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COMPARISON OF FROST AND DEFROST PERFORMANCE BETWEEN MICROCHANNEL COIL AND FIN-AND-TUBE COIL FOR HEAT PUMP SYSTEMS

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ABSTRACT

This paper presents a comparison of frost and defrost cycling performance between a microchannel heat exchanger with louvered fin and a fin-and-tube heat exchanger with straight fins employed as outdoor coils of a 4-ton (48,000 Btu/hour) heat pump system. In addition to temperature, pressure, and flow rate measurements taken at various locations of the systems, the fin-base and tube wall surface temperature were also recorded by using fine-gauge pre-calibrated thermocouples on the coils. Further, load cells were used to measure the mass of frost accumulation during heating tests.

Data showed that the frosting time of the microchannel heat exchanger is more than 50% shorter than for the fin-and-tube heat exchanger, which is chosen as the baseline system. The average heating capacity and system performance are also lower for the system with microchannel heat exchangers. Higher frost growth rate in microchannel heat exchangers are due to residual water in the coil after defrost, augmented temperature difference between air and the surface of the heat exchanger, and preferential frost nucleation sites on the louvered fins and microchannel tubes. Blowing nitrogen on the microchannel coil after defrost removed any visible water retained in the coil after the defrost cycle but the cycle time increased only by 4% with respect to wet and frost conditions. The cycle time of the same microchannel starting with dry condition was about 60% longer than the cycle time in wet and frost conditions.

1. INTRODUCTION

Microchannel heat exchangers are becoming quite popular as an alternative to bulky fin-and-tube heat exchangers. For a desired capacity, a microchannel heat exchanger is smaller in size than a fin-and-tube heat exchanger, resulting in less refrigerant charge in the system. However, microchannel heat exchangers are not widely used as outdoor coil in heat pump systems because of they build up frost rapidly during winter.

There are numerous studies in the literature that focus on frost and defrost heat transfer performance of fin-tube heat exchangers. One of the early studies on the effect of frosting on the heat pump system adopting fin-and-tube heat exchanger is by Martinez and Aceves (1999). The authors developed a model for heat exchanger with frost build up and integrated it with a heat pump system simulation model. Characteristics of the heat pump, such as COP, pressure drop, frost thickness and frosting time were studied. Kim and Groll (2003) reported a comparison between microchannel and a fin-and-tube heat exchangers when used as an outdoor coil in a heat pump system. The study included both cooling and heating tests. The authors reported frosting/defrosting time and the heating capacity of the heat pump with both coils. Effect of other variables such as heat exchanger inclination, and fins per inch (FPI) were also studied. The authors concluded that microchannel heat exchangers have a shorter frosting time compared to fin-and-tube heat exchangers, and the time decreased even further with each cycle due to residual water retained at the end of each defrost cycle. Kim and Groll investigated the effect of the orientation of the microchannel tubes and they concluded that slightly inclined tubes in the direction of the air flow was beneficial with respect to a straight vertical configuration. The work by Kim and Groll focused on the comparison of frosting times and heating performance but their study did not provide sufficient information on frost growth rate and frost build up on microchannel heat exchangers. Xia et al. (2006) studied the performance of a flat-tube heat exchanger with louvered fins in frosting conditions, which resembles a microchannel heat exchanger. An overall heat transfer coefficient was obtained for the heat exchanger using the experimental temperatures and flow rate. Frost thickness was measured

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using images by a CCD camera, and frost weight was obtained by using a high precision scale. Xia et al. developed a numerical model, which was also experimentally validated, to predict the frost thickness and blockage ratio.

Several researchers in the field have concluded that water retention is the main cause that penalizes the cycling time of microchannel heat exchangers. However, there is not a clear experimental study that supports this theory and isolates this effect. This is due to the complexity of measuring the frost mass build up on the coil during actual operating conditions. It is also difficult to vary one parameter at a time while maintaining temperatures and pressures constant along the vapor compression cycle because frost growth is a transient phenomenon. Local temperature of fins and wall tubes at various locations in the coil is the key parameter to understand the heat and mass transfer occurring during frost growth and frost melt. A relative comparison between a fin-and-tube coil and a microchannel coil would provide a better understanding of the behavior of microchannel heat exchanger during frost conditions.

