PERFORMANCE OF ADVANCED VENTILATION SYSTEMS RESULTS OF REAL TIME MONITORING IN TWELVE NORTHWEST HOMES

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A subsample of 12 houses received high-level, real time monitoring under the Bonneville Power Administration's Residential Construction Demonstration Project (RCDP), Cycle 1. Five had exhaust air heat pumps and seven had air-to-air heat exchangers.

Twenty data channels per house were recorded and archived as hourly sums or means. Among these were indoor and outdoor temperatures, supply and exhaust air flow rates, and indoor and exhaust air humidities. In addition, blower door tests, and passive tracer gas (perfluorocarbon and SF_6) ventilation tests were performed. Control system configuration was recorded and occupants were interviewed regarding their system operating practices.

The various ventilation test results are compared. The relative findings of those tests are explored for consistency with prevailing theories concerning combining stack, wind, and mechanical ventilation, and time integration of varying ventilation rates. Distinction is made between thermally relevant ventilation and air quality relevant ventilation, the latter being defined as the harmonic time average of total mechanical and natural ventilation.

An important aspect of the analysis is control, i.e., timing of ventilator use. This is dominated by hot water and space heat demand in the exhaust air heat pump houses. With the air-to-air heat exchangers, the only control design and operating goal was to obtain necessary ventilation. These differences allow greatly differing ventilation schedules that substantially affect ventilation effectiveness. The resulting range of ventilation scenarios gives insight into the efficacy of alternative control strategies currently being used for both heat recovery and non-heat recovery ventilators.

INTRODUCTION

Since 1986 Bonneville Power Administration has sponsored the Residential Construction Demonstration Project (RCDP). The project's purpose is demonstration of energy conserving innovations, but it has contained a strong research component. The initial cycle of this project sponsored the building of 165 houses with heat recovery ventilators (HRV's) and state-of-the-art envelope construction. All the houses were monitored for thermal performance. Part of this research has involved much more comprehensive thermal monitoring in a sub-sample of 12 houses in Washington, Idaho, and Montana. A general analysis of that research has been prepared with a strong focus on the energy saving performance of the HRV's (Douglass and Downey 1990).

This report is an analysis of ventilation data from the 12 houses. It addresses the issue of ventilation for control of humidity and other air contaminants. The focus is on the effect of timing and flow rate of ventilation. It is not intended as an evaluation of the heat recovery equipment used to achieve that ventilation.

Accordingly, it may lend insight into other residential ventilation strategies that do not include heat recovery.

DESCRIPTION OF MONITORED HOUSES

All houses were built to the northwest region's model conservation standards in place at the time of construction. All had heat recovery ventilators (HRV's), either air to air heat exchangers or exhaust air heat pumps. Specific configuration of each house and its HRV are presented in Table 1 and discussed below. The houses are identified throughout this report by an internal tracking number to preserve the anonymity of the occupants.

Site 101

Mechanical ventilation is provided by a rotary wheel air to air heat exchanger. Distribution is by direct connection to the heat pump return air plenum. The system has no special wiring to override the distribution fan to "on" when the AAHX operates. For control, there is a dehumidistat and a manual switch in a hallway on the downstairs main living level. The dehumidistat remained set at about 45% during monitoring.

Site 124

Mechanical ventilation is provided by an air-to-air heat exchanger with a defrost system. Several controls wired in parallel are capable of operating the system. There are three twist timers in the three bathrooms, a manual variable speed switch on the unit, and a dehumidistat in the lower floor hall. The occupants report that the dehumidistat is always set at 40%, and that it usually brings the system on when the outside temperature warms up.

Site 144

Primary space heating and mechanical ventilation is a modular, forced-air, electric resistance furnace/ air-to-air heat exchanger combination located in the crawl space. There is only one speed for all fans in the unit. Wiring is such that the AAHX comes on any time the furnace fan runs. However, the AAHX can come on without causing the furnace fan to come on. One thermostat controls the furnace and it is usually kept at 70°F with no setbacks. In addition to furnace operation, several other controls in parallel can operate the AAHX. There is a dehumidistat in the first floor hallway, three twist timers in the three bathrooms, and a manual switch in the first floor hallway. The dehumidistat is set at 40% and the occupants report that it rarely turns the AAHX on. Most AAHX operation is caused by furnace operation. There are two time periods in early 1987 when the AAHX ran continuously for more than a week as if the manual switch were on.

