Experimental and numerical study of high-pressure-swirl injector sprays in a direct injection gasoline engine

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The manuscript was received on 11 January 2005 and was accepted after revision for publication on 28 July 2005.
DOI: 10.1243/095765005X31333

Abstract: The characteristics of free spray of a new type high-pressure-swirl injector in gasoline direct injection (GDI) engine under various injection conditions are investigated. The fuel spray with hollow-cone structure, wide spreading, and large spray angle is observed under the injection condition simulating to the GDI engine operation at full load. The study shows that a vortex structure can be clearly observed in the periphery of the spray. Meanwhile, an initial spray slug also appears at the tip of the main spray. Under the injection condition of GDI engine partial load, the structure of fuel spray changes into the more compact and solid-cone shape with decreased spray width. Moreover, the influences of the injection pressures and ambient pressures on the spray characteristics of the injector are studied. Along with the experimental studies, a general numerical model for the swirl spray is developed. Then, the model is implemented into a multi-dimensional computational fluid dynamics code (KIVA-3V) to theoretically study the pressure-swirl injector sprays. Comparisons between the computed and measured spray characteristics such as spray structure, spray tip penetration, and droplet sizes are made, and good agreement has been achieved between the model prediction and measurement.

Keywords: GDI engine, high-pressure-swirl injector, spray characteristics, spray model, numerical study

1 INTRODUCTION

Gasoline direct injection (GDI) engine technology has been proved to be a good potential for future automobile engine. The advantages of GDI engine reflect in their higher thermal efficiency, better potential for reducing the specific fuel consumption, and the freedom for controlling the injection timing and in-cylinder fuel quantity. The GDI engine also has the potential for significant improvements in pollutant emissions and the start-acceleration performance when compared with those of traditional gasoline engine [1, 2].

Much research has been done for characterizing GDI injector sprays [2, 3]. It was found that fuel spray with smaller droplet size, wide spreading, and low penetration should be required at high load condition, whereas a well-atomized compact spray and a stratified charge would be desirable at low load condition. Recently, a specially designed high-pressure-swirl injector would satisfy these demands and will be widely used in the production of GDI engines progressively. For this type of injector, the fuel is injected through tangential or helical passages into a swirl chamber, from which it emerges with both tangential and axial velocity components to form a thin conical sheet at the nozzle exit [4]. Most of the recently developed GDI engines use the high-pressure-swirl injector. Two kinds of fuel preparation strategies are taken: (1) under the high load condition, early injection is adopted, where homogeneous mixture is desirable; (2) under the partial load condition, late injection and stratified lean burn are taken, and the charge stratification
and the controlling of mixture distribution near the spark plug will permit the engine to be operated at a very lean air/fuel ratios. As the spray characteristics have strong influence on engine air–fuel mixing and combustion, an in-depth understanding of the spray characteristics of this new type injector is important.

Besides the measurement, another powerful tool to get deeper insights into various subprocesses of engine combustion and their interactions is the multi-dimensional modelling, called CFD (computational fluid dynamics), codes. Great progress has been made in multi-dimensional modelling, both in submodels development and in numerical methods. Miyamoto and Kobayashi [5] used a sheet atomization model for the air-assisted hollow-cone injector, based on a sheet stability argument; however, the secondary drop break-up process was not considered. Lee and Bracco [6] used Lagrangian equations to solve the motion of an intact sheet outside the injector; their computed results were sensitive to the grid settings. Han et al. [7] used a variant of the TAB model for predicting the break-up of hollow-cone sprays in GDI engines. Papageorgakis and Assanis [8] developed an alternative low pressure break-up model, applicable for the Rayleigh and first wind-induced regimes, which reflects both aerodynamic and surface tension effects. However, all the earlier mentioned models focus only on the individual processes (atomization, break-up, etc.), a general numerical model that can describe the whole physical phenomena, such as fuel spray atomization, spray break-up, droplet distribution for the high-pressure-swirl injector sprays, has not been presented.

