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# Application Characteristics of Bioethanol as an Oxygenated Fuel Additive in Diesel Engines

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**Abstract:** In this study, pure diesel fuel (E0), 5% bioethanol blended with 95% diesel fuel (E5), 10% bioethanol blended with 90% diesel fuel (E10) and 15% bioethanol blended with 85% diesel fuel (E15) were tested on a diesel engine. The 40, 60 and 80 Nm were the main experimental variables, while the engine speed was kept constant at 1500 rpm. The main results show that the addition of ethanol slightly reduced the maximum combustion pressure and delayed the combustion start, but increased the heat release rate (HRR) to varying degrees. Although the addition of ethanol was not very helpful for reducing hydrocarbon (HC), it could reduce carbon monoxide (CO) under appropriate load conditions (60 Nm and 80 Nm). Additionally, nitrogen oxides (NOx) and smoke emissions were reduced with the addition of ethanol under all test conditions.

**Keywords:** bioethanol; diesel engine; fuel additive; engine performance; combustion and emission

## 1. Introduction

As a renewable fuel, biodiesel is widely used in diesel engines. Biodiesel has many advantages, such as a wide source of raw materials, renewable, non-toxic, safe transportation, easy mixing of diesel fuel in any proportion and a high oxygen content can promote combustion [1–3]. However, at present, most commercial biodiesel is mainly produced by transesterification. Although it is more economical than pyrolysis (thermal cracking) and micro-emulsion, it requires a large amount of strong acid or alkali in the process of transesterification, and also needs a series of water washing and purification, which causes water pollution [4,5]. In addition, the price of biodiesel is higher than that of diesel fuel due to the cost of raw materials and complex preparation process [6]. For the above reasons, many researchers began to study the application characteristics of ethanol as an additive in diesel engines.

As a kind of alcohol, ethanol is mainly used as a gasoline fuel additive in spark ignition engines based on its high octane number [7]. The production of raw materials in the form of major free sugars for large-scale ethanol production worldwide is based on low total fossil consumption. The production process of sugarcane ethanol includes five main units: feed treatment, fermentation, separation, dehydration and stillage evaporation [8].

Kurre et al. [9] investigated the influences of four diesel-bioethanol blends (D95E5B0, D90E10B0, D70E15B15 and D60E20B20) on the engine performance and emissions from a diesel engine under various engine loads. They indicated that the BTE slightly decreased and the BSFC increases with increases in ethanol. Moreover, the emissions of nitrogen oxides (NOx), carbon dioxide ($CO_2$), hydrocarbon (HC) and carbon monoxide (CO) decreased with the increase in ethanol mixing ratio. Ge et al. [10] investigated diesel–bioethanol binary blends on combustion and emissions from a common rail direct injection (CRDI) diesel engine according to various injection strategies. The results showed that increasing the ethanol mixing ratio had little effect on the peak value of combustion pressure in the cylinder, but increased the heat release rate (HRR) and reduced the emissions of NOx and smoke. Imdadul et al. [11] studied the engine performance, combustion and emission

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characteristics in a light-duty diesel engine fueled with higher alcohol–biodiesel–diesel blends. Qi et al. [12] investigated the influence of EGR on the engine performance, emission and combustion characteristics of a CRDI diesel engine fueled with diesel–palm-oil–ethanol ternary blends according to different injection strategies. They found that diesel–palm-oil–ethanol-blended fuels presented a little higher temperature than that of neat diesel fuel in the cylinder. BSFC was increased with a higher content of palm oil and ethanol in the ternary blends. The ternary blends are unfavorable to the reduction in NOx. Only when EGR was operated, NOx can be effectively reduced. Prakash et al. [13] studied the effect of bioethanol, diesel and castor oil-blended fuels on the engine performance, emission and combustion of a single-cylinder diesel engine. They reported that the addition of ethanol reduced maximum in-cylinder pressure and HRR and prolonged ignition delay. In addition, the addition of ethanol was only beneficial to the reduction in NO emissions, but did not to the reduction in other exhaust emissions, such as CO, HC and smoke.

In summary, most application studies of bioethanol on a diesel engine are based on ternary fuel, that is, diesel–biodiesel–ethanol blends. The results are different for different engines using different ethanol mixtures and operating under different experimental conditions. Therefore, in order to study the effect of directly adding ethanol to a diesel engine on engine performance, combustion and emission characteristics, ethanol was directly blended with diesel fuel according to the volume ratios of 0:100, 5:85, 10:90 and 15:85. The four test fuels were mixed without any additives.

## 2. Experimental Details

### 2.1. Fuels

In the current work, ethanol was directly blended with diesel fuel according to the volume ratios of 0:100, 5:95, 10:90 and 15:85, which were denoted as E0, E5, E10 and E15, respectively. After 24 h, no phase separation was found when ethanol and diesel were mixed at room temperature. This is consistent with what was reported by other researchers [14]. No phase separation occurs when an appropriate amount of ethanol is blended with diesel oil above 10 degrees [10,14]. The main fuel properties are shown in Table 1.

