Exploring the potential benefits of Ethanol Direct Injection (EDI) timing and pressure on particulate emission characteristics in a Dual-Fuel Spark Ignition (DFSI) engine

Xiang Li a, b, *, Dayou Li a, Jingyin Liu c, *, Tahmina Ajmal a, Abdel Aitouche d, e, Raouf Mobasherï d, e, Oyuna Rybdylova f, Yiqiang Pei b, *, Zhijun Peng g, *

a School of Computer Science and Technology, University of Bedfordshire, Luton, UK

b State Key Laboratory of Engines, Tianjin University, Tianjin, China

c School of Chemistry and Chemical Engineering, Nantong University, Nantong, China

d Univ. Lille, CNRS, Centrale Lille, UMR 9189 - CRISTAL - Centre de Recherche en Informatique Signal et

Automatique de Lille, F-59000 Lille, France

e Junia, Smart Systems and Energies, F-59000 Lille, France

f Advanced Engineering Centre, School of Architecture, Technology and Engineering, University of Brighton,

Brighton, UK

g School of Engineering, University of Lincoln, Lincoln, UK

*Corresponding author:

Xiang Li, School of Computer Science and Technology, University of Bedfordshire, Luton, LU1 3JU, UK

Email: xiang.li@beds.ac.uk

Jingyin Liu, School of Chemistry and Chemical Engineering, Nantong, China.

Email: jingyin.liu@ntu.edu.cn

Yiqiang Pei, State Key Laboratory of Engines, Tianjin University, Tianjin, China.

Email: peiyq@tju.edu.cn
Abstract

Nowadays, particulate matter emitted by vehicles severely impacts environmental quality and human health. In this paper, the potential benefits of Ethanol Direct Injection (EDI) timing and pressure on particulate emission characteristics in a Dual-Fuel Spark Ignition (DFSI) engine were initially and systematically explored. The experimental results illustrate that by delaying EDI timing from -340 °CA to -300 °CA, there is a significant benefit in both particulate number and mass concentration. Furthermore, the size distribution curve of particulate number changes from bimodal to unimodal, meantime size distribution curves of particulate mass consistently concentrate on the accumulation mode. By increasing EDI pressure from 5.5 MPa to 18 MPa, the droplet size of ethanol spray can be effectively reduced. The benefit of increasing EDI pressure is more apparent in reducing particulate number is than particulate mass. The concentration of number and mass for total particulates have a reduction of 51.15% and 22.64%, respectively. In summary, it was demonstrated that an appropriate EDI timing or high EDI pressure could be a practical and efficient way to reduce particulate emissions in a DFSI engine.

Keywords

Dual-Fuel Spark Ignition (DFSI) engine; Particulate emissions; Ethanol; Injection timing; Injection pressure
1. Introduction

Over the last decade, the impact of environmental pollution has been a global problem. One of the major pollutants is particulate matter emitted by vehicles, which would adversely affect regional air quality, climate change and human health, particularly cardiorespiratory diseases [1]. Currently, some novel vehicle powertrain technologies without Internal Combustion Engine (ICE) have been developed to reduce engine emissions, such as Battery Electric Vehicle (BEV) [2], Fuel Cell Electric Vehicle (FCEV) [3][4], Oxy-Fuel Combustion-Carbon Capture and Storage (OFC-CCS) system [5]. However, the popularisation rate of these technologies is normally subject to cost, cruising range, charging time and relevant supporting facilities [6][7]. Hence, particulate emissions from the ICE of ICE-only vehicle, Hybrid Electric Vehicle (HEV) and Plug-in Hybrid Electric Vehicle (PHEV) have attracted increasing attention from scholars in various fields [8][9].

With the advantages of superior volumetric efficiency, thermal efficiency, power output, and transient response, Gasoline Direct Injection (GDI) engine has become an increasingly prevalent option for ICE and vehicle manufacturers [10][11]. However, compared to Port Fuel Injection (PFI) engine, particulate emissions of GDI engine are usually higher owing to higher spray impingement possibility and shorter air-fuel mixing time [12]. Moreover, the regulations for particulate emissions of GDI-powered vehicles have become more stringent in recent years. For example, in 2014 and 2017, Euro 6b and Euro 6c standards limit the particulate number of GDI-powered vehicles to $6 \times 10^{12}$/km and $6 \times 10^{11}$/km, respectively.

