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Transient analysis of boiling heat transfer in periodic drying out miniature pools

S. Filippeschi, G. Salvadori *

Department of Energetics Lorenzo Poggi, University of Pisa, Via Diotisalvi 2, 56100 Pisa, Italy

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ABSTRACT

The evolution of the two-phase thermal control technique is moving in the direction of the use of devices which operate in stable periodic or chaotic unsteady regimes. In these devices the heat transfer coefficient relative to the evaporator walls therefore changes over time and it is hardly predictable, especially in the case of boiling regime. This paper deals with the analysis of the boiling coefficient evolutions over time during the operations of a thermal control devices which periodically operates, the Periodic Two-Phase Thermosyphon (PTPT). An experimental technique for measuring the transient boiling heat transfer coefficient in a thick flat evaporator is shown.

This technique has been used to measure boiling heat transfer coefficients over time in the case of evaporators with small pools of liquid (lower than 64 ml) which periodically dries out. The FC72 fluid has been used in the experiments. The results show that no nucleate boiling regime has been observed in the evaporator, which works in an unstable transitional boiling regime with relative heat transfer coefficients lower than those typical of nucleate boiling regimes (lower than 50%).

1. Introduction

In order to remove high heat fluxes (10–100 W/cm^2) from a hot surface, devices with a two-phase circulating working fluid are used. These devices can be divided into two main groups according to their operative mode: steady and unsteady state heat transfer devices.

Among the steady-state devices, the most studied are the capillary driven two-phase loops, such as the Capillary Pumped Loop (Platel et al., 1995; Chen and Lin, 2001) and Loop Heat Pipes (Hoang et al., 2003; Riehl and Dutra, 2005; Maydanik, 2005).

Among the unsteady devices, the most analysed in the literature are the Pulsating Heat Pipes (PHPs) (Akachi et al., 1996; Dobson, 2004), while the Periodic Two-Phase Thermosyphons (PTPT) (Filippeschi, 2006) are less known. All these devices periodically operate at different operative frequencies: PHPs operate with high frequencies 1–10 Hz (Khandekar and Groll, 2004), whereas PTPTs generally operate with low frequencies (0.001–10^{-4} Hz) (Filippeschi, 2006) but in the case of miniature evaporators (Filippeschi et al., 2004) they can operate with higher frequencies (0.1–0.6 Hz).

Because of their simple manufacture, their quick dynamic response to thermal variations and, more importantly, because of their high heat transport capability in systems which are able to work even against gravity (Delil, 2001), unsteady two-phase loops are a suitable solution in the case of high packaging thermal control applications.

The heat and mass transfer of a generic unsteady thermally driven loop has not been completely modelled, but there are a variety of mathematical codes that attempt to correctly interpret the physical phenomena which are involved both in PHP and in PTPT devices. Papers dealing with PHP heat and mass transfer models are more numerous than those which deal with PTPTs (Zhang and Faghri, 2003; Dobson, 2003; Sakulchangsatjatai et al., 2004; Holley and Faghri, 2005).

In the case of PTPT devices the literature offers very few mathematical models. In Filippeschi (2006) a broad review of PTPT devices and their main models have been presented.

All these models, both for PHPs and PTPTs, are developed by using an average value of heat transfer coefficient (constant over time) to predict the convective heat transfer regime inside the evaporator. But in miniature unsteady devices the fluid-dynamic conditions quickly change over time and the heat transfer coefficient can be strongly influenced by this transient, especially in the boiling regime.

In order to understand how the heat is linked with the mass transfer in a periodic natural circulation loop, the heat transfer coefficient evolution over time must be, therefore, known.

This paper deals with an experimental and theoretical analysis of periodic boiling heat transfer regime relative to a flat surface wetted by a small pool of liquid which periodically dries out. The periodic boiling heat transfer regime is typical of the PTPT evaporator operation mode but the results, obtained from the present work, can be extended to a generic flat surface where the fluid-dynamic conditions change with frequencies lower than 0.5 Hz.

* Corresponding author. Tel.: +39 0502217152; fax: +39 0502217150.
E-mail address: g.salva@ing.unipi.it (G. Salvadori).

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2. Experimental arrangements

2.1. Test apparatus

The PTPT loop used in the experiments consists of three main components: an evaporator with an internal volume of 238 ml, a condenser and an accumulator with internal volumes of 10 ml and 70 ml, respectively. All these elements are connected to each other by different pipes, as shown in Fig. 1, (internal volume of 34 ml). All the connections are flexible polyethylene pipes (internal diameter of 4 mm). Inside the loop two stainless steel solenoid electrovalves of 2-way type normally closed, with an opening response time of 5 ms are inserted.

The evaporator is a vessel made of aluminium, except for its base, which is made of copper. The shape of this base, the dissipator, is shown in Fig. 1. The upper surface of this dissipator (2.77 cm^2) is wetted with the FC-72 working fluid. The working fluid has been chosen for its following physical properties: high dielectric constant, low tension surface, low boiling temperature, good boiling heat transfer coefficients and a high saturation curve slope at environmental temperature. In Neeper (1998) the positive influence of the aforesaid working fluid properties on the PTPT heat transfer performance is analyzed.

