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Nucleate pool boiling characteristics from coated tube bundles in saturated R-134a

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Abstract

Nucleate pool boiling heat transfer from plasma coated copper tube bundles with porous copper (Cu) immersed in saturated R-134a was experimentally studied. The bundle is composed of 15 tubes (of which the number of heated/instrumented tubes was varied) arranged in four different configurations with a pitch-to-diameter ratio of 1.5. The influences of various parameters, for instance, bundle arrangements and heat flux were clarified. Tests were conducted with both increasing and decreasing the heat flux. The data presented indicated that at low heat fluxes, the vertical-in-line tube bundles have the highest bundle factor. A configuration factor was proposed which can be used to characterize the geometric arrangements of the bundles.

1. Introduction

Many water chillers of the centrifugal type have evaporators utilizing a flood type of operation with water circulated through the tube and the refrigerant evaporated on the shellside of the tubes in nucleate pool boiling. The evaporator tube bundle is fully wetted at full-loaded operating conditions. Most of previous research on nucleate pool boiling heat transfer was directed at the clarification of characteristics for a single tube on plate as well as the multi-rods results. Even though, neither sufficient experimental data nor reliable methods are available for the prediction of nucleate pool boiling in tube bundles especially for enhanced tube bundles despite its important application to process heat exchangers.

Early research on heat transfer coefficients in large multi-tube bundles was performed by Palen and co-workers [1–3]. In these studies, only tube bundle average heat transfer coefficients were obtained, but those coefficients were greater than those for a single tube in saturated pool boiling.

Significant progress has been made in understanding nucleate boiling heat transfer and two-phase convection effects on the shellsides of smooth tube bundles, by Leong and Cornwell [4], Cornwell and Schuller [5], and Jensen and Hsu [6]. In their work, either using a photographic technique to study the circulation effects in a slice of a reboiler tube bundle and high-speed photography to observe numerous bubbles sliding over and around the top tubes or the influence of two-phase convection effects in a bundle was examined, the influence of tube position within a bundle of smooth tubes has been extensively studied by Fujita et al. [7], and by Muller [8] for finned tube bundles. An experimental investigation was performed by Jensen et al. [9] to examine the effects of crossflow on the heat transfer coefficients in a tube bundle with smooth and Wolverine Turbo-B and Linde High Flux enhanced tubes. Recently, Marto and Anderson [10] made measurements for R-113 nucleate pool boiling for a bundle of 15 electrically heated, smooth copper tubes arranged in an equilateral triangular pitch with a $p/d = 2$, and higher heat transfer coefficients of the upper tubes were found. Later, Memory et al. [11] conducted almost the similar work except that a Turbo-B tube bundle was used. Significant

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Nomenclature

A	surface area, m^2
d	tube diameter, mm
h	heat transfer coefficient, W/m^2 K
h_{fg}	latent heat, kJ/kg
k	thermal conductivity, W/m K
p	tube pitch, mm
Q	heat flow rate, W
q	heat flux, W/m^2
Ra	surface roughness, μm
ΔT	temperature difference, K
ΔT_{sat}	degree of superheat, K
T	temperature, K

Greek symbols

δ	porous layer thickness, μm
ε	porosity, %
ϵ	average pore diameter, μm
η	bundle factor
ξ_{kj}	configuration factor, η_k/η_j

Subscripts

avg	average
iso	isolated single
sat	saturation
v	vapor

Superscript

–	bundle average

enhancement (up to 4.6 at a low heat flux) was found. Li and Hahne [12] experimentally studied the boiling heat transfer on finned tube bundle with lower tubes heated with constant heat flux in R-11 at a pressure of 1 bar. The heat transfer coefficient with a given pitch-to-diameter ratio were increased as much as 150% and the time-averaged liquid velocity under the bubble can be reached to maximum as the heat increases.

There has been limited work on plasma coated tube bundles [13]. For a porous coated surface, Fujita et al. [7] and Czikk et al. [14] found average bundle data agreed with single-tube data and a bundle factor about one over a wide range of heat fluxes.

It is apparent that the two-phase interactions that occur in tube bundles during boiling may be much complicated and can be changed with tube bundle arrangements, heat flux levels, refrigerants, etc. [10]. The objective of this paper is to provide a comprehensive nucleate pool boiling database for R-134a from smooth and plasma coated surfaces with triangular and square pitch tube bundles as well as to shed further light on the mechanisms which affect bundle heat transfer. Of particular interest is the information regarding the influence of the lower tubes on upper tubes and the influence of a plasma coated surface on bundle performance.

2. Experimental facility

2.1. Experimental setup

The basic experimental apparatus used in the study consisted of an evaporator and condenser arranged to provide reflux operation, as shown in Fig. 1. A rectangular vessel, 370 mm in width, 650 mm in height, and 370 mm in depth, with an electrically heated multi-tube bundle was used to simulate a portion of a refrigerant-flooded evaporator. R-134a was used as the working fluid. All the bundle tubes were made from commercially available, 20 mm diameter smooth copper tubing with outer porous copper coated. The tubes were cantilever mounted from the back wall of the evaporator to permit observation of the boiling phenomena along the axis of tubes through two glass windows mounted in the rear and side of Fig. 2. The tubes in the bundle were roughly arranged in four different configurations (triangular, rectangular, horizontal-in-line, and vertical-in-line) with a definite pitch depicted in Table 1. There are two types of tubes, heated and instrumented and dummy tube (see Table 1 for details).

Fig. 2 is a schematic sectional view of the test section that shows the sets of tubes. The heated tubes contained 1 kW electrical cartridge heaters, 11.1 mm o.d. with a heated length of 220 mm. Two auxiliary heaters, each capable of 1.5 kW, were installed on the front side of the test bundle to maintain the liquid pool at saturated conditions and to provide system pressure control. Also, the heated tubes were instrumented with thermocouples at four circumferential positions in the middle for surface temperature measurements. Four slots were machined into the sheath as shown in Fig. 3. The slots extended from one end to about 1/3 length of the heater exposed junction stainless steel sheath (1.5 mm o.d.), four type T thermocouples (1.1 mm o.d.) were placed into the slots to measure the circumferential temperatures (at four equidistant locations along the periphery). Each slot was then press fitted with an aluminum filler piece welded, and machined to restore a smooth surface. The power to the heaters was supplied by a 110 V, 30 A, DC power supply.

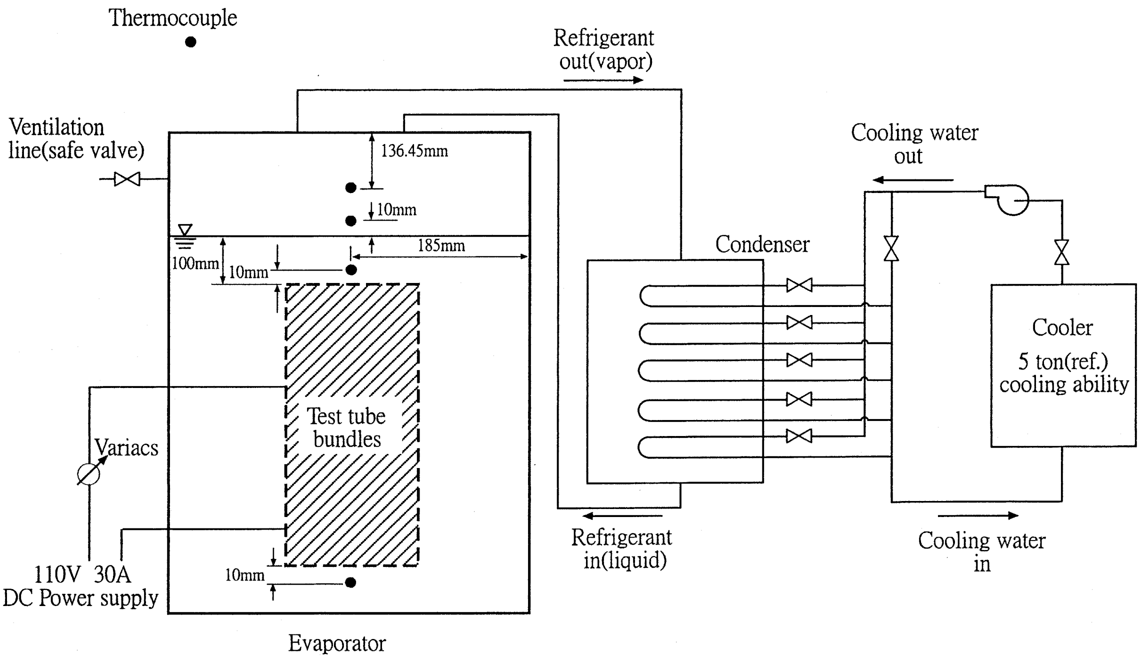


Fig. 1. Experimental apparatus.

