THE EFFECTS OF EGR AND INJECTION TIMING ON THE ENGINE COMBUSTION AND PARTICULATE MATTER EMISSION PERFORMANCES FUELLED WITH DIESEL-ETHANOL BLENDS

by

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In this paper, an experimental study was carried out to assess the effects of diesel/ethanol blends, including pure diesel, diesel (90%)/ethanol (10%) (E10, by mass) and diesel (80%)/ethanol (20%) (E20), on combustion and exhaust particulate matter emissions characteristics in a four-cylinder Diesel engine under a low engine load condition with different injection timings and exhaust gas recirculation ratios. Results indicated that ethanol addition delayed the ignition timing and shortened the combustion duration, meanwhile, produced higher brake thermal efficiency and lower brake specific fuel consumption. Moreover, the total particulate and nucleation mode particle (particle size Dp < 50 nm) number concentrations were increased and the total particulate mass and the accumulation mode (50 nm < Dp < 1000 nm) number concentration were decreased when engine fueled with diesel/ethanol blends compared to pure diesel.

Key words: *Diesel engine, diesel/ethanol, combustion characteristics, particulate matter*

Introduction

Currently, the development of alternative energy for the Diesel engine has been an effective method to cope with the shortage of fossil fuel and global environmental issues. Many eco-friendly fuels have been proposed so far, like compressed natural gas, liquefied petroleum gas, alcohols, and bio-diesel. Some of them have been applied to the vehicle engine currently. Ethanol has many advantages as a sustainable and renewable fuel [1-3]. Ethanol can be produced from many plants, such as crop straws, corn, and potato. Meanwhile, as an oxygenated fuel, it can produce lower particulate matter (PM) emission compared with diesel. However, ethanol is difficult to be used independently as a fuel for conventional Diesel engines due to its low cetane number [4]. Hence, many researchers used diesel/ethanol blends as Diesel engine fuels [5, 6]. However, the solubility of ethanol in diesel oil is limited. The amounts of ethanol in diesel/ethanol blends solutions are restricted up to 20% typically [7]. Besides that, ethanol has other disadvantages as an alternative fuel, such as its large latent heat of vaporization which will degrade cold start performance of engine.

In recent years, researchers from many research institutions have investigated combustion performances and emissions of engine fueled with diesel/ethanol blends. Prashant *et al*. [8] investigated the combustion parameters of a 4-cylinder engine operating using ethanol/diesel

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dual fuel with varying ethanol substitutions. They found that the in-cylinder pressure rise, ignition delay and maximum heat release are increased with ethanol mixed compared with pure diesel operation. Gnanamoorthi and Devaradjane [9] studied the effects of ethanol addition on combustion and emissions of a single-cylinder direct injections (DI) Diesel engine. Their results showed that ethanol addition could prolong the ignition delay, increase the NO_x emission and decrease the HC, CO, and smoke emissions. Pedrozo *et al*. [10] carried out experimental studies on a heavy-duty Diesel engine with ethanol injecting in inlet port. By adding ethanol, in-cylinder local temperatures and fuel-rich zones were reduced in most cases, resulting in lower soot and NO_x emissions. The maximum decline of soot and NO_x emissions could achieve 29% and 65%, respectively. Murcak *et al*. [11] investigated the effects of diesel blend with 5%, 10%, and 20% ethanol on combustion performances of a single-cylinder DI Diesel engine. The experimental results indicated that 5% and 10% ethanol addition improved power and torque output in the majority of cases, but power and torque showed a downward trend when fueled with ethanol (20%)/diesel (80%) blend fuel. In addition, ethanol addition increased the engine fuel consumption. Blasio *et al*. [12] studied gaseous pollutants and PM emissions of ethanol-diesel blends on a single-cylinder engine with dual-fuel configuration. The results showed that CO and HC emissions were increased at lower loads with the increase of ethanol content, but they were reduced at higher loads. In addition, the large size particles emission was reduced with ethanol addition, and ethanol addition could effectively reduce smoke under higher engine loads.