In this study, first, a measurement system for studying the fuel spray characteristics is developed. Then, the spray characteristics of the high-pressure-swirl injector under various injection conditions are obtained. Along with the experimental study, a general numerical model that can describe all physical phenomena, such as fuel spray atomization, spray break-up, droplet distribution for the high-pressure-swirl injector sprays, is developed. The improved fuel sheet model is used to simulate the spray atomization process, the high speed liquid sheet break-up model based on surface break-up theory and linear stability analysis is introduced to simulate the fuel sheet break-up processes, the Rosin–Rammler function distribution is used to calculate the droplet distribution, and the initial spray slug is also introduced by split injection consideration and separate treatment to that of the main spray. All submodels are implemented into the KIVA-3V code to perform the computations. Comparisons between the computed and measured spray characteristics such spray structure, spray tip penetrations, and droplet sizes are made.

2 EXPERIMENTAL SET-UP AND CONDITIONS

2.1 Experimental set-up

A schematic diagram of experimental set-up for the study is shown in Fig. 1. It consists of a constant volume chamber, a high-pressure-swirl injector, a high-pressure fuel-supply system, an electronic controlling circuit, and a high-speed schlieren photography system.

The inside size of the constant volume chamber is 200 × 382 mm inner diameter. The chamber can be pressurized up to 10 MPa by nitrogen gas. Two sides of the chamber are transparent to make the inside observable. The fuel injector used in this study is a swirl-type GDI injector, which is widely used in production of GDI engines. Schematic diagrams of the injector and the details of the nozzle tip, including the tangential slots, are shown in Fig. 2. The nozzle has a press-fitted swirl tip with six equally spaced tangential slots, which gives the injecting fuel an angular momentum. The hole diameter of the nozzle is 0.5 mm and the length-to-diameter ratio of the nozzle orifice is 2.3 [9].

In order to achieve fuel supply with a constant pressure for the injector, a high-pressure fuel-supply system is prepared, an accumulator containing fuel inside will be pressurized by nitrogen gas. The injection signal is controlled by a Single Chip Micyoco, which generates TTL pulses with certain time duration, and to the signal is transmitted to the injector and to the high-speed CCD camera simultaneously. The schlieren images of spray are recorded by a REDLAKE HG-100 K high-speed CCD camera, at 640 × 480 pixels and 8 bit depth of greyscale level with recording speed of 5000 frames per second. Detailed description of the schlieren method was described in reference [10].
2.2 Experimental conditions

Table 1 gives the experimental conditions and fuel properties. The spray development processes are studied under four injection pressures ranging from 2.0 to 5.0 MPa and five ambient pressures ($P_a$) ranging from 0.1 to 2.0 MPa under room temperature. The injection duration is fixed at 1.6 ms.

The spray injected at low ambient pressure (0.1 MPa) will simulate the early injection mode in GDI engine under high load condition, whereas the sprays injected at the elevated ambient pressures (0.5–2.0 MPa) will simulate the late fuel injection modes for stratified charge combustion operation.

3 NUMERICAL MODELS

The CFD-code used in this study is a version of KIVA-3V, which has been developed originally by the Los Alamos National Laboratory, USA [11–13]. The modified submodels for spray atomization, spray break-up, and droplets distribution processes are described in more detail as follows.

### 3.1 Liquid sheet model

It is assumed that a conical liquid sheet with length $L_s$ and thickness $2h$ is formed at the nozzle exit. The average cone angle of the spray is $2\theta$. As shown in Fig. 3, disintegration occurs due to the continuous growth of waves on the surface caused by the aerodynamic forces acting on the sheet. When the waves reach a critical value, fragments of liquid will break up and form the cylindrical ligaments. Furthermore, the unstable ligaments will break into droplets. The external forces make the large droplets to break into the small droplets. Later on, the droplets will undergo collision, coalescence, turbulent dispersion, and evaporation processes.

The parameter, $X$, which is defined as the ratio of the air-core area to the orifice area at the nozzle exit, is introduced into the study

$$
X = \left(1 - \frac{4h}{d_0}\right)^2
$$

where $d_0$ is the diameter of nozzle orifice.

The fuel flowrate, $Q$, according to the mass conservation, is given as

$$
Q = A_0(1 - X)V \cos \theta
$$

where $A_0$ is the nozzle orifice cross-section area.