In order to reduce particulate emissions and keep the engine power performance, Dual-Fuel Spark Ignition (DFSI) engine has been developed with the advantages of multiple fuel injection modes and fuel properties [13]. Due to higher heating value and lower vaporisation latent heat, gasoline can be used to contribute a better transient response, especially during engine cold start. As renewable fuels
with low carbon and high oxygen content, alcohol fuel can be utilised to reduce engine particulate emissions and improve anti-knock performance by the advantage of higher octane.

The important research findings relevant to Dual-Fuel Spark Ignition (DFSI) engines are summarised in Table 1. Daniel et al. [14] found that with the advantages of fuel injection flexibility, Dual-Fuel Spark Ignition (DFSI) is very beneficial to optimise engine gaseous and particulate emissions with changes in engine operating conditions. Kim et al. [15] demonstrated that both reduction of particulate emissions and knock frequency could be achieved when ethanol port injection is added. Liu et al. [16][17] compared different alcohol–gasoline and gasoline–alcohol injection modes from a DFSI combustion engine. It was found that for selected engine operating conditions, there is an optimal mass fraction for alcohol injection to optimise particulate matter emissions and simultaneously keep fuel economy and power output. Catapano et al. [18] observed particulate formation and emissions in an optical small DFSI engine fuelled with Compressed Natural Gas (CNG) and gasoline. It was demonstrated that there is a benefit in reducing particulate emissions owing to the gaseous properties of CNG. Kang et al. [19] systematically compared the effects of different injection modes on combustion and knock suppression characteristics. Yu et al. [20][21] conducted experimental studies about combustion and emissions in an SI engine, with ethanol/gasoline and hydrous ethanol/gasoline dual-fuel injection modes. The studies concluded that the synergistic effects of utilising ethanol injection and Exhaust Gas Recirculation (EGR) could effectively reduce gas and particulate emissions. Furthermore, particulate size can be reduced by increasing the water ratio in hydrous ethanol. Zhuang et al. [22][23] investigated the effects of different ethanol Direct Injection (DI) timings on air-fuel mixture formation, combustion process, knock mitigation, Nitrogen Oxide (NO) and Hydrocarbon (HC) emissions from a DFSI engine. It was indicated that ethanol DI timing strongly influences the air-fuel mixture process and quality. Moreover, with the advance of ethanol
As mentioned above, previous studies on the DFSI engine have proposed and investigated some effective solutions to reduce engine emissions. However, regarding the fuel injection strategies of DFSI engines, research findings mainly focused on the influence of fuel injection ratios and timings on gaseous emissions. Besides, the effects of ethanol injection ratio on particulate emissions are another existing hot topic. However, almost no systematic study on the effects of Ethanol Direct Injection (EDI) timing and pressure on particulate emissions in a DFSI engine was reported.

