# The Effects of Design and Operating Factors on the Frost Growth and Thermal Performance of a Flat Plate Fin-tube Heat Exchanger Under the Frosting Condition

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An experimental study of the effects of various factors (fin pitch, fin arrangement, air temperature, air humidity, and air velocity) on the frost growth and thermal performance of a fin-tube heat exchanger has been conducted under the frosting condition. It is found that the thermal performance of a heat exchanger is closely related to the blockage ratio of the air flow passages due to the frost growth. The maximum allowable blockage ratio is used to determine the criteria for the optimal operating conditions of a fin-tube heat exchanger. It is also shown that heat transfer rate of heat exchanger with staggered fin arrangement increases about 17% and the time required for heat transfer rate to reach a maximum value becomes longer, compared with those of an inline fin-tube heat exchanger under the frosting condition. The energy transfer resistance between the air and coolant decreases with the increase of inlet air temperature and velocity and with decreasing inlet air humidity.

**Key Words** : Frost, Fin-tube Heat Exchanger, Blockage Ratio, Fin Pitch, Fin Arrangement, Energy Transfer Resistance

**Nomenclature**

$A$ : Heat transfer area, $m^2$
$a$ : Flow area, $m^2$
$BR$ : Blockage ratio, %
$c_p$ : Specific heat at constant pressure, kJ/kg°C
$D$ : Diameter, m
$i$ : Enthalpy, kJ/kg
$L_H$ : Latent heat, kJ/kg
$m$ : Mass, kg
$\dot{m}$ : Mass flow rate, kg/s
$q$ : Heat transfer rate, kW
$R$ : Energy transfer resistance, °C/W
$S$ : Spacing, mm
$T$ : Temperature, °C
$t$ : Time, s or hr
$V$ : Volume, $m^3$
$w$ : Specific humidity, kg/kg′
$X$ : Thickness, m or mm
$\rho$ : Density, $kg/m^3$

**Subscripts**

$a$ : Air
$av$ : Average
$c$ : Coolant
$F$ : Fin
$f$ : Forst
$i$ : Inlet
$inl$ : Inline fin arrangement
$l$ : Latent
$max$ : Maximum
$o$ : Outlet, outer
$s$ : Sensible
$stg$ : Staggered fin arrangement
$T$ : Total
$t$ : Tube

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## 1. Introduction

The frost layer formed on surfaces of a fin-tube heat exchanger, such as an evaporator of domestic refrigerator/freezer in use today, not only blocks the heat flow between moist air and refrigerant, but also decreases the thermal performance of the heat exchanger since the air flow channel becomes smaller and hence the volumetric air flow rate is reduced with higher pressure drop. To obtain a good performance of the heat exchanger, a periodic defrosting is required. This defrost will inevitably interrupt the refrigeration process, causing the extra heating energy and temperature rise in the unit, which does eventually give rise to an additional energy consumption. Nevertheless, the heat exchangers have been designed without considering the frost formation on the heat exchanger surface. Hence, the designers should know how much additional cooling capacity is required in order to compensate for the reduced performance, and also when the defrosting process needs to be active to minimize the energy consumption.

A considerable portion of the literature about frost formation process is concerned with frost growth and properties in simple geometries. The previous studies available on the fin-tube heat exchanger operating under the frosting condition are limited and lead to different results, sometimes conflicting with one another. The contradictory results from the previous works made it difficult to predict and evaluate the performance of heat exchanger with frosting. Gatchilov & Ivanova, 1979, Hosoda & Uzuhashi., 1967, and Stoecker, 1957 reported that the heat transfer rate increases at the initial stage of frosting and then decreases with time, whereas Aoki et al., 1985 showed the opposite trend. Barrow, 1985 and Kondepudi, 1988 concluded that the insulation effect due to the frost layer was negligible and the reduction in the total heat transfer coefficient resulted from the blocking effect of the air side due to the frost buildup. Rite & Crawford, 1991 insisted that the heat transfer rate increases continuously with frost growth by the effects of frost surface roughness and the steady increase in air velocity. Many researchers (Barrow, 1985; Gates et al. 1967; Kondepudi, 1988; Niederer 1976) have indicated that the decrease of air flow rate caused by pressure drop is the index which represents the unfavorable influence of frost formation on heat transfer performance.

