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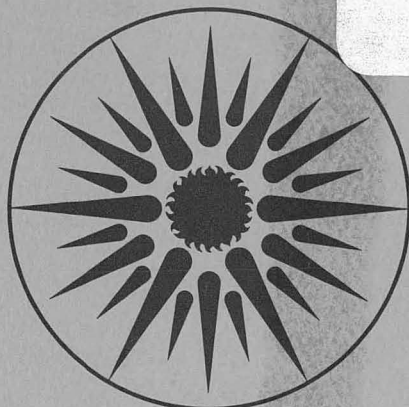
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W. Fisk, R. Chant, K. Archer, D. Hekmat,
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November 1984

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ONSET OF FREEZING IN RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS

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ABSTRACT

Mechanical ventilation of residences, with heat recovery in air-to-air heat exchangers, is an increasingly common practice. When this technique of ventilation is used in cold climates, however, freezing can occur in the air-to-air heat exchanger and substantially reduce its performance. A laboratory investigation was conducted to determine the indoor and outdoor environmental conditions that lead to freezing. In a cross-flow, counterflow, and enthalpy-type cross-flow heat exchanger, respectively, freezing was observed when the inlet temperature of the cold airstream was below -7 to -3°C , approximately -6°C , and -8 to -12°C , for a typical range of indoor humidities. These results are in fair agreement with the theoretical predictions presented in this paper and with data from two field studies conducted with similar heat exchangers. Data from a previous laboratory study of a counterflow heat exchanger and tabulated data supplied by ASHRAE, however, indicate that freezing is initiated at significantly lower cold airstream temperatures, particularly when the warm airstream is humid.

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

INTRODUCTION

One increasingly utilized method of ventilating residences that incorporates energy recovery is to use a residential mechanical ventilating system with an air-to-air heat exchanger (MVHX system). These systems supply outdoor air with a low concentration of indoor-generated pollutants to the residence and exhaust an equal amount of indoor air with a higher concentration of these pollutants. In the core of the heat exchanger, heat is transferred from the warmer airstream to the cooler airstream without mixing of the two airstreams; thus, the supply airstream is preheated in the winter and precooled in the summer before it enters the residence.

A typical residential air-to-air heat exchanger consists of a core, where the heat is transferred between the airstreams, a pair of fans (one for each airstream), and an insulated case containing the core, fans, and fittings for attachment to ducting. A duct system is generally installed so that air is withdrawn from various locations throughout the residence, and preheated or precooled outdoor air is supplied to other locations. Other air-to-air heat exchangers, designed for use without ductwork, are installed through a wall or window, much like a window air conditioner is installed. The performance and economics of residential MVHX systems have been investigated in laboratory and field studies conducted by Lawrence Berkeley Laboratory (LBL) and by researchers in Canada, Europe, and Japan (see, for

example, Fisk and Turiel 1983; Offermann et al. 1982; Svensson 1982).

From an economic point of view, the use of MVHX systems is most attractive in regions with cold climates and in houses heated with expensive forms of energy. However, when it is sufficiently cold outdoors, moisture in the exhaust airstream can freeze within the core of the heat exchanger, and the resulting accumulation of ice or frost can substantially reduce the amount of heat recovery, i.e., the amount by which the incoming air is preheated.

This paper describes the results of a simple theoretical prediction and a laboratory study to identify the indoor and outdoor environmental conditions that lead to freezing. Background information is provided, previous data on the onset of freezing are reviewed, and the experimental system is also described. In a companion paper (Fisk et al. 1985), a study of the performance of residential-to-air heat exchangers during operation with freezing and periodic defrosts is described.

BACKGROUND

Freezing will occur in an air-to-air heat exchanger if the temperature of a heat transfer surface, at some location within the core, is below 0°C and also below the dewpoint temperature of the moist air in contact with the surface at that location. (The term "dewpoint tempera-

ture" is used in this paper to represent both dewpoint and frost-point temperature.) If the temperature drop across the thickness of the plates that separate the airstreams is negligible (which is generally true), and the heat conduction along these plates parallel to the direction of flow is assumed negligible, then the temperature of the heat transfer surface is determined by the temperatures of the surrounding air and by the convective heat transfer coefficients. For this simple case, the surface temperature ($T_{s,i}$) at a point "i" within the core can be calculated from the equation

$$T_{s,i} = (h_{w,i} T_{w,i} + h_{c,i} T_{c,i}) / (h_{w,i} + h_{c,i}) \quad (1)$$

where: $h_{w,i}$ and $h_{c,i}$ are the convective heat transfer coefficients at point "i" for the warm and cold airstreams, respectively, and $T_{w,i}$ and $T_{c,i}$ are the corresponding bulk temperatures for the warm and cold air on each side of the heat transfer surface. (The "bulk temperature" is the average air temperature, across the flow channel, in the direction perpendicular to the surfaces.) If the convective heat transfer coefficients are not equal, the surface temperature will more closely approach the temperature of the air with the higher coefficient. The warm air's convective coefficient and, thus, the heat transfer surface temperature may be increased if water vapor condenses within the core. The increase in convective heat transfer coefficient is caused by the mass transfer of water to the surface and the formation of a rough film of liquid water on the surface. However, the coldest surfaces

are generally located where the cold air enters the core, and the convective coefficient for the cold air may be significantly increased in this region due to entrance effects. These two factors, condensation and entrance effects, will counteract, although one or the other may dominate in some situations. The relationships between airstream bulk temperatures, heat transfer surface temperatures, condensation, freezing, and entrance effects are illustrated qualitatively in Figure 1, which shows the temperature profiles for a counterflow exchanger. Measured temperature profiles in an actual counterflow exchanger under conditions with condensation are presented by Besant and Bugg (1982).

The inlet temperature of the cold airstream at the onset of freezing will depend on a number of factors. An increase in the inlet temperature of the warm airstream will cause freezing to be initiated at a lower temperature. In a heat exchanger that has a higher temperature efficiency (defined here as the temperature change of the cold airstream divided by the difference between the temperatures of the two airstreams entering the heat exchanger), the warm air will be cooled by a greater amount, and, thus, freezing will generally be initiated at a higher inlet cold-airstream temperature. Airflow geometry is also important. In an ideal counterflow, the temperature distribution is one-dimensional, i.e., the temperatures vary only in the direction parallel to the flow, while in an ideal cross-flow the temperature distribution is two-dimensional. Due to the different

temperature distributions, colder surface temperatures will occur in a cross-flow core compared to a counterflow core with the same temperature efficiency and identical inlet air conditions. Thus, the temperature at the onset of freezing should be higher in cross-flow cores, although it is possible that freezing may be limited to only the colder regions of the core. However, in many actual counterflow cores, the flow geometry deviates from true counterflow near the ends; thus, the differences between the onset of freezing in the two types of cores may be less than otherwise indicated.

