

Research Article

Influence of Injection Pressure on the Dual-Fuel Mode in CI Engines Fueled with Blends of Ethanol and Tamanu Biodiesel

M. Parthasarathy ^(b), ¹ P. V. Elumalai ^(b), ² M. Murunachippan, ¹ P. B. Senthilkumar, ¹ Saboor Shaik ^(b), ³ Mohsen Sharifpur ^(b), ^{4,5} and Nima Khalilpoor ^(b)

¹School of Mechanical and Construction, Vel Tech Rangarajan Dr. Sagunthala R&D Institute of Science and Technology, Chennai, India

²Department of Mechanical Engineering, Aditya Engineering College, Surampalem, India

³School of Mechanical Engineering, Vellore Institute of Technology, Vellore 632014, Tamil Nadu, India

⁴Clean Energy Research Group, Department of Mechanical and Aeronautical Engineering, University of Pretoria, Hatfield, Pretoria, South Africa

⁵Department of Medical Research, China Medical University Hospital, China Medical University, Taichung 404, Taiwan ⁶Department of Energy Engineering, Graduate School of the Environment and Energy, Science and Research Branch, Islamic Azad University, Tehran, Iran

Correspondence should be addressed to M. Parthasarathy; msarathymech@gmail.com, P. V. Elumalai; elumalaimech89@gmail.com, Mohsen Sharifpur; mohsen.sharifpur@up.ac.za, and Nima Khalilpoor; nimakhalilpoor@gmail.com

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The acceleration of global warming is primarily attributable to nonrenewable energy sources such as conventional fossil fuels. The primary source of energy for the automobile sector is petroleum products. Petroleum fuel is depleting daily, and its use produces a significant amount of greenhouse emissions. Biofuels would be a viable alternative to petroleum fuels, but a redesign of the engine would be required for complete substitution. The use of CNG in SI engines is not new, but it has not yet been implemented in CI engines. This is due to the fuel having a greater octane rating. The sole use of CNG in a CI engine results in knocking and excessive vibration. This study utilizes CNG under dual-fuel conditions when delivered through the intake manifold. In a dual-fuel mode, compressed natural gas (CNG) is utilized as the secondary fuel and a blend of 90% tamanu methyl ester and 10% ethanol (TMEE10) is used as the primary fuel. The injection pressure (IP) of the primary fuel changes between 200 and 240 bar, while the CNG induction rate is kept constant at 0.17 kg/h. The main combustion process is governed by the injection pressure of the pilot fuel. It could be affecting factors such as the vaporization characteristics of the fuel, the homogeneity of the mixture, and the ignition delay. Originally, tamanu methyl ester (TME) and diesel were used as base fuels in the investigation. As a result of its inherent oxygen content, TME emits more NOx than diesel. The addition of 10% ethanol to TME (TMEE10) marginally reduces NOx emissions in a CI mode because of its high latent heat of vaporization characteristics. Under peak load conditions, NOx emissions of TMEE10 are 6.2% lower than those of neat TME in the CI mode. Furthermore, the experiment was conducted using TMEE10 as the primary fuel and CNG as the secondary fuel. In the dual-fuel mode, the TMEE10 blend showed higher combustion, resulting in an increase in performance and a significant decrease in emission characteristics. As a result of the CNG's high-energy value and rapid burning rate, the brake thermal efficiency (BTE) of TMEE10 improves to 29.09% compared to 27.09% for neat TME. In the dual-fuel mode of TMEE10 with 20.2% CNG energy sharing, the greatest reduction in fuel consumption was 2.9%. TMEE10 with CNG induction emits 7.8%, 12.5%, and 15.5% less HC, CO, and smoke, respectively, than TME operation.

