Experimental study on oil-air multiphase flow in a right circular cylindrical channel during reciprocating motion

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Received: 21 October 2018; Revised: 18 December 2018; Accepted: 10 January 2019

Abstract
Much attention has recently been given to high-efficiency cooling of pistons in internal combustion engines by cooling channels because of improved thermal efficiency. Cooling a piston efficiently requires a grasp of the gas-liquid multiphase flow state. However, because the magnitude and direction of the inertial force applied to the piston change depending on the crank angle, the flow field in the cooling channel that forms a complex gas-liquid multiphase flow remains a problem. Therefore, we developed a rig test apparatus simulating the reciprocating motion of a piston and visualized internal flows in a clear acrylic channel using a high-speed camera. This paper examines the effects of the Reynolds number of the oil jet and oscillation frequency of the reciprocating motion on flow characteristics in a right circular cylindrical channel. The Reynolds number and oscillation frequency were tested in the ranges 1000 to 2500 and 0 to 8.33 Hz, respectively. We found that the oil flow pattern in the channel forms the complex air-oil multiphase flow via air entrainment caused by the collision between the oil jet and the channel oil at the inlet. The gas phase area ratio in the channel increases with increasing Reynolds number, but its fluctuation is dominated by oscillation frequency. The fluctuation of the gas phase center of gravity becomes larger because of increases in inertial force with increases in oscillation frequency. The average bubble diameter in the channel decreases with increasing Reynolds number and oscillation frequency. We found that bubbles of small diameter are generated because the interfacial fluctuation of the oil jet increases as the jet goes downstream.

Keywords : Internal combustion engine, Piston cooling, Cooling channel, Oscillation frequency, Thermal efficiency, Particle image analyzer

1. Introduction
Thermal management of pistons in internal combustion engines has recently received broad attention because of improvements in specific power. Although the piston has the highest thermal load in the combustion chamber (comprising a piston, cylinder head and cylinder liner), it is more difficult to cool the piston with engine coolant than other parts in the combustion chamber are.

Two types of cooling systems using engine oil have been commonly used for pistons: an oil jet inflow into a piston cooling channel (Thiel et al., 2007 and Deng et al., 2017) and an oil jet impingement on the underside of the piston (Agarwal and Goyal, 2007 and Luff et al., 2012). To improve thermal efficiency and suppress knocking, the piston cooling channel has been used in recent years. In this method, the momentum force of the oil jet inflow to the piston cooling channel and the inertial force caused by the reciprocating motion of the piston are used to promote heat exchange. Because the magnitude and direction of the inertial force applied to the piston changes with varying crank angle, the flow field in the cooling channel forms a complex gas-liquid multiphase flow. Therefore, to cool a piston efficiently, it is important to grasp the gas-liquid multiphase flow state depending on channel shape, engine conditions
such as piston frequency, dimensionless parameters of engine oil such as oil jet Reynolds numbers and so on. However, oil jets in internal combustion engines generally have a Reynolds number (Re) of 2500 or more, and a large amount of oil is injected. When the oil jet flow rate increases, the cooling performance improves because the amount of oil inflow to the channel increases, but because a high oil pressure is required to increase the flow rate, the driving energy of the oil pump increases. In addition, the increase in the amount of oil causes many problems such as increasing pumping loss, deteriorating emissions and increasing oil consumption because the amount of oil mist scattered by the reciprocating motion of the piston increases. Therefore, it is necessary to develop piston cooling technologies that control the heat transfer in the piston cooling channel with low oil jet flow rates.

