

Research Paper

An ultralight foam vapor chamber for mobile and aerial electronics cooling

Haodong Chen^a, Kiju Kang^{a,*}, Michael D. Atkins^{b,*}, Tongbeum Kim^c^a Department of Mechanical Engineering, Chonnam National University, Gwangju 61186, South Korea^b School of Mechanical, Industrial and Aeronautical Engineering, University of the Witwatersrand, Johannesburg 2000, South Africa^c College of Aerospace Engineering, Nanjing University of Aeronautics and Astronautics, Nanjing, Jiangsu 210016, PR China

ARTICLE INFO

Keywords:

Latent heat conversion
Lightweighting
Mobile and aerial electronics cooling
Thermal spreading
Foam vapor chamber

ABSTRACT

This study premieres a new type of vapor chamber that can achieve a remarkable ~80 % weight reduction over state-of-the art commercial vapor chambers (e.g., ~25 g, compared to ~120 g with similar exterior dimensions) potentially for mobile and aerial applications while providing comparable thermal spreading and junction temperature. This achievement has been made possible by incorporating a high porosity copper foam that is blackened for superhydrophilicity and encapsulated by an ultrathin-walled (~50 μm) envelope via electro/electroless plating – that is, an ultralight foam vapor chamber (uFVC). This article details its thermal absorption and spreading characteristics subjected to natural convection as well as the fabrication process. The uFVC spreads heat via latent heat conversion of an encapsulated deionized water. Blackened foam ligaments enhanced capillary forces to transport the deionized water within the ultrathin-walled envelope that further facilitates heat rejection to the surroundings. The achieved effective density of the uFVC as a measure of lightweighting is $\rho_{eff} \sim 1,000 \text{ kg/m}^3$ in sharp contrast to $\rho_{eff} \sim 9,000 \text{ kg/m}^3$ for a pure copper plate, $\rho_{eff} \sim 5,000 \text{ kg/m}^3$ for a state-of-the art commercial vapor chamber, with being even lighter than the latest developmental graphene-enhanced vapor chamber, $\rho_{eff} \sim 1,800 \text{ kg/m}^3$.

1. Introduction

To ensure reliable operation of electronic components that emit excessive heat, a variety of thermal management devices have been implemented. A heat pipe (HP) is a device that spreads the heat via latent heat conversion with subsequent rejection and is perhaps the most popular choice for cooling as a compact in-situ thermal management system. Fig. 1(a) shows a typical HP design that is tubular with a circular cross-section and equipped with various wick topologies and externally attached fins. The latent heat conversion between the liquid and vapor phases of a working fluid e.g., water, nitrogen, or sodium chambered within the tubular “envelope” spreads the excessive heat. Locally, the evaporator region is where vaporization of the working fluid takes place, and the end regions that are farther away from the heat source act as a condenser. Between these two regions, a structural medium is inserted called a wick, and the voided space is designed as vapor passages. The fluid and wick are enclosed by the envelope as a closed system. The envelope functions as a pressure vessel where internal pressure is initially set to be near vacuum. Internal pressure in turn increases from the vacuum state with the vaporization of the working fluid. The

external pressure is typically at an ambient condition. Therefore, a critical wall thickness is required to sustain the pressure differential between the inside and outside of the envelope.

The “ultrathin heat pipes (UTHPs)” were developed for compact thermal solutions, by “thinning” the heat pipe, where the external thickness is less than 2.0 mm [4,5]. UTHPs are implemented in smartphones with constrained space availability and thus fabricated either by flattening a tubular envelope of conventional HPs or by sandwiching a thin wick layer with two metallic sheets [6–8]. In both cases, the flattened or flat configuration increases vapor flow resistance, thus decreasing the capillary limit and heat spreading capacity [9,10]. To address these limitations different UTHP designs were proposed. For example, Cui et al. [11] developed a striped super-hydrophilic copper mesh wick with the total thickness of 0.68 mm, providing the thermal resistance of 0.26 K/W. In addition to flattening the HP, the thermal device is usually bent to follow a winding “S” shaped path as shown in Fig. 1(a(i)), to improve compactness and provide guided thermal spreading along a bent one-dimensional path [1–3].

A limitation of HPs relates to the case when multiple heat sources exist within a single device [12–15]. For example, in some smartphones multiple chips might emit excessive heat during operation. Thus, to cool

* Corresponding authors.

E-mail addresses: kjkang@chonnam.ac.kr (K. Kang), michael.atkins@wits.ac.za (M.D. Atkins).<https://doi.org/10.1016/j.applthermaleng.2025.128639>

Received 14 July 2025; Received in revised form 1 October 2025; Accepted 3 October 2025

Available online 9 October 2025

1359-4311/© 2025 The Author(s). Published by Elsevier Ltd. This is an open access article under the CC BY license (<http://creativecommons.org/licenses/by/4.0/>).

Nomenclature			
Symbols		V_e	External volume, m ³
A	Heat transfer area of vapor chamber, mm ²	W_e	Width of the foam sheet, mm
C_p	Specific heat capacity, J/(kg K)	$W_{envelope}$	Envelop weight, g
c_1	Thermal mass ($= C_p M$), J/K	W_{fluid}	Fluid weight, g
c_2	A dummy constant ($= Ah_e$), W/K	W_{total}	Total weight, g
d_{pore}	Pore size, μm	W_{wick}	Wick weight, g
H_e	Height of the foam sheet, mm	Greek symbols	
h_{nc}	Natural convective heat transfer coefficient, W/m ² K	δ	Envelope thickness, mm
h_{rad}	Radiative heat transfer coefficient, W/m ² K	ε	Porosity of the copper foam
h_e	Effective heat transfer coefficient including natural convection and radiation, W/m ² K	θ_∞	Difference between mean condenser surface temperature and ambient temperature in steady-state, °C
L_e	Length of the foam sheet, mm	θ	Contact angle, Deg
M	Mass, kg	ρ_{eff}	Effective density, kg/m ³
Q_{nc}	Heat loss from natural convection, W	Abbreviation	
Q_{in}	Power input, W	CP	Copper Plate
Q_{rad}	Heat loss from radiation, W	DI	Deionized
Q_s	Stored heat within the VC, W	HP	Heat Pipe
q''	Heat flux, kW/m ²	NCG	Non-Condensable Gas
T_{amb}	Ambient temperature, °C	uFVC	Ultralight Foam Vapor Chamber
T_{cm}	Average temperature on the top (or condenser) surface, °C	UTHPs	Ultrathin Heat Pipes
T_{IR}	Temperature values from both IR camera, °C	UTVCs	Ultrathin Vapor Chambers
T_j	Junction temperature, °C	VC	Vapor Chamber
T_{tc}	Temperature values from film thermocouples, °C	PMMA	Polymethyl-Methacrylate
T_∞	Steady-state temperature of a given medium, °C	PPI	Pores Per Inch
t	Timing, s	SEM	Scanning Electron Microscope

these chips a thin aluminum (or copper) plate was integrated into the smartphones' chassis. However, this solution may lead to increased overall weight and compromised thermal spreading performance.

To address the limitations of compact HP's, vapor chambers (VCs) are utilized due to their superb two-dimensional spreading capability (Fig. 1(b(i))) and reduced weight in comparison to aluminum and copper plates. VCs share their working principle with HPs as illustrated in Fig. 1(b(ii)). However, VCs are distinguished from HPs by unguided "planar" thermal spreading and substantially larger internal volume to store the working fluid. The planar surface on which a heat source is

attached, functions as an evaporator while the other planar surface is separated by a narrow gap and spacer (e.g., pins) to condense an encapsulated working fluid, simultaneously allowing the lateral spreading (or diffusion) of vapors [5]. The extended region of the condenser planar surface farther away from the attached heat source is also involved in the condensation of vapor.

Based on the review by Weibel and Garimella [16] of VC's, the developmental focus was to maximize heat load or heat flux by designing alternative wick structures. Ji et al. [17] adopted a 2 mm-thick copper foam wick, and Sudhakar et al. [18] used a 1.5 mm-thick sintered

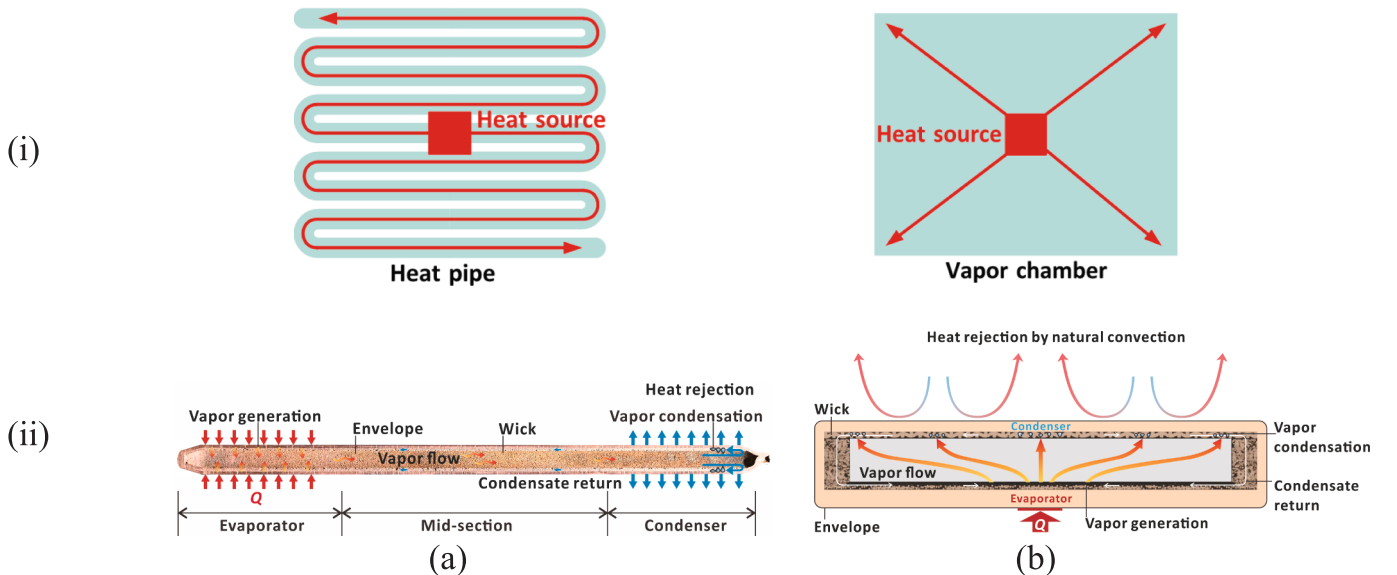


Fig. 1. Thermal spreading mode: (a) A heat pipe (HP) with (i) One-dimensional guided spreading along a winding "S" shaped heat pipe (HP) [1–3], and (ii) Heat rejection from condenser side of the HP, (b) A vapor chamber (VC) with (i) Two-dimensional unguided spreading and (ii) Heat rejection by natural convection.

