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Effect of Methanol/Water Mixed Fuel Compound Injection on Engine Combustion and Emissions

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Abstract: Due to the recent global increase in fuel prices, to reduce emissions from ground transportation and improve urban air quality, it is necessary to improve fuel efficiency and reduce emissions. Water, methanol, and a mixture of the two were added at the pre-intercooler position to keep the same charge and cooling of the original rich mixture, reduce BSFC and increase ITE, and promote combustion. The methanol/water mixing volume ratios of different fuel injection strategies were compared to find the best balance between fuel consumption, performance, and emission trends. By simulating the combustion mechanism of methanol, water, and diesel mixed through the Chemkin system, the ignition delay, temperature change, and the generation rate of the hydroxyl group (−OH) in the reaction process were analyzed. Furthermore, the performance and emission of the engine were analyzed in combination with the actual experiment process. This paper studied the application of different concentration ratios of the water–methanol–diesel mixture in engines. Five concentration ratios of water–methanol blending were injected into the engine at different injection ratios at the pre-intercooler position, such as 100% methanol, 90% methanol/10% water, 60% methanol/40% water, 30% methanol/70% water, 100% water was used. With different volume ratios of premixes, the combustion rate and combustion efficiency were affected by droplet extinguishment, flashing, or explosion, resulting in changes in combustion temperature and affecting engine performance and emissions. In this article, the injection carryout at the pre-intercooler position of the intake port indicated thermal efficiency increase and a brake specific fuel consumption rate decrease with the increase of water–methanol concentration, and reduce CO, UHC, and nitrogen oxide emissions. In particular, when 60% methanol and 40% water were added, it was found that the ignition delay was the shortest and the cylinder pressure was the largest, but the heat release rate was indeed the lowest.

Keywords: diesel engine; methanol; MSR; emissions; injection position

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1. Introduction

The water–methanol–diesel mixed fuel was one of the key technologies used to reduce the pressure, heat release rate, and emissions during the combustion of diesel engines [1,2]. Compared with diesel, when the methanol/water mixture was injected through the front of the intercooler to increase the mixed fuel charge, it played a cooling role through heat conversion and reduced exhaust emissions [3,4]. The latent heat of vaporization of water was higher than that of methanol [5], after the two fuels were mixed, it also reduced the formation of soot [6] during the combustion process, reduced in-cylinder pressure, and increased HRR. Several researchers conducted water injection experiments on small supercharged engines and observed improvements in full-load thermal efficiency, reduction in exhaust gas temperature, and improvement in braking specific fuel consumption [7–10].

Kim et al. [11] studied the effect of direct injection and observed that injected water was beneficial to BMEP and BSFC. Generally, the micro-explosion phenomenon in the water–methanol–diesel combined combustion process affected the combustion of the diesel engine. The water particles in the mixed fuel transformed into superheated steam in the combustion chamber and decomposed into small particles [12], which improved the mixing of combustion and air, thereby increasing the combustion efficiency [13,14]. The intensity of the micro-explosion was affected by factors such as water concentration, droplet size, pressure, and temperature. Due to the high intensity of the micro-explosion, the braking specific fuel consumption was reduced [4,15] and improved the braking thermal efficiency [16].

Kang Zhe et al. [17] studied the effect of high temperature water injected into diesel using the high pressure common rail injected system, and found that the high temperature and high pressure environment in the cylinder could improve the water evaporation speed, increase the effective working fluid in the cylinder, increase the work volume, increase the indicated thermal efficiency by about 8%, and reduce the cyclic fluctuation of the combustion process. Nicholls et al. [18] found that combustion with water can reduce nitrogen oxide emissions. Dryer [19] concluded that water-blended combustion can simultaneously reduce the nitrogen oxide emissions and particulate matter from diesel engines.

Fuel water emulsification technology promotes the combustion process mainly by using the "micro-explosion" theory [14]. When the water-containing fuel is introduced into the high temperature area of the combustion chamber, due to the water-in-oil phenomenon, these water particles evaporate by themselves in the hot combustion chamber air and split into scattering beams, and then the diesel oil is dispersed into tiny droplets to improve the formation of the mixture [20]. This phenomenon is called "micro-explosion" and it promotes the phenomenon of "secondary atomization" of fuel droplets. However, the water particles absorb a certain amount of heat in the form of latent heat, thereby reducing the peak temperature.

Abuzaid and Sajith et al. [21,22] found that water-blended diesel fuel can effectively increase power. The use of water-blended diesel by Basha et al. [23] increased the efficiency of the diesel engine by approximately 6.7%. Ahmad et al. [24] studied the combustion emission characteristics of different blending ratios and found that both nitrogen oxides and particulate matter were reduced.

The scope of the research was to study the effect of the water–methanol mixture on engine performance and compare them with the performance of pure water and pure methanol. The questions needed to explore whether the performance of the water–methanol mixture was better than pure water or pure methanol. According to the author's knowledge, research in this area had not been extensively studied in engine research so far. This research aimed to solve this research gap. The novelty of this research was: the experimental study to determine the influence of the water–methanol mixture on engine performance, and compare the performance of the water–methanol mixture engine with pure water and alcohol methanol to understand the influence of the water–methanol mixture on exhaust gas emissions and exhaust gas temperature.

2. Experimental Equipment and Experimental Set

This experiment was carried out on a Yunnei D30 in-line 4-cylinder enhanced intercooler common rail diesel engine. Table 1 presents the engine specifications. Figure 1 presents a schematic diagram of the engine test bench.

Methanol/water injection was controlled by a separate ECU. The ECU communicated with the original ECU via the CAN bus, synchronized the crankshaft and camshaft signals, connected to the computer via ETAS 592, and controlled the methanol injection cycle and pulse width via the INCA system. The position shown at 8 in Figure 1 was the pre-intercooler injection position at 40 cm away from the turbocharger system, to prevent the methanol injection device from getting too close to the supercharger and damaging the injector. On the other hand, the waste heat of the exhaust gas was used to reduce the negative influence caused by the high latent heat of vaporization and increase the intake air temperature, promoting the evaporation of methanol fuel, fully mixed with air. A set of thermocouples was installed 10 cm before and after each water/methanol injection position to measure the changing trend of intake air temperature before and after methanol injection.

