, and PMB50 blends is lesser by about 30%, 44.4%, 40%, 38.4%, and 52.6% when compared to diesel, respectively.
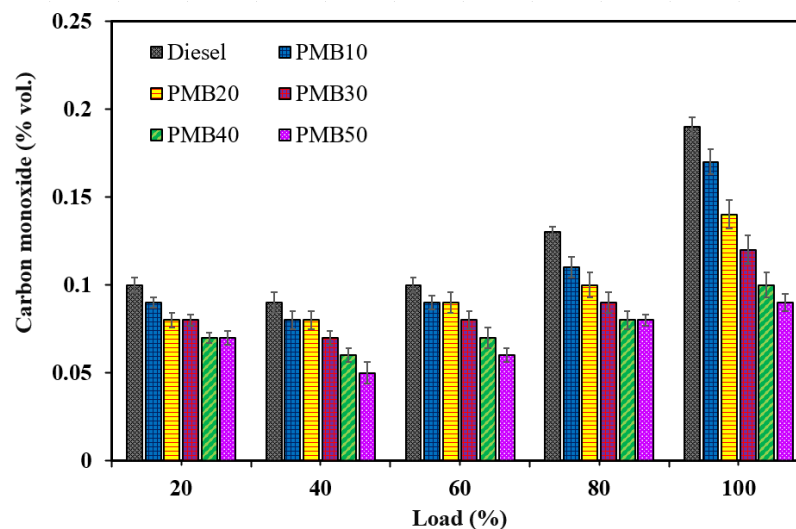
Figure 3. CO emission of ternary biodiesel blends.

The presence of innate oxygen in the pumpkin-maize biodiesel blend and the DEE, as well as their higher cetane number, is the reason behind the decreasing CO emission. The oxygen leans out of the locally rich fuel zones, whereas a higher cetane number starts the combustion early, giving more time for combustion [28]. Both these actions speed up the combustion process and promote faster oxidation of CO, thus resulting in complete combustion. This reduces the CO levels in the exhaust gases of the ternary blends. This finding agrees with the pumpkin biodiesel-diesel and maize biodiesel-diesel mixtures tested in a CI engine reported [11, 15].

Hydrocarbon emission. Figure 4 depicts the emission of HC for ternary biodiesel mixtures and diesel at different engine loads. The HC emission is the unburnt /partially burnt fuel products formed by the development of locally rich/lean mixtures, flame and wall quenching, and the accumulation of unburned fuel particles in crevice volumes the HC emission for ternary blends and diesel increases gradually with engine load [29]. The HC emission at 20% load for diesel is 32 ppm, which is increased to 69 ppm at 100% load. Due to the increased fuel injection quantity, there is less time for all the fuel to vaporise, combine, and burn, resulting in incomplete combustion [30]. As the binary biodiesel blend percentage increases from 10% to 50%, the HC emission increases irrespective of engine loads. At maximum load, the ternary blends PMB10, PMB20, PMB30, PMB40, and PMB50 showed an HC emission of 73, 79, 86, 101, and 119 ppm, respectively, which are 5.80%, 14.49%, 24.64%, 46.38%, and 72.46% higher than pure diesel. In general, the oxygen in the fuel compensates for fuel-rich regions and improves combustion, which tends to reduce HC emissions [31]. However, the oxygen in biodiesel and DEE does not diminish HC emissions in ternary blends. This is because the considerably greater density, viscosity, cetane number, and boiling temperature of ternary blends have a key influence on rising HC emissions. Fuel with high density and viscosity experiences a restriction in the flow and forms droplets with greater mass and size [32]. The bigger fuel droplets take more time to evaporate, which affects the mixing process and thereby results in an improper mixture. This action gets aggravated even further due to the slow evaporation characteristics of the ternary blends caused by the increased boiling point of the binary biodiesel blend.