powder wick to reduce condensate flow resistance. As an advanced wick design solution, bi-porous wicks were introduced for high capillarity with high permeability [19–21]. Multiple, thick lateral liquid-supply arteries were used by Hwang et al. [22] and Liu et al. [23] to feed a thin sintered-particle evaporator, sustaining high heat fluxes with delayed dry-out. Joshi et al. [24] designed a two-layer sintered-powder wick with top-down condensate feeding posts and vapor vents in a cap-layer wick and demonstrated a maximum heat flux of 451 W/cm^2 . Although these efforts could achieve relatively high heat fluxes (or heat loads), the thickness of their VCs became inevitably thick.

To meet the demands for “compact” electronics cooling, further research shifted focus towards separately thinning and lightweighting. Thus, ultrathin vapor chambers (UTVCs) with a thickness of less than 2 mm have been developed, offering geometric flexibility, temperature uniformity, and extremely high thermal conductivity [25–27]. Specifically, Luo et al. [28] employed six 1.5 mm thick UTVCs to reduce the in-plane temperature difference between the layers of a 5-cell fuel cell stack, resulting in a minimum temperature difference of 0.3 K on the cathode gas diffusion layer surface. Zhong et al. [29] fabricated a 1.5 mm thick aluminum UTVC with multiple-parallel microchannels for solar collectors, and their thermal resistance decreased by 74 % compared to an aluminum plate, achieving about 0.22 K/W under natural air convection. Huang et al. [30] designed a 1.53 mm thick UTVC for miniaturized electronics cooling, which could dissipate 50 W with a thermal resistance of 0.197 K/W: about 4.4 times lower than that of a 1.5 mm copper sheet. Their subsequent study [31] presented a 0.5 mm thick UTVC with four spirally woven meshes and a composite wick, which provided a maximum effective thermal conductivity of about 20,900 W/(mK): approximately 52 times higher than that of pure copper. However, as noted by Huang et al. [32], the reduced UTVC thickness decreases the maximum spreading, which is particularly pronounced when the thickness of an inner cavity becomes less than 0.3 mm, resulting in a sharp increase in the total thermal resistance [33].

For mobile and aerial applications, weight reduction can be a crucial design constraint together with thinning. Yet, research to investigate the lightweighting of VCs was largely overlooked in the literature. To the best of the authors’ knowledge, the first attempt was made by employing a less dense (lighter) material i.e., titanium than copper [34]. Jiang et al. [35] and Bulut et al. [36] reported that lightweight heat transfer devices are necessary for the thermal management of electronics devices in the aerospace and telecommunications industries. Present research on manufacturing lightweight vapor chambers focused on the construction of the thermal devices from high strength-to-weight materials such e.g., aluminium and titanium alloys, magnesium, aluminium-magnesium alloy, aluminium nitride, silicon carbide instead of the common materials like copper and stainless steel. Recently, Enmark et al. [37] proposed an extremely light VC, which was fabricated using a graphene-assembled film as an envelope. Commonly for capillary-driven HPs and VCs, both wick structure and envelope account for more than 90 % of their total weight [38,39]. The thickness of the graphene envelope was 200 μm , which was strong enough to withstand pressure forces exerted internally and externally. The inner structure consisted of a copper mesh acting as wick and 1.0 mm diameter copper spheres as spacers with 0.5 ml of ethanol as a working fluid. In their design, the total thermal resistance of 1.21 K/W was claimed to be achievable with a water-cooled heat sink attached, which is 21.6 % lower than an unmodified VC as reference. Three VC prototypes were tested with dimensions of 50 mm $\times$ 50 mm $\times$ (1.9 mm / 2.1 mm / 2.7 mm as thickness) and the net weight of 9.04 g / 10.41 g / 11.83 g, respectively. Therefore, the effective density was measured to vary from 1,752.6 kg/m³ to 1,982.9 kg/m³, which is substantially lighter than commercial state-of-the-art VCs ($\sim 5,000 \text{ kg/m}^3$). In addition, VCs were made “ultrathin” to reduce weight and ensure compact form factor. A technical challenge to lightweighting VCs is that the external structure is subjected to significant internal and external differential pressure, depending on whether the device is idling (i.e., no heat input) or in

operation with thermal spreading. Jiang et al. [35] described that as the thickness of the VC shell plate decreases towards an extremely thin size, the vapour channel then becomes susceptible to collapse, and plastic deformation that leads to increased vapour flow resistance. Conventional VCs, therefore, rely on increasing the wall thickness of the envelope and the placement of vertical columns to mechanically support the walls to withstand buckling and yielding loads that cause the VC wall to collapse [35,40,41]. These conflicting requirements, together with the use of copper as the typical VC construction material increases the overall weight of the VC.

This study aims to further lightweight VCs. To this end, we have applied a groundbreaking fabrication process that was recently introduced for heat pipes by our group [38,39]. Here, electro/electroless plating was implemented to simultaneously lightweight both the wick and envelope. We have thus achieved remarkable weight reduction compared to commercial VCs while providing comparable thermal spreading and junction temperature (T_j). This achievement incorporates a high porosity copper foam with blackening for superhydrophilicity and encapsulated by an ultrathin-walled ($\sim 50 \mu\text{m}$) envelope via electroless plating – that is, an ultralight foam vapor chamber (uFVC) whose effective density is $\sim 1,000 \text{ kg/m}^3$, even lighter than the latest graphene-enhanced VCs [37]. This study details the fabrication process and thermal absorption and spreading characteristics of the uFVC, where the condenser surfaces are subjected to natural convection.

2. Materials and methods

2.1. Fabrication and chemical surface treatment

We have developed a novel lightweighted VC, which achieves substantially lower effective density compared to state-of-the-art commercial, and developmental VCs [37]. To this end, the present study has encapsulated a monolithic, foam core structure in an ultrathin-walled envelope. For this purpose, we have adopted a fabrication process that begins with preparing a metal-polymer template as illustrated in Fig. 2 (a). Firstly, an open-cell Cu foam block with 60 pores per inch (PPI) was cut by an electric discharge machine according to a pre-determined shape of a VC – a 3.0 mm thick square sheet ($W_e = 90 \text{ mm} \times L_e = 90 \text{ mm}$) and was dipped in a pool of a cold-mounting resin (Fig. 2(a)). Here, W_e and L_e are the external width and length of the foam sheet. The porosity of the foam was measured to be $\varepsilon = 0.975$ [38,39].

The cold-mounting resin was a polymethyl-methacrylate (PMMA) resin that was polymerized under a pressure of 7 bar at room temperature for the suppression of bubble formation during curing. After which, the exterior surfaces of the block of a foam-PMMA specimen were machined and polished, providing clear bonded interfaces between the foam ligaments and the void filling PMMA.

Electroless plating and electroplating were subsequently implemented to deposit a thin copper layer that encapsulated the foam-PMMA specimen as an envelope of our final VC. The copper layer metallurgically joined the foam ligaments’ cross-sections that were cut flush with the specimen’s exterior surfaces. The cut individual foam pores on the exterior surfaces provide mechanical support for the deposited layer for compressive load during idling and tensile load during the vaporization of a working fluid. The fabrication process is illustrated in Fig. 2(b) and detailed as follows.

Firstly, electroless plating deposited a thin Cu layer at a pre-determined rate (of 1.5 $\mu\text{m}/\text{hour}$) on the polished surfaces of the foam-PMMA specimen. As a result, the metallurgical joints between the deposited Cu layer and the cut foam cells flush with the exterior surfaces were established. Secondly, the Ni-P coating via electroless plating was implemented to prevent the external surfaces of the Cu layer (envelope) from undergoing any chemical reaction during blackening [42].

