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Performance of an Air-to-Air Heat Exchanger and an Exhaust Air Heat Recovery Heat Pump

by A. Kim

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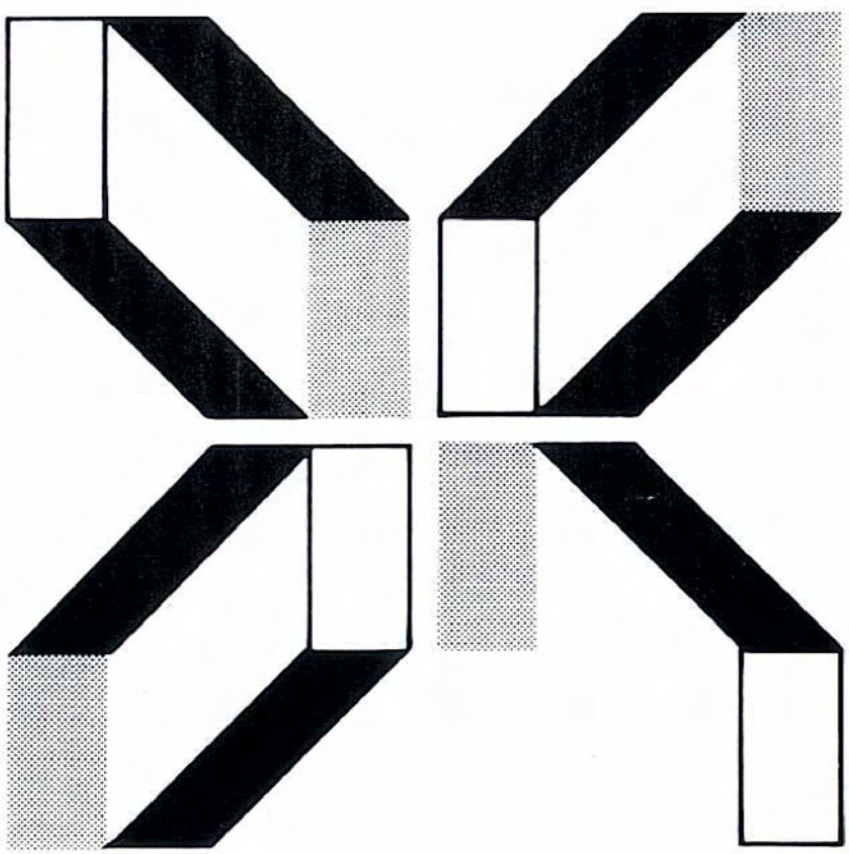

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PERFORMANCE OF AN AIR-TO-AIR HEAT EXCHANGER AND AN EXHAUST AIR HEAT RECOVERY HEAT PUMP

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ABSTRACT

The main objective of the study was to determine the thermal performance parameters of an air-to-air heat exchanger and an exhaust air heat recovery heat pump.

The experimentally determined performance parameters were used to estimate the energy savings that would be achieved by using these devices in houses located in various regions of Canada. Using the test house as a model, the predicted energy savings using the heat exchanger were between 2600 and 5000 kW·h and for the exhaust air heat recovery heat pump, between 5000 and 7100 kW·h, depending on the degree-days of the location.

RÉSUMÉ

L'étude avait pour but de déterminer les paramètres de rendement thermique d'un échangeur de chaleur air-air et d'une pompe à chaleur de récupération de l'air extrait.

Les paramètres d'essai ont servi à évaluer les économies d'énergie qui résulteraient de l'utilisation de ces appareils dans des maisons de diverses localités canadiennes. Pour une maison pilote, les économies prévues au moyen d'un échangeur de chaleur variaient entre 2600 et 5000 kW·h, et entre 5000 et 7100 kW·h à l'aide d'une pompe à chaleur de récupération de l'air extrait, le tout étant fonction des degrés-jours attribués à la localité.

INTRODUCTION

Heat recovery devices are being sold in Canada even though their capability for recovering heat from the exhaust air is not well known. A project was carried out, therefore, to determine the effect of an air-to-air heat exchanger and an exhaust air heat recovery heat pump on house energy use. The objectives of this study were to:

(1) determine the thermal performance of the two heat recovery devices;

(2) estimate the house seasonal energy savings in various regions of Canada that would be achieved by such devices.

Since the primary function of these devices is ventilation, the heat recovery aspect is an optional part of a mechanical ventilation system. The use of a heat recovery component can only be justified if the resultant energy savings are commensurate with the extra cost of having such an option. For this reason, two types of residential heat recovery devices were tested to provide information needed for an economic assessment.

An air-to-air heat exchanger consists of a heat exchanger core, a supply fan and an exhaust fan. The supply fan draws outdoor air into the house and the exhaust fan discharges indoor air to the outside, through the heat exchanger core. As the two air streams pass through the heat exchanger core, heat is transferred from the exhaust air to the supply air.

An exhaust air heat recovery heat pump replaces the heat exchanger by a heat pump with the evaporator in the exhaust air stream and the condenser in the fresh air stream. In this way, the heat extracted from the exhaust air by the evaporator, the heat dissipated by the compressor, and the fan energy, are transferred to the fresh air.

TEST HOUSES AND HEAT RECOVERY DEVICES

Two houses of the CHBA(HUDAC)/NRC Mark XI energy research project [1] were used for these experiments. House H4 was used for the heat exchanger test and house H2 was used for the EAHRHP test. These two-storey houses with full basements are located side-by-side in a developed residential area in the city of Gloucester, Ontario; they are of the same plan and construction and have similar air leakage characteristics. The thermal resistance of the walls is 3.5 $m^2 \cdot {}^\circ C/W$, of the basement it is 1.3, and of the ceiling, 5.6. The house volume is 386 m^3. Each house has a forced air heating system with an electric furnace. The furnace fan was on continuously during the experiment.

Air-to-Air Heat Exchanger

One of the air-to-air heat exchanger models commonly used in low energy houses was selected for testing. Figure 1 shows a sketch of the heat exchanger. The specifications of the device are given in Table 1.

