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Effects of Anhydrous and Hydrous Fusel Oil on Combustion and Emissions on a Heavy-Duty Compression-Ignition Engine

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Abstract: The efficient application of oxygen-containing clean fuels in engines has always been a research focus. With the increase in ethanol production, the output of fusel as a co-product is also increasing. The application of fusel is also an effective way to lessen the consumption of fossil fuels. Therefore, the influences of fusel on performance and emissions were investigated in the current study on a six-cylinder heavy-duty compression-ignition engine and revolved around the WHSC test cycle. The three test fuels were diesel, F20NW (the volume proportion of anhydrous fusel is 20%, and the rest is pure diesel), and F20WW (the volume proportion of hydrous fusel is 20%). The addition of fusel improved BTE, reduced NOx and soot emissions, and thermal efficiency and emissions were further improved in combination with EGR optimization. In terms of WHSC, the improvement effect of hydrous fusel was the best. The equivalent fuel consumption, NOx, soot, and CO_2 emissions of F20WW were reduced by 1.77%, 37.49%, 17.38%, and 1.32%, respectively, with the optimization of EGR compared with pure diesel. The addition of 20% hydrous fusel combined with the introduction of EGR can be directly applied to existing diesel engines and achieve a simultaneous reduction in fuel consumption and emissions.

Keywords: diesel engine; fusel; combustion; emissions; thermal efficiency; EGR

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1. Introduction

With the extensive application of engines in many fields of human society because of their high reliability and low cost [1–3], the engines consume large amounts of fossil fuels, which contributes to air pollution and energy shortages [4–11]. In the actual operation of an internal combustion engine, in addition to HC, CO, NOx, PM, and other harmful emissions, CO_2 as a greenhouse gas, which will cause global warming, is also a focus of current attention [7,12].

The adoption of renewable energy is an important way to relieve the dependence on fossil energy and reduce harmful emissions, especially CO_2 emissions, and the application of alcohol as an alternative oxygenated fuel to petroleum has become a hot topic and the focus of current research [13–26]. Among alcohols, ethanol has been widely researched and applied for its low production cost and nontoxic properties. As an additive to gasoline and diesel, ethanol can effectively reduce fossil fuel consumption and has the latent capacity to improve engine performance. Additionally, harmful emissions, such as NOx, soot, and carbon emissions in exhaust, are also reduced by the addition of ethanol in fuels [20,21,27–31]. Large-scale use of ethanol in engines has shown good feasibility and prospects. On the one hand, by 2020, many countries, such as Brazil, the United States, and the European Union, will have announced the compulsory use of ethanol as an oxygenated additive in gasoline. The Chinese government has also promoted the use of E10 as gasoline fuel in the market. On the other hand, the difficulties of ethanol as an additive to diesel, such as mutual solubility and cetane number, are also being gradually solved, which proves the feasibility of ethanol as an additive to diesel. It can be seen that there will be more room and potential for ethanol applications in the future.

So far, driven by the aim of reducing fossil fuel consumption and carbon emissions, the world's ethanol production is also increasing year by year. From 2009 to 2019, world ethanol production increased from 1.29 MMb/d (million barrels per day) to 1.89 MMb/d, i.e., an increase of 46.51%. In 2019, ethanol production in the United States was 1.03 MMb/d, accounting for 54.50% of global ethanol production, while that in Brazil was 0.54 MMb/d, accounting for 28.57%. When ethanol is produced by fermentation using molasses as a raw material, which contains a high proportion of sucrose, fusel oil can be obtained by distilling the by-product of fermentation broth. In the process of ethanol production, the output of fusel oil is about 1/200 of that of ethanol [32]. Therefore, as a by-product of ethanol production, the output of fusel oil will rise with the increase in ethanol production. According to the above conversion ratio between ethanol and fusel oil, it can be estimated that by the end of 2019, the output of fusel oil all over the world will be 9.43 Mb/d (thousand barrels per day), or about 550 million liters. The United States and Brazil are also the world's top two producers of fusel oil, accounting for 54.36% and 29.63%, respectively. Therefore, it is also of great significance to reduce oil consumption and CO_2 emissions in engines if fusel oil can be used.

The main components of typical fusel are i-propanol, i-butanol, and i-amyl alcohol, as well as a small amount of ethanol and water. In these alcohols, the lower heat value, density, and cetane index of higher alcohols increase with the increase of carbons in their molecules, while the oxygen contents gradually decrease at the same time [33–41]. Aiming at these specific components in fusel, some previous work has been conducted. Pinzi et al. [34] researched the effects of diesel with the additives ethanol and propanol. Additive ethanol and propanol can substitute a certain proportion of diesel, reduce diesel consumption, and improve NOx and soot emissions. Compared with ethanol, propanol showed better improvements in exhaust emissions and noise. Chen et al. [36] researched the influence of a high proportion of n-butanol as a diesel additive combined with exhaust gas re-circulation. Compared with ethanol, butanol is better mixed with diesel. By adding a large proportion of n-butanol, NOx emissions rose while soot decreased. Simultaneous reductions of NOx and soot were realized compared with diesel with the application of exhaust gas recirculation. Javier et al. [41] researched the influence of pentanol. Compared with methanol and ethanol, the fuel characteristics of pentanol are more similar to diesel. Aditive pentanol can improve combustion and BTE. Blends of 25% pentanol and 75% diesel can be used as an alternative to diesel without obvious change.

