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VENTILATION STRATEGIES FOR DIFFERENT CLIMATES

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ABSTRACT

Until recently, residential ventilation in the United States has been provided by infiltration. In this report we compare natural ventilation (ventilation by infiltration) with several mechanical ventilation strategies and examine the overall energy consumption associated with these strategies in different climatic regions in the U.S. The strategies examined are: natural ventilation, balanced ventilation with an air-to-air heat exchanger, exhaust ventilation without heat recovery, and exhaust ventilation with heat recovery via a heat pump. Two strategies for utilizing the heat pump output for domestic hot water are examined. One heat pump strategy employs exhaust fan reversal to provide space cooling whenever possible during the summer months. A modified TRNSYS residential load model incorporating the LBL infiltration model, an algorithm to calculate effective ventilation, 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 useful heat supplied by an exhaust ventilation heat pump as a function of daily hot water demand. The simulations indicate that the choice of ventilation strategy can have a significant impact on energy consumption. They show that total end-use energy consumption can be reduced as much by mechanical ventilation as by superinsulation of a house. The comparisons also show that for the same effective ventilation rate, houses with mechanical ventilation systems (especially those with exhaust fans) have better indoor air quality than those that rely on natural ventilation.

INTRODUCTION

Over half the energy used in the building sector is consumed by space heating and cooling; thus it is the largest single energy end use in buildings. Space conditioning consumption can be broken down into two major components: conduction and infiltration. In the United States, conventional houses have leaky envelopes; therefore, the ventilation occurs naturally through infiltration driven by wind and stack effects. Recently, in an effort to conserve energy, there has been a significant increase in the insulating and tightening of houses. If the building envelope is tightened, the infiltration may become too low, especially under mild weather conditions, thus causing indoor air quality problems [1].

Several strategies have been employed in order to both reduce the heat loss due to infiltration and to maintain acceptable indoor air quality. In the most sophisticated strategy, a tight building envelope is combined with a mechanical ventilation system. The mechanical ventilation technique most often used in the United States employs an air-to-air heat exchanger to connect the air streams of two fans, providing a balanced system in which flow rates are set at a specified ventilation rate and the intake air is preheated by the exhaust stream. In Scandinavia, a popular strategy is to install an exhaust fan with heat recovery. The house is depressurized by the exhaust fan, and outdoor air is drawn into the house either through leaks in the envelope or vents designed for this purpose. The heat from the exhaust stream is coupled to a heat pump and can then be used for domestic water heating and/or space heating [2]. In cases where uniform ventilation rates are desirable but space conditioning loads are small, exhaust fans are used without heat recovery.

In this study, we concentrate on the impacts of these ventilation strategies on the total energy consumption of single-family dwellings throughout the United States. Earlier work focused on the Pacific Northwest, where houses are typically all-electric [3]. As climatic considerations are important in this research, we have chosen cities representative of five different climates: hot and humid, windy and cold, calm and cold, typical East Coast, and dry desert. We will examine five ventilation strategies: natural ventilation, balanced ventilation with an air-to-air heat exchanger, exhaust ventilation without heat recovery, exhaust ventilation connected to a heat pump that heats domestic hot water, and exhaust ventilation connected to a heat pump that heats domestic hot water and is equipped with a reversible fan for cooling. These data will be analyzed with an hour-by-hour residential building simulation model.

VENTILATION

Ventilation can be either natural or mechanical. Natural ventilation can be obtained either by infiltration or by intentionally opening windows and doors. Infiltration occurs when the building envelope interacts with pressure differences resulting from wind and indoor-outdoor temperature differences (stack effect). In this report, a simplified single-zone infiltration model is used to determine the ventilation rate as a function of weather conditions [4,5]. Here, the ventilation rates obtained from wind speeds and temperature differences are added assuming quadrature.

$$Q_{tot} = \left\{ Q_{wind}^2 + Q_{stack}^2 \right\}^{\frac{1}{2}} \tag{1}$$

where

Q_{tot} = total ventilation [cfm], [L/s]
Q_{wind} = infiltration rate due to wind effect [cfm], [L/s]
Q_{stack} = is the infiltration due to stack effect [cfm], [L/s]

When a mechanical ventilation system supplements the natural ventilation, Equation 1 becomes [6]:

$$Q_{tot} = \left\{ Q_{wind}^2 + Q_{stack}^2 + Q_{unbal}^2 \right\}^{\frac{1}{2}} + Q_{bal} \tag{2}$$

where

Q_{unbal} = airflow rate of an unbalanced fan [cfm], [L/s]
Q_{bal} = airflow rate through a balanced fan system [cfm], [L/s]

Equation 2 states that balanced flows simply add to the infiltration, but that unbalanced flows add in quadrature. This occurs because the internal pressure of the house is changed by unbalanced flows, which affects the wind- and stack-induced flows.

In balanced mechanical ventilation systems, two air streams are driven by a supply fan and an exhaust fan. An air-to-air heat exchanger connects the two streams and transfers heat from the the warm air stream to the cold air stream with little or no mixing. However, under certain weather conditions, problems can occur in the core of the heat exchanger, when the moisture contained in the exhaust air stream tends to freeze [7,8]. Freezing impedes heat transfer and may cause the system to become unbalanced.

When a house is depressurized with a single exhaust fan ventilation system, the outdoor air is sucked into the house through the building envelope. If the house is supertight, vents must be placed in the envelope to allow air intake. There are several advantages to using exhaust ventilation systems instead of balanced systems; exhaust systems do not require supply ductwork because ambient air enters the house through leaks distributed over the entire envelope rather than through a single intake, and exhaust systems permit less variation in ventilation rates because these systems change the pressure in the house (see Eq. 2). In the Scandinavian countries, heat is extracted from the exhaust stream with a small heat pump that provides either domestic space heating or water heating. Although this device can extract a large amount of heat from the exhaust air, its complexity and high initial cost are major disadvantages.