1. Introduction

The world's energy demand has triggered vigorous research on nonfossil fuels, nonpolluting fuels, and renewable fuel sources. Conventional fossil fuel is widely used in the entire world for transportation applications [1]. Today, more countries depend on imports of fossil fuels to meet their energy requirements [2]. The situation is highly critical in countries such as China and India, which import 85% of the essential oil. The increasing trend in vehicle population and the industrialization of the world have caused a rise in the demand for and consumption of fossil fuels [3]. The depletion and nonrenewability of fossil fuels have necessitated the search for alternative power sources for transportation [4]. It is very essential to use alternative fuels as an energy source to sustain current needs and future requirements, and they are also environmentally friendly and energy efficient for the automobile sector [5]. It is more important to discover the viability of replacing diesel with alternative fuels, which can be produced within the nation. Neat vegetable oils are converted to biodiesel by a transesterification process [6]. In this process, neat vegetable oils are chemically treated with alcohol in the presence of catalysts such as KOH (potassium hydroxide) and NaOH (sodium hydroxide) for converting triglycerides into alkyl esters [7]. Vegetable seed ester is filtered to separate it from glycerol. Undoubtedly, transesterification is the best suitable method for utilizing vegetable oils in compression ignition engines [8]. Tamanu is a nonedible plant. The seed from a single tree ranges from 50 kg to 100 kg, having an amount of 50%–63% of the oil [9]. Tamanu oil has a lower calorific value than diesel because of the presence of oxygen molecules [10]. Additional names for this tree are Alexandrian laurel, punnai, kamani, and tamanu. It belongs to the Clusiaceae (Mangosteen) family [11]. Some of the important properties of tamanu methyl ester (TME), diesel, ethanol, and natural gas are shown in Table 1. Deepankumar conducted the experimental analysis of the performance characteristics of tamanu oil as an alternative fuel in a CI engine. The experimental research study probed the use of renewable tamanu oil as a new possible energy source of alternative fuels for the diesel engine. Biodiesel was prepared from tamanu oil by the transesterification method. Tamanu oil methyl ester and standard diesel were blended at different proportions on a volume basis [12]. In this research study, the engine load was varied from 0% to 100%. It was found that renewable tamanu biodiesel fuel was better at reducing NOx emissions [13]. Xue et al. studied the consequences of using biodiesel on engine performance analysis and emissions levels. A significant reduction in HC, CO, and PM tailpipe emissions was observed with the noticeable loss of power, NOx, and fuel consumption being increased without minor modifications. Researchers from all over the world are making serious efforts to improve the efficiency of the engine and reduce emissions. To get 100% of benefits from using biodiesel in a diesel engine, a small modification in the system is required to achieve better performance [14]. Aydin et al. have studied the consequences of adding ethanol to biodiesel on DICI engine exhaust emission and performance

behaviour characteristics. The addition of ethanol resulted in superior performance characteristics, and the least value was obtained for B20, which was stated to be due to the superior cetane number. Interestingly, the emission of nitrogen oxides increased for the ethanol blend due to the fluctuation in the cetane number [15-17]. As a result of the IP on performance, three different injection pressures, namely, 200, 220, and 240, were varied to achieve the best results. An injection pressure of 240 bar showed higher BTE and reduced emissions [18]. Higher injection pressure resulted in higher cylinder pressure owing to lower ignition delay and superior emission due to a reduction in unburnt hydrocarbons, smoke opacity, and CO with a penalty of nitrogen oxides [19, 20]. Mohsin et al. investigated the performance physiognomies of a CI engine operating on dual fuel. It is used as the main fuel. Biodiesel significantly increases CO and NOx, and a reduction in unburned hydrocarbons and carbon dioxide emissions is noticed. These results showed that biodiesel can be used without any modification of the engine, so it can be used as an alternative, renewable, and environmentally friendly fuel for engines [21].

Gharehghani et al. studied combustion, performance, and emission parameters of a reactivity-controlled CI engine dual-fueled with natural gas and fish oil biodiesel. Biodiesel-CNG average BTE is 1.6% greater than diesel-CNG. At all loads, biodiesel-CNG operation reduced combustion loss by 2%. Biodiesel-CNG reduces unburned HC by 32.5% at all engine loads, whereas its CO emissions are similar to the CO emissions of diesel-CNG. Biodiesel-CNG NOx emissions were higher than diesel-CNG, but still lower than traditional combustion with biodiesel or diesel [22]. Gómez Montoya examined the impact of biogas and natural gas on the emissions and performance of CI engines. Considering that biogas has a larger calorific value than diesel, BSFC dropped as a result of its adoption. Dual-fuel engines have a delayed peak cylinder pressure, indicating a longer ignition delay than diesel engines. The prolonged igniting time is likely owing to the increased CO2 content of gaseous fuel. Compared to diesel, biogas, and natural gas, the single-fuel mode improved peak heat release by 30% and decreased combustion time by 22%. Diesel and CNG NOx emissions were equal at higher loads. Biogas decreased NOx emissions by 37% compared to diesel. NOx reduction was associated with CO_2 concentration in biogas [23].

