VOID FRACTION CHARACTERISTICS OF ONE-COMPONENT GAS–LIQUID TWO-PHASE FLOW IN SMALL DIAMETER TUBES

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When designing a two-phase flow loop system for space structures, it is necessary to understand the effect of surface tension on the gas–liquid two-phase flow behaviors. In this paper, the effect of the tube diameter on the void fraction characteristics of vertical upward one-component gas–liquid two-phase flows in small diameter (inner diameter = 4.0, 2.0, 1.1, and 0.5 mm) circular tubes is discussed. FC-72 was used as the working fluid. The void fraction was measured by a capacitance method and the flow behaviors were observed simultaneously using a high-speed camera. The transition boundaries of the observed flow pattern agreed well with the Mishima and Ishii model, except in the case of the 0.5 mm tube. The gas volumetric flux at the transition from slug to annular flow for the 0.5 mm tube was higher than that obtained by the previous model. For the annular flow, the average void fractions were overestimated by the homogeneous model and Cioncolini and Thome model due to the high slip ratio. In the correlation based on the drift–flux model, the distribution parameter for the annular and churn flows in each tube agreed well with the Zuber annular flow model and the Ishii churn flow model, respectively. The drift velocities were overestimated by these models, except in the case of the 4.0 mm tube. The disturbance wave frequency for annular flow increased when the tube diameter was decreased to 1.1 mm; however, for the 0.5 mm tube the frequency decreased due to enhancement of the surface tension effect.

KEY WORDS: adiabatic gas–liquid two-phase flow, void fraction, small diameter tube, drift–flux model, capacitance method

1. INTRODUCTION

Recently, the two-phase flow loop thermal control system has been attracting attention for space structures in order to remove large amounts of heat with high heat flux from equipment and to transport the heat to a radiator over a long distance. In this system, a cold plate is cooled by ebullient cooling with a high heat transfer coefficient, and then the heat is transferred to a radiator with a vapor–liquid two-phase flow. For the design of the system, it is necessary to clarify the thermal hydraulic characteristics of gas–liquid two-phase flows, for example, the flow pattern, void fraction, pressure drop, and heat transfer characteristics. Specifically, to realize this system for space structures, it is necessary to clarify the two-phase flow characteristics under microgravity.

Several forces act on two-phase flows, such as gravity, inertial force, and surface tension. Since gravitational acceleration becomes close to zero in space, the effect of surface tension may become relatively larger. On the other hand, the effect of surface tension will also become relatively larger when decreasing the tube diameter regardless of the gravitational acceleration. If the surface tension is dominant in the two-phase flow characteristics under normal gravity, the difference in two-phase flow behaviors between normal and microgravity conditions will become negligible, and the system will be able to be designed using two-phase flow correlations for normal gravity conditions. Therefore, it is important to clarify the operating conditions, including the tube diameter, where the effect of gravity is negligible.
To clarify the dominant force in gas–liquid two-phase flows, three dimensionless numbers are introduced, i.e., the Bond (Bo) number (= gravity/surface tension), Weber (We) number (= inertial force/surface tension), and Froude (Fr) number (= gravity/inertial force). These three dimensionless numbers are defined as follows:

\[ \text{Bo} = \frac{(\rho_L - \rho_G) g D^2}{\sigma} \]  
\[ \text{We} = \frac{DG^2}{\rho_{tpf} \sigma} \]  
\[ \text{Fr} = \sqrt{\frac{\text{We}}{\text{Bo}}} = \frac{G^2}{\rho_{tpf} (\rho_L - \rho_G) g D} \]

where \( G \) is total mass flux, and \( \rho_{tpf} \) is average density of the two-phase mixture. The average density is defined as follows:

\[ \rho_{tpf} = (1 - \alpha) \rho_L + \alpha \rho_G \]

Since the heat transfer and pressure drop depend on the flow pattern, modeling of these characteristics should be constructed according to the dominant force map of gas–liquid two-phase flows.

