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DISCLAIMER

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Preliminary Assessment of Residential Exhaust-Air Heat Pump Applications in the Pacific Northwest

by

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CONTENTS

Abstract

We 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, compressor and controls have recently become available on the U.S. market. The study concerns applications in the Pacific Northwest with three representative locations: Portland, Oregon, Spokane, Washington and Missoula, Montana. Our approach is to simulate system performance for a full year in both a typically-insulated and a well-insulated house. 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 that of 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 range from about 5000kwh in Portland to about 7500 kwh in Missoula if the house with the heat pump has a lower average ventilation rate than the naturally ventilated house used as a basis for comparison. An economic analysis indicates that heat pump installations are cost competitive from the homeowner's point of view in the colder interior parts of the Pacific N.W. The impact of exhaust ventilation on indoor radon concentrations and radon entry rates into houses is examined using recently-developed models. Exhaust ventilation is shown to be generally suitable, from the perspective of indoor radon, in houses with crawl spaces and in houses with basements surrounded with relatively impermeable soil. Available information on the availability, reliability, and general acceptability of exhaust-air heat pumps is also reviewed.

INTRODUCTION

The topic of this study is electric exhaust-air heat pumps that provide ventilation in a controlled manner together with energy recovery. The emphasis here is on heat pumps that heat tap water because such heat pumps are relatively new to the U.S.; for example, the authors are aware of only one U.S. manufacturer, currently marketing an exhaust-air to tap-water heat pump.

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. 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 the set-point temperature, and hence no energy recovery can occur when this single heat sink is saturated. With the fan coil, energy recovery from the exhaust air stream can in principle occur 100% of the time during the heating season. However, during the cooling season the fan coil does not operate. Heat pumps that also provide cooling in summer are not studied in the present work although there are some energy conservation benefits from such an enhancement of the basic heat pump system in hot climates where air conditioning loads are significant.

Although the heat-pump system modeled here is a remote type, i.e. the heat-pump condenser is separate from the water tank, the analysis is directly applicable to integral heat pumps where the condenser is part of the tank (internal or external). Also, the fan coil may be heated directly by the refrigerant instead of the water-heated fan coil studied here. The refrigerant-heated fan coil allows space heating priority over water heating, a case that is not studied here.

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 that has been studied previously (Fisk and Turiel, 1983). Comparing the two approaches briefly, the heat pump has the main advantage of providing a large temperature difference for driving the energy recovery, i.e., energy recovery is not limited by 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 of the Pacific Northwest 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 it 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. This aspect of exhaust-air heat pumps is investigated in this report.

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 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 (4792 degree days/year), Spokane, Washington representing a colder climate (6835 degree days/year) and Missoula, Montana representing a still colder climate (7931 degree days/year). For the sake of comparing the heat pump to alternative ventilation strategies, the same houses are simulated using natural ventilation (NV) with electric space and water heating.

VENTILATION AND ENERGY PERFORMANCE SIMULATIONS

Simulated System

TRNSYS (Transient System Simulation Program) is a commercially available program for simulating the energy performance of buildings (Solar Energy Laboratory, 1984). 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) thermostatic control modules for indoor-space and hot-water temperature, and (5) a weather data module. The building module is further divided into submodules representing the building envelope, an indoor space heater, internal heat gains, an infiltration/ventilation model, and a domestic hot water heater.

The building envelope specifications for the two houses simulated 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, walls R-28 vs R-11, floor R-31 vs R-13, and triple-glazed windows vs double-glazed windows in the STD house.

An important building envelope parameter listed in Table 2 is the effective leakage area that correlates the rate of air flow through the building envelope to the pressure difference. This quantity is

$$Q = ELA \cdot \sqrt{\frac{2 \cdot \Delta P}{\rho}} \tag{1}$$

where Q = volumetric infiltration rate, ELA = effective leakage area, ΔP = pressure difference (indoor-outdoor), and ρ = 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 ELA by a fan-pressurization technique and determination ΔP allow prediction of Q (Sherman and Grimsrud, 1980). In the LBL infiltration model, ΔP 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 coupled through the combined ΔP and the total ventilation rate is given by $\sqrt{(Q^2_{infiltration} + Q^2_{exhaust})}$. For the simulations of this paper, the effective leakage area of the naturally ventilated houses is adjusted to achieve the desired total ventilation rate. This leads to the different effective leakage areas in the different locations (Table 2).

To account, at least approximately, for the effect occupants may have on the ventilation rate, a window opening strategy is used in the simulations. This strategy is temperature based according to the following rule (on/off controller with hysteresis gated by the outdoor temperature):

Windows <u>open</u> if open last time period and T_{indoor}>22.5 and T_{indoor}> $T_{outdoor}$, or if closed last time period and T_{indoor}>23.5 and T_{indoor}> $T_{outdoor}$.

When windows are open, $1.2m^2$ 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. However, our results are insensitive to the window-opening model.

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 shown in Fig. 2 where the maximum flow rate is 8 l/min for 7.5 min starting at 8:00. 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 heat-pump system for the houses. The performance data for the heat pump is taken from a Flakt AB product bulletin (Flakt AB, 1984) which contains the only detailed performance curves that we could obtain. The correlation for the heat pump coefficient of performance (COP) is redrawn in Fig. 3 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 50°C. However, the water tank is assumed to be stratified in accordance with a seven-zone TRNSYS model. 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, 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 1) 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. 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 heater deadbands referred to in Table 1 (2°C for water heater and 0.75°C for space heater) are above the given setpoints so that the electric space heater for example, is always on when the indoor temperature is below 20.25°C and always off above 21°C, and on or off in between depending on the state before entering the deadband region (hysteresis). 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), 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.

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) weather data for solar radiation, outdoor temperature, humidity and wind speed (National Climate Center NOAA, 1981) 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. However, the emphasis in this report is on integral results for the entire 1-year simulation, and not on peak power demand.

Ventilation Results

A typical ventilation result from a naturally ventilated house is shown in Fig. 4, which shows the yearly time distribution of ventilation rates for the Portland MCS/NV house with an assumed effective leakage area of 460 cm^2. The yearly mean ventilation rate is 1.4 ach, while the winter (December, January and February) mean ventilation rate is 0.51 ach. The cause of the "tail" of the distribution is the window opening that occasionally produces high ventilation rates. Window opening thus affects the yearly mean ventilation rate strongly, but does not influence the winter ventilation rates. However, a more important aspect is the front end of the distribution, i.e. ventilation rates from 0.1 ach to 0.25 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 hrs for the distribution in Fig. 4, which is clearly undesirable because the recommended minimum ventilation rate is considered to be in the range of 0.35 - 0.5 ach. (ASHRAE, 1981 and the Nordic Committee on Building Regulation, 1981). Interestingly, the front end of the distribution is almost unaffected by the window opening strategy as shown by simulations without window opening. The main reason for the low natural ventilation rates is periods of warm weather with no wind when windows are closed because the outdoor temperature exceeds the indoor temperature.

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 winter mean ventilation rates of approximately 0.7 ach. The results of Table 2 show that there is more than an order of magnitude reduction in time with a ventilation rate less than 0.25 ach. Consequently, a conclusion is that naturally ventilated houses should not be designed to the same house tightness standards as used for houses with mechanical ventilation.

In contrast to the broad distribution obtained with natural ventilation, Fig. 5 shows the sharp distribution of ventilation rates that is obtained through mechanical exhaust ventilation. A simple exhaust fan provides, of course, the same ventilation as an exhaust-air heat pump. Figure 5 shows that the ventilation rate never drops below 0.45 ach as long as the exhaust fan is on (for the Fig. 5 case, the exhaust fan is on 95% of the year; the fan is only off when windows are open and the hot water tank is at set point temperature).

Although exhaust air heat pumps (and simple exhaust fans) provide essentially a constant ventilation rate throughout the year, there is no guarantee for a uniform spatial distribution of ventilation rate (uniform 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 from the kitchen, but a question is: How well ventilated is a side bedroom with closed door? To answer questions of this nature experimental studies involving tracer gases will be required.

Energy Results

For the energy and economics sections of this study the 0.7 ach natural ventilation (NV) case is considered the reference case. The reason for this approach is two-fold: (1) there is evidence that typical naturally ventilated houses have roughly this ventilation rate (Grimsrud, Sherman and Sonderegger, 1983) 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 to be 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 in energy-efficient, mechanically-ventilated residences. A consequence of lowering the average ventilation rate from 0.7 ach in the NV houses to 0.5 ach for the houses with mechanical ventilation is energy savings for the mechanically ventilated cases. The energy savings due to the lower ventilation rate are approximately 1,100 kwh in Portland, 1,600 kwh in Spokane and 2,500 kwh in Missoula. These savings, minus a fan power charge, can be accomplished simply by house tightening and addition of an exhaust fan with appropriate ductwork.

All energy results are summarized in Tables 3-5. 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 the naturally ventilated houses. The main result of this comparison is that the mechanical ventilation strategies save energy over the natural ventilation case in the following order: Exhaust Fan Only < Heat Pump < Heat Pump with Fan Coil. The amount of energy saved is dependent on climate , heat pump configuration (Exhaust Fan, HP, and HP + FC) and on house type. The energy savings are summarized in Table 6. For the HP + FC system, the energy savings are 4900 kwh in the Portland MCS house, 5900 kwh in the spokane MCS house and 7000 kwh in the Missoula MCS house. Similarly, for the STD houses, the energy savings are 5800 kwh in Portland, 7000 kwh in Spokane and 8200 kwh in Missoula.

