AN EXPERIMENTAL STUDY ON THE EFFECTS OF TUBE POSITION AND SPACING IN A BUNDLE POOL BOILING HEAT TRANSFER

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Abstract. *In this paper, an investigation on the effect of the tube positioning on the pool boililing heat transfer coefficient is presented. Experiments were performed for R123 at a reduced pressure of 0.022, average surface roughness of 0.12 µm, and heat fluxes from 1 to 40 kW/m^2. Bundles containing 2 and 3 horizontal, electrically heated brass tubes were used in the experiments. The effect of tube spacing on the heat transfer coefficient were also investigated. Experimental results have shown that thermal performance of the bottom tube of the row presents similar thermal behavior as a single tube in a pool over the whole range of heat flux. However, the upper two tubes present distinct behavior depending on the heat flux range. Under partial nucleate boiling, thermal performance of the two upper tubes is better than the bottom one. However, as the heat flux increases and boiling becomes fully developed, thermal performance of the three tubes tends to be similar.*

Key words: *nucleate boiling , tube bundle, halocarbon refrigerant*

1. INTRODUCTION

The nucleate boiling heat transfer mechanism has been the object of massive investigation during the last 70 years since the pioneer studies of Jakob and Fritz (1931), and Nukyama (1934) were published. Since then, pool boiling heat transfer has been investigated to better understand the boiling phenomenon, viz. nucleation site characteristics, pool boiling regimes, critical heat flux, bubble growth, bubble departure diameter, bubble detachment frequency, and the development of physical models and correlations. Additionally, intensive efforts have been made towards the enhancement of nucleate boiling heat transfer by means of surfaces with special structures and plain surfaces covered with special porous matrixes (see Thome and Ribatski (2007)).

Nucleate boiling processes can be found in flooded evaporators, falling-film evaporators and centrifugal chillers in refrigeration and air-conditioning systems, and in kettle reboilers in refineries and chemical processing plants. Recently, nucleate boiling is also found in the electronics industry for cooling high-power densities micro-processors. Moreover, nucleate boiling mechanism is responsible for most parcel of the heat transferred through convective boiling at low vapor qualities and, thus nucleate boiling effects are often included in the convective boiling heat transfer predictive methods (Chen (1966), Liu and Winterton (1991), Kandlikar (1990) and Wojtan et al (2005)).

The current investigation is related to an experimental analysis of the heat transfer performance of individual horizontal tubes along a flooded evaporator. In these equipments, the heat transfer performance is a result of the complex interaction between the dynamics of the gas and liquid phases. Bubbles detached from the lower tubes may slide along the surface of the upper tubes what may promote both the activation of additional cavities and the evaporation of a superheated thin film captured at the bubble base. Moreover, in flooded evaporators additional complexity is introduced when compared with single tube pool boiling and in tube boiling due to geometrical parameters viz. tube arrangement, tube spacing, number of rows and columns, and tube diameter.

Table 1 summarizes the experimental conditions and tube array characteristics of some published investigations concerning flooded evaporation in tube bundles. These studies have focused on measuring heat transfer coefficients, h, and investigated the effect of the tube spacing and distribution on the heat transfer performance. As shown in Table 1, previous researches concerning this topic have been conducted on in-line and staggered tube arrangements. Generally the tube diameters, d, and tube pitch, s, have been similar to those used in heat exchangers. Research has been carried out for plain and enhanced surfaces. Most of the studies were conducted with CFC refrigerants which are now banned in most countries. There are no data available for halocarbon refrigerants with lubricating oil mixtures. This table also provides the ratio between $h(n)/h(1)$, where n is the tube row number numbered in the upwards direction, and $h(1)$ the heat transfer coefficient on the lower tube row, and the heat flux at which $h(n)/h(1)$ is maximum.

From the studies in Table 1, it is important to highlight that only few authors investigated in detail the effects of the tube positioning on the heat transfer coefficient. Instead, most of them obtained overall heat transfer results. It is important to highlight that overall heat transfer coefficient results are typical of the tested bundle and consequently are not useful to the development of general predictive tools. Additionally, in the design of modern heat exchanger local heat transfer coefficients are used instead of overall values. Based on the abovementioned aspects, in this paper, experimental results concerning the effect of the tube positioning on the local pool boiling heat transfer coefficient are reported. Experiments were performed for R123 at a reduced pressure, p_r, of 0.022 (saturation temperature, T_{sat}=21.7^{o}C), average surface roughness, *Ra*, of 0.12 µm, and heat fluxes, ϕ, up to 40 kW/m^2. The effect of tube spacing on the heat transfer coefficient was also investigated.

