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Effects of Ethanol–Diesel on the Combustion and Emissions from a Diesel Engine at a Low Idle Speed

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Abstract: In this study, detailed experiments were conducted on the combustion and exhaust characteristics of ethanol–diesel blended fuels. The four-stroke four-cylinder common-rail direct injection diesel engine was used. The experiment was carried out at 750 rpm at a low speed idle, and a 40 Nm engine load was applied to simulate the operation of the accessories during the low idle operation of the actual vehicles. The test fuels were four types of ethanol-blended fuel. The ethanol blending ratios were 0% (DE_0) for pure diesel, and 3% (DE_3), 5% (DE_5) and 10% (DE_10) for 3%, 5% and 10% ethanol mixtures (by vol.%). Blending ethanol with diesel fuel increased the maximum combustion pressure by up to 4.1% compared with that of pure diesel fuel, and the maximum heat release rate increased by 13.5%. The brake specific fuel consumption (BSFC) increased, up to 5.9%, as the ethanol blending ratio increased, while the brake thermal efficiency (BTE) for diesel-ethanol blended fuels remained low, and was maintained at 23.8%. The coefficient of variation (COV) of the indicated mean effective pressure (IMEP) was consistently lower than 1% when ethanol was blended. The blending of ethanol increased the ignition delay from a 12.0 degree crank angle (°CA) at DE_0 to 13.7 °CA at DE_10, and the combustion duration was reduced from 21.5 °CA at DE_0 to 20.8 °CA at DE_10. When ethanol blending was applied, nitrogen oxides (NOx) reduced to 93.5% of the level of pure diesel fuel, the soot opacity decreased from 5.3% to 3% at DE_0, and carbon monoxide increased (CO) by 27.4% at DE_10 compared with DE_0. The presence of hydrocarbon (HC) decreased to 50% of the level of pure diesel fuel, but increased with a further increase in the ethanol blending ratio. The mean size of the soot particulates was reduced by 26.7%, from 33.9 nm for pure diesel fuel, DE_0, to 24.8 nm for DE_10.

Keywords: ethanol–diesel; combustion; emission; low idle speed

1. Introduction

The development of next-generation internal combustion engines and new technologies to prevent air pollution and the depletion of resources is underway around the world. Many researchers [1–3] are studying the application of alternative fuels to improve the mechanical functioning and efficiency of internal combustion engines. Bajpai et al. [1] and Mahumdul et al. [2] summed up the features of bio-diesel produced with various kinds of raw materials, and reported that biodiesels were sufficient as a fuel for internal combustion engines and were likely to reduce emissions and prevent the depletion of fossil fuels. Lim et al. [3] studied GTL (Gas-To-Liquid), a synthetic fuel in the form of a clean diesel fuel generated by the Fischer–Tropsch process, and reported its potential as a fuel for the diesel engine. The representative alternative fuel is biofuel, such as biodiesel and bioethanol, which is produced from vegetable or animal raw materials. Biodiesel and bioethanol have the advantage of containing oxygen in the fuel itself. Biodiesel contains oxygen, ranging from 10 to 12 percent [4–7], and ethanol contains about 35 percent [8,9]. Many studies have been conducted on the application of biodiesel to diesel engines that have confirmed biodiesel's positive effect on the reduction of pollutant emissions emitted from diesel engines [2,10–12]. Ethanol is known to be an acceptable fuel for gasoline engines because it has a high octane number [13]. Pure ethanol cannot be used in diesel engines, but it can be used by in blends with diesel fuel. For use as a fuel in an internal combustion or diesel engine, ethanol has many favorable properties [14], such as low viscosity, high oxygen content, high H/C ratio, low sulfur content and high evaporative cooling, which improve its volumetric efficiency. Ethanol has a lower viscosity compared to diesel, which results in the superior atomization of fuel injected into cylinders, and improves the mixing with air when it is blended with diesel. Ethanol also has a high latent heat of evaporation, so using ethanol in a diesel engine by blending it with diesel or biodiesel fuel can increase the volume efficiency by the evaporative cooling of ethanol in the intake and compression strokes. There are three ways to apply ethanol to diesel engines [13,15]. The first method is to supply ethanol fumigation to the intake air using a carburetor or an injector on the manifold. The second method is to build a dual injection system on the cylinder head by modifying the configuration of the system and mechanically changing the engine cylinder head. Lastly, ethanol can effectively be used in diesel engines by blending alcohol and diesel, while preventing phase separation, without modifying the engine system. Many studies have been conducted on the third approach of the above methods. He et al. [16] compared the results of the experiment obtained by applying ethanol 10%-diesel and ethanol 30%-diesel fuels to a four-cylinder direct injection diesel engine. NOx, CO and smoke decreased when using the ethanol blends, but HC increased. Rakopoulos et al. [13] experimented by blending 10% and 15% ethanol (by vol.%) with conventional diesel in a six-cylinder heavy-duty direct injection diesel engine. In that study, CO and NOx tended to decrease slightly, while HC increased. Furthermore, BSFC increased but BTE decreased with increasing alcohol concentration. Sayin [14] applied ethanol–diesel blends, which had 10% ethanol and 15% ethanol by volume, to a single-cylinder four-stroke engine to conduct an experiment with an engine load of 30 Nm at 1000 rpm and 1800 rpm. The results of the experiment showed that CO, HC and smoke opacity decreased, but NOx and BSFC increased. Alptekin [15] tested the effect of a 15% ethanol–diesel blend on three engine speeds (1500, 2000, 2500 rpm) and several engine loads (BMEP 3.3, 5.0, 6.6, 8.3 bar), and reported an increase of pollutant emissions and BSFC. Li et al. [17] applied diesel blended with 5%, 10%, 15% and 20% ethanol by volume to a single-cylinder direct injection diesel engine, and conducted experiments on five engine load conditions (25%, 50%, 75%, 90% and 100%) and two engine speeds (the rated and maximum speed). The addition of ethanol reduced CO and NOx, and increased HC and BSFC. In the study of Author links open overlay panel Lü et al. [18], 15% ethanol–diesel blended fuel was applied to a four-cylinder 3.2-L diesel engine, which demonstrated that CO and HC increased and NOx decreased. Di et al. [19] applied ethanol to diesel and biodiesel to confirm its impact. The concentrations of ethanol in the test fuels were 2, 4, 6 and 8% by volume. It was reported that the BTE improved as ethanol increased, while HC and CO decreased, but NOx increased. Huang et al. [20] conducted a study using five types of ethanol–diesel blended fuels (0, 10, 20, 25 and 30% ethanol by volume). The BTE was reduced because the lower heating value (LHV) of ethanol is low, and the smoke was reduced. The CO decreased above the central load, but increased at low loads and low speeds. Furthermore, the HC increased. However, NOx had different emission trends depending on each fuel condition, engine load and engine speed condition. In addition to the studies of the application of ethanol with diesel, many studies are underway on the application of methanol, an alcohol-based fuel that is less-frequently mixed with diesel than ethanol, but is fully applicable. Jamrozik [21] and Tutak et al. [22] used various proportions of methanol and ethanol–diesel fuel to compare combustion and emission characteristics.

