HEAT TRANSFER DUE TO MICROSCALE THIN FILM EVAPORATION FROM THE STEADY STATE MENISCUS IN A COHERENT POROUS SILICON BASED MICRO-COLUMNATED WICKING STRUCTURE

Navdeep S. Dhillon∗
Department of Mechanical Engineering
University of California, Berkeley
Berkeley, California 94720
Email: dr.ndhillon@gmail.com

Jim C. Cheng
Albert P. Pisano
University of California, Berkeley
Berkeley, California 94720
Email: appisano@me.berkeley.edu

ABSTRACT

A numerical fluid flow and heat transfer model is presented in order to study the evaporation characteristics of a stationary thin film liquid-vapor meniscus. The model is used to evaluate the evaporative heat transfer performance of micron-size rectangular channels on the surface of the secondary wick, inside a micro-columnated coherent porous silicon wick design. The shape of the liquid-vapor meniscus in the channel is obtained by solving the Young-Laplace equation, using a surface energy minimizing algorithm. Mass, momentum and energy equations are then solved in the liquid domain using a discrete finite volume method-based approach. The vapor is assumed to be fully saturated and evaporation at the liquid-vapor interface is modeled using kinetic theory. The effect of wall superheat and inlet-liquid subcooling on the rate of evaporation and associated heat transfer from the evaporating meniscus is characterized.

NOMENCLATURE

γ Surface tension.
θ Contact angle.
E Energy.
V Volume.
\( \dot{m} \) Evaporation mass flux.
\( \dot{\sigma} \) Accommodation coefficient.
\( M \) Molecular weight.
\( \bar{R} \) Universal gas constant.

∗Address all correspondence to this author.
INTRODUCTION

In recent years, there has been a tremendous thrust in the area of thermal management of electronics. Previously, thermal management was usually an afterthought for the Integrated circuit design engineers, but that approach is no longer tenable. Pursuant to Moore’s law, the number of transistors on a chip doubles every 18 months [1]. This increases the heat dissipated per unit area of the chip, which is further augmented by higher operating frequencies geared towards improving computation speeds. In addition to this, the number of chips in a multi-chip module and number of devices in small confined spaces in systems are increasing continuously [2]. Conventional thermal management solutions, such as conduction and fan based convection technologies, can no longer dissipate the heat generated in densely packed electronic modules. Further, it has been seen that the thermal hotspots can be roughly 3 to 8 times the average power density on a chip [1]. This means that the junction temperatures for many of these device are now persistently perilously close to the maximum operating limits. Microprocessor designers are now having to limit clock speeds and throttle performance due to lack of adequate cooling.

A host of new and improved cooling technologies are being investigated in response to these thermal management challenges facing the electronics industry. Prominent among these are liquid cooling solutions implemented in the form of heat pipes, microchannel heat sinks, jet-impingement and spray cooling. All of these except heat pipes are active cooling technologies, where external pumping power is required to force the liquid is the cooling system. Heat pipes belong to a class of passive, capillary driven, phase change systems, which also includes loop heat pipes and capillary pumped loops. Heat is absorbed in these devices due to thin film evaporation from the liquid-vapor meniscus, which is formed in the tiny pores of the wick structure. Surface tension induced pressure drop across the curvature of the meniscus provides the pumping action that maintains a continuous supply of liquid to the evaporating region. As opposed to heat pipes, loop heat pipes keep the liquid and vapor phases separate, thereby increasing performance. They also enable heat transport over larger distances by confining the wick to the evaporation area, which in heat pipes spans the entire length of the device. We are working on a microscale loop heat pipe, with the aim of developing a localized, high-heat-flux cooling system that can be integrated more easily with modern electronic substrates. One of the critical parameters of device performance is the rate of heat transfer due to evaporation from the liquid-vapor meniscus. In order to optimize the evaporator wick design, we need a way to predict evaporation rates for different wick topologies that can be fabricated using standard MEMS microfabrication techniques.

