Film condensation heat transfer on a horizontal tube in presence of a noncondensable gas


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The double boundary layer model has been developed to study the behavior of film condensation heat transfer outside a horizontal tube in presence of air treated as a noncondensable gas. And the coupled heat and mass transfer on a smooth horizontal tube is numerically solved with the finite difference method. The local mass concentration of the noncondensable gas, the distributions of velocity and temperature in the boundary layers are presented and discussed. The numerical results have shown that the mass concentration and velocity of the noncondensable gas increase from the bulk mixture to the interface while the temperature decreases from the bulk mixture to the interface. Although the mass concentration of the noncondensable gas in the bulk mixture could be small, the reduction in average heat transfer coefficient is obvious. The comparisons of heat transfer coefficient show that the numerical predictions agree well with available experimental data.

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1. Introduction

Energy is a basic protection of the economic and social development, and energy efficiency is the key in terms of energy security. Compared with the solid fuel, a clean fuel such as natural gas is widely used, as high as 20% of volumetric fraction of water vapor in combustion products will be generated, in which the latent heat of the exhaust steam loss can be up to 10%–12%. In industry, condensation heat exchanger is widely applied to reuse remaining heat. As the phenomenon of film condensation occurs on most of industrial condensers, especially power condensers, it is necessary to recovery this part of energy by using flue gas condensation technology to achieve the goal of energy saving. There is a number of works reported on the film condensation heat transfer in the presence of a noncondensable gas.

For the condensation heat transfer in the presence of a noncondensable gas, most studies were focused on either theoretical simplification or experimental investigation on the effect of the noncondensable gas. Lebedev et al. [1] performed a combined experimental study of heat and mass transfer in condensation of vapor from humid air on a flat plate in longitudinal flow in an asymmetrically cooled slot. They observed an increase in condensation heat transfer with an increase in the relative humidity of the air. Osakabe and his colleagues [2,3] investigated real flue gas with different vapor mass fractions to study experimentally the heat and mass transfer characteristics of the condensing heat exchanger on horizontal smooth tubes and multiple stainless-steel tubes. An experimental system by means of heat-pipes has been designed for the recovery of waste-heat from exhaust gases in the literature [4]. Condensation from R113-air mixtures, and condensation from different zeotropic mixtures with pure HFC134a and HFC23 on the outside of a bank of horizontal smooth tubes were experimentally studied by Abdullah et al. [5] and Belghazi et al. [6], respectively. Maheshwari et al. [7] have developed a theoretical model to study the heat transfer coefficient of a condensing vapor in the presence of a noncondensable gas, taking into account the suction effect due to condensation, where the gas mixture flowed downward inside a vertical tube. Cheng and van der Geld [8] conducted experiments of condensation heat transfer in a polymer compact heat exchanger under various conditions for air/water heat exchange and airsteam/water heat exchange. And they found that the inlet noncondensable gas concentration has great effects on the overall heat transfer coefficient. In the study of Kim et al. [9], a theoretical model was proposed to evaluate the process of condensation heat transfer at high pressure using the heat and mass transfer analogy. More recently, Hwang et al. [10] experimentally studied latent heat recovery on a titanium heat exchanger. They compared experimental data and found that the convective heat transfer coefficient with condensation is 3–5 times higher than that of the case without condensation.

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experimentally investigated the problem of recovering waste-heat, including both the sensible and latent heat. By using humid air to simulate the exhaust, they found that the effect of water vapor concentration on heat transfer coefficient became more important with increased water vapor volume fraction, when steam volume fraction of the air-steam mixture was 5−15%.

A theoretical approach was proposed to predict the condensation in the presence of the noncondensable gas on the finned tube [12] with a modification to Peterson's condensation gas diffusion theory [13]. A steady-state heat transfer mathematical model for heat and mass conservation was proposed for the aim of the steam condensation heat transfer in the presence of a noncondensable gas, which was reported by Sparrow and Lin [15], Sparrow and Gregg [16], and Minkowycz and Sparrow [17].