In this study, bioethanol was blended with diesel fuel and applied to a four-cylinder common-rail direct injection diesel engine applied to a commercial passenger vehicle. Notably, this experiment was conducted in a low-speed idle condition. Low idling operation is one of the worst operating conditions of the engine, with poor combustion and high exhaust pollutants. Rahman et al. [23] explained that, during idling, the mixture of air and fuel is rich, and combustion is unstable due to the low operating temperature of the engine, resulting in the increased generation of exhaust pollutants, and more exhaust pollutants being emitted when devices such as air conditioners are operated. Brodrick et al. [24] explained that the engine has a thermal efficiency of 30% under high-speed driving conditions, but only 3–11% at idle. Many researchers reported increased emissions of exhaust pollutants under idle conditions using heavy-duty engines. Khan et al. [25] reported an increase in NOx under idle conditions, Brodrick et al. [24] and McCormick et al. [26] showed an increase in HC and Storey et al. [27] showed an increase in CO. The previous studies were conducted and researched with engines from heavy-duty trucks at high idle, over 1000 rpm. In the above mentioned high idle, the engine runs at a low rated speed with a low load. The experimental conditions of this study were carried out at low-idle speed, rather than a high-idle speed, with an engine for passenger cars. In addition, the fuel injection pressure was 280 bar, the lowest of the engine operating conditions, resulting in relatively poor fuel atomization.

In these poor combustion conditions, at a low idle speed, the following objectives were intended to be carried out using bioethanol–diesel fuel, which is an oxygen fuel: (1) study of the effects of combustion due to high oxygen content, low density and low LHV, which are characteristics of bioethanol; (2) check the change in emission of bioethanol; (3) check the validity of bioethanol application.

2. Methodology

2.1. Test Fuels

In this study, pure diesel fuel was blended with bioethanol with 99.9% purity. Bioethanol was blended with pure diesel fuel at 3%, 5% and 10% by volume. The test fuels were labeled DE_0 (Diesel 100% + Ethanol 0%), DE_3 (Diesel 97% + Ethanol 3%), DE_5 (Diesel 95% + Ethanol 5%) and DE_10 (Diesel 90% + Ethanol 10%). The viscosity of ethanol is about 40% lower than that of diesel fuel, and the LHV is also about 64% lower than diesel fuel. The low viscosity of ethanol is about 40% that of diesel. The cetane number of ethanol is 14.3% that of diesel. Ethanol is further known to have a higher latent heat of vaporization than diesel, by about 3–5 times [8,21,28]. The properties of the diesel and ethanol used in the tests are as shown in Table 1.

Table 1. Diesel and Bioethanol properties.

Properties	Units	Diesel	Bioethanol
Density at 15 °C	kg/m^3	836.8	799.4
Viscosity at 40 °C	mm^2/s	2.719	1.10
Lower heating value	MJ/kg	43.96	28.18
Cetane number	-	55.8	8.0
Flash point	°C	55	12.0
Oxygen content	wt.%	0	34.3
Hydrogen content	wt.%	13.0	13.1
Carbon content	wt.%	85.7	52.2

When blending ethanol and diesel fuel, preventing phase separation is critically important. As shown in Figure 1, the experimental fuels of ethanol blended in diesel fuel were observed, and no phase separation occurred. A study by Hansen et al. [29] also reported that the phase separation of a diesel-ethanol blend does not occur when the ambient air temperature is above 10 °C. Lapuerta et al. [30] also reported that phase separation began to occur when the temperature drops below 10 °C. In this experiment, the test fuels were blended at the ambient temperature of 20 °C, and were kept at about 25 °C during tests.

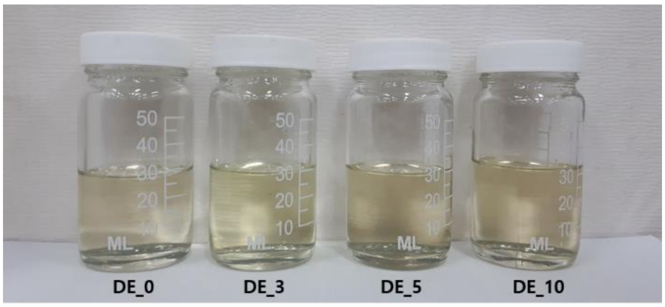


Figure 1. Experimental fuels blended with ethanol in diesel.

2.2. Experimental Setup and Measurements

2.2.1. Engine and Experimental Setups

A four-cylinder 2.0 L common-rail direct injection diesel engine was used for this test. The fuel injection systems (solenoid injectors, fuel pump, common rail) of Bosch and an ECU (engine control unit) were applied. The detailed specifications are shown in Table 2.

Table 2. Engine specifications.

