A COMPARISON STUDY ON THE COMBUSTION AND PARTICULATE EMISSIONS OF 2,5-DIMETHYLFURAN/DIESEL AND ETHANOL/DIESEL IN A DIESEL ENGINE

by

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Compared with pure Diesel fuel, Diesel engine fueled with 2,5-dimethylfuran/diesel prolonged the ignition delay, shortened the combustion duration, increased the engine efficiency and decreased the number of accumulation mode particles (50 nm < Dp < 1000 nm), but increased the nucleation mode (Dp < 50 nm) number. The mean diameter of particles was declined with 2,5-dimethylfuran addition due to the increase of small particles number. Ethanol/diesel was more effective in extending the ignition delay, shortening the combustion duration, enhancing the engine efficiency and reducing the number of large size particles, but produced more small size particles compared with 2,5-dimethylfuran/diesel.

Key words: *Diesel engine, 2,5-dimethylfuran, ethanol, combustion, particulate matter emissions*

Introduction

Ethanol has been blended with gasoline as transportation fuels in many countries because of the rapid consumption of fossil fuels and environmental pollution, but the lower heating value and the solubility in water hinder its practical utilization. Recently, 2,5-dimethylfuran (DMF) as the alternative or additive fuel in internal combustion engines is gaining public and scientific attention [1]. As a biomass-based renewable fuel, DMF has been made with significant achievements in its production techniques. Meanwhile, DMF has many attractive properties compared with ethanol, such as higher energy density, higher boiling point, better miscibility and stable storage.

Particulate matter (PM) emissions from Diesel engine have become a growing concern as it may adversely affect human health and the environment [2, 3], but the PM reduction has long been a severe challenge. As an oxygenated fuel, DMF has good potential in reducing PM emissions from Diesel engine. Zhang *et al*. [4] reported that DMF/diesel blend could reduce soot further compared to gasoline/diesel, due to its longer ignition delay and higher oxygen content. Liu *et al*. [5] investigated the effects of DMF addition on combustion and emissions in a Diesel engine compared with diesel blended with the mixture of cetane and iso-cetane, *n*-heptane, and the mixture of DMF and 2-ethylhexyl nitrate. They found DMF/diesel blends has the lowest soot emission among the tested fuels. Chen *et al*. [6] reported that DMF/diesel outper-

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form *n*-butanol/diesel and gasoline/diesel in reducing soot emissions due to the joint effects of its prolonged ignition delay and atomic oxygen. Recently, Liu *et al*. [7] studied the combustion and emission characteristics of a direct injection compression ignition engine fueled with diesel, DMF/diesel and iso-octane/diesel, respectively. The results showed that the lowest soot emissions was obtained by using DMF/diesel blends.

However, the known benefits of low soot emissions after the DMF addition into diesel are mainly related to mass basis, but little is known about the number concentration and size of exhaust particles. Thus, the objective of this work is to explore the effects of DMF/diesel blends on the number and size of exhaust particles. In addition, ethanol/diesel was studied widely, but the comparison between the combustion and emissions characteristics of DMF/diesel and ethanol/diesel are limited. Thus, the combustion and PM emissions between DMF/diesel and ethanol/diesel were especially compared in order to decide on the suitability of DMF as a diesel fuel additive.

Experimental set-up

The experimental engine employed for the present study was a four-cylinder, four-stroke, turbocharged, water-cooled, Euro-4, Diesel engine equipped with a common rail fuel injection system. A brief description, summarizing the essential features of the experimental apparatus will be given here. The specifications are listed in tab. 1 and the experimental layout is shown in fig. 1. The engine was controlled by an eddy current dynamometer, which adjusted the torque output and kept the engine speed abidingly. High pressure cooled exhaust gas recirculation (EGR) system was employed, and EGR rate can be effectively adjusted by the combined control of air-throttle valve and EGR valve. An electronic control unit was employed to control and monitor the injection timing, injection pressure and injection quantity. A fuel consumption meter (FCD-M) with a gravity scale was used to measure the fuel-flow rate. The cylinder pressure traces were detected by a Kistler 6025C pressure sensor installed in the combustion chamber. The pressure signals were conveyed to a charge amplifier (Kistler 5108A1003) and then to a CB-466 burning analyzer. The burning pressures were recorded for 100 cycles at an interval of 0.25 crank angle (CA). The intake-air temperature and pressure were controlled by an air conditioning system and a supernumerary compressor, respectively.

