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Thermofluid characteristics of frosted finned-tube heat exchangers

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Abstract

The performance of frosted finned-tube heat exchangers of different fin types is investigated by experiments in this paper. The effects of the air flow rate, the air relative humidity, the refrigerant temperature, and the fin type on the thermofluid characteristics of the heat exchangers are discussed. The time variations of the heat transfer rate, the overall heat transfer coefficient, and the pressure drop of the heat exchangers are presented. The heat transfer rate, the overall heat transfer coefficient, and the pressure drop for heat exchangers with re-direction louver fins are higher than those with flat plate fins and one-sided louver fins are. The amount of frost formation is the highest for heat exchangers with re-direction louver fins.

1. Introduction

Heat exchangers are used in many engineering applications to facilitate heat transfer between two fluids. Frost formation on heat exchanger surfaces is a major problem in low temperature devices such as refrigerator-freezers, freezers, and heat pumps. The processes of frost formation process are very complicated phenomena involving simultaneous heat and mass transfer. They are affected by many factors including air humidity, air temperature, air velocity and the cooling surface. Mao et al. [1] and Le Gall and Grillot [2] attempted to determine the empirical correlations for frost formation, while Storey and Jacobi [3] examined the effect of vortices on the frost growth rate. In spite of several decades of intensive research, many important characteristics of frost growth cannot yet be accurately predicted. Moreover, few experimental studies have been performed in the range of low humidity conditions, which indicate that the dew point of the incoming air is near or below the freezing point of water, these being typical air conditions in winter. As for the effects of air temperature on frost growth, Trammell et al. [4] found that the air temperature has a smaller influence than other parameters on the frost formation. Brian et al. [5] examined experimentally the air temperature on the frost formation. They showed that the frost thickness decreases with an increase in the air temperature. But, Tao et al. [6] and Han and Ro [7] concluded from their predictions that the frost thickness increases with an increase in the air temperature. As for the effect of air velocity on frost growth, many studies have been performed but as yet, no consensus has been reached. Tokura et al. [8] and Yonko and Sepsy [9] concluded from their experiments

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Nomenclature

A	area	t	time
C_p	specific heat	U	overall heat transfer coefficient
F	correction factor	ϕ	relative humidity
i	enthalpy		
$\dot{m}$	mass flow rate	*Subscripts*	
ΔP	pressure drop	a	air
Q	volume flow rate	i	inlet
$\dot{Q}$	heat transfer rate	o	outlet
T	temperature	r	refrigerant

that the air velocity has a negligible influence on the frost growth rate. Recently, Na and Webb [10–12] have shown that the air is supersaturated at the frost surface, as opposed to the previous assumptions that the water vapor is saturated at the frost surface.

Niederer [13] performed experiments to investigate the frosting and defrosting effects on the heat transfer of heat exchangers. He found that frost accumulation on the coil surface reduces the air flow rate and the heat exchanger capacity. Heat exchangers with wider fin spacing are affected to a lesser degree than those with closer fin spacing under frosting conditions. Heat exchangers with variable fin spacing perform better when compared with heat exchangers with constant fin spacing. Kondepudi and O'Neal [14] in a review paper discussed the effects of frost on the fin efficiency, the overall heat transfer coefficient, the pressure drop, and the surface roughness of extended surface heat exchangers. They suggested that more work is needed to determine the effects of frost on the fin performance. Kondepudi and O'Neal [15] experimentally studied the performance of louvered finned-tube heat exchangers under frosting conditions. They reported that the frost growth, the pressure drop, and the energy transfer coefficient increase with the air humidity, the air velocity, and the fin density. Kondepudi and O'Neal [16] compared the performance of finned-tube heat exchangers with different fin configurations. It was found that the louvered fin type has the best thermal performance, followed by the wavy fin type and the flat fin. Senshu et al. [17] and Yasuda et al. [18] investigated the performance of heat pumps under frosting conditions experimentally and theoretically. It was found that the speed of frost formation could be assumed constant when the air and the refrigerant conditions are specified. Raising the refrigerant evaporation temperature can reduce the possibility of frosting. The heat transfer coefficient of the air is not significantly affected by the frost. Oskarsson et al. [19,20] presented equations, correlations, and models for evaporators operating with dry, wet, and frosted surfaces. Rite and Crawford [21,22] investigated the effects of various parameters on the rate of frost formation and the performance of a domestic refrigerator-freezer finned-tube evaporator. They concluded the frosting rate increases for higher air humidities, temperatures, flow rates and lower refrigerant temperatures. The UA-value and the air side pressure drop increase as frost forms on the evaporator coil for a constant air flow rate. Kondepudi and O'Neal [23,24] proposed a model to evaluate the performance of finned-tube heat exchangers under frosting conditions. The results predicted by the model were compared with experimental data for the frost growth, the air side pressure drop, and the energy transfer coefficient. In general, this model underpredicts the experimental data by 15–20%. Thomas et al. [25] and Chen et al. [26] investigated the frost characteristics on heat exchanger fins. Recently, Yan et al. [27,28] investigate experimentally the different operating conditions on the characteristics of heat transfer and pressure drop heart in plate fin-and-tube heat exchangers with or without frosting conditions.