**Table 1.** Main fuel properties.

| Properties (Units) | Diesel | Ethanol | Test Standard |
|---|---|---|---|
| Density ($kg/m^3$ at 15 °C) | 836.8 | 800 | ASTM D941 |
| Kinematic viscosity ($mm^2/s$ at 40 °C) | 2.719 | 1.1 | ASTM D445 |
| Low calorific value (MJ/kg) | 43.96 | 28.18 | ASTM D4809 |
| Cetane index | 55.8 | 8 | ASTM D4737 |
| Flash point (°C) | 55 | 13 | ASTM D93 |
| Oxygen content (%) | 0 | 35 | - |

### 2.2. Experimental Apparatus

The schematic diagram of the experimental devices is shown in Figure 1. A turbocharged, water-cooled, 4-cylinder, 4-stroke, CRDI diesel engine was selected as the experimental test engine. The detailed information of the diesel engine is listed in Table 2. A water-cooled eddy current dynamometer with a control system was connected in series with the engine for controlling the engine speed and load. The diesel engine was connected. A kistler piezoelectric sensor was used to test the in-cylinder pressure. The diesel emissions, including CO, HC, NOx and smoke, were tested by MEXA-554JKNOx, GreenLine MK2 and OP-160 exhaust emission analyzers.

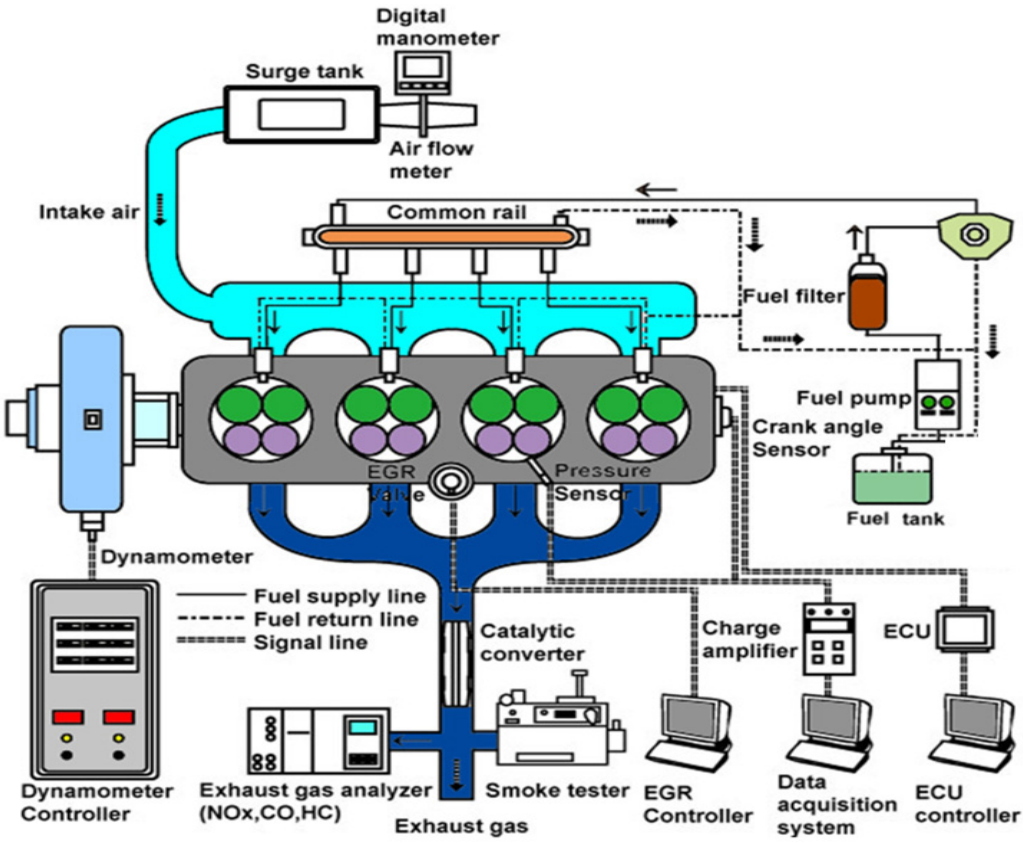



**Figure 1.** Experimental equipment.

**Table 2.** Detailed specifications of the test engine.

| Item | Units | Specifications |
|---|---|---|
| Type | - | In-line 4-cylinder |
| Number of cylinders | - | 4 |
| Bore × stroke | mm | 83 × 92 |
| Injector hole diameter | mm | 0.17 |
| Injector nozzle holes | - | 5 |
| Injection pump type | - | Bosch in-line type |
| Compression ratio | - | 17.7:1 |
| Max. power | kW/rpm | 82/4000 |

### *2.3. Experimental Conditions*

Table 3 shows the specifications of the engine. Table 4 shows the specifications of the exhaust gas analyzer. The experiments were conducted at a constant engine speed of 1600 rpm and three engine loads of 40 Nm, 60 Nm and 80 Nm. To reduce the influence of other variables, the fuel injection pressure, pilot injection and main injection timing were fixed at 35 MPa, 17 and 7 °CA BTDC, respectively. In this study, the diesel engine was not modified during all test periods. In addition, in order to minimize cross contamination between different fuels, the engine was allowed to run with new fuel for 30 min before each test to clean the fuel system. In addition, the heat release rate (HRR) was calculated using the following formula:

$$\frac{dQ}{d\theta} = \frac{k}{k-1}P\frac{dV}{d\theta} + \frac{1}{k-1}V\frac{dP}{d\theta}$$

**Table 3.** Experimental conditions.

| Item | Unit | Conditions |
|---|---|---|
| Fuel | - | E0, E5, E10, E15 |
| Load | Nm | 40, 60, 80 |
| Speed | rpm | 1600 |
| Fuel injection pressure | MPa | 35 |
| Pilot injection timing | °CA BTDC | 17 |
| Main injection timing | °CA BTDC | 7 |
| Intake air temperature | °C | 25 ± 3 |
| Cooling water temperature | °C | 85 ± 2 |