The sheet velocity, $V$, is defined as

$$
V = KV \left[\frac{2(p_1 - p_2)}{p_1}\right]^{0.5}
$$

Table 1  Experimental conditions and fuel properties

<table>
<thead>
<tr>
<th>Injector</th>
<th>Mitsubishi swirl type injector</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection duration (ms)</td>
<td>1.6</td>
</tr>
<tr>
<td>Injection pressure (MPa)</td>
<td>2.0, 3.6, 4.6, 5.0</td>
</tr>
<tr>
<td>Ambient pressure (MPa)</td>
<td>0.1, 0.5, 1.0, 1.5, 2.0</td>
</tr>
<tr>
<td>Ambient temperature (K)</td>
<td>300</td>
</tr>
<tr>
<td>Fuel density (kg m$^{-3}$)</td>
<td>770</td>
</tr>
<tr>
<td>Fuel surface tension (N m$^{-1}$)</td>
<td>$2.3 \times 10^{-2}$</td>
</tr>
<tr>
<td>Fuel viscosity (Pa s)</td>
<td>$5.0 \times 10^{-4}$</td>
</tr>
</tbody>
</table>

**Fig. 2** Schematic diagrams Mitsubishi injection nozzle and details of nozzle tip

**Fig. 3** Schematic diagram of the liquid sheet and its break-up
where \( p_1 \) is the fuel pressure inside the injector, \( p_2 \) the ambient pressure, \( \rho_l \) the liquid density, and \( K_V \) the velocity coefficient, which can be determined from the following exponential correlation [4]

\[
K_V = \frac{C}{\cos \theta} [(1 - X)Re_h^{0.25}]^{1/2}
\]  \hspace{1cm} (4)

where \( C \) is a constant that varies with the injector designs, \( Re_h = (1 - X)d_0\rho_l V \cos \theta/\mu_1 \) the liquid sheet Reynolds number, and \( \mu_1 \) the fuel dynamic viscosity.

Thus, the whole liquid sheet characteristics' parameters including sheet thickness, sheet velocity, and velocity coefficient (break-up length \( L_s \)), the equation is in the following section) at the nozzle exit can be calculated with equations (1)–(4).

It is found by both experiments and numerical simulation that \( K_V \) will have a constant value (the variation < 0.02) when the injection pressure exceeds a certain value (~3.0 MPa) [14]. This implies that the liquid sheet thickness will also remain constant (the variation < 2 \( \mu \)m). By referencing the previous studies, the value of 0.7 for \( K_V \) is used instead of equation (4) for the injection pressure of 5.0 MPa, so the model can be further simplified by this assumption.

### 3.2 Spray break-up model

Spray break-up simulation usually use the TAB break-up model [11] and WAVE break-up model [15], whereas in standard KIVA-3V code, the TAB model is used. In this study, a high-speed liquid sheet break-up model based on the surface break-up theory and the linear stability analysis is used to simulate the fuel sheet break-up processes [16].

The model assumes that the disintegration of the liquid sheet occurs due to the continuous growth of waves on the liquid sheet surfaces. The amplitude for the infinitesimal disturbance can be expressed as

\[
\eta = R[\eta_0 \exp (ikz + \omega t)]
\]  \hspace{1cm} (5)

where \( \eta_0 \) is the infinitesimal amplitude of initial perturbation, \( \omega = \omega_r + i\omega_i \) the wave growth rate, \( k = 2\pi/\lambda \) the wave number, \( \lambda \) the wavelength, and \( R \) the real part of complex function in the parentheses. By solving the mass equation and motion equation, the dispersion equation of wave growth rate and wavelength can be derived. The solution of the dispersion equation can be expressed as

\[
\omega_c = \sqrt{TV^2k^2 - \sigma k^3/\rho_l(1 + T)}
\]  \hspace{1cm} (6)

where \( T = \rho_2/\rho_1, \rho_1 \) and \( \rho_2 \) density of fuel and gas, respectively, \( \sigma \) and the fuel surface tension. The correlation of the maximum growth rate, \( \Omega \), with the corresponding wavelength, \( \Lambda \), can be obtained by curve-fitting method as follows

\[
\frac{\Omega h}{V} = \frac{2}{3} We_1 \sqrt{\frac{T}{3}}
\]  \hspace{1cm} (7)

\[
\frac{\Lambda}{h} = \frac{2\pi}{We_1}
\]  \hspace{1cm} (8)