The present work investigates the characteristics of a microchannel heat exchanger with louvered folded fins and a fin-and-tube heat exchanger with straight fins under frosting conditions. The study also aims to identify the possible root causes that penalize microchannel heat exchangers operating as the outdoor coil in heat pump systems. A commercially available 4-ton heat pump system was used in the experimental set up and the outdoor unit was placed in an environmental chamber, which was maintained at 35/33°F [1.7/0.6°C] (DB/WB)[‡]. Frost build up on the heat exchangers was measured by weighing the entire unit. Quantities such as air temperatures at the coil inlet and outlet (for indoor coil only), refrigerant temperatures at different locations, and fin and tube surface temperatures at different locations of the heat exchanger were measured. The experimental setup and methodology undertaken is explained in later sections. Results obtained from experiments are also presented in this paper.

2. EXPERIMENTAL SETUP

A 4-ton R22 unitary heat pump was used in the test set up. The unit was placed between two environmental chambers, which were controlled closely to replicate the outdoor and indoor conditions. Indoor and outdoor chambers are conditioned using glycol circuits and electric reheat coils to achieve required temperatures. During frosting tests the outdoor chamber was maintained at ARI test conditions of 35/33°F [1.67/0.56°C] (DB/WB). The indoor chamber was maintained at 70°F DB [21.1°C] and the air flow rate across the indoor coil was about 1800 cfm [0.85 m³/s] .

The outdoor unit was placed on three load cells, each with a maximum capacity of 250 lb [113.3 kg] and precision of 1% of full scale. The load cells were calibrated in-situ with standardized weights and the accuracy on the load cells was determined to be ±0.6 lb [28.3 g]. The refrigerant flow rate was measured using a coriolis type flow meter, which was mounted on the unit, and it had a full scale reading of 2400 lb/hr [0.302 kg/s]. Voltage and current readings in the supply lines to the indoor fan, outdoor fan and compressor were measured in order to calculate the power drawn. Pressure transducers and thermocouples were connected at different locations in the refrigerant pipelines and their range and accuracy is given in Table 1. In addition to the refrigerant and air temperatures, fin and tube surface temperatures were also measured using 16 fine gage T-type thermocouples. The thermocouples were attached evenly across the heat exchanger face. All the thermocouples were calibrated in-situ. The in-situ calibration resulted in improving the accuracy on the thermocouples to within ±0.4°F [0.2°C]. A videoscope is used to record the frost growth on the outdoor coil. Figure 1(a) shows the experimental set up, while Figure 1(b) the close view of the microchannel heat exchanger section that shows the placement of thermocouples. Inlet air conditions were sampled using air samplers to obtain the drybulb and wetbulb temperatures. Conditions leaving the indoor coil were also monitored to calculate the capacity of the unit from the air side. The wet bulb measurement was obtained by using the “wet sock” method.

Table 1: Range and Accuracy of Refrigerant Line Instruments

<i>Refrigerant Line Instrument</i>	<i>Range</i>	<i>Accuracy (Calibrated)</i>
Pressure Transducer	1000 psig (6.89 MPa)	±1%
Thermocouple	-50 F to 200 F (-45.5°C to 93°C)	±1 F (±0.56°C)
Mass flow meter	2400 lb/hr (0.302 kg/s)	±0.1%

[‡] ARI Standard 210/240-2005, Performance rating of unitary air conditioning and air-source heat pump equipment, H2 test condition

The defrost cycle was initiated by a demand defrost control which measures the difference between the coil surface temperature and air inlet temperature to the coil. If the temperature difference is higher than a certain threshold, which depends on the compressor specification for its safe operation, and if it stays above the limit for about 4.5 minutes, then the defrost cycle is initiated. The condenser fan was turned off during the defrost cycle while the indoor fan was allowed to run. The unit was tested under frosting conditions with a fin-tube heat exchanger as baseline outdoor coil. Then, the fin-and-tube was replaced with a microchannel coil heat exchangers, which was designed for similar capacity. Geometry details of both coils are given in Table 2. The microchannel coil had about 6% less face area compared to the fin-tube coil for a similar capacity of 4 tons.

Table 2: Parameters for Fin-Tube and Microchannel Coils

	Fin and Tube HX	Microchannel HX
Coil height	38 in (0.965 m)	36.38 in (0.924 m)
Coil width	61.3 in (1.557 m)	60.3 in (1.532 m)
Coil depth	0.866 in (22 mm)	0.708 in (18 mm)
Tube material	Copper	Aluminum
Fin material	Aluminum	Aluminum
FPI	16	17
Circuits/Passes	6	1
Number of Rows	36	N/A
Number of Ports	N/A	23

Several tests were conducted in this study; they are summarized in the test matrix of Table 3. Fin-and-tube frosting data obtained from test 1 are for dry-coil conditions (Dry), that is, for the first frost/defrost cycle of the unit. The entire outdoor unit was left in the room at ambient temperature overnight and slowly brought to ARI conditions without turning on the compressor or the fans. Test 2 measures the performance of the fin and-tube after the heat pump system reaches steady periodic (Cyclic) conditions, that is, after several frost/defrost cycles in which the behavior of the heat pump becomes periodic. During periodic cycles, the time between defrosts is quite constant. Microchannel heat exchangers are studied in test 3 to 5. Dry and steady periodic conditions are test 3 and 4. Finally, in test 5, the unit was turned off manually at the end of the defrost period and high pressure nitrogen gas was blown through the coil to remove residual water. The unit was turned back on and allowed to run one entire cycle.

