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The Use of a Regenerative Air-to-Air Rotary Heat Exchanger for Heat Recovery in Residential Ventilation Systems

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With the increasing cost of energy supply for residential houses, the economics of reclaiming waste heat from residential ventilation systems have become very significant. This paper discusses the use of a regenerative air-to-air rotary heat exchanger for heat recovery in residential ventilation systems. Some of the potential operating problems in low temperature environment, such as frosting, are also discussed.

ABSTRACT

With the increasing cost of energy supply for residential houses, the economics of reclaiming waste heat from residential ventilation systems have become very significant. This paper discusses the use of a regenerative air-to-air rotary heat exchanger for heat recovery in residential ventilation systems. Based on theoretical considerations regarding the design of small rotary heat exchangers to handle low air flow rates, typical of residential houses, a unit was designed and constructed. The heat exchanger unit is constructed of a light-weight honeycomb structure made of aluminum foil sheets. Laboratory tests of the heat exchanger showed a high sensible heat recovery effectiveness in the order of 85% with acceptable levels of pressure drop and cross leakage. Cost saving analysis showed significant savings even at today's energy cost. Some of the potential operating problems in low temperature environment, such as frosting, are also discussed.

NOMENCLATURE

A heat transfer area on the side designated by subscript

A_c cross sectional area the rotor material on the side designated by subscript

c specific heat

C heat capacity rate of air stream = $\dot{m}c$

C_r heat capacity rate of rotor - $m_r c_r N$

h convenctive heat transfer coefficient

k thermal conductivity of the rotor material

$\dot{m}$ air mass flow rate

m_r rotor mass

N rotational speed

Nu Nusselt number

Q volumetric flow rate

Subscripts

i inlet

o outlet

s supply

e exhaust

r rotor

INTRODUCTION

With the increased cost of energy supply for residential homes, considerable efforts have been directed towards energy conservation as well as waste energy utilization. Conservation houses are intended to be air tight to cut down on the energy losses associated with air infiltration into the house. However, a well-sealed home requires adequate ventilation to supply fresh air to replace stale air exhausted to the outside.

In this ventilation process, air at the room temperature is exhausted to the outside and replaced by cold or warm air (depending on the season) which ought to be heated or cooled to the room temperature respectively. The energy losses associated with this

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process are considerable and tend to make well-sealed homes uneconomical. However, some of this energy loss can be reclaimed by allowing an exchange of heat between the exhausted and intake air through a heat exchanger.

The regenerative air-to-air rotary heat exchanger, which is sometimes referred to as a "thermal wheel" can be used effectively for energy reclamation in ventilating systems because of its compactness and high effectiveness. The thermal wheel consists of a cylindrical rotor packed with an air permeable media having a large surface area that is exposed to the air streams. The supply and exhaust air streams flow in a counter-flow arrangement. As the wheel rotates across the supply and exhaust ducts, it absorbs heat from the warmer air stream and transfers it to the cooler air stream. The rotation of the wheel provides a flow of energy from the hot to the cold air stream. The rotary air-to-air heat exchanger, as an energy reclaiming device in ventilating systems, has the capability of year round service as a cooler in the summer and a heater in the winter as demonstrated by Fig. 1. The

