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Journal

International Journal of Heat and Mass Transfer, 269

ISSN

0017-9310

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Publication Date

2026-11-01

DOI

10.1016/j.ijheatmasstransfer.2026.129108

Peer reviewed

Direct-on-Chip Confined Two-Phase Jet Impingement Cooling utilizing Water with Membrane-based Phase Separation

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Abstract

Two-phase jet impingement cooling has emerged as a leading strategy for managing the extreme heat fluxes and local hot spots of modern electronics. Yet, practical application in confined geometries is often hindered by flow maldistribution and instabilities that occur due to outflow of the expanded liquid-vapor mixture. This study proposes a direct-on-chip confined two-phase jet impingement cooling strategy that incorporates a liquid-vapor phase-separation membrane to manage outflow of the two-phase mixture. Implemented through a staggered multi-jet array with distributed inlets and outlets to manage flow distribution in a three-path manifold, the membrane is positioned as the jet confinement wall above the heated surface. Phase separation is thereby achieved by selectively permeating vapor to a dedicated outlet while retaining liquid under controlled transmembrane pressure. An operational regime map is first developed based on the vapor generation rate and vapor venting capacity to identify the theoretical limiting factors. An experimental investigation then systematically characterizes the thermal-hydraulic performance using deionized water as the working fluid, examining the effect of phase separation under varying heat fluxes, flow rates, and transmembrane pressures. The results show that phase separation consistently mitigates both the magnitude and oscillation of pressure drop in two-phase flow regimes

compared to non-separation operation where the dedicated vapor outlet is closed. At the maximum steady-state heat fluxes, the implementation of phase separation achieves average pressure drop reductions of 23.0%, 13.0%, and 10.5%, and suppresses the pressure drop oscillations (sample standard deviation) by 51.9%, 57.5%, and 23.5% at flow rates of 0.06, 0.10, and 0.14 L/min, respectively. Under extreme operating conditions near the critical heat flux at low flow rates (0.06 L/min and 175 W/cm²), phase separation effectively suppresses local dryout, reducing the surface temperature by up to 6.7 °C and diminishing the surface temperature oscillations from 4.76 °C to 0.44 °C (90.8% reduction). The membrane enables stable operation by preferentially venting vapor directly from the impingement cavity, which prevents the accumulation of large vapor structures in the liquid path. This research provides key insights into phase-separation-enhanced two-phase jet impingement cooling, offering a high-performance solution having stable operation for the thermal management of next-generation electronic devices.

Keywords: jet impingement; phase separation; vapor venting; membrane; two-phase flow; water; electronics cooling

Highlights

- Proposed and evaluated membrane-based phase separation for jet impingement cooling.
- Developed operational regime maps based on vapor generation and venting models.
- Two-phase pressure drop and its oscillations mitigated by up to 23.0% and 57.5%.
- Superheat and its oscillations reduced by 28.6 % and 90.8% under extreme conditions.
- Active venting from impingement cavity prevents vapor accumulation in liquid paths.

Nomenclature

A	area, m ²
D	diameter, m
d	pore size, m
H	jet height, m
h	specific enthalpy, J/kg
k	thermal conductivity, W/(m·K)
N	number of variables
P	pressure, Pa
p	nozzle pitch, m
q	heat flux, W/m ²
Re	Reynolds number
T	temperature, °C
u	velocity, m/s
$\dot{V}$	volumetric flow rate, m ³ /s
x	vapor quality
y	distance, m

Greek letters

γ	surface tension, N/m
Δ	difference or drop
δ	thickness, m
η	phase separation efficiency
θ	contact angle, °
κ	permeability, m ²

μ	dynamic viscosity, Pa·s
ρ	density, kg/m ³
σ	sample standard deviation

Acronyms

CHF	critical heat flux
DI	deionized
HTC	heat transfer coefficient
NON	non-separation
PET	polyethylene terephthalate
PP	polypropylene
PTFE	polytetrafluoroethylene
SEP	separation

Subscripts

b	bubble
break	breakthrough
fg	latent heat of vaporization
gen	generation
in	inlet
l	liquid
mem	membrane
out	outlet
sat	saturation
sgl	single
surf	surface

1 Introduction

The rapid digitalization of industries and growth of artificial intelligence applications has transformed data centers into critical infrastructure with increasingly significant energy demands. In 2024, data centers consumed approximately 415 TWh of electricity, and this number is predicted to more than double by 2030 [1,2]. At the same time, highly integrated and power-dense electronic chips produce intense and localized heat, creating critical challenges for thermal management [3,4]. Increased operating temperatures not only deteriorate device performance but also reduce device reliability, contributing to over 50% of electronic failures [5,6]. Consequently, developing efficient and reliable cooling strategies has become essential for mitigating the energy and thermal challenges associated with modern data centers.

To address the heat dissipation demands of modern electronics, performance-enhanced cooling technologies such as jet impingement [7,8], microchannel heat sinks [9,10], and spray cooling [11,12] have garnered significant research interest as superior alternatives to conventional air cooling [5,13,14]. In jet impingement cooling, a high-velocity fluid stream is directed onto the heated surface, thereby thinning the thermal boundary layer and resulting in a high heat transfer coefficient (HTC) in the impingement stagnation zone, along with manageable flow resistance [15,16]. Among jet impingement configurations, confined jet impingement, where the flow is constrained in a gap between the heated surface and a confinement plate, is particularly promising for electronics cooling due to its compactness and scalability compared to free-surface or submerged jets [17–19]. The use of multi-jet arrays with distributed inlets and outlets can further enhance thermal performance by effectively managing localized hot spots, and simultaneously reducing the pressure drop through locally distributed fluid flow networks [20–25].

In two-phase jet impingement onto a superheated surface that causes boiling of the fluid, the latent heat of vaporization is leveraged to achieve an increased HTC at significantly lower flow rates than single-phase systems, and maintains improved surface temperature uniformity [26,27]. Research in this domain has characterized the underlying mechanisms, critical heat flux (CHF) limits, and the influence of key parameters. Clark et al. [28] identified that nucleate boiling, rather than forced convection, is the dominant

heat transfer mechanism during confined jet impingement in the fully boiling regime, where the entire impingement surface undergoes boiling. The HTC becomes insensitive to jet characteristics in this regime and the CHF exhibits a linear dependence on jet velocity. This dominance of nucleate boiling was further elucidated by Rau [29] by using flow visualization to demonstrate that vapor generation and buoyancy-driven motion are the primary drivers of the flow field within the confinement gap, overwhelming the inertia of the impinging jet itself. Devahdhanush and Mudawar [22] conducted a systematic study for a confined round jet of R134a, revealing that jet velocity exerts a more significant positive influence on CHF than the mass flow rate. Two distinct CHF regimes were identified: a subcooled CHF occurring at high flow rates and a saturated CHF at low flow rates, each characterized by a mild and sharp temperature rise, respectively.

The performance of two-phase jet impingement is strongly influenced by geometric and operational parameters. For instance, Tang et al. [30] investigated a distributed jet array impinging on a microchannel surface, finding that reducing subcooling from 40 K to 20 K enhances the HTC by 45.4% to 57.2%, at the cost of a 25.5% average increase in pressure drop. They also reported that decreasing the jet height from 4 mm to 2 mm causes a 2.5-fold increase in pressure drop and a degradation in the HTC, noting that the overall thermal-hydraulic performance can be improved by increasing subcooling and jet height or reducing the flow rate. Zhang et al. [31] examined jet impingement boiling over micro-pin-finned surfaces, confirming that enhanced surfaces consistently outperform smooth surfaces. Their work highlighted the Reynolds number as the most influential dimensionless parameter on thermal performance in both single-phase and two-phase regimes, while the dimensionless jet height (H/D) has negligible impact on the Nusselt number in the two-phase dominant regime.

A primary concern in two-phase jet impingement boiling is the potential for flow maldistribution and instability triggered by vapor generation, particularly when using water, due to the larger density difference between vapor and liquid compared to other fluids [8,32]. In two-phase water jet impingement with a single confined jet, Mira-Hernández et al. [33] visualized that the vapor agglomerates in the confinement gap into a bowl-like vapor structure that recurrently shrinks under saturated exit flow conditions. Their following

study [34] further characterized how flow disturbances in the confinement gap cause premature dryout and CHF degradation, noting that this effect is more severe at lower jet velocities. In distributed jet array impingement, Tang et al. [30] observed that increasing the flow rate of the subcooled jet induces sharper pressure fluctuations due to intensified condensation of generated vapor by the subcooled liquid, while decreasing the jet height helps stabilize the flow. Additionally, Ji et al. [35] found that instability is triggered when large vapor masses intermittently block the jet inlets and outlets, and that reducing the jet height can suppress these instabilities without increasing the pressure drop, using HFE-7100 as the working fluid on enhanced surfaces. Similarly, Cui et al. [36] observed that the flow becomes unstable when large bubbles choke the outlets at certain high heat fluxes, causing large pressure fluctuations. However, the flow can stabilize once steady vapor columns form. The authors also reported that increasing the flow rate delays the onset of instability, while reducing the jet height or enlarging the nozzles can reduce fluctuations altogether. Nevertheless, these operational and geometric adjustments do not fundamentally address the mechanism of vapor accumulation that triggers these instabilities in highly confined spaces, and often entail some trade-offs, such as excessive pressure drop from reduced jet heights or diminished cooling efficiency from increased flow rates.

Applying phase separation to remove vapor from the boiling working fluid is a promising approach to mitigate flow instabilities and enhance overall device performance. The concept of a vapor-venting microchannel heat exchanger was first proposed by Zhou et al. [37], in which a hydrophobic porous membrane atop the microchannel selectively allows vapor to permeate while repelling liquid. Building on this, a series of studies has been conducted, as summarized chronologically in Table 1. David et al. [38] demonstrated that such a design yields a lower two-phase pressure drop than a conventional non-venting system where liquid along with all generated bubbles flows through channels. Subsequent prototypes showed a 35% reduction in the single-phase thermal resistance and a delay in the onset of nucleating boiling when providing additional air flow in the vapor channel [39]. Their subsequent visualization study confirmed effective air (adiabatic testing) and vapor (diabatic testing) removal through polytetrafluoroethylene (PTFE) membranes at mass qualities of up to 7% and 20%, respectively [40].

Further testing revealed significant improvements, including an ~60% reduction in normalized pressure drop and a substrate temperature reduction of up to 4.4°C compared to a device without phase separation [41]. Numerical simulations by Fang et al. [42] demonstrated that the transient vapor-venting process effectively reduces pressure drop and suppresses local dryout inside microchannels. Li and Wei [43] similarly reported that phase separation membranes effectively reduce pressure drop and enhance the HTC by lowering the average vapor quality of the flowing stream, with performance further improved when combined with segmented microchannel structures. Mohiuddin et al. [44] investigated the role of channel aspect ratio in a microchannel manifold with phase separation membranes. They found that a high-aspect-ratio channel with a PTFE membrane achieves a 60% reduction in two-phase pressure drop with a 10% HTC penalty, while a low-aspect-ratio configuration yields a 30% increase in HTC alongside a 10% pressure drop reduction.

