# Bioethanol Injection and Air Preheating on the Performance and Exhaust Emissions of a CRDI Diesel Engine

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***Abstract*—Alternative fuels for diesel engines are being researched to reduce diesel consumption and exhaust pollution. As a result, the study examines the performance and exhaust emissions of a Common-Rail Direct Injection (CRDI) diesel engine when running on diesel combined with bioethanol injection and air preheating at a constant speed and load. The injection of bioethanol and preheated air were increased from 10 to 50 ms and 50 to 60 °C, respectively. Diesel paired with bioethanol injection at 30ms and air preheating at 60 °C improved engine performance compared to diesel-only mode, increasing brake thermal efficiency and diesel savings by 3 % and 15 %, respectively. This scenario significantly reduced carbon dioxide, black smoke, particulate matter, and nitrogen oxide emissions by 3 %, 2 %, 3 %, and 34 %, respectively. Carbon monoxide was only added in 12 %.**

***Keywords*—bioethanol, CRDI diesel engine, diesel, engine performance, exhaust emissions**

## I. INTRODUCTION

In recent decades, due to their great power and efficiency, diesel engines have been used for the majority of industrial and transportation power generation. They release Carbon Dioxide ($CO_2$) and Nitrogen Oxide (NO), which cause global warming and environmental harm, and they emit copious amounts of Carbon Monoxide (CO) and Black Smoke (BS), which Produce Particulate Matter (PM) particles. Biofuels are sustainable fuels derived from biological sources (plants, algae, or animal fat), waste oils (used cooking or plastic oil), and bioalcohols (microorganisms and enzymes fermenting waste sugars, starches, or cellulose). They are always researching ways to reduce engine emissions. The biological oils are mostly generated as biodiesel, which can be used as a standalone fuel or combined with diesel. The neat biodiesel and its blends can replace fossil fuel, but engine performance suffers. With increased biodiesel, Brake Thermal Efficiency (BTE) falls but Brake Specific Fuel Consumption (BSFC) increases. CO, BS, and PM emissions are reduced, although $CO_2$ and NO emissions vary depending on the feedstock [1–4]. Later, spent frying and waste plastic oils are examined to reduce harmful waste and greenhouse gas emissions. Used cooking oils are converted into biodiesel, while waste plastic oils are created by the pyrolysis of plastic trash. They are primarily blended with diesel because of their higher viscosity. However, mixing both oils with diesel reduces engine performance and increases CO, BS, and PM emissions proportionally to the increase in both volumes [4]. Furthermore, the addition of nanoparticles and graphene quantum dot additions to waste plastic oil improves engine performance and exhaust emissions [4, 5].

However, biodiesel made from biological sources and waste oils has poorer fuel characteristics to diesel. Other additions have been investigated for improving the physical qualities of blended fuels. Petrochemical alcohol, often known as alcohol, is commonly employed as a cosolvent to improve the physical qualities of diesel-biodiesel blends. Five additives, including ethanol, butanol, methanol, propanol, and pentanol, are blended with diesel and biodiesel, improving engine performance and exhaust emissions [6, 7]. Alcohols have several disadvantages, including their high toxicity and risk, their bad influence on the environment during manufacture and transportation, and the production process, which emits greenhouse gases. Bioalcohols have been studied as alternative alcohols since they are created through the pyrolysis of agricultural waste or algal biomass, with common examples including bioethanol, biobutanol, and biomethanol [7, 8]. Because of their low cetane number, poor lubricating qualities, and great resistance to spontaneous ignition, alcohol and bioalcohol cannot be used directly in diesel engines and must be emulsified or fumigated [7–9]. Previous research [6–12] investigated diesel-biodiesel-alcohol emulsions in a variety of liquid-liquid ternary phase diagrams as well as in diesel engines with direct injection (DI), turbocharged direct injection (TDI), and common-rail direct injection (CRDI). BTE was greatly improved, but variations in fuel injection pressure (FIP), fuel injection timing (FIT), and BSFC were dependent on the alcohol type. $CO_2$ and NO emissions differed due to the chemical features of alcohols, resulting in variable combustion characteristics. As a result, BS and PM emissions were reduced, while CO emissions increased as alcohol concentrations rose.

Importantly, diesel is not blended with more than 30% alcohol since the separation duration is less than 24 hours [13–15]. Blended fuels cost more than diesel [13], and the addition of alcohols causes wear and tear on fuel injection systems [16]. As a result, alcohol fumigation is an alternative for diesel engines because it is inexpensive and requires no engine modifications. Previous research [16–18] used carburetion techniques on dual-fuel modes, in which the principal fuels (normal diesel and neat biodiesel) were combined with carbureting ethanol, bioethanol, and biobutanol. They were carried out by increasing the main-jet diameters of carburetors. The expanded main-jet size required changes to BTE and BSFC. The diesel savings were greater than 40%. $CO_2$ and CO emissions differed, while NO, BS, and PM levels decreased. The literature [19, 20] reported the use of neat biodiesel and its mixes coupled with carbureting alcohols (ethanol, butanol, methanol, etc.), which resulted in variations in $CO_2$, CO, and NO releases. BS and PM emissions were reduced. Nonetheless, carburetor size, evaporation rate, and manifold alterations are the primary elements that contribute to a variety of complex modification methods and increased expenses. The dual-fuel alcohol injection with primary fuels is investigated by changing FIP and FIT. The alcohol injector is placed through the intake manifold without changing diesel engines. There are two methods for controlling injections: the first is to modify an engine control unit (ECU) to regulate the injection of primary fuel and alcohol. Another alternative is to utilize software for modifying FIP and FIT. Table 1 includes studies of [19–37] with different FIP and FIT. In dual-fuel modes, the injected alcohols were blended with the primary fuels, as opposed to primary fuels alone. The BTE was increased, while the BSFC was adjusted. The $CO_2$, NO, and BS emissions were reduced, while the CO emissions increased. Other researchers [30–32] employed preheated main fuel and air in combination with varying alcohol FITs. The engine characteristic findings were identical to those obtained using FIP and FIT changes. According to the literature [31], preheated ethanol should be less than 70 °C since its boiling point is 78 °C, which causes loud knocking and irregular combustion. The studies of [38, 39] reported the improvements of engine performance and exhaust emissions, found by air preheating lower than 60 °C. Air preheating at an intake manifold was able to reduce the direct heat contact between fuels and heaters to prevent irregular combustion.

To summarize, several researches have concentrated on ethanol injections since they are less expensive than other alcohols [13, 14, 16]. However, the widespread use of ethanol as an alternative fuel has an influence on food prices, environmental degradation, and increased demand for land and water resources for cultivation. Bioethanol is produced by fermenting waste sugars or starches derived from crop wastes such as corn, sugarcane, or wheat using processes such as biomass pretreatment, enzymatic hydrolysis, and recycling ethanol fermentation. The primary benefits of bioethanol include lower greenhouse gas emissions and air pollution from burning biomass waste and transforming agricultural waste into alternative alcohol [9]. Bioethanol has only been studied as an alternative fuel for diesel engines in blended fuels [7, 9, 15]. Furthermore, bioethanol injection, when combined with air preheating in the intake manifold and the addition of FIT for more than 10ms, has only been partially studied in conventional diesel engines [19, 20, 36]. Outlining the research aims and hypotheses, this work will investigate the performance and exhaust emissions of a CRDI engine using dual-fuel diesel, bioethanol injection, and air preheating. Engine tests are conducted at a constant speed and varying loads, with bioethanol injection and air preheating modes compared to standard diesel mode. An electric injector injects bioethanol, which is anhydrous ethanol, every 10 to 50 ms. A cylinder heater linked to an intake manifold preheats the air to 60 °C.

