Performance Evaluation Of Peripheral-Finned Tube Evaporators Under Frosting Conditions

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Performance Evaluation of Peripheral-Finned Tube Evaporators under Frosting Conditions

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ABSTRACT

The peripheral finned-tube (PFT) is a novel air-side geometry for compact heat exchangers. Previous studies (Pussoli et al., 2012) evaluated the thermal-hydraulic performance of the PFT geometry under dry conditions (i.e., no frost or condensate formation on the air side). The present study focuses on quantifying the thermal-hydraulic performance of the PFT geometry under frosting conditions. A PFT prototype was tested in a closed-loop wind tunnel calorimeter to determine the influence of the tube wall temperature, air velocity and psychrometric properties (temperature and relative humidity) on the heat transfer rate, air-side pressure drop and mass of frost built up on the surface. A mathematical model was developed to predict the behavior of the evaporator in a variety of operating conditions. The model treats the airflow paths as a porous medium, whose porosity, equivalent particle diameter and thermal properties change with time due to the frost accumulation. The time-dependent characteristics of rates of heat transfer and moisture deposition and of the air-side pressure drop were well correlated by the mathematical model.

1. INTRODUCTION

Frosting is an undesirable yet inevitable phenomenon in refrigeration and air-conditioning equipment. It results from the desublimation of water vapor from moist air flowing over a solid surface whose temperature lies below the freezing point of water. Frosting reduces the cooling capacity and the COP of the cooling unit by adding a low-thermal conductivity resistance to the air-side surface of the evaporator, which also decreases the airflow rate due to the narrowing of the airflow passages.

The literature on frosting can be divided into studies on (i) the measurement and correlation of physical properties (Hayashi et al., 1977; Iragorry et al., 2004; Hermes, 2014, Negrelli & Hermes, 2015), (ii) the mechanisms of frost nucleation, growth and densification (Lee et al., 1997; Na & Webb, 2004; Hermes et al., 2009), and (iii) the prediction of the performance of heat exchangers under frosting conditions (Knabben et al., 2010; Silva et al., 2011b; Hwang & Cho, 2014; Chen et al., 2016). In the latter class of studies, the proposed models are such that the air-side flow path is divided into one-dimensional (lengthwise) or two-dimensional (lengthwise and spanwise) control volumes onto which mass momentum and energy balances are applied to calculate the local rates of frost formation (vapor desublimation), heat transfer and the pressure drop. The local rates are integrated over the entire area of the heat exchanger, to give the overall instantaneous rates. Moreover, in two-dimensional approaches, it is possible to evaluate more precisely the influence of the refrigerant flow distribution and predict the zones of the air-side surface that are more affected by the frost accretion. This has a direct impact on the airflow distribution (Knabben et al., 2010; Hwang & Cho, 2014). Thermal capacity effects of the frost layer are usually neglected with satisfactory results (Hermes, 2012).

The majority of heat exchanger frosting models has been proposed for conventional finned surfaces (e.g., integral fins, plain continuous fins, wavy fins, etc.) (Seker et al., 2004; Yang et al., 2006; Silva et al., 2011b; Hwang & Cho, 2014; Chen et al., 2016). As far as the present authors are aware, no model has been proposed to predict the performance of novel (structured or unstructured) enhanced surfaces. The peripheral finned-tube (PFT) geometry is a cross-flow heat exchanger whose air-side is composed by a hexagonal arrangement of open-pore (interconnected) cells formed by radial fins whose bases are attached to the tubes and tips are connected to the peripheral fins. Each fin arrangement is constructed with six radial fins and six peripheral fins forming a hexagon-like structure. The PFT geometry was
proposed by Wu et al. (2007) and evaluated experimentally and mathematically by Pussoli et al. (2012) for sensible heat transfer conditions.

The objective of the present paper is to extend the analysis of Pussoli et al. (2012) to predict the behavior of the peripheral finned-tube geometry under frosting conditions. To this end, experiments have been conducted using one of the prototypes developed by Pussoli et al. (2012) in a closed-loop wind tunnel calorimeter specifically designed and built for thermal-hydraulic experiments under frosting conditions (Silva et al., 2011a). A mathematical model was developed and implemented to predict the time-dependent behavior of parameters such as the heat transfer rate, enthalpy effectiveness, pressure drop and mass of frost accumulated with time. The majority of the experimental data was predicted to within 20% deviation with respect to the data.

