



# Using EnergyPlus to Perform Dehumidification Analysis on Building America Homes

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# Using EnergyPlus to Perform Dehumidification Analysis on Building America Homes

## ABSTRACT

*A parametric study was conducted using EnergyPlus version 6.0 to investigate humidity issues on a typical mid-1990s reference home, a 2006 International Energy Conservation Code home, and a high-performance home in a hot-humid climate. The impacts of various dehumidification equipment and controls are analyzed on the high performance home. The study used the Walker and Wilson (1998) infiltration model to examine the combined effects of infiltration and mechanical ventilation with balanced and unbalanced mechanical ventilation systems. Indoor relative humidity excursions were examined; specifically, the number of excursions, average excursion length, and maximum excursion length. Space relative humidity, thermal comfort, and whole-house source energy consumption were analyzed for indoor relative humidity set points of 50%, 55%, and 60%.*

*The study showed and explained why similar trends of high humidity were observed in all three homes regardless of energy efficiency, and why humidity problems are not necessarily unique in high-performance homes. Thermal comfort analysis indicated that occupants are unlikely to notice indoor humidity problems. The study confirmed that supplemental dehumidification should be provided to maintain space relative humidity below 60% in a hot-humid climate. All modeled supplemental dehumidification options successfully controlled space relative humidity excursions, yet the increase in whole-house energy consumption was much more sensitive to the humidity set points than the chosen technology option.*

## INTRODUCTION

As the Building America (BA) program begins constructing homes that achieve 50% and greater source energy savings over typical mid-1990s construction, proper modeling of whole-house latent loads and operation of humidity control equipment has become a high priority. A 50% savings home has significantly reduced space-sensible cooling loads because of its high-performance envelope, ENERGY STAR® appliances, and energy-efficient lighting. An energy-efficient home must also manage a different latent load profile compared to conventional construction because of reduced natural infiltration, added ASHRAE Standard 62.2 code-compliant mechanical ventilation, and ENERGY STAR appliances.

Walker and Sherman (2007) showed that, for an International Energy Conservation Code (IECC) 2005 code-compliant home, adding ASHRAE 62.2-2004 (ASHRAE 2004a) code-compliant ventilation systems yielded insignificant changes to average indoor relative humidity (RH) *except* in a hot-humid climate. An average increase of 5%–10%

RH was observed for the hot-humid climate of Houston, yet the occurrence of hours above 70% RH was reduced.

Rudd and Henderson (2007) presented field data collected from 43 homes in various climate regions throughout the United States, including six hot-humid cities. They observed that periods of high space RH generally occurred when the air-conditioner (A/C) ran at part-load conditions. The study concluded that low sensible heat gain, coupled with continuous mechanical ventilation in high-performance homes, significantly increased the number of hours requiring supplemental dehumidification.

Henderson et al. (2008) conducted a TRNSYS study to analyze space A/C equipment with and without humidity control strategies in several hot-humid cities. The study used the TRNSYS 16 multi-zone type 56 building model to analyze either variable natural infiltration using the Sherman-Grimsrud model (ASHRAE 2009a) or constant infiltration and mechanical ventilation as prescribed by ASHRAE 62.2-2004, but did not examine the combined effects of natural infiltration and mechanical ventilation. Their study included a Home Energy Rating System (HERS) reference home (which was consistent with the 2004 IECC minimum efficiency standards) and a high-efficiency home, which had slightly lower performance than the high-performance home used in the current study. Their study also concluded that explicit dehumidification must be provided to maintain space RH below 60%.

We used new features in EnergyPlus version 6.0 (DOE 2010) to conduct a parametric study for analyzing the impacts of various dehumidification equipment and control strategies on a high-performance home in Houston at the 50% source energy savings level. A 50% savings home has significantly reduced whole-house energy consumption, so dehumidification equipment energy use becomes a larger portion of whole-house energy consumption. Improved simulation of A/C and dehumidification system interaction is expected to support the optimization of a home's features for durability, healthy indoor air quality, and minimization of energy use and system cost. A typical mid-1990s reference home and a 2006 IECC code-compliant home served as basis for comparison.

The paper presents analysis results, focusing on RH "excursions," which are defined as unique events of indoor RH above a threshold. For each mechanical system and home type in this study, the number of excursions, average excursion length, and maximum excursion length were recorded. Mold growth can begin during an excursion above 70% RH in 24 to 48 hours (EPA 2010, Occupational and Environmental Epidemiology 2005), given the appropriate temperature, mold spores, and food source. Although mold growth is directly related to the humidity and temperature of the surfaces rather than of the indoor air, the two are closely related. One 24-hour long RH excursion promotes much more mold growth than 24 intermittent 1-hour long RH excursions, and therefore, does much more harm to the house. Mold reduces the home's durability and degrades indoor air quality, so significant excursions must be controlled.

The impacts of different humidity set points (50%, 55%, and 60% RH) on RH excursions and energy consumption were studied for the 50% savings home parametric cases with explicit humidity control equipment. These set points are commonly used on dehumidification equipment for mold and dust mite control in houses (ASHRAE 2009a).

Obviously the dehumidification equipment will consume more energy to meet the lower RH set point, but the space humidity control may become more effective. Additionally, a lower RH set point may require larger dehumidification equipment.

Our study examined combined infiltration and mechanical ventilation effects with balanced and unbalanced mechanical ventilation systems in EnergyPlus. We used the infiltration model from Walker and Wilson (1998), which was developed specifically for residential buildings and includes more detailed coefficients than the Sherman-Grimsrud method. We examined living space RH, thermal comfort, and whole-house energy consumption.

## MODELING APPROACH AND ASSUMPTIONS

### Descriptions of Homes

Three homes are used in the analysis: a typical mid-1990s reference home, which is described as the BA Benchmark Home (Hendron and Engebrecht 2010), a 2006 IECC-compliant house (ICC 2006), and a high-performance home (50% source energy savings level whole-house technology package developed using a cost and performance analysis) (Anderson and Roberts 2008). The geometry from Anderson and Roberts was used for all three homes. The house used in the analysis was a two-story, three-bedroom, 2,500-ft<sup>2</sup> (232 m<sup>2</sup>), slab-on-grade structure. Shelter class 3 was assumed for modeling infiltration and it was assumed the house did not contain a flue. It was assumed to be east facing with an 18% window-to-floor area ratio. The homes' parameters with greatest impact on living space humidity levels are shown in Table 1.