The lower surface of the dissipator is heated by an electric heater which can supply heat fluxes up to 0.150 MW/m^2. The mass of the copper dissipator is 0.156 kg. A ring, made of PTFE, thermally disconnects the upper portion of the lateral surface of the copper dissipator from the aluminium case of the evaporator. The lower portion of the lateral surface (cylindrical surface with diameter $D_b = 0.038$ m and height $\Delta z = 0.01$ m, see Fig. 2) is not thermally insulated and is surrounded by air at a temperature of 297 ± 2 K. This portion is covered with a film of known emittance ($\varepsilon = 0.94$). This solution allows the two-dimensional temperature distribution of the lower surface of the dissipator to be measured by an infrared thermo-camera.

The condenser, air-cooled by a fan, is made of a $64 \times 78 \times 8$ mm^3 aluminium plate with a side covered with 22 rectangular fins, 28 mm high, 78 mm long and 1.3 mm thick. Inside the plate a serpentine groove has been manufactured, in such a way as to obtain a 4×4 mm^2 section surface. This serpentine groove is covered with a 2 mm thick plate made of Lexan©, to allow the condensing two-phase flow to be viewed.

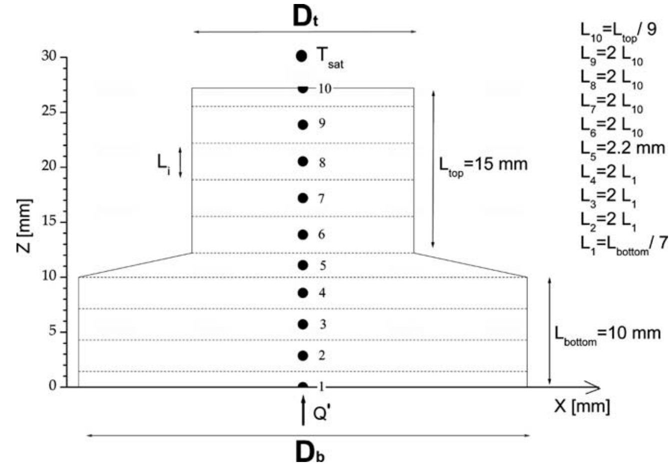

Fig. 2. Finite difference approximation of the copper dissipator.

2.2. Instruments, control systems and procedures

In addition to the test apparatus, the experimental facility consists of a control device, an acquisition system and of a set of instruments: one gauge glass for measuring the level of the liquid, five thermocouples and two pressure transmitters.

The gauge glass is located inside the accumulator: its accuracy is ±0.5 mm. The temperature of the surface wetted by the working fluid T_1 is measured by a K-type thermocouple 0.5 mm thick. It is located 1.5 mm under the surface of the copper dissipator. All the other thermocouples are of K-type, 1 mm thick, and they measure the working fluid temperatures in the loop, as shown in Fig. 1. In particular, inside the evaporator, the temperature of the liquid pool is measured by thermocouple T_2 placed 2 mm above the hot surface of the dissipator, whereas the temperature of the vapour is measured by thermocouple T_3 placed 10 mm under the top of the evaporator.

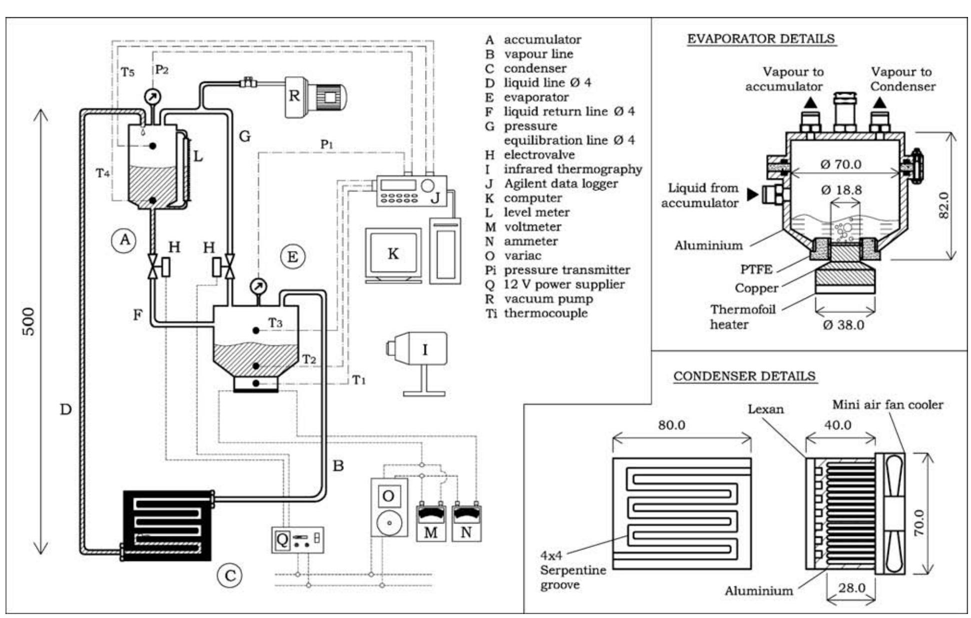

Fig. 1. Experimental facility.

The vapour pressure inside the accumulator and the evaporator are measured by two Druck PTX 75-11 transmitters (0–2.5 bar, accuracy 0.25% of full scale range).

