Initial Development of Non-evaporating Diesel Sprays in Common-rail Injection Systems

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Transient nature of common-rail diesel sprays was characterized at the initial stage of fuel injection, the effects of injection rate being taken into account. Direct photography and shadowgraph technique were used to obtain macroscopic spray images at several different experimental conditions. Highly versatile image processing software was developed to analyze the spray images. In the analysis, spray penetration, dispersion and spray angles were defined and accurately measured by the software with consistency. Discussed in detail were the effects of rail pressure and ambient gas density upon spray behaviours. Based on the measured injection rates, the asymptotic relations of spray penetration were found at early and late stage of injection. The changes of spray volume were correlated with the measured injection rates.

It was found that the initial spray penetration is well correlated with the amount of fuel injected. The increase of injection rate during injection period was observed to induce smaller spray angle.

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INTRODUCTION

Direct injection diesel engines have recently been developed for small passenger cars aiming for the advantages of fuel economy and low CO₂ emission. To produce the sufficient amount of power, small engines need to be operated at high speeds or to incorporate power-boosting techniques such as turbocharging. With the increase of engine speed, however, the ignition delay in crank angle degree becomes larger. The relative increase of ignition delay within a cycle causes the decrease of the time available for complete combustion. Moreover, the fuel injected during the ignition delay is known to be a source of diesel noise and NOₓ emissions. Thus, it is necessary to reduce the ignition delay.

Common-rail fuel injection systems that are widely used for small direct injection diesel engines can inject a small amount of fuel during the pilot injection period prior to the main injection. Therefore, the pilot injection of common-rail system can alleviate the ignition delay problem that usually occurs in the high-speed direct-injection (HSDI) diesel engines. Furthermore, the injection pressure of common-rail can be set in a broad range; up to 1350 bar in this study, independently from the engine speed and load. Thus, the common-rail injectors can be operated with more flexible injection rate control that could not be attained by the conventional injection systems.

The fuel supplying pressure of common-rail to the injector ideally remains constant during the injection period. However, the consequent injection rate curve becomes triangular or trapezoidal, since the injector needle still moves gradually during the injection period.

Many investigators have studied diesel sprays under quasi-steady or constant injection rate conditions. The mostly cited spray penetration correlations of Dent [1] or Hiroyasu and Arai [2, 3] adopted the concept of potential core or intact liquid core, respectively. Those correlations imply that eventually, spray tip velocity is inversely proportional to the penetration after break-up occurs. The effect of supplying pressure change and consequent axial momentum change during injection period was not considered. Naber and Siebers [4] quantified the effect of ambient gas density on penetration and dispersion of diesel sprays with a common-rail injector, where injection rates were maintained nearly constant with very rapid start and end of injection (a “top hat” profile). Non-dimensional penetration correlation was suggested based on the control volume analysis of simple spray model and experimental data. Heimgärtner and Leipertz [5] investigated diesel
sprays during the quasi-stationary phase of the injection period at full needle lift using a common-rail injection system for passenger car. The injection durations were more than 3 ms in every injection conditions. With highly magnified spray images at the vicinity of nozzles, they investigated the effect of common-rail pressure and nozzle type on macroscopic and microscopic spray characteristics. The constant injection rate condition is different from the reality but useful to study the effects of injection pressure on macroscopic and microscopic diesel spray characteristics.

A few studies have dealt with the effects of injection rate changes on diesel spray behaviour or structure. Chaves and Obermeier [6] performed numerical modelling to investigate the effects of modulation of injection velocity on diesel spray structure. In the initial period of injection where the injection rate increases with time, the foregoing fuel is caught up with later injected fuel due to its lower speed. As a result, a moving free stagnation point and unstable lamellae are formed. Arai and Amagai [7] studied the diesel sprays from multi-stage injection. Discussed were the effects of temporal spacing between injections on the spray volume and equivalence ratio. An interesting observation was that the momentum difference between two successive sprays resulted in either coalescence or further separation of the sprays.

Sprays with changing injection rate were characterized by authors in previous papers [8, 9]. Lacking of analysis, spray visualization results were presented for mechanical and common-rail injection system, respectively.

In this paper, diesel spray behaviour from the common-rail injection system was analyzed with macroscopic spray images and the results were discussed in correlation with injection rate profiles. With extensive analysis of the data, the behaviours of initial spray development were characterized at various ambient conditions.

