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# Assessment of Residential Exhaust-Air Heat Pump Applications in the United States

by

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ABSTRACT

The objective is to provide a technical and economic assessment of residential exhaust-air heat pumps for water heating with option for space heating. Compact one-cabinet heat-pump units containing a hot water tank with refrigerant condenser, exhaust fan, refrigerant evaporator, refrigeration compressor and controls have recently become available on the U.S. market. The emphasis is on U.S. applications with three representative climates: moderate (Northwest), cold (Midwest), and hot and humid (Southeast). The approach is to simulate detailed system performance for a full year of two typical houses equipped with the heat pump. TRNSYS, a transient building system simulation package, is used allowing detailed simulation of heat pump (and other HVAC equipment) operation including control action (on, off and hysteresis effects). Ventilation and energy performance of the heat-pump houses is compared to the same houses equipped with air-to-air heat exchangers, and to naturally ventilated houses. Exhaust-air heat pumps are found to provide a nearly constant ventilation rate throughout the year as opposed to the natural ventilation rate that is found to be highly variable. Energy savings realized with the heat pumps are greater than those obtained with heat exchangers. However, heat-pump installation and maintenance costs are also higher than those for the heat exchanger resulting in broadly similar economics for the two systems. Both are cost competitive with natural ventilation in cold climates, and clearly uneconomical in hot and humid climates with present electric rate predictions. In the moderate climate of Portland, both heat pumps and heat exchangers are roughly economically neutral. Albeit neutral or slightly negative in economic terms, heat pumps offer the certainty of a constant ventilation rate throughout the year, a benefit for indoor air quality that is disregarded by the economic analysis.

## INTRODUCTION

Indoor air pollution such as radioactive radon gas from the soil or organic compounds from building materials, has recently become a new driving force for efforts to improve the ventilation of residential buildings. Also, the popular energy conservation measure of "house tightening", i.e., the elimination of cracks and other sources of natural infiltration, has significantly reduced ventilation and has thus led to an undesirable development with regards to indoor air quality. The trade-off between adequate ventilation and energy efficiency is the underlying broad topic of this paper. Specifically, the topic here is electric exhaust-air heat pumps that provide ventilation in a controlled manner together with energy recovery. The emphasis in this paper is on heat pumps that heat tap water because such heat pumps are technically simpler than heat pumps whose main function is space heating. Also, exhaust air-to-water heat pumps are relatively new to the U.S.; for example, the authors are aware of only one U.S. manufacturer, whereas the Scandinavian countries have at least a dozen manufacturers, and Sweden alone is reported to have had 18,000 such heat pump installations by the end of 1983.[1]

The exhaust-air heat pump, shown schematically in Fig 1 and abbreviated HP in this work, transfers heat from an air stream typically driven by a single exhaust fan to the domestic hot water by means of a standard refrigeration plant. An important enhancement of the basic heat pump is a hydronic fan convector or "fan coil" heated by a pump-around tap water loop that allows space heating in addition to tap water heating. This heat pump/fan coil combination (abbreviated HP+FC) is important in this study because the fan coil acts as a "swing" sink for the energy extracted from the exhaust air, and thus allows more heat recovery. Without the fan coil, the heat pump will switch off when the water in the tank reaches set-point temperature, hence no energy recovery can occur when this single heat sink is saturated. This limitation is, however, always present during the cooling season when the fan coil is off. Heat pumps that also provide cooling in summer are not studied in the present work although there are obvious energy conservation benefits from such an enhancement of the basic heat pump system. Complex dual heating systems such as a heat pump coupled with a supplementary gas heater are also excluded from the present study.

An alternative technical solution to the problem of energy-efficient ventilation that competes with the exhaust-air heat pump is the exhaust-air to fresh-air heat exchanger (abbreviated HX) that has been reported on previously.[2] Comparing the two approaches briefly, the heat pump has the main advantage of providing a large temperature difference for driving the energy recovery and not being limited to simply the indoor-outdoor temperature difference. Consequently, the heat pump can be expected to perform better when this temperature difference is small, such as in moderate climates and during the summer months. In addition, the requirement of balanced supply and exhaust flows through the heat exchanger is a disadvantage because the balancing is a difficult operation particularly in leaky houses. (In cold climates is is desirable that house pressure be neutral or slightly negative to avoid moisture buildup in walls). In comparison, the heat pump with its single exhaust fan provides controlled ventilation without such balancing. Instead, the exhaust-air heat pump depressurizes the house somewhat leading to pressure-induced flow of fresh air through the building envelope. This mechanically induced depressurization also reduces the effect of weather on the ventilation rate. However, the depressurization may also have negative consequences in some houses in the form of an increased rate of radon entry from the soil.

The specific objective of this study is to provide both a technical and an economic analysis of the exhaust-air heat pump. To meet this objective, a TRNSYS system simulation approach[3] is taken where the ventilation and energy performance of two different electrically heated houses are calculated for an entire year. One house is a standard (STD) house built to current standard practice, and the other a model conservation superinsulated (MCS) house. Three locations are considered: Portland, Oregon representing a moderate climate, Minneapolis, Minnesota representing a cold climate and Memphis, Tennessee representing a hot and humid climate. Portland and Memphis are selected because all-electric houses are common both in the Pacific Northwest and in the area served by the Tennessee Valley Authority. Minneapolis is included simply to include a cold climate in the study. In another study[4] of exhaust-air heat pumps, it is shown that this ventilation technology does not provide primary fuel savings (assuming a 33% utility conversion efficiency) relative to naturally ventilated houses with gas space and water heating. Consequently, the potential for saving primary energy with electric heat pumps lies in regions of the U.S. where inexpensive electricity is available such as areas of abundant hydropower.

For the sake of comparing the heat pump to alternative ventilation strategies, the same houses are simulated using both natural ventilation (NV) with electric space and water heating, and ventilation by an air-to-air heat exchanger.

The heat pump characteristics (and characteristics of other HVAC equipment) are assumed based on published data, and are the same regardless of house type and climate. This implies that the heat pump is not necessarily optimal in terms of size or other characteristics for any particular application. However, the modeled applications are considered reasonable and a sensitivity study is included as the last section of this paper to strengthen this claim.

The economic assessment is based on best estimates of costs both for the installed equipment and the power saved. Two economic parameters are used; one is the cost of conserved energy that provides an economic comparison of the three mechanical ventilation strategies HP, HP+FC, and HX as energy conservation measures, and the other parameter is the net present benefit to the homeowner. Each mechanical ventilation strategy is compared to natural ventilation.

## SIMULATED SYSTEM

TRNSYS ("Transient System Simulation Program")[3] is a commercially available program for simulating the energy performance of buildings. The program includes a central differential-equation processor but is otherwise modular in structure; modules representing physical components, e.g. the heat pump, are interconnected to make up a building system much as they are in an actual building.

The primary modules employed in this study include: 1) a building module, 2) an exhaust-air heat-pump module, 3) a fan-coil module, 4) an air-to-air heat exchanger module, 5) thermostatic control modules for indoor-space and hot-water temperature, and 6) a weather data module. The building module is further divided into submodules representing the building envelope, an indoor space heater, an air conditioner, internal heat gains, an infiltration/ventilation model, and a domestic hot water heater.

The building envelope specifications for the two houses used in this study are listed in Table 1. The MCS (Model Conservation Superinsulated) house is identical to the STD (standard) house except for the following insulation characteristics: Ceiling R-38 vs. R-19 in the STD house, walls R-28 vs. R-11 in the STD house, floor R-31 vs. R-13 in the STD house and triple-glazed windows vs. double-glazed windows in the STD house.

An important building envelope parameter not listed in Table 1 is the effective leakage area that correlates the rate of air flow through the building envelope to the pressure difference, [5]

$$Q = A \cdot \sqrt{\frac{2 \cdot \Delta P}{\rho}} \qquad (1)$$

where Q - volumetric infiltration rate

A - effective leakage area

$\Delta P$ - pressure difference (indoor-outdoor)

$\rho$ - air density

Thus, the effective leakage area combines into one parameter the true leakage area of the building and the effective friction factor of all flow paths. Determination of A by a fan pressurization technique[5] and prediction of $\Delta P$ allow prediction of Q. In the LBL infiltration model[5], $\Delta P$ is correlated to indoor-outdoor temperature difference, wind speed, and a terrain/shielding factor. In the exhaust-air heat pump case, the infiltration and the exhaust are in series and therefore the total ventilation rate is given by $\sqrt{(Q^2_{infiltration} + Q^2_{exhaust})}$. In the pressure balanced case using the air-to-air heat exchanger, the mechanical and the infiltration component are in parallel and therefore add simply as $(Q_{infiltration} + Q_{heat\ exchanger})$.

For the simulations reported in this paper, the effective leakage area is adjusted to achieve the desired total ventilation rate. This implies different effective leakage areas for different climates. For example, in the NV houses, an effective leakage area of 0.08 $m^2$ in Portland, 0.06 $m^2$ in Minneapolis and 0.07 $m^2$ in Memphis give the same yearly mean ventilation rate of 0.7 ach ($200 m^3/hr$).