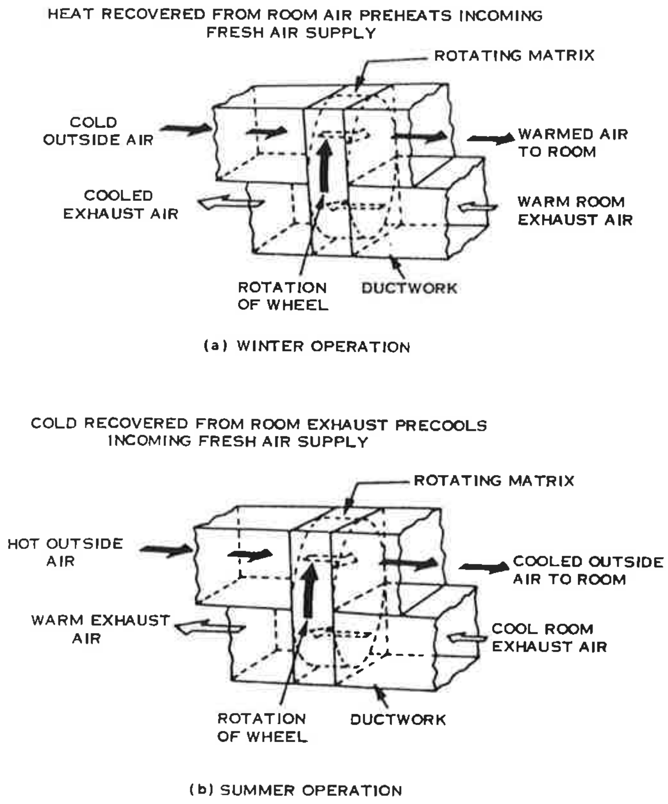

Fig. 1 Heat Recovery for Residential Ventilating Systems Using an Air-to-Air Rotary Heat Exchanger

wheel can also be utilized for latent heat (humidity) transfer between the two streams. If the rotor is made of a non-hygroscopic metallic material, it is considered a sensible heat recovery device only, although some transfer of humidity may take place should the temperature and humidity fields within the rotor be suitable for the condensation/evaporation process to take place. In applications where the transfer of latent heat is desirable, the rotor surface can be treated to make it hygroscopic and thus improve its capabilities as a latent heat transfer device as well as being a sensible heat transfer one. However, in a northern climate with long heating season, the wheel is mostly needed during the winter where the ventilation air is fairly dry. During this extended heating season, a well sealed house may experience a build-up of humidity inside the house which ought to be reduced by the mechanical ventilation system. For this application, a sensible heat recovery device is of more interest.

The performance of regenerative air-to-air rotary heat exchangers is, more or less, well established for large capacity ventilation installations. However, small units required to handle the ventilation air of residential houses are not available. This paper presents some theoretical considerations in designing small rotary heat exchangers as well as the construction and performance of a lightweight unit for sensible heat recovery designed to handle ventilation flow rates in the range of 2-3 m^3/min.

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 effectiveness is defined, with respect to Fig. 2, by

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

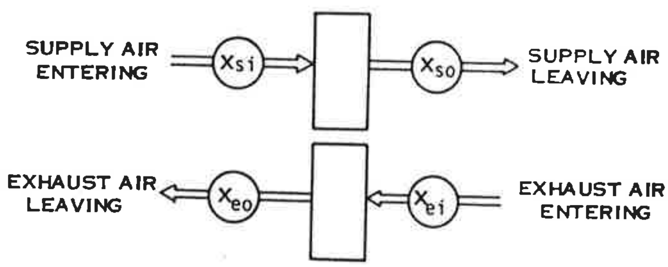

Fig. 2 Schematic Diagram of Air-to-Air Heat Exchanger

where η is the sensible, latent or total heat exchanger effectiveness and X is the dry bulb temperature, humidity ratio or enthalpy respectively. This definition of the effectiveness can accommodate sensible as well as total heat transfer devices and it should not be confused with the erroneous criteria of efficiency that sometimes appear in the literature.

Theoretical performance of regenerative rotary heat exchangers can be predicted by solving the set of partial differential equations governing the heat transfer between the two air streams and the rotor surface. Based on the solutions developed in[1,2], the effectiveness of sensible heat regenerative rotary heat exchangers can be presented in dimensionless parameters which can be used to determine the important independent variables so far as the design and performance of this type of equipment. The commonly used dimensionless parameters are[3,4].

(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) $NTU_o = \dfrac{(hA)s}{C_{min}} \left[\dfrac{1}{1+(hA)^*}\right]$ = overall number of transfer units

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

An additional dimensionless parameter was introduced in[5] to account for the effect of longitudinal heat condition through the rotor matrix material,

(4) $\lambda = \dfrac{kA_{sc}}{C_s L} \left[1 + \dfrac{1}{A_c^*}\right]$ = conduction parameter

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

The conduction parameter 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 residential homes.