As shown by Table 6, climate has an effect both on energy saved by the heat pump only (this is due to the difference in ventilation rate between the HP case and the NV reference case, 0.5 ach vs. 0.7 ach), and on the energy saved by the fan coil. A conclusion is that a cold climate favors heat pumps, particularly with fan coils. House Type (MCS vs. STD) has little effect on energy saved by the heat pump only, but has a significant effect on the energy saved by the heat pump + fan coil. Consequently, another conclusion is that thermal insulation makes the fan coil less attractive.

An observation from Tables 3-5 is that the major part of the saved energy, even in the fan-coil cases, is power saved for electric water heating and in these 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 included in the next section.

<u>Sensitivity Results</u>

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.

From an energy-efficiency point of view it is desirable to operate the heat pump compressor continuously so that maximum heat recovery occurs. However, such continuous compressor operation would require a large heat sink that never became saturated. In practice, the two heat sinks, the tap water tank and the building space become saturated, i.e. reach set point temperature, and the compressor must shut off. Tables 3-5 include the on times of both compressor and fan coil in the simulated cases. Table 3, for the Portland MSC/HP + FC case, shows that addition of the fan coil (and the building space as a second heat sink) allows the heat-pump compressor to be on 49% of the year as compared to only 39% for the case with no fan coil. The corresponding on times for the winter months (December through February) are 42% for the HP case and 70% for the HP+FC. This last figure is interesting because significant electric space heating is used in winter, although for 30% 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 allow recovery of 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.

The above example demonstrates the energy balancing that is central to exhaust-air heat-pump operation. Contrary to air-to-air heat exchanger that are intrinsically balanced in terms of energy source and sink (as long as the air flows are balanced), exhaust-air heat pumps are not. For heat-pumps, the recoverable energy from the exhaust-air stream is limited either by the flow rate of the exhaust stream (since the exhaust temperature for avoiding frosting problems of the evaporator is limited to roughly 0°C) or by available water and space heating load. For analysis of the heat-pump system (including heat sinks) the following yearly system COP is defined

$$\text{Yearly System COP} = \frac{\text{Heat Transferred by Condenser in a Year}}{\text{Power Consumed by HP incl. Fans \& Pumps in a Year}} \quad (2)$$

All power consumed by the exhaust fan is included in the denominator; hence, the yearly system COP is lowered when the exhaust fan is on but the compressor is not (heat sink saturated).

Tables 3-5 show that increasing the exhaust rate from 125 m^3/hr for the HP case (no FC) decreases the yearly system COP although the intrinsic heat-pump COP increases with exhaust-air flow rate (+12% COP increase). The reason for the decrease in yearly system COP is that the tap water heat sink is already saturated with the 125 m^3/hr exhaust air flow rate, and hence no increase can occur in condenser heat transfer. The fan power requirement however, increases with the increased exhaust-air rate. The HP + FC cases on the other hand, maintain approximately their yearly system COP with the 60% increase in exhaust ventilation rate.

Fig. 6 shows the sensitivities of the yearly system COP to four system parameters: HP intrinsic COP, HP Capacity, hot water usage and house size. It should be pointed out that constant yearly system COP does not imply constant energy savings, because the system COP is a ratio of useful heat supplied by the HP to the power input. Comparing the three locations studied, the base HP + FC case produces a yearly system COP of approximately 1.9 in all three climates for both the MCS and the STD house. Also, the sensitivities are very similar in the three climates. The main conclusion to be drawn from the sensitivity curves in Fig 6 is that the yearly system COP is strongly affected only by the HP intrinsic COP, and the proportionality is almost direct. Hot water usage has an effect but it is minor, even on the down side. In comparison, heat pumps without fan coils are strongly affected by reductions in hot water usage because of a reduced heat sink (with no hot water usage, the HP case degenerates to an exhaust fan without heat recovery).

Water usage has a strong effect on the amount of energy saved as shown in Tables 3-5. In the Portland MCS house with HP + FC, energy savings increase from 4900 kwh for a water usage of 300l/day to 5980 kwh for a water usage of 450l/day. For a reduced water usage of 150l/day, energy savings drop to 3600 kwh. Similarly in Missoula the energy savings for the MCS house with HP + FC are 5860 kwh for 150l/day, 7030 kwh for 300l/day and 8040 kwh for 450l/day. Increased house size also has a positive effect on the energy savings, but not as strong as the effect of water usage.

A good sizing criterion for heat pumps is that either the evaporator capacity be sufficient to cool the desired exhaust air flow down to the practical temperature limit of 0°C (or possibly a few degrees below zero), or if saturation of available heat sinks occurs at lesser capacity, the condenser duty be sufficient to saturate the heat sink (Here, saturation of the heat sink must be considered in a time-averaged fashion because the water tank provides some capacitance for load shifting. In addition, the controlling heat sink is the larger winter-time sink). Hence, the heat pump need only have the capacity to satisfy either the source or the sink. Overcapacity in terms of the sink is easily accommodated by cycling the heat pump on and off, whereas overcapacity in terms of the source can lead to frosting problems. Consequently it is desirable to match compressor capacity to exhaust fan rate.

Refering to Tables 3-5, a change in the hot water delivery schedule from the assumed schedule with a maximum flow rate of 8 l/min (Fig 1) to the "standard" NSDN schedule of Fig. 7 (Barvir et al., 1981) has an insignificant effect on the energy balance. Although not shown explicitly, it can be mentioned that a smaller hot water tank volume (half the "standard" 300 liter volume) also 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.

ECONOMICS

Economic Approach

The input assumptions for the cost analysis are given in Table 7 and represent our best judgment. The only publicly available document used for some of the cost data is a survey of 366 houses in the Pacific Northwest (Vine, 1986). 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 7) is equal to the purchase price ($1300), plus the cost of the installation of the ductwork ($225), minus the cost of a conventional hot water tank ($225). 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 approximately 10 years. The only other maintenance cost assumed is the cost for parts and labor ($90) to replace the fans every five years. Mechanical ventilation with a simple exhaust fan is assumed to require investment in the house tightening ($540) and in an installed fan with ductwork ($325).

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

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

where CCE = cost of conserved energy $/kwh, C = initial cost of installed heat pump (and fan coil) 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:

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

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.

Prediction of future electricity prices is very difficult; here, f_k for one set of results 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, and for another set of results f_k is assumed to be 2%. Current electricity prices used in the calculations are given in Table 8.

Economic Results

Table 9 summarizes the economic results for both houses in all three cities. Although the cost of conserved energy (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 in Table 8): 4.8¢/kwh in Portland, 4.1¢/kwh in Spokane and 5.3¢/kwh in Missoula (in wintertime). With this comparison, the mechanical ventilation strategies have relatively poor economic potential in Portland, good potential in Missoula, and are approximately neutral in Spokane. Mechanical ventilation at a high rate (0.7 ach) increases the CCE significantly relative to the lower average rate of 0.5 ach. Hence, excessive exhaust ventilation rates must be avoided, particularly in winter. The STD houses provide greater economic potential for heat pumps with fan coils than the MSC houses because of their greater heating loads. This effect is similar to the effect of house size that also favors the HP + FC case. Interestingly, the simple exhaust fan provides a CCE of only 3.4¢ in Missoula, and is therefore a viable option. In the warmer climate of Portland, this ventilation strategy has poor economic potential (CCE = 9.1 ¢/kwh).

Addition of a fan coil to the basic heat-pump is seen to lower the cost of conserved energy in all cases (Table 9). The benefit from this addition is particularly significant in houses with high heating/ventilation loads. The only case where a fan coil is not needed is when energy for hot water demand positively exceeds the recoverable energy in the exhaust air stream.

Table 9 also lists the net present benefits that homeowners are predicted to realize with the different ventilation strategies. The net present benefits are presented for both the 0% and the 2% escalation of electricity rates. Present predictions (Wallman at al., 1986) are close to 0%. As shown by Table 9, a homeowner in Portland is more likely to incur a loss than a benefit from investing in a heat pump system, particularly if the house is well insulated. Consequently, the conclusion is that either hot water demand or house size must be significantly above the average-type values considered in this economic analysis before a homeowner in the coastal region of the Pacific Northwest can benefit economically from an exhaust-air heat-pump system. In the significantly colder climate of Missoula, a homeowner can presently save money by buying a heat-pump system as shown by the net-present-benefit values of Table 9.

EXHAUST VENTILATION AND INDOOR RADON

Background

In most U.S. houses with elevated radon concentrations, the radon enters primarily due to pressure-driven flow through penetrations in the substructure. For example, in a house with a basement, soil gas can enter through a small gap between the basement floor and walls (i.e., the wall-to-floor gap) or through other penetrations to the soil which surround pipes and conduits. In a house with a crawl space, pressure-driven flow can occur from the crawl space to the house through various cracks and penetrations in the floor above the crawl space. The pressure differences that drive these flows result from wind and, during the heating season, the indoor-to-outdoor temperature difference. In addition, any device or process that causes air to be exhausted from the house without a mechanical supply -- such as operation of fireplaces, vented combustion appliances, and exhaust fans -- will contribute to the pressure difference which drives radon entry. Because exhaust ventilation is included in this latter category, its impacts on radon entry and indoor radon concentrations are of concern.

Approach to Analysis

Since there are insufficient measured data on the relationship between exhaust ventilation and indoor radon concentration, the models developed by Mowris (1986) are utilized for the analysis in this paper. Due to the complexity of the models, only a general description, selected equations, and key assumptions are presented here.

