NANOSCALE WICKING STRUCTURES

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ABSTRACT

Heat pipes are ubiquitous in various heat transfer applications due to their low maintenance and lack of moving parts. Their simplicity makes them compact and ideally suited for microelectronics use. Recirculation of the coolant in a heat pipe is done passively by means of a wicking structure that induces capillary-driven flow from the condenser to the evaporator. This fluidic scheme is highly desirable but requires precise optimization of the wicking structure geometry to provide the required coolant flow rates under different heat loads.

In this paper we present an ab initio model that simulates the capillary flow within a wicking structure of regular and periodic geometry. An energy formulation incorporating capillary equations for pressure gradient and the Stokes flow equation for frictional dissipation were used in the analysis. The feasibility of using nanostructures for capillary-driven flow was assessed using this theoretical analysis. This model is specifically designed to simulate a nanopillar array wick (or nanowick) but was also extended to incorporate commercially available homogenous wicks through the use of a general Darcy’s flow approach. A Darcy’s flow analysis requires knowledge of the porous structure permeability ($k$), which must be empirically determined. However, our first principles approach can be used to estimate the effective permeability of various commercial wicks. Only the characteristic structural dimensions of a wick are needed in our model for an accurate estimate of the permeability and the maximum flow rate the wick can sustain without the necessity for an empirical correlation.

The results of the theoretical model were corroborated through experimental measurements of baseline mesh wicks and nanowicks. Since the thermal performance of most heat pipes is usually capped by the capillary limit, this threshold was examined for each wick by measuring the mass flow over time at different heat fluxes. At high heat fluxes, the wick cannot sustain the fluid flow necessary for heat removal and burnout occurs. This phenomenon occurs at the thermal capillary limit. The mass flow ceases to increase in the case of burnout and may actually decrease if a disruptive vapor film is created. Experiments show that the baseline wicks were found to have higher mass flow rates when compared to a nanowick due to the difference in thickness of the wicks. However, when the data were normalized to produce velocity values, the nanowick was found to have a higher velocity than most of the baseline wicks. These experimental results were weighed against the theoretical model results showing very good agreement of the two.

INTRODUCTION

With the reduction in transistor size and consequential increase in chip density within the last quarter century, heat removal has become an important issue in modern day electronics [1]. Future computer chips are projected to generate a heat flux greater than 150 W/cm² with localized sub-millimeter zones with heat fluxes as high as 1 kW/cm² [2]. Although electronics have become smaller, they still produce large amounts of heat, translating into high heat fluxes that require efficient heat removal devices that are equally small, yet reliable. One way to address the reliability requirement is by heat pipes. With no moving parts, heat pipes are extremely robust components that rely solely on capillary forces to move the fluid through the system.

Although there are many factors that limit the heat flux removal capability of a heat pipe system, the most significant one is the capillary wicking limit [3]. Properly understanding the fundamental physics behind this limit is imperative to the implementation of heat pipes in future generation high power electronics.

There has been extensive research done on microgroove, sintered powder and mesh-type heat pipes [4-11]. However, despite recent advances in microfabrication and nanotechnology there has been very little work in the area of wicking assemblies based on arrays of nanostructures, such as
nanotubes and nanowires [12-14]. This research examines wicking capillary structures produced by arrays of nanopillars. Nanopillars have been found to exhibit wicking characteristics despite being potentially hydrophobic (depending on the specific dimensions and characteristics of the wick and fluid) [14]. Since standard heat pipes achieve wicking through hydrophilic means, the inherent hydrophobic characteristics of some nanostructures must be compensated for through the use of surfactants and optimized geometrical arrangements in the wicking portion.

Most heat pipe analyses consider the system as a whole. Typical models are lumped in nature, consisting of an evaporator, adiabatic/wicking and condenser section, and depict the coupled behavior of these components as a unit [3, 5, 8]. Likewise, experimental characterization work deals with fully enclosed systems, where the performance of individual components is not well understood. Since the capillary wicking limit can be associated with a critical heat flux (i.e. where dry-out/burn-out occurs), it is important to characterize this behavior on the wicking structure as a stand-alone component. For these purposes, an elementary capillary-driven model was developed for a homogenous nanopillar array to assess the limiting conditions under which dry-out or (incipient) disruptive burn-out limit would occur for existing commercially available wicks and nanostructures. The model was developed from first principles and correlated to a Darcy’s flow equivalent [15].

In addition, experiments were carried out to quantify the capillary performance of the different nanostructured wicks and their commercially available counterparts (sintered powder, mesh). These results were cross-validated against the capillary-driven model for several homogenous wicks.