The infrared thermo-camera, which is used to measure the temperature map of the lower part of the external surface of the dissipator, has a focal plane array of two-dimensional micro-bolometric sensors, characterised by a digital resolution of 320×240. These sensors are sensitive to radiation with wave lengths between 8 and14 μm. The minimum thermal resolution is 0.15 K. The optical system consists of a 35 mm lens, with a field of view of $25.8° \times 19.5°$. The distance between the hot surface and the objective lens is 0.3 m and the pixel size is 0.42×0.42 μm. The infrared system captures a frame every 0.03 s.

All the temperatures and pressures within the device are sampled by an Agilent 34970A data acquisition system (thermal resolution 0.1 K, accuracy ±0.5 K, minimum acquisition frequency 3 Hz) and stored in a personal computer.

The electrovalves are activated by a control system, which consists of a digital timer (0.01 s–1000 h), covering various operational modes. In the experiments it is set with a period when the valves are off Δt_{off} and a period when the valves are on Δt_{on}. The choice of these two periods has been made in a previous experimental activity in order to allow the complete discharge of the liquid inside the evaporator. Δt_{off} and Δt_{on} could be, however, dynamically and automatically changed in relation to the increasing and the decreasing rates of the dissipator temperature.

Before to start the experiments a vacuum (10^{-2} bar) has been realised inside the evaporator and the device is charged with the working fluid. At this time the heat load is supplied to the evaporator and the device operates for a minimum of 2 h. After this time the fluid is degased with two valves placed on the top of the evaporator and the accumulator. This operation was repeated three times before of every test. After the fluid has been degassed, the test starts as soon as the input power is supplied to the dissipator and the timer is set.

3. PTPT operating characteristics

The operational modes of a generic PTPT are described in Filippeschi (2006) and quickly summarised here. The evaporator is continuously heated by a power input, while the condenser and accumulator exchange heat with the environment. The heat dissipated by the connecting pipes is negligible.

A single heat transfer cycle can be described by observing the evolution of the temperature and pressure oscillations of the working fluid inside the evaporator.

A single cycle starts when the whole amount of the liquid, which periodically circulates in the loop, is inside the evaporator. The power input which is supplied to the working fluid causes an increase in temperature and pressure of the liquid until a saturation condition is achieved, then the liquid starts to evaporate. From this time the volume of the liquid pool changes over time.

The vapour generated inside the evaporator goes into the condenser, where it is condensed and subcooled creating a column of liquid between the condenser and the accumulator. The pressure inside the evaporator increases again to lift the liquid column up to the accumulator. As the condensed liquid reaches the accumulator, the pressure and temperature inside the evaporator no longer increase rapidly, and a liquid mass flow rate into the accumulator is observed.

As soon as a given volume V_T of working fluid is transferred from the evaporator to the accumulator, the electrovalves open the connecting pipes between the evaporator and the accumulator. The accumulated cold liquid returns to the evaporator, pushed down by the gravity force, the heat transfer cycle is closed and all the operations are repeated. After few cycles, the device achieves a stable regime (Fantozzi et al., 2003).

4. Transient analysis of the boiling heat transfer coefficient

4.1. The transient heat transfer coefficient measurement

The heat transfer coefficient h is generally defined as the ratio between the heat flux dissipated by the exchanging surface and the wall superheat $T_W - T_S$ (where T_W is the wall temperature and T_S the temperature of the wetting fluid). In the boiling regime h mainly depends on the wall superheat and in a transient boiling regime it is hard to predict.

The simplest way of measuring this parameter h over time is to suppose that the temperature differences at different points inside the copper dissipator are negligible, so that the whole mass M is at the temperature T_W. In this way $h(t)$ can be approximated by using a lumped capacitance method, and it can be indirectly measured by Eq. (1):

$$h_{(n)} = \left(Q' - \frac{M \cdot c \cdot (T_{W(n+1)} - T_{W(n)})}{\Delta t}\right) \cdot \frac{1}{S \cdot (T_{W(n)} - T_{S(n)})} \tag{1}$$

where Q' is the power supplied by the thermo-heater, n is the generic time sampling increment and Δt is the time sampling rate ($\Delta t = 3$ s in the experimental activity).

This procedure, however, introduces high prediction errors if the Biot number is higher than 0.1.

The Biot number evaluated for the PTPT object of the paper is about 0.5 and the temperature distribution inside the evaporator wall (copper dissipator) cannot be therefore neglected. The evaluation of Biot number has been made by taking into account the typical heat transfer coefficients relative to a steady-state pool boiling of FC-72 over a flat surface (10000–12000 W/m^2 K as indicated in Guglielmini et al. (2002), Rainey et al. (2003)).

To approximate the function $T(r,z,t)$ in the copper dissipator, a transient finite difference method has been used. The copper dissipator has been divided into 10 nodes, as Fig. 2 shows, with a variety of control volumes V_i (diameter D_i and length L_i, $1 \leqslant i \leqslant 10$). The control volume of each node has been chosen in order to obtain a Biot number lower than 0.1. The time increment Δt used in the time approximation is equal to the sampling rate, which, during the experimental activity, was 3 s. By achieving an energy balance on each generic node i, the implicit form of the finite difference equation, for the time increment n, is given by Eq. (2);

$$\frac{T_{i-1}^{n+1} - T_i^{n+1}}{R_{i-1,i}} - \frac{T_i^{n+1} - T_{i+1}^{n+1}}{R_{i,i+1}} = M_i \cdot c \cdot (T_i^{n+1} - T_i^n)/\Delta t \tag{2}$$

where $R_{i-1,i}$ is the conductive thermal resistance between nodes i and $i-1$, M_i is the mass of node i and $c = 385$ J/(Kg K) is the copper specific heat.