Table 1. Engine specifications

Type of engine	Four-cylinder 4-stroke
Bore, [mm]	96
Stroke, [mm]	103
Compression ratio	17.5
Displacement, [cm^3]	2982
Rated power, [kW]	85
Rated speed, [rpm]	3200
Type of ignition	Compression ignition
Method of starting	Electric start

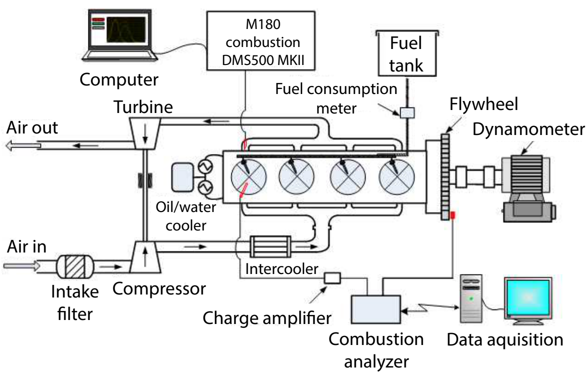

Figure 1. Research engine experimental layout

The PM was sampled from the first exhaust manifold of the engine and measured by a DMS500 MKII fast particulate spectrometer (Cambustion Ltd.) [8]. DMS500 has a fully-integrated two-stage dilution system and classified the particles by the electrical mobility diameter (Dp) from 5-1000 nm in 22 classes. The primary dilution was set to be five throughout the engine operation, and change the secondary dilution factor according to the user interface. A detailed description of the working principle can be found elsewhere [9]. The PM emissions were measured in terms of particle size and mass distribution (PSD and PMD), nucleation mode (NM, Dp < 50 nm) and the accumulation mode (AM, 50 nm < Dp < 1000 nm) particle concentration (NMC and AMC), total particle number and mass concentration (total N and total M) as well as the geometric mean diameter (GMD).

Conventional diesel, DMF and ethanol were used here, with properties shown in tab. 2. In this study, pure diesel (D0) as the based fuel, diesel blended with different mass fractions of DMF (10% and 30%), referred as to D10 and D30, respectively, for evaluation. Diesel blended with 30% mass fraction of ethanol (E30) was also prepared for comparison.

Table 2. Properties of the diesel, DMF and ethanol fuel [10, 11]

Parameters	Diesel	DMF	Ethanol
Chemical formula	C_{12}-C_{25}	C_6H_8O	C_2H_5OH
Cetane number	52.1	9	11
Boiling point, [°C]	180-360	92	79
Lower heating value, [MJkg^{-1}]	42.5	33.7	26.83
Heat of evaporation, [kJkg^{-1}]	250-290	333	1162.6
Auto-ignition temperature, [°C]	180-220	286	385
Stoichiometric A/F ratio	14.3	10.79	9
Density at 20 °C [kgcm^{-3}]	826	889.7	790.9
Carbon content, [mass %]	86.7	75.0	52.1
Hydrogen content, [mass %]	12.7	8.4	13.1
Oxygen content, [mass %]	–	16.6	34.7
C/H	6.8	9.0	4.0

The engine speed was kept at a constant speed of 1800 ± 5 rpm, the injection timing was fixed at 7.5 °CA bTDC. Engine load was changed from 10-90% at a 20% increment equal to 0.13, 0.38, 0.63, 0.88, and 1.13 MPa brake mean effective pressure (BMEP), respectively. The combustion characteristics and PM emissions of different fuels with EGR ratios from 0-25% under the 0.38 MPa engine load were also detected. Different engine conditions corresponding to different injected diesel fuel mass. For DMF/diesel and ethanol/diesel blends, the cyclic fuel mass should be recalculated according to their low heating values (see tab. 2), *i. e.*, fuels with smaller low heating values should increase the injection quantity to ensure the same energy input. To ensure the reproducibility and reliability of all measured data, the engine was first heated and maintained at steady-state for several minutes at each test condition, while keeping the coolant at 85 ± 1 °C, the lubricating oil at 87 ± 2 °C, and the intake air temperature precisely at 15 ± 0.5 °C. Then cylinder pressure, emissions and control parameters were recorded for off-line analysis. The uncertainties of the main measurements were summarized in tab. 3.

