A Modeling Study of Ventilation in Manufactured Houses

Andrew K. Persily, Ph.D.
Member ASHRAE

Samuel R. Martin

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. In order to obtain some insight into these issues, the multizone airflow and indoor air quality program CONTAM96 was used to simulate a double-wide unit under several different ventilation scenarios. Annual simulations were performed in three cities to assess ventilation rates and energy consumption associated with these scenarios. The results of these simulations are presented and discussed, and recommendations are made for changes to the HUD standards and for future research.

INTRODUCTION AND BACKGROUND

The HUD 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). This paper describes a study that addresses some questions regarding the ventilation–related requirements and their implementation. A more complete report on this study is available in Persily and Martin (2000).

The first ventilation requirement in the MHCSS 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 states that infiltration shall be considered to provide an air change rate of 0.25 h^{-1}, with the remaining 0.10 h^{-1} 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 (0.035 cfm per ft^2) of interior floor space. The standard states that this additional capacity shall be in addition to any openable window area. There are also requirements for local exhaust, including 47 L/s (100 cfm) from the kitchen and 24 L/s (50 cfm) from each bathroom and toilet. A toilet may be ventilated with 0.14 m^2 (1.5 ft^2) of openable glazed area in place of mechanical ventilation, except in northern climates.

Issues Related to MHCSS Ventilation Requirements

The objective of this project is to use multizone airflow analysis to investigate some of the technical issues related to the MHCSS ventilation requirements and their implementation. The airflow model CONTAM96 (Walton 1997) was used to perform these simulations in a manufactured home intended to be typical of current construction practice in the U.S. 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

Andrew K. Persily and **Samuel R. Martin** are mechanical engineers at the National Institute of Standards and Technology, Gaithersburg, Md.

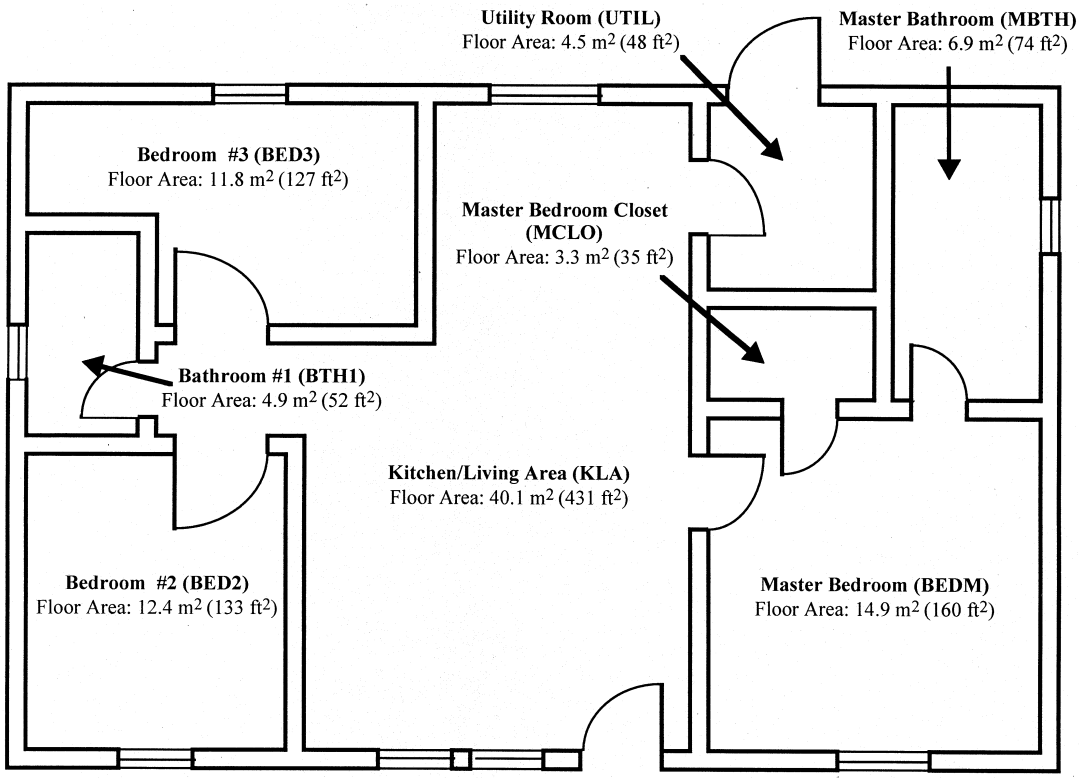

Figure 1 *Layout of living space.*

The first issue relates to the assumption of 0.25 h^{-1} from infiltration, since it is key to the supplemental ventilation requirement of 0.1 h^{-1}. The second issue addresses a common approach for meeting the MHCSS ventilation requirement, which is an outdoor air inlet duct connected to 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 the reliability of the ventilation rates achieved and the fan energy consumed. 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 whole house exhaust without inlet vents. Finally, while whole house exhaust fans can be used to meet the supplemental ventilation requirement, current interpretation of 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.

