#3544

Effect of Residential Air-to-Air Heat and Moisture Exchangers on Indoor Humidity

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ABSTRACT

A project was undertaken to develop guidelines for the selection of residential heat and moisture recovery ventilation systems (HRVs) in order to maintain an acceptable indoor humidity for various climatic conditions. These guidelines were developed from reviews on ventilation requirements, HRV performance specifications, and from computer modeling. Space conditions within three house/occupancy models for several types of HRV were simulated for three climatic conditions (Lake Charles, LA; Seattle, WA; and Winnipeg, MB) in order to determine the impact of the HRVs on indoor relative humidity and space-conditioning loads.

Results show that when reduction of cooling cost is the main consideration, exchangers with moisture recovery are preferable to sensible HRVs. For reduction of heating costs, moisture recovery should be done for ventilation rates greater than about 15 L/s and average winter temperatures less than about −10°C if internal moisture generation rates are low. For houses with higher ventilation rates and colder average winter temperatures, exchangers with moisture recovery should be used.

INTRODUCTION

Current ventilation guidelines focus on preventing excessively high concentrations of toxic gases such as formaldehyde and radon. The high ventilation rates that are characteristic of these guidelines can have a significant effect on indoor relative humidity. In dry climates, ventilation in accordance with air quality guidelines may significantly reduce indoor humidity, while the reverse may occur in humid climates.

The indoor relative humidity has important effects on health and building materials. Higher relative humidities tend to increase the survivability of airborne microorganisms, while lower humidities tend to increase the susceptibility of nasal passages to infection. High humidity levels can result in condensation on interior surfaces, which promotes the growth of molds and fungi, which can discolor interior materials and weaken wooden structures.

The indoor relative humidity also affects space-conditioning loads. In dry climates, where the indoor relative humidity tends to be low, the addition of humidifiers usually produces an increase in space heating loads. In humid climates, higher indoor humidities usually result in higher air-conditioning and/or dehumidifying loads.

A possible alternative to humidifiers and dehumidifiers is the heat and moisture recovery ventilator. These heat recovery ventilators transfer moisture in addition to sensible heat. Consequently, they act to maintain indoor humidity conditions. In dry climates, these units will prevent excessively low indoor humidities. In hot, humid climates, moisture recovery units reduce air-conditioning latent loads.

Currently, there is no information on appropriate conditions for the use of moisture recovery ventilators or on their economic benefits. Clearly there is a need for this information. The objective of this project was to establish guidelines for homeowners, contractors, and designers on the selection and operation of residential total heat recovery ventilation systems. The guidelines focus on the role of HRVs in maintaining acceptable indoor humidity in situations where conventional ventilation systems would produce very low indoor humidity in winter or very high humidity in summer. These guidelines were developed from reviews of available information on HRVs and from computer modeling of HRVs for three geographic locations.

The major tasks in this project are listed below:

- Review of residential heat recovery ventilator technology
- Review of current standards and guidelines for acceptable air quality
- Development of a moisture balance model (including models of moisture recovery ventilators)
- Modeling seasonal performance of three heat exchangers at three geographic locations
- Development of heat and moisture exchanger selection guidelines

REVIEW OF RESIDENTIAL ENTHALPY RECOVERY VENTILATOR TECHNOLOGY

This section summarizes performance characteristics of residential enthalpy recovery units. Presently, there are

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two types of residential heat and moisture recovery ventilators: the rotary heat exchanger type and the porous plate type. The subsections that follow describe the operating principles of the two types of ventilators and their cross-flow leakage and frosting characteristics.

Principles of Operation

Rotary Heat Exchanger Heat Recovery Ventilators. Rotary heat exchangers transfer heat between supply and exhaust airstreams through a rotating core that picks up and stores heat from the hot stream and releases it to the cold stream. The core may contain a desiccant material that allows increased transfer of moisture. Further details on desiccant heat exchangers are provided in the subsections below.

Non-Desiccant Wheel Heat Recovery Ventilators. If the core is made of a non-desiccant material, moisture is transferred only if it condenses from the warm airstream on the core and then evaporates in the colder stream. This will only happen if the cold stream is well below the dew point of the hot stream.

Desiccant Wheel Heat Recovery Ventilators. In this device, the wheel contains a desiccant material that will absorb moisture from the humid airstream and then release the moisture into the less humid stream. The desiccant wheel will also transfer moisture through condensation.

Porous Plate Heat Recovery Ventilators. The plates in a porous plate heat recovery ventilator are normally made from a specially treated paper that has good sensible heat transfer characteristics and a high moisture permeability.

Cross-Flow Leakage and Frosting

Cross-Flow Leakage Characteristics. The following subsections describe the cross-flow contamination characteristics of the two types of enthalpy recovery ventilators.

Rotary Heat Exchanger Heat Recovery Ventilators. Fisher et al. (1975) conducted tests on microbiological and tracer gas contamination in large (56 in and 37 in diameter) rotary heat exchangers. The large heat wheel had a randomly packed plastic hygroscopic core, while the small wheel was corrugated asbestos impregnated with lithium chloride. Both wheels had purging mechanisms to reduce contamination. It was found that microbiological contamination was 2.45% and 0.1% for the larger and smaller wheels, respectively. Tracer gas contamination was estimated to be less than 0.1%.

Fisk et al. (1985) studied the transfer of water vapor, formaldehyde, and tracer gases in desiccant wheel heat exchangers. Tracer gases were found to be transferred between airstreams by carryover and leakage. Typical tracer gas transfer efficiency was 5% to 7%. Water vapor and formaldehyde transfer rates were much higher than those of the tracer gases, usually about 55% and 10%, respectively. It was concluded that the desiccant was responsible for these higher rates. The formaldehyde transfer efficiency was approximately 20% of the water vapor transfer efficiency.

Porous Plate Heat Recovery Ventilators. Fisk et al. (1985) studied the transfer of water vapor, formaldehyde, and tracer gases in porous plate heat exchangers. Tracer gases were transferred between airstreams directly through the treated paper and by leakage. Typical tracer gas transfer effectiveness was 5% to 8%. Water vapor and formaldehyde transfer rates were much higher than those of the tracer gases, typically 30% and 10%, respectively. The formaldehyde transfer efficiency was about 30% to 50% of the water vapor transfer efficiency.

Frosting Characteristics. The following subsections describe the frosting characteristics of the two types of enthalpy recovery ventilators.

Rotary Heat Exchanger Heat Recovery Ventilators. As the exhaust stream moves through the core, moisture is removed. Consequently, the dew point of the exhaust stream is lowered, so that the temperature for frosting is reduced. Therefore, frosting on a desiccant wheel heat exchanger should occur at lower temperatures than other types of heat exchangers. This has been confirmed in measurements by Ruth et al. (1975) and Fisk et al. (1983), who found that frosting in a desiccant wheel heat exchanger occurs about 5° to 15°C lower than conventional heat exchangers.

Porous Plate Heat Recovery Ventilators. Like the desiccant wheel heat exchanger, frosting on a porous plate heat exchanger should occur at lower temperatures than other types of heat exchangers. This is because the dew point of the exhaust stream is lowered as it travels through the heat exchanger core. Measurements by Fisk et al. (1983) have shown that, compared to other conventional heat exchangers, the onset of freezing for a porous plate heat exchanger is lower by about 5°C.

A manufacturer of the paper core claims that the paper will not deteriorate if it is not exposed to condensation, direct sunlight, or large amounts of oil mist. In order to prevent condensation in cold climates, the outdoor air may require preheating before it enters the unit.