In the process of condensation for the two-component mixture of noncondensable gas and vapor, interface temperature $T_i$ between the condensate film and mixture film is a key factor. Many researches [18−20] assumed a constant temperature of the mixture-liquid interface and ignored the shear stress of mixture-liquid interface in the theoretical analysis. When water vapor condensation occurs on a wall tube, inasmuch as the interface temperature $T_i$ is unknown, this assumption for condensation heat transfer on the outside of a horizontal tube is clearly unreasonable. And another assumption that mixture-liquid interfacial shear stress is zero is incorrect even for stagnant steam either. As shear stress of liquid−mixture interface prevents the liquid film from flowing downwards, the thickness of liquid film increases and local heat transfer coefficient decreases. By analyzing shear stress at liquid−mixture interface, Koh et al. [21] concluded that it is negligible in the case of the medium with $Pr > 10$, while interfacial shear stress cannot be ignored for the medium of lower Prandtl number. In the case of water vapor condensation, large errors will be caused if the effect of shear stress is neglected.

The numerical simulation was respectively employed to study condensation heat transfer of a nonazeotropic mixture and the behavior of a pure saturated vapor on a horizontal tube in the literature [22,23]. And a recent numerical simulation of the process of condensation or evaporation has been conducted by means of the finite volume technique by Morales-Ruiz et al. [24].

In summary, most of the existing studies provide the average heat transfer coefficient while the detailed local information of the diffusion, velocity and temperature is seldom presented. The main objective of the present study, therefore, is to numerically investigate the performance of film condensation heat transfer in the presence of a noncondensable gas with the finite difference method and to particularly evaluate the influence of the noncondensable gas on distributions of the velocity, temperature, gas concentration, and interface characteristics, as well as the heat transfer coefficient, which is of great significance to engineering design and optimizing in practical condensation heat transfer process.

2. Governing equations

Since horizontal tubes have a potentially higher heat transfer capability than vertical tubes [25], film condensation heat transfer outside a single horizontal tube is studied in the presence of a noncondensable gas in this paper by numerical simulation to avoid the above mentioned defects. First, some basic assumptions in the process of numerical analysis are as follows:

(1) The tube keeps uniform wall temperature along the circumference.
(2) The bulk vapor-gas mixture is at the static state and keeps with the saturation temperature.
(3) All physical properties, except for the buoyancy force are constant.
(4) Compared to the tube diameter, the thickness of both condensate film and vapor-gas boundary layer is very small.
(5) Multi-component system is always in a state of local equilibrium and there is no slip in the liquid film or the vapor-gas boundary layer.
(6) The noncondensable gas, the vapor, and the vapor-gas mixture are all treated as ideal gases.
As the tube wall is surrounded by the vapor-gas mixture under the normal temperature and pressure circumstances, when the wall temperature \( T_w \) is maintained at a value below the saturation temperature \( T_s \) of the vapor, condensation occurs and a condensate film forms on the wall surface and flows downward along the tube under the gravity. It is natural to regard the condensate film as a boundary layer. The transport of mass and heat in the vapor-gas mixture adjacent to the condensate film is activated by the condensation process and the region in which the transports occur can also be regarded as a boundary layer. The rate of condensation is controlled by the diffusion of the vapor through the vapor-gas film, thus the problem of convective condensation is treated as a coupled problem of heat and mass transfer. It is convenient to formulate the governing equations separately for the two boundary layers, and then to couple them by applying conditions of compatibility at the interface [17,26].

As shown in Fig. 1, the coordinates along and normal to the tube wall are \( x \) and \( y \), respectively. The behavior of heat and mass transfer for condensate boundary layer is governed by the mass, momentum, and energy equations [27], which can be respectively expressed as:

\[
\begin{align*}
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} &= 0 \\
\frac{u}{\partial x} + \frac{v}{\partial y} &= \frac{\partial^2 u}{\partial y^2} + g \sin \left( \frac{2\pi}{d} \right)
\end{align*}
\]  

(1)

\[
\begin{align*}
\frac{\partial T}{\partial x} + \frac{v}{\partial y} &= \frac{\partial^2 T}{\partial y^2} \left( 1 - \frac{\rho_w}{\rho} \right)
\end{align*}
\]  

(2)

