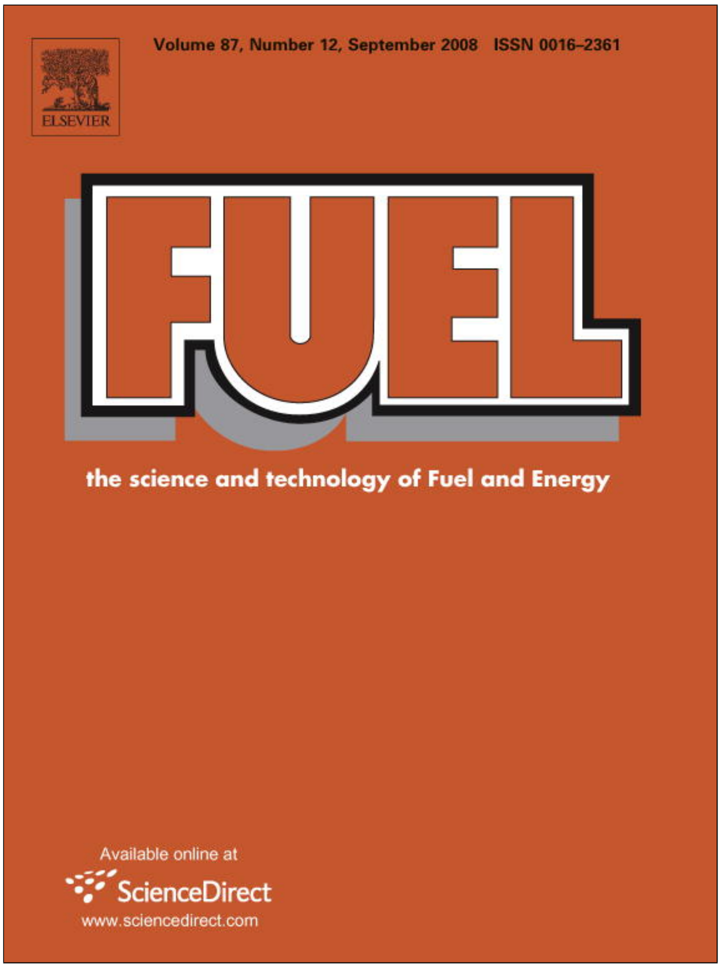
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Combustion and emissions of a DI diesel engine fuelled with diesel-oxygenate blends

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Abstract

Combustion and emissions of a DI diesel engine fuelled with diesel-oxygenate blends were investigated. The results show that there exist the different behaviors in the combustion between the diesel-diglyme blends and the other five diesel-oxygenate blends as the diglyme has the higher cetane number than that of diesel fuel while the other five oxygenates have the lower cetane number than that of diesel fuel. The smoke concentration decreases regardless of the types of oxygenate additives, and the smoke decreases with the increase of the oxygen mass fraction in the blends without increasing the NO_x and engine thermal efficiency. The reduction of smoke is strongly related to the oxygen-content of blends. CO and HC concentrations decrease with the increase of oxygen mass fraction in the blends. Unlike conventional diesel engines fueled with pure diesel fuel, engine operating on the diesel-oxygenate blends presents a flat NO_x/Smoke tradeoff curve versus oxygen mass fraction.

Keywords: Oxygenate; Combustion; Diesel engine

1. Introduction

With the more concerning for environmental protection and human health, the increasingly stringent emissions standards are implemented all over the world. The seeking of new methods in reducing the harmful exhaust emissions and improving the fuel economy of engines become urgently. The advantage of the diesel engine compared with the gasoline engine is better fuel economy benefits and high power output; however, the high NO_x and smoke emissions are still the main obstacle for the development of diesel engines. Thus, the reduction of engine emissions becomes one of major researches in the engine development.

In the conventional diesel engine, it is difficult to simultaneously reduce NO_x and smoke due to the tradeoff relationship between NO_x and smoke. For stringent emissions regulation, the wide ranges of technologies including combustion improvement, fuel improvement and exhaust treatment should be taken, in which fuel improvement has become one of the most promising approach in realizing low emissions engine. Oxygenated fuels are beneficial to the reduction of smoke or particulate matter emissions in diesel engines. Thus, using diesel-oxygenate blends can decrease the engine emissions without large modification of the diesel engine and has a wide applicability in currently used vehicle.