2. EXPERIMENTAL WORK

2.1 Heat exchanger prototype

The heat exchanger prototype evaluated in this work is shown in Fig. 1. The outer diameter of the copper tube is 8.94 mm. A staggered tube array with a mixed-mixed cross-parallel flow configuration was used. The basic dimensions of the heat exchanger and peripheral fins are shown in Table 1. The fin structure is composed of three distinct levels of fin arrangement (R1, R2 and R3), each characterized by the length of the radial fin extending from the tube. The fins were assembled with a 30° offset from their neighbors, according to the distribution shown in Table 1. Each group of six fins comprises a so-called ‘unit’, as illustrated in the insert of Fig. 1 (Pussoli et al., 2012).

<table>
<thead>
<tr>
<th>Characteristics of the prototype used in the tests</th>
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<tr>
<td>R1 / R2 / R3 length [mm]</td>
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<tr>
<td>R1 / R2 / R3 number of fins [-]</td>
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<td>Fin thickness / width [mm]</td>
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<tr>
<td>Porosity [-]</td>
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<td>Surface area [m²]</td>
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<tr>
<td>Height / Length / Width [mm]</td>
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</tbody>
</table>

2.2 Experimental facility

The experimental work was carried out in a closed-loop wind-tunnel calorimeter (Silva et al., 2011a), as shown in Fig. 2. The apparatus controls the air flow rate and its psychrometric condition (temperature and humidity) at the test section inlet. In the lower part of the wind tunnel, the air flow is measured using a calibrated nozzle (1) connected to a differential pressure transmitter with an uncertainty of ±0.5% of the full-scale (1744 Pa). The air is then cooled by a coil (2) connected to a vapor compressor refrigeration system (see Fig. 3). Next, the air is re-heated by an electric heater (3) and re-humidified by a water tray (4). The air flow rate is supplied by a computer-driven variable-speed centrifugal fan (5). In the upper part, a wire mesh was used to make the flow uniform at the inlet of the test section. The air temperature was measured by eighteen thermocouples (TT) (nine at the inlet and nine at the outlet), with an uncertainty of ±0.2°C. The thermocouples were embedded into small copper blocks (diameter and height of 10 mm) to minimize temperature oscillations. The air relative humidity was measured by two humidity transducers placed upstream and downstream of the test section (HT) (uncertainty of ±1%). The air-side static pressure drop was measured by a differential pressure transmitter (PT) with an uncertainty of ±0.5% of the full-scale (996 Pa).

The coolant (a water-glycol solution) temperature and flow rate are set by a secondary flow loop connected to a commercial chiller. The flow rate is measured with a turbine flow meter with an accuracy of 0.3% of the full scale (11.35 L/min). The inlet and outlet temperatures are measured by T-type immersion thermocouples (±0.2°C). The tubes that connect the heat exchanger to the chiller are thermally insulated. A schematic diagram of the flow loops is shown in Fig. 3.
2.3 Experimental Procedure and Test Conditions

The heat exchanger is fastened onto a wooden frame inside the test section. The desired test conditions (flow rates, inlet humidity and temperatures) are adjusted in the control system. While these conditions are not reached and stabilized (usually 3 hours are needed), the air and the coolant by-pass the evaporator to avoid the premature frost formation on the surface. The room temperature was kept at 15 ± 1°C. The test is started by directing the air and coolant streams to the evaporator and beginning the data acquisition. At the end of a test, the evaporator is removed from the test section and placed in a tray to collect and weigh the accumulated frost on a digital scale with an accuracy of 0.01 g (gravimetric method).

In total, 24 tests were carried out with the following conditions: inlet air flow rate: 20 and 30 CFM (34 and 51 m³/h); inlet air temperature: 15 °C; inlet coolant temperature: −10°C and −15°C; inlet relative humidity: 50% and 80%; test duration: 30, 60 and 90 minutes.

