



Exploring capillary limits of copper wire mesh manifold for area scaling of capillary-driven two-phase coolers

Heungdong Kwon^{a,b}, Roman Giglio^c, Daeyoung Kong^{a,d}, Muhammad Shattique^{e,f,g},
Hyoungsoon Lee^h, James W. Palko^c, Ercan M. Dedeⁱ, Mehdi Asheghi^a, Kenneth E. Goodson^{a,*}

^a Department of Mechanical Engineering, Stanford University, Stanford, CA 94305, United States

^b School of Mechanical Engineering, Kyung Hee University, Yongin 17104, Republic of Korea

^c Department of Mechanical and Aerospace Engineering, University of California, Merced, CA 95343, United States

^d Advanced Materials and Components Research Division, Electronics and Telecommunications Research Institute, Daejeon 34129, Republic of Korea

^e Graduate Program in Materials and Biomaterials Science and Engineering, University of California, Merced, CA 95343, United States

^f Department of Aeronautical Engineering, Military Institute of Science and Technology, Mirpur Cantonment, Dhaka 1216, Bangladesh

^g Department of Biomedical Engineering, Military Institute of Science and Technology, Mirpur Cantonment, Dhaka 1216, Bangladesh

^h School of Mechanical Engineering, Chung-Ang University, Seoul 06974, Republic of Korea

ⁱ Electronics Research Department, Toyota Research Institute of North America, MI 48105, United States

ARTICLE INFO

Keywords:

Two-phase cooling
Capillary-driven boiling
Copper inverse opal wick
Wire-mesh manifold
Area-scalable thermal management
Data center cooling

ABSTRACT

Data-center cooling represents 30–40% of the total data-center facility energy consumption, and the rapid growth of high-power processors is exceeding the practical limits of pumped single-phase liquid cooling. Capillary-driven two-phase cooling offers a path to chip-scale thermal management with orders-of-magnitude lower pumping power, but its area-scaling to practical chip areas remains constrained by the liquid transport limit of wick. In this study, a silicon pin array coated with a copper inverse opal (CIO) wick and copper wire-mesh (CWM) 3D manifold are combined to construct capillary-driven two-phase micro-coolers by decoupling liquid feeding and vapor extraction. We systematically investigated the limits of area scaling using three combinations of heated area and 3D manifold footprints of (case A: $5 \times 5 \text{ mm}^2$: $5 \times 5 \text{ mm}^2$), (case B: $10 \times 10 \text{ mm}^2$: $10 \times 10 \text{ mm}^2$), and (Case C: $5 \times 5 \text{ mm}^2$: $10 \times 10 \text{ mm}^2$), respectively. For all cases, A, B, and C, we achieved critical heat fluxes (CHF) of ≈ 220 and $\approx 440 \text{ Wcm}^{-2}$ at heated-area mass fluxes of 5 and 10 g/min/cm^2 , respectively, with full mass flow utilization for phase change. When excess fluid is supplied to the CWM 3D manifold, for cases A and C, slightly higher CHF level of $\approx 550 \text{ Wcm}^{-2}$ is achieved. At this heat flux and mass flow rate, high-speed camera images reveal the formation of a liquid-vapor phase front, indicating excess liquid that is not fully vaporized. However, for case B ($10 \times 10 \text{ mm}^2$: $10 \times 10 \text{ mm}^2$), the CHF value is pinned at $\approx 440 \text{ Wcm}^{-2}$, which indicates that CWM 3D manifold operates near its capillary limit, as the viscous pressure drop is highly non-linear for higher mass flow rates along CWM for the larger footprint CWM 3D manifold. At the most efficient operating point, the coolers achieved two-phase heat transfer coefficients, $HTC_{2-\phi}$ ranging from 0.4 to $0.5 \text{ MWm}^{-2}\text{C}^{-1}$ for cases A and B. The findings in this study provide crucial geometric design criteria for optimizing the hydraulic performance of two-phase micro-coolers utilizing capillary-based 3D manifold and high-performance porous wick.

1. Introduction

The demand for data processing continues to grow exponentially, driven by advancements in artificial intelligence, machine learning, and data-intensive software applications [1,2]. This significant surge in

computational load requires innovative cooling approaches to address the associated thermal management challenges [3]. Furthermore, as the computational power increases, the proportion of total data-center facility energy consumption dedicated to cooling has increased to 30–40% [4]. Efficient thermal management solutions are essential not only to

* Corresponding author at: Department of Mechanical Engineering, Stanford University, Stanford, CA 94305, United States.

E-mail addresses: hdkwon@stanford.edu (H. Kwon), rubaietshattique@ae.mist.ac.bd, mshattique@ucmerced.edu (M. Shattique), goodson@stanford.edu (K.E. Goodson).

<https://doi.org/10.1016/j.icheatmasstransfer.2026.112500>

maintain the performance and reliability of data centers and high-performance computing systems, but also to mitigate their environmental impacts and to contribute to global energy saving [5]. The traditional single-phase liquid flow cooler offers excellent cooling performance, while pumping high fluid flow rates through small passages, such as microchannels, results in substantial pressure drops and potential mechanical issues [6]. Additionally, the refrigeration infrastructure to supply chilled water increases the energy requirement and cooling costs. Two-phase cooling techniques are emerging as promising solutions due to the high latent heat of phase change. However, two-phase coolers may exhibit increased superheat above saturation temperature and are susceptible to flow instabilities and large temperature fluctuations [7].

Early application of capillary-based two-phase cooling consists of an integrated hierarchical wick that serves both as boiling and liquid delivery platforms for vapor chamber applications [8,9]. Nam et al. [8] reported that copper micropillars enhanced with nanostructures achieved a critical heat flux (CHF) of 160 Wcm^{-2} with a corresponding superheat of $\approx 30^\circ\text{C}$ over $5 \times 5 \text{ mm}^2$ heated area. Coso et al. [9] demonstrated that a square silicon pin-fin array integrated with liquid microchannels in the wick achieved capillary-driven boiling performance of a CHF of 270 Wcm^{-2} over 1 cm^2 heated area. When the heated footprint was reduced to $2.5 \times 2.5 \text{ mm}^2$, the CHF increased above 700 Wcm^{-2} with the exceptionally low superheat level $\approx 6^\circ\text{C}$, which was attributed to effective liquid transport for phase change. In both of these studies [8,9], the vapor extraction pathways are not well-defined and often interfere and disrupt the liquid delivery to the hot spots, which explains relatively low CHFs. Sudhakar et al. [10] addressed this challenge by introducing a capillary-fed two-layer evaporator ($10 \times 10 \text{ mm}^2$ footprint) with wicking posts and vapor vents to separate liquid and vapor flow paths in a vapor chamber. This configuration achieves a large CHF of 485 Wcm^{-2} over 1 cm^2 heated area but exhibits a comparatively large superheat due to the thick wick layer of the sintered copper wick. We later demonstrate in this paper that by implementing CIO wick and 3D copper wire-mesh can achieve full vapor-liquid phase separation resulting in ten times smaller two-phase thermal resistance from 0.1 to $0.01 \text{ MWm}^{-2}\text{C}^{-1}$ for 1 cm^2 heated chip.

Previous fundamental studies by Palko and Chi et al. [11–13] on very narrow ($250\text{--}1000 \mu\text{m}$) planar CIO wicking bridges demonstrated the upper limit of CHF by delivering liquid (in-plane) to both sides of the heated bridge allowing for vertical (out-of-plane) vapor extraction. However, this is of very little use for cooling large area of large chip size ($800\text{--}900 \text{ mm}^2$) [14], and high power GPUs [15]. To address the area scaling challenge, we have introduced several generations of a capillary-driven boiling micro-cooler that consists of two separate components, namely a thermally-conductive microporous wicking structure where the two-phase boiling occurs and a capillary-based 3D manifold for liquid delivery and vapor extraction over a large, heated area [16–19]. These innovative designs achieved large critical heat flux (CHF) and small superheat, leading to high two-phase heat transfer coefficients, superior flow stability, and temperature uniformity, compared to the conventional microchannel single- and two-phase micro-coolers. This superior performance is due to several reasons that include but not limited to (a) the presence of a large number of nucleation sites within thermally-conductive porous structures, (b) precise self-regulating liquid delivery to the hot spots, and (c) providing unimpeded pathways for vapor extraction [8–13,16–22]. Additionally, the co-authors of this paper have progressively advanced capillary-based 3D manifold architectures, beginning with silicon microchannels [17–19] and continuing with copper wire mesh platforms [16,23,24] to optimize for both performance (CHF, superheat, and area scalability) and manufacturing cost. Previously, we reported [17–19] several silicon-based capillary-driven (open and closed) microchannel array 3D manifold using $\approx 100 \mu\text{m}$ wide liquid delivery channels that enables smaller capillary wicking length to achieve large CHF levels. While the 3D manifold was designed to operate in a capillary-driven mode, we also

utilized a syringe pump to deliver precise flow rates to the micro-cooler, allowing us to test both (a) hybrid single-phase/capillary-driven two-phase and (b) fully capillary-driven two-phase operating regimes. The silicon 3D manifold has a smaller form factor and height but is more expensive to fabricate, although the thermal performance is nearly similar to that of copper wire-mesh 3D manifold considered in this study. However, this may not hold, as we attempt to cool large area chips exceeding 1 cm^2 heated footprint.

Giglio et al. [16] reported the first implementation of copper wire-mesh 3D manifold immersed in a pool of saturated water and on top of a AlN pin fin wick with $5 \times 5 \text{ mm}^2$ heated area. At low heat flux levels, the flow resembles pool boiling. However, as the heat flux ramps up, the vapor columns push away the liquid pool, allowing for copper wire-mesh to operate, at least partially, in a capillary-driven liquid transport regime. Nevertheless, we expect a relatively thick liquid film to be present over the AlN wicking structure as such the micro-cooler is more likely to operate in a “shallow pool” and/or “capillary-driven” boiling regime(s), which degrade the thermal performance. Subsequently, we introduced a new copper wire-mesh 3D manifold and porous wick assembly that ensure a full “capillary-driven” boiling for a $5 \times 5 \text{ mm}^2$ heated area [24,25]. In this arrangement, we used a syringe pump to deliver liquid at different flow rates to both sides of the copper wire-mesh 3D manifold. The copper wire-mesh manifold draws liquid from both sides to the heated area using capillary forces allowing the excess fluid to exit from the 3D manifold.