On a dominant force region map, Baba et al. (2012) evaluated the effect of the dominant force on the heat transfer coefficient using FC-72 as the working fluid. The effect of gravity was considered by changing the tube orientation, such as horizontal and vertical upward and downward flows. The effect of surface tension was also considered by using different tube diameters (0.51 and 0.13 mm). Baba et al. (2012) suggested from the results that the boundary between gravity and the inertial force dominant region was given at Fr ≈ 4, and the boundary between surface tension and the inertial force dominant region was given at We ≈ 5. In their paper, the mean density of two-phase mixtures was defined by the homogeneous flow model.
Regarding the effect of surface tension, flow pattern maps in millimeter-scale tubes have been reported in many studies. For instance, Mishima and Hibiki (1996) confirmed that the Mishima and Ishii (1984) prediction model of flow pattern transition boundaries were applicable to vertical upward air–water two-phase flows in 4.08, 3.12, 2.05, and 1.05 mm diameter tubes. Zhao and Rezkallah (1993) arranged the flow pattern map based on dimensionless numbers for air–water two-phase flows in a 9.525 mm diameter tube under microgravity conditions. The Weber numbers for the gas and liquid phases, \( \text{We}_G \) and \( \text{We}_L \), respectively, were applied to the flow pattern map, and it was shown that the flow patterns could be classified into three regions depending on the gas Weber number. Bubbly and slug flow \( \text{We}_G \) values of less than 1 in the region were considered to be the surface tension dominant region. Annular flow in the range of \( \text{We}_G \) values over 20 was considered to be the inertial force tension dominant region. In the intermediate region with \( \text{We}_G \) values of 1–20, frothy slug and frothy annular flows were observed. Akbar et al. (2003) proposed the flow regime transition boundaries of air–water two-phase flows in horizontal circular channels with the hydraulic equivalent diameter \( D_H \) of less than 1.0 mm. The results were arranged by using Weber numbers as the coordinates. The flow regimes were classified into four zones: (1) the surface tension dominant zone (bubbly, plug, and slug flows); (2) the first inertial force dominant zone (annular flow); (3) the second inertial force dominant zone (dispersed flow); and (4) the transition zone. Chen et al. (2006) carried out flow visualization experiments for HFC-134a adiabatic two-phase flows in vertical small diameter tubes (inner diameters of 1.10, 2.01, 2.88, and 4.26 mm) to clarify the effect of tube diameter on the flow patterns. They concluded from the results that flows for a diameter below 2.01 mm were dominated by surface tension, and the transition boundaries could be correlated to the Weber number.

Since the heat transfer coefficient and pressure drop of gas–liquid two-phase flows is strongly affected by the gas–liquid interface structure and slip velocity, it is important to clarify the flow pattern map and the void fraction characteristics. The information on void fractions in normal diameter tubes (with a diameter over 10 mm) has been well provided in previous studies, and many correlations have been proposed. On the other hand, there are some papers that have focused on two-phase flows in microchannels. However, the information on void fractions in moderate diameter tubes is still insufficient. Mishima and Hibiki (1996) used neutron radiography to measure the averaged void fraction of air–water two-phase flows in a vertical tube with varied diameters of 4.08, 3.12, 2.05, and 1.05 mm. The measured void fractions were arranged based on the drift–flux model. It was reported that for bubbly and slug flows the distribution parameter was increased with decreasing tube diameter, and the drift velocity became zero for smaller diameter tubes. Saisorn and Wongwises (2008, 2015) investigated the flow pattern, void fraction, and pressure drop characteristics of air–water two-phase flows in vertical and horizontal tubes with a diameter of 0.53 mm. Only the void fractions in slug, throat annular, and annular rivulet flows were measured from images taken by a high-speed camera. It was reported that the measured void fractions agreed well with the calculated results obtained by the homogeneous flow model for horizontal flow; however, the measured void fractions for vertical upward flow tended to be lower than those obtained by the homogeneous flow model. A similar tendency in relation to the void fraction characteristics was reported by Barreto et al. (2015), in which the void fractions of air–water two-phase flows in a 1.2 mm diameter vertical tube were measured by an impedance sensor with high sampling frequency. They reported that the measured void fractions agreed well with the calculated results obtained using the Lockhart and Martinelli (1949) correlation, and the slip ratio was larger than previous results obtained by Oliveira et al. (2009) for churn flow in a 21 mm diameter vertical tube. Several research studies on void fraction characteristics under microgravity have been reported. Elkow and Rezkallah (1997) used a capacitance method to measure the void fraction in vertical air–water two-phase flows in a 9.53 mm diameter tube under normal and microgravity conditions. The measured results for bubbly and slug flows were compared with those obtained using the drift–flux model proposed by Zuber and Findlay (1965). They reported that the average void fractions for slug flows under microgravity were higher than those under normal gravity. On the other hand, for slug annular, churn, and annular flows, the differences between the results obtained under normal and microgravity conditions were slight because these flows were dominated by inertial force. Narcy et al. (2014) measured the void fraction, pressure drop, and heat transfer coefficient of HFE-7000 two-phase flows in a 6.0 mm diameter tube, and considered the gravity effect by comparing the measured results under microgravity with those under normal gravity. They reported that the void fractions under microgravity were larger than those under normal gravity, and the void fractions under microgravity were in reasonable agreement with the calculated results obtained using the Cioncolini and Thome (2012) prediction model. Although some researchers have evaluated the effect of gravity by comparing the measured results obtained under microgravity conditions and some
researchers have evaluated the effect of surface tension from experimental results using various tube diameters, the understanding of the void fraction characteristics in small diameter tubes—especially in relation to the tube diameter boundary and whether or not surface tension dominates two-phase flow behaviors—is insufficient.

