Available online at www.sciencedirect.com
SCIENCE DIRECT®
International Journal of Heat and Mass Transfer 49 (2006) 3752–3756
International Journal of HEAT and MASS TRANSFER
www.elsevier.com/locate/ijhmt

Technical Note

Pool boiling heat transfer in a vertical annulus with controlled inflow area at its bottom

Myeong-Gie Kang *

Department of Mechanical Engineering Education, Andong National University, 388 Songchun-dong, Andong-city, Kyungbuk 760-749, South Korea

Received 1 June 2005; received in revised form 30 March 2006
Available online 9 June 2006

Abstract

To investigate effects of the inflow rate on pool boiling heat transfer in a vertical annulus, the inflow area at its bottom has been changed from 0 to 1060.26 mm^2. For the test, a heated tube of 16.5 mm diameter and water at atmospheric pressure has been used. To elucidate effects of the inflow area on heat transfer results of the annulus are compared to the data of a single unrestricted tube. The change in the inflow rate at the bottom of the annulus results in variation in heat transfer coefficients. When the inflow area is 176.71 mm^2 the deterioration point of heat transfer coefficients gets moved up to the higher heat fluxes because of the convective flow at the bottom regions.

1. Introduction

One of the effective methods to increase heat transfer coefficients of pool boiling is considering a confined space around a heat exchanging tube. To have higher heat transfer coefficients is very important if the space for the heat exchanger installation is very limited like advanced light water reactors [1]. Major geometries studied for the crevices are annuli [2–5] and plates [6,7]. Some geometry has closed bottoms [2,4–6].

It is well known from the literature that the confined boiling can result in heat transfer improvements up to 300–800% at low heat fluxes, as compared with unconfined boiling. However, a deterioration of heat transfer appears at higher heat fluxes for confined than for unrestricted boiling [2,4]. The cause for the deterioration was suggested as active bubble coalescence at the upper regions of the annulus [4]. Around the upper region of the annulus the upward movement of the bubble slugs is interrupted by the downward liquid. 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 investigation of any possible ways to prevent the deterioration is needed in advance. Recently, Kang [5] published some results considering changes in the outer tube length of the annulus and identified that reduction of the outer tube length could remove the deterioration point to a higher heat flux.

Since the major cause of the bigger bubble coalescence which results in the deterioration is partly because of the no inflow at the bottom of the annulus with closed bottoms, the present study is aimed at the investigation of the way to improve heat transfer in the annulus through changing the inflow at the bottom of the annulus. Up to the author's knowledge, no previous results concerning the ways have been published yet.

2. Experiments

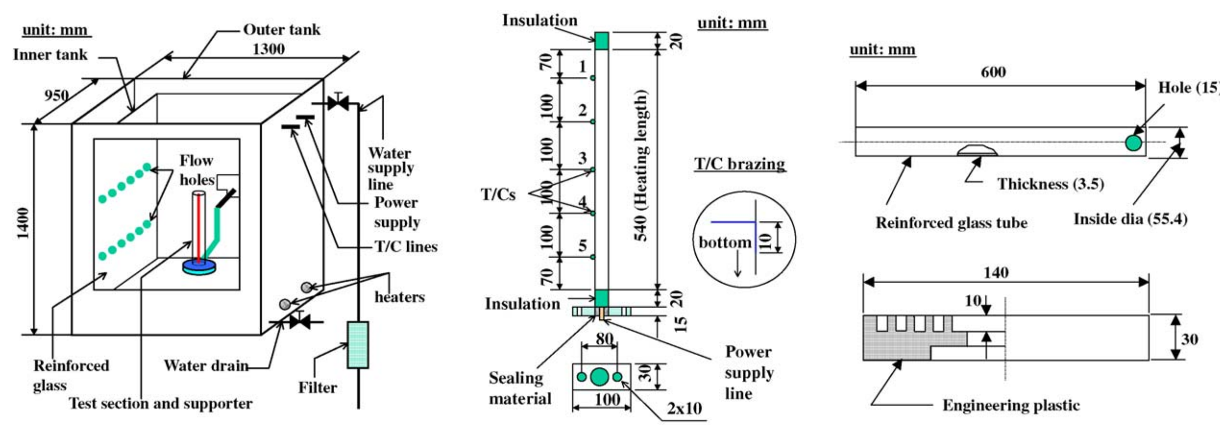
A schematic view of the present experimental apparatus and a test section is shown in Fig. 1. The water storage tank (Fig. 1(a)) is made of stainless steel and has a rectangular cross section (950 × 1300 mm) and a height of 1400 mm.