Engine Parameters	Unit	Specification
Engine Type	-	In-line 4 Cylinder, WGT Turbocharged, EGR
Maximum Power/Torque	kW/Nm	84.6(@4000 rpm)/260(@2000 rpm)
Bore × Stroke	mm × mm	83 × 92
Displacement	cc	1991
Compression Ratio	-	17.7:1
Number of Injector nozzle holes	-	5
Injector type	-	Solenoid
Injector hole diameter	mm	0.17

The test engine was installed on an eddy current dynamometer (DY-230 kW, Hwanwoong Mechatronics, Gyeongsangnam-do, Korea). A piezoelectric pressure sensor (Kistler, 6056a, Winterthur, Switzerland), located at the position of the glow plug, was used to measure the combustion pressure, and the data was recorded and analyzed by a data acquisition board (PCI 6040e, National Instruments, Austin, TX, USA). The angular position of the crankshaft for the analysis was measured by a rotary encoder (E6B2-CWZ3DE, Omron, Japan). The emission levels of NOx and CO were measured with an MK2 multi-gas analyzer (Euroton, Italy), and HC was measured with an HPC-501 analyzer (Pantong Huapeng Electronics, China). The smoke opacity level was measured using a partial flow collecting soot analyzer OPA-102 (Qurotech, Korea). The fuel flow rate used during the tests was calculated by measuring the fuel weight change over 10 min on a high-precision digital electronic weighing balance (GP-31K, A&D, Japan). The exhaust gas temperature was measured just after the turbocharger. The soot emitted from the engine was collected by a copper grid (FCF400-CU, Electron Microscopy Sciences, PA, USA), and was used to analyze the soot particle sizes by transmission electron microscopy (TEM, H-7650, HITACHI, Fukuoka, Japan). The experimental equipment diagram is shown in Figure 2.

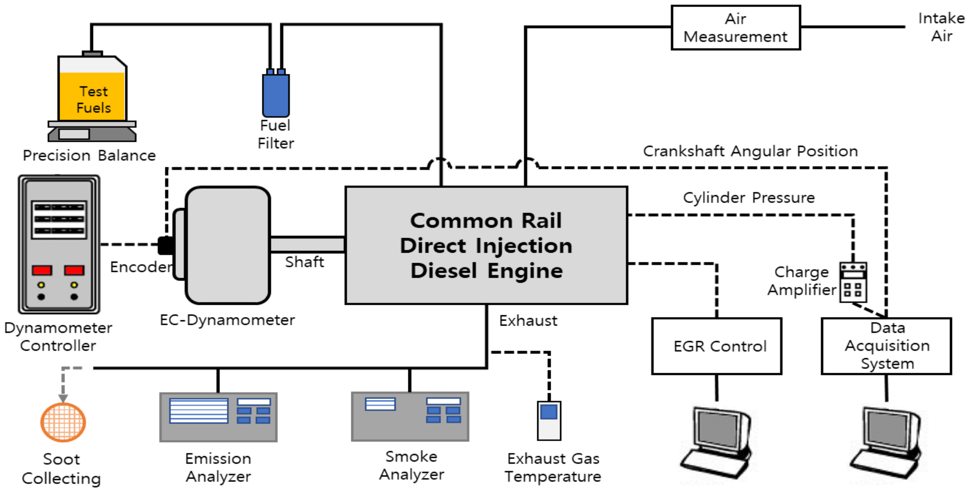


Figure 2. Schematic diagram of the experimental setups.

2.2.2. Test Procedure

In this experiment, the engine speed was set to 750 rpm, which was defined as the low idle speed. The engine load was 40 Nm, in order to produce conditions similar to those of real vehicles with accessory systems such as an alternator and an air conditioner at a low idle speed. The main injection timing was fixed at before top dead center (BTDC) 2 degree crank angle (°CA), the pilot injection timing was fixed at BTDC 20 °CA (the separation is 18 °CA) and the injection pressure was fixed at 280 bar. The main injection duration was 3.4 °CA, and the pilot injection's duration was 1.5 °CA. The combustion pressure and exhaust levels were measured when the engine speed was stabilized within 750 ± 10 rpm for each experimental condition. The coolant temperature was maintained at 85 ± 5 °C. During the experiments, the ambient temperature was kept at about 25 °C. The combustion pressure was calculated as the average of 200 cycles. The soot emitted from the test engine was directly collected from the exhaust pipe onto the copper grid. The grids were evaporated in a 60 °C-vacuum chamber for 12 h. Soot images were filmed 100,000 times magnified using TEM. The combustion pressure and exhaust measurements were initialized when the engine speed was stabilized within 750 ± 10 rpm for each experimental condition. The coolant temperature was maintained at 85 ± 5 °C. The combustion pressure was calculated as the average of 200 cycles. The experimental conditions are summarized in Table 3.

Table 3. Test conditions.

Table	Unit	Condition
Engine Speed	rpm	750 ± 10 (Low idle speed)
Engine Load	Nm	40
Cooling Water Temperature	°C	85 ± 5
Intake Air Temperature	°C	25 ± 5
Fuel Injection Pressure	bar	280
Injection Timing	°CA	Main BTDC 2/Pilot BTDC 20

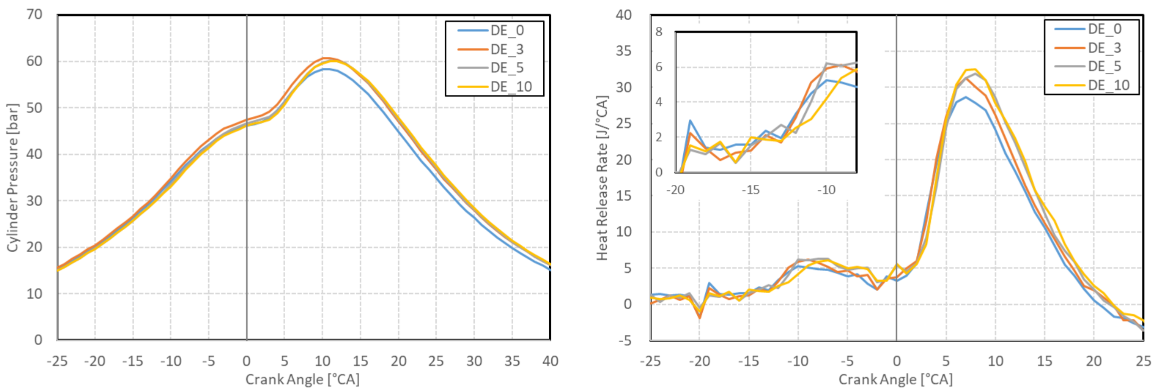
3. Results and Discussion

3.1. Combustion Characteristics

3.1.1. Combustion Pressure and Heat Release Rate

The blending of ethanol has a great effect on combustion in the cylinder. By weight, the ethanol molecule is approximately 33% oxygen, which promotes combustion. On the other hand, the low LHV of ethanol is low, at 64% of that of diesel fuel, but it has a high latent heat of vaporization, which provides an ethanol evaporation cooling effect during combustion. Also, the low density and viscosity of ethanol promote the atomization of the injected fuel.

