NUCLEATE BOILING OF n-PENTANE IN A CONFINED SPACE

Elaine Maria Cardoso, elaine@lepten.ufsc.br
Julio César passos, jpassos@emc.ufsc.br
Jhonatan Erhart, jhonatan@lepten.ufsc.br
Laboratórios de Engenharia de Processos de Conversão e Tecnologia de Energia – LEPTEN/Boiling, Departamento de Engenharia Mecânica, Centro Tecnológico, Universidade Federal de Santa Catarina, Campus Universitário, Florianópolis, SC, Brasil.

Abstract. *An experimental study was carried out on n-Pentane nucleate boiling at atmospheric pressure and saturation temperatue for different degrees of confinement. The boiling was investigated in a confined space between two horizontal discs, a heating surface and an adiabatic surface, for distances varying between 0.1mm and 13mm. Tests were carried out on heating surfaces of 12 mm and 20 mm diameter. The effects of the degree of confinement on the partial boiling curve and the diameter of the heating surface were studied. A dimensionless analysis of the most important parameters was carried out and a correlation to describe the phenomenon of confined nucleate boiling is presented.*

Keywords*: Nucleate Boiling, Heat Transfer, Confined Boiling, Bond number.*

1. INTRODUCTION

The main factors that influence the heat transfer mechanisms in nucleate boiling regime are the heat flux, the thermophysical properties of the working fluid and the surface characteristics such as the thermophysical properties, dimensions, geometric shape, thickness, orientation, and roughness of the material, among others. In general, the effect of surface characteristics on the boiling process is dependent on the thermal conductivity, interactions between the solid, liquid and vapor, and the shape and size of nucleation sites (microstructure). All of these parameters are interrelated, but due to difficulties associated with measuring them, these effects are studied separately.

Boiling in a narrow space occurs in many industrial applications such as energy conversion, cooling of electronic devices, micro heat exchangers and others. The first studies focusing on confined boiling were performed by Katto *et al.* (1977) and Ishibashi and Nishikawa (1968). The former focused on the confined boiling on horizontal plates showing that confinement enhances the boiling heat transfer at low heat flux and causes a decrease in the critical heat flux (CHF). In the latter work, the authors presented an analysis of the effect of pressure, surface tension and confinement on pool boiling in an annular vertical channel. The authors showed that the confinement leads to an increase in the heat transfer coefficients at low heat fluxes. They also established the existence of two regimes depending on the Bond number, *Bo*, defined as the ratio between the gap, *s* (between the heating surface and the confining element), and the capillary length:

$$Bo = s / \left[\frac{\sigma}{g(\rho_L - \rho_v)} \right]^{1/2} \tag{1}$$

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. The authors showed that enhancement of the heat transfer due to the confinement occurs when $Bo<1$, when the gap *s* is equal to or less than the bubble detachment diameter.

The effect of pressure was reported in Bonjour *et al*. (1997), for vertical and horizontal configurations, and the enhancement of heat transfer due to pressure in unconfined boiling tends to disappear as the confinement increases. The same authors studied the effect of active nucleation site density and showed that, in unconfined boiling, a single active nucleation site cannot enhance the heat transfer. This enhancement of the heat transfer by a single nucleation site can clearly be seen in confined boiling.

Cardoso *et al*. (2009) and Katto *et al*. (1977) reported results showing the dependence on the Bond number: for $Bo<1$ a thin liquid film is formed by the fast growing bubbles, while the liquid film is not observed for $Bo>1$.

The enhancement of heat transfer is generally attributed to two different mechanisms: enhanced microlayer evaporation (Utaka *et al*. (2009), Passos *et al*. (2004)) or enhanced liquid agitation (Fujita *et al*. (1988)). The former is due to the fast growth of the bubble forming a microlayer with a large surface. This mechanism is generally cited in studies investigating a horizontal surface or narrow space. In the latter case, this is due to the coalescence phenomenon and displacement of the bubbles along the surface. This mechanism is generally cited for a vertical heated surface.

Nishikawa *et al*. (1984) studied the effect of the orientation of the heated surface and its influence on several aspects of water nucleated boiling, at atmospheric pressure, such as the release, growth and detachment of the bubbles, and the movement of the bubble and the liquid over the surface, among others. Their results indicated that for low heat flux values and inclination angles lower than 120°, the heat transfer is controlled by the agitation of isolated vapor bubbles.

