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ARTICLE

Influence of Ethanol-Diesel Blends on Pollutant Emissions and Engine Behavior in Compression Ignition Engines

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ABSTRACT: In response to the current energy crisis and escalating concerns about pollution, research efforts have increasingly focused on reducing fuel consumption and minimizing the emission of toxic substances from combustion processes. This has led to the exploration of non-petroleum, renewable, sustainable, and environmentally friendly fuels, unlike conventional energy sources such as natural gas, oil, and coal, which are non-renewable. Renewable and sustainable resources can be used to produce alcohol, such as ethanol. This study compares the combustion and emission characteristics of neat diesel fuel with those of ethanol-diesel fuel blends. Experiments were conducted under steady-state conditions using a four-cylinder, direct-injection diesel engine operating at various load and 1500 rpm. The results showed that blending ethanol with Diesel at 10%, 15%, 20%, and 25% significantly reduced carbon monoxide emissions by 10.06%, 20.89%, 40.34%, and 49.13%, respectively. For unburned hydrocarbons (HC), the reduction rates were 6.3%, 14.86%, 19.70%, and 28.62%, while the nitrogen oxides (NOx) reduction rates were 19.73%, 10.53%, 17.65%, and 12.69%, respectively, compared to Diesel. However, due to ethanol's lower calorific value, ethanol-diesel blends increased specific fuel consumption by 12%, 8.9%, 5.4%, and 2%, respectively, when the engine was operated at the optimal injection timing for each fuel type. When the engine was operated at the optimal diesel timing, the BSFC was reduced by 2%, 5.4%, 8.9% and 12%, compared to diesel, respectively. These results suggest that ethanol can be efficiently blended with diesel and used in existing engines without modifications.

KEYWORDS: Compression ignition engines; cetane number; diesel-ethanol solution; oxygenate; noise

1 Introduction

Now, humanity faces the twofold challenge of exhausting fossil fuel resources and worsening environmental issues. Years of extensive exploitation and use of traditional energy resources have exhausted underground carbon deposits and contributed to climate change and air pollution. Consequently, the discovery of alternative fuels that can meet sustainability, energy-saving, and environmental preservation requirements has become a matter of great urgency [1,2].

A renewable, eco-friendly substance, ethanol could reduce the environmental impacts of transportation. While fossil fuels must be used as is, ethanol could be made from various renewable feedstocks and can improve how energy is combusted and its environmental impact [3]. Studies have shown that adding ethanol

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to diesel can help reduce CO, NOx, and particulate matter emissions, as well as enhance combustion in compression-ignition engines. Moreover, the addition of ethanol not only alters the nature of emissions but also alters the combustion pattern of compression ignition engines (CIE), potentially changing their performance [4]. For example, although ethanol can improve combustion efficiency, it may also increase ignition delay and shorten combustion duration, potentially negatively affecting engine performance [5]. At higher ethanol blending ratios, there is a likelihood of increased NOx emissions; therefore, it is imperative to balance ethanol ratios to regulate emissions without compromising engine performance. Such complexity underscores the importance of learning more about the combustion and emission properties of ethanol-diesel mixtures, as enhancing these aspects could make diesel engines more environmentally friendly in the future [6]. The use of ethanol-diesel blends affects the engine's performance and reduces pollutant emissions. Across all engine loads, E10 produced the least CO, whereas at 75% load, E20 produced the most NO. The engines had different performance, with E10 having the best brake thermal efficiency and E20 the worst.

All in all, CO_2 and unburnt hydrocarbon emissions were similar across the tested blends [7]. Research by Shadidi et al. [8] showed that blending ethanol with diesel reduced CO and HC emissions by 3.2%–30.6% and 7.01%–16.25%, respectively, due to ethanol's high oxygen content. Nevertheless, NOx increased by 7.5%–19.6% because higher combustion chamber temperatures improved combustion. It was found that increasing engine speed and mixing a ten percent ethanol blend led to better performance and less pollution. Jamrozik and Tutak [9] stated that using ethanol-diesel blends in compression ignition engines can help with combustion efficiency; however, they point out that this may cause a decrease in engine output and an increase in toxic pollutant emissions known as HC and NOx. Despite their promise in lessening carbon monoxide (CO) and particulate matter (PM) emissions, it is quite difficult to find the right blending of fuel since its outcome depends on both the condition and settings of the engine.

However, there are many difficulties in adapting diesel engines to run on ethanol. Because ethanol has a lower cetane number, it may take longer for the mixture to ignite; therefore, to remain stable, ethanol is frequently blended with diesel using emulsifiers or cetane improvers [10]. Also, problems with materials that bond well and lubricity should be fixed to keep the engine safe and operating smoothly. In spite of these issues, research finds that only a little tuning to engines helps them handle ethanol-diesel mixtures, so this way of using biofuels is both practical and relatively cost-efficient [11].

Research has proven that blending ethanol and diesel can lower harmful emissions and, at the same time, keep or increase the performance of some engines. Results from numerous tests in multi-cylinder diesel engines have shown that NOx and particle emissions were decreased, while there was a rise in combustion consistency, and only a little more fuel was needed [12–14]. The studies underline the fact that ethanol could be used as a good alternative or additive to diesel fuel in today's diesel engines.

According to the study by Ezhumalai and Govindasamy [15], increasing ethanol results in slower engine startup due to delays and can also reduce efficiency, but properly selecting the injection timing can mitigate these problems. According to de Souza et al. [16], the principal benefits of using ethanol-diesel blends in diesel engines include less NOx coming from the engine, less smoke, better fuel combustion at full power.

- The cleaner burning of ethanol-diesel fuel helps improve air quality by reducing the emissions typically caused by diesel engine. Using ethanol as a fuel result in cleaner combustion, which also means the environment is less polluted. Based on these points, this work will share more details about these benefits [17].
- Ref. [18] indicates that emissions of nitrogen oxides, carbon monoxide, and particulate matter decreased by 30%, 9%, and 20%, respectively. The fuel used contains ethylhexanol (specifically added as a stabilizer/solvent to prevent phase separation between ethanol and diesel). It also contains a cetane number enhancer, 2-ethylhexyl nitrate (2-EHN).

- Because both CO and HC play a big role in polluting city air and causing health problems, this reduction is very important [19].
- Under certain operating conditions, ethanol in the fuel may increase NOx emissions. However, the toxicity of the resulting exhaust gases depends on ethanol concentration and other factors [20].
- Although ethanol blends could increase emissions, they still improve air quality by reducing levels of PM, CO, and HC [21].

The novelty of this study lies in (a) the simultaneous evaluation of emissions, engine noise, incomplete combustion limits, and performance in a direct-injection, multi-cylinder diesel engine using a diesel-ethanol mixture, considering both a fixed optimum diesel injection timing and individual injection timing optimization for each mixture; and (b) the integrated systematic evaluation of the diesel-ethanol mixture under both a fixed optimum diesel injection timing and individual injection timing optimization for each mixture, considering performance, incomplete combustion limits, emissions, and engine noise under steady-state operating conditions at full load. This work quantifies the effect of adding ethanol up to the practical incomplete combustion threshold (25% by volume) on the optimal injection timing window, and how this window affects specific brake fuel consumption, brake thermal efficiency, pollutant composition (carbon monoxide, hydrocarbons, nitrogen oxides, carbon dioxide, and exhaust gas temperature), and acoustic behavior. This work includes mapping incomplete combustion zones and measuring noise, as well as a detailed characterization of fuel properties, including heat of vaporization, surface tension, viscosity, air-fuel stoichiometry, and cetane number, to provide a comprehensive picture of the thermal and physical effects on diesel and ethanol combustion—a rarely discussed topic. Therefore, for existing engines, this work not only describes the emissions trends of a given ethanol-diesel blend but also establishes practical limits for fuel mixing and injection tuning to operate an existing ethanol-diesel engine. These limits can be directly applied to achieve maximum environmental benefit without increasing incomplete combustion, or increasing engine noise.

The effect of ethanol–diesel blends on emissions of CO and HC is known to result in decreases in their concentrations, while the influence on NOx and efficiency greatly depends on the ratio of the mixture and its combustion conditions; the current research explores the influence of ethanol–diesel blends using constant and optimum injection timings in diesel engines.

It is pertinent to note that the present work is limited to experiments conducted on a single multi-cylinder, steady-state, laboratory-scale diesel engine. The results are therefore directly applicable to engines of similar size, technology, and calibration, but extrapolation to large and long-scale/long-term applications should be done with caution. Transient operation, cold-start behavior, long-term durability, injector and pump wear, fuel storage stability, and economic considerations of ethanol supply and additive usage are all factors that can impact scale-up to actual fleets and duty cycles. The present results, therefore, offer a consistent physical context and trends for diesel-ethanol blends, and full-scale implementation will require complementary studies of durability and transient response, aftertreatment interactions and life-cycle cost and emissions performance under real-world operating conditions.

In this respect, the current study explores the dynamics of various ethanol and diesel blends and the emissions formed when they are being combusted. Expert selection of these ratios was based on observing that they blended well and remained stable in diesel fuel, enhanced with special additives. The research team is conducting structured experiments to better understand ethanol's role in developing clean, environmentally friendly transport fuels.

