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# A COMPARISON OF VENTILATION STRATEGIES FOR TIGHTLY CONSTRUCTED HOUSES IN COLD CLIMATES

# ÉTUDE COMPARATIVE DE TECHNIQUES DE VENTILATION POUR MAISONS ÉTANCHES À L'AIR DANS DES CLIMATS FROIDS

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There are at least three consensus definitions of the term *standard*: (1) a concept that has been established by authority, custom, or agreement to serve as a rule in the measurement of quality or the establishment of a practice or procedure; (2) a document, established by consensus and approved by a recognized body, that provides for common and repeated use, rules, guidelines, or characteristics for activities or their results, aimed at the achievement of the optimum degree of order in a given context; and (3) a prescribed set of rules or requirements concerned with the definition of terms; classification of components; delineation of procedures; specification of dimensions, materials, performance, design, or operations; measurement of quality and quantity in describing materials, products, systems, services, or practices; or description of fit and measurement of size.

A code (in the law) is a collection of laws (regulations, ordinances, or statutory requirements) adopted by governmental (legislative) authority. Codes are further classified as model codes, building codes, energy codes, fire codes, plumbing codes, mechanical codes, and electrical codes.

Relatively few uncontested, reliable guidelines have been developed for control of moisture problems in buildings. A principal reason is that not enough understanding exists to translate the results of numerous laboratory and field tests into universal design guidelines. One widely referenced work is the *ASHRAE Handbook—Fundamentals*, in which chapters prepared by ASHRAE technical committees on such topics as thermal insulation and moisture retarders, ventilation requirements, and infiltration provide moisture control guidance.

The author cites 11 references to ASTM and industry guidelines and to a large number of standards issued by ASTM, Great Britain, industry, and others, dealing with building materials and components of building envelopes and relating to air leakage, moisture, and heat transfer.

The three model building codes in the United States do not cover the control of moisture in buildings except to warn against possible damage from water vapor condensation. Ignored is the fact that the transfer of moisture may result in increased heat transfer and energy use in the building.

More than 20 sources of guidelines, standards, and codes have been cited by the author. There is no single information source, index, data base, or compendium in which the designer or practitioner may find all standards, codes, and guidelines dealing with moisture control in buildings. Obviously, such a resource would be widely used provided that it could be available in electronic form and be periodically updated.

## CHAPTER 4.3—LEGAL CONSIDERATIONS AND DISPUTE RESOLUTION: THE WATER-RELATED CONSTRUCTION FAILURE, Bruce Ficken

A construction failure occurs any time that construction fails to perform as intended or required. Thus, not only is a construction failure the bridge that collapses or the structural support system that fails, it is also the paint that peels prematurely, the roof that leaks, or the glue that fails to hold. Every construction failure raises key issues and considerations that must be understood for the contractor, architect, specification writer, or owner to protect himself.

In every jurisdiction, the contractor is required to perform construction fully in accordance with the contract documents. He is responsible for the resulting damages. Conversely, a contractor's compliance with plans and specifications is universally recognized as a contractor's defense against liability in construction failure cases.

The contractor who knows that there is a significant risk of a construction failure if he complies with applicable plans and specifications has a duty to bring these deficiencies to the attention of the owner. If there is a subsequent construction failure as a result of the specified deficiency, the contractor can be liable for his failure to warn. Similarly, a contractor has the duty to warn of defective site conditions if the contractor knows or should know of the defect.

In virtually every jurisdiction, design professionals, including architects and engineers, are required to follow accepted standards of practice. Therefore, they are liable for damages that may arise from failing to follow these standards. In a leading case on architect liability, the Pennsylvania Supreme Court held, "An architect is bound to perform with reasonable care the duties for which he contracts. His client has the right to regard him as skilled in the science of the construction of buildings, and to expect that he will use reasonable and ordinary care and diligence in the application of his professional knowledge to accomplish that for which he is retained. Thus, it has been held that a design professional impliedly warrants that his plans and specifications are free from defects and fit for the purpose for which they were produced."

Under this doctrine it can be argued that if the finished construction is built in accordance with plans and specifications, it

(a) will not leak,
(b) will not allow the accumulation of water where the building will be adversely affected,

(c) will not be duly affected by predictable influx of moisture in the physical construction,
(d) will expel water that enters the construction predictably, and
(e) will not utilize materials that tend to trap excessive amounts of water under predictable circumstances.

In any construction failure dispute, the owner is likely to argue that he hired a design professional to design and a contractor to build the building, so that if anything goes wrong with the job, it must be the responsibility of one or the other. That perspective ignores the owner's responsibility to maintain the building and not to modify the structure in a deleterious manner.

The author presents a case study on the court settlement of a construction failure involving a 16-story oceanfront hotel in which major mold and mildew problems developed almost as soon as the hotel was occupied. The case shows that the owner was able to collect only small damages from the architect and contractor, partly because during the court hearing he did not adequately develop and present the facts surrounding the operation of the hotel.

**ABSTRACT**

This report examines three devices (exhaust fan, air-to-air heat exchanger (ATAHE) and exhaust air heat recovery heat pump (EAHRHP)) which could be used to increase the ventilation rate of a tightly constructed house to a level sufficient to keep indoor air pollutants and moisture to acceptable concentrations.

Various types of EAHRHP's were examined and the non-frosting, space heating type was deemed to be the most practical for colder climates. The space heating, non-frosting EAHRHP and the ATAHE were each compared to the exhaust fan to determine the most economical choice of device for a range of Canadian climates. Based on current energy prices, the EAHRHP was found to be the most attractive, except in the case where space heating is undertaken by low cost gas; in which case an exhaust fan only is indicated. It was found that an EAHRHP could supply a substantial portion of the annual heating requirements for an energy efficient house.

**RÉSUMÉ**

Dans ce rapport trois dispositifs sont étudiés (ventilateur d'extraction, échangeur de chaleur air-air (ECAA) et extraction d'air avec récupération de la chaleur par pompe à chaleur (EARPC)). Ces dispositifs servent à augmenter la ventilation d'une maison étanche à l'air pour limiter à des niveaux appropriés les concentrations de polluants et d'humidité à l'intérieur.

On a examiné divers types de EARPC et constaté que le type avec chauffage des espaces habités qui ne présente pas de risque de givre s'est révélé le plus pratique pour les climats très froids. On a notamment comparé les types EARPC et ECAA avec le ventilateur d'extraction pour déterminer le type le plus économique dans les divers climats du Canada. En se fondant sur les prix actuels de l'énergie, le type EARPC s'est toujours révélé le plus économique, sauf dans le cas du chauffage au gaz bon marché où le ventilateur d'extraction l'emporte. Enfin on s'est aperçu que le EARPC peut fournir une partie importante du chauffage annuel dans une maison bien isolée thermiquement.

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# A COMPARISON OF VENTILATION STRATEGIES FOR TIGHTLY CONSTRUCTED HOUSES IN COLD CLIMATES

## 1.0 INTRODUCTION

Since the early 1970's, and the advent of steeply rising energy costs, houses have been built to increasingly strict energy conservation standards. Primarily this energy conservation has been achieved by increasing the thermal insulation levels and sealing up the cracks in the building shell to reduce the infiltration of unconditioned outdoor air. Although significant energy savings have been realized, there are negative aspects associated with such tightly constructed houses.

It has been found that if the ventilation rate of the house is lowered to below 0.5 of an air change per hour (ACH), problems with indoor air quality can arise [Ref. 1 and 2]. With such a low rate of fresh outdoor air entering the house (and corresponding low rate of 'stale' air leaving the house) pollutants and moisture can build up to what may be unacceptable levels.

This report looks at three different mechanical devices which could be used to increase the ventilation rate of tightly built houses. The three devices are an exhaust fan (EF), air-to-air heat exchanger (ATAHE) and an exhaust air heat recovery heat pump (EAHRHP). EF's and ATAHE's for residential applications are relatively straightforward devices with many commercial examples available on the market. However, the EAHRHP is a new concept which is currently very much in the developmental stage. Therefore this report will also look at some alternative designs for the EAHRHP and will select the configuration considered most suitable for this application.

## 2.0 VENTILATION STRATEGIES

The three systems compared are an exhaust fan (EF), an air-to-air heat exchanger (ATAHE) and a space heating, exhaust air heat recovery heat pump (EAHRHP). Of these systems, the first and last type are exhaust only systems which create a low pressure inside the house to increase the natural ventilation rate through the 'cracks' in the building shell. The ATAHE is classified as a balanced device where the airflow forced out of the house equals the air flow rate drawn into the house by the device, and hence the pressure inside the house is affected very little. For very tightly constructed houses with natural infiltration rates much lower than say 0.2 ACH, an exhaust only system is not considered appropriate because of the very low pressure needed in the house to encourage sufficient air to leak through the 'tight' building shell. This low pressure could cause problems with the building such as in opening exterior doors and windows.

To allow comparison of the three devices, a house with a natural ventilation rate of 0.3 ACH was considered. This detached, two-storey house with a heated basement was considered typical of new house construction in 'comfortable' middle class neighbourhoods in the northern USA and Canada. Details of the house are given in Appendix A. The ventilation devices were sized to produce a total ventilation rate of 0.5 ACH in this house. Unless a major source of pollution such as urea formaldehyde foam insulation (UFFI) is present in the house, this ventilation rate is considered to represent an acceptable standard for maintaining air quality [Ref. 1 and 2].

