AN EXPERIMENTAL STUDY OF THE COMBUSITION AND EMISSION PERFORMANCES OF 2,5-DIMETHYLFURAN DIESEL BLENDS ON A DIESEL ENGINE

by

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Experiments were carried out in a direct injection compression ignition engine fueled with diesel-dimethylfuran blends. The combustion and emission performances of diesel-dimethylfuran blends were investigated under various loads ranging from 0.13 to 1.13 MPa brake mean effective pressure, and a constant speed of 1800 rpm. Results indicate that diesel-dimethylfuran blends have different combustion performance and produce longer ignition delay and shorter combustion duration compared with pure diesel. Moreover, a slight increase of brake specific fuel consumption and brake thermal efficiency occurs when a Diesel engine operates with blended fuels, rather than diesel fuel. Diesel-dimethylfuran blends could lead to higher NO_x *emissions at medium and high engine loads. However, there is a significant reduction in soot emission when engines are fueled with diesel-dimethylfuran blends. Soot emissions under each operating conditions are similar and close to zero except for D40 at 0.13 MPa brake mean effective pressure. The total number and mean geometric diameter of emitted particles from diesel-dimethylfuran blends are lower than pure diesel. The tested fuels exhibit no significant difference in either CO or HC emissions at medium and high engine loads. Nevertheless, diesel fuel produces the lowest CO emission and higher HC emission at low loads of 0.13 to 0.38 MPa brake mean effective pressure.*

Key words: *2,5-dimethylfuran, compression ignition engine, combustion, emission*

Introduction

With rapid development of industrialization and modernization, excessive consumption of fossil fuels has become the major source of air pollution and severe energy crisis [1]. Hence, it is increasingly urgent to seek out alternative and sustainable energy to overcome the challenges of energy shortage and pollutant emissions [2]. In particular, this search is highly significant in terms of the internal combustion engine (ICE) for approximately 60% of fossil fuels are consumed by transportation. According to current research, biofuels, such as ethanol [3, 4], methanol [5, 6], biodiesel [7, 8], and n-butanol [9, 10], have shown great potential to meet these challenges.

With sustained researches, increasing biofuels applicable for ICE are discovered. In the past, dimethylfuran (DMF) did not give rise to widespread concern of researchers due to

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its inconvenience in production. A highly efficient way of their mass production method has been successfully developed by investigators [11-14]. The properties of DMF are shown in tab. 1. Compared with ethanol, which is the most widely used biofuel in ICE, DMF has significant strengths as an alternative biofuel [15].

Firstly, the energy density of DMF is 40% higher than that of ethanol and much closer to gasoline (31.5 *vs.* 23 *vs.* 35 MJ/L). Secondly, compared to ethanol, DMF offers higher initial boiling point, which largely prevents its volatilization to form a vapor lock. Finally, DMF is relatively water-insoluble, which benefits its storage steady and transport. Therefore, DMF has great potential as a superior biofuel alternative.

Table 1. Properties of diesel, bioethanol, DMF, and gasoline [16-19]

Parameters	Diesel	Bioethanol	DMF	Gasoline
Chemical formula	C_{12}-C_{25}	C_2H_6O	C_6H_8O	C_4-C_{12}
Research octane number	20-30	109	101	96.8
Motor octane number	–	90	88	85.7
Octane number	–	108	119	90-99
Oxygen content [%]	0	34.78	16.67	0
Stoichiometric air-fuel ratio	14.3	8.95	10.79	14.7
Water solubility (wt.%, 20 °C)	Negligible	Miscible	Negligible	Negligible
Latent heat [$kJkg^{-1}$] at 25 °C	270-301	919.6	332	373
Lower heating value [$MJkg^{-1}$]	42.5	26.9	33.7	42.9
Initial boiling point [°C]	180-370	78.4	92	32.8
Auto-ignition temperature [°C]	180-220	434	286	420