**NOMENCLATURE**

- $\delta$: Fluid inflection distance inward
- $\kappa$: Permeability constant
- $\theta$: Contact angle of fluid on solid interface
- $\mu$: Dynamic viscosity of the fluid
- $\rho$: Density of the fluid
- $\sigma$: Surface tension of fluid
- $A_c$: Cross sectional area of fluid flow
- $A_{surf}$: Surface area
- $c_p$: Specific heat capacity
- $d_a$: Diameter of nanotube pillar
- $H$: Height of fluid at point $i$
- $h_{fg}$: Heat of formation
- $h_{air}$: Heat transfer coefficient of air
- $h_{water}$: Heat transfer coefficient of water
- $k$: Thermal conductivity of a solid
- $L_{nick}$: Length of wick
- $m$: Mass flow rate
- $P$: Pressure of fluid
- $q$: Heat loss
- $Q_m$: Mass flow rate from point $i$ to point $j$
- $Q_v$: Volumetric flow rate from point $i$ to point $j$
- $r_{eff}$: Effective pore radius
- $R$: Radius of curvature
- $s$: Spacing between pillars
- $T$: Temperature
- $v$: Velocity
- $w_n$: Center to center nanotube spacing
- $w_s$: Width of wick sample

**CAPILLARY FLOW IN HOMOGENOUS WICKS UNDER THERMAL LOAD**

**Capillarity**

The main driving forces in a standard heat pipe are the capillary wicking forces within the wick. Succinctly, capillary pressures arise as a consequence of curved free surfaces. To model the capillary forces in an array of nanopillars whose spacing is much smaller that their height, a force balance was used as shown in Figure 1. Surface tension acts on the edges of the fluid and solid interface introducing an upward force (denoted by black arrows). The surface tension forces act in the lateral and normal direction as shown in Figure 2 and can be broken down into an x and y-component with the contact angle of the liquid-solid interface. In order to balance these forces, a pressure difference between the liquid and atmosphere acts on the projected cross sectional area (denoted by the opposing green block arrows). The balance of these two forces yields the equation [16]

$$\Delta P_{cap} = \frac{2\sigma \cos \theta}{w_n}$$  \hspace{1cm} (1)

where $\sigma$ is the surface tension of the interface, $\theta$ is the contact angle, $w_n$ is the width or spacing between the nanopillars and $\Delta P_{cap}$ is the difference between the pressure of the atmosphere ($P_{atm}$) and the pressure of the liquid at the liquid-gas surface ($P_{liq}$).

![Figure 1. Balance of surface tension and pressure forces](image-url)
Figure 2. Geometric analysis of the fluid’s radius of curvature

The difference in pressure at the surface ($AP_{cap}$) is also known as the driving capillary pressure induced by the nanowick. This capillary pressure force is assumed to act on an advancing wetting front as defined by Figure 3. Although the analysis is confined to flow within a single row of nanopillars, Figure 4 shows the advancing wetting front for a whole set of nanopillar rows within the wick. This model assumes that the flow within the nanopillary array is primarily 1D, which is a valid assumption for the high aspect ratio of heat pipe wicks, where flow is primarily from the condenser end to the evaporator end. The width ($W_n$) and spacing ($s$) can be considered to be of the same order, if not equal, in the case of a homogenous wick although Figure 3 exaggerates differences for visual purposes. The wetting of a surface by a fluid in an array of pillars can be attributed to this advancing front caused by the capillary forces.

From equation (1) it can be deduced that to increase the driving force in a wick, the fluid should exhibit a high surface tension ($\sigma$) and an extremely low (0°) or high (180°) contact angle ($\theta$). However, if the contact angle was high, then the wick structure would be hydrophobic, inducing dewetting and flow in the reverse direction – i.e. the wick would dry-up and repel liquid in general. From the denominator, it can be seen that a small pillar width or spacing ($w_n$) would be ideal for maximizing capillary forces. The need for smaller pillar spacing (or effective pore radii) justifies the use of nanostructures for wicking purposes [17].

However, since there are higher frictional losses associated with smaller pore sizes, a theoretical model simulating capillary flow of a wick must also include the consideration of viscous losses.

Figure 3. Control volume for analysis of an advancing front through a pillar forest

Viscous Losses

To model the viscous pressure losses acting against the capillary flow, a flat plate approximation was applied to the pillars within the wick and the fluid was assumed to be incompressible. Figure 5 shows a top view of the 2D flow with a flat plate approximation applied to two pillars within the fluid. The frictional losses from the substrate can be neglected due to the high aspect ratio and close spacing between the nanopillars (the diameter and spacing between the pillars is in the order of nanometers whereas they are microns tall). Thus, the flow generated between the two pillars was approximated to be a steady state Poiseuille flow between two flat plates as shown in Figure 5.