A computer program was developed in[6], using the finite difference scheme presented in[5], to look at the effect of these different dimensionless parameters on the heat exchanger effectiveness. Typical results are presented in Fig. 3, which are fundamentally similar to those presented earlier in[5]. An increase in the heat exchanger effectiveness with increasing number of transfer units is evident. Fig. 3(a) shows that the effectiveness also increases with increasing the rotor heat capacity, however, no significant increase should be expected if the ratio C_r/C_{min} exceeds a value of about five. The effect of the conduction parameter λ and the capacity ratio C_{min}/C_{max} are shown in 3(b) and 3(c) respectively where the increase of λ or C_{min}/C_{max} results in lower effectiveness.

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

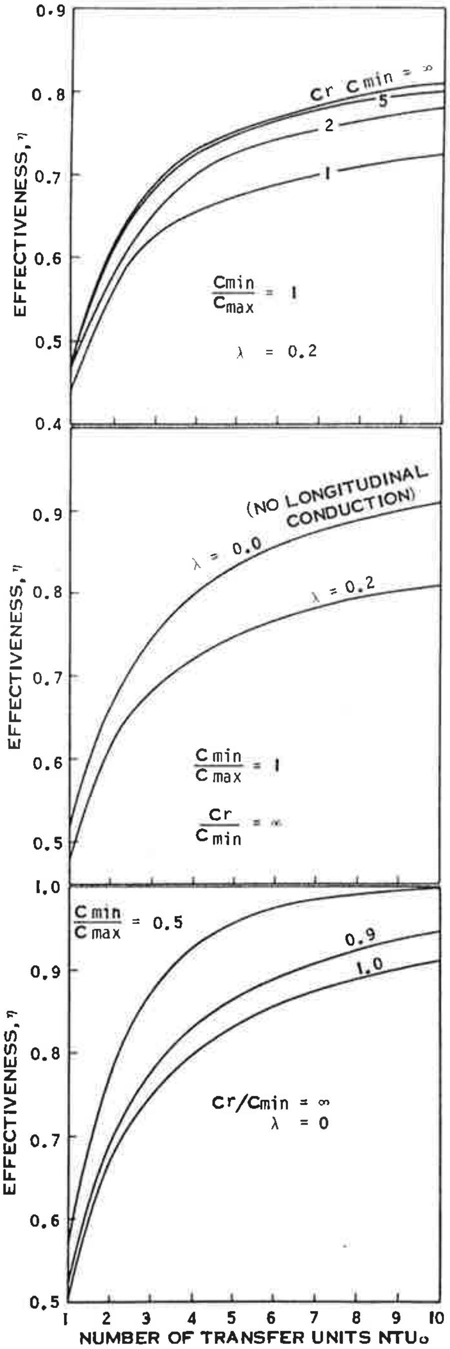

Fig. 3 The Predicted Heat Exchanger Performance

DESIGN AND CONSTRUCTION OF THE HEAT EXCHANGER

The computer program mentioned earlier was incorporated in a typical optimization procedure to design a rotary heat exchanger constructed of a light-weight honeycomb structure made of aluminum foil sheets and suitable for ventilation rates of 2-3 m^3/min. Some constraints were imposed regarding the maximum allowable pressure drop across the heat exchanger. Also, the honeycomb cell size, foil gauge, etc, were restricted to those commercially available. In designing the heat exchanger, the heat transfer coefficient was evaluated based on half the values predicted by the laminar forced convection in a tube for which Nu = 4.36[3]. This was done to account for the effect of the hexagonal flow passages as well as the rotational motion of the flow passages. The final specifications of the rotor were:

Rotor diameter	=	400 mm
Rotor length	=	180 mm
Cell size	=	1.6 mm (hexagonal)
Foil gauge	=	0.08 mm
Weight	=	2.2 kg

The honeycombe structure was supplied by HEXCEL Corporation and was formed into a cylindrical shape with a centre hole to accommodate the rotating shaft. The rotor was mounted in a reinforced aluminum frame divided into two sections to accommodate the supply and exhaust ducts and was allowed to rotate via two ball bearings. The duct sectors of the wheel were sealed using flexible rubber tapes reinforced by metal strips. The circumferential seal was improved by using 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 was changed in the range 3-12 rpm by changing the gear ratio. The final shape of the rotary heat exchanger is shown in Fig. 4.

