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Technical Note

Effects of outer tube length on saturated pool boiling heat transfer in a vertical annulus with closed bottoms

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Abstract

To improve pool boiling heat transfer in an annulus with closed bottoms, the length of an outer tube has been changed between 0.2 m and 0.6 m. For the test, a heated tube of 19.1 mm diameter and water at atmospheric pressure has been used. To elucidate effects of the outer tube length on heat transfer results of the annulus are compared with the data of a single unrestricted tube. The change in the outer tube length results in much variation in heat transfer coefficients. As the outer tube length is 0.2 m the deterioration point of heat transfer coefficients gets moved up to the higher heat fluxes because of the decrease in the intensity of bubble coalescence.

1. Introduction

The mechanism of pool boiling heat transfer has been studied extensively in the past since it is closely related with the thermal design of more efficient heat exchangers [1]. One of the effective means to increase heat transfer is to consider confined geometries. Major geometries concerning about 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 [2,4,5]. According to Kang [4], 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 K. However, increasing ΔT_{sat} more than 2 K the coefficient has almost a same value (i.e. about 20 kW/m^2 K) regardless of the heat flux increase. The cause for the deterioration was suggested as bubble coalescence at the upper regions of the annulus [4]. To apply the vertical annulus with closed bottoms to the thermal design of a heat exchanger investigation of any possible ways to prevent the deterioration is needed in advance.

Up to the author's knowledge, no previous results concerning the ways have been published yet. Therefore, the present study is aimed at the investigation of the way to improve heat transfer in the annulus with closed bottoms through changing the length of the outer tube.

2. Experiments

A schematic view of the present experimental apparatus and a test section is shown in Fig. 1. The water

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Nomenclature

D	heating tube diameter	T_{sat}	saturation temperature
h_b	boiling heat transfer coefficient	T_W	tube wall temperature
I	supplied current	t	time
L	heated tube length	V	supplied voltage
L_o	outer tube length	ΔT_{sat}	tube wall superheating ($=T_W - T_{sat}$)
q''	heat flux		

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. The sizes of the inner tank are 800 × 1000 × 1100 mm (depth × width × height). The inside tank has several flow holes (28 mm in diameter) to allow fluid inflow from the outer tank. Four auxiliary heaters (5 kW/heater) were installed at the space between the inside and the 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 $D = 19.1$ mm). The surface of the tube was finished through buffing process to have smooth surface. Electric power of 220 V AC was supplied through the bottom side of the tube.

