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November 1984
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PERFORMANCE OF RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS DURING OPERATION WITH FREEZING AND PERIODIC DEFROSTS

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ABSTRACT

In a laboratory study of the performance of residential air-to-air heat exchangers during operation with freezing and periodic defrosts, freezing caused the temperature efficiency of a cross-flow heat exchanger to decrease at a rate ranging from 1.5 to 13.2 percentage points per hour. Much smaller rates of decrease in temperature efficiency, 0.6 to 2.0 percentage points per hour, occurred during tests with a counterflow heat exchanger. The rate of decrease in efficiency depended on the airstream temperatures and humidities and the duration of the period of freezing. The amount of time required to defrost the heat exchanger's core was 6% to 26% of the total operating time. The average temperature efficiency for freeze-defrost cycles ranged from 48% to 64% in tests of the cross-flow exchanger and 70% to 82% in tests of the counterflow exchanger. When the frequency and duration of defrosts were nearly optimal, approximately a ten to fifteen percentage point decrease in average temperature efficiency was attributed to the freezing and required defrosts. The results suggested that the rate of performance deteriorations due to freezing can be reduced by avoiding small airflow passages that can easily be plugged with ice and by designing the exchanger so that condensed water does not drain toward the cold regions of the core. Based on this investigation, suggestions are made for future experimental studies of freezing and for improved control of freeze-protection systems.

Keywords: air-to-air, energy conservation, freezing, heat exchangers, residential, ventilation.
INTRODUCTION

Occupied buildings rely on ventilation, the exchange of indoor air for outdoor air, to help maintain acceptable indoor concentrations of moisture and indoor-generated pollutants. A major drawback of ventilation, however, is the energy required to heat or cool the ventilation air, and this drawback is becoming increasingly important as energy prices rise. A method to reduce the energy requirements associated with ventilation is to minimize the amount of air leakage through the building envelope, mechanically supply and exhaust approximately equal amounts of air to and from the building, and transfer heat between the incoming and outgoing airstreams in an air-to-air heat exchanger. Some air-to-air heat exchangers, often called enthalpy exchangers, are designed so that moisture, as well as heat, is transferred between airstreams. The transfer of moisture is advantageous in situations when it reduces the latent load on air-conditioning equipment or when it reduces humidification requirements during the winter.

Mechanical ventilation with heat exchange (MVHX) has been used for many years in a limited number of commercial buildings and is now becoming increasingly common in residences, particularly those located in regions with cold climates. The performance of MVHX systems, however, can be substantially degraded if moisture from the exhaust airstream freezes within the core of the air-to-air heat exchanger.
The accumulation of ice or frost within the heat exchanger can substantially reduce both the flow rate of the exhaust airstream and the amount of heat recovery.

Prior research on freezing in air-to-air heat exchangers has been directed primarily toward a determination of the indoor and outdoor environmental conditions that lead to freezing. In this paper, we provide background information on freezing, describe an experimental study of the performance of two air-to-air heat exchangers during operation with freezing and periodic defrosts, and discuss techniques to control freeze-protection systems and characteristics of heat exchanger design that are relevant to freezing. The experimental system and the heat exchangers are described only briefly; however, detailed descriptions are provided in a companion paper (Fisk et al. 1985), which focuses on the environmental conditions that lead to freezing.

BACKGROUND

Mechanisms and Rate of Freezing

In this paper the term "freezing" refers to the formation of both ice and frost on the heat transfer surfaces within the heat exchanger. The following three different mechanisms of freezing in heat exchangers have been identified: (1) Water vapor within the air may condense
on cold surfaces. If the surface temperature is reduced or the condensate drains to a colder region of the heat exchanger's core, the liquid water may freeze to ice. (2) If the dewpoint temperature of the warm moist air within the core is below 0°C, and the air contacts a sufficiently cold surface, a layer of frost will form on the surface. (3) If the air within the core is cooled below its dewpoint temperature, liquid droplets can form within the air and with further cooling the liquid droplets may freeze into a solid. With a sufficiently low dewpoint temperature, ice crystals may form directly within the air without the prior formation of liquid droplets. In either instance, the frozen material may adhere to the heat transfer surfaces.