Table 1 Overview of current literature on two-phase cooling approaches incorporating phase separation

Literature	Year	Methodology	Flow structure	Working fluid	Separation layer
David et al. [38]	2007	experimental	microchannel	water	unlaminated PTFE membrane
David et al. [39]	2008	experimental	microchannel	water	unlaminated PTFE membrane
Fang et al. [42]	2008	numerical	microchannel	water	PTFE membrane
David et al. [40]	2011	experimental	microchannel	air-water, water	unlaminated PTFE membrane
David et al. [41,45]	2011	analytical & experimental	microchannel	water	unlaminated PTFE membrane
Warrier et al. [46]	2014	analytical	microchannel	FC-72	silicon porous rib
Fazeli et al. [47]	2015	experimental	channel without liquid outlet	water	PTFE membrane
Fazeli and Moghaddam [48]	2017	experimental	channel without liquid outlet	water	PTFE membrane
Fazeli and Moghaddam [49]	2017	analytical & experimental	channel without liquid outlet	water	PTFE, acrylic copolymer, and PES membranes
Mohiuddin et al. [44]	2020	experimental	microchannel	water	PTFE membrane
Alipanah and Moghaddam [50]	2020	experimental	channel without liquid outlet	water	acrylic copolymer and custom-made membranes
Tamvada et al. [51]	2021	experimental	channel without liquid outlet	ethanol	oleophobic PTFE membrane

Li and Wei [43]	2022	experimental	microchannel	water	unlaminated PTFE membrane
Tamvada and Moghaddam [52]	2023	experimental	channel without liquid outlet	water	PTFE membrane
Li et al. [53]	2024	experimental	minichannel	glycerol-water	PTFE membrane under electric field
Li et al. [54]	2024	experimental	microchannel	water	composite PTFE membrane with PET
Li et al. [55]	2024	experimental	minichannel	glycerol-water	PTFE membrane under electric field
Wang et al. [56]	2024	experimental	minichannel	water	micro-pore porous copper rib
Zhang et al. [57]	2024	experimental	microchannel	water	composite PTFE membrane with PET
Wang et al. [58]	2025	experimental	minichannel	water	gradient porous copper rib
Fan et al. [59]	2025	experimental	minichannel	anhydrous ethanol	modified electrospun PVDF-HFP membrane

Recent studies have further advanced the understanding and performance of this membrane-venting approach. Li et al. [54] developed a microchannel heat sink with vapor venting, where vapor extraction through a 1.0 μm -pore-size PTFE membrane enabled dissipation of a heat flux of 287.8 W/cm² from the heated surface, with a transmembrane heat flux of 152.7 W/cm². In a related approach, Zhang et al. [57] demonstrated that an open microchannel heat sink with a PTFE membrane could achieve a stable heat flux dissipation of 488.6 W/cm² with minimal temperature and pressure fluctuations, significantly outperforming a closed microchannel design. Li et al. [53] further investigated the performance of porous phase separation structures by considering the synergistic effect of an electric field applied via wire electrodes positioned at the geometric center of each channel. The phase separation structure alone enhances the average HTC by up to 26.2% and reduces the pressure drop by 19.4%; adding an electric field increases the maximum heat transfer enhancement to 65.5%. A subsequent study with a 30% mass fraction glycerol-water solution reported that applying a phase separation membrane and a 1600 V electric field achieves a maximum enhanced heat transfer factor of 1.63 and a wall temperature non-uniformity reduction of 11.25% to 30.86% compared to a plain heat sink [55].

Besides top-wall separation, porous side walls have also been explored for phase separation in microchannels. Warriar et al. [46] proposed a microchannel device incorporating perforated side walls, allowing the separated vapor to flow independently within an adjacent vapor channel. Building on this concept, Wang et al. [56] utilized microporous copper ribs within minichannels and demonstrated up to a

25.5 % reduction in total pressure drop compared to non-separation operations. In their subsequent study [58], they introduced a gradient porous copper rib design where the vapor channels are filled with open-cell porous copper. This configuration enhances both thermal conduction and structural support by effectively utilizing the space previously occupied by vapor channels. Adopting a similar architecture, Fan et al. [59] integrated modified electrospun PVDF-HFP membranes within the rib structures, and developed a design allowing downstream-generated vapor enters the upstream region of adjacent counterflowing channels. This design suppresses bubble overgrowth and improves thermal uniformity without requiring external venting.

Beyond micro/minichannel manifold devices, Fazeli et al. [47] developed a membrane-assisted heat sink capable of maintaining 100% exit vapor quality and featuring no liquid outlet. This approach was shown to achieve an order of magnitude higher HTC when operating at 1 to 2 orders of magnitude lower mass flux compared to conventional heat sinks. Using water as the working fluid, a maximum CHF of 1760 W/cm² was reached at a wall superheat of less than 26 °C [48]. Further investigations quantified the impact of wickability, effective heat transfer area, and liquid pressure on CHF, with a model indicating that the thermal conductivity of surface structures ultimately limits performance [49]. In the authors' next prototype with a high-permeability membrane, a remarkable CHF value reaching 1000 W/cm² was achieved by Alipanah and Moghaddam [50]. Using an oleophobic membrane and the low-surface-tension fluid ethanol, Tamvada et al. achieved a CHF of 340 W/cm² and an HTC of 120 kW/(m²·K) [51]. In their subsequent research, a maximum heat flux of 670 W/cm² was reported for water at a sub-atmospheric saturation pressure of 16 kPa [52].

While the literature has demonstrated that phase separation can significantly enhance the thermal performance and flow stability of two-phase cooling approaches, its integration into two-phase jet impingement cooling, which is more scalable to large chip sizes with lower power drop and more uniform temperature, remains largely unexplored. Unlike microchannels where vapor expansion is highly confined and is forced in a single outflow direction, confined jet impingement involves complex stagnation flows where vapor expands radially within this narrow confinement gap and can block liquid replenishment to the

heated surface. Therefore, a solution is needed to address the fundamental trade-off between minimizing the confinement gap for compactness and enhanced heat transfer versus providing sufficient space for vapor flow to reduce pressure drop in two-phase jet impingement cooling.

To bridge this critical gap, this study proposes confined two-phase jet impingement with an integrated phase-separation membrane for direct-on-chip electronics cooling. The design employs a staggered multi-jet array and distributed inlet-outlet configuration in a three-path manifold, with a phase-separation membrane positioned as the jet confinement wall above the heated surface, to manage flow distribution and vapor removal. Using water as the working fluid, this study systematically investigates the influence of phase separation on thermal-hydraulic performance, under varying heat fluxes, flow rates, and transmembrane pressures, so as to provide key insights into the synergistic benefits of two-phase jet impingement and phase separation.

2 Design of jet impingement cooling with phase separation

The proposed concept of direct-on-chip confined two-phase jet impingement cooling integrated with a phase-separation membrane is illustrated in Fig. 1. Subcooled working fluid enters the manifold through the liquid inlet and is distributed through a nozzle array shown in Fig. 1(a), impinging directly onto the superheated surface. As the fluid absorbs heat, vapor is generated at the surface, creating a two-phase mixture. This mixture flows towards the liquid outlets in a gap between the surface and a porous hydrophobic membrane that forms the confinement wall of the impingement cavity, as shown in Fig. 1(b). The novelty of this design lies in the inclusion of this membrane that acts as a selective phase barrier. Driven by a controlled transmembrane pressure difference, a portion of the vapor in the two-phase mixture is continuously vented through the membrane as the two-phase liquid flow travels across the heated surface. This active removal of vapor directly from the confinement gap serves to reduce the local vapor fraction of the flow before it is collected and discharged through the liquid outlet manifold, mitigating flow instabilities associated with vapor slugging and thereby enhancing overall system stability and performance.

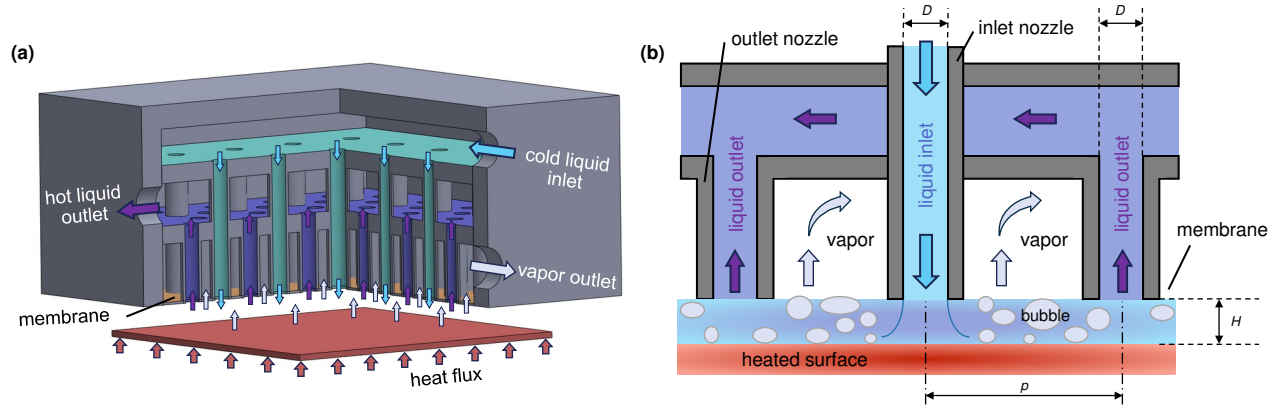


Fig. 1 Schematic diagram of proposed direct-on-chip confined two-phase jet impingement cooling with a phase-separation membrane: (a) 3D cutaway view showing the manifold architecture and internal flow paths; (b) 2D side view illustrating the jet impingement mechanism and phase separation process.

3 Experimental methodology

This section outlines the materials, prototype device architecture, and flow loop system setup used in the experimental characterization of the cooling approach. The tested operating conditions and procedures are then specified. Lastly, the methodologies for data reduction and uncertainty analysis are presented.

3.1 Membrane characterization

A composite PTFE membrane is selected as the phase separation layer due to its inherent hydrophobicity, excellent chemical stability, and heat resistance. Its porous structure permits the passage of water vapor while blocking liquid water under certain operating conditions, making it an ideal material for this application [60]. The selected composite membrane consists of an expanded PTFE layer laminated onto a polypropylene (PP) fiber support to enhance mechanical strength. As observed in the scanning electron microscope (SEM) images shown in Fig. 2, the fibers and pores in the PP layer are of a much larger length scale compared to the PTFE layer that has smaller $\sim 1 \mu\text{m}$ pore sizes. Static contact angle measurements for sessile water droplets on these surfaces are $\theta = 127.4^\circ$ for the PTFE layer and $\theta = 108.8^\circ$ for the PP layer.

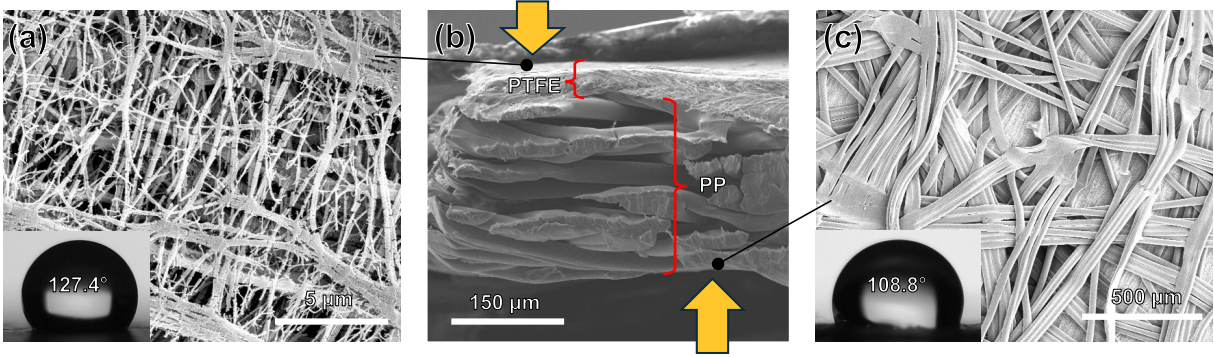


Fig. 2 SEM images and contact angle measurements for the: (a) PTFE layer with a 1 μm pore size; (b) cross-section of composite membrane; and (c) PP support layer.

