APPENDIX E

TESTING AND ANALYSIS OF A HEAT WHEEL HEAT EXCHANGER

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THE PERFORMANCE OF AIR-TO-AIR REGENERATIVE ROTARY HEAT EXCHANGERS FOR WASTE HEAT RECOVERY IN RESIDENTIAL VENTILATION SYSTEMS

by

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NOMENCLATURE

A	-	heat transfer area on side designated by subscript, ft^2 (m^2)
A_c	-	area available for longitudinal conduction on side designated by subscript, ft^2 (m^2)
c	-	specific heat, $BTU/lb_m{}^\circ F$ ($kJ/kg^\circ C$)
C	-	heat capacity rate of fluid = $\dot{m}c_p$, BTU/hr (W)
C_r	-	heat capacity rate of rotor = 60 $m_r c_r N$, BTU/hr (W)
h	-	heat transfer coefficient, BTU/hr $ft^{2\circ}F$ ($W/m^{2\circ}C$)
k	-	thermal conductivity of the rotor material, BTU/hr ft °F (W/m°C)
$\dot{m}$	-	mass flow rate of fluid, lb_m/hr (kg/h)
m_r	-	rotor mass, lb_m (kg)
N	-	rotor speed, rpm
NTU_o	-	overall number of transfer units
Q	-	Ventilation air volumetric flow rate, ft^3/min (m^3/s)
T	-	air temperature, °F (°C)
W_ℓ	-	leakage rate, (ft^3/min) (m^3/sec)
$W_{c.o.}$	-	carryover rate, ft^3/min (m^3/sec)

Subscripts

i	-	inlet
o	-	outlet
e	-	exhaust
s	-	supply

1. INTRODUCTION

With the increased cost of energy supply, considerable efforts have been directed towards energy conservation. Conservation houses are intended to be nearly air-tight to cut down on different sources of energy losses. This development will increase the need for forced mechanical ventilation systems to be provided in future homes. In such ventilation processes, air at room temperature is exhausted to the outside and replaced by cold or hot air (depending on the season) which ought to be heated or cooled to the room temperature respectively. The energy losses associated with these processes are considerable and tend to make air-tight homes less economical to operate.

However, some of this loss, hopefully most of it, can be reclaimed by providing an efficient heat exchange between the exhausted and intake air. The air-to-air regenerative rotany heat exchanger, which is sometimes referred to as "thermal wheel" can be used effectively for this purpose. It has two main advantages over other types of heat exchangers, namely, compactness and high effectiveness.

As shown in Figure 1, 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 and transfers it to the cool one. The rotation of the wheel provides a flow of energy from the hot to the cold air streams. Although the design and performance of regenerative rotary heat exchangers is, more or less, well established for large industrial applications, small units to handle residential ventilation rates are lacking.

The development of a small air-to-air rotary heat exchanger for residential applications was undertaken by the Mechanical Research Department of Ontario Hydro. Our contribution is limited to the demonstration of the principle and evaluation of its potential use for residential applications. So far, the study included the construction and testing of a prototype heat exchanger as well as the formulation of a numerical model to predict of the performance of regenerative rotary heat exchangers which proved to be very useful in optimizing future designs of this type of heat exchangers.

2. THE PROTOTYPE HEAT EXCHANGER

2.1 Construction

The rotor was constructed of 26 gauge utility grade aluminum sheet (0.4 mm thick). The aluminum sheet was corrugated by passing it between two crimping rolls. The head of a horizontal milling machine was used to turn the rolls as shown in Figure 2. As the required length of corrugated sheet was produced, the lower crimping roll was replaced by a smooth one. Two layers of the sheet metal, a corrugated one and a flat one, were passed through and tightly coiled on a 16 mm steel shaft to form the rotor. The rotor was mounted in a reinforced aluminum frame which was divided into two separate sections as shown in Figure 3. A chain and sprocket arrangement was used to rotate the wheel in the range 3 - 12 rpm by changing the geer ratio. Reinforced rubber seals were used to reduce the leakage from one stream to the other. The final dimensions of the rotor were 400 mm in diameter and 300 mm in length.

2.2 Performance Tests

The performance tests of the prototype were carried out in the winter season using the experimental set up shown in Figure 4. Outside air was drawn through duct 1 and room air was drawn through duct 3. This arrangement closely simulated winter operation in a residential installation. The simulated home conditions ie, duct 3, were controlled using a heater and humidifier. Temperature and flow measurements were carried out in the locations specified in Figure 4.

The experimental data are presented in terms of five dimensionless groups usually used to describe the performance of regenerative heat exchangers which are listed in Table I.

Figure 5 demonstrates the increase of sensible heat recovery rate with increasing ventilation rate and the initial temperature difference between the supply and exhaust ducts. By converting the heat recovery data into effectiveness data as shown in Figure 6, the effectiveness is shown to decrease slightly with increasing ventilation rate, within the tested range, and to be independent of the initial temperature difference. The average effectiveness was in the order of 73%.