However, for angles higher than 150°, the heat transfer is controlled by the removal of the superheated thermal layer when the vapor bubble slides over the surface and by the heat of vaporization due to the liquid film vaporization under the vapor bubble when it covers the surface. For high heat fluxes, the mechanisms associated with the movement of the vapor bubble are not influenced by the orientation of the surface and the vaporization of the liquid film becomes the dominant mode of heat transfer. Guglielmini *et al*. (2006) analyzed the combined effects of surface orientation and confinement on nucleate pool boiling and critical heat flux of HFE-7100 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≈1 and angles of 0° (upward-facing surface) and 45°, heat transfer was enhanced. At high wall superheats and confinements of 3.5, 2 and 1 mm, the heat flux, heat transfer coefficient and CHF decreased as the channel width decreased. This effect was more evident for 0° and 45°, whereas it was less pronounced at 90° and was almost negligible at 135°.

The objective of this paper is to analyze n-Pentane nucleated boiling, at saturated temperature under atmospheric pressure conditions, for different degrees of confinement (s=0.1, 0.2, 0.5, 0.7 and 13mm), on an upward facing surface, with two different diameters of heating discs. The aim of this study is to investigate the influences of different parameters on the boiling phenomenon and to obtain results of interest to the industry.

1.1 Correlations for nucleate boiling

The first introduction of the condition of the heating surface in a correlation for nucleate boiling was performed by Rohsenow (1952). He used different constants to express the effects of various materials, conditions and types of surface:

$$h_{Rohsenow} = \mu_l h_{lv} \left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{0.5} \left(\frac{c_{pl}}{C_{sf} h_{lv} P_r^{\,s}} \right)^3 \Delta T_p^{\,2} \tag{2}$$

where μ_l, h_{lv}, c_{pl}, and Pr_l represent the viscosity of the liquid (*kg/m s*), the latent heat of vaporization (*kJ/kg*), the specific heat of the liquid (*kJ/kg K*), and the Prandtl number of the liquid, respectively. $\Delta T_p = T_p - T_{sat}$ and C_{sf} is dependent on the material of the heating wall, the surface roughness and the working fluid: r=0.333 e s=1.7 (Carey,1992).

Correlation of Stephan and Abdelsalam (1980):

$$h_{sa} = 207 \left(\frac{k_l}{d_b} \right) \left(\frac{q d_b}{k_l T_{sat}} \right)^{0.745} \left(\frac{\rho_l}{\rho_v} \right)^{0.581} \mathrm{Pr}_l^{\,0.533} R_p^{\,0.133} \tag{3}$$

where k_l represents the thermal conductivity of the liquid. The detachment diameter of the bubble is calculated by:

$$d_b = 0.0149 \theta \left(\frac{2\sigma}{g(\rho_l - \rho_v)} \right)^{0.5} \tag{4}$$

Later, Cooper (1984) published a correlation for nucleate boiling taking into account the surface roughness and reduced pressure of the liquid:

$$h_{cooper} = 55 p_r^{\,b} (-\log p_r)^{-0.55} M^{-0.5} q^{0.67} \tag{5}$$

where $b=0.12 - 0.2logR_p$, and p_r, M and R_p represent the reduced pressure, the molecular weight and the surface roughness, respectively.

Borishanski (1969) developed a simple correlation based on experimental data. The heat transfer coefficient, h is a function of the critical pressure of the working fluid, p_c, in bar, reduced pressure p_r, and heat flux q in W/m^2:

$$h_{bo} = 0{,}1011 \mathrm{p_c}^{0.69} \mathrm{q}^{0.7} \mathrm{F(p_r)} \tag{6}$$

where:

$$\mathrm{F(p_r)} = 1{,}8 p_r^{\,0.17} + 4 p_r^{\,1.2} + 10 p_r^{\,10} \tag{7}$$

Ratiani (1972) proposed a correlation valid for various fluids (water, liquid metals, fluids and refrigerants):

$$h_{ra} = 0,7e^{-2}\frac{k_l}{r_c}\left(\frac{q\rho_v h_{lv} r_c^{\,2}}{k_l \sigma T_{sat}}\right)^{0.7}\left(\frac{\sigma\rho_l c_{pl} T_{sat}}{\rho_v^{\,2} h_{lv} r_c}\right)^{0.25}\left(\frac{r_c\rho_l\sqrt{p_{sat}\left(\frac{1}{\rho_v}-\frac{1}{\rho_l}\right)}}{\mu_l}\right)^{0.25} \quad (8)$$

where k_l is the conductivity of the liquid (*W/m K*), T_{sat} is the saturation temperature (*K*), μ_l, is the viscosity of the liquid (*kg/m s*), h_{lv}, is the latent heat of vaporization (*kJ/kg*), c_{pl} is the specific heat of liquid (*kJ/kg K*), ρ_l and ρ_v are the density of the fluid and vapor *(kg/m³)*, σ is the superficial tension *(J/m²)* and p_{sat} is the saturation pressure *(bar)*. The author recommends using the value of 20μm when the value of cavity radius, r_c, is unknown.