CURRENT STANDARDS AND GUIDELINES FOR VENTILATION AND ACCEPTABLE AIR QUALITY

Ventilation systems are intended to maintain acceptable comfort and air quality. Consequently, the guidelines for the selection of HRVs must ensure acceptable humidity levels and incorporate existing standards on ventilation. The subsections below identify both acceptable and unacceptable humidity levels and describe current ventilation requirements.

Effects of Indoor Relative Humidity on Health, Comfort, and Materials

Control of indoor humidity is an important consideration in residences since it can have a significant impact on occupant health and comfort.

Respiratory Illnesses. Respiratory illnesses have been shown by a large number of investigators to have the greatest incidence during the winter (Green et al. 1985; Green 1974). Lubart (1962) found that low relative humidities result in drying of nasal passages, which are a favorable environment for infective agents. Green et al. (1985) and Green (1974) have found that the survival time of the bacteria increased with increasing humidity. It has been shown by Lester (1948) that mice had reduced in-

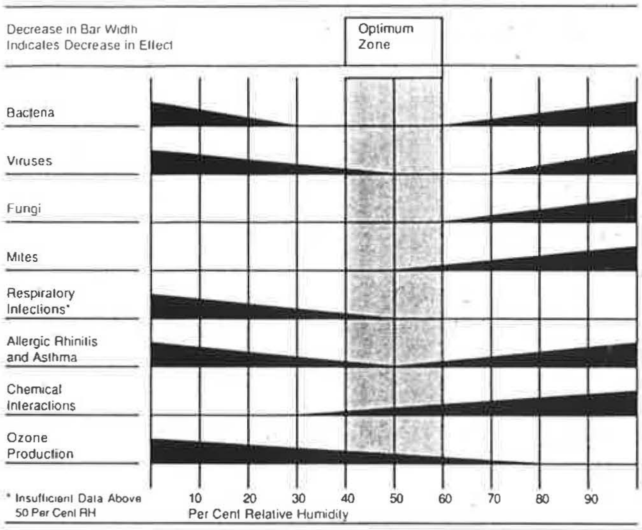

Figure 1 *Optimal relative humidity range (source: Sterling et al. [1985])*

fluenza mortality rates at relative humidities between 30% and 80%.

Allergic Diseases. The most frequently occurring indoor allergens are proteins from house dust mites, animal dander, and fungal spores. According to Andersen and Korsgaard (1984), the optimum temperature and humidity for house dust mites is 25°C and 75% to 80% RH. However, they can thrive at temperatures of 17°C to 32°C and relative humidities of 50% to 80%. Very few dust mites are found in homes with temperatures of 20° to 22°C and relative humidities of less than 45%. Fungi require relative humidities of between 75% and 95%. These fungi release spores that produce allergic reactions.

Humidity and Odors. The perception and irritation caused by suspended particles and vapors decreases with increasing humidity. According to ASHRAE (1985), the relative humidity should be between 45% and 60% if odor perception and irritation are to be minimized.

Comfort. According to the ASHRAE Comfort Chart (ANSI/ASHRAE 1981), at temperatures between 20°C and 26°C, the relative humidity should be maintained between approximately 30% and 70%.

Static Electricity. Low relative humidities promote the accumulation of electrostatic charges in materials. ASHRAE (1983) states that relative humidities greater than 45% will reduce or eliminate this.

Effects on Building Structures and Interiors. The indoor relative humidity can result in warps or shrinkage in wood floors or can produce cracks in furniture. High indoor humidities can result in condensation inside walls, which can cause damage to structural components, insulation, and outside paint.

Condensation. High humidity levels can result in condensation on interior surfaces, which promotes molds and fungi growth. Molds and fungi can discolor paints and plasters and can weaken wood structures.

Summary. The information covered in the previous sections is summarized in the chart in Figure 1 (reproduced from Sterling et al. 1985). From this information, it can be concluded that the most desirable range for the indoor relative humidity is between 40% and 60%.

Survey of Ventilation Guidelines

ASHRAE Ventilation Guidelines. ASHRAE is currently updating guidelines for acceptable indoor air quality (ANSI/ASHRAE 1986). The proposed guidelines for residences are listed below:

Living areas	.35 ach but not less than 7.5 L/s per person
Kitchens	50 L/s intermittent mechanical exhaust
Bathrooms	25 L/s intermittent mechanical exhaust

The section on humidity (Section 5.11) in this standard concerns humidity in bathrooms and bedrooms. For bathrooms, no guidelines are given beyond those quoted above. For bedrooms, the standard states that the "relative humidity shall be maintained below 55% during the heating season to minimize dust mite concentrations in bedrooms."

R2000 Ventilation Guidelines. The R2000 ventilation guidelines were developed for energy-efficient houses built to R2000 standards (developed by the Department of Energy, Mines and Resources Canada). These houses have minimal air infiltration and must have continuous mechanical ventilation. These guidelines (from Canadian Home Builders Association 1987) are summarized below:

	Minimum Continuous Ventilation	Required Capability for Intermittent Exhaust
Bedrooms, living room, dining room	5 L/s	
Kitchen	5 L/s	25 L/s
Bathroom	5 L/s	50 L/s
Living/dining room, utility rooms, basement	10 L/s	

DEVELOPMENT OF AN ENTHALPY BALANCE MODEL

The information on heat recovery ventilators in the previous sections does not provide a complete set of selection guidelines. In order to determine the impact on humidity level in a house, it is necessary to use a mathematical model for the temperature and humidity of the air within the home. Consequently, a computer simulation was developed from an existing time-dependent program to estimate the humidification load, the dehumidification load, the total space heating and cooling energy, and the range of indoor relative humidity. Subroutines were added to the program for:

- moisture recovery ventilators
- absorption/desorption in wall materials
- moisture storage and transport in basement concrete
- condensation and evaporation on windows and sills.

Details on these subroutines are given below.

Moisture Recovery Ventilator Modeling

The types of ventilators that were modeled in the subroutine were:

- ventilators with no heat recovery
- ventilators with only sensible heat recovery
- non-desiccant wheel moisture recovery ventilators
- porous plate moisture recovery ventilators
- desiccant wheel moisture recovery ventilators.

The information used to develop the models was obtained from test data, manufacturers' literature, and journal publications. The impact of defrosting or frost avoidance mechanisms was included in the models.

Ventilator Volume Flow Rate. The heat recovery ventilators were sized in accordance with the previously mentioned ASHRAE or R2000 ventilation guidelines. The ventilators were assumed to have two flow speeds, with the low-speed flow providing the required continuous ventilation rate (defined by ASHRAE or R2000 guidelines). The ventilator was assumed to switch to high-speed operation when the indoor humidity (as measured at a humidistat in one of the zones) exceeded 60%. The modeling assumed that the high-speed volume flow rate was twice the low-speed volume flow rate.