**EXPERIMENTAL APPARATUS**

**Experimental setup**

A schematic diagram of optical diagnostic system for spray visualization is shown in Fig 1. Spray images were taken by direct photography of Mie-scattering and shadowgraph techniques. Most of the presented data
except Fig. 14 were obtained from Mie scattering images. Although not shown in this paper, spray penetration and angles obtained from shadowgraph images were very similar to those from Mie scattering images because of non-evaporating ambient condition. The major components of experimental setup consist of a pressure chamber, a common-rail injection system, optics, an image acquisition system, and a computer-controlled signal generator. Optical access to the pressure chamber was available through four window glasses of 80mm diameter. The chamber was pressurized with nitrogen at room temperature.

The common-rail injection system was controlled by a programmable ECU emulator. Induction voltage across the solenoid of injector was monitored to obtain reference trigger signals for the imaging and data acquisition system. A five-hole valve-covered orifice (VCO) nozzle of 0.144 mm hole diameter was used for experiments. A five-hole sac nozzle, whose orifice diameter was 0.146 mm, was also tested for the comparison with the VCO nozzle. Figure 2 shows the geometry of two different types of nozzle. Since the dripping of the fuel remaining in the nozzle sac after injection was known to be a source of particulates or unburned hydrocarbon emissions, the design of the VCO nozzles were motivated to minimize sac volumes. The typical streamlines of fuel flow incoming the nozzle orifices are sharper in VCO nozzles than sac nozzles. Therefore, the discharge coefficients of VCO nozzles become smaller than sac nozzles. In case of small needle lift, moreover, VCO nozzles with single guided needle may cause non-symmetric spray patterns due to unbalanced pressure distributions near the holes [8].

For direct photography of Mie-scattered light, the spark light was directed to sprays through a window (the left window), while a CCD camera captured scattered light through the perpendicular window (the front window) as shown in Fig.1. A mirror was attached on the opposite side of the lighting window (the right window) so that the reflected light can uniformly illuminate the sprays. A typical spray image taken at the front window is shown in Fig. 3(a). About one microsecond of exposure time enabled freezing macroscopic images of sprays. The images were captured by a frame grabber that was synchronized with the spark light source. The delay of triggering signal for the frame grabber and the spark light source was adjusted from the start of injection with 10µs resolution to observe spray development in detail. For shadowgraph imaging technique, a continuous Ar-ion laser and an ICCD camera were used. A circular mirror, whose center part was drill-holed for mounting nozzle, a half-transparent mirror and beam-expansion optics were aligned along the beam path. This represents a typical setup of double-path shadowgraph. Both incident and reflected laser
beams pass through test sections. The gating time of ICCD camera determined an exposure time of 70 ns. Shadowgraph visualizes the interfaces with different density, while direct photography records Mie-scattered light. Thus, shadowgraph can show both gaseous and liquid phase of fuel, while direct photography shows only liquid phase of spray. For ensemble averages of the data analyzed from the images, six consecutive sprays were captured at the same timing. The maximum viewing area was limited by the chamber windows, whose diameter was 80 mm.

Experimental conditions were set to simulate the operating conditions of practical diesel engines. The high-pressure pump was operated at 600 rpm, which corresponds to 1200 engine rpm. The amount of a single fuel injection used in this study was determined at 25 % and 50 % of the maximum load, which were 10 and 20 mm³, respectively.

**Image Analysis**

Major parameters for spray behaviours were quantified using purpose-built image processing software. Figure 3 shows an example of the image processing. In the analysis, the background image taken before each injection was subtracted from macroscopic image to discern the spray region. Then the histogram of the gray levels in image was evaluated as displayed in Fig. 4(a). Each column indicates the number of pixels with respect to the corresponding gray level. In general, two peaks were observed, where the first peak corresponds to the dark background and the second one to spray region. The gradients of the histogram was calculated at each gray level between two peaks, and the gray level, which corresponded to the minimum absolute gradient was selected as the threshold gray level for generating binary image. For the typical histogram shown in Fig 4(a), which corresponds to the spray image in Fig. 3(a), the flat region of the valley between the peaks appears from the gray value of 70 to 90. When threshold gray level changed from 70 to 90 (suggested threshold level by the software was 75), the variation of spray penetration was less than 1.5% while that of geometric spray angle was 7~30%. By identifying the coordinates of spray periphery, the multi-spray images obtained from a five-hole nozzle could be analyzed efficiently. Geometric centroids of sprays were evaluated to identify the spray axis and to find a geometric spray angle. Figure 4(b) shows geometric definitions to characterize the spray as follows;
Spray penetration ($S$): The maximum distance from the origin (nozzle tip) to spray periphery.

