An experimental assessment on the influence of high fuel injection pressure with ternary fuel (diesel–mahua methyl ester–pentanol) on performance, combustion, and emission characteristics of common rail direct injection diesel engine

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Abstract
Optimization of fuel injection strategies can maximize the utilization of ternary fuel by addressing the issues concerning fuel consumption, engine performance, and exhaust gas emission. In the midst of the pervasiveness of plant-based biofuel, this paper focused on maximizing the mahua oil biodiesel usage in a diesel engine having a common rail direct injection (CRDI) system without any engine modifications. The crude oil extracted from the seeds of Madhuca longifolia is known in India as mahua butter and has shown impressive fuel properties such as lower viscosity, flashpoint, boiling point, and comparable calorific value to diesel. 1-Pentanol, which has a chain of five carbons and can easily be blended with both diesel and biodiesel, is a promising type of alcohol for the future. In this study, the influence of fuel injection pressure with ternary fuel (diesel + mahua methyl ester + pentanol) on engine characteristics of CRDI diesel engine was analyzed. The fuel injection pressure is varied from 20 to 50 MPa so that ternary fuel can be properly utilized. The high injection pressure of 50 MPa has better combustion characteristics and higher brake thermal efficiency (4.39%) value than other injection pressure values. A better mixture is formed due to well-atomized spray, and as a result, the levels of CO (22.24%), HC (9.49%), and smoke (7.5%) fall with the increase in injection pressure. The usage of ternary fuel raised the NOₓ emission (12.46%) value and specific fuel consumption (SFC) with a decrease in the BTE (brake thermal efficiency) which attributes to its properties and combustion characteristics.

Keywords Ternary fuel · Fuel injection pressure · Trade-off · Performance · Combustion and emissions

Introduction
Due to the heavy usage of fuels around the world, fuel energy demand is increasing at a speedy rate. Also, fossil fuel resources are depleting and due to increased environmental concern, the development of alternate energy sources such as biodiesel is emphasized more than anticipated. The biodiesel’s history is as old as the history of the normal engine, and moreover, the use of soluble vegetable oil (SVO) was also thoroughly studied at the time when the diesel engine was invented. Rakopoulos et al. (2008) investigated the influence of alcohol with different concentrations (10% and 5%) along with diesel fuel. There was a significant improvement in the performance of the engine and the emission characteristics while using a 10% ethanol–diesel blend when compared with the 5% blend of the same. Selvan et al. (2009) studied about the effect of cerium oxide nano additive concentration in ethanol–biodiesel–diesel blends, and results revealed that there was an increase in fuel consumption whereas brake thermal efficiency reduced when compared to conventional diesel fuel. Barabas et al. (2010) investigated multicylinder CI with bio-ethanol/biodiesel/diesel and studied about the emissions and performance characteristics and reported that at part loads, the reduction in performance because of less heating value. Carbon monoxide and hydrocarbons emissions were reduced while using biodiesel–diesel–ethanol blend. Fang et al. (2013)
and Qi et al. (2011) also submitted alike reports regarding the usage of biodiesel–ethanol–pure diesel blend in diesel engines. Nitrogen oxide emissions were considerably decreased when the ternary blend was used in comparison with pure diesel for entire load conditions. Hulwan and Joshi (2011) studied about the practicality of ethanol in high concentrations in the diesel–ethanol blends and reported that the nitrogen oxide emissions were increased considerably. However, carbon monoxide emissions increased at smaller loads, and smoke opacity was reduced for higher loads. An improved model of ethanol–biodiesel–diesel fuel blend was formulated by Lin et al. (2013). They conducted experiments with different fuel blends which can improve the entire profitability of the system, given the cost of production, prices of fuels, and so on. Datta and Mandal (2014) studied about the suitability and ANPs at varying concentrations of 100, 75, 50, 25 mg/l with DBF (diestrol-blended fuel), 70% diesel + 20% ethanol + 10% jojoba biodiesel, on the utilization of engine characteristics and reported that at a dosage of 25 mg/l, the hydrocarbons and nitrogen oxide emission decreased in comparison with other nanoparticles additives. Sandalci et al. (2014) examined the impact of ethanol on diesel engines under two operating conditions, i.e., 30% and 15% diesel fuel levels (on the basis of volume). No separation of phase was observed, and the blend of ethanol and diesel was very stable throughout the operation. They also witnessed a considerable decrease in smoke and nitrogen oxide emissions in comparison with diesel fuel. Yilmaz et al. (2014a) analyzed the impact of ternary blend and reported that changing ethanol concentrations resulted in varying emission characteristics due to varying oxygen content and cooling effect of ethanol. They concluded that the ethanol concentration in the blend decreased the carbon monoxide and hydrocarbon emissions and increased the nitrogen oxide emissions. Li et al. (2014) investigated the influence of ternary blend in a diesel engine of 1600 rpm speed (constant) and reported that the addition of pentanol results in the betterment of air–fuel mixing rate, greater thermal efficiency, and lesser emissions. Shaafi and Velraj (2015) investigated about the impact of aluminum nanoparticles (ANP) at a dosage of 100 mg/l in both ternary blend of 1% surfactant + 4% ethanol + 15% soybean diesel + 80% diesel and binary blends B2 and reported that the heat release rate and cylinder pressure of ternary blend displayed higher value due to the high value of the surface area to volume ratio of alumina. Venu and Madhavan (2016) studied about the impact of ANP in ethanol–biodiesel–diesel ternary blend at different injection timings and found that the combustion of nanoparticles was effective in delayed injection timing and decreasing the hazardous tailpipe emissions such as nitrogen oxide, carbon monoxide, smoke, and hydrocarbons. Prabu and Anand (2016) investigated the influence of nano additives blended with jatropha in a diesel engine and reported a high brake thermal efficiency of 31% and a decreased value of brake-specific fuel consumption of 0.293 kg/kWh.