2 Methods and Materials

2.1 Materials

The solvent can be a non-ionic surfactant or a commercially available auxiliary solvent. In Iraq, commonly used nonionic surfactants for improving the miscibility of diesel and alcohols, such as Span 80 (sorbitan monooleate, C24H44O6) and Tween 80 (polyoxyethylene sorbitan monooleate), are readily available. Span 80 is a sorbitan ester produced by esterification of sorbitan with oleic acid, while Tween 80 is obtained by ethoxylation of sorbitan monooleate, resulting in a more hydrophilic polysorbate surfactant. They are compatible with various hydrocarbon fuels and require low dosage. In this study, experiments were conducted using diesel fuel and 99.7% purity ethanol. diesel, ethanol, solvent, and cetane number improver were mixed to prepare an ethanol-diesel blend. Initially, 1.5% v/v of cetane number improver and solvent were added to the ethanol. Span 80, manufactured by Haihang Industries (China), was used. This substance is characterized by its oily, liquid appearance, ranging in color from yellow to amber. Its pH is ≤8.0, its saponification value (mg/g) ranges from 145 to 180, its hydroxyl content (mg/g) ranges from 193 to 210, and its water content is ≤2.0%. The residue after combustion is ≤0.25%, and the heavy metal content is <0.001%. Also, in this study, Iso-octyl nitrate and 2-ethylhexyl nitrate (2-EHN), are generally the most convenient and common additives to improve the cetane number of diesel fuels. This substance is produced by Green Industry (China). This material has a boiling point of 210.9 ± 8.0°C at 760 mmHg, a density of 0.963 g/mL, a refractive index of 1.432, and a moisture content of ≤450 mg/kg. It exhibits a corrosion resistance of ≤1 for copper (50°C, 3 h), a flow viscosity of 1.700–1.800 mm^2/s (at 20°C), and an acidity of ≤3 mg potassium hydroxide (KOH)/100 mL. This product is a cetane number improver for diesel, which reduces the ignition delay time and makes efficient combustion of low-quality diesel fuel by improving issues like poor combustion, black smoke production, and low engine power output. It is an intermediate compound in organic synthesis for the manufacturing of nitrogen-containing organic molecules, like pharmaceuticals and dye intermediates.

Cetane numbers presented in Table 1 have been measured and have not been calculated. Each neat diesel, neat ethanol, and ethanol–diesel blend was tested for CN using the standard cetane number test method (ASTM D613), and the results are the average of repeated tests conducted under identical test conditions. Neat ethanol has a low CN, as it has an intrinsically low ability to auto-ignite, while the CN of the ethanol–diesel blends decrease as the ethanol percentage increases because ethanol has lower auto-ignition ability than diesel. The mix of CN, however, is higher than that of neat ethanol due to the fuels still being diesel-biased and the cetane improver added at fuel preparation time aiding in the ignition quality. The measured CN values in Table 1 are then assumed to represent the actual ignition properties of the fuels under the test conditions used. The ASTM D613 CFR engine was used as the standard for measuring the cetane number of each blend, with a measurement certainty of ±0.98.

After that, the diesel and premixed solutions were thoroughly mixed to form a diesel fuel blend with no compositional differences. The physical characteristics of the fuel, as shown in Table 1, have been found through laboratory methods. The fuel properties listed in Table 1 were measured using standard analytical methods that are typically used for liquid fuels. The cetane number has been experimentally determined according to the appropriate diesel fuel cetane-number method (e.g., ASTM D613/ISO 5165), and the reported value is the average of repeated tests conducted under the same test conditions. Density was determined gravimetrically or by a density meter, according to ASTM D1298/ASTM D4052 and viscosity was measured by a calibrated viscometer, according to ASTM D445. Flash point was determined using either the closed-cup method, ASTM D93, or ASTM D3828, whichever was available. The measured fuel composition of the various fuels (diesel, ethanol, and their combination blends) was used to determine the oxygen content and carbon content of each fuel using compositional or elemental analysis and to determine the lower heating

value of each fuel using the measured fuel properties and the known blend composition. The uncertainty of each reported property was calculated based on the measurement repeatability, calibration limits, and instrument accuracy. Thus, the properties shown in Table 1 are experimental and obtained under identical laboratory conditions.

Table 1: Fuel properties for diesel and ethanol.

Specification	Diesel	Ethanol	10% Ethanol-Diesel	15% Ethanol-Diesel	20% Ethanol-Diesel	25% Ethanol-Diesel
Chemical formula	$C_{10.8}H_{18.7}$	C_2H_5OH	–	–	–	–
Mole weight (g)	148.3	46.1	138.08	132.97	127.98	122.75
Density (g/cm^3 at 20°C)	0.84	0.789	0.834	0.832	0.829	0.827
Boiling point (°C)	180–330	78	150–320	140–310	130–300	120–290
Heat of evaporation (kJ/kg)	280	856	337.6	366.4	395.2	424
Lower heat value (MJ/kg)	42.5	27.0	40.95	40.17	39.4	38.62
Liquid viscosity (cP at 20°C)	3.03	1.2	2.84	2.75	2.66	2.57
Surface tension (mN/m at 20°C)	34.1	28.9	33.85	33.32	33.06	32.8
Flash point (°C)	96	16.6	88.06	84.09	80.12	76.15
Stoichiometric air-fuel ratio	14.5:1	9:1	13.95	13.67	13.4	13.13
Cetane number	51	10	46.9	44.85	42.8	40.75
Auto-ignition (°C)	235	423	253.8	263.2	272.6	282.8
Carbon content (wt.%)	87.4	52.2	83.05	81.33	79.6	77.88
Oxygen content (wt.%)	0	34.3	3.43	5.14	6.86	8.57

In the current research, diesel with ethanol, solubilizer and cetane improver blends were prepared just before testing and stored in sealed, dark amber glass bottles under normal laboratory conditions of 20°C–25°C and 20%–50% relative humidity, free from any source of illumination. For blend stability assessment before each engine test, visual inspection was performed for the presence of any signs of blend instability, such as phase separation, sedimentation, turbidity, and discoloration. During the entire course of experiments, there was no indication of phase separation or sediment formation found in the examined blends, suggesting that the stability is satisfactory for the conditions used. Due to the relatively short experimental period during which fuel samples were utilized, there was no need for performing accelerated stability studies.

Table 1 lists the diesel and ethanol blend properties. It must be mentioned that the properties of the fuel indicated in Table 1 have been obtained through experiments and not by any mathematical formula based on the mixture components or any relationship whatsoever. In particular, the cetane number, percentage of oxygen, carbon, and calorific value of the fuel and the mixtures have been evaluated through experimental techniques conducted in Al-Dora refinery-Baghdad labs. Every experimental data point was recorded thrice using the same conditions during operation, and the values recorded were obtained by calculating the arithmetic average of the data collected, while the standard deviation was determined from the spread of the collected data.

It is also worth noting that the biofuel in this case is anhydrous ethanol, which has a very low dynamic viscosity value at 20°C (1.2 cP) compared to that of the diesel fuel (3.03 cP) as stated in Table 1. It is evident, therefore, that the viscosities obtained in the present study for the ethanol-diesel mixtures (2.84–2.57 cP at

20°C for 10%–25% ethanol) lie somewhere between the values of the pure components and become even lower with increased amounts of ethanol in the blend without the addition of any further viscosity reducers except for the miscibility surfactant and cetane improver additives discussed above.

Table 1 shows that as the content of ethanol in the diesel-ethanol blends rises, some primary specifications vary:

- There is a mild drop in density to 0.827 g/cm^3 (25% ethanol) relative to 0.834 g/cm^3 (10 percent ethanol).
- The range of the boiling point decreases, which suggests the rise in volatility (180°C–330°C for diesel to 120°C–290°C at 25% ethanol blend).
- The Heat of Evaporation increases by a large increment (337.60 kJ/kg at 10% to 424.0 kJ/kg at 25%), signifying that ethanol has a greater latent heat.
- Lower Heat Value is reduced; that is, less Energy per kilogram of fuel (from 40.95 MJ/kg at 10% ethanol to 38.62 MJ/kg at 25%).
- Viscosity reduces, which has an impact of enhancing atomization, and it may have weak treatment impacts on lubrication (viscosity ranging between 2.8 cP at 10% to 2.57 cP at 25%).
- Surface Tensions will decrease, and this is likely to improve spray properties (at 10% it is 33.85 mN/m and for 25% it equals 32.8 mN/m).
- Flash Point goes down, hence rendering the mixture more flammable (88.06°C at 10% and 76.15°C at 25%).
- Stoichiometric A/F ratio becomes smaller (13.91:1 at 10% to 13.12:1 at 25%), needs less air to reach full combustion as the oxygen content in the blends increases.
- Cetane Number decreases, possibly affecting the quality of ignition (down 46.9 at 10% to 40.75 at 25%).
- Auto-Ignition Temperature rises, which means that it is less combustible (253.8°C at 10% and 282.8°C at 25%).
- Carbon Content reduces, and oxygen content increases, boosting combustion; however, this lowers the energy density.

In general, higher ethanol levels make the ethanol–diesel mixtures more volatile, more oxygenated, and less energy-dense, while reducing their cetane number and ignition quality.

2.2 Instrumentation and Measurements

The engine used in the study was a direct injection (DI), water-cooled, four-cylinder, inline TD 313 Diesel engine reg (Fiat diesel engine) that operated without the aid of air compressors. The engine has a 100 mm bore, a 110 mm stroke, a 17:1 compression ratio, and delivers 130 kW at 1500 rpm. The main technical information about the engine, like displacement, fuel, and injection timing, is shown in Table 2. In order to assess the engine's performance, it was linked to a hydraulic dynamometer that offered exact control over the engine load. The torque ranges from 600 to 700 Nm, and the speed from 0 to 3000 rpm. The torque range refers to the load range applied by the dynamometer during testing rather than the engine's rated torque. Fig. 1 shows the test engine setup.

Exhaust gas concentrations were measured using an emission measurement system. Emissions of exhaust gas were checked using the Multigas Model 4880 analyzer, which measures nitrogen oxides (NOx), unburned total hydrocarbons (HC), carbon monoxide (CO), and carbon dioxide (CO_2) in real time. Only CO, CO_2, and HC concentrations are checked by the non-dispersive infrared (NDIR) spectroscopy analyzer, while a chemiluminescence detector senses NOx. The oxygen concentration (O_2) is monitored by a sensor within the system, enabling the software to measure excess air and the unit's combustion efficiency.