Sensible Heat Transfer Efficiency. The sensible heat transfer efficiency is the recovered sensible energy less the supply fan and preheater energy, divided by the exhausted sensible energy plus the exhaust fan energy (Canadian Standard Association 1985). For balanced flows, the equation for this is:

$$e_s = [m' C_p (T_2 - T_1) - Q_{sf} - Q_h] / [m' C_p (T_3 - T_1) + Q_{ef}] \quad (1)$$

where

e_s = sensible heat transfer efficiency
m' = mass flow rate of exhaust and supply airstreams (assuming balanced flows)
C_p = heat capacity of air
T_2 = temperature of air supplied to the house that is leaving the ventilator
T_1 = outdoor air temperature
Q_{sf} = power consumption by the supply fan
Q_h = heat addition to supply air by the preheater
T_3 = indoor air temperature
Q_{ef} = power consumption of the exhaust fan

Efficiency at Conditions Above the Frosting Threshold. The various types of heat recovery ventilators have slightly different values for the sensible heat transfer efficiency. In order to allow a fair comparison between the units, the following assumptions were made regarding the efficiencies at conditions above the frosting threshold:

- The sensible efficiency above the onset of frosting was assumed to be dependent only on the mass flow rate of air:

$$e_s = e_{so}(m') \quad (2)$$

- The normal sensible heat transfer efficiency at low-speed operation was set to 70%. This value is representative of heat recovery ventilators. Since the ventilators were sized to match ASHRAE or R2000 ventilation requirements, 70% sensible heat transfer efficiency would always occur at low-speed operation.
- When heat recovery ventilators switch to high-speed operation, the model uses a sensible heat transfer efficiency of 62%. This was based on an examination of the variation of efficiency in sensible heat recovery ventilators with volume flow (Barringer et al. 1989). This study indicated that sensible efficiency dropped by a factor of about 0.89 when the volume flow rate was doubled. Based on this finding, if the low-speed efficiency is to be 70%, then the high-speed efficiency would be 62%.

Modeling Efficiency Losses Due to Frosting. The heat recovery ventilator model assumed a preheater was used to prevent the formation of frost. The preheater power is equivalent to a loss in sensible heat transfer efficiency. As a consequence, the variation of sensible efficiency with temperature conditions and volume flow rate was found using the general equation below.

$$e_s(m') = e_{so}(m') - m' C_p (T_{1e} - T_1) / [m' C_p (T_3 - T_1) + Q_{ef}] \quad (3)$$

where

$e_{so}(m')$ = sensible transfer efficiency of ventilator at conditions above the frosting threshold
T_{1e} = temperature of the supply air entering the heat exchanger core

The numerator in the above equation is the preheater power. The power required by the preheater must be high enough to raise the temperature of the air entering the core to a threshold temperature at which frost begins to form. Therefore, the temperature of the air entering the heat exchanger core is:

$$T_{1e} = T_1 \text{ for } T_{frst} < T_1 \quad (4)$$
$$= T_{frst} \text{ for } T_{frst} > T_1$$

where

T_{frst} = threshold outdoor temperature for the formation of frost in the heat exchanger core.

Equations have been developed for T_{frst} (Barringer et al. 1989) for each type of ventilator. For a given ventilator. the frosting threshold outdoor temperature was found to be a function of outdoor temperature and indoor humidity and temperature conditions.

Characterization of Sensible Efficiency in the Computer Program Input Files. The data required by the program to define the performance of the heat recovery ventilators was the sensible heat transfer efficiency at five outdoor temperatures (selected by the user) at low speed and high speed. The program interpolates the efficiency at intermediate temperatures. The program implicitly assumes that the efficiency is a function of outdoor temperature and volume flow rate only. The following steps were followed to generate the required performance figures:

- Relationships were developed for the frosting threshold outdoor temperature, T_{frst}, for each type of ventilator.
- Using these T_{frst} relationships, efficiencies were calculated for a representative range of indoor and outdoor conditions for each type of ventilator.
- These efficiencies were plotted against outdoor temperature and curve fits were obtained. This produced equations of efficiency as a function of outdoor temperature.
- Five outdoor temperatures were selected, ranging from −30°C to 35°C and applied to the curve fit equations to generate the efficiency data required by the computer program.

The subsections below provide these curve fit equations for each type of ventilator.

Sensible Heat Recovery Ventilator. The efficiency was assumed to be a function of mass flow rate only. The low-speed and high-speed efficiency were set to 70% and 62%, respectively.

Non-Desiccant Wheel Heat Recovery Ventilator. The model for the non-desiccant wheel heat recovery ventilator was developed from a theoretical study by Holmberg (1977). In this model, the sensible efficiency is constant above the frosting threshold. As mentioned above, there is additional power consumption due to a preheater used to prevent frosting, which results in reduction in sensible efficiency. Curve fits to Holmberg's results yielded the following equations:

low-speed

$$e_s = 0.70\,[1 - 0.0324\,(-4.28 - T_1)] \quad \text{for } T_1 \leqslant -4.28°\text{C} \qquad (5)$$
$$= 0.70 \quad \text{for } T_1 > -4.28°\text{C}$$

high-speed

$$e_s = 0.62\,[1 - 0.0324\,(-4.28 - T_1)] \quad \text{for } T_1 \leqslant -4.28°\text{C} \qquad (6)$$
$$= 0.62 \quad \text{for } T_1 > -4.28°\text{C}$$

where

e_s = sensible efficiency
T_1 = outdoor temperature

Desiccant Wheel Heat Recovery Ventilator. The information on the performance of a desiccant wheel heat recovery ventilator was obtained from Hoagland (1986). According to this study, the efficiency of the desiccant wheel heat recovery ventilator was constant above the onset of frosting. However, like the non-desiccant wheel heat recovery ventilator, there is an effective reduction in the efficiency due to a preheater. As before, curve fits to the efficiency for representative indoor and outdoor conditions yielded the following equations:

low-speed

$$e_s = 0.70\,(1 - 9.747 \times 10^{-4}\,T_1 - 6.881 \times 10^{-4}\,T_1^2) \quad \text{for } T_1 \leqslant 0°\text{C} \qquad (7)$$
$$= 0.70 \quad \text{for } T_1 > 0°\text{C}$$

high-speed

$$e_s = 0.62\,(1 - 9.747 \times 10^{-4}\,T_1 - 6.881 \times 10^{-4}\,T_1^2) \quad \text{for } T_1 \leqslant 0°\text{C} \qquad (8)$$
$$= 0.62 \quad \text{for } T_1 > 0°\text{C}$$

Porous Plate Heat Recovery Ventilator. Information on the performance of a porous plate ventilator was obtained from a major manufacturer. Curve fits of the efficiency for representative indoor and outdoor conditions produced the following equations for the efficiency:

low-speed

$$e_s = 0.70\,[1 + 0.001714\,T_1 - 2.24 \times 10^{-4}\,T_1^2 + 6.41 \times 10^{-6}\,T_1^3] \quad \text{for } T_1 \leqslant 0°\text{C} \qquad (9)$$
$$= 0.70 \quad \text{for } T_1 \; ^6\, 0°\text{C}$$

high-speed

$$e_s = 0.62\,[1 + 0.001714\,T_1 - 2.24 \times 10^{-4}\,T_1^2 + 6.41 \times 10^{-6}\,T_1^3] \quad \text{for } T_1 \leqslant 0°\text{C} \qquad (10)$$
$$= 0.62 \quad \text{for } T_1 \; ^6\, 0°\text{C}$$

Moisture Transfer Efficiency. The moisture transfer efficiency is the ratio of the recovered moisture in the air supplied to the house, to the moisture that is expelled in the exhaust air. Assuming balanced airflows, the equation for moisture transfer efficiency is:

$$e_m = (w_2 - w_1) / (w_3 - w_1) \qquad (11)$$

where

e_m = moisture transfer efficiency
w_2 = humidity ratio of supply air leaving the ventilator and entering the house
w_1 = humidity ratio of outdoor air
w_3 = humidity ratio of indoor air.

The various types of heat recovery ventilators have differing moisture transfer efficiencies. To allow a comparison between the units, the following assumptions were made:

- It was assumed that the maximum moisture transfer efficiency at low speed was 70%. This is representative of these units.
- It was assumed that the moisture transfer efficiency would decrease with the sensible transfer efficiency as the air volume flow is increased. Consequently, the moisture transfer efficiency was assumed to be 62% at high-speed operation.