The dual-fuel engine has the potential to run on either gasoline or diesel, making it more fuel flexible than a standard diesel engine. The characteristics of a dual-fuel engine are affected by the energy distribution ratio of gaseous fuel and pilot fuel. Thermal efficiency is significantly affected by the mass share of the gaseous fuel and pilot fuel. The combustion process of a dual-fuel engine is a hybrid of the SI and CI processes. At a full load, when combustion rates are high, NOx emissions increase, whereas HC and CO emissions decrease in the dual-fuel engine. According to this analysis, the primary purposes of the study were (a) to compare the characteristics of the standard CI engine and the dual-fuel engine powered by tamanu biodiesel with CNG enrichment, (b) to operate the dual-fuel engine with TME and 10% ethanol for an effective reduction in NOx

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emissions, and (c) to improve the performance of the dualfuel engine operated by the CNG induction with TME at various injection pressures ranging from 200 to 260 bar.

2. Experimental Setup

Figure 1 depicts the experimental setup with necessary instruments for engine testing. The experiment was conducted on a single-cylinder, water-cooled, DI CI engine manufactured by Kirloskar. The engine had a compression ratio of 17.1:1 and generated a power of 5.2 kW. The standards of the engine are listed in Table 2. Dual-fuel engines are modified from CI engines that have gaseous or volatile fuels inducted into the engine's intake. The premixed fuel-air combination and the combustible mixture are ignited by pilot fuels delivered through the injector into the combustion chamber. The test was carried out at a constant speed of 1800 rotations per minute under varying load conditions. The load varied from zero to a peak load (such as 0%, 25%, 50%, 75%, and 100%). The engine's standard injection timing (IT) and injection pressure (IP) were defined by the manufacturer at 220 bar and 21obTDC, respectively. A burette was attached to the panel board's front side for measurement of fuel consumption. CNG was delivered into

the engine's inlet plenum via a nonreturn valve. This induction system includes a high-pressure CNG cylinder, control valve, pressure regulator, gas flow metre, and flame arrester. A flame arrester was used to prevent backfire from the inlet port of an engine. The constant CNG flow rate was 0.17 kg/hour for the entire test condition. A Crypton gas analyzer was used to detect engine exhaust emissions, such as UBHC, NOx, CO, and excess oxygen. Table 3 provides the exhaust gas analyzer characteristics.

2.1. Uncertainty Analysis. The result of uncertainty analysis is a function of the independent variables X1, X2, X3, and Xn. A number of measurements were subjected to experiments to determine the mean and standard deviation of any observed parameter (Xi). The engine was permitted to operate under normal conditions. At least five measurements of engine speed, torque, engine temperature, coolant temperature, in-cylinder pressure, emission gases, and fuel consumption time were obtained. The following equation was used to calculate the degree of uncertainty for a variety of experimental results, including BP, BTE, SFC, CO, HC, NOx, smoke and cylinder pressure, and HRR. An analysis of parameter uncertainty is shown in Table 4.