The wettability of the membrane is characterized by its breakthrough pressure ($\Delta P_{\text{mem, break}}$, Pa, i.e., liquid entry pressure), which can be estimated using the Young-Laplace equation [61] given by

$$\Delta P_{\text{mem, break}} = - \frac{4\gamma \cos \theta}{d} \quad 11 \setminus * \text{MERGEFORMAT } ()$$

where this breakthrough pressure depends on the liquid surface tension (γ , N/m), contact angle (θ , $^\circ$), and maximum membrane pore size (d , m). For the measured contact angle, multiple different candidate PTFE membranes with nominal pore sizes of 0.45, 1, and 3 μm are estimated to have breakthrough pressures of 318, 143, and 48 kPa, respectively, for water at 100 $^\circ\text{C}$. Experimental validation at room temperature yields comparable measured breakthrough pressures of 340, 165, and 54 kPa for the 0.45, 1, and 3 μm membranes, respectively. These measured breakthrough pressure values at room temperature are consistently higher than the theoretical calculations at 100 $^\circ\text{C}$, which is attributed to the higher surface tension at lower temperatures.

The membrane with a 3 μm pore size was prone to liquid leakage during preliminary tests, while the 0.45 μm membrane offered a lower permeability. The intrinsic permeability (κ , m^2) of the membrane can be estimated using Darcy's law [62] as

$$\kappa = \mu \frac{\delta_{\text{mem}}}{A_{\text{mem}}} \frac{\dot{V}_{\text{mem}}}{\Delta P_{\text{mem}}} \quad 22 \setminus * \text{MERGEFORMAT } ()$$

Because higher vapor permeability is desirable for phase-separation performance [54], the composite PTFE membrane with a larger 1 μm pore size was selected over the 0.45 μm membrane for all subsequent experiments. The selected 1 μm membrane has a thickness of $210 \pm 60 \mu\text{m}$ and the intrinsic permeability (κ) of the membrane for water vapor at room temperature, calculated using Eq. 2, was determined to be $1.859 \times 10^{-14} \text{ m}^2$.

3.2 Manifold design

The concept design shown in Fig. 1 depicts a flow manifold for a large array of inlet and outlet nozzles to illustrate the scalability of the approach to large surface areas. However, to investigate and demonstrate the phase separation behavior, a simplified manifold having fewer nozzles is designed for fabrication and testing. Fig. 3(a) illustrates the flow path within this prototype manifold. Subcooled working fluid enters through the top-central liquid inlet and is distributed to an array of inlet nozzles (shown in blue). The liquid jets impinge upon the heated surface where heat absorption leads to vapor generation. The two-phase mixture is then driven toward outlet nozzles, while the hydrophobic membrane is bonded to the bottom surface of the manifold with a high-temperature-resistant adhesive enables selective phase separation (Fig. 3(b)). As liquid cannot pass through the membrane, vapor preferentially vents through the membrane into a dedicated vapor cavity and exits through the vapor outlet (shown in yellow), while the liquid with any remaining unvented vapor is collected by the liquid outlet nozzles and exhausted through the liquid outlet (shown in purple). To ensure a uniform flow and temperature distribution, the design employs a staggered array of multiple inlets and outlets. The configuration consists of 5 liquid inlet nozzles and 4 liquid outlet nozzles (Fig. 3(c)), all with a diameter of 0.8 mm and a nozzle pitch of 4 mm. The impingement cavity is designed to be 10 mm $\times$ 10 mm. The cooler manifold, shown in Fig. 3(d), was fabricated via additive manufacturing using a rigid high-temperature resin (Formlabs Rigid 10K), selected for a high heat deflection temperature exceeding 200 $^{\circ}\text{C}$ that withstands the prototype testing conditions [63].

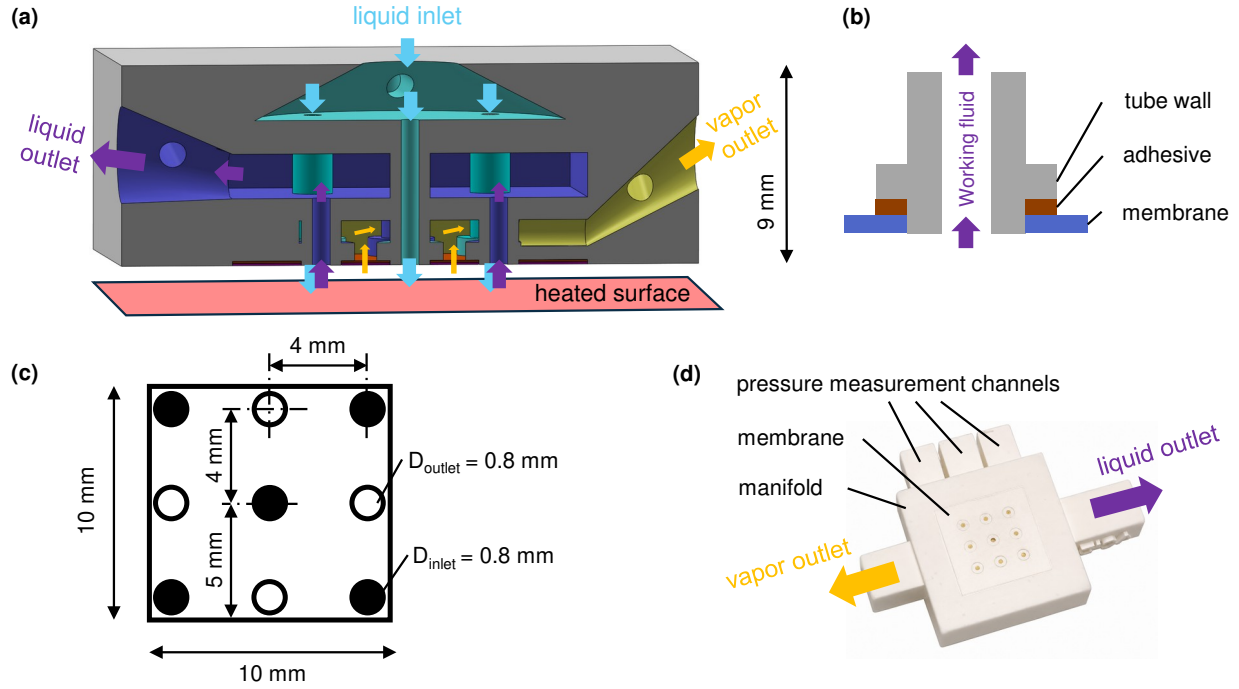


Fig. 3 Manifold design details: (a) 3D cutaway view showing the working fluid flow paths; (b) membrane attachment details; (c) nozzle array configuration; and (d) assembled cooler manifold prototype.

3.3 Test section assembly

A heater assembly was designed to simulate a high-power chip and provide the required heat flux, as illustrated in Fig. 4(a). The core of the assembly is an oxygen-free copper heater block with a 10 mm × 10 mm top surface that serves as the impingement target. A total of nine 50 W cartridge heaters are inserted into the bottom of the block and powered by DC power supply (XLN15010-GL, BK Precision). To determine the heat flux and surface temperature based on measurement of the temperature gradient, three T-type thermocouples (± 0.5 °C accuracy) are installed along the centerline of the upper section of the heater block. These thermocouples (T_1 , T_2 , T_3 , °C) are spaced at 2.5 mm intervals (Δy , m), with the uppermost thermocouple positioned 3.5 mm from the top surface ($\Delta y'$, m). The heat flux through the surface (q , W/m²) is calculated using Fourier's Law of heat conduction and a three-point backward difference Taylor's approximation, given as below:

$$q = k \frac{3T_3 - 4T_2 + T_1}{2D_y}$$

33* MERGEFORMAT ()

and the surface temperature (T_{surf} , °C) is calculated based on the temperature measured by the top thermocouple (T_1 , °C):

$$T_{\text{surf}} = T_1 - \frac{q}{k} D_y \zeta$$

44* MERGEFORMAT ()

The copper heater block is housed within a 3D-printed polymer enclosure (Formlabs Rigid 10K) comprising three parts: a lid for cooler manifold integration and sealing, a middle housing to provide lateral thermal insulation, and a bottom support for structural alignment and heat loss reduction. These parts are assembled and secured using screws and nuts around the perimeter. An additional ceramic block supports the copper heat block from the bottom as the temperature at the bottom of the heater block exceeds the temperature limit of Rigid 10K. The gap between the top edge of the copper block and the enclosure lid is sealed with epoxy to prevent flow leakage.

The jet impingement manifold is positioned directly above the heater assembly, as illustrated in Fig. 4(b). A silicone gasket is placed between the manifold and the heater assembly lid to ensure a leak-proof seal and to maintain a certain jet height (i.e., nozzle-to-target distance).

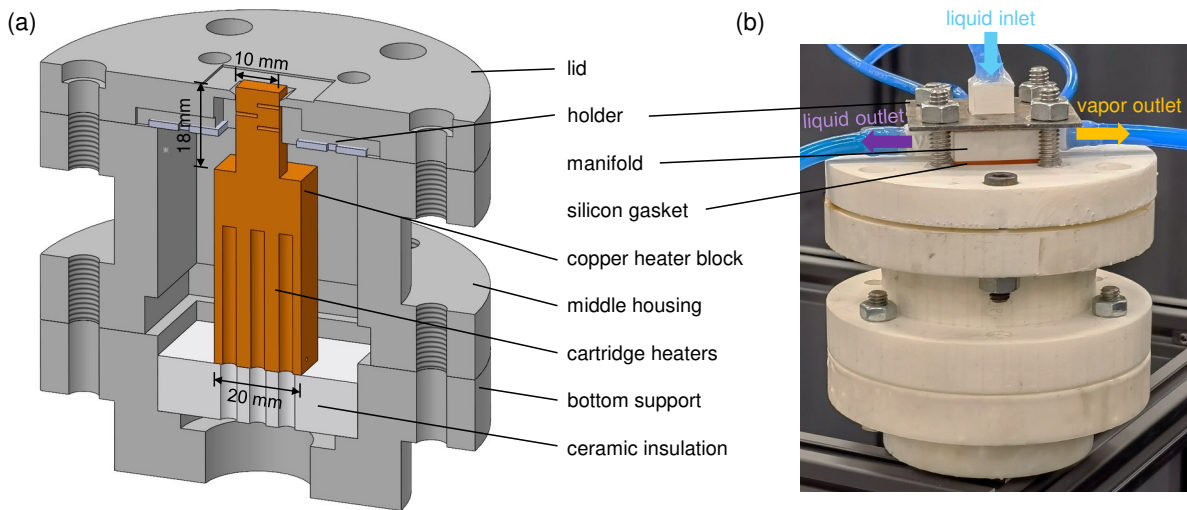


Fig. 4 Detailed configuration of the heater assembly design: (a) 3D cross-section view showing copper

heater block and enclosure; (b) experimental setup assembling the test cooler manifold.

3.4 Flow facility

The test section is integrated into a flow system where the reservoir is open to the atmosphere and remains at ambient pressure. Deionized (DI) water serves as the working fluid. As illustrated in Fig. 5(a)–(b), the system has a pump (GC-M35.JF5S.E, Micropump) that draws liquid from the reservoir. A bypass loop from the pump outlet back to the reservoir enables precise flow control and protects the pump by providing a pressure-relief path if the main test section path becomes restricted or blocked. The flow rate is measured at the pump outlet by a flowmeter (KODA K-Series, Alicat scientific, $\pm 0.1\%$ accuracy). Subsequently, the liquid passes through a plate heat exchanger that preheats the liquid temperature to a desired test section inlet subcooling level. A particulate filter is installed upstream of the test section to remove contaminants.

