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Exhaust Gas Recirculation Meets Ethanol for Advancing Performance and Reduced Emissions in Diesel-Powered Compression Ignition Engines

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ABSTRACT

Mixed high-calorific-value fuels with diesel have been proven to enhance compression engine performance and may reduce the environmental impact of exhaust gases. However, the technique will have added value if the added fuels are bio-sourced. A 10% E10 (10% ethanol, 90% diesel) fuel blend combined with 10%, 20%, or 30% EGR was used in a four-cylinder direct-injection diesel engine. To overcome the reduction in the evaporation rate and the cetane number of the ethanol blend (because of ethanol's low cetane number), a solubilizer and cetane number improver were added to enhance the fuel's performance. Running the engines with bioethanol-diesel blends and exhaust gas recirculation (EGR) reduced nitrogen oxide emissions and made them quieter. Nevertheless, the hydrocarbon (HC) and carbon monoxide (CO) emissions were increased by 24.5% and 18.8%, respectively, at 20% added EGR. Because ethanol was present, EGR could be more rigorous without causing significant increases in HC and CO. These findings indicate that blending bioethanol with diesel fuel combined with EGR can enhance diesel engine performance by reducing noise, improving efficiency, and lowering emissions; however, further improvements are needed to reduce HC and CO emissions more effectively.

1. INTRODUCTION

Environmental and health matters led to a stronger push to lower Greenhouse gas (GHG) emissions and use fewer fossil fuels. Consequently, countries are adopting more stringent emissions rules to address the damage caused by vehicle pollution. Because of these rules, companies are now looking for new ways to reduce harmful exhaust and make engines more efficient when burning fuel. While diesel engines are known for better fuel economy and greater power, their emissions of nitrogen oxides and particulates continue to pose significant challenges, hindering their wider acceptance and further development [1].

A possible strategy is the use of blended fuels made from diesel and oxygenate, which can help lower emissions from current diesel engines with minimal adaptation. Many experiments have demonstrated that oxygenated diesel can reduce smoke emissions without increasing NOx emissions [2]. Lately, biomass-based fuels have caught the attention of people worldwide. Ethanol (C_2H_5OH) is a renewable biofuel derived from corn, sugarcane, and other biomass sources. Nowadays, ethanol is sold in blends from neat to low quantities, which helps protect the environment. Using fermentable starches or sugar crops to produce ethanol benefits the agricultural sector by boosting the economy, supporting the environment, and benefiting rural communities [3].

Neat ethanol and ethanol blends with diesel spark ignition (CI) engines do not work well because of some unavoidable mechanical and chemical restrictions [4]. Ethanol has a low cetane number, which leads to poor ignition and an unstable flame. Ethanol does not lubricate pump parts as well as other fuels, which could lead to faster wear of injection components [5]. When water is present, biodiesel is not thoroughly mixed with diesel, complicating fuel storage and handling. Also, the presence of ethanol can degrade lubricating oil, leading to rusting within the engine. Because it is a carbon-neutral biofuel, ethanol has a low boiling point, which affects both vaporization and combustion, and the amount of water can shape both the ignition time and the types of emissions formed during bioethanol combustion. Also, testing combustion in diesel engines with a 15% ethanol-diesel blend revealed benefits and some difficulties [6].

Reducing diesel engine NOx emissions by lowering peak combustion temperatures is efficiently done with exhaust gas recirculation (EGR). Because NOx levels depend on combustion temperature, controlling that temperature is very important for emission control. Even so, adding EGR to engines typically results in more fuel consumption and greater emissions of HC, CO, and PM. Because EGR brings in more inert gases, there is less oxygen available for combustion [7, 8].

Varying the EGR ratio significantly affects combustion and engine operation. A major result is that the explosion's strength decreases, reducing the oxygen remaining in the cylinder. So, the fuel spray must be evenly distributed over a larger area to blend with oxygen before combustion [9]. This large flammable area includes not only the ideal fuel-air ratio but also higher levels of inert gases such as CO_2, H_2O, and N_2. These gases absorb heat from combustion, reducing the overall combustion temperature. Lower oxygen pressure slows the chemical reactions that produce NOx, thereby reducing NOx emissions. Changes in combustion due to EGR affect how quickly heat is released during the initial ignition phase, reducing NOx formation but possibly affecting engine efficiency and the output of other pollutants. It's important to understand these complex relationships to optimize EGR levels and achieve a good balance between lowering emissions and maintaining engine performance [10, 11].

The heat is absorbed by these inert gases (CO_2 and water vapor) in the air, slowing the reactions needed for combustion and thereby delaying its onset. As a result, more combustion occurs during the expansion stroke than during compression. Because the exposure time is shorter, fewer nitrogen oxides are formed, as their formation is influenced mainly by high temperatures [12, 13]. During combustion, recycled gases containing CO_2 and H_2O break apart into molecules at very high temperatures. As an endothermic reaction, dissociation removes heat from the system, helping decrease the flame temperature. These reactions, which happen during combustion in the engine cylinder, also help lower NOx emissions. Thus, the chemical interactions of diluents affect combustion and the emissions produced [14]. The ease with which EGR gases burn depends on their thermal properties. These two gases, CO_2 and H_2O, have specific heat capacities greater than those of oxygen (O_2) and nitrogen (N_2), which are the main components of fresh intake air. As a result, the turn-to-turn heat capacity of the charge in the cylinder increases with EGR, thereby reducing the final combustion temperature. EGR primarily reduces NOx by lowering combustion temperatures, which makes it harder for thermal NOx to form [15].

Ekaab et al. [16] carried out a performance and emissions assessment of a diesel engine fueled with various biodiesel and biokerosene additives to Iraqi diesel in hot summer. They concluded that using biokerosene caused the consumption to increase by 5.56% and 5.19% when 10 vol.% and 20 vol.% biokerosene were added, respectively. However, the engine's exhaust-gas temperatures and thermal efficiency decreased. The biokerosene blends KB10 and KB20 also emitted lower concentrations of particulate matter (22.4% and 25.63%), hydrocarbon (7.74% and 21.93%), and carbon monoxide by 15% and 20.31%, respectively, compared to diesel at small or medium engine loads. Nitrogen oxide concentrations increased by 2.11% and 4.57% with KB10 and KB20, respectively. The PM–NOx trade-off for biokerosene was the best among all tested blends.

Although the addition of bioethanol to diesel fuel and the use of EGR have each demonstrated the possibility of lowering emissions and enhancing the performance of diesel engines, the results of combining the effects of the optimization of bioethanol-diesel blends with the different proportions of EGR to the final blends on engine performance and emissions have never been experimentally reported exhaustively under various operating conditions.