To realize internal open-cell foam structures, the PMMA needed to be removed from the foam-PMMA composite that was enveloped by a thin Cu layer. To this end, the four corners of the foam-PMMA composite

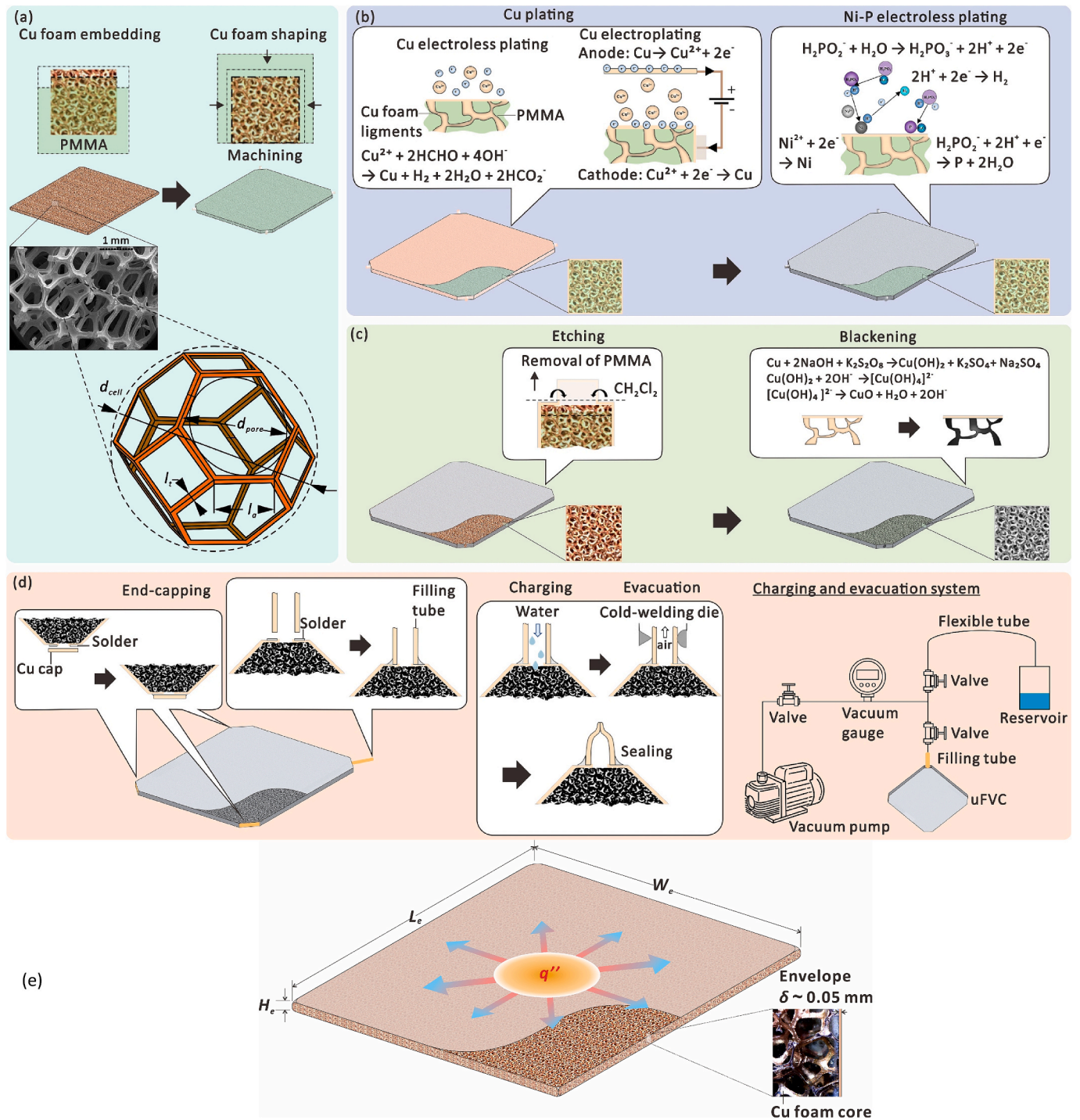


Fig. 2. Fabrication process of ultralight foam vapor chamber (uFVC). (a) Open-cell copper (Cu) foam with 60 pores per inch (PPI) with a tetrakaidekahedron unit cell, PMMA embedding and shaping, (b) Enveloping via copper plating and Ni-P electroless plating, (c) PMMA etching and blackening, (d) End-capping, DI water filling and sealing, and (e) A complete uFVC with $W_e = 90$ mm, $L_e = 90$ mm, $H_e = 3.0$ mm, $\delta \sim 0.05$ mm.

were cut to facilitate the etching of the PMMA in a dichloromethane filled pool for six days at room temperature as illustrated in Fig. 2(c). As a result, only the foam ligament structures and thin Cu layer with the joints between the Cu layer and the outermost foam cells flush bonded with the inner surface of the Cu layer remained. After which, the surfaces underwent a series of chemical treatments. Firstly, the specimen was dipped into an acid agent (5 wt% HCL) that removed copper oxide layers on the foam ligament surfaces and the inner surfaces of the envelope. Secondly, another treatment – the so-called “blackening” was implemented for super-hydrophilicity [43] of the foam wick as

illustrated in Fig. 2(c) by a blackening solution (7.5 g/L NaOH and 1.5 g/L $K_2S_2O_8$) mixture [44]. It should be pointed out that condensate is pumped back to the evaporator regions of the vapor chamber via the capillary action of the wick. Therefore, super-hydrophilicity of the wick provides improved pumping of condensate so that the dry-out can be prevented [38,39].

The specimen was next prepared for charging, evacuation, and sealing, as shown in Fig. 2(d). To make the specimen able to be fully functional, one corner was soldered with a filling tube while the other three corners were soldered with a copper cap. A custom-built

evacuation and charging system were constructed to fabricate the uFVC. The envelope encapsulating the Cu foam core was connected to the system via a filling tube. A two-stage oil-sealed rotary vane vacuum pump was used for evacuation. A digital vacuum gauge monitored the pressure inside the envelope during evacuation. A reservoir of DI water as the working fluid due to the good compatibility with the Cu foam and Cu envelope at the target working temperature ranging from 30 °C to 200 °C [45], was connected to the system with another flexible tube. In addition, to control the amount of working fluid injected into the VC, three bellow-sealed valves were used to switch the mode between evacuation and charging. The reservoir was heated and subsequently frozen by liquid nitrogen to prevent or minimize non-condensable gas (NCG) from being dissolved back in the DI water [46]. Subsequently, the pre-cleaned VC envelope was connected to the reservoir and evacuated at an absolute pressure of 2.1 Pa. The uFVC specimen was charged with working fluid via the reservoir due to the pressure difference and gravity [46]. The filling tube was sealed using a cold-welding die during evacuation. Fig. 2(e) presents our uFVC completed with an ultrathin-walled envelope having dimensions with $W_e = 90$ mm, $L_e = 90$ mm, $H_e = 3.0$ mm, $\delta \sim 0.05$ mm. In this study, the envelope thickness was chosen to be ~ 0.05 mm, based on our prior estimate of the wall buckling mode using classical theories [38].

2.2. Microstructures and wettability

Fig. 3(a) shows the anatomy of our uFVC with an ultrathin-walled envelope. Microstructures of an un-blackened inner surface of the uFVC due to an attached foam ligament (upper circle) and a blackened inner surface due to the direct exposure (lower circle) of the electroless plated envelope are shown in Fig. 3(b).

Fig. 3(c, d) are scanning electron microscope (SEM) images taken for the uFVC that was sacrificed to enable an investigation of the surface condition with and without being blackened. The blackened surfaces exhibit a nano-porous texture as shown by SEM images in Fig. 3(c, d).

A raw copper wick fabricated via electroless plating prior to blackening is hydrophobic [43]. For example, sequential images by a high-speed camera in Fig. 4(a) show that a drop of deionized (DI) water falling onto the un-blackened wick layer, clearly exhibits hydrophobicity, since the drop free falls normal to the surface, impacts the

surface, and then bounces back, but immediately returns to sit on the *un-wetted* wick surface where the contact angle was estimated to be $\theta \approx 72^\circ$. On the other hand, after blackening (Fig. 4(b)), the free-falling water drop was absorbed by and spread within the porous wick layer in its immediate contact due to superhydrophilicity ($\theta \rightarrow 0^\circ$).

2.3. Heat transfer measurements

The present study aimed to compare the thermal performance of the novel uFVC relative to a commercial VC and copper plate. Here, the heat and heat flux inputs were slightly higher than those used in Chang et al. [47] and Yu et al. [40], who investigated the thermal performance of UTVCs, with similar exterior dimensions to the specimens used in this research. Chang et al. [47] implemented a heater with 10 mm $\times$ 10 mm square area sandwiched into a 20 mm $\times$ 20 mm copper block, with heating power of up to 9.5 W, thus the heat flux was 2.375 W/cm² onto an UTVC specimen. Similarly, Yu et al. [40], applied a maximum heating power of 6.0 W using a heating unit of 15 mm $\times$ 15 mm in size (i.e., heat flux was 2.67 W/cm²).

Thermal characterization of test specimens used in this study were carried out using a purposely built test rig illustrated in Fig. 5(a). The test rig consists of a test specimen, a film heater that simulated the external heat source or electronic chip (Q_{in}) attached on the bottom central surface, a direct current power supply, an infrared (IR) camera, K-type film thermocouple, and a data acquisition unit. A chosen specimen was horizontally mounted on the top face of a copper heating base block, with a heating area of 11 mm $\times$ 12 mm and thickness of 2 mm, and an input power regulated for $Q_{in} = 10.0$ W, fixed by double-sided tape, thus the applied heat flux (i.e., ~ 7.58 W/cm²). The film heater was attached to the bottom surface of the Cu heating block by adhesive tape. Its outer side was tight fitted to a machined thermal insulation block (Formlabs, FLGPCL04) as illustrated in Fig. 5(a). Another K-type thermocouple was used to measure the ambient temperature.

To evaluate thermal spreading, the temperature distribution (i.e., $T_c(x, y)$) on the top surface of a given specimen subjected to natural convection cooling was measured using an IR camera (FLIRTM E5). Prior to measurement, the IR camera was calibrated against data obtained from film thermocouples attached at selected locations. Temperature values from both IR camera and film thermocouples were correlated linearly as

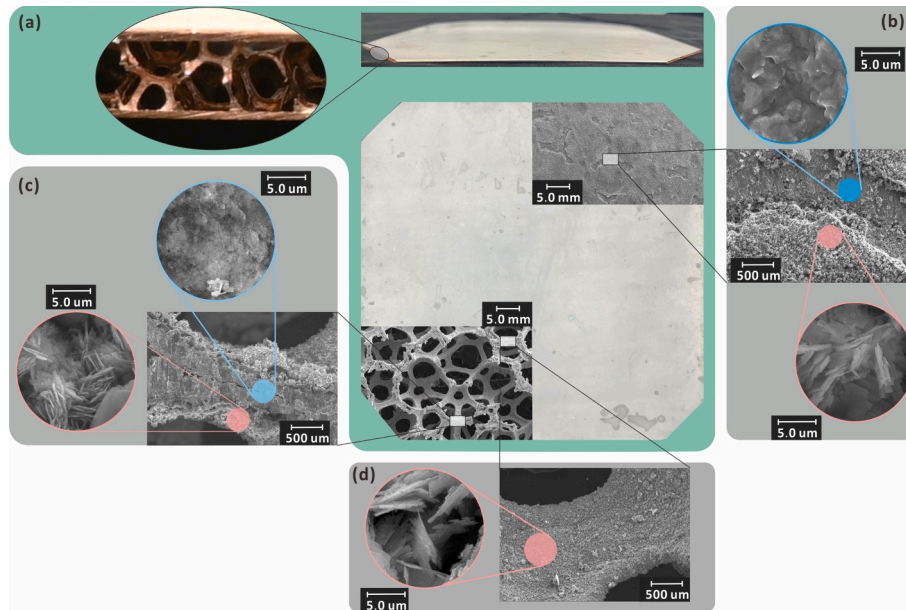


Fig. 3. Microstructures of a uFVC. (a) A thin flat VC with ultrathin-walled envelope, (b) An un-blackened inner surface due to an attached foam ligament (upper circle) and a blackened inner surface due to a direct exposure (lower circle), of the electroless plated envelope, (c) a blackened ligament's vertex due to a direct exposure (upper circle) and an un-blackened outer foam pore's ligament due to an attached envelope's surface (left circle), and (d) a blackened foam ligaments.