Furthermore, a higher cetane number tends to cause the premature start of combustion, again reducing the time available for the mixing process, and resulting in improper mixture formation [33]. Figures 7 and 9 display the pressure curve and delay in ignition findings, respectively, showing that greater biodiesel ratio ternary mixes result in an earlier combustion start. As the incorrect mixture slows down the combustion process, partial combustion occurs, and the resulting HC emission is greater for the ternary mixes.
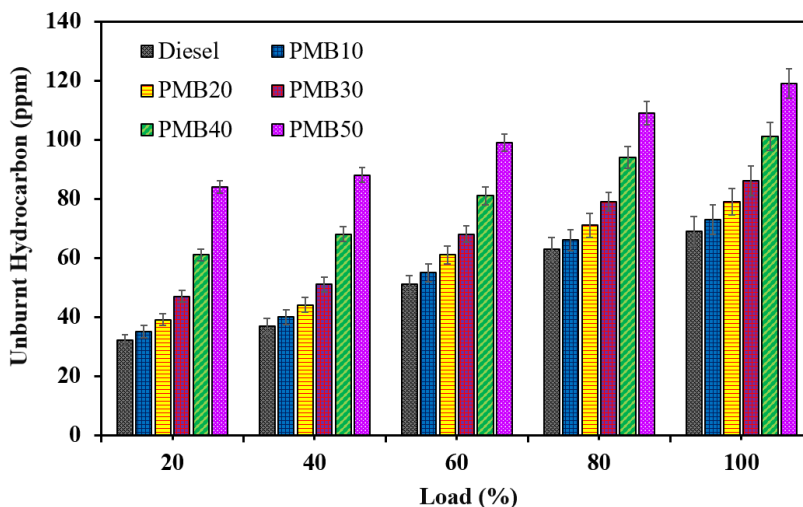

Figure 4. HC emissions of diesel and ternary mixes.

The high viscosity and density of biodiesel cause atomization issues, which result in greater emissions of smoke, HC, and CO. The higher oxygen content of biodiesel leads to more complete combustion of the fuel, but also increases the formation of NO_x and reduces the formation of CO and HC. However, when the biodiesel ratio is too high, the excess oxygen may react with the nitrogen in the air and form more NO_x, while the unburned fuel may escape as HC and smoke. The lower cetane number of biodiesel reduces the ignition quality of the fuel, which causes longer ignition delay and higher peak pressure. This may increase the combustion temperature and NOx emissions, as well as the incomplete combustion of the fuel and the resulting HC and smoke emissions. The presence of pentanol as an oxygenated additive in the ternary blends may have a positive effect on reducing the emissions of CO, HC, and smoke, but also a negative effect on increasing the emissions of NOx and CO_2. This is because pentanol has a higher oxygen content, lower cetane number, and lower boiling point than biodiesel.

NOx emission. NO_x emission of ternary blends and diesel at different engine loads is compared in Figure 5. The flame front and post-combustion gases are where a major amount of NO_x formation occurs. The NO_x formation is controlled by the oxygen content and combustion gas temperature [34]. As the load is increased from 0% to 100%, it is seen that NO_x emission rises. This is because when load increases, in-cylinder operating temperature increases and a greater proportion of the fuel is burned. As a result, the combustion temperature increases, which in turn increases the NOx formation [35]. The NO_x emission of diesel is increased from 363 ppm at 20% load to 1143 ppm at full load. The ternary blends showed a decreasing NOx emission trend with increasing binary biodiesel blend content. At 60% load, the NOx emission of diesel, PMB10, PMB20, PMB30, PMB40, and PMB50 is 893, 865, 801, 746, 712, and 636 ppm, respectively. Despite excess

oxygen inherited by the biodiesels and DEE in the ternary blends, their NOx emission is slighter than diesel. The primary reason for this action is the drop in combustion temperature, as evidenced by the decrease in in-cylinder pressure and HRR in Figures 7 and 8, respectively. The same finding has been reported by Nanthagopal *et al.* [36]. The lower heating value and incomplete combustion inherited by the ternary blends are the main reasons for the decrease in heat release rate. In addition, the in-cylinder gas temperature has been lowered by the cooling effect created by the increased latent heat of vaporisation (LHV) of DEE, which consequently reduced the NO_x emission.

