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Effects of the Width and Location of a Flow Disturbing Plate on Pool Boiling Heat Transfer on a Vertical Tube

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Abstract

Effects of the width and location of a flow disturbing circular plate, installed at a vertical tube surface, on nucleate pool boiling heat transfer of water at atmospheric pressure have been investigated experimentally. Through the tests, changes in the degree of intensity of liquid agitation have been analyzed. The plate changes the fluid flow around the tube as well as heat transfer coefficients on the tube surface. It is identified that the plate width changes the rate of the circulating flow whereas its location changes the growth of the active agitating flow. Moreover, the flow chugging was observed at the downside of the plate.

Key Words : pool boiling, flow disturbing plate, passive heat exchanger, vertical tube

1. Introduction

The mechanism of pool boiling heat transfer has been studied for a long time since it is closely related with the designs of more efficient heat exchangers. Recently, it has been widely investigated in nuclear power plants for application to the design of new passive heat removal systems employed in the advanced light water reactors [1,2]. One of the generally adopted geometries for the heat exchangers is tubular type and the orientation of the heat exchanging tubes of the heat exchangers is usually in vertical direction [2]. Although many researchers have investigated effects of the several design parameters on pool boiling heat transfer for the past several decades [3], researches for a vertical tube are relatively small. Cornwell and Houston [4] suggested that nucleate boiling on a tube differs considerably from that on a flat plate. The same is true for a wire. Therefore, results for a flat plate and a wire cannot apply to a tube without significant modification.

Jakob [5] recommended four empirical correlations for water boiling at standard atmospheric pressure under free convection condition. He set up the formulas for the horizontal and vertical heating surfaces in wide vessels. Stralen and Sluyter [6] performed a test to find out boiling curves for platinum wires in the horizontal and vertical orientations at atmospheric pressure. They reported that the major cause of the reduction in heat transfer for the vertical position is due to the formation of large vapor slugs. Nishikawa et al. [7] studied changes in the heat flux and the wall superheat (ΔT_{sat}) on a flat plate oriented at an inclination angle (θ) that varied between 0 and 175 deg from a horizontal, upward-facing position in the water. They reported that the difference in the effect of surface configuration over the whole region of nucleate boiling is presumed as changes in characteristics of bubble behavior and heat transfer mechanisms between low heat fluxes and high heat fluxes. One year later, Lienhard [8] explained the cause of the loss of orientation dependence using the Moissis-Rerenson Transition. Jung et al. [9] reported a study of nucleate and film boiling heat transfer in R-11 at 1 and 2 bar as a function of surface characterization and orientation. Fujita et al. [10] studied about the combined effects of the inclination angle and the gap size between two plates under the closed side periphery condition. Chang and You [11] investigated effects of the surface orientation from a micro-porous-enhanced square heater in FC-72. Some years later, Rainey and You [12] added effects of the heater size to Chang and You' s results.

Although many authors studied effects of the inclination angle on pool boiling heat transfer along with effects of geometry, pressure, and surface conditions, no detailed studies have been performed for tubes until Chun and Kang [3] studied effects of tube orientation on pool boiling heat transfer in combination with tube surface roughness and the diameter (D). Recently, Kang [13] carried out an experimental parametric study of a tubular heat exchanger to determine effects of the tube inclination angle on pool boiling heat transfer. Kang [14] also performed some more detailed study for the inclination angle with combining the surface roughness into the analysis. Although effects of an inclination angle on pool boiling heat transfer have been studied for various heating surface geometries, the detailed analysis on the heat transfer mechanism itself is very limited. Cornwell and Houston [4] and Kang [13] have executed studies to analysis the mechanisms affecting heat transfer on a horizontal tube. They observed variations of the local heat transfer coefficient through the tube circumference and suggested mechanisms of sliding bubble [4] and bubble coalescence [13]. Kang [1,13] also analyzed some mechanisms affecting heat transfer on a vertical tube. Kang [1] suggested a new experimental correlation containing the length of the vertical tube length. Corletti and Hochreiter [2] published some experimental results for a very lengthy vertical tube. They investigated pool boiling heat transfer on the tube surface for application to the advanced light water reactor design.

Summarizing the previous researches on heat transfer mechanisms of pool boiling it can be concluded that the major changes in the heat transfer rate come from the bubble behavior. According to Kang [1,13] and Corletti and Hochreiter [2] boiling heat transfer on a vertical tube is dependent on the intensity of (1) liquid agitation, (2) bubble coalescence, and (3) the rapid convective flow around the heated tube surface. Liquid agitation enhances heat transfer whereas bubble coalescence and the rapid convective flow decrease heat transfer from the heated surface. Bubble coalescence on a vertical tube is usually observed at the area where the density of active nucleation sites is very high and the intensity of liquid agitation is very weak. To activate more nucleation sites, the heated surface should be manufactured rougher [14] or the value of the heat flux should be higher. To observe bubble coalescence at the lower heat flux, the geometry of the heated surface should be changed (e.g., annular condition) [15]. The rapid convective flow prevents the generation of fully developed bubbles on the tube surface and, hence, decreases the heat transfer rate at the upper region of the longer tubes installed vertically [2]. Therefore, to fully observe the mechanisms of bubble coalescence and/or rapid convective flow specially manufactured surfaces and/or higher heat fluxes are necessary. However, the intensity of liquid agitation can be observed through the full heat flux range and it is very important to explain the increase in heat transfer at the fixed wall superheat.

Therefore, the present study is aimed at the analysis of the intensity of liquid agitation around a vertical tube. For the tests, the normal flow of the bubbles around the heated tube will be disturbed by a flow disturbing plate. With changing the width and the location of the plate variations in the local and average heat transfer coefficients on the tube surface will be obtained. The investigations on the movement of bubbles along the tube surface and its spreading over the water surface will be focused in the study.