Site 202

Mechanical ventilation is provided by an air-to-air heat exchanger with a defrost system. Several controls wired in parallel are capable of operating the system. There are two twist timers, two manual switches, and a dehumidistat. The unit had a variable speed capability initially, but by the end of monitoring, it had either failed or been disconnected so only high speed and off were possible. The occupants report that the dehumidistat is always set at 40%, but that the dominant control is usually the living room manual switch. Operating habits were not consistent over time. Early in the monitoring period the occupants tended to turn the unit off when people were sleeping or away. There are predominant on-times at about 8:00 a.m. and from 3:00 p.m. until about midnight. Later in the period, the data show periods of several days or weeks with continuous operation.

Site 212

Mechanical ventilation is provided by an air-to-air heat exchanger with a defrost system. It is indirectly connected to the furnace ducting. Several controls wired in parallel are capable of operating the system. They are door jam switches in the three upstairs bathrooms, a manual switch in the kitchen, and a dehumidistat in the first floor hallway. There is also an on/off variable speed switch on the unit

Table 1. Test House Summary

Site	Area (ft^2)	Location	Type	Alt. (ft.)	Degree Days (65 deg. F base) and Duration	Mechanical Ventilation	Space Heat
101	2,752	Boise, ID	2-story w/cond. unfin. bsmt.	2,400	5,140 from May 1, '87-Apr.30,'88	Rotary wheel AAHX directly connected to forced air space heat ducting	Split system air source heat pump
124	2,351	Boise, ID	2-story w/crawl space	2,300	5,515 from May 1, '87-Apr. 30,'88	AAHX with duct heater	Wall mounted electric resistance heaters
144	1,908	Boise, ID	2-story w/crawl	2,625	5,258 from May 1, '87-Apr. 30,'88	Modular combined AAHX and electric furnace	AAHX and electric furnace
202	2,696	Deer Lodge, MT	2-story w/crawl space	4,531	7,944 from May 1, '87-Apr. 30,'88	AAHX with duct heater	Ceiling electric resistance radiant system
212	3,040	Bozeman, MT	2-story w/heated unfinished bsmt.	5,000	7,944 from May 1, '87-Apr. 30,'88	AAHX indirectly connected to forced air space heat ducting	Electric resistance forced air furnace
228	2,356	Helena, MT	Split entry w/heated daylight bsmt.	3,500	7,944 from May 1, '87-Apr. 30,'88	Balanced EAHP	Electric baseboard heaters
235	1,821	Helena, MT	2-story w/vented crawl space	3,850	4,532 from Dec. 19, '87-Sep. 24,'87	Balanced EAHP	Electric baseboard heaters
423	2,521	Olympia, WA	2-story w/vented crawl space	150	2,409 from Jan. 17, '87-Sep. 17,'87	Rotary wheel AAHX indirectly connected for forced air space heat ducting	Electric resistance forced air furnace
439	2,798	Yacolt, WA	2-story w/vented crawl space	720	4,957 from May 1, '87-Apr. 30,'88	Exhaust-only EAHP	Electric resistance forced air furnace
449	1,967	Bothell, WA	2-story w/vented crawl space	500	4,639 from May 1, '87-Apr. 30,'88	Exhaust-only EAHP	Ceiling electric resistance radiant system
450	2,183	Olympia, WA	2-story w/vented crawl space	100	2,517 from Jan. 17, '87-Sep. 17,'88	Exhaust-only EAHP	Wall mounted electric resistance heaters
459	1,440	Federal Way, WA	2-story w/vented crawl space	100	4,757 from May 1, '87-Apr. 30,'88	Rotary wheel AAHX with duct heater	Wall mounted electric resistance heaters

itself. The three remote controls bring the unit "on" to the speed set by the switch on the unit. The bathroom door jam switches operate the unit when the bathroom doors are closed, and the dehumidistat is set for 60%. The latter rarely operates the AAHX. AAHX operation triggers the furnace fan to come on at low speed.

Sites 228 and 235

Mechanical ventilation is provided by an exhaust air heat pump. The unit is a balanced system, providing fan induced supply air as well as fan induced exhaust air. It works by recovering heat from exhaust air and simultaneously providing heat to the integral 80 gallon domestic hot water (DHW) tank and incoming supply air. The evaporator is connected to the compressor/tank unit by refrigerant lines. The evaporator is a novel device that has both a supply air and exhaust air passage. Supply air and refrigerant flow parallel to each other and counter to exhaust flow. The water tank is heated by a condenser jacket that encircles the tank, contacting the metal surface beneath the insulation. A backup resistance element actuates to heat the upper region of the tank if high water usage drops temperature below a set minimum at that elevation.

Site 423

Mechanical ventilation is provided by a rotary wheel air to air heat exchanger. Distribution is by indirect connection to the furnace. The heat exchanger supply air is introduced about 2 feet from the furnace cold air return in the ceiling of the lower floor.