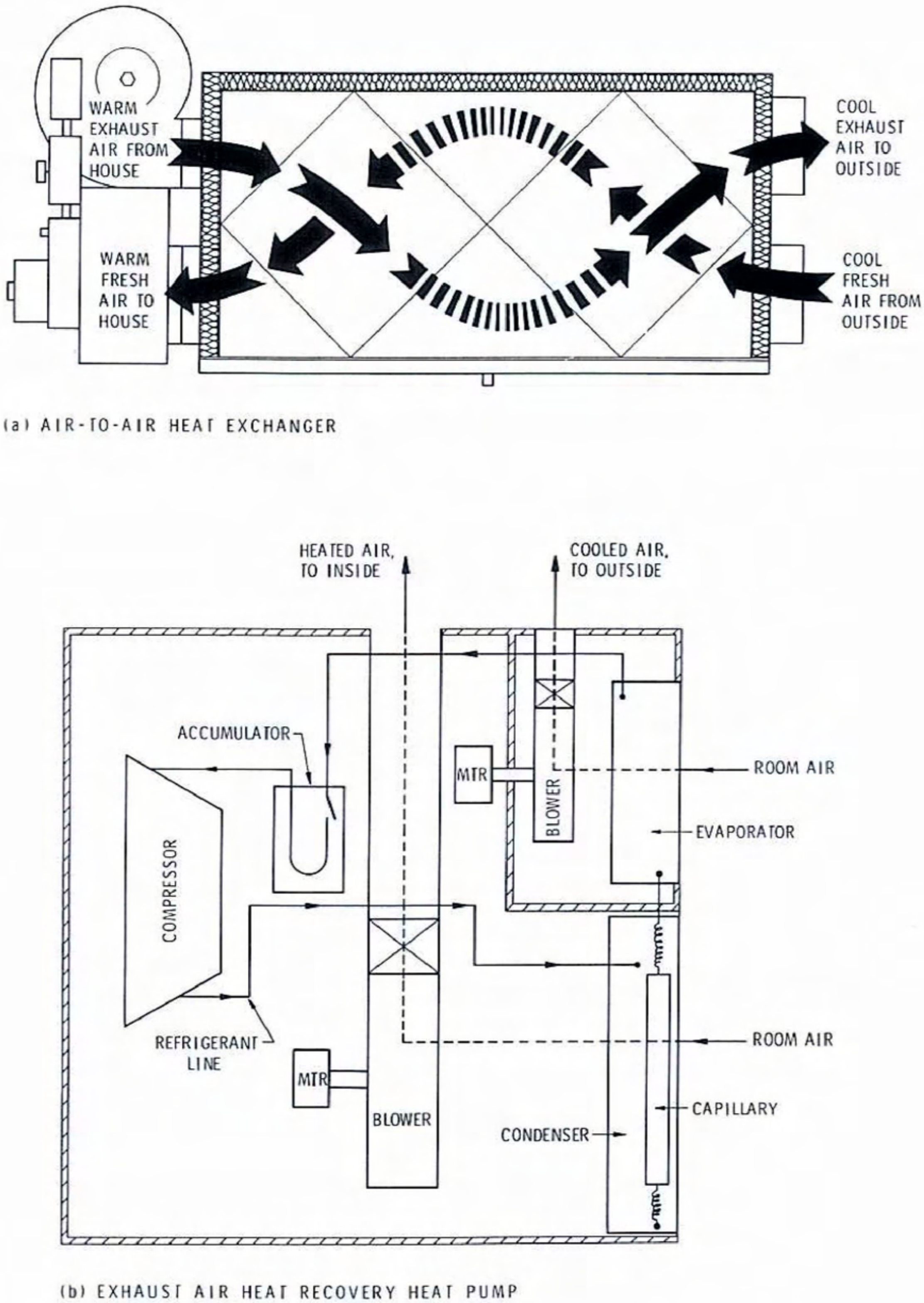

Figure 1 The heat recovery devices.

TABLE I DESCRIPTION OF THE HEAT RECOVERY DEVICES

(a) Air-to-Air Heat Exchanger

Type	double cross-flow
Size	$121 \times 38 \times 56$ cm
Core material	polyolefin plastic
Core surface area	12 m^2
Exhaust fan capacity	118 L/s without any duct attachment
Supply fan capacity	118 L/s without any duct attachment
Defrost control	demand type defrost
Manufacture's stated efficiency	71% at 94 L/s and 83% at 47 L/s

(b) Exhaust Air Heat Recovery Heat Pump

Type	air-to-air indoor unit
Size	$47 \times 69 \times 62$ cm
Exhaust fan capacity	52 L/s at 59.5 Pa of external static pressure
Condenser fan capacity	260 L/s at 59.5 Pa of external static pressure
Compressor	reciprocating R22 compressor, 7900 watt size
Defrost control	timed defrost
Compressor control	1st stage of 2-stage thermostat

This unit was equipped with a demand defrost control for freeze-protection. The supply air (i.e., outside air) fan was turned off when $T_{s,w} \leq 5°C$, where $T_{s,w}$ is the supply air temperature downstream of the heat exchanger core. The warm exhaust (i.e., indoor) air flowing through the heat exchanger core supplied the heat needed to melt the accumulated frost or ice. The supply air fan was turned on for normal heat exchanger operation, as soon as $T_{s,w} \geq 14°C$.

The installation of the air-to-air heat exchanger is shown in Figure 2. As shown, the supply air duct was not connected directly to the return plenum of the heating system. The direct connection would induce cold outdoor air flow through the heat exchanger core during the defrost cycle and this would increase the defrost period.

Exhaust Air Heat Recovery Heat Pump

The test unit was one of the two prototypes constructed specially for the Division of Building Research. Figure 1 shows a sketch of the unit. The specifications of the unit are given in Table 1.

The evaporator fan was set to operate continuously to provide the ventilation air, but the compressor, condenser fan and electric furnace were controlled by a two-stage thermostat located in the living room. The thermostat turned on the heat pump when the room temperature dropped below a set point (normally 21°C), and the electric furnace was turned on when room temperature dropped 1.1 K below this set point.

A timer-control was used to defrost the evaporator coil. For every 34 minutes of operation, the compressor was shut down for 4.5 minutes for defrosting. During the defrosting period, the exhaust fan continued to discharge the indoor air to the outside. This air flow supplied the heat needed to melt the ice off the evaporator.

Figure 2 shows the installation of the exhaust air heat recovery heat pump.

TEST CONDITIONS AND MEASUREMENT METHOD

The experiments were conducted in unoccupied houses from December to April of the 1983-84 heating season. The nominal relative humidity of the houses was 30-35% and the indoor temperature was 21°C. The forced ventilation rate, i.e., the air flow rate of the supply and exhaust fans, was set at 54 L/s. This flow rate was chosen to ensure a house air change rate of 0.5 ac/h, as recommended by the ASHRAE Standard 62-81, Ventilation for Acceptable Indoor Air Quality [2]. The mechanical ventilation systems and the furnace fan ran continuously for the whole heating season. The heat recovery devices were activated every other week so that the effect of heat recovery units on the house heating demand could be determined.

Data for the operation of the electric furnace, and for the mechanical ventilation systems with and without heat recovery devices in operation, as well as the weather data, were collected by a computer-based data acquisition system. The monitored variables were:

(a) indoor and outdoor air temperature,
(b) dry bulb and dew point temperature in the ventilation system (sensor locations are shown in Fig. 2),
(c) energy use by the electric furnace, fans and compressor,
(d) air flow rates of the supply and exhaust systems.

Manual readings were taken daily for the air temperatures in the ventilation system, and the energy uses of the electric furnace, fans and compressor, as a check on the computer system. Also, the air change rates of the houses were measured at least twice each day using the tracer gas decay method [3].

TEST RESULTS

Air-to-Air Heat Exchanger

The thermal performance of a heat exchanger is usually described by:
thermal effectiveness factor (E), %,
temperature recovery factor (TR), %,
seasonal heat recovery factor (η), %.