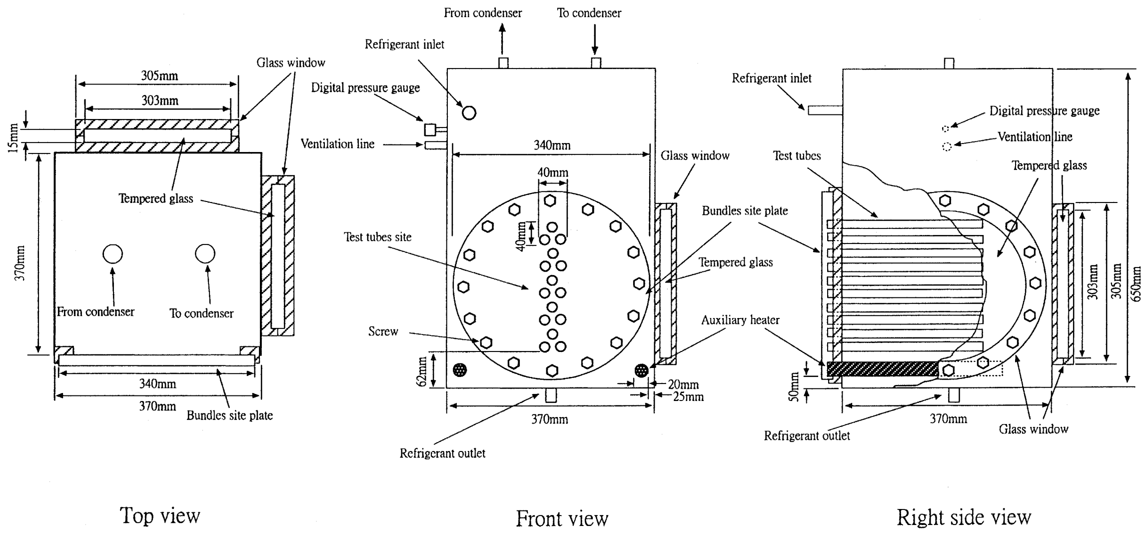


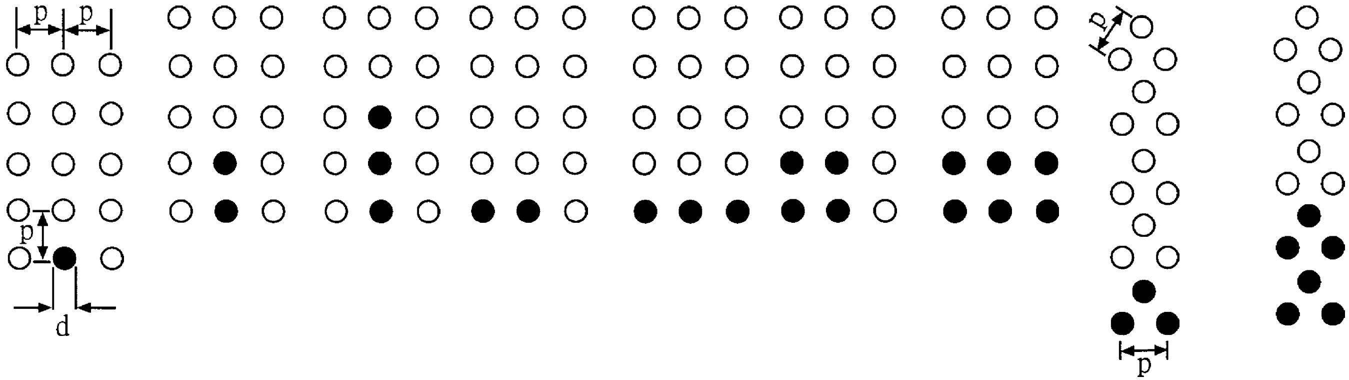
Fig. 2. Schematic sectional view of the test section.

2.2. Test section and conditions

Experiments reported in the present paper were performed using R-134a as the boiling liquid. R-134a has a boiling point of 18 °C at the pressure of 537.06 kPa. Data were obtained by both increasing and decreasing heat flux in steps and taking measurements when the steady state was reached. A steady state was determined when the temperature variations between the adjacent two tubes/or for the single tube is less than ±0.2 °C for 2 min.

In order to study the bundle effects on the boiling characteristics systematically, experiments were performed on (i) one, two, and three vertical-in-line tubes; (ii) one, two and three horizontal-in-line tubes; (iii) two rectangular type 2×2 and 2×3 array; and (iv)

Table 1
The arrangement of tube bundles

Tubes form	Group								
	(i) Vertical-in-line			(ii) Horizontal-in-line		(iii) Rectangular		(iv) Triangular	
Test tube Dummy tube d: tube diameter (mm) p: pitch (centerline to centerline spacing)									
Geometry of the unit									
Pitch (mm)	30	30	30	30	30	30	30	30	30
Tube diameter (mm)	20	20	20	20	20	20	20	20	20
Pitch to diameter ratio	1	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Bundle factor	η_{v1}	η_{v2}	η_{v3}	η_{h1}	η_{h2}	η_{r1}	η_{r2}	η_{t1}	η_{t2}
Configuration factor	–	$\xi_{2,1} = \frac{\eta_{v2}}{\eta_{v1}}$	$\xi_{3,1} = \frac{\eta_{v3}}{\eta_{v1}}$	–	$\xi_{2,1} = \frac{\eta_{h2}}{\eta_{h1}}$	–	$\xi_{2,1} = \frac{\eta_{r2}}{\eta_{r1}}$	–	$\xi_{2,1} = \frac{\eta_{t2}}{\eta_{t1}}$

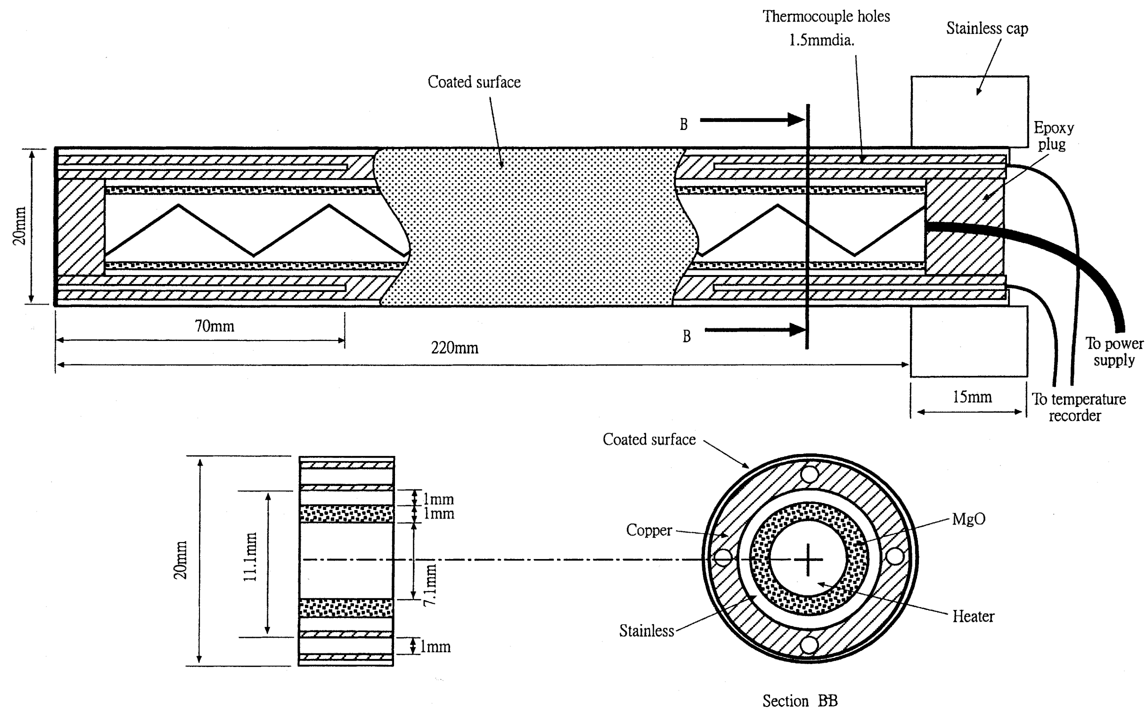


Fig. 3. Schematic sectional view of the heated tube with thermocouple position.

triangular type (three tube bundle and six tube bundle). The details for bundle arrangements with associated parameters are listed in Table 1. The nominal ranges of test conditions were as follows:

Heat flux	0.1–30 kW/m^2
Pressure	537.06 kPa
Types of tube bundle arrangement	Four types; vertical-in-line, horizontal-in-line, rectangular, and triangular
Total number of tubes in a bundle	15 tubes
Heated tube	1–6 tubes

2.3. Experimental procedure

The enhanced surface were prepared by the method described by Hsieh and Yang [15]. The resultant surface condition is listed in Table 2. Prepared test sections were cleaned with chlorinol and water and finally, with acetone. The tank was cleaned with acetone before each run. Once the evaporator tube bundle had been installed, the system was evacuated to a pressure of about 30 Pa. If no leaks were detected over a 24 h interval, the evaporator was charged with the working fluid from a reservoir to a level of 60 mm above the top of tube bundle. This resulted in a vapor pressure of 537.06 kPa (R-1134a).