Shaw [Ref. 3] has presented findings on the effect of balanced and exhaust ventilation systems on the ventilation of a typical, tightly-constructed two-storey detached house with basement. Exhaust was from the basement and supply was to the return duct of the central, forced-air heating system. The air change rate was measured using the tracer gas ($N_2O$) concentration decay method. Tracer gas was injected into the furnace duct system and air samples for measuring the tracer gas concentration were also taken from the duct system. The furnace fan was run continuously during these tests and tests were run during a heating season with a range of indoor to outdoor temperature differences, wind speeds and wind directions.

During the colder winter months (outdoor temperatures below -10°C) the natural air infiltration rate for the test house was found to vary very little with temperature difference and wind speed or direction. An average value of 0.26 air changes per hour (ACH) was found.

For the exhaust only systems Shaw found that the air change rate was relatively insensitive to outside temperature and wind conditions. Also, it was found that as the ventilation flow rate increased, the neutral pressure level (NPL) was shifted upward, thereby increasing the infiltration but decreasing the exfiltration through the cracks in the building. Eventually at an exhaust flow rate corresponding to about 0.5 ACH, the NPL had migrated to the ceiling level of the second storey and the air infiltration rate equalled the exhaust air flow rate, with the exfiltration rate through the 'cracks' in the building being essentially zero. For the purpose of this exercise it will be assumed that the natural infiltration rate without mechanical ventilation, $I_o$, is 0.3 ACH. The EF and EAHRHP will be assumed to operate with an exhaust flow rate, $V_v$, of 70.8 std l/s corresponding to 0.5 ACH and this will just equal the infiltration rate.

This exhaust rate will be maintained for the entire heating season of nine months (6552 hours) for both devices to ensure adequate ventilation. The compressor and condenser fan of the EAHRHP will cycle on and off with the demand from the first stage of the thermostat. The second stage of the thermostat would be connected to the central furnace.

For the balanced system, Shaw found that during the colder winter months the ventilation rate for the house, I, was not just the sum of the natural infiltration rate and the mechanical exhaust flow rate, $V_v$, but is a linear relationship;

$$I = I_o + RV_v \qquad (1)$$

where the proportionality constant, R, is given by

$$R = 0.13\ (\Delta t)^{1/2} \qquad (2)$$

and $\Delta t$ is the temperature difference between inside and outside. The effect of wind on the ventilation of the house was much less than the temperature difference effect and was ignored in this correlation.

From Table 1 it is apparent that to ensure adequate ventilation over the whole heating season, a mechanical ventilation rate of 0.5 ACH is required for the balanced system. This apparent anomaly is mainly due to the fact that the various volumetric flow rates were not corrected for temperature. Therefore, as the ambient air increases in temperature, the air entering the house becomes less dense and so more volumetric flow rate was required to achieve the same amount of tracer gas dilution. Also, at indoor to outdoor temperature differences below 30°C the natural ventilation rate will probably decrease also, thereby requiring an increase in the mechanical ventilation rate to compensate. This flow rate is convenient to use for this comparison, as the other two devices use the same flow rate.

For the purpose of this exercise, the ATAHE will operate continuously with a discharge flow rate of 0.5 ACH (corresponding to 70.8 std l/s) for the nine months of the heating season. Staff at Lawrence Berkeley Laboratory determined an average heat exchanger effectiveness of about 0.6 for two 'good' quality ATAHE's [Ref. 4]. It will be assumed that this figure includes any internal cross stream leakage and allows for losses due to defrosting the device.

The characteristics of the three ventilation systems compared are illustrated diagrammatically in Figure 1.

## 3.0 DESIGN OF EXHAUST AIR HEAT RECOVERY HEAT PUMPS (EAHRHP)

For heat pumps, the heat flow rate from the condenser, $Q_c$, the heat flow rate to the evaporator, $Q_E$, and the power consumed by the compressor, $P_c$, are related in the following way (ignoring the small losses present).

$$Q_c = Q_E + P_c \qquad (3)$$

A coefficient of performance (COP) can be defined as in (4) (not allowing for the power to operate fans).

$$COP = \frac{Q_c}{P_c} \qquad (4)$$

therefore, $$Q_E = Q_c \left(1 - \frac{1}{COP}\right) \qquad (5)$$

For a constant condenser heat delivery, $Q_C$, it is apparent from (5) that the heat recovery by the evaporator, $Q_E$, will increase as the COP increases and the purchased power, $P_C$, will decrease.

A conventional, vapour compression heat pump operates on a reverse Rankine Cycle and for a particular system with a fixed condensing temperature, the COP increases almost linearly with evaporating temperature. Therefore, the higher the evaporating temperature, the greater the energy recovery by the heat pump. However, in the case of the EAHRHP considered, the evaporator is working with a fixed flow rate of exhaust air for a particular size of house (70.8 std l/s (150 scfm) for 0.5 air changes per hour in a 209 $m^2$ house) at a constant inlet temperature of 21°C (70°F). Therefore, the energy recovery from the exhaust air stream will increase as the evaporating temperature decreases. With these two conflicting requirements there will obviously be an optimum evaporating temperature at which a particular system will operate most efficiently.

Figure 2 shows the energy that can be extracted from the exhaust air stream. Two evaporator curves are used, the lower one is for a fixed size evaporator; 10 mm (3/8 inch) tube, 25 mm X 22 mm (1.0 inch X 0.866 inch) staggered tube spacing, 550 fins per metre (14 fins per inch), rippled and corrugated fins, 4 tube rows with a face area of 0.14 $m^2$ (1.5 $ft^2$). The upper curve is for a variable size evaporator which maintains an evaporating temperature 2.8°C (5°F) cooler than the air exiting from the evaporator. Also shown on figure 2 are compressor capacity curves for various condenser heat capacities. A constant condensing temperature of 49°C (120°F) was assumed and the capacities were calculated using actual COP values for a typical, high efficiency, hermetic, reciprocating compressor.

To obtain more energy from the air, the evaporator temperature must decrease and the capacity of the system must increase. Therefore, the size and cost of the system must increase as the energy recovery increases. A problem associated with evaporating temperatures below about -1°C is the frosting of the evaporator. This means that the system will have to be periodically turned off and the evaporator defrosted. Typically, the defrost would be accomplished by using the warm exhaust air to melt the ice.

The Hooper and Angus report [Ref. 5] on the EAHRHP recommended a temperature reduction of the exhaust air stream by 30°C (54°F). A prototype, based on the Hooper and Angust concept was built and tested. Tests of the prototype [Ref. 6] indicated that a defrost would be required after 30 minutes of running (for 40% relative humidity and airflow rates of 70.8 std l/s (150 scfm) and 269 std l/s (570 scfm) over the evaporator and condenser respectively).

These tests also indicated that, due to frosting of the evaporator, the heat delivered by the condenser decreased to 76% of its steady state value (attained after about 5 minutes of operation) after 30 minutes of operation.

Thomas [Ref. 7] found that on start-up, a heat pump's output can be predicted fairly closely by the equation:

$$\frac{Q_c}{Q_{c_{ss}}} = 1 - e^{-t} \qquad (6)$$

where $Q_{c_{ss}}$ is the steady state condenser output which is reached after about 5 minutes of operation. Figure 3 shows the variation of condenser output as the heat pump starts up and the evaporator frosts. Integration of the energy delivery curve in Figure 3 showed that the average energy delivery over the 30-minute period from start-up was 0.91 of the steady state value. A five-minute defrost period was found to be adequate for most indoor relative humidities. Therefore, the heat pump will be off for five minutes of every 35 minutes, i.e. the heat pump will be on for a maximum of 0.857 of the time; the remainder of the time the heat pump was off because it was being defrosted or the house did not require heat.

Thomas [Ref. 7] also found that the average power demand by the compressor during the five minutes or so that it takes to reach steady state can be approximated by the steady-state power requirement. Tests of the frosting EAHRHP [Ref. 6] found the compressor power requirement dropped to about 0.84 of its five-minute value after 30 minutes of operation. Therefore, the average compressor power requirement over the 30-minute "on" period of the frosting EAHRHP was assumed to be 0.93 of the steady-state, unfrosted evaporator value. The power requirements for the fans were assumed to be nearly constant over the "on" cycle [Ref. 6].

The average evaporator capacity for the "on" cycle was approximated by the difference between the average condenser capacity and the average compressor power requirement. For the frosting EAHRHP described below, the average evaporator capacity was 0.89 of the steady-state, non-frosted value.

To determine whether a frosting or non-frosting heat pump is best as an exhaust air heat recovery unit, a theoretical analysis was undertaken using a heat pump system design method [Ref. 8] which has proved to be reliable in predicting the performance of real heat pump systems[1]. The prototype design based on the Hooper

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[1] For example, this design method was able to predict the measured, steady-state unfrosted performance of the prototype EAHRHP to within 8%.

and Angus report was used to represent a frosting heat pump and a small non-frosting unit was designed to compare with it. A comparison of the two units and their projected steady-state, non-frosted performance is given in Table 2.

The larger, frosting EAHRHP was given a condenser airflow rate of 283 std l/s (600 scfm) to improve the performance as much as possible and yet provide air of adequate temperature to the house. Airflow for the condenser of the non-frosting unit was adjusted to give the same exit air temperature as with the frosting unit. Airflow over the evaporator was maintained at a constant 70.8 std l/s (150 scfm) (to achieve 0.5 ACH in the representative house chosen) for the whole heating season from September to May (6552 hours).