<b>Table 1: Features of the Three Homes Used In the Analysis</b>			
	<b>Reference Home</b>	<b>IECC 2006 Home (Climate Region 2)</b>	<b>High-Performance Home</b>
Wall Assembly	2 × 4, 16" on center R-11 cavity	2 × 4, 16" on center R-13 cavity	2 × 6, 24" on center R-21 cavity
Ceiling Assembly	R-20 assembly	R-30 assembly	R-30 assembly
Window U-Value	1.0 Btu/hr-ft <sup>2</sup> -°F (5.7 W/m <sup>2</sup> -K)	0.75 Btu/hr-ft <sup>2</sup> -°F (4.3 W/m <sup>2</sup> -K)	0.35 Btu/hr-ft <sup>2</sup> -°F (2.0 W/m <sup>2</sup> -K)
Window Solar Heat Gain Coefficient	0.79	0.40	0.26
Specific Leakage Area	0.00057	0.00036	0.00015
Duct Location	Vented Attic	Vented Attic	Conditioned Space
Duct Insulation	R-5	R-8	N/A
Duct Leakage	10% Supply Leakage 5% Return Leakage	10% Supply Leakage 5% Return Leakage	N/A
Average Daily Internal Moisture Generation	15.0 lb/day (6.8 kg/day)	15.0 lb/day (6.8 kg/day)	14.8 lb/day (6.7 kg/day)
Heat Pump Rating	SEER 10, HSPF 7.2	SEER 13, HSPF 8.1	SEER 15, HSPF 8.8
Mechanical Ventilation	Spot Vents Only	100% ASHRAE 62.2	100% ASHRAE 62.2

The average daily internal moisture generation listed in Table 1 follow the BA Benchmark-prescribed load profiles for water use fixtures (shower, bath, and sinks) and appliances (clothes washer and dishwasher). The slight difference in the internal moisture generation among the three homes is due to ENERGY STAR appliances. Several studies in the literature assumed similar average daily internal moisture generation:

- Tenwolde and Walker (2001) found an average moisture generation of 15.9 lb/day (7.2 kg/day) for a house with one to two adults, with a standard deviation of 4.9 lb/day (2.2 kg/day) from multiple studies. The first child introduces an additional 8.8 lb/day (4 kg/day), the second child introduces 4.4 lb/day (2 kg/day), and each additional child introduces 2.2 lb/day (1 kg/day). The data set is adopted in ASHRAE Standard 160 “Design Criteria for Moisture Control in Buildings” to derive design conditions (ASHRAE 2009b).
- Walker and Sherman (2007) examined several sources of information and used 14.3 lb/day (6.5 kg/day) for their simulation. The number is based on the assumption that moisture generated by bathing, cooking, and dishwashing 8.8 lb/day (4 kg/day) for a family of four (Emmerich et al. 2005)) is directly exhausted by spot ventilation, and thus needs to be subtracted from a total daily 30.4 lb/day (13.8 kg/day) (ASHRAE 2009b). Additionally, their simulated house was occupied for only two-thirds of the day.

The heating and cooling set points are defined by BA Benchmark definition at 71°F (21.67°C) for heating and 76°F (24.4°C) for cooling. The effect of RH set point (when applicable) was examined and three RH set points were used: 50%, 55%, and 60%. Based on the BA Benchmark definition, the house had three occupants whose schedules resulted in fewer occupants during the daytime hours. Schedules and assumptions are consistent among the three homes and can be found in the BA Benchmark definition.

A moisture capacitance model is used in the analysis with a whole-house moisture capacitance multiplier of 15 (EPA 2001). This multiplier is also consistent with values used in previous studies for simulation of residential buildings (Henderson 2010). The EnergyPlus humidity capacitance multiplier was used to alter the effective moisture capacitance of the zone air volume and provided a mechanism to include the moisture buffering effect of building materials and furnishings.

### **High Performance Home Dehumidification Parametric Case Descriptions**

Parametric cases 0 through 9 are on the high-performance home in Houston. A properly sized 2-ton (7-kW) A/C unit with a sensible heat ratio of 0.8 was selected for all cases. Proper sizing of the A/C is paramount in real practice and modeling. EnergyPlus uses the latent heat degradation model (Henderson et al. 1992), so an oversized A/C system with shorter run time will remove less moisture.

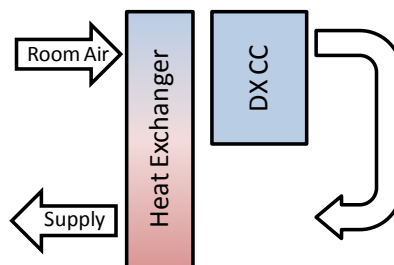
**Case 0 – A/C Only.** Case 0 represents the high-performance home equipped with a typical air conditioner for space conditioning (see Table 1). The cooling set point is constant at 76°F (24.4°C).

**Case 1 – Thermostat Reset.** Despite the A/C system maintaining a well-controlled cooling set point of 76°F (24.4°C), there are hours when space RH reaches high levels. An “enhanced” A/C control strategy commonly quoted is to overcool the space by 3°F (1.67°C) when the living space RH exceeds 60%. Overcooling the living space during high humidity periods increases A/C run time; thus, it removes more moisture and reduces the space humidity ratio. However, from a psychrometric standpoint, reduced space humidity ratio combined with reduced air temperature can result in continued high space RH, making overcooling periods excessively long. This interaction effect is

analyzed in the next section. In reality, overcooling the living space can exacerbate humidity problems because a larger outdoor-indoor temperature difference results in more moisture flow from the warm side to the cold side of the building envelope. With reduced indoor air temperature, interior surface temperatures also decrease and the surface RHs increase, approaching 70% (Lstiburek 1993). These issues, though currently not analyzed using EnergyPlus, are serious practical concerns that can make Case 1 an unrecommended strategy.