Because of its high latent heat of vaporization (LHV), this lowers the combustion temperature and leads to a reduction of NOx emission [37, 38]. A similar finding was reported by Jeevanantham *et al.* [39], stating that NO_x emission decreased because of the cooling effect induced by the high LHV characteristic of the DEE on the combustion gas temperature.
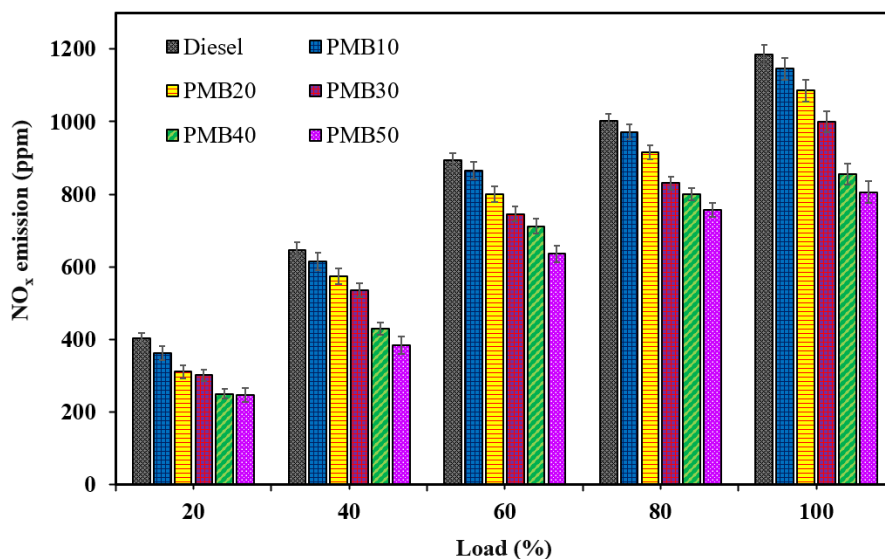

Figure 5. NOx emission of ternary biodiesel blends.

Smoke opacity. The presence of smoke is a reliable indicator of incomplete combustion in the engine cylinder. Poor fuel atomization and evaporation characteristics, rich mixture zones, engine load conditions, and accretion of surplus fuel in the cylinder are the most common causes of smoke emission [36, 40]. Smoke opacity under varying loads is seen in Figure 6 for both diesel and ternary biodiesel mixes. It is noticed that smoke emission rises with an increasing engine load. The smoke opacity is increased from 24.7% to 67.8% as the load applied on the engine is increased from 20% to 100%. This is due to an increase in fuel quantity with the load, which subsequently upsurges the quantity of fuel that is injected late in the fuel spray process, which burns directly in the combustion flame without undergoing a proper mixing process, resulting in an increased proportion of incomplete combustion and thus increasing the smoke. It is also noticed that blending biodiesel and DEE with diesel increased the smoke opacity. There is a correlation between the biodiesel ratio in the blend and the opacity of the resulting smoke for ternary blends. At maximum load, the smoke opacity of PMB10, PMB20, PMB30, PMB40, and PMB50 is 3.39%, 4.71%, 9.7%, 28.31%, and 33.33% higher than that of diesel, respectively. This is because the adverse effects caused by poorer atomization and evaporation characteristics of the binary

