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Experimental optimization of injection parameters to enhance combustion characteristics of engine on using CE40 fuel

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Recently, the automobile sector has been moving to dual-fuel vehicles for the hazardous emissions of internal compression ignition engine to be diminished. The purpose of this experimental optimization is to learn how different configurations of injector nozzle holes, fuel injection pressure, and injection time affect the performance, emission, and combustion of a CI engine using CE40 fuel. The brake thermal efficiency has been exhibited an increase to 5.48%, accompanied by a 30.65% reduction in fuel consumption under the influence of a 22 MN/m² higher injection pressure. 44% of HC has been reduced at 29° bTDC crank angle, and 31% of CO has been controlled with the help of the five-hole nozzle. This experiment raises the NO_x emission at increased injector pressure and enhances it with a five-hole injector. However, the emissions of carbon monoxide and hydrocarbons are reduced.

Keywords: Brake thermal efficiency, Combustion engine, Indicator power, Injector nozzle hole, Injection pressure, Injection timing

Introduction

The world is focusing on clean energy to become less reliant on oil. Researchers are concentrating on emission reduction in the automobile sector. In India, the government sets many standards for protecting the environment. Concerns are growing about running out of fossil fuels and environmental damage. India, for example, gets 70% of its crude oil from outside the country. Countries that import fossil fuels are affected by price increases and shortages. India needs 4–5 times as much diesel as gasoline. Due to their limited availability and rising costs, petroleum products must be replaced with other fuels, especially diesel. Alcohol fuels, like ethanol, are easy to make and can be made again and again from plants. The cetane number, energy density, heat of vaporization, and stoichiometric air/fuel ratio of ethanol are identical to those of diesel fuel. Many scientific investigations have shown that using ethanol as a fuel alternative to diesel can significantly reduce dangerous exhaust emissions[1].

The performance and emission characteristics of a compression ignition engine were studied experimentally by Thirunavukkarasu et al. using a variety of nozzle hole sizes. They concluded that the five-hole nozzle was given better brake thermal efficiency of up to 26.19%[2]. The cylinder pressure and ignition delay were found to be reduced when the number of nozzle holes was decreased, according to the investigation report by Dong et al. Lesser NO_x and Higher UHC and CO were observed in the reduced nozzle holes. The decrease in the nozzle holes caused a decrease in the combustion efficiency[3]. Vijay Kumar et al. found that 0.20 mm orifice Nozzle Hole Diameter (NHD) reduces pollutants (HC, CO, and Smoke) while increasing NO_x. The mean gas temperature and maximum peak pressure were greater for 0.20 mm NHD compared to the standard NHD. Good atomization, cone angle increase, and fuel mixing come from smaller orifice NHD, which decreases the combustion premixing process with lower orifice NHD. Smaller NHD produces smaller droplets that evaporate and mix faster than larger ones[4]. Narayanan et al. investigated how nozzle hole numbers influenced diesel engine performance and emissions with KME combinations. Three types of Injection Hole Nozzles (IHN), fuel combines, and loads were tested. The biofuel blend and load increased brake thermal efficiency, and IHN enhanced brake thermal efficiency. Nozzle holes six and B40 had the highest Brake Thermal Efficiency (BTE) and Biofuel blend boosted Brake Specific Fuel Consumption (BSFC). Better air-fuel combination decreases BSFC with IHN. Better combustion reduces CO emissions using IHN and biofuel blends. KME oxygen concentration reduces CO with a biofuel blend. Biofuel blend boosts NO_x emissions as expected. Increased nozzle hole number minimizes fuel droplet wall impingement and NO_x emissions[5].

Parravicini et al. tested two diesel-polyoxymethylene dimethyl ether blends for combustion and emissions[5]. Increased injector nozzle perforations reduced combustion characteristics. Three nozzle holes 7, 8, and 9, for reference diesel and 22% and 42% OME, were examined, and with increased OME percentage, the rate of combustion is accelerating, and consumption of fuel decreases by 5.7 percent[6]. Walle Mekonen and Sahoo concluded that at 114 °C, Palm Oil Methyl Ester (POME) viscosity and density dropped to diesel fuel levels. Preheating POME enhanced BSFC and BTE of the engine. Improved BSFC and BTE, together with reduced CO and HC emissions from baseline diesel, were achieved with an increased Fuel Injection Opening Pressure (FIOP) of 21.2 MN/m^2 and an advanced IT of 27^o before the Top Dead Center (bTDC) in conjunction with a four-hole for preheat POME (114 °C)[7]. Saraei and Khalilarya examined engine out-combustion qualities and emission levels with different nozzle hole diameters[7]. Decreasing the holes-diameter improves fuel atomization, increasing the HRR peak. D-10% cases peaked higher than others[8].

Ganippa et al. evaluated nozzles with 0% and 20% hydro grinding (HG). Sharp-inlet nozzles have considerably lower discharge coefficients and increased turbulence than rounded-inlet ones. The fuel jet angles, ignition delay, penetration, and flame volumes produced by two nozzles with very different initial cavitation and turbulence levels but the same momentum are identical[9]. Nagaraj et al. assessed the exhaust emissions of a diesel engine incorporating a 3-Hole (3NH) and 4-Hole (4NH) nozzle with an 80% diesel/20% biodiesel. 3NH and 4NH emit NO_x under no load, while 4NH releases added CO_2 and fewer HC and CO. 4NH emits additional NO_x gas than 3NH when loaded. At 2 kg for 4NH and 8 kg for 3NH, the CO gas concentration was stabilized to 0.01 percent[10]. The influence of diesel injector nozzle dimension and shape modifications on flow and spray characteristics was calculated prediction versus actual results from experiments from novel nozzle designs by Constantin Vasconi and Roland Baar. The diameter was 70-130 µm, and there were 8-14 nozzle holes. Spray penetration and mass flow are diminished with decreased nozzle hole sizes. The 70 µm hole reduces mass flow by 70% but only 30% spray penetration compared to the 130 µm orifice[11].