Figure 3a,b and Table 4 shows the combustion pressure and heat release rates for each fuel condition. The maximum combustion pressure of DE_0 is 5828 kPa, and the ethanol-blended DE_3, DE_5 and DE_10 are 4.1%, 3.1% and 2.9% higher, respectively, at 6070 kPa, 6013 kPa and 5998 kPa, respectively. The position of maximum cylinder pressure was at 11 °CA after top dead center (ATDC) under all fuel conditions. The maximum heat release rate (HRR) is also higher when ethanol is blended. The maximum HRR for DE_3 is increased by 9.3%, from 28.64 J/°CA to 31.30 J/°CA at DE_0, DE_3 is increased by 11.2% to 31.86 J/°CA, and DE_10 is increased by 13.5% to 32.53 J/°CA. The maximum heat release rate occurred at ATDC 7–8 °CA under all fuel conditions. The maximum HRR for DE_3 is increased by 9.3%, from 28.64 J/°CA at DE_0 to 31.30 J/°CA, DE_5 is increased by 11.2% to 31.86 J/°CA, and DE_10 is increased by 13.5% to 32.53 J/°CA. The maximum heat release rate occurred at ATDC 7–8 °CA under all fuel conditions. Jamrozik [21] reported that the mixture of alcohol increased the HRR due to the increase in the ignition delay because of the low cetane number and high latent heat of evaporation, thereby increasing the maximum combustion pressure. The study by Gnanamoriti et al. [31] also reported that, with an increased ethanol blending rate, the heat release rate increased dramatically due to the increase in the adiabatic flame template. On the other hand, combustion after the main injection tends to be different from the combustion of pilot injection fuel. When the main injection is sprayed, combustion is generated partially in the combustion chamber, so the diffuse combustion stage is initiated. The greater the increase in the blend ratio of ethanol, the stronger the improvement of combustion by the increase in oxygen content. Also, the addition of ethanol improves the atomization of the injected fuel. Sayin [14] said that the advantage of using alcohol as fuel is improving mixing with the air by improving the saturation of the injected fuel through its low viscosity. Li et al. [17] reported that the increase in ethanol blending resulted in the lower density and reduced surface tension of the blended fuel, which improved the air mixing due to the evaporation of the ethanol spray, which improved the macroscopic spray behavior and atomization performance. Garai et al. [32] also confirmed the injection characteristics of pure diesel fuel, pure ethanol, and ethanol 10% and 20% blended fuel, and reported that the addition of ethanol facilitates the improvement of the spray and atomization. In addition, with the addition of ethanol, fuel in parts of the pilot injection that have not been burned due to the slow burning of the pilot injection in DE_10 will be burned with the main injection, and the heat release rate will increase rapidly.



(a) (b)

Figure 3. (**a**) Combustion pressure and (**b**) heat release rate of test fuels.

Table 4. Combustion characteristics under test conditions.

Test Fuel	Max Combustion Pressure (P_{max}) (bar)	Location of P_{max} (°CA)	Max Heat Release Rate (HRR_{max}) (J/°CA)	Location of HRR_{max} (°CA)	Exhaust Gas Temperature (K)	BSFC (g/kWh)	BTE (%)
DE_0	58.3	ATDC 11	28.64	ATDC 7	501	336.9	24.3
DE_3	60.7	ATDC 11	31.30	ATDC 7	498	348.3	23.8
DE_5	60.1	ATDC 11	31.86	ATDC 8	495	350.3	23.8
DE_10	60.0	ATDC 11	32.53	ATDC 8	494	356.7	23.8

Figure 4 shows (a) the BSFC and BTE and (b) the exhaust gas temperature for each fuel condition. The BSFC increased from 336.9 g/kWh (DE_0) to 348.3 g/kWh (DE_3) as 3.3%, 350.3 g/kWh (DE_5) as 4.0% and 356.7 g/kWh (DE_10) as 5.9% with an increasing blend ratio of ethanol. This is because more fuel was consumed due to the lower LHV of ethanol. The BTE, on the other hand, is lower than that of diesel fuel when ethanol is blended, and remains at a similar level of 23.8% as the ethanol blend ratio increases. In all studies using ethanol-blended or alcohol-blended fuel, the BSFC was seen to increase. In a study by Lü et al. [18], ethanol had a lower BTE than diesel fuel, and in the study of Huang et al. [20] under low engine load conditions, the BTE tended to be lower than that for pure diesel fuel. Sayin [14] reported that the BTE of alcohol (ethanol and methanol) blended fuel is lower than that of diesel fuel because of the higher BSFC. Gnanamoriti et al. [31] reported that the reason for the reduced BTE of blended ethanol was the low LHV, resulting in more fuel being consumed to produce the same power, an increased ignition delay due to the low cetane number and a decreased combustion temperature due to the quenching effect of ethanol. Rakopoulos [13] and Li et al. [17], on the other hand, reported that the BTE increases with the blending of ethanol. Di et al. [19] explained that there are several factors that affect the BTE. (i) The combustion is improved by the increase of oxygen due to the inclusion of ethanol. (ii) The temperature of the flame decreases due to the high evaporative latent heat, reducing the heat loss of the cylinder. (iii) The increase in the ignition delay due to the low cetane number increases the fuel burned during the premixed period. (iv) In order to produce the same output with a lower LHV, more fuel is injected and burned by the expansion stroke, and the diffusion combustion is also increased. These factors have complex effects depending on the engine load, speed and fuel conditions. Additionally, the exhaust gas temperature decreases with the increasing ethanol blending ratio. A study by Tutak et al. [22] with ethanol and methanol blended fuel, an alcohol fuel, also reported that the exhaust temperature was lowered as the blend ratio of alcohol increased.