2. Experiments

A schematic view of the present experimental apparatus and test sections is shown in Fig. 1. The water storage tank (Fig. 1(a)) is made of stainless steel and has a rectangular cross section (950(1300 mm) and a height of 1400 mm. This tank has a glass view port (1000(1000 mm) which permits viewing of the tubes and photographing. The tank has a double container system. The sizes of the inner tank are 800×1000×1100 mm (depth(width(height). The bottom side of the inner tank is situated at 200 mm above the bottom of the outer tank. The inside tank has several flow holes (28 mm in diameter) to allow fluid inflow from the outer tank. To diminish effects of inflow from outside tank, holes are situated at 300 and 800 mm high from the bottom of the inside tank. Four auxiliary heaters (5 kW/heater) were installed at the space between the inside and the outside tank bottoms to boil the water and to maintain the saturated condition. To reduce heat loss to the

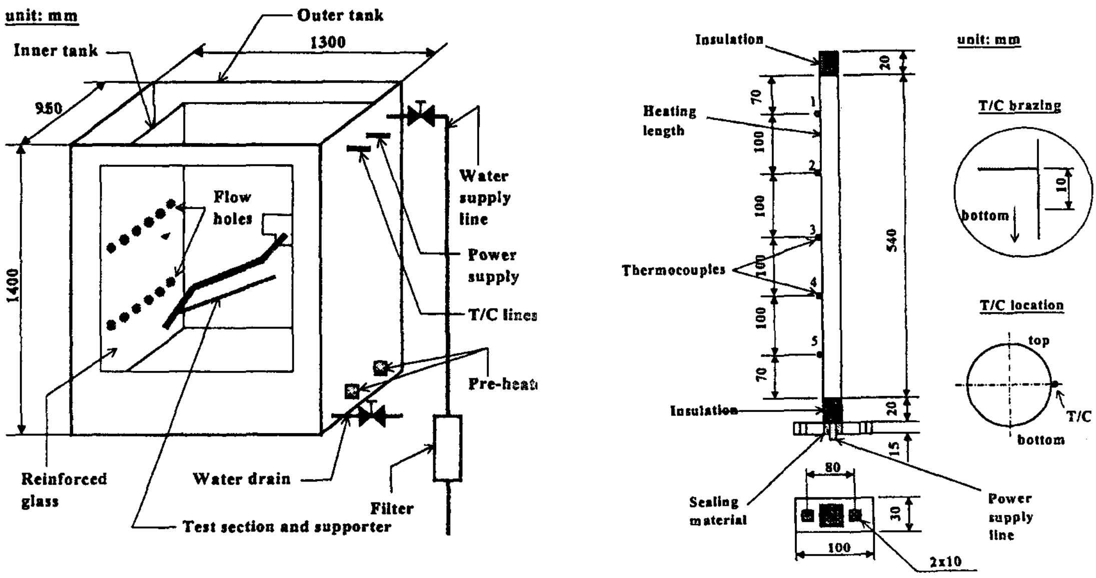

Fig. 1. Schematic Diagram of the Experimental Apparatus

environment, the left, right, and rear sides of the tank were insulated by glass wool of 50 mm thickness. The heat exchanger tubes are simulated by resistance heaters (Fig. 1(b)) made of a very smooth stainless steel tube (L=540 mm and D=19.1 mm). The surface of the tube was finished through buffing process to have smooth surface. Electric power was supplied through the bottom side of the tube. For the test, 220 V AC was used.

The tube outside was instrumented with five T-type sheathed thermocouples (diameter is 1.5 mm). The thermocouple tip (about 10 mm) was bent at a 90 degree angle and the bent tip brazed on the tube wall. The locations of the thermocouples are 70, 170, 270, 370, and 470 mm from the heated tube bottom as shown in Fig. 1(b). The water temperatures were measured with six sheathed T-type thermocouples placed vertically at a corner of the inside tank. All thermocouples were calibrated at a saturation value. To measure and/or control the supplied voltage and current two power supply systems (each having three channels for reading of both voltage and current in digital values) were used. The capacity of each channel is 10 kW.

To observe the intensity of liquid agitation and the flow spreading specially manufactured flow disturbing plates were prepared. The flow disturbing plate is divided into two parts and has an annular shape when fabricated as shown in Fig. 2. The outer diameters (D_o) of the fabricated three plates are 39.1, 59.1, and 79.1 mm, respectively. Therefore, the widths (W_p=(D_o − D)/2 of the plates are 10, 20, and 30 mm, respectively, since the outer diameter of the heated tube is 19.1 mm. For the smooth fabrication of a plate with the heated tube the inside diameter of the plate is manufactured slightly larger (about 0.1mm) than the tube outer diameter. To prevent sliding of the plate on the tube surface after fabrication, some margin was prepared at the joints. Therefore, the joint points can obtain tight contacts between the plate and the tube. To minimize any possible fin effects due to the attachment of the plate, every contact region was controlled to maintain a point-

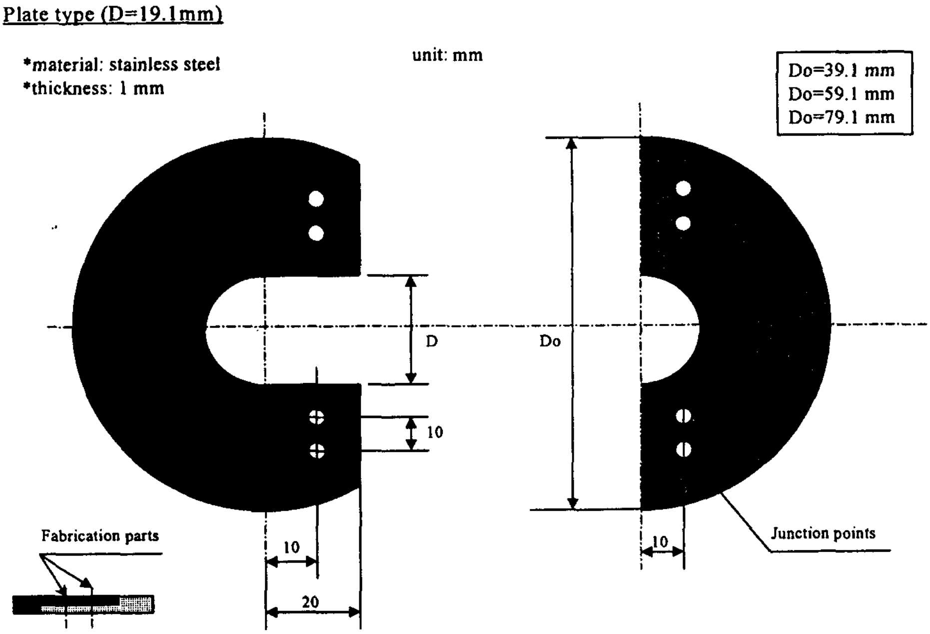

Fig. 2. Schematic of the Flow Disturbing Plate

type contact. The plates were made of stainless steel and have the thickness of 1 mm. The thickness of the fabricating regions is 0.5 mm to maintain the uniform plate thickness after fabrication. Several screws were used to assemble two parts of the plate into one.