Spray area ($A$): The projected area enclosed by the spray periphery.

Geometric spray angle ($\theta_{geo}$): The angle between two lines connecting the origin and half penetration points on the spray periphery.

Spray cone angle ($\theta_{cone}$): The angle given by an imaginary spray cone that possesses the same magnitude of cross-sectional area to the spray area ($A$), when the height of spray cone is spray penetration ($S$).

\[
\theta_{cone} = 2 \tan^{-1} \left( \frac{A}{S^2} \right) \tag{1}
\]

Spray volume ($V_s$): The volume of imaginary spray cone.

\[
V_s = \frac{\pi}{3} \tan\left(\frac{\theta_{cone}}{2}\right) S A \tag{2}
\]

Spray dispersion ($V_s/V_f$): The ratio of spray volume to the fuel volume contained in the spray.
RESULTS AND DISCUSSIONS

Injection rate measurements

Figure 5 displays the fuel injection rate profiles measured for a common-rail injector with VCO nozzle at two different common-rail pressures, 25 MPa and 120 MPa. The injection rate was measured with so-called Bosch tube method, which is in principle based on hydraulic pulse theorem [10]. In the early injection period, the injection rates measured at the same pressure condition appear to coincide each other, independent of the amount of injected fuel.

The needle lift of common-rail injectors is determined by the pressure history in the valve control chamber of injector, which occurs through interaction between the feeding and bleeding orifices and fuel supplying pressure. Although the fuel supplying pressure remains constant, the upstream fuel pressure of a nozzle hole varies during early stage of injection, because of gradual needle lift. Such a characteristic of needle lift determines the injection rate profiles as shown in Fig. 5, while the change of supplying pressure generates the typical injection rate profiles of conventional mechanical-type injectors.

Generally, needle lift is measured by modifying injectors. In most cases, the modification is complex and affects dynamics of needle motion. In an effort of tracing the needle lift, it was recently suggested that a high-power X-ray source might be used as a non-intrusive measurement tool [11]. Although needle lifts are measured, complicated internal flow characteristics including cavitation make it difficult to predict injection rates and injection velocities from needle lift data [12]. Injection rate measurement seems to be a practically effective way to estimate injection velocity changes during injection period so far. In this paper, injection rate was presented on behalf of injection velocity to avoid ambiguity about determining effective liquid flow area.

Spray structure depends on the injection velocity, which is a function of the injection pressure, the nozzle geometry, the chamber pressure etc. The identical injection rate, however, is not sufficient condition for the same injection velocity, thus care should be taken to investigate the structure and behaviour of sprays from nozzles of different nozzle hole geometry [13].

The injection rate curve obtained for the 25 MPa case represents a less steep gradient in the initial stage of
injection, compared to the higher-pressure case (120 MPa). In both cases, the profiles present several peaks in the early stage, which are distinct from the small fluctuations appearing in later injection period. These peaks are likely to be caused by the pressure waves that are generated due to the immediate injection of fuel through injection holes. Prior to a fuel injection, a high picking current (~ 20 A) is applied to the injector solenoid, while several discrete holding currents (~ 12 A) are transported during the injection. The total period of these current pulses is called the solenoid-energizing time.

The intervals between two adjacent peaks in the injection rate curve for 25 MPa become shorter with time, and the fluctuating amplitudes of peaks were damping out with time. The time interval between the first two peaks appears to be about 200 μs. Assuming the typical acoustic velocity in diesel fuel of 1380 m/s, the pressure wave generated at the onset of injection can travel 0.28 m during 200 μs. The magnitude of this distance is approximately two-fold the flow passage length in the injector. Therefore, it is suggested that the local peaks appearing on the injection rate curves could be caused by the pressure wave generated at injection start.

In the case of the fuel supplying pressure 120 MPa, the profile of injection rate in the early stage of injection was quite independent of the injection duration, similar to the 25 MPa case. The most significant peak was generated at about 100 μs after injection start. Similar results were observed in the experimental work performed by Yue et al. [14], who measured fuel mass distributions in common-rail fuel sprays using a synchrotron X-ray source. They showed that the instantaneous fuel mass measured at 1 mm from the nozzle tip rapidly increased, and then decreased till another increase of fuel mass was observed in about 120 μs.