And for the vapor-gas mixture boundary layer, equations of the mixture momentum, mixture energy, and noncondensable gas diffusion are respectively written as:

\[
\begin{align*}
\frac{\partial u}{\partial x} + \frac{v}{\partial y} &= \frac{\partial^2 u}{\partial y^2} + g \sin \left( \frac{2\pi}{d} \right) \left( \frac{1}{\rho} \right)
\end{align*}
\]  

(4)

\[
\begin{align*}
\frac{\partial T}{\partial x} + \frac{v}{\partial y} &= \frac{\partial^2 T}{\partial y^2} \left( \frac{1}{\rho c_p} \right)
\end{align*}
\]  

(5)

In the two-component mixture, it is convenient to represent local concentrations of the components in terms of their mass fractions: the local mass fraction of a noncondensable gas \( W_g \), and the vapor mass fraction \( W_v \) defined as [17]

\[
W_g = \frac{\rho_g}{\rho} W_v = \frac{\rho_v}{\rho}
\]  

(7a)

Where \( \rho \) is the local density of the mixture, and \( \rho_g \) and \( \rho_v \) are the local densities of the gas and the vapor, respectively. In the vapor-gas mixture with \( \rho = \rho_g + \rho_v \), it follows that

\[
W_g + W_v = 1
\]  

(7b)

In Eq. (4), the second term on the right side is the buoyancy force. Combining the ideal gas state equation \( p = \rho (R/M)T \) at a not high pressure, where \( R \) and \( M \) denote the local gas constant, and molecular weight of the mixture, respectively, the following equation can be obtained for the constant total pressure.

\[
\frac{\rho_w}{\rho} = \frac{M_w}{M} \frac{T}{T_w}
\]  

(8)

And combining the rule of the ideal mixture, \( 1/M = W_g/M_g + W_v/M_v \), the following correlation can be expressed as:

\[
1 - \frac{\rho_w}{\rho} = 1 - \frac{T [M_g - (M_g - M_v) W_g]}{T_w [M_g - (M_g - M_v) W_g + W_v]}
\]  

(9)

Setting the following variables:

\[
\Omega_W = \frac{M_g - M_v}{M_g - (M_g - M_v) W_g + W_v}, \quad \Omega_T = \frac{1}{T_w}
\]  

(10)

Substituting Eq. (10) into Eq. (4) yields the following equation:

\[
\begin{align*}
\frac{\partial u}{\partial x} + \frac{v}{\partial y} &= \frac{\partial^2 u}{\partial y^2} + \left[ \Omega_W (W_g - W_g) + \Omega_T (T_w - T) \right] g \sin \left( \frac{2\pi}{d} \right)
\end{align*}
\]  

(11)

And the boundary conditions for the governing equations are written as follows.

For \( y = 0 \), at the tube wall surface,

\[
u_1 = 0, v_1 = 0, T_1 = T_w
\]  

(12)

For \( y \to \infty \),

\[
u = 0, T = T_w = T_s, W_g = W_g
\]  

(13)

The mass flow rate through the interface of the two boundary layers by convection is defined as:

\[
\left[ \rho u \frac{d y}{d x} - \rho v \right]_{y = \delta}
\]  

(14)

Owing to the temperature and concentration gradients, a diffusive mass flux (per unit time and area) of the vapor and the gas can be respectively expressed as

\[
\begin{align*}
\dot{j}_v &= -\rho \frac{\partial W_g}{\partial y} - \rho \frac{\partial W_v}{\partial y} \dot{j}_g &= -\rho \frac{\partial W_g}{\partial y} - \rho \frac{\partial W_v}{\partial y}
\end{align*}
\]  

(15)

where \( D \) is the diffusion coefficient. From Eqs. (7a) and (7b), the following equation can be obtained.
\[ j_v + j_g = 0 \]

Therefore, the total mass flow rate through mixture-liquid interface is

\[ q_m = \left[ \frac{d \eta}{d x} \right] y_h - \eta \]

(17)

Owing to that the mixture-liquid interface is impermeable to a non-condensable gas, the non-condensable gas carried by the mixture to the interface with the same speed as before in order to maintain this steady-state. Therefore, the following equation can be obtained.

\[ \left\{ \left[ \rho_g \left( \frac{d \eta}{d x} \right) - \rho_g \psi \right] - j_g \right\}_i = 0 \]

