Mechanism of Condensation Enhancement by Corrugated Low Finned Tubes in Presence of Noncondensable Gas

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Abstract. Corrugated tubes are the most convenient reinforcement techniques to enhance in-tube condensation. In this paper, the average condensation heat transfer coefficients inside the corrugated low finned tube and smooth tube were firstly introduced. The distribution of local heat transfer coefficient was presented at lower and higher inlet noncondensable gas mass fractions. The results show that the major heat transfer resistance is the thermal resistance of liquid film at lower inlet gas fraction. So the mechanism of condensation enhancement by corrugated low finned tubes is the increase of liquid film condensation coefficient. But at higher inlet gas fraction gas layer mass transfer resistance becomes the control thermal resistance. The spiral ribs inside corrugated low finned tube can thinner not only the gas layer thickness but also the liquid film thickness. The mechanism of condensation enhancement is the increase of both gas layer and liquid film condensation coefficient.

1. Introduction
Condensers using in-tube condensation with air-cooled or water-cooled are widely used in refrigeration, air-conditioning, power, and process industries. With the break of energy crisis, the heat transfer enhancement is utilized to reduce heat transfer surface area, reduce temperature difference, increase heat transfer rate and reduce pumping power. Micro-fin tubes, micro-channel tubes, corrugated tubes and tube with wire insert are the commonly passive enhancement techniques for in-tube condensation [1]. Corrugated tubes are the most convenient reinforcement techniques to apply.

Laohalertdecha and Wongwises investigated the effect of pitch on the condensation heat transfer coefficient and pressure drop of R-134a inside horizontal corrugated tubes [2]. It was found that the maximum heat transfer coefficient ratio of the corrugated tube was up to 50% higher than that of the smooth tube. Li et al studied the condensation pressure drop and heat transfer coefficient of R-404A flowing through the corrugated tubes [3]. The results showed that using corrugated tube increased the heat transfer coefficient up to 59% above the plain tube values. Ren et al studied the characteristics of condensation and flow resistance inside horizontal corrugated low finned tubes [4]. They found that the maximum performance evaluation factor was 2.24 at the minimum vapour quality and mass flux.

It is well known that the presence of a small amount of noncondensable gas significantly reduces the performance of condensation [5]. Wu and Vierow experimentally studied the heat transfer and fluid flow phenomena in a horizontal condenser tube [6]. It was found that the condensation heat transfer coefficients on the tube top were much greater than the values at the bottom near the inlet of
condenser tube. Ren et al [7] studied the effect of noncondensable gas on condensation heat transfer inside horizontal tubes. They found that the overall heat transfer coefficient decreased with the increase of inlet noncondensable gas fraction and the decrease of inlet mass flux. And two correlations were developed respectively for stratified flow and annular flow regime.

However, there are few researchers who have studied the condensation phenomenon in enhanced tube in presence of noncondensable gas. Fan et al investigated the natural convection condensation heat transfer outside the vertical corrugated tube [8]. It was found that the corrugated tube showed a slight enhancement with an average increase of heat transfer coefficient about 10%. Ren et al compared the condensation heat transfer performance in corrugated low finned tubes and that in a smooth tube [9, 10]. The results revealed that corrugated low finned tube could weaken the influence of noncondensable gas. But the enhancement mechanism had not been quantitatively analysed.

In this paper, the mechanism of condensation enhancement by corrugated low finned tubes in presence of noncondensable gas was investigated using diffusion layer model. The average condensation heat transfer coefficients inside the enhanced tube and smooth tube were firstly introduced. Then the distribution of local heat transfer coefficient was described along the tube and the enhancement mechanism at different noncondensable gas mass fractions was revealed.

2. Experimental facility and numerical model

2.1 Experimental facility

The experimental facility in Ren’s paper [9], including steam/air loop, the cooling water circuit, the test section and the data acquisition system was used to study the in-tube average condensation performance. Only the inlet and outlet temperatures of the gas mixture and the coolant needed be measured by sheathed thermocouples. The test section was a double pipe heat exchanger, in which the gas mixture condensed inside the inner tube while the cooling water flowed in the annular channel. One corrugated low finned tube and one smooth tube were utilized as the inner tube. They were made of stainless steel with the outer diameter of 19mm, the thickness of 2mm and the length of 1750 mm. The enhanced tube was named tube #5 with rib height of 0.7mm and corrugation pitch of 6mm.

2.2 Numerical model

The numerical model was based on diffusion layer theory, in which the heat flux transferred from the steam/air gas layer equalled to that transferred through the condensate film. And the heat flux transferred from gas layer could be divided into the sensible and latent heat flux. Then the overall in-tube condensation heat transfer coefficient could be described as:

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

Here, the \( h_s \) was the sensible heat transfer coefficient. The \( h_{cd} \) was the latent heat transfer coefficient and the \( h_f \) was the condensate film convective heat transfer coefficient.