**Table 4.** Specifications of the exhaust emission analyzers.

| Item | Range | Resolution | Accuracy |
|---|---|---|---|
| Electrochemical $O_2$ (%) | 0–30 | 0.1% | ±0.57% |
| Electrochemical CO (ppm) | 0–4000 | 1 ppm | ±0.62% |
| Pellistor HC (ppm) | 0–10,000 | 1 ppm | ±5% |
| Electrochemical NOx (ppm) | 0–6000 | 1 ppm | ±0.25% |
| Smoke opacity (%) | 0–100 | 0.1% | ±1% |

## 3. Results and Discussion

### *3.1. Engine Performance*

Figure 2 shows the brake-specific fuel consumption (BSFC). As shown in Figure 2, all test fuels have the highest BSFC at a low engine load of 40 Nm. Additionally, the BSFC shows a decreasing trend with the increase in engine load. The average BSFC of all tested fuels is, respectively, reduced by 15.74% and 18.82% at 60 Nm and 80 Nm compared with at 40 Nm. This can contribute to the increase in engine load, which improves the combustion in the cylinder (e.g., air flow intensity, temperature and pressure), thereby increasing the combustion efficiency. This is similar to the result reported by [15]. Compared with diesel, the BSFC of the blended fuels gradually increases with the increase in the percentage of ethanol at 40 Nm and 60 Nm. However, the BSFCs of the blended fuels are lower than that of diesel under the high load of 80 Nm. More specifically, at 40 Nm, the BSFCs of the diesel engine fueled with E5, E10 and E15 were, respectively, increased by 1.16%, 3.61% and 5.92% compared with that of E0; at 60 Nm, the BSFCs of the diesel engine fueled with E5, E10 and E15 were, respectively, increased by 1.94%, 3.77% and 4.80% compared with that of E0; at 80 Nm, the BSFCs of diesel engine fueled with E5, E10 and E15 were, respectively, reduced by 3.22%, 3.22% and 2.54% compared with that of E0. The reason for the increase in BSFCs caused by the addition of ethanol under low and medium loads may be related to the low heating value and the high latent heat of ethanol. Fuel with a lower calorific value needs to consume more fuel to obtain the same power output [16]. However, the opposite results under a high load can be attributed to the fact that the higher pressure and temperature in the cylinder under a high load counteract the negative effects of a low heating value and high latent heat generation of ethanol, which together with the high oxygen content promotes full combustion.

### *3.2. Combustion*

#### 3.2.1. In-Cylinder Pressure

The in-cylinder pressure in this study was directly measured by a Kistler piezoelectric sensor, and the pressure curve as shown in Figure 3 was obtained according to the change in crank angle. Important information, such as combustion pressure, HRR, peak heat release rate, IMEP, fuel supply effective pressure, combustion duration, ignition delay and start of combustion (SOC), can be obtained from the in-cylinder pressure value. Therefore, the in-cylinder pressure is recognized as a very important information parameter, especially in the engine development and calibration phase [17]. Figure 3 shows the p–θ diagram for the diesel and diesel–ethanol-blended fuels (E5, E10 and E15) at the various engine

loads of 40 Nm, 60 Nm and 100%, respectively. From these figures, it is clear that the pressure curves for all the tested fuels show almost the same change trend under the low load of 40 Nm, the maximum pressure in the cylinder of the diesel-ethanol blended fuel is obviously lower than that of diesel under the medium load of 60 Nm and the high load of 80 Nm, and the SOC of all the diesel–ethanol-blended fuels is later than that of diesel. This may be because, under the low load of 40 Nm, the combustion environment in the cylinder, such as pressure and temperature, are relatively low, and the fuel properties, such as cetane number, oxygen content and the latent heat of evaporation, are not effectively released, resulting in almost the same p–θ diagram for all the tested fuels. Ethanol with a low calorific value causes the maximum combustion pressure of the blended fuels to be lower than that of diesel, and ethanol with a low cetane number and high latent heat of regeneration causes a late SOC.

