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Boiling of saturated FC-72 on square pin fin arrays ☆

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Abstract

Pool boiling heat transfer from finned copper surfaces immersed in a saturated dielectric fluid (Fluorinert FC-72) has been experimentally studied. Twelve extended surfaces with different geometrical configurations were tested. Each extended surface consisted of an array of pin fins with square cross-section. Fins were 3 or 6 mm long and their width varied from 0.4 to 1.0 mm. Fins were uniformly or non-uniformly spaced on the base surface: starting from the uniform configuration, in which the width and the spacing of fins were equal, non-uniform surfaces were obtained by regularly removing some rows of fins. For each extended surface, boiling curves were obtained at three different saturation pressures: 0.5, 1.0 and 2.0 bar. The effects of fin dimensions, spacing and pressure on heat transfer in saturated pool boiling were examined. In particular, the effect which the non-uniform distribution of fins produces on boiling behaviour was analysed. When fins thin out, the overall heat transfer coefficient based on the total area of the extended surface increases, but the heat transfer rate does not improve, even in the boiling region close to the maximum heat flux. The better wetting of the boiling surface by the liquid refrigerant, as a result of the sparser grouping of fins, is apparently offset by the reduction in the heat transfer area. If pressure is increased, the boiling curves of finned surfaces move towards lower wall superheats, as already observed in previous studies with regard to both plane and finned surfaces.

Keywords: Pool boiling; Extended surfaces; Highly-wetting fluid; Heat transfer

1. Introduction

In the last three decades many techniques have been developed to enhance convective heat transfer. General goals are to permit the accommodation of high heat fluxes at a moderate temperature difference or to reduce the temperature difference for a fixed heat flux. The success of a heat transfer enhancement technique is strictly related to the convective mechanism involved.

Selected passive techniques based on enhanced surface geometries have been shown to be effective at very high fluxes, and numerous enhanced surfaces have been developed specifically for nucleate pool boiling. Several theoretical and experimental studies have been carried out involving pool boiling from single fin or extended surfaces, on both micro and large scales.

Early investigations into boiling on extended surfaces soon raised the problem of evaluating reciprocal interactions among fins. Siman-Tov [1] pointed out that an analysis of single fins might be vulnerable to the effect of spacing in an array of fins. Klein and Westwater [2] experimentally analysed the effect of varying vertical and horizontal spacing between fins in water and R-113. Generally, in the case of finned heat sinks, the extended surface, from the thermal and fluid dynamic point of view, does not behave as the sum of many single fins, each independent of the others, but, as visual observations of boiling phenomena reveal, there is a marked interaction among fins. Hirono et al. [3] pointed out the relevance of interference between adjacent rectangular fins, which appears as a reduction in heat transfer performance in the high heat flux heat region. Likewise, Bergles et al. [4] observed a marked space effect for multiple square-shaped spines and pointed out that effectiveness diminished as the gap between two adjacent spines was reduced.

What appears particularly interesting is the possibility of defining—on the basis of fin geometry, fluid properties and boiling regime, and consequently of vapour flow rate—a non-interfering limit value for fin space, i.e., the clearance

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Nomenclature

a	spine thickness	mm
b	distance between two adjacent spines	mm
A_b	base area of the heat flux meter	m^2
A_{ext}	total heat transfer area of the extended surface	m^2
h_{ext}	overall heat transfer coefficient, defined by Eq. (3)	$W \cdot m^{-2} \cdot K^{-1}$
l	spine length	mm
p	pressure	bar
p_c	critical pressure	bar
q	heat transfer rate	W
q''	heat flux referred to the base area	$W \cdot m^{-2}$
q''_{ext}	heat flux referred to total heat transfer area of the extended surface	$W \cdot m^{-2}$
R	ratio between the total heat transfer area of the extended surface and the base area	
T_{sat}	saturation temperature	K
T_w	temperature of the base surface	K
ΔT_{sat}	base surface superheat ($T_w - T_{sat}$)	K
w	distance between two adjacent blocks of spines	mm

at which each fin in an array acts as a single fin. For heat transfer to boiling water or R-113 at atmospheric pressure, Klein and Westwater [2] regarded horizontal spacing of 1.6 mm as wide enough to allow fins to act independently of each other. Using FC-72, Mudawar and Anderson [5] discerned a non-interference gap of 0.6 mm between pin-fins, which is close to the bubble departure diameter; that value falls within the range considered by Hirono et al. [3] (0.28–2.0 mm) and Bergles et al. [4] (0.4–1.0 mm), though with differing fin geometries and dielectric fluids. As recent studies have emphasised [6–8], this issue continues to hold considerable practical and theoretical interest, but it is far from solved. Moreover, to our knowledge, the possibility of enhancing the boiling performance of extended surfaces by utilising finned surfaces with non-uniformly spaced fins has not been investigated.

Since the early 40's, pressure was considered to affect boiling performance and critical heat flux [9] and it was soon established that boiling performance improves when pressure rises [10]. Already theoretically quantified by Lienhard and Schrock [11], this result was experimentally proved for highly wetting fluids and plane surface [12], too, and in particular, more recently, for chlorine-free dielectric fluids [13,14]. For finned surfaces, the displacement of the boiling curve with pressure has likewise been analysed. Several experiments have been performed for a wide range of pressures with finned tubes and low contact-angle fluids [15–17]: the general result has been that the heat transfer coefficient for a given heat flux increases with an increase in reduced pressure for all fin geometries and for all refrigerants tested.

The purpose of the present study was to explore the possibility of enhancing boiling heat transfer by thinning out spines set on the surface, with a view to improving liquid flow through fin arrays, particularly at high heat fluxes, when the vapour mass flow rate is considerable. The effect of pressure on finned surface boiling performance has been also investigated in order to analyse the heat transfer rate increase with pressure and ascertain whether the methods of predicting the influence of pressure on boiling proposed in the literature for plane surfaces are also applicable to extended surfaces.