Static pressure gains required of the fans were calculated from curves supplied by a heat exchanger manufacturer [Ref. 9]. A small allowance was also included for each fan for cabinet and duct losses. Table 3 summarizes the fan requirements for all the devices considered in the report.

## 4.0 COMPARISON OF VENTILATION DEVICES

### 4.1 Basis of Comparison

For this comparison, the energy required to heat and ventilate the house described in Appendix A was determined for three cities in Canada. Vancouver, Montreal, and Saskatoon were chosen to represent the range of climates experienced in this country with heating degree (Celsius) days (18°C base) of 3007, 4471 and 6077 respectively.

In each location, four different heating and ventilating strategies were examined. In all cases, the house relied on a gas or electric furnace as the primary heating device. For the base case, ventilation of the house was accomplished with an exhaust fan (EF) only. In the three other cases, the exhaust fan was replaced by an exhaust air heat recovery device; either an air-to-air heat exchanger (ATAHE) or a frosting or non-frosting exhaust air heat recovery heat pump (EAHRHP).

The costs of the various fuels on a regional basis were supplied by Statistics Canada. Oil heating is not considered as an alternative here. Table 4 shows the cost of energy produced and includes an estimate for typical seasonal furnace efficiency values. As the new generation of high efficiency gas furnaces increases in its market share, the cost of energy from gas will decrease slightly.

The heat loss due to an infiltration rate of 14.2 std l/s (30 scfm) (which corresponds to a ventiltion rate of 0.1 ACH) is given in Table A1. The heat loss due to a ventilation rate of 0.5 ACH (70.8 std l/s (150 scfm)) is, of course, five times this figure. From Table A2 it is apparent that there is not a net heat loss at outdoor temperatures above 15.0°C (59°F) and therefore the energy lost due to ventilation is only for outdoor temperatures of 15.0°C (59°F) and below.

For the ATAHE a constant energy recovery of 60% of the heat lost is assumed [Ref. 4]. The ATAHE and EF are assumed to run continuously for 6552 hours. With the EAHRHP's, the evaporator fans are assumed to operate continuously for the heating season while the condenser fan and compressor operate on demand from the house thermostat. Other control schemes are possible, but this one was chosen to make the analysis of the various devices more uniform and less complicated.

The results of the analysis are summarized in Tables 5 and 6. Appendix B details how the various values were calculated. Estimates of the costs of the devices are given in Table 7. The costs for the ATAHE and EAHRHP were partly based on the Hooper and Angus report [Ref. 5]. The estimated installation costs in this report were considered too low. The Canadian Standards Association (CSA) currently does not permit connection of other heating devices to the inlet of furnaces. Therefore, to ensure adequate ventilation, a separate ventilation duct system is required and the cost of installation is increased for the ATAHE and EAHRHP. The larger, frosting EAHRHP was considered to be about $300 more expensive than the much smaller, non-frosting EAHRHP.

With the extreme uncertainty involved in any prediction of future energy costs and inflation rates, the simple pay back (SPB) ratio of initial cost divided by the cost savings per year was used. The more rigorous (and complex) parameters, such as rate of return on investment or life cycle cost savings, rely on these dubious projections. The use of the SPB as an economic order of merit is easy to apply and is generally understood by the public. Also, it is considered that unless a non-essential system can show a simple pay back period of less than about five years, based on current costs, then it is not economically attractive for the home-owner to purchase.

### 4.2 Frosting Versus Non-Frosting EAHRHP

Table 5 shows that the frosting and non-frosting EHRHP's recover similar amounts of energy from the exhaust airstream. The non-frosting unit tends to recover slightly more in the warmer climate of Vancouver, while in the cold climate of Saskatoon it recovers slightly less. The energy recovery of the frosting unit was degraded by the need for a defrost cycle, and also because its larger delivered energy will cause the unit to cycle on-and-off more often to meet the house load. For a well insulated house the heating demands are modest, especially in the less cold regions; hence the non-frosting unit can supply 98% of the house heating requirements in Vancouver. This is why the smaller, non-frosting unit performed better than the larger, frosting unit in warmer climates.

Referring to Table 6, it is seen that the non-frosting unit offers a lower annual heating bill than the frosting unit in all locations with either gas or electric heating, except for the one case in Saskatoon where electric heating only is used. In all cases the non-frosting unit offers a shorter pay back period than the frosting unit.

The non-frosting unit uses a much smaller compressor; therefore it will weigh less, take up less room and cost less to purchase and install. A smaller compressor also means quieter operation and the unit can operate from a standard 120-Volt wall outlet instead of a 220-Volt outlet (which may have to be specially wired in).

From the above discussions it was concluded that if an EAHRHP was to be a success in the residential market it must be of the non-frosting type. Further analysis in this report proceeds on this basis.

### 4.3 Base Case Versus ATAHE and Non-Frosting EAHRHP

Table 5 shows that the EAHRHP saves considerably more energy than the ATAHE in all the cases considered. As expected, the ATAHE recovers a fairly constant percentage of the energy lost by the exhaust air, with a slight downward trend in warmer climates as the fan energy has more influence. Conversely, the EAHRHP saves more energy as the climate gets colder, because of increased running time; but the percentage of energy saved to energy lost by the exhaust air decreases as the climate gets colder. This is because the EAHRHP recovers a constant amount of energy from the airstream, but energy lost by the airstream increases with decreasing outdoor temperature and so the EAHRHP ends up recovering a smaller percentage of the energy lost. Still, even for the coldest city, Saskatoon, the EAHRHP saves 81% of the energy lost by the exhaust air, compared to the ATAHE, which only saves 60%.

Looking at Table 6, the economic picture is not so straightforward. Because of the low cost of natural gas in Vancouver and Saskatoon (nearly 2.5 times less expensive than electricity, see Table 4), the use of exhaust air energy recovery devices is not economically justified in those areas, and only an exhaust fan need be considered for ventilation. Because the ATAHE displaces the least amount of cheap natural gas heating, it is the least undesirable of the energy recovery devices. With its low energy recovery and relatively high cost, the pay back period for the ATAHE was unacceptably high. Since the EAHRHP uses a significant amount of electricity to supply heating and its COP (2.87) is similar to the cost ratio of electricity to gas, the cost savings are not very large, and hence the pay back periods are too long to make this device attractive for gas heating situations in Vancouver or Saskatoon.

In Montreal, where the gas and electricity costs are more comparable, the EAHRHP can be justified economically, with a pay back period of 4.6 years. With its lower energy recovery, the ATAHE saves less energy cost and has a longer pay back period (6.5 years) in this city.

In the case of electric heating, the EAHRHP saves more energy cost and has a shorter pay back period than the ATAHE in all the three cities considered. With pay back periods of between 2.5 and 4 years, the EAHRHP is economically attractive in all three locations.

The EAHRHP also provides 2.93 kW of useful heating whenever it operates. Table 8 shows that this amount of heating can satisfy a significant proportion of the seasonal heating load for the well insulated house considered, especially in milder climates such as Vancouver where 96% of the heating load can be met by the EAHRHP. This means that in many locations, the EAHRHP could provide the base line heating for the house with electric resistance, gas or some other form of energy providing supplementary heat.

The above analysis has a weakness in that it assumes a constant temperature and relative humidity within the house. It would be expected that the humidity and, to a lesser extent, the temperature within the house will vary during the heating season, e.g. higher humidity in the warmer months and lower humidity in the colder months. This variation in humidity will affect the capacity of the EAHRHP. Should the humidity go too low, it is possible that the non-frosting EAHRHP will begin to frost up and hence a control should be incorporated to detect this condition. The only way of determining the 'true' performance of an EAHRHP is to build several prototypes and to test them in several realistic (i.e. occupied) residential situations.

## 5.0 OTHER APPLICATIONS OF THE EAHRHP

### 5.1 The EAHRHP as a Water Heater

The water heating EAHRHP has an attraction in that the house water heating load is fairly constant over the whole year, whereas a space heating EAHRHP recovers heat only during the heating season. It is possible to build a reversible, space heating EAHRHP which will heat in the winter time and cool in the summer; however, the heating only EAHRHP will be considered here.

From the heat loss/gain figures for the standard house given in Table A2, it is apparent that no space heating is required for outside air temperatures above 15.0°C (59°F) because of internal heat gains. The non-frosting EAHRHP described in Section 3.0 has a heat delivery rate of 2.93 kW. (10,000 Btu/hr). From Table A2, the EAHRHP will therefore operate continuously at outdoor temperatures below 3.9°C (39°F). Between ambient air temperatures of 3.9°C and 15.0°C (39°F and 59°F) the unit will cycle on and off to meet the load. The operating hours for each of the three locations considered were calculated from Tables A1 and A2. Energy saved is the energy recovery rate, $Q_E$ = 2.05 kW (7000 Btu/hr), minus the power to run the fans ($P_{FHP}$ = 0.14 kW) times the number of operating hours. The results are summarized in Table 9.

For the water heating EAHRHP, the water temperature to the condenser and hence the load will vary with the time of day. A typical heating load schedule for a family of four in Canada was presented by the Hooper and Angus report [Ref. 5]. This load schedule resulted in an average demand (including jacket losses) of 50 MJ/day (52,750 Btu/day) for a daily draw of about 200 litres (45 Imp. gallons) of water being heated from 10°C (50°F). The Hooper and Angus report also conducted a finite difference scheme analysis of a typical 180 litre (40 Imp. gallon) hot water tank. They assumed a maximum tank temperature of 65°C (150°F), at which point the EAHRHP would be turned off; and a minimum tank temperature of 50°C (120°F), when an electric resistance heater would be turned on.