### Table 1. Important research findings concerning DFSI engines

<table>
<thead>
<tr>
<th>Publication Year</th>
<th>Key Advances</th>
<th>Fuel</th>
<th>Main Authors</th>
</tr>
</thead>
<tbody>
<tr>
<td>2013</td>
<td>Particulate emissions were investigated for ethanol injection under both dual-fuel injection and DI strategies.</td>
<td>Ethanol; Gasoline</td>
<td>Daniel et al. [14]</td>
</tr>
<tr>
<td>2014</td>
<td>Effects of ethanol injection timing on knock mitigation, lean-burn, NO and HC emissions were investigated from an SI engine with PFI-gasoline and DI-ethanol.</td>
<td>Ethanol; Gasoline</td>
<td>Zhuang et al. [22]</td>
</tr>
<tr>
<td>2015</td>
<td>Particulate emissions were investigated in an SI engine with PFI-ethanol and DI-gasoline with varying engine compression ratios and ethanol injection timings.</td>
<td>Ethanol; Gasoline</td>
<td>Kim et al. [15]</td>
</tr>
<tr>
<td>2015</td>
<td>The alcohol–gasoline and gasoline–alcohol DFSI combustion was compared for particulate reduction and fuel economy with varying alcohol mass fractions.</td>
<td>Methanol; Ethanol; Gasoline</td>
<td>Liu et al. [16][17]</td>
</tr>
<tr>
<td>2017</td>
<td>Particulate emissions were investigated in an optical small DFSI engine with DI-CNG and PFI-gasoline.</td>
<td>CNG; Gasoline</td>
<td>Catapano et al. [18]</td>
</tr>
<tr>
<td>2019</td>
<td>Effects of fuel injection modes on knock suppression were compared and studied under different injection modes on a single-cylinder SI engine.</td>
<td>Ethanol; Gasoline</td>
<td>Kang et al. [19]</td>
</tr>
<tr>
<td>2020</td>
<td>Effects of ethanol injection strategies on air-fuel mixture formation and combustion process were investigated from an SI engine with PFI-gasoline and DI-ethanol.</td>
<td>Ethanol; Gasoline</td>
<td>Zhuang et al. [23]</td>
</tr>
<tr>
<td>2021</td>
<td>The combustion and emissions were investigated with varying access air ratios, ethanol direct injection ratios and exhaust recirculation ratios from an SI engine with PFI-gasoline and DI-ethanol.</td>
<td>Ethanol; Gasoline</td>
<td>Yu et al. [20]</td>
</tr>
<tr>
<td>2021</td>
<td>Effects of different water ratios in hydrous ethanol on combustion and emissions were investigated from an SI engine with PFI-gasoline and DI-hydrous ethanol.</td>
<td>Ethanol; Gasoline</td>
<td>Yu et al. [21]</td>
</tr>
</tbody>
</table>
Hence, the study presented in this paper concentrates on exploring the benefits of EDI timing and pressure on particulate emission characteristics from a DFSI engine. The findings of this study would help not only fill the gap of exploring EDI strategy on the reduction of particulate, but also provides a fresh and practical way towards controlling the particulate problem of different kinds of vehicles.

2. Experimental methodology

2.1. Experimental testbed and procedure

The experimental study was performed on a dual-injection DFSI engine with the specifications shown in Table 2. It is an advanced four-cylinder turbocharged engine with a displacement of 2.0-litre and a compression ratio of 9.6. The fuel properties of gasoline and ethanol used in this study are both presented in Table 3 [24]. Furthermore, as shown in Fig. 1, gasoline and ethanol are injected via PFI injectors and DI injectors, respectively.

<table>
<thead>
<tr>
<th>Table 2. Engine specifications</th>
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<tbody>
<tr>
<td>Items</td>
</tr>
<tr>
<td>Engine type</td>
</tr>
<tr>
<td>Bore × Stroke (mm)</td>
</tr>
<tr>
<td>Displacement (L)</td>
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<tr>
<td>Injection mode</td>
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<tr>
<td>Intake mode</td>
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<tr>
<td>Compression ratio</td>
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<tr>
<td>Rated speed (rpm)</td>
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<tr>
<td>Rated power (kW)</td>
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<tr>
<td>Maximum Torque (N·m)</td>
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<tr>
<th>Table 3. Fuel properties [24]</th>
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<tbody>
<tr>
<td>Fuel type</td>
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<tr>
<td>Chemical formula</td>
</tr>
<tr>
<td>Relative molecular mass</td>
</tr>
<tr>
<td>Gravimetric oxygen content (%)</td>
</tr>
<tr>
<td>Research octane number</td>
</tr>
<tr>
<td>Property</td>
</tr>
<tr>
<td>--------------------------------</td>
</tr>
<tr>
<td>Density (20 °C) (kg/L)</td>
</tr>
<tr>
<td>Dynamic viscosity (20 °C) (mPa·s)</td>
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<tr>
<td>Kinematic viscosity (20 °C) (mm²/s)</td>
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<tr>
<td>Surface tension (20 °C) (N/m)</td>
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<tr>
<td>Boiling range (°C)</td>
</tr>
<tr>
<td>Low heating value (kJ/kg)</td>
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<tr>
<td>Latent heat of vaporisation (kJ/kg)</td>
</tr>
<tr>
<td>Laminar flame speed (20 °C) (m/s)</td>
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<tr>
<td>Stoichiometric air-fuel ratio</td>
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</table>

