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HEAT TRANSFER OF TWO-PHASE IMPINGING JET - HEAT TRANSFER ENHANCEMENT

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ABSTRACT

To study the enhancement and degradation mechanism of impinging two-phase heat transfer, air/water two-phase jet was applied on the cooling of copper surface of 30 mm in diameter. The two-phase jet impinged vertically on the horizontal heat transfer surface from capillary nozzle holes of 2, 4 and 6 mm in inner diameter. The non-dimensional heat transfer coefficient (HTC) was defined as the experimental HTC divided with the predictive HTC where the superficial two-phase velocity j_G+j_L and the physical properties of water were used in the empirical HTC correlation for single-phase flow. The larger non-dimensional HTC and stagnation pressure fluctuation were obtained with the nozzle of larger diameter. The larger nozzle could provide the more significant enhancement of heat transfer and pressure fluctuation with an addition of air. It was considered that the enhancement of heat transfer was due to the stimulation of thermal boundary layer with an addition of air.

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].

The heat transfer experiment of impinging two-phase jet is scarce. To enhance the heat transfer coefficient and save the amount of water, an air/water two-phase jet was applied on the cooling of copper surface of 30mm in diameter. To study the enhancement and degradation mechanism of heat transfer, the flow oscillation in a capillary tube such as the nozzle tube and the oscillating pressure at the stagnation point of impinging two-phase jet was measured.

NOMENCLATURE

c: frictional pressure loss coefficient
D: inner diameter of nozzle
dP: pressure
f: nozzle flow area
g: acceleration due to gravity
H: distance between nozzle and surface
h: heat transfer coefficient
K: arrangement compensation function
j: superficial velocity
L: nozzle pitch
Nu: Nusselt number
Pr: Prandtl number
q: heat flux
Re: Reynolds number

X: Lockhart-Martinelli parameter
x_q: quality
λ: thermal conductivity
β: volumetric airflow rate

subscript

G: gas, *L:* liquid, *0*: earth gravity condition

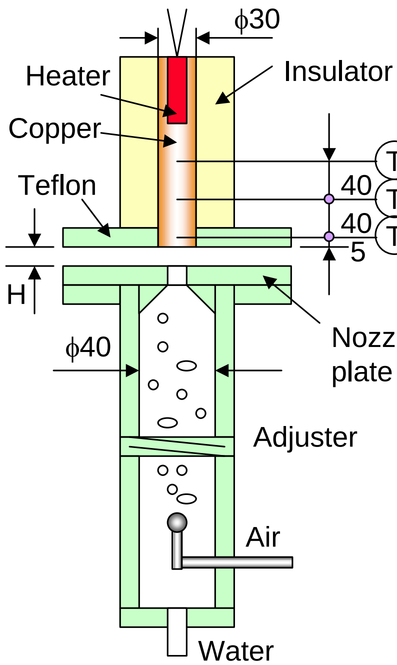

Fig. 1 Schematic of experimental apparatus

EXPERIMENTAL APPARATUS AND METHOD

Schematic diagram of experimental apparatus and the nozzle plate are illustrated in Figs. 1 and 2, respectively. The apparatus consisted of test section of heat transfer and air/water supplying system. Several nozzle plates with a single round hole at the center or seven round holes arranged in a hexagonal array of pitch 10mm as shown in Fig.2 were used in the experiments. For the single nozzle experiments, the nozzle diameters were 2, 4 and 6 mm. The nozzle diameter of 2 mm was used for the seven nozzles configuration.

The flow channel before the nozzle plate was a round pipe of 40mm in inner diameter and 495mm in length. A copper rod of 30mm in inner diameter and 200mm in length was used to provide the heat transfer surface, and to measure the surface heat flux and temperature. The purity of the copper was 99.99% to ensure a constant heat conductivity in the wide range of temperature. The thermal conductivity of the pure copper was 394 W/(mK). A cartridge heater inserted in the rod end provided a heat flow toward the heat transfer surface. To measure the inner temperature distribution, three T-type thermocouples of 1mm in diameter were embedded at locations of 5, 45 and 85mm from the surface. To decrease the heat loss, the insulator was rolled around the copper rod sufficiently. A Teflon plate was installed at the heat transfer surface to protect the insulator from the impinging water. The Teflon plate of 10mm in thickness had a hole of 30mm in inner diameter to protrude the copper rod.