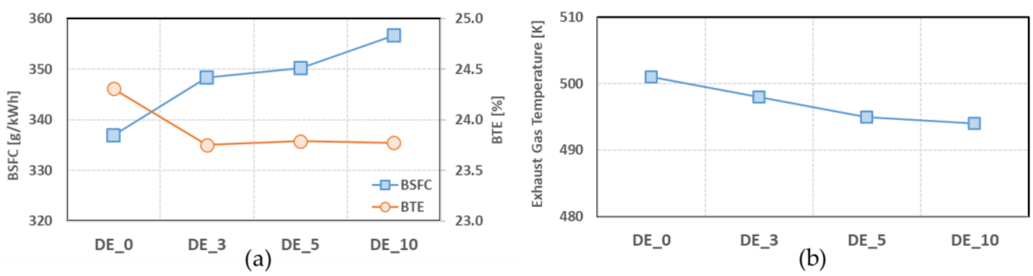


Figure 4. (**a**) The BSFC and BTE, and (**b**) exhaust gas temperature with changing ethanol concentrations.

3.1.2. Combustion Phase

The combustion phase is known to directly affect the thermal efficiency of internal combustion engines. The mass fraction burned (MFB) provides a convenient way to analyze the phase of the combustion in the cylinder by the duration of the crank angle. The MFB is calculated by dividing the amount of accumulated heat during the combustion period by the total heat released [12,33]. From the start of fuel injection, the crank position where the MFB is 10% is noted as CA10, and the point where the MFB is 90% is noted as CA90. The difference between the fuel injection start point and CA10 is called the ignition delay, or the flame-development angle, and the difference between CA10 and CA90 is called the combustion duration, or the rapid-burning angle [33–35]. CA50 denotes where the MFB is 50%, which means that 50% of the injected fuel is converted to energy [36].

Figure 5 shows the MFB for each fuel condition. Figure 5a is the initial area of combustion, which shows the CA10 location of each fuel condition, and (b) is the end of the MFB, which shows CA90. With an increasing ethanol blend ratio, CA10 is retarded from BTDC 8.0 °CA at DE_0 to BTDC 6.3 °CA at DE_10. With increasing ethanol, CA90 was retarded from ATDC 13.5 °CA, at DE_0, to ATDC 14.5 °CA, at DE_10. Figure 6 presents the ignition delay and combustion duration. As the ethanol in the blend is increased, the ignition delay is increased from 12.0 °CA to 13.7 °CA, and the combination duration decreased from 21.5 °CA to 20.8 °CA. This means that the combustion of the pilot injection becomes slower with an increased ethanol blend ratio, but the diffusion combustion after the main injection becomes faster. When ethanol is blended, the ignition delay increases due to the high evaporation latent heat and low cetane number of ethanol. As the diffusion combustion of the main injection is initiated, rapid combustion occurs due to the effect of oxygen added from the blended ethanol. Studies by Xingcai et al. [18], Jamrozik [21] and Tutak et al. [22] also reported that blending ethanol or alcohol fuels increases the ignition delay due to the low cetane number and the cooling effect caused by the high latent heat of evaporation, and decreases the combination duration. CA50 is the point where the maximum heat release rate occurs at the center of the MFB profile, and is used to set the maximum brake torque of the engine. CA50 is retarded from ATDC 6.8 °CA to ATDC 7.6 °CA as the ethanol blend ratio increases. It is noted that these points are similar to the locations of HRR_{max}.

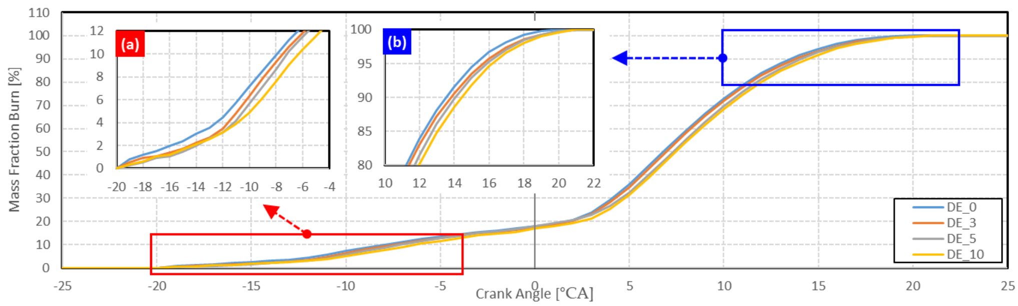


Figure 5. Mass Fraction Burned. (**a**) CA10 and (**b**) CA90.

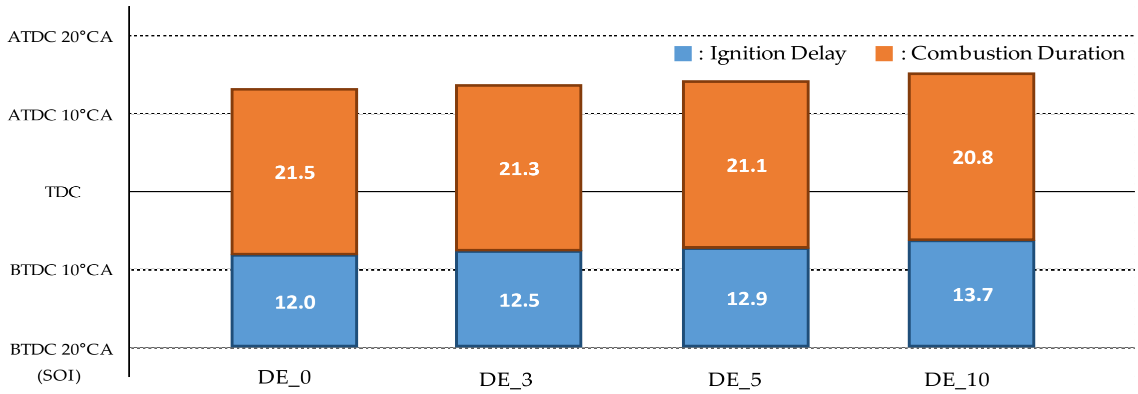


Figure 6. The ignition delay and combustion duration.

