COOLING CHARACTERISTICS OF TWO-PHASE IMPINGING JETS

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ABSTRACT

Cooling of surfaces is often carried out in devices consisting of arrays of round nozzles, through which coolant impinges vertically upon the surfaces. To enhance the heat transfer coefficient and save the amount of coolant, two-phase jets were applied to the cooling of copper surface of 30mm in diameter. Experiments were conducted with using water and air at atmospheric pressure.

The heat transfer coefficient by single-phase water jets was well described with an empirical correlation. The experimental heat transfer coefficient increased with addition of air. When the volumetric air flow ratio β is less than 0.2, the heat transfer coefficient by air/water two-phase jets was larger than the prediction where the volumetric two-phase velocity and the physical properties of water were used in the empirical correlation for single phase heat transfer. However, the experimental data gradually decreased with increase of gas phase at β>0.2.

To study the enhancement and degradation mechanism of impinging two-phase heat transfer, the flow pattern in a capillary tube such as the nozzle tube and the impinging behavior on a transparent glass plate were observed with a high-speed video camera. When β was small, numerous air micro bubbles of very small diameter were impinging on the heat transfer surface. When β was large, impinging micro bubbles could not be observed. Gas and water phases intermittently impinged on the glass plate as the heat transfer surface. In the observed photograph, a hexagonal pattern resulting from interference among the adjusting two-phase jets could be recognized.

The enhancement of heat transfer at β<0.2 is considered to be due to the micro bubbles sweeping on the surface. The degradation at β>0.2 is mainly due to the lower thermal capability of gas-phase than that of liquid-phase. The gas-phase intermittently impinged on the surface and its heat transfer was relatively low.

NOMENCLATURE

D: inner diameter of nozzle
f: nozzle flow area
H: distance between nozzle and surface
h: heat transfer coefficient
K: arrangement compensation function
j: superficial velocity
L: nozzle pitch
Nu: Nusselt number
P: pressure
Pr: Prandtl number
Q: volumetric flow rate
q: heat flux
Re: Reynolds number
T: temperature
x: distance from surface
λ: thermal conductivity
β: volumetric air flow rate ratio

subscript

f: fluid
G: gas
L: liquid
w: wall

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(1977) 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. 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.(1999) 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.(1973) 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(2001).

The heat transfer experiment of impinging two-phase jet is rares. 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 pattern in a capillary tube such as the nozzle tube and the impinging behavior on the transparent glass plate were also observed with a high-speed video camera.

EXPERIMENTAL APPARATUS AND METHOD

Schematic diagram of experimental apparatus and the array of round nozzles are illustrated in Figs. 1 and 2, respectively. The apparatus consisted of test section of heat transfer and air/water supplying system. Seven round nozzles were arranged in a hexagonal array as shown in Fig.2. Its inner diameter was 2mm and the pitch was 10mm.

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, 40 and 80mm 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.

The water flow rate was measured with a turbine type flow-meter. The air flow 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. 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