In an attempt to model the effect occupants may have on the ventilation rate, a window opening strategy is used in some of the simulations. This strategy is simply temperature based according to the following equation:

$$A_o = \begin{Bmatrix} A_w \cdot \dfrac{Tindoor - 22.4}{24.9 - 22.4} & \text{if } 22.4 < Tindoor \leq 24.9 \text{ and } Tindoor > Toutdoor \\ O & otherwise \end{Bmatrix} \quad (2)$$

where:

$A_O$ = Additional effective leakage area

$A_W$ = $1.2 m^2$ (max additional effective leakage area)

$A_O$ is added to the "standard" effective leakage area, and the total area is then used in the LBL infiltration model. This window opening model is a gross simplification because the distributed leakage area of the infiltration model is quite different from the lumped leakage area of open windows. In addition, window opening behavior may differ substantially from that implied by Eq. (2). However, a detailed and accurate model of window opening is not required for this analysis.

Another effect related to the occupants in the building is the time dependence of appliance and hot water use. The appliances including lights have an average power input of 420W, but the load is a strong function of the time of the day. The same is true for the sensible load from the four occupants themselves who are assumed to be away between 9:00 and 17:00. The assumed schedule of hot water usage is similar to the National Solar Data Network schedule[6] that has flow rates constant over 1-hour periods. Here however, higher flow rates and more intermittent flow (maximum flow rate of 8 l/min for 7.5 min starting at 8:00) are used. Both the water and the appliance use schedules are the same for every day of the year. The cold water is assumed to be at 10°C on January 1, increase linearly to 20°C on July 1, and decrease again to 10°C on December 31.

Table 1 also gives detailed information of the assumed HVAC system for the houses. Performance data for the air conditioner is based on a Carrier product bulletin,[7] while data for the heat pump is taken from a Flakt AB product bulletin which contained the only detailed performance curves that could be obtained.[8] The correlation for the heat pump coefficient of performance (COP) is redrawn in Fig. 2 for easy reference. This figure shows the advantage of high COP at low condenser temperatures. As a consequence of the strong effect of condenser temperature, the degree of stratification in the hot water tank (Fig. 1) is an important performance parameter because without tank stratification the condenser water inlet temperature is equal to the fully-mixed water temperature in the tank, or typically 52-54°C. However, the water tank is assumed to be stratified in accordance with a seven-zone TRNSYS model[3]. Such a model allows transient temperature differences up to 25° to develop between top and bottom. With reference to Fig. 1, cool water from the lowest level in the tank is pumped to the heat-pump condenser where it is heated in accordance with the heat-pump performance data[8], and returned to tank zone 4 which is just below the auxiliary electric heating element. Conforming to this physical arrangement the heat-pump compressor on/off thermostat sensor is located in the bottom of the tank and the auxiliary heater on/off thermostat sensor is located in tank zone 3 near the top of the tank. The two thermostat setpoints are offset by 2°C (Table 2) so that the auxiliary heater comes on only when the zone-3 temperature is 2°C below the heat-pump setpoint temperature.

In the "Heat-pump with Fan Coil" case the water tank has a second water pump-around loop for the fan coil in addition to the heat-pump condenser loop (Fig. 1). The fan-coil water is pumped from the top of the tank to the fan coil where it heats a forced stream of indoor air (heat exchange effectiveness = 65%), and returns to the bottom of the tank. The pump and the air fan are controlled by the master space-heating thermostat that also controls the auxiliary electric space heater and the air conditioner. The fan coil control also includes a low-water temperature override that prevents the fan coil from running when the auxiliary water heater is on because the objective is to use only excess heat-pump capacity for space heating.

The master space thermostat setpoints for the fan-coil and for the electric heater are offset by 0.75°C (Table 1) in order to use the fan coil fully before resistance heating comes on. The third setpoint in the master thermostat is 25.75°C for air conditioning with a deadband of 0.75°C, i.e., air conditioning is always on above 25.75°C, always off under 25°C, and on or off in between depending on the state before entering the deadband region (hysteresis). It should be noted that all <u>heater</u> deadbands referred to in Table 1 are <u>above</u> the given setpoints in contrast to the deadband for the air conditioner. It should also be noted that the heating setpoints for the two stages of heating (fan coil and electric resistance) are set back 5°C at night (23:00 -7:00).

Space and water heating thermostat settings are adjusted for the comparison cases HP (no fan coil), HX and NV so as to keep the space and the hot water at the same temperatures. In the HP case, for example, the setpoint for the electric space heater is raised to 21°C in order to keep the space heated to the same temperature as in the HP+FC case. The HX case also requires special modeling of the necessary defrosting of the heat transfer surface. It is assumed that for 20% of the time when outdoor temperatures are below -5°C, the heat exchanger is in defrost mode which is simply modeled as zero heat exchanger efficiency. In contrast, the HP cases do not involve any defrosting since the outlet air temperature is typically 5°C.

For accurate simulation of thermostat on/off times, a time step of 7.5 minutes is used in all TRNSYS simulations. However, typical meteorological year (TMY)[9] weather data for solar radiation, outdoor temperature, humidity and wind speed are used in 1-hour increments throughout the year. Despite the relatively long time step, calculated results for the instantaneous ventilation rate and power usage are considered to be of adequate accuracy. Integrated results for the entire 1-year simulation are believed to be particularly accurate. In contrast, an accurate measure of peak power demand during the day is more difficult to obtain from the simulations partially due to the time step used, and partially due to the fact that some HVAC equipment used can be considered over-rated.

## VENTILATION RESULTS

Typical ventilation results from a naturally ventilated house are shown in Fig. 3. This figure shows the yearly time distribution of ventilation rates for the Portland STD/NV house with an assumed effective leakage area of 0.05 $m^2$. The yearly mean and median ventilation rate are both approximately 0.45 ach (air changes per hour). However, the more important result is the wide distribution of ventilation rates; they range from a mere 0.1 ach to 1.5 ach. A useful parameter for the front end of the distribution is total time in a year with ventilation rates under 0.25 ach. This parameter is about 600 hours for the distribution in Fig. 3. This is clearly undesirable because the recommended minimum ventilation rate is considered to be in the range of 0.35 - 0.5 ach[10,11].

When the window opening strategy is incorporated into the model according to Eq. (2), the distribution of ventilation rates shown in Fig. 4 is obtained. Here, the median rate is unaltered at 0.45 ach while the mean increases to 0.78 ach. Interestingly, the front end of the distribution is almost unchanged; the total time with ventilation rates under 0.25 ach is 530 hrs, an insignificant improvement over the case with closed windows. Table 2 shows that the window opening strategy fails to substantially reduce the amount of time with low ventilation in all climates: the Memphis houses, for example, have a ventilation rate less than 0.25 ach for approximately five weeks of the year. The main reason for the low ventilation rates is periods of warm weather with no wind when windows are closed according to Eq. (2) (and air conditioning is on), and when the natural infiltration rate is low.

A more effective window opening strategy might reduce the periods of low ventilation rates, but less house "tightening" (a higher year-round leakage area) is a more common solution in the natural ventilation cases. The effect of an increased effective leakage area on the ventilation rate distributions is to shift the entire distribution toward higher ventilation rates. The right-hand side of Table 2 shows results from houses with roughly 50% greater effective leakage areas yielding yearly median ventilation rates of approximately 0.7 ach (<u>mean</u> ventilation rate for the houses with window opening ~1.2 ach for the full year, and ~0.75 ach for the winter months). The results of Table 2 show that in Portland and Minneapolis there is a factor of 5-10 reduction in time with a ventilation rate less than 0.25 ach. In Memphis however, the time with a low ventilation rate decreases by only a factor of 2 to about 2 weeks of the year.

In contrast to the broad distribution obtained with natural ventilation, Figure 5 shows the sharp distributions of ventilation rates that are obtained through mechanical ventilation. Figure 5/upper portion shows the result for the exhaust-air heat pump (HP and HP+FC). Here, the whole distribution is closely centered around the mean of 0.5 ach, and the ventilation rate never drops below 0.45 ach as long as the exhaust fan is on (in Fig. 5 the exhaust fan is on all year). Figure 5/lower portion shows the ventilation rate distribution that is obtained with the balanced air-to-air heat exchanger (HX case). This distribution is in essence the same as the one obtained with the exhaust-air heat pump. However, with the same assumed air flow rate through the heat pump and the heat exchanger (125 $m^3$/hr), the HX house has a lower effective leakage area (0.01 $m^2$) than the HP house (0.025 $m^2$) for the same total ventilation rate. The reason for this is the interaction between the mechanically induced ventilation and the natural ventilation discussed above (HP in series with natural ventilation, HX in parallel).

The exhaust-air ventilation strategy utilized by the heat pump does not involve supply and exhaust air balancing as does the HX ventilation strategy; rather, depressurization of the house induces flow through the natural leakage area, or through supply vents installed specifically for this purpose. The exact amount of depressurization is strongly dependent on the leakage area (inversely proportional to the square of the area, and directly proportional to the square of the exhaust flow rate). For the houses considered in this work with an effective leakage area of 0.025 $m^2$ (no supply vents) and an exhaust-air flow rate of 125 $m^3$/hr, the depressurization is 1.2 Pa. Such a depressurization may be of significance in increasing soil gas entry rates in houses with basements. Soil gas is the dominant source of indoor radon in the U.S.[12, 13] and consequently, exhaust-air heat pumps may have a negative effect with respect to the rate of radon entry into buildings. This aspect is not discussed further in the present paper but is currently the subject of study at the Lawrence Berkeley Laboratory.