2. EXPERIMENTAL SETUP

Figure 1 shows the experimental set up, consisting of a boiling chamber (8) installed in the center of an external chamber (9), both assembled between two horizontal stainless steel plates of 200 x 200 x 10mm (12 and 13). The boiling chamber is a vertical glass tube with a 90 mm inner diameter and 175mm height, which contains the test section and the working fluid being inside. The external chamber has a square cross section of 170x170mm and transparent lateral walls of plexy-glass (9). The test condition temperature of the working fluid is imposed by a forced flow of water in the space created between the glass tube of the boiling chamber and the plexy-glass wall of the external chamber. The water temperature is controlled by a cryostat. Inside the boiling chamber, in the upper part, there is a serpentine condenser (7) cooled by water whose temperature is controlled by a second cryostat. The boiling chamber is equipped with a pressure transducer (3) and valves (2 and 6). Two K thermocouples (4) in the liquid and another in the vapor allow the monitoring of the test condition temperature, which is controlled by cold water flowing inside the serpentine (7).

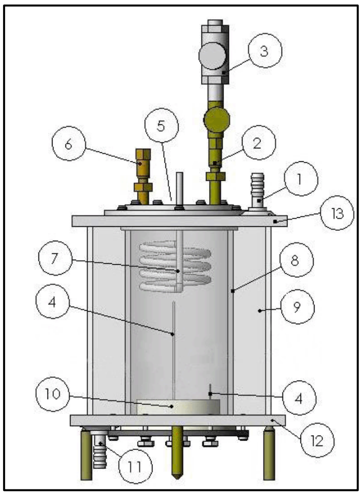

Figure 1. Scheme of the experimental apparatus:
1) Bath water inlet; 2) Valve; 3) Pressure transducer; 4) Thermocouples; 5) Plexy-glass window; 6) Valve; 7) Condenser; 8) Boiling chamber; 9) Plexy-glass chamber; 10) Test section; 11) Bath water outlet; 12) and 13) Stainless steel plates.

The test section, Fig. 2, consists of a copper block of 12mm or 20mm diameter and 60mm height cylinder, isolated on its perimeter with Teflon®. The experiments were performed using n-Pentane as the working fluid under saturated conditions at p = 1bar (T_{sat} = 35.8°C). The capillary length is close to L = 1.6mm. The section test is heated by a cartridge heater of 177Ω, Fig. 2, imbedded in the sample. This resistance must be sufficient to reach the critical heat flux which, according to Eq. (3), is 243kW/m² for n-Pentane. Four K thermocouples, fixed in the cylindrical part of the copper block, are used to determine the wall temperatures and the heat flux.

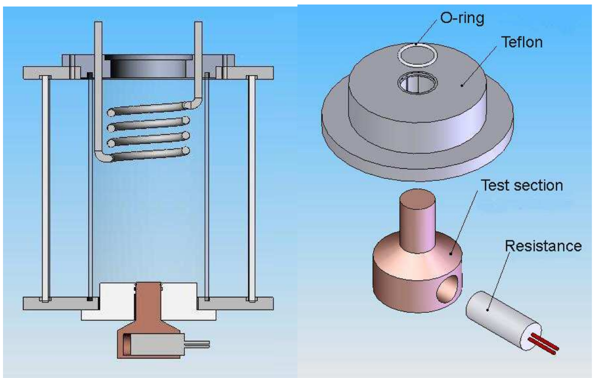

Figure 2. View of test section assembly.

The confining element, Fig. 3, consists of a transparent acrylic piece fixed to an aluminum support, and this, in turn, is fitted to the test section. This conical unheated plate is placed parallel to the heating surface (45° cone angle and 12mm or 20mm diameter at the bottom). Inside the upper part of the boiling chamber there is a transparent plexy-glass window, allowing the visualization of the boiling phenomenon on the heating surface.

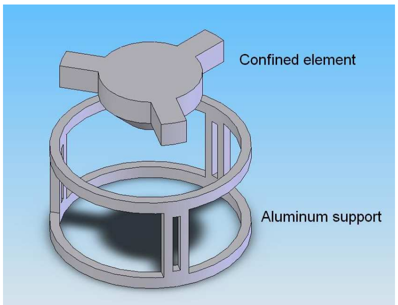

Figure 3. View of confining element.

Figure 4 shows a schematic drawing including the cryostat, power source, data acquisition system and the computer.