The thermal effectiveness and temperature recovery factors are parameters used for rating an air-to-air heat exchanger and are given by reference 4:

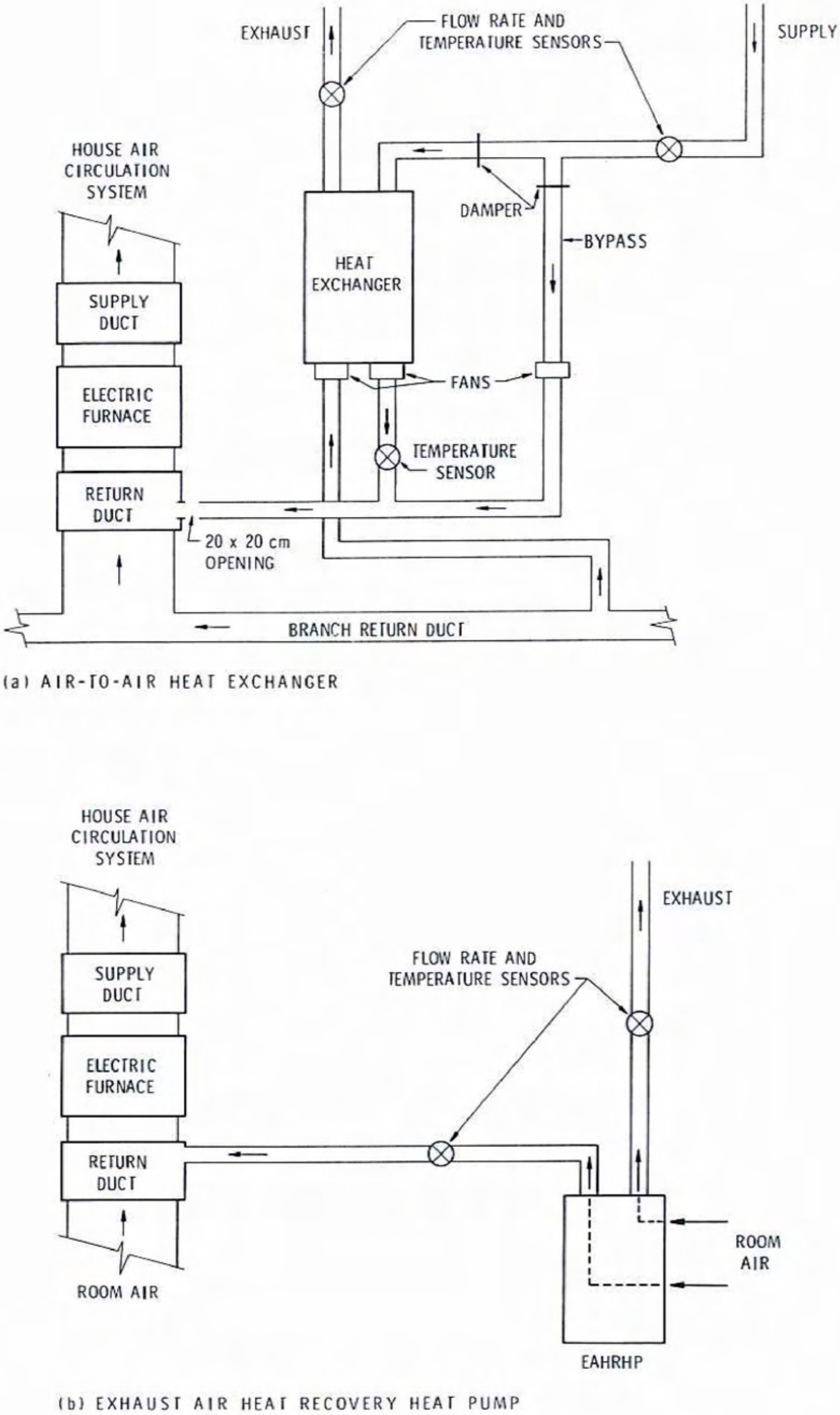

Figure 2 Heat exchanger and EAHRHP installation.

$$E = \frac{h_{s,w} - h_{s,c}}{h_{e,w} - h_{s,c}} \times 100\% \tag{1}$$

and

$$TR = \frac{T_{s,w} - T_{s,c}}{T_{e,w} - T_{s,c}} \times 100\% \tag{2}$$

where

$h_{s,w}$ = enthalpy of air supplied to house by heat exchanger, J/g,
$h_{s,c}$ = enthalpy of outside air, J/g,
$h_{e,w}$ = enthalpy of room air, J/g,
$T_{s,w}$ = temperature of air supplied to house by heat exchanger, °C,
$T_{s,c}$ = temperature of outside air, °C,
$T_{e,w}$ = temperature of room air, °C.

The two equations give identical results if the latent heat of vapourization associated with the moisture in the supply and exhaust air streams is neglected.

Figure 3 shows E and TR as a function of the outdoor air temperature, where E and TR calculations were based on measured values. As this figure indicates, E and TR are independent of outdoor temperature; the experimentally determined values are $E = 58\%$ and $TR = 72\%$. Figure 3 also shows five points which are substantially lower than the other values. These deviations occurred due to the reduction in exhaust air flow, which in turn was due to frost buildup on the heat exchanger core. This effect is shown in Figure 4, where the temperature recovery factor is plotted against the ratio of mass flow rates through the heat exchanger.

The purchased heating energy for a house can be approximated by the following equation, because basement heat loss and solar gain of a house are generally independent of outdoor temperature and approximately equal in value.

$$E_i = \overset{\text{heating season}}{\int G_i(t) \cdot dt} = \overset{\text{heating season}}{\int P_i \cdot \Delta T(t) \cdot dt} \tag{3}$$

where

E_i = purchased heating energy, kJ,
G_i = furnace power including the fan, kW,
= $P_i \cdot \Delta T$ (see Fig. 5),
t = time, s,
P_i = house heat loss factor, kW/K,
ΔT = indoor-outdoor air temperature difference, K,
i = B, mv, hx for no mechanical ventilation, mechanical ventilation only, and mechanical ventilation with the heat exchanger, respectively.

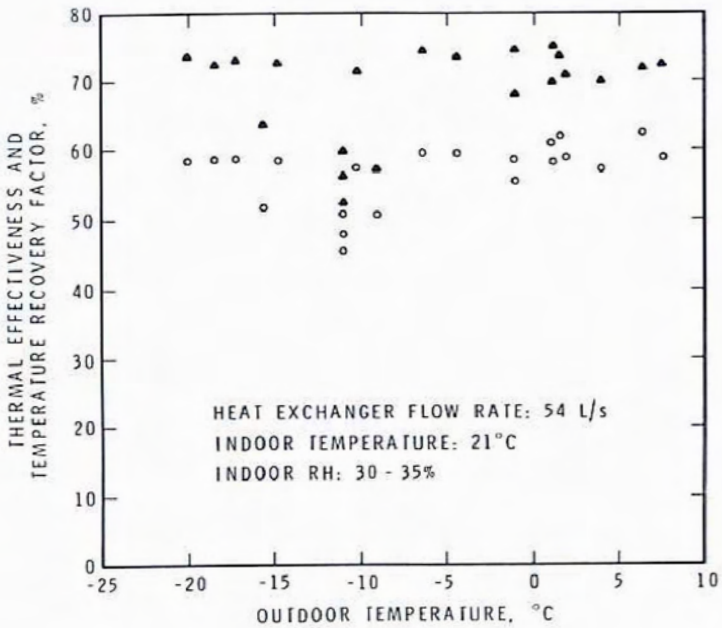

Figure 3 Heat exchanger effectiveness (E,○) and temperature recovery ratio (TR,▲) vs outdoor temperature.