Table 2: Tested engine specifications [17].

Item	Specification
Engine type	4 cyl., 4-stroke
Engine model	TD 313 Diesel engine reg
Combustion type	DI, water-cooled, natural aspirated
Displacement	3.456 L
Valve per cylinder	Two
Bore	100 mm
Stroke	110 mm
Compression ratio	17
Fuel injection pump	Unit pump 26 mm diameter plunger
Fuel injection nozzle	Hole nozzle 10 nozzle holes Nozzle hole dia. (0.48 mm) Spray angle = 160° Nozzle opening pressure = 40 MPa

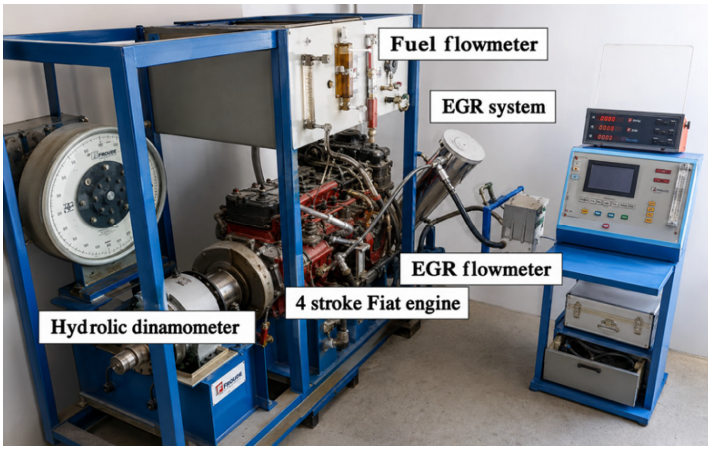


Figure 1: Testing rig. Note: EGR is exhaust gas recirculation.

The sampling probe was inserted directly into the engine exhaust to detect poisonous gases. The heated sampling line was maintained at approximately 180°C to prevent condensation and loss of gaseous species from the exhaust. A condensate separation filter was used to remove moisture before the gas entered the measurement cell. Within the measurement cell, an infrared light source emits rays that pass-through filters designed to match the wavelengths targeted by the gas. At the given wavelengths, the amount of IR light absorbed can be directly correlated with CO, CO_2, and HC levels in the sample.

Homonuclear diatomic molecules such as hydrogen (H_2), nitrogen (N_2), and oxygen (O_2) do not have a permanent dipole moment and are therefore infrared inactive, meaning they will not be detected by an NDIR sensor. This spectroscopic limitation is specific to these nonpolar gases, however, and can be avoided when measuring polar IR-active pollutants like CO, and hydrocarbons. The analyzer also features a chemiluminescence detector (CLD) for separate and critical measurement of nitrogen oxides (NO_x) that are

not reliably measured with NDIR. The CLD functions on a different principle, detecting NO concentrations as the characteristic light emission produced during the unique reaction between NO and O_3 (ozone). The NDIR and CLD modules, therefore, form orthogonal, non-redundant channels within the same analytical platform, each covering a different range of exhaust components and providing a comprehensive and precise measurement of exhaust emissions.

The exhaust emissions of CO, CO_2, HC, and NOx were analyzed using the Multigas Analyzer 4880 (Servomex, UK), which was calibrated regularly with certified span and zero gases traceable to national standards. Each test day, a two-point calibration was performed first using high-purity nitrogen as the zero gas, followed by certified span mixtures (typically 50%–70% of the full-scale reading of the analyzer) for each gas (CO, CO_2, HC, and NOx). A zero-span check was also conducted after each test sequence (about every 2–3 h) to confirm that the drift remained within the manufacturer's specified limits; no recalibration was necessary if the drift was within those limits.

The instruments uncertainty (95% confidence) of the emissions analyzer was taken from the manufacturer's specifications and checked using the calibration gas certificates. It was found to be within the following ranges: ±0.21 vol% for CO_2, ±0.42 vol% for CO, ±0.8 ppm for HC, and ±0.66 ppm for NOx. The uncertainties associated with both the torque and the speed and fuel flow were propagated through the definitions of brake power, brake specific fuel consumption, and brake thermal efficiency by the root-sum-square method, and the instruments uncertainties are summarized in Table 3 for the performance data.

Table 3: Instrumentation uncertainty.

Measurement	Instrument	Manufacturer	Unit	Uncertainty (%)
Exhaust gas temperature	Thermocouple	Pendotech, Italy	°C	±0.68
Mass flow rate	Air flow meter	Fiat, Italy	kg/s	±0.55
Mass flow rate	Fuel flow meter	Fiat, Italy	kg/s	±0.89
Engine speed	Tachometer	Ono Sokki, Japan	RPM	±0.63
Engine torque	Hydraulic dynamometer	Fiat, Italy	kg m	±1.04
CO, HC and NOx levels	Multigas mode 4880	Servomex, UK	Vol. % and ppm	±0.67
Sound pressure level	AVL 450	AVL, Italy	dB	±1.13

Note: HC is unburnt hydrocarbons, NOx is the nitrogen oxides and AVL is a manufacturer brand.

2.3 Engine Performance Parameters

The following equations were used in calculating engine performance parameters [17]:

1. Brake power (kW):

$$bp = \frac{2\pi \times N \times T}{60 \times 1000} \tag{1}$$

2. The engine load (kN/m^2):

$$Load = bp \times \frac{2 \times 60}{V_{sn} \times N} \tag{2}$$

3. Fuel mass flow rate (kg/s):

$$\dot{m}_f = \frac{V_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \tag{3}$$

4. Brake specific fuel consumption (kg/kW h) [22,23]:

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \tag{4}$$

5. Total fuel heat (kW):

$$Q_t = \dot{m}_f \times LCV \tag{5}$$

6. Brake thermal efficiency (%) [24]:

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 = \frac{3600}{BSFC \times LCV} \tag{6}$$

The uncertainty value of each instrument was estimated using the manufacturer's claimed accuracy and the repeatability of the measurements. The standard deviation of the multiple readings for each parameter measured was first determined, and then the random variation of individual measurements is generally represented by the standard deviation or by an experimentally determined measurement standard uncertainty. The uncertainty of the derived quantities was determined using the root sum square method, which involved applying the uncertainty contributions from the corresponding input variables, following the general principles of uncertainty propagation. The uncertainties in fuel flow rate, engine speed, torque, and load were propagated through the governing equations to estimate the uncertainty in the brake power, brake specific fuel consumption, and brake thermal efficiency for the performance parameters. Likewise, the uncertainty in the emission measurement was calculated using the accuracy of a gas analyzer, and the uncertainty in the sound pressure level was calculated using the specification of a microphone and the data acquisition system. The expanded uncertainty values reported were based on the uncertainty arising from the experiment to give a more accurate picture of the level of confidence of the experimental result. The Gaussian equation was used to assess the uncertainty in the tests:

$$e_R = \left(\left(\frac{\partial R}{\partial V_1} e_1 \right)^2 + \left(\frac{\partial R}{\partial V_2} e_2 \right)^2 + \ldots + \left(\frac{\partial R}{\partial V_n} e_n \right)^2 \right)^{0.5} \tag{7}$$

In Table 3, the measured engine performance and emissions variables are assigned. Also, the test uncertainty for the thermophysical parameters is added.

2.4 Testing Procedure

Experiments begin with the engine warming up and running on pure diesel fuel until it reaches a steady state, after which readings are taken. Two sets of experiments were completed for diesel and blends. The first involved running the engine at constant injection timing, i.e., at the optimum injection timing (OIT) for diesel fuel. The second involved running the engine at the optimum injection timing for each fuel. In both conditions, brake-specific fuel consumption (BSFC), brake thermal efficiency (BTE), and exhaust pollutants were measured.

The OIT for each tested fuel was established experimentally by means of an initial sweep of timing at the chosen engine speed and loading condition. Several different timing settings were evaluated for each fuel near the basic diesel timing point, ranging from advanced to retarded injection timings, along with the corresponding BSFC, BTE, exhaust gas temperature, and combustion quality. The OIT was found to be a time when all the above parameters achieved the optimal combination of values without any occurrence of misfire. The same procedure was applied to each tested fuel to find the optimum injection timing.

Two test procedures have been used to investigate the characteristics of the examined fuel blends: running with the injection timing set at a fixed point and running with the OIT for each fuel tested. For clarity of presentation, it is specified explicitly below whether the presented results belong to the case of a particular injection technique. The data obtained with fixed injection timing refer to a fixed diesel calibration for all fuels considered. At the same time, data on the case of OIT correspond to the timing calibrated individually for each fuel as the result of the optimization procedure carried out first. It should be noted that the choice of injection timing is critical for achieving optimal combustion phase, optimal BSFC, optimal BTE, minimal emissions, and stable engine operation.

During the experiments, the ambient test conditions were monitored and kept within a narrow range, with the laboratory air temperature maintained at 23°C–26°C, the relative humidity at 35%–45%, and the atmospheric pressure at 100.0 to 101.5 kPa, so as to minimize their influence on the measured engine performance, emissions, and noise characteristics and to ensure repeatable test conditions.

In this experiment, only one speed, namely 1500 rpm, was maintained for carrying out the study, meaning that the conclusions obtained are mainly based on the performance of the fuels at this particular speed. Consequently, generalization of these results cannot be done until further verification for varying engine speed and load conditions is carried out. Moreover, this study involves short-term testing, where long-term factors like injector clogging, component wear resistance, fuel storage life, and fuel system component compatibility have not been taken into consideration.