2. Extended surfaces and experimental apparatus

The experimental apparatus utilised during the tests consists of an airtight stainless steel vessel containing a saturated dielectric fluid (Fluorinert FC-72) in which the test section assembly is immersed.

The heat source module consists of a copper block, used as a heat flux meter, which is thermally insulated and heated on the lower side by a plane heater. The boiling area is the circular upper surface of the flux meter, which has a diameter of 30 mm, corresponding to a base area, A_b, of 7.07×10^{-4} m^2. The heat flux through the copper test section and the surface temperature were calculated on the basis of the temperatures measured by nine K-type thermocouples lodged inside the copper flux meter at different depths.

All temperatures were recorded by means of a high-precision data acquisition system. The chain of instruments used to detect the signals given by the thermocouples produces an accuracy of ± 0.1 K on absolute temperature values and of ± 0.02 K on differential values. The influence of errors in the measurement of temperatures and in the estimation both of thermal conductivity of copper bar and of position of the thermocouples on uncertainty in experimental results was analysed. The overall uncertainty in the determination of heat flux resulted to be dependent on operative conditions. In the region of single phase natural convection and at the onset of nucleate boiling the maximum error on heat flux was $\pm 15\%$. In the fully developed nucleate boiling region, the maximum error associated with the heat flux was calculated to be less than $\pm 5\%$. Close to the thermal crisis the overall uncertainty arose to $\pm 8\%$. The uncertainty associated with the base surface temperature in the fully developed boiling region was estimated to be less than ± 0.2 K.

During the experiments twelve surfaces were tested to examine the influence of non-uniform fin spacing on the pool

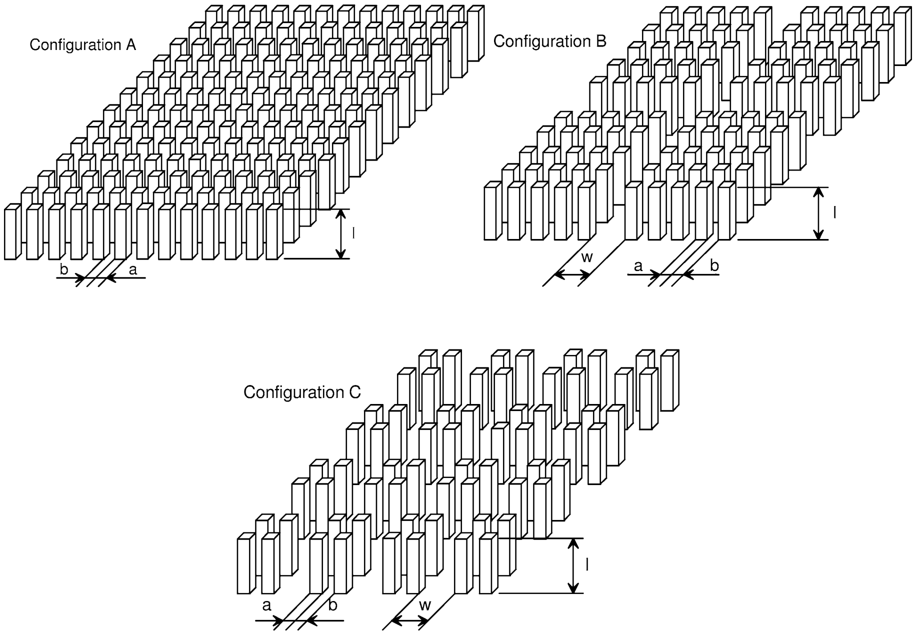

Fig. 1. Configurations of extended surfaces.

Table 1
Dimensions of extended surfaces

Surface No.	1	2	3	4	5	6	7	8	9	10	11	12
l (mm)	3	3	3	6	6	6	3	3	3	6	6	6
a = b (mm)	0.4	0.4	0.4	0.4	0.4	0.4	1.0	1.0	1.0	1.0	1.0	1.0
w (mm)	–	1.2	1.2	–	1.2	1.2	–	3.0	3.0	–	3.0	3.0
Configuration	A	B	C	A	B	C	A	B	C	A	B	C
$R = A_{\text{ext}}/A_{\text{b}}$	8.50	6.20	4.33	16.00	11.42	7.66	4.00	3.08	2.33	7.53	5.53	3.90

boiling performance of extended surfaces. Each extended surface consisted of an array of straight spines with square cross-section, uniformly or non-uniformly spaced. The fins had a length of 3 mm or 6 mm, in order to conform to typical spatial packaging limitations. The spine thickness, coinciding with the distance between two adjacent spines, was set at 0.4 mm or 1.0 mm. These finned surfaces were obtained by cutting perpendicular grooves into the plane surface of a copper bar by means of electro-discharge machining. Fins extended over the entire circular surface. Starting from a uniform configuration (configuration A; Fig. 1) for which the width and spacing of the spines were equal, non-uniform surfaces were obtained by regularly eliminating some rows of spines: one row every six in the case of configuration B and one row every three in the case of configuration C were removed in both directions.

Fig. 1 shows the three different configurations adopted, whereas Table 1 reports the geometrical dimensions of the twelve surfaces tested. For each finned surface, Table 1 indicates: spine length (l), spine thickness (a) (coinciding with the distance between two adjacent fins, (b), distance between two adjacent blocks of spines (w), surface configuration, and, finally, the ratio R between the total heat transfer area, A_{ext}, of the extended surface and the base area, A_{b}.

The experiments were conducted in steady-state conditions and the following procedure was employed. Before each test a degassing process was carried out and the heating surface was left immersed in the saturated liquid overnight, with nil heat flux. Experiments were then run with the heat flux being increased in small steps. When conditions of heat flux close to the critical value had been achieved (that flux is hereafter mentioned as “maximum heat flux”), the voltage applied to the heater was reduced by steps of the same order.

