An Investigation of the Effects of Fuel Injection Pressure, Ambient Gas Density and Nozzle Hole Diameter on Surrounding Gas Flow of a Single Diesel Spray by LIF-PIV Technique

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1. ABSTRACT

The characteristics of ambient gas motion induced by a single Diesel spray were measured quantitatively by using the Laser Induced Fluorescence-Particle Image Velocimetry (LIF-PIV) technique under the non-evaporating quiescent condition. The effects of fuel injection pressure, ambient gas density and nozzle hole diameter on the ambient gas mass flow rate into the spray through the whole spray periphery (spray side periphery and tip periphery) were investigated quantitatively according to the gas flow velocity measurements. The results show that the captured gas mass flow rate through the spray tip periphery is prominent in the whole periphery and the proportion of the gas entrainment through the spray side periphery increases with the spray development. The higher injection pressure significantly enhances the total gas mass flow rate through the whole periphery, however the increase in the ratio of ambient gas and fuel mass flow rate becomes moderate gradually with the increase in the injection pressure. The higher ambient gas density results in a slight increase in ambient gas flow velocity along the spray side periphery and the tip periphery and a reduction of the spray volume, however, the ambient gas mass flow rate was enhanced apparently. The smaller nozzle hole diameter results in the significant decrease in the ambient gas mass flow rate and the increase in the ratio of gas and fuel mass flow rate. The numerical simulation results provide more understanding of the spray induced gas flow field and validates the measurement accuracy of the LIF-PIV results.

Keywords: Fuel Spray; Diesel Engine; Ultra-high Injection Pressure; Micro-hole Nozzle; LIF-PIV; Ambient Gas Entrainment;

2. INTRODUCTION

Rapid fuel and air mixing is necessary to extend the operation range into higher load in advanced Diesel combustion modes with moderate early injection, high EGR rate and high boosting intake charge. The usage of micro-hole nozzle and ultra-high injection pressure is believed to be one of the effective ways to realize this goal [1], thus their effects on the mixture formation process deserves more comprehensive study.

Recently, the non-intrusive optical visualization methods have been widely used to investigate the Diesel spray fuel and air mixing process. Endoscope-based optical system inside an engine cylinder, PDA system and LDV system can be applied to measure the spray and surrounding gas flow velocity and droplet size [2, 3, 4]. Adrian [5] firstly developed Particle Image Velocimetry (PIV) technique for two-dimensional flow velocity measurement, and later Raffel et al. [6] systematically introduced the application of PIV technique in the experimental investigation of fluid mechanics. As for the planar measurement of spray induced gas motion, Laser Induced Fluorescence-Particle Image Velocimetry (LIF-PIV) technique has proved to be one of the most promising measuring methods [7, 8, 9]. Since only the fluorescent signal from the tracer itself can be derived without the influence of the scattered light from the spray droplets, this measuring method is feasible at the field approaching the spray periphery, even at the spray tip region where the diluted spray droplets may superpose with the tracer droplets. Furthermore, the flow velocity result from the PIV technique has higher spatial resolution compared with other techniques.

The ambient gas mass flow rate around the spray periphery was widely investigated because of its significant influence on the mixture concentration and combustion behavior. In the early time, Ricou et al. [10] and Hill [11] studied the air entrainment law of a steady gas jet by using the "porous-wall technique" and concluded that the air entrainment rate increases linearly with the axial distance after reaching the fully developed region. Kuniyoshi et al. [12] investigated the ambient gas flow field around a Diesel spray by using the smoke wire method. Cho et al. [13] also investigated the surrounding gas flow field of the Diesel spray with the injection pressure of 40MPa by means of the smoke wire method. It was shown that the "penetration part" of the jet or the spray entrains their surroundings, while the "stagnation part" of the jet or the spray pushes aside their surroundings. The ratio of the entrainment volume at the "penetration part" to the total volume of the unsteady jet or the Diesel spray is almost 50%. More recently, Sepret et al. [14] investigated the result of the local variation of fuel and air mixing rate on the side periphery of full cone fuel spray by using the LIF-PIV technique. The ambient gas velocity distribution on the side periphery of the transient Diesel spray with the injection pressure of 146MPa was also measured by using PIV technique by Rhim et al. [15]. The ambient gas flow was categorized as three regimes: (1) entrainment through side-periphery, (2) air pushed -out by head-vortex and (3) entrainment by spray-capturing at the spray tip. By comparing the volume of the non-evaporating spray and the measured mass of entrained gas along the side periphery, the author assumed that almost 60% of the total ambient gas mass in the spray is captured by the spray tip periphery. However, most of the previous studies were restricted by the lower injection pressure and in-cylinder gas density compared with the current Diesel engine conditions, or the low resolution of measurement result. Only a few works have quantitatively analyzed the velocity distribution around the whole spray periphery with various injection and ambient conditions and compared with the relevant CFD result, especially at ultra-high injection pressure and micro-hole nozzle case.