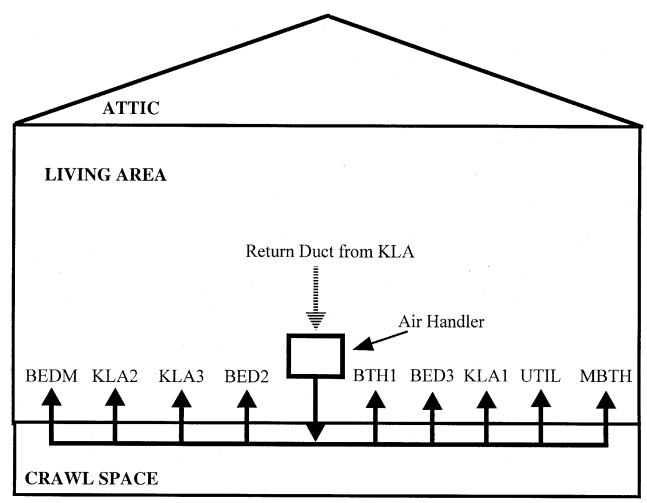

Figure 2 *Schematic of air distribution network.*

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. The house has a forced-air heating and cooling system with the supply ductwork located in the crawl space, as shown in Figure 2. The conditioned space of the house has a floor area of 98.8 m^2 (1063 ft^2) and a volume of 250.9 m^3 (8860 ft^3). The

floor areas of the crawl space and attic are identical to the conditioned space, and their volumes are 59.3 m^3 (2094 ft^3) and 39.5 m^3 (1394 ft^3), respectively. The house has a cathedral ceiling, with an 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 details of the airflow model of the house are contained in Persily and Martin (2000) and draw heavily from two previous modeling studies—Emmerich and Persily (1996) and Persily (1998). Note that the house model was assembled to be consistent with typical levels of envelope airtightness for U.S. manufactured homes, though not at the levels being achieved in the most energy-efficient buildings. The results of simulated pressurization tests, contained in Persily and Martin (2000), confirm that this goal was indeed achieved.

Ventilation Systems

This section describes the ventilation systems in the model house, including the forced-air system, the local exhaust systems, and the ventilation approaches used to meet the supplemental ventilation requirement of the MHCSS.

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, the same system layout is used in all three cities to facilitate comparisons of the various cases analyzed. A schematic of the air distribution system is shown 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 return grille is located in the kitchen/living area.

The duct system modeling capabilities of CONTAM96 were used to model airflow through the forced-air system. The system was modeled to provide 24 L/s (50 cfm) to one of the bathrooms and to the utility room, 35 L/s (75 cfm) to the master bath, 47 L/s (100 cfm) to the two smaller bedrooms, 83 L/s (175 cfm) to the master bedroom, and 142 L/s (300 cfm) to the kitchen/living area, for a total supply airflow rate of 402 L/s (850 cfm). 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, based on field measurements in a number of manufactured houses (AEC 1996; Baylon et al. 1995; Cummings et al. 1991; Davis et al. 1996).

In the simulations that include forced-air fan operation, the fan operating time is based on 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, 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:

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

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, the fractional on-time ff_c is expressed as

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

In the case of cooling, ΔT is based on an indoor air temperature of 24.4°C (75.9°F). Equations 1 and 2 were derived based on field experience that reveals 30% to 60% on-times are not unusual under design conditions and an attempt to match these "targets" with the Albany climate. Also, these same equations were used in a previous study (Persily 1998).

These fractional on-times define the extent of oversizing of the heating and cooling systems, and the use of the same on-time equations in the three cities corresponds to different levels 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. While the extent of oversizing is quite large, especially under cooling, it is not unusual in current practice. Forced-air system oversizing impacts residential ventilation rates through duct leakage and for mechanical ventilation systems that rely on the forced-air system for intake. The issue of oversizing is important and merits further study; however, this was not part of the scope of this particular effort.

Fan operation is accounted for by first determining the whole house air change rates with CONTAM96 for each hour, both with the fan on and with the fan off. 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.

Local Exhaust Fans. The house has exhaust fans in the kitchen and in each of the two bathrooms, all of which are ducted directly to the outdoors and are modeled with the CONTAM96 duct model. The kitchen fan has a capacity of 47 L/s (100 cfm) and the two bathroom fans have a capacity of 24 L/s (50 cfm), all at a pressure of 25 Pa (0.1 in. H_2O). 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. The kitchen exhaust fan is assumed to operate from 5 p.m. to 6 p.m. daily. The bathroom exhaust fan in the master bath runs from 6 a.m. to 7 a.m. on weekdays and 8 a.m. to 9 a.m. on weekends. The other bathroom 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. A 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 corresponds to the 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. Obviously, other schedules are possible, as are other automated control approaches, but these were not studied in this project.

