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Impact of Residential Mechanical Ventilation on Energy Cost and Humidity Control

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for
Building Science Corporation,
Florida Solar Energy Center/BA-PIRC, IBACOS

January 2014

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U.S. Department of Energy

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Impact of Residential Mechanical Ventilation on Energy Cost and Humidity Control

Prepared for:

The National Renewable Energy Laboratory

On behalf of the U.S. Department of Energy's Building America Program

Office of Energy Efficiency and Renewable Energy

15013 Denver West Parkway

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NREL Contract No. DE-AC36-08GO28308

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January 2014

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Definitions

AHU	Air handling unit
BSC	Building Science Corporation
CFI	Central fan integrated supply ventilation
DOE	U.S. Department of Energy
E+	EnergyPlus
EMPD	Effective moisture penetration depth
EGUSA	EnergyGauge USA
EPA	U.S. Environmental Protection Agency
ERV	Energy recovery ventilator
HAMT	Heat and moisture transport
HUD	U.S. Department of Housing and Urban Development
HRV	Heat recovery ventilator
IECC	International Energy Conservation Code
IMC	International Mechanical Code
IRC	International Residential Code
MC	Moisture capacitance
NBC	National Building Code of Canada
NREL	National Renewable Energy Laboratory
RH	relative humidity
WAVIAQ	Washington State Ventilation and Indoor Air Quality Code

Executive Summary

The U.S. Department of Energy (DOE) Building America program has been conducting research leading to cost-effective high performance homes since the early 1990s. Optimizing whole-house mechanical ventilation as part of the program's systems engineered approach to constructing housing has been an important subject of the program's research. Ventilation in residential buildings is one component of an effective, comprehensive strategy for creation and maintenance of a comfortable and healthy indoor air environment. Indoor air pollution control begins with avoiding the placement of items of known high pollutant emissions inside the living environment. It follows with local exhaust in areas such as kitchens, and bathrooms, where high emissions from pollution sources cannot be avoided, and it ends with whole-building controlled mechanical ventilation to dilute remaining indoor air pollutants with fresher outdoor air.

Several codes and standards require residential ventilation. Definitions include airflow rate requirements for local exhaust, whole-building mechanical ventilation rate requirements, sources of ventilation air, and distribution requirements. While a consistency of intent can be inferred, differences among codes and standards, along with the absence of residential ventilation requirements in many local codes, indicate the difficulty and uncertainty involved with a generic specification. If the primary indoor environment concern is the occupant's annual average exposure to pollutants, certain questions must be answered relating to the ability of ventilation to minimize that exposure:

- What are the health impacts of pollutants of concern? Are odor and moisture of concern, which have shorter term, rather than annual, exposure issues?
- What are the limits of whole-house residential ventilation systems to resolve the concerns? To what extent can such systems improve indoor air quality, by minimizing concentrations of chemical and particulate pollutants, beyond what can be accomplished through intermittent spot ventilation and source control?

In any climate, year-round indoor humidity control reduces condensation potential and relative humidity (RH), exceeding comfort conditions. As homes become more energy efficient, space conditioning systems with lower capacities can be used to maintain space temperature. Overall, this is good, and produces significant net energy and cost savings. However, this often requires a change to conventional residential space conditioning system design in humid climates. While the sensible cooling load is lower, and can be dealt with in the conventional way, the latent (moisture) load in high performance homes remains nearly unchanged due to ventilation requirements and internal moisture generation by occupants and their activities. Therefore, at times when there is no need to lower the space air temperature, supplemental dehumidification may be required to maintain acceptable RH.

The study described in this report is based on building energy modeling with an important focus on the energy and indoor humidity impacts of ventilation. The modeling tools used were EnergyPlus version 7.1 (E+) and EnergyGauge USA (EGUSA). Twelve U.S. cities and five climate zones were represented. A total of 864 simulations (2*2*2*3*3*12 = 864) were run using two building archetypes, two building leakage rates, two building orientations, three ventilation systems, three ventilation rates, and 12 climates. Modeling of rates lower than those promulgated by ASHRAE standards is not to suggest that such rates provide adequate indoor air quality—that is outside the scope of this report. The rates are chosen purely to evaluate the energy and moisture impacts of varying rates.

Energy-related conclusions of the simulation work include determining the difference in total space conditioning operating cost for a DOE Challenge Home controlled to < 60% RH (Figure 1). For homes with 75% of the ASHRAE 62.2-2013 ventilation rate requirement compared to 100% of that ventilation rate requirement, the decreased cost was about $45/yr averaged over all climates, enclosure tightness, and ventilation systems simulated, or a savings of 10% of total space conditioning energy. The study found that the 75% ASHRAE 62.2-2013 ventilation rate requirement is roughly equal to 94% of the ASHRAE 62.2-2010 requirement. At 50% of the ASHRAE 62.2-2013 ventilation rate requirement, the average additional savings is about another $35/yr for an average total savings of 15% of space conditioning energy compared with 100% of that ventilation rate requirement.

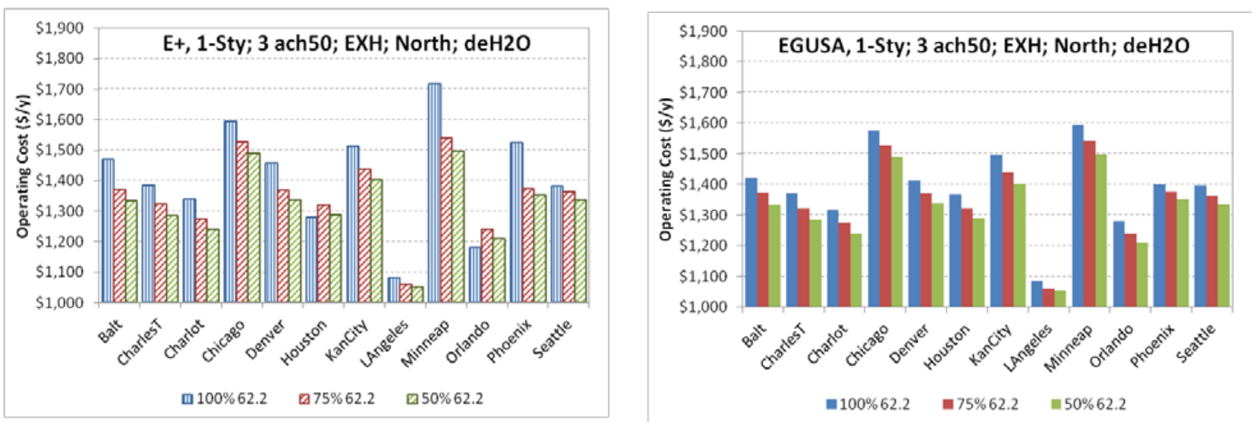


Figure 1. Total annual operating cost for a DOE Challenge Home controlled to < 60% RH over a range of ventilation rates

Total annual operating cost for a DOE Challenge Home varied less than $90/yr among all ventilation systems in every climate (Figure 2). Compared to exhaust ventilation as the base, a balanced energy recovery ventilator (installed such that it does not require coincident air handling unit fan operation to avoid short-circuiting of ventilation air) ranged from $0/yr to $40/yr more, with an average of about $20/yr more for 100% of the ASHRAE 62.2-2010 requirement at 3 ACH50. However, at the larger mechanical ventilation rates required in the 1.5 ACH50 homes, the ERV *saved* an average of $20/yr compared to exhaust ventilation, in all but the dry climate of Phoenix and mild climate of Los Angeles, where it continued to consume more energy than exhaust. Central fan integrated supply ventilation ranged from about $11/yr to $93/yr more than exhaust ventilation, with an average of about $40/yr.

Supplemental dehumidification controlled to 60% RH was predicted to be $10–$58/yr for the warm-humid climates of Charleston, Houston, and Orlando, and the marine climate of Los Angeles. However, a caveat is provided—this value is predicated on an operating dehumidifier energy factor of 1.47 L/kWh and recent field data indicate that conventional dehumidifiers operate closer to 0.8 L/kWh (Mattison and Korn 2012), which would tend to double this cost. Additionally, dehumidifiers tend to operate on a large humidity dead band, which means that maintaining humidity below 60% would likely require humidity set points near 55%, which could increase dehumidification costs by as much as 40% in humid climates. Also, with unbalanced mechanical ventilation, air change goes up as building leakage goes down from 3.0 ACH50 to 1.5 ACH50, resulting in more needed supplemental dehumidification. The increase in air exchange rate from unbalanced to balanced ventilation was about 24% for the 3.0 ACH50 houses and about 10% for the 1.5 ACH50 houses.

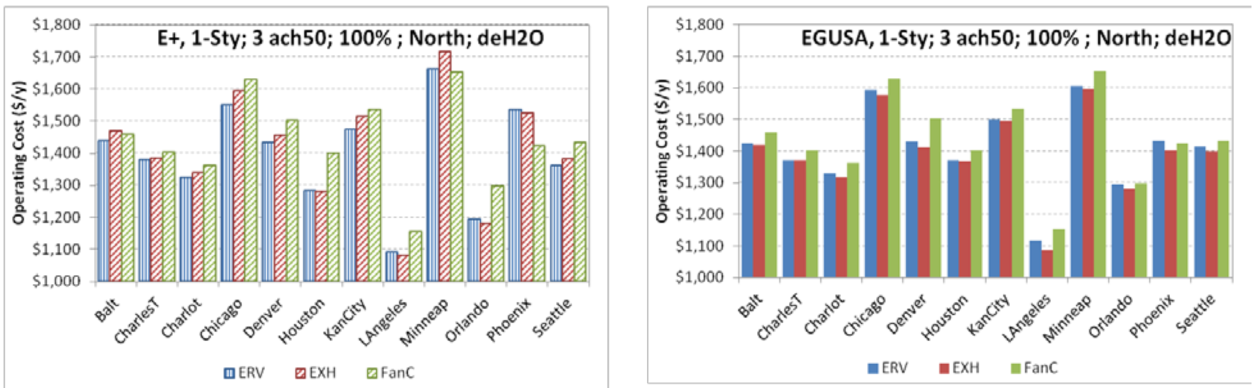


Figure 2. Total annual operating cost per year for a DOE Challenge Home over a range of ventilation system types

Humidity control related conclusions include validating that the hours of elevated indoor RH are a strong function of the selected RH limit and climate (Figure 3). For example, in Los Angeles, nearly all of the hours above 60% RH are also below 65% RH. However, this is not the case for Charleston, Houston, and Orlando. Hours above 60% RH during a particular space conditioning mode (heating, cooling, floating) were also a strong function of climate. Most of the hours significantly above 60% RH occur during floating hours, which occur mostly during fall, winter, and spring in Orlando.

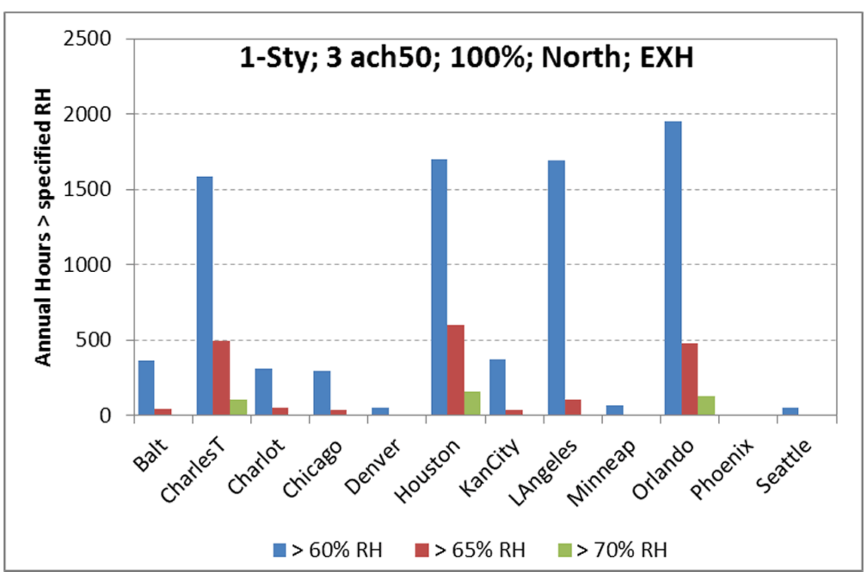


Figure 3. Hours of elevated indoor RH are a strong function of the selected RH limit and climate using EGUSA

A key gap and area of ongoing research is cost-effective application of ASHRAE Standard 62.2-2013 with systems and approaches that reduce energy consumption, improve humidity control, and improve indoor air quality. Identifying methods to achieve such objectives would benefit the U.S. Environmental Protection Agency Indoor airPLUS program required by the DOE Challenge Home. Research should be conducted into methods that include considerations for:

- Managing indoor air pollutants in ways other than outdoor air exchange.
- Quantifying exposure to pollutants, and mitigation of that exposure, over a time scale shorter than the annual average.
- Accounting for the quality of the outdoor air for different outdoor environs and ventilation and filtration systems types.
- Accounting for ventilation air distribution effectiveness.
- Application of demand-based ventilation systems.

Models that can accurately simulate the performance of advanced ventilation control systems in high performance buildings, along with associated humidity control systems in humid climates, are needed to understand a wide range of scenarios related to the economics and operational success of low-energy homes. Accurate simulation of ventilation and space conditioning control systems that operate on subhourly time scales coupled with the interacting heat and mass transfer effects is complex. Accurate and meaningful results depend on many still somewhat unknown inputs, such as:

- Internal moisture generation rates
- Other moisture loads, including effects of construction moisture and rain wetting under solar loading
- Building moisture capacitance and the impacts of building material moisture adsorption/desorption
- Detailed space conditioning equipment performance maps.

1 White Paper Introduction and Residential Ventilation Overview

The U.S. Department of Energy (DOE) Building America program has been conducting research leading to cost-effective high performance homes since the early 1990s. Optimizing whole-house mechanical ventilation as part of the program's systems engineered approach to constructing housing has been an important subject of the program's research. Tens of thousands of homes have been constructed in partnership with Building America, and many different approaches to whole-house mechanical ventilation have been incorporated and evaluated.

In the last decade, national programs have begun to utilize Building America research as a basis to define and label high performance homes, including DOE Challenge Home, U.S. Environmental Protection Agency (EPA) ENERGY STAR®, and U.S. Green Building Council Leadership in Energy and Environmental Design for Homes. These programs include whole-house mechanical ventilation as a required element. However, some programs, as well as some codes and standards, require minimum ventilation rates significantly higher than what many Building America builder partners have historically incorporated as part of the research collaboration. While ventilation is critically important to indoor air quality, whole-house mechanical ventilation can also have impacts on energy use, comfort, and durability. In northern cold climates, the energy use to condition outdoor air for ventilation can be significant; in southern climates, there are serious concerns related to indoor humidity control that may warrant additional energy use for dehumidification.