Compared with ethanol, the fuel characteristics of higher alcohols are closer to diesel and easier to blend with diesel. It has little impact on the performance of the engine while improving combustion and reducing emissions. Predictably, as a mixture of higher alcohols, the application of fusel oil can substitute a certain proportion of traditional fossil fuels and diversify the application of fuels without any modifications to existing SI and CI engines [42]. In some literature on the blending of fusel and diesel fuels [43–45], it has been pointed out that blending fusel with diesel reduced the cylinder pressure and reduced NOx and soot emissions, but the trends in fuel consumption, thermal efficiency, CO, and CO_2 emissions varied in different studies. Both a lower cetane number and a lower heating value had a negative impact on cylinder combustion and the output power of the engine.

As a coproduct of ethanol production, fusel usually contains a small amount of ethanol and water. According to a comparison of energy consumption, removing water from alcohols will consume a lot of energy, especially when the purity is increased to more than 90% [46,47]. Moreover, some studies show that the application of hydrous alcohols in diesel engines can improve performance. The main reason is that the addition of water can decrease the in-cylinder temperature and NOx emissions [20,48,49]. Correspondingly, the water contained in the fusel will reduce the cetane number of the blended fuel and affect combustion. The presence of water can also reduce the viscosity of the blended fuel and increase the wear of the fuel supply system. Considering that removing water from alcohol will consume a lot of energy and that the application of hydrous alcohol can improve the performance of engines and reduce emissions while reducing energy consumption in the fuel production process, the influences of water in fusel oil need to be evaluated to determine whether it is necessary to remove water from the fusel oil when it is used as engine fuel. In previous studies, it was rare to compare hydrous and anhydrous fusel blended with diesel fuel in the same experiment. The results of this comparison can provide a valuable reference for determining whether to remove water from the fusel. Therefore, the influences of diesel blended with hydrous and anhydrous fusel oil were compared by experiment in this research.

In addition, previous studies [20,36,50,51] showed that introducing EGR is an effective method to reduce combustion temperature, which results in a reduction of heat transfer loss and NOx emissions. By further combining EGR with oxygenated fuel, the BTE, NOx, and soot can be improved simultaneously. To improve BTE and reduce emissions, the EGR rate was varied in this experiment.

Therefore, the aim of the current research is to study the influences of hydrous and anhydrous fusel oil on diesel engines. The volume ratios of diesel in blends were set at 80%, while those of hydrous or anhydrous fusel oil were set at 20%. The volume proportion of water in hydrous fusel was 6.5%. The influence of additive hydrous and anhydrous fusel oil was analyzed based on diesel fuel. The influences of water in fusel oil were investigated as well. Moreover, weighted fuel consumption and emissions of the WHSC (World Harmonized Stationary Cycle) for the different test fuels with optimized EGR rates were also investigated.

Compared with previous studies, the effects of hydrous and anhydrous fusel on combustion and emissions were compared and analyzed in this study, and it has the potential to improve both engine economy and emissions with the introduction of EGR. This provides an effective reference for whether to remove moisture before using fusel as a fuel. A valuable reference for the efficient and clean application of fusel oil in compression-ignition internal combustion engines can be afforded.

2. Test Apparatus and Methods

2.1. Test Facility

A six-cylinder compression-ignition internal combustion engine equipped with electronically controlled high-pressure common rail fuel injection system was used as the experimental setup. The engine parameters are shown in Table 1, and the experimental device diagram is shown in Figure 1. The uncertainties of the measuring devices are shown in Table 2.

A pressure sensor (Kistler 6125C, Kistler, Winterthur, Switzerland) was used to measure the cylinder pressure together with a matching charge amplifier and data gathering system. At each measuring site, the pressure data of 100 consecutive cycles was continually recorded with an increase of 0.5 °CA (crank angle degree). To ensure the stability of the engine's operating state, the COV_{IMEP} of the engine did not exceed 2% when saving data. Then, the cylinder pressure data were examined using a single zone heat release model and the assumption that the temperature and air/fuel ratio were constant across the whole cylinder capacity. The Woschni correlation was used to determine the heat transfer coefficient. Previous research has used the heat release rate (HRR) values computed by this model [19,24,25].

Table 1. Engine Parameters.

PARAMETERS	VALUES
Engine type	6 cylinders, 4 valves, water-cooled, Turbocharger with air intercooler
Bore × stroke	110 × 135 mm
Connection rod length	215 mm
Displacement	7.7 L
Compression ratio	17.5:1
Combustion chamber shape	Reentrant
Number of nozzle holes	8
Diameter of nozzle hole	0.153 mm
Included spray angle	147°
Fuel injection system	Common rail
Max torque @ speed	1350 N·m @ 1200–1700 rpm
Rated power @ speed	230 kW @ 2200 rpm

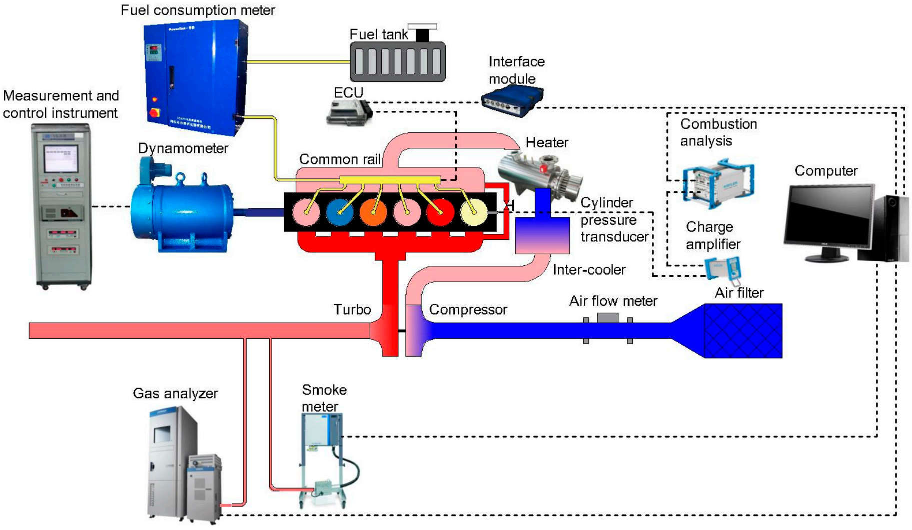


Figure 1. Schematic diagram of experiment setup.