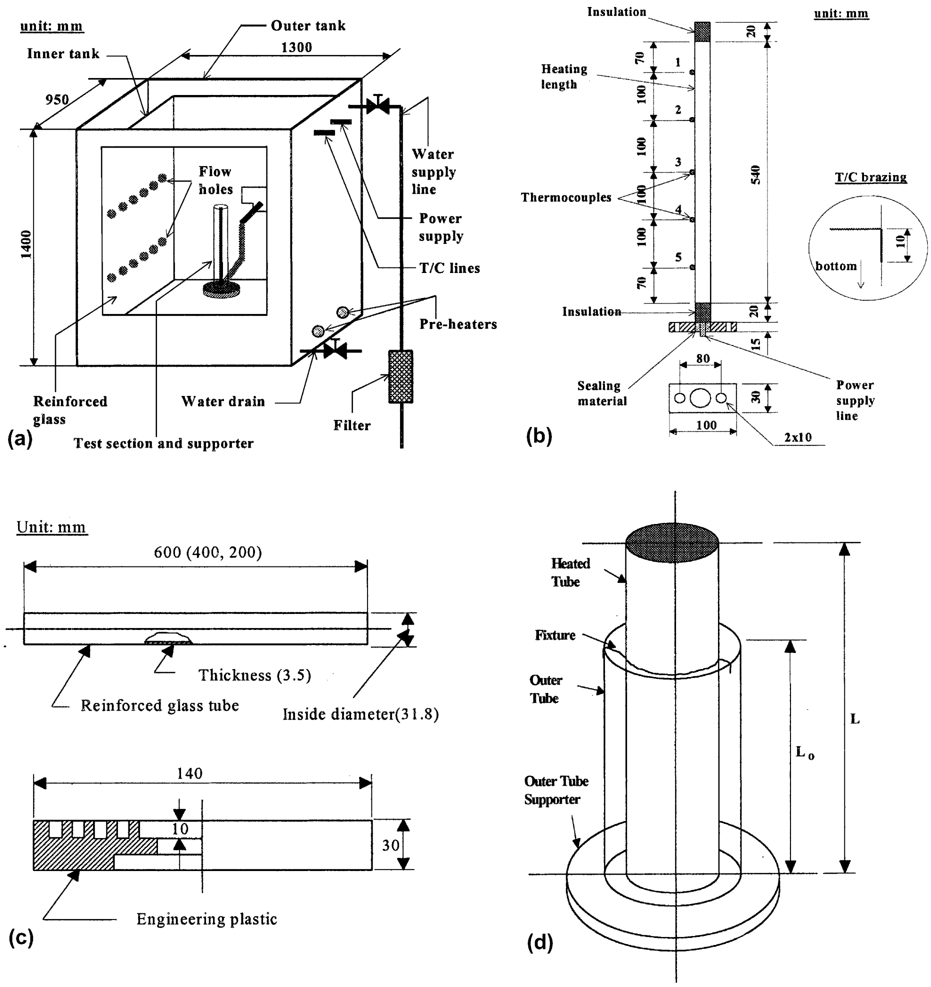

Fig. 1. Schematic diagram of the experimental apparatus.

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 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, a glass tube (gap size = 6.35 mm) of different axial length (i.e., $L_o = 0.2$, 0.4, and 0.6 m) was used. A fixture made of slim wires was inserted into the upper side of the gap to keep the space between the heating tube and 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 (i.e., $T_{sat} = 100$ °C since all the tests are run at atmospheric pressure condition), the water is then boiled for 30 minutes 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. Once a set of experiments has been performed for various heat fluxes at the fixed tube, a series of experiments has been executed for the different outer tube length. The single tube has been tested at first, and then the annulus is tested.

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

The uncertainty in the 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. 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 ±10%.

3. Results and discussion

Fig. 2 shows variations in heat transfer as the length of the outer tube changes. Experimental data for $L_o = 0.2$, 0.4, and 0.6 m are shown in the Figure and are compared with the single unrestricted tube (i.e., $L_o = 0.0$ m). At $q'' \leqslant 60$ kW/m^2, changes in ΔT_{sat} for the annulus are different from the single unrestricted tube. For the annulus, the heat flux is almost a linear function of the superheat and increases gradually as the wall superheat increases from 2 K to 6 K. For the single tube a steeper curve slope of q'' versus ΔT_{sat} is

