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FC72 and FC87 nucleate boiling inside a narrow horizontal space

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ABSTRACT

This paper presents new experimental results for saturated nucleate boiling of FC72 and FC87 on a horizontal copper disc, at atmospheric pressure, for different degrees of confinement, *s*, in the range of 0.1–13 mm, and with two kinds of confining element, for low and moderated heat fluxes ($\leqslant$40 kW/m^2), on both a downward and an upward facing heating surface. For low heat flux a decrease of the confinement gap causes an enhancement of the boiling and a decrease in the dryout heat flux. A visualization of the boiling phenomenon shows the effect of confinement and heat flux on the liquid–vapor configuration.

1. Introduction

Boiling is of great interest in all applications involving high heat flux. Because of its high heat transfer, nucleate boiling enables the transfer of a large amount of energy without requiring a strong superheat. Boiling in a narrow space occurs in many industrial applications, such as in the cooling of electronic devices, in micro-heat exchangers and in heat spreaders. The first studies focusing on confined boiling were performed by Katto et al. [1] and Ishibashi and Nishikawa [2]. The former focused on the confined boiling on a horizontal plate, and showed that confinement enhances the boiling heat transfer at low heat flux while it causes a decrease in the critical heat flux (CHF). In the latter publication, the authors presented a detailed analysis of the effect of pressure, surface tension and confinement for pool boiling in an annular vertical channel. The authors also showed that the confinement leads to an increase in heat transfer at low heat fluxes. Furthermore, they established the existence of two regimes depending on the Bond number, *Bo*, defined as the ratio of the gap, *s*, between the heating surface and the unheated surface, and the capillary length, L_c:

$$Bo = \frac{s}{L_c} \quad (1)$$

with,

$$L_c = \left[\frac{\sigma}{g(\rho_l - \rho_v)}\right]^{1/2} \quad (2)$$

where σ, g, ρ_l and ρ_v represent the surface tension, the acceleration due to gravity, the vapor density and the liquid density, respectively. Also, Ishibashi and Nishikawa showed that the enhancement of heat transfer due to confinement becomes stronger when $Bo < 1$, meaning that the gap *s* is equal to or smaller than the bubble detachment diameter.

The effect of pressure was also studied by Bonjour et al. [3] for vertical and horizontal configurations, and the authors found, in agreement with Ishibashi and Nishikawa [2], that the enhancement of heat transfer due to pressure in unconfined boiling tends to disappear as the confinement increases. Ishibashi and Nishikawa [2] also showed that the surfactant effect disappears with the confinement, a result confirmed later by Hetsroni et al. [4]. The effect of subcooling was investigated by Passos et al. [5], who showed that while in saturated boiling the confinement has a positive effect on heat transfer, its effect is negative in subcooled boiling: the subcooling increases the heat transfer for unconfined boiling at low heat flux while its influence is much lower in the confined case and heat transfer is higher during unconfined subcooled boiling than during confined subcooled boiling. The effect of the active nucleation site density was also studied by Bonjour et al. [3] and the authors showed that while in unconfined boiling a single active nucleation site does not enhance heat transfer compared to natural convection, this enhancement of the heat transfer by a single nucleation can clearly be seen in confined boiling.

The boiling patterns and regimes in confined boiling were studied in detail by Bonjour and Lallemand [6] for a vertical heating surface. According to this study, boiling before the CHF in a confined space presents three different regimes: isolated deformed bubbles at low heat flux, coalesced bubbles at moderate heat flux,

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Nomenclature

D_b	detachment diameter of vapor bubbles (mm)
e	thickness of the vapor layer on the confining element (mm)
g	acceleration due to gravity (m/s^2)
h	heat transfer coefficient (kW/m^2 °C)
h_{lv}	latent heat of vaporization of the liquid (kJ/kg)
k_l	conductivity of the liquid (W/mK)
L_c	capillary length (mm)
N_a	the density of active nucleation sites
q	heat flux (kW/m^2)
r_c	radius of cavity (μm)
S_i	the mean surface area influenced by bubbles
s	degree of confinement (mm)
T_{sat}	temperature of saturated pool liquid (°C)
T_w	wall temperature (°C)
θ	dimensionless temperature
ρ_v	vapor density (kg/m^3)
ρ_l	liquid density (kg/m^3)
α	contact angle (°)
η_{eb}	proportion of the surface influenced by bubbling
δ	thickness of the thermal layer (mm)
σ	surface tension of the liquid (J/m^2)
ΔT_{sat}	temperature difference (°C)

and partial dryout at the highest heat flux. It was shown that the ranges of heat flux at which the different regimes are observed decrease with the Bond number. However, Cardoso et al. [7] and Katto et al. [1] reported a different behavior dependence for small Bond numbers: for $Bo < 1$ there is a thin liquid film regime due to the fast growth of bubbles, and this regime is not observed for $Bo > 1$.