Slocinski et al. changed conicity angles to create five geometric nozzles: A, B1, B2, C1, and C2. At 20 mm from the nozzle's tip, they measured the spray force. The nozzle B2 initially has a rate comparable to the other nozzle sprays, and then the B2 spray differs drastically from the other nozzle sprays[12]. Using diesel fuel, waste cooking oil biodiesel, and their blends B5, B10, B20, and B30, at fuel injection pressures ranging from 17 to 22 MN/m^2, engine speeds ranging from 1000 to 3200 rpm, and full load circumstances, Yesilyurt completed the environmental and economic studies. The findings of the environmental and economic evaluations indicated that for the remaining fuel injection pressures, B20 was the worst fuel. The initial fuel injection pressure was also significantly impacted by the use of diesel fuel. Under all conditions of engine operation, B100 outperformed the other fuels tested[13].

B.P. Singh evaluated and computed the behaviour of a diesel engine operating on biodiesel by measuring parameters such as brake-specific fuel consumption, brake thermal efficiency, CO, and NOx. Increasing the percentage of biodiesel in the blends increased the brake-specific fuel consumption and decreased the brake thermal efficiency across all injection timings and injection pressures. With increasing injection time and fuel injection pressure, however, the opposite trend was shown for these parameters[14]. D. Jagadish experimented on a diesel engine with different 18, 22, and 26 MN/m^2 injection pressures. Diesel and algae biodiesel blends were selected. Lower injection pressures and supercharging lowered the fuel used by the engine brakes. Supercharging and lower injection pressures improved the thermal efficiency of the brakes. Injection pressures increased NO_x and unburned hydrocarbon emissions due to greater maximum temperatures[15]. Srivastava et al. assessed the influence of injection pressure on diesel-acetylene engine performance, emissions, and combustion. Dual-fuel experiments with 18, 19, 20, and 21 MN/m^2 pressures and acetylene flow rates of 120 LPH were compared to normalfuel (diesel) operations. In the case of 20 MN/m^2 pressure during injection, the diesel-acetylene dual fuel mode obtained the maximum brake thermal efficiency of 27.57%, compared to 23.32% for normal diesel. Hydrocarbon and Carbon monoxide, smoke emissions were reduced, while NO_x emissions were greater in dual fuel mode at 20 MN/m^2 compared to other injection pressures and baseline diesel mode[16]. Yesilyurt and Arslan concluded that biodiesel and diesel exhibit similar patterns in terms of energy and exergy analyses. At an initial injection pressure of 19 MN/m^2, the diesel engine's maximum energy and exergy efficiencies have been identified as 24.509% and 21.275%, respectively. The biodiesel engine's greatest energy and exergy efficiencies were 22.129% and 20.052% at 21 MN/m^2 injection pressure. Increased injection pressure enhances biodiesel energy and exergy efficiency[17].

Sudershan et al. demonstrated that an injection time of 27^o bTDC, injector opening pressure of 24 MN/m^2, and a five-hole injector produce higher performance in the form of BTE with lower emissions of a tire pyrolysis oil blended fuel engine[18]. Sharma et al. evaluated the diesel engine performance with polanga biodiesel 0%, 10%, 20%, 30%, and 40% blends with injection timing up to 31^o bTDC. 20% Polanga-based biodiesel blends improved at Fuel Injection Opening Timing (FIOT) of 31^o bTDC and fuel delivery pressure of 24 MN/m^2, leading to lower emissions[19].

J. Ramachander observed that using ternary fuel in a diesel engine with increasing injection pressure up to 50 Mpa resulted in 4.39% improved brake thermal efficiency. The well-atomized spray produces a superior mixture with increased injection pressure, lowering CO, HC, and smoke levels by 22.245, 9.49%, and 7.5%, respectively[20]. Tamilselvan and Periyasamy conducted a performance test, Ceiba pentandra and Mahua biodiesel mixed with regular petroleum in volumes of 10%, 20%, 30%, and 40%. Appropriate compression ratios and loads were used to evaluate the performance of blended fuels. Efficiency measures were investigated for blended fuels with compression ratios ranging from 15 to 19. The combustion process as well as engine exhaust pollutants were investigated. Using a B20 biofuel blend with an 18-compression ratio enhanced engine performance.

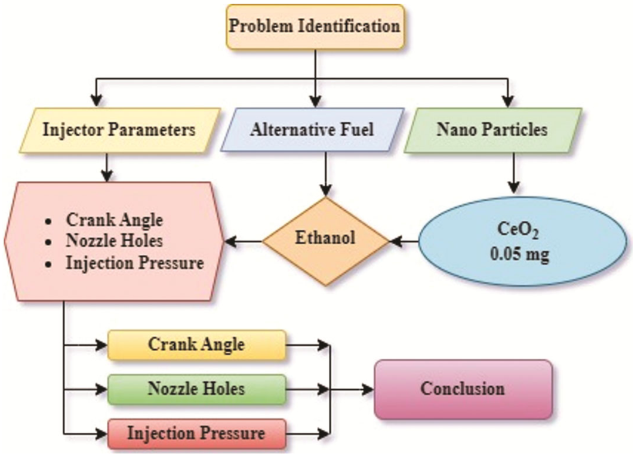
From the literature review, the authors have made many conclusions on engine performance and emission by only modifying engine components or alternative fuels. No literature report was observed on the experimental optimization of injection parameters with alternative fuels and nanoparticles combination. Hence, in this experimental investigation, the injection holes, pressure, and timing are changed, and ethanol with nanoparticles has been used as an alternative source for enhancing engine performance with lesser emission to find optimal conditions for ethanol fuel.