3 Results and Discussion

3.1 Brake Specific Fuel Consumption

BSFC is defined as the ratio of the total mass flow rate of fuel consumed by an internal combustion engine to the mechanical power output delivered at the crankshaft, measured under steady-state dynamometer conditions (Eq. (4)). Fig. 2 shows how BSFC changes with load for the tested blends, with the optimum injection timing for each blend. The fractional load values shown in Fig. 2 were used as normalized engine load levels (as a percentage of full load). That is, the dynamometer resistance was modified to apply the required fractional load, and the actual load applied to the engine was determined by the measured brake torque divided by the rated brake torque for the engine. These intermediate load points were chosen to be representative of engine performance throughout the operating range and to represent trends in performance and emissions between the low, medium, and high load operating points.

At low loads, diesel-ethanol blends exhibit a higher BSFC than regular diesel fuel. This increase is due to the need for more fuel to compensate for the blend's lower net calorific value, which is necessary for combustion or to generate an equivalent amount of heat to higher-calorific fuels. Consequently, the 25% ethanol blend, which has the lowest net calorific value among the fuels tested, exhibits the highest BSFC. Adding 10% ethanol results in a minor increase in fuel consumption, and the difference becomes even smaller at higher loads. Even at a very high load (85.78 kN/m^2), the BSFC for the 10% blend is not far from that of diesel. Adding ethanol to diesel at 10%, 15%, 20%, and 25% increased BSFC by 2%, 5.4%, 8.9% and 12%, respectively. BSFC is at its peak at low loads, while at moderate and high loads, it is enhanced. 20% and 25% ethanol additions to diesel resulted in lower BSFC at medium and high loads compared to its values at low loads. This can be referred to as the increase in the temperature inside the combustion chamber that compensates for the ethanol's high heat of vaporization and introduces better air-fuel preparation. The data illustrate a rise in diesel consumption compared to diesel-ethanol consumption. This is due to better combustion and higher energy output in this situation. These results confirm the importance of using OIT with ethanol-diesel blends.

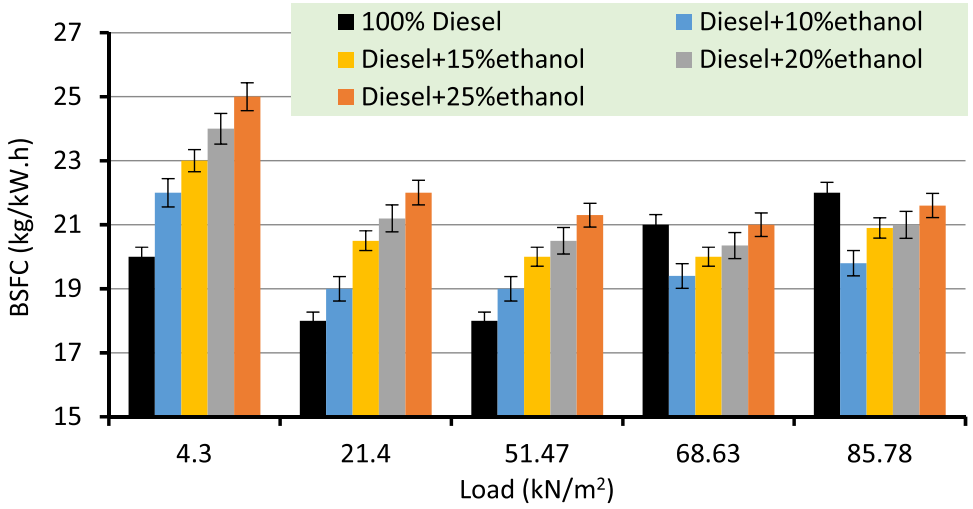


Figure 2: The effect of engine loads on the BSFC (brake-specific fuel consumption) of diesel-ethanol blends at OIT (optimum injection timing) for each fuel.

Error bars, where shown, indicate the standard deviation of three repeated measurements at each operating condition.

To avoid confusion, it should be noted that the columns in Fig. 2 were obtained by OIT for each fuel. In contrast, those in Fig. 3 were obtained by maintaining a fixed diesel injection timing. In other words, the difference observed between these two graphs is due to the injection strategy used, not to an inconsistency in the data. Given that the process of fuel injection plays a major role in the combustion phasing, ignition delay, and mixing of air with fuel, there will likely be different trends shown by the same fuel depending on the operating mode.

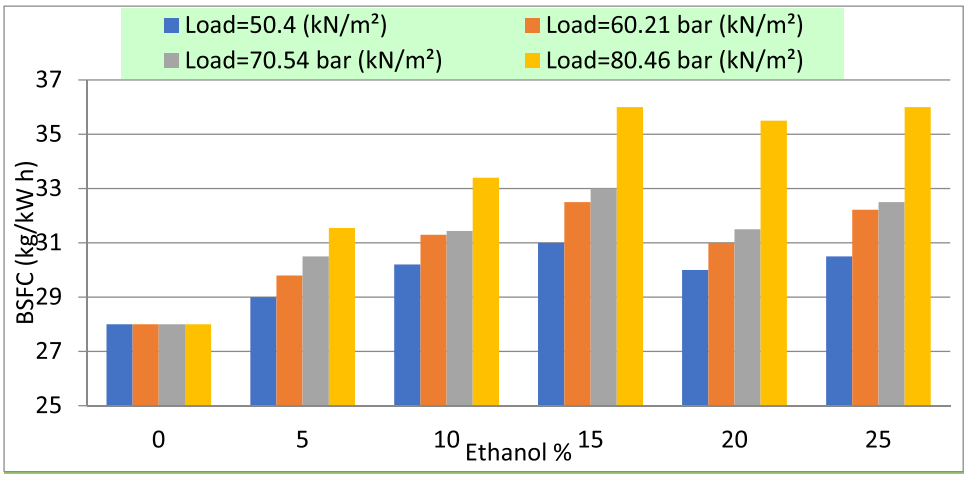


Figure 3: The relation between ethanol percentage in the blends and BSFC (brake-specific fuel consumption) at diesel's OIT (optimum injection timing).

Fig. 3 gives BSFC readings for each blend according to the load on the engine while it is operating at fixed injection timing (OIT for diesel). Diesel-ethanol blends with higher ethanol content burn more fuel per kW than diesel does and prove uneconomical at low to medium loads of operation. This is the result of

ethanol's net calorific value being lower, which requires more fuel to produce the same amount of power. Also, ethanol adds oxygen to the combustion process. This increases the likelihood of perfect combustion, which may decrease unburned hydrocarbons and raise general combustion efficiency.

Under loads of 60.21, 70.54, and 80.46 kN/m^2, the engine's BSFC increased by 3.47%, 5.45%, and 12.89%, respectively, compared to its value under a load of 50.4 kN/m^2. The BSFC, based on addition rates of 5%, 10%, 15%, 20%, and 25%, was as follows: 9.71%, 12.8%, 18.3%, 14.28%, and 17.16%, compared to diesel, respectively. Ethanol does not offer the same amount of heat as diesel does. For this reason, an engine needs to use more ethanol-blended fuel, which results in higher BSFC. At all the tested engine loads, only diesel leads to the lowest amount of fuel needed for a given unit of power, and when there is more ethanol in the blend, it results in higher BSFC. It is important to recognize that the outcomes shown in Fig. 3 might contradict those in Fig. 2. This difference arises because Fig. 2 uses the OIT for each type of fuel, whereas Fig. 3 maintains a constant injection timing set at the diesel OIT. Bhowmik et al. [23] pointed out the same behavior in their latest study.

For clarity, it is important to note that the ethanol blend was intentionally formulated and tested at each of the desired volume concentrations: 10%, 15%, 20%, and 25%, the levels where it is actually blended in the main performance and emissions testing. The reference condition is set at 0%, to serve as a baseline for comparison, which is pure diesel fuel. The 5% ratio is only listed in the evaluation for misfiring limit as an auxiliary operating point where the minimum stable combustion limit is calculated, and the onset of blend-induced instability is identified. The 0%–25% range of the misfire configuration is therefore not in conflict with the 10%–25% blend matrix applied during the main experimental campaign but represents a second stability analysis deliberately applied to extend the mapping of the combustion limits. The misfiring limits as a function of ethanol content and loading, as observed in the experiments, are shown in Fig. 4. It is intended to show the realistic limit of performance and emissions for the blends tested, which is necessary for the interpretation of the results. Thus, the presence of misfiring was identified in the engine through the unstable torque response, cycle-to-cycle variation, and by the engine being unable to operate normally during testing.

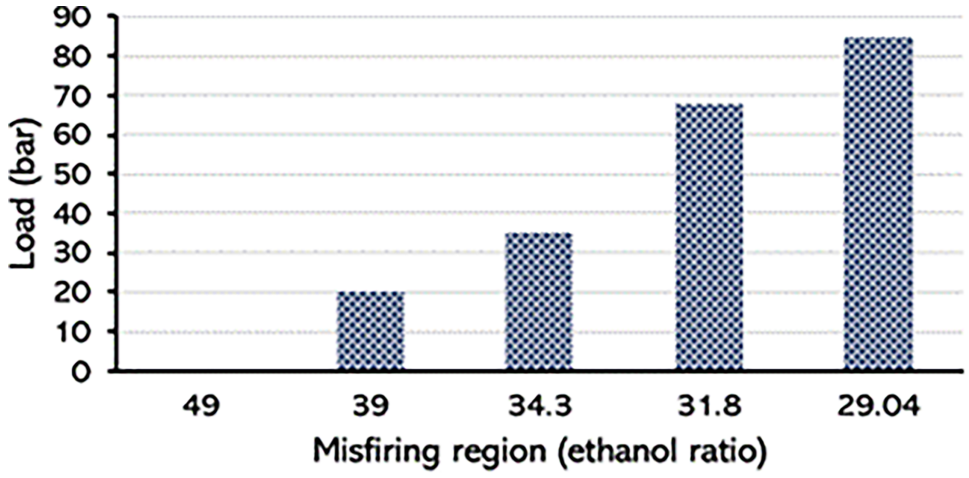


Figure 4: Variation in the misfire region with added ethanol and load.