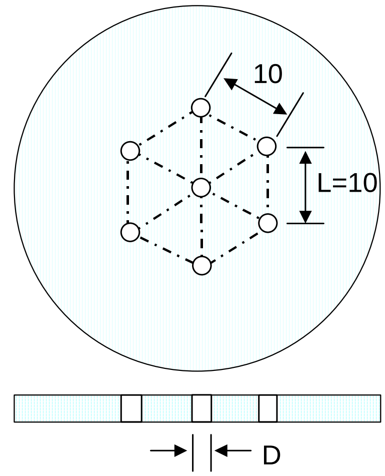

Fig. 2 Nozzle plate

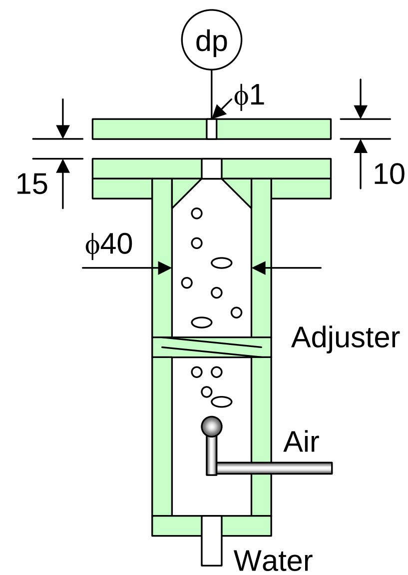

Fig. 3 Measurement of stagnation pressure

The water flow rate was measured with a turbine type flow-meter. The airflow rate was measured with a float type flow meter or an orifice flow meter.

In the present experiment, the distance between the nozzle and the heat transfer surface was fixed at 8mm for the seven nozzles array and 15mm for single nozzle. The mirror-finished heat transfer surface was wiped with absorbent cotton soaked with acetone before the each experiment. The experiments were conducted at atmospheric pressure, and different air and water flow rates. The surface temperature was maintained enough below the saturation temperature at the atmospheric pressure

The two-phase impinging flow behavior was experimentally investigated by measuring the stagnation pressure at the impinging surface as shown in Fig.3. The plate with a pressure measurement tap of 1mm in diameter was placed instead of the heat transfer surface above the nozzle. The pressure was measured at sampling rate of 5 Hz during 60 s with a reluctance-type pressure cell.

Shown in Fig.4 is the experimental condition comparing with the annular transition model by Osakabe et al.[6]. The annular transition boundaries obtained both in the capillary tubes and in microgravity were well predicted with the model without viscosity and surface tension[7]. The model is expressed as,