This study aims to fill this gap by systematically investigating bioethanol-diesel blends at varying EGR rates to assess their effects on combustion characteristics, emissions (NOx, HC, and CO), thermal efficiency, and noise levels in a multi-cylinder direct-injection diesel engine. The originality of the research lies in combining cetane number improvers and solubilizers to enhance the stability of fuel blends and combustion performance. Extensive tests of injection timing impacts under different EGR conditions provide new insights into a feasible strategy for optimizing cleaner, quieter, and more efficient diesel engine operation using sustainable biofuel blends.

2. RESEARCH METHODOLOGY

2.1 Materials and properties

For this study, we used traditional Iraqi diesel fuel with a high sulfur content (exceeding 10,000 ppm) and a cetane number of 46.9, along with bioethanol derived from Iraqi dates, with a purity of 99.7%. The required amounts of diesel fuel, bioethanol, solubilizer, and cetane number booster were purchased. Prior research by the author led to the preparation of a specific bioethanol-diesel blend just before conducting the experiments. Notably, the tests were performed on diesel engines using pure diesel and a 10% bioethanol blend (E10). To evaluate how bioethanol impacts engine performance and emissions, properties of the bioethanol-diesel mixtures were analyzed and compared with those of the standard diesel fuel. A concise summary of the test fuels is provided in Table 1.

Table 1. Fuel properties for diesel and ethanol

Property	Diesel	Ethanol	E10
Chemical formula	$C_{10.8}H_{18.7}$	C_2H_5OH	-
Molecular weight (g)	148.3	46.1	138.08
Density (g/cm^3 at 20 °C)	0.84	0.789	0.835
Boiling point (°C)	180-330	78	214.35
Heat of evaporation (kJ/kg)	280	856	337.6
Lower heat value (MJ/kg)	42.5	27.0	40.95
Liquid viscosity (cP at 20 °C)	3.03	1.2	2.847
Surface tension (mN/m at 20 °C)	34.1	28.9	33.56
Flash point (°C)	78	13.5	71.55
Stoichiometric air fuel ratio	14.4	9	13.86
Cetane number	45	5-8	41.15
Auto-ignition (°C)	235	423	444.15
Carbon content (wt%)	87.4	52.2	83.88
Oxygen content (wt%)	0	34.3	3.43

2.2 Experimental setup

The experimental engine was a direct-injection, water-cooled, four-cylinder, in-line, naturally aspirated FIAT diesel engine, with its key specifications outlined in Table 2. It was connected to a hydraulic dynamometer, which applied load by increasing torque. A photograph of the tested engine is presented in Figure 1.

Figure 1. Photo of the experimental setup used in the tests

In this study, an external EGR system was employed, which channels exhaust gases back into the intake system. The EGR ratio, representing the amount of recirculated exhaust gas relative to the intake charge in an engine cylinder, was determined using a specific equation.

$$EGR = \frac{\dot{m}_{EGR}}{\dot{m}_{air} + \dot{m}_{EGR}} \quad (1)$$

where, $\dot{m}_{EGR}$ is the flow mass rate of EGR, and $\dot{m}_{air}$ is the flow mass rate of fresh air. The EGR flow mass rate was measured by using a D.P. Engineers (US-made) pitot tube-based flow meter.

Table 2. Tested engine specifications

Engine Model	TD 313 DIESEL Engine Reg
Engine type	4cyl., 4-stroke
Combustion type	DI, water cooled, naturally aspirated
Displacement	3666 cc
Valve per cylinder	two
Bore	100 mm
Stroke	110 mm
Compression ratio	17:1
Fuel injection pump	Unit pump (26 mm diameter plunger)
Fuel injection nozzle	10 nozzle holes, nozzle hole dia. (0.48 mm), spray angle = 160°, nozzle opening pressure = 4 0 MPa

2.3 Performance formulation

The following equations were used in calculating engine performance parameters [16]:

Brake power (BP) (kW):

$$bp = \frac{2\pi * N * T}{60 * 1000} \quad (2)$$

Brake mean effective pressure (kN/m^2):

$$bmep = bp \times \frac{2 * 60}{V_{sn} * N} \quad (3)$$

Fuel mass flow rate (kg/sec):

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad (4)$$

Air mass flow rate (kg/sec):

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \left(\frac{kg}{sec}\right)$$
$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60 * 2} \times \rho_{air} \left(\frac{kg}{sec}\right) \quad (5)$$

Brake specific fuel consumption (kg/kW·hr):

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad (6)$$

Total fuel heat (kW):

$$Q_t = \dot{m}_f \times LCV \quad (7)$$

Brake thermal efficiency (%):

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 \quad (8)$$

2.4 Instruments and measurements

To measure emissions, we used a multi-gas mode 4880 analyzer, which assesses the concentrations of nitrogen oxides (NOx) with uncertainty of 0.66%, unburned total hydrocarbons (HC) with uncertainty of 0.72%, carbon dioxide (CO_2) with uncertainty of 0.49%, and carbon monoxide (CO) with uncertainty of 0.65%. This device utilizes an infrared light beam generated by a transmitter that passes through the measuring elements. The gases within the measuring cell absorb different wavelengths of light based on their concentrations, as CO, CO_2, and HC have specific absorption spectra due to their molecular structures. Additionally, the analyzer features a chemical sensor that measures oxygen concentration. An overall sound pressure level was evaluated using a precision sound level meter equipped with a microphone of type 4615, made in Italy, with an uncertainty of 1.15%. The device was calibrated with a standard calibrator type Pisto Phone 4220, measuring sound pressure in decibels (dB). The overall uncertainty was 1.714, indicating high test accuracy.

2.5 Experimental procedure and research variables

To assess the influence of EGR on engine efficiency and emissions under various engine conditions, tests were conducted with pure diesel fuel to establish baseline data. The engine was first heated until the cooling water reached 70 ℃, a state maintained for approximately 30 minutes. Performance measurements for the E10 blend were then taken and documented. Subsequently, hot EGR was introduced at 10%, 20%, and 30% to assess its effects on engine performance and emissions. These specific percentages were selected to represent low, medium, and high dilution levels. Emissions produced from the exhaust gases were measured and compared with those generated from pure diesel fuel. In a second phase of experiments, the engine speed was varied between 1000 and 2500 rpm in increments of 250 rpm to analyze how speed changes affected performance and emissions. Two loads, 50% and 100% of the maximum torque that can be achieved at each speed, were tested. After establishing the load percentage at each speed without EGR, the fuel pump rack position was held constant while EGR levels were adjusted using the EGR valve. Four EGR mass ratios of 0%, 10%, 20%, and 30% were examined in each test. The fuel injection timing was consistently set at 38° BTDC during these initial tests. In the third series of tests, the injection timing was varied while maintaining a steady engine speed of 1500 rpm. Additionally, it is important to mention that when EGR was introduced, the dynamometer slightly adjusted the engine load to maintain the desired speed. The results from the various tests were consistent, indicating that the experiments could be reliably repeated.