The resulting diameter, \( d_L \), of the new ligaments is given by

\[
d_L = \sqrt{\frac{8\Lambda h}{\pi}}
\]  \hspace{1cm} (9)

Furthermore, the unstable ligaments will break into droplets and their diameter is expressed as

\[
d_0 = 1.88d_L(1 + 3Oh)^{1/6}
\]  \hspace{1cm} (10)

where \( Oh = \mu_1/(\rho_1\sigma d_L)^{1/2} \) is the Ohnesorge number and the diameter is considered as the sauter mean diameter (SMD) of the new droplets.

The break-up time, \( T \), and break-up length, \( L_s \), of the liquid sheet are given as follows

\[
\tau = \frac{1}{\Omega} \ln \left( \frac{\eta_0}{\eta_s} \right)
\]  \hspace{1cm} (11)

\[
L_s = V_T = \frac{U}{\Omega} \ln \left( \frac{\eta_0}{\eta_s} \right)
\]  \hspace{1cm} (12)

where \( \eta_0 \) is the wave amplitude when the surface disturbance reaches the critical value of break-up and \( \ln (\eta_0/\eta_s) \) is given the value of 12 in the study.

Later on, the large droplets will undergo the secondary break-up and turn into small droplets. The TAB break-up model is used for simulating this break-up process. The SMD of the droplets after the TAB break-up is expressed as

\[
d_{32} = \frac{d_0}{(7/3) + (1/64)(\rho_1d_0^2/\sigma)(dy/dt)^2}
\]  \hspace{1cm} (13)

where \( y \) is relative deformation of droplet radii.

The liquid sheet model and the new break-up model are implemented into the KIVA-3V code, combining with the droplets collision, coalescence, turbulent dispersion, and evaporation modulations, to proceed the simulation.
3.3 Droplet distribution model

In the original KIVA code [11], it is assumed that after parent droplet break-up, the sizes of the children droplets are distributed as $\chi^2$ function with $r_{32}$ to be the sauter mean radius. In this study, a Rosin–Rammler distribution is introduced, giving less large droplets and more medium droplets when compared with the $\chi^2$ distribution. The R–R volume distribution is expressed as

$$\frac{d(V)}{d(d)} = \frac{qd^{q-1}}{\bar{d}^q} \exp\left[-\left(\frac{d}{\bar{d}}\right)^q\right]$$  \hspace{1cm} (14)$$

where $V$ is the fraction of the total volume containing the droplets with diameter less than $d$, $q$ the distribution parameter, which was set to be 3.5 [7], and $\bar{d}$ the characteristic mean droplet size that relates to the SMD $d_{32}$

$$\bar{d} = d_{32}\Gamma(1 - q^{-1})$$  \hspace{1cm} (15)$$

where $\Gamma$ is the gamma function.

Comparison of $\chi^2$ distribution function and R–R distribution function of two SMD is shown in Fig. 4.

3.4 Initial spray model

Figure 5 shows the measured and simulated spray photographs of the pressure-swirl injector at injection pressure of 5.0 MPa and ambient pressure of 0.1 MPa under room temperature. Figure 5(a) is the measured photograph and Fig. 5(b) is the two-dimension simulated result by the standard KIVA-3V code model. The results clearly show that the spray of the swirl injector has been divided into two parts, which an initial spray slug is found in front of the main spray tip. The initial spray slug is formed due to the inside structure of the swirl injector. It is believed that the initial spray slug is formed during the initial stage of the injection when the angular momentum of liquid within the injector passages has not yet been fully built up. Although the amount of fuel injected in this period is less when compared with that of the main spray, it has great

![Initial spray slug](image)

(a) measurement

![Initial spray slug](image)

(b) simulation

Fig. 5 Measured photograph and the simulation from the standard KIVA-3V (0.8 ms after start of injection) (injection pressure 5.0 MPa, ambient pressure 0.1 MPa, ambient temperature 300 K)
Influence on the spray structure and SMD (especially in the initial injection stage). The prediction from the standard KIVA-3V code model shows great difference to that of measured photograph, due to not taking the initial spray slug into consideration. In order to describe this behaviour, the initial spray model is added to the original spray model, by taking into account of preliminary discrete fuel injection before the main spray. The initial spray, in this study, is considered from the early injection with duration of 0.1 ms and spray angle of \(10^\circ\). Simulated results will be presented in the later section.