A variety of factors will affect the rate of freezing, i.e., the rate of ice or frost formation, and its impact on heat exchanger performance. The rate of freezing and the amount of heat transfer surface area over which freezing occurs should increase as the inlet temperature of the cold, i.e., supply, airstream is reduced. If airstream temperatures are sufficiently below those at which freezing is initiated, the rate of freezing will increase with an increase in the inlet humidity of the warm, i.e., exhaust, airstream. However, when conditions are near those at which freezing is initiated, the relationship between the rate of freezing and inlet humidity is complex, because heat released during condensation and freezing will increase temperatures within the core.
Various factors related to the design of the heat exchangers, e.g., airflow geometry and temperature efficiency (defined later), will also affect the temperature distribution in the core and, thus, the rate of freezing.

Methods of Freeze Protection

A variety of freeze-protection techniques are used in commercial heat recovery systems. In this section, we describe only the two freeze-protection techniques that are most commonly used for residential heat exchangers. Methods for controlling the freeze-protection systems are discussed in a later section.

A common strategy is to defrost the exchanger periodically by interrupting the flow of the supply airstream. In general, this is accomplished by turning off the supply-airstream fan. The exhaust airstream continues to flow through the exchanger, warms up the core, and melts the accumulated frost or ice. After a period of time, the supply fan is turned on and normal operation resumes. During the defrost, air continues to be exhausted from the residence; thus, an equivalent amount of outdoor air must enter the residence. A portion of this outdoor air will enter through cracks and other openings in the building envelope—this air is, therefore, not substantially preheated before it enters the residence. Some air may also enter through the supply ductwork and the supply passages of the heat
exchanger (where it will be preheated) if this path is not blocked during the defrost. For the defrost technique used in this experimental study, no cold air flowed through the exchanger during the defrosts.

The second common freeze-protection technique for residential heat exchangers is to preheat the cold supply air sufficiently to prevent freezing before this air enters the heat exchanger's core. An electric resistance heater is generally used for the preheater. Preheating the air consumes electricity and also reduces the amount of heat recovery in the exchanger, but the heat exchanger's operation is not interrupted by the freeze-protection system.

EXPERIMENTAL SYSTEM

Overview

The experiments were conducted in a laboratory with facilities for simultaneously producing warm and cold air with temperatures and humidities representative of the indoor and outdoor environments. The minimum attainable air temperature was approximately -12°C for the majority of the test program, although, procurement of a low-temperature brine chiller permitted a few experiments to be performed with inlet air temperatures of -21°C. The flow of warm air through the heat exchangers was provided by the fan for this airstream within the
exchanger. This air was withdrawn from and returned to a chamber in which the temperature and humidity were controlled to match typical indoor conditions. This open-loop configuration without an auxiliary fan was chosen to ensure that the reduction in flow rate of the warm airstream during our tests was representative of the reductions that would occur due to freezing when the exchanger is operating in a residence. Measurements of the flow rate, temperature, humidity, and static pressure of the airstreams entering and exiting the exchanger were used to determine performance. A detailed description of the testing and measuring systems is provided in the companion paper (Fisk et al. 1985).

EXPERIMENTAL PROTOCOL

The basic experimental protocol was to operate the heat exchanger under conditions with freezing and to periodically interrupt operation with a defrost. The elapsed time between defrosts, designated as the "duration of the freeze," was a test variable and equalled roughly one, two, four, six, hours or overnight. The temperature and relative humidity of the inlet warm airstream was maintained at approximately 20°C and 30%, 40%, or 55% respectively. At the beginning of the freeze-defrost cycles, the airstream flow rates were balanced and equal to approximately 200 kg/h. (Here, we are referring to the flow rates of the airstreams that exit the heat exchanger. Inlet and outlet flow rates were slightly different because of leakage between
the two airstreams. The net amount of leakage was roughly 1% and 5% of the flow rate in Heat Exchangers Nos. 1 and 2, respectively.) At the beginning of each cycle, the temperature of the inlet cold airstream decreased rapidly (as the ductwork upstream of the heat exchanger cooled) after which this temperature remained fairly stable at approximately \(-12^\circ C\) for the majority of tests and \(-21^\circ C\) for a few tests.