Power was applied to the pool to degas the test fluids, R-134a at heat flux of 30 kW/m^2 for 1.5 h. The saturation temperature at the measured pressure was compared to the pool temperature measured by the thermocouple. The power supplied to the test section was gradually, and slowly, reduced to zero. The test pool was maintained close to the saturation temperature with an auxiliary heater for about 40 min; then it was switched off to minimize convective effects. The heating power supplied

Table 2
The specification and dimensions of treated surfaces

Tube no. (designated symbol)	Surface (coated material)	Thickness of porous layer δ (μm)	Surface roughness Ra (μm)	Porosity ε	Mean pore diameter ϵ (μm)
Tube 1 (S)	Smooth	–	0.07	–	–
Tube 2 (Cu)	Cu	100	7.84	0.057	4

to the test section was slowly and gradually increased to nearly 30 kW/m^2.

The saturation temperature was kept near 18 °C for R-134a supplied by ICI during all the tests. The liquid level was kept approximately 100 mm above the test tube. All the data were obtained and reduced with a computer-controlled data acquisition system.

2.4. Precautions taken during experiments

1. The pool temperature was compared to the saturation temperature corresponding to the measured saturation pressure for refrigerants. This ensures that there are no noncondensibles in the container. It also verifies that there is no subcooling in the liquid pool within ±0.2 K.
2. To ensure that the correct wall temperature was measured, a tightly pressed thermocouple was put onto the wall of a sleeve insert with thermal jointing compound applied to the tube. In addition, a three-dimensional heat transfer model was employed to correct the wall temperature measured (i.e., to minimize the conductance and capacitance effect) to obtain a more accurate (or nearly true) wall temperature. Consequently, 0.1 K accuracy was expected.
3. The heater was tested for circumferential uniformity of heat flux. Nonuniformities in the heat flux were smoothed out followed the methodology described in [15].

3. Data reduction and uncertainty analysis

For each power input, the heat transfer coefficient was calculated on the basis of bulk fluid saturation temperature (T_{sat}), tube heat flux, and the average (T_{avg}) of the four tube wall temperatures for each instrumented tube. The heat transfer coefficient at each power input was then calculated as $h = Q/[A(T_{avg} - T_{sat})]$, where A is the heated area of the tube.

Using the method of Kline and McClintock [16], uncertainty estimates were made to consider the errors of the instruments, the measurement variance, geometry uncertainty, and calibration errors for the heat flux and temperature measurements. The uncertainty in the wall superheat was dominated by the wall temperature measurements. The values of the four wall temperatures were recorded and compared for examining the variations caused either by nonuniformities in the cartridge heater or by the test tube soldering and assembly procedure. Wall superheat uncertainty can be attributed primarily to thermocouple calibration (±0.1 K) and temperature correction from the thermocouple reading to the reference surface. The maximum variation of the four measured wall temperatures was ±0.3 K at the maximum heat flux ($\cong$30 kW/m^2). The uncertainty in the saturation temperature was estimated to be less than ±0.1 K.

Substrate conduction heat losses were quantified at different heat flux conditions by solving three-dimensional conduction problems with a finite-difference solver. This loss varied from 10.2% to 0.2% for heat flux conditions varied from 0.8 to 30 kW/m^2, respectively. The other primary contributor to heat flux uncertainty was heated surface area. Combining all these effects led to an overall uncertainty estimated in heat flux of 11.2% at the lowest heat input. It indicates the uncertainty of the wall heat transfer coefficient to be about ±15% at $q = 0.8$ kW/m^2.

4. Results and discussion

It is long recognized that the boiling characteristics of a tube bundle are different from that of an isolated single tube, and the heat transfer coefficient at the top of the bundle was found to be significantly higher than that at bottom due to the liquid circulation and turbulence caused by bubbles rising from the lower tubes in the bundle. It is also known that in pool boiling from a coated tube bundle, natural convection regime, intermediate (partial nucleate boiling) regime, and fully developed nucleate boiling regime can be identified based on the heat transfer performance. Following Memory et al. [17], a general observation that should be mentioned was the fluid oscillation/or induced vibration that consistently occurred within each bundle at heat fluxes $\geqslant 10$ kW/m^2. This can clearly be observed as a periodic side-to-side movement of the bubbly mixture as it flowed up through the bundle. Visualization indicates that at the interspace for $p/d = 1.5$, there are filled with vapor in the fully developed nucleate boiling regime. However, in the partial nucleate boiling, several tube interspaces remain without bubble generation, which indicates the effect of convection appears still dominant for smooth tubes. For plasma coated tubes, it provides very small surface openings, and the dominant heat transfer processes in nucleate boiling occur inside the cavities and internal channels even at low heat fluxes. It is found about a factor of two/or three increase in the heat flux at a given temperature difference was obtained.

4.1. Boiling characteristics

Fig. 4 shows the nucleate boiling data for the vertical-in-line arrangement with/without (plasma) coated (tube surface) in a tube bundle. It consists of one, two, and three test tubes, in Fig. 4(a)–(c), respectively. Also are