Wayner et al. [3] developed a theoretical model to calculate the heat transfer coefficient of the interline region in a wetting liquid film. Xu and Carey [4] developed an analytical model to predict heat transfer characteristics of film evaporation on a microgroove surface, and compared their results with experiments. They concluded that the disjoining pressure plays an important role in determining the rate of evaporation. Ma and Peterson [5] developed a mathematical model to predict the rate of evaporation and temperature distribution along the axial direction of a groove plate. They accounted for surface tension and disjoining pressure effects, and showed that the apparent contact angle, and hence the rate of heat transfer through the micro region, increases in the axial direction, and that temperature induced surface tension changes can be neglected in the micro region. Wang et al. [6] developed a numerical model, using the complete expression for the kinetic theory-based mass transport equation, to study heat transfer aspects of the liquid vapor meniscus in a two-dimensional microchannel. Beyond a channel width of a few microns, the rate of heat transfer in the thin film region is found to be relatively insensitive to channel size; Conversely, heat transfer in the intrinsic meniscus has a significant dependence on channel size. For the channel sizes considered, thin film accounts for only a small portion of the overall heat transfer, while most of it happens in a micro-region defined to be within a one micron thick meniscus. Morris [7, 8] analyzed the stationary meniscus of an evaporating perfectly wetting system to show that the apparent contact angle is dictated by a small scale flow, driven by evaporation near the contact line, which does not contribute much to the overall heat transfer. The apparent contact angle depends on material properties and wall superheat, and can be calculated separately in order to simplify the meniscus heat transfer problem, which can now be posed without worrying about the disjoining pressure and other nanometer scale effects. Ranjan et al. [9] developed a finite volume based 3-d numerical model to predict evaporative heat transfer rates for different wick materials used in heat pipes. They were able to characterize the effects of wick geometry, wick porosity, and liquid superheating on heat transfer from the liquid vapor interface.

OVERVIEW OF THE MICRO-COLUMNATED WICK

Rate of evaporation in the wicking structure is one of the critical operating parameters of the Micro-Columnated Loop Heat Pipe (μCLHP). Although the upper limit on the heat transfer carrying capacity of the μCLHP can be calculated by performing a pressure balance on the device flow loop [10], the actual heat transported by the device is limited by the rate of evaporation from the liquid-vapor meniscus in the wicking structure. Fig.1 shows the structure and operation of Coherent Porous Silicon (CPS) based MEMS dual-scale wicking structure used in the Micro-Columnated Loop Heat Pipe. The fabrication of the wick is tightly integrated with the Microfabrication process flow of the μCLHP device [11]. The dual-scale Micro-Columnated wick consists of two separate wick components: The primary wick is fabricated by etching vertical columns in a Coherent Porous
Evaporation occurs at the liquid-vapor interface

Silicon wafer using Deep Reactive Ion Etching (DRIE). The secondary wick is fabricated by etching rectangular microchannels on the silicon substrate, which is in contact with the heat source. The two substrates are then bonded to each other, whereby the primary wick is connected to the secondary wick.

The base of the primary wick is in contact with the liquid coming in from the liquid microchannels and the compensation chamber. Using the strong capillary action developed in the CPS pores, it wicks this liquid to the rectangular channels of the secondary wick. Here, the liquid spreads over the entire area of the hot surface, and evaporation occurs across the liquid vapor meniscus in the rectangular microchannels. Some amount of evaporation also occurs from the liquid-vapor meniscus in the CPS pores of the primary wick, but most of it happens from the secondary wick surface due to the larger surface superheat temperatures that exist close to the heat source. The primary wick, with its smaller pores, also acts as a buffer by preventing total wick dryout during sudden spikes in heat flux [12].

Fig. 2 shows the two different wick topologies that can be fabricated using MEMS Microfabrication techniques. A uni-directional-channel wick topology has parallel microchannels in the secondary wick and continuous column walls in the primary wick. The vapor formed due to evaporation has to travel along continuous column walls in the primary wick, while the primary wick consists of columns etched into the CPS base. (a) Uni-directional-channels wick topology consists of parallel channels in the secondary wick and continuous column walls in the primary wick. (b) Criss-crossing-channels wick topology consists of criss-crossing channels in the secondary wick and simple columns in the primary wick.
Liquid supply from the primary wick
Hot channel walls and bottom surface

Figure 3. The model problem of thin film evaporation in a rectangular microchannel of the secondary wick of the Micro-Columnated CPS wick structure. The liquid supply to the channel is from the CPS pores of the primary wick, and is specified as a liquid inlet boundary condition. Evaporation occurs from the liquid meniscus exposed to the ambient vapor, while heat is transferred to the liquid from the solid walls of the microchannel.

The column walls on its way out to the vapor line. In another wick topology, criss-crossing channels can be etched to form the secondary wick and cubical columns used to wick the liquid from the primary wick to the evaporating surface in the secondary wick.