Their analysis showed that a heat pump unit which delivers more than 2.5 kW (8500 Btu/hr) will not require additional electric resistance backup, i.e. it can meet the whole heating load (see Fig. 4 taken from Ref. 5). Therefore, it would seem that a unit with the same heat delivery as the space heating EAHRHP would be too large for water heating. The average heat sink fluid temperatures will be much higher for the water heating EAHRHP than the space heating one. Therefore, the COP and heat delivery will be correspondingly lower for the same sized water heating unit. Field and laboratory tests of a water heating heat pump by Oak Ridge National Laboratory [Ref.10] found an average overall COP of 2.0[1] for an average heat delivery of 2.2 kW (7500 Btu/hr), at an inlet air temperature of 21°C (70°F). For the purpose of this comparison, it will be assumed that the water heating EAHRHP is sized to be just large enough to meet the total load and will operate with an average seasonal COP of 2.0.

From Table 9 it is obvious that the space heating EAHRHP will save much more energy than the water heating unit. This is because the average domestic water heating load is much lower than the space heating load in Canadian climates. Therefore, the water heating unit is limited in size and spends much of its life turned off, waiting for the tank temperature to fall to a point where it can turn on again. The cost of a water heating unit would be slightly cheaper than the space heating unit because it would not need as much expensive, site-installed air distribution ductwork. Despite this consideration, it is concluded that of the two systems, the air heating EAHRHP offers a much more economic alternative.

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[1] COP is defined in this case as the energy required to heat the water by electric heat only divided by the energy to heat the same quantity of water with the heat pump.

### 5.2 The EAHRHP as an Airconditioner

By including a reversing valve, a few extra controls and a two-speed fan which can increase the exhaust flow rate to 118 std l/s (250 scfm), the EAHRHP can be made reversible and hence provide dehumidification and cooling in the summer. From the design of the non-frosting unit, a cooling rate of 2.67 kW (9100 Btu/hr) can be achieved, which is sufficient for the living room of most houses, especially in the mild Canadian summer. Unfortunately the size of the EAHRHP precludes it from acting as a central airconditioner for the whole house, but it can take the place of a window-mounted airconditioner.

Currently, mass produced window airconditioners of a nominal 2.9 kW (10,000 Btu/hr) cooling capacity retail for about $500. The components in a window airconditioner and EAHRHP are of similar size and cost; however, the EAHRHP would cost more because of the ductwork, added controls and reversing valve mentioned above (many window airconditioners already have multiple speed fans). Therefore, the EAHRHP is extremely attractive in many cases in that it can pay for itself in heating costs saved in under five years and will also provide airconditioning in the summer; thus saving the cost of purchase of an airconditioner and possibly a dehumidifier as well.

## 6.0 CONCLUSIONS

An analytical analysis of the heating and ventilating requirements of a typical, "comfortable", middle-class, detached house was conducted for three representative locations in Canada. The effect of four different ventilation devices on the cost of heating and ventilating this house was examined for each of the three locations. An exhaust fan (EF), air-to-air heat exchanger (ATAHE), a frosting and a non-frosting exhaust air heat recovery heat pump (EAHRHP) were examined. The results of the analysis are summarized below.

A non-frosting, space heating application is considered to be the most practical for an EAHRHP in a residential situation in cold climates.

The above EAHRHP will save more energy and is more economical than the ATAHE, except where the low cost of natural gas would make both devices impractical. As the climate gets milder, the EAHRHP will recover a greater percentage of the energy lost by the ventilation air. The estimated costs of installing the ATAHE and EAHRHP are similar. Of the cities considered, when compared to an exhaust fan, the EAHRHP can obtain a simple pay back (SPB) of four years or less, where space heating is by electricity. For the two cities where natural gas is available at relatively cheap rates (Vancouver and Saskatoon), the SPB was found to be unacceptably high, and an exhaust fan would be the most economical choice of device. The third city where gas is available, Montreal, has relatively higher gas rates and a SPB of less than five years was found for the EAHRHP. These payback periods were based on current 1984 energy prices and therefore any external factors such as a sharp reduction in the world oil supply will have a substantial effect.

An EAHRHP can also supply a substantial portion of the seasonal house heating requirements. In milder climates, the EAHRHP could even become the base line space heating device; with some other device to provide the balance of the heat required on colder days.

The EAHRHP can be made reversible for very little extra cost and hence supply cooling and dehumidification to the house during summer. This feature makes the EAHRHP even more desirable, as it not only can pay for itself in less than five years in many locations, based on its energy savings from exhaust air, but it can also function as a small airconditioner and a dehumidifier.

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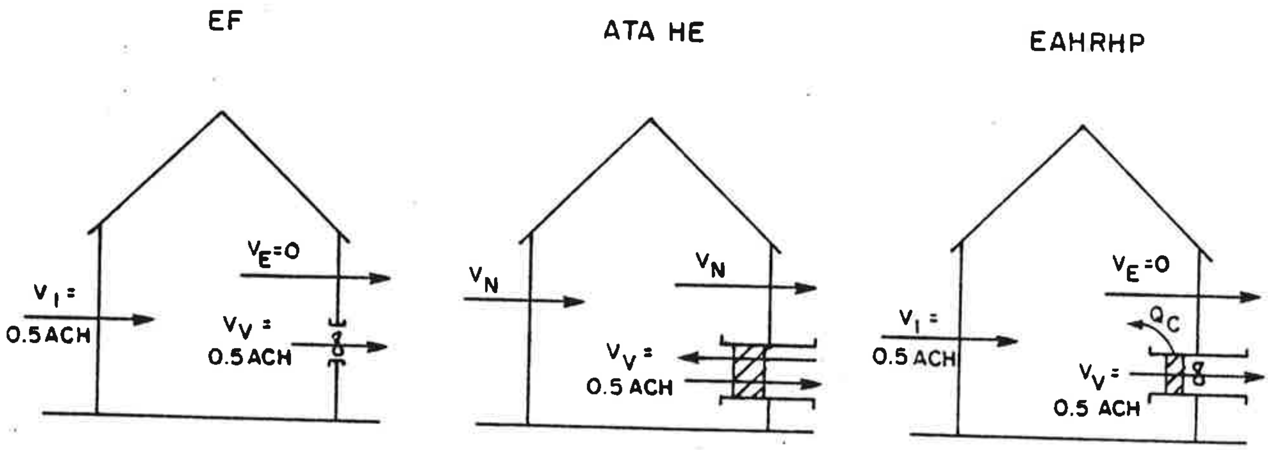


FIG. I SCHEMATIC OF THREE VENTILATION SYSTEMS

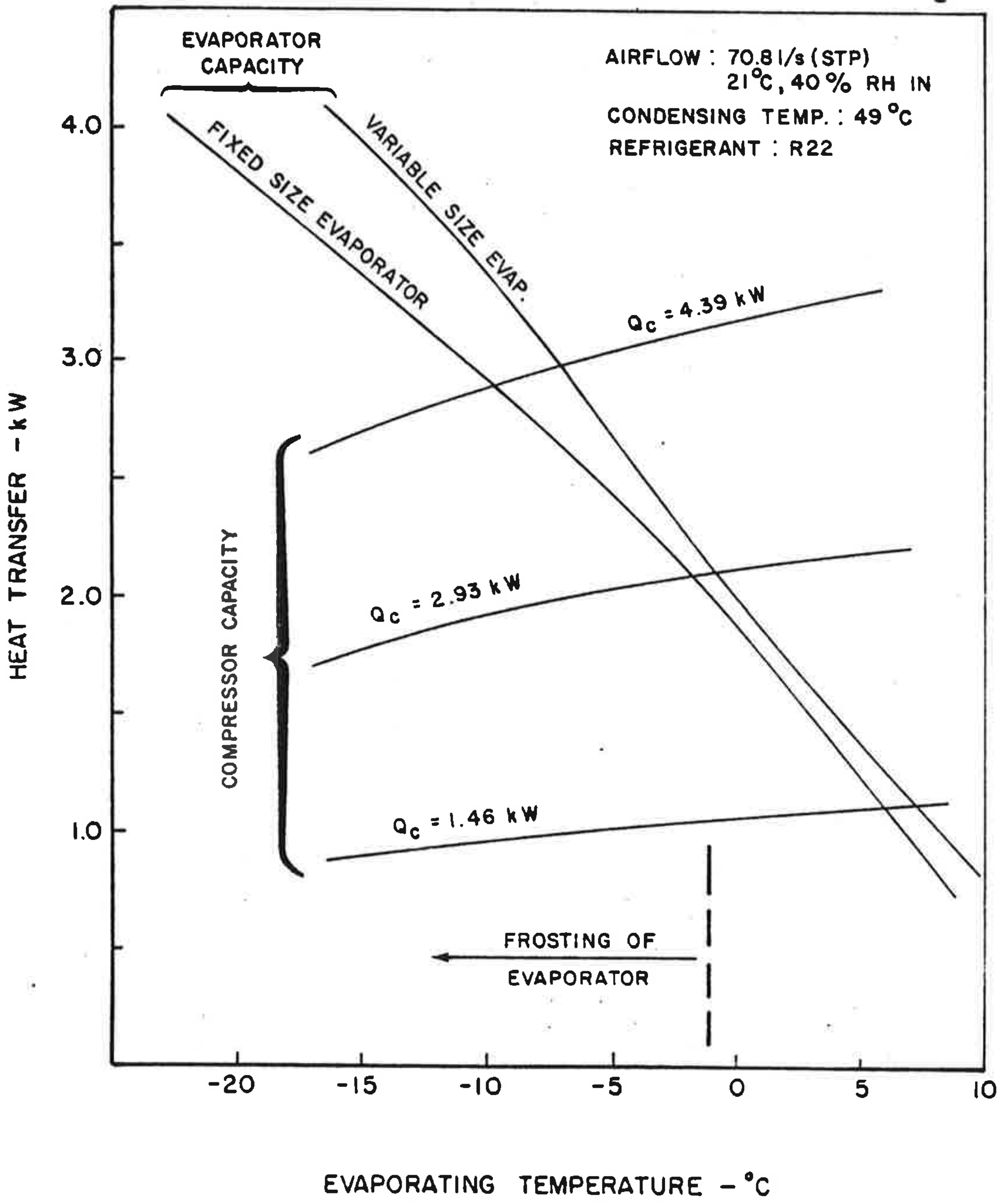


FIG. 2 HEAT RECOVERY VERSUS EVAPORATING TEMPERATURE FOR EXHAUST AIR HEAT RECOVERY HEAT PUMP

$$\frac{\int_0^t Qc\,dt}{\int_0^t Qc_{ss}\,dt} = \begin{array}{l} 0.91 \text{ for } t = 30 \text{ min} \\ 0.78 \text{ for } t = 35 \text{ min} \end{array}$$