$$R = f(x1, x2...xn)$$

$$x1, \pm \Delta x1, x2 \pm \Delta x2, xn \pm \Delta xn,$$

$$\Delta R = \left[\left(\frac{\delta R}{\delta x1} \Delta x1 \right)^2 + \left(\frac{\delta R}{\delta x2} \Delta x2 \right)^2 + ... + \left(\frac{\delta R}{\delta xn} \Delta xn \right)^2 \right]$$

$$\left[\text{[uncertainty of spee d]}^2 + [\text{uncertainty of BP]}^2 + [\text{uncertainty of the fuel flow rate]}^2 + [\text{uncertainty of HC}]^2 + [\text{uncertainty of CO}]^2 + [\text{uncertainty of Smoke]}^2 + [\text{uncertainty of CP}]^2 + [\text{uncertainty of the CA enco de r]}^2 + [\text{uncertainty of NOx}]^2 = \sqrt{[1.1]^2 + [0.21]^2 + [1]^2 + [1]^2 + [0.2]^2 + [1.01]^2 + [0.5]^2 + [1.04]^2 + [0.3]^2 + [0.5]^2} = 2.44\%.$$
(1)

3. Results and Discussion

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3.1. Performance Characteristics

3.1.1. Brake Thermal Efficiency. As shown in Figure 2, the thermal efficiency of a diesel engine is influenced by CNG induction at varying injection pressures of TMEE10.

Injection pressure can be changed by altering the tension of the needle spring in the injector. At a peak load, the BTE of diesel, TME, and the blend TMEE10 was determined to be 29.69%, 27.89%, and 25.92%, respectively. The BTE of pure TME is less than that of standard diesel. This could be due to biodiesel's weak mixture formation and atomization

TABLE 1: Comparison of fuel properties of TME, diesel, ethanol, and CNG.

Fuel	Diesel	Ethanol	CNG	TME	TMEE10	Test methods
Calorific value (kJ/kg)	43869	26900	48500	41150	39725	ASTM D240
Density (kg/m ³)	832	789	0.72	910	897.9	ASTM D1298
Autoignition temperature (°C)	273	363	568	315	319.8	_
Cetane number	55	11	—	58	53.3	ASTM D613
Flash point (°C)	48	17	_	130	118.7	ASTM D93
Carbon (%wt)	87	52.2	—	77.4	74.88	ASTM D5291
Oxygen (%wt)	0	34.8	—	11.8	14.1	ASTM D5292
Hydrogen (%wt)	13	13	—	10.9	11.11	ASTM D5293
Latent heat of evaporation (kJ/kg)	250	840	509	250	309	—



FIGURE 1: A schematic diagram of the experimental setup.

TABLE 2: Engine specification.

Description	Data
Make and model	Kirloskar and TV1
Types of engines	4 strokes, DI CI engine
Brake power	5.2 kW
Bore and stroke	88 and 110 mm
Compression ratio	17.1:1
Cubic capacity	661 cc
Nozzle opening pressure	220 bar
Fuel injection timing	21°bTDC

properties. Fuel modification (TMEE10) was accomplished by adding 1% isopropanol to the ethanol blend as a blending agent. The conventional engine with a TMEE10 blend produces the least amount of BTE. In addition, it was discovered that the BTE of the engine improved when the injection pressure was increased from its standard condition. The results for TMEE10 + CNG with IP 200 bar, 220 bar, 240 bar, and 260 bar were 26.39%, 28.46%, 29.09%, and 27.22%, respectively. It can be inferred that increasing CNG energy share results in increased combustion rates. As a result of CNG induction, the increased rate of enthalpy and chemical reaction rates enhanced combustion. This attribute improves the BTE in dual-fuel operation in contrast to the CI mode [24]. Due to the rapid burning rate and higher energy content of CNG, the BTE of TMEE10 drastically increased when compared to neat TME. During dual-fuel operation, an increase in BTE percentage can be achieved by increasing the amount of pilot fuel consumption.

3.1.2. Brake Specific Energy Consumption. CNG induction at different TMEE10 injection pressures affects the diesel engine's specific energy consumption, as seen in Figure 3. Brake power decreases BSEC because friction power drops under the increased load. Under normal operation, diesel, TME, and TMEE10 obtained BSEC values of 12.1, 12.8, and 13.8 MJ/kWh, respectively. As expected, higher density of neat TME compared to diesel leads to larger fuel droplets and incomplete combustion, resulting in higher BSEC for all load conditions [25]. When ethanol is included in the mixture, energy consumption rises substantially higher. Because ethanol has a larger latent heat of vaporization than diesel, it is inferred that it absorbs more heat during combustion. Improvements in IP up to 240 bar for TMEE10 resulted in higher BTE and lower BSEC. TMEE10's BSEC at



FIGURE 2: Variations in brake power impact the BTE of the engine.

