Evaluation of a new designed microchannel heat sink for CPU cooling based on IR – thermography synchronized with high-speed flow visualization

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Abstract

The design and implementation of an experimental boiling loop is studied in this work. MEMS fabrication techniques are used to assemble the integrated multi-microchannel evaporative cooling device (iMMECo) in a single silicon substrate with optical access to the coolant flow. Tests on thermo-fluid-dynamic performance of the iMMECo are performed making use of synchronized high-speed visualization and high-speed IR thermography, confirming the existence of flow instability, backflow and non-uniform distribution of flow among the channels, all necessary to define the optimum range of operation. Flow boiling curves experimentally obtained suggest that the smaller the inlet passage, the lower the heat flux for boiling incipience to occur, resulting in a lower overshoot of wall temperature and lower overall surface temperature. On the other hand, pressure drop is highly increased for smaller inlet passages, increasing the pumping needs.

Introduction

After William Herschel discovery of “invisible rays” lower in energy than red light in 1800 [Herschel, 1800], and Macedonio Melloni’s building of the first thermopile IR detector in around 1830 [Schettino, 1989], the first steps were made for infrared thermography to become an important technique to collect the temperature of bodies through infrared radiation in a non-intrusive and instantaneous mode. Over the past few years, the application of infrared thermography at the micro and nanoscale for quantitative temperature visualization in internal and external flows has gained increasing attention, particularly due to the difficulty to use temperature sensors at these small scales, which might affect the flow.
Usually, surface temperature measurements are performed either using thermocouples located at some depth on top or below the wall–fluid interface of the microchannel (e.g., Peng and Peterson [1994]; Qu and Mudawar [2003]), or by some thin-film resistance temperature sensors deposited directly on the surface of a silicon heat sink (e.g., Koo et al. [2001]; Popescu et al. [2002]). Recently, a few studies using IR thermography at the microscale have been reported. Hetsroni et al. [2001, 2003a, 2003b] determined the outer surface temperature of circular minichannels and of triangular microchannels using IR radiometry. Methods to reduce noise in the experiments were tested by the authors, such as reduction of background radiation by using a controlled temperature background surface. Hapke et al. [2002] used IR thermography to examine the outside wall temperature distribution for flow boiling in rectangular microchannels. The authors concluded that it is possible to detect the streamwise position of flow boiling regions using this technique. Xu et al. [2005] used IR thermography to identify thermal oscillations of the outer surface of silicon microchannels. They identified three zones within a full boiling cycle. They are the liquid refilling stage, the bubble nucleation, growth and coalescence stage and the transient annular flow stage.

In 2011, Hetsroni and coworkers [Hetsroni et al., 2011] presented a critical analysis of the application of IR thermography to microscale systems. They identified several problems in applying the technique, e.g. the need for accurate characterization of the IR system performance and its calibration, the determination of the body surface emissivity, the design of the optical access window including the choice of the most appropriate IR material, etc and demonstrated the resolution for these issues. Recently, Szczukiewicz et al. [2012, 2013] introduced a new in situ pixel by pixel technique to calibrate the raw IR image signals with an accuracy of ±0.2 ºC, and thus converting them into accurate two-dimensional temperature fields of 10,000 pixels over the heated surface of a silicon microevaporator.

In the work presented here, IR thermography and high-speed visualization are simultaneously implemented to measure spatially resolved surface temperature variations along the microevaporator, as they provide accurate information of the local heat transfer variation along the channels and relate them with dominant two-phase flow patterns and characterize hydrodynamic instabilities in the multiple microchannels systems.

**Experimental methodology**

Despite the many studies conducted to investigate the performance of electronic cooling systems, only a few have been performed in silicon chips comprising embedded sensors and heaters. Metallic blocks containing cartridge heaters in contact with silicon dies for heating
purposes pose many difficulties on heating uniformity, also due to the need of a thermal interface (usually a thermal grease), which conducts far worse than metal. Furthermore, it may trap air bubbles which in turn introduce significant temperature and pressure oscillations. Integrated heaters making use of Joule effect are by far more suitable, as micromachining technology enables structuring microheaters with no moving parts, simplifying fabrication and operational design requirements at the same time allowing their precise definition and positioning. Consequently, the integrated multi-microchannel evaporative device is fabricated in a single silicon substrate with the heater and the thermal sensors on one side and the microfluidic passages on the opposite side. This raises additional challenges to guarantee precise alignment of the sensors and heaters relatively to the microchannel in order to obtain the desired uniform heat flux boundary condition and accuracy of the temperature measurements.