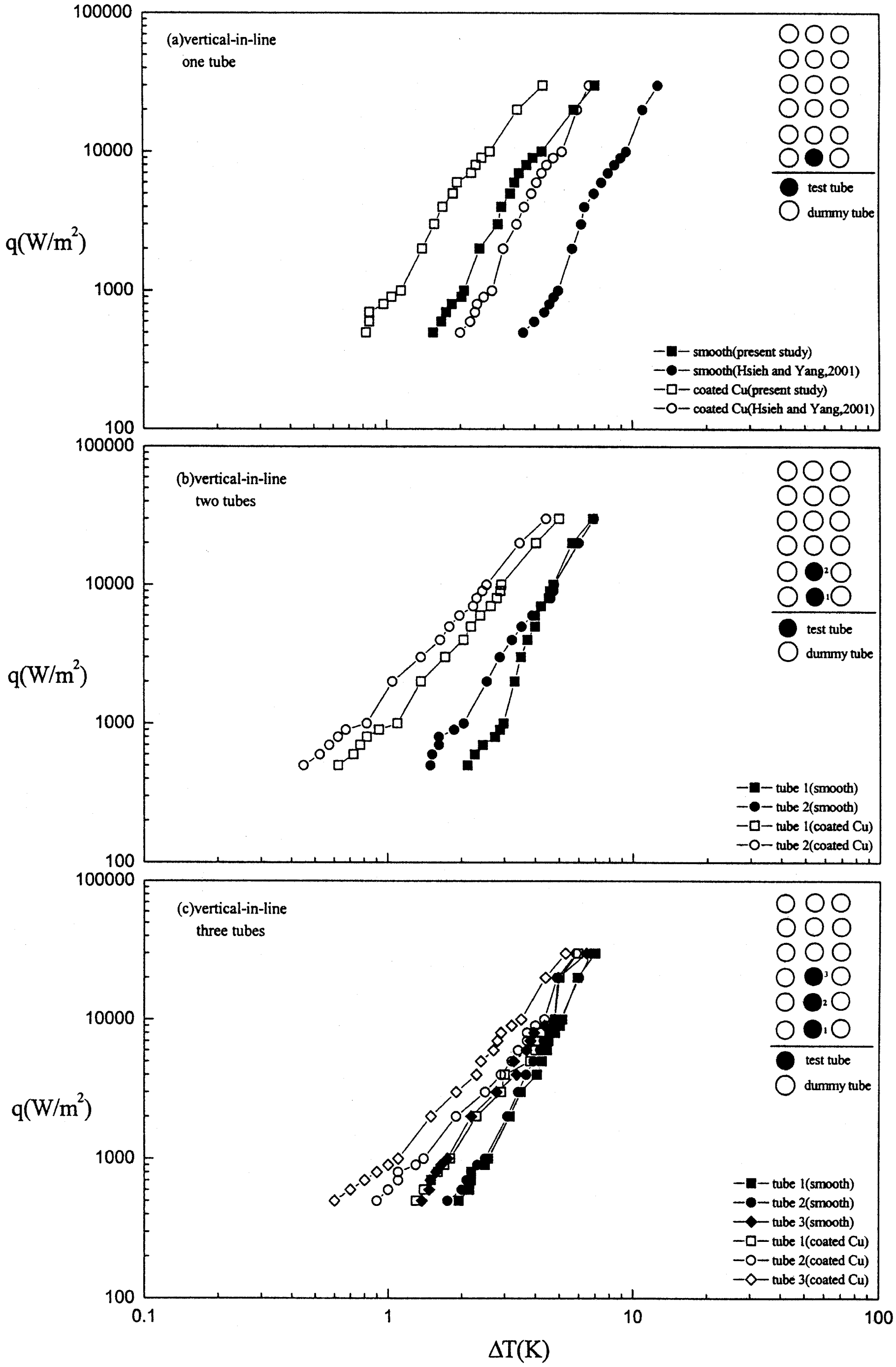


Fig. 4. Boiling curve of tube bundles (vertical-in-line).

the isolated tube pool boiling data from [15] also included in Fig. 4(a) for comparison. The difference is obviously because the latter data is for an isolated single tube. Due to the effect of strong or local convection felt in the bundle and also due to the restriction of the flow around other tubes, the present data show a much higher heat transfer performance. This has also been observed by Memory et al. [17]. For instance, for a coated tube, the degree of superheat, ΔT can be as lower as 1.3 K; while for an isolated single smooth tube $\Delta T \cong 5$ K. For the three vertical-in-line tests, the incipient boiling heat flux for tube 1 in Fig. 4(a) was found to be about 1000 W/m^2; while in Fig. 4(b), it decreased to about 800 W/m^2 with tubes 1 and 2 to be heated. The reduction seems not as large as that of Marto and Anderson [10]. This situation was rebuilt with three tubes heated. As Fig. 4(a) stands for one tube in vertical-in-line, it suggests that the boiling from a lower tube enhances the performance of an upper tube, the bubbles from lower tube impacting upon the surface of the upper tube can reduce the incipient boiling heat flux.

It can be seen also from Fig. 4 that the heat transfer from the instrumented tube is strongly enhanced by the strong two-phase flow induced from the tubes of upper five rows and two adjacents unheated. Both increasing and decreasing heat transfer rate mode were recorded but only the decreasing mode was presented here. The present boiling curves for tube bundles with/without plasma coated surfaces are experienced with large-scale boiling curve hysteresis ($\cong$3 K), an effect which does not appear to have been reported previously in the literature. This behavior becomes less noted for two tubes and three tubes operated simultaneously as evidenced by Fig. 4(b) and (c), due to the temperature gradient associated with the dummy tube so that the flow blockage would be occurred. The heat transfer enhancement compared to the isolated tube was found to be large for all the three cases considered in Fig. 4 due to the bulk upward movement of the fluid and circulation as well as turbulent effect produced by rising vapor bubbles. It can be noted also by comparing the degree of superheat. For tube 1 (coated Cu), the degree of superheat for the onset of nucleate boiling for these three cases is 1.6, 1 (average) and 0.8 (average) K, respectively. Moreover, for two tubes configuration, heat transfer enhancement for tube 2 is higher than that of tube 1. As expected, for three tubes, tube 3 will be the best and tube 1 would be the least in heat transfer performance. But, the enhancement would become less distinct as can be seen from Fig. 4(c) as the number of heated tubes in the bundle increases. This may be contributed to the influence of the closely spaced tubes on the velocity and thermal fields in the wake region of a heated tube. The upper tubes feel a warmer fluid due to the heated tubes below, the warmer fluid decreases the local thermal driving potential. On the contrary, the upper tubes are exposed to a moving fluid as to enhance the boiling heat transfer. The counter-balance effect is responsible for the preceding results.

The difference in heat transfer performance for tubes 1 and 2 cannot clearly noted in Fig. 5(a) for both smooth and coated surfaces in horizontal-in-line arrangement. The situation preserves for three coated tubes as shown in Fig. 5(b), the tube positioned at the middle has the least performance, while tube 3 has the best performance. The reason is still not understood. Further study may include this aspect. For rectangular type, with 2×2 arrangement as shown in Fig. 5(c), the upper row has a better heat transfer performance (e.g. tubes 3 and 4 in Fig. 5(c)); while for 3×3 arrangement, such behavior seems not observed.

The above-stated situation becomes more dominant for triangular arrangement of heated tube as shown in Fig. 6(a) and (b) for three tubes and six tubes arranged in staggered configuration. In fact the various curves merge in fully developed nucleate boiling.

While these results are consistent with those reported previously for tube bundle [9], it is not clear what the plasma coated surfaces heat transfer mechanisms are and how they interact. In addition, most previous bundle research have used R-113 as the working fluid. Quite few data were reported for nonCFC refrigerants like R-134a.

The present flow pattern can be considered as boiling in bubbly flow regime which is applicable to low quality and low heat flux. Mechanistic models, such as suggested by Cornwell [18] have been reported and can be applied to the present experiments with slight modifications. There are three parts of heat transfer coefficient for the present study of which it include h_{nc}, the heat transfer coefficient due to liquid natural convection at the local velocity, the sliding bubble contribution h_{sb}, and h_{nb} relates to bubbles nucleated and grew in pool boiling. This indicates that the present heat transfer mechanisms are due to liquid convection and the influence of the sliding bubbles in the middle and upper tubes for plasma coated tubes, nucleation occurs only on the surfaces of the lowest tubes.

4.2. Heat transfer coefficient

The heat transfer coefficients corresponding to Figs. 4–6 were plotted in Figs. 7–9, respectively. Like boiling curves, the heat transfer coefficient curves again exhibit the previous findings, the slope of these curves is about 0.7 (i.e. $h \sim q^{0.7}$) for fully developed nucleate boiling. As to be mentioned, there are three regimes in pool boiling from Fig. 7. For instance, in Fig. 7(a) the region between natural convection and incipient boiling was found at $q = 1000$ W/m^2; while a region of partial nucleate boiling was noted with the single tube heated at $q > 1000$ W/m^2. For two and three tubes as shown in Fig. 7(b) and (c), the traditional natural convection, partial nucleate boiling and fully nucleate boiling regimes can be identified. The highest enhancement of the upper tube was found to be about 25% for coated and 33% for smooth

