Energy and Buildings, 9 (1986) 239 - 251

Impacts of Ventilation Strategies on Energy Consumption and Indoor Air Quality in Single-family Residences

DARIUSCH HEKMAT, HELMUT E. FEUSTEL and MARK P. MODERA

Energy Performance of Buildings Group, Lawrence Berkeley Laboratory, Berkeley, CA 94720 (U.S.A.)

(Received June 2, 1985; accepted April 8, 1986; in revised form June 10, 1986)

SUMMARY

This report compares the impacts of five different ventilation strategies on the overall energy consumption of superinsulated houses in the Northwestern United States. The strategies examined are: (1) natural ventilation, (2) balanced ventilation with an air-to-air heat exchanger, (3) exhaust ventilation without heat recovery, (4) exhaust ventilation connected to a heat pump to provide space heating, and (5) exhaust ventilation connected to a heat pump to heat domestic water. A modified Transient System Simulation (TRNSYS) residential load model incorporating the Lawrence Berkeley Laboratory (LBL) infiltration model, and a modified TRNSYS domestic hot water model, are used to simulate the energy consumption associated with each strategy. The domestic hot water model is used to determine the amount of useful heat supplied by an exhaust ventilation heat pump as a function of hot water demand schedule and storage tank size. The simulations are made for cities with: (1) a moderate coastal climate, (2) a windy cold climate, and (3) a calm cold climate. They show that total energy consumption (space heat + domestic hot water) can be reduced by 9 to 21% by using mechanical ventilation systems with heat recovery. These savings, compared with energy savings of 18 to 21% achieved by superinsulating the same houses, indicate that the choice of ventilation strategy can have a significant effect on energy consumption. The comparisons also show that for the same effective ventilation rate, houses with mechanical ventilation systems (especially those with exhaust fans) have uniform ventilation and therefore better indoor air quality.

Key words: air-to-air heat exchanger, computer simulation, domestic water heating, exhaust air heat pump, heat recovery, mechanical ventilation, space heating.

INTRODUCTION

Over the past ten years, significant efforts have been made to tighten and insulate houses to conserve energy. However, tightening the envelope of a house has an important impact on the ventilation of that house, and therefore on its indoor air quality. Conventional houses in the United States have leaky envelopes, such that ventilation is provided naturally, namely by infiltration driven by wind and stack effects. As the building envelope is tightened to reduce the average infiltration, the infiltration (natural ventilation) can become very low under mild weather conditions, possibly causing indoor air quality problems [1].

To meet the combined goals of reducing the heat loss due to infiltration and maintaining acceptable indoor air quality, different strategies can be employed. A common strategy employed in the United States is to tighten the envelope so as to obtain a given average natural ventilation rate (e.g., 0.5 air changes per hour). A more sophisticated strategy is to tighten the house as much as possible and install a mechanical ventilation system. There are several options for this strategy, all of which require a rather tight

0378-7788/86/$3.50

building envelope. The most popular option in the United States is to install two balanced fans whose air streams are connected via an air-to-air heat exchanger. In this system, flow rates are set to provide a given average ventilation rate, and the incoming outdoor air is preheated by the exhaust air stream. Another mechanical ventilation option, commonly used in Scandinavian countries, employs an exhaust fan with heat recovery. The exhaust fan depressurizes the house, drawing outdoor air into the house either through leaks in the envelope or through specially designed vents. The heat is recovered from the exhaust stream by coupling it to a heat pump that can be used either for space heating or for domestic water heating [2]. In some instances exhaust fans are used without heat recovery, simply to provide a more uniform ventilation rate.

Our goal in this study is to examine the impacts of these ventilation strategies on the total energy consumption of single-family residences located in three cities in the northwest United States. The cities chosen include: (1) a moderate coastal climate, (2) a windy cold climate, and (3) a calm cold climate. Using an hour-by-hour residential building simulation model, we shall examine five ventilation strategies: (1) natural ventilation, (2) balanced ventilation with an air-to-air heat exchanger, (3) exhaust ventilation without heat recovery, (4) exhaust ventilation connected to a heat pump that provides space heating, and (5) exhaust ventilation connected to a heat pump that heats domestic hot water.

VENTILATION

Natural ventilation (or infiltration) is caused by the interaction of the building envelope with pressure differences caused by wind and by indoor-outdoor temperature differences (stack effect). We use a simplified infiltration model to determine the natural ventilation rate as a function of weather conditions [3, 4]. In this model, the equation used to add the ventilation rates obtained from wind speeds and temperature differences is:

$$Q_{\text{nat}} = (Q_{\text{wind}}^2 + Q_{\text{stack}}^2)^{1/2} \tag{1}$$

where

Q_{nat} is the natural infiltration (m^3/h)
Q_{wind} is the infiltration rate due to wind effect (m^3/h)
Q_{stack} is the infiltration due to stack effect (m^3/h).

In the case where natural ventilation is supplemented by a mechanical ventilation system, eqn. 1 takes the form [5, 6]:

$$Q_{\text{tot}} = (Q_{\text{nat}}^2 + Q_{\text{unbal}}^2)^{1/2} + Q_{\text{bal}} \tag{2}$$

where

Q_{tot} is the total infiltration (m^3/h)
Q_{unbal} is the air-flow rate of an unbalanced fan (m^3/h)
Q_{bal} is the air-flow rate through a balanced fan system (m^3/h).