The models are utilized to estimate the impacts of exhaust ventilation in two hypothetical houses. The first house has a full crawl space and the same volume, geometry, effective leakage areas, and exhaust ventilation rate (when exhaust ventilation is employed) as the crawl-space house used in the previously described TRNSYS simulations for Portland. We assume that the effective leakage area of this house is distributed uniformly between walls, ceiling, and floor in proportion to their respective areas and that the crawl space is vented. The second hypothetical house contains a full basement, has twice the volume of the house with a crawl space, and, when exhaust ventilation is employed, is exhaust ventilated at 250 m^3/h or twice the rate of exhaust ventilation in the crawl-space house. It is assumed that the only significant penetration between the basement and the soil is a wall-to-floor joint which extends around the entire perimeter of the basement floor. The permeability of the soil surrounding the basement is also assumed to be uniform.

The analysis is performed for an indoor-to-outdoor temperature difference of 20°C and a wind speed of 3 m/s. The first step is a calculation of the ventilation rate of the house using the LBL infiltration model (Sherman and Grimsrud, 1980). The magnitude of the pressure difference which drives flow from the crawl space or soil to the building's interior is then estimated, taking into account the impacts of wind, the indoor-to-outdoor temperature difference, and exhaust ventilation. The component of the total pressure difference which results from exhaust ventilation (ΔP_e) is expressed simply by the equation

$$\Delta P_e = \frac{\rho}{2} \left[\frac{Q_e}{ELA} \right]^2 \tag{5}$$

where ρ is the density of the indoor air, Q_e is the exhaust flow rate, and ELA is the total effective leakage area.

The third step of the analysis is a computation of the flow rates of crawl-space air or soil gas into the house. For a crawl space, this flow rate (Q_{cs}) is easily estimated by using the equation

$$Q_{cs} = ELA_f \left[\frac{2\Delta P_{cs}}{\rho} \right]^{1/2} \tag{6}$$

where ELA_f is the effective leakage area of the floor and ΔP_{cs} is the total pressure difference across the floor above the crawl space. In the model for a house with a basement, a more complex procedure, which accounts for the resistance to flow through both the soil and the wall-to-floor gap, is used to estimate the entry rate of soil gas into the basement.

The fourth step, a calculation of the rate of pressure-driven flow of radon into the house, is based on assumed values for radon concentrations in the crawl space or in soil gas. The pressure-driven flow of radon is, therefore, a simple product of the assumed radon concentration and the appropriate flow rate.

The final step is to use a single-zone, steady state, mass balance equation to compute the indoor radon concentration, i.e.,

$$Rn_i = \frac{\sigma_d + \sigma_f + (\lambda_v - Q_{cs}/V)\ Rn_o}{\lambda_v} \qquad (7)$$

where: σ_d is the rate of radon diffusion into the house (which is assumed to equal zero), σ_f is the rate of pressure driven flow of radon into the house, λ_v is the ventilation rate, V is the house volume, and Rn_o is the outdoor radon concentration which is assumed to equal 0.25 pCi/l (Gesell, 1983).

Results

Before discussing the results, which are presented in Tables 10 and 11, we point out the need for experimental data on exhaust ventilation and radon entry. The results generated with this model are thought to be generally representative, and Mowris (1986) has presented some favorable comparisons between modeled results and measured data. However, in instances where the basic assumptions of the model do not correspond to actual conditions, the trends indicated by the model may deviate substantially from actual tends. Two key assumptions, that may lead to discrepancies, are the physical characteristics of the penetrations between the basement and the soil and the assumption of a uniform soil permeability. In regard to the second assumption, the permeability of the soil immediately adjacent to penetrations is clearly a key factor. Also, if the ratio of exhaust flow rate to effective leakage area differs substantially from that assumed in this paper, the amount of depressurization caused by exhaust ventilation will also change substantially. Thus, the results presented here are sensitive to this ratio.

The analysis for a house with a vented crawl space, indicate that exhaust ventilation plus house tightening, which includes tightening of the floor, will lead to significantly lower indoor radon concentrations and less flow from the crawl space to the house. The tightening of the floor more than counteracts the increased driving force caused by exhaust ventilation. Crawl-space radon concentration, which should be largely unaffected by exhaust ventilation if the crawl space is vented, is shown to have the greatest influence on indoor radon concentrations. Therefore, ample ventilation of crawl spaces is recommended for houses with and without exhaust ventilation. A conclusion of this analysis is that exhaust ventilation is generally suitable, from the perspective of indoor radon, for houses with a full, vented crawl space as long as the floor does not have a disproportionate fraction of the total effective leakage area. However, balanced mechanical ventilation, when combined with house and floor tightening, would be expected to lead to even lower indoor radon concentrations.

The results for a house with a basement are more complex. Table 11 only includes results for relatively permeable soils. The model indicates that the rate of pressure driven flow of soil gas into a basement of both exhaust-ventilated and naturally-ventilated houses will be only a few hundred liters per hour if the soil permeability is lower than approximately 10^{-7} cm^2. These low soil gas flow rates will significantly affect the indoor radon concentration only if the soil gas radon concentration is unusually high (e.g., several thousand pCi/l). If diffusion and outdoor air are negligible sources of radon, the indoor radon concentration will scale directly with the soil gas radon concentration. If the soil permeability is less than approximately 10^{-8} cm^2, pressure driven flow of soil gas into a house should not cause problems even when soil gas radon concentrations are unusually high. As illustrated in Figure 8, many soils have a permeability that is less than 10^{-7} or 10^{-8} cm^2. Therefore, exhaust ventilation appears to be a suitable technology for houses with basements that are surrounded with a low-permeability soil. The presence of a layer of highly permeable aggregate beneath the basement floor, is one factor that might alter this conclusion.

This analysis also indicates that exhaust ventilation combined with house tightening could lead to substantial increases in the soil gas entry rate if the basement is surrounded with soil that has a permeability greater than approximately 10^{-6} cm^2. Therefore, exhaust ventilation is not recommended for houses with basements located in highly permeable soils unless some process is employed to limit soil gas entry.

The last four rows of Table 11, illustrate the impact of one possible process for limiting soil gas entry. As indicated, a reduction in the width of the wall-to-floor gap from 2 mm to 0.5 mm can counteract the impacts of exhaust ventilation. Although a reduction in gap width may not be a practical alternative, a similar effect would be expected from other processes which increase the resistance to flow through the slab. Examples of such processes are sealing of exposed sections of the wall-to-floor gap (thus reducing gap length) and sealing other penetrations through the slab, or, in new housing, using construction techniques that minimize penetrations to the soil. Development and evaluation of these processes would be useful for radon control in general, and might expand the range of conditions for which exhaust ventilation has no significant adverse impacts on the indoor radon concentration.

PRACTICAL ASPECTS OF EXHAUST-AIR HEAT PUMP APPLICATIONS

Availability

Exhaust-air heat pumps are technically very similar to heat-pump water heaters, the major difference being the addition of ducted air supply to and exhaust from the exhaust air heat pump, and also a lower flow rate of air in the exhaust-air case. In 1984, there were at least fifteen U.S. manufacturers of heat-pump water heaters (EPRI Report, 1984). Most models were of the remote type (for application to existing water tanks), but four manufactures also made integral units. One manufacturer (DEC International, Therma-Stor Products Group) has recently modified their integral heat-pump water heater for exhaust-air service. We expect other manufacturers to follow DEC's example, and quickly develop exhaust-air models as the market develops. We also expect foreign, particularly Canadian manufacturers to enter the U.S. market. An example is Fiberglass Canada Inc. that presently markets the Swedish Elektrostandard exhaust-air heat pump (integral unit) in Canada, but is expected to start production in Canada and to enter the U.S. market.

Although many of the manufacturers of heat-pump water heaters have discontinued their production, we believe that they can re-enter a developing exhaust-air heat pump market relatively quickly. Therefore, availability should be no obstacle to increased use of exhaust-air heat pumps.

Reliability and Service Life

Data on reliability and service life of exhaust-air heat pumps are scarce because of the novelty of the product. Even the service life of heat-pump water heaters is not known accurately (EPRI Report, 1984). However, we will make some general observations regarding reliability, and predict the service life of exhaust-air heat pumps.

We believe integral units are significantly more reliable than remote units because of the additional water pump-around loop in the remote units. The water drawn from the bottom of the tank is expected to contain high amounts of lime that with time can reduce, or even stop water flow to the condenser. Use of auxiliary resistance heating in the tank will promote liming due to higher local temperatures relative to operation with the heat pump only. Hence, the liming problem can be remedied either by avoiding use of resistance heat, or by periodic drain and flushing of the tank. With respect to the liming problem, fan-coil water loops are less sensitive because their water is generally drawn from a higher level in the tank (top in the simulations).

The trend toward integral exhaust-air heat pump units is also supported by the Scandinavian experience. In Sweden, Flakt AB and Bahco AB, both reputable manufacturers of HVAC equipment, have discontinued production of remote exhaust-air heat-pump units, while several manufacturers of integral units are competing with very similar products (Elektrostandard AB, Stromberg AB, and Nilan AB)

We predict the median service life of integral exhaust-air heat pumps to be approximately 10 years, i.e. 50% of all units will require full replacement by the time they reach 10 years of age. One input for this prediction is the fact that the exhaust-air heat pump can be considered a combination of an electric water heater and a refrigerator. The median service life of a refrigerator is 13 years, and that of an electric water heater is 12 years (Appliance, September 1985). If both service lives are assumed to follow convex distributions, and if failures of the "refrigerator component" and of the "water-heater component" are additive, the median service life of the combination is found to be 9 years. In reality, a somewhat longer service life is expected, because the electric water heater part probably has a service life longer than 12 years because of much reduced operation of the auxiliary resistance heater which is the main source of lime formation. In addition, water tanks used for heat pumps may be of better quality than average water heater tanks. A 10-year useful life has also been quoted for heat-pump water heaters (Calm, 1984).

The exhaust fan is predicted to require replacement every five years on the average (44,000 hrs of operation), but such replacement is a simple operation.