## II. Literature Review

Given the global emphasis on decreasing greenhouse gas emissions and particulate matter from internal combustion engines, research into bioethanol as a renewable, oxygenated fuel additive is urgent and important. Furthermore, the use of intake air preheating has been established as a recognized way for improving combustion efficiency and fuel-air mixing, particularly in dual-fuel modes. The combination of these strategies has the potential to serve as a feasible pathway for improving engine performance while simultaneously reducing hazardous emissions. The goal of this review is to assess the manuscript's scientific merit, methodological rigor, clarity of presentation, and overall contribution to the field of sustainable combustion technologies. The evaluation process is intended to determine the originality of the approach used, the relevance and quality of the experimental data, the robustness of the analysis, and the validity of the findings reached. Table 1 shows prior experiments on injected alcohols in dual fuel modes by altering FIP and FIT. When mixed with primary fuels, the injected alcohols were employed in DI, Reactivity-Controlled Compression Ignition (RCCI), and Homogeneous-Charge Compression Ignition (HCCI) engines at speeds and loads comparable to the main fuel mode. The literature [19, 20] described the use of Diesel (D) and biodiesels derived from raw materials as primary fuels when combined with injectable Alcohols (ethanol (E), Methanol (M), Hexanol (H), Pentanol (PE), and Butanol (Bu) by raising FIP and FIT up to 10 bar and 50ms, respectively. The BTE and BSFC were enhanced. NO and BS emissions were reduced, while CO release was significantly enhanced due to injected alcohols at 50ms, resulting in more incomplete combustion. Many researches [21–26] investigated the principal fuels (D and Moringa Oleifera Biodiesel (MOB)) combined with the addition of FIP from 1 to 7 bar of alcohols (E, M, and H), Gasoline (G), gasoline mixed with 65% E (E65), and gasoline mixed with 85% E (E85). The results of increasing alcohol FIP indicated an improvement in BTE, NO, and BS. However, adding FIP more than 3 bar resulted in a rise in $CO_2$ and CO emissions. The report [27] employed diesel mixed 20% karanja biodiesel (KB20) paired with injected PE at 3 bar in an RCCI diesel engine, resulting in an increase in CO and NO emissions. On the other hand, the literature [28] investigated the injection of Bu up to 200 bar in an HCCI engine, resulting in the addition of BTE and BSFC. $CO_2$, NO, and BS emissions were reduced, while CO emissions increased. Other researchers [29–32] primarily employed dual-fuel diesel and diesel blended with 20% lemongrass biodiesel (LB20) and ethanol infusion by changing FIT and FIP, resulting in variations in engine performance and exhaust pollutants. Furthermore, the dual-fuel diesel combined with rising FIT of ethanol and exhaust gas recirculation (EGR) was explored in the literature [33, 34]. As a result, increasing EGR lowered BTE while decreasing $CO_2$, NO, and BS emissions. Furthermore, the report of [35] employed preheated Cashew Nut Structure (CNSL) oil mixed with changing FIT of ethanol, resulting in decreased engine performance and increased CO emissions. The literature of [36] evaluated the dual-fuel diesel combination with increasing FIT of ethanol and preheating air over 70 °C, resulting in reduced engine performance and increased CO emissions. Finally, the paper [37] used dual-fuel diesel and Jatropha Biodiesel (JB) in combination with changing FIT of

Bu via engine software, demonstrating that higher Bu injection lowered engine performance while increasing $CO_2$ and CO emissions. According to the literature review research, injected alcohols in dual-fuel modes primarily resulted in the addition of BTE and BSFC when compared to primary fuels. Injecting alcohols resulted in much lower NO and BS emissions. The level of CO emission was increased; however, the $CO_2$ release was modified. In previous experimental experiments, numerous researchers focused on DID, RCCI, and HCCI engines. The utilization of CRDI engines with injected alcohols in dual-fuel mode has not yet been reported in terms of performance and emissions. The primary alcohol used in secondary fuel is ethanol. This is because ethanol is an alcohol derived from the fermentation of plant sources. Other alcohols, such as butanol, methanol, and hexanol, are created in a more difficult process, have more carbon atoms, and are more expensive than ethanol [21–24, 29–36]. Furthermore, the ethanol utilized for secondary fuel is primarily plain ethanol, while bioethanol research has yet to be detailed. The addition of FIT in conjunction with hot air is partially seen. As a result, this paper describes the experimental findings of bioethanol injection paired with preheated air and diesel in dual-fuel modes. In contrast to the diesel-only mode, they are powered by a four-cylinder CRDI diesel engine that operates at a steady speed and under varying loads.

Table 1. Tabular summary of published data

| Ref. | Fuels | | Adjustments | Engine/Conditions | Performance and emissions characteristics | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | PF | SF | | | BTE | BSFC | $CO_2$ | CO | NO | BS | PM |
| [19, 20] | D/biodiesels | E/M/H/PE/Bu | Increased FIP and FIT at 10 bar and 50ms | Single- and multi-cylinder DID/ speeds and loads | ↑ | ↑ | - | ↑ | ↓ | ↓ | - |
| [21, 22] | D | E/M | Changes of FIP | Single-cylinder DID/ fixed speed and loads | - | - | ↑ | ↑ | ↓ | - | - |
| [23] | D | E/G | Increased FIP from 2 to 7 bar | Single-cylinder DID/ fixed speed and fixed load | ↑ | ↑ | - | ↑ | ↓ | ↓ | - |
| [24] | D | E/E65/E85 | Increased FIP at 4.5 bar | Four-cylinder DID/various speeds and fixed load | ↑ | - | ↑ | ↑ | ↓ | ↑ | - |
| [25] | D | G | Increased FIP at 1 bar | Single-cylinder DID/ fixed speed and fixed load | ↑ | ↓ | - | - | - | - | - |
| [26] | MOB/D | H | Changes of FIP | Single-cylinder DID/ constant speed and loads | ↑ | ↑ | - | ↑ | ↓ | ↓ | - |
| [27] | KB20 | PE | Increased FIP at 3 bar | Single-cylinder RCCI/ fixed speed and fixed load | - | - | - | ↑ | ↑ | ↓ | - |
| [28] | D | Bu | Increased FIP up to 200 bar | Single-cylinder HCCI/ constant speed and loads | ↑ | ↑ | ↓ | ↑ | ↓ | ↓ | - |
| [29] | D | E/M | Changes of FIP and FIT | Four-cylinder DID/ constant speed and loads | ↓ | - | - | ↑ | ↓ | ↓ | ↓ |
| [30] | D | E | Changes of FIT by engine software | Six-cylinder DID/various speeds and fixed load | ↑ | - | - | - | - | - | - |
| [31] | LB20/D | E | Increased FIT from 1 to 5ms | Single-cylinder DID/ constant speed and loads | ↑ | ↑ | ↓ | ↑ | ↓ | ↓ | - |
| [32] | D | E | Changes of FIT | Single-cylinder DID/ speeds and fixed load | - | - | - | ↑ | - | ↓ | - |
| [33, 34] | D | E | Changes of FIT and EGR flows | Single-cylinder DID/ constant speed and loads | ↓ | - | ↓ | - | ↓ | ↓ | - |
| [35] | Preheated CNSL | E | Changes of FIT | Single-cylinder DID/ fixed speed and loads | ↓ | ↑ | - | ↑ | - | - | - |
| [36] | D | E | Changes of FIT and preheating air | Single-cylinder DID/ constant speed and loads | ↓ | - | ↓ | ↑ | ↓ | - | - |
| [37] | JB/D | Bu | Changes of FIT by engine software | Single-cylinder DID/ fixed speed and loads | ↓ | ↑ | ↑ | ↑ | ↓ | ↓ | - |

**Note:** PF (Primary fuel), SF (Secondary fuel), ↑ (Increase), ↓ (Decrease), and – (N/A)

## III. Materials and Methods

### A. Fuels

Diesel (D) was regular diesel (fossil diesel mixed with 7% biodiesel) purchased at local petrol stations. Bioethanol (BE) was a 99.9% water-free alcohol acquired from Thai ethanol factories. The characteristics of diesel and bioethanol were examined using ASTM procedures, as listed in Table 2.