Therefore, this study is aimed at introducing a measurement method of the ambient gas velocity distribution and quantitative analysis method of the ambient gas mass flow rate around the whole periphery of the transient Diesel spray. Furthermore, numerical simulation was performed under the same condition in order to obtain better understanding of the spray induced gas flow field. The experiment was conducted in a wide range of injection pressure from the ordinary values in the current Diesel engines (100MPa, 200MPa) to the ultra-high (300MPa) value. Three representative ambient gas density conditions (11, 15, 20 kg/m^3) were selected. A micro-hole nozzle (0.08mm) and a conventional nozzle (0.12mm) were applied as the test nozzles in this study.

3. EXPERIMENTAL SETUP

3.1 Experimental Methodology

Figure 1 shows the schematic of the experimental set-up, which mainly comprises mechanical high pressure fuel injection system, optical accessible constant volume vessel, timing control unit and LIF-PIV system. The internal space of the chamber has the height of 300mm and the radius of 100mm. The sufficient chamber's volume (9.4×10^6mm^3) ensures that the free spray develops in the nearly quiescent condition. Three quartz windows with a diameter of 100mm were installed at the three sides of chamber respectively and the tracer injector was installed at the fourth side. The injection of the tracer droplets and the incidence of the laser sheet were from two opposite sides, while a CCD camera (PCO1600, PCO Inc.) was placed perpendicular to the laser sheet. Diesel fuel was injected into the high ambient gas density condition while the CCD camera captured the spray and the tracer images at a specific timing.

The LIF-PIV technique is capable of measuring the motion of the tracers very close to the spray vicinity accurately by providing high contrast tracer image and cutting off the scattering light from the high-number-density droplets in the spray. Figure 2 shows the experimental time chart. Two delay generators (DG535, Stanford Inc.) synchronized the pulses for the tracer injection, fuel injection, laser shot and image capture. Firstly, the water solution of Rhodamine B reagent which was used as the fluorescent tracing particle was injected into high pressure chamber from left side by a swirl type injector with an injection pressure of 9MPa. After a certain interval when the tracing particles had been distributed uniformly in the whole chamber (Δt_{inj}=1300ms in Fig. 2), the fuel injection pulse was triggered. At desired timing after the start of injection (SOI), an Nd:YAG laser sheet with a wavelength of 532nm and thickness of 1mm was incident from the right side of the vessel thus making the spray plume illuminated, the tracing particles emitted the fluorescent light at wavelength of longer than 590nm. Meanwhile the CCD camera mounted with the long pass filter (>560nm) was synchronized to capture two images with high spatial resolution of 1600×1200 pixels during a short time interval (Δt_{shot}=40μs in Fig. 2). The scattering signal of the spray droplets was cut off and only the fluorescent light from the tracing particles was recorded. Figure 3 shows the principle of the image capturing process using the LIF-PIV technique in the previous work by Moon et al. [9]. The two-dimensional ambient gas flow field was calculated based on the displacement of tracers in the two images by using the cross correlation algorithm. The tracing particle size, the time interval between the dual frames and the interrogation window size were carefully adjusted. Detailed information can be found in the authors' previous work [16].

3.2 Experimental Conditions

Table 1 shows the experimental conditions. The constant volume vessel was charged with nitrogen gas at room temperature (300K) to reach the desired pressure. In this study, three ambient gas densities 11 kg/m^3, 15kg/m^3 and 20kg/m^3 were used which also correspond to the typical in-cylinder ambient gas density condition at the fuel injection timing in the D.I. Diesel engine. The JIS#2 commercial Diesel fuel in the Japanese market was used as the test fuel in this work. Fuel injection duration of 2.2ms was selected to get a quasi-steady state spray. Several measurement timings during the fuel injection duration and after the end of injection were selected. Two types of single-hole nozzle with the hole diameter of 0.08mm and 0.12mm were used to investigate the effect of nozzle hole-size on the spray-induced ambient gas motion. A wide range of injection pressure from 100, 200 to 300MPa were used, which covers the current common rail diesel injection pressure and future probable ultra-high injection pressure conditions.

4. Numerical Simulation Methodology

Basic RANS (Reynolds averaged Navier-Stokes) governing equations were analyzed based on the mass, momentum and energy conservation. Standard k-ε model was used as the turbulent flow model. Basically, the fuel spray was analyzed based on Lagrangian spray module which involves the multi-phase flow phenomenon and implements the conservation equations for liquid phase and vapor phase simultaneously. The liquid phase simulation is carried out in a statistical Discrete Droplet Method (DDM) way [17, 18], in which a group of droplets with the identical properties will be represented by a "parcel". The vapor of evaporating droplets was used as a source term of an additional transport equation for the vapor void fraction in Eulerian formulation.