The liquid inlet and liquid outlet of the test section manifold are integrated into the primary flow line, while the vapor outlet is configured differently based on the testing condition: sealed with a valve (NON), vented to the atmosphere (SEP1), or connected to a vacuum pump (SEP2 and SEP3), as illustrated in Fig. 5(c) and specified in the Section 3.5. After exiting the test section, the working fluid leaving the liquid outlet is cooled by a plate subcooler connected to a chiller, rejecting the absorbed heat before returning to the reservoir.

The test section instrumentation comprises two absolute pressure transducers (PX429-100GI, OMEGA, $\pm 0.08\%$ accuracy) at the liquid inlet and vapor outlet, and one differential pressure transducer (PX409-100DWU5V, OMEGA, $\pm 0.08\%$ accuracy) across the liquid inlet and liquid outlet to determine the pressure drop across the manifold (ΔP , Pa). Two T-type thermocouples (± 0.5 °C accuracy) measure the fluid temperature at the liquid inlet and liquid outlet.

$$Re = \frac{ruD}{m} \quad 55 \backslash * \text{MERGEFORMAT } ()$$

Four distinct configurations were applied to the vapor outlet to establish different phase-separation conditions. A non-venting benchmark case, denoted as **NON**, was established by closing the valve at the vapor outlet. Three phase-separation conditions were investigated: **SEP1**, where the vapor outlet is open to atmosphere, resulting in venting driven by the pressure difference between the manifold outlet and atmosphere; **SEP2**, where a moderate vacuum level of ~34 kPa (below atmospheric pressure) is applied at the vapor outlet; and **SEP3**, where a high vacuum level of ~69 kPa (below atmospheric pressure) is applied.

The experimental procedure was initiated in the single-phase regime at a fixed target flow rate. The SEP1 configuration was evaluated first, followed by the NON case by closing the vapor valve. Subsequently, the vacuum pump was activated to test the SEP2 and SEP3 cases. The heat flux was then increased in 25 W increments. Throughout the process, the flow rate was regulated via both the pump frequency and bypass valve. Upon reaching steady state, data were recorded at a frequency of 2 Hz for 2 minutes, and the resulting time-averages were used as the final steady state value. In the cases where significant fluctuations persisted, data were recorded after 60 min of operation.

Experimental uncertainties are evaluated as

$$\frac{dR}{R} = \sqrt{\sum_{i=1}^n \left(\frac{\partial R}{\partial X_i} \frac{dX_i}{R} \right)^2} \quad 66 \backslash * \text{MERGEFORMAT } ()$$

which accounts for the propagation of individual sensor inaccuracies into the calculated parameters [64]. The maximum uncertainties for pressure and pressure drop are $\pm 0.08\%$, as specified by pressure sensors. Based on the measurement uncertainties of the thermocouples embedded within the heater block, the maximum uncertainties for the heat flux and surface temperature are calculated as $\pm 18.8\%$ (at low heat fluxes) and $\pm 1.9\%$, respectively. The maximum uncertainty for the heat transfer coefficient (HTC) is further determined to be $\pm 19.0\%$ (at low heat fluxes). For heat fluxes above 100 W/m^2 , the uncertainties of the heat flux and HTC generally drop below 12%.

4 Results and analyses

To examine the relationship between vapor generation rate at the heated surface and vapor venting capacity through the membrane, the operational regimes and limiting factors of the phase-separation device are first established and determined using a theoretical model. Pressure drop and heat transfer performance are then evaluated experimentally to verify the actual effectiveness of phase separation and to identify the practical gaps compared to the model. Finally, transient data are analyzed to provide a more detailed understanding of the underlying flow dynamics during phase separation in two-phase jet impingement cooling.

4.1 Theoretical operational regimes from vapor generation and venting models

The operational regimes and limiting factors are theoretically identified by analyzing the balance between vapor generation at the heated surface and vapor venting through the membrane. To simplify the physical complexity of the model, the following assumptions need to be adopted:

- (a) Vapor generation is estimated globally based on an overall energy balance, where the subcooled liquid into the impingement cavity needs to be heated above thermodynamic equilibrium quality of zero for boiling to occur, i.e., subcooled boiling is not considered;
- (b) The porous membrane possesses uniform permeability and pore size distribution, and remains strictly non-wetting such that only vapor is permitted to permeate without liquid infiltration;
- (c) The device operates at 100% phase separation efficiency (η), i.e., all generated vapor bubbles can be separated and vented through the membrane.

Under the first assumption, the vapor generation rate ($\dot{V}_{\text{gen}}$, m³/s) can be evaluated for the range of experimental inlet liquid flow rates of 0.06 – 0.14 L/min, calculated as a volumetric rate as

$$\dot{V}_{\text{gen}} = \frac{qA - \dot{V}_l \rho c_p \Delta T}{h_{\text{fg}} \rho} \quad 77 \setminus * \text{MERGEFORMAT } ()$$

The vapor venting capacity ($\dot{V}_{\text{mem}}$, m³/s) is determined based on the second assumption under transmembrane pressures ranging from zero to the breakthrough pressure, using Darcy's law by rearranging

Eq. 2, where the breakthrough pressure of the membrane is calculated using Eq. 1. The third assumption leads to the vapor separation rate of the device ($\dot{V}_{SEP}$, m³/s) ultimately governed by

$$\dot{V}_{SEP}^{***} = \min \left\{ \dot{V}_{gen}, \dot{V}_{mem} \right\} \quad 88 \backslash * \text{MERGEFORMAT } ()$$

where $\dot{V}_{gen}$ and $\dot{V}_{mem}$ are eventually limited by CHF and breakthrough pressure, i.e., $\dot{V}_{gen} \leq \dot{V}_{gen,CHF}$ and $\dot{V}_{mem} \leq \dot{V}_{mem,break}$, respectively.

Fig. 6(a) plots the increasing vapor generation rate with heat fluxes, for different liquid inlet flow rates, as indicated by the different shades of brown inclined lines. At a given heat flux, higher vapor generation rates can be achieved in lower flow rates because less energy is required from subcooled liquid to saturated liquid. However, the maximum vapor generation rate for a given flow rate is in part limited by its maximum operating heat flux, i.e., CHF (estimated using the correlation of Monde and Mitsutake [65]), denoted by red long-dash lines. The CHF is enhanced by increasing the supplied liquid flow rate; as a result, the maximum vapor generation rate is also enhanced for higher flow rates. The blue inclined line in Fig. 6(a) indicates that the vapor venting capacity of the membrane increases with the applied transmembrane pressure, up to the breakthrough pressure denoted by a purple long-dash line. This line also represents the maximum venting capacity of the device under a given transmembrane pressure.

Under the above three assumptions, the limiting factors are elucidated by considering the three operational stages (following along the green dashed lines in Fig. 6(a)) for a given supplied liquid flow rate of 0.14 L/min at a given moderate transmembrane pressure of 37.9 kPa (representing a 3.4 kPa positive system running pressure plus 34.5 kPa vacuum). Initially, the device operates in a single-phase regime (Stage 1) where the heat flux is too low to initiate boiling. Upon reaching the onset of nucleate boiling, it transitions into a fully vented two-phase regime (Stage 2), where the vapor generation rate remains lower than the vapor venting capacity of the membrane. A critical limit is reached when the vapor generation rate equals the vapor venting capacity of the membrane. Beyond this point (Stage 3), the vapor separation rate of the device plateaus even though more vapor is generated due to increasing heat flux; therefore, a two-phase mixture will be forced through the liquid exit until the CHF is reached, risking flow instabilities.

Consequently, the overall vapor separation performance limit can be enhanced either by increasing the transmembrane pressure to raise the vapor venting capacity of the membrane, or by increasing the supplied liquid flow rate to extend the maximum vapor generation rate, with the optimal strategy being dependent on the specific operating conditions.

From the above assumptions and analysis, a theoretical operational regime map based on vapor generation and venting can be developed to identify the limiting factors under different operating conditions, as shown in Fig. 6(b). At a given supplied liquid flow rate, the green inclined solid line represents the condition where the vapor venting capacity equals the vapor generation rate, meaning that the transmembrane pressure perfectly matches the heat flux under the assumption of 100% phase separation efficiency. Beyond this line, the blue region indicates a vapor-generation-limited regime according to Eq. 8. In contrast, the brown region below this line represents a vapor-venting-limited regime, where the vapor venting capacity is lower than the vapor generation rate. Both regimes are ultimately constrained by the breakthrough pressure limit (purple long-dash line) and CHF limit (red long-dash line).

However, in practice, achieving 100% phase separation efficiency during flow boiling is impossible, as some generated vapor will always travel with the liquid toward the liquid outlet rather than pass through the membrane. An illustrative generation-venting matching curve under imperfect phase separation efficiency is drawn as the green dotted line, where $\dot{V}_{\text{mem}} = \eta \dot{V}_{\text{gen}}$ and phase separation efficiency $\eta < 1$. Under this imperfect efficiency, as shown in Fig. 6(b), the vapor-venting-limited regime dominates most of the two-phase operating regime, which agrees well with the experimental results presented later in the following sections, where increasing transmembrane pressure consistently enhances vapor separation performance. Further studies are still needed to explore these characteristics under imperfect phase separation efficiency.

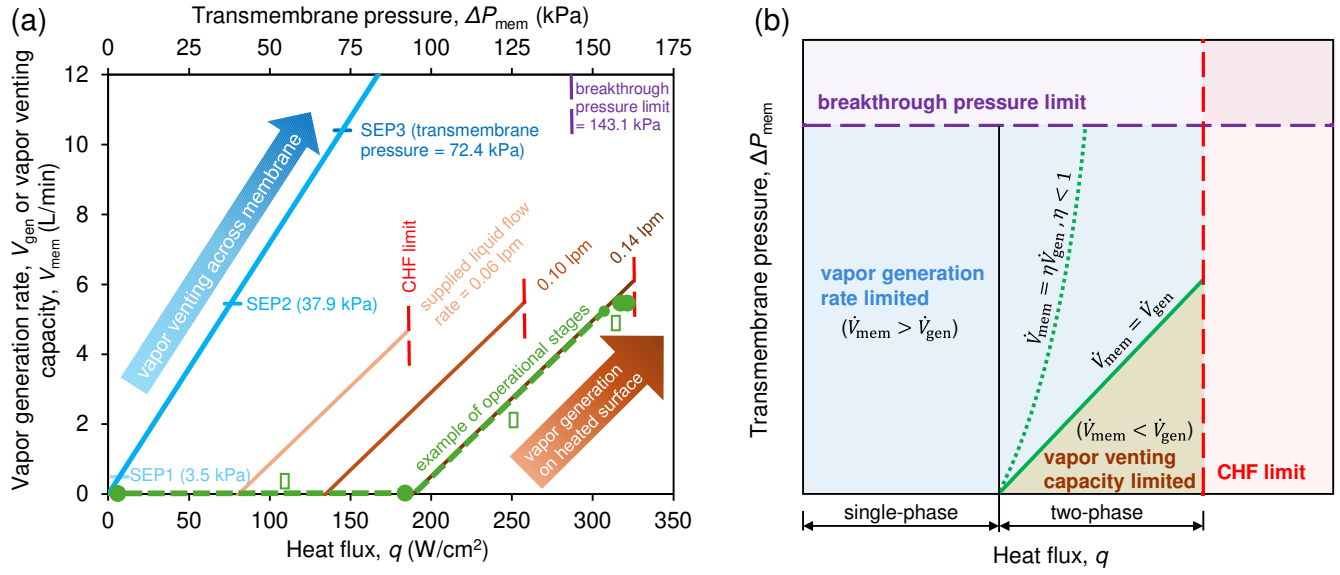


Fig. 6 Operational regimes and limiting factors governing phase separation of the cooling approach: (a) correlating the vapor generation rate and vapor venting capacity as a function of the heat flux, supplied liquid flow rate, and transmembrane pressure; (b) operational regime map at a given flow rate, including a vapor generation rate limited regime and vapor venting capacity limited regime, bounded by the breakthrough pressure limit and CHF limit.