Table 3. Uncertainties of the acquired quantities

Measurement	Uncertainty [%]
Torque	±1.0
Fuel-flow meter	±1.0
Air-flow meter	±0.5
In-cylinder pressure	±0.1
EGR	±0.5
BSFC	±1.93
BTE	±1.72
PM	±0.1

Results and discussion

Combustion characteristics

The combustion characteristics of DMF/diesel blends under different engine loads have been investigated in our previous study [10]. In this work, fig. 2 shows the ignition delay (defined as the difference in CA position between fuel injection timing and 10% of total heat is released) and the combustion duration (defined as the difference in CA position between 10% and 90% of the total heat release) *vs*. EGR rate for the test fuels. Clearly, the ignition delay is prolonged with the increasing EGR rate for every tested fuel, because the oxygen content of in-cylinder decreased and the specific heat capacity of the intake gas mixture increased with the decreasing EGR rate. The ignition delay is prolonged by the DMF addition due to the lower cetane number, higher auto-burning temperature and higher latent heat of DMF. As for E30, because of its higher auto-burning temperature and latent heat, it has longer ignition delay time than D30. Meanwhile, with the increase of EGR ratio, the ignition delays are more obviously different between D30 and E30, indicating with high EGR dilution, the fuel properties would more significantly affect the ignition and combustion.

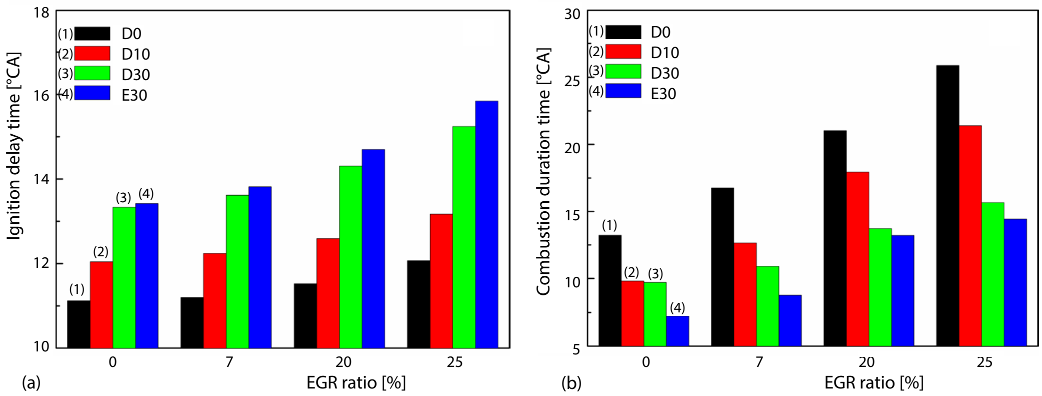

Figure 2. Effects of DMF or ethanol addition on: (a) ignition delay and (b) combustion duration under different EGR rates for the test fuels at 0.38 MPa BMEP

The combustion duration changes in an opposite trend compared to ignition delay, fig. 2(b), because an extended ignition delay can promote the air/fuel mixing process and thus accelerate the premixed burning. In addition, the shorter combustion duration of E30 than D30 can also be attributed to the lower cetane number, higher volatility and higher oxygen acontent.

Figure 3 shows the in-cylinder pressure and heat release rate (HRR) of the tested fuels at EGR ratios of 0% and 20% under the 0.38 MPa BMEP. As reported in [10], with the increasing mass fraction of DMF, the peak in-cylinder pressure declined at low engine load (10% full load), but increased at high engine load (70% full load). However, in this work, D10 has higher peak in-cylinder pressure than D0 at both operating conditions in fig. 3, while the peak in-cylinder pressure for D30 decreased slightly, fig. 3. Moreover, E30 shows higher peak in-cylinder pressures than D30.

These combustion differences can be mainly attributed to the differences in the cetane number and oxygenation of fuels [5]. As for D10, the prolonged ignition delay, fig. 2, and higher oxygen contents can enhance the premixed combustion and accelerate the burning rate (higher HRR) compared with D0, which raise the in-cylinder pressure. However, the entire combustion pressure and temperature are low at the engine load of 0.38 MPa, while more DMF addition

