Performance of Prototype High Pressure Swirl Injector Nozzles for Gasoline Direct Injection

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ABSTRACT

Prototype intermittent swirl-generating nozzles for gasoline direct injection application were fabricated by modifying MPI injector nozzles. Design parameters include geometric configuration of nozzle internal flow passage such as orifice diameter and length, needle geometry and swirl passage designs. Operating parameters are considered such as injection pressure, ambient pressure, injected fuel mass and duration of injector opening.

Performances of the nozzles have been characterized in terms of static and transient flow rate, initial and overall spray angle, penetration, mean droplet diameter and drop size distribution. Computational fluid dynamic modeling of internal flow for the nozzles provided additional insight in addition to the experimental measurements.

Sprays from the prototype nozzle used for measurement in this study exhibited the general features of swirl injection sprays. The spray drop size varied from 20µm to 28µm in Sauter mean diameter under wide range experimental conditions. Computation of internal flow of prototype nozzles was validated with the measurements in terms of flow rate and initial spray angle. After validation, the influence of various design factors on these two performance parameters was examined. The most significant factor was found to be the orifice diameter.

INTRODUCTION

One of the most serious problems in gasoline direct injection (hereafter GDI) engines is known to be the excess unburned hydrocarbon emissions, which is closely related with atomization performance [1] and spray pattern of applied fuel injectors. Because gasoline fuel is directly injected into a combustion chamber and ignited just a short time later, well-atomized spray is required to obtain the proper mixture of fuel vapor and air. High pressure swirl injectors (hereafter HPSI) have been acknowledged to be most suitable for GDI, since it was installed on the first developed engine [2]. Injector manufacturers are making efforts to develop and optimize HPSIs, and also carmakers to apply them into GDI engines [3-8].

The HPSI was evolved from the existing solenoid-driven fuel injector through the modification of the continuous-spray type pressure-swirl atomizer commonly used in various industrial applications. It has advantages such as suitable spray pattern for both the mixture homogenization and the stratification, relatively good atomization quality, simpler control compared to air-assisted fuel injection and so on. Nevertheless, optimization process is still required to enhance the spray characteristics, flow rate control and deposit-free performance of orifice tip.

Recognizing the significance of the hydrodynamic considerations of the nozzle, researchers are focusing on its design optimization considering many design parameters such as orifice diameter, swirl-generating method, swirl port geometry, needle geometry and so on. Investigations with HPSI nozzles have many difficulties. The most significant one is the trouble in fabricating the prototype nozzles. Typical HPSI nozzles are of more complicated design, and would require more sophisticated manufacturing processes. Typical manufacturing tolerances run under a few µm. Prototype injectors would demand higher level of precision manufacturing compared to the mass produced models, and all these processes are costly and time-consuming. Since 1990’s, with this background, attempts have been made to analyze the flow inside the nozzle and to predict the external spray properties from the internal characteristics using computational fluid dynamics [9,10]. Although the computational method has inherent limitations that it requires validation process, its potential of efficiency is increasing with the advancement of computer technology.

From the injector users’ viewpoint, the main considerations include the optimization of operating conditions as well as the design optimization to obtain stable combustion and to reduce fuel pumping power loss. Design optimization of the fuel injector must firstly be carried out,
and, in order to meet the requirements of spray characteristics [11], operating conditions such as injection pressure, ambient pressure and injection duration must be set appropriately.

However, the information for GDI application has not been so comprehensive so far, as compared with diesel injectors. Most of the published research results are obtained with the commercial injector assembly or at least the latest version in the developing process.

In this study, instead of using commercial nozzles, swirl-generating nozzles were fabricated by modifying MPI injector nozzles, which have similar dimensions with the current HPSI nozzles.

This paper presents the results of this development, in which design and operating parameters for the prototype injection nozzles have been investigated. For the application of computational method, validation in terms of the static flow rate and initial spray angle was carried out prior to the main computations. Note that all pressure notations in this manuscript indicate gage pressures.

