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Theoretical and Numerical Analysis of the 3D-Distributed Thermodynamic Properties in Inclined Condenser Tubes

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Steam condensation in tubes exhibits obvious thermodynamic non-equilibrium characteristics, which leads to many problems, such as the decline of condenser efficiency and the freezing crack of condenser under low temperature. Limited by the tube size and the complexity of the condensation process, previous studies were mostly conducted with lumped parameter or one-dimensional model. As a result, the basic affecting mechanism of the problems existing in the application of condensers has not been revealed. This paper introduces developed by authors calculation method, which can quickly solve the coupled control equations involved in a three-dimensional (3D) steam condensation process. As the method was adopted to study the typical thermodynamic process of the steam condensation in an inclined tube, the 3D distributions of condensation heat transfer coefficient (HTC) and liquid film thickness in the whole inclined circular tube are obtained. Based on a comparative analysis together with the results of the referenced lumped method, it is found that the liquid film is the main source of the thermal resistance during a condensation process, and its variation along the circumferential and axial directions leads to the variations of HTC correspondingly. These fundamental mechanisms may explain the complex thermodynamic processes like freezing crack of condensers under low-temperature environment. The results obtained in this paper were verified by well-accepted 2D-distributed model of HTC and relevant experimental data with high accuracy.

1. Introduction

The research demand for condensation process was increasing, and this is related to the considerable use of power, refrigeration equipment and industrial production. However, it is difficult to carry out experimental studies on condensation inside pipes with a narrow tube space, and the complex multiphase flow and heat transfer process. The inconveniences of experimental research result in immature research on the condensation process in tube.

Nusselt (Yang and Tao, 2006) was the first to analyse film condensation theoretically and summarize the calculation formula of condensation heat transfer coefficient (HTC). Chato (1960) put forward the calculation formula of condensation heat transfer coefficient suitable for Re<35,000. Based on a great quantity of previous experimental data, Shah (1979) fitted the calculation correlation which can be applied to a variety of working fluids. Through the comparison of the experimental data of condensation HTC in the tube at different inclined angles, the calculation formula of condensation HTC in inclined tubes based on the condensation HTC in the horizontal tube is proposed by Würfel (2003).

Because of the narrow space inside the tubes and the tendency for the measuring points to destroy the film condensate, the current experimental research only focuses on the evaluation of the overall efficiency of the condenser, the one-dimensional distribution of the HTC along the axial or lumped parameter model of the condensation heat transfer process in the tube. Almost all experimental studies have adopted the same experimental equipment as the experiment performed by Rifert (2020). The lack of experimental data causes the inability to perform the theoretical analysis. As a result, the 3D distribution of thermodynamic parameters of condensation in tubes has been neglected.

However, in practical engineering applications, such as direct air-cooling power plants, the phenomenon of freezing and cracking often occurs at the bottom of the tube due to the uneven distribution of thermodynamic

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parameters. Based on the situation of present study, the condensation phenomenon of steam in tube is analysed theoretically, and the 3D distribution characteristics of thermodynamic parameters are discussed.
In this study, steam condenses and equations are solved numerically on PC with developed software, the 3D distribution characteristics of liquid film thickness, HTC and heat flux are finally obtained. Considering the heat and mass transfer of the condensate layer, the distribution of the complete liquid film thickness along the circumferential direction is calculated. Through the numerical calculations, it is found that the 3D distribution of heat transfer characteristics is more uniform with the increase of the tilt angle, but when β = 90 °, the HTC reaches the minimum. Paper points out that the uniformity of a thermodynamic parameter distribution has a significant impact on the safety and performance of the condenser in practical applications and the uniformity needs to be considered when selecting the optimum inclination angle.

2. Mathematical modeling

The physical model of steam condense inside inclined tubes is shown in Figure 1. Due to the difference in temperature between the saturated steam and the wall, steam condenses onto the wall and forms condensate films. Under the action of gravity, the condensate converges along the inner periphery of the tube wall to the direction of gravity and shapes the condensate layer.