MODELING THIN FILM EVAPORATION

In order to characterize heat transfer performance of the proposed wick topologies, we need to compute the rates of evaporation in these structures. As discussed above, most of the evaporation is expected to occur from the liquid-vapor meniscus in the rectangular channels of the secondary wick, which is in direct contact with the heat source. During normal device operation, the function of the primary wicking structure is mainly to convey the liquid to the secondary wick for evaporation. It does this by using the capillary force developed across the liquid-vapor meniscus in its tiny CPS pores. Once the liquid reaches the evaporating surface, it is pulled into the rectangular channels of the secondary wick, which again is due to the capillary pressure across the meniscus formed in the channels. If no heat is applied to the secondary wick—and gravitational forces are neglected compared to those due to surface tension, due to the small length scale of the problem—the rectangular channels should be completely filled with the working liquid. In the presence of a heat source, the liquid supply to the secondary wick is balanced by the evaporation from the channels. The rate of evaporation will depend on wall superheat and the shape of the liquid vapor meniscus in the channel, which in turn will depend on the rate of liquid supply from the CPS pores of the primary wick.

Problem Statement

The thin-film evaporation problem to be solved is shown in Fig.3. It consists of a rectangular microchannel on the surface of the secondary wick, which obtains it liquid supply from the CPS pores of the primary wick, at the liquid inlet. The liquid, which wets the channel material, is pulled into the channel by the surface tension related capillary forces. The heat, supplied to the liquid by the hot channel walls, causes evaporation of liquid at the liquid-vapor interface. In steady state, the liquid supply to the channel is balanced by the rate of evaporation of the liquid, which results in a stationary evaporating meniscus.

The amount of liquid in the wick microchannel will depend on a complex interrelationship between the surface superheat—which dictates the rate of evaporation—and the thermodynamics of the flow loop—which involves a balance between the capillary pressure across the liquid-vapor meniscus in the wick microchannel and the pressure drop due to fluid flow in the rest of the µCLHP device [10, 12].

In order to pose the present problem of steady state thin-film evaporation in a secondary wick microchannel, we will assume that the amount of liquid in the channel is dictated by extraneous parameters that can be modeled separately [10, 12]. As it will be shown in the next section, using a fixed value for the amount of liquid in the channel, we can compute the shape of the liquid meniscus in the channel. A thermal fluid solver, coupled with a kinetic theory-based evaporation boundary condition at the liquid-vapor meniscus, will then be implemented in the liquid domain.

SHAPE OF THE LIQUID-VAPOR MENISCUS

The shape of a non-evaporating, static liquid-vapor meniscus can be obtained by solving the Young-Laplace equation for a given solid-liquid contact angle. This contact angle only depends on the solid, liquid, and vapor present at the contact line. For a perfectly wetting system, the contact angle is equal to zero. On the other hand, the stationary meniscus of an evaporating perfectly wetting system exhibits a non-zero apparent contact angle; This contact angle vanishes as the wall heat source, responsible for evaporation, is turned off [7]. Using a scaling analysis, Morris [7] has shown that the theory of evaporation at the liquid-vapor interface can be divided into two distinct theories of an inner and outer structure. The nanometer-scale inner structure near the contact line determines the apparent contact angle; however, it contributes only a small fraction to the total evaporation heat transfer. Most of the heat transfer happens in the larger scale
outer region, which subtends an apparent contact angle on the wall. By assuming a given value for the apparent contact angle, the problem of heat flow computation can be decoupled from the nanoscale physics governing the apparent contact angle.

To calculate the shape of the liquid-vapor meniscus, we use a numerical surface energy minimizer called Surface Evolver [13]. In this method, the surface is implemented as a union of triangles; free energy can be attributed to the surface and its edges. The surface is iterated upon to minimize the free energy, subject to geometrical constraints on vertex positions as well as constraints on integral quantities such as volume. During the iteration procedure, the surface mesh can be tuned using procedures such as equi-angulation, vertex averaging and mesh refinement [14].

Figure 4 shows the problem domain for the generation of the liquid-vapor meniscus in Surface Evolver. Vertices, edges, and faces are used to define the liquid in the open rectangular microchannel shown in Fig. 3. Due to symmetry, only half of the liquid in the channel is modeled. Geometrical constraints on the vertices and edges are used to define the channel wall and the bottom surface. Similarly, constraints are applied on the liquid-vapor meniscus surface and edges to keep the liquid within the channel (see Tab. 1).