Many experimental and numerical studies have been conducted on the cooling characteristics of piston cooling channels in internal combustion engines. Leites and Camargo (1993) measured the effects of several oil jet nozzles and skirt configurations on piston temperature in a diesel engine. Thiel et al. (2007) investigated in detail the effects of oil jet number and arrangement and cooling gallery shape on piston temperature using an automotive diesel engine. Torregrosa et al. (2010) also proposed a correlation of membrane coefficients related to convective heat transfer through an oil gallery in the piston geometry of a diesel engine. In an analytical study, Kajiwara et al. (2003) developed a new two-dimensional (2D) approach to predict piston temperature based on the amount of oil inflow to the gallery using computational fluid dynamics (CFD). Pan et al. (2005), Yi et al. (2007), Wang et al. (2017) and Deng et al. (2017) simulated oil flow and heat transfer using three-dimensional (3D) CFD for piston cooling gallery structures used in actual engines. Deng et al. (2018) also performed multi-objective optimization of the cooling channel position in the piston with temperature and thermal stress as the objective functions using 3D-CFD. Previous studies on engines have often been based on numerical simulation because it is difficult to grasp oil behavior inside a piston during operation. However, examining numerical simulation requires comparing with experimental results. For this reason, previous studies have focused on simple channel shapes and rig tests to grasp the behavior of gas-liquid multiphase flow in the channel under piston reciprocating motion and improve predictions of simulations. Nozawa et al. (2005) visualized the behavior of oil in the clear acrylic dummy piston channel and measured the amount of heat transfer around an aluminum dummy channel of the same shape. They showed that oil behavior at the channel outlet depends on backflow due to oil retention near the top dead center and that the heat transfer increases as more oil passes. Komiya et al. (2011) used light scattering to visualize the purified water-air multiphase flow inside a clear acrylic piston cooling channel installed to inject the jet from top to bottom during reciprocating motion. They observed that the oil in the cooling channel flows spirally because of the reciprocating motion of the piston. In addition, Lv et al. (2015) investigated the effects of the water filling ratio and oscillation frequency (fwa) on water-air multiphase flow using a simple closed rectangular channel. They showed that the optimum heat transfer coefficient is obtained at the water filling ratio in the range of 40 to 60%. Although there have been studies on visualizations of gas-liquid multiphase flow inside cooling channels and measurements of heat transfer at the wall, there are few studies on the effects of engine speed and oil jet flow rate on flow characteristics such as gas phase behavior and bubble diameter distribution. Therefore, to grasp the gas-liquid multiphase flow state toward optimum piston cooling method, it is first necessary to clarify the flow characteristics inside cooling channel during the reciprocating motion.

In this study, we examined the effects of Re and fwa on oil flow characteristics using a simple clear acrylic right circular cylindrical channel. We developed a reciprocating motion apparatus that injects the oil jet from the bottom to the top and visualized detailed oil flow behavior using background illumination. We analyzed the effects of Re and fwa on the gas phase area ratio and center of gravity in the channel. The bubble diameter distribution in the channel was also measured by a particle image analyzer (PIA) to assess the effects of Re and fwa on gas-liquid multiphase behavior.

Nomenclature

- $A_{air}$: Projected gas phase area in the channel (pixel)
- $A_{all}$: Projected area of the entire channel (pixel)
- $A_b$: Projected area of bubble (mm$^2$)
- $B(x, z)$: Output value after binarization ($B(x, z) = 0$ and 1 indicate the air and the oil, respectively)
- $D$: Inner channel diameter (mm)
- $D_{b/10}(\theta)$: Average bubble diameter at each crank angle (μm)
- $\bar{D}_{b/10}$: Average bubble diameter during one cycle (μm)
- $G_{air}$: Gas phase center of gravity position (mm)

[DOI: 10.1299/jfst.2019jfst0002] © 2019 The Japan Society of Mechanical Engineers
L Channel length (mm)

$L_b$ Peripheral length of bubble (mm)

N Number of class values in the bubble diameter distribution

Re Oil jet Reynolds number

Q Oil jet flow rate (m³/s)

$\bar{W}$ Area-averaged streamwise velocity at the nozzle exit (m/s)

$Z_{TH}$ Theoretical displacement of the channel during reciprocating motion (m)

d Inner diameter of the nozzle tip (m)

$d_b$ Bubble diameter (μm)

$f_{osc}$ Oscillation frequency of reciprocating motion (Hz)

l Connecting rod length of reciprocating motion apparatus (m)

n Number of bubbles at each class value

r Crank radius of the reciprocating motion apparatus (m)

$x, z$ Cartesian coordinates (mm)

$\alpha_{TH}$ Theoretical acceleration of the channel during reciprocating motion (m/s²)

$\theta$ Crank angle (degree)

$\lambda$ Ratio of the crank radius to the connecting rod length, $\lambda = r / l$

$\nu$ Kinetic viscosity of oil (m²/s)

$\omega$ Angular velocity of the crankshaft (rad/s)