The fabrication, detailed in Silvério et al., 2015, followed an uncommon process in that the heater and the thermal sensors are of different materials which, despite at increased costs and complexity, allows using materials with adequate thermo-electric response for the entire operating ranges of heaters and sensors, respectively. A brief explanation of the fabrication process is given here to contextualize. The front side of the silicon chip comprises one 1000 nm-thick Aluminum heater and six 400 Å-thick Ruthenium thermal sensors microfabricated to achieve precise and interface-free heat flux imposition and temperature measurements, respectively. The heat sink on the backside is made of 27 parallel rectangular channels with hydraulic diameter $D_h = 96 \mu m$, length to diameter ratio $L/D_h = 104$ and relative roughness of 2.4%.

<table>
<thead>
<tr>
<th>Channel</th>
<th>Width(μm)</th>
<th>Depth(μm)</th>
<th>Length(μm)</th>
<th>Area ± u/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>C_313</td>
<td>313 (±6)</td>
<td>57 (±4)</td>
<td>9997 (±6)</td>
<td>$27 \times (1.73 ± 0.09) \times 10^4$</td>
</tr>
<tr>
<td>C_241</td>
<td>241 (±2)</td>
<td>55 (±3)</td>
<td>302 (±6)</td>
<td>$27 \times (1.70 ± 0.10) \times 10^4$</td>
</tr>
<tr>
<td>C_168</td>
<td>168 (±3)</td>
<td>54 (±6)</td>
<td>301 (±6)</td>
<td>$27 \times (8.16 ± 0.40) \times 10^4$</td>
</tr>
<tr>
<td>C_109</td>
<td>109 (±2)</td>
<td>51 (±5)</td>
<td>301 (±6)</td>
<td>$27 \times (5.42 ± 0.13) \times 10^4$</td>
</tr>
<tr>
<td>C_091</td>
<td>91 (±3)</td>
<td>47 (±6)</td>
<td>302 (±6)</td>
<td>$27 \times (4.15 ± 0.20) \times 10^4$</td>
</tr>
<tr>
<td>C_048</td>
<td>48 (±2)</td>
<td>45 (±4)</td>
<td>302 (±6)</td>
<td>$27 \times (2.10 ± 0.01) \times 10^4$</td>
</tr>
<tr>
<td>Plenum</td>
<td>9672 (±2)</td>
<td>57 (±3)</td>
<td>8851 (±6)</td>
<td>$(5.34 ± 0.21) \times 10^4$</td>
</tr>
</tbody>
</table>

Table 1. Average dimensions of microchannels, plenums and inlet constrictions etched on the silicon die

The channels are closed with glass to allow visualization of the flow, and sealed by anodic bonding, instead of using adhesives, thermal fusion or mechanical sealing, in order to
guarantee resistant sealing over the entire range of operating pressures and temperatures while still preserving the depth of the microchannels. Several multiple microchannel systems are designed with inlet constrictions with different dimensions (Table 1, Fig. 1) to infer on their influence on the stabilization of the flow. The effective area or the footprint area of the microchannels is 1 cm².

![Fig. 1](image)

**Fig. 1.** Microscope images of a) inlet plenum, constrictions and microchannels b) microchannels and outlet plenum and c) schematics of the position of thermal sensors relative to channels