The purpose of this study is to clarify the transition boundaries in the dominant force map, and especially to clarify the condition in which the effect of gravity becomes less in comparison to the effect of surface tension. The void fraction characteristics of adiabatic vertical upward one-component two-phase flows in small diameter tubes were measured using a capacitance method. The tube diameter was varied from 0.5 to 4.0 mm in order to change the effect of surface tension on two-phase flows. Section 2 describes the experimental apparatus and conditions, including the details of the capacitance sensor used to measure the void fraction. In Section 3, the experimental results on the flow pattern transitions and cross-sectional average void fractions are discussed and compared with the predicted results obtained by several previous models.

2. EXPERIMENTAL APPARATUS AND CONDITIONS

A schematic diagram of the experimental setup is shown in Fig. 1(a). Vertical upward two-phase flows were examined using FC-72 as the working fluid. The physical properties of FC-72 are given in Table 1. Four tubes with diameters of 4.0, 2.0, 1.1, and 0.5 mm were used. The accuracy of the tube diameter was confirmed in a preliminary test by evaluating the frictional pressure loss characteristics of the liquid single-phase flows.

The measurement accuracies of the tube diameter were ±0.10, ±0.05, ±0.035, and ±0.02 mm for the 4.0, 2.0, 1.1, and 0.5 mm diameter tubes, respectively. Liquid FC-72 was circulated by a gear pump driven by a DC motor through

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**FIG. 1:** Schematic diagram of the experimental setup: (a) Test section, (b) Detail of test section.

Interfacial Phenomena and Heat Transfer
TABLE 1: Physical properties of FC-72 at the saturated condition

<table>
<thead>
<tr>
<th>$P$ (kPa)</th>
<th>$T_{\text{sat}}$ (°C)</th>
<th>$\rho_L$ (kg/m$^3$)</th>
<th>$\rho_G$ (kg/m$^3$)</th>
<th>$\sigma$ (mN/m)</th>
<th>$\mu_L$ (μPa·s)</th>
<th>$\mu_G$ (μPa·s)</th>
<th>$C_p$ [kJ/(kg·K)]</th>
<th>$r$ (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>101.3</td>
<td>56.8</td>
<td>1621</td>
<td>13.5</td>
<td>8.44</td>
<td>413.3</td>
<td>11.9</td>
<td>1.10</td>
<td>83.6</td>
</tr>
<tr>
<td>150.0</td>
<td>68.9</td>
<td>1591</td>
<td>19.7</td>
<td>7.37</td>
<td>351.5</td>
<td>12.4</td>
<td>1.13</td>
<td>80.0</td>
</tr>
<tr>
<td>220.0</td>
<td>81.8</td>
<td>1554</td>
<td>28.7</td>
<td>6.27</td>
<td>297.8</td>
<td>13.0</td>
<td>1.16</td>
<td>76.1</td>
</tr>
</tbody>
</table>