4.2 Pressure drop characteristics

The pressure drop across the manifold (from the liquid inlet to liquid outlet) and its oscillation (quantified by the sample standard deviation, σ , defined in Eq. 9) are illustrated in Fig. 7 as a function of applied heat flux under varying supplied liquid flow rates for all of the separation cases. In the single-phase regime, the pressure drop for all four separation configurations remains low and stable, remaining at ~0.9, 1.8, and 3.8 kPa under 0.06, 0.10, and 0.14 L/min, respectively, as shown in Fig. 7(a)–(c), with minimal oscillations observed in Fig. 7(d)–(f). Upon the onset of nucleate boiling, the pressure drop increases due to vapor generation. Applying phase separation reduces pressure drop in varying degrees depending on the operating conditions. Specifically, this reduction becomes more pronounced at lower flow rates and higher transmembrane pressures.

$$\sigma = \sqrt{\frac{1}{N-1} \sum_{i=1}^N (x_i - \bar{x})^2}$$

99* MERGEFORMAT ()

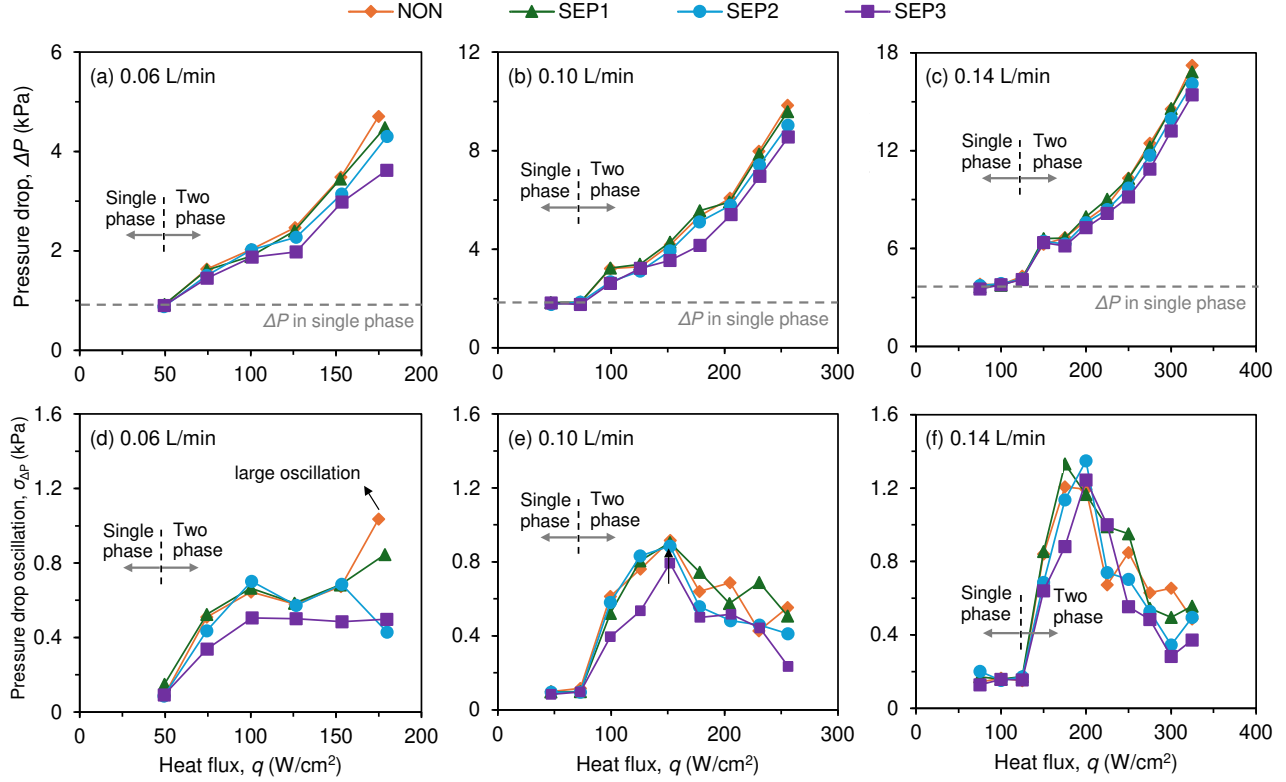


Fig. 7 Pressure drop characteristics at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min; and pressure drop oscillations at (d) 0.06 L/min, (e) 0.10 L/min, and (f) 0.14 L/min, for four separation cases: NON, SEP1, SEP2, and SEP3.

A more detailed pressure drop comparison between the NON and SEP cases is presented in Fig. 8, utilizing the relative pressure drop, defined as the total pressure drop divided by the corresponding single-phase value, to evaluate the two-phase pressure drop contribution across different flow rates. The results demonstrate that phase separation effectively reduces the pressure drop during two-phase operation, with the effect becoming more pronounced at higher transmembrane pressures. At the maximum steady-state heat flux, the pressure drop reduction from NON to SEP3 is 23.0%, 13.0%, and 10.5% for flow rates of 0.06, 0.10, and 0.14 L/min, respectively.

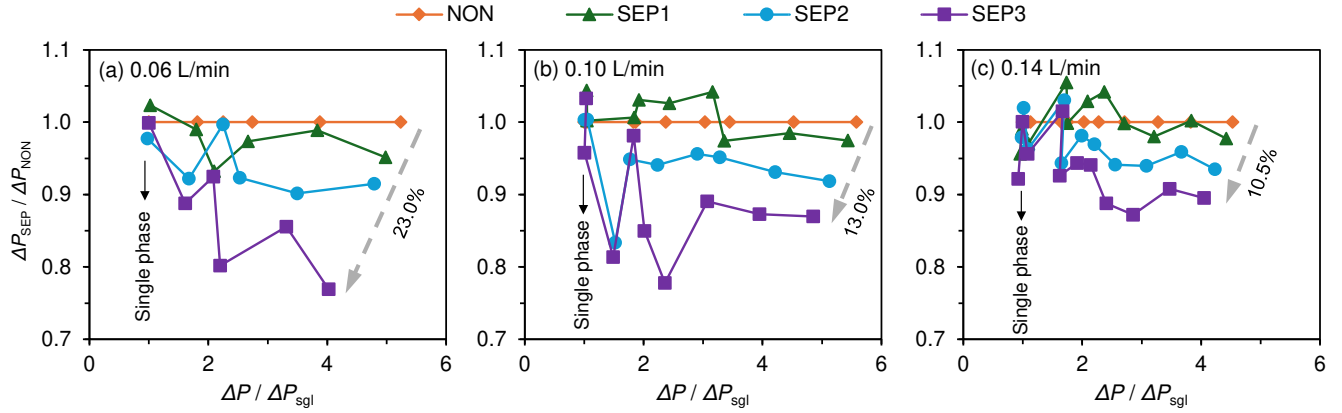


Fig. 8 Pressure drop reduction achieved through phase separation, presented as a function of relative pressure drop at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min, for four separation cases: NON, SEP1, SEP2, and SEP3

It is noteworthy that SEP1 shows little improvement over NON. This is attributed to condensation within the vapor cavity, caused by the large temperature difference between the water saturation temperature and the ambient air. The accumulated condensate blocks membrane pores, severely impeding vapor venting. In contrast, applying a vacuum (SEP2 and SEP3) not only creates a larger driving force for vapor venting, but also removes vapor from the manifold, thereby mitigating condensation. It is also important to clarify that achieving 100% separation efficiency is unrealistic in this jet impingement cooling configuration, as the liquid outlet provides an alternative path for the liquid carrying generated bubbles. Consequently, while the theoretical analysis in Fig. 6 suggests the limiting factor for SEP2 and SEP3 is the vapor generation rate, practical performance in experiments is still enhanced by a higher transmembrane pressure due to more efficient vapor removal (i.e., increasing the proportion of vapor passing through the membrane rather than traveling with the liquid flow).

The pressure oscillation significantly rises in the partial boiling regime before leveling off (at a lower flow rate in Fig. 7(d)) or decreasing (Fig. 7(e)–(f)) in the fully boiling regime. Phase separation demonstrates a significant dampening effect on these oscillations, particularly at the low flow rate of 0.06 L/min, where lower fluid momentum and higher vapor quality make the system more susceptible to instability. The active vapor removal in this regime stabilizes the flow, leading to a lower and steadier

pressure drop for the SEP cases compared to the NON case. This effect is analyzed in more detail through inspection of the transient data in Section 4.4.

Fig. 9 further reveals a positive proportional relationship between the relative pressure drop and the local vapor quality, which indicates that the increasing volume of generated bubbles is the cause of pressure drop rise in the two-phase regime. The local vapor quality x is calculated using Eq. 10, based on the heat flux, flow rate, latent heat, and the sensible heating for the subcooled supplied liquid under thermal equilibrium assumption. The average vapor quality within the impingement cavity and the exit vapor quality at the liquid outlet for phase separation cases are expected to be lower than this calculated local vapor quality, as part of vapor is vented through the membrane situated between the inlets and outlets. Although the vapor quality is less than 0.04 in most cases, the large density ratio (~ 1600) between liquid water and water vapor results in a substantial vapor volume fraction (shown in Fig. 6(a)), demonstrating the significant impact of even small amounts of vapor and the corresponding benefit of phase separation.

$$x = \frac{qA / r + \dot{m}_l c_p \Delta T}{\dot{m}_l h_{fg}}$$

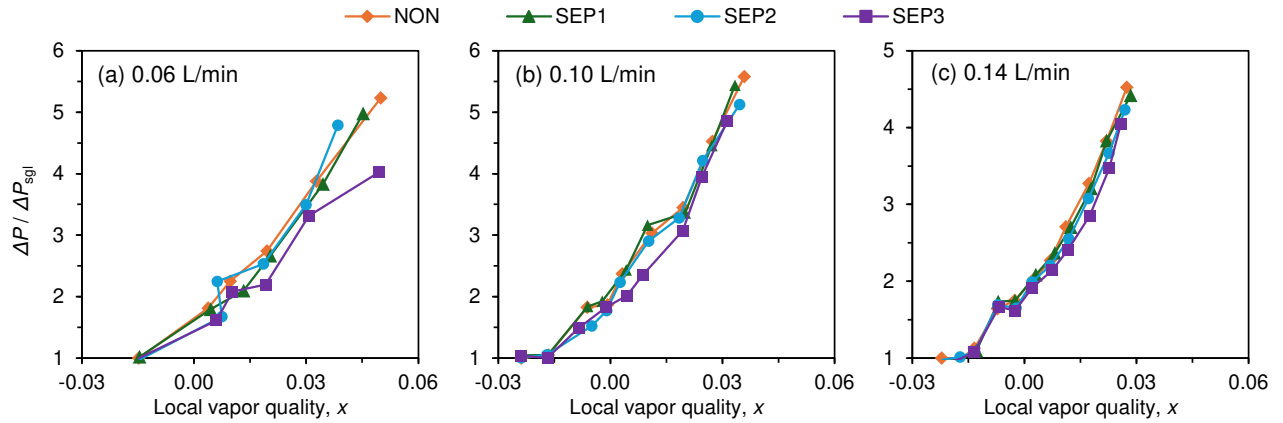


Fig. 9 Relative pressure drop with respect to local vapor quality at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min, for four separation cases: NON, SEP1, SEP2, and SEP3.