Table 2. Uncertainties of the measurement instruments.

Instrument	Uncertainties	Resolution/Sensitivity
Gaseous analyzer (HORIBA 7100DEGR, Kyoto, Japan)	0.5% full scale	1×10^{-6}
Smoke meter (AVL 415S, Graz, Austria)	0.005 FSN + 3% of measured value	0.001 FSN
In-cylinder pressure sensor (Kistler 6125C, Winterthur, Switzerland)	<±1%	−16 pC/bar
Air flow meter (vortex-shedding flow meter)	<±1%	$0.1\ \text{m}^3/\text{h}$
Fuel flow meter (AVL 733S, AVL, Graz, Austria)	<±1%	0.01 kg/h
Intake pressure (pressure transmitter)	±1 kPa	0.1 kPa
Intake temperature (K-type thermocouple)	±1 °C	0.1 °C

2.2. *Test Fuels*

Three test fuels were used, i.e., diesel, F20WW (the blends of 20% hydrous fusel oil and 80% diesel in volume fraction), and F20NW (the blends of 20% anhydrous fusel oil and 80%

diesel in volume fraction). The volume proportion of water in hydrous fuel was 6.5%. The main properties of diesel, fuel oil, and test fuels are shown in Table 3. The lower heating value, cetane number, and density of fuel oil are closer to those of diesel than those of low-carbon alcohols, and the viscosity of fuel oil is higher, which is beneficial for reducing the wear of engine components such as the oil pump and pistons [52–57]. According to the fuel properties, it can be predicted that when anhydrous fuel oil is added to diesel, the density, lower heating value, and cetane number of the blended fuel decrease, while the latent heat of evaporation and oxygen content increase. Compared with anhydrous fuel oil, the latent heat of evaporation of the hydrous fuel oil-diesel blended fuel is further increased due to the existence of water in fuel oil, which will increase the auto-ignition resistance. As can be seen, it is necessary to investigate the effects of fuel oil and the existence of water in fuel oil on diesel engines. It can also be a more reliable measure of whether it is worth consuming energy to remove water from fuel oil.

Table 3. Main properties of fuels.

	Diesel	Fuel Oil	F20NW	F20WW
Cetane number	51	42	49.2	48.65
Oxygen content (wt.%)	--	18%	3.6%	4.52%
Density (kg/L) at 20 °C	0.834	0.800	0.827	0.830
Lower heating value (MJ/kg)	43.50	35.32	41.86	41.40
Latent heat of evaporation (kJ/kg) at 25 °C	232	874	360.4	380.93
Viscosity (mm^2/s) at 40 °C	3.8	4.162	3.87	3.83
Stoichiometric air-fuel ratio	14.3	11.38	13.72	13.57

2.3. Test Conditions

First, for analysis of the effects of fuel oil and the existence of water in fuel oil in Section 3.1, the three operating points, i.e., 25% load, 100% load of 1144 r/min, and 100% load of 1765 r/min, were selected from the WHSC, and the BMEP of the three operating conditions are 0.51, 2.04, and 2.01 MPa, respectively. The reason for selecting these three operating points is to visually analyze combustion at different speeds and loads, and the operating points in the WHSC are representative of the operating points that vehicles frequently operate at. The single injection strategy (i.e., only the main injection) was used in this experiment. The injection parameters for the three working points are shown in Table 4. The effects of fuel oil and the existence of water in fuel oil can be shown by the analysis of cylinder pressures and the heat release rate curves.

Table 4. Working points and injection parameters.

Working Points	Injection Parameters
1144 r/min, 25% load BMEP = 0.51 MPa	Main injection timing = −5 °CA ATDC Injection pressure = 90 MPa EGR = 0%
1144 r/min, 100% load BMEP = 2.04 MPa	Main injection timing = −7.5 °CA ATDC Injection pressure = 105 MPa EGR = 0%
1765 r/min, 100% load BMEP = 2.01 MPa	Main injection timing = −10.5 °CA ATDC Injection pressure = 169 MPa EGR = 0%

Then, 10%, 25%, 50%, 70%, and 100% loads of 1144 r/min were selected in Section 3.2. The effects of fuel oil and the existence of water in fuel oil on BSFC, BTE, and emissions were investigated at different loads.

In Section 3.3, the EGR rates were varied at 25%, 50%, and 100% loads of 1144 r/min, and the effects of EGR on BSFC, BTE, and emissions were investigated for different test fuels.

Finally, WHSC test cycle was conducted in Section 3.4 using the initial fuel injection MAP (China VI Emissions Standard) of the test engine, and the EGR rates were adjusted as well. The weighted BSFC and emissions, such as NOx, soot, and CO_2, of diesel and the blended fuels with or without EGR optimization were compared. The specific working points of WHSC have been shown in Table 5 [58].

Table 5. The working points for the WHSC cycle.

