

## Article

# Research on Combustion, Emissions, and Fault Diagnosis of Ternary Mixed Fuel Marine Diesel Engine

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## Abstract

This study aimed to investigate the effects of diesel/ethanol/n-butanol mixed fuel on the marine diesel engine combustion and emissions at different ethanol blending ratios, different single injection times, and pre-injection times. In addition, this study takes the injector fault phenomenon as an example, simulates the three fault phenomena of the injector, and uses a variety of algorithms to optimize the probabilistic neural network model to achieve the fault state identification and diagnosis of the injector. The results of research showed that, with the increase in the ethanol blending ratio, the peak cylinder pressure shows a decreasing trend. The ignition delay period is extended, and the peak instantaneous heat release rate increases. Compared with D100, the nitrogen oxide (NO<sub>x</sub>) emissions of D50E40B10 mixed fuel are reduced by 12.3%, soot emissions are reduced by 29.18%, and carbon monoxide (CO) emissions are increased by 5.7 times. With the injection time advances, the peak values of cylinder pressure and heat release rate show an increasing trend, soot emissions gradually decrease, and NO<sub>x</sub> and CO emissions gradually increase. The peaks of the cylinder pressure and heat release rate in the pilot injection stage gradually decrease as the pilot injection time advances, while the peak heat release rate in the main injection stage increases. In terms of emissions, NO<sub>x</sub> emissions first decrease and then increase as the pilot injection time advances, while soot emissions gradually increase. The average accuracy of the PSO-PNN neural network model reaches 90%, and the average accuracy of the WOA-PNN neural network model reaches 95%. Therefore, the WOA-PNN neural network model is determined to be the optimal injector fault diagnosis model, which can be applied to the identification and diagnosis of injector fault states of diesel engines.



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**Keywords:** diesel/ethanol/n-butanol; combustion and emissions; fuel injector; fault diagnosis

## 1. Introduction

As the main power source, the diesel engine is widely used in ships, which is crucial to ensure the stability of navigation and operational efficiency. Due to the increasing energy consumption and the depletion of fossil fuels, countries around the world are actively promoting the development and application of renewable and clean energy [1]. Many researchers have paid much attention to the development of cleaner alternative fuels, such as natural gas, bio-diesel, and alcohols, for reducing the PM emission and decreasing the dependence on fossil fuels [2]. As a kind of clean, renewable energy, n-butanol is a high-quality diesel alternative fuel, which can be mixed with diesel in any proportion to avoid liquid phase separation and improve the stability of ethanol and diesel miscibility. It has a wide range of application prospects [3,4]. Ethanol and n-butanol can be made from crops such as corn, cassava, barley, sugar cane, and other agricultural products [5]. When

ethanol and n-butanol are used in marine diesel engines as alternative fuels, the stability of their operation needs to be paid more attention to. Long sailing time, bad sea conditions, and a high-intensity working environment expose ship equipment to various potential failure risks, including various failures of diesel engines [6].

Ethanol has been widely studied in the research of alternative fuels. Chen [7] studied combustion and emission characteristics under different ethanol ratios by injecting ethanol into the inlet and injecting diesel directly into the cylinder. The research results showed that with the increase in the ethanol ratio, the combustion duration of the engine was shortened, the ignition delay was prolonged, and the particle concentration in the engine was reduced. Wei [8] studied the effects of ethanol substitution rate and EGR rate on diesel engine combustion and emission under high load by numerical simulation. The research results show that with the increase in the ethanol ratio, the content of soot and CO generated in the cylinder decreases, but the content of NO<sub>x</sub> and HC generated increases. The maximum content of NO<sub>x</sub> can be reduced by increasing the EGR rate. Pedrozo [9] conducted experiments on a single-cylinder heavy-duty diesel engine with a constant speed of 1200 rpm, and studied the experimental study of dual-fuel combustion from low load to full load. The experimental results showed that when the indicated average effective pressure range was 0.6 MPa to 2.4 MPa, dual-fuel engines have a higher net indicated thermal efficiency than conventional diesel engines, while NO<sub>x</sub> emissions are up to 90% lower than conventional diesel engines.

N-butanol is a four-carbon alcohol. Its carbon content is higher than ethanol, and it has good miscibility with diesel oil, with no need to add a cosolvent. Ning [10] studied the effects of n-butanol addition on combustion and emission characteristics of diesel engines. The results showed that the addition of n-butanol made flame development slow and flame propagation fast, and the braking thermal efficiency of diesel engines decreased with the increase in the n-butanol mixing ratio. Lmani [11] studied the combustion performance of n-butanol and diesel mixture at different injection times by means of experiment and numerical simulation. The research results showed that the addition of n-butanol increased the thermal efficiency and improved the combustion emission characteristics of the diesel engine compared with diesel. Plus, with the increase in the mixing ratio of n-butanol, the emission of NO<sub>x</sub>, CO, and particulate matter decreased. Nadir [12] studied the effect of adding n-butanol to diesel on the emission of polycyclic aromatic hydrocarbons (PAH) from diesel engines. The results show that the addition of n-butanol can significantly reduce the formation of PAHs.

The numerical simulation of a diesel engine is based on heat transfer, fluid dynamics, thermodynamics, and other disciplines to explain the fuel mixing, gas flow, and combustion processes in a diesel engine. Therefore, the numerical simulation can predict the combustion emission of diesel engines more accurately, and can also provide guidance for experimental research [13–16]. The CONVERGE software has obvious advantages in numerical simulation of a diesel engine. The multi-dimensional model of CONVERGE can describe the combustion process in the cylinder more accurately, which provides important support for experimental research [17–20]. In order to ensure the safety of navigation, it is necessary to maintain the diesel engine regularly, so the fault diagnosis technology of marine diesel engines comes into being [21,22].

In order to investigate the combustion and emission performance of ethanol and n-butanol as diesel alternative fuels in marine diesel engines, and to evaluate their potential as diesel alternative fuels, they were mixed with diesel in the way of partial substitution. In addition, the application of new alternative fuels may lead to the abnormal injection state of the injector, which will affect the reliability and performance of the equipment. Therefore, in order to improve the safety of the diesel engine, the abnormal state of the

injector is diagnosed and recognized by the simulation data. By combining the combustion emission analysis of the fuel mixture with the status recognition of the fuel injector, it is expected that the combustion performance of the fuel mixture in the diesel engine can be fully understood, and the status diagnosis of the fuel injector can be realized in time, so as to improve its stability. Finally, it can expand the application of clean alternative fuels in diesel engines.

## 2. Model Calibration and Verification

### 2.1. Model Building

During the operation of marine diesel engines, the combustion process in the cylinder involves complex physical and chemical processes. In order to better understand this process, CONVERGE software was chosen to analyze it.

The computational model was established based on a marine medium-speed diesel engine. Simulation was carried out on a four-cylinder, four-stroke, direct-injection engine, which is commonly used as an auxiliary engine and propulsion unit on ships. The main specifications of the diesel engine are shown in Table 1.