Table 2. Physical properties of fuels

| Items | ASTM | D | BE |
|---|---|---|---|
| Chemical compound | - | $C_{7.12}H_{13.73}O_{0.05}$ | $C_{4.32}H_{13.36}O_{2.17}$ |
| Pour point (°C) | D 97 | -8 | N/A |
| Cloud point (°C) | D 2500 | 8 | N/A |
| Flash point (°C) | D 93 | 45 | 175 |
| Boiling point (°C) | D 86 | 283 | 79 |
| Density (kg/m$^3$) | D 1298 | 833 | 792 |
| Kinematic viscosity (mm$^2$/s) | D 445 | 3.2 | 1.4 |
| Heating value (MJ/kg) | D 240 | 45.07 | 28.33 |

The mixture of diesel and bioethanol interacted with air preheating to explain combustion kinetics, emissions, and performance, considering the reactant and product. The findings in [38] provided a theoretical framework for understanding how the reaction occurred in the combustion chamber during the power stroke. The entire reaction was depicted in the following way:

$$[C_xH_yO_z]_D + [C_xH_yO_z]_{BE} + O_2 + N_2 = CO_2 + H_2O + O_2 + N_2 + NO + CO$$

Because diesel fuel did not contain sulfur, it did not emit sulfur oxides in exhaust. Unburned hydrocarbons released by CRDI diesel engines were extremely low [8, 9].

### B. Experimental Setup of Engine Test

Fig. 1 depicts a CRDI diesel engine (Model, TOYOTA: 2KD-FTV; cylinder, 4 cyl; capacity, 2,482 cc; power (max.), 126 kW at 3,600 rpm; compression ratio, 18:1; engine systems, CRDI, turbocharging, and EGR) coupled to a generator (15 kW at 1,500 rpm). Electrical loads were increased, and electrical power was measured using a digital multi-function power meter with a USB converter and a computer. Temperatures of air intake, water coolant, and exhaust gas were measured using K-type thermocouples connected to a multiple channel data logger and displayed on a computer. An air flow meter and a venturi tube were used to monitor air flow rates. Fuel cylinders were attached to load cell sensors to detect bioethanol and diesel flow rates, which were recorded using an Arduino Uno-R3 (AU-R3) and processed on a computer. A heat regulator and a cylinder heater attached to an intake manifold provided preheated air. BE injections utilized an injector and a BE pump connected to an injection controller shown on the computer. An ECU and On-Board Diagnostic II (OBDII) examined the engine positions (speed, FIP, FIT, air-fuel flow, variable-geometry turbocharger (VGT), and so on) in order to record engine operation systems. Finally, $CO_2$, CO, and NO emissions were analyzed using a Cosber KWQ-5 emission analyzer. Black smoke was measured using a Cosber KYD-6 opacimeter by evaluating the intensity and opacity of BS discharge. The uncertainty analysis was set at ±0.05 $m^{-1}$ of black smoke intensity and ±0.20% of opacity.

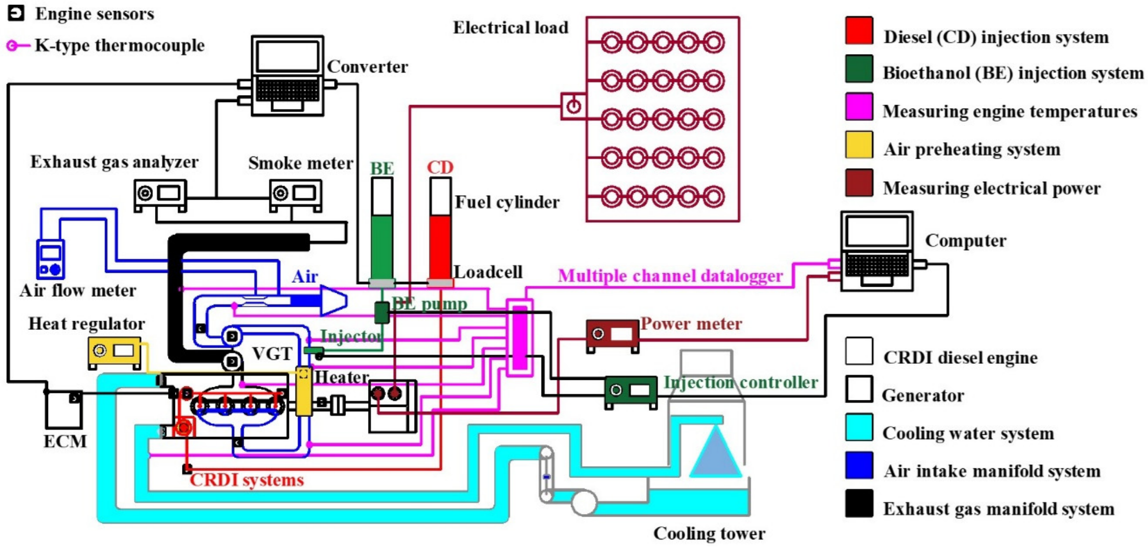


Fig. 1. Schematic diagram of experimental setup.

### C. Scenarios and Experimental Procedures

Scenarios in Table 3 describe studies of dual-fuel diesel mixed with injected bioethanol. Scenarios 1–3 used only diesel with normal air temperature of 30±3 °C and preheated air of 50–60 °C. This is due to a rise in warmed air temperature of greater than 60 °C, which caused engine knock and irregular combustion [36–39]. Scenarios 4–8 used dual-fuel diesel mixed with injected bioethanol, boosting FIT from 10 to 50ms while maintaining FIP at 3 bar. This is because using FIT below 10ms caused very moderate increases in BTE, whereas increasing FIT above 50ms resulted in large CO emissions, according to [19, 20]. In circumstances of controlled injection pressure, the use of FIP greater than 3 bar resulted in an increase in $CO_2$ and CO emissions, as studied in [23, 24, 27]. Furthermore, the use of FIP less than 3 bar resulted in very minimal alterations in BTE, as demonstrated in [23, 25]. The preheated air was kept at 60 °C since higher temperatures combined with bioethanol injection resulted in loud knocking and irregular combustion, which increased engine vibration [36, 38].

The experimental methodologies for a CRDI diesel engine with various modes were as follows. Scenario 1 initially warmed up the engine for around 15 mins. The ECM and OBD-II programs were used to manage the engine speed at 1,500±50 rpm with normal injection timing and main fuel pressure (Diesel). After the engine was stable, Scenario 1 was tested at this speed with various loads, which were modified by raising the electrical load by 20, 40, 60, 80, and 100 %, respectively. Meanwhile, engine parameters and positions (speed, air-fuel flow rates, FIT, FIP, power, sensors, temperatures, and exhaust emissions) were recorded in accordance with the various orders of electrical load as described in [8, 9]. After Scenario 1's investigations were completed, Scenario 2 and Scenario 3 began. Diesel was mixed with air preheating at 50 to 60 °C and evaluated under the same conditions as Scenario 1. Next, the modes 4 through 8 were examined in turn. Scenarios 4–8 used a dual-fuel diesel with bioethanol injection at 10, 20, 30, 40, and 50ms of FIT, respectively. They were run under the identical conditions as in Scenario 1. The weight of diesel was fixed in each mode, while the weight of bioethanol and fuel consumption time were measured using an Arduino Uno-R3 (AU-R3) that was processed on a computer. Finally, ECM and OBDII collected engine parameters and locations in Scenarios 2–8 to compare to Scenario 1 and investigate performance and emission characteristics. To investigate specifics on data collecting and analysis, all engine tests were conducted for 100 hours, and data on engine performance and exhaust emission parameters were repeated more than five times, as studied in [13, 14, 17, 18]. The cumulative uncertainty of instrumentation was determined using published references [37].

