NISTIR 6455

A Modeling Study of Ventilation in Manufactured Houses

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ABSTRACT

The HUD Manufactured Home Construction and Safety Standards (Part 3280, 1994) contain requirements intended to provide adequate levels of outdoor air ventilation in manufactured homes. In the implementation of these standards, questions have arisen regarding the impact and significance of some of these requirements. Some of these questions relate to the actual ventilation rates in homes built to the standards and the means of providing supplemental mechanical ventilation to meet the requirements of the standards. Other questions have arisen as to how specific ventilation system components such as duct leakage, local exhaust fans and ventilation inlets affect ventilation rates, air movement patterns, and building pressures. In order to obtain some insight into these issues, the multizone airflow and indoor air quality program CONTAM was used to simulate a double-wide unit under several different ventilation scenarios. These scenarios include envelope infiltration only, infiltration plus the effects of local exhaust and forced-fan operation, an outdoor air intake duct installed on the forced-air return, and whole house exhaust with and without passive inlet vents. Simulations were performed to predict outdoor air ventilation rates into the house due to infiltration and mechanical ventilation, interzone airflow rates between the rooms, building air pressures, and ventilation air distribution. Annual simulations were performed in three cities to assess ventilation rates and energy consumption associated with these scenarios. The results show that despite the assumption in the HUD standards that infiltration contributes 0.25 h^{-1}, the predicted infiltration rates are lower than this value for many hours of the year. The supplemental ventilation systems investigated in this study provide ventilation rates that meet or exceed the total ventilation requirement of 0.35 h^{-1}, but the impacts of such systems are dependent on their operating schedules. In addition, in these simulations, the impacts of a whole house exhaust fan are independent of whether this fan is located in the main living area or in a bathroom off the main living area. Also, for the case of ventilation with a whole house exhaust fan, the inclusion of passive inlet vents is not critical given the level of envelope airtightness used in these simulations. The results of these simulations are presented and discussed, and recommendations are made for changes to the HUD standards and for future research.

Keywords: building performance, manufactured homes, modeling, residential, ventilation

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EXECUTIVE SUMMARY

The HUD manufactured home standards, referred to as the Manufactured Home Construction and Safety Standards (MHCSS), cover the design and construction of all manufactured homes in the United States and contain a number of requirements related to ventilation (HUD 1994). Since these standards have been issued, questions have arisen regarding some of the ventilation–related requirements and their implementation. This report describes a study that addresses some of these questions and provides technical input relevant to their implementation and future revision.

Based on a review of the literature on ventilation in manufactured homes and discussions with individuals in the field, the following issues were identified as relevant to the study:

- Validity of the 0.25 h^{-1} assumption for infiltration
- Impact and effectiveness of an outdoor air inlet to the furnace return
- Impact and effectiveness of whole house exhaust fan with passive inlet vents
- Impact and effectiveness of whole house exhaust fan without passive inlet vents
- Location of whole house exhaust fan in the main living area versus the bathroom

In order to address these issues, simulations were performed in a double-wide manufactured house using the multizone airflow model CONTAM. The house model includes the effects of exterior envelope leakage, interior partitions, forced-air distribution and associated duct leakage, exhaust fan operation and outdoor weather. Transient annual simulations were performed for three cities, Albany, Miami and Seattle, and steady-state analyses were performed for specific conditions of weather and fan operation.

Summary of Simulation Results

The airflow simulations were focused on building ventilation rates relative to the requirements in the MHCSS. Additional simulations and analyses were performed to better understand the airflow characteristics of the simulated house, including pressurization tests to determine the airtightness of the building envelope and analyses of airflow patterns between the major volumes of the house. In addition, effective air change rates are presented as a measure of the indoor air quality impacts of different ventilation approaches, and the age of air to characterize outdoor air distribution to the different zones of the house. The energy consumption associated with the different ventilation scenarios is also discussed.

The simulations did not address contaminant concentrations in the house or occupant exposure to contaminants. While there are a number of indoor air quality issues of interest in manufactured housing, such as moisture and formaldehyde levels, contaminant analysis was beyond the scope of this project.

The results of the simulated pressurization tests reveal that the airtightness of the model house is typical of recent manufactured home construction, as intended. The effective leakage area at 4 Pa (0.016 in. H_20) is 373 cm^2 (57.8 in^2) including leakage associated with the envelope, duct system and exhaust fans. The corresponding air change rate at 50 Pa (0.2 in. H_20) is 6.5 h^{-1}. In addition, analysis of the airflow patterns within the house show that an upward airflow pattern dominates under conditions of zero wind speed and a higher indoor air temperature than outdoors. This pattern leads, in general, to most of the air entering the building at lower elevations, including from beneath the belly of the house. Air entry from beneath the house has potential indoor air quality implications, as contaminants such as water vapor, radon and pesticides can be drawn into the occupied space by such airflow.

In order to understand the impacts of fan operation and interior door position on building air change rates, steady-state airflow simulations were performed for several different conditions. Table ES-1 presents these air change rates, all of which correspond to an indoor-outdoor air temperature difference of 20 °C (36 °F) and zero wind speed. These air change rates are all for the case of open interior doors. As discussed in the report, closing the interior doors does not have a large impact on these air change rates.

Conditions	Air change rate (h^{-1})
Forced-air fan off	
All exhaust fans off	0.28
Both bath fans on; kitchen fan off	0.72
Kitchen fan on; bath fans off	0.73
Forced-air fan on	
All exhaust fans off	0.55
Both bath fans on; kitchen fan off	1.22
Kitchen fan on; bath fans off	1.22
Inlet on forced-air return	
All exhaust fans off	0.65
Both bath fans on; kitchen fan off	1.25
Kitchen fan on; bath fans off	1.25
Passive inlet vents and whole house exhaust in KLA	
Whole house exhaust fan on	0.50
Exhaust and forced air fan on	0.79
Exhaust off and forced-air fan on	0.61
Passive inlet vents and whole house exhaust in BATH1	
Whole house exhaust fan on	0.50
Exhaust and forced air fan on	0.85
Whole house exhaust in KLA, no passive inlet vents	
Whole house exhaust fan on	0.44
Exhaust and forced air fan on	0.79

All values correspond to 20 °C (36 °F) temperature difference and zero wind speed.
To convert the air change rates to L/s (cfm), multiply by 70 (148).

Table ES-1 Air Change Rates for Different House and Fan Configurations

At an indoor-outdoor air temperature difference of 20 °C (36 °F) and zero wind speed, the house has an air change rate of 0.28 h^{-1}. Operating both bath fans, or the kitchen exhaust fan, raises the air change rate to about 0.7 h^{-1}. Due to the supply duct leak into the crawl space, operating the forced-air fan depressurizes the building, increasing infiltration into the building and yielding an air change rate of 0.55 h^{-1} with all exhaust fans off and interior doors open. The supplemental ventilation strategies investigated in this study increase the air change rate of the house significantly. With an outdoor air inlet duct on the forced-air return, the air change rate is about 0.7 h^{-1}. A whole house exhaust fan in combination with passive inlet vents yields an air change rate of 0.5 h^{-1} with the forced-air fan off and about 0.8 h^{-1} with the fan on. The same whole house exhaust fan without the inlet vents results in an air change rate of 0.44 h^{-1} with the forced-air fan off and 0.79 h^{-1} with it on. Therefore, the supplemental ventilation systems all have the capacity to meet the 0.35 h^{-1} ventilation requirement. Their actual impact in practice depends on how often they are operated and how the operating time is determined.

Indoor-outdoor pressure differences were determined in the main living zones of the house for these steady-state airflow simulations under fixed weather conditions. For the first three cases in Table ES-1 with the forced-air fan off, the pressure differences are all around 1 Pa (0.004 in. H_20) and are uniform for all the building zones. Operating the forced-air fan reduces the interior pressure due to the supply duct leak into the crawl space. The combination of forced-air fan operation and exhaust fan operation leads to indoor pressures about 5 Pa (0.020 in. H_20) lower than outdoors. Operation of the outdoor air inlet on the forced-air return decreases the magnitude of house depressurization by about 1 Pa (0.004 in. H_20), since it acts like a return duct leak to balance the supply leak. The location of the whole house exhaust fan in the KLA or BATH1 zone has little impact on these pressure differences.

In order to understand the performance of these different ventilation approaches year-long simulations were conducted in Albany, Miami and Seattle for the following cases:

- Case #1: Envelope leakage only; no fans or ducts in model.
- Case #2: Bath and kitchen exhaust fans on schedules.
- Case #3: Forced-air fan operation based on outdoor temperature.
- Case #4: Outdoor air intake on forced-air return.
  - A: Forced-air operation based on outdoor temperature.
  - B: Forced-air operation during occupancy.
- Case #5: Whole house exhaust with passive inlet vents.
  - A: Whole house exhaust fan in KLA zone, operated on (Case #2) exhaust fan schedules.
  - B: Whole house exhaust fan in KLA zone, during occupancy.
- Case #6: Whole house exhaust without passive inlet vents.
  - A: Whole house exhaust fan in KLA zone, operated on (Case #2) exhaust fan schedules.
  - B: Whole house exhaust fan in KLA zone, during occupancy.

Table ES-2 contains the annual mean air change rates for all these cases, as well as the percent of hours over the year during which the air change rate is below the reference values of 0.25 h^{-1} (the infiltration assumption in the MHCSS) and 0.35 h^{-1} (based on ASHRAE Standard 62-1999). On an annual basis, the envelope infiltration only case (#1) has mean air change rates below the 0.25 h^{-1} in all three cities. The hourly air change rate for Case #1 is below this value for 56 %, 100 % and 74 % of the year in Albany, Miami and Seattle respectively. Operating the exhaust fans on an occupancy-based schedule (Case #2) increases the annual mean air change rates, and the means are consistent with the MHCSS value in Albany and Seattle. However, there are still a high percentage of hours below 0.25 h^{-1} is all three cities. Case #3 can be considered a relevant baseline case since it includes both local exhaust and forced-air fan operation. The mean air change rate is above the 0.25 h^{-1} reference in Albany and Seattle, but is below that value in Miami. The hourly air change rate is below 0.25 h^{-1} for 34 %, 78 % and 33 % of the year in the three cities. As expected, the mechanical ventilation approaches have higher mean air change rates; the relevant reference for these cases is 0.35 h^{-1}. The mean air change rates are above this value for almost all of the supplemental ventilation cases in Albany and Seattle, but there are still a significant number of hours during the year below this reference value. Case #4B has the highest air change rates and the lowest fractions of hours below 0.35 h^{-1} due to the large number of hours during which the supplemental ventilation system operates. Case #5B and #6B also have high air change rates and low percentages, again due to the operating schedule. The means in Miami are all less than or equal to 0.35 h^{-1}, except for Case #4B.

	ALBANY			**MIAMI**			**SEATTLE**		
Case/Condition	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}
1/Envelope leakage only	0.22	56	88	0.10	100	100	0.20	74	99
2/Scheduled exhaust fans	0.27	48	77	0.16	86	91	0.25	64	85
3/Forced-air operating on outdoor temperature	0.34	34	53	0.19	78	90	0.32	33	71
4A/Intake on forced-air, operating on outdoor temperature	0.37	32	46	0.20	73	90	0.33	30	60
4B/Intake on forced-air, occupancy schedule	0.59	13	18	0.51	24	33	0.55	14	24
5A/Passive inlets and whole house exhaust on exhaust schedule	0.41	28	42	0.26	66	84	0.38	24	52
5B/Passive inlets and whole house exhaust on occupancy schedule	0.50	16	29	0.34	38	72	0.47	13	29
6A/ Whole house exhaust (no inlets) on exhaust schedule	0.36	34	52	0.22	78	86	0.34	33	71
6B/ Whole house exhaust (no inlets) on occupancy schedule	0.46	21	35	0.31	52	78	0.43	12	37

Table ES-2 Summary of Annual Air Change Rates

Effective air change rates, as defined in ASHRAE Standard 136, are presented as a measure of the indoor air quality performance of the different ventilation scenarios. The effective air change rates are around 0.2 h^{-1} or less for Case #3 in all three cities, indicating that in terms of indoor air quality these baseline conditions correspond to a constant air change rate below the 0.25 h^{-1} reference value. The relevant reference for the supplemental ventilation approaches is 0.35 h^{-1} based on ASHRAE Standard 62-1999. The effective air change rates are below 0.35 h^{-1} when the supplemental ventilation systems are operated on the more limited schedules (Cases #4A, #5A and #6A) and closer to 0.35 h^{-1} when they are operated during building occupancy (the B schedules).

The predicted air change rates are compared with the limited measurements of air change rates in manufactured homes. The data for this comparison were measured in recently constructed homes, but not homes built to the most demanding energy efficiency standards. And while there are no measured data in the literature that correspond to the exact conditions of the simulations, the data that are available are consistent with the predicted air change rates.

Age of air values are used to examine the distribution of ventilation air within the building, with results presented in terms of inverse age of air in units of h^{-1}. For the case of weather-driven infiltration, the inverse ages are fairly uniform for the four zones indicating good air distribution. With the forced-air fan on, the inverse ages are also essentially uniform across the four zones with the interior doors open. With the interior doors closed, the inverse ages increase somewhat, but the uniformity across the zones is maintained. Operating the forced-air fan would be expected to mix the interior air fairly well, since the airflow rate through the forced-air distribution system corresponds to almost 6 air changes per hour. Similarly, with the intake on the forced-air return, the inverse ages are uniform across the four zones due to the forced-air fan operation. With the passive inlet vents providing ventilation air independent of the forced-air system, there are some nonuniformities in air distribution. The uniformity of air distribution as characterized by the age of air is essentially independent of whether the whole house exhaust fan is located centrally in the KLA zone or in the BATH1 zone. And for the case of whole house exhaust without the inlet vents, the level of variation among rooms is about the same as that with the vents.

The energy consumption associated with the ventilation approaches was evaluated based on estimates of the energy associated with heating, cooling and fan operation for the three cities. The total energy consumption for each ventilation case is roughly proportional to the annual mean air change rate for each city. The energy impact of the two supplemental ventilation strategies depends on the operating schedule. With the forced-air fan and intake operating based on the outdoor air temperature the increase in energy consumption is not large, but the air change rates do not increase significantly either. However, when the forced-air fan and the intake operate whenever the building is occupied, the energy use increases significantly. Relative to the case without the intake, the energy consumption increases by 70 %, 174 % and 94 % respectively in Albany, Miami and Seattle. An important portion of this increase is the fan energy, which roughly triples in Albany and Seattle, and increases by a factor of about seven in Miami. The whole house exhaust fan with passive inlet vents also increases energy consumption. With the whole house exhaust fan operating whenever any exhaust fan would otherwise operate, the energy consumption relative to the forced-air only baseline case increases by roughly 15 % in the three cities. With the whole exhaust fan operating whenever the house is occupied, the energy increase is larger. Again relative to the forced-air case, the increase is 33 %, 42 % and 39 % in Albany, Miami and Seattle, respectively. With the whole house exhaust fan alone, without the inlet vents, the energy consumption increase is slightly less than the case with the inlet vents.

Study Issues

As mentioned earlier, five issues were the primary motivation for his study. The findings of this study with respect to these issues are summarized below:

Validity of the 0.25 h^{-1} assumption for infiltration

Using a single value for a weather-driven infiltration rate is inherently problematic, given the strong dependence of infiltration on weather. As seen in these simulations, the infiltration rates vary by as much as 5 to 1 based on variations in weather conditions alone. Including the impacts of exhaust fan and forced-air fan operation more than doubles the range of variation. Nonetheless, when considering the predicted infiltration rates on an annual basis, the air change rate is below 0.25 h^{-1} for about one-third of the year in Albany and Seattle and for 70 % of the year in Miami. Note that if there were no duct leakage in the house model, these percentages would be significantly higher. Therefore, the assumption of 0.25 h^{-1} for infiltration in modern manufactured homes may be too high, but more importantly it ignores variations due to weather and fan operation.

Impact and effectiveness of an outdoor air inlet to the furnace return

Employing an outdoor air intake duct on the forced-air return duct is certainly effective in raising air change rates and distributing ventilation air throughout the house. However, the overall impact on the building air change rate is a strong function of the operating time of the forced-air system, which in turn depends on the extent of system oversizing and the use of other control strategies such as manual switches and timers. While increased forced-air fan operation provides higher ventilation rates, there is an energy cost associated with the increased fan operation. Some control strategies have been proposed to reduce this energy impact by reducing the fan speed (Lubliner 1997). Also, given the existence of significant duct leakage, this scenario suffers from excessive air change rates particularly when weather-driven infiltration is high.

Impact and effectiveness of whole house exhaust fan with passive inlet vents

The whole house exhaust with passive inlet vents provided adequate ventilation in this house and reasonable air distribution, but again the impact is highly dependent on the fan operation schedule. As implemented in the house model, these vents themselves were not particularly effective in ventilating the building. Based on the magnitude of the vent openings relative to the house airtightness, their installation basically corresponds to a 15% leakier envelope rather than a designed air intake system as they could conceivably be used. Such a system would presumably require a tighter envelope than is typically achieved in practice. Furthermore, under the conditions in these simulations, outdoor air did not necessarily enter the building through these vents, and when they did indeed act as inlets, the amount of outdoor air entering the building was not large.

Impact and effectiveness of whole house exhaust fan without passive inlet vents

The simulations with a whole house exhaust fan but without the inlet vents exhibit lower ventilation rates than with the vents as expected. However, the rates are still above the 0.35 h^{-1} requirement in the MHCSS. Again, the overall impact of the whole house exhaust fan depends on the fan operating schedule. Therefore, given the level of envelope airtightness assumed in these simulations, the passive inlet vents do not appear to be essential to the proper functioning of a supplemental ventilation system based on a whole house exhaust fan.

Location of whole house exhaust fan in the main living area versus the bathroom

For the conditions in this house model, the impact of the whole house fan did not depend much on its location. Whether the fan was in the main living area or a bathroom off the main living area did not have a significant impact on air change rates, outdoor air distribution or building pressures.

Recommendations

While the simulations performed in this study have limitations, there are a number of recommendations that can be made relevant to the construction of manufactured houses and to subsequent versions of the MHCSS.

One issue relates to the adequacy of the assumption that these houses have a base infiltration rate of 0.25 h^{-1}. These simulations show that at levels of airtightness consistent with current practice, infiltration rates are often below this value except during colder and windier weather. Also, using a single value ignores the significant variation in infiltration that exists as a function of weather. It may therefore make sense for the MHCSS to consider a more realistic treatment of background infiltration. One potential approach is to use ASHRAE Standard 136 to convert a building airtightness value from a pressurization test to an annual effective air change rate for a given climate. While this approach would have its own limitations, it would be straightforward to use since the standard contains a rather simple approach to accounting for climate and the necessary data for many U.S. cities. While Standard 136 requires the results of a pressurization test, the MHCSS could assume a conservative (low) value for envelope airtightness unless an actual pressurization test is performed. Based on the infiltration rate calculated in this manner, the amount of supplemental ventilation to achieve 0.35 h^{-1} would then be calculated. The problem of low infiltration rates during mild weather and high rates at other times would remain, but it would be an improvement over the current approach.

However infiltration is handled, it is important to also address the operation of the supplemental ventilation system. While the systems studied in this effort and presumably other systems have the capacity to achieve ventilation rates of 0.35 h^{-1} or more, the system must be operated to achieve these ventilation rates. The MHCSS and current practice do not provide sufficient attention to system operation time. This issue could be addressed by specifying that the system operate a sufficient amount of time to increase the average air change rate, or perhaps the effective air change rate, to a specified level. Different means for accomplishing this end could be identified including time clocks and occupancy sensors.

The negative impacts of duct leakage are evident in these simulation results, raising ventilation rates well above the required levels and depressurizing the building interior whenever the forced-air system is on. The higher ventilation rates result in an energy penalty, while the depressurization increases the potential for moisture problems in hot, humid climates and can draw contaminants into the conditioned space from the crawl space volume. Instituting design, construction and perhaps commissioning practices that reduce the level of duct leakage are doable with existing technology.

INTRODUCTION AND BACKGROUND

The HUD manufactured home standards, referred to as the Manufactured Home Construction and Safety Standards (MHCSS), cover the design and construction of all manufactured homes in the United States and contain a number of requirements related to ventilation (HUD 1994). Since these standards have been issued, questions have arisen regarding some of the ventilation–related requirements and their implementation. This report describes a study that addresses some of these questions and provides technical input relevant to their implementation and future revision.

MHCSS Ventilation Requirements

This section describes the portions of the MHCSS containing ventilation requirements, most of which are contained in section 3280.103 *Light and ventilation*. The first requirement states that each home shall be capable of providing a minimum air change rate of 0.35 h^{-1} continuously, or an equivalent hourly rate. This value corresponds to the residential ventilation requirement in ASHRAE Standard 62-1999 (ASHRAE 1999). The HUD standard also states that infiltration and exfiltration shall be considered to provide an air change rate of 0.25 h^{-1}. The standard does not address variation in this infiltration rate as a function of weather. The contribution of open windows to this 0.25 h^{-1} of infiltration is not clear either, however, the terms infiltration and exfiltration generally refer to airflow through unintentional leakage sites and not through intentional openings such as windows, open doors and vents.

The standard states that the remaining 0.10 h^{-1}, or its hourly equivalent, may be provided by mechanical or passive systems or a combination of the two. The additional ventilation capacity is to be calculated based on 0.18 L/s per m^2 of interior floor space (0.035 cfm per ft^2). The standard specifically states that this additional capacity shall be in addition to any openable window area, that is, open windows can not be used to meet this requirement.

The means of providing this additional ventilation shall not create a positive pressure in climate Zones 2 and 3, or a negative pressure in Zone 1. These three climate zones are used in the MHCSS to define heat loss and other requirements. Zone 1 corresponds to hot and humid climates in the southeastern portion of the U.S., Zone 2 includes the middle of the country, and Zone 3 includes the northern U.S. These pressure requirements are intended to minimize the potential for excessive moisture accumulation in the building envelope. In Zone 1, under air conditioner operation, a negative indoor pressure could increase the potential for moisture accumulation if moist incoming air condenses on cold interior surfaces. A positive indoor pressure in a predominantly heating climate (Zones 2 and 3) could lead to condensation and moisture accumulation on colder outer surfaces of the building envelope. The standard also states that mechanical systems shall be balanced, meaning that the outdoor air intake into the building be balanced by an equivalent amount of exhaust. Passive systems and combined passive and mechanical systems shall have adequate inlets and exhausts to alleviate any unbalanced pressure.

Other requirements specify that the ventilation system shall exchange air directly with the outdoors, but shall not draw from or expel air into the space underneath the home or the floor, or the wall or ceiling/roof systems. The system may be integral with the space conditioning system, but in this case must be capable of operating independently of space conditioning requirements. The system, or a portion of the system, shall be provided with a manual control and may be provided with automatic timers and humidistats. Finally, "substantiation of the ventilation capacity" to provide 0.1 h^{-1} shall be provided; however, the standard does not describe how to meet this requirement.

In a section called *Additional ventilation*, there is a requirement that at least half of the minimum required glazed area (8 % of gross floor area) shall be openable directly to outdoors. If a mechanical ventilation system is provided, then this openable glazed area of 4 % of the floor area need not be provided. Instead, the mechanical system must be capable of providing an air change in the room(s) every 30 min. There are also requirements for local exhaust, including a system capable of exhausting 47 L/s (100 cfm) from the kitchen and 24 L/s (50 cfm) from each bathroom and toilet compartment. A toilet compartment may be ventilated through the provision of 0.14 m^2 (1.5 ft^2) of openable glazed area in place of mechanical ventilation, except in Zone 3 (northern climates).

Other requirements in the standard relate to indoor air quality, but not to the ventilation issues addressed in this report. These include material in section 3280.308 *Formaldehyde emission controls for certain wood products* and section 3280.504 *Condensation control and installation of vapor retarders.*

Section 3280.708 requires that all clothes dryers be exhausted to the outside. Section 3280.710 *Venting, ventilation and combustion air*, which covers the venting of heating equipment, also has a subsection titled, *Ventilation options to improve indoor air quality*. This subsection states that one of the following options shall be made available to improve indoor air quality: a passive ventilation system, a mechanical ventilation system, a combination passive/mechanical system, or a fresh-air inlet which shall be continuously connected from a forced-air furnace to the exterior. This forced air inlet shall be capable of providing at least 12 L/s (25 cfm) with the furnace fan in normal operation.

Section 3280.715 on *Circulating air systems* requires, among other things, that there be provisions to permit the return of air from all rooms and living spaces, except toilet rooms, to the inlet of the furnace. Living areas not served by return air ducts or closed off from the return by doors or separating partitions shall be provided with permanent uncloseable openings in the doors or partitions. The openings may be grilled or louvered, and the net free area shall not be less than 14 cm^2 per m^2 (1 in^2 for every 5 ft^2) of living area closed off from the furnace by the door or partition. Undercutting doors may be used, but they must be undercut a minimum of 5 cm (2 in.) and not more than 6.4 cm (2.5 in.).

Research Related to MHCSS Requirements

The ventilation requirements in the MHCSS were based in part on measurements and analyses discussed in TenWolde and Burch (1996). This report begins with the residential ventilation requirement of 0.35 h^{-1} in ASHRAE Standard 62-1999. While there have not been many measurements of ventilation rates in manufactured homes, TenWolde and Burch (1996) note that existing infiltration rate measurements are often less than 0.35 h^{-1}, even during cold weather and even in old homes not built to energy efficient standards. Therefore, the report concludes that supplemental ventilation is needed to provide the target ventilation rate of 0.35 h^{-1}. In discussing the amount of supplemental ventilation that is needed, two reports are cited that describe ventilation rate measurements in newly constructed, energy-efficient manufactured homes in the Pacific Northwest (Hadley and Bailey 1990, Palmiter et al. 1992). These documents report average infiltration rates of 0.23 h^{-1} and 0.26 h^{-1} respectively, and provide the basis for the MHCSS assumption that infiltration provides 0.25 h^{-1}.

Based on the assumed infiltration rate of 0.25 h^{-1}, supplemental ventilation is required to provide an additional 0.1 h^{-1} in order to achieve the target ventilation rate of 0.35 h^{-1}. However, providing mechanical ventilation at a volumetric airflow rate equivalent to 0.1 h^{-1} will not result in a total ventilation rate of 0.35 h^{-1} based on the complexities of adding mechanical ventilation rates to infiltration rates. Therefore, the question exists of how much mechanical ventilation is needed to achieve 0.35 h^{-1}. A number of approaches have been proposed for combining mechanical ventilation rates and infiltration rates to calculate total ventilation rates, two of which are cited by TenWolde and Burch. For both of these calculation approaches, a mechanical rate of 0.2 h^{-1} in combination with an infiltration rate of 0.25 h^{-1} results in a total ventilation rate of 0.35 h^{-1}. This mechanical rate of 0.2 h^{-1} converts into a capacity of 0.18 L/s per m^2 (0.035 cfm per ft^2) based on an assumed ratio of actual airflow to rated airflow of 0.632 (Palmiter et al. 1992), a ceiling height of 2.3 m (7.5 ft) and a multiplicative floor area adjustment factor of 1.14. This last factor is based on an assumption that 14 % of the floor area does not need to be ventilated (e.g. closets, utility rooms, etc.).

Issues Related to MHCSS Ventilation Requirements

The objective of this project is to use multizone airflow analysis to investigate some of technical issues related to the MHCSS ventilation requirements and their implementation. Based on a review of the limited literature on ventilation in manufactured homes and discussions with individuals in the field, the following issues were identified as relevant to the study:

- Validity of the 0.25 h^{-1} assumption for infiltration
- Impact and effectiveness of an outdoor air inlet to the furnace return
- Impact and effectiveness of whole house exhaust fan with passive inlet vents
- Impact and effectiveness of whole house exhaust fan without passive inlet vents
- Location of whole house exhaust fan in the main living area versus the bathroom

The first issue relates the assumption of 0.25 h^{-1} from infiltration, since it is key to the supplemental ventilation requirement of 0.1 h^{-1}. Given that infiltration rates vary with weather and building airtightness, one might ask if and when this assumption is valid.

The second issue addresses a common approach for meeting the MHCSS ventilation requirement, that is an outdoor air inlet duct on the furnace return. In this approach, a damper in the duct opens when the furnace fan operates, drawing outdoor air into the house. Questions exist concerning this approach in terms of the reliability of the ventilation rates achieved and the fan energy consumed in providing this mechanical ventilation. This approach is most commonly implemented with no additional ventilation systems components, as is the case in the simulations discussed in this report. However, outdoor air inlet ducts can also be used with an exhaust fan to provide balanced ventilation or a relief vent to prevent pressurization of the home.

The third and fourth issues concern the use of a whole house exhaust fan to meet the MHCSS requirements. Questions exist regarding the adequacy of the resultant ventilation rates and the distribution of the ventilation air within the building. The third bullet concerns the use of a whole house exhaust fan with passive inlet vents installed. Based on questions regarding the necessity of these vents given typical levels of building envelope airtightness, the fourth bullet addresses the impact of whole house exhaust without inlet vents.

Finally, while whole house exhaust fans can be used to meet the supplemental ventilation requirement, the MHCSS does not allow bathroom fans to serve as the whole house exhaust. Some have questioned the appropriateness of this prohibition, suggesting that an exhaust fan in the bathroom can be equally effective as a centrally-located fan given sufficient airflow communication between the bathroom and the rest of the house.

Description of Project

The project described in this report used multizone network airflow analysis to investigate the ventilation issues identified in the previous discussion. The airflow model CONTAM (Walton 1997), developed at NIST, was used to perform these simulations in a manufactured home intended to be typical of current construction practice in the U.S.