BTE represents the percentage of Energy generated by fuel combustion in the cylinder converted into useful work. BTE is calculated using the formula shown in Eq. (6). This equation shows an inverse relationship between BTE and BSFC. If the former increases, the latter decreases, and *vice versa*. Fig. 5 shows that the diesel engine demonstrates higher brake thermal efficiency at low and medium loads.

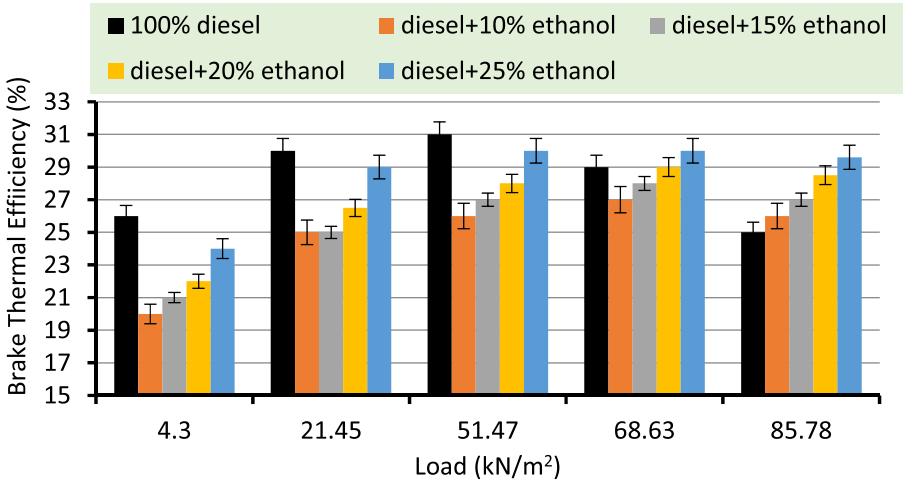


Figure 5: The effect of engine loads on the BTE (brake thermal efficiency) of the tested blends at OIT (optimum injection timing) for each fuel.

In contrast, engines using diesel-ethanol blends show enhanced performance at high loads due to improved combustion from increased oxygen availability, as illustrated in Fig. 5. When operating the engine in OIT for each blend, it is seen that at low loads, BTE decreases because the mixture temperature is lower. Additionally, ethanol absorbs a significant amount of Energy from the combustion chamber during evaporation, which lowers both temperature and pressure. This leads to a longer delay, subsequently reducing the output. Another contributing factor could be incomplete combustion. Because of the heavy load, ethanol burns with extra oxygen to achieve more effective combustion. At higher loads, the increased combustion temperature prevents ethanol's ability to act as a coolant, resulting in greater fuel burn and higher BTE in the blends. The present results are compatible with Bhowmik et al. [23] and Zhang et al. [25].

3.2 Analysis of the Misfiring Phenomena

The investigation into misfire limits is illustrated in Fig. 4. For the range of tested loads, misfire values were constrained to 49%, 39%, 34.3%, 31.8%, and 29.04% ethanol, corresponding to load values (no load, 20, 35, 67.5, and 85) kN/m^2, respectively. Long delays can lead to misfires, limiting the use of alcohol as a diesel fuel substitute. A rapid rise in pressure from the swift, intense combustion of the alcohol/air mixture can lead to self-ignition. As the first reactions occur, both the pressure and temperature in the cylinder increase, extending the diesel delay period. The use of alcohol as a coolant hinders replacement and delays issues at low to medium loads. In contrast, at high loads, quick combustion can trigger knocking, further limiting the replacement.

The experimental results show that the engine ran without misfiring when ethanol was added to the Diesel at a concentration of 29.04% up to an engine load of 85 kN/m^2. However, at a concentration of 31.8%, misfire occurred at a load of 67.5 kN/m^2. Similarly, the engine shut down at 49% ethanol concentration with no engine load. These results show that the addition of ethanol must be within limits that prevent misfires across the entire engine load range. Therefore, in this study, we adhered to a maximum ethanol concentration of 25% to ensure continuous, misfire-free combustion.

3.3 Analysis of the Injection Timing Effect

Fig. 6 shows the relationship between injection timing and BSFC for diesel, as well as for diesel and ethanol (10% ethanol) at a constant engine load of 47.44 kN/m^2 and 1500 rpm. For diesel, BSFC decreases

as the fuel injection approaches the OIT, reaching its minimum at 34° BTDC (before top dead centre) (the OIT for diesel). As injection timing is retarded (at 23°, 26°, and 30° BTDC), BSFC increases. For the 10% ethanol-diesel blend, at injection timing of 38° BTDC (which stands for the OIT for this blend), the BSFC was decreased less than for diesel.

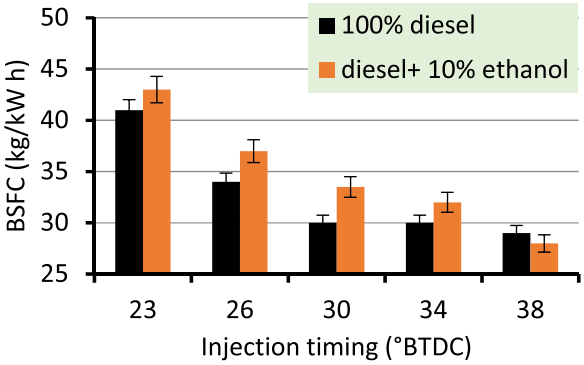


Figure 6: Injection timing effect on BSFC (brake-specific fuel consumption) for tested fuels.

Timing sensitivity analysis was carried out for E10 only since E10 was chosen as a representative case of a low-ethanol fuel blend that exhibited stability throughout the experiment and did not misfire anywhere within the test range. The use of a lower blend percentage was also necessary because using a larger blend would mean having to factor in other confounding variables that arise when there is more ethanol. Such confounding factors include increased charge cooling, increased ignition delay and instability arising from higher ethanol content in the mixture. Moreover, performing a timing sensitivity analysis for each ethanol blend percentage could have increased the experimental scope of work without necessarily adding more value to the engineering insight.

The reason for this is that, at this timing, the blend is more suitable for combustion. When injection timing is retarded, both fuels have a higher BSFC, and the ethanol-diesel blend has the highest value. This occurs because the fuel and air do not have enough time to fully mix and combust, increasing fuel consumption. At its lowest blend concentration, ethanol increases BSFC by about 6% compared to diesel because it takes longer to ignite, due to its lower cetane number and higher latent heat. The present results are consistent with those reported by Bhowmik et al. [23] and Wang et al. [26].

There is a straightforward relationship BSFC and ethanol content because it relies on the changes resulted by reduction in the blend's lower heating value (LHV). The overall trend for higher BSFC at higher ethanol levels is the result of a combination of effects such as increased oxygen availability, reduced viscosity, reduced surface tension, improved atomization, ignition delay, and combustion phasing. These effects increase as ethanol concentration decreases and intermediate, and compensate for the BSFC penalty from LHV in part, resulting in a non-linear relationship between BSFC and ethanol level. Additionally, the use of an optimum injection timing (OIT) for each fuel blend affects the overall trend of BSFC of the combustion efficiency balance. Importantly, the OIT data do not contradict the overall upward trend in BSFC but rather provide a kinetic basis for the non-linear deviations—if the onset of ignition occurs earlier in the crank angle (relative to the baseline), then the heat release will also be earlier, and the combustion will be better, locally flattening the BSFC curve for certain ratios of the blend. However, the general trend of an increase in BSFC is thermodynamically controlled by the energy density loss from ethanol, which can be reduced but not overcome by the correct injection timing.

Fig. 7 illustrates how BTE changes with injection timing for both pure diesel and a 10% ethanol-diesel blend. This analysis is conducted at an engine load of 47.44 kN/m^2 and a speed of 1500 rpm. The data reveal that as the injection timing is advanced, the BTE improves for both types of fuel, peaking at 38° before top dead center (BTDC). This increase in efficiency indicates better combustion timing, which enhances the complete burning of fuel and optimizes energy use. Conversely, at a retarded injection timing, such as 23° BTDC, BTE reaches its lowest point for both fuels. Interestingly, the diesel-ethanol blend performs slightly better than the pure diesel at this timing. This boost can be linked to the oxygen present in ethanol, which enhances combustion, particularly when ignition is delayed.

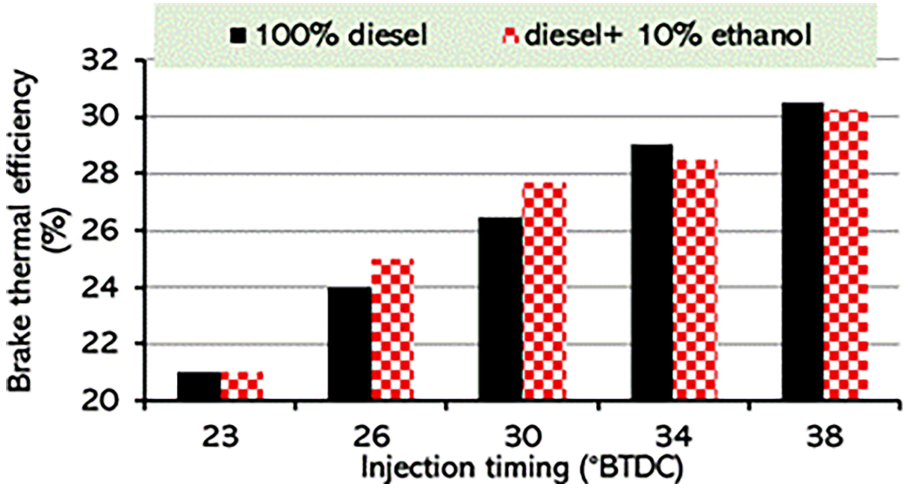


Figure 7: The effect of injection timing variation on the BTE (brake thermal efficiency) of the tested blends.

