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Contents lists available at SciVerse ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Hydraulic and thermal characteristics of a vapor venting two-phase microchannel heat exchanger

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ARTICLE INFO

Article history:

Received 21 October 2010

Accepted 19 July 2011

Available online 18 August 2011

Keywords:

Vapor separation

Microchannel heat exchanger

Two phase

Flow boiling

Porous hydrophobic membrane

ABSTRACT

In the present work we design, model and experimentally characterize a two-phase vapor venting parallel microchannel heat exchanger capped with a 220 nm pore, hydrophobic PTFE membrane that vents the vapor phase into separate vapor transport channels. We compare the performances of a traditional non-venting heat exchanger and the vapor-separating version operating at heat fluxes of up to 820 kW/m² and water mass fluxes of between 102 and 420 kg/s m². We find ~60% improvement in the normalized pressure drop and up to 4.4 °C reduction in the average substrate temperature between the control and vapor venting device under similar operating conditions.

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1. Introduction

Effective thermal management is essential for the development of next generation computational systems. The International Technology Roadmap for Semiconductors (ITRS) predicts total power dissipation requirements of over 200 W per package, junction temperatures less than 70 °C and thermal resistances less than 0.2 °C/W [1]. Reliability, size, noise and power consumption of the cooling solution also need to be considered for practical applications. The introduction of 3D IC architectures also requires that the cooling solution be integrated within the die stacks [2,3]. Traditional cooling solutions such as fans and heat pipes are reaching their limit for providing high performance cooling while meeting the variety of geometrical and cost constraints.

Liquid cooling using microfabricated structures is promising owing to small dimensions and very high thermal conductances as demonstrated by Tuckerman and Pease's seminal work using forced liquid convection in silicon microchannels to dissipate 790 W/cm² and obtain a thermal resistance of less than 0.1 °C/W [4]. However, single-phase liquid cooling using microchannels [4–7] require large pressure heads to deliver the coolant at a rate such that the temperature non-uniformity across the chip, caused due to sensible heating of the fluid, is maintained below a required level. The large pressure head requires larger pumps and more

power consumption, which increases the size and cost of the overall system. One widely researched solution to reduce this pressure head while providing large heat flux dissipation is two-phase convection in microchannels [8–11]. The large latent heat of vaporization enables comparably smaller flow rates than single-phase counterparts. Phase change also provides high heat transfer coefficients, leading to smaller thermal resistances, and may also improve temperature uniformity by maintaining the working fluid at the saturation temperature. Despite these advantages and the ease of fabrication of microchannels in a variety of substrates, two-phase cooling faces major technical challenges that limit commercial application.

The growth and advection of vapor bubbles in microchannels leads to an increase in the pressure-drop due to the added friction of the two-phase flow and the acceleration of the fluid during vaporization. This increase in pressure drop during two-phase flow in a variety of tubes and channels has been extensively studied and the fundamental aspects are discussed in several texts [12–14]. The large increase in the two-phase pressure drop with the vapor quality necessitates a larger, more power consuming pump, negating some of the benefit of using two-phase cooling over single-phase cooling. The increase in pressure results in an increase in the saturation temperatures and delays the onset of boiling; this manifests itself as an added thermal resistance in addition to the conductive and convective thermal resistances.

The large rise in the pressure drop also leads to single and multi-channel instabilities. Wu and Cheng [15] studied flow instabilities in parallel, 186 µm silicon microchannels and discuss the

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existence of multiple types of instabilities that occur at different mass flux and heat input rate combinations. Low-mass flux, high-heat flux conditions were found to result in large fluctuations in pressure-drop and wall temperatures. Static Ledinegg instabilities in parallel microchannel heat exchangers results in boiling in only a few channels leading to redistribution of liquid to the cooler single-phase channels and flow starvation in the hotter two-phase channels [16]. This problem can also manifest itself in larger systems such as server racks with blades being liquid cooled in parallel; boiling in heat exchangers on the hottest blades would result in flow mal-distribution at the rack level with less coolant being delivered to where it is needed the most.

Zhou et al. [17] proposed vapor removal through a porous hydrophobic membrane in order to mitigate some of the problems observed during flow boiling in microchannels. A schematic of the vapor venting heat exchanger discussed in this paper is shown in Fig. 1. The key component in the heat exchanger is the porous hydrophobic membrane that, through capillary forces, prevents the liquid from leaking out of the device but provides minimal flow resistance to the vapor phase. The vapor flows through the membrane into a separate set of parallel channels on the other side of the membrane. The vapor can then be reintroduced into the coolant flow downstream of the microchannels where the impact of having two-phase flow is less significant to the system. The removal of the vapor phase is predicted by both compact 1D/2D simulations (discussed in Section 4) and full 3-D [18] FLUENT (ANSYS Inc.) simulations to significantly improve the pressure drop, lower device temperatures, and delay dry-out.

Previous experimental work has focused on the separation of a dissolved gas from a gas–liquid two-phase flow. Passive gas venting from water and methanol solutions has been investigated by Meng et al. [19,20] for degassing applications in Direct Methanol Fuel Cells (DMFC), where CO₂ formed at the anode inhibits delivery of the methanol to the fuel cell. Active degassing was demonstrated by Yang et al. [21] for portable dialysis applications where ultrasonic transduction was used to coalesce dissolved gas in the working fluid, which was then removed through hydrophobic treated side channels.

In this paper we focus on the design and characterization of copper multi-microchannel vapor venting heat exchangers attached to a silicon thermal test chip. The heat exchangers were designed to operate at temperatures up to 125 °C and gage pressures of 150 kPa (20 psi) and were able to dissipate heat fluxes of 820 kW/m² for water mass fluxes ranging from 102 kg/s m² to 420 kg/s m². The key goal of this work was to experimentally characterize the hydraulic, thermal and venting performance of the fabricated venting heat exchanger and to compare the findings against experimen-

tal results from a non-venting heat exchanger and a compact two-phase vapor-venting flow model. Improved hydraulic and thermal performance is an important contribution to the field of two-phase heat exchangers and opens the way for increased adoption of practical two-phase cooling for electronics.

Heat exchanger design and fabrication is discussed in the following section, followed by the experimental setup and data reduction in Section 3. Section 4 provides details on the two-phase flow model and Section 5 presents the experimental results and discussion of the findings.

2. Design of the vapor separating microchannel heat exchanger

2.1. Venting membrane

The porous hydrophobic membrane is the key component in the vapor venting heat exchanger and enables the separation and transport of the vapor phase. Pore size, membrane thickness, intrinsic permeability, hydrophobicity, and thermo-mechanical stability are all important aspects that need to be considered when selecting or fabricating the phase separation membrane. The liquid leakage pressure (or breakthrough pressure) for the membrane is determined by the Young–Laplace equation, Eq. (1), and varies inversely with the pore diameter, d_{pore} , and directly with the cosine of the advancing contact angle, $\theta_{adv,max}$, and the surface tension, σ [19].

$$\Delta P_{leak} = \frac{4 \cdot \sigma \cdot \cos(\pi - \theta_{adv,max})}{d_{pore}} \quad (1)$$

The total pressure drop across the membrane for single-phase, viscous, laminar flow, with Reynolds and Knudsen numbers less than 1, is given by Darcy Law, Eq. (2), where t_{mem} is the thickness of the membrane, G_{mem} , the mass flux, and μ and ρ , the dynamic viscosity and density of the fluid being transported. The intrinsic permeability, κ , is an intensive property of the porous material and is a measure of the ease of viscous fluid flow through the porous material on application of a normal pressure gradient

$$\Delta P_{mem} = G_{mem} \cdot \frac{\mu}{\rho} \cdot \left(\frac{t_{mem}}{\kappa} \right) \quad (2)$$

The ideal membrane for use in two-phase heat exchangers would: (i) possess a leakage pressure, ΔP_{leak} , larger than the maximum operating pressure expected across the membrane and (ii)

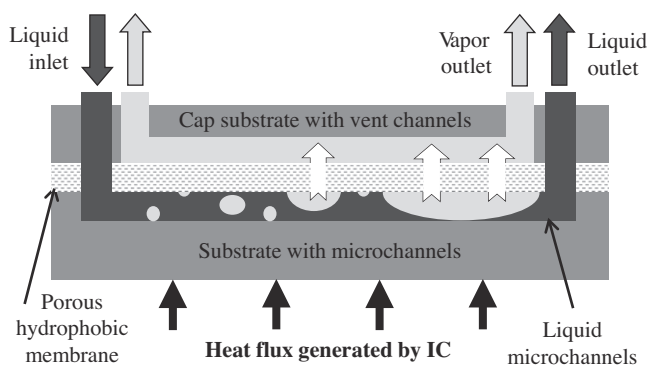


Fig. 1. Schematic of a vapor venting heat exchanger showing the three main components: the two-phase microchannels, the vapor vent channels and the porous hydrophobic membrane. The membrane separates the vapor phase from the two-phase mixture and transports it to the vent channel.

