Numerical study on condensation of steam/air mixture inside corrugated low finned tubes

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Abstract. Corrugated low finned tubes are the special spirally corrugated tubes, which can enhance the condensation heat transfer with and without noncondensable gas. In this paper, a numerical model on the basis of diffusion layer theory was established to investigate the local heat transfer characteristic inside corrugated low finned tubes. The local parameters contain bulk temperatures, gas mass fraction, heat transfer coefficient and heat flux. The results showed that the bulk temperatures decreased along the condenser tube while the noncondensable gas mass fraction displayed the opposite trend. The local heat transfer coefficient and heat flux decrease along the condenser tube especially near the outlet. Of the five enhanced tubes, #5 tube had the best overall heat transfer performance, but it also had the fastest rate of decline as condensation progressed, indicating that the rib was the main influence factor compared with pitch.

1. Introduction
According to Bergles’s perspective, the heat transfer enhancement techniques can be classified either as passive, which require no direct application of external power, or as active, which require external power. And the enhancement techniques have been developed to the third generation [1]. The common used enhanced tubes for in-tube condensation are micro-fin tubes and corrugated tubes [2]. Corrugated low finned tube (CLFT) is a kind of spirally corrugated tube with trapezoidal profile. Due to the wider spiral protrusion, its heat transfer performance is better than that of ordinary spirally corrugated tube.

Ren et al experimentally studied the condensation heat transfer and flow resistance characteristics inside corrugated low finned tubes with/without noncondensable gas [3, 4]. It was found that the heat transfer performance inside corrugated low finned tubes was greater than that inside smooth tubes even when the noncondensable gas existed. However, because of the difficulty in measuring wall temperature, the mechanism of condensation inside corrugated low finned tubes with noncondensable gas has not been understood thoroughly.

In this study, a theoretical model on the basis of diffusion layer theory was established. The theoretical model didn’t need constant wall temperature or heat flux as the solving condition. The wall temperature distribution and outlet temperature of cooling water could be evaluated by coupling the in-tube condensation and out-tube forced convection flow. The profiles of these parameters along the condenser tube was showed, including bulk temperature, noncondensable gas mass fraction, heat...
transfer coefficient and heat transfer flux. The structural parameters of these tubes and the inlet conditions can be obtained from Ren’s work [3].

2. Numerical model

2.1 Basic principle

The numerical model utilizes the diffusion layer method to study the condensation characteristic of steam/air mixture. The diffusion layer model is on the basis of heat balance at the liquid–air interface, in which the heat transferred from the steam–air boundary layer is equal to the heat transferred through the condensate film [5]. The heat flux transferred from the bulk mixture to the interface can be described as the sum of sensible and latent heat fluxes:

\[
q = q_s + q_{cd} = h_i(T_b - T_i) + m_s^n h_{fg}
\]

(1)

In the gas region, the latent heat transfer coefficient can be expressed as:

\[
h_{cd} = \frac{m_s^n h_{fg}}{(T_b - T_i)}
\]

(2)

So the heat flux from the gas mixture to the interface can be described as:

\[
q = h_s(T_b - T_i) + h_{cd}(T_b - T_i) = (h_s + h_{cd})(T_b - T_i)
\]

(3)

The heat flux is equal to that transferred through the liquid film:

\[
q = h_f(T_i - T_w)
\]

(4)

The overall condensation heat transfer coefficient can be expressed as:

\[
h_c = \frac{q}{(T_b - T_w)}
\]

(5)

Substitute Eq. (3) and Eq. (4) into Eq. (5), the overall condensation heat transfer coefficient is described as:

\[
h_c = \left[\frac{1}{h_s} + \frac{1}{h_{cd}} + \frac{1}{h_f}\right]^{-1}
\]

(6)

From Eq. (6), it is found that, to get the overall condensation heat transfer coefficient \(h_c\), the sensible heat transfer coefficient \(h_s\), the gas region condensation heat transfer coefficient \(h_{cd}\) and the film heat transfer coefficient \(h_f\) must be firstly calculated.

The sensible heat transfer coefficient \(h_s\) can be obtained from:

\[
h_s = \frac{Nu \cdot k_i}{d_i}
\]

(7)

Here, the Nusselt number can be calculated from:

\[
Nu = 1.121 \cdot Re^{0.638} \cdot Pr^{0.33} \cdot \left(\frac{P}{d_i}\right)^{-0.239} \cdot \left(\frac{e}{d_i}\right)^{0.461}
\]

(8)

And the gas region condensation heat coefficient can be obtained from [6]:

\[
h_{cd} = \frac{Sh \cdot k_i}{d_i}
\]

(9)
Here, the Sherwood number can be get from the same formulation showed in Eq. (8) by substituting the Prandtl number with the Schmidt number. And $k_c$ is the condensation thermal conductivity defined as:

$$k_c = \frac{1}{\phi} \left( \frac{h_{fg}^2 p v M^2}{R^2 T^3} \right)$$

(10)

For the annular flow, the simplest method to calculate heat transfer coefficient is the two-phase multiplier approach, where the formula can be expressed as the two-phase multiplier form:

$$h_f = 0.096 \cdot \text{Re}_{\text{eq}}^{0.83} \cdot Pr^{0.33} \cdot (\frac{d_i}{d_j})^{-0.239} \cdot (\frac{D}{d_j})^{0.668} \cdot (\frac{\lambda}{d_j})$$

(11)

2.2 Calculation procedure

The numerical model requests inlet parameters as the solving condition. The inlet parameters contain steam flow rate ($\dot{m}_{s,i}$), air flow rate ($\dot{m}_{g,i}$), gas mixture temperature ($T_{s,i,n}$), gas mixture pressure ($p_i$), coolant flow rate ($\dot{m}_{co,i}$) and coolant temperature ($T_{co,i,n}$). The calculation procedure consists of the following steps.