<u>Consumer Acceptance</u>

In a field study of heat-pump water heaters operating in "flip-flop" mode with ordinary resistance heating it has been reported (EPRI Report, 1984) that most occupants could not tell when the heat pump was operating and when it was not. This positive result implies that the heat pumps had been located properly because fan and compressor operation produces significant noise. Calm, 1984 reports operating heat-pump water heaters to produce a sound level of approximately 65 decibels five feet from the unit, or about the same as an average room air conditioner. The exhaust-air heat pump is expected to produce a lower sound level because both fan and compressor are contained in an insulated cabinet and because the air flow rate through the exhaust-air heat pump is substantially lower. Flakt (Flakt AB, 1984) reports a sound level of only 37 dB (A scale) in the installed space. Since the exhaust-air heat-pump will usually be located in an unoccupied space, noise is not expected to be an obstacle for consumer acceptance. In fact, one Scandinavian manufacturer (Stromberg AB) is confident enough in noise-free operation to recommend installation of their unit in the kitchen. The Stromberg unit is totally enclosed in one tall cabinet of quite attractive finish with all controls located in a range-hood unit that is supplied with the heat pump.

We also believe that customers will respond favorably to quiet continuous exhaust-venting from bathrooms, especially when compared to noisy ventilation by intermittently operated fans. Range-hood venting would be a great benefit, but we know of only one manufacturer (Stromberg AB of Finland) that recommends range-hood venting through the heat pump (Since they also supply the range hood, they can make sure that a high-quality grease filter is used). For most exhaust-air heat pumps however, the kitchen exhaust should not be located right at the range, but rather high near the ceiling.

CONCLUSIONS

From our 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. For a daily hot water usage of 300 l, the basic heat pump (without fan coil) saves approximately 3800 kwh in Portland, 4400 kwh in Spokane, and 5300 kwh in Missoula, as compared to naturally ventilated houses with a 50% higher average ventilation rate for indoor air quality purposes. The energy savings increase with increased hot water usage, but not linearly. 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; for an MCS house of 125 m^2 floor area, it saves approximately 1000 kwh in Portland, 1500 kwh in Spokane, and 1800 kwh in Missoula over the basic heat-pump case. Fan-coil energy savings increase with increased size of the house, and with reduced insulation.

The cost of conserved energy when comparing a heat-pump MCS house (with fan coil) to the same house with natural ventilation is 3.8 ¢, in Missoula, 4.5¢/kwh in Spokane, and 5.4¢/kwh in Portland. Hence, the potential of exhaust-air heat pumps is greatest in cold climates. From the homeowner's point of view, exhaust air heat pumps are presently economically attractive only in the colder regions of the Pacific Northwest. In the moderate climate of Portland, a small net loss is more likely than a net 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 above inflation would make an investment in a heat-pump system economically attractive in the coastal region of the Pacific Northwest.

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 an advantage when considering indoor air pollution including radon in crawl-space houses. In houses with basements however, mechanical exhaust ventilation coupled with house tightening increases indoor radon concentration when the basement is surrounded with very permeable soil (permeability greater than approximately 10^{-6} cm^2). Therefore, exhaust ventilation is not recommended for houses with basements located in highly permeable soils unless some measure is taken to limit soil gas entry such as sealing of basement wall-to-floor gaps.

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TABLE 1.

House Characteristics

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	$W/m^2 \cdot {}^\circ C$	0.18 (0.45 for STD house)
Ceiling U-Value	$W/m^2 \cdot {}^\circ C$	0.15 (0.28 for STD house)
Wall U-Value	$W/m^2 \cdot {}^\circ C$	0.20 (0.50 for STD house)
Window U-Value	$W/m^2 \cdot {}^\circ C$	1.7 (3.1 for STD house)
Space Heating		
Space Heater Power Input	kW	15(10 in Portland)
Space Heater ON Temperature	°C	20.25
Space Heater 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
Water Heater Power Input	kW	4.5
Water Heater ON Temperature	°C	48
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	600 (nominal)
Coefficient of Performance (COP)		2-3 (yearly average = 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	50
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	50

Table 2. Natural Ventilation Characteristics.

Location/ House Type	Winter Average Ventilation ~0.5 ach		Winter Average Ventilation ~0.7 ach	
	Effective Leakage Area, cm^2	Time with ventilation less than 0.25 ach % year	Effective Leakage Area, cm^2	Time with ventilation less than 0.25 ach % year
Portland/MCS	460	6.8	710	0.2
Portland/STD	460	9.8	710	0.7
Spokane/MCS	410	6.0	640	0.3
Spokane/STD	410	7.6	640	0.5
Missoula/MCS	530	2.6	840	0.3
Missoula/STD	530	5.0	840	0.8

Table 3. Yearly Energy Use in Portland, Oregon

House/Ventilation Strategy & Studied Parameter	Winter Ventilation Rate ACH	Heat Pump Compressor On-Time % of year	Fan Coil On-Time % of year	Yearly System COP	<—POWER CONSUMPTION—> Electric Water Htr. kwh	Electric Space Htr. kwh	Heat Pump & Fan Coil (inc.Exh.Fan) kwh	Lights Appliance kwh	Total kwh
MCS/NV	0.51	---	---	---	4920	4500	---	3650	13,070
MCS/NV 50% Higher Ventilation Rate	0.76	---	---	---	4920	5950	---	3650	14,520
			(For 50% Less Water Usage:		2460				12,060)
			(For 50% More Water Usage:		7380				16,980)
MCS/Exhst Fan Only	0.49	---	---	---	4920	4490	590	3650	13,650
MCS/HP	0.49	39	---	1.85	200	4150	2730	3650	10,730
MCS/HP 60% Higher Exhst Rate	0.71	39	---	1.76	200	5560	2880	3650	12,290
MCS/HP + FC	0.49	49	22	1.88	410	2180	3380	3650	9,620
MCS/HP + FC 60% Higher Exhst Rate	0.71	52	26	1.85	450	3130	3570	3650	10,800
MCS/HP + FC 50% Less Water Usage	0.49	38	25	1.71	-0-	2040	2770	3650	8,460
MCS/HP + FC 50% More Water Usage	0.49	56	20	1.99	1330	2290	3730	3650	11,000
MCS/HP + FC NSDN Water Usage Schedule	0.49	52	23	1.89	510	2100	3530	3650	9,790
MCS/HP + FC 33% Smaller Compressor	0.49	61	18	1.79	850	2640	2930	3650	10,070
MCS/HP + FC 33% Larger Compressor	0.49	41	23	1.92	190	2020	3630	3650	9,490
MCS/HP + FC 33% Lower COP	0.49	61	18	1.30	850	2640	4050	3650	11,190
MCS/HP + FC 33% Higher COP	0.49	41	23	2.41	190	2020	2900	3650	8,760
MCS/HP + FC 5°C Lower Water Temperature	0.49	45	24	1.91	230	2290	3080	3650	9,250
MCS/HP + FC 5°C Higher Water Temperature	0.49	53	21	1.84	690	2100	3620	3650	10,060
STD/NV	0.51	---	---	---	4920	9660	---	3650	18,230
STD/NV 50% Higher Ventilation Rate	0.76	---	---	---	4920	11200	---	3650	19,770
STD/HP + FC	0.49	56	34	1.88	560	5980	3770	3650	13,960
STD/HP + FC 60% Higher Exhst Rate	0.71	58	37	1.87	580	7160	3900	3650	15,290

Table 4. Yearly Energy Use in Spokane, Washington

House/Ventilation Strategy & Studied Parameter	Winter Ventilation Rate ACH	Heat Pump Compressor On-Time % of year	Fan Coil On-Time % of year	Yearly System COP	POWER CONSUMPTION: Electric Water Htr. kwh	Electric Space Htr. kwh	Heat Pump & Fan Coil (inc.Exh.Fan) kwh	Lights Appliance kwh	Total kwh
MCS/NV	0.50	---	---	---	4920	7250	---	3650	15,820
MCS/NV 50% Higher Ventilation Rate	0.75	---	---	---	4920	9400	---	3650	17,970
			(For 50% Less Water Usage:		2460				15,510)
			(For 50% More Water Usage:		7380				20,430)
MCS/Exhst Fan Only	0.49	---	---	---	4920	7440	570	3650	16,580
MCS/HP	0.49	39	---	1.86	200	7080	2720	3650	13,650
MCS/HP 60% Higher Exhst Rate	0.71	39	---	1.78	200	9110	2850	3650	15,810
MCS/HP + FC	0.49	52	27	1.89	500	4440	3520	3650	12,110
MCS/HP + FC 60% Higher Exhst Rate	0.71	55	32	1.87	540	5980	3700	3650	13,870
MCS/HP + FC 50% Less Water Usage	0.49	42	31	1.74	-0-	4210	2990	3650	10,850
MCS/HP + FC 50% More Water Usage	0.49	58	25	1.99	1450	4590	3840	3650	13,530
MCS/HP + FC NSDN Water Usage Schedule	0.49	54	29	1.89	590	4350	3660	3650	12,250
MCS/HP + FC 33% Smaller Compressor	0.49	63	22	1.80	950	5080	3010	3650	12,690
MCS/HP + FC 33% Larger Compressor	0.49	44	29	1.93	240	4180	3820	3650	11,890
MCS/HP + FC 33% Lower COP	0.49	63	22	1.30	950	5080	4180	3650	13,860
MCS/HP + FC 33% Higher COP	0.49	44	29	2.43	240	4180	3030	3650	11,100
MCS/HP + FC 5°C Lower Water Temperature	0.49	47	29	1.92	280	4610	3200	3650	11,740
MCS/HP + FC 5°C Higher Water Temperature	0.49	56	26	1.85	790	4300	3800	3650	12,540
STD/NV	0.50	---	---	---	4920	14810	---	3650	23,380
STD/NV 50% Higher Ventilation Rate	0.75	---	---	---	4920	17050	---	3650	25,620
STD/HP + FC	0.49	60	41	1.89	600	10400	3990	3650	18,640
STD/HP + FC 60% Higher Exhst Rate	0.71	62	43	1.89	610	12230	4080	3650	20,570