While no simulation studies of ventilation in manufactured houses have been conducted, a number of studies have employed computer modeling to evaluate the impacts and effectiveness of mechanical ventilation in residential buildings (Blomsterberg 1991; Carlsson and Blomsterberg 1995; Hamlin and Cooper 1991; Hamlin and Cooper 1993; Hekmat et al. 1986; Matson and Feustel 1997; Sibbit and Hamlin 1991; Yuill and Jeanson 1990; Yuill et al. 1991). Some of these studies have predicted only building ventilation rates, while others have also studied indoor contaminant concentrations and energy consumption associated with different system types, climatic conditions, and levels of building envelope airtightness. Some of these studies have employed single-zone modeling approaches, while others have used multizone building models. Of particular relevance to this study are two earlier efforts involving multizone modeling of ventilation and indoor air quality in a number of single-family residential buildings (Emmerich and Persily 1996; Persily 1998). These two studies serve as a basis for much of the work presented in this report, including the details of the house airflow model.

Another key activity related to the work presented here is International Energy Agency Annex 27 (Mansson 1995; Millet et al. 1996). IEA Annex 27 is an international research effort intended to develop methods for evaluating residential ventilation systems in terms of ventilation rates, contaminant levels, acoustics, energy consumption and economics, to validate these methods with experimental data, and to demonstrate the performance of these systems in different climates and building types. Ventilation and contaminant modeling is an important aspect of this effort (Millet et al. 1996), and some of the modeling approaches and assumptions used in Annex 27 were utilized in this effort.

DESCRIPTION OF HOUSE MODEL

The manufactured home modeled in this project is a double-wide unit intended to represent recent construction consistent with the MHCSS. The floor plan is shown in Figure 1, and is based on a floor plan provided to NIST by the National Conference of States on Building Codes and Standards (NCSBCS). While the floor area is relatively small for a double-wide unit, the results of the airflow simulations and the findings of the study would not be dramatically different if the floor area was more typical. Each room or zone is identified in Figure 1 with a short zone code in parentheses, e.g. KLA for the kitchen/living area, which was used in the CONTAM model to identify the zone. The house also has a crawl space and attic, which are vented to the outdoors as described in the section on airflow characteristics. The house has a forced-air heating and cooling system with the supply ductwork located in the crawl space.

The "crawl space" zone represents both the so-called "belly" of the house and the space below the belly. The belly is generally insulated and contains most of the air distribution ductwork, while the space below the belly generally contains the "crossover" duct connecting multiple units. Based on the expected leakiness of the belly to the space below and the fact that the most significant duct leakage is in the crossover duct, the use of a single volume to represent these two spaces is a reasonable approximation. The term crawl space is used throughout the report to describe this space, though it is not strictly a crawl space as in site-built homes.

	Floor Area		Volume	
	m^2	ft^2	m^3	ft^3
Ground Level				
Crawl Space (CrSp)	98.8	1063	59.3	2094
First Floor				
Kitchen/living room (KLA)	40.1	431	101.9	3,598
Master bedroom (BEDM)	14.9	160	37.8	1,334
Second bedroom (BED2)	12.4	133	31.5	1,112
Third bedroom (BED3)	11.8	127	30.0	1,059
Master bathroom (MBTH)	6.9	74	17.5	618
Bathroom one (BTH1)	4.9	52	12.4	437
Utility room (UTIL)	4.5	48	11.4	402
Master closet (MCLO)	3.3	35	8.4	296
Attic				
Attic (Attc)	98.8	1,063	39.5	1,394
Totals				
First Floor Only	98.8	1,063	250.9	8,860
All Levels	296.4	3,190	349	12,324

Table 1 Floor Areas and Volumes of House Zones

Table 1 contains the zone floor areas and volumes. The house has a cathedral ceiling design, with the average ceiling height of 2.54 m (8.33 ft). The crawl space has a height of 0.6 m (1.97 ft), and the attic space has a peaked roof with a height of 0.5 m (1.64 ft) in the center.

The following sections describe the details of the house model as represented in CONTAM. Figures 2 through 4 show the individual levels of the house as they appear on the CONTAM sketchpad. These levels are the ground, which contains the crawl space, the living area, and the attic.

Airflow Characteristics

CONTAM requires information on all the airflow paths included in the airflow model of the building being simulated. The leakage values used in this study are based on Table 3 in Chapter 25, Ventilation and Infiltration, of the ASHRAE Fundamentals Handbook (1997) and on information in Klote and Milke (1992). Unless otherwise noted, all airflow paths or elements are described in terms of their effective leakage area (ELA) at 4 Pa (0.016 in. H_2O) and employ a flow exponent of 0.65. The effective leakage of an opening is the area of an orifice with a discharge coefficient of 1.0 that would result in the same airflow rate as that across the opening at a pressure difference of 4 Pa (0.016 in. H_2O). As mentioned earlier, the airflow model of the manufactured house used in this study is based on residential airflow models employed in two previous studies at NIST (Emmerich and Persily 1996; Persily 1998) and the efforts of IEA Annex 27 (Mansson 1995; Millet et al. 1996).

Exterior Envelope

A number of different exterior wall airflow paths are used in the house model. These paths, expressed as ELA per unit wall area, per unit interface length, and per item, include the following:

Exterior wall: 6.7 x 10^{-2} cm^2 per m^2 of wall area (9.3 x 10^{-4} in^2/ft^2)
Floor-wall interface: 0.53 cm^2 per m of interface (2.53 x 10^{-2} in^2/ft)
Ceiling-wall interface: 0.33 cm^2 per m of interface (1.6 x 10^{-2} in^2/ft)
Corner interface: 0. 33 cm^2 per m of interface (1.6 x 10^{-2} in^2/ft)
Window #1: 2.3 cm^2 each (0.36 in^2)
Window #2: 2.1 cm^2 each (0.33 in^2)
Window #3: 2.7 cm^2 each (0.41 in^2)
Window #4: 0.9 cm^2 each (0.15 in^2)
Exterior Door: 4.0 cm^2 each (0.62 in^2)

The floor-wall and ceiling-wall interfaces are intended to represent the combined effect of all the leaks located low in the wall and high in the wall, and not necessarily specific leaks at these elevations. Air leakage paths are also included in the crawl space ceiling to the zone above, with the ELA for these paths obtained by multiplying the floor area of the zone above by 1.33 cm^2/m^2 (1.86 x 10^{-2} in^2/ft^2). Air leakage paths from the attic to the zones below are based on a value of 0.67 cm^2 per m^2 of attic floor area (9.3 x 10^{-3} in^2/ft^2).

The wind pressure coefficients for the exterior wall openings employ the wind direction correlation given by Equation 27 in Chapter 25, Ventilation and Infiltration, of the ASHRAE Fundamentals Handbook (1997). Separate expressions are used for the long and short walls of the house, with the wind pressure coefficient at a normal wind direction assumed to equal 0.6. In converting weather station wind speeds to site wind speeds, the house is assumed to be located in suburban terrain.

The attic venting to the outdoors is based on the MHCSS requirement of a minimum free area of not less than 1/300 of the attic or roof cavity floor area. This requirement also states that at least 50 % of the free area be located in the upper portion of the attic and at least 40 % in eave, soffit or low gable vents. Based on the attic floor area of 98.8 m^2 (1,063 ft^2), the 1/300 "rule" corresponds to 0.329 m^2 (3.54 ft^2) of vent area. Half of the vent area is located at the eaves and half at the ridge. Specifically, there is one ridge vent with an effective leakage area of 0.165 m^2

(1.78 ft^2), and two vents of 0.082 m^2 (0.88 ft^2) are located on each of the eaves. The wind pressure coefficients for the ridge vents are based on values used in the Annex 27 simulations (Mansson 1995), while the long-wall pressure profile used for the other openings in the exterior walls is used for the eave vents.

The venting of the crawl space is based on a ratio of vent to floor area of 1/150 (Steven Winter Associates 1999). Given the crawl space floor area of 98.8 m^2 (1,063 ft^2), this corresponds to 0.659 m^2 (7.09 ft^2) of vent area. Ten crawl space vents, each with an effective leakage area of 660 cm^2 (102 in^2), are included in the model. Three such vents are located in each of the long walls, and two in each short wall. Each vent is at an elevation of 0.3 m (0.98 ft) above the floor of the crawl space. In addition to the vents, there are three leaks on each exterior wall of the crawl space. These leaks have an effective leakage area of 10 cm^2/m^2 (0.14 in^2/ft^2) and are at heights of 0 m (0 ft), 0.3 m (0.98 ft), and 0.6 m (1.97 ft) above the crawl space floor.

Adding the effective leakage areas of all the airflow paths from the occupied zones of the building to the outdoors and to the unoccupied zones (crawl space and attic) yields a total effective leakage area for the building of 263 cm^2 (40.8 in^2). This value does not include any leakage associated with the kitchen or bath exhaust fans or with forced-air system duct leakage.

Interior Partitions

The airflow paths between the interior zones are also based on the previous NIST modeling studies (Emmerich and Persily 1996; Persily 1998). Interior walls are assumed to have an effective leakage area of 2 cm^2 per m^2 of wall area (2.9 x 10^{-2} in^2/ft^2). Additional leaks are included to account for interior doorframes and for door undercuts of 5.1 cm (2 in.) in the bathrooms and bedrooms. When closed, the bathroom doors have an effective leakage area of 330 cm^2 (51 in^2) to account for the undercuts and frame leakage, while the bedroom doors have undercut and frame leakage of 410 cm^2 (64 in^2). The bathroom doors have effective leakage areas of 1.3 m^2 (14 ft^2) when open during a simulation, while the bedroom and the utility room doors have leakage areas of 1.6 m^2 (17 ft^2).

Ventilation Systems

This section describes the ventilation systems in the model house. These include the forced-air system and the local exhaust systems that serve the kitchen area and bathrooms. In addition, the ventilation approaches used to meet the supplemental ventilation requirement of the MHCSS are also described. They include an outdoor air inlet on the forced-air return and a whole house exhaust fan with and without passive inlet vents.

Forced-Air System

The house has a forced-air heating and cooling system intended to represent typical equipment used in U.S. manufactured homes. While there are regional variations in forced-air systems, a single system layout is used to facilitate comparisons of the various cases analyzed. A schematic of the air distribution system is shown in Figure 5, while the duct layout itself is shown in the CONTAM sketchpad of the crawl space in Figure 2. The air handler is located in the utility closet on the first floor, and the supply ductwork is located in the crawl space. A single central return grille is located in the kitchen/living area.

The duct system modeling capabilities of CONTAM were used to model airflow through the forced-air system. The CONTAM duct model requires the user to input lengths of ductwork, junctions, transitions, and terminals. Each element in the duct model accounts for friction losses, leakage, and dynamic losses as appropriate, but the user must determine the parameters that account for these losses. Dynamic losses at junctions and transitions in the duct system in the house model were determined using values in the ACCA Residential Duct System Manual (ACCA 1995). This manual provides equivalent length of ductwork for a wide range of fittings based on fitting geometry, friction loss rate and air velocity.

After the duct model was created in CONTAM, simulations were performed with the forced-air system operating to assess the airflow rates in the system. "Dampers" were installed at each supply grille in the duct model to provide a means of "balancing" the system. These dampers were CONTAM airflow paths based on an orifice model, in which the orifice area could be adjusted individually. These orifice dampers were adjusted to obtain the supply airflow rates into each room from the actual forced-air system layout of the manufactured house on which the house model is based. The "design" values of the supply airflow rates are as follows:

Bathroom #1	24 L/s (50 cfm)
Utility room	24 L/s (50 cfm)
Master bath	35 L/s (75 cfm)
Kitchen/living area	142 L/s (300 cfm)
Bedroom #2	47 L/s (100 cfm)
Bedroom #3	47 L/s (100 cfm)
Master bedroom	83 L/s (175 cfm)

The total design supply airflow rate for the system is 402 L/s (850 cfm). The forced-air fan is modeled in CONTAM using a fan curve relating airflow rate to pressure difference and achieving the design airflow rate of 402 L/s (850 cfm) at a pressure of 25 Pa (0.1 in. H_2O).

Leakage in air distribution ductwork to and from unconditioned spaces has been shown to have significant impacts on residential infiltration rates (Cummings and Tooley 1989; Lambert and Robison 1989; Modera 1989; Parker 1989; Robison and Lambert 1989). Therefore, the forced-air system model also includes leakage from the supply ductwork into the crawl space. This leakage is modeled as an orifice area of 75 cm^2 (11.6 in^2), with a flow exponent of 0.5 and a discharge coefficient of 1.0. This duct leakage value is based on field measurements in a number of manufactured houses (Alternative Energy Corporation 1996; Baylon et al. 1995; Cummings et al. 1991; Davis et al. 1996). The supply leak was modeled such that it has a pressure difference across it in the range of 15 Pa to 20 Pa (0.06 in. H_2O to 0.08 in. H_2O) when the forced-air fan is on based on measurements in the referenced studies.

In the simulations that consider forced-air fan operation, the fan operating time is based on the outdoor air temperature. For outdoor temperatures between 15.6 °C and 26.7 °C (60 °F and 80 °F), the forced-air system is assumed not to operate. Under heating conditions, that is, outdoor air temperatures less than 15.6 °C (60 °F), the fractional fan on-time ff_h is determined from the hourly average outdoor air temperature based on the following expressions:

$$ff_h = 0.10 + 0.0136 \times \Delta T \qquad (1)$$

where ΔT is the indoor-outdoor air temperature difference in °C, that is, the indoor air temperature of 21 °C (70 °F) minus the outdoor air temperature. Under cooling conditions, the fractional on-time ff_c is expressed as:

$$ff_c = 0.15 + 0.0240 \times \Delta T \qquad (2)$$

In the case of cooling, ΔT is based on an indoor air temperature of 24.4 °C (75.9 °F).

These fractional on-times in essence define the extent of oversizing of the heating and cooling systems, and the use of the same on-time equations in each of the three cities corresponds to different amounts of oversizing in each city. Based on Equation (1), the fractional on-time of the heating system under winter design conditions is about 60 % in Albany, 25 % in Miami and 40 % in Seattle. The fractional on-time of the cooling system at design conditions is about 25 % in all three cities. As noted in the discussion section, the amount of oversizing is important to the simulation results given the impact of forced-air system operation on ventilation rates. This is particularly true for the ventilation approach relying on an outdoor air intake duct on the furnace return. The on-times under design conditions are not unreasonable based on field experience, but additional simulations for difference cases of oversizing are worth considering.

The fractional-fan on times are used in the analysis by first determining the whole house air change rates with CONTAM for each hour, both with the fan on and with the fan off. To determine the air change rate for that hour, the fan-on air change rate is multiplied by the appropriate value of *ff* and added to the fan-off value multiplied by (1-*ff*). While this approach to estimating the impact of forced-air operation involves significant assumptions, it does capture the dependence of on-time on thermal load in a way that is straightforward to implement and allows annual airflow simulations. A more complete treatment would involve detailed thermal analysis in conjunction with multizone airflow analysis.

Local Exhaust Fans

The house has exhaust fans in the kitchen and in each of the two bathrooms. The kitchen fan has a capacity of 47 L/s (100 cfm) at a pressure of 25 Pa (0.1 in. H_2O). It is modeled with the duct model in CONTAM, using a fan curve relating airflow rate to pressure difference and including a short length of ductwork in the outside wall. In some manufactured houses, bath fans are exhausted through a vertical duct that passes through the attic and discharges above the roof. Such a duct can lead to increased infiltration rates due to the stack flow through the duct when the fan is off, but this effect is not captured in these simulations. The two bathroom fans are based on a fan with a capacity of 24 L/s (50 cfm) at a pressure of 25 Pa (0.1 in. H_2O) and are also modeled with a fan curve in the CONTAM duct model. When these exhaust fans are off, the model assumes 18.8 cm^2 (2.91 in^2) of effective leakage area for each fan. There is also a clothes dryer in the utility room, which is assumed to act as a 47 L/s (100 cfm) exhaust fan when operating.

Note that these exhaust fans are assumed to operate at their design airflow rates despite the fact that field studies have shown otherwise. Due to a variety of installation problems and fans designed to provide their rates airflow rates at relatively low pressure drops, such fans can have installed flows well below the intended values. In one study, the average measured exhaust fan flow was 37 % below the fan rating (Palmiter et al. 1992). Such effects are not considered in this study, but should be kept in mind when considering the results.

Each of the local exhaust fans is assumed to operate on a specific schedule. The kitchen exhaust fan operates from 5 p.m. to 6 p.m. daily. The bathroom exhaust fan in the master bath run from 6 a.m. to 7 a.m. on weekdays and 8 a.m. to 9 a.m. on weekends. The other bathroom exhaust fan operates from 7 a.m. to 8 a.m. on weekdays and 9 a.m. to 10 a.m. on weekends. The dryer operates from 7 p.m. to 8 p.m. on Tuesday and Thursday, and 9 a.m. to 11 a.m. on Sunday.

Supplemental Ventilation Approaches

Two approaches to meeting the supplemental ventilation requirement in the MHCSS were investigated in this study, an outdoor air inlet duct connected to the forced-air return and whole house exhaust fans, with and without passive inlet vents. The forced-air inlet duct is modeled as a 13 cm (5 in.) diameter duct running vertically from the furnace return up through the attic to a terminal on the roof. The duct model of the inlet was created such that when the furnace fan operates, the outdoor airflow rate into the furnace return is 17.5 L/s (37 cfm). This value is based on the MHCSS ventilation requirement of 0.18 L/s per m^2 (0.035 cfm/ft^2) multiplied by the floor area of the house. The inlet is assumed to operate (be open) on two different schedules in the analysis. The first schedule is simply the schedule of forced-air fan operation based on the outdoor air temperature. For the second schedule, the forced-air fan and inlet operate whenever the building is occupied. Occupancy is assumed to occur from midnight to 8 a.m. and 5 p.m. to midnight on weekdays and from midnight to noon and 5 p.m. to midnight on weekends. When the forced-air fan is off, the inlet is assumed to close such that there is no leakage through the intake duct to or from the outdoors. Sometimes these systems are designed with no damper, so the duct communicates with the outdoors all the time. This approach obviously impacts the fan-off air change rates of the building, but these effects are not captured in this study.

Another approach to meeting the supplemental ventilation requirements involves a whole house exhaust fan with and without passive inlet vents incorporated into the window frames. The whole house exhaust fan is sized to provide 17.5 L/s (37 cfm), identical to the forced-air inlet duct. For the simulations with passive inlet vents, one such vent is located in each of the bedrooms and in the kitchen/living area zone, for a total of 4 such vents. Each vent is modeled with an effective leakage area of 13 cm^2 (2 in^2) and is located 0.3 m (1 ft) from the ceiling. As required by the MHCSS, the whole house exhaust fan is located in the main living area, in this case the kitchen/living area. Simulations with the exhaust fan were run with and without the inlet vents. Simulations of the exhaust fan with the inlet vents were also run with the fan located in bathroom #1. In all of these simulations, the whole house fan is run under the two schedules. Under the first schedule, the fan runs whenever the bath and kitchen exhaust fans would be operating, and under the other schedule that fan operates whenever the house is occupied.

SIMULATION APPROACH

The house model was used within CONTAM to investigate the ventilation issues discussed earlier. These issues were:

- Validity of the 0.25 h^{-1} assumption for infiltration
- Impact and effectiveness of an outdoor air inlet to the furnace return
- Impact and effectiveness of whole house exhaust fan with passive inlet vents
- Impact and effectiveness of whole house exhaust fan without passive inlet vents
- Location of whole house exhaust fan in the main living area versus the bathroom

This section describes the cases studied, the airflow analysis performed, and the approach used to estimate the energy consumption associated with the different ventilation strategies.

As discussed earlier, the simulations employed the multizone airflow and contaminant dispersal model CONTAM (Walton 1997). CONTAM considers a building as a system of interconnected volumes or zones, each at a uniform temperature. Airflow paths between zones, and between zones and the outdoors, are specified in the building model along with other relevant information such as ventilation system parameters, outdoor weather, and wind pressure coefficients on exterior surfaces. Based on these inputs, CONTAM calculates airflow rates between each zone under steady-state or transient conditions based on a simultaneous mass balance of air in each zone. In addition, given information on contaminant sources and removal mechanisms and on outdoor contaminant concentrations, CONTAM determines contaminant concentrations in the zones based on the calculated airflow rates and contaminant-specific information. However, contaminant analysis was not included in this study.

The simulations in this study included transient annual analyses for three cities and steady-state analyses for specific weather conditions. The annual simulations were performed for Albany NY, Miami FL and Seattle WA using TMY2 weather (Marion and Urban 1995). Table 2 presents a summary of the climate in each city, including mean monthly values of outdoor temperature, wind speed, and relative humidity. The annual simulations employed a 1 h time step, resulting in airflow rates through each airflow path at 1 h intervals for an entire year, which were then used to calculate airflow rates between zones and whole building air change rates.

	Outdoor Temperature		Wind Speed		Relative Humidity	
	°C	°F	m/s	mph	g/kg	%
ALBANY						
January	-5.2	22.7	4.4	9.8	1.9	68
February	-3.8	25.2	4.5	10.1	2.0	69
March	1.4	34.6	4.7	10.5	2.6	61
April	6.3	43.4	5.2	11.6	3.8	64
May	15.9	60.6	4.2	9.4	7.4	65
June	19.2	66.5	3.5	7.9	10.2	74
July	22.0	71.5	2.8	6.3	11.9	73
August	20.7	69.2	3.5	7.8	10.7	71
September	16.2	61.2	3.4	7.6	8.8	75
October	9.5	49.2	3.0	6.7	5.2	67
November	3.4	38.1	4.1	9.2	3.9	77
December	-3.5	25.7	4.3	9.6	2.2	76
Annual average	8.6	47.4	4.0	8.9	5.9	70
MIAMI						
January	20.0	68.0	4.3	9.7	11.3	75
February	20.8	69.4	4.8	10.7	11.1	71
March	21.6	70.8	5.6	12.5	11.3	68
April	24.5	76.1	5.6	12.6	12.2	63
May	25.8	78.4	4.5	10.0	15.8	76
June	27.3	81.1	3.6	8.1	16.3	72
July	28.0	82.3	3.9	8.8	17.9	76
August	27.9	82.2	4.0	9.0	17.3	74
September	26.9	80.4	3.0	6.6	17.3	78
October	25.1	77.1	3.5	7.9	15.3	77
November	23.2	73.8	4.8	10.8	12.5	70
December	20.6	69.1	4.4	9.8	10.6	70
Annual average	24.3	75.8	4.3	9.7	14.1	73
SEATTLE						
January	4.5	40.1	4.1	9.2	4.1	77
February	6.6	44.0	3.5	7.7	4.2	70
March	7.0	44.7	3.5	7.9	4.6	72
April	9.5	49.1	4.9	10.9	5.2	71
May	13.0	55.3	3.9	8.6	6.6	72
June	15.2	59.3	3.7	8.3	7.1	67
July	17.7	63.8	3.8	8.4	8.0	65
August	18.0	64.3	3.7	8.3	8.8	70
September	15.4	59.7	3.8	8.5	7.7	73
October	11.4	52.6	3.7	8.3	6.7	80
November	7.1	44.9	4.9	11.0	5.4	91
December	4.8	40.7	3.3	7.5	4.6	85
Annual average	10.9	51.6	3.9	8.7	6.1	73

Table 2 Monthly Average Weather Conditions

Cases Analyzed

The annual simulations analyzed the following cases to address the issues discussed earlier:

- Case #1: Envelope leakage only; no fans or ducts in model.
- Case #2: Bath and kitchen exhaust fans on schedules.
- Case #3: Forced-air fan operation based on outdoor temperature.
- Case #4: Outdoor air intake on forced-air return.
  - A: Forced-air operation based on outdoor temperature.
  - B: Forced-air operation during occupancy.
- Case #5: Whole house exhaust with passive inlet vents.
  - A: Whole house exhaust fan in KLA zone, operated on (Case #2) exhaust fan schedules.
  - B: Whole house exhaust fan in KLA zone, during occupancy.
- Case #6: Whole house exhaust without passive inlet vents.
  - A: Whole house exhaust fan in KLA zone, operated on (Case #2) exhaust fan schedules.
  - B: Whole house exhaust fan in KLA zone, during occupancy.

Case #1 was analyzed to assess building ventilation rates due to envelope leakage alone. The house model contains no fans or ducts at all, just air leakage paths in the exterior envelope and between the building zones. Case #2 includes the effects of exhaust fan operation, in which these fans operate on the previously-described schedules of one or two hours per day. In this case, the house model also includes the air distribution ductwork and supply duct leak, though the forced-air fan is not on. Case #3 includes the same scheduled exhaust fans, plus forced-air fan operation based on outdoor temperature. This case reflects the increased building air change rate due to supply duct leakage into the crawl space and represents a baseline case with no supplemental ventilation. Cases #4, #5 and #6 are all different approaches to the supplemental ventilation requirement in the MHCSS. In Case #4, an outdoor air intake duct is connected to the furnace return to simulate a common approach to meeting the supplemental ventilation requirement in the MHCSS. Two different schedules for furnace fan operation are employed in Case #4, one based on outdoor air temperature as in Case #3 and one in which the fan also operates continuously whenever the building is occupied. In Case #5 the ventilation system consists a whole house exhaust fan with passive inlet vents incorporated in the bedroom and kitchen/living area windows. This case also includes two schedules of exhaust fan operation, one in which the whole house exhaust fan operates whenever a local exhaust fan would run under the Case #2 schedules and the other in which the fan operates whenever the building was occupied. Case #6 is the same as #5, but without the passive inlet vents.

Airtightness and Ventilation

The airflow simulations focused on building ventilation rates and how they compare with the requirements in the MHCSS and ASHRAE Standard 62-1999. However, additional analyses were performed to better understand the airflow characteristics of the simulated house. These additional aspects include effective air change rates, pressurization tests to determine the airtightness of the building envelope, analyses of airflow patterns between the major volumes of the house, and determinations of the age of air to characterize outdoor air distribution.

Pressurization Tests of Envelope Airtightness

Pressurization tests of the house were simulated with CONTAM, mimicking the measurement of envelope airtightness in real buildings using ASTM test methods E779 (ASTM 1987) or E1827 (ASTM 1996). In these tests, a fan is installed in a doorway of a house and is used to pressurize (or depressurize) the interior of the building relative to the outdoors. The airflow rates required to maintain the house at a series of indoor-outdoor pressure differences are then measured and used to determine a measure of the building airtightness. Pressurization tests were simulated at indoor-outdoor pressures of 4 Pa (0.016 in. H_2O) and 50 Pa (0.20 in. H_2O). This is done in CONTAM by setting the pressure in the KLA zone at the reference pressure and running a steady-state airflow simulation with no indoor-outdoor temperature difference and zero wind speed. After the simulation, all of the airflow rates from the building to the outdoors, crawl space and attic are added. The total airflow rate at 4 Pa (0.016 in. H_2O) is converted to an effective leakage area, and the total airflow rate at 50 Pa (0.20 in. H_2O) is divided by the volume to determine the air change rate in units of h^{-1}.

Air Change Rates

Building air change or ventilation rates are calculated under fixed weather conditions to compare different cases of fan operation and on an hourly basis over an entire year for the three cities discussed previously. In both cases, the air change rates are based on the total airflow rate into the conditioned space of the building and air distribution ductwork from the outdoors as well as from the unconditioned spaces, that is, the attic and crawl space. This total airflow rate is then divided by the volume of the conditioned space to yield the air change rate in units of air changes per hour or h^{-1}. The mean hourly ventilation rate and the percentages of hours below 0.25 h^{-1} (the assumed infiltration rate in the MHCSS) and below 0.35 h^{-1} (the "target" ventilation rate in the MHCSS based on ASHRAE Standard 62) are also calculated for each month and annually.

Effective Air Change Rate

In addition to air change rates, effective air change rates are also determined for the annual simulations as an indicator of the indoor air quality performance of the different ventilation approaches. The effective air change rate is a measure of temporal ventilation effectiveness, that is, the effectiveness of ventilation over time as opposed to the spatial effectiveness at ventilating the different rooms of the house. (Spatial effectiveness is addressed by the age of air as discussed below.) The effective air change rate is defined in ASHRAE Standard 136 as "the constant outdoor air change rate that would result in the same average pollutant concentration over the same period of time as actually occurs under varying conditions" (ASHRAE 1993). In other words, assuming that there is a pollutant emitted in the house at a constant rate, consider the average pollutant concentration that would result based on the actual time history of hourly air change rates. The effective air change rate is the constant air change rate that would yield the same average pollutant concentration over the same period of time. Therefore, the effective ventilation rate is a measure of how well a given ventilation approach controls a constant pollutant source. It turns out that the effective air change rate is equal to the inverse of the mean of the inverse air change rate, and it is always less than or equal to the mean air change rate over any time period. While the effective air change rate is useful in comparing different ventilation scenarios in the same building in terms of their impacts on indoor air quality, the air change rate is the relevant quantity for calculations of the energy consumption associated with ventilation.

Airflow Patterns and Building Pressures

As mentioned earlier, airflow simulations were performed under fixed weather conditions to compare different cases of fan operation. The results of these simulations are presented in the form of schematic representations of interzone airflow rates. These interzone airflow rates are determined by summing all the airflow rates between the major volumes of the building including the first floor, attic and crawl space. These sums are determined for several conditions including forced-air fan off and on, outdoor air intake duct open, passive vents open, and various combinations of exhaust fan operation. The net airflow rates through the envelope, into and out of the air handling system, and into and out of the duct leaks are presented in schematic diagrams. As part of this analysis, the indoor-outdoor pressure difference across the exterior walls in each room is also determined. These pressures provide another indication of the impacts of different modes of fan operation.

Air Distribution

The ventilation analysis also includes the determination of the age of air in each of bedrooms and in the kitchen/living area. These age of air values provide an indication of spatial ventilation effectiveness, or the distribution of outdoor ventilation air throughout the building. The age of air in a zone is the length of time since the air in that zone entered the building from outdoors and is a measure of outdoor air distribution (Roulet and Vandaele 1991; Sandberg 1983). In a building with complete mixing between zones, the age of air will be the same in each zone and equal to the inverse of the building air change rate. Otherwise, zones with low ages of air correspond to higher outdoor airflow rates into the zones than zones with higher ages. The age of air in each zone is calculated directly from the airflow matrix used by CONTAM in the airflow calculation. Values of the age of air in the three bedrooms and in the kitchen/living area are determined at a fixed indoor-outdoor air temperature difference of 20 °C (36 °F) and under varying wind conditions. The age of air results are presented as their inverse values since the inverses have the same units as the air changes rates, that is, h^{-1}.