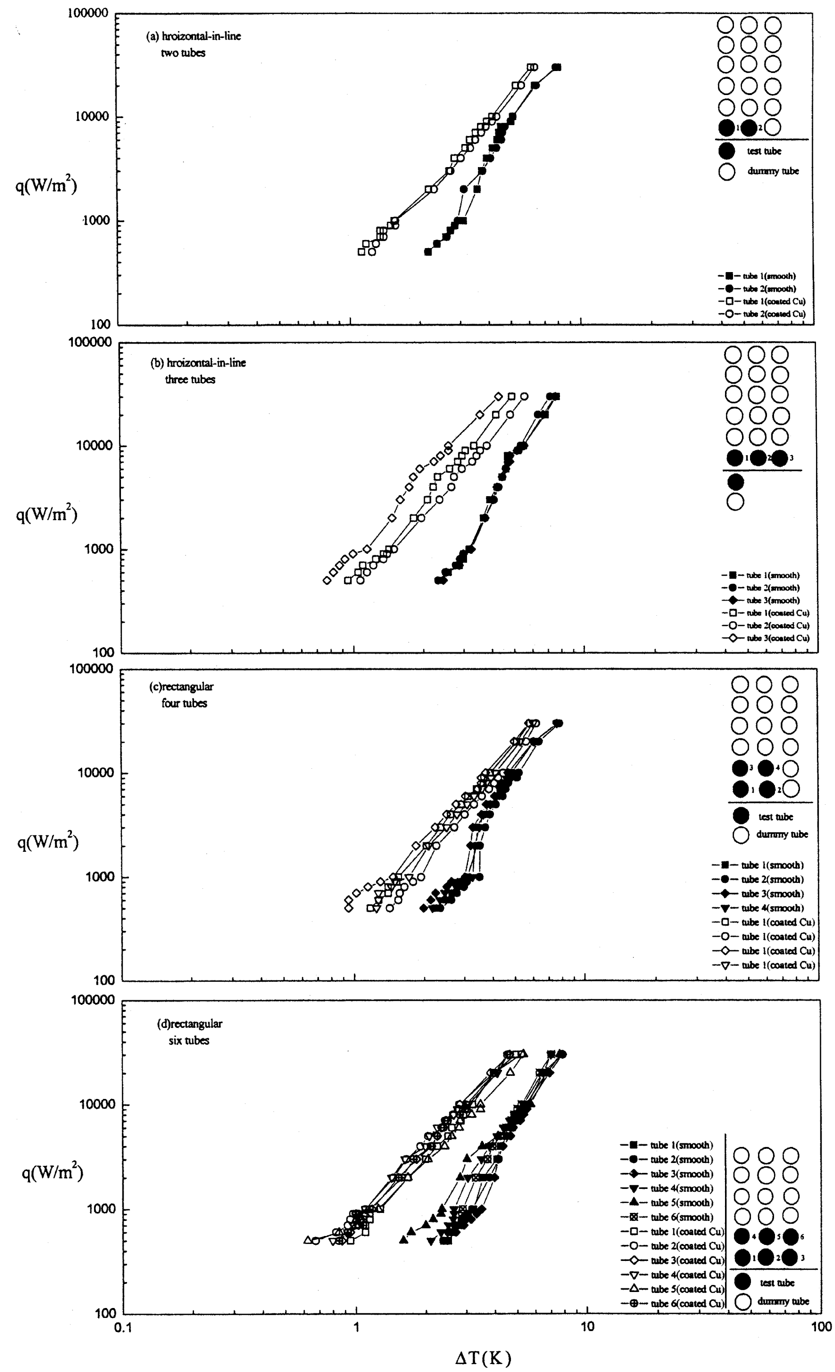


Fig. 5. Boiling curve of tube bundles (horizontal-in-line/rectangular).

tubes at $q \cong 1000$ W/m^2, respectively. As heat flux increases, such enhancement decreases as shown in Fig. 7(b). The same behavior happens for horizontal-in-line and staggered type arrangements. Also included in Fig. 7(b) are the results from [19] for comparison. The heat transfer enhancement can directly and clearly be seen for coated tubes and two/or three tubes arrangements. Moreover, it is obvious from Fig. 9, that the bundle effects are limited to the lower heat flux region. At high heat fluxes, the data merge together forming almost a

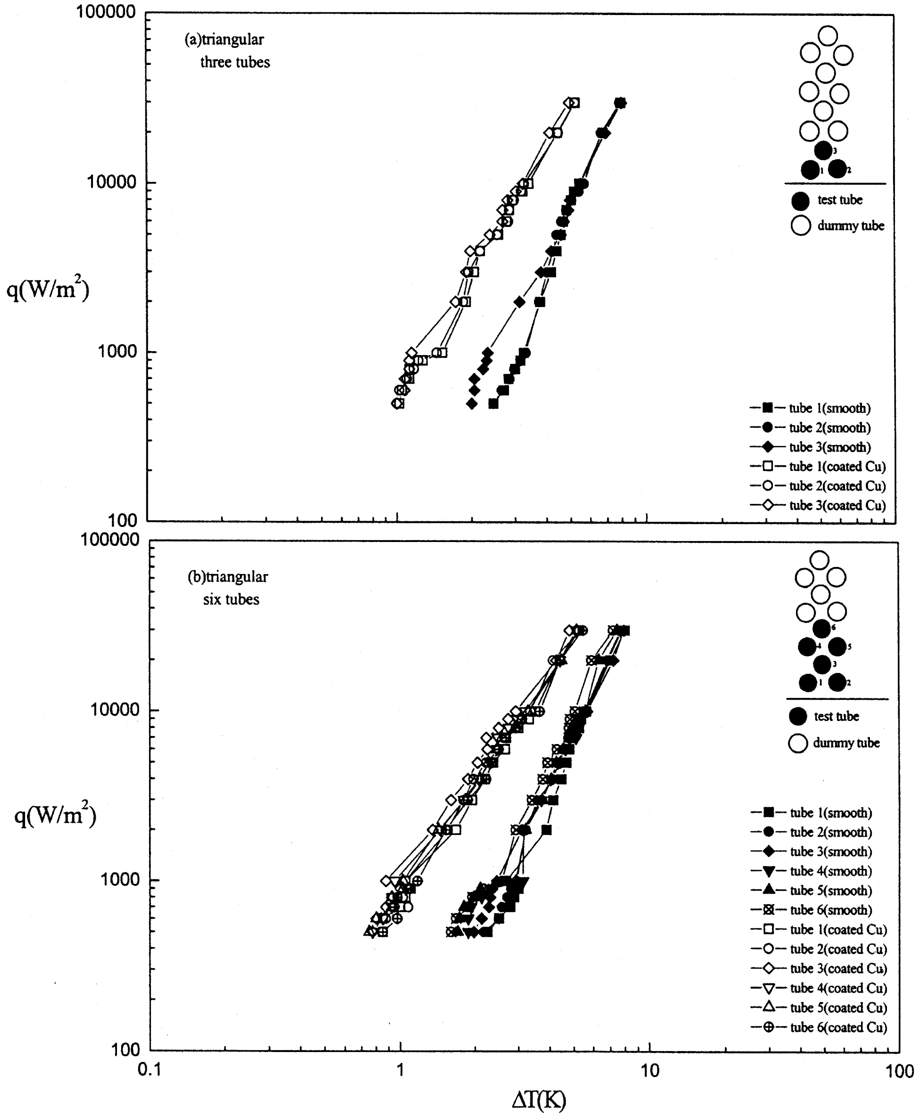


Fig. 6. Boiling curve of tube bundles (triangular).

single relationship between the heat transfer coefficient and heat flux.

Generally speaking, the heat transfer curves for the lower tube (tube 1) in two/and three tube arrangements show that there is an increase in heat transfer due to the increased agitation by a recirculatory flow. While for the upper tube (tube 2), the increase in heat transfer is due to the effect of approaching flow. Higher heat transfer are found for both natural convection and partial nucleate boiling regimes. For three types arrangements, the vertical-in-line and staggered type configuration show their superior in heat transfer performance, the present staggered type has the best heat transfer performance at $q > 3$ kW/m^2; while for $q \leqslant 3$ kW/m^2 it suggests that vertical-in-line type has a better heat transfer. Based on Browne and Bansal [20], the two main factors affecting the heat transfer performance in these configurations are (i) convection effects due to fluid velocity and rising bubbles; and (ii) the effects of static head while causes increased saturation temperatures in the lower part of the tube bundle and hence reduce the local driving temperature difference.

In summary, for the cases studied, the average bundle heat transfer coefficient, ($\overline{h}$, defined later) with q can be correlated in Table 3 if $q \geqslant 1$ kW/m^2).