Table 1. Summary of experimental conditions and tube array characteristics.

Author	**Tube bank characteristics**	**Maximum $h(n)/h(1)$ ratio**	**Heat flux related to the maximum $h(n)/h(1)$ ratio (kW/m^2)**	**Fluid/ saturation pressure (kPa)**
Wallner (1971)	Staggered / 12 tubes / 4 rows / *s*/*d* = 1.33	1.5	1.0	R11/ 100
Hahne e Muller (1983)	inline / 2 tubes / 2 rows / *s*/*d* = 2.0	1.8	4.0	R11/ 100
Muller (1986)	staggered / 18 tubes / 6 rows / *s*/*d* =1.3	4.0	2.0	R11 / 100
	s/*d* =1.5	3.3		
	s/*d* = 2.0	2.2		
Jensen e Hsu (1988)	inline / 135 tubes / 27 rows	2.4	2.17	R113 / 517
Hahne et al. (1991)	inline / 2 tubes / 2 rows / *s*/*d* =1.05	1.6	7.5	R11/ 100
	finned tube (748 fins/m) / *s*/*d* = 1.3	1.82	4.0	
	s/*d* = 3.0	2.0	4.5	
Gupta et al (1995)	inline / 3 tubes / 3 rows / *s*/*d* =3.0	2.2	10	water / 100
	inline / 2 tubes / 2 rows / *s*/*d* =3.0	1.4		

** Experiments were performed for plain tubes when not specified.*

2. EXPERIMENTAL SETUP

The experimental setup comprises the refrigerant and cooling circuits. The refrigerant circuit is schematically shown in Fig. 1. The charge of refrigerant is basically contained in the boiler in which the liquid is kept at a reasonable level above the test surfaces (tubes) so that the column head does not affect significantly the equilibrium saturation temperature. The cooling circuit is intended to control the equilibrium pressure in the boiler by condensing the refrigerant boiled on the heating surface. The condensing effect is obtained by a 60% solution of ethylene glycol/water that operates as intermediate fluid between the condenser and the cooling system, not shown in Fig. 1. The ethylene glycol/water solution is cooled by a vapor-compression refrigeration system.

The boiler is a 40 l carbon steel container with two lateral circular windows for visualization. It contains the boiling surfaces in addition to a 1500 W/220 V electrical heater, installed at the bottom, and two sheathed type T thermocouples. The boiler is also fitted with openings for connections to a pressure transducer, a safety valve (not shown in the figure), and vapor and liquid return copper lines, as shown in Fig. 1. The sheathed thermocouples are installed in such a way to measure and monitor the temperature of the liquid pool and the vapor in equilibrium with it. Under normal operating conditions, these thermocouples indicate temperatures very close to each other and to the saturation temperature at the boiler internal pressure measured by the pressure transducer.

The test (boiling) surfaces are placed in the central region of the boiler so that the boiling mechanism can easily be visualized through the glass windows. The test surfaces are made up of a 19.0 mm diameter and 3.1 mm thick brass tube, their cut way view is shown in Fig. 2. The test tubes are placed according to a vertical inline arrangement and supported by a brass device (also shown in Fig. 2) which is thread attached to the flanged cover of the boiler. By rotating this device different tube spacing are accordingly set up (*s*/*d* ratios of 1.32, 1.53 and 2.00). The electrical power to the boiling surfaces are individually controlled by manually operated voltage converters and measured by power transducers. Surface temperature is measured through 30 AWG type T thermocouples installed in grooves carved by an electro-erosion process in locations indicated in Fig. 2. Thermocouples are kept in place by a conductive epoxy resin. Electrical signals from the transducers are processed by a data acquisition system which includes two 12 bit A/D converter boards with 16 channels each, and three connection panels.