a magnet coupling (Micropump Type 187P for low flow rate, and Type 185P for high and low flow rate produced by Chuorika CO., Ltd. in Japan). The volumetric flow rate was measured by a Coriolis mass flow meter (Type CMFS010 produced by Emerson Electric Co. in US; measuring range: maximum 10 g/s, measuring accuracy: ±0.1% of the reading value). The test section consisted of a heating section and an observation section. The inlet temperature was set by a pre-heater, and then sub-cooled liquid was supplied to the test section. The details of the test section are shown in Fig. 1(b). In the heating section, the working fluid became two-phase flows by Joule heating. The input AC voltage and electric current were measured by an AC voltmeter (Type 452F-24A produced by Tsuruga Electric Corp.; measuring range: maximum 10 V$_{\text{rms}}$, measuring accuracy: ±0.2% of the reading value) and an ammeter (Type 452F-24A produced by Tsuruga Electric Corp. in Japan; measuring range: maximum 50 A$_{\text{rms}}$, measuring accuracy: ±0.5% of the reading value). The local wall temperatures were measured by K-type thermocouples mounted on the outside of a thin wall stainless steel (SUS) tube with the same diameter as the observation section and with a wall thickness of 0.2 mm. These thermocouples were made of each material wire. The wire diameter was 0.2 mm for the heating section, with 4.0 and 2.0 mm inner diameter tubes, and 0.1 mm for the heating section, with 1.1 and 0.5 mm inner diameter tubes. To decrease the heat loss to the surroundings, the heating section was covered by a heat insulator. The temperatures of the working fluid were measured by K-type sheathed thermocouples ($\phi = 0.5$ mm) inserted into the channel at the points indicated in Fig. 1(a), with an accuracy of 0.05% of the reading value. The pressures and differential pressure were measured at the points shown in Fig. 1(b) by pressure transducers (Type NS100A produced by Minebea Co., Ltd. in Japan; measuring range: maximum 500 kPa, measuring accuracy: 0.5% of the rated output) and a differential pressure transducer (DP15-38 produced by Validyne Engineering Corp. in US; measuring range: ±55 kPa, measuring accuracy: ±0.25% of the full scale). At the observation section, a capacitance sensor was used for the void fraction measurements and a high-frame-rate camera was used for flow behavior observation (Type MotionXtra N3 produced by Integrated Design Tools, Inc. in US); both the measurements and observations were conducted simultaneously at the same measuring position. The experimental loop equipment was connected by the SUS tubes with a 4.0 mm inner diameter.

The detailed structure of the observation section is shown in Figs. 2(a) and 2(b). A parallel plate--type capacitance sensor was selected for the electrode arrangement, and the surfaces of the sensing, guard, and ground electrodes were placed in contact with a glass tube. The capacitance of two-phase flows was measured between the sensing and ground electrodes by using a high-sensitivity capacitance meter (Type HC-102 produced by Yamamoto Electric Instrument Co., Ltd. in Japan; sensing resolution 10⁻¹² pF). The guard electrode was attached around the sensing electrode with insulation to reduce stray capacitance. For each tube diameter, $D$, the length of the sensing area of void fraction, $L_L$, was set as $L_L/D = 4.0$. The specifications of the test section are summarized in Table 2. The volumetric average void fraction was measured from the measured capacitance, $\varepsilon$, using the following equation:

$$\varepsilon_{\text{capa}} = \frac{(\varepsilon_L - \varepsilon)}{(\varepsilon_L - \varepsilon_G)} \tag{5}$$

where $\varepsilon_G$ and $\varepsilon_L$ are the capacitance of the vapor and liquid single phases, respectively. Equation (5) was confirmed in a static calibration by inserting Teflon rods with varied diameters simulated as the liquid phase. The relative permittivity of each material is given in Table 3.
by the quick closing valve method, the measurements were repeated 30–50 times for each flow condition. On the other hand, in the measurement by the capacitance method, the average void fraction was obtained by time averaging in 60 seconds. Figure 3 confirms that the measuring accuracy of the capacitance sensor is within 5%. Since the phase distribution can be estimated to be axisymmetric in vertical upward flows, the measurement accuracy of the void fraction by the capacitance sensor for 4.0, 2.0, 1.1, and 0.5 mm diameter tubes can be considered to be within 5%.

The experimental conditions are summarized in Table 4. The thermal equilibrium quality at the observed section can be calculated by the following equation:

$$x = \frac{h_{\text{in}} + (Q_{\text{input}} - Q_{\text{loss}})/M - h_{L_{\text{sat}}}}{h_{L_{\text{sat}}} - h_{G_{\text{sat}}}}$$  \hspace{1cm} (6)$$

where $h_{L_{\text{sat}}}$ and $h_{G_{\text{sat}}}$ denote the specific enthalpy of the saturated liquid and vapor, respectively; and $Q_{\text{input}}$ and $Q_{\text{loss}}$ denote the input heat and heat loss at the heating section, respectively. The heat loss was estimated from the
TABLE 2: Specifications of the test section (unit: mm)

<table>
<thead>
<tr>
<th>D</th>
<th>L_{ent}</th>
<th>L_{heat}</th>
<th>L_{capa}</th>
<th>L_{TC}</th>
<th>L_{L}</th>
<th>L_{W}</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.0</td>
<td>30</td>
<td>800</td>
<td>55</td>
<td>20</td>
<td>16.0</td>
<td>8.0</td>
</tr>
<tr>
<td>2.0</td>
<td>30</td>
<td>400</td>
<td>55</td>
<td>40</td>
<td>8.0</td>
<td>16.0</td>
</tr>
<tr>
<td>1.1</td>
<td>30</td>
<td>200</td>
<td>55</td>
<td>40</td>
<td>4.4</td>
<td>2.0</td>
</tr>
<tr>
<td>0.5</td>
<td>30</td>
<td>100</td>
<td>55</td>
<td>80</td>
<td>2.0</td>
<td>4.4</td>
</tr>
</tbody>
</table>