Because initial conditions, e.g., amount of moisture in the core can have a significant effect on the rate of freezing, freeze-defrost cycles less than six hours in duration were repeated two to six times in succession. Data from the last cycle or pair of cycles should be largely unaffected by initial conditions, and it is these data that are presented later in this paper. Freeze-defrost cycles of more than six hours duration could not be repeated in succession during a workday, since the defrosts were manually controlled. Tests with a six-hour duration of freeze were preceded by approximately two hours of freezing and a defrost. Overnight tests were preceded by either the defrost from a previous test of shorter duration or several hours of operation with the inlet temperature of the cold stream maintained at \(0^\circ C\). The defrosts were initiated manually by turning off the heat exchanger's cold-airstream fan and adjusting valves so that the cold airstream bypassed the exchanger. The defrost was allowed to progress until complete, as determined by a visual observation of the core. The criteria for the end of a defrost were that all visible ice and
frost had melted and that rapid dripping of water from the core had decreased abruptly. Previous to establishing this criteria, we attempted to use the flow rate of the warm airstream (which increased throughout the defrost) as the criterion to end the defrost. This criterion did not always correlate well with visual observations and yielded fairly nonrepeatable results - so it was rejected.

THE HEAT EXCHANGERS

Tests were performed with two different residential heat exchangers of substantially different design; therefore, it was expected that the rates of freezing and performance deterioration for the units might differ significantly. The heat exchangers are described briefly below; drawings of the exchangers are provided in the companion paper (Fisk et al. 1985).

Heat Exchanger No. 1. The first heat exchanger consists of a cross-flow aluminum core, two aluminum mesh filters, and two centrifugal fans mounted in an insulated case with fittings for attachment to ductwork. The parallel plates in the cross-flow core are separated by aluminium fins that divide the space between the plates into small triangular passages. The cold airstream flows through the core in an upward direction (45° from the horizontal) and the warm airstream flows downward (also 45° from the horizontal). Any water that condenses in the core drains downward in the direction of
airflow toward the colder end of the core.

Heat Exchanger No. 2. The second heat exchanger tested consists of a polypropylene counterflow core and two axial fans mounted in an insulated case with fittings for attachment to ductwork. The cold airstream enters and exits the ends of the rectangular core. The warm airstream enters the top of the core through a series of slots at one end and exits from the bottom of the core through a series of slots at the opposite end. As with most plate-type heat exchangers, the core is essentially a series of parallel plates with portions of each airstream directed through alternate passages. In this exchanger, the passages for the cold airstream are further subdivided into small rectangular channels while the passages for the warm airstream are not subdivided. Due to the unique design of this heat exchanger, much of the water that condenses within its core drains against the direction of airflow to the warmer end of the core.

**CALCULATED PERFORMANCE PARAMETERS**

A description of the most important parameters calculated from data obtained during freeze-defrost cycles is provided below. The calculations are based on complete sets of data that were collected and recorded every five minutes.

Airstream Properties. The average, maximum, and minimum values
of the temperature and mass flow rates of the airstreams entering and exiting the heat exchanger were determined, excluding data from the defrost portion of the freeze-defrost cycles. For the entering and exiting warm airstream, the average, maximum, and minimum values of humidity ratio (mass of water vapor divided by mass of dry air) and relative humidity were also determined.

**Temperature Efficiency and Rate of Change of Temperature Efficiency.**

For each five-minute data set, consisting of a measurement for all airstreams, the temperature efficiency for the supply airstream was calculated from the equation

\[
\varepsilon = 100\% \times \frac{(T_{c,o} - T_{c,i})}{(T_{w,i} - T_{c,i})}
\]  

(1)

where: \(T\) is an airstream temperature, and the subscripts \(c, w, i,\) and \(o\) refer to the cold airstream, the warm airstream, inlet, and outlet of the heat exchanger, respectively. This efficiency equals the temperature rise of the cold airstream divided by the theoretical maximum temperature rise and, in theory, can range from zero to unity. (Actually, if some heat source, such as a fan motor, is located between the upstream and downstream measuring points, the efficiency as defined above can exceed unity.) An increase in efficiency corresponds to a decrease in the heat load imposed on the home's heating system; therefore, it is desirable for the efficiency to be as high as possible.
As ice or frost forms in the core of the heat exchanger, the temperature efficiency decreases. The average rate of change of efficiency for a freeze cycle, $\Delta \varepsilon$, which is a measure of the rate of performance deterioration, was calculated from the equation