**Correlation between the injection rate and spray penetration at the early stage of injection**

Several investigators obtained correlations for non-evaporating spray penetrations ($S$) from their experimental works [1-4, 15]. These correlations can be summarized by a formula described below as equation (3), when enough time elapsed from start of injection (SOI) at room temperature.

$$S = K(\Delta P / \rho_g)^{0.25}(D_o t)^{0.5} ; t >> t_0 = (\rho_f / \rho_g)^{0.5} D_o / V_{inj}$$

(3)
Where, $\Delta P$ is the pressure difference between inlet and exit of a nozzle hole whose diameter is $D_o$. The density of fuel and ambient gas are represented as $\rho_f$ and $\rho_g$, respectively. $t$ is the time elapsed from SOI. The characteristic time, $t_0$ is a time scale for evaluating spray development, which becomes shorter as fuel injection velocity ($V_{inj}$) and ambient gas density ($\rho_g$) increase.

The correlation for spray penetration in the works of Hiroyasu and Arai [2, 3] has the form of equation (3) after 15.8 $t_0$ elapsed. Naber and Siebers [4] published non-dimensional correlation for spray penetration. They included half spray angle in characteristic time. The penetration correlation seems to coincide with long time limit form described by square root of non-dimensional time after 30 or more non-dimensional time elapsed.

In the equation (3), K is a coefficient that has different values depending on experimental conditions. This correlation can be obtained by applying momentum conservation along spray axis, with the assumptions that the spray angle remains constant along the spray axis and the spray density approximates that of ambient gas. Momentum is not conserved along spray axis while injection rate is increasing. After the maximum injection rate appears, however, no momentum exchange would take place at the spray front due to following fuels of higher injection velocity. The constant spray angle actually means a linear growth of spray width with spray penetration rather than unchanging geometric angle through injection period. When injection occurs in diesel engine, typical ambient gas density is about 30 kg/m$^3$ and the density of fuel is about 840 kg/m$^3$. However, the portion of ambient gas entrained into the spray increases with spray penetration. In this study, the volume of entrained gas in the spray became more than thousand times the volume of injected fuel within 50 mm of spray penetration. After all, spray density reaches that of ambient gas as it develops. Therefore, equation (3) could be used to estimate an asymptotic line of spray penetration at infinite time from SOI in case of unsteady injection condition.

The prediction of spray penetrations in the early stage of injection is not straightforward even for the cases of quasi-steady state, because it requires detailed information for the internal spray structure and atomization mechanism. Arai et al. [3] obtained the correlation for the initial spray penetration as equation (4), based on the hypothesis that fuel is injected as a liquid jet and disintegrated by the unstable growth of surface
Yule and Filipovic [15] pointed out that transition is rather gradual than showing clear break-up between nozzle and downstream spray. Single correlation that has hyperbolic tangent function in addition to equation (3) was derived to regress the spray penetration data. Recently, Naber and Siebers [4] formulated the correlation of spray penetration in a non-dimensional form, which was obtained by the mass and momentum conservation equations under the assumption of constant spray angle. As time approaches to zero, the correlation becomes identical to equation (4) except the experimental coefficient of 0.39 is replaced with a velocity coefficient $C_v$ (0.68–0.76).

Figures 6 and 7 display the early spray penetration data measured for a common-rail VCO nozzle with injection rate profiles in Fig. 5. The data represented with hollow circle were obtained for the common rail pressures of 25 MPa and 120 MPa at the ambient gas density of 1.23 kg/m$^3$. The data of each condition were well regressed with following equations respectively.

\[
S = 0.39 \sqrt{\frac{2 \Delta P}{\rho_f}} t ; t < \frac{28.65 \rho_f D_0}{(\rho_g \Delta P)^{0.5}} \quad (4)
\]

\[
S = 1831.8 t^{1.414} \quad at \quad P_{\text{common-rail}} = 25 \text{ MPa}
\]

\[
S = 19008.2 t^{1.544} \quad at \quad P_{\text{common-rail}} = 120 \text{ MPa} \quad (5)
\]

$S$: liquid penetration (m)

$t$: time elapsed after start of injection (sec)

The magnitudes of time exponents are greater than unity, which reveals that the velocity of spray front is increasing due to injection rate increase at the early stage of injection. As discussed above, conventional spray correlations for constant injection rate are becoming almost linear at the early stage of injection. Spray penetrations are calculated by integrating the velocity of spray front with respect to time. In case of the
constant fuel injection velocity, therefore, the spray penetration should increase linearly with time. The inherent difficulty of estimating spray penetration under the changing injection rate is on the understanding of momentum exchange between preceding and following fuels. Two extreme cases could be considered. First, dense spray core was assumed as an incompressible flow where air entrainment into the spray was negligible at the start of injection.