2.4 Data regression

The desublimation rate, \( \dot{m}_s \), was calculated based on a moisture mass balance on the air side as follows:

\[
\dot{m}_s = \dot{m}_a (\omega_{\text{out}} - \omega_{\text{in}})
\]

where the humidity ratios were computed based on the local measurements of air temperature and relative humidity. In addition to the gravimetric method, the mass of frost accumulated on the air side can be computed from the integration of Eq. (1) over the duration of the test:

\[
M_s = - \int_0^t \dot{m}_s \, dt
\]

where \( t \) is time. The heat transfer rate was computed via a heat balance on the coolant side.

3. MODELING

3.1 Fin Thermal Model

The geometric parameters of the peripheral fins are shown in Fig. 4(a) and (b). Thermal equilibrium is assumed at the connection between the tip of a radial fin and the bases of two peripheral fins, as seen in Fig. 4(b). Thus,

\[
\dot{Q}_{r,t} = 2\dot{Q}_{p,B}
\]

where \( \dot{Q}_{r,t} \) and \( \dot{Q}_{p,B} \) are the heat transfer rates at the tip of a radial fin and at the base of a peripheral fin.
The heat transfer rate through a frosted fin can be calculated based on the concept of fictitious fin enthalpy. Thus, an energy balance in a generic frosted fin (radial or peripheral) gives (Threlkeld et al., 1998):

\[
\frac{d^2 \Delta h_f}{dx^2} - m^2 \Delta h_f = 0
\]

where \( \Delta h_f \) is the fictitious fin enthalpy potential defined as the difference between the enthalpy of moist air, \( h \), and the enthalpy of saturated air at the mean fin temperature. \( m^2 = 2h_o/(k_f t_f) \), where \( h_o \) is an effective heat transfer coefficient that accounts for the heat conduction in the frost layer (Threlkeld et al., 1998).

The radial and peripheral fins are subjected to different boundary conditions. For radial fins, \( \Delta h_f(r)(x=0) = \Delta h_f(r,s) \) (fictitious enthalpy prescribed at the base) and \( \Delta h_f(r)(x=L_r) = \Delta h_f(r,t) \) (fictitious enthalpy prescribed at the tip). For peripheral fins, \( \Delta h_f(p)(x=0) = \Delta h_f(p,s) \) (fictitious enthalpy prescribed at the base) and \( d\Delta h_f(p)(x=L_p)/dx = 0 \) (symmetry). Solving Eq. (4) for the radial and peripheral fins with their respective boundary conditions, and using Fourier’s law to determine the heat transfer rates in Eq. (3) gives:

\[
\dot{Q}_{p,B} = \frac{k_f A_c}{b_1} \Delta h_{f,p,B} m_p \sinh (m_p L_p)
\]

\[
\dot{Q}_{r,L_r} = \frac{k_f A_c}{b_1} \left[ \frac{\Delta h_{f,r,B} m_r - \Delta h_{f,r,L_r} m_r \cosh (m_r L_r)}{\sinh (m_r L_r)} \right]
\]

where \( b_1 = (h_{sat,s} - h_{sat,t}) / (T_s - T_t) \) is the slope of the linear interpolation of the saturated air enthalpy with respect to temperature (Threlkeld et al., 1998). The subscripts \( s \) and \( t \) correspond to the frost layer and tube, respectively. \( A_c \) is the fin cross-section area. By substituting Eqs. (5) and (6) in Eq. (3), the fictitious fin enthalpy, \( h_f \), that satisfies the equilibrium condition at the connection, and the enthalpy potential distributions along the fins can be determined.

### 3.2 Heat Exchanger Model

To calculate the total heat transfer rate, pressure drop and desublimation rate, the evaporator is divided into uniform control volumes in the direction of the air flow, as shown in Fig. 4(c). The number of control volumes coincide with
the number of tube rows in the flow direction (Pussoli et al., 2012). The air flow is assumed uniformly distributed in the spanwise direction.

