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ABSTRACT

With the Building America (BA) program advancing to the 50% source energy savings level, it is becoming a high priority to properly model whole house moisture loads and operation of humidity control equipment. In a 50% savings home, space sensible cooling load is significantly reduced by high-performance envelopes, EnergyStar appliances, and efficient lighting. Conventional air-conditioning equipment is unlikely to maintain healthy space relative humidity during key periods of the year. In addition, with the significant reduction of the other space energy consumption, dehumidification equipment energy consumption becomes a bigger portion of whole house energy consumption.

A parametric study was conducted using EnergyPlus version 4.0 to analyze the impact of various dehumidification equipment and control strategies on a typical mid-1990's reference home, a 2006 IECC home, and a high-performance home in a hot humid climate. Space relative humidity, thermal comfort, and whole house energy consumption were analyzed. The study was compared with past published modeling results (Henderson et al., 2008). The results confirmed that supplemental dehumidification must be provided in a high-performance home in order to maintain space conditions below 60% relative humidity. A detailed analysis was conducted to examine indoor relative humidity excursions; specifically the number of excursions, average excursion length, and maximum excursion length. Recommendations were made on dehumidification options in high-performance homes.

1. INTRODUCTION

As the Building America (BA) program begins constructing homes that achieve 50% and greater source energy savings, modeling whole house moisture loads and operation of humidity control equipment properly has become a high priority. In a 50% savings home, space sensible cooling load is significantly reduced due to high-performance envelopes, EnergyStar appliances, and efficient lighting. Conventional air-conditioning equipment is unlikely to

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maintain healthy space relative humidity (RH) during key periods of the year in humid climates since a majority of the time when the space relative humidity goes out of control is when the air-conditioner (A/C) does not run or runs at part load conditions. Because of the significant reduction in whole house energy consumption, dehumidification equipment energy use becomes a bigger portion of whole house energy consumption. Improved simulation of these systems' interactions will enable optimization of a home's features for durability, healthy indoor air quality, minimizing energy use and system cost, and other similar goals.

A parametric study was conducted using EnergyPlus version 4.0 (DOE, 2009) to analyze the impact of various dehumidification equipment and control strategies on a high-performance home at the 50% source energy savings level. A typical mid-1990's reference home and 2006 code compliant home serve as basis for comparison. The analysis uses the city of Houston, which represents the hot-humid climate region. Living space relative humidity, thermal comfort, and whole house energy consumption were examined.

The paper presents analysis results examining relative humidity "excursions," defined as a unique event of indoor relative humidity above a threshold. For each mechanical system 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 as little as 24 hours, under ideal conditions. "*If the space relative humidity consistently stays above 70% for an extended period of time, mold will almost certainly grow*" (Epidemiology, 2003). One 24-hour long RH excursion promotes much more mold growth than twenty-four intermittent 1-hour long RH excursions, and therefore, does much more harm to the house. Mold is known to reduce the home's durability and result in degraded indoor air quality, so significant excursions must be controlled to ensure a healthy home.

Past studies (Henderson et al., 2008) were conducted using TRNSYS to analyze space air-conditioning equipment with and without humidity control strategies. The study used the TRNSYS 16 multi-zone type 56 building model to analyze either natural infiltration with Sherman-Grimsrud model or constant mechanical ventilation as prescribed by ASHRAE 62.2-2004. This paper examines combined infiltration and mechanical ventilation effects with both balanced and unbalanced mechanical ventilation systems in EnergyPlus. The infiltration model from Walker and Wilson (1998) was used. The Walker and Wilson infiltration model was developed specifically for residential buildings and includes more detailed coefficients compared to the Sherman-Grimsrud method. In addition, a sensitivity analysis was conducted to examine the range of house leakage areas vs. space relative humidity.

2. MODELING APPROACH AND ASSUMPTIONS

2.1 Descriptions of Homes

Three homes were used in the analysis: a typical mid-1990's reference home which was adopted from the BA Benchmark Home (Hendron and Engebrecht, 2010), a 2006 International Energy Conservation Code (IECC) 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 (2008) was used for all three homes. The house used in the analysis was a two-story, three-bedroom, 2500 square foot, slab-on-grade home. The house was assumed to be west facing with an 18% window-to-floor area ratio. The parameters that varied between the three homes that have the largest impact on the living space humidity levels are shown in Table 1.

