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Pool boiling heat transfer on tandem tubes in vertical alignment

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ABSTRACT

The combined effects of a tube pitch and the heat flux of a lower tube on saturated pool boiling heat transfer of tandem tubes were investigated experimentally. For the test, two smooth stainless steel tubes of 19 mm diameter and the water at atmospheric pressure were used. The pitch was varied from 28.5 to 114 mm and the heat flux of the lower tube was changed from 0 to 110 kW/m^2. The bundle effect was clearly observed when the heat flux of the lower tube was greater than that of the upper tube and the heat flux of the upper tube is less than 60 kW/m^2. The maximum bundle effect was 3.08 as the tube pitch was 95 mm when the heat fluxes of the upper and the lower tubes were 10 and 90 kW/m^2, respectively. In general, the bundle effect was decreased with increasing the tube pitch. The major heat transfer mechanisms are considered as the convective flow of bubbles, rising from the lower tube, and liquid agitation caused by the bubbles. As a way of quantifying the bundle effects a simple empirical correlation was suggested.

1. Introduction

Pool boiling is closely related to the design of passive type heat exchangers, which have been investigated in nuclear power plants to meet safety functions in case of no power supply [1,2]. Since the space for a heat exchanger install is usually limited, the exact estimation of the heat transfer is very important to keep up a reactor integrity. One of the major issues in the design of a heat exchanger is the heat transfer in a tube bundle. The passive condensers adopted in SWR1000 and APR+ has U-type tubes [1,2]. The heat exchanging tubes are in vertical alignment. For the cases, the upper tube is affected by the lower tube. Therefore, the results for a single tube are not applicable for the tube bundles.

Since heat transfer is closely related to the conditions of tube surface, bundle geometry, and liquid, lots of studies have been carried out for the several decades to investigate the combined effects of those factors on pool boiling heat transfer [3,4]. One of the most important parameters in the analysis of a tube array is the pitch (P) between tubes. Many researchers have been investigating its effect on heat transfer enhancement for the tube bundles [5–7] and the tandem tubes [6,8,9]. The effect of tube array on heat transfer enhancement was also studied for application to the flooded evaporators [10–13]. The upper tube within a tube bundle can significantly increase nucleate boiling heat transfer compared to the lower tubes at moderate heat fluxes. At high heat fluxes these influences disappear and the data merge onto the pool boiling curve of a single tube [13].

Cornwell and Schuller [14] studied the sliding bubbles by high speed photography to account the enhancement of heat transfer observed at the upper tubes of bundles. The study by Memory et al. [15] shows the effects of the enhanced surface and oil adds to the heat transfer of tube bundles. They identified that, for the structured and porous bundles, oil addition leads to a steady decrease in performance. The flow boiling of n-pentane across a horizontal tube bundle was investigated experimentally by Roser et al. [16]. They identified that convective evaporation played a significant part of the total heat transfer. The fouling of the tube bundle under pool boiling was also studied by Malayeri et al. [17]. They identified that the mechanisms of fouling on the middle and top heater substantially differ from those at the bottom heater.

The heat transfer on the upper tube is enhanced compared with the single tube [10]. The enhancement of the heat transfer on the upper tube is estimated by the bundle effect (h_r). It is defined as the ratio of the heat transfer coefficient (h_b) for an upper tube in a bundle with lower tubes activated to that for the same tube activated alone in the bundle [18]. The upper tube within a tube bundle can significantly increase nucleated boiling heat transfer compared to the lower tubes at moderate heat fluxes. At high heat fluxes these influences disappear and the data merge the pool boiling curve of a single tube [13]. It was explained that the major influencing factor is the convective effects due to the fluid velocity and the rising bubbles [4]. For tandem tubes having an equal heat flux, the greatest heat transfer coefficient of the upper tube

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Nomenclature

A_T	data acquisition error, °C	q''_L	heat flux on the lower tube, W/m^2
D	diameter of the heating tube, m	q''_T	heat flux on the upper tube, W/m^2
h_b	boiling heat transfer coefficient, W/m^2-°C	R_a	surface roughness, µm
h_r	bundle effect ($=h_b/h_{b,q''_L=0}$)	T_{sat}	saturation temperature, °C
I	supplied current, A	T_W	tube wall temperature, °C
L	heated tube length, m	V	supplied voltage, V
P	pitch distance, m	ΔT_{sat}	tube wall superheat ($= T_W - T_{sat}$), °C
P_T	precision limit, °C		

decreases [8], increases [9], or negligible [6] with increasing tube pitch in pool boiling. Some possible explanations for the discrepancy are due to the liquid and the difference in geometric conditions of the test section and the pool.

