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CFD evaluations on bundle effects for steam condensation in the presence of air under natural convection conditions

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ABSTRACT

Steam condensation in the presence of air is a relevant phenomenon in various industrial applications. Its heat transfer property has been broadly revealed by experimental and numerical investigations on vertical plates or single tubes. However, these results may not be directly applied to tube bundle cases, since there may exist mutual interactions among adjacent tubes. To have an insight on this problem, the present work conducted numerical simulations on various tube bundles with a tube pitch 1.5 times of the tube diameter. The cases evaluated were classified into three categories including the single row, double row, and triple row. In each category, structures with various tube columns were assessed. The results indicate that tube bundles will thicken the near-wall high concentration air layer, resulting in inhibited condensation heat transfer. This phenomenon is defined as the inhibition effect. On the other hand, a heat transfer enhancement effect caused by a strengthened natural circulation driven by the density difference between the mainstream and the high concentration air region is found and defined as the stack effect. The average condensation heat transfer for tube bundles is determined by the relative magnitude between the inhibition effect and the stack effect. The stack effect becomes much intensive with the increase of tube rows and columns, which can enhance condensation heat transfer. Typically, the results in the triple row categories show that the maximum average heat transfer coefficient can be 14% greater than that of the single tube.

1. Introduction

Vapor condensation is an important heat transfer process in various industrial applications including the coal power plant [1], refrigeration [2], seawater desalination [3], chemical engineering [4], energy efficient devices [5] etc. In some energy applications, such as the design of heat exchangers [6] and the application of passive containment cooling systems in nuclear power plants [7], the steam condensation process is in the presence of non-condensable gases like air. The existence of non-condensable gases can affect condensation heat transfer [8].

To have an insight on this problem, various experimental and numerical investigations have been performed [9–12] on single tubes or vertical plates. Via these studies, it was found that the existence of air can to a large degree deteriorate steam condensation heat transfer, leading to its heat transfer coefficient more than an order of magnitude smaller than that of the pure steam. To quantitatively describe the influence of air, some experimental studies proposed heat transfer coefficient (HTC) correlations in terms of air mass fraction, pressure, wall sub-cooling etc. [13–17]. In order to analyze the reasons caused the deterioration effect, some theoretical and numerical studies were conducted [6,7,18,19]. They found that after steam condensates at the liquid-gas interface, a high concentration air layer will be built in near-wall regions. Since the mainstream steam has to diffuse through this air layer before condensing, the high concentration air layer becomes the main thermal resistance for steam condensation [20]. In contrast, thermal resistances of liquid film and gas convection are small enough to be neglected.

It is noteworthy that the above-mentioned studies are based on single tubes or vertical plates. In some actual application cases, e.g. heat exchangers or passive condensation cooling systems, tube bundle heat exchange components are commonly used. Different from a vertical plate or a single tube, there may have mutual influences among adjacent tubes in tube bundle conditions, which may lead to some special phenomena. These characteristics may further affect the condensation heat transfer for individual tubes or tube bundles. Thus, the results concluded from a plate or a single tube cannot be directly applied to analyze tube bundle cases. Although there are many papers dealing with single tube, there is little published literature discussing steam

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condensation consisting of air for vertical tube bundles under natural convection conditions, so it is necessary to conduct such investigations.

2. Calculation models

For the convenience of local phenomena analyses and reducing the cost of investigation, the CFD method was applied to evaluate condensation heat transfer for various tube bundles. The governing equations and turbulence model employed are the same as the common CFD calculations. In comparison, the condensation model plays the key role in simulating steam condensation in the presence of air. To date, two commonly used condensation models have been developed. One is the experimental correlation model, and the other is the diffusion boundary layer model [21]. It is generally recognized that the latter model demonstrates advantages in local phenomena analyses and can describe the diffusion of steam across the near-wall high concentration non-condensable gas layer. Our previous papers have developed, discussed and validated the diffusion boundary layer model [20–22] based on the CFD software STAR-CCM+. Considering the numerical conditions, including flow state (natural convection), pressure, air mass fraction and wall sub-cooling, are within the validated parameter scope and the difference merely exists in bundle structures, it is reasonable to evaluate tube bundle cases via CFD code with the diffusion boundary layer condensation model. For the convenience of discussion, the whole model and previous validations are presented here simply.

2.1. The governing equations and turbulence model

The commercial CFD software STAR-CCM+ with user field functions was employed to investigate steam condensation characteristics for various bundle structures. This CFD code solves the local transport equations for mass, momentum, energy, and species by setting proper initial and boundary conditions. These equations are written as:

Mass conservation:

$$\frac{\partial \rho}{\partial t} + \nabla\cdot(\rho \overrightarrow{w}) = S_m \tag{1}$$

Where ρ is the density, t is the time, w is the velocity, and S_m denotes the mass source term.

