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Effects of the location of side inflow holes on pool boiling heat transfer in a vertical annulus

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Abstract

To investigate effects of the location of side inflow holes on pool boiling heat transfer in a vertical annulus, the location of the holes has been changed from 0.1 to 0.4 m. For the test, a heated tube of 34 mm diameter and the water at atmospheric pressure have been used. To clarify effects of the location of the inflow holes on heat transfer results of the annulus are compared to the data of a single unrestricted tube and the annulus with closed bottoms. Adoption of the side inflow holes results in higher heat transfer than that of the single unrestricted tube as the heat flux is less than 100 kW/m^2. Moreover, it can remove the possibility of heat transfer deterioration at higher heat fluxes, which is a general characteristic of pool boiling in the annulus with closed bottoms. As the holes are located at 0.3 m from the bottom of the tube the most attractive results have been obtained and the enhancement in the heat transfer coefficient is 26.9% comparing to the single unrestricted tube at 160 kW/m^2.

1. Introduction

The mechanism of pool boiling heat transfer has been studied extensively in the past [1] since it is closely related with the thermal design of more efficient heat exchangers. Recently, it has been widely investigated in nuclear power plants for application to the design of new passive safety systems employed in the advanced light water reactors [2]. One of the effective ways to enhance heat transfer is considering a confined space around a heat exchanging part. Having higher heat transfer rate is very important if the space for the heat exchanger installation is very limited or rapid heat removal is essential.

Although many workers have in the past two generations investigated effects of heater geometries on boiling heat transfer, knowledge on the confined spaces on pool boiling heat transfer is still very limited. However, crevice effects in flow boiling have been widely studied [3–5]. Studies on the crevices can be divided into two categories. One of them is about annuli [6–9] and the other one is about plates [10–12]. In addition to the geometric conditions, flow to the crevices can be limited. Some geometry has a closed bottom [6,9,10]. Therefore, fluid should be supplied and be discharged through the open side only.

It is well known from the literature that the confined boiling is an effective technique to enhance heat transfer. It can result in heat transfer improvements up to 300–800% at low heat fluxes, as compared with unconfined boiling [6,10]. However, a deterioration of heat transfer appears at high heat fluxes for confined than for unrestricted boiling [3,10,11]. According to Kang [8], once the flow inlet at the tube bottom is closed, a very rapid increase in the heat transfer coefficient (h_b) is observed at low wall superheat (ΔT_{sat}) less than 2 °C. However, increasing ΔT_{sat} more than 2 °C the coefficient has almost the same value (i.e. about 20 kW/m^2 °C) regardless of the heat flux increase. The boiling heat transfer coefficient usually increases when gap size decreases at low heat fluxes whereas it decreases at higher heat fluxes. However, the heat transfer coefficient increases when gap size decreases to a certain value [6–8,11]. Further decrease in gap size results in sudden decrease in the heat transfer coefficient.

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Nomenclature

D	diameter of the heating tube, m	T_{sat}	saturation temperature, °C
h_b	boiling heat transfer coefficient, W/m^2 °C	T_W	tube wall temperature, °C
I	supplied current, A	V	supplied voltage, V
L	heated tube length, m	ΔT_{sat}	tube wall superheat ($=T_W - T_{sat}$), °C
L_h	location of the inflow holes, m		
q''	heat flux, W/m^2		

The major cause of the deterioration in an annulus was suggested as active bubble coalescence at the upper regions of the annulus [8]. Summarizing the previous works about crevice effects on pool boiling heat transfer it can be said that the amount of the heat transfer coefficient is highly dependent on the geometry and confinement condition.

Around the upper region of the annulus with closed bottoms the downward liquid interrupts the upward movement of the bubble slugs. Thereafter, bubbles are coalescing into much bigger bubbles while fluctuating up and down in the annular space. To apply the vertical annulus to the thermal design of a heat exchanger a solution to prevent the deterioration is needed in advance. Recently, Kang [9] published effective results of removing the deterioration point to a higher heat flux and preventing the creation of the critical heat flux. Kang [9] considered changes in the outer tube length of the annulus to decrease the possibility of the bigger bubble generation in the annular space. To remove the coalescence of the big size bubbles around the upper region of the annulus Kang [9] controlled the length of the outer tube of the annulus. The change of the outer length results in much variation in heat transfer coefficients. As the length of the outer tube is much shorter than the length of the heated tube the deterioration point of heat transfer gets moved up to the higher heat fluxes.