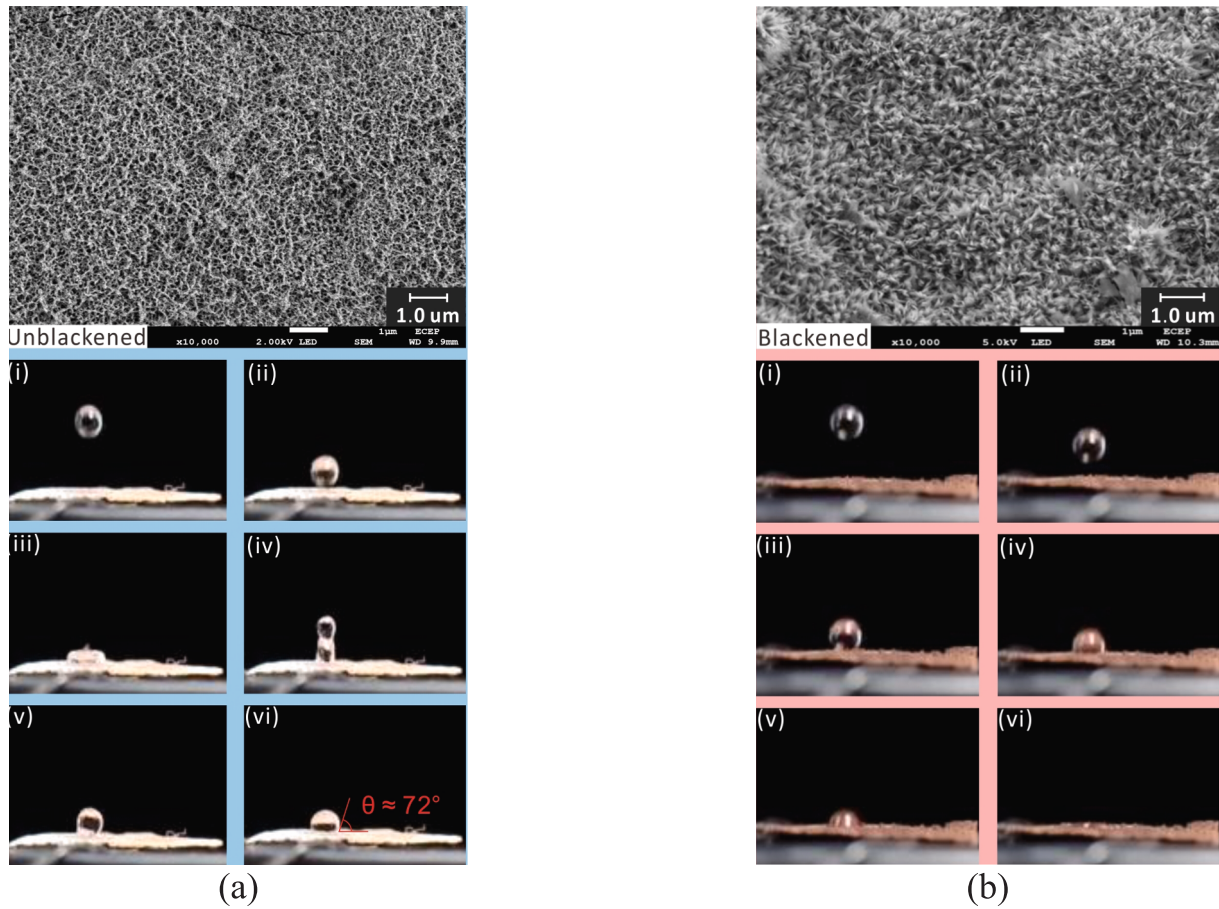


Fig. 4. Wettability of electroless plated surfaces as a wicking medium. (a) Hydrophobic droplet formation on un-blackened raw copper wick with the contact angle measured to be $\theta \approx 72^\circ$, and (b) Superhydrophilicity ($\theta \rightarrow 0^\circ$) of a porous wick layer after blackening.

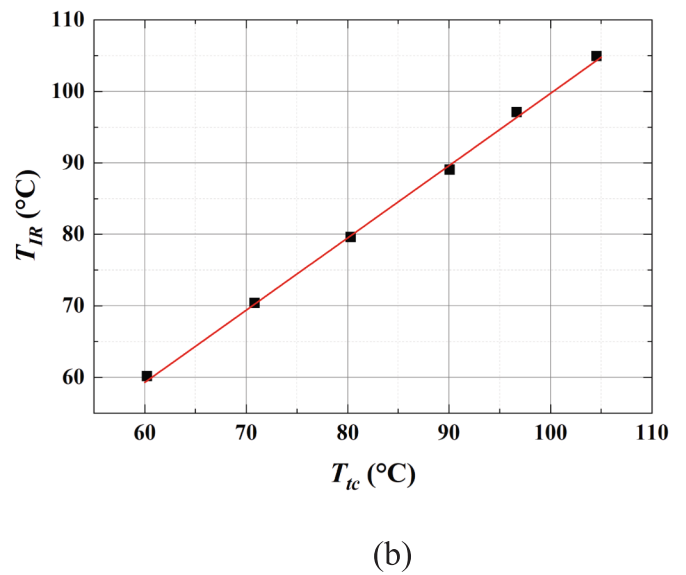
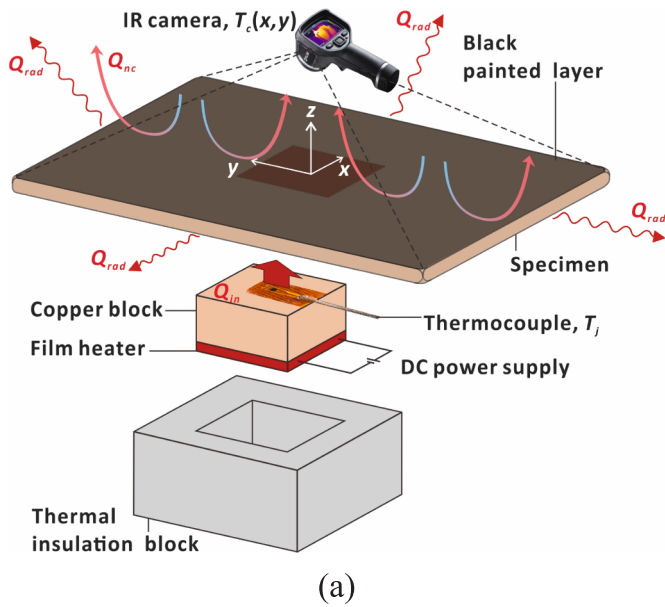


Fig. 5. Heat transfer test setup: (a) Schematic and (b) IR camera calibration against a K-type film thermocouple (T_{tc} vs. T_{IR}).

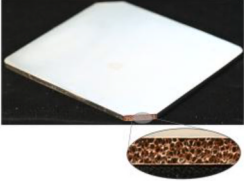
$T_{tc} = 1.01T_{IR} - 1.29$ (°C) for a wide range of heat inputs to the film heater (Fig. 5(b)).