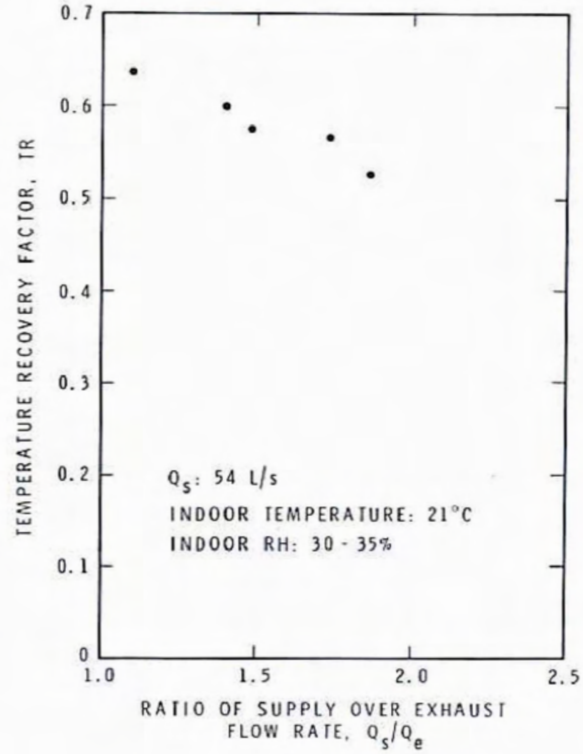

Figure 4 Temperature recovery factor vs supply and exhaust flow rate ratio

The seasonal heat recovery factor is given by

$$\eta = \frac{E_{mv} - E_{hx}}{E_{mv} - E_B} \times 100\%$$
$$= \frac{(P_{mv} - P_{hx})\overset{\text{heating season}}{\underset{\int \Delta T(t)\cdot dt}{}}}{(P_{mv} - P_B)\overset{\text{heating season}}{\underset{\int \Delta T(t)\cdot dt}{}}} \times 100\%$$
$$= \frac{P_{mv} - P_{hx}}{P_{mv} - P_B} \times 100\%. \tag{4}$$

The experimentally determined values of P_B, P_{mv} and P_{hx} were 0.119, 0.179 and 0.146 kW/K, respectively (see Fig. 5). The value of η for the heat exchanger, therefore, is 55%. This means that the heat exchanger was capable of recovering 55% of the energy increase due to the operation of the mechanical ventilation system.

Figure 6 shows the house air change rates with no mechanical ventilation, with mechanical ventilation alone, and with mechanical ventilation and heat exchanger. The house air change rates with mechanical ventilation, and with and without the heat recovery, were

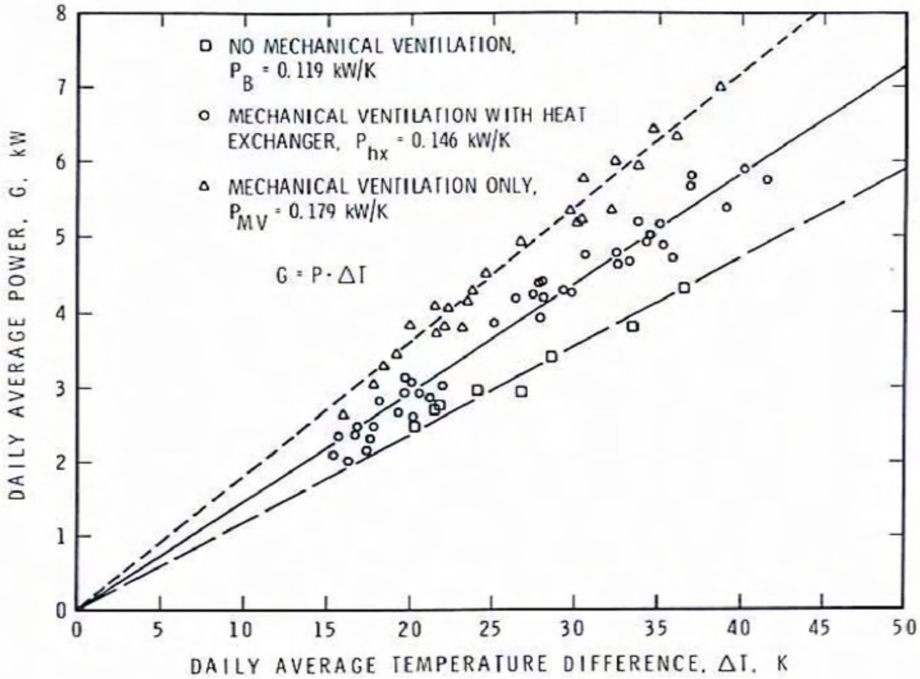

Figure 5 Daily average furnace power demand of the heat exchanger test house (H4).

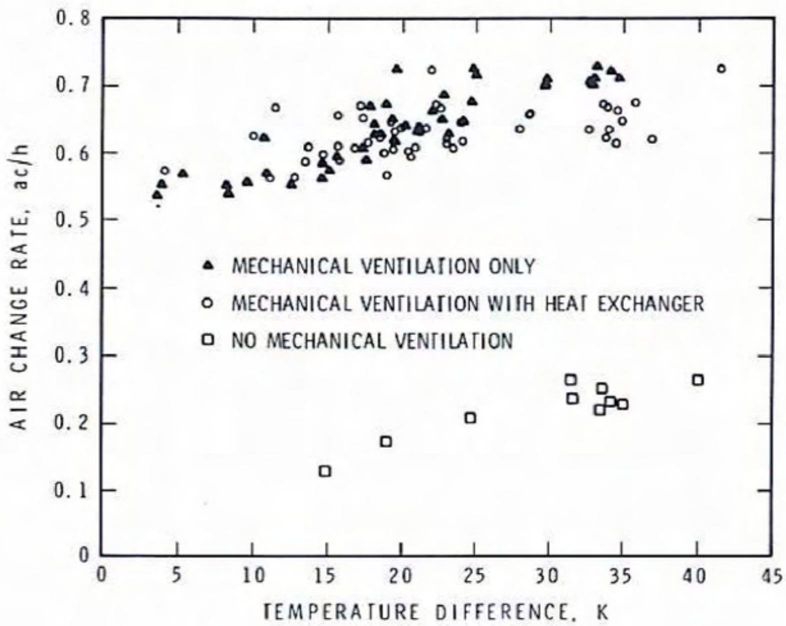

Figure 6 Air change rates of the heat exchanger test house (H4).

almost identical for an indoor-outdoor temperature difference less than 25 K. For higher temperature differences, the house air change rate without heat recovery was greater than that with heat recovery. This was caused by a reduction in the exhaust air flow rate due to ice or frost buildup in the heat exchanger core.