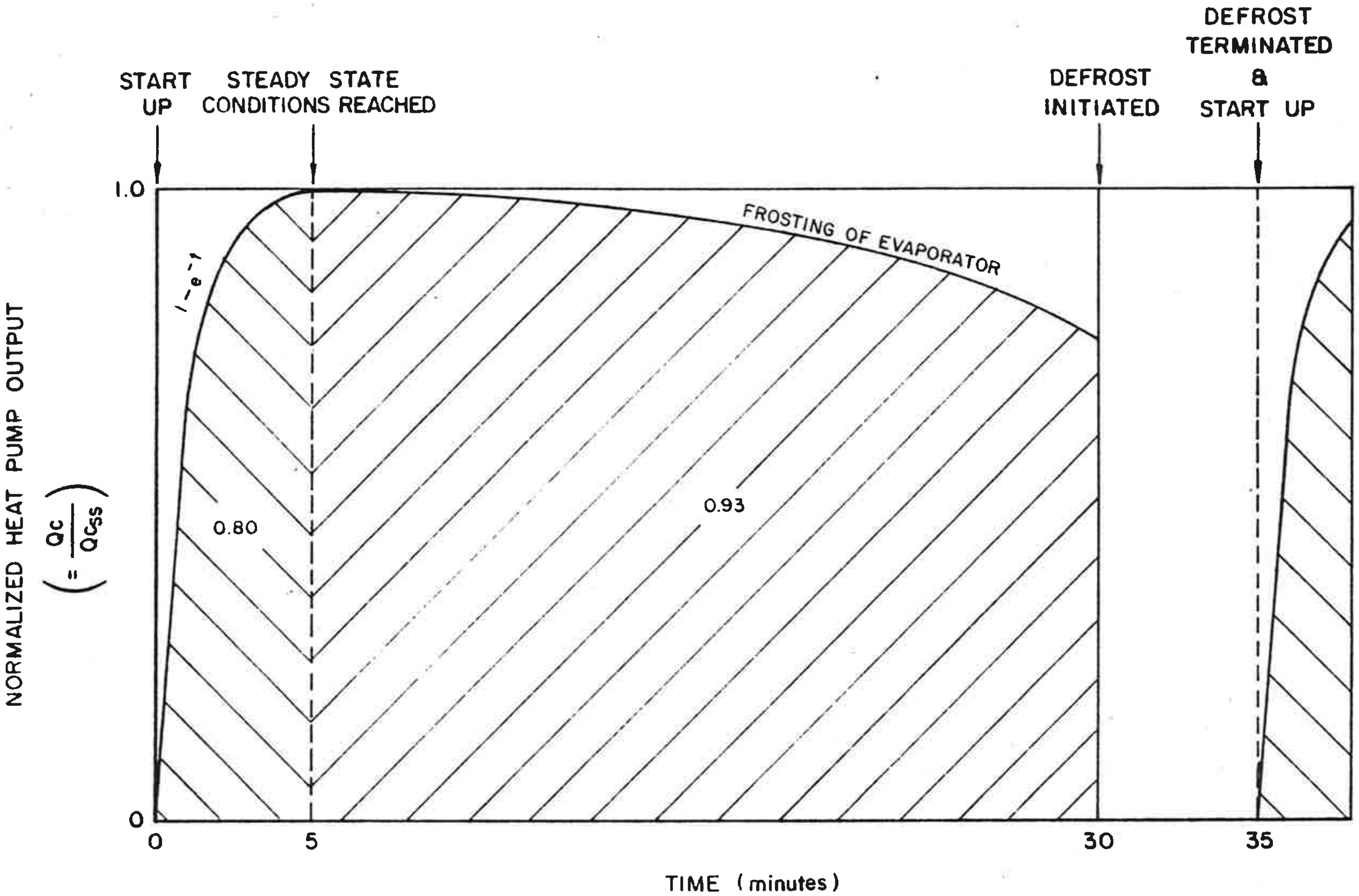


FIG. 3 OUTPUT OF PROTOTYPE HEAT PUMP ON START UP AND FROSTING OF EVAPORATOR

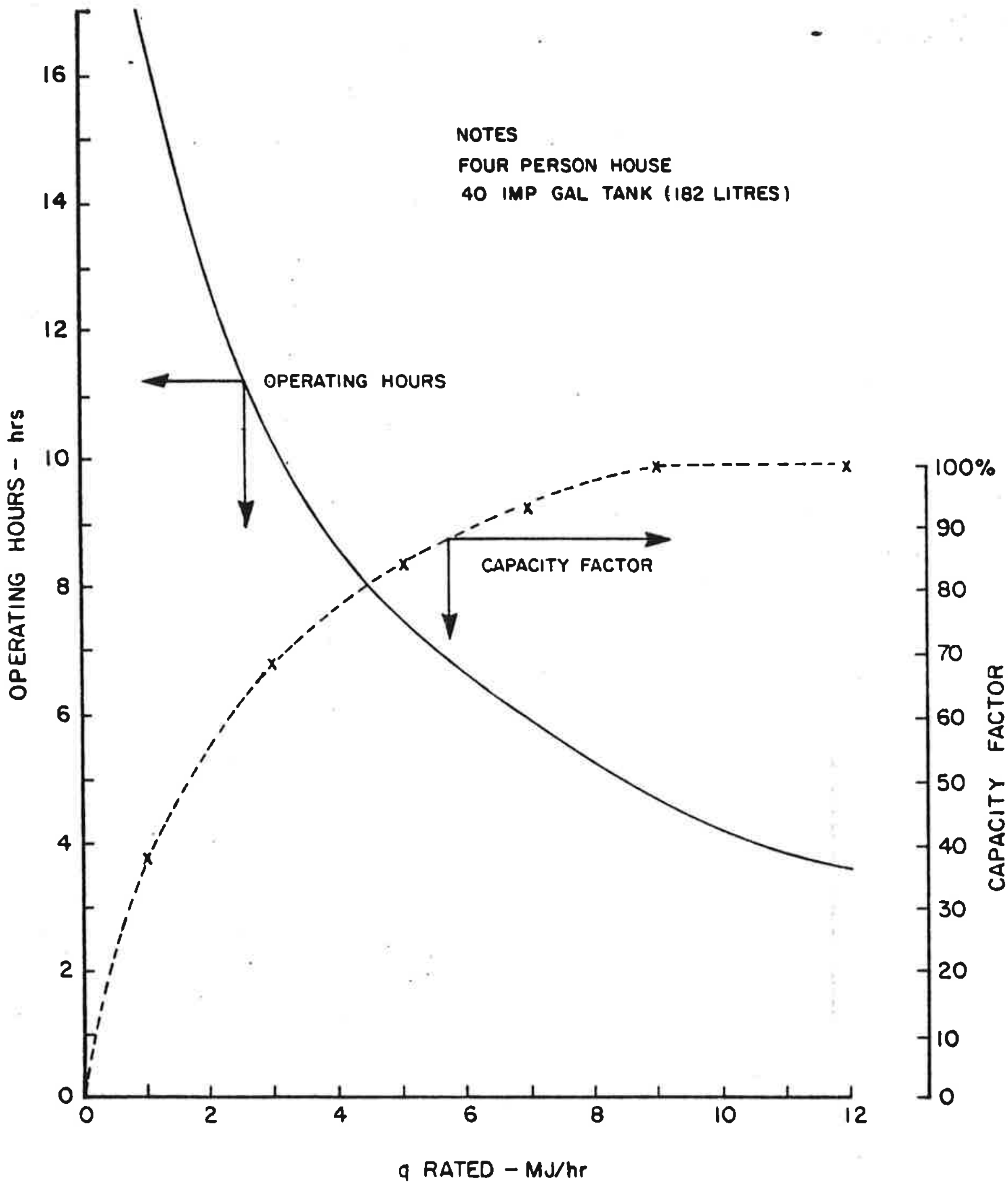


**FIG. 4 DHW EAHRHP OPERATING HOURS**

( Fig. 6.16 from Ref. 3 )

