Effect of flow in the circular 90-degree curved nozzles on ejecting oil jet behavior

Mikimasa KAWAGUCHI*, Goichi NITTA*, Keita MIMURA*, Keiya NISHIDA**, Masanobu KOUTOKU***, Ryo YAMAMOTO***, Akira NAKASHIMA*** and Yoichi OGATA**

*Department of Mechanical Systems Engineering, Hiroshima University  
**Graduate School of Advanced Science and Engineering, Hiroshima University  
1-4-1 Kagamiyama, Higashi-Hiroshima, Hiroshima 739-8527, Japan  
E-mail: kawaguchi-mik@hiroshima-u.ac.jp  
***Mazda Motor Corporation  
3-1 Shinchi, Fuchu-cho, Aki-gun, Hiroshima 730-8670, Japan

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Abstract

In general, technical methods for improving the thermal efficiency of an engine increase the heat load on peripheral components. Recently, a piston cooling gallery equipped with a flow path has been developed. The engine oil is supplied an oil jet from the nozzle, which is placed under the piston to the piston gallery entrance hall. The nozzle of the oil jet is curved to minimize its size, and the jet interface between ambient air and oil fluctuates near the nozzle exit owing to the shape. Few studies have investigated the behavior of oil jets ejecting from curved pipes. We therefore investigated the flow in two nozzles having a basic bend of 90° with radii of curvature of 15 and 60 mm. Our results clarify the effect of internal flow on the ejecting oil jet behavior. Silicone oil was used as the working fluid. The kinematic viscosity of the silicone oil at 298 K was similar to that of engine oil at 353 K. The behavior of the oil jet was investigated by visualization using background light. A light-emitting-diode displacement meter was installed to measure the jet width. We found that the width of the oil jet increased on the downstream side with large fluctuation of the interface under the condition of a small radius of curvature and large Reynolds number. Furthermore, we time-synchronously measured flow in the nozzle, two-dimensional two-component time-resolved particle image velocimetry, and visualization of the jet. The Reynolds number was set from 1000 to 3000, which is close to that of the engine oil jet. The oil flow velocity in the nozzle fluctuated in the radial direction. The fluctuation became strong under the condition of a small radius and large Reynolds number. The fluctuation propagation speed calculated from the correlation coefficient was as high as the flow speed itself. Furthermore, the jet interface fluctuation speed in the flow direction was as high as the fluctuation propagation speed in the nozzle. Our results demonstrate that the cause of the interface fluctuation is the fluctuation propagation of flow in the nozzle.

Keywords: Internal combustion engine, Piston cooling, Oil jet, Multi-phase flow, Curved circular nozzle, Interfacial fluctuation, Particle image velocimetry

1. Introduction

National governments are making efforts to reduce CO₂ emissions and thus global warming. In particular, governments in the European Union have established regulations for the emission of CO₂ from vehicles having internal combustion engines. These regulations require any new vehicle to have an average CO₂ emission below 95 g/km from 2021 (Schulz M. and Kourkoulas D., 2014). The thermal efficiency of the internal combustion engine must be increased to meet these regulations. As an example, an engine is designed with a high compression ratio (Yamakawa, M. et al. 2012) and/or using a downsizing turbocharger (Petitjean, D. et al. 2004) to increase thermal efficiency. Both methods increase the heat load
through a high combustion temperature. This is a serious problem for piston cooling because past pistons had no cooling device, such as a water jacket. A piston cooling gallery was recently proposed as a piston cooling system. Kajiwara et al. (2003) suggested a temperature prediction technique for a piston with cooling gallery. Oil is injected from a nozzle placed under the piston, and the oil jet pours into the piston gallery through an entrance hall (Deng, X., et al. 2018). The piston heated by high-temperature combustion gas can thus be cooled by low-temperature oil that flows in the gallery. It may therefore be desirable to control the shape of the oil jet such that it is stable and straight. However, there will be a Kelvin–Helmholtz instability and Rayleigh–Taylor instability at the gas (ambient air)–liquid (oil jet) interface of the jet injected from the nozzle because of the speed difference between the air and jet, and the acceleration of the jet is much higher than that of ambient gas (Rayleigh, L., 1878). Additionally, the nozzle must be curved for it to be as small as possible in the design of the engine layout. The behavior of the oil jet will therefore be wavy and unstable.

Many studies have investigated the instability of a liquid jet ejected from a straight circular nozzle. Arai et al. (1999), for example, used two laser systems and two photodetectors to measure the wave velocity and frequency and found the deformation behavior of wave break up. Tomotika (1935) calculated the instability of a cylindrical thread of viscous fluid surrounded by another viscous fluid. In that work, he analyzed the effect of each fluid viscosity for the wavelength having the maximum growth rate.