Experimental Section

This experiment optimization process was conducted in three phases. In the first phase, the injection angle changed up to 29^o bTDC, and the second phase was done with increasing injector nozzle holes from 3 to 5. In the third phase, the fuel spray (injection) pressure has changed from 18 to 24 MN/m^2. Scheme 1 depicts the process in its various stages.

Enriched ethanol

Nowadays, in India, ethanol is used as an alternative fuel, and ethanol mixed with fuels is available in public fuel stations to meet the scarcity of crude oil fuels and reduce the pollutants. 0.05 mg 50 ppm cerium oxide (CeO_2) additives were added to the ethanol blends to enrich the fuels. A probe sonicator was used to break down the particles of additives with the help of a pressure wave, and it avoided the quick settling of the additives[21]. The prepared sample is shown in Fig. 1, and blend properties have been evaluated; the same is shown in Table 1[22].



Scheme 1 — Scheme for experimental methodology

Injection parameters

Combustion in an engine is highly dependent on the trajectory of the fuel spray, and increasing injection holes creates a positive impact on the fuel burning rate. Three, four, and five holes are customized in the injectors. Fig. 2(a) shows the changing injection timing, 2(b) demonstrates the numerous injector options available, and 2(c) displays the injector pressure changing. The duration of the fuel burning depends on the fuel injection timing. In this experimental investigation, the injection timing varies from 16 to 32^o bTCD crank angle[23]. Fuel injection opening pressure influenced combustion phenomena such as ignition delay, fuel atomization, and fuel penetration with compressed air. Fuel injection pressure is modified in the range between 19 to 24 MN/m^2.

Table 1 — Fuel properties[22]

Properies	Petrolium diesel	Alternative ethanol
Molecular structure	$C_{12}H_{23}$	C_2H_5OH
Mass of oxygen present (%)	0	34.8
Fuel density (kg/l)	0.82	0.80
Lower heating value (kJ/kg)	42600	26790
Flash point (^{o}C)	70	15
Self ignition temperatur (^{o}C)	316	422

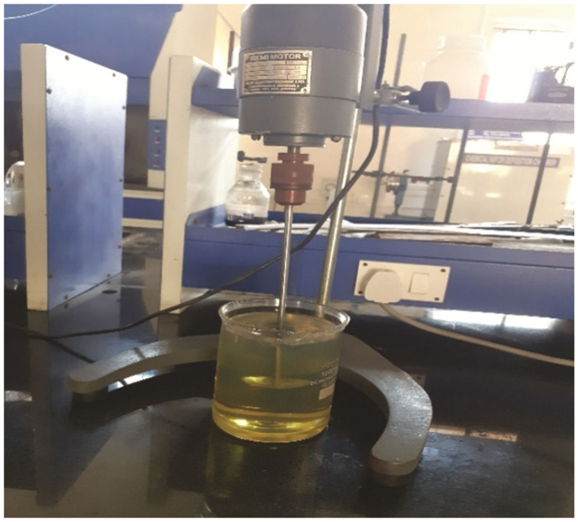


Fig. 1 — Cerium oxide,Ethanol, and Dieselblendingpreparation.

Instrumentation for an experiment

This apparatus includes an eddy current dynamometer and a single-cylinder, four-stroke, variable-compression-ratio (VCR) engine that is water-cooled. Instruments for gauging combustion pressure, crank angle, air flow, fuel flow pressure, temperatures, and load are all included. A high-speed data-acquisition device served as the interface between these signals and the computer. Technical specifications and a schematic view of the instrumentation for an experiment are depicted in Table 2 and Fig. 3.

Uncertainty and error analysis

Regular checks of engine operation and gas analyzer accuracy were performed. HC, CO_2, CO, and NO_x emissions were manually noted once the engine had stabilized. Parameter uncertainty is displayed in Table 3. Uncertainty can be measured using Eq. (1). Five replicates were conducted under identical experimental circumstances. The percentage of

Table 2 — Engine specification

Feature	Specification
Cubic capacity	662 cc
Cylinder bore diameter	87.6 mm
Stroke length	111 mm
Engine speed	1500 (Constant speed)
Compression ratio	Variable from 12.1 to 18.1
Engine capacity	3.5 kW

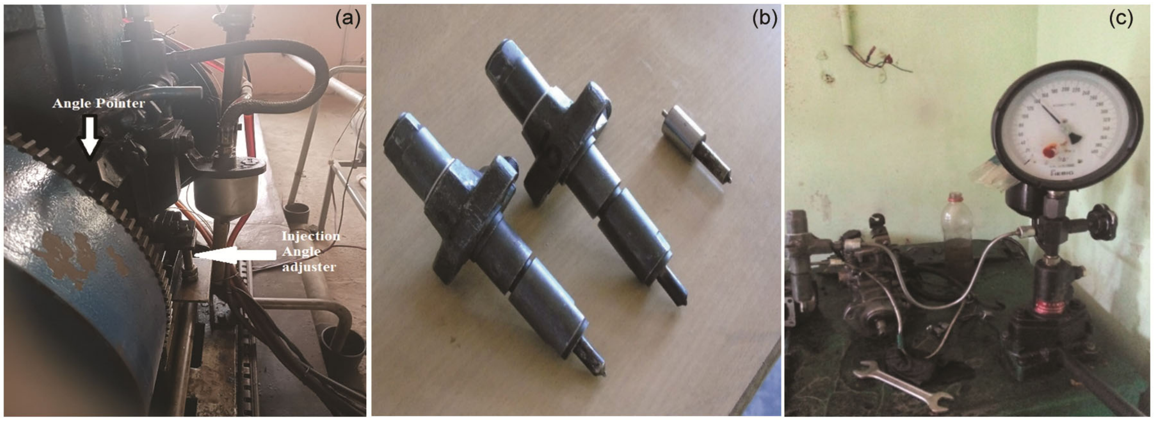


Fig. 2 — Photographs of (a) change in angle, (b) different injectors and (c) injector pressure

