Optimising microscopic spray characteristics and particle emissions in a dual-injection Spark Ignition (SI) engine by changing GDI injection pressure

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Abstract

Regarding reducing particle emissions from dual-injection spark ignition engines, most of the existing research focused on the benefits of using alcohol fuels. However, a comprehensive study of the effects of fuel injection pressure on microscopic spray characteristics and particle emissions in dual-injection spark ignition engines fuelled with gasoline has not been reported before. In this paper, with the assistance of phase Doppler particles analyser system and fast particle analyser, a study of optimising microscopic spray characteristics and particle emissions in a dual-injection spark ignition engine fuelled with gasoline by changing GDI injection pressure was conducted. The results show that by
increasing injection pressure from 5.5 MPa to 18 MPa, both normal and tangential components of droplet velocity increase, but the possibility of spray impingement would not increase a lot. Higher injection pressure would increase the probability of small droplets, and more droplets would collapse with a mode of continuous ripping or break down abruptly. From jet’s central axis to sides, Sauter mean diameter increases first, then reduces outside the spray boundary. Increasing injection pressure from 5.5 MPa to 18 MPa reduces total particle number concentration, which is 53.98% and 45.44% at 2 bar and 10 bar, respectively. Meanwhile, the peak of particle number distribution curve decreases from $3.01 \times 10^6$ to $1.43 \times 10^6$ at 2 bar, whilst reducing from $1.08 \times 10^6$ to $5.33 \times 10^5$ at 10 bar. Overall, this paper comprehensively analyses the effects of fuel injection pressure on microscopic spray characteristics and particle emissions, whilst offering a practical approach to reduce particle emissions in dual-injection SI engines fuelled with gasoline.

Keywords

Dual-injection Spark Ignition (SI) engine; Particle emissions; Microscopic spray characteristics; Injection pressure

1. Introduction

Gasoline Direct Injection (GDI) engines have been a preferred selection by the automotive industry in recent years due to the advantages of improved fuel economy, transient response and power performance [1-8]. However, particles emitted by GDI engines not equipped with Gasoline Particulate Filter (GPF) are recognised higher than that of Port Fuel Injection (PFI) engines and some diesel engines with the Diesel Particulate Filter (DPF) [9-14].

Nowadays, particle emissions from vehicles powered by GDI engines negatively affecting air
quality and human health have attracted more attention from researchers [15][16]. The emission standards have also been more stringent for the restrictions of particle emissions [10][11][14]. In 2014, Euro 6b standard released and first limited the Particle Number (PN) emissions from vehicles powered by GDI engines to $6 \times 10^{12}$/km. Afterwards, it was further limited to $6 \times 10^{11}$/km by Euro 6c standard in 2017. To effectively reduce particle emissions of GDI engines, a dual-injection system was commercially applied as a novel technology by Toyota in 2005 [17]. As shown in Fig. 1, dual-injection Spark Ignition (SI) engines could offer combined advantages of PFI and DI as required, so it has been an effective way to reduce particle emissions.

![Fig. 1. Schematic of a dual-injection system in SI engine](image)

Regarding the important research findings of particle emissions in dual-injection SI engines, Daniel et al. [18] concluded that the emission of large particles (> 50 nm) is almost removed by using ethanol fractions in dual-injection, resulting in a unimodal distribution of particle mass. Liu et al. [19][20] systematically compared the effects of "PFI-alcohols and DI-gasoline" and "PFI-gasoline and DI-alcohols" on PN reduction. It was indicated that as the alcohols mass ratio rises, PN would reduce more than 95% compared to pure gasoline DI injection. Moreover, under the selected engine operating conditions, there is an optimal mass fraction for PFI-alcohols to significantly reduce PN
and keep fuel economy and power performance in the meantime. Kim et al. [21] found that the size distribution of particle moves to a smaller range when PFI-ethanol is added. The number reduction of particles larger than 50 nm would bring a significant reduction in particle mass emissions. Catapano et al. [22] demonstrated that due to the gaseous properties (no carbon-carbon bond), there is a benefit of using Compressed Natural Gas (CNG) to reduce particle emissions in an optical CNG-gasoline dual-fuel SI engine. Yu et al. [23] found that under the conditions of excess air ratio ($\lambda$) = 1 and 1.2, the total PN emissions present an increasing trend with the increase of gasoline addition ratio. Furthermore, due to improved fuel evaporation under lean-burn conditions, the accumulation mode PN can remain at a low level. Kalwar et al. [24] found that the dual-fuel engine fuelled with methanol/gasoline could achieve the lowest particle emissions among the fuel selections of methanol/gasoline, ethanol/gasoline and butanol/gasoline. Liu et al. [25] demonstrated that when the proportion of direct injection gasoline exceeds 50% in a PFI-ethanol and DI-gasoline dual-injection engine, PN is strongly affected by the change of ignition timing or direct injection timing.