4 RESULTS AND DISCUSSIONS

4.1 Interaction between fuel spray and air motion

Figure 6 shows the spray images at injection pressure of 5.0 MPa and ambient pressure of 0.1 MPa. In this case, the spray is well developed with a wide spray-angle as time proceeds. It is found that about 2.0 ms after start of injection, a hollow-cone shape is formed around the leading edge of the spray. As the spray develops with time, a rotating vortex-like structure is clearly observed in the spray tip region, which is a typical characteristic of the swirl-type injector sprays.

Previous study showed that owing to the swirling motion of the liquid fuel, the liquid does not occupy the whole cross-sectional area of the nozzle hole, instead, air is sucked into the middle of the liquid to form an air core, thus a hollow-cone shape spray is observed [4]. Moreover, the rotating vortex-like structure is clearly observed in the spray tip region, which is a typical characteristic of the swirl-type injector sprays.

In addition, the initial spray slug is observed in front of the main spray, also known as the SAC spray [3]. The initial spray come from the fuel in the dead volume between the tangential slots and the nozzle needle and is injected without angular momentum. Emphatically, it should be pointed out that this initial spray slug mainly consists of large droplets with high axial momentum. In engine applications, it is probably one of main sources contributing to unburned hydrocarbon emissions.

4.2 Effect of injection pressure

Figure 7 shows the schlieren photographs of the pressure-swirl injector sprays at various injection pressure under room temperature (ambient pressure 0.1 MPa and temperature 300 K). As shown in the spray images, the injection pressure greatly influences on the spray structure. Generally, the spray basically has a hollow-cone structure under all conditions. However, in the case of low injection pressure (2.0 and/or 3.6 MPa), weak rotating air motion is appeared and the vortex-like structure is also weak in the periphery of the spray, especially in 2.0 MPa case. With the increase of injection pressure (4.6 and/or 5.0 MPa), a strong vortex in the periphery of the spray plume is developed. The difference vortex can be explained from the swirl intensity, indicating by the following swirl Reynolds number [2].

\[
\text{Swirl Reynolds number} = \frac{U \cdot r}{\mu}
\]  

where \(U\) is the fuel velocity in swirling grooves, \(r\) the swirling radius, and \(\mu\) the viscosity of fuel. In the fuel injection system, the fuel injection velocity is mainly determined by the injection pressure difference and the discharge coefficient of the nozzle. Practically, for a given injector discharge coefficient (generally constant), fuel velocity will be determined by injection pressure. Low injection pressure will lead to...
low fuel injection velocity and a low value of Swirl Reynolds number. This results in small local pressure difference and low value of swirl intensity and subsequently weak rotating air flow.

The spray tip penetration and spray cone-angle for various injection pressures are shown in Fig. 8. As shown in the figures, both the main tip penetration and the initial spray tip penetration give the same tendency against time at all injection pressures and the spray tip penetration shows slightly increase with the increase of injection pressure. The difference is not significant, because the pressure energy can be effectively converted into the rotating moment with the increase of injection pressure, promoting fuel atomization and depressing the spray tip penetration, which is the typical characteristic of the swirl-type injector. The experimental results also show that the initial spray slug just sustained in the initial 1.7 ms under all injection conditions.

The spray cone-angle graph shows a remarkable increase in the initial 1.5 ms and maintains a fairly constant value in the late stage. Little influence from the injection pressure on the spray cone-angle is observed except for injection pressure of 2.0 MPa. The spray cone-angle stabilizes at 60° in
the stable stage of injection for the Mitsubishi high-pressure-swirl injector, which is determined by the inside structure of the swirl injector.