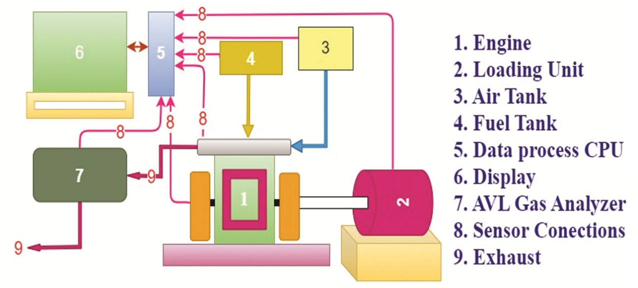


Fig. 3 — Schematic view of experimental setup

Table 3 — Accuracy of measuring instrument

Measured quality	Measuring range	Resolutions	Accuracy
Carbon dioxide (% vol)	0…20	0.1	<10: ±0.5 >10: ±5
Carbon monoxide (% vol)	0…10	0.01	<0.6: ±0.03 >0.6: ±5
Hydrocarbon (ppm vol)	0…2000	≤2000:1, >2000:10	<200: ± 10 >200: ± 5
Opacity (%)	0-100	0.1	± 1
Nitrous oxide (ppm vol)	0…5000	1	<500: ± 50

uncertainty can be computed using Eq. 2. Instruments, settings, tuning, estimating, assessing, and analyzing may introduce errors and indeterminacy. The overall uncertainty of this experiment is 2.11% percent[24, 25].

$$\Delta\varphi_i = \frac{2\sigma_i}{\sqrt{Y_i}} \times 100 \qquad \text{... (1)}$$

$$TUP = \sqrt{\Delta\varphi_1{}^2 + \Delta\varphi_2{}^2 + \cdots\cdots\cdots\cdots\cdots\cdots + \Delta\varphi_n{}^2} \qquad \text{... (2)}$$

Results and Discussions

Injection parameters such as injection timing, pressure, and nozzle hole optimization were discussed deeply for the enriched ethanol fuel blend.

Timing of fuel injection and its effect on performance and pollution

Fig. 4 exhibits the resultant of adjusting fuel opening injection timings at all loads on performance. Fig. 4 (a) reveals that at 23°bTDC (default), the highest BTE is 23.05%. On the other hand, it has the highest brake and indicates a thermal efficiency of 25.53% and 90.4 % at 29°bTDC. When using ethanol blends as a test fuel, higher ignition delay durations are observed, which causes the BTE of CE40 mixes to rise in correlation with increasing fuel injection timing and adds cerium oxide. Fig. 4(b) indicates that a BSFC of less for CE40 blends at a fuel injection timing of 29°bTDC is 0.361 kg/kWh respectively. Cerium oxide enriches the CE40 fuel's lower calorific value into a faster burning one. A portion of its value can be converted into BSFC. Fig. 4(c) illustrates the deviation in indicated thermal efficiency at engine loads between CE40 and fuel injection timings. Compared to alternative fuel injection timings, the highest ITE values were observed for CE40 blends with a fuel injection time of 29° bTDC. When the engine's fuel opening injection timing is increased from 23° to 32°bTDC, the fuel burning achieves its maximum value because it is injected at the best time for burning. Carbon monoxide is a harmful consequence of an incorrectly burned pre-mixed mixture. Fig. 4(d) depicts the influence of fuel opening injection timing on CO pollutants for CE40 blends.

CO was reduced when fuel injection was timed earlier. Due to optimizing fuel injection time results in better fuel combustion, CE40 has the lowest CO emission value (0.09% vol.) of the tested. When using CE40 test fuel with an injection timing advance of 6°CA (23° default to 29°bTDC), CO emissions are reduced by 40%. CE40 emits the least quantity of CO at 29°bTDC. CO emissions were 0.13%vol at the manufacturer-specified 23°bTDC, 30% higher than CE40 emissions at the lower 29°bTDC. Fig. 4(e) depicts the UHC emission outputs for test fuel as a function of fuel injection timings.

Due to the cerium oxide presence, increasing combustion by altering the injection timing from 23 to 32°bTDC is possible. CE40 has the lowest UHC emissions for any given fuel injection timing. Compared to the manufacturer's suggested operating FIOT of 23°bTDC, UHC for E40 is 23.3% lower at 29°bTDC. Ethanol blends have a more extended ignition period because of their low calorific value; adopting advanced fuel injection timing enhances engine performance. If the spray is improved, the physical delay will be reduced. Fig. 4(f) exhibits NO_x emissions decrease as the FIOT increases. One probable explanation is that the earlier fuel opening injection timing increases in-cylinder pressure. Reduced nitrogen oxide (NO_x) emissions have been achieved by delaying fuel injection. Due to the injection timing being advanced by 6°bTDC, ethanol and cerium oxide blends have an increase in NO_x at maximum load. The NOx emissions from the CE40 at 29°bTDC are 284 ppm vol. EGT slightly increases by increasing the injection timing for CE40 blends. It's due to early injection timings converting more heat into work. The exhaust gas temperature (EGT) grew in tandem with the biodiesel content of the test fuel.

