Estimating real-time infiltration for use in residential ventilation control

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Abstract

Minimum outdoor air ventilation rates specified in standards such as ASHRAE Standard 62.2 are generally based on envelope airtightness, building floor area, geographical location, and number of occupants. ASHRAE Standard 62.2 allows for a constant infiltration credit, which reduces the required mechanical ventilation. However, infiltration rates vary based on the weather and system operation. Thus, mechanical systems could potentially operate less if the real-time (RT) infiltration rate were known and used to adjust the mechanical ventilation rate. CONTAM models of two test houses on the campus of the National Institute of Standards and Technology were verified with measurements and used to simulate hourly infiltration rates in three cities. The infiltration rates were passed to a theoretical controller that changed the hourly mechanical ventilation rate to meet the ventilation requirement. Simulated energy use and relative annual occupant exposure for this RT control strategy was compared with ventilating at a constant rate. Implementation of the RT control strategy resulted in annual average energy savings of $66USD across both houses and three cities without increasing the annual occupant exposure compared with ventilating continuously at a constant rate. The authors discuss the advantages and limitations of the proposed real-time ventilation control strategy.

Keywords: CONTAM, residential ventilation, real-time infiltration, energy use, relative exposure factor, ASHRAE Standard 62.2

Introduction

In ASHRAE Standard 62.2-2016[1], the minimum required outdoor ventilation rate is determined based on the number of bedrooms and occupiable floor area. An infiltration credit can be used to reduce the required mechanical ventilation in a house. The credit is based on converting an envelope airtightness value to an effective annual infiltration rate using floor area, house height, and location of the house to capture the climatic conditions. The credit is applied to the entire year, even though the actual infiltration rate will be lower or higher than the credit at any given time[2]. Similarly, the infiltration credit does not account for times of the year when the mechanical ventilation plus the actual infiltration rate exceed the outdoor air ventilation requirement, leading to excessive energy use. Application of real-time ventilation control can help achieve the target total ventilation rate on an hourly basis rather than on an annual average basis. This would balance the goals of maintaining acceptable indoor air quality and reducing energy use by avoiding both under- and over-ventilation.

In studies of control strategies to reduce the energy use of mechanical ventilation, carbon dioxide and relative humidity have been used as the control indicator[3]. However, the dynamic nature of infiltration is often not addressed or quantified. One experimental study shut off mechanical ventilation at the highest indoor-outdoor temperature differences to save about 9 % in heating/cooling energy[4]. It was assumed that infiltration would be the greatest under these conditions, which would compensate for the mechanical ventilation system being off. While greater indoor-outdoor temperature differences do result in higher infiltration rates, the infiltration rate was not measured in this study. It was, however, estimated using empirical equations in Appendix C of ASHRAE Standard 62.2-2016.

Hesaraki and Holmberg[5] implemented an occupant-based ventilation control approach in newly-built Swedish houses, where the ventilation rate was decreased by 27 % during unoccupied periods to save energy. They varied the length of time in which the reduced ventilation was allowed (a range of 4 h to 10 h) and estimated what the potential exposure would be to occupants returning home after a prolonged period of reduced ventilation. Their study showed that a heating energy savings of 16 % could be achieved by operating at the reduced ventilation rate for eight hours (and returning to normal ventilation two hours before occupants returned). This study only accounted for mechanical ventilation (i.e., they assumed no infiltration) when determining the concentration of pollutants during the periods of reduced and normal ventilation. Without accounting for infiltration, their estimates of the concentrations of internally-generated contaminants may be elevated.

Turner and Walker[6] simulated potential energy savings of more than 40 % using their Residential Integrated Ventilation Controller (RIVEC), without compromising long-term or short-term exposures to pollutants with constant emission rates. The RIVEC monitored all the ventilation devices in a house, including the kitchen and bathroom exhausts, and reduced mechanical outdoor ventilation when those exhaust systems were running. The RIVEC also took into consideration times of peak electrical demand and times of the day when outdoor pollutants may be high to reduce outdoor ventilation until a more favourable time. Because the mechanical ventilation fan may be switched-off for up to four hours (i.e., during peak electrical demand), the airflow capacity of the installed fan must be 125 % of what is required by ASHRAE Standard 62.2, per the requirements of the standard for intermittent operation[7]. Turner and Walker[6] used the constant ASHRAE Standard 62.2 infiltration credit in their analysis.

A study by Walker and Less[8] used relative dose and exposure for real-time ventilation control using a model developed by Lawrence Berkeley National Laboratory (LBNL) called REGCAP. This is a mass-balance model that accounts for ventilation, heat transfer, equipment operation, and moisture. A generic pollutant was assumed emitting continuously but that occupancy varied throughout the day. Simulations were performed on a single-story home. As in the previous study, the ventilation airflow rate needed to be (2 to 2.5) times higher than the requirement in ASHRAE Standard 62.2-2016 in order to make-up for fan-off times during unoccupied periods. The fan also needed to be oversized to maintain the annual relative exposure during occupied hours per requirements in the standard for real-time control. Thus, ventilation energy savings were small because the benefits of switching-off ventilation during unoccupied hours was offset by elevated ventilation during occupied hours. Infiltration was calculated in one of two ways: (1) an annual mean effective rate which is constant for the year and (2) using the Alberta Air Infiltration Model (AIM-2)[9, 10]. Relatively small savings in heating, ventilating, and air conditioning (HVAC) energy were also reported by Clark et al.[11] due to the increase in fan operation to make-up for fan-off times. They used CONTAM-EnergyPlus coupled models[12] for their analyses.

Infiltration is dynamic, varying in time by 5-to-1, or more, with ventilation system operation, weather, and indoor conditions[13]. Despite this variability, the dynamics are often simplified in ventilation standards and ventilation control strategies. For example, a single infiltration rate is often used even though it may not be appropriate when determining whether the total outdoor air ventilation rate complies with a standard or local code requirements throughout the year. Currently, the most accurate way to determine real-time infiltration rates into a building is through tracer gas testing dilution testing[14]. Infiltration rates determined by tracer gas dilution tests are often averaged over a period of several hours and are only applicable to the weather conditions under which the tests are performed. Tracer gas dilution tests are not currently practical, however, as a means of real-time ventilation control because measurements are taken over a period of several hours and the installation and maintenance of such a system can be costly.

Study objective

The objective of this work was to determine how effectively real-time (RT) estimates of infiltration could be used to control residential mechanical ventilation while saving energy and maintaining IAQ (indoor air quality) according to ASHRAE Standard 62.2-2016. The authors propose using airflow simulations to predict RT infiltration rates so that the operation of the mechanical ventilation fan could be reduced or stopped if the RT infiltration is greater than the baseline infiltration credit. Comparisons are made among a constant and an RT ventilation control strategy in terms of metrics that describe ventilation performance, exposure, and energy use.

Infiltration estimation methods

This section describes the infiltration estimation methods used in this study, the infiltration credit in ASHRAE Standard 62.2-2016 and the CONTAM models used to simulate infiltration for RT control.

Infiltration credit

In ASHRAE Standard 62.2-2016[1], the minimum required outdoor ventilation rate ($Q_{\mathrm{tot,62.2}}$) is determined by Equation (1):

$$Q_{\mathrm{tot,62.2}}\ (L/\mathrm{s}) = 0.15 \times A_{\mathrm{floor}} + 3.5 \times (N_{\mathrm{br}} + 1)\ , \qquad (1)$$

where A_{floor} is the dwelling floor area (m^2) and N_{br} is the number of bedrooms. The standard allows the use of an infiltration credit so that the required rate of mechanical ventilation – which determines the minimum size of the fan and ducts that are installed – can be less than $Q_{\text{tot,62.2}}$. The infiltration credit can be determined in various ways. The normalized leakage (*NL*) method was used in this study by Equation (2):

$$NL = 1000 \times \frac{ELA}{A_{\text{floor}}} \times \left[\frac{H}{H_r}\right]^{z}, \tag{2}$$

where *ELA* is the effective leakage area at 4 Pa measured with a fan pressurization or blower door test (m^2)[15], *H* is the vertical distance between the lowest and highest above-grade points within the dwelling's pressure boundary (m), H_r is the reference height (2.5 m), and z is an exponent to convert *ELA* to an effective annual infiltration rate ($z = 0.4$). All of these parameters are defined in more detail in Standard 62.2. The infiltration credit ($Q_{\text{inf,62.2}}$) is defined by Equation (3):

$$Q_{\text{inf,62.2}}\ (\text{L/s}) = \frac{NL \times wsf \times A_{\text{floor}}}{1.44}, \tag{3}$$

where *wsf* is the weather shielding factor. This factor is based on geographical location (i.e., to capture the climatic conditions) and is assigned as specified in Normative Appendix B of ASHRAE Standard 62.2-2016. Thus, for a single-family detached house, Equation (4) defines the required mechanical fan flow rate ($Q_{\text{fan,62.2}}$):

$$Q_{\text{fan,62.2}}\ (\text{L/s}) = Q_{\text{tot,62.2}} - Q_{\text{inf,62.2}}\ . \tag{4}$$

Real-time estimate using CONTAM

Infiltration rates in homes change with time depending on weather, indoor conditions and equipment operation. In this study, real-time infiltration rates are estimated using the multizone airflow simulation software, CONTAM, developed at the National Institute of Standards and Technology (NIST)[16]. CONTAM is particularly useful for whole-building simulation because it fully captures the airflow physics related to building airflow and is computationally efficient. CONTAM also has the advantage in that it can model the impacts of HVAC system operation on infiltration and interzonal airflows. CONTAM has been validated in terms of program integrity[17], and comparisons with laboratory experiments[17] and field studies[17-20].