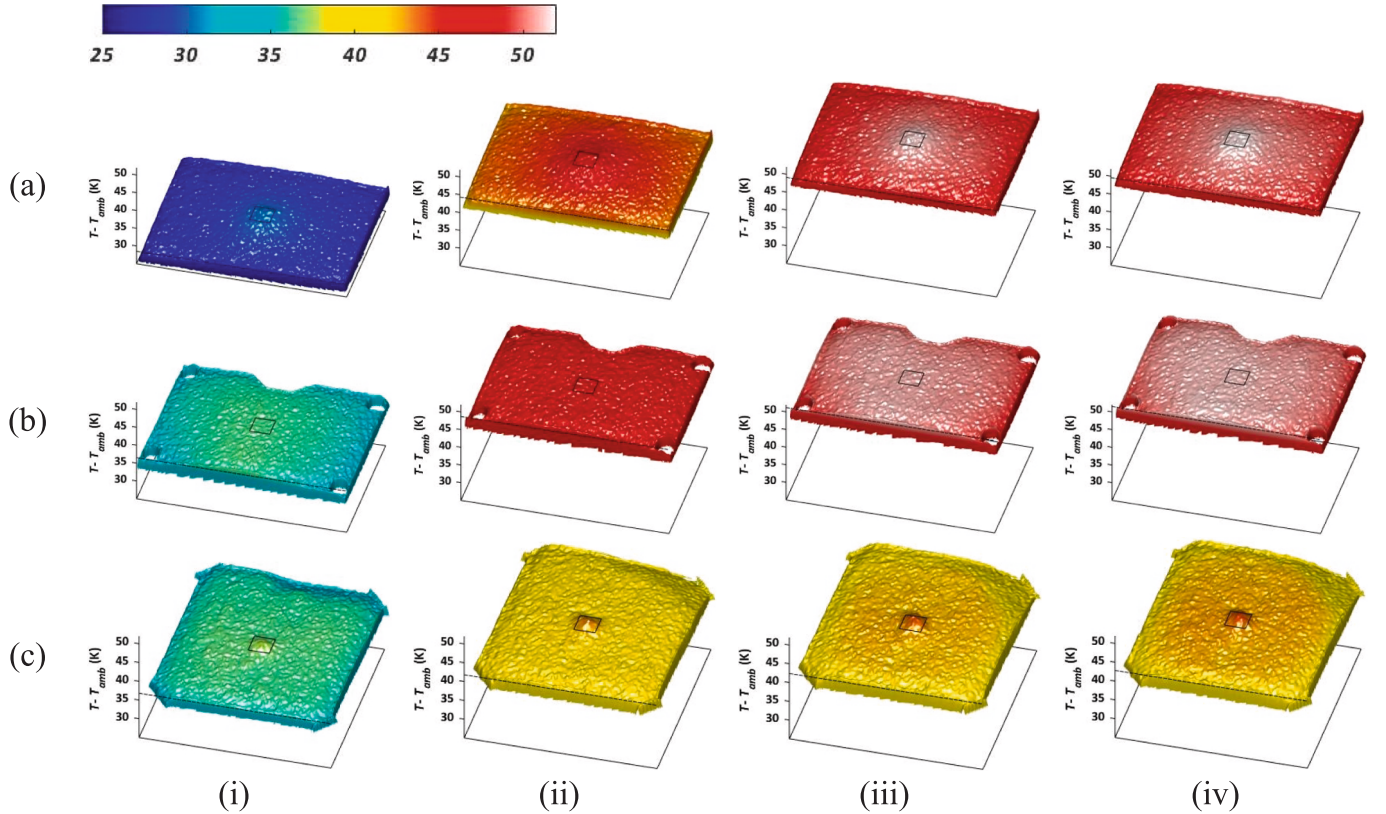
Three specimens were chosen: a copper plate (CP) as reference, a commercial VC, and our uFVC. Geometric parameters and material

properties of these three specimens are detailed in Table 1. In this setup, when the central square region of the specimen's bottom surface (indicated in Fig. 6(a)) is heated, vapor is generated, functioning as an evaporator for both VC and uFVC, where convection in the x-y plane is

Table 1

Geometric parameters and material properties of the copper plate (CP), commercial vapor chamber (VC) and ultralight vapor chamber (uFVC) tested.

Specimen	CP	Commercial VC	uFVC
			
Width, W_e (mm)	70	70	90
Length, L_e (mm)	106	106	90
External thickness, H_e (mm)	3	3	3
External volume, $V_e (=W_e \times L_e \times H_e)$ (m ³)	$\sim 22.26 \times 10^{-6}$	$\sim 22.26 \times 10^{-6}$	$\sim 24.3 \times 10^{-6}$
Envelope	N/A	Material: Pure copper Evaporator side wall thickness: 1.0 mm Condenser side wall thickness: 570 μm	Material: Ni-P coated copper Wall thickness: $\delta \sim 50 \mu\text{m}$
Wick	N/A	Material: Sintered copper mesh Evaporator side thickness: 122 μm with 61 μm diameter beads Condenser side thickness: 200 μm with 10 μm diameter beads	Material: Blackened 60 PPI copper foam Pore size: $d_{\text{pore}} \sim 423 \mu\text{m}$ Porosity: $\varepsilon \sim 0.965$ Blackened mean particle size $\sim 5 \mu\text{m}$
Working fluid	N/A	DI water $W_{\text{fluid}}: \sim 1.2 \text{ g}$ Filling ratio (FR): $\sim 12 \%$	DI water $W_{\text{fluid}}: \sim 3.63 \text{ g}$ Filling ratio (FR): $\sim 15 \%$
Weight and proportion	$W_{\text{total}} = 195.86 \text{ g}$	$W_{\text{total}}: 121.43 \text{ g}$ Proportion: $W_{(\text{wick} + \text{envelope})}/W_{\text{total}}: 0.991$ $W_{\text{fluid}}/W_{\text{total}}: 0.009$	$W_{\text{total}}: 25.43 \text{ g}$ including a soldering agent & filling tube, $\sim 8.7 \text{ g}$ Proportion: $W_{(\text{wick} + \text{envelope})}/W_{\text{total}}: 0.78$ $W_{\text{fluid}}/W_{\text{total}}: 0.22$
Effective density, $\rho_{\text{eff}} (=W_{\text{total}}/V_e)$	$\sim 8,887 \text{ kg/m}^3$	$\sim 5,455 \text{ kg/m}^3$	$\sim 1,046 \text{ kg/m}^3$

**Fig. 6.** Temperature contours on the upper (condenser) surface measured by a pre-calibrated IR camera at (i) $t = 420 \text{ s}$, (ii) $t = 1,100 \text{ s}$, (iii) $t = 2,000 \text{ s}$, and (iv) $t = 2,620 \text{ s}$ with $Q_{\text{in}} = 10 \text{ W}$ ($q'' \sim 7.58 \text{ W/cm}^2$): (a) CP, (b) Commercial VC and (c) uFVC.

evaluated. Furthermore, thermal absorption and rejection were analyzed using the classical first-order unsteady-state energy balance whose details will be discussed in III. Results and Discussion. Thermal absorption and rejection are a measure of how a vapor chamber responds to a thermal surge, where the temperature rise is characterized after a step heat input is imposed on the VC. In this analysis, the initial rate of temperature rise is important, and the steady-state operation temperature is crucial during idling and typical continued operations. While a VC absorbs heat Q_{in} which is emitted by an attached electronic chip, heat is stored within the VC (Q_s) and simultaneously dissipated via natural convection (Q_{nc}) and radiation (Q_{rad}) from the external surfaces as shown in Fig. 5(a). Thus, for a given heat input, the central film heater temperature – junction temperature (T_j) and surface average temperature (i.e., T_{cm}) on the top (or condenser) surface that was measured by the IR camera (i.e., $T_c(x, y)$) were synchronized to give $(T_j - T_{amb})$ and $(T_{cm} - T_{amb})$ at any given moment (t).

Measurement uncertainties were estimated using a method detailed by Coleman and Steele [48], as follows. If $y = f(x_1, x_2, \dots, x_n)$, then the uncertainty propagated by x_i in the variable y is given by:

$$\Delta y = \sqrt{\left(\frac{\partial y}{\partial x_1} \Delta x_1\right)^2 + \left(\frac{\partial y}{\partial x_2} \Delta x_2\right)^2 + \dots + \left(\frac{\partial y}{\partial x_n} \Delta x_n\right)^2} \quad (1)$$

where Δx_i is the absolute uncertainty in x_i . Equation (3) was used to estimate the uncertainty of surface temperatures on both the evaporator and condenser (i.e., T_j and T_c) whose accuracy and resolution are $\pm 2.0\%$ and 0.06°C , respectively.

3. Results and discussion

3.1. Thermal spreading

A unique feature that distinguishes a VC from a HP is perhaps multi-dimensionality in thermal spreading via latent heat conversion (between liquid and vapor) of a working fluid. Both external and internal volumes of a commercial VC are typically one order of magnitude larger than a commercial HP (e.g., $\sim 1.0 \times 10^{-5} \text{ m}^3$) for VCs (see, Table 1) vs. $\sim 1.0 \times 10^{-6} \text{ m}^3$ for HPs with a proportionally more working fluid contained [39]. For generic cylindrical HPs typically with a circular cross-section, thermal spreading takes place only along their span – that is, one-dimensional spreading. In sharp contrast, thermal spreading in VCs can occur either two or three-dimensionally, depending on their thickness relative to lateral dimensions.

For a thin VC i.e., $H_e \ll (W_e \text{ and } L_e)$ (see, Table 1), the thickness-wise spreading may be insignificant in comparison to that in the lateral directions: along the VC's surfaces perpendicular to the imposed heat. On the other hand, the spreading could be three-dimensional if the VC is sufficiently thick compared to its lateral dimensions i.e., $H_e \sim (W_e \text{ and } L_e)$. However, it entirely depends on the individual VC's internal structures. For both cases, the spread heat is subsequently rejected to the surroundings by either natural convection or forced convection e.g., a fan and fins.

The present evaluation was carried out by a series of transient measurements under natural convection for three specimens. Fig. 6(a) shows the temporal variation of CP's upper surface temperatures (T_c) that were mapped by the pre-calibrated IR camera at $Q_{in} = 10.0 \text{ W}$ emitted by a film heater through the heating area of $11 \text{ mm} \times 12 \text{ mm}$ (i.e., $q'' \sim 7.58 \text{ W/cm}^2$). Four instances were selected for discussion: (i) $t = 420 \text{ s}$, (ii) $t = 1,100 \text{ s}$, (iii) $t = 2,000 \text{ s}$, and (iv) $t = 2,620 \text{ s}$. The surface temperature relative to the ambient temperature (i.e., $T - T_{amb}$) increases as time elapsed (i) $\rightarrow$ (iii) but there is no change from (iii) to (iv). Locally, a slightly higher temperature region at the center of the CP is seen (i.e., iii to iv), which is coincident with the heat source attached to the lower (bottom) surface as indicated by a square box on each contour plot.

Next, we consider the temperature ($T - T_{amb}$) contours for the

commercial VC in Fig. 6(b) at the same four selected temporal instances. Similarly, the temperature increases as time elapsed (i) $\rightarrow$ (iii) with no change from (iii) to (iv). For the first two instances i.e., (i) $t = 420 \text{ s}$ and (ii) $t = 1,100 \text{ s}$, the upper (condenser-side) surface temperatures are higher than those on the CP. However, at (iii) $t = 2,000 \text{ s}$ and (iv) $t = 2,620 \text{ s}$, they have comparable values with the CP. A unique feature is that the surface temperature is fairly uniform without showing a central thermal peak unlike the foregoing case.