Data directly relating the effect of ventilation on occupant health are scarce. Ventilation systems have often been deemed effective in homebuilding practice as long as occupants are satisfied with odor and moisture control. However, this has little to do with whether or not there are pollutants present that may cause health issues because many pollutants are colorless and odorless, and many can be controlled through direct source removal or abatement. While evaluating details related to the effect of whole-house mechanical ventilation on occupant health is outside the scope of the Building America program, evaluating details relating to the effect of whole-house mechanical ventilation on energy use, comfort, and durability remains a priority of the program. This report intends to review how ventilation is handled in residential codes and standards, review some of the Building America program's practical experience with the application of mechanical ventilation in high performance research homes, and describe the results of simulations that were conducted to answer the following questions involving the effect of application of ventilation in high performance Building America Homes:

- What is the relationship between ventilation rate and energy required to maintain comfort?
- How do those relationships vary as a function of enclosure airtightness?
- How do those relationships vary climatically?
- How do those relationships vary as a function of ventilation system type?

1.1 Ventilation Goals

Ventilation in residential buildings is one component of an effective, comprehensive strategy for creation and maintenance of a comfortable and healthy indoor air environment. Indoor air pollutant control begins with avoiding the placement of items of known high pollutant emission inside the living environment. It follows with local exhaust in areas where high emission from pollutant sources cannot be avoided, such as kitchens, toilet rooms, and bathrooms (wet rooms), and it ends with whole-building controlled mechanical ventilation to dilute remaining indoor pollutants with fresher outdoor air.

Whole-building ventilation is accomplished where outdoor air is distributed in a controlled manner to dilute more polluted indoor air. Since pollutants are dispersed and usually at low concentration, there is no practical way to capture and exhaust them as is typical in a toilet room, over a shower, or over a kitchen cooktop. Rather, dilution through delivery and distribution of outside air is used to reduce the concentration of dispersed pollutants inside the home. Whole-building ventilation can be operated continuously at a lower rate, or intermittently at a higher rate and still deliver the same volume of dilution air.

While whole-building ventilation is needed for improved indoor air quality, ventilation rates that are higher than needed will waste energy. In dry climates, and during wintertime in cold climates, high ventilation rates can cause the house to be too dry, causing occupants to want humidification equipment that may not have been necessary if the ventilation rate were lower. Likewise, in hot, humid climates, high ventilation rates will increase indoor humidity, requiring additional dehumidification.

Complications begin when establishing the whole-building ventilation rate needed while also accounting for control of dangerous sources, system effectiveness factors related to balanced versus unbalanced ventilation systems, the distribution of ventilation air, and the source of ventilation air. There is a real need to establish a true accounting for these factors because excessive ventilation impacts energy consumption and humidity control in cold (too dry) and hot humid climates (to wet) alike.

1.2 The Role of Pollutant Source Control

The impact of source control should be further investigated, as it could reduce ventilation requirements in various programs that promote healthy indoor environments such as the EPA Indoor airPLUS and DOE Challenge Home programs. In this context, pollutant source control refers to controlling the introduction of materials into the dwelling, and controlling the frequency of activities inside the dwelling, that cause emissions of pollutants.

If the primary indoor environment concern is the occupant's annual average exposure to pollutants, certain questions must be answered relating to the ability of ventilation to minimize that exposure:

- What are the health impacts of pollutants of concern? Are odor and moisture of concern, which have shorter term, rather than annual, exposure issues?
- What are the limits of whole-house residential ventilation systems to resolve that concern? To what extent can such systems improve indoor air quality, by minimizing concentrations of chemical and particulate pollutants, beyond what can be accomplished through intermittent spot ventilation and source control?

For example, research has shown that formaldehyde can act as a constant concentration pollution source; i.e., as outdoor air exchange is increased the concentration of formaldehyde remains mostly unchanged because the emission rate increases (Weisel et al. 2005). This presents a limitation with respect to the ability of ventilation to minimize an occupant's acute exposure, however may reduce the extent of long term exposure by allowing for faster offgassing, such that the total offgassing of materials is completed sooner.

A recently published study by Lawrence Berkley National Laboratory measured a decrease in formaldehyde concentration with increasing air exchange rate, but found the reduction to be less than proportional, indicating that emission rates may have been buffered by the elevated indoor concentration of formaldehyde (Willem et al. 2013). Such studies point to the importance of source control as a primary strategy to minimize exposure to indoor pollutants. If introduction of sources of pollutants into the living space is not prevented, or if subsequent emission is not removed at the source, even the internal mixing caused by a residential ventilation system, or a heating/cooling system, or internal temperature differences could exacerbate the local exposure, and the problem.

1.3 Ventilation Considerations in Humid Climates

In any climate, year-round indoor humidity control reduces condensation potential and relative humidity (RH), exceeding comfort conditions. Areas of high moisture generation such as kitchens, bathrooms, toilet rooms, and laundries, should be exhausted at the source. When outdoor conditions are right, whole-house ventilation can serve to dilute remaining indoor moisture with drier outdoor air.

As homes become more energy efficient, space conditioning systems with lower capacities can be used to maintain space temperature. Overall, this is good, and produces significant net energy and cost savings. However, this situation requires a change to conventional residential space conditioning system design in humid climates. While the sensible cooling load is lower, and can be dealt with in the conventional way, the latent (moisture) load in high performance homes remains nearly unchanged due to ventilation requirements and internal moisture generation by occupants and their activities. Therefore, at times when there is no need to lower the space air temperature, supplemental dehumidification may be required to maintain RH at acceptable levels.

1.4 Review of Ventilation in Residential Codes and Standards

Several codes and standards require residential ventilation. Definitions include airflow rate requirements for local exhaust, whole-building mechanical ventilation rate requirements, sources of ventilation air, and distribution requirements. While a consistency of intent can be inferred, differences among codes and standards, along with the absence of residential ventilation requirements in many local codes, indicate the difficulty and uncertainty involved with a generic specification.

1.4.1 Local Mechanical Exhaust Flow Requirements

1.4.1.1 Kitchen

The 2012 International Residential Code (IRC), the 2012 International Mechanical Code (IMC), the Minnesota Building Code (which points to the IRC), the Washington State Ventilation and Indoor Air Quality Code (WAVIAQ), and ASHRAE 62.2-2010 require 100 CFM to outside intermittent, or 25 CFM continuous in a kitchen. The U.S. Department of Housing and Urban Development (HUD) Code and the National Building Code of Canada (NBC) require rated exhaust capacity of 100 CFM in a kitchen.

1.4.1.2 Bathroom

The 2012 IRC, the 2012 IMC, the Minnesota Building Code, the WAVIAQ, and ASHRAE 62.2-2010 require 50 CFM to outside intermittent, or 20 CFM continuous in a bathroom (having a shower, tub, spa, or other bathing fixture). The HUD Code and the NBC require rated exhaust capacity of 50 CFM in a bathroom.

The 2012 IRC, the 2012 IMC, the Minnesota Building Code, and the WAVIAQ require 50 CFM to outside intermittent, or 20 CFM continuous in a toilet room. The HUD Code and the NBC require rated exhaust capacity of 50 CFM in a toilet room (water closet).

While ASHRAE 62.2-2013 does require intermittent ventilation of bathrooms, it does not require mechanical ventilation for toilet rooms, and does not consider odor an indoor pollutant of concern.

1.4.2 Whole-Building Mechanical Ventilation Rate Requirements

1.4.2.1 International Residential Code 2012

For all new construction with measured building air leakage of < 5.0 ACH50, the IRC requires a whole-house mechanical ventilation system consisting of one or more supply or exhaust fans, or a combination of such, providing a whole-house mechanical ventilation rate according to Table 1 below, either continuous or intermittent within a time period not less than 1 hour in every 4 hours. The rates shown in the table are based on the ASHRAE 62.2-2010 rate equation, but are generally higher than that because the groupings cover a wide range of floor area. For example, the 1,501–3,000 ft^2 house category captures most U.S. housing in one category with a ventilation rate requirement that is 60% higher than required by ASHRAE 62.2-2010 for a 1,501-ft^2, two-bedroom house. For an average 2,000-ft^2, three-bedroom house, the rate is 20% higher.

Table 1. Continuous Whole-House Mechanical Ventilation System Airflow Rate Requirements of the 2012 IRC*

Dwelling Unit Floor Area (ft^2)	Number of Bedrooms				
	0–1	2–3	4–5	6–7	> 7
	Airflow in CFM				
< 1,500	30	45	60	75	90
1,501–3,000	45	60	75	90	105
3,001–4,500	60	75	90	105	120
4,501–6,000	75	90	105	120	135
6,001–7,500	90	105	120	135	150
> 7,500	105	120	135	150	165

*Adapted from Table M1507.3.3(1)

1.4.2.2 International Mechanical Code 2012

For all new construction with measured building air leakage of < 5.0 ACH50, the IMC 2012 requires a balanced mechanical ventilation system while not prohibiting negative or positive building pressure with respect to outdoors. The mechanical ventilation must be provided by a method of supply air and return or exhaust air, and the amount of supply air shall be approximately equal to the amount of return and exhaust air. For single and multiple private dwellings, the required airflow rate must be calculated based on 0.35 ACH, but not less than 15 CFM/person, with two people counted for the first bedroom and one for each additional bedroom.

1.4.2.3 Minnesota Building Code 2009

The Minnesota Building Code requires a total ventilation rate (capacity) equal to total ventilation rate (CFM) = (0.02)(square feet of conditioned space) + (15)(number of bedrooms + 1) for each 1-hour period. For a 2000-ft^2, three-bedroom house, that would be 100 CFM. For heat recovery ventilators (HRVs) and energy recovery ventilators (ERVs), the average hourly ventilation capacity must account for any reduction of exhaust or outdoor air intake, or both, for defrost or other equipment cycling per Home Ventilation Institute Standard 920. However, the continuous (or averaged over each hour) ventilation rate (operating), must be not less than 50% of the total ventilation rate and not less than 40 CFM. In other words, the Minnesota code requires a total ventilation rate *capacity* of two times the ASHRAE 62.2-2010 rate and an *operating* ventilation rate equal to the ASHRAE 62.2-2010 rate. A maximum airflow limit occurs when local ventilation requirements are being met by the continuous ventilation system, in which case shall not be capable of operating at more than the total ventilation rate.

1.4.2.4 Washington State Ventilation and Indoor Air Quality Code

The WAVIAQ requires minimum and maximum whole-house ventilation rates according to the table below. The rates shown in the Table 2 are also generally higher than the ASHRAE 62.2-2010 rates. For example, the 1,501–2,000 ft^2 house category captures most U.S. housing in one category with a minimum ventilation rate requirement that is 73% higher than required by ASHRAE 62.2-2010 for a 1,501-ft^2, two-bedroom house. For an average 2,000-ft^2, three-bedroom house, the minimum rate is 60% higher.

Table 2. Ventilation Rates for All Group R Occupancies Four and Fewer Stories (CFM)*

Floor Area (ft^2)	Number of Bedrooms													
	≤ 2		3		4		5		6		7		8	
	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
< 500	50	75	65	98	80	120	95	143	110	165	125	188	140	210
501–1000	55	83	70	105	85	128	100	150	115	173	130	195	145	218
1001–1500	60	90	75	113	90	135	105	158	120	180	135	203	150	225
1501-2000	65	98	80	120	95	143	110	165	125	188	140	210	155	233
2001–2500	70	105	85	128	100	150	115	173	130	195	145	218	160	240
2501-3000	75	113	90	135	105	158	120	180	135	203	150	225	165	248
3001–3500	80	120	95	143	110	165	125	188	140	210	155	233	170	255
3501–4000	85	128	100	150	115	173	130	195	145	218	160	240	175	263
4001–5000	95	143	110	165	125	188	140	210	155	233	170	255	185	278
5001–6000	105	158	120	180	135	203	150	225	165	248	180	270	195	293
6001–7000	115	173	130	195	145	218	160	240	175	263	190	285	205	308
7001–8000	125	188	140	210	155	233	170	255	185	278	200	300	215	323
8001–9000	135	203	150	225	165	248	180	270	195	293	210	315	225	338
> 9000	145	210	160	240	175	263	190	285	205	308	220	330	235	353

*Adapted from Table 3-2 of the WAVIAQ

1.4.2.5 National Building Code of Canada 2005

The NBC requires a principal ventilation fan to have minimum and maximum exhaust capacities shown in Table 3 below, depending solely on the number of bedrooms.

Table 3. Normal Operating Exhaust Capacity of Principal Ventilation Fan*

Number of Bedrooms in Dwelling Unit	Normal Operating Exhaust Capacity of Principal Ventilation Fan (CFM)	
	Minimum	Maximum
1	34	51
2	38	59
3	47	68
4	55	81
5	64	95

*Adapted from Table 9.32.3.3 of the 2005 NBC

1.4.2.6 Housing and Urban Development Code 2008

The HUD Code requires that each manufactured home be provided with whole-house ventilation having a minimum capacity of 0.035 CFM/ft^2 of interior floor space or its hourly average equivalent, but not less than 50 CFM or more than 90 CFM.

1.4.2.7 ASHRAE Standard 62.2-2013

The ASHRAE Standard 62.2-2013 determines the whole-house ventilation rate based on two factors—the number of bedrooms and the conditioned floor area. The bedroom part amounts to 7.5 CFM for each occupant, with the number of occupants being counted as the number of bedrooms plus one. The “plus one” comes from presuming that the first bedroom will likely have two occupants. The floor area part amounts to 3 CFM per 100 ft^2 of floor area. These two parts (the people and floor area parts) are added together. For example, a three-bedroom, 2,000-ft^2 house would require 90 CFM, calculated as: $(7.5)(3+1) + (2000)(0.03) = 30 + 60 = 90$ CFM. Without taking any credit for measured infiltration, that is an 80% increase over the 62.2-2010 requirement. Where a blower door test is conducted, and the annual average infiltration rate is calculated, up to two-thirds of the infiltration rate can be deducted from the whole-building ventilation rate to arrive at a mechanical ventilation fan rate.