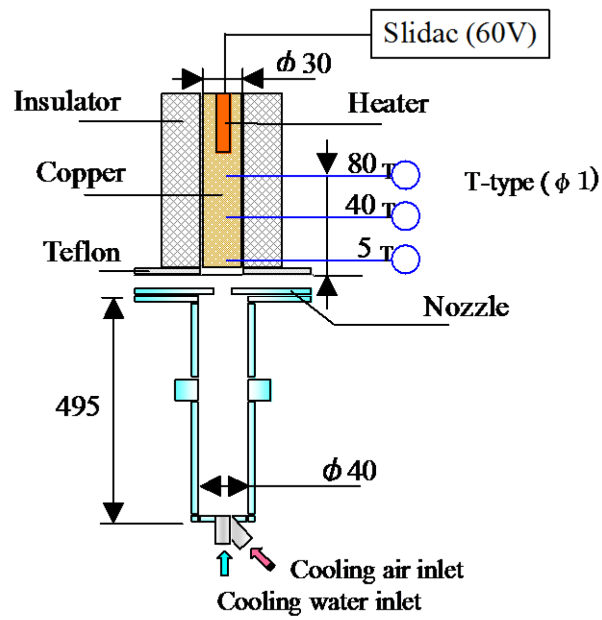

Fig. 1 Schematic of experimental apparatus

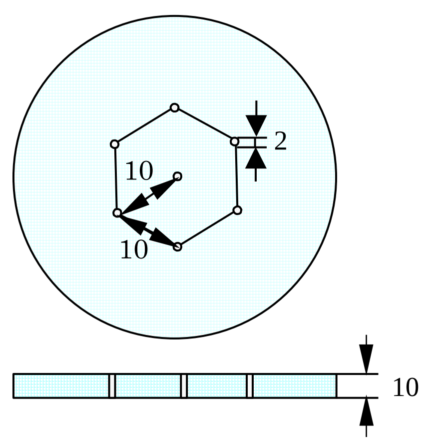

Fig. 2 Array of round nozzles

Shown in Fig.3 is the relation between the measured temperatures and the distance from copper surface. At the steady state the temperatures were measured with the thermocouples at the three locations in the copper rod. Then the linear temperature distribution was determined by least square method as,

$$T = T_W + \frac{dT}{dx}x \tag{1}$$

where T_w is the copper surface temperature, dT/dx is the temperature gradient in the copper rod, x is the distance from the surface. The intersection of the liner line and Y-axis is the copper surface temperature.

The surface heat flux can be calculated from Fourier's law with the temperature gradient and the copper thermal conductivity.

$$q = \lambda_c \frac{dT}{dx} \tag{2}$$

The copper thermal conductivity λ_c is 394 W/(mK). The heat transfer coefficient h is defined as,

$$h = \frac{q}{T_W - T_f} \tag{3}$$

where T_f is the impinging water temperature.

Single-phase impinging jet experiments with the array of nozzles as same as the present experiment have been conducted and an empirical correlation is established. Martin(1977) 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 $f = \frac{\pi}{2\sqrt{3}}\left(\frac{D}{L}\right)^2$,

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

This correlation is available at 2000 < Re < 100000, 0.004 < f < 0.04 and 2 < H/D <12.

The present experiments were conducted at 3000 < Re < 12500 for single-phase water jet experiments. The two-phase air/water jet experiments were conducted at the superficial water velocity of 0.7 < j_L(m/s) < 7.8 and the superficial air velocity of 0.06 < j_G(m/s) < 60. The volumetric air flow rate ratio β was 0.01 < β <0.98. In the two-phase jet, the Reynolds number was defined as,

$$Re = \frac{j_L D}{\nu_L} \tag{5}$$

where j_L is the superficial water velocity. The two-phase experiments were conducted at 2000 < Re_{jL} < 20000.

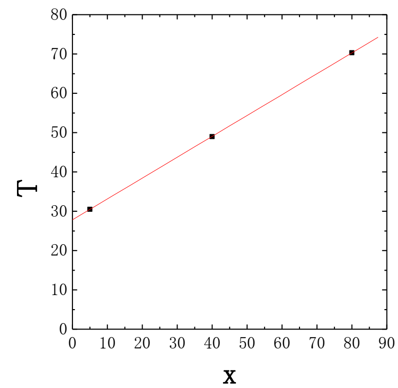

Fig. 3 Relation of surface temperature and distance from surface

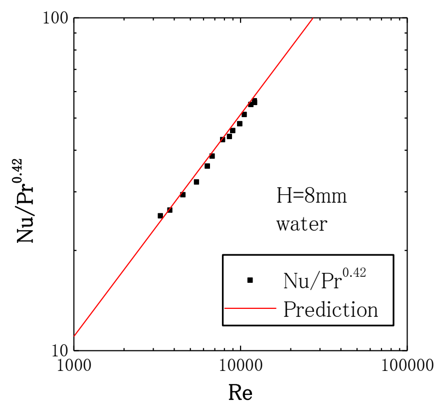

Fig. 4 Relation of Nu and Re in single-phase condition

EXPERIMENTAL RESULT AND DISCUSSION

Heat transfer experiments

Shown in Fig.4 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). The experimental data agreed well with the prediction. Therefore, present experimental apparatus and measurement system are reliable for the evaluation of the two-phase heat transfer.