For a temperature difference greater than 20 K, typical Canadian winter conditions, the house air change rate was about 0.69 ac/h with mechanical ventilation and about 0.23 ac/h without mechanical ventilation.

Exhaust Air Heat Recovery Heat Pump

The thermal performance of the exhaust air heat recovery heat pump is described by the coefficient of performance (COP), which is given by

$$\mathrm{COP} = \frac{\int^{T^*} Q_r \cdot dt}{\int^{T^*} W \cdot dt} \tag{5}$$

where

Q_r = heat rejection rate of the condenser, W,
= $m_c \cdot \rho \cdot C_p(T_c - T_i)$,
W = power demand of the heat pump, W,
T^* = time period, s,
m_c = air flow rate through the condenser, L/s,
ρ = density of air flowing through the condenser coil, kg/m^3,
C_p = specific heat of air, J/(g·K),
T_c = air temperature downstream of the condenser coil, °C,
T_i = air temperature at the condenser fan intake, °C.

Equation 5 was used to calculate the COP from the experimental data for time periods when the outdoor air temperature was below the balance point temperature of the house - heat pump system (i.e., when the heat demand by the house to maintain room set point temperature was greater than the output of the heat pump). The average COP for this condition was 2.0; this value was based on the following measured average values,

$$\overline{W} = 1.65\ \mathrm{kW} \qquad \overline{m}_c = 203\ \mathrm{L/s}$$
$$T_c = 40^\circ\mathrm{C} \qquad T_i = 24^\circ\mathrm{C}$$

for the heat pump operation cycle of 34 minutes, during which the compressor and supply fan were "OFF" for 4.5 minutes for defrost.

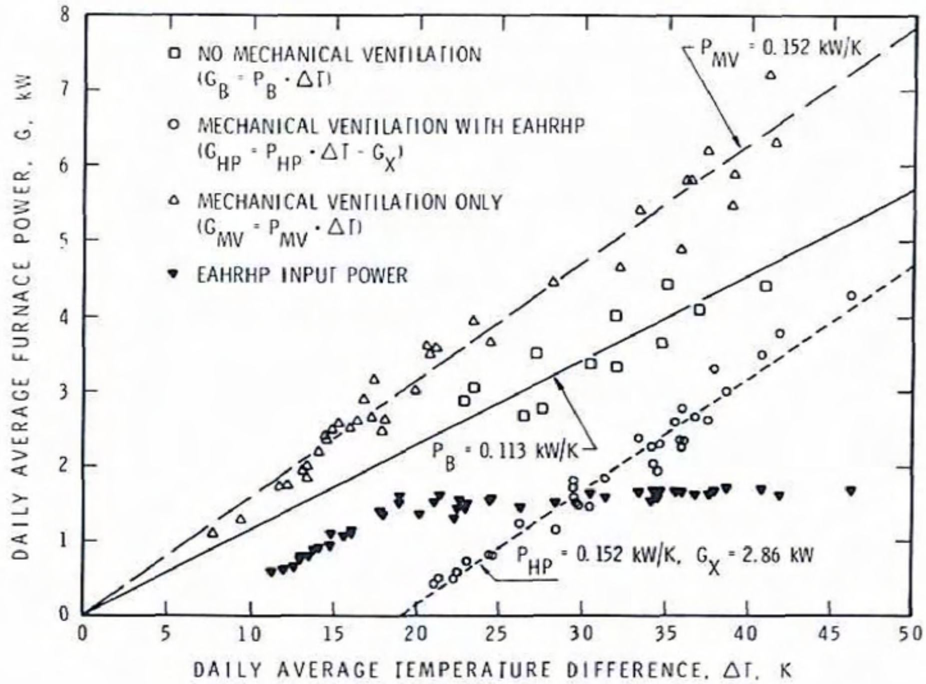

Figure 7 Daily average furnace power demand of the EAHRHP test house (H2).

The daily average furnace power for house #2 is shown in Figure 7 for the three test conditions: with no mechanical ventilation (house as is, i.e., 0.2 ac/h), with the exhaust-only ventilation system operating alone (0.5 ac/h), and with the exhaust air heat recovery heat pump in operation (0.5 ac/h).

The daily average furnace power increased linearly with the indoor-outdoor temperature difference. At a temperature difference of about 19 K, the daily average furnace power with the heat pump in operation is nearly zero (Fig. 7). This indicates that for an outdoor temperature of 2°C or higher (i.e., balance point temperature), the heat recovered from the exhaust air alone was sufficient to meet the house heating demand.

Measurements indicate that the house air change rate without any mechanical ventilation is 0.2 ac/h (see Fig. 8). Because of this low natural ventilation, the exhaust-only mechanical ventilation system determines the house air change rate. Thus when the exhaust-only mechanical ventilation system (heat pump) is installed and the exhaust flow rate is set at 0.5 ac/h, the total house air change rate also becomes 0.5 ac/h. (The operation of the compressor and the air circulating fan of the heat pump does not affect the house air change rate.)

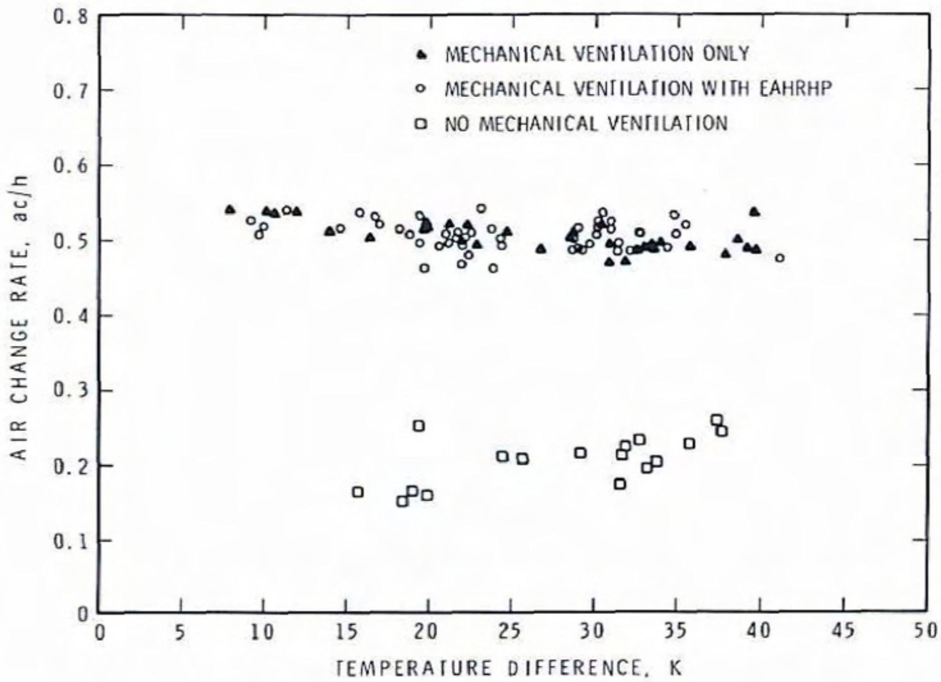

Figure 8 Air change rates of the EAHRHP test house (H2).