For the tests, the heat exchanging tube is placed vertically at the tube supporter. After the water storage tank is filled with water until the initial water level is at 1200 mm from the outer tank bottom, the water is heated using four pre-heaters at constant power (5 kW/heater). Through the heating process, temperatures of the water were measured. When the water temperature reaches the saturation temperature, the water is then boiled for 30 minutes to remove the dissolved air. Thereafter, the supply of electricity to the heated tube started. The temperatures of the water and tube surfaces are measured when they are at steady state while controlling the heat flux on the tube surface with input power. In this manner a series of experiments has been performed for various combinations of the width and the location of the flow disturbing plate. Throughout the tests, auxiliary heaters were maintained on condition to certain the water to be saturated.

After a test for the given set of the width and the location is over, the switches to supply electricity to the heaters were turned off and, then, the water was drained. After changing the location or the width of the plate, a new test is started according to the above mentioned test procedure. Every experimental data were measured as the heat flux is decreasing since the heat transfer data show more stable as the heat flux decreases comparing to the corresponding values as the heat flux increases. According to Kang [16], no significant hysteresis is observed in the boiling tests of the combination of a smooth stainless steel tube and water like present tests. Therefore, the present results might be applied to the tube as the heat flux increases, too.

The heat flux from the electrically heated tube surface is calculated from the measured values of the input power as follows:

$$q'' = \frac{q}{A} = \frac{VI}{\pi DL} = h_b(T_W - T_{sat}) = h_b \Delta T_{sat} \quad (1)$$

where V and I are the supplied voltage (in volt) and current (in ampere), and D and L are the outside diameter and the length of the heated tube, respectively. T_W and T_{sat} represent the measured temperatures of the tube surface and the saturated water, respectively. The tube surface temperature T_w used in Eq. (1) is the arithmetic average value of the temperatures measured by five thermocouples brazed on the tube surface.

The error bounds of the voltage and current meters used for the test are 0.5% of the measured value. Therefore, the calculated heat flux is estimated to be 1.0%. The measured temperature has uncertainties originated from the thermocouple probe itself, thermocouple brazing, and translation of the measured electric signals to digital values. To evaluate the error bound of a thermocouple probe, three thermocouples brazed on the tube surface were submerged in an isothermal bath containing water. The measured temperatures were compared with the set temperature (80 ℃) of the isothermal bath of 0.01 K accuracy. According to the results, the deviation of the measured values from the set value is within ±0.1 K including the accuracy of the isothermal bath. Since the thermocouples were brazed on the tube surface, the conduction error through the brazing metal must be evaluated. The brazing metal is a kind of brass and the averaged brazing thickness is less than 0.1 mm. The maximum temperature decrease due to this brazing is estimated as 0.15 K. To estimate the total uncertainty of the measured temperatures the translation error of the data acquisition system must be included. The error bound of the system is ± 0.05 K. Therefore, the total uncertainty of the measured temperatures is defined by adding the above errors and its value is ± 0.3 K.

The uncertainty of the calculated heat transfer coefficients (i.e., $h_b = q''/\Delta T_{sat}$) depends on the wall superheat and the heat flux. The maximum possible error is at the highest heat flux (120 kW/m^2 for the case). Since the error bound of the heat flux at its highest value is calculated as ± 1.2 kW/m^2 and the uncertainty of the wall superheat is ± 0.6 K. The uncertainty of the superheat has been obtained by the equation of the wall superheat (i.e., ΔT_{sat} $T_W - T_{sat}$). Therefore, the calculated boiling heat transfer coefficient has its maximum error at q''=120 kW/m^2 and ΔT_{sat}=10 K. The value of h_b for the given values of and ΔT_{sat} is 12 kW/m^2-K and the deviation of it, (including errors of the heat flux and the wall superheat) is ± 0.5 kW/m^2-K. Therefore, the maximum possible uncertainty of the heat transfer coefficient can be estimated as $\pm 4.2\%$.

3. Results and Discussion

In order to investigate effects of the location of a flow disturbing plate on pool boiling heat transfer, a plate of 30 mm width was installed at the tube surface. Fig. 3 shows h_b versus ΔT_{sat} curves as the location of the flow disturbing plate changes from 160 mm to 460 mm. To clarify effects of the plate on heat transfer, the data were compared with the data of a single tube without the plate. As the plate is located at the tube bottom side, decreases in the heat flux and the heat transfer coefficient were observed at the same tube wall superheat. The slopes of h_b versus ΔT_{sat} curves of the single tube are changed by the presence of the plate. Therefore, it can be supposed that the presence of a flow disturbing plate and its location along the tube can change the mechanisms

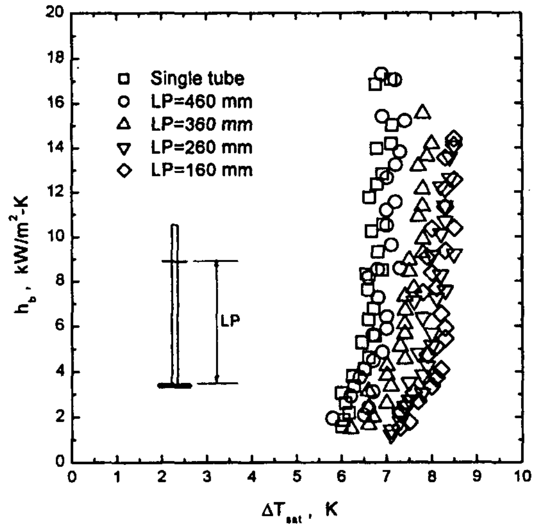

Fig. 3. Changes in Heat Transfer Due to Location of the Flow Disturbing Plate(Do=79.1mm)

affecting heat transfer. The major mechanism is liquid agitation due to the flow spreading and the active bubble movement. The bubbles generated at the tube bottom regions coalesce together with the other bubbles and activate environment liquid strongly while moving toward the water surface. This movement of bubbles changes heat transfer. In a vertical tube condition, enough distance is necessary for the flow of bubbles to generate active liquid agitation. If the flow of bubbles developed fully, strong swirling flow of bubbles were observed along the tube. The presence of the plate in front of the flow disturbs the flow of bubbles much before the flow gets fully developed. The dispersed flow has no enough momentum to affect the environment liquid. Therefore, the bubbles from the bottom of the plate spread over the plate circumference and, accordingly, the intensity of liquid agitation decreases much. The decrease in the heat transfer coefficient due to the plate can be observed as the location of the plate goes down to LP=360 mm. As ΔT_{sat} =7 K, the value of the heat transfer coefficient for the single