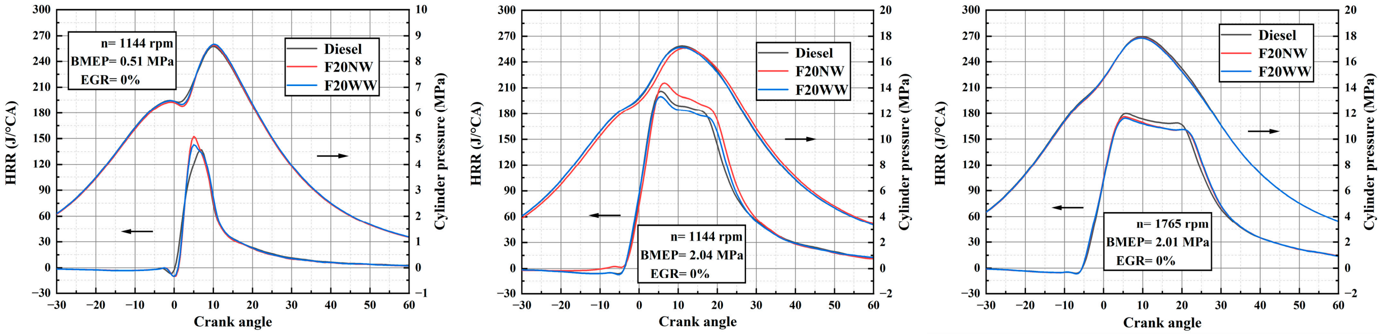
	Speed (rpm)	Load (N·m)	Weight (%)
0	Motoring	24	
1 (cold idle)	650	2	8.5
2	1454	1250	2
3	1454	313	10
4	1454	875	3
5	1144	1250	2
6	988	296	8
7	1299	875	3
8	1299	313	6
9	1454	625	5
10	1765	1230	2
11	1144	625	8
12	1144	313	10
13 (hot idle)	600	2	8.5

The engine's working conditions were stabilized for a short while at each test operating point before the data were collected. The test was repeated for each operating point three times, and the average value was recorded.

3. Results and Discussions

3.1. Effect of Fusel Oil and the Existence of Water in Fusel Oil on Combustion

The effects of fusel oil and the existence of water in fusel oil on combustion are shown in Figure 2. The injection strategies for different fuels remained consistent. The combustion duration is defined as the interval between the crank angle where 10% of the total heat is released (CA10) and the crank angle where 90% of the total heat is released (CA90).



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

Figure 2. Combustion analysis at different operating conditions. (**a**) Combustion under 25% load of 1144 r/min; (**b**) Combustion under 100% load of 1144 r/min; (**c**) Combustion under 100% load of 1765 r/min

Figure 2a shows that under low load and low speed, the ignition delay periods of F20NW and F20WW were slightly longer than those of pure diesel; this is attributed to the lower cetane number and higher latent heat of evaporation of fusel oil. Therefore, after the addition of fusel oil, the ignition delay of the blended fuel was prolonged, the premixed combustion proportion was increased, and the combustion was more concentrated. In addition, because of the high oxygen content in fusel oil, the heat release rate (HRR) at the initial stage of combustion and peak HRR of the blended fuel were higher than those of pure diesel fuel. Since F20WW contains water, which absorbs heat during evaporation, the peak value of HRR for F20WW was slightly lower than that of F20NW.

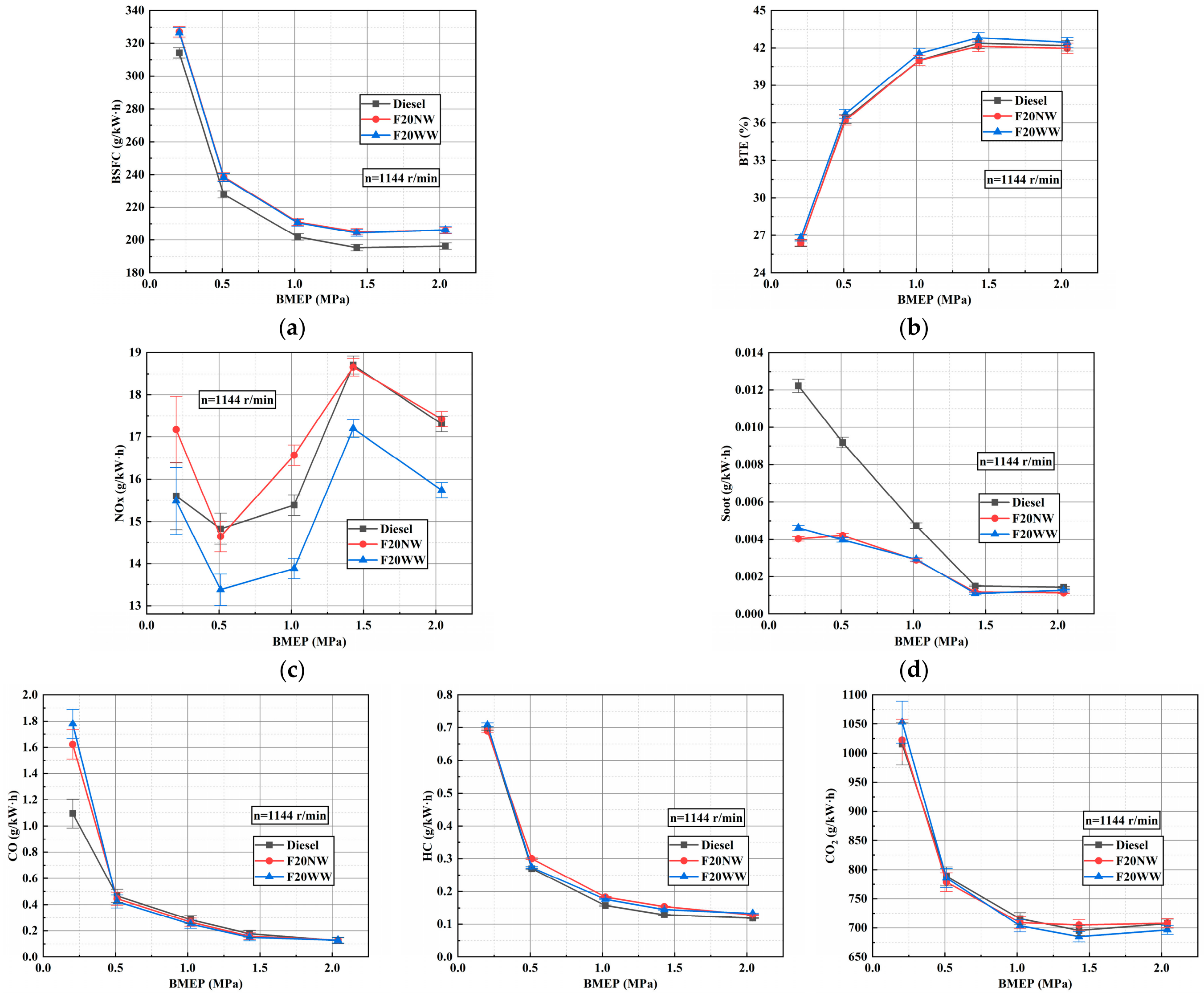
It can be seen from Figure 2b that under high load and low speed, the ignition delay periods of the three fuels were basically the same, mainly due to the higher cylinder temperature, and the cetane number has little effect on in-cylinder combustion, which has been pointed out in the previous study [59]. Therefore, under high load conditions, the latent heat of evaporation and a small change in cetane number had little effect on the ignition delay. In contrast to low-load conditions, the quantity of fuel injected per cycle was large under high load. Because the lower heating value of fusel oil was lower than that of pure diesel, the fuel injection duration of the blended fuel was extended. Therefore, the combustion duration of the blended fuel was prolonged. Due to the high oxygen content and high volatile components (propanol and butanol) in fusel oil, although the combustion duration was prolonged, the peak HRR of F20NW was higher than that of pure diesel. Due to the water content in F20WW, the peak HRR decreased.