Since the major cause of the bigger bubble coalescence which results in the deterioration is partly because of the no inflow through the lower regions of the annulus, the present study is aimed at the investigation of improving heat transfer in the annulus through changing the location of the inflow holes along the height of the annulus. Upto the author's knowledge, no previous results concerning to this effect have been published yet.

2. Experiments

A schematic view of the present experimental apparatus and a test section is shown in Fig. 1. The water storage tank (Fig. 1a) is made of stainless steel and has a rectangular cross section (950 × 1300 mm) and a height of 1400 mm. The sizes of the inner tank are 800 × 1000 × 1100 mm (depth × width × height). Four auxiliary heaters (5 kW/heater) were installed at the space between the inside and outside tank bottoms. The heat exchanging tube is a resistance heater (Fig. 1b) made of a very smooth stainless steel tube ($L = 0.5$ m and $D = 34$ mm). The surface of the tube was finished through a buffing process to have a smooth surface. Electric power of 220 V AC was supplied through the bottom side of the tube.

The tube outside was instrumented with five T-type sheathed thermocouples (diameter is 1.5 mm). The thermocouple tip (about 10 mm) was brazed on the tube wall. The water temperatures were measured with six sheathed T-type thermocouples brazed on a stainless steel tube that placed vertically at a corner of the inside tank. All thermocouples were calibrated at a saturation value (100 °C since all tests were done at atmospheric pressure). To measure and/or control the supplied voltage and current, two power supply systems were used. The capacity of each channel is 10 kW.

For the tests, the heat exchanging tube is assembled vertically at the supporter (Fig. 1a) and an auxiliary supporter is used to fix a glass tube (Fig. 1b). To make the annular condition, glass tubes (gap size = 10.7 mm) of 55.4 mm inner diameter and 600 mm length were situated around the heated tube. To maintain the gap size between the heated tube and the glass tube a spacer made of a thin wire has been installed at the upper region of the test section. The inflow into the annular space was controlled by the location of the inflow holes changing from 0.1 to 0.4 m. The diameter of the inflow hole is 15 mm and the number of the holes is four, which are located at every 90° along the circumference of the glass tube.

After the water storage tank is filled with water until the initial water level is reached at 1100 mm, the water is then heated using four pre-heaters at constant power. When the water temperature is reached at a saturation value, the water is then boiled for 30 min to remove the dissolved air. The temperatures of the tube surfaces (T_W) are measured when they are at steady state while controlling the heat flux on the tube surface with input power.

