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Pool boiling heat transfer in a vertical annulus as the bottom inflow area changes

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Abstract

To identify effects of the bottom inflow area on pool boiling heat transfer in a vertical annulus, three tube diameters (16.5, 25.4, 34.0 mm) and the saturated water at atmospheric pressure has been tested. The inflow area has been changed from 0 to 1060.3 mm^2. To clarify effects of the inflow area on heat transfer results of the annulus are compared to the data of a single unrestricted tube. The inflow area changes heat transfer coefficients much and moves the deterioration point of heat transfer coefficients to the higher heat fluxes. To quantify effects of the inflow area on heat transfer, a new empirical correlation has been developed in terms of the area ratio, inflow area, and the heat flux. The correlation predicts the heat transfer data of boiling within $\pm 10\%$.

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 [2,3]. To determine the required heat transfer surface area as well as to evaluate the system performance during postulated accidents, overall heat transfer coefficients applicable for the passive heat exchangers are needed. Although many researchers 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 [4–6].

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]. Studies on the crevices can be divided into two categories. One of them is about annuli [3,7,8] and the other one is about plates [9–11]. In addition to the geometric conditions, flow to the crevices can be limited. Some geometry has a closed bottom [3,7,10,12]. 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 [7,9]. However, a deterioration of heat transfer appears at high heat fluxes for confined than for unrestricted boiling [7,10]. The effect of gap sizes (s) on pool boiling is fluid dependent and this can change the general trend mentioned above [7,8]. The boiling heat transfer coefficient (h_b) usually increases when gap size decreases at low heat fluxes whereas it decreases at higher heat fluxes. However, h_b increases when gap size decreases to a certain value [7,8,10]. Further decrease in gap size results in sudden decrease in h_b. Summarizing the previous works about crevice effects on pool boiling heat transfer it can be said that the amount of h_b is highly dependent on the geometry and confinement condition.

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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
A_r	ratio of the areas $(=A_f/A_o)$	s	gap size
D	diameter of the heated 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 superheat $(=T_W - T_{sat})$

Improved heat transfer characteristics with the restriction might be attributed to an increase in the heat transfer coefficient due to vaporization from the thin liquid film on the heating surface or increased bubble activity [8,9]. In the confined spaces, at a fixed heat fluxes, the mass of vapor generated is constant so that with decreasing gap size higher vapor velocities are induced. With the increased vapor velocities, the shear stress on the liquid film at the heated surface increases and the liquid film is reduced in thickness. Since the major heat transfer resistance is the heat conduction across the liquid film, the reduced film thickness increases the heat transfer coefficient [8]. According to Cornwell and Houston, the bubbles sliding on the heated surface agitate environmental liquid [13]. In a confined space a kind of pulsating flow due to the bubbles is created and, as a result very active liquid agitation is generated [3]. The increase in the intensity of the liquid agitation increases heat transfer.

The cause of the deterioration is suggested as active bubble coalescence at the upper regions of the annulus [3]. When too many coalesced bubbles are generated at high heat fluxes, the heating surface is almost covered with a single mass of vapor [9]. Around the upper region of the annulus the downward fresh liquid interrupts the upward movement of the coalesced bubbles. 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. Kang [12] 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 area at the bottom of the annulus. Recently, Kang [14] published some preliminary results on the effects of the bottom inflow area on pool boiling heat transfer in the vertical annulus using the 16.5 mm diameter tube and the saturated water. Kang identified that the change in the inflow area resulted in some improvement in the heat transfer coefficient. Although Kang [14] observed the possible advantage of the inflow area change to enhance heat transfer in vertical annulus, some more detailed study is needed to extend the applicability of the results. Since Kang [14] used somewhat larger gap size (i.e., 16.5 mm), the change in heat transfer due to the inflow area was not significant. Therefore, the present study is focused on the extension of the Kang's work [14] with considering other gap sizes and tube diameters. In addition, an empirical correlation will be suggested to quantify the effects of the major test parameters on pool boiling heat transfer.

2. Experiments

A schematic view of the present experimental apparatus and a test section is shown in Fig. 1. The water 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) are installed at the space between the inside and outside tank bottoms. The heat exchanger tubes are simulated by a resistance heater (Fig. 1b) made of a very smooth stainless steel tube (see Table 1). The surface of the tube is finished through a buffing process to have a smooth surface. Electric power of 220 V AC is supplied through the bottom side of the tube.