Table 3: Test Matrix showing different tests performed and Indoor coil flow rate

Test #	Fin-Tube Heat Exchanger		Microchannel Heat Exchanger		
	Dry	Cyclic	Dry	Cyclic	N ₂
1	X				
2		X			
3			X		
4				X	
5					X

3. DATA REDUCTION

After periodic cyclic conditions were achieved, the unit runs continuously between heating and defrost modes. The heating capacity of the unit was calculated from the air side measurements as follows:

$$\dot{Q}_a = \dot{m}_a C_p (T_{a,id,o} - T_{a,id,i}) \quad 1$$

The average capacity and power input are calculated as given by equation (2) respectively.

$$\bar{\dot{Q}} = \frac{\int_0^t \dot{Q}_a dt}{t} \quad \bar{P} = \frac{\int_0^t P dt}{t} \quad 2$$

The unit's coefficient of performance (COP) was calculated as the ratio of the air side capacity and the total power input to the compressor and outdoor fan. Indoor fan power was not accounted for in the COP because only the

outdoor coil was replaced with between tests while the indoor coil was the same for all tests. The COP was calculated as:

$$COP = \frac{\dot{Q}_a}{\dot{P}_{comp} + \dot{P}_{od, fan}} \quad 3$$

The EER of the unit is then calculated with the standard conversion formula:

$$EER = 3.412 \cdot COP \quad 4$$

Figure 1: Schematic of the Experimental Setup: (a) Chambers with Unitary Equipment and Load cells; (b) Close view of Microchannel with thermocouples and videoscope

4. RESULTS AND DISCUSSIONS

Experimental results are presented in dimensionless form using constants to normalize the plots. Dimensionless quantities preserve the same trend and characteristics of the original data and they protect the proprietary nature of the tests. In this paper, the data are analyzed under four sections; cycle time (frost/defrost), frost weight, coil temperatures, and system performance. The experimental results are also grouped under two different conditions of the coil; dry coil and steady periodic (referred as cyclic). Dry coil was obtained by leaving the unit at the ambient temperature overnight at about 77°F. The outdoor chamber was then cooled by the auxiliary conditioning equipment of the psychrometric facility and the ARI H2 test standard conditions were achieved inside the room. Dry coil test refers to the first cycle of the heat pump system after the unit was off overnight. Cyclic test refers to the behavior of the heat pump systems after a few cycles from the first one when near steady-periodic performance had been achieved.

4.1. Frost Cycle Time

Frosting time and defrost time for each test are given in Table 4. The microchannel coil starting dry, that is, test #3 in Table 3, had a frost cycle time of 39m 43 s and a defrost time of about 2m 50s. The cyclic, steady periodic behavior of the same unit during test #4 had a frost cycle time of 23m 13s while the defrost time was 3m 30s. It is clear that the microchannel coil operating at steady state in frosting conditions switches to defrost mode quicker than the same coil starting dry. The cycling time of the microchannel starting dry was about 60% longer than the same coil starting wet after a defrost cycle. The frost time of microchannel heat exchanger does not increase significantly when the retained water is blown off of coil using pressurized nitrogen gas (test #6). Nitrogen gas was blown through the coil at the end of the defrost cycle and cleared the coil of all visible water. Then, when the unit was turned on, the microchannel heat exchanger accumulated frost at a growth rate similar to the cyclic case (without nitrogen blowing) as described later in section 4.2 of this paper. With Nitrogen blowing about 95% of the water was removed from the coil while in cyclic operations without Nitrogen blowing about 85% of the water drains from the coil. This suggests that water retention is not the primary factor that impacts the frosting cycle time. The thin layer of water or small water droplets is believed to act as preferential nucleation sites for the frost formation. The

significant amount of water that is visibly retained in the coil during normal operation does not practically contribute to shortening the heating period.

Fin-tube coil starting dry on the other hand had a frost cycle time of 49m 13s and a defrost time of 2m 02s. Thus a microchannel heat exchanger with dry starting conditions had a frost cycle time which is 25% less than a fin-tube coil starting dry. Microchannel heat exchanger operating under steady periodic conditions (test #4) performed even worse with about 50% less frost cycle time compared to the fin-and-tube heat exchanger operating under steady periodic conditions (test #2).