The whole-building ventilation can be continuous or intermittent, and must have on/off override control readily accessible to the occupant. For a continuous ventilation fan flow rate requirement of 90 CFM, the intermittent ventilation fan flow rate for a system operating one-half of the time would have to be 180 CFM. Also, the full effective amount of ventilation must be provided within a 4-hour cycle time or a higher ventilation rate must be used based on an effectiveness factor. Whole-building ventilation fans that are not remotely mounted, or are not part of the central space conditioning system, must comply with a sound level requirement of 1 sone or less.

ASHRAE Standard 62.2 accounts for ventilation effectiveness only in regards to system runtime. While the maximum ventilation air delivery cycle time is truncated to 1 day, the runtime effectiveness values are based on calculations that would allow the ventilation system to be off for months without any decrease in effectiveness, because the evaluation metric is locked to annual average exposure. That approach ignores shorter term indoor air quality effects of odor and sensory irritation, which are nevertheless stated parts of an acceptable indoor air quality approach in the ASHRAE Standard 62.2 Scope, and definitions of "acceptable indoor air quality" and "air cleaning."

1.4.3 Whole-Building Mechanical Ventilation Air Distribution Requirements

1.4.3.1 International Mechanical Code 2012

The 2012 IMC requires outdoor air such that the minimum outdoor airflow rate shall be determined in accordance with a specification table in the IMC Section 403.3. Ventilation supply systems shall be designed to deliver the required rate of outdoor airflow to the breathing zone within each occupiable space.

1.4.3.2 Minnesota Building Code 2009

The Minnesota Building Code requires ventilation air distribution and circulation such that outdoor air is delivered to each habitable space by a forced air circulation system, separate duct system, individual inlets, or a passive opening. When outdoor air is directly ducted to a forced-air circulation system, circulation of 0.075 CFM/ft^2 must be maintained on average each hour. When outdoor air is not directly ducted to a forced-air circulation system, circulation of 0.15 CFM/ft^2 must be maintained on average each hour.

1.4.3.3 Washington State Ventilation and Indoor Air Quality Code 2009

The WAVIAQ requires the introduction and distribution of outdoor air and the removal of indoor air by mechanical means. It further requires that outdoor air be distributed to each habitable room by means such as individual inlets, separate duct systems, or a forced-air system. Conflictingly, in homes with exhaust-only ventilation systems without outdoor air inlets, the home must have a ducted forced-air heating system that communicates with all habitable rooms and the interior doors must be undercut to a minimum of ½ in. above the surface of the finish floor covering; however, nothing is mentioned about a minimum interval of ducted forced-air heating system communication with all habitable spaces. This will typically leave days and weeks on end with minimal ventilation air distribution.

1.4.3.4 National Building Code of Canada 2005

In the NBC, for ventilation systems not used in conjunction with a forced-air heating system, an outdoor air supply ventilation fan is required with the same rated capacity as the principal (exhaust) ventilation fan to distribute outdoor air directly to all bedrooms through a system of supply ducts. Where an exhaust-only system is installed via the principal ventilation fan, the exhaust fan control must be wired so that activation of the exhaust fan automatically activates the circulation fan of the forced-air distribution system required at its rated capacity but not less than 5 times the rated capacity of the exhaust fan. Alternatively, interlocking the forced-air distribution system's circulation fan with the principal (exhaust) ventilation fan can be accomplished where the forced-air distribution system is equipped with a control that automatically activates the circulation fan at user-selected intervals.

1.4.3.5 Housing and Urban Development Code 2008

The HUD Code requires that ventilation system be designed to ensure that outdoor air is distributed to all bedrooms and main living areas.

1.4.3.6 ASHRAE Standard 62.2-2013

ASHRAE 62.2 does not attempt to address the issue of delivery of outdoor airflow to each space, or the breathing zone within each occupiable space, or forced-air circulation of ventilation air at all. It simply makes an assumption that for all ventilation system cases, the entire house is a single, well-mixed zone, focusing only on annual average occupant exposure.

1.4.4 Whole-Building Mechanical Ventilation Air Source Requirements

1.4.4.1 International Mechanical Code 2012

The IMC 2012 requires an approximately balanced ventilation system with the ventilation supply system designed to deliver the required rate of outdoor airflow to the breathing zone within each occupiable space.

1.4.4.2 Washington State Ventilation and Indoor Air Quality Code 2009

The WAVIAQ requires that the ventilation system have direct outdoor air inlets, and that they be screened and located so as not to take air from the following contaminated areas, unless an exhaust only ventilation system has a ducted forced-air heating system that communicates with all habitable rooms and the interior doors are undercut to a minimum of ½- in. above the surface of the finish floor covering:

- Closer than 10 ft from an appliance vent outlet, unless such vent outlet is 3 ft above the outdoor air inlet
- Where it will pick up objectionable odors, fumes, or flammable vapors
- A hazardous or unsanitary location
- A room or space having any fuel-burning appliances therein
- Closer than 10 ft from a vent opening of a plumbing drainage system, unless the vent opening is at least 3 ft above the air inlet
- Attics, crawlspaces, or garages.

This provision creates a significant inconsistency, since it is presumably acceptable to take outdoor air from all the above contaminated locations for exhaust-only systems but not for any other system. To be consistent, the WAVIAQ would need to also require that for exhaust-only ventilation systems, the path of least resistance must always be the outdoor air inlet(s) and not elements of the building enclosure, while understandably excluding high wind infiltration conditions. To make that possible, very large inlets would be required, and would create significant comfort and energy problems.

1.4.4.3 National Building Code of Canada 2005

The NBC stipulates that outdoor supply air supply be connected directly to the outside.

1.4.4.4 Housing and Urban Development Code 2008

The HUD Code requires that the ventilation system be balanced, and designed to exchange air directly with the exterior of the home. It specifically prohibits air drawn from the space underneath the home, through the floor, walls, or ceiling/roof systems. Except for high wind effects, positive pressure is not allowed in cold climates, and negative pressure is not allowed in warm-humid climates.

1.4.4.5 ASHRAE Standard 62.2-2013

ASHRAE Standard 62.2-2013 requires that supply and balanced ventilation systems draw outdoor air from a known fresh air location, but it does not include any requirement for exhaust ventilation systems. Therefore, makeup air for exhaust ventilation air comes from the paths of least resistance, which could be through a garage, attic, crawlspace, basement, or other soil contact location. To be consistent, the Standard would need to require intentional makeup air inlets, or require a supply system that provides makeup air from a known fresh air location whenever the whole-building exhaust ventilation system was operating.

1.5 Review of Building America Teams' Experience With Ventilation Approaches

As stated in the preceding review of codes and standards, ASHRAE Standard 62.2 whole-building ventilation does not account for shorter than annual average indoor air quality effects (such as odor, moisture, and sensory irritation), nor does it address system effectiveness factors related to outdoor air distribution, mixing with filtration, or source of outdoor air. Research conducted by the DOE Building America program has resulted in widespread use of systems that combine reduced rates with various elements of effectiveness.

Building Science Corporation (BSC) experience with whole-building controlled mechanical ventilation in tens of thousands of high performance homes in locations all across the United States has shown that drawing outdoor air from a known fresh air location, conditioning that air by filtration and sometimes heating or cooling, tempering that ventilation air by mixing it with central system return air, and fully distributing that air on at least an hourly average basis is a practical and effective way to mitigate odor complaints in all climates and an effective way to mitigate moisture buildup in mixed and cold climates. For more than 15 years, BSC builder partners have been installing systems capable of meeting more than ASHRAE Standard 62.2 ventilation rates, but typically running those systems at one third to one half that rate, resulting in satisfied builders and homeowners in both production and custom housing (Rudd and Lstiburek 1999, 2001, 2008). BSC attributes that satisfaction at the lower ventilation rates to the full distribution and whole-house mixing of outdoor air drawn from a known fresh air source with filtration (Hendron et al. 2006, 2007; Rudd and Lstiburek 2000; Townsend et al. 2009a, 2009b).

BA-PIRC (formerly BAIHP) worked with site and factory builders constructing custom, production, affordable, and multifamily homes to implement supply-based mechanical ventilation through the introduction of outdoor air into the return side of centrally ducted, forced-air, space conditioning systems. This approach, combined with rightsized heating/cooling systems and properly operating bathroom and kitchen exhaust fans (ducted to the outdoors) has been implemented in thousands of homes, primarily in the southeastern United States, since 1997 and has effectively controlled odors, maintained comfort, and proven effective at minimizing wintertime moisture buildup (Chandra et al. 2008). Similar to BSC's approach, these systems draw outdoor air from a known fresh air location, filter the air, temper the air by mixing it with central system return air, and fully distribute the air. Systems have been commissioned to deliver approximately 30%–70% of ASHRAE Standard 62.2-2010 rates, enough to create a slight positive pressure in the home with respect to outdoors; however, only while the central HVAC system is running to satisfy a heating or cooling requirement. Therefore, operation of the ventilation system is intermittent, especially during periods of limited to no HVAC runtime. In the Southeast, these periods typically coincide with increased natural ventilation through more frequent window operation, and the system has gained the acceptance of homeowners and builders alike in terms of comfort, durability, energy consumption, and perceived odor and moisture control. However, most of these systems do not meet the whole-house mechanical ventilation requirements of ASHRAE 62.2-2010.

1.6 Review of Other Previous Modeling Efforts

In the publication by BSC (Rudd and Walker 2007), a comprehensive set of simulations were performed to examine the relative operating costs and air change rates of various residential ventilation methods. The simulations included six major U.S. climate zones, three house sizes, two house types (standard-performance International Energy Conservation Code [IECC] and higher-performance), and 13 ventilation systems. The key results for the high performance houses (most comparable to the houses in this study) were:

- For both the Standard-Performance and Higher-Performance house, ASHRAE Standard 62.2 can be met using a simple exhaust-only system for an annual operating cost ranging from a savings of $225 to a cost of $150, all compared to a Standard-Performance house having approximately the same annual average air change through natural infiltration alone. The highest cost increase and the highest annual average air change increase occurred in the milder climates.
- For the same rated fan flow, the greatest increase in air change rate is for balanced systems (HRV/ERV), the least change is for exhaust-only systems, and supply-only systems are between those two.
- HRVs/ERVs sized to meet ASHRAE Standard 62.2, and as commonly installed to require coincident operation of the central air handling unit (AHU) fan, tend to require the greatest operating cost due to higher air change and fan energy consumption.
- Depending on climate and electricity cost, the cost to provide ventilation air distribution and thermal comfort mixing using a central fan cycling system at a 20 minute per hour minimum was between –$20 and +$90 compared to the same house in the same climate without the minimum central fan cycling. The higher costs were in hotter climates with more expensive electricity.
- In humid climates, the ASHRAE Standard 62.2 compliant ventilation systems increased the median RH by about 15% compared to the Standard-Performance house without mechanical ventilation. It was clear that, with or without mechanical ventilation, supplemental dehumidification would be required to control elevated indoor RH year-round.

ASHRAE Research Project (RP) 1449 (Rudd et al. (2013) completed a comprehensive simulation analysis of energy and humidity control performance of various options in humid climates for residential buildings with and without ventilation. Results show the effects of duct location, thermostat set points, and modeled moisture capacitance on humidity control:

- Duct location has a major influence on humidity control, and locating the air distribution system ducts inside conditioned space saves energy overall, but, with the reduced sensible cooling load, and hence reduced latent capacity, also comes an increased need for supplemental dehumidification in warm-humid climates. Moving the ducts inside conditioned space compared to ducts in the attic with 5% leakage (3% supply side, 2%

return side) increases the hours above 60% RH indoors by 30%–50% for a DOE Challenge Home. That was simulated without accounting for the moisture desorption from wood framing materials that typically increases the attic humidity ratio over that of the outdoors during the late morning to early afternoon hours in warm-humid climates (about 10°F dew-point temperature over outdoors from 10:00 a.m. to 1:00 p.m. between May 15 and October 15). That effect would tend to lessen the difference, since return duct leakage with higher attic humidity would increase the moisture source for the ducts-in-attic configuration. However, additional simulations done on this as a sensitivity study showed that the results were nearly the same. The reason is that most elevated indoor RH hours occur in night and early morning hours between late November and March. Some nighttime and rainy periods during mild summer conditions also produce elevated indoor RH.

    - Similar to ducts in hot attics, very high internal sensible heat gain experienced by older homes (simulated at 21 kWh/day) drives the cooling system to operate more often and for longer runtimes, reducing indoor humidity. High performance homes, with compact fluorescent lamps or light-emitting diodes and ENERGY STAR-rated appliances, have lower internal sensible heat generation than older homes (simulated at 15 kWh/day), further complicating humidity control.

- Thermostat set points have a major influence on humidity control. For the warm-humid climate, shifting the heating and cooling thermostat set points up by 2°F (to 73°F and 78°F, respectively) would tend to increase the hours above 60% RH at the edge of cooling demand due to less space conditioning runtime, and decrease the hours above 60% RH at the edge of heating demand. Raising the heating set point has a significant impact on reducing indoor RH (even though it does nothing to reduce the absolute humidity) because it keeps the air from getting as cold, stopping the rise in RH.

- Moisture capacitance is a secondary factor in increasing hours above 60% RH, and moisture capacitance factors of 10–30 times the air mass capacity do not show significant difference in indoor RH.

2 Technical Approach for Assessment of Energy and Indoor Humidity Impacts of Residential Ventilation

This study was based on building energy modeling with an important focus on indoor humidity impacts. The modeling tools used were EnergyPlus version 7.1 (E+) and EnergyGauge USA (EGUSA). Twelve U.S. cities and five climate zones were represented. A total of 864 simulations (2*2*2*3*3*12 = 864) were run using two building archetypes, two building leakage rates, two building orientations, three ventilation systems, three ventilation rates, and 12 climates. For both the E+ and EGUSA simulations, custom scripts were written to automate parameter modification for the hundreds of simulations, which could be run overnight. The following sections explain the details of these parametric inputs.

The BEopt program (E+ version) was used to generate the building geometry and the base IDFs (input data files) for the E+ simulations. Figure 4 shows the building geometry for the two-story house.