The effect of unequal mass flow rates in the supply and exhaust ducts is also demonstrated in Figure 7 where decreasing the parameter Cmin/Cmax resulted in increasing effectiveness. So far as the effect of rotational speed is concerned, it was expected that an increase in the rotor speed would lead to higher effectiveness. However, when tests were carried out in the range of 3 - 12 rpm, no significant difference in effectiveness was observed apparently because the rotor heat capacity was too high due to its large mass. A quantitative explanation of this point will be shown later.

One of the important considerations in designing and operating rotaty heat exchangers, is the air leakage and carry over from one duct to the other. Figure 8 shows a schematic of the leakage paths in the thermal wheel. Leakage at the rotor face across the seals is a function of the pressure difference between the two ducts and can be reduced by minimizing this pressure difference and by the use of proper seal. However, in situations where leakage from the exhaust duct to the supply duct cannot be tolerated, the supply duct should intentionally be kept at slightly higher pressure than the exhaut one. Carry over is the air which is entrapped in the rotor passages as it rotates from one duct to the other. Carry over rate is easily predicted as the product of the void volume times the rotational speed. Tracer gas analysis technique was used to assess both leakage and carry over in the prototype. The results are shown in Figures 9 and 10. Details of the construction and performance data of the prototype heat exchanger were included in reference/1/.

Although the exercise of constructing and testing the prototype rotary heat exchanger showed good potential. Two problems were noticeable:

1. The 73% effectiveness observed was less than the expected effectiveness based on the design procedures specified in different heat exchanger handbooks/2,3/.

2. The total weight of the rotor was about 30 kg which may be undesirable for the use in residential applications.

3. THE NUMERICAL MODEL

In order to better understand the interaction between the different design parameters so that our design can be optimized, a numerical model was formulated in which the governing differential equations for the heat transfer between the rotor matrix and the air streams were solved using a finite difference scheme similar to that presented in/4/. However, the effect of heat conduction in the rotor material in the longitudinal direction, parallel to the flow direction, was included in the formulation which was neglected in/4/. Details of the numerical model were presented in/5/.

The results demonstrated that the longitudinal heat conduction is an important parameter and that a conduction parameter defined as:

$$\lambda = \frac{kAsc}{C_s} \left[1 + \frac{Aec}{Asc} \right]$$

should be included into the list presented in Table I. Some of the results of the numerical model as related to the prototype performance are shown in Figures 11 and 12.

The effect of longitudinal heat conduction is shown in Figure 11 which shows the predicted effectiveness when the longitudinal conduction is neglected (λ = o) as compared to the case when λ = 0.2. By using the value of 0.2 for the conduction parameter, which corresponds to the prototype condition for flow rate of 0.04 m^3/s (85 cfm), the effectiveness is much lower than that obtained by neglecting the conduction effects. This explains why the prototype showed a lower effectiveness than the designed value.

The effect of the rotor heat capacity is shown in Figure 12. It is clear that increasing the rotor capacity Cr results in higher effectiveness up to a value of 5 for the parameter Cr/Cmin beyond which no significant improvement is achievable. Since Cr is proportional to the product of the rotor mass and its rotational speed, and since the prototype was operated beyond this critical ratio, no significant effect of changing the rotor speed was observed.

4. CLOSURE

The results of this work were used to design a new rotary heat exchanger using a honeycomb structure made of foil papers having a thickness of 1/10 of the metal sheet used for the prototype. This will result in a much lighter heat exchanger. The reduction in the area available for longitudinal conduction will result in higher effectiveness while the reduction of the rotor capacity should not affect the effectiveness significantly as long as the ratio Cr/Cmin is properly chosen as shown earlier.

The numerical model was used to optimize the new heat exchanger design for ventilation rates of 0.02 - 0.045 m^3/sec (50-100 scfm). The final specifications were:

Rotor diameter = 40 cm
Rotor length = 18 cm
Total weight = 5 kg
Average effectiveness = 85%

Performance tests are currently carried out for this heat exchanger. It will be installed in a HUDAC conservation house No 2 in Ottawa in the near future so that field experience could be also established.

ACKNOWLEDGEMENTS

To Dr. T. Ellis for his involvement in the early stages of this work.