Many researchers have also researched the combustion and pollutants formation of diesel/ethanol blends using the simulation method. Achmad *et al*. [13] used AVL Boost to simulate the combustion performance and exhaust emissions of engine fueled with diesel/ethanol blends under different operating conditions. The simulation results showed that diesel blend with ethanol can produce lower CO, soot, and NO_x emissions. Meanwhile, the engine brake powers are slightly higher than those of pure diesel, especially for the engine speeds higher than 1400 rpm. Datta and Mandal [7] carried out a numerical simulation and observed that the ethanol blends into diesel prolong ignition delay and improve the combustion condition, resulting in a slight increase in brake thermal efficiency (BTE) but increase the blend fuel consumption. They noted a decrease in CO, HC, NO_x, and soot emissions, but an increase of CO_2 emissions in the exhaust gas.

Due to the frequent occurrence of haze phenomenon and the huge health hazards of PM, researchers have paid more attention to particle measurements of engine exhaust in recent years, especially for the measurements of particle size and number [14]. In addition, the emission regulations in many countries and regions have also limit the amount of particle number in vehicle exhaust [14]. Although an extensive experimental and numerical investigation on engine combustion and emissions fueled with diesel/ethanol blends have been performed, the effects of ethanol addition on particle size distribution, particle number and mass concentration in a Diesel engine still lack a deep investigation. In this study, the effects of ethanol additive (10% and 20% by mass) on combustion performance, NO_x and PM emissions under different injection timing and exhaust gas recirculation (EGR) ratio at a low engine load in a four-cylinder Diesel engine were investigated, especially the particle size distribution, particle number, and mass concentration were analyzed.

Experimental apparatus and methods

Engine and instrumentation

A four-cylinder water-cooled DI Diesel engine equipped with high pressure common rail fuel injection system and EGR system was used in this experimental study. Test equipment

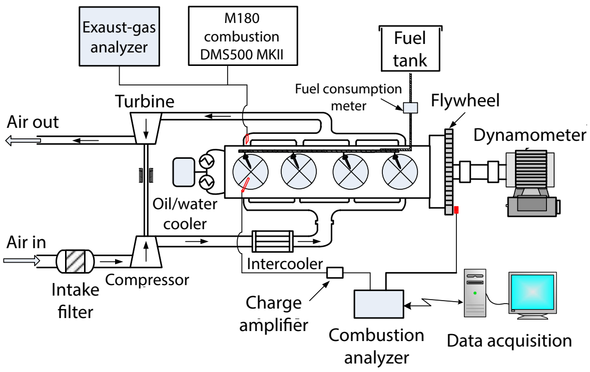


Figure 1. Schematic of engine and instrumentation set-up

Table 1. Engine specification

Engine type	Four-cylinder 4-stroke
Bore × Stroke, [mm]	96 × 103
Displacement volume, [L]	2.982
Compression ration	17.5
Rated power, [kW]	85
Rated speed, [rpm]	3200
Cooling type	Water-cooled
Injection system	Common rail

layout is shown in fig. 1 and specifications of the Diesel engine are listed in tab. 1. The speed and torque output of engine were controlled and changed by an eddy current dynamometer. An electronic control unit module was used to control and monitor the engine operating parameters, such as fuel injection timing and fuel injection quantity. In addition, high pressure cooled EGR system was employed, the EGR ratios could be effectively adjusted by the combined control of air-throttle valve and EGR valve, and the EGR ratios was calculated [15]:

$$\text{EGR\%} = \frac{(\text{CO}_2\%)_{\text{intake}}}{(\text{CO}_2\%)_{\text{exhaust}}} 100\% \tag{1}$$