**Fig. 1.** Schematic view of GPI plus EDI dual-injection system

Fig. 2 shows the schematic view of engine testbed. By using a programmable Electronic Control Unit (ECU) and calibration software (INCA), an electrical dynamometer can measure and control engine speed, torque and power output in real-time. Combustion characteristics can be calculated by the transient signals of in-cylinder pressure, which were recorded and analysed via a combustion measurement platform, including four high-precision spark-plug pressure sensors (AVL-GH13Z), an encoder (Kistler-2614CK1), charge amplifiers (Kistler-5018A) and a combustion analyser (AVL-641). In addition, the emission of particulates ranging from 5 nm to 1000 nm was measured by a fast particulate analyser (Cambustion-DMS 500) connected to a sampling point in front of the engine’s three-way catalytic converter. In this study, the engine operated at the speed of 2000 revolutions per minute (rpm) and a typical low load of 2 bar Brake Mean Effective Pressure (BMEP). In order to
make the research process more efficient, the fuel injection mass ratio of PFI to DI was fixed at 1:1, representing 50% gasoline-PFI plus 50% ethanol-DI.

![Fig. 2. Schematic view of engine testbed](image)

In order to assure the accuracy of experimental results, Maximum Brake Torque (MBT) spark timings were applied to all the engine operating conditions. The lambda, intercooler outlet temperature and coolant were maintained at 1 ± 0.01, 298 ± 2 K and 360 ± 2 K, respectively. The combustion characteristics and particulate emissions data for each engine operating condition should be recorded after the engine stabilises for two minutes. Furthermore, to minimise the impacts of cycle-to-cycle variations, cylinder pressure result was averaged based on two hundred consecutive engine cycles. Meanwhile, the result of particulate emissions was obtained from the average of repeated
measurements three times. Table 4 presents the uncertainties of some key parameters calculated by Holman's root mean square method [25].

Table 4. Uncertainties of measured parameters

<table>
<thead>
<tr>
<th>Measured Parameters</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>± 0.1</td>
</tr>
<tr>
<td>BMEP</td>
<td>± 0.1</td>
</tr>
<tr>
<td>BSFC</td>
<td>± 0.2</td>
</tr>
<tr>
<td>Pressure</td>
<td>± 0.1</td>
</tr>
<tr>
<td>Crank angle</td>
<td>± 0.1</td>
</tr>
<tr>
<td>Lambda</td>
<td>± 0.3</td>
</tr>
<tr>
<td>Coolant temperature</td>
<td>± 0.4</td>
</tr>
<tr>
<td>Intercooler output temperature</td>
<td>± 0.4</td>
</tr>
<tr>
<td>Particulate number</td>
<td>± 1.7</td>
</tr>
</tbody>
</table>

In order to make this research more comprehensive, the effects of EDI pressure on the droplet diameter of ethanol spray from the engine’s DI injector were studied via microscopic spray characteristics investigation. Fig. 3 shows the Phase Doppler Particles Analyser (PDPA) system for this investigation. With the advantage of an argon-ion laser, a 180 MHz high-frequency signal processor and some other accessories, the PDPA system can provide accurate measurement results of droplet diameter with a high resolution of 0.1 μm and a range from 0 to 236 μm. In order to match with engine experiments, EDI pressure was set to be 5.5 MPa, 10 MPa, 14 MPa and 18 MPa in this test. Furthermore, ethanol was injected into an ambient condition, which is 293 ± 0.5 K and 0.1 MPa. An air extractor was utilised to help ensure the safety of experimental site.

Besides, as shown in Fig. 4, according to the Society of Automotive Engineers (SAE) J2715 standard [26], measurement points are selected at 50 mm downstream from the axial direction of the nozzle. For minimising the interference of suspended fuel droplets of the latest injection, the ethanol injection pulse and injection width were set to 0.1 Hz and 1.2 ms, respectively. To ensure
measurement accuracy, 20000 validated sample droplets were collected for each experimental condition, and the collection should be repeated three times.