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

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

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. Every temperature used

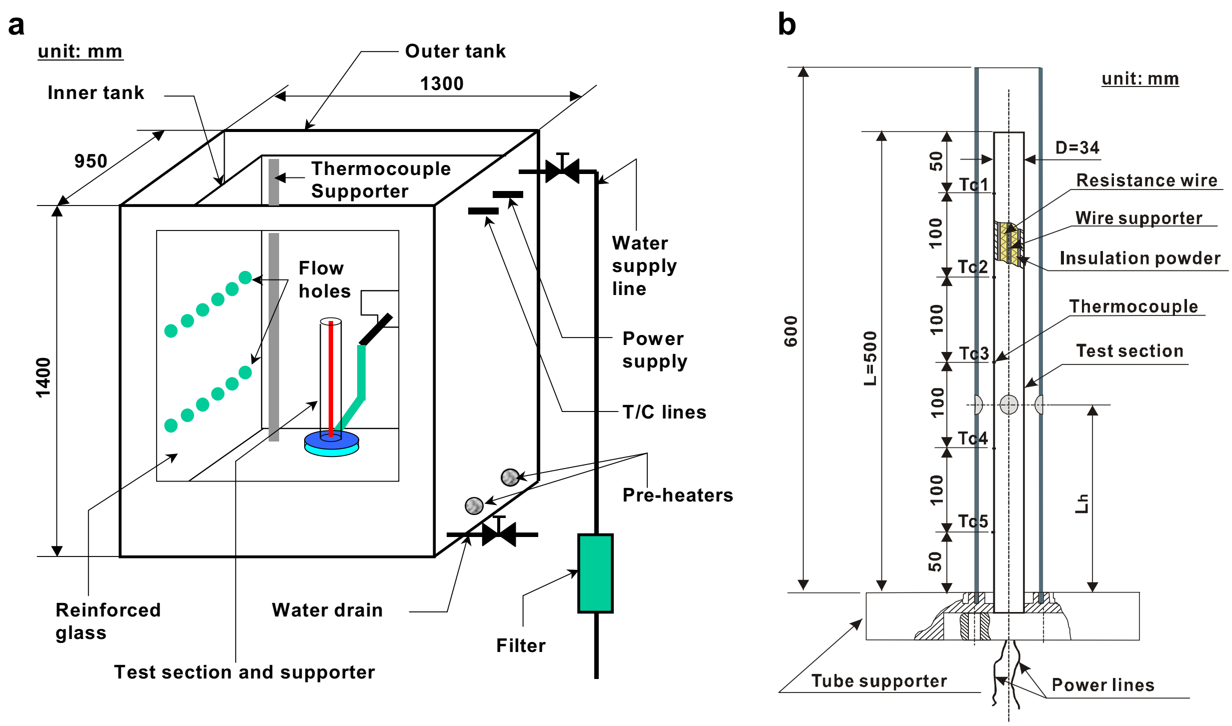

Fig. 1. Schematic of experimental apparatus and test section.

in Eq. (1) is the arithmetic average values of the temperatures measured by thermocouples.

The error bounds of the voltage and current meters used for the test are ±0.5% of the measured value. Therefore, the calculated power (voltage × current) has ±1.0% error bound. Since the heat flux has the same error bound as the power, the uncertainty in the heat flux is estimated to be ±1.0%. When evaluating the uncertainty of the heat flux, the error of the heat transfer area is not counted since the uncertainties of the tube diameter and the tube length are ±0.1 mm and its effect on the area is negligible.

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 °C) of the isothermal bath of ±0.01 °C accuracy. Since the duration to finish a set of the present test took less than 1 h, the elapsed time to estimate the uncertainty of the thermocouple probes were set as 1 h. According to the results, the deviation of the measured values from the set value is within ±0.1 °C 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 calculated as 0.15 °C. 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 °C. Therefore, the total uncertainty of the measured temperatures is defined by adding the above errors and its value is ±0.3 °C. The uncertainty in the heat transfer coefficient can be determined through the calculation of $q''/\Delta T_{sat}$ and is within ±10%.

3. Results and discussion

Fig. 2 shows variations in heat transfer as the location of the inflow holes changes. The amount of heat transfer for the annulus, except the annulus of closed bottoms (i.e., no side inflow holes), is higher than the single unrestricted tube. Results for the closed bottoms without side holes show deterioration of heat transfer comparing to the single unrestricted tube. At $q'' \geqslant 70\ \mathrm{kW/m^2}$ the tube wall superheat for the closed bottoms is higher than that of the single tube at the same heat fluxes. This means that the adoption of the bottom-closed annulus has no advantage in heat transfer enhancement at theses heat fluxes. Addition of the side inflow holes removes deterioration of heat transfer at higher heat fluxes and, moreover, still maintains higher heat transfer at low heat fluxes comparing to the single unrestricted tube. The amount of heat transfer enhancement depends on the location of the inflow holes. The most attractive results are obtained for the annulus with inflow holes at $L_h = 0.3$ m. As Kang [9] explained the major causes of the deterioration are the formation of large bubble slugs and the interference between the upward bubbles and the downward inlet liquid around the upper region of

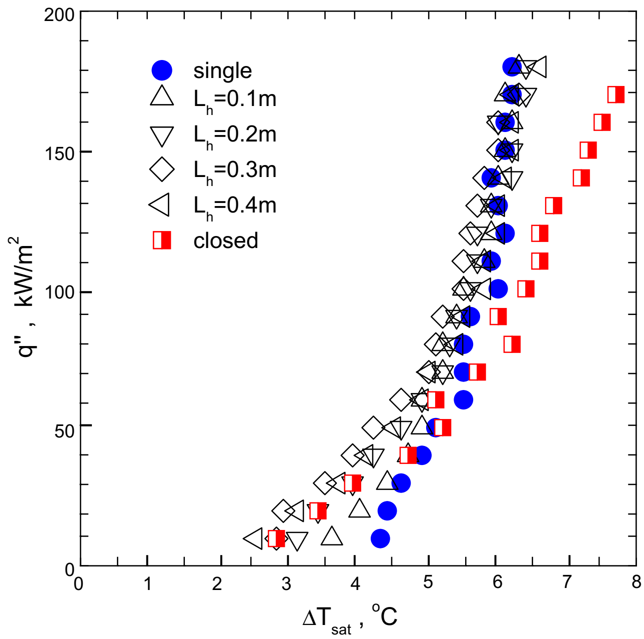

Fig. 2. Plots of q'' versus ΔT_{sat} data.

the annuls with closed bottoms. The creation of the bigger bubbles is accelerated by the liquid inflow since the liquid reduces the upward velocity of the bubbles. Once a bubble is interfered by the liquid it coalesces with relevant bubbles to generate much bigger bubbles. Two cases should be coincided with each other to generate bigger size bubbles. One is generation of enough bubbles at the lower region of the inflow location and the other one is liquid interference to reduce the velocity of the bubbles. The introduction of the side inflow holes effectively removes the possibility of the bigger bubble generation around the upper region of the tube since no enough bubbles are generated under the lower region of the inflow holes. The inlet liquid through the side inflow holes still interferes upward bubble movement at the lower regions of the tube. It generates pulsating flow to occur active liquid agitation in the annular space. This is the major cause of heat transfer enhancement at low heat fluxes.