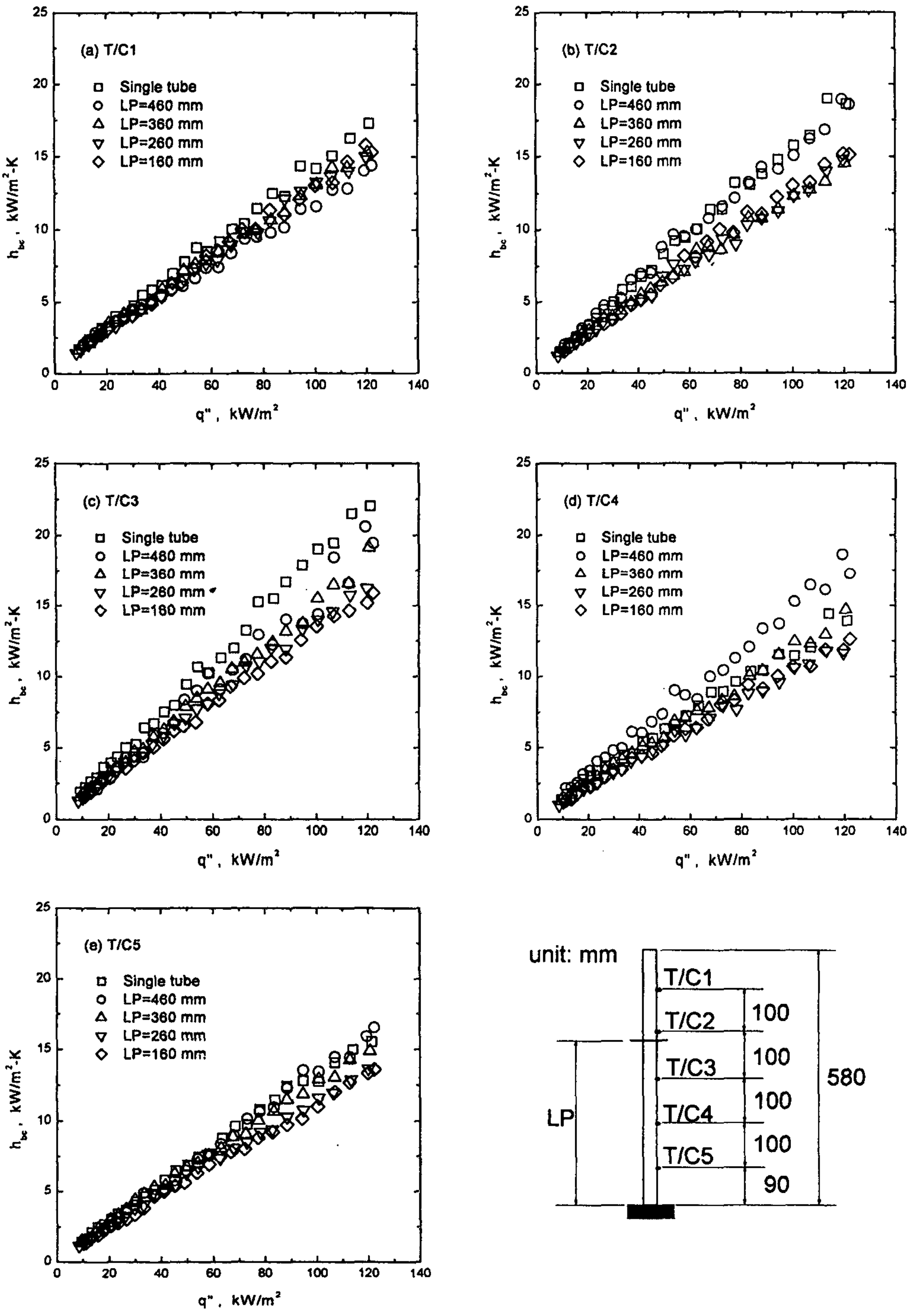

Fig. 4. Changes in Local Heat Transfer Coefficients due to Location of the Flow Disturbing Plate(Do=79.1mm)

tube is 17 times greater than the corresponding values of LP=160 or 260 mm. However, as the plate is situated at LP=460 mm the presence of the plate gives no significant change in the heat transfer coefficient comparing with the single tube without the plate. Therefore, it can be concluded that the minimum distance from the tube bottom is indispensable to generate effective liquid agitation around the heated tube.

Local heat transfer coefficients (h_{bc}) on the tube surface at different thermocouple locations were measured to clear up the difference in heat transfer coefficients due to the plate location, and the results are shown in Fig. 4. As shown in the figure some discrepancies exist between the trends of the averaged and the local heat transfer coefficients. Observing the experimental results for the single tube shown in Fig. 4(a), no symptom of effective bubble coalescence is identified on the tube surface. If active bubble coalescence exists on the tube surface, there should be sudden decrease in the slope of h_{bc} versus q'' curve [15]. Therefore, it can suggest that the major heat transfer mechanism of the present tests has no relevance with bubble coalescence on the tube surface. At T/C1, the local heat transfer coefficient of the single tube was compared to the corresponding value of the tube with the plate at LP=460 mm. The value for the tube with the plate is much less than the value for the single tube. There are about 20% difference between two heat transfer coefficients for the tubes with or without the plate at T/C1 and q''=120 kW/m^2. Since the rapid flow of bubbles creates strong liquid agitation, the slower flow results in weak liquid agitation. This explanation can be supported by the local heat transfer coefficients for the tubes with the plate at the location lower than LP=460 mm. As the plate is at lower than 460 mm, the local heat transfer coefficient increases at T/C1. As the plate is at the lower side of the tube the flow of bubbles around the upper side of the tube gets faster and results in stronger liquid agitation comparing to the tube with the plate at LP=460 mm. Comparing the local heat transfer coefficients for the tubes with the plate at 360, 260, and 160 mm, it is identified that there is no significant variation among them. Although the flow rate of bubbles at T/C1 for the tube with the plate at LP=160 mm is supposed to be faster than the corresponding value for the tube with the plate at LP=360 mm, no significant changes in local heat transfer coefficients were observed at T/C1. This suggests the importance of the fully developed flow to generate strong liquid agitation.