The impact of the humidity of the warm airstream on the onset of freezing is more complex. In many instances, a higher inlet humidity will lead to more condensation within the core and a smaller decrease in the warm airstream temperature. Furthermore, the condensation may increase the warm airstream's convective heat transfer coefficient, resulting in an increase in surface temperature. These factors cause the temperature of the cold airstream at the onset of freezing to be reduced as the humidity is increased. However, this trend may be reversed if the humidity of the warm airstream is low. In the ideal case of zero moisture in the warm air, freezing cannot occur. As the moisture content of the incoming warm airstream is gradually increased from this idealistic dry state, the temperature at which freezing occurs will also increase to some maximum at an intermediate humidity.

Under some conditions, the transfer of water vapor between air-

streams in enthalpy-type air-to-air heat exchangers can reduce the temperature at which freezing is initiated. In these heat exchangers, the dewpoint temperature of the warm moist air will be reduced by the water vapor transfer through the heat transfer surfaces or by the adsorption of water vapor on hygroscopic surfaces as it flows toward the colder ends of the heat transfer surfaces. The dewpoint temperature of the warm air can be reduced to below 0°C if the inlet humidity is low and/or the water vapor transfer rate is high; this, in turn, reduces the heat transfer surface temperature and the inlet temperature of the cold airstream at which freezing is initiated.

Some other factors that affect the onset of freezing are mentioned briefly: (1) the method by which the core is installed in the case can affect the temperature of heat transfer surfaces because of heat conduction between the case and core; (2) in actual heat exchangers there will be some heat conduction parallel to the direction of airflow in the plates that separate the airstreams, and this conduction will affect the surface temperatures; (3) the airstream flow rates and the ratio between the flow rates of the two airstreams will affect the temperatures within the core and, thus, the onset of freezing; and (4) heat released by fans and fan motors located within the heat exchanger can increase the temperature of the air that enters the core and thus reduce the onset of freezing.

REVIEW OF PREVIOUS WORK

Early studies of freezing in air-to-air heat exchangers were conducted by staff of the University of Manitoba, who encountered freezing problems in a hospital installation of three rotary air-to-air heat exchangers with 3.7 m diameter, aluminum heat-wheel cores. The wheels were being operated below optimal rotational speed to reduce freezing and because of imbalances of the wheels caused by the accumulated ice. Tests indicated that freezing was initiated with outside temperatures between -16°C and -26°C when the exhaust airstream's inlet relative humidity was between 25% and 30%. Additional tests were performed in the laboratory with a hygroscopic-wheel exchanger (Ruth et al. 1975). The onset of freezing, determined by an increase in pressure drop for the exhaust airstream of 50 Pa, was highly dependent on the humidity of the exhaust airstream, and the humidity of the supply airstream could not be ignored. Freezing did not occur with supply airstream inlet temperatures above -12°C , and supply temperatures as low as -26°C could be tolerated without freezing when the exhaust airstream had a temperature of 24°C and a relative humidity below 22%.

Following the work in Manitoba, Sauer et al. (1981) at the University of Missouri-Rolla determined the onset of freezing for a "counterflow pure-plate" heat exchanger. Numerous tests were conducted with an inlet exhaust airstream temperature of 24°C . The onset of freezing linearly ranged from -23°C with an inlet exhaust

stream relative humidity of 60% to -9°C with an inlet relative humidity of 30%. The criterion used to determine the onset of freezing was not given. These results agreed fairly well with tabulated data in the ASHRAE Handbook--1983 Equipment (ASHRAE, 1983).

During the winter of 1980-1981, Lawrence Berkeley Laboratory (LBL) staff studied the performance of MVHX systems in nine occupied residences located in Rochester, New York (Offermann et al. 1982). The thermal performance of the heat exchanger in each residence was monitored for approximately a one-week period by measuring inlet and outlet airstream temperatures every half-hour. Occasional checks for freezing were also made visually. Freezing was observed in the cores of both a counterflow and a cross-flow heat exchanger. The onset of freezing, as indicated by rapid deterioration in heat exchanger performance, was approximately -8°C for both exchangers. The indoor relative humidities during the study were in the range of 30% to 40%.

LBL staff also participated in monitoring the performance of a counterflow heat exchanger installed in a Minnesota residence. The onset of freezing, determined by performance measurements and periodic visual observations, was approximately -7°C with an indoor relative humidity of 30%.

In addition to the experimental studies, Rostami (1982) completed a Ph.D. dissertation on condensation and frosting in air-to-air heat

exchangers. His dissertation includes a review of previous work on frost formation on uniform-temperature flat plates, development of an improved model for frost growth rate, and use of the model to predict the performance of a parallel-flow heat exchanger under conditions with frosting.

EXPERIMENTAL SYSTEM

Overview

Experiments to study freezing in residential air-to-air heat exchangers 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. Measurements of the flow rate, temperature, humidity, and static pressure of the airstreams entering and exiting the exchanger were used to determine performance. A description of the testing and measuring systems is provided below.

Environmental Control of Air

The cold airstream flowed in a closed loop through the heat exchanger, an external fan, and conditioning equipment. Air returning from the exchanger was dried as it passed through a desiccant dehumidifier. The dried airstream then passed through cooling coils,

where it gave up heat to a chilled brine. The chilled air then returned to the exchanger through an insulated duct. 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 exchanger was provided by the fan 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. The temperature of the inlet warm air was generally maintained within 0.5°C of the desired value, although larger deviations occurred occasionally and some test data were rejected due to poor temperature control. Relative humidity (RH) was generally maintained stable within $\pm 3\%$ RH for tests at 30% and 40% RH and within $\pm 5.5\%$ RH for tests at 55% RH.

Measuring System

Temperature. Air temperature was measured in the insulated ducts adjacent to the inlet and outlet fittings of the heat exchanger. A

pair of air-mixing vanes, upstream of each measuring location, ensured that the air temperature was uniform within approximately 0.25°C at the measuring location. A thermistor-based temperature probe was placed in the center of each duct and its output voltage was read by a microcomputer system. A grid of five copper-constantan thermocouples at each location, connected to a microvoltmeter was read manually once during each test and served as an alternate measuring system. The thermistor-based probes were calibrated twice at multiple temperatures during the measuring period by comparison to precision thermometers. The thermocouple system had been calibrated twice prior to this study by comparison to a platinum resistance thermometer that is traceable to the National Bureau of Standards. Based on the daily comparisons between the thermocouples and the thermistor-based probes, small corrections (generally 0.1 to 0.3°C) were made to the probe readings during final data analyses. The estimated uncertainty in the temperature measurements is $\pm 0.3^{\circ}\text{C}$.

Flow Rate. The flow rates of the airstreams were measured with orifice plate flowmeters. The pressure difference between the pressure taps upstream and downstream of the orifice plates was sensed by an electrical pressure transducer, and the output signal from the transducer was read by the microcomputer system. The pressure transducer was calibrated at multiple pressures twice during the measurement period using a micromanometer with a sensitivity of 0.5 Pa. For each test, the differential pressures were also determined

manually one time using the micromanometer. The estimated uncertainty in the flow rate measurements using a procedure specified by the American Society of Mechanical Engineers (1971) is approximately $\pm 2\%$. As a check on uncertainty of measurement, the mass flow rate of all air entering the heat exchanger was compared to the mass flow rate of all air exiting the exchanger and the difference was generally less than 2% to 3%.

