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HEAT TRANSFER OF TWO-PHASE IMPINGING JET - TWO-PHASE FLOW IN CAPILLARY NOZZLE TUBES

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ABSTRACT

To enhance the heat transfer coefficient and save the amount of coolant, two-phase jets have been applied to the cooling of heat transfer surface. The slugging in capillary nozzle tubes and the gushing behavior at outlet of tubes would strongly affect the impingement heat transfer. The slug (or churn) to annular flow transition dominated by the slugging mechanism was investigated in the previous experimental data in capillary nozzle tubes. Two-phase experiments were conducted with vertical tubes of 1, 2, 4, 5 and 10mm in inner diameter. The oscillatory behavior of bubble, slug and churn flow was observed. The peculiar phenomena in capillary nozzle tubes were not observed. The differential pressures between 500mm vertical span were measured. The root mean square (RMS) value of differential pressure was well correlated with Lockhart-Martinelli parameter. The RMS value was expected to be a key parameter related to the heat transfer enhancement with the two-phase impinging jet.

INTRODUCTION

Impingement jet cooling is often used to provide a high heat transfer coefficient. For example, the cooling of gas turbine blades is well known. Also the method is used to cool down the high temperature objects such as in the case of metalworking. As the impingement jet cooling is widely used in the modern engineering and technology fields, the improvement of efficiency is very important.

Numerous researchers performed the theoretical and experimental studies for heat transfer coefficient using a single-phase impinging jet. The cooling mechanism is well understood and several kinds of correlation for the heat transfer coefficient have been established for single-phase flow. The review by Martin[1] shows an empirical correlation for single-phase heat transfer on impinging jets from arrays of round smooth nozzles. To enhance the heat transfer coefficient and save the amount of water, the applicability of air/water two-phase jets was investigated [2]. When air was included in water jets, the mixture velocity of two-phase could be increased and the increase of turbulence could be expected.

The heat transfer of air/water (or solid/water) two-phase flow in pipes or ducts has been investigated in the mechanical or ship engineering fields. Recently, the skin friction reduction with the injection of air near the ship wall was proposed and discussed by numerous researchers. Kato et al.[3] showed that air micro bubbles increased the wall shear stress at the low volumetric ratio of air, but decreased it at the high volumetric ratio. Subramanian et al.[4] reported the enhancement of the heat transfer due to particle motion near the wall. That is caused by “film scraping” and “particle convection”. The effect of particles on the temperature distribution and heat transfer was also discussed in detail in the experimental study by Hetsroni[5].

To study the enhancement and degradation mechanism of impinging two-phase heat transfer, the air-water two-phase flow experiments were conducted with vertical tubes of 1 to 10mm in inner diameter. The oscillation of differential pressure dp in capillary nozzle tubes was very important for the better understanding of two-phase impinging heat transfer.

The differential pressure dp was measured at the long observation span more than fifty times of diameter. For the comprehensive understanding of oscillatory behavior, FFT analysis was conducted and spectrum peaks were obtained. Furthermore root mean square (RMS) value of dp was considered to be an important parameter to describe the impinging two-phase characteristics and was studied carefully in this study.

The prediction of flow patterns is the basis for modeling the two-phase flow and heat transfer. As the body forces largely control the flow behavior, the gravity is a key parameter in the flow patterns in relatively large pipe where the effect of viscosity and surface tension are considered to be negligible. However, the effect of viscosity and surface tension are considered to be important in capillary nozzle tubes. When gravity disappeared in previous annular transition models, the predicted flow pattern always becomes annular flow.

In microgravity condition, it is considered that the effect of viscosity and surface tension becomes significant due to the lack of buoyancy force and the peculiar phenomena on capillary nozzle tubes are emphasized because the capillary length of two-phase becomes large. As some useful similarity between the flow pattern transitions in microgravity and in microchannels is expected as pointed out by Akbar et al.[6], the effect of gravity is very important also for the correct understanding of viscosity or surface tension effects dominating the flow in capillary nozzle tubes.