Table 5. Yearly Energy Use in Missoula, Montana

House/Ventilation Strategy & Studied Parameter	Winter Ventilation Rate ACH	Heat Pump Compressor On-Time % of year	Fan Coil On-Time % of year	Yearly System COP	Electric Water Htr. kwh	Electric Space Htr. kwh	Heat Pump & Fan Coil (inc.Exh.Fan) kwh	Lights Appliance kwh	Total kwh
					<——— POWER CONSUMPTION				———>
MCS/NV	0.48	---	---	---	4920	8700	---	3650	17,270
MCS/NV 50% Higher Ventilation Rate	0.74	---	---	---	4920	11410	---	3650	19,980
			(For 50% Less Water Usage:		2460				17,520)
			(For 50% More Water Usage:		7380				22,440)
MCS/Exhst Fan Only	0.46	---	---	---	4920	8530	590	3650	17,690
MCS/HP	0.46	39	---	1.85	200	8120	2740	3650	14,710
MCS/HP 60% Higher Exhst Rate	0.69	39	---	1.75	200	10550	2880	3650	17,280
MCS/HP + FC	0.46	55	32	1.88	540	5070	3690	3650	12,950
MCS/HP + FC 60% Higher Exhst Rate	0.69	57	36	1.87	580	6970	3870	3650	15,070
MCS/HP + FC 50% Less Water Usage	0.46	45	36	1.74	-0-	4810	3200	3650	11,660
MCS/HP + FC 50% More Water Usage	0.46	60	30	1.98	1520	5260	3970	3650	14,400
MCS/HP + FC NSDN Water Usage Schedule	0.46	57	34	1.89	630	4980	3820	3650	13,080
MCS/HP + FC 33% Smaller Compressor	0.46	65	26	1.79	1010	5810	3120	3650	13,590
MCS/HP + FC 33% Larger Compressor	0.46	46	33	1.93	260	4800	4000	3650	12,710
MCS/HP + FC 33% Lower COP	0.46	65	26	1.29	1010	5810	4330	3650	14,800
MCS/HP + FC 33% Higher COP	0.46	46	33	2.42	260	4800	3180	3650	11,890
MCS/HP + FC 5°C Lower Water Temperature	0.46	49	34	1.91	300	5290	3350	3650	12,590
MCS/HP + FC 5°C Higher Water Temperature	0.46	59	31	1.84	830	4920	3960	3650	13,360
STD/NV	0.48	---	---	---	4920	17290	---	3650	25,860
STD/NV 50% Higher Ventilation Rate	0.74	---	---	---	4920	20100	---	3650	28,670
STD/HP + FC	0.46	62	45	1.88	650	12060	4140	3650	20,500
STD/HP + FC 60% Higher Exhst Rate	0.69	64	47	1.88	660	14260	4230	3650	22,800

Table 6. Yearly energy savings in kwh realized with exhaust air systems at 0.5 ach relative to natural ventilation at 0.7 ach.

	MCS House				STD House		
	Exhst Fan Only	HP	+	FC	HP	+	FC
Portland	870	3790	+	1110	3880	+	1930
Spokane	1390	4320	+	1540	4410	+	2570
Missoula	2290	5270	+	1760	5370	+	2800

Table 7. 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
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 8. Current Electric Rate Structure

PORTLAND		SPOKANE		MISSOULA	
3.84¢/kwh	0-300kwh	3.40¢/kwh	0-600kwh	5.28¢/kwh	Dec-Mar
4.80¢/kwh	>300kwh	4.11¢/kwh	600-1300kwh	4.06¢/kwh	Apr-Nov
		4.99¢/kwh	>1300kwh		

Table 9. Economic results for houses with heat pumps compared to naturally ventilated houses at 0.7 ach.

Location/ House Ventilation Strategy	Cost of Conserved Energy ¢/kwh	Net Present Benefit $ with Electric Rate Escalation 0%	2%
PORTLAND			
MCS/0.5 ach Exh. Fan Only	9.1	-460	-350
MCS/0.5 ach HP	5.5	-350	100
MCS/0.5 ach HP + FC	5.4	-380	210
STD/0.5 ach HP + FC	4.6	170	860
MCS/0.7 ach HP	7.5	-1280	-1010
MCS/0.7 ach HP + FC	7.1	-1090	-640
STD/0.7 ach HP + FC	6.0	-640	-110
SPOKANE			
MCS/0.5 ach Exh. Fan Only	5.7	-130	40
MCS/0.5 ach HP	4.9	-130	360
MCS/0.5 ach HP + FC	4.5	80	760
STD/0.5 ach HP + FC	3.8	880	1710
MCS/0.7 ach HP	9.7	-1400	-1160
MCS/0.7 ach HP + FC	6.5	-950	-470
STD/0.7 ach HP + FC	5.3	-300	310
MISSOULA			
MCS/0.5 ach Exh. Fan Only	3.4	420	710
MCS/0.5 ach HP	4.0	480	1100
MCS/0.5 ach HP + FC	3.8	840	1670
STD/0.5 ach HP + FC	3.3	1490	2450
MCS/0.7 ach HP	7.8	-1050	-740
MCS/0.7 ach HP + FC	5.4	-440	140
STD/0.7 ach HP + FC	4.5	100	780

Table 10 Predicted indoor radon concentrations, flow rates from the crawl space to the house, and related parameters for a house with a vented crawl space, with and without exhaust ventilation. Assumed indoor-to-outdoor temperature difference is 20°C and wind speed is 3 m/s.

Total ELA cm^2	Floor ELA cm^2	Exhaust Flow Rate m^3/h	Air Exchange Rate h^{-1}	Pressure Diff. Across Floor pa	Flow Rate Through Floor m^3/h	Indoor Rn Conc. pCi/l with Rn_{cs}=5*	Rn_{cs}=10	Rn_{cs}=50
710	246	0	0.58	1.69	148	4.2	8.4	41.7
460	159	0	0.38	1.69	96	4.2	8.4	41.7
250	86	0	0.20	1.69	52	4.2	8.4	41.7
250	86	125	0.46	2.85	68	2.5	5.0	24.3

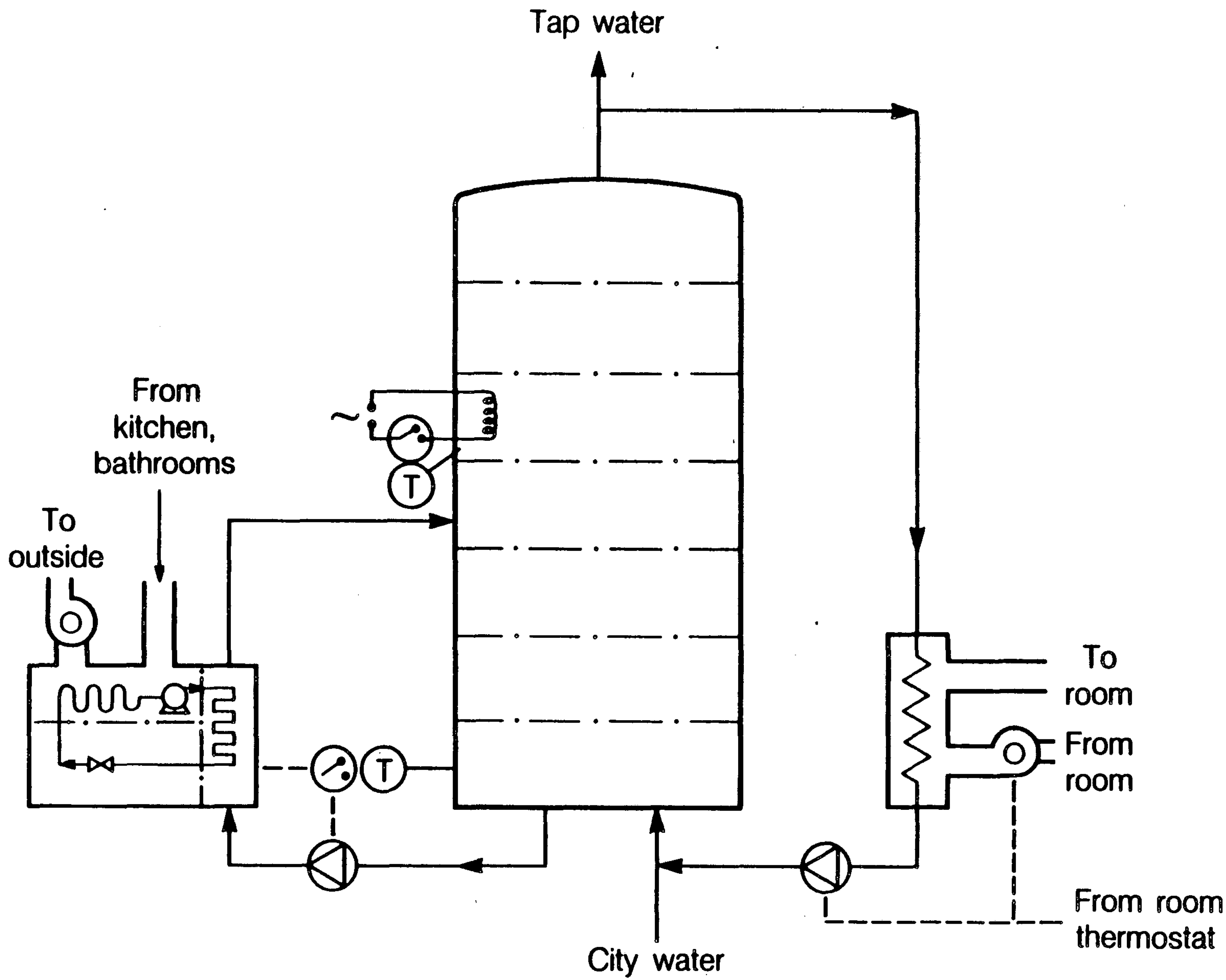
|<——Inputs to Model——>|<——————————Outputs from Model——————————>|

*Values for Rn_{CS}, the radon concentration in the crawl space, are inputs to the model.