biodiesel blend present in the ternary blend are dominant over the positive effect induced by its innate oxygen content on the smoke emission [41, 42]. Due to biodiesel addition, the rise in viscosity, density, and boiling point of ternary blends are the primary causes of their poor atomization and evaporation characteristics [43]. In addition to this, a high cetane number could increase smoke emissions. This is because fuels with high cetane numbers have comparatively lesser ignition delay, thereby starting the combustion earlier [44]. This action, in turn, decreases the time available for the air-fuel mixing process and burns more amount of fuel in the diffusion phase; some fuel in the locally rich zone might undergo pyrolysis due to lack of time for evaporation, mixing and oxidation, resulting in incomplete combustion manifested as dark smoke [45]. The ternary blends presented comparatively higher smoke emissions because the cetane number of all ternary blends, except the 10% blend, is slightly higher than diesel. The reduction in ignition delay for the ternary blends is also evident from the ignition delay graph shown in Figure 9. To overcome this effect on smoke emission, the injection timing can be slightly advanced for ternary blends, thereby giving more time for the formation of the mixture.

Furthermore, adding DEE contributes to smoke emission because of its high cetane number and LHV. This claim is in agreement with the study by Nanthagopal *et al.* [28], which reported that the smoke emission increased when the concentration of DEE in the blend was increased from 2.5% to 12.5% in the ternary blend containing biodiesel, diesel, and DEE. Additionally, unsaturated fat in biodiesel further aggravates smoke emissions when they burn in fuel-rich zones.
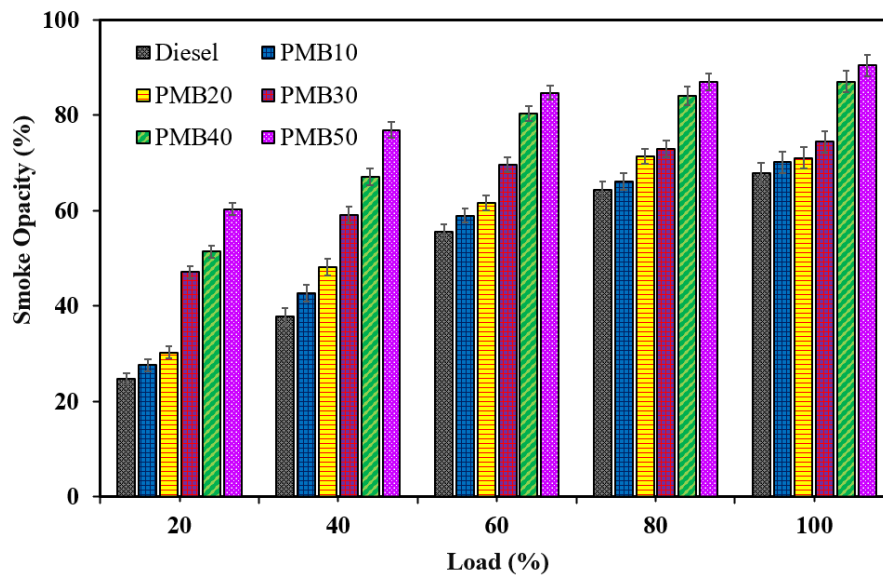

Figure 6 Smoke emission of ternary biodiesel blends.

At various loading situations, the engine's ability to emit smoke is significantly influenced by increased oxygen. It indicates a suitable amount of oxygen and a flawless combustion process. Consequently, the fact that the exhaust temperature is crucial for describing the smoke behaviours was coincidental. At a 30% blending ratio, it can be concluded that optimal opacity values were obtained [46].

Combustion characteristics

The parameters that give inferences about the combustion process for the test fuels are discussed in this section with proper technical reasons.