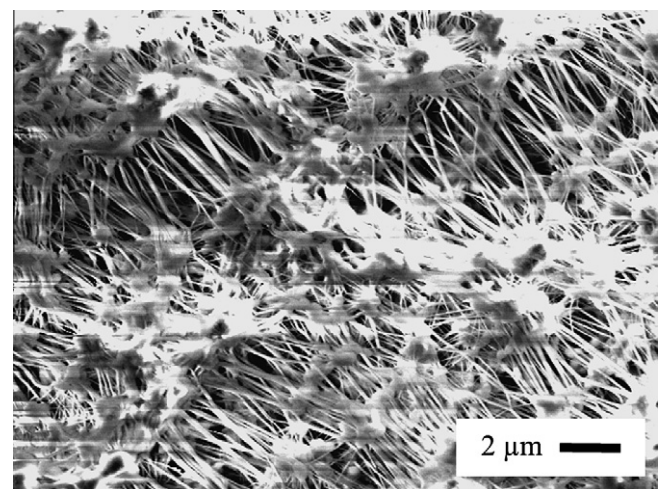


Fig. 2. SEM image of an unused PTFE membrane with manufacturer stated pore diameter of 220 nm and porosity of 0.5–0.8. Surface charging of the non-conductive organic membrane leads to variable brightness in the image and is a non-physical imaging artifact.

have a low pressure drop across the membrane, ΔP_{mem} , to increase the vapor removal rate. This requires that the membrane have a large contact angle with the working fluid, high intrinsic permeability and be thin. Hydrophobic membranes with graded pore diameters is generally desirable, with smaller pores at the separation surface facing the two-phase flow to ensure that liquid does not penetrate into the membrane and larger pores in the bulk of the membrane to increase vapor permeability ($\kappa \sim d^2$).

Based on the requirements and our previous study into the temperature dependant behavior of various hydrophobic membranes [22] we chose to use a 220 nm pore diameter PTFE membrane obtained from GE Water, USA and Sterlitech, USA, as the vapor separation layer. A SEM image of a new PTFE membrane is shown in Fig. 2. The membrane has a porosity of 0.5–0.8, measured thickness of $65 \mu\text{m} \pm 5 \mu\text{m}$, static contact angle of 123° for water at room temperature [23] and manufacturer stated upper limit temperature of 260°C . The theoretical breakthrough pressure for this membrane is approximately 600 kPa for water at 100°C , assuming the contact angle remains the same.

2.2. Copper heat exchanger

The copper heat exchanger fabricated for this study is shown in Figs. 3 and 4. One major advantage of this heat exchanger is the ability to easily insert or remove membranes such that different types of membranes could be used and then studied after use to gauge the impact of wear and fouling. The same microchannel device is used for all the experiments thus removing a source of variability between different device configurations. The copper microchannels and PTFE membrane were capped with either high temperature abrasion resistant polycarbonate/ULTEM or with copper. The liquid and vapor microchannels are fabricated out of 2.4 mm thick polished copper using conventional machining. The copper heat exchanger has 19 channels with a length of 19 mm, average width of $130 \mu\text{m} (\pm 5 \mu\text{m})$ and a depth of $134 \mu\text{m} (\pm 5 \mu\text{m})$. The mating vapor channels have the same length, a width

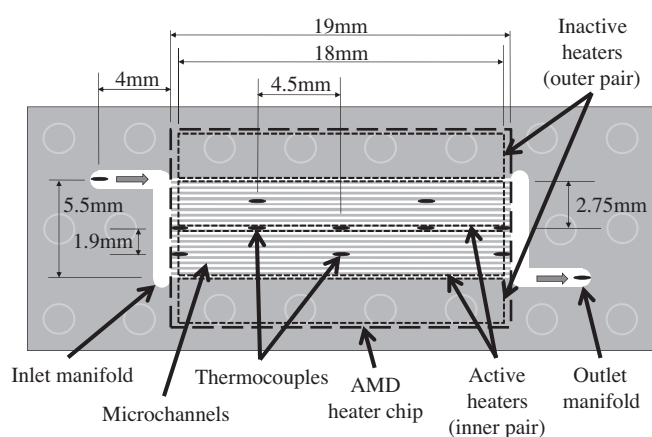


Fig. 4. Top view of tested device showing the manifold, microchannels, thermocouple measurement points and the position of the 4-strip AMD heater chip. The 12 thermocouples are embedded 1.2 mm below the top of the channel walls and heat flux is applied 2.4 mm below the top surface using only the middle two heaters on the test chip.

of $125 \mu\text{m} (\pm 5 \mu\text{m})$ and a depth of $132 \mu\text{m} (\pm 10 \mu\text{m})$. Heat exchanger temperatures on the liquid-side are measured using twelve thermocouples embedded approximately 1.2 mm below the top of the channel walls in the arrangement shown in Fig. 4. Fluid outlet, copper cap and ambient temperatures were also measured using thermocouples. Heat flux was generated by an AMD Thermal Test Chip, 19 mm long and 11 mm wide, with four heaters, each 18 mm long and 2.75 mm wide. Only the middle two heater strips were used, resulting in a heated area of approximately 1 cm^2 .

Three different device configurations were experimentally characterized as shown in Fig. 5. In the control device, Fig. 5A, the channels are capped with a $400 \mu\text{m}$ thick, non-porous high temperature silicone sheet and then with a flat copper cap. The sheet provides better sealing and prevents inter-channel leakage due to variations

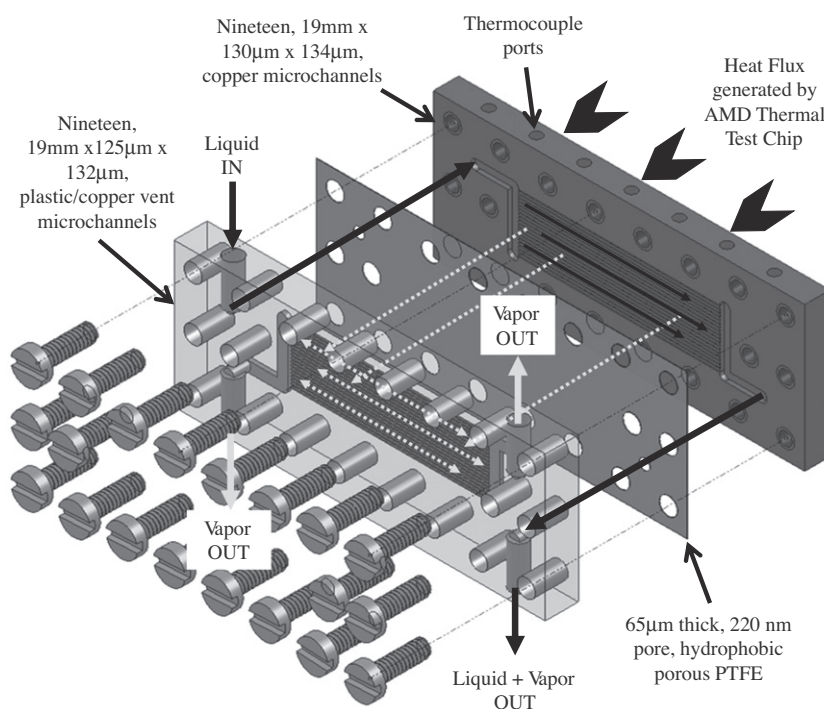


Fig. 3. Design and assembly of the copper microchannel heat exchanger used in this work. Screws and edges of device are coated with high temperature RTV to provide sealing. The channel region is 19 mm long and 5.5 mm wide with nineteen $130 \mu\text{m} \times 134 \mu\text{m}$ microchannels.

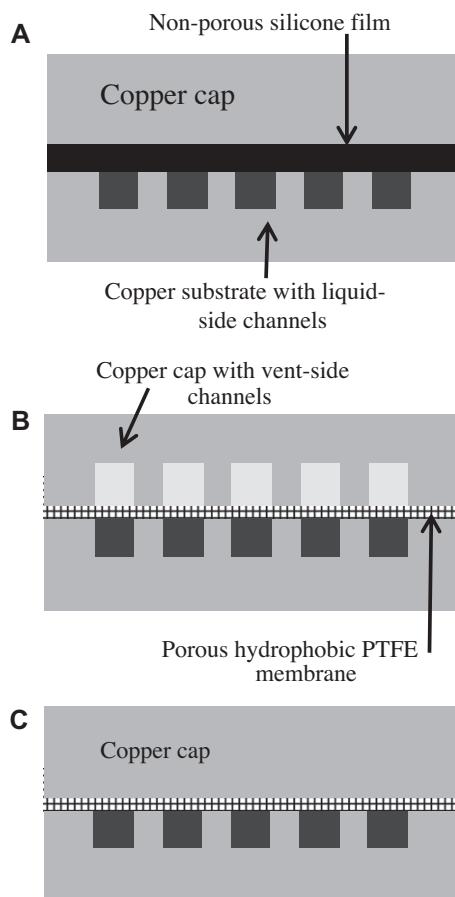


Fig. 5. Schematics of the three different types of copper devices that were assembled and tested in this work. The PTFE membrane is 65 μm thick and the silicone film is 400 μm thick.

in surface planarity between the tops of the channel walls and the flat cover. In the venting heat exchanger, Fig. 5B, the porous membrane is sandwiched between mating sets of channels on the liquid and vent side. And finally, in the non-venting membrane device (NVMD), the porous PTFE membrane used in the venting device is instead capped with the flat copper plate.