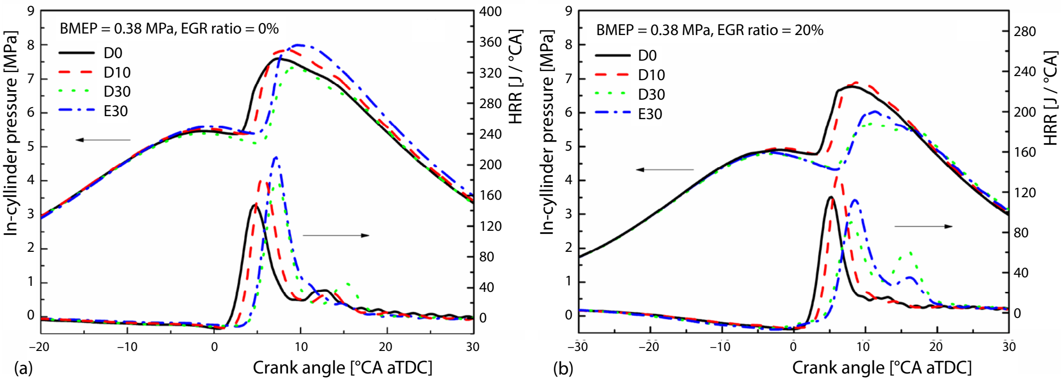

Figure 3. In-cylinder pressure and HRR of test fuels under engine load of 0.38 MPa BMEP

results in more heat absorption for fuel evaporation. In addition, the D30 is burned further away from the TDC in the expansion stroke. Thus, D30 shows an apparently stronger diffusion phase and thereby a lower peak cylinder pressure than D0 and D10. Compared to D30, E30 has longer ignition delay, higher oxygen content, higher volatility and lower viscosity, which contributed to the stronger premixed combustion and higher peak in-cylinder pressure.

At 1.13 MPa BMEP, shown in fig. 4, it is observed that the combustion starting points of D10 and D30 are also delayed compared to D0, while with the increasing mass fraction of DMF, the peak of HRR and in-cylinder pressure increased. As stated before, the combustion start points of blend fuels are delayed on account of the lower cetane number, the higher latent heat of vaporization and the higher autoignition temperature of DMF with respect to pure diesel. The delayed ignition time of blends lead to the premixed combustion proportion increased, which resulted in higher values of the peak HRR and the peak in-cylinder combustion pressure. The E30 has the latest combustion start point and the most abundant oxygen content, which could explain why it produce the highest peak HRR and in-cylinder combustion pressure.

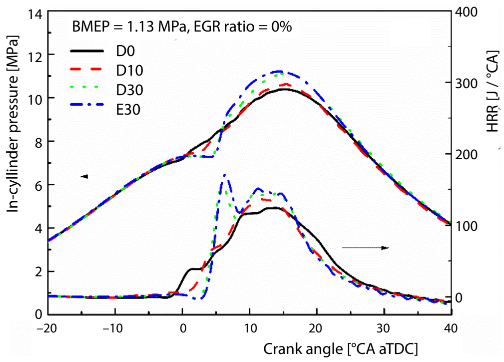

Figure 4. In-cylinder pressure and HRR of test fuels under engine load of 1.13 MPa BMEP

Figure 5 displays the variations of brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) *vs*. the EGR ratio with both fuels. The BSFC of D10, D30, and E30 are slightly higher than that of D0. In comparison with D0, the BSFC are increased by a mean of 1.52% for D10, 5.80% for D30, and 10.32% for E30. This is mainly due to the lower energy content of DMF (33.7 MJ/kg) and ethanol (26.8 MJ/kg) as compared to diesel (42.5 MJ/kg). Meanwhile, DMF addition produce slightly higher BTE values compared to D0, and the thermal efficiencies are further improved by adding ethanol into diesel, mean increasements in BTE of 0.59% for D10, 0.78% for D30 and 1.95% for E30 are observed, which can be explained that the addition of DMF or ethanol can provide additional fuel lubricity, reduce fuel viscosity, improve atomization, and provide more oxygen contents for improving the in-cylinder combustion process in converting fuel chemical energy into useful engine work.

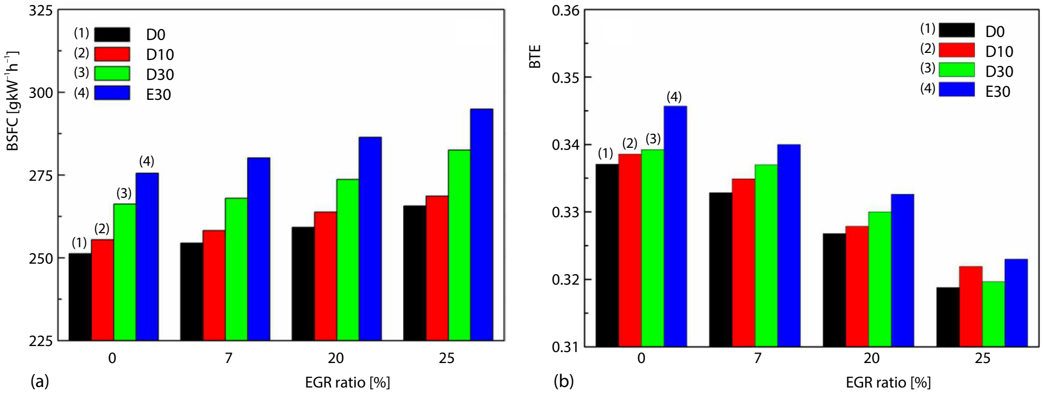

Figure 5. The BSFC and BTE *vs.* EGR rate of different fuels under engine load of 0.38 MPa BMEP

