Residential Ventilation with Heat Recovery

H 941

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The use of a regenerative, air-to-air, rotary, heat-exchanger as part of a controlled ventilation system in a modern tightly built house is discussed. Analytical studies and construction of a prototype were carried out. A four-month trial in a test-house indicated that the unit would be capable of recovering 5000 kW.h of waste energy if operation were extended over the full heating season.

The continuing increase in energy costs has prompted considerable effort to lessen requirements for use of energy to counter the effects of air infiltration. As a result some houses now being built are virtually airtight, but these still require a controlled inflow of fresh air to replace stale and humid air that must be exhausted.

In this ventilation process, the incoming air must be brought to room temperature and the associated energy consumption may largely offset the economic benefit of tight construction. However, this energy consumption can be avoided to some extent by use of a device that transfers heat between outgoing and incoming air streams.

Because it is compact and efficient, the regenerative, air-to-air, rotary heat exchanger, or thermal wheel, is well suited for energy reclamation in residential energy systems. The thermal wheel is a cylindrical rotor packed with an air-permeable medium with high thermal capacity that exposes a large surface area to air streams moving through it in opposite directions. As it rotates across the exhaust and supply ducts, the packing material absorbs heat from the warmer air stream and transfers it to the cooler air stream (see Figure 1). The rotary heat-exchanger, therefore, can serve year-round as an energy reclaiming device, warming incoming air in winter and cooling it in summer. Of course, its winter operation is of more interest in the Canadian climate.

The performance of regenerative air-to-air rotary heat exchangers is quite well established for large capacity ventilation installations. However, the small units required for residential ventilation systems are not available in North America. This article presents some theoretical considerations in the design of small rotary heat exchangers and describes the construction and performance of a light-weight unit intended for use with ventilation flow rates of 2 to 3 m^3/min. The performance of the heat exchanger in a four-month field trial is reported.

Theoretical Considerations

Regenerative air-to-air heat exchangers are rated by their effectiveness in recovering sensible and latent heat. The effectiveness is defined as the ratio of the actual heat-transfer rate to the thermodynamically limited maximum heat-transfer rate in a counter-flow heat exchanger of infinite surface area. The sensible heat effectiveness is defined, with respect to Figure 2, by:

$$\eta = \frac{\dot{m}_s (T_{so} - T_{si})}{\dot{m}_{min} (T_{ei} - T_{si})} \quad (1)$$

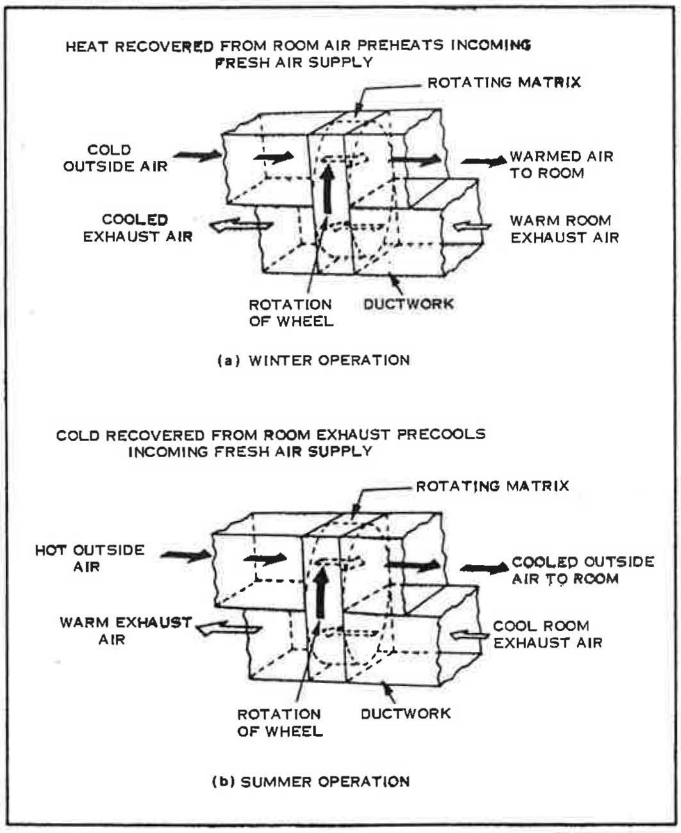

FIGURE 1—Heat recovery for residential ventilating systems using an air-to-air rotary heat exchanger.