In recent years, many experiments on oxygenated fuels in the diesel engine were performed, and the researches proved that smoke emissions from the engine could be reduced when oxygenated fuels were blended with diesel fuel [1–9]. Moreover, previous reports showed that comparing with diesel fuel, the use of diesel-oxygenate blends in the diesel engine did not bring the increase of NO_x emission [10–13]. Miyamoto et al. [9] found that the amount of smoke reduction depended on the total oxygen mass frac-

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Notation

ATDC	after top-dead-center	Hu	lower heating value (MJ/kg)
bmep	brake mean effective pressure (MPa)	H, wt.%	mass fraction of hydrogen in fuel blend
BTDC	before top-dead-center	m	mass of cylinder gases (kg)
bsfc	brake specific fuel consumption (g/kW h)	O, wt.%	mass fraction of oxygen in fuel blend
θ_{fd}	fuel delivery advance angle (CA BTDC)	TDC	top-dead-center
C, wt.%	mass fraction of carbon in fuel blend	φ_c	crank angle of the center of heat release curve (CA degrees ATDC)
$dQ_B/d\varphi$	heat release rate with crank angle (kJ/CA)		

tion in the blends, and species of each oxygenated fuel added into the diesel fuel had approximately the same impact in reducing the smoke emissions with the same total oxygen-content of the blends. Adelbert [14] investigated the effects of oxygenates blended with the diesel fuel on the particulate matter (PM) emissions in a compression–ignition engine, and they found that PM reduction was only correlated to the oxygen-content of the blends and PM was reduced by about 3.5% for each 1% of fuel oxygen by mass. Westbrook et al. [15] used the detailed chemical kinetic modeling to simulate soot reduction in diesel engine and their results showed that oxygenate species influenced the smoke reduction level since the amount of soot precursor was varied for different oxygenate additives

In practice, comparing with gaseous oxygenates like DME, adding some liquid oxygenate into diesel fuel in reducing engine emissions without modifying fuel system seems to be a more attractive method. Up to now, reports on utilization of oxygenate additives in diesel engine give individual influence and lack the comprehensive study on combustion and emissions of the diesel engine fuelled with diesel-oxygenate blends. The general behaviors of oxygenate additives in engine combustion and emissions have not clarified yet, making it difficult to provide the guidance for engine operation and selection of oxygenates.

Based on the authors' previous study [16–19], the objectives of this study are to investigate the general behavior in combustion and emissions of a compression–ignition engine operating on various types of diesel-oxygenate blends. The study provide the common insights on combustion and emissions operating on the diesel-oxygenate blends.

2. Experimental setup and procedures

In this study, diesel fuel is the base fuel while dimethoxymethane (DMM), diglyme (DGM), dimethyl carbonate (DMC), diethyl carbonate (DEC), diethyl adipate (DEA) and ethanol are used as the oxygenic additive. The six selected oxygenated fuels can reflect ethers, esters and alcohols, respectively. For the structure of molecule, esters include the $C–O_2$ moiety with one C=O structure, ethers have the $C_2–O$ moiety, and alcohols have the C–OH moiety. These oxygenates were individually blended with pure diesel fuel to make the diesel-oxygenated blends, and the volume fractions of oxygenates in the blends were from 0% to 20%. These blended fuels and pure diesel fuel were tested in a direct-injection diesel engine. For the test engine, the original fuel delivery advance angle is 25 CA BTDC, and engine speed at maximum torque is reached at 1400 rpm. The detailed specifications of the test engine are listed in Table 1, and fuel properties are listed in Table 2. The tested diesel fuel was provided by China Petroleum

Table 1
Engine specifications

Bore (mm)	100
Stroke (mm)	115
Displacement (cm^3)	903
Compression ratio	18
Shape of combustion chamber	ω shape in the bottom of bowl-in-piston
Rated power/speed	9.5 kW/2000 rpm
Nozzle hole diameter (mm)	0.3
Number of nozzle holes	4

Table 2
Fuel properties of diesel and oxygenic fuels

	Diesel	Dimethoxymethane	Diglyme	Diethyl adipate	Dimethyl carbonate	Diethyl carbonate	Ethanol
Chemical formula	$C_{10.8}H_{18.7}$	$C_3H_8O_2$	$C_6H_{14}O_3$	$C_{10}H_{18}O_4$	$C_3H_6O_3$	$C_5H_{10}O_3$	C_2H_6O
Molecular weight	148.3	76	134	202	90	118	46
Theory air/fuel ratio kg/kg	14.36	7.24	11.24	13.25	4.59	8.79	12.15
Density (g/cm^{-3})	0.83	0.85	0.94	1.00	1.06	0.97	0.79
Cetane number	45	30	126	–	35	–	8
Lower heating value (MJ/kg)	44	22.4	24.5	25.5	13.5	21.1	26.78
Heat of evaporation (kJ/kg)	260	385	322	295.1	369	360.4	854∼904
Boiling point (°C)	180–330	43	161.3	127	90.9	126.8	78.4
C (wt.%)	86	47.4	53.7	59.4	53.3	50.8	52.2
H (wt.%)	14	10.5	10.5	8.9	6.7	8.5	13
O (wt.%)	0	42.1	35.8	31.7	40	40.7	34.8