The PM emission characteristics

Figures 6 and 7 illustrate the PM emission characteristics of pure diesel under tested conditions. The PSD curves are bimodal-shaped under all engine conditions and dominated by the NM particles. As shown in fig. 6, the particle number concentration (dN/dlogDp) and particle mass concentration (dM/dlogDp) of the exhaust particles in the whole size range tends to be slightly higher with the engine load increases, In addition, the PSD curves shifted to larger diameters, indicating the proportion of larger particles and thus the total mass concentration increased with the elevation of engine load, fig. 6(b), but the GMD are changed unobviously.

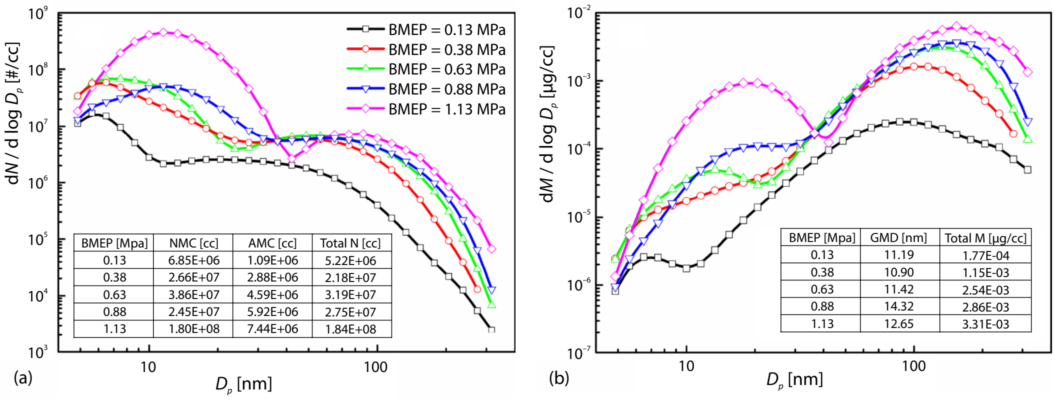

Figure 6. The PM emission characteristics of pure diesel at different engine loads with no EGR
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Under low engine loading, the air/fuel blends are lean and generate relatively low combustion temperatures, which result in the formation of less carbonaceous particles. With the increase of engine load, a larger amount of fuel burned in the diffusion mode, which induce and accelerate the soot nucleation, Moreover, Tsolakis [12] thought that the increased particle formation at high engine load is attributed to the decline of soot oxidation at low oxygen content.

It is obviously observed that, as the EGR ratio increases, the number concentration of large particles increases gradually, fig. 7. It is well-known that the NO_x emissions can be reduced by using the EGR technique, but it is favorable for soot formation and inhibiting soot oxidation. In addition, higher amount of soot emission can also due to EGR induced creates an environment that is more likely to promote the process of particles' coagulation, accumulation, condensation of volatile fractions on the particles. However, it is worth to notice that the number of smaller-size particles (<12 nm) decreases slightly with higher EGR. This is because the increased AM particles act as *sponges* for condensation or adsorption of volatile materials, and thereby suppress the soot nucleation [13]. In addition, the use of high level EGR promote the coagulation among small size particles [14], so the small size particles decreases.