In the present study, using a 2-row, 2-column fin-tube heat exchanger, the effects of design factors (fin pitch and fin arrangement) and operating conditions (temperature, humidity, and velocity of air) on the frost growth and thermal performance of the heat exchanger under the frosting condition are investigated to obtain useful data for designing a high performance heat exchanger and to find optimum operating condition.

## 2. Experiment

### 2.1 Experimental apparatus

Figure 1 shows the experimental apparatus used in the present work. The apparatus is composed of four sections; the test section, circulation section, climate chamber and cooling section. To see the frosting phenomena occurring under various conditions, the test section was constructed of acrylic plate with 0.02 m thickness and insulated by detachable styrofoam with 40 mm thickness. By altering the number of revolutions of the fan with 0.5 HP, the circulation section could control air flow rate induced into the test section. The


**Fig. 1** Schematic diagram of experimental apparatus.

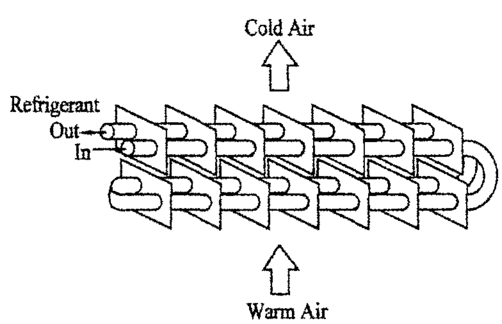


**Fig. 2** Fin-tube heat exchanger used in this study.

volume of the climate chamber was 252 ℓ, and it controlled the temperature and humidity of air. The cooling section was composed of a pump with 1 HP and a refrigerator with 5 HP to control the flow rate and temperature of the refrigerant circulated in the test heat exchanger. Figure 2 shows the fin-tube heat exchanger used in the present work.

### 2.2 Experimental method

The temperature and humidity of the inlet and outlet air of the heat exchanger were measured using five type-T thermocouples and ceramic humidity sensors, respectively, placed in the inlet and outlet of the test section. The volumetric air flow rate was measured by the flow nozzle located in the circulation section. The inlet and outlet temperatures and flow rate of the refrigerant were measured using RTD placed in the heat exchanger and a turbine flow meter, respectively. The tube wall temperature of the test heat exchanger was measured using the type-T thermocouples (wire diameter=0.01 mm) set in three locations (inlet, middle, and exit) on the surface of tube. The fin temperature was obtained as the arithmetic average of the root and edge temperatures of the fin located at the middle of tube, which were measured using type-T thermocouples. Aluminium tape with thickness of 50 $\mu$m was used to attach a thermocouple to the surface of the heat exchanger. The necessary data were monitored by a personal computer through data recording every 15 seconds. The thickness of the frost layer was measured through a photograph of frost layer, and the pressure drop across the test section was measured using a precision manometer every 15 minutes. The amount of frost was correlated with the mass flow rate of air and the difference between the inlet and exit humidities of the air. In this experiment, the volumetric air flow rate was maintained at a constant value for the duration of each test.

### 2.3 Definition of the various quantities

#### 2.3.1 Heat transfer rate

Total heat transfer rate of the air passing through the heat exchanger can be expressed as the sum of the sensible heat transfer due to the variation of air temperature and the latent heat transfer due to the phase change of water vapor. Since the mass of water vapor contained in humid air was order of $10^{-3}$ in mass fraction, the sensible heat transfer of the remaining water vapor could be neglected. Therefore, the total heat transfer rate can be written as

$$q_T = q_l + q_s \cong \dot{m}_a (i_{a,i} - i_{a,o}) \tag{1}$$

where

$$i_{a,i} = c_{p,a} T_{a,i} + L_H w_{a,i} \tag{2a}$$

$$i_{a,o} = c_{p,a} T_{a,o} + L_H w_{a,o} \tag{2b}$$

#### 2.3.2 Energy transfer resistance

The energy transfer resistance between the air and coolant is given by

$$R = \frac{(i_a - i_c)/c_{p,a}}{q_T} \tag{3}$$

based on the enthalpy difference of air and coolant. The average temperature of coolant is used to obtain the average enthalpy of coolant.

