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Simulations of saturated boiling heat transfer on bio-inspired two-phase heat sinks by a phase-change lattice Boltzmann method

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ABSTRACT

Pool boiling heat transfer from four types of micro-pillar heat sinks with different wettability patterns is simulated numerically with the latest version of liquid-vapor phase-change lattice Boltzmann model. Effects of pillar geometry and wettability on bubble dynamics are investigated. It is found that bubbles will nucleate either on the hydrophobic pillar top or on the hydrophilic cavity bottom between micro-pillars, depending on wettability and local wall temperature. Among the four types of micro-pillar heat sinks with hybrid wettability patterns, it is found that the bio-inspired heat sink (with hydrophobic pillar tops and hydrophilic base) has the best boiling heat transfer performance with the following desirable features: (i) unique characteristics of orderly separation of vapor and liquid paths at low superheats, (ii) hydrophobic surface characteristics where residual bubbles on hydrophobic pillar tops provide faster bubble departure frequency, (iii) triple phase lines are pinned at corners of micro-pillar, restricting expansion of bubbles into film boiling at high superheats. Simulated results show that geometry of micro-pillars and wettability patterns greatly influence transition boiling regime including the maximum heat flux (CHF) and the Leidenfrost temperature, resulting in pool boiling curves with widely different shapes.

1. Introduction

Boiling heat transfer, being one of the most effective modes of heat transfer, has a wide range of applications in energy conversion systems and thermal management devices [1]. In recent years, a great deal of interest has been given to fabrication of *rough* surfaces with microstructures [2–5] or *smooth* surfaces with mixed wettabilities [6–8] for pool boiling heat transfer enhancement. As early as in 2003, Wei and Honda [2] performed pool boiling experiments with FC-72 on micro-pin-fin structured chips, showing that micro-structured surfaces could enhance nucleate boiling heat transfer. Yu and Lu [3] investigated experimentally bubble dynamics and pool boiling heat transfer performance of copper rectangular fin array surfaces immersed in saturated FC-72, and found that the CHF on the test surface with optimal geometry parameters was five times higher than that of the plain surface. Chu et al. [4] performed pool boiling experiments on structured surfaces and found that structured surfaces enhanced CHF owing to roughness-amplified capillary effects. Dong et al. [5] conducted pool boiling of ethanol on micro-structured horizontal heated surfaces, and found that micro-structures enhanced nucleate boiling and critical heat flux significantly.

Bitz et al. [6] fabricated hydrophobic and hydrophilic regions on oxidized ultra-*smooth* silicon wafers, and found that the critical heat flux and boiling heat transfer were 65% and 100% higher than on a pure hydrophilic surface at the same superheat. Jo et al. [7] fabricated a mixed wettability *smooth* surface composed of hydrophobic dots on a hydrophilic surface, and concluded that the number of hydrophobic dots and the pitch distance between dots were critical parameters to influence pool boiling performance. Kumar et al. [8] conducted boiling experiments on mixed wettability *smooth* surfaces, showing improved boiling performance on these surfaces.

Most recently, many researchers conducted experiments on *rough* surfaces with hybrid wettability patterns to enhance pool boiling heat transfer. Suroto et al. [9] investigated effects of the edge (micro structures and length) of hydrophobic spots on nucleate pool boiling from a mixed-wettability surface. They concluded that boiling heat transfer at saturated and subcooled conditions was enhanced with the increased peripheral length of the hydrophobic-spot. Dai et al. [10] conducted boiling experiments on triangular hydrophilic cavities with mixed wettability tops and found better nucleate boiling performance than a bared surface. Jo et al. [11] conducted boiling experiments to investigate nucleate

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boiling heat transfer with controlled wetting patterns and micro-posts, and they found that boiling heat transfer was enhanced without critical heat flux deterioration. Kandlikar [12] proposed that separation of the vapor and liquid path would enhance boiling heat transfer, and they designed a fin structured surface for this purpose. They found eightfold increase in nucleate boiling heat transfer coefficient and two-and-half times higher CHF over a plain copper surface. Rahman et al. [13,14] fabricated a bi-conductive surface, comprising rows of low-conductivity epoxy embedded into high-conductivity copper substrates. Thus, a spatial ordering of the flow field was formed on the heat sink, resulting in fivefold improvements of boiling heat transfer coefficient and two times higher critical heat flux than that of a bared copper surface, respectively.