and Chemical Corporation and the solidifying point of diesel fuel is 273.15 K. It should be noted that the oxygenate fuels have the lower energy density compared with that of pure diesel fuel as shown in Table 2.

In the experiment, the beginning timing of nozzle valve lifting was measured by the needle lift detecting apparatus; AVL Di-smoke was used to measure the exhaust smoke, and the exhaust NO_x was measured by AVL Di-Gas 4000 light. The cylinder pressure was recorded with the Kistler type cylinder pressure sensor, and the data were recorded for every 0.1 crank angle. The crank angle was measured with the Kistler type crank angle sensor. Meanwhile, the cylinder pressure and emissions were measured and analyzed under the same brake mean effective pressure (bmep) and engine speed, and combustion analysis was made based on the cylinder pressure information.

3. Results and discussion

3.1. Combustion characteristics

The ignition delay is defined as the time interval from the beginning timing of nozzle valve lifting (i.e. the start of fuel injection) to the separation timing between rate of cylinder pressure rise and rate of motoring pressure rise (it is regarded as the start of combustion). The total combustion duration is the duration from the start of combustion to the end of combustion.

Thermodynamic model is used to calculate the thermodynamic parameters in this paper, the model neglects the leakage through the piston rings, and thus the energy conservation in cylinder is written as follows:

$$\frac{dQ_B}{d\varphi} - \frac{dQ_W}{d\varphi} = \frac{d(mu)}{d\varphi} + p\frac{dV}{d\varphi} = mC_V\frac{dT}{d\varphi} + mT\frac{dC_V}{d\varphi} + p\frac{dV}{d\varphi} \quad (1)$$

Gas state equation is

$$pV = mRT \quad (2)$$

The variation of gas state equation with crank angle is given by:

$$p\frac{dV}{d\varphi} + V\frac{dp}{d\varphi} = mR\frac{dT}{d\varphi} \quad (3)$$

Heat release rate $\frac{dQ_B}{d\varphi}$ can be derived from formula (1) and (3) as follows:

$$\frac{dQ_B}{d\varphi} = p\cdot\frac{C_p}{R}\frac{dV}{d\varphi} + \frac{C_V V}{R}\frac{dp}{d\varphi} + mT\frac{dC_V}{d\varphi} + \frac{dQ_W}{d\varphi} \quad (4)$$

where heat transfer rate is given by

$$\frac{dQ_W}{d\varphi} = h_c\cdot A\cdot(T - T_W) \quad (5)$$

Heat transfer coefficient h_c uses the Woschni's correlation formula, and C_p and C_V are temperature-dependent parameters, their formulae are given in literature [20].

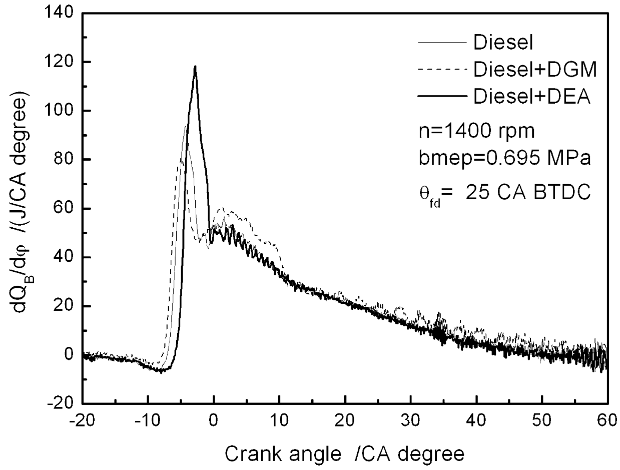

Fig. 1. Heat release rate of the blends.