Table 3. Scenarios to investigate engine characteristics

| Scenarios | Modes |
|---|---|
| 1 | D used to normal air temperature at 30±3 ºC |
| 2 | D used to preheated air at 50 ºC |
| 3 | D used to preheated air at 60 ºC |
| 4 | D used to FIT of BE at 10ms and preheated air at 60ºC |
| 5 | D used to FIT of BE at 20ms and preheated air at 60ºC |
| 6 | D used to FIT of BE at 30ms and preheated air at 60ºC |
| 7 | D used to FIT of BE at 40ms and preheated air at 60ºC |
| 8 | D used to FIT of BE at 50ms and preheated air at 60ºC |

### D. Methods

Combustion kinetics was studied by the mass fraction of species $i$ ($Y_i$) and the amount of heat removed ($Q_{cv}$) related to the reactant and product standardized enthalpies, called the enthalpy of reaction ($\Delta H_R$). Since the $\Delta H_R$ depended on the temperature chosen for evaluation, the enthalpies of both reactants and products varied with temperatures, leading to changes in various fundamental performance and emissions parameters [40]. They were analyzed from [40] as follows:

$$Y_i = \frac{m_i}{m_{total}} \tag{1}$$

$$m_i = N_i . MW_i \tag{2}$$

$$\Delta H_R \equiv Q_{cv} = H_{prod} - H_{reac} \tag{3}$$

where $m_i$ and $m_{total}$ are mass of species $i$ and $total$, $N_i$ is mole number of species $i$, and $MW_i$ is molecular weight of species $i$. $H_{prod}$ is enthalpy of product; $H_{prod} = \left| \sum N_i.h_i \right|_{\text{prod}}$. $H_{reac}$ is enthalpy of reactant; $H_{reac} = \left| \sum N_i.h_i \right|_{\text{reac}}$. $h_i$ is standardized enthalpy at temperature of species $i$, depended on enthalpy of formation at standard reference state and sensible enthalpy change. The performance and emission parameters were investigated based on the brake thermal efficiency (BTE), the brake specific fuel consumption (BSFC), and the European Vehicle Emissions regulations, as calculated below:

$$BTE = \frac{P_{ele}}{(\dot{m}_D.Q_{HV,D}) + (\dot{m}_{BE}.Q_{HV,BE})} \tag{4}$$

$$BSFC = \frac{\dot{m}_D + \dot{m}_{BE}}{P_{ele}} \tag{5}$$

$$CO_2 \left(\frac{g}{kW-h}\right) = 63.47 \times CO_2\ (\%vol) \tag{6}$$

$$CO \left(\frac{g}{kW-h}\right) = 35.91 \times CO\ (\%vol) \tag{7}$$

$$NO \left(\frac{g}{kW-h}\right) = 6.64 \times 10^{-3} \times NO\ (ppm) \tag{8}$$

$$PM \left(\frac{g}{kW-h}\right) = \frac{C\left(\frac{mg}{m^3}\right) \times 3.6 \times VFR}{P_{ele}} \tag{9}$$

where $P_{ele}$ is electrical power, $\dot{m}_D$ and $\dot{m}_{BE}$ are mass flow rates of diesel and bioethanol, $\dot{Q}_{HV,D}$ and $Q_{HV,BE}$ are heating value of diesel and bioethanol (Table 2). The literature [13] was used to study the universal conversion of $CO_2$, CO, NO, and PM (% v/v or ppm) to BSFC (g/kWh) for European vehicle emissions requirements. $C$ is correlation of filter smoke number to black smoke intensity, and $VFR$ was the volume flow rate of exhaust gases.

## IV. Results and Discussions

### A. Engine Performance

Scenarios using a CRDI diesel engine at 1,500 rpm and electrical loads of 20, 40, 60, 80, and 100% resulted in 1.76±0.07, 3.53±0.06, 5.71±0.02, 6.87±0.04, and 8.50±0.02 kW, respectively. Power measurements were accurate to within ±0.07 kW. The overall uncertainty of experimental performance and emission parameters was found to be ± 4.65%, which is similar to [37]. BTE increased with increasing power, as shown in Fig. 2. BSFC dropped as power was added, as illustrated in Fig. 3. This study discovered the highest BTE and the lowest BSFC at maximum power, which conform to literature [8, 9], due to the highest combustion efficiency and the lowest energy losses, and fuel consumption was properly converted into power output. This condition resulted in the best engine performance, thus variables were compared at this condition. In the instance of BTE, the various modes resulted in changes in BTE for each electrical power. The BTE was added to each electrical power source to warm the air.

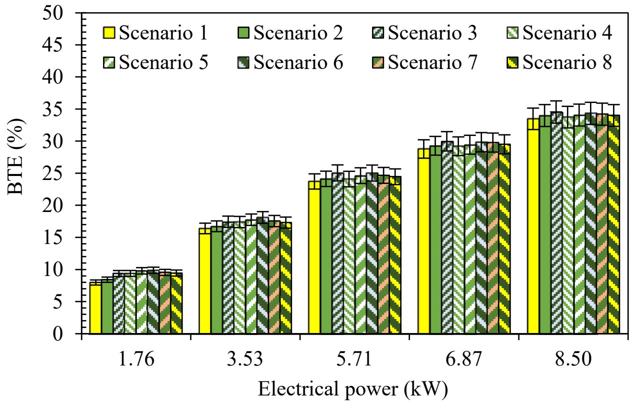


Fig. 2. BTE with increasing electrical power.

When compared to Scenario 1 at maximum power, Scenarios 2–3 increased the BTE by 1.53 and 3.16 %, respectively. These results were consistent with [39], which investigated air preheating alone at 55 °C using a heater coil. Heat input decreased as air temperature increased. Using Scenarios 4–8 in comparison to Scenario 1, dual-fuel diesel along with raising FIT of bioethanol from 10 to 50 ms and adding air temperature resulted in an increase of BTE in each electrical power. Outstandingly, Scenario 6 produced the greatest BTE, increasing by up to 2.54 % at maximum power. Because bioethanol was injected at greater ambient temperatures, it evaporated more quickly. As a result, bioethanol and air were mixed better, resulting in a shorter ignition delay and faster flame propagation velocity in burning zones. The burning rate was rapidly increased, and energy losses were reduced, resulting in a lower energy input to the engine while keeping the same power output [23–26]. Furthermore, these results outperformed those of [32], which examined diesel mixed with ethanol injection alone, raising replacement from 10 % to 20 %.

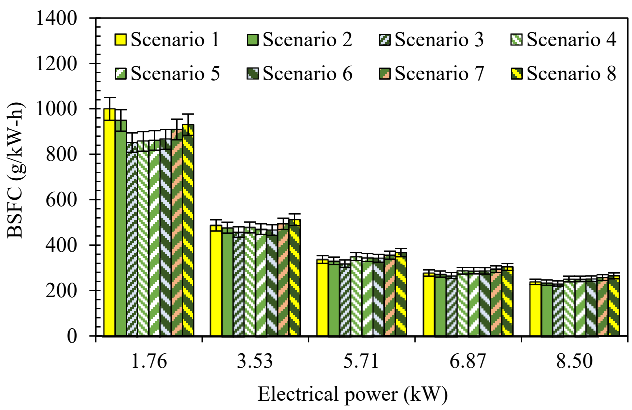


Fig. 3. BSFC with increasing electrical power.

