

Expert Meeting: Recommended Approaches to Humidity Control in High Performance Homes

Armin Rudd

Building Science Corporation (BSC)

July 2013

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Office of Energy Efficiency and Renewable Energy

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Acknowledgements

The author appreciates the presentations made by the invited speakers including: Hugh Henderson of CDH Energy Corp.; Philip Fairey of Florida Solar Energy Center; and Michael Sypolt of IBACOS.

Definitions

BSC	Building Science Corporation
AHRI	Air Conditioning Heating and Refrigeration Institute
AHAM	Association of Home Appliance Manufacturers
RH	Relative humidity
NRC	Natural Resources Canada
USDOE	United States Department of Energy
NREL	National Renewable Energy Laboratory
CFI	Central-fan-integrated [supply ventilation]
ERV	Energy recovery ventilator
HRV	Heat recovery ventilator

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Unless otherwise noted, all tables were created by BSC.

1 Introduction

As homes get more energy efficient, cooling systems have longer off-times due to lowered sensible heat gain. Overall, this is good, and produces a 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 will be required to maintain the relative humidity (RH) below acceptable levels.

Extensive field testing was done with builder partners in Texas and Florida in 2001 to 2007 (Rudd et al. 2003, BSC 2004, Rudd et al. 2005). That testing revealed that supplemental dehumidification was required in high performance Building America-level homes in order to maintain indoor RH below 60% year-round.

Off-the-shelf, stand-alone supplemental dehumidification systems were employed to address this problem while working with manufacturing partners on supplemental dehumidification integrated with the central space conditioning system. These companies began to offer integrated supplemental dehumidification solutions that allow year-round indoor RH control between 50% and 60%. This was intended to enable further reduction in sensible cooling load through further efficiency improvements, without the risk of elevated indoor humidity.

While these advancements have been important and needed in the residential space conditioning industry, supplemental dehumidification technology continues to improve and evolve, and the market for these products is still in its infancy. Design capacity prediction is subject to many unknowns and requires continued research to fully quantify.

Models that can accurately simulate the performance of humidity control systems in high performance buildings are needed to understand a wide range of scenarios related to the economics and operational success of low-energy homes with supplemental dehumidification. Simulation of supplemental dehumidification needs and performance in buildings is complex. It requires a model that controls both indoor temperature and RH, and depends on many still somewhat sketchy inputs, such as: internal moisture generation and other moisture loads including effects of construction moisture drying and rain wetting under solar loading; building moisture capacitance and the effects of building material moisture adsorption/desorption under solar loading; and detailed dehumidification 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 programs.

Building Science Corporation (BSC) hosted an expert meeting, “Recommended Approaches to Humidity Control in High Performance Homes,” on October 16, 2012, in Westford, Massachusetts, to bring together experts in the field of residential humidity control to address modeling issues for dehumidification.

2 Logistics

The expert meeting was held at the Westford Regency Inn and Conference Center, in Westford, Massachusetts, on October 16, 2012. The meeting was also made available to attendees by webinar, and was recorded. See Table 1 for a list of attendees.

Table 1. List of Meeting Participants.

Last name	First name	Company	Email	Attended In Person (P) or by Webinar (W)
Bergey	Daniel	BSC	daniel@buildingscience.com	P
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3 Research Questions

A list of research questions that were compiled and distributed with the meeting invitation is as follows:

- What are the important humidity control conclusions from recent residential dehumidification systems modeling, with a wide range of building and equipment sensitivity, using a customized <2 minute time-step TRNSYS model with temperature and humidity control?
- What are the important humidity control conclusions from recent residential ventilation modeling efforts using the temperature and humidity control logic of the Energy Plus version of the NREL BEopt program?
- What are the important indoor humidity conclusions that can be drawn from recent residential ventilation modeling efforts using the temperature-only control logic of the FSEC EnergyGauge USA program?

- How do the results from these different programs compare?
- Is it generally agreed that controlling to less than 60% RH is the appropriate humidity control point for high performance homes, and why?
- Is it generally agreed that annual hours above 60% RH is the appropriate humidity control performance metric to use to compare system performance and to compare required supplemental dehumidification energy? Does that metric give generally the same result as looking at 4-hour and 8-hour events above 60% RH?
- For a range of climates, ventilation systems, and space conditioning equipment in high performance homes, what is the magnitude of hours above 60% RH, the magnitude of supplemental dehumidification energy required to control to less than 60% RH, the time of year occurrence of elevated indoor humidity and supplemental dehumidification, and the space conditioning mode (heating, cooling, floating) during which most periods of elevated indoor humidity and supplemental dehumidification occur?

4 Objectives

The topic of this meeting was “Recommended Approaches to Humidity Control in High Performance Homes,” and focused on dehumidification requirements. Presentations and discussions centered on computer simulation and field experience with these systems, with the goal of developing foundational information to support the development of a Building America Measure Guideline on this topic.

The expert meeting was designed to bring together experts in the field of residential indoor humidity control by dehumidification to discuss the following objectives:

- Compare and contrast the state-of-the art in modeling residential supplemental dehumidification requirements in high performance homes.
- Come to agreement on an acceptable RH control system within a model and the primary metric to be used for evaluating the success of a given humidity control system.
- Quantitatively identify general trends as to the performance of, and cost to operate, supplemental dehumidification systems in high performance homes in humid climates.

5 Agenda

Table 2 shows the expert meeting agenda.

Table 2. Expert Meeting Agenda.

Time	Speaker	Topic
8:00 to 8:10	Armin Rudd Building Science Corp	Welcome and Introduction
8:10 to 9:00	Hugh Henderson CDH Energy Corp	ASHRAE RP 1449 Energy Efficiency and Cost Assessment of Humidity Control Options for Residential Buildings
9:00 to 9:50	Philip Fairey Florida Solar Energy Center	Humidity Control and Ventilation
9:50 to 10:00		Break
10:00 to 10:50	Michael Sybolt and Duncan Prah IBACOS	EnergyPlus Humidity Control and Ventilation Modeling Analysis
10:50 to 12:00	Armin Rudd Building Science Corp	Facilitated Open Discussion, Action Items, and Final Remarks

6 Agenda Summaries

Discussions occurred during and after each of the three invited presentations and in the facilitated discussion period that followed.

ASHRAE RP 1449: Energy Efficiency and Cost Assessment of Humidity Control Options for Residential Buildings

(Hugh Henderson, CDH Energy Corp.)

A number of fundamental computer modeling improvements were made to the TRNSYS/TRNSED model that CDH had previously developed for desiccant and dehumidifier systems. The model improvements, listed in Figure 1, allowed for more accurate, real-world simulation results. The short time-step model allowed precise determination of when mechanical equipment would be on or off according to actual control strategies. Referring to Figure 2, space conditioning equipment, ventilation equipment, and dehumidification equipment could all be operating at the same time, with all the heat and moisture consequences accounted for in the same simulation time interval.

Modeling Approach

- Using TRNSYS model first developed for Desiccant and DH systems
- Uses small time step (72 sec)
 - No more part load curves (either ON or OFF)
 - Thermostat dynamics (for cooling)
 - Detailed control of all components (e.g., CFIS)
- Type 56 for multi-zone building
 - Combined duct leakage & infiltration calcs
- Robust equipment models (performance maps)

Figure 1. Computer modeling improvements allowing more realistic simulation results.

Updated Simulation Framework

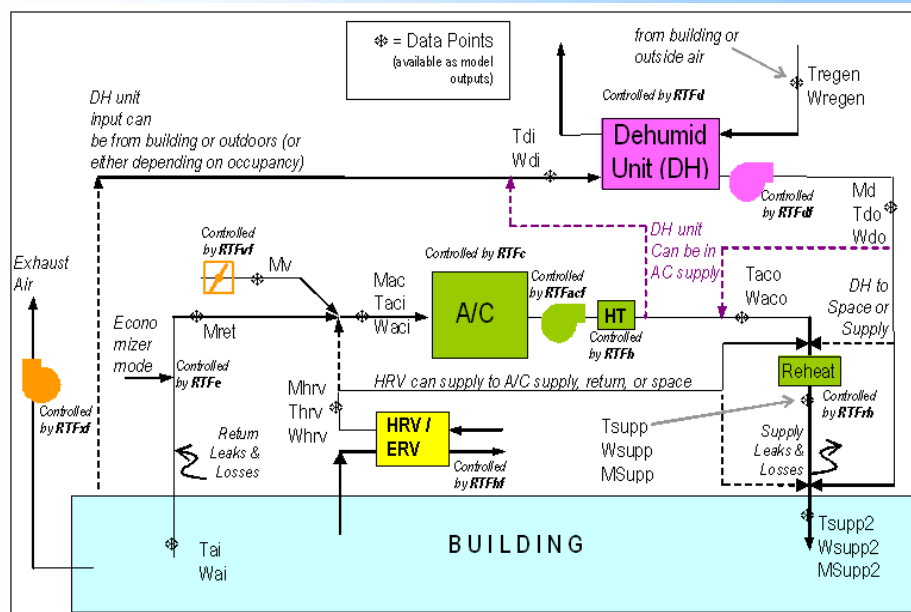
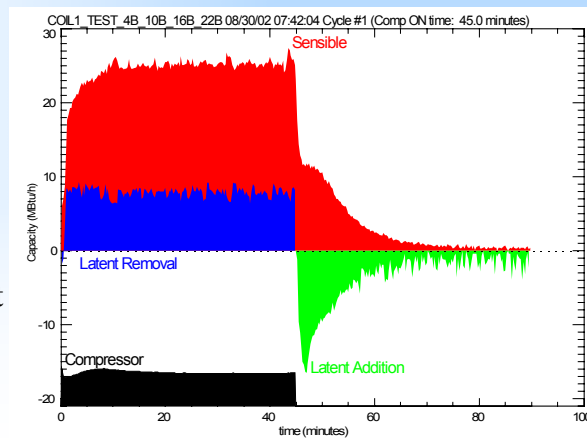


Figure 2. Simulation framework that allows for multiple temperature control, humidity control, ventilation, and fan recirculation systems to operate at the same time.

Moisture evaporation from wet cooling coils is modeled to realistically account for cooling and dehumidification system latent degradation (moisture added back to the ducts and conditioned space) when the cooling compressor is off.

Moisture Evaporation Models

- Some degradation in AUTO fan mode
 - Off-cycle airflow is assumed to be ~3 cfm
- Significant degradation when fan is ON



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Figure 3. Simulation framework that allows for multiple temperature control, humidity control, ventilation, and fan recirculation systems to operate at the same time.

The combined airflow of infiltration, duct leakage, and balanced and unbalanced ventilation is accounted for, as shown in Figure 4.

Figure 5 illustrates the conventional cooling system and indoor temperature and RH response with heating, cooling, and floating hours in the same day. Floating hours are times when no space conditioning is active because the house remains between the heating and cooling control setpoints. For the warm-humid climate, the heating and cooling thermostat setpoints were 73°F and 78°F). The expected space conditioning equipment operation, and the temperature and RH response, is shown.