TABLE 3: Relative permittivity of each material at 25°C

<table>
<thead>
<tr>
<th>Material</th>
<th>Air</th>
<th>FC-72</th>
<th>Glass</th>
<th>Teflon</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative permittivity</td>
<td>1.00</td>
<td>1.00</td>
<td>1.74</td>
<td>3.7</td>
</tr>
</tbody>
</table>

FIG. 3: Comparison of the measured average void fractions by the capacitance method with those obtained using the quick valve closing method for a 4.0 mm diameter tube.

TABLE 4: Experimental conditions

<table>
<thead>
<tr>
<th>D (mm)</th>
<th>G [kg/(m² · s)]</th>
<th>x (%)</th>
<th>P_{in} (kPa)</th>
<th>Bo (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.0</td>
<td>30–500</td>
<td>–0.1–0.98</td>
<td>105–160</td>
<td>30.5–34.5</td>
</tr>
<tr>
<td>2.0</td>
<td>50–600</td>
<td>–0.1–0.93</td>
<td>105–180</td>
<td>7.63–9.06</td>
</tr>
<tr>
<td>1.1</td>
<td>70–600</td>
<td>–0.1–0.91</td>
<td>105–220</td>
<td>2.31–2.95</td>
</tr>
<tr>
<td>0.5</td>
<td>200–600</td>
<td>–0.1–0.75</td>
<td>105–185</td>
<td>0.48–0.56</td>
</tr>
</tbody>
</table>

mass flux of the refrigerant and the average wall temperature of the heating section for each experimental condition. The measurement accuracies of the mass flux and quality were ±5% and ±11%, respectively. The Bond numbers defined by Eq. (1) for the 4.0, 2.0, 1.1, and 0.5 mm diameter tubes were calculated to be about 32, 8.0, 2.5, and 0.51 for each pressure condition.
3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Flow Pattern Transition

The flow patterns observed at the observation section were classified as bubbly, slug, churn, semi-annular, and annular flows. In this study, the churn and semi-annular flows were distinguished based on whether or not intermittent liquid reverse flows existed. Churn flow is more affected by gravity. While all flow patterns, except churn flow, were observed for each tube diameter, churn flows were observed in 4.0 and 2.0 mm tubes only at low mass flux conditions. Examples of typical flow behaviors in the 2.0 mm tube are shown in Figs. 4(a)–4(d) with void fraction fluctuations measured by the capacitance sensor. It can be clearly seen that the void fraction was successfully measured in synchronization with the flow behavior.

The flow pattern maps for each tube diameter are shown in Fig. 5, where the horizontal and vertical axes show the volumetric gas and liquid fluxes, respectively. The broken lines show the flow pattern transition boundaries proposed by Mishima and Ishii (1984). These boundaries were based on theoretical relations for normal diameter tubes. The volumetric gas and liquid fluxes are defined as follows:

\[ j_G = \frac{G_x}{\rho_G} \quad j_L = \frac{G(1-x)}{\rho_L} \]  

(7)

Churn flows were observed only at low mass flux conditions in 4.0 and 2.0 mm diameter tubes because the gravity effect was relatively large. For low-quality conditions, the transition boundary from slug to bubbly flow was not well predicted by the Mishima and Ishii (1984) model. The reason for this might be a result of the difference between the thermal equilibrium quality and the actual quality in the experiments.

For the 4.0, 2.0, and 1.1 mm tubes under high-quality conditions, the transition boundaries from slug to churn, slug to annular, and churn to annular flows were well predicted by the Mishima and Ishii (1984) model when semi-annular flows were categorized as intermittent flow (similar to slug flow). On the other hand, for the 0.5 mm tube the transition boundary from slug to annular flow shifted to a higher \( j_G \) value than with the prediction model. Since the surface tension on a meniscus in the 0.5 mm tube was larger, higher vapor velocity might be required to form a vapor core. Therefore, the effect of surface tension on two-phase flow dynamics might become larger for the 0.5 mm tube.