**EXPERIMENTAL AND COMPUTATIONAL SETUP**

**PROTOTYPE NOZZLES** – The prototype nozzle was fabricated by inserting a swirler into a commercial MPI nozzle (KEFICO EV 1.3 model). Evolution of test nozzles is summarized in Fig. 1. Three types of nozzles were made. A base nozzle was the first version and employed a needle of the same feature as that in the MPI nozzle. A ball type nozzle was the second version, which was made use of in validation of computation. The final version was a pin type nozzle. All research concentrated on this configuration except for the validation process and comparison study with the MPI nozzle. Fig. 2 shows typical images of sprays from this nozzle.

The body of a prototype injector was also modified from the body of the EV 1.3 MPI injector as shown in Fig. 3. It was designed as an adjustable injector to easily exchange components such as a solenoid coil, a spring, a fuel supply connector and so on. All parts were adjusted to operate under high pressure conditions up to 7MPa.

**SPRAY TEST SYSTEM** – An Injector operation and spray measurement system is shown in Fig. 4. Photographic image method was used for measuring the drop size in spray. A spark light was utilized as a light source to freeze moving particles. Image was grabbed by a CCD camera and saved in a personal computer through an image grabber. Information of spray in the captured image, such as drop size, liquid fuel mass in measured volume, was evaluated through image processing [12]. Calibration of droplet measurement was carried out using a reticle prior to main experiments.

Overall spray image was also obtained by adjusting the optical magnification factor in the same system. Back light method was used for spray image grabbing with a stroboscope as a light source.
Liquid fuel was pressurized by means of both the high pressure nitrogen cylinder and the high pressure fuel pump. When it is required to operate in a static condition for a long period, the nitrogen cylinder connected with a high pressure fuel tank was used as a pumping tool considering durability and stability.

COMPUTATIONAL CONDITIONS – Fig. 5 shows the inside of the pin type nozzle and the grid setup of the metering part. Circled region in the nozzle was analyzed because the internal pressure and the flow fields dominantly start to change just from the upstream of the swirl port inlet. Because the swirl ports were placed in four corners at intervals of 90 degrees in the nozzle, only a quarter of the total nozzle was considered by imposing the cyclic boundary conditions on both side sections. Prior to the main computation, the effects of grid density, turbulent intensity and analyzed region of the nozzle part were examined [13]. CFD-ACE+ 4.0 was used as a computational code.

Governing equations of internal flow consists of the continuity and Navier-Stokes equations. They can be written in Cartesian tensor form as follows:

\[
\frac{\partial u_i}{\partial x_i} = 0
\]  

\[
\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho \overline{u_i u_j} \right]
\]  

The term \( \rho \overline{u_i u_j} \) is the Reynolds stress in the above equation. Closure of these equations was achieved using the standard k-ε model owing to its simplicity and robustness, which is based on Launder and Spalding [14]. Because it is a high Reynolds model and does not intend to be used in the near-wall regions where viscous effects dominate the effects of turbulence, wall functions are combined with it.

Computational conditions are summarized in Table 1. Turbulent intensity was set to 3%, which is adequate for internal flow [15]. Fluid was n-heptane similar to gasoline fuel in its properties. All cases were analyzed under singe-phase flow assumption although air core is developed in the orifice and cavitation may occur at the edge or throat in real swirl injection flows. Computations on the influence of nozzle design parameters were mainly carried out under steady-state condition to examine the static flow rate and initial flow rate. For the case of transient flow variation during early or late period of injection, moving grid utility was applied to simulate needle motion.
Fig. 6 shows the geometric parameters of the prototype nozzle. An upper needle-seat angle (θ1), a lower needle-seat angle (θ2), a needle-seat contact diameter (Dc), a needle lift (Ln), an orifice diameter (Do) and an orifice length (Lo) were chosen as the parameters. Other parameters such as a needle diameter, a seat angle, swirl port geometry and so on could be considered but not presented here. One of the reasons was that the analyzed parameters were more easily adjustable ones for the existing test nozzle.