The enhancement of heat transfer is generally attributed to two different mechanisms: enhanced microlayer evaporation (Stutz et al. [8], Utaka et al. [9], Katto et al. [1]) or enhanced liquid agitation (Ishibashi and Nishikawa [2], Fujita et al. [10]). The former is due to the fast bubble growth forming a microlayer with a large surface. This enhanced mechanism is generally cited in studies carried out on a horizontal surface and in a very narrow space (typically when the bubbles have a long residence time in the confined space). The latter is due to the coalescence phenomenon and displacement of the bubbles along the surface and is generally reported for a vertical heating surface. However, according to Stutz et al. [8], this mechanism also plays a role in the case of a horizontal heating surface when the heat flux approaches the critical heat flux.

Thus, the orientation of the heating surface plays a role in the importance of these two mechanisms. Nishikawa et al. [11] demonstrated that for low to moderate heat flux, and inclination angles lower than 120° (0° indicating that the heater is horizontal and upward facing), the heat transfer is controlled by the agitation of isolated vapor bubbles, and for angles greater than 150° the vaporization of the liquid layer plays a more significant role. Guglielmini et al. [12] analyzed the combined effects of surface orientation and confinement on the nucleate pool boiling and critical heat flux of HFE7100 on a smooth copper surface. At low wall superheats, for $Bo > 1$, the effect of confinement was negligible for all surface orientations, while for $Bo \approx 1$ and angles of 0° (upward facing surface) and 45°, heat transfer was enhanced. At high wall superheating and with gaps of 3.5, 2 and 1 mm, the heat transfer coefficient and CHF decreased as the channel width decreased. This effect was more evident for 0° and 45°. Kannengieser et al. [13] showed that parameters like gravity or subcooling can modify considerably the behavior of the boiling mechanism without, however, modifying significantly the overall heat transfer. Nevertheless, they explained also that the temperature at the transition from one regime (natural convection, isolated bubble regime, fully-developed boiling, etc.) to another is influenced by these parameters which can lead to a significant difference in the heat transfer.

The maximum heat flux, represented by the critical (CHF) or dryout (DHF) heat flux, allowing a system to operate in nucleate boiling can be highly dependent on the confinement and the general trend is that it decreases with a decrease of the confinement gap, as reported by Bonjour and Lallemand [14].

In this study, the boiling of FC72 and FC87 for different degrees of confinement on a horizontal upward facing (UF) heater and on a downward facing (DF) heater was investigated in order to better understand the mechanism responsible for boiling heat transfer in a narrow space. Also, the boiling heat transfer of FC72 is compared to that of FC87, dielectric fluids which have the same physical properties differing only in terms of the surface tension: $\sigma_{FC72} = 0.008$ N/m, $\sigma_{FC87} = 0.014$ N/m.

2. Experiment

The test section consists of a copper disc with a 12 mm diameter and 1 mm thickness, with three type-E thermocouples set in the disc close to its center. The copper disc is heated by an 11.8 Ω electrical resistance skin heater fixed by Araldite® epoxy resin to one side of the disc. This disc, in turn, is fixed to a piece of PVC beveled to an angle of 45° with an outside diameter of 20 mm. For the case where the test section is mounted with a downward facing heating surface, the PVC support is mounted on the end of an aluminum tube which the thermocouple cables pass through. This ensemble is mounted inside a boiling vessel which is placed inside a second vessel, called the external chamber, as shown in Fig. 1. The lateral walls of the external chamber are transparent plexy-glass plates, allowing the lateral visualization of the boiling space, and the upper and lower bases consist of two aluminum plates. Mounted in the lower aluminum base there is a rectangular transparent plexy-glass window, allowing the visualization of the boiling phenomenon on the copper disc.

The boiling vessel is filled with 150 ml of a dielectric fluid, FC72 (C_6F_{14}) or FC87 (C_5F_{12}), at atmospheric pressure. Two other type-E thermocouples are placed inside the boiling vessel in order to measure the liquid and the vapor temperatures. The surface of the copper disc which is in contact with the working fluid was polished using 600-grit emery paper, corresponding to a roughness, R_a, of 1.1 μm. The distance between the copper disc and the plexy-glass window can be adjusted by turning the aluminum tube and is controlled by means of a dial. For a downward facing heating disc the geometric factor of the confining element is the PVC support of the copper disc, with a 45° bevel, and this configuration of the test section will be denoted as DF45° (Fig. 1). The working fluid (FC72 or FC87) is heated up to the saturation temperature by water flowing in the external chamber, whose temperature is controlled by a cryostat (LAUDA RK20 KP).

For the case where the test section is mounted with an upward facing heating surface, the confinement of the boiling space was