4.3 Effect of ambient pressure

As the ambient pressure increases from 0.1 to 0.5 MPa or even higher pressures, the spray structure will change into the more compact and solid-cone shape with decreased spray width as shown in Fig. 9. It is considered that under the same injection pressure, a higher ambient pressure results in the less pressure difference in the nozzle exit, decreasing fuel injection velocity and reducing swirling motion of the liquid fuel. Meanwhile, the complete liquid sheet and the liquid sheet under disintegration form a boundary layer between the ambient air and spray’s interior volume, preventing air from moving into the spray [9]. Moreover, increase in ambient pressure also brings high ambient density and large drop-drag force, all these factors make the solid-cone spray at high ambient pressure, a favourable spray for wall-guided stratified combustion for partial load of GDI engines. The vortex-like structure is also formed in the periphery of the spray and their differences of high ambient pressure in vortex reflect on the size and shape when compared with that at ambient pressure of 0.1 MPa.

Similar spray development is observed when the ambient pressure further increases; the overall spray structure still maintains the solid-cone shape while the spray tip penetration and spray cone-angle are decreased, which bring a more compact spray at high ambient pressure.

Figure 10 shows the spray tip penetration and cone angle of the main spray against time at various ambient pressures. The spray tip penetration monotonically increases with time after injection regardless of ambient pressure. Spray tip penetration is decreased at high ambient pressures of 0.5–2.0 MPa when compared with that at low ambient pressure of 0.1 MPa. This can be explained by the shifting of spray axial momentum to radial momentum as increase ambient air resistance to the spray.

Large spray cone-angle is presented at ambient pressure of 0.1 MPa as the existence of hollow-cone shape spray and the large-size vortex-like structure. However, little difference in spray cone-angle is observed between the high ambient pressure cases (0.5–2.0 MPa).

5 COMPARISONS OF MODEL SIMULATION AND EXPERIMENTAL RESULTS

All models described in section 3 are implemented into the KIVA-3V code to proceed the computation. The initial conditions and fuel properties for injection process in calculation are listed in Table 2. Simulation is carried out corresponding to the experimental cases. They are low ambient pressure case that simulates high load case and high ambient pressure case that simulates low load case. The duration of the initial spray slug is set to 0.1 ms, and the late spray is regarded as the hollow-cone type. The parameters of fuel liquid sheet parameters are listed in Table 3.
Fig. 9  Spray images at various ambient pressures at injection pressure of 5.0 MPa and room temperature of 300 K

(a)Spray tip penetration  
(b)Spray cone angle

Fig. 10  Influence of the ambient pressure ($P_i = 5.0$ MPa, $T_a = 300$ K)
5.1 Spray images of model simulation and measurements

Figure 11 shows the photographs and simulation of spray development under low ambient pressure of 0.1 MPa, and Fig. 12 shows the results under high ambient pressure of 0.5 MPa.

Two-dimensional slice of the simulated spray is illustrated for describing the spray structure. The comparison shows a good prediction of spray with that of experiment, indicating the reasonable models for simulating the spray behaviour. In the early stage of spray at high ambient pressure \( (t = 0.8 \text{ ms}) \), the spray has a perfect cone shape with the cone angle specified in Table 2. With the progress of spray \( (t = 1.6 - 3.2 \text{ ms}) \), the spray will deviate from its initial perfect cone shape, and the cone front will contract and a re-circulating vortex structure will form at the edges of spray. Particularly, the vortex clouds will fully developed and are captured in the computation. In the case of low ambient pressure (Fig. 11), good prediction is also realized for the spray and initial spray slug development.