This paper is intended to investigate the effects of frost formation on the performance of finned-tube heat exchangers with different fin types.

2. Experimental apparatus

The experimental setup is shown in Fig. 1. The major components are the psychrometric room, the heat exchanger test section, the wind tunnel, the refrigerant system, and the data acquisition system. The psychrometric room provides conditioned air of constant temperature and relative humidity in the range −10 °C to 45 ± 0.3 °C and 40% to 95 ± 3%, respectively, for the required test conditions. Conditioned air from the psychrometric room is drawn through the wind tunnel by a 2.24 kW centrifugal fan with an inverter. Finned-tube heat exchangers of two tube rows with three different fin types are used in this paper to investigate the effects of frost on the performance of heat exchangers. The fin types and geometrical parameters of the heat exchangers

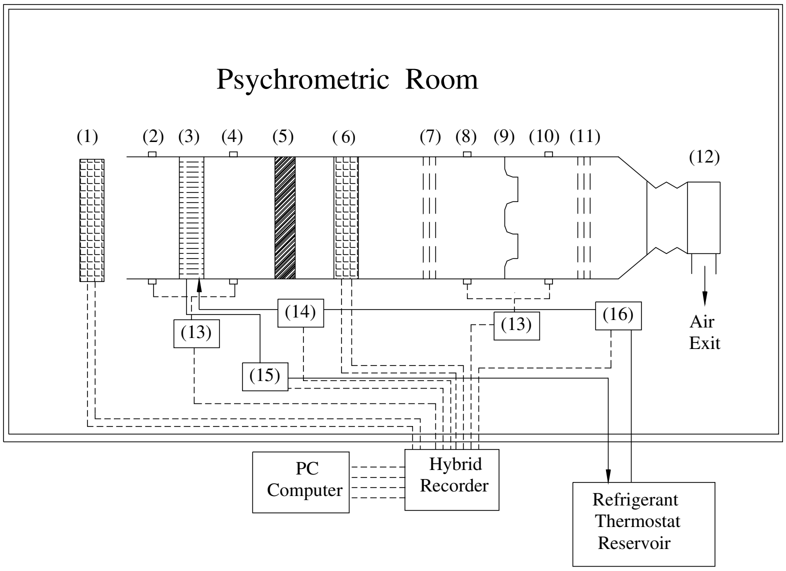

Fig. 1. Schematic of the experimental apparatus. (1) Dry and wet-bulb thermometer; (2) inlet pressure measuring section; (3) heat exchanger test section; (4) outlet pressure measuring section; (5) air mixer; (6) dry and wet-bulb thermometer; (7) flow straightener; (8) nozzles inlet pressure measuring section; (9) multiple nozzles; (10) nozzles outlet pressure measuring section; (11) flow straightener; (12) exhaust fan system; (13) micro manometer; (14) refrigerant inlet RTD; (15) refrigerant outlet RTD; (16) magnetic flow meter.