It can be seen from Figure 2c that under high load and high speed, the combustion duration of the blended fuel was still prolonged compared with the diesel. At high speeds, the peak HRR of blended fuel was lower than that of pure diesel. This is mainly because at high speed, the reaction time per cycle was shortened, and the longer fuel injection duration became the main influencing factor. But in general, the difference in HRR between the three fuels was small at high loads and speeds. Under these three operating conditions, there was no significant difference in the cylinder pressures of the three fuels. It may be due to the small volume proportion of fusel oil (20%), which would not influence combustion or cylinder pressure largely.

3.2. Effect of the Fusel Oil and the Existence of Water in Fusel Oil at Different BMEP

Figure 3a–g show the BTE, BSFC, and emissions at different BMEPs; the EGR rate was 0%. As can be seen from Figure 3a,b, with the increase in BMEP, BSFC gradually decreased, while BTE showed the opposite trend, which was mainly due to the change in mechanical efficiency under different loads. Since the lower heating value of fusel oil was lower than that of pure diesel, the BSFC of the blended fuel was higher than that of pure diesel. On the one hand, the addition of fusel oil reduced the lower heating value, prolonged the fuel injection duration, and then prolonged the combustion duration, which may lead to a decrease in thermal efficiency. On the other hand, the addition of fusel oil increased the volatility and oxygen content of the blended fuel and improved the mixing of fuel and air, which is beneficial for efficient combustion. Under the combined effects of the above factors, the BTE of F20NW was equivalent to that of pure diesel. For the three fuels, the BTE of F20WW was the highest because the water in the fuel absorbs heat during evaporation, which reduces the temperature in the cylinder and reduces the heat transfer loss to some extent. With the increase in BMEP, the improvement in BTE of F20WW relative to pure diesel gradually decreased from 1.71% to 0.64%. This was because under high load, the temperature in the cylinder was high and the influence of water was decreased.

It can be seen from Figure 3c that with the increase in load, NOx emissions of the three fuels showed a trend of first decreasing, then increasing, and then decreasing again. This was mainly due to the effect of the cylinder temperature and different injection strategies under different loads. There is no obvious rule for the change in NOx emissions for the three fuels with the change of load. Compared with pure diesel, on the one hand, the addition of fusel oil increased the oxygen content, which was conducive to the generation of NOx. On the other hand, it increased the latent heat of vaporization, which reduced the temperature in the cylinder and consequently decreased the generation of NOx. The combined effects of these two aspects made the NOx emissions of F20NW slightly higher than those of pure diesel.