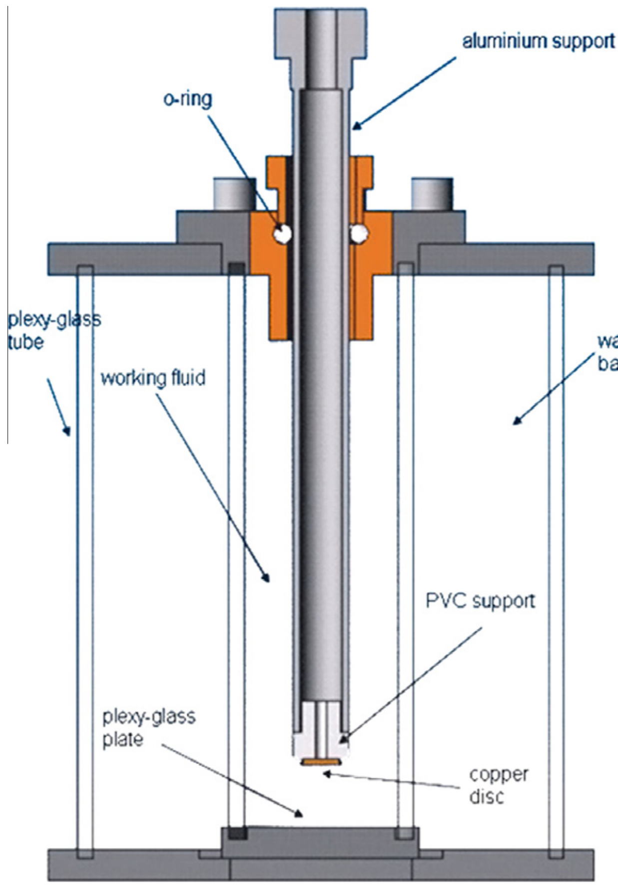

Fig. 1. Scheme of the experimental setup (DF45°).

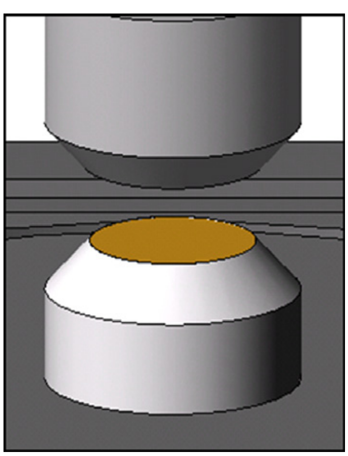

Fig. 2a. Scheme of the PVC confinement element and the test section (UF45°).

imposed by a PVC element, with a 20 mm outside diameter, and with a 45° bevel or without beveling, as shown in Fig. 2a (UF45°) and Fig. 2b (UF90°), respectively, mounted on the end of the aluminum tube, where the distance from the copper disc is controlled as in the downward facing case.

The thin gap between the periphery of the copper disc and the PVC support is filled with Araldite® epoxy resin, however, this is not sufficient to avoid the presence of natural parasite sites at the periphery of the copper disc. Moreover the polishing treatment of the copper surface, after the boiling tests, can contribute to creating new parasite nucleation sites. This can adversely affect the quality of the experimental results.

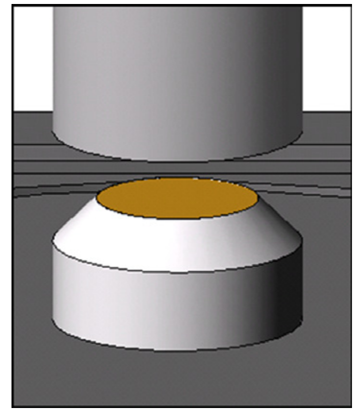

Fig. 2b. Scheme of the PVC confinement element and the test section (UF90°).

The DC power supply, HP6030A, is connected to the skin heater and controlled by a PC using LABVIEW and the acquisition and initial treatment of the data are carried out with an HP34970A system. In this study the heating mode involves increasing the heat flux.

The uncertainty levels associated with the temperature and heat flux are ±0.6 °C and 2%, respectively. The experimental uncertainty for the heat transfer coefficient, in nucleate boiling, varies from 2% to 12%.

Table 1 shows the different degrees of confinement and the corresponding Bond number for the working fluids that were analyzed in this study.

3. Results

3.1. Boiling regime

Fig. 3 shows the partial boiling curves for FC72 and $s = 0.1$, 0.2, 0.3, 0.4, 0.5, 1 and 13 mm, for the upward facing heating surface and the confining element with a 45° bevel (UF45°). For heat fluxes between 2.5 kW/m^2 and 25 kW/m^2, the heat transfer is higher in confined than unconfined boiling. As a general trend (between 10 kW/m^2 and 25 kW/m^2) the heat transfer increases with a decrease in s. For the unconfined case or $s = 13$ mm, the region of heat transfer dominated by natural convection extends to a superheat of 13.5 K. Above this superheat the slope of the boiling curve increases considerably indicating the transition to the nucleate boiling regime. For the confined case, the slope of the boiling curve is elevated at a lower superheat (6 K): the confinement decreases the wall superheating at the transition from the natural convection to the nucleate boiling regime.

For $s = 13$ mm, the Bond number is much greater than unity. For the experiments with the upward facing plate, the bubbles detach from the wall and can coalesce on the confining element to form a layer of vapor. The thickness of this vapor layer is limited by

Table 1
Bond numbers for different degrees of confinement.