Table 4: Frost and Defrost Times for Different Cases

	Fin-Tube-Dry (Test #1)	Fin-Tube-Cyclic (Test #2)	Microchannel Dry (Test #3)	Microchannel-Cyclic (Test #4)	Microchannel-N ₂ Blow (Test #5)
Frost time	49m 13s	41m 07s	39m 43s	23m 13s	24m 25s
Defrost time	2m 02s	2m 32s	2m 50s	3m 30s	3m 02s

It must be noted that the frost times shown in Table 4 are a combination of the effect of water retention in the heat exchangers, lower saturation pressure inside microchannel tubes, and refrigerant distribution. The comparison between test #5 (microchannel cyclic) and test #6 (microchannel N₂ blow) aims to isolate the effect of excess water retention at the end of the defrost cycle.

In addition, the capacity degradation of fin-and-tube and microchannel coils at the end of the heating (frosting) period is not the same. The capacity reduction for the fin-tube heat exchanger at the end of heating period is around 16% for the dry coil and 19% for the wet coil. For the microchannel heat exchanger, the heating capacity degradation was about 25%. The defrost controller of the unit measured the coil surface temperature at outlet to initiate the defrost cycle, regardless of whether the heat exchanger is a fin-and-tube or a microchannel type. This suggests that a different defrost control should be adopted in case of microchannel heat exchangers.

4.2. Frost Weight

Weight of frost growing on the heat exchanger surfaces was measured using load cells. The load cells were sensitive to several system related factors in addition to the weight of the frost. These factors included the refrigerant shift when switching between heating and defrost modes, the absence of downward fan force when turned off during defrost periods and dynamic stress caused by the transient loading of the flexible duct that connected the unit to the indoor coil. A calibration procedure was carried out before the experiments to estimate the influence of these factors on the readings from the load cells. Several calibrated weights were placed on the unit to simulate the frost growth. Air conditions were well above freezing point at about 43°F and the unit was manually controlled to switch from heating to defrost modes without having real frost build up on the coil. From the calibration data, a curve fit approach was used to estimate the hysteresis of the load cell and the load induced by the flexible duct. These were found to be within ± 0.6 lb.

The frost mass that is accumulated on the fin-and-tube and microchannel heat exchanger is shown in Figure 3. Figure 3(i) shows the frost build up of fin-and-tube and microchannel heat exchangers starting dry. Frost build up on fin-and-tube and microchannel heat exchangers starting under wet coil conditions is shown in Figure 3(ii). The data are presented in form dimensionless frost weight per unit area of the face of coil. Fin-and-tube heat exchanger under both dry and steady periodic conditions accumulated approximately same amount of frost before going into defrost. However, the steeper slope of the microchannel curve shows that the rate of frost growth of the microchannel was significantly higher than the rate of frost growth of the fin-and-tube. During the first 5 minutes of the heating cycle, the rate of frost growth of fin-and-tube is about 0.006 lb/(lb-ft²-min) while the rate is about 0.0086 lb/(lb-ft²-min) for microchannel. This suggests that initially frost growths about 43% faster in microchannel than in fin-and-tube heat exchangers. In both fin-and-tube and microchannel heat exchangers the frost growth under dry starting conditions flattened before going into defrost. Fin-and-tube has the lowest frost growth rate and it builds up about 0.06 lb/(lb-ft²) in about 40 minutes (50m for dry starting condition). Microchannel heat exchanger accumulated about 0.065 lb/(lb-ft²) of frost in approximately 24 minutes (40m for dry starting condition). Both tests #4 and #5 gave similar

results. The frost weight measurements of Figure 3 also indicate that the frost accumulates during the first part of the heating cycle, that is, approximately during the first 10 minutes.

The frost accumulation was recorded by using a videoscope positioned at about 1 foot from the lower header of the microchannel heat exchanger, as shown in figure 1b. The camera was kept still for the entire cycle and the frost appeared as shown in Figure 4. The images were captured at different time during frost growth. The growth of frost layer begins from the microchannel tubes and progress irregularly toward the fins.