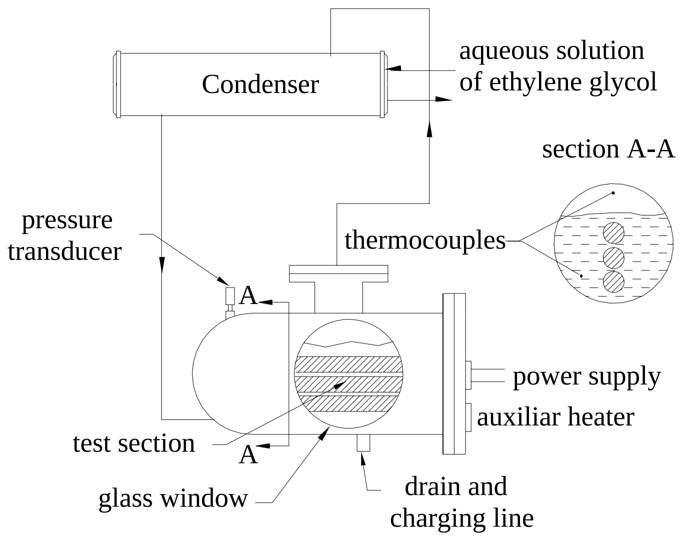

Figure 1. Schematic diagram of the experimental apparatus

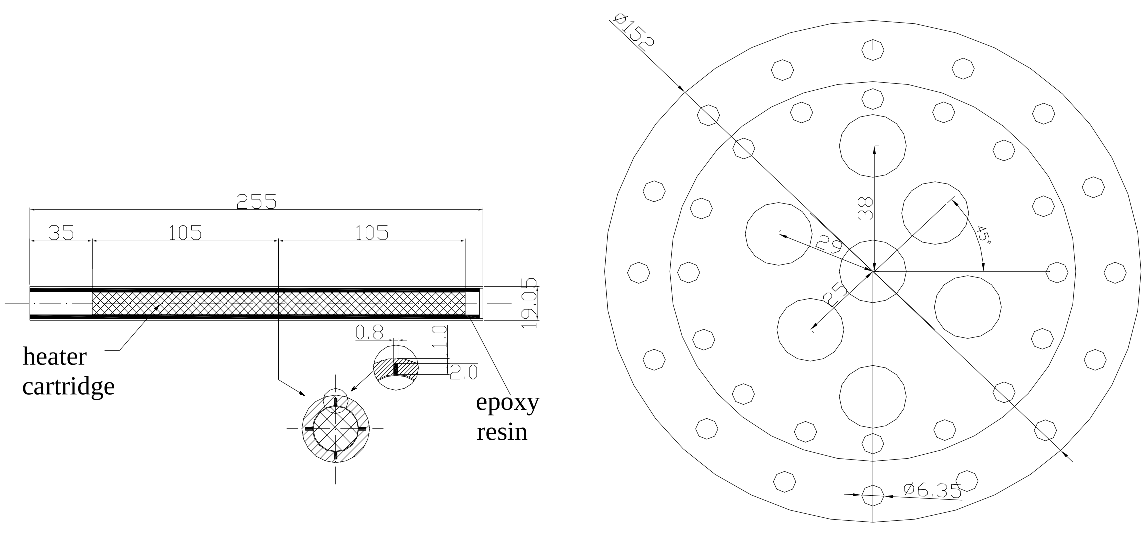

Figure 2. Longitudinal and transversal cut views of the cylindrical heating surface and schematic of the device supporting the test sections (measurements in mm).

3. EXPERIMENTAL PROCEDURE

The boiling surfaces used to be treated prior to the beginning of the tests. Sandpaper, applied through a regular late machine run at 1200 rpm, was used to obtain the final surface roughness. After such a treatment, the boiling surfaces used to be thoroughly cleaned with a solvent and their roughness measured at 10 randomly selected regions before attaching it to the boiler. After testing, other 10 randomly selected regions of the boiling surfaces were again taken for roughness measurement so that conditions of the surface before and after the tests could be compared. Experiments were performed for a roughness measured in terms of the CLA arithmetic average, Ra of 0.12± 0.03.