Ventilation is important for keeping the concentration of contaminants in the indoor air below certain limits. To examine the effectiveness of different ventilation strategies, a simple relationship between ventilation and contaminant concentration is needed. Given a constant contaminant source strength (i.e., rate of contaminant generation), the steady-state concentration of that contaminant is proportional to the inverse of the ventilation rate. However, under real (non-steady-state) conditions, the volume of air in the room tends to damp out the impacts of sudden changes in ventilation rate. Taking these effects into account, a single parameter for describing the average concentration of (or exposure to) a constant-source-strength contaminant has been developed [9]:

$$Q_{e,i} = \frac{1}{\dfrac{1-\alpha_i}{Q_i} + \dfrac{\alpha_i}{Q_{e,i-1}}}. \tag{3}$$

where

$$\alpha_i = e^{-A_i \Delta t}$$

with:

- $Q_{e,i}$ = effective ventilation rate at time i $[m^3/h]$
- $Q_{e,i-1}$ = effective ventilation rate at time i-1 $[m^3/h]$
- Q_i = ventilation rate at time i $[m^3/h]$
- A_i = air change rate at time i $[ach]$
- Δt = time step $[h]$

The statistical spread of the effective ventilation rate describes the expected fluctuations, and is, therefore, a measure of acute concentration peaks and valleys. For contaminants for which acute exposure (rather than integrated exposure) is the major health hazard, the ventilation spread factor is an important measure of indoor air quality.

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

where

- S = spread factor [dimensionless]
- Q = ventilation rate [ach]
- n = number of points [dimensionless]

As the spread increases, the frequency of occurrence of low ventilation rates and concomitant poor air quality will also increase for a given effective ventilation rate. The indoor air quality resulting from different ventilation strategies is, therefore, described by the effective ventilation rate and its spread. From the energy perspective, the total flow rate of outdoor air into the house is the important quantity to determine. The average ventilation rate is used to describe this quantity.

ANALYTICAL PROCEDURE

We chose to simulate each of the strategies in a superinsulated ranch-style house [3] at five sites, each representative of a different U.S. climatic region. Figure 1 shows the sites for the simulations. Table 1 in the Appendix contains a comparison of the climatic data for the five sites and Table 2 gives the thermal properties of the house.

We performed an hour-by-hour simulation of energy consumption and ventilation rate using Typical Meteorological Year (TMY) weather data [10] for each site for each of the strategies. We also performed one additional simulation for each site, for a naturally ventilated house that conformed to typical new-house construction standards. From this simulation we can compare the energy impacts of superinsulating a house with the energy impacts of different ventilation strategies. The naturally ventilated superinsulated house is the base case for all comparisons. Table 2 in the Appendix provides the detailed specifications for the houses. Adjustments were made to the ventilation rates used for the different strategies in order to provide the same air quality (to first order) by assuring that the average effective ventilation rates were equal. Effective leakage area values (see Reference 1) for the naturally ventilated houses were chosen by obtaining an average effective ventilation rate of 0.5 ach. The effective leakage area for the mechanically ventilated houses was assumed to be 150 cm^2. Adjustments were then made to the fan flows of the mechanical ventilation systems to insure that the average effective ventilation rates would be the same.

For the fourth and fifth ventilation strategies, it is necessary to simulate hot water consumption, as the total hot water demand, the hot water demand profile, and the size of the storage tank determine how much of the required energy can be supplied by the heat pump. The total hot water demand (242 L/day) and its profile were chosen from the literature [11,12]. The hot water tank chosen was a commercially available solar hot water system tank (310 L) [13].

COMPUTER SIMULATION

In order to compare the different ventilation strategies, we used an existing computer simulation program, TRNSYS (Transient System Simulation) [14]. In this program, a central differential equation solver and a set of independent component modules can be interconnected to simulate a particular system, which gives us a high degree of flexibility. The residential load and domestic hot water models in this comparison of ventilation strategies are of particular interest.

Residential Load Model

The residential load model in TRNSYS consists of roof and zone models that use the ASHRAE response-factor method for calculating the heat transfer through the walls [15,16]. Instead of using the air infiltration model used in the TRNSYS zone model, we used the simplified single-zone model. For each site, we used Typical Meteorological Year (TMY) hourly weather tapes. The assumptions made in the residential load model are listed below.

- 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 was 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 was assumed to strike only the floor,
- Set point for heating was $T = 20^{\circ}C$.

Because of the relatively small capacity of exhaust air heat pumps supplying space conditionin the shoulder season has an important effect on the total energy delivered by such systems. Therefor internal gains and thermostat operation had to be well thought out. Internal gains were separate specified for people and equipment, both of which were specified according to an hour-by-ho schedule. We assumed that 70% of the sensible heat gain from people is radiative. We also assume that appliances and lighting would deliver 4779 kilowatt hours (kWh) per year distributed over the da according to the schedule given in Figure 2. Twenty-five percent (25%) of the total heat gain from th appliances is assumed to be radiative and 75% convective. We also included as internal gains th standby losses from the domestic hot water tank. The schedule, total loads, and underlying assump tions are contained in the Appendix.

The thermostat setpoints are particularly important for the strategy that employs exhaust fa reversal to provide space cooling. Care must be exercised so as to avoid flow reversals.

We assumed that the heating coefficient of performance (COP) is variable between 2.1 and 2.8 anc delivers 920 W for the heat pump heating the domestic hot water. The heat from the 100 W exhaus fan heats the exhaust air, which results in a smaller temperature difference between the hot water anc the exhaust air and changes the COP. As both the COP and the on-time of the heat pump, and. therefore, the energy delivered, depend on the tank temperature, the choice of hot water setpoint, hot water demand, demand schedule, and tank size is crucial. The switch points for the fan reversal were set at T_{press} = 23 °C for cooling (pressurization) and $T_{depress}$ = 21 °C for exhaust (depressurization). When the airflow is reversed for summer cooling, the fan energy is subtracted from the cooling provided by the heat pump, and the air stream temperature before it hits the cooling coil is equal to the outdoor temperature.

For the ventilation strategy using an air-to-air heat exchanger, we assumed that the heat exchanger has a seasonal heat transfer efficiency of 65% (including freeze-defrost cycles) [17], and that it has two 50 W fans, the supply fan located downstream and the exhaust fan located upstream of the heat exchanger core. This gave us a recovery of 50 W times (1+ 0.65), or 82.5 W, when the fans operate during the heating season. For more details see Table 7 in the Appendix.