are shown in Fig. 2 and Table 1. The refrigerant in the tube used in this paper is ethylene glycol water solution. The inlet refrigerant temperature is controlled by means of a thermostat reservoir. The inlet and outlet temperatures of the refrigerant, and the inlet and outlet (dry-bulb and wet-bulb) temperatures of the air are measured by pre-calibrated RTDs (Pt-100) which have an accuracy of 0.2 °C. The measurements of the dry-bulb and wet-bulb temperatures of the air across the heat exchanger are based on ASHRAE 14.1 standard [29] with two psychrometric boxes. The air flow rate is measured by multiple nozzles based on the ASHRAE 41.2 standard [30]. Precision differential pressure transducers with 0.1 Pa resolution are used to detect the pressure drops across the heat exchanger and the multiple nozzles, respectively. The flow rate of air is maintained constant throughout each test by adjusting the speed of the centrifugal fan to keep the pressure drop across the multiple nozzles constant. The refrigerant flow rate is measured by a calibrated magnetic flow meter with 0.002 l/s resolution. The data are recorded every 5 min with the acquisition system that transmits the data to the personal computer for further operation.

The experimental setup described above is used to investigate the thermofluid characteristics of the frosted heat exchangers.

3. Data reduction

The variables measured are the inlet and outlet air dry-bulb and wet-bulb temperatures of the heat

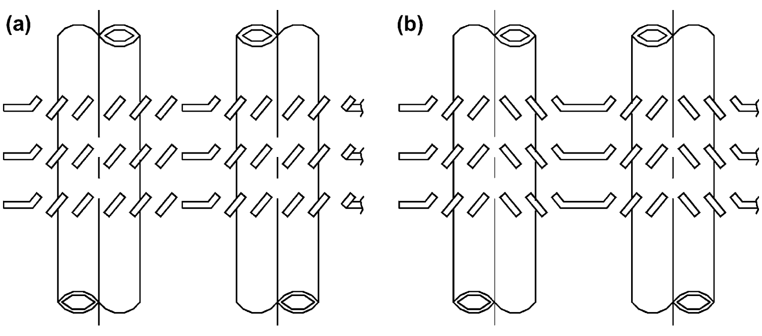

Fig. 2. Schematic of the fin types. (a) One-sided louver fins; (b) re-direction louver fins.

Table 1
Fin geometries

No.	Fin type	Pitch no.	Fin pitch (mm)	Outer diameter (mm)	Width (mm)	Height (mm)	Depth (mm)	Row no.
1	Flat plate	292	2.0	10.3	590	356	37.2	2
2	One-sided louver	292	2.0	10.3	590	356	37.2	2
3	Re-direction louver	292	2.0	10.3	590	356	37.2	2

Tube material: copper; fin material: aluminum; horizontal tube pitch: 25.4 mm, vertical tube pitch: 19.05 mm; tube thickness: 0.35 mm; fin thickness: 0.12 mm.

exchanger, the air flow rate, the pressure drop of the air across the heat exchanger, the inlet and outlet refrigerant temperatures, the flow rate of the refrigerant, and the air pressure drop across the nozzles. The heat transfer rate and the overall heat transfer coefficient of the heat exchanger are obtained from the experimental data.

3.1. Heat transfer rate

Since frosting process includes both sensible and latent heat transfer for the air side, the heat transfer rate of the air side can be calculated by

$$\dot{Q}_a = \dot{m}_a(i_{a,i} - i_{a,o}) \tag{1}$$

where $\dot{Q}_a$ is the heat transfer rate, $\dot{m}_a$ is the mass flow rate, $i_{a,i}$ and $i_{a,o}$ are the enthalpies of the air at the inlet and the outlet of the heat exchanger, respectively.

The heat transfer rate of the refrigerant side can be computed by

$$\dot{Q}_r = \dot{m}_r C_{p,r}(T_{r,o} - T_{r,i}) \tag{2}$$

where $\dot{Q}_r$ is the heat transfer rate, $\dot{m}_r$ is the mass flow rate, $C_{p,r}$ is the specific heat, $T_{a,i}$ and $T_{a,o}$ are the temperatures of the refrigerant at the inlet and the outlet of the heat exchanger, respectively.

