Numerical Simulation of Frosting on Wavy Fin-and-tube Heat Exchanger Surfaces

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Abstract. Frost on fin surfaces of the heat exchanger increases thermal resistance and blocks air flow passage, which reduces the system energy efficiency. In this paper, a frosting model based on Euler multi-phase flow proposed before is used to simulate the frost layer growth process on wavy fin-and-tube heat exchanger surfaces. The model predicts the frost layer and temperature distributions on the heat exchanger surfaces. The air flow pressure drops before and after frosting have been obtained. The results show that the frost layer is unevenly distributed and no frost appears on the fin surfaces in the tube wake region. Frost on the wavy fin-and-tube heat exchanger surfaces restricts the airflow and the pressure drop increases about 140% after 45 min frosting. The simulation results are in good agreement with the experimental results.

1. Introduction

Wavy fins are more appropriate for improving fin-and-tube heat exchanger performance since slit fins and louver fins are easily blocked by dust particles. When the heat exchangers operate in cold and wet air, frost layer may appear on the fin surfaces. Frost on fin surfaces increases thermal resistance and blocks air flow passage, which reduces the system energy efficiency. Thus, the prediction of frost distribution on fin surfaces and the calculation of air flow pressure drop are necessary.

A number of frosting models have been proposed in the past two decades. Lee et al. [1] treated the frost layer growth process as a one-dimensional growth process. In his model, the mass transfer rate from water vapor to ice was proportional to the difference between the absolute humidity of the air and saturated absolute humidity of the frost surface. Hermes [2] proposed a frosting model based on macroscopic heat and mass balances in the frost layer. Lee et al. [3] divided the simulation domain into two parts, the humid air sub-domain and the frost layer sub-domain. The governing equations were solved for the air-side and the frost layer respectively. Na et al. [4] presented a frosting model, which assumed the humid air at the frost surface was supersaturated. Yang et al. [5], Lenic et al. [6] and Armengol et al. [7] also established governing equations for the humid air sub-domain and the frost layer sub-domain to simulate frosting on flat plates. Cui et al. [8, 9] proposed a frosting model based on the nucleation theory and multi-phase flow method and calculated frost growth on flat plates and fin surfaces. The frosting models proposed by Kim et al. [10] and Wu et al. [11-13] also based on the multi-phase flow method. Wu et al. [11] developed a phase change mass transfer model based on the phase change driving force, which can better reflect the frosting mechanism. A frost layer growth
criterion was proposed to illustrate the frost layer growth and the densification processes. The frosting processes on a cold plate with local cooling and on plain fin-and-tube heat exchanger surfaces were simulated and the simulated results of frost distribution, frost thickness, frost weight and local temperature agreed well with experimental results [11, 12]. Later, an improved frosting model with non-dimensional phase change driving force and a criterion for simulating frost growth was developed, and the model was validated by comparing not only the simulated frost thickness but also the corresponding frost weight with experimental data for various conditions. [13]

In this paper, the frosting model proposed by Wu et al. [13] is used to simulate the frosting process on wavy fin-and-tube heat exchanger surfaces.

2. Frosting Model

The Euler multi-phase model was used to calculate the multi-phase flow and two phases were set in the model: the primary phase was humid air containing dry air and water vapor, the secondary phase was ice. In the frosting process, the water vapor in humid air conducted mass transfer to the ice, simultaneously the momentum transfer and energy transfer basing on the mass transfer generated. In order to simplify the simulation, the following assumptions were applied:

1. Humid air was assumed to be incompressible Newtonian fluid;
2. The heat radiation was neglected;
3. Only the mass transfer from water vapor to ice is taken into consideration i.e., the frost sublimation and melting processes were neglected.

2.1. Governing equations

The calculation domain was divided into two parts, the humid air domain and the aluminum fin domain. The governing equations for the humid air domain contained the mass conservation equation, the momentum conservation equation and the energy conservation equation for each phase and the species conservation equation for water vapor. Each phase was represented by the volume fraction, \( \alpha \). There were two phases exist in one control volume, as relation (1).

\[
\alpha_a + \alpha_i = 1
\]  

where \( \alpha_a \) and \( \alpha_i \) are volume fraction of humid air and volume fraction of ice respectively.

The governing equation for the aluminum fin domain contained the energy conservation equation.

2.2. Mass transfer model

Since the phase-change driving force for frosting equals to the decrease of Gibbs free energy when a single water molecule transfers from gas state to solid state. Expression \( \frac{w_c - w_{vs}}{w_{vs}} \) was used to express the non-dimensional phase change driving force for frosting, where \( w_c \) is the mass fraction of water vapor in humid air, while \( w_{vs} \) is the saturated mass fraction of water vapor corresponding to the temperature of the control volume.