2.2. Key parameters in this study

In this study, some key parameters are introduced and defined to help better understand the characteristics of combustion and particulate emissions. $t_{EI}$ and $P_{EI}$ denote EDI timing and EDI pressure, respectively. $\theta_F$ denotes ignition delay, representing Crank Angle (CA) interval between
spark timing and $\varphi_{CA10}$ (where 10% of cumulative heat has released). $\theta_c$ denotes combustion duration, representing CA interval between $\varphi_{CA10}$ and $\varphi_{CA90}$ (where 90% of cumulative heat has released). $\varphi_{CA50}$ denotes the CA where 50% of cumulative heat has released. $R_M$ and $T_M$ denote maximum heat release rate and maximum in-cylinder temperature, respectively. $\eta_B$ denotes engine brake thermal efficiency, as shown in Equation (1).

$$\eta_B = \frac{p_B}{M_G \times H_G + M_E \times H_E} \times 100\%$$ (1)

Here, $p_B$ is engine brake power. $M_G$ and $M_E$ are the mass flow rate of gasoline and ethanol, respectively. $M_G$ and $M_E$ are the mass flow rate of gasoline and ethanol, respectively. $H_G$ and $H_E$ are the low heating value of gasoline and ethanol, respectively.

Regarding the parameters of particulate emissions, $D_p$, $N_p$ and $M_p$ denote particulate diameter, particulate number and particulate mass, respectively. For each particulate, the calculation of converting $D_p$ to $M_p$ is shown in Equation (2) [27][28].

$$M_p(\mu g) = 1.72 \times 10^{-15} \times D_p^{2.65}(nm)$$ (2)

Besides, in the PDPA test, $D_d$ denotes diameter of droplet; $D_{SMD}$ denotes Sauter mean diameter of droplets; $D_{sub}$ denotes the difference of $D_{SMD}$ between $P_I = 5$ MPa and other $P_I$ (10 MPa, 14 MPa and 18 MPa) conditions.

3. Results and discussion

3.1. Optimising engine particulate emission characteristics by changing EDI timing

In the section, the characteristics of particulate emission are investigated by changing $t_i$ from -340°C to -280°C. Meantime, $P_I$ is fixed at base value, which is 5.5 MPa.

Fig. 5 shows the effects of $t_i$ on $\theta_F$, $\theta_c$, $\varphi_{CA50}$ and $\eta_B$. On the whole, combustion phasing characterised by $\theta_F$, $\theta_c$ and $\varphi_{CA50}$ is not significantly affected by $t_i$, but some key features can also be observed.
With the delay of $t_i$ from -340 °CA to -280 °CA, $\theta_f$ slightly increases from 27.06 degrees to 27.37 degrees, meantime $\theta_c$ has a reduction of 0.68 degrees. Besides, the variation of $\varphi_{CA50}$ is generally stable with the delay of $t_i$. As a result, $\eta_B$ shows a slight improvement of 0.21 percent. This can be attributed to a combined effect of three factors. First, as the earliest injection condition ($t_i = -340$ °CA) is very near to Top Dead Centre (TDC), it is easy to cause fuel impingement on the piston crown, slowing down the rate of fuel vaporisation. Hence, delaying $t_i$ to -280 °CA would help reduce the possibility of fuel impingement, which is beneficial to promote air-fuel mixture. Second, under $t_i = -280$ °CA, $\theta_c$ becomes shorter which helps reduce the heat transfer to combustion chamber walls. Third, by delaying $t_i$, the homogeneity of air-fuel mixture would be reduced owing to a shorter mixing time. It would lead to a negative impact on $\eta_B$, but it cannot offset the first two benefits.