This assumption is valid in the sense that the injection pressure is by far lower than common-rail pressure at the initial stage of injection, because of throttling in the opening between the needle and nozzle. Therefore, the acoustic velocity of fuel is much higher than the injection velocity and consequently, injected fuel flow can be considered as an incompressible flow. Furthermore, air entrainment into the spray increases along spray axis. Therefore, the volume of ambient air entrained into the spray near the nozzle could be neglected. Under this assumption, spray penetration could be expressed as follows.

\[ S = \int_0^t \left( \frac{\dot{m}}{\rho_f C_a A_o} \right) dt \]  

(6)

Solid lines in Figs. 6 and 7 were obtained by integrating the injection rate curves displayed in Fig. 5 and dividing it by \( \rho_f C_a A_o \) based on equation (6). Here, \( C_a \) is the area contraction coefficient and \( A_o \) is the area of nozzle hole exit. The area contraction coefficient was assumed unity for simplification. Good accordance was shown in both figures between the solid lines and the spray penetration measured from spray images.

The second case deals with the unit mass of fuel injected with velocity \( V_x(0) \) when the time is \( \tau \). If spray atomization occurs immediately for the fuel exiting the nozzle hole and no interactions are considered between unit masses of fuels that are moving at different velocities, the spray penetration could be determined by the maximum distance travelled by all unit masses at time \( t \). This idea is represented in equation (7).

\[ S = MAX_{0 \leq \tau \leq t} \left[ \int_0^{t-\tau} V_x(t) dt \right] \]  

(7)
Chaves and Obermier [6] reported the picture of diesel sprays injected into atmosphere with fluctuating injection pressure. Their result showed that the positions of four different structures in the spray were moving linearly. Moreover, the origin of each trace was well correlated with distinct peaks on injection pressure curve. This observation provides the motivation to formulate equation (7). In steady-state conditions, $V_\tau(t)$ term is constant; therefore, equations (6) and (7) are identical. The dashed lines in Figs. 6 and 7 represent the data calculated based on equation (7) with injection rate curve in Fig. 5.

If the initial phase of injection rate is approximated as $\dot{m}(t) = Kt$, then Eqs. 6 and 7 become $S = Kr^2 / 2 \rho_f A_o$ and $S = Kr^2 / 4 \rho_f A_o$, respectively. However, the differences between the solid lines and dashed lines in Figs. 6 and 7 were not so significant because of ripples on the measured injection rate profiles.

In summary, the solid lines obtained from the first assumption showed reasonable agreements with the spray penetration measured with spray images at the early stage of injection, regardless of common-rail pressure.

Figure 8 shows the spray penetrations (hollow circles) and smoothed injection rate profile in arbitrary unit. The data were acquired for the sprays injected from a VCO nozzle at a typical ambient gas density of diesel engine. At the beginning of this paragraph, it was discussed that the liquid penetration is expected to follow the equation (3) as spray develops with ambient gas entrainment. The dashed line in Fig. 8 represents Hiroyasu’s correlation [2] for spray penetrations after the spray break-up occurred. It was chosen as an example among other correlations that have the form of equation (3). Experimental data were well represented by the correlation after maximum injection rate occurs. The solid line in Fig. 7 was redrawn in Fig. 8, which presented the initial spray penetrations at atmospheric ambient condition. As shown in Fig. 7, at atmospheric ambient condition the experimental data were in good accordance with the solid line until spray fronts penetrated over the viewing limit that was about 40mm. At elevated ambient pressure condition in Fig. 8, however, spray penetrations began to deviate from the solid line after 70~80 $\mu$s from start of injection. For experimental condition of Fig. 8, the characteristic time $t_o$ was about 2 $\mu$s at maximum injection rate point, thus 15.8 times of characteristic time was about 32 $\mu$s. It seems that spray penetrates according to the square root of time after this amount of time is elapsed from maximum injection rate point.

This paper deals with non-evaporating diesel spray injected into quiescent environment in conjunction with
injection rate profile. In real engine condition, however, evaporation characteristic of fuel and the effect of swirl flow should be considered additionally for describing spray trajectory.