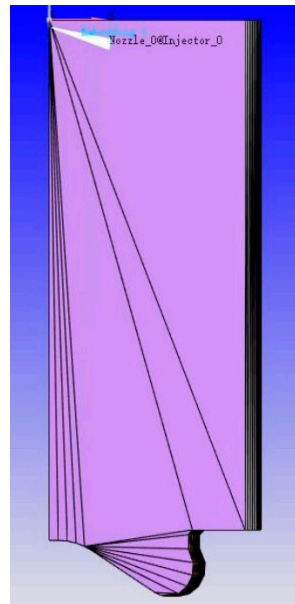
**Table 1.** Main parameters of the marine diesel engine.

| Name                            | Parameter |
|---------------------------------|-----------|
| type                            | In-line   |
| cylinder number                 | 4         |
| cylinder bore/mm                | 190       |
| Stroke/mm                       | 210       |
| Speed/r/min                     | 2000      |
| rated power/kW                  | 220       |
| connecting rod length/mm        | 410       |
| compression ratio               | 14        |
| fuel injection hole diameter/mm | 0.26      |
| Fuel injection hole number      | 8         |

The combustion process of the diesel/ethanol/n-butanol mixture and the emission of pollutants are mainly studied, without considering the air exchange process of the diesel engine. In order to reduce the calculation time, considering the symmetry of the combustion chamber of a diesel engine, the calculation area is set to 1/8 of the whole combustion chamber according to the number of injector holes. Under the make-surface module in CONVERGE software, the geometric model of some combustion chambers of a diesel engine is constructed according to relevant parameters of the diesel engine and combustion chamber profile, etc. The obtained model is shown in Figure 1.

The crankshaft angle range of the model covers the entire process from the time the intake valve is closed ( $-126.5^{\circ}\text{CA}$ ) to the time the exhaust valve is opened ( $121^{\circ}\text{CA}$ ). In the Simulation time parameters module of CONVERGE software, the crankshaft angle range of the model needs to be set to ensure that the simulation covers the required time period. When the SAGE model in CONVERGE software is used to simulate the combustion process of diesel/ethanol/n-butanol mixed fuel, the corresponding chemical reaction mechanism file needs to be imported. The import of the chemical reaction mechanism file needs to be completed in the Gas simulation module of the CONVERGE software. In the CONVERGE software, the initial parameters in the cylinder of diesel engines generally include initial temperature, cylinder pressure, gas composition in the cylinder, and turbulence parameters. The initial temperature is 355 K, and the initial cylinder pressure is 197 KPa. The initial cylinder gas composition is air. The parameters are set in the Regions and initialization module of the CONVERGE software. In the CONVERGE software, injector parameter

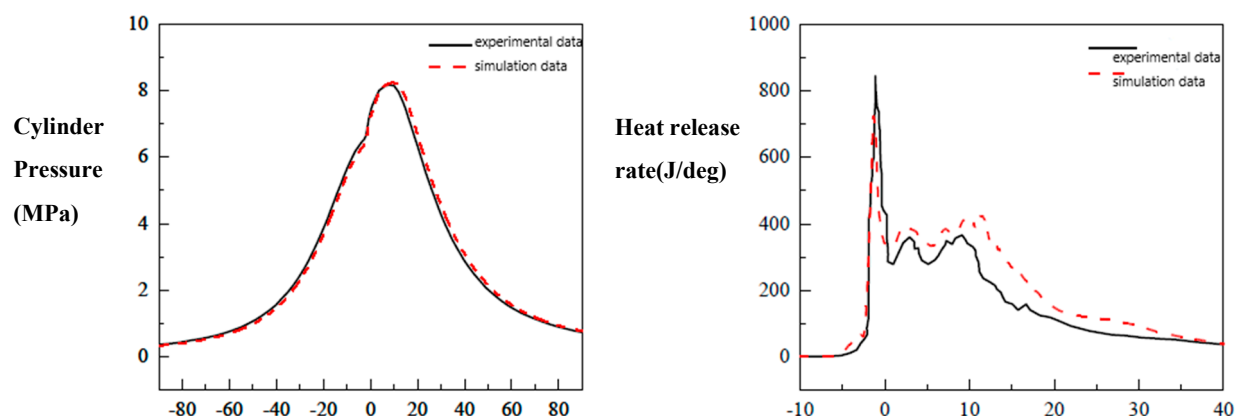
setting is carried out in the Injectors part of the Spray modeling module. These parameters usually include fuel composition, injection time, injection temperature, and the number of injection nozzles. The fuel is divided into diesel, ethanol, and n-butanol. The injection time is  $-9^{\circ}\text{CA}$ , the injection temperature is 341 K, and the number of injection nozzles is 8.



**Figure 1.** Geometric model of a marine diesel engine.

## 2.2. Model Verification

The simulation verification was carried out at a constant engine speed of 2000 r/min and load levels of 75% with the experimental data provided by Zhang [23]. The comparison between the experimental values and simulation values of cylinder pressure and heat release rate in the pure diesel mode is shown in Figure 2. The simulated cylinder pressure curves for the three load levels show an error range of less than 5% compared to the experimental values. In terms of heat release rate, there is a difference between the simulation values and experimental values, but the overall difference remains within an acceptable range. The reasons for the error are as follows: Firstly, the simplified combustion mechanism used in the SAGE model coupled in the simulation cannot fully represent the actual fuel combustion process. Secondly, the selection of calculation models will also bring certain errors, but the errors are within the allowable range. In summary, the model built in this paper is relatively accurate, and the model can accurately predict the engine's performance.



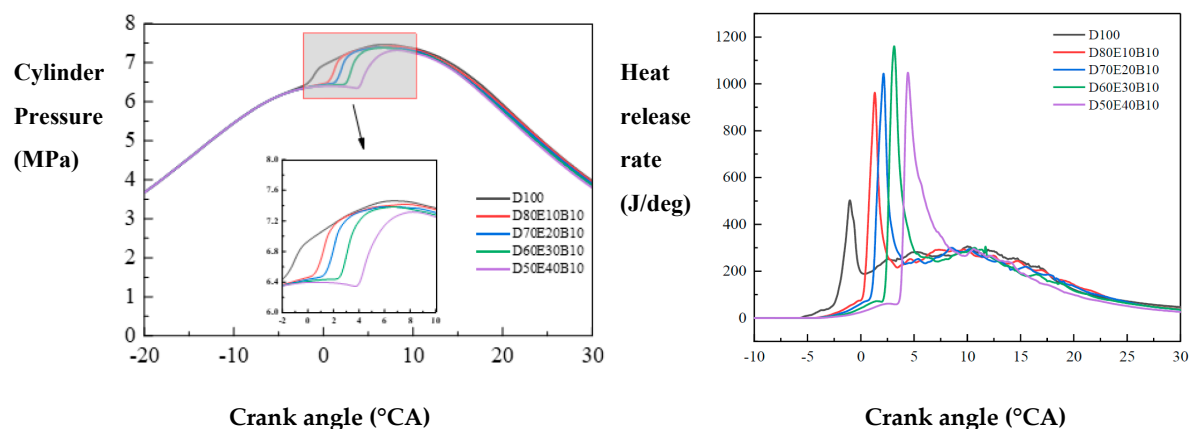
**Figure 2.** Comparison of experimental and simulation data of cylinder pressure and heat release rate.