Table 11 Predicted indoor radon concentrations, soil gas entry rates, and related parameters for a house with a basement and wall-to-floor gap, with and without exhaust ventilation. The soil gas radon concentration was assumed to equal 700 pCi/l.

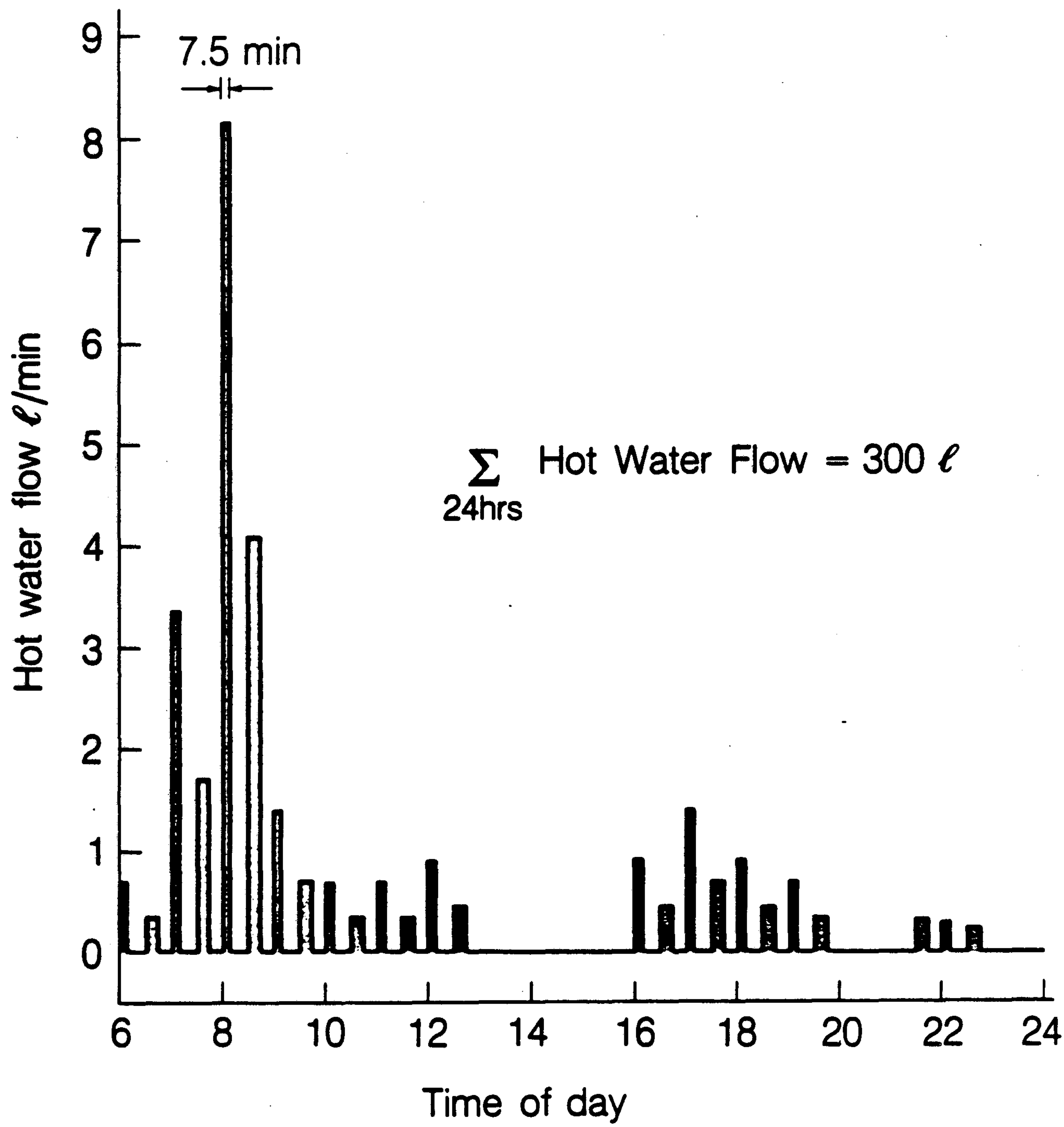
Soil Permeability cm^2	Total ELA cm^2	Exhaust Flow Rate m^3/h	Gap Width mm	Air Exchange Rate h^{-1}	Effective Pressure Diff. pa	Soil Gas Entry Rate m^3/hr	Indoor Rn conc. pCi/l
10^{-4}	1420	0	2.0	0.75	3.6	78.2	119
" "	920	" "	" "	0.49	" "	" "	184
" "	500	" "	" "	0.26	" "	" "	338
10^{-5}	1420	" "	" "	0.75	" "	13.2	20
" "	920	" "	" "	0.49	" "	" "	31
" "	500	" "	" "	0.26	" "	" "	57
10^{-6}	1420	" "	" "	0.75	" "	1.42	2.4
" "	920	" "	" "	0.49	" "	" "	3.6
" "	500	" "	" "	0.26	" "	" "	6.4
10^{-7}	1420	" "	" "	0.75	" "	0.14	0.5
" "	920	" "	" "	0.49	" "	" "	0.6
" "	500	" "	" "	0.26	" "	" "	0.9
10^{-4}	500	250	" "	0.49	4.8	103	242
" "	250	" "	" "	0.43	8.3	178	473
10^{-5}	500	" "	" "	0.49	4.8	17.4	41
" "	250	" "	" "	0.43	8.3	30.0	80
10^{-6}	500	" "	" "	0.49	4.8	1.87	4.6
" "	250	" "	" "	0.43	8.3	3.22	8.8
10^{-7}	500	" "	" "	0.49	4.8	0.19	0.7
" "	250	" "	" "	0.43	8.3	0.32	1.1
10^{-5}	500	" "	0.5	0.49	4.8	1.96	4.8
" "	250	" "	" "	0.43	8.3	3.37	9.2
10^{-6}	500	" "	" "	0.49	4.8	0.94	2.5
" "	250	" "	" "	0.43	8.3	1.62	4.6

|<——Inputs to Model——>|<——Output from Model——>|



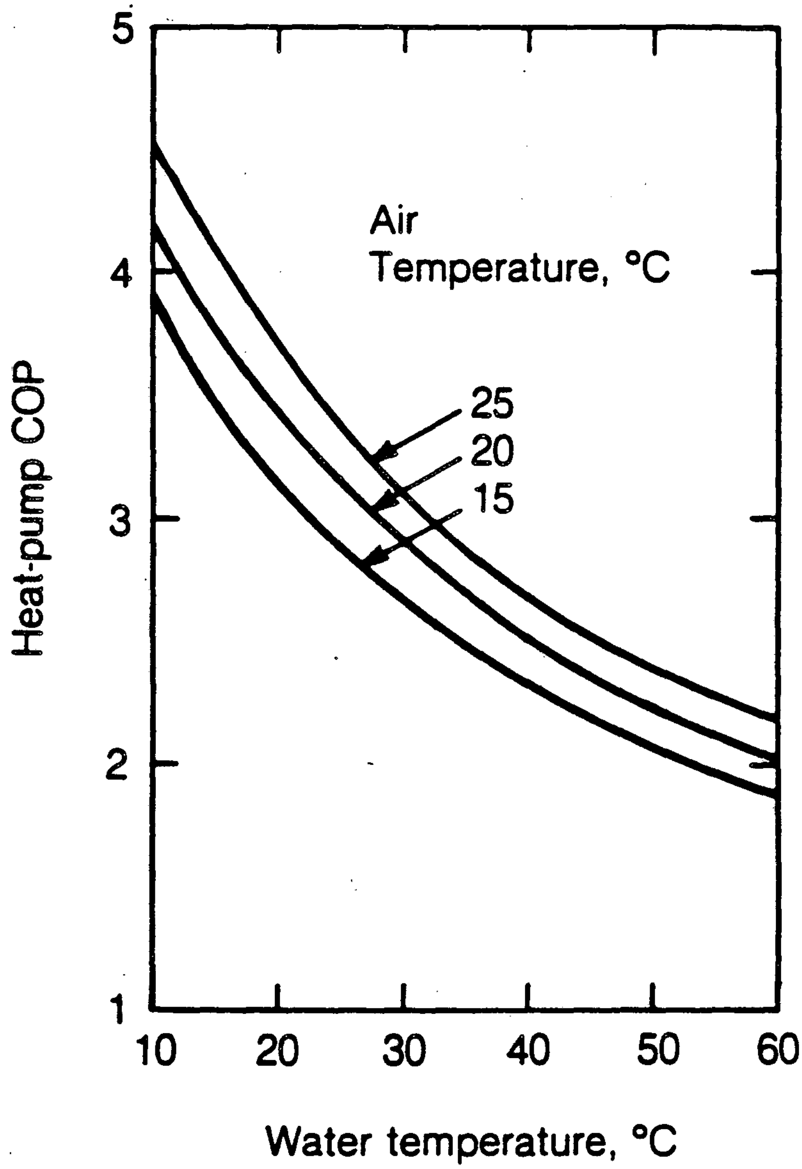
XBL 866-9833

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.



