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Effect of non-condensable gas on laminar film condensation of steam in horizontal minichannels with different cross-sectional shapes☆

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ABSTRACT

In the present study, a 3-D numerical simulation of laminar film condensation of steam in the presence of non-condensable gas is performed in horizontal minichannels with six cross-sectional shapes based on the volume of fluid (VOF) method. Mixture of steam and oxygen enters the channel with uniform temperature, while the inlet volume fraction of oxygen increases from 0% to 3%. It is shown that the existence of non-condensable gas results in significant reduction in the mass transfer rate from vapor to liquid along the interface in the axial direction. And then the heat transfer coefficient of condensation with oxygen is demonstrated to decline sharply compared with that of the pure vapor condensation.

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1. Introduction

During vapor condensation in the industrial applications, such as turbine plant, condenser in air conditioner, and electronic devices, condensation plays an important role because of its excellent heat transfer performance. However, vapor always mixes with some species that may not condense depending on different working conditions [1], and the non-condensable gas will have a negative effect on condensation heat transfer. For example, less than 1% (by mass) of air in steam condensation can make the condensation heat transfer coefficient drop by more than half, induced by a concentration of non-condensable gas near the vapor–liquid interface [2]. Therefore, condensation in the presence of non-condensable gas has attracted great attention in the recent years.

Minkowycz and Sparrow [3] developed an analytical investigation of laminar film condensation of steam on a vertical isothermal plate in the presence of air. The results indicated that the influence of the non-condensable gas on the heat transfer was accentuated at lower pressure levels, and the effect of superheating on condensation in the presence of a non-condensable gas was much more significant than that in the case of pure vapor condensation. In the recent years, numerical simulation of condensation with non-condensable gas mainly depended on the solution of the boundary layer equations [4–8]. The double boundary layer model was developed by Tang et al. [4] to study the film condensation with air outside a horizontal tube. By solving the coupled heat and mass transfer simultaneously with the finite difference method, the performance of condensation heat transfer was found to deteriorate dramatically even with a little air, which agreed well with the experimental data.

Some experiments have also been performed to investigate the condensation heat transfer in the presence of non-condensable gas. The effect of air on condensation heat transfer in horizontal tubes was investigated experimentally by Ren et al. [9]. The results showed that the overall heat transfer coefficient decreased with higher inlet non-condensable gas mass fraction and higher inlet pressure. In addition, two correlations were developed for the stratified flow and annular flow regimes, respectively. An experiment at atmospheric pressure varying the air concentration was conducted by Chung et al. [10] to analyze the condensation heat transfer of steam–air mixture on a vertical flat plate. It was found that the mixture flow would enhance the heat transfer substantially.

The studies above indicate that the laws of condensation have not been totally clarified yet. In addition, the influence of non-condensable gas in non-circular minichannels has hardly been discussed before. Therefore, a numerical simulation of steam condensation with non-condensable gas in horizontal minichannels with different cross-sectional shapes is conducted by the VOF method in ANSYS FLUENT [11].

2. Numerical model

2.1. Physical model

Totally, six three-dimensional horizontal minichannels with different cross-sectional shapes are adopted. The hydraulic diameter and length of the channels are 1 mm and 50 mm, respectively. The detailed cross-sectional shapes include circular, square, rectangular (aspect ratio of 2 in two directions), and equilateral triangular (regular and inverted),

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Nomenclature

c_p	Specific heat, J/(kg · K)
$\vec{F}$	Surface tension, N/m^3
$\vec{g}$	Gravity, m/s^2
h	Heat transfer coefficient, W/(m^2 · K)
k	Thermal conductivity, W/(m·K)
L_H	Latent heat of vapor, J/kg
$\dot{m}$	Mass source due to phase change, kg/(m^3·s)
n	Categories of phases
p	Pressure, Pa
r	Mass transfer coefficient, s^{-1}
T	Temperature, K
$\vec{v}$	Velocity, m/s
W	Mass fraction of non-condensable gas, %

Greek symbols

α	Volume fraction, %
μ	Dynamic viscosity coefficient, Pa·s
ρ	Density, kg/m^3
φ	Physical property

Subscripts

g	Non-condensable gas
l	Liquid phase
oxy	Oxygen
s	Saturation

Abbreviation

VOF	Volume of fluid

as shown in Fig. 1. The temperature of the mixture at the inlet is the saturation temperature of the vapor at atmospheric pressure. The inlet volume fraction of the non-condensable gas varies from 0% to 3%, while the inlet velocity of the mixture and the wall temperature of the tube wall remain constant at 10 m/s and 353.15 K, respectively.

