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Mesoscale simulations of boiling curves and boiling hysteresis under constant wall temperature and constant heat flux conditions

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ABSTRACT

Mesoscale simulations for pool boiling curves and boiling hysteresis on hydrophilic/hydrophobic surfaces, under constant wall temperature/constant wall heat flux conditions, are presented in this paper. It is found that simulated boiling curves in dimensionless form under these two different heating modes are identical in nucleate boiling and film boiling regimes, and they differ only in the transition boiling regime. Saturated temperatures have relatively small effects on boiling curves up to the fully-developed nucleate boiling regime, but have pronounced effects on critical heat flux and on film boiling. Boiling hysteresis between increasing heating and decreasing heating are also simulated numerically. It is confirmed numerically that boiling hysteresis exists in transition boiling regime for both hydrophilic and hydrophobic surfaces under controlled wall heat flux conditions. Under controlled wall temperature conditions, however, boiling hysteresis exists only on a hydrophobic surface during decreasing heat flux but no boiling hysteresis exists on a hydrophilic surface. Rohsenow's classical correlation equation for nucleate boiling heat transfer matches well with simulated nucleate boiling heat transfer results for smooth horizontal superheated surfaces. Simulated critical heat fluxes are in qualitative agreement with those predicted by Zuber's hydrodynamic model and by Kandlikar's analytical model.

1. Introduction

Pool boiling is a complex heat transfer process as it is affected by many factors, including heating modes (constant wall heat flux or constant wall temperature), thermal properties of vapor and liquid phases, size, orientation and surface properties of the heater (contact angle and surface microstructures), heating conditions (increasing or decreasing wall heat flux or wall temperature), and saturated temperatures. For high heat flux thermo-fluid systems using phase-change working media, boiling curve is the most important information for the design and safe operation.

In 1934, Nukiyama [1] performed well-known experiments on pool boiling from horizontal wires in water under *controlled heat flux conditions*. When heat flux was increased to the maximum heat flux (which has been called the critical heat flux (CHF)), a temperature jump occurred after a slight increase in heat flux. He classified the heat transfer process into nucleate boiling regime, transition boiling regime and film boiling regime [1], and presented his data in terms of heat flux versus wall superheat which has since been called a boiling curve. Shortly after, Drew and Muller [2] studied boiling heat transfer in transition boiling regime under *controlled temperature conditions*. Subsequently, numerous researchers conducted experiments on different boiling regimes to study their heat transfer characteristics [3]. For example, Rohsenow [4] obtained a correlation equation for the nucleate boiling regime with different working fluids and surface conditions. Kutateladze [5] obtained a correlation equation for CHF based on a dimensionless analysis, and subsequently Zuber [6] proposed a CHF model based on a hydrodynamic instability analysis. Lienhard [7,8] refined the CHF model and studied heater size's effects on critical heat flux. Kandlikar [9] performed a force balance analysis to study contact angle effects on CHF for smooth surfaces, and found that CHF decreases as the contact angle increases, which is in agreement qualitatively with pool boiling experiments [10].

Relatively less work has been published on transition boiling regime in comparison with other boiling regimes. Much of the transition boiling heat transfer data were obtained from quenching experiments [11]. One of the most interesting phenomena in transition boiling is the boiling hysteresis phenomena, i.e., boiling curves between increasing and decreasing wall temperatures (or wall heat flux) are different depending on wettability of the heater surface. Sakurai and Shiotsu [12] were probably the first to document the boiling hysteresis phenomena. Subsequently, Ramilison and Lienhard [13] carried out experiments to study hysteresis phe-

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nomena in transition boiling regime under controlled wall temperature conditions on surfaces with different advancing contact angles. They found that the heater surface with larger advancing contact angles had a larger boiling hysteresis than those of well-wetted surfaces. Based on the vapor volume fraction on the wall, Liaw and Dhir [14,15] obtained a correlation for transition boiling heat transfer on surfaces with different wettabilities under increasing and decreasing wall temperatures. However, Auracher and Marquardt [16,17] performed experiments with well-wetting fluids (such as FC-72, isopropanol and water) under controlled wall temperature conditions, and found that boiling hysteresis occurred only in transient boiling conditions. They claimed that no boiling hysteresis existed on clean surfaces (hydrophobic) under steady-state conditions.

Recently, lattice Boltzmann methods have been successfully developed to study pool boiling phenomena numerically. Gong and Cheng [18,19] proposed an improved phase-change lattice Boltzmann model based on Hazi and Markus's paper [20,21] with a simpler heat source term. Based on this improved model, Gong and Cheng [22] obtained boiling curves for smooth surfaces with different wettabilities under *controlled wall temperature conditions* numerically for the first time. Based on the modified version of this model, Zhang and Cheng [23] studied effects of subcooling and heater size on boiling curves under *controlled wall heat flux conditions*. Most recently, Gong and Cheng [24] investigated roughness effects on boiling curves under controlled wall temperature conditions numerically. Apart from Cheng and co-workers' work [18,19,22–24], others [25–28] have also used similar methods to simulate bubble nucleation and boiling curves numerically.

As a continuation of our previous work, we will use the improved version of the phase-change lattice Boltzmann method to simulate effects of the saturated temperature and boiling hysteresis on boiling curves in this paper. Simulated results for nucleate boiling regime are compared with Rohsenow's correlation equation [4]. The good agreement between simulated results and existing correlation for nucleate boiling heat transfer suggests that the newly developed LB phase-change method is a promosing tool for studies of boiling heat transfer phenomena. Simulated critical heat fluxes are also compared with those predicted by Zuber's hydrodynamic model [6] and by Kandlikar's analytical model [9].

