International Journal of Heat and Mass Transfer 61 (2013) 409–418

Contents lists available at SciVerse ScienceDirect

International Journal of Heat and Mass Transfer

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

Vapor chambers with jumping-drop liquid return from superhydrophobic condensers

Jonathan B. Boreyko, Chuan-Hua Chen *

Department of Mechanical Engineering and Materials Science, Duke University, Durham, NC 27708, USA

ARTICLE INFO

Article history:
Received 19 October 2012
Received in revised form 16 January 2013
Accepted 28 January 2013
Available online 6 March 2013

Keywords:
Jumping drops
Vapor chamber
Superhydrophobicity
Wick
Interfacial resistance
Non-condensable gases

ABSTRACT

In closed-loop phase-change systems, self-propelled jumping drops on a superhydrophobic condenser offer a new mechanism to return the working fluid to the evaporator, eliminating the requirement for either external forces or wick structures along the return path. Here, we report the heat transfer performance of a jumping-drop vapor chamber consisting of two parallel plates, a superhydrophobic condenser and a superhydrophilic evaporator. With proper removal of the non-condensable gases including those dissolved in the working fluid, the overall thermal resistance of the jumping-drop chamber is primarily governed by the conduction resistance of the wicked evaporator in series with the interfacial resistances at the phase-change interfaces. The lumped resistance model was verified by systematically varying the parameters of the chamber such as wick thickness, vapor spacing, vapor temperature, and heat flux. As an alternative mechanism to transport the working fluid, the jumping drops can enable novel phase-change heat transfer systems such as planar thermal diodes and heat spreaders, for which this work provides practical design guidelines.

1. Introduction

Vapor chambers offer passive yet effective phase-change heat transfer in a planar form factor useful for microelectronics cooling [1–5]. Conventional vapor chambers are essentially flat-plate heat pipes [1–3], relying on wicked or grooved walls to return the working fluid from the condenser back to the evaporator by capillarity. For the capillary return to be effective, porous wick structures or fine grooves are required along the entire return path which poses design constraints and manufacturing complexities; see for example [6–9]. Alternative mechanisms such as gravitational and electrohydrodynamic return have their own limitations, rendering the heat transfer devices dependent on orientation or active power input [3]. Planar heat spreaders can also be obtained by arranging one-dimensional pipes such as thermosyphons and pulsating heat pipes into a plate [1,2], but such arrangements are less versatile because of the inherent one-dimensional tubular design, and are not effective for certain applications such as planar thermal diodes [10].

In our previous work [10], a novel jumping-drop mechanism for condensate return was utilized to enable an inherently planar vapor chamber, circumventing the drawbacks of gravity- or capillarity-driven return mechanisms. Rather than lining the entire vapor chamber with a wick structure, a wickless superhydrophobic plate was placed parallel to a wicked superhydrophilic plate, such that water condensing on the superhydrophobic surface would spontaneously jump across the gap directly back to the wicked evaporator (Fig. 1). The kinetic energy required for the jumping-drop phenomenon is harvested from the surface energy released upon coalescence [11–17], and the out-of-plane directionality is a result of the superhydrophobic surface breaking the symmetry of energy release [12,18].

Our jumping-drop chamber builds upon conventional vapor chambers with a few notable differences owing to the new liquid return mechanism. (i) The superhydrophilic evaporator design is directly borrowed from wicked vapor chambers, with the same capillary redistribution mechanism of the working fluid within the wick structures; however, the mass flow rate is no longer dictated by the cross sectional area of the wick (i.e. proportional to the thickness of the wick), but by the frequency and size distribution of the jumping condensate drops. The jumping return therefore circumvents the capillary wicking limit, which is typically the most significant limit of moderate temperature heat pipes and vapor chambers [1]. (ii) The perpendicular jumping return path between the condenser and the evaporator resembles the directionality of gravity-driven return in themosyphons; however, the jumping drops are much smaller and faster and therefore independent of gravitational orientation [12]. The out-of-plane jumping return is

* Corresponding author. Tel.: +1 919 660 5343; fax: +1 919 660 8963.
E-mail address: chuanhua.chen@duke.edu (C.-H. Chen).

0017-9310/$ - see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.01.077

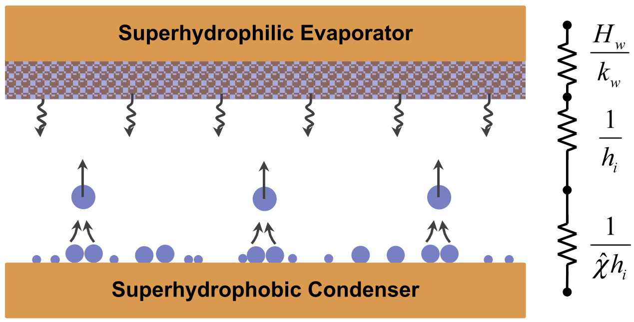

Fig. 1. Schematic of the jumping-drop vapor chamber consisting of a superhydrophobic evaporator and a superhydrophilic condenser in parallel. Heat is removed by evaporation of liquid water contained in the wicked evaporator, and rejected at the condenser by condensation of the water vapor. The working fluid is returned back to the evaporator by the self-propelled jumping phenomenon, where condensate drops spontaneously jump out-of-plane upon drop coalescence on the superhydrophobic surface. The major thermal resistances across the vapor chamber include the conduction resistance of the wick and the interfacial resistances across the evaporating and condensing interfaces.

scalable and particularly suitable for planar systems, unlike conventional vapor chambers with capillary return along wicked walls, where longer liquid return paths are expected for devices of larger areas. (iii) The jumping-drop vapor chamber consisting of condensers and evaporators of asymmetric wettability is a thermal diode by design [10], in contrast to conventional wicked vapor chambers with symmetric (interchangeable) condenser and evaporator.