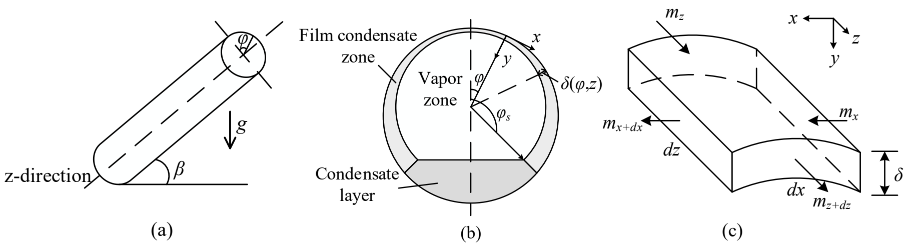

Figure 1: Physical modelling of stratified condensation in an inclined tube: (a) illustration of an inclined tube, (b) distribution of condensate inside the tube and (c) element of the condensate film inside the tube

To simplify the analysis, the following assumptions are made based on the classic Nusselt theory:

- Lose sight of the acceleration term in the momentum equation.
- The temperature gradient in the condensate is linear.
- The main flow pattern of the multiphase flow is stratified flow, and the condensate is smooth without fluctuation.
- The flow direction of condensate is only axial and tangential, and there is no radial flow.
- There is no shear force between the two phases.
- Ignoring the pressure drop, the steam is always saturated.
- The convective heat transfer between the condensate and the wall is approximated as heat conduction.

2.1 Condensate film flow analysis, $0<\varphi<\varphi_s$

Based on the above assumptions, the momentum conservation equations in x-direction and z-direction can be obtained by performing conservation analysis on the micro element in Figure 1c, as shown in Eq(1) and Eq(2):

$$\eta_l \frac{\partial^2 u_x}{\partial y^2} + g(\rho_l - \rho_v) sin\varphi cos\beta = 0 \quad (1)$$

$$\eta_l \frac{\partial^2 u_z}{\partial y^2} + g(\rho_l - \rho_v) sin\beta = 0 \quad (2)$$

To complete the formulation, the following boundary conditions are introduced:

$$\text{at } y = 0: u_x = u_z = 0 \quad (3)$$

$$\text{at } y = \delta: \frac{\partial u_x}{\partial y} = \frac{\partial u_z}{\partial y} = 0 \quad (4)$$

Integrating Eq(1) and Eq(2) across the film thickness with boundary conditions Eq(3) and Eq(4) for the velocity components in the axial and tangential directions:

$$u_x = \frac{g(\rho_l - \rho_v) sin\varphi cos\beta}{\eta_l} (\delta y - \frac{y^2}{2}) \quad (5)$$

$$u_z = \frac{g(\rho_l - \rho_v)sin\beta}{\eta_l}(\delta y - \frac{y^2}{2}) \tag{6}$$

The average velocities in x-direction and z-direction can be obtained by integrating and averaging the Eq(5) and Eq(6) in [0, δ], as shown below:

$$u_{xm} = \frac{g(\rho_l - \rho_v)sin\varphi cos\beta}{3\eta_l}\delta^2 \tag{7}$$

$$u_{zm} = \frac{g(\rho_l - \rho_v)sin\beta}{3\eta_l}\delta^2 \tag{8}$$

In the calculation, the whole circle is divided into 36,000 micro-elements in the circumferential direction, so the curvature can be ignored in the subsequent calculation and the mass flow rate of condensate in the x-direction and z-direction can be expressed as follows:

$$m_x = u_{xm}\rho_l\delta dx \tag{9}$$

$$m_z = u_{zm}\rho_l\delta dz \tag{10}$$

According to the previous hypothesis, the heat released by steam condensation is transferred to the wall through the heat conduction of condensate film and the energy conservation equation is described as:

$$\frac{\lambda_l(T_{sat} - T_{wall})dxdz}{\delta} = m_{con}h_{fg} \tag{11}$$

According to Eq(9), Eq(10) and Eq(11), the mass conservation equation can be rewritten as:

$$\frac{g\rho_l(\rho_l - \rho_v)}{3\eta_l}[sin\varphi cos\beta dx(\delta(i,j)^3 - \delta(i-1,j)^3) + sin\beta dz(\delta(i,j)^3 - \delta(i,j-1)^3)] = \frac{\lambda_l \Delta T dxdz}{h_{fg}\delta(i,j)} \tag{12}$$

Before starting the calculation, the following conditions need to be added:

- at $z = 0$: $\delta = 0$.
- at $\varphi = 0$: $\frac{\partial\delta}{\partial\varphi} = 0$ (Fiedler and Auracher, 2004).
- $dx = 2\pi R/360{,}00$.

