Water Pool boiling on Aluminum Metal Foams

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Abstract. With current growth in electronic systems, their physical sizes decrease, and the spacing between components decreases; both the total amount of heat generated and the power density increase significantly. There is a general agreement in the scientific community that current air-cooling technologies are asymptotically approaching their limits imposed by available cooling area, available air flow rate, fan power, and noise. Pool boiling is widely used in many different engineering systems: chemical and nuclear reactors, refrigerating and air conditioning equipment, and thermal management of electronic devices. Most of these applications have a common limitation: the maximum heat flux that can be rejected by the cooling systems under safe, reliable, and efficient operation. In this paper experimental data pertinent to deionized water pool boiling across 10 mm thick aluminum foams are presented. Three foam samples with different pore densities, 5, 10, and 40 PPI, yet with an identical mean porosity of 0.92 are tested. Compared to a heated flat plate, the foams offer higher heat transfer area albeit at induced bubble escaping resistance. The tradeoff between these two effects is investigated. Through the use of high speed video camera recording, bubble generation, trajectory and growth rate were analyzed and critically discussed.

1. Introduction

With current growth in electronic systems miniaturization the spacing between components decreases; therefore, both the total amount of heat generated (hence to be dissipated) and the power density (the heat generated per unit volume) increase significantly. There is a general agreement in the scientific community that current technologies based on single-phase heat transfer are asymptotically approaching their limits imposed by available cooling area, available air flow rate, fan power, and noise. Pool boiling is widely used in many different engineering systems: chemical and nuclear reactors, refrigerating and air conditioning equipment, and thermal management of electronic devices. Most of these applications have a common limitation: the maximum heat flux that can be rejected by the cooling systems under safe, reliable, and efficient operation. The characterization of pool boiling has been a topic of worldwide research since Nukiyama [1] conducted experiments to measure a ‘boiling curve’ of the heat rejected from a surface submerged in stagnant liquid as a function of its temperature. It is well-documented that surface treatments may provide effective boiling heat transfer enhancement, and many variants of treated surfaces have been developed. In particular, it was experimentally demonstrated that microparticle coatings have promising capabilities for the enhancement of nucleate boiling heat transfer coefficients and critical heat flux (CHF) [2-3]. Furthermore, recent works have led to new concepts for surface modification at the nanoscale. Nanostructured materials (e.g., nanowire coatings, nanoporous layers, carbon nanotube arrays, etc.) have been shown to enhance nucleate boiling and critical heat flux (CHF) [4-6]. A comprehensive review can be found in Ref. [7].
Porous layers with different pore sizes (mini-, micro-, nano-size) have been proposed for possible application in boiling. Accordingly, a great deal of information is available in the literature, as surveyed by Nield and Bejan [8], on improving the pool boiling heat transfer in terms of heat transfer coefficient and CHF by means of different kinds of porous layers.

Among the porous media that can be used to enhance pool boiling, high porosity metal foams represent a different and alternative possibility. In fact, metal foams have been demonstrated to be very effective in enhancing the heat transfer during both single-phase and two-phase heat transfer processes [9-12]. Metallic foams are a class of stochastic cellular structured materials with open cells, randomly oriented and almost homogenous in size and shape. A typical structure of an aluminum foam is shown in Fig 1. Nonetheless, in the open literature, only a few studies can be retrieved on pool boiling heat transfer enhancement with metallic foam structures. Recently, Yang et al. [13] investigated the water pool boiling heat transfer on plain surfaces covered by different copper foam layers. The pore density varied from 30 to 90 PPI, while the porosity was ranged between 0.88 to 0.95. The foam layer welded on the surface was from 1.0 to 5.0 mm thick. The authors found that the copper foams enhanced pool boiling heat transfer, with heat transfer coefficients two to three times higher than those of the plain surface. Furthermore, it was found that the optimal foam thickness decreased with an increase in the foam pore density.