Table 1. Engine description.

Engine Type	D30 4 Cylinder in-line diesel engine
Air intake form	Turbocharged inter-cooled
Cylinder NO.	4
Fuel	Diesel
Displacement	2.977 L
Maximum power output	115 kW
Speed	3200 rpm
Max Hp	156 Hp
Max Torque	350 N·m
Max torque speed	1500–2700 rpm
Min fuel consumption for full load	<208 g/kW·h
Bore × Stroke	95 mm × 105 mm
Valve train	4

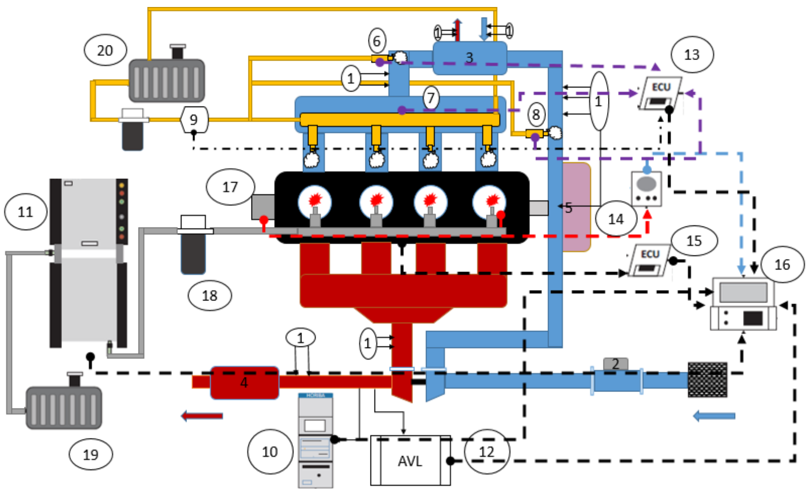


Figure 1. A schematic diagram of the engine test bench; (1) pressure/temperature sensor, (2) airflow meter, (3) intercooler, (4) silencer, (5) dynamometer, (6) intercooler injection point, (7) intake manifold injection point, (8) intercooler injection point, (9) methanol flow Meter, (10) MEXA-584L, (11) FCMM-3 fuel consumption measuring instrument, (12) AVL DISMOKE 4000, (13) methanol control unit, (14) KISTLER Charge Amplifier 5018, (15) diesel injection control unit, (16) computers, (17) HRC-F15-720 angle encoder, (18) filters, (19) diesel oil container, (20) methanol container.

The exhaust gas was measured by the HORIBA MEXA-584L analyzer, and the measuring hole was set at the rear end of the turbocharger. The cylinder pressure data in the experiment had an average value of 100 consecutive data cycles. The cylinder pressure signal collected through the KISTLER 6056A pressure sensor, amplified by KISTLER Charge Amplifier 5018, and collected by the HR-CA-B1 combustion analyzer. The shaft encoder used an HRC-F15-720 angle encoder. The sampling resolution of the cylinder pressure signal was 0.5 CAD.

3. Fuel Properties

The diesel used in the experiment was commodity 0# diesel, and the sulfur content of the diesel was less than 10 ppm. The methanol used was chemical with a purity of 99.99%. The temperature always fixed at 25 °C during methanol injection. The water mixed with methanol was distilled water at room temperature and pressure. Table 2 shows the properties of methanol and diesel. The data of Table 2 was from the Chemical database and China National Petroleum Corporation Laboratory [25].

Table 2. Characteristics of diesel and methanol [5,9,22].

Properties	Methanol	Diesel	Water
Density g/mL	0.7866	0.829	1.0
Boiling point °C	65	187–343	99.975
Octane number	136	15–25	-
Cetane number	3.8	46	-
Energy density MJ/L	16	46	+∞
LHV MJ/kg	20.1	42.8	-
Latent heat of evaporation MJ/kg	1.2	0.23–0.60	2.257
Melting point °C	−97	−40–34	0
Cooling point °C	12	74	0
Auto-ignition temperature °C	463	235	-
Viscosity Pa·s	0.5445×10^{-3}	-	100

4. Experimental Methods and Procedures

The experiment mainly studied the effect of methanol/water injection at pre-intercooler injection position on engine performance, combustion, and emissions. The engine parameters were at the same load (200 N·m) at the pre-intercooler injection position was at a constant speed of 1600 rpm. Water/methanol blending volume test point was 100% methanol, 90% methanol + 10% water, 60% methanol + 40% Water, 30% methanol + 70% water, 100% water. Each water/methanol blending volume test point was injected according to five kinds of methanol/water substitution rate (MSR) 0%, 10%, 20%, 30%, and 40%. After each experimental point was stable for 3 min, the data would record, and the data of each experimental point recorded three times in a row.

The constant volume adiabatic model of CHEMKIN software was used to simulate the reaction process of fuel ignition and combustion after the piston of an internal combustion engine reached the top dead center. In order to measure the methanol substitution situation in the methanol–diesel dual–fuel combustion mode, MSP was defined as the percentage of the calorific value of methanol complete combustion to the total calorific value, expressed by the methanol substitution rate, calculated by Equation (1) [26]. The calculation method of the equivalence ratio is shown in Formula (2).

$$\text{MSR} = \frac{m_a \cdot Q_{LHV,a}}{m_d \cdot Q_{LHV,d} + m_a \cdot Q_{LHV,a}} \times 100\% \tag{1}$$

$$\varphi = \frac{m_a AFR_a^{th} + m_d \cdot AFR_d^{th}}{m_{air}} \tag{2}$$

where m_a and $\mathrm{m_d}$ represent the initial mass of methanol and n-heptane methanol in kg; where $Q_{LHV,a}$ and $Q_{LHV,d}$ represent the low heating values of methanol and n-heptane, MJ/kg; where AFR_a^{th} and AFR_d^{th} represent the stoichimeric air-fuel ratio for methanol and n-heptane; where m_{air} represents the mass of air.