Energy

The energy loads associated with ventilation are estimated in order to compare the energy impacts of the different ventilation approaches examined in the study. These estimates employ an approximate technique based on the hourly air change rates determined from the CONTAM simulations. The sensible heating load for each hour is based on the following expression:

$$\rho C_p I V \Delta T \times 1\ hour \tag{3}$$

where

ρ = air density, 1.2 kg/m^3 (0.075 lb/ft^3)
C_p = specific heat of air, 1000 J/kg-°C (0.239 Btu/lb-°F)
I = house air change rate, h^{-1}
V = house volume, 251 m^3 (8,860 ft^3)
ΔT = indoor-outdoor air temperature difference, °C (°F)

Equation (3) is applied to all hours of the year for which the outdoor air temperature is below 15.6 °C (60 °F). The sensible cooling load for each hour is also based on Equation (3), but is applied to only those hours for which the outdoor air temperature is above 26.7 °C (80 °F). In these calculations, the indoor air temperature is assumed to be 21.0 °C (69.8 °F) under heating and 24.4 °C (75.9 °F) under cooling. The sensible heating and sensible cooling energy calculated with Equation (3) is summed over each hour for each month of the year.

The latent cooling load for each hour is based on the following expression:

$$\rho h_{fg} IV\Delta W \times 1\ hour \tag{4}$$

where

h_{fg} = latent heat of water vapor, 2.34 x 106 J/kg (1010 Btu/lb))
ΔW = indoor-outdoor air humidity ratio difference, kg water vapor/kg dry air (lb/lb)

In these calculations, the indoor relative humidity is assumed to be constant at 50 %, which corresponds to 9.64 g/kg at an air temperature of 24.4 °C (75.9 °F). Equation (4) is applied to all hours of the year for which the outdoor air temperature is above 26.7 °C (80 °F) and the outdoor relative humidity is greater than 9.64 g/kg.

The energy consumed by the forced-air and exhaust fans is computed by multiplying the hours that each fan operates, based on the assumed operating schedules, by the power consumed by each fan. The fans are assumed to consume energy at the following rates: forced-air fan 350 W; kitchen exhaust fan 60 W; bathroom exhaust fans 40 W; and, whole house exhaust fan 30 W. In these energy calculations the impact of the forced-fan on heating and cooling loads is not considered. Under heating, the energy consumed by the forced-air fan and released as heat will contribute to meeting the heating load, and during cooling this energy will increase the cooling load. Neither effect is considered in this analysis.

RESULTS

This section presents the results of the simulations, specifically building envelope airtightness, airflow patterns, ventilation rates, effective air change rates, air distribution, and the energy consumption associated with infiltration and ventilation.

Airtightness

Simulated pressurization tests were performed on the house for the following four cases: envelope leakage only (no fans or ducts in the house model); envelope leakage plus the supply duct leakage into the crawl space; envelope and duct leakage, plus the leakage associated with the exhaust fans when not operating; and, envelope, duct and exhaust fan leakage, plus the passive inlet vents. The results are presented in Table 3 as the effective leakage area (ELA) at 4 Pa (0.016 in. H_2O) in units of cm^2 (in^2), the airflow rate at 50 Pa in L/s (cfm), and the air change rate at 50 Pa (0.2 in. H_2O) in units of h^{-1}.

Test Case	**ELA at 4 Pa (0.016 in. H_2O)**		**Airflow rate at 50 Pa (0.2 in. H_2O)**		**Air change rate at 50 Pa (0.2 in H_2O)**
	cm^2	in^2	L/s	cfm	h^{-1}
Envelope leakage only	266	41.2	354	750	5.1
Envelope and duct leakage	339	52.5	421	892	6.0
Envelope, ducts and exhaust fan leakage	373	57.8	452	958	6.5
Envelope, ducts, exhaust fan and passive inlet vents	425	65.9	520	1102	7.5

Table 3 Simulated Pressurization Test Results

As mentioned earlier, the sum of the effective leakage areas of all the exterior envelope openings included in the house model is 263 cm^2 (40.8 in^2). As noted in the table above, the effective leakage area for envelope leakage only is 266 cm^2 (41.2 in^2) for the simulated pressurization test. This agreement between the input values and those obtained from a simulated pressurization test is expected and serves as a check on the accuracy of the house model as represented within CONTAM. Adding the supply duct leakage into the crawl space increases the effective leakage area by 73 cm^2 (11.3 in^2) or 27 % relative to the envelope-only leakage. Adding the leakage associated with the kitchen and bathroom exhaust fans increases the effective leakage area by another 34 cm^2 (5.3 in^2) or 10 % relative to the envelope-plus-duct leakage case. Including the passive inlet vents results in an effective leakage area of 425 cm^2 (65.9 in^2), an increase of 52 cm^2 (8.1 in^2) or 14 % over the leakage without the passive vents. This increase is identical to the expected impact of the four passive vents based on 13 cm^2 (2 in^2) per vent.

Several sets of pressurization data have been reported for manufactured homes that are comparable to the airtightness for the case of envelope, duct and exhaust fan leakage in the third line of Table 3, that is, 6.5 h^{-1} at 50 Pa (0.2 in. H_2O). In a large study of residential airtightness and ventilation, Davis et al. (1996) presents airtightness values for four groups of manufactured homes in the Pacific Northwest. Two groups are of conventional construction, one group of 21 homes built between 1965 and 1980 and another group of 29 homes built in the late 1980s. The mean air change rate at 50 Pa (0.2 in. H_2O) for the older homes is 14.3 h^{-1}, and the mean for the more recently-constructed group is 8.8 h^{-1}. Two groups of energy-efficient manufactured homes are also included in the study. One is a group of 131 homes built in 1988 and 1989, with a mean airtightness value of 6.1 h^{-1} at 50 Pa (0.2 in. H_2O). The second group consists of 157 homes built in 1992 and 1993, with a mean airtightness of 5.5 h^{-1} at 50 Pa (0.2 in. H_2O). Chandra et al. (1998) report values of about 5.5 h^{-1} in two recently-constructed energy efficiency homes, and a range from 5.5 h^{-1} to 7.5 h^{-1} in six conventional homes. Cummings et al. (1991) report airtightness values for a collection of 21 manufactured homes of various ages, from 5 years to 17 years old at the time of the study. The average air change rate at 50 Pa is 12.6 h^{-1}. The Alternative Energy Corporation (1996) reports airtightness values for 8 homes in North Carolina and 6 homes in New York State, all constructed in 1994 and 1995. The average value of the North Carolina homes is 10.2 h^{-1}, and the average for the New York homes is 12.0 h^{-1}. The objective in creating the house model used in this study is to achieve a level of airtightness typical of current construction built to the MHCSS, but not as tight as the most energy-efficient houses being built. Based on the published data available, the simulated house is definitely tighter than the pre-1995 houses but not as tight as houses where efforts were made to achieve a high level of energy efficiency.

The airtightness results, expressed as normalized leakage, are 0.27 for envelope leakage only and 0.38 with duct and exhaust fan leakage included. Normalized leakage is a non-dimensional measure of airtightness based on the effective leakage area at 4 Pa (0.016 in. H_2O), normalized by floor area and building height, and is used in ASHRAE Standard 119 on air leakage in single-family residential buildings (ASHRAE 1988). This standard establishes performance requirements for air leakage in the form of classes from A to J, with each class comprising a range of normalized leakage values and Class A being the tightest. The more severe the climate, the lower the class required by the standard. For Albany, the air leakage is required to be Class F or below, which corresponds to a normalized leakage below 0.57. For Miami and Seattle, the air leakage is required to be Class G or less, which corresponds to normalized leakage below 0.80. The results of the simulated pressurization tests correspond to class D for envelope leakage only and Class E with the duct and exhaust fan leakage included. Therefore, the model house complies with the requirements of ASHRAE Standard 119.

Airflow Patterns

In order to understand the air movement patterns in the house, including the impacts of fan operation and interior door position, steady-state airflow simulations were performed and analyzed for several different conditions. All of these steady-state simulations were performed at an indoor-outdoor air temperature difference of 20 °C (36 °F) and a wind speed of zero. When considering the results presented in this section, it is important to note that if there were a nonzero wind speed, the airflow patterns would change and the impact of fan operation and door opening would be more complex. Similarly, if the sign of the indoor-outdoor temperature difference were reversed, the airflow patterns would also change. This discussion of airflow patterns makes reference to whole building air change rates for the different conditions. These rates are summarized in Table 4. These air change rates can be converted to L/s (cfm), by multiplying by 70 (148). Therefore, for example, the air change rate of 0.55 h^{-1} with the forced-air fan on and the exhaust fans off, corresponds to 39 L/s (81 cfm), which for a family of four is 10 L/s (20 cfm) per person. The requirement of 0.35 h^{-1} in ASHRAE Standard 62 corresponds to 25 L/s (52 cfm) per person based on four occupants.

Conditions	Interior door position	Air change rate (h^{-1})
Forced-air fan off (Figure 6)		
All exhaust fans off	Open	0.28
Both bath fans on; kitchen fan off	Open	0.72
Kitchen fan on; bath fans off	Open	0.73
Forced-air fan on (Figure 7)		
All exhaust fans off	Open	0.55
	Closed	0.64
Both bath fans on; kitchen fan off	Open	1.22
	Closed	1.21
Kitchen fan on; bath fans off	Open	1.22
	Closed	1.21
Inlet on forced-air return (Figure 8)		
All exhaust fans off	Open	0.65
	Closed	0.71
Both bath fans on; kitchen fan off	Open	1.25
	Closed	1.25
Kitchen fan on; bath fans off	Open	1.25
	Closed	1.28
Passive inlet vents and whole house exhaust in KLA (Figure 9)		
Whole house exhaust fan on	Open	0.50
	Closed	0.50
Exhaust and forced air fan on	Open	0.79
	Closed	0.92
Exhaust off and forced-air fan on	Open	0.61
	Closed	0.71
Passive inlet vents and whole house exhaust in BATH1 (Figure 10)		
Whole house exhaust fan on	Open	0.50
	Closed	0.51
Exhaust and forced air fan on	Open	0.85
	Closed	0.92
Whole house exhaust in KLA, no passive inlet vents (Figure 11)		
Whole house exhaust fan on	Open	0.44
	Closed	0.44
Exhaust and forced air fan on	Open	0.79
	Closed	0.85

All values correspond to 20 °C (36 °F) temperature difference and zero wind speed.
To convert the air change rates to L/s (cfm), multiply by 70 (148).

Table 4 Air Change Rates for Different House and Fan Configurations

Figure 6 is a schematic of the airflow patterns for three different cases with the forced-air fan off. The flows in this figure are airflow rates in units of sL/s (L/s calculated from the mass flow rates output by CONTAM using a standard air density of 1.2 kg/m^3) between the two zones connected by each arrow, as well as the flows associated with selected ventilation system elements. In the first case, the three exhaust fans are also off. In the second case, both bath exhaust fans are on, and the kitchen exhaust fan is off; in the third case the bath fans are off, and the kitchen fan is on. With all of the fans off, outdoor air tends to enter the conditioned space at lower elevations such as openings to the crawl space and in the exterior walls, and then move up within the building. Air exits the conditioned space from the upper portions of the exterior walls and into the attic. Air also flows from the crawl space into the supply ductwork through the duct leakage site, and into the space through both the supply vents and the return grille.

Referring to Table 4, the air change rate with all the fans off and the interior doors open is 0.28 h^{-1}. This rate is determined by adding all of the flows into the conditioned space from outdoors, the attic (which in this case is zero), and the crawl space (including the flow into the supply duct leak). This net airflow rate inward is divided by the building volume of 251 m^3 (8860 ft^3) to yield air changes per hour. In the second case in Figure 6, both bathroom exhaust fans are on and the kitchen exhaust is off. Given the bathroom exhaust airflow rate of 49.8 sL/s (105.6 scfm), all of the airflows out of the building via envelope leakage are reduced and the airflows into the building are increased. The building air change rate with the both bath fans on is 0.72 h^{-1}. The increased air change rate relative to the previous case corresponds to 30.7 sL/s (65.0 scfm), which is 62 % of the exhaust fan airflow rate. All of the exhaust airflow does not convert directly into an increase in air change rate because some of the exhaust airflow simply "replaces" exfiltration that exists without the exhaust operating. In the third case in Figure 6, the kitchen exhaust fan is on and the bathroom exhaust fans are off. Since the exhaust airflow rate of the kitchen fan is the same as the two bathroom fans combined, the airflow pattern is almost identical to the previous case, though some small differences exist due to the locations of the different exhaust fans.

Figure 7 shows six cases with the forced-air fan on, that is, three states of fan operation with the interior doors open and closed. The closed interior doors include all of the bedroom and bathroom doors. In the first state of fan operation, none of the local exhaust fans are operating. In the second case, both bath exhausts are on and the kitchen exhaust is off. And in the third case the bath fans are off, and the kitchen fan is on. With the forced-air fan on, the conditioned space becomes depressurized due to the supply duct leakage into the crawl space. Air flows into the conditioned space from the outdoors, attic and crawl space, and no airflows exist in the opposite direction. Turning on either the bathroom exhaust fans or the kitchen fan further increases this depressurization, which in turn increases the airflows into the building from outdoors, the attic and the crawl space. With the interior doors open, the air change rate with the forced-air fan on and the exhaust fans off is 0.55 h^{-1}, almost twice the rate with the forced-air fan off. Turning on either the bath fans or the kitchen exhaust fan increases the air change rate to 1.22 h^{-1}, again with the interior doors open. This increase in air change corresponds to 46.7 sL/s (99.0 scfm), which is 95 % of the exhaust airflow rate. The exhaust airflow rate converts almost directly to an increase in air change in this situation because there is no exfiltration to be "replaced" by the exhaust airflow; the building is already depressurized at all sites on the building envelope. With the interior doors closed, there is little change in the airflow pattern in the building. The only significant difference occurs for the situation with all of the exhaust fans off, where closing the interior doors leads to airflow into the conditioned space from the attic (as well as airflow into the attic). With the interior doors open, the pressure difference across the attic floor is very small and uniform throughout the building. Closing the interior doors leads to an increase in this pressure in rooms with supply vents, specifically the bedrooms, and a decrease in the KLA zone where the forced-air return is located. In either of the exhaust fan situations with the interior doors closed, the airflow pattern is essentially the same as with the doors open. The increased depressurization caused by the exhaust fan, in combination with the significant door undercuts, overrides any effect of door closure seen with the exhaust fans off.

Figure 8 shows the airflow patterns with an outdoor air inlet on the forced-air return and the forced-air fan on. The six cases again correspond to no exhaust fans operating, the two bathroom exhausts on, and the kitchen exhaust on, again with the interior doors open and closed. With the exhaust fans off and the interior doors open, the existence of the outdoor air intake partially balances the depressurization resulting from the supply duct leakage to the crawl space. Relative to the same fan conditions in Figure 7, there is less airflow into the conditioned space with the intake duct operating and a nonzero airflow into the attic. The airflow rate through the intake duct is 17.5 sL/s (37.1 scfm) of outdoor air, based on the supplemental ventilation requirement in the MHCSS. The air change rate for this first case is 0.65 h^{-1}, which compares with a value of 0.55 h^{-1} without the intake damper. The increase of 0.1 h^{-1} is equal to the increase intended to occur from the MHCSS requirements and corresponds to 7.0 sL/s (14.8 scfm), which is 40 % of the airflow rate through the intake duct. With either the bathroom exhaust fans or the kitchen fan on, the depressurization of the conditioned space and the infiltration airflows both increase. The whole house air change rate increases to 1.25 h^{-1} for both cases of exhaust fan operation. This increase of 0.6 h^{-1} corresponds to 41.7 L/s (88.4 cfm), 84 % of the exhaust airflow rate. With the exhaust fans off, closing the interior doors change the airflow patterns somewhat, with additional infiltration into the KLA zone and exfiltration from the rooms with supply vents and closed doors. The air change rate increases by only 0.06 h^{-1}, which corresponds to 4.2 sL/s (8.9 scfm). For both exhaust fan cases, the airflow rates with the doors closed are very close to those with the doors open.

Figure 9 shows the airflow patterns with a whole house exhaust fan in the KLA zone and passive inlet vents in the bedrooms and the KLA zone. The six cases correspond to only the whole house exhaust operating, the whole house exhaust and forced-air fans on, and only the forced-air fan on, again with the interior doors open and closed. For the first case, passive vents open, whole house exhaust on and forced-air fan off, air tends to flow into the conditioned space at low elevations, including the crawl space and the supply duct leak, and leave the building at higher elevations, including the passive "inlet" vents. Due to the strength of the stack effect, even the operation of the whole house exhaust fan does not counteract the airflow out of the passive vents, which are located 0.3 m (1 ft) from the ceiling. This case can be compared with the first case in Figure 6, in which all fans are off. The air change rate increases by 0.22 h^{-1}, which corresponds to 15.3 sL/s (32.4 scfm). Turning on the forced-air fan in the second case in Figure 9 leads to the depressurization of the building interior and the elimination of exfiltration. The airflow through the passive vents reverses, converting them to their intended performance as ventilation air inlets. However, the inlet vents provide only 5.7 sL/s (12.1 scfm). In the third case, the whole house exhaust fan is off, but the forced-air fan is on. The passive vents still act as intakes, but provide only 3.0 sL/s (6.4 scfm). With the interior doors closed, the airflow rates change very little for the case with the whole house exhaust on and the forced-air fan off relative to having the interior doors open. The air change rate does not change at all. With the forced-air fan also on, closing the interior doors does change the airflow patterns. Specifically, rooms with supply vents only are at higher pressures, increasing the exfiltration flow rate. And the KLA zone, with the exhaust fan and the furnace return grille, is at a lower pressure with the doors closed, increasing the infiltration flows. Relative to the case with the interior doors open, the air change rate increases by 0.13 h^{-1} to 0.92 h^{-1}. With the forced-air fan on, but the whole house exhaust off, the closed interior doors increase the air infiltration rate by 0.1 h^{-1}.

Figure 10 shows the airflow patterns with the same passive vents as in Figure 9, but with the whole house exhaust fan in the BATH1 zone, located off the KLA zone. The four cases correspond to the whole house exhaust operating and to the whole house exhaust and forced-air fans on, with the interior doors open and closed. The airflow pattern with the whole house fan on and forced-air fan off is independent of whether the doors are open or closed. Furthermore, the airflow and air change rates are the same as in Figure 9 with the whole house fan in the KLA zone. With the forced-air fan on as well, the airflow pattern and air change rate are again very close to those in Figure 9. This similarity between the results in Figures 9 and 10 addresses one of the questions motivating this study, whether it is important for whole house exhaust fans to be located in the main living area or whether it is acceptable for a bathroom exhaust fan to fill that function. Based on these results, the airflow patterns in the house do not depend on whether the whole house exhaust fan is located centrally or in a bathroom.

Figure 11 shows the airflow patterns with the whole house exhaust fan in the KLA zone, but without the passive inlet vents. The four cases correspond to the whole house exhaust operating and to the whole house exhaust and forced-air fans on, with the interior doors open and closed. The airflow patterns for both cases of fan operation are again independent of whether the doors are open or closed.

The calculated pressures for these steady-state airflow simulations under fixed weather conditions are also of interest. Table 5 presents the indoor-outdoor air pressure differences in Pa for the main living zones of the house at mid-height on the exterior walls. For the first three cases with the forced-air fan off, the pressure differences are all around 1 Pa (0.004 in. H_2O) and are uniform for all the building zones. Of course, for a nonzero wind speed these pressures would have been larger in absolute value and variable among the zones. Operating the forced-air fan with no exhaust fans on reduces the indoor pressure by about 1 Pa (0.004 in. H_2O) due to the supply duct leak into the crawl space. All of the cases with interior doors open exhibit uniform pressure differences among the rooms, while having the interior doors closed yields variability in magnitude and direction. The combination of forced-air fan operation and exhaust fan operation, leads to indoor pressures about 5 Pa (0.020 in. H_2O) lower than outdoors. Operation of the outdoor air inlet on the forced-air return decreases the magnitude of house depressurization by about 1 Pa, since it acts like a return duct leak to balance the supply leak. The whole house exhaust fans have a small effect on the building pressures given their small capacity relative to the level of envelope airtightness. And as was the case with whole house air change rates, having the whole house exhaust fan in the KLA or BATH1 zone has little impact on these pressure differences.

Figures 6 through 11 and the pressure differences in Table 5 reveal a few key aspects of the airflow patterns in this building. First, an upward airflow pattern dominates under conditions of zero wind speed and a higher indoor air temperature than outdoors. This pattern leads, in general, to most of the air entering the building at lower elevations. Due to the supply duct leak into the crawl space, the operation of the forced-air fan depressurizes the building, increasing infiltration into the envelope. For the operating conditions studied, the house tends to be at a negative pressure relative to outdoors, though the zero wind speed must be noted. The operation of the local exhaust fans further depressurizes the building, by as much as 5 Pa (0.020 in. H_2O) with the interior doors and slightly more in specific rooms with the interior doors closed. In general, whenever the interior doors are open, the pressure difference across the exterior walls is the same from room to room. With the doors closed, variations exist among the rooms depending on which fans are operating. Little difference is seen in airflow patterns or indoor-outdoor pressure differences for the whole house exhaust fan in either the main living area (KLA zone) or in the bathroom off the main living area (BATH1). Obviously these airflow patterns and pressures will differ for nonzero wind speeds and will depend on wind direction. And while the impact of wind speed and direction was not considered in this discussion, their influences are reflected in the annual ventilation rate simulations presented in the next section.

Referring again to Table 4, the fans-off rate is indeed consistent with the rate of 0.25 h^{-1} assumed within the MHCSS. However, the rates in Table 4 were determined at a significant indoor-outdoor temperature difference. As seen in Table 2, the mean monthly outdoor temperatures result in temperature differences that are typically well below the value of 20 °C (36 °F) used to determine the values in Table 4. As seen later, the mean air change rate for envelope infiltration alone for each city is below the "All exhaust fans off" value in the first line of Table 4 and below the infiltration rate assumed in the MHCSS, even accounting for wind speed. Operating the exhaust fans raise the air change rate by about 0.5 h^{-1}, while operating the forced-air fan raises the air change rate by 0.2 h^{-1}. Operating either the bath or kitchen fans, with the forced-air fan running, increases the air change rate by about 1.0 h^{-1}. However, these fans are generally on only on a fraction of the day, limiting their impact on the average air change rate. Both of the supplemental ventilation options, inlet on forced-air return and whole house exhaust with or without passive inlet vents, both increase the air change rate sufficiently to meet the overall MHCSS requirement of 0.35 h^{-1}. Again, fan operation time becomes critical to making these mechanical ventilation approaches and presumably other approaches effective in meeting the MHCSS requirements.

Conditions	Indoor-Outdoor Pressure Difference (Pa) KLA	MBED	BED2	BED3	MBATH	BATH1
Forced-air fan off						
All exhausts off	0.7	0.7	0.7	0.7	0.7	0.7
Bath exhausts on	-1.3	-1.3	-1.3	-1.3	-1.3	-1.3
Kitchen exhaust on	-1.2	-1.2	-1.2	-1.2	-1.2	-1.2
Forced-air fan on/Interior doors open						
All exhausts off	-1.3	-1.3	-1.3	-1.3	-1.3	-1.3
Bath exhausts on	-4.8	-4.8	-4.8	-4.8	-4.8	-4.8
Kitchen exhaust on	-4.6	-4.6	-4.6	-4.6	-4.6	-4.6
Forced-air fan on/Interior doors closed						
All exhausts off	-2.6	0.4	-1.6	-1.6	0.9	-1.9
Bath exhausts on	-5.9	-3.1	-4.8	-4.8	-3.0	-5.8
Kitchen exhaust on	-6.2	-2.4	-5.1	-5.0	-1.7	-5.3
Outdoor inlet on forced-air return/Interior doors open						
All exhausts off	-0.6	-0.6	-0.6	-0.6	-0.6	-0.6
Bath exhausts on	-3.0	-3.0	-3.0	-3.0	-3.0	-3.0
Kitchen exhaust on	-2.9	-2.8	-2.9	-2.9	-2.8	-2.9
Outdoor inlet on forced-air return/Interior doors closed						
All exhausts off	-1.5	1.2	-0.7	-0.7	1.6	-0.1
Bath exhausts on	-3.8	-1.4	-2.8	-2.8	-1.3	-3.8
Kitchen exhaust on	-4.3	-0.9	-3.2	-3.2	-0.4	-3.6
Passive inlet vents/Whole house exhaust in KLA zone/Interior doors open						
KLA exhaust on	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2
KLA and forced-air on	-1.9	-1.8	-1.9	-1.9	-1.8	-1.9
KLA off, forced-air on	-1.2	-1.2	-1.2	-1.2	-1.2	-1.2
Passive inlet vents/Whole house exhaust in KLA zone/Interior doors closed						
KLA exhaust on	-0.2	-0.1	-0.2	-0.2	-0.1	-0.2
KLA and forced-air on	-3.3	-0.2	-2.3	-2.3	0.3	-2.7
KLA off, forced-air on	-2.4	0.5	-1.5	-1.5	0.9	-1.8
Passive inlet vents/Whole house exhaust in BATH1 zone/Interior doors open						
BATH1 exhaust on	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2
BATH1 and forced-air on	-1.9	-1.8	-1.9	-1.9	-1.8	-1.9
Passive inlet vents/Whole house exhaust in BATH1 zone/Interior doors closed						
BATH1 exhaust on	-0.1	-0.1	-0.1	-0.1	-0.1	-0.4
BATH1 and forced-air on	-3.3	-0.2	-2.3	-2.3	0.3	-3.1
Whole house exhaust in KLA zone (No passive inlet vents)/Interior doors open						
KLA exhaust on	0.0	0.0	0.0	0.0	0.0	0.0
KLA and forced-air on	-2.2	-2.2	-2.2	-2.2	-2.2	-2.2
Whole house exhaust in KLA zone (No passive inlet vents)/Interior doors closed						
KLA and forced-air on	0.0	0.0	0.0	0.0	0.0	0.0
KLA off, forced-air on	-3.7	-0.5	-2.7	-2.7	0.0	-3.0

All values correspond to 20 °C (36 °F) temperature difference and zero wind speed.
To convert the pressure differences from Pa to in. H_2O, multiply by 250.

Table 5 Indoor-Outdoor Pressures for Different House and Fan Configurations

Ventilation Rates

This section discusses the whole building air change rates that were calculated for the three cities for each hour of the year. These rates are based on the sum of the airflow rates into the living area from the outdoors, the crawl space (including any airflow from the crawl space into the duct system via duct leakage) and the attic, divided by the volume of the living area.

Case #1 Envelope Leakage Only

Airflow simulations were performed for Case #1, all fans off and all interior doors open, to evaluate the ventilation characteristics of the building due to envelope leakage alone. For these simulations, the house model contained no ducts or fans. Figure 12 is a plot of the hourly mean air change rate over one year versus the indoor-outdoor air temperature difference for Albany. The data all lie above an envelope of minimum air change rates, which corresponds to stack-dominated ventilation of the building. The spread in the direction of higher air change rates is due to wind effects. Figure 13 is a plot of the same data restricted to hours for which the wind speed is less than 2 m/s (4.5 mph), and shows the expected dependence of the air change rate on temperature difference. Similar patterns exist for Case #1 for the other two cities, with the only difference being the range of outdoor temperatures. For Miami, the indoor-outdoor temperature difference ranges from about –12 °C (-22 °F) to about 18 °C (32 °F), and for Seattle from -15 °C (-27 °F) to 25 °C (45 °F). The maximum air change rate for Miami is just under 0.4 h^{-1} and below 0.45 h^{-1} for Seattle. Since the same house model is used in the three cities, the dependence of air change rate on temperature difference is identical for the three cities.

	ALBANY			MIAMI			SEATTLE		
Month	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}
January	0.35	2	51	0.10	98	100	0.26	30	99
February	0.34	4	60	0.11	100	100	0.24	60	98
March	0.29	17	89	0.11	100	100	0.24	58	98
April	0.25	42	98	0.10	100	100	0.23	67	98
May	0.16	93	100	0.10	100	100	0.18	91	100
June	0.12	100	100	0.10	100	100	0.15	99	100
July	0.09	100	100	0.12	100	100	0.13	100	100
August	0.11	99	100	0.12	100	100	0.12	100	100
September	0.14	98	100	0.10	100	100	0.16	98	100
October	0.20	74	99	0.09	100	100	0.19	95	100
November	0.27	41	94	0.09	100	100	0.25	49	99
December	0.34	1	64	0.09	100	100	0.26	40	99
Annual	0.22	56	88	0.10	100	100	0.20	74	99

Table 6 Summary of Air Change Rates for Case #1 (Envelope Leakage Only)

Table 6 contains a summary of the air change rates for Case #1 for the three cities. For each city, the table contains the mean air change rate for each month and the percent of hours during which the air change rates for that month are below 0.25 h^{-1} and below 0.35 h^{-1}. These two reference values correspond to the infiltration rate assumed to exist in the MHCSS and the residential ventilation rate requirements contained in ASHRAE Standard 62-1999. However, the MHCSS infiltration value is the relevant reference for this case. Means and these two percentages are also given on an annual basis. For this case, there are many hours during which the air change rates are below the two reference values, particularly during the months with milder temperatures. In the milder climate of Miami, the rates are essentially always below the MHCSS infiltration assumption of 0.25 h^{-1}. While these rates are lower than might be expected or desired, note that the Case #1 simulations do not account for the impact of exhaust or forced-air fan operation or window and door opening.