The internal surface of the boiler used to be cleaned and kept under a vacuum of less than 2 kPa during a period of 12 h before attaching the boiling surfaces and introducing the refrigerant. Tests were conducted under saturated conditions of the refrigerant. The datum point would only be logged if the readings of the sheathed thermocouples were close enough (within 0.2 K) to each other and to the saturation temperature corresponding to the boiler pressure. For analysis purposes, the saturation temperature, T_{sat}, of the pool was determined as the average of the readings of the sheathed thermocouples. Wall temperatures given by the thermocouples were corrected according to Fourier's law in order to adjust the measured value within the tube wall to the temperature on the surface of the tube, T_w. The tube wall superheating (T_w-T_{sat}) was defined as the arithmetic mean of the local wall superheating along the tube circumference of each boiling surface. Assuming only heat flow in the radial direction, the heat flux, ϕ was defined as the ratio of the power supplied to the electrical heater and the external area of the tube referred to the heater length. The heat transfer coefficient was calculated as the ratio between the heat flux and the tube wall superheating.

In order to not have heat transfer data significantly affected by hysteresis effects, the surfaces were submitted to a heat flux about 40kW/m^2 for some time before starting the experiments. Moreover, tests were conducted by gradually decreasing the heat flux down to zero. Instruments were calibrated and the uncertainty of measured parameters evaluated according to the procedure suggested by Abernethy and Thompson (1973) with results summarized in Table 2.

Table 2. Uncertainty of measured and calculated parameters.

Parameters	**Uncertainties**
Minimum heat flux, ϕ = 0.9 kW/m^2	± 1.75 %
Maximum heat flux, ϕ = 40kW/m^2	± 0.80 %
Wall superheating	± 0.3 K
Saturation temperature	± 0.2K
Tube surface area	± 0.26%
Heat transfer coefficient at the minimum heat flux, (h=0.20 kW/m^2.K)	± 9.45 %
Heat transfer coefficient at the maximum heat flux, (h=2.85 kW/m^2.K)	±2.77 %

4. EXPERIMENTAL RESULTS

According to the different behavior of bubble dynamics and heat transfer characteristics, the nucleate boiling regime may be categorized into, partial nucleate boiling and fully developed nucleate boiling (see Dhir (1998)). The two-phase flow dynamics and heat transfer characteristics are distinct in the two regimes. In the stage of partial nucleate boiling, isolated vapor bubbles are released from randomly located active sites on the heated surface, the free convection mechanism is still responsible for most of the heat transferred from the surface and the relationship between the heat flux and wall superheating is given according to $\phi = a + b\Delta T^n$. The fully developed nucleate boiling regime is characterized by bubbles beginning to merge at a given site, the natural convection effects become negligible, and the relationship between the heat flux and wall superheating is given by $\phi = b\Delta T^n$, with an exponent substantially higher than in partial nucleate boiling.

At partial nucleate boiling regime, Ribatski and Saiz Jabardo (2003) observed for single tubes that nucleate boiling occurs in an intermittent fashion, with clusters of bubbles appearing and disappearing in different regions of the heating surface. It has been observed that the active site density is higher and the bubbles size smaller in the uppermost region with respect to those at the bottom of the boiling surface. They also observed that such a behavior was affected by the bubbles dynamics along the tube perimeter. Thus, for low heat fluxes, it seems reasonable to expect that in a bundle bubbles generated on the lower tubes may affect the boiling and heat transfer processes at the upper tubes. It is important to emphasize that actually most of the heat exchanger manufacturers from the refrigeration and air-conditioning industries are keeping as goal working at heat fluxes up to 20kW/m^2, which, under some circumstances are characterized as partial nucleate boiling regime. Such an aspect highlights the importance of carrying out experimental studies at relatively low heat fluxes.