Methods

The objective of this work was to determine how effectively RT estimates of infiltration could be used to control residential mechanical ventilation while saving energy and maintaining IAQ according to the standard. The authors used CONTAM to predict RT infiltration rates in two test houses on the NIST campus and then used those rates to emulate RT mechanical ventilation system controllers. The two test houses, their ventilation requirements, and the measurements of air change rates, pressure differences and other parameters are described in *Test houses, ventilation requirements, and measurements*. Detailed CONTAM models of the test houses are described which include every interior zone and closely match the actual floor plans. The tracer gas decay tests used to verify the predictive ability of the detailed CONTAM models was described next. Finally in this section, the RT ventilation control strategy that was investigated was described along with the metrics used to evaluate the performance of the control strategy.

Test houses, ventilation requirements, and measurements

This section describes the two test houses on the NIST campus in Gaithersburg, MD, USA, their ventilation requirements, and the measurements that were conducted in them.

IAQ Manufactured House. The IAQ Manufactured House (MH) is a one-story, double-wide, manufactured house built on the NIST campus in 2002 (Figure 1). It has three bedrooms, two bathrooms, and an open space containing a living room, dining room, family room, and kitchen. The MH is conditioned with a central, forced-air 22 kW gas furnace and 15 kW electric air conditioner. The supply air distribution ductwork is in an insulated portion of the crawlspace (i.e., belly) that spans the entire floor area of the house (Figure 1). Based on prior measurements, air within the supply ducts leaks into the belly[21], such that when the space conditioning system is on, the conditioned areas of the MH are slightly depressurized.

(a) (b)

Figure 1. (a) Photograph and (b) schematic cross-section of MH showing crawlspace and belly.

The MH has two bath exhausts, a kitchen exhaust, and a whole-house ventilation exhaust fan. These four exhausts were turned off during the testing. The temperature of the house was measured and the operation of the central heating and cooling system, including its indoor blower, were controlled by a commercially available thermostat. The front and back doors of the attached garage were open during the testing to reduce the garage's impact on the pressure boundary of the house. Table 1 summarizes the characteristics of the MH. Additional details of the MH can be found in Nabinger and Persily[22] and Nabinger and Persily[21].

Table 1. Characteristics of MH and NZERTF

Building characteristics	MH	NZERTF
Year of construction	2002	2012
Floor area (m^2)	140	245 (habitable area) 490 (all floors) [a]
Building volume (m^3)	357	1301 (all floors)
Stories	1	3 above ground, includes attic
Height (m)	2.5	6.3
Exterior surface area, above grade (m^2)	301	367
ELA at 4 Pa (cm^2) (from blower door test)	663	137
Heating/Cooling system	22 kW gas furnace 15 kW electric air conditioner	Air-to-air heat pump 7.6 kW cooling capacity 7.8 kW heating capacity
Mechanical ventilation (Design value, L/s)	Components in place, but disabled during this study	HRV, 38[b]

a) The habitable area of the NZERTF is used to calculate the infiltration credit in ASHRAE Standard 62.2-2016. The total area of the house is used to calculate *NL*.
b) The mechanical ventilation of 38 L/s is based on ASHRAE Standard 62.2-2010, the standard in effect when the building was designed.

Net-Zero Energy Residential Test Facility. The Net-Zero Energy Residential Test Facility (NZERTF) at NIST was built in 2012 to support the development and adoption of cost-effective net-zero energy designs, technologies, and construction methods (Figure 2). It is two-story, and has four bedrooms, three bathrooms, and an open living room, dining room, and kitchen space. The NZERTF also has a basement and attic, both located within the conditioned space because the thermal and air-moisture barriers encompass the basement walls and attic roof. Transfer grilles link the living spaces to these two zones. The central heating and cooling system includes an air-to-air heat pump, which delivers air to the basement, first and second floors. The heat pump has a cooling capacity of 7.6 kW and a heating capacity of 7.8 kW. The indoor unit is in the basement and ductwork runs along the basement ceiling. A balanced heat recovery ventilator (HRV) is installed in the basement and has its own dedicated ductwork. The HRV supplies 47 L/s of outdoor air to the house, with supplies on the first floor (in the kitchen/dining area) and in each of the three second-floor bedrooms. Air from the first-floor bathroom and both second-floor bathrooms are exhausted to the HRV before being exhausted. During the testing, the HRV operated on an intermittent schedule (40 min on, 20 min off) to deliver an hourly average of 38 L/s. The house also has a kitchen exhaust and a clothes dryer exhaust, both of which were turned off. The temperature of the house was measured and the operation of the space conditioning system, including its recirculating air distribution fan, were controlled by a commercially available thermostat. Table 1 summarizes the characteristics of the NZERTF. Additional design, construction, equipment, and energy performance details for the NZERTF can be found in Pettit et al.[23], Fanney et al.[24] and Fanney et al.[25]

Figure 2. Photograph of NZERTF.

Ventilation requirements. The ASHRAE Standard 62.2-2016 requirements for both houses, $Q_{tot,62.2}$, $Q_{inf,62.2}$ and $Q_{fan,62.2}$, are summarized in Table 2 for Atlanta (mixed-humid), Baltimore (mixed-humid), and Chicago (cold). These cities were chosen to represent a range of climatic conditions. Due to its larger size, the NZERTF requires 54 L/s of total outdoor air per ASHRAE Standard 62.2-2016, compared to the MH, which only requires 35 L/s.

Table 2. ASHRAE Standard 62.2-2016 ventilation requirements and infiltration credit for MH and NZERTF in three cities

MH			
Value/City	Atlanta	Baltimore	Chicago
$Q_{tot,62.2}$ (L/s)	35		
$Q_{inf,62.2}$ (L/s)	21	23	23[a]
$Q_{fan,62.2}$ (L/s)	14	12	12
NZERTF			
Value/City	Atlanta	Baltimore	Chicago
$Q_{tot,62.2}$ (L/s)	54		
$Q_{inf,62.2}$ (L/s)	6	7	8
$Q_{fan,62.2}$ (L/s)	48	47	46

a. Using Eq. (3), the infiltration credit in Chicago for the MH was calculated as 28 L/s but due to the 2/3s limitation in the standard, its credit is set to 23 L/s.

Blower door tests were performed on both houses in accordance to ASTM E779-10[15]. The measured ELA of the MH was 663 $cm^2 \pm 84$ cm^2 and that of the NZERTF was 137 $cm^2 \pm 7$ cm^2, which are 95 % confidence intervals. As shown in Table 2, because the NZERTF is tighter than the MH, the $Q_{fan,62.2}$ required (Equation (4)) at the NZERTF is larger than what is required at the MH. The infiltration credit was determined using Equation (2) and Equation (3), where the *wsf* for Atlanta, Baltimore, and Chicago are respectively 0.46, 0.50, and 0.60. Note that ASHRAE Standard 62.2-2016 sets a limit on the infiltration credit so that it cannot exceed 2/3 of $Q_{tot,62.2}$ for new construction (i.e., 36 L/s for the NZERTF and 23 L/s for the MH). For existing homes, Appendix A of ASHRAE Standard 62.2-2016 provides guidance on determining the required mechanical ventilation. For the purposes of this study, both test houses were assumed as new construction. Using Equation (3), the infiltration credit in Chicago for the MH was calculated as 28 L/s, but was set to 23 L/s due to the 2/3's limitation in the standard.

Measurements. Tracer gas decay tests were conducted to measure the whole building air change rate (envelope infiltration plus any mechanical outdoor air intake) in each house under varying conditions, including with the air distribution fans always-on and always-off. Exhaust fans were turned off during the testing. The tracer gas decay tests complied with ASTM E741-11[14], with sulphur hexafluoride (SF_6) automatically injected at specified time intervals into the return stream of the space conditioning system in the MH and into the HRV supply duct in the NZERTF. In both houses, this injection approach led the tracer gas to be delivered and mixed throughout the house. The tracer gas was sampled in six locations in each house ( ) at 30 s intervals with the Lumasense INNOVA 1412 photoacoustic infrared sampler. This instrument has a measurement range of 3.6 mg/m^3 to 18.2 mg/m^3 (0.6 ppm_v to 3 ppm_v), and the manufacturer's reported accuracy is 5 % and its rated repeatability is within 1 %.

Differential pressure (ΔP), indoor and outdoor relative humidity (*RH*), indoor and outdoor temperature (T_{in} and T_{out}, respectively), and wind speed (W_s) were collected during the tests at one-minute intervals (Figure 3). At both test houses, T_{in}, T_{out}, and *RH* (indoor and outdoor) were measured using humidity and temperature probes. ΔP was measured using differential pressure transducers. In the MH, ΔP was measured across each exterior wall and on the wall between the house and open garage at heights of 0.3 m and 1.8 m from the floor ( ). In the NZERTF, ΔP was measured across each exterior wall at heights of 0.8 m and 4.3 m from the ground level (i.e., middle of the wall on each floor). W_s was measured at the MH using a Climatronics sonic anemometer placed 4 m from the south wall and 9 m above the ground. W_s was measured at the NZERTF using an ultrasonic wind sensor located 90 cm above the roof line. The measurement range and accuracy of the sensors are listed in Table 3. Different products were used at the two test houses so that the specifications for measurement range and accuracy in Table 3 are different.