Site 439

Mechanical ventilation is provided by an exhaust air heat pump. The unit is an exhaust-only model installed in a standard configuration. Make up air provisions are two through-the-wall vents on each floor. The unit works by recovering heat from exhaust air and alternately providing heat to the integral 80 gallon domestic hot water tank and a remote supplemental space heat condenser. The supplemental space heat condenser is connected to the compressor/tank unit by refrigerant lines. The water tank is heated by a condenser jacket that encircles the tank, contacting the metal surface beneath the insulation. A backup resistance element actuates to heat the upper region of the tank if high water usage drops the temperature below a set minimum at that elevation.

Controls give priority to water heating. A space heat thermostat (separate from the main furnace thermostat) activates the supplemental space heat condenser, which performs space heating if water heating demand is satisfied.

The operating instructions specify that the supplemental space heat condenser thermostat be set slightly higher than the primary space heating thermostat. This is done to ensure that the unit's supplemental space heating comes on first, before the electric furnace. This priority is prescribed not only for ventilation, but because the heat recovery function provides space heating more efficiently than the electric furnace.

A timer on the exhaust air heat pump can be set to provide a fixed minimum of fan-only ventilation during times when there is no space or water heating demand. It is the only control capable of activating ventilation in the absence of water or space heat demand. The timer on the site 439 unit is set to run the exhaust fan 15 minutes every hour, from 4:00 a.m. until 9:00 p.m., although from the data, it did not appear to be working. There is no exhaust fan flow during hours when the there is no space or water heating.

The unit exhaust air is damped down per the manufacturer's installation instructions. This is to ensure more nearly continuous running at a lower air flow rate, thus fewer and shorter off-cycles. During off cycles, exfiltration and timed fan-only cycles purge air without benefit of heat recovery.

Site 449

Mechanical ventilation is provided by the same kind of exhaust air heat pump system used for site 439, above. Make up air provisions are four through-the-wall vents, but these were not installed until April 1988, just before monitoring ended.

Site 450

Construction is timber framing with 6 3/8" Stress-skin wall panels and 8 3/8" Stress-skin vaulted roof panels. The monitoring period did not span an entire heating season. Primary space heating is by wall-mounted electric resistance fan coil heaters. Mechanical ventilation is provided by the same kind of exhaust air heat pump system used for site 439 and site 449, above. Make up air provisions are through-the-wall vents, but the occupant perceived their effect as drafty and blocked them.

Site 459

Mechanical ventilation is provided by a rotary wheel air-to-air heat exchanger. Two controls wired in parallel are capable of operating the system. They are a dehumidistat and a time clock wired in parallel with each other and a manual switch on the unit. The unit switch, which is set to "low," determines the speed at which the unit operates when either switch is on. The dehumidistat is placed in the closet with the AAHX. This appears to have been a mistake of the electrician. It was set at 50% at the time of equipment removal. Smoke testing revealed significant cross leakage from upstream exhaust into the downstream supply air. This is corroborated by a duct traverse, which showed 26% more supply air flow downstream of the heat exchanger compared to upstream. There are no balancing dampers on any of the ducts and the unit is substantially out of balance with higher supply flow.

DESCRIPTION OF MONITORING SYSTEM

Monitoring System Overview

Site data were retrieved by both real time data acquisition techniques and one-shot on-site testing. The monitoring protocol for real-time data acquisition has been discussed in detail in another report. (Douglass 1989). Data are accumulated and stored as hourly totals and running averages. At least once per week, the units are interrogated by a central computer at the monitoring contractor's home office and reset. Hourly data are archived by the monitoring contractor and transmitted by floppy disk to the client's office.

In addition to the real time data from the above automatic logging system, other ventilation data were recorded at the sites. These include passive tracer gas testing, and blower door testing of envelope leakage. The passive tracer gas testing was by a process developed in a national laboratory using perfluorocarbon (PFT) tracer chemicals. (Dietz et al. 1983). Various emitters are placed in a building. These consist of a metal capsule with a permeable rubber-like plug. The emitters give off perfluorocarbon gas at a rate determined by indoor temperature. Receivers are placed elsewhere in the building. These are capillary adsorption tubes that absorb the perfluorocarbon gas at a rate proportion to its concentration in the air. After a period of weeks the receivers are retrieved. The amount of perfluorocarbon adsorbed is determined by precise gas chromatography. From the results, the time-integrated concentration of perfluorocarbon in the building is determined. From that concentration, and the known source emission rate, the effective air change rate is inferred.