Energy balances between the inlets and outlets of the moist air and coolant streams of a control volume are given by:

\[
\dot{Q}_{CV} = \dot{m}_a (h_{in} - h_{out}) - \dot{m}_c (h_{c,in} - h_{c,out})
\]

where \(\dot{m}_a\) is the water vapor desublimation rate and \(\dot{m}_c\) is the enthalpy of the frost surface. The heat transfer rate in the control volume can be calculated from the concept of log-mean enthalpy difference as follows (Threlkeld et al., 1998):

\[
\dot{Q}_{CV} = UA_{CV} \Delta h_{LM,CV} = UA_{CV} \left[ \frac{(h_{in} - h_{f,c,out}) - (h_{out} - h_{f,c,in})}{\ln \left( \frac{h_{in} - h_{f,c,out}}{h_{out} - h_{f,c,in}} \right)} \right]
\]

where \(h_{in}\) and \(h_{out}\) are the enthalpies of moist air at the inlet and outlet of the control volume, respectively. \(h_{f,c,in}\) and \(h_{f,c,out}\) are the fictitious enthalpies of the coolant at the inlet and outlet of the control volume. The total heat transfer rate can be calculated by summing up the contributions of the \(N_{CV}\) control volumes. The overall heat transfer coefficient associated with each control volume is given by (neglecting the conduction resistance of the tube wall):

\[
\frac{1}{UA_{CV}} = \frac{b_c}{h_c} + \frac{b_t}{h_t} + \sum_{k=1}^{3} N_{r,k} A_{o,k} \eta_{o,k}
\]

where \(b_c = (h_{sat,s} - h_{sat,c}) / (T_s - T_c)\). In Eq. (9), \(k\) is a counter for the number of array types (R1, R2 and R3). \(\eta_o\) is the overall efficiency of the (frosted) finned surface calculated based on the frosted fin efficiency. \(A_i\) is the internal surface area (tube) and \(A_o\) is the external surface area, which includes the area of the frosted fins and frost-covered bare tube, as seen in Fig. 4(b). The internal heat transfer coefficient, \(h_t\), was estimated using the Gnielinski correlation (Shah & Sekulic, 2003). The air-side heat transfer coefficient, \(h_{o,k}\), was estimated using the Whitaker (1972) correlation. It should be noted that \(h_{o,k}\) is different for each fin type \(k\) due to distinct values of frost thickness on each fin surface.

The water vapor desublimation rate is calculated based on a species mass balance between the inlet and the outlet of each control volume,

\[
\dot{m}_{s,CV} = \dot{m}_a (\omega_{in} - \omega_{out}) = \bar{h}_{m,o} \left( \sum_{k=1}^{3} A_{p,o,k} \Delta \omega_{LM,P,k} N_{p,k} + \sum_{k=1}^{3} A_{r,o,k} \Delta \omega_{LM,R,k} N_{r,k} + A_{b,o} \Delta \omega_{LM,b} \right)
\]

where \(\bar{h}_{m,o}\) is the average convective mass transfer coefficient in the control volume calculated using the Lewis analogy for convective heat and mass transfer on the air side. \(\Delta \omega_{LM,j,k}\) is the log-mean humidity ratio given by:

\[
\Delta \omega_{LM,j,k} = \frac{(\omega_{in} - \omega_{sat,j,k}) - (\omega_{out} - \omega_{sat,j,k})}{\ln \left( \frac{\omega_{in} - \omega_{sat,j,k}}{\omega_{out} - \omega_{sat,j,k}} \right)}
\]

In a control volume, each fin type \((j = r \text{ or } p)\), in each level (R1, R2 or R3), has its own fictitious enthalpy distribution, resulting in a different mean surface saturation temperature. Hence, there are different log-mean humidity ratios for each fin, plus the bare-tube part, resulting in seven different contributions to the desublimation rate. Again, the total vapor desublimation rate is calculated summing up the rates associated with each control volume.

The total air-side pressure drop is calculated as follows:

\[
\Delta P = \frac{1}{2} \rho_1 u_1^2 \left( 1 - \epsilon_1^2 + K_e \right) + \sum_{n=1}^{N_{CV}} \int_0^{L_n} \frac{\rho_n u_n^2}{d_{p,n}} \left( \frac{1 - \epsilon_n}{\epsilon_n} \right) - \frac{1}{2} \rho_{NCV} u_{NCV}^2 \left( 1 - \epsilon_{NCV}^2 - K_e \right)
\]

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where the first and third terms are the entrance and exit pressure losses, respectively. The contraction and expansion coefficients, $K_c$ and $K_e$, where obtained from Shah & Sekulic (2003), assuming that the air side behaves as stack of parallel plates. $u_1$ and $u_{N_{CV}}$ are the in-situ velocities in the first and last control volumes. The second term accounts for the frictional pressure drop (skin friction and form drag) in the $n = 1$ to $N_{CV}$ control volumes of the porous matrix. $u_D$ is the Darcian velocity of the moist air. The equivalent particle diameter was calculated as six times the ratio of the solid volume (metal and frost) to the interstitial area. As in the heat and mass transfer calculation, five control volumes were considered. The friction factor was calculated using the Ergun (1952) correlation.