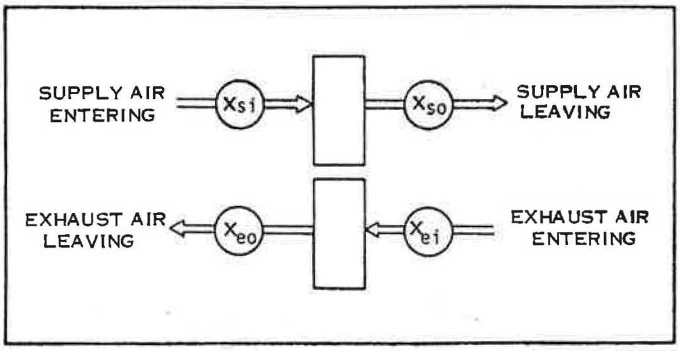

FIGURE 2—Schematic diagram of air-to-air heat exchanger.

Symbols and subscripts are defined in the nomenclature on page 8.

The performance of regenerative rotary heat exchangers can be predicted by solving a set of partial differential equations governing the heat transfer between the two air streams and the rotor surface. As described in Reference 2, the governing equations can be derived if no leakage or carryover is assumed. The formulation of the problem takes into account the effect of longitudinal heat-conduction in the rotor which has been neglected in earlier investigations.[3,4] A completely implicit finite-difference scheme is used to solve the problem.

Based on the solution of the governing equations, the effectiveness of the heat-exchanger can be presented in dimensionless parameters which can be used to determine the important independent variables for design optimization purposes. The commonly used dimensionless parameters[5] are:

(1) C_{min}/C_{max} = capacity-rate ratio of air streams

(2) C_r/C_{min} = capacity-rate ratio of rotor matrix to the minimum fluid

(3) $\text{NTU}_o = \dfrac{(hA)s}{C_{min}} \dfrac{1}{1 + (hA)^*}$

= overall number of transfer units

where $(hA)^* = (hA)_s/(hA)_e$

(4) $\lambda = \dfrac{kA_{sc}}{C_sL} \quad 1 = \dfrac{1}{A_c^*}$

= conduction parameter

where $A_c^* = A_{sc}/A_{ec}$

= conduction area ratio

The conduction parameter (λ) represents the relative significance of the longitudinal heat conduction through the rotor material as compared to the heat transferred to the air stream. This can be a significant parameter in cases where the cross sectional area subject to longitudinal heat conduction is large or when low flow rates are encountered, eg, ventilation rates in residences.

Typical results are shown in Figure 3. These indicate that some of the design parameters are acting in different directions so far as improving the effectiveness is concerned. As an example, increasing the rotor heat-capacity (Cr) by increasing its metal content in order to improve its effectiveness (Figure 3a) may lead to the increase of the conduction parameter which tends to act in the opposite direction, ie, results in a decrease in effectiveness (Figure 3b). For a specific heat exchanger, although an increase in the ventilation rate should cause a decrease in effectiveness due to the decrease of C_r/C_{min} and NTU_o, it would also lead to the decrease of λ which results in higher effectiveness. This suggests that for specific ventilation requirements an optimum design should exist.

Design and Construction of the Heat Exchanger

The computer program developed in Reference 2 was incorporated in a typical optimization procedure and used to design a rotary heat exchanger suitable for ventilation rates of 2 to 3 m^3/min. The rotor, 400 mm in diameter, 180 mm in length, and 2.2 kg in mass, consisted of 0.08-mm-thick aluminum foil in a structure of 1.6-mm hexagonal cells.

The rotor was mounted in an aluminum frame divided into two sections to accommodate the supply and exhaust ducts. The duct sectors of the wheel were sealed by flexible rubber