The pressure signals were obtained by a Kistler 6025C pressure sensor mounted in the cylinder head. Then signals were transmitted through a charge amplifier to the CB-466 combustion analyzer, and finally the pressure curve was obtained. Based on the measured pressure curves, the heat release rate (HRR) was calculated according to the First thermodynamic law [16, 17]:

$$\frac{\text{d}Q_g}{\text{d}\varphi} = \frac{\text{d}Q_n}{\text{d}\varphi} + \frac{dQ_w}{\text{d}\varphi} \tag{2}$$

where the net HRR, $\text{d}Q_n/\text{d}\varphi$ was determined by traditional first law, eq. (3):

$$\frac{\text{d}Q_n}{\text{d}\varphi} = \frac{\gamma}{\gamma\text{-}1} p \frac{\text{d}V}{\text{d}\varphi} + \frac{1}{\gamma - 1} V \frac{\text{d}p}{\text{d}\varphi} \tag{3}$$

where $\text{d}Q_w/\text{d}\varphi$ is the heat loss rate, and it was obtained using eq. (4):

$$\frac{\text{d}Q_w}{\text{d}\varphi} = A_{ht} h_c \frac{T - Tw}{6n} \tag{4}$$

The engine cooling water and lubricating oil temperature were monitored by temperature sensors, and was controlled by a temperature control device. The intake-air temperature and pressure of the engine were controlled by an air conditioning system.

Gaseous pollutants (CO_2 and NO_x) and PM were sampled from the first engine exhaust manifold and measured by AVL DiGas 4000 gas analyzer and DMS500 MKII fast particulate spectrometer (Cambustion Ltd.), respectively. All emissions were measured during steady-state engine operation. The DMS500 classified the particles to be measured by the electrical mobility diameter from 5-1000 nm in 22 classes. It has been equipped with a fully-integrated two stage diluter. The first diluter worked at the sampling position to avoid water condensation and particle agglomeration, and the second one allowed aerosol sampling in a very wide range by a rotating disc. More details about the equipment can be found in [18].

In this study, PM emissions were investigated by DMS500 in terms of particle size distribution function (PSDF), particle number, and mass concentration. The PSDF is expressed as dN/dlogDp, where N is the particle number concentration and Dp is the particle diameter. The particle mass concentration measured by DMS500 was calculated from the soot density of 1.2 g/cm^3 [19]:

$$\text{Soot mass } (\mu g) = 1.53 \cdot 10^{-16} Dp^{3.19} \text{ nm} \tag{5}$$

Test fuels and experimental procedures

Conventional diesel and ethanol (99.9% purity) were used in this experiment study, with properties shown in tab. 2. In order to investigate the combustion and emissions at different mixing ratios, pure diesel was prepared as based fuel and two blends were prepared and denoted as E10 (90% diesel; 10% ethanol, mass basis) and E20 (80% diesel; 20% ethanol) for evaluation.

Experiments were carried out at 30% engine loads, corresponding to brake mean effective pressures (BMEP) of 0.38 MPa, and the engine was kept at 1800 rpm. In order to better investigate the effects of ethanol addition on combustion and emissions, the same thermal value in every cycle was input. When blending fuels with smaller low heating values, the test should increase the injection fuel quantity to ensure the same energy input. Meanwhile, the test was carried out under different fuel injection timings and EGR ratios: firstly, the EGR valve was kept closed and the start of injection (SOI) time was changed from 2.5 to 22.5 °CA bTDC at a 5 °CA increment. Secondly, the SOI fixed at 7.5 °CA bTDC, the EGR ratios were set at 0%, 9%, 25% and 27%. In order to ensure the reliability and repeatability of the test data, the engine was firstly warmed up to steady condition at each testing condition, keeping the cooling water temperature and the lubricating oil at 85 °C and 87 °C, respectively. After switching to a new blend, the engine was allowed to run for 15 min before data collection, to ensure the new fuel was not contaminated by the remnants of the old fuel. During the experiment, each measurement was repeated twenty times to guarantee the reliability of results. The uncertainties of the main measurements were summarized in tab. 3.