3. Experimental method

3.1. Experimental setup and apparatus

The apparatus used to test the heat exchangers is shown in Fig. 6. For the non-venting control and non-venting membrane heat exchangers, the vent side vacuum is not used. The fluid is pumped at a constant flow-rate using a Shimadzu LC10-AD peristalsis pump. The liquid-side inlet pressure is measured using an Omega PX219-30V45G5V pressure sensor and the vent-side vacuum pressure measured using an Omega PX219-30V15G5V pressure sensor. The syringe provides a vacuum of approximately 18 kPa gage (83 kPa absolute) when fitted with a 2 kg mass (used by default); the vacuum can be adjusted by varying the attached mass. The vacuum is used to remove any condensed vapor and to increase the vapor removal rate by increasing the driving pressure across the membrane. Since the liquid is in a closed loop, vapor venting through the membrane results in a mass loss that is measured using a VICON electric mass scale with RS-232 attachment. The ice bath condenser ensures that the measured mass loss is solely due to vented vapor. To minimize heat loss to the ambient,

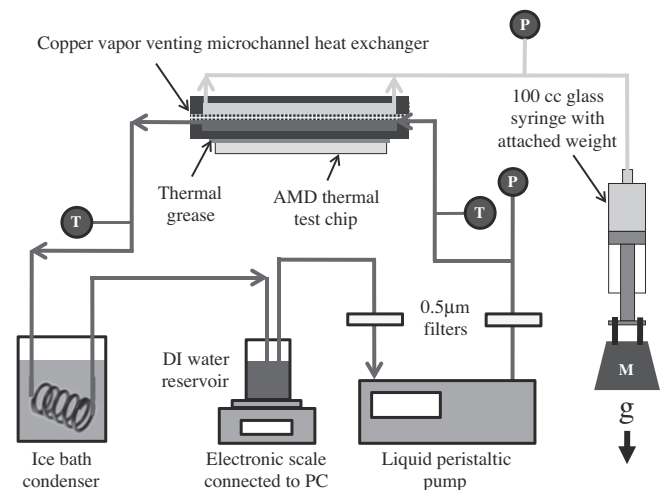


Fig. 6. Experimental setup. 'P' represents pressure sensors and 'T' thermocouples. Heat flux is generated by an AMD test chip and control platform. Vapor vent rate is calculated using the electronic scale. Vacuum on vent side generated by syringe and mass. For control and non-venting membrane devices the vent side apparatus is not used.

the exposed copper cover of the heat exchanger is insulated with fiberglass and polyimide. The inlet and outlet tubing in the vicinity of the device is also insulated with fiberglass. All the thermocouples used in the study, save the inlet thermocouple, are Omega 40-gauge type-K insulated thermocouples. A type-T thermocouple is used to measure the inlet fluid temperature and is read by the AMD control system. Input heat flux was controlled and measured using the AMD control software. The temperature and pressure measurements were gathered at 10 Hz using a NI E-series PC DAQ board via a low noise NI SCXI signal conditioning board and breakout box.

3.2. Data reduction and error analysis

3.2.1. Input and effective heating power

The total input heating power is delivered using the thermal test chip and control hardware. The control system self-calibrates at the beginning of every run resulting in an uncertainty of 1 mW. The heat flux reported in this work is based on the *actual* power absorbed by the fluid (after heat losses) divided by the heater area of 1 cm^2 . The sensible heating of the fluid during single-phase runs at the beginning of each experiment was used to determine the heat loss fraction at each tested mass flux. The average heat loss was found to be 24% at 102 kg/s m^2 , 10% at 208 kg/s m^2 , 6% at 307 kg/s m^2 and 3% at 420 kg/s m^2 . Due to the increased convective heat transfer coefficient during two-phase flow the actual heat loss is expected to be smaller during flow boiling. Therefore, the single-phase heat loss fraction results in an overestimation of the thermal resistance and an under-estimation of the heat transfer coefficient during flow boiling.

3.2.2. Pressure drop

The total simulated and experimental pressure drop reported in this paper is determined using Eq. (3)

$$\begin{aligned} \Delta P_{\text{tot}} &= \Delta P_{\text{chan,sp}} + \Delta P_{\text{chan,tp,acc}} + \Delta P_{\text{chan,tp,fric}} \\ &= \Delta P_{\text{measured}} - \Delta P_{\text{ext.losses}} - \Delta P_{\text{chan,exp/con}} \end{aligned} \quad (3)$$

Due to the comparatively larger hydraulic diameters of the tubing (12 $\times$ larger than microchannels) and plenums (4 $\times$ larger), calculations found the viscous loss in the tubing, minor losses due to contractions, expansions and turns, and loss in the plenums, all

combined as $\Delta P_{ext. losses}$, to constitute less than 1% of the total pressure drop. The contraction and expansion at the plenum-channel and channel-plenum region, $\Delta P_{chan exp/con}$, are the largest minor loss terms and using the expressions presented by Qu and Mudawar [24] were found to be less than 1.4% of the total pressure drop. The external and channel minor losses are subtracted out of the measured pressure drop to obtain the total experimental pressure drop in the microchannels due to single and two-phase flow. The expressions for the single phase and two-phase accelerative and frictional pressure drop in the channels are discussed in more detail in Section 4.2 and are used to determine the equivalent simulated pressure drop.

3.2.3. Temperature and heat transfer coefficient

The type-K thermocouples were calibrated while embedded in the device against the type-T thermocouple (pre-calibrated by AMD Inc.), and an ice bath at 0 °C. In-device calibration accounts for any contact thermal resistances and results in an uncertainty of ± 0.3 °C.

The two-phase heat transfer coefficient is calculated using Eq. (4)

$$h_{tp} = \frac{q_{sat}}{A_{chan,tp} \cdot (T_{d,tp} - T_{sat,m})} \quad (4)$$

where q_{sat} is the fraction of the effective input power, q_{in} , that is used for phase change and can be estimated as $q_{sat} = q_{in} - m \cdot C_p \cdot (T_{sat,in} - T_{in})$. $T_{sat,in}$ is the saturation temperature at the channel inlet pressure and is used with the assumption that the pressure drop in the single phase region is comparatively small such that the saturation temperature at the inlet is approximately equal to the saturation temperature at the point where phase change occurs. Simulations using the model described in Section 4 were used to estimate the relative error in q_{sat} associated with this simplification and the average error for the four tested flow-rates were found to vary from 0.2% to 1.2% as the flow-rates increase. This estimation error is included in the total uncertainty in the calculation of h_{tp} . $T_{sat,m}$ is the mean of the calculated inlet saturation temperature, $T_{sat,in}$, and the outlet saturation temperature, assuming that saturation temperatures vary linearly along the two-phase region of the channel. Simulations were used to determine the relative error in $T_{sat,in}$ due to this simplification, and was found to be on average between 1.5% and 2.3% for the four flow-rates with the simplification tending to underestimate the actual saturation temperature except at low qualities. The device temperature, $T_{d,tp}$, is calculated from the average of the two downstream center-line thermocouples since the three upstream thermocouples were found to be more strongly influenced by the single phase flow at low exit qualities. The area, $A_{chan,tp}$, is assumed to be the wetted surface area in the microchan-

nels experiencing two-phase flow and does not include the area covered by the capping material.

Using error propagation rules, the uncertainty in the heat transfer coefficient, averaged over all the runs where two-phase flow was produced, is found to be 44%, 43% and 27%, for the control, non-venting membrane and venting membrane devices, respectively. The leading contributor to the uncertainty in h_{tp} is the uncertainty introduced in using the linear variation assumption to determine $T_{sat,m}$. Since the simplification tends to underestimate the mean saturation temperature, the heat transfer coefficient has a larger positive uncertainty.

Table 1 summarizes the errors in the raw and calculated values discussed in this study.

4. Two-phase vapor venting model

4.1. Description of computational domains and solution technique

Fig. 7 highlights the computational element and the key external boundary conditions used in the two-phase venting model. This 2-D element is assumed to be an interior element in the device and does not consider the impact of the edges of the device. Regions A, B and C are solved separately (only A and C in the case of non-venting microchannel) using three nested iterative loops to reach the final converged solution for the domain as a whole.

Region A considers 1-D two-phase flow in the microchannel with heat flux boundary condition provided from the solution of Region C. The interface with Region B is a pressure boundary condition generated from the solution of Region B. Since fluid properties can change quite significantly with pressure and temperature, a MATLAB based steam-table [25] as well as several property tables are used to calculate the new fluid properties at each step. The pressure drop is modeled using Poiseuille flow for single-phase flow and a separated-flow model for two-phase flow. The impact of the square cross-section is captured using the hydraulic diameter approximation as well as the appropriate friction factor. Similarly, we use single phase and two-phase heat transfer relations developed for pipe flow. Due to the comparatively much lower thermal conductivity of the capping material (silicone film or PTFE membrane) in our devices we assume that only three of the four walls participates in heat transfer and so make the Nusselt number correction to the heat transfer coefficients as described in Qu and Mudawar [8]. Mass and energy loss due to venting is accounted for in the given relations at each element with the assumption that these relations remain valid despite asymmetric suction.

We also consider flow in the membrane within Region A, and treat it as 1-D, single-phase and adiabatic. Since the thickness of the membrane is more than two orders of magnitude smaller than

Table 1
Summary of experimental uncertainties.