Fig. 6 and Fig. 7 show the effects of $t_i$ on cylinder pressure and heat release rate, respectively. It is demonstrated that there is no obvious change in the curves of cylinder pressure and heat release rate by changing $t_i$. The main feature is that with the delay of $t_i$ from -340 °CA to -280 °CA, there is a slight rise in the peak of the curve. Fig. 8 presents the effects of $t_i$ on $R_M$ and $T_M$. It can be seen that with the delay of $t_i$, $R_M$ shows a general gradual increase of 0.56 J/CA. In the meantime, $T_M$ is a bit lower when $t_i$ is -320 °CA and -300 °CA, but it keeps around 2575 K on the whole, which indicates that the probability of particulate oxidation is not greatly affected by the variation of $T_M$ under different conditions of $t_i$. 
Fig. 5. Effects of $t_I$ on $\theta_F$, $\theta_C$, $\varphi_{CA50}$ and $\eta_B$

Fig. 6. Effects of $t_I$ on cylinder pressure
Fig. 7. Effects of $t_I$ on heat release rate

Fig. 8. Effects of $t_I$ on $R_M$ and $T_M$

Fig. 9 shows the effects of $t_I$ on $N_p$ size distribution. It can be seen that the $N_p$ size distribution is quite sensitive to $t_I$. With $t_I$ from -340 °CA to -320 °CA, the peak of curve for nucleation mode decreases from $1.27 \times 10^6$ to $8.36 \times 10^5$, whilst the peak of accumulation mode decreases from $1.48 \times 10^6$ to $6.45 \times 10^5$. By delaying $t_I$ to -300 °CA, the curve changes from bimodal to unimodal. However, it would increase again and change to be bimodal again with the further delay of $t_I$ to -280 °CA. This is mainly because when $t_I$ is -340 °CA, the piston has just passed through
the TDC. The spray impingement possibilities will be significantly increased, as the fuel injector is relatively close to piston crown. When \( t_i \) is -280 °CA, heterogeneous mixture will also be enhanced by less time between \( t_i \) and spark timing compared to that of \( t_i = -300 \) °CA. Besides, Hydrogen-Abstraction-Acetylene-Addition (HACA) growth rates would be increased with a higher \( T_M \) when \( t_i \) is -340 °CA and -280 °CA [29][30]. Thus, the quality of air-fuel mixture has deteriorated, leading to higher emissions of \( N_p \).

Fig. 10 shows the effects of \( t_i \) on \( M_p \) size distribution. Regardless of \( t_i \), \( M_p \) size distribution almost concentrates on the accumulation mode, meantime \( M_p \) of nucleation mode is very low. Besides, the highest \( M_p \) curve can be seen under the condition of \( t_i = -340 \) °CA due to the largest \( N_p \) of accumulation mode under this condition.

Fig. 9. Effects of \( t_i \) on \( N_p \) size distribution
In order to better understand the effects of $t_I$ on particulate emissions at a macroscopic level, Fig. 11 and Fig. 12 present the $N_P$ and $M_P$ concentrations with varying $t_I$, respectively.

It can be seen that both $N_P$ and $M_P$ concentrations are very sensitive to changing $t_I$. On the whole, an appropriate $t_I$ is very helpful to decrease particulate emissions. By delaying $t_I$ from -340 °CA to -300 °CA, there is a significant reduction of 54.65 % and 89.15% in $N_P$ concentration and $M_P$ concentration of total particulates, respectively. But the immediate cause of reduction for $N_P$ concentration is not very similar to that of $M_P$. Under $t_I = -300$ °CA, although $N_P$ of nucleation mode is a bit higher than that of $t_I = -320$ °CA owing to the less air-fuel mixture time, $N_P$ concentration of total particulates is still lower under $t_I = -300$ °CA by the significant reduction in accumulation particulates. Regarding the $M_P$ concentration, $M_P$ of nucleation mode can be neglected, so $M_P$ concentration of total particulates is closely related to accumulation mode.
Fig. 11. Effects of $t_I$ on $N_p$ concentration for nucleation, accumulation and total particulates.
Fig. 12. Effects of $t_I$ on $M_p$ concentration for nucleation, accumulation and total particulates

3.2. Optimising engine particulate emission characteristics by EDI pressure

This section mainly focuses on the experimental results by changing $P_I$ from 5.5 MPa to 18 MPa, which covers the $P_I$ common range of most commercial GDI and DFSI engines. Moreover, $t_I$ is fixed at -300 °CA in the meantime.