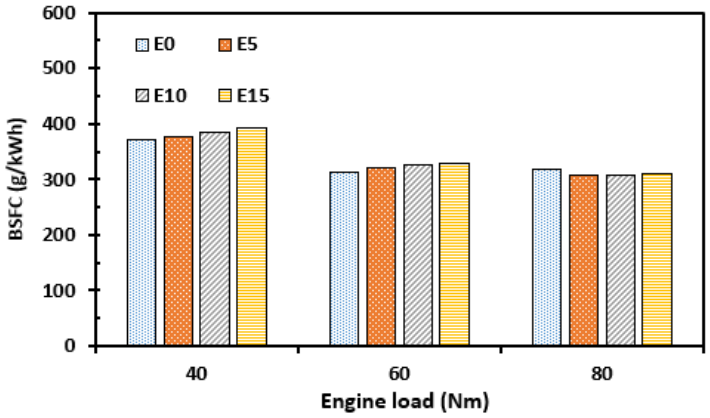



**Figure 2.** Brake-specific fuel consumption.

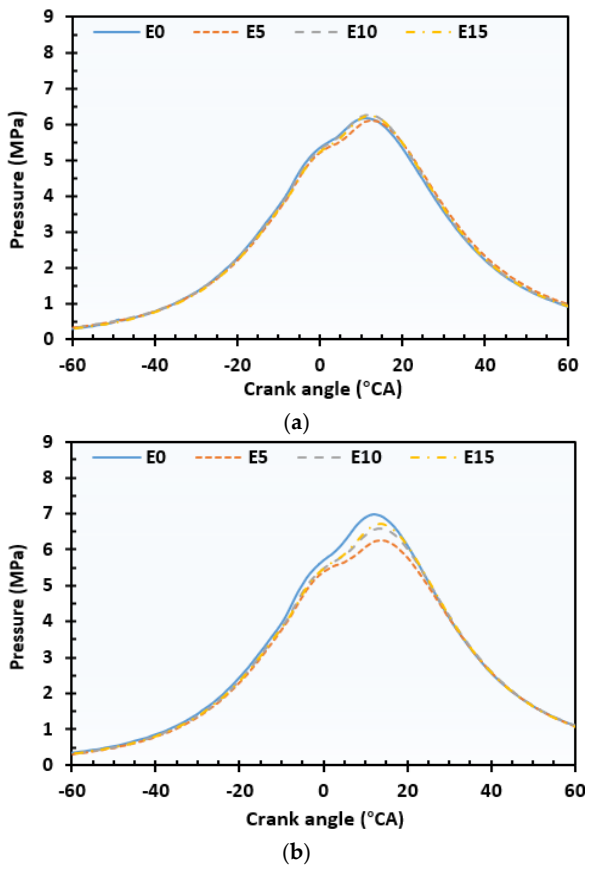



**Figure 3.** *Cont.*

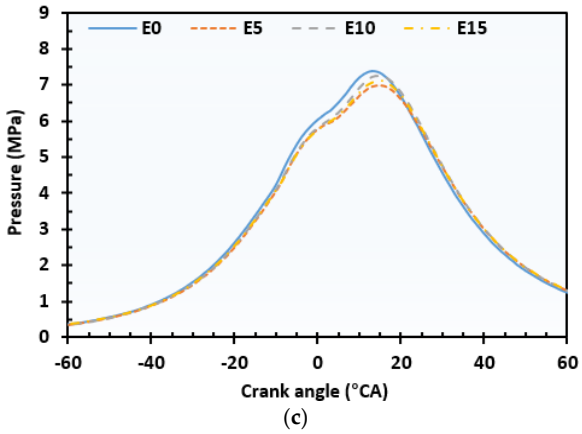



**Figure 3.** In-cylinder pressure: (**a**) 40 Nm; (**b**) 60 Nm; (**c**) 80 Nm.

### 3.2.2. Peak Combustion Pressure

Figure 4 shows that peak combustion pressure for all the tested fuels at different engine loads in the cylinder. The peak combustion pressure of all the tested fuels shows a significant increase trend with the increase in engine load. In addition, the peak combustion pressure of all the tested fuels has no obvious change under the low load of 40 Nm, but the peak combustion pressure of the diesel–ethanol-blended fuels is lower than that of diesel under 60 Nm and 80 Nm. The specific result is that, at 60 Nm, the peak combustion pressures of E5, E10 and E15 were, respectively, decreased by 10.44%, 5.58% and 3.72% compared to that of E0; at 80 Nm, the peak combustion pressures of E5, E10 and E15 were, respectively, decreased by 5.41%, 2.03% and 3.52% compared to that of E0. This may be mainly caused by the increase in ignition delay caused by the low cetane number and high evaporation latent heat of ethanol [18]. In addition, ethanol is a low reactive fuel, so it delays the start of combustion. The mass flow rate of ethanol will delay the combustion stage, and more combustion will be completed in the expansion stroke, so it leads to the decrease in the peak combustion pressure [19].

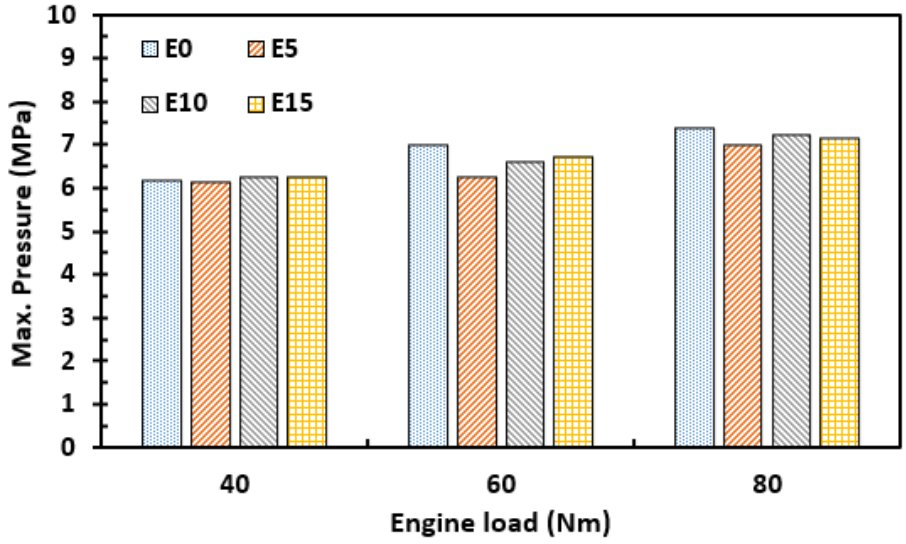
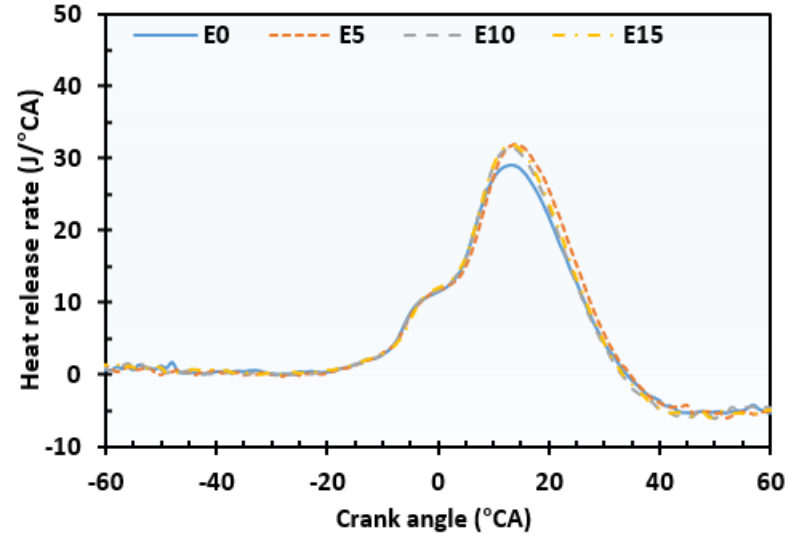