Table 2 Initial conditions and fuel properties for two cases

<table>
<thead>
<tr>
<th>Simulated engine operating condition</th>
<th>Low load</th>
<th>High load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection pressure (MPa)</td>
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<td>6.0</td>
</tr>
<tr>
<td>Initial spray angle (°)</td>
<td>60</td>
<td>60</td>
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<tr>
<td>Fuel sheet dispersion angle (°)</td>
<td>10</td>
<td>10</td>
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<tr>
<td>Injection period (ms)</td>
<td>1.6</td>
<td>1.6</td>
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<tr>
<td>Ambient pressure (MPa)</td>
<td>0.5</td>
<td>0.1</td>
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<tr>
<td>Fuel flow rate ((\text{mm}^3\text{s}^{-1}))</td>
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<td>12.0</td>
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<tr>
<td>Fuel density ((\text{kg m}^{-3}))</td>
<td>770</td>
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<td>Fuel surface tension ((\text{N m}^{-1}))</td>
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<td>Fuel viscosity ((\text{Pa s}))</td>
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<td>Ambient temperature (K)</td>
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<td>Initial spray slug injection</td>
<td>01</td>
<td>01</td>
</tr>
<tr>
<td>period (ms)</td>
<td></td>
<td></td>
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<tr>
<td>Initial spray slug angle (°)</td>
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</table>

Table 3 Computed sheet thickness and velocity for the two cases

<table>
<thead>
<tr>
<th>Simulated engine operating conditions</th>
<th>Low load</th>
<th>High load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sheet thickness ((\text{mm}))</td>
<td>43.4</td>
<td>48.9</td>
</tr>
<tr>
<td>Sheet velocity ((\text{m s}^{-1}))</td>
<td>89.258</td>
<td>88.895</td>
</tr>
</tbody>
</table>

5.2 Spray characteristic parameters of model simulation and measurements

The simulated and measured spray tip penetrations for low and high ambient pressures are given in Fig. 13. The results show that better agreement are achieved between the prediction and measurement under two ambient pressure conditions including the initial spray slug at low ambient pressure case. It can been that due to the absence of initial atomization model and adopting of the TAB break-up model in the standard KIVA-3V code, the break-up process

![Fig. 11](https://example.com) Photographs and simulation of spray development under low ambient pressure. (Upper experimental images: injection pressure 5.0 MPa; ambient pressure 0.1 MPa; ambient temperature 300 K. Lower model simulation: injection pressure 5.0 MPa; ambient pressure 0.1 MPa; ambient temperature 300 K)
is generally overestimated, where the prediction gives lower value than that of experiment. Good agreement between the computed results and experimental results is obtained after introducing the liquid sheet model and the new break-up model. The persisting duration of the initial spray lasts about 1.7 ms under 0.1 MPa ambient pressure.

To the penetration of the main spray body, increase in the ambient pressure will result in the increased ambient density and increased drop-drag force, leading to the short penetration of spray.

As this study did not measure the SMD of spray, the prediction of SMD is made under the experimental conditions of Chang and Farreu [17]. Chang used

![Fig. 12 Measured photographs and simulation of spray development under high ambient pressure. (Upper experimental images: injection pressure 5.0 MPa; ambient pressure 0.5 MPa; ambient temperature 300 K. Lower model simulation: injection pressure 5.0 MPa; ambient pressure 0.5 MPa; ambient temperature 300 K)](image1)

![Fig. 13 Comparison of simulated and measured spray tip penetration ($P_a = 5.0$ MPa, $T_a = 300$ K)](image2)
a single hole, pintle, swirl type ‘Zexel’ injector. Measurement under two different load cases were taken in the study of Chang, and the main experimental conditions are listed in Table 4.

Figure 14 gives the computed overall SMD of the sprays. It can be seen that the computed SMD is in a good agreement with that of measurements. However, the computed SMD using the standard KIVA-3V code gives lower value of SMD than that of measured, especially in the initial 0.5 ms period. The reason would be the absence of the sheet atomization model in the standard KIVA-3V code, and fuel is treated as the discrete droplets at the nozzle exit. In addition, $x^2$ function droplet distribution model and TAB break-up model are used in the standard KIVA-3V code, where the break-up process is overestimated. Furthermore, the initial spray slug is not taken into account. The modified models can improve the accuracy of computation obviously. The early large droplet sizes can be predicted by introducing of the initial spray model. An overall good agreement between the prediction and experiment is achieved, and this indicates that the developed spray submodels are appropriate for simulation of high-pressure-swirl injection spray.

### 6 CONCLUSIONS

The spray characteristics of a new type high-pressure-swirl injector in GDI engine under various injection conditions are investigated. Along with the experimental studies, a general numerical model for the swirl spray is also developed. From the experimental and numerical study, the following conclusions could be made.