Lastly, thermal spreading in the uFVC is shown in Fig. 6(c). The surface temperature relative to the ambient temperature i.e., $T - T_{amb}$ increases as time elapsed (i) $\rightarrow$ (iii) but no change occurs from (iii) to (iv). Similar to the case of the CP, there is a thermal peak at the center. For the first instance i.e., (i) $t = 420 \text{ s}$, the distributed surface temperatures are higher than those on both CP and VC but at (ii) $t = 1,100 \text{ s}$, (iii) $t = 2,000 \text{ s}$ and (iv) $t = 2,620 \text{ s}$, they turn to become lower than the two foregoing cases.

In all three cases considered thus far, the steady-state operation appears to establish for (iii) and (iv) i.e., $t > 2,000 \text{ s}$. For a direct comparison, the contours of the three specimens in Fig. 6 were extracted along the length of each specimen i.e., the x -axis. Fig. 7 shows the local peak at the central region (i.e., solid line) of the CP followed by a gradual decrease towards both edges. The area-averaged surface temperature (i.e., dashed line) is $T_{cm} - T_{amb} = 50.2^\circ\text{C}$ with its maximum local deviation of $T_c - T_{cm} \sim 1.5^\circ\text{C}$ occurring at the center. The imposed heat on the bottom center ($0.435 \leq x/L_e \leq 0.565$) is conducted through the 3-mm thick Cu plate and reaches the upper surface while being spread laterally, exhibiting the observed distribution.

On the VC, however, there is no apparent peak that coincides with the central region. The distribution is rather uniform: a gradual decrease from the center to both edges. This distribution may imply that the concentrated heat input from the bottom center ($0.435 \leq x/L_e \leq 0.565$) is discontinued by the void space between bottom and upper substrates (envelope walls). Further, circular pillar (or pin) inserts that are discretely distributed play an insignificant part in spreading the heat to the upper substrate, and vapor flow's thickness-wise thermal transport is ineffective unlike its effective lateral spreading. The area-averaged surface temperature is $T_{cm} - T_{amb} = 50.7^\circ\text{C}$ with its maximum local deviation of $T_c - T_{cm} \sim 0.6^\circ\text{C}$ at the center.

With the same heat imposed on the uFVC i.e., $Q_{in} = 10.0 \text{ W}$, the uFVC's upper surface temperature is substantially lower, e.g., $\sim 16\%$ lower than both CP and VC: $T_{cm} - T_{amb} = 42.5^\circ\text{C}$ with its maximum local deviation of $T_c - T_{cm} \sim 2.0^\circ\text{C}$ at the center. A thermal peak is apparent for the uFVC specimen, which coincides with the heat source. The

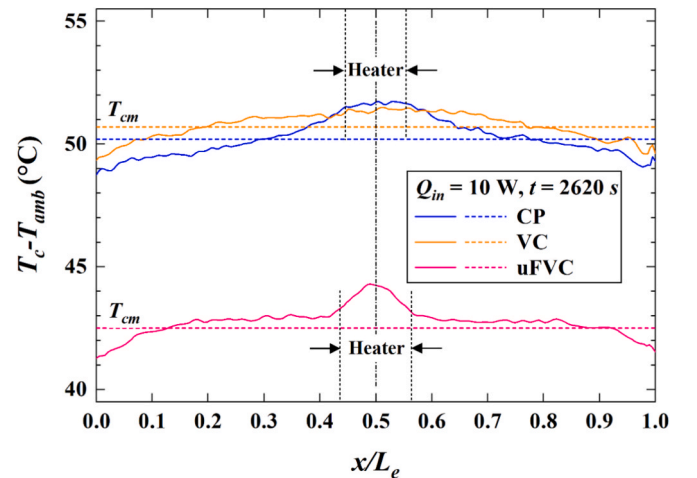


Fig. 7. Comparison of temperature distributions along the length of each specimen's condenser (upper) surface at $t = 2,620 \text{ s}$, representing the steady-state operation with $Q_{in} = 10 \text{ W}$. Here, the solid line shows the local surface temperature variation and dashed line indicates the area-averaged temperature.

presence of this central peak indicates that heat transfer through the uFVC's 3-mm thickness is substantial via both integral vertical foam ligaments and end-wall and rising vapor flow. Away from the peak, the temperature distribution over the condenser surface is substantially uniform due to thermal spreading. The temperature drop occurs from $x/L_e = 0.1$ and $x/L_e = 0.9$ to both edges.

3.2. Thermal absorption

The thermal absorption property of a given medium can be characterized based on its transient response. To this end, additional area-averaged data points on the condenser surface at different instances were obtained from $t = 0.0$ s to $t = 2,700$ s as plotted in Fig. 8 together with the temporal variation of junction (or evaporator) temperatures, T_j for the three specimens. Immediately after the thermal load was imposed, both junction temperatures i.e., $T_j - T_{amb}$ at $x = 0, y = 0, z = 0$, and area-averaged condenser temperature i.e., $T_{cm} - T_{amb}$ increased, and approached respective steady-state values. The junction temperatures are much higher than those on the condenser surfaces. Thermal absorption is measured by determining how much heat from the input heat imposed on the evaporator is stored by each VC specimen.

Referring to the junction temperature, T_j that was measured by a film-type thermocouple inserted between the heater and the bottom central surface of a given specimen. Firstly, the CP shows the most gradual increase among the three specimens and eventually approaches $T_j - T_{amb} \sim 89^\circ\text{C}$ roughly after $t = 2,000$ s. Secondly, the commercial VC's junction temperature increases steeper than the CP. There is a cross-over at $t = 1,100$ s, after which, the junction temperature becomes lower than the CP, approaching $T_j - T_{amb} \sim 84^\circ\text{C}$. Lastly, the junction temperature of the uFVC rises more steeply than the other two specimens. However, the uFVC exhibits a cross-over (i.e., at $t = 1,500$ s) in relation to the CP data, and thereafter reaches a lower steady-state temperature (i.e., $T_j - T_{amb} \sim 85^\circ\text{C}$).

The initial rate of the temperature rise at the junction and over the condenser surface indicates thermal absorption. The gradient of the initial rise is compared to be $\text{CP} < \text{VC} < \text{uFVC}$, while approaching $T_{cm} - T_{amb} = 50^\circ\text{C}$, 51°C , and 42.5°C for the CP, VC, and uFVC, respectively.

Parameters that govern the initial temperature rise and steady-state operation temperature were quantified by the classical first-order unsteady-state analysis. While a medium (CP, VC, or uFVC) absorbs the imposed heat, Q_{in} is emitted by the attached heat source, the heat is stored within the medium (Q_s) and simultaneously dissipated by

external cooling via natural convection (Q_{nc}) and radiation (Q_{rad}) over the surfaces, as indicated in Fig. 5(a). The energy balance in a given medium (CP, VC and uFVC) may be expressed as,

$$Q_{in} = Q_s + Q_{nc} + Q_{rad} \quad (2)$$

where

$$Q_s = C_p M \frac{dT}{dt} \quad (3)$$

$$Q_{nc} + Q_r = A(h_{nc} + h_{rad})(T(t) - T_{amb}) \quad (4)$$

here, A is the heat transfer area of a given vapor chamber, h_{nc} is the convective heat transfer coefficient (by natural convection), h_{rad} is the radiative heat transfer coefficient, T is the representative temperature of a given medium, which varies with time (t), and T_{amb} is the ambient temperature, which is invariant with time.

For convenience, let

$$\theta(t) = T(t) - T_{amb} \quad (5)$$

$$c_1 = C_p M \quad (6)$$

and

$$c_2 = A(h_{nc} + h_{rad}) \quad (7)$$

Since the present analysis aims particularly to identify the governing parameters that characterize the transient thermal behavior, the terms for c_2 in Eq. (7) are lumped into effective values, i.e. $c_2 = Ah_e$ where Ah_e may be estimated in the steady state as,

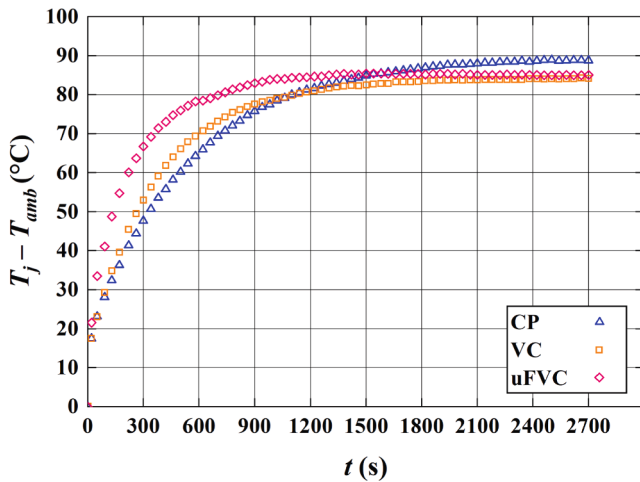
$$c_2 = Ah_e = \frac{Q_{in}}{T_{\infty} - T_{amb}} = \frac{Q_{in}}{\theta_{\infty}}, c_2 = A(h_{nc} + h_{rad}) \quad (8)$$

Finally, the energy balance in the vapor chamber (Eq. (8)) is in the simple form,

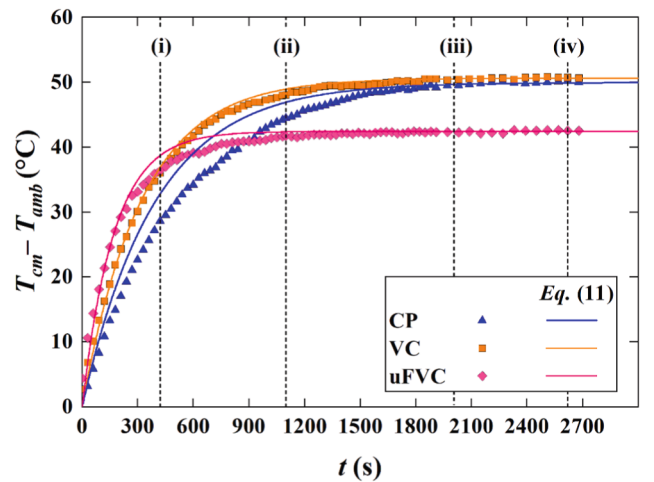
$$Q_{in} = c_1 \frac{d\theta}{dt} + c_2 \theta \quad (9)$$

The general solution of Eq. (9) is,

$$\theta(t) = Ce^{-\left(\frac{c_2}{c_1}\right)t} + \frac{Q_{in}}{c_2} \quad (10)$$



(a)



(b)

Fig. 8. Transient and steady thermal absorption with $Q_{in} = 10$ W: (a) $T_j - T_{amb}$ and (b) $T_{cm} - T_{amb}$ where (i) $t = 420$ s, (ii) $t = 1,100$ s, (iii) $t = 2,000$ s, and (iv) $t = 2,620$ s.