Starting from the momentum equation, the pressure loss between two flat plates can be derived to be [18]
Figure 5. Flat plate approximation applied to two pillars and the resulting velocity profile

\[ \Delta P_{\text{loss}} = \frac{12 \mu d_n}{w^2_A} Q_v \]  

(2)

where \( \Delta P_{\text{loss}} \) is the pressure drop past two pillars caused by frictional losses. \( Q_v \) is the volumetric flow rate past the two pillars, \( \mu \) is the dynamic viscosity of the fluid, \( d_n \) is the diameter of a pillar, \( A_c \) is the cross sectional area of the fluid between the two pillars and \( w_n \) is the center to center spacing of the nanotubes. Equation (2) can be extended to incorporate the viscous losses over any specific length of wick \( x \) by introducing an effective viscous length parameter \( L_e \)

\[ L_e = \frac{d_n x}{w_n} \]  

(3)

This represents the actual length over which viscous losses are occurring for a given nominal wick length, such that

\[ \Delta P_{\text{loss,wick}} = \frac{12 \mu L_e}{w^2_n A_c} Q_v = \frac{12 \mu d_n x}{w^2_n A_c s} Q_v \]  

(4)

**Advancing Front Equilibrium**

Under steady state conditions, capillarity must be balanced by the viscous losses. Thus, equating equation (1) and equation (4) for a single row of pillars that is \( w_n \) wide within a wick length \( x \), a formulation for the wetting front velocity as a function of \( x \) can be obtained.

\[ Q_v = A_e \frac{dx}{dt} = \frac{w^2_n A_c \Delta P_{\text{cap}}}{12 \mu L_e} \]  

(5)

and

\[ v = \frac{dx}{dt} = \frac{w^2_n \Delta P_{\text{cap}}}{12 \mu L_e} = \frac{w^2_n s \Delta P_{\text{cap}}}{12 \mu d_n x} \]  

(6)

This equation gives the velocity of the wetting fluid front as function of its location for an array of pillars without the use of empirical permeability data. If \( x \) is substituted by the total wick length, \( L_{\text{wick}} \), equation (6) provides the maximum velocity attainable by the wick such that

\[ v_{\text{max}} = \frac{w^2_n s \Delta P_{\text{cap}}}{12 \mu d_n L_{\text{wick}}} \]  

(7)

Thus, equation (7) can be used to calculate the maximum mass flow rate that a given wick can sustain such that

\[ \dot{m}_{\text{max}} = \rho A_{e,\text{wick}} v_{\text{max}} = \rho w_n H v_{\text{max}} \]  

(8)

where \( A_{e,\text{wick}} \) is the total cross sectional area of the wick or the total width of the wick (\( w_n \)) multiplied by the height of the pillars (\( H \)).

The corresponding maximum heat dissipation that can be achieved by the system can be found from the combined enthalpy change due to phase change and heat convection of the fluid. From general thermodynamics, the equation [19]

\[ q_{\text{heat}} = \dot{m} \Delta h = \dot{m} (h_{fg} + c_p \Delta T) \]  

(9)

gives the total dissipated heat where \( q_{\text{heat}} \) is the heat flux, \( \dot{m} \) is the mass flow rate, \( h_{fg} \) is the latent heat of evaporation, \( c_p \) is the specific heat and \( \Delta T \) is the difference between the wick operating temperature (i.e., evaporator or test section temperature) and the initial fluid temperature (i.e., condenser or fluid reservoir temperature). Thus combining equation (8) and equation (9), one gets

\[ q_{\text{heat,max}} = \dot{m}_{\text{max}} \Delta h = \rho w_n H v_{\text{max}} (h_{fg} + c_p \Delta T) \]  

(10)
which is the maximum heat flux that the wick can sustain before dry-out starts to occur.

Darcy’s Flow Equivalent

Although an \textit{ab initio} method has been used to calculate the wick flow resistance for the specific nanopillar array, the same analysis can be used to generate a more general formulation based on Darcy’s flow through porous media. If the structural details of the wicking structure are not well known we can resort to Darcy’s law to express equation (5) and equation (7) in terms of the permeability constant (\(\kappa\)) – which is the foundation of Darcy’s law – such that [20]:

\[
Q_{\text{wick}} = A_{c,\text{wick}} \frac{dx}{dt} = \frac{\kappa A_{c,\text{wick}} \Delta P_{\text{cap}}}{\mu L_{\text{wick}}} \quad (11)
\]

where the capillary pressure expression of equation (1) must be replaced by the more general formula

\[
\Delta P_{\text{cap}} = \frac{\frac{2\sigma \cos \theta}{r_{\text{eff}}}} \quad (13)
\]

where \(r_{\text{eff}}\) is the effective pore radius of the porous medium. Comparing equation (7) and equation (12) it can be seen that for a nanopillar array of regular and repeating structure

\[
\kappa_{\text{nanopillar}} = \frac{w_{s}^{2} s}{12d_{n}} = \frac{s^{3}}{12d_{n}} \quad (14)
\]

