International Journal of Heat and Mass Transfer 101 (2016) 445–451

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Pool boiling heat transfer from an inclined tube bundle

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ARTICLE INFO

Article history:
Received 26 January 2016
Received in revised form 23 May 2016
Accepted 23 May 2016

Keywords:
Pool boiling
Heat transfer
Inclination angle
Tube bundle
Advanced reactors

ABSTRACT

The combined effects of an inclination angle and the heat flux of a lower tube on pool boiling heat transfer of a tube bundle were investigated experimentally. For the test, two heated stainless steel tubes of 19 mm diameter and the water at atmospheric pressure were used. The increase of the heat flux of the lower tube and the decrease of the inclination angle increases the heat transfer coefficient of the upper tube. The increase of the heat transfer coefficient is clearly observed when the inclination angle is less than 30° and the heat flux of the lower tube is greater than that of the upper tube. The major reason of the heat transfer enhancement on the upper tube is due to the convective flow and liquid agitation caused by the bubbles departed from the lower tube.

1. Introduction

The mechanism of pool boiling heat transfer has been studied for the several decades since it is related to the design of passive type heat exchangers. These heat exchangers are of importance in nuclear power plants to meet safety functions in case of no power supply [1]. The exact estimation of the heat removal capability of the heat exchangers is essential to keep up the integrity of a nuclear reactor. One of the major issues in the design of a heat exchanger is the heat transfer in a tube bundle.

Lots of studies have been carried out to investigate the combined effects of geometric parameters of a tube bundle on pool boiling heat transfer [2–4]. Table 1 shows a summary of the major parameters investigated. Ribatski et al. [4] identified that the effect of tube spacing on the local heat transfer coefficient along the tube array was negligible. The spacing effects on the heat transfer became relevant as the tubes came closer to each other due to bubble confinement between consecutive tubes [4]. Gupta et al. [5] found that the heat transfer coefficients of the upper tube were increased as the pitch (P) decreased due to the larger number of bubbles intercepted by the upper tube when the pitch was lowered. Hahne et al. [6] found that the largest heat transfer of the upper tube was increased with increasing the pitch of a tube bundle.

Since the source of the convective flow in pool boiling is the tube situated at the lower side of a bundle, the heat flux of the lower tube (q''_L) is of interest. Kumar et al. [7] developed a model to predict the heat transfer coefficient of an individual tube in a multi-tube row and the bundle heat transfer coefficient. Ustinov et al. [8] also investigated effects of the heat flux of the lower tube using microstructure submerged in R134a or FC-3184 and identified that the increase in the heat flux of the lower tube decreased the superheating (ΔT_{sat}) of the upper tube. Along with the tube spacing, its location is also of interest. Kang investigated that the bundle effect was dependent on the elevation angle [9], tube pitch [10], and the heat flux of the lower tube. The bundle effect was clearly observed when q''_L was greater than the heat flux of the upper tube (q''_T) [9]. The bundle effect was increased when the elevation angle was increased [9] and the pitch was decreased [10].

One of the key parameters is the inclination angle (ϕ) of the heated surfaces. According to the published results, it is identified that the effects of the inclination angle on pool boiling are closely related with the geometries [11]. Many researchers had in the past generations investigated the effects of the orientation of a heated surface for the various combinations of geometries and liquids as listed in Table 2 [11–19]. The restriction of a fluid flow is of interest, too [16,19]. El-Genk and Bostanci [12] studied the effects of the inclination angle of a copper specimen shaped as a plate on pool boiling of HFE-7100 for application to the design of an electric chip. Stralen and Sluyter [13] performed a test to find out boiling curves for platinum wires and concluded that the horizontal type was more effective than the vertical type. The major cause of the reduction in heat transfer for the vertical position is due to the formation of large vapor slugs. According to Nishikawa et al. [14], the effect of the surface configuration is remarkable at low heat fluxes. Fujita et al. [16] observed that the effects of the inclination angle are closely related with the gap size. Narayan et al. [18] studied the effect