Regarding the particle emissions in dual-injection SI engines, it can be concluded that most of the existing research focused on the benefits of using alcohol fuels, which is an effective way to reduce particle emissions. However, in these cases, the lower heating value of alcohols as a disadvantage should be considered, and the implementation of two fuel tanks and accessories would increase the engine cost. Hence, it would make sense to further explore the potential of particle reduction in dual-injection SI engines fuelled with gasoline.

Although the effects of injection pressure on particle emissions in SI engines has always been a research hotspot in academia [26-33]. Wang et al. [26] demonstrated that increasing gasoline injection pressure can help GDI engines reduce particle emissions. By a numerical study on a GDI engine, Reddy et al. [28] found that there is an apparent decrease in the soot emissions with the increase of
injection pressure from 11 MPa to 20 MPa. Sharma et al. [29] concluded that injection pressure could significantly affect the air-fuel mixture preparation and particle emissions from a GDI engine fuelled with gasohol. By both simulation and experimental studies from a GDI engine fuelled with gasoline or bioethanol-gasoline blended fuels, Park et al. [30][31][33] demonstrated that increasing injection pressure is an effective way to reduce particle emissions, particularly under the wall wetting conditions. However, there is almost no study on the effects of injection pressure on particle emissions in dual-injection SI engines. Furthermore, a comprehensive study of the effects of fuel injection pressure on microscopic spray characteristics and particle emissions in dual-injection SI engines has not been reported before.

Therefore, a study of optimising microscopic spray characteristics and particle emissions in a dual-injection SI engine by changing GDI injection pressure up to 18 MPa was conducted in this paper. Although nowadays the maximum injection pressure of some latest generation engines has been improved to 35 MPa, the maximum injection pressure of 20 MPa is still one common option for commercialised GDI engines [26][28][32]. The findings of this paper would provide a deep understanding of the effects of fuel injection pressure on microscopic spray characteristics and particle emissions in dual-injection SI engines fuelled with gasoline, whilst offering a practical approach to the reduction of particle emissions. The rest of the paper is organised as follows: Section 2 mainly introduces the experimental setup of microscopic spray characteristics and engine testbed. Section 3 provides the experimental results and discussion, followed by the conclusions given in Section 4.

2. Experimental setup and procedure

2.1. Microscopic spray characteristics testbed and test procedure

The experiment of microscopic spray characteristics was conducted using a Phase Doppler
Particles Analyser (PDPA) system, as shown in Fig. 2. It mainly consists of an argon-ion laser, a beam separator, a transmitter, a receiver, a signal processor with 180 MHz frequency and a 3-D motion traverse system with high accuracy of 0.1 mm.

The GDI injector used in this study is from a dual-injection engine, which specifications are shown in Table 1. The orifice geometry and spray sketch of the five-hole injector are shown in Fig. 3. The information of the injector orifices is obtained from X-ray computed tomography, which indicates that the inner diameter of each hole is 0.174 mm. The fuel used in this study is commercial gasoline, which properties are shown in Table 2. The injector was installed in a customised metal holder, which fully kept the injector perpendicular to the ground and its position stability during the test. Fuel was injected into room conditions, which is 293 ± 0.5 K and 0.1 MPa. Fuel injection pressure was controlled and maintained with a high resolution of 0.1 MPa with the assistance of a high-pressure fuel pump. Besides, an air extraction hood was used to frequently extract the air out of the test site to eliminate the safety risks.