The specific performance is the use of numerical calculation methods to simulate the ignition and combustion process of n-heptane/methanol mixed fuel, including the influence of different methanol substitution rates on the ignition delay and temperature rise of the fuel. With the aid of chemical elementary reaction paths, concentration changes of main components, and chemical reaction rate analysis, the influence factors of methanol blending on the ignition characteristics of n-heptane were explored from the aspects of macro-combustion characteristics and chemical reaction kinetics. In this paper, when the equivalent ratio was 1, the initial pressure was 20 bar, and different proportions (20%, 40%, 60%, 80%) of methanol were mixed, respectively. The mixed flame conditions were compared. We ensured that the total mole fraction of fuel remained unchanged, and the total heating value of the fuel after combustion was the same. The setting of each working condition is shown in Table 3.

Table 3. Simulation conditions (% molar volumes of each constituent).

φ	MSR/%	N-Heptane	CH_3OH	O_2	N_2
1	0	1	0	11	41.36
1	20	1	1.77665	13.66497	51.3803
1	40	1	4.737733	18.1066	68.08081
1	60	1	10.6599	26.98985	101.4818
1	80	1	28.4264	53.63959	201.6849

5. Results and Discussion

This paper used the Chemkin simulation system combined with experiments to study engine combustion and performance indicators, such as ID, BSFC, cylinder pressure (CP), HRR, and ITE, and discussed and analyzed engine emissions, such as nitrogen oxides, unburned hydrocarbons, and carbon monoxide.

5.1. Ignition Delay (ID)

The ignition delay was measured by connecting the cylinder pressure sensor and the combustion analyzer to the computer through the CAN line, using the combustion analysis software to collect the cumulative heat release rate data, and integrate the heat release rate data. The crank angle when the heat release rate reached 10% was compared with the crank angle when the pure diesel combustion heat release rate reached 10%, and the ignition delay value was calculated.

In diesel engines, the ignition delay was a very important issue, because it played a vital role in the presence of NOx in starting energy, combustion rate, knocking, noise, and exhaust emissions. Hydroxyl (−OH) was an important reason to affect ignition delay. Hydroxyl (−OH) was the main free radical involved in the dehydrogenation of n-heptane and had hydrophilicity. Through the Chemkin simulation system, it could be observed (from Figure 2) that the addition of methanol increase was a very intuitive suppression of the amount of the growth of OH free radicals in the low-temperature exothermic stage of n-heptane [27]; the Chemkin simulation parameters are shown in Table 3.

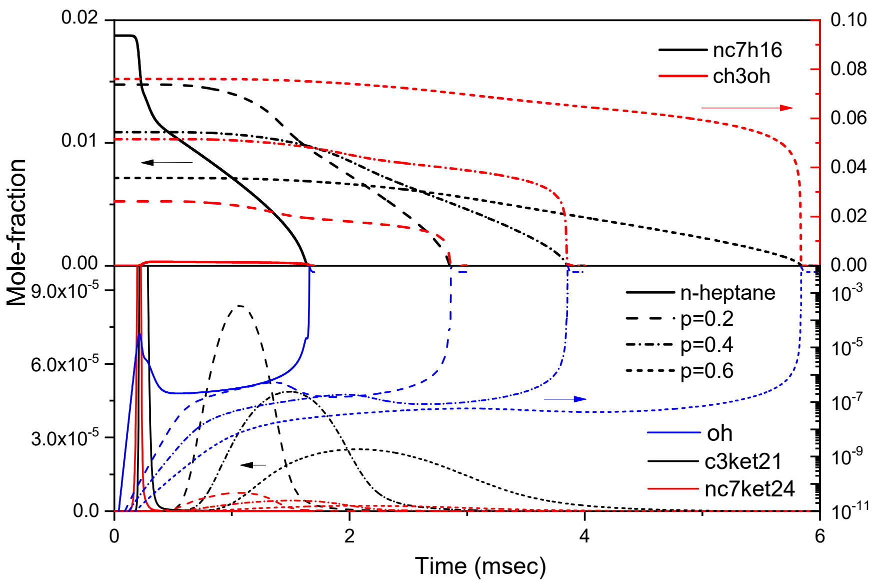


Figure 2. Under low temperature, the concentration change curve of the fuel substance.

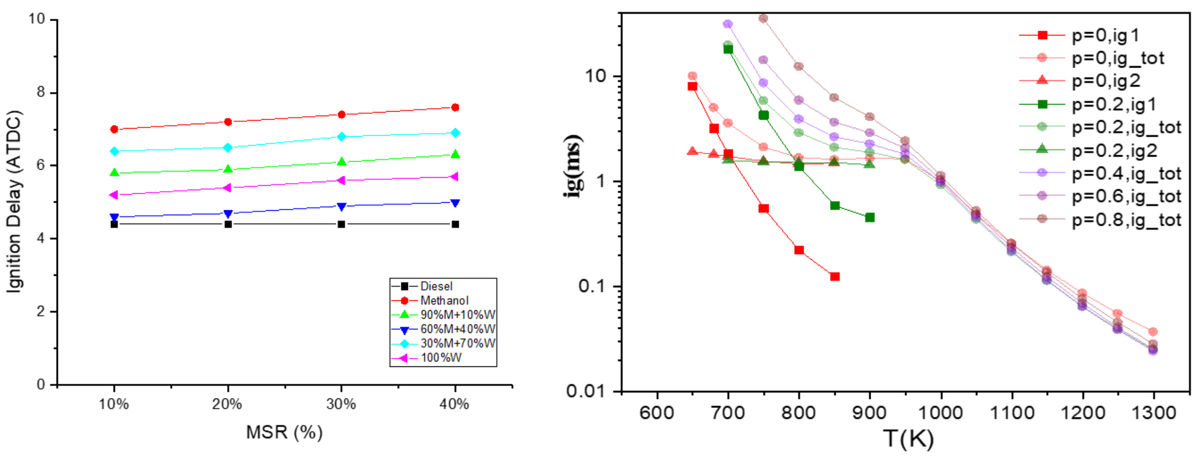
When MSP = 20%, the peak concentration of OH in the low-temperature exothermic phase was 100 times lower than that in pure n-heptane. The higher the methanol substitution rate, the more suppressed the increase in the OH radicals in the low-temperature exothermic phase.