The energy balances of the boundary nodes 1 and 10 are expressed by the following equations:

$$Q' - \frac{T_1^{n+1} - T_2^{n+1}}{R_{1,2}} = M_1 \cdot c \cdot (T_1^{n+1} - T_1^n)/\Delta t \tag{3}$$

$$\frac{T_9^{n+1} - T_{10}^{n+1}}{R_{9,10}} - h^{n+1} \cdot S_d \cdot (T_W^{n+1} - T_S^{n+1}) = M_{10} \cdot c \cdot (T_W^{n+1} - T_W^n)/\Delta t \tag{4}$$

where T_W^{n+1} and T_S^{n+1} are the temperatures experimentally measured at time $t = (n+1) \cdot \Delta t$, respectively by the thermocouples T_1 and T_2. The energy balance written for every node creates a linear system of 10 equations with nine unknown temperatures (T^{n+1} in the nodes 1–9) and one unknown heat transfer coefficient h^{n+1}.

The temperature of the node 10, at the generic time step n, is considered equal to $T_{1(n)}$, which is experimentally measured at every sampling time. The set of linear equations can therefore be solved with the following initial condition:

$$T_{i(0)} = T_{W(0)} \quad \text{for } 1 \leqslant i \leqslant 10 \tag{5}$$

where $T_{W(0)}$ is the temperature measured by the thermocouple T_1 at the start of the test (environmental temperature).

In this way, it is possible to determine the evolution of the heat transfer coefficient over time during the start-up and, if a periodically steady-state regime is reached, along a portion of the cycle. This technique supposes that the heat losses from the external surface of the copper dissipator are negligible: this is true as the heat transfer coefficient is over 1000 W(m^2 K). However, this contribution can be determined by sharing all the heat losses from the external surface of the copper dissipator into two rates, the top and the bottom rate. At the top (L_{top} in Fig. 2), the external surface is separated from the environmental air by a thick layer of insulating material (the heat losses are neglected) whereas the heat losses of the bottom surface (L_{bottom} in Fig. 2) are due to free convection from vertical surface of a cylinder. These heat losses can be, therefore, accounted by modifying the Eq. (2) which becomes, for each node within the range $1 \leqslant i \leqslant 5$:

$$\frac{T_{i-1(n+1)} - T_{i(n+1)}}{R_{i-1,i}} - \frac{T_{i(n+1)} - T_{i+1(n+1)}}{R_{i,i+1}} - Q'_d = M_i \cdot c \cdot (T_{i(n+1)} - T_{i(n)})/\Delta t \tag{6}$$

where on the "left hand side" there is the environmental heat losses Q'_d, which has been calculated using the generic Squire–Eckert correlation (Lienhard, 1981), as applied to vertical cylinders (Cebeci, 1974).

4.2. Experimental results

The experimental results show that three different heat transfer coefficients have been measured during the test: h_1 is evaluated by neglecting the temperature distribution in the copper dissipator (Eq. (1)), h_2 is calculated by taking into account the temperature distribution in the copper dissipator and neglecting the heat losses into the environment (Eqs. (2)–(4)) and, lastly, h_3 is calculated by taking into account both the contributions (Eq. (6)).

Fig. 3 shows the three different heat transfer coefficient on dependence of the non-dimensional time $\tau = t/t_c$ as soon as a stable periodic regime has been reached, in the case of a power supply of 40 W (specific heat flux $q' = 0.144$ MW/m^2) and a transferred liquid volume $V_T = 64$ ml. Fig. 3 shows that the temperature distribution in the copper dissipator can be neglected, except for the time at the start and the end of the single period, because the evolutions of h_1 and h_2 are in good agreement but the evolution of h_3 is quite different. The environmental heat losses introduce errors higher than 7% especially at the start and the end of each single heat transport cycle.

In the case of the calculation of h_3, the temperature distribution in the copper dissipator has been calculated and compared with those measured with a thermo-camera at three different times. The method used to measure this temperature distribution, along with the uncertainties affecting this measurements, is described in Filippeschi et al. (October 2006). An example of this comparison is shown in Fig. 4 in the case of a power supply of 40 W and a volume V_T of 3 ml. The agreement between the measured and calculated temperatures is good with differences lower than 2%. The same comparison has been repeated for each volume V_T which has been tested and the differences between the measured and calculated values are, however, lower than 2%.

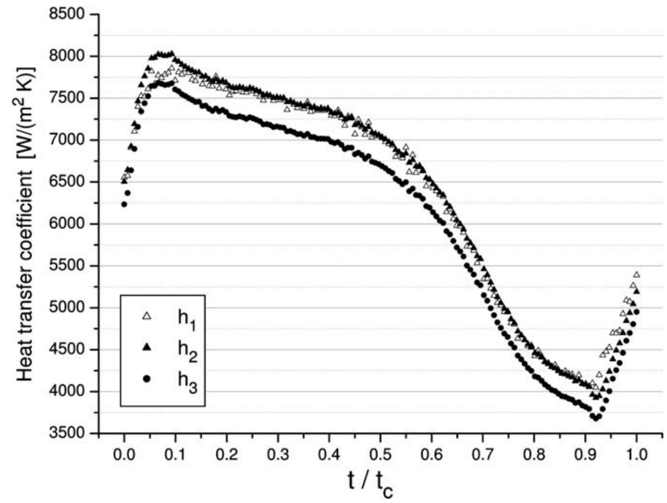

Fig. 3. Evolutions of the heat transfer coefficients calculated by taking into account different contributions ($Q' = 40$ W, $q' = 0.144$ MW/m^2, $V_T = 64$ ml).