	FC72 $L_c = 0.73$ mm	FC87 $L_c = 0.98$ mm
s (mm)	*Bo*	*Bo*
0.1	0.14	0.10
0.2	0.27	0.20
0.3	0.41	0.30
0.4	0.55	0.41
0.5	0.68	0.51
1.0	1.36	1.01
13.0	17.74	13.19

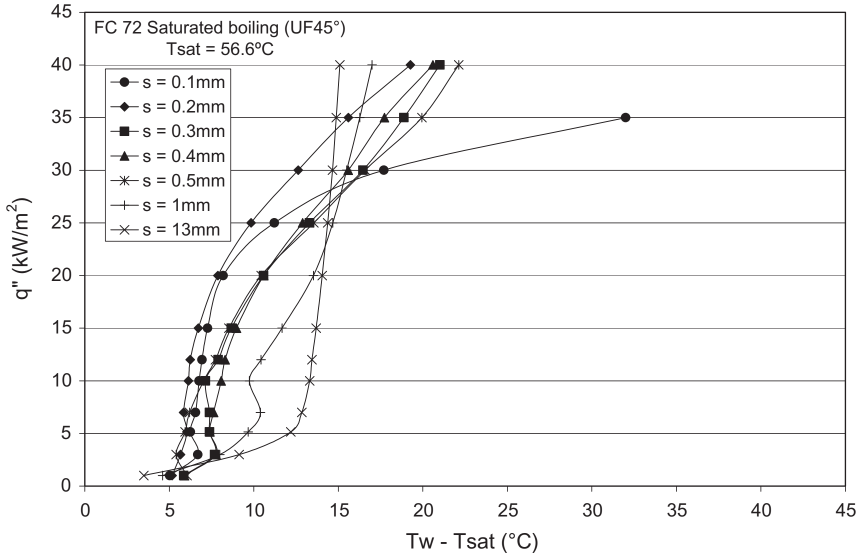

Fig. 3. Partial boiling curves as a function of *s*, for UF45° and FC72.

gravity. Considering a stationary equilibrium (unstable in this case), the thickness of the vapor layer on the confining element, *e*, is given by De Gennes et al. [15]:

$$e = \sqrt{2(1 + \cos\alpha)}L_c \tag{3}$$

where α is the contact angle. Thus, the thickness of such a layer, independently of the contact angle, is less than $2L_c$, which is less than 2 mm considering the capillary length for FC72 and FC87, as shown in Table 1. In our case, the dynamic phenomena can lead to a layer with a thickness larger than this value but will not change the order of magnitude of this thickness. Thus, the interface of such a layer will be far from the wall and their thermal interaction will be very limited.

Nevertheless, we can observe a difference between the boiling curves for experiments UF90° (Fig. 4) and UF45° for *s* = 13 mm. This variation can be partially attributed to the influence of the two different confining elements on the natural convection. At a superheat of 15 K we measured a difference of 20 kW/m^2. Most of this variation in the heat flux appears to be a consequence of the increase in the density of active nucleation sites due to the aging of the surface. It should also be noted that this difference between the boiling curve of experiments UF45° and UF90° disappear for a high degree of confinement in the nucleate boiling regime.

For *s* < 1 mm, the Bond number is less than one and the vapor layer thickness will be limited by the gap size. A microlayer of liquid can be trapped between the surface and this vapor layer. In the model of Cooper and Lloyd [16], the initial thickness of the microlayer is dependent on the velocity profile of the liquid close to the foot of the bubble. This profile varies as a function of the bubble growth dynamics. In confined boiling, once the bubble reaches the size of the gap, its growth becomes bi-dimensional and its foot radius varies considerably over a short time. Moreover the non-slip condition imposed by the confining element will modify the velocity profile of the liquid. Thus, the thickness of the microlayer is affected by confinement and the heat transfer due to microlayer evaporation will not be comparable to that observed when the bubble growth is tri-dimensional.

Therefore, in horizontal confined boiling with an upward facing heating surface, the two boiling regimes differ in terms of the heat transfer mechanism involved. For *Bo* > 1, the layer of vapor formed on the confining element will not interact with the surface heater and the boiling will be comparable to unconfined boiling. For

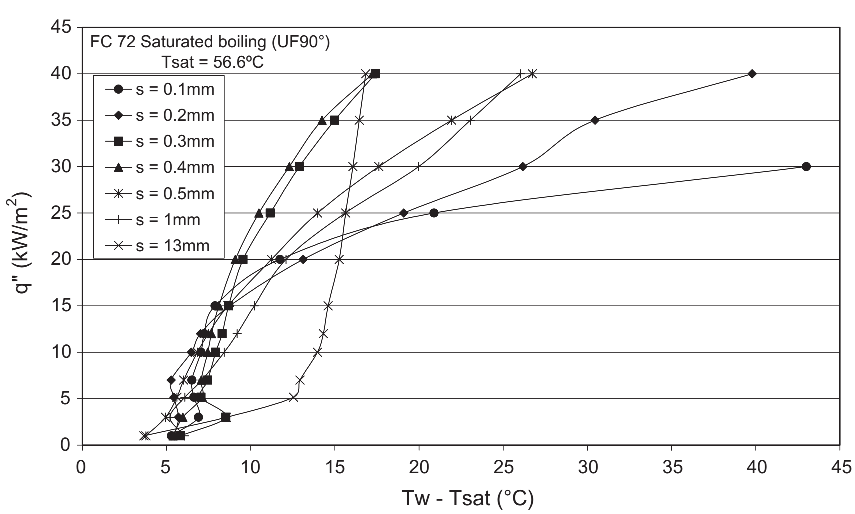

Fig. 4. Partial boiling curves as a function of *s*, for UF90° and FC72.