Fig. 2. Experimental setup of microscopic spray characteristic
Table 1. Engine specifications

<table>
<thead>
<tr>
<th>Items</th>
<th>Content</th>
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</thead>
<tbody>
<tr>
<td>Engine type</td>
<td>four-cylinder, four-stroke</td>
</tr>
<tr>
<td>Bore × Stroke (mm)</td>
<td>82.5 × 92</td>
</tr>
<tr>
<td>Displacement (L)</td>
<td>2.0</td>
</tr>
<tr>
<td>Injection type</td>
<td>Dual-injection system (PFI plus GDI)</td>
</tr>
<tr>
<td>Intake type</td>
<td>Turbocharged</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>9.6:1</td>
</tr>
<tr>
<td>Rated speed (rpm)</td>
<td>5500</td>
</tr>
<tr>
<td>Rated power (kW)</td>
<td>160</td>
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<tr>
<td>Maximum Torque (N·m)</td>
<td>320</td>
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Fig. 3. Injector orifice geometry and spray sketch

Table 2. Fuel properties

<table>
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<tr>
<th>Fuel type</th>
<th>Gasoline</th>
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<tr>
<td>Chemical formula</td>
<td>C5-C12</td>
</tr>
<tr>
<td>Relative molecular mass</td>
<td>95-120</td>
</tr>
<tr>
<td>Gravimetric oxygen content (%)</td>
<td>&lt; 1</td>
</tr>
<tr>
<td>Research octane number</td>
<td>95</td>
</tr>
<tr>
<td>Density (20 °C) (kg/L)</td>
<td>0.73</td>
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<tr>
<td>Dynamic viscosity (20 °C) (mPa·s)</td>
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<td>Kinematic viscosity (20 °C) (mm²/s)</td>
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<td>Surface tension (20 °C) (N/m)</td>
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<td>Boiling range (°C)</td>
<td>30-200</td>
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<tr>
<td>Low heating value (kJ/kg)</td>
<td>44300</td>
</tr>
<tr>
<td>Latent heat of vaporisation (kJ/kg)</td>
<td>370</td>
</tr>
</tbody>
</table>
In this research, fuel injection pressure was set to be 5.5 MPa, 10 MPa, 14 MPa and 18 MPa, covering the normal range of commercial GDI injection pressure. Based on the recommendation of the Society of Automotive Engineers (SAE) J2715, spray measurement points are typically 50 mm downstream along the injector axial direction [34]. Fig. 4 shows the coordinate system of measurement points during the test. The rightward and downward directions are defined to be the positive direction of "Y-axis" and "X-axis", respectively. For example, (4, 50) represents the point which is 4 mm right and 50 mm below the nozzle. Besides, the injector actuation signal was sent by a programmable Electronic Control Unit (ECU), which is also used to synchronise fuel injection with PDA data acquisition. During the test, laser wavelength of PDPA system ranged from 488 nm to 514.5 nm, and laser power was set to be 1.3 W. The measurement range of droplet velocity and size was -151.95 to 238.77 m/s and 0 to 236 µm, respectively. Meanwhile, the PDPA system can provide a high measuring accuracy, with resolutions of 0.01 m/s and 0.1 µm for droplet velocity and size, respectively. Furthermore, in order to avoid the interference of suspended fuel droplets, injection pulse width and injection frequency were set to be 1.2 ms and 0.1 Hz, respectively. To minimise the deviations by injection variations, data collection was completed with 20,000 validated sample droplets for each measurement point, and was repeated three times.
In addition, some parameters used in the study of microscopic spray characteristics are introduced as follows. $P_I$ denotes fuel injection pressure of GDI; $t$ denotes the time After Start of fuel Injection (ASOI); $V_N$ denotes normal component of droplet velocity; $V_T$ denotes tangential component of droplet velocity; $D_d$ denotes droplet diameter.

According to the research work by Arcoumanis et al. [35], droplets can be identified to be different regimes based on Weber number ($We$) which is commonly used to analyse the breakup mechanism, as shown in Equation (1).

$$We = \frac{\rho_L U^2 D_d}{\sigma}$$  \hspace{1cm} (1)

Here, $We$ denotes Weber number; $\rho_L$ denotes fuel density; $U$ denotes normal incident velocity; $D_d$ denotes droplet diameter; $\sigma$ denotes surface tension coefficient of the fuel.

As shown in Equation (2), droplets can be defined by Sauter Mean Diameter (SMD), which means the diameter of a sphere that has the same volume-surface area ratio as a droplet of investigation [36]. Based on the recommendation of SAE J2715, SMD is very effective to provide a better visual understanding for the diameter of a large cluster of droplets [34].
\[ D_{SMD} = \frac{\int_{D_{min}}^{D_{max}} D^3 dN}{\int_{D_{min}}^{D_{max}} D^2 dN} \] (2)

Here, \( D_{SMD} \) denotes SMD of droplet; \( D_d \) denotes droplet diameter; \( N \) denotes number of droplets.