Case #2 Scheduled Exhaust Fans

Figure 14 is a plot of the Case #2 hourly mean air change rates for Albany, in which the bath and kitchen exhaust fans and the clothes dryer operate on occupancy-based schedules, but the forced-air fan is always off. However, the air distribution ductwork including the supply duct leak is included in the model, which does impact the airflow rates in the house. The data in this plot fall into four groups, with the lower group corresponding to conditions in which all the exhaust fans are off. These air change rates are the same as those calculated in Case #1. The next higher group contains hours for which one of the bath fans operates at 24 L/s (50 cfm). This exhaust airflow raises the air change rate by about 0.2 h^{-1} relative to that with no exhaust fans operating. The next group, with air change rates between about 0.6 h^{-1} and 0.8 h^{-1}, corresponds to hours during which the kitchen exhaust or the clothes dryer is operating at 47 L/s (100 cfm). The highest group of rates, around 1.0 h^{-1}, corresponds to the dryer and one of the bath fans operating at the same time. At these high exhaust airflow rates, the building air change is dominated by exhaust-induced infiltration and the air change rate does not depend on temperature difference. The plots of hourly air change rates versus temperature difference are similar for the other two cities, with the only difference again in the range of temperature differences.

	ALBANY			MIAMI			SEATTLE		
Month	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}
January	0.40	2	43	0.16	84	89	0.31	26	85
February	0.39	3	52	0.17	85	88	0.29	52	84
March	0.34	15	77	0.17	86	88	0.29	50	84
April	0.31	36	84	0.16	86	90	0.28	58	84
May	0.21	81	87	0.15	86	92	0.23	78	86
June	0.18	86	88	0.16	86	93	0.20	85	86
July	0.15	86	91	0.17	86	93	0.18	86	87
August	0.16	85	90	0.17	86	93	0.17	86	88
September	0.20	85	87	0.16	86	93	0.21	84	86
October	0.26	64	85	0.15	86	92	0.24	81	86
November	0.32	36	81	0.15	86	90	0.30	42	84
December	0.38	1	55	0.15	86	90	0.31	35	85
Annual	0.27	48	77	0.16	86	91	0.25	64	85

Table 7 Summary of Air Change Rates for Case #2 (Scheduled Exhaust Fans)

Table 7 contains a summary of the air change rates for Case #2. The scheduled exhaust fan operation raises the mean monthly air change rates, as expected, relative to the envelope leakage only conditions in Table 6. The mean air change rates increase by about 0.05 h^{-1}, and the percentages of hours below the reference air change rates of 0.25 h^{-1} and 0.35 h^{-1} decrease. However, the numbers of hours below these reference rates are still quite high, particularly in Miami and Seattle and during the nonwinter months in Albany. It is worth noting that the annual mean rates for Albany and Seattle are consistent with the assumed infiltration rate of 0.25 h^{-1} in the MHCSS, while the annual mean for Miami is below that assumed rate. However, the hourly rates are less than 0.25 h^{-1} at least half of the year in Albany and Seattle.

Case #3 Exhaust and Forced-Air Fan Operation

Operating the forced-air fan raises the building air change rate due to the supply duct leakage into the crawl space. As seen earlier, this leakage lowers the pressure in the conditioned space, increasing infiltration flows from outdoors and unconditioned spaces. Figure 15 is a plot of the Case #3 hourly air change rates for Albany. In this case the exhaust fans operate on the same schedules as Case #2 and the forced-air fan operates for a fraction of each hour based on outdoor temperature. As in Case #2, the data in this plot lie in four groups, corresponding to different combinations of exhaust fan operation. For temperature differences between -7 °C (-13 °F) and 6 °C (11 °F), no heating or cooling load is assumed to exist and the forced-air fan does not operate. Note that the windows are closed for all of these simulations, even during hours with no heating or cooling. The air change rates in this range are the same as in Case #2. When the temperature difference is outside of this range, the forced-air fan operates for a fraction of time proportional to the temperature difference and the air change rates in the figure increase above those with the forced-air fan off. The dependence of the hourly air change rates on temperature difference is similar for the other two cities, however the impact of forced-air fan operation differs for the three cities based on climate and therefore the amount of time that the forced-air fan operates.

	ALBANY			MIAMI			SEATTLE		
Month	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$
January	0.51	0	6	0.18	76	89	0.41	0	43
February	0.49	0	6	0.18	81	88	0.38	4	66
March	0.44	0	23	0.18	80	88	0.37	10	60
April	0.39	9	46	0.18	79	90	0.36	16	67
May	0.26	57	85	0.19	79	92	0.29	40	82
June	0.21	76	88	0.21	79	93	0.25	60	86
July	0.17	82	91	0.24	63	91	0.21	80	87
August	0.18	81	89	0.24	72	91	0.20	85	86
September	0.23	63	87	0.20	77	93	0.26	58	86
October	0.32	30	71	0.17	84	92	0.31	32	84
November	0.41	3	40	0.16	85	90	0.39	6	51
December	0.49	0	4	0.16	85	90	0.40	0	51
Annual	0.34	34	53	0.19	78	90	0.32	33	71

Table 8 Summary of Air Change Rates for Case #3 (Forced-Air Fan Operating)

Table 8 contains a summary of the air change rates for Case #3. The addition of forced-air fan operation raises the mean monthly air change rates relative to the rates in Case #2. The magnitude of the increase depends on the outdoor air temperatures during the month, with colder or hotter outdoor temperatures leading to more forced-air fan operation and larger increases in the air change rates. For example in Albany, the mean air change rate during January increases by 0.11 h^{-1}, while the means during June, July and August only increase by about 0.02 h^{-1}. In contrast, the most significant increases in Miami occur during the summer, but the increase in the annual mean is not as large as in Albany due to the milder outdoor temperatures overall. The annual means increase and the percent of hours below 0.25 h^{-1} and 0.35 h^{-1} decrease for all cities, but there are still a significant number of hours below the reference rates.

Cases #4A and #4B Outdoor Air Intake on Forced-Air Fan

Airflow simulations were performed with an outdoor air intake connected to the forced-air return duct in Cases #4A and #4B. This intake opens whenever the forced-air fan operates, and as noted earlier the air change rate increases by about 0.1 h^{-1} with the intake open. Therefore, the impact of this ventilation approach is a strong function of operating time. Two cases of forced-air fan operation were investigated. In Case #4A the forced-air fan operates as in Case #3, that is, whenever the outdoor air temperature creates a demand for heating or cooling. In Case #4B, the forced-air fan operates whenever the house is occupied, and as dictated by heating or cooling demands when the house is unoccupied. As noted earlier, the house is assumed to be occupied from midnight to 8 a.m. and 5 p.m. to midnight on weekdays and from midnight to noon and 5 p.m. to midnight on weekends. Figure 16 is a plot of the Case #4A hourly air change rates for Albany. These air change rates are not very different from Case #3 due to the limited time of forced-air fan operation, and therefore intake duct opening. While the outdoor air intake increases the air change rate by 0.1 h^{-1}, it never operates for more than about two-thirds of each hour under the most extreme weather conditions. More typically, the forced-air fan runs for one-quarter to one-half of each hour. As in Case #3, the data are in four groups, corresponding to different combinations of exhaust fan operation. The hourly air change rates' dependence on temperature difference is similar for the other two cities, with the previously-noted differences based on the impact of climate on forced-air fan operation.

Table 9 contains a summary of the air change rates for Case #4A. The addition of an outdoor air intake duct on the forced-air return does increase the air change rates relative to Case #3, but as already noted the increase is not very large based on the limited forced-air on time. The monthly means increase by at most about 0.05 h^{-1} for the months with the highest heating or cooling demand, and do not increase at all for milder months. The annual means increase by only 0.01 h^{-1} to 0.03 h^{-1}. The percent of hours below the two reference air change rates decrease by a moderate amount, with the largest decreases in the colder and warmer months.

Month	ALBANY			MIAMI			SEATTLE		
	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$
January	0.57	0	1	0.18	75	88	0.44	0	18
February	0.55	0	2	0.18	79	88	0.40	3	44
March	0.48	0	7	0.18	78	87	0.39	8	45
April	0.43	8	26	0.19	75	90	0.38	12	53
May	0.27	51	81	0.20	71	92	0.30	35	78
June	0.21	72	87	0.22	70	93	0.25	56	86
July	0.17	80	91	0.26	55	91	0.21	76	87
August	0.19	78	89	0.25	60	91	0.20	82	86
September	0.24	59	84	0.21	70	93	0.27	51	85
October	0.35	27	57	0.18	81	92	0.33	25	80
November	0.45	3	24	0.16	85	90	0.41	5	35
December	0.55	0	0	0.16	83	90	0.42	0	26
Annual	0.37	32	46	0.20	73	90	0.33	30	60

Table 9 Summary of Air Change Rates for Case #4A (Intake on Forced-Air, Exhaust Schedule)

Figure 17 is a plot of the Case #4B hourly air change rates for Albany. With the forced-air fan operating whenever the building is occupied, the impact of the intake duct is much more significant that in Case #4A. The data again split into four groups depending on which exhaust fans are operating. Note that the impact of forced-air operation is only seen for the lower group corresponding to no exhaust fans on. For all the other data, the exhaust fans only operate when the building is occupied and therefore the forced-air fan is on, eliminating the discontinuity seen in the lower group of points.

Table 10 contains a summary of the air change rates for Case #4B. With the forced-air fan operating whenever the house is occupied, the air change rates increase significantly and the percentages of hours below the reference air change rates decrease. Relative to Cases #3 and #4A, the annual means increase by 0.2 h^{-1} to 0.3 h^{-1}. The monthly means are all between about 0.5 h^{-1} and 0.6 h^{-1}, which is well above the reference air change rates. And many of the hours during which the rates are below the reference rates occur when the building is not occupied. During occupancy, the rates are often well above the reference rates, particularly during exhaust fan operation. The energy impacts of this "overventilation," as well as of the forced-air fan operation, are discussed later in this report.

	ALBANY			MIAMI			SEATTLE		
Month	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$
January	0.70	0	0	0.50	29	31	0.61	0	11
February	0.69	0	0	0.50	30	33	0.58	3	20
March	0.65	0	4	0.50	29	33	0.57	5	19
April	0.62	4	12	0.52	23	32	0.58	7	21
May	0.53	24	32	0.51	21	33	0.52	18	31
June	0.50	28	33	0.51	21	33	0.50	27	33
July	0.49	28	32	0.53	10	31	0.49	31	32
August	0.49	30	33	0.52	15	33	0.48	32	33
September	0.51	26	32	0.51	18	33	0.51	23	32
October	0.56	15	26	0.50	28	33	0.54	14	31
November	0.62	2	11	0.49	32	33	0.60	3	15
December	0.69	0	0	0.49	31	32	0.60	0	12
Annual	0.59	13	18	0.51	24	33	0.55	14	24

Table 10 Summary of Air Change Rates for Case #4B (Intake on Forced-Air, Occupancy Schedule)

Cases #5A and #5B Whole House Exhaust Fan with Passive Inlet Vents

In Cases #5A and #5B, the house is ventilated by a whole house fan in combination with passive inlet vents. As noted earlier, one inlet vent is located in each bedroom and in the KLA zone. The whole house exhaust fan is located in the KLA zone and has a capacity of 17.5 L/s (37 cfm). In Case #5A, the whole house exhaust operates on the Case #2 exhaust fan schedule, that is, it is on whenever a bath or kitchen exhaust fan is operating. The intention is to represent a situation of low usage of the whole house fan. In Case #5B, the whole house exhaust fan operates whenever the house is occupied. The four passive inlet vents are always open in these simulations, regardless of any fan operation schedule, and the windows are closed. As noted earlier, operating the whole house exhaust fan with passive inlet vents open increases the building air change rate (for a temperature difference of 20 °C (36 °F) and zero wind speed) to 0.50 h^{-1} relative to 0.28 h^{-1} for the case with no passive vents and all fans off. With the forced-air fan on, the air change rate with the inlet vents and whole house fan is 0.79 h^{-1} relative to 0.55 h^{-1} for the case with no passive vents and only the forced-air fan on. Figure 18 is a plot of the Case #5A hourly air change rates for Albany. The pattern is similar to that seen for Case #3 with the forced-air fan operating based on heating and cooling demand except all the rates are increased based on the existence of the whole house exhaust fan and the inlet vents.

	ALBANY			MIAMI			SEATTLE		
Month	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}
January	0.59	0	1	0.24	70	82	0.48	0	10
February	0.57	0	2	0.25	70	84	0.44	1	33
March	0.52	0	4	0.25	71	83	0.43	5	36
April	0.47	5	19	0.26	69	84	0.43	8	38
May	0.33	35	73	0.25	63	84	0.35	27	71
June	0.26	65	84	0.27	58	85	0.30	45	83
July	0.22	76	85	0.31	46	79	0.27	64	86
August	0.24	72	85	0.30	46	84	0.25	72	86
September	0.29	51	82	0.26	64	85	0.32	40	79
October	0.39	22	52	0.23	74	85	0.38	16	65
November	0.48	2	18	0.23	81	86	0.46	4	23
December	0.57	0	0	0.22	77	86	0.46	0	16
Annual	0.41	28	42	0.26	66	84	0.38	24	52

Table 11 Summary of Air Change Rates for Case #5A (Whole House Exhaust and Passive Inlets, Exhaust Schedule)

Table 11 contains a summary of the air change rates for Case #5A. The addition of the whole house exhaust fan and the inlet vents increases the air change rates. Relative to Case #3, the monthly means increase fairly consistently by 0.05 h^{-1} to 0.10 h^{-1}, and the annual means increase by about 0.06 h^{-1}. The percent of hours below the two reference air change rates decrease as well, but there are still a significant fraction of hours below both reference rates, particularly in Miami and during the milder months in the other two cities. Relative to Case #4A an outdoor air intake on the forced-air return, this case has higher air change rates. The annual means are about 0.05 h^{-1} higher for Case #5A, and consequently the percentages of hours below the reference air change rates are lower.

Figure 19 is a plot of the Case #5B hourly air change rates for Albany. With the exhaust fan operating whenever the building is occupied, the rates increase for many of the hours of the year relative to Case #5A. Otherwise the pattern is similar to that seen in Figure 18.

Table 12 contains a summary of the air change rates for Case #5B. With the whole house exhaust fan operating whenever the house is occupied, the air change rates increase about 0.1 h^{-1} relative to those seen in Table 11 for Case #5A. The percentages of hours below the reference rates decrease below 80% for all but one month, and are still highest in Miami and during the milder months in Albany and Seattle. In comparison to Case #4B with an outdoor air intake on the forced-air return, the whole house exhaust and passive inlet has lower monthly mean air change rates by about 0.1 h^{-1} to 0.2 h^{-1}. Despite the leakier envelope in Case #5B, Case #4B has higher average rates due to the high air change rate with the forced-air intake and the large number of hours during which it operates.

	ALBANY			MIAMI			SEATTLE		
Month	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}
January	0.67	0	0	0.34	38	69	0.57	0	7
February	0.65	0	0	0.34	36	66	0.53	1	15
March	0.60	0	3	0.35	32	68	0.52	4	16
April	0.56	3	8	0.35	29	72	0.52	5	17
May	0.42	23	41	0.33	41	76	0.44	15	37
June	0.35	33	61	0.34	39	75	0.39	25	47
July	0.31	47	75	0.37	27	62	0.36	32	57
August	0.33	41	70	0.37	27	64	0.34	33	62
September	0.38	28	51	0.33	46	78	0.40	23	44
October	0.47	13	28	0.30	56	82	0.46	10	29
November	0.56	1	9	0.32	44	78	0.55	2	10
December	0.65	0	0	0.32	45	75	0.55	0	9
Annual	0.50	16	29	0.34	38	72	0.47	13	29

Table 12 Summary of Air Change Rates for Case #5B (Whole House Exhaust and Passive Inlets, Occupancy Schedule)

Cases #6A and #6B Whole House Exhaust Fan without Passive Inlet Vents

Based on questions as to the need for passive inlet vents given typical levels of envelope leakage, airflow simulations were also performed with a whole house exhaust fan but without the inlet vents. As in Cases #5A and #5B, the whole house exhaust fan is located in the KLA zone and has a capacity of 17.5 L/s (37 cfm). In Case #6A, the whole house exhaust operates on the same schedule as the other local exhausts. In Case #6B, the whole house exhaust fan operates during occupancy. As noted earlier, operating the whole house exhaust fan without passive inlet vents increases the building air change rate (at a 20 °C (36 °F) temperature difference and zero wind speed) to 0.44 h^{-1} relative to 0.50 h^{-1} with the passive inlets. While the air change is lower without the inlets, it is still above the MHCSS requirement of 0.35 h^{-1}.

Table 13 contains a summary of the air change rates for Case #6A. Relative to Case #5A with the same exhaust fan schedule, but with the inlet vents, the monthly means decrease by about 0.03 h^{-1} to 0.06 h^{-1} and the annual means decrease by about 0.05 h^{-1}. The percent of hours below the two reference air change rates are higher, but only by 10% to 20%.

Table 14 contains a summary of the air change rates for Case #6B. With the whole house exhaust fan operating whenever the house is occupied, the air change rates increase about 0.1 h^{-1} relative to those seen in Table 13 for Case #6A. The percentages of hours below the reference rates decrease below 80% for almost all months. Relative to Case #5B with the same exhaust fan schedule plus the passive inlet vents, the removal of the inlet vents decrease the mean air change rates by only about 0.03 h^{-1} to 0.05 h^{-1}. The percentages of hours below the reference air change rates increase by around 5% to 10%. Therefore, for the house model used in these simulations, the use of passive inlet vents in conjunction with a whole house exhaust is not dramatically different from the case of a whole house exhaust without the inlet vents.

Month	ALBANY			MIAMI			SEATTLE		
	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$
January	0.53	0	6	0.21	76	85	0.43	0	44
February	0.51	0	6	0.21	81	86	0.40	4	66
March	0.46	0	23	0.21	80	86	0.39	10	60
April	0.41	9	46	0.21	79	86	0.38	16	67
May	0.28	57	84	0.22	79	86	0.31	40	82
June	0.23	77	86	0.24	80	86	0.27	61	86
July	0.20	83	86	0.27	64	86	0.23	81	86
August	0.21	81	85	0.27	72	86	0.22	85	86
September	0.26	65	86	0.23	77	86	0.28	58	86
October	0.35	31	71	0.20	83	86	0.34	32	85
November	0.43	3	42	0.19	85	86	0.41	6	55
December	0.51	0	4	0.19	85	86	0.42	0	52
Annual	0.36	34	52	0.22	78	86	0.34	33	71

Table 13 Summary of Air Change Rates for Case #6A (Whole House Exhaust, Exhaust Schedule)

Month	ALBANY			MIAMI			SEATTLE		
	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$	Mean air change rate (h^{-1})	Percent of hours $< 0.25\ h^{-1}$	Percent of hours $< 0.35\ h^{-1}$
January	0.61	0	2	0.31	48	75	0.52	0	21
February	0.60	0	3	0.30	47	75	0.49	3	27
March	0.55	0	12	0.31	45	77	0.48	6	26
April	0.51	5	20	0.31	45	81	0.47	8	26
May	0.38	33	49	0.30	58	81	0.40	23	42
June	0.32	42	67	0.32	56	80	0.36	33	50
July	0.29	58	78	0.35	37	72	0.33	41	61
August	0.30	51	75	0.34	41	71	0.31	40	66
September	0.35	37	55	0.31	56	82	0.38	30	47
October	0.44	18	37	0.29	70	84	0.43	16	34
November	0.52	2	19	0.28	58	84	0.50	3	22
December	0.60	0	2	0.29	58	78	0.51	0	23
Annual	0.46	21	35	0.31	52	78	0.43	17	37

Table 14 Summary of Air Change Rates for Case #6B (Whole House Exhaust, Occupancy Schedule)

Summary and Comparison to Measured Ventilation

This section summarizes the annual air change rates predicted for the different ventilation approaches and compares these predictions to measured ventilation rates in manufactured buildings. Table 15 contains the annual mean air change rates for all cases, as well as the percent of hours over the year during which the air change rate is below the reference values of 0.25 h^{-1} (the infiltration assumption in the MHCSS) and 0.35 h^{-1} (based on ASHRAE Standard 62).

	ALBANY			**MIAMI**			**SEATTLE**		
Case/Condition	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}
1/Envelope leakage only	0.22	56	88	0.10	100	100	0.20	74	99
2/Scheduled exhaust fans	0.27	48	77	0.16	86	91	0.25	64	85
3/Forced-air operating on outdoor temperature	0.34	34	53	0.19	78	90	0.32	33	71
4A/Intake on forced-air, operating on outdoor temperature	0.37	32	46	0.20	73	90	0.33	30	60
4B/Intake on forced-air, occupancy schedule	0.59	13	18	0.51	24	33	0.55	14	24
5A/Passive inlets and whole house exhaust on exhaust schedule	0.41	28	42	0.26	66	84	0.38	24	52
5B/Passive inlets and whole house exhaust on occupancy schedule	0.50	16	29	0.34	38	72	0.47	13	29
6A/ Whole house exhaust (no inlets) on exhaust schedule	0.36	34	52	0.22	78	86	0.34	33	71
6B/ Whole house exhaust (no inlets) on occupancy schedule	0.46	21	35	0.31	52	78	0.43	12	37

Table 15 Summary of Annual Air Change Rates

Case #1, corresponding to envelope infiltration only, has mean air change rates below the 0.25 h^{-1} MHCSS assumption for all three cities. The hourly air change rate is below this value for 56 %, 100 % and 74 % of the year in Albany, Miami and Seattle respectively. Operating the local exhaust fans on an occupancy-based schedule (Case #2) increases the mean air change rates, which are consistent with the MHCSS value in Albany and Seattle. But there still are a significant number of hours below 0.25 h^{-1} in all three cities. Case #2 would correspond to a house with no air distribution duct leakage and no pressure effects due to forced-air fan operation. Case #3 can be considered a more relevant baseline case since it includes both local exhaust and forced-air fan operation. The mean air change rate is above 0.25 h^{-1} in Albany and Seattle, but the hourly air change rate is below 0.25 h^{-1} for 34 %, 78 % and 33 % of the year for the three cities. As expected, the two supplemental ventilation approaches have higher mean air change rates. The relevant reference air change rate for cases #4A through #6B is 0.35 h^{-1} based on the MHCSS and ASHRAE Standard 62-1999. The mean air change rates are above this value for all of these cases in Albany and all but two in Seattle, but there are still a significant number of hours during the year below this reference value. Only Case #4B has a mean above 0.35 h^{-1} in Miami. Case #4B does the best job in meeting the required ventilation rates of Standard 62, but the energy impacts of this ventilation strategy are significant as noted below.

	ALBANY			MIAMI			SEATTLE		
Case/Condition	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}
1/Envelope leakage only	0.23	55	87	0.09	100	100	0.20	72	99
2/Scheduled exhaust fans	0.30	43	70	0.18	79	87	0.28	57	79
3/Forced-air operating on outdoor temperature	0.37	29	48	0.20	76	86	0.35	27	64
4A/Intake on forced-air, operating on outdoor temperature	0.40	27	41	0.21	73	86	0.37	24	54
4B/Intake on forced-air, occupancy schedule	0.72	0	0	0.67	0	0	0.69	0	0
5A/Passive inlets and whole house exhaust on exhaust schedule	0.45	24	38	0.28	68	78	0.43	18	46
5B/Passive inlets and whole house exhaust on occupancy schedule	0.58	7	18	0.40	28	61	0.56	2	12
6A/ Whole house exhaust (no inlets) on exhaust schedule	0.41	29	47	0.25	76	79	0.39	27	64
6B/ Whole house exhaust (no inlets) on occupancy schedule	0.54	10	21	0.38	36	68	0.52	4	14

Table 16 Summary of Annual Air Change Rates Based on Occupied Hours

For some of the ventilation strategies, particularly those operated on an occupancy-based schedule, many of the lower ventilation rates occur when the building is unoccupied. While the MHCSS and ASHRAE Standard 62 do not state that the ventilation rate requirements only apply to occupied hours, annual mean air change rates and percentages of hours below the two reference values were calculated for only occupied hours. Table 16 presents these mean values, along with the percent of occupied hours below the reference air change rates. These percentages are based on only the occupied hours of the year, that is 5895 h based on the assumptions in this analysis versus 8760 h for the whole year. In almost all cases, the means increase and the percentages decrease. For cases #1, #2, #3, #4A, #5A and #6A, the mean air change rate increases by only a few hundredths of an air change per hour, if at all. For cases #4B, #5B and #6B, the ventilation system operates whenever the building is occupied, and therefore the increase is larger, around 0.15 h^{-1} for case #4B and between 0.05 h^{-1} and 0.10 h^{-1} for cases #5B and #6B. The percent of hours below the reference air change rates also drop for all cases, with the most significant changes for cases #4B, #5B and #6B. In these cases, the supplemental ventilation operates whenever the building is occupied, and lower percentages are expected. While lower, the percentages are still significant for the most of the cases, with the exception of #4B for which they are all zero. While the air change rate during occupied hours may be relevant for some contaminants, particularly those emitted for short periods of time from occupant activities, the unoccupied rates are still relevant for other contaminants, particularly those emitted on a more continuous basis. Water vapor is a good example of the latter, as humidity control from non-occupant sources requires ventilation during more than just occupied hours.

	ALBANY			MIAMI			SEATTLE		
Case/Condition	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}	Mean air change rate (h^{-1})	Percent of hours < 0.25 h-1	Percent of hours < 0.35 h^{-1}
1/Envelope leakage only	0.27	39	83	0.14	100	100	0.23	67	99
2/Scheduled exhaust fans	0.32	33	71	0.19	87	89	0.27	57	84
3/Forced-air operating on outdoor temperature	0.42	13	38	0.29	64	87	0.36	17	66
4A/Intake on forced-air, operating on outdoor temperature	0.46	10	28	0.31	49	87	0.38	13	52
4B/Intake on forced-air, occupancy schedule	0.64	4	11	0.49	20	46	0.58	4	18
5A/Passive inlets and whole house exhaust on exhaust schedule	0.49	6	25	0.35	30	82	0.43	6	42
5B/Passive inlets and whole house exhaust on occupancy schedule	0.57	2	9	0.41	10	56	0.52	2	11
6A/ Whole house exhaust (no inlets) on exhaust schedule	0.44	13	39	0.31	64	87	0.38	17	66
6B/ Whole house exhaust (no inlets) on occupancy schedule	0.53	5	15	0.37	30	65	0.48	6	23

Table 17 Summary of Annual Air Change Rates During Heating or Cooling

As noted earlier, all of these simulations have been done with the windows closed, and this issue is discussed in the Summary and Discussion section. However, the means and percentages were also determined only for those hours during which heating and cooling is assumed to occur. These hours correspond to outdoor air temperatures below 15.6 °C (60 °F) and above 26.7 °C (80 °F). While this analysis does not address the window issue directly, it does focus on hours during which window opening is much less likely to occur. Table 17 presents these mean values, along with the percent of occupied hours below the two reference air change rates. These percentages are based on only the hours during which heating or cooling is assumed to occur, which comprise 71%, 32% and 78% of the year in Albany, Miami and Seattle respectively. In almost all cases, the means increase and the percentages decrease relative to the values based on the entire year in Table 15. And for all but one of the supplemental ventilation cases, the mean air change rate is greater than or equal to 0.35 h^{-1}. However, the percent of hours below this value, while less than for the whole year, are still significant for many of these cases.

Table 18 contains the annual effective air change rates for all of the cases. As described earlier, the effective air change rate quantifies the impact of a given ventilation approach on indoor air quality. It is defined in ASHRAE Standard 136 as the constant air change rate that would yield the same average pollutant concentration over a year, given a constant pollutant source subject to the actual variation in air change rate over a year. Compared to the values in Table 15, the effective air change rates are below the mean air change rates for all the cases, which is always true for effective air change rates. The effective air change rates for Case #3 are around 0.2 h^{-1} or less, indicating that in terms of indoor air quality, these baseline conditions correspond to a constant air change rate below the 0.25 h^{-1} reference value. The relevant reference air change rate for the supplemental ventilation cases is 0.35 h^{-1}, and the effective air change rates are close to or below 0.35 h^{-1} for all the mechanical ventilation cases in all three cities, with the lowest values in Miami. Therefore, within the limitations of the concept of effective air change rates, all of the ventilation cases could be considered marginal in terms of indoor air quality control. This is due in part to the intermittent nature of the ventilation approaches. Ventilation strategies that maintain more constant ventilation rates will exhibit less difference between effective air change rates and mean air change rates.

Case/Condition	**Effective air change rate (h^{-1})**		
	Albany	**Miami**	**Seattle**
1/Envelope leakage only	0.15	0.09	0.16
2/Scheduled exhaust fans	0.17	0.10	0.18
3/Forced-air operating on outdoor temperature	0.19	0.11	0.21
4A/Intake on forced-air, operating on outdoor temperature	0.19	0.11	0.22
4B/Intake on forced-air, occupancy schedule	0.34	0.27	0.34
5A/Passive inlets and whole house exhaust on exhaust schedule	0.24	0.15	0.27
5B/Passive inlets and whole house exhaust on occupancy schedule	0.34	0.25	0.35
6A/ Whole house exhaust (no inlets) on exhaust schedule	0.19	0.11	0.22
6B/ Whole house exhaust (no inlets) on occupancy schedule	0.28	0.20	0.29

Table 18 Summary of Effective Air Change Rates

There have been limited measurements of air change rates in manufactured homes for comparison to the values predicted in these simulations. The most relevant data are those measured in recently constructed homes, but not homes built to the most demanding energy efficiency standards. The available measured data include a group of 131 homes constructed under an energy efficiency construction program in the Pacific Northwest (Palmiter et al. 1992). These homes are older than may be relevant, but nonetheless the mean air change rate over a two-week period is 0.27 h^{-1}. This value can be compared with the annual mean of 0.32 h^{-1} for Seattle under Case #3 in Table 15, though the mean temperature difference for the measurements was about 15 °C (28 °F) while the annual mean for the simulations in Seattle would be closer to 10 °C (18 °F). For comparison, another study in the Pacific Northwest focused on more recent construction, again under an energy efficiency construction program, in which the houses were tighter than the test house in this study based on the pressurization results. The mean air change rate for these homes with all fans off is 0.12 h^{-1} (Davis et al. 1996). With the forced-air fan on the mean rate is 0.26 h^{-1}, and 0.58 h^{-1} with two bath fans on and the forced-air fan off. The measured values are lower than the corresponding values in Table 4, but the houses were tighter and the temperature differences during the air change rate measurements were about one-half of those in Table 4. A study of homes in North Carolina and New York reported mean air change rates with the forced-air fan off of 0.33 h^{-1} and 0.39 h^{-1} in the two states. Based on pressurization test results, these homes were almost two times as leaky as the house in this simulation effort. While there are no measured data in the literature that correspond to the exact conditions of the simulations, the available data are consistent with the predicted air change rates.