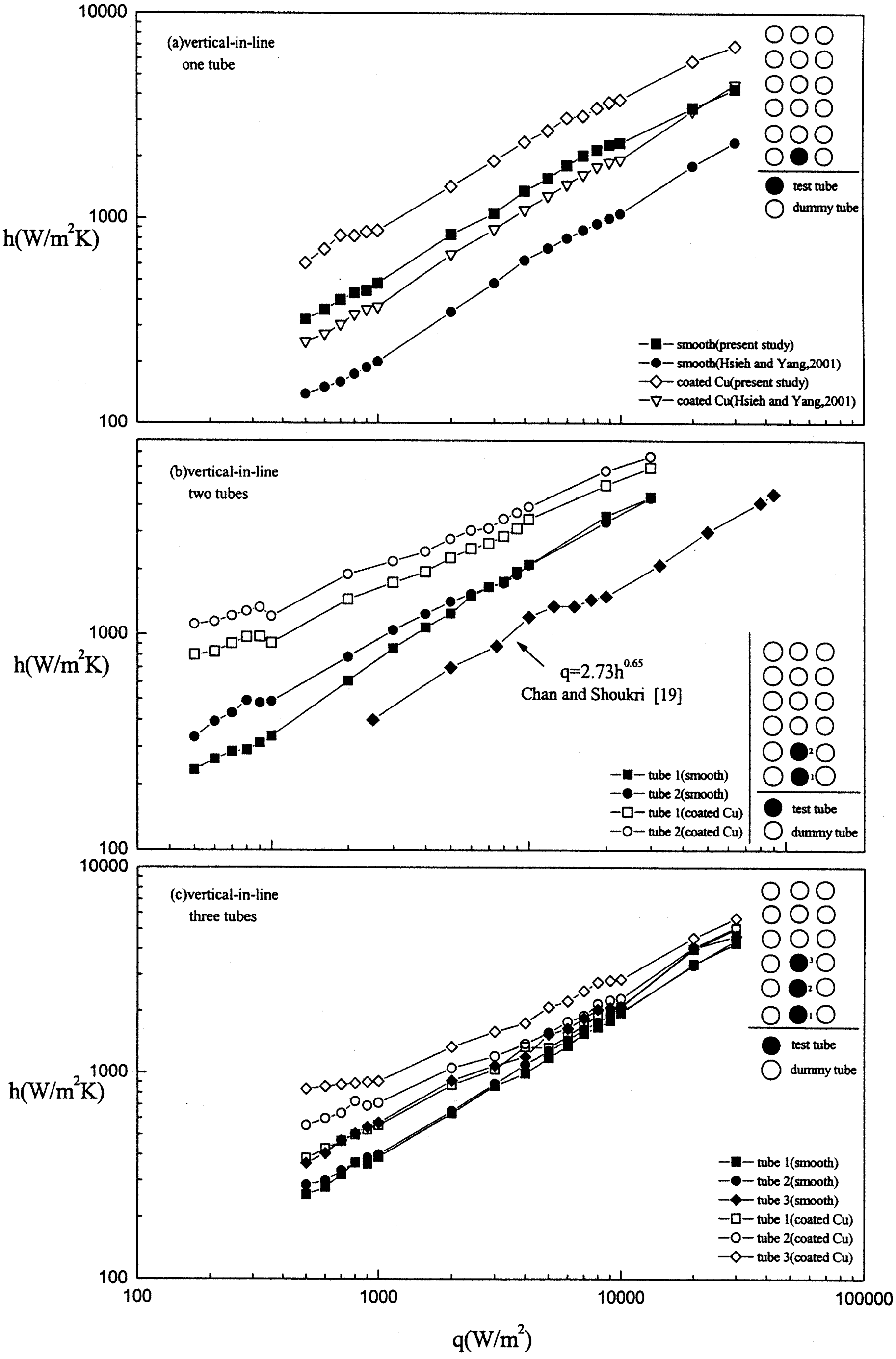


Fig. 7. Heat transfer coefficients of tube bundles (vertical-in-line).

The range of the exponent of q is about 0.57–0.73, which is in good agreement with those of Muller [8] ($\bar{h} = 10.0q^{0.53}$ for an in-line internal-finned tube bundle), but the enhancement of the present coated tube bundle are clear compared to Muller's results.

4.3. Bundle factor effect

As the heat transfer coefficient of the tube bundle is different from that of an isolated single tube, a bundle factor η, was thus defined as

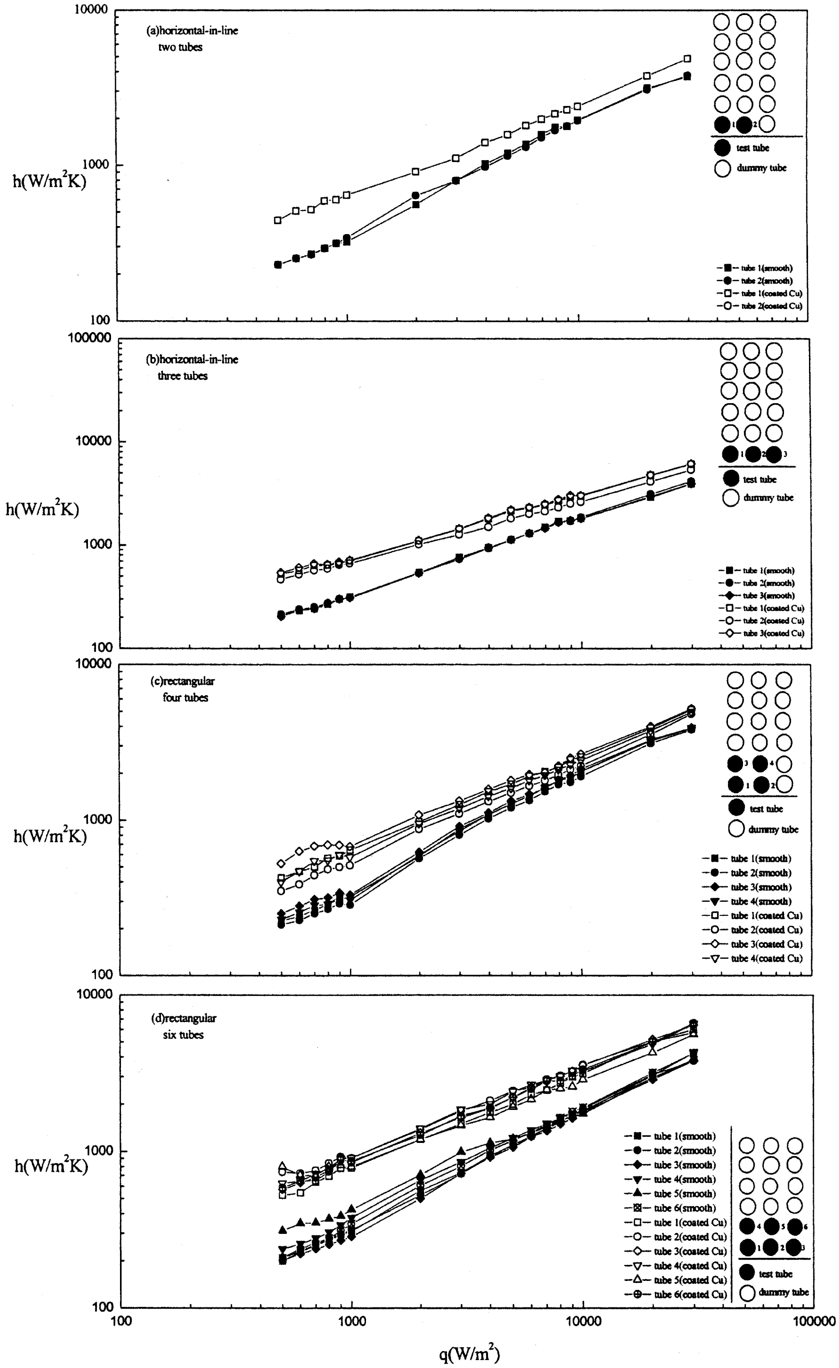


Fig. 8. Heat transfer coefficients of tube bundles (horizontal-in-line/rectangular).

$$\eta = \frac{\bar{h}}{h_{\mathrm{iso}}} \qquad (1)$$

and

$$\bar{h} = \frac{\sum A_i h_i}{\sum A_i} \qquad (2)$$

where $\bar{h}$ is an area (tube bundle surface) average and h_{iso} indicates the heat transfer coefficient of an isolated single