### Surface Free Energy

In order to obtain the shape of the stationary meniscus, Surface Evolver needs to minimize the surface tension related free energy associated with the liquid-vapor, liquid-solid, and solid-vapor interfaces. Since the solid-vapor interface is not defined in the problem domain (Fig. 4), a virtual surface tension is instead assigned to the liquid-solid interface, which depends on the liquid-vapor surface tension $\gamma$ and the liquid-solid contact angle $\theta$ as follows:

$$
\gamma_{virtual} = -\gamma \cos(\theta) \tag{1}
$$

But, assigning surface tension values to all surfaces in this way leads to evolution problems for vertices that are located on flat faces. The solution is to use Green’s theorem to convert surface free energy integrals to line integrals over the edges bounding the surface. Using this method, only the liquid-vapor surface needs to be iterated upon to minimize the free energy of the system. Edge surface tension-related free energies for the different faces can be calculated as follows:

**Channel bottom:** The following energy term is applied to edge 4, through constraint 1

$$
E_1 = \int_{\text{face 4}} (-\gamma \cos \theta_b \hat{j}) \cdot d\vec{A} = \int_{\text{edge 4}} \gamma \cos \theta_b \hat{j} \cdot d\vec{s} + c \tag{2}
$$

**Liquid symmetry boundary** Since this boundary lies inside the liquid, it has no associated free energy.

**Channel wall:** The following energy term is applied to edge 3, through constraint 3

$$
E_3 = \int_{\text{face 3}} (-\gamma \cos \theta_w \hat{j}) \cdot d\vec{A} = \int_{\text{edge 3}} \gamma \cos \theta_w \hat{j} \cdot d\vec{s} + c \tag{3}
$$

---

**Table 1. VERTEX, EDGE, AND FACE CONSTRAINTS FOR MENISCUS SHAPE DEFINITION**

<table>
<thead>
<tr>
<th>Index</th>
<th>Constraint</th>
<th>Vertices</th>
<th>Edges</th>
<th>Faces</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Fixed</td>
<td>2, 3</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>$x = 0$</td>
<td>1, 4</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>$y = 0$</td>
<td>1</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>$y = w$</td>
<td>4</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>$x \leq h$</td>
<td>1, 4</td>
<td>1, 3</td>
<td>1</td>
</tr>
<tr>
<td>5</td>
<td>$x \geq 0$</td>
<td>1, 4</td>
<td>1, 3</td>
<td>1</td>
</tr>
</tbody>
</table>

5 Copyright © 2011 by ASME
Fluid inlet: Since this boundary lies inside the liquid, it has no associated free energy.

Liquid-vapor meniscus: A surface energy term is applied to the liquid-vapor meniscus by specifying the value of liquid-vapor surface tension $\gamma$ for face 1.

Volume of the Liquid
To constrain the amount of liquid inside the meniscus, Gauss's theorem is used to express volume as a surface integral as follows:

$$V = \iiint_V dV = \iiint_A \hat{F} d\hat{A} \text{ where } \nabla \hat{F} = 1$$ (4)

By default, Surface Evolver specifies $\hat{F} = \hat{z}k$. The volume of the liquid is a sum of the surface integral of $\hat{F}$ over all the liquid faces. This integral can be calculated for the individual faces as follows:

Channel bottom:

$$\iiint_{\text{face 4}} \hat{F} d\hat{A} = \iiint_{\text{face 4}} \hat{z}k d\hat{A} = 0 \text{ since } \hat{k} \perp d\hat{A}$$ (5)

Liquid symmetry boundary:

$$\iiint_{\text{face 2}} \hat{F} d\hat{A} = \iiint_{\text{face 2}} \hat{z}k d\hat{A} = 0 \text{ since } \hat{k} \perp d\hat{A}$$ (6)

Channel wall:

$$\iiint_{\text{face 3}} \hat{F} d\hat{A} = \iiint_{\text{face 3}} \hat{z}k d\hat{A} = 0 \text{ since } \hat{k} \perp d\hat{A}$$ (7)

Fluid inlet:

$$\iiint_{\text{face 5}} \hat{F} d\hat{A} = \iiint_{\text{face 5}} \hat{z}k d\hat{A} = 0 \text{ since } z = 0$$ (8)

Liquid-vapor meniscus: Surface Evolver calculates the surface integral of $\hat{F}$ automatically.