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

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

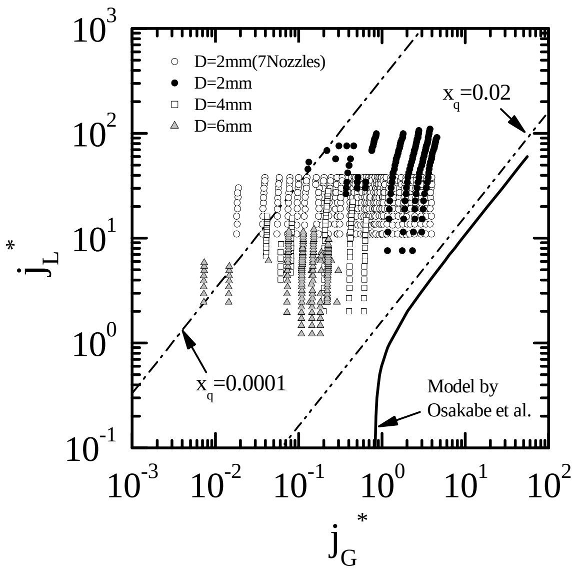

Fig. 4 Experimental conditions

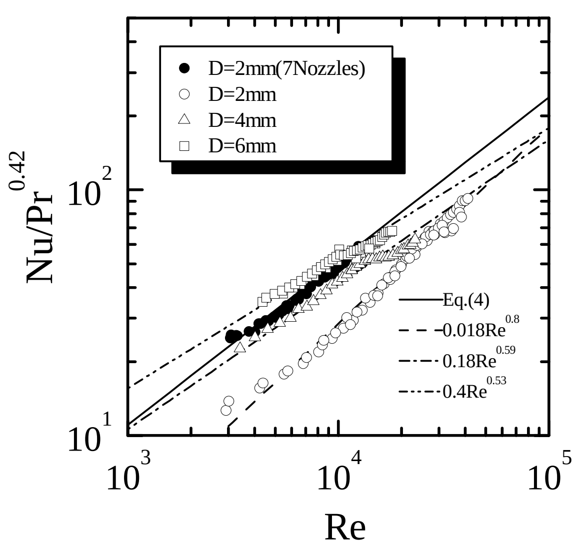

Fig. 5 Relation of Nu and Re in single-phase condition

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

The present experiments were conducted at non-annular flow region jugging from 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

$$j_G^* = \frac{x_q}{1-x_q}\sqrt{\frac{\rho_L}{\rho_G}}\, j_L^* \tag{3}$$

The dashed dotted 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.

EXPERIMENTAL RESULT AND DISCUSSION

Heat transfer experiments

Single-phase impinging jet experiments with the array of nozzles as same as the present experiment have been conducted and an empirical correlation was proposed. Martin[1] described the following empirical correlation in a review report.

$$\frac{Nu}{Pr^{0.42}} = K\sqrt{f}\,\frac{1-2.2\sqrt{f}}{1+0.2(H/D-6)\sqrt{f}}\,Re^{2/3} \tag{4}$$

where $f = \frac{\pi}{2\sqrt{3}}\left(\frac{D}{L}\right)^2$,

$$K = \left[1+\left(\frac{H/D}{0.6/\sqrt{f}}\right)^6\right]^{-0.05}.$$

This correlation is available at 2000 < *Re* < 100000, 0.004 < *f* < 0.04 and 2 < *H/D* <12, and applicable to the present experiment.

Shown in Fig.5 is the relation between the modified Nusselt number, $Nu/Pr^{0.42}$, and Reynolds number, *Re*, in the single-phase water jet impinging experiments. The solid line is prediction by the empirical correlation Eq.(4) for the 7-nozzles jet. The experimental data for the 7-nozzles jet agreed well with the prediction. Therefore, the present experimental apparatus and measurement system are reliable for the evaluation of the impinging heat transfer.

The heat transfer for the single nozzle is described with the following correlations as shown in Fig.5.

$$\frac{Nu}{Pr^{0.42}} = 0.018Re^{0.8} \quad \text{for D= 2 mm} \tag{5}$$

$$\frac{Nu}{Pr^{0.42}} = 0.18Re^{0.59} \quad \text{for D= 4 mm} \tag{6}$$

$$\frac{Nu}{Pr^{0.42}} = 0.4Re^{0.53} \quad \text{for D= 6 mm} \tag{7}$$

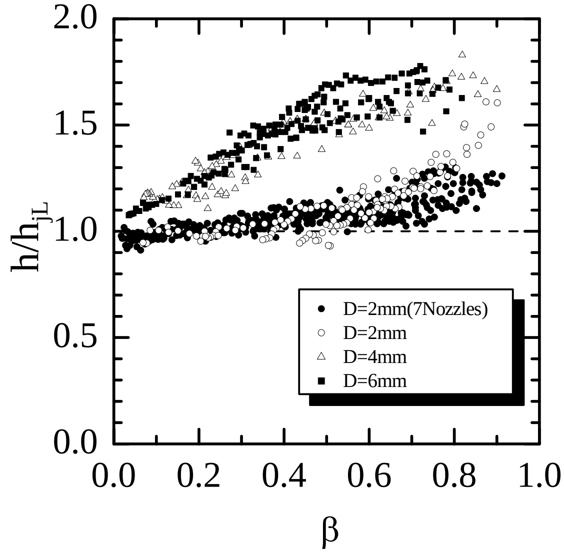

Fig. 6 Relation of heat transfer ratio and volumetric airflow ratio β in two-phase condition