The CFD code AVL FIRE was used to carry out the numerical simulation of the spray induced ambient gas motion. Time step was selected as the run mode with the interval of 0.1ms. A cylindrical computational domain with the height of 100mm and the radius of 30mm was formed as shown in Fig. 4. The cell size in the central region was refined to 1mm. In accordance with the experiment carried out in the constant volume vessel, the "static wall" was used as boundary condition and the initial condition was set to 293K of ambient temperature, $0.001m^2/s^2$ of turbulent kinetic energy and no initial air flow. Table 2 shows main models used in this study. Several parameters in the sub-models, such as the break up time in WAVE model, R/D value in nozzle flow model, the discharge coefficient in core injection model, the initial droplet size and other modeling parameters, require adjustment in order that the value of spray tip penetration, spray spreading angle and equivalence ratio distribution coincide with that in the experiment.

5. RESULTS AND DISCUSSION

5.1 Velocity Vector Distribution of Ambient Gas

Figure 5 shows the examples of the spatial distribution of ambient gas velocity vector synthesized with the spray image in various typical conditions. All the gas velocity data shown in this study is obtained by averaging 5-6 samples with good repeatability. The high injection pressure keeps the spray tip an approximately hemispherical shape even near the end of injection if eliminating the shot by shot variation by averaging several spray samples. If only taking the air flow along the side periphery into account, there is no obvious change of air flow velocity when the ambient gas density rises from $15kg/m^3$ to $20kg/m^3$. The spray tip penetration decreases due to the increased air drag force, thus decelerating the spray development. The increase in both the injection pressure and the nozzle hole diameter significantly enhances the air flow velocity since more momentum transfers from the spray droplets to the ambient gas. It is assumed that there is a large amount of gas mass captured by the spray development at the tip periphery because the major spray momentum is still propagating downstream along the nozzle axis. The mean gas velocity at the tip periphery is 1-2m/s higher compared with that along the side periphery, which is still much lower than the spray penetrating velocity.

5.2 Method of Calculating Ambient Gas Mass Flow Rate

As shown in Fig.6, the spatial distribution of spray-ambient gas interaction is categorized as three regions. Section 1: Entrainment section at the side periphery. Here the ambient gas is entrained into the spray periphery because of the pressure gradient from the surrounding air to the spray induced by fuel injection. The side periphery in this section is quite smooth and almost stable. Section 2: Capturing section at the spray tip periphery. The spray droplets at the spray tip region move downstream with sufficient momentum. A large amount of ambient gas is captured into the spray plume due to the substantial relative velocity of spray penetration and ambient gas motion. Section 3: Recirculation section downstream of the side periphery. In this section, the droplets at the spray tip gradually lose momentum and are pushed aside by the upstream droplets and ambient gas flow, resulting in the simultaneous movement of droplets and ambient gas in the radial direction. At the same time, due to the influence of the relatively lower pressure of the ambient gas outside of the entrainment section, the droplets and ambient gas flow towards the upstream direction. Ikegami et al.[19] thought that in the Diesel combustion condition the pushed-out air movement would convey the soot formed in the spray tip region to the outside which improved soot oxidation. In the non-reacting spray case, the large vortex motion in this section implies strong turbulent mixing of fuel and air inside spray plume. In contrast, Cho et al. [13] divided the spray periphery into two parts with smooth line: the penetration part and stagnation part. The current model refines the classification of the spray-ambient gas interaction according to the measurements of the instantaneous ambient gas flow field.

In order to analyze the gas mass flow rate at the side periphery and the tip periphery, a calculating model is introduced as shown in Fig.7. The mass flow rate calculation is based on the following assumption: (1) The spray plume is axisymmetric; (2) In the recirculation section downstream of the entrainment section, the ambient gas moves with the spray droplets under the same velocity. Koo et al. [20] also found that the velocity of spray droplets decreases significantly along the radial coordinate. According to the laser illuminated spray image, the radial expanding speed of the width of the recirculation section (defined by the position where the upward gas velocity vector parallel to the nozzle axis locates) ranges from 2 to 4m/s since the main momentum propagation is in the axial direction, which also supports the assumption that the velocity of ambient gas and spray droplets are in the same order of magnitude. (3) The spray tip periphery is a half sphere whose radius is determined by the position where the spray periphery crosses the gas flow velocity vector perpendicular to the nozzle axis. The coordinate of spray periphery was obtained by averaging the images of several spray shots. The difference in distance between the ideal half sphere and the real irregular tip periphery is not greater than 3mm and the difference of the measured ambient gas velocity in this region is less than 0.4m/s. Therefore the assumption of the ideal half sphere is acceptable. The velocity vector along the side periphery is subdivided into two components, the normal velocity which is perpendicular to the control surface and the tangential velocity which is parallel to the control surface. The normal velocity $V_n(x,\alpha)$ is used to quantify the gas mass flow rate through the side periphery. The control surface was drawn according to the spray spreading angle at different timings.