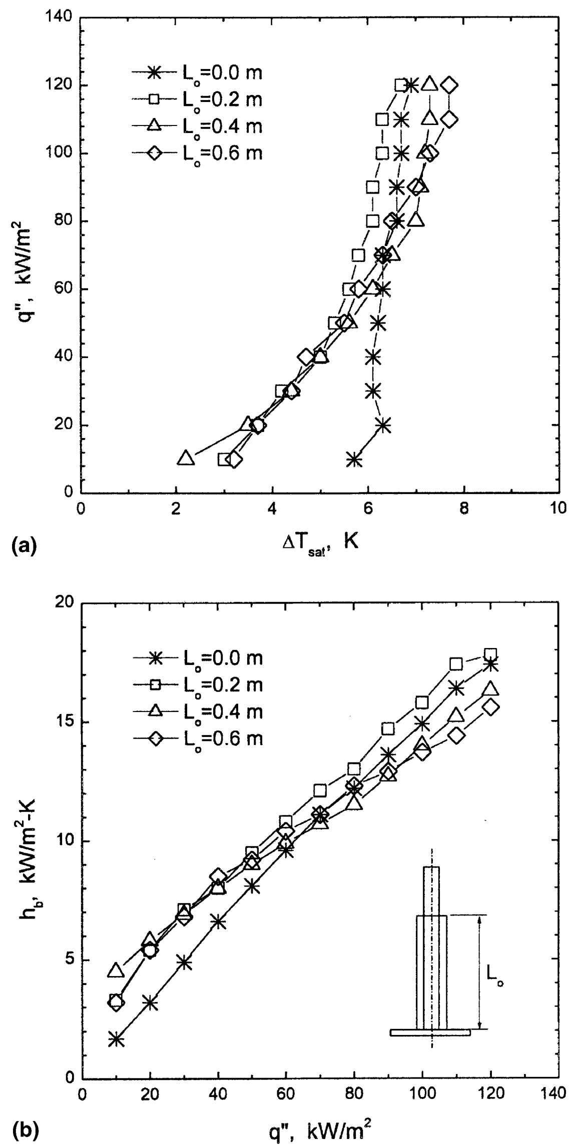

Fig. 2. Plots of experimental data as the outer tube length changes. (a) Curves of q'' versus ΔT_{sat} and (b) curves of h_b versus q''.

observed and small changes in ΔT_{sat} result in much changes in q''. As the length of the outer tube gets shorten the slope of q'' versus ΔT_{sat} curve approaches to the single tube at $q'' > 60\ \mathrm{kW/m^2}$. As $L_o > 0.2$ m the slopes of q'' versus ΔT_{sat} curves are smooth and results in deterioration in the slope of h_b versus q'' curves. At $q'' > 60\ \mathrm{kW/m^2}$ the gradient of h_b versus q'' curves for $L_o = 0.4$, and 0.6 m get decreased and at $q'' \geqslant 70\ \mathrm{kW/m^2}$ h_b for the annulus is less than the single tube. At $q'' < 70\ \mathrm{kW/m^2}$ heat transfer coefficients for the annulus is higher than the single tube. The major mechanisms affecting on the heat transfer changes from liquid agitation to bubble coalescence for the annulus [4]. At lower heat fluxes less than $60\ \mathrm{kW/m^2}$ active liquid agitation in the annular space increases heat transfer. As the heat flux increases the incoming liquid interrupts bubbles escaping through the exit. Therefore, bubbles are coalescing in the space. However, as the outer tube length gets shorten (for the present $L_o = 0.2$ m) active bubble coalescence is not observed and the heat transfer coefficient is not deteriorated. Therefore, the heat transfer coefficients for the annulus with $L_o = 0.2$ m is larger than the single tube because of active liquid agitation at $q'' \leqslant 60$ kW/m and the slope of h_b versus q'' is maintaining at higher heat fluxes because active bubble coalescence is not observed.

Fig. 3 shows some photos of boiling in the annulus as L_o changes. The closed bottom has restricted net-flow of liquid and fluid chugging was observed. Those photos were taken at around the mid-point of the tube length. As shown in the photos no active bubble coalescence is observed at $L_o = 0.2$ m. Bubbles coming