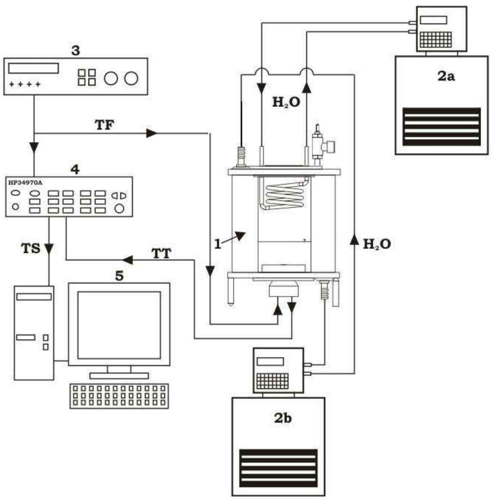

Figure 4. Schematic drawing of the apparatus.
1) Boiling Chamber; 2a) Serpentine Cryostat; 2b) Liquid Circulation Cryostat; 3) Power Source; 4) Data Acquisition System; 5) Computer; TS) All Signals; TT) Thermocouples and pressure transducer signals; TF) Power source signal.

2.1 Experimental Procedure

Before each test run the working fluids were heated to very close to the saturation temperature in order to degas them. No evidence of significant amounts of gas dissolved in the working liquids was detected on the boiling curves. A vacuum was created in the boiling chamber, before each series of measurements, which was then fed with the working fluid. By setting the temperature of the water bath, the test conditions represented atmospheric pressure and the saturation temperature. This condition was regulated by monitoring the pressure and the temperature inside the boiling chamber. Once the test conditions had stabilized, the heat flux was imposed in the range of 15 to 190 kW/m².

The experimental procedure was programmed in LABVIEW and each test had 180s duration for each imposed heat flux, followed by an interval of 300s with the power supply turned-off. Only the temperature data for the last 90s of the test interval were acquired, at a rate of 3 points/s. The test section was polished using #600 emery paper and the surface was then cleaned using acetone and dried with an air jet. This procedure was repeated before each series of measurements.

The uncertainty of the temperature was ± 0.2°C. Table 1 shows the experimental uncertainties for the heat transfer coefficient (the heat flux had similar values).

Table 1. Experimental uncertainties for n-Pentane.

	D = 12mm	D = 20mm
Natural convection ($2 \leq q_{fluid} \leq 10$ kW/m^2)	$46.3 \geq h \geq 13$ %	$48 \geq h \geq 16.5$%
Confined nucleate boiling ($20 \leq q_{fluid} \leq 180$kW/m^2)	$16 \geq h \geq 2.5$%	$18 \geq h \geq 3.5$%
Unconfined nucleate boiling ($20 \leq q_{fluid} \leq 200$kW/m^2)	$11.3 \geq h \geq 2.2$%	$15 \geq h \geq 3.7$%

3. RESULTS

Figure 5 shows the partial boiling curves for n-Pentane and s = 0.1, 0.2, 0.5, 0.7 and 13mm, on a heating surface facing upward with 12mm diameter. The capillary length, Eq. (1), is approximately 1.6mm. For these values of s, the Bond numbers are 0.06, 0.13, 0.32, 0.45 and 8.35, respectively. We can observe for s = 0.1mm and 0.2 a particular dependence on the heat transfer coefficient (or ΔT) of s and q.

For a heat flux of between 30kW/m^2 and 130kW/m^2 the heat transfer is higher in the case with higher confinement than for the case without confinement. As a general tendency the heat transfer increases with a decrease in s. For the unconfined case, the region of heat transfer dominated by natural convection extends to a superheat of 22.5K. Above this superheat the slope of the boiling curve increases considerably indicating the transition to the boiling regime. For the confined case, the slope of the boiling curve is elevated at a lower superheat (except for the case s = 0.1mm): the confinement decreases the superheat of the transition from the convective regime to the boiling regime.