Description	Data		
Make and model	Crypton and 290 SERIES		
Measured gas	HC, CO, CO ₂ , NOx		
Ranges			
HC	0 to 10000 ppm		
CO	0 to 10% vol		
CO_2	0 to 20% vol		
NO	0 to 5000 ppm		
Accuracy/Performance			
HC	±10 ppm		
CO	±0.03% vol		
CO ₂	±0.5% vol		
NO	±10 ppm		
Flow rate	5 litres/min nominal		

200, 220, 240, and 260 bar IP in the dual-fuel mode varied considerably from 13.6 to 12.4 to 13.2 MJ/kWh. It is possible that this is because of improved combustion caused by greater atomization. For injection pressures above 240 bar, preignition of fine fuel droplets in excess air leads to a decrease in BTE and an increase in BSEC.

3.1.3. CNG Energy Share. Figure 4 depicts the proportion of CNG energy for each test fuel condition. It was determined by comparing the energy acquired from blended TMEE10 with CNG at different loads to the total

TABLE 4: Analysis of parameter uncertainty.

S. no	Parameters	Uncertainty in percentage		
1	Brake power	±0.21		
2	Temperature sensor	±0.16		
3	Speed sensor	±1.1		
4	Thermal efficiency	± 1.04		
5	Pressure sensor	±0.5		
6	Crank angle encoder	±0.3		
7	Smoke metre	± 1.0		
8	Flow rate of fuel	± 1.0		
9	NO _X emission	±0.5		
10	CO emission	±0.2		
11	HC emission	± 1.0		

energy acquired. The figure reveals that the amount of CNG inducted is greater at initial loads, despite the fixed flow rate, because the governor adjusted TMEE10 quantity to maintain the constant speed. Increasing the amount of injected TMEE10 results in a lower percentage of CNG gas substitution during peak load conditions. CNG induction was found to have a proportion of energy of 22.05% at 200 bar, 20.6% at 220 bar, 20.2% at 240 bar, and 21.4 at 260 bar for TMEE10. According to the results, the percentage of the total energy supplied by CNG has decreased as a result of increased pilot fuel quantity. This is because the increased injection pressure improves atomization and chemical processes, reducing energy usage [26].



FIGURE 3: Variations in brake power impact the BSEC of the engine.



FIGURE 4: Variation of the CNG energy share with respect to BP.

3.2. Emission Characteristics

3.2.1. CO Emission. CO formation is shown in Figure 5 as a result of CNG induction with a varying injection pressure of TMEE10. The figure clearly shows that the CO value decreases up to midload conditions, further increasing the load until it reaches its maximum. It was mostly due to the availability of rich-fuel zones in higher load conditions. Diesel, TME, and blend TMEE10 CO emissions without CNG induction were 4.3 g/kWhr, 4.0 g/kWhr, and 4.1 g/kWhr, respectively. At a peak load, TME emits the least amount of CO because of its built-in oxygen and superior air utilization up to midload. An increase in CO emissions was observed after ethanol was added to TMEE10, due to a greater latent heat of vaporization of ethanol. The TMEE10 blend at various IP pressures of 200 bar, 220 bar, 240 bar, and 260 bar was determined to be 4.3 g/kWhr, 3.8 g/kWhr, 3.8 g/kWhr, and 4.3 g/kWhr, respectively. With an increase in the injection pressure, combustion quality improved and CO emissions dropped. It was mostly attributable to improved fuel atomization, which leads to improved homogeneity of charge preparation, hence improving combustion. Induction of CNG leads to a further decrease in CO emissions [21]. This is due to the fact that a high flammability limit improves combustion. With 20.2% CNG energy contribution, CO emissions for TMEE10 decrease from 4.05 g/kWh to 3.80 g/ kWh at a peak load. This decrease in CO emissions is mostly because of the increased heat energy liberated during combustion, which raises the in-cylinder temperature and exponentially accelerates the reaction rate, resulting in enhanced CO2 promotion [9].