REFERENCES

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TABLE I

$$\eta = F\left(C_{min}/C_{max},\ C_r/C_{min},\ NTU_o\right)$$

$$\text{EFFECTIVENESS} = \eta = \frac{(\dot{m}C_p)_s(\Delta T)\ \text{ACROSS THE WHEEL IN THE SUPPLY DUCT}}{(\dot{m}C_p)_{min}\ (\Delta T)\ \text{SUPPLY AND EXHAUST INLETS IN THE WHEEL}}$$

$$\frac{C_{min}}{C_{max}} = \frac{(\dot{m}C_p)_{min}}{(\dot{m}C_p)_{max}} = \text{CAPACITY RATE RATIO OF THE TWO STREAMS}$$

$$\frac{C_r}{C_{min}} = \frac{m_r C_{pr} N}{(\dot{m}C_p)_{min}} = \text{CAPACITY RATE RATIO OF THE ROTOR MATRIX TO THE MINIMUM FLUID}$$

$$NTU_o = \frac{(hA)_s}{C_{min}}\left[\frac{1}{1 + (hA)^*}\right] = \text{OVERALL NUMBER OF TRANSFER UNITS}$$

$$(hA)^* = \frac{(hA)_s}{(hA)_e} = \text{CONDUCTANCE RATES}$$

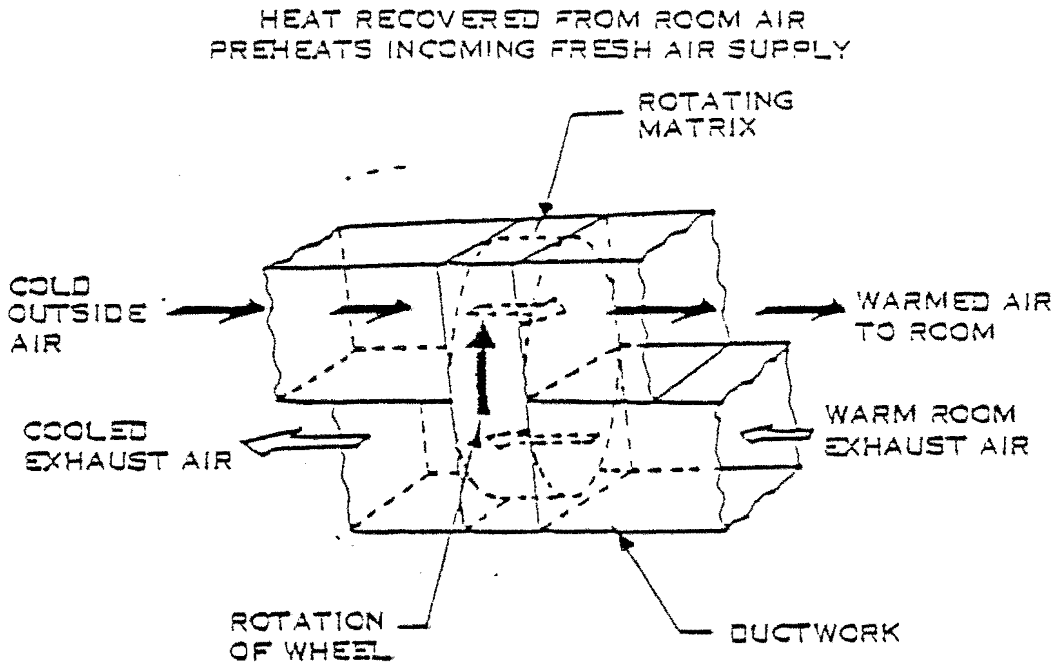

FIGURE I

OPERATION OF THE THERMAL WHEEL

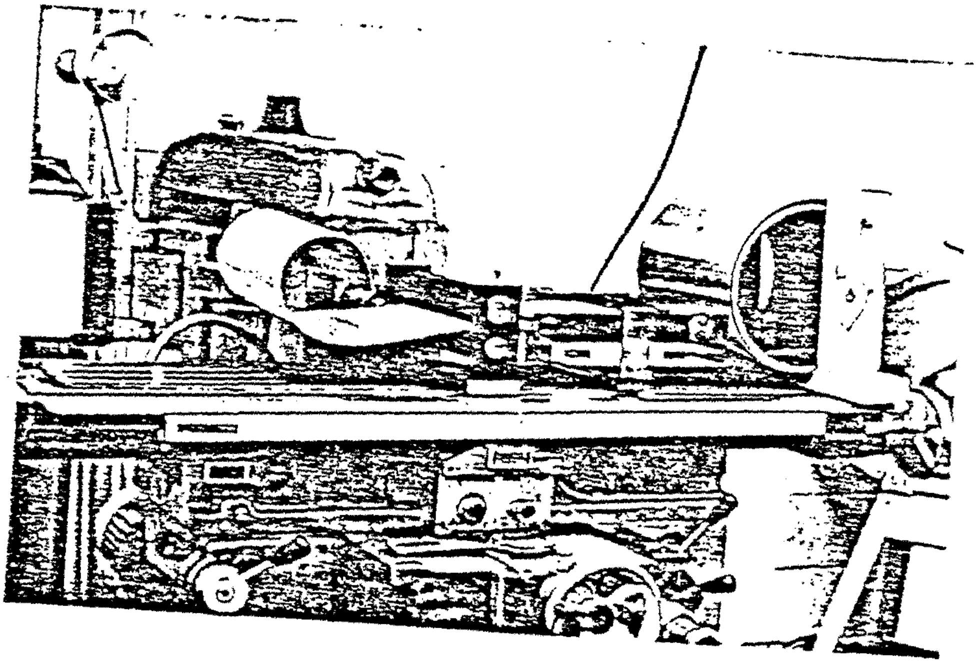

FIGURE 2
PRODUCING THE CORRUGATED SHEET METAL

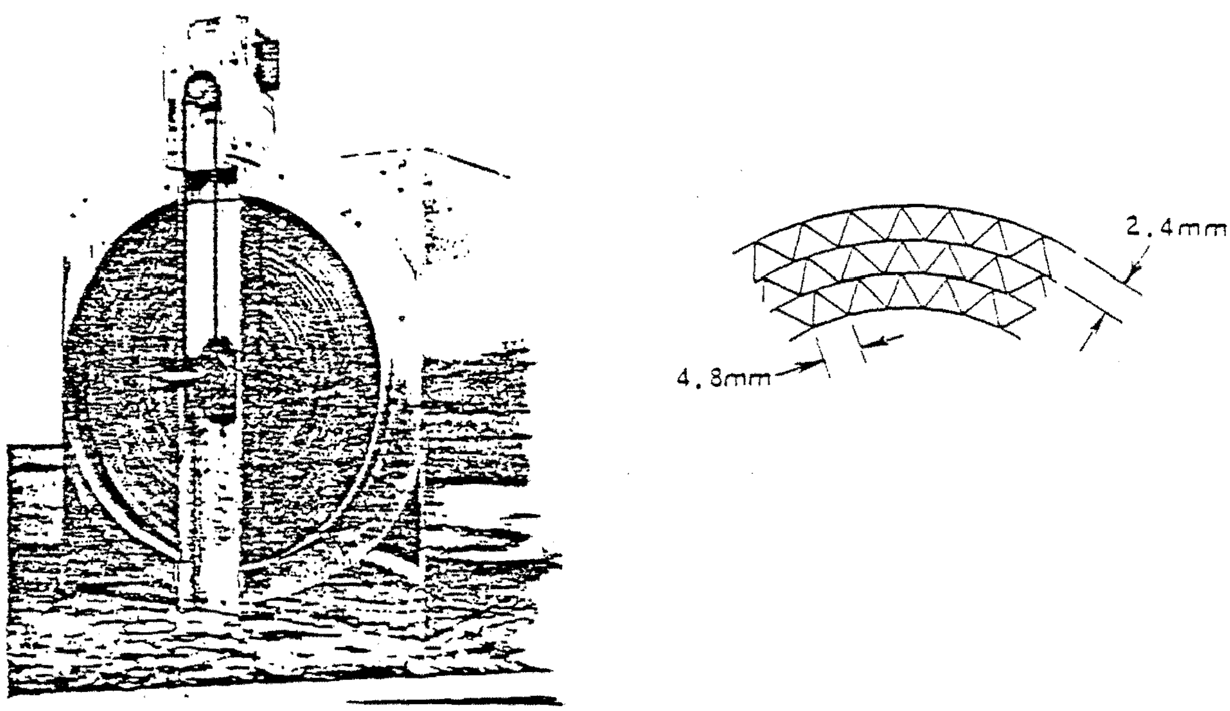