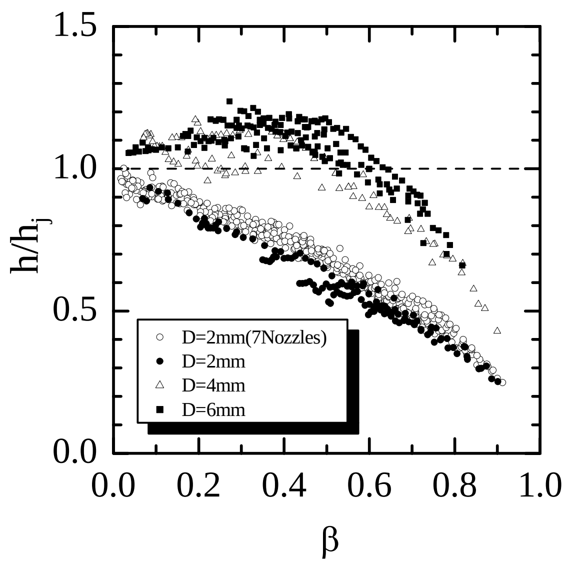

Fig. 7 Relation of heat transfer ratio and volumetric airflow ratio β in two-phase condition

To clarify the two-phase heat transfer, two kinds of heat transfer coefficients are defined. The heat transfer coefficient h_j is obtained with the single-phase correlation using the velocity j of two-phase mixture. The velocity j is defined as,

$$j = j_G + j_L \tag{8}$$

The heat transfer coefficient h_{jL} is obtained with the single-phase correlation using the superficial velocity j_L. The physical properties of water are used to obtain both the heat transfer coefficients. The heat transfer coefficient h_j is considered to be appropriate when the physical properties of two-phase mixture can be approximated with those of water, and the slip velocity between water and air is negligibly small. The heat transfer coefficient h_{jL} is considered to be the minimum value evaluated with the superficial water velocity. The actual heat transfer coefficient should be larger than h_{jL} due to an addition of air into the water jet.

The experimental data are divided by the heat transfer coefficient h_{jL} to confirm the increase of heat transfer due to an addition of air. Shown in Fig.6 is the relation between the heat transfer ratio h/h_{jL} and the volumetric airflow rate β. All the experimental data are larger than 1.0 indicating the increase of heat transfer with an addition of air. For the nozzle of 2 mm, approximately same the heat transfer ratios were obtained both for the single nozzle and the array of 7 nozzles. When the volumetric airflow rate β is less than 0.4, the heat transfer ratio is approximately 1.0 indicating the no increase of heat transfer with an addition of air for the single and multiple nozzles of 2 mm. The heat transfer ratio increases with increasing the diameter of nozzle. For the single nozzle of 4 and 6 mm, the two-phase heat transfer significantly increases with an addition of air as shown in Fig.6.

The experimental data are also divided by the heat transfer coefficient h_j. Shown in Fig.7 is the relation between the heat transfer ratio h/h_j and the volumetric airflow rate β. For the single and multiple nozzles of 2 mm, the heat transfer ratio is smaller than 1.0 in all the range of β indicating the smaller heat transfer than the prediction using the mixture velocity j. The degradation is mainly due to the lower thermal capability of gas-phase than that of liquid-phase.

However the heat transfer ratio is larger than 1.0 when β is smaller than 0.5 for the single nozzles of 4 and 6 mm. In these region, the higher two-phase heat transfer than that obtained by the water jet of the same velocity as the mixture velocity can be expected. The amount of water to cool the hot surfaces can be saved with an addition of air. The enhancement of heat transfer at β<0.5 is considered to be due to the stimulation of thermal boundary layer with an addition of air. The enhancement due to the micro bubbles sweeping on the surface is doubtful as the degradation for the nozzle of 2 mm takes place. The heat transfer ratio gradually decreased with the increase of gas phase atβ<0.5.

Measurement of pressure oscillation

To study the enhancement and degradation mechanism of heat transfer, the pressure at the stagnation point of impinging two-phase jet was measured at sampling rate of 5 Hz during 60 s with a reluctance-type pressure cell. Shown in Fig.8 is a typical transient differential pressure measured for the single nozzle of 2 mm. The superficial water and air velocities were 3.2 and 2.9 m/s, respectively. From the pressure oscillation, important information such as the time-averaged (mean) value and the root mean square (RMS) value can be obtained.