3.2.2. HC Emission. Figure 6 shows the formation of HC in response to CNG induction at varying TMEE10 injection pressures. At normal operation, HC emission of diesel, TME, and TMEE10 is determined to be 0.122 g/kWh, 0.112 g/kWh, and 0.114 g/kWh, respectively. It is evident from the above results that diesel has higher HC emissions, followed by TME and TMEE10. The deficiency of oxygen and carbon composition of diesel is higher than that of TME, which results in inferior air entrainment and combustion, eventually resulting in more HC formation in diesel compared to TME and TMEE10. TMEE10 showed 1.75% higher HC emission than TME. Adding ethanol to TMEE10 suppresses the adiabatic temperature, which results in low cylinder temperature and flame quenching, thereby increasing HC formation. It results in slow rate of oxidation combustion of ethanol compared with diesel, and adiabatic flame temperature was dropped, inhibiting hydrocarbon oxidation. With CNG inducted at full loads, the TMEE10 blend at various IP pressures of 200 bar, 220 bar, 240 bar, and 260 bar was found to be 0.110 g/kWhr, 0.104 g/kWhr, 0.098 g/kWhr, and 0.108 g/kWhr, respectively. The percentage reduction in HC emissions of TMEE10 is 3.5% at 200 bar, 8.7% at 220 bar, 14.0% at 240 bar, and 5.2% at 260 bar for a 20.1% CNG energy share. The increased heat energy released during combustion, which raises the temperature inside the cylinder and exponentially accelerates the pace of reaction, leading to improved oxidation, is largely responsible for the noticeable drop in hydrocarbon emissions [27].

3.2.3. NOx Emission. NOx formation during CNG induction is shown in Figure 7 for a variation of TMEE10 injection pressures. NOx of diesel, TME, and blend TMEE10 in conventional engines was found to be 4.51 g/kWh, 5.26 g/ kWh, and 4.94 g/kWh, respectively. Many aspects such as high cylinder temperature, oxygen availability, heat transfer behaviour, and quality of the fuel impact NO formation [28]. The result clearly showed that NOx emission is 14.3% higher for TME than that for diesel. A higher air-fuel ratio with a decreasing degree of unsaturation of TME is associated with features such as the enhanced adiabatic flame temperature, which tends to promote better combustion, subsequently releasing more heat energy than diesel, followed by TMEE10. In addition, the oxidation of molecular nitrogen in the after flame and the flame zone is considered to be the main mechanism for NO formation. Reduced NOx formation occurs as a result of the lower flame temperature and quenching zone after the addition of ethanol to TMEE10. Furthermore, NOx emission increased when the injection pressure was increased from the standard IP. The TMEE10 blend at injection pressures of 200 bar, 220 bar, 240 bar, and 260 bar was determined to be 4.73 g/kWhr, 5.08 g/kWhr, 5.40 g/kWhr, and 4.94 g/kWhr, respectively, when CNG was inducted at full loads. An increase in NOx emission is due to an increase in the quality of combustion of fuel at a higher pressure of injection. Inducing CNG increases the overall cylinder temperature. This leads to high peak temperature of combustion, which leads to an increase in NOx emission [29]. However, the formation of NO in this way is governed by the concentration of H and O atoms in a combustion environment up to 1800 K, after which thermal NO formation predominates. CNG-TMEE10 combustion results in higher NO emissions than neat TMEE10 combustion because CNG contributes more energy to reaction and produces leaner mixture.

3.2.4. Smoke Opacity. The influence of CNG gas at varying injection pressures on smoke emission formation is graphically depicted in Figure 8. The percentage of smoke opacity is 60, 54, and 57% for diesel, TME, and TMEE10, respectively. A lack of atomization and air entrainment results in smoke formation when test fuels are burned improperly [30–34]. Additionally, the fuel-air ratio rises as the amount of fuel injected increases, especially when using diesel fuel. Thus, in the absence of sufficient oxygen, fuel droplets in the spray domain are more likely to undergo thermal cracking to low boiling components of carbon, leading to substantial smoke generation in the diffusive combustion stage. Moreover, TME without the ethanol blend showed less smoke formation for all load conditions due to the cooling effect of high LHV of ethanol suppressing the postcombustion temperature, thereby resulting in inferior combustion. As a result of the cooling effect of the high LHV of ethanol suppressing the postcombustion temperature, TME without the ethanol blend demonstrated more