### 3. Results and Discussions

#### 3.1. Combustion and Emissions with Different Fuel Blending Ratios

##### 3.1.1. Cylinder Pressure and Heat Release Rate

The effects of the ethanol blending ratio on the combustion characteristics of the diesel engine at 100%, 75%, and 50% load are shown in Figure 3. The peak cylinder pressures of D80E10B10, D70E20B10, and D60E30B10 blended fuels were 0.67%, 0.6%, and 0.48% higher than that of D100, respectively, while the peak cylinder pressure of D50E40B10 was 0.46% lower than that of D100. With the increase in the ethanol blending ratio, the peak cylinder pressure of D80E10B10, D70E20B10, D60E30B10, and D50E40B10 gradually decreased. Meanwhile, the ignition timing of the blended fuels gradually shifted back. This phenomenon is due to the obvious improvement in the atomization effect of the blended fuels at low ethanol blending ratios. At the same time, the blended fuels have higher oxygen content, which helps to promote complete combustion of the fuel in the cylinder, thereby increasing the peak combustion pressure in the cylinder, making the cylinder pressure peak of the blended fuels higher than that of diesel. However, as the ethanol blending ratio increased, on the one hand, the heat value of the blended fuels was lower than that of pure diesel, and on the other hand, due to the high evaporation enthalpy of ethanol and n-butanol, the ignition timing of the blended fuels was delayed, the delay in ignition period was increased, and the ignition timing was shifted to after the top dead center (the highest point of the piston stroke, when the crankshaft angle is 0 °CA), at which time the cylinder volume increased and cylinder pressure decreased, ultimately leading to a continuous decrease in the peak cylinder pressure.

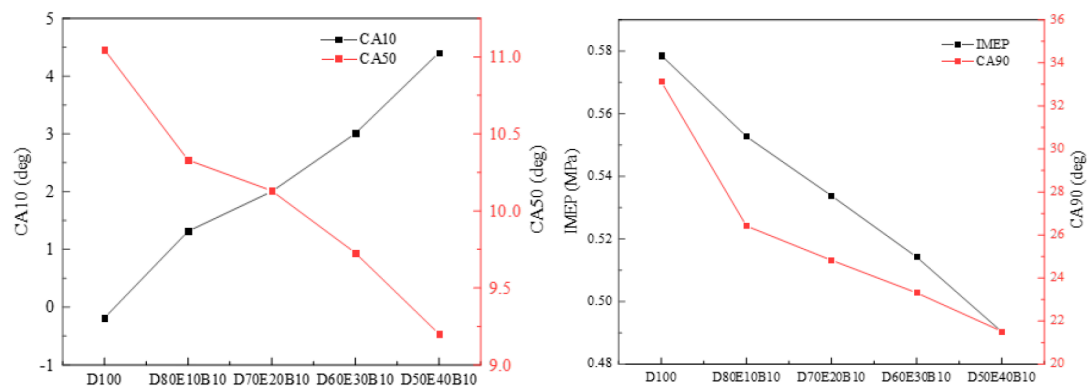


**Figure 3.** Variation curve of cylinder pressure and heat release rate with the increase in the ethanol blending ratio.

Meanwhile, it can be seen from the figure that the peak instant heat release rate is significantly higher than that of pure diesel, and the corresponding crankshaft angle at each peak also gradually shifts later. In addition, the peak instant heat release rate of the same mixture of fuel also gradually decreases. The peak instant heat release rates of D80E10B10, D70E20B10, D60E30B10, and D50E40B10 mixture fuels are 1.35, 1.43, 1.44, and 1.24 times higher than that of D100, respectively. The corresponding crankshaft angle is delayed from 9.61 °CA to 9.12 °CA because of the low cetane number and high latent heat of the vaporization of ethanol, which can delay the ignition of the mixture fuel and prolong the diffusion time of the mixture fuel in the cylinder. Thus, the number of combustible mixtures is increased during the premixing stage, and heat release is more concentrated during combustion, which increases the peak instant heat release rate.

### 3.1.2. CA10, CA50, CA90, and IMEP

Figure 4 shows the effects of different blending ratios of the diesel/ethanol/n-butanol mixture on CA10, CA50, CA90, and IMEP. In this study, CA10, CA50, and CA90 represent the crankshaft angles corresponding to 10%, 50%, and 90% of the cumulative heat release in the cylinder, respectively, with CA10 representing the ignition start point, CA50 representing the combustion center, and CA90 representing the ignition end point. The delay period is defined as the crankshaft angle from the start of fuel injection to CA10, and the combustion duration is defined as the crankshaft angle between CA10 and CA90. It can be seen that, with the increases in the ethanol blending ratio, the CA10 of the diesel engine increases, while CA50, CA90, and IMEP decrease. The ignition start point undergoes a gradual shift from  $-0.6^{\circ}\text{CA}$  to  $4.21^{\circ}\text{CA}$ . This is mainly because when the ethanol content increases, the cetane number of the mixture decreases, causing delayed combustion, prolonged delay period, and increased diffusion time of the mixture in the cylinder. As a result, the combustion duration decreases. In addition, since the ignition start point of the mixture gradually moves away from the top dead center, the cylinder volume increases, and the heat released by the fuel is less effectively converted into work, causing a decrease in IMEP.