Integrated Infiltration, Ventilation and Duct Leakage

- Currently integrate infiltration and ventilation (balanced, unbalanced)

$$\begin{aligned}
 \text{cfm}_{\text{in}} &= \text{Vent inlet} + \text{return duct leaks} \\
 \text{cfm}_{\text{out}} &= \text{Exhaust} + \text{Supply duct leaks} \\
 \\
 \text{cfm}_{\text{balanced}} &= \text{MIN}(\text{cfm}_{\text{in}}, \text{cfm}_{\text{out}}) \\
 \text{cfm}_{\text{unbalanced}} &= \text{MAX}(\text{cfm}_{\text{in}}, \text{cfm}_{\text{out}}) - \text{cfm}_{\text{balanced}} \\
 \text{cfm}_{\text{inf}} &= \text{infiltration flow calculated for building for the timestep} \\
 \text{cfm}_{\text{combined}} &= \text{MAX}(\text{cfm}_{\text{unbalanced}}, \text{cfm}_{\text{inf}} + 0.5 * \text{cfm}_{\text{unbalanced}}) \\
 &\quad + \text{cfm}_{\text{balanced}}
 \end{aligned}$$

Figure 4. Calculation of the combined airflow of infiltration, duct leakage, and balanced and unbalanced ventilation.

Conv AC (System 1)

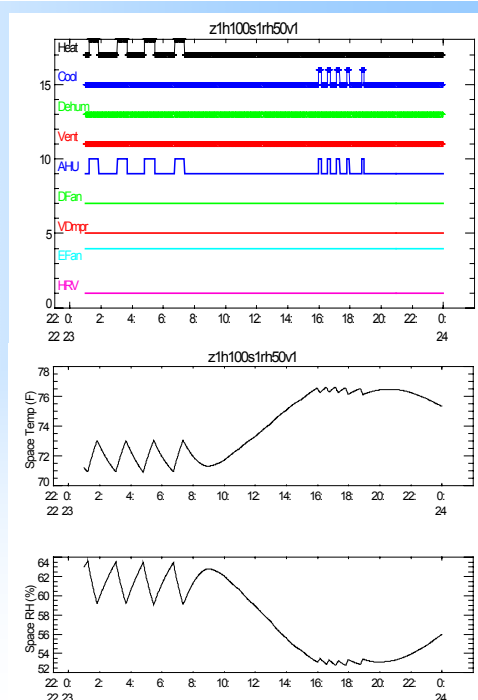


Figure 5. Conventional cooling system and indoor temperature and RH response with heating, floating, and cooling hours in the same day.

Hrs Over 60%

No of 4 hr Events

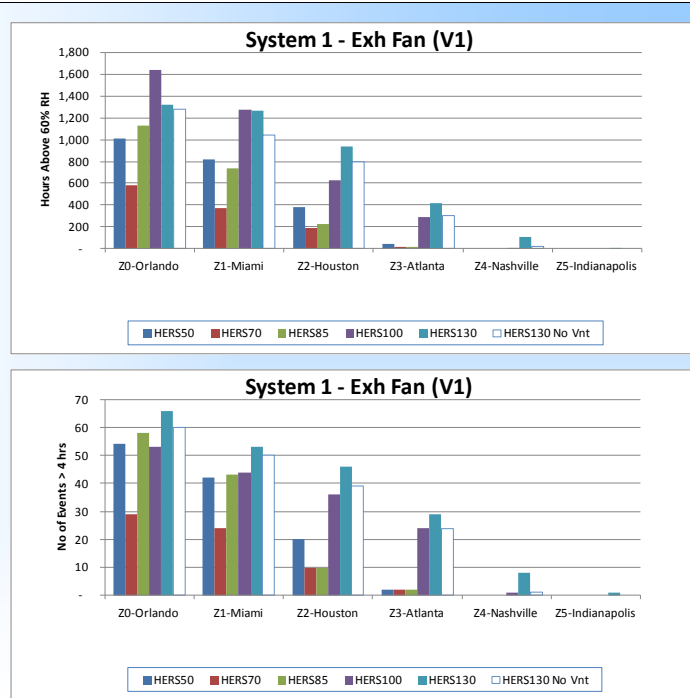


Figure 6. RH limit expressed as hours over 60% RH and 4 hour events over 60% RH.

A number of references (ASHRAE Standard 55-2010, Balaras and Balaras 2007, Wolkoff and Kjaergaard 2007) refer to indoor RH between 30% and 60% as comfortable, healthy, and recommended for human occupancy. In Rudd and Henderson 2007, based on measured data from over 40 sites, an analysis approach was presented whereby hours over 60% RH and 4-hour events over 60% RH were examined. The logic was that if many of the hours over 60% RH occurred sporadically, and did not accumulate to long events, then perhaps both ways of characterizing unacceptably elevated indoor humidity would be needed. It was shown that hours over 60% RH and 4-hour events over 60% RH had essentially the same result profile and gave essentially the same answer. That was because most hours over 60% RH did not occur sporadically, but in lumped time periods. A mostly similar comparison was found when extending the analysis to 8-hour events over 60% RH. Referring to Figure 6, the same result was found in the computer simulations presented here.

Figure 7 shows a sensitivity of ventilation rate for a house rated at a HERS Index of 70 with exhaust ventilation and no cooling system dehumidification enhancements or supplemental dehumidification. The ventilation rate was simulated at 50%, 100%, and 150% of the ASHRAE Standard 62.2-2010 rate. Total space conditioning energy cost typically went down about 10% when the ventilation rate was reduced by 50%, and went up about 10% when the ventilation rate was increased by 50%. However, for the hot-humid climates of Miami, Orlando, and Houston, the hours where indoor RH was above 60% varied much more, and sometimes in different directions. In Orlando, reducing the ventilation rate increased the hours above 60% RH by 72 hours because of less drying potential in wintertime, but increasing the ventilation rate increased the hours above 60% RH by 252 hours. In both Miami and Houston, reducing the ventilation rate

reduced the hours above 60% RH by 63% and 23%, respectively, and increasing the ventilation rate increased the hours above 60% RH by 76% and 35%, respectively.

Impact of Ventilation Rate HERS70

	Exh Vent	Hours Above 60% RH	AC Runtime (hrs)	AC EER (Btu/Wh)	AC Energy (kWh)	Htg Energy (kWh)	AHU Fan Energy (kWh)	Exh Fan Energy (kWh)	HRV Energy (kWh)	Total Electric w/o HT (kWh)	Total Costs w Furnace (\$)	
Orlando	50% Vent	678	1,774	16.6	3,093	554	594	91	-	3,778	419	91%
	100% Vent	606	1,851	16.7	3,270	686	622	203	-	4,095	460	100%
	150% Vent	858	1,915	16.8	3,419	862	646	305	-	4,370	500	109%
Miami	50% Vent	159	2,981	16.6	4,181	25	783	91	-	5,055	512	93%
	100% Vent	433	3,122	16.7	4,430	42	820	203	-	5,453	554	100%
	150% Vent	763	3,247	16.7	4,647	55	853	305	-	5,805	590	107%
Houston	50% Vent	164	2,307	16.3	3,241	1,792	635	91	-	3,967	411	92%
	100% Vent	214	2,391	16.4	3,403	2,045	661	203	-	4,267	446	100%
	150% Vent	288	2,461	16.5	3,537	2,522	687	305	-	4,529	488	109%
Atlanta	50% Vent	27	1,525	16.5	2,098	4,126	468	91	-	2,657	512	91%
	100% Vent	15	1,558	16.6	2,167	4,612	485	203	-	2,855	560	100%
	150% Vent	40	1,577	16.6	2,214	5,251	500	305	-	3,019	614	110%

Figure 7. Impact of exhaust ventilation rate at 50%, 100% and 150% of the ASHRAE Standard 62.2-2010 whole-building ventilation rate.

A sensitivity comparing a conventional cooling system in a HERS 70 level house with cooling systems modified to increase latent capacity and a 50% RH setpoint is shown in Figure 8.

Enhanced AC represents three degrees of overcooling and lowered airflow (210 cfm/ton vs. 375 cfm/ton). This showed a reduction in hours over 60% RH of about 95% in Miami, 75% in Houston, and 50% in Orlando. However, field experience has shown that three degrees of overcooling (cooling to three degrees below the requested setpoint) causes comfort complaints. Note that the total space conditioning system operating cost is not directly comparable between the conventional system and the enhanced system because the enhanced system was modeled with a more efficient cooling system, as typically dictated in the market for systems with variable airflow.

Heat pipes (passive system for pre-cooling air entering the evaporator and reheating air leaving the evaporator) show a reduction in hours over 60% RH of about 90% in Miami, 60% in Houston, and 30% in Orlando. Due to the increased static pressure and associated increase in fan energy consumption associated with the heat pipe system, the total space conditioning system energy consumption and operating cost increased by 20%-25%.

With an indoor RH setpoint of 50%, the Humiditrol system by Lennox (employing overcooling by three degrees and refrigerant subcooling reheat) was able to control indoor RH below 60% in Miami and Houston, but still showed 32 hours above 60% RH in Orlando. Those hours occurred when overcooling had reached its limit.

Enhanced Systems HERS70

		Hours Above 60% RH	AC Runtime (hrs)	AC EER (Btu/Wh)	AC Energy (kWh)	Htg Energy (kWh)	AHU Fan Energy (kWh)	Exh Fan Energy (kWh)	Total Electric w/o HT (kWh)	Total Costs w Furnace (\$)
Orlando	Conv AC	606	1,851	16.7	3,270	686	622	203	4,095	460
	Enhanced AC	308	3,826	20.8	2,756	699	196	203	3,155	366
	AC w/ HPs	418	2,311	16.0	3,799	689	1,126	203	5,128	565
	Lennox Humiditrol	32	4,000	17.0	3,374	703	265	203	3,843	436
Miami	Conv AC	433	3,122	16.7	4,430	42	820	203	5,453	554
	Enhanced AC	20	5,721	19.9	3,772	48	339	203	4,315	439
	AC w/ HPs	46	3,846	16.0	5,064	42	1,482	203	6,749	685
	Lennox Humiditrol	-	5,686	16.9	4,413	42	432	203	5,048	513
Houston	Conv AC	214	2,391	16.4	3,403	2,045	661	203	4,267	446
	Enhanced AC	53	4,016	19.3	2,914	2,045	342	203	3,459	378
	AC w/ HPs	93	2,918	15.9	3,837	2,044	1,157	203	5,197	525
	Lennox Humiditrol	-	4,061	17.3	3,233	2,039	380	203	3,817	408
Atlanta	Conv AC	15	1,558	16.6	2,167	4,612	485	203	2,855	560
	Enhanced AC	15	2,619	19.7	1,819	4,594	270	203	2,292	501
	AC w/ HPs									
	Lennox Humiditrol	-	2,662	19.2	1,879	4,595	274	203	2,356	508

Figure 8. Simulation results for enhanced Systems modeled in HERS 70 level houses.