3. RESULTS AND DISCUSSION

3.1 Performance analysis

3.1.1 Performance analysis under varied engine loads

Figure 2 illustrates how EGR utilization with E10 impacts volumetric efficiency. The addition of ethanol enhanced volumetric efficiency due to its higher oxygen content. However, the inclusion of EGR resulted in a decline. EGR works by substituting some of the fresh air's oxygen and nitrogen entering the combustion chamber with carbon dioxide and water vapor from the exhaust. Recirculating a portion of exhaust gases back into the intake reduces the oxygen level in the intake air.

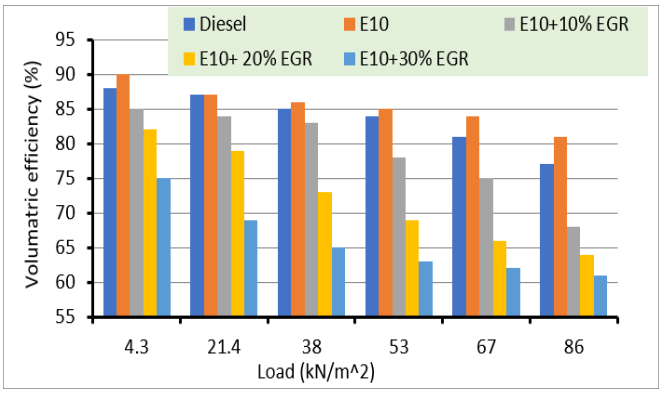

Figure 2. Exhaust gas recirculation (EGR) effect on volumetric efficiency for variable engine loads

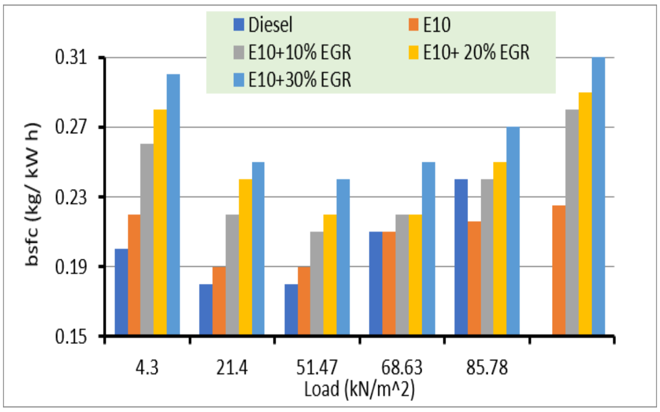

Figure 3. Exhaust gas recirculation (EGR) effect on brake-specific fuel consumption (BSFC) for variable engine loads

Figure 3 depicts the effects of EGR on E10 compared to diesel fuel. As anticipated, the brake-specific fuel consumption (BSFC), measured in kg/kWh, increased with the addition of ethanol due to its lower energy content. The enhancements in BSFC at higher loads were marginal and likely stemmed from improvements in thermal cycle efficiency.

When EGR was used, there was a significant increase in BSFC, as substituting some of the fresh air with recirculated exhaust gases reduced BP output, necessitating a higher fuel-air mixture to achieve the desired load while maintaining a constant engine speed of 1500 rpm. The use of E10 led to an increase of approximately 3.96% in BSFC when compared to diesel. Incorporating EGR at 10%, 20%, and 30% resulted in BSFC increases of about 19.8%, 23.7%, and 38.6%, respectively.

The oxygen in the ethanol blend can enhance combustion and thermal efficiency, particularly at high loads. However, operating the engine with EGR negatively impacted combustion by increasing ignition delays, decreasing mixture temperatures, and lowering oxygen levels, as shown in Figure 4.

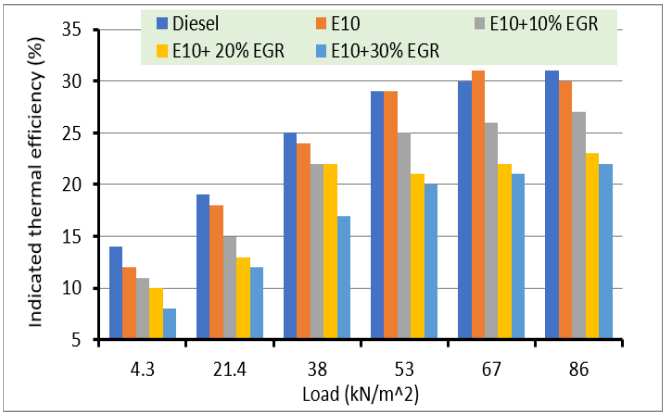

Figure 4. Exhaust gas recirculation (EGR) effect on indicated thermal efficiency for variable engine loads

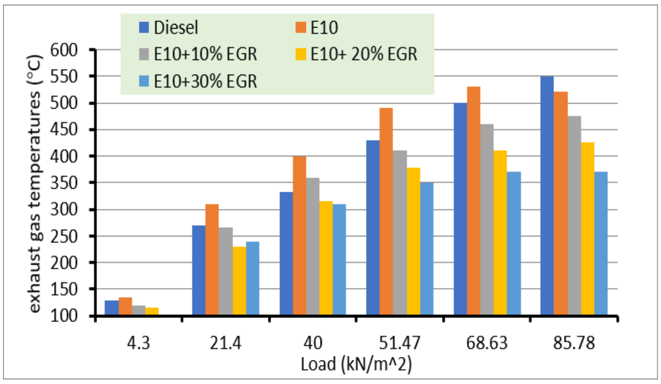

Figure 5. Exhaust gas recirculation (EGR) effect on exhaust gas temperatures for variable engine loads

The addition of 10% bio-ethanol resulted in a slight reduction of 2.7% in indicated thermal efficiency. In contrast, applying EGR resulted in efficiency drops of 14.86%, 25%, and 32.4% at 10%, 20%, and 30%, respectively. EGR replaces some of the fresh air's oxygen with exhaust gas from the engine, which lowers the oxygen concentration. This reduction reduces combustion temperatures, thereby decreasing exhaust gas temperatures, as illustrated in Figure 5. Figure 6 illustrates NOx emission levels for both diesel and E10 fuels with and without EGR. The extent of NOx reduction was greater under higher loads. The decrease in NOx emissions when using EGR in diesel engines can be attributed to reduced oxygen concentration and lower combustion chamber temperatures. However, E10 without EGR produced higher NOx emissions than diesel at high loads due to elevated combustion-chamber temperatures.