Quantity	Uncertainty
Mass flux	5%
Mass flux velocity ^a , G	9%
Pressure drop	1 kPa (liquid-side) 0.5 kPa (vent-side)
Normalized liquid-side pressure drop ^a	0.3 (103 kg/s m ²), 0.1 (208 kg/s m ²)
Normalized trans-membrane pressure drop ^a	0.09 (307 kg/s m ²), 0.07 (420 kg/s m ²)
Temperature	0.3 °C
Venting rate	0.12 g/min
Heat transfer coefficient ^a	27–44% (average), 48–80% (max.)
Total input power	0.1 mW
Heat flux ^a	5%

^a Calculated values.

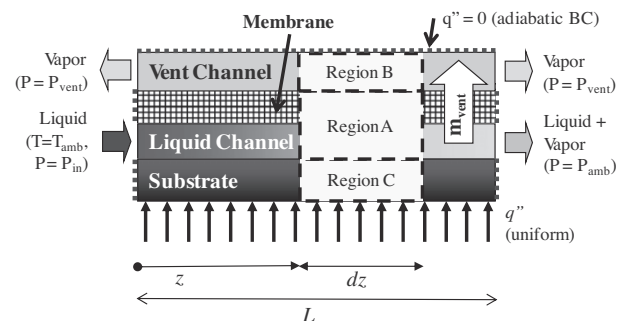


Fig. 7. Schematic of computational element used in simulations presented in this study. External boundary conditions are presented. Region A: 1D two-phase flow in the microchannels with single-phase flow in the membrane. Region B: 1D single-phase vapor flow in vent channel. Region C: 2D solid conduction in substrate.

the length of the membrane along the channel, the flow in the membrane is assumed to be 1-D and normal to flow in the channels. We assume that there is no entrained liquid mass in the venting vapor and that the high temperature of the surrounding substrate and membrane prevents condensation of the venting vapor within the membrane, thereby maintaining single-phase vapor flow in the membrane. Due to the low maximum Reynolds number of 0.01 and a Knudsen number < 1 for the steam flow through the membrane pores, we ignore any non-linear inertial and molecular flow effects. Given these assumptions, the flow in the membrane can be described by the Darcy equation for viscous flow in porous media, Eq. (2), with the intrinsic permeability of $8 \times 10^{-15} \text{ m}^2$ determined from experimental data. Vapor venting from the manifold can be significant and is included in the model as well. From a thermal standpoint, we assume that there is little heat transfer between the membrane and the vapor and treat flow in the membrane as adiabatic.

Region B evaluates 1D, adiabatic, single-phase vapor flow in the vent channel with a mass-flow boundary condition at the interface with Region A and pressure boundary conditions at either end of the vent channels. Averaged fluid properties are used since neither temperature nor pressure changes significantly in the vent channel for our experimental conditions. We assume flow in the vent channel to be single-phase as the temperature of the top surface of the copper cap in which the vent channels are machined were measured to be between 1°C and 10°C higher than the saturation temperature for the vented vapor. Flow in the vent channel is also assumed adiabatic due to the small convective heat transfer coefficient and temperature difference during vapor flow. Similar to Region A, we again assume fully developed pipe-flow pressure drop relations are applicable with mass influx from the liquid channel treated as variable mass flux along the length of the vent channel.

Region C deals with 2D conduction in the substrate with constant material properties and a convective boundary condition at the channel-wall interface provided from the solution of Region A. Since the AMD chip is designed to generate a uniform heat flux and spreading in the silicon chip is small in comparison to that in the copper substrate we assume a constant heat flux boundary condition at the bottom of the copper substrate. We assume that there is no conduction from the lower liquid-channel containing substrate to the vent-side capping substrate due to the top substrate being well insulated and heat flow to the vapor minimal.

4.2. Pressure drop models

Single-phase liquid and vapor flow in the liquid channel and single-phase vapor flow in the vent channel is modeled using standard Poiseuille flow, Eq. (5), with the laminar fanning friction factor, f , for rectangular channels given by Kakac et al. [26]

$$\left(\frac{dP}{dz}\right)_{sp} = -\frac{2 \cdot f \cdot G^2}{\rho \cdot D_h} \quad (5)$$

The subcooled boiling region is also treated hydraulically as single-phase and modeled using Eq. (5). We verified the validity of this simplification by determining the subcooled region in our simulated results and found it to occupy about 4% of the total device length on average. Treating this region as single-phase thus has little impact on the total pressure drop.

The two-phase pressure drop is modeled using a separated flow approach. The separated flow model for a horizontal channel [12], given by Eq. (6), originally developed by Lockhart and Martinelli for adiabatic air–water flows, allows for different velocities in the liquid and vapor phase, unlike homogenous models which treat the two-phases as a single-phase with averaged properties and

equal velocities for both phases. However, unlike the homogenous model, this model requires correlations for the two-phase multiplier, ϕ_t , in the frictional component (1st term on the right) and for the void-fraction, ε , in the accelerative component (2nd term on the right)

$$\left(\frac{dP}{dz}\right)_{tp,SFM} = -\phi_t^2 \cdot \left(\frac{dP}{dz}\right)_l + \frac{d}{dz} \left(\frac{G^2 \cdot x^2}{\rho_v \cdot \varepsilon} + \frac{G^2 \cdot (1-x)^2}{\rho_l \cdot (1-\varepsilon)} \right) \quad (6)$$

The Zivi void fraction correlation, Eq. (7), is an analytical model derived from a kinetic energy minimization approach for annular flow (common in microchannel flow) [27]. Empirical void fraction models such as the Smith separated flow correlation [28], the Rouhani–Axelsson drift-flux correlation (as described by Thome [29]) and the Chung and Kawaji correlation [30], developed from two-phase flow in $100 \mu\text{m}$ capillaries, were compared against the Zivi model. For the range of qualities and mass fluxes in our study the calculated variation in the total pressure drop when using the different models was found to be less than 2%; thus we chose to use the analytically derived Zivi model for void fraction

$$\varepsilon = \left(1 + \frac{(1-x)}{x} \cdot \left(\frac{\rho_v}{\rho_l} \right)^{2/3} \right)^{-1} \quad (7)$$

Lockhart and Martinelli developed a graphical correlation relating the two-phase multiplier with the Martinelli parameter. Chisholm et al. then presented this graphical correlation mathematically as shown by Eq. (8), where C is the Chisholm parameter and X is the Martinelli parameter, given by Eq. (9)

$$\phi_l^2 = \left(1 + \frac{C}{X} + \frac{1}{X^2} \right) \quad (8)$$

$$X^2 = \frac{(dP/dz)_l}{(dP/dz)_v} = \frac{\left(\frac{2f_l \cdot G^2 \cdot (1-x)^2}{\rho_l \cdot D_h} \right)}{\left(\frac{2f_v \cdot G^2 \cdot x^2}{\rho_v \cdot D_h} \right)} = \frac{f_l}{f_v} \cdot \frac{\rho_v}{\rho_l} \cdot \left(\frac{1-x}{x} \right)^2 \quad (9)$$

The separated flow model has had wide use in the modeling of two-phase flow at the macroscale and microscale with authors modifying C to capture additional effects. Refs. [31–33] provide recent reviews on previous correlations for two-phase flow in microchannels.

To determine the correct form of the two-phase pressure drop equation for use in our model we compared our control data for the 102 kg/s m^2 mass flux against several models. We found that our data falls mostly between the values predicted by the homogenous model and the separated flow model proposed by Lockhart and Martinelli with $C = 5$. In trend however, the data agreed with the correlation proposed by Lee et al. [31] with a Chisholm parameter given by $C = 121.6(1 - \exp(-22.7Bo))x_e^{1.85}$. Modifying their model, we obtain a new correlation for the pressure drop with the Chisholm parameter given by Eq. (10) where the liquid only Reynolds number is given as $Re_{lo} = GD_h/\mu_f$. The Bond number was not included since we did not run experiments using channels of different hydraulic diameters or using different fluids to confirm or determine its functional form in our correlation

$$C = 1.84 \cdot Re_{lo}^{0.3} + 1.5 \cdot Re_{lo} \cdot x^{1.85} \quad (10)$$

This correlation agrees well with our experimental data, suggesting that the Chisholm parameter has the same power relationship with the vapor quality as proposed by Lee et al. They suggest that the reason behind this dependence is the increased velocity ratio and thus interaction between the phases as quality increases; however a more detailed study on the physical nature of this dependence is required. Our experimental data at the higher mass fluxes of 208 kg/s m^2 , 307 kg/s m^2 and 420 kg/s m^2 suggests that the Chisholm parameter is also increasing with the flow velocity. Qu and

Mudawar [24] and Lee and Garimella [33] also find velocity dependence in the pressure drop though the relationships are different. The velocity dependence is captured in our correlation using the Reynolds number, however, the Capillary number or Weber number may be more appropriate. Further studies are needed using different channel diameters and fluids to determine the relationship between the inertial, viscous, surface tension and gravitational forces and the Chisholm parameter.

4.3. Convective heat transfer models

The single-phase fully developed heat transfer coefficients for the liquid and vapor are modeled using a mean Nusselt number, Nu_m , of 2.98 [34]. This Nusselt number represents constant wall temperature in a square channel. Assuming a constant heat transfer coefficient along the perimeter of the channel the fin efficiency for the 150 μm thick copper channel walls is calculated to be unity. Since the thermal entry length can be significant we model the single-phase thermally developing heat transfer coefficient using the series solution provided in Kays and Crawford [34].