Humidity. The humidity (dewpoint temperature) of the warm airstream was measured both upstream and downstream of the heat exchanger. The dewpoint temperature of the cold airstream was generally less than -18°C and was only measured occasionally. The dewpoint probes used for this study utilize a thermistor-based temperature sensor surrounded by a lithium chloride bobbin. The output voltages from the probes were read by the microcomputer. The temperature sensor in each probe was calibrated by the method previously described for air-temperature probes. During each test, humidity was also measured once at each location using precision wet- and dry-bulb thermometers. For the inlet warm airstream, the difference between the dewpoints calculated from wet- and dry-bulb measurements and from the lithium chloride probes was as great as 1.5°C for a few tests but was generally between 0.5 and 1.0°C . In the final data analysis, the measured dewpoints were corrected to the extent possible using the wet- and dry-bulb measurements; however, the estimated uncertainty in the final calculated dewpoints is still on the order of $\pm 1.0^{\circ}\text{C}$, which

causes approximately a +3% uncertainty in inlet relative humidity. The humidity probe in the outlet warm airstream gave erratic results during some of the tests. This airstream was nearly saturated with water vapor during many of the tests, and condensation of water on the probes may have caused the measuring problems. For the final few tests, capacitance-type relative humidity probes were used. These were calibrated by comparison to an instrument that uses a chilled mirror to sense dewpoint temperatures. However, this type of probe also gave poor results when placed in the nearly saturated outlet air.

Microcomputer. The microcomputer was controlled by a BASIC program. Four channels of data were read in sequence, each channel consisting of all signals for an airstream (e.g., the signals from sensors for temperature, pressure, humidity, and flow rate of the inlet warm airstream constituted one channel). After cycling through the four channels, the zero reading of the two pressure transducers was checked automatically. A complete cycle was completed every five minutes. The computer controlled the opening and closing of solenoid valves so that the pressure transducers sensed the appropriate differential and static pressures.

Data Processing. Substantial data processing was performed by the BASIC program loaded into the microcomputer system (e.g., calculations of flow rates, temperatures, pressures, dewpoint temperatures, relative humidities, mass balance, etc.). The processed and

unprocessed data were printed on paper and recorded on magnetic tape. Using a larger computer system, computer-generated plots were prepared for each test and further calculations were performed.

EXPERIMENTAL PROTOCOL

To determine the onset of freezing, the temperature and relative humidity of the warm airstream entering the exchanger were established at approximately 20°C and 30%, 40%, or 55% RH. (To determine the onset of freezing of Heat Exchanger No. 1, tests were performed at 25% RH instead of 30% RH.) The temperature of the cold airstream entering the exchanger was adjusted so that it equalled the desired value, and damper valves were adjusted so that the mass flow rate of the two airstreams were approximately equal (i.e., within 10%). The flow rates chosen for the tests were approximately 210, 190, and 130 kg/h for tests of Heat Exchangers Nos. 1, 2, and 3, respectively. (The exchangers are described later.) Static pressures were adjusted to minimize leakage of air between airstreams. Conditions were maintained stable for up to six hours. The face of the heat exchanger's core, where the exhaust airstream exits the core, was periodically inspected for frost or ice through small windows installed in the case. If frost or ice was not visible after six hours, the test was designated as one without freezing, and if ice or frost was visible, the test was designated as one with freezing. For tests of Heat Exchanger No. 1, which were completed before the test system became

partially automated, the waiting period was only 5, 4 1/2, and 4 hours respectively for tests at 25%, 40%, and 55% relative humidity. Freezing appeared more rapidly in this exchanger; however, it would have been preferable to wait for six hours as in tests of the other units.

It should be noted that a visual observation of frost or ice always preceded any definitive changes in performance attributable to freezing. Small (a few percent) changes in static pressures and small reductions in the flow rate of the warm airstream were considered to be unreliable as indicators of freezing because a buildup of condensate within the core could have been the cause of these changes. It may also be noted that these tests did not determine whether the freezing would progress sufficiently to cause a substantial reduction in performance. The choice of a visual criterion for the onset of freezing is further discussed later in this paper.

THE HEAT EXCHANGERS

Tests to determine the onset of freezing were performed with two different residential heat exchangers plus the core from a third exchanger. The three exchangers were of substantially different design, therefore, it was expected that the onset and rates of freezing for the units might differ significantly.

Heat Exchanger No. 1. The first heat exchanger tested (Figure 2) 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. Before testing this exchanger, an effort was made to seal between the core and the case in order to minimize leakage between airstreams.

Heat Exchanger No. 2. The second heat exchanger tested (Figure 3) consists of a polypropylene counterflow core and two axial fans mounted in an insulated case with fittings for attachment to ductwork. 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. The unit tested was one of the first produced by the manufacturer, and considerable sealing was

required to decrease the rate of air leakage between airstreams.

Heat Exchanger No. 3. The third product tested is a cross-flow, enthalpy-type heat exchanger core (Figure 4) constructed from a treated paper. Water vapor is transferred directly through the paper surfaces from the more humid to the less humid airstream. The exact mechanism for the moisture transfer is not specified. The core is utilized in a heat exchanger that is designed for installation through a wall or window and, therefore, has no provisions for attachment to ductwork. For testing, the core was removed from the exchanger and installed in a sheet metal housing that could easily be connected to ductwork. In the tests and in the actual heat exchanger, the warm airstream flowed in an upward direction (45° from the horizontal) and the cold airstream flowed in a downward direction (45° from the horizontal). Condensed water drained against the direction of airflow toward the warmer end of the core.

RESULTS AND DISCUSSION

Analytical Prediction

To obtain an appreciation for the impact of some of the parameters involved in the onset of freezing, a simple theoretical prediction of the onset of freezing in a counterflow exchanger was made based on the following simplifying assumptions:

1. Equal airstream mass flow rates.
2. The convective heat transfer coefficients are equal, i.e., the plate temperature is the mean of the bulk airstream temperatures.
3. Freezing occurs when the temperature of the heat-transfer surface is less than the freezing point (0°C) and the inlet dewpoint temperature of the warm air.
4. The temperature efficiency of the heat exchanger is independent of humidity when the inlet relative humidity of the warm airstream is less than 30%, and the temperature efficiency increases linearly by eight percentage points, due to increased condensation within the core, as the inlet humidity increases from 30% to 70%. (This assumption is an approximation based on data from this study and work by others.)

An iterative procedure was used for the calculations. In each iteration, the assumed value for the temperature of the inlet cold airstream was reduced by 0.25°C , and the minimum plate temperature was calculated using an energy balance to equate the heat gained by the cold airstream to the heat lost by the warm airstream. The calculations were terminated when the calculated plate temperature was less than both 0°C and the inlet dewpoint temperature of the warm airstream.

