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Numerical and analytical study of film boiling in a planar liquid jet☆

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ABSTRACT

Direct numerical simulation of film boiling in a planar liquid jet is performed by solving the conservation equations of mass, momentum and energy in the liquid, vapor and air phases. The liquid–air and liquid–vapor interfaces are tracked by a sharp-interface level-set method, which is modified to include the effect of phase change at the liquid–vapor interface. An analytical model to predict the vapor film thickness and wall heat flux in the stagnation region is also developed by simplifying the momentum and energy equations in the liquid and vapor phases. The computational results show a stable vapor film formation on the wall. The effects of jet subcooling, jet velocity, and wall temperature on the vapor film thickness and boiling heat transfer are investigated.

1. Introduction

Liquid jet impingement is one of the primary cooling techniques of hot metal plates in the metal production industry. When the plate has a very high temperature, as in its initial cooling stage, film boiling occurs and a continuous vapor film is formed between the hot plate and the impinging liquid. An understanding and prediction of the boiling process is essentially important to achieve the desired mechanical and metallurgical properties of metals.

Several experimental studies of film boiling in liquid jet impingement were reported in the literature [1–4], but most of them were conducted under transient (or quenching) conditions, where film, transition, and nucleate boiling modes appear sequentially or simultaneously on the wall surface and thus the hydrodynamic and thermal characteristics of film boiling are not clearly separated from the other boiling modes. Also, considering that the temperature and heat flux distributions on the wall surface were obtained from the extrapolation of the measured data below the wall surface using an inverse heat conduction technique, the transient experimental data are expected to include the transient thermal effect of heater solid as well as the characteristics of boiling.

Only a few experimental studies were conducted under steady-state conditions. Robidou et al. [5,6] performed steady-state experiments of entire boiling regimes in a planar water jet, having a width of 1 mm. Their data showed that, under the conditions of a jet velocity of 0.8 m/s and a subcooling of 16 °C, the film boiling regime started at a wall temperature of about 450 °C, which is called the Leidenfrost temperature (or the minimum film boiling temperature). The Leidenfrost temperature and the film boiling heat transfer were strongly dependent on the jet subcooling. The wall temperature signals also indicated no significant fluctuations in time, which means the vapor film in liquid jet impingement is stable.

Bogdanic et al. [7] used a miniaturized optical probe of 1.5 μm tip diameter to investigate the two-phase structures underneath a planar water jet, whose experimental configuration was similar to that used by Robidou et al. [5,6]. The liquid or vapor contact signals obtained from the optical probe showed that the liquid contact frequency was very high in the transition boiling regime whereas the liquid contacts disappeared in the film boiling regime. The measured vapor film thickness was 8 ± 2 μm under the experimental conditions of a jet velocity of 0.4 m/s and a subcooling of 20 °C.

Efforts were made to predict the film boiling heat transfer in liquid jet impingement. Nakanish et al. [8] analyzed film boiling in the stagnation region of a planar water jet by solving the similarity equations for the conservation of mass, momentum, and energy in the liquid and vapor phases. They also presented a simplified analytical model to predict the film boiling heat transfer. Applying the potential-flow approximation to the liquid layer and the lubrication approximation to the thin vapor film, they derived the vapor film thickness δ_v and the wall heat flux q_w in the stagnation region as

$$\rho_v V_j h_{lv} \frac{a\delta_v}{2W_j}\left(1+\frac{a\mu_l\delta_v^2}{6\mu_v W_j^2} Re_l\right) = \frac{\lambda_v \Delta T_w}{\delta_v} - \frac{\lambda_l \Delta T_{sub}}{W_j}\sqrt{\frac{2a}{\pi} Re_l Pr_l} \quad (1)$$

$$q_w = \frac{\lambda_v \Delta T_w}{\delta_v} \quad (2)$$

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Nomenclature

a	dimensionless gradient of the free-stream velocity, $(u_\infty/V_j)/(x/W_j)$
c_p	specific heat at constant pressure
g	gravity
h	grid spacing
h_{lv}	enthalpy of vaporization
p	pressure
Pr_l	liquid Prandtl number, $c_{pl}\mu_l/\lambda_l$
q	heat flux
Re_l	liquid Reynolds number, $\rho_l V_j W_j/\mu_l$
t	time
T	temperature
ΔT_{sub}	jet subcooling, $T_{sat} - T_j$
ΔT_w	wall superheat, $T_w - T_{sat}$
u	horizontal flow velocity
u_∞	free-stream velocity
u_δ	liquid or vapor velocity at the interface
v	vertical flow velocity
V_j	jet velocity
W_j	jet width
x, y	horizontal and vertical coordinates

