CONFINED BOILING OF THE n-PENTANE IN A HORIZONTAL SPACE

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RESUMO

This paper presents experimental results for saturated nucleated boiling of n-Pentane on a heating surface facing upward, at atmospheric pressure, for different degrees of confinement, s = 0.2, 0.5, 0.7 and s = 13mm, corresponding to Bond numbers 0.13, 0.32, 0.45 and 8.35.

Comparative studies with results from literature, together with analysis of the experimental data allowing the improvement of the experimental apparatus. The results show the enhancement of boiling heat transfer with a decreasing distance *s* between the heating surface and an unheated surface. The experimental heat transfer coefficients for unconfined boiling, s=13mm, are compared with three empirical correlations.

Keywords: Confined nucleate boiling, Bond number, dryout heat flux.

INTRODUCTION

The development of new technologies associated with a reduction in manufacturing and installation costs requires the dissipation of increasing heat flux. The heat transfer associated with the phase change continues to be studied with the aim of achieving high heat transfer coefficients, with applications such as the cooling of electronic components or as a basis for diverse energy conversion systems that rely on the efficiency of the evaporator.

A deep analysis is necessary in the case of confined nucleate boiling in order to perfect its use as an intensification technique and also to minimize the risks associated with confinement when it is imposed on a design, because it can cause problems linked to the premature occurrence critical heat flux, which represents the limit of operation of the system in the nucleate boiling regime.

The effect of the confinement on the bubbles can be characterized by a dimensionless parameter known as the Bond number, *Bo*, defined as the ratio of the characteristic length to the confined space, *s*, and the capillary length, *L*. The latter is proportional to the detachment diameter of the vapor bubble in a pool and defined as [1]:

$$L = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \qquad (1)$$

where σ, g, ρ_l and ρ_v represent the surface tension, the acceleration due to gravity, the liquid density and the vapor density, respectively.

According to Eq. (2), a decrease in the superheating $\Delta T_{sat} = (T_w - T_{sat})$ for the same heat flux, *q*, causes an increase in the heat transfer coefficient, *h*.

$$h = \frac{q}{\Delta T_{sat}} \qquad (2)$$

Several studies were focused on the effects of contact angle, thermophysical properties, dimensions, shape, thickness, orientation in space, roughness and microstructure of the boiling surfaces.

The objective of this work is the experimental investigation of n-Pentane in a confined and unconfined space, with upward facing surface pool boiling heat transfer and for saturated conditions. The motivation of this study is the understanding of the mechanisms and effects under the nucleate boiling regime.

LITERATURE REVIEW

The increase in the heat transfer coefficient when *Bo*<1 is explained by the evaporation of a thin liquid film, which is present between the heated surface and the base of the deformed bubble [2]. For a low Bond number, as consequence of the coalescence phenomenon, the bubble becomes large and deformed allowing an increase in the bubble area pressed against the heated wall. According to Straub [3], at the base of the bubble a thin liquid film called the microlayer is attached to the surface by intermolecular attractive forces between the liquid and the surface and because of these, the film does not evaporate completely. Only a high superheating will evaporate the microlayer, with a dry area then forming and the heat transfer coefficient slowly decreasing.

When *Bo*>1, the bubbles are isolated and the boiling mechanism, tends to be similar to that on an upward facing heating surface in a liquid pool. However, for a downward facing heating surface there is an increase in the heat transfer coefficient because of the thicker thermal layer that increases the nucleation site density [4, 5].

The boiling crisis, when the imposed heat flux attains the maximum value for the nucleate boiling regime, before the decrease in the heat transfer coefficient, and for an infinite upward facing heating surface cooled by a liquid pool, is calculated by the Zuber's equation, [1]:

$$q_{máx,Z} = 0,131\rho_v^{0,5} h_{lv}[\sigma g(\rho_l - \rho_v)]^{\frac{1}{4}} \qquad (3)$$

The literature contains a number of publications that report the studies of factors that affect pool boiling heat transfer. These include the effects of surface characteristics such as thermophysical properties, dimension, thickness, etc., and also, thermophysical properties of the working fluid.