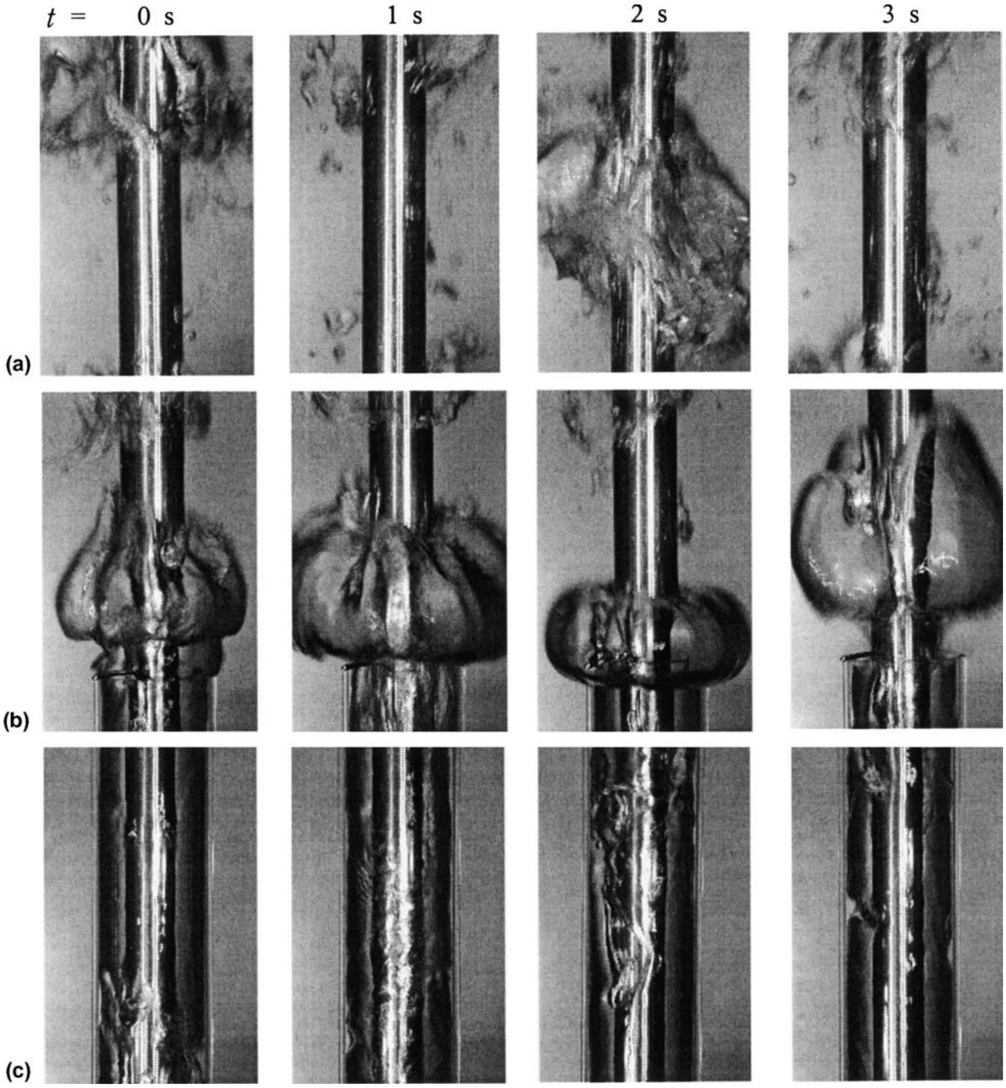

Fig. 3. Photos of boiling on tube surface in the annulus at $q'' = 70\ \mathrm{kW/m^2}$. (a) $L_o = 0.2$ m; (b) $L_o = 0.4$ m; (c) $L_o = 0.6$ m.