The other approach to meeting the supplemental ventilation requirements involves a whole house exhaust fan with or without passive inlet vents in 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, and is located in the main living area (the kitchen/living area). 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 four such vents. Each vent is modeled as an effective leakage area of 13 cm^2 (2 $in.^2$) and is located 0.3 m (1 ft) from the ceiling. Simulations with the exhaust fan were run with and without the inlet vents. Simulations of the exhaust fan 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 or kitchen exhaust fans would be operating. Under the other schedule, the exhaust fan operates whenever the house is occupied. Again, other approaches to ventilation system control, such as the use of timers, are possible and used in practice, but these were not examined in this study.

SIMULATION AND ANALYSIS APPROACH

As discussed earlier, the simulations employed the multizone airflow and contaminant dispersal model CONTAM96 (Walton 1997), which considers a building as a system of interconnected volumes or zones. Each zone is assumed to be at a uniform temperature, pressure, and contaminant concentration, sometimes referred to as an assumption of "perfect mixing." 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, CONTAM96 calculates airflow rates between each zone under steady-state or transient conditions based on a simultaneous mass balance of air in each zone. Given information on contaminant sources and removal mechanisms and on outdoor contaminant concentrations, CONTAM96 can determine contaminant concentrations in the zones; 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 N.Y., Miami Fla., and Seattle Wash., using TMY2 weather data (Marion and Urban 1995). The annual simulations analyzed 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.

Case #1 was analyzed to assess building ventilation rates due to envelope leakage alone. The house model contains no fans or ducts, 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 as described earlier. In this case, the house model also includes the air distribution ductwork and supply duct leakage, though the forced-air fan is not on. Case #3 includes the same exhaust fans plus forced-air fan operation based on outdoor temperature. This case reflects the increase in 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.

When using simulation studies to compare different ventilation strategies, it is not totally straightforward as to how one presents the results and compares the different approaches. This study is focused primarily on how these strategies compare with the requirements of the MHCSS. In general, two key issues when comparing different strategies are energy consumption and indoor air quality. Energy consumption is accounted for in this study through hourly air change rates as described below. As noted later in this paper, the annual mean air change rate for each case (in each city) is roughly proportional to the total energy consumption so that mean is a useful indicator of energy. Evaluating and compar-

ing indoor air quality is more problematic. Ideally, one would examine occupant exposure to all relevant contaminants. Unfortunately, we do not know enough about which contaminants are of most interest from either a health or comfort perspective, the sources and source strengths of these contaminant, and the concentration limits for evaluating occupant exposure. Therefore, we use ventilation rate as a surrogate for indoor air quality, as is done in ASHRAE Standard 62 and in the MHCSS.

In this study, we first examine steady-state air change rates for the various ventilation strategies and compare them to these standards, but these values do not consider variations in air change rate due to system operation and weather conditions. It turns out that neither standard specifies how to account for variations over time in assessing compliance. Multiple approaches are used in this paper to summarize the annual air change rate simulations—annual and monthly mean air change rates, effective air change rates over the year, and the percentage of hours during the year for which the air change rate is below 0.25 h^{-1} (the infiltration assumption in the MHCSS) and 0.35 h^{-1} (the residential requirement in Standard 62). The effective air change rate over a period of time is the constant air change rate that would yield the same average contaminant concentration over the same period as does the actual time-varying air change rates. It is determined by averaging the inverse of the individual air change rates (in this case hourly) and then taking the average of this inverse (Yuill 1991). The determination and use of effective air change rates as a measure of residential air change characteristics are covered by ASHRAE Standard 136 (1993). In fact, while ASHRAE Standard 62 does not discuss variations in ventilation rate within the context of compliance, an official interpretation of the standard allows the use of effective air change rates to demonstrate compliance with the residential ventilation requirement. The effective air change rate provides a well-established assessment of indoor air quality impacts that is most relevant to sources that are constant in time and uniformly distributed throughout the building volume. Many important contaminant sources do not fit this description, for example, those that occur over short periods of time in specific locations within a building (e.g., cooking, use of consumer products). While the annual mean of the hourly air change rates do not capture the indoor air quality implications of different ventilation strategies as well, they are still used in this report. It is important to note that given the dependence of the predicted air change rates on many model inputs and the great range of situations that exist in real buildings, the absolute values of the average air change rates, effective rates or any other parameters are less important than their relative value. This fact makes absolute comparisons to ASHRAE or MHCSS less reliable than relative comparisons among the different cases to each other and, therefore, whether the average or effective value is being compared becomes somewhat less critical.