(18)

Where \( i \) denotes the interface. From Eqs. (7a) and (7b), the following equation can be written as:

\[ \left( \rho B \frac{d W_g}{d \eta} \right)_i = -\left( \rho_g B \left( \frac{d \eta}{d x} - \frac{d \psi}{d x} \right) \right)_i = -\rho W_g \left( \frac{d \eta}{d x} - \frac{d \psi}{d x} \right)_i \]

\[ = -W_{g_i} q_m \]

(19)

Therefore, boundary conditions on the mixture-liquid interface are:

\[ y = \hat{y}_i, \quad u_h = u_i; \quad \left( \frac{\partial u}{\partial y} \right)_i = \left( \frac{\partial u}{\partial y} \right)_i, \quad T_h = T_i \]

(20)

\[ \left[ \rho \left( \frac{d \theta_i}{d x} - \frac{d \psi}{d x} \right) \right]_i = \left( \frac{\partial \theta}{\partial y} \right)_i \]

(21)

\[ \left( \frac{\partial T_i}{\partial y} \right)_i = \lambda_h q_m + \left( \frac{\partial T}{\partial y} \right)_i \]

(22)

\[ \left( \rho \frac{d W_g}{d \eta} \right)_i = -W_{g_i} q_m \]

(23)

Where \( \lambda_h \) denotes the latent heat. As the thickness of both the liquid film and vapor-gas mixture boundary layers is unknown, it is difficult to discretize the grid directly. Thus, the non-dimensional coordinates are introduced and defined as

\[ \eta = \frac{y}{\delta_i}; \quad \eta = \frac{y - \hat{y}_i}{\delta} \]

(24)

The dimensionless variables are introduced as follows:

\[ \eta_l = \frac{u_i d}{v}; \quad \eta_l = \frac{v d \hat{y}_i}{v_i} = \frac{T_l - T_w}{T_w} \]

\[ W_g = \frac{W_g - W_g_w}{W_g_w} \eta \]

\[ \bar{\theta} = \frac{T_v - T}{T_w} \theta; \quad \bar{y}_i = \frac{\theta_i}{\delta} \bar{\theta} = \frac{\delta_i}{\delta} \bar{\theta} = \frac{x}{d} \bar{y} = \frac{y}{d} \]

\[ \bar{u}_i = \frac{q_m d}{\mu_i} \bar{u}_m = \frac{q_m d}{\mu} \]

By introducing dimensionless stream functions \( \bar{\psi}_i \) and \( \bar{\psi} \) for the condensate and mixture, respectively, and from the relationship between the stream function and velocity, we have:

\[ \bar{\psi}_l = \frac{1}{\delta_i} \frac{\partial \bar{\psi}_i}{\partial \eta_l}; \quad \bar{\psi} = \frac{1}{\delta} \frac{\partial \bar{\psi}}{\partial \eta} \]

(26)

After transformation, Eqs. (2)–(6) can be rewritten as:

\[ \left( \frac{1}{\delta_1} \frac{\partial \bar{\psi}_i}{\partial \eta_1} \right) + \left( -1 \frac{\partial \bar{\psi}_i}{\partial \eta_1} \right) \frac{\partial \bar{\psi}_i}{\partial \eta_l} = \left( \frac{1}{\delta} \frac{\partial \bar{\psi}}{\partial \eta} \right) \frac{\partial^2 \bar{\psi}}{\partial \eta^2} + \frac{G_\delta \sin(2\xi)}{C_0} \]

(27)

\[ \left( \frac{1}{\delta_1} \frac{\partial \bar{\psi}_i}{\partial \eta_1} \right) + \left( -1 \frac{\partial \bar{\psi}_i}{\partial \eta_1} \right) \frac{\partial \bar{\psi}_i}{\partial \eta_l} = \left( \frac{1}{\delta} \frac{\partial \bar{\psi}}{\partial \eta} \right) \frac{\partial^2 \bar{\psi}}{\partial \eta^2} \]

(28)

\[ \left( \frac{1}{\delta_1} \frac{\partial \bar{\psi}_i}{\partial \eta_1} \right) + \left( -1 \frac{\partial \bar{\psi}_i}{\partial \eta_1} \right) \frac{\partial \bar{\psi}_i}{\partial \eta_l} = \left( \frac{1}{\delta} \frac{\partial \bar{\psi}}{\partial \eta} \right) \frac{\partial^2 \bar{\psi}}{\partial \eta^2} \]

(29)

In terms of the transformed variables, the corresponding boundary conditions are as follows.