In contrast to our earlier work [23], which considered the fixed $5 \times 5 \text{ mm}^2$ heater and manifold footprints, the present study adds a conformal $30 \mu\text{m}$ CIO monolayer on the silicon pin-fin array, implementing a bi-porous wick, and independently varies the heater and manifold footprints, so that the device-level transition between wick-limited and manifold-limited operation is experimentally validated. The focus of the present work is to investigate the area scalability of the copper wire-mesh from $5 \times 5 \text{ mm}^2$ to $10 \times 10 \text{ mm}^2$ chip area, as well as to understand the capillary limits of copper wire-mesh by conducting three sets of experiments. The first experiment is to establish a baseline where the 3D manifold footprint is the same as the heated area ($5 \times 5 \text{ mm}^2$) augmented with silicon pin-fin array coated with CIO layer. The second experiment demonstrates the feasibility of area scaling of the heated area and 3D manifold footprint from $5 \times 5 \text{ mm}^2$ to $10 \times 10 \text{ mm}^2$. Finally, in the third experiment, we tested a $10 \times 10 \text{ mm}^2$ 3D manifold on a $5 \times 5 \text{ mm}^2$ heated area. The combination of these three experiments provided a great deal of insight into the physics of operation and area scalability of micro-coolers with copper wire-mesh 3D manifolds. We also demonstrate these advancements yield a self-regulated capillary-driven two-phase cooling architecture with excellent thermal performance, facilitated by enhanced liquid supply to the heated area during phase change.

2. Experimental setup

2.1. Cooler description

Fig. 1 illustrates the evolution of the current micro-cooler concept from a conventional capillary-driven two-phase evaporator. The conventional capillary-driven evaporator (Fig. 1a), which typically supplies the liquid from the sides of the heated wick from the liquid pool, enhances performance by optimizing the wick structure to simultaneously achieve both high capillary pressure and permeability. However, these two requirements are fundamentally contradictory, because decreasing characteristic pore size increases capillary pressure but significantly reduces permeability, whereas increasing pore size improves permeability but decreases capillary pressure. Additionally, it is more challenging to design separate liquid delivery and vapor extraction pathways to achieve complete liquid-vapor phase separation. To overcome this inherent trade-off, this work introduces a 3D manifold architecture that establishes an additional liquid-supply pathway while

Fig. 1. Description of the capillary-driven two-phase cooler and its operating principle. (a) Comparison between a conventional capillary-driven two-phase cooler (left) and the cooler design in this study (right). (b) Schematic representation of the heat and mass transfer mechanisms in the capillary-driven two-phase cooler investigated in this study. The cooler features a copper wire-mesh manifold placed directly atop a copper inverse opal (CIO) wick. Capillary action within the wire-mesh transports the liquid, which is transferred to the top surface of the CIO wick via contact. A heat flux applied to the bottom of the wick induces boiling within the porous structure. The generated vapor escapes through the gaps between the wire-meshes, ensuring effective phase separation. The inset provides a microscopic view of the boiling process, illustrating how liquid supplied by the wire-mesh continuously replenishes the wick to sustain nucleate boiling.

integrating interspersed vapor extraction passages (Fig. 1b). In this study, the manifold employs a coarse copper wire-mesh array (capillary porous structure) to deliver liquid to the wick, thereby enabling capillary-driven two-phase boiling with high-heat-flux. The advantages of this architecture become even more pronounced as the heated area is scaled up. The copper wire-mesh manifold consists of periodically spaced copper wire-meshes, which serve as the separate liquid transport channels across the cooling area. This manifold is similar to the wire-mesh manifold developed in Ref. [16]. The manifold is overlayed or simply placed on the CIO wick where boiling takes place. It should be noted that no permanent solid-solid bonding is required between the wick and the copper wire-mesh 3D manifold. Liquid is supplied from the

manifold to the CIO wick via a liquid meniscus bridge formed at the interface between the polished wire-mesh bottom and the conformally-grown CIO wick surface (Fig. 1b). This unbonded configuration allows for various combinations of wicking structures and manifold materials, thereby eliminating concerns regarding thermomechanical mismatch [23]. Liquid supplied to the manifold is drawn into the wick by capillary action and subsequently vaporized by the heat flux applied to the opposite side of the CIO wick. The resulting vapor escapes through the gaps between neighboring wire-meshes, achieving near-complete liquid-vapor phase separation.

Fig. 2. (a) Description of the copper wire-mesh manifold fabrication. (b) Schematic illustration of the fabricated wire-mesh manifold. (c) Optical image of the copper wire-mesh manifold for the $5 \times 5 \text{ mm}^2$ heated area. (d) Optical image of the wire mesh manifold for the $10 \times 10 \text{ mm}^2$ heated area. (e) Microscopy images of the copper wire-mesh 3D manifold revealing the boundary between the plenum and the active area.

2.2. Wire-mesh manifold and copper inverse opal wick

Fig. 2a illustrates an overview of the copper wire-mesh manifold fabrication. A solid copper spacer ($8 \times 8 \text{ mm}^2$) with tube insertion holes and #100 copper wire-meshes are cut using UV laser ablation for subsequent assembly. The capillary transport limit and permeability of the same single-layer wire-mesh were systematically characterized in Ref. [26], which reported the permeability-thickness product $K \cdot \xi$ as a continuous function of the applied capillary pressure, together with a maximum available capillary pressure of $\approx 1.03 \text{ kPa}$ after surface cleaning and $\approx 0.37 \text{ kPa}$ in the as-received condition. These directly measured properties are the only transport input to our liquid-transport model, which will be discussed later. The prepared copper meshes are then stacked to achieve mesh center-to-center spacing of $650 \mu\text{m}$. Finally, the assemblies are pressed and sintered to form the integrated manifold in reducing atmosphere using activated carbon at 800°C for 1 h. The wire-mesh count (#100, wire diameter $114 \mu\text{m}$, opening $152 \mu\text{m}$) and the mesh-to-mesh stacking pitch ($650 \mu\text{m}$) were selected based on the permeability-capillary pressure characterization of free-standing single-layer copper meshes which identifies #100 plain-weave copper as offering an attractive compromise between capillary pressure and thickness permeability ($K \cdot \xi$). The $650 \mu\text{m}$ pitch was chosen based on the reported capillary-transport length scale of CIO layer with $\sim 30 \mu\text{m}$ thickness and $5 \mu\text{m}$ pore size [12]. The manifold features two holes at either end to accommodate 6-mm-wide perforated PVC tubes (Fig. 2b). These tubes effectively wet the wire-mesh manifold supplying liquid to the wick via capillary-driven flow. It should be noted that the wire-mesh manifold is an assembly of stacked porous structures (copper wire-meshes). Because the liquid paths in the manifold are fully open to the environment, fluid transport within the manifold is driven by capillary force rather than the pressurized flow typical of pumped single-phase cooling systems [27,28]. Finally, the bottom of the wire-mesh manifold, which contacts the top surface of the wick, is lapped with abrasive paper on a flat granite surface plate. Planarity is confirmed visually before each experiment. Lapping is continued until the brown oxide has been removed uniformly over the entire face, leaving a uniformly bright finish that indicates the whole area has been brought into a single plane.

Fig. 2c presents top and cross-sectional optical images of the $5 \times 5 \text{ mm}^2$ wire-mesh manifold. Liquid supplied through tubes at both ends is distributed throughout the stacked wire-mesh and spacer assembly, facilitating capillary-driven flow along the meshes. Fig. 2d shows the

similar $10 \times 10 \text{ mm}^2$ manifold, which retains the same $650 \mu\text{m}$ mesh period as the $5 \times 5 \text{ mm}^2$ version. As shown in Fig. 2e, the microscopy image of the copper wire-mesh 3D manifold reveals the boundary interface between the plenum and the active area. The tube with hole arrays wets the plenum region, allowing liquid to be drawn by the wire-mesh to supply the CIO wick over the active area during phase change. The copper wire-mesh stack is held against the CIO wick by a light mechanical clamp rather than a permanent metallurgical bond. This unbonded configuration was deliberate, as it preserves a continuous capillary path across the manifold/wick interface, which a thin solder layer would block. It also permits independent replacement and characterization of either layer and is mechanically tolerant of thermal-expansion mismatch between delicate CIO layer and wire-meshes at the soldering temperatures relevant to package integration.

As shown in Fig. 3a and b, the silicon chip incorporates the CIO wick at the top side and a micro-fabricated mesh-type metal heater at the bottom side. This heater applies a uniform heat flux to the wick at the opposite side. The footprint areas correspond to the $5 \times 5 \text{ mm}^2$ and $10 \times 10 \text{ mm}^2$ dimensions shown in Fig. 3a and b, respectively. The cooler operates by positioning the manifold (Fig. 2) directly atop the wick, thereby enabling efficient liquid distribution during two-phase cooling. The heater features two or four electrical pads (for the $5 \times 5 \text{ mm}^2$ and the $10 \times 10 \text{ mm}^2$ heaters, respectively) to supply electrical current and accurately measure the voltage drop across the heated area. Fig. 3c illustrates the CIO wick structure with annotated dimensions and presents the three SEM images acquired at different magnifications. SEM images indicate the conformal CIO layer electroplated on the dense silicon square pin-fin array, with a thickness of $\approx 30 \mu\text{m}$, an average pore diameter of $\approx 5 \mu\text{m}$, and a pore-to-neck diameter ratio of ≈ 0.4 . The methods used to fabricate the CIO structure are described in detail in prior publications [17,29]. The CIO geometry was adopted based on the parameter sweeps reported by Zhang et al. [12]. This specific geometry provides a capillary transport length scale of $\approx 300 \mu\text{m}$, which served as the design basis for setting the wire mesh spacing in the manifold to $600 \mu\text{m}$.