The effects of cavity size, shape, and density of nucleation have been the most investigated for many researches [6]. Cavity radius and depth, contact angle of the fluid, and initial fluid penetration velocity are important parameters determining the stability of a cavity in boiling.

Some experiments showed that, with increasing surface roughness, starting from a smooth polished surface, the h increases and reach some maximum value; after this point, a further increase in surface roughening has no effect on boiling heat transfer [7].

In the case of a rough surface that has quite a large number of potential vapor generating centers, the wall thermal conductivity does not significantly affect the boiling heat transfer.

Wang and Dhir [8] conducted experiments with water boiling at atmospheric pressure on vertical copper surfaces with different wetabilities (contact angles of 90°, 35° and 18°). They concluded that the number of active vapor generating centers decreased as the wettability of the boiling surface improved, or either, as values of the contact angle decreased.

The effect of boiling surface orientation on the h is noticeable. Kang [9] conducted experiments with pool boiling on a horizontal, inclined, and vertical tube and found that the orientation effect on the heat transfer could be significant. According to his findings, the h is higher on an inclined tube (about 45° from the horizontal plane) compared to horizontal and vertical orientations. This effect seems to be due to decreased bubble slug formation on a boiling surface and improved liquid access to the surface.

This trend was also observed by Nishikawa et al. [10], who studied the influence of the orientation of the heated surface, from 0° (horizontal upward facing heated plate) to close to 180° (downward facing heated plate) and the heat flux on water nucleate boiling, at atmospheric pressure. Their results indicated that for low heat flux values and $\theta < 120°$, the heat transfer is controlled by the agitation of isolated vapor bubbles. However, for $\theta > 150°$, 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 orientation of the surface.

You et al. [11] conducted FC-72 and R-113 saturated pool boiling experiments at atmospheric pressure, using a 0.13 mm diameter chromel wire and 0.51 mm diameter platinum cylinder. The properties of the liquid affects wall superheat values but the material effect was less significant than the surface microgeometry for highly wetting dielectric fluids. The capacity of such fluids to flood microcavities induces large superheats at incipience.

Liang and Yang [12] illustrated the fundamental importance of the contact angle comparing boiling incipience of n-Pentane on a flat copper surface and on a porous graphite/copper surface. The use of graphite/copper reduced considerably the hysteresis at boiling incipience, by increasing the number and the size of the nucleation sites and its capacity to preserve the vapor nucleus. The authors explained these results by the contact angle, 2° for the n-Pentane on copper versus 35° for the n-Pentane on graphite.

EXPERIMENT

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 10 mm (12 and 13). The boiling chamber is a vertical glass tube with a 90 mm inner diameter and 175 mm height, the test section and the working fluid being inside. The external chamber has a square cross section of 170x170 mm 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 the other in the vapor, allow the monitoring of the test condition temperature, which is controlled by cold water flowing inside the serpentine (7).
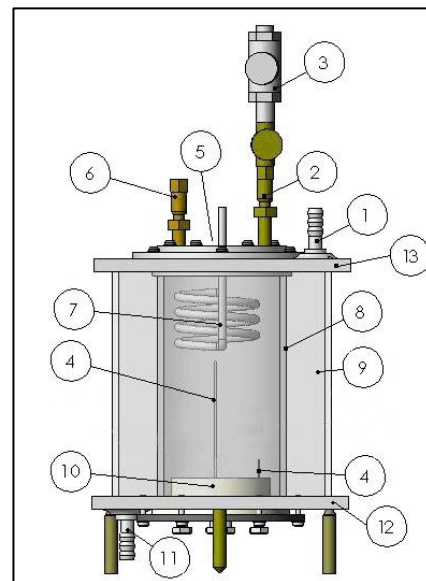

Figure 1. Scheme of the experimental apparatus:
1) Bath water inlet; 2) Valve; 3) Pressure transducer; 4) Thermocouples; 5) Plexy-glass window; 6) Valve; 7) Condenser; 8) Boiling chamber; 9) Plexy-glass chamber; 10) Test section; 11) Bath water outlet; 12) and 13) Stainless steel plates.