**Effects of injection rate on spray volume**

Figure 9 shows that both geometric spray angle and spray cone angle change with spray penetration at different common-rail pressures; 25 MPa and 120 MPa, and ambient gas densities; 1.23 kg/m$^3$ and 33.8 kg/m$^3$. Spray angles in (a) and (b) seemed to decrease with spray penetration increase. Spray angles in (c), however, were observed to increase with spray penetration. The different tendency between (a), (b) and (c) suggests that spray penetration is not an exclusive parameter for spray angle changes. The correlation between spray angles and injection rate curve was shown in Fig. 10. It should be noted that spray angles in (a) and (b) were measured at early stage of injection while injection rates were increasing. On the contrary, spray angles in (c) were obtained at late stage of injection while injection rate was decreasing. To assess the degree of linear dependence between injection rate and spray cone angle, a correlation coefficient was calculated. The correlation coefficient varies on a scale from –1 to +1; zero means there is no correlation between two variables and negative value means the relationship is inverse [16]. The correlation coefficients of (a), (b) and (c) were –0.38, -0.4 and –0.1 respectively. Consequently, injection rate increase resulted in spray angle decrease and injection rate decrease led overall increase of spray angle even with weak correlation.

The spray volume change according to injection rate profile could be discussed with the microscopic images shown in Fig. 11. The microscopic spray images were taken with long distance microscope and nano-pulse light source of 10ns duration. The sprays were injected from the 5-hole VCO nozzle with specially built nozzle cap. The magnification factor was more than six and the width of picture was about 1.3mm (nozzle hole diameter was 0.144mm). Figure 11 shows microscopic spray images near the nozzle tip at three different injection times in the initial injection period. Fuel was injected into the chamber at atmospheric pressure condition at the rail pressure of 120 MPa. Careful observation of the images revealed that the ligaments initially formed around the spray periphery disappeared with time as the spray breaks up into droplets.
Another observation was that the geometric spray angle near the nozzle decreased with injection rate. Figure 10 illustrated that spray angle measured from macroscopic spray images has negative correlation with injection rate profile. Lai et al. [17, 18] also observed that spray angle oscillated during injection period and oscillations of injection pressure, needle movement, and cavitation bubble inside injectors were mentioned as possible causes while the individual mechanisms were uncertain.

There could be several explanations about spray angle decrease with the increase of injection rate. Firstly, the disappearance of ligament at spray surface could make the spray look slimmer along the spray axis. Secondly, because the fast moving fuels penetrate into the clouds of fuel parcels proceeding slowly, the chance to entrain air into the spray becomes relatively low when injection rate increases. And lastly, it could be also explained from the analogy between diesel spray and incompressible gas jet [19]. One of the important features of incompressible gas jet is the linear increase of the volume of flow along jet axis. For laminar jet, the volume of flow at a certain distance from the orifice is independent of injection velocity as long as kinematic viscosity is kept constant. As a result, higher injection velocity leads to narrower jet. For turbulent jet, the volume of flow increases with injection velocity. This implies that the ambient gas entrainment into turbulent jet increases with injection velocity, and it can be shown that cone angle of turbulent jet is independent of injection velocity. It could be induced from the above considerations that the increase of injection velocity would result in narrower spray angle of diesel spray as far as the degree of mixing with ambient gas is not as vigorous as turbulent gas jet. Heimgärtner and Leipertz [5] also observed macroscopic spray cone angle decrease with common-rail pressure increase.

The spray dispersion, defined as the ratio between spray volume to the contained fuel volume, is presented in Fig.12. Within viewing area, the effect of ambient gas density was not clearly distinguishable. The mass ratio between total spray and fuel, however, was quite different for each case. In case of high ambient gas density ($\rho_g=33.8 \text{ kg/m}^3$), total mass of entrained ambient gas was about twenty seven times larger than the other case ($\rho_g=1.23 \text{ kg/m}^3$).

Spray volume changes calculated from spray images are plotted in Fig.13. Only spray images with the penetration of more than 10mm were processed because the spray volume might be underestimated at low penetration. Because spray volume became more than hundred times of contained fuel volume from 10mm penetration as shown in Fig. 12, volume change rate could be mostly due to ambient gas entrained into the
From the two cases of spray injected into atmospheric ambient condition, it was clear that fast moving spray mixed with ambient gas more vigorously. The volume flow rate into the spray moving in high-density gas was represented with thick lines. Difference between VCO and sac nozzle disappeared as spray penetration increased. Furthermore, the volume flow rate into the spray seems to increase as a power law ($S^{1.58}$). Differentiating this power law reveals that ambient gas entrainment rate per unit distance increase with spray penetration. It is known that the volume flow rate into the steady gas jet per unit distance is constant [19]. However, Cossali et al. published that the derivative of entrained mass flow rate with distance increased proportional to the square root of distance [20]. This result was obtained with transient diesel spray.