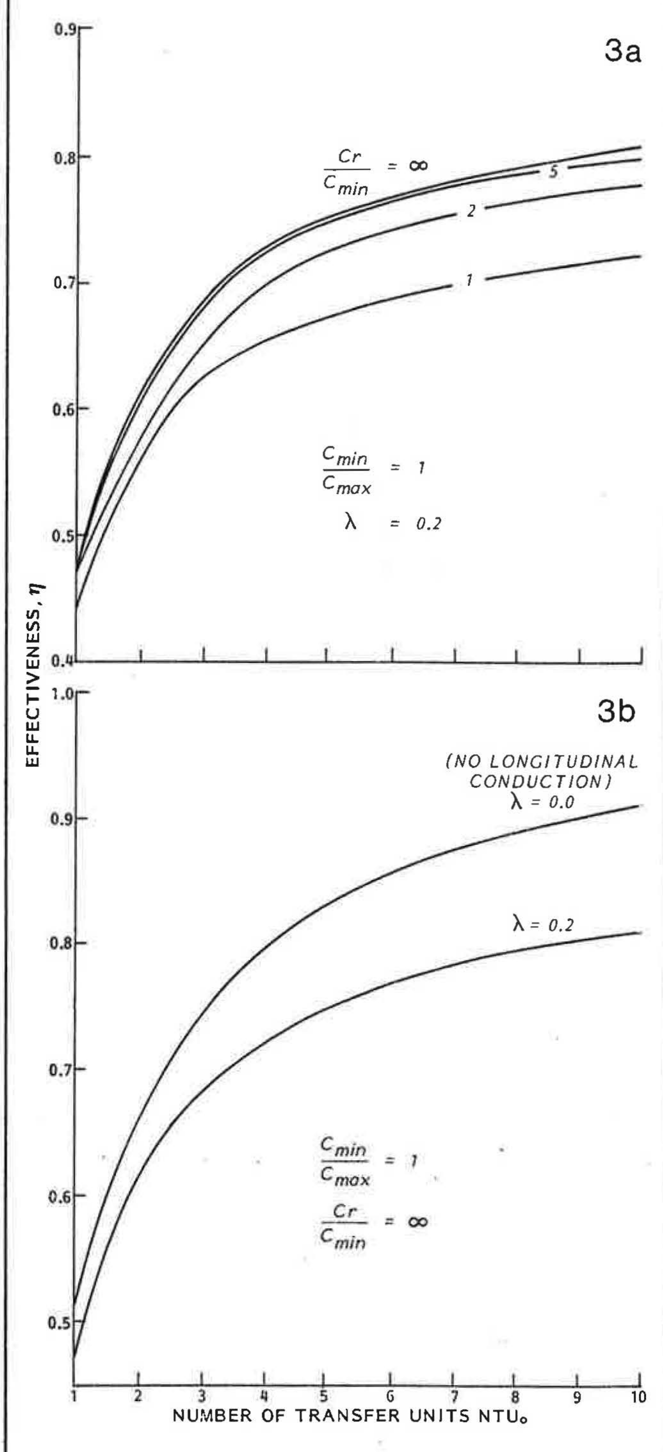

FIGURE 3—Predicted heat-exchanger performance.

tapes reinforced by metal strips. The circumferential seal used two overlapping seals, one fixed to the frame and the other fixed to the rotor. The wheel was driven by a 6-rpm, 40-watt gear motor mounted on the top of the frame via a chain-sprocket arrangement. The rotor speed could be changed in the range from 3 to 12 rpm by changing the gear ratio. The final shape of the rotary heat exchanger is shown in Figure 4. Further details about the theoretical analysis and design of the unit can be found in Reference 6.

Laboratory Testing of the Heat Exchanger

The performance of the rotary heat-exchanger was tested in the laboratory with the experimental setup shown in Figure 5. Air flow rates in the ducts were measured by the pitot tube traverse method. Temperature measurements were carried out at four locations (inlet and outlet of the heat exchanger in each duct). The pressure drop across the wheel (ΔP_w) and the pressure differential between the two ducts (ΔP) were measured as well.

Air leakage from one duct to the other through the seals, which is caused by the pressure differential between the two ducts (ΔP), and the air carry-over within the rotor voids were also measured using tracer gas technique.

The thermal performance results are shown in Figures 6 and 7. The heat exchanger effectiveness in terms of ventilation rate at 6-rpm rotational speed is shown in Figure 6. The average effectiveness was found to be about 85 per cent in the range of ventilation rates considered. The values of λ, NTU_o and C_r/C_{min} are superimposed on Figure 6. It is evident that the initial increase in heat-exchanger effectiveness with increase in air flow rate is due to the decrease of the conduction parameter, λ, in spite of the decrease of both NTU_o and C_r/C_{min}. However, with further increase in air flow rate, the influences of the parameters NTU_o and C_r/C_{min} prevail and effectiveness declines.