The program models the moisture transfer efficiency as a function of humidity ratio differential (difference between the indoor and outdoor air humidity ratio) and fan speed. The inputs for the program are the moisture transfer efficiency at five humidity ratio differentials at low speed and high speed. The efficiency is interpolated intermediate differentials. The procedure for calculating the required moisture transfer efficiencies is described below:

- Moisture transfer efficiencies were calculated for a representative range of indoor and outdoor conditions for each type of ventilator.
- These efficiencies were plotted against humidity ratio differential, and curve fits were obtained. This produced equations of efficiency as a function of humidity ratio differential.

- Five humidity ratio differentials were selected, ranging from 0 to 0.008, and applied to the curve fit equations to generate the efficiency data required by the computer program.

The moisture transfer efficiency equations resulting from the above procedure are given in the subsections below.

Sensible Heat Recovery Ventilator. The sensible heat recovery ventilator has zero moisture transfer efficiency.

Non-Desiccant Wheel Heat Recovery Ventilator. The moisture transfer efficiency of the non-desiccant wheel heat recovery ventilator is a function of a broad range of variables. Consequently, defining moisture transfer efficiency only in terms of the humidity differential is a simplification. However, using the above-described procedure, the moisture transfer efficiency is given by:

low-speed

$$
\begin{aligned}
e_m &= 0 && \text{for } \Delta w \leqslant 0.003358 \\
&= 0.70\,[1660.6\,(\Delta w - 0.003358)]^{.769} && \text{for } 0.003358 < \Delta w < 0.0039602 \\
&= 0.70 && \text{for } 0.0039602 < \Delta w
\end{aligned}
\qquad (12)
$$

high-speed

$$
\begin{aligned}
e_m &= 0 && \text{for } \Delta w \leqslant 0.003358 \\
&= 0.62\,[1660.6\,(\Delta w - 0.003358)]^{.769} && \text{for } 0.003358 < \Delta w < 0.0039602 \\
&= 0.62 && \text{for } 0.0039602 < \Delta w
\end{aligned}
\qquad (13)
$$

where

$\Delta w = w_3 - w_1$ = humidity ratio differential
e_m = moisture transfer efficiency
w_3 = humidity ratio of the indoor exhaust air
w_1 = humidity ratio of the outdoor air.

Desiccant Wheel Heat Recovery Ventilator. According to Hoagland, the moisture transfer efficiency of the desiccant wheel heat recovery ventilator was constant. As mentioned at the beginning of this section, the moisture transfer efficiency is set to 0.70 at low speed, and 0.62 at high speed.

Porous Plate Heat Recovery Ventilator. Like the non-desiccant wheel ventilator, the moisture transfer efficiency of the porous plate ventilator is a function of several variables. Using the above-described procedure, the equation for the efficiency as a function of humidity ratio differential is:

low-speed

$$
\begin{aligned}
e_m &= 0.70\,(0.5596 + 112.97\,\Delta w - 6571\Delta w^2) && \text{for } 0 < \Delta w < .006 \\
&= 0.70 && \text{for } \Delta w \geqslant .006
\end{aligned}
\qquad (14)
$$

high-speed

$$
\begin{aligned}
e_m &= 0.62\,(0.5596 + 112.97\,\Delta w - 6571\Delta w^2) && \text{for } 0 < \Delta w < .006 \\
&= 0.62 && \text{for } \Delta w \geqslant .006
\end{aligned}
\qquad (15)
$$

Moisture Absorption and Desorption in Walls, Floors, Ceilings, and Furnishings

Several studies have shown that moisture storage in building interiors can have a significant impact on indoor relative humidity. For example, according to research done at the Florida Solar Energy Center (Fairey et al. 1986), if a house is cooled at night with humid outdoor air, the building interior will absorb moisture from the air. If an air conditioner is used during the day to cool the house, it will experience an additional latent load as stored moisture is released from interior materials.

Most of the available moisture storage data for materials are limited to graphs showing the variation of stored moisture with relative humidity. Little attention has been given to dynamic behavior. However, Kusuda and Miki (1985) investigated the response of various materials to step increases in air humidity. Plots of moisture content vs. time indicate that the absorption rates appear to follow an exponential decay. Consequently, it was decided to model the rate of moisture absorption into materials with a simple first-order differential equation. The only variables in this equation are the hygroscopic constant (which is the ratio of water concentration in the material to the air's relative humidity) and a time constant. The time constants were estimated from Kusuda's step response curves.

Moisture Storage and Transport in Basement Concrete

Basement concrete contributes to indoor moisture characteristics by storing moisture and by allowing diffusion of moisture from the ground. There is very little information on the dynamic moisture storage characteristics of concrete. However, data collected by Kusuda and Achenbach (1963) indicate time response characteristics very similar to those of walls described previously. Therefore, as in the case of wall moisture storage, it was decided to model the rate of moisture absorption into concrete with a simple first-order differential equation.

A time constant for concrete was estimated to be one day from the data obtained by Kusuda and Achenbach (1963). However, due to the scatter in these data, this time constant was not considered to be reliable. Consequently, test runs were done in which the concrete time constant was varied from 0.1 days to 100 days. The results showed that this time constant had little effect on heating and cooling loads.

The model for the diffusion of moisture through basement floors assumes that the concrete slab is not in contact with liquid water. Consequently, the flow of moisture is governed by the water vapor pressure difference across the concrete and the concrete's permeability to water vapor. Estimates for the vapor diffusion flow rate range from close to zero to about 3 kg/day.

Window Condensation and Evaporation

This model included a model for the sill. If the condensed water on the window reached a critical mass (per unit area), subsequent condensation was deposited on the sill. Evaporation could then occur at the sill.

SIMULATION OF ENTHALPY RECOVERY VENTILATORS IN THREE RESIDENCES AT THREE GEOGRAPHIC LOCATIONS: INPUT FILE SPECIFICATION

Climates Used for the Simulation

There is a broad range of climatic types in North America. The climates selected for the simulations were intended to include widely differing climates. The selected climates are described below:

	heating degree-days (°C-day)	winter design dry-bulb (°C)	summer design dry-bulb (°C)	summer design wet-bulb (°C)
Winnipeg	5874	−34	30	23
Seattle	2881	−6	28	19
Lake Charles	797	−3	34	25

Winnipeg was chosen to represent a cold climate with dry winters. Seattle was selected for having a cold, relatively moist climate. Lake Charles was chosen to represent a warm, humid climate. Since there was a limit to the number of climates that could be modeled, it was decided not to include hot, dry climates.

General Description of Houses Used in the Simulation

House 1. House 1 has a floor area of approximately 100 m^2 with no basement and is occupied by two adults. The floor plan for house 1 was based on a design from Canada Mortgage and Housing Corporation (1972). The zones in this house were: living room/master bedroom, kitchen/dining room/childrens' room, and crawlspace. Significant features of this house are: the living room and the master bedroom face south, the living room and the master bedroom are carpeted, and the crawlspace is unheated.

House 2. House 2 has a floor area of approximately 200 m^2, including basement, and is occupied by two adults and two young children. The floor plan for house 2 is the same as house 1 except that a full basement is added. The zones in this house were: living room/master bedroom, kitchen/dining room/childrens' room, and finished basement. The only significant difference between this house and house 1 is the addition of the finished insulated basement.

House 3. House 3 has a floor area of approximately 300 m^2, including basement, and is occupied by two adults and four children. The floor plan for house 3 was obtained from CMHC (1972). The zones in this house are: living room/den, washroom/dining room/kitchen, basement, and master bedroom/childrens' bedroom/main bathroom. Significant features of this house are: the living room and den face south; the living room, den, master bedroom, and childrens' bedroom are carpeted; the basement is heated; and the basement may or may not have significant vapor diffusion from the ground.