2.3. Experimental procedure

The inlet liquid is supplied by a syringe pump that sets a prescribed volumetric flow rate at the manifold plenum entrance. We emphasize that the syringe pump is used here strictly as a means of imposing a

Fig. 3. (a) and (b) Top and bottom views of the silicon chip, showing the CIO wick on the top side and the micro-patterned heater on the bottom side. In this study, the wick and heater have matched dimensions of either (a) $5 \times 5 \text{ mm}^2$ or (b) $10 \times 10 \text{ mm}^2$. Each heater incorporates four electrical probes to accurately determine the imposed heat flux. (c) Schematic of cross-section view of the CIO layer formed on the dense square pin-fin array and scanning electron microscopy (SEM) images of the CIO layer at three different magnifications.

known, controllable condition for the inlet mass flow. The capillary-driven micro-cooler is designed for passive deployment, drawing liquid directly from a reservoir into the wire mesh manifold by capillary suction. In the present experiments, a syringe pump is used not as a flow driver but as a metrological instrument. By supplying liquid at a known inlet flow rate and measuring the drain flow rate, the two-phase mass flow utilized by the wick for nucleate boiling is quantitatively determined. The inlet flow does not pressurize the manifold, as evidenced by the absence of a seal between the PVC supply tubes and the manifold plenum. The liquid free surface in the plenum remains at ambient pressure, which is negligible compared with the ≈ 1 kPa capillary pressure of the wire-mesh. This allows excess liquid to freely bypass through large clearances rather than flowing through the micrometer-sized pores of each wire-mesh sandwiched by the solid copper spacers in the plenum (Fig. 2e). Therefore, the syringe-pump-fed and the capillary-fed modes of operation are hydrodynamically equivalent at the manifold inlet. The cooler itself is therefore a self-regulated, capillary-driven device and the syringe pump simply sets the flow rate at which liquid is made available to the plenum. As illustrated in Fig. 4a and b, a 3D-printed test vehicle is fabricated to accommodate the wire-mesh manifold on the silicon chip cold plate, while providing the fluidic delivery and the drainage lines. The electrical current through the power supplies is supplied from the backside of the cold plate, and the voltage drop across the heater is measured using the digital multimeter (Model: DMM 6500, Manufacturer: Keithley, and accuracy: ± 25 PPM). Temperature maps of the heater are monitored with an infrared (IR) camera (Model: A600, Manufacturer: FLIR, and accuracy: ± 2 °C). The two plastic tubes are inserted at both sides of the manifold plenums. The tube ends are sealed to ensure that the supplied liquid fully wets the manifold. The liquid mass flow rate supplied to the manifold is denoted by $\dot{m}_{in}$. The tiny hole arrays along the tubes are exposed to the inner surface of the manifold plenum. The liquid that wets the manifold plenum is distributed along the individual wire-meshes, which deliver the liquid to the CIO wick on the silicon cold plate. It should be noted that liquid can

continue to be supplied from the manifold to the heated wick to compensate for phase-change-induced liquid loss via capillary flow, which corresponds to $\dot{m}_{2-\phi}$ (the two-phase mass flow rate). Any excess liquid supplied through the two tubes, which is not vaporized during boiling, leaks along the contact line between the tube and the tube hole in the manifold plenum, as well as through the individual wire-meshes that are sandwiched between the solid copper spacers in the plenum. This flow rate is denoted by $\dot{m}_{out}$. The excess water collected through the drainage pipe is measured (averaged over a set time) using a digital scale. Subsequently, the manifold and tube assembly are placed on top of the cold plate, with a gentle normal force applied to prevent lateral movement of the manifold using silicone rubber. Fig. 4(c-e) shows this setup for the different manifold/cooling (wick) area combinations of $5 \times 5 \text{ mm}^2/5 \times 5 \text{ mm}^2$, $10 \times 10 \text{ mm}^2/10 \times 10 \text{ mm}^2$, and $10 \times 10 \text{ mm}^2/5 \times 5 \text{ mm}^2$, respectively.

Liquid flow behavior above the manifold was recorded with an i-SPEED 720 high-speed camera (iX Cameras) at 5000 fps with an exposure time of 100 μs . Frames were retained at a 1:5 ratio, giving an effective rate of 1000 fps, and recording was started only after the base temperature had stabilized at each operating point. A 0.2 s time window (200 frames) is analyzed for every condition. The liquid agitation is quantified by the moving-area fraction, defined as the fraction of pixels inside the manifold active area whose intensity changes between consecutive frames. A pixel is counted as pixels changing by more than 16 grey levels seed regions that grow into connected pixels changing by more than 8 grey levels.

2.4. Data reduction

In this work, deionized water was used as the coolant. To evaluate the heat flux removed by the cooler in this study, the total heat input to the heated area, given by the product of the heater voltage drop (V_{heater}) and electrical current (I_{heater}), is divided by the heater area to determine the heater heat flux, q''_{heater} . For temperature measurement, an IR

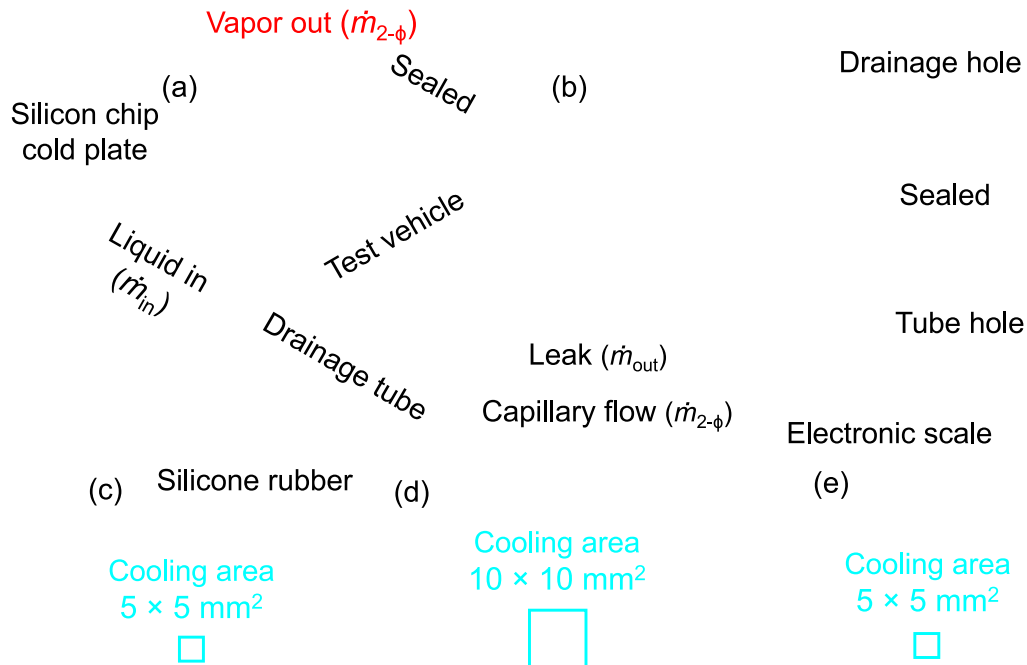


Fig. 4. Thermal test setup. (a) Isometric and (b) top views of the experiment setup illustration. (c–e) Top view pictures showing the manifold secured to the silicon cold plate for three manifold/cooling area combinations: (c) $5 \times 5 \text{ mm}^2/5 \times 5 \text{ mm}^2$, (d) $10 \times 10 \text{ mm}^2/10 \times 10 \text{ mm}^2$, and (e) $10 \times 10 \text{ mm}^2/5 \times 5 \text{ mm}^2$.

camera captures the two-dimensional temperature profile over the heater footprint. Based on the heater temperature, the temperature over the base of the CIO wick is estimated by subtracting the temperature drop due to the heat conduction across the thickness of the silicon substrate from the heater temperature. The base temperature of the wick is given by

$$T_{\text{base}} = T_{\text{heater}} - \frac{d_{\text{Si}}}{k_{\text{Si}}} q''_{\text{heater}} \quad (1)$$

From the IR image of the heater, the temperature value is evaluated at each pixel. The camera views the black-spray-coated heater with an emissivity setting of 0.97 and a pixel pitch of 180 μm . The temperature reading was calibrated against a thermocouple on an identically coated copper shim over 45–140 $^{\circ}\text{C}$, agreeing within 1.6 $^{\circ}\text{C}$ (Supplementary Information S2). Lateral conduction in the 285 μm silicon substrate smooths only features comparable to its thickness. Therefore, the one-dimensional correction of Eq. (1) is accurate for the mm-scale temperature fields reported here. Therefore, the mean temperature of the wick base (the base of the silicon pin array), $T_{\text{mean,base}}$, is calculated by averaging T_{base} over the heated area. The mean heat transfer coefficient attributed to the phase change ($HTC_{2-\phi}$) is calculated from

$$HTC_{2-\phi} = \frac{q''_{2-\phi}}{T_{\text{mean,base}} - T_{\text{sat}}} \quad (2)$$

where $q''_{2-\phi}$ is the heat flux attributed to the phase change and T_{sat} is the water saturation temperature at the atmosphere condition. The heat flux attributed to the phase change is calculated from

$$q''_{2-\phi} = \frac{\dot{m}_{2-\phi} h_{\text{fg}}}{A} \quad (3)$$

in which h_{fg} is the latent heat of vaporization of water and $\dot{m}_{2-\phi}$ is the mass converted from liquid to vapor. $\dot{m}_{2-\phi}$ is measured by subtracting the drain mass flow rate ($\dot{m}_{\text{out}}$) from the inlet flow rate ($\dot{m}_{\text{in}}$) as follows:

$$\dot{m}_{2-\phi} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \quad (4)$$

To quantify the overall capillary-driven two-phase cooling performance, we introduce the two-phase mass-flow-rate utilization (η), which indicates how efficiently the wick utilizes the coolant supplied to the wire-mesh manifold for the phase change. This parameter is calculated as follows:

$$\eta = \frac{\dot{m}_{2-\phi}}{\dot{m}_{\text{in}}} \quad (5)$$

3. Results and discussions

Fig. 5 presents thermal performance data, showing heater heat flux as a function of the mean base temperature of the wick for three footprint combinations of copper wire-mesh manifold/CIO wick. In each plot, the values of $\dot{m}_{\text{in}}$, $\dot{m}_{2-\phi}$, $HTC_{2-\phi}$, and η (defined in Section 2.3 and Section 2.4) are provided at the heat flux level just prior to critical heat flux (CHF). Fig. 5a depicts the experimental results where the manifold and wick (heater) footprint areas are $5 \times 5 \text{ mm}^2$. Liquid was supplied to the manifold plenums via a syringe pump at mass flow rates of $\dot{m}_{\text{in}} = 1.25, 2.50$, and 3.75 g/min . For all three inlet flow rates, the mean base temperature increases linearly with heat flux prior to reaching the saturation temperature (T_{sat}), attributed to the prevalence of heat conduction through the liquid-saturated mesh. In the wick base temperature range between T_{in} and T_{sat} , the inlet mass flow mostly leaks from the plenums rather than entering the wire-mesh array to replenish the wick

Fig. 5. Illustration of the experimental configurations and corresponding thermal performance (q''_{heater} versus $T_{\text{mean,base}}$). Three footprint configurations were tested: (a) $5 \times 5 \text{ mm}^2$ manifold with $5 \times 5 \text{ mm}^2$ wick, (b) $10 \times 10 \text{ mm}^2$ manifold with $10 \times 10 \text{ mm}^2$ wick, and (c) $10 \times 10 \text{ mm}^2$ manifold with $5 \times 5 \text{ mm}^2$ wick. The heated area is located at the center of each manifold footprint. The inlet liquid mass flow rate supplied by syringe pump into the copper wire-mesh manifold ($\dot{m}_{\text{in}}$), the mass flow rate distributed throughout the wire-meshes in the manifold ($\dot{m}_{2-\phi}$), which is used for vaporization, the mean two-phase heat transfer coefficient ($HTC_{2-\phi}$), and the two-phase mass-flow utilization (η) are shown at the heat flux just prior to critical heat flux (CHF).

with subcooled liquid, as the spaces between the interspersed wire-meshes are filled with water. This bypassed fluid contributes negligibly to the cooling process at the wick surface, leading to the characteristically small slope of the thermal performance data in this region. It is noted that the porous wire-mesh architecture is unable to confine liquid flow, preventing the flow path from being pressurized. Therefore, the liquid exiting the perforated PVC tube largely bypasses the manifold, leaking out regardless of the mass flow rate imposed by the syringe pump. However, a slight increase in the slope is observed. This occurs because a higher inlet flow rate delivers marginally more liquid to the plenums, where heat is extracted from the wick mainly through conduction across the wetted copper wire-meshes.