Fig. 2 shows the experimental apparatus. The hydrofluoroether coolant (HFE-7000) is pumped (NE-1010 programmable high pressure syringe pump, New Era Pump Systems Inc., dispensing accuracy ± 1 %, 58.3 µL.h⁻¹ to 7635 mL.h⁻¹ with 60 cm³ luer-lock Norm-Ject® disposable syringes) at a given constant mass flux and constant wall heat flux (programmable DC power supply GEN150-5, from TDK-Lambda®, accuracy and linearity ± 5 % of rated \( V_{\text{out}} \), \( V_{\text{max}} = 150 \text{ V}, I_{\text{max}} = 5 \text{ A} \)). Synchronized high-speed visualization and thermal imaging are used to capture the resulting surface temperature and conclude about the influence of flow dynamics on the surface temperature evolution. The high-speed camera (Phantom V4.2 high-speed camera, Vision Research, Inc., 8 bit CMOS sensor, 4800 ISO monochrome, frame rate 2200 Hz at full resolution 512 · 512 to 90 kHz at 32 · 32 resolution, Gigabit Ethernet control, Phantom 640 camera control software) coupling microscope optics is used to visually capture the dynamical process of boiling. With this apparatus is possible to determine dominant two-phase flow patterns and characterize hydrodynamic instabilities in the multiple microchannels systems. A high-speed Onca-MWIR-InSb-320 infrared imaging camera (Xenics, frame rate 460 Hz at full resolution 320 · 256, pixel dimension 30 · 30 µm, thermal sensitivity 17 mK, spectral sensitivity 3.5 to 5 µm, Xeneth64 user interface) records the thermal evolution of the chip surface. The infrared energy emitted by an object differs
according to the composition of the object’s surface and the physical state of the object. Consequently, the chip outer surface is uniformly coated with thermal black matte ink (904 Nero Alte, Solcolor) with emissivity 0.94 to maximize the chip surface signal.

![Diagram of experimental setup](image)

**Fig. 2.** a) Schematics of the Experimental Setup. The test section represented here is the multiple silicon channel device, iMMECo; b) and c) pictures of the experimental apparatus; d) detail of the experimental apparatus e) detail of the flow circuit
Both the flow circuit and the fluid may retain a substantial quantity of air entrapped. This can lead the flow inside the microchannels to deviate from classical theory as is likely that hydrophobicity of the liquid-solid interface may be induced by air pockets entrapped by surface tension within surface grooves [Silvério and Moreira, 2008]. To prevent such phenomena, both the reservoirs and the flow loop are degassed using a mechanical Charles Austen Capex vacuum pump. Additionally, the working fluid is heated to boiling in a custom made degassing glass reservoir, while the vacuum pump exhausts gas non-condensables, which may otherwise induce premature bubble nucleation and affect heat transfer performance [Steinke and Kandlikar, 2004; Chen and Garimella, 2005].

The temperature of the flow circuit is monitored by k-type thermocouples (80 µm tip, Omega® Engineering). Pressure transmitters Eco-1 from Wika®, calibrated from 0 to 4 bar, presenting best fit straight line (BSFL) accuracy lower than 0.5 % of span and non-linearity of 0.3 % of span, with a response time lower than 5 ms are used to measure the pressure drop in the microevaporator.

The experimental setup is fully automated through a PC for simultaneous measurements of pressure and temperature, high-speed visualization and IR thermal imaging. Matlab® and NI Labview SignalExpress softwares are used to monitor and acquire temperature and pressure, while images are collected through dedicated graphical user interfaces. Post-processing is accomplished via Matlab® routines.

Results and discussion

One of the most important issues in electronics cooling is the need to guarantee working temperatures within functioning limits as well as to evaluate the electronic parts capability to withstand the cumulative hours of heating during their lifetime.

<table>
<thead>
<tr>
<th>Class</th>
<th>Type</th>
<th>Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Static</td>
<td>Leducng instability</td>
<td>Flow under sudden, large amplitude excursion to a new stable operation condition</td>
</tr>
<tr>
<td></td>
<td>Boiling crisis</td>
<td>Wall temperature excursion and flow oscillation</td>
</tr>
<tr>
<td></td>
<td>Flow pattern transition instability</td>
<td>Cyclic flow pattern transition and flow rate variation</td>
</tr>
<tr>
<td></td>
<td>Bumping, geysering or chugging</td>
<td>Periodic process of superheat and violent evaporation with possible expulsion and refilling</td>
</tr>
<tr>
<td>Dynamic</td>
<td>Acoustic oscillations</td>
<td>High frequencies related to time required for pressure wave propagation in the system</td>
</tr>
<tr>
<td></td>
<td>Density wave oscillation</td>
<td>Low frequencies related to transit time of a continuity wave</td>
</tr>
<tr>
<td></td>
<td>Thermal oscillations</td>
<td>Occur in film boiling</td>
</tr>
<tr>
<td></td>
<td>Coupling instabilities</td>
<td>Strong only for a small time constant and under low pressures</td>
</tr>
<tr>
<td></td>
<td>Pressure drop oscillations</td>
<td>Very low frequency periodic process</td>
</tr>
</tbody>
</table>