XBL 864-11111

Fig. 2 Assumed hot water usage schedule.



XBL 864-11110

Fig. 3 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 for an exhaust ventilation rate of 125 m^3/hr. For an exhaust ventilation rate of 200 m^3/hr, COP curves are assumed 12% higher.

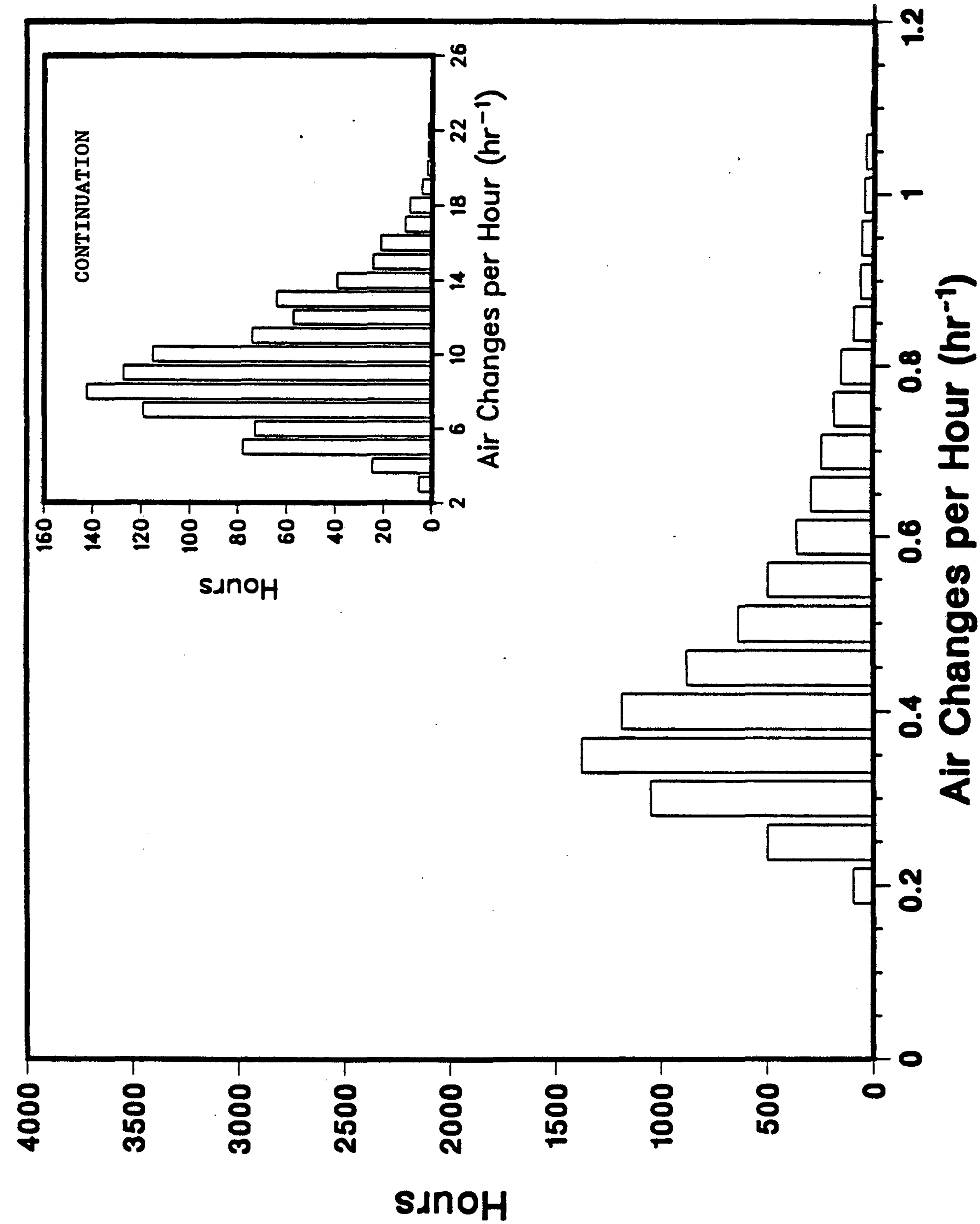


Fig. 4 Time histogram of ventilation rates in naturally ventilated MCS Portland house with an effective leakage area of 460 cm^2. Winter mean ventilation rate is 0.51 ach.

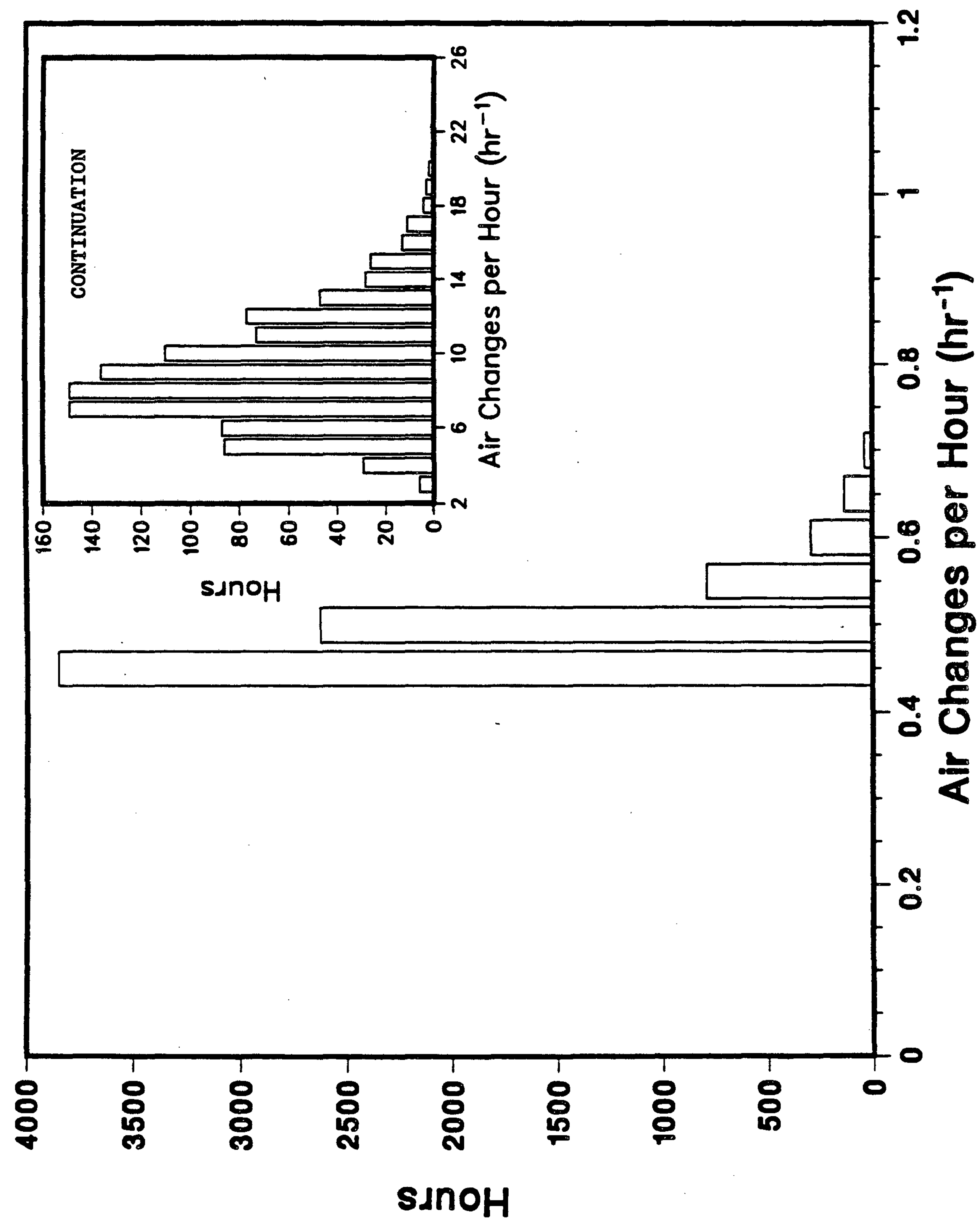
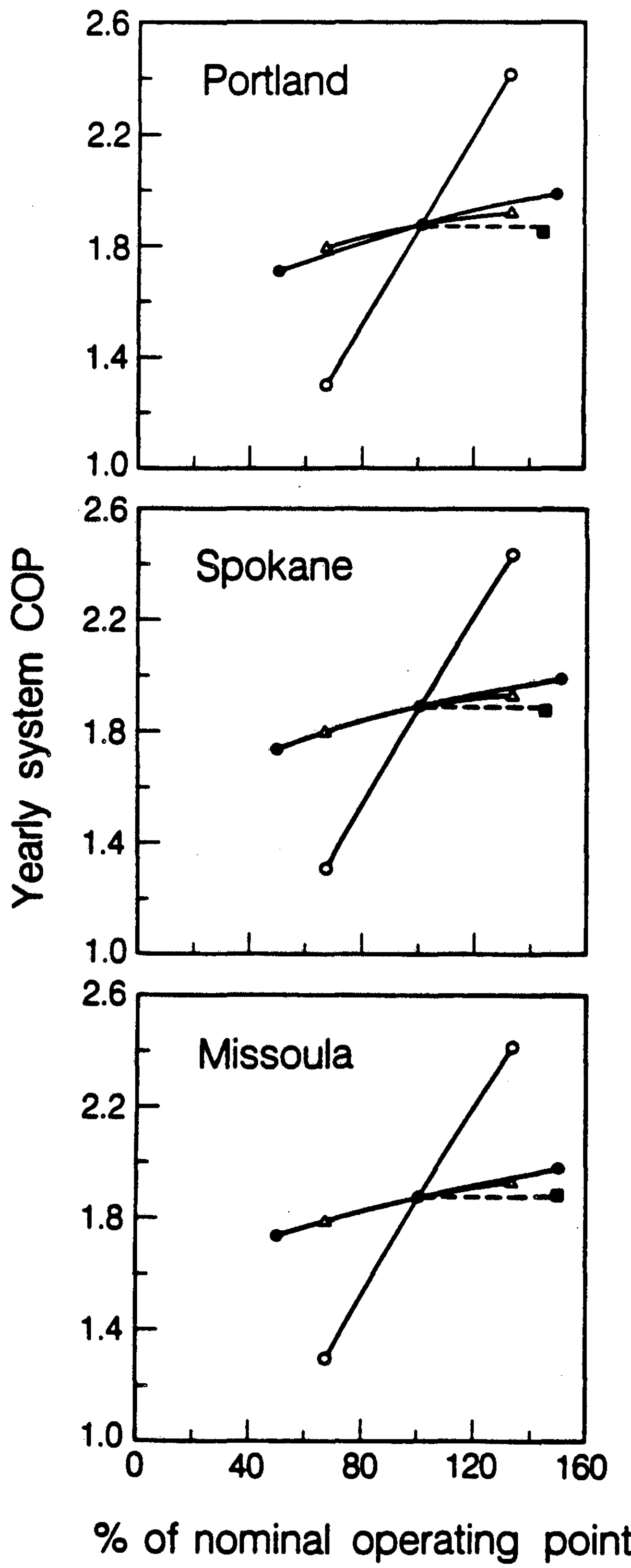