The test section, Fig. 2, consists of a copper block of 12 mm diameter and 60 mm height cylinder, isolated on its perimeter with Teflon®. The experiments are performed using n-Pentane as working fluid in saturated conditions at $p = 1$bar ($T_{sat} = 35.8°$C). The capillary length is close to $L = 1.6$mm. 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 that, according to Eq. (3) is 243kW/m^2 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.
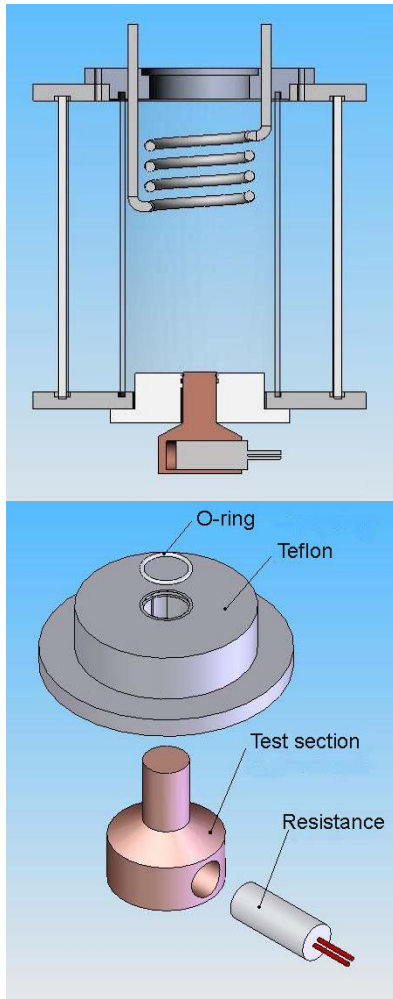

Figure 2. View of test section assembly.

The confined element, Fig. 3, consists of a transparent acrylic piece fixed to an aluminum support, and this is in turn is fitted to the test section. This conical unheated plate is placed parallel to the heating surface (45° cone angle and 12mm diameter at 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.
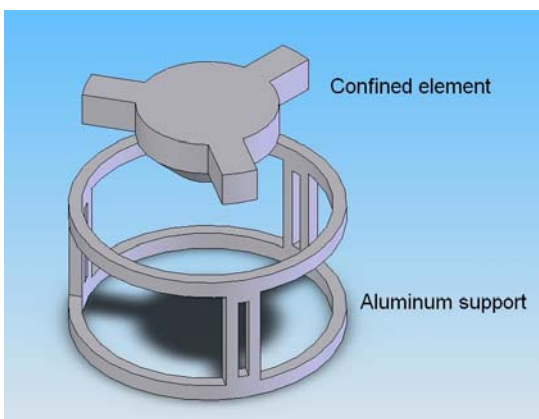

Figure 3. View of confined element.

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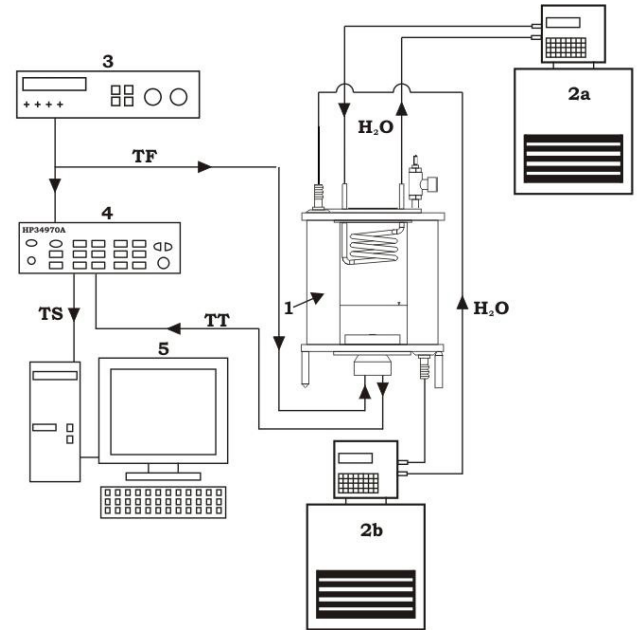