Figure 5 was plotted from the calculations performed and serves to demonstrate the relationship of some of the variables involved. First, temperature efficiency can have a significant impact on the onset of freezing, although previous data on the onset of freezing were often published without any reference to temperature efficiency. The curves demonstrate the concept sometimes used to control freezing, namely, that a reduction in the efficiency lowers the inlet temperature of the supply airstream at which freezing is initiated. The reasons for the reduction of the temperature at the onset of freezing with high and low humidities of the exhaust airstream were explained previously. A peak temperature at which the onset of freezing occurs is predicted at a relative humidity of 25% to 35%. The model predicts that freezing occurs at temperatures that are significantly higher than the "frost threshold temperatures" listed by ASHRAE (1983). ASHRAE does not indicate the assumptions for their data.

Experimental Results

Figure 6 shows the experimental results of tests with Heat Exchanger No. 1, which has a cross-flow core. The temperature efficiency during these tests varied from 70% to 77%. A peak temperature for the onset of freezing of -2.7°C is indicated at a mid-range relative humidity, but the actual location of the peak cannot be determined because tests were performed at only three different humidities. At the high relative humidity, the onset of freezing was reduced

because condensation tends to increase the surface temperatures. At low humidities, the inlet dewpoint temperatures were close to the freezing temperature, and the formation of a white frost, in contrast to a more translucent ice, was observed.

Experimental data from tests of Heat Exchanger No. 2, which has a counterflow core, are also plotted in Figure 6. The temperature efficiency varied from 85% to 90% during these tests. Although both airstreams flowed in a predominately horizontal direction in this heat exchanger, some of the condensate drained against the direction of flow and presumably some reevaporation of the condensate occurred. The results indicate that the onset of freezing is almost independent of the relative humidity of the exhaust air. Compared to the onset of freezing in Heat Exchanger No. 1, some improvement is evident, since freezing did not take place with inlet temperatures above -5°C . The plate temperature at the cold end of the core was calculated for all the data points by assuming equal heat transfer coefficients, i.e., averaging the supply inlet and exhaust outlet temperatures. In all cases the calculated plate temperature was below 0°C for the tests where freezing was observed and above 0°C for tests without freezing, within the limits of experimental error.

For comparison to the results of tests with Heat Exchanger No. 2, the work completed at the University of Missouri-Rolla by Sauer et al. (1981) was included in Figure 6. In their study, the inlet tempera-

ture of the exhaust airstream was held at 24°C , but no mention was made of the mass flow ratios used or the range of temperature efficiencies. In the LBL tests, the onset of freezing was observed visually. The authors of the Missouri-Rolla paper described the technique for determining the onset of freezing, based on a change in the exhaust airstream's pressure drop, that was previously used by Ruth et al. (1975) of the University of Manitoba. If Missouri-Rolla researchers used this method to determine the onset of freezing, it may account for the lower temperatures determined for the onset of freezing. Also, the inlet temperature of the exhaust airstream was higher during the tests by Sauer et al. (1981), and the temperature efficiency of the exchanger used for their tests may have been considerably lower than the efficiency of the exchanger used in LBL's tests.

The experimental results of tests with the core of Heat Exchanger No. 3 are shown in Figure 7. During the tests, the temperature efficiency was approximately 59% prior to any performance deteriorations caused by freezing. As anticipated, the results do not indicate any lowering of the temperature of the onset of freezing as the relative humidity of the exhaust air increases. Moisture is transferred out of the exhaust airstream, lowering the specific humidity and the dewpoint of the air as it progresses toward the outlet of the core. In tests with low inlet relative humidities, it is apparent that the dewpoint of the exhaust airstream has been lowered considerably below the

freezing temperature because the onset of freezing is significantly lower than for the other units tested. This fact is confirmed by the measurements of outlet humidity.

For purposes of comparison, the University of Manitoba results by Ruth et al. (1981) have been plotted in the same figure. The similarity in the shapes of the characteristic curves are of interest. The University of Manitoba tests were performed on a rotary-type enthalpy exchanger consisting of a hygroscopic wheel. The onset of freezing temperature was determined by the previously described technique based on exhaust airstream pressure drop. The low temperatures at the onset of freezing are presumably due to the removal of the moisture from the exhaust airstream and the resultant lowering of the dewpoint.

General Observations

Comparing the work performed in the LBL facilities with the previous work highlighted some shortcomings. In this work, the onset of freezing was determined visually, which, although positive, may not be convenient and does not assure that sufficient freezing would occur to reduce performance significantly. Other researchers have chosen a 50 Pa increase in the static pressure drop of the exhaust airstream as the criterion for the onset of freezing; however, this criterion is not appropriate for all heat exchangers. The establishment of an

acceptable and universal method of determining the onset of freezing is recommended. Since temperature efficiency is the variable of primary interest, a useful criterion for the onset of freezing might be a reduction in temperature efficiency by a specified amount (e.g., 5%) from a reference value of efficiency that is measured with the same test system. The reference value of temperature efficiency could be the measured value with the same inlet temperature and humidity for the warm airstream and with an inlet cold airstream temperature slightly above freezing (e.g., 2°C).

SUMMARY AND CONCLUSIONS

An experimental investigation of the onset of freezing in residential air-to-air heat exchangers was conducted. The inlet temperature of the cold airstream at the onset of freezing and its dependence on the inlet relative humidity of the warm airstream was determined for three exchangers of different design. As determined by visual observation, the onset of freezing ranged from approximately -3°C to -12°C and was a function of both heat exchanger design and the humidity of the warm airstream. The results are in fair agreement with determinations of the onset of freezing in the limited number of reported field studies with similar heat exchangers and with theoretical predictions for a counterflow heat exchanger presented in this paper. Data from a previous laboratory study of the onset of freezing in a counterflow exchanger and tabulated data supplied by ASHRAE, however, indicate

that freezing is initiated at significantly lower cold airstream temperatures, particularly when the warm airstream is humid. In future studies, it would be beneficial if different investigators used the same criterion for the onset of freezing. A suggested criterion, based on a drop in the heat exchanger's thermal performance, is provided in this paper.

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REFERENCES

ASHRAE. 1983 ASHRAE handbook--1983 Equipment. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASME. 1971. Fluid meters, Their theory and application. American Society of Mechanical Engineers.

Besant, R.W; and Bugg, J.D. 1982. "The performance of a counterflow air-to-air heat exchanger with water vapor condensing and frosting." Department of Mechanical Engineering, University of Saskatchewan, Saskatchewan, Canada S7N 0W0.

Fisk, W.J; and Turiel, I. 1983. "Residential air-to-air heat exchangers: Performance, energy savings, and economics." Energy and Buildings, 5, pp. 197-211.

Fisk, W.J.; Chant, R.E.; Archer, K.M.; Hekmat, D.; Offermann, F.J.; and Pedersen, B.S. 1985. "Performance of residential air-to-air heat exchangers during operation with freezing and periodic defrosts." ASHRAE Transactions, Vol. 91, Pt. 1B.