Equations (1)-(3) show the process of calculating the gas mass flow rate entrained through the side periphery and captured by the spray tip periphery. The total gas mass flow rate was obtained by integrating the local gas mass flow rate with a series of discrete segments. The interval of each segment is 1mm at the side periphery and 1 degree at the spray tip. The calculation of gas mass flow rate at the entrainment section ($\dot{M}a_{ent}$) focused only on the entrainment section while the recirculation section was in balance state—no gain or loss of ambient gas flow. The captured gas mass flow rate at the tip periphery ($\dot{M}a_{cap}$) was obtained by subtracting ambient gas mass flow rate pushed out of the tip periphery ($\dot{M}at_{pushing}$) from the ideal gas mass flow rate due to the spray penetration ($\dot{M}at_{penetrating}$). Within a short time interval Δt_{shot}, the spray tip periphery was assumed to penetrate downwards without radial expansion, thus the spray volume change equals $V_{sp} \cdot \pi \cdot R^2$. The spray penetrating velocity V_{sp} was obtained by calculating the temporal change of the power fitting curve of the spray tip penetration length according to the spray images. The position along the nozzle axis in which the light intensity is less than 10% of the maximum value was defined as the spray tip. Correspondingly, $\dot{M}at_{pushing}$ refers to the integral of the local gas mass flow rate at different θ position based on the axial component of gas velocity $V_y(\theta)$ and axial projected area of tip periphery $A(\theta)$.The sum of the mass flow rate at the side periphery ($\dot{M}a_{ent}$) and mass flow rate at the tip periphery ($\dot{M}a_{cap}$) was considered to be the total gas mass flow rate through the whole periphery of the spray ($\dot{M}a$). The detailed definition of the variables is listed in the nomenclatures section.

Entrainment Section at Spray Side Periphery:

$$\dot{M}a_{ent} = \int_0^{X_1} \rho_a \; V_n(x,\alpha)\; dA(x,\alpha)\; dx \qquad (1)$$

Capturing Section at Spray Tip Periphery:

$$\dot{M}a_{cap} = \dot{M}at_{penetrating} - \dot{M}at_{pushing} = \rho_a \; V_{sp}\,\pi\, R^2 - \int_0^{\theta_1} \rho_a \; V_y(\theta)\; dA(\theta)\; d\theta \qquad (2)$$

At the Whole Spray Periphery:

$$\dot{M}a = \dot{M}a_{ent} + \dot{M}a_{cap} \qquad (3)$$

5.3 Effect of Injection Pressure on Ambient Gas Flow

5.3.1 Velocity Distribution and Mass Flow Rate at Spray Peripheries

The effect of injection pressure on the normal component of the ambient gas flow velocity along the control surface is shown in Fig.8. The control surface is drawn along the spray side periphery from the nozzle tip, and extended to the far field, as shown in Fig.7. The position with negative velocity value refers to the entrainment section, while the positive value refers to the pushed out flow at the recirculation section and a part of capturing section. The gas velocity shows very low value near the nozzle tip region and gradually increases downstream in the quasi-steady state. This is because the entrainment motion is not fully developed near the nozzle due to the gas inertia. The smooth side periphery near the nozzle tip demonstrates the weak interaction in the interface of the spray and ambient gas. When go downstream, the large vortex motion at the recirculation section acts as the source to improve the gas entrainment upstream. It is evident that the higher injection pressure increases the gas flow velocity and the length of gas entrainment section along the side periphery. The increased gas flow velocity under the high injection pressures is attributed to the increased aerodynamic force along the side periphery. Figure 9 shows the results of ambient gas mass flow rate into the spray entrainment section $\dot{M}a_{ent}$ and the capturing section $\dot{M}a_{cap}$ under the condition with the injection pressure from 100MPa to 300MPa and the ambient gas density of 15 kg/m^3. The triangle dots show the ratio of instantaneous ambient gas and fuel mass flow rate ($\dot{M}a$ / $\dot{M}f$). The instantaneous fuel mass flow rate $\dot{M}f$ was obtained from the results of injection rate measurement. Due to the injection ends at 2.2ms, there is no $\dot{M}a$ / $\dot{M}f$ value at the timing of 2.5ms ASOI.