Fig. 1 shows the heat release rate of the blended fuels, respectively. Due to the variation of fuel properties of the blended fuels with the addition of oxygenates, the heat release rate of the diesel-oxygenated blends gives different behaviors compared to those of pure diesel fuel. The figures reveal that the combustion initial phase is postponed and the maximum rate of heat release is increased for the most diesel-oxygenate blends except the diesel-DGM blend. The high cetane number of DGM leads to the advancement of the combustion initial phase and the decrease of the maximum rate of heat release. Thus, the combustion phase and the heat release rate are strongly related to the cetane number of the oxygenate fuel.

Fig. 2 illustrates the ignition delay versus oxygen mass fraction in the blends. The ignition delay strongly depends on the cetane number of fuels. As shown in the figure, the ignition delay of the diesel-DGM blend decreases and the ignition delay of other diesel-oxygenate blends increases with the increase of the oxygen mass fraction in the blends. This is also due to the influence from the cetane number of the oxygenate fuel. The cetane number of DGM is higher

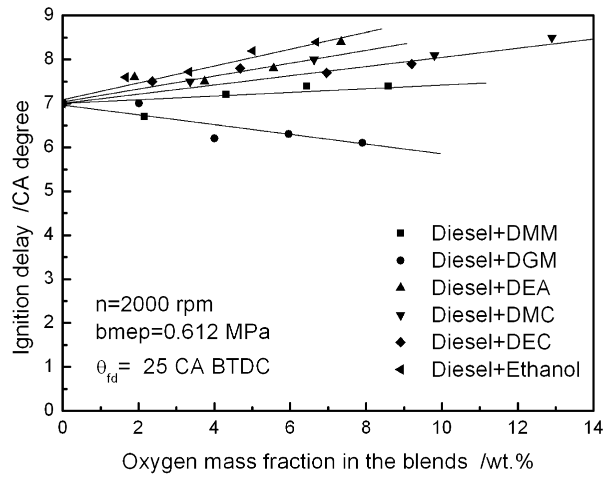

Fig. 2. Ignition delay of the blends.

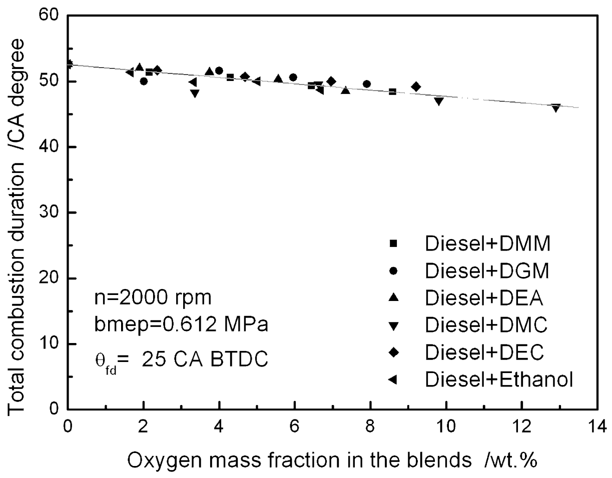

Fig. 3. Total combustion duration of the blends.

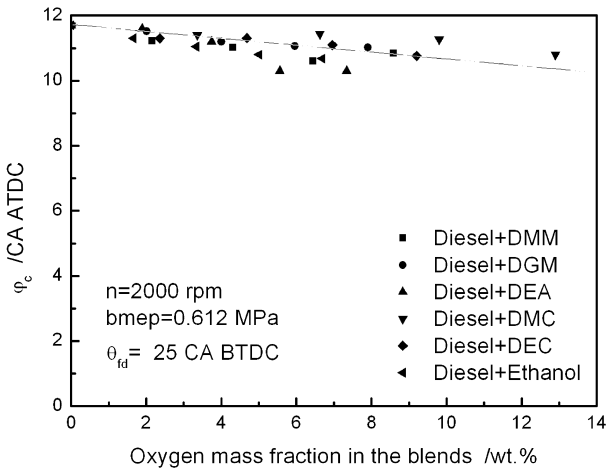

Fig. 4. Crank angle of the center of heat release curve (φ_c) of the blends.

than that of diesel fuel and the cetane numbers of other oxygenates are lower than that of diesel fuel. It is well known that the fuel with the high cetane number gives the short ignition delay while the fuel with the low cetane number gives the long ignition delay.

Fig. 3 shows the total combustion duration versus oxygen mass fraction in the blends. The total combustion duration decreases with the increase of oxygenate fraction in the blends. The decrease of the heating value of the blends by adding oxygenates requires more fuels to be injected into the cylinder to obtain the same power output. This is favorable to the extension of the total combustion duration of the diesel-oxygenate blends. However, adding oxygenate additive in the diesel creates more oxygen available for promoting combustion, especially the diffusive combustion. The comprehensive influence makes the decrease of the total combustion duration for the diesel-oxygenate blends.