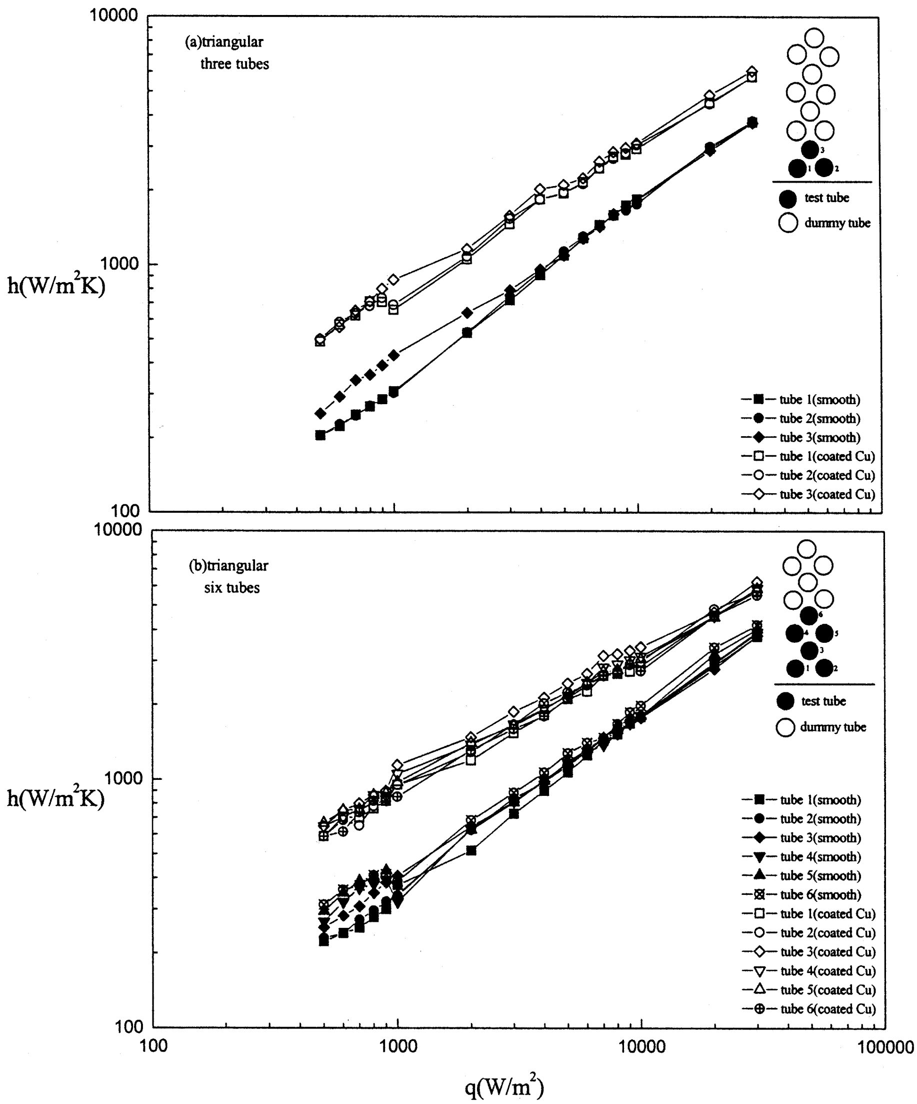


Fig. 9. Heat transfer coefficients of tube bundles (triangular).

Table 3
Correlation of tube bundles

Types of configuration	Number of tubes	Smooth	Coated
Vertical-in-line	One tube	$q = 5.26\bar{h}^{0.66}$	$q = 14.29\bar{h}^{0.61}$
	Two tubes	$q = 4.27\bar{h}^{0.67}$	$q = 4.26\bar{h}^{0.66}$
	Three tubes	$q = 4.29\bar{h}^{0.66}$	$q = 20.31\bar{h}^{0.53}$
Horizontal-in-line	Two tubes	$q = 4.04\bar{h}^{0.72}$	$q = 11.45\bar{h}^{0.58}$
	Three tubes	$q = 1.93\bar{h}^{0.74}$	$q = 11.35\bar{h}^{0.6}$
Rectangular	Four tubes	$q = 2.17\bar{h}^{0.74}$	$q = 10.9\bar{h}^{0.59}$
	Six tubes	$q = 2.6\bar{h}^{0.71}$	$q = 17.49\bar{h}^{0.57}$
Triangular	Three tubes	$q = 2.63\bar{h}^{0.71}$	$q = 12.45\bar{h}^{0.6}$
	Six tubes	$q = 4.08\bar{h}^{0.66}$	$q = 22.21\bar{h}^{0.54}$

Table 4
Comparison of bundle factor and configuration factor

Types of configuration	Heat flux (W/m^2)									
	500		1000		5000		10,000		30,000	
	η	$\xi_{k,j}$	η	$\xi_{k,j}$	η	$\xi_{k,j}$	η	$\xi_{k,j}$	η	$\xi_{k,j}$
Smooth										
Vertical-in-line										
One tube	2.32	–	2.41	–	2.19	–	2.21	–	1.81	–
Two tubes	2.05	0.88	2.06	0.85	1.87	0.85	1.99	0.9	1.85	1.02
Three tubes	2.17	0.94	2.27	0.94	1.86	0.85	1.91	0.86	1.9	1.05
Horizontal-in-line										
Two tubes	1.66	–	1.66	–	1.64	–	1.85	–	1.61	–
Three tubes	1.51	0.91	1.56	0.94	1.57	0.96	1.73	0.94	1.7	1.06
Rectangular										
Four tubes	1.65	–	1.56	–	1.78	–	1.94	–	1.66	–
Six tubes	1.65	1	1.72	1.1	1.6	0.9	1.73	0.89	1.69	1.02
Triangular										
Three tubes	1.58	–	1.73	–	1.55	–	1.72	–	1.61	–
Six tubes	1.89	1.2	1.81	1.04	1.63	1.05	1.73	1.01	1.68	1.04
Coated										
Vertical-in-line										
One tube	2.42	–	2.35	–	2.08	–	1.96	–	1.54	–
Two tubes	3.82	1.58	2.86	1.22	1.97	0.95	1.91	0.97	1.45	0.94
Three tubes	2.36	0.98	1.96	0.84	1.29	0.62	1.25	0.64	1.18	0.77
Horizontal-in-line										
Two tubes	1.69	–	1.73	–	1.2	–	1.22	–	1.07	–
Three tubes	2.05	1.21	1.87	1.08	1.6	1.34	1.5	1.23	1.31	1.23
Rectangular										
Four tubes	1.7	–	1.63	–	1.29	–	1.28	–	1.12	–
Six tubes	2.57	1.51	2.31	1.41	1.72	1.33	1.71	1.33	1.37	1.22
Triangular										
Three tubes	1.98	–	1.99	–	1.56	–	1.58	–	1.31	–
Six tubes	2.48	1.25	2.67	1.34	1.73	1.11	1.58	1	1.3	1

tube. Such data was from [15]. Subscript i denotes number of the tubes and A_i is the surface heat transfer area for entire tube bundle.

The values calculated for vertical-in-line, horizontal-in-line and rectangular, and staggered configurations were detailedly listed in Table 4 and plotted in Fig. 10(a)–(d), respectively. The tendency of these curves exhibit a common trend with a decrease in heat transfer coefficient as q increases as one expected for coated tubes. This is because the tube bundle is actually operating under the combined effect of nucleate and two-phase convective boiling with the relative importance of each component varying as a function of heat flux and location in the bundle. Moreover, it is evident that the two-phase convective effects are only significant at lower fluxes, while at higher heat fluxes nucleate boiling is dominant. Namely, η approaches 1 at $q \cong 30$ kW/m^2. A cross-over behavior was found at $q \cong 1$ kW/m^2 for coated and smooth tubes which indicates the η for smooth tube is nearly a constant ($\cong$1.67). However, it starts from about 4 and 3 at lower heat fluxes and drops to unity at higher heat fluxes for coated tubes which results in the above results.

It is found from Table 4 that the number of tubes in each bundle configuration has strongly influences on heat transfer.

Like Memory et al. [17], bundle factor decreases, reaching value of unity at the highest heat fluxes. This is where boiling from the surface itself begins to dominate, diminishing the influence of the other factors including convection pattern, different size rising bubbles, and secondary nucleation [21]. Note that the bundle factor of the present coated surfaces show the different behavior as that of porous surfaces in [17] where a bundle factor is nearly about 1 for the entire range of heat fluxes covered. However, for smooth surfaces of the present study, showing a bundle factor $\cong$1 regardless of heat flux level,