It is well known that flow in a curved pipe generates twin vortices because of secondary flow. Eustice (1910) investigated the stream pattern in a circular curved bend and found twin vortexes generated by secondary flow. Dean (1927, 1928) found a twin vortex structure in secondary flow theoretically using incompressible Navier–Stokes equations, with the analysis agreeing with the experiments conducted by Eustice.

However, few studies have investigated the effect of the instability of a liquid jet ejected from a curved circular nozzle to an atmosphere at ambient pressure. Nakashima et al. (2018) investigated the behavior of an oil jet ejected from a nozzle with a diameter of 6 mm and a 90° bend through a 30-mm radius. They found that the oil jet ejected from the bent pipe nozzle became unstable with an increasing Reynolds number $Re$, while the jet from a straight pipe nozzle was stable even at a large Reynolds number. The rotation direction of twin vortexes generated by secondary flow in the bend pipe changed as shown in Fig.1. In their work, when the Reynolds number was less than 2000, a Dean vortex formed and flowed from the inner side to the outer side (Dean, 1928). Meanwhile, twin vortices formed and flowed from the outer side to the inner side when the Reynolds number exceeded 2500. These twin vortexes had a rotation opposite to that of a Dean vortex. However, only one curved nozzle shape was considered in the study.

Various studies have investigated the behavior of a jet ejected from a straight pipe and the secondary flow pattern for a bent pipe, but there have been few studies on the effect on oil jet behavior of the fluctuation state of the nozzle internal flow due to secondary flow. The present study thus examines how the oil jet behavior is affected by the nozzle shape. Two nozzles having different curvatures of radius were prepared for investigation. We selected these two nozzles with half the radius of curvature (15-mm radius) and twice the radius of curvature (60-mm radius) the respectively nozzles that was used in the study by Nakashima et al. (2018). Three series of experiments were conducted: visualization of the oil jet behavior adopting background illumination, measurement of the width of the oil jet adopting a light-emitting-diode (LED) displacement gauge, and velocimetry of the nozzle internal flow adopting time-resolved particle image velocimetry.

![Fig. 1 Vortices components that were made from secondly flow in curved bend pipe with a diameter of 6 mm and a 90° bend through a 30-mm radius were taken at the 6mm after bend by Nakashima et al (2018).](image-url)
2. Experimental setup

All experiments reported in this paper were conducted using two nozzles made of acryl, silicone oil as the working fluid, and an oil flow system. The experimental methods refer to Nakashima (2018).

Figure 2 (a) and (b) shows the shapes of the two nozzles having different radii of curvature. Both nozzles have a 90° bend and inner diameter \( d = 6 \) mm. The bend portions of curved nozzles (a) and (b) have a radius of curvature \( R \) of 15 and 60 mm, respectively, a length after the bend exit of 30 mm, and a length before the bend of 300 mm. These nozzles are denoted \( R_{15} \) and \( R_{60} \) in the results of this paper.

Table 1 compares physical properties of the silicone oil (KF-56A, Shin-Etsu Silicones, Tokyo, Japan) used in the present experiments with those of engine oil (SAE 0W-30). Kinematic viscosity and surface tension of KF-56A at 298 K are similar to those of SAE 0W-30 at 353 K. In addition, because the refractive index of KF-56A is similar to that of acrylic compounds as shown in Fig. 3, velocimetry of the internal nozzle is possible through index matching.

The oil flow system used in all experiments is shown in Fig. 4. The flow rate of oil supplied by the vortex pump (20KHD07Z, Nikuni, Kanagawa, Japan) is controlled by changing the pump frequency set by the inverter (FR-D720, Mitsubishi Electric, Tokyo, Japan). A pump sucks oil from a tank and discharges the oil to the nozzle through a Coriolis flowmeter (FD-SS20A, Keyence, Osaka, Japan) and a flow straightener (MODEL FS, OVAL, Tokyo, Japan). The flowmeter was used to measure the oil flow rate and oil temperature. The flow straightener having an inner diameter of 35.5 mm and a pipe length of 400 mm was used to rectify the flow. In addition, a pipe with an inner diameter of 14.9 mm and a length of 1200 mm was installed in front of both nozzle inlets such that flow was fully developed.

In this paper, the coordinate axes are defined as shown in Fig. 5. \( X, Y, \) and \( Z \) axes move to end of the sentence refer to the side direction, in–out direction of the curve, and injection direction. The positive \( X \) direction is from the center to the inner side. The coordinate \( Z \) is positive away from the nozzle exit from \( Z = 0 \) to 148.2 mm and is negative inside the nozzle from \( Z = 0 \) to −19.6 mm.

