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Thermal performance of microchannel heat exchangers according to the design parameters under the frosting conditions

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ABSTRACT

The thermal performance and uniformity of frost growth between the front- and rear-sides of microchannel heat exchangers was investigated as a function of the depth of the heat exchanger, the pitch of the fins, and the pitch of the channels. The microchannel heat exchangers exhibited frost formation that was concentrated on the front-side, resulting in significant frost growth non-uniformity between the front- and rear-sides, which led to rapid performance degradation. During the defrosting cycle, compared with natural convection, forced convection defrosting resulted in the unit retaining less water between the fins, leading to improved thermal performance. However, after a number of frosting and defrosting cycles, the relative improvement of forced convection became smaller. Thinner heat exchangers (i.e., with a smaller coil depth) exhibited more uniform frost growth between the front- and rear-sides of heat exchangers, and exhibited approximately 5% increase in the heat transfer rate per unit volume. A larger fin pitch showed less airflow blockage due to larger air pathway. In addition, heat exchangers with a larger fin pitch showed more uniform frost growth, as well as approximately 6% increase in the average heat transfer rate. Heat exchangers with a larger channel pitch exhibited a larger heat transfer rate during the early stages of the experiment; however, the frost growth was less uniform between the front- and rear-sides, leading to a rapid reduction in the heat transfer rate mid-way through the experiment.

1. Introduction

Electric vehicles have recently been commercialized; because there is considerably less waste heat than vehicles with internal combustion engines, they require additional heat sources for passenger heating. This can be achieved using heat pumps with the same vapor compression cycle as is used for air conditioning. Since both heating and cooling are then possible from a single module, heat pumps can be used in electric vehicles for combined air-conditioning and heating systems. When heat pumps operate in heating mode, a frost layer often forms on the outdoor heat exchanger, leading to the performance degradation of heat pumps. A defrosting cycle is used to prevent the performance loss that accompanies frost growth; however, during this defrosting cycle, additional energy is required to melt the frost layer and the heating system cannot function. Therefore, frost growth should be minimized by careful design of the heat exchanger.

There have been a number of reports of frosting and defrosting phenomena for round-tube heat exchangers [1–3]; however, studies of microchannel (flat-tube) heat exchangers, which are typically used for exterior vehicle systems, have mainly focused on cooling applications, which do not lead to frost growth. Only recently have a small number of studies of frosting in microchannel heat exchangers published [4–9]. Wu et al. [4] attached thermocouples to a parallel-flow evaporator (a microchannel heat exchanger), and investigated the surface temperature distribution of the heat exchanger, imaging the front-side using a charge-coupled device (CCD) array camera. Xu et al. [5] compared the performance of vertical refrigerant flow with horizontal refrigerant flow in a parallel flow evaporator, and reported that the vertical unit showed better thermal performance than the horizontal one because there was less residual water following the defrosting cycles. Moallem et al. [6,7] imaged the front-side of a microchannel heat exchanger to investigate the surface temperature and water retention effects on frost growth. They found that the surface temperature of the fin and the humidity of the air were significant factors in frost formation. Zhang et al. [8] reported empirical correlations of the heat transfer rate (Colburn *j*-factor) and the air-side pressure drop (Fanning friction *f*-factor) during the early stages of frost growth in a parallel-flow heat exchanger as a function of the Reynolds number, fin shapes and fin pitches. However, the correlations only considered the performance of the heat exchanger during the early stages of frost growth. Moallem et al. [9] studied the design parameters for microchannel modules during frost growth. They carried out frosting experiments with simplified geometries and

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Nomenclature

c_p	specific heat of air, kJ/kg · k
L_h	latent heat of sublimation, kJ/kg
$\dot{m}_a$	mass flow rate of air, kg/s
ΔP_a	air-side pressure drop, pa
$\dot{Q}$	heat transfer rate, kW
T	temperature, °C
t	operating time, min
$u_{a,ini}$	initial air velocity, m/s

Subscripts

a	air
d	dry bulb temperature
def	defrosting
f	frosting
in	inlet
ini	initial
lat	latent heat
out	outlet
sen	sensible heat
w	wet bulb temperature

modified the shape of the heat exchanger, imaging the front-side of the modules.

Most studies of frost growth in microchannel heat exchanges have only reported observations of the frost layer on the front-side. Microchannel heat exchangers employ high-density folded fins, and non-uniform frost growth occurs, which is typically concentrated on the front-side of the heat exchanger. The frost growth uniformity between the front- and rear-sides has not yet been examined. Here, we investigate the frost growth uniformity between the front- and rear-sides and the corresponding thermal performance of microchannel heat exchangers. The depth of the heat exchanger, the fin pitch, and the channel pitch were varied, and the frost growth was studied during repeated frosting/defrosting cycles.