Fig. 4 gives the crank angle of the center of the heat release curve (φ_c) versus oxygen mass fraction in the blends. For all test-blends, the center of the heat release curve moves close to the top-dead-center (TDC) with the increase of the oxygenate fraction in the blends. This is due to the improvement combustion process and decrease in combustion duration. A compact heat release process is presented in the case of the diesel-oxygenate blends.

Fig. 5 plots the brake specific fuel consumption (bsfc) versus oxygen mass fraction in the blends. The results show that bsfc increases linearly with the increase of the oxygen mass fraction in the blends. Two factors are considered to explain the behavior. One is the decrease in the heating value of the blends by adding oxygenates, and this requires more fuel to be injected into the cylinder to get the same power output, leading to the increase in the bsfc. Another is the improvement of the combustion by oxygen enrichment, and this will decrease the bsfc. The comprehensive influence makes the increase of bsfc in the case of diesel-oxygenate blends.

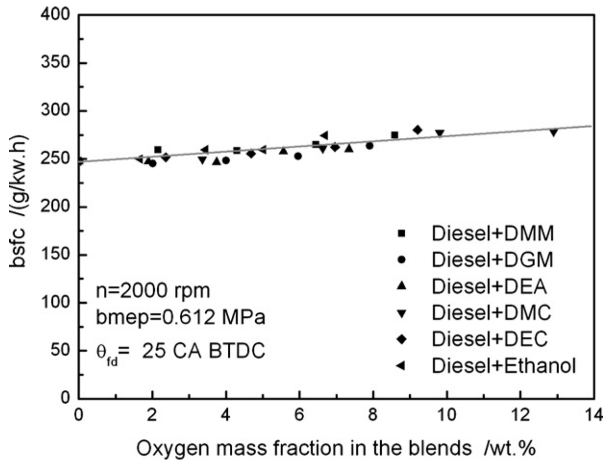

Fig. 5. Brake specific fuel consumption (bsfc) of the blends.

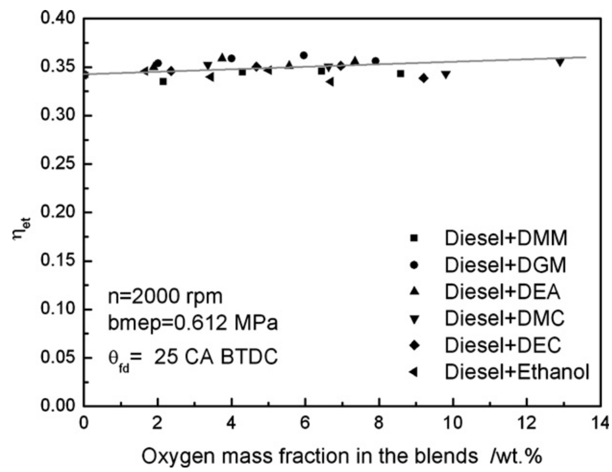

Fig. 6. Effective thermal efficiency (η_{et}) of the blends.

The effective thermal efficiency η_{et} versus oxygen mass fraction in the blends is illustrated in Fig. 6. The effective thermal efficiency shows a slight increase with the increase of the oxygen mass fraction in the blends. This indicates that the addition of the oxygenated fuels can increase the engine thermal efficiency. Short and compact heat release process results in an increase of engine thermal efficiency.

Figs. 7 and 8 give the cylinder maximum mean gas temperature (T_{max}) and the duration of the temperature over 1800 K versus oxygen mass fraction in the blends. In this study, the temperature was calculated with the thermodynamic model based on the cylinder pressure. Little variation in T_{max} is presented at the engine speed of 2000 rpm and a slight increase in T_{max} is presented at engine speed of 1400 rpm. The duration of temperature over 1800 K will decrease remarkably with increasing the oxygenate fraction in the blends. The combustion improvement by oxygen enrichment leads to the increase of gas temperature, but the decease of the combustion duration leads to the decrease of the duration of gas at high temperature.