Blower door testing was performed per ASTM E779-87 to determine the equivalent leakage area and the relationship of leakage to pressure differential. These data, along with other site geographical and meteorological data, are used as inputs to a model that predicts time averaged air change rate for a heating season, or other specified time period. (Sherman and Grimsrud 1980).

List of Data Collected

Monitored parameters vary by house. The following is a generalized list of parameters monitored and reported by the data loggers:

- Indoor ambient temperature
- Outdoor ambient temperature
- HRV upstream exhaust temperature
- HRV downstream exhaust temperature
- HRV supply or supplemental space heat condenser upstream air temperature

- HRV supply or supplemental space heat condenser downstream air temperature
- HRV relative humidity exhaust warm side, i.e., upstream
- HRV relative humidity exhaust cold side, i.e., downstream
- Exhaust air in pounds mass for the hour ending with the current time
- Supply or supplemental space heat condenser air in pounds mass for the hour ending with the current time
- Heat transfer to supply or supplemental space heat condenser air for the hour ending with the current time
- Heat transfer, including latent to exhaust air for the hour ending with the current time
- kWh to space heat for the hour ending with the current time.
- kWh used by HRV supply fan or compressor/fan unit for the hour ending with the current time
- Total kWh used by exhaust air heat pump tank resistance element and compressor/fan unit for the hour ending with the current time
- Exhaust air heat pump water flow in gallons for the hour ending with the current time
- Water temperature upstream of the exhaust air heat pumps averaged over the hour ending with the current time
- Water temperature downstream of the exhaust air heat pumps averaged over the hour ending with the current time
- Water heat delivered by exhaust air heap pumps, in thousand BTU's, for the hour ending with the current time

Other data used for this analysis are the time averaged air change rates inferred from blower door testing and time averaged effective air change rates inferred from the perfluorocarbon testing.

Data Quality

A comprehensive error analysis has been performed on the data from this study in the general report on data logger monitoring (Douglass and Downey 1990). That error analysis is too lengthy for inclusion in this report, but it can be summarized. All but one of the above parameters were recorded with probable errors too small to be of significant concern in this study. The exception was air mass flow. Mass flow was obtained by a sophisticated and precise hot wire mass flow sensor, commonly used in commercial HVAC control applications. Each of these was rigidly fixed into ducting at a location and depth of penetration that was found to have velocity closely matching the mean determined by a duct traverse. Theoretically, the root sum square of instrument accuracy, quantization error, and calculation error yielded a probable error of only 2%. However, a problem not addressed in the mass flow formal error analysis was sensor contamination.

The indicated mass flow dropped as sensors became more contaminated. This was found to be a serious problem. The worse case flow drops, inferred after cleaning, were on the order of 50%. Flow degradation was too variable depending upon location, and time elapsed since cleaning, for a trivial quantification. The flow degradation was studied extensively. It only appeared to be a problem in the exhaust air flow. Exhaust air was drawn from places where there were high concentration of aerosols and humidity, i.e., kitchens and bathrooms.

A method was devised for correcting the mass flows. Fortunately, the degradation appeared to be linear with operating hours elapsed from the time of installation or sensor cleaning. This was the key used for correcting the readings. Each sensor was cleaned at least twice, with the last time being at the time of equipment removal. The increase of reading, immediately following cleaning, was used as the benchmark for correction. This increase, prorated by the fraction of operating hours elapsed since installation or prior cleaning, was added to the recorded value to produce the corrected value.

There is too much uncertainty for rigorously quantifying mass flow error. Clearly the 2% theoretical probable error is too optimistic. However, the following considerations suggest that the corrected mass flows should yield time averaged mechanical air flow accuracy well within 15%, or less, of true flow. The supply flow was considered equally with the exhaust flow in determining mechanical air change rate, and the supply flows indicated no significant degradation. The worse case degradations of as much as 50% were noted at the time of cleaning and were likely double the mean degradation of the time interval since the prior cleaning. Extensive inspection of scatter plots of corrected and uncorrected mass flows generally revealed a flat curve of corrected mass flow, over time, during hours of 100% on-time.

It is likely that the accuracy of these mechanical ventilation measurements is at least comparable to that achieved from passive tracer gas techniques and infiltration models based on blower door testing. These respectively require delicate gas chromatography and difficult assumptions about mixing, and difficult assumptions about distribution of leaks, site terrain, and shielding.