2.2. VOF method

VOF method is a widely used approach for the numerical simulation of multiphase flow, which belongs to the Euler–Euler multiphase models [11]. It is a surface-tracking technique that can model two or more immiscible fluids by solving a single set of momentum equations and tracking the volume fraction of each of the fluids throughout the domain.

In the VOF method, a phase will be designated as the primary phase, and the other phases are the secondary phases. The volume fractions of the secondary phases are obtained by solving the volume fraction equation of the corresponding phase, while that of the primary phase is achieved based on the theory that the volume fractions of all phases equal to unity. In the present study, the vapor is defined as the primary phase. Therefore, the volume fraction equations of the liquid; non-condensable gas and mixture are presented in the following way:

$$\nabla \cdot \left(\alpha_l \rho_l \vec{v}\right) = \dot{m} \tag{1}$$

$$\nabla \cdot \left(\alpha_g \rho_g \vec{v}\right) = 0 \tag{2}$$

$$\nabla \cdot \left(\rho \vec{v}\right) = 0 \tag{3}$$

In the above equations, subscripts l and g represent the liquid and non-condensable gas, respectively. In Eq. (1), $\dot{m}$ is the mass source due to the phase change from vapor to liquid. However, there is no mass transfer between the gas and other phases in the condensation process with non-condensable gas. As a result, the mass source in Eq. (2) is 0. The velocity in the equations above is shared among all the phases, as well as for the temperature distribution. Hence, the momentum equation and energy equation are as follows:

$$\nabla \cdot \left(\rho \vec{v}\vec{v}\right) = -\nabla p + \nabla \cdot \left[\mu\left(\nabla \vec{v} + \nabla \vec{v}^{T}\right)\right] + \rho \vec{g} + \vec{F} \tag{4}$$

$$\nabla \cdot \left(\vec{v}\left(\rho c_p T + p\right)\right) = \nabla \cdot (k \nabla T) + L_H \dot{m} \tag{5}$$

where $\vec{F}$ and L_H represent the surface tension and the latent heat of vapor, respectively. The physical properties of the multiphase flow shown in the equations above will be assigned based on the local volume fraction and physical properties of each phase:

$$\varphi = \sum_{1}^{n} \alpha_q \varphi_q \text{ with } \varphi = \rho, k, \mu \tag{6}$$

$$\varphi = \frac{1}{\rho} \sum_{1}^{n} \alpha_q \rho_q \varphi_q \text{ with } \varphi = c_p \tag{7}$$

where n represents the categories of phases.

In the present study, a steady state simulation with implicit formulation is performed. The pressure–velocity coupling is handled by means of the SIMPLEC algorithm, while the PRESTO! scheme is adopted for the pressure interpolation. The QUICK scheme is employed for the momentum, VOF, and energy equations.

2.3. Phase change model

As the standard VOF method is applied for the multiphase flow without phase change, a phase change model is needed during the application of the VOF method for the numerical simulation of condensation.

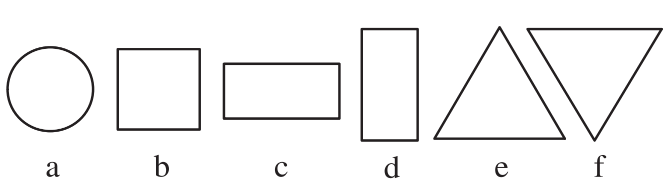

Fig. 1. Schematic of the minichannels with different cross-sectional shapes: (a) circular, (b) square, (c) horizontal rectangular, (d) vertical rectangular, (e) regular equilateral triangular, (f) inverted equilateral triangular.

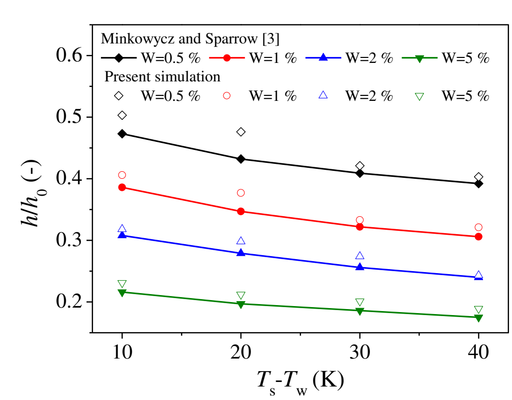

Fig. 2. Validation of the numerical model.

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