2. Experimental setup

2.1 Reciprocating motion apparatus for visualizing oil behavior

Figure 1 shows a schematic of the reciprocating motion apparatus. The oil jet was injected upward by a vortex pump (20KHD07Z, Nikuni), and the flow rate was adjusted by the vortex pump frequency using an inverter (FR-D720, Mitsubishi Electric). The flow rate and oil temperature were precisely measured using a Coriolis flowmeter (FD-SS20A, Keyence). The vortex pumps generate drift or vortex flow downstream after the pump. Therefore, to rectify the oil flowing into the nozzle, we installed a flow straightener (FS Series, Oval) and a pipe between the nozzle and flowmeter. The flow straightener had an inner diameter of 35.5 mm and length of 400 mm and the pipe had an inner diameter of 14.9 mm and length of 1200 mm. A crank mechanism was applied for the reciprocating motion of the channel. The crankshaft was installed in the upper part of the apparatus so as not to affect the visualization area by oil scattering during rotation. Therefore, the channel displacement during reciprocating motion was reversed between the top dead center (TDC) and bottom dead center (BDC) compared with the piston displacement of the engine. In the reciprocating motion, the drive frequency of the motor (TFO-LK 4P 3.7kW, Hitachi Industrial Equipment Systems Co.) was adjusted by the inverter (FR-E720, Mitsubishi Electric), and the crankshaft was rotated via the two pulleys (Pulley-A/Pulley-B = 2). The crank radius, connecting rod length and ratio of the crank radius to the connecting rod length were $r = 40$ mm, $l = 160$ mm and $\lambda = 0.25$, respectively. In the channel section, four guide rods were installed and slid by ball splines. In addition, a signal for rotation of the crankshaft was acquired by an optical detector (LG-9200, Ono Sokki Co.) and displayed on a digital tachometer (TM-3110, Ono Sokki Co.).

Fig. 1 Schematic of the experimental apparatus for visualizing oil behavior.

Fig. 2 Sketch of the clear acrylic nozzle.
Figure 2 shows the dimensions of the oil jet nozzle. A clear acrylic curved nozzle was used as a test nozzle. The inner diameter of the nozzle outlet was $d = 6$ mm, the radius of curvature was 30 mm, and the straight portion length after bending was 30 mm. Figure 3 (a) shows the channel shape used in this study. We designed a clear acrylic right circular cylindrical channel with one inlet and two outlets. Figure 3 (b) shows the dimensions of the channel. The channel had a cylindrical shape with a diameter of $D = 20$ mm and a length of $L = 100$ mm, with an inlet with a diameter of 12 mm at the center and an outlet with a diameter of 8.5 mm. We used a space simulating a skirt with a height of 20 mm in the lower part of the channel. In addition, to avoid oil scattering to the visualized region during reciprocating motion, a cover was installed in the lower part. The distance between the channel inlet at BDC and the nozzle tip was set to 30 mm.

![Figure 2](image1.png)

(a) Three-dimensional model.  
(b) Dimensions.

**Fig. 3** Simple clear acrylic right circular cylindrical test channel.

### 2.2 Measurement method

Oil flow behavior and bubble diameter distributions in the channel were visualized by background illumination. Figure 4 shows the method of visualizing oil-air multiphase flow. A high-speed camera (SA-Z, Photron) with 1024 × 1024 pixels was used to film the flow behavior in the channel during reciprocating motion. A manual focus lens (AI Micro-Nikkor 105mm f/2.8S, NIKON CORPORATION) with a focal length of 105 mm was used for the lens of the camera. Light from an LED (LLBK Series, Aitec System Co.) was diffused by a diffusion plate (#47-952, Edmond Optics). The camera inputted the start trigger from the tachometer acquiring the rotation pulse of the crankshaft. To analyze the gas-liquid multiphase flow, the visualized image was binarized to classify the gas and liquid phases on the 2D image as $B(x, z)$. Binarization was carried out using the Otsu method (Otsu, 1979) to obtain a threshold value representing the maximum separation metric.