Spray cone angles measured at various experimental conditions are shown in Fig. 14. Solid circle represent the value from shadowgraph images and hollow squares represent the result from direct photography of Mie-scattered liquid spray region. In bottom-right of each figure, abbreviation of experimental condition is written. The first letter distinguishes nozzle type (V: VCO, S: sac), the middle one represents common-rail pressure (H: 120 MPa, L: 25 MPa), and the last letter is for ambient pressure condition (A: atmospheric $\rho_g$=1.23 kg/m$^3$, P: pressurized $\rho_g$=33.8 kg/m$^3$, V: vacuum $P_g$=6 mmHg). The experiments in this paper were done at room temperature condition where the evaporation does not seriously affect the liquid region. Therefore, the differences between the data acquired from shadowgraph and direct photography was hardly noticed. As mentioned above, spray cone angle changed inversely to injection rate profile. Spray cone angle of sac nozzle sprays were more scattered than that of VCO nozzle sprays at the same experimental conditions.
CONCLUSIONS

This paper discussed the effect of injection rate on macroscopic characteristics of diesel spray from common-rail injector suited for high-speed direct injection diesel engine. It has been observed that injection rate change resulted in different diesel spray behaviours compared to that under quasi-steady injection rate. This study found that macroscopic and microscopic characteristics of diesel spray are affected by injection rate profiles as follows.

1. Fuel injection rate with time at typical engine operating condition was triangular or trapezoidal shape. Regardless of injection duration, the gradient of injection rate curve was identical for each common-rail pressure while injection rate is increasing at the initial stage of injection.

2. Spray front was accelerated at the early stage of injection in contrast to the conventional observations for quasi-steady injection that it moves with constant velocity at the start of injection. This difference was due to injection rate changes.

3. Spray penetration at the early stage injection was proportional to the integration of injection rate curve or the injected fuel mass.

4. After maximum injection rate appears, spray penetration follows the conventional correlations describing spray penetration increases according to the square root of time.

5. There is a transition period between two different penetration trends at early and late stage of injection described in conclusion 3 and 4. Spray penetration in transition period can be estimated by two asymptotic lines obtained at early and late stages of injection from injection rate curve.

6. Spray cone angle was found to have inverse relationship with injection rate. Consequently, spray cone angle decreases during early stage of injection.

7. Microscopic images of sprays taken from nozzle hole to about $7D_o$ show that, as injection rate increases, spray width decreases and ligaments around the surface became shorter and finer.
NOMENCLATURE

$A$ : Spray area (m$^2$)

$A_{eff}$ : Effective liquid flow area of nozzle hole exit (m$^2$)

$A_o$ : Area of nozzle hole exit (m$^2$)

$C_d$ : Discharge coefficient $\quad C_d = C_a \cdot C_v$

$C_a$ : Area contraction coefficient $\quad C_a = \frac{A_{eff}}{A_o}$

$C_v$ : velocity coefficient $\quad C_v = \frac{V_{inj}}{(2(P_f - P_g)/\rho_f)^{0.5}}$

$D_o$ : Nozzle hole diameter (m)

$K$ : Proportional coefficient

$M_{inj}$ : The mass of fuel injected per hole (kg)

$m, m'$ : Fuel injection rate (kg/sec)

$P_{common-rail}$ : Pressure of common-rail (Pa)

$P_f$ : Fuel supplying pressure to the nozzle (Pa)

$P_g$ : Ambient gas pressure (Pa)

$S$ : Liquid penetration (m)

$t, \tau$ : Time after injection start (s)

$t_0$ : Characteristic time for transition (s) $\quad t_0 = (\rho_f/\rho_g)^{0.5}D_o/V_{inj}$

$T_{energizing}$ : Energizing time of the injector solenoid (s)

$V_{inj}$ : Injection velocity (m/s)

$V_s(t)$ : The velocity of unit mass of fuel injected at $t=\tau$ (m/s) $\quad [0 \leq \tau \leq t]$

$V_s$ : Spray volume (m$^3$)

$V_f$ : Fuel volume (m$^3$)

$\Delta P$ : Pressure difference between inlet and exit of the nozzle hole (Pa)

$\theta_{geo}$ : Geometric spray angle (rad)

$\theta_{cone}$ : Imaginary spray cone angle (rad)