### 2.2. Engine testbed and test procedure

Fig. 5 shows the schematic diagram of the engine testbed. An electrical dynamometer was coupled with a turbocharged dual-injection engine, which has been introduced in Table 1 before.

Piezo-electric spark-plug pressure sensors (AVL-GH13Z), a crank encoder (Kistler-2614CK1), a charge amplifier (Kistler-5018A) and a combustion analyser (AVL-641) were used to measure and analyse the transient in-cylinder pressure signals. A fast particle analyser (Cambustion-DMS 500) was connected to the exhaust pipe in front of the three-way catalytic converter to measure engine particle emissions, ranging from 5 nm to 1000 nm. In order to ensure the consistency of measurement standard for particle emissions under all conditions. The dilution factor of first-stage and second-stage in the particle analyser are kept at 1:4 and 1:100, respectively. In order to eliminate the impacts of cycle-to-cycle variations, cylinder pressure data were averaged by 200 consecutive cycles, and particle emissions data was measured three times for each measurement condition. The air-fuel equivalence ratio (lambda), intercooler outlet temperature and coolant kept constant at 1 ± 0.01, 298 ± 2 K and 360 ± 2 K, respectively. Besides, based on Holman's root mean square method [37], Table 3 presents that the uncertainties of the engine test are very low and completely acceptable.

In order to make the experimental procedure more efficient, the fuel mass ratio of PFI and GDI was kept at 1:1. Regarding the operating conditions, "2000 revolutions per minute (rpm)-2 bar Brake Mean Effective Pressure (BMEP)" and "2000 rpm-10 bar BMEP" were chosen to represent the engine typical low load and mid-high load, respectively. Meanwhile, a programmable ECU with software INCA can accurately control the GDI fuel injection pressure, spark timing and other operating
parameters. The spark timing was optimised to be the minimum advance for Maximum Brake Torque (MBT) or Knock Limited Spark Advance (KLSA). Besides, the engine speed, torque, stoichiometric air-fuel ratio and fuel injection timing are all kept constant under a fixed engine operating condition.

Fig. 5. Schematic diagram of engine testbed

<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>
</tbody>
</table>
3. Results and discussion

3.1. Optimising microscopic spray characteristics by changing GDI injection pressure

Fig. 6 shows $V_N$ at (0, 50) with varying $P_I$. It can be seen that when the tip of jet firstly reaches the measurement point, $V_N$ is relatively high. Afterwards, $V_N$ demonstrates a decline to a very low level. This is mainly because as time progresses, the spray would gradually collapse, reducing the regularity of droplets movement direction.

Besides, although $V_N$ curves show similar overall trends under different $P_I$, but some discrepancies can also be observed. First, with the increase of $P_I$, $V_N$ would be significantly improved. Under $P_I = 18$ MPa, $V_N$ of spray tip can be up to around 42.5 m/s, which is much higher than the 30 m/s under $P_I = 5$ MPa. Second, the beginning of decline of $V_N$ advances from 1.9 ms ASOI to 1.2 ms ASOI, leading to a shorter head section of $V_N$ curve. Third, the decreasing rate of $V_N$ curve would increase under higher $P_I$. These discrepancies can be mainly attributed to that by increasing $P_I$ from 5.5 MPa to 18 MPa, the gap of the velocity between the jet surface and ambient gas would be enhanced. Hence, the increased instabilities for the droplets would have a negative impact on the kinetic energy of droplets movement along the axial direction. Therefore, it can be concluded that increasing $P_I$ does not substantially increase $V_N$, which would not significantly raise the possibility of spray impingement.

Fig. 7 shows $V_T$ at 50 mm of jet downstream. It can be seen that due to the interaction effects between adjacent plumes, the absolute value of $V_T$ for measurement point (-16, 50) is a bit larger than (16, 50). With $P_I$ increases from 5.5 MPa to 18 MPa, the absolute values of $V_T$ shows a trend of increase. This can be attributed to that higher $P_I$ increases the instability of spray, enhancing the tangential kinetic energy of droplets. Meantime, due to the stronger effects of vortex and interaction around the spray boundary, the increase of $V_T$ becomes more obvious for the measurement points at
both ends. Consequently, with a higher $P_I$, both $V_N$ and $V_T$ of droplets increase, improving the diffusion rate of spray.