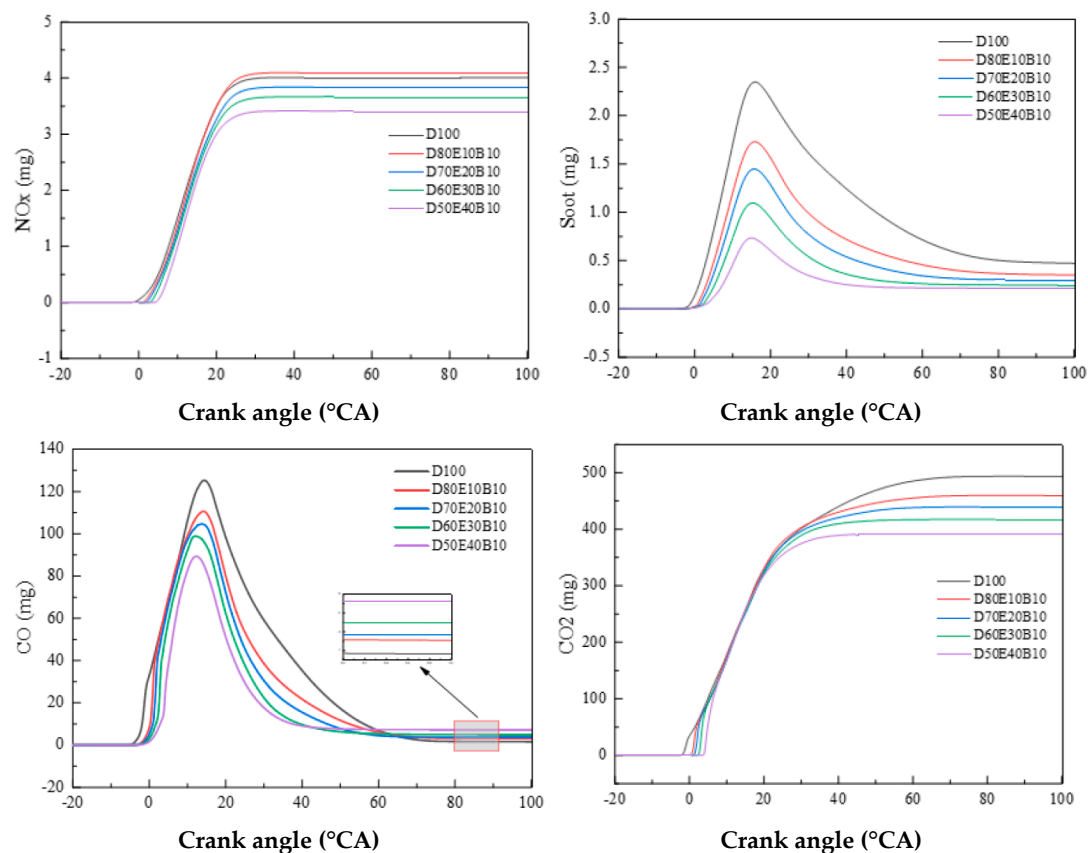
**Figure 4.** Variation curve of CA10, CA50, CA90, and IMEP with the increase in the ethanol blending ratio.

### 3.1.3. NO<sub>x</sub>, Soot, CO, and CO<sub>2</sub> Emissions

The effects of different ethanol blending ratios on emissions of diesel engines are shown in Figure 5. The NO<sub>x</sub> emissions decrease, and the ethanol blending ratio increases. The NO<sub>x</sub> emissions of D80E10B10 fuel are slightly higher than those of pure diesel, and they gradually decrease and become lower than those of D100 because NO<sub>x</sub> formation is affected by high temperature, oxygen enrichment, and longer combustion duration. The reduction in NO<sub>x</sub> emissions is due to the high latent heat value of ethanol, which absorbs heat during combustion and lowers the cylinder temperature. On the other hand, the increase in ethanol reduces the viscosity of the blended fuel, thereby improving the quality of the in-cylinder mixture and increasing the combustion speed, which shortens the combustion duration and reduces NO<sub>x</sub> emissions in a comprehensive manner.

When the ethanol blending ratio increases, the soot emissions gradually decrease. Compared with D100, the soot emissions of D80E10B10, D70E20B10, D60E30B10, and D50E40B10 blended fuels are reduced by 11.93%, 19.64%, 28.2%, and 29.18%, respectively. This is because ethanol has a lower octane number, which delays the ignition of the blended fuel, prolonging the diffusion time of the fuel and improving the premixing effect, thereby promoting combustion. Meanwhile, the addition of ethanol and n-butanol reduces the viscosity of the blended fuel, thereby improving its atomization and evaporation effects,

further promoting the combustion process. In addition, the high oxygen content of ethanol and n-butanol also plays a positive role in oxidizing carbon smoke.



**Figure 5.** Variation curve of different emissions with the increase in the ethanol blending ratio.

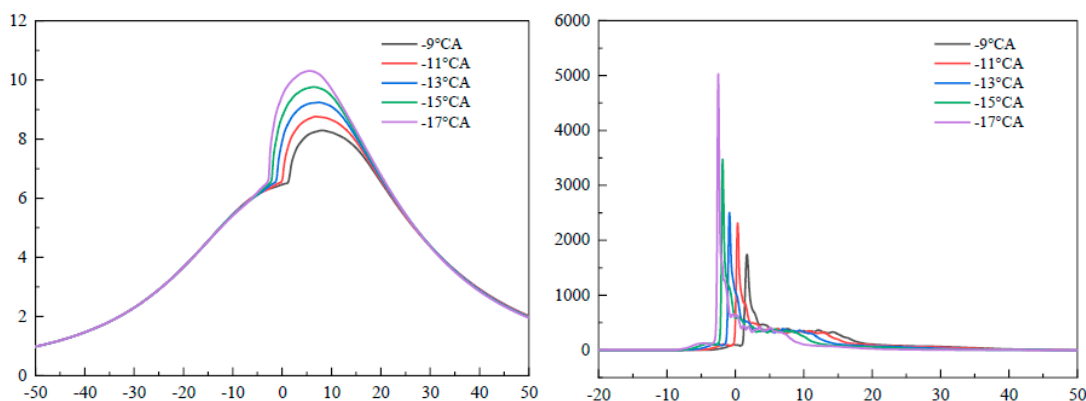
When the ethanol blending ratio increases, the CO<sub>2</sub> emissions decrease, and the maximum in-cylinder CO generation decreases, while the final emissions increase. This is mainly due to the fact that during the peak generation stage of CO, the rapid heat release phase is in the cylinder; at this time, the mixed fuel is fully burned, and the maximum value of CO generation decreases. However, due to the higher latent heat value of ethanol than diesel, the mixed fuel will absorb a large amount of heat and generate a large amount of water vapor during the combustion stage. Subsequently, as the cylinder pressure and temperature decrease, CO cannot be fully oxidized to CO<sub>2</sub>, ultimately causing the final CO emissions to increase while the CO<sub>2</sub> emissions decrease.

### 3.2. Combustion and Emissions with Different Injection Times

#### 3.2.1. Cylinder Pressure and Heat Release Rate

The effect of different injection times on the in-cylinder pressure and heat release rate when a single injection strategy is adopted is illustrated in Figure 6. It can be observed from the figures that as the injection time advances, both the peak values of cylinder pressure and the instantaneous heat release rate increase. Specifically, when the injection time is advanced to −17 °CA, there is a more uniform gas mixture in the cylinder, resulting in a rapid combustion-induced pressure rise with a peak value of 10.31 MPa at 5.51 °CA. Moreover, compared to the original model, an increase of 3729.94 J/deg can be seen in terms of peak instantaneous heat release rate while its corresponding crankshaft angle shifts from 1.7 °CA to −2.47 °CA. Due to extended fuel diffusion time caused by piston movement away from the top dead center, the volume within the cylinder is increased, which leads to

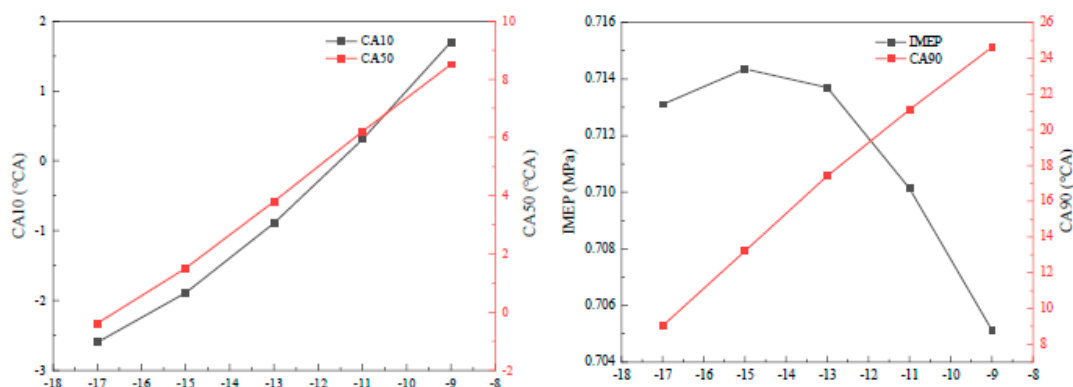
lower pressure and temperature conditions inside it, along with enhanced premix effect facilitated by ethanol and n-butanol volatilization during the compression phase.