(**a**) (**b**) (**c**) (**d**) (**e**) (**f**) (**g**)

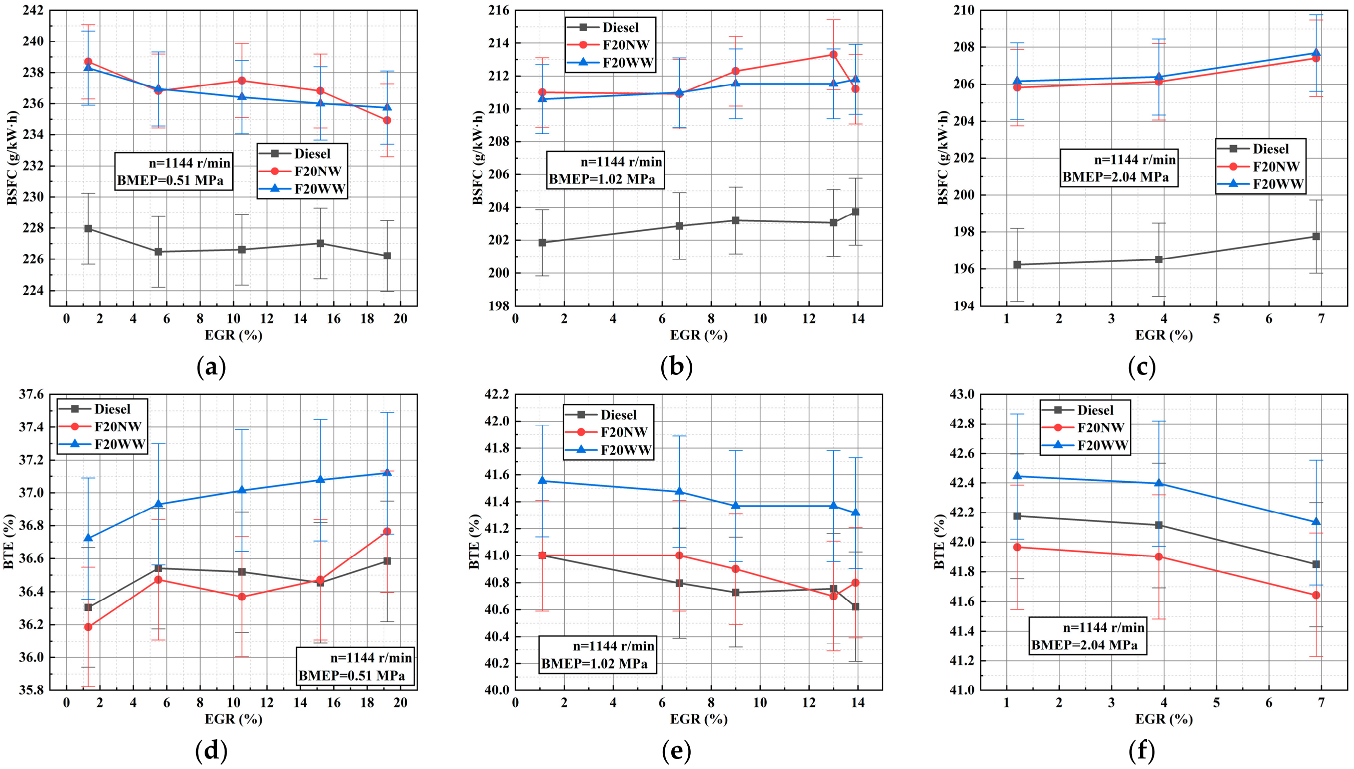
Figure 3. BSFC, BTE, and emissions of the test fuels at different BMEP. (**a**) BSFC at different BMEP; (**b**) BTE at different BMEP; (**c**) NOx emissions at different BMEP; (**d**) Soot emissions at different BMEP; (**e**) CO emissions at different BMEP; (**f**) HC emissions at different BMEP; (**g**) CO_2 emissions at different BMEP.

However, when hydrous fusel was added to diesel, the water had a great impact on the cylinder temperature, so the NOx emissions of F20WW were significantly lower than those of pure diesel. According to Figure 3d–f, with the increase in BMEP, the soot, CO, and HC emissions of the three fuels gradually decreased, mainly due to the increase in cylinder temperature promoting complete combustion. The soot emissions of the blended fuels were significantly lower than those of pure diesel due to the higher oxygen content and better volatility of the blended fuel, which improved the fuel-air mixing and combustion process, reduced soot generation, and promoted soot oxidation. Among them, the oxygen content was the main factor. Other emissions of HC and CO were similar to those of soot. Only under low loads were the CO emissions of the blended fuels higher than those of pure diesel. This was mainly because the cylinder temperature was low under 10% load, and the addition of fusel oil and water further reduced the cylinder temperature so that the CO emissions were not completely oxidized. For CO_2 emissions, the CO_2 emissions of F20NW were equivalent to those of pure diesel. The CO_2 emissions of F20WW were lower than those of pure diesel except for the operating condition of 10% load, with a maximum decrease of 1.67%. In general, the error lines of the CO_2 emissions of the three fuels showed that there was little difference between the CO_2 emissions of the three fuels.

3.3. Effect of the EGR on Fuel Consumption and Emissions of Different Test Fuels at Different BMEP

Figure 4a–l show the BTE, BSFC, and emissions under different EGR rates when BMEP was 0.51, 1.02, and 2.04 MPa, respectively. In the experiment, while adjusting different EGR rates under a certain operating condition, the engine speed and output torque remained unchanged. It can be seen that no matter the BSFC, BTE, or emissions, the trends of different fuels with EGR rates were consistent.