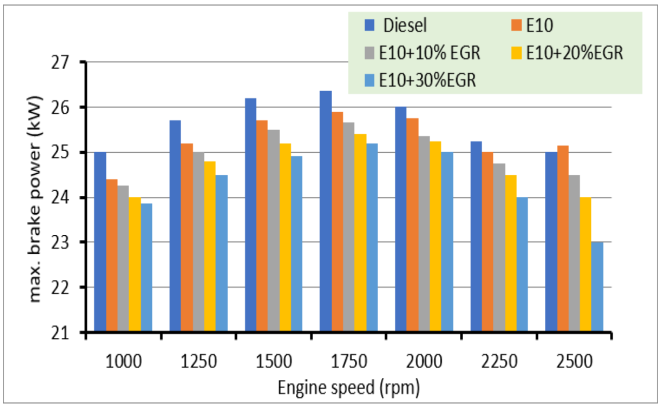

Figure 6. Exhaust gas recirculation (EGR) effect on engine maximum brake power (BP) for variable engine speeds

3.1.2 Performance analysis under varied engine speed

Figure 6 illustrates how EGR influences the maximum BP of an engine under varying loads. The combination of EGR flow in the multi-cylinder engine system and the use of E10, which contains a high level of oxygen, results in higher BP at mid-range speeds between 1,500 and 2,250 rpm. The adverse impacts of increased EGR on maximum BP are closely tied to engine speed. Specifically, when the engine runs at either low or excessively high speeds, the changes in the mixture composition due to EGR significantly affect performance. This correlation is particularly evident in the increase in ignition delay associated with EGR use.

Figure 7 depicts the influence of EGR on minimum brake specific fuel consumption (BSFC) across different engine speeds. It was found that BSFC rose by 6.52%, 10.33%, and 13.55% for EGR rates of 10%, 20%, and 30%, respectively, in comparison to diesel fuel.

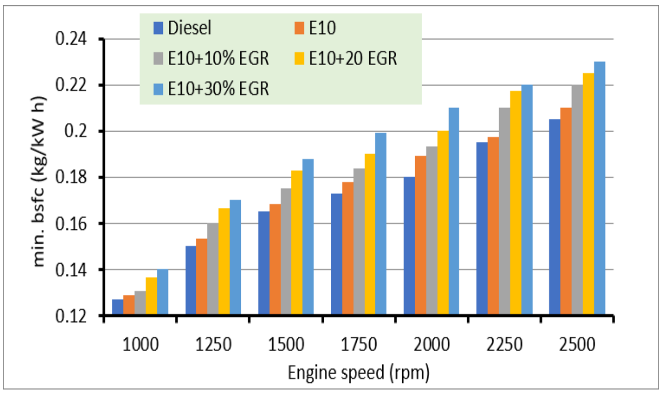

Figure 7. Exhaust gas recirculation (EGR) effect on minimum brake specific fuel consumption for variable engine speeds

3.1.3 Performance analysis under various injection timing

Figure 8 illustrates how injection timing affects the maximum BP across five test modes. Using ethanol-diesel mixtures can lead to a slight advance in injection timing, improving engine efficiency compared to standard diesel. This improvement occurs because the diffusion flame speed increases, offsetting the delayed combustion onset caused by prolonged auto-ignition periods. The analysis of combustion timing indicates that significant adjustments to the injection system are unnecessary when using blends containing 10% or less bioethanol. These findings align with the existing literature and imply that incorporating cetane boosters and co-solvent additives to improve blend stability, along with a reconfiguration of the injection approach, could further enhance performance and emissions. When the injection timing advances from 38° BTDC to 45° BTDC, the resulting bp improves by 14%, 9.23%, and 5.6% for EGR rates of 10%, 20%, and 30%, respectively.

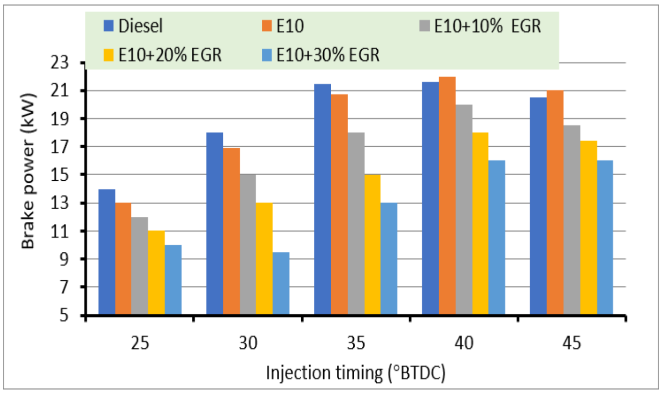

Figure 8. Exhaust gas recirculation (EGR) effect on brake power (BP) for variable engine injection timing

As shown in Figure 9, advancing injection timing significantly reduces BSFC. Ignition delay, a critical factor in the combustion process, is the duration (or crank angle) from the start of fuel injection to the point at which heat release becomes detectable. Increasing the mixing time of the direct-injected fuel and fresh oxygen is achieved by reducing oxygen levels. This adjustment aims to prolong the ignition delay and decrease the combustion rate once diffusion combustion begins, resulting in lower maximum BP and higher minimum BSFC with elevated EGR ratios. Moreover, advancing injection timing provides more time for fuel and air to mix, leading to combustion at higher temperatures and higher NOx levels, even with EGR in use.

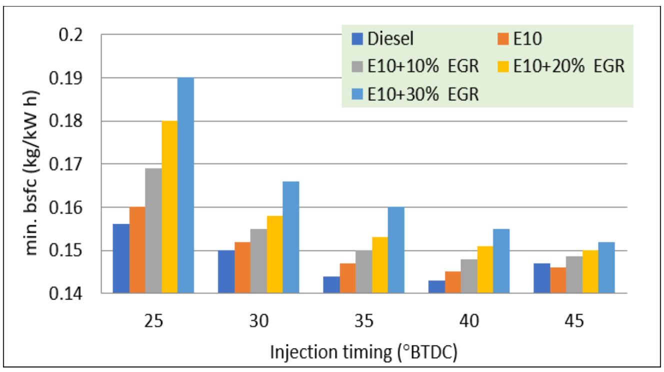

Figure 9. Exhaust gas recirculation (EGR) effect on brake-specific fuel consumption (BSFC) for variable engine injection timing

3.2 Emission and environmental analysis

3.2.1 Emission and environmental analysis under varied engine load

From Figure 10, it is evident that introducing EGR at 10%, 20%, and 30% resulted in lower NOx emissions of approximately 13.27%, 33.54%, and 45.27%, respectively, compared to diesel fuel without EGR. These findings demonstrate that EGR effectively manages NOx emissions. Additionally, EGR contributed to increased HC and CO emissions, as seen in Figures 8 and 9. Excessive dilution from EGR's inert gases leads to poor combustion quality, partial burning, and even misfires, resulting in significantly higher HC and CO emissions. Thankfully, E10's high oxygen content helps mitigate these increases.