Although the jumping-drop vapor chamber is essentially the forward mode of the jumping-drop thermal diode we previously reported [10], the work reported here represents a more systematic study of the heat transfer performance of vapor chambers enabled by the jumping-drop return mechanism. In particular, we carefully minimized the effects of non-condensable gases, and investigated the heat transfer coefficients while varying system parameters such as wick thickness, vapor spacing, vapor temperature, and heat flux. In this mechanistic study, we focused on the unique aspects of our jumping-drop vapor chamber. The inherently planar design justifies the simplification of the system as serial resistances due to wick conduction and interfacial phase change. The capability of the jumping drops to travel against gravity and/or vapor flow presents new operating limits to our vapor chamber.

2. Theoretical model

As schematically shown in Fig. 1, the jumping-drop vapor chamber consists of two parallel plates enclosing water in both liquid and vapor phases. The working fluid is circulated as follows: the liquid water evaporates from the wick structure embedded on the superhydrophilic evaporator, the water vapor condenses on the opposing superhydrophobic condenser, and the condensate jumps back to the evaporator upon drop coalescence. The parallel-plate configuration justifies the following one-dimensional model for the effective heat transfer coefficient and the operational limits specific to the jumping return.

2.1. Overall heat transfer coefficient

For phase-change heat transfer, non-condensable gases can present a significant thermal resistance due to their accumulation at the phase-change interface [19–21]. In the subsequent analysis, we assume negligible presence of non-condensable gases and ignore the associated gas diffusion resistance, in which case the interfacial resistance will predominantly govern the heat transfer at the phase-change interface. Kinetic theory yields an interfacial heat transfer coefficient [21],

$$h_i \approx \frac{2\hat{\alpha}}{2-\hat{\alpha}} \frac{\rho_v h_{lv}^2}{T_v} \sqrt{\frac{\bar{M}}{2\pi \bar{R} T_v}}, \qquad (1)$$

where $\hat{\alpha}$ is the accommodation coefficient, ρ_v, h_{lv}, and $\overline{M}$ are the density, latent heat, and molecular weight of the saturated water vapor at a temperature T_v, and $\overline{R}$ is the universal gas constant. The accommodation coefficient is the fraction of molecular collisions that result in actual phase change across the interface, and is assumed to be equal for condensation and evaporation [21]. The accommodation coefficient ($\hat{\alpha}$) for water varies dramatically in different systems, and is reported to be typically below 0.1 for stagnant interfaces and above 0.1 for dynamically renewing interfaces [22]. Because of this large variation, the value of the accommodation coefficient is typically obtained by fitting the data measured on a particular phase-change system [22]. For a given $\hat{\alpha}$, the interfacial heat transfer coefficient (h_i) is approximately an exponential function of the vapor temperature.

For purely evaporative cooling on the superhydrophilic evaporator, the phase-change process is expected to be governed by the interfacial heat transfer coefficient in Eq. (1). For dropwise condensation on the superhydrophobic condenser, the phase-change heat transfer can be additionally affected by the capillary resistance (important for nanometric drops) and conduction resistance (important for millimetric drops) [20,21]. However, since the average diameter of self-propelled condensate drops on superhydrophobic surfaces tends to be micrometric [12], the interfacial resistance is expected to dominate on the condenser as long as non-condensable gases are negligible.

As shown in Fig. 1, the overall heat transfer coefficient of the jumping-drop vapor chamber (h_o) is dependent on the conduction resistance across the wick which is almost independent of temperature, and the interfacial resistances across the evaporator and condenser which strongly depend on the vapor temperature,

$$\frac{1}{h_o} \approx \frac{H_w}{k_w} + \frac{1}{h_i} + \frac{1}{\hat{\chi} h_i} \equiv \frac{H_w}{k_w} + \frac{1}{h_{pc}}, \qquad (2)$$

where H_w is the thickness of the wick structure, k_w is the effective conductivity of the water-saturated wick, and $1/h_{pc}$ lumps together the phase-change resistances on both the evaporator and the condenser. The geometrical parameter $\hat{\chi}$ accounts for the partial coverage of dropwise condensate on the condenser surface and also the fact that the actual surface area of a drop open to additional condensation is larger than its projected area on the condenser. In this simplified model, we neglect the flow resistances for transporting the working fluid in both liquid and vapor phases, on the ground that the cross-sectional area for the fluid transport (the plate area) is very large and the distance of transport (the inter-plate spacing) is very short.

For our system, the geometrical parameter $\hat{\chi}$ is expected to be approximately constant and close to unity. The steady-state surface coverage is reached after the condensate drops start to coalesce and spontaneously jump; The surface coverage is expected to be constant for self-propelled condensation, e.g. around 40% for tests in the open air [12]. The actual surface area open to additional condensation is related to the projected surface area on the condenser by the apparent contact angle, which can be assumed constant for condensate drops with micrometric average diameter; For a contact angle of 90°, a factor of 2X is expected for hemispherical drops [23]. These two effects combine to yield an approximately constant $\hat{\chi}$ on the order of 1.

Combining Eqs. (1) and (2), the lumped heat transfer coefficient due to phase change on both the evaporator and the condenser is given by,