The heating and cooling set points were established by BA benchmark definition at 71°F (21.67°C) for heating and 76°F (24.4°C) for cooling constant 24/7 year round and a constant relative humidity (RH) set point of 55%, when applicable. Based on the BA benchmark definition the house had 3 occupants that followed an occupancy schedule such that the house had fewer occupants during the daytime hours. The average daily internal moisture loads listed in Table 1 were generated using the benchmark prescribed load profiles for water use fixtures (such as the shower, bath, and sinks) and appliances (such as the clothes washer and dishwasher). The difference in the internal moisture loads among the three homes is due to EnergyStar appliances. Remaining schedules and assumptions are consistent among the three homes and can be found in the BA benchmark definition.

Table 1: Features of the three homes used in the analysis

	Reference Home	IECC 2006 Home (Climate Region 2)	High-performance Home
Wall Assembly	2x4, 16" on center, R-11 cavity	2x4, 16" on center, R-13 cavity	2x6, 24" on center, R-21 cavity
Ceiling Assembly	R-20 assembly	R-30 assembly	R-30 assembly
Windows	U-value = 1.0 SHGC = 0.79	U-value = 0.75 SHGC = 0.40	U-value = 0.35 SHGC = 0.26
Specific Leakage Area (SLA)	0.00057	0.00036	0.00015
Duct Location	Vented Attic	Vented Attic	Conditioned Space
Duct Leakage	R-5 10% Supply Leakage 5% Return Leakage	R-8 10% Supply Leakage 5% Return Leakage	N/A
Average Daily Internal Moisture Loads	8.6 kg/day	8.3 kg/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

2.2 Dehumidification Parametric Case Descriptions

Case 0 represents the high-performance home equipped only with a typical air-conditioner for space conditioning as described in Table 1. The cooling set point is constant at 76°F (24.4°C).

2.2.1 Thermostat Reset (Case 1): Despite the air-conditioning system maintaining a well controlled cooling set point of 76°F (24.44°C) there are hours where space relative humidity reaches high levels, resulting in reduced thermal comfort. Under these circumstances, it is fairly common for occupants to dial down the thermostat set point a couple of degrees, causing the air conditioner to run which reduces indoor temperature and humidity based on the equipment's Sensible Heat Ratio (SHR). This comfort control practice brings down the space relative humidity and the quantity of hours exceeding 60% relative humidity. The model set a home Predicted Mean Vote (PMV) limit of ±0.5 when the occupants are expected to actively change the thermostat (7:00am – 7:00pm) and a relaxed PMV limit of ± 1.0 during the rest of the day.

2.2.2 A/C with Energy Recovery Ventilator (ERV) (Case 2): In the summer time, an ERV cools and dries the hot and humid outside ventilation air by exchanging heat and moisture into the (conditioned) house exhaust air. Pre-treating the ventilation air permits downsizing the A/C cooling capacity. Using an ERV for mechanical ventilation will reduce the space humidity level and the amount of hours of RH exceeding 60%. The modeled ERV is a static enthalpy heat exchanger with an average winter/summer effectiveness of 75% / 60%.

2.2.3 Heat Exchanger Assisted A/C (Case 3): A heat-exchanger-assisted cooling coil has a heat exchanger wrapped around the direct-expansion (DX) cooling coil, as shown in Figure 1. The air-conditioner inlet air is first pre-cooled at the passive heat exchanger. This process improves the latent removal performance of the cooling coil by allowing it to dedicate more of its cooling capacity toward dehumidification (lower SHR). The cold air leaving the coil then is rewarmed somewhat by the passive heat exchanger and exits the equipment. Similar to the previous TRNSYS model (Henderson et al., 2008), a heat pipe heat exchanger was modeled with an average sensible effectiveness of 32%.

2.2.4 A/C with Condenser Reheat (Case 4): A DX cooling coil with condenser reheat is a type of equipment that can actively control the space relative humidity. Under normal operation, the A/C operates to meet the space thermostat cooling set point. When the space RH exceeds the set point, the DX cooling coil further reduces the leaving air temperature down to meet the space RH set point. The dehumidified supply air is then reheated by the condenser coil downstream. A desuperheater module was set in the EnergyPlus model with maximum allowed heat reclaim efficiency of 30%. Because the modeled package needs to dehumidify the entire supply air volume to meet the space humidity set point, an excessive amount of air conditioning energy may be required.