Table 2. Properties of diesel and ethanol

Parameters	Diesel	Ethanol
Chemical formula	C_{12}-C_{25}	C_2H_5OH
Cetane number	53	8-10
Oxygen content, [%]	0	34.7
Stoichiometric air/fuel ratio	14.3	8.98
Density at 15 °C, [kgcm^{-3}]	845	790
Latent heating, [kJkg^{-1}]	265	840
Lower heating value, [MJkg^{-1}]	42.6	27.2
Auto-ignition temperature, [°C]	180-220	434

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

Cylinder combustion characteristics

The cylinder pressure and the corresponding HRR for E10 at various injection timings and EGR ratios are presented in fig. 2. As shown in fig. 2(a), with the advance of injection timing from 2.5 °CA to 22.5 °CA bTDC, the peak values of cylinder pressure increased and appeared in advance. Advance of the injection timing means that the injected fuel has sufficient time to mix with the air, and there are more fuel and gas mixture in the cylinder when the ignition condition is reached. At the same time, the proportion of burned fuel increased during the compression stroke, which increases the peak value of cylinder pressure. When the injection timing at 2.5 °CA bTDC, most of the fuel burns after TDC in the power stroke, and the proportion of constant volume combustion decreased, resulting in the pressure curve shows a significant reduction.

With the EGR ratio increasing, the peak of the cylinder pressure decreased. Meanwhile, the peak of the cylinder pressure corresponding to the CA gradually deviates from the TDC, see fig. 2(b). The main reasons for this trend can be mainly described by the following aspects. On the one hand, with the increase of EGR, the oxygen concentration decreased, which prolonged the ignition delay time even decreased the combustion rate. At the same time, the inert gas in EGR has a retarding effect on the rate of chemical reaction, leading to a decrease in pressure peak. On the other hand, the inert gases make the specific heat capacity of the intake gas increase, eventually reducing the cylinder pressure.

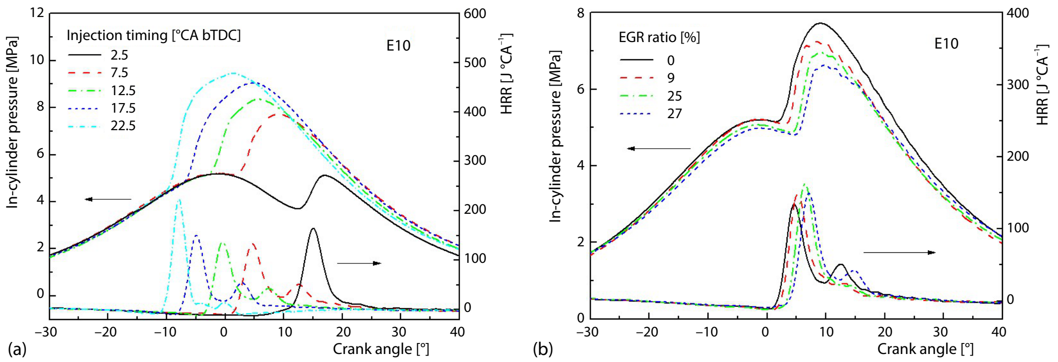


Figure 2. Effect of SOI and EGR ratio on cylinder pressure and HRR of E10; (a) injection timing, (b) the ERG ratio

Effects of ethanol addition on the cylinder pressure and HRR at SOI of 7.5 °CA bTDC with EGR ratios of 0% are shown in fig. 3. It was noted that ethanol addition causes a slight decrease in the cylinder pressure peak under this operating condition. It can be explained by the following reasons. At 0.38 MPa BMEP engine load, the temperature in the engine cylinder is relatively low, at the same time, due to the high latent heat of vaporization of ethanol, more heat will be absorbed for fuel evaporation by ethanol addition. In addition, because eth-

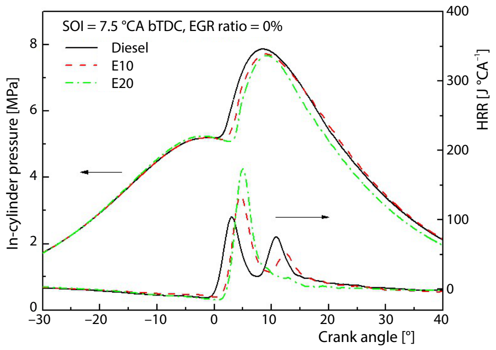


Figure 3. Effect of ethanol addition on cylinder pressure and HRR at SOI of 7.5 °CA bTDC

anol addition prolonged the ignition delay time, so a relative low temperature and pressure environment may be produced due to combustion happens away from TDC at the expansion stroke for E10 and E20.