The averaged heat transfer coefficient, as the plate is at LP=460 mm, is larger than the heat transfer coefficients as the plate is at the lower side of the tube. The cause for the tendency should be observed at the other thermocouple locations. Observing the results at T/C2, the local heat transfer coefficient as the plate is at =460 mm is rapidly increasing as the heat flux increases. The cause is the flow chugging at the lower regime of the plate. If the flow of bubbles flowing upward along the tube surface encounters the disturbing plate, the flow rebounds to downward and this flow mixes actively with the upcoming flow. Then, escapes to the circumference of the plate. The serial process of the bubbles of impact, rebound, mixing to a large bubble slugs, and escape results in the flow chugging under the plate. Thereafter, very active liquid agitation is generated around this area like the pool boiling in an annular space as the tube bottom is closed [15]. The intensity of the flow chugging is magnified, as the flow rate of the bubbles gets bigger. Other important factors contributing to the intensity of liquid agitation is the flow circulation inside the tank. Two major clues observed in this study to explain the circulating flow are (1) the flow of bubbles departed from the upper most region of the tube and (2) the area of bubbles spreading over the water surface. Somewhat clear increases in local heat transfer coefficients of the single tube at T/C3 (Fig. 4(c)) and of the tube with the plate at LP=460 mm at T/C4 (Fig. 4(d)) are resulted from the circulating flow inside the water tank. The circulating flow can generate liquid agitation at the locations.

To identify the characteristics of the flow spreading and its effects on heat transfer coefficients, the width ($W_p=(D_o-D)/2$) of the plate has been changed as the plate is at the fixed location of LP=460 mm. Fig. 5 shows changes in the slopes of hb versus ΔT_{sat} curves as the values of the width are changing from 10 mm to 30 mm. According to the results, the heat transfer rate is biggest at the largest plate width (W_p=30 mm). This is caused by the changes in the degree of the flow spreading over the water surface. As the width of the plate gets wider, much stronger circulating flow would be resulted in. However, the effect of the flow from the tube lower side on the heat transfer gets increased as the width of the plate decreases. Therefore, the heat transfer

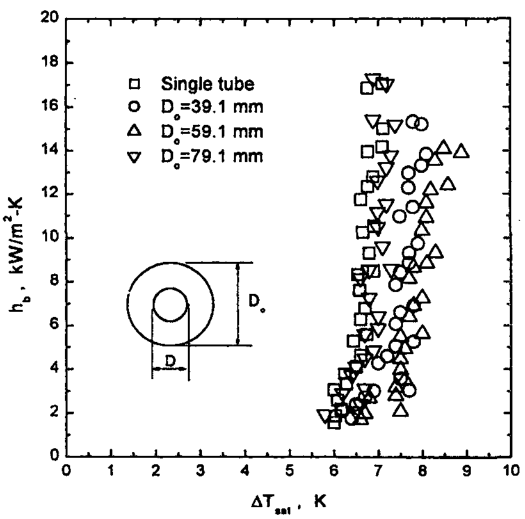

Fig. 5. Changes in Heat Transfer Due to Size of the Flow Disturbing Plate(LP=460mm)

coefficient decreases at its first stage as the plate width decreases. When the width of the plate is between 10 and 20 mm, the threshold of the increase and the decrease in the heat transfer coefficient exists.

Fig. 6 shows variations in local heat transfer coefficients at the thermocouple locations as the width of the plate changes. At T/C1, the single tube without the plate and the tube with the plate of 10 mm width (D_o=39.1 mm) have larger heat transfer coefficients comparing to the other two cases with wider plates. The cause is due to liquid agitation created by the faster flow of bubbles around this region. However, at T/C4 location the plate should be wider to have strong liquid agitation (see Fig. 6(d)). As the width of the plate gets narrower the rate of the circulating flow decreases due to the smaller spreading of bubbles at water surface. Comparing the three plate cases with the single tube at T/C4 location, the case of W_p=30 mm (D_o=39.1 mm) has the biggest heat transfer coefficient whereas the other three cases have the similar values. Therefore, it can be concluded that, the size of the plate should be wider to generate faster circulating flow at the tube lower side. Once a plate is located at LP=460 mm, it is difficult for the flow to recover the similar intensity of liquid agitation comparing to the cases where the plate is located at the lower side of the tube. Comparing the curves at T/C2 in Fig. 4 with in Fig. 6, the present local heat transfer coefficients are bigger than the local values of the tubes with the plate at lower regions of the tube. This tendency is resulted from the generation of the chugging flow under the plate. If the flow chugging occurs, active liquid agitation is generated at this region, and this results in heat transfer increase. Larger values in the slopes of h_{bc} versus q'' curves are observed at T/C3 for the single tube and at T/C4 for the tube with a plate of 30 mm width. However, this tendency is not