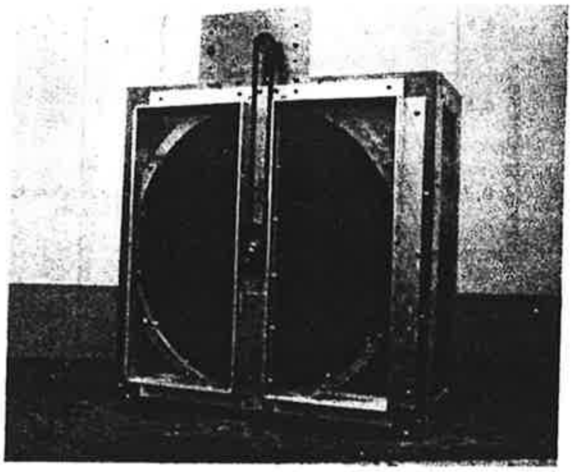

Fig. 4 The Heat Exchanger

EXPERIMENTAL ARRANGEMENTS

The performance of the rotary heat exchanger was tested using the experimental setup shown in Fig. 5. Four air ducts were connected to the self-contained thermal wheel unit. Tests were carried out during winter time where supply air was drawn from the outside atmosphere through duct 1 while room air or exhaust air was drawn through duct 3. The experimental arrangement provided the possibility of simulating summer operation as well. Air drawn from the room was heated and humidified at duct 3 to simulate the summer supply air while room air was drawn into duct 1 simulating the summer exhaust air.

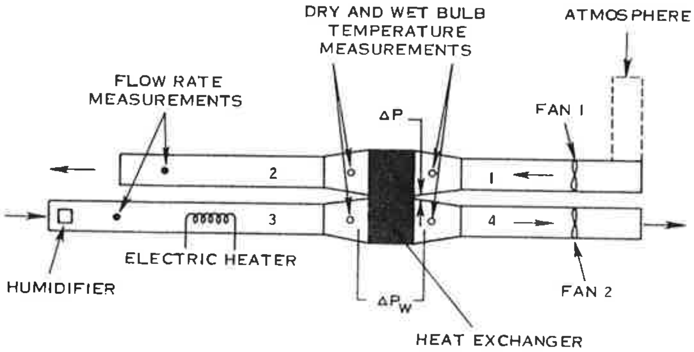

Fig. 5 Experimental Arrangement

Air flow rates in the ducts were measured by the pitot tube traverse method. The pressure differentials of the pitot tubes were read on two inclined water-manometers. Dry bulb and wet bulb temperature measurements at four locations (inlet and outlet of the heat exchanger in each duct) were carried out using Chromel-Alumel thermocouples and wet bulb thermometers respectively. The thermocouple outputs were read on a digital voltmeter.

The pressure drop across the wheel (ΔP_W) as well as the pressure differential between the two ducts (ΔP) were measured with inclined water-manometers which were connected to static pressure taps located in the duct walls.

Air leakage from one duct to the other through the seals, which is caused by the pressure differential between the two ducts (ΔP), as well as the air carryover within the rotor voids were also measured. The leakage paths can be best demonstrated by Fig. 6.

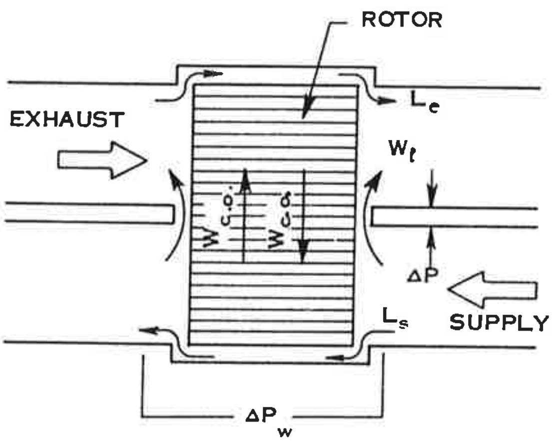

Fig. 6 Leakage and Carryover

The air leakage and carryover in the rotary heat exchanger were determined by means of the tracer gas technique (sulphur dioxide). With reference to Fig. 5, the supply duct (1-2) was kept at a higher static pressure than the exhaust duct (3-4) to ensure that air leakage due to the pressure differential was taking place only from the supply duct (1-2) to the exhaust duct (3-4), which means that the transfer of air in the opposite direction was due to carryover only. The rate of carryover was evaluated by injecting the tracer gas into the exhaust duct (3) at a known constant rate by means of a pressure regulator and a nozzle. Then the supply air in duct (2) was isokinetically sampled and analyzed via a Meloy SA-185 Sulphur Analyzer to determine the tracer gas concentration in duct (2) from which the carryover rate was calculated. Using the same technique, the total rate of both leakage and carryover was determined by injecting the tracers into the supply duct (1). The transfer of air from the supply duct to the exhaust duct was obviously due to both carryover and leakage. Since the carryover rate is equal in both directions, the leakage rate was determined accordingly as function of the pressure difference between the two ducts P.