Figure 5 shows the method of visualizing the bubble diameter distribution in the channel. In this measurement, a CCD camera (AM-200GE, JAI) with 1200 × 1600 pixels was used to obtain a cycle average image at an arbitrary crank position. The optical interference photometer (Nd:YAG laser) was used to measure the bubble diameter distribution in the channel.
angle. The lens of the camera used a lens common to oil behavior visualization measurement. To visualize a distinct bubble interface, the light source used a Nd: YAG pulsed laser (20Hz, LS-2132, LOTIS TII), the stripe was removed by an optical interference reducer, and diffused light was irradiated by the diffuser. Because the output signal pulses from the tachometer included not only the rotation signals of the crankshaft but also high-frequency noises, low-pass filter processing was performed by a programmable filter (3624, NF Corporation). The signals of the camera and the laser were synchronized using a digital delay generator (VSD2000, Flowtech Research). The bubble diameter was calculated using particle tracking velocimetry software FtrPIV (Flowtech Research), which can measure spherical and nonspherical bubble diameters (Nishino, 1993). Therefore, the bubble diameter was indicated by the equivalent diameter obtained from the bubble area.

2.3 Validation of channel displacement in reciprocating motion apparatus

Figure 6 shows the comparison between the theoretical and measurement values (4.17 and 8.33 Hz). The theoretical displacement $Z_{TH}$ during the reciprocating motion in our apparatus can be approximated by the following function of the crank angle $\theta$:

$$Z_{TH} \cong r \left(1 + \cos \theta \right) + \frac{\lambda}{4} (1 - \cos 2\theta) ,$$

(1)

where $\theta = 0$ indicates the TDC position and $Z_{TH} = 0$ the BDC position. For the measured values, the displacement was acquired from three cycle images by cross-correlation under the condition that the oil jet was not injected using the method of Fig. 4. From this result, we found that measured values corresponded reasonably well with the theoretical values. Therefore, we investigated oil behavior in the channel under the reciprocating motion using this apparatus.

![Fig.6 Comparison of theoretical and experimental values in reciprocating motion apparatus.](image)

![Fig.7 Observation ranges for visualizing oil behavior and determining bubble diameter distribution. The area-A and area-B are defined as the red and orange frames, respectively.](image)

2.4 Experimental conditions

Ambient pressure was 99.6 ± 0.3 kPa, and ambient temperature and oil temperature were adjusted within 298 ± 1.5 K to reduce the influence of the temperature difference between them. For a surrogate we used silicone oil (KF56A, Shin-Etsu Chemical) close to the kinetic viscosity of the engine oil (SAE 0W30) at 353 K. The density, kinetic viscosity, surface tension and refractive index of this oil were 995 kg/m$^3$, 1.5×10$^{-5}$ m$^2$/s, 24.4 mN/m and 1.498, respectively, at 298 K. The three $f_0$ values used in this study were 0 (stationary condition), 4.17 and 8.33 Hz.

To visualize oil behavior, $Re$ was set in the range from 1000 to 2500 and the nozzle diameter was taken as the representative length. $Re$ was calculated using the equation

$$Re = \frac{Wd}{\nu} ,$$

(2)
where the area-averaged streamwise velocity at the nozzle outlet is \( \bar{W} = 4Q/\pi d^2 \). Measurements consisted of 100 images of the stationary condition \( \theta_{\text{in}} = 0 \) Hz at a rate of 50 fps at crank angles 0, 90, 180 and 270 deg. and 5 cycles of reciprocating motion \( \theta_{\text{in}} = 4.17 \) and 8.33 Hz) at 5000 fps. The diaphragm of the lens was set to F8 and the spatial resolution of the images were measured at 0.1493 mm/pix.

To determine the bubble diameter distribution, the phase-averaged images were acquired at crank angles 0, 90, 180 and 270 deg. We set \( Re \) in the range from 1000 to 2000 and acquired 100 images of the stationary condition at 20 fps and 100 cycles of reciprocating motion. The diaphragm of the lens was set to F8 and the spatial resolution of the images were measured at 0.0192 mm/pix.

The observation ranges were visualized in two areas for each measurement as shown in Fig. 7. Measurements visualized the range from the oil jet nozzle tip to the channel at the TDC (Area-A). To obtain the bubble diameter distribution, we performed enlarged photography in a range of \( \pm 0.75D \) from the channel center (Area-B).