* Tel.: +82 54 820 5483; fax: +82 54 823 1766.
E-mail address: mgkang@andong.ac.kr

0017-9310/$ - see front matter © 2006 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2006.03.030

Nomenclature

A_f	inflow area at the bottom of the annulus	L	heated tube length
A_o	flow area in the annulus $(\pi(D_i^2 - D^2)/4)$	q''	heat flux
D	diameter of the heating tube	T_{sat}	saturation temperature
D_i	inner diameter of the outer tube	T_W	tube wall temperature
h_b	boiling heat transfer coefficient	V	supplied voltage
I	supplied current	ΔT_{sat}	tube wall superheating $(T_W - T_{sat})$



(a) water tank (b) heated tube (c) glass tube and supporter

Fig. 1. Schematic diagram of the experimental apparatus.

The sizes of the inner tank are $800 \times 1000 \times 1100$ mm (depth × width × height). Four auxiliary heaters (5 kW/heater) were installed at the space between the inside and outside tank bottoms. The heat exchanger tubes are simulated by a resistance heater (Fig. 1(b)) made of a very smooth stainless steel tube ($L = 0.54$ m and $D = 16.5$ mm). The surface of the tube was finished through a

Table 1
Geometries of the outer tube

Number of holes	0	1	2	3	4	5	6
A_f (mm^2)	0	176.7	353.4	530.1	706.8	883.6	1060.3
A_f/A_o	0	0.08	0.16	0.24	0.32	0.04	0.48

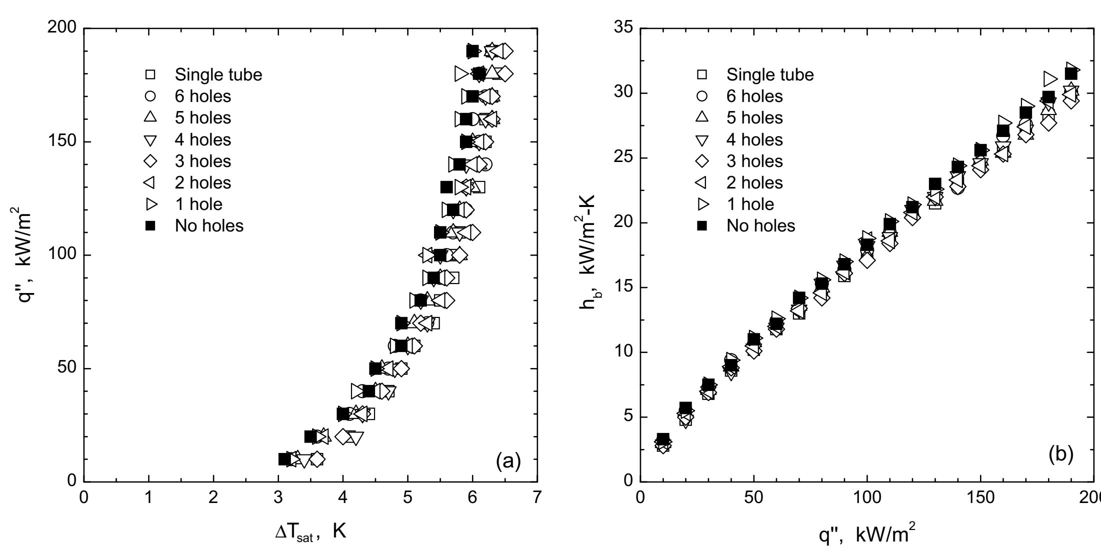


Fig. 2. Boiling heat transfer in an annulus as the inflow area at the bottom is changed.

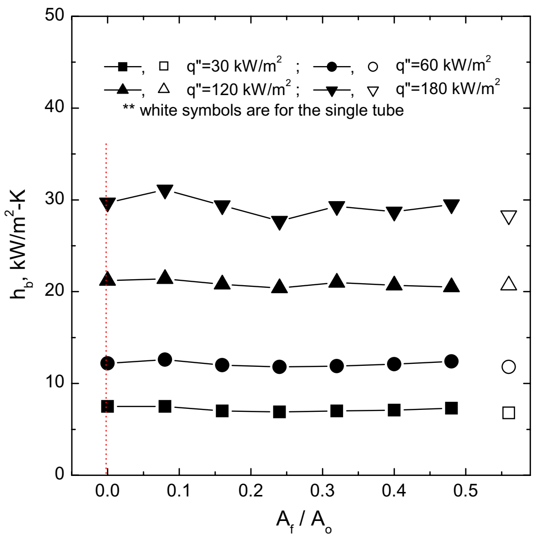


Fig. 3. Variations in heat transfer coefficients as the ratio of the inlet flow area to the annulus area increases.