Air Distribution

In order to examine the distribution of outdoor ventilation air within the house, the age of air was calculated in each of the bedrooms and in the KLA zone. As mentioned earlier, the age of air for a room is the mean amount of time that has passed since the air in that room entered the building, and is an indicator of the distribution of ventilation air. Low values of the age of air in a zone, relative to the building mean, indicate that a relatively large proportion of the outdoor ventilation air reaches that zone. Higher values indicate that less outdoor air reaches the zone. While little or no outdoor air may enter a zone directly, the zone can still have a low age of air if ventilation air reaches the zone from another building zone with a high airflow rate from outdoors. Also, if the air within the building is well-mixed by the forced-air distribution system, the ages of air will generally be fairly uniform in the zones served by that system, even if there is no air flowing directly from the outdoors into all the zones.

Table 19 presents a summary of the age of air values for several different configurations of the house and its ventilation systems. Rather than present the age of air, this table contains the inverse of the age or air, which has the same units as air change rate values, that is, h^{-1}. Two values are presented for each condition, one with zero wind speed and the other at 5 m/s (11.2 mph). The nonzero wind speed values are averages of the four values determined at wind directions from the north, south, east and west. All the values correspond to an indoor-outdoor temperature difference of 20 °C (36 °F). For reference, if the house ventilation rate is 0.35 h^{-1} and the outdoor air is distributed in proportion to the volume of each room, or the interior air is perfectly mixed, then the inverse age of air will be 0.35 h^{-1} in every zone. Regardless of the magnitude of the inverse age, similar values among the zones are an indication of good ventilation air distribution.

House conditions	**Wind speed (m/s)**	**Inverse age of air (h^{-1})**			
		KLA	MBED	BED2	BED3
All fans off	0	0.16	0.17	0.18	0.18
	5	0.17	0.19	0.18	0.19
Forced-air fan on; all exhaust fans off; interior doors open	0	0.19	0.19	0.19	0.19
	5	0.53	0.53	0.52	0.52
Forced-air fan on; all exhaust fans off; interior doors closed	0	0.24	0.24	0.24	0.24
	5	0.55	0.55	0.57	0.57
Intake on forced-air return; interior doors open	0	0.32	0.32	0.32	0.32
	5	0.63	0.64	0.62	0.63
Intake on forced-air return; interior doors closed	0	0.38	0.38	0.38	0.38
	5	0.65	0.63	0.65	0.65
Whole house exhaust in KLA zone; passive inlet vents; interior doors open	0	0.40	0.37	0.31	0.35
	5	0.55	0.42	0.39	0.44
Whole house exhaust in KLA zone; passive inlet vents; interior doors closed	0	0.40	0.33	0.31	0.35
	5	0.55	0.41	0.39	0.44
Whole house exhaust in BATH1 zone; passive inlet vents; interior doors open	0	0.40	0.37	0.31	0.35
	5	0.54	0.42	0.39	0.44
Whole house exhaust in BATH1 zone; passive inlet vents; interior doors closed	0	0.39	0.32	0.30	0.30
	5	0.51	0.39	0.39	0.41
Whole house exhaust in KLA zone; no passive inlet vents; interior doors open	0	0.36	0.33	0.28	0.32
	5	0.47	0.37	0.31	0.37
Whole house exhaust in KLA zone; no passive inlet vents; interior doors closed	0	0.36	0.29	0.27	0.30
	5	0.47	0.34	0.30	0.36

All values correspond to 20 °C (36 °F) temperature difference. The values for nonzero wind speed are averages over winds from the north, south, east and west.

Table 19 Summary of Inverse Age of Air Values

The first case in Table 19 corresponds to weather-driven infiltration. The inverse ages are below the reference value of 0.25 h^{-1} for infiltration in the MHCSS, but the values are fairly uniform for the four zones indicating good air distribution. In the next two cases, the forced-air fan is on and all the exhaust fans are off. With the interior doors open, the inverse ages are uniform in the four zones. The values with zero wind speeds are below the infiltration-only value in the MHCSS, and the values at 5 m/s (11.2 mph) are well above the 0.35 h^{-1} reference value in ASHRAE Standard 62. With the interior doors closed, the inverse ages increase somewhat, but the uniformity across the zones is maintained. Operating the forced-air fan is expected to mix the interior air fairly well, since the airflow rate through the air distribution system corresponds to almost 6 air changes per hour. Similarly, in the next two cases with the intake on the forced-air return, the inverse ages are uniform in the four zones for both cases of door position. The values are close to the ASHRAE value at zero wind speed and well above 0.35 h^{-1} at the elevated wind speed. As expected, the forced-air distribution system does a good job of mixing the ventilation air throughout the building. With the whole house exhaust and passive inlet vents providing ventilation air independent of the forced-air system, there is some variation in the inverse age of air among the four rooms. With the whole house exhaust fan in either the KLA or BATH1 zone, the highest inverse ages are seen in the KLA zone, which is expected given its direct connection to both of the whole house fans. The lowest inverse ages are generally seen in BED2. With the interior doors closed, the inverse ages stay the same or decrease relative to their values with the doors open. Also, the values are similar with the whole house exhaust fan in either the KLA zone or the BATH1 zone. For the case of the whole house exhaust without the inlet vents, the level of variation among rooms is about the same as that seen with the vents. Therefore, for these conditions, the lack of inlet vents does not negatively impact ventilation air distribution.

Energy

The energy consumption associated with the ventilation approaches is shown in Table 20. These same data are presented graphically in Figure 20, which also contains the mean annual air change rate. For each ventilation approach, the table and the figure present the energy associated with heating, cooling and fan operation for the three cities. The heating load for each hour, as discussed earlier, is determined from the hourly building air change rates times the heat capacity of air and the indoor-outdoor air temperature difference. These hourly heating loads are summed over the year, but are not converted to primary energy consumption by assuming a value for the heating system efficiency. The cooling load consists of the sensible load, which is determined similarly to the heating load, and the latent cooling load, based on the indoor-outdoor humidity ratio difference instead of the temperature difference. Again, the cooling loads are not converted to primary energy consumption values. The fan energy values in Table 20 are based on the operating schedules of the various fans and the energy consumption associated with each fan. No credit is taken for fan energy in the heating load calculations, nor any penalty under cooling. In addition to the various ventilation approaches analyzed in the simulations, the table also contains a reference case in which the ventilation rate is 0.35 h^{-1} for every hour of the year. The fan energy in this case is assumed to be the same as the third case in which the forced-air fan operation is based on the outdoor temperature.

The energy consumption for the first case includes only heating and cooling loads, but no fan loads, as this is the case of envelope leakage only. This case shows the clear difference between the three climates, with Miami being dominated by cooling and Seattle by heating. Albany has the highest heating loads and a small cooling load. While the cooling load in Miami is indeed significant based on climate, the envelope-based air change rates are 0.1 h^{-1} on average, resulting in the low energy load for this case. The second case includes the impacts of exhaust fan operation, but not the forced-air fan. The heating and cooling loads increase due to the higher air change rates with the exhaust fans operating, and of course the energy consumed by the fans is seen. Exhaust fan operation increases the energy consumption by 18 %, 49 % and 23 % in Albany, Miami and Seattle respectively. The larger percentage increase in Miami is due to the high outdoor humidity levels that compound the impact of the increased the air change rate. The third case includes the operation of the forced-air fan based on the outdoor air temperature. All three categories of energy consumption increase relative to the second case, with the total energy consumption increasing by 51 %, 78 % and 60 % respectively in the three cities. Again, the increase in Miami is larger that the other two cities due to its higher outdoor humidity levels.

CASE/CITY	ANNUAL ENERGY CONSUMPTION							
	Heating		Cooling		Fans		Total	
	MJ	kWh	MJ	kWh	MJ	kWh	MJ	kWh
Envelope leakage only (Case #1)								
Albany	10200	2834	172	48	0	0	10372	2882
Miami	189	53	2012	559	0	0	2201	612
Seattle	6150	1708	14	4	0	0	6164	1712
Scheduled exhaust fans (Case #2)								
Albany	11781	3273	263	73	184	51	12228	3397
Miami	243	68	2843	790	184	51	3270	909
Seattle	7392	2053	26	7	184	51	7602	2111
Forced-air fan operating on outdoor temperature (Case #3)								
Albany	15139	4206	393	109	2927	813	18459	5128
Miami	328	91	4422	1228	1080	300	5830	1619
Seattle	9634	2676	36	10	2521	700	12191	3386
Intake on forced-air, operating on outdoor temperature (Case #4A)								
Albany	16799	4667	414	115	2927	813	20140	5595
Miami	342	95	4686	1302	1080	300	6108	1697
Seattle	10228	2841	38	11	2521	700	12787	3552
Intake on forced-air, occupancy schedule (Case #4B)								
Albany	22334	6204	565	157	8440	2345	31339	8706
Miami	628	174	7273	2020	8054	2237	15955	4431
Seattle	15329	4258	54	15	8275	2299	23658	6572
Passive inlets, whole house exhaust on exhaust schedule (Case #5A)								
Albany	17697	4916	475	132	3045	846	21217	5894
Miami	407	113	5310	1475	1198	333	6915	1921
Seattle	11385	3163	44	12	2639	733	14068	3908
Passive inlets, whole house exhaust on occupancy schedule (Case #5B								
Albany	20573	5715	509	141	3564	990	24646	6846
Miami	511	142	6056	1682	1716	477	8283	2301
Seattle	13786	3830	47	13	3157	877	16990	4720
Whole house exhaust on exhaust schedule (Case #6A)								
Albany	15888	4414	428	119	3045	846	19361	5379
Miami	367	102	4731	1314	1198	333	6296	1749
Seattle	10246	2846	41	11	2639	733	12926	3590
Whole house exhaust on occupancy schedule (Case #6B)								
Albany	18984	5274	465	129	3564	990	23013	6393
Miami	479	133	5569	1547	1716	477	7764	2157
Seattle	12822	3562	44	12	3157	877	16023	4451
Constant air change rate of 0.35 h^{-1}								
Albany	11604	3224	439	122	2927	813	14970	4159
Miami	368	102	5362	1490	1080	300	6810	1892
Seattle	9002	2501	34	9	2521	700	11557	3210

Table 20 Summary of Energy Consumption

The impact of the first mechanical ventilation option, that is, the outdoor air intake duct on the forced-air return depends on the schedule of forced-air fan operation. With the forced-air fan and intake operating based on the outdoor air temperature (Case #4A) the increase in energy consumption is not large, as is the case with the air change rates discussed earlier. The total annual energy use increases by only 9 %, 5 % and 5 % in the three cities relative to Case #3. However, when the forced-air fan and the intake operate whenever the building is occupied (Case #4B), the energy use increases significantly. Relative to the case without the intake, the energy consumption increases by 70 %, 174 % and 94 % respectively in Albany, Miami and Seattle. An important portion of this increase is the fan energy, which roughly triples in Albany and Seattle, and increases by a factor of about seven in Miami.

The second mechanical ventilation approach, the whole house exhaust fan with passive inlet vents, also increases energy consumption with the increase dependent on the exhaust fan operation schedule. With the whole house exhaust fan operating whenever any exhaust fan would otherwise operate (Case #5A), the energy consumption relative to the forced-air only baseline Case #3 increases by roughly 15 % in each of the three cities. With the whole exhaust fan operating whenever the house is occupied (Case #5B), the energy increase is larger. Again relative to Case #3, the increase is 33 %, 42 % and 39 % in Albany, Miami and Seattle respectively. With the whole house exhaust fan alone, without the inlet vents, the energy consumption increase is slightly less than the case with the inlet vents. With the exhaust fan operating on the exhaust fan schedule (Case #6A), the energy consumption increases by 5% to 8% in the three cities, and with the exhaust fan operating during occupancy (Case #6B) the energy increase ranges from 25% to 33%.

The last case in the table is the idealized case of a constant air change rate of 0.35 h^{-1}. Note that the energy consumption of most of the other cases (other than envelope leakage only and exhaust fans with no forced-air operation) are at or well above the constant air change rate case. This "excess" energy consumption exists even though all of the other cases experience air change rates below the ASHRAE Standard 62 reference value of 0.35 h^{-1} during a significant portion of the year.

SUMMARY AND DISCUSSION

The study described in this report was performed to address a number of questions related to the ventilation requirements in the HUD MHCSS. Based on a review of the literature on ventilation in manufactured homes and discussions with individuals in the field, the following issues were identified as relevant to the study:

- Validity of the 0.25 h^{-1} assumption for infiltration
- Impact and effectiveness of an outdoor air inlet to the furnace return
- Impact and effectiveness of whole house exhaust fan with passive inlet vents
- Impact and effectiveness of whole house exhaust fan without passive inlet vents
- Location of whole house exhaust fan in the main living area versus the bathroom

In order to address these issues, airflow analyses were performed in a manufactured home using the multizone airflow and indoor air quality model CONTAM. The house model includes the effects of exterior envelope leakage, interior partitions, forced-air distribution and associated duct leakage, exhaust fan operation and outdoor weather. Transient annual simulations were performed for three cities, Albany, Miami and Seattle, and steady-state analyses were performed for specific conditions of weather and fan operation.

Summary of Simulation Results

The airflow simulations were focused on building ventilation rates relative to the requirements in the MHCSS. Additional simulations and analyses were performed to better understand the airflow characteristics of the simulated house, including pressurization tests to determine the airtightness of the building envelope and analyses of airflow patterns between the major volumes of the house. In addition, effective air change rates are presented as a measure of the indoor air quality impacts of different ventilation approaches, and the age of air to characterize outdoor air distribution to the different zones of the house. The energy consumption associated with the different ventilation scenarios is also discussed.

The simulations did not address contaminant concentrations in the house or occupant exposure to contaminants. While there are a number of indoor air quality issues of interest in manufactured housing, such as moisture and formaldehyde levels, contaminant analysis was beyond the scope of this project.

The results of the simulated pressurization tests reveal that the airtightness of the model house is typical of recent manufactured home construction, as intended. In addition, analysis of the airflow patterns within the house show that an upward airflow pattern dominates under conditions of zero wind speed and a higher indoor air temperature than outdoors. This pattern leads, in general, to most of the air entering the building at lower elevations, including from beneath the belly of the house. Air entry from beneath the house has potential indoor air quality implications, as contaminants such as water vapor, radon and pesticides can be drawn into the occupied space by such airflow.

At an indoor-outdoor air temperature difference of 20 °C (36 °F) and zero wind speed, the house has an air change rate of 0.28 h^{-1}. Operating both bath fans, or the kitchen exhaust fan, raises the air change rate to about 0.7 h^{-1}. Due to the supply duct leak into the crawl space, operating the forced-air fan depressurizes the building, increasing infiltration into the building and yielding an air change rate of 0.55 h^{-1} with no exhaust fans operating. The supplemental ventilation strategies investigated in this study increase the air change rate of the house significantly. With an outdoor air inlet duct on the forced-air return, the air change rate is about 0.7 h^{-1}. A whole house exhaust fan in combination with passive inlet vents yields an air change rate of 0.5 h^{-1} with the forced-air fan off and about 0.8 h^{-1} with the fan on. The same whole house exhaust fan without the inlet vents results in an air change rate of 0.44 h^{-1} with the forced-air fan off and 0.79 h^{-1} with it on. Therefore, the supplemental ventilation systems all have the capacity to meet the 0.35 h^{-1} ventilation requirement. Their actual impact in practice is a function of how often they are operated and how the operating time is determined. Only two operating schedules were examined in this study, which could be considered extreme cases of low and high use. Several other options exist for controlling the operation of these systems, including time clocks, temperature controls, and occupancy sensors. One of the cases studied involves an outdoor air intake on the forced-air return with the forced-air fan and intake operation controlled by the heating and cooling load, a common approach used in actual manufacturers homes. In this approach, the operation of the forced-air system and therefore the ventilation rates achieved are strongly dependent on the sizing of the system relative to the heating and cooling loads of the building. Heating and cooling systems are often oversized in manufactured houses, which can lead to relatively short on-times and lower ventilation rates for supplemental ventilation systems controlled by the thermostat alone.

On an annual basis, the envelope infiltration only case (#1) has mean air change rates below the 0.25 h^{-1} MHCSS assumption for all three cities. The hourly air change rate is below this value for 56 %, 100 % and 74 % of the year in Albany, Miami and Seattle, respectively. Operating the exhaust fans on an occupancy-based schedule (Case #2) increases the annual mean air change rates, which are consistent with the MHCSS value in Albany and Seattle. However, there are still a high percentage of hours below 0.25 h^{-1} is all three cities. Case #3 can be considered a relevant baseline case since it includes both local exhaust and forced-air fan operation. The mean air change rate is above the 0.25 h^{-1} reference in Albany and Seattle, but is below that value in Miami. The hourly air change rate is below 0.25 h^{-1} for 34 %, 78 % and 33 % of the year in the three cities. As expected, two mechanical ventilation approaches have higher mean air change rates; the relevant reference for these cases is 0.35 h^{-1} based on ASHRAE Standard 62-1999. The mean air change rates are above this value for almost all of the supplemental ventilation cases in Albany and Seattle, but there are still a significant number of hours during the year below this reference value. Case #4B has the highest air change rates and the lowest fractions of hours below 0.35 h^{-1} due to the large number of hours during which the supplemental ventilation system operates. Case #5B and #6B also have high air change rates and low percentages, again due to the operating schedule. The means in Miami are all less than or equal to 0.35 h^{-1}, except for Case #4B.

Effective air change rates, as defined in ASHRAE Standard 136, are presented as a measure of the indoor air quality performance of the different ventilation scenarios. The effective air change rates are around 0.2 h^{-1} or less for Case #3 in all three cities, indicating that in terms of indoor air quality these baseline conditions correspond to a constant air change rate below the 0.25 h^{-1} reference value. The relevant reference for the supplemental ventilation approaches is 0.35 h^{-1} based on ASHRAE Standard 62-1999. The effective air change rates are below this value when the supplemental ventilation is operated on the more limited schedules (Cases #4A, #5A and #6A) and closer to 0.35 h^{-1} when they are operated during building occupancy (the B schedules).

The predicted air change rates are compared with the limited measurements of air change rates in manufactured homes. The data for this comparison were measured in recently constructed homes, but not homes built to the most demanding energy efficiency standards. And while there are no measured data in the literature that correspond to the exact conditions of the simulations, the data that are available are consistent with the predicted air change rates.

Age of air values are used to examine the distribution of ventilation air within the building, with results presented in terms of inverse age of air in units of h^{-1}. For the case of weather-driven infiltration, the inverse ages are fairly uniform for the four zones indicating good air distribution. With the forced-air fan on and the exhaust fans off, the inverse ages are also essentially uniform across the four zones with the interior doors open. With the interior doors closed, the inverse ages increase somewhat, but the uniformity across the zones is maintained. Operating the forced-air fan would be expected to mix the interior air fairly well, since the airflow rate through the forced-air distribution system corresponds to almost 6 air changes per hour. Similarly, with the intake on the forced-air return, the inverse ages are uniform across the four zones in both cases of door position. With the passive inlet vents providing ventilation air independent of the forced-air system, there are some nonuniformities in air distribution. The uniformity of air distribution as characterized by the age of air values is essentially independent of whether the whole house exhaust fan is located centrally in the KLA zone or in the BATH1 zone. For the case of whole house exhaust without the inlet vents, the level of variation among rooms is about the same as that seen with the vents.

The energy consumption associated with the ventilation approaches was evaluated based on estimates of the energy associated with heating, cooling and fan operation for the three cities. The total energy consumption for each ventilation case is roughly proportional to the annual mean air change rate for each city. The energy impact of the two supplemental ventilation strategies depends on the operating schedule. With the forced-air fan and intake operating based on the outdoor air temperature the increase in energy consumption is not large, but the air change rates do not increase significantly either. However, when the forced-air fan and the intake operate whenever the building is occupied, the energy use increases significantly. Relative to the case without the intake, the energy consumption increases by 70 %, 174 % and 94 % respectively in Albany, Miami and Seattle. An important portion of this increase is the fan energy, which roughly triples in Albany and Seattle, and increases by a factor of about seven in Miami. The whole house exhaust fan with passive inlet vents also increases energy consumption. With the whole house exhaust fan operating whenever any exhaust fan would otherwise operate, the energy consumption relative to the forced-air only baseline case increases by roughly 15 % in the three cities. With the whole exhaust fan operating whenever the house is occupied, the energy increase is larger. Again relative to the forced-air case, the increase is 33 %, 42 % and 39 % in Albany, Miami and Seattle, respectively. With the whole house exhaust fan alone, without the inlet vents, the energy consumption increase is slightly less than the case with the inlet vents.

While the simulations did not address contaminants, the study does have implications for at least moisture control in manufactured housing. Specifically, the existence of supply duct leakage in this house was shown to cause significant depressurization of the building when the forced-air fan is operating. In a hot, humid climate, this condition could increase the potential for moisture accumulation in the exterior walls and elsewhere in the building as hot, humid air is drawn inward and moisture condenses on cold interior surfaces.

Impact of Open Windows

These simulations were all performed with the windows closed, and it is worth considering how the use of window opening would impact the results. However, it is important to note that the MHCSS states that the supplemental ventilation shall be in addition to any operable window area. This statement is one reason that the simulations were performed with the windows closed. Another reason is that the simulations with windows open would be highly dependent on the window opening schedule employed, and there are no accepted opening schedules available for use in airflow simulations. Also, current airflow modeling techniques do not reliably represent airflow through open windows.

Nonetheless, simulations were performed to get some sense of the impact of open windows. These were done for Case #3, scheduled exhaust fans and forced-air operation based on outdoor temperature. In these simulations, each window was replaced by an airflow path using an orifice model based on one-quarter of the gross window area, multiplied by two-thirds to account for the open area of a screen, and employing an orifice discharge coefficient of 0.6 and a flow exponent of 0.65. This approach clearly is not a physically-sound window model, but rather a simplified attempt to estimate the magnitude of the impact of open windows. In these simulations, all the windows were open whenever the outdoor temperature fell between 15.6 °C and 26.7 °C (60 °F and 80 °F), the range over which neither heating nor cooling was assumed to occur. In essence, an idealized situation was modeled in which all the windows were automatically opened whenever no heating and cooling demand existed.

Figure 21 is a plot of the hourly air change rate versus the indoor-outdoor air temperature difference for Albany with the windows "operated" as described. The simulations predict rates above 20 h^{-1} at these low temperature differences, driven primarily by wind. Most of the air change rates with the windows open are 10 h^{-1} or less. Given the lack of air change rate measurements under these conditions in manufactured or site-built homes, it is not possible to determine if these predictions are realistic or not. For these simulations, the average air change rates for May through September are about 3 h^{-1}, relative to 0.2 h^{-1} to 0.3 h^{-1} for Case #3 with the windows shut. Even at these high rates, the air change rate is below 0.25 h^{-1} for 20 % to 30 % of the hours during these five months, and below 0.35 h^{-1} for 30 % to 50 % of the time. Therefore, based on these assumptions on window usage and airflow characteristics, the air change rates increase dramatically under windy conditions, but there are still a significant fraction of hours during which the rates fall below the reference values even under these idealized conditions.

Study Issues

As mentioned earlier, five issues were the primary motivation for his study. The findings of this study with respect to these issues are summarized below:

Validity of the 0.25 h^{-1} assumption for infiltration

Using a single value for a weather-driven infiltration rate is inherently problematic, given the strong dependence of infiltration on weather. As seen in these simulations, the infiltration rates vary by as much as 5 to 1 based on variations in weather conditions alone. Including the impacts of exhaust fan and forced-air fan operation more than doubles the range of variation. Nonetheless, when considering the predicted infiltration rates on an annual basis, the air change rate is below 0.25 h^{-1} for about one-third of the year in Albany and Seattle and for 70 % of the year in Miami. Note that if there were no duct leakage in the house model, these percentages would be significantly higher. Therefore, the assumption of 0.25 h^{-1} for infiltration in modern manufactured homes may be too high, but more importantly ignores variations due to weather and fan operation..

Impact and effectiveness of an outdoor air inlet to the furnace return

Employing an outdoor air intake duct on the forced-air return duct is certainly effective in raising air change rates and distributing ventilation air throughout the house. However, the overall impact on the building air change rate is a strong function of the operating time of the forced-air system, which in turn depends on the extent of system oversizing and the use of other control strategies such as manual switches and timers. While increased forced-air fan operation provides higher ventilation rates, there is an energy cost associated with the increased fan operation. Some control strategies have been proposed to reduce this energy impact by reducing the fan speed (Lubliner 1997). Also, given the existence of significant duct leakage, this scenario suffers from excessive air change rates particularly when weather-driven infiltration is high.

Impact and effectiveness of whole house exhaust fan with passive inlet vents

The whole house exhaust with these vents provided adequate ventilation in this house and reasonable air distribution, but again the impact is highly dependent on the fan operation schedule. As implemented in the house model, these vents themselves were not particularly effective in ventilating the building. Based on the magnitude of the vent openings relative to the house airtightness, their installation basically corresponds to a 15% leakier envelope rather than a designed air intake system as they could conceivably be used. Such a system would presumably require a tighter envelope than is typically achieved in practice. Furthermore, under the conditions in these simulations, outdoor air did not necessarily enter the building through these vents, and when they did indeed act as inlets, the amount of outdoor air entering the building was not large.

Impact and effectiveness of whole house exhaust fan without passive inlet vents

The simulations with a whole house exhaust fan but without the inlet vents exhibit lower ventilation rates than with the vents as expected. However, the rates are still above the 0.35 h^{-1} requirement in the MHCSS. Again, the overall impact of the whole house exhaust fan depends on the fan operating schedule. Therefore, given the level of envelope airtightness assumed in these simulations, the passive inlet vents do not appear to be essential to the proper functioning of a supplemental ventilation system based on a whole house exhaust fan.

Location of whole house exhaust fan in the main living area versus the bathroom

For the conditions in this house model, the impact of the whole house fan did not depend much on its location. Whether the fan was in the main living area or a bathroom off the main living area did not have a significant impact on air change rates, outdoor air distribution or building pressures.

Recommendations

While the simulations performed in this study have limitations, there are a number of recommendations that can be made relevant to the construction of manufactured houses and to subsequent versions of the MHCSS.

One issue relates to the adequacy of the assumption that these houses have a base infiltration rate of 0.25 h^{-1}. These simulations show that at levels of airtightness consistent with current practice, infiltration rates are often below this value except during colder and windier weather. Also, using a single value ignores the significant variation in infiltration that exists as a function of weather. It may therefore make sense for the MHCSS to consider a more realistic treatment of background infiltration. One potential approach is to use ASHRAE Standard 136 to convert a building airtightness value from a pressurization test to an annual effective air change rate for a given climate. While this approach would have its own limitations, it would be straightforward to use since the standard contains a rather simple approach to accounting for climate and the necessary data for many U.S. cities. While Standard 136 requires the results of a pressurization test, the MHCSS could assume a conservative (low) value for envelope airtightness unless an actual pressurization test is performed. Based on the infiltration rate calculated in this manner, the amount of supplemental ventilation to achieve 0.35 h^{-1} would then be calculated. The problem of low infiltration rates during mild weather and high rates at other times would remain, but it would still be an improvement.

However infiltration is handled, it is important to also address the operation of the supplemental ventilation system. While the systems studied in this effort and presumably other systems have the capacity to achieve ventilation rates of 0.35 h^{-1} or more, the systems must be operated to achieve these ventilation rates. The MHCSS and current practice do not provide sufficient attention to system operation time. This issue could be addressed by specifying that the system operate a sufficient amount of time to increase the average air change rate, or perhaps the effective air change rate, to a specified level. Different means for accomplishing this end could be identified including time clocks and occupancy sensors.

The negative impacts of duct leakage are evident in these simulation results, raising ventilation rates well above the required levels and depressurizing the building interior whenever the forced-air system is on. The higher ventilation rates result in an energy penalty, while the depressurization increases the potential for moisture problems in hot, humid climates and can draw contaminants into the conditioned space from the crawl space volume. Instituting design, construction and commissioning practices that reduce the level of duct leakage are all achievable with existing technology.

Additional Research

In considering the results of this study, it is important to note that only one house in three climates was studied, with specific inputs defining the house, ventilation approaches, and weather. While many of the results are generalizable, it is important that similar analyses be conducted in other buildings and in other climates. Also, other ventilation options that have been proposed to comply with the MHCSS merit consideration. In particular, given the importance of ventilation system operation in determining the impact of supplemental ventilation, different operating strategies and control approaches need to be studied. These could include time clocks, temperature or humidity based controls, and other approaches. Further studies of exhaust-based ventilation are also needed to better understand the impact of envelope airtightness, the use of passive inlet vents, and the potential for operating the exhaust fan continuously

Other research that should be considered includes the simulation of indoor contaminant levels to understand the indoor air quality implications of the various strategies for meeting the MHCSS ventilation requirements. A number of different contaminants and contaminant sources have been discussed within the context of manufactured houses including moisture, combustion products, and formaldehyde and other organic compounds from building materials and furnishings. The house model developed in this project could be used to examine the indoor air quality impacts of various contaminant sources under different ventilation scenarios. In addition, the validation of these airflow simulation results, as well as future contaminant simulations, through field studies are important for extending the usefulness of simulation as a means of investigating ventilation and indoor air quality performance issues in manufactured houses.