The airflow simulations focused on ventilation rates and how they compare with the requirements in the MHCSS. However, additional analyses were performed to better understand the airflow characteristics of the simulated house. These additional aspects include pressurization tests to determine the airtightness of the building envelope, effective air change rates (sometimes referred to as "temporal ventilation effectiveness"), analyses of airflow patterns between the major volumes of the house, and determinations of the age of air to characterize outdoor air distribution or mixing within the building. Pressurization tests, effective air change, airflow patterns, and air distribution are discussed in the full report on this study (Persily and Martin 2000).

As noted earlier, 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 rates are calculated annually.

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

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

where

ρ = air density, 1.2 kg/m^3 (0.075 lb$_m$/ft^3),

C_p = specific heat of air, 1000 J/kg·°C (0.239 Btu/lb$_m$·°F),

I = house air change rate, h^{-1},

V = house volume, 251 m^3 (8,860 ft^3), and

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 Q_L is based on the following expression:

TABLE 1 *
Steady-State Air Change Rates for Different House and Fan Configurations

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

* Note: All values correspond to 20ºC (36ºF) temperature difference and zero wind speed.

$$Q_L = \rho h_{fg} IV\Delta W \times 1 \text{ hour} \quad (4)$$

where

h_{fg} = latent heat of water vapor, 2.34×10^6 J/kg (1010 Btu/lb_m), and

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 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. 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. As seen later, the forced-air fan energy is significant in some cases and, therefore, could have significant impact on heating and cooling loads. The extent of this impact will depend on climate, system operation, and other factors, but it tends to be larger for cooling than for heating.

RESULTS

This section presents the results of the simulations, specifically ventilation rates and the energy consumption associated with infiltration and ventilation.

Air Change Rates

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 1 presents these steady-state air change rates, all of which correspond to an indoor-outdoor air temperature difference of 20°C (36°F) and zero wind speed. These rates are determined by adding all of the flows into the conditioned space from outdoors, the attic, and the crawl space, including any 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. These air change rates can be converted to L/s (cfm) by multiplying by 70 (148). The requirement of 0.35 h^{-1} in ASHRAE Standard 62 corresponds to 25 L/s (52 cfm).

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 leakage into the crawl space, operating the forced-air fan depressurizes the building, increasing infiltration and yielding an air change rate of 0.55 h^{-1} with all exhaust fans off. 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.

Ventilation Rates

This section discusses the whole building air change rates that were calculated for the three cities for each hour of the year and then averages by month and annually. As noted

earlier, mean air change rates are an imperfect indicator of long-term ventilation rates, as they do not account for variations, but they are still useful for comparing the different approaches.

Case #1—Envelope Leakage Only. Airflow simulations were performed for Case #1, all fans off and all interior doors open, to evaluate ventilation rates due to envelope leakage alone. For these simulations, the house model contained no ducts or fans. Figure 3 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 rates is due to wind. Similar patterns exist for the other two cities.

Table 2 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

TABLE 2
Summary of Air Change Rates for Case #1 (Envelope Leakage Only)

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	0.35	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