$$\Delta \varepsilon = (\varepsilon_3 - \varepsilon_n) / (t_n - t_3)$$

where: $\varepsilon_3$ is the value of $\varepsilon$ calculated from data collected during the third five-minute data set after completion of the previous defrost; $\varepsilon_n$ is calculated from data taken in the last set prior to initiating the next defrost; and $t_3$ and $t_n$ are the corresponding times. The value of temperature efficiency from the third data set was used in the calculation because the first one or two values were often abnormally high due to heat storage in the core during the previous defrost. The time-weighted average temperature efficiency ($\varepsilon_{avg}$) for the entire freeze-defrost cycle (or pair of consecutive cycles) was also calculated, assuming that $\varepsilon$ equaled zero during the defrost.

Rate of Change of Mass Flow Rate. The mass flow rate of the warm airstream was reduced as ice or frost formed within the core. The rate of change of mass flow rate ($\Delta M$) is another indicator of the impact of freezing on performance and was calculated from the equation

$$\Delta M = (1/M_1)(M_1 - M_n)/(t_n - t_1) \times 100\%$$

(3)
where: \( M \) and \( t \) refer to mass flow rate and time and the subscripts "1" and "n" refer to the first and last five-minute data set, respectively.

**Defrost Time Fraction.** One last parameter calculated was the defrost time fraction \( (R) \). This parameter equals the elapsed time during the defrost divided by the total elapsed time for the freeze-defrost cycle.

**RESULTS AND DISCUSSION**

**Trends with Time**

Two parameters that were highly affected by the freezing within these exchangers are the temperature efficiency and the mass flow rate of the warm airstream. As typical examples, these parameters and the inlet temperature of the cold airstream are plotted versus time in Figures 1 and 2.

Figure 1 shows these plots for a test of Heat Exchanger No. 1 with the inlet relative humidity of the warm airstream maintained at approximately 30%. In this test, the duration of the freeze, i.e., the elapsed time between the end of the previous defrost and the start of the subsequent defrost, was 13.3 hours and the defrost of the core required approximately 0.8 hours. At the start of the test, the inlet
temperature of the cold airstream fell rapidly, after which it decreased slowly. The average temperature of the cold airstream was \(-13^\circ\text{C}\). There was a sharp peak in temperature efficiency at the start of the test, which is attributed to heat storage in the core during the previous defrost and subsequent transfer of this heat to the cold airstream. Neglecting data during this initial peak, the efficiency decreased gradually from 72% to 42% and the rate of decrease in efficiency leveled off over time. After the defrost, the efficiency returned to approximately its initial value.

The trend in mass flow rate during this test was similar to the trend in temperature efficiency, although there was no initial peak in flow rate and no leveling off of the rate of decrease in flow rate.

Figure 2 shows the same curves for two successive freeze-defrost cycles with approximately a four-hour duration of the freeze. The inlet relative humidity of the warm airstream was maintained at approximately 55%. No sharp peak in temperature efficiency is evident in this figure. We expect that a peak in efficiency occurred but was not recorded due to its short duration and the periodic nature of our measurements. The average rate of decrease in temperature efficiency and mass flow rate was greater during this test than during the previously described test due to the higher inlet humidity and shorter duration of the freezing period.
The trends in temperature efficiency and mass flow rate during tests of Heat Exchanger No. 2 were similar; however, the rates of decrease in these parameters were generally much smaller. In addition, the rate of decrease in temperature efficiency did not level off significantly with time, even during tests of 14 hours in duration.

Visual Observation.

During tests of each exchanger, the face of the core, where the warm airstream exits, was visually inspected through windows installed in the case. In the cross-flow core of Heat Exchanger No. 1, the coldest surfaces should theoretically be located at the lower edge of this outlet face. Frost or ice appeared first along this edge and then progressed upward with increasing time. A white, opaque solid, assumed to be frost, was frequently observed in the colder regions of the core especially during tests with low inlet relative humidities. A more translucent solid, assumed to be ice, was also observed and was generally present in greater quantities than the frost. Initially, the ends of individual triangular flow passages became plugged with ice or frost but, as time progressed, large portions of the face of the core became covered with ice and icicles extended from the surface of the core. The face of the core was never entirely obstructed because the warmer, top regions of the core remained free of ice or frost.
Based on the visual observations, we suspect that during these tests many of the flow passages became obstructed at their cold end before substantial ice or frost formed deeper within the core. Obstruction of the end of a flow passage would prevent flow of air along the entire passage and, therefore, render the surface area in that passage useless for heat transfer. We also suspect that the direction of water drainage had a large impact on performance. Because condensed water drained toward the cold end of the core where it could freeze, the rate of ice formation and performance deterioration was probably more rapid. Under conditions with freezing, this exchanger might perform better if the warm air flowed upward and condensed water drained against the direction of flow toward the warmer end of the core, despite the adverse impact on performance that would result from reevaporation of condensed water.