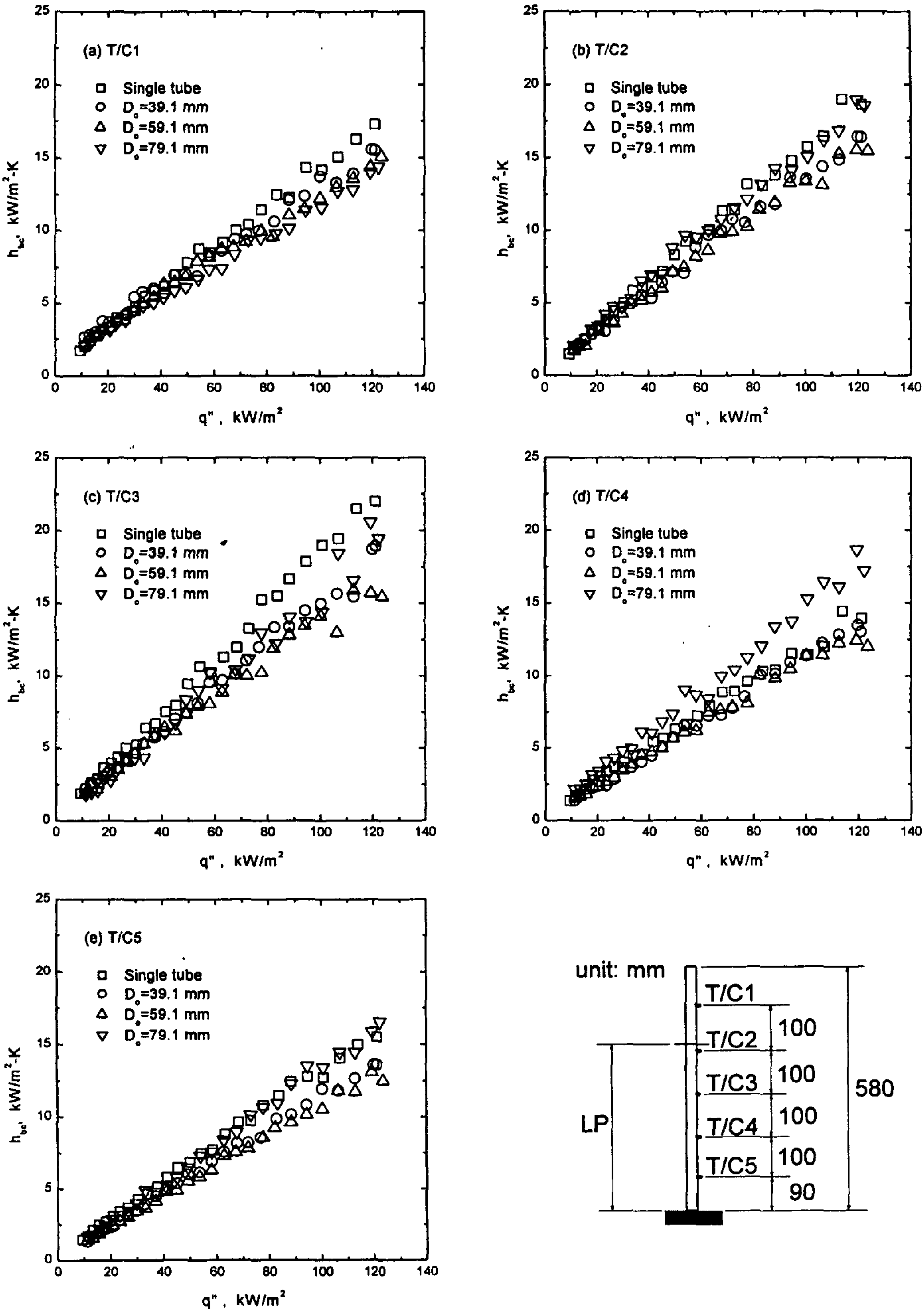

Fig. 6. Changes in Local Heat Transfer Coefficients due to Size of the Flow Distributing Plate(LP=460mm)

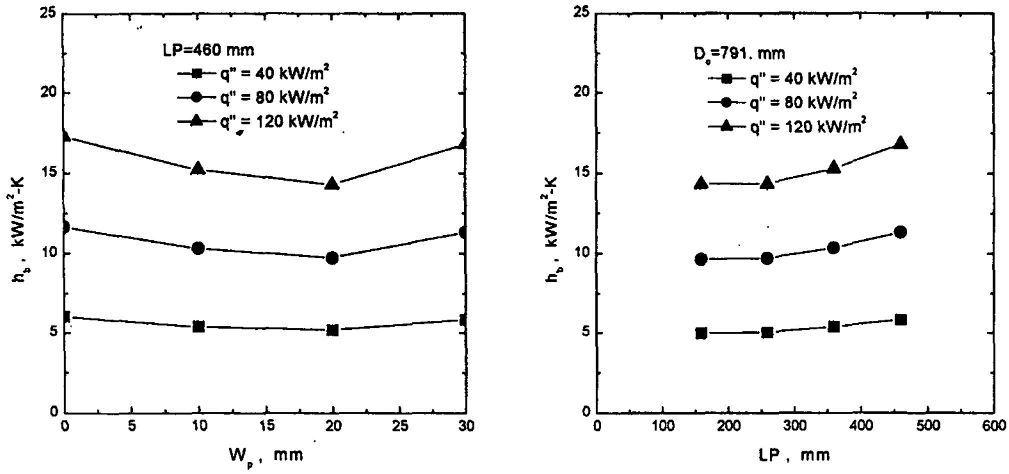

Fig. 7. Changes in Heat Transfer Coefficient as the Plate Width or Location Changes

observed, as the width of the plate is narrower (W_p=10 or 20 mm) than W_p=30 mm. Comparing the local heat transfer coefficients at T/C5 as shown in Fig. 6(e), the cases of the single tube and the tube with the wider plate have the values more than 30% bigger than the cases of the narrower width (10 and 20 mm). To magnify the intensity of liquid agitation (1) the flow should be concentrated along the tube like the single tube or (2) it should be spread over wider area like the tube with the plate of 30 mm width.

To clearly identify the changes of the heat transfer coefficient as the heat flux changes, the measured data of the heat transfer coefficients are correlated as a function of the heat flux by using the least square method. The heat transfer coefficients and the heat fluxes satisfy the linear equation form of $h_b=A+Bq''$ where A and B are constants depending on the plate conditions (width and location). The deviation of $h_{b,measured}/h_{b,calculated}$ from the exact value is within 10%. Fig. 7 shows the calculated curves of h_b versus W_p and h_b versus LP as a function of the heat fluxes. As the size of the plate width increases from 0 (single tube without the plate) to 20 mm the heat transfer coefficients decrease for the fixed plate location (LP=460 mm). After then, the heat transfer coefficient increases as the size of the plate width increases from 20 to 30 mm. This tendency is more clearly observed as the heat flux increases since much stronger liquid agitation is expected at the higher heat fluxes. Although the size of the plate is same, the location of it can change the heat transfer coefficient as explained above. The heat transfer coefficient increases as the plate location is situated at the upper side of the tube length for the fixed plate width (D_o=79.1 mm). Its effect is much more clearly observed as the heat flux increases.

Summarizing the above-mentioned mechanisms affecting the variations of heat transfer, those can be categorized as two groups of enhancing and decreasing heat transfer, respectively, as shown in Fig. 8. The mechanisms of enhancing heat transfer on the tube surface are as followings.