Several empirical correlations for the two-phase heat transfer coefficients now exist in the literature for a variety of experimental conditions. We compared our control device data against the heat transfer models proposed by Kandlikar and Balasubramanian [35] and by Lee and Mudawar [36] and found that our data was higher than either predicted at qualities <20% and fell between the two models at higher qualities. Similar to Lee and Mudawar we developed a correlation that better fits our data based on the Martinelli parameter, X . The new correlation is given by Eq. (11) and matches the experimental control device data at all mass fluxes with a mean approximate error of 13%

$$h_{tp} = 20 \cdot X \cdot h_{sp,fd} \quad (11)$$

The heat transfer coefficient data collected at different input heat-fluxes and mass fluxes were found to collapse to a single curve when plotted against the vapor quality, suggesting little dependence of the heat transfer coefficient on either the heat-flux or mass-flux. Agostini and Thome [37] in their review find similar behavior in the literature though it is typically only observed for high quality flows of over 50%.

5. Results and discussion

5.1. Hydraulic characteristics

5.1.1. Steady state pressure drop

The reduction in the steady state pressure drop after the onset of boiling in microchannels is one of the key demands for the vapor venting technology as it leads to: (i) smaller pumping power (and thus pump size), (ii) reduced fluid saturation and heat exchanger temperature and (iii) potentially reduced impact of Ledinegg instability.

Fig. 8 illustrates both the experimentally measured and simulated normalized pressure drop, $\Delta P_{tot}/\Delta P_{lo}$, against heat flux at the lowest and highest of the four tested mass fluxes. The pressure is normalized to account for the slight variations in the pressure drop between the different devices due to small differences in the hydraulic diameter (<10 μm). A minimum of two-data sets, taken non-sequentially, is presented to indicate repeatability. Fig. 8 also includes the pressure drop that is measured when using the non-venting membrane device (NVMD), Fig. 5C. The maximum heat flux tested is lower for the venting and non-venting membrane devices due to poorer application of the thermal interface material between the chip and heat exchanger that led to higher

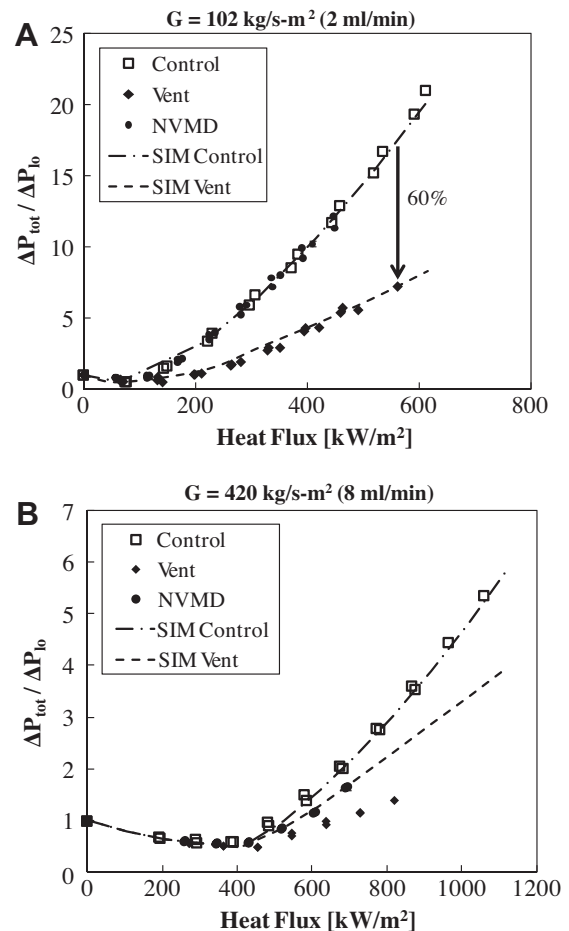


Fig. 8. Comparison of experimental results for the non-venting control device, non-venting membrane device (NVMD) and venting device shows average improvement ~60% between the control and venting devices. NVMD performance and numerical prediction deviate at higher mass fluxes (B) due to the formation of complex churn-annular type flows.

chip temperatures for a given input power. Testing higher heat fluxes would have exceeded the safe temperature limits of the test chip.

The improvement in the normalized pressure drop is found to be approximately 60% between the control and venting membrane devices. Using the numerical model described in Section 4 we find excellent agreement with the experimental data at the lowest mass flux of 103 kg/s m^2 but find an increasing under-prediction of the pressure drop improvement during venting as the mass flux increases from 208 kg/s m^2 to 420 kg/s m^2 . Interestingly we also observe a deviation in the non-venting membrane device from the control data at 420 kg/s m^2 that is not observed at the lowest tested mass flux. The results suggest that the additional pressure reduction is related to both mass-flux increase and the presence of the membrane.

The primary and most easily understood source of improvement in the pressure drop in the venting devices is the removal of vapor from the two phase flow, resulting in a reduction in mass flux, G , as well as quality, x . As described by Eqs. (6)–(9) this results in a reduction in both the frictional and accelerative two-phase pressure drops. Another important source of pressure drop improvement is the change in flow-regime due to the hydrophobic membrane. Visualization work that we recently carried out and discussed in Ref. [38], found that as mass-flux was increased the two-phase flow transitioned from mostly smooth stratified flow at a mass flux of 140 kg/s m^2 to increasingly more chaotic and

churn-annular type flows at a mass flux of 340 kg/s m^2 (shown in Fig. 9). Unlike during stratified flow, the churn-annular flow was found to deposit a significant amount of fluid in the form of droplets on the membrane surface that would then be pushed along the membrane surface. The amount of pressure required to push non-wetting droplets on the membrane surface is smaller than that required to move wetting films. Since our model does not capture this complex behavior it under-predicts the pressure drop improvement as mass fluxes increase. This also explains the reduction in pressure drop at the higher mass-flux measured in the non-venting membrane device.

Other sources of pressure-drop improvement that were considered but found to be ultimately insignificant include membrane deformation (estimated to be only about 50 nm given a Young's Modulus of 30 MPa for the porous membrane [39]), vapor flow along the length of the membrane (whose hydraulic resistance is estimated to be five orders larger than that of the channel) and vapor slip at the channel-membrane interface (slip-length is determined to be only 650 nm using the expression presented in the text by Kaviani [40]).

5.1.2. Impact of trans-membrane pressure on pressure drop improvement

We also investigated the impact of varying the trans-membrane pressure on the pressure drop improvement in the venting devices. The vent side pressure was reduced by increasing the mass suspended on the syringe plunger as shown in the setup in Fig. 6. The results are summarized in Fig. 10 with the normalized pressure drop compared to the normalized trans-membrane pressure with the non-venting limit included for comparison. The normalized pressure drop reduces almost linearly as the trans-membrane pressure is increased and the results and trend are well characterized by the model. In the self-venting case, the vent side outlets were left open to the atmosphere such that vapor venting is solely driven by liquid-side pressure with no additional assistance. This is

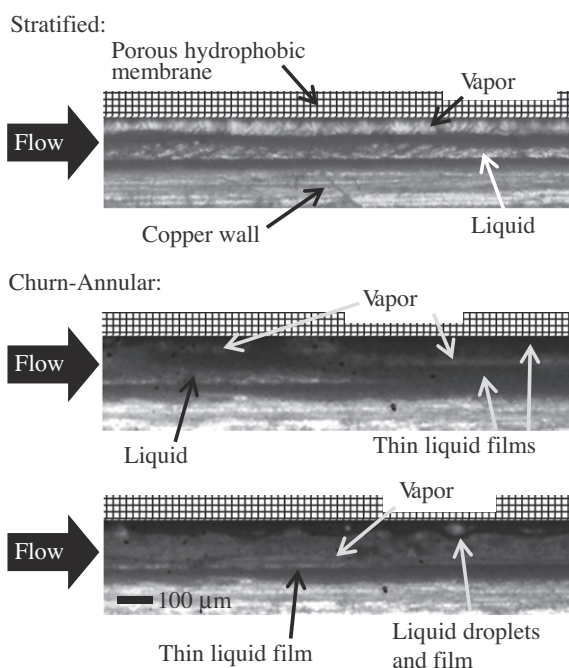


Fig. 9. Representative images from previous visualization work showing the stratified flow and churn annular flow encountered at low and higher mass fluxes. The average film thickness is larger during stratified flow but much smaller and on multiple walls during churn annular flow. The third image highlights the droplet flow along the membrane.

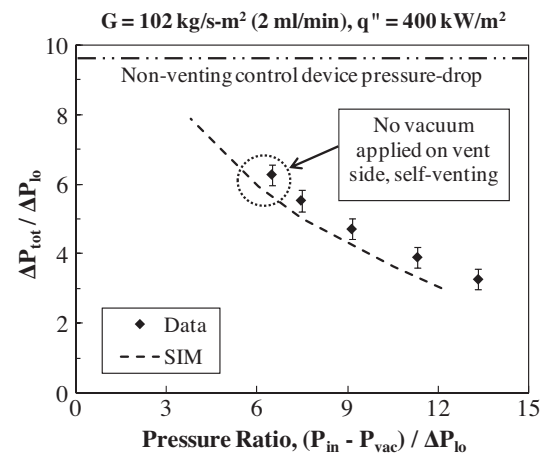


Fig. 10. Increasing the trans-membrane pressure by reducing the vacuum pressure results in increased venting and thus improved device pressure drop. When no vacuum is applied and the vapor self-vents due to the pressure in the liquid-side channel the improvement in the pressure drop is ~30%.

practically important since the application of vacuum on the vent side may not always be feasible. For best hydraulic performance however a high vacuum on the vent side is necessary.