NOMENCLATURE

c: frictional pressure loss coefficient
D: inner diameter of tube
g: acceleration due to gravity
h: length of tube
j: superficial velocity
dp: pressure difference
Re: Reynolds number
X: Lockhart-Martinelli parameter
x_q: quality
ρ: density
σ: surface tension

subscript
G: air, *L*: liquid, *0*: earth gravity condition

ANNULAR FLOW TRANSITION

Osakabe et al.[7] proposed the following annular transition model in the vertical pipes of various shapes and sizes. The model gave good prediction for channels whose hydraulic diameters were between 4 and 20mm.

$$j_G = 0.412\left[4D\left(\frac{\rho_L}{\rho_G}-1\right)g + 5\frac{\rho_L}{\rho_G}j_L^2\right]^{1/2} \quad (1)$$

where j: superficial velocity, D: hydraulic diameter, ρ: density, g: acceleration due to gravity. The suffix G and L indicate gas phase and liquid phase, respectively. The correlation was obtained by assuming the transition void fraction of 0.8 in the simple annular flow model using the Wallis[8] interfacial shear correlation. The transition void fraction of 0.8 had been empirically used to avoid the dryout of heating tubes in the design and operation of water tube boilers.

The non-dimensional form of Eq.(1) is

$$j_G^* = 0.412\left[4\frac{g}{g_0} + 5(j_L^*)^2\right]^{1/2}, \quad (2)$$

where the suffix 0 indicates earth gravity condition. The non-dimensional superficial velocity is defined as,

$$j_i^* = \frac{\rho_i^{1/2} j_i}{\left[g_0 D(\rho_L - \rho_G)\right]^{1/2}} \quad (i = L\ or\ G). \quad (3)$$

The transition gas velocity is expressed with the gravitational and shear stress terms.

Hewitt et al.[9] proposed the following empirical correlation for the annular transition. The transition gas velocity depends on the liquid velocity as,

$$j_G^* = 0.9 + 0.6 j_L^* \quad \text{at} \quad j_L^* < 1.5. \quad (4)$$

Taitel et al.[10] proposed the following annular transition model based on the lifting criterion of droplets. In the model, the annular transition takes place when the large droplets described as the Weber number of 30 can be lifted with gas phase stream. It was considered that the large droplets falling down in the vertical tube reasonably prevent the annular transition.

$$j_G = 3.1\left[\frac{g\sigma(\rho_L - \rho_G)}{\rho_G^2}\right]^{1/4} \quad (5)$$

The annular transition model by Ishii[11] was obtained with the condition for the disappearance of falling water on the wall normally observed in slug flow.

$$j_G = \left[\frac{1}{1.2 - 0.2(\rho_G/\rho_L)^{0.5}} - 0.1\right]\left[\frac{(\rho_L - \rho_G)gD}{\rho_G}\right]^{1/2} \quad (6)$$

In Eqs.(5) and(6), it should be noted that the transition gas velocity is expressed only with the gravitational term and indicates zero at microgravity.

Shown in Fig.1 is the comparison of annular transition models and experimental data in relatively small tubes at earth gravity condition. The steam/water experiments by Bergles et al.[12] were conducted in vertical tubes of 10.3 and 9.7 mm in diameter at pressure of 6.9 and 3.4MPa, respectively. Mishima et al.[13] and Sekoguchi[14] conducted air/water experiments at atmospheric pressure using small vertical tubes of 2.05, 4.08, 6 and 10 mm in inner diameter. Triplett et al.[15] used a small horizontal tube of 1.1 mm in inner diameter at atmospheric pressure. It should be noted that the annular transition boundary obtained in the small tube experiments are well predicted with the model without the surface tension.