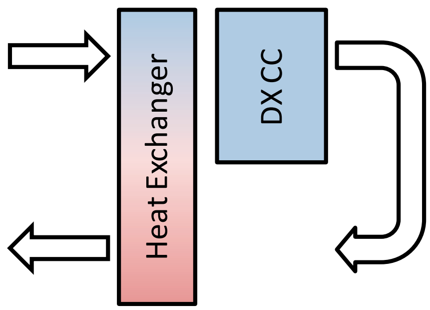

Figure 1: Heat Exchanger Assisted Cooling Coil

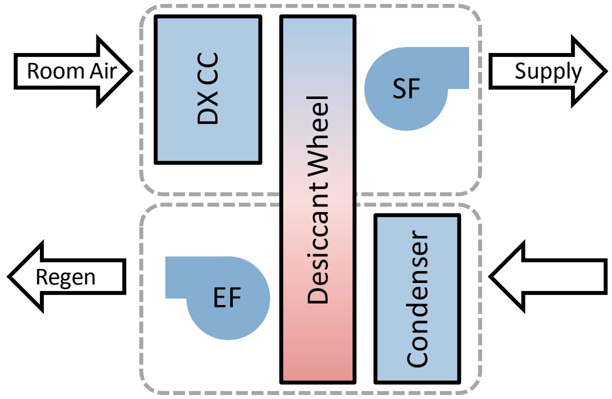

Figure 2: A/C with Desiccant Wheel Dehumidifier

2.2.5 A/C with Desiccant Wheel Dehumidifier (Case 5): A desiccant dehumidifier regenerates the desiccant wheel with heat rejected by the condenser, as shown in Figure 2. By separating the supply airstream from the regenerative airstream, the unit supplies cool and dry air to the home along with reduced A/C runtime. A room air recirculating desiccant wheel dehumidifier was used in the model. The model had a SEER 13 DX cooling coil (DXCC), a moisture removal capacity of 56.8 L/day, energy factor 3.3 L/kWh, and rated air flow at 250 cfm. Estimated supply fan and regeneration fan efficiency is 0.55 W/cfm, counting pressure drop across the desiccant wheel. The default desiccant wheel performance curve from EnergyPlus was used.

2.2.6 A/C with High Efficiency DX Dehumidifier (Case 6): A fairly simple solution for controlling relative humidity is a vapor compression dehumidifier. A high-performance 30.75 L/day dehumidifier was used in the analysis with a rated energy factor of 2.0 L/kWhr. Performance data was taken from Christensen and Winkler (2009) and it was assumed that the normalized performance curves generated from test data could be applied to a similarly configured unit of smaller capacity. The DX dehumidifier supplies warm and dry air to the living space since it converts the latent heat removed and the electric power consumed to heat rejected to the supply airstream. As a result, operating a DX dehumidifier is seen to increase A/C runtime.

2.2.7 A/C with Standard Efficiency DX Dehumidifier (Case 7): A standard efficiency dehumidifier was included in the analysis to determine the energy savings of using a high efficiency unit. The standard efficiency unit was assumed to have the same capacity as the high efficiency unit at reduced energy factor of 1.5 L/kWhr (minimal EnergyStar cutoff point). Due to the lack of performance data on such a unit, the normalized performance curves for the high efficiency unit were applied.

2.2.8 A/C with ERV and High Efficiency DX Dehumidifier (Case 8): This case combines technologies used in Cases 2 and 6.

2.2.9 A/C with ERV and Standard Efficiency DX Dehumidifier (Case 9): This case combines technologies used in Cases 2 and 7.

3. MODELING RESULTS AND ANALYSIS

3.1 Predictions of Relative Humidity Excursions

An RH set point of 55% was used for cases that actively controlled the space relative humidity. Table 2 and Table 3 show simulation results for RH excursions above 60% and 70%, respectively.

Relative humidity levels reach and maintain unhealthy levels for the high-performance home equipped with only a typical air-conditioner (Case 0). The RH exceeds 60% for an 86 hour period and exceeds 70% for a 35 hour period, almost guaranteeing mold growth. Adjusting the thermostat in attempt to maintain comfort (Case 1) does little to reduce the space relative humidity over the base case. Incorporating an ERV (Case 2) into the house reduces the overall number of hours of high relative humidity, but does nothing to reduce the maximum excursion length. Similarly, the heat-exchanger-assisted cooling coil (Case 3) also reduces the total number of hours with high humidity, but the maximum excursion length is not significantly affected.