Venu and Madhavan (2017), through their studies, found that the addition of nanoparticles into ternary fuel blend is much better than the addition of oxygenated additives. Hosseini et al. (2017) studied about the impact of ANPs as catalysts in blends of 95% diesel and 5% waste cooking oil (WCO) biodiesel (B5 blend) and 90% diesel + 10% WCO biodiesel blend (B10 blend) in a diesel engine and reported that B10AL90 (90 ppm ANP in B10 blend) shows higher brake thermal efficiency, power, exhaust gas temperature (EGT), and torque of values of 10.63%, 5.36%, 5.8%, and 5.36%, respectively, when compared to conventional diesel fuel, while brake-specific fuel consumption decreased to 14.66%. Venu and Madhavan (2017) investigated the influence of biodiesel blends on CI engine characteristics which use the blend of diesel–ethanol–biodiesel as ternary fuel. From their study, it was clear that the addition of nano additives resulted in the betterment of engine performance in terms of various parameters. Chockalingam et al. (2017) investigated diesel engines by adding 10% ethanol to diesel, and it was observed that the performance was similar to that of diesel fuel. Nour et al. (2018) studied about the ANP at different concentrations of 100, 75, 50, and 25 mg/l mixed into a DEB (diestrol-blended 70% diesel + 20% ethanol + 10% jojoba diesel) fuel and their effect on various diesel engine characteristics. When a 25-mg/l dosage of ANP was used, there was a decrease in hydrocarbons and nitrogen oxides emissions when compared with other nanoparticles addition. Wu et al. (2018) investigated the influence of nanoparticle additives used as fuel catalyst in the blend of diesel–biodiesel to study the performance characteristics of the engine and reported that BSFC (brake-specific fuel consumption) is reduced by 6%, emissions of HC decreased by 14.5%, and CO reduced by 10% when compared with the biodiesel blend of 10%. Sivakumar et al. (2018) studied about the impact of ANP for a fuel blend of 75% diesel–25% Pongamia biodiesel inside a diesel engine(single cylinder) and reported that the B25 blend with 100 ppm alumina doping resulted in a reduction of brake-specific fuel consumption by 16.67% and increased brake thermal efficiency to 8.36% when compared to normal B25 blend without alumina doping, which has issues such as high evaporation rate, extended flame sustenance, delay in the ignition, and high flame temperatures of ANPs. Charoensaeng et al. (2018) conducted an experiment in a diesel engine with palm biodiesel–diesel–ethanol blend as fuel and studied about emissions from the engine tailpipe and revealed that the microemulsion biofuels presented an increase in SFC and a reduction in nitrogen oxide emissions to an extent. Soudagar et al. (2018) investigated the impact of nanoparticles on performance characteristics of biodiesel-blended diesel engines and reported that the incorporation of nano additives resulted in an increase in improvement of
thermophysical properties, decrease in emissions from engine tailpipe, and better stabilization of air–fuel mixture. El-Seesy et al. (2018) studied the impact of nanoparticle additives-blended biodiesel to study the performance characteristics of the diesel engine on various loading conditions by adding different ANPs (aluminum oxide nanoparticles) concentrations and reported that at full load condition, there was a decrease in various exhaust emissions: hydrocarbons reduced by 80%, nitrogen oxides by 70%, and carbon monoxide reduced by 60%. Ghadikolaei et al. (2019) studied the formation of particulate matter and chemical properties of ternary fuel blend of ethanol–biodiesel–diesel at various load conditions. They found out that this type of blend has a better impact on DPFE (diesel particulate filter efficiency), and the obtained value of metal elements was 0.7% and ions 1.9% and diesel particulate matter was 85.8%.

Yesilyurt and Aydin (2020) studied the performance characteristics while using the DEE (diethyl ether) and cottonseed oil blend in the diesel engine at various load conditions. Diethyl ether was added to a 20% biodiesel–cottonseed blend at various concentrations such as 10%, 7.5%, 5%, and 2.5% (on the basis of volume). While adding 10% diethyl ether to 20% cottonseed–biodiesel blend, it was observed that thermal efficiency was decreased by 17.39% and brake-specific fuel consumption showed an increase of 29.15%. Dogan et al. (2020) studied the diesel engine characteristics which incorporate alcohol as an oxygenated fuel by considering various parameters such as sustainability, enviro-economic, and exergoeconomic; exergy and energy were analyzed and reported that 20%, 10%, and 5% 1-heptanol was added (on the basis of volume) to the pure diesel, and test results show that SFC is more for 20% 1-heptanol blend of 0.221 kg/kWh.