2.2 Condensate layer flow analysis, $\varphi_s < \varphi < \pi$

Due to the effect of gravity, the condensate will converge along the wall towards the bottom of the tube and form the condensate layer. In order to calculate the condensate film thickness and other parameters of the condensate layer, approximate it to the smooth flow pattern without fluctuation. The condensate film thickness at each discrete point in the condensate layer portion can be obtained using the geometric method.

2.3 Heat transfer coefficient

With the calculated thickness of the condensate, the local heat transfer coefficient is obtained by Eq(13) (Kekaula and Chen, 2020). Similarly, the average HTC of the entire tube and the one-dimensional distribution of HTC along the z-direction can also be calculated.

$$\alpha(i,j) = \frac{\lambda_l}{\delta(i,j)} \tag{13}$$

2.4 Solution

The numerical calculation is based on the finite element method, hence the first step is discretize the condenser in space. The values of n are selected as 360, 3,600 and 36,000 for calculation. The results show that when n equal to 36,000, the value of φ_s can be accurately captured. The initiated parameters required in the calculation process are the temperature and mass flow rate of the saturated steam, the temperature of the tube wall, the inclined angle, total tube length and inner radius of the tube. The calculation procedure is explained in the following steps:

(1) Discretize the condenser in the circumferential and axial directions respectively. In this study, $d\varphi$ = 0.01 ° and dz = 0.15 m. Therefore, there are 1 ≤ i ≤ 36,001 and 1 ≤ j ≤ 28.
(2) Calculate the thermal and physical properties of the saturated steam and water by the initial parameters.
(3) Assuming φ_s = 180 ° at j = 2 (i.e. z = 0.15 m), calculate the circumferential distribution of condense film thickness at j = 2 by Eq(12).
(4) Calculate the vertical distance from the two-phases interface to the bottom of the tube at each discrete point.
(5) Iterate the φ_s until the calculated film thickness distribution can meet the mass conservation and the simplification of condensate layer.

(6) Make j = j + 1 and assign $\varphi_s(j-1)$ to $\varphi_s(j)$, so as to replace the assumption in step 3, and repeat steps 3-5 until z equals to the length of the tube.

3. Results and discussion

The model calculation results are compared with the experimental data from Ye (2014) and the data calculated with empirical formulas from Chato (1998). The comparison results are shown in Figure 2. The reliability of the model is verified by comparing the experimental data with the classical empirical formulas. The variation trend of the HTC calculated by this study along the z-direction is basically the same as the previous research data. Compared with the data calculated by empirical formulas from Chato, the average errors are 7.1 %. As for compared with the experimental data from Ye, the average error is 13.4 %. As such, conclude that this model is reliable.

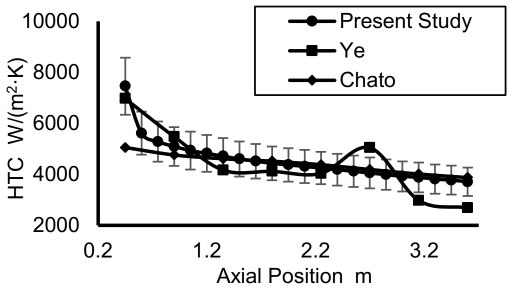

Figure 2: Compared with the data of the experimental data from Ye (2014) and calculated by empirical formulas from Chato (1998)