Before conducting the experiments, preliminary tests show that the differences between $\dot{Q}_a$ and $\dot{Q}_r$ for heat exchangers without frost formation are within 5%. Since there are difficulties to calculate $\dot{Q}_a$ for air with temperature below 0 °C , therefore, $\dot{Q}_r$ is adopted for the results presented in this paper.

3.2. Overall heat transfer coefficient

The overall heat transfer coefficient U can be expressed as

$$U = \frac{\dot{Q}}{A(\mathrm{LMTD})F} \tag{3}$$

where F is the correction factor, A is the area, and the log-mean temperature difference (LMTD) is defined as

$$\mathrm{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{4}$$

Table 2
Summary of estimated uncertainties

Parameter	Uncertainty
Q_r	±0.04 l/s
ρ_r	±2 kg/m^3
T_r	±0.15 °C
T_a	±0.07 °C
ϕ	±3% RH
$\dot{Q}$	±5.18%
U	±6.6%
ΔP	±5.13%

where

$$\Delta T_1 = T_{a,i} - T_{r,o} \tag{5}$$

$$\Delta T_2 = T_{a,o} - T_{r,i} \tag{6}$$

The uncertainties for the experimental results are calculated according to the procedure outlined by Kline and McClintock [31]. The results of the uncertainty analysis are tabulated in Table 2.

4. Results and discussion

To check accuracy of the present experimental study, a series of limiting cases without frosting conditions was first measured. It is found that the measured Colburn j factor and Fanning f factor agree with those of Ref. [27].

The effects of the air flow rate, the air relative humidity, and the refrigerant temperature on the heat transfer rate, the overall heat transfer coefficient, and the air side pressure drop of the frosted heat exchangers are investigated. Three fin types are used in this paper, i.e. flat plate fins, one-sided louver fins, and re-direction louver fins. The experiments are performed at the baseline conditions as listed in Table 3 except for the variable being evaluated. The effects of the air flow rate on the performance of the heat exchangers with flat plate fins, one-sided louver fins, and re-direction louver fins are shown in Figs. 3–5, respectively. In Figs. 3(a) and (b), 4(a) and (b), 5(a) and (b), it is found that the heat transfer rate and the overall heat transfer coefficient increase as the

Table 3
Baseline testing conditions

Parameter	Value
Inlet refrigerant temperature	−15 °C
Inlet air temperature	5 °C
Inlet air relative humidity	70%
Air flow rate	24 m^3/min
Refrigerant flow rate	4.17 l/min

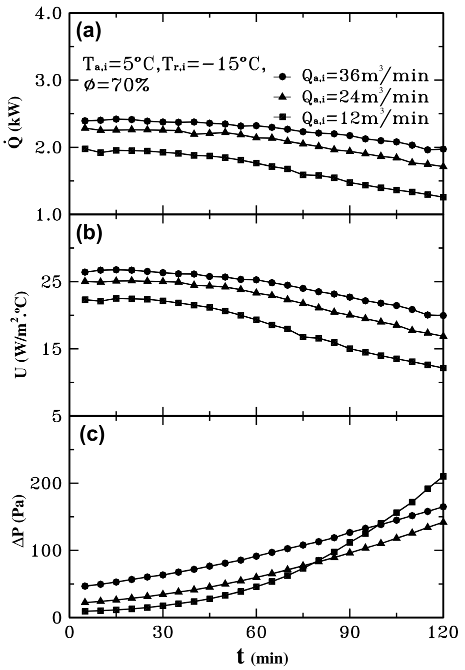

Fig. 3. The effects of the air flow rate on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with flat plate fins.

air flow rate increases. The experimental data indicate that a higher air flow rate leads to a higher pressure drop initially as shown in Fig. 3(c). This is similar to the trends of dry heat exchangers. However, after 100 min the pressure drop for $Q_a = 12$ m^3/min becomes the largest. This phenomenon is also found for heat exchangers with one-sided louver fins and re-direction fins as shown in Figs. 4(c) and 5(c). It indicates that the amount of frost formation increases as the air flow rate decreases for the heat exchangers. This is because the surface of the heat exchanger becomes colder for a lower air flow rate due to a lower heat transfer rate. Thus, the heat transfer rate and the overall heat transfer coefficient decrease as the air flow rate decreases. In the separate experimental runs, it is observed that the frost accumulation is highest for heat exchangers with re-direction louver fins.