Figure 5 shows that both the exhaust-air heat pump and the air-to-air heat exchanger provide essentially a constant ventilation rate throughout the year. The present work does not address the issue of <u>spatial</u> distribution of ventilation rate which can be referred to as "local ventilation efficiency". Typically, the exhaust air is drawn from only a few locations in the house such as the bathrooms and the kitchen. This, no doubt, is advantageous for removal of water vapor from bathrooms and cooking-generated pollutants form the kitchen, but a question is: How well ventilated is a side bedroom with closed door? To answer questions of this nature further experimental studies involving tracer gases will be required.

## ENERGY RESULTS

For the energy and economics sections of this paper only the 0.7 ach natural ventilation (NV) case is considered. The reason for this approach is two-fold: (1) there is evidence that typical naturally ventilated houses have roughly this ventilation rate[14] and (2) house tightening down to 0.5-ach level is not recommended for natural ventilation because of the resulting low ventilation during certain times of the year. It should be noted that, in this context, "house tightening" is not considered a retrofit but rather, a new construction practice. In contrast, the mechanically ventilated houses that do not suffer from extremely low-ventilation rates can advantageously be built to tighter "leakage standards". For comparison to the 0.7-ach NV houses, a ventilation rate of 0.5 ach is selected in the mechanically ventilated houses; such a ventilation rate is a typical goal or "target value" in energy-efficient, mechanically-ventilated residences. A consequence of lowering the average ventilation rate from 0.7 ach for the NV houses to 0.5 ach for the HP, HP+FC and HX houses is implicit energy savings for the mechanically ventilated cases. These yearly energy savings are approximately 1,700 kwh in Portland, 2,700 kwh in Minneapolis and 1,000 kwh in Memphis. The window-opening strategy is used only for the NV houses, where it lowers the air-conditioning power requirement by 100 - 300 kilowatt hours.

All energy results are summarized in Fig. 6. An immediate observation is the significant energy savings in space heating that is realized when going from the STD (standard) house to the MCS (model conservation superinsulated) house. However, the objective here is not to investigate the effects of insulation, but rather to compare the exhaust-air heat pump houses to both the naturally ventilated house and the house with the air-to-air heat exchanger. The main result of this comparison is that the mechanical ventilation strategies save energy over the natural ventilation case in the following order: Heat Exchanger < Heat Pump < Heat Pump with Fan Coil regardless of climate and house type. However, the amount of energy saved is dependent on climate (and somewhat on house type): For the Portland STD house, the heat exchanger saves 3,100 kwh over the NV case, the heat pump 4,200 kwh and the heat pump with fan coil 5,400 kwh, also relative to the NV case. The energy savings obtained with the heat pumps in the other houses are listed in Table 3 for easy reference. A first observation (Table 3) is that less energy is saved for the MCS house than for the STD house particularly in the HP+FC case. A second observation (Fig. 6) is that the major part of the saved energy, even in the fan-coil case is power saved for electric water heating. Third, in the HP+FC cases significant electric space heating demands remain in all climates. This raises the question of increased heat-pump capacity but a discussion of this and other sensitivity issues are deferred to a subsequent section. Besides reduced power demand for water heating, the heat pump cases benefit from the previously discussed reduced average ventilation (1,700 kwh in Portland, 2,700 kwh in Minneapolis and 1,000 kwh in Memphis).

Table 3 clearly shows that the Memphis houses have the smallest energy savings, and the cause lies in the low heating load and the fact that neither the heat exchanger nor the heat pump reduces the air conditioning load relative to the naturally ventilated house. This is because the mechanical ventilation strategies provide ventilation in hot weather when the naturally ventilated case shows a very low infiltration rate. In addition, the heat pump increases the air conditioning load because the convective heat loss from the heat pump compressor is assumed to enter the living space. The conclusion is that heat pumps, and the type of heat exchangers considered here, are not well suited for hot and humid climates in the sense that energy savings are low. Heat pumps with reversed air flow direction in summer would probably do better. Such heat pumps would provide cooling in summer, but would also require more elaborate ductwork since the cool air would have to be supplied to living spaces other than the bathrooms and the kitchen. Reversible flow is not included in the present study but is included in another investigation.[4]

## ECONOMIC RESULTS

For a technology assessment, economic aspects are as important as the technical aspects such as the predicted energy savings of the previous section. The task in this section is to combine information on energy savings with cost information.

The input assumptions for the cost analysis are given in Table 4 representing the best judgment of the authors. Installed cost can be expected to vary from city to city in the U.S., but the quoted figures are considered good estimates for the three cities of this study. The only publicly available document used for some of the cost data is a survey of 366 houses in the Pacific Northwest.[15] However, manufacturers, builders and retailers were consulted in order to develop realistic cost estimates. The installed cost of the heat pump of $1,300 (Table 4) is equal to the purchase price, plus the cost of the installation of the ductwork, minus the cost of a conventional hot water tank. Similarly, the installed cost of the heat exchanger is $1,040. The cost of constructing a more air-tight house, as required for the mechanical ventilation options, is estimated to be $540. A complete replacement of the heat pump is assumed at the end of 10 years because the service life of a heat pump is expected to be similar to the service life of a standard water heater, which is approximately 10 years. The only other maintenance cost assumed is the cost for parts and labor to replace the fans every five years (HP+FC, HP and HX).

The first economic parameter is the cost of conserved energy CCE that is of interest to energy conservationists in general, and is defined as follows:

$$CCE = \left\{ \left[ C + \sum_{i=1}^{N} \frac{Mi + Ri}{(1+d)^i} \right] \left[ \frac{d}{1-(1+d)^{-N}} \right] \right\} / \Delta E \qquad (3)$$

where:

CCE - cost of conserved energy \$/kwh

C - initial cost of installed heat pump (and fan coil) or (heat exchanger) system, plus cost of house tightening

Mi - maintenance cost in year i (zero in some years)

Ri - replacement cost in year i (zero in some years)

N - planning horizon, 20 years

d - discount rate, 0.05

ΔE - yearly energy savings

The CCE parameter is an amortization of total cost of the heat-pump project according to a constant discount rate d over the planning horizon N, expressed per saved kilowatt hour. Present and predicted future electricity costs can be compared to the CCE for judging the economic merit of the heat pump.

A second economic parameter that is of more importance to the homeowner considering buying a heat pump is the net present benefit NPB defined as:

$$NPB = S \cdot \sum_{i=1}^{N} \frac{\prod_{k=1}^{i} (1 + f_k)}{(1+d)^i} - \sum_{i=1}^{N} \frac{Mi + Ri}{(1+d)^i} - C \qquad (4)$$

where:

NPB - net present benefit, \$

S - \$ savings in electricity for year 1 based on total electricity usage of mechanically ventilated house and current electricity price schedule

$f_k$ - real escalation rate for electricity in year k relative to previous year

With the present world situation of uncertain oil prices, prediction of future electricity prices is very difficult; here, $f_k$ is assumed to be zero for the entire time horizon, i.e. the escalation rate for electricity prices is assumed to equal the rate of inflation. This assumption is based on consultation with utilities and other organizations that forecast future electricity prices.

Figure 7 shows the cost of conserved energy (CCE) for both houses in all three cities. The results for the MCS house are somewhat higher due to the somewhat smaller energy savings in the denominator of Eq. (3). Although the CCE is best compared to the marginal cost of electricity, or the so-called avoided cost, it is here simply compared to current residential electricity costs (rates above baseline limit): 4.8¢/kwh in Portland, 5.7¢/kwh (4.4¢/kwh winter time) in Minneapolis and 5.2¢/kwh in Memphis. With this comparison all three ventilation strategies have poor economic potential in Memphis, good potential in Minneapolis, and are economically neutral in Portland. It is interesting to note that addition of the fan coil does not appreciably affect the economics. This implies that the added energy savings from the fan coil roughly balances the added costs. A second observation is that the heat exchanger is very attractive in Minneapolis implying that heat exchangers that are significantly cheaper in initial and replacement cost than heat pumps, do well in cold climates where the temperature driving force is large during a large portion of the year.

Figure 8 compares the net present benefit that a homeowner is predicted to realize with the three ventilation strategies (all referenced to the 0.7 ach NV case). The benefit is seen to turn to a significant net cost in both houses in Memphis, a small net cost in the Portland MCS house, and essentially zero benefit in the Portland STD house. A significant benefit is realized only in Minneapolis; both houses benefit from the heat exchanger, but only the STD house benefits significantly from a heat pump-based strategy (particularly with fan coil). From the homeowner's point of view therefore, the heat pump at present is economically attractive only in a house with "standard" insulation (not superinsulated) located in the northern part of the U.S. However, it must here be emphasized that the net present benefit is very sensitive to the assumed electric rate escalation factor (assumed zero in Fig. 8 results). The results are also sensitive to the price of the heat pump.

## HEAT PUMP PARAMETRIC EFFECTS

The reason for including a section on heat-pump parametric effects is to ascertain that this assessment is not overly sensitive to the assumptions made regarding the heat-pump/water tank system. The results presented in this paper _are_ sensitive to the control set points for water and space heating. However set points given in Table 1 and discussed in a previous section were the result of extensive testing, and are considered near optimal.