Domestic Hot Water Model

Originally, the domestic hot water model chosen for TRNSYS used a solar collector system as a heat source. In this study we use instead an air-to-water heat pump whose heat source is the exhaust airflow. Due to the input requirement of the TRNSYS module, we needed to specify both the heat rejected at the condenser of the heat pump and the flow rate of the hot water loop. We obtained the condenser heat rejection from the specifications of a commercially available heat pump, and we determined the water flow rate from an average heat rejection and the size of the heat exchanger at the condenser. We also had to consider the heat-exchanger design parameters (i.e., UA-value and flow velocity) to make this calculation.

To make the simulations realistic, we set an upper limit of 55 °C for the water temperature at which the heat pump was turned off. (This constraint is partially due to the operating characteristics of the small heat pumps currently available.) As the condenser temperature and therefore, the refrigerant pressure increases, the useful lifetime of the compressors decreases. Thus, the heat pump cycles on and off, depending on the storage tank temperature.

To size the exhaust-air heat pumps used for domestic hot water, we calculated the amount of heat that could be extracted from the exhaust air without causing freezing at the evaporator of the heat pump. A supertight house with an exhaust ventilation system was found to have a fan flow rate of 150 m^3/h. We chose an exhaust-air temperature drop of 11 K to avoid freezing and, using the above flow rate, obtained approximately 550 W of heat from the exhaust air. As we used a stratified hot water tank model, we designed the heat pump loop so that water was pumped from the lowest level in the tank through the condenser and back into the tank at half the tank height. The top of the tank, where the auxiliary heater is located, was kept between 52 °C and 55 °C.

The hourly hot water demand profile used in the simulations, the National Solar Data Network (NSDN) profile [12], is described in Figure 3. This profile was found to fit well with the assumed occupant internal gain schedules. The total demand was assumed to be 242 L/day [11]. The hot water simulation assumptions are summarized below.

- The storage tank is stratified,
- The storage tank is located in the heated section of the house and heat losses occur to temperature $T = 20\,^{\circ}C$,
- The tank was a 310-liter commercially available tank,
- The feed-water temperature is constant over the year at $10\,^{\circ}C$,
- The daily hot water demand profile does not change over the course of the year,
- The heat pump can heat water up to $T = 55\,^{\circ}C$,
- The dead band for the water temperature controller at the condenser of the exhaust-air heat pump is $3\,^{\circ}C$.

In arriving at the final assumptions used for the comparisons in this report, sensitivities of the results to different parameters were tested. It was found that using a stratified tank model rather than a mixed tank model [3] implied significantly smaller auxiliary energy requirements. With the stratified tank model, the delivery temperature never dropped below $50\,^{\circ}C$, even without the additional electric resistance heat source. Preliminary studies showed only a slight difference (< 5%) in energy consumption for domestic hot water heating using the monthly average feed water temperature, compared to using the annual average. We also found that changing the hot water demand over the course of the year did not significantly affect the results.

RESULTS

Tables 9 through 13 present the results of simulating space conditioning and water heating loads, including a comparison of total end-use (not primary energy) conditioning consumptions (water heating, space heating and cooling) for the five ventilation strategies in superinsulated 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 comparison of the ventilation achieved with each of these strategies yieldes some important results. (Remember that we kept the effective ventilation rate constant.) First, for naturally ventilated houses, the average ventilation rate is higher than the effective ventilation rate, whereas for mechanically ventilated buildings, these values are essentially the same. For mechanically ventilated houses, the spread of the effective ventilation is much smaller than for houses not mechanically ventilated. Because the pressure field changes, houses having exhaust ventilation show the smallest spread. For naturally ventilated houses, spread values for the five locations are in the range of 38%-43%, compared to 3%-11% for the mechanically ventilated buildings. The average monthly effective ventilation rates that have been calculated for three different strategies for a house in Bismarck are shown in Figure 4. We see that for natural ventilation the effective ventilation rate averages below 0.4 ach in the summer, whereas during the winter monthly averages can reach 0.65 ach. These ventilation fluctuations are not uncommon. The spread of the effective ventilation is reduced significantly by the installation of an air-to-air heat exchanger with two fans and a tighter building envelope. However, houses with exhaust fans have the best distribution. The unnecessary variation of effective ventilation rates is the shaded area of Figure 4.

The bottom line of Tables 9 through 13 shows that houses using mechanical ventilation strategies with heat recovery consistently use less heating energy than those relying on natural ventilation. Houses using exhaust ventilation strategies with heat recovery also consume the same or less energy than houses using the balanced flow strategy. In hot, humid climates (Table 10), however, end-use energy consumption for systems using air-to-air heat exchangers might not save any energy compared to the naturally ventilated base case. Even though there are savings in space heating consumption, the electrical energy used to drive the two fans makes this system a loser for that particular climate. For cold, windy climates, however, even exhaust systems without any heat recovery device perform better than the base case.

To get a better perspective, we compare the energy savings described above with those achieved by superinsulating a house. If we compare cases 0 and 1, we see that superinsulating has reduced the end-use energy consumption by 11%-22%. When a mechanical ventilation system is added to the house, the savings increase to 27%-34%.

Although all energy consumption data were based on the end-use consumption, we made the following comparison for the houses in Bismarck and Lake Charles based on primary energy use. We assumed that the oil/electricity conversion efficiency was 33%, the gas furnace efficiency 80%, the gas DHW efficiency (including standby losses) 60%, and we assumed an air-conditioner coefficient of performance of 2.3 for cooling and 2.8 for heating. We assumed the use of gas domestic hot water heating (except for cases 4 and 5) as well as for space heating in Bismarck. We used an air conditioner for the space conditioning in Lake Charles and for the cooling for Bismarck. In Table 14, we see that the system using an air-to-air heat exchanger for Bismarck showed minimal primary energy savings, whereas the primary energy consumption for the three different exhaust strategies differed little from the consumption for the naturally ventilated house.

The energy costs for the different ventilation strategies are shown in Table 15. We derived these costs from the average local energy cost documented by the U.S. Department of Labor [18]. For electricity, the cost for Bismarck was 6 cents/kWh, and for Lake Charles, 7 cents/kWh. Gas prices, given in cents per kWh, were 2.2 for Bismarck and 2.1 for Lake Charles. Based on the primary energy usage given above, operational costs seem to be most favorable for the air-to-air heat exchanger system in Bismarck. In Lake Charles, however, we found no savings in operational costs for the mechanical systems, compared to the naturally ventilated houses. In this case, an exhaust fan without heat recovery is the best choice for maintaining reasonable indoor air quality.