Figs. 9 and 10 show the exhaust smoke concentration and its reduction rate versus oxygen mass fraction in the blends, respectively. In this experiment, a smoke meter of AVL Di-smoke 4000 was used to measure the exhaust

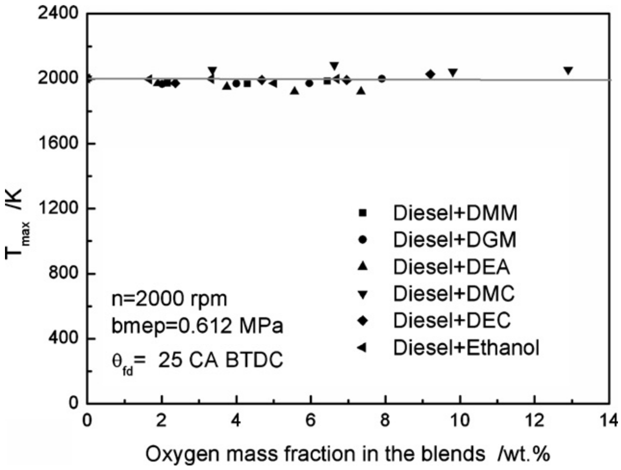

Fig. 7. Maximum temperature (T_{max}) of the blends.

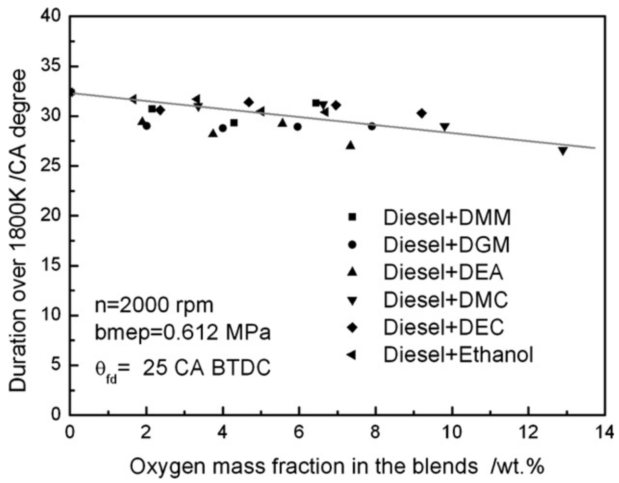

Fig. 8. Duration of gas temperature over 1800 K of the blends.

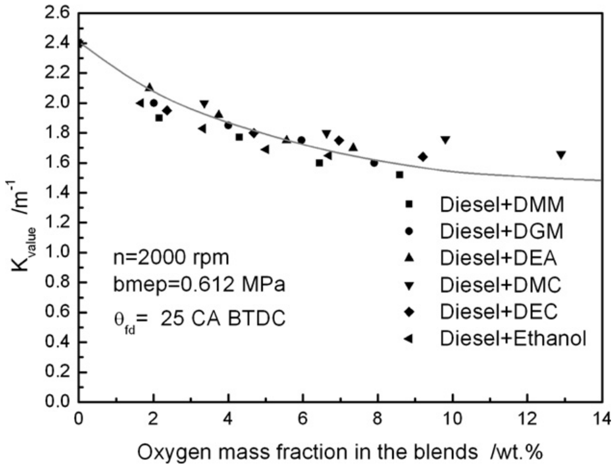

Fig. 9. Extinction coefficient (K_{value}) of the blends.

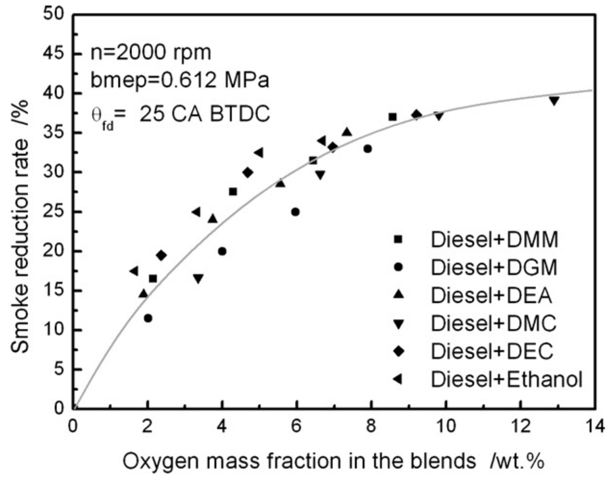

Fig. 10. Smoke reduction rate of the blends.