Table 5 shows key results from simulation runs for the Portland STD house provided with modified heat-pump/water tank systems. In addition, the first two rows refer to the "standard" HP and HP+FC cases, and are included for reference purposes. Some interesting observations can be made regarding the heat-pump on times in the reference cases. Addition of the fan coil (and the building space as a second heat sink) allows the heat-pump compressor to be on 69% of the year as compared to only 55% for the case with no fan coil. The corresponding on times for the winter months (December through February) are 59% for the HP case and 87% for the HP+FC. This last figure is interesting because significant electric space heating is used in winter although for 13% of the time no energy is recovered from the exhaust air stream. The cause of this imbalance is the dynamics of the system: both the fan coil and the water tank would have to be several times larger in capacity in order to recover energy from the exhaust air 100% of the time. Such a system would be an overdesign for the rest of the year when the extractable energy in the exhaust air (assuming constant exhaust-air rate) exceeds the energy required for water heating.

Row 3 of Table 5 refers to a case where the heat-pump rating has been doubled to 800W implying a doubling of compressor power, water and air flow rates through the fan coil, but no change in ventilation rate (0.5 ach) and no change in COP. The assumption of a constant COP with HP capacity is somewhat unrealistic because COP generally drops with temperature in the evaporator. Another consequence of reduced evaporator outlet air temperature is a buildup of frost; this would require occasional defrosting which would lead to a somewhat lower efficiency. Despite the somewhat favorable assumptions made for the 800-Watt heat pump, it is clear from Table 5 that significant energy savings can be accomplished by increasing the heat-pump rating. Even an increase of heat pump rating without increasing the fan-coil rating has a significant positive effect on the energy balance (10,170 kwh total water and space heating power usage).

Assuming a fully mixed hot water tank volume, i.e. no stratification, is seen to have a negative effect on the energy balance (Row 4 of Table 5) although not as strong as might be expected from the steep COP curves in Fig. 1. It is not known whether the effect indeed is only 1,000 kwh, or if the TRNSYS model of a stratified tank[3] that neglects both buoyancy effects and the momentum of the feed water is inaccurate. The model does predict relatively large temperature differences (15-25°C) between top and bottom, but they dissipate quite quickly when hot water is demanded. Accurate information on this issue must come from future experiments.

Figure 9 shows the effect of hot water usage on the energy savings of the three mechanical ventilation cases as compared with the natural ventilation case. The heat pump without fan coil is seen to be very sensitive to water usage; the energy savings would, in fact, go to zero at zero water usage if it were not for the different average ventilation rates. (The heat pump at zero water usage "degenerates" into a simple exhaust-ventilation case with no energy recovery. Such a case can be implemented simply by installing an exhaust fan with ductwork, and tightening the house). The heat pump with fan coil is much less sensitive, although low water demands are clearly unfavorable. The economics of the HP + FC option would suffer from a reduced water demand, particularly when the heat pump is sized for the higher water demand. Increased water usage, on the other hand is favorable until the capacity limit for heat transfer from the exhaust-air stream is approached. This limit is determined by heat pump size and exhaust-air flow rate. Also, the attractiveness of the fan coil is reduced as water usage is increased because the primary tap water heat sink alone can balance the heat source. For comparison purposes, the heat exchanger case is included in Fig. 9: It is, of course, unaffected by water usage.

In contrast to the relatively strong effect of hot water usage, Table 5 shows that changing the hot water delivery schedule from the assumed schedule with a maximum flow rate of 8 l/min to the "standard" NSDN schedule[6] with a maximum flow rate of 2 l/min has an insignificant effect on the energy balance. Also, a smaller hot water tank volume (half the "standard" 300 liter volume) has little effect on the energy balance. When the tank is made even smaller (100 liters) the delivered water temperature begins to fall off significantly during certain times of the day.

## CONCLUSIONS

The previous section on heat-pump parameter effects demonstrate that further optimization of the exhaust-air heat pump system can probably yield better performance than the "average" type of performance that was obtained from applying the same heat pump to the two houses in three different climates. Heat pump rating relative to water demand for example, is a critical parameter which can be optimized for each application. However, for any meaningful optimization, more detailed operating data would be required for the heat pump/water tank system. Such data will be collected in a complementary experimental program on exhaust-air heat pumps now underway at the Lawrence Berkeley Laboratory.

From the computational assessment of exhaust-air heat pumps several important conclusions can be drawn. From an energy conservation point of view, exhaust-air heat pumps offer controlled ventilation with heat recovery from the exhaust-air stream. The basic heat pump (without fan coil) saves approximately 3000 kwh in Memphis, 4000 kwh in Portland, and 5000 kwh in Minneapolis, as compared to naturally ventilated houses with a 40% higher average ventilation rate for indoor air quality purposes. A fan coil that transfers some of the energy stored in the hot water to the building space is a good idea from an energy conservation point of view; it saves approximately 1000 kwh in both Memphis and Portland, and 1300 - 2000 kwh in Minneapolis (higher saving for STD house, lower for MCS house) over the basic heat-pump case. However, the marginal cost of conserved energy for the fan coil is approximately the same as the cost of conserved energy for the basic heat pump.

The cost of conserved energy when comparing a heat-pump house (both with and without fan coil) to a standard naturally ventilated house is about 4¢/kwh in Minneapolis, 5¢/kwh in Portland, and 7¢/kwh in Memphis. Hence, the potential of exhaust-air heat pumps is greatest in cold and moderate climates. For hot and humid climates like Memphis, the simple exhaust-air heat pump that provides only water heating (and space heating through the fan coil) is less attractive from an energy-conservation economy point of view. In hot climates, a heat pump with reversible air flow to provide cooling in summer should prove more economical.

The main result from the comparison of the air-to-air heat exchanger to the exhaust-air heat pump is that the heat exchanger is even more climate sensitive than the heat pump; it is more economical than the heat pump in cold climates, about the same in moderate climates, and less economical than the heat pump in hot climates.

From the homeowner's point of view, exhaust air heat pumps are economically attractive only in cold climates. Even in the moderate climate of Portland, a small net present loss is more likely than a net present benefit. However, these predictions are made assuming a zero real electric rate escalation for the next twenty years. Even a small escalation in electric rates would make an investment in a heat-pump system economically attractive in moderate climates.

For the homeowner however, the constant ventilation rate provided by the exhaust-air heat pump may be more important than a small net loss or net benefit in economy. As compared to natural ventilation houses where ventilation rates sometimes drop to very low levels, the house with an exhaust-air heat pump has essentially a constant ventilation rate throughout the year. This is definitely an advantage when considering indoor air pollution except for the possibility of increased radon entry rates that was discussed in a previous section. The indoor air quality aspects of mechanical ventilation including the spatial distribution of ventilation rate within the house will be addressed in future work.

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Table 1. Input Assumptions for TRNSYS Simulations

| | | |
|---|---|---|
| **Building Envelope** | | |
| Volume | $m^3$ | 306 |
| Total Floor Area | $m^2$ | 125 |
| Total Wall Area | $m^2$ | 96 |
| Total Window Area | $m^2$ | 14 |
| Floor U-Value | $W/m^2 \cdot {}^\circ C$ | 0.45 (0.18 for MCS house) |
| Ceiling U-Value | $W/m^2 \cdot {}^\circ C$ | 0.28 (0.15 for MCS house) |
| Wall U-Value | $W/m^2 \cdot {}^\circ C$ | 0.50 (0.20 for MCS house) |
| Window U-Value | $W/m^2 \cdot {}^\circ C$ | 3.1 (1.7 for MCS house) |
| **Space Heating & Cooling** | | |
| Space Heater Power Input | kW | 10(20 in Minneapolis) |
| Air Conditioner Cooling Capacity | kW | 7.0 - 10.0* |
| Air Conditioner Power Input | kW | 3.2-3.8 * |
| Space Heater ON Temperature | °C | 20.25 |
| Space Heater Thermostat Deadband | °C | 0.75 |
| AC ON Temperature | °C | 25.75 |
| AC Thermostat Deadband | °C | 0.75 |
| Thermostat Night Setback | °C | 5 |
| Average Light & Appliance Power Input | W | 420 |
| **Water Heating** | | |
| Tank Volume | l | 300 |
| Tank Height | m | 1.4 |
| Number of Stratified Zones | - | 7 |
| Jacket U-Value | $W/m^2 \cdot {}^\circ C$ | 0.36 |
| Hot Water Demand | l/24 hr | 300 |
| Max. Hot Water Flow Rate | l/min | 8 |
| Water Heater Power Input | kW | 4.5 |
| Water Heater ON Temperature | °C | 50 |
| Water Heater Thermostat Deadband | °C | 2 |
| Water Heater and Thermostat Location | (zone) | 3 (from top) |
| **Heat Pump (HP)** | | |
| Exhaust Air Flow Rate | $m^3/hr$ | 125 |
| Exhaust Fan Power Input | W | 75 |
| Compressor Power Input | W | 370-410 * |
| Coefficient of Performance (COP) | | 2-4 * |
| Condenser Loop Water Flow Rate | kg/hr | 156 |
| Condenser Loop Pump Power Input | W | 25 |
| Condenser Water from Tank Zone | - | 7 (Tank Bottom) |
| Condenser Water Return to Tank Zone | - | 4 |
| HP ON Temperature | °C | 52 |
| HP Thermostat Deadband | °C | 2 |
| HP Thermostat Sensor Location, Tank Zone | - | 7 (Tank Bottom) |
| Fan Coil Loop Water Flow Rate | kg/hr | 150 |
| Fan Coil Air Flow Rate | $m^3/hr$ | 133 |
| Fan Coil Pump & Fan Power Input | W | 35 |
| Fan Coil Heat Transfer Effectiveness | % | 65 |
| Fan Coil Water from Tank Zone | - | 1 (Tank Top) |
| Fan Coil Water Return to Tank Zone | - | 7 (Tank Bottom) |
| Fan Coil ON Temperature (Space Heater Thermostat) | °C | 21 |
| Thermostat Deadband | °C | 0.75 |
| Thermostat Night Setback | °C | 5 |
| Fan Coil Shutoff on Low Water Temperature (from Water Heater Thermostat) | °C | 52 |
| **Heat Exchanger (HX) Alternative** | | |
| Number of Centrifugal Fans | | 2 |
| Air Flow Rate | $m^3/hr$ | 125 |
| Fan Power Input (each) | W | 75 |
| Heat Transfer Effectiveness | % | 65 |
| Defrost Timer Trigger Temperature | °C | -5 |
| Defrost Time as Fraction of Timer ON Time | - | 0.20 |