In-cylinder pressure. Figure 7 compares the in-cylinder pressure based on the crank position for diesel and the ternary blends at 100% load. The pressure trends of all the test fuels are very similar. Peak cylinder pressure is based on the quantity of fuel consumed during the premixed combustion phase, which is determined by the rate at which it is consumed. The burn rate is based on the type of fuel and air-fuel mixture formation. The peak pressure of ternary blends decreased with increasing biodiesel proportion in the blend. The test blend's lower calorific values than diesel mean less heat is released, leading to lower peak cylinder pressure. Additional factors contributing to lower peak pressures include the ternary mixes' higher viscosity and density than diesel, which hinders the air/fuel mixing process and leads to inappropriate mixture formation and inefficient combustion. So, it stands to reason those ternary mixes have a lower peak cylinder pressure than diesel fuel. Diesel, PMB10, PMB20, PMB30, PMB40, and PMB50 all have peak pressures of 72.59 bar, whereas PMB20, PMB30, PMB40, and PMB50 all have peak pressures of 68.95 bar, 66.97 bar, and 65.93 bar. The current study's findings are reliable, with similar results reported by Anwar *et al.* [47] and Krishnamoorthi *et al.* [48]. Figure 7 compares the in-cylinder pressure based on the crank position for diesel and the ternary blends at 100% load. The pressure trends of all the test fuels are very similar. Peak cylinder pressure is based on the quantity of fuel consumed during the premixed combustion phase, which is determined by the rate at which it is consumed. The burn rate is based on the type of fuel and air-fuel mixture formation. The peak pressure of ternary blends decreased with increasing biodiesel proportion in the blend. The test blend's lower calorific values than diesel mean less heat is released, leading to lower peak cylinder pressure. Additional factors contributing to lower peak pressures include the ternary mixes' higher viscosity and density than diesel, which hinders the air/fuel mixing process and leads to inappropriate mixture formation and inefficient combustion. So, it stands to reason that ternary mixes have a lower peak cylinder pressure than diesel fuel.
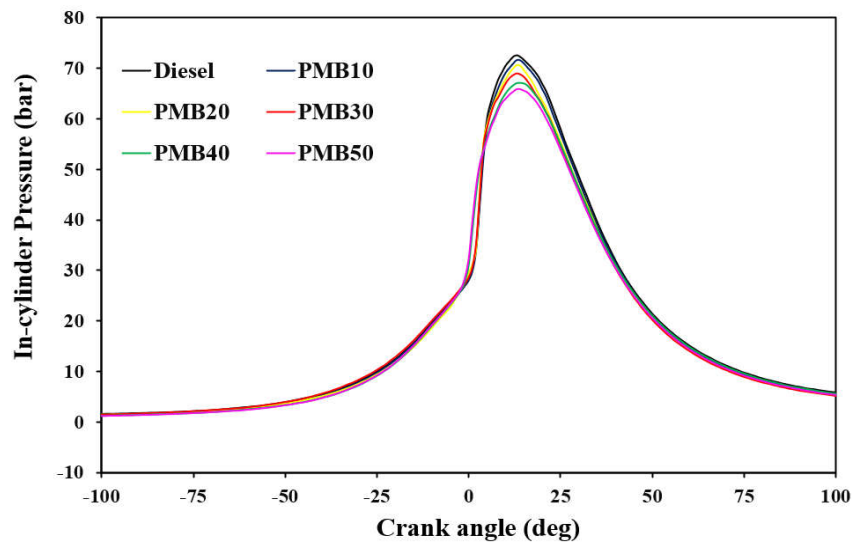

Figure 7. In-cylinder pressure for ternary biodiesel blends.

Diesel, PMB10, PMB20, PMB30, PMB40, and PMB50 all have peak pressures of 72.59 bar, whereas PMB20, PMB30, PMB40, and PMB50 all have peak pressures of 68.95 bar, 66.97 bar, and 65.93 bar. The current study's findings are reliable, with similar results reported [47, 48]. For PMB blends with a blend ratio of up to 30%, the crank angle corresponding to peak pressure is the same as diesel's; however, for PMB40 and PMB50, the crank angle of peak pressure is retarded by 1°CA despite the early beginning of ignition. This is because certain mixtures tend to burn slowly. The delayed combustion is mostly due to the production of the incorrect mixture, which is produced by the physiochemical features of the pumpkin-maize biodiesel blend.