In Fig. 5a, as the heat flux increases over the wick base temperature above $T_{\text{sat}} \approx 100^\circ\text{C}$, the slope of the heat flux curve rises sharply, indicating the onset of nucleate boiling. For the lowest flow rate of 1.25 g/min, a CHF of 237 Wcm^{-2} was achieved. Doubling the inlet flow rate to 2.5 g/min resulted in roughly two-fold increase in CHF. This suggests that all supplied liquid was consumed for vaporization, as evidenced by the absence of liquid flow at the outlets of the test platform ($\dot{m}_{\text{out}} \approx 0$). Tripling the inlet flow rate to 3.75 g/min increases CHF levels to 532 Wcm^{-2} , at which a finite outlet flow rate is measured (Table 1).

It should be emphasized that such a liquid supply imposed by the syringe pump is not representative of practical two-phase cooler operation with the copper wire-mesh manifold used in this study. In practical implementations, the present cooler would be configured as a self-regulated, capillary-driven cooling device, in which the manifold plenum is simply wetted by an external reservoir without a syringe pump. The manifold's intrinsic capillary pressure would draw liquid through the mesh at a rate set by the local vaporization demand at the CIO layer, so the imposed inlet flow is hydrodynamically indistinguishable from a self-regulated, capillary-driven supply at the same flow rate. In the experiments, however, the pump is employed solely to systematically characterize the cooling performance and to elucidate the heat and mass transfer behavior of the capillary-driven cooler using the copper wire-mesh manifold. Therefore, for any given heater heat flux level, the inlet flow rate ($\dot{m}_{\text{in}}$) should be above a certain limit to ensure enough liquid supply available to be transferred from the copper wire-mesh 3D manifold to the CIO wick. The present measurements characterize the device-level limits of a self-regulated capillary architecture, not the performance of an externally pressurized cooler.

To investigate heat and mass transfer in the cooler, we divide the total heater heat flux into parasitic heat conduction through the cold plate to the test platform that supports the cooler (q''_{cond}), sensible heating of the two-phase mass flow (q''_{sens}), convective heat loss to the leaking liquid flow ($q''_{\text{conv,loss}}$), and two-phase heat transfer ($q''_{2-\phi}$). Prior to the onset of boiling (Fig. 6a), the entire supplied liquid mass flow, $\dot{m}_{\text{in}}$, drains into the test platform outlets. The temperature of the draining outlet flows, $\dot{m}_{\text{out}}$, increases to T_{out} , as the inlet mass flow absorbs heat from stagnant liquid in the manifold and the heated wire-mesh manifold while leaking from the manifold ($q''_{\text{conv,loss}}$). After the onset of boiling, two-phase heat transfer is more complex because a fraction of the supplied liquid is routed to the vaporizing area of the heated wick ($\dot{m}_{2-\phi}$), while the remainder drains into the outlets, $\dot{m}_{\text{out}}$, as shown in Fig. 6b. A portion of the heat is transferred to the coolant entering the heated wick during phase change (q''_{sens}). This fluid is heated to T_{sat} and subsequently

fully vaporized across the porous wick. In addition, the remaining fraction ($\dot{m}_{\text{out}}$) leaks from the plenums around the heated area when the inlet mass flow rate exceeds the vaporization rate ($\dot{m}_{2-\phi}$). This leaking flow removes heat from the cooling system via convective heat transfer from the manifold plenum, raising its temperature to T_{out} before it exits the manifold. $\dot{m}_{2-\phi}$ is drawn by capillary force into the copper wire-mesh manifold and supplied to the heated section. At near CHF, $\dot{m}_{\text{out}}$ drops to zero only in specific cases, such as when $\dot{m}_{\text{in}}$ is perfectly balanced with $\dot{m}_{2-\phi}$ (Fig. 6c). CHF is triggered when the liquid supply to the manifold is insufficient to satisfy $\dot{m}_{2-\phi}$. However, even when $\dot{m}_{\text{in}}$ is sufficient, the hydraulic limitations of the wire-meshes and the wick can lead to CHF. In this scenario, excess liquid leaks out of the manifold without vaporization through several leakage pathways in the manifold, as described in Fig. 4a and b.

Based on the heat transfer mechanisms in the capillary-driven two-phase cooler using the wire-mesh manifold, the heater heat flux can be described as the following equation:

$$q''_{\text{heater}} = q''_{\text{cond}} + q''_{\text{sens}} + q''_{\text{conv,loss}} + q''_{2-\phi} \quad (6)$$

The heat conduction loss (q''_{cond}) flows through the silicon chip to the environment, specifically transferring through the chip boundary to the test vehicle or the environment. This is shown as the black lines in the plots in Fig. 5 and is quantified by measuring the mean temperature of the heater as a function of the applied heat flux for a dry cooler (no liquid present), so that the temperature rise is exclusively due to heat conduction through the silicon substrate. The dry-cooler data are fitted by $q''_{\text{cond}} = C1(T_{\text{mean,base}} - T_{\infty})$ with $T_{\infty} = 21.4^\circ\text{C}$ and $C1 = 0.203\text{ Wcm}^{-2}\text{ K}^{-1}$ for the $5 \times 5\text{ mm}^2$ heater and $0.073\text{ Wcm}^{-2}\text{ K}^{-1}$ for the $10 \times 10\text{ mm}^2$ heater, measured over $21\text{--}90^\circ\text{C}$. Sensible heating (q''_{sens}) is the heat flux required to raise the two-phase mass flow rate ($\dot{m}_{2-\phi}$) to the saturation temperature, T_{sat} .

$$q''_{\text{sens}} = \dot{m}_{2-\phi} C_p (T_{\text{sat}} - T_{\text{in}}) A_{\text{heater}}^{-1} \quad (7)$$

where C_p is the specific heat of water, and T_{in} is the inlet liquid temperature. It is noted that q''_{sens} is zero, when the wick base temperature is below the saturation temperature, since phase change is negligible in this regime. $q''_{\text{conv,loss}}$ accounts for the convective heat transfer of the outlet flow ($\dot{m}_{\text{out}}$), raising the inlet liquid temperature to T_{out} prior to exiting the manifold plenum without routing to the wick area. T_{out} is the temperature of the liquid leaking from the manifold plenums. T_{out} cannot be directly measured, since the liquid flow is not confined to channels but spreads as intermittent flow and dispersed liquid films over the silicon chip, which makes a well-defined outlet bulk flow temperature unavailable. However, this contribution is negligible during the two-phase regime at high heat flux level, as will be discussed later. Because $T_{\text{in}} < T_{\text{out}} < T_{\text{sat}}$, setting $T_{\text{out}} = T_{\text{sat}} = 100^\circ\text{C}$ is an upper bound that maximizes $q''_{\text{conv,loss}}$. At the six operating points with $\dot{m}_{\text{out}} = 0$ the term vanishes identically, and at the three points with finite drainage (15 g/min/cm^2), sweeping T_{out} over its full physical range from T_{in} to T_{sat} changes the calculated total heat flux by at most 16.8, 26.0 and 15.6 Wcm^{-2} , which are 3.1, 5.9 and 2.9% of CHF for the $5 \times 5\text{ mm}^2/5 \times 5\text{ mm}^2$, $10 \times 10\text{ mm}^2/10 \times 10\text{ mm}^2$ and $10 \times 10\text{ mm}^2/5 \times 5\text{ mm}^2$ heated area/manifold area configurations, respectively. The reported CHF, $HTC_{2-\phi}$ and η contain no T_{out} input and are unaffected. $q''_{\text{conv,loss}}$ is expressed in Eq. (8).

Table 1

Thermal performance at near CHF for different flow rates using the wire-mesh manifold and heated area having the same size as $5 \times 5\text{ mm}^2$.

	$\dot{m}_{\text{in}}$ g(min) ⁻¹	$\dot{m}_{\text{in}}/A_{\text{heater}}$ g(min) ⁻¹ cm ⁻²	$\dot{m}_{\text{in}}/A_{\text{manifold}}$ g(min) ⁻¹ cm ⁻²	$\dot{m}_{\text{out}}$ g (min) ⁻¹	$\dot{m}_{2-\phi,\text{CHF}}$ g(min) ⁻¹	$q''_{\text{heater,CHF}}$ Wcm ⁻²	$Q_{\text{heater,CHF}}$ W	$T_{\text{mean,base,CHF}}$ °C	$q''_{2-\phi,\text{CHF}}$ Wcm ⁻²	$HTC_{2-\phi,\text{CHF}}$ MWm ⁻² °C ⁻¹	η
Manifold 5	1.25 ± 0.03	5.00 ± 0.12	5.00 ± 0.12	0	1.25 ± 0.03	237.3 ± 0.2	59.3	105.3 ± 2.1	188.3 ± 3.8	0.35 ± 0.14	1
× 5 mm ²	2.50 ± 0.05	10.00 ± 0.21	10.00 ± 0.21	0	2.50 ± 0.05	455.4 ± 0.5	113.9	106.7 ± 2.1	376.5 ± 7.5	0.56 ± 0.18	1
Heater 5	3.75 ± 0.08	15.00 ± 0.25	15.00 ± 0.25	0.79 ± 0.01	2.96 ± 0.08	532.4 ± 0.5	133.1	107.5 ± 2.2	445.2 ± 11.3	0.59 ± 0.18	0.79
× 5 mm ²											

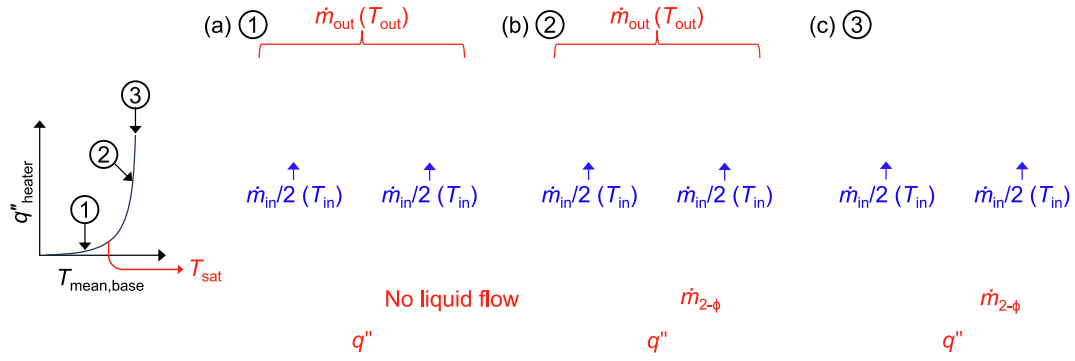


Fig. 6. Schematic representation of the heat and mass transfer in three distinct regimes: (a) Heat transfer regime prior to the onset of boiling, (b) two-phase heat transfer regime with finite leakage, and (c) two-phase heat transfer regime without leakage, where the capillary-driven flow is completely balanced with the vaporization rate.

$$q''_{\text{conv,loss}} = \dot{m}_{\text{out}} C_p (T_{\text{out}} - T_{\text{in}}) A_{\text{heater}}^{-1} \quad (8)$$

The remaining two-phase heat flux ($q''_{2-\phi}$) is derived from the measured vaporization mass flow rate ($\dot{m}_{2-\phi}$). The uncertainty analysis is presented in Supplementary Information S1.