FIGURE 3
THE ROTOR AND ITS DRIVE

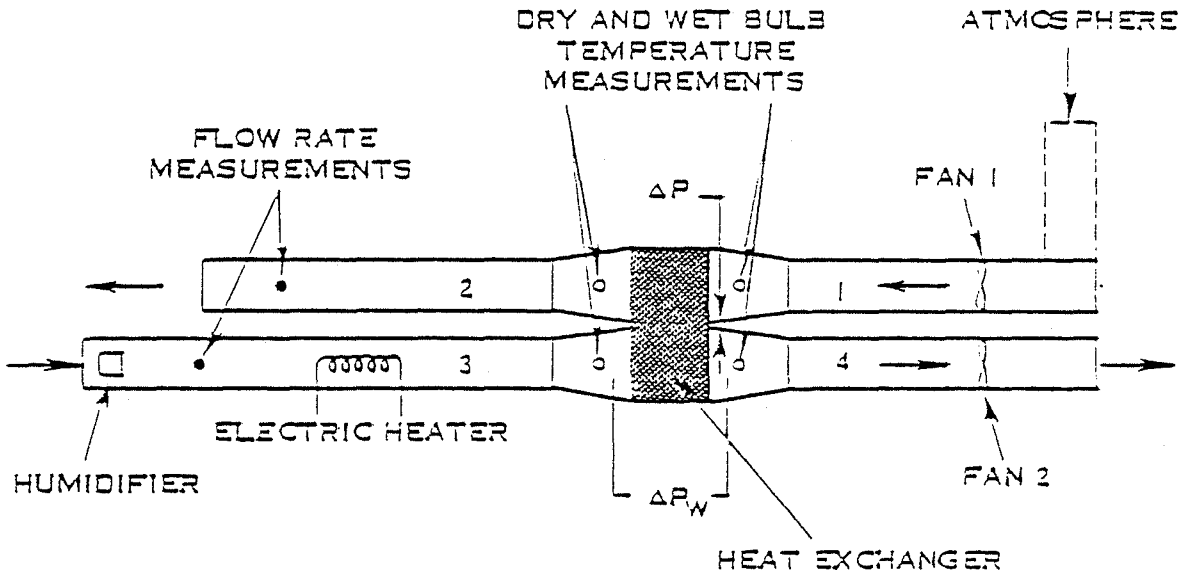

FIGURE 4

SCHEMATIC DIAGRAM OF THE TEST ARRANGEMENT

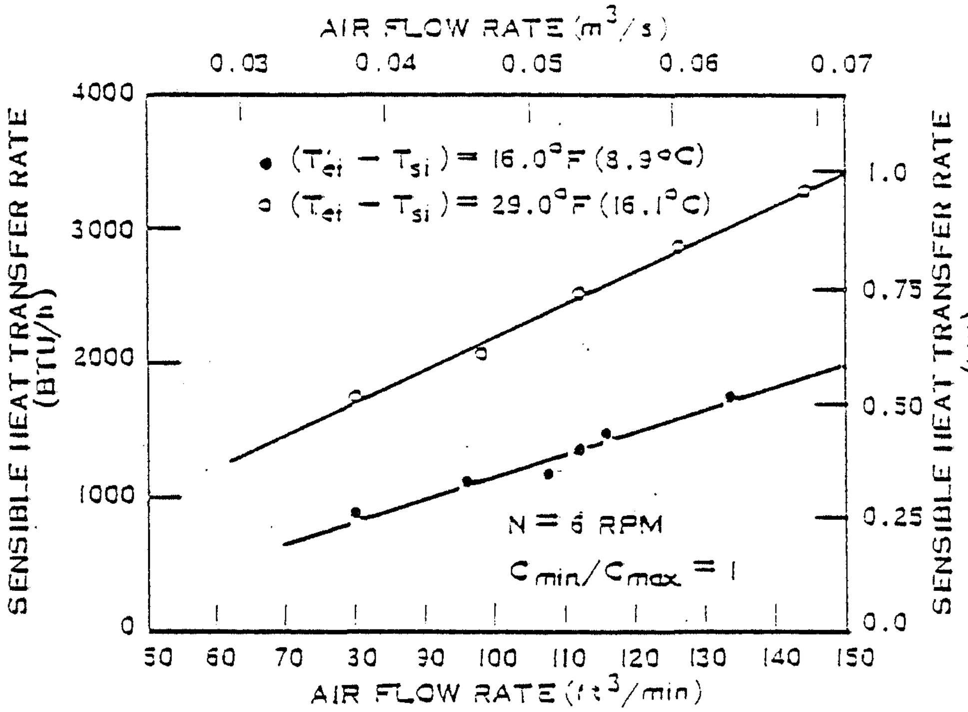

FIGURE 5

THE EFFECT OF THE AIR FLOW RATE AND AVAILABLE TEMPERATURE DIFFERENCE ON THE HEAT TRANSFER RATE

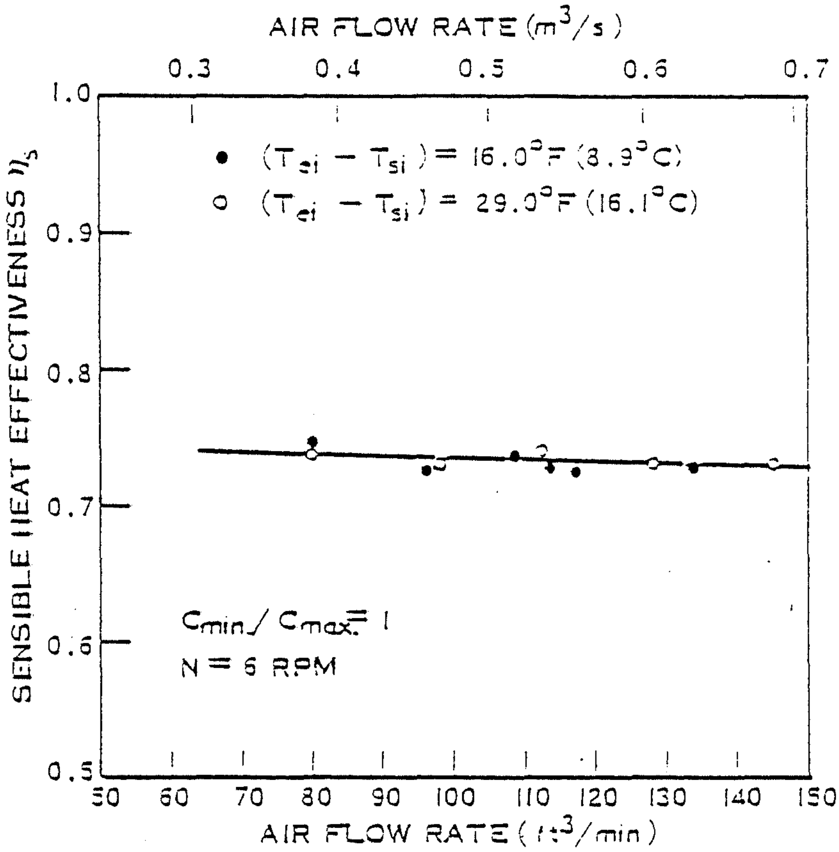

FIGURE 6

THE EFFECT OF THE AIR FLOW RATE
ON THE HEAT EXCHANGER EFFECTIVENESS

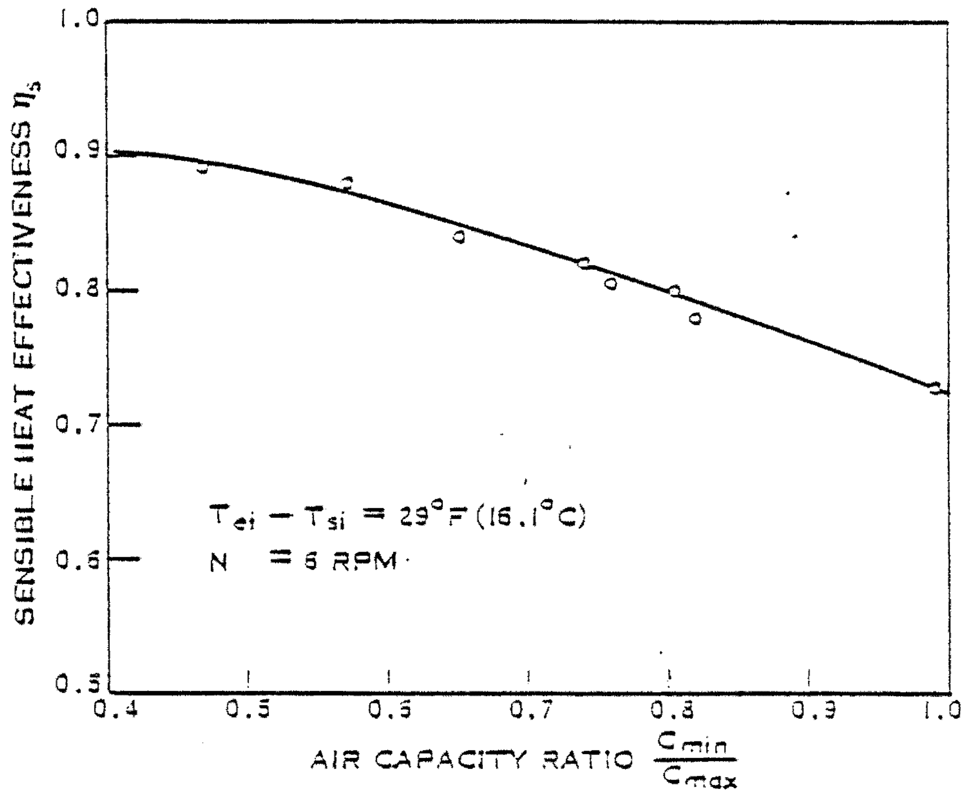

FIGURE 7

THE EFFECT OF THE AIR CAPACITY RATIO