Several studies have been conducted to examine the combustion and emission performance of engines fueled with DMF [20]. Researches on the combustion intermediates [21], knocking propensity [22], laminar burring velocities, and Markstein length in a premixed laminar DMF flame [23-25] are quite important to understand the fundamental combustion characteristics. Wang *et al.* [16] and Zhong *et al.* [26] compared the combustion performances and emissions of DMF with gasoline and ethanol in a single-cylinder gasoline direct injection spark ignition (DISI) engine at different engine loads. The results showed that because of the similar physicochemical properties, DMF exhibited similar combustion and emissions characteristics to those of gasoline. Authors [17, 18, 27] researched combustion and emissions under low temperature combustion fueled with diesel-DMF blends in a Diesel engine, and the results demonstrated that DMF20 (20% blending ratio by volume) led to much longer ignition delay, which was the main factor in soot reduction, when the DMF blending ratio was 40%, the trade-off relationship between NO_x and soot disappeared and soot emissions were close to zero. Daniel *et al.* [28] experimented the effect of spark timing and load on direct injection engine fueled with gasoline, ethanol, and DMF. The results showed that DMF presented comparable combustion efficiency and emission qualities to gasoline and surpassed ethanol in certain cases. All results highlight that DMF has the potential to be a competitive alternative.

Currently, to the authors' knowledge, a little research has been conducted with diesel-DMF blends in different proportions and relatively comprehensive operating conditions in

DICI engines. In present research, different mass fractions of DMF were blended with pure diesel. The study focused on the combustion and emission characteristics at different mass fractions and operating conditions.

Experimental

Engine and instrumentation

All experiments were conducted on a modified 4-cylinder, 4-stroke, water-cooled, DICI engine coupled with a common rail fuel injection system, as illustrated in fig. 1(a). The real experimental set-up is shown in fig. 1(b). The primary specifications of the engine are given in tab. 2. The engine was coupled to an eddy current dynamometer to maintain a constant speed of 1800 rpm (±5 rpm) regardless of the engine torque output. To determine and monitor the desired engine parameters, such as injection timing and injected fuel mass, the engine was modified with an electrical control module (provided by Cheng Du ELECK Company, China). The injection timing was fixed at a 7.5 crank angle (°CA) before top dead center (bTDC) by the ECTEK ECU manager software.

Table 2. Engine specification

Type of engine	4-cylinder, 4-stroke
Bore	96 mm
Stroke	103 mm
Compression ratio	17.5
Displacement	2982 cm^3
Rated power	85 kW
Rated speed	3200 rpm
Type of ignition	Compression ignition
Method of starting	Electric start
Initial injection	7.5 °CA bTDC

The in-cylinder pressure was measured using a Kistler 6025C pressure sensor, which was flush-fitted with the wall of the cylinder head. The signals were delivered to a charge amplifier and later received by a CB-466 combustion analyzer. The pressure data was taken every 0.25 °CA intervals for 100 consecutive cycles, and then averaged. The signal of the TDC position was taken by a Holzer sensor fitted in the crankshaft free wheel. In order to control the pressure and temperature of the intake air,

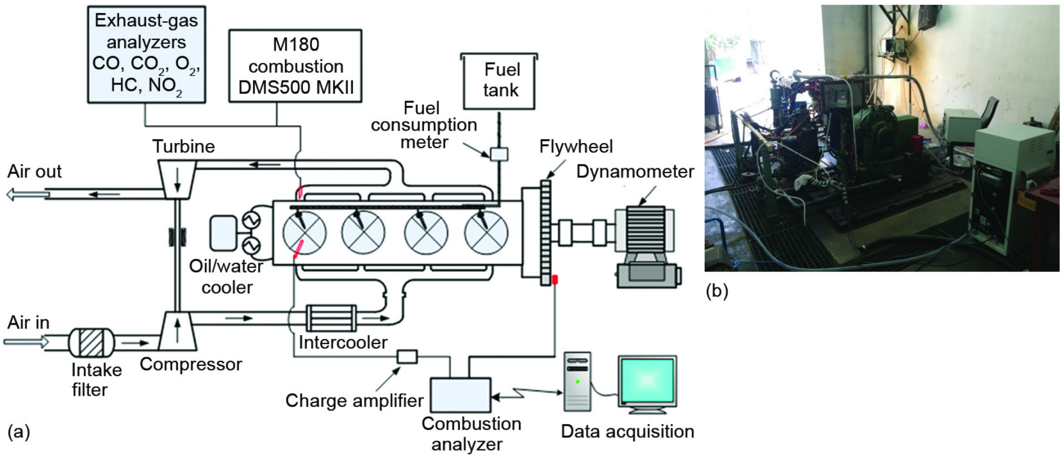

Figure 1. (a) Schematic and (b) real experimental set-up

a supernumerary compressor and an air conditioning system were used. The intake air temperature is stably maintained at 25 °C (±0.5 °C). The coolant temperature of engine was precisely stabilized at 85 °C by a temperature controller, while the lubricating oil temperature varied from 87 °C (±2 °C) as the load increases.