Thus, since the lateral spacing (\(s\)) is approximately equal to the normal spacing (\(w_{s}\)) in a homogeneous nanopillar array wick, the permeability can be simplified to two dimensional parameters, spacing (\(s\)) and diameter (\(d_{n}\)). Although equation (14) is mainly relevant for nano and micropillar arrays of a specific geometry and repeating nature, it will be used as a first order approximation in the estimation of the permeability of other wicking structures of a more random nature.

EXPERIMENTAL SETUP

Critical Heat Flux Meaasuresments

Since the capillary limit is typically the limiting factor in moderate temperature heat pipes the thermal capillary limit of various wicks was investigated experimentally. At high heat fluxes, the wick can no longer sustain fluid flow necessary for heat removal and the wick dries out. This phenomenon is called burnout or dry-out and is indicated by an increased superheat and a fluid flow rate that does not increase with increasing heat flux. The superheat is defined as the surface temperature subtracted by the fluid temperature. In some circumstances, a vapor film caused by excessive nucleate boiling may form and reduce the effective area for heat transfer and actually lower the flow rate of the fluid [16].

In order to compare the capillary performance of different wick structures, the thermal capillary limit of various wicks was investigated by finding their critical heat flux and corresponding flow rate. In order to simulate a heat pipe evaporator surface, cartilage heaters were used to heat up a copper block that was insulated with mineral wool from the environment as shown in Figure 6. A 1.9 cm by 1.5 cm surface area was extruded from the copper block in order to provide a controlled heat flux testing region for the wick. By insulating the copper block with mineral wool, the majority of the heat is directed to the face of this testing region. The wick was then attached to the surface of this region by a ceramic piece made of MACOR®. MACOR® was used since it provided insulation and therefore limited the heat loss through the surface area that was not in contact with the wick. The subsequent exposed surface area was approximately 1 cm by 1 cm and was used to calculate the heat flux. Since the experiment’s goal is the investigation of the capillary limit, the wick was exposed to the air in order to create an open loop system. This allows direct and precise investigation of the thermal capillary limit without the coupled vapor saturation and circulation effects associated with a closed system. An experimental data point was established by taking the mean of a set of measurements taken over a 15-minute interval with a sampling rate of two or four seconds depending on the instrument’s capabilities. The uncertainties for the data point were associated with the Gaussian distribution of the measurement sets and the inherent instrument uncertainties.

The uncompensated heat flux was calculated by dividing the heat cartridges power by the surface area of the exposed wick in contact with the copper block. To monitor the applied power, a wattage transducer was used to convert the watts going to the heating element into a 0-10V DC voltage logged by LabVIEW as shown in Figure 7 approximately every two seconds. However, at lower heat flux trials (as is the case with nanowicks), the error associated with the wattage transducer was too high and a DC power supply was used instead of a variable autotransformer. The current was measured with the use of a 5Ω shunt resistor and the voltage drop across the heaters was measured and logged via LabVIEW. However, the DC power supply cannot provide more than 2.92 W without endangering the data acquisition cards and therefore could not be used for higher heat fluxes, in which case the variable autotransformer was employed.

The working fluid is provided by a reservoir that is resting on a precision analytical balance. Since the water is wicked upwards and evaporated away from the reservoir as shown in
Natural Convection Losses

Despite the use of insulation around the copper block, some of the cartridge heaters power is dissipated into the surrounding environment by natural convection. These were assessed and quantified with the use of conduction models coupled with thermocouple temperature measurements on both sides of the insulation.

Since the insulation is thick relative to the dimensions of the system, a 1D conduction approximation would not be valid. There would be 2D conduction through the edges and 3D conduction through the corners which complicates the matter [19]. In order to simplify the problem, conduction shape factors were used to calculate the heat loss through the insulation. Since this method requires 27 calculations per test run, matrix computations were used to expedite the analysis.

The insulation heat loss matrix calculations required temperature measurements at the surface of the insulation and between the copper block and insulation on all surfaces. After the temperatures of the system reached steady state, the data was logged via LabVIEW and analyzed as matrices by an auxiliary computational program with respect to the dimensions of the experimental setup.