Turn off capacitance, Check Gains

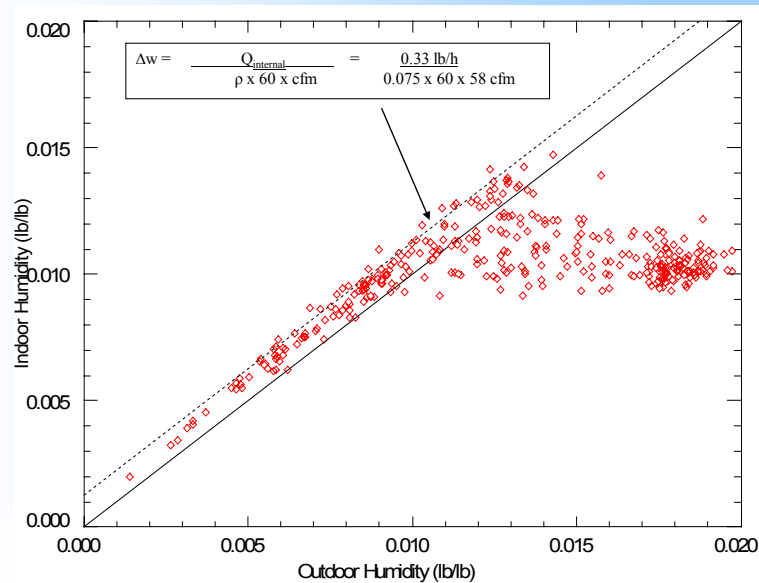


Figure 9. A check of the simulation response with moisture capacitance turned off (set to zero).

Figure 9 provides a graphical check of the expected moisture modeling result with moisture capacitance turned off (set to zero). All values to the right of the solid line represent hours where moisture was being removed from the indoor air by the cooling system. All values between the dotted and solid lines represent hours where cooling was not active and internal moisture generation pushed the indoor humidity up beyond the outdoor humidity. If there was only ventilation and no internal moisture generation, the values would follow the solid line. Values to the left of the dotted line would be indicative of moisture capacitance, as shown in Figure 10 with measured and simulated indoor vs. outdoor humidity trends with indoor moisture capacitance.

A graphic illustration of the impact of overcooling is shown in Figure 11. Comparing the two plots, one can see how overcooling by three degrees moves or “sweeps” the higher RH hours down and to the left.

Comparing Humidity Trends

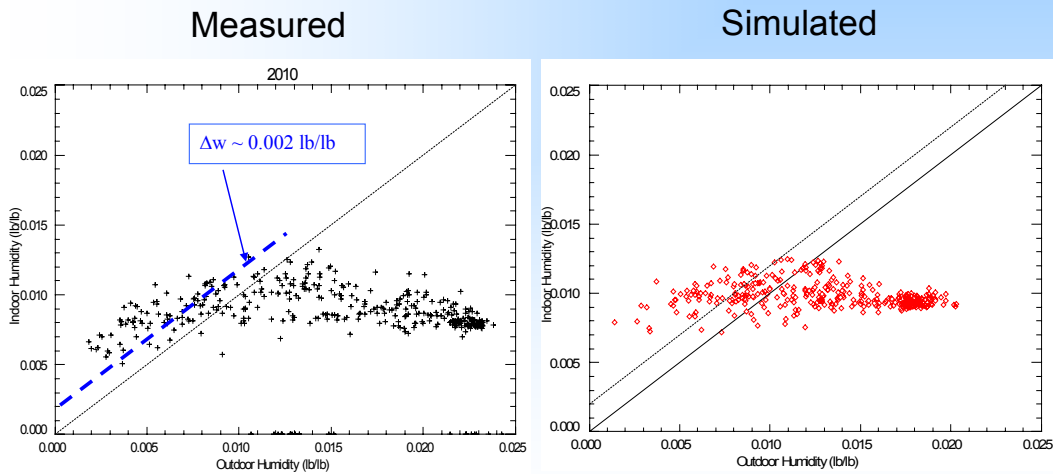


Figure 10. Measured and simulated indoor vs. outdoor humidity trends with indoor moisture capacitance.

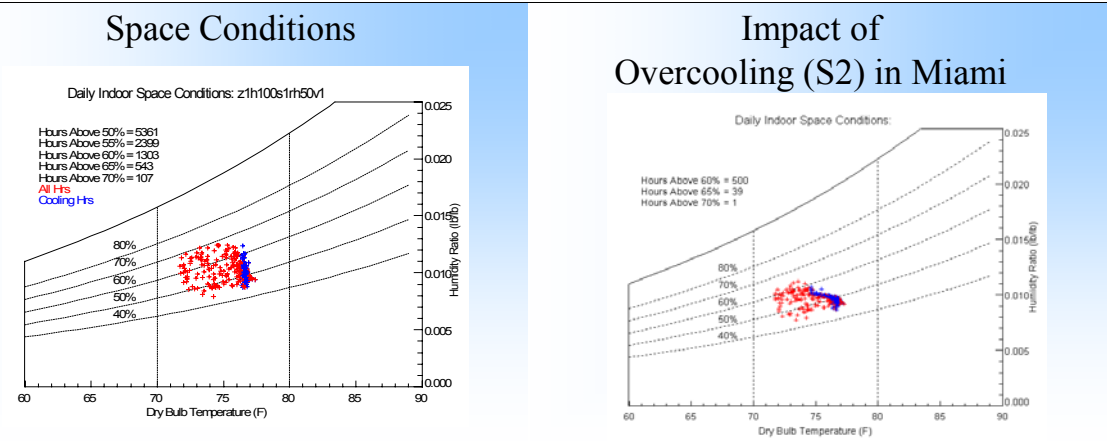


Figure 11. The impact of overcooling by three degrees “sweeps” the higher RH hours down and to the left.

Locating the air distribution system ducts inside conditioned space saves energy overall, but, with the reduced sensible cooling load, also comes an increased need for supplemental dehumidification. This impact is illustrated in Figure 12, where ducts in the attic were modeled

with 5% leakage (60% supply side, 40% return side). For the HERS 70 and HERS 50 efficiency levels, moving the ducts inside conditioned space increases the hours above 60% RH indoors by 25%-50%. A point of discussion in the meeting was that this difference may not be so large if the model were to account for the moisture desorption from wood framing materials that typically increase the attic humidity ratio over that of the outdoors during the late morning to early afternoon hours in warm-humid climates. In that case, return duct leakage would increase the moisture source for the ducts-in-attic configuration. This will be addressed in a simplified way by running an additional sensitivity with the attic dewpoint temperature being forced to 10°F over outdoors from 10 am to 1 pm between May 15 and October 15. However, this may not change the result that much since examining Figure 13 shows that, in Miami, for example, most elevated indoor RH hours occur in night and morning hours between late November and March. Some nighttime and rainy periods during mild summer conditions also produce elevated indoor RH.

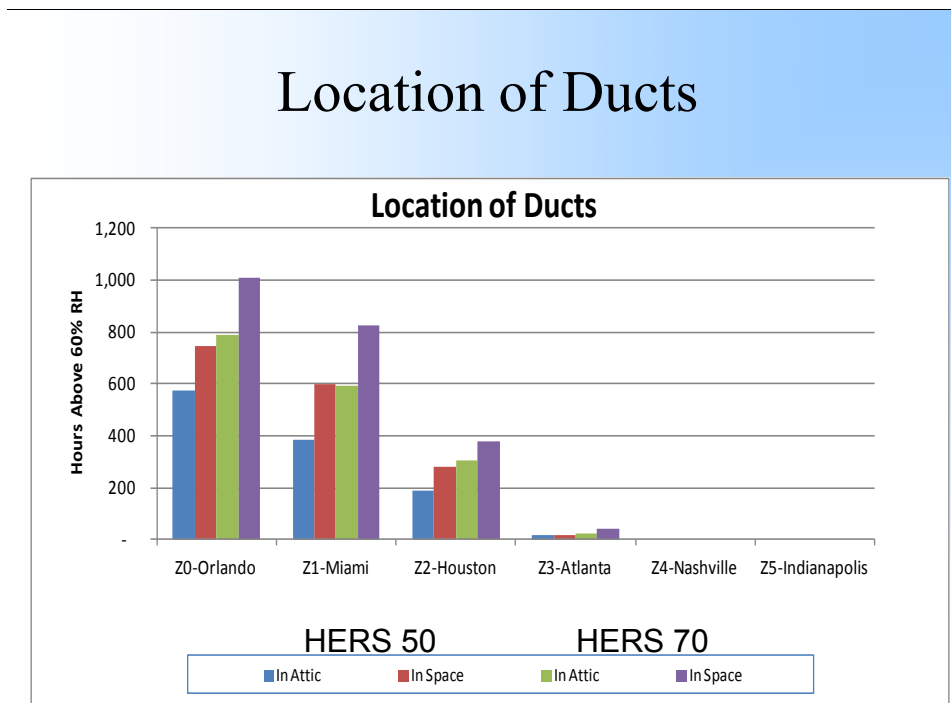


Figure 12. Locating the air distribution system ducts inside conditioned space saves overall energy, but also increases the need for supplemental dehumidification.

Shade Plot of Humidity Bins

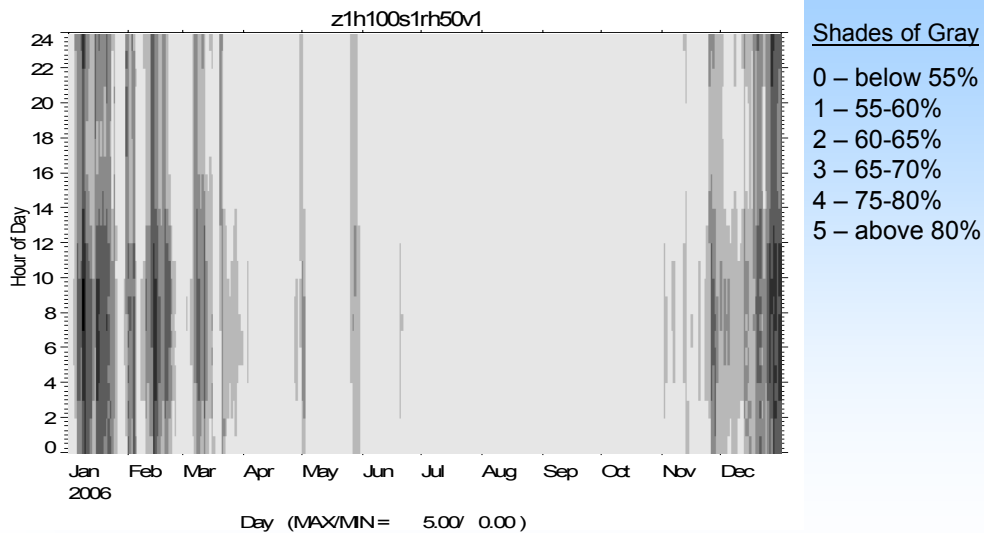


Figure 13. In Miami, hours of elevated indoor humidity occur mostly in morning hours between late November and March.