When MSP > 20%, although the concentration of OH radicals in the early stage of the low-temperature reaction was still rising, its upward trend slowed down, and there was no obvious peak, which was reflected in the heat release rate, i.e., the low-temperature ignition of the mixed gas was not monitored. However, the low-temperature reaction still existed and worked at this time, but the intensity was greatly reduced.

Due to the hydrophilicity of the hydroxyl group—when water was added, the hydroxyl group (−OH) first contacted with water, which reduced the inhibitory effect on n-heptane and shortened the ignition delay [28]. The specific heat capacity of water was larger and the ambient temperature was lowered. Compared with pure diesel, the ignition delay was longer. So with the addition of water, the ignition delay time increased.

Wuethrich et al. [29] experimentally studied the spray and combustion of water added to diesel engines and found the same results. The increase in the ignition delay of the mixed droplets should be physical because the larger heat capacity of water slowed down the temperature rise. In addition, when the mixed fuel was burned, due to the evaporation of water on the surface of the droplets, the ignition delay significantly increased and shortened the burning time [14]. When the temperature of a certain point on the interface of the mixed droplet reaches the superheat limit, a micro-explosion may occur.

Figure 3 illustrates the relationship between the change in ignition delay and the fuel mixture type of the engine. Figure 3b presents the results obtained using the Chemkin simulation system, showing that the addition of methanol prolongs the ignition delay. As methanol was injected into the diesel engine, the −OH molecule in the reaction with the diesel was robbed for the first time, which reduced the reaction efficiency of the diesel. When water was injected, methanol snatched −OH from the water molecules, reducing the inhibition of −OH growth in diesel, and the addition of water diluted the concentration of methanol and alleviated the impact of methanol's low calorific value on the mixed fuel. From the experiments—results showed that the amount of water added at 10% and 40% would lead to the ignition delay in advance, as shown in Figure 3a. Due to the micro-explosion of water in the cylinder, the greater the water content, the more obvious the micro-explosion. Thus, with an increase in the amount of water added, the ignition delay was shorter. From Figures 2 and 3b, we can see that, when MSR = 0.6, the ignition was longer than MSR = 0.2 and 0.4, because the methanol robbed the −OH in the reaction with the diesel, causing the ignition delay. However, as seen in Figure 3a, water injected into the diesel engine caused the delay in advance because of the micro-explosion combustion of water/diesel blending. Compared with methanol, the ignition delay advanced by about 7%, 14%, 21%, and 37% after water injection.



(**a**) (**b**)

Figure 3. The relationship between the change of ignition delay and the type of engine fuel mixture (**a**) ignition delay data of experiment, (**b**) ignition delay data of simulation.

5.2. Braking Specific Fuel Consumption (BSFC)

Figure 4 illustrates that, as the percentage of water/methanol fuel in the engine increase, the diesel BSFC decreased. The reason for lowering diesel BSFC was the mixing ratio of water–methanol–diesel oil and the micro-explosion caused by superheated steam and water droplets. Due to the increase in viscosity, after the injection of water and calorific value was lower, the specific heat capacity of water was higher than that of methanol, and the lower cylinder temperature caused by the evaporation of water caused a long-term mixed reaction, and avoided the occurrence of knocking, and increased diesel BSFC.

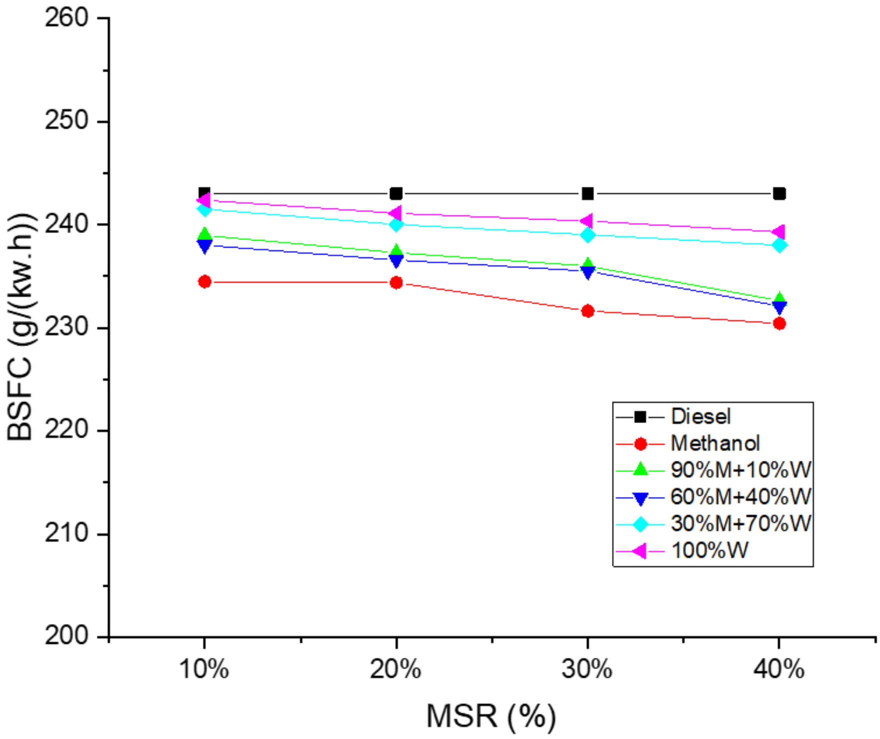


Figure 4. Changes of diesel BSFC under different mixing ratios.