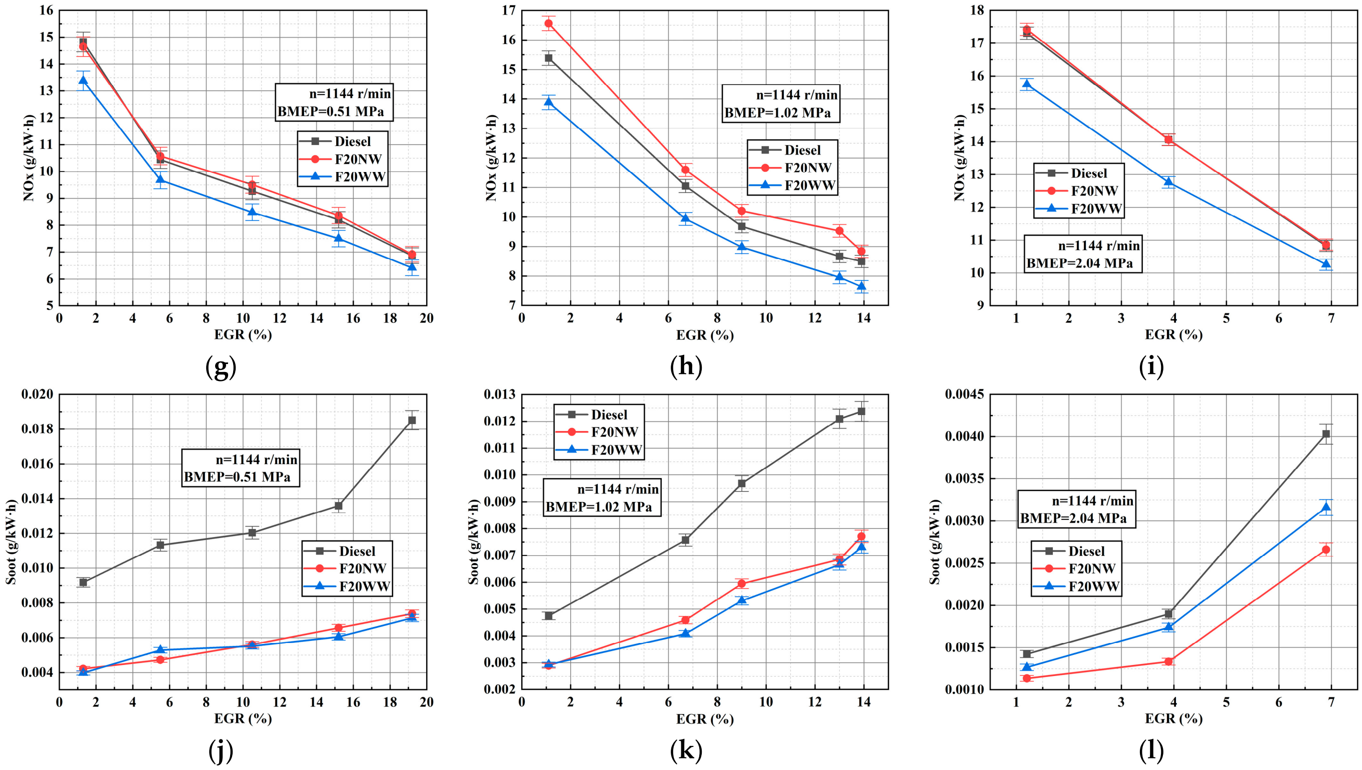
From Figure 4a–f, it can be seen that with the increase in the EGR rate, the change ranges of BSFC and BTE were all about 1%, which was relatively small. This was mainly because, with the increase in the EGR rate, although the pumping loss and heat transfer loss were reduced, the combustion efficiency might be reduced. Rakopoulos et al. [60] noted in the study pertaining to the exergy perspective that when the EGR rate rose, the heat transfer and working transfer exergy terms somewhat reduced, but the equivalent terms for irreversibility and net exhaust transfer (flow out) slightly increased. Therefore, the BSFC and BTE were basically unchanged.



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

(**d**) (**e**) (**f**)

Figure 4. *Cont.*



(g) (h) (i)

(j) (k) (l)

Figure 4. BSFC, BTE and emissions of the test fuels under different EGR rates. (**a**) BSFC at 0.51 MPa; (**b**) BSFC at 1.02 MPa; (**c**) BSFC at 2.04 MPa; (**d**) BTE at 0.51 MPa; (**e**) BTE at 1.02 MPa; (**f**) BTE at 2.04 MPa; (**g**) NOx emissions at 0.51 MPa; (**h**) NOx emissions at 1.02 MPa; (**i**) NOx emissions at 2.04 MPa; (**j**) Soot emissions at 0.51 MPa; (**k**) Soot emissions at 1.02 MPa; (**l**) Soot emissions at 2.04 MPa.

Under different EGR rates, the BTE of F20WW was always higher than that of pure diesel and F20NW. In terms of emissions, with the increase in the EGR rate, the NOx emissions of the three fuels gradually decreased and the soot emissions gradually increased under all operating conditions. It can be seen that the variation range of BSFC, BTE, and NOx emissions of the three fuels with EGR rates was basically the same, while the variation range of soot emissions of the blended fuel was smaller than that of pure diesel, mainly because the oxygen content of the blended fuel was higher and the EGR rate had less influence on the soot emissions of both fuels with fusel addition. It can also be seen from the test results that the BTE of F20WW was improved by increasing the EGR rate, and compared with pure diesel, the NOx and soot emissions were reduced at the same time.

3.4. Experiment of WHSC Test Cycle

The tests on the three fuels over the WHSC cycle were performed. The WHSC test cycle differs from the European Stationary Cycle (ESC). Compared with ESC, the WHSC test cycle focuses on medium and low speeds, and medium and small loads, which is more consistent with urban operating conditions.