(1) Flow spreading: The increase in the area of the flow spreading at downstream of the plate results in faster circulating flow at the upstream regions of the plate. This generates active liquid agitation at the mid-regions of the tube.

(2) Flow chugging: This is observed just at upstream of the plate. As the heat flux

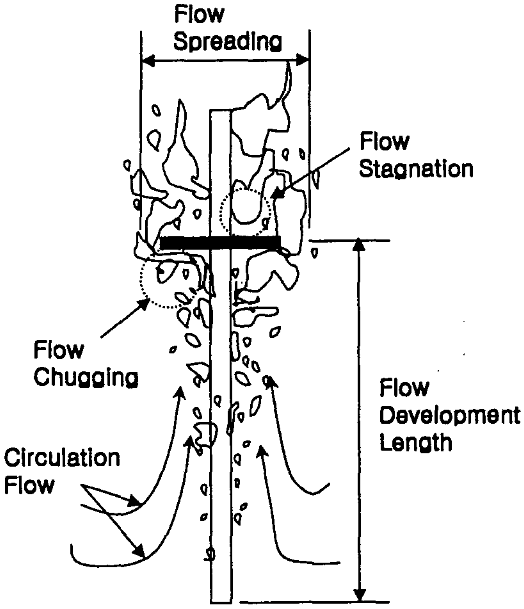

Fig. 8. Schematic View of the Mechanisms Related with Pool Boiling Heat Transfer

increases, many bubbles generated at the lower regions of the tube move upward and merge to the downside of the plate. Then, the bubble slugs come out to the plate circumference along the plate surface. If the size of bubble slugs increases, the impacted bubbles on the plate rebound to the downward. The rebounded bubbles are, then, mixing with the bubbles coming from the downside of the tube. This series of bubble movement results in the chugging flow just at downside of the plate. The generation of the chugging flow creates stronger liquid agitation at the downside of the plate.

(3) Flow development length: To get stronger liquid agitation at the upper regions of the tube, enough distance is needed for the flow of bubbles to be fully developed. As the flow rate gets faster, stronger liquid agitation at the upper regions of the tube as well as faster circulating flow at the lower regions of the tube can be observed.

The mechanism decreasing heat transfer is the flow stagnation. The flow rate of bubbles gets slower at downstream of the plate. Then, the retarded bubbles can be a cause of the flow stagnation. As the flow is stagnant, the intensity of liquid agitation decreases rapidly. For visual observation of the above-mentioned descriptions some photos of boiling on the tube surface are shown in Fig. 9 and 10 as the width or the location of the flow disturbing plate changes.

5. Conclusions

To identify effects of the size and location of a flow disturbing plate on pool boiling heat transfer of water at atmospheric pressure, three widths (W_p=10, 20, and 30 mm) and four locations (LP=160, 260, 360, and 460 mm) of the circular plate have been studied experimentally. In addition, the results were compared with those of the single tube without the plate and some empirical correlations were suggested. The major conclusions of the present study are as followings:

1. The size and/or location of the flow disturbing plate could change the value of the heat transfer coefficient. Moreover, the local heat transfer coefficient also can be changed by the adoption of the plate.
2. As the width of the plate is increasing from 0 to 20 mm the heat transfer coefficient decreases. After then, the heat flux increases as the width increases more than 20 mm. This is related with the intensity of the liquid agitation and flow spreading through the inside of the water tank.
3. The heat transfer coefficient increases as the location of the plate (W_p=30 mm) is changed from LP=160 mm to 460 mm for the given plate width of 30 mm. Therefore, it can be said that the development of the active liquid agitation is closely related with the flow

generated at the tube lower regimes.

4. As the plate is situated at =460 mm where the fully developed bubble flow is observed, the flow chugging and the flow stagnation are observed at the downside and upside of the plate, respectively. The chugging flow increases the