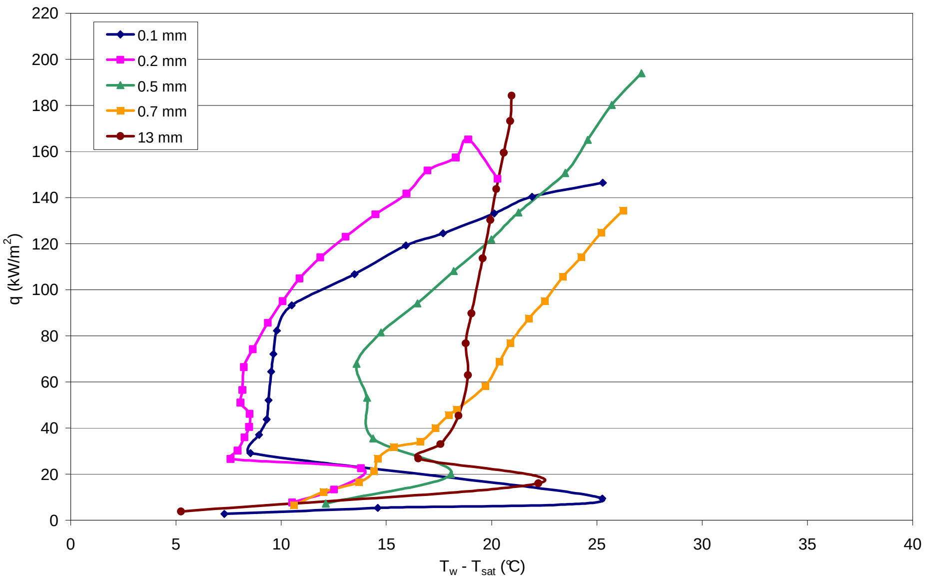

Figure 5. Partial boiling curves as a function of s, for D = 12mm.

Figure 6 shows the effect of the confinement on the partial boiling curve, at saturation temperature, for n-Pentane, for s = 0.1,0.2, 0.5, 0.7 and 13mm, on a heating surface facing upward with 20mm diameter .

For $s \leq$ 1mm, the Bond number is less than unity 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 *et al.* (1969), the initial thickness of the microlayer is dependent on the velocity profile of the liquid close to the foot of the bubble. This velocity profile is a function of the dynamics of the growing bubble. In confined boiling, once the bubble reaches the size of the gap, its growth becomes bi-dimensional and the variation of its foot radius in term of time must change considerably. Thus, the thickness of the microlayer is affected by the confinement and the heat transfer due to microlayer evaporation will not be comparable to that observed when the growth of the bubble is tri-dimensional.

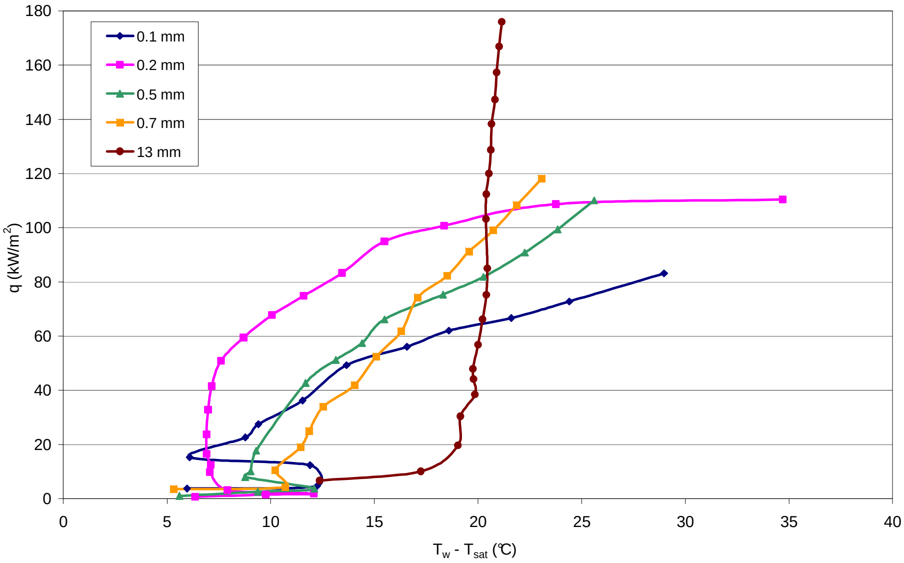

Figure 6. Partial boiling curves as a function of s, for D = 20mm.

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

The effect of the heating surface diameter on the heat transfer for the case with higher confinement (s = 0.1mm) and for the case without confinement (s = 13mm) was determined. Figures 7 and 8 show this effect for s = 0.1mm, and s = 13mm, respectively.