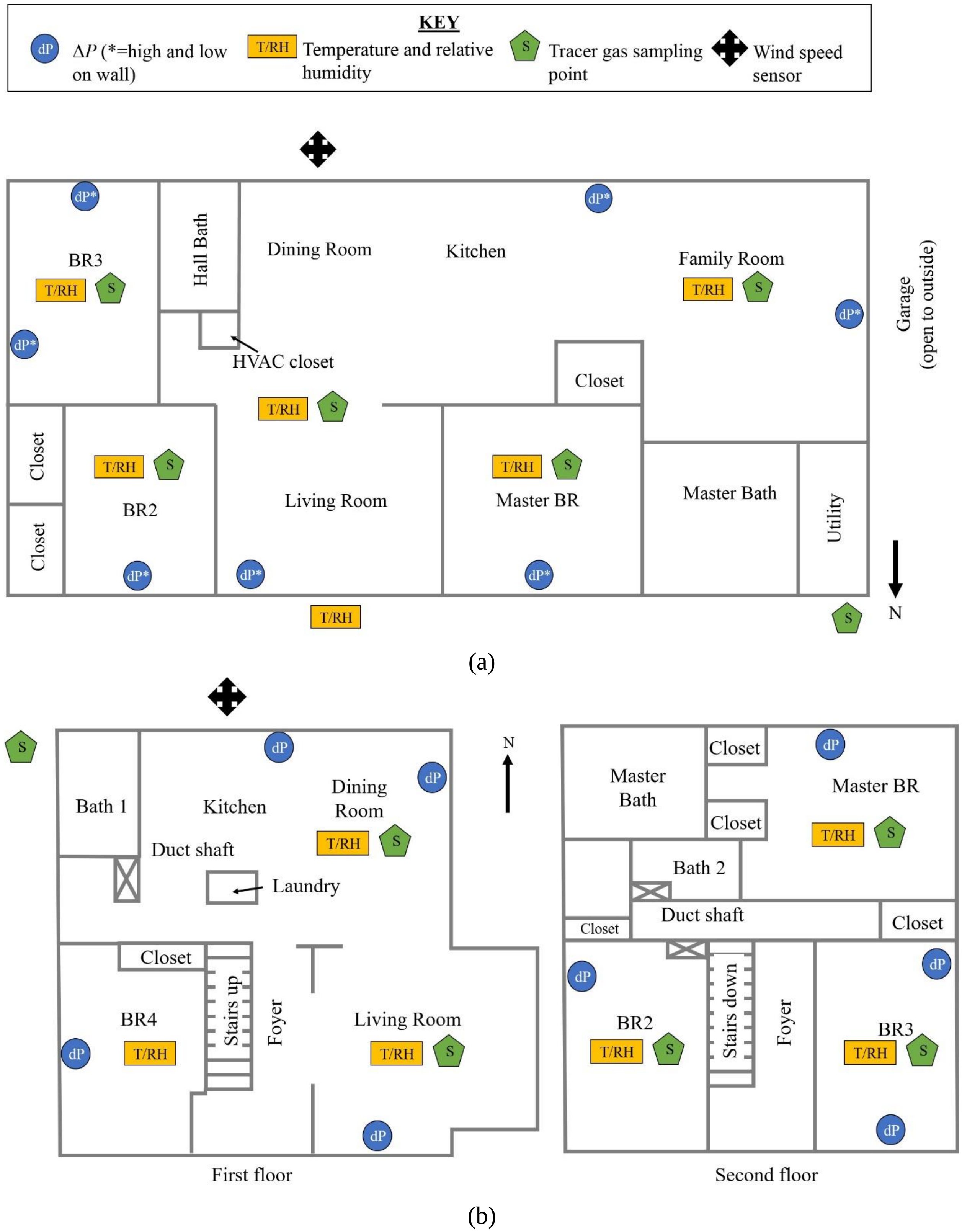


Figure 3. Location of sensors in (a) MH and (b) NZERTF

Table 3. Data collected and specifications of instruments

Data collected	**Measurement range**	**Accuracy**
Differential pressure	± 25 Pa	± 1 % full scale
Indoor dry bulb temperature and relative humidity	MH: -39.2 °C to 60 °C 0.8 % RH to 100 % RH	±0.2 °C ±2 % RH (0 % to 90 % RH) ±3 % RH (90 % to 100 % RH)
	NZERTF: -20 °C to 70 °C 1 % RH to 95 % RH	±0.21 °C (0 °C to 50 °C) ±2.5 % RH (10 % to 90 % RH) ±5 % RH (<10 % and >90 % RH)
Outdoor dry bulb temperature and relative humidity	MH: -39.2 °C to 60 °C 0.8 % RH to 100 % RH	±0.2 °C ±2 % RH (0 % to 90 % RH) ±3 % RH (90 % to 100 % RH)
	NZERTF: -40 °C to 70 °C 0 % RH to 100 % RH	±0.21 °C (0 °C to 50 °C) ±2.5 % RH (10 % to 90 % RH) ±5 % RH (<10 % and >90 % RH)
Outdoor wind speed	MH: 0 m/s to 65 m/s 0 ° to 360 °	± 0.5 m/s or 5 % ± 5 ° at W_s > 2.2 m/s
	NZERTF: 0 m/s to 60 m/s 0 ° to 360 °	± 3 % at 10 m/s ± 3 °

CONTAM models

Hourly CONTAM simulations were run for each test house using annual Typical Meteorological Year 3 (TMY3) weather files for Atlanta, Baltimore, and Chicago[26]. The detailed CONTAM models of the test houses include every room, including closets and duct shafts in the case of the NZERTF. A detailed description of the MH model, as well as its validation, can be found in Nabinger and Persily[22] and Nabinger and Persily[21]. Validation of a coupled CONTAM-EnergyPlus model of the NZERTF is available in Ng et al.[27]

In CONTAM, ventilation systems can be modelled either as "simple" or ducted air handling systems (AHS). In the MH, the central heating and cooling system was modelled in CONTAM as a ducted air handling system to capture airflow and contaminant transport between the belly and the habitable areas of the MH. Figure 4 shows the ductwork at the MH in the belly, which supplies the habitable areas through floor vents. The recirculating fan (labelled as "HVAC fan" on the first floor) was modelled using the Fan Performance fan type in CONTAM, with a cut-off ratio of 0.1, meaning the simulated fan turns off if the calculated airflow is less than 10 % of the maximum airflow. The fan then becomes a simple orifice with an area of 0.02 m^2 to allow airflow through the duct system, to and from the belly, when the fan is off.

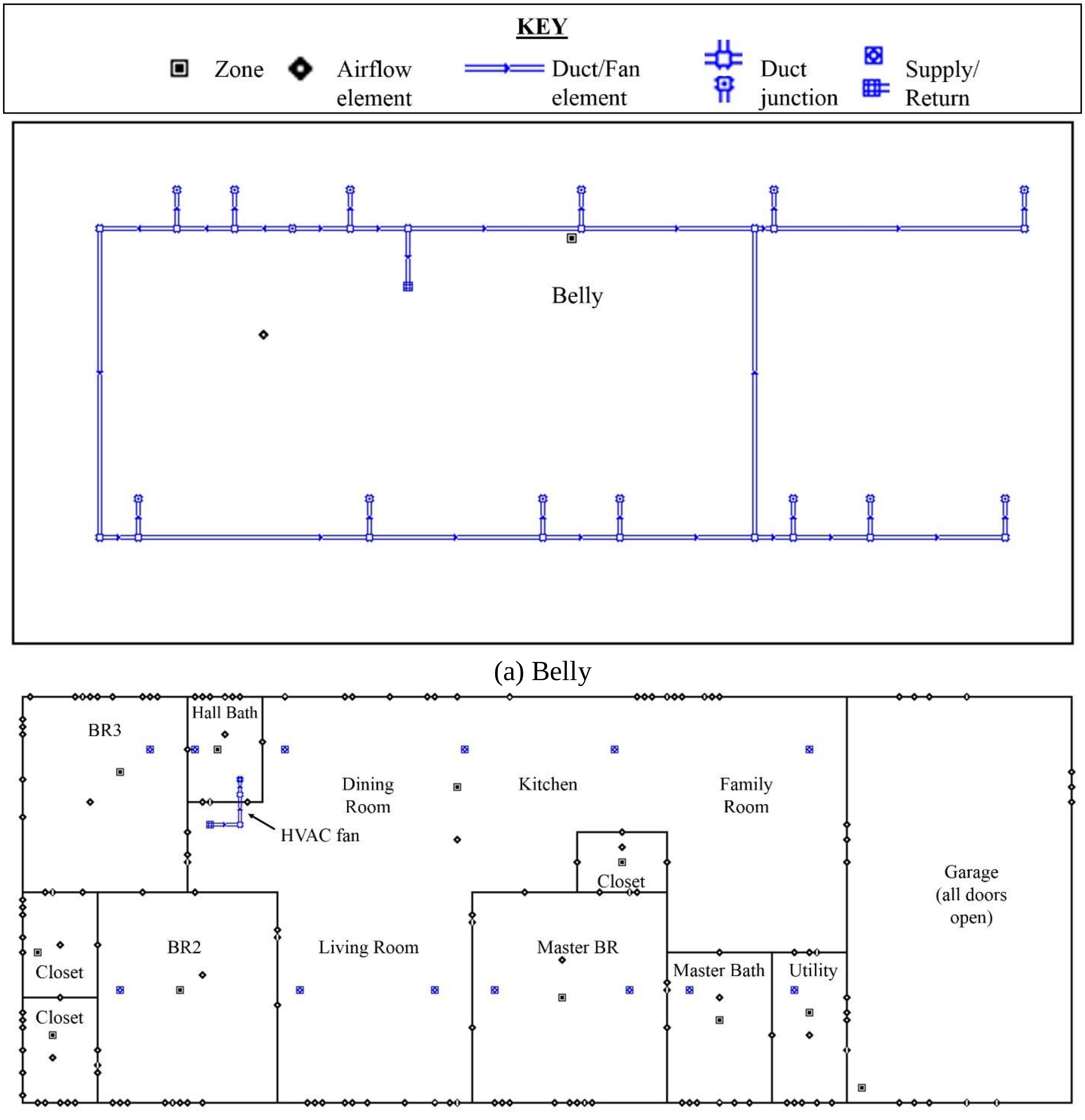


Figure 4. CONTAM model of MH (crawlspace and attic levels not shown).