![Fig. 2 Illustrations and photographs of nozzles used in visualization: (a) smaller radius of curvature \((R = 15 \text{ mm})\) and (b) larger radius of curvature \((R = 60 \text{ mm})\)]
2.1 Visualization and width measurement of the oil jet

Figure 6 is a schematic of the experimental apparatus for the visualization of oil jet behavior. Background illumination was adopted. An LED strobe light (LLBK1–LA-W-0001, AITEC SYSTEM, Kanagawa, Japan) with a diffuser (#47-952, Edmond Optics, New Jersey, USA) was used as a light source. A high-speed camera (SA-Z, Photron, Tokyo, Japan) was controlled by a computer to record video. The high-speed camera had a maximum image size of 1024 × 1024 pixels and a maximum frame rate of 20,000 fps at the maximum image size.

In the quantitative assessment of the width of the oil jet, an LED displacement gauge (LS-9030MR, Keyence, Osaka, Japan) was installed as shown in Fig. 7. The gauge had a resolution of ±2.0 µm and sampling frequency of 16,000 Hz. The jet width was measured at Z = 1.0 mm and at intervals of 5.0 mm in the Z direction from Z = 5.0 mm at the nozzle exit. Figure 8 shows the correlation of the jet visualization and width measurement. The red points in the figure show measurements made by the LED displacement gauge, and the points are 3.0 s time averaged interface point between oil and air at each distance from nozzle exit. The photograph shows the jet behavior averaged over a period of 1.0 s. All red points lie on the air–oil interface, confirming the accuracy of the measurement of the LED displacement gauge.

Table 1 Physical properties of silicone oil

|                      | KF-56A  | SAE 0W-30 |
|----------------------|---------|-----------|
| Density [kg/m³]      | 995     | 846       |
| Kinematic viscosity [m²/s] | 1.5×10⁻⁵ | 1.44×10⁻⁵ |
| Surface tension [N/m] | 28.1×10⁻³ | 26.0×10⁻³ |
|                      | KF-56A  | Acryl     |
| Refractive index [-] | 1.498   | 1.49 - 1.53 |

Fig. 3 Test for index matching:
(left) acryl nozzle without silicone oil,
(right) acryl nozzle with silicone oil

Fig. 4 Oil flow system used in the experiments

Fig. 5 Definitions of X, Y, and Z axes

Fig. 6 Experimental apparatus for visualization of the oil jet behavior

Fig. 7 Experimental apparatus for measurement of the oil jet width

Fig. 8 Correlation of the jet visualization and width measured using an LED displacement gauge (R = 15, Re = 1500)
2.2 Time-synchronous measurement of 2D2C time-resolved PIV and visualization of jet behavior

Two high-speed cameras were used to synchronize two-dimension, two-component (2D2C) time-resolved PIV; i.e., PIV for jet visualization in the XZ plane. This experiment investigated the effect of flow in the nozzle on the jet interface. Figure 9 shows the time-synchronous experimental setup. The PIV system was added to the oil flow system in Fig. 6. A digital delay generator (VSD2000, Flowtech Research, Kanagawa, Japan) was used for the time synchronization of the Q-switch of a Nd:YAG high-frequency double-pulsed laser (MESA-PIV, Amplitude Systemes, Bordeaux, France) and the camera exposure. Fluorescent tracer particles (Fluostar0459, average particle size: \( d_p = 15 \text{ um} \), density of the tracer particles: \( \rho_p = 1100 \text{ kg/m}^3 \), EBM Corporation, Tokyo, Japan) were excited at a wavelength of 532 nm to emit light at 580 nm and seeded. A long-pass filter (SCF-50S-560, Sigmakoki, Saitama, Japan) was attached to a camera lens (Micro-Nikkor 105-F2.8, Nikon, Tokyo, Japan) to photograph the fluorescent emission at 580 nm.

Figure 10 shows the Principle image of PIV using frame straddling method. The aperture of a camera is shut at regular intervals. In this experiment, the camera shutter speed was set to 20,000 fps and the double-pulsed laser illuminated particles for each half frequency of the camera shuttering (10,000 Hz). The particles illuminated by the two lasers were thus photographed at a sampling rate of 10,000 Hz. The light interval time (\( \Delta t \)) was the time for particle movement. Depending on the Reynolds number, the time interval between the two laser pulses was adjusted between 10 and 40 \( \mu \text{s} \) such that tracer particles moved two pixels or more with minimal movement and 15 pixels or less with maximum movement within the paired imaging shown in Table 2. PIV analysis software (FtrPIV, Flowtech Research, Kanagawa, Japan) that made use of direct cross-correlation was adopted. A window of \( 33 \times 33 \) pixels with a 50% overlap was set for interrogation and 10 pixels were added to the interrogation window for the search range.