The tube outside is instrumented with five T-type sheathed thermocouples (diameter is 1.5 mm). The thermocouple tip (about 10 mm) is brazed on the tube wall. The water temperatures are 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 are 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 are 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 tube supporter (Fig. 1b) is used to fix a glass tube (Fig. 1b). To make the annular condition, glass tubes of 55.4 mm inner diameter and 600 mm length are used. Accordingly, three different gap sizes (i.e., 10.7, 15.0, and 19.5 mm) have been generated according to the combination of the heated tube and the outer glass tube. The number and the diameter of

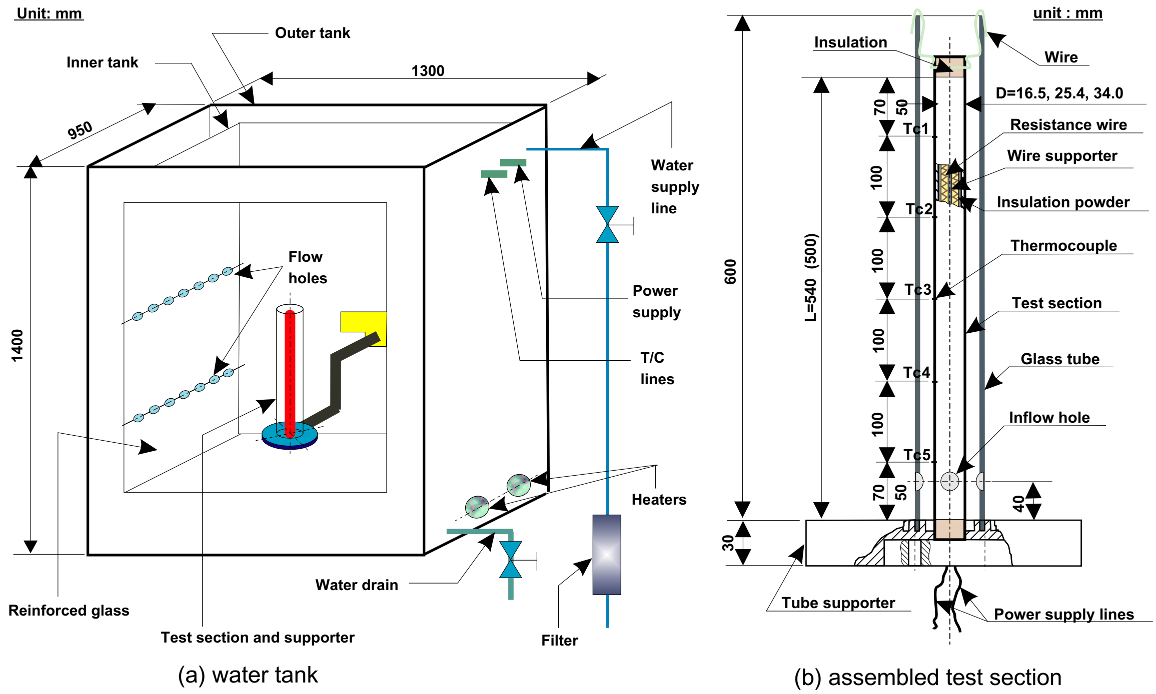


Fig. 1. Schematic diagram of the experimental apparatus.

Table 1
Test matrix and q'' versus ΔT_{sat} data

D (mm)	L (m)	s (mm)	A_f (mm^2)	A_o (mm^2)	A_r (A_f/A_o)	q'' (kW/m^2)	Remark	Number of data
16.5	0.54	–	–	–	–	0–190	Single	19
16.5	0.54	19.5	0	2196.7	0	0–190	Annulus	19
16.5	0.54	19.5	176.7	2196.7	0.08	0–190	Annulus	19
16.5	0.54	19.5	353.4	2196.7	0.16	0–190	Annulus	19
16.5	0.54	19.5	530.1	2196.7	0.24	0–190	Annulus	19
16.5	0.54	19.5	706.8	2196.7	0.32	0–190	Annulus	19
16.5	0.54	19.5	883.6	2196.7	0.40	0–190	Annulus	19
16.5	0.54	19.5	1060.3	2196.7	0.48	0–190	Annulus	19
25.4	0.54	–	–	–	–	0–100	Single	10
25.4	0.54	15.0	0	1903.8	0	0–100	Annulus	10
25.4	0.54	15.0	50.3	1903.8	0.03	0–100	Annulus	10
25.4	0.54	15.0	176.7	1903.8	0.09	0–100	Annulus	10
25.4	0.54	15.0	530.1	1903.8	0.28	0–100	Annulus	10
25.4	0.54	15.0	1060.3	1903.8	0.56	0–100	Annulus	10
34.0	0.50	–	–	–	–	0–170	Single	17
34.0	0.50	10.7	0	1502.6	0	0–170	Annulus	17
34.0	0.50	10.7	19.6	1502.6	0.01	0–170	Annulus	17
34.0	0.50	10.7	50.3	1502.6	0.03	0–170	Annulus	17
34.0	0.50	10.7	113.1	1502.6	0.08	0–170	Annulus	17
34.0	0.50	10.7	176.7	1502.6	0.12	0–170	Annulus	17
34.0	0.50	10.7	353.4	1502.6	0.24	0–170	Annulus	17
34.0	0.50	10.7	530.1	1502.6	0.35	0–170	Annulus	17
34.0	0.50	10.7	706.8	1502.6	0.47	0–170	Annulus	17
34.0	0.50	10.7	883.6	1502.6	0.59	0–170	Annulus	17
34.0	0.50	10.7	1060.3	1502.6	0.71	0–170	Annulus	17

inflow holes changed the amount of the fresh water inflow into the annular space at its bottom. The detailed information of the tests is listed in Table 1. The ratio (A_r) between the inflow area (A_f) at the bottom of the annulus and the flow area (A_o) in the annulus is ranging from 0 to 0.71.