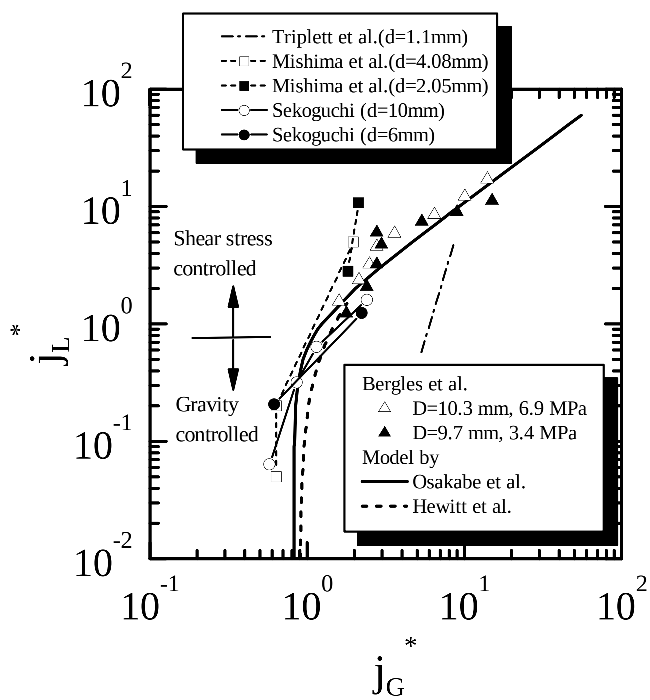

Fig. 1 Annular transition in earth gravity condition

The transition line by Osakabe et al.[7] can be divided into two regions, which are shear stress controlled region and gravity controlled region. As the gravitational term is dominant in the gravity controlled region, the transition gas velocity is approximately constant and agrees well with the empirical correlation by Hewitt et al.[9]. In the shear stress controlled region, the transition gas velocity increases with increase of liquid velocity and agrees well with the experimental data by Bergles et al.[12], Mishima et al.[13] and Sekoguchi[14].

The transition models were also compared with experimental results obtained at microgravity conditions during parabolic trajectory flights on aircraft. All the experiments were conducted at

$$\frac{g}{g_0} \leq 0.03 . \tag{7}$$

In the model applying for the microgravity condition, the gravitational term was approximately set to zero.

Shown in Fig.2 is the comparison of annular transition models and experimental data by Zhao & Rezkallah[16], Dukler et al.[17] and Ohta et al.[18] in the microgravity condition. The experiments were conducted at the atmospheric pressure in tubes of 9.525, 12.7 and 8 mm in diameter, respectively. The solid and dashed lines are annular transition lines by Osakabe's model[7] with g/g_0 =1 and 0, respectively. The experimental data with solid key is slug (slug-annular) flow and those with open key are annular flow in the microgravity condition. Due to the lack of gravity, the region of annular flow pattern tends to spread to the left of map indicating the gravity is one of factors to promote the liquid slugging. The annular transition boundary of experimental data generally agrees with the model using zero gravity.

Ohta's experiment[18] using R113 was conducted at the same j_G and j_L also in earth gravity condition. The observation showed the annular flow in earth gravity condition except the one data point indicated in Fig.2. At the point, the slug flow was observed in earth gravity though the annular flow was observed in microgravity. The transition line shifts to the left due to the disappearance of gravity and indicate the flow pattern change from slug to annular at the point, which supports the experimental observation.

EXPERIMENTAL APPARATUS AND METHOD

Shown in Fig.3 is a schematic of experimental apparatus. Air and water were supplied into the lower plenum and flow through test tubes of 1, 2, 4, 6 and 10mm in inner diameter. The test tubes were made of cupper or glass and the tolerance of the inner diameter was ±50μm. The measurement by a microscope showed that their accuracies were all in the range of the tolerance. By using the same tubes, the pressure losses of single-phase and two-phase flow were measured and good agreements with the conventional correlations were reported [19]. The two-phase flow from the test tubes entered to the upper plenum where the water level was kept with water drain tubes. The air was separated in the upper plenum and released to the atmosphere. The flow rates of air and water were measured with several sets of rotor-meters before entering the lower plenum. The differential pressures between 500mm vertical span were measured at sampling rate of 5 Hz during 60 s with a reluctance-type differential pressure cell. The measurement error was within ±0.13% and the time constant was 0.0025s.