#### 2.3.3 Frost layer density

With the frost thickness estimated through a photograph and the amount of frost, the frost layer density is calculated from the following equation:

$$\rho_f = \frac{m_f}{V_f} \tag{4}$$

where

$$m_f = \dot{m}_a \int_0^t (w_{a,i} - w_{a,o})\, dt \tag{5}$$

The volume of frost is estimated by calculating the total volume of frost deposited on the tubes and fins

$$V_f = X_{f,F}\left[A_F - \frac{\pi\{(D_o + 2X_{f,t})^2 - D_o^2\}}{4}\right] + X_{f,t} A_{t,o} \tag{6}$$

## 3. Results and Discussion

In this section, the effects of design factors (fin pitch and fin arrangement) and operating conditions (temperature, humidity, and velocity of air stream) on the frost growth and thermal performance of the fin-tube heat exchanger are

**Table 1** Coil geometry of fin-tube heat exchanger.

<table>
<tr><th></th><th>Component</th><th colspan="2">Spec.</th><th>Component</th><th>Spec.</th></tr>
<tr><td rowspan="5">Geometric Conditions</td><td>Number of steps</td><td colspan="2">2</td><td>Number of rows</td><td>2</td></tr>
<tr><td>Transverse tube spacing, mm</td><td colspan="2">27</td><td>Longitudinal tube spacing, mm</td><td>30</td></tr>
<tr><td>Tube ID, mm</td><td colspan="2">6</td><td>Tube OD, mm</td><td>8</td></tr>
<tr><td>Tube length, mm</td><td colspan="2">370</td><td>Fin pitch, mm</td><td>20</td></tr>
<tr><td>Fin type</td><td colspan="2">Flat type</td><td>Fin thickness, mm</td><td>0.2</td></tr>
<tr><th>Symbol</th><th>Fin spacing, mm</th><th colspan="2">Operating condition</th><th colspan="2">Fin arrangement</th></tr>
<tr><td>●*</td><td>20.0</td><td colspan="2">Baseline</td><td colspan="2" rowspan="4">Inline type</td></tr>
<tr><td>☆</td><td>10.0</td><td colspan="2">Baseline</td></tr>
<tr><td>★</td><td>7.5</td><td colspan="2">Baseline</td></tr>
<tr><td>✡</td><td>5.0</td><td colspan="2">Baseline</td></tr>
<tr><td>○</td><td>20.0</td><td colspan="2">Baseline</td><td colspan="2" rowspan="4">Staggered type</td></tr>
<tr><td>△</td><td>10.0</td><td colspan="2">Baseline</td></tr>
<tr><td>□</td><td>7.5</td><td colspan="2">Baseline</td></tr>
<tr><td>◇</td><td>5.0</td><td colspan="2">Baseline</td></tr>
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* : Baseline design condition

**Table 2** Baseline operating conditions.

| | | | |
|---|---|---|---|
| Air inlet temperature, ℃ | 6 | Air inlet humidity, % | 70 |
| Air inlet velocity, m/s | 1.0 | Coolant mean temperature, ℃ | −30 |

examined. The specifications of the test heat exchanger are shown in Table 1, and the baseline operating conditions in Table 2.

### 3.1 Frost growth

The temporal variation of the thickness of frost formed on the fin and tube for the inline type fin-tube heat exchanger with fin spacing of 20 mm for baseline operating conditions is illustrated in Fig. 3. In this figure, the average thickness of frost given by eq. (7) is also presented:

$$X_{f,av}=\frac{X_{f,F}A_F+X_{f,t}A_{t,o}}{A_F+A_{t,o}} \tag{7}$$

The thickness of frost formed on the fin and tube as shown in Fig. 3 is average values formed on the first and second row. It is shown that the frost thickness on tube surface is greater than the one on fin. Since the surface temperature of tube is lower than that of fin, the water vapor flux onto the tube surface has a larger value than that onto the fin.