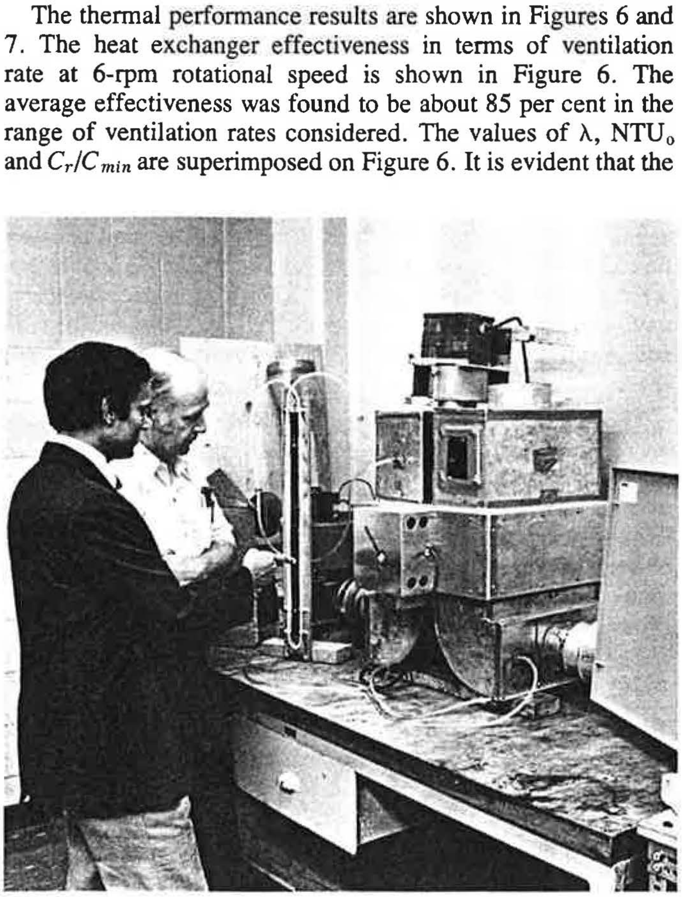

FIGURE 4—The heat exchanger.

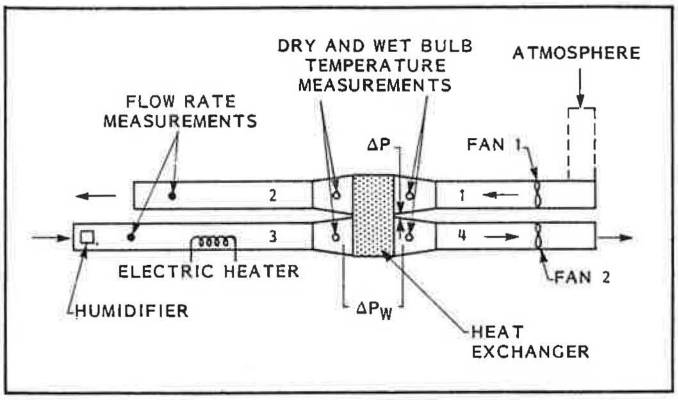

FIGURE 5—Experimental arrangement of rotary heat exchangers.

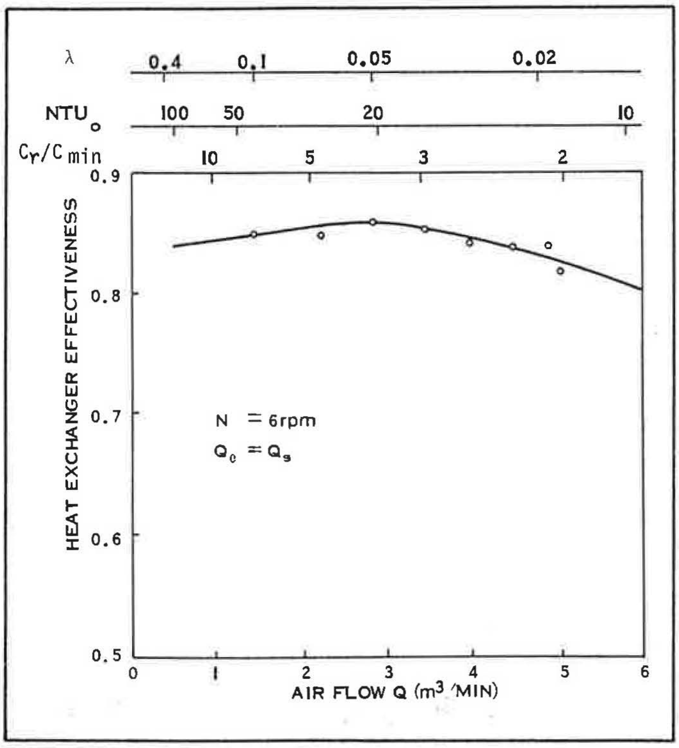

FIGURE 6—The effect of air flow.