In Heat Exchanger No. 2, the warm airstream exits the core through a series of long narrow slots instead of very small flow passages. The flow passages for the warm airstream in this heat exchanger are not subdivided into smaller passages, thus complete blockage of the end of a flow passage with ice would appear less likely than in the core of the first exchanger. Ice or frost (primarily ice) first formed at the coldest end of the slots through which the warm airstream exited and progressed with time toward the warmer end of the slots. A slot never became completely obstructed with ice, although it is possible that flow passages became substan-
tially or completely obstructed at locations within the core that we could not observe. The rate of ice formation at the outlet face of this core was considerably less than the rate of ice formation observed during tests of the other exchanger.

Based on the visual observations, two factors may help to explain why performance deteriorations were less rapid in Heat Exchanger No 2. First, the flow passages for the exhaust airstream were not small enough to be easily obstructed by ice. Second, much of the water that condenses within this exchanger's core drains toward the warmer end of the core instead of the colder end where it could freeze.

Tests with an Inlet Supply Temperature of Approximately $-12^\circ C$

Rate of Change of Temperature Efficiency. The average rate of change in temperature efficiency ($\Delta \varepsilon$) during the periods of freezing, i.e., the periods of time between defrosts, as well as other calculated results and the average properties of the inlet airstreams are tabulated in Tables 1 and 2. The data on rate of change of temperature efficiency are also plotted versus duration of freeze in Figures 3 and 4.

In tests of the Heat Exchanger No. 1, $\Delta \varepsilon$ ranged from 1.5 %/h (percentage points decrease in $\varepsilon$ per hour) to 13.0 %/h. There is significant scatter in the data, but the highest values of $\Delta \varepsilon$ occurred
for tests with a high inlet humidity and a fairly short duration of freeze (one to four hours). In tests with the inlet relative humidity of the warm airstream maintained at 40% and 55%, the average value of $\Delta e$ decreased significantly with an increase in test duration. This trend may be explained by the decrease $\Delta e$ that was observed as time progressed during an individual test. It is also clear from the figure that an increase in $\Delta e$ is associated with an increase in the inlet humidity of the warm airstream at least for tests with a duration of freeze from one to four hours.

In tests of Heat Exchanger No 2., $\Delta e$ ranged from 0.6 %/h to 2%/h. On the average, $\Delta e$ was much smaller during tests of this exchanger when compared to its value during tests of the first exchanger. There are insufficient data to indicate correlations between $\Delta e$ and inlet humidity or the duration of freeze.

Rate of Change in Mass Flow Rate. The rates of change in mass flow rate of the initially warm airstream as a percentage of initial flow rate ($\Delta M$) are tabulated in Tables 1 and 2. The relationships between $\Delta M$ and test conditions, i.e., duration of freeze and inlet humidity, were generally similar to the relationships discussed above for $\Delta e$.

For tests of Heat Exchanger No. 1, the ratio $\Delta M/\Delta e$ (not tabulated) averaged 0.9 (with a standard deviation of 0.6) and ranged
from 0.3 to 1.6. The average of $\Delta M/\Delta t$ from tests of Heat Exchanger No. 2 was 2.2 (with a standard deviation of 0.7) and the range was from 1.1 to 3.2. Thus, the temperature efficiency of the second heat exchanger appeared to be less affected by the reductions in mass flow rate.

Defrost Time Fraction. Another calculated parameter is the defrost time fraction $R$ (Figures 5 and 6), which equals the time required to defrost divided by the total elapsed time during the freeze-defrost cycle. The value of $R$, expressed as a percentage, ranged from 4% to 26% in tests of Heat Exchanger No. 1 and from 6% to 17% in tests of Heat Exchanger No. 2. These ranges are not directly comparable, however, because tests of shorter and longer duration were conducted with the first exchanger. If tests of the same duration are compared, the values of $R$ for the two exchangers are not highly different.