For \( \eta_l = 0; \quad \eta_l = 0, \bar{\psi}_i = 0, \bar{\psi} = 0 \)

(32)

For \( \delta \to \infty \) and \( \eta = 1 : \quad \eta_l = 0, \bar{\psi}_i = 0, \bar{\psi} = 0 \)

(33)

For \( \eta_l = 1 \) and \( \eta = 0; \quad \bar{\psi}_i = \bar{\psi}, \bar{\psi} = 1 \)

(34)
Fig. 3. Iteration scheme flowchart.
\[ \tau_{ij} = \left( \frac{\mu_i \nu}{\delta} \frac{\partial \eta_{ij}}{\partial \xi} \right) = \tau_1 = \left( \frac{\mu \nu}{\delta} \frac{\partial \eta}{\partial \eta} \right) \]  \hspace{1cm} (35) \\

\[
\frac{1}{\beta} \left( \frac{\partial \bar{\eta}}{\partial \eta} \right)_i = \frac{Pr_{mi}}{H_i} \left( \frac{\lambda}{\lambda_i} \left( \frac{1}{\beta} \right) \left( \frac{\partial \bar{\eta}}{\partial \eta} \right)_i \right)
\]  \hspace{1cm} (36)

\[
\frac{1}{\beta} \left( \frac{\partial \bar{W}_g}{\partial \eta} \right)_i = -q_{in} Sc (\bar{W}_{gi} + 1)
\]  \hspace{1cm} (37)

At the top point of \( \beta = 0 \), boundary conditions are:

\[
\eta_i = 0, \eta = 0, \bar{\eta}_i = 0, \bar{\eta} = 0, \frac{\partial \bar{\eta}_i}{\partial \xi} = 0, \frac{\partial \bar{\eta}}{\partial \xi} = 0, \frac{\partial \bar{W}_g}{\partial \xi} = 0
\]  \hspace{1cm} (38)

3. The discrete equations

Eqs. (27)–(31) can be written in the unified form as:

\[
A \frac{d\phi}{d\xi} + B \frac{d^2\phi}{d\eta^2} + C \frac{d\phi}{d\eta^2} + D = 0
\]  \hspace{1cm} (39)

Eq. (39) is a steady-state non-linear parabolic equation, which can be calculated by the domain discretization after grid difference generation as shown in Fig. 2. The step lengths for \( \xi \), \( \eta \), and \( \eta \) in the simulation are all 1/50. The range of \( \xi \) is \( 0 - \pi/2 \) from the relationship between \( \xi \) and \( \pi \). The central difference scheme is employed for all the second derivative terms. And the convective terms are discretized using upwind difference scheme in order to maintain the transfer characteristics of the discrete equation. Convective terms in the direction of \( \xi \) are discretized by the first-order upwind difference scheme. And, the velocity component in the direction of \( \eta \) is larger than zero. The first-order derivative of convective terms in the direction of \( \eta \) is discretized by the second order upwind difference scheme. For the first internal discrete node in the direction of \( \eta \) (the equation of the liquid film) and the last internal one (the equation of the mixture), quadratic interpolation is employed. To ensure the computational accuracy and the stability of numerical process, all the terms of \( \eta \) are discretized using the fully implicit scheme. Thus, for each partial difference equation, a group of linear equations is obtained with a tridiagonal matrix as follows:

\[
A_{j-1} \phi_{j-1} + B_j \phi_j + C_j \phi_{j+1} = D_j
\]  \hspace{1cm} (40)

Fig. 3 shows the iteration process of the numerical simulation. Assuming the mixture-liquid interface temperature \( T_i \), from which can obtain the partial vapor pressure and density at the interface from the standard saturation vapor table. Since the total pressure is constant, the partial pressure of the noncondensable gas at the interface is derived by \( p_{gi} = p - p_v \), and then based on the known values of \( T_i \) and \( p_{gi} \), the noncondensable gas density \( \rho_{gi} \) can be obtained in terms of ideal gas rule. Consequently, the following equation can be induced:

\[
W_{gi} = \frac{p - p_{vi}}{p - (1 - M_v/M_g) p_{vi}}
\]  \hspace{1cm} (41)