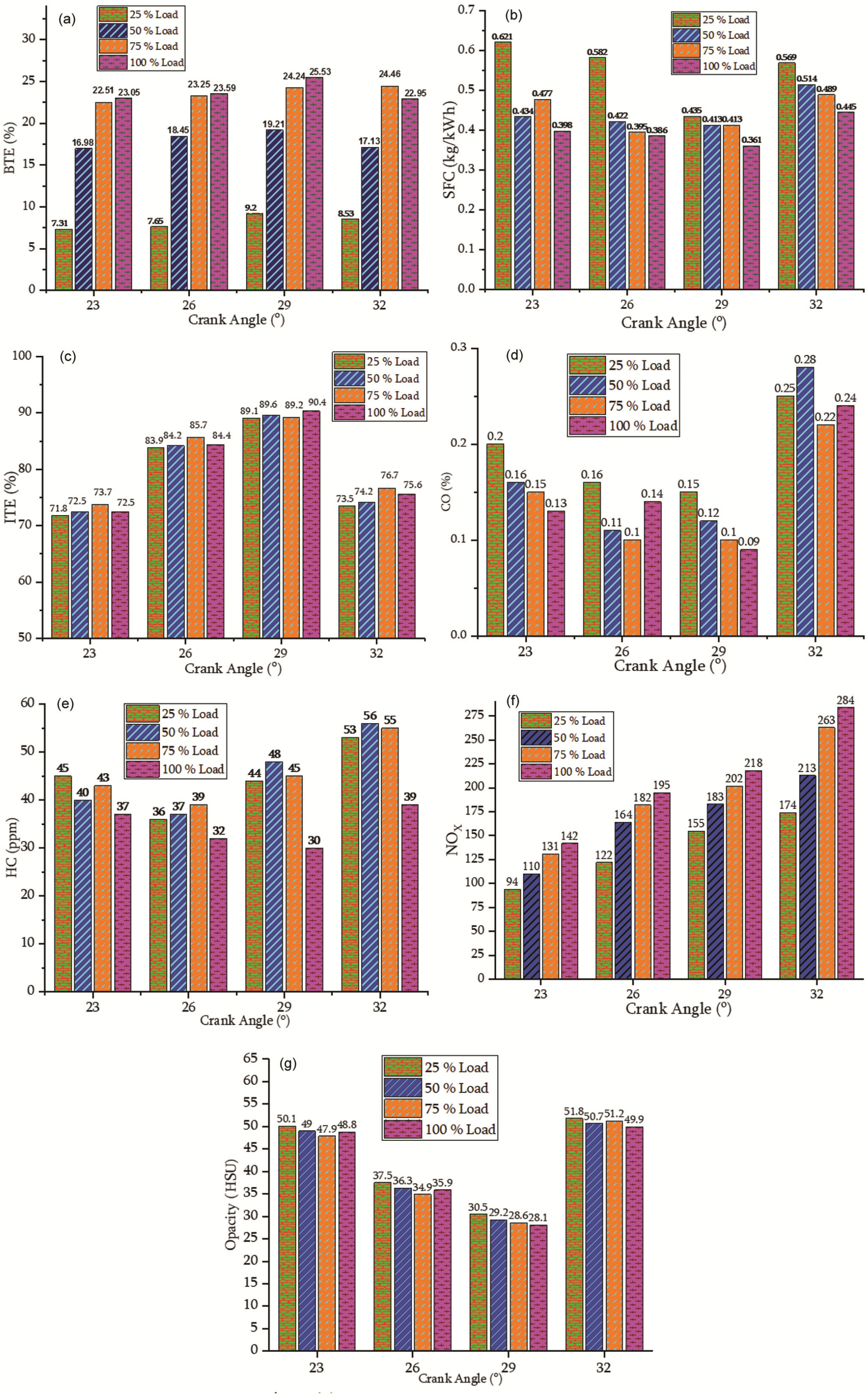


Fig. 4 — Graphs for (a) impact of FIOT on BTE, (b) impact of FIOT on SFC, (c) impact of FIOT on ITE, (d) impact of FIOT on CO, (e) impact of FIOT on HC, (f) Impact of FIOT on NO_x and (g) impact of FIOT on opacity

When oxygen is scarce, smoke may be formed. Fig. 4(g) depicts the difference in smoke opacity as a function of fuel injection timing for CE40. Operating at full loads, smoke emission levels at 29°bTDC are lower than the standard setup. Under full load, smoke opacities of 28.1 HSU were measured at 29°bTDC. Engine smoke opacity is lowered by altering the injection of fuel timing from 23°to 29°bTDC due to enhanced combustion and more efficient use of the oxygen contained in ethanol and cerium oxide mixes. The fuel injection timing from 23° to 29°bTDC reduces smoke opacity, but it rapidly increases from 29° to 32°bTDC at all loads.

Performance and emissions: The role of nozzle hole number

The influence of nozzle holes with a default injection pressure of 18 MN/m^2 and optimized timing 23^o bTDC on engine combustion and emission parameters under varied loads is depicted in Fig. 5.

The maximum BTE with a five-hole nozzle is seen in Fig. 5(a). The spray properties, air mixing, and atomization are all enhanced by the five-hole nozzle. Delayed injection occurs when the nozzle hole is less than five, decreasing the BTE, and according to the results, the BTE is 27.05% at 100% load using a 5-hole nozzle. However, at 18 MN/m^2, BTE was calculated to be 23.57% for 3-hole nozzles and 25.48% for 4-hole nozzles. The engine performed better and BTE can be obtained by increasing the fuel-air mixing rate, which can be accomplished by increasing the number of holes.

The improved ITE of 92.27% at 75% of load, shown in Fig. 5(b), was obtained using a nozzle with five holes, and improved combustion quality results from burned gas being expelled in several ways. The burned gas ratio was increased to achieve more efficiency within the cylinder. More penetration by the fuels with air and better combustion inside the chamber reduced the consumption of more fuels to deliver the same output by the engine.