**Figure 4.** Peak combustion pressure in the cylinder.

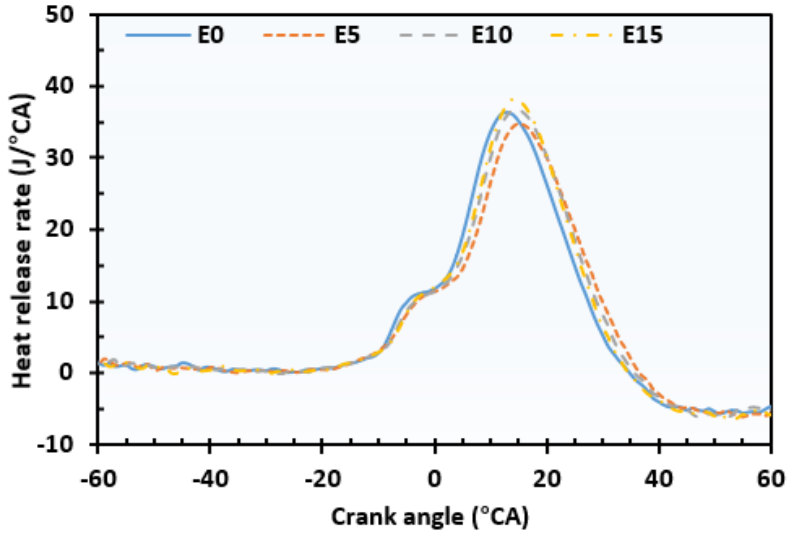
### 3.2.3. HRR

Figure 5 shows the effect of the diesel–ethanol-blended fuels on the heat release rate (HRR) according to different engine loads. On the whole, similar to the above combustion pressure curve in the cylinder, the SOC of all tested fuels is the same under the low load of 40 Nm, while under the load of 60 Nm and 80 Nm, the SOC of the diesel–ethanol-blended

fuel is later than that of diesel. This is because the low cetane number and high latent heat of ethanol lead to a higher temperature of spontaneous combustion, which is not easy to spontaneous combustion. In addition, with the addition of ethanol, the peak value of HRR corresponds to different crank angles, but all of them are distributed at 13 to 16 °CA ATDC. The specific result is that, at 40 Nm, the peak HRR of E0, E5, E10 and E15 appears at 13, 14, 13 and 14 °CA ATDC, respectively; at 60 Nm, the peak HRR of E0, E5, E10 and E15 appears at 13, 15, 15 and 14 °CA ATDC, respectively; at 80 Nm, the peak HRR of E0, E5, E10 and E15 appears at 14, 16, 16 and 16 °CA ATDC, respectively. On the whole, with the increase n engine load and the addition of ethanol, the peak HRR is far away from TDC at the crank angle. This may cause more fuel to be consumed due to the increase in load, thus increasing the combustion duration and delaying the location of the HRR peak. Moreover, the high latent heat of ethanol (cooling effect) further reduces the in-cylinder temperature during vaporization and increases the ignition delay. This is similar to the results reported by other researchers [20,21].

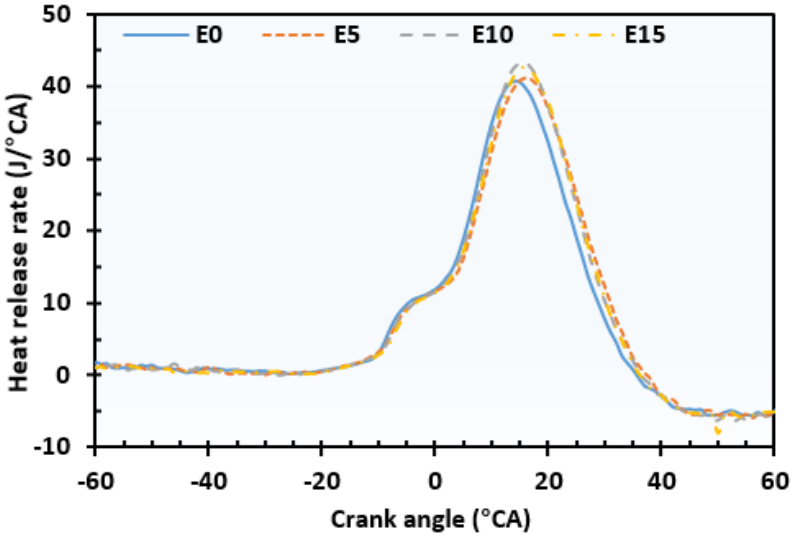



(**a**)