PREDICTED ENERGY SAVINGS

The energy savings by the heat recovery devices for various regions of Canada were calculated by a computer program (EASI3), developed at the Division of Building Research, and using experimentally determined performance factors of the heat recovery devices. The EASI3 program gives the monthly house heating requirement based on the hourly calculation of house heat loss rate and internal and solar gains.

The following data which describe the test house are required as input to the EASI3 program.

(i) dimensions of the house (see Ref.1),
(ii) area and R value of all building components (see Ref.1),
(iii) shading coefficient and orientation of each window (north = 0.53, south = 0.51 and east = 0.47),
(iv) seasonal average house air change rate (described in previous chapter),
(v) internal heat gain (1.0 kW), and
(vi) number of occupants (zero).

As a check on EASI3 program calculations, Figure 9 compares the energy use predicted by EASI3 and that calculated from Eq. 3 using the measured heat loss factor for the two test houses. The agreement is within 3%.

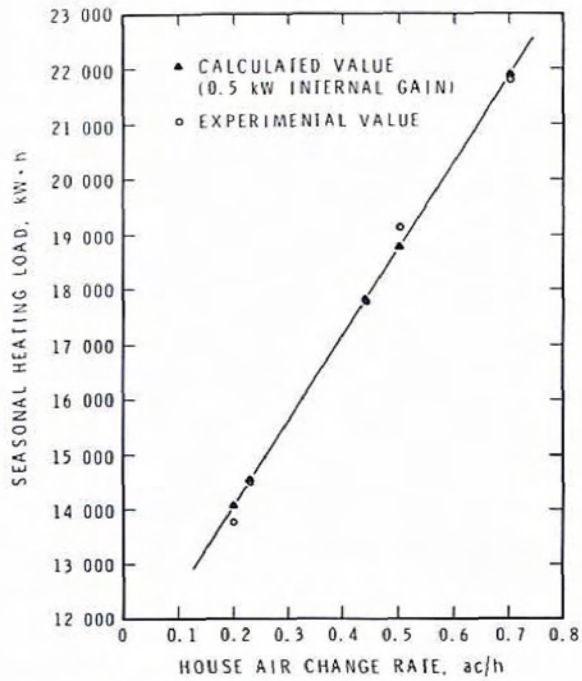

Figure 9 Comparison of seasonal heating load calculated by EASI3 program and that obtained by experiment.

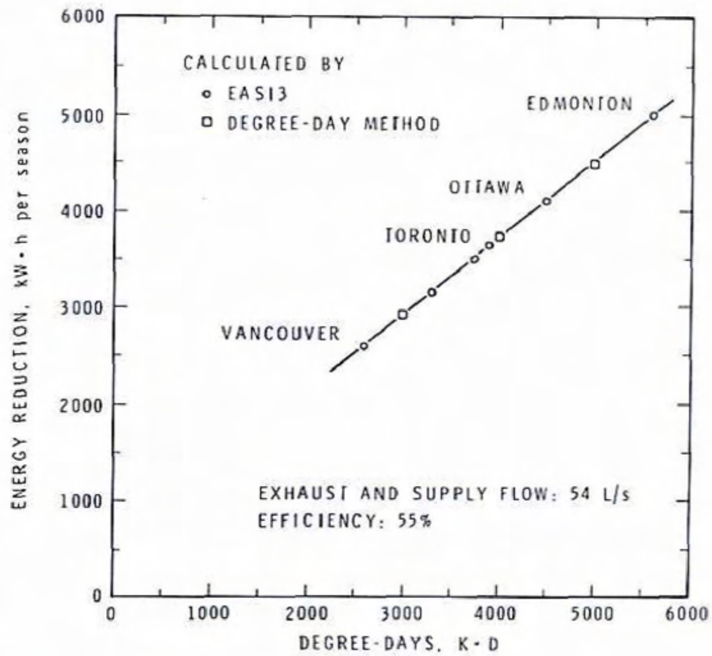

Figure 10 House seasonal heating energy reduction by the heat exchanger.

Air-to-Air Heat Exchanger

EASI3 does not have an option to account for any heat recovery devices directly. The effect of a heat exchanger, however, can be accounted for indirectly by replacing the actual air change rate with the apparent air change rate. The apparent air change rate that accounts for the operation of the heat exchanger in the test house was calculated to be 0.44 ac/h (calculation procedure is given in Appendix A).

The calculated house heating energy reduction as a result of the use of the heat exchanger is shown in Figure 10 for various places in Canada. It indicates that the heating energy reduction per season increased linearly from 2500 kW·h to 5000 kW·h as the heating degree-day (based on 18°C) increased from 2600 to 5600 K·D.

For comparison, Figure 10 also shows the energy savings determined by EASI3 (in kW·h per season) and by the following equation

$$\begin{aligned} E_s &= \rho \cdot C_p(I_{mv} - I_{hx})V \cdot D \cdot 24 \\ &= 1.21(I_{mv} - I_{hx})V \cdot D \cdot 24 \end{aligned} \tag{6}$$

where

E_s = energy saved by heat exchanger, kJ/season,*
I_{mv} = air change rate with mechanical ventilation, ac/h,
I_{hx} = apparent air change rate, ac/h,
V = house volume, m^3,
D = heating degree days (based on 21°C), K·D.

As expected, energy savings predicted by Eq. 6 and by EASI3 are equal.

Exhaust Air Heat Recovery Heat Pump

EASI3 was modified to include the option to estimate the energy savings by a heat pump. It was assumed that the output of the heat pump was 3.22 kW and the COP was 2 (i.e., measured performance parameters of the test EAHRHP). Also, a constant air change rate of 0.5 ac/h was used in these calculations.

Figure 11 shows the energy savings per heating season by the heat pump for various regions of Canada. The energy savings increased from 5000 kW·h to 7100 kW·h as the heating degree days increased from 2600 to 5600 K·D. The heat recovered per heating season from the exhaust air by the heat pump was sufficient to meet 70% of the house heating energy requirement even at Edmonton, where the winter climate is severe. In addition, calculations were carried out for a range of COP values to determine the effect of the COP on energy savings. The results of these calculations are shown in Figure 12.

DISCUSSION

Comparison Between Heat Exchanger and Heat Pump

Figure 13 compares the calculated energy savings obtained by the two heat recovery devices. To make such a comparison, it was assumed that the house under consideration was tightly built and therefore the natural air infiltration through the house envelope was negligible. Based on a constant house air change rate of 0.5 ac/h supplied completely by the mechanical ventilation system, the energy savings obtained by the heat pump would be 1.4 to 2 times those obtained by the heat exchanger.

Cost Effectiveness of Heat Recovery Units

There is a wide variety of methods by which the benefits of an energy conserving option can be weighed against its cost. Each has its advantages and disadvantages. In this study, the "Present Worth" method seems to be most applicable because the installed cost of the EAHRHP is not known yet.

The present worth is an amount of money which, if borrowed and repaid annually at the same rate as the monetary value of the annual energy savings by the heat recovery unit,

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* 1 kJ = 2.778×10^{-4} kW ·h.