3. Results and discussion

3.1 Visualization of oil behavior during reciprocating motion

3.1.1 Oil flow pattern for different values of the oil jet Reynolds number

Figure 8 shows the oil behavior images at each crank angle for different values of \( Re \) at \( \theta_{\text{in}} = 4.17 \) Hz. The black region in the channel indicates the gas phase. Conversely, the black region between the nozzle tip and the skirt indicates oil because it is directly irradiated by the background light. The oil jet interface behavior became complicated with increasing \( Re \). For \( Re = 2500 \), the interfacial fluctuation of the inner (right) side of the jet became larger than the fluctuation on the outer (left) side of the jet. Our previous study (Nakashima et al., 2018) on oil jets injected by curved nozzles also found fluctuation due to the flow toward the inner side of the bend in the curved pipe. However, the variation of the gas phase inflow distribution at the inlet portion of the channel was small, and there was no significant difference in multiphase flow patterns in the left and right regions within the channel. It was found that the effect of the asymmetry of the oil jet interface by the curved nozzle on gas-liquid multiphase flow patterns in the channel decreases under the condition that the gas phase inflow is large.

During reciprocating motion, the region occupied by the gas phase (black region in the channel) increased with increasing \( Re \). At \( Re = 2500 \), the entire flow channel was occupied by gas phase area. We observed that the increase of gas phase area was caused by air entrainment due to collision between the oil jet and oil in the channel at the inlet. These results demonstrate that inflow amount of the gas phase into the channel can be controlled by the difference in the shape specification of the channel inlet portion. For \( Re = 1000 \) where the channel interior could be clearly monitored, the behavior of the gas-liquid phase for each crank angle changed greatly because of the difference in direction of inertial force. For the crank angle at which the channel descends \( (\theta = 90 \text{ deg.}) \), the liquid phase was biased toward the upper part of the channel, and the gas phase was biased toward the lower part. Conversely, for the crank angle at which the channel rises \( (\theta = 270 \text{ deg.}) \), the gas-liquid phase had the opposite behavior. These behaviors were similar to the observations of Nozawa et al. (2005). Moreover, the amount of oil discharged from the outlet tended to increase with increasing \( Re \). For the crank angle at which the channel descends \( (\theta = 90 \text{ deg.}) \), the oil discharge from the outlet was stagnant for \( Re = 1000 \). When \( \theta \) reaches 90 deg., it is important to replace the high-temperature oil with fresh oil after the combustion stroke before the next cycle. Therefore, we suggest that oil exchange efficiency deteriorates at low \( Re \).

The effect of the reciprocating channel motion on jet behavior decreased because of the increase of the jet momentum with increasing \( Re \). At \( Re = 1000 \), the shape of the jet was deformed by the oil returning from the inlet at \( \theta = 270 \text{ deg.} \), when the channel rises. Conversely, a decrease in the oil returning from the inlet was observed due to the increase in the momentum of the jet and the spread of the jet width under the condition of \( Re = 2500 \). These results demonstrate that fresh oil before heat exchange in the channel during the compression stroke is likely to be discharged from the channel inlet under condition of a low flow rate.

Furthermore, we observed that the amount of the oil mist between the nozzle tip and the skirt of the channel increased with increasing \( Re \). Oil mist was caused by the splashing of oil adhering to the skirt when the channel rises. We found that the amount of scattered oil increased because of the increase in oil attached to the inner wall of the skirt with increasing \( Re \). In addition, we observed that the generated oil mist diameter did not change significantly with varying \( Re \).
3.1.2 Oil flow pattern for different oscillation frequencies

Figure 9 shows the oil behavior at each crank angle for different values of $f_{os}$ at $Re = 1000$. We observed that the behavior of oil varied with $f_{os}$. When the stationary condition $f_{os} = 0$ Hz, the inflow of the gas phase from the channel inlet portion was observed similar to conditions of the reciprocating motion, and an annular flow was formed in the channel irrespective of the crank angle. As $f_{os}$ increased, the annular flow collapsed and the occurrence of fine bubbles in the channel was observed. This is because the effect of the cocktail shaker increased with the increasing inertial force. The bias of the gas phase increased with increasing $f_{os}$ due to the effect of the inertial force. In particular, at $\theta = 90$ deg. (descending channel), the gas phase concentrated in the lower region of the channel with increasing $f_{os}$, but the gas flowing from the inlet was distributed in the upper region of the channel inlet portion.

In observing oil jet behavior during the reciprocating motion, we found that the jet shape changed with increasing $f_{os}$. For $f_{os} = 4.17$ and 8.33 Hz, we observed that the oil jet was obstructed by return oil from the inlet when the channel was rising ($\theta = 0$ and 270 deg.). These results show that the increase in $f_{os}$ affects the oil inflow to the channel.