**Case 2 – A/C with Energy Recovery Ventilator (ERV).** In the summertime, an ERV cools and dries the hot and humid outside ventilation air by exchanging heat and moisture into the (conditioned) house exhaust air. In a hot-humid city such as Houston, where mechanical ventilation increases space RH on average by 5%–10% (Walker and Sherman 2007), using an ERV for mechanical ventilation should reduce the house humidity level. Rudd and Henderson (2007) revealed an ERV may have helped control the latent load from ventilation; however, the ERV was insufficient to control humidity alone or when the A/C was running at part-load conditions. The modeled ERV is a static enthalpy heat exchanger with an average winter and summer effectiveness of 75% and 60%, respectively. The supply and exhaust fans are assumed to have balanced flow rates equal to ASHRAE 62.2 requirements. Explicit humidity controls were not used to control the ERV.

**Case 3 – Heat Exchanger (HX)-Assisted A/C.** An HX-assisted cooling coil (CC) has an HX wrapped around the direct-expansion (DX) CC (see Figure 1). The A/C inlet air is first precooled by the passive HX. This process improves the latent removal performance of the DX CC by allowing it to further depress the supply air dewpoint temperature. The cold air leaving the coil is then rewarmed by the passive HX and exits the equipment. Similar to the Henderson et al. (2008) TRNSYS model, a heat pipe HX was modeled with an average sensible effectiveness of 32%.



*Figure 1: Heat Exchanger-Assisted Cooling Coil*

**Case 4 – A/C with Condenser Reheat.** A DX CC with condenser reheat can actively control space temperature and RH (see Figure 2). Under normal operation, the A/C operates to meet the space thermostat cooling set point. When the space RH exceeds the humidistat RH set point, the DX CC further reduces the leaving air temperature to meet the space RH set point. The overcooled and dehumidified leaving air is then reheated by the condenser coil downstream.

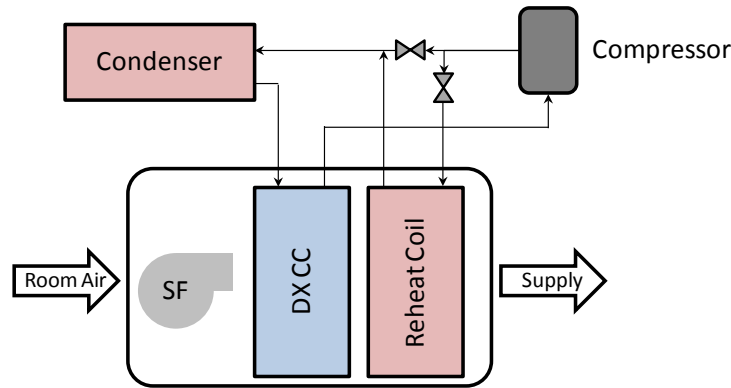


Figure 2: A/C with Condenser Reheat

The component model available in EnergyPlus version 6.0 to simulate this process is a desuperheater; however, the heat reclaim efficiency is limited to 30%. Thus, the system cannot simultaneously meet the cooling space temperature and humidity set points. To overcome this limitation, an unmetered gas heating coil was used to reheat the supply air (simulating condenser waste heat recovery) downstream of the DX CC. The condenser coil inlet air conditions were adjusted during dehumidification mode using the DX CC outlet and outdoor air temperatures.

During dehumidification mode, EnergyPlus reheats the CC outlet air to meet the space heating set point. In actual residential equipment, the reheat temperature during dehumidification mode is either the cooling set point or  $0.5^{\circ}\text{F}$  ( $0.28^{\circ}\text{C}$ ) below cooling set point (Henderson et al. 2008). Besides occupant discomfort, overcooling the living space during dehumidification mode results in the same problem described in Case 1. This problem worsens because the living space temperature is further reduced to the heating set point. To avoid this problem, an energy management system procedure was created in EnergyPlus to raise the space heating set point to the cooling set point from April to October. Because the modeled package needs to overcool and dehumidify the entire cooling supply air volume to meet the space humidity set point, an excessive amount of A/C energy may be required.

**Case 5 – A/C with Desiccant Wheel Dehumidifier.** A desiccant dehumidifier regenerates the desiccant wheel with heat rejected by the condenser (see Figure 3). By separating the supply airstream from the regenerative airstream, the unit supplies cool and dry air to the home and reduces A/C run time. A room air recirculating desiccant wheel dehumidifier was used in the model. Performance curves used for both the desiccant wheel and the DX coil were generated using the manufacturer’s publicly available sizing software.

At Air-Conditioning, Heating and Refrigeration Institute (AHRI) rated conditions, the system simulated has a moisture removal rate of 120 pints/day (56.8 L/day) and an energy factor of 7.0 pints/kWh (3.3 L/kWh) at an air flow rate of 200 cfm ( $0.09447\text{ m}^3/\text{s}$ ) for process and regeneration sides.



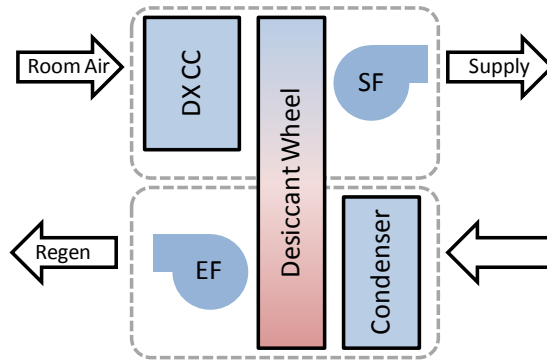


Figure 3: Desiccant Wheel Dehumidifier

**Case 6 – A/C with High-Efficiency DX Dehumidifier.** A common solution for controlling RH is to use a standalone vapor compression dehumidifier. We used three dehumidifiers to determine the RH control and energy consumption of different size units.

Table 2: High-Efficiency Dehumidifier Test Cases		
	Moisture Removal Rate, pints/day (L/day)	Energy Factor, pints/kWh (L/kWh)
High-Efficiency Dehumidifier #1 (Small)	65 (30.8)	4.2 (2.0)
High-Efficiency Dehumidifier #2 (Medium)	90 (42.6)	5.3 (2.5)
High-Efficiency Dehumidifier #3 (Large)	150 (71.0)	7.8 (3.7)

We used performance data from Christensen and Winkler (2009) and assumed that the normalized performance curves generated from test data could be applied to a smaller, similarly configured unit. The DX dehumidifier supplies warm, dry air to the living space because the latent heat removed from the airstream and the electric power consumed by the unit is rejected as heat to the supply airstream. As a result, operating a DX dehumidifier will increase A/C cooling run time and offset heat pump heating demand.