ON THE HEAT EXCHANGER EFFECTIVENESS

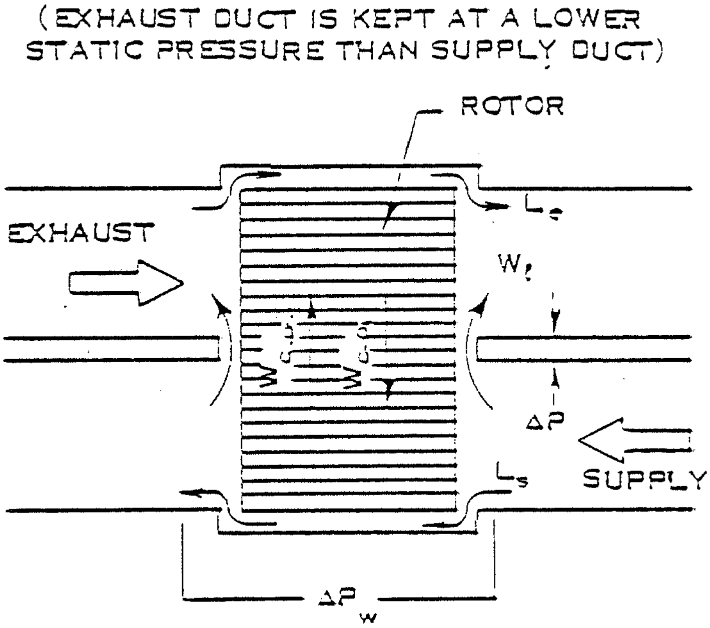

FIGURE 3

LEAKAGE AND CARRY—OVER IN
ROTARY HEAT EXCHANGERS

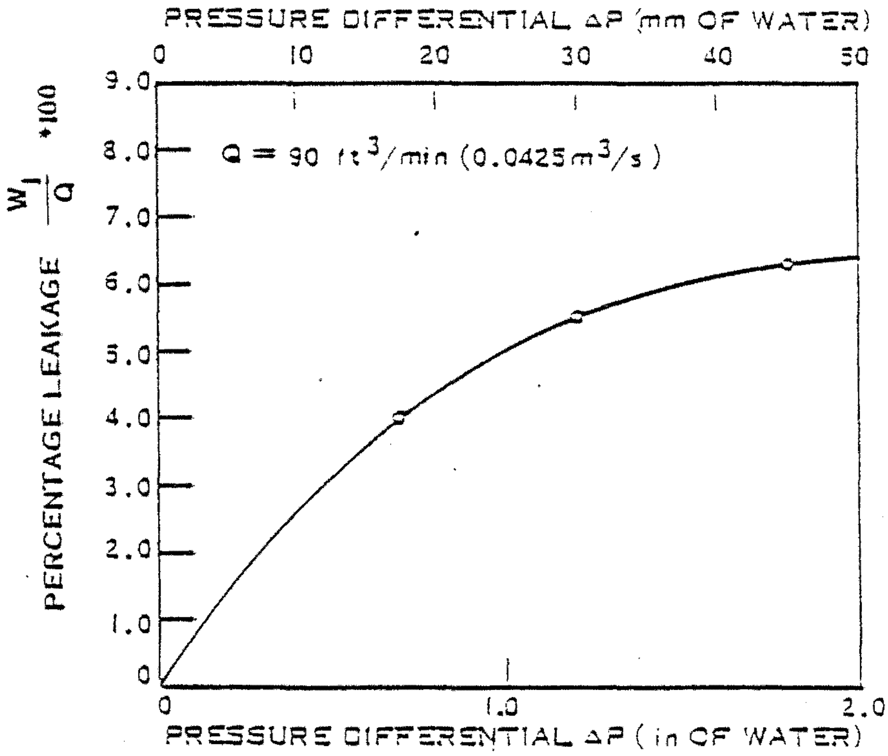

FIGURE 9

LEAKAGE RATES AS A FUNCTION OF
THE PRESSURE DIFFERENCE
BETWEEN THE TWO DUCTS

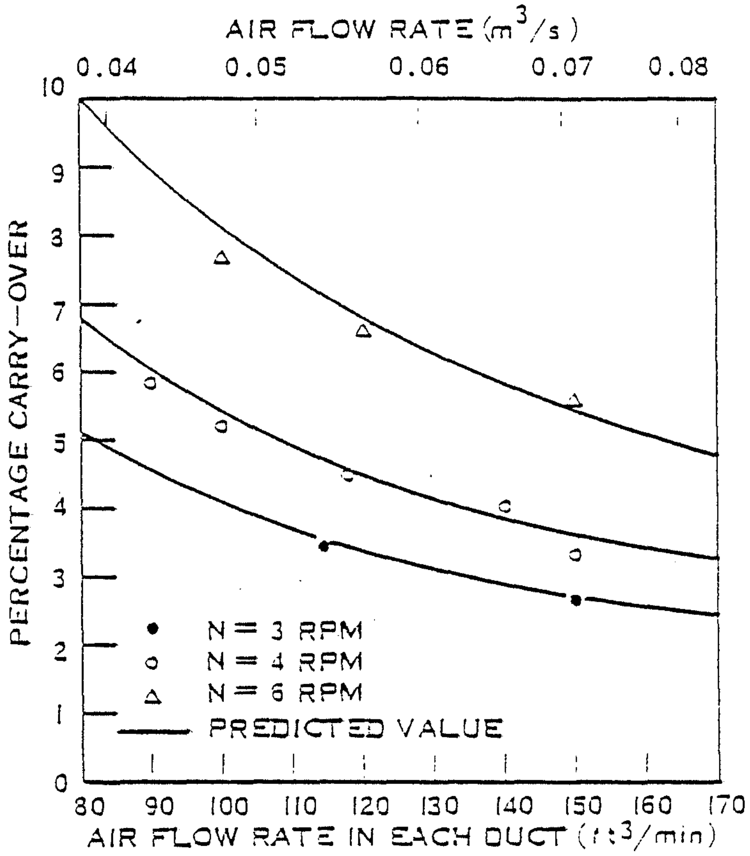

FIGURE 10

CARRY—OVER RATES IN THE TESTED WHEEL

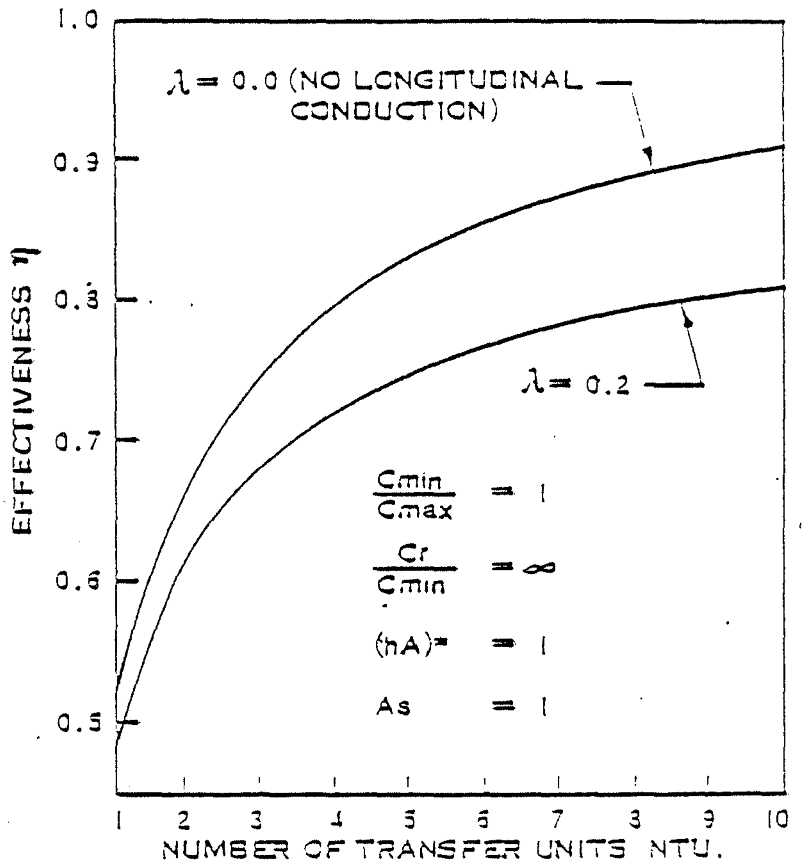

FIGURE II

THE EFFECT OF LONGITUDINAL CONDUCTION

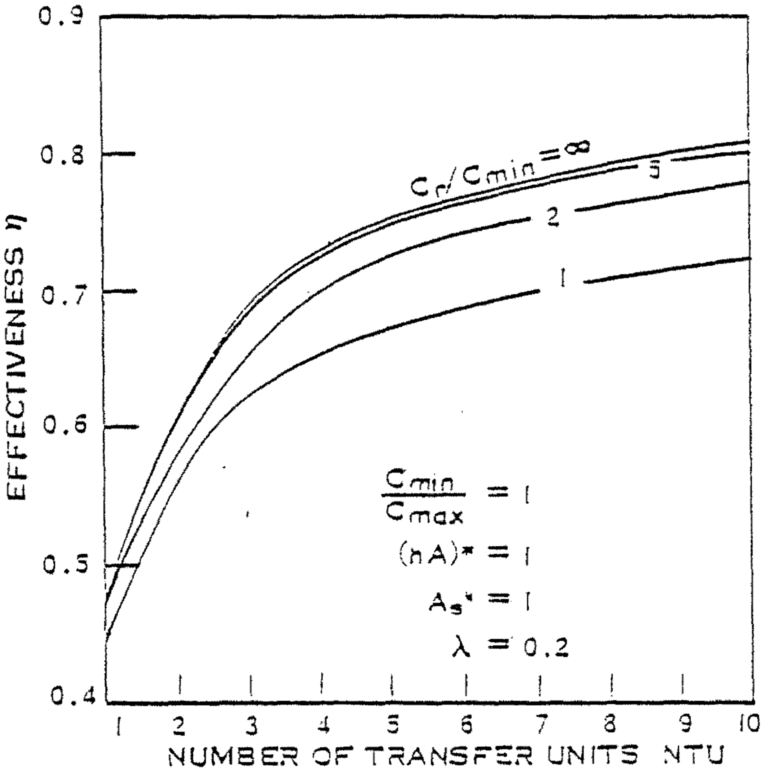

FIGURE 12

THE EFFECT OF C_r/C_{min} ON THE ROTARY HEAT EXCHANGER EFFECTIVENESS

E - 2 DISCUSSION

Dr. Shoukri passed around a sample of aluminum foil honeycomb similar to the type which they planned to use in their second prototype. There was discussion regarding the possible difficulty in cleaning the small cells of the honeycomb. The wheel is intended to be demountable and weigh about 10 lb. It could be removed and washed in the laundry tub but it is unlikely that it would be cheap enough to be considered disposable.

Dr. Shoukri was not prepared to predict the cost of this design although he didn't think it would be very expensive.

He expected the second prototype to produce a pressure drop along its length of 0.15 in. of water and have a face velocity of 300 ft./min. at a flow of 50 standard cfm. The first prototype produced a pressure drop of 0.4 in. of water. The second prototype is designed to operate at a rotational speed of 6 R.P.M.