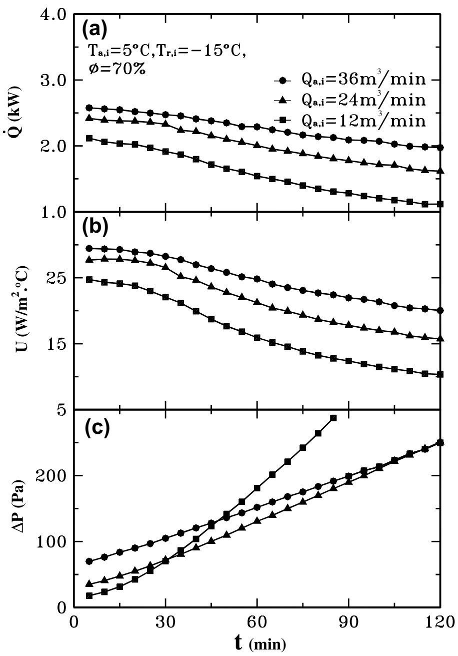

Fig. 4. The effects of the air flow rate on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with one-sided louver fins.

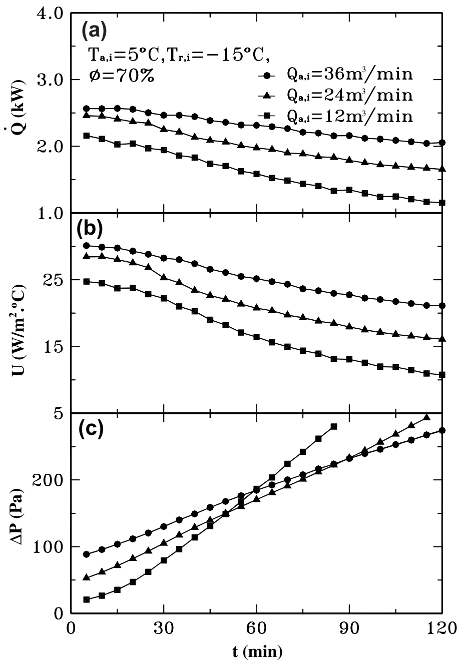

Fig. 5. The effects of the air flow rate on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with re-direction louver fins.

Figs. 6–8 show the effects of the air relative humidity on the heat transfer and the pressure drop characteristics of the heat exchangers with different fin types. Initially, the heat transfer rate and the overall heat transfer coefficient are higher for higher relative humidities as shown in Figs. 6(a) and (b), 7(a) and (b), 8(a) and (b). As the relative humidity increases, there is higher moisture content and the frost accumulation increases. Consequently, the heat transfer rate and the overall heat transfer coefficient drop more quickly for higher relative humidities. After a period, the heat transfer rate and the overall heat transfer coefficient become higher for lower relative humidities. The effects of the relative humidity on the pressure drop are presented in Figs. 6(c), 7(c) and 8(c). The pressure drop across the heat exchanger increases as the relative humidity increases. It is noted the pressure drop is largest for the re-direction louver fins as compared with other types of fins. The rate of the pressure drop increases in the following order: flat plate fins, one-sided louver fins, and re-direction louver fins. Thus, the amount of frost formation is largest for heat exchanger with re-direction louver fins.