**Case 7 – A/C with Standard-Efficiency DX Dehumidifier.** A standard efficiency dehumidifier was included to determine the energy savings from using a high-efficiency unit. The standard efficiency unit was assumed to have the same capacity as the smallest high-efficiency unit (65 pints/day, 30.8 L/day) at reduced energy factor of 3.2 pints/kWh (1.5 L/kWh) (minimal ENERGY STAR cutoff point). Experimental test data generated by the National Renewable Energy Laboratory for a similar unit were used to create the necessary performance curves.

**Case 8 – A/C with ERV and High-Efficiency DX Dehumidifier.** This case combines technologies used in Case 2 and the smallest dehumidifier used in Case 6.

**Case 9 – A/C with ERV and Standard-Efficiency DX Dehumidifier.** This case combines technologies used in Cases 2 and 7.

## MODELING RESULTS AND ANALYSIS

### Predictions of Relative Humidity Excursions

Tables 3 and 4 show simulation results for RH excursions above 60% and 70% for the reference home, 2006 IECC home, and high-performance home without humidity control equipment. These homes undergo lengthy RH excursions above 60% and 70%, and thus are prone to mold growth. Reducing the thermostat when the space RH exceeds 60% (Case 1) does little to reduce the number of hours with high humidity levels. Incorporating an ERV (Case 2) into the house reduces the overall number of hours of high RH, but does little to reduce the maximum excursion lengths. The ERV's ability to reduce the overall number of high RH hours is a result of the ventilation latent load being a significant percentage of the overall latent load of the building. Of the three passive techniques (Cases 1–3), the HX-assisted CC (Case 3) best controls the space RH. Even though the number of hours of high RH is significantly reduced in Case 3, the maximum excursion length in Case 3 is 66 hours.

Case Description	Number of Hours		Number of Excursions		Average Excursion Length (h)		Maximum Excursion Length (h)	
	> 60%	> 70%	> 60%	> 70%	> 60%	> 70%	> 60%	> 70%
	Reference Home	1,729	96	139	17	12	6	86
2006 IECC Home	2,069	83	162	17	13	5	90	14

Case	Description	Number of Hours		Number of Excursions		Average Excursion Length (h)		Maximum Excursion Length (h)	
		> 60%	> 70%	> 60%	> 70%	> 60%	> 70%	> 60%	> 70%
		0	A/C Only	1,528	39	103	6	15	6
1	Thermostat Reset	1,666	13	93	1	18	13	99	13
2	A/C w/ERV	688	37	60	3	11	12	101	16
3	HX-Assisted A/C	353	16	30	2	12	8	66	15

Table 5 shows the RH excursions above 60% for the various types of humidity control equipment controlled to the three humidistat set points of 50%, 55%, and 60%. All types of active humidity control equipment control space RH levels very well. The RH did not exceed 70% for cases 4–9 with the listed humidistat RH set points. The number of hours when the living space RH exceeds 60% increases as the humidistat RH set point increases from 50% to 60% for all cases. This is because the equipment does not have the capacity to maintain the RH set point for all hours of the year. As the set point increases, the likelihood of the living space RH exceeding 60% also increases. The number of hours when the living space RH exceeds 60% is highest for the A/C system with condenser reheat (Case 4); however, the RH exceeds 60% for only short periods throughout the year. Using different humidistat set points (50%, 55%, or 60%) has minimal impact on space air RH excursions above 60% or 70%, assuming well-mixed room air conditions. A *small* dehumidifier controls the room RH as well as a large dehumidifier.

Case	Description	Number of Hours > 60% RH			Number of Excursions			Average Excursion Length (h)			Maximum Excursion Length (h)		
		50%	55%	60%	50%	55%	60%	50%	55%	60%	50%	55%	60%
4	A/C w/Condenser Reheat	0	36	88	0	17	63	0	2.1	1.4	0	4	6.7
5	A/C w/Desiccant Wheel Dehumidifier	0	0	40	0	0	48	0	0	0.8	0	0	5
6.1	A/C w/ <i>Small</i> High-Efficiency Dehumidifier	0	0	3	0	0	18	0	0	0.2	0	0	0.3
6.2	A/C w/ <i>Medium</i> High-Efficiency Dehumidifier	0	0	3	0	0	18	0	0	0.2	0	0	0.3
6.3	A/C w/ <i>Large</i> High-Efficiency Dehumidifier	0	0	1	0	0	7	0	0	0.2	0	0	0.2
7	A/C w/Standard-Efficiency Dehumidifier	0	0	5	0	0	22	0	0	0.2	0	0	0.5
8	A/C w/ERV and Small High-Efficiency Dehumidifier	0	0	1	0	0	7	0	0	0.2	0	0	0.2
9	A/C w/ERV and Standard Efficiency Dehumidifier	0	0	2	0	0	13	0	0	0.2	0	0	0.3

\* A humidistat deadband of  $\pm 2\%$  RH was assumed in determining excursion statistics for the 60% RH set point.

Henderson et al. (2008) examined humidity in a HERS reference home. That simulation had an internal moisture generation rate of 10.4 lb/day (4.7 kg/day) and assumed a year-round constant ventilation rate of 97.5 cfm (0.046 m<sup>3</sup>/s). The simulation results predicted 1,557 hours when the space RH exceeded 60%. The 2006 IECC home in the current study is similar to this house. The differences include additional internal moisture generation, a specific leakage area of 0.00036, modeling of the interaction effects between natural infiltration (using the Walker and Wilson infiltration model), and mechanical ventilation. Based on these differences, the increase in the number of hours when the space RH exceeded 60% from 1,557 to 2,069 is expected.

The results presented in Tables 3 and 4 demonstrate that indoor humidity problems are not unique to high-performance homes and likely appear in conventional homes as well. Previous studies suggested humidity problems might be unique to high-performance homes because the sensible load is drastically reduced, but the latent load is changed little (Rudd and Henderson 2007). Therefore, A/C can no longer be expected to remove the necessary amount of moisture from the living space in a high-performance home. The A/C run times, coupled with the infiltration and ventilation latent loads from the current study, can explain why the humidity problems are not unique to the high-performance home.