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. 1(a)) and an auxiliary supporter (Fig. 1(c)) is used to fix a glass tube (Fig. 1(c)). To make the annular condition, glass tubes (gap size = 19.45 mm) of 55.4 mm inner diameter and 600 mm length were used. The inflow into the annular space at its bottom was controlled by the number of inflow holes (15 mm diameter) changing from 0 to 6 as listed in Table 1. The ratio of the inflow area at the bottom of the annulus (A_f) to the flow area in the annulus (A_o) is between 0 and 0.48.

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:

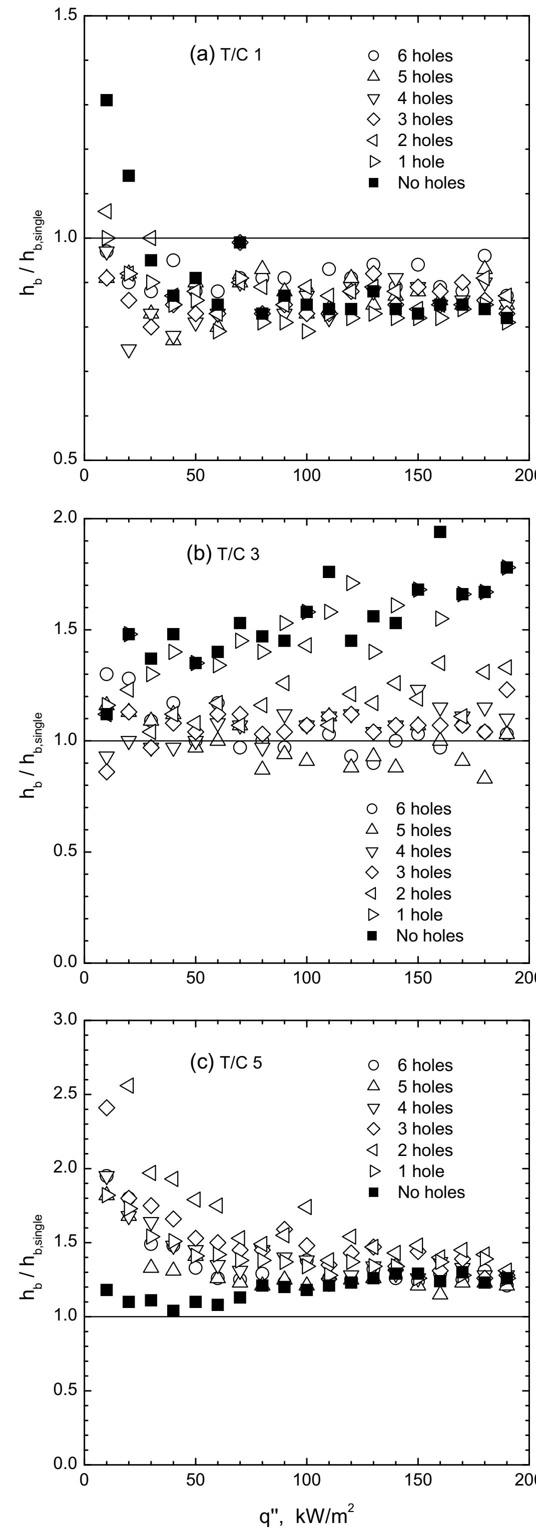


Fig. 4. Variations in local coefficients as the inflow area at the bottom is changed.

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

where V and I are the supplied voltage (in volt) and current (in ampere) and D and L are the outside diameter and the