None of these comparisons accounts for the first costs of the mechanical systems. These costs were determined in a previous study [19] of all-electric houses with resistance heat in the Pacific Northwest, where payback times were found to be between 10 and 20 years.

Another observation in Tables 9-13 is that additional end-use energy savings between 2% and 5%, depending on climate, were attained with the ventilation strategy of using a reversible fan. A second advantage of this technique, in addition to the energy savings, is that it switches from the depressurization mode during the heating season to the pressurization mode during the cooling season. Not only does this supply cool air to the conditioned space, it also changes the flow direction through the building envelope. In the depressurization-only mode, hot, humid air is sucked through the walls, condensing on its way while being cooled down before approaching the air-conditioned space. Reversing the airflow presses the cool, dry air through the walls; the air is then heated on its way to the outside. This method produces no additional condensation. It is believed that this measure will actually significantly *reduce* damages due to moisture inside the building structure.

CONCLUSIONS

Our first set of conclusions from this comparison of ventilation strategies is based on the total airflow and indoor air quality resulting from each strategy. We found that all the mechanical ventilation strategies examined provided more uniform ventilation rates than natural ventilation and, thus, lower total airflow and potentially better indoor air quality. In addition, the excess ventilation extremes in winter are lower for the mechanical ventilation strategies and, therefore, the excess ventilation heat loss is also lower.

Comparisons of the ventilation strategies also confirm that exhaust ventilation is less weather-dependent than balanced ventilation, which suggests that it provides better indoor air quality. This conclusion, that exhaust ventilation systems provide better indoor air quality than balanced ventilation systems, does not, however, consider the short-circuiting that occurs in all balanced systems, especially those not fitted with ductwork. Short circuiting will further decrease the ventilation effectiveness of balanced systems.

The most important conclusion we draw from this investigation is that mechanical ventilation systems not only provide better ventilation but also reduce end-use energy consumption significantly. However, we must also conclude that end-use comparisons do not tell the whole story. For all-electric regions, the end-use comparison is the bottom line. For regions with gas service, cost and primary energy comparisons do not look favorably on mechanical ventilation systems. We do find, however, that mechanical ventilation systems can provide significantly better air quality with little or no penalty, even under the most adverse price and energy supply conditions.

Finally, a method of quantifying indoor air quality must be established in order to make an accurate comparison of ventilation strategies. The effective ventilation and its spread are a first step in that direction, although much remains to be done. Only by quantifying and specifying a minimum level of ventilation effectiveness can first costs and operating costs be used to compare the economics of different ventilation strategies.

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APPENDIX: Tables 1-15

Table 1:
Sites and Climates for the Simulation Runs

Site	HDD* [°C-days]	CDD* [°C-days]	IDD* [°C-days]	Remarks
Bismarck	5041	262	6899	cold and windy
Lake Charles	885	1477	2957	hot and humid
Minneapolis	4534	418	6033	cold
New York	2731	664	4176	east coast climate
Albuquerque	2475	732	2696	hot, very cold, dry

* Base temperatures: HDD and CDD = 18.5 °C; IDD = 24 °C, 50% rel. humidity

Table 2: Areas and Thermal Resistance of the House			
Type	Area $[m^2]$	Normal U-Value $[W/m^2 K]$	Superinsulated U-Value $[W/m^2 K]$
Floor	125	0.52	0.30
Ceiling	125	0.19	0.16
South Wall	27.9	0.46	0.24
East Wall	17.9	0.46	0.24
North Wall	29.4	0.46	0.24
West Wall	19.1	0.46	0.24
Total Wall	94.2	0.46	0.24
South Window	3.7	3.0	2.0
East Window	2.6	3.0	2.0
North Window	4.0	3.0	2.0
West Window	3.3	3.0	2.0
Total Window	13.6	3.0	2.0
South Door	1.9	2.2	0.97
East Door	1.9	2.2	0.97

Table 3: Heat Gain from Occupants (ASHRAE Handbook 1985 Fundamentals)					
			Sensible Heat only		
Action	Total for Male [W]	Ratio Sens/Lat [-]	Male [W]	Female [W]	Child [W]
Resting	117	1.5	70	60	53
Light Work	234	1.0	115	98	86
Light Machine Work	304	.5	101	86	76
Heavy Machine Work	468	.55	166	141	125

Table 4: Annual Internal Gains from Appliances (Sensible)		
Type	Unit	Value
Light	kWh/yr	967
Refrigerator	kWh/yr	964
Range	kWh/yr	1200
Television	kWh/yr	220
Dryer	kWh/yr	900
Dishwasher	kWh/yr	256
Heat Loss from DHW Tank	kWh/yr	272
Radiative Portion	%	25

Table 5: Daily Schedule for Internal Gains (Sensible)				
Time	Type			
	Father	Mother	Children	Lights
[h]	[-]	[-]	[-]	[% peakload]
1	S	S	S	14
2	S	S	S	14
3	S	S	S	4
4	S	S	S	4
5	S	S	S	7
6	S	S	S	7
7	U	U	U	18
8	U	U	U	18
9	N	U	N	18
10	N	U	N	18
11	N	U	N	14
12	N	U	N	14
13	N	N	N	14
14	N	N	N	14
15	N	N	N	14
16	N	U	N	14
17	N	U	U	29
18	U	U	U	29
19	U	U	U	57
20	U	U	U	57
21	U	U	S	75
22	U	U	S	75
23	U	U	S	54
24	S	S	S	54

S = sleeping; U = up; N = not home; peakload = 0.77 kW; Lighting schedule includes outside lighting;

Table 6:
Daily Schedule for Internal Gains (Sensible)