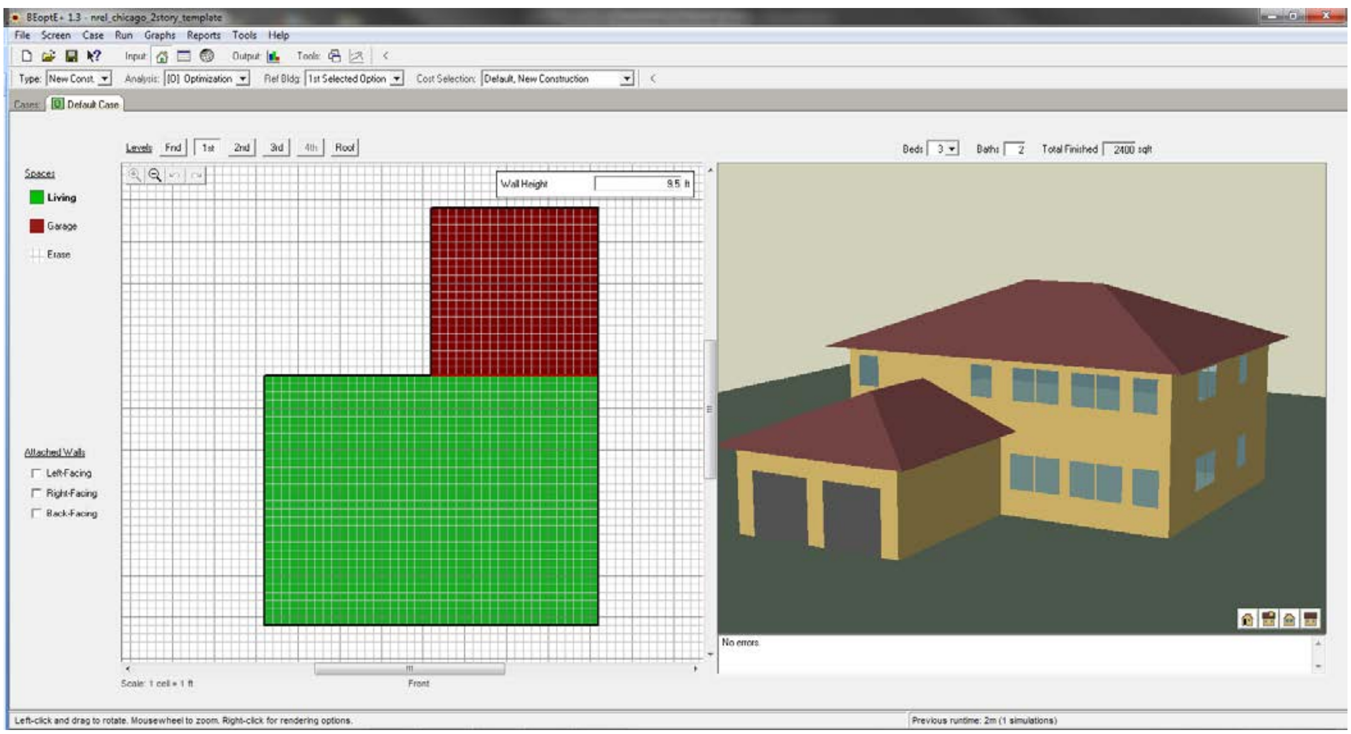


Figure 4. NREL BEopt E+ program used to generate the building geometry and the base IDF files (input description)

The EGUSA program is based on the DOE-2 hourly simulation engine and was a custom version for this project (V3.0.01P). Some of the custom routines were designed to improve the modeling of indoor humidity while using the temperature-only control capability of DOE-2. Latent degradation due to evaporation of moisture from wet cooling coils during cooling system off cycles (Shirey et al. 2006) was not modeled (moisture added back to the ducts and conditioned space when the cooling compressor is off and fan remains on).

2.1 Residential Whole-Building Ventilation Systems Evaluated

2.1.1 Exhaust Only

Exhaust whole-house ventilation systems expel inside air directly to outdoors, tending to depressurize the interior space relative to outdoors (Figure 5). Exhaust systems draw outdoor air from whatever building enclosure leaks create the path of least resistance. The exhaust ventilation system modeled here was a single-point exhaust system such as a high quality exhaust fan installed in a master bathroom, family bathroom, toilet room, or laundry room.

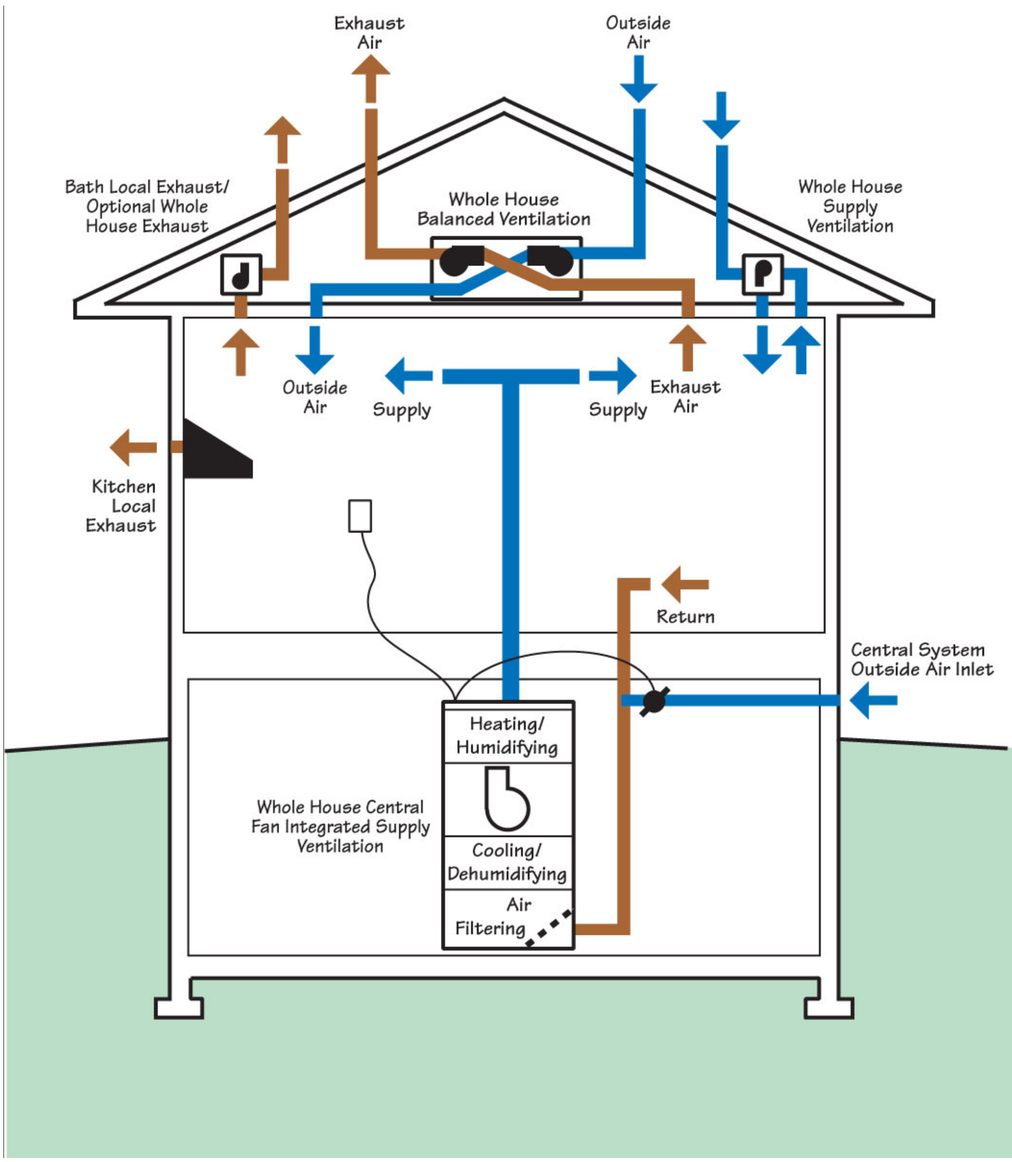


Figure 5. Schematic overview of typical local and whole-house ventilation equipment (a home typically has only one whole-house system)

(Rudd 2011)

2.1.2 Supply Only

Supply whole-house ventilation systems draw outdoor air from a known location and deliver it to the interior living space. This known location should be selected to maximize the ventilation air quality. The air can be treated before being distributed to the living space (heated, cooled, dehumidified, filtered, or cleaned).This study used a central fan integrated (CFI) supply ventilation system that provides ventilation air through a duct that extends from outdoors to the return air side of the central heating and cooling system AHU, with a motorized outdoor air damper, and with an automatic timer control to ensure ventilation air is periodically supplied when heating and cooling have been inactive and to limit outdoor air introduction to a maximum regardless of how long the fan operates.

2.1.3 Balanced Heat Recovery Ventilation and Energy Recovery Ventilation

Balanced whole-house ventilation systems both exhaust and supply in roughly equal amounts. Inside air is exhausted to the outdoors and outdoor air is supplied indoors. In current practice, most balanced ventilation systems are HRVs or ERVs. Balanced ventilation systems with HRVs use a heat exchanger to transfer heat between the exhaust air stream and the outdoor air supply stream. With HRVs, no moisture is exchanged between the air streams. This means that in cold months, the heating load due to ventilation will be less, and in hot months, only the sensible cooling load due to ventilation will be less.

The balanced ventilation system modeled here is an ERV. ERVs operate the same as HRVs with the exception that both heat and moisture are exchanged between the exhaust air stream and the outdoor air supply stream. This means that in cold, dry months, the heating load due to ventilation will be less, and the house indoor moisture level will be higher than it otherwise would have been without energy recovery. In hot, humid months, the total cooling load (both sensible and latent) due to ventilation will be less. While less heat and moisture will come in from outdoors, an ERV can neither cool nor dehumidify the interior space. A good way to think of this is that the heat and moisture tend to remain on the side from which they came. Another important point about using ERVs in humid climates is that, at times of the year when indoor humidity is the highest, there is usually a small difference between indoor and outdoor humidity, minimizing the latent exchange effect of the ERV.

The ERV modeled here was configured with ducts completely separate from the central space conditioning system. Therefore, no central AHU operation was needed coincident with the ERV operation as is the case in the majority of installations.

2.2 Modeling Assumptions

2.2.1 Constants

2.2.1.1 No Window Ventilation, Only Mechanical Ventilation and Natural Infiltration

The models assumed that windows were closed the entire year. Window openings were not included in the simulations because they could easily confound the results. The worst case scenario would be that occupants may choose not to open windows for a variety of reasons. In addition, the simulations assumed that there was no added infiltration induced by duct leakage to the outside. This assumption was made for similar reasons of confounding the results as well as the goal that duct leakage to the outdoors should be effectively eliminated in a DOE Challenge Home.

2.2.1.2 Infiltration Model and Combined Airflow

The infiltration rate in EGUSA is simulated using the Sherman-Grimsrud model and Shelter class 4, and is simulated in E+ using the Walker Wilson model. Unbalanced mechanical ventilation airflow and the infiltration airflow are added in quadrature (by summing the squares, and then taking the square root). Balanced mechanical ventilation airflow and infiltration airflow are summed directly. The combined airflow of infiltration and balanced and unbalanced ventilation was accounted for.

2.2.1.3 Thermal Enclosure Specifications

The objective was to make each simulated home qualify for the DOE Challenge Home. The DOE Challenge Home is generally a combination of EPA's ENERGY STAR version 3, Indoor airPLUS, and Water Sense requirements, and the IECC 2012 requirements, resulting in a Home Energy Rating System Index in the low- to mid-50s range. The Home Energy Rating System Index for the homes being simulated here fell in the mid- to upper-50s range depending on climate zone.

2.2.1.4 Space Conditioning Equipment

Following the requirements of the DOE Challenge Home, the space conditioning air duct system and AHU were located in conditioned space with zero leakage to the outdoors. The space conditioning equipment efficiency matched the DOE Challenge home requirements.

2.2.1.5 Ventilation System Airflow and Power

The mechanical ventilation airflow met the requirement of ASHRAE Standard 62.2-2013 every hour. Exhaust fan power was set to 0.3 Watts/CFM. The ERV airflow was left slightly unbalanced, with a 10% difference between supply and exhaust, with the larger value set equal to the Qfan requirement (regarding Qfan, refer to the Section 2.2.2.6). The ERV fan power was set equal to 0.5 W/CFM*(supply CFM + return CFM). ERV effectiveness was set to 60% per the DOE Challenge Home criteria. The CFI supply ventilation duty cycle was set to 33% for both the minimum and maximum operation times, and only AHU energy in excess of that needed for heating and cooling was counted as ventilation energy. The CFI outdoor airflow was set equal to Qfan/duty cycle fraction.[1] AHU fan power was set at 0.5 Watts/CFM for SEER < 14 (assuming permanent split capacitor fan at > 0.5 in. water column external static pressure), and 0.375 watts/CFM for SEER >14 (assuming electronically commutated motor fan). Ventilation fan heat was added to the indoor space for supply ventilation air stream (CFI and supply side of ERV) but not for exhaust ventilation air stream (exhaust fan and exhaust side of ERV).

2.2.1.6 Building Geometry and Glazing Area

A 15% window-floor-area ratio was used for all homes.

2.2.1.7 Internal Heat and Moisture Generation and Schedule

Internal gains for all homes were set at: sensible heat gain = 57,717 Btu/day, and latent heat gain = 12.09 lb/day (12,698 Btu/day). The hourly schedule for both heat and moisture generation is shown in Figure 6.

[1] The CFI systems modeled in this study met the airflow requirements of ASHRAE 62.2-2013. CFI systems are typically not operated with airflows required by ASHRAE 62.2, due to various practical reasons, although there is nothing inherent with this system to preclude that. The system is included in this modeling study, as it is a popular system with builders for some reasons not accounted for in 62.2.