Because the HVAC fan affects the depressurization of the MH, it was important to capture its operation in the CONTAM model. This accounting was accomplished by simulating control of a hypothetical heating and cooling system that is uniquely sized based on the local heating design temperature (HDT) and cooling design temperature (CDT), and a chosen indoor setpoint of 23.5 °C, and a representative thermostat deadband of ± 2 °C[28]. HDT and CDT are the temperatures that are exceeded 1 % of the hours in a typical weather year according to ASHRAE[29]. The system was assumed to operate two-thirds of the hour when the outside temperature was equal to its HDT or CDT. For temperatures between 23.5 °C and HDT, and between 23.5 °C and CDT, the runtime fraction varied linearly between 0.0 and 0.67. The fan runtime fraction was not allowed to exceed 1.0 at any temperature. Also, the fan runtime fraction was set to 0.0 when the outdoor temperature was between 21.5 °C and 25.5 °C. An example of the controller runtime fraction as a function of outdoor temperature is given in Figure 5 for Baltimore, MD, USA (HDT = -7.6 °C and CDT = 33.0 °C). The other two U. S. cities included in this study and their respective HDT and CDT were: Atlanta, GA where HDT = -3.0 °C and CDT = 33.1 °C, and Chicago, IL where HDT = -15.3 °C and CDT = 31.4 °C.

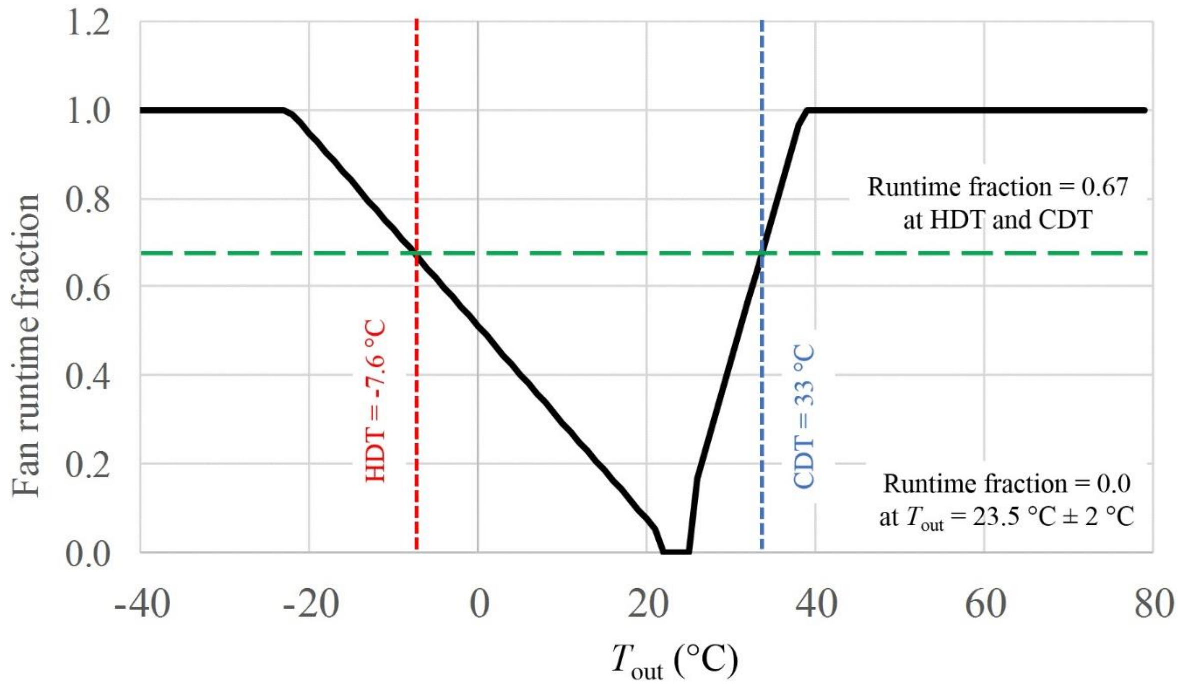


Figure 5. Simulated fan runtime fraction controller at MH in Baltimore.

In the NZERTF, both the heating/cooling and HRV systems are modelled as simple AHS so only supplies and returns are included in the CONTAM model (Figure 6). The only ductwork included in the CONTAM model of the NZERTF are the kitchen and dryer exhausts on the first floor. The airflow rate at each supply diffuser and return outlet were specified in the model based on airflow rates measured using a low-flow capture hood. Because any duct leakage at the NZERTF is within the conditioned space, fan operation does not impact indoor-outdoor air pressures or infiltration rates. Therefore, a fan runtime controller was not implemented in the CONTAM model, as was done for the MH. Instead, the central heat pump fan was simulated as on continuously. Further, the HRV is a balanced system so its operation does not affect the pressurization of the NZERTF.

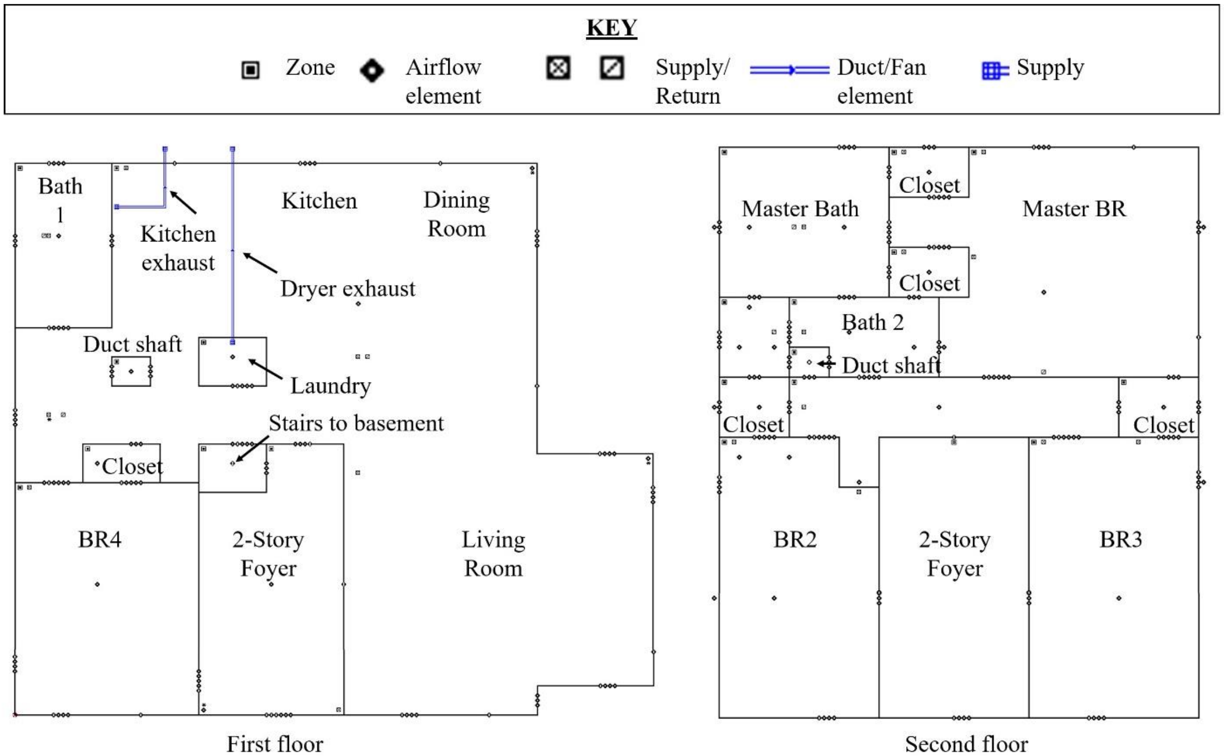


Figure 6. CONTAM model of NZERTF (basement and attic levels not shown).

There are three leakage sites per section of wall in each model to better capture the stack effect. One leak is placed one-quarter up the height of the wall, another is placed at the middle of the wall, and the third is placed three-quarters up the height of the wall. All the exterior windows and doors were closed and all interior doors were open in the simulations. Results from simulations of the detailed MH and NZERTF models are compared with measurements in *Results*.

CONTAM verification tests

Tracer gas decay measurements of whole building air change rates were used to verify the detailed CONTAM models of the test houses. Fourteen decay tests were performed in the MH in October 2017 (8 with the system heating/cooling system on, 6 with the heating/cooling system off). At the NZERTF, eight tests were performed in December 2017 (6 with the heating/cooling system and HRV on, 2 with the heating/cooling system and HRV off). The outdoor air change was calculated using the procedure in ASTM E741-11[14]. Table 4 and Table 5 show the average temperature difference (T_{out}-T_{in}) and wind speed (W_s) during the tests. On average, the tests at the MH were conducted in milder weather than the tests conducted at the NZERTF.

Table 4. Measured and predicted CONTAM total outdoor airflow rates for MH

System status	Test number	Average T_{out}-T_{in} (°C)	Average W_s (m/s)	Measured total outdoor airflow rate (L/s)	95 % confidence interval of measured value (L/s)	Predicted total outdoor airflow rate (L/s)	Percentage difference (%)
On	1	3	1	21	0.3	14	33
On	2	-7	1	15	0.1	15	0
On	3	4	4	30	0.2	25	17
On	4	-1	3	23	0.2	21	9
On	5	1	1	17	0.2	13	24
On	6	-2	4	24	0.1	23	4
On	7	4	2	23	0.3	19	17
On	8	-5	0	12	0.1	13	-8
On	**Average tests 1 to 8**	**0**	**2**	**21**	**0.2**	**18**	**14**
Off	9	-2	1	9	0.1	8	11
Off	10	-3	3	14	0.1	18	-29
Off	11	-6	4	19	0.4	22	-16
Off	12	-7	2	23	0.2	18	22
Off	13	-8	2	21	0.2	17	19
Off	14	-7	1	20	0.1	15	25
Off	**Average tests 9 to 14**	**-5**	**2**	**18**	**0.2**	**16**	**20**

Table 5. Measured and predicted CONTAM total outdoor airflow rates for NZERTF

System status	Test number	Average T_{out}-T_{in} (°C)	Average W_s (m/s)	Measured total outdoor airflow rate (L/s)	95 % confidence interval of measured value (L/s)	Predicted total outdoor airflow rate (L/s)	Percentage difference (%)
On	1	-21	3	77	3	59	23
On	2	-22	2	68	3	54	21
On	3	-24	4	66	3	59	11
On	4	-31	5	42[a]	2	41	2
On	5	-28	3	63	2	58	8
On	6	-30	1	43[a]	1	40	7
On	**Average of tests 1 to 6**	**-26**	**3**	**60**	**2**	**52**	**12**
Off	7	-25	2	17	0.3	18	-6
Off	8	-31	1	27	0.4	19	30
Off	**Average of tests 7 and 8**	**-28**	**2**	**22**	**0.5**	**18**	**18**

Note a: The HRV was in recirculation mode during part of these tests to prevent frosting of the heat exchanger.

