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Thermal design optimization of evaporator micropillar wicks

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ABSTRACT

Heat pipes and vapor chambers act as efficient heat spreaders since they rely on phase change of the coolant. The evaporator design is critical and typically high performance is characterized by the high heat dissipation capability with low thermal resistance. In the past, there have been numerous experimental and modeling studies focused on the design of evaporator wicks of different geometries, but systematic studies to simultaneously optimize both the heat flux and thermal resistance have been limited. In this work, we developed a comprehensive model that considers both aspects to provide design guidelines for evaporator micropillar wicks. We show that capillary limited heat dissipation is best captured with a recently developed numerical model as compared to previous analytical models. We also developed a numerical model to obtain the effective wick thermal conductivity, which is a function of pillar diameter, pitch, and height. Smaller diameters with smaller pitches of the pillars had more thin film area and had larger effective wick thermal conductivities. Our parametric investigations show that trade-offs between lowest thermal resistance and maximum heat carrying load exists, and the actual wick geometry will be dictated by application specific requirements. Finally, we highlight the importance of accurately obtaining the accommodation coefficients to predict the effective wick thermal conductivity. The present work would enable in optimal design of micropillar wicks (with low thermal resistance and high dry-out heat flux) and the same methodology can be extended to other types of wick structures as well.

1. Introduction

Next generation high performance electronics devices have severe thermal management challenges associated with the high heat dissipation and low temperature rise requirements. For example, smartphones or tablets are becoming more functional, yet they are turning smaller in size. They typically need to dissipate 1–10 W/cm^2 and still need to maintain the mobile skin temperature within 40–45 °C [1]. Vapor chambers and heat pipes are promising solutions, whereby they can efficiently spread the heat via capillarity and liquid-vapor phase change heat transfer. Heat pipes are widely used in laptops for transferring heat from the processor to the condenser/fan unit. The main objective in designing a vapor chamber or heat pipe is to be able to spread as much of heat as possible without allowing liquid dry-out. Simultaneously the temperature drop across the heat spreader (heat pipe/vapor chamber) has to be kept low to maintain the device within operating temperatures. The evaporator wick design plays a crucial role in the overall performance of the heat spreader, both in terms of maximum heat carrying capacity and thermal resistance of the spreader.

To achieve high performance vapor chambers and heat pipes, the evaporator wick needs to be carefully designed. Therefore, many modeling papers have focused on maximizing the capillary pressure for various wick structures to achieve high heat fluxes. However, many simplifications and assumptions on the evaporation processes have been made.

Evaporators relying on capillary wicking force for coolant circulation use either sintered particles or other micromachined (channels/pillars) etc. as the wick structure to provide the capillary force required for coolant circulation. Though the work described here can be applied to all types of wicks in principle, the discussion will be based on cylindrical micro-pillars as examples.

Capillary dry-out heat flux prediction has been widely made using Darcy's law [2–4]. This method is accurate for heat pipes which are long and have only a small evaporator section at the end. However for cases where evaporator size is larger (relative to the total length of flow) or where there is continuous evaporation as in the case of vapor chamber the actual local mass flow at each point in the wick is a variable and a modified form of Darcy's law or Brinkman's law needs to be used to

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describe the physics better. Recently there has been some reports which use Brinkman's law to describe the fluid flow in an evaporator wick [5]. A recent report [6] on an improved numerical model estimated the dry-out heat flux accurately by including the effect of actual local meniscus shape. Since the permeability and capillary pressure were locally calculated based on the actual meniscus shape the results predicted from this numerical model is expected to be more accurate than other analytical models.

There have been studies focusing on thin film region to predict corresponding thermal resistance [7,8]. A report of a numerical study of different types of wick geometries by observing the available thin film area and capillary pressure in each case was made [7]. A detailed model to understand the thin film evaporation mechanism in wick structures has also been provided [8].

Though there have been some efforts in the past [3,4] to optimize the wick geometry to maximize the heat flux both experimentally and numerically there are not many reports which attempted to simultaneously optimize the heat flux and thermal resistance. So, in summary, there have been few different models to calculate the dry-out heat flux in evaporators and there are few models to describe the thin film evaporation dynamics. Capillary dry-out heat flux is generally considered in the macro design of a system and thermal resistance arising from thin film evaporation area is a local design aspect. However since both these factors are governed by wick geometry, methods to optimize the geometric sizes for a given wick type, to simultaneously maximize the dry-out heat flux and to minimize the thermal resistance is required.

In this paper, we first review the existing models and compared the results of different models with the recently reported numerical model [6]. The overall thermal performance of a device does not exclusively depend on the capillarity limited heat carrying capacity but also on the thermal conductivity of the wick structure which will govern the device temperature for a given heat load. So, a detailed methodology is presented to accurately estimate the thermal conductivity of the wick. Finally, we show that wick geometry that has very high capillary limited heat flux also has a high thermal resistance. So an optimal geometry that balances both must be chosen for the best device thermal performance.