Figure 4. Schematic drawing of the apparatus.
1) Boiling Chamber; 2a) Serpentine Cryostat; 2b) Liquid Circulation Cryostat; 3) Power Source; 4) Data Aquisition System; 5) Computer; TS) All Signals; TT) Thermocouples and pressure transducer signals; TF) Power source signal.

Experimental Procedure

Before each test run the working fluids was 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 by 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 is programmed in LABVIEW and each test has a 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 temperature was ± 0.2°C. In the Tab.1 is showed the experimental uncertainties for the heat transfer coefficient and heat flux.

Table 1. Experimental uncertainties for n-Pentane.

Natural convection ($10 \leq q \leq 30 kW/m^2$)	$57.6 \geq h \geq 11.3$ %	$57.5 \geq q \geq 11\%$
Confined nucleate boiling ($30 \leq q \leq 190 kW/m^2$)	$12.1 \geq h \geq 2.8\%$	$12 \geq q \geq 2.7\%$
Unconfined nucleate boiling ($30 \leq q \leq 140 kW/m^2$)	$8.2 \geq h \geq 2.8\%$	$8.1 \geq q \geq 2.7\%$

RESULTS

Partial Boiling Curves

Figure 5 shows the effect of the confinement on the partial boiling curve, at saturation temperature, for n-Pentane, for s = 0.2, 0.5, 0.7 and 13 mm. The capillary length, Eq. (1), is near 1.6mm. For these values of s, the Bond number is equal to 0.13, 0.32, 0.45 and 8.35, respectively. We can observe for s = 0.2 mm, a particular dependence on the heat transfer coefficient (or ΔT) of s and q.

For s = 0.2mm and heat fluxes lower than 140kW/m², the experimental points are shifted to the left compared with the case of s = 13mm, characterizing an enhancement in the heat transfer for the confined case. However, for a heat flux higher than 160kW/m² the wall temperature increases, that leads to consider the possibility of starting the dryout heat flux earlier than in the case with s = 13mm. Due to this very thin gap, corresponding to a Bond number of 0.13, the bubbles are deformed and the frequency of bubble detachment is not sufficient to cool the heating surface.

For the cases with a very high level of confinement, $s \leq$ 0.5mm, the enhanced boiling is a consequence of the deformed bubbles which increase the area of the liquid film between the vapor bubble and the wall, allowing an efficient heat transfer, as explained by Katto [13] and Ishibashi and Nishikawa [2].
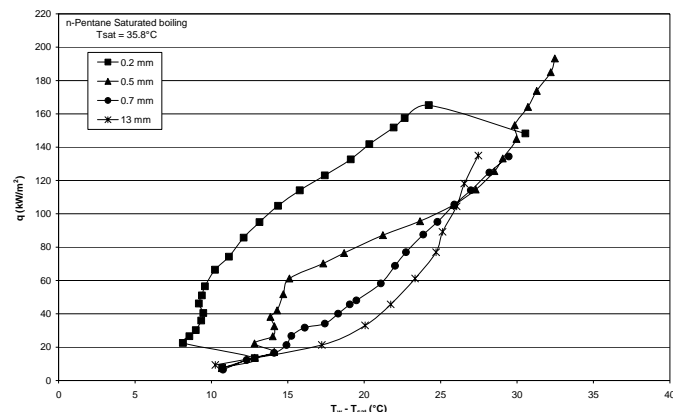

Figure 5. Partial boiling curves for n-Pentane.