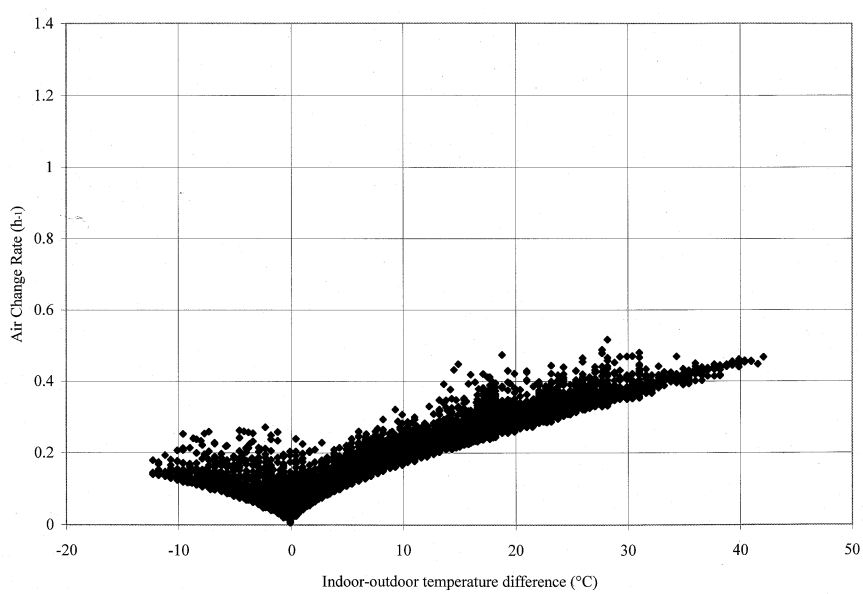

Figure 3 *Hourly air change rate for Albany: envelope leakage only (Case #1).*

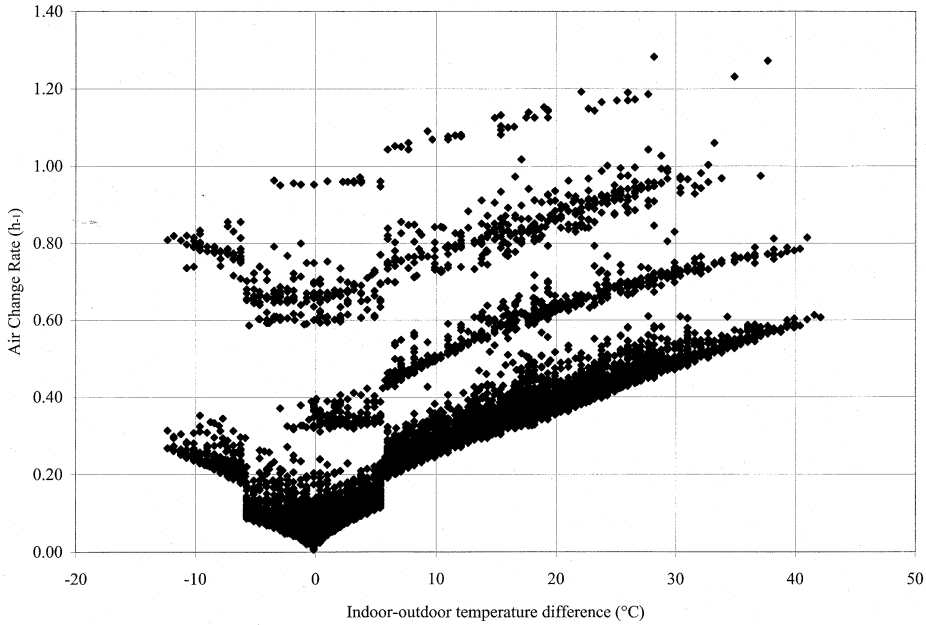

Figure 4 *Hourly air change rates for Albany: exhaust and forced-air fan operation (Case #3).*

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. In Case #2, the bath and kitchen exhaust fans and the clothes dryer operate on occupancy-based schedules, but the forced-air fan is always off. The scheduled exhaust fan operation raises the mean monthly air change rates, relative to the envelope leakage only conditions, by about 0.05 h^{-1}. And as expected, 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. This duct leakage lowers the pressure in the conditioned space, increasing infiltration flows from outdoors and unconditioned spaces. Figure 4 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. The data in this plot fall into four groups, with the lower group corresponding to conditions in which all the exhaust fans are off. The next higher group contains hours for which one of the bath fans operates. 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 1.0 h^{-1}, corresponds to hours during which the kitchen exhaust or the clothes dryer is operating. The highest group of rates, 1.0 h^{-1} to 1.2 h^{-1}, corresponds to the dryer and one of the bath fans operating at the same time. 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.

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

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	0.51	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 3 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 Case #2. The magnitude of the increase depends on the outdoor air temperature, 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}. 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. Note that the air change rates under conditions of heating and cooling are highly dependent on the extent of system oversizing. All else being equal, a more properly sized system would result in longer operating times under given weather conditions and, therefore, higher air change 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 increases the air change rate by about 0.1 h^{-1} with the intake open. Therefore, the impact of this ventilation approach is a strong function of operating time. 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 demand when the house is unoccupied. The air change rates for Case #4A are not very different from Case #3 due to the limited time of forced-air fan operation. 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 an hour under the most extreme weather conditions. More typically, the forced-air fan runs for 15 to 30 minutes each hour. Obviously, these results depend on the extent of system oversizing. A more properly sized system would have longer operating times under given weather conditions and, therefore, higher air change rates.

Table 4 contains a summary of the air change rates for Case #4A. The addition of an outdoor air intake duct on the forced-air return increases 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 change for milder months. The annual means increase by only 0.01 h^{-1} to 0.03 h^{-1}. The percent of hours below the reference air change rates decrease by a moderate amount, with the largest decreases in the colder and warmer months. Figure 5 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 than 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 operate only when the building is occupied and, therefore, the forced-air fan is on, eliminating the discontinuity in the lower group of points.

Table 5 contains a summary of the air change rates for Case #4B. With the forced-air fan operating whenever the

TABLE 4
Summary of Air Change Rates for Case #4A (Intake on Forced-Air, 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	0.57	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