In recent years, 2D liquid-vapor phase-change lattice Boltzmann models have been developed successfully to study boiling heat transfer phenomena [15–18]. In particular, Gong and Cheng have used early versions of these phase-change LB models to study pool boiling from mixed wettability smooth surfaces [15] as well as mixed wettability micro-cavities surfaces with hydrophilic tops and hydrophobic bottom [16]. They found that although these mixed wettability rough heat sinks (with hydrophilic pillar tops and hydrophobic cavity bottom) had higher nucleate boiling heat transfer than rough hydrophilic surface, it did not have higher critical heat flux. Li et al. [17] also studied heat transfer characteristics from mixed wettability rough surfaces, and found that the mixed wettability heat surface could improve heat transfer during nucleate boiling regime. Most recently, Ma and Cheng [18,19] found that it is important to impose the conjugate boundary condition by Li et al. [20] to simulate boiling from heat sink surfaces when the liquid-solid interface temperature or heat flux is unknown. This latest version of liquid-vapor LB model was used to study film boiling by Gong and Cheng [21].

In this paper, we propose a two-phase micro-pillar heat sink with hydrophobic pillar tops on a hydrophilic base to achieve separate vapor and liquid paths which is different from previous studies [15,16,22]. This heat sink design is inspired by *O.microdasys* (from the Chihuaha Desert) [23] and desert beetle (from Namib Desert) [24], in which these insects can survive in highly arid environments. This is because interlaced wettability surface of both cactus and beetle's shell help collecting water from highly arid environments. The latest version of the liquid-vapor phase-change lattice Boltzmann model [18,19,21] is used to study effects of structure parameters of these bio-inspired micro-pillar heat sinks on pool boiling heat transfer under controlled wall temperature conditions numerically. Effects of wettability, pillar width, height, and pitch distance on bubble dynamics above these bio-inspired heat sinks are investigated, and boiling curves from these bio-inspired heat sinks are obtained numerically. It is shown that bio-inspired heat sinks have many desirable features to achieve not only higher nucleate boiling heat transfer but also higher critical heat flux as well.

2. Liquid-vapor phase-change lattice Boltzmann model

In this paper, 2D simulations of pool boiling on bio-inspired heat sinks under constant wall temperature conditions are carried out with the latest version of liquid-vapor phase-change LB model [18,19,21]. This liquid-vapor phase-change LB model will be briefly summarized below.

The evolution for density is the LBGK model, which is given by

$$f_i(\mathbf{x}+\mathbf{e}_i\delta_t, t+\delta_t) - f_i(\mathbf{x},t) = -\frac{1}{\tau}(f_i(\mathbf{x},t) - f_i^{(eq)}(\mathbf{x},t)) + \Delta f_i(\mathbf{x},t) \tag{1}$$

where $f_i(\mathbf{x}, t)$ is the particle distribution function at position $\mathbf{x}$ and time t, $\mathbf{e}_i$ is the particle velocity at the direction of i, τ is the relaxation time, and $f_i^{(eq)}(\mathbf{x}, t)$ is corresponding equilibrium distribution function which is given by

$$f_i^{(eq)} = \omega_i\rho\left[1 + \frac{\mathbf{e}_i\cdot\mathbf{u}}{c_s^2} + \frac{(\mathbf{e}_i\cdot\mathbf{u})^2}{2c_s^4} - \frac{\mathbf{u}^2}{2c_s^2}\right] \tag{2}$$

where c_s is the lattice sound speed, ω_i is the weighting coefficients ($\omega_i = 4/9$ for $i = 0$, $\omega_i = 1/9$ for $i = 1$–4 and $\omega_i = 1/36$ for $i = 5$–8).