The calculation procedure consisted of nine steps. The condenser tube was divided into \( n \) sections. The most important part was the determination of gas-liquid interface temperature of each section. The principle was the heat balance between gas layer and condensate film. The predicted inlet temperature of coolant was compared with measured value. The detailed steps could be found in Ren's work [10].

3. Results and discussion

3.1 Comparison of condensation heat transfer coefficient

The variation of average condensation heat transfer coefficient in tube #5 and smooth tube is showed in Fig. 1. It is clearly that both the heat transfer coefficient in tube #5 and that in smooth tube decreases as increasing the inlet noncondensable gas concentration. The reason is that the noncondensable gas boundary layer acts as the mass transfer resistance and decreases the temperature
difference between steam and liquid film. The higher inlet noncondensible gas mass fraction means the larger mass and heat transfer resistance. As described in Reference [9], the descending rate of enhance tube is slightly slower than that of smooth tube. The heat transfer coefficient of tube #5 decreases by 17.31% when gas fraction changes from 0.05 to 0.3. But the descending rate of smooth tube is 28.37%. The enhancement mechanism can be obtained by analysing the distribution of local heat transfer coefficient.

![Figure 1. Variation of average condensation heat transfer coefficient](image1)

### 3.2 Distribution of local heat transfer coefficient at lower inlet gas fraction

Figure 2 depicts the distribution of overall condensation, gas region condensation and liquid film condensation heat transfer coefficient in tube #5 at the inlet noncondensible gas fraction of 5%. The gas region forced convective heat transfer coefficient is too small to be ignored. It is observed that all the heat transfer coefficients decrease along the condenser tube. The gas region condensation heat transfer coefficient $h_{cd}$ is much larger than film region condensation coefficient $h_f$. This indicates that the major heat transfer resistance is the thermal resistance of liquid region at lower inlet gas fraction. The decrease of in-tube condensation coefficient is due to the reduction of liquid film coefficient.

![Figure 2. Distribution of local heat transfer coefficient at inlet gas fraction of 5%](image2)

The distribution of enhancement ratios of in-tube condensation, gas region condensation and liquid film condensation coefficient at the inlet gas fraction of 5% is illustrated in Fig. 3. It is found that all the enhancement ratios decrease along the condenser tube. The enhancement factor curve of in-tube condensation coefficient basically coincides with that of liquid film condensation coefficient. The
average enhancement factor of $h_c$ is 1.88. The average enhancement factor of $h_f$ is 1.86 and the average enhancement factor of $h_{cd}$ is 1.69.

Figure 3. Distribution of enhancement factor at inlet gas fraction of 5%

3.3 Distribution of local heat transfer coefficient at higher inlet gas fraction
Figure 4 shows the distribution of overall condensation, gas region condensation and liquid film condensation coefficient at inlet gas fraction of 30%. It is found that although gas region condensation coefficient is larger than liquid film condensation coefficient, the liquid film condensation coefficient cannot be ignored. The total thermal resistance is determined by heat and mass resistance.

Figure 4. Distribution of local heat transfer coefficient at inlet gas fraction of 30%
The distribution of enhancement ratios of in-tube condensation, gas region condensation and liquid film condensation coefficient at higher inlet gas fraction is demonstrated in Fig. 5. It is observed that the enhancement factor of gas region condensation coefficient is larger than that of liquid film condensation coefficient. The average enhancement factor of $h_c$ is 2.33, the average enhancement factor of $h_f$ is 1.97 and the average enhancement factor of $h_{cd}$ is 2.45. The enhancement factor at higher inlet gas fraction is larger than that at lower inlet gas fraction. At higher inlet gas fraction, the spiral rib not only thinner the condensate film which increases the liquid film condensation coefficient but also destroy the stability of gas film which increases the gas region condensation coefficient. And the thermal resistances of gas and liquid films are both control thermal resistance.

4. Conclusions
In this paper, the average condensation heat transfer coefficients inside enhanced tube and smooth tube were introduced. The distribution of local heat transfer coefficient was presented and the enhancement mechanism at different inlet noncondensable gas mass fractions was revealed. The results show that the enhancement factor at higher inlet gas fraction is larger than that at lower inlet gas fraction. This is because at higher inlet gas fraction gas layer mass transfer resistance becomes the control thermal resistance. The spiral ribs inside corrugated low finned tube can thinner not only the gas layer thickness but also the liquid film thickness.

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