Equation 2 shows that balanced flows add simply to the total infiltration, whereas unbalanced flows add in quadrature. The reason for this is that unbalanced flows change the internal pressure of the house and therefore interact with wind-induced and stack-induced flows.

Balanced-flows mechanical ventilation systems have two air streams driven by a supply fan and an exhaust fan. The two streams are usually connected by means of an air-to-air heat exchanger which transfers heat from the warm air stream to the cold air stream with little or no mixing. One problem with these systems is that the moisture contained in the exhaust air stream sometimes freezes in the core of the heat exchanger. Another problem is that ventilation effectiveness is often reduced by short-circuiting of exhaust and intake air streams within the house (i.e., unless the air streams are ducted, re-entrainment may occur, thereby preventing full mixing) [7, 8].

Exhaust ventilation systems usually depressurize the house with a single exhaust fan, thereby sucking outdoor air into the house through the building envelope (In supertight Swedish houses, vents are placed in the envelope to allow air intake). In principle, exhaust ventilation systems have several advantages over balanced air-to-air systems: (1) an exhaust system does not require supply ductwork because fresh air enters the house through leaks distributed over the entire envelope rather than through a single fan, and (2) ventilation peaks and valleys are less pronounced for an exhaust system because it adds in quadrature with natural infiltration. The strategy employed in the Scandinavian countries to extract heat from the exhaust stream is to use a small heat pump that provides either space heating or domestic water heating. Although such a device can extract a large amount of heat from the exhaust air (possibly cooling it below outdoor temperature), its major disadvantages are its complexity and high initial cost.

The main purpose of ventilation is to minimize the concentration of contaminants in the indoor air. For a given contaminant source strength (i.e., rate of contaminant generation), the steady-state concentration of that contaminant is proportional to the inverse ventilation rate, implying that the *average* contaminant concentration is proportional to the *average inverse* ventilation rate. Because indoor air quality is inversely proportional to the average contaminant concentration, we shall use the *inverse* of the *average inverse* ventilation rate as a measure of the indoor air quality resulting from different ventilation strategies. This quantity shall be referred to as the *effective ventilation rate*:

$$Q_{\mathrm{eff}} = 1 \Big/ \left(\sum \frac{1}{Q} \Big/ n \right) \tag{3}$$

As a second measure of indoor air quality, we shall use the statistical spread of the effective ventilation rate, which describes the expected fluctuations. The spread of a log-normal distribution is analogous to the standard deviation of a normal distribution and is defined as:

$$S = \exp \sqrt{\frac{\sum \left(\ln \frac{1}{Q} \right)^2 - \left(\sum \ln \frac{1}{Q} \right)^2 \Big/ n}{n+1}} \tag{4}$$

where

S is the spread factor (dimensionless)
Q is the ventilation rate (ach)
n is the number of points (dimensionless).

For a given effective ventilation rate, the frequency of occurrence of low ventilation rates, and concomitant poor air quality, will increase as the spread increases.

From the energy perspective, the important quantity is the total flow rate of outdoor air into the house. We shall use the average ventilation rate to describe this quantity.

ANALYSIS PROCEDURE

To evaluate the five ventilation strategies, we chose to simulate each of the strategies in a superinsulated ranch-style house [9] at three sites: Portland, Oregon (2840 Heating Degree Days (°C)), which has a moderate coastal climate, Missoula, Montana (4603 HDD (°C)), which has a cold calm climate, and Great Falls, Montana (4497 HDD (°C)), which has a cold windy climate. For each of the strategies, we performed an hour-by-hour simulation of energy consumption and ventilation rate using TMY (Typical Meteorological Year) [10] data for each site. One additional simulation was performed for each site for a naturally-ventilated house that follows typical new construction standards. This simulation allows us to compare the energy impacts of different ventilation strategies with the energy impacts of superinsulating a house. The base case for all comparisons is the naturally-ventilated superinsulated house. (The detailed specifications for all houses are provided in the Appendix.)

The ventilation rates used for the different strategies were adjusted to provide the same air quality (to first order) by assuring that the effective ventilation rates were equal. The effective leakage areas of the houses (a parameter that describes the air tightness of a building envelope, see ref. 3) were chosen to reflect current building practice. The values chosen for the naturally ventilated houses, 700 cm^2 for Portland and Missoula, and 500 cm^2 for Great Falls, provide a typical average ventilation rate, namely 0.5 air changes per hour for the effective ventilation rate. For the mechanically-ventilated houses, values regarded as typical for supertight construction were used to arrive at an effective leakage area of 150 cm^2. The fan flows of the mechanical ventilation systems were then adjusted to assure the same effective ventilation rates for all strategies (i.e., the same as for the naturally-ventilated houses).

For the fifth ventilation strategy, an exhaust fan with a heat pump used to heat domestic water, it was necessary to simulate the hot water demand, inasmuch as the demand and the size of the storage tank determine how much of the required energy can be supplied by the heat pump. The major parameters are: (1) daily hot-water demand profile, (2) total daily hot-water demand, and (3) tank size. For our comparisons we chose two typical tank sizes, two total hot-water demands, and three daily demand profiles to determine the amount of hot-water heating that could be supplied by the heat pump.