Figures 3 and 4 show, respectively, the variation in the local heat transfer coefficient with the heat flux for two and three horizontal tubes vertically aligned. Additionally, boiling curves, h vs. ϕ , from Saiz Jabardo et al. (2004) and Ribatski and Saiz Jabardo (2003) correlations are also overlaid in the same plot. In general, the lowest tube within the row provides heat transfer coefficients similar to those observed in experiments with single tubes. As noted in these figures, results from both correlations over predict experimental data for the lowest tube, displaying an unexpected result. It must be noted that both correlations have been developed considering wider ranges of heat fluxes data (up to 120 kW/m^2) what affects the performance in the extremes of the heat fluxes range. However, the expressions by Saiz Jabardo et al. and Ribatski and Saiz Jabardo compare very well with experimental data in at higher heat fluxes. In the case of the upper tubes and similar experimental conditions (ϕ, T_{sat}, Ra, surface material), the following two distinct trends are suggested: (i) at partial nucleate boiling regime, the heat transfer coefficient increases in the upward direction along the tube row; and (ii) at fully developed nucleate boiling regime, the heat transfer coefficient changes slightly with the relative tube positioning and has a value similar to the one observed in single-tube experiments. Although not investigated in this study, it should be expected that parameters such as reduced pressure, surface roughness, *s*/*d*, and fluid refrigerant, which affect the bubble dynamics, would also affect the transition from partial to fully developed nucleate boiling. Thus, in the case of heat transfer enhanced surfaces, high reduced pressures and high pressure refrigerants, conditions for which the boiling process is intense even at low wall superheating, the heat flux range at which the tube positioning affect the heat transfer coefficient is probably reduced.

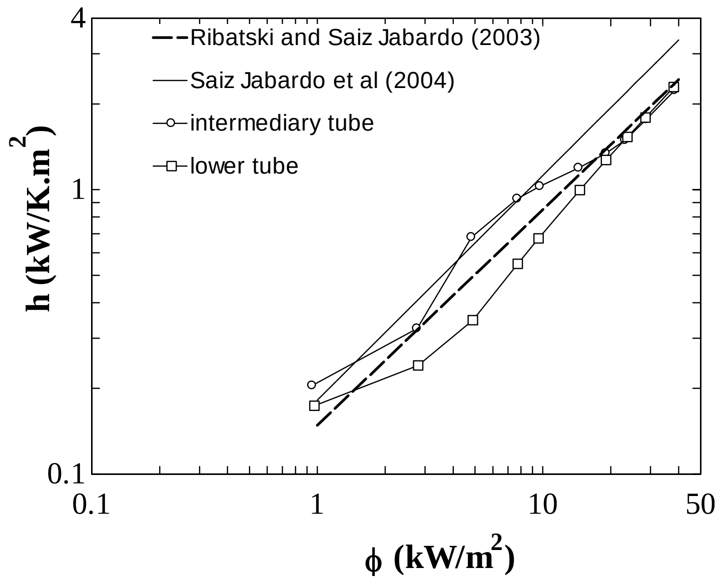

Figure 3. Boiling curves for two vertically aligned horizontal tubes, R123 , p_r =0.023, s/d =2.0

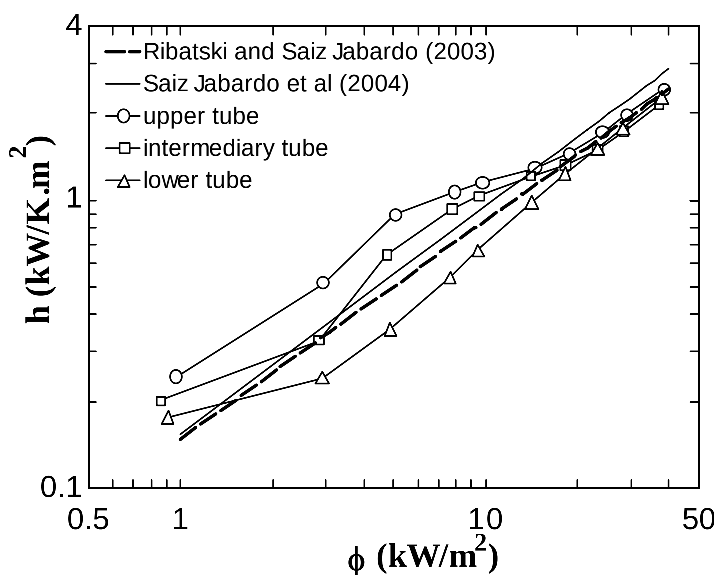

Figure 4. Boiling curves for three vertically aligned horizontal tubes, R123, p_r =0.023, s/d = 1.32.

Figure 5 shows the variation with the heat flux of the ratio between the heat transfer coefficients of one tube and another below it. According to this figure, this ratio reaches a maximum at a heat flux about 4 kW/m^2. Additionally, it can be concluded that the ratio $h(n)/h(n\text{-}1)$ decreases as n increases, where n is the tube row numbered in an upwards direction. Based on this behavior, it is speculated that there is a limiting tube row number above which $h(n)/h(n\text{-}1)$=1.