Each extended surface was tested for horizontal orientation of the base surface (vertical fin orientation). In all tests, the liquid level was maintained at approximately 80 mm above the boiling surface. Boiling curves were obtained at three different saturation pressures: 0.5, 1.0 and 2.0 bar.

3. Results and discussion

Before proceeding to the presentation and discussion of the experimental results, some preliminary remarks should be made:

(1) The pool boiling curves of highly wetting dielectric fluids involve the phenomena of temperature overshoot and

hysteresis effects associated with the onset of nucleate boiling and with the progressive activation (or deactivation) of nucleation sites. The boiling curves obtained by increasing or decreasing heat fluxes are therefore different. This hysteresis effect, previously observed for plane surfaces by several authors [18], was noted during the present experiments for all surfaces tested. This paper reports only boiling curves at decreasing heat flux, starting from the maximum heat flux value, because these curves seem to describe more properly the behaviour of heat sinks in operating conditions.

(2) All of the data obtained have been analysed in such a way as to evaluate both the pool boiling heat transfer performance of the heat sinks and, as far as possible, the effects of each single parameter that was varied during the experiments.

The heat transfer rate, q, transmitted by the flux meter refers both to the base (projected) area of the extended surface, A_b, and to the total heat transfer area (fins and free base) of the extended surface, A_{ext}, thus defining the following heat fluxes:

– heat flux based on base surface area

$$q'' = \frac{q}{A_b} \tag{1}$$

– heat flux based on total surface area

$$q''_{ext} = \frac{q}{A_{ext}} \tag{2}$$

These heat fluxes were then related with ΔT_{sat}, i.e., the difference between the temperature of the base surface, T_w (obtained by extrapolating the temperature profile inside the flux meter) and the saturation temperature of the fluid, T_{sat}. To analyse the heat transfer effectiveness, the overall heat transfer coefficient, h_{ext}, was then defined as equal to:

$$h_{ext} = \frac{q}{A_{ext}\Delta T_{sat}} = \frac{q''}{R\Delta T_{sat}} \tag{3}$$

where R represents the ratio between A_{ext} and A_b, as reported in Table 1.

The overall heat transfer coefficient h_{ext} does not represent the convective coefficient averaged on total surface A_{ext}, as it also takes into account the temperature profile along the fins. However, it allows us to describe the overall behaviour of the heat sink and, at the same time, the effect of some geometrical parameters.

The results of the experimental tests are presented in diagram form, in order to show the effects produced on boiling phenomena by changes in surface geometry and by variations in pressure.

3.1. Uniformly spaced fins

Fig. 2 reports typical boiling curves (q'', ΔT_{sat}), at the pressure of 1.0 bar, for all extended surfaces with uniformly spaced fins, in the horizontal orientation of the base surface.

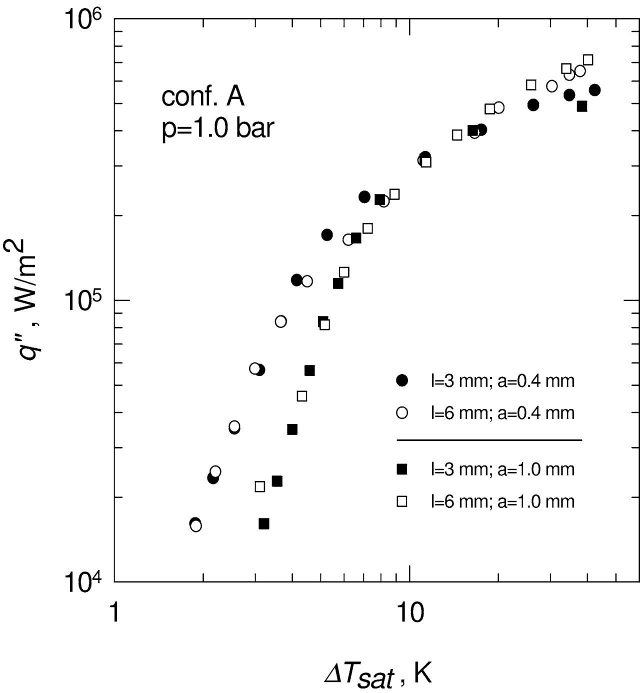

Fig. 2. Boiling curves based on base surface area, at the pressure of 1.0 bar, for extended surfaces No. 1, 4, 7 and 10, with uniformly spaced fins (configuration A).

Fig. 2 shows that, at low heat fluxes, the heat transfer rate increases as fin size decreases from 1.0 mm (surfaces No. 7 and No. 10) to 0.4 mm (surfaces No. 1 and No. 4), because whenever fin width diminishes, the heat transfer area, A_{ext}, increases, as can be observed in Table 1. In the central zone of the boiling curves, the differences among the different finned arrays are negligible, even though the surface areas of the finned arrays tested differ markedly from one another. At high heat fluxes, however, longer spines (6 mm) appear to work slightly better, particularly in proximity to the maximum heat flux. When spine length increases, the maximum heat flux increases, too.

In Fig. 3 the values of overall heat transfer coefficient related to the extended area, h_{ext}, versus wall superheat are plotted. Fig. 3 corresponds to the experimental data reported in Fig. 2. Represented in diagrammatic form (h_{ext}, ΔT_{sat}), the boiling data show a typical maximum trend, which evidences a decline in heat transfer effectiveness at higher base surface superheats. The trend with a maximum can be explained by the fact that boiling intensifies as the wall temperature rises, becoming gradually more efficient until large clusters and accumulations of vapour form among the spines; at this point, the overall heat transfer coefficient begins to decline. The maximum value of the overall heat transfer coefficient depends on the width and length of fins. The ΔT_{sat} value that corresponds to the peak value of the overall heat transfer coefficient varies with fin geometry, too, and corresponds to the wall superheat that determines the slope change in the boiling curves.