COMPUTER SIMULATION

An existing computer simulation program called TRNSYS (Transient System Simulation) [11], developed at the University of Wisconsin at Madison, was chosen for comparing the different ventilation strategies. This program was chosen because it is also well-suited for examining other waste heat utilization strategies for residences. The program consists of a central differential equation solver and a set of independent component modules that can be interconnected for simulating a particular system, thereby allowing a high degree of flexibility. Of particular interest to this comparison of ventilation strategies are its residential load model and its domestic hot water model.

Residential load model

In TRNSYS, the residential load model consists of a roof model and a zone model (see ref. 10) that uses the ASHRAE response-factor method to calculate heat transfer through the envelope [12]. We replaced the air infiltration model used in the TRNSYS zone model with the infiltration model developed at Lawrence Berkeley Laboratory (LBL) (see refs. 4 and 5).

Typical Meteorological Year (TMY) hourly weather tapes were used for each site. The reduced weather data consisted of: (1) direct normal solar radiation, (2) total horizontal solar radiation (standard year corrected), (3) dry-bulb temperature, (4) dew-point temperature, and (5) wind speed.

The following assumptions were made to simplify the analysis:

- the house was modeled as a single zone;
- the crawlspace walls were not insulated and the crawl space was assumed to be at outdoor temperature;
- framing of walls and floors were not taken into account for heat-transfer calculations;
- overhangs were not modeled;
- furnishings were not included (i.e., small thermal mass);
- area ratios of wall surfaces were used to determine view factors for calculating radiation exchange;
- beam radiation through windows assumed to strike only the floor;
- heating setpoint was 20 °C.

The internal gains were specified separately for people and equipment. It was assumed that four people occupy the house and add an average of 65 W sensible heat and 55 W latent heat per person (activity level 2 from Table 18, Chapter 26, *1981 ASHRAE Handbook of Fundamentals*). It was also assumed that 70% of the sensible heat gain from people is radiative. Appliances and lighting were assumed to deliver 4000 kWh/year distributed evenly over the year, 25% of which is radiative gain and 75% of which is convective gain. Standby losses from the domestic hot water tank were also included as internal gains.

For the ventilation strategy that uses an exhaust fan with a heat pump for space heating, we assumed that the heating coefficient of performance (COP) is 3.0 and that it delivers 920 W. We also assumed that the heat pump output of 920 W includes any heat recovered from the 100-W fan. For the ventilation strategy using an air-to-air heat exchanger we made the following assumptions: (1) the heat exchanger has a seasonal heat transfer efficiency of 65% (including freeze-defrost cycles) [13], and (2) it has two 50-W fans, the supply fan located downstream and the exhaust fan located upstream of the heat exchanger core. Thus, 50 W times (1 + 0.65), or 82.5 W, is recovered when the fans operate during the heating season.

Domestic hot water model

The existing TRNSYS water-heater model has a solar collector system as a heat source. This model has been modified to substitute the heat input from an exhaust air heat pump for that supplied by a solar collector. With this substitution, the heat rejected at the condenser of the heat pump and the flow rate of the hot water loop must be specified. The condenser heat rejection was obtained from the specifications of a commercially available heat pump, and the water flow rate was determined from an average heat rejection and the size of the heat exchanger at the condenser. Heat-exchanger design parameters (i.e., UA-value and flow velocity) have to be taken into consideration for this calculation. The input data for the simulation runs are presented in the Appendix.

For all domestic hot water simulations, the following assumptions were made:

- the storage tank is fully mixed;
- the storage tank is located in the heated section of the house and heat losses occur to $T = 20$ °C;
- the feed-water temperature is constant over the year at 13 °C;
- the daily hot water demand profile does not change over the course of the year;
- the heat input from the exhaust air heat pump is either zero (off) or 920 W (on);
- the heat pump can heat water up to T_{max} = 55 °C;
- the minimum hot water delivery temperature required is 60 °C (with the temperature boost provided by an electric resistance heater);
- the dead band for the water temperature controller at the condenser of the exhaust-air heat pump is 4 °C.

Although we know that assumptions 2, 3 and 4 are not quite valid, they were made to keep the level of detail on a par with the other simulations. Limited data from a study on domestic hot water energy consumption indicates that seasonal variations in the energy consumed to heat water can be as high as 40% [14]. These variations are due to feed-water temperature variations, stand-by loss variations (approximately 45% of water heaters in the United States are located in unheated spaces), and variations in the amount of hot water consumed (increases approximately 20% increase in winter). These effects could easily be incorporated in our present simulation framework.

The fifth assumption, that the heat input from the heat pump is constant, is reasonable as long as there are no substantial variations in the temperature at the evaporator and/or condenser of the heat pump, and as long as dynamic effects are minimal. The evaporator temperature is constant if there is no night setback and if the design temperatures successfully avoid freezing on the coils. On the other hand, the condenser temperature depends on the storage tank temperature, which depends on the hot water demand schedule and the size of the tank. From manufacturer's compressor data [15], however, the heat output (450 W nominal input) varies only 17% when the storage tank (i.e. condenser) temperature is reduced from 60 °C to 32.2 °C at a constant evaporator temperature of 4.4 °C. The COP increases from 2.7 to 3.7 and the compressor input drops by about 17%. Taking these considerations into account, we assumed a constant value for the heat supplied by the heat pump as a reasonable first-order approximation.