Wick Conduction Losses

In addition to natural convection losses, there is heat conduction loss through the wick itself. Although it is an inherent benefit in a functional heat pipes, the goal of this research is to investigate thermal capillary flow. Because of this, the heat loss by the wicking structure causes inaccuracies when comparing capillary flow rates and heat fluxes.

There are two simple approaches that can be used to calculate the conduction losses by the wick; one is by considering the wick as a rectangular fin in two stagnant fluids...
and the other is through a conduction analysis by assuming the heat loss is primarily into the water (i.e., neglecting fin convection losses in the portion of the wick exposed to air). However, both require knowledge of the temperatures along the wick itself. Since the wick is often delicate, the only method to measure the temperature on the wick is through the use of an infrared camera.

Figure 8 shows an example of the temperature imaging along a wick obtained from the IR camera. The section of the wick that was analyzed for conduction losses is outlined in red dashed lines. The top of the analyzed section is where the wick meets the base of the copper block and the temperature is typically around 100°C. The water and air interface is at the bottom of the red dashed box and is typically around 50°C. If the wick is considered a rectangular fin in two fluids, the two aforementioned temperatures would be needed for calculations. The convective losses can be calculated by the expression

\[ q_{\text{convective loss}} = h_{\text{air}} A_{\text{surf},1} \left( \frac{T_{\text{base}} + T_{\text{mid}}}{2} - T_{\text{air}} \right) + h_{\text{water}} A_{\text{surf},2} \left( \frac{T_{\text{mid}} + T_{\text{water}}}{2} - T_{\text{water}} \right) \] (15)

where \( T_{\text{base}}, T_{\text{mid}} \) and \( T_{\text{water}} \) are the temperatures of the base of the fin, the interface of the two fluids and the water respectively. The tip of the fin is assumed to be the same temperature as the water which is measured by a thermocouple.

The aforementioned convective fin approach is the more accurate methodology since it accounts for convective heat transfer to both fluids in contact with the wick. However, since the heat transfer coefficients for air and water can range drastically from 10-100 W/m²K for air and 500-10,000 W/m²K for water, the error associated with using this method can pose significant problems considering phase change and saturated vapor is typically involved in the experiment [21].

The alternative approach is to calculate the conduction through the upper part of the wick under the assumptions that the air is insulating and the heat loss by the wick can be modeled by 1D conduction into the water reservoir. As Figure 9 shows, this assumption holds true since the temperature profile across the length of the wick exposed to the air is extremely linear. The linear trend line fit to the plot has a coefficient of determination very close to 1, thus justifying the aforementioned conduction assumptions. With the slope generated from the linear fit, the conduction loss through the wick can be calculated from

\[ q_{\text{conduction loss}} = k A \frac{dT}{dx} \] (16)

where the \( k \) is the thermal conductivity of the wick taken from various published sources for their respective temperatures [19, 22]. Because of the lower uncertainty associated with the thermal conductivity relative to the uncertainties of the heat transfer coefficients of the two fluids, this was the method that was chosen to account for wick conduction losses. The cross sectional area of the wick can be obtained from a scanning electron microscope (SEM) and \( dT/dx \) is directly extracted from the linear fit to the IR temperature data.

From the heat losses, the total heat flux that is dissipated by the wick is found by the equation

\[ q_{\text{actual}} = q_{\text{measured}} - q_{\text{insulation}} - q_{\text{wick}} \] (17)

where \( q_{\text{actual}} \) is the total heat dissipated by the wick, \( q_{\text{measured}} \) is the power that was measured to the heaters, \( q_{\text{insulation}} \) is the calculated heat loss through the insulation and \( q_{\text{wick}} \) is the heat loss via conduction through the wick. The heat dissipated by the wick can be found by dividing the power by the surface area and is given by the equation:

\[ \phi_q = \frac{q_{\text{measured}} - q_{\text{insulation}} - q_{\text{wick}}}{A_{\text{surf},\text{wick}}} \] (18)

where \( \phi_q \) is the heat flux through the wick and \( A_{\text{surf},\text{wick}} \) is the surface area of the wick exposed to the heaters. The heat flux would be used for performance comparisons between the examined wicks.
RESULTS

Capillary Limits (Dry-Out/Burn-Out)

Tests were conducted in order to assess the capillary limit of different wicks. In order to create a baseline for the nanowicks different samples were tested ranging from non-heat pipe wicking materials to wicks found in commercially available heat pipes. The experimental data acquired was compared against theoretical predictions and error analysis was done via sequential perturbation [23]. The critical heat flux was found by increasing the heat flux slowly with the variable autotransformer or DC power supply. Before the onset of dry-out/burn-out, the experimental mass flow rate versus heat flux data should follow a linear trend, as per equation (9). When the critical heat flux is achieved, the mass flow rate is no longer linear with heat flux, and depending on specific dry-out/burn-out conditions it might plateau or decrease altogether. The plateau or point of linear deviation was compared against the theoretically predicted capillary limit for a homogeneous pillar array wick, as per equations (7), (8) and (10).