*depending on operating conditions

Table 2. Natural Ventilation Characteristics: Percent of year with ventilation less than 0.25 ach.

| Climate/ House Type | Median Ventilation ~0.5 ach Windows Closed | Windows Open* | Median Ventilation ~0.7 ach Windows Closed | Windows Open* |
|---|---|---|---|---|
| Portland/STD | 6.9 | 6.1 | 0.4 | 0.3 |
| Portland/MCS | 4.2 | 2.8 | 0.3 | 0.2 |
| Minneapolis/STD | 7.2 | 5.1 | 1.2 | 0.9 |
| Minneapolis/MCS | 6.3 | 3.8 | 1.0 | 0.7 |
| Memphis/STD | 11.4 | 9.4 | 4.9 | 4.6 |
| Memphis/MCS | 10.6 | 8.5 | 4.6 | 4.1 |

*Temperature-based window opening strategy defined in the text.

Table 3. Yearly Energy Savings in kwh Obtained with Exhaust-Air Heat Pumps at 0.5 ach Relative to Natural Ventilation at 0.7 ach.

| House/ Heat Pump Configuration | Portland | Minneapolis | Memphis |
|---|---|---|---|
| STD/HP | 4,200 | 5,000 | 3,200 |
| STD/HP+FC | 5,400 | 7,000 | 4,100 |
| MSC/HP | 3,900 | 4,900 | 3,100 |
| MSC/HP+FC | 4,700 | 6,200 | 3,500 |

Table 4. Input Assumptions for the Economic Analysis

| Input Parameter | Value |
|---|---|
| Heat Pump Installed Cost | $1300 |
| Heat Pump Service Life | 10 years |
| Heat Pump Replacement Cost | $1075 |
| Fan Coil Installed Cost | $700 |
| Fan Coil Service Life | > 20 years |
| Heat Exchanger Installed Cost | $1040 |
| Heat Exchanger Service Life | > 20 years |
| Fan Service Life | 5 years |
| Fan Replacement Cost (per fan) | $90 |
| House "Tightening" Cost | $540 |
| Planning Horizon | 20 years |
| Real Discount Rate | 0.05 |

Table 5. Heat pump Parameter Effects for Standard House in Portland (all figures on an annual basis)

| | | | | <——— POWER CONSUMPTION | | | ———> |
|---|---|---|---|---|---|---|---|
| Heat Pump Configur-ation | Heat Pump Compressor On-Time % of year | Fan Coil On-Time % of year | Heat Transferred by HP Condensor | Electric Water Htr. kwh | Electric Space Htr. kwh | Heat Pump & Fan Coil (inc.Exh.Fan) kwh | Total for Water & Space Heating kwh |
| No Fan Coil | 55 | -- | 4720 | 420 | 9390 | 2640 | 12,450 |
| Base Case (w Fan Coil) | 69 | 28 | 5810 | 1190 | 6750 | 3250 | 11,190 |
| Double HP+FC Rating | 52 | 30 | 8690 | 640 | 4190 | 4690 | 9,520 |
| Fully Mixed Water Tank | 59 | 25 | 4670 | 2280 | 6920 | 2950 | 12,150 |
| Small (1501) Water Tank | 62 | 29 | 5220 | 1750 | 6580 | 3000 | 11,330 |
| NSD Water Use Schedule (lower flow rates) | 71 | 29 | 6000 | 1220 | 6640 | 3290 | 11,150 |

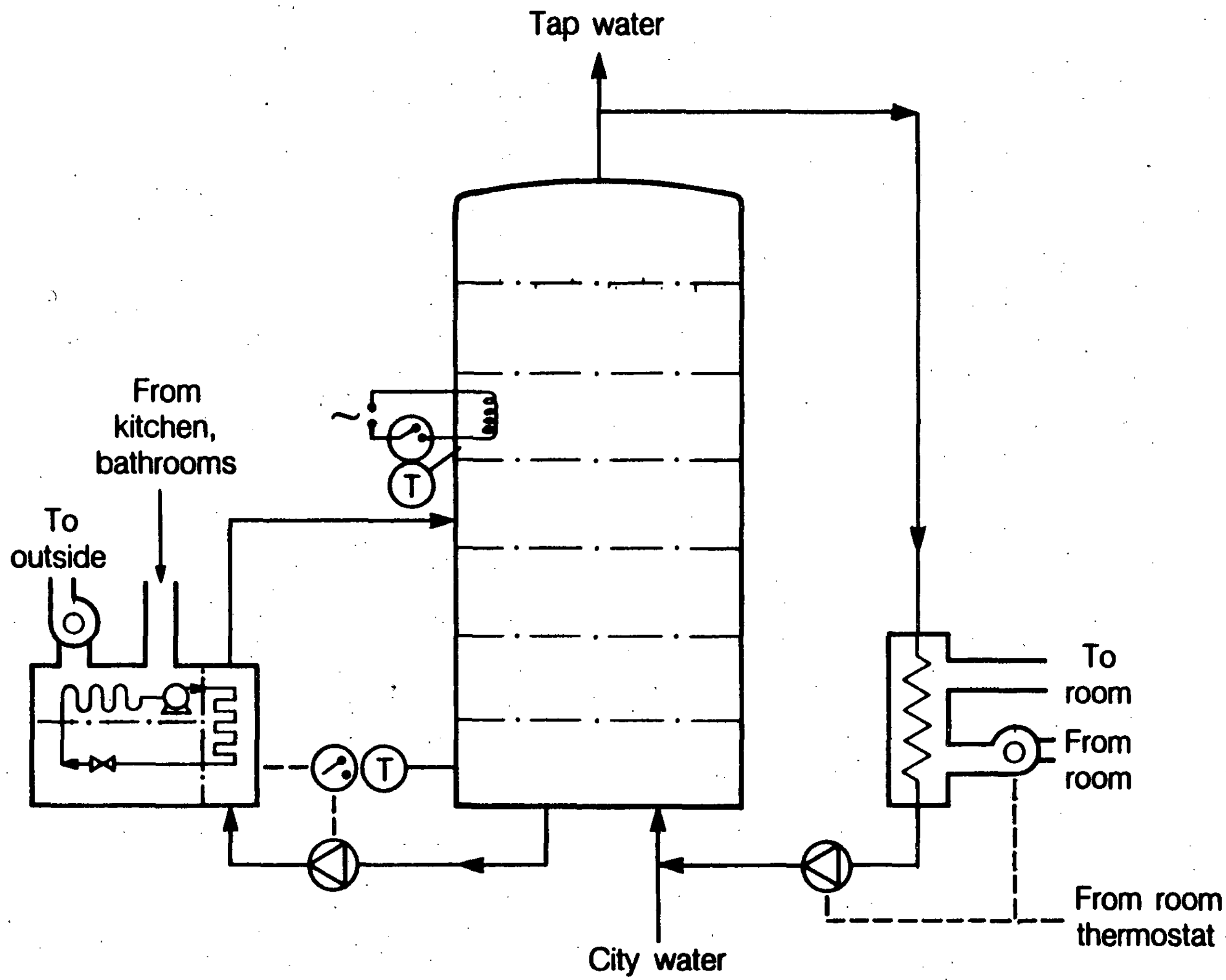


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Fig. 1 Schematic of Exhaust-Air Heat Pump System showing Water Tank in the Middle, Heat Pump to the Left and Fan Coil for Space Heating to the Right.

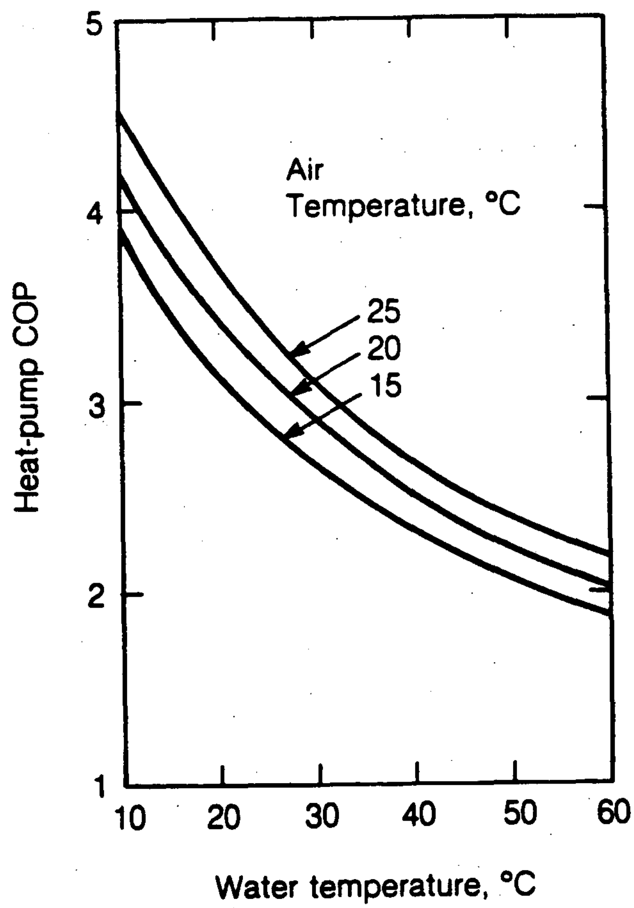


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Fig. 2 Assumed Heat-Pump Coefficient of Performance (COP = Condenser Heat Flux/Compressor Power Input) as a Function of Condenser Water Inlet Temperature and Evaporator Air Inlet Temperature.