The living space RH exceeded 60% for the most hours during April, when the A/C run times for the reference home, IECC 2006 home, and high-performance home were 130, 109, and 83 hours, respectively. Thus, for all three homes the A/C did not run for a

significant amount of time (compared to July when the A/C ran two to three times longer). Figure 4 shows the number of hours when the living space RH exceeds 60% as a function of A/C run time fraction (DOE 2010). Nearly 50% of the hours when RH exceeds 60% occur when the A/C is barely running (0–0.1 run time fraction).

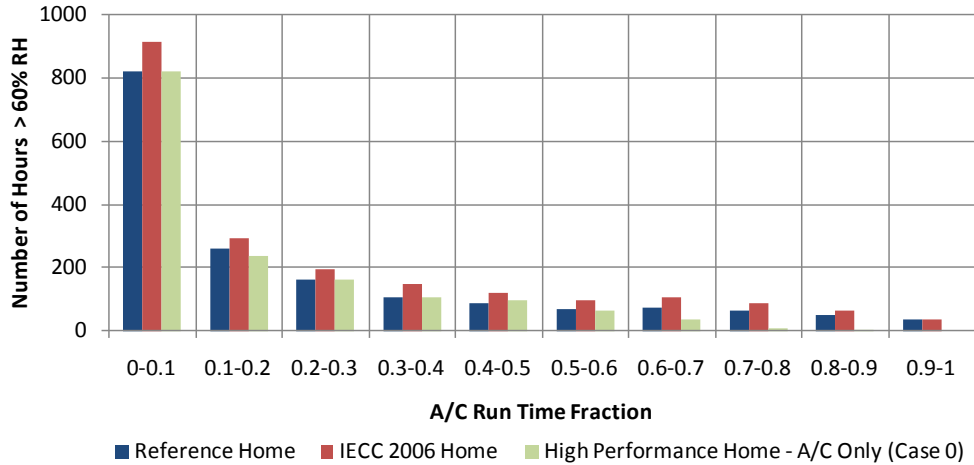


Figure 4: Number of Hours > 60% RH Based on A/C Run Time Fraction

Figures 5 and 6 show the average daily profiles for the space sensible cooling load and latent load for April. The space sensible load follows the expected trend with a peak in mid- to late afternoon. The sharp peak at 6 A.M. in the space latent load is due to the bathroom exhaust ventilation. For April the average latent load for the high-performance home is approximately 42% lower than for the reference home compared to an average reduction of 35% in the sensible cooling load. Because of a much tighter building envelope and despite whole-house mechanical ventilation, the latent load for the high-performance home was reduced by a greater percentage than the sensible cooling load compared to the reference home. Thus, the current analysis shows humidity problems are not necessarily unique to high-performance homes.

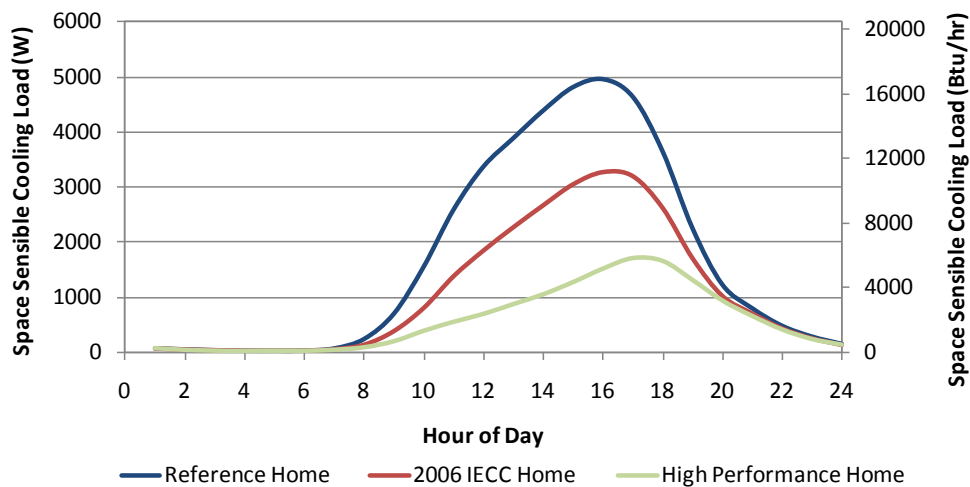


Figure 5: Space Sensible Cooling Load Average Daily Profile for April

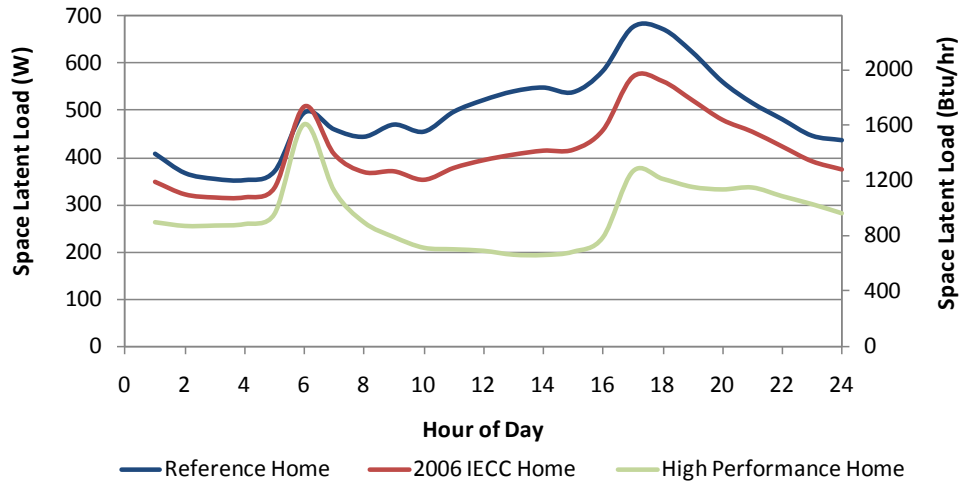


Figure 6: Space Latent Load Average Daily Profile for April

### Predictions of Thermal Comfort

EnergyPlus offers three thermal comfort models: Fanger (Fanger 1970), Pierce 2-node (Gagge et al. 1986) and KSU 2-node (Azer and Hsu 1977). Similar thermal comfort trends with respect to space temperature and RH were observed in all three models. Results displayed in Table 6 are Fanger Predicted Mean Vote (PMV) results only. A clothing (clo) schedule was created to take into account seasonal variation: December through February, clo = 1.0; March through April and October through November, clo = 0.75; May through September, clo = 0.5.