The influence of the starting volume of small working fluid pools, which periodically dry out, upon the transient boiling regimes on a flat surface has been studied by measuring the heat transfer coefficient h_3 over time. Fig. 3 shows that the heat transfer coefficient h_3 varies slightly over time for more than 50% of a single heat transport cycle; it is approximately 7250 ± 500 W/m^2 K. If t/t_c is included within the range 0.6–0.9, the heat transfer coefficient falls abruptly.

The main technical interest is, therefore, focused on the knowledge and prediction of the heat transfer regime which is present in the evaporator during the time range (t_4–t_1), where the temperature of the liquid pool remains constant and the volume of the liquid pool gradually decreases over time. Can this particular heat transfer regime be considered a steady-state periodic pool boiling regime or it is considered a transient boiling regime, with the heat transfer coefficients lower than the expected ones?

In order to answer this question several heat transfer coefficient functions $h_3(\tau)$ have been compared with the data which are in the literature relative to a steady-state pool boiling heat transfer from an infinite flat surface to the fluid FC-72.

In a previous paper (Filippeschi et al., June 2006) the authors have carried out tests on a steady-state pool boiling on a flat surface with FC-72, to investigate the effect of sidewall-confinement on the nucleate boiling in miniature pools. These data have been compared with some of the correlations used to predict the nucleate boiling heat transfer coefficients. The correlation that has the best agreement with the experimental data has been that of Rosenhow and Hartnett (1973) which depends on the constant C_{sf}, an empirical correction for typical surface condition. It varies between 0.006 and 0.018 for most fluids with a copper surface and typically has errors in heat flux of 100% and errors in wall superheat of about 25%. If this constant is determined by fitting the experimental data shown in Filippeschi et al. (2004) in a least squared sense ($C_{sf} = 0.054$), the errors on the wall superheat falls to 13%. Rohsenow's correlation has been used to predict the heat transfer coefficient h_R relative to a steady-state pool boiling heat transfer regime for FC-72 on an unconfined flat surface.

The evaporator temperature evolutions shows a transient boiling heat transfer regime, but the saturation temperature remains approximately constant over time (±1 K) and the pressure is almost constant for large portions of the cycle. The influence of the saturation pressure on the heat transfer can, therefore, be neglected.

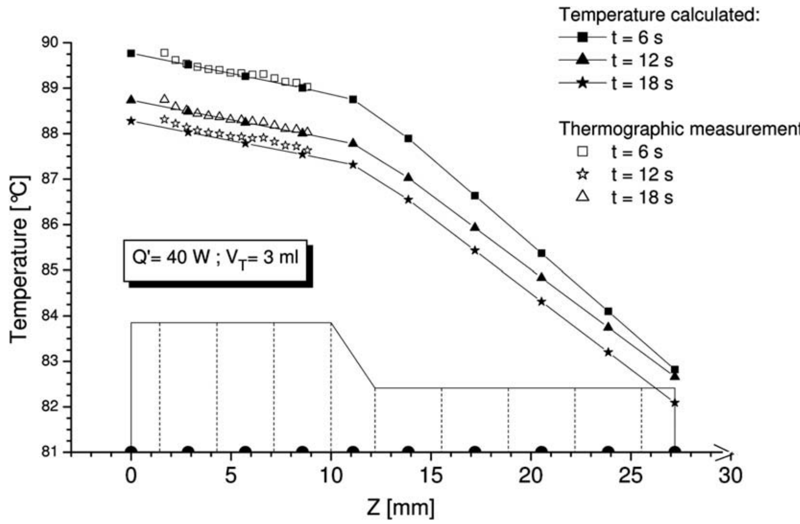

Fig. 4. Comparison between temperature distributions calculated and measured with a thermo-camera in the copper dissipator ($q' = 0.144$ MW/m^2).

Fig. 5 shows the evolution over time of the ratio between the heat transfer coefficient h_3 previously calculated and the Rohsenow heat transfer coefficient h_R for different values of working fluid transferred volume, with $Q' = 40$ W. Instead of time, the non-dimensional parameter τ ($\tau = t/t_c$) has been used, because different transferred volumes shows different time periods t_c.

Fig. 5 shows that the heat transfer coefficient reaches a maximum and then gradually falls over time. For every V_T investigated, the maximum value of h is observed in the range $0.05 < \tau < 0.3$. For $V_T = 64$ ml the maximum value of h is close to h_R (h_3/h_R reaches the value 1.05, see Fig. 5), whereas for $V_T \leqslant 40$ ml it is remarkably lower than h_R ($h_3/h_R < 0.5$ for $V_T = 3$ ml).