Ventilation Rates

Two ventilation rates were used in the modeling: the ASHRAE Standard 62-1981R ventilation rate and the R2000 ventilation rate. The calculated ventilation rates are given below:

	ASHRAE low-speed (L/s)	ASHRAE high-speed (L/s)	R2000 low-speed (L/s)	R2000 high-speed (L/s)
house 1	20	40	40	80
house 2	30	60	50	100
house 3	55	110	70	140

Power consumption by the ventilation fans was assumed to be 0.5 W/(L/s). This was based on a survey of power consumption figures for several heat recovery ventilators. Additional ventilation is supplied by the exhaust fans.

The ventilators switched to high speed when the indoor relative humidity at the ventilator inlet exceeded 60%. Additional runs were carried out for Winnipeg at other setpoints to determine the ventilation rate that would result from the use of humidity level to control ventilation.

All three houses were assumed to have essentially no infiltration except for that required to make up for kitchen and bathroom exhaust fans. Only one exception to this was the crawlspace of house 1. In this case, the infiltration rate was set at 0.3. Some simulations were carried for the Winnipeg area with infiltration rates of 0.1 to determine the effect of infiltration on the humidity levels.

Space Heating, Humidifying, and Cooling Equipment

In all the houses, the space heating was done by a central electric forced-air furnace. The capacity of the furnace was selected to meet all space heating loads. The humidifier was assumed to be connected to the furnace. Consequently, the humidifier could only operate when the furnace was in operation. The capacity of the humidifier was 4 L/h (the capacity of standard furnace humidifiers).

The air conditioner was sized to exceed the maximum cooling load. The cooling coil surface temperature and the bypass factor were set at representative values of 11°C and 0.1, respectively.

In each of the houses, the heating, cooling, and ventilating systems were controlled by the temperature and humidity in the zone that contained the dining room and kitchen.

Exterior Walls, Ceilings, and Floors

Characteristics common to all the buildings are listed below:

Interior wall and ceiling material	Gypsum drywall
Thermal resistance (RSI)	
outer walls	2.12
interior walls	
windows	0.35
roof	3.56
doors	0.96
crawlspace floor	2.12
uncarpeted floors between heated zones	0.65
carpeted floors between heated zones	1.03

According to Barakat (1985), the air exchange rate between zones is 150 L/s for an open doorway. For different sizes of openings, it was assumed that the air exchange rate was directly proportional to the opening area.

THERMAL CAPACITY AND MOISTURE STORAGE DATA

Internal Sensible Load Profile

The sensible internal load profiles for each zone were derived from profiles developed by the Florida Solar Energy Center (FSEC) (Fairey et al. 1986). The house with the greatest similarity to the FSEC house is house 2.

Internal Latent Load Profile

It was assumed that the latent load profile matches the sensible load profile. The total daily water generation rates for each zone were derived from the FSEC data. The daily total generation rates are listed below:

	House 1 (L/day)	House 2 (L/day)	House 3 (L/day)
Zone 1	2.29	2.84	1.89
Zone 2	2.75	6.05	6.25
Zone 3	–	0.63	0.00
Zone 4	–	–	5.99
Total	5.04	9.52	4.13

Kitchen and Bathroom Exhaust Fan Schedule

The program requires for each zone: the maximum exhaust flow rate (in L/s) and the fraction of maximum exhaust flow rate for each hour of the day. The maximum exhaust flow rate in each zone was taken as the sum of exhaust fan capacities in each of the zones. The exhaust fan capacity was taken as that recommended by Kusuda and Achenbach (1963): 50 L/s for kitchens and 25 L/s for each bathroom. The schedule of exhaust fan usage was derived from the FSEC data.

RESULTS OF SIMULATIONS

Simulation runs were carried out for a one-year period for the three house types in the three climatic locations: Lake Charles, LA; Seattle, WA; and Winnipeg, MB. Many additional simulation runs were required for Winnipeg to develop the selection guidelines. The results of the simulations are presented below.

Results for Lake Charles, LA

Lake Charles is situated on the Gulf of Mexico. It was selected to investigate the performance of heat and moisture exchangers in a warm, humid climate. In this case the primary purpose of the heat exchanger is to cool and

TABLE 1
Results of Simulations for Lake Charles, LA

	HRV Description	Ventilation Rate L/s	Heating Latent Load GJ	Heating Net Energy GJ	Cooling Latent Load GJ	Cooling Net Energy GJ	Annual Cost @ $0.045/kWh $/yr	Annual Cost @ $0.10/kWh $/yr
House 1	none	20	0.06	6.8	11.6	18.3	314	697
	plate	20	0.08	5.3	11.6	18.3	295	656
	rotary, non-desiccant	20	0.06	5.3	11.6	18.3	295	656
	porous plate	20	0.01	5.3	9.6	17.5	285	633
	rotary, desiccant	20	0.00	5.4	7.9	16.8	278	617
	none	40	0.40	9.4	17.1	20.5	374	831
	plate	40	0.50	6.5	17.2	20.6	339	753
	rotary, non-desiccant	40	0.50	6.4	17.2	20.6	338	750
	porous plate	40	0.14	6.1	12.8	18.8	311	692
	rotary, desiccant	40	0.00	5.9	9.6	17.5	293	650
House 2	none	30	0.00	5.2	17.8	22.9	351	781
	plate	30	0.02	3.5	17.8	23.1	333	739
	rotary, non-desiccant	30	0.02	3.4	17.9	23.1	331	736
	porous plate	30	0.00	3.5	15.1	21.9	318	706
	rotary, desiccant	30	0.00	3.8	13.1	21.2	313	694
	none	50	0.13	7.2	23.0	25.1	404	897
	plate	50	0.20	4.1	23.2	25.3	368	817
	rotary, non-desiccant	50	0.14	3.9	23.2	25.3	365	811
	porous plate	50	0.02	4.0	18.2	23.2	340	756
	rotary, desiccant	50	0.00	4.2	14.4	21.6	323	717
House 3	none	55	0.00	7.1	32.0	32.6	496	1,103
	plate	55	0.00	3.8	32.3	32.5	454	1,008
	rotary, non-desiccant	55	0.00	3.7	32.3	32.6	454	1,008
	porous plate	55	0.00	3.9	26.1	29.9	423	939
	rotary, desiccant	55	0.00	4.2	21.4	27.9	401	892
	none	70	0.00	8.5	25.8	34.4	536	1,192
	plate	70	0.00	4.2	34.0	1.4	70	156
	rotary, non-desiccant	70	0.00	4.0	34.0	31.4	443	983
	porous plate	70	0.00	4.2	29.1	31.2	443	983
	rotary, desiccant	70	0.00	4.6	22.5	28.4	413	917