Figure 6 shows the heat transfer coefficient, $h = q/(T_w - T_{sat})$, against the heat flux for saturated n-Pentane. For the case with higher confinement we can observe a decrease in h when q is higher than 110kW/m².

The high values for heat flux can be explained by thermophysical properties of n-Pentane, as mentioned in Peng *et al.* [14]. These authors had found that liquids with greater liquid/vapor density differences, higher latent heats and larger thermal diffusion coefficients need larger heat fluxes to initiate nucleation.
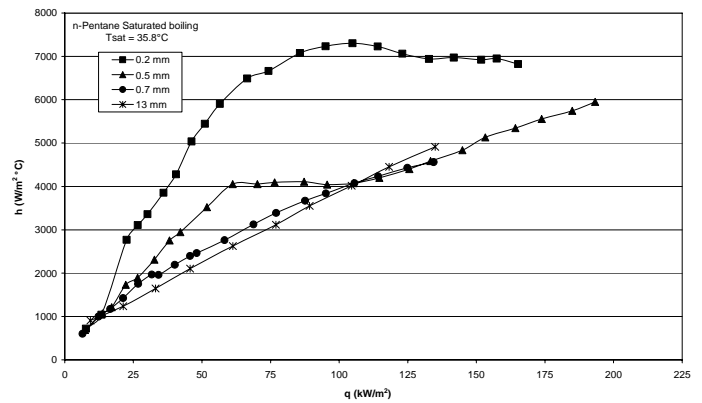

Figure 6. Confinement effect on the heat transfer coefficient against the heat flux.

Dryout Heat Flux

Figures 7 and 8 show the diagrams representing the method used to determine the experimental dryout heat flux, DHF, and the wall superheating at the beginning of the wall dryout, ΔT_{DHF}, for s = 0.2mm. In Fig. 7, the horizontal line slope on the maximum point of the curve of h versus q, obtained by fitting the experimental points, allows the determination of the h_{DHF}, and the corresponding heat flux, q_{DHF}. Taking a horizontal line from q_{DHF} and crossing the experimental partial boiling curve q x ΔT_{sat}, Fig. 8, the corresponding experimental wall superheating ΔT_{DHF}, can be obtained at the point of abscissa intercepted by the vertical line.
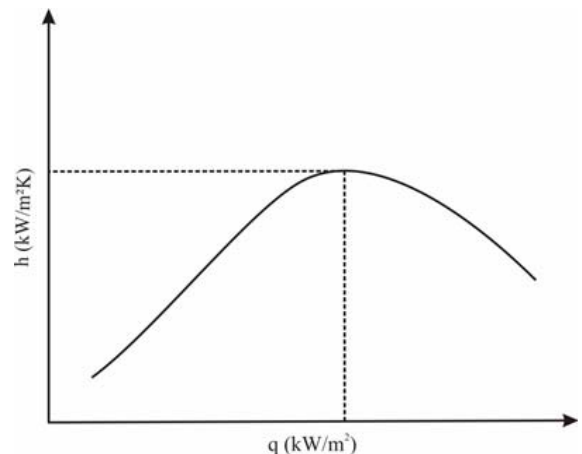

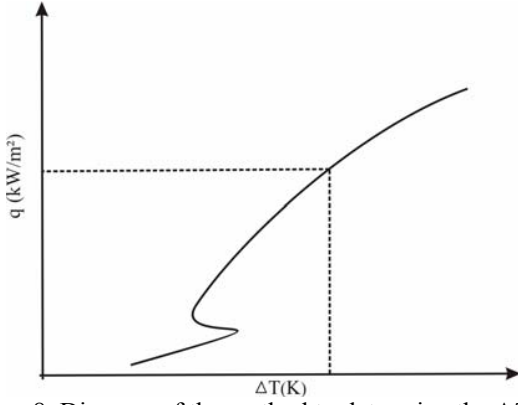Figure 7. Diagram of the method to determine the q_{DHF}.