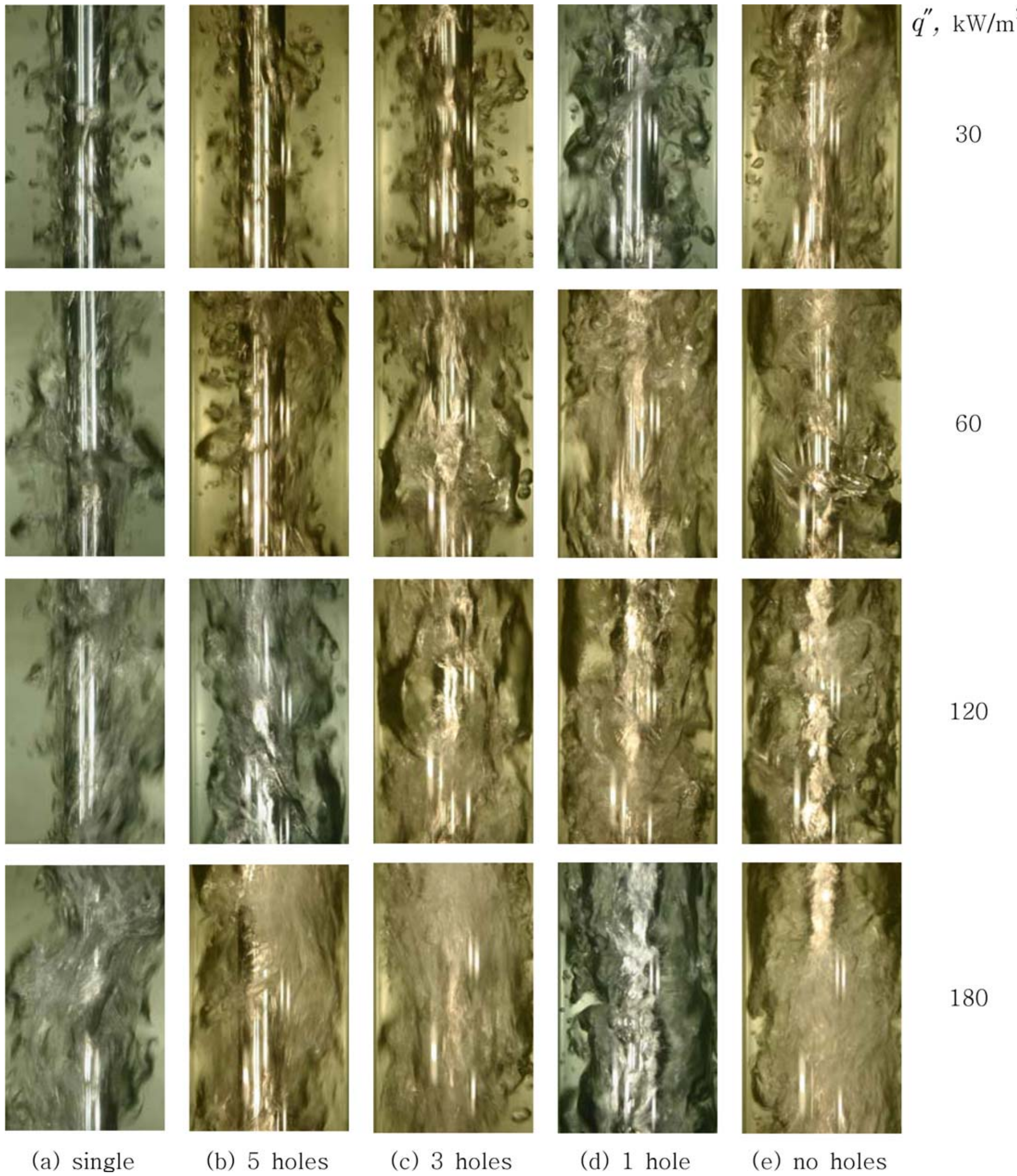


Fig. 5. Photographs of pool boiling in the annulus as the heat flux changes.

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 uncertainty in the heat flux is estimated to be $\pm 1.0\%$. The total uncertainty of the measured temperatures is estimated as ± 0.3 K. The uncertainty in the heat transfer coefficient can be determined through the calculation of $q''/\Delta T_{sat}$ and is within $\pm 10\%$.

3. Results and discussion

Fig. 2 shows variations in the heat transfer as the inflow area of the outer tube changes. The amount of the heat transfer for the annulus is higher comparing to the single tube. As the heat flux increases the difference between the results get increased. The annulus with closed bottoms (i.e., no holes) has the highest heat transfer coefficients among the test cases except the annulus with one hole. At $q'' \leqslant 150\ \mathrm{kW/m^2}$, two cases of no holes and one hole have almost same heat transfer coefficient. However, as the heat flux increases more than $150\ \mathrm{kW/m^2}$ the heat transfer coefficient for the annulus with one hole gets greater than the annulus with closed bottoms. Variations in the heat transfer coefficient as the ratio of A_f/A_o increases are shown in Fig. 3 as a function of the heat flux. At $q'' = 180\ \mathrm{kW/m^2}$ the heat transfer coefficient has the almost same value as A_f/A_o decreases to 0.24. After then, the coefficient increases as A_f/A_o decreases to 0.08. When the ratio is 0.08 the heat transfer coefficient has the highest value of $31.1\ \mathrm{kW/m^2}$ and is greater than that of the annulus with closed bottoms.