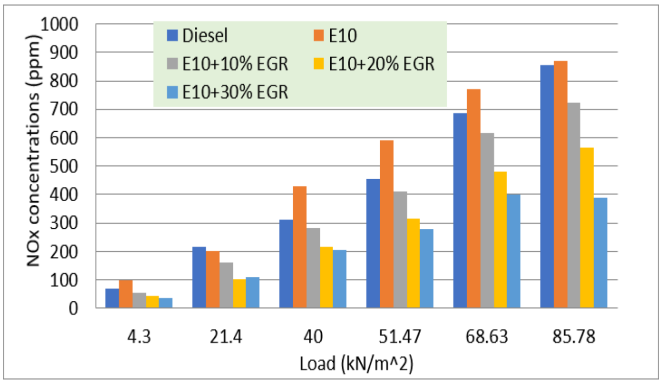

Figure 10. Exhaust gas recirculation (EGR) effect on NOx concentrations for variable engine loads

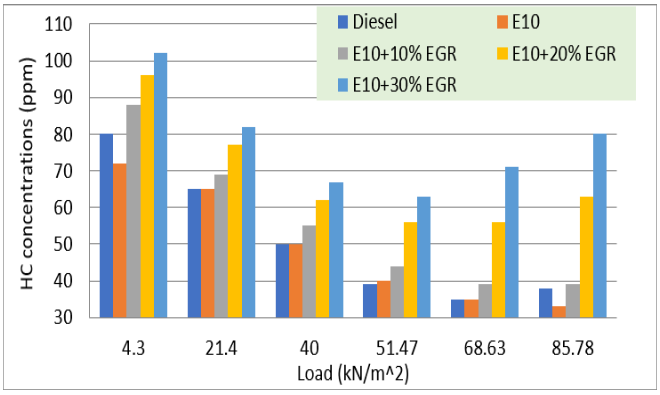

Figure 11. Exhaust gas recirculation (EGR) effect on unburnt hydrocarbon concentrations for variable engine loads

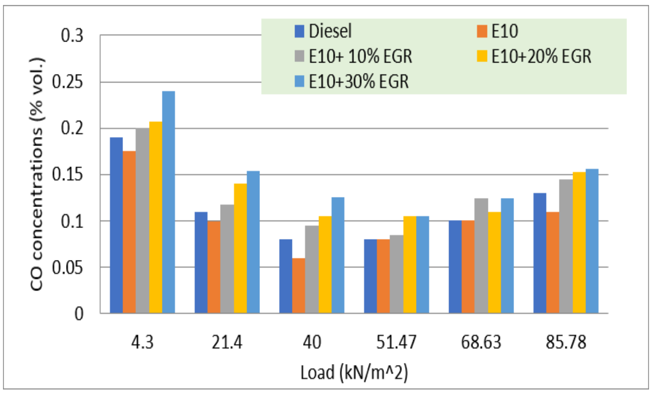

Figure 12. Exhaust gas recirculation (EGR) effect on carbon monoxide (CO) concentrations for variable engine loads

Figure 11 indicates that the penalties due to HC concentrations were increased by 8.79%, 24.5%, and 31.46% for 10%, 20%, and 30% EGR, respectively, compared to diesel. Notably, the use of E10 without EGR resulted in a 3.9% reduction in HC concentrations. CO emissions increased with the EGR ratio, as shown in Figure 12.

The increase was around 31% with 30% EGR, while the lowest EGR ratio of 10% led to a minimal rise in CO emissions of just 2.4%. On the other hand, the EGR ratio of 20% increased emissions by 18.8%. E10 showed a 9.35% reduction in CO concentrations compared to diesel fuel, highlighting the benefits of its higher oxygen content. The noticeable increases in CO and HC emissions correlate to a decrease in CO_2 emissions, as shown in Figure 13. Both diesel and E10 resulted in similar CO_2 levels. EGR application decreased the resulting CO_2 concentrations by roughly 18.18%, 26.1%, and 32.6% for the 10%, 20%, and 30% configurations, respectively.

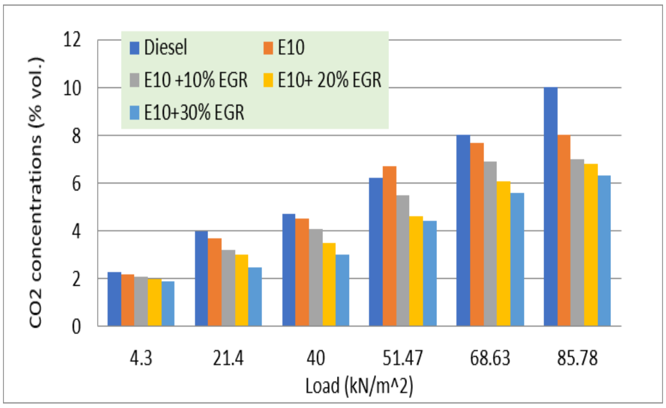

Figure 13. Exhaust gas recirculation (EGR) effect on carbon dioxide (CO_2) concentrations for variable engine loads

Noise, a byproduct of combustion, can directly impact those nearby. It often leads to immediate discomfort and changes in body response. There are two types of combustion noise: direct and indirect. The direct noise originates from areas with turbulent combustion, resulting from temporary changes in the heat release rate. Although the fluctuations are minor, they produce pressure waves. Conversely, indirect noise forms further along the flow, arising from interactions between different temperature streamlines. Depending on the engine's design, either type of noise may be more prominent. It is well recognized that, in diesel engines, both the pressure-time characteristics and combustion-related turbulence play crucial roles in noise production [17].

Employing EGR in dual-fuel engines is a promising strategy to reduce NOx emissions. However, EGR alters the combustion rate and the pressure rise inside the combustion chamber, which is linked to another harmful pollutant: combustion noise. Diesel engines are known to generate significantly more noise than spark-ignition engines. Noise travels through the engine block as vibrations, which the human ear can detect across various frequencies. Besides airflow and mechanical vibrations, combustion noise is a primary noise source, especially in engines with high compression ratios, where combustion pressure increases rapidly. A key factor influencing combustion noise is the rate of pressure rise during combustion. Studies have shown a direct relationship between the maximum pressure rise rate and the sound pressure level (SPL) in decibels in the main chamber of a diesel engine [18, 19].

Figure 14 illustrates how EGR influences engine noise across different engine loads. By raising the EGR rate, engine noise can be reduced by approximately 8.8%, 13.6%, and 16.6% for EGR levels of 10%, 20%, and 30%, respectively.

Experimental findings validate that the maximum cylinder pressure diminishes as the EGR ratio increases. The exhaust gas contains a mixture of combustion byproducts, primarily CO_2 and H_2O, with smaller amounts of CO, NOx, hydrocarbons, and particulate matter. Increasing the concentration of spent gases in the intake air results in lower oxygen levels, leading to longer ignition delays. Consequently, combustion occurs later, and the combustion rate slows. This drop in combustion efficiency causes the maximum cylinder pressure to decrease as EGR levels rise, thereby significantly reducing engine noise [20, 21].