The sixth assumption helps make the simulations more realistic, that is, we set an upper limit of 55 °C for the temperature at which the heat pump was turned off and used an electric resistance heater to provide the heat required to bring the water up to the 60 °C delivery temperature. This limitation stems from the operating characteristics of the small heat pumps currently available. The useful lifetime of the compressors drops off very quickly as the condenser temperature (and therefore the refrigerant pressure) increases. Thus, the heat pump cycles on and off depending on the storage tank temperature.

The exhaust-air heat pumps used for space heating and domestic hot water were both sized by calculating the amount of heat that could be extracted from the exhaust air without causing freezing at the evaporator of the heat pump. For a supertight house with an exhaust ventilation system, the fan flow rate is 150 m^3/h. Choosing an exhaust-air temperature drop of 11 K to avoid freezing, and using the above flow rate, we obtain approximately 550 W of heat from the exhaust air. For space heating we assumed a heating COP of 3.0, but for water heating the COP is lower due to the higher temperature at the condenser. With a COP of 2.5, a condenser heat output of approximately 920 W results.

Three different hourly hot-water-demand profiles were used for the simulations. These are the Rand Corporation (RAND) profile, The National Solar Data Network (NSDN) profile [16], and an extreme profile in which 80% of the total daily demand occurs during

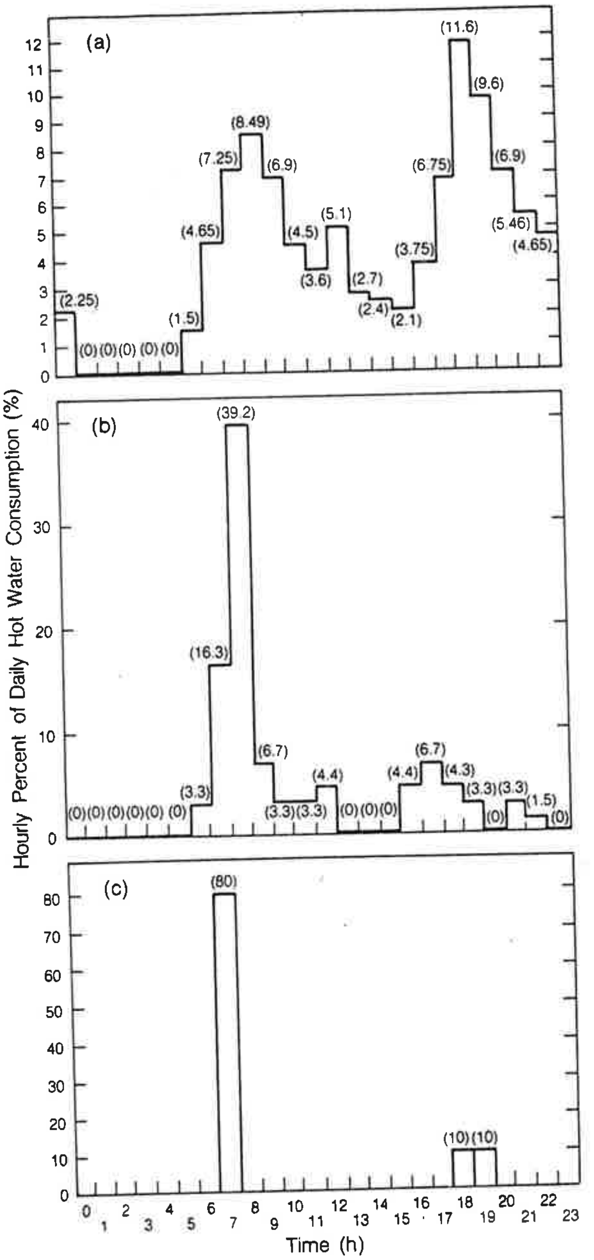

Fig. 1. Hourly profiles of domestic hot water consumption; (a) RAND, (b) NSDN, (c) extreme.

one hour of the day (see Fig. 1). The two total daily hot water demands and storage tank sizes chosen come from ref. 16 and are listed in the Appendix. Varying these parameters changes the hourly temperature profile of the storage tank, and thus the on-time of the heat pump.

Our simulations assume year-round operation of the exhaust fan, but do not take into account the possibility of the heat pump being used to provide space cooling during the summer. If the indoor temperature exceeds a certain comfort level (e.g., 25 °C) the air flow from the exhaust fan could be thermostated to reverse, thereby supplying space cooling as well as hot water heating.

RESULTS

The results of simulating space heating and water heating loads are presented in Tables 1 - 3. These Tables compare the total heating consumptions (space heating + water heating) for the five ventilation strategies in super-insulated houses, and for a house built to typical new construction specifications. An evaluation of the strategies based on the bottom line energy consumptions is only strictly valid for houses with electric heating and hot water. A fair comparison for houses with gas heating or hot water should take into account the price differences between gas and electricity, or possibly compare primary energy consumption to account for the difference between the two forms of energy.