Ribatski et al. [6] performed an experiment with R123 and smooth brass tubes of 19.05 mm outer diameter (D). Through the investigation, the effects of reduced pressure (p_r) and tube pitch were studied. They identified that the effect of tube spacing on the local heat transfer coefficient along the tube array was negligible. At low heat fluxes, the tube positioning shows a remarkable effect on the heat transfer because of the free convection. However, at high heat fluxes the effect of natural convection is negligible and the effects of bubbles become dominant. According to Ribatski et al. [6] the spacing effects on the heat transfer became relevant as the tubes come closer to each other due to bubble confinement between consecutive tubes.

Gupta et al. [8] investigated the effect of tube pitch on pool boiling of two tubes placed one above the other. They used a stainless steel tube of commercial finish having 19.05 mm outer diameter and the distilled water at 1 bar. They found that the heat transfer coefficients of the upper tube were increased as the pitch distance decreased due to the larger number of bubbles intercepted by the upper tube when the pitch was lowered. The authors also investigated three horizontal tubes, which were stacked one above another at a constant pitch distance and got the similar results.

Hahne et al. [10] used finned type copper tubes submerged under R11 at 1 bar. The tube pitch was varied between P/D = 1.05 and 3.0. They found that the largest heat transfer of the upper tube increases with increasing tube pitch. This is expected as the convective flow of bubbles and liquid, rising from the lower tube, enhances the heat transfer on the upper tube. However, as the heat flux increases the heat transfer of the upper tube decreases with increasing tube pitch.

Since the source of the convective flow in pool boiling is the lower heated tube, the heat flux of the lower tube (q''_L) is of interest. Kumar et al. [19] carried an experimental study using the combination of distilled water and two horizontal reentrant cavity copper tubes. They used a fixed spacing and developed a model to predict the heat transfer coefficient of individual tube in a multi-tube row and the bundle heat transfer coefficient. Ustinov et al. [20] investigated effects of the heat flux of the lower tube on pool boiling of the upper tube for the fixed tube pitch. They used microstructure-R134a or FC-3184 combinations and identified that the increase in the heat flux of the lower tube decreased the superheating (ΔT_{sat}) of the upper tube.

Summarizing the previous results it can be stated that heat transfer coefficients are highly dependent on the tube geometry and the heat flux of the lower tube. As shown in Table 1 the aim of the published results for tandem tubes were to investigate the effect of pitch or lower tube heat flux individually. In general, there are many tube arrays where the analysis of the combined effects is necessary. Therefore, the focus of the present study is an identification of the effects of tube pitch as well as the heat flux of the lower tube on the heat transfer.

2. Experiments

A schematic view of the present experimental apparatus is shown in Fig. 1. The water 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). Four auxiliary heaters (5 kW/heater) are installed in the space between the inside and the outside tank bottoms. The heat exchanging tubes are resistance heaters (Fig. 1(b)) made of very smooth stainless steel tubes of 19 mm outside diameter and 400 mm heated length (L). The surface of the tube is finished through a buffing process to have a smooth surface. The value of the surface roughness was measured by a stylus type profiler. The arithmetic mean of all deviations from the center line over the sampling path has the value of R_a = 0.15 µm. Electric power of 220 V AC was supplied through the bottom side of the tube.

For the tests, the assembled test section was in the water tank as shown in Fig. 1(a). The pitch was regulated from 28.5 to 114 mm by adjusting the space between the tubes, which were positioned one above the other and were assembled using bolts and nuts to the supporter. The values of the tube pitches and the heat fluxes of the lower tube are listed in Table 2. The heat flux of the lower tube was (1) set fixed values of 0, 30, 60, and 90 kW/m^2 or (2) varied equal to the heat flux of the upper tube (q''_T). The water tank was filled with water until the initial water level reached 1.1 m; the water was then heated using four pre-heaters at constant power. When the water temperature was reached the saturation value (100 °C since all tests were done at atmospheric pressure), the water was then boiled for 30 min to remove the dissolved air. The temperatures of the tube surfaces (T_W) were measured when they were at steady state while controlling the heat flux on the tube surface with the input power.

The tube outside was instrumented with six T-type sheathed thermocouples (diameter is 1.5 mm). The thermocouple tip (about 10 mm) was brazed on the sides of the tube wall. The brazing metal is a kind of brass and the averaged brazing thickness is less than 0.1 mm. The temperature decrease along the brazing metal was calibrated by the one dimensional conduction equation. Since the thermal conductivity of the brass is nearby 130 W/m-°C at 110 °C [21], the maximum temperature decrease through the brazing metal is 0.08 °C at 110 kW/m^2. The value was calculated by the product of the heat transfer rate and the thermal resistance. The measured temperatures were calibrated considering the above error. The water temperatures were measured with six sheathed T-type thermocouples attached to a stainless steel tube that placed vertically in a corner of the inside tank. To measure and/or control the supplied voltage and current, two power supply systems was used.