4.3 Heat transfer characteristics

The boiling curves plotting the dissipated heat flux as a function of the surface superheat are presented in Fig. 10 (a)–(c) for each of the test flow rates. The data generally indicate that phase separation has a

limited effect on the average steady-state surface temperature in most cases. A notable exception is observed at the maximum steady-state heat flux for the 0.06 L/min case, where phase separation reduces the surface superheat by 6.7 °C comparing NON to SEP3. Consistent with findings for microchannel cooling with phase separation membranes by David et al. [41], the application of phase separation in this jet impingement system does not alter the CHF for any given flow rate. This result contrasts with phase separation configurations that feature no liquid outlet [49], where a significant CHF increase is achieved. In such configurations with no liquid outlet, the bubble-liquid interface is subjected to an omnidirectional pressure that physically limits bubble growth and coverage of the heated surface. Conversely, in flow boiling configurations such as the present jet impingement study or for microchannels, bubbles are not mechanically constrained and can freely expand and coalesce near the surface, ultimately leading to dryout under the same hydrodynamic limits.

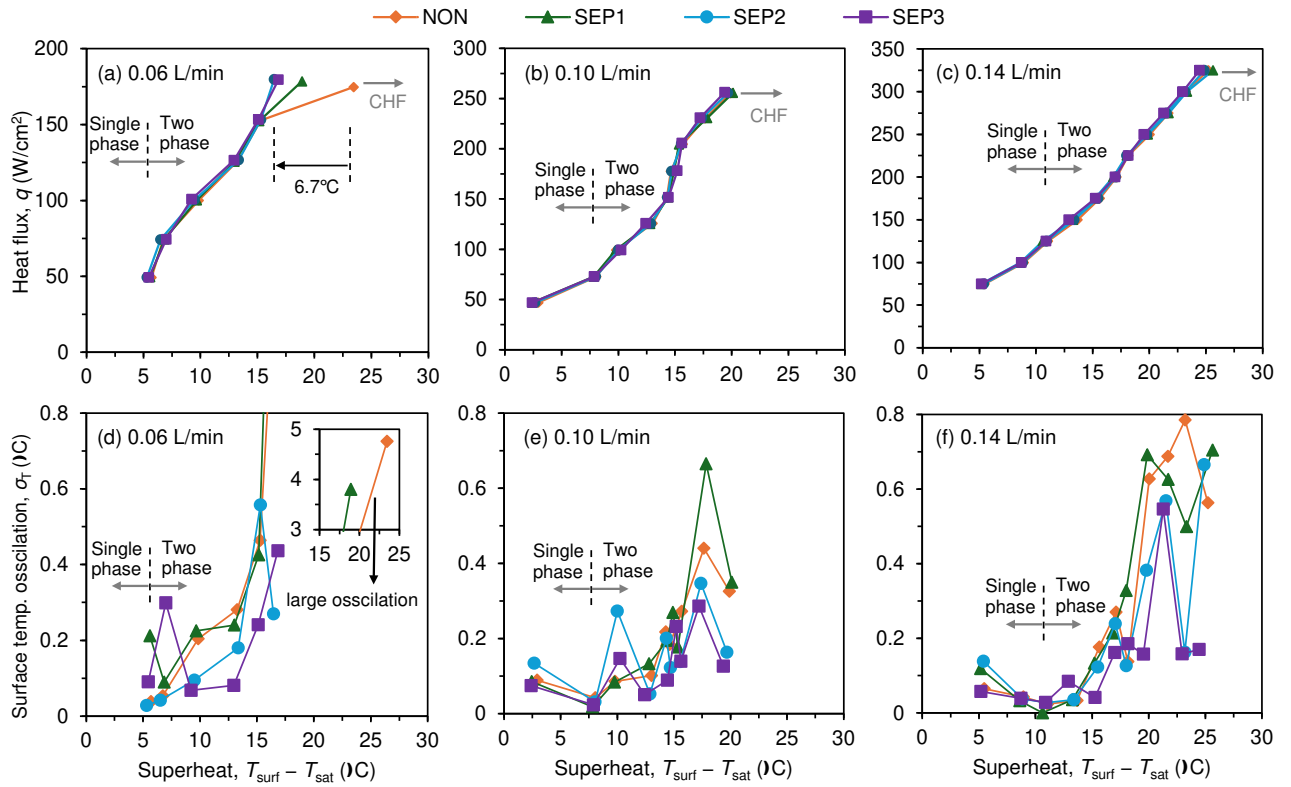


Fig. 10 Heat transfer characteristics at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min; and surface temperature oscillations at (d) 0.06 L/min, (e) 0.10 L/min, and (f) 0.14 L/min, for four separation cases:

NON, SEP1, SEP2, and SEP3.

While phase separation does not affect the average steady surface temperature in most cases, it does have a more noticeable impact on the thermal dynamics. For all cases, the surface temperature oscillation generally exhibits an increasing trend with the wall superheat, as shown in Fig. 10 (d)–(f). The application of phase separation proves highly beneficial in dampening these thermal oscillations, particularly at higher heat fluxes where pressure instabilities are more pronounced. For the 0.06 L/min case, the surface temperature oscillation is reduced from 4.76 °C for NON to 0.44 °C for SEP3. More details will be depicted using transient data in Section 4.4.

A comparison of the heat transfer coefficient (HTC, W/m²·K) defined by Eq. 11 is provided in Fig. 11. The inlet temperature is used as the reference temperature to maintain consistency in HTC calculations across both single-phase and two-phase regimes. The calculated HTC would be approximately 2–3 times higher if the saturation temperature were used as the reference temperature instead. This comparison in Fig. 11 leads to similar conclusions as the boiling curves, where HTC is almost identical in most cases of different separation conditions except for extreme conditions. For the 0.06 L/min flow rate and ~175 W/cm² heat flux, the NON case achieves an HTC of ~33,000 W/(m²·K) at a superheat of 23.4 °C. In contrast, SEP3 achieves a 14.7% higher HTC of ~37,800 W/(m²·K) at a significantly lower superheat of 16.7 °C.

$$HTC = \frac{q}{T_{\text{surf}} - T_n} \quad 1111 \backslash * \text{MERGEFORMAT } ()$$

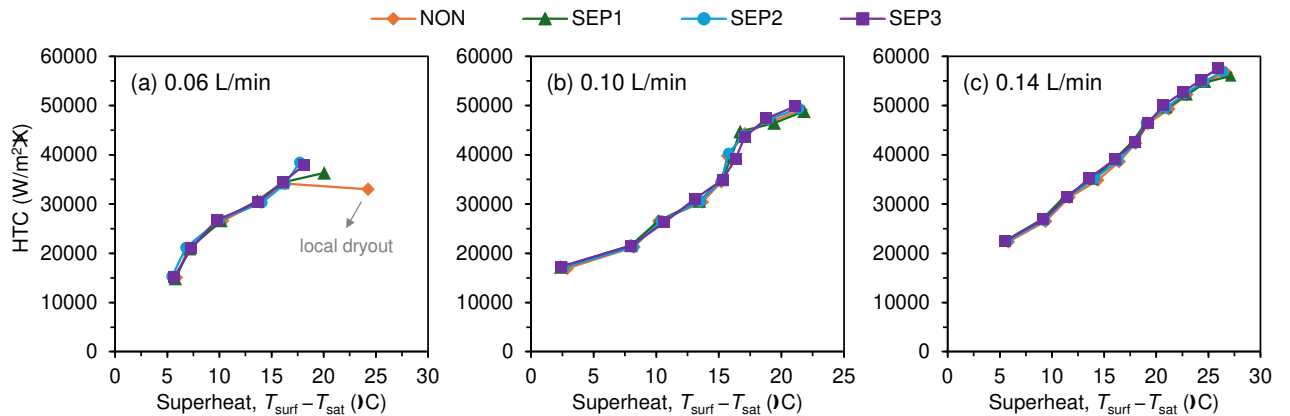


Fig. 11 Heat transfer coefficient with respect to superheat at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min, for four separation cases: NON, SEP1, SEP2, and SEP3.

4.4 Transient thermal-hydraulic behavior

As two-phase flow visualization is challenging in such a confined jet impingement manifold with a phase-separation membrane, the underlying mechanisms of phase separation are inferred through the analysis of transient measurement data. Fig. 12 exhibits the transient pressure drop signals at selected heat flux levels corresponding to the maximum steady-state heat flux, a moderate heat flux within the nucleate boiling regime, and in single-phase.

At the maximum steady-state heat flux (Fig. 12(a-1)), a clear periodic pressure drop oscillation is observed without phase separation (NON), with an average pressure drop of 4.70 kPa and oscillation of 1.036 kPa. The SEP1 configuration shows a marginal improvement, because phase separation remains limited due to the low transmembrane pressure and condensation issue discussed above. In contrast, SEP2 and SEP3, which utilize the vacuum pump to increase the transmembrane pressure, demonstrate significant performance and stability gains, reducing the average pressure drop by 8.5% and 23.0%, and reducing the oscillation by 58.6% and 51.9%, respectively. This trend of pressure drop and oscillation reduction is consistently observed across the flow rates, as depicted in Fig. 12(b-1) and (c-1).

At a moderate heat flux in the nucleate boiling regime (Fig. 12(a-2), (b-2), (c-2)), the oscillation period appears to be shorter compared to the maximum heat flux condition. For these cases, the most advantageous phase separation configuration (SEP3) effectively lowers the average pressure drop by 19.5%, 15.1%, 7.5%, and reduces the oscillation by 13.3%, 13.6%, 26.9% compared to the NON cases, at flow rates of 0.06, 0.10, and 0.14 L/min, respectively. This behavioral shift across boiling regimes is attributed to the underlying vapor dynamics. At this moderate heat flux, the higher frequency oscillation is likely associated with individual bubble growth and departure, and such high-frequency events are less significant as they do not trigger large flow instability and severe local dryout. However, as the heat flux increases toward the maximum condition, vapor agglomeration likely leads to longer-period oscillation. Phase separation is most effective in this latter regime, as the membrane-based venting actively removes the

accumulated vapor, preventing the formation of large vapor structures and thereby stabilizing the flow. When the heat flux decreases to the single-phase regime, as expected, all SEP configurations perform almost identically to NON, as no vapor is generated to be separated.