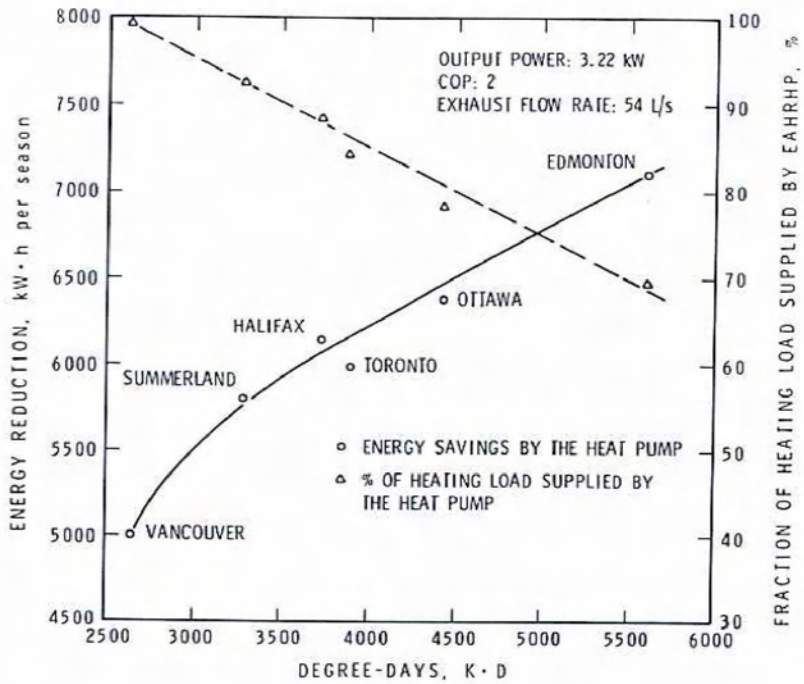

Figure 11 House seasonal heating energy reduction by the EAHRHP.

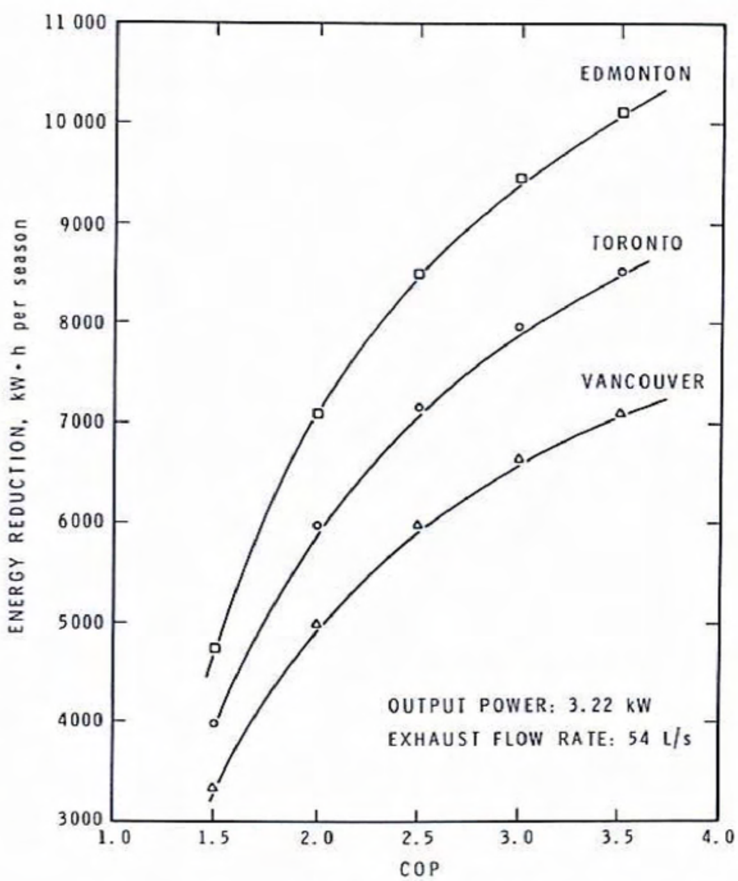

Figure 12 Effect of COP on energy savings by EAHRHP.

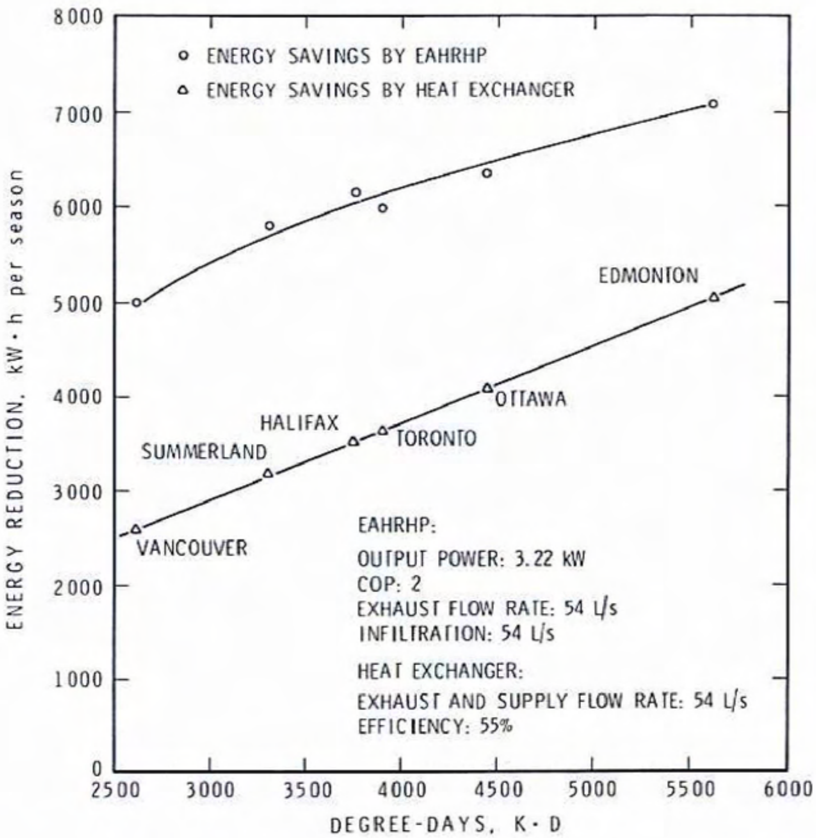

Figure 13 Comparison of house heating energy reduction attained using the heat exchanger and the EAHRHP.

would just be repaid by the time the device had to be replaced. It is calculated using the following equation [5]:

$$PW = \frac{C(1-(1+a)^{-n})}{a} \tag{7}$$

where

$$a = \frac{i-e}{1+e} \tag{8}$$

PW = the present worth of the annual energy savings over n years,
C = the first year saving,
a = the effective interest rate,
e = the rate at which energy costs are expected to increase,
i = the interest rate on money borrowed to purchase the energy-saving option,
n = the lifetime of the equipment.

The service life of the heat recovery unit is difficult to estimate. For this study, both 5 years and 10 years have been considered. Ten years was selected because many building owners are unwilling to look beyond 10 years and, therefore, this represents their maximum allowable n value.