The effects of the refrigerant temperature on the heat transfer rate, the overall heat transfer coefficient, and the pressure drop are presented in Figs. 9–11. As the refrigerant temperature decreases, the surface temperature of the heat exchanger decreases and the amount of frost formation increases. Therefore, it has a larger heat

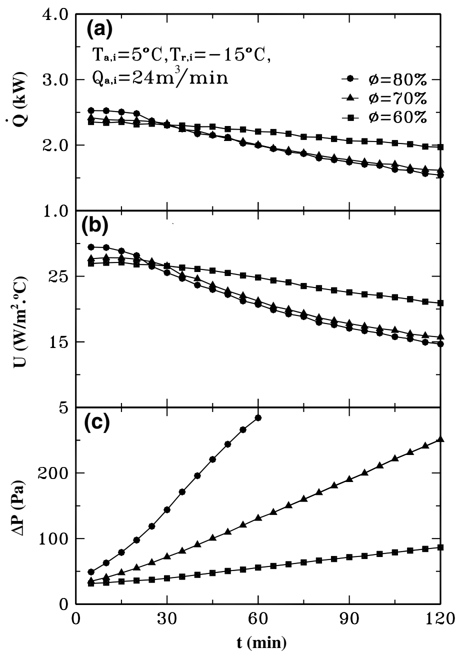

Fig. 7. The effects of the air relative humidity on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with one-sided louver fins.

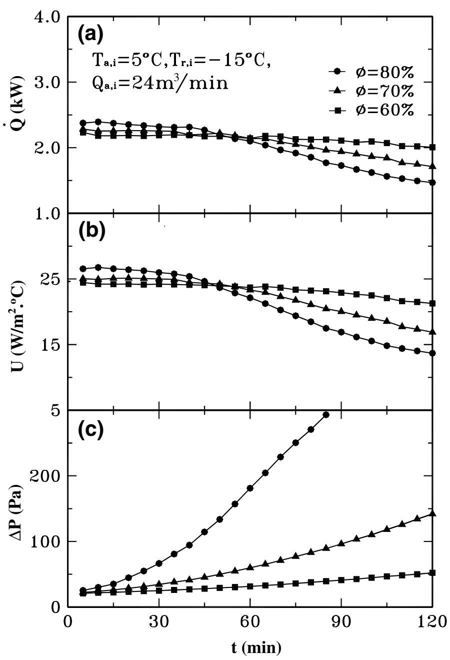

Fig. 6. The effects of the air relative humidity on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with flat plate fins.

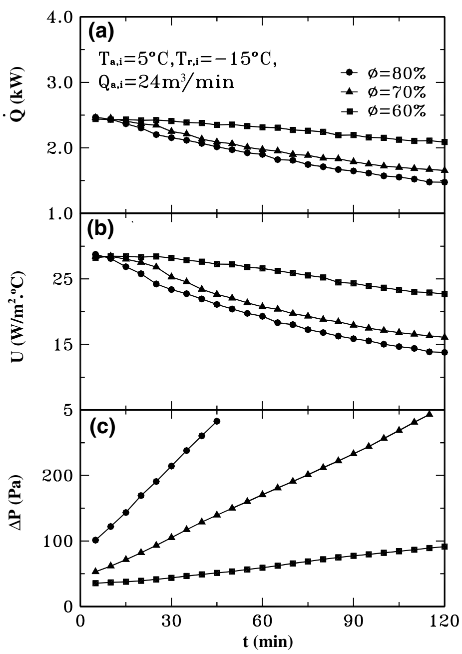

Fig. 8. The effects of the air relative humidity on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with re-direction louver fins.

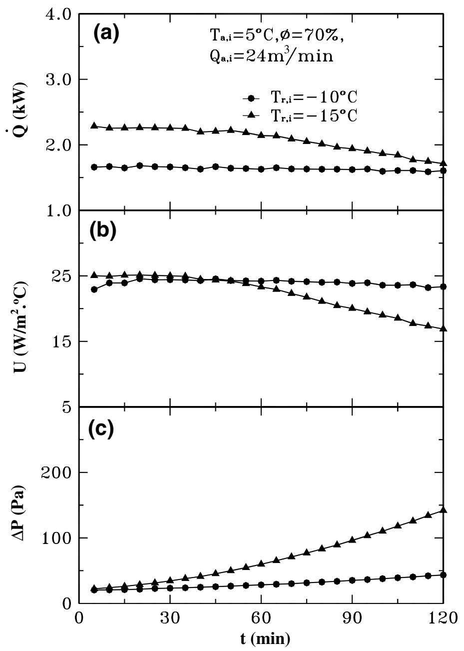

Fig. 9. The effects of the refrigerant temperature on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with flat plate fins.