The defrost time fraction ($R$) decreases with an increasing test duration, indicating that the defrost time is used more effectively after a long freeze. Optimal performance is not achieved by minimizing $R$, however, but instead by maximizing the time-averaged temperature efficiency (defined later). With this criterion for optimal performance, the optimal length of time between defrosts of Heat Exchanger No. 1 appeared to be in the range of one to six hours. Thus, defrost time fractions of roughly 15% would generally be re-
quired when operating the exchanger under similar conditions. (Defrost time fractions up to 25% may be required if the inlet humidity is high and the duration of the freeze is one to two hours.)

The optimal time between defrosts of Heat Exchanger No. 2 under the test conditions appeared to be in the range of 6-13 hours. No data are available with freeze periods in this range, but the available data suggest that defrost time fractions of 6% to 12% would be required under these conditions.

It should be noted that the test protocol did not permit determination of the optimal value of R for any given test. Also, the decision of when to terminate defrosts was subjective because it was based on visual indicators. More extensive testing would be required to precisely determine the optimal frequency and duration of defrosts.

Time-Averaged Temperature Efficiency. One final calculated parameter (plotted in Figures 7 and 8) is the time-averaged temperature efficiency ($\varepsilon_{\text{avg}}$). Obviously, it is desirable to operate the exchanger in a manner that will maximize the average temperature efficiency.

The value of $\varepsilon_{\text{avg}}$ ranged from 48% to 64% in tests of Heat Exchanger No. 1. The average temperature efficiency was higher when the time between defrosts was one to six hours compared to 10-20 hours. An average efficiency of 62% was typical for tests of one to
four hours duration with inlet relative humidities of approximately 30% and 40%. The temperature efficiency during initial stages of the freeze cycles was typically in the range of 72% to 82% depending on inlet humidity. Therefore, a decrease in average temperature efficiency of roughly 15 percentage points can be attributed to the freezing and the required defrosts when the inlet humidity was 30% and 40%. A greater reduction in performance occurred with higher inlet humidities.

In tests of Heat Exchanger No. 2, the average temperature efficiencies were significantly higher, i.e., 70% to 82%. The optimal time between defrosts appeared to be in the range of 6 to 13 hours, although no tests were conducted in this range. Properly operated under conditions similar to those in the tests, average temperature efficiencies of 73% to 82% appear achievable, although the higher efficiencies would occur only with a high inlet humidity. It should be noted that there was significant leakage of air (approximately 5%) during the tests from the warm to the cold airstream, thus the measured temperature efficiencies cannot be attributed entirely to heat transfer across surfaces in the core. The temperature efficiencies of this exchanger during the initial periods of freeze-cycles were in the range of 86% to 91%. Therefore, roughly a 10% reduction in temperature efficiency can be attributed to the freezing and the required defrost cycles under these operating conditions. The reductions in temperature efficiency noted above (10% and 15%) are based on the
assumption that the periodic defrosts are controlled in a fairly optimal manner. The methods currently used to initiate and terminate periodic defrosts in residential heat exchangers may, in many cases, not provide optimal performance. Thus, larger reductions in average temperature efficiency due to freezing may be common.

Tests with an Inlet Cold-Airstream Temperature of Approximately $-20^\circ C$

A limited number of tests were performed with the average inlet temperature of the cold airstream maintained at a significantly lower value, i.e., approximately $-20^\circ C$ compared to $-12^\circ C$. These tests were conducted only with Heat Exchanger No. 1 and only with warm airstream relative humidities of approximately 32% and 42%. The results are tabulated in Table 1 and plotted as open data points in Figures 3, 5, and 7.

During most of these low-temperature tests, there were small reductions (approximately 8%) in the mass flow rates of the cold airstream due to freezing on the cooling coils of the test system, and these reductions undoubtedly affected the results. The rates of decrease in temperature efficiency were two to three times greater during low-temperature tests compared to the rates of decrease during corresponding tests with a higher cold airstream temperature. Despite this fact, for tests with a two-hour freeze duration, the defrost time fractions and average temperature efficiencies were not significantly
different from those observed with the higher cold-airstream temperatures. In tests with a three- and four-hour duration of freeze, however, the defrost time fraction was increased and the average effectiveness was reduced compared to the higher-temperature tests. The number of tests performed was insufficient for definite conclusions, but the results indicate that it is desirable to defrost more frequently when the inlet temperature of the cold airstream is low.