As shown in Fig. 7a, for the three supplied inlet flow rates, the measured CHF is compared with the heat flux contributions, which are broken down into heat conduction, sensible heating of the two-phase mass flow, convective heat transfer attributed to the leaking mass flow, and two-phase heat transfer. The sum of these heat transfer components matches well with the measured heater heat flux just prior to critical heat flux. At the high heat flux, two-phase heat transfer is the dominant heat transfer mechanism, while the others are significantly smaller. At the largest inlet flow rates (3.75 g/min) where a finite outlet flow is observed, the outlet temperature is assumed to be 100 °C, since direct measurement was unavailable. Even with this conservative assumption, its contribution is negligible, as a large portion of the supplied liquid is vaporized, leaving a small draining mass flow rate. Therefore, the sum of the heat transfer components closely follows the measured heater heat flux at the highest inlet mass flow rate, with the calculated value slightly exceeding the measured one. This small excess may originate from liquid escaping as fine droplets, which never absorb their latent heat but are counted in $\dot{m}_{2-\phi}$ as vaporized. Furthermore, no droplet ejection is discernible in the high-speed recordings, and thus the

excess is treated as an upper bound on such losses. Across the nine near-CHF points, the bound is at most 0.04, 0.59 and 0.03 g/min (1.3, 5.8 and 1.2% of $\dot{m}_{2-\phi}$) at the three points where the calculated value is the larger one, and zero within uncertainty at the others. This could overestimate the heat transfer coefficient $HTC_{2-\phi}$, the vaporization rate and η by at most these fractions. Even at the worst point, the vaporization plateau of the $10 \times 10 \text{ mm}^2$ heater/manifold changes from 10.1 to no less than 9.5 g/min. No liquid accumulates during the steady-state recordings, and no evaporation occurs outside the heated area, which stays well below the saturation temperature. Additionally, the heat flux contributing to heat conduction loss is estimated by analyzing the thermal performance data in the temperature region (where $T_{\text{mean,base}} < T_{\text{sat}}$). By fitting a linear curve to these data points and extrapolating to the heater temperature at CHF, we extract the heat flux contribution due to heat conduction loss at CHF level (shown in white in Fig. 7). For the two low flow rates (1.25 g/min and 2.5 g/min), no outlet flow was measured. Therefore, the supplied liquid is raised to saturation temperature and mostly vaporized. The calculated two-phase mass flow rate utilization remains $\eta \approx 1$. Furthermore, increasing the inlet flow to 3.75 g/min results in $\eta \approx 0.79$, indicating that 21% of the supplied liquid is not utilized for vaporization and is drained. These results suggest that the boiling performance is limited either by the wick's capacity (despite sufficient liquid transport) or by the wire-mesh's ability to supply liquid to the heated area.

To investigate the limiting factors of the cooler performance, we

Fig. 7. Comparison of measured heater heat flux (grey) against calculated heat flux for various supplied inlet mass flow rates ($\dot{m}_{\text{in}}$). The data shown in the plots correspond to the heat flux measured immediately prior to wick dry-out. The calculated components include sensible heat transfer of the two-phase mass flow (blue), convective heat loss of the outlet flow (magenta), two-phase heat transfer (red), and conduction loss (white). Calculations of sensible heating and two-phase heat transfer rely on the associated mass flow rates and the water thermophysical properties. The outlet flow temperature is assumed to be 100 °C (saturation temperature) as a conservative estimate, as direct measurement was not available. (a), (b), and (c) correspond to three manifold/cooling area combinations: $5 \times 5 \text{ mm}^2 / 5 \times 5 \text{ mm}^2$, $10 \times 10 \text{ mm}^2 / 10 \times 10 \text{ mm}^2$, and $10 \times 10 \text{ mm}^2 / 5 \times 5 \text{ mm}^2$, respectively. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

performed similar thermal experiments using an enlarged manifold and a silicon cold plate with a matching footprint of $10 \times 10 \text{ mm}^2$. The experiment configuration and the thermal performance are shown in Figs. 4d and 5b. For this experiment, inlet flow rates of 5, 10, and 15 g/min were used. These flow rates were selected to maintain the same mass flux over the heated area as a cooler with a matching $5 \times 5 \text{ mm}^2$ footprint. The data curves in the single-phase heat transfer region exhibit a significantly lower slope compared to those of the $5 \times 5 \text{ mm}^2$ cooler (Fig. 5a), because the mass flux ($\dot{m}_{\text{in}}/A_{\text{heater}}$) does not influence the heat transfer prior to the onset of boiling, since there is no forced liquid convection. Instead, the liquid in the active area of the wire-mesh manifold remains effectively stagnant due to the absence of liquid loss over the heated wick area. However, boiling heat transfer shows similar trend to that of the smaller cooler (Fig. 5a). As shown in Fig. 5b, at $\dot{m}_{\text{in}} \approx 5 \text{ g/min}$, the CHF is measured to be 233 Wcm^{-2} . Increasing the inlet mass flow rate to 10 g/min dramatically raises the CHF to 441 Wcm^{-2} , which is almost directly proportional to the increase in inlet flow rate. This behavior indicates that at 5 g/min and 10 g/min mass flow rates, most of the supplied liquid using the syringe pump is being transported through the wire-meshes, and the CHF remains liquid-limited at these lower flow rates. The further increase in the inlet flow rate from 10 g/min to 15 g/min, the CHF nearly unchanged, indicating that the system does not approach the CHF achieved with the $5 \times 5 \text{ mm}^2$ cooler ($\approx 532 \text{ Wcm}^{-2}$). This supports that the cooling performance over the larger footprint is limited by the wire-mesh manifold rather than by the wick, since the wick design is identical to that used in the smaller $5 \times 5 \text{ mm}^2$ cooler. Based on this experiment result, we estimate the maximum liquid transport capacity of the wire-mesh. Given that the $5 \times 5 \text{ mm}^2$ cooler has half the number of wire-meshes compared to the $10 \times 10 \text{ mm}^2$ design, the maximum liquid transport capacity of $5 \times 5 \text{ mm}^2$ wire-mesh manifold is estimated to be at least 5 g/min, corresponding to a two-phase heat flux of 752 Wcm^{-2} . This conservative estimate disregards the smaller viscous pressure drop across the wire-mesh and thus the $5 \times 5 \text{ mm}^2$ manifold can support a capillary-liquid driven liquid flow rate greater than 5 g/min. However, the measured CHF for the $5 \times 5 \text{ mm}^2$ case using 3.75 g/min is much lower than 752 Wcm^{-2} , indicating that the wick limits the cooling performance in this scenario. For the $10 \times 10 \text{ mm}^2$ cooler, the breakdown of heat transfer pathways (Fig. 7b) shows that the sum of the calculated heat flux components matches well with the measured CHF. It is noted that at a 15 g/min inlet flow rate, the calculated heat flux is slightly higher than the measured CHF. This discrepancy is attributed to the assumption of a 100°C outlet temperature, which likely overestimates the sensible heat contribution, resulting in a higher total calculated heat flux. The two-phase mass flow rate utilization (η) is calculated for the corresponding inlet flow rates. For two cases of low flow rates, η remains nearly identical and close to 1, indicating that most of the liquid drawn through the wire-meshes is utilized for vaporization with minimal leakage. This behavior contrasts with the results observed for the higher inlet flow rate (15 g/min), showing $\eta \approx 0.67$ and a large portion of the liquid cannot be transported to the wick for phase change. The performance of the cooler with a matched $10 \times 10 \text{ mm}^2$ manifold and heater (wick) footprint near the CHF point is summarized in Table 2. Scaling the heated footprint from $5 \times 5 \text{ mm}^2$ (Table 1) to $10 \times 10 \text{ mm}^2$ (Table 2), enabled by the larger manifold area, increases the heated area by a factor of four and,

correspondingly, significantly increases the total heat dissipation rate (Q_{heater}) with the given heat flux level. This demonstrates substantially enhanced heat removal capability.

To better understand the liquid delivery capacity of the wire-mesh manifold, a smaller $5 \times 5 \text{ mm}^2$ cold plate is tested with a larger $10 \times 10 \text{ mm}^2$ manifold, as shown in Fig. 5c. Since the heater size is one-fourth the size of the $10 \times 10 \text{ mm}^2$ manifold, the flow rates were scaled down by a factor of four (1.25, 2.50, and 3.75 g/min) to fix the mass flux. These rates match those used in the configuration where both the manifold and heater are $5 \times 5 \text{ mm}^2$ (Fig. 5a). It is noted that the mass flux, based on the manifold size, is one-fourth that of the matched $5 \times 5 \text{ mm}^2$ configuration and also one-fourth that of the matched $10 \times 10 \text{ mm}^2$ case. First, in the single-phase region prior to the onset of boiling (ONB), the slope of the heat flux curve is steeper than that of the matched $10 \times 10 \text{ mm}^2$ case (Fig. 5b), but it closely follows the trend of the $5 \times 5 \text{ mm}^2$ case (Fig. 5a), despite the lower mass flux based on the manifold size. This suggests that sensible heating prior to ONB is negligibly correlated with the mass flux (defined as the inlet mass flow rate divided by the manifold footprint area). Instead, it appears to be governed by conduction rather than convection, as discussed in previous sections. This is because effective delivery of fresh coolants to the wick surface requires active vaporization to deplete the fluid residing within the wire-mesh screens, creating a capillary-driven suction that draws fresh coolant from the plenum. In addition, as shown in Fig. 5b, the larger manifold and heater exhibit a smaller slope prior to ONB, because the increased distance between the heated zone and the plenum further weakens the heat transfer from the heated wick to the leaking flow in the manifold plenums.