Hence, the lower SFC values of 0.462, 0.367, 0.331, and 0.304 kg/kWh for 25%, 50%, 75%, and 100% of loads are observed in the five-hole nozzle. Variations of SFC on various nozzle holes are shown in Fig. 5(c).

Fig. 5(d) depicts how nozzle geometry affects HC emission at various loads. Increased combustion quality resulted in a substantial drop in HC at the five-hole nozzle. Ignition Delay (ID) decreases in proportion to the degree of atomization. This is especially useful for CE40, which has a higher ID than regular gasoline because of its higher percentage of ethanol. The HC level minimized from 37 to 22 ppm when increasing the nozzle holes from 3 to 5 at maximum power. The lower count of nozzle holes raising HC levels is likely due to decreased BTE. HC emissions were lowered by increasing the injector nozzle's hole count from three to five. It was revealed that a 5-hole nozzle reduced unburned HC while the engine was running.

Fig. 5(e) depicts the effect of nozzle geometry on CO emission during various load conditions. CO emissions showed similar trends to those reported for HC emissions.

Fig. 5(f) reveals the higher the combustion temperatures and, consequently, the NOx emissions. More energetic combustion within the engine's cylinders results in more NO_x(Ref. No. 26). Greater NO_x levels were seen during premixed combustion at these settings due to improved combustion, and this was observed with a 5-hole with the similar orifice size as when the BTE was greater.

The impact of nozzle geometry is shown in Fig. 5(g) on smoke opacity at various loads. It has been found that enhanced mixture formation due to a well-atomized spray reduces smoke levels. The smoke threshold is at its lowest, with a five-hole nozzle. When the nozzle holes were increased from 3 to 5, the smoke level fell from 48.8 HSU to 29.2 HSU at full load. By raising the fuel-air mixing rate, it seems that increasing the injector hole count from 3 to 5 enhances combustion and minimizes smoke emissions.

Performance and emissions influenced by the opening pressure of the fuel injection

Fig. 6 depicts the combustion characteristics for cerium oxide, and E40 blend fuels for the various loads and fuel injection opening pressures.

Fig. 6(a) illustrates the variation in BTE for various fuel injection opening pressures with default conditions of the engine by cerium oxide and ethanol fuel. For CE40 blends, BTE increases as injection opening pressure and nozzle holes are raised. Due to the magnificent fuel spray by high injection pressure and the presence of nanoparticles (cerium oxide) resulting in improved fuel atomization, the highest value of BTE for the CE40 blend was 28.53 % at 22 MN/m^2 injection pressure. Furthermore, as the injection opening pressure rises, the braking thermal efficiency cuts; this could be because the fuel particles become smaller and a very thin CE40 fuel spray is injected. As a result of the increased diffusion of the CE40, less fuel is sprayed, and the momentum of the blend droplets decreases.