After the water 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 (q'') 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:

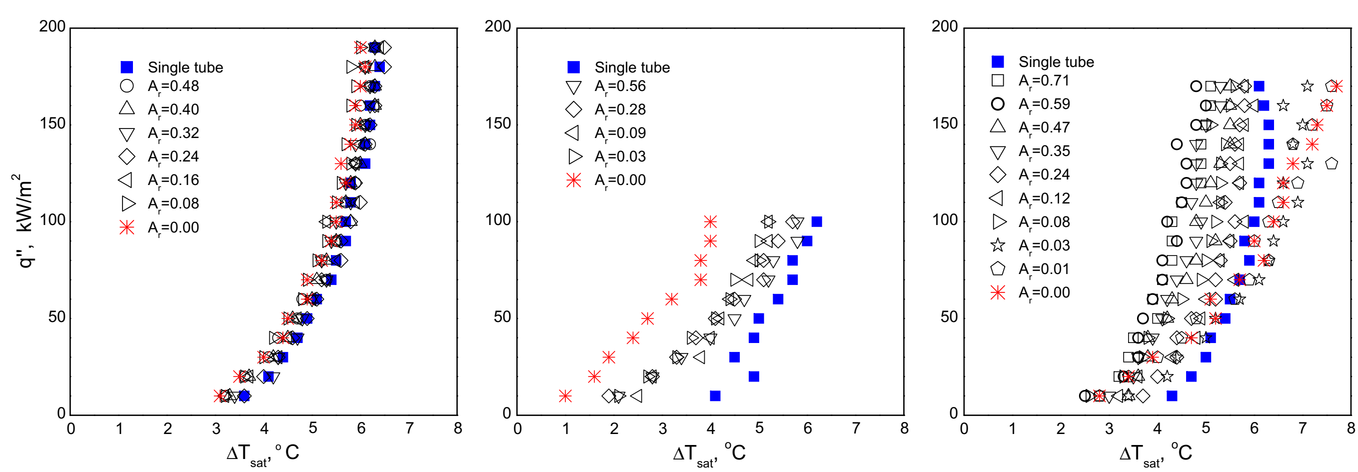
$$q'' = \frac{VI}{\pi DL} = h_b \Delta T_{sat} = h_b(T_W - T_{sat}) \tag{1}$$

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.

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 taken into account 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 the thermocouple probe, three thermocouples brazed on the tube surface were submerged in an isothermal bath of ±0.01 °C accuracy containing 80 °C water. Since the time to complete one set of the present test was less than 1 h, the elapsed time to estimate the uncertainty of the thermocouple probes was 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 through the brazing metal must be evaluated. The brazing metal is a type of brass and the averaged brazing thickness is less than 0.1 mm. The maximum temperature decreases due to this brazing is estimated 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 possible maximum uncertainty of the measured temperatures is defined by adding the above errors, giving a value of ±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 curves of the heat flux versus tube wall superheat as the bottom inflow area into the annulus changes. The heat transfer of the annulus is very much dependent on the inflow area. As the tube diameter is 16.5 mm ($s = 19.5$ mm), the annulus with closed bottoms ($A_r = 0.00$) has the highest heat transfer coefficients among the tests except the annulus with $A_r = 0.08$. At $q'' \leqslant 150$ kW/m^2, two cases have almost same heat transfer coefficient. However, as the heat flux increases more than 150 kW/m^2 the heat transfer coefficient for the annulus of $A_r = 0.08$ gets greater than the annulus of $A_r = 0.00$. The amount of heat transfer for $D = 25.4$ mm ($s = 15.0$ mm) is slightly different from the results of $D = 16.5$ mm. The annulus with closed bottoms ($A_r = 0.00$) has the highest heat transfer coefficients among the inflow area tested. The difference in ΔT_{sat} is much larger than the results for $D = 16.5$ mm. When the tube diameter is 34.0 mm, which has the smallest gap size of 10.7 mm, much different heat transfer feature is observed comparing to the other two cases. When $q'' \leqslant 50$ kW/m^2 results for the annuli show enhanced heat transfer than the unrestricted single tube. When $q'' > 50$ kW/m^2 heat transfer characteristic trends can be divided into two categories. Results of the annuli show improved heat transfer than the single tube as



(a) D=16.5 mm (b) D=25.4 mm (c) D=34.0 mm

Fig. 2. Curves of q'' versus ΔT_{sat}.