Fig. 13 presents the effects of $P_I$ on $p_d$ of $D_d$ at (0, 50). It can be seen that with the increase of $P_I$ from 5.5 MPa to 18 MPa, a steady decline can be found in the $p_d$ of large droplets which $D_d$ is more than 20 µm. Furthermore, the concentration of $D_d$ moves to smaller size droplets. The position and $p_d$ of curve’s peak respectively change to 6 µm and 14.8% under $P_I = 18$ MPa, while the corresponding values are respectively 10 µm and 11.55% under $P_I = 5.5$ MPa.

Regarding the whole view of droplet size at 50 mm of jet downstream, Fig. 14 shows the effects of $P_I$ on both $D_{SMD}$ and $D_{Sub}$ for different locations. It was found that increasing $P_I$ can effectively decrease $D_{SMD}$ regardless of the locations, which would promote the progress of secondary atomisation, evaporation and air-fuel mixing. Besides, the comparison of $D_{sub}$ of
Different $P_I$ denotes that a reduction of around 1.7 µm can be observed for every 4 MPa increase of $P_I$. Another interesting observation is that $D_{SMD}$ of (-12, 50) and (12, 50) is larger than locations. This explanation can be that the bulk of the spray breaks into filaments and droplets during the primary atomisation, increasing the collision probability between droplets inside the spray boundary. In the meantime, the breakup rate is increased by the shearing force outside the spray boundary, reducing the $D_{SMD}$ of (-16, 50) and (16, 50). As a result, the curves present a general distribution of bimodal under all conditions of $P_I$.

**Fig. 13.** Effects of $P_I$ on $p_d$ of $D_d$ at (0, 50)
Fig. 14. Effects of $P_I$ on $D_{SMD}$ and $D_{Sub}$ at 50 mm of jet downstream

Fig. 15 presents the effects of $P_I$ on $\theta_F$, $\theta_C$, $\varphi_{CAS0}$ and $\eta_B$. It can be seen that increasing $P_I$ could reduce the combustion duration, even though the effect is not apparent. With the increase of $P_I$ from 5.5 MPa to 18 MPa, $\theta_F$ and $\theta_C$ each reduces 0.72 degrees and 0.47 degrees, meanwhile $\varphi_{CAS0}$ slightly advances from 8.63 °CA to 8.51 °CA. This is because the air-fuel mixture quality is improved with the reduction of $D_{SMD}$ owing to higher $P_I$. A shorter combustion duration would mitigate the waste heat between high-temperature gases and in-cylinder wall, leading to a benefit of 0.13 percent in $\eta_B$. Fig. 16 and Fig. 17 present the variations of cylinder pressure and heat release rate with varying $P_I$. On the whole, both of them are not significantly affected by $P_I$. The main characteristic is that a slight advance and improvement can be observed for the curve’s peak by increasing $P_I$. Fig. 18 further shows that there is a gradual increase of $R_M$ and $T_M$ with the increase of $P_I$, which could help promote fuel burn rate, leading to a benefit in $\eta_B$. 
Fig. 15. Effects of $P_I$ on $\theta_F$, $\theta_C$, $\varphi_{CA50}$ and $\eta_B$

Fig. 16. Effects of $P_I$ on cylinder pressure
Fig. 17. Effects of $P_I$ on heat release rate

Fig. 18. Effects of $P_I$ on $R_M$ and $T_M$

Fig. 19 and Fig. 20 show the effects of $P_I$ on size distribution of $N_p$ and $M_p$, respectively. It can be observed that every $N_p$ curve presents an approximate unimodal distribution with a peak located around 17 nm nucleation mode. Furthermore, by increasing $P_I$ from 5.5 MPa to 18 MPa, the peak of $N_p$ curve gradually reduces from $9.83 \times 10^5$ to $4.66 \times 10^5$. Regarding the trend of $M_p$, it demonstrates that increasing $P_I$ has a positive effect on $M_p$ reduction, but the benefit is not very obvious. The curve shape of $M_p$ size distribution remains stable, and the curve peak concentrates...
around 300 nm $D_p$. Besides, because a large nucleation particulate is much heavier than nucleation particulates, an increment can be seen for $D_p$ more than 600 nm in the $M_p$ curves. But due to the very small $N_p$, the absolute value of $M_p$ is still very low, which is less than 70 at 1000 nm.