In comparing the ventilation achieved with each of the strategies, and remembering that we kept the effective ventilation rate constant, we find two important results. The first is that the average ventilation rate is equal to the effective ventilation rate for all mechanical ventilation strategies but not for natural ventilation. This finding indicates that the average air flow through the house is higher under natural ventilation, implying that the energy load is higher. This is due to the larger fluctuations in ventilation rate associated with natural ventilation, which can be seen directly in the spread of the effective ventilation rate, row four. The spread values are in the range of 37 - 47% for the naturally ventilated structures as compared with 2 - 13% for the mechanical ventilation strategies. This observation leads us to our second important result, namely, that indoor air quality is better in houses using mechanical ventilation strategies. Larger fluctuations in effective ventilation rate imply that there are more periods when ventilation rates are low and, presumably, when indoor air quality suffers as a result. For

TABLE 1

Results of residential load model simulation, Portland, Oregon

	Case number*					
	0	1	2	3	4	5
Average ventilation rate (ach)	0.52	0.52	0.50	0.50	0.50	0.50
Effective ventilation rate (ach)	0.49	0.49	0.50	0.50	0.50	0.50
Spread of effective ventilation rate (%)	38	38	8	3	3	3
Space heating consumption (kWh/year)	6790	4960	3250	4450	2230	4450
Water heating consumption (kWh/year)	5330	5330	5330	5330	5330	2830
Ventilation system consumption (kWh/year)	0	0	490	880	880	880
Total heating consumption (kWh/year)	12110	10290	9070	10650	8430	8160
Relative consumption percentage (%) (to Case 1)	118	100	88	104	82	79

*Case number:
0 new medium-tight house, no mechanical ventilation;
1 superinsulated medium-tight house, no mechanical ventilation (base case);
2 superinsulated, supertight house, with air-to-air heat exchanger;
3 superinsulated, supertight house, with exhaust ventilation but no heat recovery;
4 superinsulated, supertight house, with exhaust ventilation and air-to-air heat pump;
5 superinsulated, supertight house, with exhaust ventilation and air-to-water heat pump.

pollutants which have threshold limits on exposure (such as organics), periods of low ventilation are clearly undesirable. However, the effects of pollutants for which integrated exposure is the major risk (such as radon) do not depend on the spread of the effective ventilation. Comparing percentage fluctuations for the exhaust air systems with those for the balanced system (case 2), we see that the exhaust systems are consistently better, especially in the windier Great Falls climate, undoubtedly because of the stronger weather dependence of balanced systems.

If we now turn to the bottom line, we see that houses using mechanical ventilation strategies with heat recovery consistently consume less energy than houses that rely on natural ventilation. We also see that houses using exhaust ventilation strategies with heat recovery consume less energy than houses using the balanced flow strategy, except in Missoula where the balanced flow strategy provides slightly better results than the exhaust air-to-water heat pump. This finding can probably be attributed to the fact that the weather is less variable in the Missoula climate, making the larger weather dependence of the balanced system less important. The two heat-pump strategies provide comparable savings, although the air-to-water strategy (case 5) has the advantage of being more easily adapted to provide space cooling, thereby promising even larger energy savings.

To put these energy savings into perspective, they should be compared with the energy savings achieved by superinsulating a house. Comparing cases 0 and 1, we see that superinsulating these houses reduces their energy consumption by 18 - 21%. If we now look at what can be achieved in these houses by adding mechanical ventilation systems, we find that the savings are comparable, 12 - 21%. (Although less than 20% of the households in the region have space-cooling equip-

TABLE 2

Results of residential load model simulation, Missoula, Montana

	Case number*					
	0	1	2	3	4	5
Average ventilation rate (ach)	0.51	0.51	0.50	0.50	0.50	0.50
Effective ventilation rate (ach)	0.49	0.49	0.50	0.50	0.50	0.50
Spread of effective ventilation rate (%)	37	37	7	2	2	2
Space heating consumption (kWh/year)	12630	9640	6910	9170	6170	9170
Water heating consumption (kWh/year)	5330	5330	5330	5330	5330	2830
Ventilation system consumption (kWh/year)	0	0	430	880	880	880
Total heating consumption (kWh/year)	17950	14970	12670	15370	12370	12880
Relative consumption percentage (%) (to Case 1)	120	100	85	103	83	86

*Case number:
0 new medium-tight house, no mechanical ventilation;
1 superinsulated medium-tight house, no mechanical ventilation (base case);
2 superinsulated, supertight house, with air-to-air heat exchanger;
3 superinsulated, supertight house, with exhaust ventilation but no heat recovery;
4 superinsulated, supertight house, with exhaust ventilation and air-to-air heat pump;
5 superinsulated, supertight house, with exhaust ventilation and air-to-water heat pump.

ment, we should note that by including space cooling, these percentage savings would be somewhat higher [17]).