Shown in Fig.9 are mean and RMS values of stagnation point pressure for the single nozzle of 4 mm. The non-dimensional values divided by the dynamic pressure of the superficial liquid velocity are obtained. The mean stagnation pressure is approximately the same as the dynamic pressure of superficial water velocity at β<0.4.

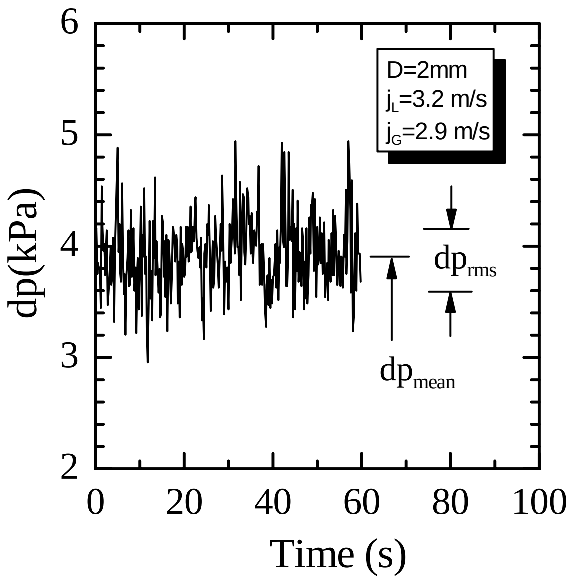

Fig. 8 Relation of heat transfer ratio and volumetric airflow rate β in two-phase condition

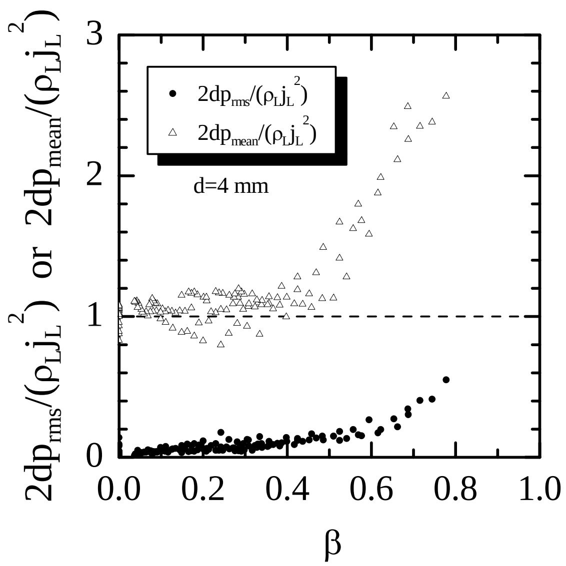

Fig. 9 Mean and RMS values of stagnation point pressure

As the actual water velocity increases with an addition of air, the gas phase are considered to play a roll to mitigate the pressure increase at the stagnation point. Further increasing the β, the stagnation pressure increases above the dynamic pressure. The RMS value of oscillating pressure gradually increases with increase of β indicating the stimulation of boundary layer with an addition of air.

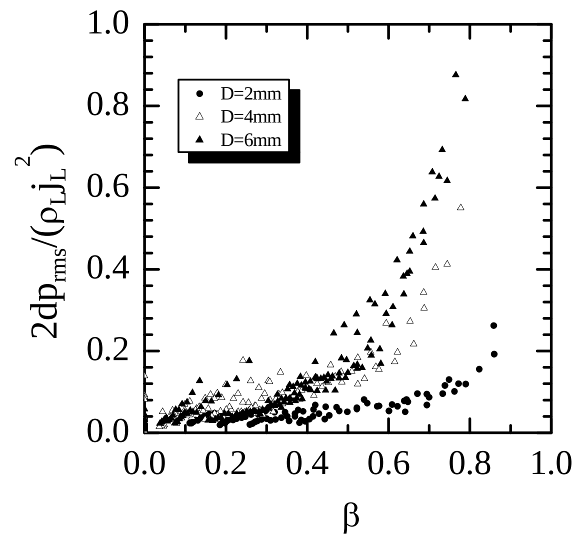

Fig. 10 RMS values of stagnation point pressure

Shown in Fig.10 are the RMS values of stagnation point pressure for the single nozzles of different diameters. The RMS values increase with increase of β and the larger values are obtained at the larger nozzle. As the heat transfer ratio in Fig.6 is also larger at the larger nozzle, the oscillation of impinging jet is strongly related to the heat transfer enhancement.