A high portion of the gas mass flow rate into the spray is from the spray tip periphery. In the early stage of injection, for example at the timing of 0.5ms ASOI, the proportion of gas mass flow rate at the side periphery account for 8%, 7.5% and 7% of the total value $\dot{Ma}$ under the injection pressure of 100MPa, 200MPa and 300MPa, respectively, because the entrainment motion at the side surface is not yet developed and the entrainment section length is short. With the gradual development of the spray-ambient gas momentum interaction, this ratio rises to 16%, 18% and 24% at the EOI timing under the injection pressure of 100MPa, 200MPa and 300MPa, respectively. In contrast, Cho et al. [13] measured the ambient gas flow rate of the Diesel spray with much lower injection pressure (20~40MPa). Their conclusion was that almost 50% of the ambient gas mass flow was taken in from the side periphery. The spray with the increasing fuel injection pressure tends to capture more ambient gas through the tip periphery due to the larger velocity difference of the spray droplets and the ambient gas. As reported by Han et al. [21], under the combustion condition, the proportion of gas entrainment at side periphery tends to be larger owing to the heat release at the tip periphery restricts the gas entrainment. The gas mass flow rate continues increasing even after the end of injection, this can be attributed to the spray plume with sufficient momentum developing downwards along the nozzle axis.

The total ambient gas mass flow rate $\dot{Ma}$ increases with the injection pressure significantly since the improved momentum interaction between the fuel spray and the ambient gas at the side periphery and the larger velocity difference at the tip periphery. However, the increase in instantaneous ratio of gas and fuel mass flow rate ($\dot{Ma} / \dot{Mf}$) becomes moderate when the injection pressure shifts from 200MPa to 300MPa. According to the Bernoulli equation, the fuel jet velocity from the nozzle orifice follows a square root function of the difference between the fuel pressure inside the nozzle sac and the pressure of ambient gas. Moreover, higher injection pressure results in lower area contraction coefficient of nozzle and higher possibility of the cavitation flow inside the nozzle hole [22]. As a result, the increase in spray tip penetration becomes moderate in higher injection pressure, which suppresses the increase in gas mass flow rate captured by the spray penetration. Several studies [16, 23] reported the similar phenomena that the positive effect of ultra-high injection pressure on promoting fuel and air mixing process is gradually weaken after the pressure reaches a certain point.

5.3.2 Comparison of Measurement and Predictive Model

In order to verify the accuracy of LIF-PIV measurement and calculation method, the comparison of the instantaneous equivalence ratio ϕ_{inst} and the mean equivalence ratio ϕ_{mean} of the whole spray are taken into account in Figs.10 and 11 respectively. The instantaneous equivalence ratio was calculated from the instantaneous ambient gas and fuel mass flow rate by LIF-PIV results in Fig.10 (a) and from the predictive model suggested by Wakuri et al. [24] in Fig.10 (b). The main equation is given as follows:

$$\emptyset_{inst} = \frac{L_{th}\sqrt{C_a}}{2\tan(\alpha)}\sqrt{\frac{\rho_f}{\rho_a}}\frac{D}{X} \quad (4)$$

The L_{th} stands for the stoichiometric air/fuel ratio, X is the spray tip penetration. ρ_f and ρ_a is the fuel density and ambient gas density, respectively. D is the nozzle hole diameter. C_a is the area contraction coefficient which is calculated based on the discharge coefficient of C_d and cavitation number [22]. C_d is defined as the ratio of the fuel mass flow rate to the theoretical one computed from the Bernoulli equation. In this study, the value of C_a 0.85 was adopted in all the conditions, since C_a can be regarded as almost constant with the increase in the injection pressure [25]. The principle of this model is based on the ratio of gas and fuel mass flow rate across the cross section downstream from the nozzle tip. Fuel injection mass M_f was the integrated result of injection rate profile. The mean equivalence ratio ϕ_{mean} is calculated according to the temporal integration of instantaneous ambient gas flow rate by LIF-PIV result in Fig.11 (a) and the spray volume in Fig.11 (b).

For all the cases, the equivalence ratio decreases with the spray development, the high injection pressure has the potential to achieve the lean mixture more rapidly. The instantaneous equivalence ratio result from Wakuri's model shows the similar trend however a bit higher value compared with that from the PIV result. Two of the main assumptions regarding the deduction of Wakuri's model are: (1) In any cross section downstream of nozzle tip, the ambient gas and spray droplets flow together without relative velocity; (2) the initial momentum of the spray droplets transfers to the fuel and gas mixture downstream. The first assumption implies that all the gas inside the spray results from the entrainment at the side periphery, which differs from the actual spray behavior according to the PIV measurement in this study. Some other study [26] also argued that this assumption may be doubtful but stands for approximate prediction for the steady spray. Regarding the second assumption, some part of momentum of the droplets is also transferred to the ambient air outside the spray which causes the entrainment and pushed-out motion. As a conclusion, the predictive model tends to underestimate the ambient gas flow rate through the whole spray periphery. The mean equivalence ratio results in Figs.11 (a) and (b) are in good agreement in both the results from the PIV measurement and the spray volume. The difference at the early stage of injection duration is supposed to result from the underestimation of the entrained gas mass in the entrainment section due to the inertia of the tracer particles. The accuracy can be improved by using smaller tracer particles with higher tracking ability and increasing the number of measurements. Generally, the calculation method proposed in this study was proved to have the ability to characterize the ambient gas flow into the spray accurately.