As shown in Figure 14, internal moisture generation (latent gain) and internal heat generation (sensible gain) strongly impact 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. Internal sensible heat generation of 21 kWh/day is typical for older homes with less efficient lighting and appliances, while 15 kWh/day is more typical of high performance homes with compact fluorescent or LED lighting and ENERGY STAR[®] rated appliances. Similar to ducts in hot attics, very high internal sensible heat gain drives the cooling system to operate more often and for longer runtimes, reducing indoor humidity.

Moisture capacitance does smooth hourly fluctuations in indoor humidity but it turns out that it has a relatively weak impact on hours of elevated indoor humidity (Figure 15). In Figure 15, the 15x, 30x, and 45x designations refer to internal moisture mass of 15, 30, and 45 times the moisture mass held in the house air volume alone. Moisture capacitance has its greatest impact when the cooling system is operating enough to intermittently drive a significant difference in indoor humidity, but that is not when most hours of elevated indoor humidity occur. Most of those hours occur when the space is floating between the cooling and heating setpoints. In that case, referring to Figure 16, raising the heating setpoint 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.

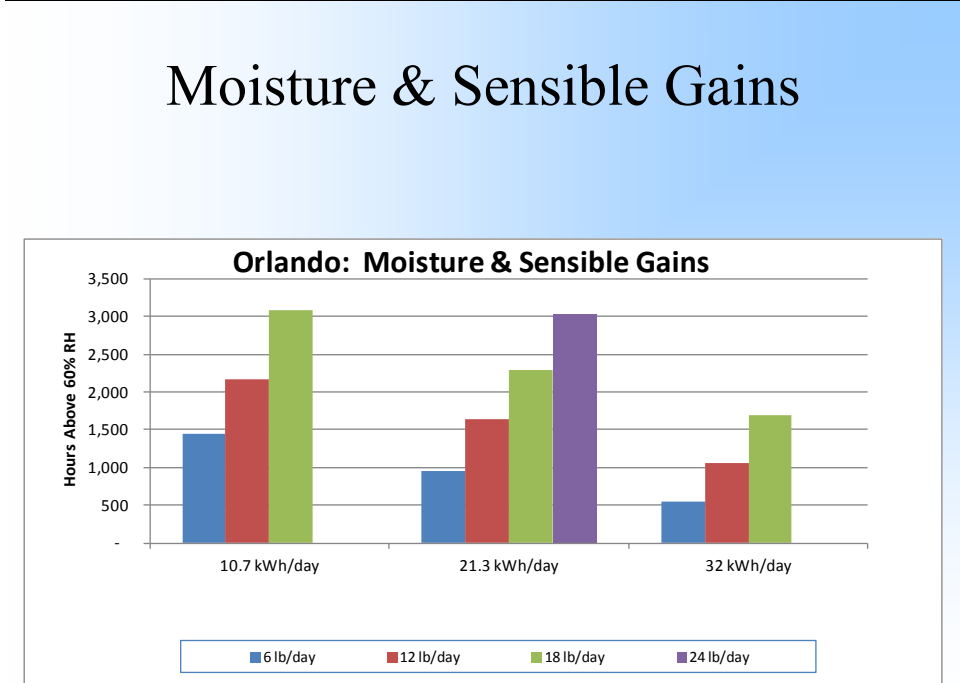


Figure 14. Hours of elevated indoor humidity is strongly related to internal moisture gains and internal sensible gains.

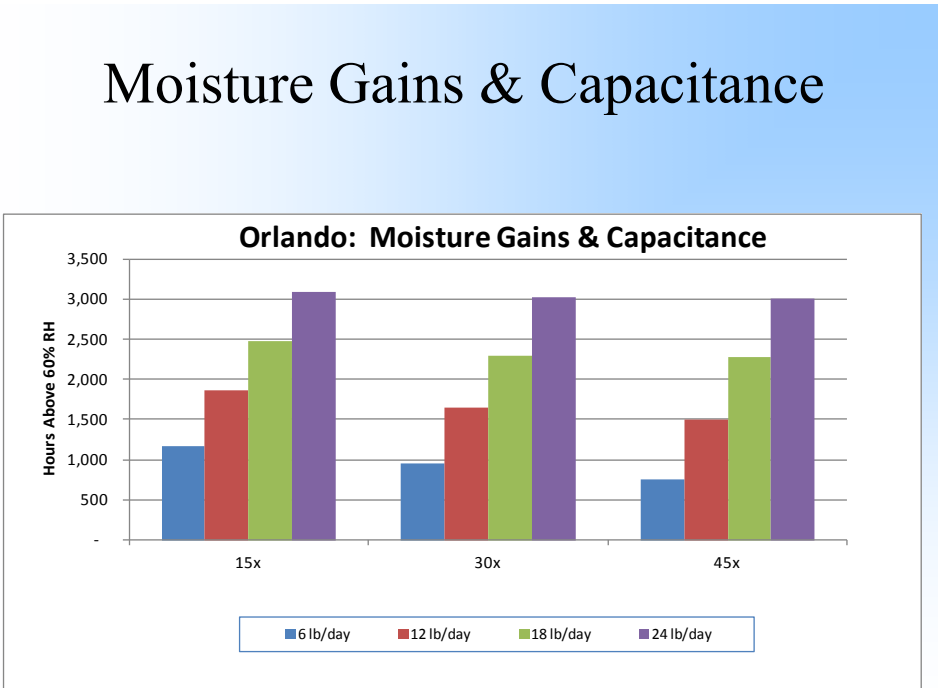


Figure 15. Hours of elevated indoor humidity is weakly related to moisture capacitance.

Sensible & Heating Set Pts

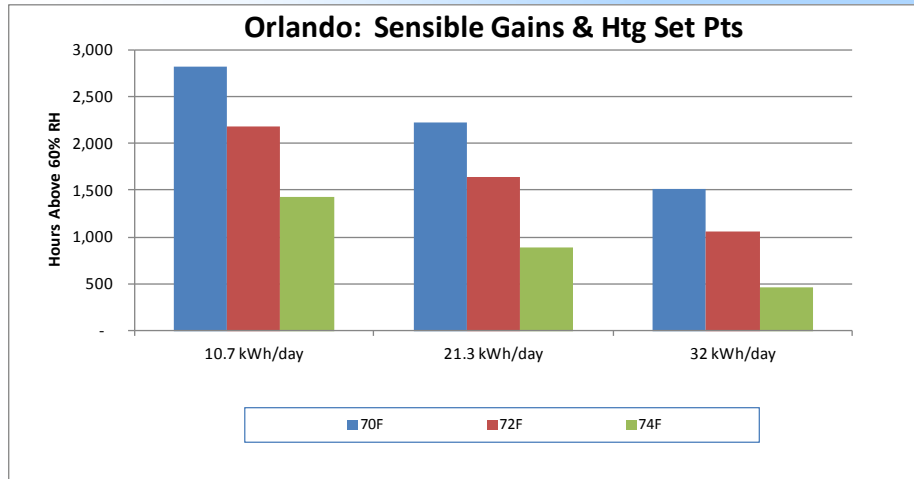


Figure 16. Hours of elevated indoor humidity is strongly related to heating setpoint.

Figure 17 shows how these data are being made available on the Internet via an interactive web application where the user can choose simulation inputs and outputs to suit a particular interest.

Web Access to Results

cloud.cdhenergy.com/rp1449

RP1449

Location	HERS Rating	System Nr.	Relative Humidity	Vent Type
Orlando	HERS50	System 1	50% RH	No Vent
Miami	HERS51	System 2	60% RH	Exh Fan
Houston	HERS70	System 3		Exh 50%
Atlanta	HERS71	System 4		Exh 150%
Nashville	HERS85	System 4L		CFI
Indianapolis	HERS100	System 5		HRV
	HERS130	System 6		ERV

Set Criteria

2: Hours Above 60% RH
 3: Hours Above 55% RH
 4: Hours Above 50% RH
 5: No of Events > 4 hrs
 6: No of Events > 8 hrs
 7: Space Temp Min (F)

Add Data Point

Save
 (Saving Requires Login)

Table11_85_100_130 Load

Data Points Selected
 None

Figure 17. Interactive Internet access is made available to retrieve customized sets of the simulation data.

Humidity Control and Ventilation

(Philip Fairey, Florida Solar Energy Center)

For a Building America ventilation white paper study with an important focus on indoor humidity impacts, twelve U.S. cities and five climate zones (Figure 18) were simulated using a customized version of the FSEC EnergyGauge USA (V3.0.01P) computer program, based on the DOE2 hourly simulation engine. Some of the custom routines were designed to improve the modeling of indoor humidity while using the temperature-only control capability of DOE2. Lumped moisture capacitance was modeled as 10 times the air mass capacitance. Latent degradation due to evaporation of moisture from wet cooling coils during cooling system off cycles was not modeled.

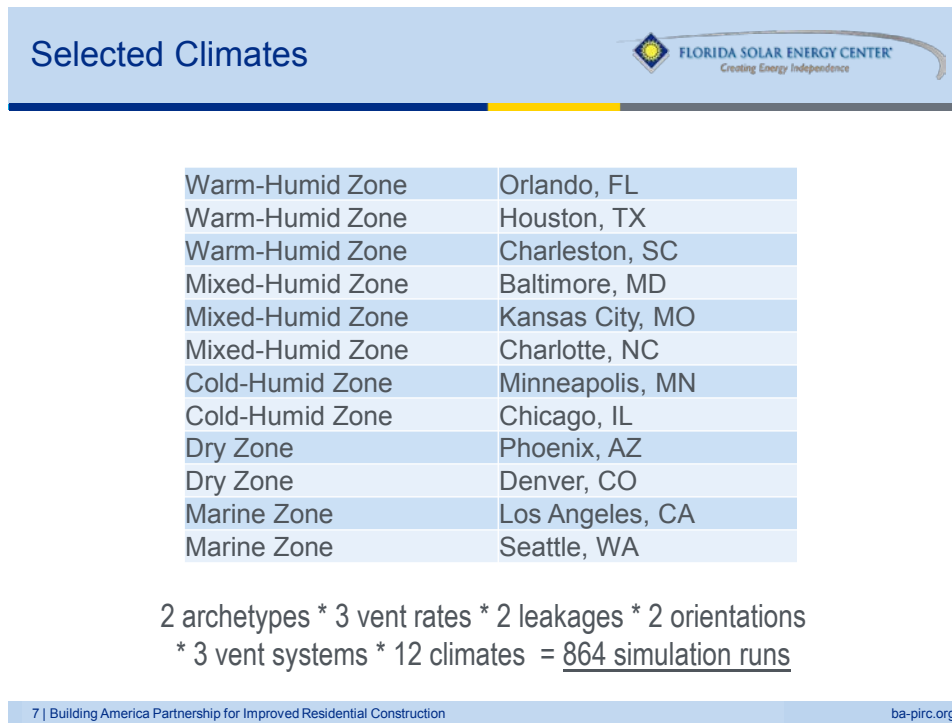


Figure 18. Twelve cities and five climate zones simulated for a ventilation study including humidity impacts.