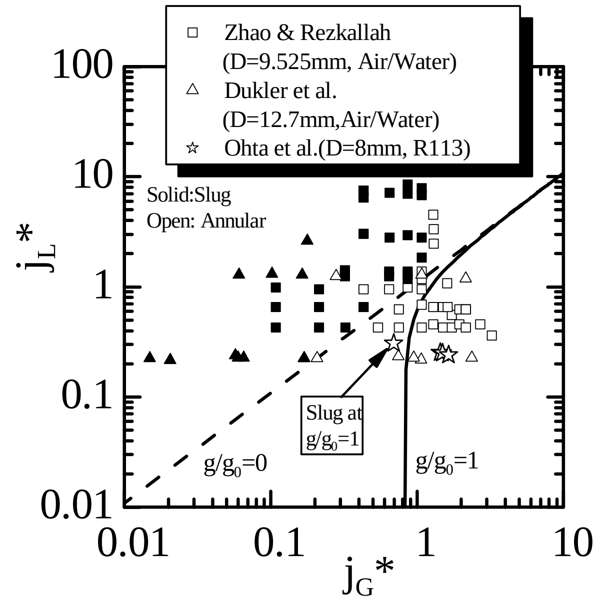

Fig. 2 Annular transition in microgravity condition

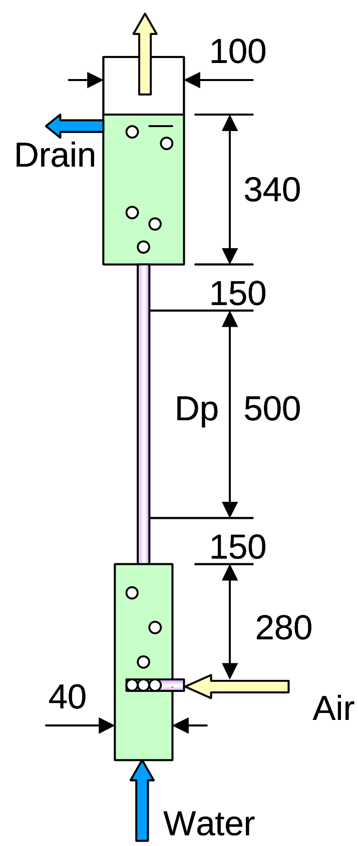

Fig. 3 Schematic of experimental apparatus

OBSERVATION OF FLOW PATTERN

Shown in Fig.4 is the experimental condition comparing with the annular transition model and experimental annular transition boundaries of air/water two-phase flow. The present experiments were conducted at non-annular flow region jugging from the previous experimental observations and the transition model. When the flow quality x_q is constant, the relation between the non-dimensional superficial gas and liquid velocities is given by

$$j_G^* = \frac{x_q}{1-x_q}\sqrt{\frac{\rho_L}{\rho_G}}\, j_L^* \tag{8}$$

The dashed doted lines in Fig.4 are the relation at x_q =0.0001 and 0.02 using the densities of air and water at atmospheric pressure. It can be confirmed that the almost data in the present experiment exist between the quality of 0.0001 and 0.02.