DISCUSSION OF FINDINGS

An objective of this research has been to compare the thermally relevant ventilation rate with the temporally effective ventilation rate. The former is the arithmetic mean air change rate over a heating season, and it has a directly proportional effect upon heating load. The latter is also a mean air change rate, but it is the harmonic mean, i.e., it is computed from the mean of the reciprocals. This is appropriate because, given a constant source rate, air contamination is proportional to the inverse of ventilation. This reciprocal is known as mean residence time. It can also be thought of as the mean ratio of contaminant concentration to contaminant specific source rate.

Both arithmetic and harmonic mean air change rates are the same if the air change rate remains constant during the period of averaging. Unfortunately, it does not. As a consequence, the practice of discussing air quality relevant residence time in terms of its reciprocal, air change rate, can be misleading. It would be simple to compare the respective thermal and air quality relevant air change rates, of mechanical ventilation systems, if there were not coincident natural infiltration. Difficulty arises because contaminant removal is not linear with ventilation. Each increment of increased ventilation has diminishing air quality benefit. (This should be obvious if one considers that at low ventilation rate, each cubic foot of expelled air is relatively more contaminated than a cubic foot at high ventilation rates.) The analytical problem this poses is that a mechanical system with periods of off-hours would actually generate a zero seasonal average air quality relevant air change rate. This is not merely a mathematical aberration. If indeed there is absolutely zero natural infiltration, a significant period of no mechanical ventilation would saturate the indoor air with contaminants from any indoor sources.

It was decided that mechanical ventilation's effect on air quality could only be appropriately analyzed by *combining* it with natural ventilation. This begs hourly data for natural ventilation, which were not recorded. To record these data would have required an extremely sophisticated, sensitive, and costly real time tracer gas system. Even if the data had been recorded, they would have been unique for the given home and might not lead to results readily generalizable to other structures with the same HRV and control system installed.

An appropriate answer to the air quality vs. thermal effective ventilation question may simply be that they should not be compared. At least they should not be compared as if they are the same thing. Important differences are as follows:

- Thermally relevant air change rate is computed or predicted for the heating season. It is intended to be an effective mean for those hours during which heating is required and it may be totally different from the mean that would be obtained over all hours.
- Air quality relevant ventilation rate must pertain to all hours in the year because air quality is a year-round need. Also many contaminants are presumed to have a cumulative effect rather than

a chronic effect. This makes it doubly important to have insight into the year round conditions affecting exposure.

- Negative effects from air contaminant sources vary inversely with ventilation, and are aggregated with a harmonic mean. The harmonic mean is mathematically not equivalent to the arithmetic (conventional) mean.
- As they have been measured in RCDP, the blower door test and infiltration model predict an arithmetic mean. Even more important, it only predicts the natural ventilation that would occur if there is no mechanical system functioning. By contrast, the RCDP PFT test is a passive tracer gas technique for measuring (not predicting) total effective ventilation. Since it works as an accumulating receptor for a proxy contaminant, it is inherently computing a mean residence time that must be inverted to yield air change rate. Accordingly it represents a harmonic mean of *combined* mechanical and natural infiltration.

The three measures of ventilation are presented together in Table 2. They should be compared only with appropriate cognizance of the differences noted above. PFT whole house results are given for the time period for which they are available. They represent the air quality relevant air change rate for a given time period. The thermally relevant mechanical air change rate is computed for the same time period except as noted. There are a total of 14 comparisons for nine houses. These represent the total number of cases for which good PFT data were obtained.

Figure 1 depicts the data in Table 2 in clustered bar format. The "W" or "S" site number suffix identifies test periods as winter or spring respectively. These designations are nominal; the exact dates are as shown in Table 2. For reference, the seasonal blower door test results are shown as a third series in the clustered bars. It should be emphasized that blower door figures are a prediction of the thermally relevant ACH for the entire heating season, not for the specific duration of the PFT and monitored mechanical ventilation testing. Also the blower door model predicts what the air change rate would be in the absence of HRV operation.

Unusual Site Observations

In some of the houses things happened that could have bearing on the test results. Where these are known, they are discussed below.

Site 144. This site is unusual in that the blower door test generated a high ACH prediction compared to the other sites, and compared to the PFT results. It is possible that something was not sealed properly for the testing. The house had a modular forced air furnace/air-to-air heat exchanger system. The blower door technician's notes indicated that the greatest air leakage was from the heating registers and return. He did not check connections at the unit because it is inconveniently located in the crawlspace. He did seal the supply and exhaust hoods at the exterior of the house before testing. It is entirely plausible that all the tests are valid. A substantial leak, confined to one location, could score high on a blower door test and low on a PFT test.