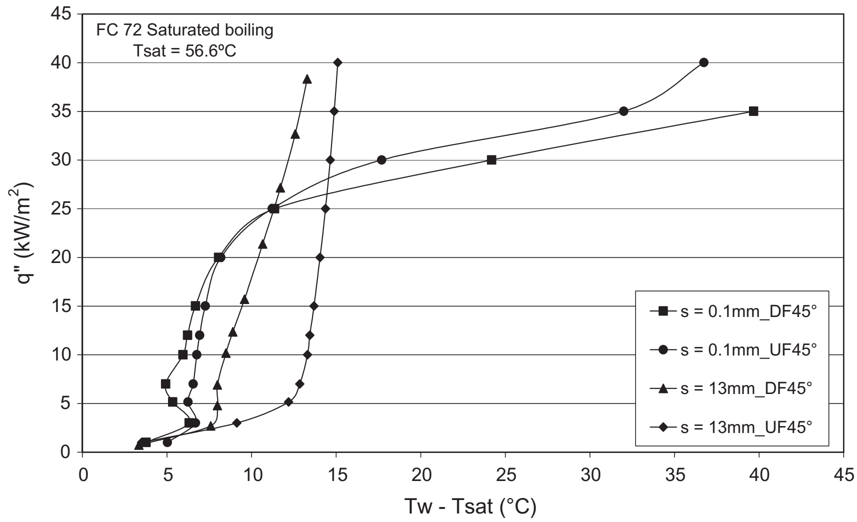

Fig. 5. Effect of heater orientation on the boiling curve.

Bo < 1, the layer of vapor will interact strongly with the heating surface. The dynamic and the shape of this vapor layer will affect the mechanism of heat transfer.

For a downward facing heating surface, the change in the boiling pattern according to the Bond number being above or below one does not appear. In fact, in this configuration gravity works as the confining element by flattening the bubble over the heating surface. The effect of the heater orientation on the boiling curve can be seen in Fig. 5. For $s = 13$ mm, the heat transfer is higher for the downward facing heater than for the upward facing heater. This effect can be attributed to the confining effect due to gravity. In fact, the buoyancy forces acting on the bubble press it against the heating surface increasing the area of the microlayer. For greater confinement, the effect of the heater orientation tends to disappear. Indeed, the smaller is the Bond number, the weaker the confining effect due to gravity.

FC87 has very similar properties to FC72 and these two fluids differ mainly in terms of their surface tension (8 mN/m for FC72 and 14.8 mN/m for FC87). The effect of the confinement described for FC72 was the same in the case of FC87 (see Fig. 6). For all degrees of confinement, the wall superheating at the transition to the boiling regime is lower for FC87 than for FC72, however, this difference is smaller for confined experiments than for unconfined experiments. This vanishing of the influence of the fluid surface properties in confined boiling has also been reported by Ishibashi and Nishikawa [2] and Hetsroni et al. [4] for the surfactant effect.

The surface tension plays a major role in the nucleation and vertical detachment of the bubbles. In confined boiling there is no vertical detachment but the surface tension continues to affect the nucleation temperature. This influence of the surface properties on the nucleation is probably the cause of the lower wall superheating at the transition to the boiling regime observed for FC87, even for $s = 0.1$ mm.

This lower wall superheating at the transition to the boiling regime is a general phenomenon which is accompanied by the enhancement of heat transfer due to the orientation of the heater, the surface properties of the fluid or the degree of confinement. Kannengieser et al. [13] reported that enhancement of the heat transfer due to microgravity was a result of a lower wall superheating at the transition to the fully-developed regime.

3.2. Onset of nucleate boiling

We have seen above that confinement acts on heat transfer by decreasing the wall superheating at the onset of nucleate boiling, *ONB*. The *ONB* can be observed on the boiling curve if a sufficient proportion of the heating surface is influenced by bubbling, assuming that whatever the degree of confinement, bubbling increases

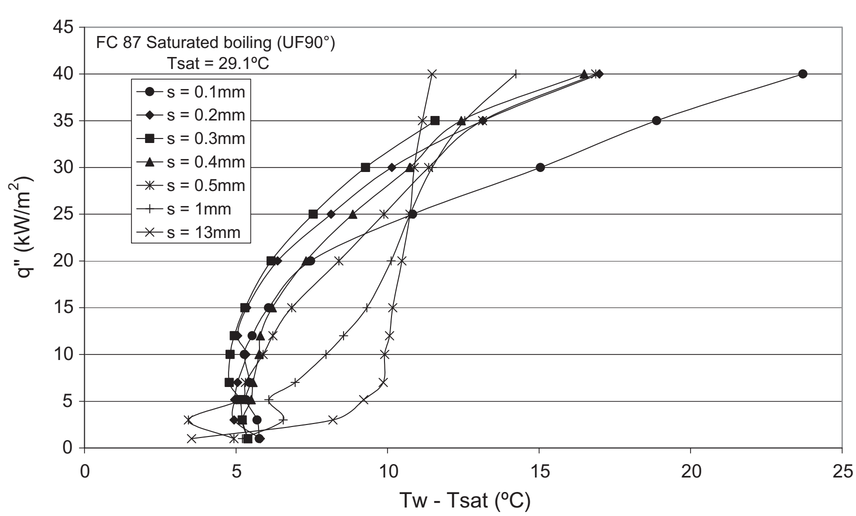

Fig. 6. Partial boiling curves as a function of *s*, for FC87 and UF90°.