It may be argued that a local nucleate boiling heat transfer can be observed if the thermal capacity of the miniature pool is high; otherwise different unstable boiling regimes are observed. This assumption is graphically illustrated in Fig. 6, where the curves of transient boiling for different volumes are compared with the steady-state boiling data reported in Rainey et al. (2003), Filippeschi et al. (2006).

Fig. 6 shows that in the case of 64 ml the cycle starts with a high wall superheat and the cold liquid coming from the accumulator gradually leads the heating surface towards boiling points typical of steady-state curves. At a certain time, something occurs within the pool which causes a deterioration of the local heat transfer regime, so the heating surface tends to find an equilibrium close to film boiling regimes. The reduction of the heat transfer performances with reduction of the volume of the liquid pool has been studied by Westwater et al. (1986) and by Elkassabgi and Lienhard (1987). The first group noticed that any reduction in the confinement diameter of the pool determines an unstable condition of the heat transfer. They characterised a critical diameter below which the heat transfer coefficient falls steeply: this diameter is correlated with Taylor wavelength. Lienhard et al. studied the reduction of the fluid volume effect and the sidewall-confinement effect on the CHF. A more thorough experimental investigation must, however, be carried out on the effect of these two parameters.

The transient behaviour of the thick flat sample observed in the present experimental activity is in agreement with what Duluc et al. have observed in Duluc et al. (2004), who observed a wall superheat overshoot before a stable boiling regime was reached during the heating of the sample. During the heating time, therefore, an

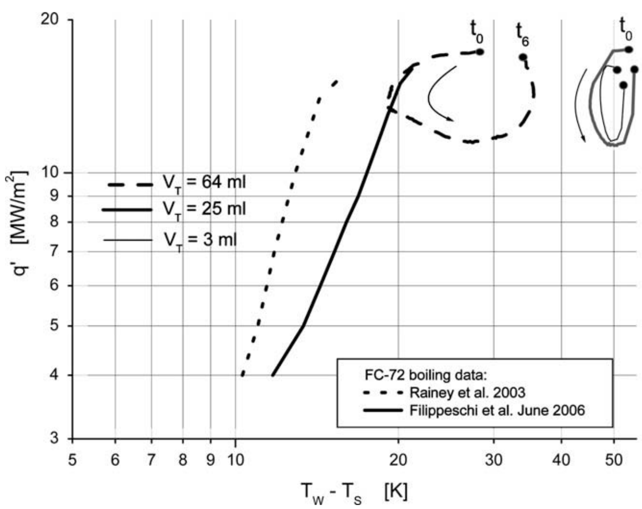

Fig. 6. Boiling curve for a thick flat surface with FC-72 at 1bar compared with the evolutions in time of the boiling regimes in a PTPT.

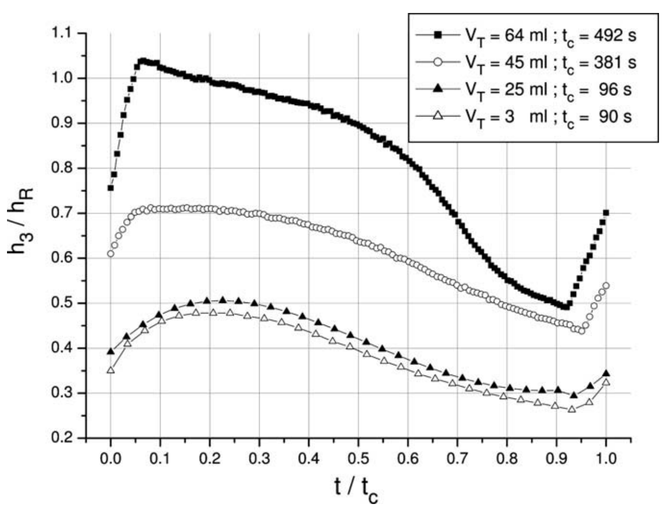

Fig. 5. Evolutions of the ratio h_3/h_R for different volume for an input supply of 40 W ($q' = 0.144$ MW/m^2).

amount of heat was stored in the sample and it was supplied again to the fluid as soon as the boiling started.

In the case of a miniature evaporator which operates in the periodic regime, the very frequent fluid-dynamic changes prevent the fluid from reaching a nucleate boiling regime and consecutive heat charges and discharges bring about wall superheat oscillations. The heat transfer coefficients are, therefore, higher than those typical of nucleate boiling regimes. In conclusion, in the case of a miniature evaporator with a miniature pool of liquid, no nucleate heat transfer coefficients can be used to predict the real boiling heat transfer. In order to predict these coefficients an experimental inquiry into the boiling onset of a thick flat sample must be carried out by taking into account liquid pools with finite thermal capacity in general and miniature pools in particular.

5. Theoretical analysis of evaporator working with a periodically drying-out liquid miniature pool

In order to analyse how the fluid-dynamic regimes inside the miniature pool of an evaporator influence the thermal evolution of the heating surface in a PTPT device, the evaporator can be schematised according to the following simplified model. The evaporator can be viewed as a vessel with a metallic base, with the other walls made of insulating material, as shown in Fig. 7.

The bottom surface of the metallic base is heated by a constant heat flux q', whereas the top surface heat transfer rate is free convection with a two-phase fluid (saturation temperature and pressure indicated by T_S, p_S, respectively). It is assumed that the top surface convective heat transfer coefficient $h(t/t_c)$ varies periodically with time with period t_c and these variations are determined by different fluid-dynamic conditions of the working fluid close to the surface.