A detailed set of simulation prototype assumptions were laid down to facilitate identical inputs for the FSEC and IBACOS simulations. Figure 19 lists many of those assumptions. The HERS Index of the configured home (which varied some by climate) was in the mid to upper 50's, falling between the HERS 50 and HERS 70 homes of the CDH simulations. The heating and cooling thermostat setpoints were shifted 2 degrees lower for the FSEC and IBACOS simulations (71°F and 76°F) which would tend to decrease the hours above 60% RH at the edge of cooling demand and increase the hours above 60% RH at the edge of heating demand.

Archetype Homes



- Two home archetypes:
 - 2000 ft², 1-story, 3-bedroom, slab-on-grade, frame
 - 2400 ft², 2-story, 3-bedroom, slab-on-grade, frame
- Both homes:
 - 15% window-floor-area ratio
 - DOE Challenge Home qualification
- HERS Index: mid to upper 50s
- Duct system and AHU located in conditioned space with zero leakage
- Thermostat set points: heating = 71F; cooling = 76F
- Mechanical ventilation is continuous (every hour)
- Window opening for natural ventilation is not allowed

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Other Archetype Assumptions



- Two enclosure leakage rates:
 - 3 ach50 (tight)
 - 1.5 ach50 (very tight)
- Best case and worst case window orientations
- Best case window area distribution:
 - N & S window area = 35% of total
 - E & W window area = 15% of total
- Worst case is rotated 90°
- 16” roof overhang on all sides of hip roof with vented attic
- Infiltration heights (requires modification of DOE-2)
 - 1-story = 9 feet
 - 2-story = 17.5 feet

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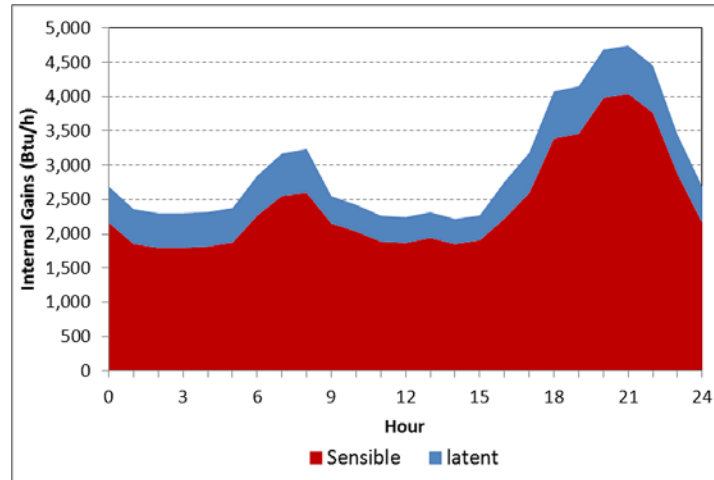
Figure 19. Outline of simulation prototype assumptions.

Internal moisture and sensible heat gains were modeled according to the magnitude and schedule provided in Figure 20. The daily total sensible gain (16.9 kWh) was 20% less than that used in the CDH Energy simulations (21.3 kWh/day), which, according to Figure 14, would tend to increase hours above 60% RH by about 8%. The latent gains of 12 lb/day were the same for both.

Internal Gains Schedule



57,717 Btu sensible gains + 12,698 Btu (12.09 lb) latent gains



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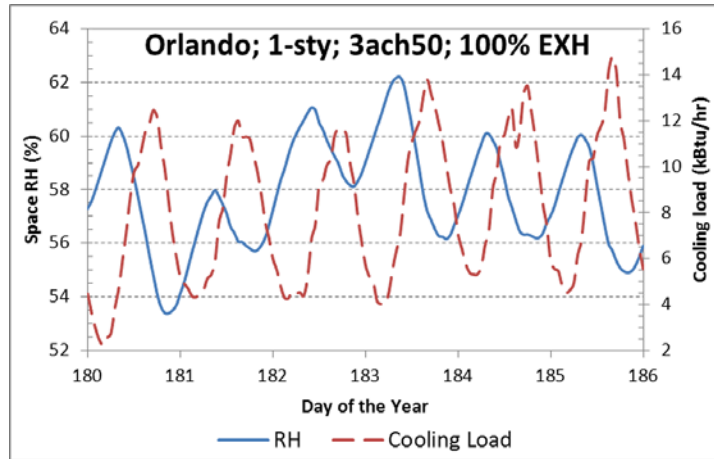
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Figure 20. Internal sensible (16.9 kWh/day) and latent gains schedule used in the simulations.

Figure 22 shows analysis of simulation results for exhaust and ERV ventilation in Orlando. The hours above 60% RH are hours where the RH was above 60%. They 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. In an hour where both heating and cooling occurred, the one with the greater runtime was designated. The simulation assumed that the windows were closed and the space conditioning system was used all year.

As shown in Figure 22, most of the hours significantly above 60% RH occur during floating hours, which occur during fall, winter, and spring in Orlando. The ERV nearly eliminated the cooling hours above 60% RH, but it had little effect on the floating hours above 60% RH. That is because the cooling system forces a greater absolute 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 absolute humidity side would be transferred to the lower absolute humidity side, 60% of a small absolute humidity difference is still a small amount of moisture. In other words, the ERV is ineffective in keeping indoor RH down during floating hours when the difference between indoor and outdoor absolute humidity is small.

Cooling Load and RH

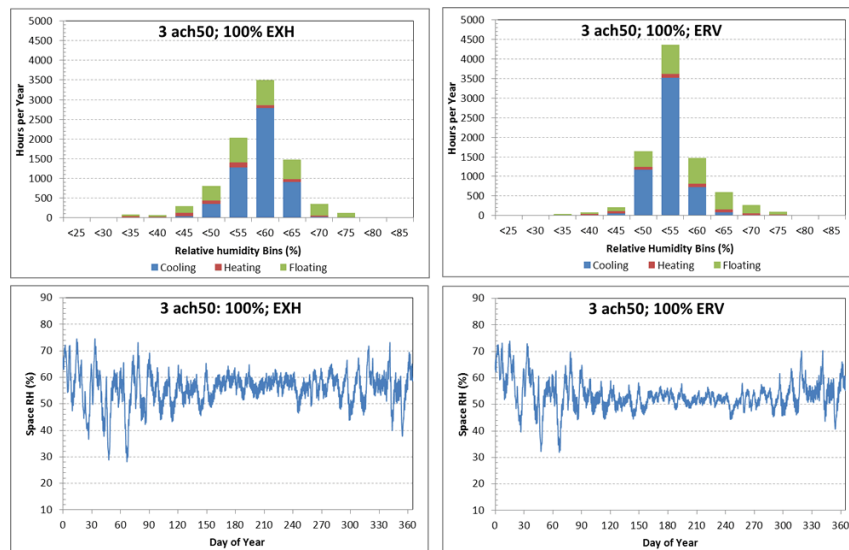


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Figure 21. For a week at the end of June and beginning of July on Orlando, the cooling system and space RH is showing an expected response.

Orlando: Exhaust vs. ERV



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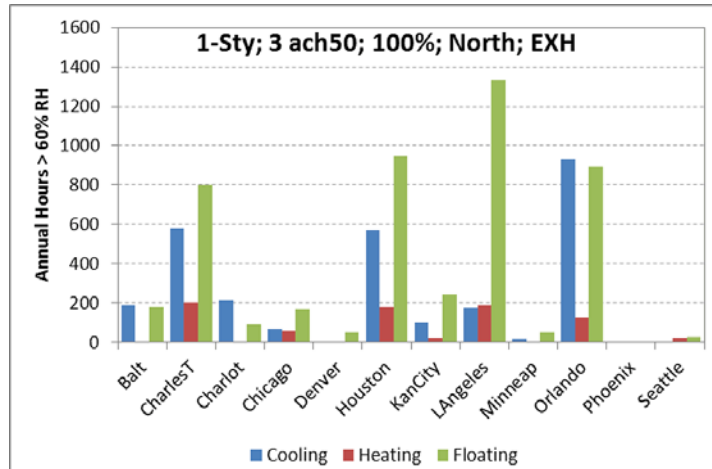
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Figure 22. Hours above 60% RH for exhaust and ERV ventilation in Orlando.

Seasonal Impacts



HVAC condition for hours above 60% is strong function of climate



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Figure 23. Hours above 60% by climate and space conditioning mode.

The hours above 60% RH during a particular space conditioning mode (heating, cooling, floating) was a strong function of climate. Figure 23 shows 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% to 50% greater for floating hours. However, the cooling hours above 60% RH are generally less than 65% RH while, in the warm-humid climates, the floating hours above 60% RH are generally in the range of 65% to 75% RH.

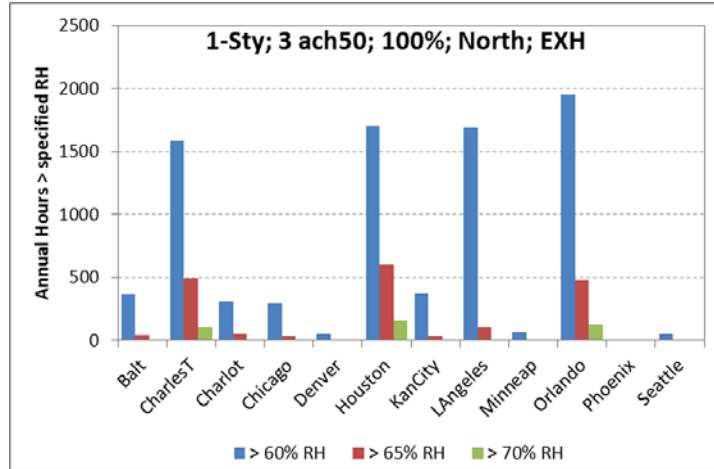
Figure 24 shows that, the hours of elevated indoor RH is a strong function of the selected RH limit and climate. For example, in Los Angeles, nearly all of the hours above 60% RH are also below 65% RH, but that is not true for Charleston, Houston, and Orlando.

Referring to Figure 25, mechanical ventilation, operated at the ASHRAE 62.2-2010 addendum r rate, in a 3 ach50 house, raises the annual median indoor RH by almost 10% RH compared to a 7 ach50 house without mechanical ventilation in Orlando. That is because infiltration drivers are generally weak in that climate during floating hours (when it is still humid outside and the cooling system is not removing moisture), but mechanical ventilation forces a minimum air exchange. Note that most of the RH increase due to ventilation is between 60%-65% RH; the hours between 65%-75% RH mostly remain the same. This indicates that some supplemental dehumidification would be needed in either case anyway.