Step 1: Obtain the structure parameters of condenser tube and inlet parameters of gas mixture and coolant.

Step 2: Assume the outlet temperature of coolant $T_{co,out}$, which corresponds to the boundary condition at the inlet of condenser tube.

Step 3: Divide the condenser tube into $n$ control cells along the axial direction, using the converging criterion of 0.05 m.

Step 4: Assume the heat transfer rate $Q$ of each control part represented by "$i$". Calculate physical property parameters and flow conditions such as quality, Reynolds number etc.

Step 5: Determine the interface temperature $T_i$. This step is made up of three steps:

Step 5-1: Estimate the interface temperature $T_i$ and the condensation mass flux $m_c$.

Step 5-2: Obtain the sensible and condensation heat transfer coefficients in the gas region from Eq. (7) and Eq. (9).

Step 5-3: Obtain the heat transfer rate $Q_g$ from Eq. (12). Calculate the relative error between $Q_g$ and $Q$ in step 4. If this value exceeds a threshold (0.05), adjust the assumed $T_i$ until the relative error is less than the threshold.

$$Q_g = (h_i + h_{co}) \cdot (T_{i} - T_i) \cdot (\pi \cdot d_{in} \cdot L/n)$$

(12)

Step 6: Obtain the heat transfer rate $Q_f$ through the liquid film from Eq. (13). The heat transfer coefficient $h_i$ and inner wall temperature $T_{w,in}$ can be respectively obtained from Eq. (11) and Eq. (14). Calculate the relative error between $Q_f$ and $Q$ in step 4. If this value exceeds a threshold (0.05), adjust the assumed heat transfer rate $Q$ in step 4, until the relative error is less than the threshold.

$$Q_f = h_i \cdot (T_i - T_{w,in}) \cdot (\pi d_{in} \cdot L/n)$$

(13)

$$T_{w,in} = T_{co} + \frac{Q / (\pi \cdot d_{in} \cdot L/n)}{h_{co} + 1 / R_w}$$

(14)

Step 7: Calculate the steam mass flow rate at the outlet of this control part from Eq. (15). If the mass flow rate is smaller than zero, the heat transfer mode transforms into forced convection. Otherwise go to step 8. Calculate the coolant temperature at outlet of current control part from Eq. (16)

$$\dot{m}_i (i + 1) = \dot{m}_i (i) - Q / h_{fg}$$

(15)
\[ T_{co}(i + 1) = T_{co}(i) - \frac{Q}{(c_{p,co} \cdot \dot{m}_{co})} \]  

(16)

Step 8: If \( i < n \), repeat step 2 to step 7 for the rest control parts. Otherwise go to step 9.

Step 9: Compare the calculated inlet temperature of coolant with the measured value. If the gap exceeds 0.1°C, adjust the assumed outlet temperature of coolant in step 2 until the temperature gap is less than 0.1°C.

3. Results and discussion

Figure 1 shows the distribution of steam/air bulk temperatures along the corrugated low finned tubes at the same inlet condition. Obviously, the bulk temperatures decrease along the condenser tube. Because as the condensation proceeds, the steam flow rate decreases, leading to the reduction of steam pressure and saturated temperature. In this figure, Tube #1, #2 and #3 have the same rib height but different pitch and Tube #2, #4 and #5 have the same pitch height but different rib. It can be found that the rate of descent for tube #5 is the fastest, followed by tube #1, #2, #4 and #3.

Figure 1. Variation of bulk temperature along the CLFTs

Figure 2 displays the variation of noncondensable gas mass fraction along the corrugated low finned tubes. The variation tendency of noncondensable gas mass fraction is opposite to that of bulk temperature. The noncondensable gas mass fraction increases along the condenser tube because of the reduction of steam flow rate. The growth rate for tube #5 is the highest, followed by tube #1, #2, #4...
and #3. This indicates that #5 tube has the best heat transfer performance during pure steam condensation, which is consistent with the results in Ren’s work [4].

Figure 3 displays the distribution of condensation heat transfer coefficient along the corrugated low finned tubes. Obviously, the heat transfer coefficient decreases along the condenser tube, due to the decrease of gas mixture Reynolds number and the increase of condensate film thickness and local noncondensable fraction. The heat transfer coefficient of #5 tube is the largest at the beginning of condenser tube. But as the condensation progresses, the heat transfer coefficient decreases rapidly, and at the outlet of condenser tube it is even smaller than that of #1, #2 and #3 tubes.

Figure 4 illustrates the variation of local heat flux along the corrugated low finned tubes. As expected, the local heat flux of #5 tube is the highest at the entrance. This indicates that the rib is the main factor to enhance condensation heat transfer even in the presence of noncondensable gas. It can also be found that the heat flux deteriorates remarkably near the outlet of condenser tube. This is caused by two factors, one is the decrease of heat transfer coefficient and the other is the reduction of temperature difference of heat transfer.

4. Conclusions
In this study, a numerical model on the basis of diffusion layer theory was established. The distribution of these parameters along the condenser tube was analyzed, including bulk temperatures, gas mass fraction, heat transfer coefficient and heat flux. The results showed that the bulk
temperatures decreased along the condenser tube while the noncondensable gas mass fraction displayed the opposite trend. The local heat transfer coefficient and heat flux decrease along the condenser tube especially near the outlet. Among the five enhanced tubes, #5 tube had the best overall heat transfer performance, but it also had the fastest rate of decline as condensation progressed, indicating that the rib was the main influence factor compared with pitch.

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