Nevertheless, the overall performance is still not ideal at retarded timings due to ethanol's lower cetane number and slower ignition rate. When the injection timing is advanced from 26° to 34° BTDC, a significant increase in efficiency occurs. Pure diesel OIT (34° BTDC) provides a higher BTE compared to the ethanol blend. This result is anticipated since diesel ignites more easily under compression, promoting efficient combustion and minimizing energy loss. On the other hand, the BTE of the blend is relatively close to that of the diesel one. However, ethanol's longer ignition delay and its higher heat of vaporization limited the increase of BTE under these circumstances.

The noted shift in the ignition delay, slightly longer than with reference ethanol content, and a slight shift of the main heat release towards higher crank angles with an increase in ethanol content may be directly attributed to the fundamental fuel characteristics stated in Table 1. Ethanol has a significantly lower cetane number and a higher autoignition temperature than diesel, which reduces its chemical reactivity under compression and consequently slows the initiation of the autoignition process in the premixed charge. On the other hand, the latent heat of ethanol vaporization provides a greater cooling effect on the in-cylinder mixture during injection and mixing, leading to lower temperatures prior to ignition, which, in turn, supports the delay. Therefore, the amount of cumulative fuel-air mixture during the delay period is increased, leading to a more forceful premixed combustion at the onset of ignition, followed by a slightly shorter duration of the diffusion-controlled combustion than that of pure diesel.

It is also these combined effects that lead to the increase in the peak cylinder temperature as ethanol blends with them. At low- to moderate-blending ratios, increased premixed combustion and greater mixture homogeneity are likely to elevate the peak flame temperature compared to diesel, but at the highest ethanol level, the large charge cooling and the mixture's lower heating value compensate somewhat for this effect.

In this case, the maximum pressures and temperatures do not vary discontinuously with the proportion of ethanol but rather indicate the competition between the improvement of the premixed burn and the cooling, reactivity-reducing properties of the ethanol.

In conclusion, the "benchmark" utilized for interpreting the performance and emissions results obtained in this study is the combination of the impact of the fuel properties listed in Table 1 along with the optimized injection timings and misfire ranges for each blend. The observed changes in BSFC, BTE, exhaust gas temperature (EGT), NOx, CO, and HC can be attributed to the decreased lower heating value and carbon content; increased oxygen content; and high latent heat of vaporization of ethanol while the reduced viscosity and surface tension have resulted in enhanced atomization and mixture formation, and the reduced cetane number and higher auto-ignition temperature have accounted for the prolonged ignition delays, misfire ranges, and timing sensitivity seen in Figs. 4–7.

Based on the findings, ethanol-diesel blends have the capability to enhance brake thermal efficiency depending on various operating factors such as the load and fuel injection timing, but when these factors are unfavorable, the high latent heat value and long ignition delay time for ethanol will decrease thermal efficiency. Present-day tendencies correspond to recent studies; however, the current work demonstrates that the advantages or drawbacks under consideration are significantly dependent on the injection timing and load of the engine, which can explain contradictions in other works.

It is important to note that the direct measurement of flame temperature was not performed in this study; thus, any information regarding it is based on inferences from their behavior, the results and findings presented, and literature such as [26,27] for ethanol-diesel blends.

3.4 Analysis of the Exhaust Emissions

Fig. 8 shows how the ethanol content in the fuel blend affects EGT. As the ethanol content of the diesel-ethanol mix increases, EGT decreases. This drop in EGT results from various physical and combustion traits of ethanol. First, ethanol has a lower heating value than diesel, meaning that for the same amount of fuel, it produces less Energy when burned. Consequently, as ethanol concentration increases, the peak combustion temperature and the exhaust temperature decline.

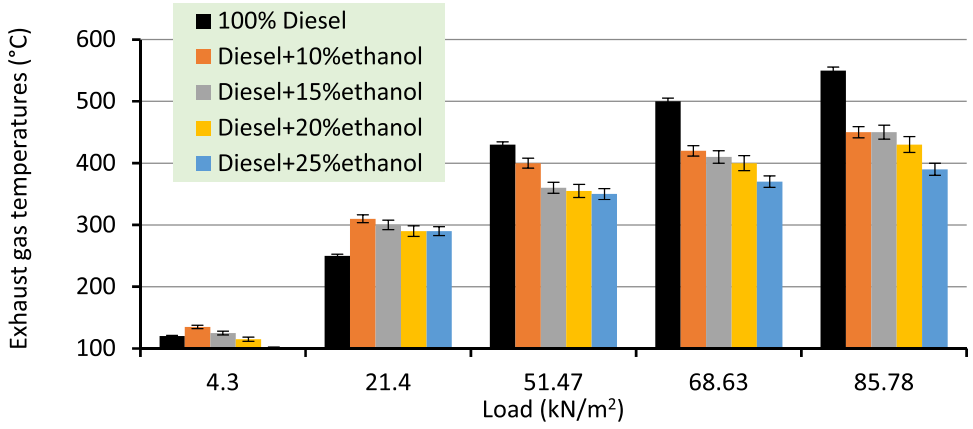


Figure 8: The effect of load variation on exhaust gas temperatures for the tested blends.

Additionally, ethanol has a higher latent heat of vaporization, which consumes a notable amount of thermal Energy during its vaporization. This phenomenon results in local cooling within the combustion chamber, further reducing exhaust gas temperature. The presence of ethanol can also lead to an extended ignition delay, particularly under lower loads or non-ideal injection conditions. This postponed combustion could shift some of it into the expansion stroke, where thermal efficiency is reduced, and peak temperatures are lower. The present results are compatible with Karthikeyan and Srithar [28] results.

It is also necessary to note that no measurements were performed to determine the temperature of combustion; thus, any information concerning them can only be considered as an inference drawn based on the observed results along with those in the literature regarding the ethanol-diesel fuel blend [29,30].

Fig. 9 presents the NOx emissions from pure diesel and diesel-ethanol blends containing 10% to 25% ethanol across different load conditions. NOx emissions are mainly driven by thermal processes, resulting from nitrogen oxidation at elevated combustion temperatures (according to the Zeldovich mechanism). The figure shows that under light-load conditions, NOx emissions from diesel-ethanol blends are similar to those from pure diesel. This indicates that the combustion temperatures for all fuels tested do not vary significantly at lower loads, where thermal conditions are less intense. However, at higher loads, 75%–100%, a decrease of 20%–26% in NOx levels are noted for ethanol-blended fuels compared to pure diesel. This trend can be attributed to a reduction in adiabatic flame temperature resulting from ethanol's substantial latent heat of vaporization, leading to cooler charge temperatures and, in turn, lower maximum combustion temperatures. Chen et al. [31] explained that nitrogen oxide emissions consistently decreased as the proportion of ethanol in the blend increased. The main reason is that although adding ethanol increased the mixture's oxygen content, the lower average combustion temperature resulting from its strong cooling effect reduced the average combustion temperature in the combustion chamber. For the whole loads range of loads, ethanol concentrations of 10%, 15%, 20%, and 25%, the reduction rates were 19.73%, 10.53%, 17.65%, and 12.69%, respectively, compared to diesel.

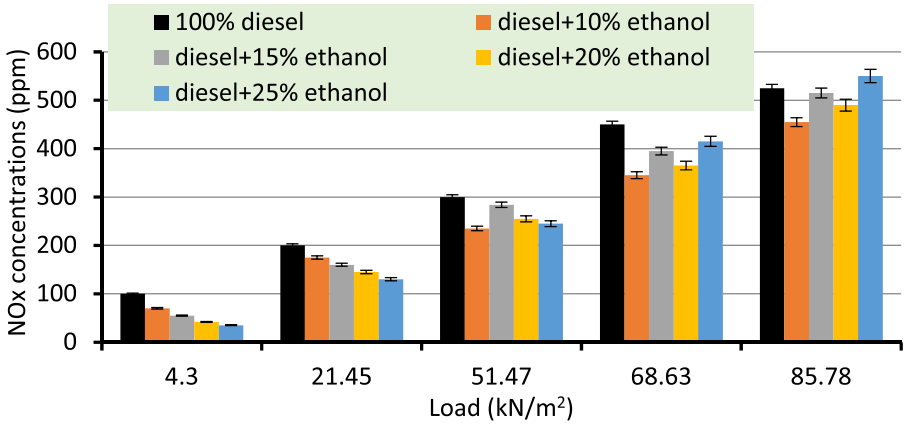


Figure 9: The effect of load variation on NOx concentrations for the tested blends.

The nonmonotonic dependence of NOx on ethanol content demonstrates the dual action of ethanol on in-cylinder temperature and oxygen availability. On the one hand, the latent heat of vaporization of ethanol is high, lowering the temperature of the intake charge and the local flame temperatures, which is likely to inhibit the production of thermal NO via the Zeldovich mechanism. Conversely, the delay in ignition due to a lower cetane number increases the level of the premixed fuel-air mixture and may result in higher-intensity burning of the premixed fuel at localized, higher peak temperatures under certain operating conditions, biased toward NOx. In the case of the intermediate blends, the synergistic effect of enhanced premixed combustion and

the added oxygen level appears to raise local temperatures in the primary reaction region, leading to partial recovery of NOx following the initial NOx loss observed at low ethanol ratios.