(**b**)

**Figure 5.** *Cont.*




(**c**)

**Figure 5.** Heat release rate: (**a**) 40 Nm; (**b**) 60 Nm; (**c**) 80 Nm.

### 3.2.4. Maximum Heat Release Rate

Figure 6 shows that maximum heat release rate (HRRmax) for all the tested fuels at different engine loads. The HRRmax of all tested fuels shows a significant increase trend with the increase in engine load from 40 Nm to 80 Nm. In addition, except the HRRmax of E15 at 60 Nm, the HRRmax of the diesel–ethanol-blended fuels is slightly higher than that of diesel. The specific result is that, at 40 Nm, the HRRmax of E5, E10 and E15 was, respectively, increased by 9.98%, 9.09% and 9.70% compared to that of E0; at 60 Nm, the HRRmax of E5, E10 and E15 was, respectively, increased by −4.30% (decrease), 0.87% and 5.06% compared to that of E0; at 80 Nm, the HRRmax of E5, E10 and E15 was, respectively, increased by 1.53%, 6.43% and 5.18% compared to that of E0. This may be because the low cetane number of ethanol increases ignition delay, shortens premixed combustion and increases diffusion combustion. Moreover, ethanol is also a fuel with a high oxygen content. The oxygen of ethanol promotes the complete combustion of the fuel and improves the combustion efficiency. Similar studies also show that ethanol has a high oxygen content, which improves the in-cylinder oxygen-deficient combustion atmosphere [21].

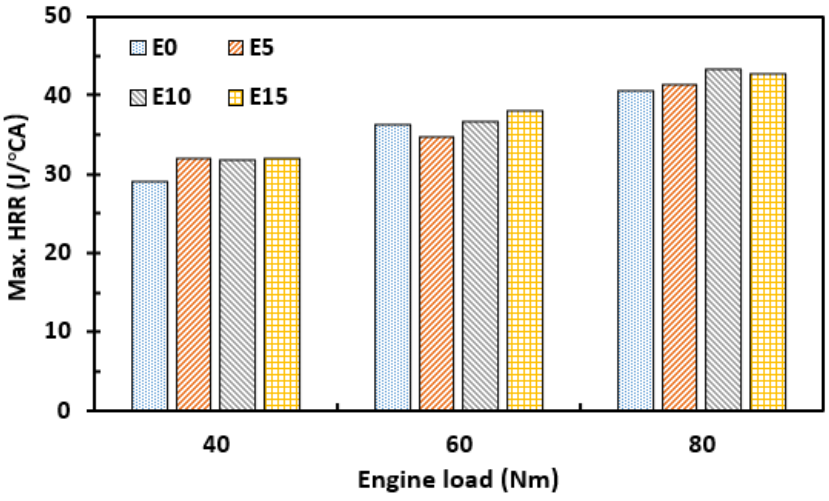



**Figure 6.** Maximum heat release rate.

### *3.3. Emissions*

#### 3.3.1. CO

Figure 7 displays the CO emissions for all the tested fuels under different engine loads. CO emissions from diesel engines are mainly caused by incomplete combustion, which largely depends on the cylinder temperature. The lower the temperature, the more CO is generated [21]. In Figure 7, it can be clearly seen that the CO emissions of all the tested fuels decrease with the increase in the engine load. The specific result is that the CO emissions of all fuels at 60 Nm and 80 Nm were reduced by 35.83% and 39.72% on average compared with that at 40 Nm, respectively. This finding is due to the increase in the combustion temperature in the cylinder with the increase in engine load, which is beneficial to the oxidation reaction of CO. On the other hand, at the low load of 40 Nm, CO emissions for blended fuels show a decreasing trend with the addition of ethanol. This may be related to the high evaporation latent heat of ethanol. However, with the increase in the engine load, the temperature in the cylinder is increased, which counteracts some negative impacts of the high evaporation latent heat of ethanol. Therefore, under the high load of 80 Nm, the CO emissions of blended fuels were lower than that of diesel. Specifically, the CO emissions of E5, E10 and E15 were decreased by 16.13%, 8.07% and 25.81% compared with E0, respectively. Moreover, the reduction in CO emissions caused by the addition of ethanol is also related to the oxygen content of ethanol, which improves the shortage of air in the rich oil area and the post-flame oxidation of CO in the late combustion process. This is similar to the result from [21].