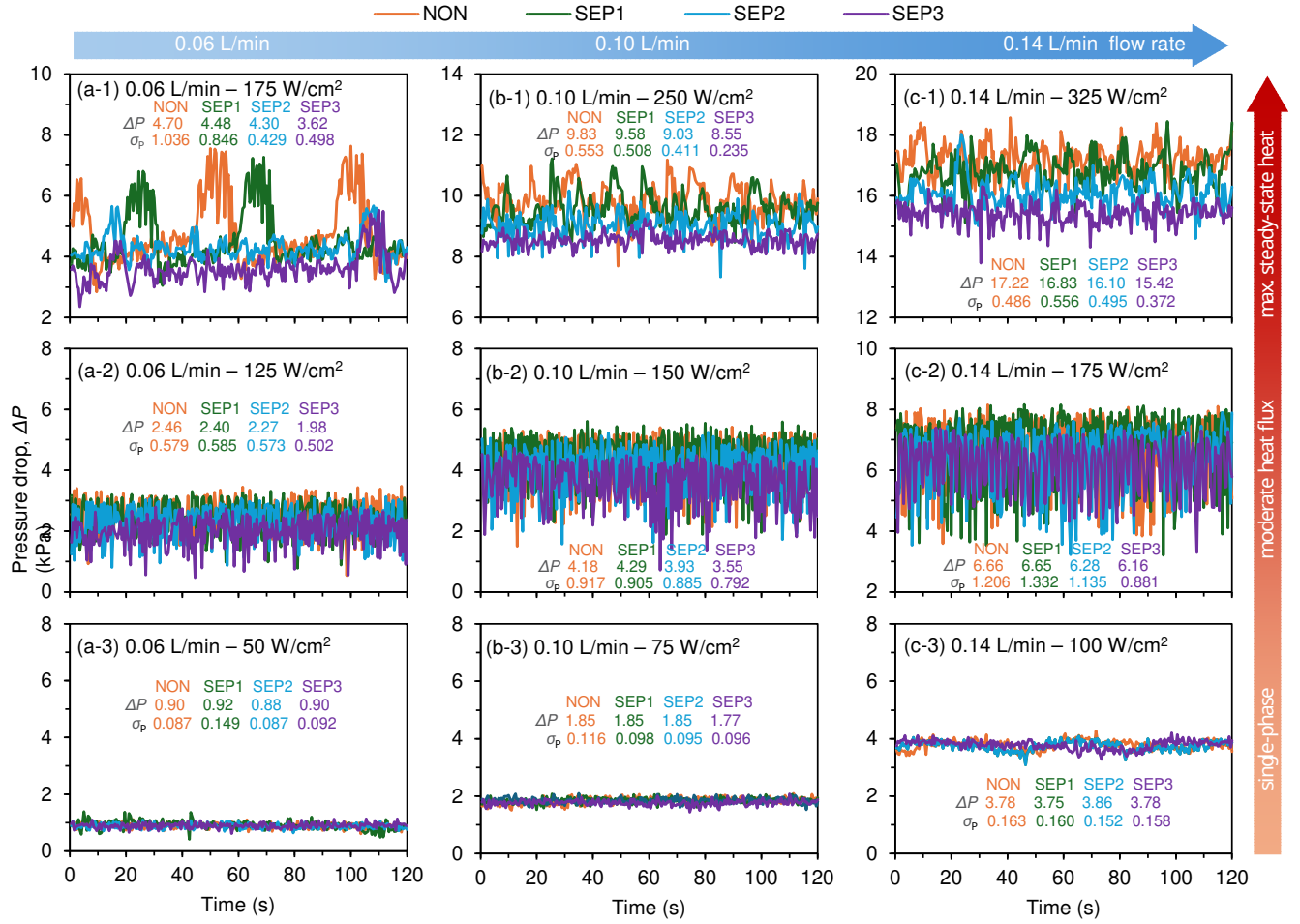


Fig. 12 Transient pressure drop data at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min. For each flow rate, the data are categorized into three regimes: at the maximum steady-state heat flux ((a-1), (b-1), (c-1)), a moderate heat flux in the nucleate boiling regime ((a-2), (b-2), (c-2)), and for a low heat flux in the single-phase regime ((a-3), (b-3), (c-3)).

Fig. 13 presents the transient data of the surface temperature. At the maximum steady-state heat flux for the 0.06 L/min case (Fig. 13(a-1)), a significant temperature excursion is observed for NON and SEP1. These spikes are indicative of local dryout, where vapor bubbles coalesce and temporarily clog the impingement cavity, preventing liquid from reaching the heated surface. In contrast, the application of a vacuum pump in SEP2 and SEP3 effectively suppresses this dryout, maintaining a stable surface temperature. Particularly, SEP3 reduces the surface temperature by 6.7 °C and its oscillation by 90.8%. A

similar stabilizing effect is evident at higher flow rates, as shown in Fig. 13(b-1) and (c-1). Notably, the oscillation frequency is higher in these higher flow rates, as the higher liquid momentum accelerates the cycle of vapor clogging and pressure build-up before releasing. By actively venting these vapor slugs, the SEP3 configuration reduces the surface temperature by 0.6 °C and 1 °C, and reduces the oscillation by 61.2% and 69.8%, for 0.10 and 0.14 L/min, respectively. This highlights phase separation to prevent large-scale vapor agglomeration, ensuring a more continuous liquid-to-surface contact.

At a moderate heat flux in the nucleate boiling regime (Fig. 13(a-2), (b-2), (c-2)), phase separation effectively suppresses temperature oscillations while the mean surface temperature remains largely unchanged. This occurs because, although vapor is actively removed, the fundamental boiling mechanism remains unaltered; even without phase separation nucleating bubbles remain unconstrained and can still freely expand, coalesce near the surface, and flow out of the cavity. As anticipated, in the single-phase regime, the surface temperature profiles for all configurations are nearly identical, with minor variations attributable only to slight fluctuations in the inlet temperature.

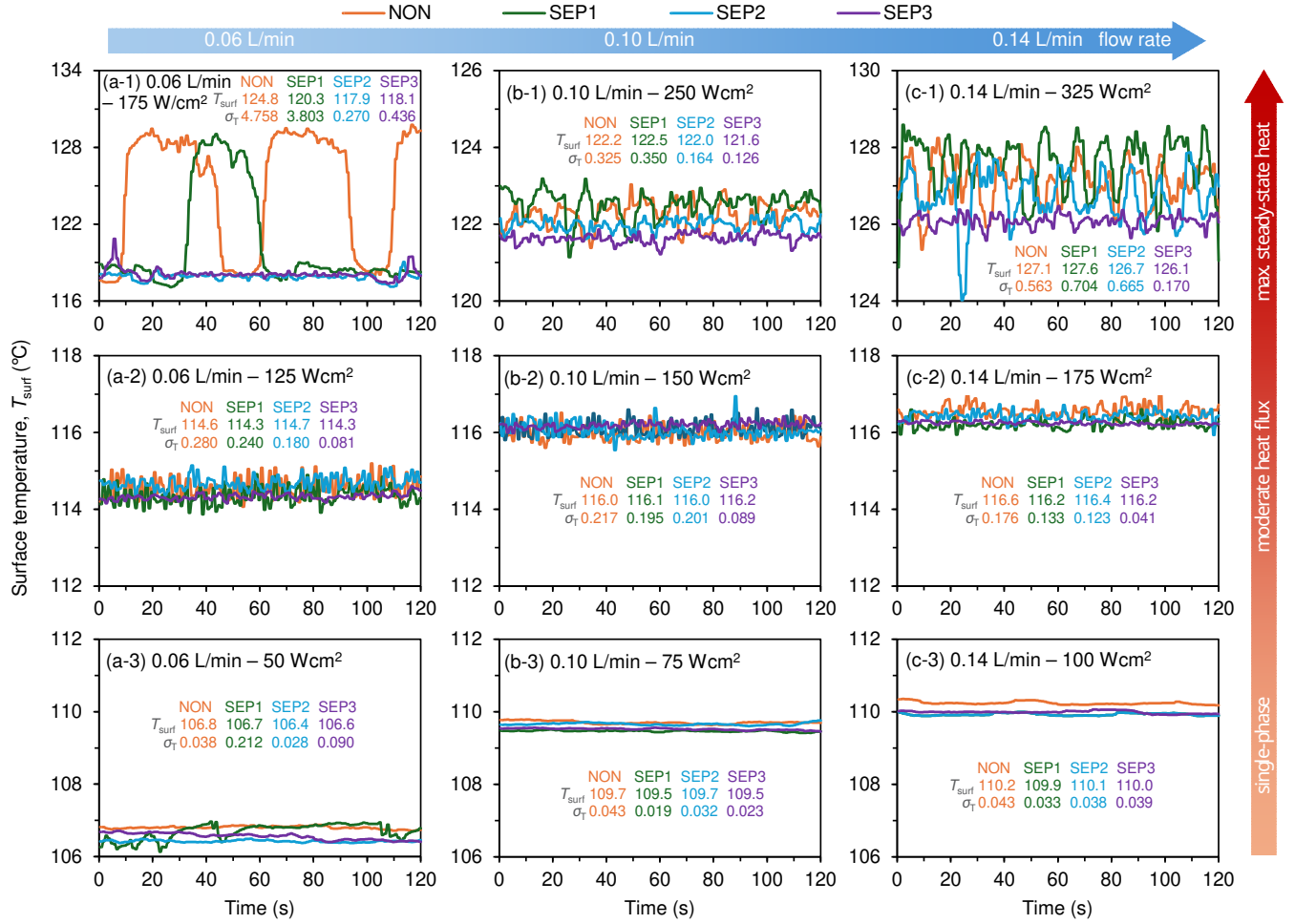


Fig. 13 Transient surface temperature data at (a) 0.06 L/min, (b) 0.10 L/min, and (c) 0.14 L/min. For each flow rate, the data are categorized into three regimes: at the maximum steady-state heat flux ((a-1), (b-1), (c-1)), a moderate heat flux in the nucleate boiling regime ((a-2), (b-2), (c-2)), and for a low heat flux in the single-phase regime ((a-3), (b-3), (c-3)).

Analysis of the transient data reveals distinct performance impacts of phase separation across different operational regimes. As expected, no performance improvement is observed in the single-phase regime, as no vapor is generated to be separated. However, in the two-phase regime, phase separation significantly reduces oscillations in both the pressure drop and surface temperature by actively removing vapor, thereby enhancing flow stability. The reduction in pressure drop becomes more pronounced from the partial to the fully developed boiling regime. This trend is attributed to the higher vapor generation rates at elevated heat

fluxes, which allow for a greater absolute amount of vapor to be vented through the membrane, thereby further reducing the vapor volume fraction within the flow passages. Furthermore, the shift toward lower frequencies and higher amplitudes at elevated heat fluxes indicates vapor agglomeration. In this regime, discrete bubbles coalesce into large-scale structures that trigger flow instability and local dryout. The phase-separation membrane mitigates these effects by continuously venting vapor to disrupt coalescence. In contrast, the higher frequencies at lower heat fluxes represent individual bubble dynamics, where the impact of phase separation is naturally less pronounced.

The mean surface temperature is largely unaffected by phase separation in most operating conditions, except when it prevents local dryout. This is attributed to two primary design factors. First, since vapor removal occurs at the upper confinement wall, it is essential to determine whether the phase-separation membrane interferes with bubble growth and departure. To evaluate this, the bubble departure diameter (D_b , m) is calculated using the correlation proposed by Abdelfattah [66] for the stagnation zone and the correlation by Omar [67] for the wall jet region. Although these correlations were originally developed for free jet impingement and the operating conditions do not fully match those of the present study, they can still provide meaningful guidance when the confinement gap is not extremely small, since no well-established correlations for confined jet impingement are currently available.

As illustrated in Fig. 14, due to the jet momentum in the stagnation zone, the bubble departure diameter remains consistently lower than the jet height despite increasing with surface superheat. In the wall jet region, the bubble departure diameter is typically larger and increases more significantly with increasing flow rate compared to those in the stagnation zone. At a low flow rate of 0.06 L/min and superheats above 5 °C, the bubble departure diameter exceeds the jet height. This indicates the confinement wall physically constrains bubble departure under these specific conditions. At high heat fluxes, where massive bubble generation leads to vapor accumulation within the impingement cavity, putting the phase-separation membrane effectively prevents flow blockage, as shown in Fig. 12(a-1) and Fig. 13(a-1).