The high ethanol content causes a strong charge-cooling effect and a lower heating value, lowering the adiabatic flame temperature enough to further reduce NOx relative to neat diesel. This is why the increase or decrease in NOx is not always proportional to ethanol content but rather reflects a balance between temperature-lowering and temperature-increasing processes associated with ignition delay, heat-release timing, and the local oxygen supply.

Fig. 10 shows the hydrocarbon (HC) emissions of a pure diesel engine as well as an engine using diesel-ethanol blends with 10–25% ethanol under various loads. All ethanol blends have lower HC emissions than pure diesel when averaged over the whole load spectrum, with the overall HC reduction increasing monotonously with ethanol content and equal to 6.3%, 14.86%, 19.70%, and 28.62% for 10%, 15%, 20%, and 25% ethanol, respectively. The main reason for this worldwide trend is the oxygen-rich structure of ethanol (C_2H_5OH), which creates more oxygen for the environment in which unburned hydrocarbons must be oxidized, which is well established as an "oxygenate effect," and the lower viscosity of ethanol, which helps to improve air–fuel mixing at moderate loads. This general decline is a masking effect, however, of an anomaly related to critical load. In particular, the 25% ethanol blend in the highest load tested found a surprising increase in HC emissions when compared with both the pure diesel fuel and the lower ethanol blends. Under these extreme thermal conditions, the charge cooling effect associated with the high ethanol fraction (because of its large latent heat of vaporization) starts to dominate its oxygen-enhancing properties, extending ignition delay and inducing local flame quenching near the combustion chamber walls, resulting in incomplete combustion. The localized HC increase at the highest operating condition does not contradict the overall reduction trend. Rather, it shows that HC emissions can increase locally despite a substantial reduction in the maximum rate of heat release zone, the shortcoming in the high MRHZ is countered by the improvements in the low- to moderate load. Therefore, on average, the overall relationship is positive but very non-linear and load sensitive.

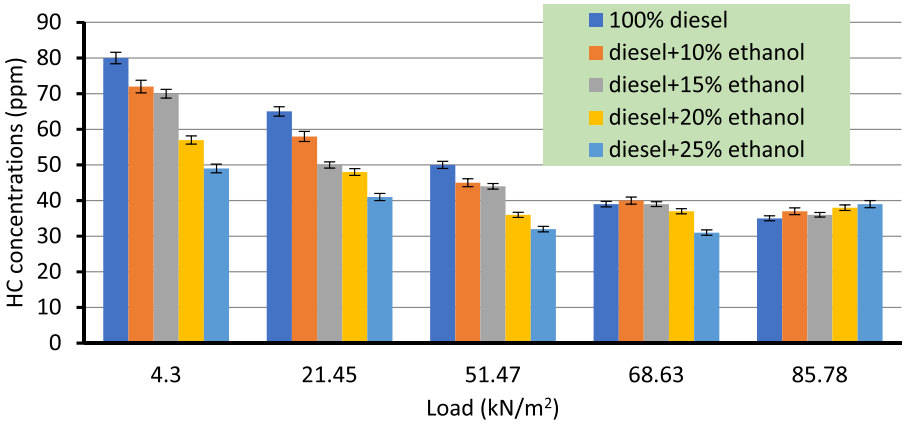


Figure 10: The effect of load variation on HC concentrations for the tested blends.

At first glance, there appears to be a contradiction between the above results: the HC increases in the E25 blend at the maximum load, whereas earlier it was stated that HC emissions decrease with increasing ethanol content. This seeming contradiction can be explained from two points of view, where each statement represents a different level of analysis. On the one hand, if we compare the values of HC emissions across the entire range of loads, it can be seen that the use of any ethanol-diesel blends (E10–E25) gives a smaller value of the integral HC emission than the neat diesel engine, and furthermore, the difference increases with the

growth in the content of ethanol, which is clearly illustrated by the percent reduction in the graph. On the contrary, from Fig. 10, one can observe that E25 also displays a localized rise in HC emission at the specific operating point of 85.78 kN/m^2 load due to the cooling effect of ethanol dominating the combustion process, resulting in partial oxidation.

Fig. 11 shows CO emissions from both pure diesel and diesel-ethanol blends (with 10%–25% ethanol) across different loads at a fixed injection timing (diesel OIT). Blends containing 10%–20% ethanol consistently emitted lower CO than pure diesel at most load conditions. The molecular oxygen in ethanol (C_2H_5OH) enhances combustion efficiency, promoting the conversion of CO-to-CO_2. The higher oxygen content in ethanol reduces the formation of fuel-rich areas, thereby minimizing the production of partial combustion products such as CO. Additionally, ethanol's lower viscosity and increased volatility facilitate better air-fuel mixing, reducing areas conducive to CO formation. Reductions in carbon monoxide (CO) emissions can be associated with enhanced combustion efficiency due to ethanol's oxygen content. However, the observed rise in CO levels at a 25% ethanol blend (85.78 kN/m^2 load) indicates a limit where the cooling effects of ethanol start to outweigh its beneficial oxygen properties, leading to more incomplete combustion. The reduction rates were 10.06%, 20.89%, 40.34%, and 49.13% compared to diesel, respectively.

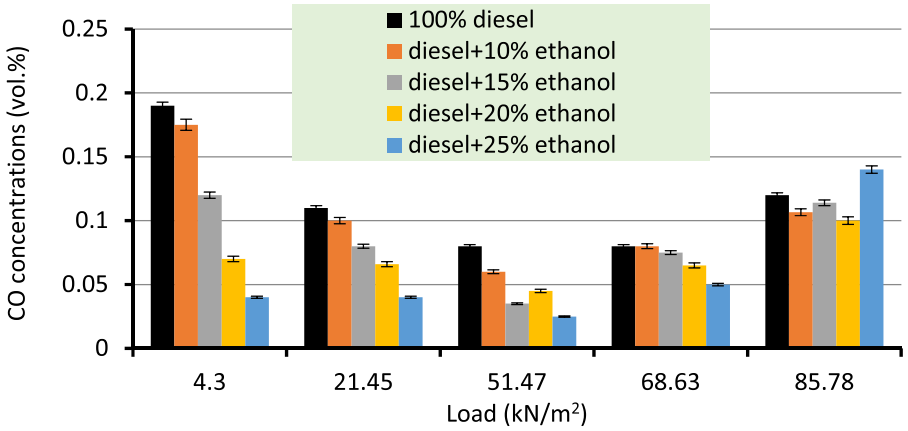


Figure 11: The effect of load variation on CO concentrations for the tested blends.

The percentages of reductions in CO emissions were calculated based on the comparison of CO emission values for each ethanol blend and the corresponding CO emission value for diesel at the same operating point according to the following:

$$Reduction\ (\%) = \frac{CO_{diesel} - CO_{blend}}{CO_{diesel}} \times 100$$

In this definition, CO_{diesel} represents the carbon monoxide concentration measured when the engine operates on neat diesel fuel, and CO_{blend} is the CO concentration measured under exactly the same speed, load, and injection-timing conditions when running on a given ethanol-diesel blend. This expression is the conventional relative-change formulation used in engine-emissions studies, as it quantifies the fractional decrease in CO with respect to the diesel baseline for each specific operating point, ensuring that the reported reduction percentage directly reflects the change in CO attributable to the fuel blend only, while other parameters are held constant. Consequently, a positive value of "Reduction" indicates that the blend emits less CO than neat diesel at that operating condition, whereas a negative value would indicate an increase in CO emissions. Therefore, the percentage reduction value of 49.13% for E25 is a general figure representing

the total reduction percentage in relation to diesel for the entire range of operation tested, based on the CO data measured for the comparison. As far as the increase in CO at high loads, this is simply a local operating point effect, and it does not make the total percentage reduction invalid; instead, it shows that at high loads, the cooling ability of ethanol can somehow compensate for the oxygenation effect of ethanol, resulting in a CO emission increase due to incomplete oxidation.

The significant decrease in CO and HC at higher ethanol concentrations may be explained primarily by the added chemically bound oxygen and the enhanced spray and mixing properties of the mixtures. From Table 1, it can be seen that the viscosity and surface tension of ethanol-diesel fuel mixtures are lower than those of naked diesel fuel, which makes the fuel atomization finer and the fuel-air mixing more efficient. This, together with an increased oxygen content of the blends, promotes local equivalence ratios in rich zones and allows greater oxidation of CO and unburned hydrocarbons, especially in the late-burn and near-wall zones. These effects prevail under full load and high-temperature operation, despite the marginally higher ignition delay, such that total CO and HC emissions would decrease with increasing ethanol fraction.

The noise emitted by the engines was measured using a Class 1 integrating sound level meter (IEC 61672 standard) with a free-field microphone. The measurement was performed by positioning the microphone 1.0 m from the engine and 1.0 m above the floor, on the side where the injection pump is located, with the microphone pointing towards the center of the engine block. The A-weighted sound pressure level at steady-state operation was measured for each case after 30 s, and the value given represents the average of three measurements when the background noise was less than 10 dB(A). This does not require an ambient noise level of 10 dB(A); rather, it is the minimum difference in sound pressure level between the signal level and the background noise level (engine noise minus the ambient noise level in the laboratory), which is 10 dB(A). This is the minimum difference between the signal level and the background noise level (engine noise minus the ambient noise level in the engine test laboratory), and it is not a requirement that the ambient noise level be 10 dB(A), which is practically impossible in a test laboratory of a running engine.