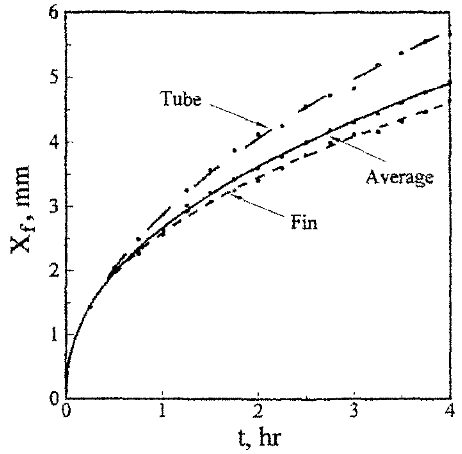


**Fig. 3** The variation of frost thickness with time.

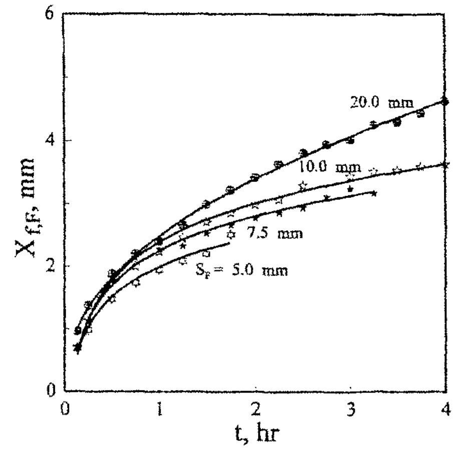


**Fig. 4** The variation of frost thickness with respect to fin pitch.

### 3.2 Effects of design factors

#### 3.2.1 Effects of fin pitch

The variations of the frost thickness formed on the fin surface and the frost density with respect to fin pitch are represented in Figs. 4 and 5. respectively. The higher velocity of air flowing between fins results from the smaller fin spacing due to more blockage of air flow passage with time. The higher air velocity will cause higher frost surface temperature. Therefore, the gradients of temperature and water vapor in the frost layer become steeper, and the more water vapor are diffused into the frost layer. Hence decreasing fin spacing causes frost layer to be thinner and denser.

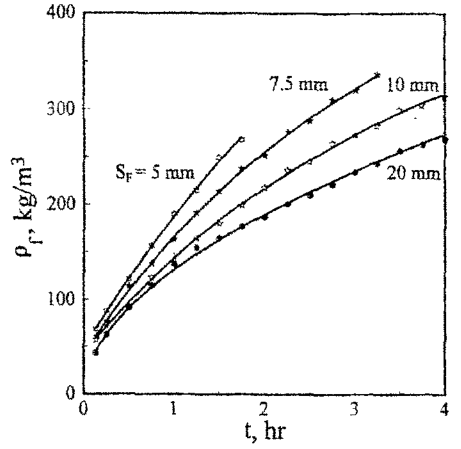


**Fig. 5** The variation of frost density with respect to fin pitch.

Figure 6 depicts the variation of heat transfer rate with fin spacing. There is little change in the amount of heat transfer with time in the case of a large fin spacing, but the significant change of it with time is noticed in a small fin spacing. The heat transfer rate increases at the initial stage of frost formation. This results from the fact that, as Hayashi et al., 1977 asserted, the ice-like frost nuclei generated during the crystal growth period acts as a small fin which results in the increase in roughness and surface area. During the frost layer growth period, frost nuclei grow to form a porous frost layer, and the thermal insulation effect of the frost layer increases because it acts like a thermal insulator. Hence, heat transfer rate decreases. During the latter term of the frost layer growth