**TABLE 1**

**MECHANICAL VENTILATION REQUIRED FOR A BALANCED VENTILATION SYSTEM TO ACHIEVE 0.5 ACH IN A HOUSE WITH A NATURAL INFILTRATION RATE OF 0.3 ACH**

| $\Delta t$ (°C) | R* | $V_v$ (ACH) |
|---|---|---|
| 10 | 0.41 | .49 |
| 20 | 0.58 | .34 |
| 30 | 0.71 | .28 |
| 40 | 0.82 | .24 |
| 50 | 0.92 | .22 |
| 60 | 1.00 | .20 |

*Correlation for calculating R is based on data for $20 < \Delta t < 50$

**TABLE 2**

**COMPARISON OF FROSTING AND NON-FROSTING EAHRHP FOR SPACE HEATING**

| | | Frosting | Non-frosting |
|---|---|---|---|
| Evaporator: | Nom. tube size (mm) | 10 | 10 |
| | No. of rows | 5 | 4 |
| | No. of fins per metre | 390 | 550 |
| | Frontal area ($m^2$) | 0.052 | 0.186 |
| | Airflow rate (std l/s) | 70.8 | 70.8 |
| | Fan power req'd (W) | 35 | 20 |
| Condenser: | Nom. tube size (mm) | 10 | 10 |
| | No. of rows | 5 | 2 |
| | No. of fins per metre | 470 | 550 |
| | Frontal area ($m^2$) | 0.163 | 0.093 |
| | Airflow rate (std l/s) | 283 | 189 |
| | Fan power req'd (W) | 176 | 120 |
| Compressor: | Weight (kg) | 26.4 | 11.8 |
| | Power supply req'd | 1 ph/240V | 1 ph/120V |
| | Locked rotor current (A) | 77.0 | 45.8 |
| | Power req'd at steady state non-frosted conditions (W) | 1560 | 880 |
| Air temp. to cond. and evap. (°C) | | 21.1 | 21.1 |
| Relative humidity of air to evap. and cond. (%) | | 40 | 40 |
| Air temp. from evaporator (°C) | | -3.6 | 2.8 |
| Air temp. from condenser (°C) | | 34 | 34 |
| Refrigerant evaporating temp. (°C) | | -18.3 | -2.0 |
| Refrigerant condensing temp. (°C) | | 37 | 49 |
| Heat recovered by evaporator (kW) | | 2.93* | 2.05 |
| Heat delivered by condenser (kW) | | 4.49* | 2.93 |
| System COP | | 2.53* | 2.87 |

* Values quoted are for steady state conditions and an evaporator free of frost.

## TABLE 3

## FAN REQUIREMENTS

| Fan | Airflow rate (std l/s) | Pressure loss (Pa) | Power req'd by fan (Watts) |
|---|---|---|---|
| Furnace fan | - | - | 250 |
| Exhaust fan | 70.8 | 25 | 15 |
| ATAHE - delivery | 70.8 | 50 | 35 |
| - exhaust | 70.8 | 50 | 35 |
| Frosting EAHRHP | | | |
| - delivery (condenser) | 283.1 | 62 | 176 |
| - exhaust (evaporator) | 70.8 | 50 | 35 |
| Non-frosting EAHRHP | | | |
| - delivery (condenser) | 188.8 | 62 | 120 |
| - delivery (evaporator) | 70.8 | 25 | 20 |

**TABLE 4**

**ENERGY COSTS IN 1984 DOLLARS**

| | Vancouver B.C. | | Montreal Que. | | Saskatoon Sask. | |
|---|---|---|---|---|---|---|
| | Gas[1] | Elect[2] | Gas | Elect | Gas | Elect |
| Energy Cost ($/GJ) | 6.47 | 15.16 | 9.88 | 12.39 | 5.57 | 14.93 |

1. Gas prices are for July 1984 for a monthly consumption of 283 $m^3$. Energy value of gas is 37.262 $MJ/m^3$. Average furnace efficiency of 70% is assumed.

2. Electricity prices are based on the average bill for a consumption of 750 kW-hrs/month as of April 1984; but includes a price rise of 13% for Saskatchewan effective from Aug. 1, 1984.

3. Information supplied by Statistics Canada.

**TABLE 5**

**COMPARISON OF ENERGY REQUIREMENTS FOR FOUR VENTILATION DEVICES**

| Location & recovery device used | Total heat req'd (GJ) | On time of recovery device (hrs) | On time of furnace (hrs) | Energy recov'd from exhaust (GJ) | Energy del'd by recovery device (GJ) | Elect. energy to recovery device (GJ) | Elect. energy for furnace fan (GJ) |
|---|---|---|---|---|---|---|---|
| **VANCOUVER, B.C.** | | | | | | | |
| Base case | 46.66 | - | 1433 | - | - | - | 1.64* |
| ATAHE | | 6552 | 912 | 16.15 | 16.97 | 1.65 | 0.82 |
| Frosting EAHRHP | | 2997 | 21 | 28.44 | 45.96 | 18.40 | 0.02 |
| Non-frosting EAHRHP | | 4068 | 61 | 30.02 | 44.67 | 15.12 | 0.06 |
| **MONTREAL, Que.** | | | | | | | |
| Base case | 73.59 | - | 1521 | - | - | - | 1.37 |
| ATAHE | | 6552 | 1036 | 22.62 | 23.45 | 1.65 | 0.93 |
| Frosting EAHRHP | | 3821 | 310 | 36.26 | 58.60 | 23.23 | 0.28 |
| Non-frosting EAHRHP | | 4773 | 396 | 35.23 | 54.41 | 17.66 | 0.36 |
| **SASKATOON, Sask.** | | | | | | | |
| Base case | 99.15 | - | 2049 | - | - | - | 1.84 |
| ATAHE | | 6552 | 1428 | 29.23 | 30.06 | 1.65 | 1.29 |
| Frosting EAHRHP | | 4371 | 664 | 41.48 | 67.04 | 26.46 | 0.60 |
| Non-frosting EAHRHP | | 5342 | 837 | 39.42 | 58.66 | 19.70 | 0.75 |

* Includes energy for exhaust fan

**TABLE 6**

**COMPARISON OF ENERGY COSTS ASSOCIATED WITH FOUR VENTILATION DEVICES**

| Location and recovery device used | Gas Furnace | | | Electric Furnace | | |
|---|---|---|---|---|---|---|
| | Cost of heating & ventilation (1984$) | Energy cost savings with recovery device (1984$) | Pay back period for recovery device (yrs) | Cost of heating & ventilation (1984$) | Energy cost savings with recovery device (1984$) | Pay back period for recovery device (yrs) |
| **VANCOUVER, B.C.** | | | | | | |
| Base case | 318.41 | - | - | 712.73 | - | - |
| ATAHE | 224.25 | 94.16 | 14 | 475.10 | 237.63 | 5.9 |
| Frosting EAHRHP | 283.65 | 34.76 | 52 | 289.56 | 423.17 | 4.3 |
| Non-frosting EAHRHP | 242.62 | 75.79 | 20 | 259.39 | 453.34 | 4.0 |
| **MONTREAL, Que.** | | | | | | |
| Base case | 734.84 | - | - | 916.17 | - | - |
| ATAHE | 518.22 | 216.62 | 6.5 | 641.74 | 274.43 | 5.1 |
| Frosting EAHRHP | 436.62 | 298.22 | 6.0 | 473.55 | 442.62 | 4.1 |
| Non-frosting EAHRHP | 409.21 | 325.63 | 4.6 | 456.45 | 459.72 | 3.3 |
| **SASKATOON, Sask.** | | | | | | |
| Base case | 574.86 | - | - | 1485.60 | - | - |
| ATAHE | 421.53 | 153.33 | 9.1 | 1056.22 | 429.38 | 3.3 |
| Frosting EAHRHP | 579.52 | -4.66 | ∞ | 874.45 | 611.15 | 3.0 |
| Non-frosting EAHRHP | 526.67 | 48.19 | 31 | 898.64 | 586.96 | 2.6 |

**TABLE 7**

**ESTIMATED COST OF VENTILATION DEVICES**

| | EF | ATAHE | Frosting EAHRHP | Non-Frosting EAHRHP |
|---|---|---|---|---|
| Capital Cost ($) | 50 | 700 | 1000 | 800 |
| Installation Cost ($) | 100 | 700 | 800 | 700 |
| Total Cost | 150 | 1400 | 1800 | 1500 |

**TABLE 8**

**HEATING PROVIDED BY NON-FROSTING EAHRHP FOR HEATING SEASON**

| | Vancouver B.C. | Montreal Que. | Saskatoon Sask. |
|---|---|---|---|
| Useful heat supplied by EAHRHP (GJ) | 44.67 | 54.41 | 58.66 |
| Heating required by house (GJ) | 46.66 | 73.59 | 99.15 |
| Percentage of house heating load met by EAHRHP | 96 | 74 | 59 |

**TABLE 9**

**COMPARISON OF ENERGY SAVED ANNUALLY BY SPACE AND WATER HEATING EXHAUST AIR HEAT RECOVERY HEAT PUMPS**

| | | Location | | |
|---|---|---|---|---|
| | | Vancouver, B.C. | Montreal, Que. | Saskatoon, Sask. |
| Water Heater | Operating Hours | 1935 | 1935 | 1935 |
| | Energy Saved (GJ) | 9.13 | 9.13 | 9.13 |
| Space Heater | Operating Hours | 4068 | 4773 | 5342 |
| | Evergy Saved (GJ) | 27.97 | 32.82 | 36.73 |