FIGURE 5: Varying braking power affects CO formation in the engine.

smoke generation for all load conditions. Smoke emission decreased on increasing the injection pressure of TMEE10. From the results, it was observed that an IP of 240 bar showed minimum smoke emission [35-38]. When compared to an IP of 200 bar, 220 bar, and 260 bar, smoke emission of an IP of 240 bar dropped by 11.3%, 6%, and 9.6%, respectively. This may be attributed to the proper collaboration of fuel and air that enhanced the combustion process. However, it was observed that when the injection pressure increased beyond 240 bar, there was a slight increase in smoke emission compared to the standard operation of 220 bar. Increasing spray angle while increasing injection pressure which affects the penetration of mixture to the combustion chamber was the reason behind this [25]. When the injection pressure was decreased to 200 bar, smoke emission increased when compared to the standard IP of 220 bar. This is because of the presence of larger droplets in the combustion chamber.

3.3. Combustion Characteristics

3.3.1. Cylinder Pressure. The impact of CNG gas at various injection pressures on cylinder pressure generation is depicted graphically in Figure 9. With an increase in IP from 220 bar to 240 bar, the peak pressure inside the cylinder simultaneously increased. Diesel at CA 10aTDC had a maximum CP of 65.6 bar, while TME at CA 7aTDC peaked at 60.3 bar. The maximum cylinder pressure (CP) was recorded at CA 14aTDC for TMEE IP 200 bar, CA 12aTDC

for TMEE IP 220 bar, CA 10aTDC for TMEE IP 240 bar, and CA 11aTDC for TMEE IP 260 bar. CNG with the TMEE10 blend at IP 200, 220, 240 and 260 bar increased the peak pressure by 7.7%, 15.7%, 17.7%, and 13.4% when compared with the TMEE10 blend operated at 220 bar. An in-cylinder peak pressure of 240 bar was increased by 10.8%, 2.3%, and 4.9% compared to an IP of 200 bar, 220 bar, and 260 bar, respectively. The peak pressure was 8.6% lower at an IP of 200 bar than at an IP of 220 bar. This was because of a lower injection pressure, which resulted in poorer air-fuel combination, which in turn reduced combustion efficiency, brake thermal efficiency, and peak pressure. When the IP was raised to 240 bar, the cylinder's maximum pressure also rose. This occurred because a more uniform combination of fuel and air was formed at higher IP (240 bar vs. 260 bar), resulting in a higher peak pressure [23–25].

3.3.2. Heat Release Rate. The effect of CNG gas at varying injection pressures on HRR generation is graphically depicted in Figure 10. The highest HRR for diesel at CA 1°bTDC was 77.7 J/CA, whereas the maximum CP for TME at CA 4°bTDC was 68.3 bar. The TMEE10 blend at injection pressures of 200 bar, 220 bar, 240 bar, and 260 bar was determined to be 62.8, 69.9, 71.3, and 68.3 J/CA respectively, when CNG was inducted at full loads. The maximum HRR was recorded at CA 1°bTDC for TMEE IP 200 bar, CA 2°bTDC for TMEE IP 220 bar, CA 4°bTDC for TMEE IP 240 bar, and CA 3°bTDC for TMEE IP 260 bar. HRR of CNG with the TMEE10 blend at IP 200 bar, 220 bar, 240 bar, and



FIGURE 6: Varying braking power affects HC formation in the engine.



FIGURE 7: Varying braking power affects NOx formation in the engine.



FIGURE 8: Varying braking power affects smoke formation in the engine.



FIGURE 9: The variation of the cylinder pressure with respect to the crank angle.



FIGURE 10: The variation of HRR with respect to the crank angle.