In the present study, electricity has been chosen as the source of house heating energy for simplicity in calculation. Utility rates are different for different parts of Canada and vary from 3¢/kW·h in Winnipeg to 6.3¢/kW·h in Halifax, according to an energy costs survey done in late 1983 [6]. However, for most cities, the utility cost is about 4¢/kW·h and this value was used in this study.

The annual interest rate (i) was assumed to be 13% over the period under consideration based on the prime lending rate of 1984. The average annual rate of expected utility cost increase (e) for various locations in Canada was estimated at 8%, based on the recent study by Energy, Mines and Resources Canada, which developed estimates of escalation rates for gas, oil and electricity for a major city in each province for the period from 1983 to 1997 [7].

The calculated present worth values of the energy savings by the heat recovery units over 5-year and 10-year periods are shown in Figure 14 for various locations in Canada. If either of these types of heat recovery device can be purchased and installed at a cost that does not exceed the present worth of the savings that it will produce, it is "cost effective". The best option is the one that has the largest difference between present worth and the cost of the unit.

The result in Figure 14 was obtained for a house which is same size as the CHBA/NRC Mark XI test house and has a negligible natural air infiltration rate. If the heat recovery units are installed in a house which is leakier and/or smaller than the test house, the air flow rates (both exhaust and supply flow rate) of the heat recovery units should be reduced and, therefore, the energy savings by the units would also be reduced. This results in reduced present worth of the energy savings.

Defrost Performance

When the indoor (exhaust) air flows through the heat recovery units, the exhaust air temperature drops as the heat from the exhaust air is removed. Depending on the operating conditions, the exhaust air temperature may drop below the dew point temperature of the air, and if the dew point temperature is below the freezing point, then ice or frost starts to form in the exhaust side of the heat recovery unit. Occasional defrost, therefore, is needed to melt the ice off the heat recovery core to maintain the design values of air flow and heat recovery core thermal resistance.

The heat exchanger was defrosted by stopping the flow of outside air through the unit and using the heat from the exhaust air to melt ice off the heat exchanger core. Defrost operation was controlled by the temperature of the supply air downstream of the heat exchanger core as described before. Experimental observations indicated that this type of

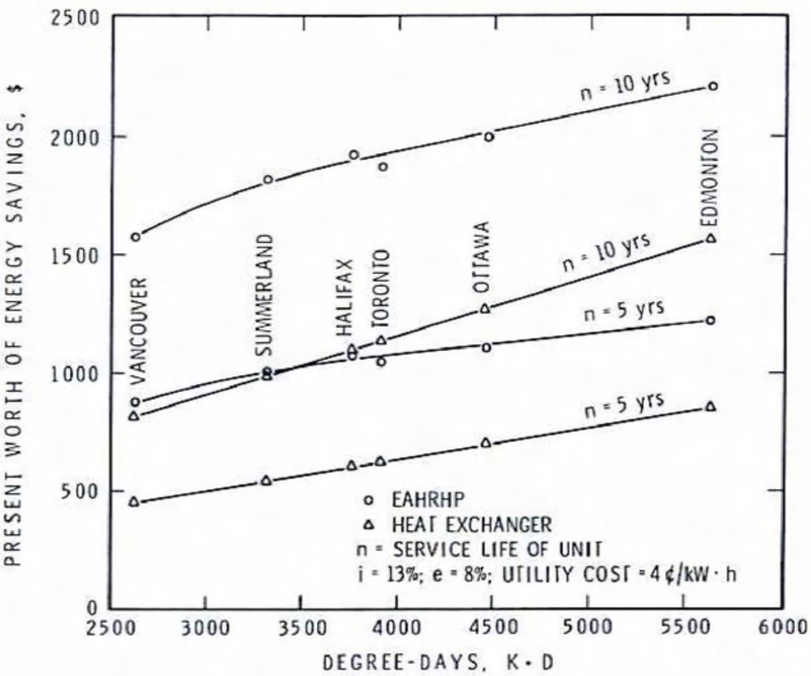

Figure 14 Present worth of the annual energy savings obtained by the heat exchanger and the EAHRHP for various locations in Canada.

defrost control creates problems because the supply air temperature downstream of the heat exchanger core is not a good indicator of the conditions of frost accumulation.

During the test of the heat exchanger, there was a heavy accumulation of ice on the outside surface of the uninsulated supply and exhaust duct when the outdoor weather was very cold (below -15°C). This ice melted and wetted the floor as the outdoor temperature rose. Therefore, both the supply and exhaust duct of the heat exchanger should be insulated.

The heat pump was defrosted by shutting down the compressor for 4.5 minutes after every 29.5 minutes. The defrost operation was controlled by a timer regardless of the condition of the evaporator. Under cold weather conditions, the exhaust air flow rate through the evaporator started to decrease 20 minutes after the end of the last defrost cycle. The air exhaust rate decreased gradually; the maximum reduction sometimes was as much as 15% of the original value. The exhaust air flow rate would normally be back to its original value 4 minutes after the defrost operation was initiated.

The exhaust air temperature leaving the evaporator was -3°C, which was 7 K higher than the design value.

The above were the only major problems encountered in the operation of these two heat recovery devices.

SUMMARY

(1) The measured average seasonal heat recovery factor of the heat exchanger was 55%, while the value stated by the manufacturer is 70 to 80% at steady state conditions.

(2) The measured output of the heat pump was 3.22 kW and the COP was 2.0. The heat pump could meet all the heating demand of the test house for an indoor-outdoor temperature difference of up to 19 K.

(3) The predicted energy savings by the heat exchanger for the test house with an air change rate of 0.69 ac/h increased from 2600 to 5000 kW·h as the degree days (based on 18°C) increased from 2600 to 5600 K·D for various locations in Canada.

(4) The predicted energy savings by the exhaust air heat recovery heat pump for the test house with an air change rate of 0.5 ac/h increased from 5000 to 7100 kW·h as the heating degree days increased from 2600 to 5600 K·D for various locations in Canada.

(5) For the same house with the same air flow rate through the ventilation system, the predicted energy savings by the heat pump were 1.4 to 2 times greater than those by the heat exchanger.

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APPENDIX A APPARENT AIR CHANGE RATE

By definition, the value of the apparent air change rate (I_a) is such that the calculated purchased heating energy for a house without heat exchanger and with air change rate I_a equals the purchased heating energy for the same house with heat exchanger and with actual air change rate I_{mv} (i.e., $E_a = E_{hx}$).

Equation 4 relates E_{hx}, E_{mv}, E_B and η and, therefore, it can be used to calculate I_a as follows:

Solution of Eq. 4 for E_{hx} gives

$$E_{hx} = E_a = E_{mv} - \eta(E_{mv} - E_B). \tag{1A}$$

House heat transmission loss (E_T) is independent of air change rate. Purchased energy for heating ventilation air is

$$E_x = I_x \cdot K$$

where I_x = air change rate and K is a constant; substitution of the above in Eq.1A gives

$$I_a = I_{mv} - \frac{\eta}{100}(I_{mv} - I_B). \tag{2A}$$

Sample calculation:

Calculate I_a for $I_{mv} = 0.69$ ac/h and $I_B = 0.23$ ac/h with $\eta = 55\%$.

Eqn. 2A gives $I_a = 0.44$ ac/h .