Yesilyurt et al. (2020a, b) studied the impact of ternary blends of safflower–diesel on the various characteristics of a diesel engine. Through the studies, they concluded that brake-specific fuel consumption shows a decrease and brake thermal efficiency shows an increase when ternary fuel blend was used. The addition of pentanol also reduced the emissions of smoke, hydrocarbons, and carbon monoxide. Yesilyurt and Aydin (2020) conducted various tests on diesel engines with a tri-fuel blend of diesel–biodiesel–cooking oil at various fuel injection pressures (220, 210, 190, 180, and 170 bar) and reported that BTE has increased considerably at higher injection pressure. Yesilyurt et al. (2020a, b) studied about the blend of four fuels, i.e., vegetable oil–alcohol–biodiesel–diesel blend, to analyze the emission, combustion, and performance characteristics of a diesel engine. From the results obtained, the author recommends using a 4-fuel blend to achieve better performance in diesel engines in the future. Yesilyurt (2019) investigated the influence of injection parameters on diesel engines and reported that while using diesel fuel at 190 bar, it was found that a maximum of exergy and energy efficiencies of 21.27% and 24.5% were obtained.

According to ASTM standards, the chemical and physical characteristics of this ternary fuel are very close to that of conventional diesel fuel. But there was a lack of a decent number of technical literatures on the emission, combustion, and performance characteristics of ternary fuel blends for application in a diesel engine. The fuel characteristics of ternary fuel are superior, so we can utilize the usage of viable energy resources such as biodiesel and bioethanol to their full extent. In this way, we can satisfy the emerging demand for fossil fuels in the present scenario. From this study, a sincere effort has been made to understand the major impacts of ternary fuel at different fuel injection pressure (20 to 50 MPa) on exhaust emissions, combustion, and engine performance in a four-stroke, single-cylinder, common rail direct injection diesel engine (with similar operating conditions), and the obtained results are compared with that of conventional diesel (D100).

### Materials and methodology

#### Mahua methyl ester preparation

Table 1 represents the fatty acid composition of mahua oil. Vegetable oil can be tailored in such a way to decrease its density and viscosity so that the final product produced will have adequate properties to be used as a diesel engine fuel. The process of using alcohol to break down the molecules of untreated vegetable oil into methyl in the presence of a catalyst (KOH or NaOH) is known as transesterification. This process produces glycerin as a by-product which can be used for other purposes. For this process, the catalyst used is KOH at a concentration of 7 g/l dissolved with methanol through powerful stirrings inside a reactor. The next step involves mixing the catalyst–methanol mixture with crude mahua oil. Finally, the vigorous stirring of the final mixture for 60 min at 60°C in ambient pressure. After the completion of the transesterification process, two different liquid phases are formed, i.e., glycerin and methyl ester. Being on the heavier side, liquid raw glycerin will sink into the bottom after a few hours of settling process. After settling, phase separation gets completed in 2–3 h. Whereas complete settling of methyl ester can take up to 8–10 h. Methyl ester washing requires two steps. A wash solution made with 1-gm tannic acid per liter

| Fatty acid | Formula | Structure | Weight (%) |
|-----------|---------|-----------|------------|
| Arachidic | C20H36O2 | 20:0      | 1.5        |
| Linoleic  | C18H32O2 | 18:2      | 14.3       |
| Oleic     | C18H32O2 | 18:1      | 37.0       |
| Steraic   | C18H32O2 | 18:0      | 22.7       |
| Palmitic  | C16H32O2 | 16:0      | 24.5       |
of water and vegetable oil (26%) is mixed with the methyl ester and then emulsified. Stirring is maintained until the methyl ester becomes transparent. The viscosity values of both methyl esters were found to be decreased after the transesterification process, and the value is closer to that of diesel fuel. Methyl ester made through the abovementioned process (20%) is then blended with plain diesel (70%) for the preparation of biodiesel blends that can be used in the CRDI engine for performing various tests.

### Preparation of ternary fuel blend

As mentioned in the “Introduction,” a TF blend was put forward with 70% diesel, 20% MMe, and 10% pentanol (Yilmaz and Atmanli 2017a; 2017b). Magnetic stirring is done for 2 h for 1 l of this mixture. The obtained blend is named TF, and various tests were conducted according to ASTM standards for deciding various properties such as the flashpoint, cetane number, calorific value, kinematic viscosity, and density; and values are given in Table 2. Cetane number, viscosity, and density were decreased by 8.16%, 26.73%, and 2.52%, respectively, when compared to 100% MMe fuel and higher than diesel fuel by 0.83%, 11.97%, and 1.43%, respectively. Still, the properties of the TF blend are not a match to conventional diesel fuel.

### CHN elemental analysis

A CHN analyzer is used to determine the key biodiesel components, such as nitrogen (N), hydrogen (H), and carbon (C). In a CHN analyzer, the biodiesel sample is subjected to oxidative decomposition and consequently, nitrogen reduction takes place along with final product formation such as nitrogen, CO₂, and water. The samples of diesel and blend of MMe were analyzed with the help of thermo-Finnigan CHN. The percentages of carbon in D100 and MMe20 were found to be 86.0301% and 84.3253%, respectively. So it is evident from the results that MMe has less carbon content when compared to diesel. After combining MMe with diesel, the MMe20, the hydrogen element percentage was 11.8024%. So it is evident that the percentage of hydrogen content is more in MAHUAME20.