Table 6: Thermal Comfort Analysis (based on Fanger PMV)			
Case	Description	Maximum PMV Value	Hours Exceeding PMV = +0.5
-	Reference Home	0.95	718
-	2006 IECC Home	0.79	306
0	A/C Only	0.66	121
1	Thermostat Reset	0.57	23
2	A/C w/ ERV	0.65	134
3	HX Assisted A/C	0.64	109
4	A/C w/Condenser Reheat	0.56	28
5	A/C w/Desiccant Wheel Dehumidifier	0.58	71
6.1	A/C w/Small High-Efficiency Dehumidifier	0.61	112
6.2	A/C w/Medium High-Efficiency Dehumidifier	0.61	110
6.3	A/C w/Large High-Efficiency Dehumidifier	0.61	111
7	A/C w/Standard-Efficiency Dehumidifier	0.61	114
8	A/C w/ERV and Small High-Efficiency Dehumidifier	0.61	119
9	A/C w/ERV and Standard-Efficiency Dehumidifier	0.61	120

Note: Cases 4 –9 PMV are examined for RH set point of 55% only.

Table 6 displays the maximum PMV value and total number of hours exceeding a PMV of +0.5. Trends in maximum PMV values and total hours exceeding a +0.5 PMV do not correlate to the trends in RH presented in Tables 3 and 4.

Figure 7 examines the Fanger PMV trend for a space with uniform temperature (mean radiant temperature = air temperature), human metabolic rate of 1.0, clo 1.0, and air velocity 19.7 ft/min (0.1 m/s). The relatively low slopes of the PMV bands with respect

to RH show the insensitivity of Fanger PMV to space RH change. It is also calculated that a 10% increase in RH results in the same amount of increase in Fanger PMV as a 0.5°F (0.26°C) increase of air temperature, given the above set conditions. Alternately speaking, a 10% increase in RH is predicted to be offset by a drop in air temperature of 0.5°F (0.26°C) (Fountain and Huizenga 1997).

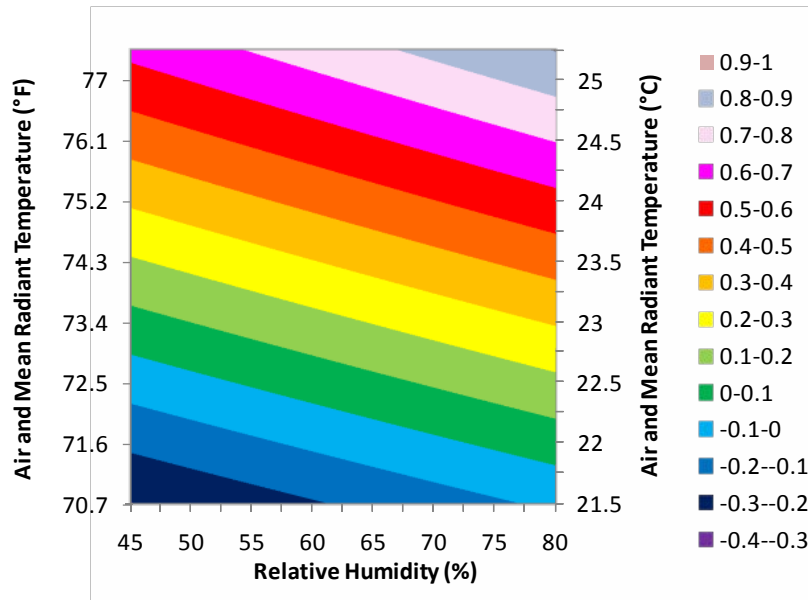


Figure 7: Fanger PMV Trend With Respect to Relative Humidity and Temperature

Figure 8 displays the ASHRAE Standard 55 (ASHRAE 2004b) thermal comfort regions for 80% of the occupants to be satisfied. The thermal comfort regions extend over a fairly large range in RH compared to temperature, meaning humans are less sensitive to humidity than to temperature. The comfort regions also set an upper limit of humidity ratio at 0.012 kg/kg dry air. With the modeled thermostat set points, the hours when PMV exceeds +0.5 are essentially equivalent to space humidity ratio exceeding 0.012 kg/kg dry air, unless clo or metabolic rate exceeds the range in Standard 55, where skin sweating regulation fails (ASHRAE 2009a). This indicates that an occupant is unlikely to notice the humidity problem in a home from a thermal comfort standpoint. Despite the variation of hours exceeding a PMV of 0.5 in the high-performance home, the PMV exceeds a value of +0.5 for less than 1.4% of the year for all the cases.