VALIDATION OF COMPUTATIONS – The validation of computation was executed in terms of the static flow rate and initial spray angle. Fig. 7 shows the static flow rate variation with respect to the needle lift. Measurement was carried out under injection pressure of 0.3MPa due to the limitation of a measuring tool. Comparison exhibited an error level less than 10% between measurement and computation. When it was measured under injection pressure of 5MPa with the needle lift of 60 µm, it was found to be 6% higher than that by computation.

Initial spray angle is usually called theoretical spray angle because it is influenced not by external factors such as ambient air flow but by internal factors such as nozzle geometry and upstream pressure. Cousin et al [16] introduced the method to predict it through the computed velocity profile at the orifice outlet of the HPSI. In this study, to reduce the influence of grid setup, the previous equations were slightly modified as below:

\[ 2\theta = 2 \cdot c \cdot \tan^{-1}\left(\frac{W}{U}\right) \]

(3)

where \( U = \frac{\dot{q}}{\pi \rho (r_o^2 - r_{ac}^2)} \) and \( W = \int_{r_{ac}}^{r_o} w_dr \)

Comparison between this computation and measurement is shown in Fig. 8. The value of a correction factor (c) for constant velocity profile assumption was 0.8, which is larger than 0.556 [17] for continuous swirl spray.

RESULTS AND DISCUSSION

STATIC FLOW RATE – Fig. 9 shows the variation of the static flow rate with respect to design parameters. The upper needle-seat angle (θ1) and the lower needle-seat angle (θ2) were found to have little influence on the static flow rate compared to other parameters. As θ1 increased, the static flow rate converged. On the other hand, the increase of θ1 causes the expansion of non-swirl volume [18], which harmfully influences initial transient flow as shown in Fig. 10. The figure was obtained with changing θ1, by unsteady flow simulation without moving grid utility, i.e., only with imposing initial condition on each domain under needle lift of 60µm. Fuel portion near the ‘first peak’ in the flow rate plot is usually injected without sufficient rotational momentum. This portion must be reduced, because it will form a core spray in the middle of a HPSI spray, which has large droplets with high penetration speed. Additionally, this peak has injurious influence on linearity of dynamic flow rate with short injection duration. Therefore, θ1 is set to be less than 5° in the current commercial nozzles. Computational method is often exploited to minimize or eliminate such influence [10].

Table 1. Computational conditions

<table>
<thead>
<tr>
<th>Operating fluid (n-heptane)</th>
<th>( \rho = 685 \text{kg/m}^3 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent model</td>
<td>( \kappa - \varepsilon ) (turbulent intensity: 3%)</td>
</tr>
<tr>
<td>Inlet &amp; outlet</td>
<td>Constant total pressure</td>
</tr>
<tr>
<td>Ambient pressure</td>
<td>0.0MPa</td>
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</table>

Figure 6. Design parameters of pin type nozzle

Figure 7. Comparison of the static flow rate of measurement and calculation in ball type (up) and pin type (down) nozzles
Figure 8. Comparison of initial spray angles of measurement and calculation

When $\theta_2$ is wide, a vortex can be generated under the needle. It perturbs the flow field, and reduces flow rate [19]. The result showed this tendency as shown in Fig. 9. The increase of $\theta_2$ also increases sac volume. It is recommended that the value of $\theta_2$ should not exceed 20° in the test nozzle.

The most important parameter influencing the static flow rate is found to be the orifice diameter ($D_o$). In fact, the needle-seat contact diameter ($D_c$) and the orifice length ($L_o$) had little effect on the static flow rate because they are set to 1.2~1.5μm and 0.2~1.5μm, respectively, in a real situation. If they exceed these ranges, problems such as excess solenoid load or strength can occur. On the other hand, the applicable range of $D_o$ is 0.4~0.9μm. In this range, the static flow rate increased almost linearly with the orifice diameter. In real application, $D_o$ decreases with the increase of injection pressure to meet the static flow rate requirement.