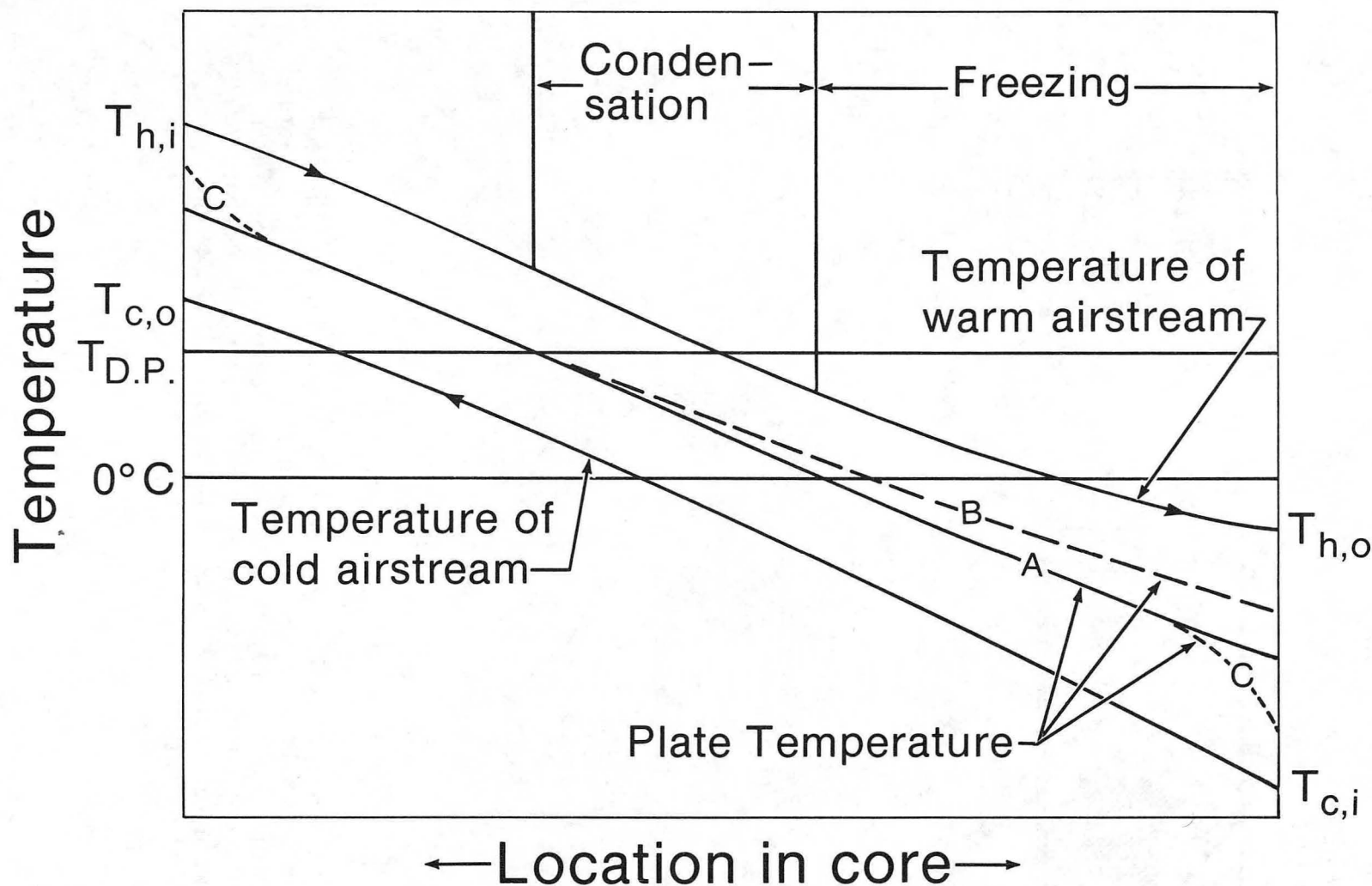
Offermann, F.J.; Dickinson, J.B.; Fisk, W.J.; Grimsrud, D.T.; Hollowell, C.D.; Krinkel, D.L.; Roseme, G.D.; Desmond, R.M.; DeFrees, J.A.; and Lints, M.C. 1982. "Residential air-leakage and indoor air quality in Rochester, New York." Lawrence Berkeley Laboratory Report, LBL-13100, Berkeley, CA 94720.

Rostami, A.A. 1982. "Condensation and frost formation in heat exchangers." Ph.D. dissertation, Department of Mechanical Engineering, University of California, Berkeley, CA 94720.

Ruth, D.W.; Fisher, D.R.; and Gawley, H.N. 1975. "Investigation of frosting in rotary air-to-air heat exchangers." ASHRAE Transactions, Vol. 81, Pt. 1, pp. 410-417.

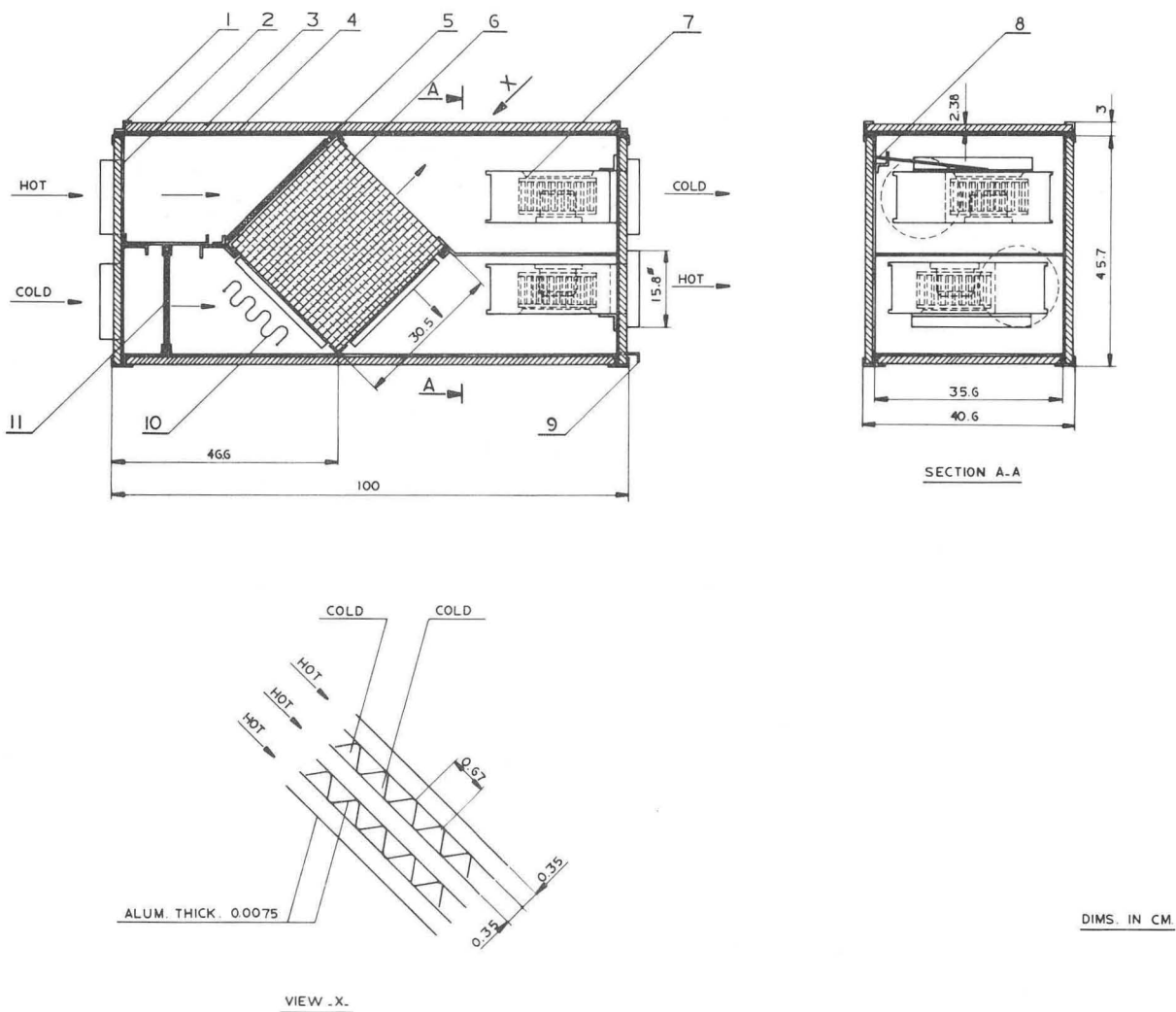
Sauer, H.J.; Howell, R.H.; and Wray, J.R. 1981. "Frosting and leak testing of air-to-air energy recovery systems." ASHRAE Transactions, Vol. 87, Pt. 1, pp. 211-234.