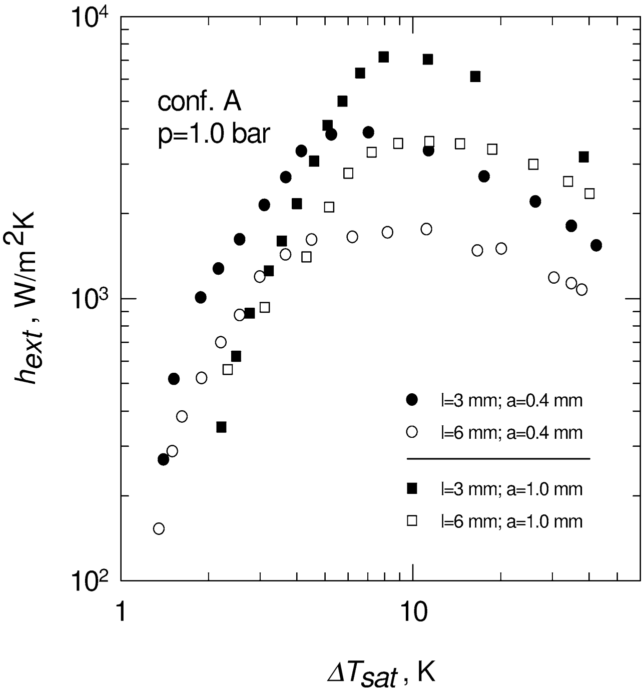

Fig. 3. Overall heat transfer coefficient at the pressure of 1.0 bar, for extended surfaces No. 1, 4, 7 and 10, with uniformly spaced fins (configuration A).

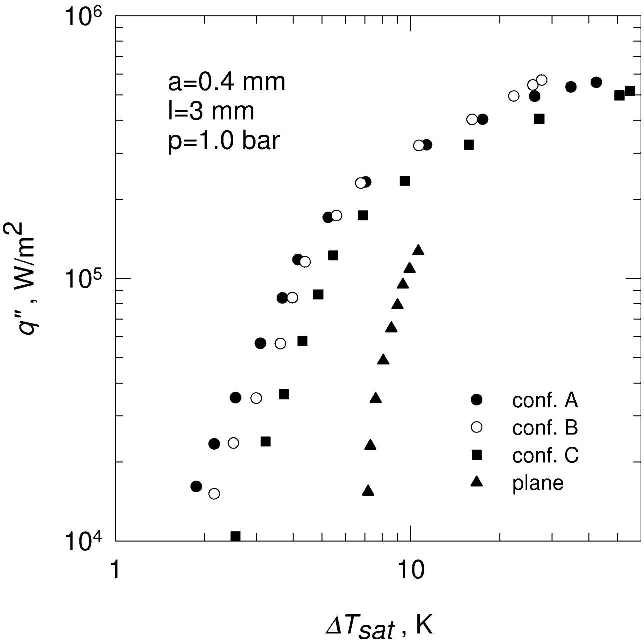

Fig. 4. Boiling curves based on base surface area, at the pressure of 1.0 bar for plane and extended surfaces No. 1, 2 and 3, with uniformly and non-uniformly spaced fins (configuration A, B and C).

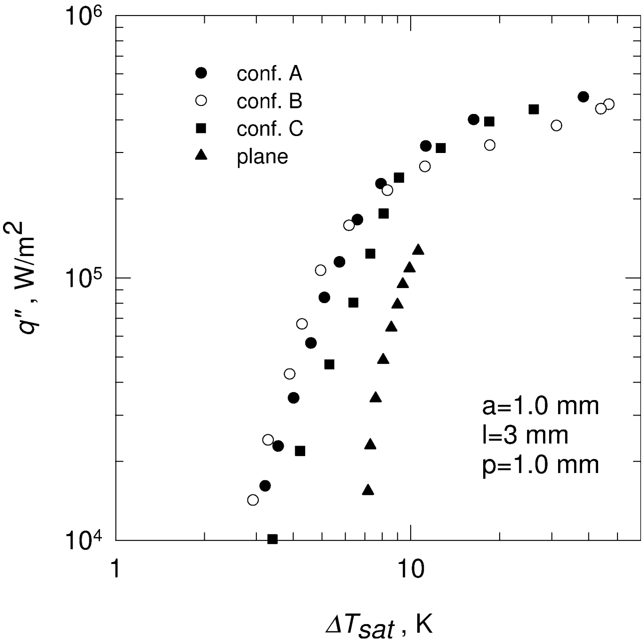

Fig. 5. Boiling curves based on base surface area, at the pressure of 1.0 bar for plane and extended surfaces No. 7, 8 and 9 with uniformly and non-uniformly spaced fins (configuration A, B and C).

At low base surface superheats, heat transfer performances, expressed in terms of the overall heat transfer coefficient, are quite similar for the various spine geometries, and prove to be slightly better for thin fins (No. 1 and No. 4). By contrast, at higher heat fluxes, fin width and spacing appear to influence boiling heat transfer efficiency: for the same fin width, h_{ext} is greater for shorter fins.

The effect of increasing fin length is always a reduction in h_{ext}, but this effect is more marked for thin fins. When fin lengths are equal, h_{ext} is greater for surfaces with thicker fins (side 1 mm). Thermal conduction along thicker fins is more effective, and the residence time of the vapour is reduced owing to the effect of the bigger spacing; therefore, when fin width and gap are increased, better flow conditions occur and the wetting of the surface by the fluid is made easier, as suggested by Rainey and You [19].

At all pressures investigated during the experiments, similar results were obtained.

Further results regarding the effects of fin geometry on boiling for uniformly spaced finned arrays are reported in [8].

3.2. Non-uniformly spaced fins

Several tests were performed on arrays with fins non-uniformly spaced, obtained by regularly and orthogonally eliminating some rows of fins from uniformly spaced arrays, as previously described.

Figs. 4 and 5 report boiling curves based on base surface area (q'', ΔT_{sat}) for extended surfaces with fins 3 mm long and, respectively, 0.4 mm or 1.0 mm wide. In both figures, experimental data obtained for uniformly spaced fins (configuration A) are compared with those obtained for non-uniformly spaced fins (configuration B and configuration C). All finned arrays were tested at the pressure of 1.0 bar and for horizontal orientation of the base surface.