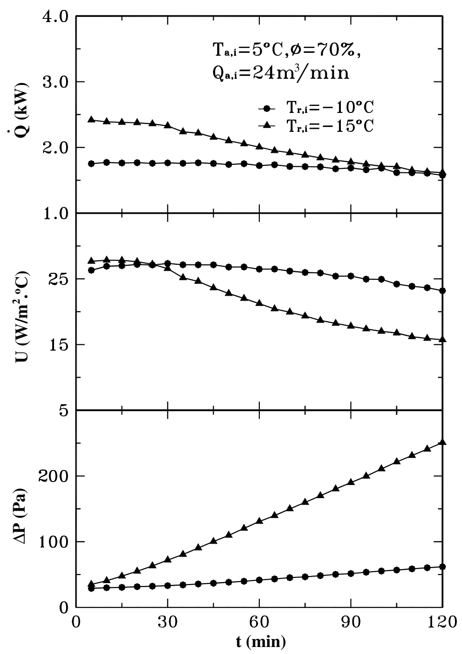

Fig. 10. The effects of the refrigerant temperature on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with one-sided louver fins.

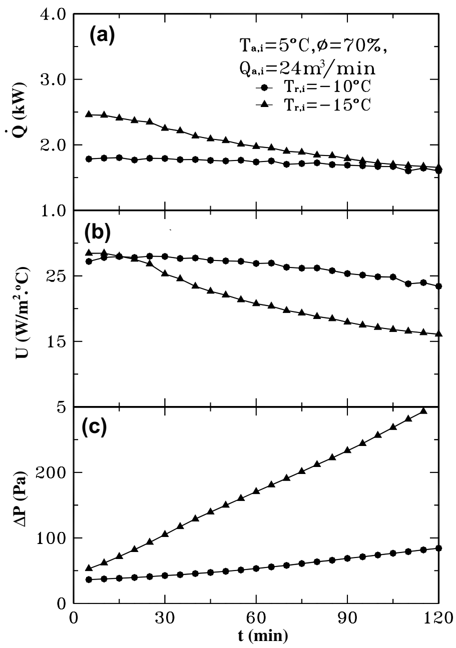

Fig. 11. The effects of the refrigerant temperature on (a) the heat transfer rate, (b) the overall heat transfer coefficient, and (c) the pressure drop of the heat exchanger with re-direction louver fins.

transfer rate and a higher pressure drop. It is also noted that for a lower refrigerant temperature, the heat transfer rate declines more quickly and the pressure drop increases more rapidly. It is because of a higher frost formation, the frost insulates and blocks the heat exchanger more quickly. The pressure drop increases more rapidly for re-direction louver fins followed by one-sided louver fins and flat plate fins.

5. Conclusions

The performance of frosted finned-tube heat exchangers with three different fin types has been investigated by experiments. The fin types used in this work are flat plate fins, one-sided louver fins and re-direction louver fins. The effects of the air flow rate, the air relative humidity, and the refrigerant temperature on the heat transfer rate, the overall heat transfer coefficient, and the air side pressure drop have been discussed. In the early stage of the experiments, the heat transfer rate increases as the air flow rate and the relative humidity increase, while it decreases as the refrigerant temperature increases. The heat transfer rate drops more quickly as the relative humidity increases or as the air flow rate and the refrigerant temperature decrease. The rate of frost formation and the rate of pressure drop increase as the relative humidity increases or as the air flow rate and the refrigerant temperature decrease. The pressure drop increases most rapidly for heat exchanger with re-direction fins. Thus, its amount of frost formation is the largest.

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