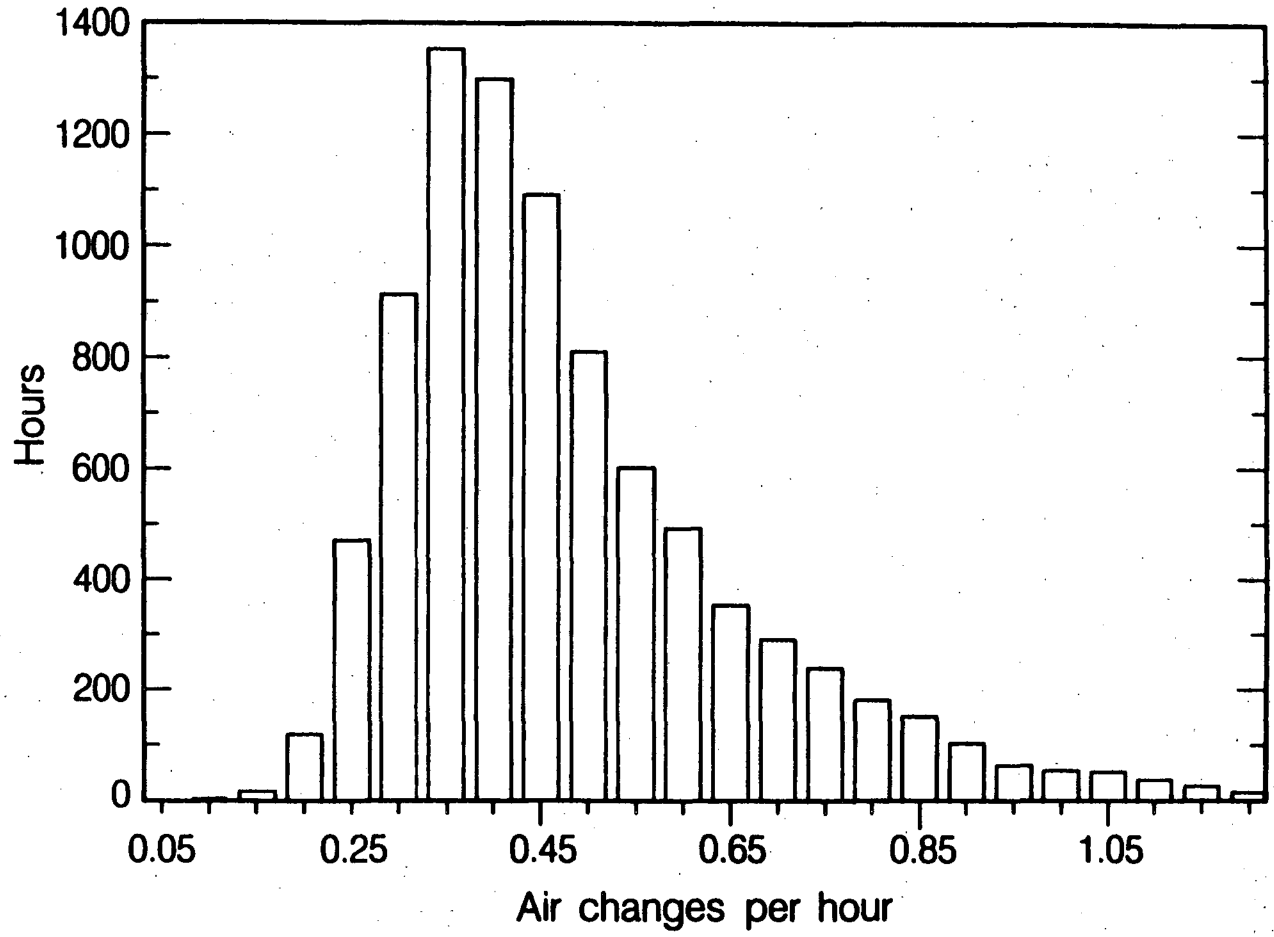


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Fig. 3 Time Histogram of Ventilation Rates in Naturally Ventilated (Infiltration Only) STD Portland House with an Effective Leakage Area of $0.05 m^2$. Yearly Median Ventilation Rate is 0.45 ach.

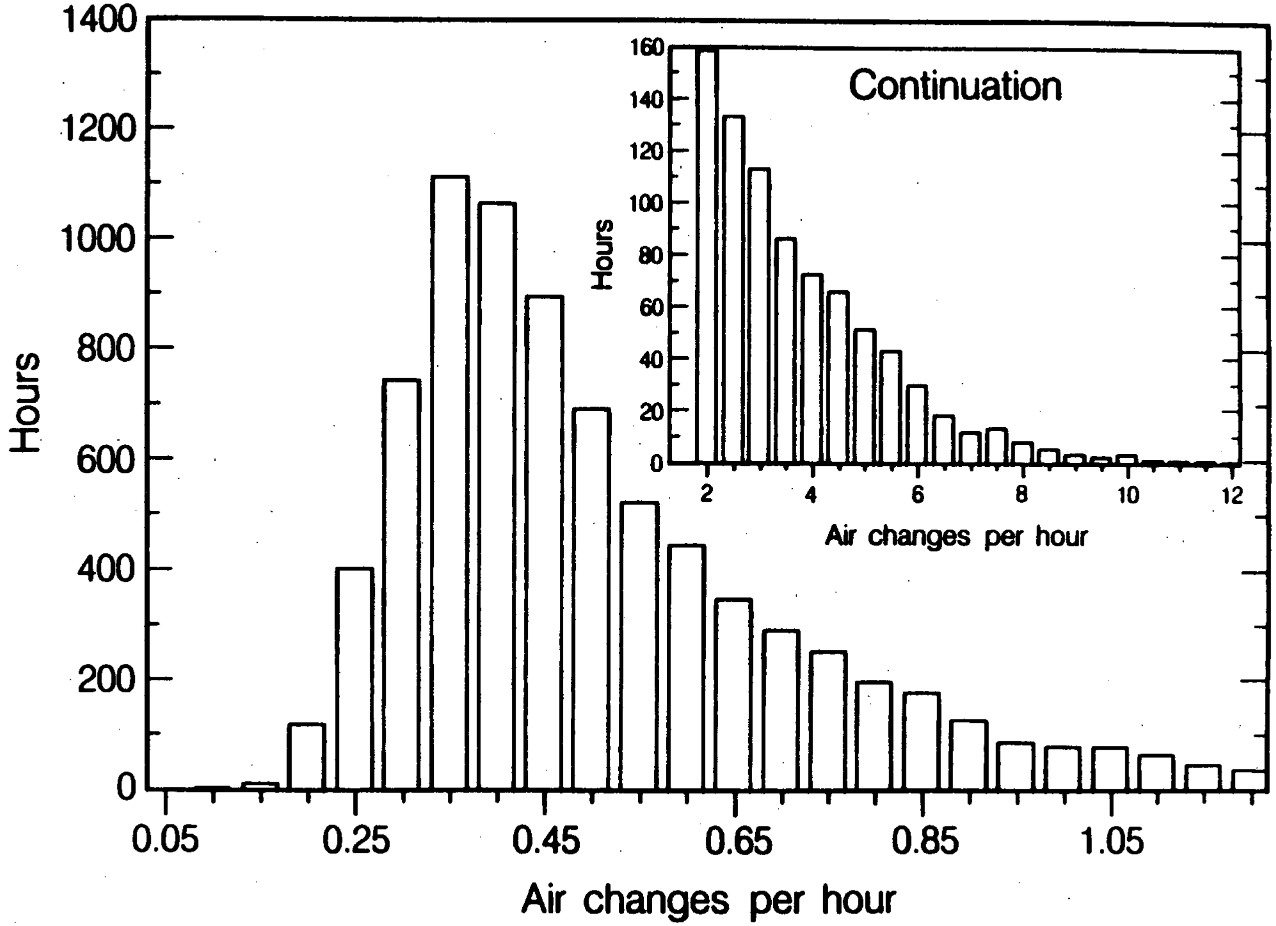


Fig. 4 Time Histogram of Ventilation Rates in Naturally Ventilated (Infiltration and Open Windows) STD Portland House with an Effective Leakage Area of $0.05 m^2$ plus an Additional Amount due to Window Opening. Yearly Median Ventilation Rate is 0.45 ach.