3.1.3. Combustion Stability

The coefficient of variation (COV) of the indicated mean effective pressure (IMEP) represents the robustness of the engine to cyclical variability per cycle, and is a way of verifying the stability of combustion. If the COV_{IMEP} exceeds 10% in the internal combustion engines used in automobiles, it is judged that there is a problem with the operation of the engine, and if it is less than 5%, it is judged to have a stable combustion state [21,22]. Figure 7a shows the IMEP for 200 cycles under each combustion condition, and (b) shows the COV_{IMEP}. Under pure diesel fuel conditions, the COV_{IMEP} of DE_0 is less than 1.5%, and those of the ethanol-blended fuels were lower than 0.8%, which demonstrates very stable combustion. For ethanol additions above 3%, further increases in ethanol very slightly increase the COV_{IMEP}. Due to the low LHV of bioethanol and the high latent heat of evaporation, the combustion temperature and pressure in the cylinder were reduced when injected with fuel, and rapid combustion occured subsequently. It was believed that COV_{IMEP} increased slightly as the blend ratio of bioethanol increased. However, it varies less than 0.2%. Similar to the results of this study, Tutak et al. [22] reported that the COV_{IMEP} was 2.5% at low ethanol blending ratios, and did not exceed 10% until the ethanol fraction was 35%. The study of Jamrozik [21] also showed that the COV_{IMEP} increased with an increasing mixture of alcohol fuels, and that the COV_{IMEP} increased by more than 10% when the ethanol blend ratio was more than 30%.

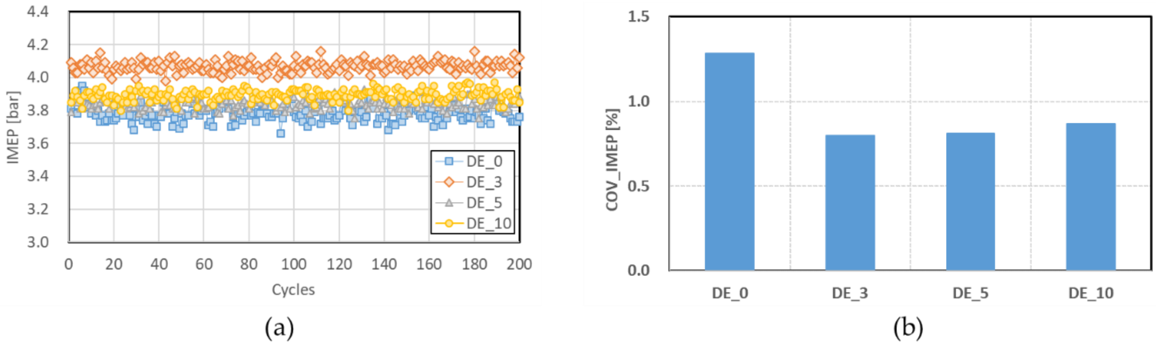


Figure 7. (**a**) Distributions of the IMEP and (**b**) the COV of the IMEP.

3.2. Emissions Characteristics

3.2.1. Gaseous Emissions

The mixture of ethanol has a very substantial effect on combustion. The properties of ethanol, such as its low cetane number, lower LHV, high latent heat of evaporation and low viscosity, affect the generation of exhaust pollutants. Figure 8 shows the emissions characteristics of exhaust pollutants on the gaseous emissions of the test fuel conditions. Figure 8a shows NOx and PM, (b) shows the emission rates of NO_2 and NO_2/NOx, and (c) shows the emission characteristics of HC and CO by test fuel conditions.

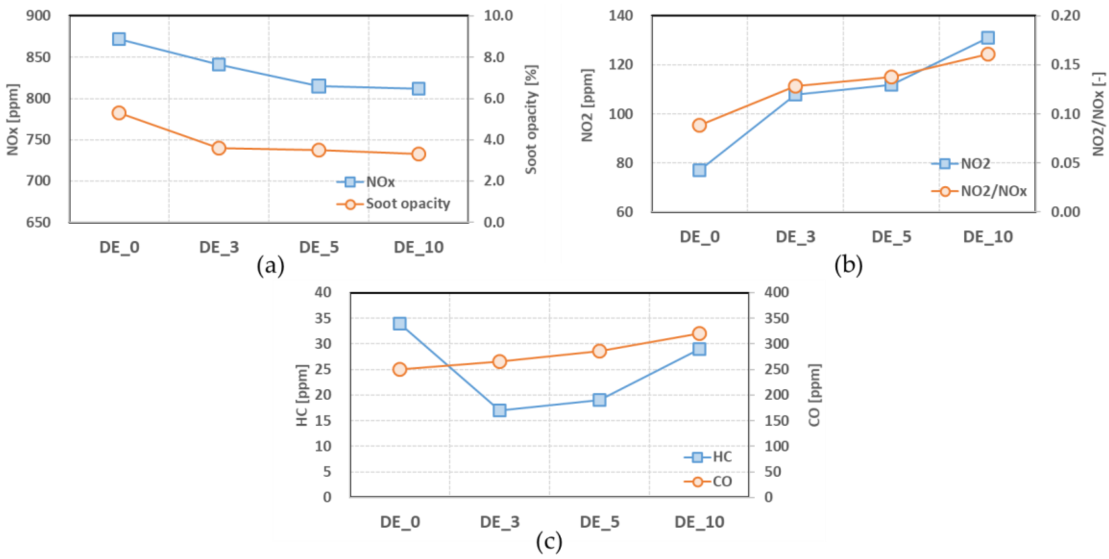


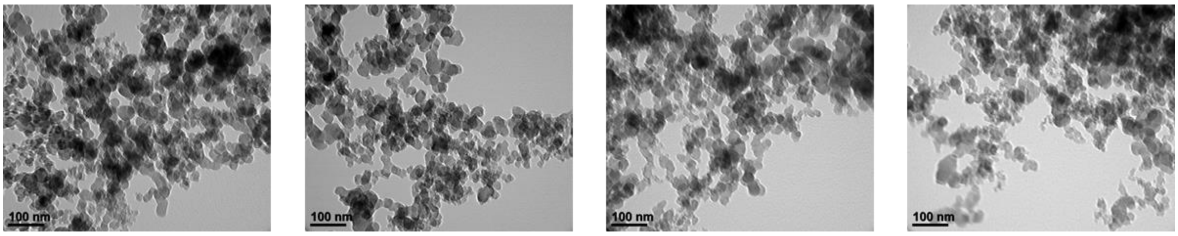
Figure 8. Emissions of (**a**) NOx and soot opacity, (**b**) NO_2 and NO_2/NOx, and (**c**) HC and CO.