Figure 5 is the relation of the modified Nusselt number and Reynolds number in the two-phase air/water jet experiments. The axis and abscissas are the same as those of Fig.4. The solid line is the prediction where the superficial water velocity and the physical properties of water are used in the empirical correlation Eq.(4) for single phase heat transfer. The experimental data was generally larger than the prediction due to the injection of air. The volumetric air flow rate ratio β is defined as,

$$\beta = \frac{j_G}{j_G + j_L} \tag{6}$$

The velocity of the two-phase mixture is

$$j = j_G + j_L \tag{7}$$

To clarify the two-phase heat transfer, two kinds of heat transfer coefficients are defined as,

$$h_j = K\sqrt{f}\,\frac{1-2.2\sqrt{f}}{1+0.2(H/D-6)\sqrt{f}}\left(\frac{jD}{\nu}\right)^{2/3} Pr^{0.42}\,\frac{\lambda}{D} \tag{8}$$

$$h_{jL} = K\sqrt{f}\,\frac{1-2.2\sqrt{f}}{1+0.2(H/D-6)\sqrt{f}}\left(\frac{j_L D}{\nu}\right)^{2/3} Pr^{0.42}\,\frac{\lambda}{D} \tag{9}$$

where the physical properties of water are used. The heat transfer coefficient h_j is the prediction using the velocity j of two-phase mixture. The heat transfer coefficient h_{jL} is the prediction using the superficial velocity j_L of water. 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 the addition of air into the water jet. The heat transfer coefficient ratio h_j/h_{jL} is

$$\frac{h_j}{h_{jL}} = \left(\frac{j}{j_L}\right)^{2/3} = \left(\frac{1}{1-\beta}\right)^{2/3} \tag{10}$$

The experimental data are divided by the heat transfer coefficient h_{jL} to confirm the increase of heat transfer due to the addition of air. Shown in Fig.6 is the relation between the non-dimensional heat transfer coefficient h/h_{jL} and the volumetric air flow rate ratio β. All the experimental datas are larger than 1.0 indicating the increase of heat transfer with an addition of air. The solid line is h_j/h_{jL} using the two-phase mixture velocity j. When the volumetric air flow rate ratio β is less than 0.2, the experimental heat transfer coefficient is larger than the heat transfer coefficient h_j using the mixture velocity j. On the contrary, the experimental heat transfer coefficient is smaller than the h_j at the larger β.

The experimental datas are divided by the heat transfer coefficient h_j. Shown in Fig.7 is the relation between the non-dimensional heat transfer coefficient h/h_j and the volumetric air flow rate ratio β. When the volumetric air flow rate ratio β is less than 0.2, the experimental heat transfer coefficient is larger than 1.0 indicating the larger heat transfer than the prediction using the mixture velocity j. The experimental data gradually decreases with an increase of β. The enhancement of heat transfer at β <0.2 is considered to be due to the micro bubbles sweeping on the surface. However it gradually decreased with the increase of gas phase at β >0.2. The degradation is mainly due to the lower thermal capability of gas-phase than that of liquid-phase.

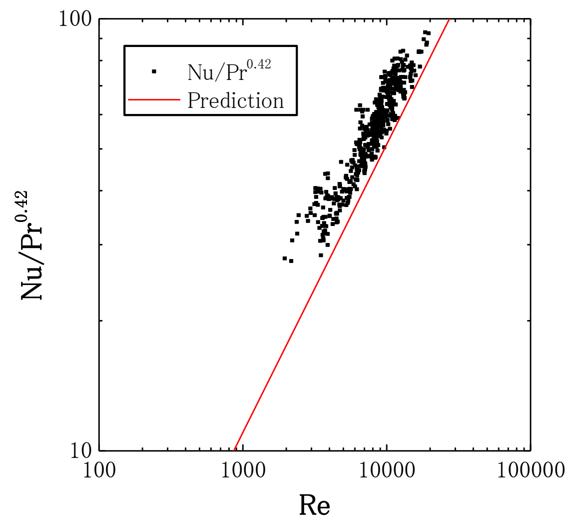

Fig. 5 Relation of Nu and Re in two-phase condition

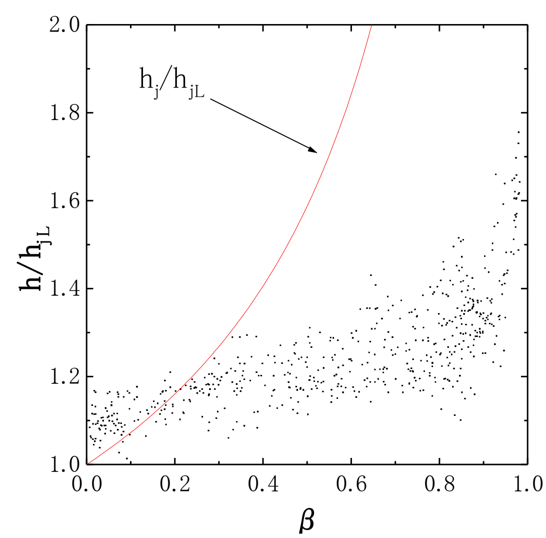

Fig. 6 Relation of heat transfer ratio and volumetric air flow ratio β in two-phase condition