This paper presents an experimental analysis of water pool boiling heat transfer in a vertical channel partly filled by aluminum foam covering a horizontal aluminum plate.

2. Experimental setup and foam samples

As shown in Figure 2, tests were run in an experimental setup designed and built to study water pool boiling on smooth or enhanced surfaces. The heater assembly consists of two different heater blocks: the first one, made of aluminum, is the test sample while the second one, made of copper, containing 9 electrical cartridge heaters (200 W/240 V) controlled by a motorized Variac. The aluminum blocks are designed to have the main size equal to the Rayleigh-Taylor wavelength, being 27.2 mm in the case of water. Each aluminum block is glued with high temperature epoxy resin in a Peek plate (λ=0.028 W m⁻¹ K⁻¹) and presents a smooth or enhanced (27.2 x 27.2 mm) top surface exposed to the working fluid (distilled water). Two different boiling chambers were designed to analyse the confined and unconfined pool boiling: both chambers consist of two glasses and two Peek 200 mm high walls, the sealing is accomplished by means of a set of silicon O-ring cords. One chamber has exactly the same size as the heating block to study the confined pool boiling.
The boiling chamber is closed by a top Peek plate and it is directly connected to the condenser which is located on the top and fed with tap water. On the top Peek plate, other four threaded holes are drilled to allow the direct access to the boiling chamber. One is used to insert a temperature probe equipped with a T-type thermocouple (uncertainty (k=2) ±0.05 K) to monitor the water temperature 20 mm away from the heated surface while another one is used to pressure taps connected to an absolute pressure transducer (uncertainty (k=2) ±0.065% f.s., f.s.=20 bar) to monitor the saturation pressure. The third one is used to convey the vapor to the top of the condenser, while the last one is used to charge the boiling chamber. As illustrated in Figure 3, four thermocouples at 1 mm, 6 mm, 11 mm, and 16 mm from the surface were inserted at the depth of 13.6 mm, every 5 mm. The imposed heat flux and wall superheat can be estimated from the evaluation of the temperature gradient measured by the thermocouples.

The data logging frequency was set at 1 Hz and the regressed data refer to time-averaged values over 100 s interval collected at each imposed heat flux once the system had reached the steady state conditions. The temperature along the sample can be approximated by the following linear regression of the measured temperature readings:

$$t_{wall} - t_{sat} = \frac{dt}{dx}$$
where \( t_{\text{wall}} \) and \( \frac{dt}{dx} \) are the y-intercept and the slope estimated by a simple linear regression of the recorded temperatures. Thus, the heat flux \( HF \) is given by:

\[
HF = \lambda \cdot \frac{dt}{dx}
\]  

(2)

where \( \lambda \) is the thermal conductivity of the aluminum sample, which is equal to 126 W m\(^{-1}\) K\(^{-1}\) being the base made of Al5182. The heat transfer coefficient \( h \) can be calculated by dividing the heat flux by the wall superheat, as:

\[
h = \frac{HF}{t_{\text{wall}} - t_{\text{sat}}}
\]  

(3)

Furthermore, four additional calibrated T-type thermocouples were located in the main heating copper block to verify the estimated heat flux. The uncertainties in surface temperature and heat flux are determined by evaluating the error in the linear regression. The surface temperature, heat flux, and heat transfer coefficient uncertainties are approximately ±0.06 °C, ±5.0 kW/m\(^2\), and ±2% over the range of operating conditions investigated. Finally, a high speed video camera is used to visualize the boiling phenomenon through the glass walls.