### Experimental setup and methodology

A 4S (4 stroke), single-cylinder, Kirloskar AV1 diesel engine was used to perform the experiments, which was assisted by the CRDI system. The power of the diesel engine was rated as 3.7 kW. The injection pressure was kept at the range of 20–100 Mpa, and the speed of the engine was 1500 rpm with different load scenarios. In order to maintain the engine speed according to pressure, fuel injection duration is maintained between 650 and 1200 μs. During the initial stage, in order to obtain the baseline data, conventional diesel was used in the engine; then a ternary fuel (pentanol 10% + MMe 20% + diesel 70%) was used. Table 4 describes the specifications of the engine. A model of Kirloskar’s pump (high pressure) powered by a motor running constantly at 1500 revolutions/min. This fuel pump is operated distinctly to make the injection pressure range of injection wider and more independent of engine speed. In the process of production, the control of pressure is done by an inlet valve which only allows the required amount of fuel to be pressurized. The pressure adjustments happen by activating spill valves with the help of a signal of extreme frequency. For these adjustments, a controller (16-bit) and an adequate spill flow are required, but since none of them were accessible, the pressure was regulated using a supplementary high-pressure valve. Additionally, a pressure sensor (injection) was mounted on the rail which facilitates the monitoring and controlling of pressure in fuel. In order to offer high-pressure data, in between the injector and the rail, a pressure transducer (fast response) was also arranged. Bosch provided the common rail system which was used on the test bed. A volume of 18 cm³, with the highest pressure up to 100 Mpa, can be endured by this linear rail having four injector ports. A pressure sensor is fixed on the single side of the tube. The data recorded by the pressure sensor are sent to the engine ECU for calculation of injection flow, timing, and closed-loop control of the injection pressure. In order to load the engine, ECD

### Table 2 Properties of diesel, MMe, and ternary fuel

| Properties                  | Fuels   | D100 | MMe20 | TF   | ASTM standard |
|-----------------------------|---------|------|-------|------|---------------|
| Density at 20°C (kg/m³)     |         | 840  | 874   | 852  | D1298         |
| Viscosity at 35°C (cSt)     |         | 2.84 | 4.34  | 3.18 | D445          |
| Cetane number               |         | 48   | 52.7  | 48.4 | D976          |
| Calorific value (kJ/kg)     |         | 44700| 42673 | 43176| D240          |
| Flashpoint (°C)             |         | 68   | 130   | 59   | D93           |

### Table 3 CHN element analysis for D100 and MMe20 blends

| Element | D100 0.1871 | MMe20 0.2122 |
|---------|-------------|--------------|
| C (%)   | 86.0301     | 84.3253      |
| H (%)   | 12.0732     | 11.8024      |
| N (%)   | 0.1871      | 0.2122       |
An Eddy current dynamometer was incorporated. The density of smoke was measured using an AVL smoke meter. Nitrogen oxide emissions, hydrocarbons, and carbon monoxide were measured using AVL five gas analyzer. HRR (heat release rate) and ICP (in-cylinder pressure) are measured with the help of a data acquisition system (dual-core processor interface) as mentioned in Table 5. Figure 1 represents the schematic of the experimental setup. The in-cylinder pressure was measured with the help of a piezoelectric transducer mounted on an engine cylinder head whereas the HRR (heat release rate) generated during power stroke was calculated.

According to the first law of thermodynamics:

\[
\frac{dU}{dt} = Q - W
\]

\[
mC_v \frac{dT}{dt} = Q - P \frac{dV}{dt}
\]

where \(Q\) is the combination of the heat release rate and heat transfer rate across the cylinder wall and \(W\) is the rate of work done by the system due to system boundary displacement. To simplify Eq. (2), the ideal gas assumption can be used.

\[
PV = mRT
\]

Assuming constant mass, Eq. (3) is differentiated as:

\[
\frac{dT}{dt} = \frac{1}{mR} \left[ P \frac{dV}{dt} + V \frac{dP}{dt} \right]
\]

After combining these two equations, the heat release equation becomes

\[
Q = \left[ \frac{C_v}{R} + 1 \right] P \frac{dV}{dt} + \frac{C_v}{R} V \frac{dP}{dt}
\]

After replacing time \((t)\) with the crank angle \((\theta)\), Eq. (5) becomes

\[
Q_t = \frac{k}{k-1} P \frac{\partial V}{\partial \theta} + \frac{k}{1-k} V \frac{\partial P}{\partial \theta}
\]

where \(k\) is the ratio of specific heats and taken as 1.35 for diesel heat release analysis, \(\theta\) is the crank angle, \(P\) is the cylinder gas pressure, and \(V\) is the cylinder volume.

BSFC was calculated in order to evaluate the engine performance depending on the fuel types used in the study. For this purpose, Eqs. (2) and (3) were used to achieve the BSFC data [43].

\[
BP = \frac{2\pi \omega T}{1000}
\]

\[
BSFC = \frac{m_f \times 10^6}{BP}
\]

Eq. (4) was also used to determine BTE values (%) in the present paper. BTE represents the brake thermal efficiency, and LHV (MJ/kg) represents the lower heating value of the used fuel.

\[
BTE = \frac{3600}{BSFC \times LHV} \times 100
\]

**Uncertainty analysis**

Precise calibration with optimal atmospheric conditions is equally important while using precision measurement devices for reliability. Uncertainty analysis gives a broad view of experimental repeatability by counting necessary errors during measurement (Yesilyurt and Arslan 2018; Yetter et al. 2009).