Heat release rate (HRR). Figure 8 represents the HRR for the ternary blends and diesel at full load. The HRR was an indicator of the combustion events occurring at various phases; the initial peak characterizes the premixed combustion phase, whereas the second peak represents the controlled combustion phase [49]. The trend shown by ternary blends is that the HRR decreases with increasing the binary biodiesel blend ratio in the blend. It is perceived that the peak HRR of all ternary blends is lesser than diesel. The observed peak HRR for diesel, PMB10, PM20, PMB30, PMB40, and PMB50 is 63, 61, 60, 59, 55, and 53 J/°CA, respectively. The reason behind this descending HRR trend is a diminution in calorific value with decreasing diesel content in the blend. In addition, the shorter ignition delay observed with ternary blends due to relatively higher cetane number also lowers the HRR by limiting the premixed combustion phase.
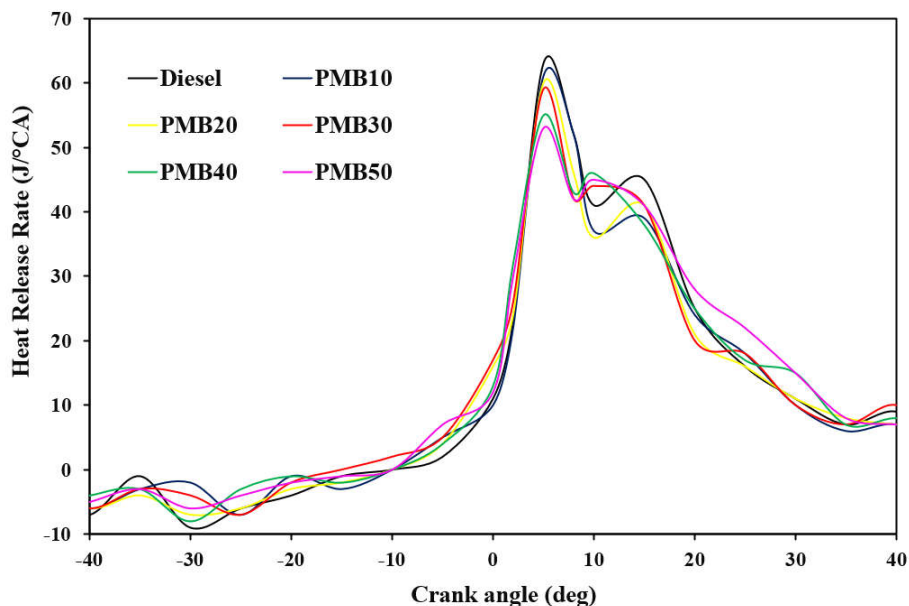

Figure 8. Heat release rate curves for ternary biodiesel blends.

Also, improper mixture formation associated with ternary mixtures due to their insufficient atomization and slow evaporation features, caused by the physical properties of biodiesels and their DEE content, reduces the combustion speed and lowers the HRR.

Ignition delay (ID). ID is the crank angle degree that distinguishes the beginnings of fuel injection from the beginning of combustion. The crank angle (CA) required reaching 5% cumulative heat release was determined using the heat release rate curve [50]. The fundamental factors responsible