2. Heat flux models

There have been numerous models in the literature to determine the capillary-limited -dry-out heat flux of an evaporator wick. We first review some of these models and compare the solutions to a recently developed numerical solution that shows good agreement with experiments [6]. The capillary limited heat flux of a wick is dependent on the permeability, maximum capillary pressure and fluid properties. Permeability and capillary pressure are in turn functions of d, h and l (pillar diameter, height and pitch respectively). Fig. 1 shows a typical evaporator wick with liquid flowing from the reservoir into the wick by

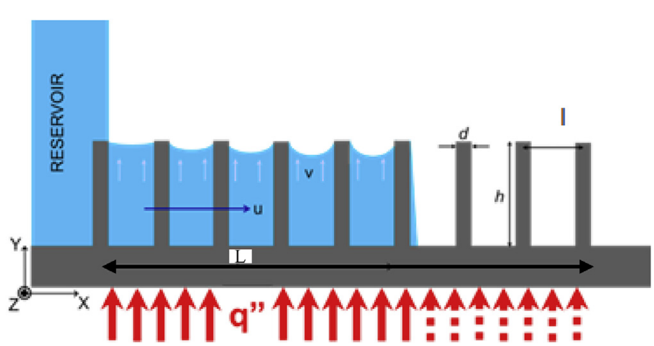

Fig. 1. Schematic showing capillary limited dry-out in an evaporator wick of length L, with micropillars of diameter d, height h and pitch l, with a surface heat flux q”.

capillary pressure. The heat transferred from the base of the evaporator enables thin film evaporation in the meniscus thin film region. The wick propagation length is L (distance between the reservoir to end of evaporator (or the dry-out point)) and the wick has permeability K and maximum capillary pressure (P_{cap}).

2.1. Darcy's law with a constant velocity

Darcy's law relates the superficial flow velocity with the pressure drop for a flow through a porous medium. But usually, there is no accompanying mass loss due to evaporation. Heat pipe based literature widely uses Darcy‘s equation based on average velocity over the entire length of evaporator [2–4,9]. A constant maximum superficial velocity, u is assumed without accounting for the local evaporation effect.

$$u = \frac{K}{\mu} * \frac{dp}{dx} = \frac{K}{\mu} * \frac{P_{cap}}{L} \tag{1}$$

Based on energy balance the superficial velocity, u or mass flow rate, $\dot{m}$ can be related to the heat load Q by the following equation.

$$u = \frac{\dot{m}}{\rho A_{wick}} = \frac{Q}{h_{fg}\rho A_{wick}} \tag{2}$$

So the final equation relating maximum dry-out heat load with capillary pressure, permeability and other geometric parameters of the wick becomes

$$\frac{K}{\mu} * P_{cap} * \frac{h_{fg}\rho A_{wick}}{L} = Q \tag{3}$$

In the above derivation, the superficial velocity u is defined based on a constant liquid mass flow rate, $\dot{m}$. For long heat pipes with ratio of evaporator length to the total flow length being small, this may be useful, but for scenarios like vapor chamber where this ratio is high, this assumption of using a constant flow rate over predicts the viscous losses and reduces the capillary limited heat flux.

2.2. Darcy's law with actual local velocity

Here we attempt to modify the above equation with the superficial velocity as a function of x-coordinate to accommodate mass lost due to evaporation in the flow path.

The local mass flow rate, $\dot{m}$ is related to the maximum flow rate $\dot{m}_o$ ([$q(L*w)/h_{fg}$] which is near the condenser or reservoir) by the amount of liquid mass evaporated, by energy balance as

$$\dot{m} = \dot{m}_o - q(x*w)/h_{fg} \tag{4}$$

Substituting (4) in (1) and (2) leads to the following expression

$$\frac{K}{\mu} * dp = \frac{(q*w)(L-x)}{h_{fg}\rho A_{wick}} dx \tag{5}$$

Integrating the above equation and rearranging leads to the following relation for dry-out heat load

$$\frac{K}{\mu} * P_{cap} * \frac{h_{fg}\rho A_{wick}}{L} * 2 = Q \tag{6}$$

Accounting for a uniform mass loss across the propagation length results in an increase of capillary limited heat flow by a factor of 2.

2.3. Brinkman's equation

Brinkman's equation is modified form of Darcy's equation with an extra term added to account for viscous losses from wall shear stress [5].

$$\frac{\partial^2 u}{\partial y^2} = \frac{\varepsilon}{\mu}\frac{\partial p}{\partial x} + \frac{\varepsilon u}{K} \tag{7}$$

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