Comparison of the various curves indicates that generally the boiling performance of surface configuration A is better than that of configuration B, which is, in turn, better than that of configuration C. In general, a slight worsening of the heat transfer rate can be observed as the fins are thinned out. Only modest variations in boiling behaviour, however, occur on passing from one configuration to another.

The analysis of the boiling data for the three surface configurations obtained at the pressure of 0.5 bar and 2.0 bar and for spines of different sizes, seems to lead to similar conclusions: thinning out the spines yields slightly lower heat transfer rates, even in the higher fluxes boiling region. The pertinent diagrams are not reported here for reasons of space.

It is interesting to observe that the clearance between fins is connected with the possible formation of large clusters and accumulations of vapour among fins. As the critical gap between fins is connected with vapour generation, it has been proposed that the critical clearance is correlated with the break-off diameter of bubbles [20]. Park and Bergles [21] observed for boiling R-113 that, although heat transfer performance did not vary much with fin gap, the best performance could be obtained with a heat sink having a fin gap of 0.49 mm. For arrays of longitudinal fins, Hirono et al. [3] noticed the existence of an optimum fin spacing; when fin spacing was reduced below that value the peak heat flux decreased.

Previous remarks about Figs. 4 and 5 can be verified in Figs. 6 and 7, in which the values of the overall heat transfer coefficient related to the extended area, h_{ext}, versus heat flux based on total (fin and base) finned surface area, q''_{ext}, are plotted, in comparison with plane surface data.

Figs. 6 and 7 correspond to the experimental data reported in Fig. 4 and Fig. 5, respectively. Fig. 6 shows typical increasing-decreasing curves with the peak value in the range 2.5–5 $\times$ 10^4 W·m^{-2}, above which the overall heat transfer coefficient decreases. For spines of the same length (3 mm) but 1 mm thick (Fig. 7), the peak value of h_{ext} is about twice as high and is obtained at heat fluxes in the range 5.5–10 $\times$ 10^4 W·m^{-2}.

Similar trends are shown by the curves obtained for spines 6 mm long (Figs. 8 and 9), with peak values of h_{ext} in the range 1–2.5 $\times$ 10^4 W·m^{-2} and 4–6 $\times$ 10^4 W·m^{-2}, respectively, for widths of 0.4 and 1.0 mm.

Below the heat flux that corresponds to the h_{ext} peak values, the finned surfaces show an appreciably greater overall heat transfer coefficient than the flat surface. Rainey and You [19] explained this effect by supposing a larger number of active nucleation sites on finned surfaces than on the flat surface, created by the machining of the fins. At higher heat fluxes, the flat surface shows higher overall heat transfer coefficient than the extended surfaces.

The diagrams $(h_{\text{ext}}, \Delta T_{\text{sat}})$ relative to non-uniformly spaced finned arrays (not reported here for the sake of space) show that transition from an increasing trend to a decreasing one corresponds to the slope change which occurs in the boiling curves of Figs. 4 and 5, similarly to what was previously observed for uniformly spaced fins. For saturated boiling of a highly wetting fluid and a plane surface, Heindel et al. [22] identified two boiling regions between the departure from natural convection (DNC) and the critical heat flux: the two zones were characterised by different slopes of the boiling curve and the transition point was defined as de-

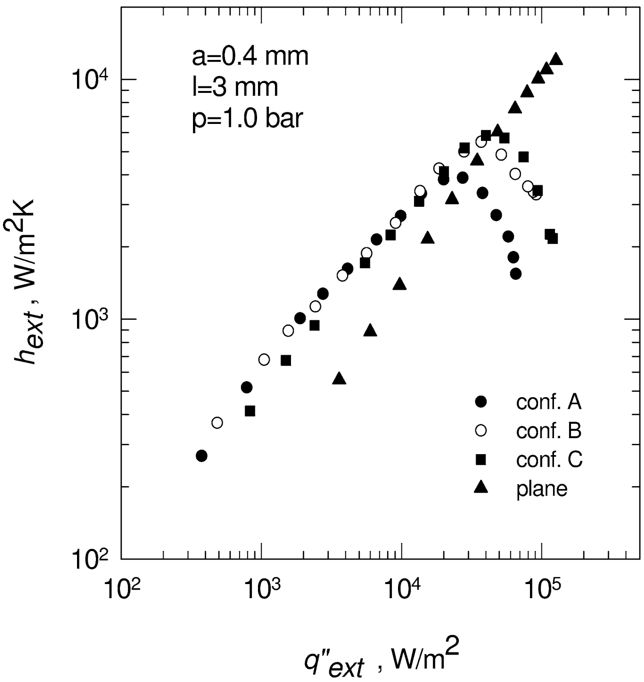

Fig. 6. Effect of surface configuration on overall heat transfer coefficient, at the pressure of 1.0 bar for extended surfaces No. 1, 2 and 3.

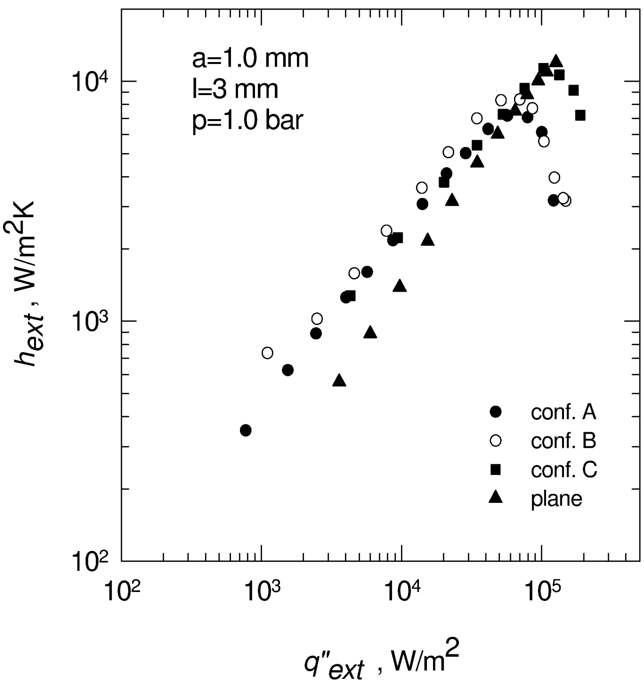

Fig. 7. Effect of surface configuration on overall heat transfer coefficient, at the pressure of 1.0 bar for extended surfaces No. 7, 8 and 9.