The area averaged nozzle internal flow speed of Eq. (1) \( \bar{W}_{PIV} \) obtained by the PIV result at 6mm before the curved bend is compared to the one of Eq. (2). \( \bar{W} \) obtained by the flow meter. Here, \( u_x \) and \( \Delta r \) in Eq. (1) are flow velocity component in x-direction and calculation grid, and Q in Eq. (2) is flow rate, respectively. Figure 11 shows the comparison between \( \bar{W} \) and \( \bar{W}_{PIV} \). The accuracy of PIV measurement is validated because the difference between them was as small as 3.0% at the maximum in Fig. 11. \( \bar{W}_{PIV} \) was assumed that PIV result was an axial symmetry. Thus, the accuracy of PIV measurement is considered to be good. Moreover, we confirmed the Stokes number \( St \) that is defined by Eq. (3). Figure 12 shows that \( St \) in the present experiments is the order of \( 10^4 \) to \( 10^3 \), and it is much smaller than one. Therefore, it is confirmed that tracer particles can follow the flow in the present experiments.

Table 2 Adjustment of the light interval for each Reynolds number

| \( Re \) [-] | 1000 | 1500 | 2000 | 2500 | 3000 |
|-------------|------|------|------|------|------|
| R15         | 30.0 | 30.0 | 25.0 | 20.0 | 10.0 |
| R60         | 40.0 | 35.0 | 30.0 | 25.0 | 12.0 |

\[
\bar{W}_{PIV} = \frac{\sum u_x \Delta r}{\sum \Delta r}
\]  
\[
\bar{W} = \frac{4Q}{\pi d^2}
\]  
\[
St = \frac{\rho_p d_p^2 \bar{W}}{18 \rho v d}
\]
2.3 Method of tracking the velocity distribution in the nozzle using the cross-correlation of PIV images

The cross-correlation coefficient $S_z$ is defined by Eq. (4) for analysis of the flow speed fluctuation state of internal nozzle flow. This method of flow analysis has been used in several studies; e.g., by Paquier (2015). Here, $\bar{U}_{mag}$ is the area-averaged flow speed of the base window and the search window each is defined by Eq. (5). The location of the base window along the Z axis $z_i$ ($i = 0–15$) from $-3.27$ to $-19.6$ mm is given by Eq. (6). The offset length $\Delta z$ for calculating $S_z$ is displaced in the Z direction from 0 to 3.27 mm. Figure 13 shows the example of two PIV images taken at a certain time $t = t$ (base image) and after a certain time $t = t + t'$ (slave image). Table 3 shows calculation parameters used to obtain the cross-correlation coefficient. We investigated the dependence of $Re$ and $R$ on $S_z$ to determine characteristics of wave propagation inside the nozzle. The base window with dimensions of $1.8$ mm $\times$ $1.8$ mm is set at the outside edge of the base image. The PIV image comprises $16 \times 47$ square cells with a cell size of $0.36$ mm. A search is made for the offset length having the maximum correlation coefficient $S_z$. Moreover, to confirm the dependence of results on the position of the base window, the coefficient is calculated for 16 points at intervals of three cells in the Z direction. The time shift $t'$ is adjusted from $100$ $\mu$s for $Re = 2000$ and $Re = 3000$ to $300$ $\mu$s for $Re = 1000$. The calculation results are discussed for the time average of 500 PIV images.

$$S_z = \frac{\sum_{i=0}^{15} (U_{mag, t=t+\Delta t}(z_i) - \bar{U}_{mag, t=t+\Delta t, searched window}) (U_{mag, t=t}(z_i) - \bar{U}_{mag, t=t, base window})}{\sqrt{\sum_{i=0}^{15} (U_{mag, t=t+\Delta t}(z_i) - \bar{U}_{mag, t=t+\Delta t, searched window})^2 \sum_{i=0}^{15} (U_{mag, t=t}(z_i) - \bar{U}_{mag, t=t, base window})^2}}$$

(4)

$$\bar{U}_{mag} = \frac{\sum_{A} U_{mag} dA}{\sum_{A} dA}$$

(5)

$$z_i = -1.09 \times i - 3.27$$

(6)

Fig. 13. Example of obtaining the cross-correlation coefficient $S_z$ ($R = 15, Re = 3000$) for $t' = 100$ $\mu$s

[Fig. 11 Comparison of the area averaged nozzle internal flow velocity by flow meter and PIV result]