Considering the limitation of solenoid force and needle behavior, the majority of injector manufacturers set the needle lift ($L_n$) to around 60μm. When the fabrication error of $L_n$ directly affects the static flow rate, there can be serious problems in mass production. The result implies that $L_n$ of 60μm is not so critical to the fluctuation of the static flow rate with the fabrication error of a few μm.

Figure 9. Influence of various design parameters on static flow rate
TRANSIENT FLOW RATE – A transient flow characteristic of the applied injector is closely related with the combustion stability of GDI engines. In order to make adequate mixture in GDI engines, the precise control of fuel mass is required, especially for the case of lean burn with small quantity of fuel. The precise control of small quantity injection is ruled by the linearity of injected mass with respect to short driving signal duration. The needle motion and the evolution of internal pressure field at the initial stage of injection usually cause non-linearity for short duration injection. Another point is that the rapidly penetrating core spray including large droplets is generated at this initial transient period, as stated in the previous section.

Fig. 11 shows the needle motion for the EV 1.3 nozzle when it is operated without fluid, and the assumed needle motion is used for moving grid simulation. Most of bouncings are eliminated with liquid fuel [20]. In the case of unsteady calculation with a non-moving grid in Fig. 9, the needle opening time can be assumed as 0ms. In this case, the needle is assumed to rise up vertically as shown in Fig. 11.

Fig. 12 shows the evolution of the internal total pressure during the needle opening time. Axial and rotational velocity fields in the orifice outlet at 0.06ms after injection start are shown in Fig. 13. Both the velocity exhibited inhomogeneous fields in a peripheral direction, and this inhomogeneity remained by 0.1ms. The non-uniform fuel mass distribution in lateral cross section as examined by Yu et al [21] is believed to be generated during this period.
Fig. 14 shows needle motions and calculated transient flow rates under short duration injections. Comparison with real dynamic flow rate is shown in Fig. 15. The result implies that the trend of dynamic flow for the used prototype nozzle is quite linear even under short duration condition. The influence of the needle opening time on initial transient flow is exhibited in Fig. 16. The case of 0ms opening time means that the needle opens instantaneous. Simulated needle motion of this case is shown in Fig. 11. As the needle opening time shortens, the 'first peak' increases, and it has an unfavorable effect on the linearity of dynamic flow.

As seen in the static flow rate calculation, $\theta_1$, $\theta_2$ and $D_c$ also had little influence on the initial spray angle, and the most significant parameter was $D_o$. In the case of $L_o$, a dramatic change was found when $L_o$ was shorter than 0.2$\mu$m. However, because such length does not make sense in real situation as stated in the previous section, the effect of $L_o$ can be neglected.

In addition to these design parameters, other parameters such as seat angle, needle diameter, swirl port area and so on, can be considered. When needle diameter of the prototype nozzle was increased from 2$\mu$m to 2.5$\mu$m, the static flow rate reduced by 14%, and the initial spray angle increased by 9%. Ren et al [9] presented that the spray angle increased with the decrease of swirl port area.
OVERALL SPRAY ANGLE – When the injection pressure is fixed and ambient pressure varies, the initial spray angle is fixed and the overall spray angle varies in general because of ambient air interaction with the spray. The overall spray angle, which is called simply spray angle in general, was investigated by measurement. Although the effective spray angle [22] is used for continuous swirl sprays, the definition of spray angle [23] for diesel sprays, which are intermittent as same as HPSI sprays, were used in this study. Fig. 18 shows the variation of the overall spray angle with respect to the injection pressure and ambient pressure. As the injection pressure increased, the overall spray angle increased by some extent, and then, decreased slowly and stabilized, which is similar to that for a simplex swirl atomizer [24]. The decrease of spray angle is mainly caused by ambient air entrainment into the spray. Ambient air entrainment is strengthened with the rise of both the injection pressure and the ambient pressure.