smoke, and the exhaust smoke concentration is scaled by the extinction coefficient (K_{value}). Smoke reduction ratio is defined by the formula of $[K_{value}(\text{diesel}) - K_{value}(\text{blends})]/K_{value}(\text{diesel})$. The figures reveal that the smoke concentration decreases with the increase of the oxygen mass fraction in the blends, and this behavior is more obvious at high engine loads. As smoke mainly produces during the diffusive combustion phase, adding the oxygenates to diesel fuel can reduce the engine smoke duo to the improvement of the diffusive combustion and the promotion of post-flame oxidation of smoke in the late expansion and exhaust processes. Meanwhile, the decrease of the amount of the fuel burned in the diffusive combustion duration is also beneficial to the reduction of the exhaust smoke. The results reveal that the smoke reduction is strongly related to the oxygen mass fraction in the blends and is less related to the type of oxygenate. Thus, smoke reduction of diesel-oxygenate blends can be controlled by adjusting the oxygen fraction in the blends. The smoke reduction rate increases, but its increment shows a decrease trend with the increase of the oxygen mass fraction in the blends Ten percent of the oxygen mass fraction in the blends can decrease the engine smoke by 30–40%.

Fig. 11 illustrates engine NO_x emission versus oxygen mass fraction in the blends. Under the same engine speed, engine load and fuel delivery advance angle, the engine NO_x concentration shows a slight decrease with the increase of the oxygen mass fraction in the blends. Two factors influence the NO_x for diesel-oxygenate blends, one is the gas temperature and another is the duration of gas at high temperature. In one aspect, using diesel-oxygenate blends increases the cylinder maximum mean gas temperature and this tends to increase the NO_x concentration. In other aspect, the usage of diesel-oxygenate blends decreases the duration of gas at high temperature and this tends to decrease the NO_x concentration. The comprehensive influence makes the NO_x behaviors of the diesel-oxygenate blends.

Fig. 12 gives the exhaust CO concentration and the exhaust HC concentration versus oxygen mass fraction in the blends, respectively. Similar to the behavior of the smoke, the engine CO concentration of the engine with the diesel-oxygenate blends decreases with the increase of the oxygen mass fraction in the blends. This is also due to the combustion promotion from the oxygen enrichment. HC concentration decreases with oxygenate flues addition, and this suggests that adding oxygenate fuels can decrease HC from the locally overrich mixture. Furthermore, oxy-

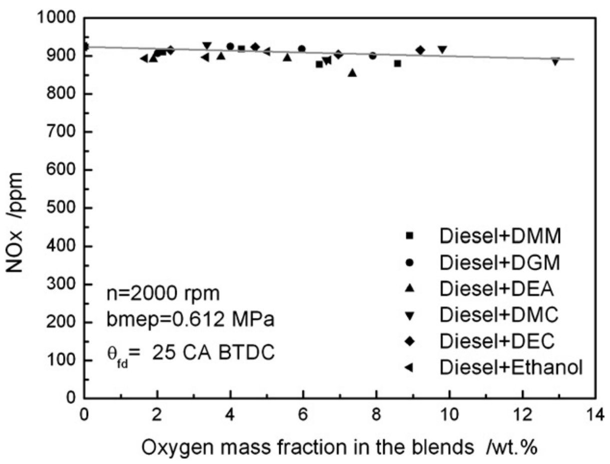

Fig. 11. NO_x concentration of the blends.

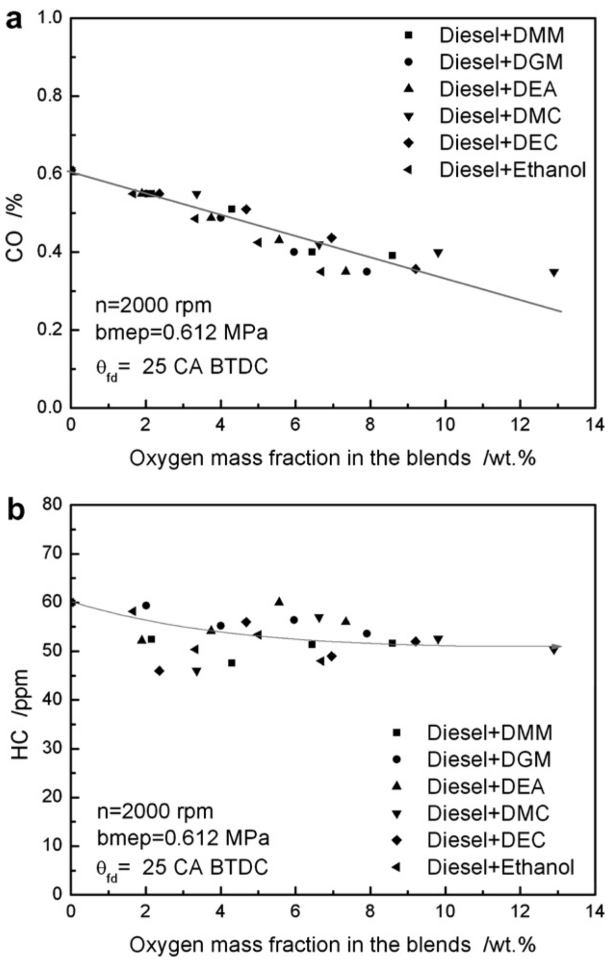

Fig. 12. CO concentration and HC concentration of the blends.