A diagram of the flow pattern map of the liquid and gas Weber numbers (\( W_{EL} \) and \( W_{EG} \), respectively) is shown in Fig. 6 with the prediction model proposed by Akbar et al. (2003). The \( W_{EL} \) and \( W_{EG} \) values are defined for the volumetric flux by the following equations:

\[ W_{EL} = \frac{\rho_L j_L^2 D}{\sigma} \quad W_{EG} = \frac{\rho_G j_G^2 D}{\sigma} \]  

(8)

In the prediction model, flow regimes are categorized into four regimes, including a transition region. Annular flow is categorized as the inertia-dominant region. Intermittent flow, including bubbly and slug flows, is categorized as the surface tension dominant region. In the range of \( W_{EL} \) values less than 3.0, the transition boundary between the inertia-dominant zone and the transition zone was well predicted for each tube diameter. However, the other transition boundaries did not agree with the results obtained by the prediction model. Since the prediction model was constructed based on the experimental results of air–water two-phase flows, the prediction model might not be applicable to adaptation for one-component flows with low surface tension and smaller density ratios of the liquid to gas phase, as with FC-72.

3.2 Average Void Fraction

In this section, the average void fraction characteristics for semi-annular and annular flows are discussed. The measured void fractions are plotted against vapor quality for varied mass flux conditions in Fig. 7. The calculated results obtained using the homogenous, Cioncolini and Thome (2012), and Awad and Muzychka (2014) models for annular flow are also plotted by red, blue, and pink broken lines, respectively. The equations for the Cioncolini and Thome (2012) and Awad and Muzychka (2014) models are given in Eqs. (9) and Eq. (10), respectively:

\[ \]
FIG. 4: Typical relations between flow behaviors and change in void fraction for each flow regime in 2.0 mm tube: 
(a) Churn flow ($G = 50 \text{ kg/(m}^2\text{ s)}$, $x = 0.08$, $\alpha_{ave} = 0.75$), (b) Churn flow ($G = 50 \text{ kg/(m}^2\text{ s)}$, $x = 0.09$, $\alpha_{ave} = 0.75$), (c) Semi-annular flow ($G = 150 \text{ kg/(m}^2\text{ s)}$, $x = 0.15$, $\alpha_{ave} = 0.75$), (d) Annular flow ($G = 50 \text{ kg/(m}^2\text{ s)}$, $x = 0.53$, $\alpha_{ave} = 0.89$).
FIG. 5: Flow pattern maps with transition boundaries proposed by Mishima and Ishii (1984).

\[
\alpha = \frac{hx^n}{1 + (h - 1)x^n}; \quad 0 < x < 1, \quad 10^{-3} < \frac{\rho_G}{\rho_L} < 1
\]

\[
h = -2.129 + 3.129 \left(\frac{\rho_G}{\rho_L}\right)^{0.2186}, \quad n = 0.387 + 0.6513 \left(\frac{\rho_G}{\rho_L}\right)^{0.515}
\]

\[
\alpha = \frac{0.5}{1 + \left(\frac{1 - x}{x}\right)^{7/8} \left(\frac{\rho_G}{\rho_L}\right)^{1/2} \left(\frac{\mu_G}{\mu_L}\right)^{1/8}}^{16/19} + \frac{0.5}{1 + 0.28 \left(\frac{1 - x}{x}\right)^{7/8} \left(\frac{\rho_G}{\rho_L}\right)^{1/2} \left(\frac{\mu_G}{\mu_L}\right)^{1/8}}^{0.71}
\]

The measured values were smaller than those obtained using the homogeneous model, and the Cioncolini and Thome (2012) and the Awad and Muzychka (2014) models tended to overestimate the values. The root-mean-square errors (RMSEs) of the measured void fractions compared with the predicted results using these three models are shown in Fig. 8 for each tube diameter and mass flux. The differences between the measured and calculated results increased when the mass flux and tube diameter were decreased. Under the high mass flux conditions, the effect of the mass flux on the void fraction was minimal for each tube diameter. However, under the low mass flux conditions [less than 100 kg/(m² · s)], the void fractions became smaller when the mass flux was decreased. The effect of surface tension might become relatively larger than the effect of inertial force at low mass flux.

Next, the average void fractions were arranged based on the drift–flux model. In the drift–flux model, the mean gas velocity is expressed as the total volumetric flux by the following equation:

\[
u_{xG} = j_G / \alpha_{ave} = C_0 j + v_G j
\]
where $C_0$ and $v_{Gj}$ are the distribution parameter and mean drift velocity, respectively. Generally, correlations are made for each flow pattern. For bubbly and slug flows, Mishima and Hibiki (1996) proposed the drift–flux model for flows in small diameter tubes, and Colin et al. (1991) proposed the drift–flux model for flows under microgravity conditions. However, the information available on churn and annular flows in small diameter tubes is insufficient.