EXPERIMENTAL RESULTS

Thermal Performance

The results are shown in Figs. 7 and 8. The heat exchanger effectiveness in terms of ventilation rate at 6 rpm rotational speed is shown in Fig. 7. The average effectiveness was found to be about 85% within the range of ventilation rates of interest.

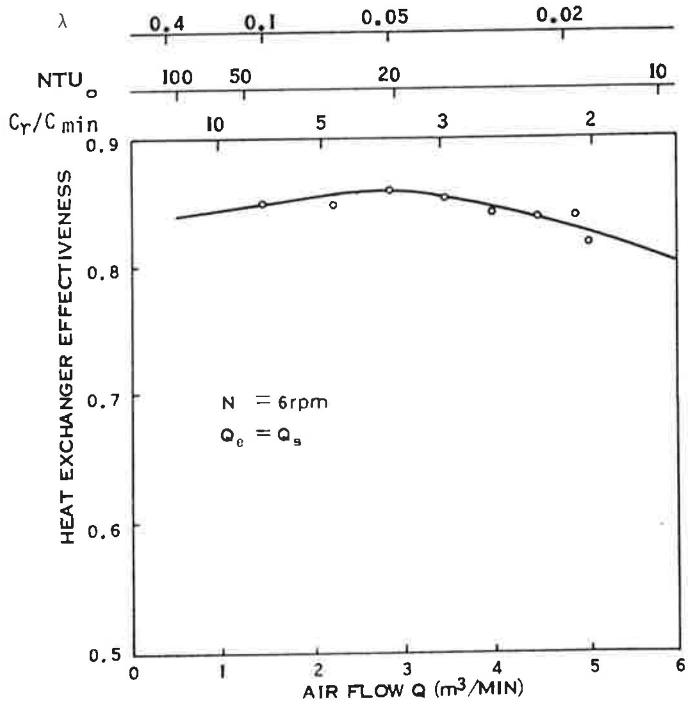

Fig. 7 Heat Exchanger Effectiveness vs Air Flow Rate

With these experimental data, it was possible to determine the actual heat transfer coefficients so that both the experimental and numerical results would agree with each other. The heat transfer coefficient was found to be 0.65 of that based on laminar flow in a circular tube, ie, for air flow in the hexagonal passages of the rotating matrix at 6 rpm, the heat transfer coefficient can be determined from Nu = 2.8, based on the equivalent diameter of the matrix passages. Using the calculated values of the heat transfer coefficient, the values of λ, NTU_o and C_r/C_{min} are superimposed on Fig. 7. It is evident that the early increase of the heat exchanger effectiveness with increasing the 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 effectiveness drops as the influence of the parameters NTU_o and C_r/C_{min} prevails. The effect of rotational speed is shown in Fig. 8. A sharp decrease in effectiveness was encountered by decreasing the rotor speed below 4 rpm while an insignificant gain in effectiveness was observed by increasing the speed beyond 8 rpm. These results demonstrated the validity of our design procedure.