In this study, to calculate the shape of the meniscus, a contact angle of $29^\circ$ is assumed for both the channel bottom and walls. The volume of the half-meniscus is assumed to be $50 \mu m \times 50 \mu m \times 100 \mu m$.

EVAPORATION AT THE LIQUID-VAPOR INTERFACE
Mass flux due to evaporation at the liquid-vapor interface is given by [6,9]

$$\dot{m}'' = \frac{2\sigma}{2-\sigma} \left( \frac{M}{2\pi R} \right)^{1/2} \left( \frac{p_{v,\text{equ}}(T_{lv}) - p_v}{T_v^{1/2}} \right)$$ (9)

where $M$ is the molecular weight of the fluid in kg/kmol, $\sigma$ is the accommodation coefficient, $R$ is the universal gas constant in J/molK, $p_{v,\text{equ}}$ is the equilibrium vapor pressure, and $T_{lv}$ is the temperature at the liquid-vapor interface. The value of the accommodation coefficient for non-polar liquids is usually equal to one, but its value for water is a matter of much debate [15]; therefore, in our study we will assume $\sigma = 1$. The equilibrium vapor pressure, i.e., the pressure at which the vapor is in equilibrium with the liquid, is given by

$$p_{v,\text{equ}}(T_{lv}) = p_{\text{sat}}(p_{lv}) \exp \left[ \frac{p_{v,\text{equ}} - p_{\text{sat}}(T_{lv}) - (p_d + p_c)}{p_l T_{lv} R / M} \right]$$ (10)

where $p_l$ is the density of the liquid, $p_d$ is the disjoining pressure, $p_c$ is the capillary pressure, and $p_{\text{sat}}$ is the saturation pressure of the liquid. In the absence of capillary and disjoining pressures, the equilibrium vapor pressure is given by

$$p_{v,\text{equ}} = p_{\text{sat}}(T_{lv})$$ (11)

The saturation pressure of the liquid $p_{\text{sat}}$ is a function of temperature, and can be found at the vapor-liquid interface as follows:

$$p_{\text{sat}}(T_{lv}) = p_{\text{sat,ref}}(T_{\text{sat,ref}}) \exp \left[ \frac{M h_f g}{R} \left( \frac{1}{T_{\text{sat,ref}}} - \frac{1}{T_{lv}} \right) \right]$$ (12)

where $T_{\text{sat,ref}}$ and $p_{\text{sat,ref}}$ are reference saturation temperature and pressure respectively.

GOVERNING EQUATIONS AND BOUNDARY CONDITIONS

Governing Equations
Non-isothermal, incompressible, laminar, and unsteady mass, momentum, and energy equations are solved in the liquid domain. The vapor is assumed to be saturated, with a given vapor temperature. The effects of disjoining and capillary pressures are not modeled explicitly; instead, disjoining pressure related flow dynamics is accounted for by assuming a Capillary number-dependent apparent contact angle [7,8] for the evaporating meniscus. Capillary pressure related heat transfer suppression effects are negligible for high superheat values [6], and can...
Mass conservation For incompressible flow with a mass source term, the mass conservation equation reduces to

$$\nabla \cdot \vec{u} = \frac{S_m}{\rho}$$  \hspace{1cm} (13)

where $\rho$ is the liquid density and $S_m$ is a mass source term applied to the cells adjacent to the liquid-vapor interface boundary to simulate evaporation at this interface. The mass source term is negative and is given by:

$$S_m = -\frac{n^m A_f}{V_{cell}}$$  \hspace{1cm} (14)

where $n^m$ is the evaporative mass flux at the interface (see Eqn. 9), $A_f$ is the area of the cell face adjacent to the liquid-vapor interface, and $V_{cell}$ is the volume of the corresponding cell.

Momentum conservation The incompressible momentum conservation equation is given by

$$\frac{\partial \vec{u}}{\partial t} + \vec{u} (\nabla \vec{u}) = -\frac{\nabla p}{\rho} + \nabla (\nu \nabla \vec{u})$$  \hspace{1cm} (15)

where $\vec{u}$ is the liquid velocity, $p$ is the pressure, and $\nu$ is the kinematic viscosity.