$\rho_f$ : Liquid fuel density (kg/m$^3$)
$\rho_g$: Ambient gas density (kg/m$^3$)
References


18. Han, J., Lu, P., Xie, X., Lai, M. and Henein, A. Investigation of diesel spray primary break-up and


**Fig. 1** Schematic diagram of experimental setup for spray visualization

**Fig. 2** Geometries of sac and VCO nozzle (left: sac nozzle, right: VCO nozzle)
Fig. 3 An example of image processing
(a) A raw macroscopic spray image
(b) Background image
(c) Detected spray boundaries

Fig. 4 The definitions of spray geometry constructed by image normalization
(a) A typical histogram of normalized spray image showing two distinct peaks corresponding to background and spray, respectively
(b) A binary image showing spray boundaries with geometric definitions
Fig. 5 Fuel injection rate of a common-rail injector with VCO nozzle. The data represent the amount from a nozzle hole.

a) common-rail pressure: 250 MPa  
b) common-rail pressure: 120 MPa
Fig. 6 The initial spray penetration calculated from injection rate compared to the measured values (common-rail injector with VCO nozzle, common-rail pressure: 25 MPa, ambient gas density: 1.23 kg/m³)

Fig. 7 The initial spray penetration calculated from injection rate compared to the measured values (common-rail injector with VCO nozzle, common-rail pressure: 120 MPa, ambient gas density: 1.23 kg/m³)
Fig. 8 Spray penetration of the common-rail injector with a VCO nozzle (common-rail pressure: 120 MPa, ambient gas density: 33.8 kg/m$^3$)
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(c) $P_{\text{common-rail}}=120 \text{ MPa}$, ambient gas density=33.8 kg/m$^3$
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(b) $P_{\text{common-rail}}=120$ MPa, ambient gas density=1.23 kg/m$^3$
(c) $P_{\text{common-rail}}=120$ MPa, ambient gas density=33.8 kg/m$^3$
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A) 20 $\mu$s, B) 50 $\mu$s, C) 120 $\mu$s from start of injection
Fig. 12 Spraying dispersion of VCO nozzle with 120MPa supplying pressure at different ambient gas density.
Fig. 13 Spray volume change along the spray axis at various conditions
Acronyms

<table>
<thead>
<tr>
<th>Nozzle type</th>
<th>Common-rail pressure</th>
<th>Ambient density</th>
</tr>
</thead>
<tbody>
<tr>
<td>V: VCO</td>
<td>H: 120 MPa</td>
<td>P: $\rho_g=33.8$ kg/m$^3$</td>
</tr>
<tr>
<td>S: sac</td>
<td>L: 25 MPa</td>
<td>A: $\rho_g=1.23$ kg/m$^3$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>V: $P_g=6$ mmHg</td>
</tr>
</tbody>
</table>

Fig. 14 Spray cone angle at various conditions

(a) VCO, $P_{\text{common-rail}}=25$ MPa, $\rho_g=1.23$ kg/m$^3$
(b) VCO, $P_{\text{common-rail}}=120$ MPa, $\rho_g=1.23$ kg/m$^3$
(c) VCO, $P_{\text{common-rail}}=120$ MPa, $\rho_g=33.8$ kg/m$^3$
(d) sac, $P_{\text{common-rail}}=25$ MPa, $\rho_g=1.23$ kg/m$^3$
(e) sac, $P_{\text{common-rail}}=120$ MPa, $\rho_g=1.23$ kg/m$^3$
(f) sac, $P_{\text{common-rail}}=120$ MPa, $\rho_g=33.8$ kg/m$^3$
(g) VCO, $P_{\text{common-rail}}=25$ MPa, Vacuum(6 mmHg)
(h) VCO, $P_{\text{common-rail}}=120$ MPa, Vacuum(6 mmHg)
(i) sac, $P_{\text{common-rail}}=120$ MPa, Vacuum(6 mmHg)

● : shadowography
□ : Direct Mie-scattered imaging
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Fig. 1 Schematic diagram of experimental setup for spray visualization

Fig. 2 Geometries of VCO and sac nozzle (left: sac nozzle, right: VCO nozzle)

Fig. 3 An example of image processing
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   (b) Background image
   (c) Detected spray boundaries

Fig. 4 The definitions of spray geometry constructed by image normalization
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       background and spray, respectively
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Fig. 5 Fuel injection rate of a common-rail injector with VCO nozzle. The data represent the amount from
   a nozzle hole
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