260 bar improved by 7.6%, 16.9%, 18.9%, and 15.6%, respectively, when compared to the TMEE10 blend at 220 bar. HRR reduced by 9.7% when the IP was lowered from 220 to 200 bar. This was possibly due to the big droplet size of the fuel, which impacted the mixture's combustion. Increasing the IP from 220 bar to 240 bar enhanced the HRR by 2.4%. As the length of the flame increased, proper diffusion of mixture and complete combustion occurred. In addition, as the IP increased from 240 bar to 260 bar, HRR decreased by 3.8%. As a result of the reduced ignition delay, more mixture burned during the diffusion phase [39–41].

4. Conclusion

In the current investigation, TME was used as the primary energy source and CNG as the secondary energy source for dual-fuel operation. The effects of increasing the TME injection pressure from 200 bar to 260 bar on the characteristics of a dual-fuel engine were studied. To optimize the performance of the dual-fuel engine and reduce emissions, compressed natural gas (CNG) was drawn into the inlet alongside air. Additionally, NOx emissions were reduced, and premature fuel ignition was prevented by mixing 10% ethanol with TME biodiesel. Therefore, the following conclusions are drawn based on the experimental work:

(i) At the standard mode of operation, tamanu methyl ester produces a lower brake thermal efficiency than diesel fuel. This is primarily due to the biodiesel's poor mixture formation and low calorific value. The blend of TMEE10 in conventional engines produced least BTE, but it was enhanced with a CNG energy share. Moreover, the BTE of the engine was increased while increasing the injection pressure from standard conditions. With an increase in the injection pressure up to 240 bar, BTE increased and BSEC decreased. Although TMEE10 is adequate for high loads, a reduction in the CNG energy share ratio results in smooth engine operation.

- (ii) Neat TME produces lower CO emission than other test fuels in the CI mode. Addition of ethanol to TMEE10 increased CO emission because of a higher latent heat of vaporization of ethanol suppressing the cylinder temperature. Furthermore, the CNG energy share leads to a decrease in CO emission due to high-flame velocity and the flammability limit of CNG.
- (iii) HC emissions of TME were drastically decreased when compared to diesel and TMEE10 in standard engines. HC emission decreased at the higher rate of TMEE10 injection. It is 0.110 g/kWh for 200 bar, 0.104 g/kWh for 220 bar, 0.098 g/kWh for 240 bar, and 0.108 g/kW for 260 bar of IP TMEE10 at a rate of CNG induction 0.17 kg/hr. It was mainly due to better atomization and a high rate of air utilization, which reduce HC formation.
- (iv) NOx emission of normal engines increased when TME was used in place of diesel, but by adding ethanol to TME, it dropped. TMEE10 showed least

NOx emission about 4.94 g/kWh in the CI mode. Further increase in NOx emission was observed when CNG was inducted in TMEE10.

- (v) In the dual-fuel mode, the TMEE10 blend at an IP of 240 bar produced 12.2%, 6.0%, and 8.5% more NOx than at an IP of 200, 220, and 260 bar, respectively. Diesel has an average smoke opacity of 65%, but TME also reaches 57% under the same conditions. TMEE10 produced the least amount of smoke while operating in the dual-fuel mode, with a CNG induction rate of 0.17 kg/hr.
- (vi) Inducting CNG with TMEE10 increased the overall cylinder temperature, which leads to the enhanced cylinder pressure and HRR at higher loads. At dual-fuel operation, CP and HRR increase with an increase in an injection pressure of 200 bar to 240 bar. The highest combustion pressure for IP 240 bar was about 63.2 bar at 15aTDC. For IP 240 bar, the maximum HRR was about 71.3 J/° CA at a CA of 1°bTDC.

Based on the results obtained, it was concluded that an injection pressure of 240 bar was the best while using TMEE10 with a 21% energy share of CNG. The effects of third-generation biodiesel (algal oil) on the performance and knock limit of CI engines powered by various gaseous fuels such as hydrogen and acetylene can be investigated in a more in-depth study. Improved NOx emission reduction can be achieved by utilizing a mixture of lower alcohols and higher alcohols (propanol and 6b2 hexanol) in the dual-fuel mode, as investigated in this study. Hence, a blend of tamanu and ethanol would be an effective alternative for diesel fuel in the CI engine.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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