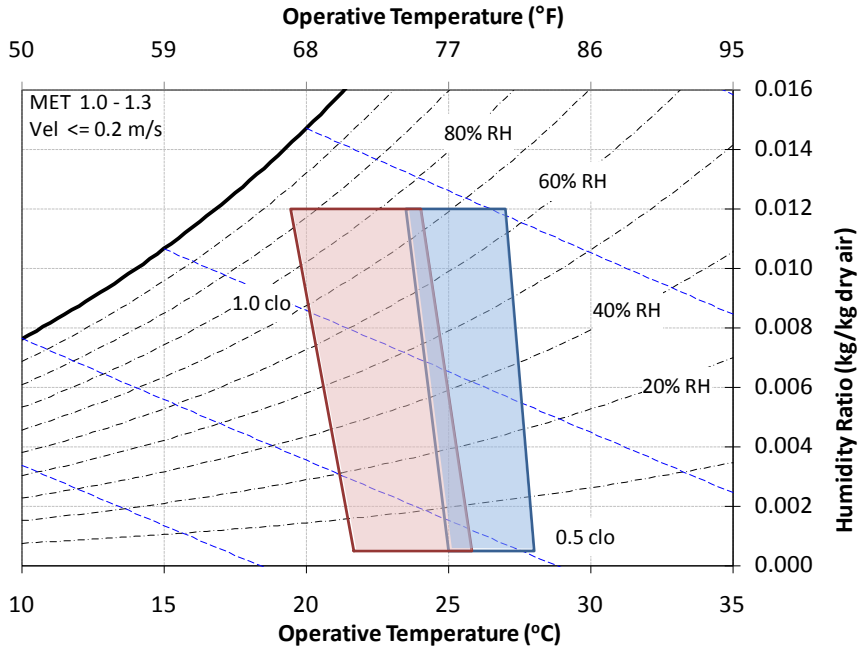


Figure 8: ASHRAE Standard 55 Range of Operative Temperature and Humidity

## Energy Consumption Summary

Tables 7 and 8 show the annual source energy consumption for the reference home, the IECC 2006 home, and the high-performance home for all the parametric cases. Source-to-site conversion ratios are 3.365 for electricity and 1.092 for natural gas (Deru and Torcellini 2007).

Case	Description	Misc. + Large Appl.	Lights	Hot Water	Vent Fans	Cooling	Heating	H/C Fan	Total
-	Reference Home	68.0 (71.8)	30.9 (32.6)	18.3 (19.3)	0.3 (0.3)	93.1 (98.2)	29.9 (31.5)	29.6 (31.6)	270.1 (285.0)
-	2006 IECC Home	68.1 (71.8)	30.9 (32.6)	15.9 (16.8)	2.8 (2.9)	59.9 (63.2)	28.1 (26.6)	12.1 (12.8)	216.4 (228.3)
0	A/C Only	62.9 (66.4)	8.6 (9.1)	9.3 (9.9)	1.8 (2.0)	29.4 (31.0)	8.9 (8.4)	3.7 (3.9)	124.2 (131.1)
1	Thermostat Reset	62.9 (66.4)	8.6 (9.1)	9.3 (9.9)	1.8 (2.0)	33.3 (35.1)	8.6 (9.1)	4.1 (4.3)	128.7 (135.8)
2	A/C w/ERV	62.9 (66.4)	8.6 (9.1)	9.3 (9.9)	4.3 (4.5)	28.8 (30.4)	7.3 (7.7)	3.5 (3.7)	124.7 (131.6)
3	HX Assisted A/C	62.9 (66.4)	8.6 (9.1)	9.3 (9.9)	1.8 (2.0)	32.9 (34.7)	8.5 (8.9)	4.1 (4.3)	128.2 (135.3)

Adding explicit humidity control equipment (Cases 4–9) increases whole-house source energy consumption (Table 8). Table 8 and Figure 9 indicate the same trend: an RH set point of 50% results in much more energy consumption than an RH set point of 60%. Table 10 indicates that an RH set point of 50% results in increased run time for both the dehumidifier and the A/C compared to an RH set point of 60%, except for A/C with desiccant wheel dehumidifier (Case 5). In A/C w/ condenser reheat (Case 4), the A/C run time includes the dehumidification run time because the A/C operates during both the cooling and the dehumidification modes.

Case	Description	Cooling and Dehumidification			Total*			Total Percent Increase vs. Case 0		
		50%	55%	60%	50%	55%	60%	50%	55%	60%
4	A/C w/Condenser Reheat	54.0 (57.0)	39.0 (41.1)	32.3 (34.1)	152.0 (160.4)	135.0 (142.4)	127.4 (134.4)	22.4%	8.7%	2.6%
5	A/C w/Desiccant Wheel Dehumidifier	40.7 (42.9)	34.3 (36.2)	30.6 (32.3)	141.2 (149.0)	131.6 (138.8)	126.0 (133.0)	13.7%	5.9%	1.5%
6.1	A/C w/Small High-Efficiency Dehumidifier	53.5 (56.4)	39.3 (41.5)	32.5 (34.3)	148.2 (156.4)	133.9 (141.3)	127.1 (134.1)	19.3%	7.8%	2.3%
6.2	A/C w/Medium High-Efficiency Dehumidifier	50.3 (53.1)	38.1 (40.2)	32.2 (34.0)	145.0 (153.0)	132.8 (140.1)	126.9 (133.9)	16.7%	6.9%	2.2%
6.3	A/C w/Large High-Efficiency Dehumidifier	45.3 (47.7)	36.4 (38.4)	32.0 (33.7)	139.9 (147.6)	131.0 (138.2)	126.6 (133.6)	12.7%	5.5%	1.9%
7	A/C w/Standard-Efficiency Dehumidifier	60.8 (64.1)	41.7 (44.0)	32.9 (34.7)	155.5 (164.1)	136.4 (143.9)	127.5 (134.5)	25.2%	9.8%	2.7%
8	A/C w/ERV and Small High-Efficiency Dehumidifier	44.3 (46.7)	33.7 (35.5)	30.7 (32.4)	147.5 (139.8)	129.3 (136.4)	126.4 (133.4)	12.6%	4.1%	1.8%
9	A/C w/ERV and Standard-Efficiency Dehumidifier	48.9 (51.6)	34.7 (36.6)	32.7 (31.0)	144.5 (152.4)	130.3 (137.5)	126.7 (133.7)	16.3%	4.9%	2.0%

\* Total column is the summation of all end uses: miscellaneous and large appliances, lights, hot water, ventilation fans, heating, cooling and dehumidification, heating and cooling fans.