According to the spray tip penetration equation proposed by Hiroyasu and Arai [27] shown as follows:

$$X = 2.95 \left(\frac{P_{inj} - P_a}{\rho_a}\right)^{0.25} (D\ t)^{0.5} \qquad t > t_{break} \tag{5}$$

P_{inj} is fuel injection pressure and P_a is the ambient gas pressure, t_{break} indicates the liquid break-up time defined in the reference [27]. The relationship between the penetration length and the injection pressure in the far field is $X \propto (\Delta p)^{0.25}$, thus if assuming that the value of C_a and θ

do not change too much with injection pressure, the asymptotic relationship of $\phi_{inst} \propto (\Delta p)^{-0.25}$ can be deduced. However, the effect of ultra-high injection pressure is still apparent in the reacting spray condition, because the air entrainment at the side periphery becomes more prominent under the evaporating condition, the higher injection pressure enhances the momentum transfer at the side periphery. The small scale turbulent mixing particularly in the head vortex region is improved with the rise of injection pressure, which leads to a lower soot level [28].

The gas mass flow rate at series of cross sections with different distances from the nozzle tip can be obtained according to the above mentioned Wakuri theory. Because it is assumed that all the entrained gas comes from the side periphery, the local normal velocity along the side periphery can be calculated by dividing the difference of gas mass flow rate of the adjacent cross sections by the surface area of side periphery and the gas density. Equations (6) and (7) show the process:

$$\dot{M}a = \rho_a \, \pi \, d^2 \left[\frac{C_a \, V_0}{8} \left(\sqrt{\left(\frac{\rho_f}{\rho_a} - 1\right)^2 + \frac{16 \, \rho_f}{C_a \, \rho_a} \left(\frac{x}{d}\right)^2 \tan^2\alpha} - \frac{\rho_f}{\rho_a} + 1 \right) - \frac{C_a}{4} V_0 \right] \quad (6)$$

$$V_n(x, \alpha) = \frac{\dot{M}a(x + dx) - \dot{M}a(x)}{\rho_a \, dA(x, \alpha)} \quad (7)$$

V_0 is the fuel injection velocity at the nozzle hole exit, which can be calculated according to the Bernoulli equation. The spray half angle α was obtained from the spray images. The interval of the adjacent sections was set to 1mm. Figure 12 shows the comparison of the normal velocity distributions from the Wakuri theory and the PIV measurement, respectively. In accordance with the definition in Fig.8, the entrained ambient gas velocity is marked positive, while the pushed-out gas velocity in the recirculation section is not shown here. The predictive result shows that the normal velocity starts with a much larger value from the nozzle tip due to the very small side area value, which conflicts with the break-up process of the actual spray. When go downstream to the quasi-steady state, both of the results are gradually consistent with each other. Because the Wakuri theory assumes a Diesel spray with extremely long injection duration, the pushed-out gas velocity downstream is not able to be predicted. Wakuri theory can properly reflect the gas entrainment feature if the spray reaches the quasi-steady state.

5.4 Effect of Ambient Gas Density on Ambient Gas Mass Flow Rate

The effect of ambient gas density on the velocity of the ambient gas is shown in Fig.13 under the fuel injection pressure of 300MPa. The maximum value of the normal velocity (absolute value) at the entrainment section shows the similar value in all of the three ambient gas density conditions. The similar result can also be found in the study by Sepret et al. [13]. That is because the ambient gas drag force is linearly proportional to the gas density which tends to accelerate the gas flow, on the other hand, the increased gas density restrains the gas phase acceleration. Moreover, the entrainment section is considerably restrained in the high ambient gas density condition. Figure 14 shows the ambient gas mass flow rate into the entrainment section $\dot{M}a_{ent}$ and the gas mass flow rate through the whole spray periphery $\dot{M}a$ as a function of injection pressure and gas density. It is clear that the higher ambient gas density leads to the larger ambient gas mass flow rate, which demonstrate that the gas density value is the key factor to determine the ambient mass flow rate even though the similar gas flow velocity and decreased spray periphery length in the high gas density condition. If considering the same fuel injection rate, the ratio of instantaneous gas and fuel mass flow rate increases apparently in the higher ambient gas density condition. The increasing drag force on the spray periphery enhances the ambient gas flow, which promotes the liquid phase disintegration into small droplets. It is also supposed that the fuel and air mixing process inside the spray plume is improved in the high ambient gas density condition. However the increase in $\dot{M}a_{ent}$ becomes moderate in the high ambient gas density condition, which results from the influence of the decreased entrainment section length. According to the above mentioned equations by Wakuri and Hiroyasu, an approximate relation $\phi_{inst} \propto (\rho_a)^{-0.25}$ can be assumed.