In terms of BSFC, this study discovered a reduction in BSFC by employing preheated air in all power. When compared to Scenario 1 at 8.5 kW of electrical power, Scenarios 2–3 reduced BSFC by 1.51 and 3.06%, respectively. These results were consistent with those reported in [38], which investigated air preheating alone at 48°C using an air preheating system made up of two concentric pipes. The warmed air boosted fuel atomization and reduced fuel consumption while enhancing combustion rates due to correct air-fuel mixing. In Scenarios 4 through 8, the diesel combination with injected bioethanol and preheated air resulted in a steady increase of BSFC at all power levels. Specifically, the BSFC at maximum power increased from 5.19 to 10.98% compared to Scenario 1. These results were congruent with [23], which investigated diesel mixed with ethanol injection by raising the injection pressure from 2 to 7 bar. Because bioethanol has a lower heating value than diesel (Table 2), the flow rate of bioethanol was increased to compensate for the lost diesel volume, resulting in a power output equal to diesel alone.

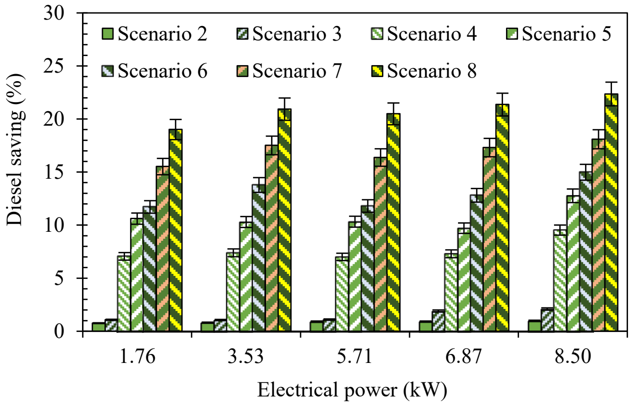


Fig. 4. Diesel saving with increasing electrical power.

Notably, this study discovered that diesel savings were growing across all power sources, as seen in Fig. 4. At maximum power, the best diesel savings were achieved by using Scenarios 4–8, which increased the diesel savings from 9.54 to 22.36 % as compared to Scenario 1. This is because the bioethanol injection was replaced with diesel injection, resulting in lower diesel usage [16].

### B. *Exhaust Emissions*

The principal exhaust pollutants from diesel engines include $CO_2$, CO, NO, BS, and PM, which have an impact on climate change, the environment, and human health. First, excessive $CO_2$ emissions from fossil fuel combustion are creating catastrophic climate change and contributing to global warming. Fig. 5 demonstrates that the release of $CO_2$ increased as electrical power increased. These findings were consistent with [8, 9, 16–18], as full combustion of the air-fuel combination increased with increasing power and BTE. $CO_2$, vapor water, and nitrogen were the only exhaust products produced after complete combustion. However, the usage of different modes resulted in variations in $CO_2$ emissions. The hot air caused the addition of $CO_2$ in all power. At 8.50 kW electrical output, Scenarios 2–3 increased $CO_2$ release from 2.31 to 3.95 % compared to Scenario 1. In this study, air preheating at 50 °C was used, as described in [39]. Preheating the air resulted in higher $CO_2$ emissions because it boosted fuel atomization and vaporization, resulting in faster fuel ignition and complete combustion. In contrast, the dual-fuel diesel combination with injected bioethanol and preheated air resulted in lower $CO_2$ emissions across all power levels, despite the engine compressing air via turbocharging. At maximum power, Scenarios 4–8 reduced $CO_2$ emissions from 1.23 to 6.98 % when compared to Scenario 1. These findings were consistent with those reported in [33, 34], where ethanol injections ranged from 10 to 20 %. This is because the diesel mixed with vaporized bioethanol caused a total fuel flow rate that exceeded the air flow rate, reducing the air/fuel ratio. $CO_2$ levels reduced due to the fuel-rich burning, but CO levels increased.

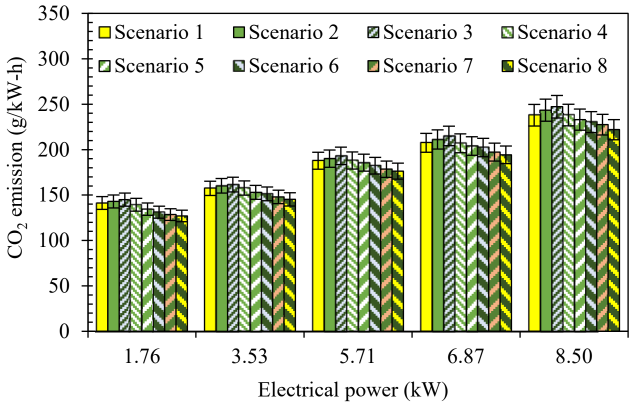


Fig. 5. $CO_2$ emission with increasing electrical power.

The results of $CO_2$ release were consistent with the results of CO emission displayed in Fig. 6. Carbonaceous particulate levels are determined by the results of CO, BS, and PM emissions, and they constitute a health risk to people [8, 9]. Adding power lowered CO emissions (Fig. 6). However, at maximum power, CO emissions rose, as reported in [26, 31]. This is due to higher fuel consumption and lower air-fuel ratio as electrical power increases. More fuel burning occurred at maximum power, resulting in less complete combustion and higher CO emissions. In circumstances of preheated air, CO emissions decreased at all power levels. At 8.50 kW of electrical power, Scenarios 2–3 reduced CO emissions by 5.27 and 12.30% compared to Scenario 1, respectively. These results were similar to [38, 39] because the combustion was more thorough.

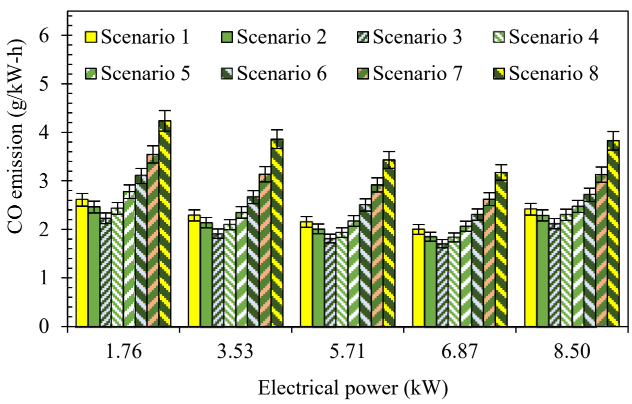


Fig. 6. CO emission with increasing electrical power.

In terms of dual-fuel diesel paired with injected bioethanol and preheated air, this study discovered that Scenario 4 produced less CO than Scenario 1 in each electrical power. At maximum power, Scenario 4 reduced CO emissions by 4.68 %. It is defined as the entire mass of fuel burned with adequate air. As a result, the CO generation process was only minimally affected [19, 20]. Scenarios 5–8 resulted in a large increase in CO emissions, rising from 2.45 to 58.40 % compared to Scenario 1. These findings were consistent with [23], as adding bioethanol infusion boosted fuel-rich combustion despite the engine employing turbocharging air. Fig. 7 demonstrates that BS emissions rose as electrical power increased, owing to higher fuel use at constant air velocity. As a result, incomplete combustion was observed in the combustion zones, notably in the non-premixed combustion zone [8, 9]. The BS emission for preheated air increased with each electrical power. At 8.50 kW electrical power, the preheated air in Scenarios 2–3 increased BS emissions from 0.69 to 1.45 % compared to Scenario 1. These results are explained by the fact that preheated air enhanced fuel-rich combustion in the non-premixed zone, resulting in an increase in BS emissions [13, 19]. Importantly, the dual-fuel diesel combination with injected bioethanol and preheated air in Scenarios 4–8 resulted in a continual reduction of BS emissions in all settings. BS emissions were reduced by 1.09 to 4.17% compared to Scenario 1. These findings were consistent with [23], because the reduction in BS emissions was achieved by decreasing the diesel flow rate while increasing the percentage of bioethanol vaporization. Incomplete combustion in the diffusive combustion zone was eliminated, and BS generation was minimized [31–34].