Furthermore, the amount of oil mist in the lower channel region increased with increasing $f_{os}$. The amount of oil adhering to the inner wall of the skirt increased because of the increase in the return oil from the inlet with increasing $f_{os}$, and the oil mist scattering increased. In addition, we observed that the generated oil mist diameter decreased as $f_{os}$ increased.

Fig.8 Oil behavior at crank angles 0, 90, 180 and 270 deg. for different values of $Re$ at $f_{os} = 4.17$ Hz.
3.1.3 Effects of Reynolds number and oscillation frequency on channel flow

To investigate the multiphase flow of air and oil in the channel, the gas phase area ratio \((A_{\text{air}}/A_{\text{all}})\) and gas phase center of gravity \((G_{\text{air}})\) were examined using binarized images. Figure 10 shows the results for the area ratio (a) and the center of gravity (b) at \(f_{\text{os}} = 0, 4.17\) and \(8.33\) Hz. The channel acceleration \((a_{TH})\) was obtained by second-order time differentiation of the channel displacement in Eq. (1), and is given by

\[
 a_{TH} = \omega^2 \frac{d^2 Z_{TH}}{d\theta^2} \approx r \omega^2 (-\cos \theta + \dot{\lambda} \cos 2\theta),
\]

(3)

where the angular velocity of the crankshaft is \(\omega = 2\pi f_{\text{os}}\). The gas phase center of gravity was obtained in the stroke direction \((z)\) using the channel center as the origin (Fig. 7) via the following equation:

\[
 G_{\text{air}} = \frac{\int_{A_{\text{air}}} (1 - B(x, z))zdA}{A_{\text{air}}} = \frac{\int_{A_{\text{air}}} (1 - B(x, z))zdA}{\int_{A_{\text{air}}} (1 - B(x, z))dA}.
\]

(4)

At \(f_{\text{os}} = 0\) Hz, the channel flow characteristics changed with varying crank angle even though the effect of the crank
angle on the gas phase area ratio and center of gravity was small (Fig. 9) for reciprocating motion \( f_{\text{os}} = 4.17 \) and 8.33 Hz. The gas phase area ratio increased irrespective of \( f_{\text{os}} \) with increasing \( Re \). Furthermore, when \( Re \) exceeded 2000, the gas occupied a region of 90% or more, and the effect of \( f_{\text{os}} \) on the gas phase area ratio became small. This is because the oil mixed with small air bubbles caused by collision between the oil jet and channel oil at the inlet was distributed throughout the entire channel as shown in Figs. 8 and 9. The gas phase center of gravity approached the channel center because of the increase in the gas phase area with increasing \( Re \). Although the area ratio for \( f_{\text{os}} = 4.17 \) and 8.33 Hz gradually decreased with increasing \( Re \), the behavior of the fluctuation tended not to change during the cycle. Therefore, we found that the gas phase area ratio increased with increasing \( Re \) and its fluctuation during reciprocating motion did not depend on \( Re \).

When \( Re = 1000 \), the gas phase area ratio behaved differently with increasing \( f_{\text{os}} \). Fluctuations in the ratio were noticeable, and the trends for \( f_{\text{os}} = 4.17 \) and 8.33 Hz were similar. The amplitude of the gas phase center of gravity increased with increasing \( f_{\text{os}} \). However, although the shift in the maximum gas phase center of gravity toward the upper region channel decreased, the shift in the minimum gas phase center of gravity toward the lower region was significantly different. In addition, at \( f_{\text{os}} = 4.17 \) and 8.33 Hz, the gas phase area ratio reached its minimum during channel descent, and both the area ratio and center of gravity became constant from 180 to 270 deg. After 270 deg., the area ratio began to increase.