Substitution of the initial conditions, i.e. $\theta(t) = 0$ at $t = 0$, yields $C = -Q_{in}/c_2$ in Eq. (10). Finally, the variation in the temperature of the medium with time is expressed as,

$$\theta(t) = \frac{Q_{in}}{c_2} \left(1 - e^{-\left(\frac{c_2}{c_1}\right)t} \right) \quad (11)$$

where the constants Q_{in} , c_1 and c_2 need to be determined from the test conditions (Q_{in}), the thermophysical properties ($c_1 = C_p M$) and the experimentally measured data (steady-state condenser temperatures, i.e. $c_2 = Ah_e$), as listed in Table 2.

By differentiating Eq. (11), it can be shown that the initial gradient of the medium's surface temperature is inversely proportional to the thermal capacity ($c_1 = C_p M$) and is independent of the effective heat transfer coefficient of h_e and the heat transfer area of A , according to

$$\frac{d\theta}{dt}(t=0) = \frac{Q_{in}}{c_1} \quad (12)$$

On substituting $t = \infty$ into Eq. (11), it can be shown that the steady-state temperature of the medium is inversely proportional to the effective heat transfer coefficient of h_e and the heat transfer area of A and is independent of the thermal capacity, according to

$$\theta(t = \infty) = \frac{Q_{in}}{c_2} = \frac{Q_{in}}{Ah_e} \quad (13)$$

Equation (11) provides the curves that fit the temporal variation of the condenser's averaged surface temperature for all three specimens included in Fig. 8. The analytical expression of Eq. (11) shows good agreement with the experimental data. The initial increases in the averaged surface temperatures following the order, CP < VC < uFVC are due to the thermal capacities ($c_1 = 90.1$ J/K for the CP, $c_1 = 63.8$ J/K for the VC, and $c_1 = 40.7$ J/K for the uFVC: the initial gradient of the surface temperature was inversely proportional to $c_1 = C_p M$ by Eq. (12), as summarized in Table 2.

On the other hand, in the steady-state regime, external cooling by

both radiation and natural convection from each specimen's surface determines the average surface temperatures. Fig. 7 shows that the uFVC surface temperature is significantly lower than the commercial VC and CP by $\sim 16\%$. Thus, the steady-state temperature was used in Eq. (13) to estimate $c_2 = Ah_e = 0.193$ W/K (for the CP), 0.196 W/K (for the VC) and 0.226 W/K for the uFVC, indicating that the uFVC is cooled $\sim 16\%$ more effectively than both CP and VC.

In summary, we revert back to the classical first-order unsteady-state differential equation to explain the above transient response of the newly introduced uFVC; the steeper increase in the temperature is due to the reduced product of the thermal capacity and mass of the uFVC, and the reduced steady-state temperature is due to the enhanced thermal spreading. The higher $Ah_{e,uFVC}$ value of 15.3 % compared to the $Ah_{e,comVC}$ and 17.1 % for the copper plate specimen means that the uFVC is more effective at spreading and dissipating heat. The increased thermal dissipation of the uFVC is attributed to the internal foam ligaments that enables the uFVC to utilize the available heat transfer surface area of the condenser surfaces to a greater extent than the commercial VC and CP, since these surfaces were not insulated in the experiments. It is important to note that previous studies characterized the thermal performance of VCs by insulating the condenser surfaces in the experiments to apply the adiabatic wall boundary condition. For example, the experimental investigation of Yu et al. [40] demonstrated the thermal performance of an uTVC, where a copper heating block and constant temperature water-cooled block were attached to the VC. The other external surfaces were well insulated to implement an adiabatic wall boundary condition. Based on this setup the thermal resistance ($R = \Delta T/Q_{in}$) was subsequently determined using the input heat (Q_{in}) and the temperature difference ($\Delta T = T_h - T_c$) between the heated surface and the cooled surface, since the heat loss from the VC walls to the surroundings was minimized. In contrast, the present study did not insulate the external surfaces of the VC. Here, heat loss from the available external condenser surfaces to the ambient surroundings were allowed to occur, since it is more representative of a real-world setup. Consequentially, substantial heat transfer from the walls of the uFVC ensures that the overall condenser temperature is lower than the reference commercial VC and copper plate as demonstrated by the theoretical analysis.

4. Conclusions

For mobile and aerial applications, this study introduced a uFVC that is ultralight e.g., ~ 25 g, compared to ~ 120 g of a commercial VC with similar dimensions, achieving a remarkable $\sim 80\%$ weight reduction with a comparable junction temperature. This advancement was made possible by a new fabrication process – incorporating a high porosity Cu foam with blackened surfaces for superhydrophilicity and encapsulation by an ultrathin-walled envelope via electroless plating – uFVC. This article detailed the fabrication process, and thermal characteristics of the uFVC subjected to natural convection. The imposed heat on the uFVC was spread via latent heat conversion of DI water whose capillary transport was promoted by the blackened foam ligaments within the ultrathin-walled envelope. In particular, the faster transient thermal response of the uFVC indicated a smaller $C_p M$ compared to the two other specimens ensuring a faster heat-up and cool-down period. For uFVC's steady-state operations, the ultrathin-walled envelope facilitated heat rejection to the surroundings by natural convection. Overall, the results suggest that the uFVC can be an alternative thermal management solution that provides extreme lightweighting, in addition to mechanical strength due to the pore-base mechanical support provided by the foam ligaments. There is potential for structural integration of the uFVCs into existing mobile and aerial systems that are equipped with electronic and electrochemical devices, e.g., electronic chips and batteries.

Future research aims to evaluate the thermal performance of the uFVC using copper metallic foam wicks with differing pore sizes (e.g., 130 PPI and 150 PPI). In addition, to characterise the ultra-thin uFVC manufactured in the range of 0.25 mm to 0.5 mm thickness. Further

Table 2

Thermophysical and measured parameters of a copper plate (CP), a commercial vapor chamber (VC), and our ultralight vapor chamber (uFVC).

Specimen	Parameters
CP	$C_p = 392$ J/(kgK) at 60 °C and 401 J/(kgK) at 100 °C, measured $M = 0.19586$ kg, measured $A = 0.00735$ m ² $h_e = 26.21$ W/m ² K $c_1 = C_p M = 90.1$ J/K $c_2 = Ah_e = 0.193$ W/K $\theta_{\infty} = T_{\infty} - T_{amb} = 51.9$ K (with $Q_{in} = 10$ W) $\rho = 8940$ kg/m ³ at 60 °C and 100 °C (measured) $K = 400.3$ W/(mK) at 60 °C and 404.1 W/(mK) at 100 °C (measured)
Commercial VC	$C_p = 525$ J/(kgK) $M = 0.12143$ kg, measured $A = 0.00707$ m ² $h_e = 27.72$ W/m ² K $c_1 = C_p M = 63.8$ J/K, estimated by fitting using Eq. (13) $c_2 = Ah_e = 0.196$ W/K, estimated by fitting using Eq. (13) $\theta_{\infty} = T_{\infty} - T_{amb} = 51.0$ K (with $Q_{in} = 10$ W)
uFVC	$C_p = 1,600$ J/(kgK) $M = 0.02543$ kg, measured $A = 0.00787$ m ² $h_e = 28.67$ W/m ² K $c_1 = C_p M = 40.7$ J/K, estimated by fitting using Eq. (13) $c_2 = Ah_e = 0.226$ W/K, estimated by fitting using Eq. (13) $\theta_{\infty} = T_{\infty} - T_{amb} = 44.3$ K (with $Q_{in} = 10$ W)

research into the manufacturing processes is required to reduce the complexity and streamline the present production process.

CRedit authorship contribution statement

Haodong Chen: Writing – review & editing, Writing – original draft, Methodology, Formal analysis. **Kiju Kang:** Writing – review & editing, Writing – original draft, Supervision, Conceptualization. **Michael D. Atkins:** Writing – review & editing. **Tongbeum Kim:** Writing – review & editing, Writing – original draft, Formal analysis.

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.

Acknowledgments

This work was supported by Ministry of Science and ICT of Korea via the National Research Foundation of Korea (Grant No. 2021R1A2C3007705).

Data availability

Data will be made available on request.