# APPENDIX A

## CHARACTERISTCS OF HOUSE MODEL, WEATHER DATA, ENERGY LOST BY INFILTRATION AND HOUSE HEATING REQUIRED

### Characteristics of House Model

The standard house chosen for comparison of the various ventilation practices is meant to represent a typical detached, middle class, two-storey house with heated basement as found in Canada and the northern U.S.A. Two adults and two children are assumed to occupy the house. References A1 and A2 were used as a guide. Details are given below.

| | |
|---|---|
| Air infiltration for winter conditions | 0.3 air changes/hour |
| Living Area | 149 $m^2$ (1,600 $ft^2$) |
| Basement Area | 60 $m^2$ (650 $ft^2$) |
| Total house volume, for 2.44 m (8 ft) ceilings | 510 $m^3$ (18,000 $ft^3$) |
| Thermal conductance of building skin above grade | 120 W/K (227 Btu/hr-°F) |
| Below grade heat loss (assumed constant) | 1.00 kW (3414 Btu/hr) |
| Gain from appliances, lights, people and sun (assumed constant) | 1.82 kW (6,213 Btu/hr) |
| Loss due to an infiltration rate of 0.3 ACH | 51.3 W/K (97.2 Btu/hr-°F) |

For this house, 0.1 ACH corresponds to a flow rate of 14.1 l/s (30.0 scfm). The heat loss due to an infiltration rate of 0.1 ACH is calculated for three locations in Table A1. The house heating required for 0.3 ACH and 0.5 ACH is calculated in Table A2 for the various temperature bins for each location. Note that the original temperature data was in Imperial units and 5°F temperature bins were used. Conversion of these Fahrenheit temperatures to Celsius resulted in Tables A1, A2 and A3 being somewhat inconsistent because of round-off error. It was therefore decided to keep the Fahrenheit temperatures for the outdoor temperature bins in these tables.

A1. Hooper & Angus — "Development of an Exhaust Air Heat Recovery Heat Pump". Study prepared for the National Research Council of Canada under Department of Supply and Service Contract No. OSX800-00148, April 1982.

A2. Cane, R.L.D. — "A 'Modified' Bin Method for Estimating Annual Heating Requirements of Air Source Heat Pumps". ASHRAE Journal, September 1979, pp.60-63.

TABLE A1

Heat Loss Due to Infiltration Rate of 14.2 l/s (STP)

| Outdoor Temp. Range (°F) | Heat Loss Rate (kW) | Vancouver, B.C. | | Montreal, Que. | | Saskatoon, Sask. | |
|---|---|---|---|---|---|---|---|
| | | No. of Hours | Energy Loss (kW.h) | No. of Hours | Energy Loss (kW.h) | No. of Hours | Energy Loss (kW.h) |
| 65 - 69 | 0.029 | 105 | 3.0 | 227 | 6.5 | 126 | 3.6 |
| 60 - 64 | 0.076 | 289 | 22.0 | 329 | 25.0 | 192 | 14.6 |
| 55 - 59 | 0.124 | 642 | 79.3 | 432 | 53.4 | 287 | 35.4 |
| 50 - 54 | 1.171 | 1042 | 178.2 | 504 | 86.2 | 382 | 65.3 |
| 45 - 49 | 0.219 | 1505 | 328.8 | 519 | 113.4 | 429 | 93.7 |
| 40 - 44 | 0.266 | 1398 | 371.8 | 588 | 156.4 | 468 | 124.5 |
| 35 - 39 | 0.313 | 852 | 267.1 | 717 | 224.7 | 518 | 162.3 |
| 30 - 34 | 0.361 | 457 | 165.0 | 674 | 243.3 | 603 | 217.6 |
| 25 - 29 | 0.408 | 150 | 61.3 | 532 | 217.3 | 527 | 215.2 |
| 20 - 24 | 0.456 | 50 | 22.8 | 452 | 206.1 | 481 | 219.3 |
| 15 - 19 | 0.503 | 24 | 12.1 | 391 | 196.8 | 416 | 209.4 |
| 10 - 14 | 0.551 | 10 | 5.5 | 325 | 179.0 | 355 | 195.6 |
| 5 - 9 | 0.598 | 4 | 2.4 | 262 | 156.8 | 325 | 194.5 |
| 0 - 4 | 0.646 | 2 | 1.3 | 171 | 110.5 | 283 | 182.8 |
| -5 - -1 | 0.693 | | | 111 | 77.0 | 272 | 188.6 |
| -10 - -6 | 0.741 | | | 63 | 46.7 | 233 | 172.6 |
| -15 - -11 | 0.788 | | | 22 | 17.3 | 195 | 153.7 |
| -20 - -16 | 0.836 | | | 7 | 5.9 | 147 | 122.9 |
| -25 - -21 | 0.883 | | | 3 | 2.7 | 94 | 83.0 |
| -30 - -26 | 0.931 | | | 0.5 | 0.5 | 44 | 41.0 |
| -35 - -31 | 0.978 | | | 0.5 | 0.5 | 20 | 19.6 |
| -40 - -36 | 1.026 | | | | | 8 | 8.2 |
| -45 - -41 | 1.073 | | | | | 1 | 1.1 |
| Totals | | 6528 | 1520 | 6329 | 2126 | 6401 | 2725 |

June, July and August not included.

TABLE A2

Heat Loss/Gain for House

| Outdoor Temp. Range (°F) | Heat Loss (.3ACH) (kW) | Heat Loss (.5ACH) (kW) | Below Grade Heat Loss (kW) | Heat Gain (kW) | House Heating Req'd (.3ACH) (kW) | House Heating Req'd (.5ACH) (kW) |
|---|---|---|---|---|---|---|
| 65 – 69 | 0.29 | 0.34 | | | 0 | 0 |
| 60 – 64 | 0.76 | 0.91 | | | 0 | 0 |
| 55 – 59 | 1.24 | 1.48 | | | 0.42 | 0.66 |
| 50 – 54 | 1.71 | 2.06 | | | 0.89 | 1.24 |
| 45 – 49 | 2.19 | 2.63 | | | 1.37 | 1.81 |
| 40 – 44 | 2.66 | 3.20 | | | 1.84 | 2.38 |
| 35 – 39 | 3.14 | 3.77 | | | 2.32 | 2.95 |
| 30 – 34 | 3.62 | 4.34 | | | 2.80 | 3.52 |
| 25 – 29 | 4.09 | 4.91 | | | 3.27 | 4.09 |
| 20 – 24 | 4.57 | 5.48 | | | 3.75 | 4.66 |
| 15 – 19 | 5.04 | 6.05 | 1.00 | 1.82 | 4.22 | 5.23 |
| 10 – 14 | 5.52 | 6.62 | | | 4.70 | 5.80 |
| 5 – 9 | 6.00 | 7.19 | | | 5.18 | 6.37 |
| 0 – 4 | 6.47 | 7.76 | | | 5.65 | 6.94 |
| -5 – -1 | 6.95 | 8.33 | | | 6.13 | 7.51 |
| -10 – -6 | 7.42 | 8.91 | | | 6.60 | 8.09 |
| -15 – -11 | 7.90 | 9.48 | | | 7.08 | 8.66 |
| -20 – -16 | 8.37 | 10.05 | | | 7.55 | 9.23 |
| -25 – -21 | 8.85 | 10.62 | | | 8.03 | 9.80 |
| -30 – -26 | 9.33 | 11.19 | | | 8.51 | 10.37 |
| -35 – -31 | 9.80 | 11.76 | | | 8.98 | 10.94 |
| -40 – -36 | 10.28 | 12.33 | | | 9.46 | 11.51 |
| -45 – -41 | 10.75 | 12.90 | | | 9.93 | 12.08 |

TABLE A3

Total Seasonal Heating Requirements for House

| Temp. Range (°F) | Heating Req'd. (kW) | Vancouver B.C. | | Montreal Que. | | SasKatoon Sask. | |
|---|---|---|---|---|---|---|---|
| | | hrs | GJ | hrs | GJ | hrs | GJ |
| 65 - 69 | 0 | 105 | 0 | 227 | 0 | 126 | 0 |
| 60 - 64 | 0 | 289 | 0 | 329 | 0 | 192 | 0 |
| 55 - 59 | .66 | 642 | 1.53 | 432 | 1.03 | 287 | 0.68 |
| 50 - 54 | 1.24 | 1042 | 4.65 | 504 | 2.25 | 382 | 1.71 |
| 45 - 49 | 1.81 | 1505 | 9.81 | 519 | 3.38 | 429 | 2.80 |
| 40 - 44 | 2.38 | 1398 | 11.98 | 588 | 5.04 | 468 | 4.01 |
| 35 - 39 | 2.95 | 852 | 9.05 | 717 | 7.61 | 518 | 5.50 |
| 30 - 34 | 3.52 | 457 | 5.79 | 674 | 8.54 | 603 | 7.64 |
| 25 - 29 | 4.09 | 150 | 2.21 | 532 | 7.83 | 527 | 7.76 |
| 20 - 24 | 4.66 | 50 | 0.84 | 452 | 7.58 | 481 | 8.07 |
| 15 - 19 | 5.23 | 24 | 0.45 | 391 | 7.36 | 416 | 7.83 |
| 10 - 14 | 5.80 | 10 | 0.21 | 325 | 6.79 | 355 | 7.41 |
| 5 - 9 | 6.37 | 4 | 0.09 | 262 | 6.01 | 325 | 7.45 |
| 0 - 4 | 6.94 | 2 | 0.05 | 171 | 4.27 | 283 | 7.07 |
| -5 - -1 | 7.51 | | | 111 | 3.00 | 272 | 7.35 |
| -10 - -6 | 8.09 | | | 63 | 1.83 | 233 | 6.79 |
| -15 - -11 | 8.66 | | | 22 | 0.69 | 195 | 6.08 |
| -20 - -16 | 9.23 | | | 7 | 0.23 | 147 | 4.88 |
| -25 - -21 | 9.80 | | | 3 | 0.11 | 94 | 3.32 |
| -30 - -26 | 10.37 | | | .5 | 0.02 | 44 | 1.64 |
| -35 - -31 | 10.94 | | | .5 | 0.02 | 20 | 0.79 |
| -40 - -36 | 11.51 | | | | | 8 | 0.33 |
| -45 - -41 | 12.08 | | | | | 1 | 0.04 |
| Total | | 6528 | 46.66 | 6329 | 73.59 | 6401 | 99.15 |