Table 2. Classification of flow instabilities (based on Boure et al. [1973])
The thermo-fluid-dynamic performance of the microevaporators are assessed with HFE-7000 at four different mass fluxes \( (G_{in} = 500, 1000, 1500 \text{ and } 2000 \text{ kg.m}^{-2}\text{s}^{-1}) \) and constant wall heat fluxes \( q'' \) up to 31 W.cm\(^{-2}\). Single-phase or two-phase flows are obtained depending on the mass flux and heat flux conditions. Additionally, flow instabilities (Table 2) have been experimentally observed in evaporative flows.

![Flow visualization](image1)

**Fig. 3.** Flow visualization (left) and surface temperature (right) for HFE-7000 flow in a) \( C_{168}, G_{in} = 2006 \text{ kg.m}^{-2}\text{s}^{-1}, q''_{\text{input}} = 23.7 \text{ W.cm}^{-2}, T_{\text{surf,avg}} = 59.1 \pm 7.4 ^\circ\text{C}, \Delta p_{\text{total}} = 82.58 \text{ kPa}; \) b) \( C_{241}, G_{in} = 2019 \text{ kg.m}^{-2}\text{s}^{-1}, q''_{\text{input}} = 24.2 \text{ W.cm}^{-2}, T_{\text{surf,avg}} = 57.3 \pm 5.8 ^\circ\text{C}, \Delta p_{\text{total}} = 62.42 \text{ kPa}; \) c) streamwise temperature distribution along the centerline of the central channel. The vertical red lines represent the position of the channel inlet (left) and outlet (right) and d) transverse temperature distribution at \( y\)-position = 0.77 cm.
Fig. 3 shows the comparison of simultaneous high-speed visualization and surface IR thermography results of two different microevaporators (C_168 and _241) at similar experimental conditions ($G_{in} \sim 2000 \text{ kg.m}^{-2}\text{s}^{-1}, q'' \sim 24 \text{ W.cm}^{-2}$). The single-phase flow regime is observed in the inlet constrictions for both cases. However, differences can be perceived at the channels outlet. For the first case, C_168 (Fig. 3a), evaporation occurs in ten channels, on the bottom part of the figure. Due to the strong evaporation occurring, the liquid film close to the walls is pushed out of the channels in such a vigorous way that the two-phase flow leaves the channels in a jet like configuration. At this point the confined flow expands and the jets are projected at an angle relative to the channel side wall. The interaction between outlet jets disrupts the jet development, and mixing of vapor and liquid occurs. For the second case, C_241 (Fig. 3b), the behavior described above is observed in all the 27 channels, and the surface temperature distribution tends to be lower and transversally more homogeneous than in the previous case (see Fig. 3c,d where the streamwise and transverse temperature distributions are presented for the central channel and $y$-position = 0.77 cm respectively). Moreover, no back flow is observed in either case, and as such, the flow is considered stable.

Fig. 4 results from the analysis implemented to experimental results obtained such as those presented in the previous figure. Fig. 4a shows the surface temperature of C_313 device resulting from imposing different heat fluxes to the channels outer walls at a mass flux $G_{in} = 500 \text{ kg.m}^{-2}\text{s}^{-1}$. Color bars represent the surface temperature range observed in the area of interest of the chip surface for each set of conditions. Spatial and temporal standard deviations to the spatial-temporal average are also presented in the error bars for further insight.