[Fig. 12 Stokes number that was calculated by tracer particles and working fluid for each Reynolds number]
Table 3 Calculation parameters used to obtain the cross-correlation coefficient

| Parameter                  | Value                     |
|----------------------------|---------------------------|
| Grid size                  | 0.36mm × 0.36mm           |
| PIV image size             | 16grids × 47grids        |
| Base area                  | 1.8mm × 1.8mm             |
| Search window size         | 1.8mm × 3.6mm             |
| PIV images                 | 500                       |

Time difference of comparison images $t'$ [μs]

| $Re$ | 1000 | 2000 | 3000 |
|------|------|------|------|
| $R15$ | 300  | 100  | 100  |
| $R60$ | 300  | 100  | 100  |

3. Experimental conditions

Table 4 summarizes the experimental conditions. All experiments were conducted at room temperature and atmospheric pressure, and the oil temperature was controlled at 298 ± 5.0 K. The ambient pressure and oil temperature were controlled, and the effects of the temperature on the viscosity of oil and surface tension were thus eliminated. In this paper, the Reynolds number $Re$ is defined by Eq. (7).

The area-averaged streamwise velocity is given by $\overline{W} = 4Q/\pi d^2$. The oil jet behavior was visualized for 500 intervals of the Reynolds number ranging from 1500 to 3000 and an observation range from $Z = 0$ mm at the nozzle exit to $Z = 148.2$ mm. The jet width was measured for 500 intervals of the Reynolds number ranging from 1000 to 3000 and an observation range from $Z = 1.0$ to 85.0 mm. PIV was performed for 500 intervals of the Reynolds number ranging from 1000 to 3000 and observation range from $Z = 0$ mm at the nozzle exit to $-25.0$ mm in the nozzle. As an example, a Reynolds number of 1000 is denoted $Re_{1000}$ in the results of this paper.

Table 4 Experimental conditions

| Parameter                  | Value                     |
|----------------------------|---------------------------|
| Ambient pressure           | [kPa] 99.3 - 100.3        |
| Ambient temperature        | [K] 296 ± 1.0             |
| Working fluid              | Silicone oil (KF56A, Shin-Etsu Chemical) |
| Fluid temperature          | [K] 298 ± 5.0             |

| | Visualization | Jet width measurement | PIV |
|-----------------|---------------------|---------------------|
| Reynolds number  | [-] 1500 - 3000     | 1000 - 3000         | 1000 - 3000 |
| Observation range | [mm] 0 - 148.2   | 1.0 - 85.0          | -25 - 0     |
| Sampling rate    | [Hz] 20000          | 16000               | 10000       |
| Light interval time | [μs] -             | -                   | 10 - 60     |

$$Re = \frac{\overline{W}d}{v}$$ (7)

4. Results and discussion

4.1 Visualization and width measurement of the oil jet

Figure 14 shows the visualization of the oil jet behavior in the outside–inside direction of the pipe bend and straight pipe. In Fig. 14, the photos of Reynolds number from 1500 to 2500 of each curved nozzles and straight pipe were taken by Nakashima (Nakashima, 2019). In addition, we added the visualization taken by the same ways as Nakashima et al. under the condition of Reynolds number 3000 for each curved nozzles. The visualization results of oil jet behavior show that as the Reynolds number increases, the interface becomes wavy downstream for both bent nozzles. Furthermore, a comparison of the starting points of wave generation outside and inside the bend shows that the outside is wavy from upstream. However, the interface in the case of the straight nozzle is stable despite the large Reynolds number. These results indicate that the wavy interface at a short distance from the nozzle exit in the case of the bent nozzle is not created by Kelvin–Helmholtz instability but by nozzle internal flow.

Figure 15 shows the width of the jet in the outside–inside direction. Each point in Fig. 15 is a time average of the
measured width of the oil jet for \( n = 48,000 \) measuring points during 3.0 s obtained at a sampling frequency of 16,000 Hz and the error bar is the standard deviation (\( \sigma_w \)) defined by Eq. (8). Figure 15 shows that the time average of the width of the jet tends to decrease immediately after the nozzle exit. At large Reynolds numbers, the jet width begins to increase early upstream and becomes large downstream. The jet for \( R = 15 \) mm is wider than that for \( R = 60 \) mm at any distance from the nozzle tip regardless of the Reynolds number. The standard deviation in Fig. 16 indicates that the time variation of the width is large for a large Reynolds number and small radius of curvature. This tendency is the same for a long time average of the jet width. These results reveal that the average width increases downstream with large time fluctuation. It is thought that the jet behavior is affected by the flow in the bend pipes, owing to the bend of the pipes and Reynolds number. The flow in the nozzle is discussed in the next section.

\[
\sigma_w = \frac{1}{n} \sum_{j=1}^{n} (w_j - \bar{w})^2
\]

(8)

Fig. 14 Oil jet behavior for different Reynolds numbers: (left) \( R = 15 \) mm, (center) \( R = 60 \) mm, and (right) straight pipe. The visualization results for both curved pipes oil jet behavior show that as the Reynolds number increases, the interface becomes wavy downstream. However, the interface in the case of the straight nozzle is stable despite the large Reynolds number 3000.