The exact differential method (EDM) is used to incorporate the force term $\Delta f_i(\mathbf{x}, t)$, which is given by

$$\Delta f_i(\mathbf{x},t) = f_i^{(eq)}(\rho(\mathbf{x},t), \mathbf{u}+\Delta\mathbf{u}) - f_i^{(eq)}(\rho(\mathbf{x},t), \mathbf{u}) \tag{3}$$

where δ_t is the time step, $\Delta\mathbf{u} = \mathbf{F}\delta_t/\rho$ is the change of velocity due to the action of body force after the collision step, where $\mathbf{F}$ includes the fluid-solid interaction force $\mathbf{F}_s$, gravitational force $\mathbf{F}_g$ and the inter particle interaction force $\mathbf{F}_{int}$, the detailed calculation of these force can be found in [25]. It should be pointed here that the "effective mass" $\psi(\mathbf{x})$ used for calculating $\mathbf{F}_{int}$ is given by

$$\psi(\rho) = \sqrt{\frac{2(p-\rho c_s^2)}{c_0 g}} \tag{4}$$

with $c_0 = 6.0$. The Peng-Robinson (P-R) equation of state is used for the real gas, which is given by

$$p = \frac{\rho RT}{1-b\rho} - \frac{a\rho^2\phi_0(T)}{1+2b\rho-b^2\rho^2} \tag{5}$$

where

$$\phi_0(T) = \left[1 + (0.37464 + 1.54226\omega - 0.26992\omega^2)\left(1-\sqrt{T/T_c}\right)\right]^2.$$

In this work, the acentric factor $\omega = 0.344$ is chosen for water.

Thus, the density and velocity are obtained by

$$\rho = \sum_i f_i \tag{6a}$$

$$\rho\mathbf{u} = \sum_i \mathbf{e}_i f_i \tag{6b}$$

The macroscopic fluid velocity $\mathbf{U}$ can be obtained by

$$\rho\mathbf{U} = \sum_i \mathbf{e}_i f_i + \frac{\delta t}{2}\mathbf{F} \tag{7}$$

The temperature evolution equation given by [26] is

$$\begin{aligned} g_i(\mathbf{x}+\mathbf{e}_i\delta_t, t+\delta_t) - g_i(\mathbf{x},t) = &-\frac{1}{\tau_T}(g_i(\mathbf{x},t) - g_i^{(eq)}(\mathbf{x},t)) \\ &+ \delta_t\omega_i\left(\left[T\nabla\cdot\boldsymbol{U} + \frac{1}{\rho^2 c_v}\left(\frac{\partial p}{\partial T}\right)_\rho \frac{\mathrm{d}\rho}{\mathrm{d}t}\right]\right. \\ &\left.+\left[\frac{1}{\rho c_v}\nabla\cdot(\lambda\nabla T) - \nabla\cdot\left(\frac{\lambda}{\rho c_p}\nabla T\right)\right]\right) \end{aligned} \tag{8}$$

where $\tau_T = 0.5 + \alpha/c_s^2\delta_t$ is the relaxation time, and the equilibrium distribution function $g_i^{(eq)}(\mathbf{x}, t)$ is given by

$$g_i^{(eq)} = \omega_i T\left[1 + \frac{\mathbf{e}_i\cdot\mathbf{U}}{c_s^2} + \frac{(\mathbf{e}_i\cdot\mathbf{U})^2}{2c_s^4} - \frac{\mathbf{U}^2}{2c_s^2}\right] \tag{9}$$

At the two-phase interface, the physical properties χ (such as thermal diffusivity and viscosity) are given by

$$\chi = \chi_{liquid}\cdot\frac{\rho-\rho_{vapor}}{\rho_{liquid}-\rho_{vapor}} + \chi_{vapor}\cdot\frac{\rho_{liquid}-\rho}{\rho_{liquid}-\rho_{vapor}} \tag{10}$$

The temperature is calculated by

$$T = \sum_i g_i \tag{11}$$