Energy conservation The energy equation for incompressible fluid flow is given by

$$\frac{\partial T}{\partial t} + \nabla . (\vec{u} T) - (\nabla \vec{u}) T = \nabla . (\alpha \nabla T)$$  \hspace{1cm} (16)

where $T$ is the liquid temperature and $\alpha$ is the thermal diffusivity. Using Eqn. 13 we can write this as

$$\frac{\partial T}{\partial t} + \nabla . (\vec{u} T) = \nabla . (\alpha T) + \left( \frac{S_m}{\rho} \right) T$$  \hspace{1cm} (17)

where the last term on the R.H.S. is a heat source term, which appears due to the evaporative mass flux in the cells adjacent to the liquid-vapor meniscus.

Boundary Conditions

Fig. 5 shows the problem domain and boundary conditions on which the numerical solution is implemented. The domain consists of the stationary liquid-vapor meniscus in an open rectangular channel that is 100 $\mu$m wide and 50 $\mu$m deep. This channel is supplied with liquid by an orthogonally connected 25 $\mu$m $\times$ 25 $\mu$m rectangular channel.

Liquid-vapor meniscus The evaporative mass and heat fluxes through the meniscus are taken care of by incorporating appropriate source terms in the mass and energy conservation equations. Therefore, the normal velocity and temperature gradient is set to zero at this boundary. A zero-gradient pressure boundary conditions is also applied.

$$\frac{\partial p}{\partial n} = p' = 0; \quad \vec{u} . \hat{n} = u_n = 0; \quad \frac{\partial T}{\partial n} = T' = 0$$  \hspace{1cm} (18)

Liquid inlet At the liquid inlet, a zero-gradient pressure and velocity boundary condition is applied. The temperature of the incoming liquid is set to $T_i$.

$$\frac{\partial p}{\partial n} = 0; \quad \frac{\partial \vec{u}}{\partial n} = 0; \quad T = T_i$$  \hspace{1cm} (19)
Wall and bottom A no-slip velocity boundary condition and a zero-gradient pressure boundary condition is applied at the wall and bottom. The temperature of the wall and bottom is set to $T_w$.

$$\frac{\partial p}{\partial \hat{n}} = 0; \quad \vec{u} = 0; \quad T = T_w$$ \hspace{1cm} (20)

Adiabatic walls A no-slip velocity boundary condition and a zero-gradient pressure boundary condition is applied at the adiabatic walls. The normal temperature gradient is set to zero.

$$\frac{\partial p}{\partial \hat{n}} = 0; \quad \vec{u} = 0; \quad \frac{\partial T}{\partial \hat{n}} = 0$$ \hspace{1cm} (21)

Liquid symmetry boundary Since we are modeling only half of the channel, a symmetric boundary condition is applied at the face that aligns with the middle of the channel.

Numerical Solution

The above governing equations and boundary conditions are solved in OpenFOAM, an open source CFD toolbox. The solution domain is discretized with a hexahedral mesh, using the blockMesh and snappyHexMesh utilities in OpenFOAM. Since there are mass source terms in the solution domain, mass and momentum equations are first solved for a given uniform liquid temperature $T_i = T_w$, in order to obtain a divergence-less velocity field. This is accomplished by solving the steady form of these equations using a SIMPLE algorithm, which uses under-relaxation. Using this result as a starting point, the complete set of equations is now solved in time using the PISO loop until a steady state is reached. The time step during the solution procedure is limited by the maximum Courant number in the domain.

RESULTS AND DISCUSSION

Thin film evaporation of water in a rectangular channel of the secondary wick was studied for different values of wall superheat and liquid inlet subcooling temperatures. The temperature $T_i$ and pressure $p_v$ of the saturated vapor were, in all cases, set to 373.15 K and $1.013 \times 10^5$ Pa respectively.

In case I, the incoming liquid is assumed to be saturated with a temperature of $T_i = 373.15$ K, while a wall superheat of $T_w - T_i = 5$ K is applied at the channel walls. The effect of increase in wall superheat is studied in case II, where a wall superheat of 10 K is applied to the saturated incoming liquid. In case III, a wall superheat of 10 K is applied to the channel walls while the inlet liquid is subcooled by $T_v - T_i = 5$ K.