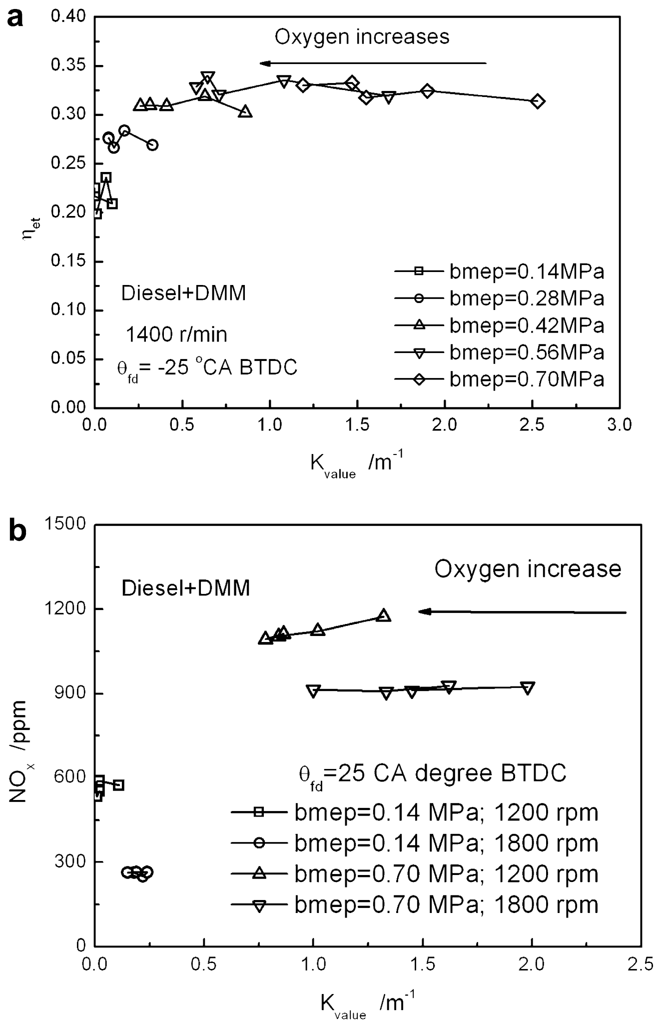

Fig. 13. Relationship between η_{et}/NO_x and smoke (K_{value}).

gen enrichment is also favorable to the oxidation of HC in the expansion and exhaust processes.

The relationship between thermal efficiency and smoke, between NO_x and smoke are plotted in Fig. 13. Adding the oxygenate fuels in diesel fuel can decrease the exhaust smoke concentration without decreasing the effective thermal efficiency. In conventional diesel engines fueled with pure diesel fuel, there exists the tradeoff curve between the smoke and the NO_x. However, in the case of diesel-oxygenate blends, a flat tradeoff cure between NO_x and smoke is presented. This suggests that smoke can be decreased by adding the oxygenate in diesel fuel without increasing the NO_x concentration. Thus, exhaust gas recirculation (EGR) can be used to eliminate the NO_x emissions. Oxygenate fuels have high EGR tolerance due to oxygen enrichment.

4. Conclusions

Combustion and emissions of a compression–ignition engine fuelled with diesel-oxygenate blends were investigated, and the main results are summarized as follows:

(1) Ignition delay decreases with the increase of diglyme fraction in the blends due to the high cetane number of diglyme. Ignition delays of the other five diesel-oxygenate blends increase with the increase of oxygenate fuels fraction in the blend the low cetane numbers of the oxygenate fuels. Combustion duration decreases with the increase of oxygenate fraction in the blends and a compact heat release process is present in the case of diesel-oxygenate blends.
(2) The centre of heat release curve moves close to the TDC with the increase of the oxygen mass fraction in blends.
(3) Smoke concentration decreases with the increase of the oxygen mass fraction in the blends without increasing NO_x CO and HC decrease with the increase of the oxygen mass fraction in the blends. A flat NO_x/smoke tradeoff curve is presented when operating on the diesel-oxygenate blends.

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