Table 4 shows the measured and predicted air change rates for the MH, where the average measured total outdoor airflow rate (mechanical ventilation plus infiltration) with the heating/cooling system on was 21 L/s ± 0.2 L/s (95 % confidence interval), and the average predicted rate was 18 L/s (average absolute difference of 14 %). With the heating/cooling system off, the measured total outdoor airflow rate was 18 L/s ± 0.2 L/s (95 % confidence interval), and the average predicted rate was 16 L/s (average absolute difference of 22 %). As noted earlier, the mechanical ventilation system in the MH was off, so the space conditioning system on and off air change rates were expected to be similar, except for the additional infiltration and duct leakage into the belly.

Table 5 shows the measured and predicted air change rates for the NZERTF. The average measured total outdoor airflow rate with the system on was 60 L/s ± 2 L/s, and the average predicted rate was 52 L/s (average absolute difference of 12 %). Both the measured and predicted rates included the outdoor mechanical ventilation provided by the HRV. There was one 4 h period during Test #4, and one 2 h period during Test #6, when the HRV recirculation mode was activated because the outside temperature was below -10 ° C. The average measured total outdoor airflow rate with the system off was 22 L/s ± 0.5 L/s, and the average predicted rate was 18 L/s (average absolute difference of 18 %).

In summary, the absolute differences between the measured and predicted rates ranged between 4 L/s and 7 L/s at the MH, and between 1 L/s and 18 L/s at the NZERTF, including both the system-on and system-off values. These differences were on average 15 % of the average measured outdoor airflow rate at each test house. These rates translate to an average of 0.05 h^{-1} difference at the MH, and 0.03 h^{-1} difference at the NZERTF. As noted in Ng, et al.[30], at rates this low, the measurement accuracy of the tracer gas decay measurements needs to be considered. Further, ASTM E741-11[14] states that following its procedure, measurements of gas concentrations will provide air change rate values within 10 % of the true value. Given the low air change rates and stated uncertainty in ASTM E741-11, the results of the CONTAM models were considered to be within reasonable accuracy. Comparisons of measured and predicted infiltration rates in the literature have also yielded differences with similar magnitude[31-33].

Ventilation control strategies

This study involved the evaluation of one continuous and one RT control strategy:

1. $\underline{\text{Vent}_{\text{cont}}}$: This strategy continuously supplied each test house with $Q_{\text{fan,62.2}}$ as defined by Equation (4). The hourly infiltration rates for this strategy are predicted by CONTAM model.
2. $\underline{\text{Vent}_{\text{RT}}}$: This strategy implemented RT ventilation control by supplying each test house with $Q_{\text{fan,RT}}$ that varied with each timestep depending on the infiltration rate calculated by CONTAM, such that $Q_{\text{fan,RT}} = Q_{\text{tot,62.2}} - Q_{\text{inf,RT}}$. If $Q_{\text{inf,RT}} > Q_{\text{tot,62.2}}$, then $Q_{\text{fan,RT}} = 0$

If $Q_{\text{inf,RT}} > Q_{\text{tot,62.2}}$, then $Q_{\text{fan,RT}} = 0$. Otherwise, Equation (4) is used to calculate $Q_{\text{fan,RT}}$. To meet $Q_{\text{tot,62.2}}$ when infiltration is close to zero, the maximum airflow rate of the RT fan was equal to

$Q_{tot,62.2}$. When the simulated infiltration was greater than the infiltration credit, the total outdoor ventilation ($Q_{inf,RT}$ + $Q_{fan,RT}$) for the Vent$_{cont}$ strategy would be greater than $Q_{tot,62.2}$. There can also be times of the year when $Q_{inf,RT} > Q_{tot,62.2}$.

Performance metrics for comparing ventilation control strategies

To compare the performance of the ventilation control strategies, the following five metrics were used: predicted infiltration rates, average Q_{fan} flow rate, ventilation hours, relative exposure factor, and energy impacts of ventilation.

Predicted infiltration rates. The annual average predicted infiltration rate for the two test houses in three cities is compared with the infiltration credit in ASHRAE 62.2-2016.

Average Q_{fan} flow rate. For the RT ventilation control strategy, the infiltration rate predicted by CONTAM was subtracted from the total ventilation requirement using Equation (4) to obtain $Q_{fan,RT}$ at every hour of the year. The annual average $Q_{fan,RT}$, which include the times when the fan was off (i.e., $Q_{fan,RT} = 0$), was calculated and compared with $Q_{fan,62.2}$.

Ventilation hours. At each hourly timestep, whether and how $Q_{tot,62.2}$ was met was evaluated. Outdoor air ventilation was either met by infiltration-alone or by the sum of Q_{fan} and Q_{inf}. Any timesteps where the sum of Q_{fan} and Q_{inf} was less than $Q_{tot,62.2}$ were considered unmet ventilation hours.

Relative exposure factor. ASHRAE Standard 62.2-2016 allows RT ventilation control only if the annual average relative exposure factor (R_{avg}) during occupied periods does not exceed 1.0 when compared with the exposure that would result from ventilating continuously at $Q_{tot,62.2}$. Further, the relative exposure at any time step (R_i) must not be greater than 5.0 for time steps not to exceed one hour. Based on ASHRAE Standard 62.2-2016 Normative Appendix C, and assuming a generic constant contaminant source that is spatially uniform, R_i is defined by Equation (5):

$$R_i = {Q_{\text{tot},62.2}}/{Q_{\text{tot},i}} + \left(R_{i-1} - {Q_{\text{tot},62.2}}/{Q_{\text{tot},i}}\right) \times e^{-Q_{\text{tot},i}/V}, \tag{5}$$

where i is the ith timestep, and V is conditioned volume (m^3) of the home. In this study, relative exposure at every hour of the year was evaluated. Normative Appendix C to Standard 62.2 provides several methods for determining Q_{inf}, including converting NL from Equation (2) to a constant infiltration rate using Equation (3). The user may also calculate the stack- and wind-driven flows using empirical equations for every time step, given the weather and T_{in} at every time step, for an infiltration rate at every timestep. The standard states that these infiltration estimation methods can only be used if a blower door test has been performed. Otherwise, Q_{inf} is set to zero. However, the first of these methods for determining Q_{inf} (using NL with Equation (3)) assumes that infiltration is constant throughout the year. The empirical equations used in the second method vary infiltration with weather but may not fully capture wind effects on infiltration because the wind coefficients used are averaged over the entire building and not dependent on the specific wind direction. In this study, infiltration is predicted using detailed CONTAM models for the specific buildings being studied, which more accurately accounts for variations in indoor and outdoor conditions. Note that in Equation (5), the occupant exposure is relative to ventilating at the total ventilation rate required by ASHRAE Standard 62.2-2016.

Energy impacts of ventilation. The energy impact of ventilation was evaluated in this study in terms of: (1) the energy to operate the mechanical ventilation (MV) fan and (2) the energy to condition the MV air. The reason that the energy required to condition the infiltration air was not considered is because the two ventilation strategies studied both used infiltration rates from CONTAM. Thus, the energy required to condition infiltration air would be the same in both ventilation strategies and no additional costs or savings could be reported.

The energy was converted to cost using an assumed cost of electricity per kilowatt-hour (kWh). Hourly values of the energy to operate the MV fan, E_{MV}, were calculated using Equation (6), assuming a ventilation fan efficiency (e_{MV}) of 0.5 L/(s•W). This is the average efficiency of HRVs in the Heating Ventilating Institute (HVI) equipment database for fans with the capacity to supply $Q_{\mathrm{tot,62.2}}$ in both houses [34]:

$$E_{\mathrm{MV}}[\mathrm{kWh}] = {}^{Q_{\mathrm{fan}}}\!/_{e_{\mathrm{MV}}} \times \frac{1}{1000} \times 1\ \mathrm{h}, \qquad (6)$$

where Q_{fan} is the hourly fan airflow rate and 1/1000 is a conversion factor from W to kW.

Hourly values of the energy required to condition the MV air were determined by calculating the sensible heat of the air being delivered to the test house at T_{out} and conditioned to the indoor setpoint (T_{in}) of 23.5°C. The effects of latent heat and heat recovery on the heating/cooling load of the test houses was not considered in this study. The sensible heat associated with the MV air, $q_{\mathrm{sens,MV}}$, is given by Equation (7):

$$q_{\mathrm{sens,MV}}[\mathrm{kWh}] = Q_{\mathrm{fan}} \times \rho \times c_{p,\mathrm{air}} \times (T_{\mathrm{in}} - T_{\mathrm{out}}) \times \frac{1}{1000} \times 1\ \mathrm{h}, \qquad (7)$$

where $c_{\mathrm{p,air}}$ is the specific heat of air (1.003 kJ/kg•°C), ρ is the density of air (1.2 kg/m^3), and 1/1000 is a conversion factor from L/s to m^3/s.