5.5 Effect of Nozzle Hole Diameter on Ambient Gas Mass Flow Rate

Figure 15 shows the spatial distribution of normal velocity with the nozzle hole diameter of 0.12mm under 100MPa injection pressure condition. In contrast to the result of 0.08mm nozzle hole shown in Fig. 8, there is a considerable increase in air flow velocity and the length of entrainment section with the increase in nozzle hole diameter. Larger nozzle hole diameter results in the increase in the initial spray droplet size, the air drag force is proportional to the droplet size, which accounts for the increase in air flow velocity. The ambient gas mass flow rate and the ratio of instantaneous ambient gas and fuel mass flow rate ($\dot{M}a / \dot{M}f$) of the 0.12mm nozzle hole under 100MPa injection pressure is shown in Fig.16. Comparing with the result of 0.08mm nozzle shown in Fig. 9, the gas mass flow rates in both the entrainment section $\dot{M}a_{ent}$ and the capturing section $\dot{M}a_{cap}$ significantly increase. On the other hand, the ratio of instantaneous ambient gas and fuel mass flow rate decreases inversely due to the increase in fuel injection rate. This implies that the spray atomization deteriorates in the larger nozzle hole diameter case. What's more, the dependency of the ratio of the ambient gas and fuel mass flow rate on the ambient gas density becomes insensitive under the larger nozzle hole diameter condition. According to Eq.5, the instantaneous equivalence ratio increases linearly with the nozzle hole diameter.

5.6 Investigation of Spray Droplets and Ambient Gas Velocity by CFD Simulation

Some studies [29, 30] successfully measured the spray internal velocity distribution by capturing the image of the diluted spray droplets or the fluorescent tracers dissolved in the fuel. However, most of the results are still ambiguous in the qualitative analysis as it is difficult to measure the droplets velocity and the ambient gas flow velocity simultaneously inside the dense non-evaporating diesel spray. Alternatively, in this study, the CFD simulation was carried out under the condition of the injection pressure of 100MPa, ambient gas density of 15 kg/m^3 and nozzle hole diameter of 0.08mm to investigate the droplets velocity and gas velocity inside the spray.

Figure 17 shows the comparison of spray images from the experiment and the simulation by using the RANS model, in which the color level denotes the velocity value and the size scale denotes the droplets size level. The spray shape shows good agreement with the experimental results. It is evident that spray droplet velocity dramatically decreases when reaching the far field due to the break-up process. However, the droplet velocity is still nearly 15-20m/s at the tip periphery at 2.0ms ASOI. Figure 18 shows the comparison of the spray tip penetration and the spray angle. The results from the measurement and the CFD simulation coincide with each other. Figure 19 shows the gas flow field inside and outside the spray and the absolute pressure distribution which is not possible to be measured using the LIF-PIV method. The result suggests that the velocity of entrained gas is accelerated to a high value after entering the spray periphery due to the existence of pressure gradient from the spray periphery to the center line. On the other hand, a comparatively higher pressure region can be observed at the tip periphery, which leads to the pushed-out gas motion. Tanabe et al. [31] measured the total pressure and static pressure distribution inside a gas jet and predicted the gas flow feature. That study shows the similar trend as the simulation result. Figure 20 shows the ambient gas velocity distribution along the nozzle hole axis at several timings ASOI. When the spray reaches the quasi-steady state within the injection duration (1.0ms and 2.0ms ASOI), the ambient gas accelerates from 0 at the nozzle hole exit to the peak value of 90m/s at 10mm from the nozzle tip, the gas flow field keeps approximately constant during the fuel injection period. On the other hand, the ambient gas velocity decreases dramatically after the end of injection (2.5ms ASOI). The ambient gas velocity in the downstream gradually decreases due to the momentum dissipation induced by the break-up of the spray droplets. The spray tip penetrating velocity is apparently larger than the velocity of ambient gas at the same position, the spray tip penetrating velocity is shown by the plots in Fig.20. It is proved that a large amount of ambient gas is captured through the tip periphery in the non-evaporating spray case. Figure 21 and 22 shows the axial component of the ambient gas velocity along the nozzle axis downstream from the spray tip and the radial component of the ambient gas velocity along the cross sections with different distances from the nozzle tip, respectively. Due to the influence of the dense fuel droplets, the ambient gas flow filed inside the spray plume can hardly be observed in PIV measurement, the ambient gas flow outside the spray periphery obtained by LIF-PIV and CFD is in good agreement. The radial velocity of the ambient gas is able to represent the tendency of the normal velocity along the spray side periphery due to the small spray spreading angle. The distances from the nozzle tip 20mm, 40mm and 60mm correspond to the typical positions in the entrainment section, recirculation section and capturing section, respectively. At 20mm axial distance, the ambient gas outside the spray periphery flows towards the nozzle hole axis due to the entrainment flow (Negative radial velocity for radial distance >2mm or <-2mm). Then the gas flow shifts to the reverse direction when entering the spray because the gas flows together with the spray droplets whose velocity vectors are in the outward and downward directions (Positive radial velocity for radial distance <2mm and >-2mm).