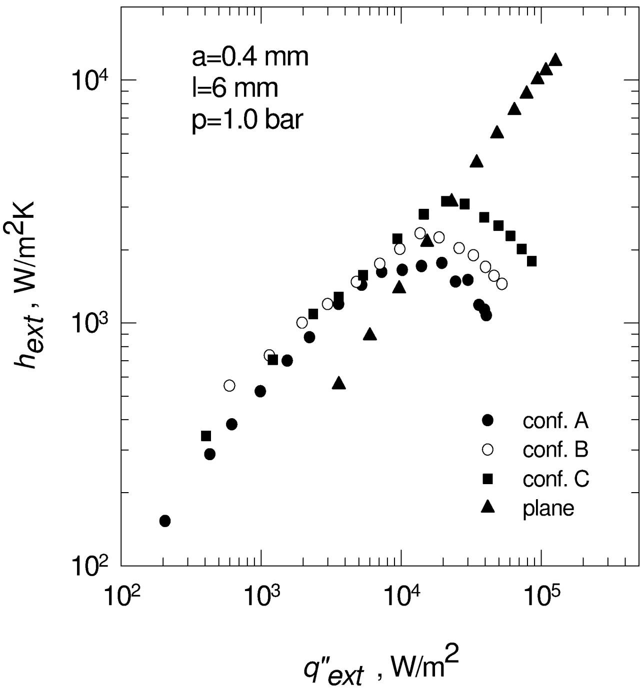

Fig. 8. Effect of surface configuration on overall heat transfer coefficient, at the pressure of 1.0 bar for extended surfaces No. 4, 5 and 6.

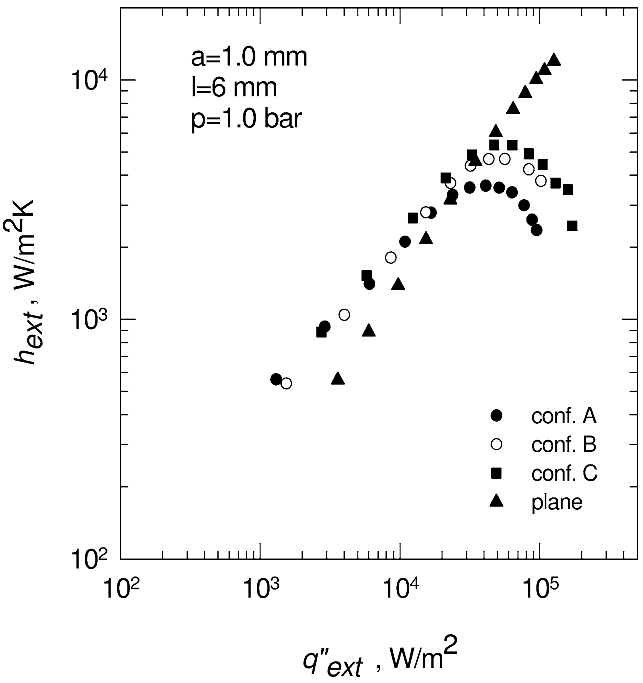

Fig. 9. Effect of surface configuration on overall heat transfer coefficient, at the pressure of 1.0 bar for extended surfaces No. 10, 11 and 12.

viation from nucleate boiling. Carvalho and Bergles [23], describing the boiling flow patterns on vertically oriented small heaters, highlighted a point of transition from nucleate boiling called deviation from nucleate boiling (DNB). That point marks a transition characterised by a deterioration of the heat transfer mechanism due to the agglomeration of vapour on

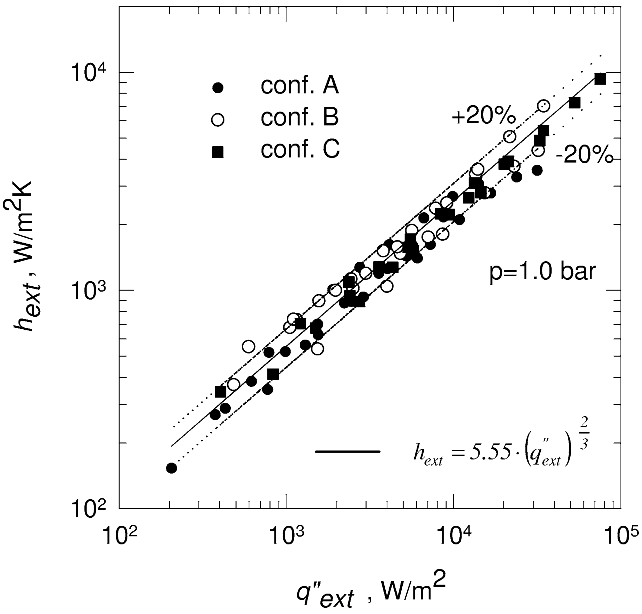

Fig. 10. Overall heat transfer coefficients h_{ext} versus q''_{ext} for different surface geometries and configurations.

the surface. Although the surface configuration in the present experiments is very different from those considered in [20, 23], a similar trend in boiling curves was noted.

All tests performed on the twelve finned arrays indicate that thinning out the fins did not lead to higher heat transfer rates, even in the boiling region close to the burn out.