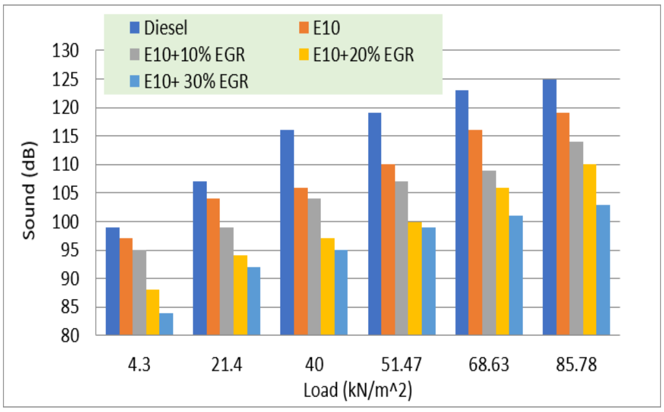

Figure 14. Exhaust gas recirculation (EGR) effect on engine noise for variable engine loads

3.2.2 Emission and environmental analysis under varied engine speed

The relationship between maximum NOx emissions and EGR rates across various engine speeds is demonstrated in Figure 15. In all three tested conditions, NOx concentration decreased as EGR levels increased. Notably, at high speeds, the initial emission and the rate of NOx reduction are significantly higher than under the other conditions. Therefore, two key factors influence NOx levels: the dilution effect of EGR and the added oxygen resulting from ethanol. The figure indicates that ethanol's presence does not alter EGR's effectiveness in reducing NOx emissions, which decreased by 13.77%, 17.18%, and 21.69% at EGR rates of 10%, 20%, and 30%, respectively.

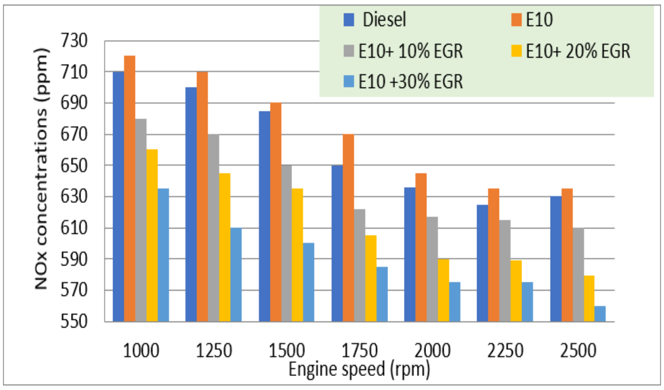

Figure 15. Exhaust gas recirculation (EGR) effect on NOx concentrations for variable engine speeds

In addition to NOx output, emissions of unburned HC and CO were analyzed and shown with respect to engine speeds in Figures 16 and 17. HC levels rose with the addition of EGR at both low and high speeds, indicating a deterioration in combustion quality. A significant surge in CO levels occurred at 2000 rpm and higher. These dramatic increases in CO and HC are notably pronounced with more than 10% EGR. The primary effect of EGR is a reduction in oxygen levels, which increases CO and HC emissions.

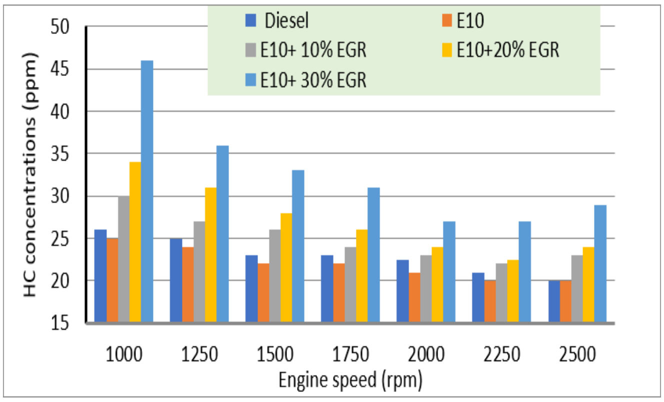

Figure 16. Exhaust gas recirculation (EGR) effect on hydrocarbons (HC) concentrations for variable engine speeds

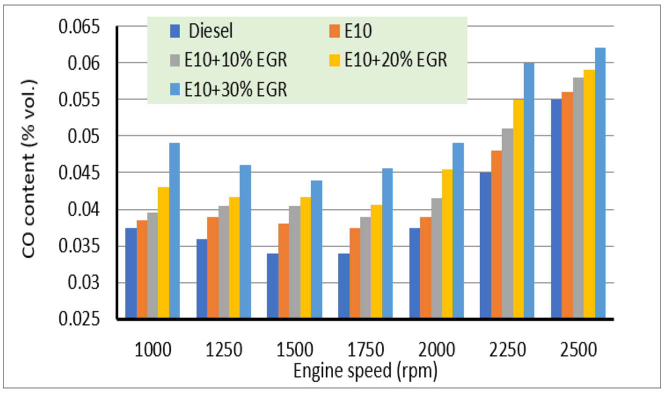

Figure 17. Exhaust gas recirculation (EGR) effect on carbon monoxide (CO) concentrations for variable engine speeds

3.2.3 Emission and environmental analysis under various Injection timings

Figure 18 clearly shows how varying injection timings affect NOx concentrations across different EGR conditions. Additionally, the experimental data shown in Figures 17 to 20 indicate that the presence of inert gas in the intake manifold and low oxygen levels extend the ignition delay when exhaust gases are recirculated.

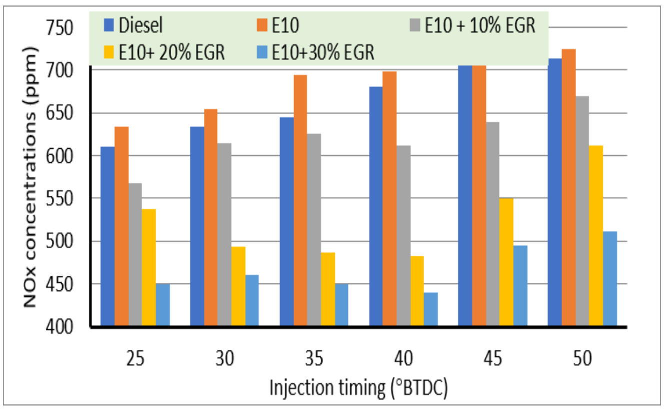

Figure 18. Exhaust gas recirculation (EGR) effect on NOx concentrations for variable engine injection timing

As EGR ratios increase, the ignition delay increases. The trends of HC and CO concentrations with injection timing (illustrated in Figures 19 and 20) parallel those observed with engine speeds, suggesting that the effects of EGR are more pronounced than those of the engine parameters.