A calculation of pumping power efficiency, q_{in}/q_{pump} , as the trans-membrane pressure is increased finds that the efficiency is always better than the non-venting control device due to the improvement in the pressure drop, though this improvement is not monotonic. There is initially an increase in the efficiency when the normalized trans-membrane pressure is increased from 6 to 9, as the reduction in the pump power on the liquid side is greater than the increasing work done on the venting side of the device. However, beyond this point, the efficiency begins to reduce as the work done on the vent side increases compared to the reduction in work by the liquid side pump. Thus the improvement in hydraulic (and thermal) performance needs to be weighed against overall efficiency when considering the amount of trans-membrane pressure to apply using a vacuum pump.

5.1.3. Dynamic and static Ledinegg instabilities

The different types of dynamic and static instabilities that exist during boiling in tubes and channels are reviewed in detail by Kakac and Bon [41]. Dynamic instabilities such as pressure drop and density wave oscillations and rapid bubble growth are major sources of flow instabilities leading to fluctuations in the pressure drop and wall temperature, as observed experimentally by Kuo and Peles [42] and Wu and Cheng [15]. Methods to mitigate these instabilities include addition of inlet constrictors [43], adding nucleation sites to reduce the superheat and explosive bubble growth [44], raising the system pressure [42] and active system control [45]. Simulations and discussion of dynamic pressure fluctuations during the growth and venting of bubbly flow are provided by Fang et al. [18]. Though the static pressure drop was lower, they found large pressure variations at low frequencies (~1 kHz) due to venting of the discrete vapor bubbles. These pressure fluctuations are not expected to be critical when vapor venting occurs from larger vapor structures such as during annular flow. For the rapid bubble growth instability, if the vapor venting rates through the membrane are comparatively large it might be expected that the vapor vent channels will absorb some of the rapid bubble growth and mitigate the instability. However, if the venting rates are low then the venting channel would appear as a non-venting channel at the short bubble growth time scales and thus unlikely to impact channel instabilities. The impact of

vapor venting on the rapid bubble growth instability and the pressure drop and density wave instabilities requires further investigation before any conclusions can be drawn.

The improvement in pressure drop is expected to reduce the occurrence of static Ledinegg instability that can have highly detrimental effects on the operation of two-phase heat exchangers by starving channels experiencing two-phase flow of coolant. Tong and Tang provide background [46] and Zhang et al. [47] experimental and numerical treatment of Ledinegg instability during two-phase flow in microchannels. Ledinegg instability occurs when the magnitude of the negative slope in the two-phase demand curve is greater than that of the supply curve for the pump, shown algebraically in Eq. (12), causing the system to become unstable

$$\left(\frac{\partial(\Delta P)}{\partial G}\right)_{\text{channel}} \leq \left(\frac{\partial(\Delta P)}{\partial G}\right)_{\text{pump}} \quad (\text{criterion for unstable case}) \quad (12)$$

Tong and Tang and Zhang et al. suggest the use of inlet constrictors to increase the pressure and reduce the negative slope of the demand curve such that Ledinegg instability is not possible. However, this increases the pressure drop of the system significantly. In our current work the occurrence of Ledinegg instability is reduced due to the smaller change in pressure that occurs on boiling, resulting in a smaller negative slope in the two-phase demand curve. Additionally, unlike the use of constrictors, the overall pressure drop of the system is also lowered. The impact of vapor venting is highlighted in Fig. 11, which shows the simulated demand curves of the non-venting control and vapor venting devices along with an example supply curve. The simulation results suggest that vapor venting could potentially be very useful in mitigating this static instability mechanism.

5.2. Thermal characteristics

5.2.1. Temperature

With the reduction in pressure drop, the saturation temperature in the venting device also drops, resulting in an improvement in the device temperatures. Fig. 12 shows the mean device temperatures as a function of the input power for the lowest and highest tested mass fluxes. Fig. 12A also includes temperature data from the non-venting membrane device. The device temperature is im-

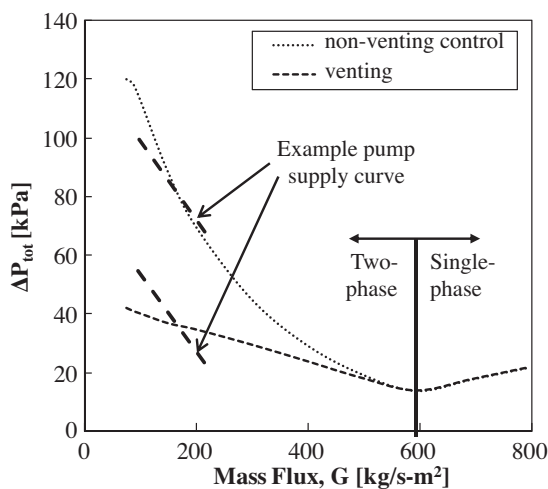


Fig. 11. Simulated demand curves for the non-venting and vapor venting devices discussed in this study. The negative slope $-\partial(\Delta P)/\partial G$ for the non-venting device is greater than that for an example pump, resulting in Ledinegg instability. However, the negative slope is smaller for the venting device making it less susceptible to this static instability.

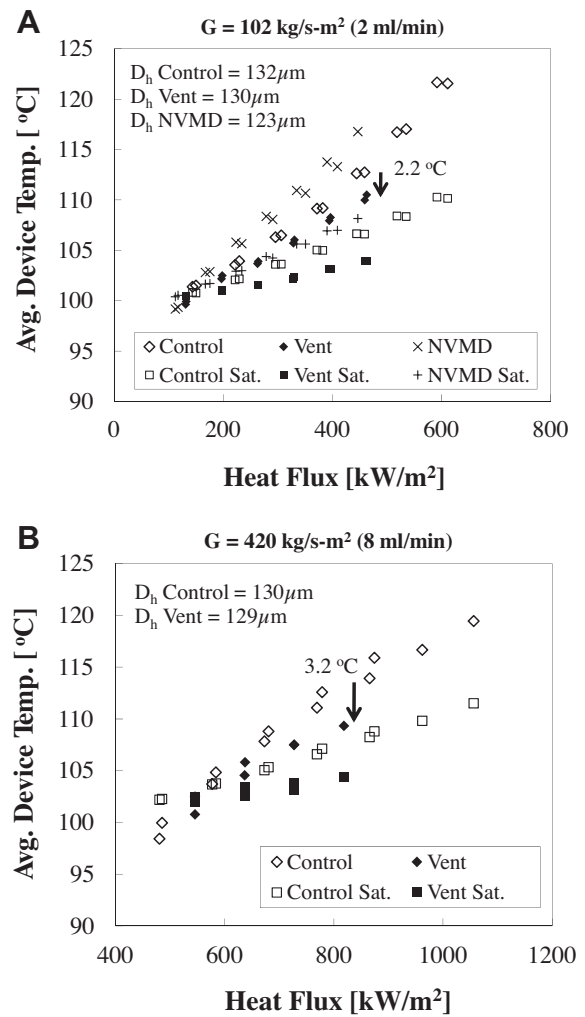


Fig. 12. Average temperatures in the control, non-venting membrane (NVMD) and venting devices at two of the four mass fluxes. Improvements in temperatures in the venting devices are mostly due to improvement in saturation temperatures. NVMD had higher temperatures due to higher saturation temperature and poorer heat transfer coefficient.

proved in all the venting cases with improvements as high as 4.4 °C at a mass flux of 208 kg/s m² and heat flux of 730 kW/m². This improvement is primarily due to an improvement in the saturation temperatures as seen comparing the average saturation temperature in the control and venting devices. Fig. 12A also indicates that the non-venting membrane device suffers from poorer thermal performance with device temperatures larger than the difference in the saturation temperatures between the non-venting membrane device and the control device. The saturation temperature in the non-venting membrane device is higher due to poorer heat transfer coefficients as discussed in the next section and due to a slightly smaller channel hydraulic diameter (~123 μm vs. 132 μm for the control device) leading to larger absolute inlet pressure and saturation temperature. Fig. 13 shows the improvement in the temperature in the venting device run at 102 kg/s m² and heat flux of 400 kW/m² as the trans-membrane pressure is increased by application of vacuum leading to a reduction in the pressure-drop and saturation temperature.