Time [h]	Type: People [Wh]	Lights [Wh]	Fridge [Wh]	Range [Wh]	TV [Wh]	Dryer [Wh]	Dishwasher [Wh]	Waterheater [Wh]	Total [Wh]
1	236		110					31	377
2	236		110					31	377
3	236		110					31	377
4	236		110					31	377
5	236		110					31	377
6	236		110					31	377
7	385	144	110	500				31	1170
8	385	144	110					31	670
9	98		110					31	239
10	98		110					31	239
11	98		110					31	239
12	98		110					31	239
13	0		110					31	141
14	0		110					31	141
15	0		110					31	141
16	98		110	1400				31	1639
17	270	360	110	1400				31	2171
18	385	360	110		100			31	986
19	385	360	110		100		700	31	1686
20	385	360	110		100	170		31	1156
21	319	360	110		100			31	920
22	319	280	110					31	740
23	319	280	110					31	740
24	236		110					31	377

Table 7: Ventilation		
Type	Units	Value
Exhaust		
Airflow	m^3/h	150
Indoor Air Temperature	$^\circ C$	20
Fan Power	W	100
Number of Reversible Axial Fans	-	1
Air-to-Air Heat Exchanger		
Airflow	m^3/h	150
Indoor Air Temperature	$^\circ C$	20
Night Setback Air Temperature	$^\circ C$	15
Fan Power (each)	W	50
Number of Centrifugal Fans	-	2
Heat Transfer Efficiency	%	65

Table 8: Hot Water		
Type	Units	Value
Stratified Tank		
Tank Size	L	310
Tank Height	m	1.6
Tank Diameter	m	0.64
Ratio Height/Diam.	-	2.52
Surface Area	m^2	3.2
Hot Water Demand (high)	$L/(day\ pers)$	60.6
Hot Water Demand (high)	$L/(day\ house)$	242.4
Feedwater Temperature	$^\circ C$	10
Jacket U-value	$W/(m^2 K)$	.28
Heat Pump		
El. Power	W	450
Coefficient of Performance	-	2.1 - 2.3
Thermostat Setting High	$^\circ C$	55
Thermostat Setting Low	$^\circ C$	52
Exhaust Airflow	m^3/h	150
Indoor Air Temperature	$^\circ C$	20
Water Loop	kg/h	156
Water Pump Power	W	50
Auxiliary Heater		
Heating Power	W	4500
Thermostat Setting High	$^\circ C$	55
Thermostat Setting Low	$^\circ C$	52

Table 9:

Annual *End Use* Energy Consumption for House in: **Bismarck**

	new constr.	superinsulated				
				exhaust air		
	nat. vent.	nat. vent.	air-to-air	w/o heat rec.	HP to DHW	HP to DHW + cooling
case number	0	1	2	3	4	5
Average Vent. Rate [ach]	0.57	0.57	0.51	0.50	0.50	0.50
Effective Vent. Rate [ach]	0.50	0.51	0.50	0.50	0.50	0.50
Spread of Vent. [%]	43	42	11	3	3	3
Space Heating Cons. [MWh/yr]	22.2	15.7	11.2	14.0	14.0	14.0
Space Cooling Cons. [MWh/yr]	2.3	2.3	22.3	2.2	2.2	2.0
Water Heating Cons. [MWh/yr]	4.7	4.7	4.7	4.7	2.4	2.4
Vent. System Cons. [MWh/yr]	-	-	0.9	0.9	0.9	0.9
Total Cond. Cons. [MWh/yr]	29.2	22.6	19.1	21.8	19.6	19.3
Rel. Cons. to Case # 1	1.28	1.00	0.84	0.96	0.86	0.85

Table 10:

Annual *End Use* Energy Consumption for House in: **Lake Charles**

	new constr.	superinsulated				
				exhaust air		
	nat. vent.	nat. vent.	air-to-air	w/o heat rec.	HP to DHW	HP to DHW + cooling
case number	0	1	2	3	4	5
Average Vent. Rate [ach]	0.59	0.58	0.50	0.50	0.50	0.50
Effective Vent. Rate [ach]	0.51	0.50	0.49	0.50	0.50	0.50
Spread of Vent. [%]	47	47	8	3	3	3
Space Heating Cons. [MWh/yr]	3.0	1.7	1.0	1.3	1.3	1.3
Space Cooling Cons. [MWh/yr]	7.7	7.3	7.5	7.3	7.3	6.6
Water Heating Cons. [MWh/yr]	4.7	4.7	4.7	4.7	2.4	2.4
Vent. System Cons. [MWh/yr]	-	-	0.9	0.9	0.9	0.9
Total Cond. Cons. [MWh/yr]	15.4	13.7	14.0	14.2	11.9	11.2
Rel. Cons. to Case # 1	1.13	1.00	1.03	1.04	0.87	0.82

Table 11: Annual *End Use* Energy Consumption for House in: **Minneapolis**						
	new constr.	superinsulated				
				exhaust air		
	nat. vent.	nat. vent.	air-to-air	w/o heat rec.	HP to DHW	HP to DHW + cooling
case number	0	1	2	3	4	5
Average Vent. Rate [ach]	0.56	0.56	0.50	0.50	0.50	0.50
Effective Vent. Rate [ach]	0.51	0.51	0.50	0.50	0.50	0.50
Spread of Vent. [%]	38	38	9	3	3	3
Space Heating Cons. [MWh/yr]	20.0	14.1	10.0	12.6	12.6	12.6
Space Cooling Cons. [MWh/yr]	2.7	2.7	2.7	2.6	2.6	2.3
Water Heating Cons. [MWh/yr]	4.7	4.7	4.7	4.7	2.4	2.4
Vent. System Cons. [MWh/yr]	-	-	0.9	0.9	0.9	0.9
Total Cond. Cons. [MWh/yr]	27.4	21.5	18.3	20.8	18.6	18.3
Rel. Cons. to Case # 1	1.25	1.00	0.85	0.97	0.86	0.85

Table 12: Annual *End Use* Energy Consumption for House in: **New York City**						
	new constr.	superinsulated				
				exhaust air		
	nat. vent.	nat. vent.	air-to-air	w/o heat rec.	HP to DHW	HP to DHW + cooling
case number	0	1	2	3	4	5
Average Vent. Rate [ach]	0.56	0.56	0.51	0.50	0.50	0.50
Effective Vent. Rate [ach]	0.50	0.50	0.50	0.50	0.50	0.50
Spread of Vent. [%]	41	41	10	3	3	3
Space Heating Cons. [MWh/yr]	11.9	8.1	5.5	7.0	7.0	7.0
Space Cooling Cons. [MWh/yr]	2.4	2.5	2.7	2.5	2.5	2.5
Water Heating Cons. [MWh/yr]	4.7	4.7	4.7	4.7	2.4	2.4
Vent. System Cons. [MWh/yr]	-	-	0.9	0.9	0.9	0.9
Total Cond. Cons. [MWh/yr]	19.0	15.3	13.8	15.1	12.8	12.5
Rel. Cons. to Case # 1	1.24	1.00	0.90	0.99	0.84	0.82