The effect of rotational speed is shown in Figure 8. Effectiveness declined sharply at rotor speeds below 4 rpm and increased only marginally at speeds above 8 rpm. These results demonstrated the validity of our design procedure.

Installation of the thermal wheel in a ventilating system

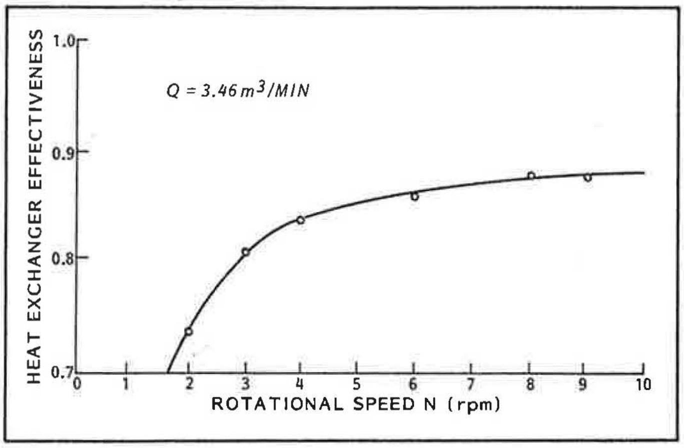

FIGURE 7—The effect of rotor speed.

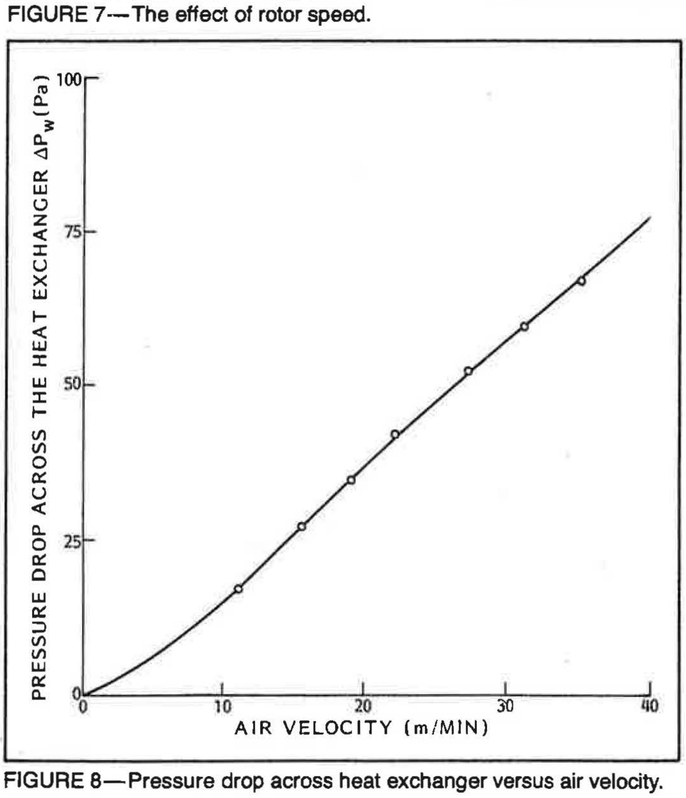

FIGURE 8—Pressure drop across heat exchanger versus air velocity.

increases the pressure loss that has to be overcome by the system fans. Figure 8 shows the observed pressure drop as a function of the air-face velocity, which is based on the actual net flow area through the matrix, ie, the mean velocity inside the flow passages. The pressure drop is small and the increase in power requirements is insignificant.

The leakage rate, W_l, is dependent on the performance of the seals used. This was measured as a function of the pressure difference between the supply and exhaust ducts which was varied by introducing restrictions on the flow in the two ducts

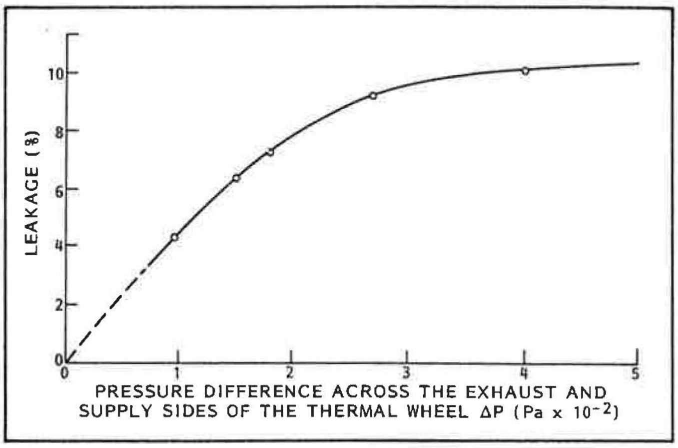

FIGURE 9—Leakage versus pressure difference between inlet supply and outlet exhaust ducts.