the heat transfer. Let η_{eb} be the proportion of the surface influenced by bubbling, N_a, the density of active nucleation sites and S_i the mean surface area influenced by bubbles, thus, we have:

$$\eta_{eb} = N_a S_i \tag{4}$$

Let us define that the boiling transition occurs at $\eta_{eb} = \eta_{tr}$, where η_{tr} is a constant independent of the degree of confinement. Using this definition, the confinement can decrease the wall superheating at the transition if it increases N_a at the same superheat or it increases S_i (or both).

In the theory of nucleation described by Hsu [17], the range of cavity radii that can be activated, r_c^*, is a function of the thickness of the thermal layer, δ:

$$\begin{bmatrix} r_{c,\min} \\ r_{c,\max} \end{bmatrix} = \frac{\delta}{2C_1}\left[\left(1-\frac{\theta_{sat}}{\theta_w}\right) \pm \sqrt{\left(1-\frac{\theta_{sat}}{\theta_w}\right)^2 - \frac{8\sigma T_{sat} C_3}{\rho_v h_{lv} \theta_w \delta}}\right] \tag{5}$$

where $\theta_w = T_w - T_\infty$, $\theta_{sat} = T_{sat} - T_\infty$, C_1 and C_3 are constants dependent on the shape of the cavities and the contact angle, and δ is the thickness of the thermal layer. This thermal layer thickness is a function of the local heat flux. In the non-boiling region where natural convection takes place, this thickness can be estimated using the coefficient of natural convection h_{NC}:

$$\delta = \frac{k_l}{h_{NC}} \tag{6}$$

where k_l is the conductivity of the liquid. The confinement modifies the natural convection coefficient. In our experiment, we obtained $h_{NC} = 290$ W/m^2 K for $s = 13$ mm and $h_{NC} = 185$ W/m^2 K for $s = 0.1$ mm. With these values for the natural convection coefficient, considering $C_1 = C_3 = 1$ and $\theta_{sat} = 5$, we find that:

$$\begin{aligned} &0.9\ \mu\text{m} < r_c > 187\ \mu\text{m}, \quad \text{for } s = 13\text{ mm} \\ &0.9\ \mu\text{m} < r_c > 290\ \mu\text{m}, \quad \text{for } s = 0.2\text{ mm} \end{aligned} \tag{7}$$

The roughness of the heating surface is around 1 μm and the radius of the cavities on the surface must be of the order of 1 μm. The confinement changes very slightly the value of $r_{c,\min}$ which is equal to 0.9 μm while it increases strongly the value of $r_{c,\max}$ which is of the order of 200 μm. This latter extension of the domain of the site that can be activated, given the roughness, will not affect N_a.

The squeezing of the bubbles and the sliding of the bubbles over the surface obviously increases S_i. On the horizontal upward facing surface without confinement, the refrigerant bubbles have a detachment diameter, D_b, of the order of 0.5 mm. The area of influence of these bubbles must be less than $S_i = \pi D_b^2/4K = 1.13$ mm^2 (in this equation $K = 2.4$ is the maximum value, which can be used to compute the area thermally influenced by the bubble given in [18]). In confined boiling, a single bubble can spread over the entire surface. In our experiments the surface covered by the flattened bubbles was of the order of 1 cm^2, 2 orders of magnitude greater than in unconfined boiling. Thus, the increase in the surface area of influence explains the lower wall superheating at the transition to the boiling regime.

The transition to the boiling regime as defined above is based on the boiling pattern and is not related to the heat transfer. Bonjour et al. [3] showed that a single nucleation site enhances the heat transfer in confined boiling. This confirms the assumption that bubbling enhances the heat transfer for all degrees of confinement.

Thus, the enhancement of the heat transfer due to confinement is a consequence of the lower wall superheating at the transition to the boiling regime. This wall superheating at the transition is decreased because of the increased surface area of influence of the bubbles in confined boiling.

This transition to the boiling regime is clearly visible on the boiling curve due to the good capacity of the flattened bubbles to remove heat. This capacity is certainly due to the mechanisms cited in the literature: increased agitation and enhanced microlayer evaporation.

3.3. Partial dryout

For a heat flux higher than 20 kW/m^2, for the majority of experimental points with $s < 0.5$ mm, a small increase in the heat flux causes a large increase in wall superheating and the enhancement of the heat transfer due to confinement disappears rapidly. For unconfined boiling, this behavior at low heat flux is not observed. Figs. 7 and 8 show photographs for $s = 0.1$ mm and $s = 13$ mm for heat fluxes of 20 kW/m^2 and 30 kW/m^2, respectively, with the downward facing heating surface and for FC72. For unconfined boiling, bubbles nucleate on the surface, coalesce and form a large bubble which is then removed from the surface of the heater. For confined boiling it can be observed that a large portion of the heating surface is covered by vapor. This area for a heat flux of 20 kW/m^2 represents 65%, and for $q = 30$ kW/m^2 this ratio increases to 75%. For the confined case shown in Fig. 8 the wall temperature is 70.4 °C, higher than the unconfined case for the same heat flux.