Table 13:

Annual *End Use* Energy Consumption for House in: **Albuquerque**

	new constr.	superinsulated				
				exhaust air		
	nat. vent.	nat. vent.	air-to-air	w/o heat rec.	HP to DHW	HP to DHW + cooling
case number	0	1	2	3	4	5
Average Vent. Rate [ach]	0.56	0.56	0.50	0.50	0.50	0.50
Effective Vent. Rate [ach]	0.50	0.50	0.50	0.50	0.50	0.50
Spread of Vent. [%]	41	40	9	3	3	3
Space Heating Cons. [MWh/yr]	7.7	4.6	3.1	3.9	3.9	3.9
Space Cooling Cons. [MWh/yr]	5.9	5.8	5.9	5.8	5.8	5.3
Water Heating Cons. [MWh/yr]	4.7	4.7	4.7	4.7	2.4	2.4
Vent. System Cons. [MWh/yr]	-	-	0.9	0.9	0.9	0.9
Total Cond. Cons. [MWh/yr]	18.3	15.0	14.5	15.2	13.0	12.5
Rel. Cons. to Case # 1	1.22	1.00	0.97	1.01	0.86	0.83

Table 14a:
Annual *Primary* Energy Consumption for House in: **Bismarck**

case number	0	1	2	3	4	5
Space Heating Cons. [MWh/yr]	27.7	19.6	14.0	17.5	17.5	17.5
Space Cooling Cons. [MWh/yr]	3.0	3.0	3.0	2.9	2.9	2.6
Water Heating Cons. [MWh/yr]	7.8	7.8	7.8	7.8	7.3	7.3
Vent. System Cons. [MWh/yr]	-	-	2.6	2.6	2.6	2.6
Total Cond. Cons. [MWh/yr]	38.5	30.4	27.4	30.8	30.3	30.0
Rel. Cons. to Case # 1	1.26	1.00	0.90	1.01	1.00	0.99

Table 14b:
Annual *Primary* Energy Consumption for House in: **Lake Charles**

Space Heating Cons. [MWh/yr]	3.2	1.8	1.0	1.4	1.4	1.4
Space Cooling Cons. [MWh/yr]	10.0	9.5	9.7	9.5	9.5	8.6
Water Heating Cons. [MWh/yr]	7.8	7.8	7.8	7.8	7.3	7.3
Vent. System Cons. [MWh/yr]	-	-	2.6	2.6	2.6	2.6
Total Cond. Cons. [MWh/yr]	21.0	19.1	21.1	21.3	20.8	19.9
Rel. Cons. to Case # 1	1.10	1.00	1.10	1.11	1.09	1.04

Table 15a:

Annual Energy Cost for House in: **Bismarck**

case number	0	1	2	3	4	5
Space Heating (Gas) [$/yr]	615	435	310	388	388	388
Space Cooling (El.) [$/yr]	45	45	45	43	43	39
DHW (Gas/El.) [$/yr]	174	174	174	174	147	147
Vent. Syst. Cost [$/yr]	--	--	51	51	51	51
Total Energy Cost [$/yr]	834	654	580	656	629	625
Rel. Cost to Case # 1	1.28	1.00	0.89	1.00	0.96	0.96

Table 15b:

Annual Energy Cost for House in: **Lake Charles**

Space Heating (El.) [$/yr]	70	35	20	27	27	27
Space Cooling (El.) [$/yr]	234	222	227	222	222	200
DHW (Gas/El.) [$/yr]	160	160	160	160	171	171
Vent. Syst. Cost [$/yr]	--	--	61	61	61	61
Total Energy Cost [$/yr]	464	417	468	470	481	459
Rel. Cost to Case # 1	1.11	1.00	1.12	1.13	1.15	1.10