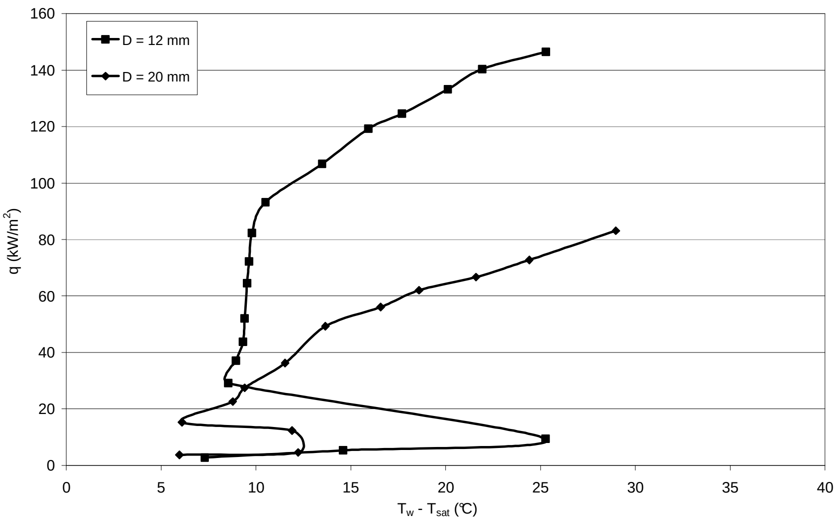

Figure 7. Effect of diameter of the heating surface, for s = 0.1mm.

In Fig. 7, for s = 0.1mm, the boiling curve for the heating surface with 12mm diameter is shifted to the left compared with the curve for 20mm diameter, indicating a better heat transfer for the case where the diameter of the heating surface is smaller. This can be explained by the fact that the smaller the diameter of the heating surface, for the confined case, the lower the residence time of the vapor bubbles inside the channel. After the departure of the bubble, a front of cold liquid enters the channel and cools the surface. This mechanism is highly dependent on the conditions imposed by the geometric characteristics of the heating surface and support. Therefore, the ratio between the diameter of the support and the test section can influence the residence time of vapor bubbles inside the gap.

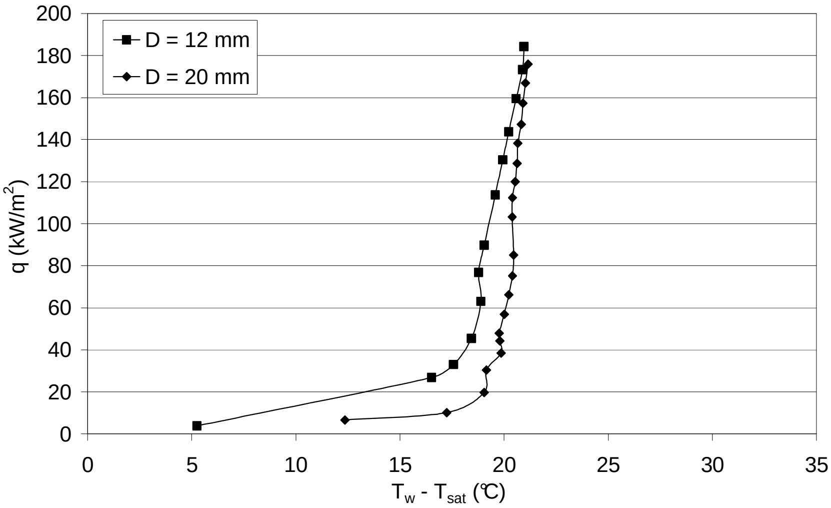

Figure 8. Effect of diameter of the heating surface, for s = 13mm.

In Fig. 8, for the case without confinement, the heat transfer coefficient increases as the diameter of the surface decreases, for natural convection. This result has the same tendency observed in the literature, as the heat flux increases this difference decreases and the CHF tends to be higher in the surface with 12mm of diameter.

In order to obtain a correlation using the minimum of dimensionless numbers we used the Buckingham (π) theorem. To apply this theorem it is firstly important to decide which of the parameters are important.

The boiling phenomenon is dependent on the heat flux, q, the superheating, ΔT, the gravity, g, the latent heat of vaporization, h_{lv}, the surface tension, σ, the characteristic length (capillary length), L_b, the confinement degree, s, and the thermophysical properties of the fluid: density of the liquid and vapor phases, ρ_l and ρ_v, specific heat of the liquid, c_{pl}, thermal conductivity, k_l, and viscosity of the liquid, μ_l. Thus, our problem can be written as:

$$(q, \Delta T, g, h_{lv}, \sigma, L_b, s, \rho_l, \rho_v, c_{pl}, k_l, \mu_l) = 0 \tag{9}$$

The functional relationship involves 12 variables. Considering a system of units with four variables (M, L, t, T), where: [M] mass, [L] length, [t] time and [T] temperature. According to the Buckingham (π) theorem, we have eight dimensionless parameters or settings in dimensionless groups. Through algebraic manipulation of the π values the main dimensionless groups are:

$$Nu = Nu(Fr, We, \mathrm{Pr}_l, Ja, Bo) \tag{10}$$

where the Nusselt number is expressed as a function of the Froude, Weber, Prandtl, Jakob and Bond numbers, respectively. The Weber and Froude numbers can be rewritten as a function of the heat flux imposed as:

$$Fr = \frac{q''^2}{\rho_v{}^2 h_{lv}{}^2 g L_b} \tag{11}$$

$$We = \frac{q''^2 \rho_l L_b}{\rho_v^2 h_{lv}^2 \sigma} \tag{12}$$

In Eqs. (13) and (14) the correlations obtained for the unconfined and confined cases, respectively, are presented.