5.2.2. Heat transfer coefficient

Fig. 14 shows the measured and simulated heat transfer coefficient at all four mass fluxes tested. The experimental data at the three higher mass fluxes found that the heat transfer coefficient

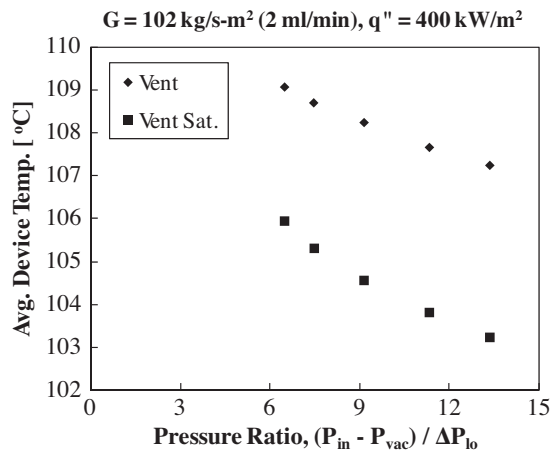


Fig. 13. Device temperatures improve almost linearly with increasing trans-membrane pressure as saturation temperature reduces.

in the venting device is comparable or higher than the control for a given input power with significant variability in the venting data. The venting data occasionally matches the predicted venting heat transfer coefficients as seen in Fig. 14B and C and the difference between control and venting predictions are similar to the predicted

difference. The picture presented by the data obtained at $420 \text{ kg/s}\cdot\text{m}^2$ is less clear with greater variability also existing in the control data. In the case of the lowest mass flux of $102 \text{ kg/s}\cdot\text{m}^2$, the two-phase heat transfer coefficient in the venting device and non-venting membrane devices are found to be consistently lower than the control device and that predicted by the simulation. Varying the trans-membrane pressure at a mass flux of $102 \text{ kg/s}\cdot\text{m}^2$ and heat flux of 400 kW/m^2 had negligible impact on the two-phase heat transfer coefficient which was found to be approximately $42 \pm 3 \text{ kW/m}^2\cdot\text{K}$.

Due to vapor removal in the venting devices, the quality, mass flux and saturation pressure all reduce. These factors and the change in flow regimes in the venting devices can potentially impact the heat transfer coefficient. The heat transfer coefficient correlation developed in this work, and used in the model, and three of the leading models in the literature given by Kandlikar and Balasubramanian [35], Lee and Mudawar [36] and Thome et al. [48] all predict higher heat transfer coefficients at lower vapor quality. Experimentally, we only occasionally find this enhancement in the venting devices at the mass fluxes of 208 and $307 \text{ kg/s}\cdot\text{m}^2$ and not at all at the lowest mass flux, even when the trans-membrane pressure, and thus vapor removal rate, increased. These results can be explained by considering the impact of saturation pressure and flow regime changes on the heat transfer coefficient. The model developed by Thome et al. [48] and confirmed

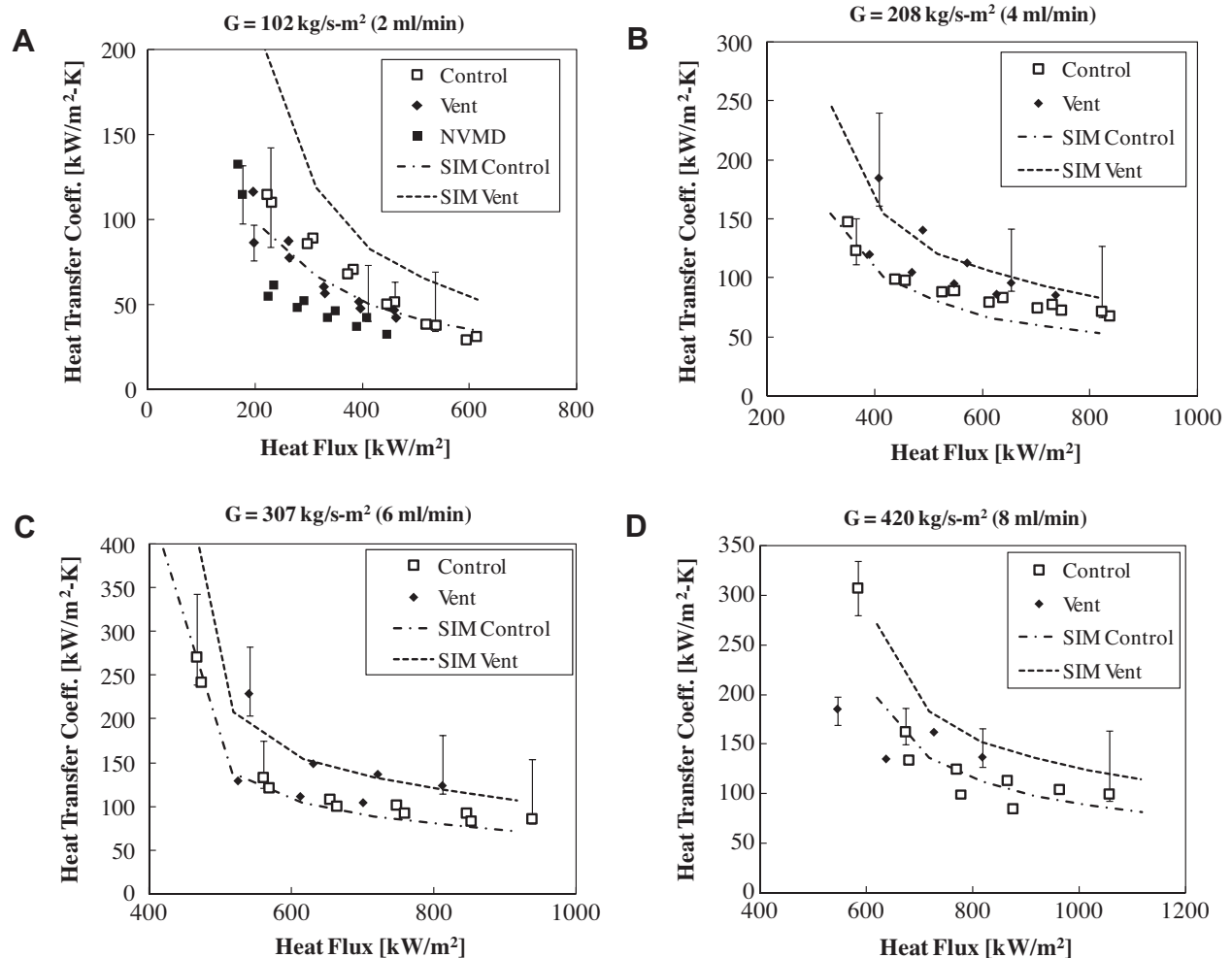


Fig. 14. Two-phase heat transfer coefficients as a function of heat flux in the control, non-venting membrane (NVMD) and venting devices at the four tested mass fluxes with representative error bars. The heat transfer coefficients in the venting device are equal or better than in the control for the three higher mass fluxes but lower for the lowest mass flux due to differences in the flow regime.

with experimental data by Dupont et al. [49], suggests that the heat transfer coefficient should decrease with reducing saturation pressures. They explain this to be due to reduced frequency in the liquid-annular-vapor cycle. Thus the improvement in the heat transfer coefficient due to reducing quality is potentially being negatively offset by this pressure effect. More significant we believe is the change in flow regime that occurs when a hydrophobic membrane replaces one of the channel walls. As discussed in Ref. [38] and in the discussion on pressure drop in Section 5.1.1, stratified flow is observed at low mass flux and churn-annular flow at higher mass flux, with liquid-annular-vapor cyclical flow found to be dominant during visual observations of two-phase flow in the control device used in this work. Stratified flow is thermally less beneficial as compared to annular flow due to the replacement of four thin evaporating films with one thicker film, as seen in Fig. 9. This explains the poorer heat transfer coefficients in the venting and non-venting membrane devices at the lowest tested mass flux. As mass flux increases churn-annular flow replaces stratified flow in the venting device and this flow regime is expected to have thermal characteristics similar to the annular flow experienced in the control device, thus explaining the similarity in the measured heat transfer coefficients.

Though not investigated in our work, it is also useful to consider the possible impact of the membrane on the dryout behavior in the channel. During stratified venting flow, though the vapor phase is more prevalent at the top of the channel due to the presence of the hydrophobic membrane, the venting of the vapor phase through the membrane will actually decrease the vapor volume fraction in the main channel section and thereby delay dryout conditions. Furthermore, because the membrane tends to draw the vapor phase away from the primary heat transfer surfaces (bottom and side walls), the problem of heat transfer coefficient reduction, associated with dryout, will likely also be suppressed. During annular venting flow, though dryout is again potentially suppressed due to the reduced vapor fraction in the channel due to vapor venting, the liquid droplets moving along the membrane surface would not participate as strongly in evaporation and this could instead manifest as premature dryout. Liquid mass loss through the membrane due to entrainment of the liquid on the membrane surface during vapor venting may also potentially lead to premature dry-out.

The complexity of heat transfer during two-phase flow makes it difficult to predict the expected heat transfer in a vapor venting device using the simple correlation presented here or others available in the literature. However, the picture presented by the experimental data is encouraging with the two-phase heat transfer coefficient in the venting devices either equivalent or slightly higher than the control devices except at the lowest mass flux where we believe stratified flow is dominant. When considering dryout, stratified flow may be beneficial due to better confinement of the liquid to the heated wetting surfaces unlike during churn annular flow. Further experimental work is necessary to draw any conclusions regarding the dryout conditions in the vapor venting channels experiencing different flow regimes.