The gaseous emissions were measured using an AVL DIGAS 4000 gas analyzer with resolution of 1 ppm for HC and CO and 0.1% for NO_x. The smoke was measured by an NH-T6 smoke opacimeter with resolution of 0.01 m^{-1}. Exhaust samples were pumped from the exhaust port through a long tube to the analyzer. A mean of a differential mobility spectrometer (DMS500) was used to estimate the particle size distribution function (PSDF).

Test fuels and experimental procedures

The fuels in this study were conventional diesel fuel (China Petroleum and Chemical Corporation) and DMF (99% purity, Ten Zhou Hong Li Biological Technology Co. Ltd.), with properties shown in tab. 3. Different mass fractions of DMF (10, 20, 30, and 40%) were mixed with diesel fuel, marked as D10, D20, D30, and D40, respectively.

Table 3. Properties of test fuels

Parameters	Diesel	DMF
Density at 15 °C [$kgcm^{-3}$]	906.36	832.62
Flash point [°C]	63	–1
Viscosity at 40 °C [Pa·s]	2.7	0.4285
Cetane number	52.1	9
Recovered at 95 °C [°C]	345-350	–

All tests were carried out at 1800 rpm, fixed injection timing (7.5 °CA bTDC), ambient air intake conditions (approximately 25 °C ±1 °C). Engine load was changed from 10% to 90% at a 20% increment corresponding to brake mean effective pressure (BMEP) of 0.13, 0.38, 0.63, 0.88, and 1.13 MPa, respectively. To guarantee the reliability and repeatability of all experimental data, the engine was firstly warmed up to steady condition, while temperatures of coolant and lubricating oil were kept at 85 °C and 87 °C, respectively. Then all the measurement data including control parameters, in-cylinder pressure data, and emission were recorded for off-line analysis. To purge the fuel supply system and ensure no residual fuel after each experiment, the engine is sustained to operate for long enough before data acquisition after changed to a new fuel. Each measurement data was acquired for twenty times so as to guarantee the reliability of results.

Results and discussion

Combustion characteristics

Both in-cylinder pressure and corresponding heat release rate curves of tested fuels were compared at engine loads of 0.13 MPa, 10% load, fig. 2(a), or 0.88 MPa, 70% load, fig. 2(b). For comparison, the motoring pressure curve is also given. For different fuels, the in-cylinder pressures closely follow the motoring pressure before –7.5 °CA aTDC. The figures show that the shapes of the in-cylinder pressure and heat release rate curves of the diesel-DMF blends are similar to that of pure diesel at the same operating conditions. At 10% engine load, the maximum in-cylinder pressure, $p_{\max}$, decreases and occurs further from TDC with the increase of DMF mass fraction in the blended fuels. However, at 70% engine load, $p_{\max}$ rises with the increase of DMF mass fraction, which is opposite to the changing trend at 10% engine load.

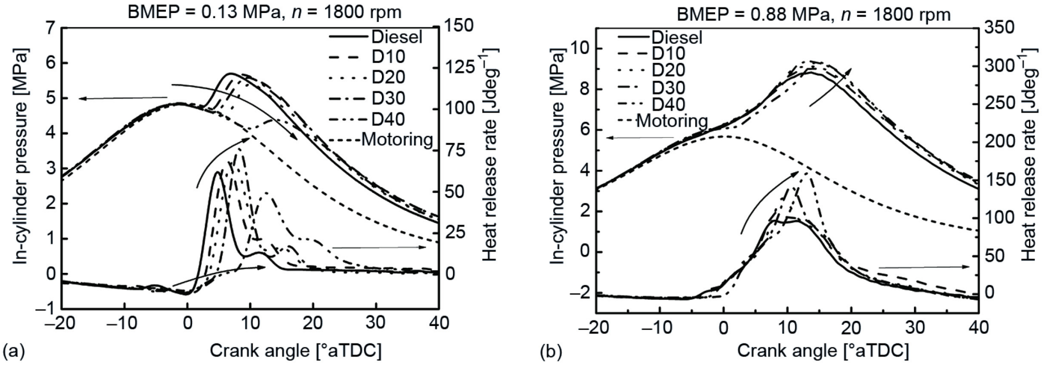

Figure 2. In-cylinder pressure and heat release rate at: (a) 0.13MPa BMEP and (b) 0.88 MPa BMEP