Figure 18. Spray angle variation of the pin type nozzle with respect to injection pressure (up) and ambient pressure (down)

PENETRATION – Fig. 19 shows the development of spray tip penetration and width with respect to operating parameters. Evolution in both directions appeared to have a similar trend. Since the HPSI injection is intermittent, injected fuel mass, which is proportional to the injection duration, influences the spray tip penetration. Measurement showed that the pattern of spray tip penetration was not changed by the injection duration. However, it had an effect on the maximum dispersion of sprays as shown in the figure. When the amount of injected fuel mass was less than 15mg (injection duration of 2ms) per injection, spray tip did not penetrate far. It seemed to be caused by the lack of drop momentum,
rapid evaporation of droplets and air motion. The smaller amount of injected fuel will lose its momentum and vaporize more quickly because more ambient air will contact each droplet. Additionally, air entrainment motion, which helps the penetration of small drops swept by this air stream around the center [10], is weak.

Initial spray tip penetration increased with the injection pressure. However, penetration speed diminished more rapidly as injection pressure increased. As a result, when injection pressure was higher than 2MPa, sprays did not penetrate any further. Penetration speed decreased with ambient pressure due to the increase of air drag force.

ATOMIZATION PERFORMANCE – In order to examine the influence of swirl generation on atomization performance, the pintle type MPI nozzle was compared with the base prototype HPSI nozzle of same geometry and the static flow rate of corresponding level. Measurement was executed up to 2MPa due to the limitation of solenoid force. Overall reduction of mean drop diameter was found to be about 35%. Both sprays exhibited that the drop diameter was proportional to the square root of injection pressure, which coincides with Decorso’s results as below [25].

$$SMD = \frac{k}{\sqrt{\Delta P_{inj}}}$$  \hspace{1cm} (4)

Figure 20. Comparison of SMD to injection pressure between the commercial pintle-type MPI nozzle and the base prototype swirl nozzle

Figure 21. Prediction of SMD with respect to injection pressure by obtained empirical equations

Figure 22. Drop size variation under injection pressure of 5MPa and ambient pressure of 0.0MPa (left) and 0.4MPa (right)
A constant \( k \) represents a nozzle constant of an injection nozzle. When the derived correlation was stretched out to higher pressure conditions as shown in Fig. 21, a value of 21.5 \( \mu m \) was obtained as the spray SMD of the prototype nozzle at 5MPa of injection pressure. The real measured SMD was found to be 22.2 \( \mu m \). The obtained correlation implies that an injection pressure of up to 10MPa is required in order to get SMD of 15\( \mu m \).

Figure 23. Overall spray SMD with respect to ambient pressure

The spray drop diameter was measured with varying ambient pressure in the time scanning mode [26] with measuring points composing a line array of 30\( \mu m \) at 30\( \mu m \) below the nozzle tip. In this measuring mode, delay time is varied, while measuring points are fixed. Injection pressure was kept at 5MPa while ambient pressure varied form 0 to 0.4MPa. Results under ambient pressure of 0MPa and 0.4MPa are shown in Fig. 22. Small droplets were concentrated on the center region as revealed by Yamauchi and Wakisaka [27]. Fig. 23 shows the summed-and-averaged mean diameter with respect to ambient pressure. The spray SMD ranged from 20\( \mu m \) to 28\( \mu m \). It increased with ambient pressure. As spray field area becomes narrower, coalescence occurs more actively than the secondary drop break-up.