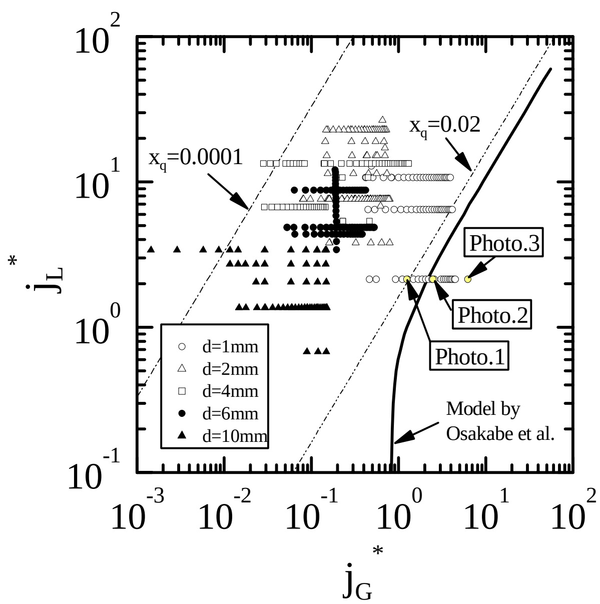

Fig. 4 Experimental condition on flow pattern map

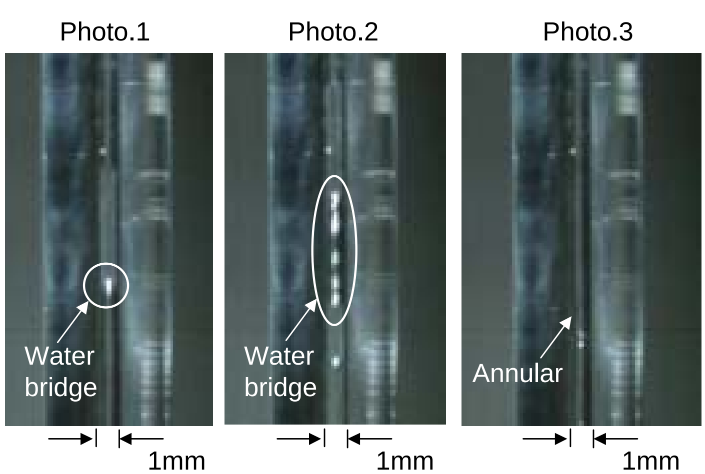

Fig. 5 Flow pattern in 1mm tube

Shown in Fig.5 are photographs in transparent tube of 1mm. These photographs are corresponding to the flow pattern map of Fig.4. The water bridges in the annular flow in Photo.1 and 2 could be frequently recognized when gas superficial velocity was relatively small. Increasing the gas velocity, the frequent water bridges disappeared as in Photo.3. In 1 mm tube, it was difficult to clearly define the annular flow transition but the appearance frequency of water bridges apparently decreased around at the transition line by Osakabe et al.'s model.

The oscillatory behavior of bubble, slug and churn flow was observed in the present experimental range. The peculiar phenomena in capillary nozzle tubes were not observed on the flow pattern in the present experimental range.

OSCILLATION BEHAVIOR

Shown in Fig.6 is a typical transient differential pressure measured in a pipe of 10 mm in diameter. The superficial water and air velocities were 1.06 and 0.12 m/s, respectively. From the differential pressure oscillation, important information such as the time-averaged (mean) value and the root mean square (RMS) value can be obtained. In the conventional two-phase flow system, the mean value of the differential pressure has been used to control the water mass content in tubes. The RMS value is considered to be a representative parameter to express the oscillation behavior.

Figure 7 shows power spectrums of oscillating differential pressure in the tube of 10 mm obtained by FFT method. The spectrums at j_G= 0.12 and 0.5 m/s were multiplied by 20 for the comparison in the same figure. When the gas velocity is low, the relatively large spectrum peaks exist at the low frequency range. However when the gas velocity is increased, the spectrum peaks become larger and tend to shift to the higher frequency region. It is possible to notice the gas velocity change if the oscillation behavior can be properly detected but the quantitative estimation is considered to be difficult.