### 3.3 Frost Formation Model

The following assumptions have been adopted in the development of the frost formation model, whose aim is to quantify the change in frost thickness and density with time, on fins of each type and level in every control volume: (i) the frost thickness and density are uniform; (ii) the pressure is uniform inside the frost layer; (iii) heat and mass diffusion in the frost are one-dimensional and quasi-steady over a time step; (iv) the thermophysical properties of the frost are uniform.

For each fin type and level, a frost mass balance is given by:

$$
\rho_{s,j,k} \frac{d\delta_{j,k}}{dt} + \delta_{j,k} \frac{d\rho_{s,j,k}}{dt} = \bar{h}_{m,a} \Delta \omega_{LM,j,k}
$$

where the first term on the left-hand side is due to frost growth and the second is due to frost densification. The frost density was calculated based on the correlation of Hayashi et al. (1977) given by $\rho_s = 650 \exp(0.277 T_s)$, where $T_s$ is the temperature of the frost surface in °C. A linearized form of Eq. (13) is used to calculate the frost thickness at every integration time step (Hermes, 2012).

The temperature of the frost surface on each fin is calculated through an energy balance at the interface between the frost layer and the moist air. The thermal capacity of the frost layer was neglected and a linear temperature distribution was assumed. Thus:

$$
T_{s,j,k} = T_{f,j,k} + \frac{\bar{Q}_{f,j,k} \delta_{j,k}}{k_{s,j,k} A_{o,j,k}}
$$

where $A_{o,j,k}$ is the surface area of the frosted fin, $T_{f,j,k}$ is the average fin temperature and $\bar{Q}_{f,j,k}$ is the total heat transfer rate through each fin. The thermal conductivity of the frost was calculated using the empirical relationship proposed by Hermes et al. (2009) as a function of the frost density.

### 4. RESULTS

The mathematical model was fully implemented on the EES platform. Fig. 5 compares the experimental and calculated heat transfer rate as a function of time for a specific run. The heat transfer rates in each control volume are also shown. For this run, the model gives a satisfactory prediction of the experimental data (although it over predicts the data by around 10%). The data decrease with time as a result of the increasing thermal resistance of the frost layer. As expected, the heat transfer rate is higher in the first control volume and decreases downstream with the reduction of the air-coolant enthalpy difference. The behavior of the frost mass accumulated on the air-side surface is shown in Fig. 6. An almost linear increase is predicted, which under estimated by around 10% the mass measured with the scale at the end of the test. Similarly to the heat transfer rate, the mass accumulated on the surface is higher in the first control volume and reduces downstream due to the decreasing mass transfer potential (humidity ratio difference).

Figure 7 shows the pressure drop predictions for the same condition of Figs. 5 and 6. An excellent agreement is observed between the model and the experimental data for this case. Although the pressure drop is almost the same in all control volumes at the beginning of the run, the differences between the control volumes increase with time, and are higher in the last control volume. This is due to the increase in frost thickness (reduction of the heat exchanger porosity) shown in Fig. 8, which causes an acceleration of the air flow through the heat exchanger.

The pressure drop results considering the entire dataset are shown in Fig. 9. As in the purely sensible heat transfer tests of Pussoli et al. (2012), i.e., no frosting, the model is in good agreement with the data, with a slight tendency for
Figure 5: Heat transfer rate. Inlet relative humidity, air and coolant temperatures: 80\%, 15\°C, −10\°C.

Figure 6: Frost accumulation. Inlet relative humidity, air and coolant temperatures: 80\%, 15\°C, −10\°C.