# APPENDIX B

## DESCRIPTION OF CALCULATIONS MADE FOR COMPARISON OF VENTILATION DEVICES

### B1.0 BASE CASE

For the base case it was assumed that the house described in Appendix A was ventilated using an exhaust fan (EF) requiring 15 Watts of power. This fan was run continuously for the 6552 hours of the heating season and thus required 0.354 GJ of electrical energy to maintain the house ventilation rate at 0.5 ACH. Heating was accomplished by either an electric or gas furnace of the same capacity. Typically, gas furnaces come in discrete sizes of 4.4 kW (15,000 Btu/hr), i.e. in 4.4, 8.8, 13.2 and 17.6 kW delivered heating capacity. From Table A3 in Appendix A, the heating requirements for the house in each location were calculated. Furnace size was selected to provide sufficient heat for the coldest temperatures encountered in the particular location, i.e. 8.8 kW in Vancouver and 13.2 kW in Montreal and Saskatoon. It was assumed that a 250 Watt furnace fan was required for both size furnaces and that the fan was "fully loaded". The assumption was also made that all the electrical power to the furnace fan, $P_{ff}$, ended up as heat to the house.

The operating hours for the furnace, $\Delta t_f$, were calculated by:

$$\Delta t_f = \frac{Q_h}{(\dot{Q}_f + P_{ff})} \tag{B1}$$

where $Q_h$ is the total energy (GJ) required to heat the house and $\dot{Q}_f$ is the heating capacity of the furnace (kW). Note that in this and subsequent equations the conversion of units to make the values compatible has not been explicitly detailed for the sake of readability.

The cost of heating and ventilation for the base case was calculated by:

$$D_b = (Q_h - P_{ff} \times \Delta t_f) \times C_f + (P_{ff} \times \Delta t_f + Q_{ef}) \times C_e \tag{B2}$$

where: $Q_{ef}$ = total energy required to operate the exhaust fan (= 0.354 GJ)

$C_f$ = cost of energy for the furnace (1984 dollars per GJ delivered)

$C_e$ = cost of electrical energy (1984 dollars per GJ).

## B2.0 AIR-TO-AIR HEAT EXCHANGER (ATAHE)

It was assumed that the same furnace would be used to heat the house as in the base case. The ATAHE will operate continuously for the heating season; therefore the two 35 Watt fans will consume a total of 1.651 GJ of electrical energy. The heat recovered by the ATAHE, $Q_r$, will be 60% of the heat lost by the exhaust air, $Q_i$, i.e.

$$Q_r = 0.60 \times Q_i \tag{B3}$$

where $Q_i$ is obtained from Table A1 in Appendix A. Note that heat losses for ambient temperatures greater than 15°C were neglected because the house does not require heating at these temperatures.

The heat delivered by the ATAHE, $Q_d$, will be the heat recovered, $Q_r$, plus the electrical energy consumed by the delivery fan, $Q_{df}$, i.e.

$$Q_d = Q_r + Q_{df} \tag{B4}$$

where $Q_{df}$ = 1.651/2 GJ

Since the ATAHE will reduce the heating load of the house, the furnace will have to supply less energy and will operate for a shorter time.

$$\therefore \quad \Delta t_f = \frac{(Q_h - Q_d)}{(\dot{Q}_f + P_{ff})} \tag{B5}$$

Therefore, the total cost of heating and ventilating the house with an ATAHE and regular furnace is given by:

$$D_{HE} = (Q_h - (Q_d + P_{ff} \times \Delta t_f)) \times C_f + (P_{ff} \times \Delta t_f + 2 \times Q_{df}) \times C_e \tag{B6}$$

## B3.0 EXHAUST AIR HEAT RECOVERY HEAT PUMP (EAHRHP)

The EAHRHP will operate from the first stage of the house thermostat and the furnace (same furnace as in the base case) will be operated on the second stage of the thermostat. Once again, the electrical power to the delivery (condenser) fan, $P_{cf}$, will be considered as a contribution to space heating. The average capacity, $Q_d$, of the EAHRHP is given by:

$$\dot{Q}_d = \dot{Q}_c \times K_c + P_{cf} \tag{B7}$$

where $\dot{Q}_c$ is the steady-state capacity of the condenser with a non-frosted evaporator, and $K_c$ is the degradation factor to allow for the reduction in performance as the evaporator frosts. For the frosting EAHRHP described in the report,

$$K_c = 0.91 \tag{B8}$$

and for the non-frosting EAHRHP,

$$K_c = 1.00 \tag{B9}$$

The house will not need heating when ambient temperature are greater than 15.0°C as the heat lost by the house will be less than the internal house gains. The EAHRHP will cycle on and off with the thermostat for ambient temperatures between 15.0°C and the balance point. The balance point is defined as the ambient temperature at which the heat lost from the house is just "balanced" by the output of the heat pump. For the non-frosting unit the balance point was 3.9°C and for the frosting unit it was -4.4°C. For each temperature bin, j, the following relations must be satisfied:

$$\dot{Q}_d \times \Delta t_{on} \leqslant \dot{Q}_j \times \Delta t_j \tag{B10}$$

$$\Delta t_j = \Delta t_{on} + \Delta t_{def} + \Delta t_{off} \tag{B11}$$

$$\frac{\Delta t_{on}}{\Delta t_{on} + \Delta t_{def}} = K_d \tag{B12}$$

where: $\dot{Q}_j$ = the average rate of heat lost from the house for temperature bin j (kW)
$\Delta t_j$ = number of hours in temperature bin j
$\Delta t_{on}$ = 'on' time of the EAHRHP (hrs)
$\Delta t_{def}$ = defrost time of the EAHRHP (hrs)
$\Delta t_{off}$ = 'off' time of the EAHRHP, i.e. accumulated time (hrs) that the first stage of the house thermostat was not activated
$K_d$ = factor to allow for defrost time. (B13)

Therefore:

for $\dot{Q}_j < \dot{Q}_d \times K_d$

$$\Delta t_{on_1} = \sum \frac{\dot{Q}_j}{\dot{Q}_d} \times \Delta t_j \tag{B14}$$

and for $\dot{Q}_j \geqslant \dot{Q}_d \times K_d$

$$\Delta t_{on_2} = \sum \Delta t_j \times K_d \tag{B15}$$

and

$$\Delta t_{on} = \Delta t_{on_1} + \Delta t_{on_2} \tag{16}$$

For the frosting EAHRHP,

$$K_d = 0.857 \tag{B17}$$

For the non-frosting EAHRHP

$$\Delta t_{def} = 0 \tag{B18}$$

$$\text{and} \quad K_d = 1.00 \tag{B19}$$

The average power required by the compressor is given by:

$$P_c = (\dot{Q}_c - \dot{Q}_e) \times K_p \tag{B20}$$

where $K_p$ is the factor which allows for the reduction in power requirement as the evaporator frosts. For the frosting unit considered in this report,

$$K_p = 0.93 \tag{B21}$$

and for the non-frosting unit,

$$K_p = 1.00 \tag{B22}$$

The time during which the furnace will operate is given by:

$$\Delta t_f = \frac{Q_h - \dot{Q}_d \times \Delta t_{on}}{(\dot{Q}_f + P_{ff})} \tag{B23}$$

Therefore, the total cost of heating and ventilating the house with an EAHRHP and regular furnace is given by:

$$D_{HP} = (Q_h - (\dot{Q}_d \times \Delta t_{on} + P_{ff} \times \Delta t_f))\, C_f + (P_{ff} \times \Delta t_f + Q_{vf} + (P_c + P_{cf}) \times \Delta t_{on})\, C_e \tag{B24}$$

where $Q_{vf} = P_{vf} \times \Delta t_s$ (B25)

$P_{vf}$ = power required by evaporator fan

$\Delta t_s$ = number of hours in the heating season (6552).

Energy recovery rate from the exhaust air stream, $\dot{Q}_r$, is given by:

$$\dot{Q}_r = \dot{Q}_c \times K_c - P_c \tag{B26}$$

## B4.0 FANS

Each device will have one or two fans. The power to operate these fans will be a direct charge against the cost of operating the device. For these small, inefficient fans and motors an overall electricity to air efficiency, $E_F$ of 10% will be assumed. The power required by each fan (Watts) is calculated using the equation B27; where $\Delta p$ is the pressure differential supplied by the fan to move the air ($P_a$) and $V_v$ is the airflow rate (l/s).

$$P_F = \frac{\Delta p \ V_v}{E_F} \times \frac{1}{1000} \tag{B27}$$

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