Effects of ethanol addition on the ignition delay time and combustion duration time under different operating conditions are shown in fig. 4. It can be seen that ethanol addition prolongs the ignition delay and shortens the combustion duration. Because of lower cetane number, higher latent heat of vaporization and higher auto-ignition temperature of ethanol, the ignition delay period of diesel/ethanol is longer with respect to pure diesel. An extended ignition delay can promote the air/fuel mixing process and thus accelerate the premixed burning. Meanwhile, ethanol has lower viscosity, higher oxygen content and better volatility than that of diesel, so blended fuels are atomized and volatized more easily in the cylinder, which can also promote uniform air/fuel premixing and accelerate the combustion speed.

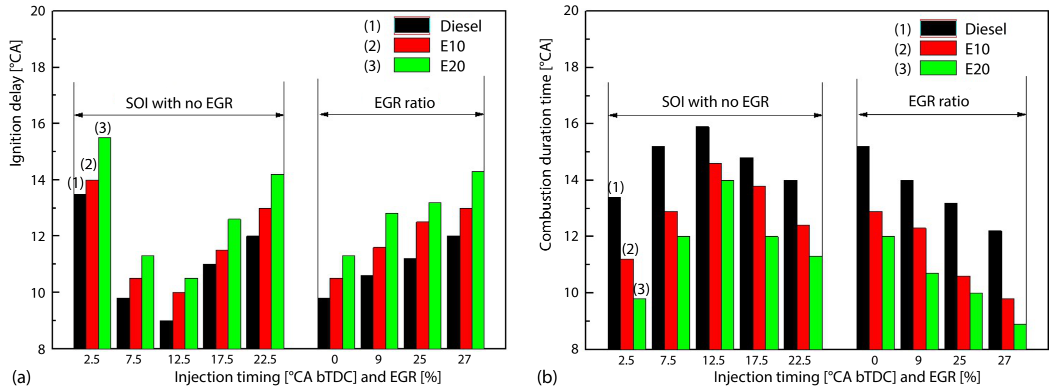


Figure 4. Effects of ethanol addition on ignition delay and combustion duration time; (a) ignition delay time (b) combustion duration time

The BTE and equivalent BSFC

Figure 5 presents the effects of ethanol addition on the BTE and diesel-equivalent equivalent BSFC for all blends. Compared with pure diesel, the addition of ethanol produces lower equivalent BSFC of Diesel engine and higher thermal efficiency. As previously mentioned, the combustion process can be improved after mixing ethanol, which burning more quickly and making the fuel consumption reduced. Though the heating value of the blend fuels decreased by

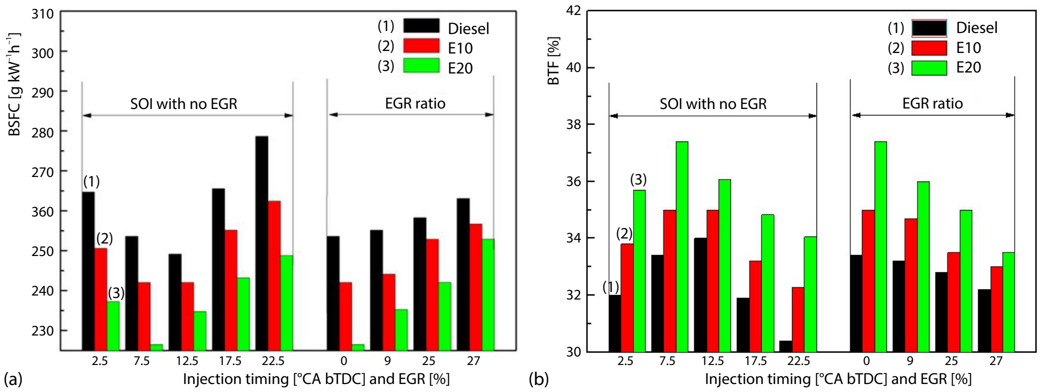


Figure 5. Effects of ethanol addition on BSFC and BTE; (a) BSFC (b) BTE

adding ethanol into diesel, the engine thermal efficiencies are improved. This can be explained that the addition of ethanol can provide additional fuel lubricity, reduce fuel viscosity, improve atomization, and provide more oxygen contents for improving the cylinder combustion process in converting fuel chemical energy into useful engine work.