$A_r > 0.03$. When $A_r \leqslant 0.03$, obvious heat transfer deterioration is observed on the results for the annulus comparing to the results of the single tube. The tendency in heat transfer variation due to the bottom inflow area is highly dependent on the gap size. The decrease in A_r results in heat transfer increase when the tube diameters are 16.5 mm and 25.4 mm. As $D = 34.0$ mm the decrease in A_r results in heat transfer increase until the value gets to 0.59. At $A_r = 0.59$ it shows the most improved heat transfer characteristics. After then, the decrease in A_r results in continual decrease in heat transfer and shows, finally, deterioration in heat transfer as $A_r \leqslant 0.03$. In summary, Fig. 2 shows that the smaller gap size (i.e., larger diameter for the case) is more sensitive to the bottom inflow area. The reasons are because the bubbles in the smaller annular space (a) agitate more actively at low heat fluxes and (b) generate bigger bubble slugs through coalescing with relevant bubbles at higher heat fluxes. The former enhances heat transfer, whereas the latter deteriorates heat transfer on the heated tube surface.

To examine the combined effect of the inflow area and the gap size on nucleate pool boiling, h_b versus q'' data obtained for the tests of three different tube diameters are plotted in Fig. 3. In these figures the following observations can be made:

(1) It is particularly interesting to note that the difference between heat transfer coefficients for the annuli and the single tube at $q'' = 100$ kW/m^2 and $A_r = 0.08$ becomes larger as the gap size is small. That is, when $D = 16.5$ mm and only 7.4% (from 17.5 to 18.8 kW/m^2 °C) increase in h_b is observed, whereas more than 42.2% (from 16.6 to 23.6 kW/m^2 °C) increase in h_b is obtained when $D = 34.0$ mm.

(2) The effects of the inflow area on the heat transfer coefficients are clearly observed, as the annular gap size is small and the tube diameter is large. At $q'' = 100$ kW/m^2 and $s = 10.7$ mm ($D = 34.0$ mm) the increase of A_r from 0.01 to 0.71 results in more than 45.3% (from 15.9 to 23.1 kW/m^2 °C) increase in h_b. As the gap size gets larger, the variation rate in heat transfer coefficient becomes small. In other words, as the gap size is 15.0 mm ($D = 25.4$ mm) the increase of A_r from 0.03 to 0.56 results in more than 11.4% (from 19.3 to 17.1 kW/m^2 °C) decrease in h_b at the same heat flux. The similar tendency is observed for $D = 16.5$ mm. The reason for this is partly because the number of bubbles on the tube surface increases more rapidly for the larger diameter. The bubbles generate more active liquid agitation, which increases heat transfer rate, in the smaller annular space.

To observe the generation and agitation of bubbles some photos of boiling are shown in Fig. 4 as area ratio changes. Those photographs are taken at around the mid-point of the tube length. As shown in the photographs the increase in the heat flux and the decrease in the inflow area create 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. The bubble coalescence causes the deterioration in heat transfer, as the gap size is small. When an annulus has a small gap size and the inflow area at the bottom region is not enough for the inflow of the fresh liquid the liquid should come into the space through the upper side of the annulus. The inflow through upper region disturbs the outward bubble flow from the annular space. Accordingly, larger bubbles are generated in the space.

To explain the difference in heat transfer among the tube diameters, local heat transfer coefficients at the thermocouple locations are calculated. The ratios ($h_b/h_{b,single}$) of the local heat transfer coefficients for the annulus over the coefficient for the single tube are shown in Fig. 5. For the comparison results of the similar area ratios for the different diameters have been selected. Although the average heat transfer coefficients for the three diameters at a given heat flux are larger than that of the single tube as shown in