All changes in percentages in Figure 5 were compared with those of pure diesel. It can be seen that compared with pure diesel, the equivalent BSFC of F20NW was increased by 1.07%, and the equivalent BSFC after EGR optimization was slightly reduced, but it was still 0.36% higher than that of pure diesel. The equivalent BSFC of F20WW was reduced by 0.95% compared with pure diesel. After EGR optimization, the equivalent BSFC was further reduced by 1.77% compared with pure diesel. In terms of CO_2 emissions, the CO_2 emissions of F20NW were higher than those of pure diesel. The CO_2 emissions of F20WW were reduced by 0.65% compared with those of pure diesel. After adjusting the EGR rate, the CO_2 emissions of F20WW were further reduced due to the improvement of BTE. In terms of NOx and soot emissions, the NOx emissions of F20NW were equivalent to those of pure diesel, and the NOx emissions of F20WW were reduced by 8.20% compared with those of pure diesel. After EGR optimization, the NOx emissions of blended fuels were significantly reduced, and the NOx emissions of F20WW were 37.49% lower than those of pure diesel. The soot emissions of blended fuels were significantly lower than those of pure diesel, and the soot emissions increased slightly as EGR was introduced.

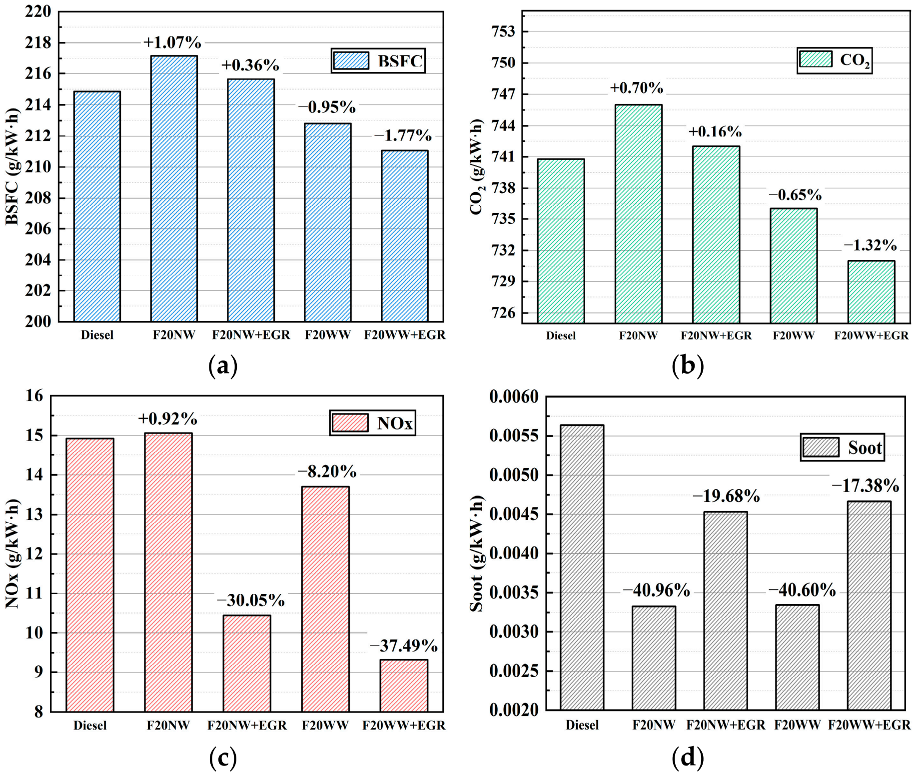


Figure 5. Weighted equivalent BSFC and emissions. (**a**) Weighted equivalent BSFC versus pure diesel; (**b**) Weighted CO_2 emissions versus pure diesel; (**c**) Weighted NOx emissions versus pure diesel; (**d**) Weighted soot emissions versus pure diesel.

In conclusion, with the application of F20NW, the equivalent fuel consumption increased by 0.36%. NOx and soot emissions were reduced by 30.05% and 19.08%, respectively, with the optimization of EGR compared with pure diesel. There was no significant difference in CO_2 emissions between pure diesel and F20NW. And with the application of F20WW, the equivalent BSFC was reduced by 1.77%. NOx, soot, and CO_2 emissions were reduced by 37.49%, 17.38%, and 1.32%, respectively, with the optimization of EGR compared with pure diesel. That means improved thermal efficiency and emissions can be achieved by adding hydrous fusel oil and combustion optimization. However, it should be noted that the impact of adding water to fuels on engine reliability and not removing water from fuel preparation costs should also be considered in future applications.

4. Conclusions

Due to the increasing production of fusels as by-products in the preparation of ethanol, the application of fusel is an effective way to lessen the consumption of fossil fuels. The effects of fusel oil and the existence of water in fusel oil on combustion, performance, and emissions at different loads were investigated on a heavy-duty diesel engine. EGR optimization was also carried out for the test cycle of WHSC. The main conclusions are as follows:

(1) The addition of fusel oil prolonged the ignition delay period, increased the peak value of heat release rates at low speed, and prolonged the combustion duration under high load.

(2) Under different engine loads, the addition of hydrous fusel improved the break thermal efficiency and reduced NOx and soot emissions, and it could be further improved in combination with EGR optimization.

(3) In terms of the WHSC test cycle, with the application of F20NW, the equivalent fuel consumption was increased by 0.36%. NOx and soot emissions were reduced by 30.05% and 19.08%, respectively, with the optimization of EGR compared with pure diesel. There was no significant difference in CO_2 emissions between pure diesel and F20NW. And with the application of F20WW, the equivalent fuel consumption was reduced by 1.77%. NOx, soot, and CO_2 emissions were reduced by 37.49%, 17.38%, and 1.32%, respectively, with the optimization of EGR compared with pure diesel.

Therefore, reasonably adding 20% hydrous fusel oil combined with combustion optimization is an effective way to improve the thermal efficiency and emissions of the engine. This provides an effective reference for whether to remove moisture before using fusel as a fuel. A valuable reference for the efficient and clean application of fusel oil in compression-ignition internal combustion engines can be afforded.

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