A cross-comparison of savings for space and water heating for houses using different ventilation strategies is presented in Fig. 2, which compares the annual energy bills for houses heated by gas and by electricity, with gas-heated houses assumed to have gas water heaters. The local energy prices used are 4 cents/kWh for electricity and \$5.69/GJ for gas. Gas space and water heaters are assumed to have a 70% efficiency and electric space and water heaters are assumed to have 100% efficiency. This comparison shows that the total bill (space heat + domestic hot water) is reduced by 9 - 21%, or in absolute values, \$27 - \$98 per year, for superinsulated houses employing mechanical ventilation systems with heat recovery compared to naturally ventilated superinsulated houses. An exhaust system without heat recovery yields slightly negative savings because of the costs of fan operation; however, the system assures better indoor air quality. The savings in an all-electric house are larger for heat pump systems and favor an air-to-water heat pump, whereas for a gas-heated house located in Great Falls, an air-to-air heat exchanger system yields slightly higher savings than do heat pump systems. The higher savings for the heat exchanger in the gas-heated Great Falls house is due to the higher electricity consumptions of the heat pumps compared to the heat exchangers. All of the results in Fig. 2 could be combined with first-cost estimates for the ventilation systems and additional house tightening to determine the cost effectiveness of the different strategies. As the overall consumptions for Missoula are comparable to those for Great Falls, the heating costs were also comparable.

The results presented in Tables 1 - 3 for the exhaust air heat pump supplying domestic hot water are for the most representative hot water situation (NSDN profile, total hot water demand 232 kg/day, 265 liter tank). Many other combinations of demand profile,

TABLE 3

Results of residential load model simulation, Great Falls, Montana

	Case number*					
	0	1	2	3	4	5
Average ventilation rate (ach)	0.53	0.53	0.48	0.49	0.49	0.49
Effective ventilation rate (ach)	0.49	0.49	0.48	0.49	0.49	0.49
Spread of effective ventilation rate (%)	47	47	13	5	5	5
Space heating consumption (kWh/year)	12680	9620	7010	8800	6210	8800
Water heating consumption (kWh/year)	5330	5330	5330	5330	5330	2830
Ventilation system consumption (kWh/year)	0	0	450	880	880	880
Total heating consumption (kWh/year)	18010	14940	12790	15010	12410	12510
Relative consumption percentage (%) (to Case 1)	121	100	86	100	83	84

*Case number:
0 new medium-tight house, no mechanical ventilation;
1 superinsulated medium-tight house, no mechanical ventilation (base case);
2 superinsulated, supertight house, with air-to-air heat exchanger;
3 superinsulated, supertight house, with exhaust ventilation but no heat recovery;
4 superinsulated, supertight house, with exhaust ventilation and air-to-air heat pump;
5 superinsulated, supertight house, with exhaust ventilation and air-to-water heat pump.

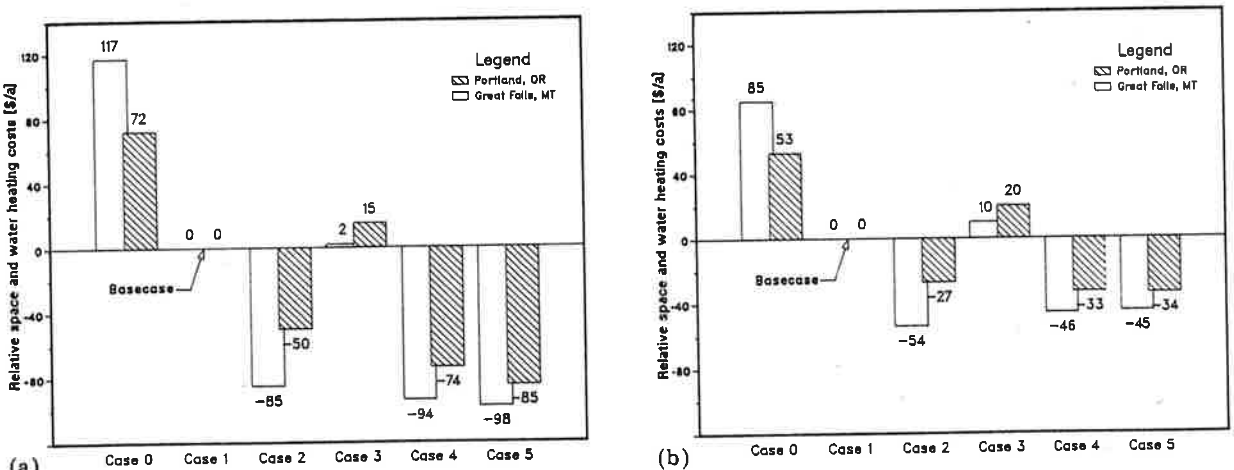

Fig. 2. Comparison of costs for space and water heating for different ventilation strategies; (a) all-electric, (b) gas space and domestic hot water. Cases: 0, new medium-tight house, no mechanical ventilation; 1, superinsulated medium-tight house, no mechanical ventilation (base case); 2, superinsulated, supertight house, with air-to-air heat exchanger; 3, superinsulated, supertight house, with exhaust ventilation but no heat recovery; 4, superinsulated, supertight house, with exhaust ventilation and air-to-air heat pump; 5, superinsulated, supertight house, with exhaust ventilation and air-to-water heat pump. Fuel prices: electricity = 4 ¢/kWh, gas = 2.1 ¢/kWh. Heater efficiencies: electric = 100%, gas = 70%.