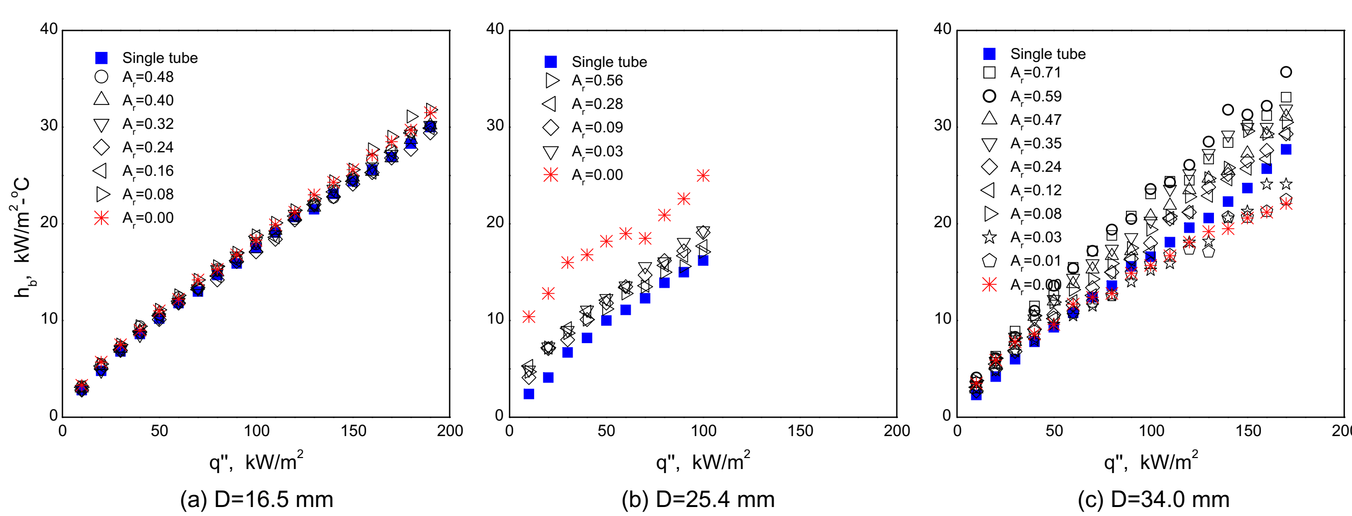


Fig. 3. Curves of h_b versus q''.

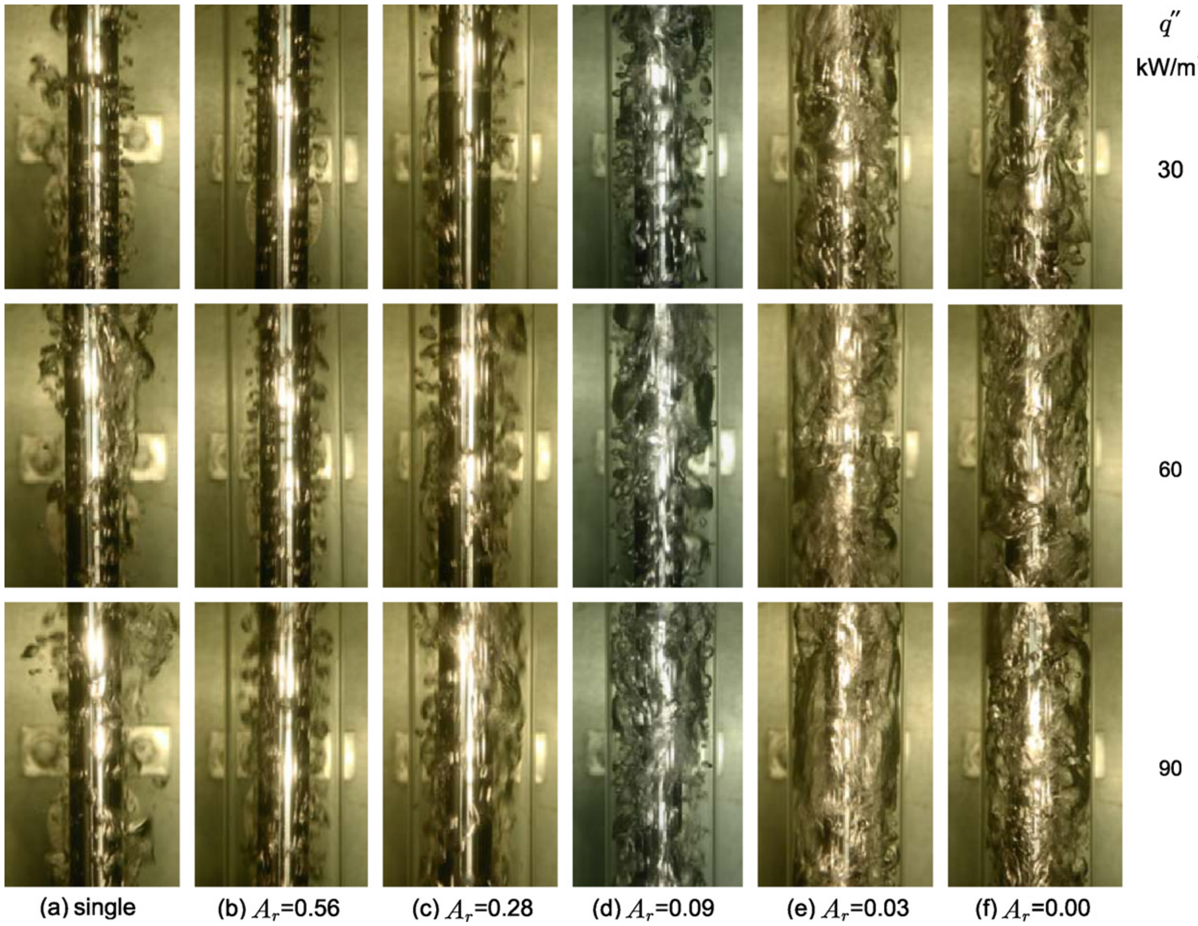


Fig. 4. Photos of pool boiling as the heat flux changes ($D = 25.4$ mm).