As noted earlier, the ability to model the impact of window opening on ventilation was inhibited by the lack of a good model for airflow through windows as a function of pressure and weather conditions. Data on window usage patterns as a function of weather, time of day, occupancy and other factors are also lacking. Research in both areas is needed before reliable simulations of window opening impacts can be performed.

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REFERENCES

ACCA. 1995. Residential duct systems. Manual D. Air Conditioning Contractors of America.

Alternative Energy Corporation. 1996. Air of Importance. A Study of Air Distribution Systems in Manufactured Homes.

ASHRAE. 1999. ANSI/ASHRAE Standard 62-1999, Ventilation for acceptable indoor air quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1997. ASHRAE Handbook Fundamentals, Chapter 25 Ventilation and Infiltration. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1993. ANSI/ASHRAE Standard 136-1993, A method of determining air change rates in detached dwellings. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1988. ANSI/ASHRAE Standard 119-1988, Air leakage performance for detached single-family residential buildings. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASTM. 1987. E779-87, Standard test method for determining air leakage rate by fan pressurization. American Society for Testing and Materials.

ASTM. 1996. E1827-96, Standard test method for determining airtightness of buildings using an orifice blower door. American Society for Testing and Materials.

Baylon, D., B. Davis, and L. Palmiter. 1995. Manufactured Home Acquisition Program. Analysis of Program Impacts. Ecotope, Inc.

Blomsterberg, A. 1991. Ventilation control within exhaust fan ventilated houses. Proceedings of 12th Air Infiltration and Ventilation Centre Conference, Air Movement & Ventilation Control within Buildings, 2: 285-305.

Carlsson, T.; and A. Blomsterberg. 1995. Improvement of mechanical ventilation systems regarding utilization of outdoor air. Proceedings of 16th Air Infiltration and Ventilation Centre Conference, Implementing the Results of Ventilation Research, 2: 315-326.

Chandra, C., D. Beal, and B. McKendry. 1991. Energy efficiency and indoor air quality in manufactured housing. Affordable Comfort 98 Selected Readings.

Cummings, J.B., Jr. J.J. Tooley, and N. Moyer. 1991. Investigation of Air Distribution System Leakage and Its Impact on Central Florida Homes. Florida Solar Energy Center, FSEC-CR-397-91.

Cummings, J.B.; and J.J. Tooley, Jr. 1989. Infiltration and pressure differences induced by forced air systems in Florida residences. ASHRAE Transactions, 95 (2): 551-560.

Davis, B., J. Siegel, L. Palmiter, and D. Baylon. 1996. Field Measurements of Heating System Efficiency in Nine Electrically-Heated Manufactured Homes. Ecotope, Inc.

Emmerich, S.J.; and A.K. Persily. 1996. Multizone Modeling of Three Residential Indoor Air Quality Control Options. National Institute of Standards and Technology, NISTIR 5801.

Hadley, D.L. and S.A. Bailey. 1990. Infiltration/Ventilation Measurements in Manufactured Houses - Residential Construction Demonstration Program. Pacific Northwest Laboratory, PNL-7494.

Hamlin, T.; and K. Cooper. 1991. The potential for residential demand controlled ventilation. Proceedings of 12th AIVC Conference, Air Movement & Ventilation Control within Buildings, 2: 235-243.

Hamlin, T.; and K. Cooper. 1993. CMHC residential indoor air quality - parametric study. Proceedings of 13th AIVC Conference, Ventilation for Energy Efficiency and Optimum Indoor Air Quality, 207-216.

Hekmat, D.; H.E. Feustel; and M.P. Modera. 1986. Impacts of ventilation strategies on energy consumption and indoor air quality in single-family residences. Energy and Buildings, 9 (3): 239-251.

HUD. 1994. Part 3280, Manufactured Home Construction and Safety Standards. U.S. Department of Housing and Urban Development.

Klote, J.H. and J.A. Milke. 1992. Design of Smoke Management Systems. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Lambert, L.A.; and D.H. Robison. 1989. Effects of ducted forced-air heating systems on residential air leakage and heating energy use. ASHRAE Transactions, 95 (2): 534-541.

Lubliner, M.; D.T. Stevens; and B. Davis. 1997. Mechanical ventilation in HUD-code manufactured housing in the Pacific northwest. ASHRAE Transactions, 103 (1): 693-705.

Mansson, L.-G. 1995. Evaluation and Demonstration of Domestic Ventilation. State of the Art. Swedish Council for Building Research, Report A12:1995.

Marion, W.; and K. Urban. 1995. User's Manual for TMY2s, Typical Meteorological Years, Derived from the 1961-1990 National Solar Radiation Data Base. National Renewable Energy Laboratory, NREL/SP-463-7668, E95004064.

Matson, N.E.; and H.E. Feustel. 1997. Residential Ventilation Systems. Lawrence Berkeley National Laboratory, LBL-40859.

Millet, J.-R.; J.G. Villenave; and J. Riberon. 1996. French ventilation system performances in residential buildings. Proceedings of 17th Air Infiltration and Ventilation Centre Conference, Optimum Ventilation and Air Flow Control in Buildings, 1: 167-173.

Modera, M.P. 1989. Residential duct system leakage: magnitude, impacts, and potential for reduction. ASHRAE Transactions, 95 (2): 561-569.

Palmiter, L.S.; I.A. Brown; and T.C. Bond. 1991. Measured infiltration and ventilation in 472 all-electric homes. ASHRAE Transactions, 97 (2): 979-987.

Palmiter, L., T. Bond, I. Brown, and D. Baylon. 1992. Measured Infiltration and Ventilation in Manufactured Homes. Ecotope, Inc.

Parker, D.S. 1989. Evidence of increased levels of space heat consumption and air leakage associated with forced air heating systems in houses in the Pacific northwest. ASHRAE Transactions, 95 (2): 527-533.

Persily, A.K. 1998. A Modeling Study of Ventilation, IAQ and Energy Impacts of Residential Mechanical Ventilation. National Institute of Standards and Technology, NISTIR 6162.

Robison, D.H.; and L.A. Lambert. 1989. Field investigation of residential infiltration and heating duct leakage. ASHRAE Transactions, 95 (2): 542-543.

Roulet, C.-A.; and L. Vandaele. 1991. Air flow patterns within buildings measurement techniques. Air Infiltration and Ventilation Centre, Technical Note AIVC 34.

Sandberg, M. 1983. Ventilation efficiency as a guide to design. ASHRAE Transactions, 89 (2B): 455-477.

Sibbitt, B.E.; and T.L. Hamlin. 1991. Meeting Canadian residential ventilation standard requirements with low-cost systems. ASHRAE Transactions, 97 (2): 969-978.

Steven Winter Associates. 1999. Manufactured home installation training manual.Prepared for U.S. Department of Housing and Urban Development.

TenWolde, A. and D.M. Burch. 1996. Ventilation, Moisture Control, and Indoor Air Quality in Manufactured Houses. Forest Products Laboratory, National Institute of Standards and Technology.

Walton, G.N. 1997. CONTAM96 users manual. National Institute of Standards and Technology, NISTIR 6056.

Yuill, G.K.; and M.R. Jeanson. 1990. An analysis of several strategies for four ventilation systems. Proceedings of 5th International Conference on Indoor Air Quality and Climate, 4: 341-346.

Yuill, G.K.; M.R. Jeanson; and C.P. Wray. 1991. Simulated performance of demand-controlled ventilation systems using carbon dioxide as an occupancy indicator. ASHRAE Transactions, 97 (2): 963-968.

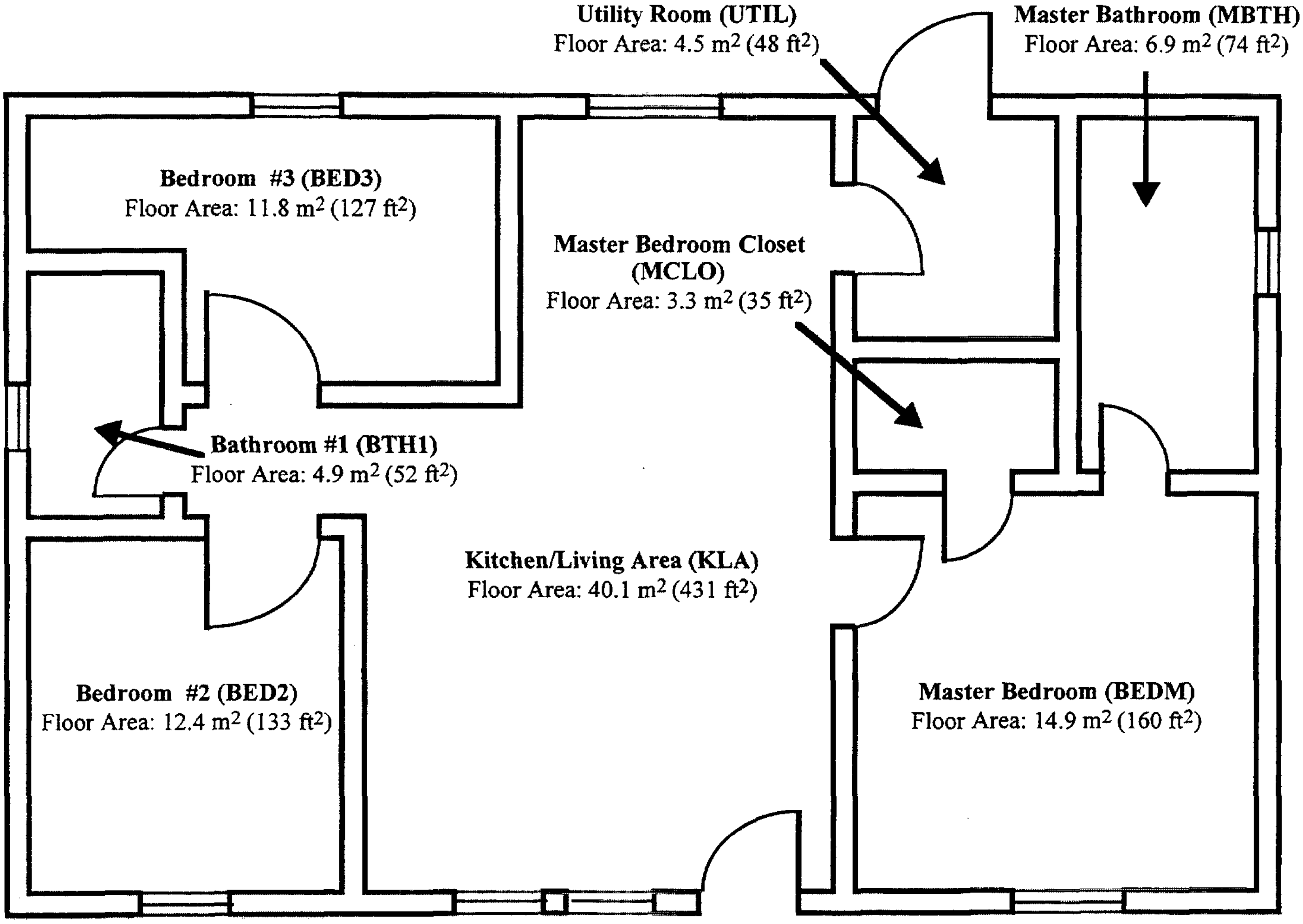


Figure 1 Layout of Living Space

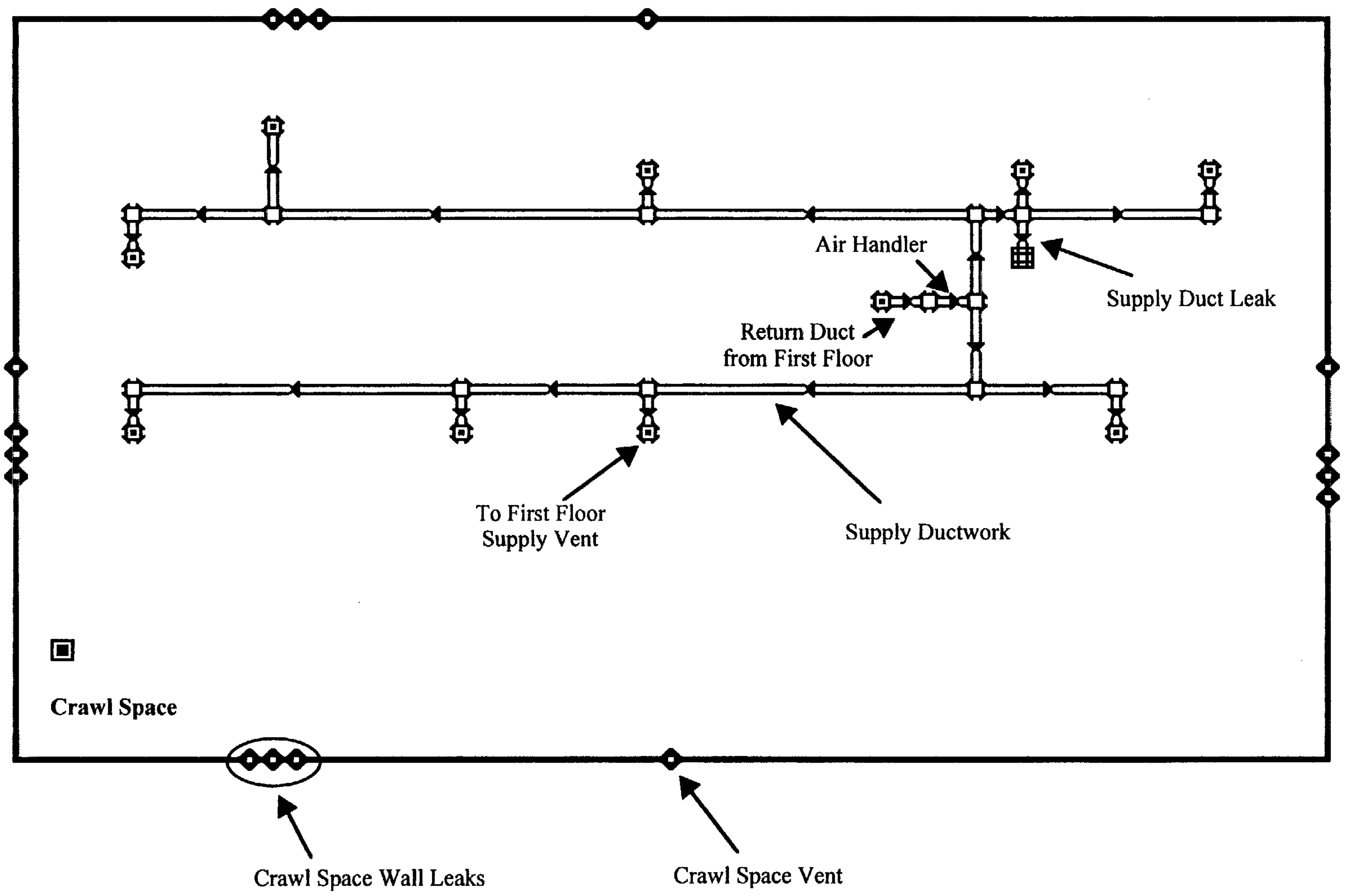


Figure 2 CONTAM Sketchpad of Crawl Space Level

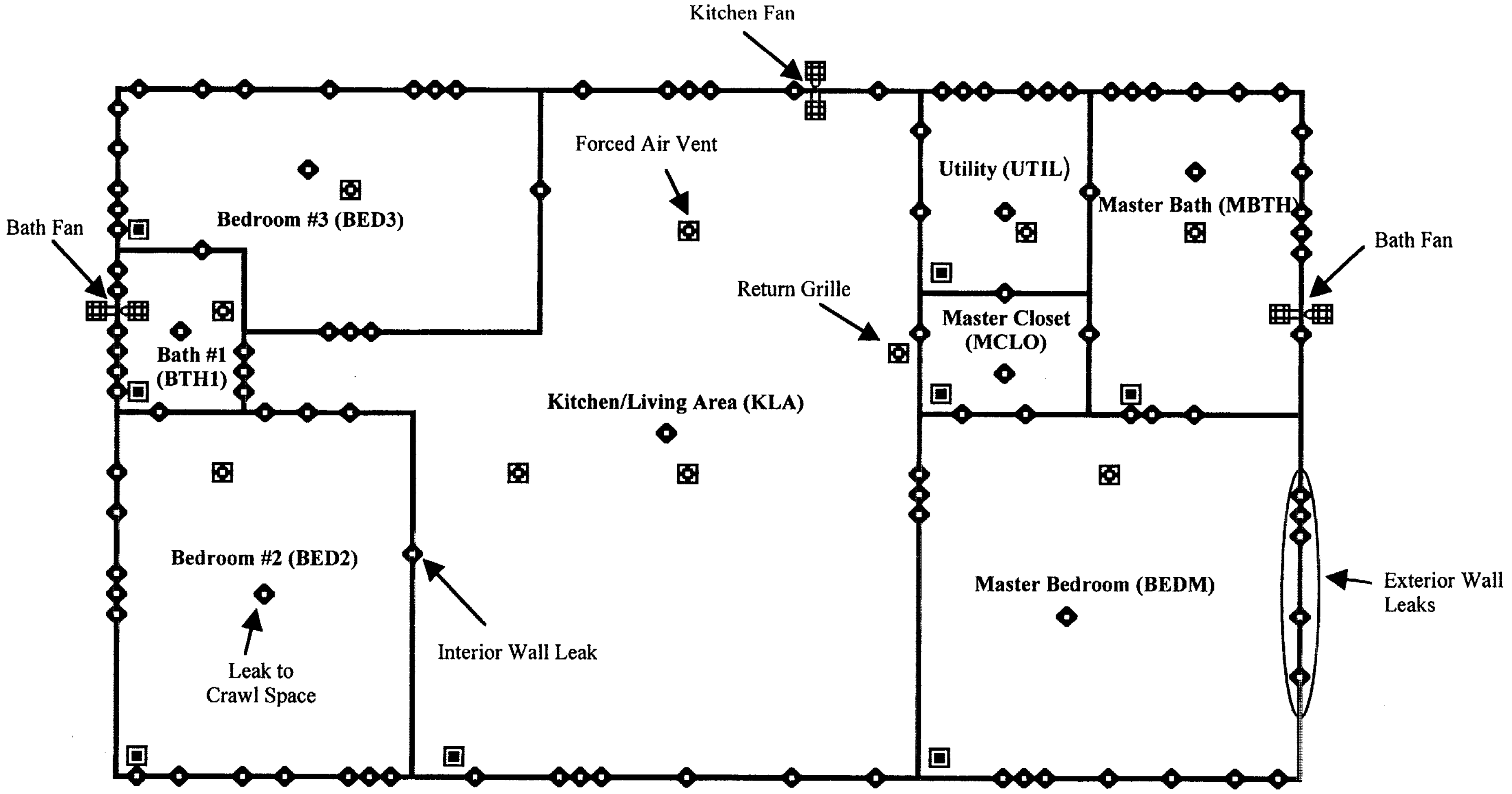


Figure 3 CONTAM Sketchpad of Living Area

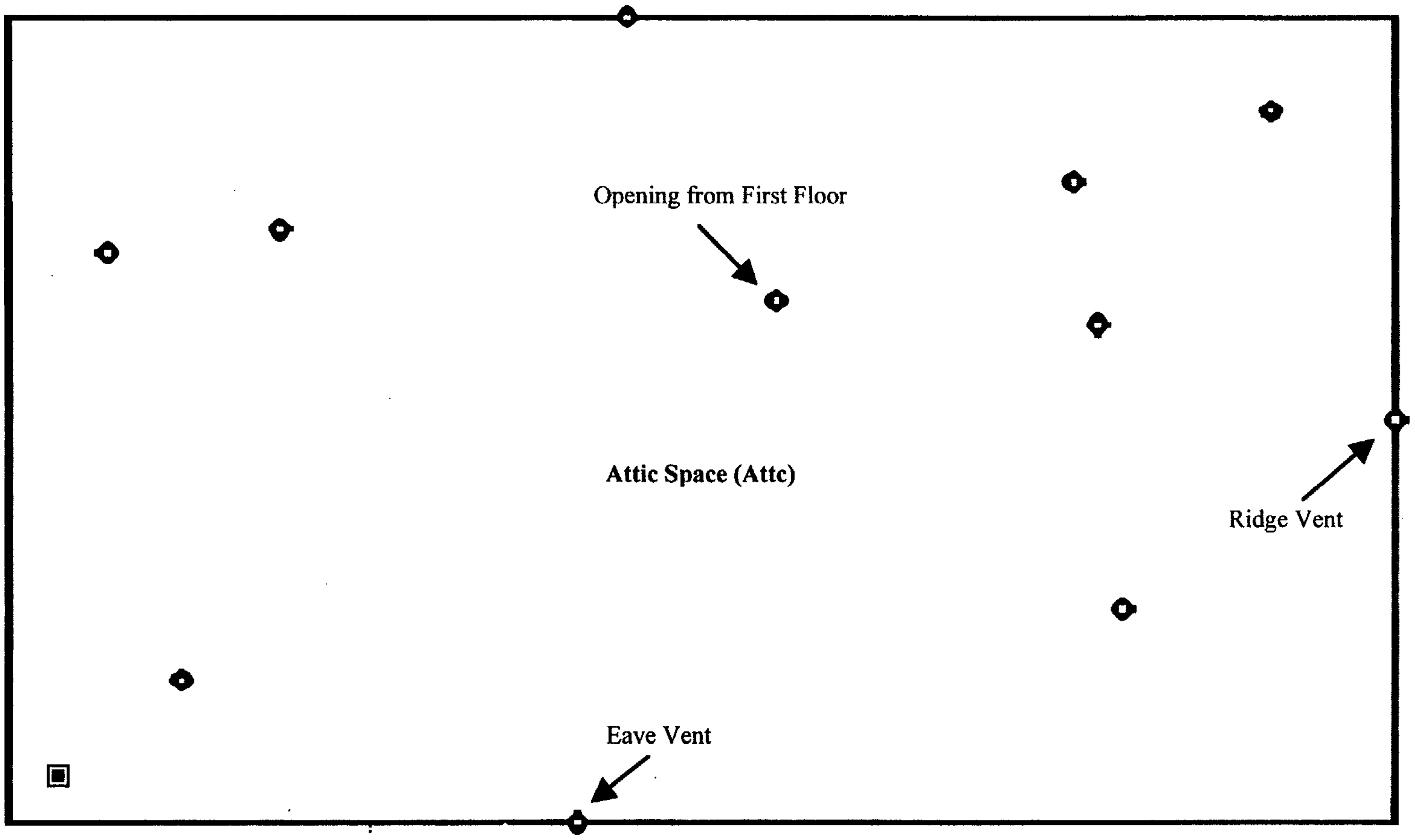


Figure 4 CONTAM Sketchpad of Attic

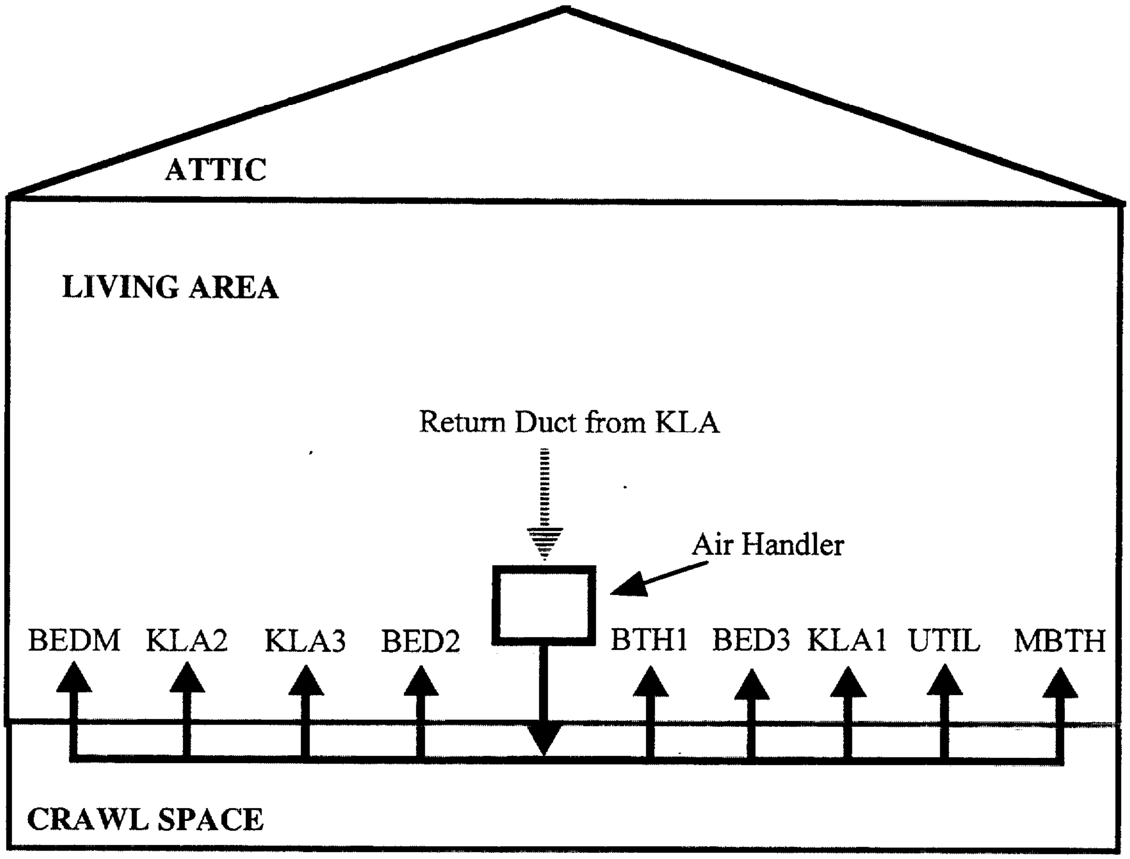
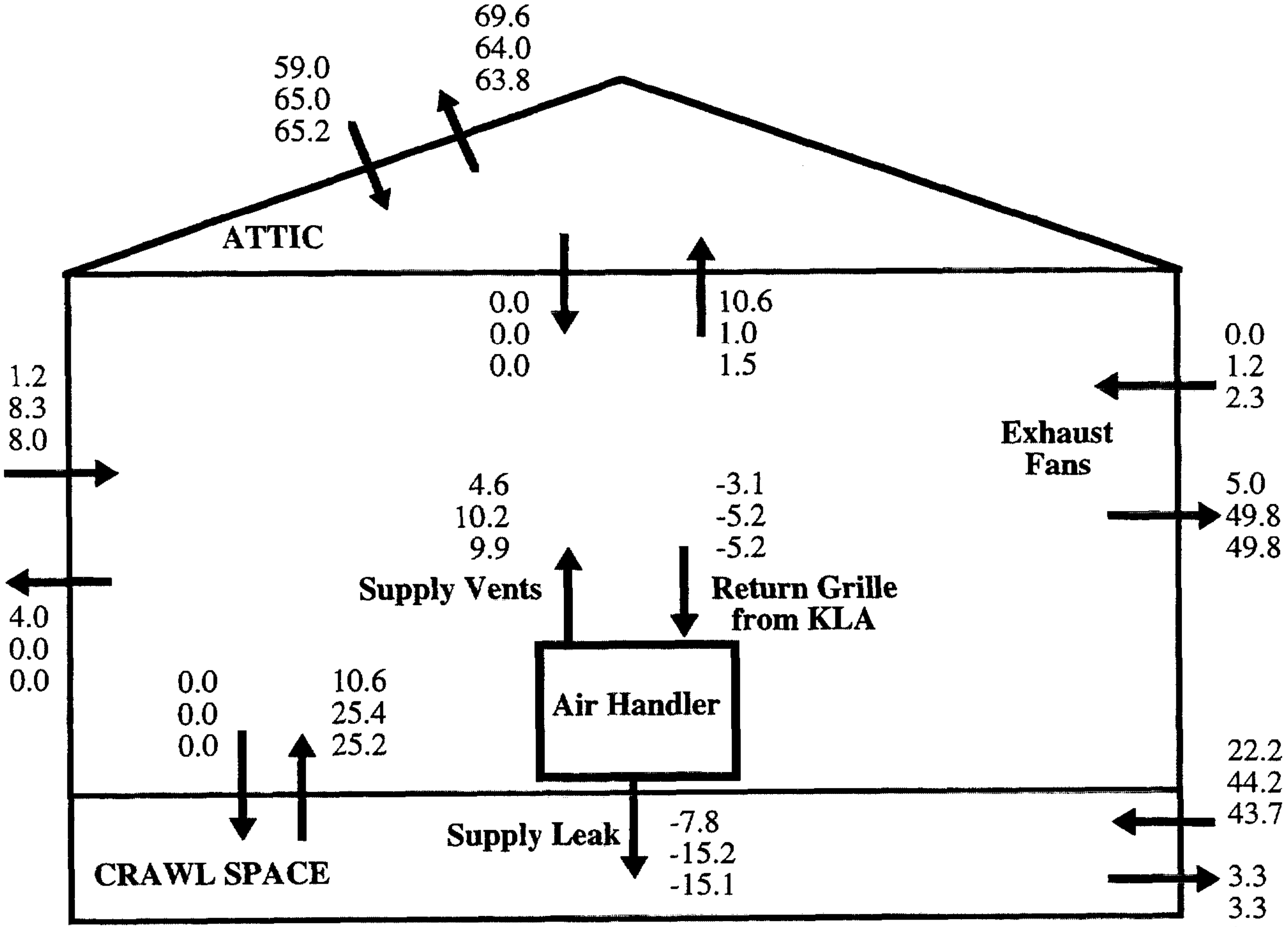


Figure 5 Schematic of Air Distribution Network

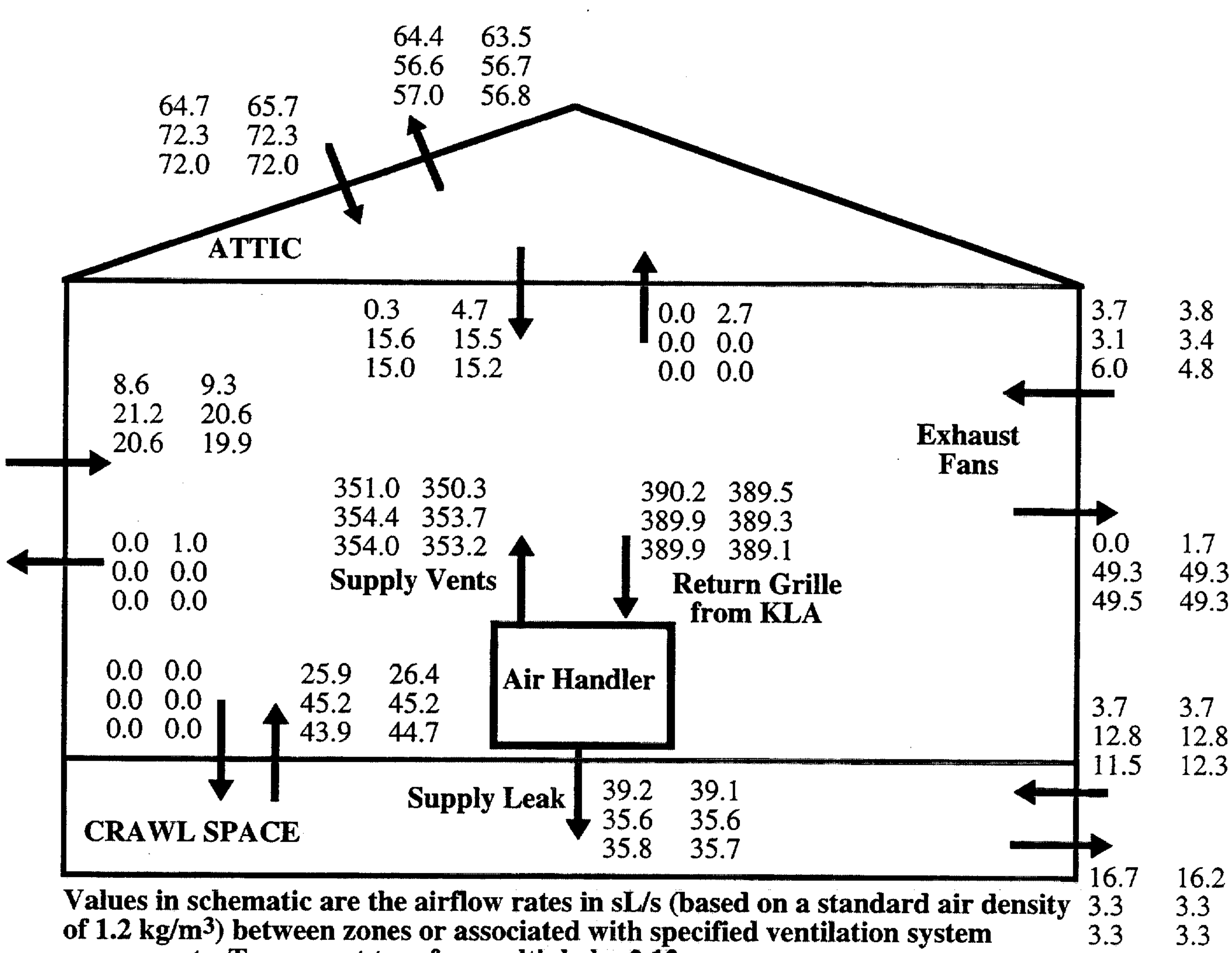


Values in schematic are the airflow rates in sL/s (based on a standard air density of 1.2 kg/m^3) between zones or associated with specified ventilation system components. To convert to scfm, multiply by 2.12.