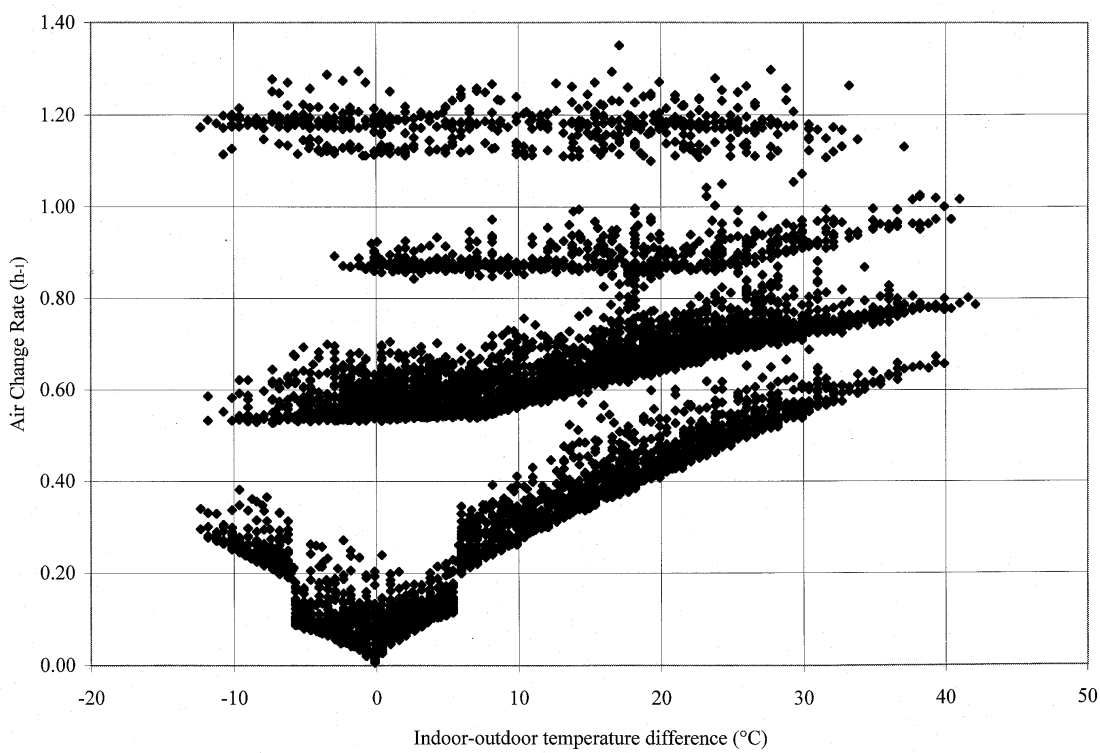

Figure 5 *Hourly air change rates for Albany: exhaust and forced-air intake operated during occupancy (Case #4B).*

TABLE 5
Summary of Air Change Rates for Case #4B (Intake on Forced-Air, Occupancy 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	0.70	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

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.

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. 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. 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 fan operation, 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} from 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. The dependence of air change rate on temperature difference is similar to that seen in Figure 5 for Case #4B, except the rates are increased based on the existence of the whole house exhaust fan and the inlet vents. 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.

For Case #5B, 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. With the whole house exhaust fan operating whenever the house is occupied, the monthly mean air change rates increase about 0.1 h^{-1} relative to those 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.

Cases #6A and #6B—Whole House Exhaust Fan without Passive Inlet Vents. Airflow simulations were also performed with a whole house exhaust fan but without inlet vents. As in Cases #5A and #5B, the exhaust fan is located in the KLA zone. In Case #6A, the whole house exhaust operates on the same schedule as the other local exhausts. In Case #6B, the whole house exhaust operates during occupancy. As noted earlier, operating the whole house exhaust 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} from 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}.

Relative to Case #5A with the same exhaust schedule but with the inlets, the monthly means for Case #6A decrease by about 0.03 h^{-1} to 0.06 h^{-1} and the annual means decrease by about 0.05 h^{-1}. The number of hours below the two reference air change rates is higher, but only by 10% to 20%. 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 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 decreases the mean air change rates by only about 0.04 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 with a whole house exhaust is not dramatically different from a whole house exhaust without the inlets.

Summary and Comparison to Measured Ventilation. Table 6 contains the annual mean air change rates for all cases, the percentages 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), and the effective air change rate over the year. As mentioned earlier, the rates for cases with the forced-air fan operating are highly dependent on the extent of system oversizing, and the system studied was extremely oversized. All else being equal, a more properly sized system would have had higher air change rates under most circumstances.

Case #1, corresponding to envelope infiltration only, has annual mean air change rates below the 0.25 h^{-1} MHCSS assumption for all three cities. And, as expected, the effective air change rates are even lower. 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. The effective air change rates for Case #3 are below 0.25 h^{-1} for all three cities. As expected, the two supplemental ventilation approaches have higher mean and effective 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. The effective rates are close to or below 0.35 h^{-1} for all cases, with the "A" schedules having the lowest values. Case #4B does the best job in meeting the required ventilation rates of Standard 62, but the energy impacts of this ventilation strategy can be significant as noted below.

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. These percentages are based on only the occupied hours of the year—that is, 5895 hours based on the assumptions in this analysis versus 8760 hours for the whole year. The details of this calculation are presented in the full report on this study (Persily and Martin 2000). 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 the increase is 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 decrease 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 from occupant activities, unoccupied rates are relevant for other contaminants. Water vapor is a good example of the latter, as humidity control from non-occupant sources requires ventilation during more than just occupied hours.