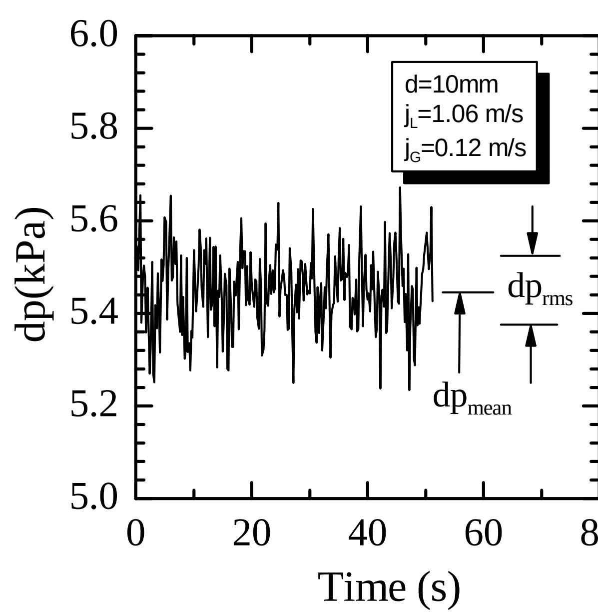

Fig. 6 Typical oscillation of pressure difference

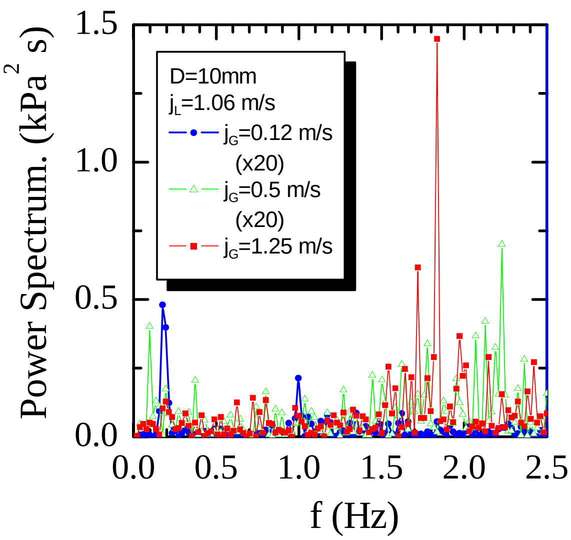

Fig. 7 Power spectrums of oscillating differential pressure in tube of 10 mm in inner diameter

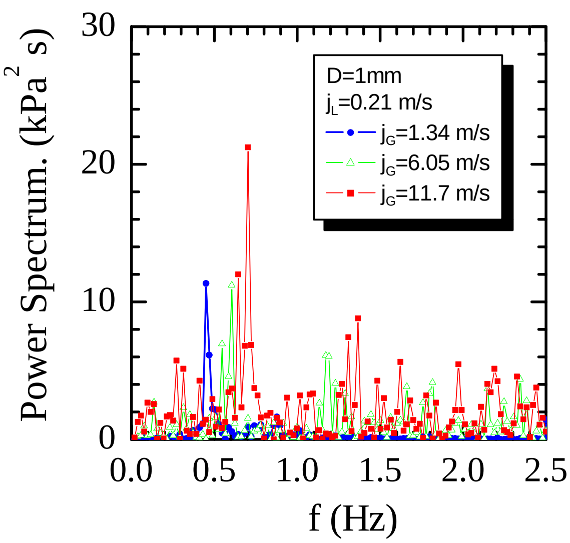

Fig. 8 Power spectrums of oscillating differential pressure in tube of 1 mm in inner diameter

The skilled engineer sometimes can determine the two-phase flow behavior in tubes by detecting the noise that is translated and amplified through an acoustic rod.

Figure 8 shows power spectrums of oscillating differential pressure in tubes of 1 mm. The spectrums at j_G= 1.34, 6.05 and 11.7 m/s were much higher than those in 10mm tube in Fig.7 due to the higher two-phase pressure loss. Generally the relatively large spectrum peaks exist at the low frequency range. When the gas velocity is increased, the several spectrum peaks appear at the high frequency region.