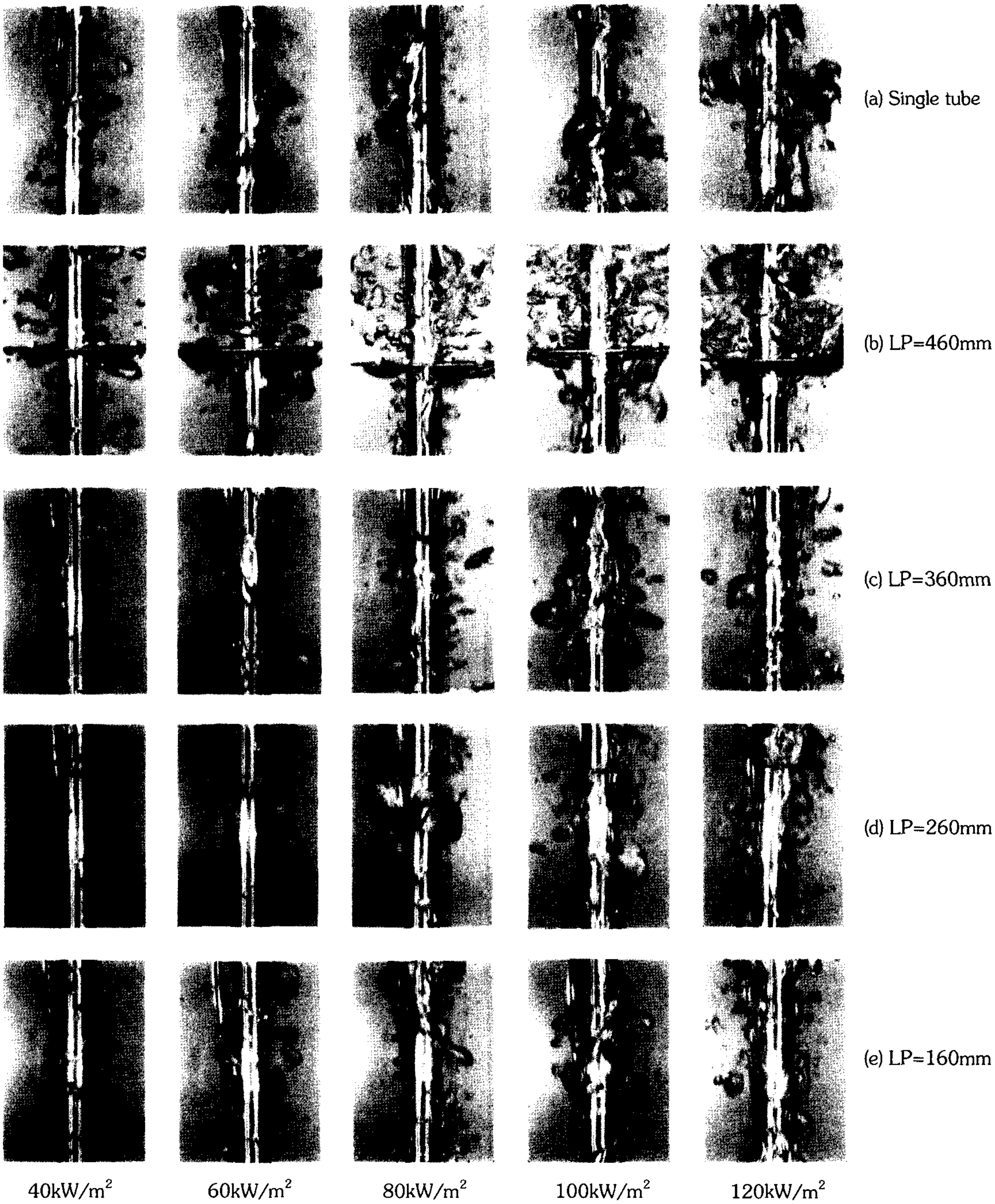

Fig. 9. Photos of Pool Boiling as the Location of the Flow Disturbing Plate (D0=79.1mm) Changes

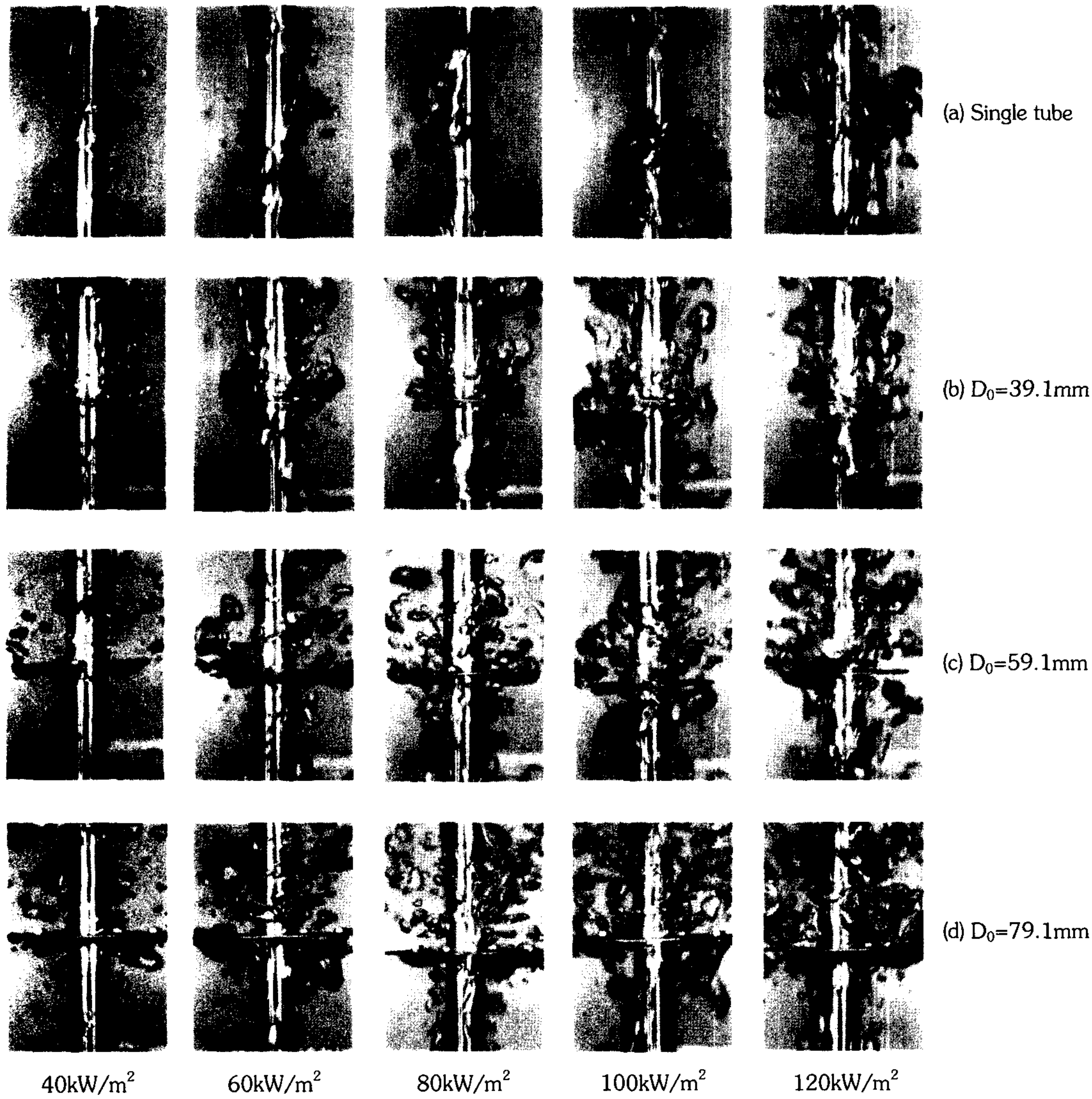

Fig. 10. Photos of Pool Boiling as the Size of the Flow Disturbing Plate Changes(LP=460mm)

heat transfer coefficient and the flow stagnation decreases heat transfer coefficient at the adjacent regions.

5. The measured heat transfer coefficients and heat fluxes can be correlated as a linear function of $h_b = A+Bq''$, and the empirical correlations can predict the experimental data within 10% error bound.

Nomenclature

A heat transfer area
D heating tube diameter
D_o diameter of the flow disturbing plate
h_b average boiling heat transfer coefficient
h_{bc} local boiling heat transfer coefficient
I supplied current
L tube length

LP height from the tube bottom
q input power
q'' heat flux
t time
T_W tube wall temperature
T_{sat} saturated water temperature
V supplied voltage
W_p width of the flow disturbing plate $(=(D_o-D)/2)$

Acknowledgments

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