5.3. Indicated Thermal Efficiency (ITE)

Thermal efficiency was an indicator of the conversion of fuel energy into useful work. Figure 5 shows that the indicated thermal efficiency decreases with the increase in mixed fuel consumption. Compared to the combustion of the pure diesel compound injection engine, when the water–methanol–diesel mixture was used, the micro-explosion of water droplets caused lower mixed fuel consumption, and thus obtained the maximum ITE. Figure 5 shows that with 90% methanol–10% water and 60% methanol–40% water, ITE had a significant increase, because the micro-explosion process was the main reason for the improvement of the indicated thermal efficiency.

Figure 4 shows that methanol had the lowest fuel consumption ratio, but in Figure 5, it shows that the indicated thermal efficiency (ITE) of methanol was also the lowest, mainly because of the micro-explosion playing a key role during the combustion process in the mixed fuel of water and diesel methanol. During the micro-explosion, it promoted the mixed combustion of fuel increasing the pressure rise ratio and expansion ratio in the cylinder, accelerating the combustion, and increasing ITE. Compare with methanol, the ITE increased about 0.5%, 1%, 1.6%, and 2.2% after water was injected.

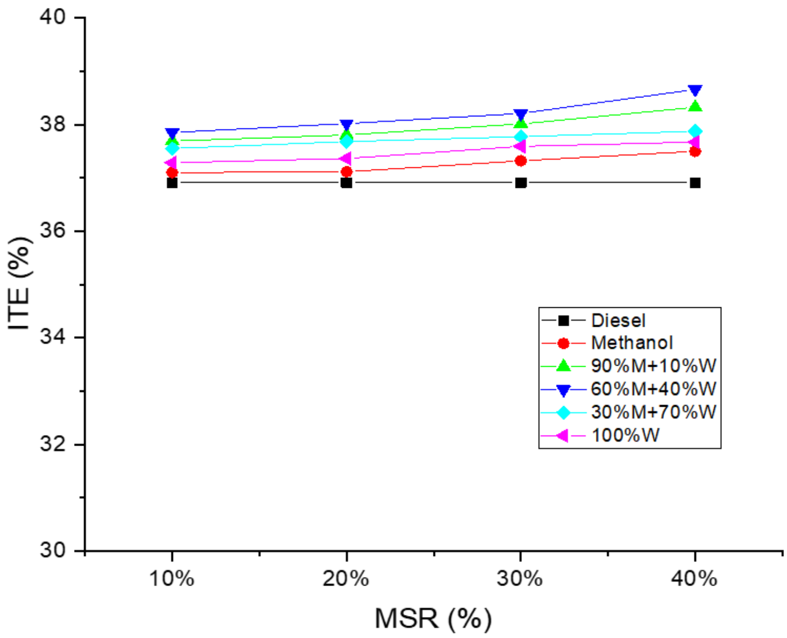


Figure 5. The change of indicated thermal efficiency (ITE) with different fuel mixture ratios.

5.4. Cylinder Pressure and Heat Release Rate

For the three water–methanol concentrations in the methanol fuel used, the pressure peaks increased due to the evaporation of water, which was caused by the micro-explosion of water causing the premature combustion of the diesel fuel. Figure 6 showed higher cylinder pressure of 60% methanol + 40% water and 100% water, and a shorter ignition delay caused lower HRR. As the injection ratio increased, a lower cylinder pressure and higher HRR was found, because the high latent heat of vaporization of water reduced the temperature in the cylinder and caused ignition delay, and prolonged the diesel and methanol mix in the combustion chamber, making the combustion more uniform in the combustion chamber instead of just spreading around the spray [30]. It can also be observed that, as the methanol fraction increased, the maximum cylinder pressure crank angle was delayed, which can be attributed to the ignition delay caused by the presence of more methanol fuel. In the early stage of diesel combustion, due to methanol containing a lower cetane number and subsequent combustion of methanol, the high flame speed of methanol caused the pressure in the cylinder to rise rapidly. The combustion in the compression stage would also aggravate the pressure increase. The maximum cylinder pressure should not be too high to avoid high noise from the engine and avoid the phenomenon of engine knocking. Adding water could advance the time of the premixed combustion. With a small amount of water injection, water evaporation increased the density of gas in a finite volume, and increased the cylinder pressure. As the amount of water increased, the pressure peak in the cylinder and the heat release rate decreased, which minimized the number of free radicals in the combustion chamber and delayed the formation of free radicals.

Figure 6 shows the change in the heat release rate of the test fuel from the combustion stage to the beginning of expansion under different crank angles and loading conditions. Due to the enhanced ignition delay period, more fuel was burned, increasing the heat release rate of the mixed combustion. As the amount of water injected increased, the methanol content reduced—reducing the negative impact of methanol on diesel, reducing methanol robbed the –OH in reaction with diesel, shortened the ignition delay, and enhanced the combustion duration for fuel mixing and combustion. The evaporation of the water and methanol mixture caused the cylinder pressure to rise, promoted combustion, increased combustion duration, lost heat, and caused the heat release rate to lower. Therefore, the highest cylinder pressure and the lowest heat release rate were found in pure water or a 60% methanol/40% water mixture.