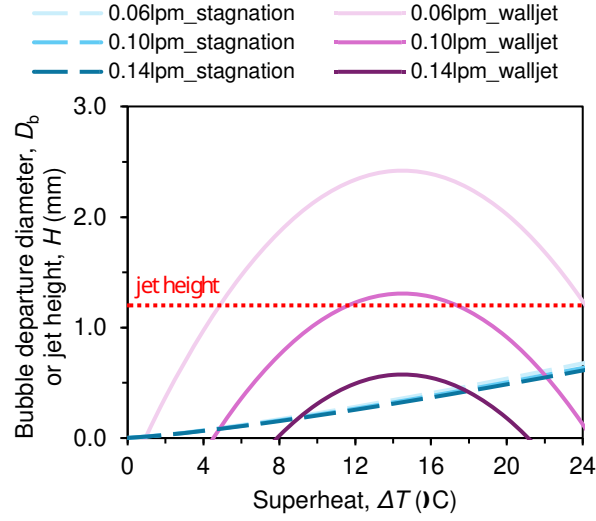


Fig. 14 Bubble departure diameter determined by Abdelfattah's correlation [66] for the stagnation zone and by Omar's correlation [67] for the wall jet region.

The second factor is that the distributed inlet-outlet manifold design inherently shortens the flow path above the heated surface, shown in Fig. 15(a), and heated fluid and bubbles are efficiently evacuated through the nearest outlet, minimizing their impact on the downstream surface temperature. The critical exception occurs at low flow rates and high heat fluxes, where large, stable vapor structures can form, as illustrated in Fig. 15(b). For water, the large capillary length scale (2.5 mm @ 1atm and 100 °C) compared to other dielectric fluids (1.0 mm @ 1atm and 61 °C for HFE-7100) results in corresponding large vapor bubble departure diameters. Under these conditions, a bowl-like vapor structure can bridge the confinement gap, periodically isolating the surface from the liquid and interfering with jet impingement [34]. Besides, as shown in Fig. 15(c), these structures tend to be stable in the center of the unit cell, where the local momentum of the wall jet is lowest, facilitating vapor accumulation. This phenomenon leads to local dryout and thermal instabilities. Phase separation directly mitigates this by disrupting the formation of these macroscopic vapor structures. At higher flow rates, the inherent jet momentum is sufficient to break up the vapor and rewet the surface, rendering the system less susceptible to this failure mode. Future flow visualization needs to be developed to further verify this mechanism for this confined case.

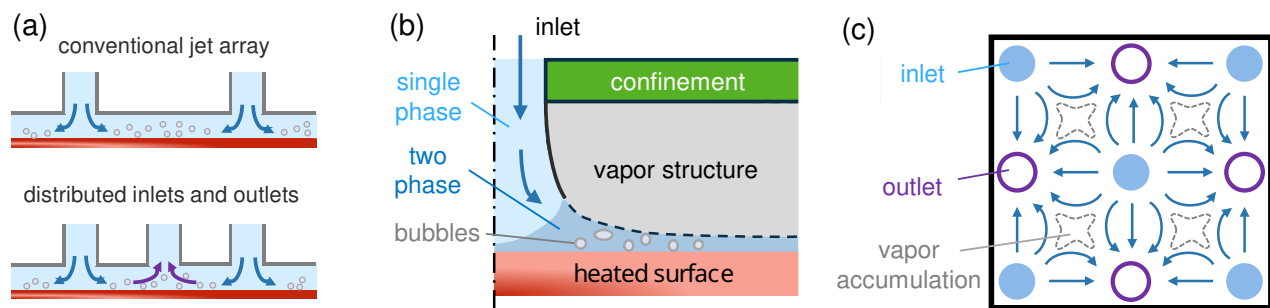


Fig. 15 Flow instability mechanisms occurring during two-phase jet-impingement: (a) distributed outlets provide additional flow paths between inlets; (b) a large amount vapor accumulates in the confinement gap to form a vapor structure during high heat flux operation; (c) flow direction using the designed manifold and vapor accumulation area within impingement cavity.

5 Discussion

5.1 Membrane condensation mitigation and structural durability

A significant challenge in membrane-based phase separation is condensation. After vapor passes through the membrane and moves away from the heated surface, its temperature is prone to fall below the saturation point. If this leads to the formation of condensate droplets on or within the hydrophobic membrane, the water will become trapped in place and block the membrane pores. This issue is particularly acute when using water at ambient pressure, where the large temperature gradient relative to the ambient environment promotes condensation within the vapor cavity. This phenomenon was directly observed in the SEP1 configuration, where venting is driven by the pressure difference between the manifold outlet and atmosphere. Visible vapor flux from the outlet diminished significantly after approximately 20–40 min of operation and eventually ceased, indicating pore blockage. While previous studies have used sweep gas streams [39,43] or vacuum pumps [54] to mitigate this, our results confirm that active vapor removal via vacuum (SEP2, SEP3) is essential for sustained performance. For future practical implementations, design changes or strategies to avoid membrane condensation or facilitate condensate drainage should be sought out.

Given the harsh operating conditions of high temperature and transmembrane pressure, the durability

of the tested PTFE membrane was assessed. SEM images of the membrane before and after ~24 hours of testing are shown in Fig. 16. The post-test images indicate a slight reduction in pore size and a flattening of the surface morphology, suggesting potential mechanical compaction or minor degradation. Despite these observable changes, the membrane remained functional, demonstrating its robustness under these demanding conditions.

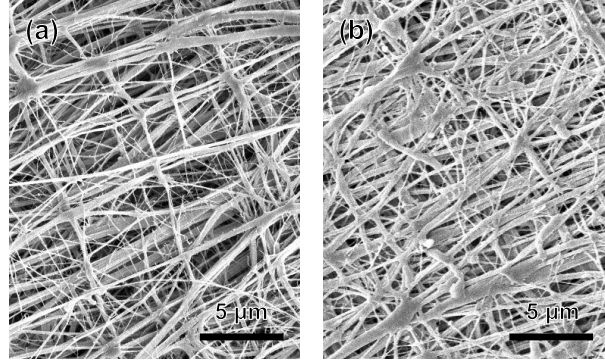


Fig. 16 SEM images of the PTFE membrane (a) before and (b) after ~24 hours of testing

5.2 Phase separation features across cooling architectures

The phase-separation-enhanced jet impingement cooling system developed in this study represents a distinct configuration within the spectrum of two-phase cooling technologies. A comparative summary of different membrane-based two-phase cooling architectures is provided in Table 2. In contrast to membrane-assisted heat sinks that achieve complete vapor exit through a single vapor outlet [49], the present design maintains conventional flow boiling characteristics with separate liquid and vapor outlets, similar to other phase-separation microchannel systems [41]. This solution preserves the fundamental advantages of flow boiling, fully utilizing both nucleate boiling mechanisms and forced convection from the continuous supply of subcooled liquid for efficient heat removal. The fundamental compromise in this configuration is the inherent limitation in achieving complete vapor separation (100% phase separation efficiency) due to the competing liquid outlet pathway.

Table 2 Comparison of membrane-based two-phase cooling architectures

	Microchannel	Single-outlet heat sink	Jet impingement
Manifold configuration	1 inlet & 2 outlets	1 inlet & 1 outlet	1 inlet & 2 outlets

Flow characteristic	unidirectional parallel flow	pressure-driven liquid replenishment	vertical stagnation flow & radial wall-jet flow
Vapor expansion behavior	confined by channel walls	restricted by surrounding liquid	constrained by jet momentum and wall shear
Heat transfer mechanism	convection + boiling	boiling	convection + boiling
Phase separation efficiency	< 1	1	< 1

Furthermore, the hydrodynamics of jet impingement introduce unique design considerations compared to microchannel systems. In microchannels, the liquid flow and vapor separation occur in perpendicular directions. In jet impingement, both liquid impingement and vapor separation occur along the vertical axis, creating a design tradeoff between the area available for the nozzles versus the membrane. Additionally, the presence of a single-phase stagnation zone directly beneath each jet creates a fundamentally different two-phase distribution. Consequently, the optimal distribution of nozzle area and membrane area requires a distinct design perspective from microchannels. A promising modeling approach, as suggested in [68], would be to treat the unit cell of each jet as being divided into a single-phase stagnation region and a surrounding boiling region, coupling this with the vapor venting through the overlying membrane. As the current manifold structure was designed primarily to probe these underlying mechanisms rather than to achieve peak performance, such models can be used to optimize architectural parameters in the future.

6 Conclusions

This study proposes a direct-on-chip confined two-phase jet impingement cooling integrated with a phase-separation membrane. An operational regime map is developed based on the vapor generation rate and vapor venting capacity to identify the theoretical limiting factors. The pressure drop and heat transfer performance of a staggered multi-jet array manifold with distributed inlets and outlets were experimentally investigated, considering the effects of supplied liquid flow rate (0.06, 0.10, 0.14 L/min) and transmembrane pressure (NON, SEP1, SEP2, SEP3). The main conclusions are summarized as follows:

- (1) While theoretical analysis identifies performance limits assuming 100% phase separation

efficiency, experimental results demonstrate that increasing the transmembrane pressure consistently enhances separation performance. This consistent improvement is attributed to more effective vapor removal, which mitigates practical separation limitations caused by the inherent competition between vapor venting and liquid discharge in flow. And it accelerates vapor removal from the vapor cavity, reducing vapor condensation on the membrane surface within the manifold.

- (2) In both the partial and fully developed boiling regimes, phase separation significantly reduces both the magnitude and oscillation of the pressure drop across the manifold. At the maximum steady-state heat flux, SEP3 achieves pressure drop reductions of 23.0%, 13.0%, and 10.5%, and reduces pressure drop oscillations (sample standard deviation) by 51.9%, 57.5%, and 23.5% compared to NON, for flow rates of 0.06, 0.10, and 0.14 L/min, respectively.
- (3) Phase separation effectively suppresses local dryout caused by the accumulation of macroscopic water vapor structures in extreme conditions. The membrane, serving as a porous confinement wall, provides an alternative path for vapor to vent directly from the heated surface, preventing vapor from becoming trapped within the impingement cavity or obstructing the liquid flow path.
- (4) At the maximum steady-state heat flux of $\sim 175 \text{ W/cm}^2$ for the 0.06 L/min case, where local dryout occurs because of vapor agglomeration, as evidenced by the transient data, SEP3 reduces the surface temperature by up to $6.7 \text{ }^\circ\text{C}$ (28.6% reduction in superheat) and diminishes the temperature oscillation by 90.8% from $4.76 \text{ }^\circ\text{C}$ to $0.44 \text{ }^\circ\text{C}$ compared to NON.

This work provides experimental validation of the pressure drop reduction and flow stability enhancement achievable through phase separation in multi-jet two-phase impingement cooling. For chip cooling applications, in practice, this strategy can be implemented at sub-ambient pressures to maintain the device below its maximum operating temperature. Future research could focus on elucidating the underlying mechanisms through flow visualization and developing predictive models to further optimize the device design for maximized dryout limits.

Acknowledgement

The authors would like to express their sincere gratitude to Prof. David M. Warsinger, Md Ashiqur Rahman, Dr. Yingru Song, and Prof. Chang-Jin Kim for their invaluable insights into vapor venting behaviors through membrane. Special thanks to Ian Quan, Ye Yang, and Shuhang Lyu for their assistance in experiments and acquiring the SEM images.

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