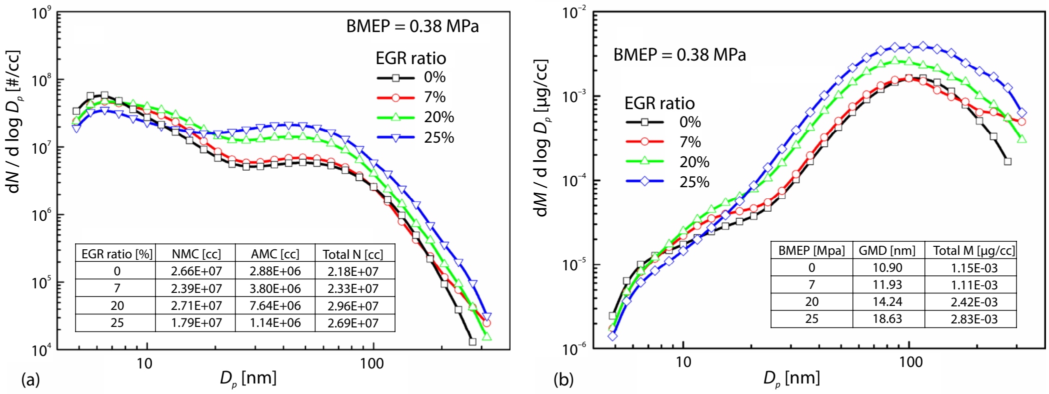


Figure 7. Effects of EGR on PM emissions of pure diesel at engine load of 0.38 MPa BMEP

Figures 8-10 show the effects of DMF and ethanol addition on the PM emission characteristics. The number of large size particles decreases after the addition of DMF or ethanol compared with pure diesel, but the number of smaller size particles increases. Table 4 displays the reduction percentage in PM emissions when engine fueled with D10, D30, and E30 compared to D0 under those test conditions. It can be see that DMF or ethanol addition can decrease the AMC, GMD, and total M, but increase the NMC and total N.

Table 4. The reduction ratio [%] in PM emissions

Fuel	NMC	AMC	Total N	GMD	Total M
D10	–5.02	45.82	1.08	10.85	30.48
D30	–60.55	77.91	–51.47	15.83	58.26
E30	–170.18	91.69	–166.18	33.79	84.77

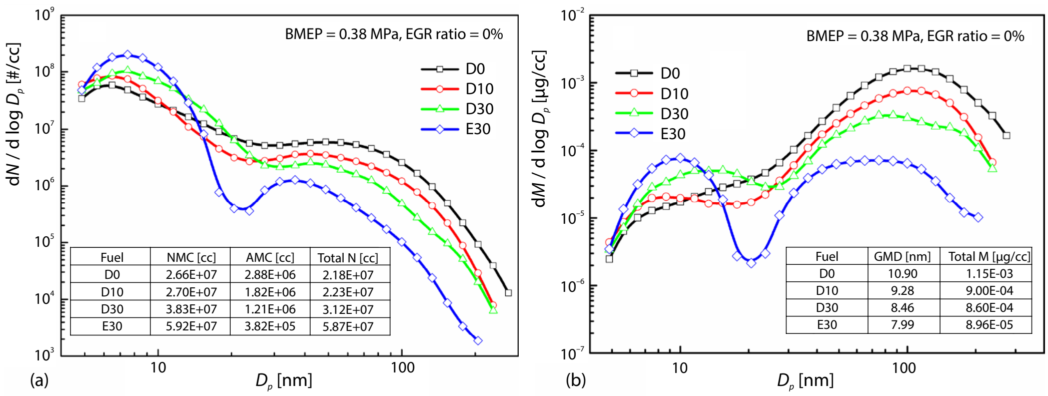
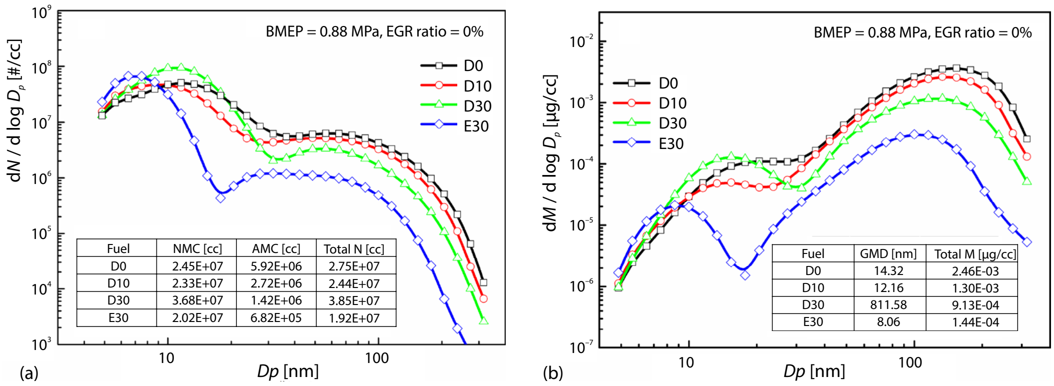