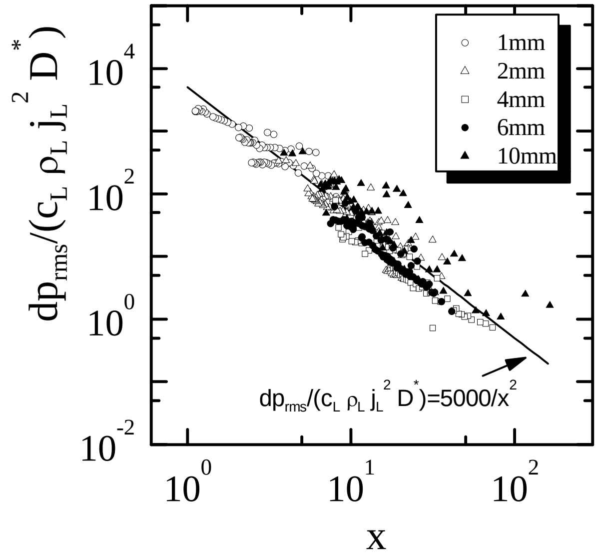

Fig. 9 Measured non-dimensional RMS value

Though the spectrum behavior is different from that in 10 mm tube, it is also possible to notice the gas velocity change if the oscillation behavior can be properly detected. Figures 7 and 8 shows the integration of spectrum with frequency increases with increasing the gas velocity at a given liquid velocity. These suggest the fluctuation energy is proportional to the amount of gas phase. The integration of spectrum with frequency can be represented by the root mean square (RMS) value of oscillation.

RMS VALUE OF DIFFERENTIAL PRESSURE

The oscillation of differential pressure can be attributed to the liquid flow fluctuation affected with the gas phase. It is considered that the fluctuation behavior strongly affected with the non-dimensional tube sizes defined as,

$$D^* = \frac{D}{\sqrt{\dfrac{\sigma}{g(\rho_L - \rho_G)}}} \tag{9}$$

Considering the oscillation of differential pressure is significantly affected with the liquid wall shear and no-dimensional tube size, the following non-dimensional RMS value was proposed.

$$\frac{dp_{rms}}{c_L \rho_L j_L^2 D^*} \tag{10}$$

where c_L is frictional pressure loss coefficient of liquid, ρ_L is liquid density, j_L is superficial liquid velocity, D^* is non-dimensional tube diameter. The frictional pressure loss coefficient c_L is expressed as,

$$c_L = 16/Re \quad \text{(laminar)} \tag{11}$$

$$c_L = 0.079\,Re^{-0.25} \quad \text{(turbulent)} \tag{12}$$

Shown in Fig.9 is relation of non-dimensional RMS value and Lockhart-Martinelli parameter X [20]. The non-dimensional RMS value obtained in the vertical tubes can be well correlated with the following correlation.

$$\frac{dp_{rms}}{c_L \rho_L j_L^2 D^*} = \frac{5000}{X^2} \tag{13}$$

The parameter X can be defined as,

$$X = \sqrt{\frac{dp_L}{dp_G}} \tag{14}$$

where dp_L and dp_G is the single-phase frictional pressure loss evaluated with superficial liquid and gas velocities, respectively.

Equation (13) indicates the RMS value is proportional to nozzle diameter at the turbulent condition when the water and air superficial velocity is given. If the heat transfer enhancement is strongly related to the oscillatory behavior in capillary nozzle tubes, the higher heat transfer with an addition of air can be expected in a larger diameter nozzle.

CONCLUSION

To study the enhancement and degradation mechanism of impinging two-phase heat transfer, the air-water two-phase flow experiments were conducted with vertical capillary tubes of 1 to 10mm in inner diameter. The followings are major results obtained in this study.

1. As the body forces largely control the flow behavior, the gravity is a key parameter in the flow patterns in relatively large pipe where the effect of viscosity and surface tension are considered to be negligible. On the other hand, the effect of viscosity and surface tension are considered to be important in capillary tubes and in microgravity. However, the annular transition boundaries obtained both in the capillary tubes and in microgravity were well predicted with the model without viscosity and surface tension. The peculiar phenomena in capillary nozzle tubes were not observed on the flow pattern in the present experimental range.
2. The differential pressures between 500mm vertical span were measured at sampling rate of 5 Hz during 60 s with a reluctance-type differential pressure cell. The FFT analysis on the oscillatory differential pressure showed the spectrum peak shifts due to the change of flow. The root mean square (RMS) value of differential pressure was well correlated with Lockhart-Martinelli parameter. The RMS value was expected to be a key parameter for the heat transfer enhancement with the two-phase impinging jet. If the heat transfer enhancement is strongly related to the oscillatory behavior in capillary nozzle tubes, the higher heat transfer with an addition of air would be obtained in a larger diameter nozzle.

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