Correlations based on the drift–flux model were made for the experimental results of churn and semi-annular flows and annular flow. The obtained results are shown in Figs. 9 and 10 for each flow pattern, respectively. The experimental results are plotted according to the flow pattern by using the same symbols as in Fig. 5. The experimental results are compared with the previous correlations. The results for churn and semi-annular flows were compared with those obtained using the Ishii (1977) model and the results for annular flow were compared with those obtained using the Zuber et al. (1967) model. In the previous models, the distribution parameter, $C_0$, and drift velocity, $v_{Gj}$, are expressed by Eqs. (12) and Eqs. (13). The correlations by each model are plotted by solid and those obtained from the experimental results by the least-square method are plotted by broken lines in Figs. 9 and 10.
FIG. 7: Comparison of the measured void fractions with the predicted results obtained by the homogeneous, Cioncolini and Thome (2012), and Awad and Muzychka (2014) models for annular flow.

- The Ishii model for churn flow:

\[ C_0 = 1.2 - 0.2 \sqrt{\frac{\rho_G}{\rho_L}}, \quad v_{Gj} = \sqrt{2 \left\{ \frac{\sigma g (\rho_L - \rho_G)}{\rho_L^2} \right\}^{0.25}} \]  

(12)

- The Zuber model for annular flow:

\[ C_0 = 1.0, \quad v_{Gj} = 23 \sqrt{\frac{\mu_L J L}{\rho_G D}} \left( \frac{\rho_L - \rho_G}{\rho_L} \right) \]  

(13)

In the calculations for the distribution parameter and drift velocity, the average values of the physical properties at the observation section were used.

Figures 9 and 10 confirm that the experimental results for the values of \( C_0 \) agree well with the prediction models for churn and annular flows. Mishima and Hibiki (1996) showed in their results that the distribution parameters for annular flow in small diameter tubes, with diameters of 1.0–4.1 mm, agreed well with the results obtained by the previous model, although for bubble and slug flows the distribution parameter increased when the diameter in this range was decreased.
FIG. 8: Comparison of the RMSEs of the measured void fractions to the predicted results obtained by the homogeneous, Cioncolini and Thome (2012), and Awad and Muzychka (2014) models for each tube diameter.

FIG. 9: Average void fractions for churn (crosses) and semi-annular flow (open circles) arranged based on the drift-flux model compared to the Ishii (1977) model for churn flow.
FIG. 10: Average void fractions for annular flow arranged based on the drift-flux model compared to the Zuber et al. (1967) model for annular flow.

On the other hand, the experimental results for the drift velocity, \( v_{Gj} \), for churn flow agreed well with the Ishii (1977) model only for the 4.0 mm tube. For the smaller tubes with 2.0 and 1.1 mm diameters, the obtained values became close to zero. A similar tendency was reported by Mishima and Hibiki (1996) for bubble and slug flows. The effect of surface tension might become larger in such smaller diameter tubes, making the slip velocity lower. For the 0.5 mm tube, the drift velocity became higher for the churn flow (up to 0.65 m/s) and annular flow (up to 1.01 m/s), despite the decrease in the tube diameter. For the annular flow, the drift velocity for each tube diameter was overestimated by the Zuber et al. (1967) prediction model. It can be said that the effect of surface tension should be considered for these conditions. It was also confirmed that the drift velocity became larger with decreasing tube diameter, especially for the 0.5 mm diameter tube (\( v_{Gj} = 1.01 \text{ m/s} \)). The tendency in the drift velocity was almost the same among churn, semi-annular, and annular flows.

The experimental results of the void fraction for the annular flow were compared with the calculated results obtained by the prediction models proposed by Cioncolini and Thome (2012) and Zuber et al. (1967). The results are plotted for each mass flux in Fig. 11, using the same symbols as in Fig. 7, where it can be seen that the Zuber et al. (1967) model predicts the void fractions within 5% for the 4.0 mm diameter tube; however, for the 2.0, 1.1, and 0.5 mm diameter tubes, the model underestimates the void fractions in the range of \( \alpha < 0.9 \). On the other hand, the Cioncolini and Thome (2012) model estimates the void fractions within \( \sim 5\% \) under high mass flux conditions; however, under low mass flux conditions, the model overestimates the void fractions.