Greek symbols

δ_l	liquid velocity boundary layer thickness
δ_t	liquid thermal boundary layer thickness
δ_v	vapor film thickness
λ	thermal conductivity
μ	viscosity
ρ	density

Subscripts

j	jet
l, v	liquid, vapor
sat	saturation
w	wall

where the dimensionless constant a is used to express the free-stream velocity u_∞ outside of the liquid boundary layer, which is defined as

$$\frac{u_\infty}{V_j} = a\frac{x}{W_j} \tag{3}$$

Another theoretical model was developed by Liu and Wang [9] for film boiling in the stagnation region of a circular water jet. They used the cubic polynomial profiles for the velocity and temperature in the liquid layer, the Reynolds analogy for the liquid thermal boundary layer thickness and the lubrication approximation for the velocity and temperature profiles in the vapor film. The model was modified by using an empirical correlation factor, which was determined from their experimental data. The wall heat flux for a highly subcooled liquid, which is also applied to a planar jet, was derived as

$$q_w = 2Re_l^{1/2}Pr_l^{1/6}(\lambda_v\lambda_l\Delta T_w\Delta T_{sub})^{1/2}/W_j \tag{4}$$

Karwa et al. [10] proposed a theoretical model for determining the Leidenfrost condition in liquid jet impingement. The Leidenfrost condition was assumed as the zero shear stress at the liquid–vapor interface. They used a parabolic profile for the liquid temperature and the lubrication approximation for the vapor film. The wall superheat and heat flux at the Leidenfrost condition were derived for a planar jet as

$$\Delta T_w = 0.99\left(\frac{\lambda_l}{\lambda_v}\right)\left(\frac{\mu_v}{\mu_l}\right)^{1/2}Pr_l^{1/2}\Delta T_{sub} \tag{5}$$

$$q_w = 0.7\frac{\lambda_l\Delta T_{sub}}{W_j}Re_l^{1/2}Pr_l^{1/2} \tag{6}$$

Direct numerical simulation (DNS) is another way further clarifying the physics of film boiling in liquid jet impingement. Computational efforts were made for film boiling on flat or cylindrical surfaces using a moving-grid method [11], a front-tracking method [12,13], a level-set (LS) method [14–16], and volume-of-fluid method [17,18]. Very recently, Kim and Son [19] presented the LS method for computation of liquid jet impingement without and with film boiling.

In this study, DNSs are further extended for film boiling in a planar water jet by using the sharp-interface LS method, which is modified to track the liquid–air and liquid–vapor interfaces and to treat the effect of phase change at the liquid–vapor interface. An analytical model to predict the vapor film thickness and wall heat flux in the stagnation region is also developed by simplifying the momentum and energy equations in the liquid and vapor phases. The effects of jet subcooling, jet velocity and wall temperature on the vapor film thickness and wall heat flux during film boiling are investigated.

2. Analysis

2.1. DNS

Fig. 1 shows the configuration used for simulation of film boiling in a planar (or two-dimensional) jet. The flow is assumed to be laminar. The liquid–air and liquid–vapor interfaces are tracked by the LS function, which is defined as a signed distance from the interface. The positive sign is chosen for the liquid phase, and the negative sign for the gas (vapor or air) phase. To solve the conservation equations of mass, momentum and energy in the liquid and gas regions as well as the LS advection and reinitialization equations, we employ the numerical approach developed in the previous study [19].

2.2. Analytical model

An analytical model to predict the vapor film thickness and wall heat flux in the stagnation region of a planar jet is developed by

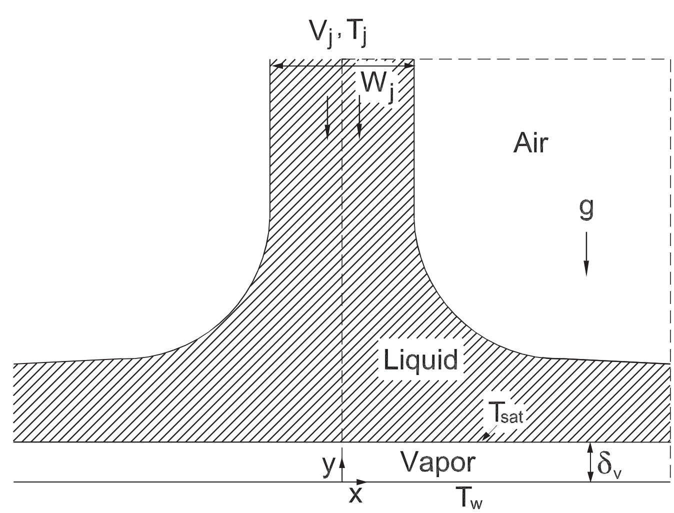

Fig. 1. Configuration used for computation of film boiling in a planar liquid jet.