To understand the tendency of mean drop diameter to spray angle, simple model was introduced applying the general knowledge that mean drop diameter is proportional to the square root of the liquid sheet thickness [28]. Fig. 24 shows the simplified mechanism of liquid sheet formation. The length \( L \) indicates the arbitrary standard position for considering sheet thickness from the nozzle tip, that is, a sort of characteristic length somewhat meaningless. The cross section area of the initial sheet and the standard sheet can be defined as below.

\[
2\pi L \sin \theta = \pi (r_o^2 - r_{ac}^2)
\]  

(5)

Considering the fact that only a spray angle influences the mean diameter,

\[
SMD \propto \sqrt{t} \propto \sqrt{\sin \theta}
\]  

(6)

From the literature survey, measurements and simple analysis, atomization trend with respect to influencing factors can be summarized into an empirical correlation as below.

\[
SMD = \frac{k_1 ((P_2)^{k_2})}{\sqrt{\Delta P_{inj} \cdot \sin \theta}}
\]  

(7)

DROP SIZE DISTRIBUTION – Fig. 25 shows volume fraction drop size distributions for the selected experimental conditions, and Fig. 26 shows cumulative volume distribution with respect to normalized drop diameter by mass median diameter (hereafter MMD). Rosin-Rammler function was well-fitted.

\[
Q = 1 - \exp \left\{ -0.693 \left( \frac{D}{MMD} \right)^{y} \right\}
\]  

(8)
The constant $q$ indicates the uniformity of droplet size and was found to be 2.0 in this study. It is larger than the value of 1.724 in the air-assisted fuel injector [26]. It means that the spray drop size distribution of the HPSI is more uniform than that of the air-assisted fuel injector. Under this drop size distribution, $D_{v0.9}$ is 1.823 times of MMD. $D_{v0.9}$ was found to be 46.3 $\mu$m under the atmospheric ambient condition in this study. It is larger than a required value of 45 $\mu$m [11]. In order to reduce it, increasing spray angle or injection pressure is recommended.

CONCLUSIONS

Prototype HPSI nozzles and an injector body for GDI were fabricated, and their characteristics with respect to various design and operating parameters were investigated using both the measurement and the calculation. From this study, several conclusions were obtained as below.

1. The most important parameter for the static flow rate was found to be the orifice diameter.
2. Linearity characteristic of the dynamic flow in short injection duration was found to be influenced by the upper needle-seat angle and the needle opening time. The non-uniform fuel mass distribution in lateral cross section is believed to be generated during the initial transient period.
3. Initial spray angle was stabilized with respect to injection pressure, when injection pressure increased over 2MPa. The most significant parameter was found to be the orifice diameter as same as the static flow rate.
4. Overall spray angle reduced with the increase of both the injection pressure and ambient pressure.
5. When small quantity of fuel was injected, maximum penetration depth increased with injected fuel mass. When injection pressure was higher than 2MPa, sprays did not penetrate any further. An empirical correlation describing the influence of injection and ambient pressures was introduced.

6. Mean drop diameter of the HPSI spray decreased with the increase of injection pressure and spray angle, and decrease of ambient pressure. An empirical correlation showing this trend was introduced.

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NOMENCLATURE AND ABBREVIATION

c: correction factor for initial spray angle prediction
D: diameter
Dc: needle-seat contact diameter
Do: orifice diameter
Dv0.9: Drop diameter of 90% volume point
GDI: gasoline direct injection
HPSI: high pressure swirl injector
k: nozzle constant
k1, k2: coefficients for drop diameter prediction
L: length
Ln: needle lift
Lo: orifice length
MMD: mass median diameter
MPI: multi-point injection
P: pressure
Pamb: ambient pressure
Pinj: injection pressure
Q: volume fraction
qm: mass flow rate
rac: air core radius
ro: orifice radius
Sp: penetration depth
SMD: Sauter mean diameter
Tj: injection duration
t: thickness
U: axial velocity
ui, uj: velocity
W, w: rotational velocity
xi, xj: coordinate
Δ: differential
µ: kinematic viscosity
µl: laminar viscosity
θ: spray angle
θ1: upper needle-seat angle
θ2: lower needle-seat angle
ρ: density
ρa: ambient air density