Fig. 15 Ejected oil jet width and standard deviation for different Reynolds numbers: (left) \( R = 15 \) mm and (right) \( R = 60 \) mm. Time average of the measured width of the oil jet for \( n = 48,000 \) measuring points during 3.0 seconds obtained at a sampling frequency of 16,000 Hz and the error bar is the standard deviation (\( \sigma_w \)) defined by Eq. (8).
Fig. 16 Standard deviation ($\sigma_u$) was defined by Eq. (8) at each measurement points. (left) $R = 15$ mm and (right) $R = 60$ mm. These results indicate that the time variation of the width is large for a large Reynolds number and small radius of curvature.

4.2 Time-synchronous PIV and visualization of jet behavior

4.2.1 Time-averaged nozzle internal flow

For the time-averaged distribution, 2500 pairs of PIV images that were taken with the high-speed video camera over a period of 0.25 s were used. In the case of $Re=1000$, which has the smallest flow velocity case, area averaged velocity is 2.5 m/s, and the measurement region is 20 mm in front of the nozzle exit. Therefore, the measurement time of 0.25 s is sufficient because it takes 0.008 s for the particles to pass through the region. Furthermore, the residuals of the flow speed ($U_{mag}$) that was calculated using Eq. (9) at 2.0 mm upstream from the exit were calculated using Eq. (10) to verify the validity of the number of PIV images. At First, in this calculation Eq. (10), the summation of the absolute values of the difference between the averaged values of 100 pairs from the start of measurement of each calculation grid and the averaged values of 200 pairs including the following 100 pairs is obtained. In addition, the summation result is divided by 16 that is the number of calculation grid. Secondly, the total absolute value of the difference among the average values of 300 pairs, which is the sum of the average values of the 200 pairs and the following the 100 pairs, is obtained from the start of measurement. The same method was repeated to calculate the difference of the average values between $(k+1)\times100$ and $k\times100$ $(k=1$ to 24) pairs from the start of measurement to verify the convergence. As a result, the residual $e$ converges to about 0.05 or less with 500 pairs or more as shown Fig. 17. Therefore, it is considered enough to use 2500 pairs to obtain the averaged velocity.

Figure 18 shows the time-averaged nozzle internal flow speed ($U_{mag}$) distribution that was calculated using Eq. (9) from 15 mm before the bend to 2 mm before the exit. Flow speed in the nozzles are separately obtained in several parts expressed by solid squares using PIV. As the Reynolds number increases, the centrifugal force becomes strong at the bend for both $R = 15$ mm and $R = 60$ mm because of the high flow speed $U_{mag}$. The main stream therefore approaches the outside.

In this paper, we focused on the 20 mm before the exit as shown in Fig. 19 for investigation of the effect of the nozzle internal flow on oil jet behavior. Figure 19 shows the time-averaged distribution 20 mm before the exit area of the flow speed ($U_{mag}$) in the measurement XZ plane of each nozzle for each Reynolds number. The $X$ axis gives the radial position of the nozzle, with the positive direction inside and the negative direction outside in Fig. 19.

The results for $R = 15$ mm under the condition of large $Re$ value 3000 in Fig. 19 show that the steep distribution of the mainstream flow speed is likely to disappear near the nozzle outlet. It is thought that this is because the flow in the pipe radial direction becomes relatively large as the $Re$ number increases in $Re$, and the mainstream spreads in the pipe radial direction due to the influence of viscosity. Meanwhile, the results for $R = 60$ mm in Fig. 12 show that the steep distribution of mainstream flow speed becomes clear at $Re$ values ranging 1000 to 3000.

Figure 20 shows the flow speed at 2.0 mm upstream from the nozzle outlet. The error bar shows the standard deviation $\sigma U_{mag}$ calculated using Eq. (11). It is confirmed that the main flow is biased to the outside for all conditions of $Re$ and $R$ owing to the centrifugal force as shown in Fig. 19.