As a reminder, the MH is conditioned by an electric air-conditioner and heated by a gas furnace. The NZERTF is heated and cooled by an air-to-air heat pump. For simplicity in comparing the energy impacts in the two test houses, the heating/cooling systems in both test houses were both assumed to be heat pumps with the same performance. The hourly energy required for the heat pump to condition the MV outdoor air ($E_{\mathrm{hp,MV}}$) was calculated assuming a heat pump (hp) coefficient of performance (COP) of 3.6 kW/kW. This value is the average COP of heat pumps in the Air-Conditioning, Heating and Refrigeration Institute (AHRI) database with heating and cooling capacities between 7.3 kW and 15 kW, which covers the capacities of the equipment in the test houses[35]. The energy required by the heat pump to condition MV air was thus expressed in Equation (8) as:

$$E_{\mathrm{hp,MV}}[\mathrm{kWh}] = {}^{q_{\mathrm{sens,MV}}}/_{\mathrm{COP}} \times 1\ \mathrm{h} \tag{8}$$

The hourly cost to operate the MV fan and condition the outdoor air were calculated by multiplying the energy use (kWh) by the national average cost of electricity for 2018 to 2019 as determined by the U. S. Energy Information Administration. This average value is \$0.13/kWh[36]. The energy savings predicted for the RT ventilation control strategy was compared with the $\mathrm{Vent_{cont}}$ strategy.

Results

Analyses of the RT ventilation control strategy included comparisons of five performance metrics: predicted infiltration rates, average Q_{fan} flow rate, ventilation hours, relative exposure factor, and energy impacts of ventilation.

Predicted infiltration rates

Table 6 show the average predicted infiltration rate from CONTAM, as well as the ASHRAE Standard 62.2-2016 infiltration credit ($Q_{\mathrm{inf,62.2}}$) for each test house and city studied. The predicted annual infiltration rate was greater than the ASHRAE Standard 62.2-2016 infiltration credit in both test houses. The largest difference between the predicted infiltration rate and $Q_{\mathrm{inf,62.2}}$ was in Chicago, where CONTAM estimated an average infiltration rate that is a little more than double $Q_{\mathrm{inf,62.2}}$ at the MH and average 2.5 times more than $Q_{\mathrm{inf,62.2}}$ at NZERTF. Table 6 also shows the standard deviation in the annual infiltration rates, which are relatively large with regards to the average value. This indicates that infiltration varies greatly throughout the year, which the ASHRAE Standard 62.2-2016 infiltration credit does not capture.

Table 6. Predicted annual average infiltration rate and infiltration credit for MH and NZERTF

Test House	Annual average infiltration rate (L/s)	Atlanta	Baltimore	Chicago
MH	Simulation: $Q_{\mathrm{inf,RT}}$ (mean±standard deviation)	33±20	37±22	47±26
	Infiltration credit: $Q_{\mathrm{inf,62.2}}$	21	23	23[a]
NZERTF	Simulation: $Q_{\mathrm{inf,RT}}$ (mean±standard deviation)	16±10	18±11	21±11
	Infiltration credit: $Q_{\mathrm{inf,62.2}}$	6	7	8

a. As noted also in *Methods*, the infiltration credit in Chicago for the MH was calculated as 28 L/s but due to the 2/3s limitation in the standard, its credit is set to 23 L/s.

Most of the difference between the predicted infiltration and infiltration credit in Chicago can be attributed to the heating months as seen in Figure 7 and Figure 8. These figures show the monthly average infiltration rate predicted by CONTAM, as well as the infiltration credit ($Q_{inf,62.2}$) and total ventilation rate required by ASHRAE Standard 62.2-2016 ($Q_{tot,62.2}$) for the MH and NZERTF, respectively, in Chicago.

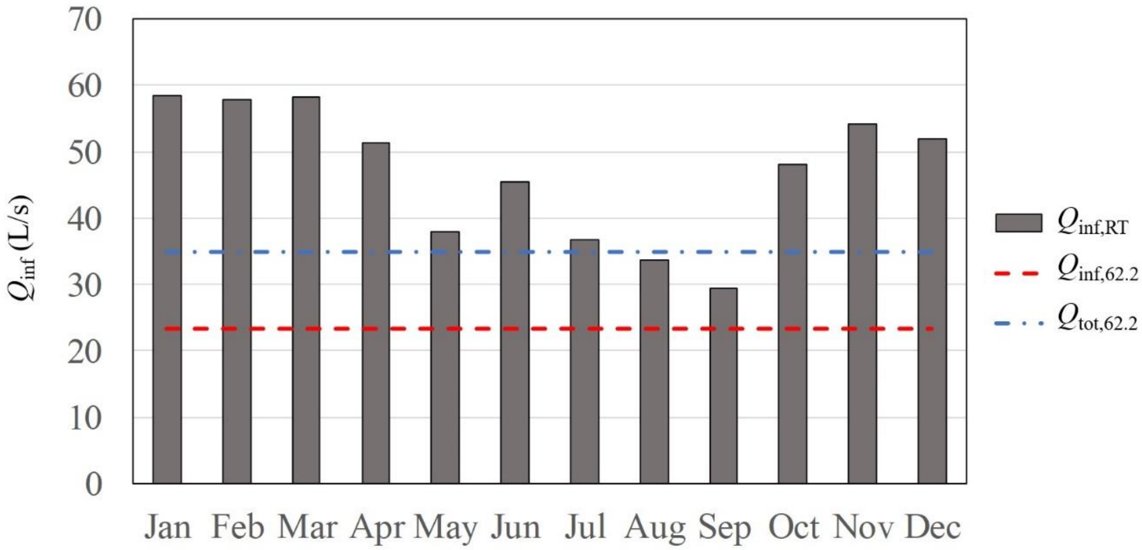


Figure 7. Monthly average infiltration predicted CONTAM in Chicago at the MH. The horizontal lines show the infiltration credit and total outdoor airflow rate required by ASHRAE Standard 62.2-2016.

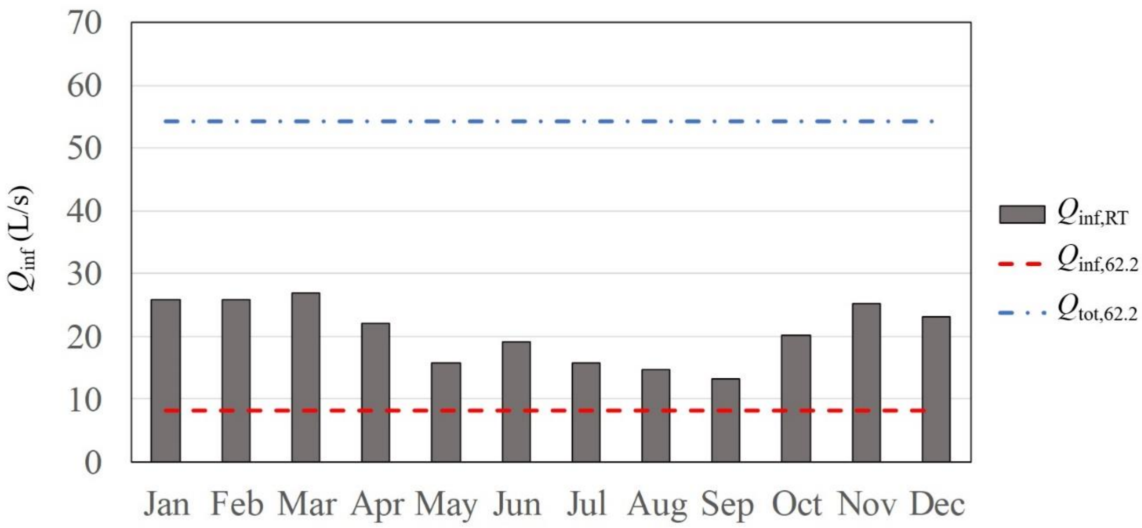


Figure 8. Monthly average infiltration predicted by CONTAM in Chicago at the NZERTF. The horizontal lines show the infiltration credit and total outdoor airflow rate required by ASHRAE Standard 62.2-2016.

For the heating months (October to March), the infiltration credit for the MH underestimates the infiltration rate by between 106 % and 150 % (average 135 %) (Figure 7). In cooling months, the infiltration credit also underestimates the infiltration rate, but only by between 26 % and 120 % (average 68 %). Further, for many of the heating months, infiltration-alone was on average greater than $Q_{tot,62.2}$. Graphs of monthly average infiltration rates for Atlanta and Baltimore show similar trends and are not presented. Results showed that in the heating months, the predicted infiltration alone could on average meet $Q_{tot,62.2}$ at the MH for all three cities.

Figure 8 shows the monthly average infiltration rate predicted by CONTAM, as well as the infiltration credit ($Q_{inf,62.2}$) and total ventilation rate required by ASHRAE Standard 62.2-2016 ($Q_{tot,62.2}$) at the NZERTF. In the heating months (October to following March), the infiltration credit underestimated the infiltration rate between 145 % and 226 % (average 197 %). In cooling months, the infiltration credit underestimated the infiltration rate between 61 % and 167 % (average 103 %). Monthly average infiltration rates for Atlanta and Baltimore showed similar trends but are not presented here. These findings of predicted infiltration rates below $Q_{inf,62.2}$ indicate that energy savings may be possible if RT control of an MV fan is implemented, especially during times when the infiltration rate was greater than $Q_{inf,62.2}$ or met $Q_{tot,62.2}$.