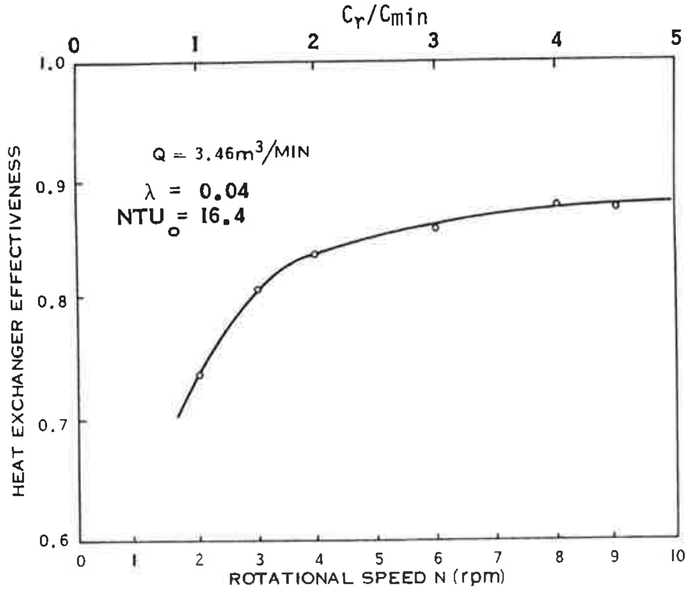

Fig. 8 The Effect of Rotor Speed

Static Pressure Drop Across the Rotor

Installing the thermal wheel in a ventilating sytem will result in an increased pressure loss in the system which has to be overcome by the system fans. The pressure drop across the wheel matrix is a function of the air velocity. Measurements of the pressure drop across the rotor were carried out at different air flow rates. Fig. 9 shows the observed pressure drop as a function of the air face velocity which was based on the actual net flow area through the matrix, ie, the mean velocity inside the flow passages.

Leakage and Carryover

Fig. 6 is a schematic of the leakage paths in the thermal wheel unit. The exhaust duct was kept at a lower static pressure than the supply duct which

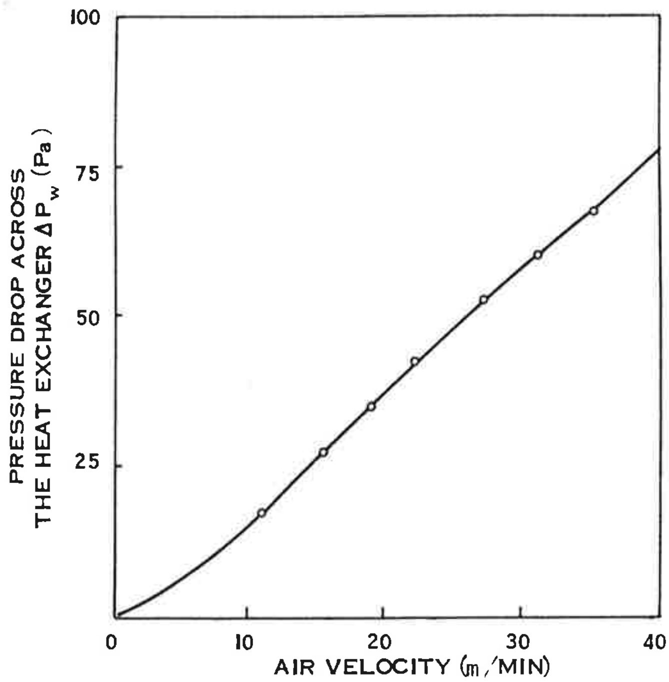

Fig. 9 Pressure Drop Across the Heat Exchanger vs Air Velocity

forced the air leakage through the rotor seals (W_ℓ) to be from the supply to the exhaust ducts. Both the carryover rate $W_{c.o.}$ and the leakage rate W_ℓ were measured using the tracer technique outlined earlier. The carryover rate is directly proportional to the rotor speed and the rotor void volume, and the percentage carryover may be evaluated by the equation.

$$W_{c.o.}/Q = \frac{\text{Rotor Void Volume x N}}{Q} \times 100 \qquad (2)$$

The observed carryover results were found to agree with this equation.

The leakage rate W_ℓ, which is dependent on the performance efficiency of the seals used, was measured as a function of the pressure difference between the supply and exhaust ducts. The pressure difference between the two ducts was changed by introducing restrictions on the flow in the two ducts keeping the air flow rate constant in each duct. Typical results are shown in Fig. 10. Although the leakage rates observed may be reasonable, an obvious room for improvement is available by improving the sealing arrangement and carefully designing the ventilation system so that no excessive pressure differentials between the two ducts are encountered.

It was difficult to evaluate experimentally the by-pass leakage, marked L_s and L_e in Fig. 6. However, the effect of this type of by-pass leakage is implicity accounted for in the measured effectiveness.