Although the aforementioned capillary flow and permeability models are mainly applicable to nanopillar arrays such as CNTs and nanowires mat wicks, they might provide good order of magnitude estimates for wicks with other nano and micro-pore architectures. This might hold particularly true for wicks composed of cylindrical fibers even if they are randomly oriented, as is the case for the wool felt. Thus, equation (14) was used to compute the permeability of all the wicks samples by choosing appropriate “effective” pillar diameter and spacing values. For these purposes, a SEM was used. Numerous measurements were taken via ImageJ and the fiber diameter and spacing at various points along the wick was measured. Figure 10 shows an SEM picture for the wool felt sample. Although the wick is not composed of perfectly straight standing pillars of uniform size and spacing, it will be shown that when using an statistical average for the effective fiber diameter and spacing, the experimental results agree well with the formulations presented in this paper.

Since the maximum mass flow rate is dependent on measured dimensions via SEM, there are uncertainties associated with the calculation. From the measurement samples recorded, a distribution of dimensions was created. For most distributions, approximately five percent of values fall outside of two standard deviations [24]. As a result, the uncertainties for measured dimensions were assumed to be two standard deviations. The characteristic structural dimensions of the wicks and their standard deviations are shown in Table 1.

A general-purpose piece of wool felt was used as a baseline material due to its considerable wicking abilities. The wool felt data is plotted in Figure 11. The solid orange line is the theoretical mass flow rate as a function of total heat flux imposed on the sample, as per equation (9), and the gold dashed line is the theoretical maximum mass flow rate that the wick can sustain according to the nanopillar array model, given by equations (1), (7) and (8). The two gray dashed lines in Figure 11 are the resulting maximum mass flow rates associated with the uncertainty from the SEM measurements.

<table>
<thead>
<tr>
<th>Table 1. Measured Dimensions of Wicks</th>
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<tr>
<td>Wool Felt</td>
</tr>
<tr>
<td>Sintered Powder</td>
</tr>
<tr>
<td>Mesh Wick</td>
</tr>
<tr>
<td>Si Nanowires</td>
</tr>
</tbody>
</table>

Mean values of spacing ($s$) and pillar diameter ($d$) with uncertainties calculated from a distribution of samples at various points along the wick.

Figure 10. SEM picture of a wool felt wick and measurements of spacing ($s$) and pillar diameter ($d$)

Figure 11. Mass flow rate as a function of heat flux for a wick made of wool felt
Figure 12. Mass flow rate as a function of heat flux for a commercial sintered powder wick

It can be seen from Figure 11 that the experimental data points follow the theoretical mass flow rate line and dry-out occurs very close to the capillary limit model’s predicted maximum mass flow rate. The drop in mass flow can be associated with a vapor film forming and disrupting the flow. The mass flow rate of the wool felt reached 77.44 g/hr at its peak while the capillary limit models predicts a maximum mass flow rate of 66.09 g/hr.

Tests were also conducted on commercially available heat pipe wicks. A sintered powder wick was found to have a peak mass flow rate around 35.09 g/hr as shown by Figure 12. Similar to the wool felt, the sintered powder wick shows an upward trend following the theoretical mass flow predicted by equation (9) until it approaches the predicted maximum mass flow. While the sintered powder wick did not reach the maximum mass flow predicted by the average dimensions, dry out did occur within the range of maximum mass flow predicted by uncertainty values. This can potentially be associated with the fact that a sintered powder wick more closely resembles porous structure composed of spheres; the model nonetheless provided a good approximation of the expected maximum mass flow. It is also important to note that this data was not corrected for wick conduction losses and as a result is shifted away from the theoretical line.