Average Q_{fan} flow rate

The average of $Q_{fan,RT}$ at the NZERTF predicted by the RT ventilation control strategy for all the cities (36 L/s) is larger than for the MH (7 L/s). These fan flow rates were on average 48 % less than what is required by ASHRAE Standard 62.2-2016 at the MH, and 23 % less at the NZERTF when the fan-off flows were included (Figure 9). While the average $Q_{fan,RT}$ fan flow rate is less than what is required in the standard, the maximum fan airflow required is equal to $Q_{tot,62.2}$. This is to ensure that when the infiltration was close to zero, the total outdoor ventilation rate could still be met. If fan-off flows were not included in the average, the annual average $Q_{fan,RT}$ at the MH would be 12 L/s across the three cities, which is a 2 % reduction from what is required by ASHRAE Standard 62.2-2016. Because the NZERTF requires more MV, its annual average $Q_{fan,RT}$ is not affected by exclusion of the fan-off flows.

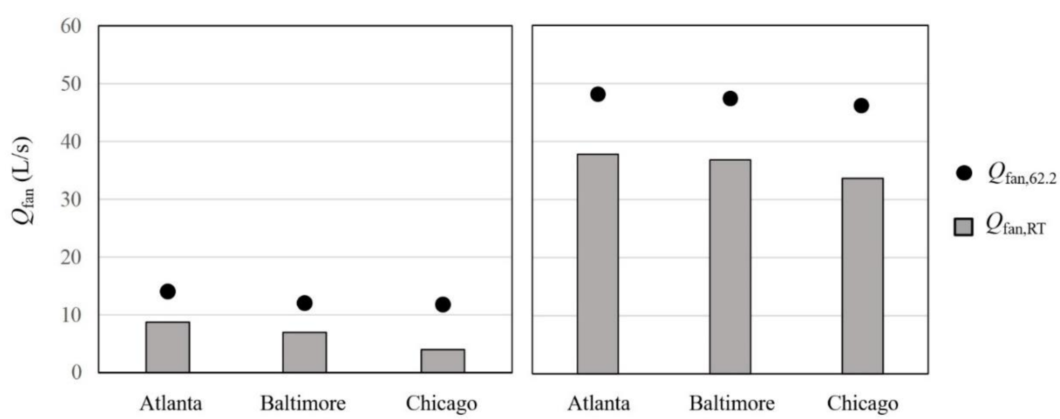
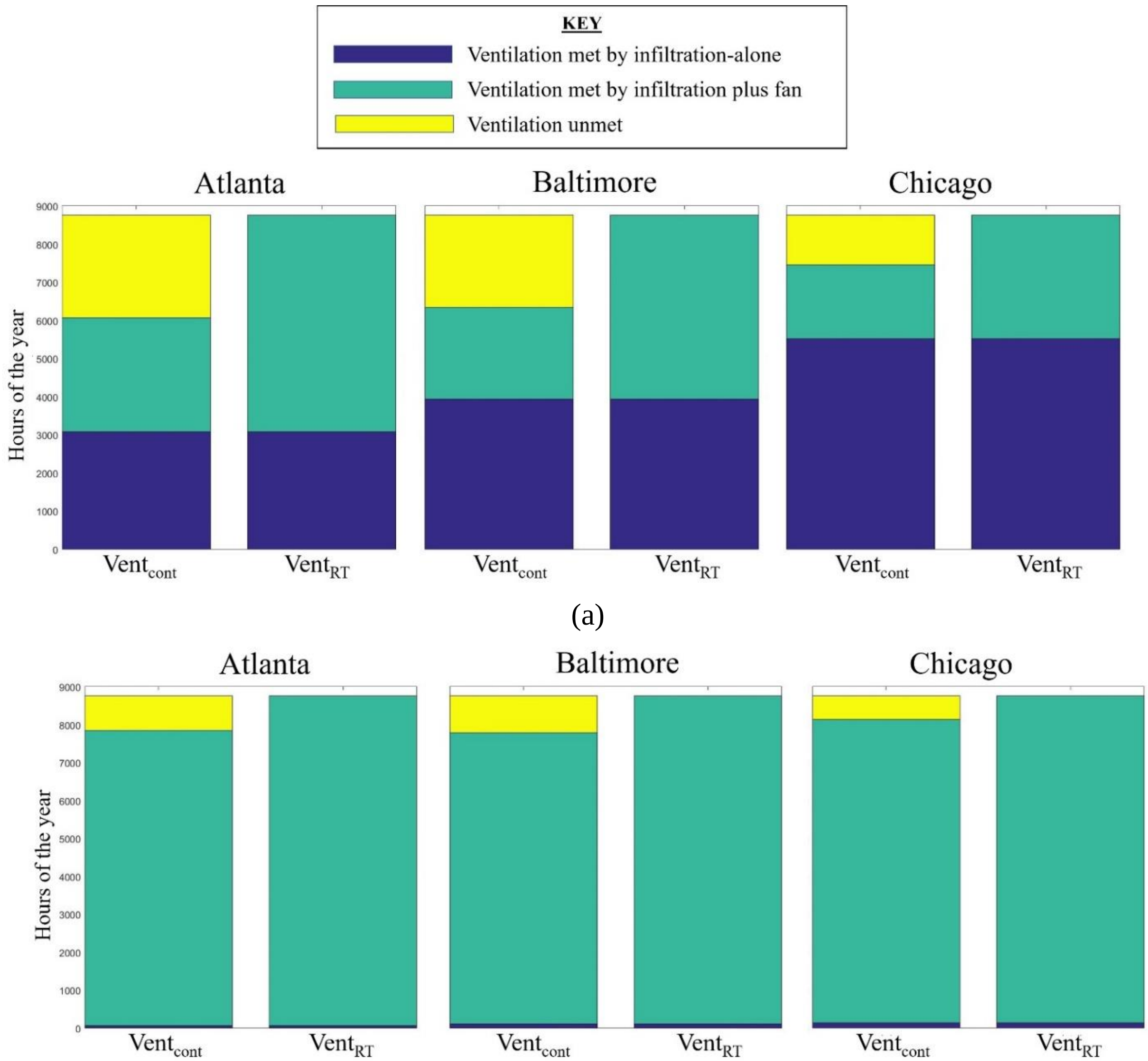


Figure 9. Annual average $Q_{fan,RT}$ flow rate at (a) MH and (b) NZERTF. $Q_{fan,62.2}$ is shown for

reference.

Ventilation hours

Figure 10 shows the number of hours when the total ventilation ($Q_{tot,62.2}$) was: (1) met by infiltration-alone; (2) met by infiltration plus mechanical ventilation; and (3) was unmet. The total of these hours is equal to 8760 h (i.e., one year). Even though the MV fan ran continuously at a rate of $Q_{fan,62.2}$ for the $Vent_{cont}$ strategy, there were unmet ventilation hours when the predicted CONTAM infiltration rate was less than $Q_{inf,62.2}$ (Figure 10). Unmet ventilation hours can be eliminated with the RT ventilation control strategy and by allowing $Q_{fan,RT} > Q_{fan,62.2}$ to make-up for the times when infiltration was lower than the infiltration credit. $Q_{tot,62.2}$ was unmet 24 % of the year (89 days) at the MH using the $Vent_{cont}$ strategy. At the NZERTF, it was unmet 10 % of the year (35 days) using the same ventilation strategy.



(a)

(b)

Figure 10. Hours of the year $Q_{tot,62.2}$ met by infiltration-alone, met by MV plus infiltration,

and unmet at (a) MH and (b) NZERTF

The most notable difference between Figure 10a (MH) and Figure 10b (NZERTF) is that in the MH there are more hours of the year where infiltration-alone met the ventilation requirement ($Q_{\mathrm{inf,RT}} \geq Q_{\mathrm{tot,62.2}}$) than in the NZERTF. This is because the NZERTF is much tighter than the MH, thus the infiltration rates are lower. In addition, the NZERTF requires more ventilation because it is larger. As expected, the colder the climate, the greater the number of met-ventilation hours by infiltration-alone in both houses. This is more obvious in Figure 10a for the MH than it is in Figure 10b for the NZERTF. Nevertheless, in Chicago, there was a 17 % increase in the number of hours where infiltration-alone met the ventilation requirement at the NZERTF compared with in Baltimore. On average, $Q_{\mathrm{tot,62.2}}$ was met by infiltration-alone at the MH across the three cities for almost 50 % the year. At the NZERTF, ventilation was on average met by infiltration-alone only 1 % of the year.

Relative exposure factor

Table 7 shows the annual relative exposure factor (R_{avg}) for the RT ventilation control strategy across the three cities for both test houses. The annual relative exposure factor of the RT ventilation strategy was less than or equal to 1.0 for both test houses in all three cities. The higher R_{avg} values were at the NZERTF.

Table 7. R_{avg} at MH in three cities using RT ventilation strategy compared with constant ventilation.

MH			NZERTF		
Atlanta	Baltimore	Chicago	Atlanta	Baltimore	Chicago
0.9	0.8	0.7	1.0	1.0	1.0

There were no hours of the year at the MH or the NZERTF when R_i was greater than 5.0, which is the limit in ASHRAE Standard 62.2-2016. The findings of reduced average $Q_{\mathrm{fan,RT}}$ and the relatively small impact on R_{avg} indicate that potential energy savings may be realized with reduced fan operation, while not significantly increasing occupant exposure. On the contrary, at the MH, R_{avg} per the exposure assumed in this study was reduced with RT ventilation control compared with constant ventilation at $Q_{\mathrm{tot,62.2}}$. At the NZERTF, RT ventilation control remained similar to the exposure using constant ventilation at $Q_{\mathrm{tot,62.2}}$.

Energy impacts of ventilation

The energy required for ventilation is evaluated in this study in terms of: (1) the energy to operate the mechanical ventilation (MV) fan and (2) the sensible energy to condition the MV air. The MV system was assumed with no heat recovery. The energy was converted to cost using an assumed cost of electricity per kilowatt-hour (kWh). In order to understand the energy impacts, a discussion of fan runtimes is presented first.

Figure 11 presents the percentage reduction in fan runtime of the RT ventilation control strategy compared with the more typical case of continuous fan operation (i.e., the $\mathrm{Vent}_{\mathrm{cont}}$ strategy).