4. Numerical results and discussion

4.1. Concentration distribution of the noncondensable gas

As shown in Figs. 4 and 5, the noncondensable gas increases from the bulk vapor-gas mixture to the interface of the two boundary layers as schematic in Fig. 1. When the vapor condensates, the noncondensable gas is also brought to the interface, however, it has been assumed that the interface is impermeable to the noncondensable gas. The noncondensable gas must return from the interface at the same speed to maintain the balance state and part of this movement is finished via diffusion. It is known that the diffusion process always occurs from the higher concentration to the lower. Thus the noncondensable gas concentration at the interface must keep a higher concentration. When a certain value reaches and a concentration gradient forms in the vapor-gas mixture film, the gas balance attains between the diffusion from the interface to the bulk vapor-gas mixture and the convection movement of the gas.

![Fig. 4. Noncondensable gas concentration distribution in vapor-gas mixture film at bulk noncondensable gas concentration of \( W_{gi} = 1\% \).](image-url)
Massive accumulation of the noncondensable gas is observed at the condensate–mixture interface, though the bulk mixture contains a small amount of noncondensable gas. In consequence, vapor partial pressure at the interface reduces sharply, and then the interface temperature $T_i$ compared to the main steam temperature is reduced largely. The behavior of condensation heat transfer will be greatly weakened by the reduced $\Delta T = T_i - T_w$. Meanwhile, the gas concentration at the interface increases with increasing $\Delta T$ at the same bulk noncondensable gas concentration. It is because that the larger $\Delta T$ will produce more condensate, which requires more mixture to participate in the condensation process and thus more noncondensable gas is taken to the interface. Comparing Fig. 5 with Fig. 4, we can see that the gas concentration at the interface is larger for a higher bulk concentration of the noncondensable gas. However, it is not so obvious at a larger $\Delta T$. For example, at the location of $\beta = 1.6$, when $\Delta T = 20$ K, the interface $W_{gi}$ increases slightly from 0.31 to 0.37 when the bulk $W_{g\infty}$ increases from 1% to 15%. However, at the same location when $\Delta T = 5$ K, $W_{gi}$ increases largely from 0.03 to 0.17 when $W_{g\infty}$ increases from 1% to 15%. Since at larger $\Delta T$, more noncondensable gas is brought to the interface and accumulates around the interface even though the bulk gas concentration is not high.
Fig. 6 presents average noncondensable gas concentration distribution at the interface. We can clearly see that the average noncondensable gas concentration at the interface increases as the temperature difference $\Delta T$ increases (in Fig. 6(a)) and the bulk gas concentration increases (in Fig. 6(b)).

4.2. Velocity distribution

The velocity distribution for different temperature differences of saturation temperature to wall temperature $\Delta T$ is shown in Figs. 7 and 8. We can see that the velocity of condensate film increases from the wall to the interface and it drops slightly close to the interface. The velocity of the vapor-gas mixture film decreases from the interface until the bulk mixture at rest. The vapor-gas mixture velocity at the interface is not zero since the flowing downwards along the wall of the condensate causes a drag to the vapor-gas film. Furthermore, this velocity increases as the increase of $\Delta T$ since more condensate will enhance the drag velocity. When the value of $\beta$ is less than $\pi/2$, i.e., on the upper half of the circumference, velocity in both the condensate film and vapor-gas boundary layer increases with increasing $\beta$. However, it decreases with the increasing value of $\beta$ when $\beta$ is larger than $\pi/2$. This is because that

**Fig. 7.** Velocity distribution for condensate and mixture at $W_{g,N} = 1\%$.

**Fig. 8.** Velocity distribution for condensate and mixture at $W_{g,N} = 15\%$. 
the condensate film flows downward due to the gravity, in which the force component in the direction of $\xi$ increases gradually when $\beta$ changes from 0 to $\pi/2$, resulting in the increasing velocity. Whereas, the gravity force component in the direction of $\xi$ decreases gradually in the range of $\beta > \pi/2$, which causes the velocity to drop. Comparing Fig. 8 with Fig. 7, we can see that at the same temperature difference $\Delta T$, the velocity of condensate film or vapor-gas film gets smaller when the bulk concentration of the
noncondensable gas is larger. This is because that the smaller condensation heat transfer coefficient due to a larger bulk concentration of the noncondensable gas results in a smaller flow rate of condensate.