While at 40mm and 60mm axial distance, the ambient gas tends to be merely pushed out of the spray periphery, which shows different radial velocity distribution.

6. CONCLUSIONS

The characteristics of ambient gas motion induced by a single Diesel spray were measured quantitatively by using the Laser Induced Fluorescence-Particle Image Velocimetry (LIF-PIV) technique under the non-evaporating quiescent condition. The ambient gas velocity and mass flow rate into the spray through the whole spray periphery (spray side periphery and tip periphery) were investigated quantitatively. The CFD simulation validated the measurement by using LIF-PIV technique and provided insightful understanding of the ambient gas flow process.

Based on this study, the following can be concluded:

1. The LIF-PIV technique is capable of measuring the 2-dimensional gas flow field around the spray periphery with high resolution and eliminating the circumstance noise. The spray model proposed in the study divides the spray body into entrainment section, recirculation section and capturing section, which quantitatively describes the ambient gas flow field of a non-evaporating transient Diesel spray.
2. The entrained gas mass flow rate at the spray tip periphery is prominent in the whole periphery in non-evaporating spray condition and the proportion of the ambient gas entrainment along the side periphery increases as the spray developing.
3. Higher injection pressure improves the ambient gas mass flow rate into the spray, both at the spray side periphery due to the increased drag force and at the tip periphery due to the larger droplets/gas velocity difference. However the increase in total entrained gas-fuel ratio tends to be moderate when the injection pressure rises to the ultra-high level.
4. Higher ambient gas density suppresses the entrainment section length and the spray penetrating velocity, on the other hand, the ambient gas mass flow rate into the spray increases apparently. As a result, the equivalence ratio of the spray decreases under the higher ambient gas density condition.
5. The smaller nozzle hole diameter results in the significant decrease in the ambient gas mass flow rate into the spray, on the other hand, the ratio of the ambient gas and fuel mass flow rate increases.
6. The CFD calculation validates the characterization of the ambient gas flow field of the Diesel spray measured by LIF-PIV technique.

7. NOMENCLATURES

AMD	Arithmetic mean diameter (um)
$A(x,\alpha)$	Surface area of spray side from x to x+dx
$A(\theta)$	Axial projected surface area of spray tip from θ to θ +d θ
C_a	Area contraction coefficient

C_d	Discharge coefficient
CCD	Charge-coupled device
D	Nozzle hole diameter(mm)
EOI	End of injection
L_{th}	stoichiometric air/fuel ratio
M_a	Ambient gas mass in spray(mg)
$\dot{M}_a$	Total ambient gas mass flow rate (mg/s)
$\dot{M}a_{ent}$	Ambient gas mass flow rate at entrainment section (mg/s)
$\dot{M}a_{cap}$	Ambient gas mass flow rate at capturing section (mg/s)
$\dot{M}at_{penetrating}$	Ideal ambient gas mass flow rate at tip periphery due to spray penetration (mg/s)
$\dot{M}at_{pushing}$	Ambient gas mass flow rate pushed out of the tip periphery (mg/s)
M_f	Injected fuel mass (mg)
$\dot{M}_f$	Fuel injection mass flow rate (mg/s)
P_a	Ambient gas pressure (MPa)
P_{inj}	Fuel injection pressure measured in common rail (MPa)
R	Radius of ideal spray tip half sphere (mm)
SOI	Start of injection
t	Time after start of injection (ms)
t_{break}	Break-up time defined by Hiroyasu and Arai [26]
$V(x,\alpha)$	Gas velocity at side periphery (m/s)
$V_n(x,\alpha)$	Normal velocity at side periphery (m/s)
$V(\theta)$	Gas flow velocity at tip periphery (m/s)
$V_y(\theta)$	Axial component of gas velocity at tip periphery (m/s)
V_{sp}	Spray tip penetrating velocity (m/s)
V_0	Fuel injection velocity (m/s)
x	Distance from nozzle tip (mm)
X	Spray tip penetration (mm)
X_1	Coordinate of downstream of entrainment section (mm)
α	Half of spray angle
θ	Arc angle relative to the horizontal direction in spray tip region
θ_1	Arc angle between horizontal direction and spray axis (90 deg.)
ρ_a	Ambient gas density (kg/m^3)
ρ_f	Fuel density (kg/m^3)
ϕ_{mean}	Mean equivalence ratio
ϕ_{inst}	Instantaneous equivalence ratio
Δt_{shot}	Time interval of two frames

Δt_{inj} Time interval from tracer injection to fuel injection

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