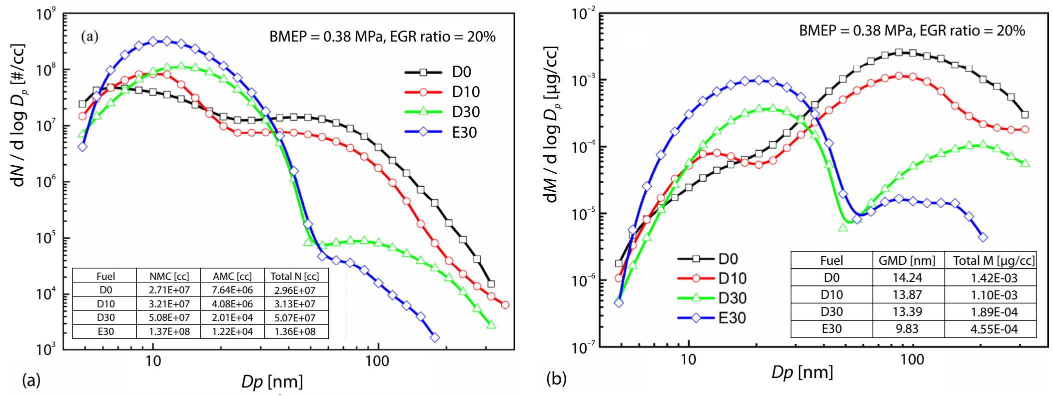
Figure 8. Effects of DMF or ethanol addition on the PM emission characteristics at engine load of 0.38 MPa BMEP



Fuel	NMC [cc]	AMC [cc]	Total N [cc]
D0	2.45E+07	5.92E+06	2.75E+07
D10	2.33E+07	2.72E+06	2.44E+07
D30	3.68E+07	1.42E+06	3.85E+07
E30	2.02E+07	6.82E+05	1.92E+07

Fuel	GMD [nm]	Total M [µg/cc]
D0	14.32	2.46E-03
D10	12.16	1.30E-03
D30	811.58	9.13E-04
E30	8.06	1.44E-04

Figure 9. Effects of DMF and ethanol addition on PM emission characteristics at engine load of 0.88 MPa BMEP



Fuel	NMC [cc]	AMC [cc]	Total N [cc]
D0	2.71E+07	7.64E+06	2.96E+07
D10	3.21E+07	4.08E+06	3.13E+07
D30	5.08E+07	2.01E+04	5.07E+07
E30	1.37E+08	1.22E+04	1.36E+08

Fuel	GMD [nm]	Total M [µg/cc]
D0	14.24	1.42E-03
D10	13.87	1.10E-03
D30	13.39	1.89E-04
E30	9.83	4.55E-04

Figure 10. Effect of DMF and ethanol addition on PM emissions at engine load of 0.38 MPa BMEP with 20% EGR ratio

Several factors may lead to the decrease of soot precursors and the subsequent soot emission after the addition of DMF or ethanol [15]. First, the DMF/diesel or ethanol/diesel blends, with extended ignition delay, higher volatility and larger oxygen content compared to pure diesel can improve the air/fuel mixing process, and then promote combustion and raise the temperature during diffusion combustion stage, which can reduce soot formation and promote soot oxidation at the late expansion and exhaust stages. Second, aromatic ring growth and soot particle inception are inhibited by the abundant radicals (primarily OH) produced after the addition of oxygenated fuel into diesel. Third, the carbon content in the blended fuels declines with the increase of oxygen content, which would reduce the concentrations of C-C bonds (source of soot formation) in the blended fuel. Meanwhile, high concentrations of radicals after the oxygenate addition promote the carbon oxidation to form CO and CO_2, which reduces carbon availability for formation of soot precursor. Moreover, aromatics and sulfur content promotes the soot formation and PM emissions. Thus, the addition of DMF and ethanol, which are aromatics-free and sulfur-free, would dilute the contents of aromatics and sulfur in the blended fuels and thus reduce soot and PM emissions.

With the increase of DMF or ethanol into the blended fuel would reduce soot nucleation, but more small-size particles produced as stated above. The reasons for this trend may be attributed to three causes: The soot formation suppressed by DMF or ethanol addition also slows down the coagulation and aggregation of soot particles to form larger particles, increasing the formation of smaller-size particles and decreasing GMD; more mass of the blended fuels with DMF or ethanol addition will be consumed due to the lower heat value than pure diesel, which also promotes particles formation; the reduction of large-size particles relieves the absorption of volatile or semi-volatile substances and promotes the formation of primary particles.