Site 212. This house had the greatest excess of blower door predicted ACH over PFT results. The blower door technician's notes indicated that the greatest leakage was at a masonry to frame wall joint. The best explanation for the discrepancy was a message received from the homeowner during the PFT monitoring period, subsequent to the blower door test, stating, "The house is being tightened for holes." As with site 144, it is entirely plausible that all the tests are valid. A substantial leak, confined to one location, could score high on a blower door test and low on a PFT test.

Site 423. This site had one of the more variable mechanical ventilation profiles. Occupants tended to turn the heat exchanger off when they were away at work, which provided a strong variation by hour of day. Frequently they forgot to turn it off, which produced a strong day to day variation. This may have resulted in less effective ventilation, because it was the only balanced system house that had a PFT

Table 2. Alternative Air Change Measures

Site	Winter '86-'87 Start	Stop	PFT ACH	Mech ACH	BD ACH	HRV
101	06-Dec	15-Mar	0.48		0.13	Rotary Wheel
101*	18-Dec	15-Mar		0.40	0.13	Rotary Wheel
124	08-Dec	27-Feb	0.49		0.33	AAHX
124*	18-Dec	27-Feb		0.42	0.33	AAHX
124	27-Feb	08-May	0.42	0.24	0.33	AAHX
144	05-Dec	27-Feb	0.43		0.61	Packaged
144*	18-Dec	27-Feb		0.31	0.61	Packaged
212	15-Dec	25-Feb	0.21		0.52	AAHX
212*	18-Dec	25-Feb		0.07	0.52	AAHX
235	17-Dec	25-Feb	0.13		0.14	Balanced EAHP
235*	19-Dec	25-Feb		0.13	0.14	Balanced EAHP
235	25-Feb	08-May	0.15	0.12	0.14	Balanced EAHP
423	06-Dec	04-Mar	0.35		0.24	Rotary Wheel
423*	17-Jan	04-Mar		0.35	0.24	Rotary Wheel
423	04-Mar	16-May	0.32	0.29	0.24	Rotary Wheel
439	05-Dec	27-Feb	0.62		0.51	EAHP
439	27-Feb	11-May	0.59	0.47	0.51	EAHP
449	10-Dec	25-Feb	0.50		0.10	EAHP
449*	17-Jan	25-Feb		0.61	0.10	EAHP
449	25-Feb	05-May	0.48	0.40	0.10	EAHP
450	07-Dec	05-Mar	0.42		0.14	EAHP
450*	17-Jan	05-Mar		0.49	0.14	EAHP
450	05-Mar	12-May	0.19	0.25	0.14	EAHP

*This line is for comparison with the line above. Two lines were used because of a mismatch in start dates.

ACH lower than the mean mechanical ventilation rate. Even this difference was not severe considering the extreme variability of the mechanical ventilation. In the spring the PFT ACH became higher than the mechanical ventilation rate. This is likely influenced by many warm days, which encouraged open windows. The inferred lower effectiveness, in winter, does not indicate less healthful air for the occupants, because the times of zero mechanical ventilation corresponded with periods of occupant absence.

Site 439. The first test period occurred prior to installation of data logger equipment. The second PFT test period occurred from February 27 to May 11, 1987 and indicated an average effective air change rate of 0.64 ACH. Mechanical ventilation for the corresponding time period is 0.46 ACH. These numbers are difficult to compare because natural infiltration could have been quite large in the final weeks of the second PFT test as weather conditions were moderate and the occupants may have opened windows.

Site 450. Figure 1 shows a dramatic change in the ventilation of site 450 between the winter and the spring PFT tests. There was a reduction in EAHP operation caused by the mild spring weather demanding less heating from the space heat condenser. The PFT air change rate dropped proportionately with the mechanical ventilation (Figure 1). As the weather became warmer, there were more hours when neither space heat nor water heat were in demand. This change in mechanical ventilation