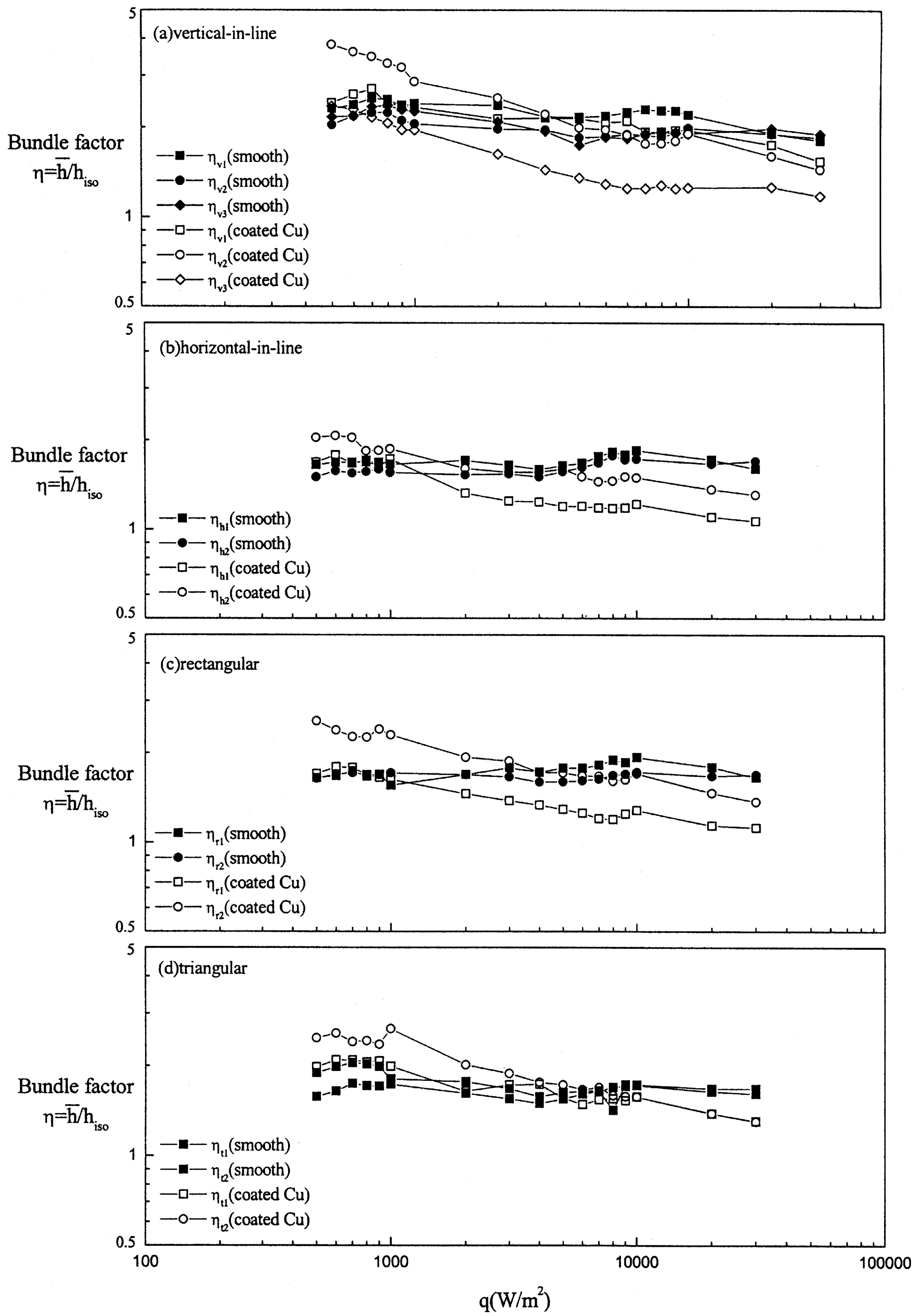


Fig. 10. Bundle factors of tube bundles.

which indicates the nucleation from the smooth surfaces is the dominant mechanism at low as well as high heat fluxes. The tubes below make little or even no difference to the heat transfer performance. Fig. 11 shows the configuration factor distributions for different geometric arrangements which are quietly consistent with those of Fig. 10.

5. Conclusions

An experimental study has been performed for pool boiling of R-134a from smooth and plasma coated tube bundles with a pitch-to-diameter ratio (p/d) of 1.5 at $100\ \mathrm{W/m^2} \leqslant q \leqslant 30,000\ \mathrm{W/m^2}$. The conclusions can be drawn as below:

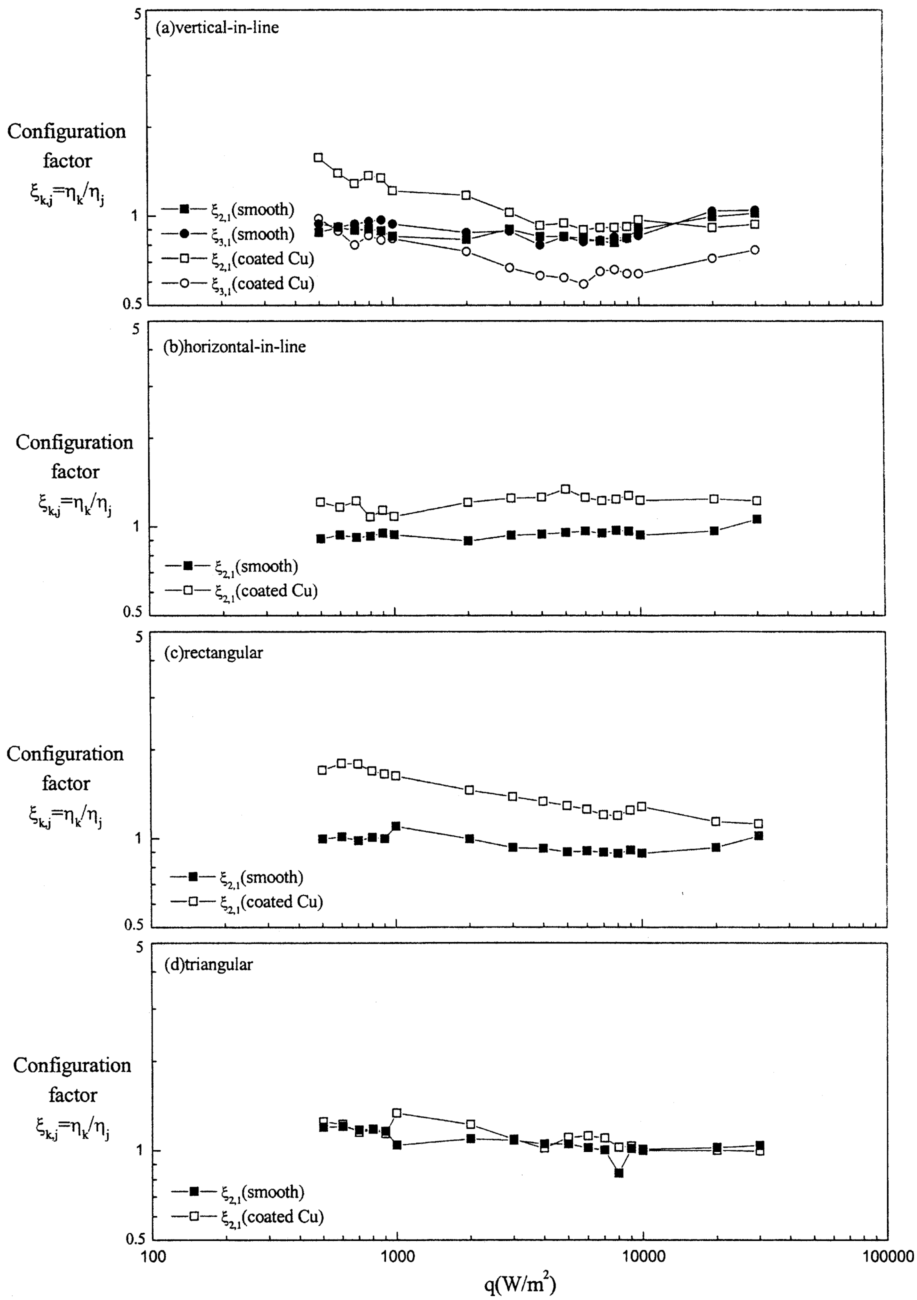


Fig. 11. Configuration factors of tube bundles.

1. Significant bundle factor effect has been found at low heat fluxes, and decreases to unity at high heat fluxes. For a smooth bundle, it is about unity for all heat fluxes studied which are different from that of Memory et al. [17].
2. Remarkable enhancement of heat transfer of the top rows in vertical-in-line and staggered types were observed upon the heat fluxes imposed.
3. The average bundle heat transfer coefficient was found to be three times greater than for a smooth tube bundle and five times greater than an isolated single smooth tube. In fact, a bundle factor of up to 3.82 was found for vertical-in-line coated tube bundles at $q \geqslant 500$ W/m^2.
4. A configuration factor was defined, its value is less than 1 for several cases in some geometric arrange-

ments. Both bundle factor and configuration factor show a consistent trend for the associated tube bundle configurations, and an optimum geometry arrangement could be selected based on the complex effect of the types of tube bundle configurations and heat fluxes.

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