2. Experiments

Fig. 1 shows an overview of the experimental apparatus. It consisted of a test section, a recirculation section, a refrigeration section, a constant temperature bath (i.e., a defrosting section), and a climate chamber. The microchannel heat exchanger was installed in the test section, and the recirculation section formed air pathways connecting the other sections. The refrigeration section controlled the temperature and flow rate of the cold refrigerant to investigate frosting, and the constant temperature bath supplied warm refrigerant for defrosting [10,11]. The climate chamber controlled the temperature, humidity, and flow rate of the air. The range of the air temperature was from −10 °C to +50 °C, the relative humidity was controlled in the range 40–95%, and the maximum airflow rate was 65 m^3/min. The blower maintained constant power during the frosting experiments. A bypass air tunnel and air-dampers were used to circulate the air, with or without it passing through the test section during the defrosting experiments. The coolant (refrigerant) used in the experiments was an ethylene glycol/water mixture with a mass ratio of 1:1. The refrigerant-side temperature was measured using a resistance thermometer (RTD) inserted in the refrigerant pipe. The temperature difference between the inlet and outlet of the heat exchanger was small (less than 2.1 °C). Bypass valves were installed at the inlet and outlet of the test section to alternately supply cold refrigerant during frosting cycles and warm refrigerant during defrosting cycles.

Fig. 2 shows a schematic diagram and an optical image of the base-design microchannel heat exchanger. The width of the heat exchanger was W = 300 mm, the height was H = 300 mm, and the depth was D = 20 mm. There was a bypass air gap of 4 mm between the fin part and the test section, located at the left- and right-hand sides of the heat exchanger, as shown in Fig. 1, so that when the fins became completely blocked by the frost layer, humid air could still pass through this gap. It should be noted that the air-side pressure drop was reduced due to the existence of this bypass air gap. The gap between the microchannels (i.e., channel pitch) was 7 mm. Folded-louvered fins with a pitch of 80 fins per decimeter (FPDM) were located in these channels.

To investigate the effects of the design parameters, i.e., the depth of the heat exchanger, the fin pitch, and the channel pitch, we fabricated four heat exchangers. Three holes were drilled in the top of the test section to observe the frost growth on the front-side of the heat exchanger, and another three holes were drilled to observe the frost growth on the rear-side of the heat exchanger. A rigid borescope was inserted through the holes to capture the images [11,12]. In addition, to determine the macroscopic behavior of the frost growth, miniature high-quality webcams were installed on both the front- and rear-side of the test section [12]. The experimental conditions were based on the heat pump operating conditions and airflow of a moving electric vehicle. To compare the performance of the heat exchangers, repeated frosting/defrosting cycles were investigated. Warm refrigerant was supplied to melt the frost layer, and defrosting occurred with both forced and natural convection of the air. Natural convection was used to simulate defrosting when the vehicle was stationary, and forced convection was used to simulate defrosting when the vehicle was moving. The temperature, humidity, and velocity of the air were maintained constant during the experiments, and the warm refrigerant was only supplied to melt the frost layer.

To evaluate the heat transfer rate, the air-side sensible and latent heat transfer rates were calculated from

$$\dot{Q}_a = (\dot{Q}_{sen} + \dot{Q}_{lat}) = \dot{m}_a C_{p,a}(T_{a,in} - T_{a,out}) + \dot{m}_a L_h(w_{a,in} - w_{a,out}) \quad (1)$$

and the refrigerant-side heat transfer rate was calculated using

$$\dot{Q}_{ref} = \dot{m}_{ref} C_{p,ref}(T_{ref,out} - T_{ref,in}) \quad (2)$$

The heat balance between the air-side and refrigerant-side was matched to within 5% [13], except during the early and final stages of the experiments. The air-side temperature was measured using a 5 × 5 grid of thermocouples installed in the inlet and outlet of the test section, where the cross-sectional area of the test section was 0.09 m^2. The air-side velocity was measured using a multi-pitot tube coupled with an air-side pressure transducer (KIMO Instruments). The accuracy of the multi-pitot tube was ±2.5%, the accuracy of the thermocouple grid was ±0.14 °C, and the accuracy of the humidity sensors was ±1%. The experiments were repeated six times to statistically assess the error in the data, and the level of confidence was 95%. The uncertainty of the average heat transfer rate was 3.2% until the middle stage of the experiments (around t_f = 16 min) [14,15]. The uncertainty increased to 8.7% during the latter stages of the experiments due to the reduced airflow rate because of the formation of frost.