THERMAL PERFORMANCE OF A SCALED-UP LOUVERED FIN HEAT EXCHANGER WITH UNEQUAL LOUVER PITCH

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ABSTRACT

We analyzed the thermal performance of louvered fin heat exchanger with equal and unequal louver pitch which continuously increased by 20% from air inlet area to the redirection region and difference of thermal performance of the louvered fin heat exchanger with unequal louver pitch was compared with that of equal louver pitch. The frost blocking of the spaces between louvers at front side of the louvered fin heat exchanger with unequal louver pitch was delayed and the time, the heat transfer rate of louvered fin heat exchanger reached 50% of the maximum heat transfer rate, of the unequal louver pitch louvered fin heat exchanger was also delayed than that of the equal louver pitch louvered fin heat exchanger. The operating time of louvered fin heat exchanger with unequal louver pitch was longer than that of equal louver pitch and the total heat transfer until reaching 50% of the maximum heat transfer rate increased at 18% than that of equal louver pitch.

NOMENCLATURE

c_p	[J/kg°C]	Specific heat at constant pressure
F_{th}	[mm]	Fin thickness
F_p	[mm]	Fin pitch
h_{sv}	[J/kg]	Latent heat of sublimation
L_l	[mm]	Length of louver
L_θ	[°]	Louver angle
$\dot{m}$	[kg/s]	Mass flow rate
N_l	[-]	Number of louver
$\dot{Q}$	[W]	Heat transfer rate
RH	[%]	Relative humidity
S_1	[mm]	Length of air inlet region
S_2	[mm]	Length of redirection region
T	[°C]	Temperature
U	[m/s]	Air velocity
w	[g/g$_{DA}$]	Absolute humidity

Subscripts

a	Air
DA	Dry air
f	Frost
in	Inlet
lat	latent
out	Outlet
r	Refrigerant
sen	Sensible

INTRODUCTION

Louvered fin heat exchangers are widely used in heat pump systems and automotive air conditioning systems. They are compact and, in general, exhibit excellent thermal performance. However, when a louvered fin heat exchanger is used as an evaporator under frosting conditions, frost develops on the surface of the fin. Frost formation decreases the thermal performance of the heat exchanger due to a reduced airflow and increases the thermal resistance between the air and heat exchanger. Thus, a new design of the louvered fin is required to delay frost formation in the louvered fin heat exchanger.

There have been numerous investigations of the thermal performance of louvered fin heat exchangers [1–4]. Park et al.[1] studied the frost behavior in the louvered fin heat exchanger using a scaled-up model. They reported the spaces between louvers at the redirection region were the first to become to blocked by frost formation, followed by blockage at the air inlet region. Xu et al.[2] compared the thermal performance of the vertical tube microchannel heat exchanger with the horizontal tube microchannel heat exchanger. They reported that the thermal performance of the vertical tube microchannel heat exchanger was better than the horizon tube because of little water retention in the microchannel heat exchanger. Moallem et al.[3] studied the thermal performance of the louvered fin heat exchanger according to surface treatment. Hsieh and Jang evaluated a louvered fin heat exchanger with variable louver angle under dry condition using a three-dimensional numerical analysis method. However, these studies considered primarily louvered fin heat exchangers with a constant pitch between louvers. Here we report a heat exchanger with louvered fins in which the pitch successively decreases by 20% from the air inlet region to the redirection region. We analyzed and compared the thermal performance and frost formation of louvered fin heat exchangers with equal and unequal louver pitch.

EXPERIMENTS

Figure 1 shows photographs of the two arrays of ten louvered fins being compared in this study: one with equal louver pitch(=constant louver pitch between louvers) and the other with an unequal louver pitch(=different louver pitch

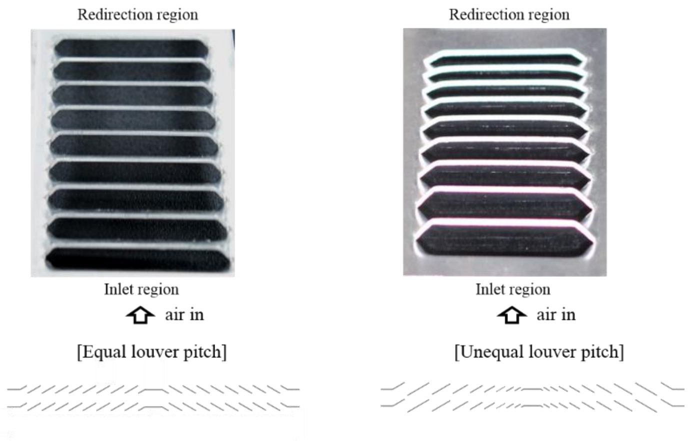

Figure 1. Arrays of ten louvered fins with equal and unequal louver pitch.