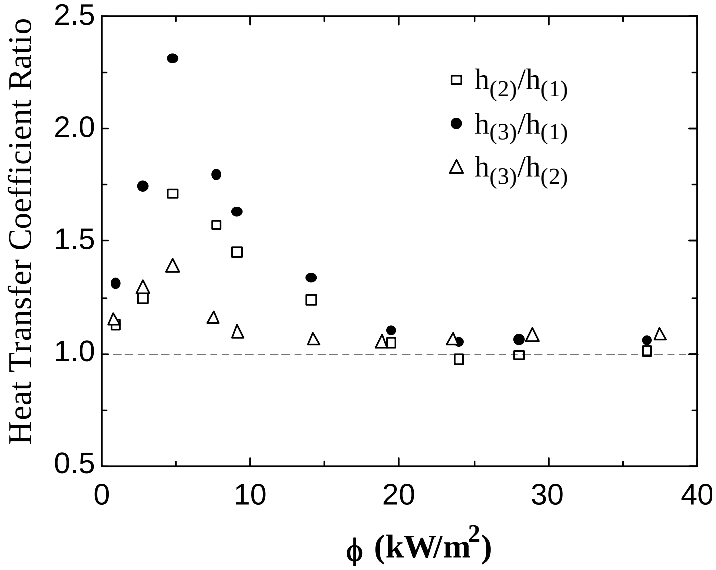

Figure 5. Illustration of the effect of the tube positioning and heat flux on the specified heat transfer coefficient ratio, R123, p_r =0.023, s/d = 1.32.

Figure 6 illustrates the effect of the tube spacing on the ratio between the heat transfer coefficients of the second and first tubes. According to this figure, this ratio varies within the uncertainty range of measured results by increasing *s/d* from 1.32 to 2.00. The limited effect of *s/d* on the heat transfer coefficient may be related to the range of tube spacing considered in the present study. As pointed out by Liu and Qiu (2002), it is expected that the tube spacing effects on the heat transfer coefficient become relevant as the tubes come closer to each other. It is important to underline that the contrasting heat transfer behavior with varying the inter-tube distance was already pointed out in the literature without any explanation that could be systematically identified and used to justify the differences between authors. As an example, for partial nucleate boiling it was found by Wallner (1971), Muller (1986), Gupta et al (1995) and Liu e Qiu (2002) an increase in the heat transfer coefficient with decreasing inter-tube spacing . A decrease in the heat transfer coefficient was observed by Hahne et al. (1991).

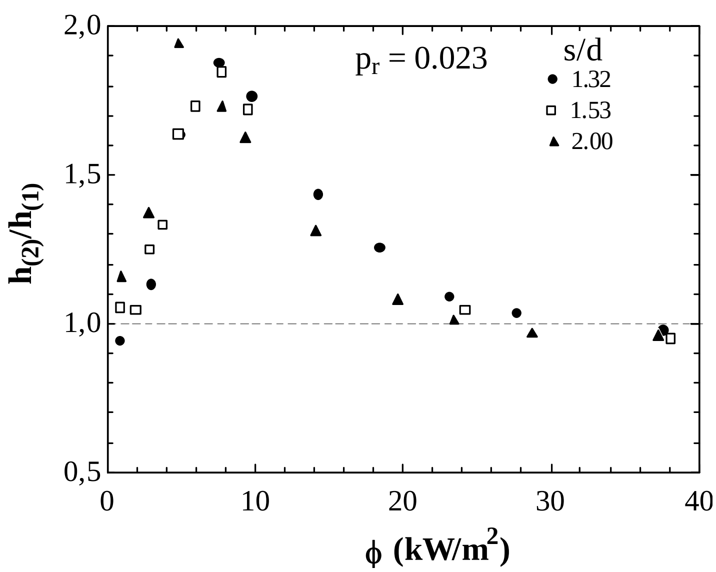

Figure 6. Illustration showing the effect of the tube spacing and heat flux on the $h(2)/h(1)$ ratio, R123, p_r =0.023.