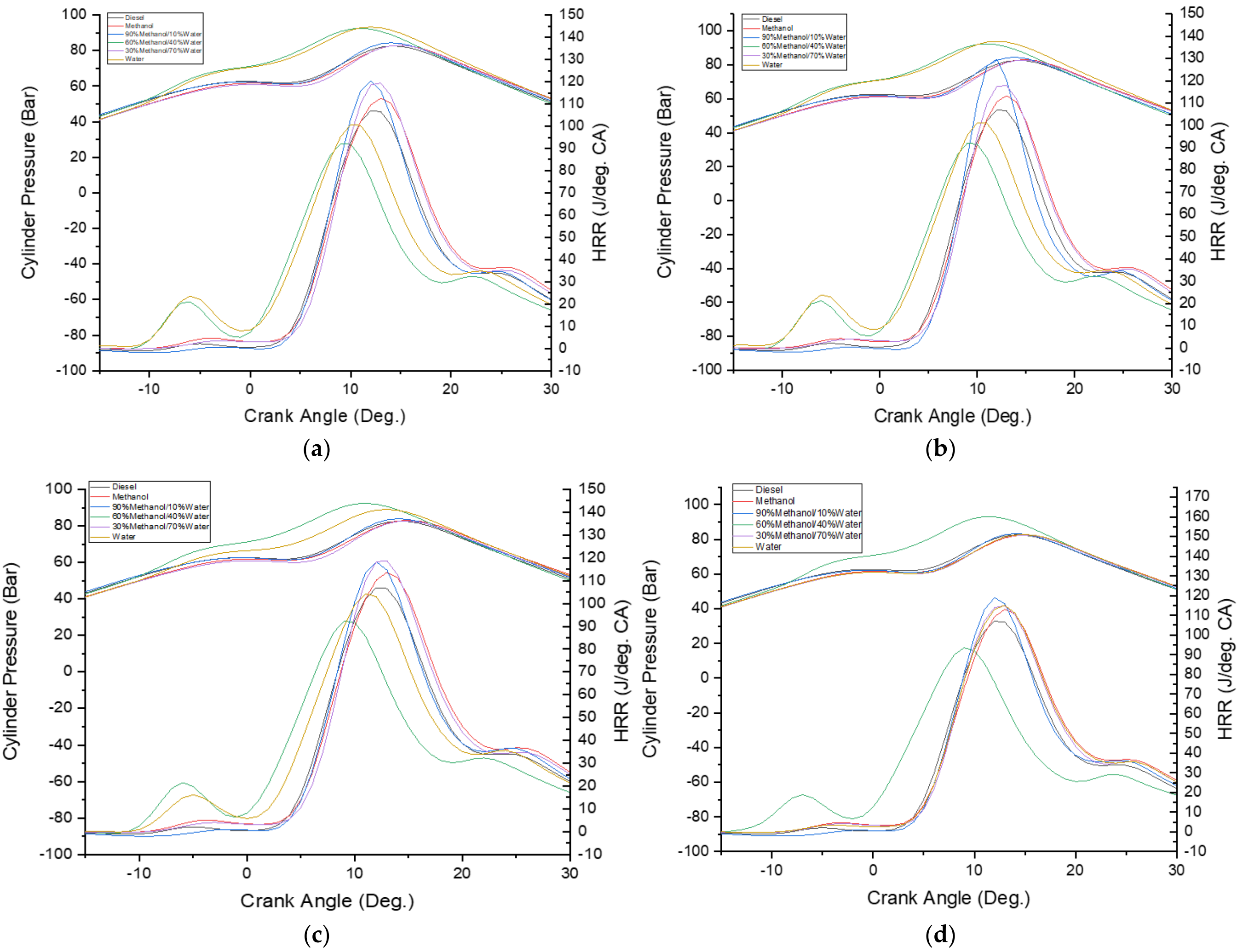
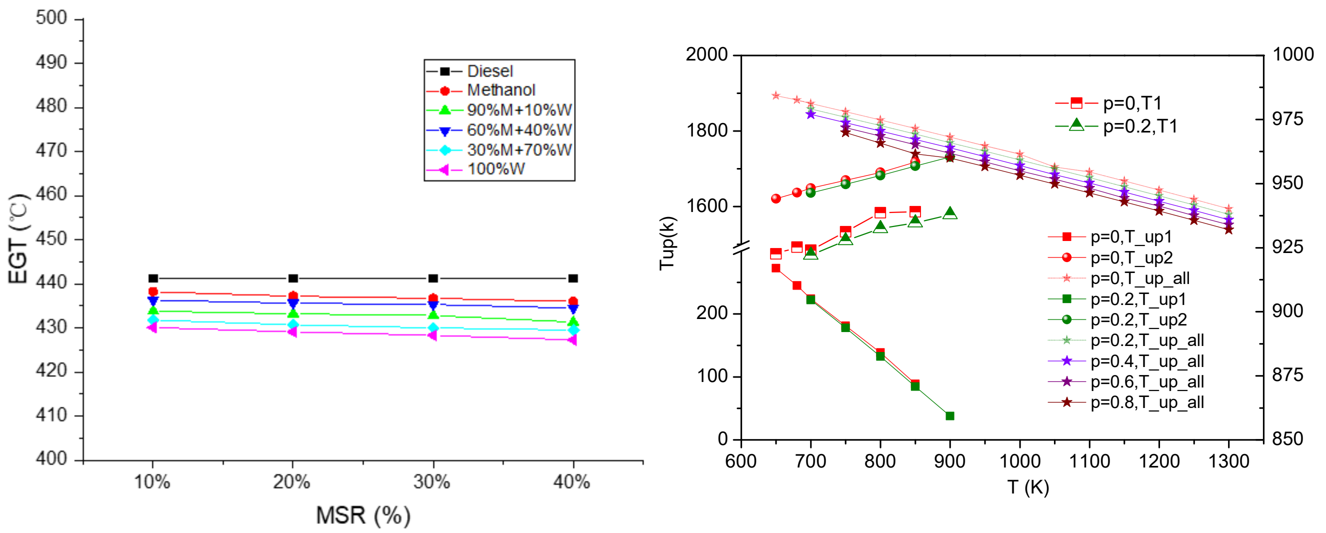


Figure 6. Changes in cylinder pressure and heat release rate under different fuel mixture ratios and substitution rates ((**a**) substitution rate is 10%, (**b**) substitution rate is 20%, (**c**) substitution rate is 30%, (**d**) substitution rate is 40%).