### Table 4 Details of the experimental test rig

| Make and type          | Kirloskar 3.7-kW vertical CRDI engine |
|------------------------|--------------------------------------|
| Engine type            | Automotive (multispeed)              |
| Stroke length          | 110 mm                               |
| Swept volume           | 625 cm³                              |
| Compression ratio      | 18.0:1                               |
| Torque/Power           | 30 Nm at 2000 RPM, 9 bhp at 3000 RPM |
| Injectors              | Solenoid                             |
| Fuel Injection         | Common rail direct injection system  |

| Dynamometer            | Make Power mag                       |
|------------------------|--------------------------------------|
| Type                   | Eddy current                         |
| Load measurement method| Strain gauge                         |
| Maximum load           | 12 kg                                |
| Cooling                | Water                                |

Table 4 Details of the experimental test rig

### Table 5 Precision for five gas analyzers

| Emission apparatus | Range for measurement | Resolution | Instrument accuracy | % uncertainty in sampling |
|--------------------|-----------------------|------------|---------------------|--------------------------|
| NO                 | 0–5000 ppm vol        | 1 ppm vol  | ± 5 ppm             | ± 0.2                    |
| HC                 | 0–3000 ppm vol        | ≤ 2000:1 ppm vol, > 2000:10 ppm vol | 0–4000 ppm ± 8 ppm, 4001–10,000 ppm | ± 0.15, ± 0.2             |
| CO                 | 0–15% vol             | 0.01% vol  | ± 0–10% ± 0.02% abs, 10.01–15% | ± 0.2                   |

Assuming constant mass, Eq. (3) is differentiated as:

\[
\frac{dT}{dt} = \frac{1}{mR} \left[ P \frac{dV}{dt} + V \frac{dP}{dt} \right]
\]

After combining these two equations, the heat release equation becomes

\[
Q = \left[ \frac{C_v}{R} + 1 \right] P \frac{dV}{dt} + \frac{C_v}{R} V \frac{dP}{dt}
\]

After replacing time \((t)\) with the crank angle \((\theta)\), Eq. (5) becomes

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Q_t = \frac{k}{k-1} P \frac{\partial V}{\partial \theta} + \frac{k}{1-k} V \frac{\partial P}{\partial \theta}
\]

where \(k\) is the ratio of specific heats and taken as 1.35 for diesel heat release analysis, \(\theta\) is the crank angle, \(P\) is the cylinder gas pressure, and \(V\) is the cylinder volume.

BSFC was calculated in order to evaluate the engine performance depending on the fuel types used in the study. For this purpose, Eqs. (2) and (3) were used to achieve the BSFC data [43].

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\[
BTE = \frac{3600}{BSFC \times LHV} \times 100
\]
In the given atmospheric conditions, the process of quantifying performance and emission parameter measurement errors due to the methodology of the experiment, observation accuracy, calibration, and instrumentation employed is called the experimental uncertainty analysis. Two major components are identified with uncertainty analysis: one of which is repeated measurement random variation and the second is accuracy bias. The values of uncertainty are shown in Table 5 for the gas analyzer used and performance parameters. Equation (10) shows the total uncertainty as below:

\[
\text{Overall uncertainty of the experiment} = \sqrt{(BP)^2 + (FC)^2 + (BSFC)^2 + (BTE)^2 + (HC)^2 + (CO)^2 + (NO)^2 + (Load)^2 + (Pressure\ transducer)^2} \\
\sqrt{(1.11)^2 + (0.66)^2 + (1.29)^2 + (1.29)^2 + (0.2)^2 + (0.2)^2 + (0.46)^2 + (0.1)^2 + (1.2)^2} = \pm 2.595\
\]

The evaluation of uncertainty values of each equipment was conducted, which established the experimental uncertainty (current) to be ± 2%. This value is obviously within the acceptable uncertainty range, i.e., less than ± 5%.

**Economic analysis comparison of biodiesel and diesel**

Economic factors that affect biodiesel production can be the cost of raw material, capital, and chemicals used and also the capacity of plant and technology used. Out of these, 80% is the raw material cost, and the rest includes the cost of chemicals (catalyst and methanol) and labor. The estimated costs for preparing biodiesel using MME oil and feedstock are 0.54/l and 0.16/l US$. These costs can be lessened if there are some ways to reuse the by-product glycerol. Also, if the feedstock is selected appropriately, the overall cost may reduce. Nowadays, since nonfood crops are popular, this can be seen as an opportunity to increase biodiesel production. Other socioeconomic factors such as regional development, sustainability, agriculture with social structure, and supply security can be an advantage of biodiesel over diesel and other petroleum products. The power generated by these biofuels will improve the local economy (Table 6).

**Results and discussion**

The working of diesel engine was aided by a common rail direct injection system with fuel as ternary fuel (pentanol 10% + MME 20% + diesel 70%) blends and was observed to be pretty clean for the entire rated load and varying pressure during fuel injections of 22–88 MPa. SFC, BTE, emissions (nitrogen oxide emissions, HC, CO), and density of smoke are plotted against load. With the available data regarding combustion, HRR and cylinder pressure were plotted against crank angle.