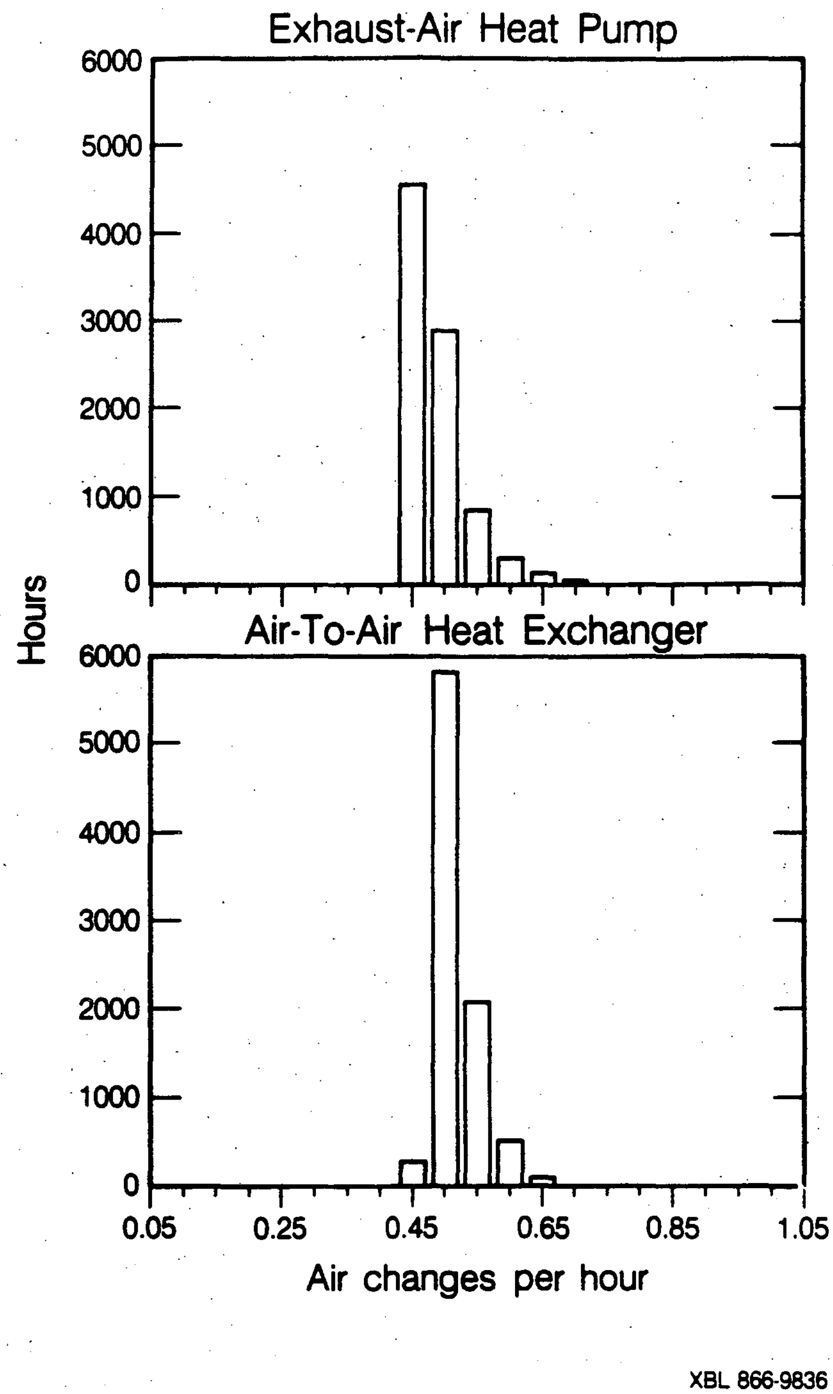


Fig. 5 Time Histogram of Ventilation Rates in STD Portland House Equipped with Mechanical Ventilation. Exhaust-Air Heat Pump Case has an Exhaust-Air Flow Rate of $125 m^3/hr$ and an Effective Leakage Area of $0.025\ m^2$. Air-to-Air Heat Exchanger Case has a Balanced Supply and Exhaust Air Flow Rate of $125\ m^3/hr$ and an Effective Leakage Area of $0.01\ m^2$.

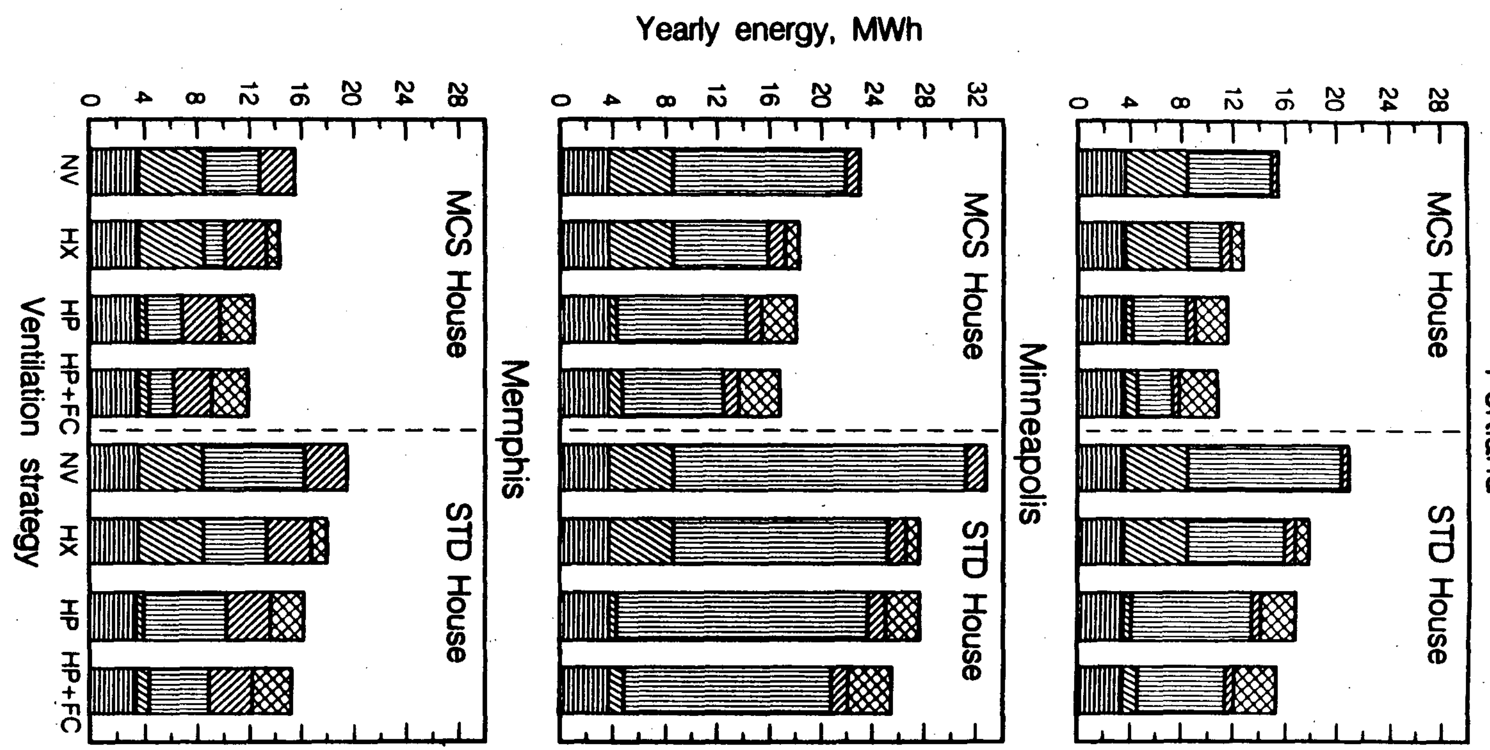


Fig. 6 Yearly Energy Use.

STD - Standard House defined in Table 1

MCS - Model Conservation Superinsulated House defined in Table 1.

NV - Natural Ventilation by Infiltration and Open Windows (0.7 ach)

HX - Air-to-Air Heat Exchanger (0.5 ach)

HP - Exhaust-Air Heat Pump (0.5 ach)

HP+FC - Exhaust Air Heat Pump with Fan Coil (0.5 ach)