5.3. Venting characteristics

Fig. 15 summarizes the experimental and simulated venting rates against the heat flux showing that as the heat flux is increased the venting rate increases due to the larger amount of vapor generated and the increasing pressure on the liquid side. We find good agreement between the experiment and simulation for all the mass fluxes though the predicted change in venting rate with heat flux is lower than measured for the highest mass flux of 420 kg/s m². The venting rate and venting fraction (vapor vented/vapor generated) as a function of trans-membrane pressure is shown in Fig. 16. We find that both the venting rate and fraction

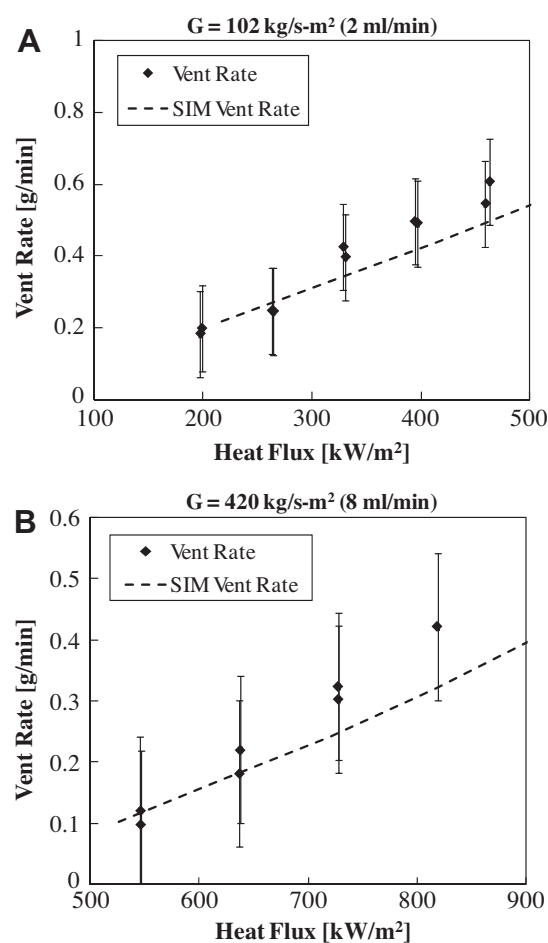


Fig. 15. Experimental and numerical results of venting rates at two of the four tested mass fluxes. Venting rates increase as more vapor is being generated and because of the rising pressure on the liquid side of the device.

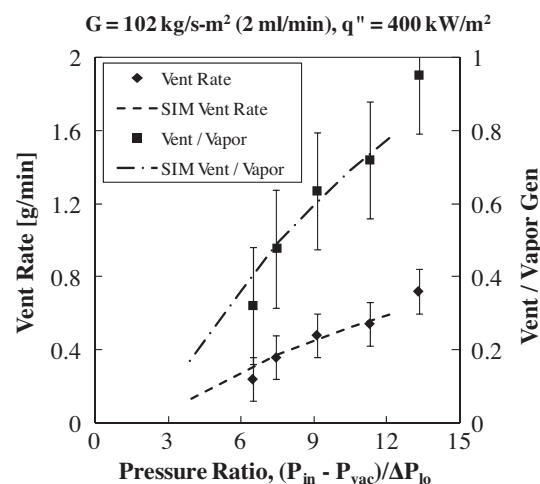


Fig. 16. Venting rates and venting fractions increase as the trans-membrane pressure increases. The increased vapor removal leads to improved pressure drops and device temperatures.

increase almost linearly with the trans-membrane pressure and use this data to determine the intrinsic permeability of the membrane, which is an input for the numerical model.

To better understand the factors that influence the venting fraction such that it can be maximized to improve the pressure drop

and device temperatures, we compare the hydraulic resistance of the channel with those in the membrane and vent channel. For simplicity we model the two-phase hydraulic resistance in the liquid channel using the homogenous description of two-phase flow. The ratio of the resistances is shown in Eq. (13), where B_1 and B_2 are property constants and B_x is a quality dependant property constant that increases with quality and is maximized when flow in the liquid-side channel is vapor only. The third term in the denominator captures the effect of unequal outlet pressures on the liquid and vent side of the device

$$\frac{R_{h,chan}}{R_{h,mem} + R_{h,vent} + R_{h,press}} = \frac{\left(B_x \cdot \frac{L_{chan}}{D_{h,chan}^4} \right)}{\left(B_1 \cdot \frac{L_{mem}}{\kappa} \cdot \frac{1}{w \cdot L_{chan}} \right) + \left(B_2 \cdot \frac{L_{vent}}{D_{h,vent}^4} \right) + \left(\frac{P_{out,vent} - P_{out,chan}}{\dot{m}} \right)} \quad (13)$$

To improve the venting fraction this resistance ratio must be maximized, and from the relation given, can be achieved through:

- Thin, highly permeable membranes.
- Long, small hydraulic diameter liquid-side channels. This is also thermally beneficial as reducing the hydraulic diameter improves the heat transfer coefficient.
- Low aspect ratio liquid channels, where width $\gg$ depth, to maximize the venting area. However this also leads to a loss in heat transfer area.
- Short, large hydraulic diameter, vent channels.
- Increasing the static trans-membrane pressure through addition of an outlet choke or vent side vacuum, the latter's effect is shown experimentally in Figs. 10 and 13. However, the trans-membrane pressure is limited by the leakage pressure limit for the given membrane.

5.4. Condensation and membrane wear/fouling

Two practical issues that need to be considered in the use of vapor venting membranes in two-phase heat exchangers is vapor condensation within the membrane and membrane wear and fouling over time.

Condensation occurs when the temperature of the membrane drops below the local saturation temperature of the vapor. If the rate of condensation in the membrane is higher than the rate of evaporation the membrane becomes saturated with liquid over time and impedes vapor transport. The impact of condensation has been studied numerically by Fang et al. [18] and experimentally by David et al. [50] and it was found that a small amount of condensation effectively stops vapor from venting. Condensation occurring on the surface of the PTFE membrane and within the membrane was also observed during diabatic operation of a silicon vapor-venting device [51]. Vapor continued to vent in the silicon device however, due to the use of a single, thin and highly hydrophobic PTFE membrane that allowed the condensate to be easily removed. Condensation did not play a measurable role in the devices presented in the current study either. Surface insulation and a large contact area between the heated substrate and the capping substrate helped maintain a high temperature in the vent channels and membrane during operation. Maintaining the temperature of the membrane over that of the saturation temperature as well as enabling easy removal of condensate from the membrane, are key to maintaining device performance.

Membrane wear and fouling over time is an important issue to consider as the membrane properties could be changed to a degree that render the heat exchanger ineffective. Changes in the hydrophobicity [52], porosity, pore diameter or pore structure can

change the permeability and leakage prevention of the membrane as well as flow-regime and heat transfer characteristics. Membrane fouling by particulates would result in eventual pore blockage and the loss of venting ability. We carried out SEM imaging of the membranes before and after approximately 30 hours of cyclical operation and found little observable physical change or fouling in the membrane. Repeatability in venting and pressure drop data between runs confirm that wear and fouling is not significant during the course of our work. However, much longer duration studies of device performance and membrane fouling and wear are required before any conclusions can be drawn regarding the long term vapor venting heat exchanger performance under real world operating conditions.

6. Summary of findings and conclusion

Our experimental and numerical work on the vapor separation and venting from a two-phase flow led to several findings that are summarized below.

- Pressure drop in the venting devices were experimentally found to be $\sim 60\%$ lower than for the non-venting control devices with the model showing excellent agreement at the lowest mass flux of 102 kg/s m^2 . Additional pressure drop improvement, believed to be due to changes in the flow regime from stratified to churning-annular, was found as the mass flux was increased to 420 kg/s m^2 . This additional pressure drop is found to be as much as 50% when using the non-venting membrane device at the highest mass flux.
- Reduction in the pressure drop is predicted to mitigate the occurrence of detrimental static Ledinegg instabilities. Simulation of demand curves of non-venting and venting devices find that the negative slope, $-\delta(\Delta P)/\delta G$, for the venting devices is reduced and could potentially be lowered below that of the supply curve.
- Temperatures in the copper venting devices were improved by an amount equivalent to the reduction in the saturation temperature and as much as 4.4°C at 208 kg/s m^2 and 730 kW/m^2 total input heat flux. Temperatures in the non-venting membrane devices were found to be worse than in the non-venting control.
- Two-phase heat transfer coefficients in the venting devices were found to be approximately the same or higher as that in the control devices for mass fluxes of 208 , 307 and 420 kg/s m^2 but lower at a mass flux of 102 kg/s m^2 . Though reduction in quality is expected to improve heat transfer rates during venting, reduction in saturation pressure and changes in flow regime could negatively impact it. Churn-annular type flows at higher mass fluxes is preferred due to similar thermal characteristics as typical annular flows.

The vapor venting heat exchanger presented here is currently applicable to water based heat exchangers. Sub-ambient pressure operation is required to lower the boiling point of the water to levels suitable for electronics cooling. A key area of future research is the experimental and numerical study of vapor venting when using refrigerants and dielectric fluids, more commonly used in two-phase cooling of electronics.

The true advantage of such a device over a hydraulically or thermally equivalent non-venting device is manifest when we have very small liquid channels to maximize the heat transfer coefficients coupled with very large vent channels. In such a case we enjoy the thermal benefits of having small channels with the hydraulic and stability benefits of larger channels. The hydraulic, thermal and potential stability improvements as a consequence

of vapor venting opens the door for further work into the practical implementation of two-phase microfluidic heat exchangers for high performance cooling applications.

Acknowledgements

The authors would like to acknowledge the support of the MARCO Interconnect Focus Center, one of five research centers funded under the Focus Center Research Program, a Semiconductor Research Corporation (SRC) program. The author also appreciates support from the SRC as well as the Stanford Graduate Fellowships. The authors also thank the Stanford TSD Machine-shop for fabricating the copper samples.

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