TABLE 4

Results of domestic hot water simulation

	Demand profile					
	RAND[a]		NSDN[b]		Extreme[c]	
Total demand (kg/day)	232	296	232*	296	232	296
Tank size (l)	265	114	265	114	265	114
Heat pump heat (kWh/day)	12.5	14.1	11.4	11.1	10.3	8.5
Jacket heat loss (kWh/day)	2.0	1.1	1.9	1.0	1.8	1.0
Compressor running time (h/day)	14	15	12	12	11	9
Minimum tank temperature (°C)	50	46	41	29	35	21
Percentage of hot water heating supplied by heat pump (%)	85	81	78	64	71	49

[a]RAND = Rand Corporation, relatively flat profile.
[b]NSDN = National Solar Data Network, 40% peak at 08:00.
[c]Extreme = 80% peak at 07:00.
* = the most representative case (used in Tables 1 - 3).

total demand, and tank size were tested, the results of which are presented in Table 4.

The combinations of total demand and tank size shown in Table 4 represent the extremes of the largest tank in combination with the smallest demand, and the smallest tank in combination with the largest demand. The results for all other tank-demand combinations will be in between these two. For the Rand Corporation profile, a small 114-liter tank allows the heat pump to provide more than 80% of the daily domestic hot water energy demand for both total water demands. The heat pump runs 14 - 15 hours per day. A plot of the storage-tank temperature over a 24-hour period, in Fig. 3(a), shows that the heat pump is able to maintain a constant temperature in the storage tank, with a minimum value of 46 °C. As mentioned in ref. 16, however, this profile gives overly optimistic results; the hot water draw peak occurs between 19:00 and 20:00 and is only 11.6% of the total daily hot water demand.

The NSDN profile is claimed to be the most representative demand profile (see ref. 16). For this profile, the demand peak is 39.2% between 08:00 and 09:00, and the heat pump runs approximately 12 hours per day, supplying 64 - 78% of the total energy

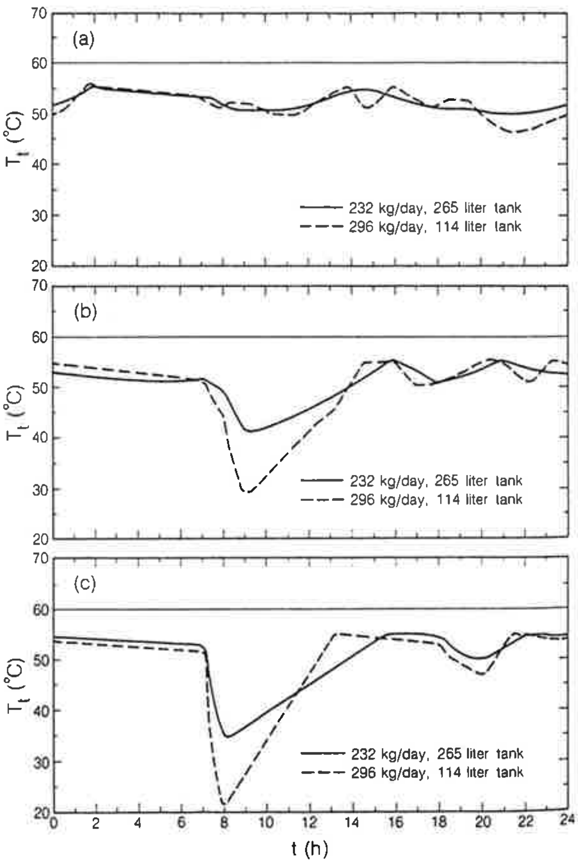

Fig. 3. Storage-tank temperature over time; (a) RAND, (b) NSDN, (c) extreme.

demand. The storage-tank temperature drops down to 29 °C, for the smallest tank and a high total demand. (See Fig. 3(b) for a plot of storage-tank temperature for a 24-hour period.)

The last demand profile was used to study the dynamics of the hot water system under extreme demand conditions. The reason for examining this profile is that the exhaust-air heat-pump water heater has a much smaller output than conventional water heaters, implying that supply-demand mismatch may be significant. The results of the simulations with this profile indicate that a very large storage tank is necessary to provide the required water temperature with an auxiliary heater of reasonable size. Because of the long periods with no hot water demand, the heat pump run time is small (9 - 11 h/day), and only 49 - 71% of the total energy is supplied by the heat pump. During the large demand peaks the storage-tank temperature drops as low as 21 °C. Transient storage-tank temperature is plotted in Fig. 3(c).

CONCLUSIONS

Our first set of conclusions from our comparison of ventilation strategies relates to the total air flow and indoor air quality resulting from each strategy. We found that all of the mechanical ventilation strategies examined provided more uniform ventilation rates than natural ventilation, implying lower total air flow and better indoor air quality. For the same effective ventilation rate, the mechanical ventilation strategies have fewer periods of low ventilation, implying a lower chance of indoor air quality problems. On the other hand, the excess ventilation extremes are lower for the mechanical ventilation strategies.

The ventilation comparisons of the strategies also confirmed that exhaust ventilation is less weather-dependent than balanced ventilation, suggesting that it provides better indoor air quality. This conclusion, that exhaust ventilation systems provide better indoor air quality than balanced ventilation systems, does not even take into account the short circuiting that can occur when balanced systems are not fitted with ductwork.

The most important conclusion to be drawn from this investigation is that mechanical ventilation systems not only provide better ventilation, but can reduce energy consumption significantly. The energy saved by installing a mechanical ventilation system with heat recovery, 9 - 21% of the total heating bill (space heating + water heating), is comparable to the energy saved by super-insulating a house. This finding holds true even in the extreme climate of Great Falls, Montana (4500 HDD °C).