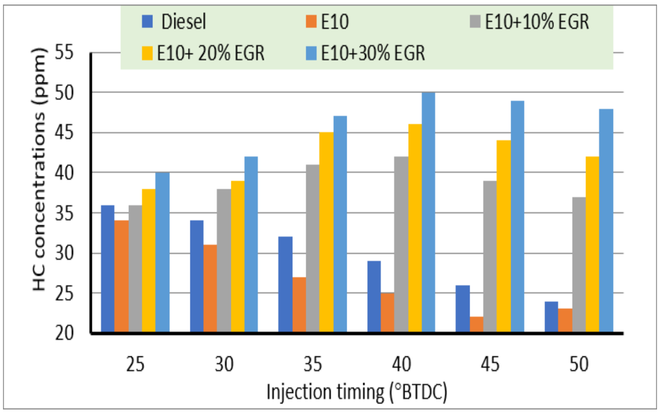

Figure 19. Exhaust gas recirculation (EGR) effect on hydrocarbons (HC) concentrations for variable engine injection timing

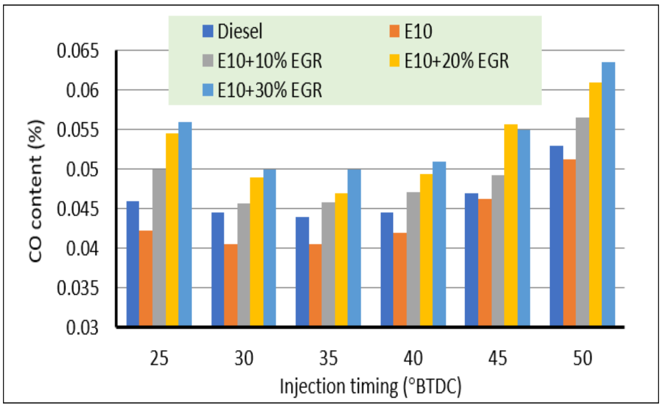

Figure 20. Exhaust gas recirculation (EGR) effect on carbon monoxide (CO) concentrations for variable engine injection timing

3.3 Comparison with previous works

When comparing studies that use different fuels and engine types, certain caveats and advantages should be considered. Among the more important caveats in making such comparisons is that various types of engines and uses of technology (diesel, gasoline, and other fuels) result in differences in the combustion of the fuel and the amount of work done by the engine compared with the amount of fuel consumed, which generally would make direct comparisons unfair. Furthermore, varying conditions such as engine speed, engine load, and fuel characteristics (chemical composition, quality, gasoline octane rating, and other fuels' cetane rating) also tend to affect the results, making genuine comparability difficult. Also, there will be differences in the types of measuring devices used for the actual measurements, as well as in analysis techniques. Then there will be differences in engines, control devices, and fuel characteristics, so there will also be differences in efficiency and emissions in all cases, even though the same fuel is used in the engine being researched as in the other cases. So that, in making comparisons without considering the results of these considerations, very misleading and erroneous conclusions can be drawn. As to some of the advantages connected with such comparisons, however, it may be pointed out that they give a reasonably general and comparative presentation of the better ways shown by the different fuels and types of engines in terms of efficiency and emissions under various conditions. They would also provide an idea of some of the general trends that yield the best results from an economic and environmental standpoint. They would also give a broader view of the types of engines and their operating variations when alternate fuels are considered. It might provide a sort of background for future work that this comparison might yield, based on the weaknesses identified in previous research. It may be seen as helping in the work to evolve sound remedies that produce minimal emissions and maximize fuel efficiency, once the differences and advantages of the various systems are understood. Table 3 compares the current study's findings with published literature.

3.3.1 Bio-blend diesel effect on engine performance

The effect of EGR on BTE and BSFC, as shown in Table 3, is complex and does not exhibit consistent trends across fuel and engine conditions. The effects of EGR on brake thermal efficiency have not been consistent in the literature, but have evidently depended to a large degree on the fuel mixture and operating conditions. Cases such as [15, 17, 22, 23, 24] have shown an improvement in brake thermal efficiency with EGR. Case [25] shows an effect that is adverse to the brake thermal efficiency in this respect. In other cases, the present study shows a new BTE effect with EGR of 29% more BTE at maximum load with EGR.

The trend in BSFC appears more uniform in previous studies. Since BSFC is the inverse of BTE, or the poorer engine requires more fuel for the same power output, the trend is usually opposite to that for the BTE. Cases such as [25] show a rise in BSFC with EGR (which means an unproven efficiency). In the present study, the MSFC increase at maximum load is approximately 46.66%, indicating a considerably greater increase in fuel consumption at the same power output. The rise in input BSFC with EGR suggests that, in some cases, more fuel is required to achieve the same output. On the other hand, the results of specific experiments, particularly those in the studies of Elkelawy et al. [23] and Kalwar et al. [24], show a significant decrease in BSFC, indicating that the proper fuel is used and that combustion is more complete, resulting in improved efficiency of the combined engine.

3.3.2 Bio-blend-diesel impact on environmental consequences

The data shown in Table 3 allow consideration of the effects of EGR on engine emissions and their resultant environmental costs, in terms of trade-offs. These effects do not occur in the same way across all fuels. It is apparent from the data that generally, though not in every case, there is a substantial and uniformly favorable decline in nitrogen oxides, because every study cited in which there was a reported effect of EGR on NOx shows a decline. For instance, Abdul Wahhab and Al-Kayiem [9] reported that cold EGR reduced NOx by 57.1% and 42.8% with hot EGR, whereas in the study under consideration, the maximum NOx reduction was 54.5%. The cause of the reduction in NOx is a decrease in peak combustion temperatures and a reduction in in-cylinder oxygen content, the two major factors in NOx formation. This is a very favorable environmental effect because NOx is the chief contributor to smog, acid rain, and respiratory diseases. This is the primary environmental benefit of EGR use. On the other hand, at lower temperatures and in the absence of oxygen, incomplete combustion can be greater, resulting in a higher carbon dioxide content. However, it is observed that using some oxidizing fuels (i.e., D70B20E10 [23] or DEE30 [23]) can compensate for the oxygen in the fuel, enabling more complete oxidation of CO to CO_2.