In the dry area the heat transfer is very weak and an increase in this surface decreases the mean heat transfer. In this case, which represents the partial dryout regime, the weak slope of the boiling curve is due to the increase in the dry area on the boiling surface.

The beginning of the dryout regime represents an important design condition for two-phase flow heat exchange, which is of interest because it represents the limit above which confined

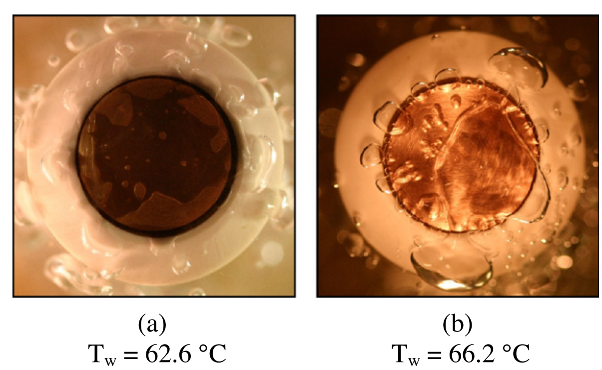

Fig. 7. Effect of confinement for $q = 20$ kW/m^2 and DF45°: (a) $s = 0.1$ mm and (b) $s = 13$ mm.

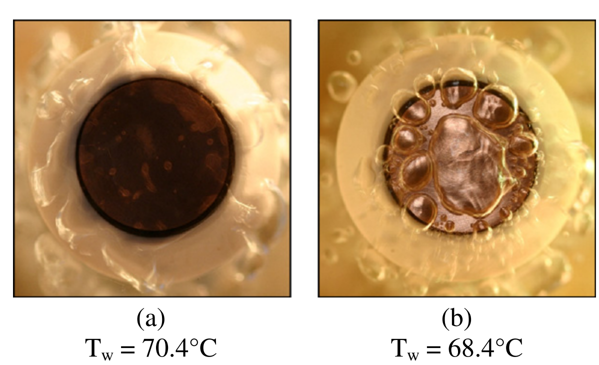

Fig. 8. Effect of confinement for $q = 30$ kW/m^2 and DF45°: (a) $s = 0.1$ mm and (b) $s = 13$ mm.

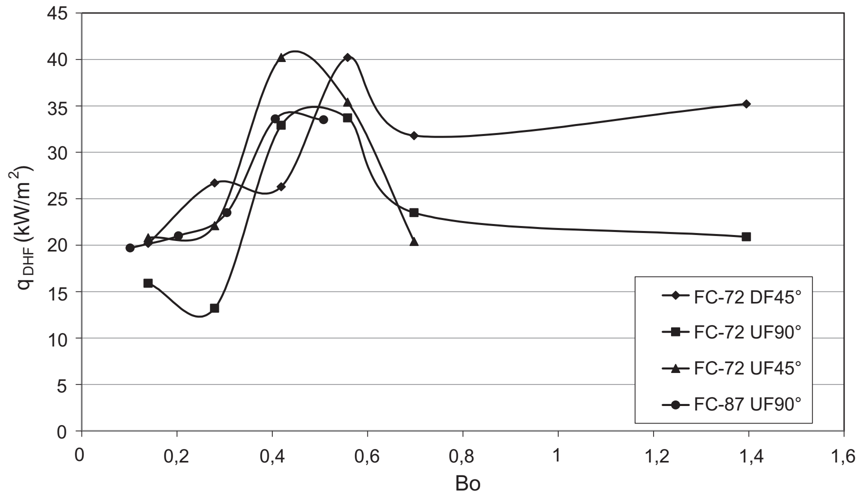

Fig. 9. Dryout heat fluxes for FC72 (UF90°, UF45°, DF45°) and for FC87 (UF90°) as a function of the Bond number.

nucleate boiling becomes inefficient. This limit varies considerably with the critical heat flux. The critical heat flux is the first maximum of the heat flux on the boiling curve. In unconfined boiling, this is of great interest from the engineering perspective as it is associated with the highest heat transfer coefficient. This maximum heat flux, in horizontal confined boiling, occurs at high superheat where the confinement decreases significantly the heat transfer.

The focus of this study is the regime where the confinement enhances the heat transfer and it was observed that this enhancement disappears rapidly when the boiling enters into the partial dryout regime. The limit between the partial dryout regime and the coalesced boiling regime is not as clear as in the case of the critical heat flux because of the progressive development of the dry patch with the increase in heat flux. Thus, it would be better to consider this as a boiling regime dominated by the presence of a dry patch in which a small increase in the heat flux leads to a large increase in the wall superheating.