**Effect of fuel injection pressure on engine combustion characteristics**

In this section, heat release rate, in-cylinder pressure, and ignition delay are discussed for all the fuels. These are the important parameters that help understand the behavior of the ternary fuel in the combustion chamber. The average temperature of the gas in the cylinder and the heat release rate are calculated using the “Enginesoft/LabView” software that uses the 0D model of a single combustion zone with a constant specific heat ratio.

![Schematic of experimental layout](image-url)
Cylinder pressure

Figure 2 illustrates the variation of the inside cylinder pressure of ternary fuel with respect to crank angle at various injection pressures. It is observed from the plot that pure diesel at a pressure of 51.62 bar is at the least point, whereas the maximum cylinder pressure is observed for TF50 at 60.10 bar because of the shorter ignition delay. Pentanol’s presence in ternary fuel helps the oxygen molecules present in it for effective combustion with improvement in-cylinder pressure. To sustain burning in the main combustion phase and for the complete combustion of the fuel during the precombustion phase, higher content of oxygen in ternary fuel is enough. It is evident that TF50 has a 0.91% higher peak pressure when compared with conventional diesel fuel.

While testing fuel during the running of the engine at no load and partial load scenarios, similar tendencies were seen. The fuel-air combination generation is fairly constant which results in increased pressure when the cetane number is improved and the viscosity of fuel remains higher. It is observed from the plot that with an increase in the injection pressure from 20 to 50 Mpa, cylinder pressure also increased. For TF20, cylinder pressure was found to be 52.68 bar; for TF30, it is 55.273 bar and for TF40, it is 58.112 bar. The reason for the same could be the production of better atomization, thereby, reducing physical delay. The cylinder pressure of TF50 is found to be better, which is 60.10 bar measured among various fuel injection pressures, and it is 4.39% greater than TF20 (Yilmaz 2012; Atmanli et al. 2015; Yilmaz and Morton 2011).

Heat release rate

Figure 3 illustrates the variation of heat release rate with respect to crank angle for different fuel injection pressure at full load conditions. When compared to other fuel injection pressures, the heat release rate of D100 is observed to be 110.36 kJ/m^3.deg; this is due to the accumulation of diesel fuel at the primary combustion phase which is in the premixed stage. Due to better thermal physical characteristics of ternary fuel, resulting in the advancement of combustion process as mentioned in the open literature. The combustion phase of ternary fuel advances, and the heat release rate reaches its peak as the injection pressure increases from 20 to 50MPa because of ignition delay shortening through improved fuel–air mixing. Because of improved atomization of injected fuel, higher dispersion, disintegrate length, and average diameter of the fuel droplet, the ignition delay period decreases to an extent when injection pressure is increased. A maximum heat release rate of 126.89 kJ/m^3.deg is observed in TF50 whereas a heat pressure...
release rate of 120.33 kJ/m³deg is observed at the fuel injection pressure of TF20 (Yilmaz et al. 2014b, 2018).

**Effect of fuel injection pressure on engine performance characteristics**

In this section, the effect of fuel injection pressure with ternary fuel on brake thermal efficiency and brake-specific fuel consumption is discussed. The two parameters explain the energy conversion efficiency of the engine.

**Brake thermal efficiency**

For ternary fuel (pentanol 10% + MME 20% + diesel 70%) with different fuel injection pressures from 20 to 50 MPa, it was evident that the BTE has increased with an increase in load as illustrated in Fig. 4. The brake thermal efficiency of ternary fuel at FIP (fuel injection pressure) of 50 MPa (TF50) was much greater when compared with other FIPs and also conventional diesel. The brake thermal efficiency of diesel is minimum (at all loads) with lower levels of viscosity and density. Brake thermal efficiency at all loads is seen to increase because of the presence of pentanol in ternary fuel. This is mostly due to sufficient oxygen molecules in pentanol, resulting in better combustion. As brake thermal efficiency of ternary fuel is still lower, enhancement in the same can be sought with an increase of fuel injection pressure. It is observed from the figure that with an increase in the FIP, an increase of 1.58%, 1.62%, 2.34%, and 4.39% was observed for TF20, TF30, TF40, and TF50 in brake thermal efficiency. Simultaneously, as the evaporation rate is increased and physical delays are reduced which leads to improved BTE along with an increase in combustion efficiency. In comparison with TF30 and TF40, better thermal efficiency is observed at TF50 at various engine loads. These results are in accordance with the increase in FIP in the ternary fuel blend with the increased thermal efficiency as mentioned in the open literature (Singh et al. 2019a, b; Yilmaz and Morton 2011).

**Brake-specific fuel consumption (SFC)**

Figure 5 represents the variations in BSFC (specific fuel consumption) for ternary fuel with different fuel injection pressure and conventional diesel with load. Corresponding to the increase in the heating value, the BSFC of ternary fuel is also expected to increase. Comparing the properties, ternary fuel has got a higher heating value than diesel fuel. As compared to diesel fuel, SFC while fuelling with the ternary fuel is increased, which can be seen in Fig. 5, and this states an agreement with many literatures [24-28] as well. Figure 5 also shows the increase of injection pressure along with the reduction of consumption of specific fuel. Atomization takes place as fuel droplet diameter decreases because of an increase in the injection pressure which leads to better combustion. At an FIP of 20 MPa, conventional diesel has higher BSFC than ternary fuel at different FIPs. A substantial lessening of around 7.5% was observed in BSFC of ternary fuel at 50 MPa when compared with ternary fuel of 20 MPa. The BSFC becomes 0.34, 0.33, 0.31, and 0.30 kg/kWh for TF20, TF30, TF40, and TF50, respectively, for different fuel injection pressures at maximum load (Singh and Chauhan 2017; Singh et al. 2019a, b).