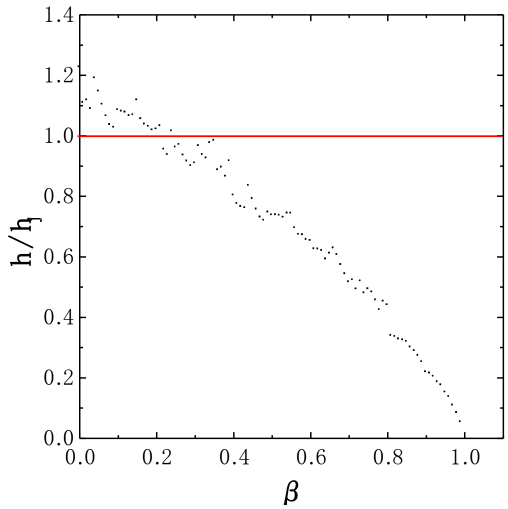

Fig. 7 Relation of heat transfer ratio and volumetric air flow ratio β in two-phase condition

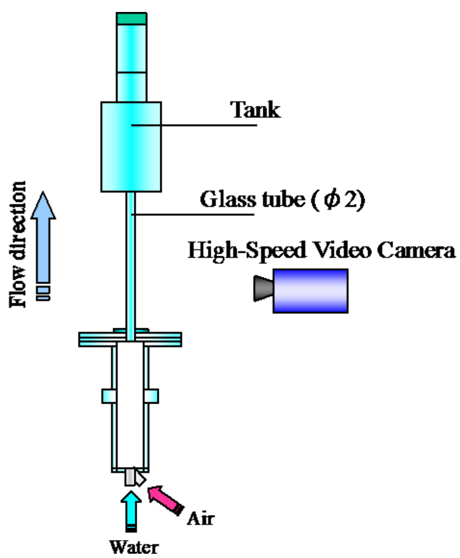

Fig. 8 Capillary tube experiment

Flow observation with high-speed video camera

To study the enhancement and degradation mechanism of heat transfer, the flow pattern in a capillary tube such as the nozzle tube and the impinging behavior on the heat transfer surface were observed with a high-speed video camera.

The experimental apparatus for the observation of flow in the capillary tube is shown in Fig.8. A glass tube of 2mm in inner diameter was used as the capillary tube. Instead of the nozzle plate, the flange of glass tube was installed in the apparatus. The inner diameter of glass tube was the same as that of nozzles. The flow pattern was observed with the high-speed video camera at the different water and air flow rates.

Shown in Fig.9 is a typical flow pattern of air/water two-phase flow in glass tube of 2mm in inner diameter. The previous study[Carey(1992)] indicated that the gas-phase only exist as plugs occupying almost the whole flow area in the capillary tube, leaving the wall at that location completely dry. But the photograph indicates that the air micro bubbles can exist in the water slug region in the capillary tube.