Svensson, A. 1982; "Efficiency of air-to-air heat exchangers in occupied houses." Proceedings of the 3rd Air Infiltration Centre Conference on Energy Efficient Domestic Ventilation Systems for Achieving Acceptable Indoor Air Quality, London, U.K., September 20-23, 1982.



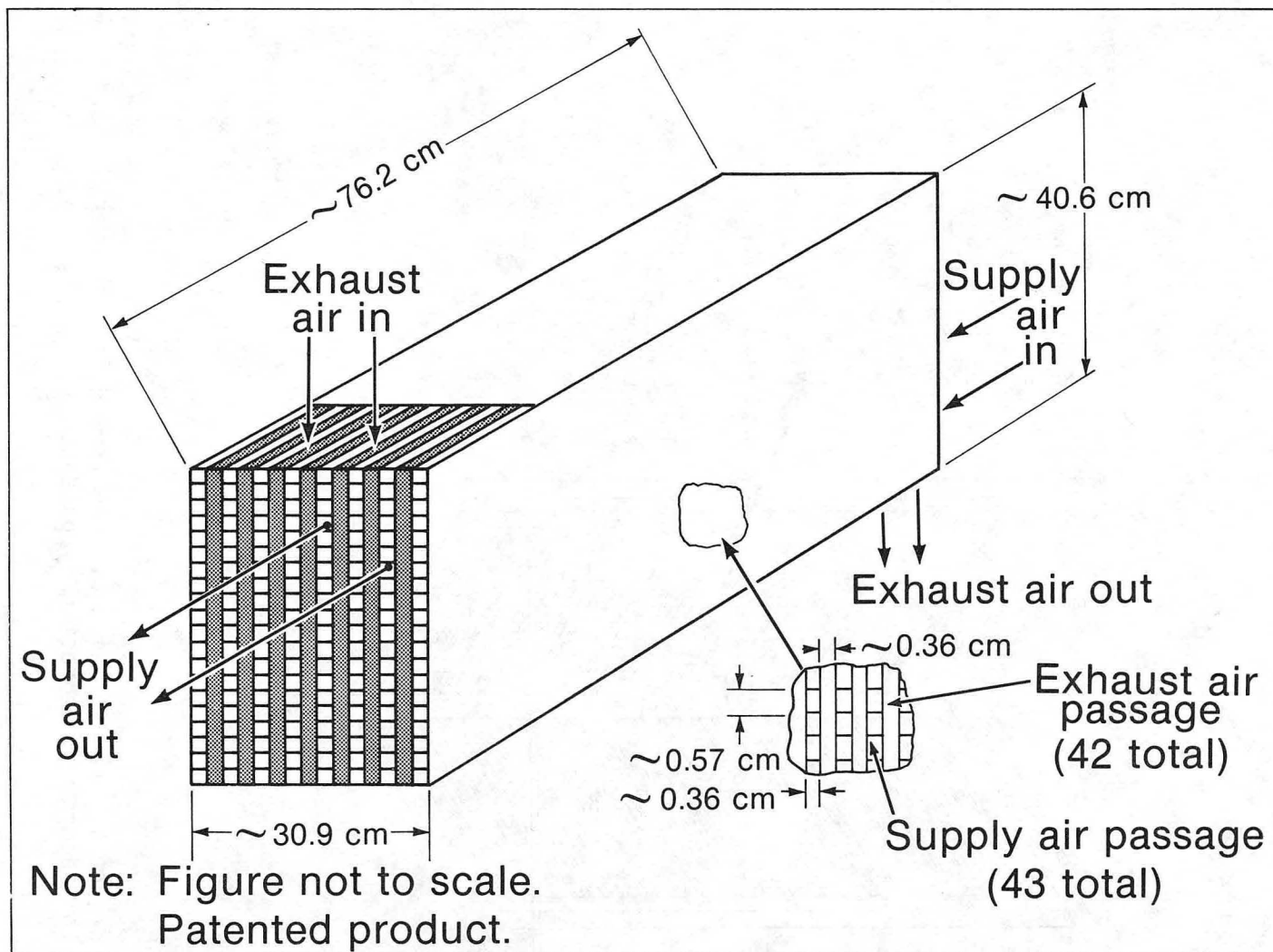
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Figure 1. Illustration of bulk airstream and plate temperatures versus location in core for a counterflow air-to-air heat exchanger with condensation and freezing. Line A represents the plate temperature if the convective heat transfer coefficients for the warm and cold airstreams are equal. Line B represents the plate temperature if the convective heat transfer coefficient of the warm airstream is increased due to condensation. Line C illustrates the impact of hydrodynamic and thermal entrance effects on plate temperature. $T_{h,i}$, $T_{h,o}$, $T_{c,o}$, and $T_{D.P.}$ are the warm inlet, warm outlet, cold inlet, cold outlet, and dewpoint temperatures, respectively.