From the analysis it is clear that the surface temperature range in each experiment is always larger than the averaged surface temperature plus standard deviation, either spatial or temporal. Moreover, for the experiments presented the spatial standard deviation is always larger than the temporal standard deviation meaning within the area of interest the temperature variation is always larger that the temporal variation of each location in time. The surface temperature is seen to increase with increasing heat flux up to the point where evaporation starts ($q'' = 12.24 \text{ W.cm}^{-2}$). At this point the evaporation is not homogeneously present in all channels, which is well depicted by the large spatial standard deviation obtained. Further increasing the heat flux, leads to homogeneous evaporation in the whole stack (lower spatial and temporal standard deviations) and the average surface temperature decreases as well as the range of surface temperature observed in the experiment. The increase in heat flux ultimately leads to the surface temperature exceeding the intended upper limit of 80 °C due to rapid expansion of vapor bubbles and complete dryout inside the
channels (see Silvério and Moreira, 2012). This will most of the times impede further usage of the device as the pronounced temporal and spatial temperature fluctuations ($T_{\text{max}} > 100 \, ^\circ\!\!\!\!\_C$ and $\Delta T > 40 \, ^\circ\!\!\!\!\_C$) lead to chip fracture.

![Graph showing temperature evolution](image1)

![Graph showing comparison of inlet contraction effect](image2)

![Graph showing effect of inlet contraction on surface temperature and pressure drop](image3)

![Graph showing dependence of surface temperature on refrigerant mass flux](image4)

**Fig. 4.** Experimental results for evaporative cooling: a) evolution of C_313 iMMECo surface temperature with imposed heat flux; b) comparison of the effect of inlet contraction on iMMECo surface temperature for $G_\text{in} = 500 \, \text{kg.m}^{-2}\text{s}^{-1}$; c) effect of inlet contraction on iMMECo surface temperature and resulting pressure drop and d) dependence of iMMECo or surface temperature on refrigerant mass flux and resulting pressure drop. Black error bars symbolize spatial standard deviation while grey error bars represent temporal standard deviation of average surface temperature. The colored columns indicate the experimental surface temperature interval obtained for each set of experimental conditions. Averaged pressure drop is plotted in red together with standard deviation bars.

Fig. 4b shows the influence of the inlet constriction on the surface temperature. The smaller the inlet constriction passage ($C_{109} < C_{313}$, Table 1) the smaller the surface temperature
under the same conditions of heat and mass flux. Furthermore, boiling inception occurs at lower heat fluxes for the chip with the smaller inlet constriction ($C_{109}$) resulting in a lower overshoot of wall temperature. The Venturi effect acting on the liquid flow through the constriction causes a decrease of the liquid local pressure consequently diminishing the saturation temperature of the liquid and enabling early evaporation. This behavior is further emphasized in Fig. 4c where the influence of inlet constriction passage on surface temperature is depicted at higher mass flux ($G_{in} = 2000 \text{ kg.m}^{-2}\text{s}^{-1}$) and heat flux ($q'' = 25.4 \text{ W.cm}^{-2}$). The figure also illustrates the associated pressure drop, which decreases steeply from $C_{109}$ to $C_{313}$. Finally, Fig. 4d shows a decreasing tendency of surface temperature and increasing tendency of pressure drop with increasing mass flux for $C_{313}$ under 27.6 W.cm$^{-2}$. From both Fig. 4c and Fig. 4d it is clear not only is important to take into account the limiting value of surface temperature but also the pressure drop when deciding the best settings to tackle heat dissipation in a real chip device.

**Summary**

The paper addresses research and development of new chip cooling technologies to tackle the challenge and achieve powerful cooling with less energy, thereby paving the way for cooling next-generation chips. The design and optimization of microchannel heat sinks are dependent of macroscale empirical laws, however several studies that have addressed the validity of macroscale theories to describe momentum and heat transfer both in single phase and phase change in microtubes have results often inconsistent. Therefore, local measurements of experimental parameters such as wall temperature and flow pressure combined with visualization techniques can support the correct assessment of heat transfer.

Flow visualization has been conducted with and without a constriction insert at the inlet of microchannels. Visualization confirms flow patterns in the microchannels and their evolution with increasing heat flux. It also confirms the existence of flow instability, back flow and non-uniform distribution of flow among the channels, more pronounced for larger inlet passages. The flow boiling curves suggest that when constrictions are included, the boiling incipience occurs at a lower heat flux resulting in a lower overshoot of wall temperature. Nonetheless this decrease in surface temperature comes at the expense of increasing pressure drop.

As a final point, improvement of state-of-the-art experimental capabilities is therefore a requisite to further progress current knowledge of phase change heat transfer in channels.
which can contribute to enhance the use of microchannel heat sinks as a viable cooling technique for high power density devices.

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References