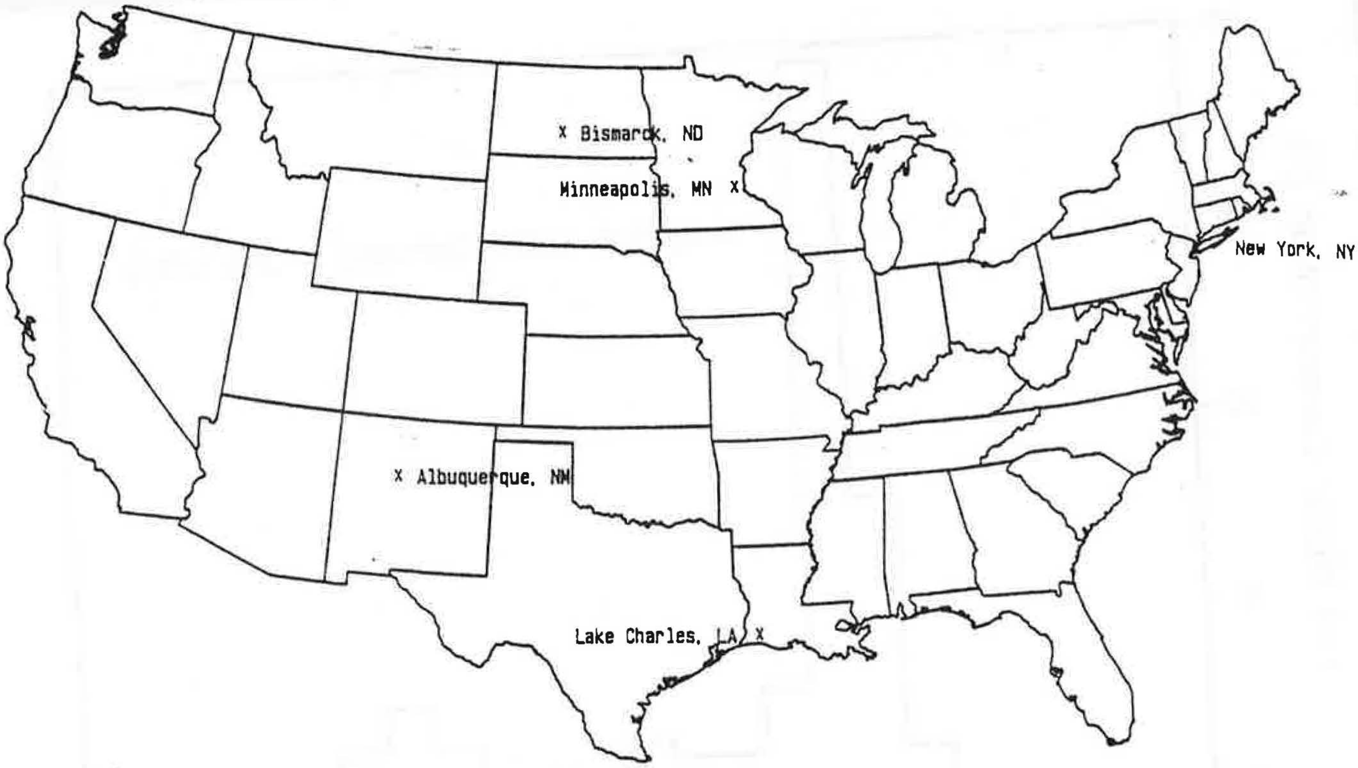

Figure 1. Map of site locations

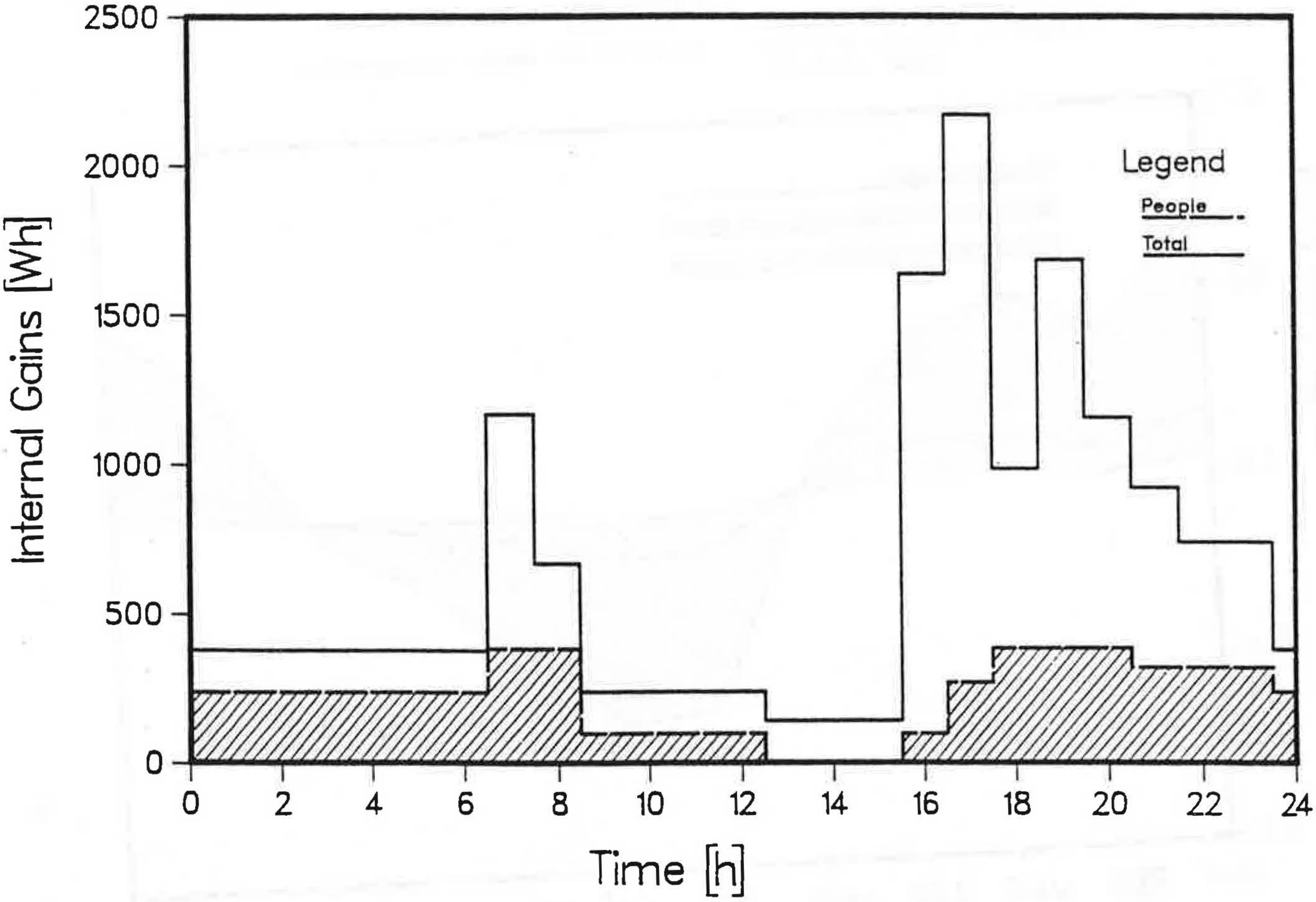

Figure 2. Schedule for internal gains

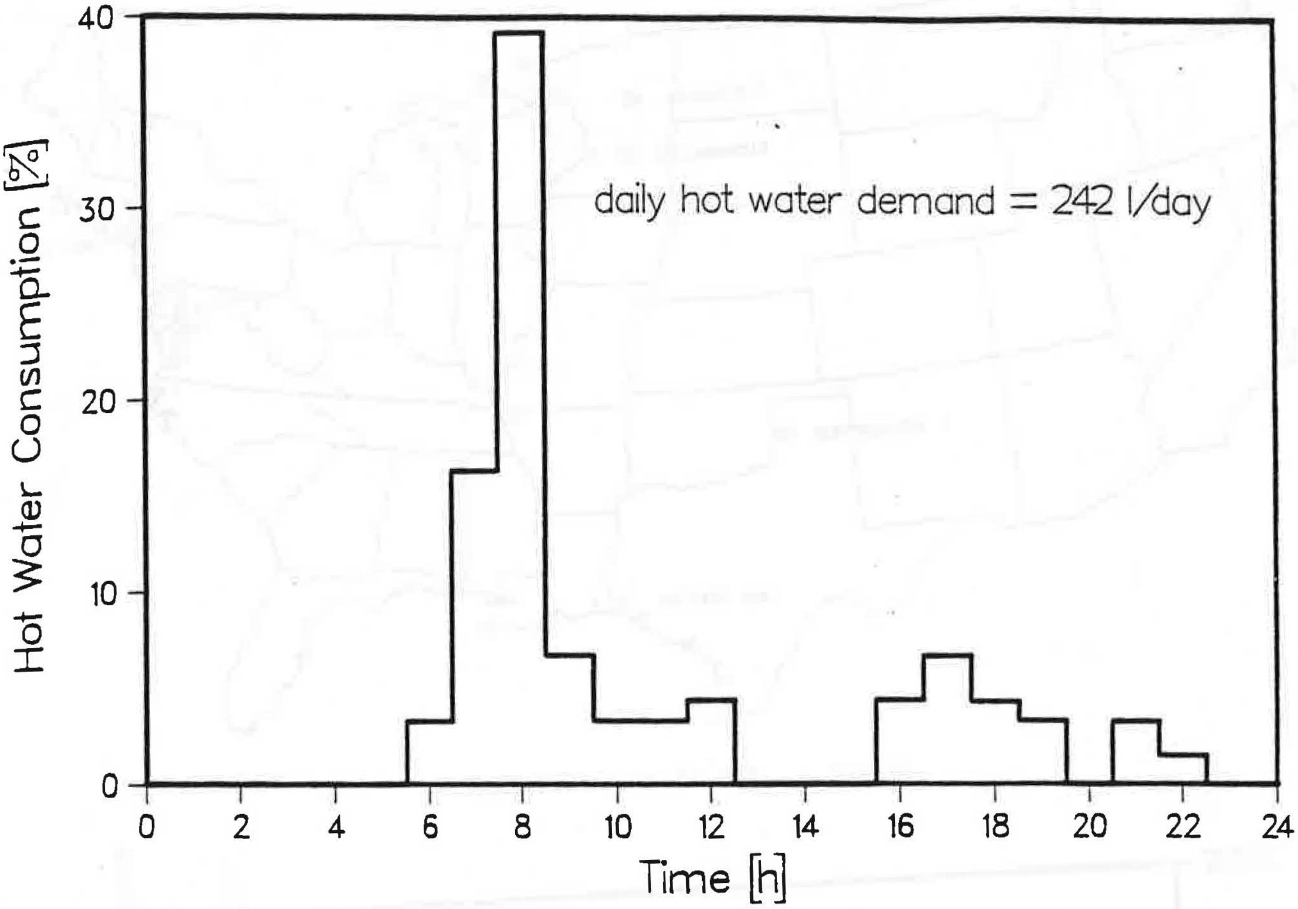