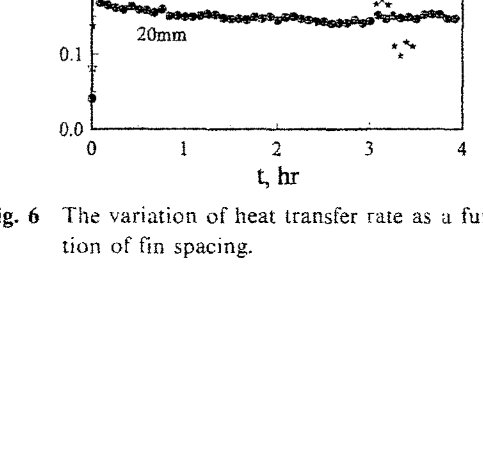


**Fig. 6** The variation of heat transfer rate as a function of fin spacing.

period, however, the heat transfer rate increases and reaches its maximum value due to the increase of the air velocity and frost thermal conductivity. However, as the thermal resistance of the frost layer increases and heat transfer area decreases, heat transfer rate decreases again. Since the period during which nuclei of frost act like small fins is very short compared with the total operating time of heat exchanger, it partially agrees with the results of Aoki et al., 1985, who reported that the heat transfer rate decreased at the initial stage of frosting and then increased. Previous works (Gatchilov & Ivanova, 1979; Hosoda & Uzuhashi, 1967; and Stoecker, 1957) have also stated that the heat transfer rate increased to a maximum value and then decreased. Therefore, the present results partially agree with the previous results (Gatchilov & Ivanova, 1979; Hosoda & Uzuhashi, 1967; and Stoecker, 1957). As fin spacing decreases, the heat transfer rate increases, and the time required to reach a maximum value becomes shorter.

To set up the criteria for the optimal operation of a fin-tube heat exchanger, the blockage ratio ($BR$) is introduced and defined as

$$BR = 2\frac{X_{f,F}}{S_F} \times 100 \ [\%] \tag{8}$$

It represents the fraction of the air flow passage which is blocked by the frost layer. The blockage ratio increases with decreasing fin spacing. Especially, in the case of 5 mm fin spacing, the flow passage is almost fully blocked after 2 hours, and in the case of 7.5 mm fin spacing, more than 80 percent of the flow passage is blocked after 3 hours.

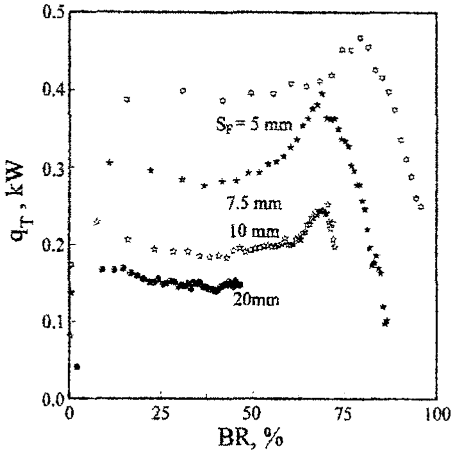


**Fig. 7** The variations of heat transfer rate with blockage ratio for several fin spacings.

Figure 7 represents the variation of heat transfer rate with the blockage ratio. It shows that when the blockage ratio is less than 10%, the heat transfer rate increases during the initial stage of frosting by the influence of frost nuclei. Then the heat transfer rate decreases, due to the influence of thermal resistance of the frost layer, until the blockage ratio reaches about 35~45%. After that, it increases to a maximum value and then decreases with a relatively constant gradient irrespective of fin spacing. Therefore, it is desirable to operate the fin-tube heat exchanger at the particular value of blockage ratio at which the heat transfer rate has its maximum value. The maximum allowable thickness of frost layer can be represented as follows:

$$X_{f,F,max} = \frac{S_F}{2} BR_{max} \tag{9}$$

where the maximum allowable blockage ratio, $BR_{max}$, is derived from the least square method

$$BR_{max} = 89.1 - 2.24 S_F \tag{10}$$

where the applicable range of eq. (10) is $5 \leq S_F \leq 20$ mm.

### 3.2.2 Effect of fin arrangement

Figure 8 depicts the variation of the ratio of the time-averaged total heat transfer rate of a staggered fin-tube heat exchanger to that of an inline type heat exchanger. It shows that heat transfer rate of a heat exchanger with a staggered fin array increases about 17% compared with the inline type heat exchanger. This is due to the fact that the heat transfer coefficient on the air side is enhanced with the staggered fin array. The case with fin spacing of 5 mm shows less enhancement in heat transfer. This is attributed to the fact that the blockage effect of the flow passages increases due to the frost layer formed on the fins. Figure 9 represents the time required to reach a maximum value of a heat transfer rate for each fin array. One can conclude that the thermal performance of

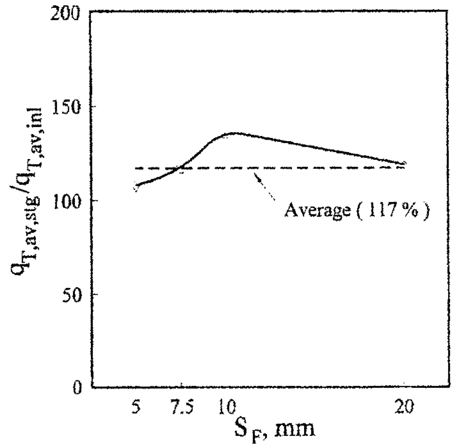


**Fig. 8** The variation of the ratio of time averaged total heat transfer rate of a staggered fin-tube heat exchanger to that of an inline type heat exchanger as a function of fin pitch.