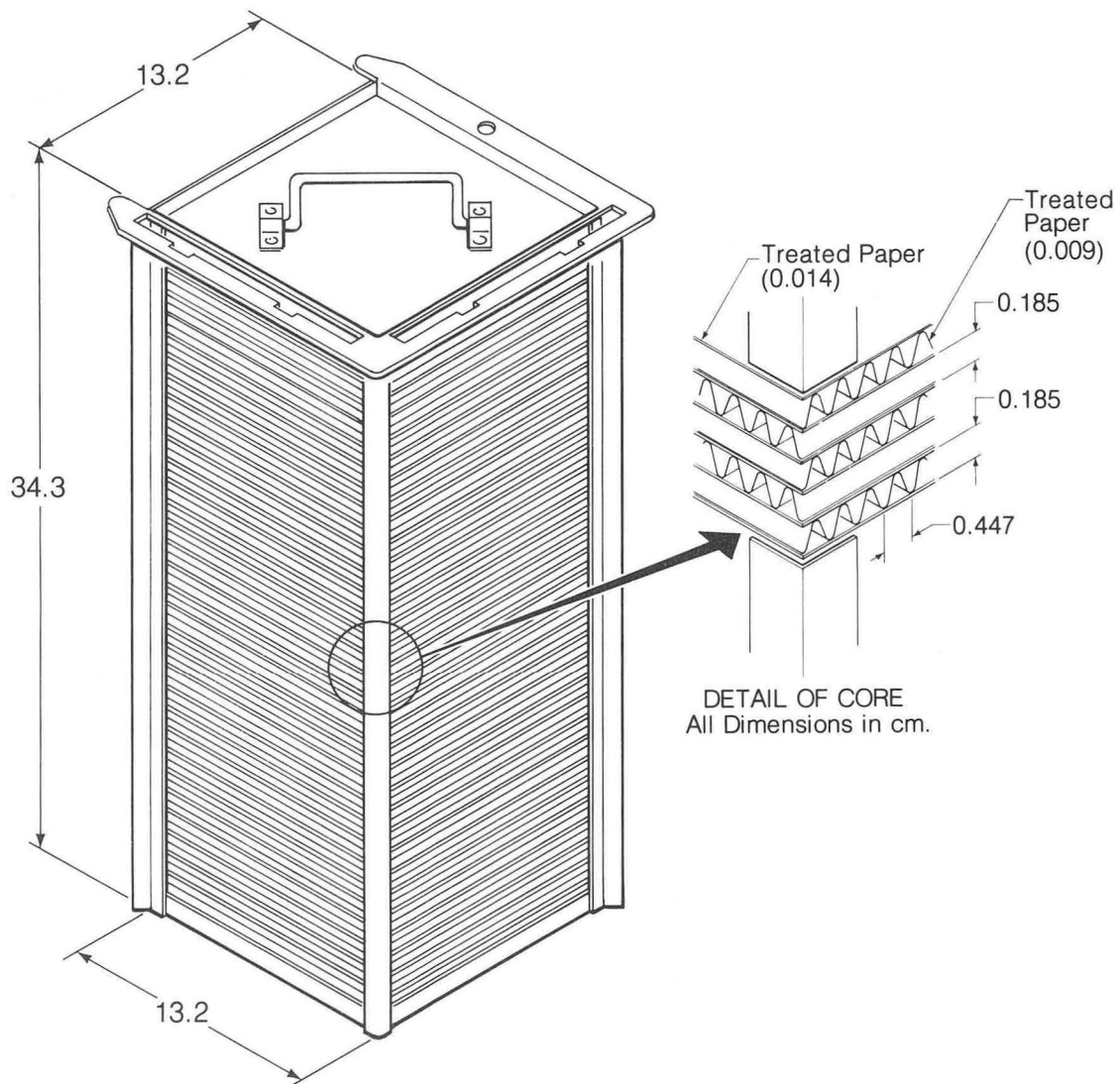
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Figure 2. Engineering drawing of Heat Exchanger No. 1. The components are as follows: 1=cover, 2=mainframe of case, 3=insulation, 4=gasket, 5=filter, 6=core, 7=fan and motor, 8=temperature sensor, 9=condensate drain, 10=heating element, 11=filter. Components No. 8 and 10 were not included in the unit tested. This design is protected by patents.



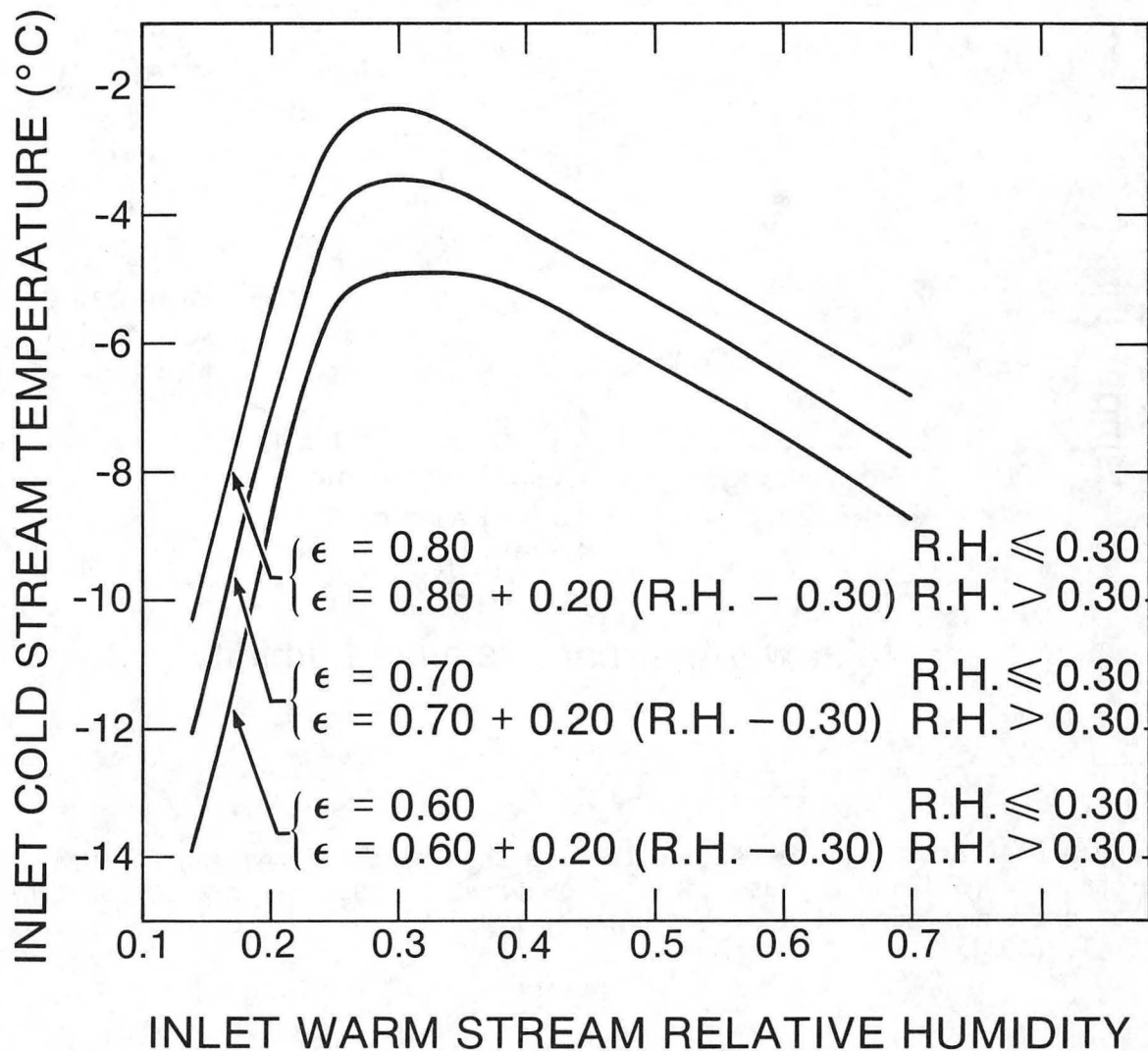
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Figure 3. Illustration of core of Heat Exchanger No. 2. This design is protected by patents.