Figure 7 shows the influence of different water–methanol mixture fuel ratios on exhaust gas temperatures through the experiments. It was found that the exhaust gas temperature of the mixed fuel decreased with the increase of the mixed fuel ratio, and showed a linear trend. Figure 7 showed the EGT of the mixture as a function of combustion delay. The increase in water and water/methanol could reduce the discharge temperature. The main reason for the decrease in EGT was the increase in work done during the expansion process and heat transfer. Therefore, more energy was extracted from the gas, thereby reducing the exhaust temperature. In addition, due to the energy exchanged between the air and the water–methanol mixture, the additional air charge lowered the temperature. As the amount of water and water–methanol increased, the increased air charge cooling caused the exhaust temperature to decrease.



(**a**) (**b**)

Figure 7. The influence of different water–methanol mixture fuel ratios on exhaust gas temperature (**a**) EGR data of experiment, (**b**) EGT data of simulation.

As presented in Figure 7b—the Chemkin simulation system was used to simulate and calculate the temperature changes at different substitution rates. It was found that as the substitution rate increased, the exhaust gas emission temperature could be reduced [27]. The main reason was because the high latent heat of vaporization caused a large amount of heat loss during evaporation [28], lowered the cylinder temperature, thereby lowering the exhaust gas discharge temperature.

6. Emissions

When running with better test fuels on different engine loads, a flue gas analyzer was used to record various diesel engine exhaust emission components, such as nitrogen oxides (NOx), unburned hydrocarbon (HC), and carbon monoxide (CO), discussed as follows.

6.1. Carbon Monoxide (CO) Emissions

In diesel engines, the measurement of carbon monoxide emissions was very important, because carbon monoxide could provide fuel produced by incomplete combustion in the combustion chamber. Carbon monoxide emissions were the result of incomplete oxidation of the carbon present in the fuel due to lack of oxygen.

Figure 8 shows that, after the injection of methanol–water in the engine, at the same speed, the high temperature in the cylinder had slowed CO emissions. Due to the influence of the micro-explosion of water droplets, the formation of better mixing and excessive oxygen promoted the conversion of CO to CO_2, causing the reduction of CO emissions. Therefore, adding an amount of water reduced CO, due to micro-explosion, which increased the probability of fuel contact with air, and more complete combustion, leading to less CO production.

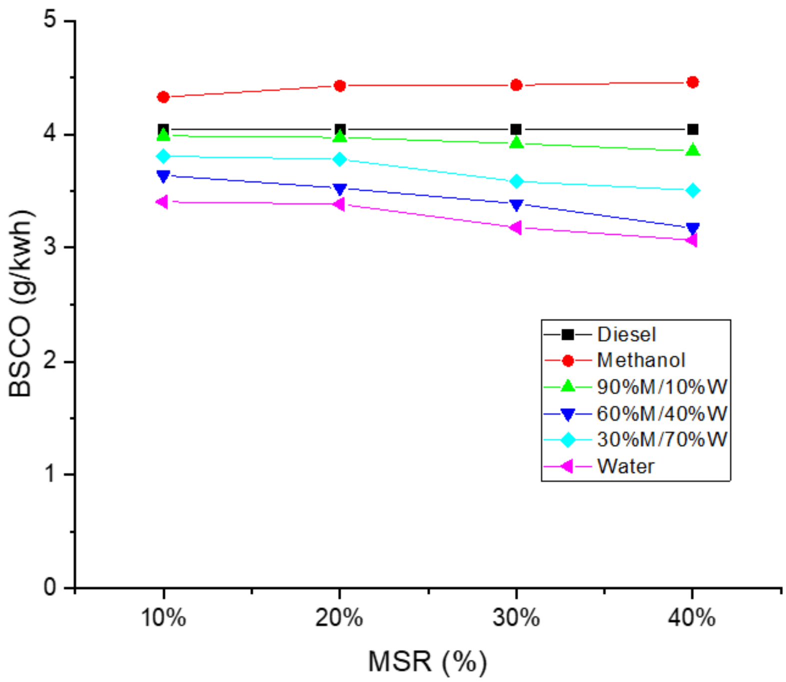


Figure 8. Changes in CO emissions with different fuel mixture ratios.

6.2. Hydrocarbon Emissions (HC)

Hydrocarbon emissions were unburned or partially reacted fuels. The emissions of hydrocarbons had harmful effects on the environment and human health. Figure 9 shows that the HC concentration decreased with the decrease of the methanol mix ratio. With a torque of 200 N·m, when methanol was injected at the injection position, the amount of fuel injected was large, resulting in a low air–fuel ratio (rich mixture), and caused insufficient oxygen, or led to incomplete combustion, thereby increasing the concentration of hydrocarbons. However, after the water was injected into the cylinder, the micro-explosion phenomenon of the water droplets enhanced the air combustion mixture, thereby improving the fuel combustion, reducing HC emissions. Therefore, compared with methanol, it reduced the BSHC after water injection, because of the micro-explosion phenomenon of water.