Table 2: Relative humidity excursions above 60% RH

	Ref. Home	2006 IECC Home	High-performance Home Cases									
			0	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**
Hrs. Above 60% RH	3,056	3,495	3,141	3,110	2,489	2,023	87	6	0	0	0	0
Number of Excursions	302	263	247	249	254	250	313	19	0	0	0	0
Avg. Exc. Length (hrs)	10.1	13.3	12.7	12.5	9.8	8.1	0.3	0.3	0	0	0	0
Max. Exc. Length (hrs)	70	87	86	75	93	75	0.8	0.8	0	0	0	0

Table 3: Relative humidity excursions above 70% RH

	Ref. Home	2006 IECC Home	High-performance Home Cases									
			0	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**
Hrs. Above 70% RH	1,014	1,304	968	956	717	606	0	0	0	0	0	0
Number of Excursions	175	225	191	192	83	83	0	0	0	0	0	0
Avg. Exc. Length (hrs)	5.8	5.8	5.1	5.0	9.7	7.3	0	0	0	0	0	0
Max. Exc. Length (hrs)	20	32	35	35	43	34	0	0	0	0	0	0

The remaining technologies control space relative humidity quite well. Even though there are periods throughout the year where the A/C with condenser reheat (Case 4) and the desiccant wheel dehumidifier (Case 5) cannot maintain the 55% RH set point, the maximum excursion length in both cases is less than one hour and does not pose a concern for mold growth. The results show that both the standard and high efficiency DX dehumidifier (Cases 6 – 9) control the space relative humidity extremely well, which is evident by there being zero hours above 60% RH in the living space.

Henderson et al. (2008) examined humidity in a Home Energy Rating System (HERS) reference house (which was consistent with the 2004 IECC minimum efficiency standards). That simulation used a single story house with SLA of 0.00047, incorporated the Sherman-Grimsrud infiltration model, had no mechanical ventilation and contained a typical air-conditioner. That work reported that indoor relative humidity exceeded 60% for only 1,017 hours. However, modeling results for the reference home and 2006 IECC home in the current study showed that relative humidity exceeded 60% for 3,056 and 3,495 hours, respectively, as shown in Table 2.

Table 4 shows a progression of modifying the simulation of the reference house from the current study to match the HERS reference house presented by Henderson et al. (2008). The number of stories dictates the stack and wind coefficients used by the Sherman-Grimsrud model. The results show that slight modifications in the simulation inputs lead to large differences in the predicted relative humidity. By changing the weather data from the typical meteorological year (TMY) 3 to TMY2, and the number of stories from 2 to 1, the modeling results showed much closer approximation to the past study results. Based on the previous publication dates of similar work by the authors, it is likely that weather data from the typical meteorological year (TMY) 2 dataset was used. The remainder of the discrepancy in the amount hours exceeding a relative humidity of 60% can be attributed to differences in the housing characteristics and assumed occupant behavior.

Table 4: Predicted relative humidity sensitivity based on simulation input parameters

Ventilation	SLA	Infiltration Model	Weather Data	Number of Stories	Hrs. Above 60% RH
Spot Ventilation	0.00057	Walker & Wilson	TMY3	2	3,056
None	0.00057	Walker & Wilson	TMY3	2	3,017
None	***0.00048***	Walker & Wilson	TMY3	2	2,747
None	0.00048	***Sherman-Grimsrud***	TMY3	2	2,243
None	0.00048	Sherman-Grimsrud	***TMY2***	2	1,457
None	0.00048	Sherman-Grimsrud	TMY2	***1***	1,359

3.2 Predictions of Thermal Comfort

Figure 3 displays the percentage of people dissatisfied (PPD) based on the predicted mean vote (PMV) with the most comfortable region falling between PMV values of ±0.5. The PMV is dependent on the dry-bulb temperature, relative humidity, air speed, activity level, and clothing level. A clothing schedule was created to take into account of seasonal variation. PMV will generally exceed a value of +0.5 during hot and humid periods. Table 5 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 PMV of 0.5 do not correlate to the trends in relative humidity presented in Table 2 and Table 3. This can be explained by taking a closer look at the ASHRAE comfort regions displayed in Figure 4. The comfort regions extend over a fairly large range in relative humidity compared to temperature; meaning humans are not as sensitive to humidity as to temperature. This also indicates that the humidity problem within a home may not be noticed by the occupants. 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 2.2% of the year for all the cases.