**Figure 6.** Variation curve of cylinder pressure and heat release rate under single injection.

### 3.2.2. CA10, CA50, CA90 and IMEP

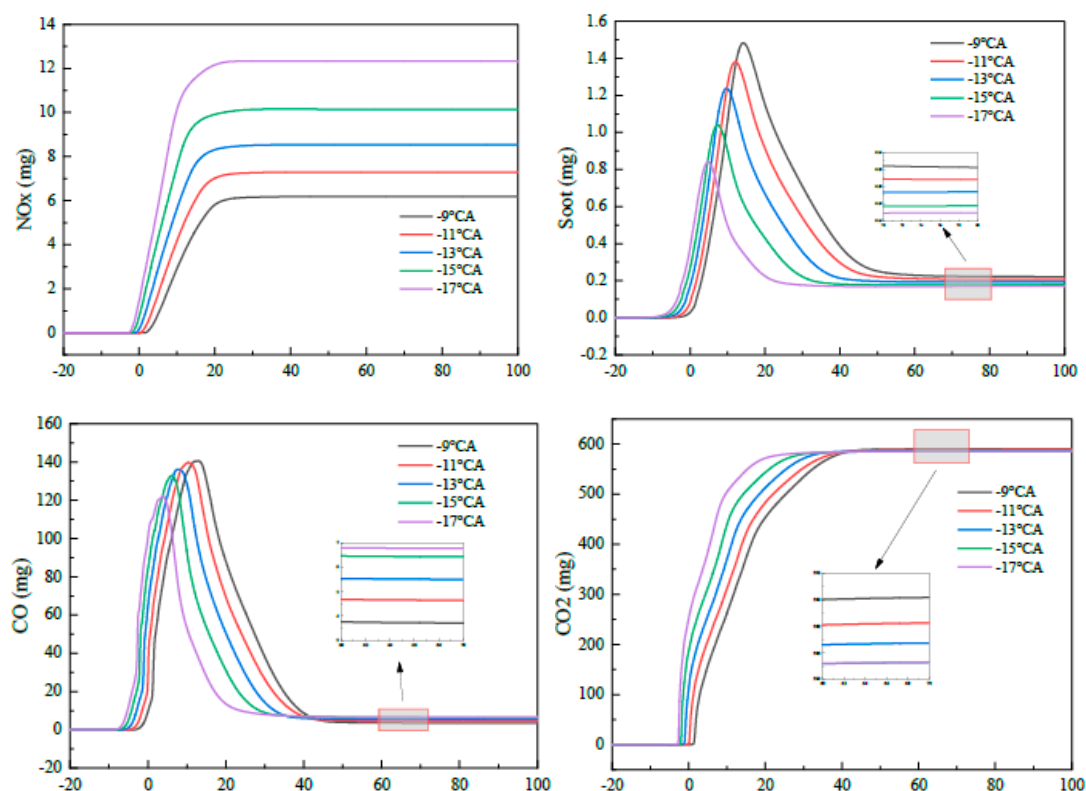
Figure 7 shows the effects of different injection times on CA10, CA50, CA90, and IMEP. CA10, CA50, and CA90 advance correspondingly with the advance of fuel injection time, and IMEP first increases and then decreases. The combustion delay period of the mixed fuel gradually increases, and the combustion duration gradually decreases. At  $-17^{\circ}\text{CA}$ , the longest fuel injection ignition delay period is  $14.4^{\circ}\text{CA}$ , and the shortest combustion duration is  $11.63^{\circ}\text{CA}$ . The IMEP of  $-17^{\circ}\text{CA}$  is increased by  $0.01\text{ MPa}$  compared with that of  $-9^{\circ}\text{CA}$ . This is mainly due to the advance of the injection time. When the mixed fuel enters the cylinder, the pressure and temperature in the cylinder are relatively low; when the spray penetration distance is increased, the fuel diffusion time is extended, and the ignition delay period is increased. At the same time, as CA10 gradually approaches TDC, the volume in the cylinder decreases, the pressure increases, and the temperature rises, which increases the proportion of fuel combustion, thus speeding up the combustion rate and shortening the combustion duration. The reason why IMEP increases first and then decreases is that with the advance of fuel injection time, the center of gravity of combustion gradually approaches TDC from far away, and the work done by the heat released by combustion on the piston increases, resulting in the gradual increase in IMEP [24]. But then the IMEP decreases as the starting point of combustion changes from near TDC to away from TDC.



**Figure 7.** Variation curve of CA10, CA50, CA90, and IMEP under single injection.

### 3.2.3. NO<sub>x</sub>, Soot, CO, and CO<sub>2</sub> Emissions

The effect of injection time on emissions from the diesel engine is shown in Figure 8. It can be observed from the figures that NO<sub>x</sub> emission gradually increases with the advance of oil injection time, while both the maximum production and total emission of soot continuously decrease. Compared to −9 °CA, NO<sub>x</sub> emission at −17 °CA nearly doubles, whereas soot emission decreases by 23.35%. This phenomenon can be attributed to the improved mixing effect of mixed fuel due to the advance of fuel injection time and extension of the combustion delay period, resulting in a reduced proportion of local concentrated mixture. Additionally, ethanol and n-butanol, as oxygen-containing fuels, contribute oxygen atoms during fuel combustion, leading to more complete burning and higher cylinder temperature, which subsequently elevates NO<sub>x</sub> emissions while reducing soot generation. With the advance of the injection time, the maximum production of CO in the cylinder decreases, while the emissions gradually increase, and the CO<sub>2</sub> emissions gradually decrease. The CO emission at −17 °CA increased by 81.5% compared to the −9 °CA injection. The reason for this phenomenon is that with the advance of the injection time, the premix effect of the mixture is good, and CO is fully oxidized to CO<sub>2</sub> in the main stage of combustion, and the maximum amount of CO production decreases. However, due to the advance of the injection time, the pressure and temperature of the fuel entering the cylinder are low, resulting in a prolonged ignition delay period. In this case, part of the gas mixture enters the piston gap and the cylinder boundary. Due to the cold shock effect, these gas mixtures are not sufficiently burned in the gap, ultimately leading to an increase in CO emissions and a decrease in CO<sub>2</sub> emissions.