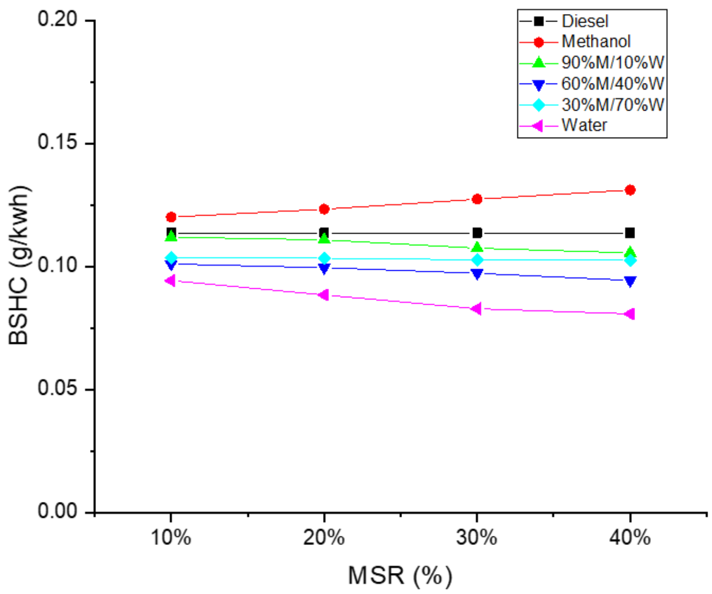


Figure 9. HC emission changes with different fuel mixture ratios.

6.3. Emissions of Nitrogen Oxide (NOx)

During the compression combustion process, as the temperature increased (>1600 °C), stable nitrogen (N_2) reacted with O_2 present in the air, mainly forming nitric oxide (NO) and a small amount of nitrogen dioxide (NO_2) and traces of other nitrogen oxides [31]. The temperature rise in the combustion cylinder, the availability of oxygen, and the residence time were the main influencing factors of nitrogen oxides. The use of direct water injection in diesel engines could significantly reduce nitrogen oxide emissions. The water entered the cylinder in a very fine atomized form and evaporated quickly. The water vapor in the cylinder reduced the partial pressure of oxygen during combustion. In addition, the maximum combustion temperature was reduced because the surrounding heat was absorbed during the vaporization process. Therefore, water with a high molar heat capacity was the reason for the reduction in the formation of nitrogen oxides [32,33]. It caused an increase in the heat capacity of the fuel in the cylinder. Therefore, as the temperature in the cylinder decreased, nitrogen oxide emissions significantly reduced.

Figure 10 shows that, under the conditions of a higher water–methanol mixture ratio, the amount of fuel injected into the combustion chamber increased, but due to the high latent heat of vaporization of the water–methanol mixture, the temperature in the cylinder decreased, the amount of heat release decreased, and the gas temperature lowered, contributing to the reduction of NOx emissions. Among them, under the mixed fuel of 100% water, the heat release rate generated was lower, and the NOx emission generated was lower.

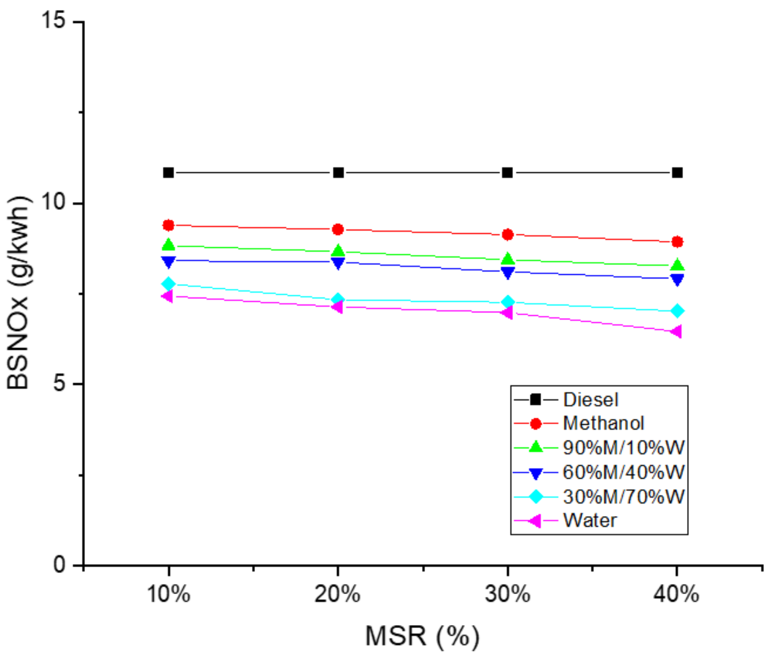


Figure 10. Changes in NOx emissions with different fuel mixture ratios.

7. Conclusions

The combustion reaction mechanism of methanol and n-heptane was simulated by the Chemkin calculation example, and it was found that hydroxyl (−OH) was the main factor affecting ignition delay. After methanol was added, the generation of n-heptane hydroxyl radicals was inhibited and the ignition delay was prolonged. After adding water, the inhibition of the growth of the n-heptane hydroxyl group was reduced, the ignition delay was shortened, and the heat release rate was reduced.

Since the energy density of methanol was lower than that of diesel, the fuel consumption rate of the mixed gas containing methanol was higher than that of the mixed gas without methanol. After spraying water, it could replace part of the air, thereby reducing fuel consumption.

The effects of the mixed fuel ratio, combustion stage, and total fuel flow rate on exhaust temperature were studied. As the fuel mixture ratio increased, the exhaust gas temperature of each fuel mixture decreased linearly. Improving the combustion phase and increasing fuel charge cooling were the reasons for the decrease in exhaust gas temperature. However, for the same combustion stage, since the heat of vaporization of methanol was lower than that of water, the higher the content of methanol in the mixed gas, the higher the exhaust temperature. It showed that fuel cooling had a certain effect on exhaust gas temperature.

From the experiment, it was easy to find that 60% methanol + 40% water had the lowest brake-specific fuel consumption and the highest indicated thermal efficiency, and had higher in-cylinder pressure and lower HRR. The high latent heat of vaporization methanol and the micro-explosion of water, with the interaction, would provide the best results.

Due to the impact of the micro-explosion of water droplets, increasing the water–methanol concentration could increase the braking-specific fuel consumption and the indicated thermal efficiency. The water–methanol injection led to a lower combustion temperature, reduced NOx emissions, the effect of perfect mixing, and atomization of water–methanol–diesel, and reduced CO and UHC emissions. After using the water–methanol–diesel compound injection, the knocking phenomenon disappeared and the load range was expanded.

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