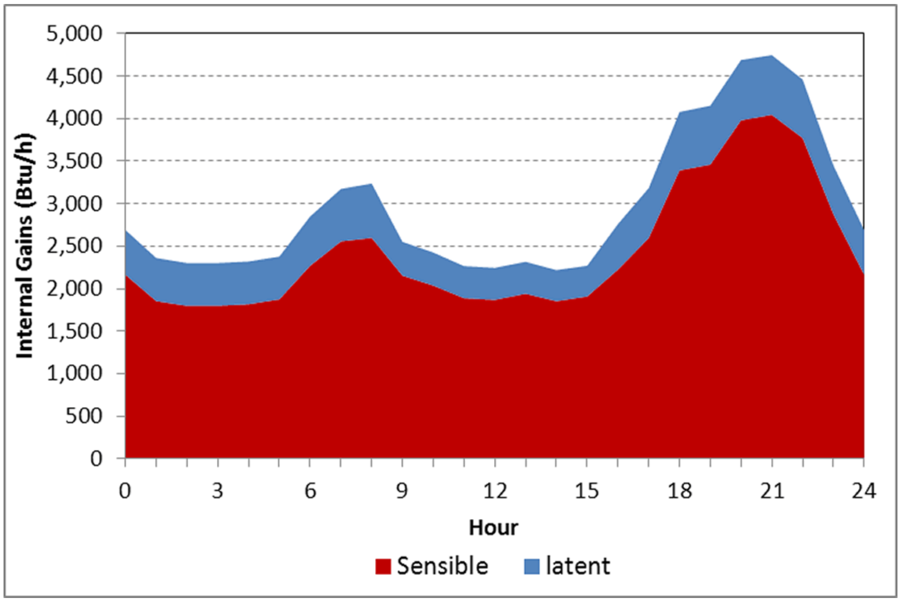


Figure 6. Hourly internal gains schedule

2.2.1.8 Internal Moisture Capacitance

Indoor moisture capacitance was simulated using an air mass multiplier of 10 (i.e., the indoor air mass for moisture storage purposes is increased by 10 times).

2.2.1.9 Temperature Control Set Points

Thermostat temperature control set points were: heating = 71°F and cooling = 76°F.

2.2.1.10 Indoor Humidity Control Set Point

The indoor humidity control set point was 60% RH.

2.2.1.11 Electricity and Gas Utility Costs

Electricity and gas utility costs were applied equally for all cities as $0.12/kWh and $1.20/therm (1 therm is approximately equal to 1 ccf or 100 ft^3 of gas at standard pressure and temperature). Figure 7 shows those rates to be generally reasonable assumptions according to the U.S. Energy Information Agency data for 2011.

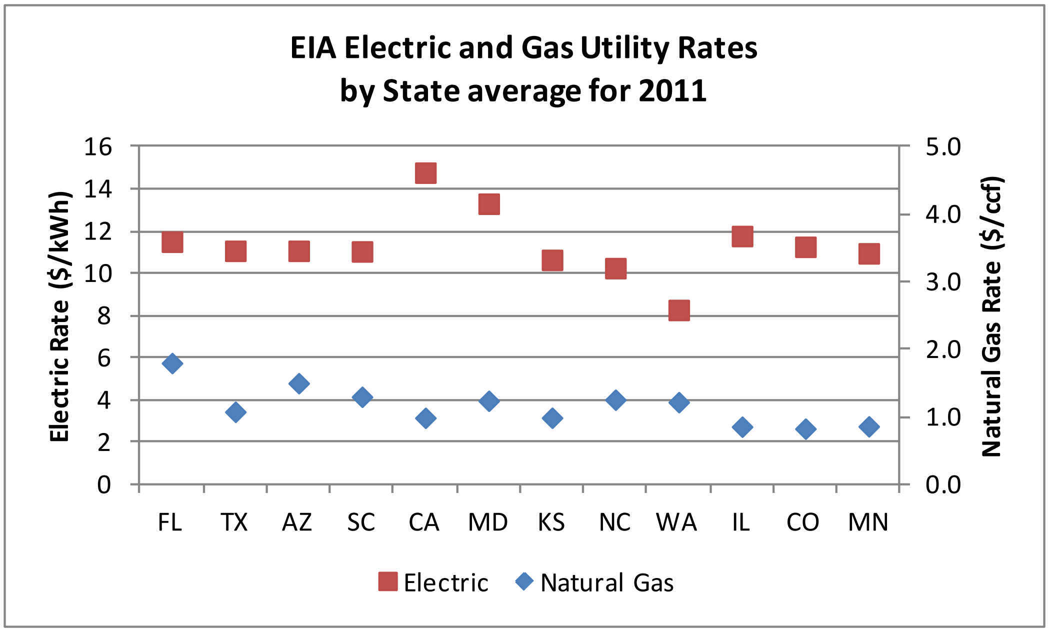


Figure 7. Average residential electricity and gas utility rates by state for 2011

(Source: EIA.gov)

2.2.2 Variables

2.2.2.1 U.S. Climates/Cities

- **Warm-humid zones (2A, 3A):** Orlando, Florida; Houston, Texas; Charleston, South Carolina
- **Mixed-humid zones (4A):** Baltimore, Maryland; Kansas City, Missouri; Charlotte, North Carolina
- **Cold-humid zones (5A and 6A):** Chicago, Illinois; Minneapolis, Minnesota
- **Dry zones (2B and 5B):** Phoenix, Arizona; Denver, Colorado
- **Marine zones (3C and 4C):** Los Angeles, California; Seattle, Washington.

2.2.2.2 House Type and Size

Two house types and sizes were simulated as follows:

- 2000-ft^2, one-story, three-bedroom, slab-on-grade, wood frame
- 2400-ft^2, two-story, three-bedroom, slab-on-grade, wood frame.

2.2.2.3 Glazing Orientation

The worst-case and best-case solar gain orientations were simulated to evaluate the impact that could have on indoor humidity levels. The indoor humidity levels would be impacted due to more or less cooling demand driven by solar gain.

2.2.2.4 Building Airtightness

Buildings with airtightness of 1.5 ACH50 and 3.0 ACH50 were simulated.

2.2.2.5 Ventilation System Type

Exhaust-only, supply-only, and balanced HRV/ERV ventilation systems were simulated.

2.2.2.6 Mechanical Ventilation Rate

Three mechanical ventilation rates were evaluated: 100%, 75%, and 50% of ASHRAE 62.2-2013. Modeling of the reduced rates is not to suggest that such rates provide adequate indoor air quality—that is outside the scope of this report. The rates are chosen purely to evaluate the energy and moisture impacts of varying rates. The 75% rate was chosen as it closely represents ASHRAE 62.2-2010 prior to the recent 62.2-2013, and the 50% rate was chosen as representative of what many Building America production builder partners have historically incorporated in various research and demonstration homes. The ventilation fan flow rate is climate dependent because a credit for measured infiltration is allowed as follows:

Qfan = Qvent – Qinfil

Qvent = 0.03*CFA + 7.5*(Nbr+1)

Qinfil = climate dependent calculation based on procedures specified by ASHRAE 62.2-2013, where Qinfil cannot be more than two thirds of Qvent

On average, for all sets of ACH50 conditions, the fan flow rates required by 62.2-2013 compared with 62.2-2010 requirements as follows:

100% of 2013 ≈ 140% of 2010

75% of 2013 ≈ 94% of 2010

50% of 2013 ≈ 48% of 2010

Table 4. Comparison of Fan Flows Required by ASHRAE 62.2-2010 and 2013

			2-story, 62.2-2010 fan cfm=54				1-story, 62.2-2010 fan cfm=50			
			3.0 ach50		1.5 ach50		3.0 ach50		1.5 ach50	
CLIMATE ZONE	**LOCATION**	**ASHRAE WSF***	**62.2-2013 fan cfm**	**% diff from 62.2-2010 fan cfm**	**62.2-2013 fan cfm**	**% diff from 62.2-2010 fan cfm**	**62.2-2013 fan cfm**	**% diff from 62.2-2010 fan cfm**	**62.2-2013 fan cfm**	**% diff from 62.2-2010 fan cfm**
Warm-Humid Zone	Orlando, FL	0.39	73	35%	88	62%	71	42%	81	61%
Warm-Humid Zone	Houston, TX	0.40	72	34%	87	61%	71	41%	80	61%
Warm-Humid Zone	Charleston, SC	0.43	70	30%	86	59%	69	38%	80	59%
Mixed-Humid Zone	Baltimore, MD	0.50	65	20%	83	55%	66	31%	78	56%
Mixed-Humid Zone	Kansas City, MO	0.60	58	7%	80	48%	61	22%	75	51%
Mixed-Humid Zone	Charlotte, NC	0.43	70	30%	86	59%	69	38%	80	59%
Cold-Humid Zone	Minneapolis, MN	0.63	55	2%	79	46%	59	19%	75	49%
Cold-Humid Zone	Chicago, IL	0.60	58	7%	80	48%	61	22%	75	51%
Dry Zone	Phoenix, AZ	0.43	70	30%	86	59%	69	38%	80	59%
Dry Zone	Denver, CO	0.61	57	5%	79	47%	60	21%	75	50%
Marine Zone	Los Angeles, CA	0.42	71	31%	86	60%	70	39%	80	60%
Marine Zone	Seattle, WA	0.56	61	12%	81	50%	63	26%	76	53%
		average of climates:	65	20%	83	55%	66	31%	78	56%

3 Modeling Results

3.1 Energy Performance

3.1.1 Energy Performance as a Function of Ventilation Rate

Removal of indoor moisture by supplemental dehumidification was approximated from the EGUSA simulations by a post-processing routine that assumed that during hours over an RH limit, the net moisture gain during that hour due to air exchange and internal moisture generation would have to be removed. That strategy is only an approximation because the heat and mass transfer interactions that a real-time dehumidifier would have on the space conditions and the cooling system are not accounted for. For example, while a dehumidifier is operating, it is removing moisture and adding heat to the space, both of which drive the RH down. Additionally, heat added from the dehumidifier may raise the space temperature enough to cause the cooling system to come on, which then will also remove moisture. Those interactions are not accounted for in the EGUSA model. They are accounted for in the E+ model; however, they were found to have a minor effect on annual space conditioning energy use.

The predicted lb/yr values from the EGUSA modeling were converted to $/yr by an assumed dehumidifier energy factor of 1.47 L/kWh and $0.12/kWh. Based on that, supplemental dehumidification was predicted to be $10–$30/yr for the warm-humid climates of Charleston, Houston, and Orlando, and the marine climate of Los Angeles. The E+ modeling yielded similar results.

Figure 8 shows the total annual operating cost for a DOE Challenge Home controlled to < 60% RH over a range of ventilation rates from 100% of the ASHRAE 62.2-2013 requirement to 75% and 50% of that. The 75% ventilation rate values generally correspond to the 62.2-2010 rates. As shown in Figure 9, that difference is exclusively space conditioning energy use, and is about $45/yr averaged over the climates, enclosure tightness and ventilation systems simulated, or a savings of 10% of total space conditioning energy. Dropping to 50% of the ASHRAE 62.2-2013 rate can save about another $35/yr on average for a total average savings of 15% of space conditioning energy.

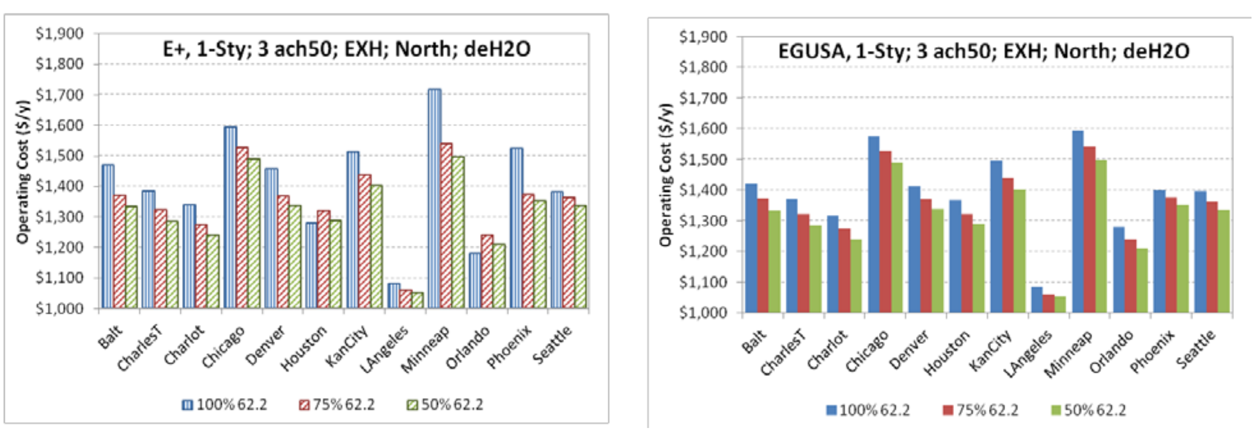


Figure 8. Total annual operating cost for a DOE Challenge Home controlled to < 60% RH over a range of ventilation rates

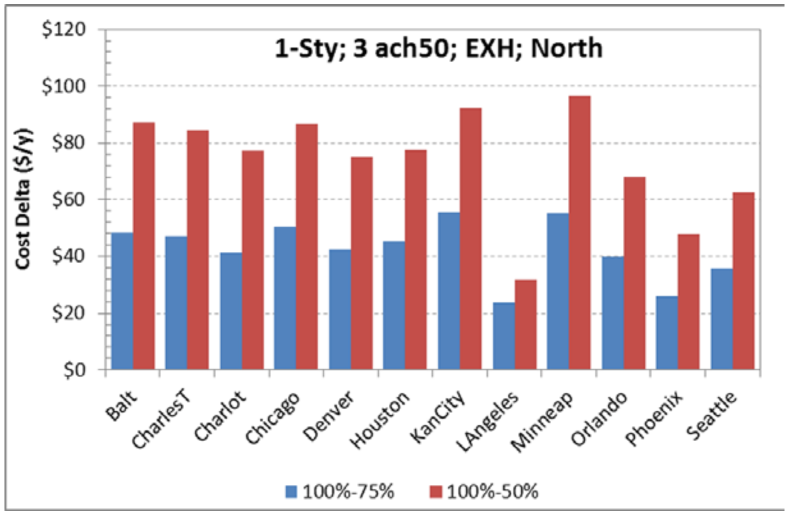


Figure 9. Total annual space conditioning cost difference for a DOE Challenge Home controlled to < 60% RH over a range of ventilation rates using EGUSA

3.1.2 Energy Performance as a Function of Ventilation System Type

Total annual operating cost per year for a DOE Challenge Home varied less than $90/yr among all ventilation systems in every climate (Figure 10). Compared to exhaust ventilation as a reference, EGUSA results show balanced ERV ventilation (installed such that it does not require coincident AHU fan operation to avoid short-circuiting of ventilation air) ranged from $0/yr to $40/yr more, with an average of about $20/yr more for 100% of the ASHRAE 62.2-2010 requirement at 3 ACH50. However, at greater mechanical ventilation rates required in the 1.5 ACH50 house,[2] the ERV *saved* an average of $20/yr over exhaust ventilation, in all but the dry climate of Phoenix and the mild climate of Los Angeles, where it continued to consume more energy than exhaust (not shown). E+ results show savings from balanced ERV ventilation compared to exhaust ventilation at 3 ACH50, primarily in cold/mixed climates. The cause of the discrepancy between E+ and EGUSA is unknown. CFI supply ventilation ranged from about $11/yr to $93/yr more than exhaust ventilation, with an average of about $40/yr.