![Fig. 6. $V_N$ at (0, 50) with varying $P_I$.](image)

![Fig. 7. $V_T$ at 50 mm of jet downstream](image)

Fig. 8 shows the probability of $D_d$ with varying $P_I$. It shows that all the curves present unimodal
distributions. With the increase of $P_I$ from 5.5 MPa to 18 MPa, the position of curve’s peak moves from 10 µm to 6 µm. Moreover, the reduced probability of $D_d$ larger than 20 µm can be found under $P_I = 18$ MPa. The higher probability of smaller $D_d$ demonstrates that secondary atomisation of spray becomes more complete under higher $P_I$, which leads to a more homogeneous air-fuel mixture.

Fig. 9 presents the probability of droplet distribution at (0, 50) according to $We$. It shows that with the increase of $P_I$ from 5.5 MPa to 18 MPa, the probability of droplets in the regime of "$We < 100$" has a reduction of 0.495%. The "$We < 100$" regime mainly includes vibrational and bag regimes, which represent that some droplets are deformed, shapeless or break into a number of fragments, but the breakup process is not very drastic compared to that of higher $We$ droplets. In the meantime, the probability of droplets in "$100 \leq We < 1000$" and "$We \geq 1000$" increase by 0.482% and 0.013%, respectively. It means that more droplets would collapse with a mode of continuous ripping of the surface layer, even abruptly breaking down into a huge number of smaller and tiny droplets.

![Fig. 8. Probability of $D_d$ at (0, 50)](image_url)
Fig. 9. Probability of droplet distribution at (0, 50) according to $We$

Fig. 10 presents $D_{SMD}$ at 50 mm of jet downstream with varying $P_I$. It can be seen that regardless of $P_I$, all the curves present bimodal distribution. From jet’s central axis to sides, $D_{SMD}$ increases first by around 3 µm, then it has a reduction at both end measurement points (-16, 50) and (16, 50). This is because the probability of droplets collision and coalescence inside the spray boundary increases a lot, leading to an improvement in $D_{SMD}$ [38][39][40]. However, the droplets outside the spray boundary are more affected by the shearing force between the jet surface and ambient gas, resulting in a higher possibility of breakup and a reduction in droplet size.
3.2. Optimising engine particle emissions by changing GDI injection pressure

Based on the particle diameter ($D_p$) boundary of 30 nm to 50 nm, engine emitted particles can be classified into nucleation particle ($5 \text{ nm} \leq D_p < 30 \text{ nm}$) and accumulation particle ($0 \text{ nm} \leq D_p < 1000 \text{ nm}$) [41][42][43][44].

Fig. 11 shows the effects of $P_l$ on PN concentration for nucleation, accumulation and total particles. A declining trend in particle emissions can be seen with the increase of $P_l$ from 5.5 MPa to 18 MPa. The total PN concentration has a reduction of 53.98% and 45.44% at 2 bar and 10 bar, respectively. This is because the primary and secondary droplets atomisation can be accelerated, improving the homogeneity of air-fuel mixture. Moreover, the possibility of spray impingement would not increase a lot under a higher $P_l$, as stated in Section 3.1. Hence, PN concentration can be effectively restricted, as soot is generally formed as a by-product of incomplete combustion in fuel-rich mixture areas. Besides, because the impingement possibility is higher at 10 bar by the longer injection pulse, the decrease of PN concentration for accumulation and total particles at 10 bar is smaller than those of 2 bar.

These characteristics can be further explained by the effects of $P_l$ on PN size distribution, engine
exhaust temperature \( T_E \) and maximum in-cylinder temperature \( T_M \).

Fig. 12 (a) shows all the PN curves present unimodal distributions at 2 bar. The curves mainly concentrate on the ultrafine particles ranging from 5 nm to 100 nm, and the peaks of curves centred around 15 nm. Fig. 12 (b) shows that under 10 bar conditions, the PN curves present approximate bimodal distributions, which peaks are around 15 nm and 50 nm. By increasing of \( P_I \) to 18 MPa, the bimodal form becomes more obvious, as the reduction of accumulation PN is less than that of nucleation particles.