The introduction of the annulus firstly increases the intensity of the liquid agitation around the heated tube. However, at $A_f/A_o > 0.24$ the intensity of the liquid agitation is not strong. At $0.08 \leqslant A_f/A_o \leqslant 0.24$ the intensity increases enough to change the heat transfer coefficient. This means the decrease in the inflow area results in stronger liquid agitation. The restriction in the inflow area decreases effects of the convective flow at lower regions of the annulus. This results in the deterioration as shown in Figs. 2 and 3 at higher heat fluxes. This tendency would be clearly observed, as the gap size gets smaller.

To explain the cause of the tendency in detail, local heat transfer coefficients at the thermocouple locations are calculated and shown in Fig. 4. At T/C1 ratios for the annulus with no holes has the higher value than the other cases at lower heat fluxes less than 30 kW/m^2. This means that no inflow at the bottom of the annulus generates active liquid agitation even at lower heat fluxes. As the heat flux gets higher the intensity of bubble coalescence gets stronger and deterioration in the ratio is observed at the results for the annuli. However, increase in the inflow area (A_f) reduces the intensity of the bubble coalescence at the region. Restriction in the inflow area results in stronger liquid agitation around the T/C3 regions. As shown in Fig. 4(b) much higher ratios are observed for the cases of no holes and one hole comparing to the results of the annuli with several holes. This region is relatively free from the stronger bubble coalescence comparing to the T/C1 location. Reduction in the inflow rate at the bottom of the annulus generates active pulsating flow in the annulus and this results in stronger liquid agitation. At T/C5 every ratios of the annuli is greater than 1. However, mechanisms affecting the heat transfer are different among the cases. The faster convective flow due to the inflow area has the major role to increase the heat transfer coefficient at these regions. The annulus with closed bottoms has no effects due to the inlet flow and has smaller ratios at $q'' \leqslant 80$ kW/m^2. The major mechanism for the closed bottom is liquid agitation by the pulsating flow in the annulus. Its effect increases as the heat flux increases. However, the portion of the effects of the convective flow on the heat transfer decreases as the heat flux increases and, accordingly, effects of the nucleation become dominant [8]. Through the heat fluxes the ratio for the annulus with one hole is larger than the annulus with closed bottoms. This can explain the tendency shown in Fig. 2.

Fig. 5 shows some photographs of boiling in the annulus as the number of flow holes changes. Those photographs were taken at around the mid-point of the tube length. As shown in the photographs increase in the heat flux and the decrease in the number of flow holes results in bigger bubbles. Bubbles coming from the bottom side generate not only active liquid agitation in the space but also stronger bubble coalescence around the upper regions.

4. Conclusions

To identify effects of the inflow area at the bottom of the annulus on pool boiling heat transfer in a vertical annulus (gap size = 19.45 mm), a heated tube of 16.5 mm diameter and water at atmospheric pressure has been tested. The change in the inflow area results in variations in heat transfer coefficients. As A_f/A_o is 0.08 a deterioration in the heat transfer at higher heat fluxes is not observed while the coefficients is still much higher than the unrestricted single tube. Therefore, the annulus with restricted inflow at the bottom region could be recommended as a useful way to improve pool boiling heat transfer.

References

[1] M.H. Chun, M.G. Kang, Effects of heat exchanger tube parameters on nucleate pool boiling heat transfer, ASME J. Heat Transfer 120 (1998) 468–476.
[2] S.C. Yao, Y. Chang, Pool boiling heat transfer in a confined space, Int. J. Heat Mass Transfer 26 (1983) 841–848.
[3] Y.H. Hung, S.C. Yao, Pool boiling heat transfer in narrow horizontal annular crevices, ASME J. Heat Transfer 107 (1985) 656–662.
[4] M.G. Kang, Pool boiling heat transfer in vertical annular crevices, Int. J. Heat Mass Transfer 45 (15) (2002) 3245–3249.
[5] M.G. Kang, Effects of outer tube length on saturated pool boiling heat transfer in a vertical annulus with closed bottom, Int. J. Heat Mass Transfer 48 (13) (2005) 2795–2800.
[6] Y. Fujita, H. Ohta, S. Uchida, K. Nishikawa, Nucleate boiling heat transfer and critical heat flux in narrow space between rectangular spaces, Int. J. Heat Mass Transfer 31 (1988) 229–239.
[7] J.C. Passos, F.R. Hirata, L.F.B. Possamai, M. Balsamo, M. Misale, Confined boiling of FC72 and FC87 on a downward facing heating copper disk, Int. J. Heat Fluid Flow 25 (2004) 313–319.
[8] W.M. Rohsenow, A method of correlating heat transfer data for surface boiling of liquids, ASME J. Heat Transfer 74 (1952) 969–976.