Fig. 5 Time histogram of ventilation rates in MCS Portland house equipped with mechanical ventilation. Exhaust-air flow rate is $125 m^3/hr$ and the effective leakage area is 250 cm^2. Winter mean ventilation rate is 0.49 ach.



Symbol	Variable	Nom. operating point
○	HP average COP	2.4
△	HP capacity	600 W compressor
●	Hot water usage	300 ℓ/24 hrs
■	House size	125 m^2

XBL 869-9909

Fig. 6 Sensitivity of yearly system COP (= Condenser heat flux/HP + FC total power input including fans) to system parameters in MCS/HP + FC houses.

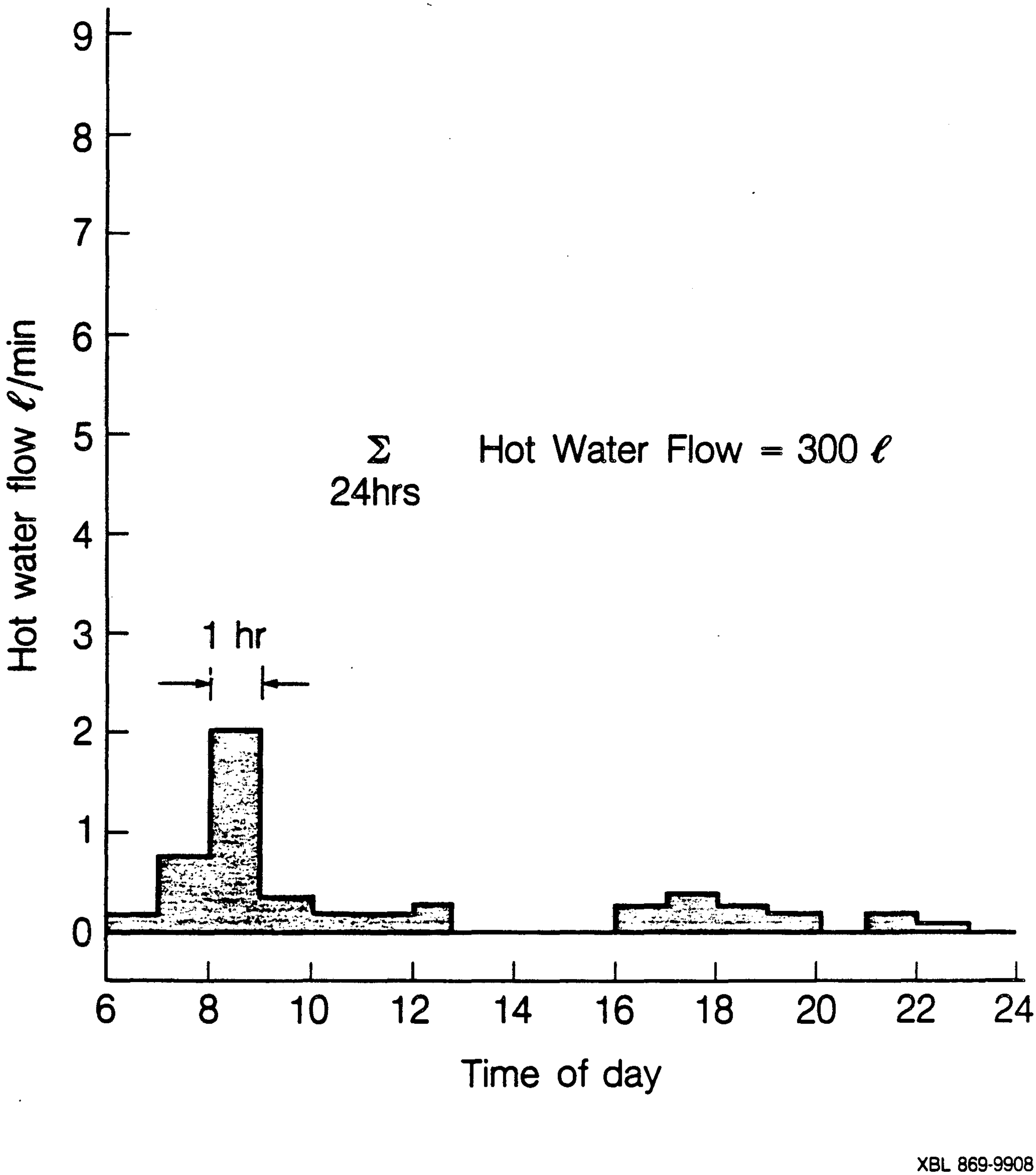


Fig. 7 Alternative hot water usage schedule.

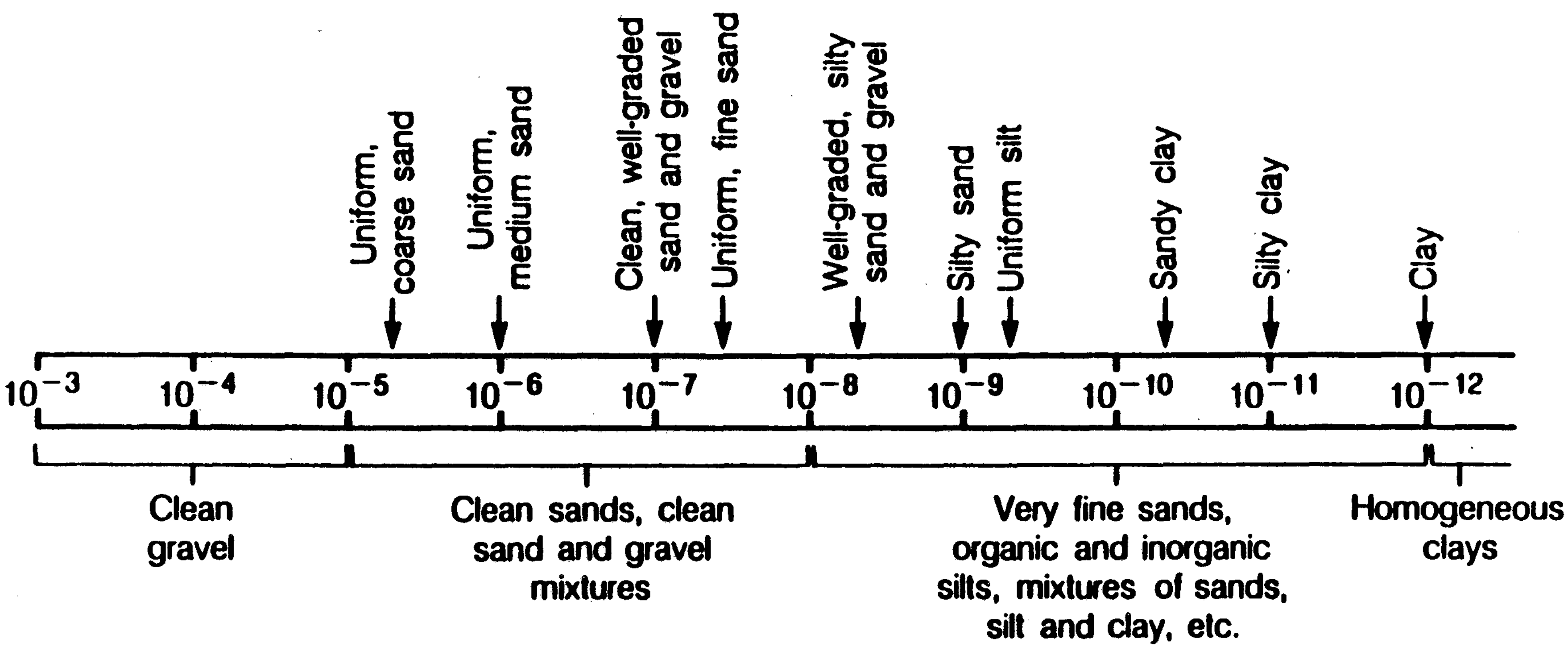


Fig. 8 Typical soil permeabilities, cm^2.