Fig. 12 also presents that with the increase of \( P_I \), from 5.5 MPa to 18 MPa, the PN size distribution curves show a clear downward trend. The peak of the curve decreases from \( 3.01 \times 10^6 \) to \( 1.43 \times 10^6 \) at 2 bar, and reduces from \( 1.08 \times 10^6 \) to \( 5.33 \times 10^5 \) at 10 bar. This is mainly because of the benefits from the improved homogeneity of air-fuel mixture and higher in-cylinder temperature. As shown in Fig. 13, \( T_E \) and \( T_M \) have an increase of around 20 K and 60 K with the increase of \( P_I \).

With regards to reducing particle emissions, a higher temperature is usually helpful, which can be attributed to three factors. First, more complete combustion could decrease the production of particles from the development of unburned carbon elements. Second, the nucleation tendency of unburned hydrocarbon is potentially reduced. Third, the probability of particle oxidation can be increased, which could reverse the production of some particles.
Fig. 11. Effects of $P_I$ on PN concentration for nucleation, accumulation and total particles.

(a) 2 bar
The present study optimises the microscopic spray characteristics and particle emissions in a dual-injection SI engine fuelled with gasoline by changing GDI injection pressure. The study results not only present a comprehensive resource about the effects of injection pressure on microscopic spray characteristics and particle emissions, but also contribute to provide a practical approach to reduce particle emissions from dual-injection SI engines fuelled with conventional fuel (gasoline). The obtained results can be summarised as follows:
(1) With the increase of $P_I$ from 5.5 MPa to 18 MPa, both $V_N$ and $V_T$ of droplets increase, which would enhance the diffusion of spray. However, the beginning of decline advances and the rate of decline increases. Hence, the possibility of spray impingement would not increase a lot under a higher $P_I$.

(2) By increasing $P_I$ from 5.5 MPa to 18 MPa, the probability of smaller droplets is higher. The probability of droplets in the regime of "$We < 100$", "$100 \leq We < 1000$" and "$We \geq 1000$" have an increase of -0.495%, 0.482% and 0.013%, respectively. It is demonstrated that by increasing $P_I$, more droplets would collapse with a mode of continuous ripping or break down abruptly.

(3) From jet’s central axis to sides, $D_{SMD}$ increases first by around 3 µm, then reduces outside the spray boundary due to the strong shearing force between the jet surface and ambient gas.

(4) At 2000 rpm-2 bar, all the PN curves present unimodal distributions which peak most centred around 15 nm regardless of $P_I$. At 10 bar, PN curves present approximate bimodal distributions, which peaks are around 15 nm and 50 nm.

(5) By increasing $P_I$ from 5.5 MPa to 18 MPa, $T_E$ and $T_M$ have an increase of around 20 K and 60 K. The peak of PN distribution curve decreases from $3.01 \times 10^6$ to $1.43 \times 10^6$ at 2 bar, whilst reducing from $1.08 \times 10^6$ to $5.33 \times 10^5$ at 10 bar. Correspondingly, total PN concentration significantly reduced 53.98% and 45.44% at 2 bar and 10 bar, respectively.

**CRediT authorship contribution statement**

**Xiang Li:** Conceptualization, Methodology, Software, Formal analysis, Investigation, Data curation, Visualization, Writing - original draft. **Dayou Li:** Writing - reviewing & editing. **Yiqiang Pei:** Methodology, Project administration, Funding acquisition. **Zhijun Peng:** Methodology, Writing
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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Reference


Appendix

Abbreviations

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<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>ASOI</td>
<td>After Start of Injection</td>
</tr>
<tr>
<td>BMEP</td>
<td>Brake Mean Effective Pressure</td>
</tr>
<tr>
<td>CNG</td>
<td>Compressed Natural Gas</td>
</tr>
<tr>
<td>DPF</td>
<td>Diesel Particulate Filter</td>
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<tr>
<td>ECU</td>
<td>Electronic Control Unit</td>
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<td>GDI</td>
<td>Gasoline Direct Injection</td>
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<td>Gasoline Particulate Filter</td>
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<td>KLSA</td>
<td>Knock Limited Spark Advance</td>
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<tr>
<td>MBT</td>
<td>Maximum Brake Torque</td>
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<td>PDPA</td>
<td>Phase Doppler Particles Analyser</td>
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<td>PFI</td>
<td>Port Fuel Injection</td>
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<td>revolutions per minute</td>
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