Climate Severity



Climate Severity is a function of selected RH limit



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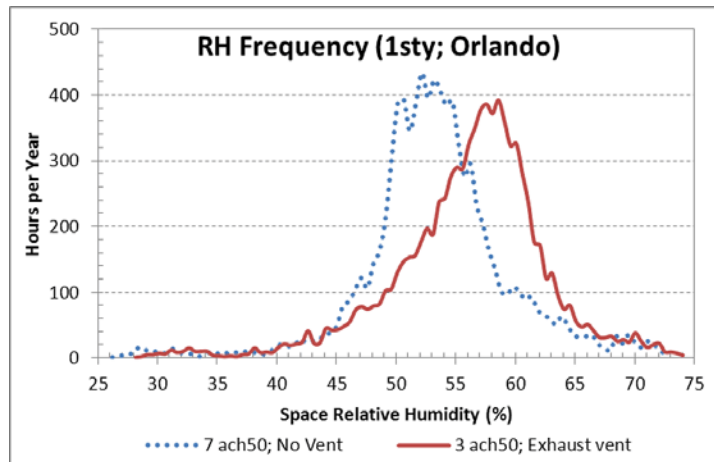
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Figure 24. Hours of elevated indoor RH is a strong function of the selected RH limit and climate.

2009 IECC Envelope Leakage



ASHRAE 62.2 exhaust ventilation raises median RH almost 10%



Internal gains: 57,717 Btu/day + 12.09 lb H₂O/day

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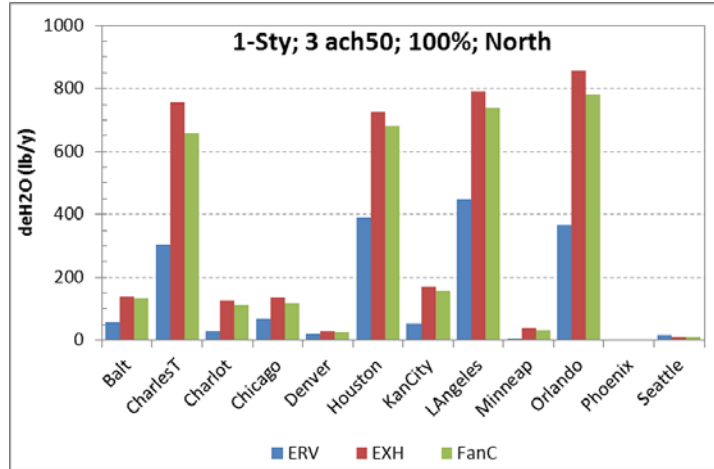
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Figure 25. ASHRAE 62.2-2010 addendum r ventilation rate raises median RH compared to conventional dwelling without mechanical ventilation.

Moisture Excess by Vent Type



ERV systems significantly reduce moisture excess at 60% RH limit



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Figure 26. Predicted required moisture removal by supplemental dehumidification, by ventilation system type (energy recovery ventilator, exhaust, CFI supply).

Shown in Figure 26, removal of indoor moisture by supplemental dehumidification was approximated by the FSEC simulations by a post-processing routine that assumed that during hours over a 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 FSEC model, but are accounted for in the CDH model.

The predicted lb/y values shown in Figure 26 are converted to \$/y in Figure 27 by an assumed dehumidifier Energy Factor of 1.47 L/kWh and \$0.12/kWh. Based on that, supplemental dehumidification was predicted to be between \$10 and \$30/yr for the warm-humid climates of Charleston, Houston, and Orlando, and the marine climate of Los Angeles.

Cost of 60% RH Limit



Theoretical costs could be a factor of 2 low with poor field performance

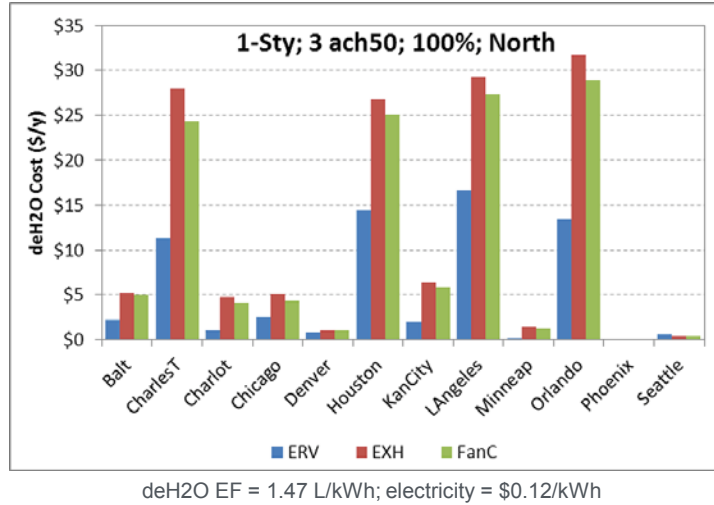


Figure 27. Predicted cost of supplemental dehumidification.

EnergyPlus Humidity Control and Ventilation Modeling Analysis

(Michael Sypolt and Duncan Prael, IBACOS)

IBACOS ran simulations in EnergyPlus that are parallel to those that FSEC ran in the customized version of EnergyGaugeUSA. The NREL BEopt E+ program was used to generate the geometry and base IDF files. A custom script modified parameters for about 860 runs which could be run overnight.

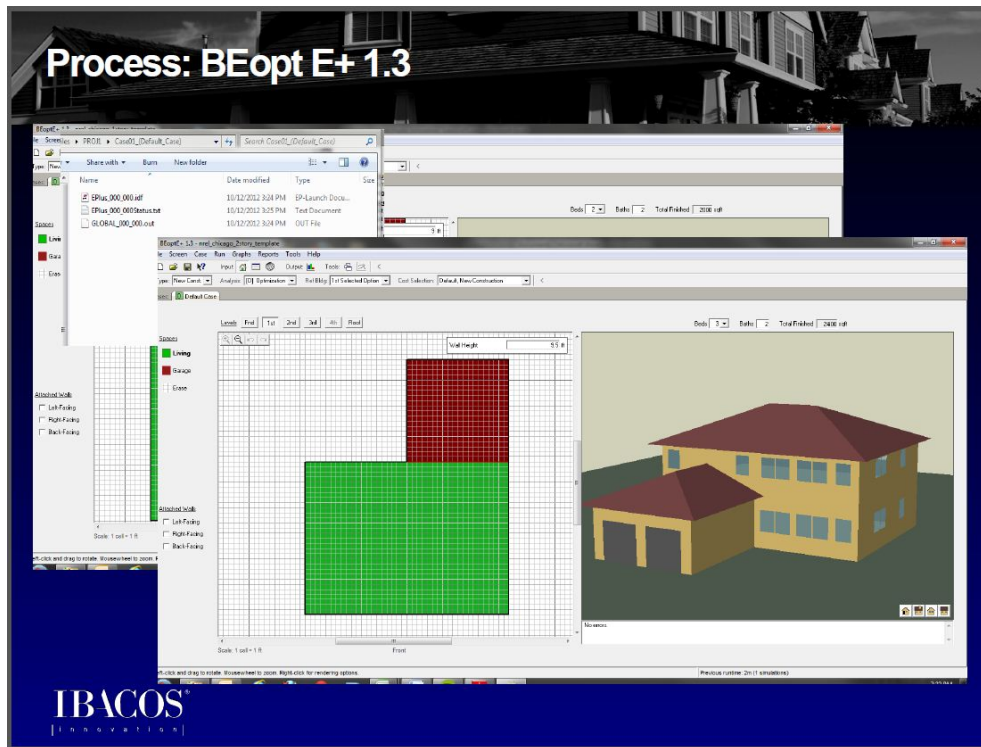


Figure 28. NREL BEopt E+ program used to generate the building geometry and the base IDF files (input description).

The model results were compared to the FSEC model results, and some input differences were corrected. Some result differences still exist. The EnergyPlus results generally show a greater number of hours over 60% RH indoors, and hours that extend to higher RH, compared to the FSEC (see Figure 29) and CDH simulations, and compared to much of BSC field data (Rudd et al. 2003, BSC 2004, Rudd et al. 2005). These might be due to differences in the infiltration model, or the infiltration-ventilation superposition. There may also be differences in the thermal energy balance model, which could lead to different loads, as another possibility. The EnergyPlus moisture modeling inputs were double checked by IBACOS and re-checked by FSEC without finding any significant discrepancies. The lumped moisture capacitance factor and moisture generation rate were consistent with the FSEC and CDH inputs.

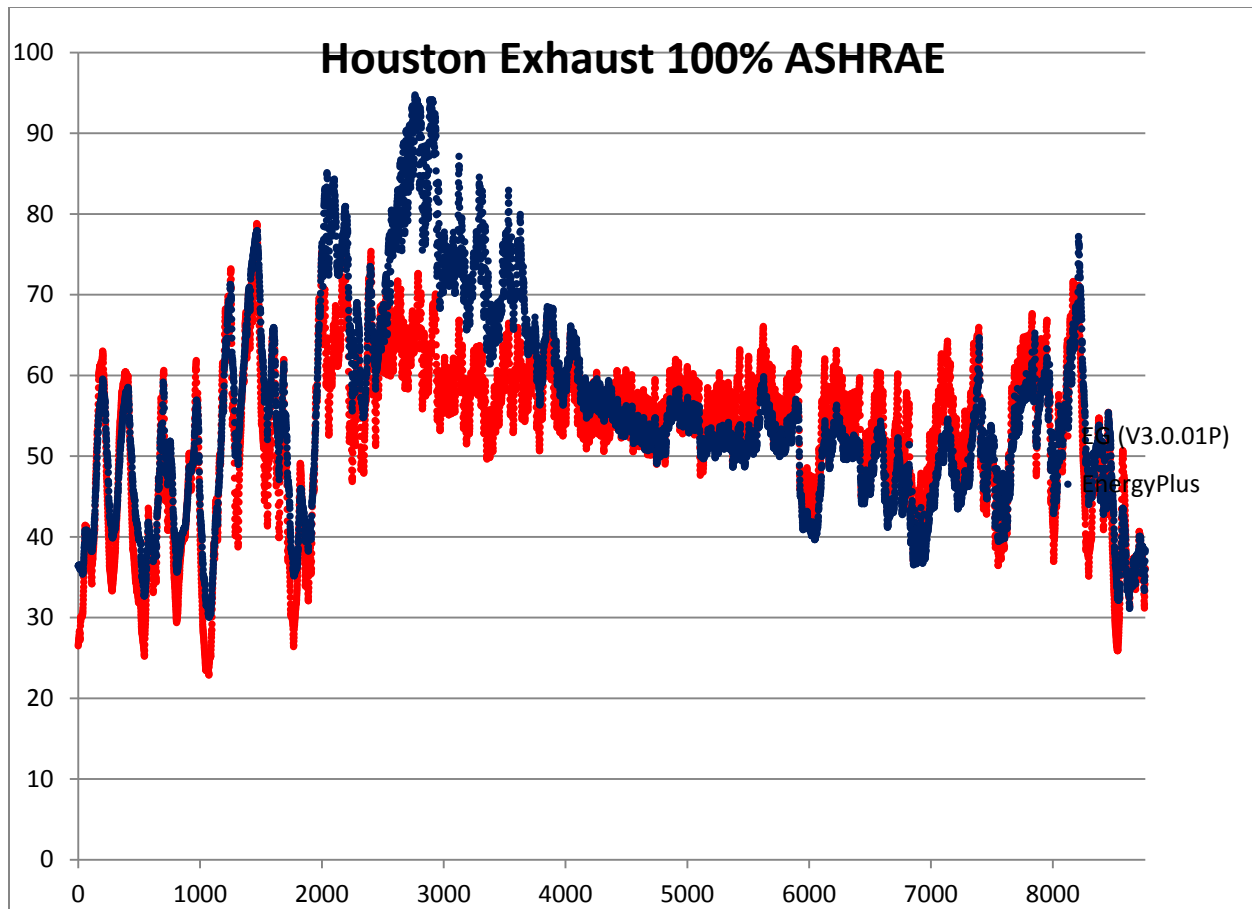


Figure 29. Indoor RH vs. hour of year comparison between EnergyPlusPlus (blue symbols) and EnergyGauge USA (V3.0.01P) simulation results.