Fig. 21 and Fig. 22 can visually explain the effects of $P_I$ on $N_p$ and $M_p$ in a macroscopic view. With the increase of $P_I$ from 5.5 MPa to 18 MPa, the concentration of $N_p$ and $M_p$ for total particulates decrease by 51.15% and 22.64%, respectively. The reduction of $M_p$ is far less than that of $N_p$. Moreover, $M_p$ concentration of total particulates is closely related to that of accumulation mode, which presents a gradual decline trend with increased $P_I$. This is because regardless of $P_I$, EDI mode would have the appearance of spray impingement, which causes the heterogeneous mixture around piston crown region. Although higher $P_I$ is helpful to air-fuel mixture quality, the heterogeneous mixture by spray impingement and pool fires cannot be entirely avoided, which is a dominant source of accumulation mode particulates. Besides, Fig. 18 has demonstrated that higher in-cylinder temperature can be seen with increased $P_I$, which would further gain the advantage of promoting particulate oxidation by ethanol as an oxygenated fuel, providing a benefit in the reduction of $N_p$ and $M_p$.

Fig. 19. Effects of $P_I$ on $N_p$ size distribution
Fig. 20. Effects of $P_I$ on $M_P$ size distribution

Fig. 21. Effects of $P_I$ on $N_P$ concentration for nucleation, accumulation and total particulates
4. Conclusions

In order to develop more environmentally friendly vehicles, particulate emissions from ICE have been a serious problem to solve. The importance of this study is that the potential benefits of EDI timing and pressure on particulate emissions are systematically explored in a DFSI engine under PFI-gasoline and DI-ethanol mode. The findings can offer some original and fresh insights into the implementation of ethanol combined with the injection strategy of controlling particulate emissions regarding the DFSI engine. The main results of this study can be drawn as follows.

(1) With the delay of $t_i$ from -340 °CA to -280 °CA, $\theta_c$ has a reduction of 0.68 degrees. The variations of $\theta_F$ and $\varphi_{CA50}$ are generally stable. In the meantime, a slight improvement of 0.21 percent in $\eta_B$ can be achieved.

(2) It is a highly effective way to optimise engine particulate emission characteristics by changing $t_i$. By delaying $t_i$ from -340 °CA to -300 °CA, there is a significant reduction of 54.65 %
and 89.15% in $N_p$ concentration and $M_p$ concentration of total particulates, respectively. Furthermore, the curve of $N_p$ size distribution changes from bimodal to unimodal. Under all conditions, $M_p$ size distribution almost concentrates on the accumulation mode.

(3) Increasing $P_l$ would promote the progress of secondary atomisation, evaporation and air-fuel mixing for the ethanol spray. By increasing $P_l$ from 5.5 MPa to 18 MPa, the position and $p_d$ of curve’s peak respectively change to 6 µm and 14.8% from 10 µm and 11.55%. $D_{SMD}$ can be effectively reduced.

(4) By increasing $P_l$ from 5.5 MPa to 18 MPa, the gradual increase of $R_M$ and $T_M$ could help promote fuel burn rate. $\theta_F$ and $\theta_C$ each reduce 0.72 degrees and 0.47 degrees, meanwhile $\varphi_{CAS0}$ slightly advances from 8.63 °CA to 8.51 °CA, leading to a benefit of 0.13 percent in $\eta_B$.

(5) Regardless of $P_l$, $N_p$ curve presents an approximate unimodal distribution with a peak located around 17 nm nucleation mode. With the increase of $P_l$ to 18 MPa, it is more apparent in the reduction of $N_p$ is than that of $M_p$. The concentration of $N_p$ and $M_p$ for total particulates decrease by 51.15% and 22.64%, respectively.

CRediT authorship contribution statement

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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