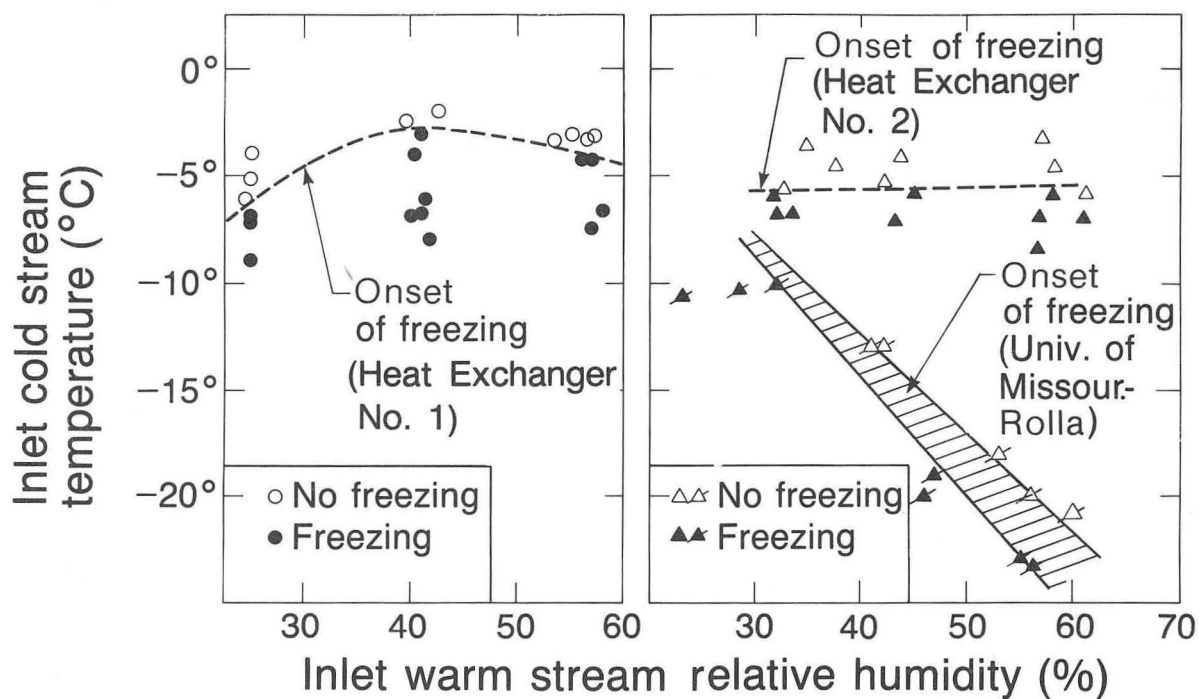
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Figure 4. Illustration of core from Heat Exchanger No. 3.
This design is protected by patents.



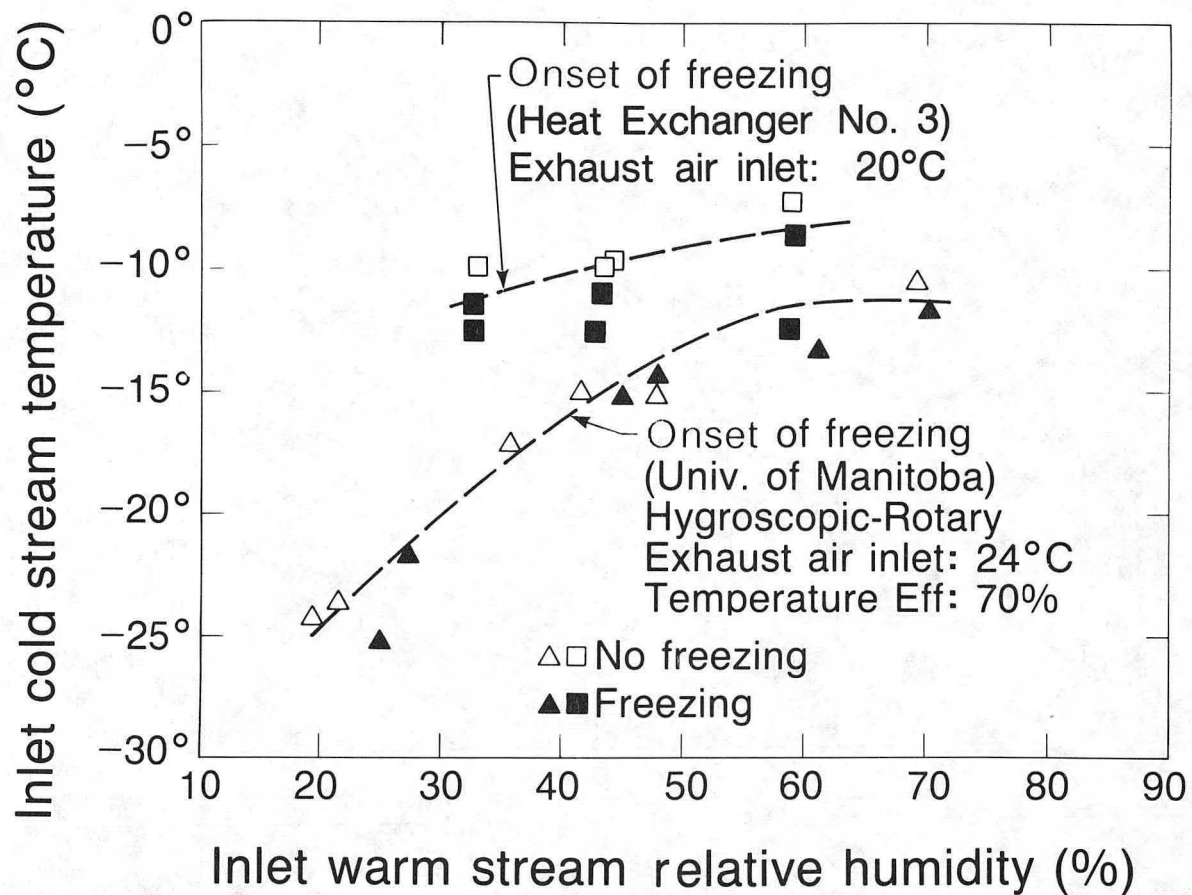
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Figure 5. Predicted temperature at onset of freezing versus humidity and temperature efficiency for a counterflow heat exchanger. The convective heat transfer coefficients for the warm and cold airstreams were assumed to be equal. The inlet temperature of the warm airstream was assumed to equal 20°C. ϵ is the temperature efficiency of the heat exchanger.



XBL 839-3294A

Figure 6. Results of tests to determine the onset of freezing for Heat Exchanger No. 1 and 2 plus results from previous tests with a counterflow heat exchanger conducted at the University of Missouri - Rolla by Sauer et al. (1981).



XBL 839-3289A

Figure 7. Results of tests to determine the onset of freezing for the core of Heat Exchanger No. 3 plus data from a previous study of a hygroscopic-rotary exchanger at the University of Manitoba by Ruth et al. (1975).

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