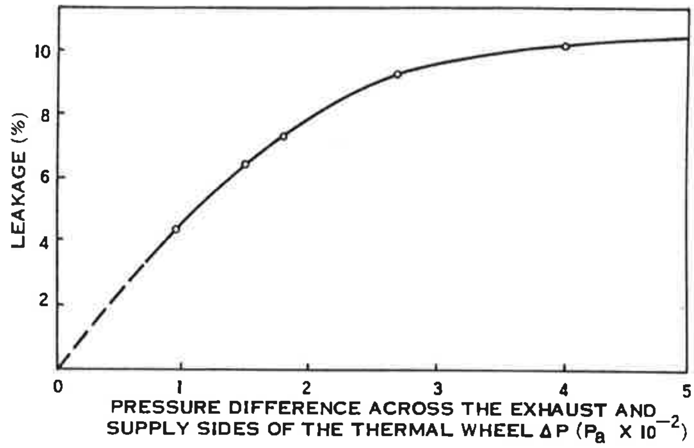

Fig. 10 The Per Cent Leakage vs the Pressure Difference Between the Inlet Supply and Outlet Exhaust Ducts

FROSTING IN AIR-TO-AIR HEAT RECOVERY DEVICES

Frosting in any type of air-to-air heat recovery device, including the rotary heat exchanger, is a potential problem that may take place when the device is used in locations where the winter ambient temperature drops below the ice point. As the matrix rotates from the supply (cold) duct into the exhaust duct the air passing through it is warmer than the matrix surface. If the metal surface is at a lower temperature than the dew point of the flowing air, condensation will take place on the matrix surface or if during this process the air becomes saturated. moisture will be deposited on the surface of the matrix. If the matrix surface temperature is below the ice point, frosting will occur. The rate at which the frost will accumulate is a function of the temperature and humidity distribution within the thermal wheel. However, it must be noticed that the cyclic changes in rotary heat exchanger surface temperature, as it rotates between the cold and warm streams, may make the frosting problem less severe than anticipated.

Theoretical analysis of the problem to establish quantitatively, with a reasonable degree of accuracy, the operating conditions under which frosting is expected, is very difficult and requires a large number of assumptions. However, field tests may be carried out to assess this problem and to investigate practical methods of predicting and eliminating frosting in rotary heat exchangers. Possible methods to avoid frosting when it is anticipated are preheating the intake air or operating the thermal wheel at lower effectiveness by reducing the speed or bypassing some of the intake air.

ANALYSIS OF THE POTENTIAL ENERGY SAVINGS

The potential energy savings due to the utilization of air-to-air heat recovery devices in ventilation systems depends on many factors. Those factors are: the local weather conditions, the ventilation rate, the effectiveness of the heat recovery device itself as well as the overall performance of the complete ventilation system.

The number of degree-hours available for recovery per year is calculated by,

$$\text{Recoverable degree-hour} = \Sigma \begin{bmatrix} (T_{room} - T_{ambient}) \times \\ \text{annual observed hours} \end{bmatrix} \quad (3)$$

For Toronto area, where the hourly ambient temperatures were obtained from[7] and based on 18°C room temperature, the recoverable degree-hour was found to be about 100 x 10^3 degree C-hour during the heating season while the recoverable degree-hour during summer is negligible.

Assuming a typical ventilation rate of 2.5 m^3/min[8], the total energy available for recovery during a typical heating season in Toronto area is about 5000 kWh. The use of the rotary heat exchanger presented earlier should make it possible to recover 85% of that (at 3¢/kWh this should total $127.50 per heating season), with negligible energy consumption in excess fan power and driving motor. However, it must be noted that the overall design and performance of the complete ventilation system may affect the final saving figure.

CONCLUDING REMARKS

Based on an extensive analysis of the performance of regenerative air-to-air rotary heat exchangers, a small unit, suitable for heat recovery in residential ventilation systems was designed and built. The unit is constructed of ligh-weight honeycomb structure made of thin aluminum foil sheets. The performance of the heat exchanger was tested under laboratory conditions and showed about 85% effectiveness and acceptable levels of pressure drop and leakage.

The rotary heat exchanger presented in this paper is currently operating in the ventilation system of a well-sealed experimental house operated by the National Research Council of Canada. The field performance of this unit will be reported later.

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