The peak heat release rate is observed to increase and occurs further from TDC, both at 10% and 70% engine loads, with the increase of DMF mass fraction, which is similar to the results of Zhang *et al.* [17]. These results indicate the delayed start of combustion and significantly prolonged ignition delay, which leads to more fuel combusted in the premixed phase, resulting in higher peak heat release rate [27]. At the diffusion combustion phase, the diesel-DMF blends provide a higher heat release rate than pure diesel does, indicating that the diffusive combustion is improved because of the higher oxygen content in the blended fuels.

Figure 3(a) presents the ignition delay *vs.* engine load for the five tested fuels. Ignition delay is defined as the crank angle interval between the start of injection and 10% of the total heat release. It can be observed from fig. 3(a) that ignition delay is prolonged with the increase of DMF fraction, which could be explained from the perspective of fuel properties. The octane number of DMF is considerably higher than that of diesel fuel. Generally, a higher octane number corresponds to a lower cetane number, which together with higher auto-ignition temperature mainly explains the longer ignition delay of blended fuels. Moreover, the higher latent heat from DMF vaporization is another important cause of lower combustion temperature and thereupon increased ignition delay. The differences of ignition delay among tested fuels become small at high loads because of the high temperature and pressure that suppress the influence of fuel properties on ignition delay. Especially at 10% engine load, the longer ignition delay of all fuels may be attributed to the lower equivalence ratio, as shown in fig. 4.

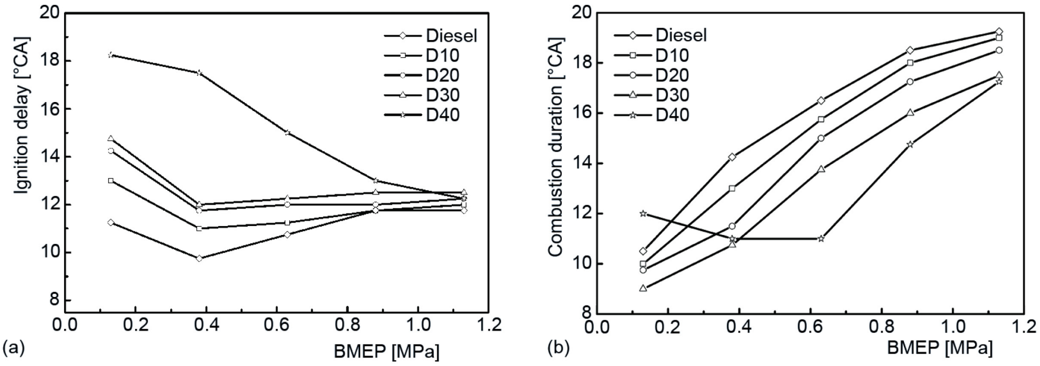

Figure 3. (a) Ignition delay and (b) combustion duration at various loads

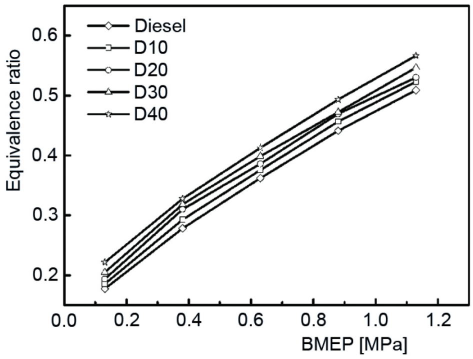

Figure 4. Equivalence ratio

Figure 3(b) illustrates the effects of fuel properties and engine loads on the crank angles corresponding to combustion duration. Combustion duration is defined as the difference in crank angle position between the 10% and 90% mass fractions burned. The combustion duration is evidently prolonged with the increase of engine load, which is mainly caused when more fuel is burned. However, the increase of DMF fraction leads to prolonged ignition delay and curtailed combustion duration. Low cetane fuels have an overly long ignition delay. Therefore, most of the fuel is injected before ignition occurs, leading to the generation of a more combustible mixture and a notably rapid burn rate [27]. According to the study by Donahue and Foster [29], blended fuels with higher oxygen content in the spray reduce pyrolysis and promote oxidation, thereby shortening the combustion. The longest combustion occurs to D40 at 0.13 MPa BMEP, primarily due to its overly long ignition delay.