Figure 21 compares the standard deviation at the point of maximum flow in Fig. 20 for each Reynolds number and each nozzle. It is found that the standard deviation, indicating the degree of time variation of flow at the mainstream position, tends to be larger for $R = 15$ mm than for $R = 60$ mm as the Reynolds number increases. It is considered that there is a fluctuation state of the flow speed internal nozzle.
\[ U_{mag} = \sqrt{U_x^2 + U_z^2} \]  
\[ \varepsilon_{(k+1)\times100} = \left( \frac{\sum_{i=0}^{15} |U_{magi}^{(k+1)\times100} - U_{magi}^{k\times100}|}{16} \right) \]

Fig. 17 Residual ε histories of the sum of the absolute values of the difference between the averaged values of each calculation grid. Each R and Reynolds number 2.0mm upstream from the exit were calculated Eq. (10). The residuals of all condition almost converged with 500 pairs or more images. (left) R = 15 mm and (right) R = 60 mm

![Image](image1.png)

(a) Re =1000 (left) R = 15mm, (right) R = 60mm

![Image](image2.png)

(b) Re =3000 (left) R = 15mm, (right) R = 60mm

Fig. 18 The time-averaged nozzle internal flow speed (\( U_{mag} \)) distribution that was calculated using Eq. (9) from 15mm before bend to 2mm before the exit. These results were taken several parts surrounded by a solid square separately.
Fig. 19 PIV result of the 0.25s-time-averaged distribution of the flow speed ($U_{\text{mag}}$) calculated using Eq. (9) in the measurement XZ plane of each nozzle for each Reynolds number. The X axis gives the radial position of the nozzle, with the positive direction inside and the negative direction outside. (a) $R = 15$ mm and (b) $R = 60$ mm.

Fig. 20 Flow speed in the nozzle

2.0 mm upstream of the exit: (left) $R = 15$ mm and (right) $R = 60$ mm

$$\sigma_{U_{\text{mag}}} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (U_{\text{mag},i} - U_{\text{mag}})^2}$$  (11)
4.2.2 Time-synchronous measurement of nozzle internal flow and oil jet behavior

The cross-correlation coefficient was calculated from the PIV explained in Section 2.3 as shown in Fig. 12 to analyze the fluctuation state of the flow speed. The cross-correlation coefficient was calculated for a position from -19.6 to -3.3 mm at 1.09 mm intervals in the Z direction of the reference range set at 1.8 mm × 1.8 mm (5 cells × 5 cells) on the outside edge in the base PIV image. Each line in Fig. 22 shows typical calculation result. In the case of large offset length, the position at -4.4 mm and -3.3 mm, the cross-correlation coefficient cannot be calculated correctly because it was out of range. The horizontal axis gives the offset length while the vertical axis gives the correlation coefficient. Error bars indicating the standard deviation are calculated using Eq. (12). The result shows that the maximum correlation coefficient Ș appeared at almost the same offset length for each base window position for each case. Figure 23 compares the fluctuation Lagrangian speed U_L calculated using Eq. (13) obtained by dividing the arithmetic mean offset length using time-averaged offset length that has the maximum correlation coefficient at each calculated position ΔZ(max S_z) by t', and the average flow speed in the five cells that is the same area as each window for cross-correlation coefficient on the outside obtained by PIV in Section 2.3. As the error bar of U_L, the value defined by dividing a half cell by t' is used. It was found that the fluctuation speed is almost identical to the flow speed obtained from PIV. Figures 24 and 25 show the time history of the flow speed distribution. The squares drawn with blue broken lines and having dimensions of 1.8 mm × 1.8 mm in Figs. 24 and 25 move at the Lagrangian speed for each Reynolds number in Fig. 23. The fluctuation is due to flow changing with time owing to the secondary flow similarly to shown by Nakashima (2018) in Fig. 1, and this is the cause of the small and large standard deviations seen in Section 4.2.1.

As the Reynolds number increases, there is a large fluctuation in flow speed in the radial direction. In contrast, there is little mainstream fluctuation for R = 60 mm in Fig. 25. Figures 26 and 27 show the ratio U_ratio_x of the flow velocity component U_L in the radial direction of the pipe to the absolute value U_mag of the flow speed expressed by Eq. (14), and the simultaneous jet behavior with the ratio U_ratio_x. Positive U_ratio_x means the flow has velocity components from the outside to the inside. The rectangle drawn with red broken lines moves a fixed amount Δz (max S_z) per unit time for the color contours of U_ratio_x in Figs. 26 and 27 and moves on the jets that were photographed synchronously with PIV. The U_ratio_x distribution after the bend does not change if vortices shapes generated by secondary flow would be similar under all flow conditions, and secondary flow intensity that was defined by Timité (2010) is proportional to Reynolds numbers. Meanwhile, it is expected that the U_ratio_x distribution changes when the secondary flow is affected by the Reynolds number and/or shape of the pipe bend. Figure 26 shows that the absolute U_ratio_x is smaller than 0.1 when the Reynolds number is small (Re = 1000) for R = 15 mm, but the absolute value of U_ratio_x increases over 0.2 and the U_ratio_x fluctuation inside the nozzle strengthens with a Reynolds number increasing beyond 2000. Meanwhile, under the present experimental conditions, the distribution of U_ratio_x at R = 60 mm hardly changes and is not affected by the Reynolds number as shown in Fig. 27. Next, the fluctuation in the nozzle and the behavior of the jet are discussed. At Reynolds numbers of 2000 and 3000 for R = 15 mm and Reynolds numbers of 3000 for R = 60 mm where the jet remarkably undulates, U_ratio_x in the outside area becomes larger than that at a low Reynolds number. The mark position in the jet interface corresponds to the fluctuation in the nozzle and the wave growth. It is considered that the fluctuation of flow in the nozzle destabilizes the jet wave.