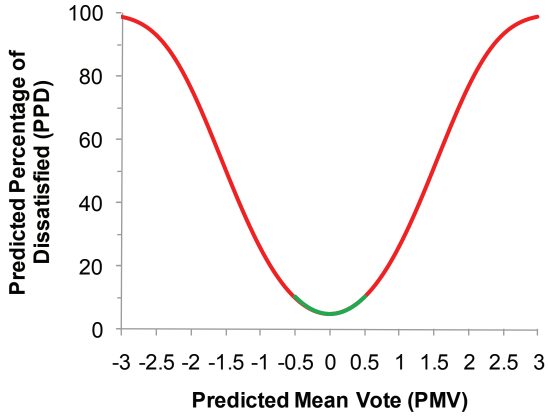

Figure 3: Predicted percentage dissatisfied as a function of predicted mean vote (ASHRAE, 2004)

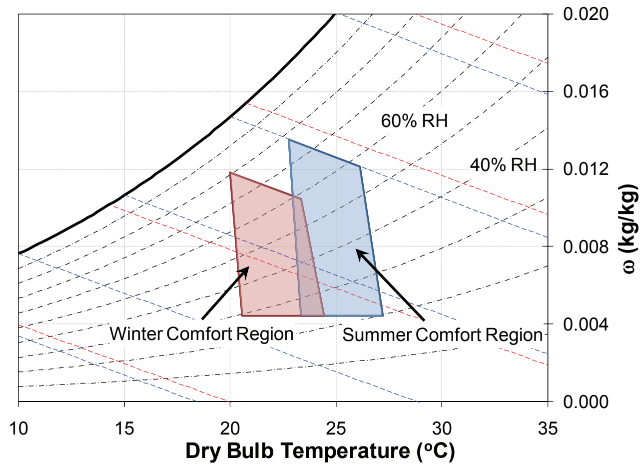

Figure 4: Psychrometric chart with seasonal comfort regions (ASHRAE, 2009)

Table 5: Thermal Comfort statistics based on PMV

	Ref. Home	2006 IECC Home	High-performance Home Cases									
			0	1	2	3	4	5	6	7	8	9
Maximum PMV Value	0.86	0.73	0.69	0.64	0.69	0.68	0.56	0.56	0.60	0.60	0.60	0.60
Hrs. Exceeding PMV=0.5	618	308	161	73	186	140	23	54	142	146	152	155

3.3 Energy Consumption Summary

Table 6 shows the annual source energy consumption for the high-performance home cases. Source-to-site conversion ratios are 3.365 for electricity and 1.092 for natural gas. Adding active humidity control equipment (Case 4-9) increases whole house source energy consumption. In the current analysis, 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. As expected, A/C with Condenser Reheat (Case 4) results in the highest increase of source energy consumption.

Table 6: Source energy consumption summary for parametric cases (GJ/Yr)

	Case 0	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9
Misc+ Large Appl.	66.40	66.40	66.40	66.40	66.40	66.40	66.40	66.40	66.40	66.40
Lights	9.05	9.05	9.05	9.05	9.05	9.05	9.05	9.05	9.05	9.05
Ventilation Fans	1.95	1.95	4.50	1.95	1.95	1.95	1.95	1.95	4.50	4.50
H/C Fans	4.21	4.27	4.14	4.64	6.19	7.37	4.58	4.61	4.30	4.35
Cooling & Dehumidification	30.76	30.89	30.25	33.85	47.05	36.45	44.02	47.05	38.57	40.38
Heating	8.91	9.32	7.71	8.98	10.94	9.32	8.51	8.48	7.30	7.27
Hot Water	9.86	9.86	9.86	9.86	9.86	9.86	9.86	9.86	9.86	9.86
Total	131.15	131.73	131.92	133.84	151.43	140.40	144.38	147.41	140.00	141.81
Percent Increase	-	0.5%	0.6%	2.1%	15.5%	**7.1%**	10.1%	12.4%	**6.7%**	8.1%