We can also conclude that exhaust ventilation with heat recovery is a viable ventilation alternative for single-family residences, providing better indoor air quality and larger energy savings than balanced ventilation systems. Other questions to be explored are how the large first costs associated with heat recovery for exhaust ventilation systems can be reduced, how these systems compare in other climates (such as those that have large cooling loads), and how an exhaust-air heat pump can be used to recover other forms of waste heat (e.g., dryer vents, refrigerator exhaust).

ACKNOWLEDGEMENTS

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Building and Community Systems, Building Systems Division of the U.S. Department of Energy under Contract No. DE-ACO3-76SF00098, and by the Bonneville Power Administration, Portland, Oregon.

REFERENCES

1 *ASHRAE Standard 62-81*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1981.
2 Product literature *Exhaust Air Heat Pump for Single-family Houses*, Flakt Products, Inc., Winston-Salem, NC, 1984.
3 *Handbook of Fundamentals*, ASHRAE, 1981, Ch. 22.
4 M. H. Sherman and D. T. Grimsrud, *A Comparison of Alternate Ventilation Strategies, Report LBL-13678*, Lawrence Berkeley Laboratory, Berkeley, CA, 1982.
5 M. H. Sherman and D. T. Grimsrud, *Measurement of Infiltration Using Fan Pressurization and Weather Data, Report LBL-10582*, Lawrence

Berkeley Laboratory, Berkeley, CA, 1980.

6 M. P. Modera and F. Peterson, *Ventilation and Infiltration, Report LBL-18955*, Lawrence Berkeley Laboratory, 1985.

7 F. J. Offerman, W. J. Fisk, D. T. Grimsrud, B. Pederson and K. L. Revzan, *Ventilation Efficiencies of Wall- or Window-Mounted Residential Air-to-Air Heat Exchangers, Report LBL-15376*, (Presented at ASHRAE meeting, Washington, DC, June 1983) Lawrence Berkeley Laboratory, Berkeley, CA, 1983.

8 F. J. Offerman *et al.*, *Residential Air-Leakage and Indoor Air Quality in Rochester, New York, Report LBL-13100*, Lawrence Berkeley Laboratory, Berkeley, CA, June, 1982.

9 Data provided by Mark McKinstry, Bonneville Power Administration, Portland, OR, 1984.

10 *Hourly Solar Radiation — Surface Meteorological Observations, Solmet Volume 2 - Final Report TD-9724*, National Oceanographic and Atmospheric Administration, 1979.

11 *TRNSYS Manual*, Solar Engineering Laboratory, University of Wisconsin, Madison, WI, 1984, Version 12.1.

12 ASHRAE *Handbook of Fundamentals*, 1977, p. 25.27; and 1981, p. 26.34.

13 W. J. Fisk *et al.*, *Performance of Residential Air-to-Air Heat Exchangers during Operations with Freezing and Periodic Defrosts, Report LBL-18024*, Lawrence Berkeley Laboratory, Berkeley, CA, 1984.

14 Personal communication with Anthony Usibelli, Building Energy Data Group, Lawrence Berkeley Laboratory, Berkeley, CA, 1984.

15 Product literature for Copelaweld R-12 compressor, *#JRN1-0025*, Copeland Corporation, Sidney, OH, 1976.

16 E. J. Barvir *et al.*, Hourly use profiles for solar domestic hot water heaters in the National Solar Data Network, *Proc. Conference, Solar Engineering, Reno, NV, 1981*, ASME.

17 Personal communication with Grant Vincent, Bonneville Power Administration, Portland, Oregon, 1984.

Appendix

TABLE A-1

House specifications

House type	Ranch-style, single-story, wood-frame construction		
Location	1) Portland, OR, latitude = 45.6°, HDD = 2840 °C 2) Missoula, MT, latitude = 46.9°, HDD = 4603 °C 3) Great Falls, MT, latitude = 47.5°, HDD = 4497 °C		
Floor area (m^2)	125		
Window area (m^2)	13.6		
Slope of roof	14°		
Volume of living space (m^3)	306		
Thermal capacitance (kJ/K)	7600		
	House		
	New construction	Superinsulated	
Ceiling insulation (W/m^2K)	0.19	0.16	
Wall insulation (W/m^2K)	0.46	0.24	
Floor insulation (W/m^2K)	0.30	0.30	
Windows (W/m^2K)	double glazed 3.0	triple glazed 2.0	
Doors (W/m^2K)	2.2	0.97	
	New construction	Superinsulated	Supertight
Effective Leakage Area (cm^2)	500 - 700*	500 - 700*	150

*500 is for Great Falls, 700 is for Portland and Missoula.

TABLE A-2

Domestic hot water system specifications

Tank size (l)	114 - 265
Heat loss coefficient of tank (W/m^2K)	1.0
Ratio of tank height to diameter (dimensionless)	2.75
Input from heat pump (including fan heat) (W)	920
Flow rate for heat pump water loop (l/h)	400
Total daily demand for hot water (kg/day)	232 - 296
Feed-water temperature (°C)	12.8
Maximum temperature of storage tank (°C)	55
Minimum hot water delivery temperature (°C)	60
Heat pump controller dead band (°C)	4