TABLE 2
Results of Simulations for Seattle, WA

	HRV Description	Ventilation Rate L/s	Heating Latent Load GJ	Heating Net Energy GJ	Cooling Latent Load GJ	Cooling Net Energy GJ	Annual Cost @ $0.045/kWh $/yr	Annual Cost @ $0.10/kWh $/yr
House 1	none	20	0.20	25.7	1.3	3.4	364	808
	plate	20	0.30	21.4	1.3	3.6	313	694
	rotary, non-desiccant	20	0.20	21.1	1.3	3.6	309	686
	porous plate	20	0.00	21.0	1.5	3.6	308	683
	rotary, desiccant	20	0.00	21.5	1.8	3.7	315	700
	none	40	1.30	34.0	1.0	3.1	464	1,031
	plate	40	1.40	25.1	1.0	3.4	356	792
	rotary, non-desiccant	40	1.40	24.6	1.0	3.4	350	778
	porous plate	40	0.50	23.5	1.2	3.5	338	750
	rotary, desiccant	40	0.00	22.9	1.5	3.6	331	736
House 2	none	30	0.00	26.9	1.7	3.5	380	844
	plate	30	0.00	20.8	1.7	3.9	309	686
	rotary, non-desiccant	30	0.00	20.5	1.7	3.9	305	678
	porous plate	30	0.00	21.0	2.2	4.0	313	694
	rotary, desiccant	30	0.00	22.2	2.7	4.2	330	733
	none	50	0.80	34.1	1.2	3.1	465	1,033
	plate	50	0.80	23.7	1.3	3.6	341	758
	rotary, non-desiccant	50	0.80	23.1	1.3	3.6	334	742
	porous plate	50	0.00	22.5	1.7	3.8	329	731
	rotary, desiccant	50	0.00	23.0	2.3	3.9	336	747
House 3	none	55	0.20	35.5	1.6	3.7	490	1,089
	plate	55	0.20	24.0	1.6	4.1	351	781
	rotary, non-desiccant	55	0.10	23.3	1.6	4.1	343	761
	porous plate	55	0.00	23.8	2.1	4.3	351	781
	rotary, desiccant	55	0.00	25.0	2.9	4.5	369	819
	none	70	0.60	40.9	1.4	3.5	555	1,233
	plate	70	0.60	26.0	1.3	4.0	375	833
	rotary, non-desiccant	70	0.70	25.3	1.4	4.0	366	814
	porous plate	70	0.00	25.0	1.8	4.2	365	811
	rotary, desiccant	70	0.00	25.9	2.6	4.4	379	842

dehumidify the ventilation air entering the house. Simulations were carried out for the three house sizes with two different ventilation rates and five different ventilation systems, giving a total of 30 runs. A summary of the results is given in Table 1. The table shows the ventilation rate provided by the ventilator, the heating energy required to evaporate water for humidification, the heating energy provided by an electric furnace, the latent cooling load, and the electrical energy required to operate the air conditioner. The energy costs are total annual costs for heating and cooling using electricity at the rates shown.

There are two base cases for each house:

- mechanical ventilation with no heat recovery; ventilation in accordance with ASHRAE standards.
- mechanical ventilation with no heat recovery; ventilation in accordance with R2000 standards.

The results of the simulations of systems with the various heat exchangers were then compared with the base cases. For the Lake Charles location, the sensible heat exchanger showed no significant savings over the base case in the cooling season; however, it did provide a savings in the heating season. The magnitude of the savings is likely to be too small to justify the use of a sensible heat exchanger. One method to improve the operation of a sensible exchanger in this climate would be to provide a water spray in the exhaust airstream entering the heat exchanger core. This spray would cool the airstream by evaporation. This option was not considered in the modeling. The performance of the rotary non-desiccant heat exchanger was very similar to that of the sensible heat exchanger, with most of the savings in the heating season. The porous plate exchanger performed better, with some reduction in the cooling requirements. In most cases, savings of about 17% in energy costs could be expected with the porous plate exchanger as compared to the straight ventilation case.

For the Lake Charles climate, the greatest savings are achieved with the desiccant wheel heat exchanger. In most cases, the savings would be in the range of 20% to 23% of total heating and cooling costs. Use of the heat and moisture exchangers also lowers the relative humidities in the houses, resulting in a potential increase in comfort levels.

The results of the simulations for Lake Charles indicate that heat and moisture exchangers are the only type of exchanger that should be considered for the warm, humid climate found in the southeastern United States near the Gulf of Mexico.

Results for Seattle, WA

Seattle was chosen as representative of a cool, relatively moist climate. The intent was to determine whether moisture recovery was necessary or desirable for this type of climate. The simulations carried out for Seattle were the same as those for Lake Charles. A summary of the results is given in Table 2.

The porous plate exchanger and the rotary non-desiccant exchanger performed slightly better than the sensible plate exchanger in terms of energy savings. The desiccant wheel exchanger provided slightly less savings than the sensible plate exchanger in most cases. The porous plate exchanger and the desiccant wheel exchanger did, however, increase the humidity levels in the houses to the top of the desirable range.

The results of the simulations showed that there is no real advantage in using exchangers with high moisture recovery efficiency in a climate such as that of Seattle unless a high humidity level is desirable. In small houses with relatively high moisture loads, the moisture recovery exchanger would produce humidity levels above the desirable range. In larger houses with low moisture loads or high ventilation rates, the heat recovery will help to maintain humidity levels at the upper end of the desirable range. Rotary, non-desiccant exchangers appear to be suitable for all applications in Seattle.

Results for Winnipeg, MB

Winnipeg was chosen as representative of a cold climate with dry winters. It is situated close to the center of the North American continent and has weather that is similar to that over much of the plains of the northern U.S. and western Canada. The simulations carried out for Winnipeg were similar to those for Seattle and Lake Charles. The results of these runs are contained in Table 3.

Runs were initially carried out with a winter humidifier setpoint of 40% RH. This setpoint resulted in condensation on the windows and window sills during periods of weather when the temperature was −30°C or below. The setpoint was subsequently changed to 35%. With this setting, there was some condensation on windows during extreme cold weather; however, the model showed no condensation collecting on the window sills for the base ventilation cases.

The rotary non-desiccant heat exchanger was able to recover a small amount of moisture and produced savings slightly greater than those for the sensible plate type exchanger. The porous plate exchanger substantially reduced the humidification load requirements without

TABLE 3
Results of Simulations for Winnipeg, MB

	HRV Description	Ventilation Rate L/s	Heating: Latent Load GJ	Heating: Net Energy GJ	Cooling: Latent Load GJ	Cooling: Net Energy GJ	Annual Cost @ $0.045/kWh $/yr	Annual Cost @ $0.10/kWh $/yr
House 1	none	20	2.80	63.8	1.3	3.0	835	1,856
	plate	20	2.80	56.2	1.3	3.2	743	1,650
	rotary, non-desiccant	20	1.70	55.6	1.3	3.2	735	1,633
	porous plate	20	0.50	52.4	1.4	3.2	695	1,544
	rotary, desiccant	20	0.00	51.9	1.6	3.3	690	1,533
	none	40	7.40	84.3	1.2	2.9	1,090	2,422
	plate	40	7.50	68.6	1.3	3.1	896	1,992
	rotary, non-desiccant	40	5.30	67.7	1.3	3.1	885	1,967
	porous plate	40	2.50	60.6	1.3	3.1	796	1,769
	rotary, desiccant	40	0.50	58.0	1.4	3.1	764	1,697
House 2	none	30	3.20	77.3	1.5	2.7	1,000	2,222
	plate	30	3.20	65.7	1.6	2.9	858	1,906
	rotary, non-desiccant	30	2.00	65.6	1.6	2.9	856	1,903
	porous plate	30	0.05	61.4	1.8	3.0	805	1,789
	rotary, desiccant	30	0.00	61.1	2.2	3.1	803	1,783
	none	50	7.50	97.3	1.4	2.6	1,249	2,775
	plate	50	7.50	77.7	1.4	2.8	1,006	2,236
	rotary, non-desiccant	50	5.10	76.9	1.4	2.8	996	2,214
	porous plate	50	2.20	69.0	1.5	2.9	899	1,997
	rotary, desiccant	50	0.00	66.2	1.9	2.9	864	1,919
House 3	none	55	7.30	114.4	1.4	2.4	1,460	3,244
	plate	55	7.10	91.8	1.4	2.5	1,179	2,619
	rotary, non-desiccant	55	4.70	91.6	1.4	2.5	1,176	2,614
	porous plate	55	1.60	83.3	1.6	2.6	1,074	2,386
	rotary, desiccant	55	0.00	82.0	1.9	2.7	1,059	2,353
	none	70	10.50	129.7	1.4	2.4	1,651	3,669
	plate	70	10.20	100.9	1.4	2.5	1,293	2,872
	rotary, non-desiccant	70	6.90	100.0	1.4	2.5	1,281	2,847
	porous plate	70	2.90	90.6	1.5	2.5	1,164	2,586
	rotary, desiccant	70	0.00	85.9	1.7	2.6	1,106	2,458