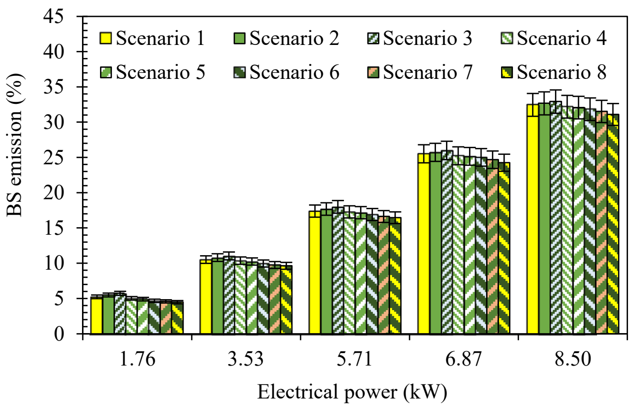


Fig. 7. BS emission with increasing electrical power.

Fig. 8 shows that PM emission rose as electrical power increased. These findings were consistent with previous research [13], as PM emissions were caused by the release of BS (carbon particles) in the diffusive combustion zone. If BS emissions were raised, so were PM emissions. Furthermore, PM emissions were concentrated in the fuel-rich combustion zone when the fuel flow rate increased and the air-fuel ratio decreased in each power output. The usage of different modes resulted in variations in PM emission. In the case of preheated air, Scenarios 2–3 included PM emissions at all power levels. At 8.5 kW of electrical power, PM emissions increased from 1.12 to 2.38 % compared to Scenario 1. They are explained by the increased expansion of incomplete combustion throughout the prolonged duration of the diffusive combustion phase, which is caused by the limiting of preheated air supplied into this zone [19, 31, 37]. In Scenarios 4–8, the dual-fuel diesel was combined with bioethanol injection and air preheating to improve PM emissions. Scenarios 4 to 8 reduced PM emissions from 1.28 to 6.58 % compared to Scenario 1. They are considered to be caused by a decrease in diesel volume with bioethanol substitution, which reduces diffusion-combustion duration and hence reduces PM production as alcohol substitution increases [13, 17]. The results of PM emission were consistent with the results of BS emission. Above all, nitrogen oxides ($NO_x$) are another pollutant produced by diesel engine exhaust that has an impact on the environment. $NO_x$ is the most prevalent nitric oxide (NO) produced in diesel engines, occurring at high oxygen concentration and during flame propagation in the premixed zone [8, 9]. NO emission increased with increasing electrical power, as shown in Fig. 9; the high flame propagation and temperature were caused by more complete combustion due to the faster engine power, resulting in a continuous addition of NO emission as electrical power increased [8, 9, 13, 17]. The usage of modes resulted in variations in NO emission.

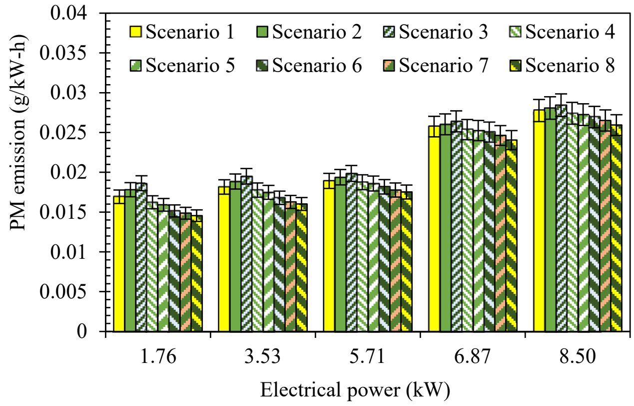


Fig. 8. PM emission with increasing electrical power.

In the case of preheated air, this study discovered a continual reduction in NO emissions in each electrical power. At 8.50 kW electrical power, Scenarios 2–3 reduced NO emissions by 5.35 and 12.49 %, respectively, compared to Scenario 1. These findings differed from the literature [38, 39], as this study studied NO release in a CRDI diesel engine paired with increased exhaust gas recirculation (EGR). As a result, the EGR flow reduced NO generation by lowering the oxygen concentration in the combustion chamber, resulting in fewer NO emissions [27, 33, 34].

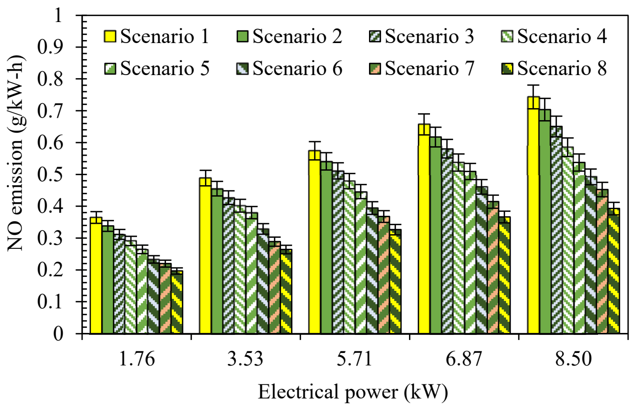


Fig. 9. NO emission with increasing electrical power.

In terms of diesel paired with bioethanol injection and air preheating, NO emissions were lowered in each electrical power. NO emissions from Scenarios 4 to 8 decreased by 21.27 to 47.15 % compared to Scenario 1. These findings were same as [33, 34], as detailed below: First, the bioethanol infusion mixed with preheated air boosted bioethanol vaporization, resulting in fuel-rich combustion. The amount of incomplete combustion in the premixed zone rose, whereas flame propagation and temperature decreased. Next, the inclusion of EGR flow, together with Scenarios 4 to 8, resulted in a limitation of air admission in burning zones, resulting in a constant decrease in flame propagation and temperature. Finally, the NO emission measurements matched the results for $CO_2$ and CO emissions.

### *C. Relationship between Combustion Kinetics, Performance, and Emissions*

This study examined the relationship between combustion kinetics, performance, and emissions at maximum power due to optimal engine performance as shown in Table 4. In cases of combustion kinetics, mass fraction of fuel ($Y_{fuel}$) reduced with increasing air preheating, but increased with increasing air preheating and bioethanol injection. These results were consistent with the changes of mass fraction of air ($Y_{air}$). The fuel-air ratio (FA ratio) was different, corresponding to the changes of mass fraction of fuel and air. The fuel-air ratio from air preheating at 60 °C (Scenario 3) reduced to 0.08% compared to Scenario 1. As a result, there were the escalation of $CO_2$ emission and the reduction of CO and NO emissions due to more complete combustion. However, the BS and PM emissions were increased due to the diffusive combustion zone was dropped because of air limitations during combustion [13, 19].