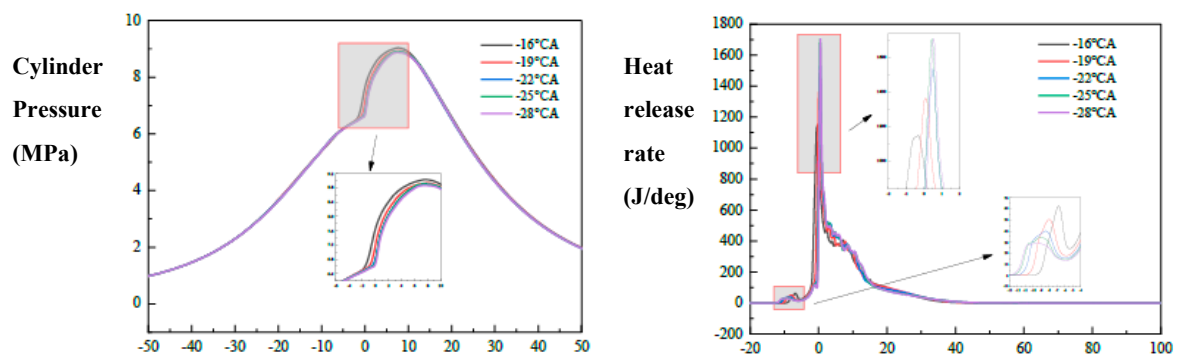


**Figure 8.** Variation curve of different emissions under a single injection.

### 3.3. Combustion and Emissions with Different Prespray Times

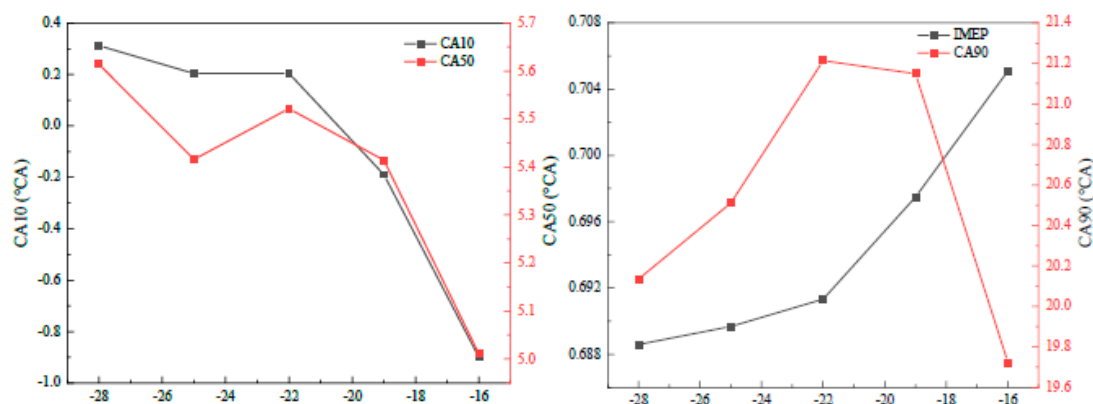
The impact of pre-injection timing on cylinder pressure and instantaneous heat release rate under two injection strategies is shown in Figure 9. The peak values of cylinder pressure and instantaneous heat release rate during the pre-injection stage gradually

decrease with earlier pre-injection times, while those during the main injection stage increase. This phenomenon is mainly attributed to a larger main pre-injection interval at earlier timings, resulting in a greater distance between the piston top dead center and injected fuel, leading to lower temperature and slower combustion of the formed premix due to a longer combustion delay period. Moreover, thinner premix further slows down combustion speed and reduces the peak value of the heat release rate during the pre-injection stage. Longer mixing time for the unburned part with air in this stage results in more mixtures conducive to fire during the primary injection stage when main injection fuel is introduced into the cylinder, causing a faster reaction rate and leading to higher peak heat release rate concentration. Additionally, as the piston descends during the main injection combustion phase with advancing pre-injection time, the volume inside the cylinder increases, which causes a reduction in cylinder pressure.



**Figure 9.** Variation curve of cylinder pressure and heat release rate at different pre-spray times.

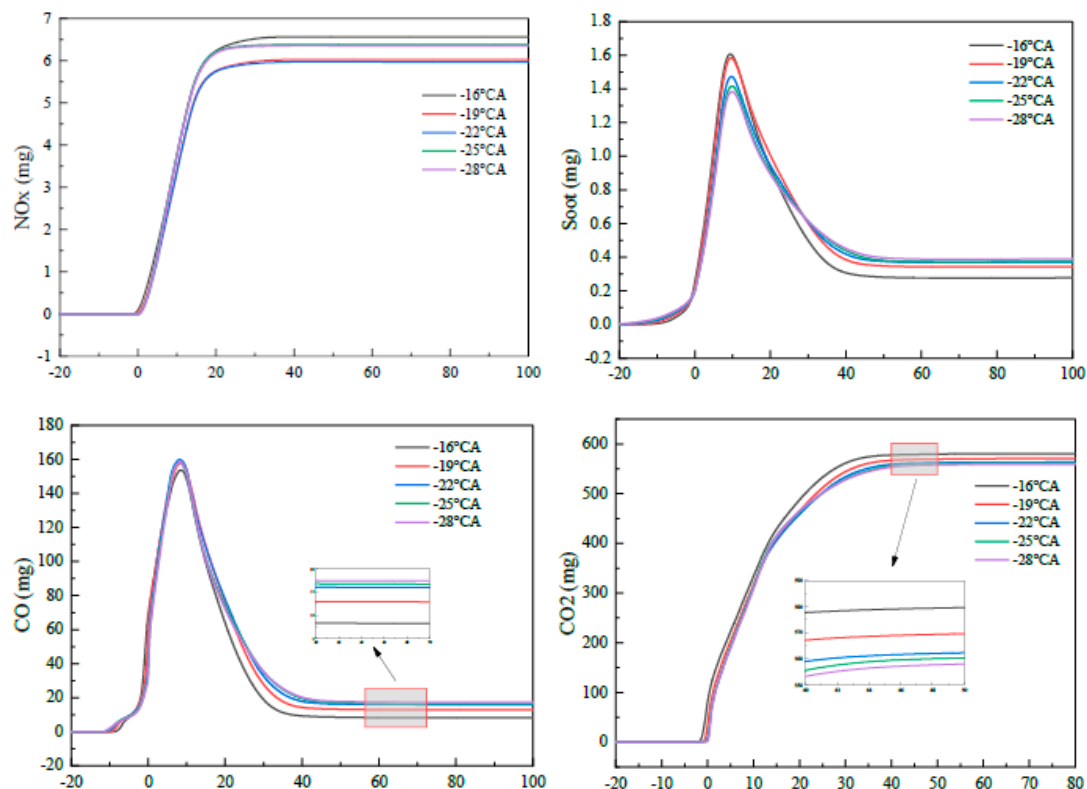
Figure 10 shows the effects of different pre-spray times on CA10, CA50, CA90, and IMEP. With the advance of the pre-spray time, CA10 and CA50 gradually increase. CA90 first increases and then decreases, but the changes for CA10, CA50, and CA90 are not obvious. This is mainly due to the advance of the pre-injection time. The pre-injection fuel in the cylinder mixing time is longer, and a relatively thin mixture is formed. The main injection fuel needs more time to diffuse to improve the equivalent ratio of the mixture, and the fuel will undergo compressed combustion. Since the combustion center of gravity is delayed after the top stop point, the heat released by the combustion of fuel in the cylinder does less work on the piston.