Fig. 2, much difference among the local heat transfer coefficients are observed. Comparing the results of the annulus with $A_r = 0.00$ to the heat transfer coefficients of the single tube, deterioration in heat transfer is observed at the results of T/C1 (thermocouple 1) for $D = 16.5$ mm and T/C1, T/C3, and T/C5 for $D = 34.0$ mm as the heat flux increases. However only a symptom of the deterioration is observed at the results of T/C3 for $D = 25.4$ mm. At T/C1 for $D = 25.4$ mm the heat transfer coefficients for $A_r = 0.00$ are higher than the results for $A_r = 0.09$. This is partly because the increase in liquid agitation. For the case, the gap size for the annulus is enough for the upside inflow throughout the heat fluxes. In other words, since the gap size is not enough for the inflow through the upside of the annulus, a kind of interference occurs between the outward bubble flow and the inward fresh liquid. Thereafter, the bubbles stay in the annular space longer than the annulus of the larger gap size. The ratios of the heat transfer coefficients at T/C5 for $A_r = 0.08$ or 0.09 are higher than the annulus with no inflow area at its bottom (i.e., $A_r = 0.00$). The cause for the tendency is the existence of the convective flow due to the inward fresh liquid through the inflow holes.

Fig. 6 shows curves of q'' versus ΔT_{sat} as a function of the bottom inflow area (A_f). Results of the three different diameters are shown for a comparison. Results of the single tubes show a similar trend as the results of Cornwell et al. [16]. Cornwell et al. identified that the heat transfer decreased with increasing D until $D = 30$ mm, then increased slightly. However, results of the annuli are very much different from the tendency. As A_f decreases from 1060.3 to 0.0 mm^2 effects of the bubbles become stronger. The effects get magnified as the annular gap size becomes smaller. When $D = 16.5$ mm ($s = 19.5$ mm) no clear variations in heat transfer is observed regarding the changes in the inflow area. For the smallest gap size ($D = 34.0$ mm, $s = 10.7$ mm) curves of q'' versus ΔT_{sat} shifts to the higher wall superheat as the inflow area gets smaller. When $A_f \leqslant 176.7$ mm^2 effects of the convective flow and liquid agitation enhance heat transfer comparing to the single tube. However, effects of the convective flow on heat transfer are decreased as the heat flux increases and, accordingly, effects of the nucleation become dominate [17]. As the heat flux becomes higher, lots of bubbles are generated. Since the bubbles come out from the upper exit of the annulus, bunches of bubbles can be interfered by the inlet liquid through the upper region of the annulus. When the gap size is large enough, the interference does not decrease heat transfer much. As the gap size is small the interference becomes the dominant factor when the heat flux increases and the inflow area decreases. When the gap size is moderate