Figure 3. Hourly profile of domestic hot water consumption (NSDN profile)

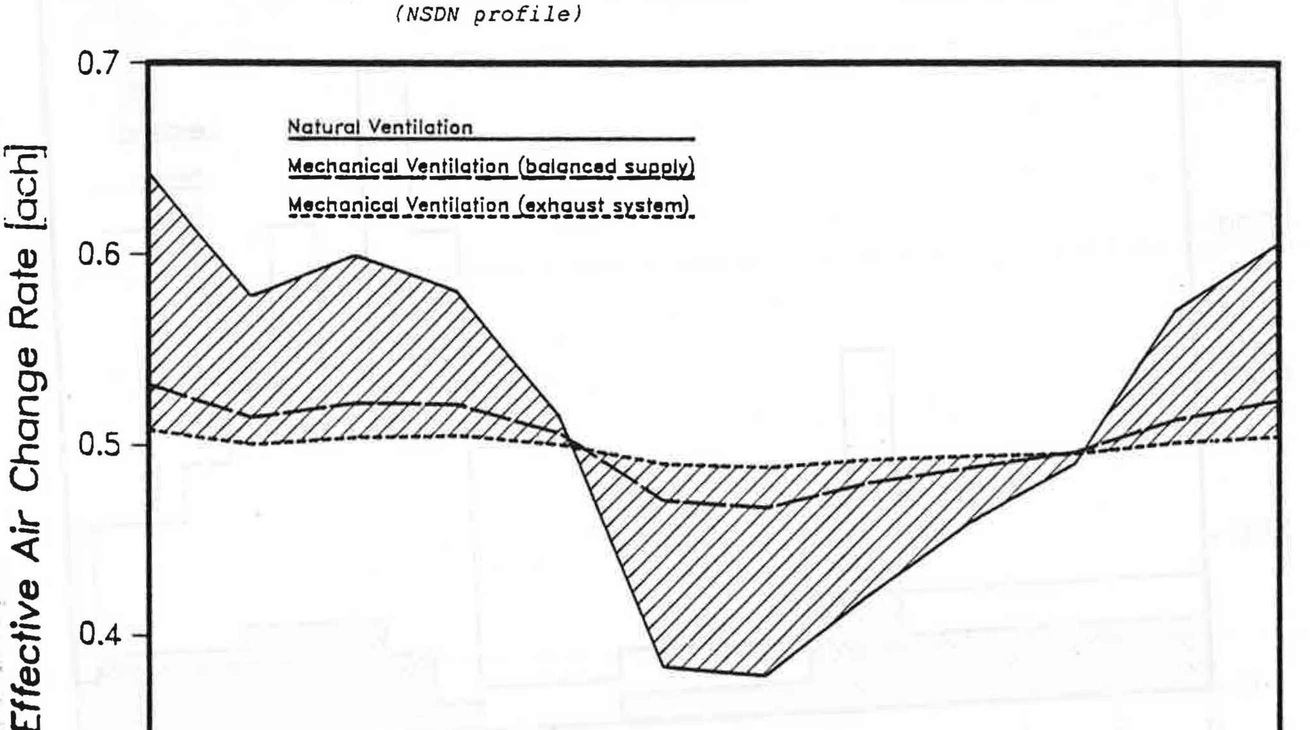

Figure 4. Effective air change rate for Bismark