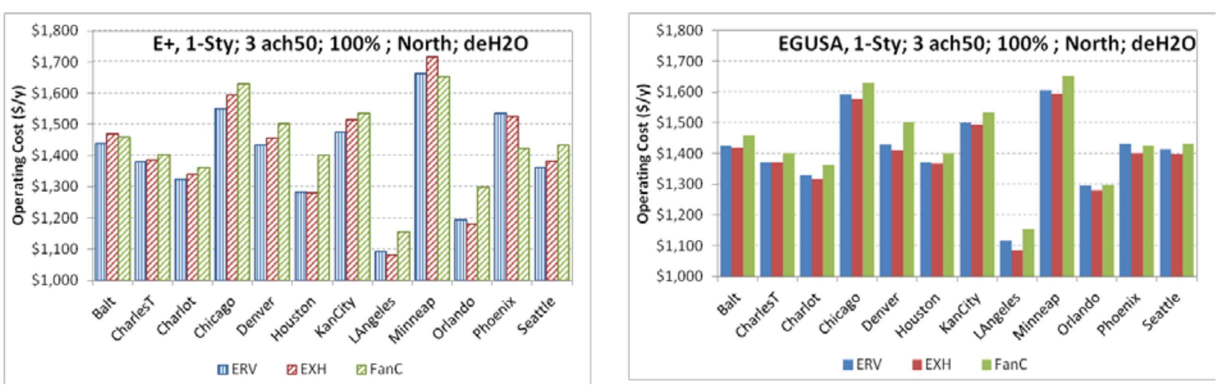


Figure 10. Total annual operating cost per year for a DOE Challenge Home over a range of ventilation system types

[2] The weather-dependent, natural component of the total ventilation requirement is smaller in the 1.5 ACH50 house. Therefore, a larger continuous, mechanical component is required.

3.1.3 Air Change Rates as a Function of Building Air Leakage

For the three ACH50 houses, air change rates were slightly higher (about 0.01 ACH) for one-story, unbalanced ventilation system houses than for two-story houses. Air change rates were mostly the same between one- and two-story, balanced ventilation system houses. The difference between balanced and unbalanced ventilation was about 0.05 ACH, with balanced being higher as expected.

For the 1.5 ACH50 houses, air change rates were about 0.02 ACH higher for one-story, unbalanced ventilation system houses than for two-story houses, and were about 0.01 ACH higher for one-story houses with balanced ventilation.

The difference in air exchange rate between balanced and unbalanced ventilation was about 24% for the 3.0 ACH50 houses and about 10% for the 1.5 ACH50 houses (Figure 11).

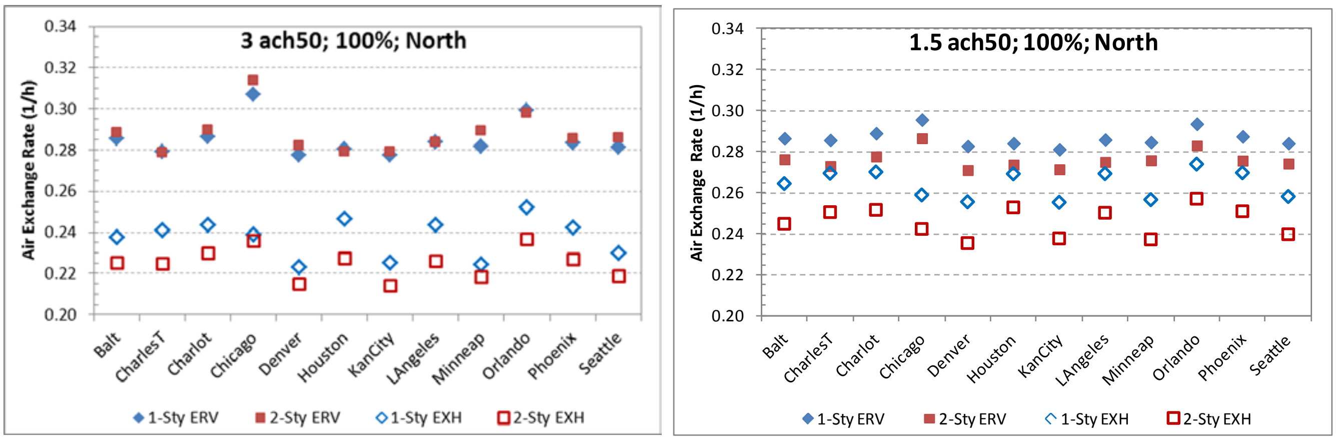


Figure 11. EGUSA differences in air exchange rate between balanced and unbalanced ventilation

Figure 11 shows that for unbalanced ventilation, total air exchange increases as building leakage decreases. The reason is that the ventilation and infiltration flows were combined in quadrature in the simulations while the required mechanical ventilation was calculated using the ASHRAE procedure, which is a simple subtraction of the estimated infiltration rate from the total ventilation rate requirement. As a result, as the envelope leakage is reduced, the portion of the total ventilation rate coming from mechanical ventilation increases. Simultaneously, the use of quadrature to combine the ventilation and infiltration rates in the simulation model causes the infiltration contribution to be discounted more in the 3 ACH50 home than in the 1.5 ACH50 home, resulting in a slightly greater total air exchange rate for unbalanced ventilation in the tighter home.

3.2 Humidity Control Performance

The E+ model results were compared to the EGUSA model results, and initially some input differences were corrected. However, some result differences still exist and remain mostly unexplained. Figure 12 shows an example of the E+ results generally showing a greater number of hours over 60% RH indoors, and hours that extend to higher RH, compared to the EGUSA results and much of BSC's field data (Rudd et al. 2003, 2005; Rudd and Henderson 2007). These might be due to differences in the infiltration model. There may also be differences in the thermal energy balance model, which could lead to different loads, as another possibility. The E+ moisture modeling inputs were double checked without finding anything significantly wrong. The moisture capacitance factor and moisture generation rate were consistent with the EGUSA inputs. While there is a noticeable difference between the two models, the trends are for the most part very similar.

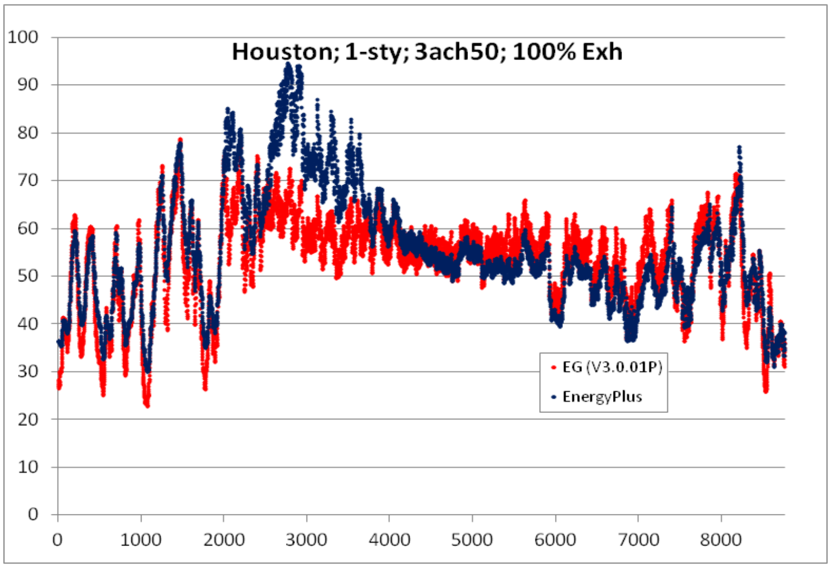


Figure 12. Indoor RH versus hour of year comparison between E+ (blue symbols) and EGUSA (V3.0.01P) simulation results

3.2.1 Humidity Control Performance as a Function of Space Conditioning System Activity

Figure 13 illustrates the indoor RH response with cooling load using EGUSA. The expected space conditioning equipment operation, and the temperature and RH response, are shown.

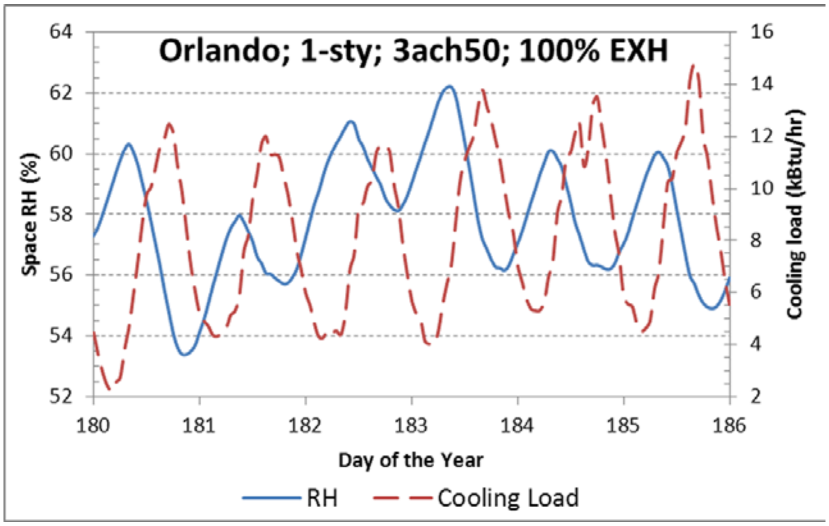
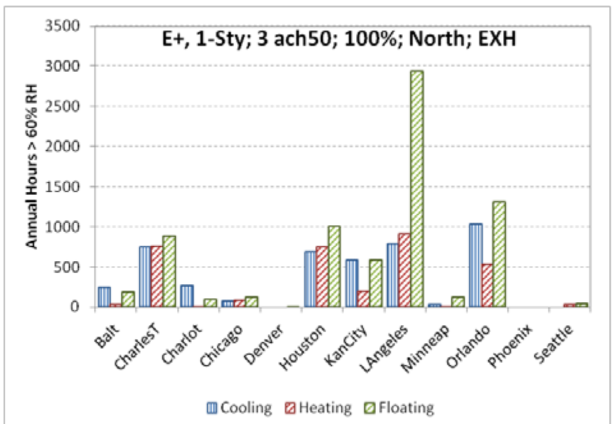


Figure 13. For a week at the end of June and beginning of July on Orlando, the cooling system and space RH are showing an expected response using EGUSA

The hours above 60% RH during a particular space conditioning mode (heating, cooling, floating) was a strong function of climate. Floating hours are times when no space conditioning is active because the house remains between the heating and cooling control set points. The EGUSA results in Figure 14 show that the hours above 60% RH during heating are below 200 in every climate, but the hours above 60% RH while floating can be equal to or significantly greater than the hours above 60% RH during cooling and that is very dependent on the climate. For example, in Orlando, the hours above 60% RH are nearly equal between cooling hours and floating hours, but in Charleston and Houston, the hours above 60% RH are 35%–50% greater for floating hours. However, the cooling hours above 60% RH are generally < 65% RH while, in the warm-humid climates, the floating hours are 5%–15% RH above 60% RH.



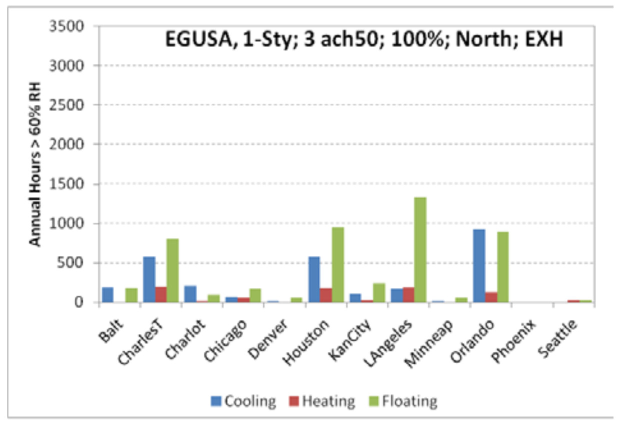


Figure 14. Hours of elevated indoor RH is a strong function space conditioning system activity and climate

The E+ results, while having a greater number of hours over 60% RH, show similar trends compared with the EGUSA results. The exception is in Los Angeles, where the floating hours are more than double those predicted by the EGUSA model.

3.2.2 Humidity Control Performance as a Function of Infiltration Rate

Referring to Figure 15, mechanical ventilation, operated at the ASHRAE 62.2-2013 rate, in a 3 ACH50 DOE Challenge level house, raises the annual median indoor RH by almost 10% compared to a 7 ACH50[3] house without mechanical ventilation in Orlando. That is because infiltration drivers are quite seasonally dependent with greater infiltration rates in winter and smaller infiltration rates in summer, but mechanical ventilation forces a minimum air exchange year round. Note that most of the RH increase due to ventilation is 60%–65% RH; the hours between 65% and 75% RH mostly remain the same. This indicates that some supplemental dehumidification would be needed in either case to maintain RH below 60%.

[3] High performance houses built to 2012 code or the DOE Challenge Home program would likely not be built with natural infiltration > 5 ACH50.

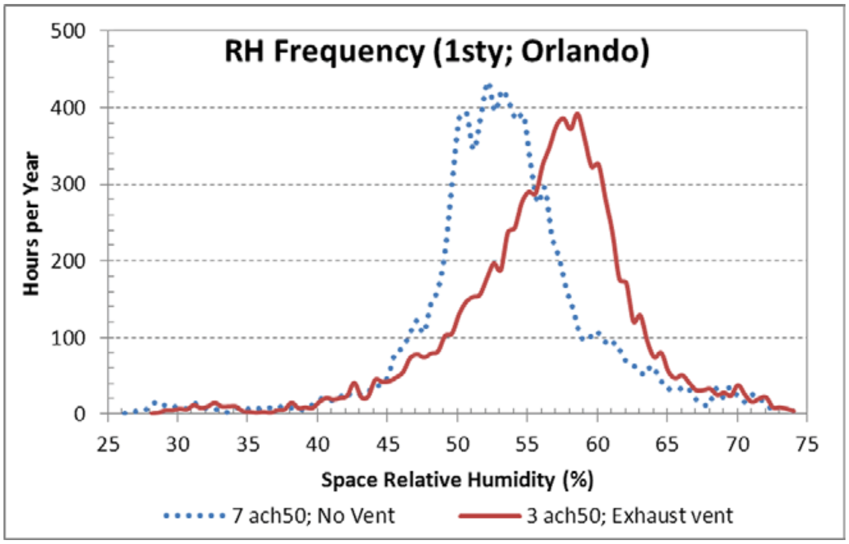


Figure 15. ASHRAE 62.2-2013 ventilation rate raises median RH compared to a conventional dwelling without mechanical ventilation using EGUSA

3.2.3 Humidity Control Performance as a Function of Ventilation System Type

Figure 16 shows analysis of simulation results for exhaust and ERV ventilation in Orlando. The hours above 60% RH are distinguished by heating, cooling, and floating hours. A heating hour was where any heating occurred during that hour, a cooling hour was where any cooling occurred during that hour, and a floating hour was an hour where no heating or cooling occurred during that hour.