In order to find the starting point of the partial dryout regime, we need to define a limit based on objective criteria. We decided to base this definition on the heat transfer coefficient ($h = q/\Delta T$) and to consider that the boiling enters into the dry patch-dominated regime when h is maximum. We will thus call q_{DHF}, the dryout heat flux, which is the heat flux at which the heat transfer coefficient is maximum. This definition is consistent with our observations and also has the advantage of being of interest in engineering applications: q_{DHF} is the limit heat flux above which confinement will rapidly decrease the heat transfer.

Fig. 9 shows the dryout heat fluxes for the boiling of FC72 in experiments UF90°, UF45°, and DF45°, and of FC87 in experiment UF90° as a function of the Bond number. For $Bo < 0.4$, we can observe that with increasing s, the dryout heat flux also increases. For $Bo > 0.4$, we observe an optimum dryout heat flux, which is specific to the horizontal confined boiling experiment and can be found on the boiling curve of other experiments [5,7].

For horizontal confined boiling, two mechanisms responsible for the evacuation of the vapor from the confined space can be identified:

- The first is the removal of the bubble or the entire removal of the vapor layer due to gravity. This is certainly the main mechanism for a vertical surface. For a horizontal surface this mechanism either involves the destabilization of the unstable equilibrium of the vapor layer trapped in the narrow space, or occurs when the interface of the bubble reaches the end of the confining element or the beveled slope.
- The second mechanism is the vapor blow off. This mechanism occurs when the first mechanism is not sufficient to remove the entire layer of vapor. In this mechanism, the strong evaporation leads to a high vapor velocity which breaks the interface and some vapor is evacuated from the narrow space without carrying the entire vapor layer. This mechanism occurs mainly for a low Bond number and high heat flux, Fig. 8a, where we observe dendrites which connect the vapor layer to the exterior of the narrow space.

The first mechanism occurs mainly for a large Bo and the second for smaller Bo numbers. The maximum q_{DHF} in terms of the Bo represents an optimum value, which is probably due to the combination of the other two mechanisms.

For low Bond numbers, the difference between the q_{DHF} values for experiments DF45° and UF45° is small and becomes undiscernibly for $Bo = 0.14$. On the other hand, this difference for experiments UF45° and UF90° is large for small Bond numbers and it tends to disappear for $Bo = 0.6$ and $Bo = 0.75$. Experiments UF45° and DF45° both have a bevel on the upper surface but differ in terms of the orientation of the heating surface which is on the bottom surface for UF45° and on the upper part for DF45°. In experiment UF90° the bevel is absent.

For $Bo > 0.5$, the peripheral condition does not influence q_{DHF}: for this size of heater, there is gravitational instability, as described for the first mechanism which takes place before the bubble reaches the side of the confined space.

For $Bo < 0.3$, the small thickness of the gap decreases the gravity influence and thus the vapor layer needs to reach the exterior of the confined space for the gravity to play a role. Moreover, the absence of a bevel will increase the quality in the peripheral region limiting the flow of liquid to the narrow region. Thus, the peripheral condition influences the two mechanisms of vapor evacuation for a small Bond number.

As described herein, the mechanism of vapor evacuation takes place in the peripheral region when the Bond number is sufficiently small. This mechanism is dependent on the conditions imposed by the geometric characteristics of the heating surface and its support. Thus, the ratio between the diameters of the support and of the test section can influence the dryout heat flux.

4. Conclusions

An experimental study was performed on the effect of confinement on the curves for the partial FC72 and FC87 saturated boiling on a copper disc of diameter 12 mm, for a downward and upward facing heating surface with two types of confinement elements. The main results are the following:

(i) As a general tendency the heat transfer coefficient increases when the confinement increases, as a function of the decrease in the distance between the heating surface and the nonheating surface.
(ii) For the experiments performed with an upward facing heating surface and $Bo > 1$, the bubbles detach from the wall and coalesce on the confining element forming a layer of vapor with a thickness limited by gravity.
(iii) For $Bo < 1$, a liquid microlayer is present between the surface and the vapor layer. The thickness of this microlayer is affected by the confinement and the heat transfer due to the microlayer evaporation will not be comparable to that observed when the bubble growth is tri-dimensional.
(iv) For $Bo > 1$, the layer of vapor formed on the confining element will not interact with the heating surface and the boiling will be similar to unconfined boiling.
(v) For the experiments performed with a downward facing heating surface, gravity works as the confining element by flattening the bubble over the heating surface.
(vi) Without confinement and at a low heat flux, the heat transfer is higher for a downward facing heating surface than for an upward facing heating surface. For a higher degree of confinement, the effect of the orientation of the heater tends to disappear.
(vii) The wall superheating for the *ONB* is lower for FC87 than for FC72. In confined boiling there is no vertical detachment and the surface tension will affect the nucleation temperature.
(viii) The enhancement of the heat transfer due to confinement is a consequence of the lower wall superheating at the onset of nucleate boiling. The wall superheating decreases because of the increased surface area under the influence of the bubbles in confined boiling.
(ix) The experimental results show the existence of an optimum value for the confinement (confinement for which q_{DHF} is maximum) at *Bo* around 0.4, for the majority of cases.
(x) For $Bo > 0.5$, the peripheral condition does not influence q_{DHF}. For $Bo < 0.3$, q_{DHF} is strongly influenced by the peripheral conditions.

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