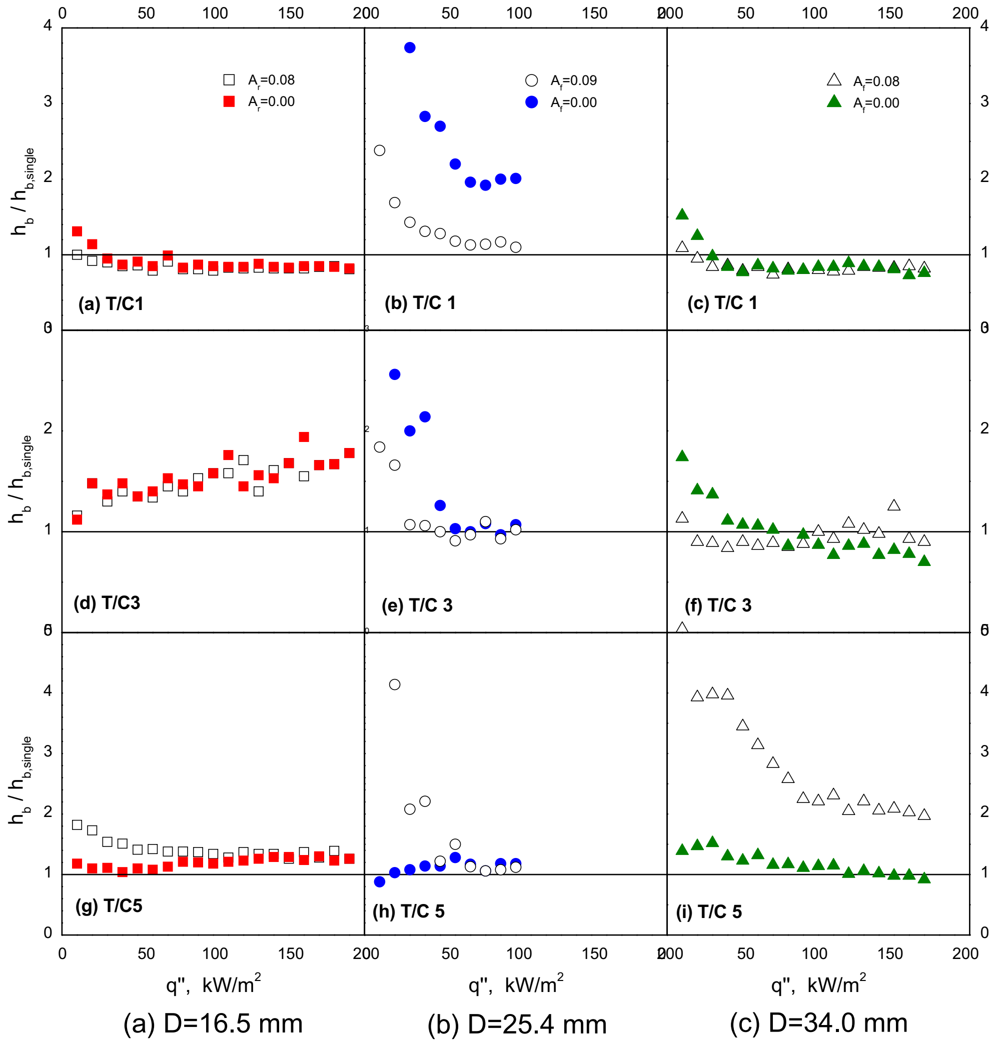


Fig. 5. Variations in local coefficients as the inflow area at the bottom is changed.

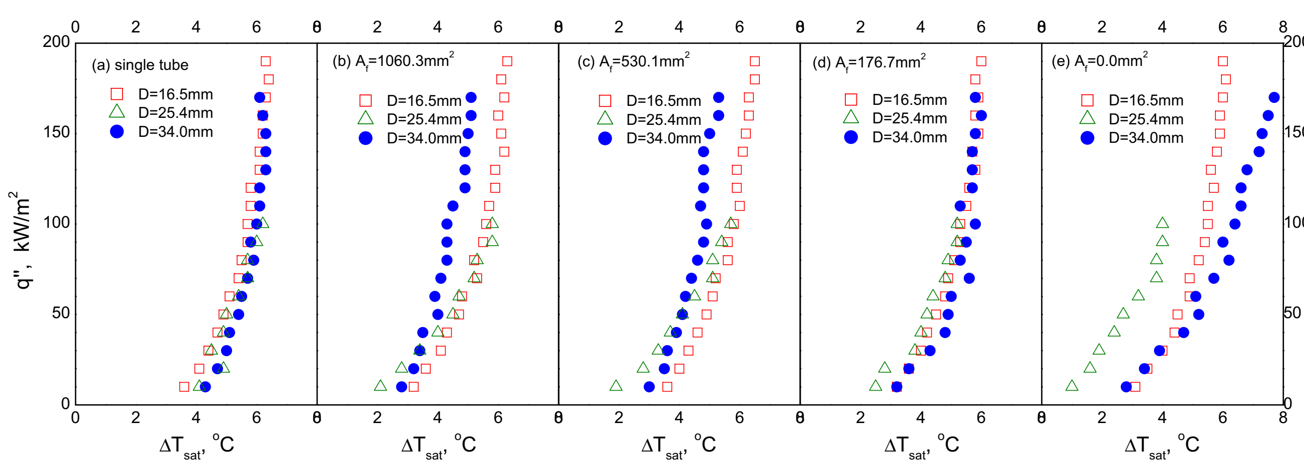


Fig. 6. Comparison of the results for the different tube diameters.

(i.e., $D = 25.4$ mm and $s = 15.0$ mm) no visible effects of the flow interference is observed even at the annulus of $A_f = 0.0\ mm^2$. For the case only adverse effects of the bubbles are observed in heat transfer enhancement. In summary, Fig. 6 shows that the smaller gap size is more sensitive to the adoption of the revised annular space.

Variations in the heat transfer coefficient as the ratio of A_f/A_o increases are shown in Fig. 7 as a function of the

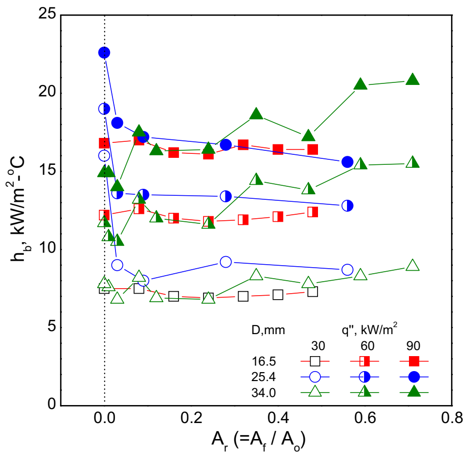


Fig. 7. Variations in heat transfer coefficients as the ratio of the inlet flow area to the annulus area increases.

heat flux. A clear difference in heat transfer tendency is observed because of the A_r change. When $A_r \leqslant 0.2$ heat transfer enhancement is observed for 16.5 and 25.4 mm tube diameters. As $D = 34.0$ mm enhanced heat transfer is observed at $A_r > 0.2$ for the heat fluxes of 60 and 90 kW/m^2. The tendency is regarded as a result of the competing heat transfer mechanisms. The major mechanisms affecting on present pool boiling heat transfer can be counted as (a) nucleation site density, (b) liquid agitation, (c) convective flow, and (d) the interference between the coalesced bubbles and the fresh liquid. When the area ratio is smaller than 0.2 the liquid agitation and the interference are the dominant mechanisms. The interference becomes activated, as the gap size is small. When the area ratio is larger than 0.2 the convective flow becomes dominated. Moreover, the interference becomes less since lots of inflow comes into the space through bottom inflow holes. The site density is increased as the heat flux increases. The increase in the intensity of liquid agitation and the convective flow and the site density enhances heat transfer. In other words, the increase in the interference followed after large bubble coalescence in the narrow space results in heat transfer deterioration.