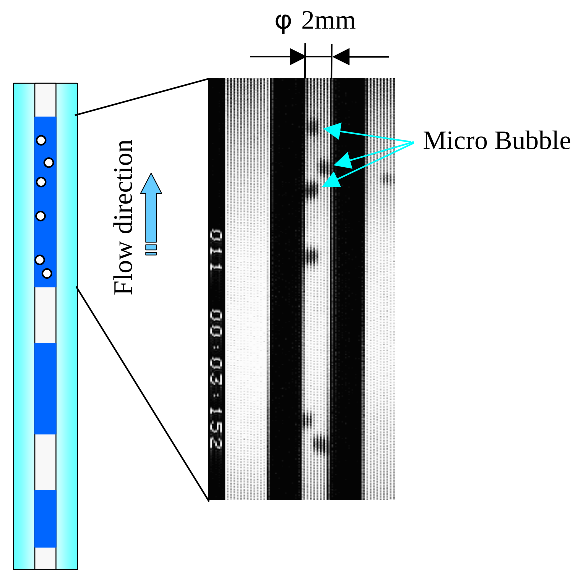

Fig. 9 Micro bubbles in capillary tube at β=0.19

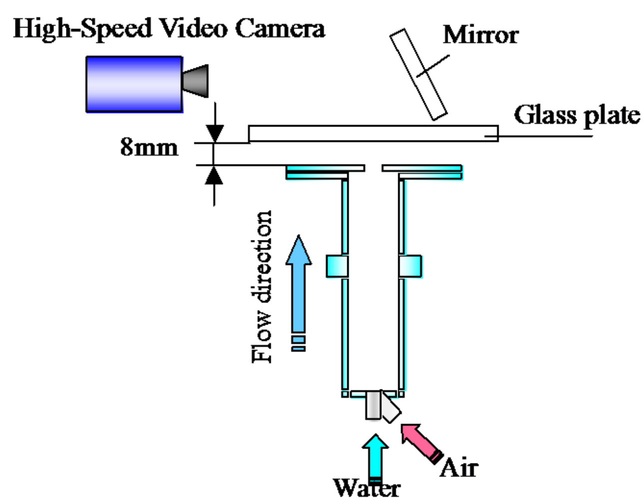

Fig. 10 Transparent wall experiment

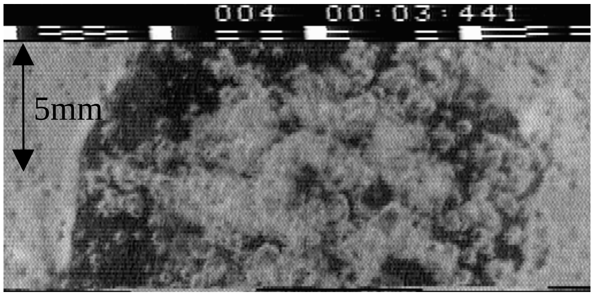

Fig. 11 Flow pattern at impinging surface at β=0.07

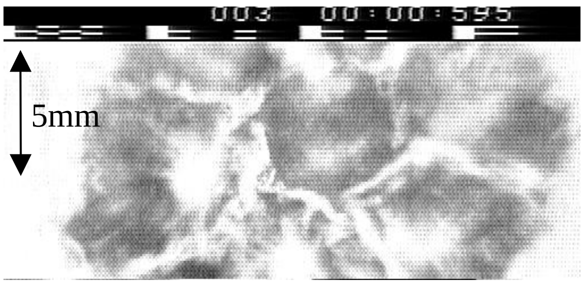

Fig. 12 Flow pattern at impinging surface at β=0.8

The experimental apparatus for the observation of the impinging behavior on the heat transfer surface is shown in Fig.10. Instead of the heat transfer surface, a glass plate was installed at 8mm from the nozzles. The impinging flow patterns were observed with a reflection by mirror through the transparent glass plate with a high-speed video camera.

Shown in Fig.11 is a photograph of impinging air/water two phase jets on the transparent glass plate at the volumetric air flow ratio β of 0.07. When β was small, numerous air bubbles of very small diameter were impinging on the glass plate. In the region, the experimental heat transfer was larger than the heat transfer coefficient h_j where the velocity j of two-phase mixture was used. The enhancement of heat transfer is considered to be due to the micro bubbles sweeping on the surface.

Shown in Fig.12 is a photograph of impinging air/water two phase jet on the transparent glass plate at the volumetric air flow ratio β of 0.8. When β was large, impinging micro bubbles could not be observed. Gas and water phases intermittently impinged on the glass plate. In the photograph, a hexagonal pattern resulting from interference among the adjusting jets can be recognized. The degradation of the heat transfer is mainly due to the lower thermal capability of gas-phase than that of liquid-phase.

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 main results obtained in this study.

1. The heat transfer coefficient by a single-phase water jet was well described with an empirical correlation. The experimental heat transfer coefficient increased with an addition of air.
2. The previous study indicated that the gas-phase only exists as plugs occupying almost the whole flow area in the capillary tube. But the photograph indicates that the air micro bubbles can exist in the water slug region in the capillary tube.
3. When the volumetric air flow rate ratio β was less than 0.2, the heat transfer coefficient by an air/water two-phase jet was greater than the prediction where the volumetric two-phase velocity j and the physical properties of water were used in the empirical correlation for single phase heat transfer. In the region, numerous air bubbles of very small diameter were impinging on the glass plate. The enhancement of heat transfer is considered to be due to the micro bubbles sweeping on the surface.
4. The heat transfer coefficient gradually decreased with the increase of gas phase at β>0.2. When β was large, impinging micro bubbles could not be observed. Gas and water phases intermittently impinged on the glass plate. In the observed photograph, a hexagonal pattern resulting from interference among the adjusting jets could be recognized. The degradation of the heat transfer is mainly due to the lower thermal capability of gas-phase than that of liquid-phase.

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