The PM emission characteristics

Figure 6 shows the effects of SOI and EGR ratio on PSDF when engine fueled with E10. It can be seen that the particle number is less when the injection time is near the TDC. However, at SOI of 12.5 °CA bTDC, the particle number is higher. Soot particles are largely affected by the combustion process and are formed from incomplete burning in a locally rich region of the cylinder. The ignition delay is shortened when the fuel injection timing changes from 22.5-12.5 °CA bTDC. The premixed combustion is weakened while the diffusion combustion is strengthened, which promotes soot formation, so more large particles produced by coagulation and aggregation of small particles. Additionally, the lower cylinder temperature caused by the lower premixed combustion suppresses soot oxidation. However, a contrary tendency appears with the injection timing form 12.5-2.5 °CA bTDC, which can also be explained by the variation of ignition delay time.

It is clearly seen that the number concentration of large particles increases gradually as the EGR ratio increases, fig. 6(b), so the total particulate mass increased with the EGR ratios increase, see fig. 7. We know that there is a trade-off relationship between the NO_x and soot. Though emissions can be reduced by using the EGR technique, it is beneficial for soot formation and restraining soot oxidation. Meanwhile, higher amount of soot particles due to EGR induced can also accelerate the process of particles' coagulation, accumulation, condensation of volatile fractions on the particles. However, the number of smaller-size particles decreases slightly with higher EGR. This is because the increased large particles act as *sponges* for condensation or adsorption of volatile materials, and thereby suppress the soot nucleation. In addition, the use of high level EGR promote the coagulation among small size particles, so the small size particles decrease.

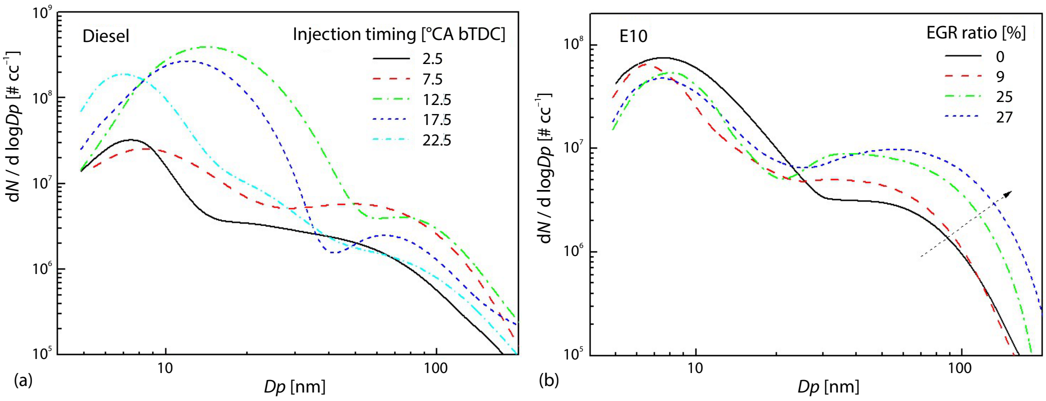


Figure 6. Effects of SOI and EGR ratio on PSDF for E10; (a) SOI (b) EGR ratio

Figure 7 is the effects of ethanol addition on the total particulate mass. The total particulate mass is decreased with ethanol addition, E10 and E20 produce mean reductions in total particulate mass of 42.56%, and 65.32%, respectively. Figure 8 shows the effects of ethanol addition on the PSD. The large particles are reduced and the small particles are increased with the ethanol addition whether in any SOI and EGR ratios. Several factors may lead to the decrease of soot precursors and the subsequent soot emission after the addition of ethanol [20]. Firstly, the diesel/ethanol blends has the longer ignition delay, higher volatility and larger oxygen content with compared to pure diesel, which can improve the air/fuel mixing process. So the combustion is promoted and the temperature during diffusion combustion stage is increased, which can reduce soot formation and promote soot oxidation. Secondly, aromatic ring growth and soot particle inception are inhibited by the abundant radicals (primarily OH), which strengthened by the addition of oxygenated fuel into diesel. Thirdly, the carbon content in the blended fuels declines with the increase of oxygen content, so the concentrations of C-C bonds (source of soot formation) in the blended fuel are reduced.