TABLE 4
Ventilation Rate Produced by Humidity Control

House 1, Winnipeg: Ventilation Rate Controlled By Humidity Level

Month	Humidity Set Point	Moisture Generation Rate (kg/day) 5		10.1		15.1		20.2	
		Ventilation Rate							
	% RH	ACH	L/s	ACH	L/s	ACH	L/s	ACH	L/s
January	35	0.21	12.68	0.38	22.95	0.56	33.82	0.73	44.09
February	35	0.21	12.68	0.38	22.95	0.56	33.82	0.73	44.09
March	40	0.23	13.89	0.44	26.58	0.64	38.66	0.83	50.13
April	40	0.42	25.37	0.67	40.47	0.90	54.36	1.08	65.23
May	40	0.54	32.62	0.80	48.32	1.00	60.40	1.17	70.67
June	50	0.67	40.47	1.01	61.00	1.17	70.67	1.27	76.71
July	50	1.28	77.31	1.42	85.77	1.48	89.39	1.50	90.60
August	50	0.49	29.60	0.98	59.19	1.16	70.06	1.26	76.10
September	50	0.53	32.01	0.79	47.72	0.96	57.98	1.09	65.84
October	50	0.50	30.20	0.74	44.70	0.93	56.17	1.08	65.23
November	35	0.33	19.93	0.63	38.05	0.89	53.76	1.11	67.04
December	35	0.23	13.89	0.42	25.37	0.62	37.45	0.80	48.32

House 2, Winnipeg: Ventilation Rate Controlled By Humidity Level

Month	Humidity Set Point	Moisture Generation Rate (kg/day) 4.8		9.5		14.3		19.1	
		Ventilation Rate							
	% RH	ACH	L/s	ACH	L/s	ACH	L/s	ACH	L/s
January	35	0.10	12.00	0.19	22.80	0.29	34.80	0.38	45.60
February	35	0.10	12.00	0.19	22.80	0.29	34.80	0.38	45.60
March	40	0.11	13.20	0.23	27.60	0.34	40.80	0.45	54.00
April	40	0.22	26.40	0.37	44.40	0.50	60.00	0.61	73.20
May	40	0.29	34.80	0.45	54.00	0.56	67.20	0.65	78.00
June	50	0.55	66.00	0.64	76.80	0.73	87.60	0.77	92.40
July	50	0.80	96.00	0.82	98.40	0.83	99.60	0.83	99.60
August	50	0.52	62.40	0.64	76.80	0.71	85.20	0.77	92.40
September	50	0.38	45.60	0.49	58.80	0.57	68.40	0.64	76.80
October	50	0.29	34.80	0.43	51.60	0.54	64.80	0.63	75.60
November	35	0.15	18.00	0.32	38.40	0.47	56.40	0.59	70.80
December	35	0.10	12.00	0.21	25.20	0.32	38.40	0.42	50.40

producing significant condensation on the window sills. The desiccant wheel exchanger was able to maintain the humidity levels in the houses without any humidifier input for all cases except house 1 with a 40 L/s ventilation rate. Significant window sill condensation did occur with the desiccant wheel heat exchanger for the lowest ventilation rate with house 2 and for both ventilation rates with house 3. Additional simulations were carried out with an infiltration rate of 0.1 to determine if this would eliminate the condensation on the window sills. While the condensation was reduced, it still occurred at the lowest ventilation rates.

The savings resulting from use of the desiccant wheel exchanger range from 3300 to 12,000 kWh per year for ventilation rates from 20 to 70 L/s as compared to the case of ventilation with no heat recovery. At an energy cost of $0.05 per kWh, the annual savings from the desiccant wheel exchanger would be from $165 to $600. The average annual saving per L/s of ventilation is 167 kWh per year. At $0.05 per kWh, this would be $8.35 per year. In comparison, the average annual saving per L/s for the sensible plate exchanger is 110 kWh per year, or $5.50 at $0.05/kWh.

The above analyses were done based on what was considered to be likely moisture generation rates for each house modeled. In order to study the effect of moisture generation rates higher or lower than those considered typical, additional runs were made. It was found that in Winnipeg, about 2.5 L/s of ventilation with outdoor air is required for each kilogram per day of moisture generated within the house to avoid condensation buildup. Thus, with 5 kg of moisture produced per day, 12.5 L/s would provide acceptable humidity control, whereas for 20 kg per day of moisture, the required ventilation rate would be 50 L/s. Ventilation or infiltration in excess of these rates during cold weather will reduce humidity levels below the desirable range or result in a humidification load. Table 4 shows the ventilation rates that would result for houses 1 and 2 in Winnipeg if the ventilation system were controlled to maintain the humidities shown. It can be seen that for low moisture-generation rates, ventilation rates to control humidity will be below the rates recommended for acceptable air quality by ASHRAE or R2000.

DEVELOPMENT OF HEAT EXCHANGER SELECTION GUIDELINES

Based on the work carried out above, tentative guidelines for the selection of residential heat and moisture exchangers can be developed. It should be noted, however, that this study was confined to only three climates. Consequently, the guidelines should not be applied to hot, dry climates.

The first step is to identify the factors that impact upon the need for moisture recovery. The factors to consider include the following:

- the humidity ratio and temperature of the outdoor air
- the rate of moisture generation in the house, including moisture from basements
- the ventilation rate required for acceptable air quality
- the natural ventilation rate
- the desired indoor humidity range
- the maximum humidity level allowable to avoid significant condensation on interior surfaces of windows in cold weather
- whether the main energy use is for heating or cooling.

Moisture Recovery Ventilators in Residences Requiring Extensive Cooling in a Warm, Humid Climate

From the analyses done for Lake Charles, it is apparent that when a heat exchanger is being considered mainly for reduction of cooling energy costs in a humid climate, enthalpy heat exchangers are preferable to sensible-only exchangers and that the higher the moisture transfer effectiveness, the better.

This conclusion is not applicable to residences in hot, dry climates.

Heat Recovery Ventilators in Residences Requiring Heating

The guidelines for which type of exchanger to use in heating applications are more complex than those for the cooling application mentioned above (warm, humid climate). Four factors affecting the selection are: climatic conditions, internal moisture generation, ventilation rate, and desired indoor humidity conditions. Elaboration of these factors is given below.

Climatic Conditions. The important factor in determining whether or not heat exchangers are suitable is the outdoor humidity ratio during the heating season. If the ratio is relatively high during the heating season, there is likely to be no need for moisture recovery. Conversely, there would likely be a need for moisture recovery at locations with low outdoor humidity ratios during the heating season. This is confirmed by the runs for Seattle (high outdoor humidity ratios) and Winnipeg (low outdoor humidity ratios).

Information on heating season outdoor humidity ratios is not readily available. However, the average winter temperature can be used as a guide on the use of moisture recovery units. For example, since the average winter temperature in Winnipeg is about −8°C, the corresponding outdoor humidity ratio must be very low. Therefore, in cold climates, the average winter temperature is proposed as a guide to the use of heat and moisture exchangers. In climates where the winters are more moderate, the average winter temperature can be used as an approximate indicator for the suitability of heat exchangers.

Internal Moisture Generation Rates. The internal moisture generation rate is a crucial factor in the decision to use heat exchangers. Clearly, if high moisture generation rates occur in the home, the use of heat exchangers may result in extremely high indoor humidity.