The observations of E30 under those operating conditions can further increase the small size particles number and decrease the large size particles number compared with D30. As for more small size particles are formed by E30, even increase the total particle mass emission, fig. 10(b), compared to D30, this because of the heat value of ethanol is lower than DMF, more E30 mass consumed, fig. 5.

Compared with D30, E30 can further reduces the number of larger-size particles and increases the number of smaller-size particles. Because of longer ignition delay, higher oxygen content, lower carbon content and higher volatility than D30. Moreover, the cyclic structure of DMF makes for the formation of 1,3-cyclopentadiene, which promotes the formation of polycyclic aromatic hydrocarbon and soot [16]. The E30 produces more small particles compared with D30, which can be explained from three aspects: the higher fuel viscosity of E30 favoring the small particles formation [17]; the formation of soluble organics is promoted by the rise of oxygen content in the fuel [18], resulting more fine particles produced [8]; higher mass of ethanol/diesel is consumed and more small-size particles are produced.

In comparison between figs. 8 and 9, the reduction in total M is more evident at higher engine load. At low engine load, lower cetane number, higher latent heat of vaporization and higher auto-burning temperature of DMF or ethanol greatly influence the combustion and PM emissions due to the lower in-cylinder pressure and temperature. At high engine load, however, these properties weaken the impacts on the combustion and emissions due to the higher in-cylinder pressure and temperature, while other fuel properties (*e. g.* oxygen content and volatility) enhance such impacts, so the total M declines more obviously at the higher engine load after the addition of DMF or ethanol.

As shown in fig. 10(a), D10, D30, and E30 both reduce the smaller particles (<7 nm) number, which are different from the results under other operating conditions (at engine load of 0.38 and 0.88 MPa without EGR ratio, *i. e.* figs. 8 and 9). The possible reasons are that the processes of coagulation and accumulation between small-size particles, condensation of volatile fractions on the particles and surface growth were enhanced by importing the exhaust gas.

Conclusions

- Under 0.38MPa BMEP engine load, compared to pure diesel, small amount of DMF addition (D10) can raise the in-cylinder pressure, but D30 lower peak in-cylinder pressure. At 1.13 MPa BMEP, the peak of HRR and in-cylinder pressure increased with the increasing mass fraction of DMF. While E30 increases the proportion of premixed combustion and improves the diffusion combustion compared to D30.
- In comparison with D0, the BSFC are increased by a mean of 1.52% for D10, 5.80% for D30, and 10.32% for E30, and mean increasements in BTE of 0.59% for D10, 0.78% for D30, and 1.95% for E30, are observed at engine load of 0.38MPa with EGR ratios form 0-25%. Increase in fuel consumption is attributed to lower heating value of DMF and ethanol and increase in engine efficiency due to improvement of the in-cylinder combustion process. The E30 is more effective in enhancing the engine efficiency compared to D30, but shows higher BSFC values.

- With the increase of engine load, particulate number and mass concentrations are increased due to a larger amount of fuel burned in the diffusion mode and the decline of soot oxidation at low oxygen content, while the GMD is changed unobviously. In addition, particulate mass concentration, GMD and the number concentration of larger size particles are increased with the increase of EGR ratio, but the number of smaller-size particles (<12 nm) decreases slightly.
- Compared to D0, D10, D30, and E30 produce average increases in NMC of about 5.02%, 60.55% and 170.18%, respectively, mean increasements in total N of –1.08% for D10, 51.47% for D30, and 166.18% for E30 are observed. In addition, D10, D30, and E30 produce mean reductions in AMC of 45.82%, 77.91%, and 91.69%, in GMD of 10.85%, 15.83%, and 33.79%, in total M of 30.48%, 58.26%, and 84.77% compared to D0, respectively. Compared to D30, E30 produce lower AM particles number, GMD and total particles mass, due to its longer ignition delay, higher oxygen content and the cyclic structure of DMF, but it produce more NM particles number and total particles number, due to it has higher fuel viscosity, promote more soluble organics formation and consume more fuel mass during combustion.

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Nomenclature

AM – accumulation mode
AMC – accumulation mode concentration
BMEP – brake mean effective pressure
BSFC – brake specific fuel consumption
BTE – brake thermal efficiency
CA – crank angle
Dp – electrical mobility diameter
DMF – 2,5-dimethylfuran
EGR – exhaust gas recirculation
GMD – geometric mean diameter
HRR – heat release rate
NM – nucleation mode
NMC – nucleation mode concentration
PM – particulate matter
PMD – particle mass distribution
PSD – particle size distribution
total M – total particle mass concentration
total N – total particle number concentration

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