Figure 11 shows that at the MH, there was an average 48 % reduction in fan runtime across the three cities using the RT control strategy. At the NZERTF, the average fan runtime reduction was only 1 %. The reduction was greater in the MH because it is leakier and subsequently, infiltration-alone could fulfil the ventilation requirement more of the time.

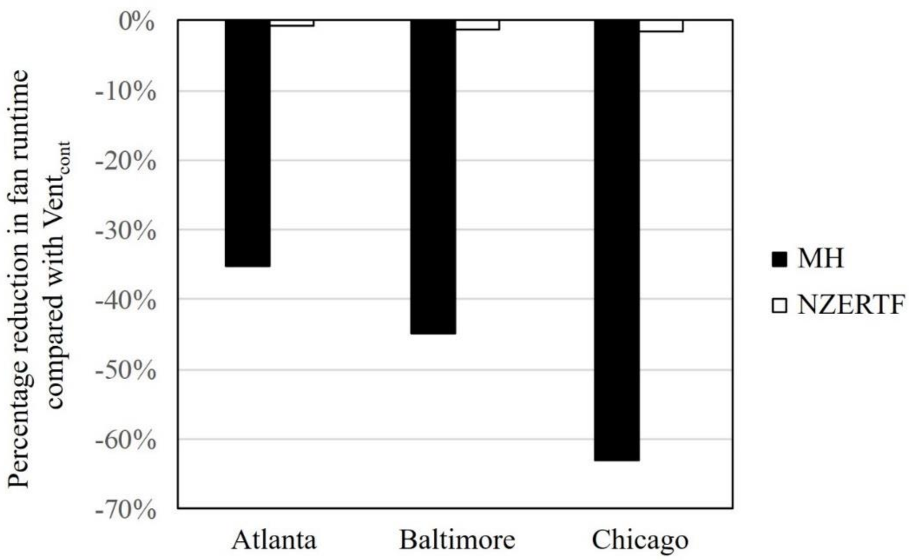


Figure 11. Percentage reduction in fan runtime for the RT control strategy at MH and NZERTF in three cities compared with Vent$_{cont}$ strategy.

Figure 12 shows the cost difference of implementing the RT ventilation control strategy compared with the Vent$_{cont}$ strategy. The cost differences are calculated for both test houses, in each city, and split into the differences resulting from (1) the energy to operate the MV fan and (2) the sensible energy to condition the MV air. Figure 12 shows that smaller savings came from reducing the MV fan operation. There was also a trend that in the coldest city of those studied, i.e., Chicago, the total energy (operation and conditioning) cost savings was the greatest. Across the three cities, the total annual energy cost savings at the MH using the RT ventilation control strategy ranged from $37USD to $69USD (average = $51USD). At the NZERTF, the total annual energy cost savings ranged from $62USD to $104USD (average = $82USD).

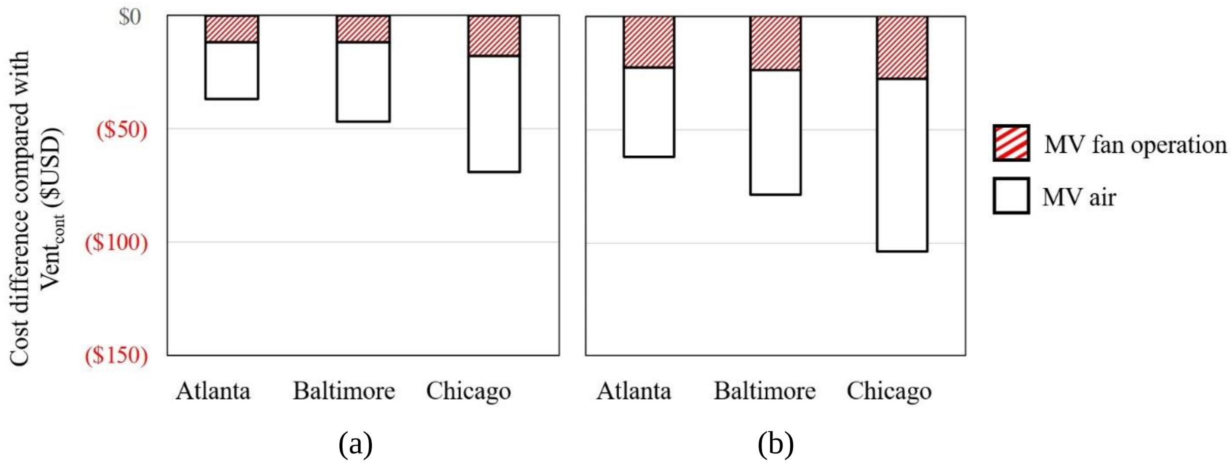


Figure 12. Energy cost difference for (a) MH and (b) NZERTF.

As noted earlier, the energy required to condition the infiltration air was not considered because the two ventilation strategies studied both used infiltration rates from CONTAM. Nevertheless, the cost to condition the infiltration air predicted by CONTAM was 2 to 5 times higher than the cost to condition the infiltration rate based on the ASHRAE Standard 62.2-2016 infiltration credit throughout the year. Thus, energy use estimates using the infiltration credit may underestimate actual energy use because it is a conservative value.

Discussion and Limitations

In this study, the authors studied the performance of an RT ventilation control strategy using detailed CONTAM models to predict hourly infiltration rates and incorporating them into a theoretical controller at the MH and NZERTF. When using CONTAM models to predict RT infiltration rates, as opposed to assuming a constant infiltration rate, we found an average 36 % reduction in the average MV airflow compared with the requirement in ASHRAE Standard 62.2.-2016 for the two test houses and three cities studied. This was when fan-off flow rates were also included in the average. When fan-off flow rates were excluded from the average, the reduction in MV across the two test houses and three cities was 12 %. For the RT control strategy in all cities, reducing the average MV airflow did not increase annual relative exposure above the limits in the standard. The RT control strategy resulted in an average annual savings of $66USD across both test houses and three cities. While the savings in an individual home may not seem large, across a larger community of homes, the total savings would be more substantial.

The RT ventilation control strategy had advantages and disadvantages. This strategy used infiltration rates from a verified, detailed CONTAM model of the house. However, developing and verifying the detailed model required expertise and time. Thus, a method to create useful, simplified CONTAM models that can be more easily applied in the field will be included in future work. Nevertheless, the real-time weather needed for RT ventilation control is currently available via a range of mechanisms and is already applied in many smart thermostats.

Future work

Additional measurement and simulation exercises would be useful to achieve a better understanding of the performance of RT ventilation controllers using infiltration rates estimates. In particular, the test houses are examples of particularly tight and leaky construction; it would be useful to evaluate homes that are more typical in terms of envelope airtightness. The analyses also assumed that the MV system was balanced. However, other types of ventilation systems should be evaluated, such as exhaust-only, supply-only, and those integrated with the heating/cooling system.

When evaluating the impacts of outdoor air ventilation on energy, only sensible heat was considered in this study. The impact of latent heat load on a home, especially in summer, needs to be considered, particularly for cities like Atlanta, where the latent load could exceed the sensible load. For this type of analysis, a coupled airflow-energy model is ideal, such as the CONTAM-EnergyPlus model[24, 37].

As noted above, the downside of developing and verifying a detailed CONTAM model of a house requires expertise and time. Thus, a follow-up to this study is one in which simplified CONTAM models are developed and used to provide RT infiltration rates. There may also be methods to estimate real-time infiltration that would not rely on a CONTAM model, such as regression models, artificial intelligence, or a combination of CONTAM simulations and data-driven models. These approaches would require more measurements to be taken in more houses and in more locations. There are also other models of infiltration, such as the AIM-2[9] that can be used to estimate infiltration and may be easier to program into a controller than CONTAM.

This study evaluated occupant exposure using a uniformly-distributed, constant and generic contaminant source as outlined in ASHRAE Standard 62.2-2016. In reality, airborne contaminant sources exhibit spatial and temporal variation, and have very different health and comfort impacts on building occupants. With the increased availability of consumer-grade air monitors and the continual improvement in their measurement accuracy, it may be possible to implement ventilation control that is more responsive to individual indoor environments and to the occupants' unique health and comfort needs and preferences.

Conclusions

Outdoor air ventilation rates specified in ASHRAE Standard 62.2-2016 account for infiltration using a single, constant value. However, infiltration rates vary significantly with ventilation system operation, weather, and indoor conditions in accordance with known physical relationships. Thus, a single assumed or measured infiltration rate may not be appropriate when determining whether the total outdoor air ventilation rate requirement is met throughout the year. The authors proposed the use of CONTAM airflow models to determine real-time infiltration rates, which could then be passed to an RT ventilation system controller to reduce or eliminate MV when the infiltration is greater than the credit assigned per ASHRAE Standard 62.2-2016. The method was evaluated for two test houses on the NIST campus in Gaithersburg, MD, USA.

The implementation of the theoretical controller resulted in predicted annual energy cost savings ranging from $37USD to $69USD (average = $51USD) at the MH and $62USD to $104USD (average = $82USD) at the NZERTF. These savings were realized without an increase in annual occupant exposure to a simple, generic contaminant, relative to ventilating continuously at a single rate at the NZERTF. At the MH, the annual occupant exposure improved with RT ventilation control.

Acknowledgements

The authors would like to thank Jonathan Wilson and David Jacobs from the National Center for Healthy Housing for partially funding this research in conjunction with Beetle Management, Inc under CRADA CN-17-0080. The authors would also like to thank Gabriel Grajewski and Daniel Greb at NIST for their key contributions to the experimental efforts. No conflicts of interest exist in the publication of this manuscript. Any mention of commercial products within this manuscript is for information only; it does not imply recommendation or endorsement by the authors, the National Institute of Standards and Technology, or Beetle Management, Inc.

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