Fig. 3 shows some photos of boiling in the annulus with inflow holes at $L_{\mathrm{h}} = 0.3$ m. Two types of photos are shown. The above four photos were taken at around the inflow holes and the bottom four photos were taken at just lower side of the inflow holes. Every photo was taken at the same camera condition. The photos are showing very rapid bubble flow inside the annular space while the bottom side photos are showing somewhat slower bubbles. The bubbles at the lower side are not enough to generate bigger bubble slugs around these regions and the upward flow is interfered by the inflowing liquid. Thereafter, flow fluctuation is generated at these lower regions. The generation of the pulsating flow is clearly observed at $L_{\mathrm{h}} \geqslant 0.2$ m. Consequently, the flowing bubbles at the upper region and the pulsating flow at the lower region of the inflow holes are considered as the major causes of liquid agitation at those regions.

To clearly observe effects of the location on pool boiling heat transfer $h_{\mathrm{b}}/h_{\mathrm{b,single}}$ versus q'' are plotted in Fig. 4. As the heat flux is less than 100 kW/m^2, the annulus with side inflow holes enhances the heat transfer rate much comparing to the single unrestricted tube. The enhancement is magnified, as the heat flux gets small. The enhancement

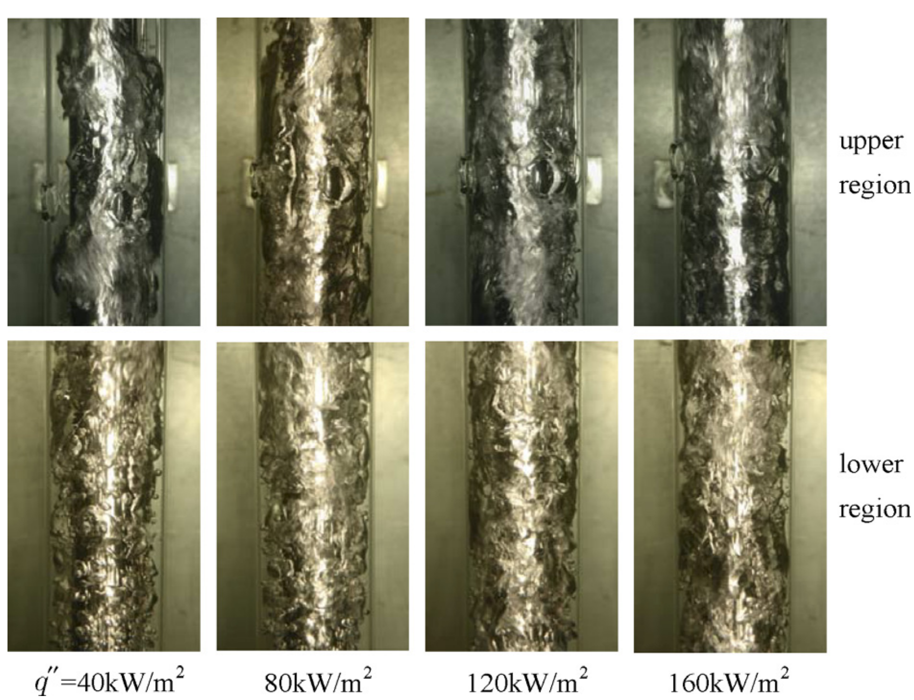

Fig. 3. Photos of pool boiling in annulus with inflow holes at $L_{\mathrm{h}} = 0.3$ m.