**Figure 10.** Variation curve of CA10, CA50, CA90, and IMEP at different prespray times.

The influence of pre-injection time on pollutant emission of the diesel engine is shown in Figure 11. The  $\text{NO}_x$  emission decreases first and then increases with the advance of pre-injection time, while the soot emission increases gradually. This is because the interval

between the main injection and pre-injection is small at the initial stage. The diffusion time of pre-injection fuel is short, and most of the fuel is burned together with the main injection stage. The initial combustion temperature is low, resulting in a decrease in  $\text{NO}_x$  emissions. With the increase in the main injection and pre-injection interval, the pre-injection fuel is fully diffused, which helps to form a better mixture, resulting in an increase in local high temperature areas during the combustion process. The reason why soot emission increases gradually is that the piston is still far away from TDC, and the hot atmosphere in the cylinder is poor as the pre-injection time advances. At this time, part of the early combustion occurred at a lower temperature and pressure, which is not conducive to soot oxidation and combustion, thus increasing soot emission.



**Figure 11.** Variation curve of the different emissions at different prespray times.

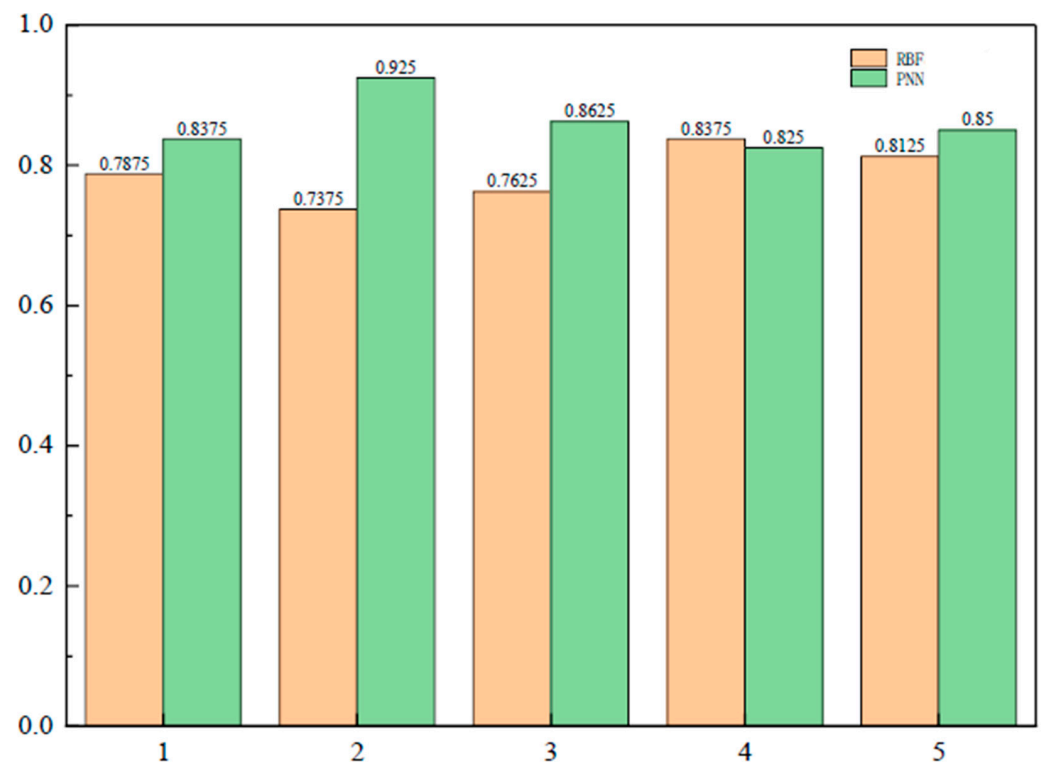
With the advance of pre-injection timing, there is a gradual increase in CO emission and a corresponding decrease in CO<sub>2</sub> emission. Firstly, as the pre-injection time increases, the pressure and temperature within the cylinder decrease, leading to poor fuel atomization and inadequate fuel-air mixing. Consequently, wet wall formation becomes more likely, impeding the complete combustion of the pre-injected fuel mixture. Additionally, ethanol and n-butanol vaporization absorbs heat energy during combustion, resulting in lower overall combustion temperatures. As a consequence of these conditions, an increased amount of incompletely burned fuel enters the gap between the cylinder and the piston. Consequently, incomplete oxidation of generated CO into CO<sub>2</sub> occurs, contributing to elevated levels of CO emissions while simultaneously reducing levels of CO<sub>2</sub> emissions.

### 3.4. Fault Diagnosis of Fuel Injector

In this paper, two sets of fault simulation data are established for the fuel injection advance and delay fault phenomena. Firstly, the simulation data range for fuel injection advance failure is set from  $-13^{\circ}\text{CA}$  to  $-23^{\circ}\text{CA}$  at  $1^{\circ}\text{CA}$  intervals, resulting in a total of 10 datasets. Secondly, the simulation data range for the injection lag fault extends

from  $-7^{\circ}\text{CA}$  to  $3^{\circ}\text{CA}$  at  $1^{\circ}\text{CA}$  intervals, also comprising 10 datasets. Additionally, for the insufficient fuel injection fault phenomenon, the fuel injection range is reduced from 20.25 mg to 6.75 mg with a dataset created every 1.35 mg incrementally. The normal working state is defined as ranging from  $-8^{\circ}\text{CA}$  to  $-10^{\circ}\text{C}$  with an interval of  $0.2^{\circ}\text{C}$ , resulting in a total of ten groups of data as well. Furthermore, during the process of fault diagnosis, six thermodynamic parameters were selected as characteristic indicators: average effective pressure (F1), explosion pressure (F2), crankshaft angle corresponding to explosion pressure (F3), maximum combustion temperature (F4), ignition delay period (F5), and combustion duration (F6).