There have been limited measurements of air change rates in manufactured homes that can be compared 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). For these homes, 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 6, 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

TABLE 6
Summary of Annual Air Change Rates

Case/Condition	ALBANY				MIAMI				SEATTLE			
	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Effective air change rate (h^{-1})	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Effective air change rate (h^{-1})	Mean air change rate (h^{-1})	Percent of hours < 0.25 h^{-1}	Percent of hours < 0.35 h^{-1}	Effective air change rate (h^{-1})
1/Envelope leakage only	0.22	56	88	0.15	0.10	100	100	0.09	0.20	74	99	0.16
2/Scheduled exhaust fans	0.27	48	77	0.17	0.16	86	91	0.10	0.25	64	85	0.18
3/Forced-air operating on outdoor temperature	0.34	34	53	0.19	0.19	78	90	0.11	0.32	33	71	0.21
4A/Intake on forced-air, operating on outdoor temperature	0.37	32	46	0.19	0.20	73	90	0.11	0.33	30	60	0.22
4B/Intake on forced-air, occupancy schedule	0.59	13	18	0.34	0.51	24	33	0.27	0.55	14	24	0.34
5A/Inlets and whole house exhaust, exhaust schedule	0.41	28	42	0.24	0.26	66	84	0.15	0.38	24	52	0.27
5B/Inlets and whole house exhaust, occupancy schedule	0.50	16	29	0.34	0.34	38	72	0.25	0.47	13	29	0.35
6A/ Whole house exhaust (no inlets), exhaust schedule	0.36	34	52	0.19	0.22	78	86	0.11	0.34	33	71	0.22
6B/ Whole house exhaust (no inlets), occupancy schedule	0.46	21	35	0.28	0.31	52	78	0.20	0.43	12	37	0.29

in this study. 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 in this study 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 1, but the houses were tighter and the temperature differences during the air change rate measurements were about one-half of those in Table 1. 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 (AEC 1996). 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.

Energy

The energy consumption associated with the ventilation approaches is shown in Table 7 and presented graphically in Figure 6, 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 and cooling loads are not converted to primary energy consumption values by assuming a value for the heating and cooling system efficiencies. The fan energy values in Table 7 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, the table 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 in Case #3 in which the forced-air fan operates based on the outdoor temperature.

The energy consumption for the first case includes only heating and cooling loads, but no fan loads. 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. 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 and, of course, the energy consumed by the fans. 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 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 Case #2, 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.

The impact of the first mechanical ventilation option—that is, the outdoor air intake on the forced-air return—depends on the schedule of forced-air fan operation. With the forced-air fan and intake operating based on outdoor air temperature (Case #4A), the increase in energy consumption is not large. 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 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 supplemental ventilation strategies investigated in this study all have the capacity to meet the 0.35 h^{-1} ventilation requirement in the MHCSS. Their actual impact in practice is a function of how often they are operated. On an annual basis, Case #3, which is an important baseline case, has a mean air change rate above the 0.25 h^{-1} reference value for infiltration in Albany and Seattle, but is below that value in Miami. The effective air change rates are even lower. 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 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. The effective

TABLE 7
Summary of Energy Consumption

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 $0.35h^{-1}$								
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