**Effect of fuel injection pressure on engine emission characteristics**

In this section, the effect of fuel injection pressure with ternary fuel on CO, HC, NOₓ, and smoke opacity is discussed. The exhaust emissions are the reflection of the combustion characteristics of the fuel in the engine. The appropriate combustion exhibits higher NOₓ and lower HC, CO, and smoke emissions.
Carbon monoxide emissions

Figure 6 illustrates the variation of carbon monoxide emissions for ternary fuel for different fuel injection pressure, clearly indicating that in comparison to plain diesel, ternary fuel blends outperform it by emitting a significantly lesser quantity of CO (Raju et al. 2018; Tyagi et al. 2008). This reduction is due to the combustion enhancer, pentanol, which amplifies the combustion process. Apparently, due to the presence of high oxygen quantity, the rate of conversion of carbon monoxide to dioxide gets accelerated while using ternary fuel in an engine. Initial decrease of CO emission at inferior loads up to 30% and a significant increase for the ternary fuel are observed. Due to excess oxygen present in methyl esters, combustion is relatively smooth than diesel [30]. It is observed from the figure that for ternary fuel at a fuel injection pressure of 20 MPa, a 15.78% decrease in CO emission is reported in comparison with conventional diesel. The by-product of incomplete combustion is called carbon monoxide which is generated from the incomplete oxidation of compounds containing carbon. When there is insufficient oxygen to make CO₂, carbon monoxide is produced. The condition wherein CO emission appears due to rich mixture and the pressure surge happens during injection of fuel which in turn directs the mixtures to progress towards lean. Here, CO emission is decreased to 22.24% when there is an increase in FIP of 50 MPa.

Hydrocarbon emissions (UHC)

At different fuel injection pressures, the variation in emissions of unburnt hydrocarbon (UHC) against engine load is presented in Fig. 7. To determine the emission behavior of the engine, UHC is also an important limitation. At all the engine loads, UHC emission is observed to be highest in conventional diesel as compared to ternary fuel which emitted 6.84%, 10.36%, 11.31%, and 9.49% (TF20, TF30, TF40, and TF50) of unburnt hydrocarbons. This trend matches with CO emission of ternary fuel with respect to plain diesel. An increase in the load affects the UHC emission because of high air–fuel mixture generation in wall films and crevices (cold quench areas) and plenty of available fuel in the combustion zone. Here, we can conclude that hydrocarbon emission further decreases as pressure during injection increases (Kapoor et al. 2020; Kumar et al. 2019).

Nitrogen oxides

The variation of pure diesel, blended with MME20 and pentanol against load for NOₓ emission, is illustrated in Fig. 8. NOₓ generation is a sophisticated process. A number of factors are responsible for NOₓ formation, such as working conditions, response time, combustion temperature, features of engine design, and fuel properties. From the plot, it can be observed that diesel produced lesser NOₓ emissions than ternary fuel. The reason could be the oxygen molecule presence which amplified the combustion process and thereby raising the combustion chamber temperature. Due to the very high temperature inside the cylinder resulting from better combustion, NOₓ emissions raised for ternary fuel by 12.46% when compared with diesel when the load is increased to a maximum. In the case of injection pressure, NOₓ emissions were increased with the increase in injection pressure. It is because fuel injection pressure increases the rapid atomization and shortens the ignition delay period, resulting in better combustion of fuel. Due to better combustion, at high temperatures, N₂ will react with oxygen and produce more amount of NOₓ emissions. Nitrogen and oxygen combinedly yield a high quantity of nitrogen oxide at high temperatures due to better combustion (De Luca et al. 2005; Kao et al. 2008).

Smoke opacity

In Fig. 9, the different fuel injection pressures of the smoke opacity variation in relation to engine load are illustrated.

![Fig. 6 Variation of carbon monoxide emissions against load for different fuel injection pressures](image_url)

![Fig. 7 Variation of unburnt hydrocarbon against load against different fuel injection pressures](image_url)
The highest smoke emission is observed in diesel for all the load conditions as compared to ternary fuel. Excess accumulation of fuel in the cylinder, deficiency of oxygen in the combustion-rich areas, and poor atomization result in the formation of smoke in CI engines which is mostly due to partial combustion. Thus, it means that the low smoke emissions found in ternary fuel are due to better fuel oxidation inside the combustion chambers present near the fuel-rich zones. The smoke level is further reduced with the increment of pressure during the injection of fuel. Further, smoke emissions were seen to be nominally reduced by 6.15%, 6.92%, 5.37%, and 5.12% at full load for TF20, TF30, TF40, and TF120, respectively. The lower smoke emission is probably due to the lower delay period due to which before ignition, surplus fuel is collected inside the cylinder, making sure that combustion rate is high enough and better fuel–air mix is facilitated (Ashok et al. 2020; Bhowmik et al. 2017).