Figures 3(a), 3(b), and 4 demonstrate that there is considerable difference between D40 and other tested fuels at low loads, such as much longer ignition delay and combustion, considerably lower in-cylinder pressure, and lower heat release rate. A lower cetane number and higher latent heat of vaporization cause D40 to be more difficult to combust completely and even may cause misfire [30], which indicates 40% of DMF in the blended fuel is close to the limit ratio at which blended fuels can burn in-cylinder.

Brake specific fuel consumption and thermal efficiency

Figures 5(a) and 5(b) show the brake power, brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) which are calculated from the engine torque, engine speed, fuel consumption rate, and lower heating value.

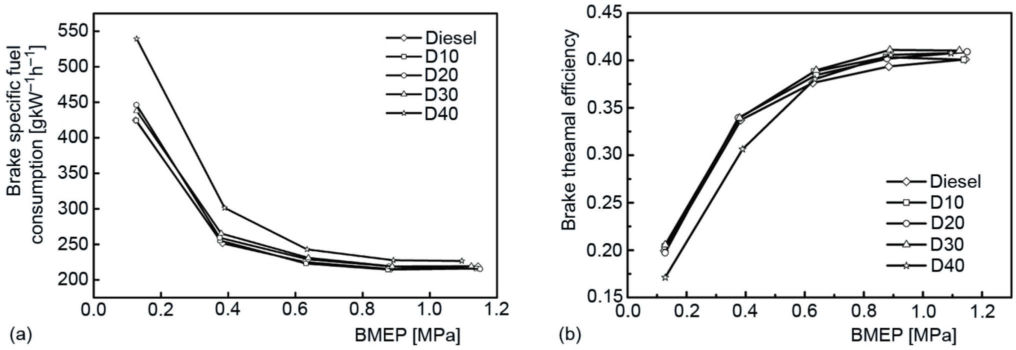

Figure 5. (a) BTE and (b) BSFC at various loads

As shown in fig. 5(a), the BSFC curves of all tested fuels have the same trend that BSFC decreases with the increase of engine load. The BSFC increases with the blended proportion of DMF. For D10, the BSFC has less difference than that of diesel fuel, however, both values are lower than that of the remaining tested fuels, especially D40. The lower calorific value of DMF compared with that of diesel fuel primarily contributes to the increase in BSFC.

Obviously, the BSFC of D40 is much higher than that of the other tested fuels, particularly at low and medium engine loads. These results are due to the excessive DMF in the blended fuel that leads to poor in-cylinder combustion.

In contrast with BSFC, the BTE increases with engine load rise, as shown in fig. 5 (b). However, the BTE of diesel fuel decreases at 1.13 MPa, indicating the deterioration in engine performance at this engine load. The BTE of D40 is 2.8% lower than that of pure diesel fuel at low engine loads, which results from poor D40 combustion. The BTE of all of the blends expect D40 are higher than that of diesel fuel, the values increase with engine load rise. Otherwise, BTE also increases with the increase of DMF proportion, which is primarily due to the improvement of the combustion process on account of increased oxygen content in the blends. Meanwhile, the increase of DMF fraction leads to shorter combustion duration, which increases the degree of constant volume combustion. With the increase of DMF fraction, the blends obtain better volatility, which allows the blends to form more-uniform charged mixtures and improve the combustion process. The BTE of D30 increases by 6.4% at an engine load of 0.88 MPa compared to pure diesel.

Effects of DMF fuel properties on pollutant emissions

The engine emissions are compared between the tested fuels at various loads. The legislated emissions are evaluated, including NO_x and soot emission, and HC and CO emissions are subsequently analyzed.