NOx was emitted at 872 ppm under the conditions of DE_0. With the ethanol-mixed fuel DE_3, NOx was reduced by 3.5% to 841 ppm; with DE_5, it decreased by 6.5% to 815 ppm; and with DE_10, NOx decreased 6.9% to 812 ppm. As the ethanol blend ratio increases, the emissions of NOx tend to decrease. It was judged that NOx decreases in this manner because the low LHV and high latent heat of evaporation of ethanol reduce the combustion temperature, even though the pre-mixed combustion is increased due to the longer ignition delay caused by the low cetane number of ethanol. Many studies [13,16,19,20] have also reported that the blend of ethanol or alcohol fuels reduces the production of NOx due to their lower LHV and high latent heat of evaporation, which makes the splitting of nitrogen molecules less likely. In the studies of He et al. [16], the generation of NOx was reduced by blending ethanol in fuels, due to the high latent heat of vaporization and low LHV, and the suppression of thermal NO generation by the trace amounts of H_2O in ethanol-blended fuel. Also, Huang et al. [20] reported NOx reduction by ethanol blending for a low engine load condition. Huang explained that the reason for NOx reduction is that the evaporation and lower LHV of ethanol in the blends after injection reduced the gas temperature in the cylinder. Di et al. [19] showed that NOx decreased by 14.1% and 6.1% at low engine loads (0.08 MPa and 0.2 MPa) when using ethanol–diesel blend fuels. In the study of Li et al., using a single-cylinder direct-injection engine, NOx concentrations decreased by 2.2% and 4.2%, respectively, when using 10% and 15% ethanol–diesel fuels. Rakopoulos [13] reported that the emission of NOx was the same or very slightly lower, when an ethanol-blended fuel was used, than that of pure diesel fuel. The reasons were that the oxygen content and the longer ignition delay from the low cetane number of ethanol, and the low combustion temperature due to the lower caloric value of ethanol, counterbalanced each other for NOx generation. On the other hand, there are other studies [14,22] that show that NOx increases with the blend of ethanol. Sayin [14] showed that NOx emissions increased with the use of ethanol and methanol blends, and explained that the NOx increase was due to the fact that the low cetane number and oxygen content were more effective than the latent heat vaporization. Based on the studies above, several factors affect the generation of NOx. First, the high combustion temperature caused by the high oxygen content of the alcohol generates thermal NOx. Second, the reduced combustion temperature caused by the cooling effect of the high evaporation cooling of alcohol reduces NOx. More NOx is produced with the rapid increase of the combustion temperature caused by the increase of premixed combustion, which is related to the low cetane number of ethanol. Tutak et al. [22] reported that the generation of NOx increased with the increase in the ethanol blending ratio, and that it was discharged to a maximum of 5.5 g/kWh from 30% ethanol–diesel fuel. Another reason for the decrease in NOx is the hydrous ethanol that can occur during fuel production or management. Due to its molecular structure, ethanol can be dissolved in water [37]. Kun-Balog et al. [38] reported that aqueous ethanol could reduce NOx. In the study, the combustion of aqueous ethanol and natural gas was compared using a Swirl burner, and the NOx generated by the combustion of aqueous ethanol was found to be reduced by 25%. Tutak explained that this was because the low cetane number of the blended ethanol increased the ignition delay. It is noted that NOx includes both NO and NO_2. NO_2 reacts with volatile organic compounds (VOCs) in the atmosphere to generate optical smog, known as fine dust [39,40]. As shown in Figure 8b, NOx decreases but NO_2 increases as the ethanol mix increases. This is due to the increase in NO_2 generation due to the added oxygen from the ethanol in the mixture. The smoke opacity is reduced by ethanol blending, as shown in Figure 8a. The levels of smoke opacity were reduced from 5.3% for DE_0 to 3% for DE_10 by ethanol blending, and slightly decreased as the ethanol blending ratio increased. This is because of the effect of oxygen from ethanol blending resulting in more complete combustion.

The HC emissions, addressed by Figure 8c, initially decrease as 3% ethanol is added to pure diesel fuel, and then increase as the ethanol blending ratio increases. The HC at DE_0 was 34 ppm, and these levels were reduced by 50% to 17 ppm for DE_3, by 44% to 19 ppm for DE_5 and by 15% (vs. diesel) to 29 ppm for DE_10. By blending ethanol, the HC emissions are lower than the diesel fuel condition. Under idle conditions with a low rotational speed and low fuel injection pressure, slightly rich combustion conditions are created for the stability of combustion. Blending ethanol with diesel reduces HC emissions because of the reduction of partial fuel-rich areas due to the effect of the oxygen and the improved atomization caused by the reduced viscosity of the injection fuel. Also, according to the report of Sayin [14], the addition of alcohol increases the laminar flame speed, which reduces the temperature of the combustion but promotes the oxidation of the HC. However, in this study, the level of HC increases with the further increase in the ethanol blending ratio. This is believed to be caused by the partial over-lean area being increased by the blending of ethanol and the bulk quenching caused by the high latent heat of the evaporation of ethanol. Most studies [13,15–22] also report an increase in HC as the blending ratio of ethanol increases. CO emissions increased as the blending ratio of ethanol increased. The CO concentration at DE_0 was 252 ppm, that of DE_3 increased by 5%to 266 ppm, DE5 increased by 13.5% to 286 ppm, DE10 increased by 27.4% to 321 ppm. Figure 8c shows that CO emissions for the blended fuels are higher than for unmixed diesel fuel. Several studies have reported that CO increases with the blending of ethanol. Lü et al. [18] explained that the increase in CO due to the ethanol mixture is caused by the incomplete combustion caused by the higher latent heat of the evaporation of ethanol. On the other hand, several other studies have reported that CO decreases with ethanol blending. Sayin [14] reported that the addition of alcohol adds oxygen to the combustion gas, resulting in a partially leaner fuel, which converts more CO to CO_2 than under the diesel fuel conditions.