The other commercially available wick was a mesh wick supplied by the same company. Although only 207 microns thick, the mesh wick still produced a peak flow rate of 35.28 g/hr as shown in Figure 13 comparable to the thicker wool felt and powder wicks (1.74 mm and 1.20 mm thick respectively). The mesh wick followed both theoretical predictions within the uncertainties presented. The structure of a mesh wick is inherently relatively uniform when compared to other wicks and resembled a lateral pillar composition as shown in Figure 14. As a result, the mesh wick agreed much better with the predicted maximum mass flow rate. However, the mesh wick experienced oxidation over time as shown in Figure 15 and the
Table 2. Mass flow rates for the various wicks

<table>
<thead>
<tr>
<th>Wick Type</th>
<th>( \dot{m} ) [g/hr]</th>
<th>( \delta \dot{m} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wool Felt</td>
<td>77.44</td>
<td>5.15</td>
</tr>
<tr>
<td>Sintered Powder</td>
<td>35.09</td>
<td>0.41</td>
</tr>
<tr>
<td>Mesh Wick</td>
<td>35.28</td>
<td>0.54</td>
</tr>
<tr>
<td>Si Nanowires</td>
<td>0.32</td>
<td>0.28</td>
</tr>
</tbody>
</table>

Wool felt provided the highest mass flow due to its thickness while silicon nanowires produced a mass flow rate that is magnitudes lower.

regularity of the structure dimensions became less consistent. Since the purpose of this study is to analyze the effects of wick dimensions and geometry on capillary forces, the data obtained from the oxidized wick was used in this paper and the effects of oxidation on the characteristic structure dimensions were accounted for in the theoretical model.

A silicon nanowire wick was tested with the same methodology and a similar setup with a DC power supply. The change in power supply was necessary due to the uncertainties associated with low wattage readings using AC power. The nanowires wick was found to have a peak mass flow of 0.31 g/hr and like the previous plots followed the predicted maximum mass flow from the homogenous model. Figure 16 shows the data against the theoretical mass flow rate and capillary limit model predictions. As the figure shows, the sustainable heat fluxes and the capillary limit for the nanowick are an order of magnitude smaller than those for the other wicks. As Table 2 shows, the mass flow rate of the silicon nanowick is 1% of the next lowest mass flow rate. This is a consequence of the thinness of the nanowick as compared to the other wicks. This translates into a much smaller cross sectional area and therefore a much smaller mass flow rate, hence reducing the heat removal capability of the nanowick. A consequence of these reduced operating conditions is that the error due to evaporation at lower mass flow rates is too large for accurate measurements and must be addressed in the future by reducing the surface area, and hence evaporation, of the reservoir. Due to the uncertainty involved, conclusions cannot be drawn from the plot. However, the experimental data does show a trend similar to the values predicted.

The thinness of the nanowick also prevents the fluid from reaching its boiling point and poses another problem. Because of the film thickness, the nanowick cannot provide enough liquid mass flow rate to maintain a sustained phase change condition at the heater testing section. As a result, the wick dries out at low heat fluxes and the surface temperature of the wick never exceeded 25°C before the mass flow rate begins to deviate from the theoretical mass flow rate. Figure 17 shows the surface temperature of the silicon nanowick as a function of the total heat flux. As Figure 17 shows, dry-out occurs well before the fluid approaches the boiling point of 100°C. Although it is possible for the surface temperature to reach 100°C by increasing the heat flux, the wick would already be dried and the heat transfer mechanism would be purely due to conduction in the substrate and convection into the air. This poses a problem for microelectronics cooling where computer chips temperatures can exceed 80°C [2].

The four data sets are plotted together in Figure 18 for comparison. As mentioned before, the nanowires have a much lower critical heat flux and corresponding mass flow rate than the baseline tests. Notice how on direct comparison the nanowire data has nearly disappeared. The rather premature critical heat flux of the nanowick is associated with its thickness. Since the nanowire wick is significantly thinner than the baseline wicks, the cross sectional area for the fluid flow is consequently smaller. In order to provide a better comparison between the various wicks, the data must be normalized with respect to its thickness.

Figure 16. Mass flow rate as a function of heat flux for an array of silicon nanowires

Figure 17. Mass flow and surface temperature of a Si nanowick with respect to power dissipation. When the nanowick dries out, the average surface temperature is not near the boiling temperature.
Fluid Velocity Comparisons

In order to properly compare the potential efficacy and benefits of the nanowick against the baseline wicks, the data from the four tests was normalized to account for the effects of different cross-sectional areas. The mass flow rate was first converted to volumetric flow rate by assuming a constant fluid density of 1000 kg/m³. The volumetric flow rate was then divided by the cross-sectional area perpendicular to the fluid flow to yield a velocity. Since the width of the wick was uniform between all four wicks, the only effect of this was from the thickness of the wick. This produced values of the actual velocity of the fluid at various heat fluxes.