Table 4. Relationship between combustion kinetics, performance, and emissions at maximum power

| Case | Parameter | Unit | Scenario 1 | Scenario 2 | Scenario 3 | Scenario 4 | Scenario 5 | Scenario 6 | Scenario 7 | Scenario 8 |
|---|---|---|---|---|---|---|---|---|---|---|
| Combustion kinetics | $Y_{fuel}$ | $kg_{fuel}/kg_{total}$ | 0.06451 | 0.06449 | 0.06447 | 0.06811 | 0.06885 | 0.06957 | 0.07060 | 0.07248 |
| | $Y_{air}$ | $kg_{air}/kg_{total}$ | 0.93549 | 0.93551 | 0.93553 | 0.93189 | 0.93115 | 0.93043 | 0.92940 | 0.92752 |
| | FA ratio | $kg_{fuel}/kg_{air}$ | 0.06896 | 0.06894 | 0.06891 | 0.07309 | 0.07394 | 0.07477 | 0.07597 | 0.07814 |
| | $\Delta h_R$ | $MJ/kg_{fuel}$ | 2.289 | 1.986 | 1.762 | 1.670 | 1.654 | 1.632 | 1.626 | 1.656 |
| Performance | BTE | % | 33.49 | 34.00 | 34.54 | 33.75 | 34.08 | 34.34 | 34.21 | 34.01 |
| | BSFC | kg/kW-h | 238.53 | 234.93 | 231.23 | 250.91 | 251.22 | 251.94 | 256.94 | 264.72 |
| Exhaust Emissions | $CO_2$ | g/kW-h | 237.96 | 243.46 | 247.32 | 238.08 | 233.09 | 230.40 | 227.61 | 222.06 |
| | CO | g/kW-h | 2.42 | 2.29 | 2.12 | 2.30 | 2.48 | 2.71 | 3.13 | 3.83 |
| | BS | % | 32.45 | 32.67 | 32.92 | 32.19 | 32.07 | 31.85 | 31.52 | 31.10 |
| | PM | g/kW-h | 0.0278 | 0.0281 | 0.0284 | 0.0274 | 0.0272 | 0.0270 | 0.0265 | 0.0260 |
| | NO | g/kW-h | 0.7436 | 0.7038 | 0.6507 | 0.5854 | 0.5373 | 0.4925 | 0.4527 | 0.3930 |

The use of diesel combined with air preheating and bioethanol injection from Scenario 4 to Scenario 8 increased the fuel-air ratio from 5.99 to 13.31 % compared to Scenario 1. They led to the accretion of CO emission, because the fuel-rich combustion increased [23]. The levels of $CO_2$, BS, PM, and NO were reduced corresponding to the decrease in mass fraction of air, insufficient for combustion reactions. Although the fuel-air ratio increased, the enthalpy of combustion also known as enthalpy of reaction ($\Delta h_R$) was different. In terms of air preheating at 60 °C (Scenario 3), enthalpy of reaction reduced to 23.04 % compared to Scenario 1. This resulted in the amount of heat removed from a combustion chamber decreased, corresponding to the addition of BTE and the reduction of BSFC. Additionally, this resulted in the reduction of CO and NO emissions due to the ideal combustion products for burning were complete with mass fraction of fuel and air used. Outstandingly, the use of diesel combined with increasing air preheating and bioethanol injection resulted in the continuous reduction of enthalpy of reaction. The enthalpy of reaction from using Scenario 4 to Scenario 8 decreased from 27.03 to 28.95 % compared to Scenario 1. These results were consistent with the addition of BTE and the reduction of BSFC, leading to changes of exhaust emissions. In particular, CO emission had increased significantly.

## V. CONCLUSION

The experimental investigations of a CRDI diesel engine running in dual-fuel mode versus diesel-only mode can be summarized as follows:

First, the engine performed best at 8.50 kW of electrical power due to the highest BTE and lowest BSFC. The use of bioethanol injection combined with air preheating in dual-fuel modes enhanced BTE more than the use of diesel alone. The maximum BTE was seen with diesel mixed with bioethanol infusion at 30 ms (Scenario 6). The diesel paired with increased bioethanol infusion resulted in the ongoing accretion of BSFC. This study shown that the addition of bioethanol injection could minimize diesel usage. Diesel savings improved by 22.36% after introducing bioethanol at 50 ms (Scenario 8).

Next, the exhaust emissions from dual-fuel modes were modified. This study discovered that using dual-fuel diesel with increased bioethanol injection and air preheating resulted in a continual reduction in $CO_2$, BS, PM, and NO emissions, but an increase in CO emissions. This is due to the fuel-air ratio and enthalpy of reaction were changed. The maximum bioethanol injection was employed in Scenario 8, which lowered $CO_2$, BS, PM, and NO emissions by 6.98 %, 1.45 %, 6.58 %, and 47.15 % correspondingly. However, it raised $CO_2$ emissions by 58.40 %.

Finally, future research will look into the combustion characteristics of a CRDI diesel engine fed by dual-fuel diesel, bioethanol injection, and air preheating. Furthermore, a study effort on increasing bioethanol injection pressure and timing, as well as increasing preheating air and EGR flow, will be begun.

## CONFLICT OF INTEREST

The authors declare no conflict of interest.

## AUTHOR CONTRIBUTIONS

Pisak Chermprayong is the first author who installed instruments and wrote methodology. Ekkachai Sutheerasak is the corresponding author who studied combustion kinetics, engine performance, and emissions, analyzed data, and wrote the introduction and results and discussions. Worachest Pirompugd and Mattana Santasnachok studied fuel properties. Sathaporn Chuepeng investigated data analysis and wrote conclusion, recommendation, and writing-review & editing draft; all authors had approved the final version.


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## REFERENCES


[1] C. S. Damian, Y. Devarajan, R. Jayabal, and T. Raja, "Enhancing marine diesel engine compatibility with sustainable fuels: Keyfactors and adjustments," *Mar. Pollut. Bull*., vol. 215, 117836, pp. 1–13, June 2025.

[2] S. K. Nayak, and Y. Devarajan, "Evaluating ignition improvers on performance and emissions of Calophyllum inophyllum biodiesel in turbocharged diesel engines," *Results Eng*., vol. 24, 103664, pp. 1–21, December 2024.

[3] Y. Devarajan, and D. C. Selvam, "Utilization of sterculia foetida oil as a sustainable feedstock for biodiesel production: Optimization, performance, and emission analysis," *Results Eng*., vol. 24, 103196, pp. 1–11, December 2024.

[4] V. G. Nguyen, M. T. Pham, N. V. L. Le, H. C. Le, T. H. Truong, and D. N. Cao, "A comprehensive review on the use of biodiesel for diesel engines," *Int. J. Renew. Energy Dev*., vol. 12, no. 4, pp. 720–740, July 2023.

[5] G. Soundararajan, C. Bibin, R. A. Kumar, S. Arunkumar, K. Rajesh, Y. Devarajan, and R. Mishra, "Synergistic effects of graphene quantum dot additives in waste plastic oil blends: Combustion stability and emission reductions analysis," *Results Eng*., vol. 25, 104130, pp. 1–11, March 2025.

[6] D. C. Selvam, and Y. Devarajan, "Performance and emission analysis of sterculia foetida biodiesel enhanced with butanol: Combustion efficiency and emission mitigation," *Results Eng*., vol. 25, 104586, pp. 1–18, March 2025.

[7] M. Q. Chau, V. V. Le, T. H. Le, and V. T. Bui, "A review on the role and impact of typical alcohol additives in controlling emissions from diesel engines," *Inter. J. Renew. En. Dev*., vol. 11, no. 1, pp. 221–236, February 2022.

[8] P. Chermprayong, E. Sutheerasak, W. Pirompugd, and S. Chuepeng, "POE20-biobutanol blends fueled for a CRDI diesel engine as a renewable power generation," *IEEE*, vol. CPESE62584, 10840785, pp. 194–200, January 2025.

[9] M. Santasnachok, C. Chinwanitcharoen, and E. Sutheerasak, "Performance of a CRDI diesel engine generator fueled with b20-bioethanol-ethyl acetate blends," *IOS Press Ebooks: Advan. Trans. Eng*., vol. 54, pp. 351–360, June 2024.

[10] K. Ayyappan, and D. R. Srinivasan, "Effect of fuel injection pressure variations on engine performance-emission-particulate matter with diesel-iso-butanol-nanoparticle fuels in compression ignition engine," *Indian J. Sci. Tech*., vol. 18, no. 5, pp. 313–327, February 2025.

[11] M. D. Tanwar, F. A. Torres, A. M. Alqahtani, P. K. Tanwar, Y. Bhand, and O. Doustdar, "Promising bioalcohols for low-emission vehicles," *Energies*, vol. 16, no. 2, 597, pp. 1–22, January 2023.