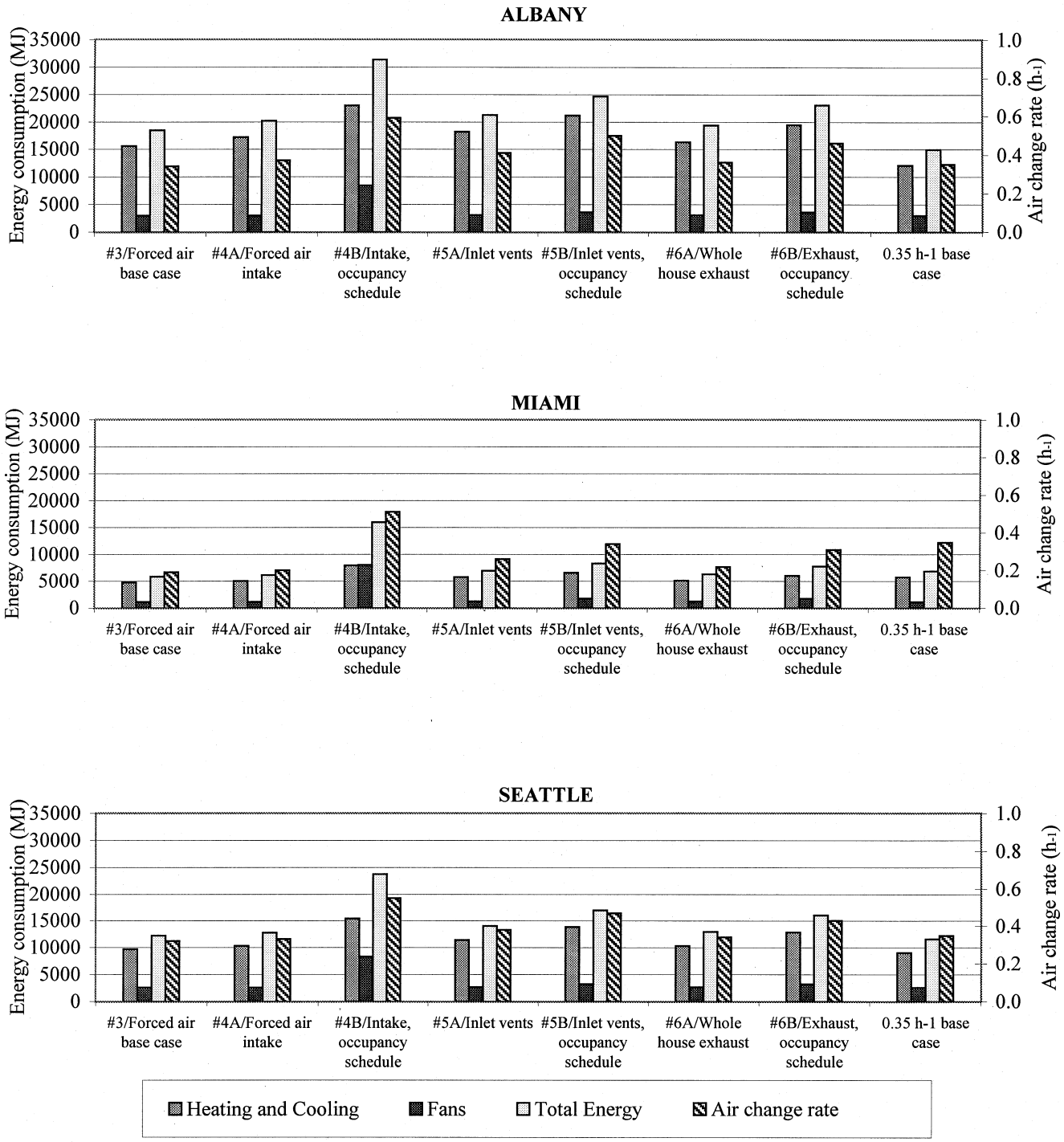

Figure 6 *Summary of energy consumption and air change rates.*

air change rates, however, are below 0.35 h^{-1} for most of the cases.

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.

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 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. Cases with limited operation of the mechanical ventilation system do not exhibit much increase in energy consumption, but they do not raise the ventilation rates much either. However, when the system operates whenever the building is occupied, the energy use increases significantly.

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.

Study Issues

This study was motivated primarily by five issues. 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.

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 effective in raising air change rates and distributing ventilation air throughout the house. The overall impact 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 et al. 1997) and by operating the fan based on a timer or "duty cycler" control. Such a duty cycler would "guarantee" a certain amount of operation per hour and therefore could limit the extent of underventilation under milder weather conditions.

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 in which they could be implemented. Such a system would presumably require a tighter envelope than the level used in this study.

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 mean air change rates are still above the 0.35 h^{-1} requirement in the MHCSS, though the effective rates are not. Again, the overall impact of the whole house exhaust fan depends on the fan operation 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.

Recommendations

While the simulations performed in this study have limitations, there are a number of recommendations that can be made.

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 (1993) to convert a building airtightness value from

a pressurization test to an annual effective air change rate for a given climate.

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, and this merits attention. Related to the issue of system operation time is that of heating and cooling system oversizing. For many of these approaches, the extent of oversizing has a strong impact on the building air change rates. Better matching of equipment sizing to load would tend to increase building air change rates for ventilation approaches that employ the forced-air system for air intake. Better sizing would tend to increase the occurrence of overventilation during periods of high heating and cooling demand. At the same time, more proper sizing could decrease the occurrence of underventilation during mild weather. However, the impact of system sizing is complex, requiring consideration of system airflow rates, duct sizes, and controls, and more detailed examination is needed to fully understand the issue.

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 beneath the house. Instituting design, construction, and commissioning practices that reduce the level of duct leakage are all achievable with existing technology.

As noted earlier, both ASHRAE Standard 62 and the MHCSS do not address the issue of the time period to which the ventilation requirements apply and how to account for variations in ventilation rate. These standards need to account for variation in some way, and effective air change rates are one way to do that.

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, system capacity, and weather. While many of the results are generalizable, it is important that similar analyses be conducted in other buildings and systems 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, duty cyclers, 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 approaches employing continuous exhaust fan operation.

Other research that should be considered includes the simulation of indoor contaminant levels to understand the indoor air quality implications of 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 is important for extending the usefulness of simulation as a means of investigating ventilation and indoor air quality performance issues in manufactured houses.

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