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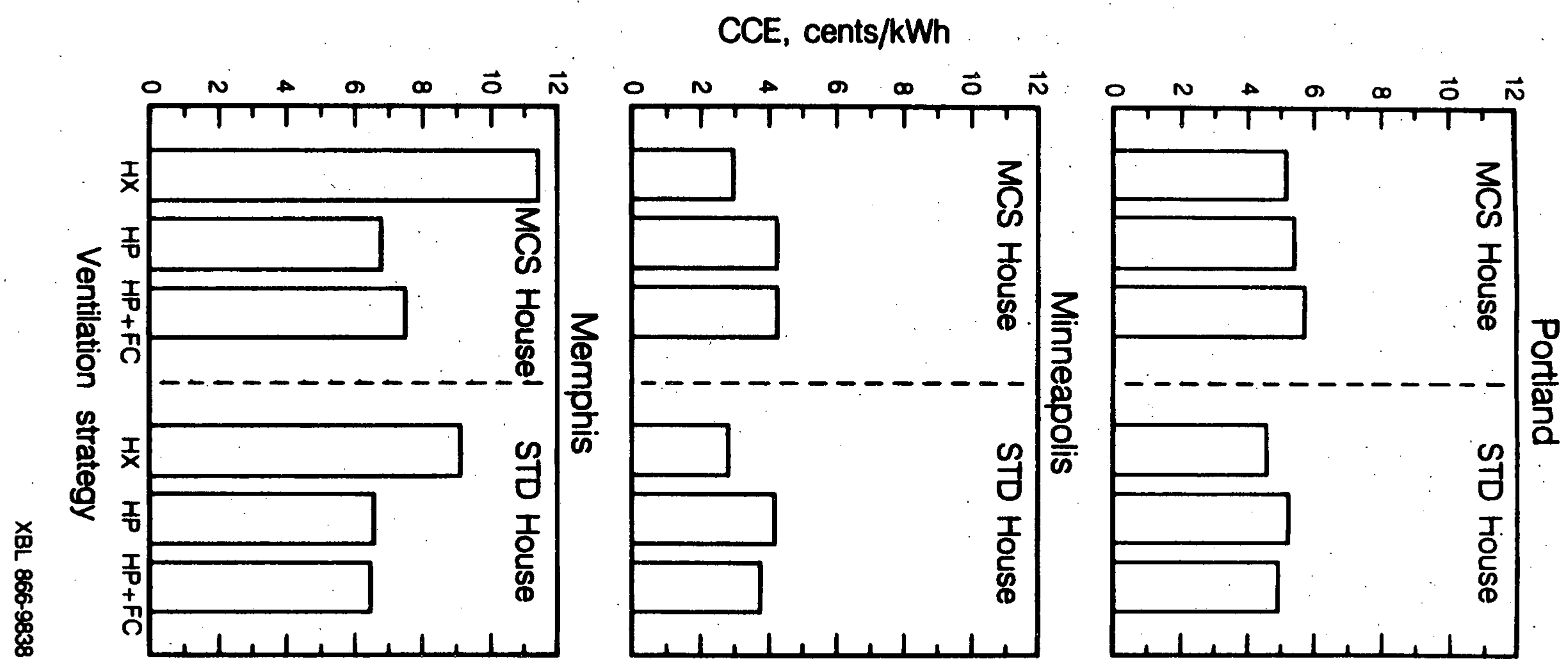


Fig. 7 Cost of Conserved Energy CCE, defined in Eq. (3) associated with Mechanical Ventilation Strategies (Relative to Natural Ventilation by Infiltration and Open Windows at 0.7 ach) as a Function of House Type and Climate.

| | | |
|---|---|---|
| STD | - | Standard House defined in Table 1 |
| MCS | - | Model Conservation Superinsulated House defined in Table 1. |
| HX | - | Air-to-Air Heat Exchanger (0.5 ach) |
| HP | - | Exhaust-Air Heat Pump (0.5 ach) |
| HP+FC | - | Exhaust Air Heat Pump with Fan Coil (0.5 ach) |

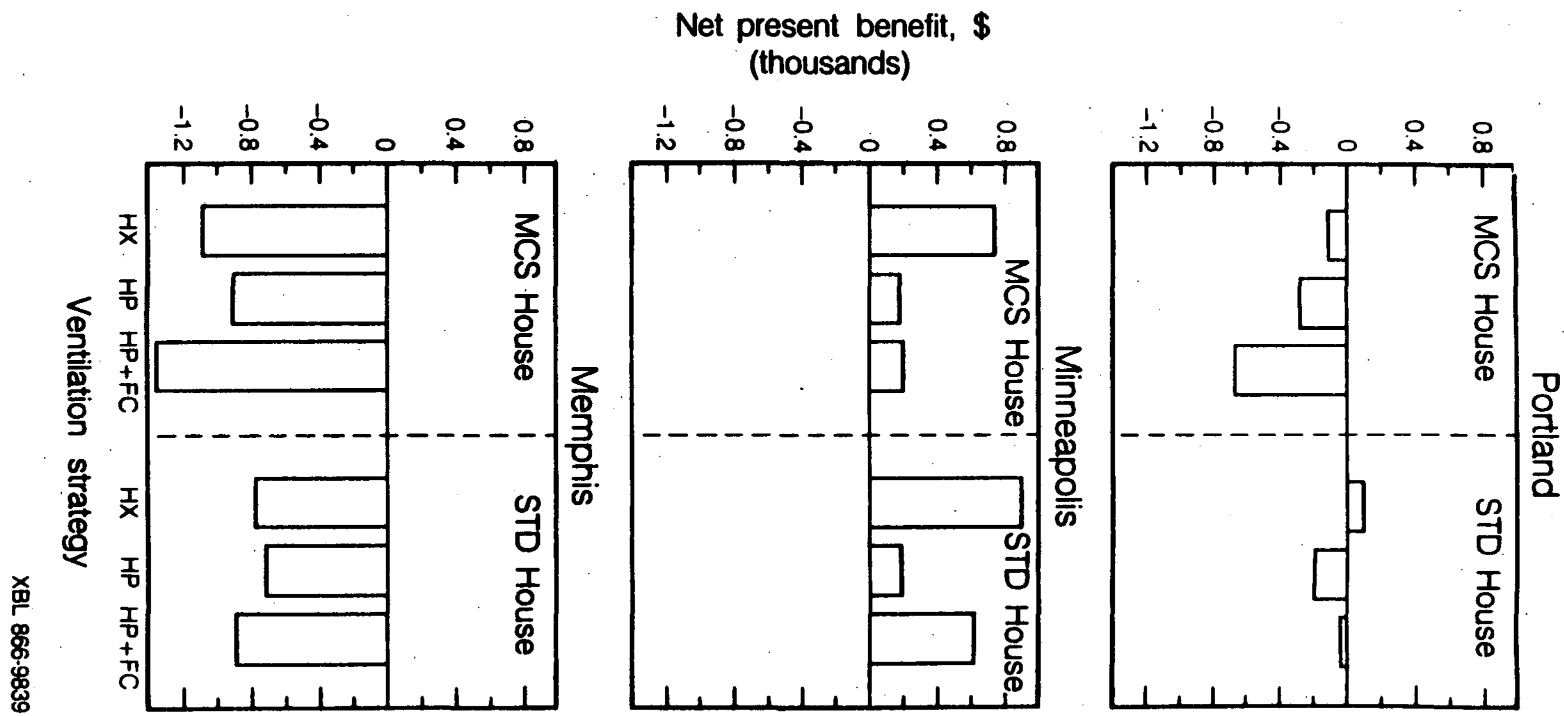


Fig. 8 Home Owner's Net Present Benefit, defined in Eq. (4), of Mechanical Ventilation Strategies at 0.5 ach Relative to Natural Ventilation by Infiltration and Open Windows at 0.7 ach as a Function of House Type and Climate.

STD - Standard House defined in Table 1
MCS - Model Conservation Superinsulated House defined in Table 1.
HX - Air-to-Air Heat Exchanger (0.5 ach)
HP - Exhaust-Air Heat Pump (0.5 ach)
HP+FC - Exhaust Air Heat Pump with Fan Coil (0.5 ach)

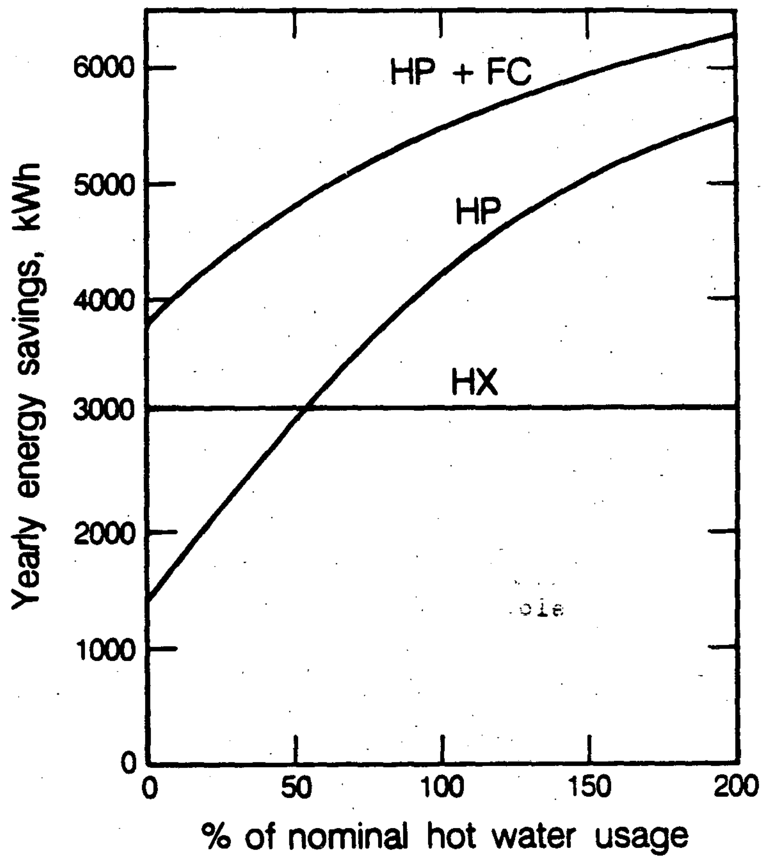


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Fig. 9 Yearly Energy Savings in STD Portland House Obtained with Mechanical Ventilation at 0.5 ach Relative to Natural Ventilation by Infiltration and Open Windows at 0.7 ach as a Function of Hot Water Usage. Nominal Hot Water Usage is 300 1/24 hrs.

HX - Air-to-Air Heat Exchanger (0.5 ach)
HP - Exhaust-Air Heat Pump (0.5 ach)
HP+FC - Exhaust-Air Heat Pump with Fan Coil (0.5 ach)