References

- [1] Baknor The Way to Cool, Ultra-thin loop heat pipe a two-phase heat transfer device. <https://www.baknorthermal.com/ultra-thin-heat-pipes-phase-change-cooling-for-electronics/>, 2025 (accessed 04 July 4 2025).
- [2] K. Dinh, 1999. Serpentine heat pipe in air conditioning systems. European patent. EP0647307B1.
- [3] X. Chen, S. Chen, Z. Zhang, D. Sun, X. Liu, Heat transfer investigation of a flat-plate oscillating heat pipe with tandem dual channels under nonuniform heating, *Int. J. Heat Mass Transf.* 180 (2021) 121830.
- [4] T. Zhao, Z. Hu, Opportunities and challenges for the development of ultra-thin heat pipe, *Front. Energy Res.* 10 (2022) 923454.
- [5] H.-W. Chiou, Y.-M. Lee, H.-H. Hsiao, L.-C. Cheng, Thermal modeling and design on smartphones with heat pipe cooling technique, in: 2017 IEEE/ACM International Conference on Computer-Aided Design (ICCAD), 2017, pp. 482–489.
- [6] A. Chen, F. Jiang, J. Dong, J. Chen, Y. Zhu, Design, fabrication and thermal performance of a novel ultra-thin loop heat pipe with printed wick structure for mobile electronics cooling, *Appl. Therm. Eng.* 200 (2022) 117683.
- [7] J. Jose, T. Kumar Hotta, A comprehensive review of heat pipe: its types, incorporation techniques, methods of analysis and applications, *Therm. Sci. Eng. Prog.* 42 (2023) 101860.
- [8] L. Jiang, Y. Tang, W. Zhou, L. Jiang, L. Lu, Fabrication of flatten grooved-sintered wick heat pipe, *Trans. Nonferrous Met. Soc. China* 23 (2013) 2714–2725.
- [9] S.H. Moon, G. Hwang, H.G. Yun, T.G. Choy, Y.I. Kang, Improving thermal performance of miniature heat pipe for notebook PC cooling, *Microelectron. Reliab.* 42 (2002) 135–140.
- [10] H. Aoki, N. Shioya, M. Ikeda, Y. Kimura, Development of ultra-thin plate-type heat pipe with less than 1 mm thickness, in: 2010 26th Annual IEEE Semiconductor Thermal Measurement and Management Symposium (SEMI-THERM), 2010, pp. 217–222.
- [11] Z. Cui, L. Jia, Z. Wang, C. Dang, L. Yin, Thermal performance of an ultra-thin flat heat pipe with striped super-hydrophilic wick structure, *Appl. Therm. Eng.* 208 (2022) 118249.
- [12] D. Ansari, K.-Y. Kim, Hotspot thermal management using a microchannel-pinfin hybrid heat sink, *Int. J. Therm. Sci.* 134 (2018) 27–39.
- [13] K.F. Wiedenheft, H.A. Guo, X. Qu, J.B. Boreyko, F. Liu, K. Zhang, F. Eid, A. Choudhury, Z. Li, C.-H. Chen, Hotspot cooling with jumping-drop vapor chambers, *Appl. Phys. Lett.* 110 (2017) 141601.
- [14] X. Chen, H. Ye, X. Fan, T. Ren, G. Zhang, A review of small heat pipes for electronics, *Appl. Therm. Eng.* 96 (2016) 1–17.
- [15] X. Cheng, G. Yang, J. Wu, Recent advances in the optimization of evaporator wicks of vapor chambers: from mechanism to fabrication technologies, *Appl. Therm. Eng.* 188 (2021) 116611.
- [16] J.A. Weibel, S.V. Garimella, Chapter four - recent advances in vapor chamber transport characterization for high-heat-flux applications, in: E.M. Sparrow, Y. I. Cho, J.P. Abraham, J.M. Gorman (Eds.), *Advances in Heat Transfer*, Elsevier, 2013, pp. 209–301.
- [17] X. Ji, J. Xu, A.M. Abanda, Copper foam based vapor chamber for high heat flux dissipation, *Exp. Therm Fluid Sci.* 40 (2012) 93–102.
- [18] S. Sudhakar, S.N. Joshi, J.A. Weibel, Investigation of boiling behaviors in vapor chambers in response to transient heat input profiles, *Appl. Therm. Eng.* 198 (2021) 117426.
- [19] T. Semenic, Y.Y. Lin, I. Catton, D.B. Sarraf, Use of biporous wicks to remove high heat fluxes, *Appl. Therm. Eng.* 28 (2008) 278–283.
- [20] 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, *ASME J. Heat Transf.* 134 (2012) 101501.
- [21] F. Zhou, Y. Liu, E.M. Dede, Design, Fabrication, and performance evaluation of a hybrid wick vapor chamber, *ASME J. Heat Transf.* 141 (2019) 081802.
- [22] G.S. Hwang, E. Fleming, B. Carne, S. Sharratt, Y. Nam, P. Dussinger, Y.S. Ju, M. Kaviany, Multi-artery heat-pipe spreader: Lateral liquid supply, *Int. J. Heat Mass Transf.* 54 (2011) 2334–2340.
- [23] C. Liu, D. Hu, Q. Li, X. Chen, Z. Zhang, F. Zhou, Vapor chamber with two-layer liquid supply evaporator wick for high-heat-flux devices, *Appl. Therm. Eng.* 190 (2021) 116803.
- [24] S.N. Joshi, F. Zhou, E.M. Dede, D.J. Lohan, S. Sudhakar, J.A. Weibel, Thermal performance evaluation of a two-layer wick vapor chamber for high heat flux dissipation by air cooling, in: 19th IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm), 2020, pp. 91–96.
- [25] J. Yu, Y. Li, Z. Xin, Z. Chen, L. Deng, X. Guo, H. Chen, H. He, Experimental investigation on the thermal characteristics of ultrathin vapour chamber with in-plane bending, *Appl. Therm. Eng.* 217 (2022) 119175.
- [26] K. Rezk, M.A. Abdelrahman, A.A.A. Attia, M. Emam, Thermal control of temperature-sensitive electronic components using a vapor chamber integrated with a straight fins heat sink: an experimental investigation, *Appl. Therm. Eng.* 217 (2022) 119147.
- [27] Z. Zhou, X. Wang, Y. Zhou, Performance study of a leaf-vein-like structured vapor chamber, *Materials* 16 (2023) 4482.
- [28] L. Luo, B. Huang, X. Bai, Z. Cheng, Q. Jian, Temperature uniformity improvement of a proton exchange membrane fuel cell stack with ultra-thin vapor chambers, *Appl. Energy* 270 (2020) 115192.
- [29] G. Zhong, Y. Tang, X. Ding, L. Rao, G. Chen, K. Tang, W. Yuan, Z. Li, Experimental study of a large-area ultra-thin flat heat pipe for solar collectors under different cooling conditions, *Renew. Energy* 149 (2020) 1032–1039.
- [30] G. Huang, W. Liu, Y. Luo, Y. Li, H. Chen, Fabrication and thermal performance of mesh-type ultra-thin vapor chambers, *Appl. Therm. Eng.* 162 (2019) 114263.
- [31] G. Huang, W. Liu, Y. Luo, Y. Li, A novel ultra-thin vapor chamber for heat dissipation in ultra-thin portable electronic devices, *Appl. Therm. Eng.* 167 (2020) 114726.
- [32] C.-K. Huang, C.-Y. Su, K.-Y. Lee, The Effects of Vapor Space Height on the Vapor Chamber Performance, *Exp. Heat Transf.* 25 (2012) 1–11.
- [33] G. Huang, W. Liu, Y. Luo, T. Deng, Y. Li, H. Chen, Research and optimization design of limited internal cavity of ultra-thin vapor chamber, *Int. J. Heat Mass Transf.* 148 (2020) 119101.
- [34] S.Y. Semenov, K.A. Burke, Titanium heat pipe thermal plane, *AIP Conf. Proc.* (2008) 131–137.
- [35] H. Jiang, X. Wang, C. Ding, D. Shan, B. Guo, H. Qi, J. Xu, A review of emerging design and theoretical progress on vapor chamber for efficient thermal performance, *Int. J. Heat Mass Transf.* 231 (2024) 125814.
- [36] M. Bulut, S.G. Kandlikar, N. Sozbir, A review of vapor chambers, *Heat Transfer Eng.* 40 (19) (2019) 1551–1573, <https://doi.org/10.1080/01457632.2018.1480868>.
- [37] M. Enmark, M. Murugesan, H. Zhang, T.M.J. Nilsson, K.J. Kallio, A. Kamal, J. Liu, Design and characterization of novel graphene-enhanced vapor chambers for lightweight and high-performance electronics cooling, *Nanotechnology* 36 (2025) 105701.
- [38] Y.J. Lee, Y.X. Wang, Y.C. Jeong, M.D. Atkins, K. Kang, T. Kim, An ultralight capillary-driven heat pipe, *Langmuir* 40 (2024) 8194–8204.
- [39] Y.X. Wang, C.Q. Yu, K. Kang, M.D. Atkins, T. Kim, An ultrathin-walled foam heat pipe, *ASME J. Heat Mass Transf.* 146 (2024) 081602.
- [40] J. Yu, Y. Li, Z. Chen, Q. Luo, H. Chen, X. Tang, Effect of the passage area ratio of wick on an ultra-thin vapour chamber with a spiral woven mesh wick, *Appl. Therm. Eng.* 196 (2021) 117282.
- [41] Z. Zhang, X. Wang, Y. Yan, A review of the state-of-the-art in electronic cooling, *Adv. Electr. Eng. Electron. Energy* 1 (2021) 100009.
- [42] S.C. Han, J.W. Lee, K. Kang, A new type of low-density material: Shellular, *Adv. Mater.* 27 (2015) 5506–5511.
- [43] H. Tang, Y. Xie, L. Xia, Y. Tang, Y. Sun, Review on the fabrication of surface functional structures for enhancing heat transfer of heat pipes, *Appl. Therm. Eng.* 226 (2023) 120337.
- [44] J. Min, X. Wu, L. Shen, F. Gao, Hydrophilic treatment and performance evaluation of copper finned tube evaporators, *Appl. Therm. Eng.* 31 (2011) 2936–2942.
- [45] B. Zohuri, Heat Pipe Theory and Modeling, Heat Pipe Design and Technology, CRC Press, Florida, 2011, pp. 144–151.
- [46] Y.J. Youn, S.J. Kim, Fabrication and evaluation of a silicon-based micro pulsating heat spreader, *Sensor Actuat. A: Phys.* 174 (2012) 189–197.
- [47] S.W. Chang, K.F. Chiang, W.L. Cai, Thermal performance evaluation of thin vapor chamber, *Appl. Therm. Eng.* 149 (2019) 220–230.
- [48] H.W. Coleman, W.G. Steele, Uncertainty in a Result Determined from Multiple Variables, Experimentation, Validation, and Uncertainty Analysis for Engineers, Wiley, Hoboken, NJ, 2018.