The non-dimensional RMS value of differential pressure obtained in the vertical capillary tubes could be well correlated with the following correlation [Horiki et al.(2004)].

$$\frac{dp_{rms}}{c_L \rho_L j_L^2 D^*} = \frac{5000}{X^2} \tag{9}$$

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 is c_L is expressed as,

$$c_L = 16/Re \quad \text{(laminar)} \tag{10}$$

$$c_L = 0.079\, Re^{-0.25} \quad \text{(turbulent)} \tag{11}$$

As the stagnation pressure fluctuation was strongly related to the differential pressure fluctuation in capillary nozzle tubes, the same non-dimensional parameter can be applicable. Shown in Fig.11 is relation between the non-dimensional RMS value and Lockhart-Martinelli parameter X [8]. The non-dimensional RMS value of stagnation point pressure can be well correlated with the following correlation similar to Eq.(9).

$$\frac{dp_{rms}}{c_L \rho_L j_L^2 D^*} = \frac{80}{X} \tag{12}$$

The parameter X can be defined as,

$$X = \sqrt{\frac{dp_L}{dp_G}} \tag{13}$$

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

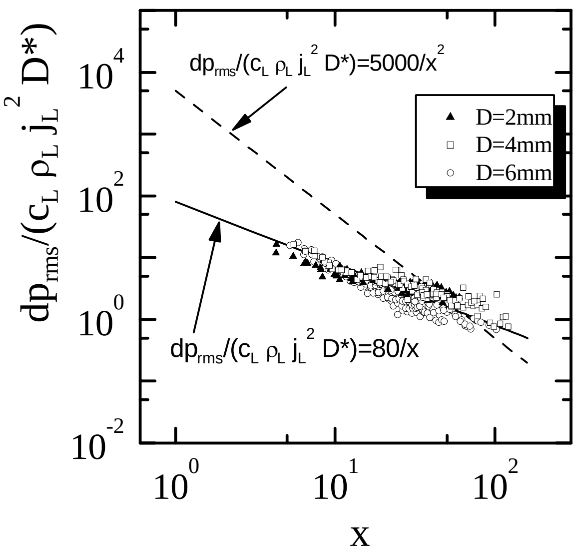

Fig. 11 Measured non-dimensional RMS value

Equation (12) indicates the RMS value is proportional to nozzle diameter at the turbulent condition when the water and air superficial velocity is given. The higher enhancement of heat transfer with an addition of air can be expected in a larger single nozzle than in array of small nozzle when the total flow area of nozzles are same. This enhancement mechanism is strongly related to the oscillatory behavior of impinging jet.

CONCLUSION

To enhance the heat transfer coefficient and save the amount of water, an air/water two-phase jet was applied to the cooling of copper surface of 30mm in diameter. The followings are major results obtained in this study.

1. The heat transfer coefficient (HTC) by single-phase water jets was well described with an empirical correlation. The experimental HTC increased with an addition of air at a given water flow rate. The non-dimensional HTC was defined as the experimental HTC divided with the predictive HTC where the superficial two-phase velocity j_G+j_L and the physical properties of water were used in the empirical HTC correlation for single-phase flow. The non-dimensional HTC monotonously decreased with increasing the volumetric airflow ratio β at the smallest nozzle of 2 mm. On the other hand, by using the nozzles of 4 and 6 mm, the non-dimensional HTC increased, took a maximum value larger than 1 and decreased with increase of β. The larger maximum of non-dimensional HTC was obtained with the nozzle of larger diameter.
2. The pressure at the stagnation point of impinging two-phase jet was measured at sampling rate of 5 Hz during 60 s with a reluctance-type pressure cell. The standard deviation of oscillating pressure increased with increase of β and the larger deviation was obtained at the larger nozzle. The stagnation pressure fluctuation was strongly related to the differential pressure fluctuation in capillary nozzle tubes. The larger nozzle could provide the more significant enhancement of heat transfer and pressure fluctuation with an addition of air. It was considered that the enhancement of heat transfer was due to the stimulation of thermal boundary layer with an addition of air.

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