In this model the environmental heat losses have been neglected. The temperature of the metallic base over time can be calculated by the following differential equation:

$$q' = \frac{C}{S} \cdot \frac{\mathrm{d}T(t/t_c)}{\mathrm{d}t} + h(t/t_c) \cdot (T(t/t_c) - T_S) \tag{7}$$

where C is the thermal capacity of the metallic base and S is the exchanging surface. Moreover, Eq. (7) can be rewritten by defining the temperature difference $\theta = T(t/t_c)\text{-}T_S$ and by using the non-dimensional parameter τ, so that it becomes:

$$\frac{\mathrm{d}\theta(\tau)}{\mathrm{d}\tau} + h(\tau) \cdot \frac{S \cdot t_c}{C} \cdot \theta(\tau) - \frac{q' \cdot S \cdot t_c}{C} = 0 \tag{8}$$

If the function $h(\tau)$ is known, Eq. (8) becomes a first-order linear differential equation with variable coefficients and its solution can be exactly calculated (Smirnov, 1999) by using the starting condition $\theta(0) = \theta_0$:

$$\theta(\tau) = \mathrm{e}^{\left(-\frac{S \cdot t_c}{C} \cdot \int_0^\tau h(u)\mathrm{d}u\right)} \cdot \left(\theta_0 + \frac{q' \cdot S \cdot t_c}{C} \cdot \int_0^\tau \mathrm{e}^{\left(\frac{S}{C} \cdot \int_0^v h(u)\mathrm{d}u\right)} \mathrm{d}v\right) \tag{9}$$

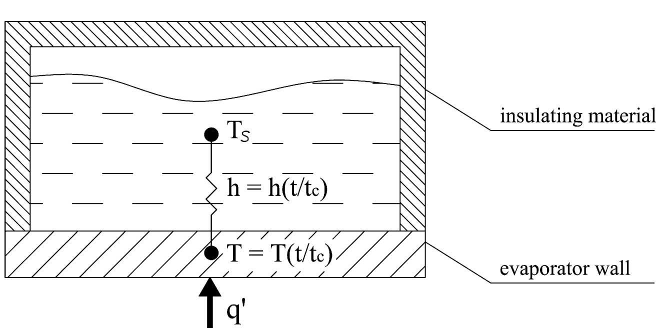

Fig. 7. The evaporator: a simplified scheme.

In the case of complex fluid-dynamic motions inside a PTPT evaporator, the function $h(\tau)$ is not known a priori; a few considerations can, however, be made according to the experimental results which are shown in the Section 4.

The typical heat transfer coefficient functions can, therefore, be supposed according to the experimental results previously described. These functions assume the shape marked with the letter (a)–(d) in Fig. 8.

The curve with a long period where the heat transfer coefficient remains constant over time represents the theoretical expected evolution of the heat transfer coefficient. The foregoing experiments have shown that the heat transfer coefficient evolves similarly over time in the case of large pool volume, but in the case of small liquid pool the time when the heat transfer coefficient is almost constant falls to zero.

Curve (a) is, therefore, characteristic of large volume evaporators, with large volumes of liquid pool. Curve (d) refers to small volume evaporators, with a volume of liquid pool for which the time period characterized by constant heat transfer coefficient is not observed. Curves (b) and (c) are referred to intermediate volumes of liquid pool.

By hypothesising that the evolution over time of the heat transfer coefficient are described by the polygonal curves of Fig. 8, Eq. (9) is well posed and can be solved. The comparison between the wall superheat evolutions experimentally obtained with $q' = 0.144\ \mathrm{MW/m^2}$ and those calculated with Eq. (9) by using the curve (b) is shown in Fig. 9.

A good agreement between the experimental and the theoretical data is observed for a large portion of the cycle. As the amount of liquid inside the evaporator is very low the heat transfer function which has been assumed does not correctly interpret the real behaviour of the device. The same topic has been observed for other volumes and other curves.

The wall superheat evolution over time of a single period are shown in Fig. 10. The periodic evolution of the heat transfer coefficient is strongly influenced by the evaporator wall superheat θ, even if the maximum and minimum values of the heat transfer coefficient in a single period are constant for each curve ($h_{\max} = 8000\ \mathrm{W/m^2\,K}$, $h_{\min} = 1000\ \mathrm{W/m^2\,K}$). Fig. 10 shows that the maximum wall superheat is approximately 30 K for curve (a) and 60 K for curve (d). The shape of the heat transfer coefficient function influences the amplitude oscillations and the average value of the wall superheat. In the case of curve (a) the amplitude of the wall superheat oscillation is 13 K, instead of 40 K for curve (d). In order

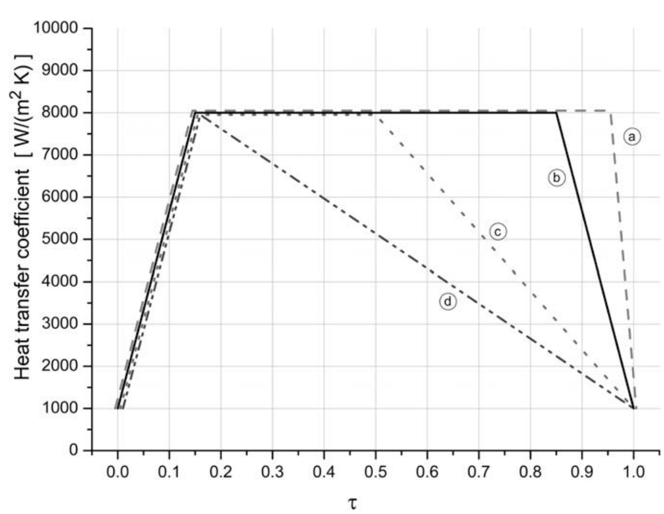

Fig. 8. Theoretical evolutions of the heat transfer coefficients in a PTPT evaporator which periodically dries out.