Fluid Flow and Heat Transfer Characteristics

Fig. 6 shows the numerical simulation results for case I, where the incoming liquid is saturated at $T_i = 373.15$ K, while the channel wall is 5 K above the liquid temperature, at $T_w = 378.15$ K. From Fig. 6a we can see that, the liquid enters the domain from the liquid inlet and flows towards the liquid-vapor meniscus, where it undergoes evaporation to the vapor phase. We note that the maximum velocity in the domain is 0.448 m/s, with comparatively larger values existing in the adiabatic supply channel due to its smaller cross-section. The total pressure drop in the domain, from the liquid inlet to the evaporating meniscus, is approximately 336 Pa (see Fig. 6b).

Fig. 6c plots the temperature distribution in the domain; the first thing that we readily see is that most of the liquid in the supply channel is close to the inlet temperature $T_i$. This is easily explained, since the supply channel walls are adiabatic and wa-
A plot of the liquid velocity vectors inside the evaporating meniscus for different wall superheats.

**Effect of Wall Superheat on Heat Transfer**

Fig. 8 shows the numerical simulation results for case II, where a higher wall superheat of 10$K$ is applied to the incoming saturated liquid. The liquid inlet temperature is still kept at 373.15$K$, while the wall temperature is increased to 383.15$K$. Compared to case I, overall, we see an increase in all the relevant field quantities such as velocity, pressure drop across the domain, and evaporation heat flux values. This is just a consequence of the fact that there is more fluid evaporation across the meniscus as a result of the increased fluid temperatures, brought about by a higher wall superheat. The average heat flux over the entire meniscus increases to $8.35 \times 10^7 W/m^2$, a little less than double the value obtained for case I. The total rate of heat loss due to evaporation is now 0.75$W$.

**Effect of Liquid Sub-cooling on Heat Transfer**

Fig. 9 shows the numerical simulation results for case III, where the wall superheat is maintained at 10$K$, but the inlet liquid is subcooled by 5$K$ compared to the saturated vapor. The wall temperature in this case is 383.15$K$, while the liquid inlet temperature is 368.15$K$. Interestingly, there is a large drop in the overall rate of evaporation compared to case II, and this is reflected in the lower values of fluid velocity and pressure gradients as well. As to why this is the case, we look at Fig. 9d and note that although the maximum evaporative heat flux in the domain ($1.7 \times 10^8 W/m^2$) is the same as in case II, now we also have a considerable amount of condensation in part of the incoming liquid that is subcooled. The average evaporation heat flux and the total rate of evaporative heat loss, at $4.63 \times 10^7 W/m^2$ and 0.42$W$ respectively, are even less than the corresponding values for case I.
Application of the Results

The presented numerical results need to be interpreted in the context of thin-film evaporation in the micro-columnated wick. It should be kept in mind that the channel size for the secondary wick needs to be optimized, in order to adequately promote its wetting and maximize the evaporative heat transfer. This can be done by integrating the above evaporation model with a broader flow loop model for the micro-columnated loop heat pipe [10].

Decreasing the secondary wick channel size promotes wetting by increasing capillary forces, which draw more liquid into the channel; This will lead to a larger fluid volume, and hence a large surface area for evaporation. On the other hand, the evaporative heat flux, per unit area of the meniscus, will decrease due to capillary suppression effects. As a result, more liquid superheat will be required in order to maintain the rate of evaporation; this can lead to wick dry-out due to spontaneous homogeneous bubble nucleation.

Large secondary wick channels, despite promising higher thin-film evaporation rates due to lower capillary suppression effects, will be susceptible to dry-out due to insufficient liquid wetting of the wick. An appropriate channel size should be chosen in order to provide the predicted rate of evaporation, while maintaining the capillary pressures necessary for device operation.
CONCLUSIONS

A numerical model was implemented in order to solve for the rate of evaporative heat transfer from the surface of a stationary liquid-vapor meniscus. The aim of the model is to study the process of microscale thin-film evaporation inside a dual-scale micro-columnated coherent porous silicon wicking structure. The shape of the meniscus inside a rectangular channel, on the surface of the secondary wick, is obtained by solving the Young-Laplace equation using a surface energy minimizing algorithm called Surface Evolver. This approach ignores the effect of actual flow mechanics on the shape of the meniscus. Mass, momentum, and energy equations, containing source terms to account for evaporation from the meniscus surface, are discretized and solved using the finite volume approach. Numerical simulation results show that most of the evaporative heat transfer from the meniscus happens near the channel walls. Increasing wall superheat increases the overall rate of evaporation, whereas subcooling the inlet liquid decreases the same.

Acknowledgments


REFERENCES