$$Nu_{calc,sconf} = 33.3 FrWe^{-0.59} \Pr_l^{-0.03} Ja^{-0.48} \quad (13)$$

$$Nu_{calc,conf} = 30.6 FrWe^{-0.59} \Pr_l^{-0.08} Ja^{-0.56} Bo^{0.012} \quad (14)$$

In order to validate the correlations, the values calculated for the heat transfer coefficient were compared with the experimental values. Figure 9 shows the comparison between h_{calc} and h_{exp}, for the case without confinement. For this case it was possible to compare the results with the experimental data obtained for FC 72 obtained by Cardoso (2005), for n-Pentane obtained by the Lyon Centre of Thermal Science (CETHIL / INSA-Lyon) and for n-Hexane obtained by Benjamin and Balakrishnan (1996).

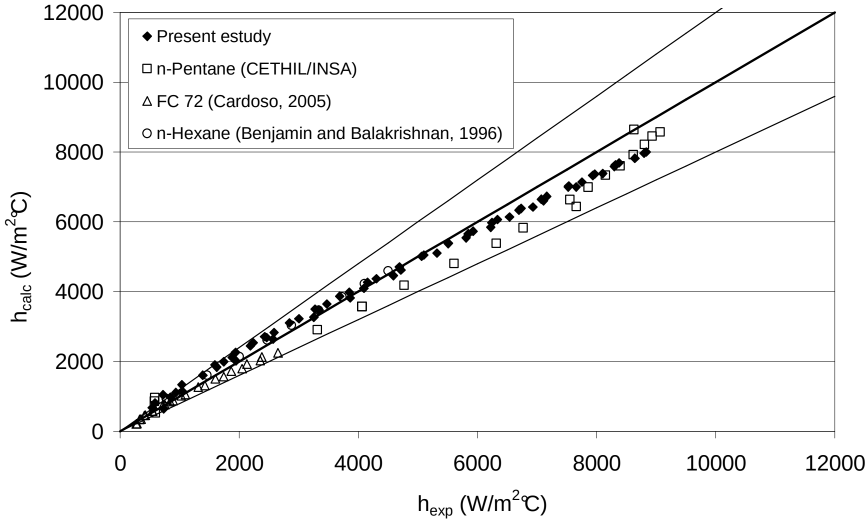

Figure 9. h_{calc} versus h_{exp}, for unconfined case.

The correlation for the unconfined case had good agreement with the experimental data. It is possible to predict the heat transfer coefficient during nucleate boiling without confinement with an accuracy of ± 10% for the experimental conditions described in section 2. The absolute standard deviation for FC 72 was 8%, for n-Pentane by CETHIL/INSA was 15% and for n-Hexane was 7%.

For the confined case, it is important to emphasize the importance of the Bond number, which represents the influence of the confinement. Figure 10 shows the heat transfer coefficients obtained for the experimental data and the data calculated using the correlation.

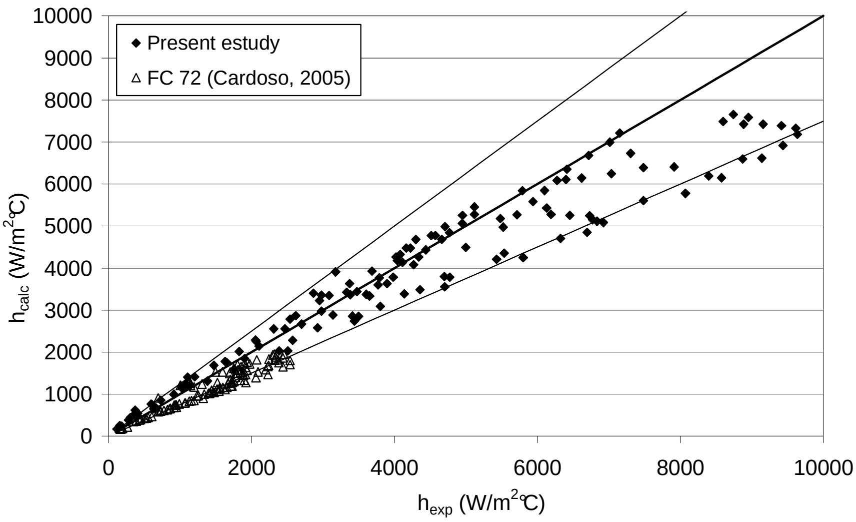

Figure 10. h_{calc} versus h_{exp}, for confined case.