Figure 7: Pressure drop. Inlet relative humidity, air and coolant temperatures: 80\%, 15\°C, −10\°C.

Figure 8: Frost thickness and porosity. Same conditions as Fig. 7.

it to under predict the data at the early stages of the tests (lower pressure drops). Fig. 10 compares the predictions of frost mass with the experimental data for the entire data set. Again, the model behaves satisfactorily (within 20\% error bands). Nevertheless, the experimental data of the gravimetric method are consistently larger, except for the short (30-minute) tests. In these tests, the model clearly over predicts the frost mass obtained via the experimental mass balance. Because of the transient nature of the experiments, a few minutes are necessary until the equilibrium conditions are reached. In these early minutes, the vapor desublimation on the heat exchanger surface may not be as intense as predicted due to thermal capacity effects of the heat exchanger itself, which are neglected in the model.

The prediction of the enthalpy effectiveness, \( \epsilon_h \), for the entire dataset is shown in Fig. 11. The enthalpy effectiveness is defined as the ratio of the actual heat transfer rate to the maximum heat transfer rate as follows:
\( \epsilon_h = \frac{\dot{Q}}{\bar{m}_a (h_{\text{in}} - h_{\text{min}}) - \bar{m}_a (\omega_{\text{in}} - \omega_{\text{min}}) h_s, T_c} \)

where \( h_{\text{min}} \) and \( \omega_{\text{min}} \) are the enthalpy and humidity ratio of saturated air at the inlet coolant temperature, \( T_c \). \( h_s \) is the enthalpy of ice at \( T_c \).

As can be seen, the agreement between the model and the experimental data is satisfactory for the earlier times (when the frost thickness is small), which is consistent with the findings of Pussoli et al. (2012) for purely sensible heat transfer in peripheral-finned tube heat exchangers. However, for some test conditions, much larger deviations were observed during the final stages of the test. These may be due to a number of reasons. Firstly, at the later stages of the frost formation, the calculated frost surface temperature is higher than the highest temperature in the range of validity of the frost density correlation (Hayashi et al., 1977), i.e., \(-18.6^\circ\text{C} \) to \(-5^\circ\text{C}\). This can affect not only the accuracy of the prediction of the frost thickness, but also the estimation of the frost thermal conductivity itself, which also depends on the choice of a specific correlation (Hermes et al., 2009). Secondly, as the frost thickness is under predicted and the flow passages through the radial fins are actually more obstructed by the frost than what the model estimates, the model gradually loses the ability of accurately predicting the reduction of the effective heat transfer surface area. As a result, the enthalpy effectiveness (and the heat transfer rate) is over predicted. In reality, as the effective surface area associated with the radial fins is reduced, the parallel-flow channels between the smaller fins (see Fig. 12) become the primary flow path. The fact that this flow path is the least thermally effective is another reason for the reduction of the heat transfer rate observed experimentally. Different correlations for the interstitial Nusselt number are currently being evaluated. The conditions of the photographs in Fig. 12 are as follows: (a) Inlet air temperature: 15 °C, inlet relative humidity: 80%, inlet air volume flow rate: 20 CFM, inlet coolant temperature = \(-10^\circ\text{C}\), test duration: 60 min; (b) Inlet air temperature: 15 °C, inlet relative humidity: 50%, Inlet air volume flow rate: 20 CFM, inlet coolant temperature = \(-15^\circ\text{C}\), test duration: 90 min.

### 5. CONCLUSIONS

A thermal-hydraulic analysis of the peripheral finned-tube geometry under frosting conditions was developed in this paper. A wind-tunnel calorimeter was used to evaluate the performance of a prototype developed by Pussoli et al. (2012) at twenty-four different test conditions. Parameters such as the total heat transfer rate, pressure drop and frost accumulation were acquired. A detailed mathematical model was developed to predict the spatiotemporal behavior of
the parameters evaluated experimentally. A satisfactory agreement was found between the model predictions and the experimental data, with most of the data for the mass of frost, pressure drop and enthalpy effectiveness lying within ±20% error bands.

NOMENCLATURE

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<td>$c$</td>
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Figure 11: Enthalpy effectiveness predictions.

Figure 12: Photographs of the frost formation.
REFERENCES


ACKNOWLEDGMENTS

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