Figure 30, shows the energy consumption results by ventilation system type, ventilation rate, and building enclosure tightness. EnergyPlus shows the differences to be relatively small in each case.

Referring to Figure 31, using the IBACOS EnergyPlus model, an ERV (not integrated with the central AHU) reduces the number of hours with lowest and highest indoor RH in Orlando. Overall, there is a reduction in indoor RH, and a very large reduction in hours above 60% RH compared to the FSEC (not integrated with the central AHU) and CDH (integrated with the central AHU) models. This result is suspect, but the cause is unclear. In each case, the ERV moisture performance was modeled pretty simply as a constant effectiveness value applied to the actual indoor to outdoor humidity ratio difference. Of the three, the CDH model shows the highest hours above 60% RH because the ERV was configured as being connected between the return and supply of the central AHU, as most ERVs are installed. That installation configuration requires the AHU fan to operate coincident with the ERV fan (in this case a 50% duty cycle), which increased energy consumption and moisture evaporation when the cooling coil was wet. The CDH model calculates the amount of water retained on the cooling coil as a function of cooling runtime. Then, it calculates coil moisture evaporation starting each time the compressor

stops, and when the fan runs without the compressor. The FSEC and IBACOS models configured the ERV as separately ducted from the central air distribution system, and did not model coil moisture evaporation when the compressor was off.

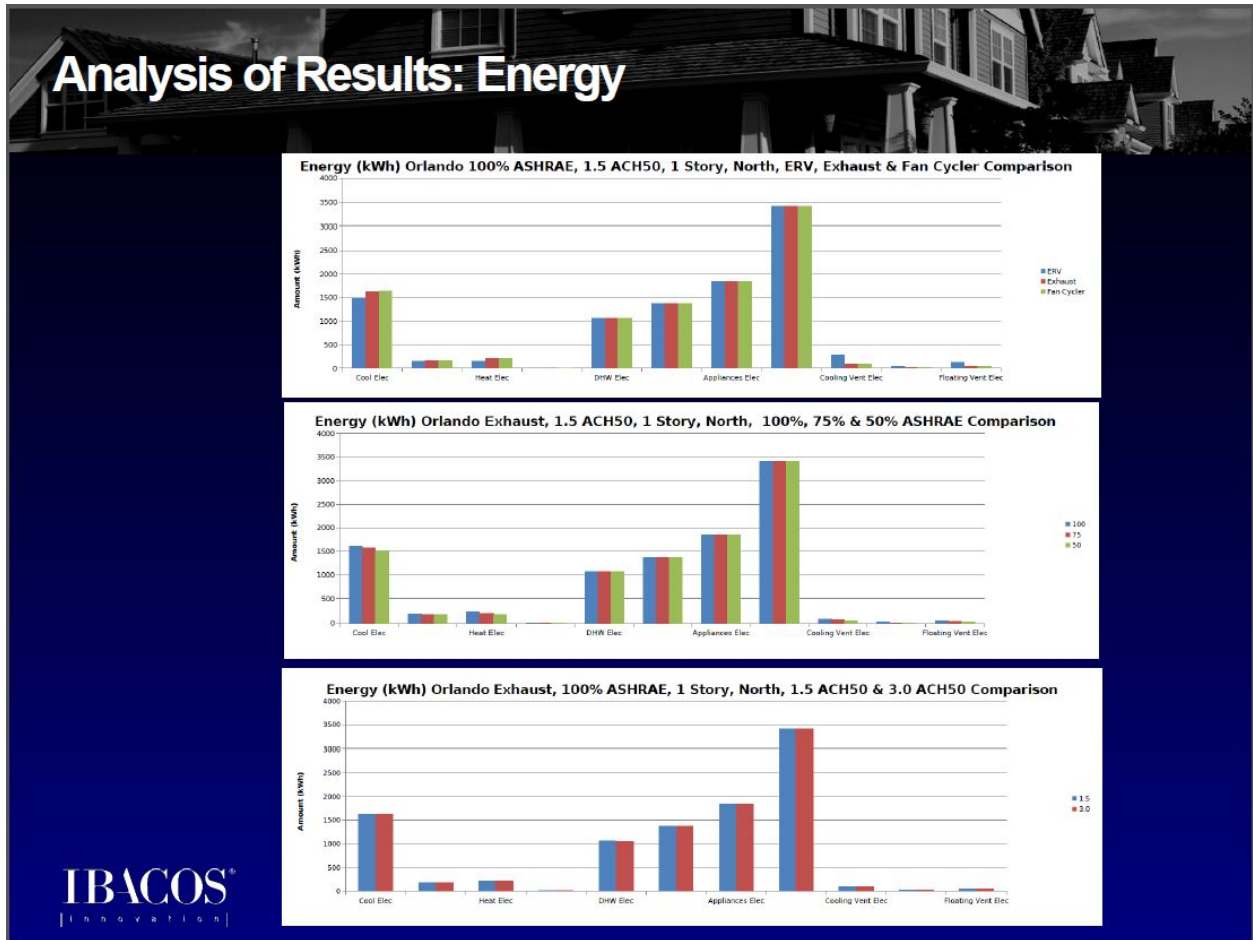


Figure 30. Energy consumption results by ventilation system type, ventilation rate, and building enclosure tightness.

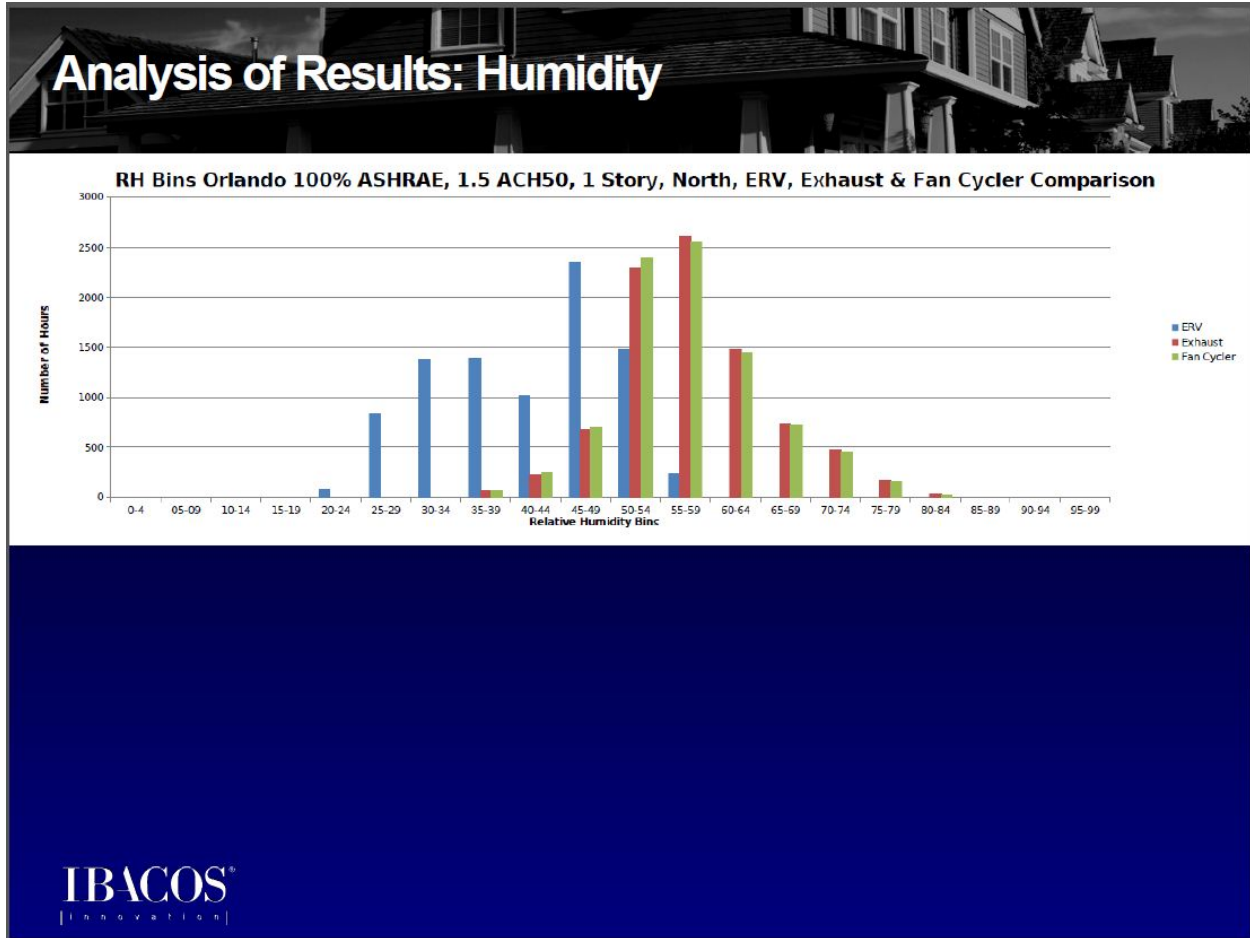


Figure 31. An ERV reduces the number of hours with lowest and highest indoor RH, and a very large reduction in hours above 60% RH compared to the FSEC and CDH models.

Analysis was conducted to look at window condensation potential and how that could impact a selected indoor RH limit from a durability perspective. Previous research (Arena et al. 2010) reported the highest occurrence of visible mold or moisture damage was on or around windows and in bathrooms. The window that was modeled was standard double pane with vinyl frame. For that window in Orlando, indoor RH above 75% shows risk of center-of-glass window condensation, especially during hours when the space conditions were floating between the heating and cooling set points (Figure 32). With a less thermally efficient window frame (non-thermally broken aluminum is common in Florida due to high wind resistance requirements), a lower indoor RH threshold would be needed to avoid condensation on or near the window frame.