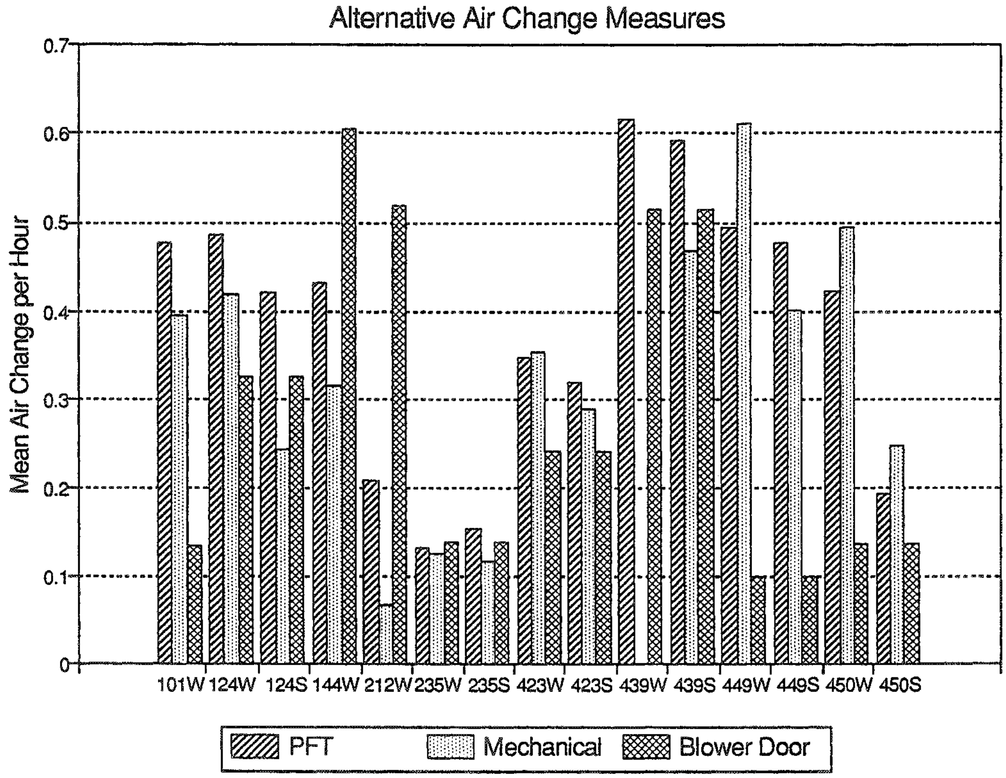

Figure 1.

with the weather is very apparent in Figure 2. This shoulder season of relative under-ventilation could have easily been mitigated if the exhaust air heat pump ventilation override timer had been set for more "on" hours. However, it should be noted that the occupant perceived the house to be drafty and even plugged some of the slot vents.

Ventilation operation patterns coincide well with seasonal periods of space heat demand. The heating season is defined as the months of October through March. The mean mechanical ventilation rate is 0.44 air changes per hour during the heating season and 0.24 air changes per hour during the shoulder and summer seasons. During the heating season the ventilation system is operated almost constantly. During the summer and shoulder seasons the system is operated only as needed for water heating or ventilation only. These figures do not include effects of natural infiltration. Long periods of "off" hours are not found.

General Observations

For most houses, it is difficult to say whether the comparative results are reasonable. PFT ventilation is consistently lower than the sum of mechanical plus natural infiltration, and it should be. Time variable ventilation tends to make the PFT results lower. Interaction between mechanical and natural infiltration also causes the combination to be less than purely additive. Both influences occur in all the houses. However, there are certain patterns worthy of note. Alone, they are not conclusive, but they support current theories and suggest areas where further investigation is appropriate.

The PFT or air quality effective ACH appears to be strongly correlated to the mechanical ventilation. This relationship is depicted in Figure 1. The relationship appears surprisingly strong considering the various reasons the two could differ. This implies that HRV mechanical ventilation systems, with