4.3. Temperature distribution

As shown in Figs. 9 and 10 at two bulk concentrations of the noncondensable gas, the dimensionless condensate temperature keeps increasing from the wall to the interface and the dimensionless temperature of the mixture film also increases from the interface to the bulk mixture. In addition, the dimensionless temperature at the same location (the same value of $\beta$) decreases with the increase of $\Delta T$ when the bulk concentration of the noncondensable gas is small (1% shown in Fig. 9). However, it increases slightly with the increase of $\Delta T$ when the bulk concentration of the noncondensable gas is large (15% shown in Fig. 10). At lower $W_{g\infty}$, the increase of $\Delta T$ leads to the sharp rise of the local noncondensable gas concentration and thus the drop of the partial vapor pressure. Consequently, the corresponding local saturation temperature drops. However, at larger $W_{g\infty}$, the rise of local noncondensable gas concentration due to the increase of $\Delta T$ is not so obvious. Compared to Fig. 9, Fig. 10 shows that local temperature at each location at the same $\Delta T$ is lower at a larger bulk noncondensable gas concentration, which is caused by the rise of the local noncondensable gas concentration. Fig. 11 presents the average temperature distribution at the condensate–mixture interface by integral calculation at twelve bulk noncondensable gas concentrations ($W_{g\infty} = 15, 12, 10, 8, 6, 5, 4, 3, 2, 1, 0.5\%$) and six temperature differences ($\Delta T = 20, 15, 12, 10, 8, 5\ K$). As shown in Fig. 11, the average temperature at the interface decreases with the increasing $\Delta T$ and $W_{g\infty}$, which is contrary to the average

![Fig. 11. Average temperature distribution at the condensate–mixture interface.](image)

![Fig. 12. Average condensation heat transfer coefficient.](image)
noncondensable gas concentration distribution at the interface shown in Fig. 6.

4.4. Condensation heat transfer coefficient

Fig. 12 compares the distribution of the average condensation heat transfer coefficient. We can see that although the bulk mixture contains a small quantity of the noncondensable gas, condensation heat transfer coefficient decreases obviously. From Fig. 12, condensation heat transfer coefficient decreases with increasing temperature difference $\Delta T$, when the noncondensable gas is in a small proportion of the bulk mixture. Nevertheless, when at a large noncondensable proportion, heat transfer coefficient shows a trend of increasing first, after reaching its maximum, decreases with the increasing temperature difference $\Delta T$. The average heat transfer coefficient against the temperature difference is given in Fig. 13(a), which shows comparisons of present numerical results and experimental data from the literature [28]. It can be found that the maximum deviation is less than 28%. The average heat transfer coefficient against the mass flow rate of the cooling is shown in Fig. 13(b), in which the present numerical results and experimental data in the literature [29] are compared and the deviation is within $\pm$23%.

5. Conclusions

The physical and mathematical models of the film condensation heat transfer are developed by simultaneously solving the equations of mass, momentum, energy and diffusion to simulate condensation heat transfer outside a horizontal tube in the presence of a noncondensable gas. Particularly, the method of dimensionless transformation, which is the key to solve the equations, is employed in the case of unknown boundary layer thickness.

Local distributions of the noncondensable gas concentration, velocity and temperature in the condensate liquid boundary layer and vapor-gas mixture boundary layer, and the mixture–condensate interface properties have been obtained using the finite difference method. And the heat transfer coefficient against temperature difference of saturation temperature to wall temperature, and against the noncondensable gas concentration of bulk mixture have also been presented. Due to the presence of the noncondensable gas, the performance of condensation heat transfer deteriorates dramatically, although the mass concentration of the noncondensable gas is very small. The predicted average condensation heat transfer coefficients agree well with the available experimental data.

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