The better wetting of the boiling surface by the liquid refrigerant, as a result of the sparser grouping of spines, is apparently offset by the reduction in the heat transfer area. However, the overall heat transfer coefficient increased as the fins were thinned out; this result proves that the effectiveness of finned surfaces rises when spines are more sparsely set, while the efficiency does not improve. In particular, thinning out the fins leads to greater overall heat transfer coefficients at high heat fluxes, while h_{ext} values are closely grouped for the different configurations of extended surfaces at low heat fluxes.

At low heat fluxes the three configurations (A, B, C) yield practically the same values of h_{ext}, meaning that the heat transfer rate mainly depends on the area of the heat transfer surface of the heat sink. If we consider only the low heat flux region, at q''_{ext} lower than values which determine the maximum of the curves in Figs. 6–9, then it will be seen that all experimental data, at the pressure of 1 bar, lie on a single straight line, which is represented by the following equation:

$$h_{\text{ext}} = 5.55 q''^{2/3}_{\text{ext}} \tag{4}$$

as shown in Fig. 10. Eq. (4) can also be written in the form:

$$q''_{\text{ext}} = 170 \Delta T^3_{\text{sat}} \tag{5}$$

which evidences the dependence on the wall superheat. The exponent which appears in Eq. (5) is the same as that which appears in Rohsenow's correlation for nucleate boiling [24].

It is important to point out that the range of validity of Eq. (4) and Eq. (5), obtained for the pressure of 1 bar, is different according to the surface geometry. As the deviation from nucleate boiling occurs for different heat fluxes, and therefore for different walls superheats, according to the finned arrays geometry the range of validity of Eq. (4) and Eq. (5), which extends along the nucleate boiling region up to the DNB point, varied with the extended surface geometry. In the range of validity, the experimental data are fitted by Eq. (4) with an uncertainty of ±20%.

3.3. Effect of pressure

In 1952 Rohsenow [24] proposed an empirical correlation for nucleate boiling under different pressures. You et al. [13] applied that correlation to FC-72 and observed a good agreement between their experimental data for different pressures and Rohsenow's equation. Cooper's correlation [25] explicitly considers the dependence of the heat transfer coefficient on the reduced pressure. Webb and Pais [16] confirmed the ability of this correlation to predict the boiling of highly wetting fluids at different pressures on plane surface. On investigating square pin-fin arrays in saturated R-113, Abuaf et al. [26] found that the boiling curves moved to higher base surface superheats as pressure decreased. Guglielmini et al. [27] also described a similar dependence of boiling curves on pressure for the same extended surface geometry.

An effective approach to this issue was proposed by Gorenflo in the VDI Heat Atlas method [28], which examines the influence of the main groups of variables on boiling heat transfer separately: heat flux, microstructure of heating surface, material of heating surface and, finally, saturation pressure in terms of reduced pressure. Gorenflo [29] derived from measurements on halocarbon refrigerants the following expression for the reduced heat transfer coefficient on single plain tubes:

$$h/h_0 = F_q \cdot F_p \cdot F_{\mathrm{WR}} \cdot F_{\mathrm{WM}} \tag{6}$$

with

$$F_q = (q/q_0)^{n(p/p_c)}, \qquad n = 0.9 - 0.3(p/p_c) \tag{7}$$

and

$$F_p(p/p_c) = 1.2(p/p_c)^{0.27} + 2.5(p/p_c) - (p/p_c)/\big((p/p_c) - 1\big) \tag{8}$$

where F_q is the heat flux factor, F_p is the pressure factor and p_c represents the critical pressure. F_{WR} and F_{WM} are terms that contain the influence of the microstructure or material properties, respectively, of the heating surface. The influence of the thermophysical properties of the boiling liquid is contained in the reference value h_0 under reference conditions for the heat flux (q_0) and the reduced pressure. The effects of the saturation pressure p are taken into account by F_p and, in lesser extent, by the index n that decrease steadily with arise in pressure. For finned tubes [28] the value obtained for the index n was less than for plain tubes, and the effect of pressure in the term F_p was weaker.

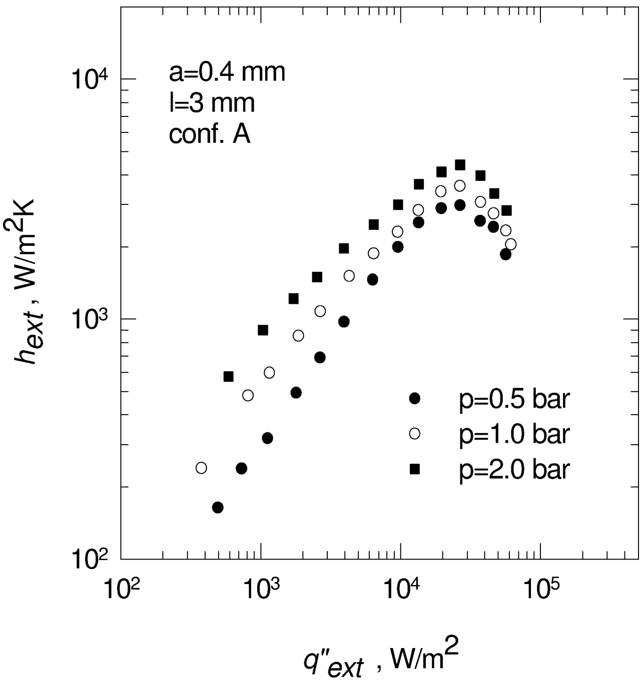

Fig. 11. Effect of pressure on overall heat transfer coefficient.

A similar approach was followed by Nishikawa et al. [30], who proposed a correlation in which the relative influences of the main groups of variables are treated separately. The effect of pressure for freons was expressed by means of the relationship:

$$F_p(p/p_c) = (p/p_c)^{0.23}/\big[1 - 0.99(p/p_c)\big]^{0.9} \tag{9}$$

which is also valid for water and organic fluids.

In the Nishikawa's correlation the heat transfer coefficient results to be proportional to $q^{4/5}$ throughout the entire pressure range investigated ($0.03 < p/p_c < 0.98$).