between louvers). Table 1 lists the geometric parameters of the louvered fins. As shown Figure 2, the experimental set up used in this study is as described by Jhee *et al*. [5]; the experimental conditions are listed in Table 2. The experimental set up consisted of a climate chamber to control air temperature and humidity, a circulation section controlled by a blower with an inverter, a test section for evaluating the heat and mass transfer of the louvered fin heat exchanger and a refrigerator for supplying a clod refrigerant(ethylene glycol solution with a mass ratio of 4:6). Average air temperature was measured using nine T-type thermocouple in the duct. A humidity sensor(accuracy ±1%) was installed at the center of the duct after the air humidity gradient was minimized for measuring air humidity. The pressure drop through the louvered fin heat exchanger was measured with four pressure taps connected to a pressure transducer at the inlet and outlet of the louvered fin heat exchanger. An RTD sensor(accuracy ±0.15℃) was inserted in to the flowing refrigerant for measuring the temperature of the refrigerant. The pitch of the scaled-up louvered fin heat exchanger was 10 times that of the 19-FPI(Fin per inch) prototype(a conventional louvered fin heat exchanger). Table 2 shows the experimental conditions in this study. These are the heat pump conditions. The air velocity was 0.7 m/s, which is the smallest velocity for minimized natural heat convection in the louvered fin heat exchanger. The energy

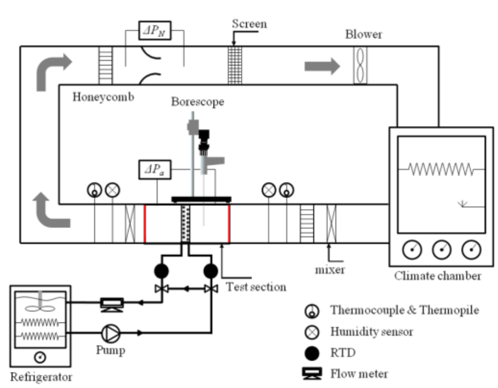

Figure 2. Experimental set up.

Table 1. Geometric parameters of the louvered fins.

S_1 [mm]	S_2 [mm]	F_{th} [mm]	F_p [mm]	N_l [−]	L_l [mm]	L_θ [°]
10	14	0.8	13.37	18	60	30

balance between the air side and refrigerant side was maintained within 5% during the experiment in this study, as described in the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 33–78[6], with the exception of the initial unbalanced period due to the thermal inertia of the heat exchanger. Following the experiments, the frost mass was measured and the louvered fins were photographed. The mass transfer rate in the heat exchanger was calculated using the airflow rate and the humidity of the inlet and outlet air, as follows:

$$\dot{m}_f = \dot{m}_a (w_{a,in} - w_{a,out}) , \qquad (1)$$

where $\dot{m}_a$ is the airflow rate, and $w_{a,in}$ and $w_{a,out}$ are the absolute humidity of the inlet and outlet air, respectively.

The total heat transfer rate at the air side is given by the sum of the sensible and latent heat transfer rates:

$$\begin{aligned} \dot{Q}_a &= \dot{Q}_{sen} + \dot{Q}_{lat} \\ &= \dot{m}_a c_{p,a} (T_{a,in} - T_{a,out}) + \dot{m}_a (w_{a,in} - w_{a,out}) h_{sv} \end{aligned} \qquad (2)$$

where $\dot{Q}_{sen}$ is the sensible heat transfer rate, $\dot{Q}_{lat}$ is the latent heat transfer rate, $c_{p,a}$ is the specific heat of air at a constant pressure, and h_{sv} is the latent heat of sublimation.
The heat transfer rate at the refrigerant side was calculated from the refrigerant flow rate, specific heat, and the temperature of the inlet and outlet air at the refrigerant side, as follows:

$$\dot{Q}_r = \dot{m}_r c_{p,r} (T_{r,out} - T_{r,in}) , \qquad (3)$$

where $\dot{m}_r$ is flow rate of the refrigerant and $c_{p,r}$ is the specific heat of the refrigerant. The total heat transfer rate was determined based on the arithmetic mean of the air side and the refrigerant side.

RESULTS AND DISCUSSION

Figure 3 shows the heat transfer rate of the louvered fin heat exchangers as a function of time. The heat transfer rates of both heat exchangers were similar up to 40 min; however, after this time, the heat transfer rate of the heat exchanger with the unequal louver pitch was higher than that with a constant pitch.

Table 2. Experimental conditions.