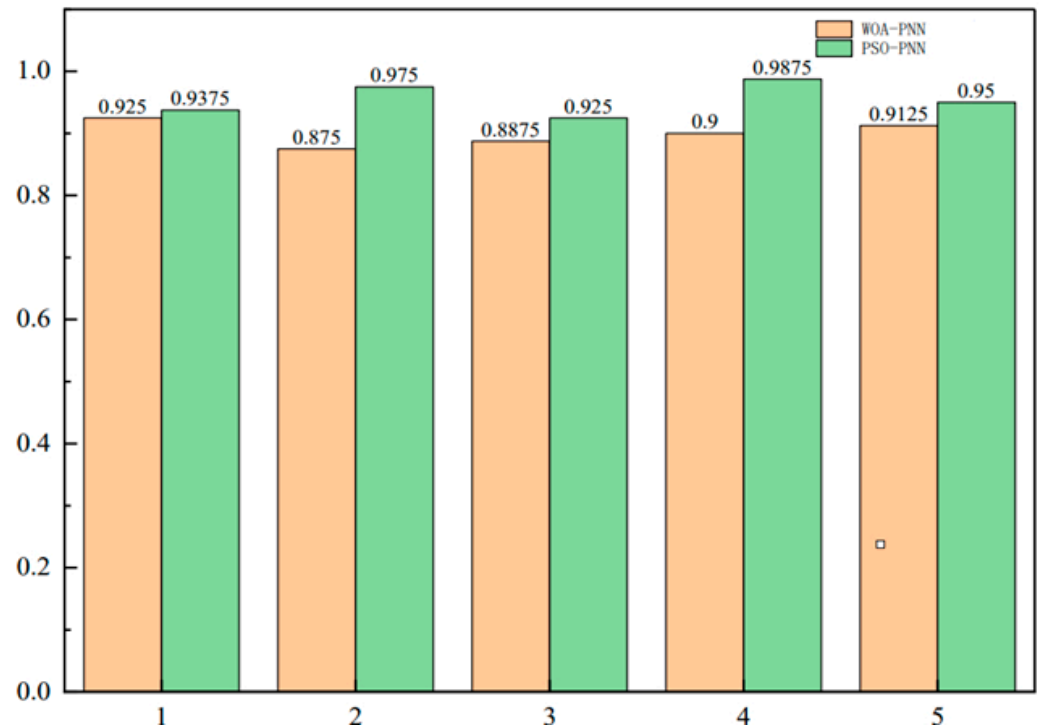
The data of three fault conditions and one normal condition are extended using simulation and data interpolation technology. A total of 400 sets of data are obtained, with 100 sets for each condition. The data was divided into a training set containing 320 sets of data (80% of the total sample) and a test set containing 80 sets of data, 20 for each condition. After that, the data set is substituted into the PNN model for the classification task, and the smoothing factor parameter is set to 1. Then, it is compared with a radial basis function (RBF) neural network. The diagnostic accuracy of the PNN/RBF model is shown in Figure 12. The PNN diagnostic model outperforms the RBF model in effectively identifying fuel injector faults; however, its average accuracy remains at 86%. Therefore, it is imperative to enhance the reliability and accuracy of the PNN diagnostic model for practical engineering applications.



**Figure 12.** Comparison of classification accuracy of PNN/RBF models.

The PSO-PNN and WOA-PNN models are employed for classification tasks, with data sets being utilized. In both optimization algorithms, the initial population is set to 50, the number of iterations is 100, and the variables have a lower bound of 0.001 and an upper bound of 100. Figure 13 presents a comparison of the classification accuracy between these two models. The PSO-PNN model achieves an average accuracy of 90% in accurately identifying injector faults, while the WOA-PNN model demonstrates a higher average accuracy of 95%. In conclusion, the WOA-PNN model exhibits superior advantages in

fault status identification and diagnosis for Marine diesel engines, enabling more precise detection of engine faults. Additionally, the exploration of the innovative hybrid FSO/RF communication system has illuminated a promising path towards addressing the challenges posed by atmospheric turbulence [25].



**Figure 13.** Comparison of classification accuracy of PSO-PNN/WOA-PNN models.

#### 4. Conclusions

The effects of the diesel/ethanol/n-butanol mixture on diesel engine combustion and emission under different ethanol blending ratios, single injection time, and pre-injection time were studied by constructing a marine diesel engine model with CONVERGE software. In addition, this paper takes the injector fault as an example to diagnose and identify three fault phenomena of the injector: fuel injection advance, fuel injection lag, and fuel injection insufficiency. The probabilistic neural network model is optimized by the particle swarm optimization algorithm and whale optimization algorithm, and the fault state identification and diagnosis of the injector are realized. The main research contents and conclusions of this paper are as follows:

1. The calculation model of the marine diesel engine is built with CONVERGE software, and the curves of cylinder pressure and instantaneous heat release rate are obtained by simulation. Then, the reliability of the model is verified by comparison and analysis with the experimental data.
2. The combustion and emission characteristics of D100, D80E10B10, D70E20B10, D60E30B10, and D50E40B10 were studied by modifying the fuel component parameters of the constructed diesel simulation model. The results show that with the increase in the ethanol blending ratio, the peak value of the instantaneous heat release rate increases. The combustion starting point of D50E40B10 fuel mixture is delayed from  $-0.6^{\circ}\text{CA}$  of D100 to  $4.21^{\circ}\text{CA}$ , and  $\text{NO}_x$  emissions are reduced by 12.3%.
3. The combustion and emission characteristics of the diesel engine were studied by changing the injection times of  $-9^{\circ}\text{CA}$ ,  $-11^{\circ}\text{CA}$ ,  $-13^{\circ}\text{CA}$ ,  $-15^{\circ}\text{CA}$ , and  $-17^{\circ}\text{CA}$  at 2000 r/min with mixed fuel D70E20B10. The results show that the peak values

of cylinder pressure and instantaneous heat release rate gradually increase with the advance of injection time, while the carbon smoke emissions gradually decrease, but the NO<sub>x</sub> and CO emissions show a gradually increasing trend. In particular, when the injection time is advanced to  $-17^{\circ}\text{CA}$ , the pressure in the cylinder during combustion rises sharply, reaching a peak value of 10.31 MPa at  $5.51^{\circ}\text{CA}$ , the peak instantaneous heat release rate increases by 3729.94 J/deg compared with the original model, and the crankshaft angle corresponding to the peak value also advances from  $1.7^{\circ}\text{CA}$  to  $-2.47^{\circ}\text{CA}$ . In addition, soot emissions decreased by 23.35%, while NO<sub>x</sub> and CO emissions increased by nearly 1 and 81.5%, respectively.

4. With the advance of the pre-injection time, the peak value of cylinder pressure decreases gradually, and the peak value of heat release rate in the pre-injection stage also decreases, while the peak value of heat release rate in the main injection stage increases. In addition, NO<sub>x</sub> emission decreased at the beginning, and then increased as the pre-injection time continued to advance, while soot emission showed a gradual increase trend.
5. Data input into the PNN/RBF diagnostic model verified that the accuracy of the PNN model was better than that of RBF, but the average accuracy was only 86%, so the model needed to be optimized. Then, the smoothing factor of PNN is optimized using PSO and WOA, respectively. The average accuracy of the PSO-PNN neural network model is 90%, and the average accuracy of the WOA-PNN neural network model is 95%. Therefore, the WOA-PNN neural network model is determined to be the best fault diagnosis model for the fuel injector.

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