Momentum conservation:

$$\frac{\partial (\rho \overrightarrow{w})}{\partial t} + \nabla\cdot(\rho \overrightarrow{w}\,\overrightarrow{w}) = \nabla\cdot P + \rho f + S_{\rho w} \tag{2}$$

Where P is the surface force, τ is the shear stress, f is the volume force, and $S_{\rho w}$ denotes the momentum source term.

Energy conservation:

$$\frac{\partial (\rho E)}{\partial t} + \nabla\cdot(\rho \overrightarrow{w} E) = \rho f\cdot\overrightarrow{w} + \nabla\cdot(P\cdot\overrightarrow{w}) + \nabla\cdot(k_{eff}\nabla T) + S_h \tag{3}$$

Where E is the energy, k_{eff} is the effective thermal conductivity, T is the temperature and S_h denotes the energy source term.

Species conservation:

$$\frac{\partial (\rho \omega_i)}{\partial t} + \nabla\cdot(\rho \overrightarrow{w}\omega_i) = \nabla\cdot(\rho D_i \nabla\omega_i) + S_i \tag{4}$$

Where ω is the mass fraction, and D denotes the diffusion coefficient. The subscript i represents species.

As for the turbulence model, various models have been applied in the previous literature, and there is no best guideline for which one is most suitable for simulating steam condensation in the presence of air. Thus, in the subsequent simulations, the CFD code recommended realizable k-ε turbulence with two layers all y + treatment was employed. The equations are described by:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho \overrightarrow{w_i} k)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right] + P_k + G_b - \rho\varepsilon - Y_M \tag{5}$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \overrightarrow{w_i}\varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right] + \rho C_1 \overline{S}\varepsilon - C_2\rho\frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}} + C_{\varepsilon 1}\frac{\varepsilon}{k}C_{\varepsilon 3}G_b \tag{6}$$

Where k is the turbulent kinetic energy, x_i is the coordinates, μ is the dynamic viscosity, μ_t is the turbulent dynamic viscosity, σ_k is the generation of turbulence kinetic energy due to mean velocity gradients, G_b is the generation of turbulence kinetic energy due to mean buoyancy, ε is the turbulence dissipation rate, Y_M is the dissipation rate due to the fluctuation in compressible turbulence, σ_ε is the generation of the turbulence dissipation rate due to mean velocity gradients, C_1, C_2, $C_{\varepsilon 1}$, and $C_{\varepsilon 3}$ are the constants, $\overline{s}$ is the average strain rate tensor, and ν denotes the kinetic viscosity.

2.2. Steam condensation model

The mass, momentum, and energy source terms in Eqs. (1)–(4) are as follows.

(1) Mass source term:

$$S_m = S_i = m_{cond}/\Delta \tag{7}$$

Where,

$$m_{cond} = -\left(\frac{\rho D}{1-\omega_v}\right)\frac{\partial \omega_v}{\partial n}\bigg|_i \tag{8}$$

$$D = D_0\left(\frac{T}{T_0}\right)^{1.75}\left(\frac{P}{P_0}\right)^{-1} \tag{9}$$

Where m_{cond} is the condensation mass flux, Δ is the thickness of the cells close to the condensation wall, n is the normal direction of the condensation wall, and P is the pressure. The subscript 0 represents the standard conditions.

(2) Momentum source term:

$$\overrightarrow{S}_{\rho w} = S_m \overrightarrow{w} \tag{10}$$

(3) Energy source term:

$$S_h = S_m h_v \tag{11}$$

Where h_v is the enthalpy flux.

2.3. Model validations and analyses

The applicability of the above-mentioned diffusion boundary layer steam condensation model has been validated and analyzed in a large pressure, wall sub-cooling and air mass fraction scope in our previous studies [20–22]. In these studies, we assessed the applicability of the steam condensation model via various experimental results, including the COPAIN [23], Uchida [16] and Su [24] at various pressure, velocity, mass fraction and wall sub-cooling conditions. For the COPAIN experiment [21], we mainly compared the calculated local heat flux with the experimental ones. The results indicated that they match well with each other and the deviations are generally within ± 25%. For the Uchida and Su experiment [20,22], much attention was paid to the accuracy of the average condensation heat transfer coefficient. The predicted results agree well with the experimental data and deviations for the Uchida and the Su experiment are ± 20% and ± 15%, respectively. In addition, local phenomena analyses were also performed and local temperature gradients were compared with the experimental ones [20]. The results show that the condensation model demonstrates the advantages of local phenomena analyses. According to the above-mentioned evaluations, we drew the conclusion that the diffusion boundary layer steam condensation model is applicable in simulating steam condensation process consisting of non-condensable gases.

In our previous work [21], the performance of the steam condensation model at various mesh conditions was also assessed. In the