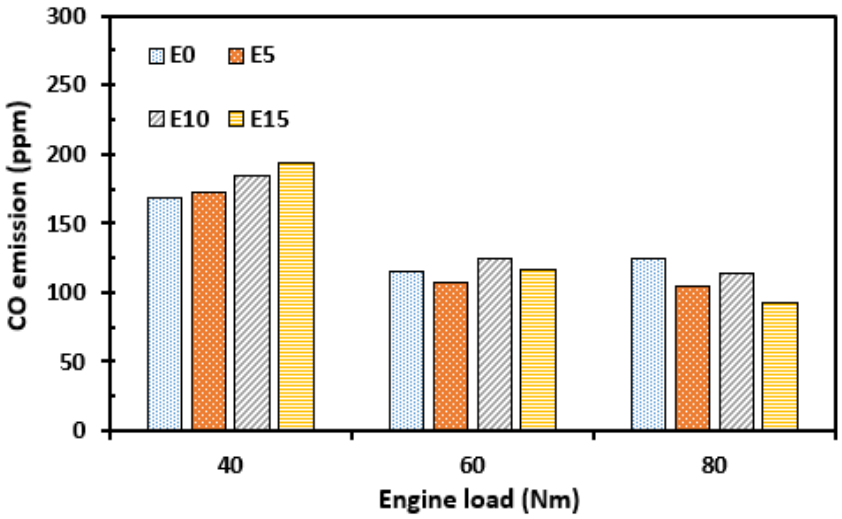



**Figure 7.** CO emissions.

#### 3.3.2. HC

Figure 8 displays the HC emissions for all the tested fuels according to various engine loads. Incomplete combustion and a low oxygen utilization rate are the main causes of HC emissions. For diesel engines with complex combustion modes, different oxygenated fuels and different operating conditions lead to different results on HC emissions [22]. On the whole, the increase in load leads to a slight increase in HC emissions. HC emissions of all fuels at 80 Nm are reduced by 10.00% on average compared with that at 40 Nm. This finding can be interpreted as the increase in load reduces the oxygen utilization rate. Under most conditions, the addition of ethanol leads to higher HC emissions than diesel, especially under the low load of 40 Nm, and the increase in HC emissions is the most significant. The specific result is that, at 40 Nm, the HC emissions of E5, E10 and E15 were, respectively, increased by 14.29%, 57.14% and 100.00% compared to that of E0; at 60 Nm, the HC emissions of E5, E10 and E15 were, respectively, increased by −22.22% (decreased), 22.22% and 44.44% compared to that of E0; at 80 Nm, the HC emissions of E5, E10 and E15 were, respectively, increased by 22.22%, 22.22% and 44.44% compared to that of E0. This

may be related to the high latent heat of evaporation of ethanol. The high evaporation latent heat causes the fuel to absorb the surrounding heat during the atomization process, thereby reducing the temperature in the cylinder. The low cetane number can greatly prolong the ignition delay, resulting in the injected fuel not being burned in time, thus sneaking into the cylinder gap. The high oxygen content further dilutes the mixture, which may cause a partial misfire in severe cases.

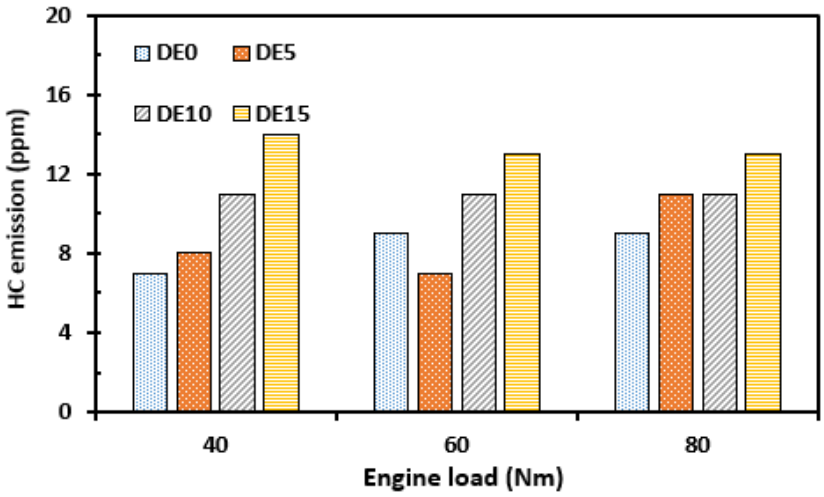



**Figure 8.** HC emissions.

3.3.3. NOx

Figure 9 displays the NOx emissions under different engine load conditions. In diesel exhaust emissions, NOx emissions are predominantly composed of NO, with lesser amounts of $NO_2$. Other NOx emissions, such as $N_2O$, $N_2O_5$ and $NO_3$, are negligible in most conditions [23]. Hoekman and Robbins [23] reviewed the generation mechanism of NOx emissions emitted from diesel engines. They pointed out that NOx emissions from diesel engines are mainly related to Thermal NOx, Prompt NOx and Fuel NOx. In most combustion conditions for diesel engine, thermal NOx is considered to be the main contributor to total NOx emissions. Generally, oxygenated fuel can provide additional oxygen atoms to improve combustion efficiency, which is beneficial to generating more NOx emissions. However, not all oxygenated fuels show the above rules, such as ethanol. In addition to its high oxygen content, ethanol also has a high latent heat of evaporation, playing an interactive role in the formation of NOx emissions. In Figure 9, as many researchers reported, with the increase in engine load, NOx emissions for all the tested fuels show a gradual and obvious increase trend. This is because the increase in load causes more fuel to participate in combustion, and the combustion duration is also increased. However, with the addition of ethanol, NOx emissions of all diesel–ethanol-blended fuels are lower than that of diesel under the same load condition. The specific result is that, at 40 Nm, the NOx emissions of E5, E10 and E15 were, respectively, decreased by 4.16%, 7.43% and 9.36% compared to that of E0; at 60 Nm, the NOx emissions of E5, E10 and E15 were, respectively, decreased by 6.86%, 6.97% and 5.82% compared to that of E0; at 80 Nm, the NOx emissions of E5, E10 and E15 were, respectively, decreased by 4.36%, 5.13% and 3.06% compared to that of E0. This result can be interpreted as the high evaporation latent heat of ethanol causing a low temperature in the combustion chamber during the atomization process, which hinders the generation of a large amount of Thermal NOx emissions [15,24].