(Negative numbers represent flows in opposite direction of arrow.)

Cases Displayed
All exhaust fans off
Both bath fans on, kitchen fan off
Kitchen fan on, bath fans off

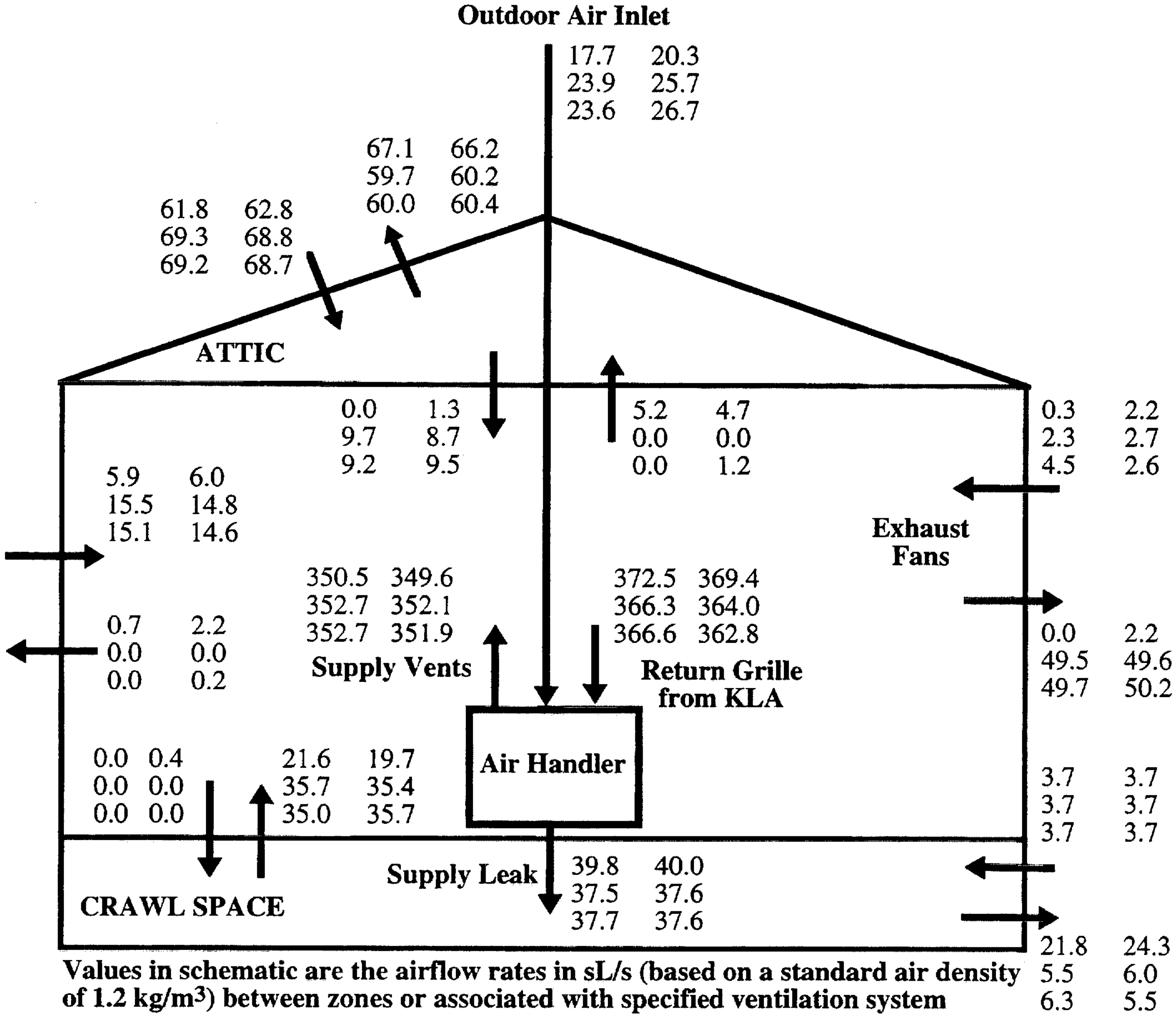
Figure 6 Airflow Patterns with Forced-air Fan Off



Values in schematic are the airflow rates in sL/s (based on a standard air density of 1.2 kg/m^3) between zones or associated with specified ventilation system components. To convert to scfm, multiply by 2.12.

Cases Displayed	Interior Doors	
All exhaust fans off	Open	Closed
Both bath fans on, kitchen fan off	Open	Closed
Kitchen fan on, bath fans off	Open	Closed

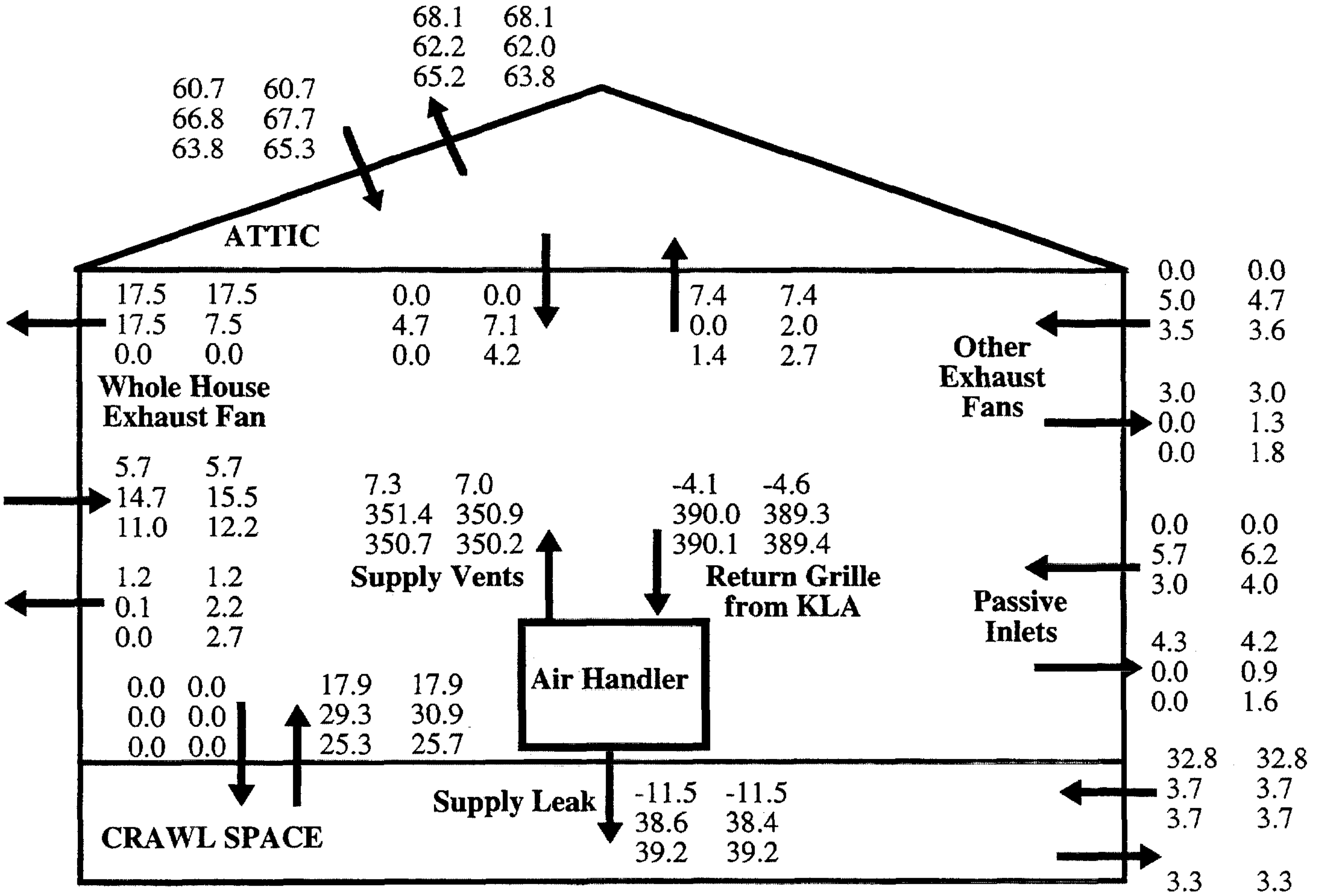
Figure 7 Airflow Patterns with Forced-air Fan On



Values in schematic are the airflow rates in sL/s (based on a standard air density of 1.2 kg/m^3) between zones or associated with specified ventilation system components. To convert to scfm, multiply by 2.12.

Cases Displayed	Interior Doors	
All exhaust fans off	Open	Closed
Both bath fans on, kitchen fan off	Open	Closed
Kitchen fan on, bath fans off	Open	Closed

Figure 8 Airflow Patterns with Outdoor Air Inlet on Forced-air Return

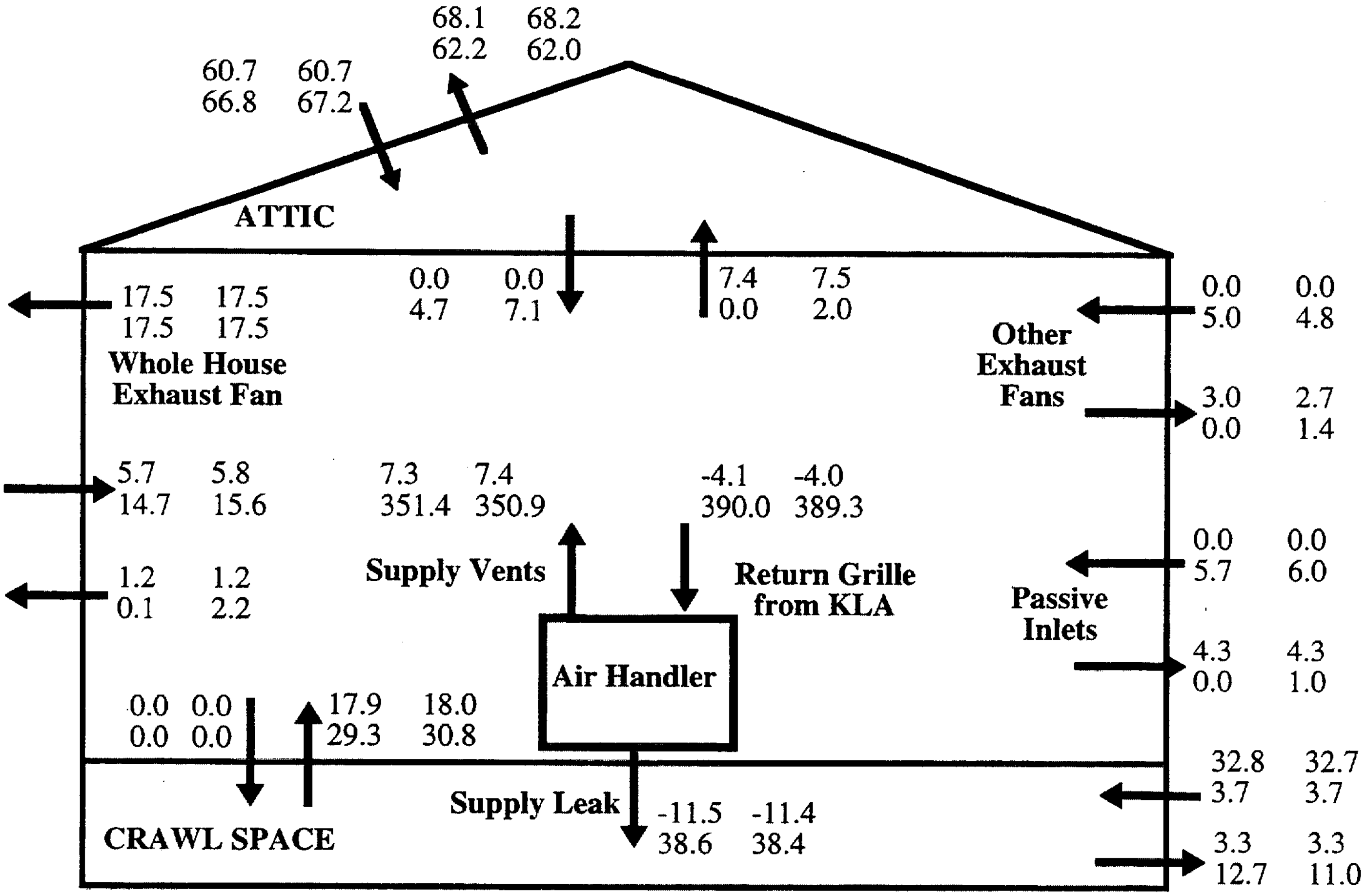


Values in schematic are the airflow rates in sL/s (based on a standard air density of 1.2 kg/m^3) between zones or associated with specified ventilation system components. To convert to scfm, multiply by 2.12.

(Negative numbers represent flows in opposite direction of arrow.)

Cases Displayed	**Interior Doors**	
Whole house fan in KLA on	Open	Closed
KLA whole house and forced-air on	Open	Closed
KLA whole house off, forced-air on	Open	Closed

Figure 9 Airflow Patterns with Passive Inlet Vents and Whole House Exhaust in KLA

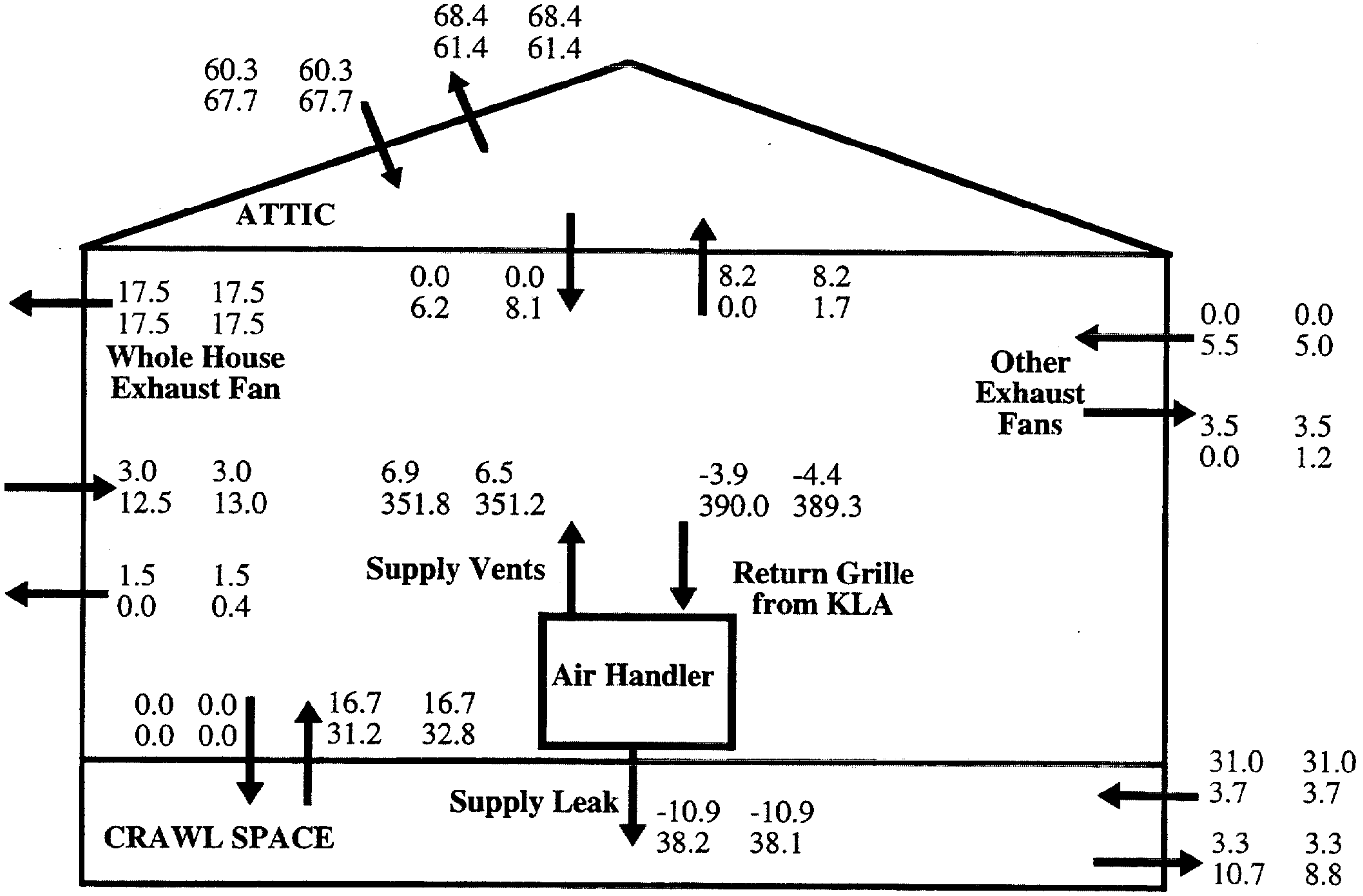


Values in schematic are the airflow rates in sL/s (based on a standard air density of 1.2 kg/m^3) between zones or associated with specified ventilation system components. To convert to scfm, multiply by 2.12.

(Negative numbers represent flows in opposite direction of arrow.)

Cases Displayed	Interior Doors	
Whole house fan in BATH1 on	Open	Closed
BATH1 whole house and forced-air on	Open	Closed

Figure 10 Airflow Patterns with Passive Inlet Vents and Whole House Exhaust in BATH1



Values in schematic are the airflow rates in sL/s (based on a standard air density of 1.2 kg/m^3) between zones or associated with specified ventilation system components. To convert to scfm, multiply by 2.12.

(Negative numbers represent flows in opposite direction of arrow.)

Cases Displayed	Interior Doors	
Whole house fan in KLA on	Open	Closed
KLA whole house and forced-air on	Open	Closed

Figure 11 Airflow Patterns with Whole House Exhaust in KLA, No Passive Inlet Vents

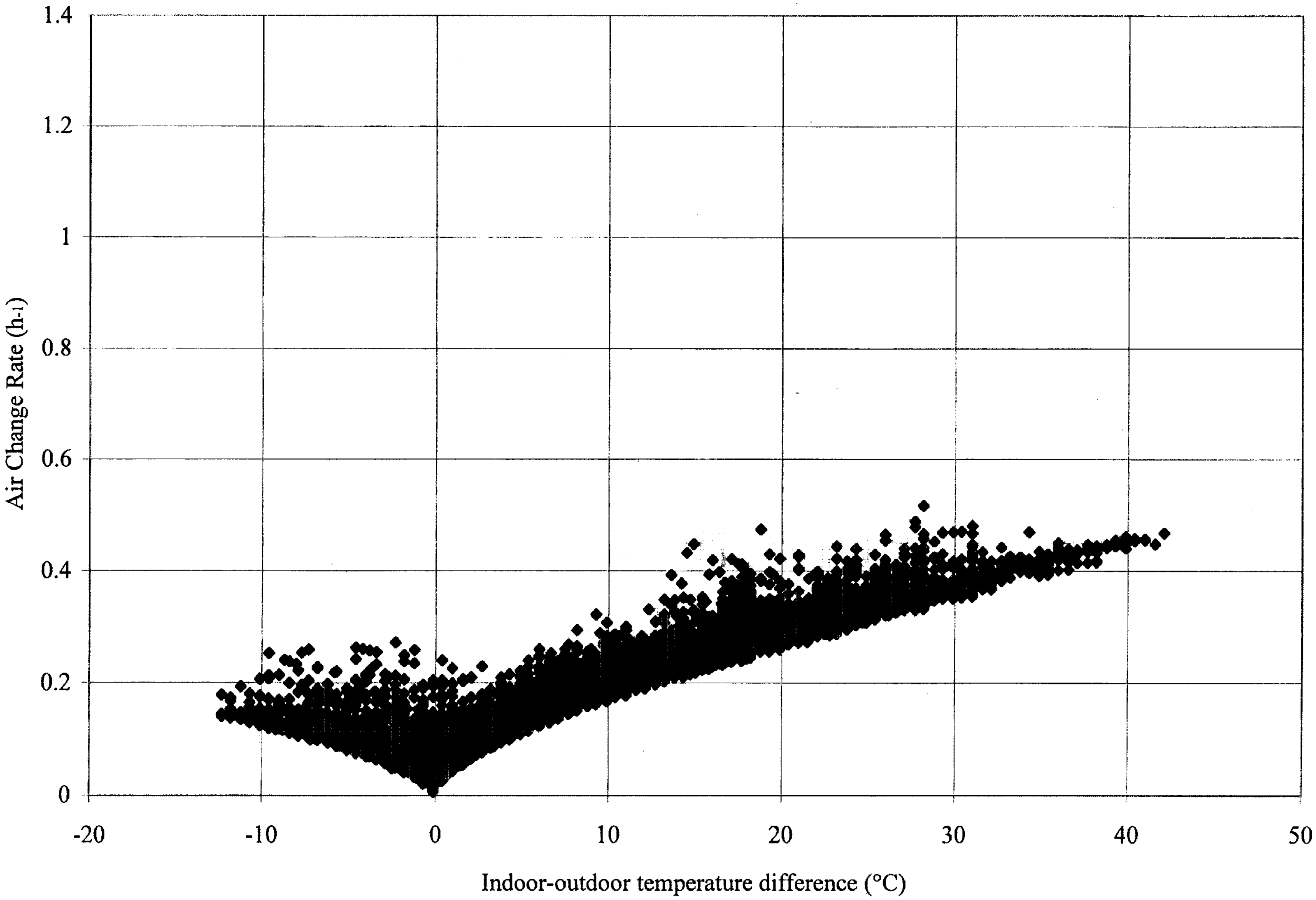


Figure 12 Hourly Air Change Rate for Albany: Envelope Leakage Only (Case #1)

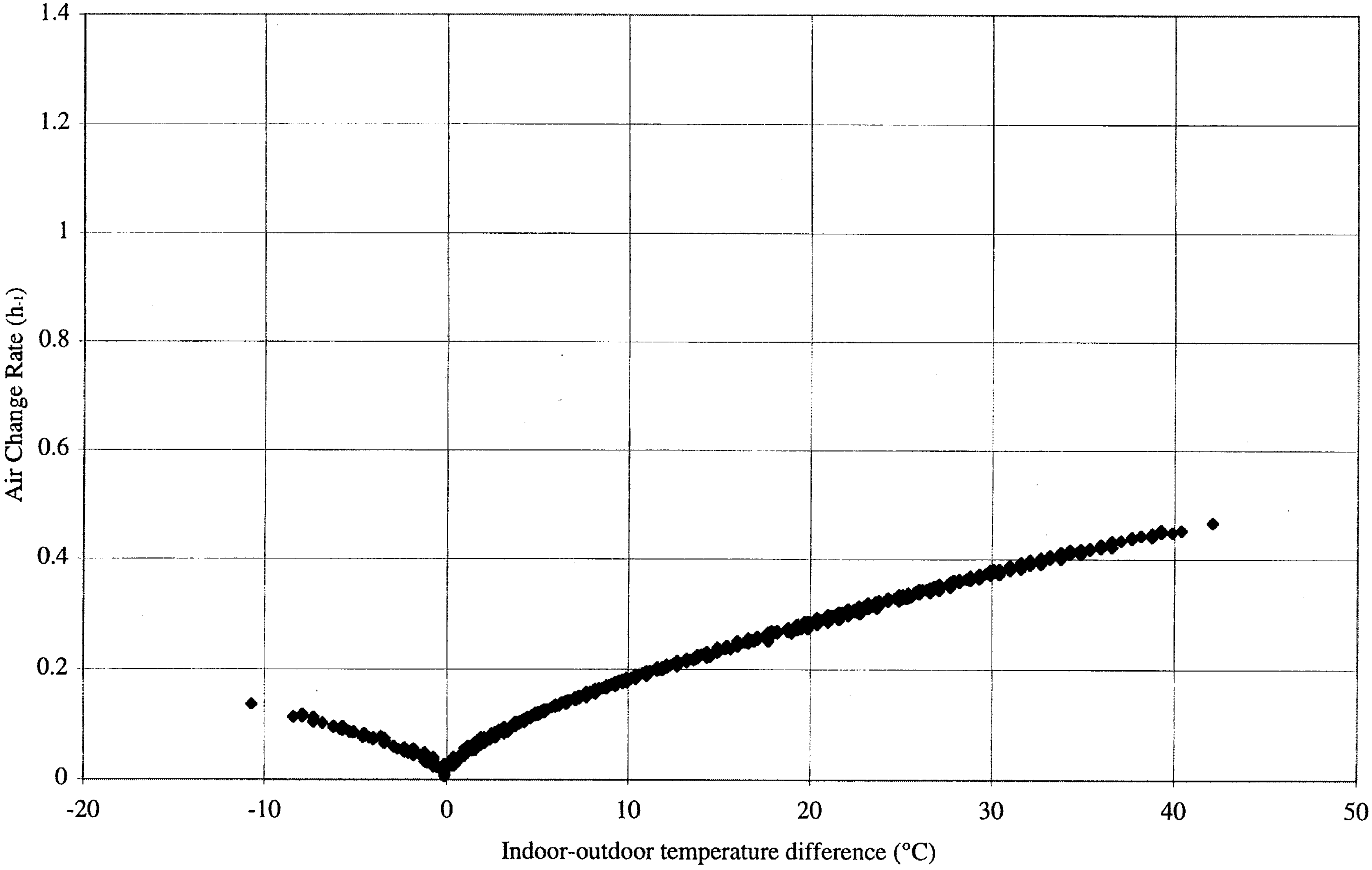


Figure 13 Hourly Air Change Rates for Albany: Wind Speed < 2 m/s, Envelope Leakage Only (Case #1)