To consider these characteristics, channel images were compared. Figure 11 shows images in the channel at \( \theta = 45, 135, 165, 225 \) and 315 deg. for \( Re = 1000 \). We observed that two liquid phases in the lower region and both sides of the channel increased at the crank angles for which the gas phase area ratios decreased (\( \theta = 45 \) and 135 deg.). The liquid phase in the lower channel region increased because of the downward inertial force caused by the acceleration of the upward channel. We also observed an increase in inflow and discharge at the outlet. Therefore, we found that the liquid phases on both sides increased because the discharge from the outlet increased with increasing relative velocity at channel descent. At \( \theta = 165 \) deg., when the gas phase area ratio increased, the gas flowing from the inlet spread toward both sides. At \( \theta = 225 \) deg., when this area ratio became constant, we observed that the gas from the inlet continuously flowed into the upper channel region. In addition, the gas phase center of gravity at this crank angle was at its maximum position, and the gas in the channel formed the most biased flow field in the upper region. Therefore, we found that the gas phase area ratio became constant because both the gas in the channel and the gas from the inlet

![Fig. 10 Observed gas phase area ratio (a) and center of gravity (b) at \( f_{\text{os}} = 0 \) (left), 4.17 (center) and 8.33 (right) Hz. The black dotted line indicates the acceleration of the channel at each \( f_{\text{os}} \). The error bars at \( f_{\text{os}} = 0 \) Hz indicate the maximum and minimum values of 100 data. The error bars at \( f_{\text{os}} = 4.17 \) and 8.33 Hz indicate the maximum and minimum values in 5 cycles.](image-url)
formed a flow biased toward the upper channel region. At $\theta = 315$ deg., when the gas phase area ratio increased again, we observed that the gas was distributed throughout the entire channel. This is because the distribution observed at $\theta = 225$ deg. diffused when the acceleration switched from positive to negative.

![Flow bias](image)

Fig. 11 Channel images at crank angles 45, 135, 165, 225 and 315 deg. for $\omega = 4.17$ (left) and 8.33 (right) Hz at $Re = 1000$.

### 3.2 Measurement of bubble diameter distribution in the channel

As shown in Figs. 8 and 9, from the results of oil behavior visualization in the channel, it was found that gas phase distributions were different in the channel depending on $Re$ and $f_{os}$. However, since it was difficult to grasp the detailed gas phase behavior such as bubble diameter distributions in the channel, we performed enlarged photographing at the inlet portion of the channel using Nd: YAG pulse laser as shown in Fig. 5. Figure 12 shows the example of enlarged photographed images for $Re = 1000$ and 2000 at $f_{os} = 4.17$ Hz. The gas phase in the channel was clearly visualized by enlarged photographing using the high-power pulse laser as a light source.

![Bubble diameter](image)

Fig. 12 Example of enlarged photographing images for $Re = 1000$ and 2000 at $f_{os} = 4.17$ Hz. The analysis range of the bubble diameter is defined as the blue dotted frame.
Therefore, to study the effects of \(Re\) and \(f_{oa}\) on bubble characteristics in the channel, we obtained the bubble diameter distributions in the liquid phases. The analysis range of the bubble diameter was performed in a region of 5 mm from the upper end of the channel and 5 mm from its lower end near the channel inner wall important for piston cooling in the range of \(\pm 0.5D\) from the channel center as shown in Fig. 12. Moreover, to exclude the overlap of bubbles, only bubbles satisfying the condition of roundness > 0.3 were obtained. The roundness is given by the following equation:

\[
\text{Roundness} = \frac{4 \pi A_b}{L_b^2}.
\]  

Figure 13 shows the average bubble diameter \(D_{b10}\) during one cycle using the average bubble diameter \(D_{b10}\) at the crank angles 0, 90, 180 and 270 deg. at \(f_{oa} = 0\), 4.17 and 8.33 Hz. We defined \(D_{b10}(\theta)\) and \(\overline{D}_{b10}\) as follows:

\[
D_{b10}(\theta) = \frac{\sum_{i=1}^{N} n_i d_{bi}}{\sum_{i=1}^{N} n_i},
\]

\[
\overline{D}_{b10} = \frac{1}{4} \{D_{b10}(\theta = 0) + D_{b10}(\theta = 90) + D_{b10}(\theta = 180) + D_{b10}(\theta = 270)\}.
\]

The average bubble diameter decreased with increasing \(Re\) and \(f_{oa}\). However, although the average bubble diameter decreased with increasing \(f_{oa}\) at low \(Re\), the effect of \(f_{oa}\) became small when \(Re\) reached above 1500. This is because the effect of stirring by reciprocating motion decreased as the gas phase area ratio in the channel increased.