Control of Freeze-Protection Systems

One of the objectives of this study and the study of conditions that lead to freezing (Fisk et al. 1985) was to identify parameters that could be used as input signals for a control-system that initiates (and possibly terminates) some technique of freeze protection. The use of periodic defrosts or preheating for freeze protection was described previously.

A common control strategy is to sense the outdoor temperature and activate the freeze-protection system when this temperature is below the onset of freezing. When the freeze-protection system is activated, a timer is used to turn off the cold airstream's fan for a fixed period of time at regular intervals. The data on onset of freezing, defrost frequency, and defrost time fraction in this report and the companion paper should be useful as design parameters for such a control system. An attractive feature of this control system is its
simplicity and low cost; however, it cannot provide optimal performance, since it does not adjust for changes in indoor temperature and humidity or outdoor temperature. Also, considerable information on the freezing characteristics of the exchanger is required, and testing to determine this information is time-consuming and expensive.

Another strategy that is not unusual is to sense the outlet temperature of the supply airstream and to activate the freeze-protection system, generally a preheat system, to maintain this temperature above some value. A weakness of this strategy is that a number of factors besides freezing could possibly reduce the supply temperature below the set point, e.g., low outdoor temperatures and fouling of the heat exchange surfaces.

A strategy that has been suggested is to use the drop in static pressure of the exhaust airstream as it passes through the core as the controlling parameter. As ice or frost forms in the core, this pressure drop will increase and a defrost could be initiated after a certain increase. This pressure drop will decrease as the ice or frost melts and the control system could terminate the defrost when the pressure drop returns to its value prior to freezing. To initiate the defrost when a 10% drop in flow rate occurs, which corresponded to roughly a 5% to 10% drop in temperature efficiency in our tests, would require that the control system respond to a very small change in pressure drop. For example, based on the fan performance data, the
increase in pressure drop would be roughly 10 and 5 Pa, in Heat Exchangers Nos. 1 and 2, respectively, for a 10% decrease in flow rate from 200 to 180 kg/h.

With the decreasing cost of electronic systems, it may be cost effective to develop more "intelligent" control systems for freeze protection. An intelligent control system would (1) activate the freeze-protection system only when needed, (2) automatically adjust for changes in indoor humidity, outdoor temperature, and flow rate, (3) not be activated by performance changes caused by factors other than freezing, and (4) eliminate the need for extensive testing of each unit under conditions of freezing. For example, if a defrost technique is employed, the control system might sense the appropriate airstream temperatures to allow a determination of temperature efficiency and also sense the temperature of a heat-exchange surface at the cold end of the core. The defrost could be initiated only when the surface temperature was below freezing and the temperature efficiency had dropped below a certain value. The defrost could be terminated when the temperature of the heat exchange surface increased to a few degrees centigrade. A preheat system might be controlled by a similar set of sensors. Alternately, the control system might sense only the temperature of the heat exchange surface and provide sufficient preheat to maintain this temperature above freezing. In fact, a simple thermostatic-timer control system based on the temperature of the heat exchange surface might be preferable to a thermostat-timer
system based on air temperature. The actual desirability of more "intelligent" control systems will depend on their cost and performance. It is more important to optimize the freeze protection system if the heat exchanger will be used in a very cold climate.

Suggestions for Future Experimental Studies

Based on our studies, the following are suggestions for future experimental investigations of freezing in air-to-air heat exchangers. Further information would be gained by experimental studies of freezing in additional models of heat exchangers and under a wider variety of operating conditions. Additional investigations of freezing in heat exchangers with low inlet cold-airstream temperatures would be particularly valuable. Also, if residential heat exchangers are to be used in regions with very cold climates, improved techniques for freeze protection and improved control systems for these techniques are desirable; thus, experimental evaluation of various methods of freeze protection would be valuable. Finally, information on temperature and humidity profiles in the cores of exchangers under conditions with freezing would be of value for model development and verification.