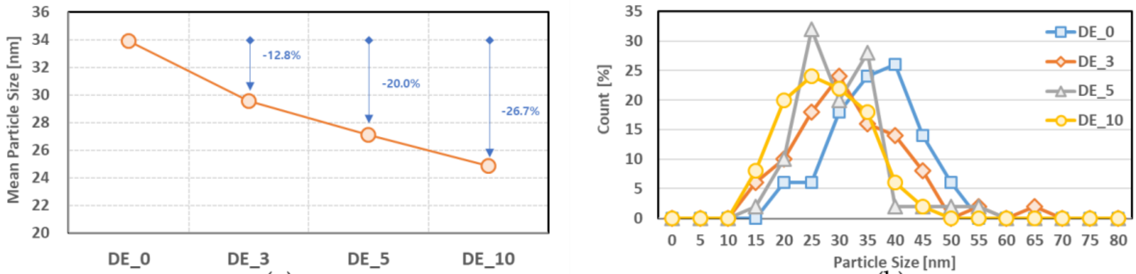
3.2.2. Soot Morphology

The size and morphology of the soot particles are changed by the operating conditions of the engine, for example, the engine load, the speed of rotation, the injection timing and the effect of the fuel used. The sizes of the soot particles emitted under each fuel condition can be identified by TEM images, as shown in Figure 9. As the blend ratio of ethanol increases, the size of the particles decreases. As shown in Figure 10, the mean size of the particles was 33.9 nm for DE_0, down 12.8% to 29.6 nm for DE_3, down 20.0% to 27.1 nm for DE_5, and down 26.7% to 24.9 nm for DE_10. The sizes of particles is between 10 nm and 60 nm, and the distribution of small particles increases with the increase in the ethanol blending ratio. This is believed to be due to the promotion of soot oxidation due to the introduction of more oxygen from the blending of ethanol. Lapuerta et al. [41] applied ethanol in the same way as this test, and reported that the oxidation of the soot nuclei produced during combustion decreases with the increase in the ethanol blending ratio. Likewise, Wei et al. [42] experimented by applying methanol 5%, 10% and 15% (by vol.%) to diesel fuel. The study reported that, due to the increase in the methanol blending ratio, the fractal dimension of soot particles decreased, because the methanol promoted branches of soot aggregates, and the soot activation occurred in the outer area, thus inhibiting soot maturity. Wei et al. [43] applied oxygenated fuel in a methanol blend, a dimethyl carbonate blend and a dimethyl methane blend to confirm the distribution of the primary particle diameter for each fuel condition. The distribution for diesel fuel was 9.5 nm to 50 nm. For the methanol blend, the size range was 7.5 nm to 42.5 nm; for the dimethyl carbonate blend, it was 12.5 nm to 43.5 nm; and for the dimethyl methane blend, the range was 6.5 nm to 41.0 nm. When using an oxygenated fuel, the soot has a smaller particle size distribution. Also, the mean primary particle meter of the methanol blend was the smallest, at 21.66 nm.



(a) (b) (c) (d)

Figure 9. Images of TEM (**a**) DE_0, (**b**) DE_3, (**c**) DE_5, (**d**) DE_10.



(a) (b)

Figure 10. (**a**) Mean soot particle sizes, and (**b**) distributions of particles.

4. Conclusions

Different ethanol–diesel fuel blends were applied to a diesel engine operating at a low idle speed to investigate their combustion characteristics, exhaust pollutants and soot morphology. Based on the experimental results, it is possible to draw the following conclusions:

- The maximum combination pressure and maximum heat release rate of ethanol-blended fuels were higher than those of pure diesel fuel. By increasing the ethanol blend ratio, the maximum combustion pressure decreased and the maximum heat release increased.
- The BSFC increased when ethanol was blended, and increased with the blend ratio; however, the BTEs of ethanol-blended fuels were lower than when pure diesel fuel was used.
- As ethanol was blended and the blending ratio increased, the ignition delay increased, and the combustion duration decreased.
- The COV_{IMEP} values of the ethanol-blended fuels were lower than those of the pure diesel fuel, and tended to increase as the ethanol blending ratio increased above 3%.
- When ethanol was blended and the blending ratio increased, the NOx and soot opacity decreased, but CO emissions increased. The emission ratio of NO_2 in NOx also increased with more ethanol. The levels of HC showed a tendency to increase as the ethanol blending ratio increased, although the HC emissions of ethanol–diesel blended fuels are lower than those of pure diesel fuel.
- As the ethanol blending ratio increased, the mean size of the soot particles decreased, and the distribution of small particles increased.

In low idling conditions with poor combustion conditions, the mixture of bioethanol, a high-oxidant fuel, improves combustion and improves idle stability. Increasing NOx generation due to a longer ignition delay due to the low cetane number of the bioethanol was suppressed by the cooling effect of the high latent heat of evaporation, and as the blend ratio of bioethanol increased, NOx was reduced still further.

This experiment confirmed its effect by blending pure diesel with bioethanol at a low rate. The low viscosity of bioethanol can adversely affect the lubrication of diesel engines. In the future, it is thought that we will need to study not only the limits of the blending ratio of bioethanol, but also the methods for higher blending ratios (e.g., the addition of biodiesel).

Author Contributions: H.Y.K. suggested this research, performed the experiments, analyzed all experimental data, and wrote this paper. J.C.G. performed the experiments and contributed to combustion analysis. N.J.C. performed the data analysis and contributed to the discussion, and supervised the work and the manuscript. All authors participated in the evaluation of the data. All authors have read and agreed to the published version of the manuscript.

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Conflicts of Interest: The authors declare no conflicts of interest.

Abbreviations

The following abbreviations are used in this manuscript.

°CA	Crank Angle
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Center
BTE	Brake Thermal Efficiency
CA10	The crank angle of 10% Mass Fraction Burned
CA50	The crank angle of 50% Mass Fraction Burned
CA90	The crank angle of 90% Mass Fraction Burned
CO	Carbon Monoxide
COV	Coefficient of Variation
DE_0	100% Diesel + 0% Ethanol, Pure petroleum diesel
DE_3	97% Diesel + 3% Ethanol
DE_5	95% Diesel + 5% Ethanol
DE_10	90% Diesel + 10% Ethanol
HC	Hydrocarbon
IMEP	Indicated Mean Effective Pressure
LHV	Lower Heating Value
MFB	Mass Fraction Burned
NOx	Nitric Oxide
TEM	Transmission Electron Microscopy

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