Table 3. A comparison between the current study findings and others from the literature

Ref.	Tests Engine	Tests Conditions	BP	BSFC	CO	HC	NOx
[9]	Single cylinder DI diesel engine	The effect of EGR temperature on the performance and emissions operating with methane in diesel at ratios of 30%, 50%, and 80%.	-	-	+50%	-	57.14% for cold EGR and 42.8% for hot EGR
[15]	Weichai Land King 6-cylinder diesel engine, model WP12, with an EGR system and two-stage turbochargers	Constant speed = 1300 rpm, fuel flow rate = 19.2 kg/h, with EGR rate controlled from 0 to 51%.	+ 6.58%	-	-	-	-
[17]	4-cylinder turbocharged diesel engine	Two engine speeds: 2000 and 2500 rpm. - Effective Mean Pressure (BMEP) at 2 and 5 bar. - EGR control. - Fuel injection pressure (rail pressure) control at multiple levels.	+8.8% when pure WCOME used at 15% EGR	+8.3% when 40% WCOME added to D. -22.2% when the engine ran with pure WCOME	-	-	+26.3 when 70% WCOME was added to D, and the EGR rate was 15%
[18]	Single-cylinder, water-cooled, direct-injection diesel engine,	- A constant speed = 1600 rpm. - At low, medium, and high loads. - EGR addition in 15%, 30%, and 40%.	+5.7 at DB30 and 40%.	+36.8 at DB45 and EGR	-11.7% at DB30 and 40% EGR	-23.07 at DB60 and 40% EGR	-65% at DB45 and 40% EGR
[26]	Perkins 854E (4-cylinder, inline, turbocharged diesel engine)	Using the ESC procedure to evaluate the engine's performance and emissions	-	-3%	-	-	-2.2% measured over the entire ESC test. +20% at engine load
[23]	Fixed-speed diesel engine	Fixed-speed, continuous load operation	+6%	-5%	-40%	-	-15%
[25]	Fiat TD 313, 4-cylinder, DI	-Constant engine Speed (1500 rpm) -EGR Rates: 10%, 20%, 30%, and from low to high BMEP	-2.6% for E10 and -31.3% at 30 EGR	+3.7% for E10 and +37.4% at 30% EGR	-8.23% for E10 and +30.6% at 30% EGR	-6.4% for E10 and +30.82% at 30% EGR	-13.3% for E10 and -43.4% at 30% EGR
[27]	AVL Diesel Engine (Single-cylinder)	Fixed engine Speed (1200 rpm), injection Pressure: 60 MPa, injection Timing: -8° to -2°CA ATDC, and EGR rate from 0% to 35%	-	-	+7.4% at ABE20	+10.7% at ABE20	+10% at ABE20
[24]	Medium-duty Tractor Engine (3-cylinder, 3298 cc)	Fixed engine Speed of 1500 rpm, variable engine load of 1.25, 2.5, 3.75, 5.0 bar, and EGR rated from 0% to 30%	+40.6 at 30% EGR and highest load	-45.9% at 30% EGR and the highest load	-34.78 at 30% EGR and the highest load	-80% at 30% EGR and the highest load	-70% at 30% EGR and the highest load
Current study	Fiat diesel engine (4-cylinder)	Adding EGR in 10%, 20%, and 30% at fixed speed and variable load, and at fixed load and variable speed. Also, injection timing variation is used with fixed engine load and speed.	29% at the maximum load operation	+46.66% at the maximum load operation	+20.3% at the maximum load operation	+120% at the maximum load operation	-54.5% at the maximum load operation

Note: BP = brake power; BSFC = brake-specific fuel consumption; EGR = exhaust gas recirculation; ESC = European Stationary Cycle; BMEP = Brake mean effective pressure.

CO is a toxic gas that reduces the blood's ability to take up oxygen. Increased carbon monoxide emissions are one undesirable effect of EGR operation in many instances. Hydrocarbon emissions are also a result of incomplete combustion, as is carbon monoxide. Hydrocarbon emissions may increase due to fuel being trapped in cracks or flame quenching caused by a charge that has become diluted. The large decrease in DEE30 may be due to the fuel's high cetane number and volatility, which promote more complete combustion even with EGR. Unburned hydrocarbons also increase ground-level ozone (smog) and contribute to air pollution.

3.3.3 Influence of EGR on CI performance and emissions

Based on the comprehensive data presented in Table 3, the impact of EGR on CI engine performance and emissions is a complex trade-off, with outcomes highly dependent on the type of fuel used. EGR is a highly effective method for significantly reducing NOx emissions, but its impact on other parameters creates a classic trade-off. The overall impact can be summarized as follows: For emissions, an apparent reduction in NOx, but with a standard increase in carbon monoxide and hydrocarbons. For performance: a common decrease in efficiency (increased BSFC, decreased BTE), but some improved fuel blends can reverse this trend.

4. CONCLUSIONS

This paper shows that adding bioethanol to diesel fuel with EGR improves engine performance and reduces specific emissions, such as NOx, by up to 45 percent. A combination of cetane improvers and solubilizers was very effective in overcoming the problems associated with ethanol's low cetane number and inadequate vaporization, allowing the engine to run reliably with minimal modification. The positive features of this technique include lower NOx emissions, reduced engine noise, and enhanced fuel vaporization, enabled by the oxygen atom in ethanol, which allows the engine to burn more efficiently at intermediate loads. Nevertheless, there are also disadvantages to the approach: higher HC and CO emissions due to incomplete combustion at high EGR ratios, and an increase in BSFC of up to 38.6% at the highest EGR level, indicating a trade-off between emissions control and efficiency. Furthermore, there is the issue of fuel stability and engine wear introduced by the reliance on ethanol blends, particularly in the absence of optimized injection timing. Further studies are needed to optimize EGR ratios, evaluate other oxygenates, and refine injection parameters to further reduce HC and CO emissions without negatively affecting fuel economy. Also, the long-term durability of the engine and its performance under real-world operating conditions should be investigated to make it applicable to industry.

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NOMENCLATURE

BMEP	Brake mean effective pressure, kN/m^2
bsfc	Brake specific fuel consumption, kg/kW·h
BTE	Brake thermal efficiency
BP	Brake power, kW
CA	Crank angle, deg
CO	Carbon monoxide
CO_2	Carbon dioxide
HC	Unburned hydrocarbons
EGR	Exhaust gas recirculation
ID	Ignition delay
LHV	Lower heating value
$\dot{m}_f$	Fuel mass flow rate, kg/sec
$\dot{m}_{a,act.}$	Actual air mass flow rate, kg/sec
$\dot{m}_{a_{theo.}}$	Theoretical air mass flow state, kg/sec
N	Engine speed, rpm
NOx	Nitrogen oxides
OIT	Optimum injection timing, msec
P	Engine power, Hp
Q_t	Total fuel heat, kJ
T	Engine torque, N·m
TDC	Top dead center
v_f	Fuel flow volume, l/min
Vsn	Swept Volume per cycle, m^3
W_R	The uncertainty in the results
$\eta_{bth.}$	Brake thermal efficiency
ρ_{air}	Air density, kg/m^3
ρ_f	Fuel density, kg/m^3