4. Correlations of experimental data

As summarized in Table 1, a total of 307 data points have been obtained for the heat flux versus the wall superheating for various combinations of the annular gap and the bottom inflow area. Data points for the single tube and the annulus with closed bottoms have been removed from the correlation development. It is not realistic to obtain any general theoretical correlation for heat transfer coefficients in nucleate boiling. This is because the boiling occurs at nucleation sites, and the number of sites is very dependent upon (a) the physical condition and preparation of the surface; and (b) how well the liquid wets the surface and how efficiently the liquid displaces air from the cavities [15]. Moreover, geometric conditions make the situation to be more complicated. Although it contains inherent unidentified uncertain parameters, we continue the development of the correlation nevertheless. This is because the quantification of the experimental results could broaden its applicability to the thermal designs. To take account of effects of the gap size, the heat flux, and the inflow area a simple correlation is sought and, as a result, an empirical correlation has been obtained using present experimental data (containing the Kang's previous data [14]) and the statistical analysis computer program (which uses the least square method as a regression technique) as follows:

$$h_b = 4.954 A_r^{0.342} q''^{0.770} / A_f^{0.281}$$
$$A_r = A_f / A_o \tag{2}$$

In the above equations, the dimensions for h_b, q'', and A_f are kW/m^2 °C, kW/m^2, and mm^2, respectively. The parameter A_r is dimensionless. Apparently the correlations only apply for the testing pressure and parameters shown in Table 1. The above correlation is only valid for boiled water on a smooth stainless steel surface. To confirm the validity of the correlation the statistical analyses on ratios of the measured and the calculated heat transfer coefficients (i.e., $h_{b,exp}/h_{b,cor}$) have been performed. The mean and the standard deviation are 1.0153 and 0.1084, respectively.

A comparison between the heat transfer coefficient from the tests ($h_{b,exp}$) and the calculated value ($h_{b,cor}$) by Eq. (2) is shown in Fig. 8. This figure indicates that the scatter of the present experimental data is between +10% and −10%, with some exceptions, from the fitted curve of Eq. (2). The scatter of the present data is of similar size to that found in other existing pool boiling data. As noted by others [13], there seems to be some inherent randomness in pool boiling due to the uncertainties associated with nucleation site density, physical conditions of the tube surface and others. This fact precludes greater accuracy of both theoretical and empirical correlations for heat transfer coefficients in nucleate boiling.

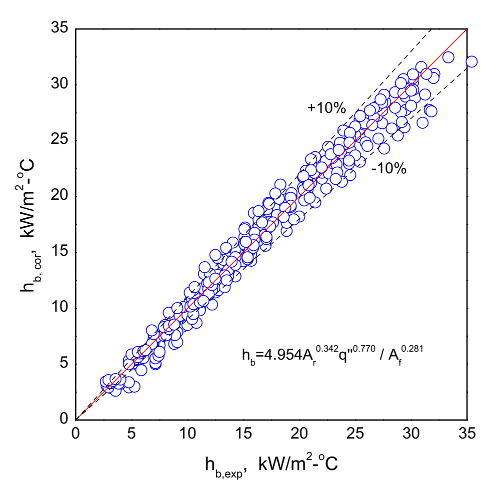


Fig. 8. Comparison of the experimental data to the calculated values.

5. Conclusions

An experimental study has been carried out to identify effects of the bottom inflow area on pool boiling heat transfer in a vertical annulus. To determine the combined effects of major parameters of heat exchanging tubes on the nucleate pool boiling heat transfer, a total of 399 data for heat flux versus wall superheat has been obtained with various combinations of tube diameter (16.5, 25.4, and 34.0 mm) and bottom inflow area (0–1060.3 mm^2). Main conclusions of the present experimental results are as follows:

(1) The bottom inflow area changes heat transfer much and moves the deterioration point of heat transfer coefficients to the higher heat fluxes. The convective flow generated by the bottom inflow and the prevention of the flow interference could be regarded as the major cause.
(2) The difference between heat transfer coefficients for the annuli and the single tube at $q'' = 100\,\mathrm{kW/m^2}$ and $A_r = 0.08$ becomes larger as the gap size is small. That is, when $s = 19.5$ mm only 7.4% (from 17.5 to 18.8 $\mathrm{kW/m^2\,°C}$) increase in h_b is observed whereas more than 42.2% (from 16.6 to 23.6 $\mathrm{kW/m^2\,°C}$) increase in h_b is obtained when $s = 10.7$ mm.
(3) To quantify effects of the inflow area on heat transfer, a new empirical correlation has been developed in terms of the area ratio, inflow area, and the heat flux. The correlation predicts the heat transfer data of boiling within ±10%.

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