With a further increase in heat flux, the cooler system enters the nucleate boiling regime, where the heat transfer behavior converges with those of the matched $5 \times 5 \text{ mm}^2$ manifold/heater configuration case. Specifically, the heat flux reaches 242 Wcm^{-2} at a flow rate of 1.25 g/min and increases to 437 Wcm^{-2} at 2.50 g/min. The thermal data prior to CHF is presented in Table 3. At these inlet flow rates to the manifold, no outlet flow is observed, indicating full utilization of the supplied liquid. This result is consistent with the matched $5 \times 5 \text{ mm}^2$ configuration, despite the manifold extending beyond the heated area. Notably, it does not exhibit the performance degradation observed in the experiments using the larger $10 \times 10 \text{ mm}^2$ manifold and heated area. Fig. 8 illustrates the hydrodynamics during both sensible heating and two-phase heat transfer. As shown in Fig. 8a, during the sensible heating regime where the wick base temperature is well below the saturation temperature, the liquid fills the entire manifold footprint, resulting in minor heat transfer to the leaking flow from the manifold plenums. However, during significant nucleate boiling, the liquid meniscus in the peripheral regions in the wire-meshes remains intact, because the heater is confined to the central $5 \times 5 \text{ mm}^2$ area. Therefore, the liquid within the wire-meshes outside this heated area is maintained without interacting with the generated vapor. In addition, at low flow rates (1.25 g/min and 2.50 g/min), all supplied liquid is consumed by vaporization. This is enabled because the liquid saturating the wire-mesh around the heated area can be drawn into the heated zone to replenish the liquid loss owing to phase change. The liquid meniscus form in the gaps between neighboring wire-meshes and is subsequently transported toward the heated area, as illustrated in Fig. 8b. At the highest flow rate (3.75 g/

Table 2

Thermal performance at near CHF for different flow rates using the wire mesh-manifold and heated area having the same size as $10 \times 10 \text{ mm}^2$.

	$\dot{m}_{\text{in}}$ $\text{g}(\text{min})^{-1}$	$\dot{m}_{\text{in}}/A_{\text{heater}}$ $\text{g}(\text{min})^{-1}\text{cm}^{-2}$	$\dot{m}_{\text{in}}/A_{\text{manifold}}$ $\text{g}(\text{min})^{-1}\text{cm}^{-2}$	$\dot{m}_{\text{out}}$ $\text{g}(\text{min})^{-1}$	$\dot{m}_{2-\phi, \text{CHF}}$ $\text{g}(\text{min})^{-1}$	$q''_{\text{heater, CHF}}$ Wcm^{-2}	$Q_{\text{heater, CHF}}$ W	$T_{\text{mean, base, CHF}}$ $^\circ\text{C}$	$q''_{2-\phi, \text{CHF}}$ Wcm^{-2}	$HTC_{2-\phi, \text{CHF}}$ $\text{MWm}^{-2}\text{C}^{-1}$	η
Manifold $10 \times 10 \text{ mm}^2$	5.0 ± 0.1	5.00 ± 0.12	5.0 ± 0.1	0	5.0 ± 0.1	223.1 ± 0.2	223.1	110.0 ± 2.1	188.3 ± 3.8	0.20 ± 0.05	1
	10.0 ± 0.2	10.00 ± 0.21	10.0 ± 0.2	0	10.0 ± 0.2	441.1 ± 0.4	441.1	109.3 ± 2.1	376.5 ± 7.5	0.41 ± 0.10	1
Heater $10 \times 10 \text{ mm}^2$	15.0 ± 0.3	15.00 ± 0.25	15.0 ± 0.3	4.93 ± 0.01	10.1 ± 0.3	441.2 ± 0.4	441.2	108.6 ± 2.2	379.1 ± 11.3	0.44 ± 0.12	0.67

Table 3

Thermal performance at near CHF for different flow rates using the wire-mesh manifold (10 mm × 10 mm) and heater (5 mm × 5 mm).

	$\dot{m}_{in}$ g(min) ⁻¹	$\dot{m}_{in}/A_{heater}$ g(min) ⁻¹ cm ⁻²	$\dot{m}_{in}/A_{manifold}$ g(min) ⁻¹ cm ⁻²	$\dot{m}_{drain}$ g (min) ⁻¹	$\dot{m}_{2-\phi,CHF}$ g(min) ⁻¹	$q'_{heater,CHF}$ Wcm ⁻²	$Q_{heater,CHF}$ W	$T_{mean,base,CHF}$ °C	$q'_{2-\phi,CHF}$ Wcm ⁻²	$HTC_{mean,2-\phi,CHF}$ MWm ⁻² °C ⁻¹	η
Manifold 10 × 10 mm ²	1.25 ± 0.03	5.00 ± 0.12	1.25 ± 0.03	0	1.25 ± 0.03	241.6 ± 0.2	60.4	107.6 ± 2.1	188.3 ± 3.8	0.25 ± 0.07	1
Heater 5 × 5 mm ²	2.50 ± 0.05	10.00 ± 0.21	2.50 ± 0.05	0	2.50 ± 0.05	436.9 ± 0.4	109.2	108.8 ± 2.2	376.5 ± 7.5	0.43 ± 0.11	1
	3.75 ± 0.08	15.00 ± 0.25	3.75 ± 0.08	0.74 ± 0.01	3.01 ± 0.08	545.6 ± 0.6	136.4	109.4 ± 2.2	453.6 ± 11.3	0.48 ± 0.12	0.80

Fig. 8. Illustration of the hydrodynamics within the micro-cooler using the wire-mesh manifold with a 10 × 10 mm² footprint and a 5 × 5 mm² heater. (a) Single-phase heat transfer regime. (b) Two-phase heat transfer regime under significant phase change.

min), a CHF of 546 Wcm⁻² is achieved with a leaking flow rate of 0.74 g/min. These results are comparable to the matched 5 × 5 mm² case (Fig. 5a), which is attributed to the fact that the wick limits the boiling performance. As shown in Fig. 7c, the calculated total heat flux consists of heat conduction loss, heat flux owing to sensible heating and convection loss, and two-phase heat flux agrees well with the measured CHF. Comparing thermal data across the three manifold/heater configurations, using the same mass flux based on heater size (5 g/min/cm² and 10 g/min/cm²), we observe that both CHF and $HTC_{2-\phi}$ are comparable to each other. This indicates that the wire-mesh array sustains an efficient liquid supply over the wick, because the wire-mesh configuration on the CIO wick is identical in all three cases at low flow rates. However, significant deviations occur at higher mass flux based on heater size (15 g/min/cm²). This comparison reveals that overall cooling performance is ultimately constrained by the ability of the manifold to supply sufficient liquid across the larger heated footprint. Two-phase mass flow utilization (η) decreased from 0.8 to 0.67 as the heater size increased from 5 × 5 mm² to 10 × 10 mm², because liquid transport for phase change could not be sustained over the larger area due to the large viscous pressure drop of liquid flow along the wire-meshes.

Liquid flow behavior above the wire-mesh were measured using high-speed imaging at several heat fluxes and inlet mass flow rates for different manifold/cooling area configurations, as shown in Fig. 9. Fig. 9a shows the high-speed camera images of the matched 5 × 5 mm² manifold and wick configuration. Using the low inlet mass flow rate of 1.25 g/min, the manifold is totally wetted at a heat flux of 0 Wcm⁻², although the liquid is difficult to distinguish in the image. With a slight increase in heat flux to 68 Wcm⁻², tiny bubbles fill the gaps within the wire-mesh, confirming the presence of stagnant liquid between the mesh-screens. As the heat flux increases further toward CHF, the liquid recedes completely, and only vapor exits through apertures in the manifold. At this CHF level, increasing the flow rate to 2.5 g/min causes bubbles to form and eject from the wire-mesh manifold due to the excess

liquid supply. Again, the liquid in the manifold receded as the heat flux increased to near CHF ($\approx$ 440 Wcm⁻²) at an inlet mass flow rate of 2.5 g/min, without any observed bubble formation. The higher inlet flow rate of 3.75 g/min shows a slightly different trend. At a CHF heat flux near 535 Wcm⁻², significant bubble formation is still observed, due to limitations imposed by the wick. The matched 10 × 10 mm² configuration shows a similar trend with low inlet flow rates of 5 g/min and 10 g/min (Fig. 9b). At zero heat flux, the manifold is observed to be fully saturated with liquid. A small increase in heat flux to 55 Wcm⁻² leads to the formation of tiny bubbles that become trapped within the manifold. As the heat flux is further increased to 240 Wcm⁻², the excess liquid recedes completely, indicating a balance between the vaporization rate and the inlet mass flow rate. Using a large inlet flow rate at this heat flux leads to a periodic cycle of vibrating bubble formation and breakage. Further increasing the heat flux to near the CHF level causes the liquid to recede again. The cooler performance is limited to a heat flux of 440 Wcm⁻², and the increase in the heat flux leads to CHF. This CHF level is much lower than that of the 5 × 5 mm² case, implying that the cooler performance is limited by the liquid transport capacity of the wire-mesh. Lastly, for the 10 × 10 mm² manifold and 5 × 5 mm² heater configuration (Fig. 9c), a key observation is that, at CHF with liquid flow rates of 1.25 g/min and 2.50 g/min, excess liquid is not observed over the entire manifold area, even though the wick size is much smaller than the manifold footprint. This implies that liquid from the surrounding region is drawn into the wick during vaporization. At a higher flow rate of 3.75 g/min, bubbling around the smaller heated area continues near CHF due to the phase change limitations due to the wick, displaying the trend seen in the cooler with the smaller manifold size (Fig. 9a).

The flow behavior of Fig. 9 is quantified in Fig. 10. Fig. 10a and b compares consecutive frames (1 ms apart) at CHF of the matched 10 × 10 mm² configurations, with the pixels that moved between the two frames marked in blue inside the manifold active area. At 5 g/min (223 Wcm⁻²) the receded manifold is motionless. The moving-area fraction is

Fig. 9. Speed camera images at the various inlet flow rates and heat fluxes for: (a) manifold (a) $5 \times 5 \text{ mm}^2$ manifold with $5 \text{ mm} \times 5 \text{ mm}$ wick, (b) $10 \times 10 \text{ mm}^2$ manifold with $10 \times 10 \text{ mm}^2$ wick, and (c) $10 \times 10 \text{ mm}^2$ manifold with $5 \times 5 \text{ mm}^2$ wick. The corresponding two-phase mass flow utilization values are shown on the right of images, evaluated near the CHF condition for each case.