The mass transfer rate from water vapor to ice \( \dot{m}_a \) can be expressed as follows:

\[
\dot{m}_a = \tau_c \alpha_a \rho_a w_c \frac{w_v - w_{vs}}{w_{vs}}
\]  

where \( \tau_c \) is the time relaxation coefficient for frosting, \( \rho_a \) is the density of humid air in the control volume.

2.3. Simulation object

Figure 1 presents a part of a wavy fin-and-tube heat exchanger. Two rows of circular tubes staggered in a regular triangular arrangement. The fins were made of aluminum and the tubes were made of copper. The outer diameter of the tube is 4.00 mm and the tube pitch is 22.00 mm. The fin thickness
and fin pitch are 0.11 mm and 1.50 mm respectively. The simulation region includes a unit of the wavy fin-and-tube heat exchanger, an inlet section and an outlet section.

The frontal air velocity \( u \) is 1.0 m/s, the air temperature \( T_m \) is 2°C and the relative humidity \( RH \) is 85%. The wall temperature of the tube \( T_w \) maintains −5°C.

![Image](a) Oblique view  
(b) Side view

**Figure 1.** A part of the wavy fin-and-tube heat exchanger.

3. Results and Discussion

3.1. Temperature and water vapor concentration distributions

Figure 2 shows the temperature distribution of the wavy fin. The tube wall is the cooling wall with constant temperature. Thus, the fin temperature near the tube is relatively low. Since the air temperature is relatively high, the fin temperature is higher near the air inlet.

Figure 3 shows the water vapor mass fraction distribution in the humid air phase near the wavy fin surface. The water vapor mass fraction is approximately relatively low downwind of the tubes, which is because there is very little air flow in the wake region of the tubes and the water vapor in the wake region is mostly depends on the diffusion.

**Figure 2.** Wavy fin surface temperature distribution \((T_w=-5^\circ\text{C}, T_m=2^\circ\text{C}, RH=85\%, u=1.0 \text{ m/s, } t=45 \text{ min})\).  

**Figure 3.** Water vapor mass fraction distribution after frosting \((T_w=-5^\circ\text{C}, T_m=2^\circ\text{C}, RH=85\%, u=1.0 \text{ m/s, } t=45 \text{ min})\).
3.2. Frost distribution on the fin surface

Figure 4 presents the simulated frost distribution on the wavy fin surface and figure 5 presents the frost layer thickness on the wavy fin surface. The frost consists of ice and air. Thus, higher ice volume fraction means higher density of the frost.

From figure 4, there is no frost appears on the fin surface in the tube wake region, which is due to the low water vapor concentration. The simulation result of frost distribution on fin surface agrees with the experimental result, which is presented in figure 6.

![Simulated frost distribution on the wavy fin surface](image)

**Figure 4.** Simulated frost distribution on the wavy fin surface  

t_F=5°C, T_w=2°C, RH=85%, u=1.0 m/s, t=45 min).

![Frost layer thickness on the wavy fin surface](image)

**Figure 5.** Frost layer thickness on the wavy fin surface  

t_F=5°C, T_w=2°C, RH=85%, u=1.0 m/s, t=45 min).

![Experimental observed frost distribution on the wavy fin surface](image)

**Figure 6.** Experimental observed frost distribution on the wavy fin surface.

3.3. Air flow pressure drops

The experimentally measured air flow pressure drops are 14.2 Pa and 34.2 Pa before frosting and after 45 min frosting for the wavy fin-and-tube heat exchanger, respectively. However, the experimentally measured values are 10.8 Pa and 20.0 Pa before frosting and after 50 min frosting for the plain fin-and-tube heat exchanger at the same geometrical size [12]. The air flow pressure drops for the wavy fin-and-tube heat exchanger drop are larger than that for the plain fin-and-tube heat exchanger. The results also show that the frost layer on the heat exchanger surfaces increase the pressure drop, which not only due to the narrower flow passage but also the rougher frost layer surface. The simulated pressure drops are 13.1 Pa and 31.2 Pa before frosting and after 45 min frosting for the wavy fin-and-tube heat exchanger, respectively. Therefore, the average relative error between the simulated and measured pressure drops is ~8.2%.
4. Conclusion
In this paper, the frosting process on wavy fin-and-tube heat exchanger surfaces is numerical simulated with the help of the frosting model proposed previously. The frost distribution on the wavy fin is obtained and no frost appears on the fin surfaces in the tube wake region due to the low water vapor concentration there. The simulated frost distribution agrees with the experimental observed frost distribution. The frost layer on the heat exchanger surfaces restricts the air flow and the pressure drop increases about 140% after 45 min frosting. The air flow pressure drop is larger for the wavy fin-and-tube heat exchanger than that for plain fin-and-tube heat exchanger. The average relative error between the simulated and measured pressure drops for the wavy fin-and-tube heat exchanger is 8.2%.

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