In this case, the maximum deviation found for the correlation was ± 14%. Therefore, the experimental data were well represented by the correlation described in Eq. (14). For this case it was possible to compare the results with the experimental data obtained for FC 72 obtained by Cardoso (2005), with accuracy of 19%.

The heat transfer coefficients obtained experimentally with *s*=13mm, for the n-Pentane, were compared with the values of *h* calculated from the correlations of Cooper (1984), Rohsenow (1952), Stephan and Abdelsalam (1980), Borishanski (1969) and Ratiani (1972).

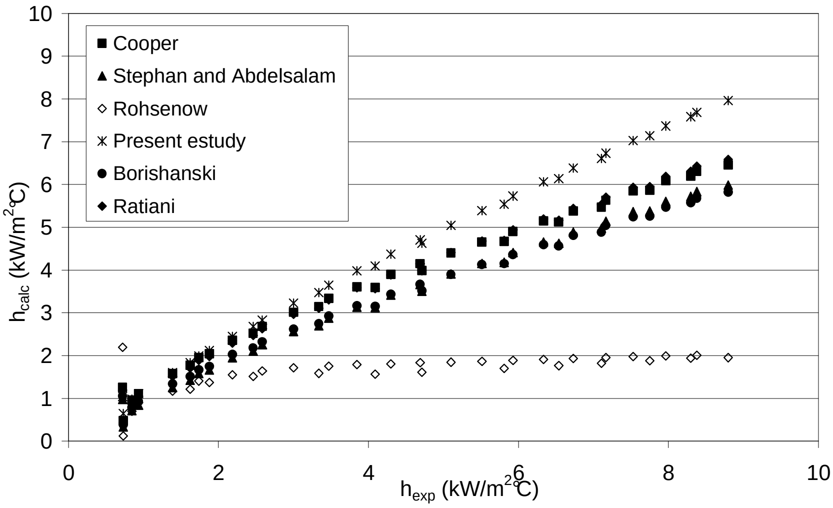

Figure 11. Comparison of *h* calculated using the proposed correlation and correlations reported in the literature.

Figure 11 shows that the values for the heat transfer coefficient, *h*, calculated using the correlation proposal in Eq. 13 are close to those calculated via the Ratiani (Eq. 8) and Cooper (Eq. 5) correlations, the absolute standard deviations being 12.3% and 11.9%, respectively. The absolute standard deviation for the Borishanski correlation (Eq. 6) was 21.8%. The Stephan and Abdelsalam (Eq. 3) showed absolute standard deviations of 23.1%. The largest discrepancy occurred for the values of *h* calculated with the Rohsenow (Eq. 2) correlation, where the value of the C_{sf} coefficient used was 0.0154, which corresponds to the copper/n-Pentane combination as suggested by Pioro (1999). In this case the absolute standard deviation was found to be 57%.

4. CONCLUSIONS

An experimental analysis of the effect of confinement on the partial saturated boiling curves for n-Pentane on a copper block of diameter 12mm or 20mm, for an upward facing heating surface, was carried out. The main results are the following:

- As a general tendency the heat transfer coefficient increases when the confinement increases, corresponding to a decrease in the distance between the heating surface and the confining element.

- For *Bo*<1, a microlayer of liquid exists between the heating surface and the vapor layer. 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 *Bo*>1.

- For the confined case, the heat transfer coefficient is higher when the diameter of the heating surface is smaller. This can be explained by the fact that the smaller the diameter of the heating surface the lower the residence time of vapor bubbles inside the channel.

- For unconfined cases, the heat transfer coefficient increases as the diameter of the surface decreases. This may be due to the fact that decreasing the surface area decreases the resistance of the liquid located near the surface, flooding it again.

- The correlation for unconfined and confined cases had good agreement with experimental data. Thus, it is possible to predict the heat transfer coefficient with an accuracy of ± 15%, for both cases.

- The heat transfer coefficient for unconfined case calculated using the proposal correlation is close to the values calculated via the Ratiani and Cooper correlations. The absolute standard deviation for the Borishanski correlation was 21.8%. The Stephan and Abdelsalam correlation showed absolute standard deviation of 23.1%. The largest discrepancy occurred for the values of h calculated with the Rohsenow correlation.

5. ACKNOWLEDGEMENTS

The authors are grateful for the support of CAPES and the Research Council of Brazil (CNPq) which allowed this study to be carried out. The authors also extend their gratitude to Mr. D. R. Souza and Mr. A. Dalmaz for their important contribution to the laboratory work.

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