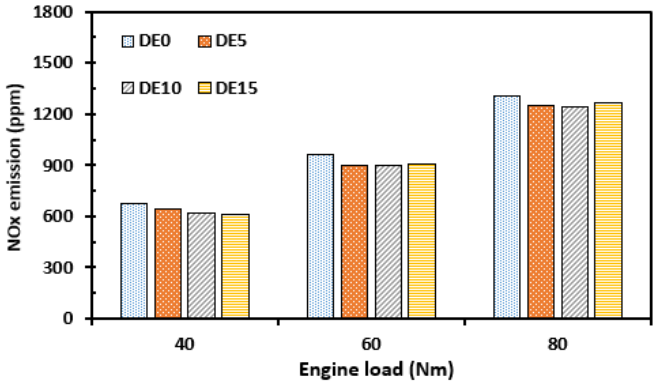



**Figure 9.** NOx emissions.

### 3.3.4. Smoke

Figure 10 displays the smoke emissions under different engine load conditions. In this study, the smoke emissions from the CRDI diesel engine were evaluated by smoke opacity (%), which is the percentage of light absorbed into the smoke [25]. As shown in Figure 10, with the increase in the engine load from 40 Nm to 80 Nm, the smoke opacity was slightly reduced at first and then increased. The smoke opacity of all fuels at 60 Nm and 80 Nm were reduced by 10.00% on average compared with that at 40 Nm. The specific result is that the smoke opacity of all test fuels was reduced by 1.29% and increased by 30.09% at 60 Nm and 80 Nm on average, respectively, compared with that at 40 Nm. This is because properly increasing the engine load can increase the temperature in the cylinder, which is conducive to the oxidation of smoke particles. However, when an excessive load is operated, a large amount of fuel is injected into the combustion chamber, resulting in a reduction in the air–fuel ratio. The formation of smoke opacity occurs at the extreme air deficiency, and carbon smoke (soot) is produced by the thermal cracking of long-chain molecules under anoxic conditions. When there is a local lack of air or oxygen in the diesel engine, smoke increases with the reduction in the air–fuel ratio [26]. On the other hand, with the addition of ethanol, the smoke opacity of all diesel–ethanol-blended fuels is lower than that of diesel under the same load conditions. The specific result is that, at 40 Nm, the smoke opacities of E5, E10 and E15 were, respectively, reduced by 12.37%, 27.84% and 41.24% compared to that of E0; at 60 Nm, the smoke opacities of E5, E10 and E15 were, respectively, reduced by 6.98%, 9.30% and 29.07% compared to that of E0; at 80 Nm, the smoke opacities of E5, E10 and E15 were, respectively, reduced by 80.00%, 24.80% and 45.60% compared to that of E0. It can be explained by ethanol being an oxygenated fuel, and the oxygen atom carried by ethanol itself improving the problem of local hypoxia. The addition of ethanol promotes the complete combustion of the blended fuel, which is conducive to the oxidation of soot in the combustion process.

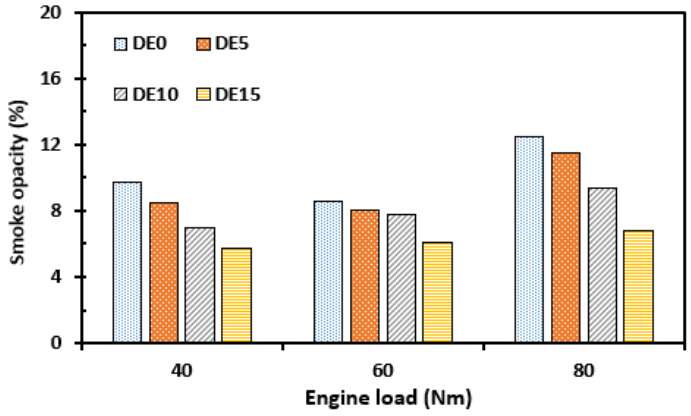



**Figure 10.** Smoke emissions.

## 4. Conclusions

In this study, to explore the application characteristics of ethanol in a CRDI diesel engine, ethanol was directly blended with diesel fuel without any additives according to the volume ratios of 0:100, 5:95, 10:90 and 15:85. These test fuels were recorded as E0, E5, E10 and E15, respectively. The engine performance, combustion and emission characteristics of the CRDI diesel engine were compared and analyzed as the engine load changes. The main experimental results are as follows:

i. For the engine performance: Only under low (40 Nm) and medium (60 Nm) engine load conditions, the brake-specific fuel consumption (BSFC) of the CRDI engine was increased with the addition of ethanol. However, under the high load of 80 Nm, the BSFC was reduced with the addition of ethanol.
ii. For the combustion characteristics: Under the low engine load of 40 Nm, the start of combustion (SOC) and maximum in-cylinder pressure of all test fuels were almost the same, while under the load of 60 Nm and 80 Nm, the maximum in-cylinder pressure gradually decreased with the addition of ethanol. Additionally, the maximum heat release rate (HRR) of most blended fuels was higher than that of diesel.
iii. For the emission characteristics: The addition of ethanol was beneficial to the reduction in carbon monoxide (CO) only under the medium and high loads of 60 Nm and 80 Nm, but under most test conditions, the addition of ethanol led to the increase in hydrocarbons (HCs) in varying degrees compared with diesel only. It is relevant to mention that nitrogen oxides (NOx) and smoke emissions were simultaneously reduced with the addition of ethanol.

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