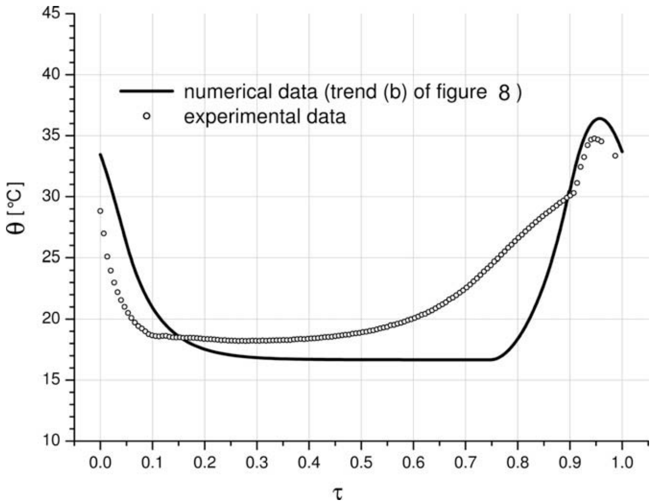

Fig. 9. Comparison of experimental data with numerical data evaluated by considering the theoretical trend curve (b).

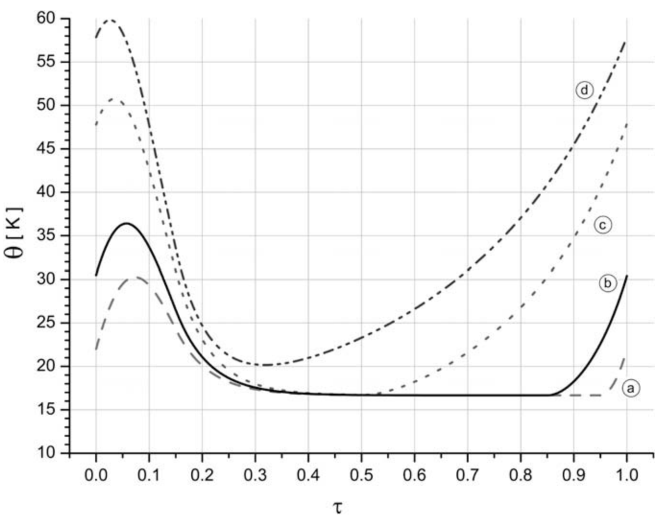

Fig. 10. Wall superheat evolutions in a single heat transfer period according to different heat transfer functions.

to avoid dangerous superheating in the evaporator wall and in the electronic component, the evolution in time of the heat transfer coefficient must, therefore, be known. The only knowledge of the maximum and minimum values of the heat transfer coefficient, in the case of a transient heat transfer regime, could lead to large wall superheat prediction errors (over 100%), but in the case of a miniature evaporator, these prediction errors could be even larger.

6. Conclusions

The trend shown by thermal control techniques is, at the moment, towards the use of miniature two-phase loop devices, both capillary and thermally driven. If the size of the single elements included in the loop device decreases, an efficient way of controlling the temperature is provided by devices operating in an unsteady-state regime, such as LHP with a pressure regulator, PHP and Periodic Two-Phase Thermosyphons.

This paper offers a theoretical and experimental analysis of the boiling heat transfer phenomena which occur in a miniature evaporator within a PTPT. The results of this experimental analysis have shown that a stable periodic heat transport regime can be achieved even when a FC-72 liquid volume of 3 ml circulates into the loop, with maximum heat flux supplied of 16.2 W/cm^2 and maximum evaporator wall temperatures lower than 100 °C. This heat flux is close to CHF for FC-72 dielectric fluid. It has been observed that the periodic, complex fluid dynamic motions inside the evaporator lead the thick evaporator walls to work in a continuous transient boiling regime.

This paper presents a technique for measuring the heat transfer coefficient over time for a thick flat surface which operates in a boiling regime by using an approximate solution of the heat transfer inverse problem. Comparison between the boiling heat transfer coefficients measured over time and those reported in the literature has shown that if the volume of the liquid pool in the evaporator, which periodically dries out, is large (more than 64 ml in this experiments) a stabilised nucleate boiling regime can be observed in a portion of the heat transport cycle. As the volume decreases, no nucleate boiling regime can be observed at most only unstable transitional or film boiling can be observed.

In conclusion in the case of evaporators with miniature pools of liquid, no nucleate heat transfer coefficient can be used to predict the boiling heat transfer, but lower coefficients must be used. In order to predict these coefficients an experimental inquiry into the boiling onset of thick flat samples must be carried out by taking into account liquid pools with finite thermal capacity in general and miniature pools in particular.

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