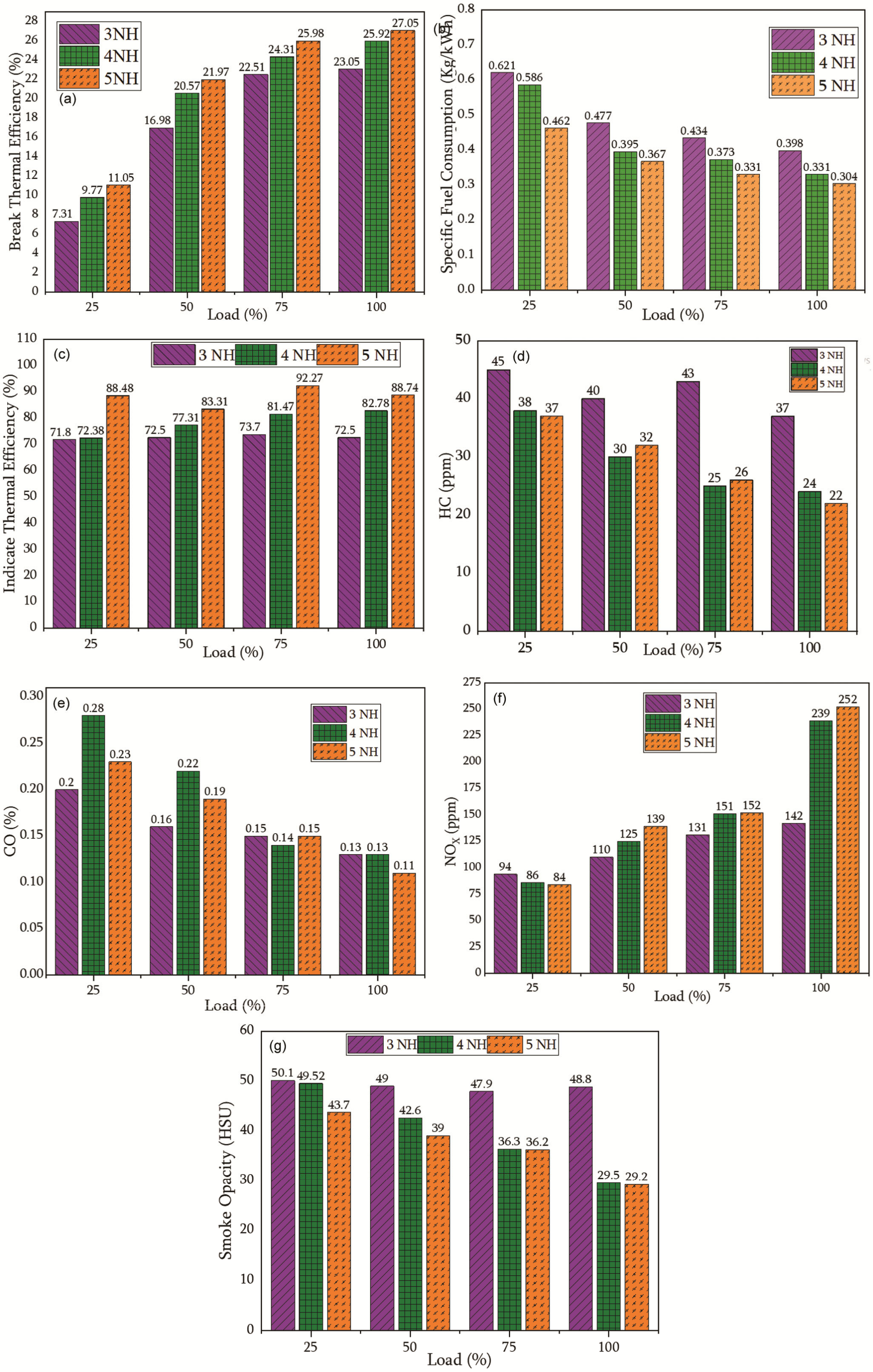


Fig. 5 — Graphs for (a) impact of NH on BTE, (b) impact of NH on SFC, (c) impact of NH on ITE, (d) impact of NH on CO, (e) impact of NH on HC, (f) impact of NH on NO$_x$ and (g) impact of NH on opacity

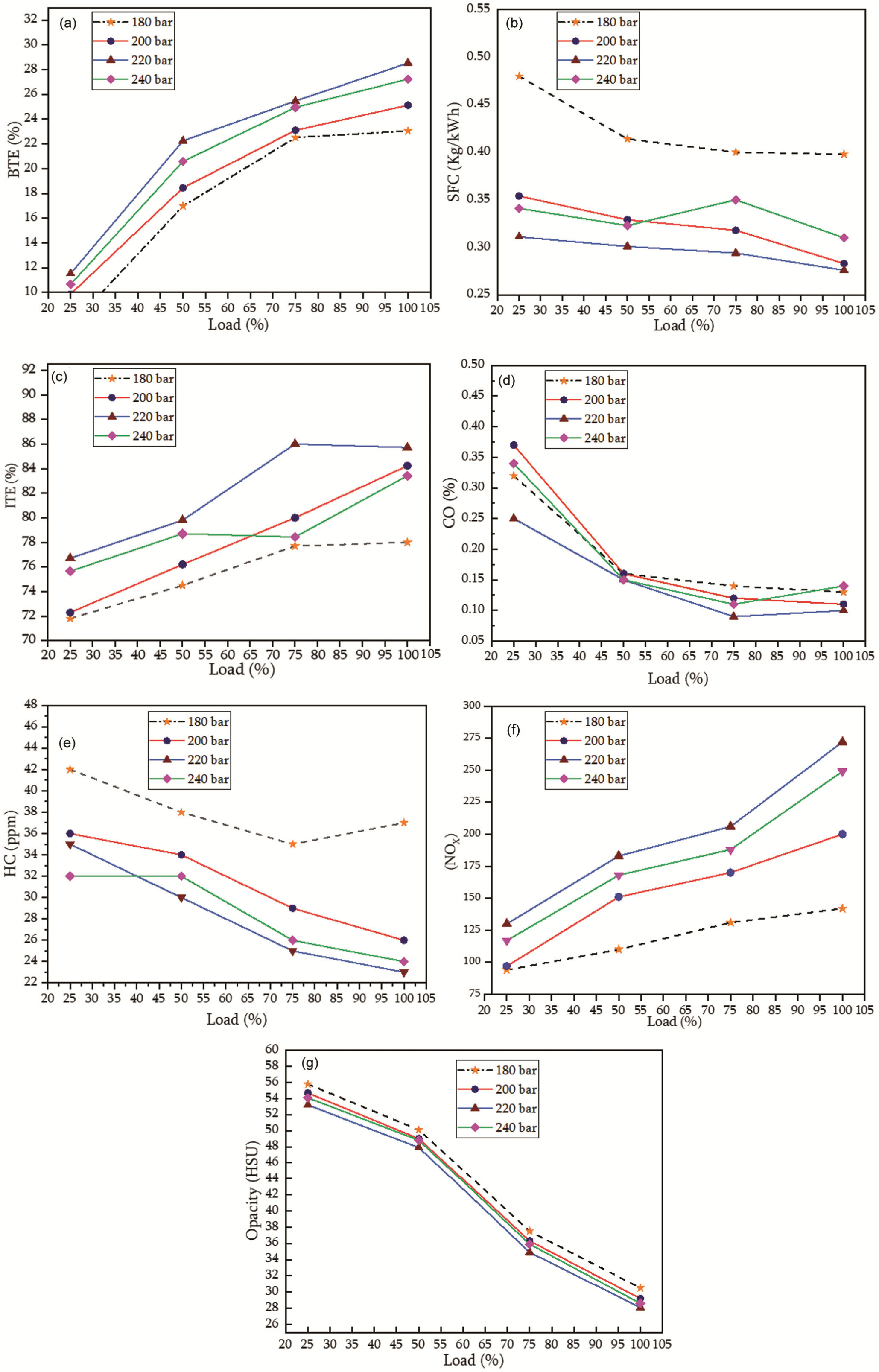


Fig. 6 — Graphs for (a) impact of FIOP on BTE, (b) impact of FIOP on SFC, (c) impact of FIOP on ITE, (d) impact of FIOP on CO, (e) impact of FIOP on HC, (f) impact of FIOP on NO$_x$ and (g) impact of FIOP on opacity