Figure 14 shows the bubble diameter distribution for different values of \(Re\) at \(f_{oa} = 0\), 4.17 and 8.33 Hz. The distributions varied with varying crank angle. However, the bubble diameter formed distributions with a peak at 50 \(\mu\)m regardless of \(Re\) and \(f_{oa}\). In addition, the generation frequency of bubbles of mode diameter increased with increasing \(Re\) and \(f_{oa}\). Under the stationary condition of \(f_{oa} = 0\) Hz, the bubble diameters at 0 and 90 deg. formed a sharp distribution compared with the bubble diameter at 180 deg. In other words, when the distance from the nozzle tip to the channel was increased, the number of small bubbles increased. This shows that bubbles of small diameter were generated because the interfacial fluctuation of the oil jet increased as the jet went downstream as shown in Fig. 8. However, under the reciprocating motion with \(f_{oa} = 4.17\) and 8.33 Hz, the behavior of the bubble diameter distributions was the same as that at \(f_{oa} = 0\) Hz except for \(\theta = 270\) deg. For \(\theta = 90\) deg., we concluded that the bubble diameter became small because air entrainment at the inlet was strong when the channel descended. Conversely, at \(\theta = 270\) deg., we concluded that the effect of air entrainment decreased because the oil jet was obstructed by the oil outflow in the channel at the inlet, as shown in Figs. 8 and 9.

![Fig.13 Average bubble diameter \(\overline{D}_{b10}\) during one cycle using the average bubble diameter \(D_{b10}\) of crank angles 0, 90, 180 and 270 deg. at \(f_{oa} = 0\) (red), 4.17 (blue) and 8.33 (green) Hz](image-url)
The effect of reciprocating motion on oil jet behavior increases with increasing motion on oil jet behavior.

4. Conclusions

We studied the effects of $Re$ and $f_{in}$ on channel flow using a simple clear acrylic right circular cylindrical channel having one inlet and two outlets. In addition, we analyzed the bubble diameter distribution in the channel. The main conclusions from this work are as follows:

1. The oil flow pattern in the channel forms a complex air-oil multiphase flow. The increase of gas phase area is caused by air entrainment due to collision between the oil jet and oil in the channel at the inlet. It is demonstrated that inflow amount of the gas phase into the channel can be controlled by the difference in the shape specification of the channel inlet portion. On the other hand, it is found that the effect of the asymmetry of the oil jet interface by the curved nozzle on gas-liquid multiphase flow patterns in the channel decreases under the condition that the gas phase inflow is large. The effect of reciprocating motion on oil jet behavior decreases with increasing $Re$. However, the effect of reciprocating motion on oil jet behavior increases with increasing $f_{in}$. It is demonstrated that fresh oil before heat exchange in the channel during the compression stroke likely to be discharged from the channel inlet under condition of a low flow rate. It is found that oil mist...
is generated as the oil adhering to the skirt scatters when the channel rises. The amount of oil mist increases with increasing $Re$ and $f_{oa}$, but the increase in $f_{oa}$ also decreases the oil mist diameter.

(2) The gas phase area ratio in the channel increases with increasing $Re$, but its fluctuation during the cycle is dominated only by $f_{oa}$. The fluctuation of the gas phase center of gravity increases because of the increase in inertial force with increasing $f_{oa}$. However, the fluctuation decreases because the gas phase area ratio increases with increasing $Re$.

(3) The average bubble diameter in the channel decreases with increasing $Re$ and $f_{oa}$. However, the bubble diameter distributions have a peak at 50 $\mu$m regardless of $Re$ and $f_{oa}$. Small bubbles are generated because the interfacial fluctuation of the oil jet increases as the jet goes downstream. At $\theta = 90$ deg., the bubble diameter becomes small because air entrainment at the inlet is strong when the channel descends. Conversely, at $\theta = 270$ deg., it is found that the effect of air entrainment decreases because the oil jet is obstructed by the oil outflow in the channel at the inlet.

Oil behavior in the channel affects the gas phase area ratio and bubble diameter via air entrainment caused by the collision between the oil jet at the inlet and oil in the channel. Therefore, inlet specifications are important for controlling oil behavior in the channel. Our future work will focus on the effect of specifications such as inlet diameter and shape on multiphase channel flow.

Acknowledgments

The authors thank Mark Kurban, M. Sc., from Edanz Group (www.edanzediting.com/ac) for editing a draft of this manuscript.

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[DOI: 10.1299/jfst.2019jfst0002] © 2019 The Japan Society of Mechanical Engineers
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