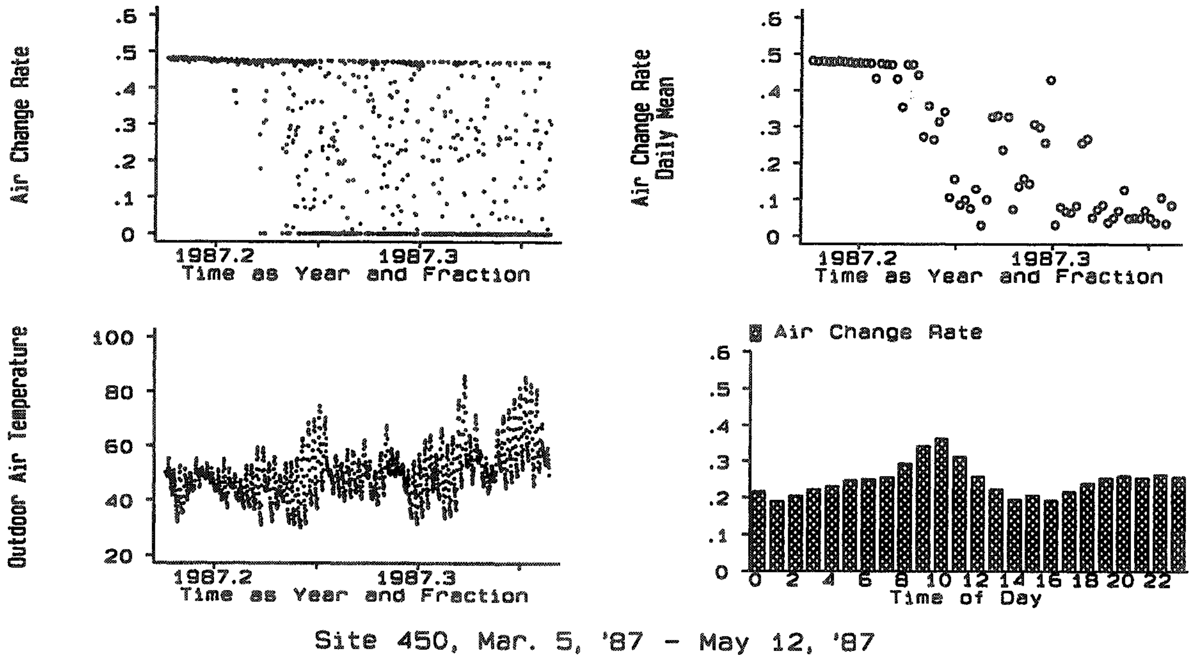

Figure 2.

adequate installed flow capacity, are working to achieve air quality relevant effective ventilation in the test houses.

The small number of test houses does not support extrapolating this expectation to all houses with mechanical ventilation. However, it allows direct comparison of a known mechanical ventilation profile with achieved ventilation effectiveness. This suggests that similar results may be achieved in other houses with HRV's or non-heat-recovery ventilators provided the installed capacity, and spatial and temporal distribution patterns are similar.

To visualize the temporal distribution pattern, cluster graphs of mechanical ACH were created for each house showing respectively, actual hourly ACH, daily mean ACH, outside temperature, and mean ACH by time of day. These were prepared for time periods coinciding with the PFT tests. These are too voluminous for inclusion in entirety, but the one with greatest variation, Site 450, Spring testing, is shown in Figure 2. The outside temperature was plotted because it is indicative of times when mild weather weakens stack effect and warm weather increases likelihood of open windows.

PFT testing may understate the benefit of good spatial distribution provided by mechanical systems. Exhaust removal points were in bathrooms and kitchens where humidity, odors, and cooking aerosols are known to originate. The PFT sources, acting as proxy contaminants, were not placed near the exhaust locations where many real contaminants originate.

With the sole exception site 423, the PFT ACH was always higher than the mechanical in houses with balanced ventilation. This is a reasonable trend since any natural ventilation should be additive with the mechanical. All of the sites above number 420 in Figure 1 have exhaust only ventilation systems. Some of these have mechanical ventilation ACH greater than the PFT ACH. This is consistent with prevailing theory that exhaust-only ventilation eclipses part of the would-be natural ventilation.

It is somewhat of a mystery why, even in balanced systems, PFT ventilation is consistently more closely correlated to mechanical ventilation alone than to the sum of mechanical plus natural. The sum exceeds PFT ACH in all cases, but it exceeds it by a highly variable amount. The greatest excess was in Montana and Idaho sites, where temperature and exposure cause the infiltration model to yield higher values of both stack and wind infiltration for a given leakage area. This raises the possibility of either or both of the following: (1) The model is overstating infiltration. (2) The extreme variability of natural ventilation is not as effective at diluting perfluorocarbon as a uniform rate would be. The possibility of the latter is consistent with the harmonic averaging mechanism inherent in the PFT test method, and should reflect the effectiveness of natural ventilation for purging real contaminants.

CONCLUSIONS

The 12 case studies imply that mere existence of high infiltration (as inferred from blower door tests with infiltration model simulations) may be inadequate for healthful ventilation.

Mechanical ventilation works to achieve effective ventilation, but only if systems have adequate capacity and are operated. This qualifier, though seemingly obvious, can be easily forgotten. It has implications for building codes or other programs to promote conservation or enhanced indoor air quality. These must ensure that systems achieve intended installed capacity and are "user friendly" enough, in terms of noise, drafts, and controls, to be operated at high utilization.

The close connection observed between perfluorocarbon testing and measured mechanical ventilation needs further investigation. A common assumption in the energy conservation community is that building tight houses with controlled (i.e., mechanical) ventilation is much better than building leaky houses and trusting infiltration to give appropriate ventilation. This has been supported by other research. (Feustel, Modera, and Rosenfeld 1986). The results of this study tend to confirm the ability of mechanical systems to provide more effective ventilation than natural leakage.

However, many in the building community remain skeptical about this matter. It would be desirable to extend the population of houses with real time mechanical ventilation data so that the contribution of mechanical ventilation toward air quality effective ventilation can be compared with that of predicted natural infiltration.

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