Figure 6(a) shows the comparison of NO_x emissions under different engine loads for all tested fuels. Generally, the rising engine load, which leads to higher in-cylinder tempera-

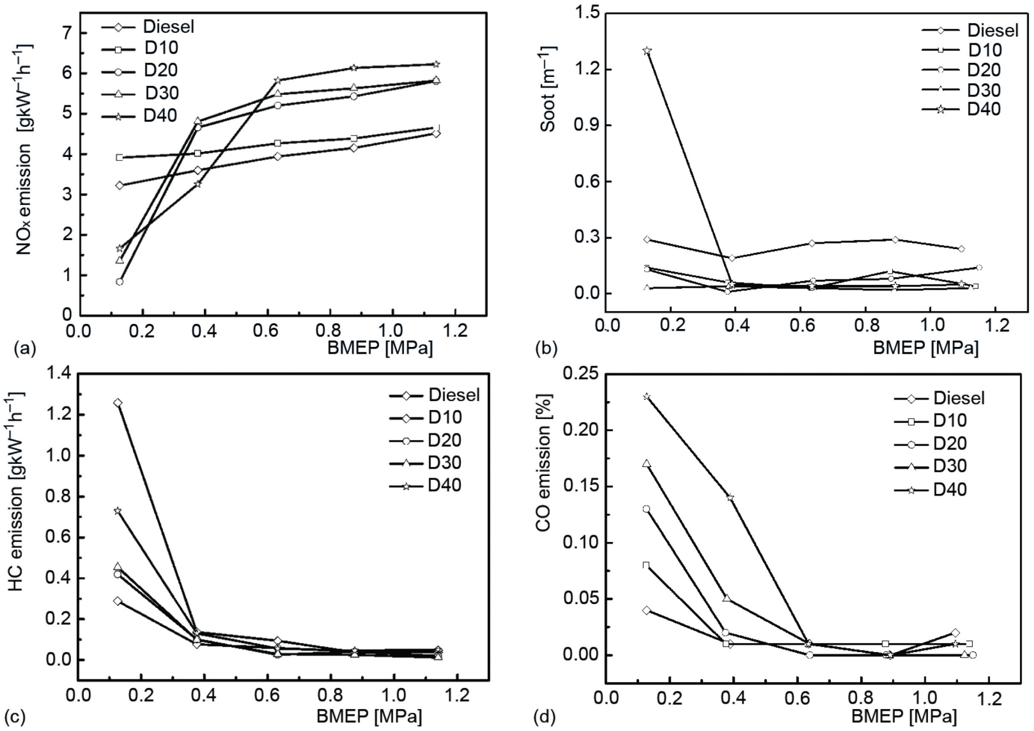

Figure 6. Emissions of different fuels at various loads: (a) NO_x, (b) soot, (c) HC, and (d) CO

ture, largely promotes the formation of NO_x [31]. The increased DMF proportion in the blends causes two circumstances for NO_x formation. Firstly, at low engine loads (0.13 MPa), the NO_x emission of diesel fuel is lower than that of D10 but higher than that of the other fuels.

The oxygen content and cetane number are two main influence factors on the formation of NO_x. Compared with diesel fuel, the 10% DMF increases the oxygen content in D10 which, in turn, can significantly improve the combustion process and lead to higher in-cylinder temperature. Despite the higher oxygen contents, D20, D30, and D40 also have lower cetane numbers than D10, which aggravate combustion and counteracts the effect of higher oxygen content. The interaction of these two factors leads to lower in-cylinder temperature. Secondly, at medium and high loads (from 0.63 to 1.13 MPa, respectively), the formation of NO_x is promoted by the increase of DMF fraction. These results occur because the in-cylinder temperature becomes higher at medium and high loads, so the oxygen content imposes a much more significant effect than cetane number does.

Figure 6(b) presents the effects of engine load and blends on soot emissions. Obviously, with the increase of DMF fraction, soot emissions decrease greatly, similar to the results of Zhang *et al.* [17] for the following reason. Firstly, DMF addition leads to longer ignition delays. The extended ignition delay makes more time available for premixing, which reduces soot formation. Meanwhile, the decrease of the amount of the fuel burned reduces the exhaust smoke. Secondly, DMF is an oxygenated fuel. Soot is primarily produced during the diffusive combustion phase [32]. Therefore, adding oxygenates to diesel fuel can reduce engine soot emissions because oxygenates improve the diffusive combustion and promote the post-flame smoke oxidation in the late expansion and exhaust processes. Ren *et al.* [33] found that soot reduction was strongly related to the oxygen mass fraction in the blends and less related to the type of oxygenate.