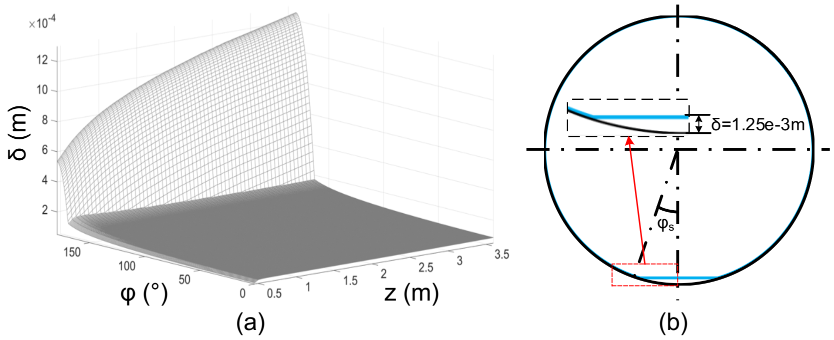

Figure 3: Distribution of condensation film: (a) 3D distribution of condensation film thickness when β = 30 ° in z∈[0.45,3.6] and (b) the condensate film thickness distribution at z = 3.6 m

Figure 3a shows the 3D distribution of condensate film thickness formed by steam condense inside tubes with β = 30 °. For the film condensate zone ($0 \le \varphi \le \varphi_s$), the condensate film thickness only changes dramatically near the entrance, and then it is basically stable. The main changes are concentrated in the condensate layer ($\varphi_s \le \varphi \le 180$ °). The condensate film is the main source of heat resistance in the heat transfer process on the steam side of the steam condense inside pipes and the HTC is inversely proportional to the thickness of the condensate film.

Figure 3b is the distribution diagram of liquid film thickness at z = 3.6 m drawn according to the calculation results. The calculation conditions are β = 30 °, T_{sat} = 354.15 K and R = 0.025 m. According to the figure, it can be clearly seen that the thickness of the condensate film at the top of the pipe is the thinnest and thickens continuously along the circumferential direction.

The conclusion of $\delta_{cl} \gg \delta_{fc}$ can be clearly obtained from Figure 3b, which is also reflects that the HTC at the condensate layer is the smallest in the circumferential direction. In reality, the wall temperature has a gradient in the circumferential direction. The bottom of the pipe is the area with the lowest wall temperature, and the temperature difference between the place with the highest temperature (at the top of the tube) is between 1 and 4 K (Du and Wang, 2001). Up to now, almost all experimental studies have adopted equipment similar to those used by Feng et al. (2020), and studies conducted with such equipment have shown that this impact can be ignored. However, in many application scenarios, local shock convection would lead to more uneven

temperature distribution on the tube wall of condensers. In such cases, ignoring the wall undercooling will lead to a significant decline in safety and efficiency.
Keeping the initial parameters unchanged and only changing the inclined angle, the calculated results with $\Delta T = 10$ K are shown in Figure 4. Figure 4 shows the variation in condensate film thickness at z = 4.05 m along the tube periphery with different tilt angles. With an increase in the inclined angle, the proportion of condensate layer area gradually decreases.

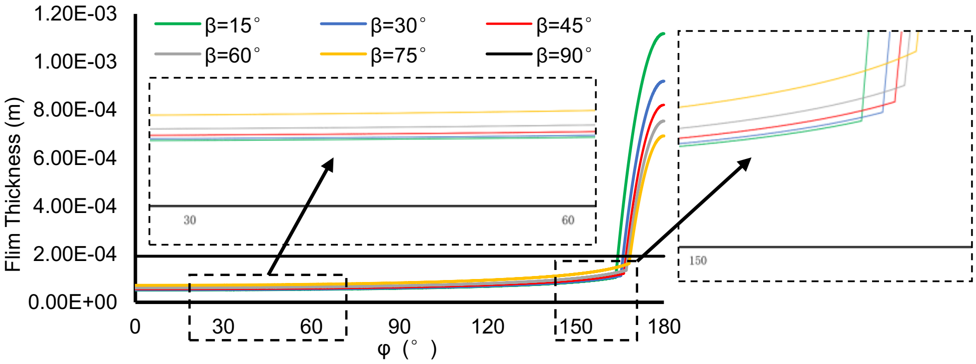

Figure 4: Variation along the tube periphery of the condensate film thickness with different inclination angles at z = 4.05 m

The condensation heat transfer coefficient was normalized as shown in Eq. 14, and the nonuniformity of condensation heat transfer coefficient in circumferential distribution can be well reflected by the ratio of maximum and minimum condensation heat transfer coefficient.