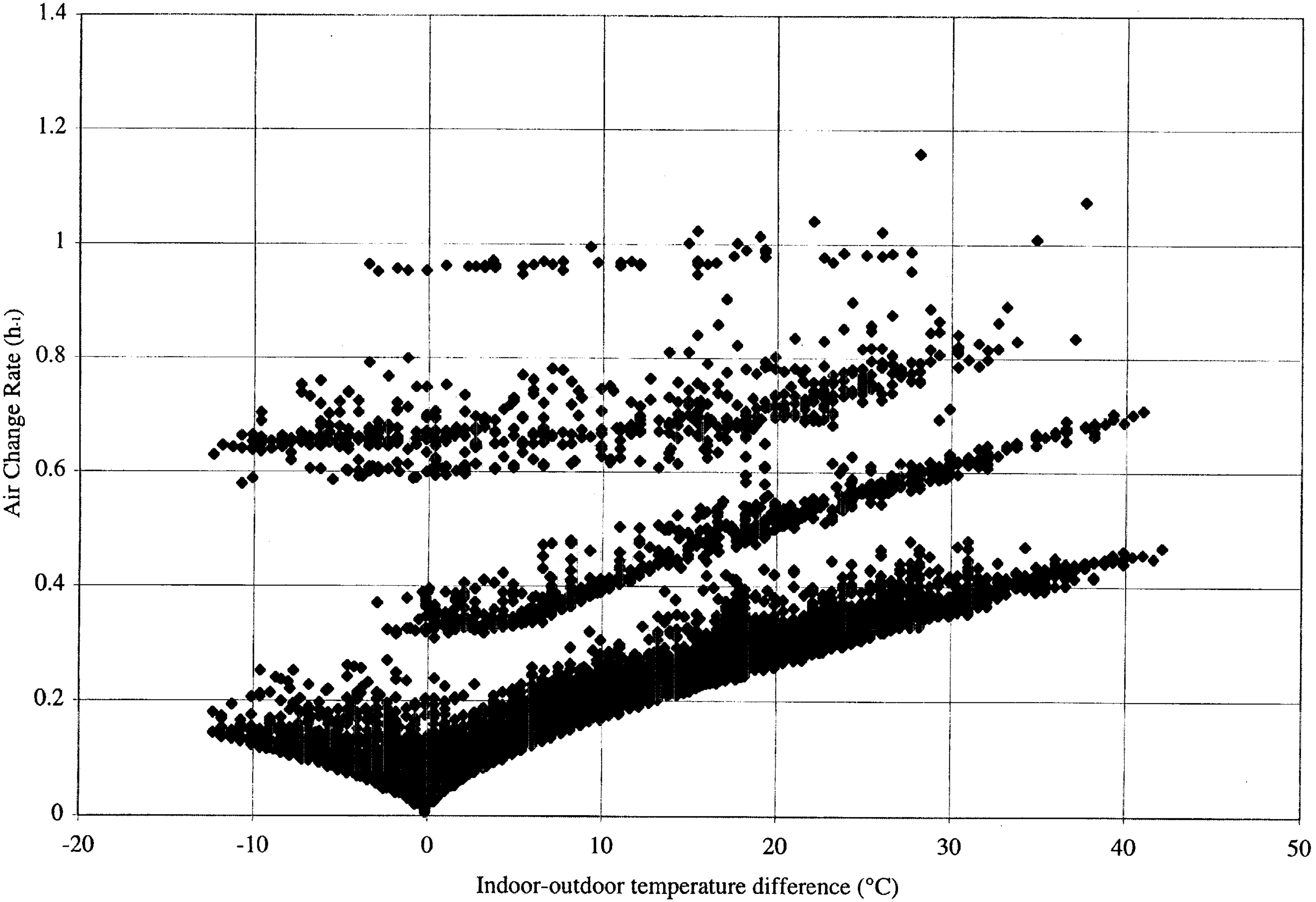


Figure 14 Hourly Air Change Rates for Albany: Scheduled Exhaust Fan Operation (Case #2)

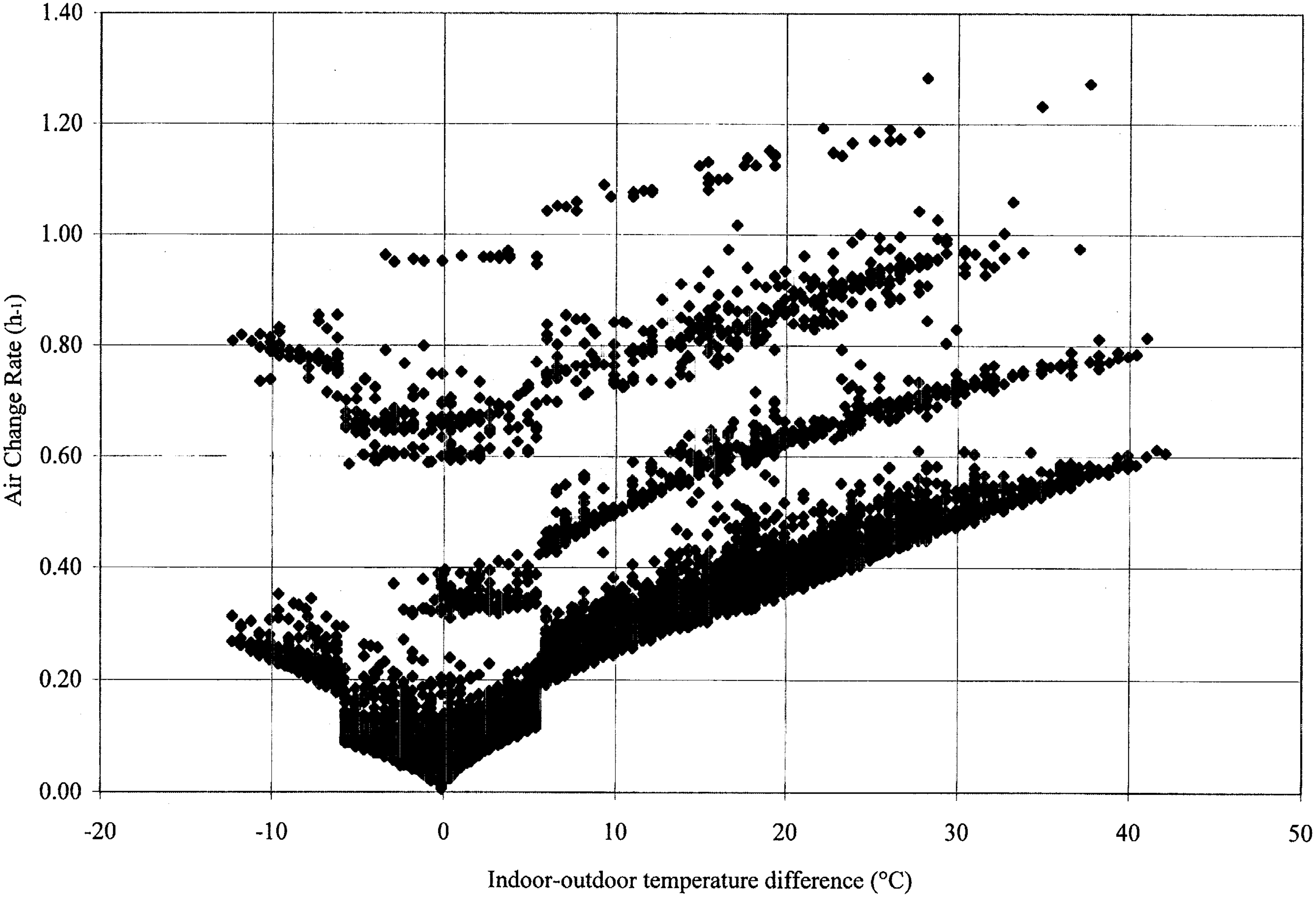


Figure 15 Hourly Air Change Rates for Albany: Exhaust and Forced-Air Fan Operation (Case #3)

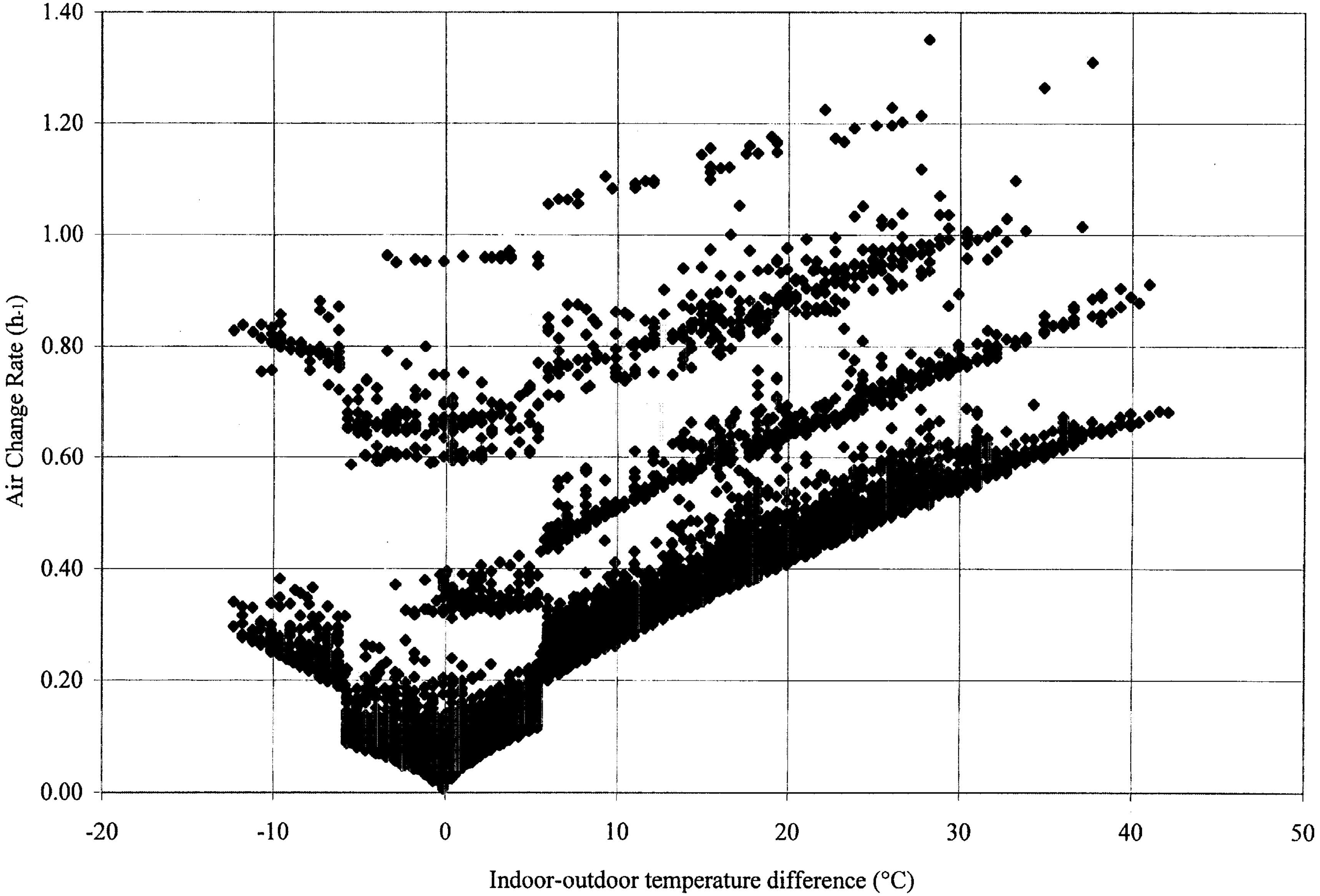


Figure 16 Hourly Air Change Rates for Albany: Exhaust and Forced-Air Inltake Operated Based on Outdoor Temperature (Case #4A)

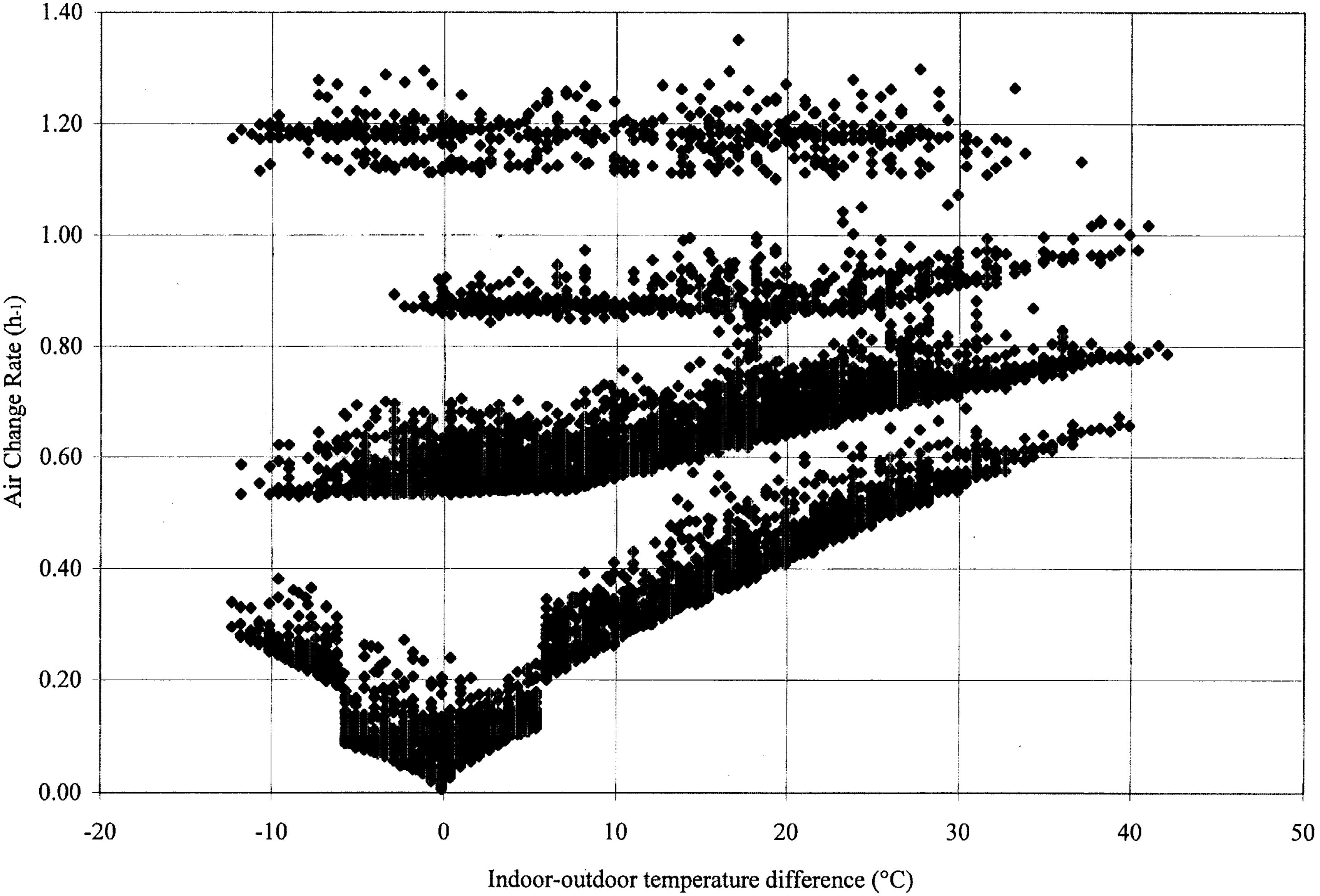


Figure 17 Hourly Air Change Rates for Albany: Exhaust and Forced-Air Intake Operated During Occupancy (Case #4B)

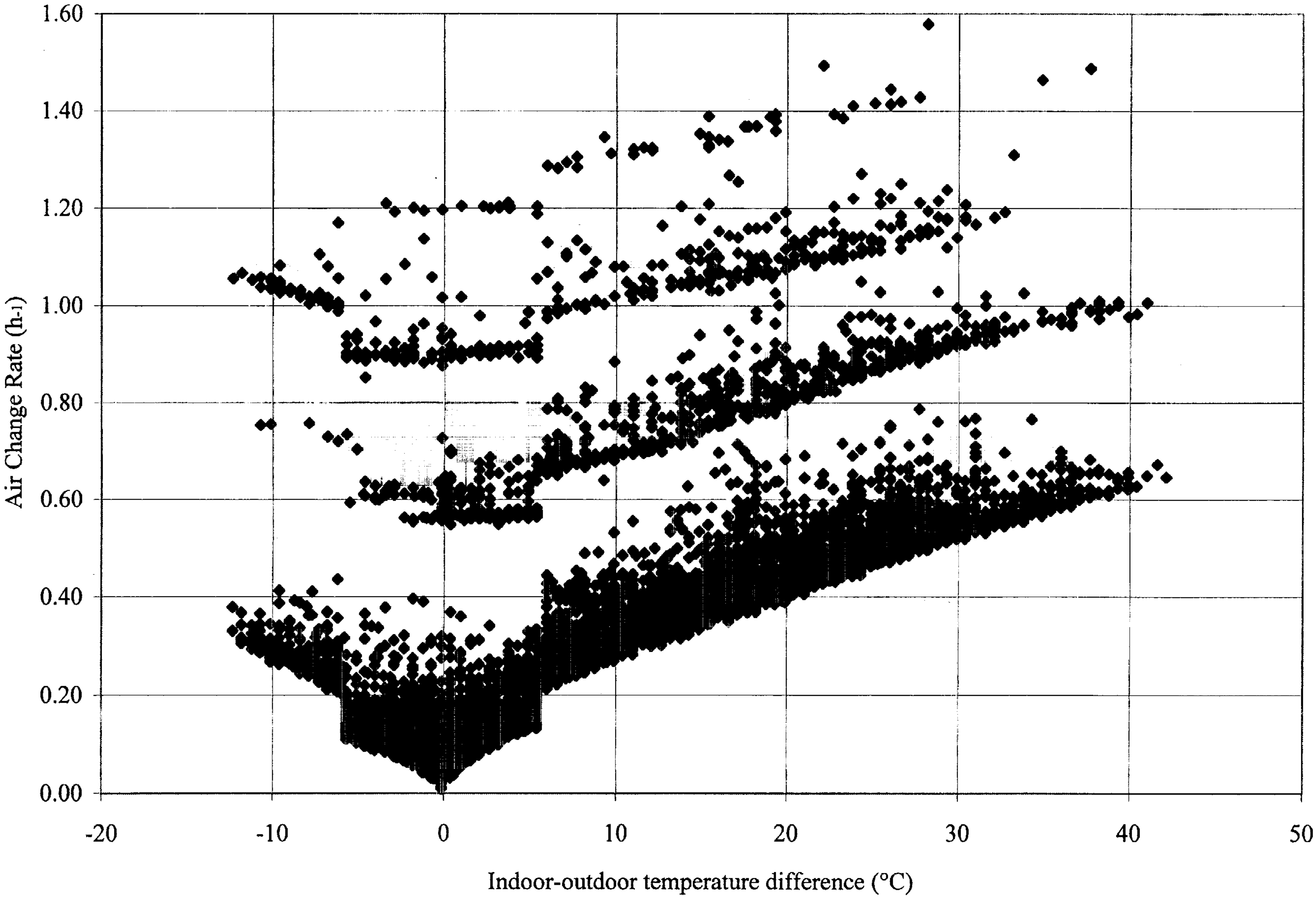


Figure 18 Hourly Air Change Rates for Albany: Whole House Exhaust Fan in KLA Zone Operated on Exhaust Fan Schedule (Case #5A)

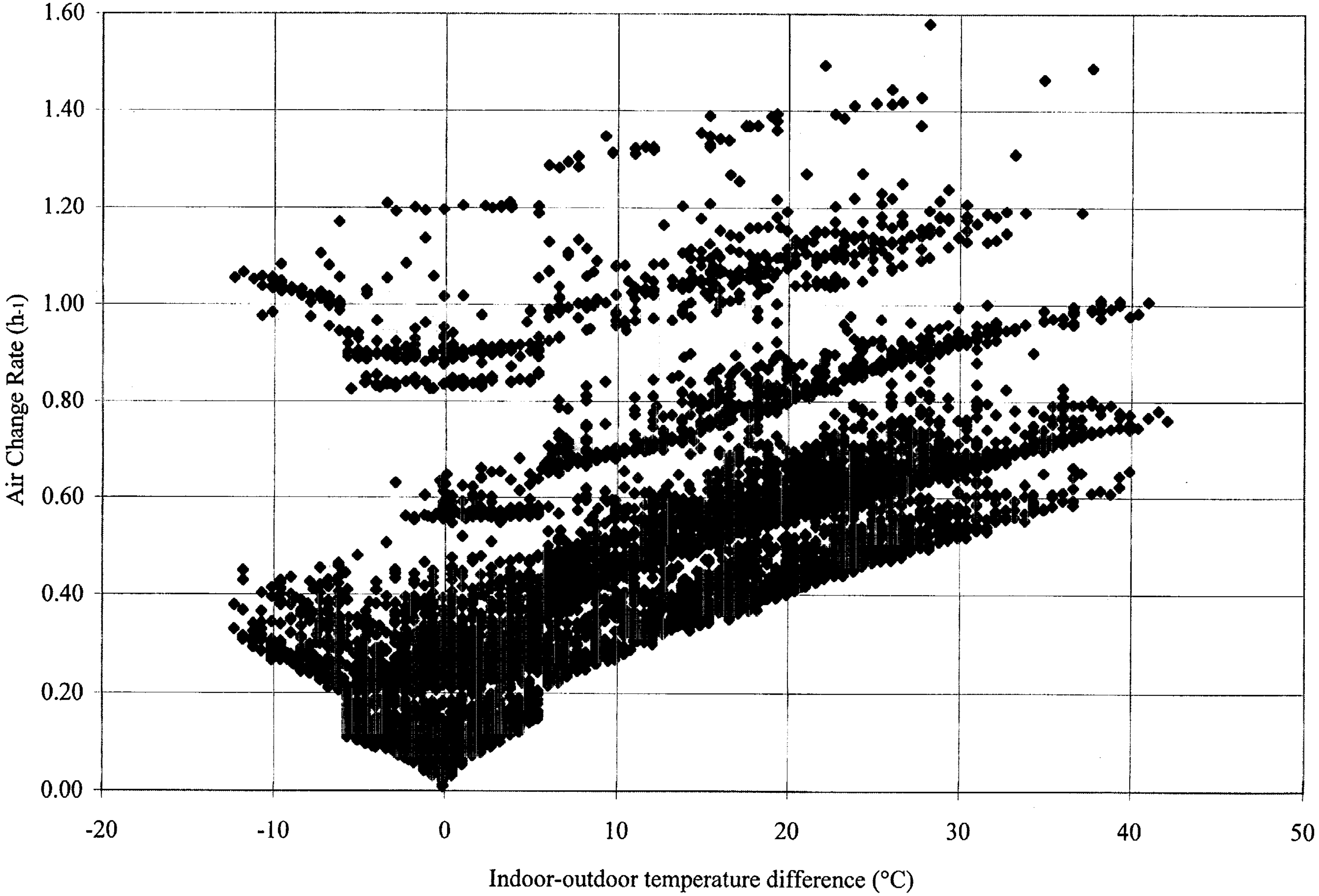
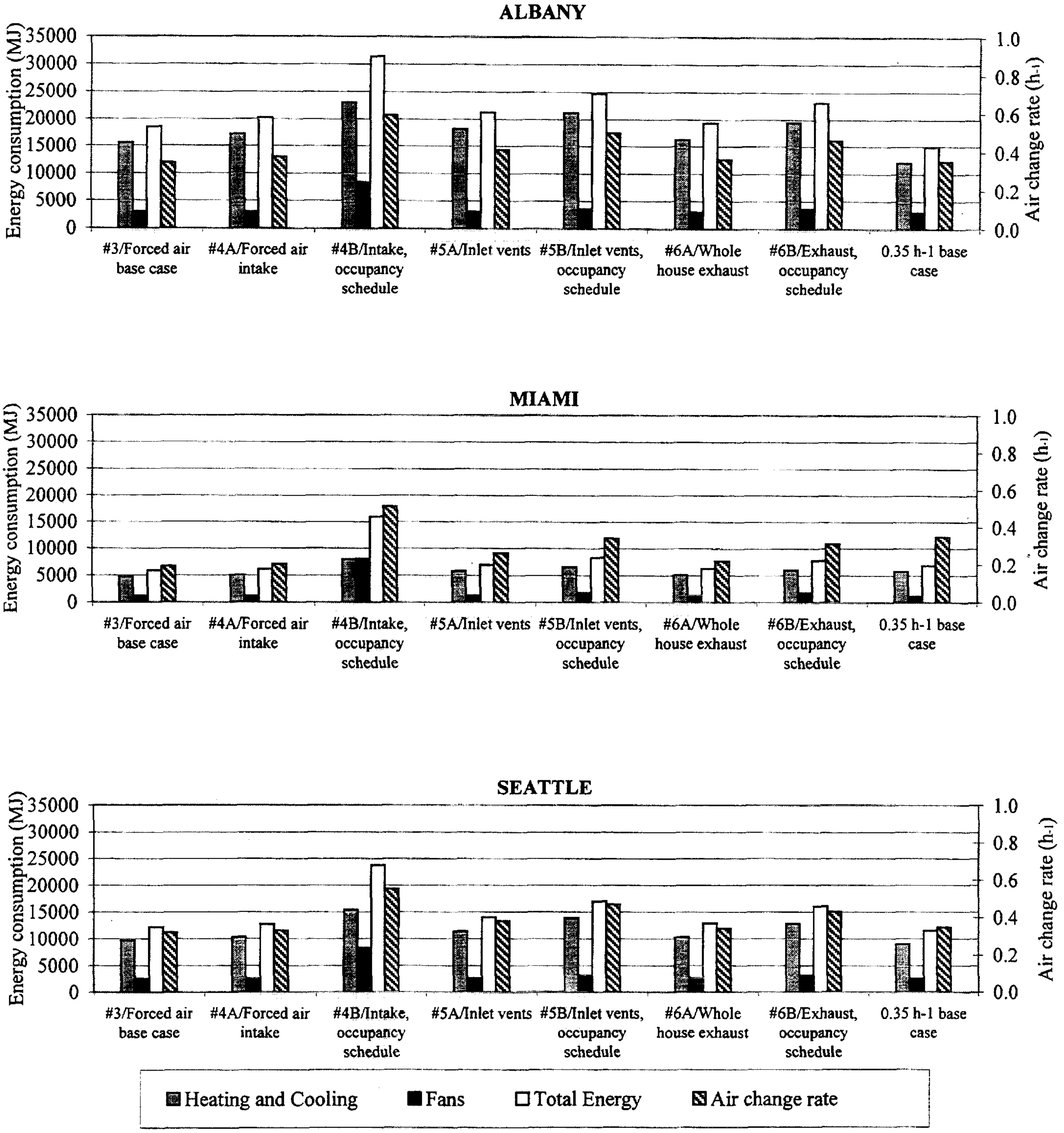


Figure 19 Hourly Air Change Rates for Albany: Whole House Exhaust Fan in KLA Zone Operated During Occupancy (Case #5B)



The three energy values (Heating and Cooling, Fans and Total Energy) refer to left-hand scale and air change rate values refer to right-hand scale.

Figure 20 Summary of Energy Consumption and Air Change Rates

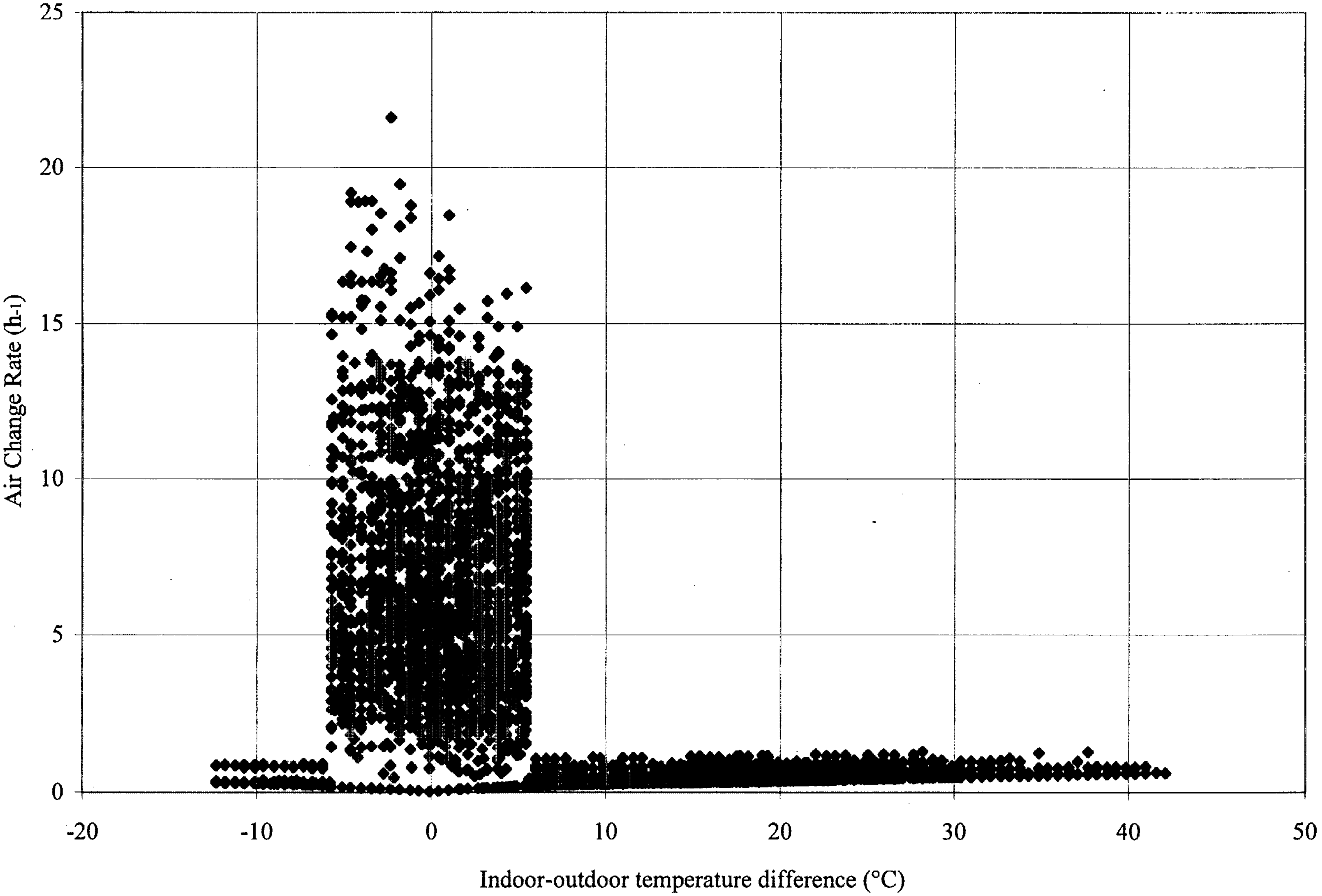


Figure 21 Hourly Air Change Rates for Albany: Window Operation