Fig. 6(b) depicts the effect of fuel opening injection pressure on a running engine BSFC. Specific fuel consumption increases when fuel injection opening pressure decreases to less than 20 MN/m^2, particle diameter increases, and ignition delay time increases. In contrast, as fuel injection pressure is raised from the starting point (18 MN/m^2), the BSFC figures decrease. Higher fuel injection pressure of 22 MN/m^2 increased test fuel combustion. As the ignition delay time lowered and injection pressure increased, the BSFC values dropped. The SFC for CE40 is 0.276 kg/kWh at 22 MN/m^2; however, increasing the fuel injection pressure to 24 MN/m^2 maximizes the BSFC by 0.31 kg/kWh due to short ignition latency. As a result, when the fuel injection pressure climbs to 24 MN/m^2, there is less possibility of homogeneous mixing and the combustion efficiency declines.

Fig. 6(c) depicts the ITE variation with fuel injection pressures for CE40 blends at various loads. Increased fuel injection pressure causes the combustion process to take longer, resulting in a higher ITE. The peak combustion pressure in engine cylinders for the CE40 mixture and fuel injection pressures under varying loads. At 22 MN/m^{2}of injection pressure, the highest measured cylinder gas pressure was 68.29 MN/m^2. Fine fuel spray is created when fuel injection pressure is high, which may explain why atomization improves after increasing from 18 to 22 MN/m^2. Furthermore, at 24 MN/m^{2}fuel injection pressure, P_{max} tends to decrease, possibly because the size of test fuel droplets reduces, resulting in decreased spray penetration and droplet momentum.

Fig. 6(d) shows CO emission patterns as a function of fuel injection pressure for a CE40 of test fuels. The enhanced oxygen content of CE40 accounts for the observed decrease in CO emissions when a higher fraction of ethanol is used in the test fuel. The carbon monoxide emissions were drastically cut when the fuel injection pressure was increased. High pressure and nanoparticles permit test fuel to atomize properly. However, increasing the fuel injection pressure from 18 to 22MN/m^2 significantly reduced CO emissions by 0.13 – 0.10% by volume for CE40 test fuel.

Fig. 6(e) shows the results for HC emissions at full load for various CE40 blends and fuel injection pressures. Fuel injection pressure increases improve fuel-air mixing in the combustion chamber, resulting in fewer unburned hydrocarbon (HC) emissions. At 22 MN/m^2, the optimum HC level of CE40 biodiesel was determined to be 23 ppm, 6 ppm lower than at 22 MN/m^2. Cerium oxide nanoparticles decrease HC emissions primarily through improved vaporization and atomization.

Fig. 6 (f) depicts NO_x levels at various loads for numerous mixes of CE40 diesel with varying fuel injection pressures. The graph shows NO_x emissions rise as fuel opening injection pressure increases because the fuel burns more efficiently at higher temperatures and pressures. Test fuel particles with a smaller diameter vaporized faster at increasing fuel injection opening pressure. The fuel particle, on the other hand, cannot penetrate very far into the engine's combustion chamber. As a result, NO_x levels rise when fuel injection opening pressure rises because it first causes faster fuel combustion rates, resulting in a greater EGT, as described in the prior section. The maximum NO_x emissions were observed at 22 MN/m^2 fuel injection pressure and 272 ppm vol., respectively.

Fig. 6(g) shows that the amount of cerium oxide in experimental fuels reduces smoke opacity. Increasing the fuel opening injection pressure exacerbates this impact. One probable explanation is that cerium oxide blends include more oxygen and lower carbon content. Increasing the fuel opening injection pressure is also important since it lowers the size of the fuel particles, resulting in more thorough combustion.

As demonstrated in Fig. 6 (g), increasing the fuel opening injection pressure reduced the opacity of the smoke. The CE40 blend improves smoke transparency regardless of fuel injection opening pressure. The smoke opacity values are 28.6 at 24 MN/m^2, 28.1 at 22 MN/m^2, 29.2 at 20 MN/m^2, and 30.5 at 18 MN/m^2 at 100% load.

Conclusion

The fuel injection parameters were experimentally optimized for a one-cylinder compression ignition engine to be operated by 40% ethanol blends with cerium oxide nanoparticles. Initially, 40% ethanol was blended with diesel fuel, and 0.05 mg cerium oxide (CeO_2) additives were mixed with blended fuels to enhance the combustion quality of the engine. CeO_2 nanoparticles were supported in all aspects and improved the engine performance. The optimization process started with a change in the fuel injection angle for CE40 blends; the FIOT changed from 23 to 32^{o}bTDC. At 29^o bTDC, the BTE increased to 2.48 % and decreased SFC to 15.49%.The percentages of HC and CO, two byproducts of incomplete combustion, dropped dramatically to 23% and 44%, respectively. The second optimization phase increased FINH from 3 to 5. The 5-hole nozzle showed better performance with CE40 fuels. 4 % of BTE gained, and 25.9 % of fuel consumption was reduced. There was a 31% drop in CO levels and a 42% drop in HC levels. Changing the injection pressure was the third step in the optimization process. CE40 fuels improved performance by 22 MN/m^2 in this case. There was a boost of 5.48% in BTE and a decrease of 30.65% in fuel use. The levels of CO and HC were dropped by 30 and 38%, respectively. Overall optimization process, the NO_x was increased with optimum level compared to other modifications and the effect of efficient combustion decreased smoke opacity. It was found that the injection parameters and nanoparticles influenced the combustion process in the engine when it was used with alternative fuels. During this optimization process, a 5-hole nozzle with 29^o bTDC FIOT and 22 MN/m^2 FIOP is recommended to operate 40 % ethanol with 0.05 mg CeO_2.

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