\[
\sigma_{S_z} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (S_{z,i} - \bar{S}_z)^2} \tag{12}
\]

\[
U_L = \frac{\Delta Z_{(max \ S_z)}}{t'} \tag{13}
\]

\[
U_{ratio_x} = \frac{U_x}{U_{mag}} \tag{14}
\]
Fig. 23 Flow speed comparison between Lagrangian Speed that was calculated by using cross-correlation method Eq (4) in section 2.3 and PIV each Reynolds number.
Fig. 24 The time history of the flow speed distribution of $R = 15\text{mm}$. The squares drawn with blue broken lines and having dimensions of $1.8\text{ mm} \times 1.8\text{ mm}$ in Figure move at the Lagrangian speed for each Reynolds number in Fig. 23.
Fig. 25 The time history of the flow speed distribution of $R = 60$ mm. The squares drawn with blue broken lines and having dimensions of $1.8 \text{ mm} \times 1.8 \text{ mm}$ in Figure move at the Lagrangian speed in Fig. 23.
Fig. 26 Time-synchronous PIV and visualization of jet behavior for \( R = 15 \) mm. The squares drawn with red broken lines and having dimensions of 1.8 mm \( \times \) 1.8 mm in Figure move at the Lagrangian speed in Fig. 23.
Fig. 27 Time-synchronous PIV and visualization of jet behavior for $R = 60$ mm. The squares drawn with red broken lines and having dimensions of 1.8 mm $\times$ 1.8 mm in Figure move at the Lagrangian speed in Fig. 23.
5. Conclusions

Characteristics of oil jets ejected from 90° curved nozzles with different radii of curvature were investigated by measuring the jet width using an LED displacement gauge and simultaneously measuring the oil jet behavior and nozzle internal flow through visualization and time-resolved PIV, respectively. The following results were obtained.

1. Visualization of the jet revealed large interfacial fluctuation under the condition of a small radius of curvature of the bend \( R = 15 \text{ mm} \) and large Reynolds number, and the oil jet interface became unstable in the downstream direction.

2. Measurement of the jet width revealed that not only the jet width but also its standard deviation downstream became large, and the oil jet interface tended to be unstable for small \( R \) and a large Reynolds number.

3. The time average of the flow speed obtained by PIV revealed that the steep distribution of flow speed was distinct at \( R = 60 \text{ mm} \) even when the Reynolds number was large, whereas the steep distribution at \( R = 15 \text{ mm} \) almost disappeared from the upstream to the vicinity of the nozzle outlet under the condition of large Reynolds number \( Re = 3000 \). Furthermore, it was found that the standard deviation in the calculation cell having the maximum flow speed under each condition was larger for \( R = 15 \text{ mm} \) than for \( R = 60 \text{ mm} \) at all Reynolds numbers, and the flow field fluctuated greatly inside the nozzle.

4. The PIV time history revealed that the flow in the nozzle was accompanied by a fluctuation state of flow speed in the nozzle, and the fluctuation state of flow speed in the nozzle propagated at a constant speed downstream. The cause of the fluctuation state of flow speed in the nozzle is considered to be secondary flow, and in the case that \( R = 15 \text{ mm} \), the fluctuation state of flow speed in the nozzle becomes strong as the effect of the flow in the radial direction of the pipe increases with the Reynolds number. As a result, the time average of the steep distribution of the flow speed almost disappears.

5. Synchronous measurement of the jet behavior and PIV revealed that the fluctuation state of flow speed in the nozzle triggered the generation of waves at the jet interface. Furthermore, when the absolute ratio of the flow velocity component in the radial direction of the pipe to the absolute value of the flow speed increases, the fluctuation of the interface between the air and oil becomes stronger.

In this paper, we investigated two radii of curvature regarding the behavior of an oil jet ejected from a curved pipe. Then, as described above, the effect of the fluctuation caused by the secondary flow on the oil jet interface, which was unknown in the previous studies, was clarified.

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