Moisture is generated internally from occupant activity such as cooking, showering, and washing, as well as moisture entering the house from the ground through crawlspaces or basement floors and walls. The moisture generated from occupant activities is described in chapter 5 of the 1983 ASHRAE Equipment Handbook (ASHRAE 1983). The average rate of moisture production for a family of four is given as 7.6 kg per day. The moisture gains from basements and crawlspaces are not easily determined but are estimated to range from 1 or 2 kg per day up to 30 kg or more.

The moisture gains from the ground can be a significant portion of the total gain. Therefore it is necessary to have a reasonably good estimate of moisture flow that is entering the house from the ground. A lower rate of about 1 to 2 kg per day can be assumed for areas with dry ground or houses with vapor barriers under basement floors and on crawlspace floors. Higher rates should be used where ground water is a problem or where crawlspaces have no vapor barrier.

Ventilation Rate. The ventilation rate used in the house is another important factor affecting the selection of heat exchangers. The ventilation rate for the house can be estimated using ASHRAE Standard 62-81 (ASHRAE 1986) or other similar ventilation standards. This ventilation can then be broken down into that supplied by infiltration and exhaust fans and that to be supplied through the heat exchanger. Infiltration will reduce the effect of the heat exchanger. In cold, dry climates, infiltration will reduce humidity levels in the house and increase the need for a heat exchanger.

Desired Indoor Humidity Level. The most desirable range for relative humidity, as identified previously, is 40% to 60%. A maximum humidity level may be dictated by potential condensation problems on interior surfaces, especially windows. The winter design temperature can be used to identify the humidity levels at which condensation may become a problem (Wilson 1960).

SUMMARY

Once all this information is assembled, a method for determining which type of heat exchanger to select is required. Tables 5 and 6 show some calculated values of moisture generation rates and ventilation rates required to achieve 40% RH in a house at various outdoor humidity levels. Using these tables, along with the results of the simulation modeling for Winnipeg and Seattle, a selection guide such as that shown in Figure 3 can be generated. From this figure, if the ventilation rate and average outdoor

TABLE 5
Moisture Generation Rate Required to Maintain 40% RH @ 20°C

Outdoor Conditions: Temperature °C	Humidity Ratio at 80 %RH	Ventilation Rate (L/s) 10.00	20.00	30.00	40.00	50.00	60.00	70.00	80.00	90.00	100.00
		Moisture Generation Rate (kg/day)									
-35	0.0001	6.12	12.24	18.37	24.49	30.61	36.73	42.86	48.98	55.10	61.22
-22	0.0005	5.71	11.41	17.12	22.83	28.54	34.24	39.95	45.66	51.36	57.07
-13	0.0010	5.19	10.38	15.56	20.75	25.94	31.13	36.32	41.51	46.69	51.88
-8	0.0015	4.67	9.34	14.01	18.68	23.35	28.02	32.69	37.36	42.03	46.69
-5	0.0020	4.15	8.30	12.45	16.60	20.75	24.90	29.05	33.21	37.36	41.51
-2	0.0025	3.63	7.26	10.90	14.53	18.16	21.79	25.42	29.05	32.69	36.32
0	0.0030	3.11	6.23	9.34	12.45	15.56	18.68	21.79	24.90	28.02	31.13
2	0.0035	2.59	5.19	7.78	10.38	12.97	15.56	18.16	20.75	23.35	25.94
4	0.0040	2.08	4.15	6.23	8.30	10.38	12.45	14.53	16.60	18.68	20.75
6	0.0045	1.56	3.11	4.67	6.23	7.78	9.34	10.90	12.45	14.01	15.56
7	0.0050	1.04	2.08	3.11	4.15	5.19	6.23	7.26	8.30	9.34	10.38

TABLE 6
Ventilation Rate Required to Maintain 40% RH @ 20°C

Outdoor Conditions: Temperature °C	Humidity Ratio at 80 %RH	Moisture Generation Rate (kg/day) 5.00	10.00	15.00	20.00	25.00	30.00	35.00	40.00	45.00	50.00
		Ventilation Rate (L/s)									
-35	0.0001	8.17	16.33	24.50	32.67	40.83	49.00	57.17	65.34	73.50	81.67
-22	0.0005	8.76	17.52	26.28	35.04	43.80	52.57	61.33	70.09	78.85	87.61
-13	0.0010	9.64	19.27	28.91	38.55	48.19	57.82	67.46	77.10	86.73	96.37
-8	0.0015	10.71	21.42	32.12	42.83	53.54	64.25	74.95	85.66	96.37	107.08
-5	0.0020	12.05	24.09	36.14	48.19	60.23	72.28	84.32	96.37	108.42	120.46
-2	0.0025	13.77	27.53	41.30	55.07	68.84	82.60	96.37	110.14	123.90	137.67
0	0.0030	16.06	32.12	48.19	64.25	80.31	96.37	112.43	128.49	144.56	160.62
2	0.0035	19.27	38.55	57.82	77.10	96.37	115.64	134.92	154.19	173.47	192.74
4	0.0040	24.09	48.19	72.28	96.37	120.46	144.56	168.65	192.74	216.83	240.93
6	0.0045	32.12	64.25	96.37	128.49	160.62	192.74	224.86	256.99	289.11	321.23
7	0.0050	48.19	96.37	144.56	192.74	240.93	289.11	337.30	385.48	433.67	481.85

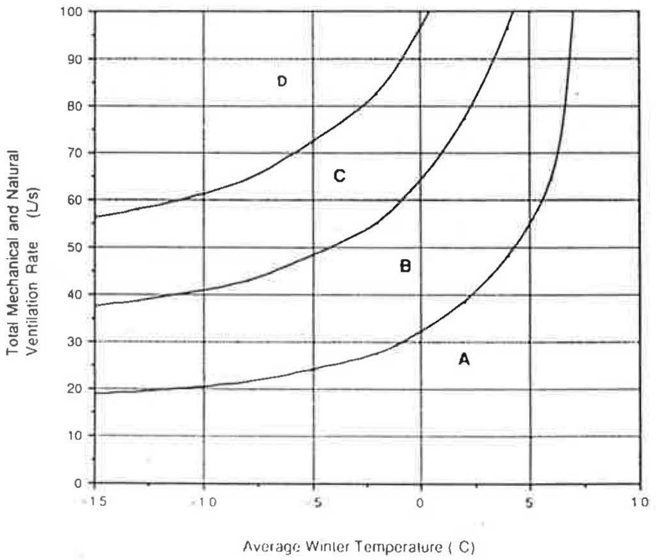

Figure 2 *Enthalpy exchanger selection chart*

winter temperature are known, a preliminary selection can be made using the criteria given below for the areas shown on the graph.

AREA A: Heat and moisture exchangers are not recommended in this area since moisture recovery is not required and may lead to excessively high humidity levels.

AREA B: Heat and moisture exchangers may be used in houses where low moisture generation rates are expected, on the order of 5 kg per day or less.

AREA C: Heat and moisture exchangers can be used in houses where moderate moisture generation rates occur (5 to 15 kg/day) or where half or more of the ventilation is supplied by means other than a heat exchanger or where an exchanger with medium moisture recovery effectiveness is used.

AREA D: Heat and moisture exchangers can be used in most homes unless unusually high moisture generation rates occur (e.g., the presence of a hot tub or indoor pool).

With this selection guide, it is relatively easy to determine where heat and moisture exchangers could be applied. The final step is to carry out an economic analysis to determine the financial benefits of using a heat exchanger.

Users should be cautioned that the above selection guide has been developed based on simulations for two locations only and has not been validated for a wide variety of climatic types.

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