The plots of Fig. 7 have been developed with the purpose of investigating the behavior of the temperature of the fluid flowing from the lower tube and reaching the tube just above it, and so obtain experimental knowledge to better explain the abovementioned trends. This figure shows, for different tube spacings, how heating the lower tube affects the temperature difference between the underside of the upper tube without heating and the saturation temperature, ΔT_{free}. The following behavior can be spotlighted from Fig. 7: (i) in general, ΔT_{free} increases with decreasing the heat flux; (ii) at lower heat fluxes and thus under the partial nucleate boiling regime, ΔT_{free} decreases with increasing the inter-tube spacing, while (iii) at higher heat fluxes, an almost negligible effect of inter-tube spacing on ΔT_{free} is observed. This behavior could be related to an interaction among effects of the inter-tube distance and the mechanisms of natural and forced convection, the latter related to the degree of agitation in the liquid imposed by the bubbles.

By increasing the heat flux, the bubbly intensity increases and the mixing between the warm liquid from the lower tube and the liquid at saturation temperature in the pool also increases. This mechanism would cause a reduction in the temperature of the liquid reaching the lower part of the upper tube. At low heat fluxes, when the free convection mechanism is dominant, the liquid heated on the lower tube flows from the uppermost part of the lower tube to the bottom of the upper tube. The upward current of warm liquid exchanges heat and mixes with the surrounding liquid at saturation temperature. Based on the fact that the amount of heat exchanged is proportional to the inter-tube distance, it is reasonable to conclude that the shorter the inter-tube distance the higher would be the value of ΔT_{free}. In the case of a heated upper tube, the warm liquid from below would favor the nucleation of new sites and, consequently, the enhancement of the heat transfer coefficient on the upper tube. This mechanism would explain higher heat transfer coefficients on the upper tubes, but apparently contradicts the almost negligible effect of *s*/*d*. A possible explanation to the reduced inter-tube distance effect could be related to the limited range of values of the present investigation that would result in variations of the order of magnitude of the measurement uncertainties.

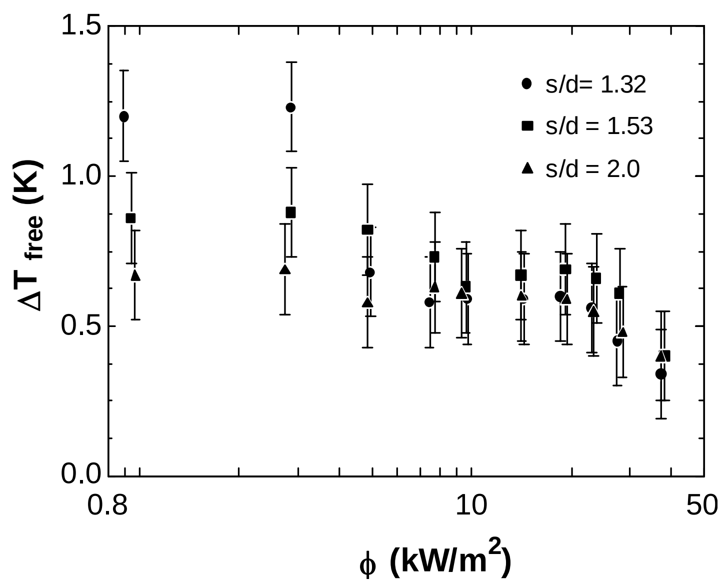

Figure 7. The effect of the tube spacing and heat flux of the lower tube on the temperature difference between the underside of the upper tube without heating and the saturation temperature, ΔT_{free}, R123, p_r =0.023.

5 - CONCLUSIONS

From this experimental study, the following general conclusions can be drawn:

- At the partial nucleate boiling regime, the heat transfer coefficient increases in the upwards direction along a vertical column of horizontal tubes until achieve a tube row after which the heat transfer coefficient becomes almost constant.
- At the fully developed nucleate boiling regime, the effects of the tube positioning on the heat transfer coefficient were almost negligible.
- No significant effect of the inter-tube spacing on the local heat transfer coefficient was observed.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the support given to the investigation reported herein by FAPESP (Fundação de Amparo à Pesquisa do Estado de São Paulo) through a doctoral assistantship to the first author and a post-doctoral assistantship to the third author, modality “young researcher” (contract number 06/52089-1), during the preparation of this paper.

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