while keeping the air-flow rate constant in each duct. Typical results are shown in Figure 9. The observed leakage rates are acceptable, but some improvement could be achieved by modifying the sealing arrangement and carefully re-designing the ventilation system to lessen pressure differentials between the two ducts. The carry-over leakage was indicated by both estimate and measurement to be less than 0.35 per cent for a ventilation rate of 2.5 m^3/min and a rotational speed of 6 rpm.[6]

Frosting in Air-To-Air Heat Recovery Devices

Frosting is a potential problem in any air-to-air, heat-recovery device that is used where the winter ambient temperature falls below freezing. The rate at which frost will accumulate is a function of the temperature and humidity distribution within the thermal wheel. However, the changes in surface temperature of the rotary heat exchanger as it rotates between the cold and warm airstreams may make the frosting problem less severe than in other types of heat exchangers.

The theoretical analysis used to determine the conditions under which frosting occurs and the effects on performance of the heat exchanger are described in Reference 7. This analysis will be used to optimize design parameters for avoidance of frosting at predetermined outdoor temperatures. Manipulation of design parameters is thought to be a better approach than the more conventional procedures using preheaters and/or defrost cycles.

Field Demonstration

To determine the viability of the heat exchanger as a means for energy recovery in residential ventilation systems, the prototype was installed in one of the Mark XI experimental homes built by HUDAC in the Ottawa area. The installation, test procedure and results are given in Reference 7.

The ventilation system was designed to control the indoor humidity level and was operated at 2.3 m^3/min. Despite long

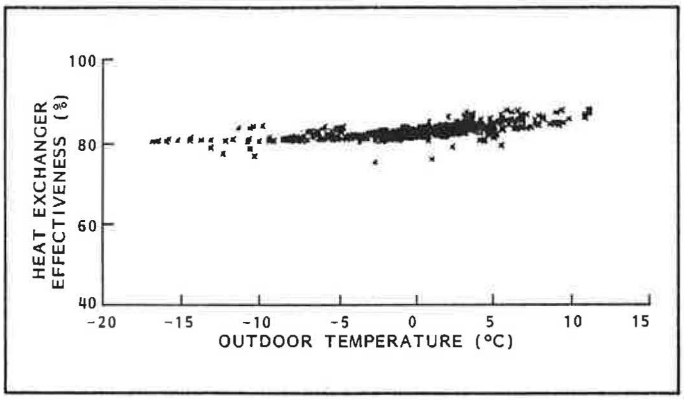

FIGURE 10—Effectiveness of the rotary heat exchanger in a four-month field test.

periods of cold weather with temperatures as low as $-17°C$, the effectiveness of the heat exchanger was found to be between 80 and 85 per cent (see Figure 10). Structural integrity of the heat exchanger components was good; the motor, chain-drive, matrix and static seals were found to be intact at the end of the four-month test period.

Monitoring of the unit in operation indicated that condensation and frost formation in the heat transfer passages of the matrix were not sufficient to have measurable effects on heat-exchanger performance. Based on the ventilation rate of 2.3 m^3/min and actual temperature measurements, 4500 kW.h were recovered during four months of operation. However, the ventilation rate of 2.3 m^3/min was found to be more than necessary to control the indoor humidity level. A rate of 1.7 m^3/min was predicted to be sufficient, and based on this rate the heat exchanger would have recovered 5000 kW.h over the six-month heating season.

References

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NOMENCLATURE

A	heat transfer area on the side designated by subscript
A_c	cross-sectional area of the rotor material on the side designated by subscript
c	specific heat
C	heat capacity rate of air stream = $\dot{m}c$
Cr	heat capacity rate of rotor = $m_r c_r N$
h	convective heat transfer coefficient
k	thermal conductivity of the rotor material
m	air mass flow rate
m_r	rotor mass
N	rotational speed
Nu	Nusselt number
Q	volumetric flow rate
T	temperature

Subscripts

i	inlet
o	outlet
s	supply
e	exhaust
r	rotor