Figure 8. Diagram of the method to determine the ΔT_{DHF}.

In Tab. 2 is showed the experimental values of q_{DHF}, h_{DHF}, ΔT_{DHF} (determinate from Figs. 5 and 6) and the ratio between q_{DHF} and the theoretical value of $q_{max,Z}$, calculated by Eq. (3).

Table 2. Experimental values for n-Pentane.

s (mm)	q_{DHF} (kW/m²)	h_{DHF} (W/m²°C)	ΔT_{DHF} (°C)	$q_{DHF}/q_{máx,Z}$ (%)
0.2	105	7303	14.36	43.20

The theoretical values of $q_{máx,Z}$, for n-Pentane is 243kW/m² and the value of $q_{DHF}/q_{máx,Z}$, shown in Tab. 1 for $s = 0.2$mm is 43.2%. For s = 0.5 and 0.7mm, is not possible to define the q_{DHF} for this method because the experimental points are not enough clear.

For $s = 13.0$mm, the graphics indicate unconfined boiling, where, the heat transfer coefficient increases as the heat flux increases. In this work, because of technical limitation from the heating system, only the dryout heat flux for the confined case has been obtained.

This procedure to calculate the dryout heat flux is more conservative as show the results.

Empirical correlations for nucleate boiling

The heat transfer coefficient, in boiling heat transfer, is defined by: $h=q(T_w-T_{sat})^{-1}$, where, q, T_w and T_{sat} represent the heat flux, the temperature of the heating surface in contact with the working fluid and the saturation temperature.

Among the great number of existing empirical and semi-empirical correlations to calculate h, for the nucleate boiling regime developed, those of Cooper (1984), Rohsenow (1952) and Stephan and Abdelsalam (1980) continue to be employed by several research groups and are presented as follows [1]

Correlation of Cooper (1984):

$$h_{cooper} = 55 p_r^{\ b} \left(-0.4343 \ln p_r\right)^{-0.55} M^{-0.5} q^{0.67} \qquad (4)$$

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

Correlation of Rohsenow (1952):

$$h_{Rohsenow} = \mu_l h_{lv} \left(\frac{g(\rho_l - \rho_v)}{\sigma}\right)^{0.5} \left(\frac{c_{pl}}{C_{sf} h_{lv} P_r^{\ s}}\right)^3 \Delta T_p^{\ 2} \qquad (5)$$

where ρ_l, c_{pl}, Pr_l represent the viscosity, the specific heat and the Prandtl number of the liquid. ΔT_w represents the superheating of the heating wall, $\Delta T_w=(T_w-T_{sat})$, and C_{sf} is dependent on the material of the heating wall, the surface

roughness and the working fluid and $r=0.333$ e $s=1.7$, see [1].

Correlation of Stephan and Abdelsalam (1980):

$$h_{sa} = 207\left(\frac{k_l}{d_b}\right)\left(\frac{q d_b}{k_l T_{sat}}\right)^{0.745}\left(\frac{\rho_l}{\rho_v}\right)^{0.581} Pr_l^{\ 0.533} R_p^{\ 0.133} \qquad (6)$$

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

$$d_b = 0.0149\theta\left(\frac{2\sigma}{g(\rho_l - \rho_v)}\right)^{0.5} \qquad (7)$$