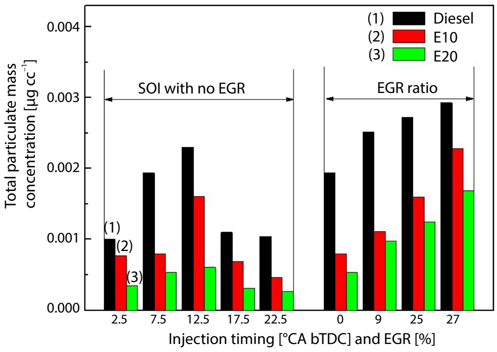


Figure 7. Effects of SOI and EGR on total particulate mass

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 ethanol, would dilute the contents of aromatics and sulfur in the blended fuels and thus reduce soot and PM emissions.

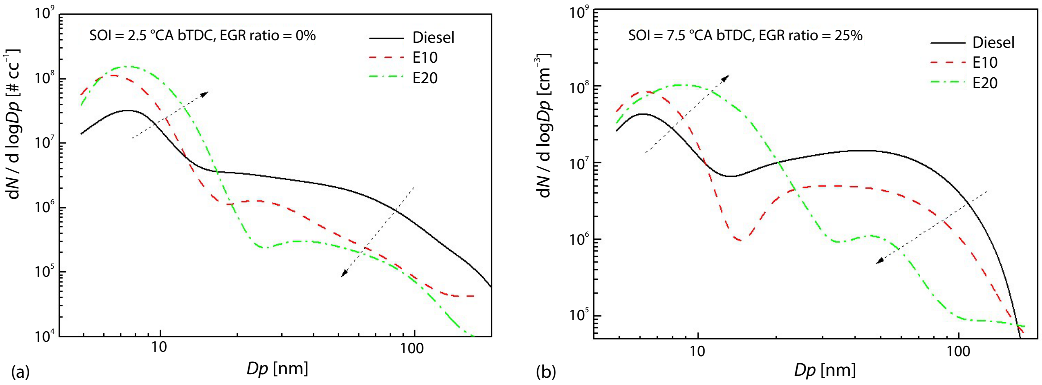
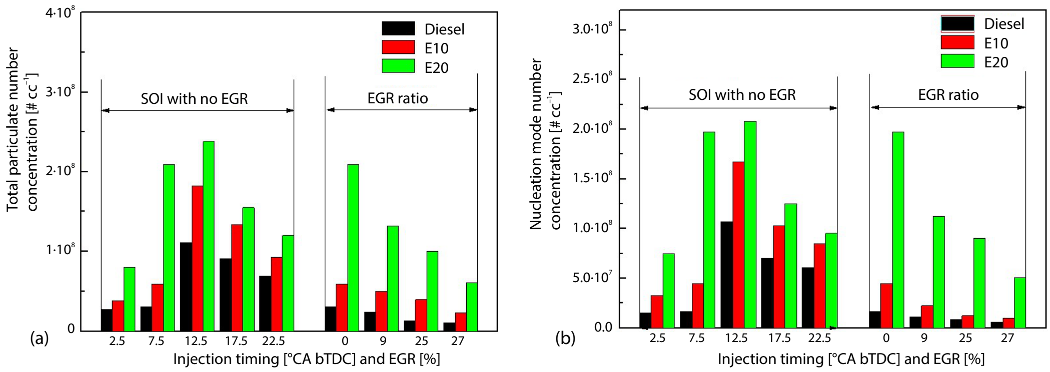


Figure 8. Effects of ethanol addition on PSDF in different SOI and EGR ratios, (a) SOI = 2.5, EGR = 0%, (b) SOI = 7.5, EGR = 25%

The soot nucleation process is retarded with the increase of ethanol addition, but more small-size particles are produced as previosly stated. The reasons for this trend may be attributed to three causes: soot formation process is suppressed by ethanol addition also slows down the coagulation and aggregation of particles to form larger soot particles, formation of smaller-size particles increased; due to the lower heat value than pure diesel, the blended fuels with ethanol addition will be consumed more, which also promotes particles formation; and the reduction of large-size particles relieves the absorption of volatile or semi-volatile substances and promotes the formation of primary particles.