2. The computation model

2.1. Gong-Cheng's improved phase-change LB model

The improved lattice Boltzmann phase-change LB model used in this paper is presented in details in our previous paper [23] and we will only be briefly mentioned here. Based on the Gong-Cheng phase change model [18,19], the density's evolution equation in half-implicit scheme is given by

$$f_i(\boldsymbol{x}+\boldsymbol{e}_i\delta t, t+\delta t) - f_i(\boldsymbol{x}, t) = -\frac{1}{\tau}\left(f_i(\boldsymbol{x}, t) - f_i^{eq}\left(\boldsymbol{x}, t+\frac{\delta t}{2}\right)\right) + \Delta f_i\left(\boldsymbol{x}, t+\frac{\delta t}{2}\right), i = 0, 1, \ldots N \tag{1}$$

where τ is dimensionless relaxation time decided by the kinetic viscous of the fluid with $\tau = 3\nu + 0.5$. $f_i^{eq}(\boldsymbol{x}, t+\frac{\delta t}{2})$ and $\Delta f_i(\boldsymbol{x}, t+\frac{\delta t}{2})$ are the corresponding equilibrium distribution and exact different term [29] given by

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

$$\Delta f_i\left(\boldsymbol{x}, t+\frac{\delta t}{2}\right) = f_i^{eq}(\bar{\rho}(\boldsymbol{x}, t), \bar{\boldsymbol{u}} + \Delta\boldsymbol{u}) - \boldsymbol{f}_i^{eq}(\bar{\rho}(\boldsymbol{x}, \boldsymbol{t}), \bar{\boldsymbol{u}}) \tag{3}$$

where $\bar{\rho}$ and $\bar{\boldsymbol{u}}$ are the mean value of time t and the estimated value of time $t + \delta t$ [23].

The evolution of temperature distribution $g_i(\boldsymbol{x}, t)$ and its equilibrium distribution are given by

$$g_i(\boldsymbol{x}+\boldsymbol{e}_i\delta t, t+\delta t) - g_i(\boldsymbol{x}, t) = -\frac{1}{\tau_T}(g_i(\boldsymbol{x}, t) - g_i^{eq}(\boldsymbol{x}, t)) + \delta t\omega_i\varphi \tag{4}$$

$$g_i^{eq}(\boldsymbol{x}, t) = \omega_i T\left[1 + \frac{\boldsymbol{e}_i \cdot \boldsymbol{U}}{c_s^2} + \frac{(\boldsymbol{e}_i \cdot \boldsymbol{U})^2}{2c_s^4} - \frac{\boldsymbol{U}^2}{2c_s^2}\right] \tag{5}$$

where dimensionless relaxation $\tau_T = 3\alpha + 0.5$, with α being the thermal diffusivity. In the last term in Eq. (4), φ is the source term which will be discuss in the next section.

2.2. The source term

Base on entropy balance equation and the thermodynamic relationship, Hazi and Markus [20] have derived the source term of the following form

$$\varphi = -\frac{1}{\rho c_v}\left(\frac{\partial p}{\partial T}\right)_\rho \nabla \cdot \boldsymbol{U} \tag{6a}$$

In the above equation, $(\partial p/\partial T)_\rho$ is evaluated from the equation of state and $\boldsymbol{U}$ is the real fluid velocity. When this source term was applied to the problem of vapor bubble rise from a superheated horizontal surface [20], they found that two unsymmetrical circulations exist in the flow field which is physically unrealistic.

Subsequently, other source terms are proposed by different investigators. For example, Gong and Cheng [18] obtained the following source term for the phase-change lattice Boltzmann method

$$\varphi = T\left[1 - \frac{1}{\rho c_v}\left(\frac{\partial p}{\partial T}\right)_\rho\right]\nabla \cdot \boldsymbol{U} \tag{6b}$$

which has been applied to a number of boiling and condensation problems [22–24,30]. Recently, Li et al. [26] obtained the following source term

$$\varphi = T\left[1 - \frac{1}{\rho c_v}\left(\frac{\partial p}{\partial T}\right)_\rho\right]\nabla \cdot \boldsymbol{U} + \left[\frac{1}{\rho c_v}\nabla \cdot (\lambda\nabla T) - \nabla \cdot \left(\frac{\lambda}{\rho c_v}\nabla T\right)\right] \tag{6c}$$

where c is the specific heat, λ is the thermal conductivity. Markus and Hazi [21] as well as Tao and coworkers [27,28] also proposed similar source terms.

After a careful derivation of the energy equation, we have derived the following exact expression for the source term.

$$\varphi = T\left[1 - \frac{1}{\rho c_v}\left(\frac{\partial p}{\partial T}\right)_\rho\right]\nabla \cdot \boldsymbol{U} + \left[\frac{1}{\rho c_v}\nabla \cdot (\lambda\nabla T) - \nabla \cdot \left(\frac{\lambda}{\rho c_p}\nabla T\right)\right] \tag{6d}$$

where c_p and c_v are thermal specific heat at constant pressure and constant volume, respectively. It should be noted that Eq. (6b) or Eq. (6c) is just a special case of Eq. (6d): (i) Eq. (6d) reduces to Eq. (6c) if $c_p = c_v$, and (ii) Eq. (6d) reduces to Eq. (6b) if $c_p = c_v$ and thermal conductivity as well as ρc_v is constant. The physical interpretation of the source terms given by Eq. (6d) are as follows: the first square blanket represents the source term owing to compressibility effects while the second square blanket represents the source terms due to property variations. In the following, it will be shown numerically that the second square blanket term in Eq. (6d) is also small in comparison with the first square blanket term if physical properties are not constant and c_p and c_v are not the same.