Figures 9 shows the comparison of experimental heat transfer coefficients, for s=13mm, with the values of h calculated by Eqs.(4-7), corresponding to the empirical correlations of Rohsenow (1952), Stephan and Abdelsalam (1980) and Cooper (1984). The values of h are close to those calculated by the correlation of Stephan and Abdelsalam with an average deviation of 5%. For the correlation of Cooper and Rohsenow with $C_{sf}=0.0154$, the average deviation were 55%.
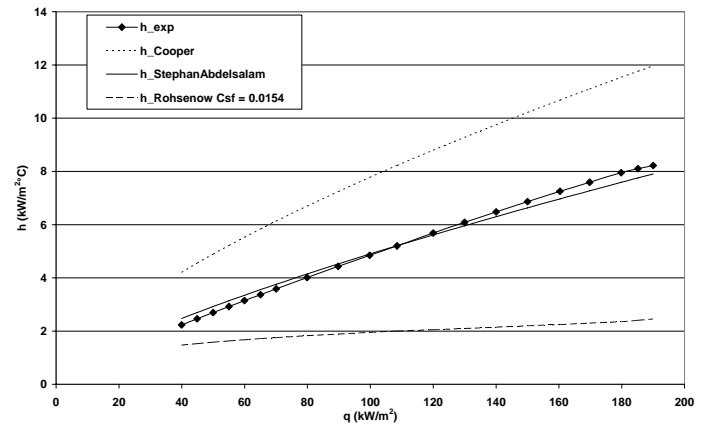

Figure 9. Comparison of experimental heat transfer coefficient with the correlations, for s=13mm.

A heating function was analyzed and the results for n-Pentane with s = 0.2 and 13 mm for both heating modes, increasing and decreasing, are shown in Fig. 10. In general, the differences in the wall superheating were less than 2°C with a lower superheating for the case with decreasing heat flux.
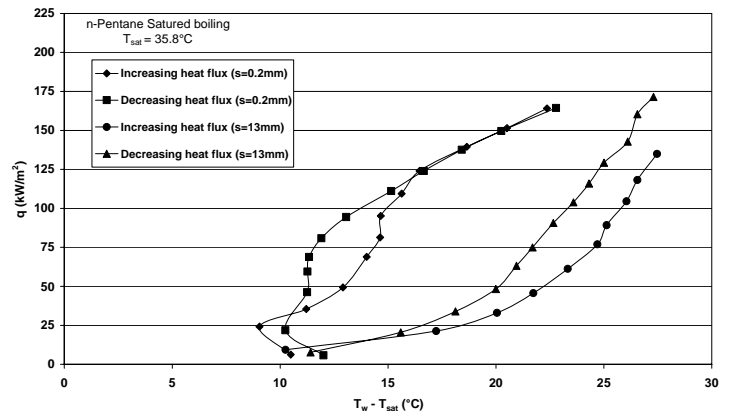

Fig. 10. Influence of the heating mode.

CONCLUSIONS

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

(i) As a general trend the heat transfer coefficient increases when the confinement increases, corresponding to the decrease in the distance between the heating surface and an unheated horizontal surface.

(ii) The value of $q_{DHF}/q_{máx,Z}$ was 43.2% and the trend of a decrease in the dryout heat flux in the confined case, was only partially found in this study. This uncertainty in the tendency of DHF is attributed to the chaotic phenomenon of coalesced bubble retention in the confined space.

(iii) The comparison of our results for unconfined boiling with some well known correlations such as those of Cooper (1984) and Rohsenow (1952) shows great differences between them. However the experimental values practically overlaps the correlation values of Stephan and Abdelsalam (1980).

ACKNOWLEDGEMENTS

NOMENCLATURE

Symbol	Quantity	SI
c_{pl}	specific heat	kJ/kgK
g	acceleration due to gravity	m/s^2
h	heat transfer coefficient	kW/m^2°C
h_{lv}	latent heat of vaporization	kJ/kg
k	thermal conductivity	W/mK
L	capillary length	mm
M	Molecular height	k/kmol
p	pression	Pa
P_r	Prandtl number	
q	heat flux	kW/m^2
R	roughness	μm
s	gap of confinement	mm
T	temperature	°C

Greek symbols

ΔT	wall superheating	K, °C
θ	inclination angle	°
ρ	density	kg/m^3
μ	viscosity	kg/ms
σ	surface tension	N/m

Subscripts

CHF	Critical heat flux
DHF	Dryout heat flux
l	liquid
lv	liquid-vapor
r	reduced
sat	saturation
v	vapor
w	wall

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