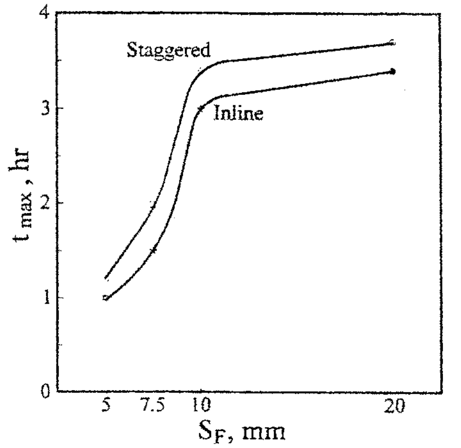


**Fig. 9** Comparison of the time required to reach a maximum total heat transfer rate between staggered and inline fin-tube heat exchanger.

**Table 3** Operating conditions.

| Experimental varialbe | Air inlet temperature, ℃ | Air inlet humidity, % | Air inlet velocity, m/s | Coolant mean temperature, ℃ |
|---|---|---|---|---|
| Baseline | 6 | 70.0 | 1.00 | −30 |
| Temperature | 10 | 53.3 | 1.00 | −30 |
| | 8 | 61.0 | 1.00 | −30 |
| | 4 | 80.5 | 1.00 | −30 |
| Humidity | 6 | 55.0 | 1.00 | −30 |
| | 6 | 63.0 | 1.00 | −30 |
| | 6 | 77.0 | 1.00 | −30 |
| Velocity | 6 | 70.0 | 0.50 | −30 |
| | 6 | 70.0 | 0.75 | −30 |
| | 6 | 70.0 | 1.50 | −30 |

the heat exchanger with a staggered fin arrangement is better compared to the one with an inline type fin array since the time required to reach a maximum heat transfer rate is longer in the case with a staggered fin array. Another important factor to evaluate the heat exchanger performance is the magnitude of the pressure drop of the air. The results also show that the pressure drop of air in the staggered fin-tube heat exchanger has a value about 1-2 $mmH_2O$ higher than that of the inline type heat exchanger. Therefore, the staggered fin-tube heat exchanger may be better than the inline type heat exchanger in the case that the increase of 1-2 $mmH_2O$ in air side pressure drop does not cause a severe reduction of air flow rate.

### 3.2.3 Effects of operating conditions

In this section, the effects of temperature, humidity and velocity of the air stream on the frost growth and energy transfer resistance are examined. The operating conditions of a fin-tube heat exchanger are summarized in Table 3. In the