Figure 6(c) displays the comparison of HC emissions *vs.* engine loads for the tested fuels. Above engine loads of 0.38 MPa BMEP, HC emissions are not significantly different and are very low. Prior to these findings, the HC emissions decrease rapidly from 0.13 to 0.38 MPa BMEP for each fuel. Compared to diesel fuel, the HC emissions from D10, D20, and D30 decrease sharply at low loads, and are maximized from D40. The high temperature leads to slight HC emissions from each fuel at medium and high loads. The fuel oxygenation that enhances the HC oxidation and lowers the volatility may primarily explain why blended fuels produce less HC emissions at low loads compared with pure diesel. Daniel *et al.* [31] suggest that the lower volatility mainly accounts for the difference of HC emission at low engine loads. The incomplete combustion of D40 leads to the highest HC emissions at low loads.

The CO emissions of the studied fuels at various loads are displayed in fig. 6(d). Similar to HC emissions, CO emissions decrease rapidly with the rising load from 0.13 to 0.38 MPa BMEP. Compared with diesel fuel, the blended fuels produce higher CO emissions at low loads of 0.13 to 0.38 MPa. Meanwhile, the higher the fraction of DMF is, the higher the CO emissions are. Above 0.38 MPa BMEP, CO emissions are relatively consistent and are notably low.

The lower CO emissions are the indicators of complete combustion; thus, more complete combustion occurs. A notably higher auto-ignition temperature of blended fuels leads to more unburned fuel. Thus, the CO emissions at low in-cylinder temperature at low engine loads occur. In general, CO emissions as the intermediate products of chemical kinetics can be rapidly oxidized to CO_2 when the combustion temperature exceeds 1400 K. At medium and high loads, as more fuel is burned, the high in-cylinder temperature reduces the CO emissions.

The PSDF of pure diesel, D10, and D30 at engine loads of 0.88 MPa (70% load) are shown in fig. 7. Compared with pure diesel, the blended fuels produce less total number of particles. The maximum peaks which reflect mean geometric diameter, D_p, of the particles decrease with the increase of DMF mass fraction in the blended fuels. Also, blended fuels showed the lower particle concentrations at all size than pure diesel. The oxygenated fuel such D10 and D30 may conduce to better fuel oxidation in the fuel-rich zone where the soot likely forms [34]. This would result in lower total number and smaller size of particles. Furthermore, as biofuel, DMF has no C-C chemical bond which would help to reduce particles formation.

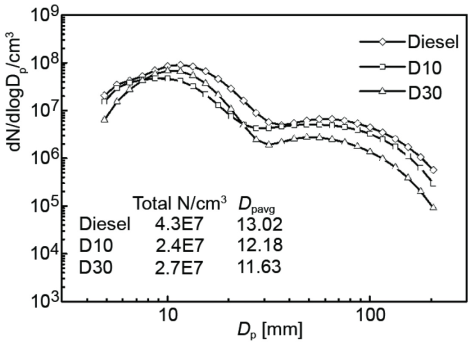

Figure 7. Particle number size distribution at 0.88 MPa BMEP

Conclusions

A series of experiments on a common-rail medium-duty commercial Diesel engine were conducted to examine the combustion and emissions characteristic of DMF. Five types of DMF-diesel blends were tested during the experiments, including pure diesel fuel, D10, D20, D30, and D40 blends. The main conclusions are as follows.

- The D10, D20, and D30 show similar combustion characteristics as diesel fuel. With increase of DMF mass fraction, the maximum in-cylinder pressure occurs further from TDC, leading to longer ignition delay and shorter combustion duration.
- Due to high auto-ignition temperature and oxygen content, the diesel-DMF blends exhibit longer ignition and shorter combustion duration than pure diesel.
- The BSFC increases with the increase of DMF proportion in blends fuel. However, except for D40, the BTE of diesel-DMF blends is higher than pure diesel.
- Blends with DMF produce lower HC but higher CO than pure diesel at low loads of 0.13 to 0.38 MPa BMEP. Above engine loads of 0.38 MPa BMEP, HC, and CO emissions are not significantly different and are very low. Meanwhile, NO_x emissions increases with the increase of DMF fraction in blends fuel at medium and high loads of 0.63 to 1.13 MPa BMEP.
- Soot emission from blends with DMF is significantly reduced compared to pure diesel. Meanwhile, both the total number and D_p of emitted particle are lower when fueled with DMF-diesel blends, making DMF superior fuel for Diesel engine.
- High DMF fraction in fuel blends is not suitable for DICI engine. D40 exhibit significant poor combustion performance (much higher BSFC and lower BTE) compared with pure diesel, indicating that indicating that D40 arrived reached at the critical point of which the engine operation.

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