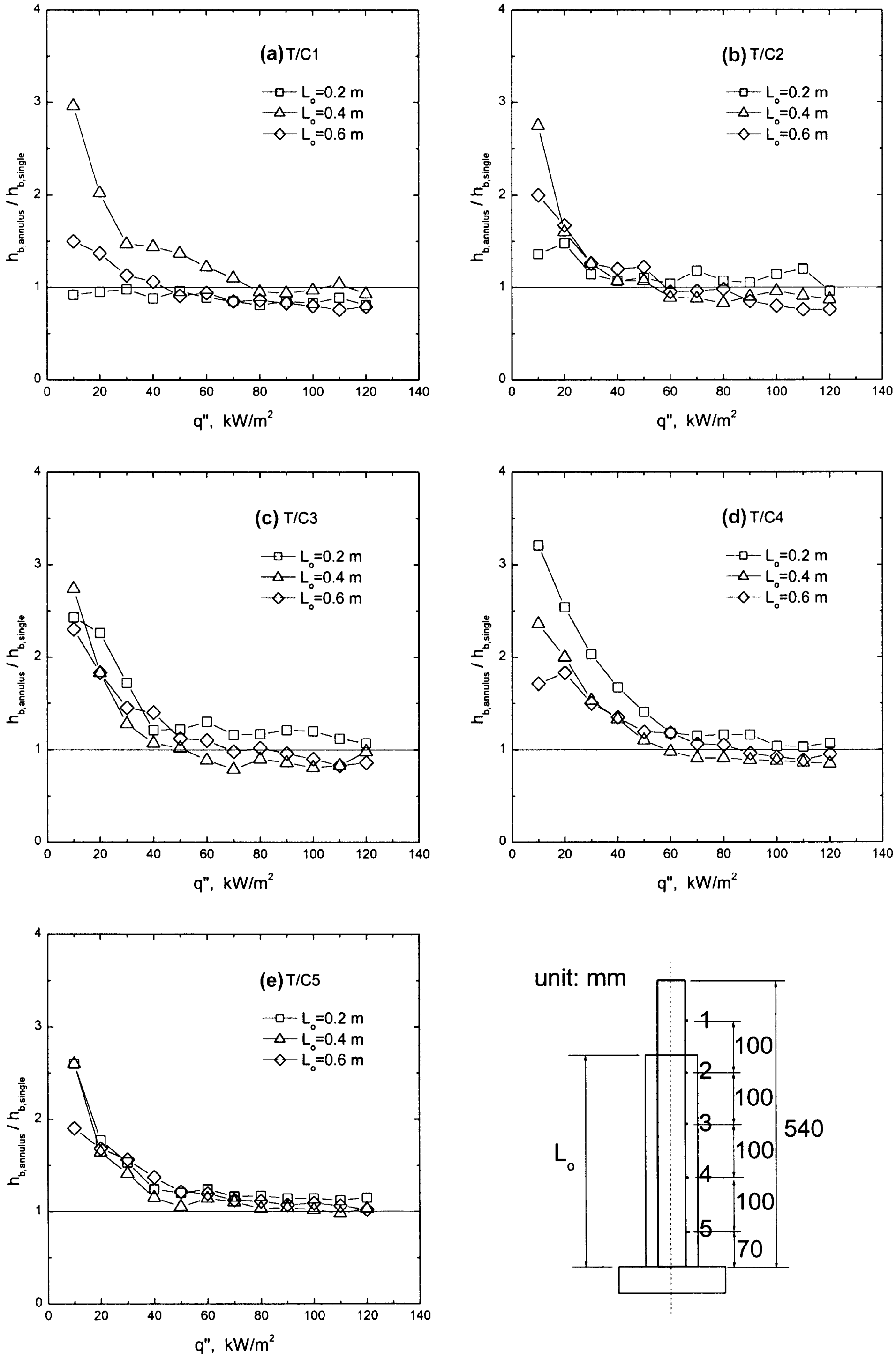

Fig. 4. Ratios of local coefficients in the annulus to those of the unrestricted single tube as the heat flux increases.

from the bottom side generates liquid agitation around the upper regions. As the length changes to 0.4 m tulip type bubbles are observed at the exit. At $t =$ 2 s bubble slugs fulfill the exit and prevent liquid inflow to the annular space. At the consecutive time step bubble slugs are moving to the upper regions and liquid inflows to the space. Same process is also observed at $L_o = 0.6$ m.

Local heat transfer coefficients at the thermocouple locations are shown in Fig. 4. In the figure ratios of $h_{b,annulus}/h_{b,single}$ are shown as the heat flux increases. At T/C1 ratios for $L_o = 0.4$ m has the highest value because of strong liquid agitation. At this location no restriction by the outer tube is existed for $L_o = 0.4$ m. As $L_o = 0.2$ m no enough liquid agitation is shown at this location since the bubbles generated at the lower regions are dispersed at this location. For $L_o = 0.6$ m liquid agitation is active at lower heat fluxes and, then, bubble coalescence gets effective as the heat flux increases. As $L_o = 0.2$ m active liquid agitation is observed at T/C3 and T/C4. Local heat transfer coefficients are higher around the locations nearby the outer tube length where changes of outflow and inflow are observed. At T/C5 the ratios are nearby 1 as the heat flux is higher than 60 kW/m^2. This means that lower liquid agitation at this region. Throughout the tube length, the ratio is larger than 1 (except some data for $L_o = 0.6$ m where bubble coalescence is activated at an early stage) at $q'' < 60$ kW/m^2 where liquid agitation is activated. Ratios for $L_o = 0.4$ and 0.6 m become less than as the heat flux is larger than 60 kW/m^2 at the regions above the T/C4. This means a creation of active bubble coalescence around this region. At $L_o = 0.2$ m the symptom of bubble coalescence is only observed at T/C1.

4. Conclusions

To identify effects of the outer tube length on pool boiling heat transfer in a vertical annulus (gap size = 6.35 mm) with closed bottoms, a heated tube of 19.1 mm diameter and water at atmospheric pressure has been studied experimentally. The change in the length results in much variation in heat transfer coefficients. As the length is 0.2 m a deterioration in heat transfer at higher heat fluxes is not observed while the coefficients is still much higher than the unrestricted single tube at lower heat fluxes. Therefore, the annulus with closed bottoms and a short outer tube length could be recommended as a very useful way to improve pool boiling heat transfer.

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