Orlando: Float

		Indoor Relative Humidity Bins																		Grand Total		
		0to4	5to9	10to14	15to19	20to24	25to29	30to34	35to39	40to44	45to49	50to54	55to59	60to64	65to69	70to74	75to79	80to84	85to89	90to94	95to99	Grand Total
Outdoor Temperature Bins	-20to-16	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	-15to-11	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	-10to-6	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	-5to-1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	0to4	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	5to9	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	10to14	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	15to19	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	20to24	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	25to29	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	30to34	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	35to39	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	40to44	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	45to49	0	0	0	0	0	0	0	0	0	0	1	8	2	9	0	0	0	0	0	0	0
	50to54	0	0	0	0	0	0	0	0	0	6	21	33	46	20	4	0	0	0	0	0	0
	55to59	0	0	0	0	0	0	0	0	6	26	46	72	80	43	22	24	7	0	0	0	0
	60to64	0	0	0	0	0	0	0	8	22	47	117	161	143	88	68	36	6	0	0	0	0
	65to69	0	0	0	0	0	0	0	12	21	43	81	85	123	152	87	48	4	0	0	0	0
	70to74	0	0	0	0	0	0	0	10	10	26	60	62	96	79	50	14	0	0	0	0	0
	75to79	0	0	0	0	0	0	0	8	8	16	41	41	46	44	24	16	0	0	0	0	0
	80to84	0	0	0	0	0	0	0	3	5	10	14	16	15	15	2	0	0	0	0	0	0
	85to89	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	0	0	0	0	0	0
	90to94	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	95to99	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
100to104	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
105to109	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
110to114	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
115to119	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Total RH bin		0	0	0	0	0	0	44	96	205	425	491	496	405	269	123	17	0	0	0	2571	

Figure 32. Window condensation potential during Orlando floating hours

7 Answers to Research Questions

- What are the important humidity control conclusions from recent residential dehumidification systems modeling, with a wide range of building and equipment sensitivity, using a customized <2 minute time-step TRNSYS model with temperature and humidity control?
 - Internal moisture generation, internal sensible heat generation, heating setpoint temperature, and air distribution system duct location (inside or outside of conditioned space) are the major influencers of elevated indoor RH in high performance homes in warm-humid climates.
 - Mechanical ventilation is a secondary factor in increasing hours above 60% RH.
 - Moisture capacitance is a secondary factor, and lumped moisture mass factors of 10 to 30 times the air mass capacity do not show significant difference in indoor RH.
 - Supplemental dehumidification by either a stand-alone dehumidifier, ducted dehumidifier, or gas-fired desiccant dehumidifier (e.g. NovelAire ComfortDry 400 in supply air stream) was effective in eliminating hours over 60% RH. In Orlando, Miami, and Houston, the difference in total HVAC operating cost between a HERS 70 level house with those systems was 15%, 11%, and 1%, respectively. The gas-fired desiccant system showed the lowest operating cost, followed by the ducted dehumidifier and the stand-alone dehumidifier.
 - Cooling system enhancement by subcooling condenser reheat was effective in eliminating hours over 60% RH in Miami and Houston, and nearly so in Orlando.

- Heat pipe cooling system enhancement was less effective than three degrees overcooling plus low evaporator coil airflow (210 cfm/ton). Both were more effective in Miami and Houston than in Orlando, but even in Orlando the hours over 60% RH were reduced by about half (600 down to about 350 on average). However, questions remain as to whether the overcooling can be tolerated from a comfort perspective.
- Two-speed and variable speed cooling systems do not appreciably reduce hours above 60% RH unless coupled with lower than standard cooling airflow (cfm/ton).
- What are the important indoor humidity conclusions that can be drawn from recent residential ventilation modeling efforts using the temperature-only control logic of the FSEC EnergyGauge USA program?
 - Most of the hours significantly above 60% RH occur during floating hours, which occur during fall, winter, and spring in Orlando.
 - In Orlando, an ERV nearly eliminated the cooling hours above 60% RH, but it had little effect on the floating hours above 60% RH.
 - Hours above 60% RH during a particular space conditioning mode (heating, cooling, floating) was a strong function of climate.
 - Hours of elevated indoor RH is 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.
 - Mechanical ventilation, operated at the ASHRAE 62.2-2010 addendum r rate (about 25% more than before addendum r), in a 3 ach50 house, raises the annual median indoor RH by almost 10% RH compared to a 7 ach50 house without mechanical ventilation in Orlando. However, most of the RH increase due to ventilation is between 60%-65% RH; the hours between 65%-75% RH mostly remain the same. This indicates that some supplemental dehumidification would be needed in either case anyway.
 - Supplemental dehumidification was predicted to be between \$10 and \$30/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 EF of 1.47 L/kWh and recent field data indicates 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 deadband, which means that maintaining humidity below 60% would likely require humidity setpoints near 55%, which would dramatically increase dehumidification costs.

- What are the important humidity control conclusions from recent residential ventilation modeling efforts using the temperature and humidity control logic of the Energy Plus version of the NREL BEopt program?
 - Energy consumption by ventilation system type, ventilation rate, and building enclosure tightness showed relatively small differences in each case compared to total HVAC energy consumption.
 - Durability analysis for Orlando, for a standard double pane window with vinyl frame, showed that indoor RH above 75% risks center-of-glass window condensation, especially during hours when the space conditions are floating between the heating and cooling set points. With a less thermally efficient metal window frame, the risk would be greater.
- How do the results from these different programs compare?

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 CDH, FSEC, and IBACOS results show hours above 60% RH in the range of 1000, 2000, and 3000, respectively. In terms of maximum indoor RH, the CDH and FSEC results both show about 75% while the IBACOS results show about 85%. However, in all cases, supplemental dehumidification would be required for high performance, low-energy houses, and the difference in supplemental dehumidification energy would probably be less than 150 kWh/yr or less than \$20/yr.

The IBACOS ERV modeling using EnergyPlus is suspect of showing too much reduction in elevated indoor humidity hours compared to both the CDH and FSEC models, however, the CDH model did not model the ERV as a completely separate system from the central system ducts. After a sensitivity run by CDH is done with an ERV as a completely separate system, the results will be more directly comparable.

- Is it generally agreed that controlling to less than 60% RH is the appropriate humidity control point for high performance homes, and why?

It was generally agreed that, a dehumidification control setpoint of 55%, in order to keep indoor RH from exceeding a 60% RH limit, was the correct strategy for high performance, low-energy homes. While it is clear that everything will not fail at once if the indoor RH goes over 60%, a 60% RH limit provides the best practice coverage for providing comfort and durability over a reasonable range of varying factors, such as internal moisture generation rate, and occupant comfort perception and susceptibility to illness stemming from elevated indoor humidity. Included in the variability of internal moisture generation rate is construction moisture drying. It has been BSC's experience that limiting indoor RH to 60% via supplemental dehumidification is a generic enough limit to remove moisture concerns related to the seasonal timing of building closure and occupancy in warm-humid climates.

- Is it generally agreed that annual hours above 60% RH is the appropriate humidity control performance metric to use to compare system performance and to compare

required supplemental dehumidification energy? Does that metric give generally the same result as looking at 4-hour and 8-hour events above 60% RH?

It was generally agreed that annual hours above 60% RH is the single most appropriate humidity control performance metric to use to compare system performance and to compare required supplemental dehumidification energy. That metric does give generally the same result as looking at 4-hour and 8-hour events above 60% RH.

- For a range of climates, ventilation systems, and space conditioning equipment in high performance homes, what is the magnitude of hours above 60% RH, the magnitude of supplemental dehumidification energy required to control to less than 60% RH, the time of year occurrence of elevated indoor humidity and supplemental dehumidification, and the space conditioning mode (heating, cooling, floating) during which most periods of elevated indoor humidity and supplemental dehumidification occur?

The answers to this question are given in more detail in section 6, Agenda Summaries, but are summarized here. The warm-humid climates of Miami, Orlando, Houston, and Charleston show a clear need for supplemental dehumidification for high performance homes. Without supplemental dehumidification, hours above 60% RH were in the range of 800 to 1800, with hours above 65% being about half of that. Most of the hours of elevated indoor humidity occur in the mild temperature but humid outdoor conditions of fall and spring, but also occur in winter in Orlando and Miami. A smaller number of hours occur during some summer nights and days-long rainy periods. Few hours above 60% RH occur during heating hours. Most hours between 60%-65% RH occur during either cooling or floating hours, and most hours above 65% RH occur during floating hours. The supplemental dehumidification energy consumed to keep indoor RH below 60%, taken as the difference in total HVAC cost for the same building with and without supplemental dehumidification, is relatively small. It is in the range of 250 kWh/yr or less (\$30/yr or less), but necessary to enable deep cuts in sensible heat gain without incurring long periods of elevated indoor RH.

8 Action Items

The following action items were discussed at the end of the meeting:

- Continue to do more checking to understand the EnergyPlus moisture modeling differences compared to the FSEC and CDH modeling.
- Add a sensitivity run using the CDH model for an ERV not integrated with the central air distribution system.
- Add a sensitivity run using the CDH model to further detail the difference between ducts in the attic and ducts in the conditioned space, where the attic dew point temperature is increased 10°F between 10 am and 1 pm between May 15 and October 15.

- Contact Association of Home Appliance Manufacturers (AHAM) to explore the prospect of developing a new residential dehumidifier standard encompassing two test points in addition to the one test point in their existing standard, and reporting of additional test results.
 - From discussion of Figure 33, the group proposed a new dehumidifier test condition of 75°F and 55% RH to represent spring/fall part load cooling conditions, and a new test condition of 65°F and 50% RH to represent basement conditions where avoiding high RH or condensation on cool below-grade concrete walls and floors is needed. These new test conditions could replace the AHAM test conditions or could be in addition to the legacy test condition of 80°F and 60% RH in the current AHAM standard if considered necessary on a consensus basis, primarily as a transitional measure for comparison to legacy test results. The AHAM test point of 80°F/60% RH is not a common indoor environmental condition for occupied spaces, limiting its usefulness to consumers and designers. Thermodynamics dictate that for the same RH set point, dehumidifier energy consumption will be higher at lower, more common indoor dry-bulb temperatures. NREL reported performance maps of several dehumidifiers at various operating conditions (Winkler et al. 2011).

Test condition represents:		Inlet T/RH/Tdp (F/%/F)	Outlet T/RH/Tdp (F/%/F)	Indoor Wet-coil Airflow (cfm)	Sensible Cooling Capacity ¹ (Btu/h)	Latent Cooling Capacity (Btu/h)	Moisture Removal Capacity (L/h)	Total Power (kW)	Moisture Removal Efficiency ² (L/kW-h)
Current AHAM test	Test 1	80/60/65							
Spring/Fall part load cooling ³	Test 2	75/55/58							
Basement	Test 3	65/50/46							
¹ Negative cooling capacity denotes net heat added from inlet to outlet									
² Same units as the USDOE and USEPA Energy Factor for dehumidifiers									
³ All tests with steady wet coil									

Figure 33. Overview of recommended dehumidifier test conditions.

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