Table 7: Parametric cases HVAC energy consumption summary

	Case 0	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9
A/C Run Time (Hrs)	1,641	1,654	1,750	1,841	2,625	1,386	1,829	1,856	1,863	1,880
Dehumidifier Run Time (Hrs)	0	0	0	0	0	1,288	1,177	1,128	732	688
Vent Fan (kWh/Yr)	161	161	372	161	161	161	161	161	372	372
H/C Fan (kWh/Yr)	347	353	356	383	511	608	378	381	356	358
Heating (kWh/Yr)	736	769	636	742	903	769	703	700	603	600
Cooling & Dehumidification (kWh/Yr)	2,539	2,550	2,497	2,794	3,883	3,069	3,633	3,883	3,183	3,333
Total HVAC (kWh/Yr)	**3,783**	**3,833**	**3,861**	**4,080**	**5,458**	**4,607**	**4,875**	**5,125**	**4,514**	**4,663**

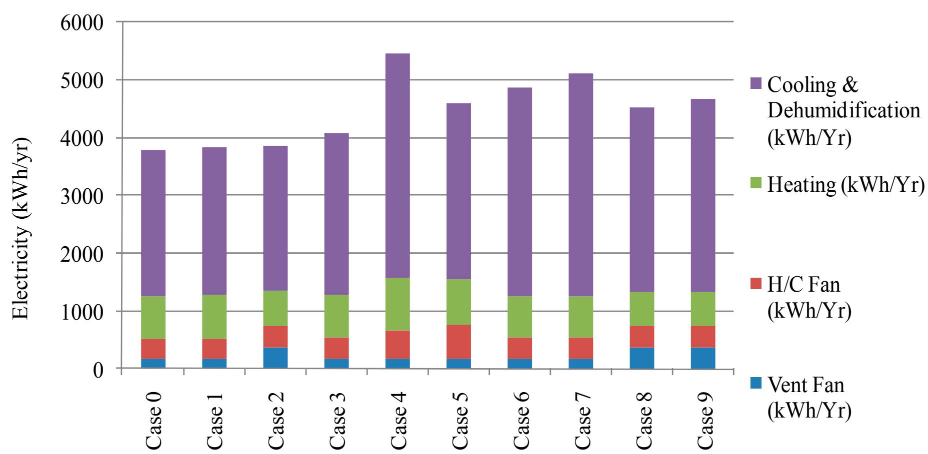

Figure 5: Parametric cases HVAC summary

Table 7 and Figure 5 are a close up look at the cases heating, ventilation, and air-conditioning (HVAC) annual electricity consumption. The energy consumption seen in Figure 5 is a necessary cost of maintaining home durability and healthy indoor air quality. Higher-efficiency solutions to space conditioning and humidity control may help achieve these requirements. A few observations were made:

- A/C and Standard Efficiency DX Dehumidifier (Case 7) showed a slight increase in A/C runtime over the A/C and High Efficiency DX Dehumidifier (Case 8). A/C with Desiccant Wheel Dehumidifier (Case 5) reduced A/C runtime.
- The system auto-size option in EnergyPlus reduced A/C cooling capacity when an ERV was used. The corresponding A/C runtime did not change much, but overall A/C energy consumption was reduced.
- Operating a standard efficiency DX dehumidifier (Case 7 and 9) showed minimal electrical energy increase compared to operating a high efficiency DX dehumidifier (Case 6 and 8). A/C with Desiccant Wheel Dehumidifier (Case 5) showed more energy savings than A/C with High Efficiency DX Dehumidifier (Case 6).

4. CONCLUSION

The following conclusions were made from the parametric study:

- Prediction of space relative humidity is very sensitive to modeling of natural infiltration and the method of mechanical ventilation employed. The relative humidity excursion analysis showed that all three homes (mid-1990's reference home, IECC 2006 home, and high-performance home) were prone to mold growth due to long excursions of high humidity.
- Thermal comfort analysis indicates that humidity problems within a home may not be noticed by the occupants.
- In the high-performance home (50% source energy savings level), adding active humidity control equipment (Cases 4-9) succeeds in effectively controlling relative humidity to a safe level, at a cost of increasing whole house source energy consumption by 6.7% to 15.5%.
- The annual HVAC electrical energy analysis showed that standard efficiency DX dehumidifiers (Cases 7 and 9) have minimal electrical energy consumption increase than high efficiency DX dehumidifiers (Cases 6 and 8).

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