3.3 Effect of the Tube Diameter on the Liquid Film Structure in Annular Flow

To consider the void fraction characteristics of the annular flows in detail, the liquid film structures were estimated from the void fraction fluctuations measured by the capacitance sensor. A measured example of void fraction fluctuation is shown in Fig. 12. A momentary decrease in the void fraction was caused by the passing of a disturbance wave. Disturbance waves were detected in the fluctuations using the method proposed by Hazuku et al. (2008). In this method, whether or not a disturbance wave exists, the threshold value of the void fraction is defined as the average value of void fractions lower than the time-averaged value during a measurement period. In Fig. 12, shows that six disturbance waves were detected for 200 ms. The disturbance frequency was measured for 60 seconds from 60,000 data points.

The void fraction fluctuations of the annular flows with the same quality and mass flux of 200 and 400 kg/(m\(^2\)·s), respectively, are shown with the probability density functions of the void fraction fluctuations. Figure 13(a) shows that
FIG. 11: The measured void fractions for annular flow obtained with the predicted model compared to the results obtained with the Cioncolini and the Thome (2012) and drift–flux model with the Zuber et al. (1967) correlation (plotted symbols are the same as in Fig. 7).

for a mass flux of 200 kg/(m$^2 \cdot$ s) the disturbance wave frequency becomes higher when decreasing the diameter from 4.0 to 1.1 mm. However, for a diameter of 0.5 mm, the disturbance wave frequency becomes lower. Almost the same tendency was observed for the higher mass flux of 400 kg/(m$^2 \cdot$ s). Since the velocity of the disturbance wave would be higher than the liquid film velocity in annular flow, the increase in the disturbance wave frequency would lead to an increase in the liquid phase mean velocity. Therefore, the fact that the drift velocities for the 2.0 and 1.1 mm tubes were larger than those obtained by the Zuber model would indicate they were caused by increasing the disturbance wave frequency. On the other hand, for the 0.5 mm diameter tube, since the effect of surface tension on the liquid film flow behavior became relatively larger than the effect of gravity and inertial force, the occurrence of disturbance waves were prevented; therefore, the disturbance wave frequency became lower and the liquid film thickness became thicker. As the result, the drift velocity for the 0.5 mm tube became larger.
FIG. 12: Example of the void fraction fluctuation of an annular flow measured by the capacitance sensor.

FIG. 13: Comparison of liquid film structures of annular flow among tube diameters: (a) $G = 200 \text{ kg/}(\text{m}^2 \cdot \text{s})$, $x = 0.41$ to 0.46, (b) $G = 400 \text{ kg/}(\text{m}^2 \cdot \text{s})$, $x = 0.23$ to 0.28.

4. CONCLUSIONS

To clarify the transition boundaries in the dominant force regime, especially the effect of surface tension on gas–liquid two-phase flows, the void fraction characteristics of adiabatic vertical upward FC-72 one-component gas–liquid two-phase flows in small diameter tubes (with 4.0, 2.0, 1.1, and 0.5 mm inner diameters) were measured using the capacitance method. The obtained results can be summarized as follows.

The flow patterns were classified into bubbly, slug, churn, semi-annular, and annular flows for each tube diameter. The churn flows in which the gravity effect could be observed on the liquid flows were observed only for the 4.0 and 2.0 mm inner diameter tubes under low mass flux conditions. Under high-quality conditions, the transition boundaries...
agreed well with the results obtained using the Mishima and Ishii (1984) model, except for the 0.5 mm inner diameter tube. For the 0.5 mm tube, the transition boundary from slug to annular flow shifted to a higher \(j_G\) value than that obtained using the prediction model. In the flow pattern map using liquid and gas Weber numbers, only the transition boundary to the inertia-dominant zone of annular flow agreed well with that obtained by Akbar et al. (2003) for each tube diameter.

The homogeneous, Cioncolini and Thome (2012), and Awad and Muzychka (2014) models tended to overestimate the average void fraction for each tube diameter, especially under lower mass flux conditions of less than 100 kg/(m\(^2\)·s). The reason for this might be related to the increase in the surface tension effect. The average void fractions for churn and annular flows were also arranged based on the drift–flux model. The distribution parameters for churn and annular flows agreed well with the results obtained using the Ishii (1977) and Zuber et al. (1967) models, respectively. The drift velocity of the churn flow agreed well with the Ishii model for the 4.0 mm inner diameter tube; however, for churn flows in other smaller inner diameter tubes and annular flows, these previous models overestimated the drift velocities. In the 0.5 mm inner diameter tube the drift velocities were higher than in the other diameter tubes due to the decrease in the disturbance wave frequency as a result of the effect of the surface tension.

REFERENCES