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2016.05.099
0017-9310/

Nomenclature

D	diameter of the heating tube, m	q''_T	heat flux of 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
q''_L	heat flux of the lower tube, W/m^2	ϕ	inclination angle, °

of nanoparticles, suspended in pure liquids, on nucleate pool boiling heat transfer at various surface orientations. Kang [19] carried out an experimental study to investigate the pool boiling heat transfer in inclined annuli with both open and closed bottoms. Recently, Kang [11] carried out an experiment to investigate the changes in pool boiling heat transfer coefficients on the inside surface of a circular tube.

Summarizing the previous results it can be stated that heat transfer coefficients of a heated specimen are dependent on the heat flux of the lower tube and the inclination angle. As listed in Table 1, most published studies for the tandem tubes were installed horizontally. Therefore, the present study is aimed at the identification of the combined effects of ϕ and q''_L on pool boiling of the upper tube in a tube bundle. The inclination of a bundle can be considered when space for the installation of a heat exchanger is limited. To the present author's knowledge, no results of this effect have as yet been published.

2. Experiments

For the tests, the assembled test section was located in a water tank which had a rectangular cross section (950 × 1300 mm) and a height of 1400 mm as shown in Fig. 1. The sizes of the inner tank are 800 × 1000 × 1100 mm (depth × width × height). Four auxiliary heaters (5 kW/heater) were installed in the space between the inside and the outside tank bottoms. The heat exchanging tube was a resistance heater made of a very smooth stainless steel tube of 19 mm outside diameter (D) and 400 mm heated length (L). Several rows of resistance wires were arrayed uniformly inside the heated tube to supply power to the tube. Both sides of the tube are thermally insulated to prevent any possible heat loss through the ends. Moreover, insulation powder was packed into the space between the tube inside wall and the wires (1) to prevent any possible current flowing to the data acquisition system through the thermocouple lines and (2) to heat the outer stainless steel tube uniformly. The tube was 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.

The inclination angle was changed from 0° to 90° by rotating the assembled test section. The heat flux of the lower tube was (1) set fixed values of 0, 30, 60, and 90 kW/m^2 or (2) varied equally to the heat flux of the upper tube. The water tank was filled with the filtered tap water until the first 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 temperatures measured at the sides can be recommended as the average value of

Table 1
Summary of published results for tandem tubes.

Author	Liquid		Tube			Parameters
	Type	Pressure	Surface	Material	Diameter	
Ribatski et al. [4]	R123	0.023* 0.063*	Smooth	Brass	19.05 mm	– P/D = 1.32, 1.53, 2.0 – ϕ = 0°
Gupta et al. [5]	Distilled water	1 bar	Smooth	Stainless steel	19.05 mm	– P/D = 1.5, 3.0, 4.5, 6.0 – ϕ = 0°
Hahne et al. [6]	R11	1 bar	Finned type (19 fpi, 26 fpi)	Copper	18.74 mm 18.89 mm	– P/D = 1.05, 1.3, 3.0 – ϕ = 0°
Kumar et al. [7]	Distilled water	35.36–97.5 kPa	Finned type (reentrant cavity)	Copper	32 mm	– P/D = 0.5 – q''_L=19–45 kW/m^2 – ϕ = 0°
Ustinov et al. [8]	R134a FC-3284	1.5–9 bar	Microstructure	Copper	18 mm	– P/D = 1.5 – q''_L = 5–125 kW/m^2 – ϕ = 0°
Kang [9]	Boiled water	1 bar	Smooth	Stainless steel	19 mm	– P/D = 1.5 – q''_L = 0–120 kW/m^2 – ϕ = 0° – θ = 0–90°
Kang [10]	Boiled water	1 bar	Smooth	Stainless steel	19 mm	– P/D = 1.5 ~ 6.0 – q''_L = 30–90 kW/m^2 – ϕ = 0°

* Reduced pressure.