The tests carried out confirm that heat transfer increases as pressure rises. Fig. 11, for example, shows the curves ($h_{\mathrm{ext}}, q''_{\mathrm{ext}}$) for surface No. 1 at three different pressures: 0.5, 1.0 and 2.0 bar. The effect of the saturation pressure is more marked at low heat fluxes and tends to lessen at high heat fluxes. Indeed, at high heat fluxes, the fluid-dynamic interactions produced by the presence of the fins could modify the pressure dependence determined by the variation in the physical properties of the fluid. Pressure influences how vapour is formed and develops, but, at high heat fluxes, this process is also influenced by the geometry of the extended surface.

If we limit our observation to the low heat flux region, in which the effect of the geometry on h_{ext} is modest, we notice that the dependence of the heat transfer coefficients on reduced pressure can be represented by the above equations. In Fig. 12 the ratio $h_{\mathrm{ext}}/(h_{\mathrm{ext}})_{\mathrm{Eq.\ (4)}}$ evaluated at a certain heat flux q''_{ext}, in which the denominator is the overall

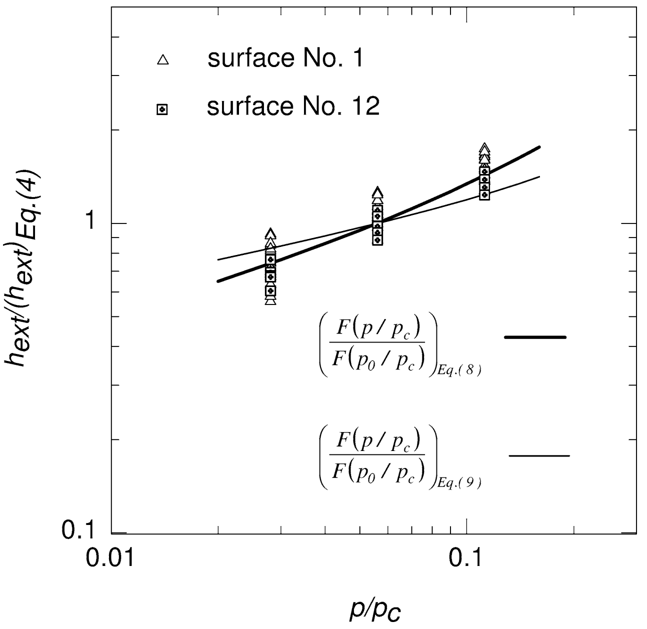

Fig. 12. Effect of pressure on boiling heat transfer for finned arrays.

heat transfer coefficient calculated by means of Eq. (4), is reported as a function of (p/p_c). In the range of heat flux values for which Eq. (4) is valid, the experimental data have been compared with the ratio $[F_p(p/p_c)/F_p(p_0/p_c)]$ calculated by means of Eqs. (8) and (9), assuming p_0 to be 1 bar. As can be seen from Fig. 12, Eq. (8) correlates the experimental data with a maximum uncertainty of $\pm 20\%$. The comparison in Fig. 12, indeed, neglects the influence of the reduced pressure on the exponent n in Eq. (7) and therefore on the heat flux factor F_q; nevertheless it confirms that factor F_p is able to properly express the dependence on pressure of heat transfer coefficient, particularly in the case of the extended surfaces, when the dependence of n on p/p_c is small.

Therefore, in those conditions in which surface geometry has little influence on the heat transfer coefficient, the dependence on pressure is well described by the correlations obtained for smooth plane and cylindrical surfaces.

4. Conclusions

This paper reports an experimental study on the saturated boiling of FC-72 on twelve different patterns of finned arrays, at various pressures. Analysis of the experimental results leads to the following conclusions:

(1) In the case of extended surfaces composed of uniformly spaced fins, longer fins appear to work slightly better, particularly in proximity to the maximum heat flux. When fin width and spacing decrease, the heat transfer rate increases, but, at high heat fluxes, the overall heat transfer coefficient diminishes.

(2) Represented in the diagrammatic forms $(h_{ext}, \Delta T_{sat})$ or (h_{ext}, q''_{ext}), the boiling data show a typical maximum trend, which evidences a degradation in heat transfer at higher base surface superheat. The maximum values of the overall heat transfer coefficients depend on fin geometry and surface configuration, increasing with thickness and space and decreasing with length.

(3) At low heat fluxes, the finned surfaces show an appreciably higher overall heat transfer coefficient than the flat surface. In these conditions the heat transfer efficiency is quite similar for the various arrays of fins, whether uniformly or non-uniformly spaced. At high heat fluxes, fin width and surface configuration appear to influence boiling heat transfer efficiency strongly.

(4) The dependence of the overall heat transfer coefficient on q''_{ext}, at the pressure of 1.0 bar, for low heat fluxes is ascertained; h_{ext} does not substantially depend on the surface geometry.

(5) Thinning out does not produce any noticeable result in terms of heat transfer rate: only a slight reduction is observed as fin spacing increased. The better wetting of the boiling surface by the liquid refrigerant, as a result of the sparser grouping of fins, is apparently offset by the reduction in heat transfer area.

(6) The effect of the thinning out of fins on heat transfer effectiveness, expressed by means of the overall heat transfer coefficient h_{ext}, is negligible at low heat fluxes, but becomes evident at high heat fluxes, where the heat transfer efficiency is enhanced by thinning out fins.

(7) Increasing the pressure improves heat transfer performance and raises maximum heat flux. This phenomenon was also noticed in the case of non-uniform finned arrays.

(8) For tested extended surfaces, the influence of pressure on boiling heat transfer has been correlated by means of the relationships proposed by Gorenflo and Nishikawa et al. for plane surfaces and smooth tubes; this yields an efficacious description of the pressure effect at low heat fluxes. At high heat fluxes the fluid-dynamic interactions produced by the presence of the fins alter the law of dependence on pressure.

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