**Ignition delay**

The word ignition delay or ignition delay period was commonly termed as a divergence from the start of combustion and the start of injection of test fuel to the engine, and it also reveals the physical and chemical delays of the fuel. The ignition delay suggests fuel mixing and atomization at the final stage, while the full precombustion process is shown at the initial stage. Depending on engine load for different fuel injection pressures, the change in ignition delay is depicted in Fig. 10. The highest ignition delay is observed in the conventional diesel at all loads. But for lower loads, TF30 and TF40 showed a similar delay profile. The reason behind this was mainly due to the viscosity, density, and better fuel mix rate. The next blend having a lower delay period was TF20 due to its low compressibility factor, high cetane number, and biodiesel composition. The reason behind the low delay period is attributed to better fuel atomization through the high surface tension and calorific range of the blend.

**Trade-off (BSFC–BTE–NOx)**

The detrimental effects of soot and NOx are well known to everyone. They cause a plethora of respiratory illnesses and degrade the environment by causing global warming through smog formation. Furthermore, a fuel which is consumed in lesser quantity by the engine is one of the factors to choose a fuel; another reason being the depleting fuel reserves. Thus, to get a clearer picture, a trade-off study is required for comparing the emission and performance of engines using various fuels with respect to SFC, brake thermal efficiency, and NOx emission (Bhowmik et al. 2018; Chen et al. 2018). It also gives scope for further explanation of intrinsic issues regarding the previously mentioned factors. Figure 11 shows the 20 to 100% load trade-off for different combinations of ternary fuels.
fuel (pentanol 10% + MME 20% + diesel 70%) with different injection pressures (TF20, TF30, TF40, and TF50). It can be noticed clearly that the trade-off shifts to the extreme left corner (minimum fuel consumption) from the extreme right corner (maximum fuel consumption). From the graph, the TF50 is seen to push the trade-off to a high-NOx emission zone and BTE with a reduction of BSFC. Ternary fuel operation reduces the equivalent BSFC along with NOx emission.
Of the other blend, the ternary blend produces lesser NO\textsubscript{x} and more BTE. When the load is increased to 40%, the TF50 blends the smoke opacity, and equivalent BSFC reduction is seen which is indicated through the shifting of the trade-off zone near to the origin. Based on the trade-off pattern for the fuel sample considered, the following results can be concluded: (1) whenever the percentage of ternary fuel blended with iron oxide additives increases, it is seen that BSFC decreases but when BTE rises, NO\textsubscript{x} also goes up; (2) on the other hand, TF50 shows high emission of NO\textsubscript{x} and BTE but a relatively low BSFC as portrayed in the top area of the graph. (3) Interestingly, TF40 shows the optimum trade-off zone with higher BTE and lower BSFC and lowest NO\textsubscript{x} emission from the current study.

Conclusions

The present work is focused on the influence of high fuel injection pressure on ternary fuel, and D100 is studied for its effect on engine emission, performance, and combustion characteristics. The following conclusions are drawn based on the extensive experimental study:

- Ternary fuel (TF) is obtained by mixing 10% pentanol, 20% MME biodiesel, and 70% diesel together. High NO\textsubscript{x} and brake-specific energy consumption (kJ/kWh) were observed for ternary fuel than conventional diesel.
- Due to poor mixing characteristics, the time for combustion of ternary fuel was higher than the conventional diesel in current research. With an increase in the FIP, cylinder pressure also increased which also led to an increase in heat release rate.
- BSFC drastically reduced to 7.5% when the pressure during injection of fuel was increased from 20 to 50 MPa. With an increase in FIP, brake thermal efficiency also increased.
- As compared to D100, ternary fuel showed lower CO, HC, and smoke. Also, when the pressure during injection of fuel was increased to 50 from 20 MPa, CO, HC, and smoke emissions were further reduced.
- It is concluded that ternary fuel usage in CRDI diesel engines with a fuel injection pressure of 50 MPa leads to better thermal efficiency (BTE) and reduction in harmful gas emissions.
- In future work, the NO\textsubscript{x} emission of ternary fuel can be reduced by using exhaust gas recirculation and optimization of injection timing.

Nomenclature

- BTE, brake thermal efficiency (%); BSFC, brake-specific fuel consumption (kg/kWh); DI, direct injection; CI, compression ignition; NO\textsubscript{x}, oxides of nitrogen (ppm); CO, carbon monoxide (% vol); HC, hydrocarbon (ppm); CO\textsubscript{2}, carbon dioxide (% vol); CP, cylinder pressure (bar); FIP, fuel injection pressure

Author contribution

Jatho Ramachander: conceptualization, methodology, writing, and review and editing; Santhosh Kumar Gugulothu: formal analysis and investigation and writing original draft and preparation; Gadepalvi Ravikiran Sastry: supervision; Burra Bhasker: supervision.

Data availability

All data generated or analyzed during this study are included in this article.

Declarations

The present study work was not conducted on human or experimental animals where national or international guidelines are used for the protection of human subjects and animal welfare.

Ethical approval

Not applicable.

Consent to participate

Not applicable.

Consent for publication

We confirm that the manuscript has been read and approved by all named authors and that there are no other persons who satisfied the criteria for authorship but are not listed. We further confirm that all the authors listed in the manuscript have been approved by all of us.

Competing interests

The authors declare no competing interests.

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