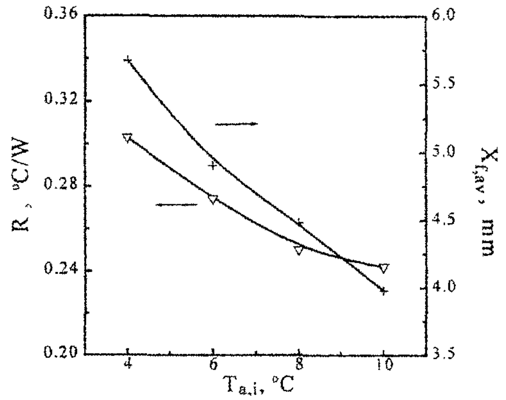


**Fig. 10** The variation of $R$ and $X_{f,av}$ with inlet air temperature.

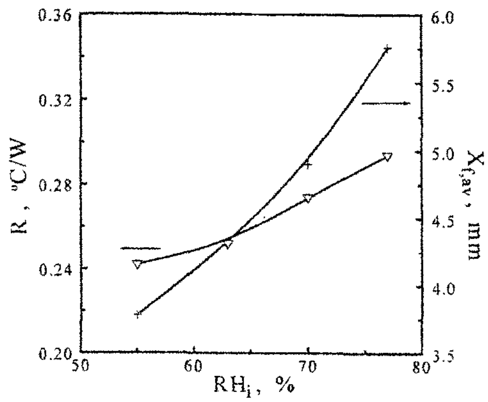


**Fig. 11** The variation of $R$ and $X_{f,av}$ with inlet air relative humidity.

following discussion, the energy transfer resistance($R$) represents an average value during the frosting time and $X_{f,av}$ means the overall average frost thickness at 200 minutes.

### 3.2.4 Effects of inlet air temperature

Figure 10 shows the effects of inlet air temperature on the frost growth and energy transfer resistance. A frost layer with large density and thin thickness is formed with increasing air temperature. This is due to the fact that the amount of vapor transferred into frost layer increases with increasing temperature gradient and increasing vapor pressure gradient in the frost layer. This results from the fact that the vapor condensed onto the frost surface has more contributions to the frost density over the frost thickness. The energy transfer resistance decreases with increasing inlet air temperature since the dense and thin frost has lesser insulation effect inside the frost layer.

### 3.2.5 Effects of inlet air humidity

Figure 11 shows the effects of inlet air humidity on the frost growth and energy transfer resistance. The higher inlet air humidity gives rise to an active mass transfer although it does not cause the heat transfer to increase at the same rate as the mass transfer, the frost layer grows rapidly and temperature gradient in the frost layer becomes small. Therefore, the frost layer becomes thick and the frost density becomes low. Hence, the conductive energy transfer resistance of the frost layer increases and the driving force of the heat transfer between the air and frost surface decreases due to increasing temperature of the frost surface. Based on the above results, the energy transfer resistance has a large value with increasing inlet air humidity.

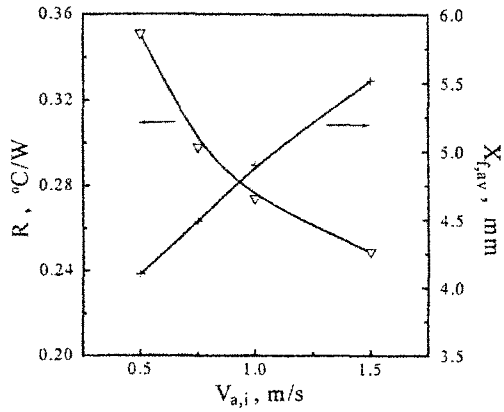


**Fig. 12** The variation of $R$ and $X_{f,av}$ with inlet air velocity.

### 3.2.6 Effects of initial inlet air velocity

Figure 12 shows the effects of inlet air velocity on frost growth and energy transfer resistance. Frost thickness and density increase with increasing inlet air velocity. These phenomena can be attributed to the fact that the increase of the mass transfer rate and temperature gradient in the frost layer results from a large amount of mass and energy transfer due to high velocity of air. The energy transfer resistance has a small value with high inlet air velocity because the convective resistance decreases with increasing air velocity.

## 4. Conclusions

In this study, the effects of various factors (fin pitch, fin arrangement, air temperature, air humidity, and air velocity) on the frost growth and thermal performance of a flat plate fin-tube heat exchanger have been examined under the frosting condition. The conclusions from the present work are as follows:

(1) The frost layer formed on the tube surface is thicker than the one on the fin surface.

(2) With decreasing fin pitch, the frost layer is thinner and denser.

(3) There is little change in the amount of heat transfer with time in the case of a large fin pitch, but the significant change of it with time is noticed for small fin spacings.

(4) The thermal performance of the heat exchanger is closely related to the blockage ratio of air flow passage due to frost growth, and the maximum allowable blockage ratio was obtained for several fin spacings.

(5) The staggered fin-tube heat exchanger may be better than the inline type heat exchanger in the case that the increase of 1-2 $mmH_2O$ in air side pressure drop does not cause a severe reduction of air flow rate.

(6) The energy transfer resistance decreases with the increase of inlet air temperature and velocity and with decreasing inlet air humidity.

## Acknowledgements

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