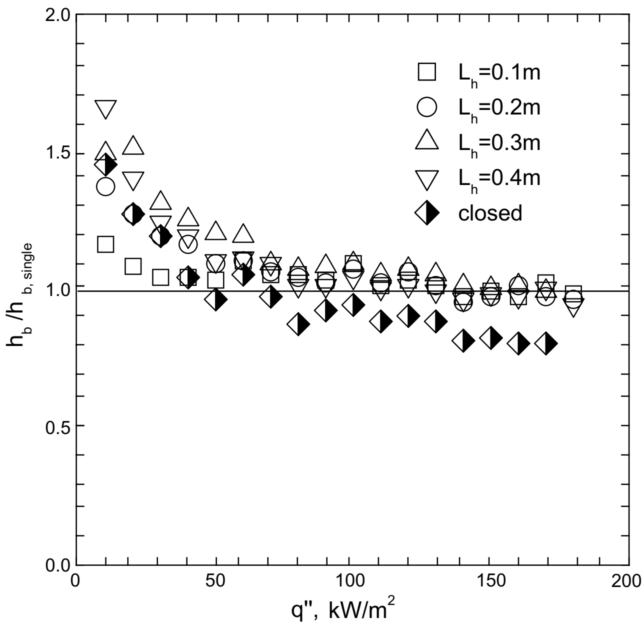

Fig. 4. Plots of $h_b/h_{b,single}$ versus q''.

is more than 50% at $q'' = 20\ kW/m^2$ and $L_h = 0.3$ m. If the heat flux gets increased more than $100\ kW/m^2$ the values (except the annulus with closed bottoms) reach asymptotic values. This is because of the increase in the effects of bigger bubbles coalesced around the upper region of the tube. The bubbles decrease the effect of liquid agitation in the annular space. As shown in the figure the deterioration of heat transfer at a higher heat flux for the annulus without side holes is not observed for the annulus with side holes. Based on the results, the adoption of the side inflow holes could be suggested as a good method to increase pool boiling heat transfer.

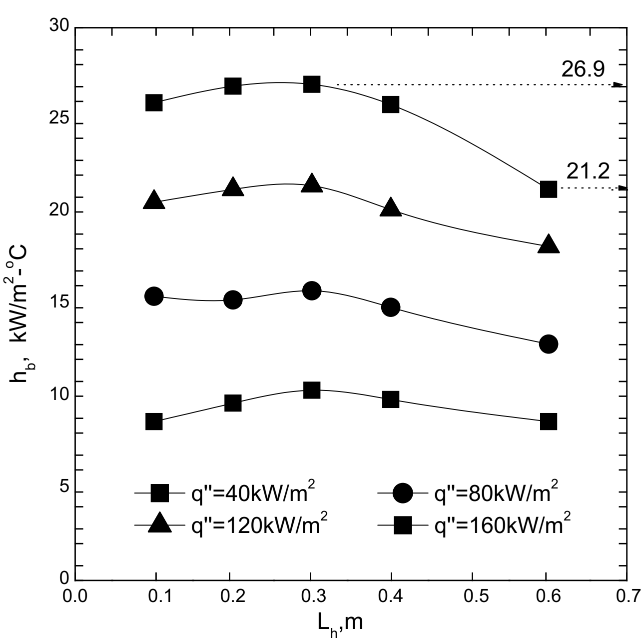

Fig. 5. Changes in h_b as L_h increases.

Changes in the heat transfer coefficients versus the location of the inflow holes are shown in Fig. 5. Four different heat fluxes have been investigated to observe the tendencies. Among the cases, $L_h = 0.6$ m at x-axis means the annulus without side holes. The heat transfer coefficients get increased at $0.1\ m \leqslant L_h \leqslant 0.3\ m$, and then get decreased at $0.3\ m < L_h$. Through the heat fluxes, the heat transfer coefficients for $L_h = 0.3$ m have the highest values comparing to the other cases. The heat transfer coefficient ($10.3\ kW/m^2\ °C$) for the annulus with side inflow holes at 0.3 m is 19.8% greater than the coefficient ($8.6\ kW/m^2\ °C$) for the annulus with side holes at 0.1 m at $q'' = 40\ kW/m^2$. To identify the enhancement two heat transfer coefficients for the annuli with inflow holes at $L_h = 0.3$ m and without the holes have been compared each other at $q'' = 160\ kW/m^2$. The heat transfer coefficients increased 26.9% (from 21.2 to $26.9\ kW/m^2\ °C$) in case of the inflow holes are applied.

4. Conclusions

To identify effects of the location of the side inflow holes on pool boiling heat transfer in a vertical annulus (gap size = 10.7 mm), a heated tube of 34 mm diameter and the water at atmospheric pressure have been tested. The change in the location of the holes results in much variations in heat transfer coefficients at low heat fluxes less than $100\ kW/m^2$. Moreover, the side inflow holes could remove the deterioration of heat transfer in the vertical annulus without side inflow holes. Locating the inflow holes at 0.3 m from the bottom of the tube results in the best results for the heat fluxes tested. Therefore, the annulus with side inflow holes could be recommended as a useful way to improve pool boiling heat transfer.

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