[12] F. Zheng, H. M. Cho, "Exploring the effects of synergistic combustion of alcohols and biodiesel on combustion performance and emissions of diesel engines: A review," *Energies*, vol. 17, no. 24, 6274, pp. 1–23, December 2024.

[13] S. Chuepeng, C. Chinwanitcharoen, W. Ruengphrathuengsuka, and E. Sutheerasak, "Performance and emission characteristics of a high-speed diesel engine using a 20% palm oil ester and ethyl alcohol blend," *Int. J. Automot. Mech. Eng*., vol. 21, no. 2, pp. 11372–11385, June 2024.

[14] M. Santasnachok, E. Sutheerasak, C. Chinwanitcharoen, W. Ruengphrathuengsuka, and S. Chuepeng, "Use of diesel mixed with ethanol and ethyl acetate for alternative fuel in a high-speed diesel engine," *ASEAN Eng. J*., vol. 11, no. 2, pp. 25–36, March 2021.

[15] E. Sutheerasak, C. Chinwanitcharoen, and S. Chuepeng, "Comparison between PEE10 and PEE10-bioethanol blends on performance and emission characteristics of a HSDI diesel engine," *Trends Sci*., vol. 18, no. 22, pp. 1–13, November 2021.

[16] E. Sutheerasak, W. Pirompugd, and S. Chuepeng, "Study of diesel combining with vaporizing ethanol via a carburetor for a diesel-engine generator," *IEEE*., vol. CPEEE54404, 9738713, pp. 54–58, February 2022.

[17] E. Sutheerasak, W. Pirompugd, and S. Chuepeng, "The performance and emission of a generator-diesel engine fueled with palm oil methyl ester combined with carbureting biobutanol," *Energy Rep*., vol. 9, no. 3, pp. 210–218, May 2023.

[18] S. Chuepeng, W. Pirompugd, and E. Sutheerasak, "Performance and emissions of a diesel engine fueled with palm oil ethyl ester combined with fumigated ethanol on a dual fuel mode," *Energy Rep*., vol. 9, no. 1, pp. 470–477, March 2023.

[19] A. Jamrozik, and W. Tutak, "Alcohols as biofuel for a diesel engine with blend mode–a review," *Energies*, vol. 17, no. 17, 4516, pp. 1–30, September 2024.

[20] A. Imran, M. Varman, H. H. Masjuki, and M. A. Kalam, "Review on alcohol fumigation on diesel engine: A viable alternative dual fuel technology for satisfactory engine performance and reduction of environment concerning emission," *Renew. Sustain. Energy Rev*., vol. 26, pp. 739–751, October 2013.

[21] H. R. Anand, and M. K. Dubey, "Effects of ethanol fumigation on the performance and emissions of diesel engines," *J. Emer. Tech. Innov. Res*., vol. 11, no. 10, pp. 1–51, October 2024.

[22] T. Ahilan, C. Selvamani, P. Suresh, S. Selvaraj, R. S. M. Malard, A. R. Kumarc, and P. Dhakshnamurthi, "Effects of ethanol fumigation on the performance and emissions of diesel engines," *Phys. Chem. Res*., vol. 12, no. 1, pp. 135–144, March 2024.

[23] M. Tongroon and S. Chuepeng, "Adjacent combustion heat release and emissions over various load ranges in a premixed direct injection diesel engine: A comparison between gasoline and ethanol port injection," *Energy*, vol. 243, no. 122719, pp. 1–13, March 2022.

[24] A. Damyanov, and P. Hofmann, "Operation of a diesel engine with intake manifold alcohol injection," *Automot. Engine Technol*., vol. 4, pp. 17–28, June 2019.

[25] C. Vipavanich, S. Chuepeng, and S. Skullong, "Heat release analysis and thermal efficiency of a single cylinder diesel dual fuel engine with gasoline port injection," *Case Stud. Therm. Eng*., vol. 12, pp. 143–148, September 2018.

[26] R. Pradeepraj, K. Rajan, and S. Nallusamy, "Impact of 1-hexanol fumigation on diesel engine emissions using moringa oleifera biodiesel," *Int. J. Eng. Trends Technol*., vol. 68, no. 11, pp. 150–155, November 2020.

[27] V. R. Sabu, J. J. Thomas, and G. Nagarajan, "Experimental investigation on the effects of multiple injections and EGR on n-pentanol–biodiesel fuelled RCCI engine," *RSC Adv*., vol. 10, no. 49, pp. 29498–29509, August 2020.

[28] S. Kommana, K. M. Babu, and A. H. R. Madhuri, "Performance and emission characteristics of fumigated butanol on a duel fuel mode HCCI engine," *E3S Web Conf*., vol. 309, 01228, pp. 1–7, October 2021.

[29] Z. H. Zhang, K. S. Tsang, C. S. Cheung, T. L. Chan, and C. D. Yao, "Effect of fumigation methanol and ethanol on the gaseous and particulate emissions of a direct-injection diesel engine," *Atmos. Environ*., vol. 45, no. 11, pp. 2001–2008, April 2011.

[30] K. N. Trung, and H. H. Tuan, "Effects of ethanol port injection timing and delivery rate on combustion characteristic of a heavy-duty V12 diesel engine," *Therm. Sci*., vol. 26, pp. 137–137, January 2021.

[31] M. Vijayakumar, and P. C. M. Kumar, "Performance and emission characteristics of compression-ignition engine handling biodiesel blends with electronic fumigation," *Heliyon*, vol. 5, no. 4, e01480, pp. 1–17, April 2019.

[32] M. Mohamed, and E. G. El-Hagar, "Exhaust emissions of a single cylinder diesel engine with addition of ethanol," *Therm. Eng*., vol. 72, pp. 25230–25233, June 2014.

[33] G. S. Hebbar, and A. K. Bhat, "Control of NOx from a DI diesel engine with hot EGR and ethanol fumigation: An experimental investigation," *Int. J. Automot. Technol*., vol. 14, no. 3, pp. 333–341, May 2013.

[34] G. S. Hebbar, and A. K. Bhat, "Diesel emission control by hot EGR and ethanol fumigation; an experimental investigation," *Int. J. Mod. Eng. Res. Technol.*, vol. 2, no. 4, pp. 1486–1491, July 2012.

[35] J. Ravikumar, D. C. Selvam, R. Devanathan, and S. Kannan, "Performance and emission analysis of a CI Engine fuelled with preheated CNSL oil with diesel blends and ethanol fumigation," *Inter. J. App. Eng. Res*., vol. 10, no. 10, pp. 2382–42387, January 2015.

[36] A. Pannirselvam, M. Ramajayam, V. Gurumani, S. Arulselvan, and G. Karthikeyan, "Experimental studies on the performance and emission characteristics of an ethanol fumigated diesel engine," *Inter. J. Eng. Res. App*., vol. 2, no. 2, pp. 1519–1527, January 2012.

[37] S. E. Gillani, M. Ikhlaq, M. U. Khan, D. Awan, and A. Hamza, "Effects of butanol blending and fumigation with Jatropha biodiesel on

combustion, performance, and emissions of diesel engine," *Int. J. Environ. Sci. Technol*., vol. 18, pp. 819–834, August 2020.
[38] M. S. H. K. Tushar, M. I. Mannan, and A. Azmain, "Exhaust gas after treatment using air preheating and selective catalytic reduction by urea to reduce NOx in diesel engine," *Heliyon*, vol. 11, no. e42399, pp. 1–11, February 2025.
[39] R. Vijayakumar, T. G. Sakthivel, T. Rajendiran, and S. L. Praveen, "Performance emission characteristics of air preheated diesel engine," *Australian J. Bas. App. Sci*., vol. 10, no. 1, pp. 117–123, January 2016.
[40] S. R. Turns, *An Introduction to Combustion: Concepts and Applications*, 3rd ed. New York: McGraw-Hill, ch. 2, 2012.