$$\alpha_{nd}(j) = \frac{\alpha(1,j)}{\alpha(n+1,j)} \tag{14}$$

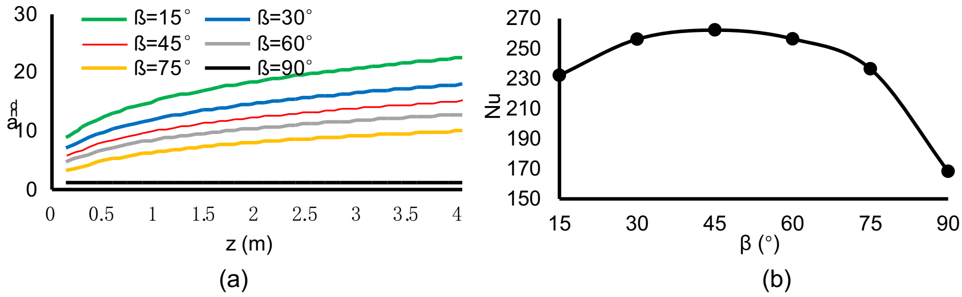

Figure 5: Analytic results: (a) the axial variation of α_{nd} calculated with different tilt angles and (b) the mean Nu for entire tube in different inclined angles

As shown in Figure 5a, the axial variation of α_{nd} under different inclination angles is shown. With the increase in inclination angle, the distribution of condensation heat transfer coefficient in the tube is more uniform. The non-equilibrium characteristic of the condensation heat transfer coefficient is closely related to the distribution of wall temperature mentioned above. Combining the two, this imbalance would have a significant impact on the performance of the condenser.
However, blindly pursuing the better equilibrium characteristic would also lead to a significant decline in the heat exchange capacity of the condenser. Figure 5b shows the variation of condensation heat transfer coefficient under different inclination angles. When only considering the heat transfer capacity, the optimal inclination angle is between 30 - 60°. This means that there is a certain contradiction between the heat exchange capacity and the equilibrium of the thermodynamic parameter distribution of the condenser, which has not been paid attention to when selecting the best inclination angle in the past.

4. Conclusions

In this study, the film condensation of steam inside pipes is investigated. Based on the classical Nusselt theory, the governing equations are solved, and the phenomenon of the condensation in the tube is numerically calculated by programming using the basic idea of micro element method. Through the research results, it can be clearly found that the distribution of thermodynamic parameters in the condensation process is uneven. From the results, the following conclusions are drawn:

- With the increasing of the inclined angle, the proportion of condensate layer area decreases gradually.
- The greater the tilt angle, the more balance distribution of thermodynamic parameters in the tube.
- In practical application, the heat transfer capacity and uniformity of the condenser need to be considered together.
- The calculation results obtained by the model are in good agreement with the experimental data and the data calculated by the mature empirical formula.

This study still has the same deficiency as the existing theoretical research, that is, the wall boundary conditions are being artificially assumed to be insufficient. If more reasonable wall boundary conditions can be calculated according to the external heat transfer conditions, the accuracy of the research results of in-tube condensation will be greatly improved, and the analysis of the problems of condensers in practical engineering applications will be improved.

Nomenclature

g – gravity, m/s^2
h_{fg} – specific enthalpy of vaporization, J/kg
m – mass flow rate, kg/s
n – number of circumferential discrete points, -
Nu – Nusselt number, -
T – temperature, K
u – velocity, m/s
x,y,z – coordinate, -
ΔT – temperature difference ($=T_{sat} - T_{wall}$), K

Greek symbols
α – heat transfer coefficient, $W/(m^2 \cdot K)$
β – inclination angle, °
δ – film condensate thickness, m
η – dynamic viscosity, kg/(m·s)
λ – thermal conductivity, W/(m·K)
ρ – density, kg/m^3
φ – peripheral angle, °

Subscripts and superscripts
bot – bottom of the tube
con – condensate
cl – condensate layer
fc – film condensate
l – liquid
m – mean
nd – normalized condensation heat transfer coefficient
sat – saturation
top – top of the tube
v – vapor
wall – the tube wall

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