1. The fuel spray with hollow-cone structure, wide spreading, and large spray angle is observed under the injection condition simulating to the GDI engine operation at full load. The study shows that a vortex structure could be clearly observed in the periphery of the spray. Meanwhile, an initial spray slug also appears at the tip region of the main spray. Under the injection condition of GDI engine partial load, the structure of fuel spray changes into a more compact and solid-cone shape with decreased spray width.

2. The results show that in the case of low injection pressure, weak rotating air motion is appeared and the vortex-like structure is also weak in the periphery of the spray. The spray tip penetration shows slight increase with the increase of injection pressure, and little influence from the injection pressure on the spray cone-angle is observed, except for injection pressure of 2.0 MPa.

3. As the ambient pressure further increases, the overall spray structure still maintains the solid-cone shape. The spray tip penetration is decreased at high ambient pressure when compared with that at low ambient pressure of 0.1 MPa. Large spray cone-angle is presented at
ambient pressure of 0.1 MPa as the existence of hollow-cone shape spray and the large-size vortex-like structure. However, little difference in spray cone-angle is observed among the high ambient pressure cases.

4. The developed models are implemented into KIVA-3V code to theoretically study the pressure-swirl injector sprays. Comparisons between the computed and measured spray characteristics such as spray structure, spray tip penetration, and droplet sizes are made, and good agreement has been achieved between the model prediction and measurement, which indicates that the developed spray submodels are appropriate for simulation of high-pressure-swirl injection spray.

ACKNOWLEDGEMENTS

This research is supported by the Chinese state key project of fundamental research plan titled as ‘New Generation of Engine of Alternative Fuels’ Grant no. 2001CB209208, the project of NSFC titled as ‘Research on Combustion Characteristic and Control Techniques of Alternative Clean Fuels in ICE’ Grant no. 50136040 and NSFC Awarding Program for excellent state key laboratory Grant no. 50323001.

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APPENDIX

Notation

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>velocity coefficient constant</td>
</tr>
<tr>
<td>C_D</td>
<td>drag coefficient</td>
</tr>
<tr>
<td>d_0</td>
<td>diameter of nozzle orifice</td>
</tr>
<tr>
<td>GDI</td>
<td>gasoline direct injection</td>
</tr>
<tr>
<td>h</td>
<td>half of liquid sheet thickness</td>
</tr>
<tr>
<td>k</td>
<td>wave number</td>
</tr>
<tr>
<td>Ku</td>
<td>velocity coefficient</td>
</tr>
<tr>
<td>L_s</td>
<td>liquid sheet break-up length</td>
</tr>
<tr>
<td>Oh</td>
<td>Ohnesorge number</td>
</tr>
<tr>
<td>q</td>
<td>distribution parameter</td>
</tr>
<tr>
<td>Q</td>
<td>fuel flowrate</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>SMD</td>
<td>sauter mean diameter</td>
</tr>
<tr>
<td>(d_32)</td>
<td>Taylor’s analogy break-up</td>
</tr>
<tr>
<td>V</td>
<td>liquid sheet velocity</td>
</tr>
<tr>
<td>y</td>
<td>relative deformation of drop radii</td>
</tr>
<tr>
<td>χ</td>
<td>geometry parameter of an injector</td>
</tr>
<tr>
<td>η</td>
<td>wave amplitude</td>
</tr>
<tr>
<td>Γ</td>
<td>gamma function</td>
</tr>
<tr>
<td>λ</td>
<td>wave length</td>
</tr>
<tr>
<td>Λ</td>
<td>wavelength corresponding to Ω</td>
</tr>
<tr>
<td>μ</td>
<td>dynamic viscosity</td>
</tr>
<tr>
<td>ω</td>
<td>complex growth rate</td>
</tr>
<tr>
<td>Ω</td>
<td>maximum growth rate</td>
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<tr>
<td>ρ</td>
<td>density</td>
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<tr>
<td>σ</td>
<td>surface tension coefficient</td>
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<tr>
<td>τ</td>
<td>liquid sheet break-up time</td>
</tr>
<tr>
<td>θ</td>
<td>half of the spray cone angle</td>
</tr>
</tbody>
</table>