### Table 1 Main characteristics of the tested aluminum foams.

| Type   | PPI  | \( \varepsilon \) | \( \rho^* \) | \( a_{sv} \) |
|--------|------|-------------------|--------------|-------------|
| Al_5_7.9 | 5    | 0.921             | 7.9          | 339         |
| Al_10_7.3 | 10   | 0.927             | 7.3          | 731         |
| Al_40_8.6 | 40   | 0.914             | 8.6          | 1834        |

3. Experimental results

This section presents the experimental results collected during water pool boiling on three aluminum foam samples brazed on heaters. The tests were run at atmospheric pressure (i.e. at around 100 °C of saturation temperature). The system was heated up till the water reached the saturation condition and then the heat flux was increased from around 50 kW m\(^{-2}\) up to 900 kW m\(^{-2}\).

The experimental results reported in Figure 4 show that the foam samples performed better as compared to the reference smooth aluminum surface; the boiling curves for the aluminum foam samples presented remarkably lower wall superheat values at any given heat fluxes. Moreover, from the analysis of the plotted boiling curves, it appears that the 10 PPI and the 40 PPI exhibited the same behavior which is slightly different from that measured for the 5 PPI. In particular, the wall superheat of the latter tended to continuously increase and it is similar to that of the reference smooth surface. Comparing the boiling curves for 5 PPI and 10 PPI, one notes a similarity up to 150-200 kW m\(^{-2}\), and then the 5 PPI one exhibits better heat transfer performance as compared to the 10 PPI foam sample. This might be due to the presence of the thick layer of foam, which when the heat flux increases, reduces the liquid drainage to the hot surface and the vapor release when the heat flux increases.
Moreover, when decreasing the number of pores per inch, the surface tends to the smooth one (to the limit, when the PPI tends to 0, the surface tends to the plain one). It is also interesting to point out that the performance of the 10 PPI tends to stand closer to that of the smooth surface; in fact, at around 20 K of wall superheat, the heat flux is similar and then the smooth surface presents even a better heat transfer behavior. These observations are confirmed by the results reported in Figure 5, where the measured heat transfer coefficients are plotted against the wall superheat. As seen, the samples presented two different behaviors; in the case of 5 PPI the heat transfer coefficient continuously increased with the wall superheat while for 10 PPI and 40 PPI ones, it constantly increased up to around 10–12 K of wall superheat and then it flattened, meaning that the boiling heat transfer performance of these two samples would not improve with an increase in the wall superheat.

When comparing 5 PPI and 10 PPI foam samples, it appears that at up to 8–10 K of wall superheat (i.e. 150-200 kW m$^{-2}$), they performed similarly; then, the 5 PPI showed higher heat transfer coefficients as compared to the 10 PPI sample, because the latter tended to settle as the wall superheat increases.

This behavior can be further explained considering that at low heat flux, the bubbles generated at the surface and within the foam are relatively small and can easily escape from the foam. However, when increasing the heat flux, the bubbles grow, experiencing larger flow resistance through the porous layer when they are detached from the base plate; this resistance is larger in the case of 10 PPI foam as compared to that of the 5 PPI sample.

On the other hand, the performance of the 40 PPI sample is notably better compared to that of the 5 PPI one, especially at low-to-medium heat fluxes. This might be due to the high capillary pressure generated by the small pores, which allows for a continuous and efficient liquid drainage and vapor release at low-medium heat fluxes. But when increasing heat flux, the bubbles grow experiencing larger flow resistance through the porous layer when detached, similar to what happens with the 10 PPI sample, and the performance decreases. At 900 kW m$^{-2}$, the 40 PPI and the 5 PPI samples presented the same wall superheat, meaning that they are performing identically.

Figs. 6, and 7 compare some visualization pictures extracted from the linked high-speed videos recorded during water pool boiling on 5 PPI (left column) and 40 PPI (right column) aluminum foams. The comparisons are conducted at constant wall superheat. In particular, Figure 6 reports two pairs of images taken at around 5.6 K, (a) and (b), and at around 9.7 K, (c) and (d), while Figure 7 reports other two pairs of images taken at higher wall superheat: around 14.1 K and around 19 K, (e) and (f), and, (g) and (h), respectively.