SUMMARY AND CONCLUSIONS

In two residential air-to-air heat exchangers (a cross-flow and a
counterflow), performance under conditions with freezing and periodic defrosts was studied in detail. In tests of the cross-flow heat exchanger, freezing caused the temperature efficiency to decrease at a rate from 1.5 to 13.2 percentage points per hour (%/h). More rapid decreases in efficiency were associated with lower cold airstream temperatures, higher warm airstream humidities, and shorter periods of freezing. Temperature efficiency decreased at a slower rate (0.6 to 2.0 %/h) during tests of a counterflow heat exchanger. From observations of the cores during freezing, it appeared that the performance of the counterflow heat exchanger decreased at a slower rate because the core contained larger flow passages which are not easily obstructed by ice and because much of the condensed water drained to the warm end of the core where it could not freeze.

The amount of time required to defrost these heat exchangers divided by the total operating time, i.e., the defrost time fraction, ranged from 0.06 to 0.26 and did not differ greatly between the cross-flow and counterflow exchangers when corresponding tests were compared. With an inlet cold airstream temperature of approximately -12°C, the optimal amount of time between defrosts appeared to be in the range of one to six hours for the cross-flow exchanger and six to thirteen hours for the counterflow exchanger. With lower cold airstream temperatures, more frequent defrosts may yield a better average performance.
When the frequency of defrosts was in the optimal range, the average temperature efficiency for freeze-defrost cycles was in the range of 62% for the cross-flow exchanger and 75% for the counterflow exchanger. The counterflow exchanger had a higher average efficiency because of its superior efficiency without freezing and because its performance was less rapidly affected by freezing. It was estimated that the freezing and periodic defrosts reduced the average temperature efficiency of the cross-flow and counterflow exchangers by 15% and 10%, respectively. Larger reductions in average performance will occur when the frequency or duration of defrosts is far from optimal.

This investigation indicates that the performance of residential air-to-air heat exchangers need not be drastically reduced when operated under conditions that lead to freezing. A freeze-protection system is clearly desirable if these exchangers are to be used in cold climates. Heat exchanger designs that minimize the effects of freezing are recommended, as well as improved controls for freeze-protection systems.

ACKNOWLEDGMENTS

The efforts of a number of people who contributed to this study are greatly appreciated. Lloyd Davis and Al Robb designed and fabricated many components of the experimental system. Assistance provided by James Koonce helped greatly to keep the project moving forward. Re-
ports by Professor Ralph Seban and Ali Rostami at the University of California, Berkeley, were helpful for the planning of this research and for interpretation of the data. The efforts by Dr. Harry Keller in an early field study of freezing in a residential heat exchanger are greatly appreciated. We would also like to thank Gayle Milligan and Dolores Henricksen for preparation of the typescript, Moya Melody for supervising the preparation of figures, and Dave Grimsrud, Ralph Seban, Les Christiansen, and Fred Bauman for their reviews of the original document on which this paper is based.

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REFERENCE

TABLE 1

Results from Tests of Cross-flow Heat Exchanger No.1.

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Figure 1. Mass flow rate of warm airstream, temperature efficiency for cold airstream, and inlet temperature of cold airstream versus time for Test No. 1-6 of Heat Exchanger No. 1. The dashed lines connect data from periods before and after the defrost.
Figure 2. Mass flow rate of warm airstream, temperature efficiency for cold airstream, and inlet temperature of cold airstream for Test No. 1-5 of Heat Exchanger No. 1. The dashed lines connect data points from periods before and after the defrost.
Figure 3. Rate of decrease in temperature efficiency versus duration of freeze for tests of Heat Exchanger No. 1.
Figure 4. Rate of decrease in temperature efficiency versus duration of freeze for tests of Heat Exchanger No. 2.
Figure 5. Defrost time fraction versus duration of freeze for tests of Heat Exchanger No. 1.
Figure 6. Defrost time fraction versus duration of freeze for tests of Heat Exchanger No. 2.
Figure 7. Average temperature efficiency versus duration of freeze for tests of Heat Exchanger No. 1.
Figure 8. Average temperature efficiency versus duration of freeze for tests of Heat Exchanger No. 2.
This report was done with support from the Department of Energy. Any conclusions or opinions expressed in this report represent solely those of the author(s) and not necessarily those of The Regents of the University of California, the Lawrence Berkeley Laboratory or the Department of Energy.

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