≈ 0 , indistinguishable from a zero-heat-flux case. At 15 g/min (441 Wcm^{-2}), by contrast, about half of the manifold area is in motion within a single millisecond (≈ 0.49). Fig. 10c and d shows the time traces of the moving-area fraction for the matched configurations at increasing heat flux. Under vigorous boiling, the fractions fluctuate with dominant frequencies of $15\text{--}30 \text{ Hz}$. Two distinct CHF signatures emerge. Whenever the supplied liquid is fully vaporized at CHF, the moving fraction collapses below 0.01 , as the liquid is exhausted and CHF coincides with a receded, motionless manifold. At the highest mass flux of $15 \text{ g(min)}^{-1} \text{ cm}^{-2}$, however, the manifold remains in vigorous motion up to CHF, while part of the supplied liquid drains unused.

Fig. 11 shows the two-dimensional wick base temperature distributions at heat flux levels just below CHF for the same cooler

configurations in Fig. 5 as the inlet liquid mass flow rate supplied through the two PVC tubes is increased. For all cases, the temperature fields are uniform over the heated footprint, indicating effective lateral liquid supply through the wire-mesh manifold and strong capillary driven boiling at the wick surface. As the inlet mass flow rate increases, the overall temperature rises, correlating with the increase in CHF observed for the $5 \times 5 \text{ mm}^2$ heater cases (Fig. 11a and c). In contrast, for the matched heater/manifold case with the larger footprint $10 \times 10 \text{ mm}^2$ (Fig. 11b), the temperature profiles at 10 and 15 g/min are similar to each other, reflecting the comparable CHF values achieved and two-phase vaporization rate at these flow rates. As increasing the inlet mass flow rate leads to only a slight change in the mean base temperature (Fig. 5) and in the temperature pattern (Fig. 11), its influence is

Fig. 10. (a) and (b) Consecutive frames (1 ms apart) at CHF of the $10 \times 10 \text{ mm}^2$ manifold with the $10 \times 10 \text{ mm}^2$ heater at (a) 5 g/min (223 Wcm^{-2}) and (b) 15 g/min (441 Wcm^{-2}). The left image of each pair is the raw frame. In the right image, the pixels that moved between the two frames are marked in blue within the manifold active area (yellow box). The moving-area fraction is the blue fraction of the box ≈ 0 in (a) and ≈ 0.49 in (b). (c) and (d) Time traces of the moving-area fraction for the matched configurations at the labeled heat fluxes and inlet flow rates.

negligible during intense phase change. At 5 and 10 g/min, the two-phase mass-flow utilization is unity, so the CHF simply follows the supply rate. At 15 g/min, 4.93 g/min drains away unused while the vaporization rate saturates at 10.1 g/min, so liquid is available in excess but is not converted to vapor. Vapor blockage is not responsible, since the vapor escape path is identical in all three manifolds. Local dry-out is not a separate mechanism but the consequence of reaching the capillary limit of the liquid over the wire-meshes, and the infrared maps at the onset of runaway show it initiating at the center of the heated area, the point farthest from both plenums, as shown in Fig. S3 in Supplementary Information S4.

To understand the observed transport limit, the capillary-driven liquid flow along a single mesh sheet of the manifold was evaluated with a two-dimensional, thickness-averaged Darcy model, as shown in Fig. 12. Transport input is the measured permeability-thickness product $K \cdot \xi$ as a function of capillary pressure (P_{cap}) of the identical #100 copper mesh [26]. The formulation and boundary conditions are given in Supplementary Information S5. The measured $K \cdot \xi$ corresponds to a freshly cleaned mesh (Fig. 12a). Therefore, the curves of Fig. 12c represent an optimistic estimate. A mesh performing below that condition could shift the characteristic of Fig. 12a toward the lower left and reaches its limit at a lower heat flux. At the highest vaporization rates measured on the $5 \times 5 \text{ mm}^2$ manifolds, the mesh is required to generate approximately 200 Pa, which is a fifth of its cleaned-mesh capillary pressure. This indicates the CHF is potentially limited by the wick. At the plateau of the $10 \times 10 \text{ mm}^2$ manifolds, the required capillary pressure is 376 Pa, comparable to the capillary pressure of the mesh in its as-received state. Beyond $\approx 600 \text{ Wcm}^{-2}$, the predicted pressure drop rises steeply as $K \cdot \xi$ de-wets. It should be noted, however, that both the $K \cdot \xi$ characteristic and the maximum capillary pressures indicated in Fig. 12a correspond to a freshly cleaned mesh, which is the most favorable condition. A mesh that has been assembled, sintered, and operated for an extended period may fall below this condition, which shifts the characteristic toward the lower left of Fig. 12a. The plateau permeability is reduced, and de-wetting begins at a lower capillary pressure, while the maximum capillary pressure available is likewise

lower. In that case the steep rise would set in below $\approx 600 \text{ Wcm}^{-2}$ and the margin against the transport limit would be smaller than Fig. 12c suggests, so the values calculated here are an optimistic estimate rather than a strict prediction. The CHF of the $10 \times 10 \text{ mm}^2$ cooler could be therefore attributed to the capillary transport limit of the manifold.

Table 4 compares the present cooler with representative capillary-fed evaporators. At the 25 mm^2 footprint, the wire-mesh manifold coolers with CIO pin-fin wicks (Cases A, B, and C) sustain CHF levels of 441–546 Wcm^{-2} at superheats below 10°C , compared with 159–492 Wcm^{-2} for the manifold-free or pool-fed counterparts of the same footprint [8,16]. At the 100 mm^2 footprint, the present cooler removes 441 W (441 Wcm^{-2}), comparable to the heat high removal reported for a capillary-fed evaporator of this footprint [10], but at approximately one third of the superheat (8.6°C versus 25°C) and more than twice the two-phase heat transfer coefficient (0.44 versus $0.19 \text{ MWm}^{-2}\text{C}^{-1}$), due to the thin CIO wick ($\approx 30 \mu\text{m}$) with its small pore size ($\approx 5 \mu\text{m}$) used in this study. The $\approx 20\%$ CHF penalty relative to the 25 mm^2 footprint quantifies the capillary transport limit of the present manifold.

4. Conclusion

In this study, we systematically investigated the thermal performance and liquid-delivery mechanisms of a capillary-driven micro-cooler that integrates a copper wire-mesh manifold with silicon cold plates with square pin-fin wick coated with a conformal $30 \mu\text{m}$ -thick CIO layer on silicon pin-fin wick. By varying the manifold footprint ($5 \times 5 \text{ mm}^2$ and $10 \times 10 \text{ mm}^2$) and heater size ($5 \times 5 \text{ mm}^2$ and $10 \times 10 \text{ mm}^2$), we explored how liquid supply, leakage, and transport within the wire-mesh manifold affect two-phase boiling performance. For the $5 \times 5 \text{ mm}^2$ cooler, where the manifold and wick (heater) have the same footprint, we achieved a high CHF exceeding 500 Wcm^{-2} with a total supplied flow rate of only 3.75 g/min (15 g/min cm^{-2}). In this study, the liquid supply was intentionally pressurized to precisely control and quantify the mass flow rate delivered to the cooler and thereby isolate the underlying physical behavior. This approach differs from conventional capillary-driven boiling experiments that rely solely on passive liquid

Fig. 11. At heat flux levels just below CHF, the temperature profiles at the wick base are presented for the three cooler configurations (each row) with the increasing inlet liquid flow rate (5, 10, and 15 g/min) through the two PVC tubes (each column). For each map, the mean, maximum, and minimum base temperature over the heated footprint are presented.

Fig. 12. Capillary transport model of a single wire mesh in the manifold. (a) Permeability-thickness product of a cleaned single-layer #100 copper mesh as a function of capillary pressure, measured in Ref. [26] and used here as the only transport input. The vertical lines mark the maximum capillary pressure of the same mesh in the as-received (≈ 0.37 kPa) and cleaned (≈ 1.03 kPa) states. (b) Model domain and boundary conditions for one mesh sheet above the wick. The two plenum edges are held at the reference pressure ($P_l = 0$), the top edge is impermeable, and liquid is withdrawn uniformly over the heated length L_{heater} . (c) Predicted liquid pressure drop ΔP_l along the mesh as a function of the two-phase heat flux, for the matched $5 \times 5 \text{ mm}^2$ ($L_{\text{heater}} = 5 \text{ mm}$) and $10 \times 10 \text{ mm}^2$ ($L_{\text{heater}} = 10 \text{ mm}$) configurations.

reservoirs, but it enables a more accurate assessment of the liquid-delivery limitations that will guide the design of future capillary-

driven cooler systems. Furthermore, the vaporization rate is completely balanced with the inlet mass flow rate below 3.75 g/min.

Table 4

Comparison of thermal performance, liquid-driving modes, and pumping requirements with previous studies.

Study	Heated area (mm ²)	CHF (Wcm ⁻²)	Total heat removal (W)	Superheat (°C)	HTC (MW m ⁻² °C ⁻¹)	Liquid-driving mode	Pumping requirement
Nam et al. [8]	25	159	40	33	0.05	Capillary, integrated Cu micropost wick, pool-fed	None
Ćoso et al. [9]	100	277	277	14	0.2	Capillary, biporous Si pin-fin wick with liquid channels	None
Ćoso et al. [9]	6.25	733	46	7	–	Capillary, biporous wick	None
Sudhakar et al. [10]	100	485	123	25	0.19	Capillary-fed two-layer sintered Cu evaporator with vapor vents	None
Zhou et al. [21]	79	203	159	20	0.12	Capillary, gradient wire-mesh wick	None
Giglio et al. [16]	25	492	37	37	0.13	Cu wire-mesh manifold on AlN pin-fin wick, saturated pool	None (pool-fed)
Kong et al. [23]	25	487	122	6.2	0.33	Cu wire-mesh manifold on Si micro-pin-fin wick	Syringe pump for metering only
Kwon et al. [17]	25	289	72	8	0.42	Si microchannel manifold on CIO wick, hybrid single/two-phase	Pressurized supply
This work, case A	25	532	133	7.5	0.59	Cu wire-mesh manifold on CIO-coated Si pin-fin wick, capillary	None (pump for metering only)
This work, case B	100	441	441	8.6	0.44	As above	None (pump for metering only)
This work, case C	25	546	136	9.4	0.48	As above	None (pump for metering only)

This behavior is confirmed by high-speed imaging of the wire-mesh manifold near CHF and is reflected in a two-phase mass flow utilization of unity (defined as the ratio of vaporization rate to inlet mass flow rate).