T_a[℃]	U_a[m/s]	RH[%]	T_r[℃]
4	0.7	72	-20

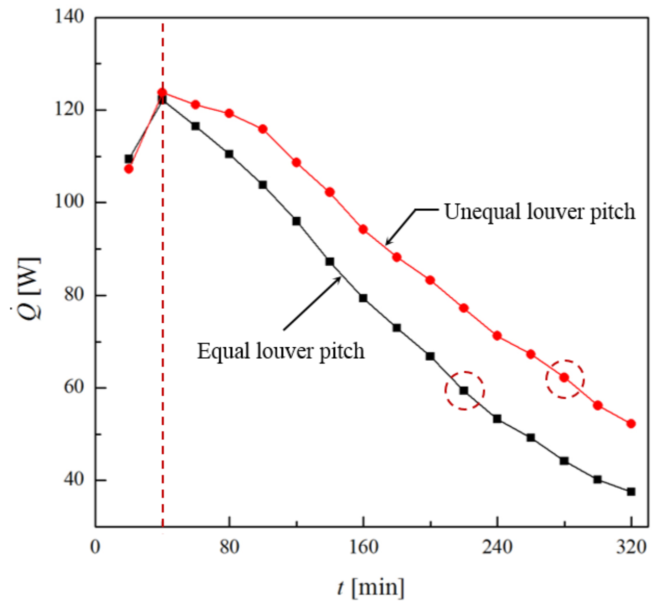

Figure 3. Heat transfer rate according to louvered fin types.

The heat transfer rate of the heat exchanger with an unequal louver pitch dropped by 50% after 280 min, whereas the heat transfer rate of the heat exchanger with the constant pitch. Figure 4 shows the frost formation on the louvers when the heat transfer rate dropped by 50%. The blocking ratio between louvers due to frost formation was 92% for the heat exchanger with the constant louver pitch and 84% for the heat exchanger with the unequal louver pitch. Because the blocking ratio of the heat exchanger with the unequal louver pitch was smaller, there was greater airflow in this heat exchanger. Thus, the thermal performance of the heat exchanger with the unequal louver pitch was enhanced.

Figures 5(a) and 5(b) show side views of the two heat exchangers. The frost layers at the front and rear sides were non-uniform for the heat exchanger with the constant louver pitch, but were more uniform for the heat exchanger with the unequal louver pitch. In addition, the blocking ratio between the louvers was higher for the heat exchanger with the constant louver pitch, in contrast to the results for the front side. At the rear side, the blocking ratio between louvers was 68% for the

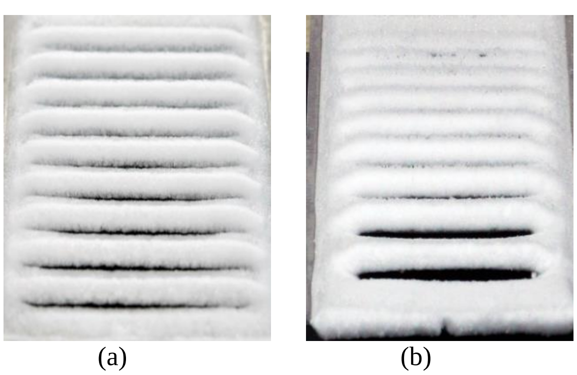

Figure 4. Frost formation on the louvered fins at the front side: (a) equal and (b) unequal louver pitch.

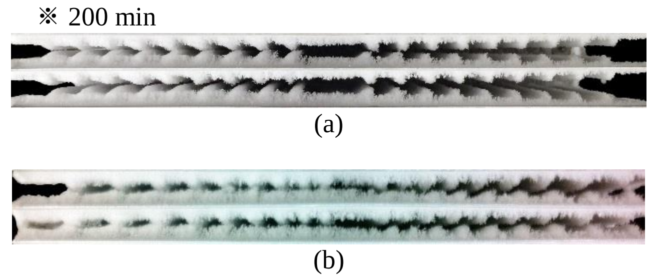

Figure 5. Side-view of frost formation on the heat exchangers: (a) equal and (b) unequal louver pitch.

heat exchanger with the constant louver pitch, which indicates that the air that entered the heat exchanger did not flow between louvers were not completely blocked. It follows that the louvers at the rear side were ineffective due to the blockage between louvers at the front side. However, frost formation in the spaces between the front-side louvers was delayed for the heat exchanger with the unequal louver pitch, enabling better effectiveness of the louvers at the rear side. This improved the uniformity of frost growth, and resulted in the heat exchanger with the unequal louver pitch having better thermal performance. In this study, the thermal performance and frost behaviors of the louvered fin with unequal louver pitch which continuously increased by 20% from air inlet area to the redirection region were investigated. In the future work, it is need to the comparison of the thermal performance and frost behaviors of the louvered fin heat exchanger with unequal louver pitch which continuously increased and decreased by 20% from air inlet area to the redirection region.

CONCLUSION

We analyzed the frost behavior and thermal performance of louvered fin heat exchangers with equal and unequal louver pitches. For the heat exchanger with the unequal louver pitch, the time for the heat transfer rate to fall by 50% was longer than that for the heat exchanger with the equal louver pitch, because blocking of the spaces between the louvers at the front side was delayed. As a result, the total heat transfer rate for the heat exchanger with the unequal louver pitch was 18% larger than that with an equal louver pitch. Furthermore, the frost growth was more uniform for the heat exchanger with the unequal louver pitch.

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