Figure 3: Dimensionless Frost Weight per Unit Coil Face Area: (i) Dry Starting Condition; (ii) Steady Periodic Condition

Figure 4: Stages of Frost Growth in a Microchannel HX (Defrost to Defrost). [Images taken at following times: (a) start of frost cycle; (b) 3 mins; (c) 7 mins; (d) 11 mins; (e) 15 mins; (f) 20 mins; (g) 23 mins; and (h) just after defrost completion]

4.3. Coil Surface Temperatures

Frost formation on the heat exchanger depends on the heat exchanger surface temperature. Lower surface temperature initiates frost growth early (Tao et al, 1993). Early initiation of frost growth will lead to a reduction in time between defrost cycles. Coil surface temperature measurements for both fin-tube and microchannel coils are made using fine gage (40-gage) thermocouples. Figure 5 shows the fin and tube temperatures at two different locations (near the inlet and the outlet) for both fin-tube and microchannel cyclic tests. Thermocouples were located about one foot from the inlet header and outlet headers of the microchannel heat exchanger. The inlet header thermocouple is close to the location of the videoscope used to take the pictures shown in figure 1b. The local

microchannel tube wall and corresponding fin surface temperatures are significantly lower at the inlet of the coil. The temperature difference between inlet air and the surface of the coil is higher for the microchannel coil than for the fin-and-tube coil. This high temperature difference impacts the frost growth rate but further investigation is necessary to quantify this effect.

Tube and fin thermocouples are located near the circuit inlet and outlet for the fin-tube heat exchanger. Similar temperature profiles at the inlet and the outlet (as shown in Fig 5(a)) suggests that frost grows uniformly on the entire face area of the fin tube coil. On the other hand, for the microchannel heat exchanger, the local temperatures near the inlet and outlet are significantly different, as shown in Fig 5(b). This suggests that the frosting on the microchannel coil is not uniform. Near the inlet header frost begins immediately, but onset of frosting is delayed near the outlet. Visual observation of the microchannel coil during frosting confirms these conclusions. The weight measurements and local temperature curves together suggest that the frost growth pattern in fin-tube and microchannel heat exchangers are quite different. Controlled wind tunnel experiments are planned to investigate the frost mechanism in microchannel coils.

Figure 5: Fin, and Tube Temperatures near Inlet and Outlet Header: (a) Fin-Tube; (b) Microchannel

4.4. Performance Evaluation

The heating capacity and the EER for the unit were calculated using the equations (1) through (4) discussed in section 3. The quantities were normalized by using the maximum capacity and maximum EER for the fin-tube dry run, which was chosen as the baseline case. Figure 6 shows the variation of capacity (Figure 6(a)) and EER (Figure 6(b)) for the unit over the entire frosting cycle, defined as time from termination to termination of the defrost cycle. The data presented are for fin-and-tube dry test, fin-and-tube cyclic test, and microchannel cyclic test. For all the three cases the average capacity increases initially and flattens out in the final stages of frost growth period. The drops of the capacity is due to the defrost stage, at which point the heating capacity of the unit becomes zero. The microchannel heat exchanger also has less capacity and EER compared to fin-tube coil. The average capacity of the microchannel coil over its entire cycle is about 27% less than the dry fin-tube coil and 22% less than the cyclic fin-tube case. The EER of the heat pump system with microchannel coil is also about 13% lower than the same unit with fin-and-tube coil.

5. CONCLUSIONS

A 4-ton heat pump system using a fin-tube outdoor coil was compared with the same unit using a microchannel coil. Tests were conducted at the ARI 210 H2 test conditions and local surface temperatures and weight of the coil were taken in real time during the experiments. The fin and tube heat exchanger had considerably longer time between defrost cycles, and about double than the microchannel coil used in our experiments. Blowing nitrogen on the microchannel coil removed all visible water retained in the coil after the defrost cycle but the time between defrost cycles increased only by 4% with respect to wet and frost conditions. The cycle time of the same microchannel starting with dry condition was about 60% longer than the same coil starting wet. It was also observed that the frost growth rate in the microchannel heat exchanger is higher compared to the fin-tube coil. The surface temperatures are

lower, promoting high heat transfer rate but also high frost growth rate. The frost growth patterns were also found to be quite different for fin-tube and microchannel coils. The average capacity of the microchannel coil over its entire cycle was lower by about 22% with respect to the fin-tube coil. The EER of the heat pump system with microchannel coil is also about 13% lower than the same unit with fin-and-tube coil

Figure 6: Performance data for different cases: (a) Average Capacity in dimensionless form; (b) Average EER in dimensionless form

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NOMENCLATURE

$\dot{m}$ = Mass Flow Rate (Kg/s)

C_p = Specific heat of air (KJ/Kg-K)

T_a = Air Temperature ($^{\circ}$ C)

$\dot{Q}_a$ = Airside Capacity (KW)

$\bar{Q}$ = Average Capacity (KW)

$\bar{P}_{comp}$ = Average compressor power (KW)

$\bar{P}_{od, fan}$ = Average outdoor fan power (KW)

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