The area-scaling performance of the manifold with a larger 1 cm × 1 cm footprint shows a reduction in critical heat flux by 20% lower than that of the 5 mm × 5 mm cooler using the same mass flux to the heated area (15 g/min/cm²). This decrease indicates that the capillary liquid-delivery limit of the wire-mesh manifold is reached, since the same type of copper wire-mesh is used for both the small and large manifold. In this configuration, the cooling performance is therefore limited by the manifold's ability to deliver sufficient liquid over the larger transport distance, rather than by the wick itself. However, at lower flow rates, the two-phase mass flow utilization approaches unity. This behavior is consistent with the 5 × 5 mm² manifold/heater cooler and confirms that essentially most of liquid drawn through the wire-meshes is vaporized during nucleate boiling. Scaling the manifold footprint to 10 × 10 mm² while maintaining the same 5 × 5 mm² heated area revealed no measurable degradation in CHF compared to the counterpart with the matched manifold and heater size. This is because there is no liquid loss in the unheated outer region. Consequently, the effective flow rate over the longer wire-mesh is lower compared to the case with a larger heater size at the same heat flux. Overall, these results establish the primary limiting mechanism for cooling performance, depending on the footprint configuration of the manifold and the heater.

This study demonstrates that the copper wire-mesh manifold has potential for area scaling and passive operation, provided that the capillary characteristic of the wire-meshes and the liquid supply architecture are properly engineered. The present results demonstrate footprint scaling from 25 to 100 mm², over which the peak CHF at the highest mass flux decreases by ≈ 20%, quantifying the capillary transport limit of the present copper wire-mesh manifold. The insights gained here lay the groundwork for designing next-generation, high-heat-flux, capillary-driven two-phase micro-coolers for high-power computing electronics, power electronics systems, and data-center applications.

CRedit authorship contribution statement

Heungdong Kwon: Writing – review & editing, Writing – original draft, Visualization, Validation, Methodology, Investigation, Formal analysis. **Roman Giglio:** Writing – review & editing, Visualization, Validation, Methodology. **Daeyoung Kong:** Writing – review & editing, Validation, Methodology, Data curation. **Muhammad Shattique:**

Writing – review & editing, Validation, Methodology, Investigation, Conceptualization. **Hyoungsoo Lee:** Writing – review & editing, Supervision, Project administration. **James W. Palko:** Writing – review & editing, Writing – original draft, Supervision, Project administration, Investigation, Funding acquisition, Conceptualization. **Ercan M. Dede:** Writing – review & editing, Writing – original draft, Project administration, Investigation, Funding acquisition, Conceptualization. **Mehdi Asheghi:** Writing – review & editing, Writing – original draft, Supervision, Project administration, Investigation, Funding acquisition, Conceptualization. **Kenneth E. Goodson:** Writing – review & editing, Writing – original draft, Supervision, Project administration, Funding acquisition.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

This work was supported in part by the Advanced Research Projects Agency-Energy (ARPA-E) under agreement DE-AR0001055 (“Exploring the limits of cooling for extreme heat flux applications: data centers and power electronics”), and by the National Science Foundation (NSF) under Grant No. 2345655 within the Partnerships for Innovation (PFI) program. This work was also supported by the International Cooperative R&D Program of the Korea Institute of Energy Technology Evaluation and Planning (KETEP), funded by the Ministry of Climate, Energy & Environment, Republic of Korea (Grant No. RS-2024-00436521). Part of this work was performed at the Stanford Nanofabrication Facility (SNF) and the Stanford Nano Shared Facilities (SNSF), supported by the NSF under Award No. ECCS-2026822.

Appendix A. Supplementary data

Supplementary information to this article can be found online at <https://doi.org/10.1016/j.icheatmasstransfer.2026.112500>.

Data availability

No data was used for the research described in the article.

References

- [1] Cisco, Cisco Global Cloud Index: Forecast and Methodology, 2016–2021 White Paper, Cisco, 2018 document 1513879861264127.
- [2] J. Morley, K. Widdicks, M. Hazas, Digitalisation, energy and data demand: the impact of internet traffic on overall and peak electricity consumption, *Energy Res. Soc. Sci.* 38 (2018) 128–137.
- [3] A. Bar-Cohen, Gen 3 “embedded” cooling: key enabler for energy efficient data centers, *IEEE Trans. Compon. Packag. Manuf. Technol.* 7 (2017) 1206–1211.
- [4] ARPA-E, COOLERCHIPS. U.S. Advanced Research Projects Agency-Energy (ARPA-E), 2026.
- [5] N. Jones, How to stop data centres from gobbling up the world’s electricity, *Nature* 561 (2018) 163–166.
- [6] I. Mudawar, Assessment of high-heat-flux thermal management schemes, *IEEE Trans. Comp. Packag. Technol.* 24 (2001) 122–141.
- [7] I. Mudawar, Recent advances in high-flux, two-phase thermal management, *J. Therm. Sci. Eng. Appl.* 5 (2013) 021012.
- [8] Y. Nam, S. Sharratt, G. Cha, Y.S. Ju, Characterization and modeling of the heat transfer performance of nanostructured Cu micropost wicks, *J. Heat Transf.* 133 (2011) 101502.
- [9] D. Coso, V. Srinivasan, M.-C. Lu, J.-Y. Chang, A. Majumdar, Enhanced heat transfer in biporous wicks in the thin liquid film evaporation and boiling regimes, *J. Heat Transf.* 134 (2012) 101501.
- [10] S. Sudhakar, J.A. Weibel, F. Zhou, E.M. Dede, S.V. Garimella, Area-scalable high-heat-flux dissipation at low thermal resistance using a capillary-fed two-layer evaporator wick, *Int. J. Heat Mass Transf.* 135 (2019) 1346–1356.
- [11] C. Zhang, J.W. Palko, M.T. Barako, M. Asheghi, J.G. Santiago, K.E. Goodson, Enhanced capillary-fed boiling in copper inverse opals via template sintering, *Adv. Funct. Mater.* 28 (2018) 1803689.
- [12] C. Zhang, J.W. Palko, M.T. Barako, M. Asheghi, K.E. Goodson, Design and optimization of well-ordered microporous copper structure for high heat flux cooling applications, *Int. J. Heat Mass Transf.* 173 (2021) 121241.
- [13] J.W. Palko, C. Zhang, J.D. Wilbur, T.J. Dusseault, M. Asheghi, K.E. Goodson, et al., Approaching the limits of two-phase boiling heat transfer: high heat flux and low superheat, *Appl. Phys. Lett.* 107 (2015).
- [14] Nvidia, NVIDIA Hopper Architecture In Depth. <https://resources.nvidia.com/en-us-hpc-ai/nvidia-hopper-architecture>, 2026.
- [15] M. Ozkan, L. Pompa, M.S. Bin Iqbal, Y. Chan, D. Morales, Z. Chen, et al., Performance, efficiency, and cost analysis of wafer-scale AI accelerators vs. single-chip GPUs, *Device* 3 (2025).
- [16] R. Giglio, M.R. Shattique, E.M. Dede, S. Narumanchi, M. Asheghi, K.E. Goodson, et al., Porous mesh manifold for enhanced boiling performance, *Appl. Therm. Eng.* 260 (2025) 124918.
- [17] H. Kwon, Q. Wu, D. Kong, S. Hazra, K. Jiang, S. Narumanchi, et al., Capillary-enhanced two-phase micro-cooler using copper-inverse-opal wick with silicon microchannel manifold for high-heat-flux cooling application, *Int. Commun. Heat Mass Transfer* 156 (2024) 107592.
- [18] H. Kwon, D. Kong, H. Lee, J. Palko, E.M. Dede, M. Asheghi, et al., Scalable large-area two-phase capillary-enhanced micro-cooler using silicon microchannel fin array with 3D silicon manifold for high-heat-flux electronics cooling application, in: ASME 2024 International Technical Conference and Exhibition on Packaging and Integration of Electronic and Photonic Microsystems, 2024.
- [19] H. Kwon, Q. Wu, D. Kong, S. Hazra, K. Jiang, C. Ahn, et al., Development of a hybrid single/two-phase capillary-based micro-cooler using copper inverse opals wick with silicon 3D manifold for high-heat-flux cooling application, in: 2024 23rd IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm), Denver, CO, USA, 2024, pp. 1–10.
- [20] J.A. Weibel, S.V. Garimella, M.T. North, Characterization of evaporation and boiling from sintered powder wicks fed by capillary action, *Int. J. Heat Mass Transf.* 53 (2010) 4204–4215.
- [21] F. Zhou, J. Zhou, X. Li, Q. Chen, X. Huai, Enhanced capillary-driven thin film boiling on cost-effective gradient wire meshes for high-heat-flux applications, *Exp. Thermal Fluid Sci.* 149 (2023) 111018.
- [22] M.S. El-Genk, A.F. Ali, Enhanced nucleate boiling on copper micro-porous surfaces, *Int. J. Multiphase Flow* 36 (2010) 780–792.
- [23] D. Kong, R. Giglio, M.R. Shattique, Q. Wu, H. Kwon, J.W. Palko, et al., Development of a capillary-driven two-phase microcooler using copper wiremesh 3D manifold and silicon micropin fin wicks, *Int. J. Mech. Sci.* 305 (2025) 110781.
- [24] H. Kwon, R. Giglio, D. Kong, M. Shattique, H. Lee, J. Palko, et al., Capillary-driven two-phase cooler for high-heat-flux electronics using copper wire mesh manifold and enhanced copper inverse opal wick heat sink, *Appl. Therm. Eng.* 303 (2026) 131914.
- [25] D. Kong, H. Kwon, H. Lee, H. Lee, M. Asheghi, K. Goodson, Computational fluid dynamics modeling and optimization of large-scale (3 cm × 3 cm) silicon-based embedded microchannels with 3D manifold micro-coolers, *J. Electron. Packag.* (2025) 1–8.
- [26] M.R. Shattique, R. Giglio, E.M. Dede, S. Narumanchi, M. Asheghi, K.E. Goodson, et al., Permeability of single-layer-free-standing meshes at varying capillary pressure via a novel method, *Adv. Mater. Interfaces* 10 (2023) 2300326.
- [27] D.B. Tuckerman, R.F.W. Pease, High-performance heat sinking for VLSI, *IEEE Electron. Device Lett.* 2 (1981) 126–129.
- [28] K.W. Jung, C.R. Kharangate, H. Lee, J. Palko, F. Zhou, M. Asheghi, et al., Embedded cooling with 3D manifold for vehicle power electronics application: single-phase thermal-fluid performance, *Int. J. Heat Mass Transf.* 130 (2019) 1108–1119.
- [29] M.T. Barako, A. Sood, C. Zhang, J. Wang, T. Kodama, M. Asheghi, et al., Quasi-ballistic electronic thermal conduction in metal inverse opals, *Nano Lett.* 16 (2016) 2754–2761.