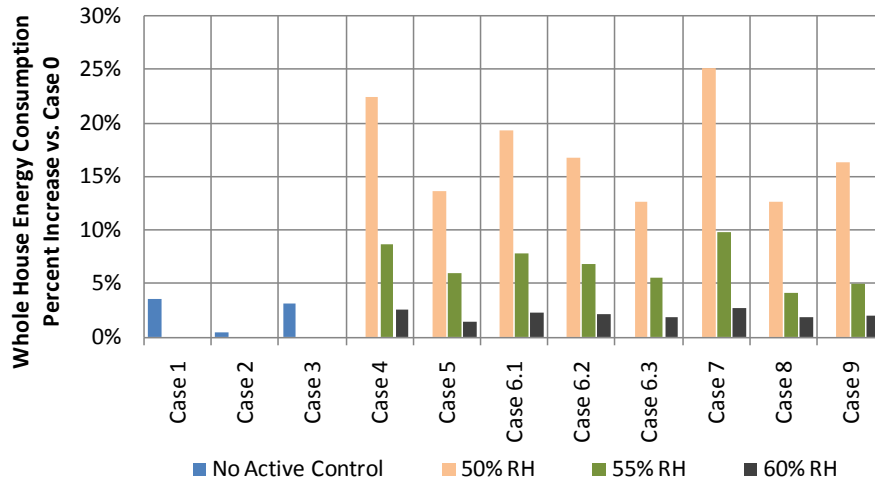


Figure 9: High-Performance Home Percent Increase of Whole-House Energy Consumption vs. Case 0

Case	Description	A/C Run Time (h)	Dehumidifier Run Time (h)
–	BA Benchmark Home	2,204	0
–	2006 IECC Home	2,068	0
0	A/C Only	1,674	0
1	Thermostat Reset	1,912	0
2	A/C w/ERV	1,647	0
3	HX-Assisted A/C	1,910	0

Case	Description	A/C Run Time (h)			Dehumidifier Run Time (h)		
		50%	55%	60%	50%	55%	60%
4	A/C w/Condenser Reheat	3,356	2,307	1,844	1,785	628	139
5	A/C w/Desiccant Wheel Dehumidifier	1,467	1,591	1,659	2,344	1,018	241
6.1	A/C w/Small High-Efficiency Dehumidifier	2,006	1,803	1,701	2,447	948	225
6.2	A/C w/Medium High-Efficiency Dehumidifier	1,979	1,794	1,700	1,854	720	174
6.3	A/C w/Large High-Efficiency Dehumidifier	1,931	1,782	1,701	1,180	471	123
7	A/C w/Standard-Efficiency Dehumidifier	2,075	1,825	1,704	2,745	1,014	231
8	A/C w/ERV and Small High-Efficiency Dehumidifier	1,846	1,695	1,654	1,563	431	112
9	A/C w/ERV and Standard-Efficiency Dehumidifier	1,889	1,703	1,655	1,757	462	117

The percent increase of source energy consumption (see Figure 9) is the required energy penalty of achieving home durability and maintaining healthy indoor air quality. A few observations were made:

- In the high-performance home with explicit dehumidification control (Cases 4–9), using a 50% RH set point increased the whole-house energy consumption by 12.6%–

22.4%; a 60% RH set point increased the whole-house energy consumption by only 1.5%–2.7%.

- Using an RH set point of 60% significantly reduced the dehumidifier or A/C dehumidification mode run time (Cases 4–9) as opposed to using RH set points of 50% and 55%. This in turn reduces the A/C run time, except for Case 5.
- Among all the active dehumidification technologies, A/C with desiccant wheel dehumidifier (Case 5) is the only option that reduced A/C run time. The desiccant wheel dehumidifier supplies cool air, so the A/C load is partially offset during dehumidifier operation. A/C with standard-efficiency DX dehumidifier (Case 7) showed the highest percent increase in the whole-house source energy consumption.
- Among the three dehumidifiers (Cases 6.1, 6.2, and 6.3), the most efficient and largest capacity (Case 6.3) resulted in the lowest energy consumption compared to the smaller dehumidifiers, despite more part-load losses.

## CONCLUSION

EnergyPlus modeling results with RH excursion analysis showed that all three homes (mid-1990s reference home, IECC 2006 home, and high-performance home) are prone to mold growth from long excursions of uncontrolled high humidity. Humidity problems are not unique to the high-performance home in the hot-humid climate compared to the mid-1990s reference home or the IECC 2006 home. Thermal comfort analysis indicates that occupants are unlikely to feel the humidity problems in a home; thus, humidity can pose a risk to home durability and health.

Adding active humidity control equipment (Cases 4–9) effectively controls RH to a safe level in the high-performance home. The 50% RH set point results in a significant source energy consumption penalty (12.6%–22.4% increase vs. Case 0) compared to the 60% set point (1.5%–2.7% increase vs. Case 0). At RH set points of 50% and 55%, A/C with desiccant wheel dehumidifier (Case 5) and A/C with ERV and high-efficiency DX dehumidifier (Case 8) stand out as the two options resulting in the smallest increase of source energy consumption. At an RH set point of 60%, all explicit dehumidification technologies (Cases 4–9) result in similar insignificant increases in source energy consumption and thus are equally competitive. Other factors such as cost, system configuration and complexity, and maintenance would therefore be dominant in determining which strategy to employ.

The study has several limitations:

- It is limited to Houston. Other cities in the hot-humid climate and cities in the mixed-humid climate can result in different indoor humidity levels.
- House internal operating conditions such as moisture generation rates and space temperature set points can vary significantly from house to house, and thus can lead to different levels of humidity in individual houses.
- Building envelope interactions with indoor and outdoor air humidity conditions are not analyzed. The moisture buffering behavior of building materials and furnishings was assumed to be constant and was modeled as a lumped capacitance.

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# REPORT DOCUMENTATION PAGE

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<b>14. ABSTRACT (Maximum 200 Words)</b> A parametric study was conducted using EnergyPlus version 6.0 to investigate humidity issues on a typical mid-1990s reference home, a 2006 International Energy Conservation Code home, and a high-performance home in a hot-humid climate. The impacts of various dehumidification equipment and controls are analyzed on the high performance home. The study examined the combined effects of infiltration and mechanical ventilation with balanced and unbalanced mechanical ventilation systems. Indoor relative humidity excursions were examined; specifically, the number of excursions, average excursion length, and maximum excursion length. Space relative humidity, thermal comfort, and whole-house source energy consumption were analyzed for indoor relative humidity set points of 50%, 55%, and 60%. The study showed and explained why similar trends of high humidity were observed in all three homes regardless of energy efficiency, and why humidity problems are not necessarily unique in high-performance homes. Thermal comfort analysis indicated that occupants are unlikely to notice indoor humidity problems. The study confirmed that supplemental dehumidification should be provided to maintain space relative humidity below 60% in a hot-humid climate.					
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