Since the thickness of the wick also played an essential role on the critical heat flux, the heat flux was normalized by the following equation

\[ V_{\text{theo}} = \frac{q}{\rho A_e (h_{fg} + c_p \Delta T)} \]  

(19)

where \( q \) is the power dissipated, \( \rho \) is the fluid density, \( A_e \) is the cross-sectional area perpendicular to the fluid flow, \( h_{fg} \) is the heat of vaporization of the fluid, \( c_p \) is the specific heat capacity of the fluid and \( \Delta T \) is the difference of temperature between the measured fluid temperature and the boiling point of the fluid. The equation is essentially the power dissipated divided by the enthalpy change of the fluid. The equation results in a value with the units of mm/s similar to the previous normalization. These values of velocity are the theoretical velocities required by the heat flux. These values were compared to the actual velocities acquired during the experiment in Figure 20.

Table 3. Maximum velocities achieved by the wicks

<table>
<thead>
<tr>
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<th>( V_{\text{max}} ) [mm/s]</th>
<th>( \delta V_{\text{max}} )</th>
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<tbody>
<tr>
<td>Wool Felt</td>
<td>1.40</td>
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<tr>
<td>Sintered Powder</td>
<td>0.92</td>
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</tr>
<tr>
<td>Mesh Wick</td>
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<td>0.57</td>
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<tr>
<td>Si Nanowires</td>
<td>2.83</td>
<td>5.06</td>
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</table>

The commercial mesh wick produced the highest velocity of the wicks.

Figure 18. Mass flow rate as a function of heat flux for all wicks

Figure 19. Normalized plot comparing all wick data sets

The plot shows that the silicon nanowire wick performs as well as commercial heat pipe wicks when the data is normalized. The orange line in the plot defines the ideal line the points should fall on if the experiment was perfectly insulated and there was no error involved. All four data sets follow the 1:1 line until they reach their critical heat flux and their velocities no longer increase. The maximum velocities of the four wicks are tabulated in Table 3.

Silicon nanowires produced higher velocities than the wool felt and powder wick but did not match the velocity of the mesh wick. However, from the uncertainties presented, it cannot be for certain that the nanowires produced a lower velocity.

Capillary Limit Model Validation

To compare the validity of the capillary limit model between the wicks, the actual maximum velocity was plotted against the predicted maximum velocities. Figure 20 shows the four wicks plotted with a 1:1 line. If the model was completely accurate, the point would fall directly on top of the line, if the theoretical model under predicted the maximum velocity, the point would fall above the line and an overestimated value would fall below the line.

From the plot, the model seems to provide an accurate model for the nanowick and a respectable simulation for the mesh wick. Although the model was less accurate for the powder and wool felt, it still provided a good estimation of the
Figure 20. Actual maximum velocities plotted against the theoretical maximum velocity predicted by the homogenous wick model.

maximum velocities capable by the wicks and their values are well within error.

In Table 4 the percent difference between the two maximum velocities are given. The model generated a prediction that was off by 1.66% for the silicon nanowires. From these values, it seems as though the model works fairly well with the silicon nanowires wick despite not being composed of freestanding pillars. Since the mesh wick’s structure was found to be fairly uniform, it was expected that the mesh wick would also be robustly predicted by the model.

This speculation was confirmed with a 3.28% difference between the actual and predicted values for the mesh wick. A surprising result from the model predictions comes from the sintered powder wick. Although the sintered powder wick more closely resembled a porous structure of spheres, the model was still able to predict the maximum velocity to a reasonable value.

CONCLUSIONS

A model for capillary flow within a nanopillar array was developed and used in conjunction with a heat transfer model to predict the capillary limit of different wicks. The model was validated against experiments performed with different types of wick materials: a wool felt, a mesh wick, a powder wick and a silicon nanowire based wick structure (nanowick). Good agreement between the experimental data and the model was found.

The nanowicks achieved lower mass flow rates than conventional wicks, but provided higher or similar velocities within the wick itself once the data was normalized. Although nanowicks performed worse than conventional wicks in some cases, their low performance can be attributed to the inherent thinness of the nanowicks. This is an important parameter to address in the future in order for nanowicks to be considered as a reasonable alternative for heat pipes. In the case of the silicon nanowick, the subsequent film thickness prohibited the fluid from ever achieving boiling temperatures. In order for micro and nanoscale heat pipes to be a feasible option, addressing potential hydrophobicity and growth limits of the structures is required.

The capillary limit model presented in the paper provided reasonable maximum mass flow rates despite the different wick geometry and structures analyzed. Although the model may not be completely applicable for some wicks, it provides a rational first order prediction of flow rate limits and would provide insight for wick design.

This study focused on the capillary flow of wick structures; however, it is important to note that to maximize performance of a heat pipe, a highly conductive material should be used for the structure (i.e. copper). This would be an important parameter to address in future studies.

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### THERMOCOUPLE LOCATION LIST

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