In general, the boiling visualizations confirm the behaviors previously described for the 5 PPI and 40 PPI samples.
Starting from the first two frames taken at around 5.5 K, it clearly appears that the amount of vapor generated and released by the 40 PPI (b) is much higher as compared to that observable in the case of 5 PPI.

Figure 6 Frames (a, b, c, d) taken from the high speed video recorded during the water pool boiling on 5 PPI (left-column) and 40 PPI (right column) foam samples at constant wall superheat.

In this case, the 5 PPI rejects around 65 kW m$^{-2}$ while the 40 PPI corresponds to a 60% higher heat flux, 105 kW m$^{-2}$. Similarly, at higher wall superheat, around 9.7 K, the difference between the vapor generated by the 5 PPI and the 40 PPI is even more evident: the bubbles produced by the 40 PPI foam are bigger and start to coalesce forming a vapor column. Furthermore, at these operating conditions, the heat flux transferred by the 40 PPI, being 314 kW m$^{-2}$, is remarkably greater (+49%) than that measured for the 5 PPI sample, 211 kW m$^{-2}$. Examining Figure 7 (e) - (f), which correspond to a wall superheat of around 14.1 K,
the vapor coalescence to form an efficient vapor column can also be observed in the case of 5 PPI, while the amount of vapor generated by the 40 PPI heated samples is quickly saturating the porous layer, impairing the heat transfer. At these operating test conditions, the heat flux exchanged by the 40 PPI samples

\[
\Delta t_{wall} - t_{sat} = 14.2 \text{ K}, \ HF = 460 \text{ kW m}^{-2}
\]

(592 kW m\(^{-2}\)) remains higher than that measured for the 5 PPI sample but the difference (+28%) is more than halved as compared to the first wall superheat (+60%). When the wall superheat is increased even more, around 19 K, the two samples ((g) and (h)) resemble very similar; in the case of 5 PPI a large vapor column is generated but part of the foam structure is still visible while in the case of 40 PPI the vapor saturates the porous layer and both liquid and vapor face high flow resistance. This can explain the reason

\[
\Delta t_{wall} - t_{sat} = 19.2 \text{ K}, \ HF = 886 \text{ kW m}^{-2}
\]

\[
\Delta t_{wall} - t_{sat} = 18.9 \text{ K}, \ HF = 898 \text{ kW m}^{-2}
\]

**Figure 7** Frames (e, f, g, h) taken from the high speed video recorded during the water pool boiling on 5 PPI (left column) and 40 PPI (right column) at constant wall superheat.
behind the underperformance of the 40 PPI foam which, at these operating test conditions, tends to be similar to that measured for 5 PPI one (being around 890 kW m\(^{-2}\)).

4. Conclusions
This paper presents an experimental analysis of the water pool boiling heat transfer in a vertical channel partly filled by a metal foam. The experimental measurements were run by boiling the water at ambient conditions, by increasing the heat flux up to 900 kW m\(^{-2}\). Three 10 mm thick foam layers having 5 PPI, 10 PPI, and 40 PPI with a porosity around 0.92 were brazed on aluminum heaters and then tested. An additional reference smooth aluminum sample was tested. The results show that the foams always performed better as compared to the smooth surface and 40 PPI sample exhibited the best performance. Furthermore, the experimental measurements highlighted a different heat transfer behavior among the tested samples; in fact, the boiling performance of the 10 PPI and 40 PPI decreased as the wall superheat increased, while that measured for 5 PPI continuously increased. The associated boiling visualization permitted to explain the described different behaviors of the porous layer, which is mainly due to a different interaction between the vapor-liquid two-phase boiling regime and the foam substrate and size. Besides, from the water boiling visualization, a more homogeneous nucleation sites’ distribution can be observed when using metal foams instead of smooth surfaces, limiting the edge effects commonly occurring in small size evaporators. However, additional heat transfer measurement and flow visualization are needed to investigate the effects of the key parameters including foam thickness, pore density, and porosity.

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