Generally, PM emissions in engine exhaust are classified into the nucleation mode (NM, $Dp < 50$ nm) and the accumulation mode (AM, 50 nm $< Dp <$ 1000 nm). Figure 9 shows



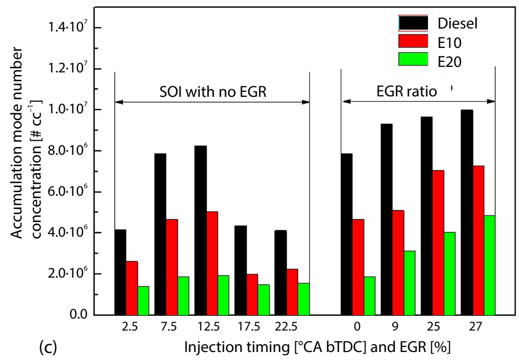


Figure 9. Effects of SOI and EGR on particulate number, nucleation mode number and accumulation mode number; (a) total particulate number (b) nucleation mode number (c) accumulation mode number

the effects of ethanol addition on total particulate number concentration (TNC), nucleation mode and accumulation mode number concentration (NMC and AMC) under different SOI and EGR ratios.

As shown in fig. 9, the TNC and NMC rise gradually and the AMC declines obviously with the ethanol blend ratios increase compared with diesel. For E10, the NMC and TNC have a mean increase by 89.75% and 90.11%, respectively. The AMC has a mean decrease by 39.58%. For E20, the NMC and TNC have a mean increase by 608.57% and 360.32%, respectively. The AMC has a mean decrease by 66.66%. As reported by Di *et al*. [21], with the amount of oxygenated fuel increased in blended fuel, the carbon content reduced and the oxygen content increased correspondingly, which lead to suppress the soot formation and reduce particle number. However, the processes of coagulation and agglomeration of the small particles will also be slowed down, which can increase the small particles. Moreover, more fuel will be consumed with the addition of ethanol due to the lower calorific value of ethanol, which also induces the small particles number increase. The effects of those factors may contribute to the results mentioned above.

Conclusions

- For pure diesel, E10 and E20, the ignition delay and BSFC decrease with delaying injection timing from 22.5 to 12.5 °CA bTDC, and then increase with the injection timing retarded more, while the combustion duration and BTE changed in an opposite trend correspondingly. Compared with pure diesel fuel, Diesel engine fueled with blending fuel (diesel/ethanol) prolong the ignition delay, shorten the combustion duration. Meanwhile, BTE increases and BSFC decreases with the rise of ethanol ratio under all combustion scenarios due to different fuel properties of blends.

- Injection timing has a great effect on PSDF due to different combustion characteristic, when the fuel injection timing change from 22.5 to 12.5 °CA bTDC, particulate number and mass concentrations are increased due to the premixed combustion is weakened while the diffusion combustion is strengthened, while a contrary tendency appears with the injection timing form 12.5 to 2.5 °CA bTDC, which can also be explained by the variation of ignition delay time. In addition, the number concentration of larger size particles and particulate mass concentration are increased with the increase of EGR ratio, but the number of smaller-size particles decreases slightly.
- Compared to pure diesel, E10 and E20 produce average increases in NMC of about 89.75%, and 608.57%, respectively, mean increment in TNC of 90.11% for E10 and 360.32% for E20 are observed. In addition, E10 and E20 produce mean reductions in AMC of 39.58%, and 66.66%, in total particulate mass of 42.56%, and 65.32%, respectively.

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
DI – direct injections
EGR – exhaust gas recirculation
NM – nucleation mode
NMC – nucleation mode concentration
PSD – particle size distribution
TNC – total particle number concentration

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