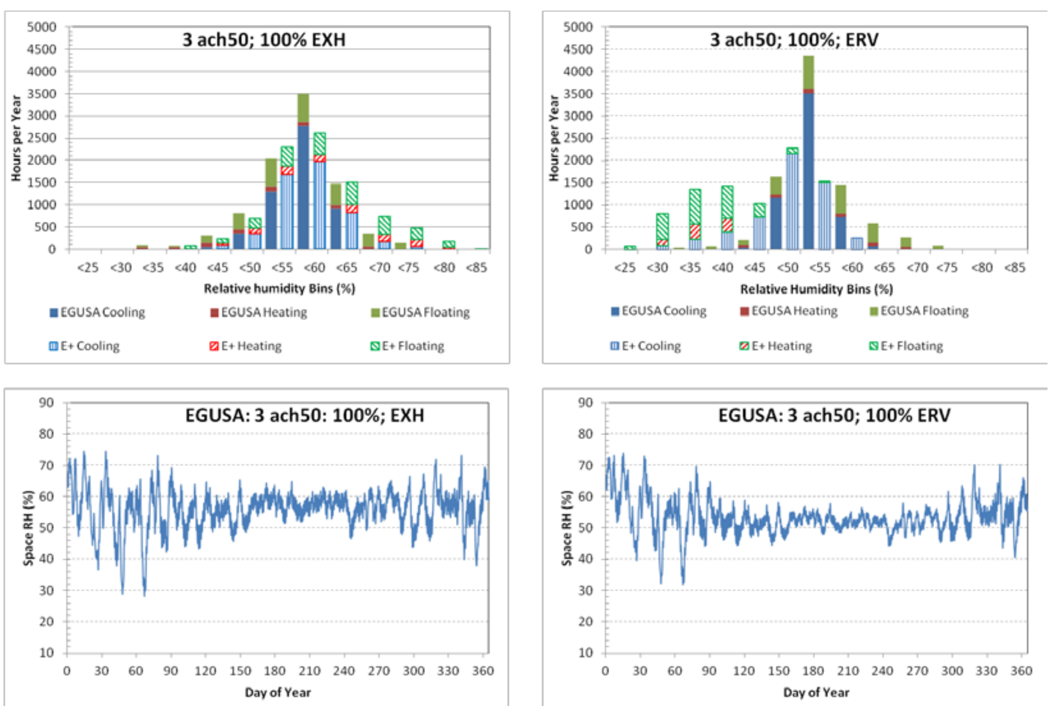


Figure 16. Hours above 60% RH for exhaust and ERV ventilation in Orlando

These data demonstrate the difference between the results from E+ and EGUSA. The exhaust-only cases are similar, however E+ results indicate more hours above 60% RH and hours with higher RH compared to EGUSA. The ERV case shows that E+ predicts much lower RH due to the use of an ERV than EGUSA. EGUSA also predicts most of the hours significantly above 60% RH occur during floating hours, which occur mostly during fall, winter, and spring in warm-humid climates, and sometimes during summer nights and summer extended rainy periods where sensible heat gain is lowered and internal moisture generation stays the same. EGUSA shows that in Orlando, the ERV nearly eliminates the cooling hours above 60% RH, but it has little effect on the floating hours above 60% RH. E+ shows that the floating hours are well below 60% RH, and that most hours above 60% RH are cooling hours. The E+ model is suspect, as the cooling system forces a greater humidity difference between the indoors and outdoors, which makes the ERV transfer more moisture to the exhaust stream. However, even though the ERV was modeled with a constant 60% effectiveness, meaning that 60% of the moisture from the higher humidity side would be transferred to the lower humidity side, 60% of a small humidity difference is still a small amount of moisture. The EGUSA model shows that the ERV is ineffective in keeping indoor RH down during floating hours when the difference between indoor and outdoor absolute humidity is small, as would be expected. The ERV can be slightly counterproductive in winter in Orlando and Houston due to recovery of indoor moisture when humidity is lower outdoors than indoors.

Figure 17 illustrates the annual hours above 60% RH by ventilation system type. The EGUSA results from the three warm-humid climates of Charleston, Houston, and Orlando stand out with more hours of elevated indoor RH. ERV ventilation generally showed one third to one half fewer hours above 60% RH compared to CFI supply and exhaust ventilation. The CFI system shows slightly fewer hours above 60% than the exhaust system because it is a supply ventilation system that adds fan heat to the space, increasing the call for cooling, which removes additional moisture.

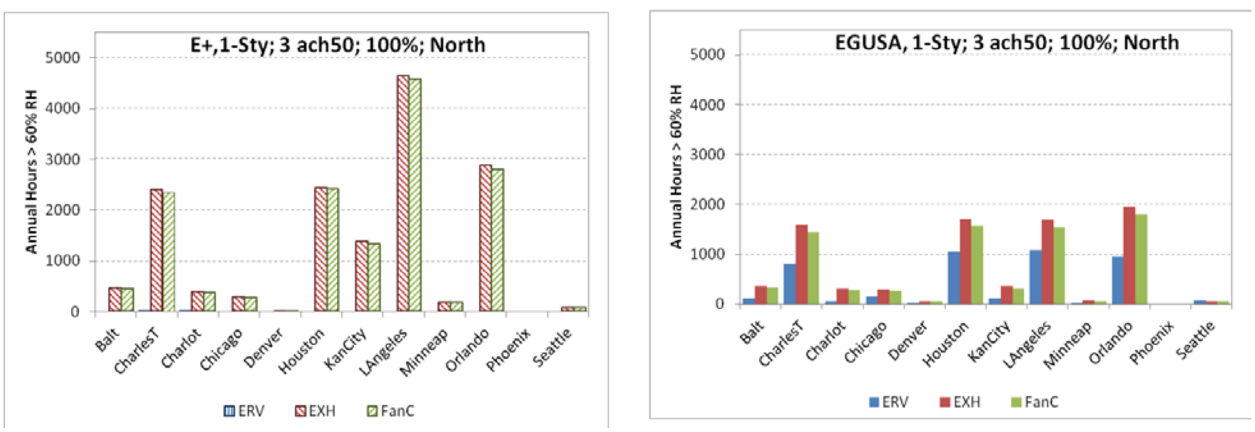


Figure 17. Annual hours above 60% RH by ventilation system type

The E+ results show that an ERV appears to virtually eliminate the number of hours with RH above 60% compared to the EGUSA results. The amount of reduction in elevated RH by the E+ model is suspect, but the cause is unclear. For both models, the ERV moisture performance was modeled pretty simply as a constant latent recovery effectiveness value applied to the actual indoor to outdoor humidity ratio difference. The EGUSA and E+ models configured the ERV as separately ducted from the central air distribution system.

If the ERV was configured as being connected between the return and supply of the central AHU (as most ERVs are to reduce installation cost), requiring that the AHU operate coincident with the ERV, it is expected that the hours above 60% RH would be higher due to moisture evaporation from the cooling coil.

3.2.4 Humidity Control Performance as a Function of Building Air Leakage

With unbalanced mechanical ventilation, total air change goes up as the building gets tighter and leakage decreases from 3.0 ACH50 to 1.5 ACH50. This is because the variable, natural component is smaller, and the consistent, mechanical component is larger. The result of that is shown in Figure 18 where the amount of supplemental dehumidification needed increases as the building air leakage decreases.

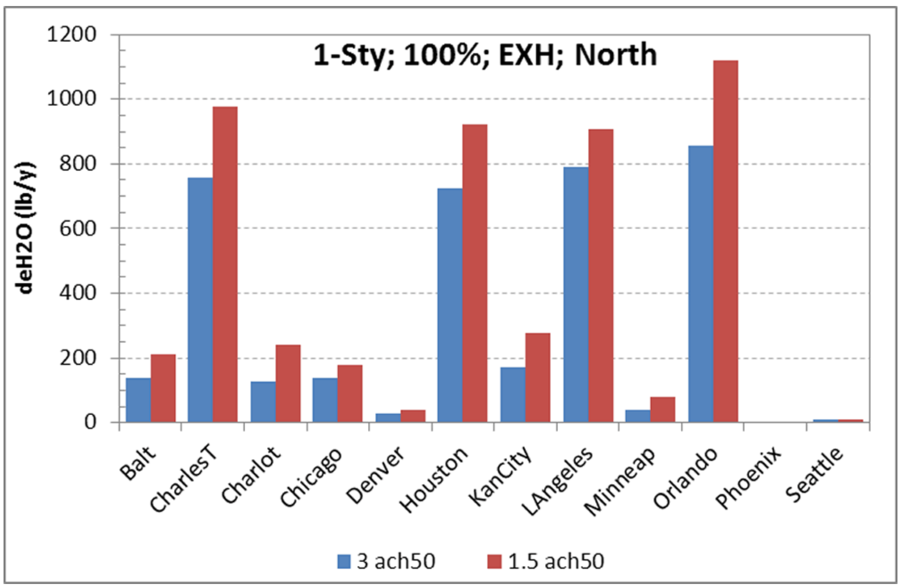


Figure 18. For unbalanced ventilation, the supplemental dehumidification requirement increases as the building air leakage decreases using EGUSA

3.2.5 Humidity Control Performance as a Function of Internal Moisture Generation

Internal moisture generation (latent gain) strongly impacts the predicted hours of elevated indoor humidity. Moisture generation of 12 lb/day produces indoor humidity results that seem to fit reasonably well with monitored data for three- to four-bedroom dwellings. More than twice that is a worst-case peak design value suggested by ASHRAE Standard 160. While that may be useful for sizing considerations in extreme cases, it is not useful for annual simulation of indoor humidity and supplemental dehumidification energy consumption. Figure 19 illustrates the sensitivity of the hours above 60% RH to an internal moisture generation change from 12 lb/day to 15 lb/day.

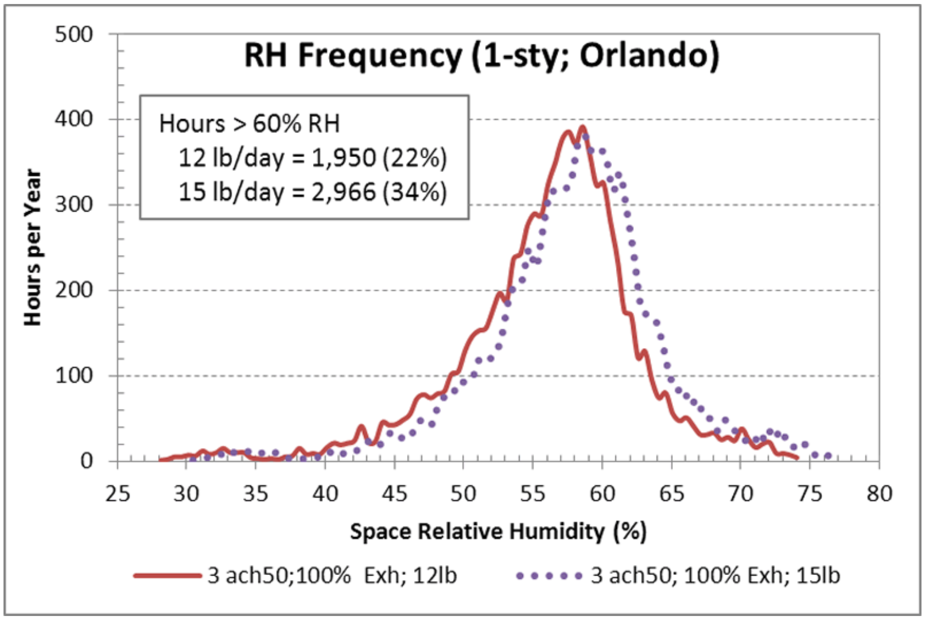


Figure 19. Effect of increasing internal moisture generation from 12 to 15 lb/day using EGUSA

3.2.6 Humidity Control Performance as a Function of Relative Humidity Set Point

Figure 20 shows that the hours of elevated indoor RH are a strong function of the selected RH limit and climate. For example, in Los Angeles nearly all of the hours above 60% RH are also below 65% RH, but that is not true for Charleston, Houston, and Orlando.

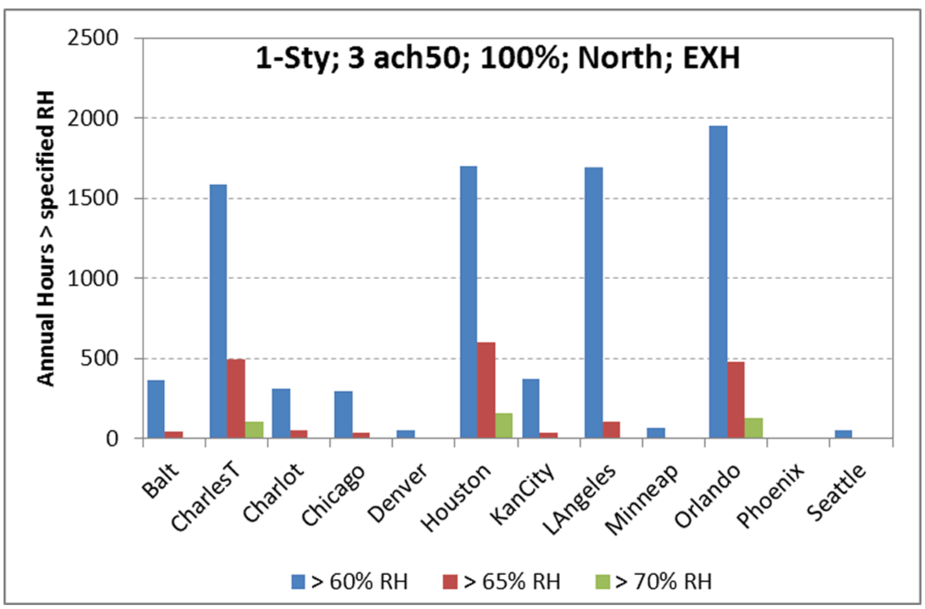


Figure 20. Hours of elevated indoor RH are a strong function of the selected RH limit and climate using EGUSA

4 Conclusions

4.1 Energy Consumption Related Conclusions

The difference in total space conditioning operating cost for a DOE Challenge Home controlled to < 60% RH with 75% of the ASHRAE 62.2-2013 requirement compared to 100% of that ventilation rate was about $45/yr averaged over all climates, enclosure tightness and ventilation systems simulated, or a savings of 10% of total space conditioning energy (Figure 21). (The 75% ventilation rate generally corresponds to the 62.2-2010 rate.) At 50% of the ASHRAE 62.2-2013 ventilation rate requirement, the average additional savings is about another $35/yr for an average total savings of 15% of space conditioning energy compared with 100% of that ventilation rate requirement.