for the delay period are the fuel's physiochemical properties and operating conditions [28]. Figure 9 shows the variation of ID for the tested fuels. The common trend observed with diesel and ternary blends is that the ID decreases with increasing engine load. This is because of increased cylinder temperature with the load. The ID of diesel at 20% load is 14.5°CA, which is decreased to 12.5°CA at maximum load. The ID for ternary blends PMB10, PMB20, PMB30, PMB40, and PMB50 at full load is observed to be 12.3, 11.7, 11.9, 11.5, and 11.6°CA, respectively. As observed, the ID of ternary blends is lesser than diesel because the former showed higher cetane number and oxygen content than the latter. All the test blends showed the same trend at all engine loads. Additionally, the higher relative volatility of the DEE in the blend promotes the development of the premixed gaseous mixture. It allows for an earlier start of combustion by decreasing the ignition delay for the ternary blends. These accords well with other research of a similar nature previously reported by Uslu and Aydın [51].
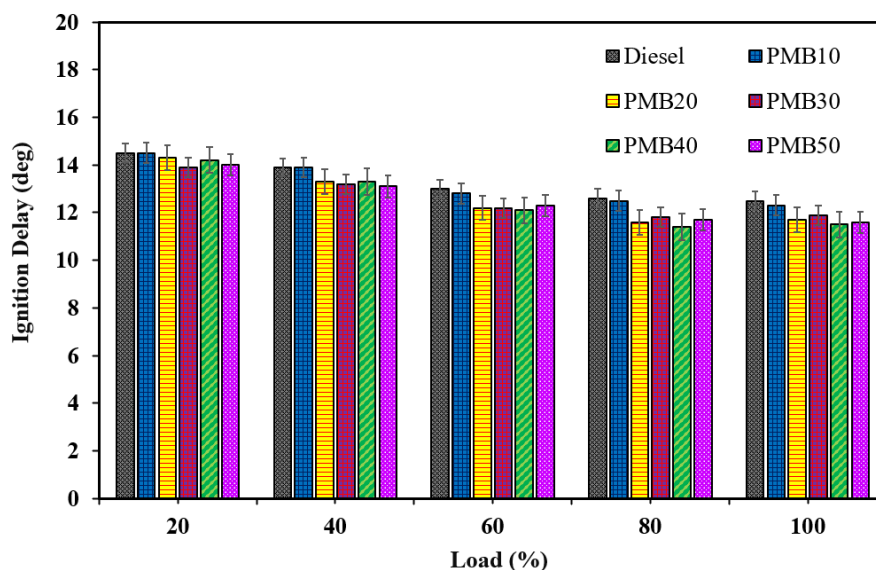

Figure 9. Ignition delay for ternary blends and diesel at tested load conditions.

CONCLUSION

This research focused as the effects of ternary blends made up of binary biodiesel blends of pumpkin-maize biodiesels, diesel and DEE on a single-cylinder CI engine's performance, emissions, and combustion attributes. The percentage of pumpkin-maize binary biodiesel blend in the ternary blend samples varied at 10%, 20%, 30%, 40%, and 50% by volume, while the DEE concentration was fixed at 0.5%. The outcomes of the investigations were analysed and compared to diesel fuel. The following are the results of this experimental study: (1) Tested ternary blends showed lower BTE than diesel. However, up to a 30% blending ratio, the BTE exhibited by ternary blends was within the range of less than 0.5% concerning diesel fuel. (2) With an increase in the binary biodiesel mix, the BSFC of the ternary blends decreased. The BSFC exhibited by 10%, 20%, and 30% ternary blend is 1%, 2.2%, and 3.4% higher than diesel. (3) Compared to diesel, the ternary mix's carbon monoxide and nitrogen oxide emissions decreased by 52.6% and 32.5%, respectively, at 50% full load. The CO and NO_x emissions lowered by around 36.8% and 15.5%, respectively, with the 30% blend. (4) The smoke and HC emissions were increased with

increasing binary biodiesel blend concentration in the blend. The HC emission was increased by 7.8%, 19.6%, and 33.3% for 10%, 20%, and 30% binary biodiesel blend blended ternary blends when compared to pure diesel, whereas, for the same blends at full load, the smoke opacity was 3.3%, 4.7%, and 5% higher than diesel, respectively. (5) All ternary combinations, even at full throttle, released less heat and had lower in-cylinder gas pressure than diesel. Until the blend ratio of 30%, the decreases in peak CP and HRR were slightly smaller. (6) A diesel-binary-biodiesel-DEE ternary blend with a binary biodiesel mix ratio of up to 30% may be a viable alternative biofuel for diesel engines. However, certain modifications are required on the engine parameters, such as injection duration, pilot injection strategies, combustion chamber shape, nozzle hole geometry, number of nozzles, and fuel preheating to achieve better performance. Optimizing these characteristics might maximize the efficacy of the investigated ternary blends as a diesel replacement. This study could be carried forward to optimize those parameters to achieve improved performance for those ternary blends.

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