The sound pressure levels (SPL) of pure diesel are compared to diesel–ethanol blends with 10%–25% ethanol at different loads in Fig. 12. The reductions ranged from 8.5% to 26.05%. The maximum reduction in SLP was at the highest load (26.05%) for all ethanol blends, compared to pure diesel. The ignition characteristics for blended fuels are determined by the high cetane number fuel (diesel), with the ethanol, having a much lower cetane number (as shown in Table 1), also influencing the combustion process mainly through its oxygen content and significant charge-cooling effect. As the previous results showed, ethanol modifies combustion phasing and reduces the rate of pressure rise, which diminishes high-frequency pressure oscillations and lowers combustion noise. This causes reduction of the pressure oscillations at high frequency, which produce combustion noise. The lower energy density and higher latent heat of vaporization of ethanol aid in controlling the peak cylinder pressures and reduce the amplitude of pressure waves generated during combustion, while its volatility can be beneficial for mixture formation and spatially uniform combustion. As a whole, the SPL reduction is attributed primarily to smoother combustion phasing and a lower pressure-rise rate caused by ethanol's charge-cooling and dilution effects, rather than to a general reduction in the premixed combustion fraction.

The presence of ethanol alone generally causes an increase in ignition delay, but the overall ignition behavior of the blends is determined by a balance between its cooling and oxygenating effect and the high reactivity of the diesel, resulting in less abrupt combustion phasing than is realized using neat diesel.

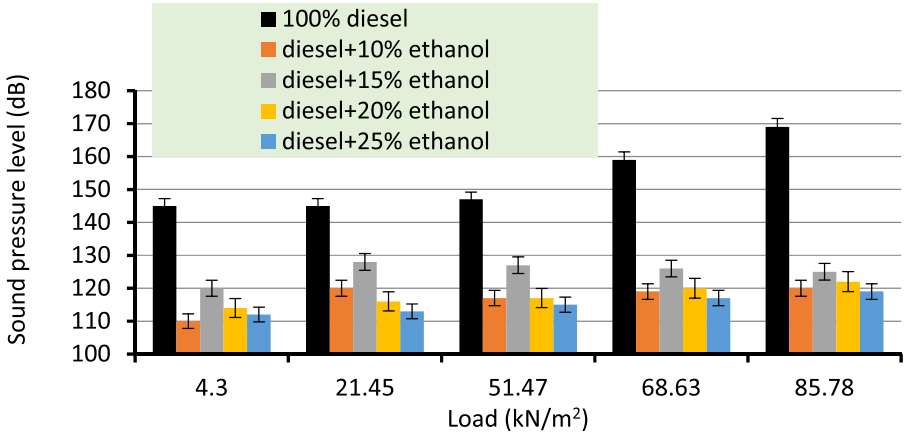


Figure 12: The effect of load variation on engine noise for the tested blends.

3.5 Comparison with Literature

Table 4 compares the results of the current study with those of some of the latest studies in this field. Certainly, this comparison is not entirely fair due to differences in the engines used, operating conditions, and even the fuel. However, these comparisons remain useful for identifying public patterns, assessing work accuracy, and evaluating the validity of their results. The table shows several directions, for example, Hamdi et al. [24], and Chen et al. [31], whose results showed an increase in BTE against a decline in BSFC, which is due to the opposite relationship between the two. The results of the current study, as well as those of Muniyappan and Krishnaiah [32], Theinnoi et al. [33] and Zhang et al. [34], showed a different behavior: BTE increased with increasing BSFC. Both results are acceptable despite their contradiction because the first group worked in OIT for each type of fuel, and the second group worked in a fixed OIT (diesel fuel). Also, all studies in the table except for [33,34] have agreed on the results of the current study, as NOx concentrations have decreased. In Theinnoi et al. [33] results, CO and HC concentrations were increased. Zhang et al. [34] findings showed a clear increase in HC and NOx concentrations. From this comparison, it can be concluded that the results of the current study were practically consistent with the conclusions.

Table 4: Comparison with the literature showing the percentages of changes of the performance parameters and the exhaust gases contains compared to neat diesel.

Ref.	Blend Type	Engine Type	Test Type	BTE (%)	BSFC (%)	CO (%)	HC (%)	NOx (%)
[22]	Diesel + waste plastic pyrolysis oil (WPPO) + ethanol	single-cylinder, 4-stroke, air-cooled, indirect injection diesel engine	Load (rated for full load)	−0.5%	−3.66	−25	−10.35	−27.1
[24]	Diesel + biodiesel l + ethanol	single-cylinder, 4-stroke, air-cooled LOMBARDINI diesel engine	50% loaded at 1000–2500 rpm	+2.5	−5	−32	–	−2.5

(Continued)

Table 4 (continued)

Ref.	Blend Type	Engine Type	Test Type	BTE (%)	BSFC (%)	CO (%)	HC (%)	NOx (%)
[31]	Diesel + ethanol	Inline, four-cylinder, turbocharged, water-cooled, common rail diesel engine ω type	1600 rpm and medium load	+0.9	−1.5	−0.25		−0.5
[32]	Diesel + ethanol	Single-cylinder unmodified diesel engine	Load (rated for full load)	+0.5	+0.5	+2	+2.5	−20.5
[33]	Diesel + ethanol	4-cylinder common-rail diesel engine	Variable engine speeds	+7.5	+6.726	+25	+31.5	+1.8
[34]	Diesel + ethanol	4-stroke diesel engine	Variable engine speeds	+6.37%	+6.72	−34.69	+44.5	+34.5
Recent study	Diesel + ethanol	4-cylinder, direct injection diesel engine	Variable loads at fixed engine speeds (1500 rpm)	−12	+18.3	−68.75	−62	−21.6

Note: BTE: Brake thermal efficiency; BSFC: Brake specific fuel consumption (kg/kW h); CO: Carbon monoxide (vol.%); HC: Unburned hydrocarbons (ppm); NOx: Nitrogen oxides (ppm).

It should be emphasized that the calculation of all relative variations has been performed against the value presented in the reference literature. If the relative variation is positive, the blends analyzed in this research yield values greater than that of pure diesel fuel. Otherwise, the relative variation is negative, meaning that its value was lower. The comparison has been carried out using the specific data on the same available operation parameters: specific fuel consumption (BSFC), brake thermal efficiency (BTE), or emission concentrations. In case the values have been presented in the reference papers at different loads, speeds, and/or blend proportions, the comparison was made in the form of a trend.

This study (i) compares engine operation with fixed diesel injection timing to operation with injection timing optimized for each diesel–ethanol blend and shows that, for the tested engine, much of the BSFC and BTE penalty reported in the literature for ethanol–diesel operation can be substantially reduced when blend-specific optimum timing is applied, particularly at medium–high loads; and (ii) explicitly maps misfire limits as a function of ethanol volume fraction and load, demonstrating that practical, misfire-free operation in this configuration is limited to ethanol contents of ≤25 vol. Moreover, the detailed fuel property data together with comprehensive performance and emission measurements provide a mechanistic understanding of the nonmonotonic NOx response, which arises from the interplay between charge cooling and improved premixed combustion. Methodologically, this implies that future research should employ blend-specific calibration and explicit misfire mapping. Practically, it indicates that ethanol blending windows can be defined that limit CO and HC emissions in a reasonable manner without compromising combustion stability.

4 Conclusions

Ethanol-diesel blends have the potential to lower greenhouse gas (GHG) outputs, but the current commercially available ethanol only reduces emissions to a level that still necessitates additional control equipment. Moreover, ethanol tends to slightly increase brake-specific fuel consumption (BSFC). Due to their lower cetane rating, these blends require the use of cetane enhancers to maintain reliable performance, and the need for emulsifiers to keep ethanol mixed with diesel also drives up fuel costs. The effect of ethanol-diesel fuel blends on brake thermal efficiency can only be achieved by optimizing the fuel ratio and fuel injection timings, which have the best results under conditions of medium to high loads; otherwise, ethanol's increased latent heat energy may decrease the efficiency.

Nonetheless, incorporating 10% ethanol (by volume) into diesel effectively reduces hydrocarbon emissions by 6.3%, carbon monoxide by 10%, and NOx by 19.73%, along with a 23.4% reduction in engine noise. The results indicate an increase in BSFC; however, the diesel BSFC declined, and a significant improvement in thermal efficiency was observed when operating on these blends. Furthermore, NOx emissions decreased when these ethanol blends were used in diesel engines. It should be noted that the study dealt with short-term operational stability rather than long-term storage stability.

Future research efforts will involve measurements of smoke opacity and particulates since this aspect was not covered in this study due to the limitations of the current measurement setup, which could only measure gaseous emissions, engine performance, and noise at steady state.

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Abbreviations

BMEP	Brake mean effective pressure (kN/m^2)
BP	Engine brake power (kW)
BSFC	Brake specific fuel consumption (kg/kW h)
BTE	Brake thermal efficiency (%)
CA	Crank angle (°)
CIE	Compression ignition engine
CO	Carbon monoxide (vol. %)
CO_2	Carbon dioxide (vol. %)
DI	Direct injection
HC	Unburned hydrocarbons (ppm)

ID	Ignition delay (°CA)
KOH	Potassium hydroxide
LHV	Lower heating value (MJ/kg)
NOx	Nitrogen oxides (ppm)
OIT	Optimum injection timing (°BTDC)
TDC	Top dead center

Symbols

$\frac{\partial R}{\partial V_i}$	The measured result sensitivity for a single variable
e_i	Uncertainty interval in the n-th variable
e_R	Uncertainty in the results
$\dot{m}_f$	Fuel mass flow rate (kg/s)
N	Engine speed (rpm)
R	Function of the independent variable ($x_1, x_2, \ldots, x_n$)
T	Engine torque (N·m)
Q_t	Total fuel heat (kW)
v_f	Fuel flow volume
V_{sn}	Swept Volume per cycle (m^3)

Greek letters

$\eta_{bth.}$	Brake thermal efficiency
ρ_f	Fuel density

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