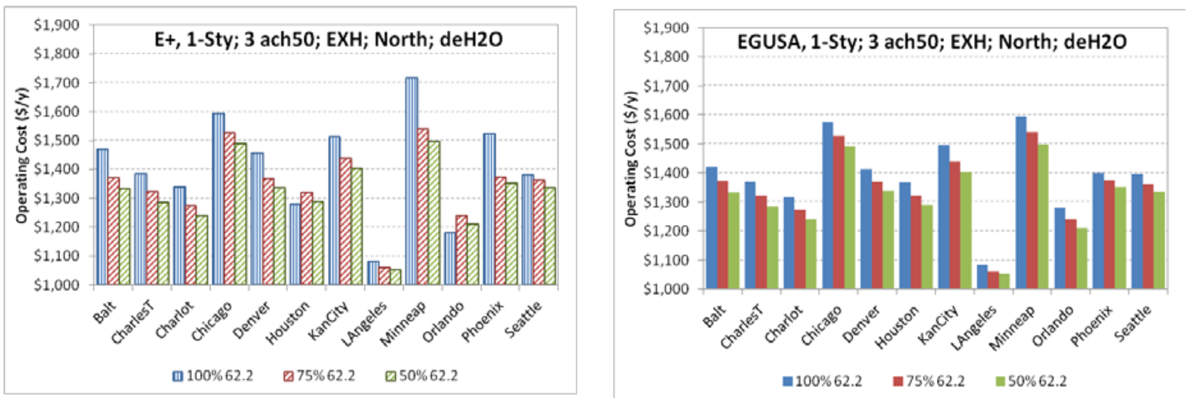


Figure 21. Total annual operating cost for a DOE Challenge Home controlled to < 60% RH over a range of ventilation rates

Total annual operating cost for a DOE Challenge Home varied less than $90/yr among all ventilation systems in every climate (Figure 22). Compared to exhaust ventilation as a reference, balanced ERV ventilation (installed such that it does not require coincident AHU fan operation to avoid short-circuiting of ventilation air) ranged from $0/yr to $40/yr more, with an average of about $20/yr more for 100% of the ASHRAE 62.2-2010 requirement at 3 ACH 50. However, at greater mechanical ventilation rates required in the 1.5 ACH50 house, the ERV *saved* an average of $20/yr over exhaust ventilation, in all but the dry climate of Phoenix and the mild climate of Los Angeles, where it continued to consume more energy than exhaust. CFI supply ventilation ranged from about $11/yr to $93/yr more than exhaust ventilation, with an average of about $40/yr.

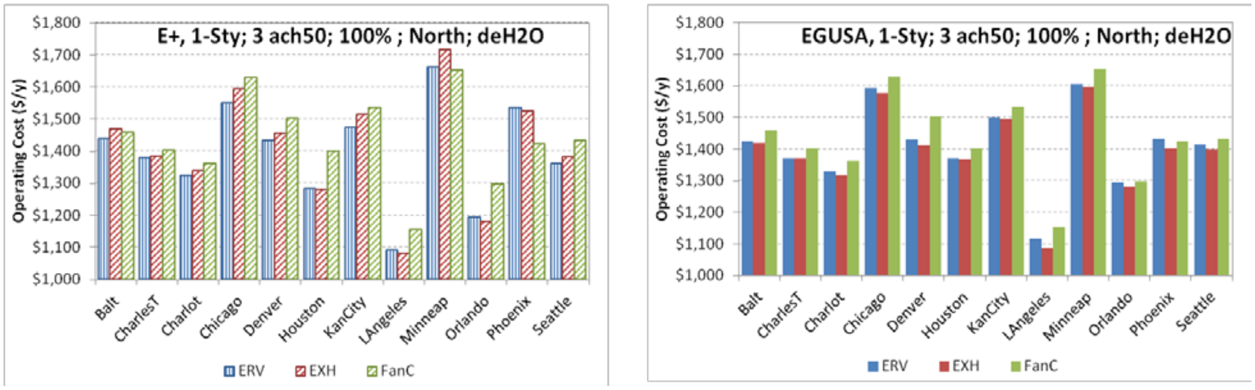


Figure 22. Total annual operating cost per year for a DOE Challenge Home over a range of ventilation system types

Supplemental dehumidification controlled to 60% RH was predicted to be $10–$58/yr for the warm-humid climates of Charleston, Houston, and Orlando, and the marine climate of Los Angeles. However, a caveat was provided that indicates that this value is predicated on an operating dehumidifier energy factor of 1.47 L/kWh, and recent field data indicate that conventional dehumidifiers operate closer to 0.8 L/kWh (Mattison and Korn 2012), which would tend to double this cost. Additionally, dehumidifiers tend to operate on a large humidity dead band, which means that maintaining humidity below 60% would likely require humidity set points near 55%, which could dramatically increase dehumidification costs. Also, with unbalanced mechanical ventilation, air change goes up as building leakage goes down from 3.0 ACH50 to 1.5 ACH50, resulting in more needed supplemental dehumidification. The increase in air exchange rate from unbalanced to balanced ventilation was about 24% for the 3.0 ACH50 houses and about 10% for the 1.5 ACH50 houses.

4.2 Humidity Control Related Conclusions

Hours of elevated indoor RH are a strong function of the selected RH limit and climate. For example, in Los Angeles, nearly all of the hours above 60% RH are also below 65% RH, but that is not true for Charleston, Houston, and Orlando. Hours above 60% RH during a particular space conditioning mode (heating, cooling, floating) was also a strong function of climate. Most of the hours significantly above 60% RH occur during floating hours, which occur mostly during fall, winter, and spring in Orlando, and sometimes during summer nights and summer extended rainy periods when sensible heat gain is lowered and internal moisture generation stays the same.

Mechanical ventilation, operated at the ASHRAE 62.2-2013 rate (about 40% more than ASHRAE 62.2-2010), in a 3 ACH50 house, raises the annual median indoor RH by almost 10% RH compared to a 7 ACH50 house with approximately the same annual average air exchange but without mechanical ventilation in Orlando. However, as seen in Figure 23, most of the RH increase due to ventilation is between 60% and 65% RH; the hours between 65% and 75% RH mostly remain the same. This indicates that the frequency of extreme RH events, which are of greater health/durability concern than median events, is not increased. This also indicates that some supplemental dehumidification would be needed in either case to maintain RH below 60%. In the 7 ACH50 house, the ventilation rate throughout the year is variable, and dependent on weather related driving forces. As seen in Figure 23, in Orlando weak summertime driving forces result in little need for supplemental dehumidification during the cooling season.

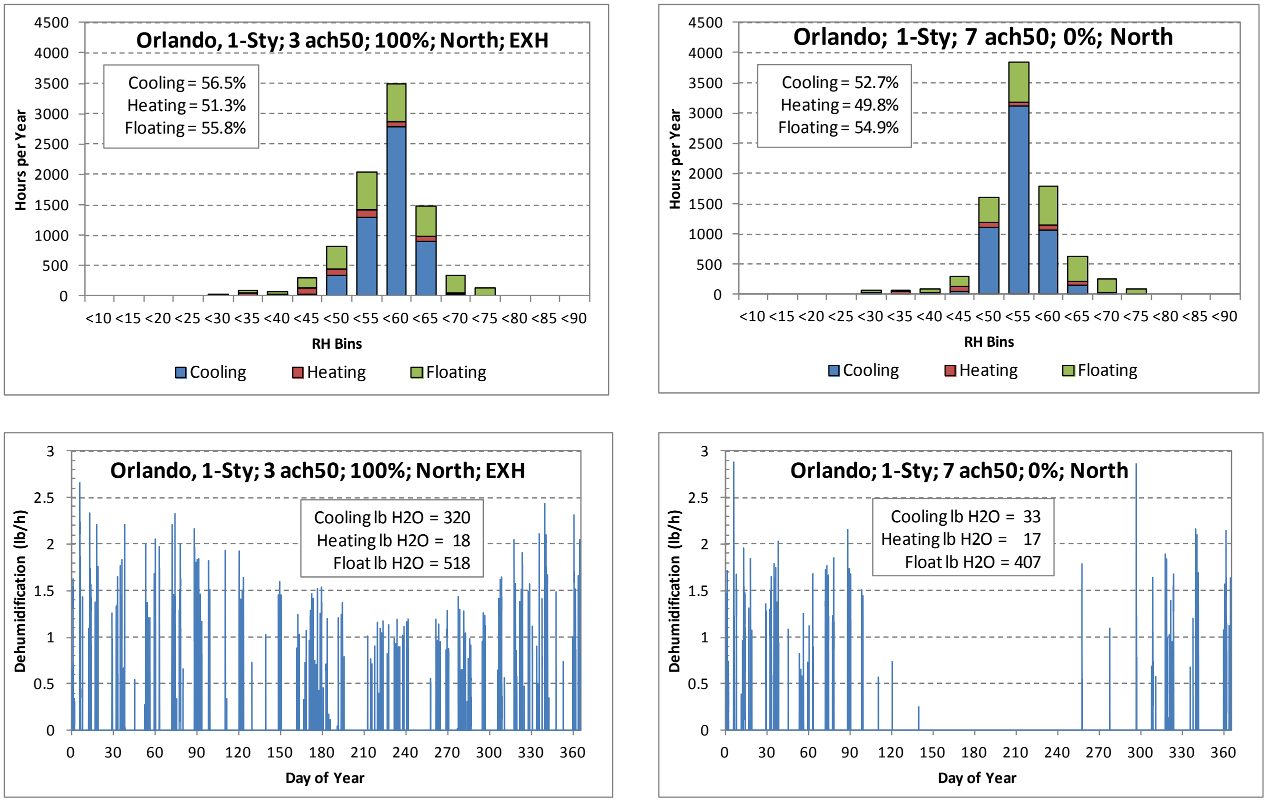


Figure 23. ASHRAE 62.2-2013 ventilation rate significantly increases number of hours between 60% and 65% RH compared to a conventional dwelling without mechanical ventilation using EGUSA

Durability analysis for Orlando, for a standard double-pane window with vinyl frame, showed that center-of-glass window condensation problems are highly unlikely to occur because the simulations show that it would require wintertime indoor RH of ≥ 75%, which is highly unlikely to occur in homes. With a less thermally efficient metal window frame, the condensation risk would be greater.

In Orlando, an ERV nearly eliminated the cooling hours above 60% RH, but according to EGUSA it had little effect on the floating hours above 60% RH. ERV modeling using E+ is suspect, as it shows too much reduction in elevated indoor humidity hours compared to the EGUSA models. The most important way in which the results differ between the simulation programs is in the frequency and magnitude of elevated indoor RH. For example, in Orlando, the EGUSA results show roughly 2000 hours above 60% RH, while the E+ results are closer to 3000 hours. In terms of maximum indoor RH, the EGUSA results show about 75% while the E+ results show about 85%. However, in all cases, supplemental dehumidification would be required to maintain RH below 60% in high performance, low-energy houses, and the difference in supplemental dehumidification energy between the simulation models would probably be less than 150 kWh/yr or less than $20/yr.

5 Key Gaps and Areas of Ongoing Research

A key gap and area of ongoing research is cost-effective application of ASHRAE Standard 62.2-2013 with systems and approaches that reduce energy consumption, improve humidity control performance, and improve indoor air quality. Identifying methods to achieve such objectives would benefit the EPA Indoor airPLUS program required by the DOE Challenge Home. Research should be conducted into methods that include considerations for:

- Managing indoor air pollutants in ways other than outdoor air exchange.
- Quantifying exposure to pollutants, and mitigation of that exposure, over a time scale shorter than annual average.
- Accounting for the quality of the source of outdoor air for outdoor environs and different ventilation systems types.
- Accounting for ventilation air distribution effectiveness.
- Application of demand-based ventilation systems.

Models that can accurately simulate the performance of advanced ventilation control systems in high performance buildings, along with associated humidity control systems in humid climates, are needed to understand a wide range of scenarios related to the economics and operational success of low-energy homes. Accurate simulation of ventilation and space conditioning control systems that operate on sub-hourly time scales coupled with the interacting heat and mass transfer effects is complex. It depends on many still somewhat unknown inputs, such as:

- Internal moisture generation rates
- Other moisture loads including effects of construction moisture and rain wetting under solar loading
- Building moisture capacitance and the impacts of building material moisture adsorption/desorption
- Detailed space conditioning equipment performance maps.

This capability is improving, but still has a long way to go to be adequately integrated into commonly used building design and performance rating software. Conducting research leading to quantification of unknown inputs is an important need.

Refinement to methods to simulate moisture capacitance is also needed. The simulations conducted for this study were conducted using the moisture capacitance (MC) modeling method where the indoor MC was set equal to 10 times the indoor air capacitance. A recent study by Woods et al. (2013) has examined the three principal methods of representing MC in building energy simulation models. They are generically referred to as the MC method, the effective moisture penetration depth (EMPD) method and the detailed, coupled heat and moisture transport (HAMT) method. Using HAMT as the basis of comparison, the other two methods were evaluated by Woods et al. In addition, the authors postulated a second EMPD2 model that uses two, instead of one, effective penetration depths. The EMPD2 method was found to have significant advantages over the EMPD1 model and the MC model. The second layer in the EMPD2 model provides for a longer-term moisture storage capacitance in addition to the short term capacitance of the EMPD1 model. This substantially improves estimation accuracy compared with both the MC model and the EMPD1 model. The EMPD2 model, like the MC model and the EMPD1 model, can be used with conduction transfer functions, rather than requiring finite element or finite difference HAMT numerical solutions. This significantly reduces simulation times compared to HAMT modeling.

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