Procedures for Calculating Residential Dehumidification Loads

Jon Winkler and Chuck Booten

National Renewable Energy Laboratory (NREL)

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June 2016

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Executive Summary

Residential building codes and voluntary labeling programs are continually increasing the energy efficiency requirements of residential buildings. Improving a building's thermal enclosure and installing energy-efficient appliances and lighting can result in significant reductions in sensible cooling loads, leading to smaller air conditioners and shorter peak cooling seasons. However, due to fresh air ventilation requirements and internal gains, latent cooling loads are not reduced by the same proportion. Thus, it's becoming more challenging for typical air-conditioning equipment to control indoor humidity at part-load cooling conditions, which poses the potential risk of high indoor humidity in moist climates.

Conventional air-conditioning equipment is typically controlled based on space sensible cooling loads. As a result, the amount of moisture removed by the air conditioner depends on the space sensible load and latent loads are not explicitly controlled. Thus, indoor humidity can reach high levels during periods with small sensible cooling loads that are common in today's newly constructed homes. However, cooling equipment is selected based on meeting peak sensible and latent cooling loads and part-load humidity control is typically not considered during the design and equipment selection process. As we progress toward more energy-efficient construction, whole-house humidity control during periods with part-load air conditioner operation will need to be considered when designing a home's space conditioning system.

The objective of this project was to investigate the impact the chosen design condition has on the calculated part-load cooling moisture load and to compare calculated moisture loads and the required dehumidification capacity to whole-building simulations. Procedures for sizing whole-house supplemental dehumidification equipment have yet to be formalized; however, minor modifications to current Air-Conditioner Contractors of America (ACCA) Manual J load calculation procedures are appropriate for calculating residential part-load cooling moisture loads. Though ASHRAE 1% DP design conditions are commonly used to determine the dehumidification requirements for commercial buildings, an appropriate DP design condition for residential buildings has not been investigated.

Two methods for sizing supplemental dehumidification equipment were developed and tested. The first method closely followed Manual J cooling load calculations whereas the second method made more conservative assumptions impacting both sensible and latent loads.

Parametric study results using EnergyPlus for three house efficiency levels in ten U.S. cities indicated residential dehumidification equipment can be appropriately selected by slightly modifying Manual J cooling load calculation procedures. Humid and dry climates were included in the analysis to test each method's ability to predict when supplemental dehumidification equipment would not be required.

Figure ES-1 plots the calculated rated supplemental whole-house dehumidifier capacity for three house designs with increasing efficiency using three dew-point (DP) temperature design conditions commonly used for sizing dehumidification equipment. The dehumidifier sizes were determined using a moisture load calculation procedure that closely follows Manual J cooling load calculations. The results indicate that using the 2% DP design condition to size supplemental dehumidification equipment in residential buildings is most appropriate and using the 1% DP design condition resulted in slightly oversized dehumidifiers in several of the cases. Using the 2% DP design condition to calculate the part-load cooling moisture load correctly identified which locations would not require supplemental dehumidification.

When using the 2% DP design condition to size supplemental dehumidification equipment, the results indicated:

- Indoor humidity was controlled to 55% RH for more than 94% of the year for all cases without being oversized
- Annual moisture loads in dry climates were small enough to be handled by the primary cooling equipment
- When using an RH set point of 55%, the indoor RH never exceeded the 60% level.

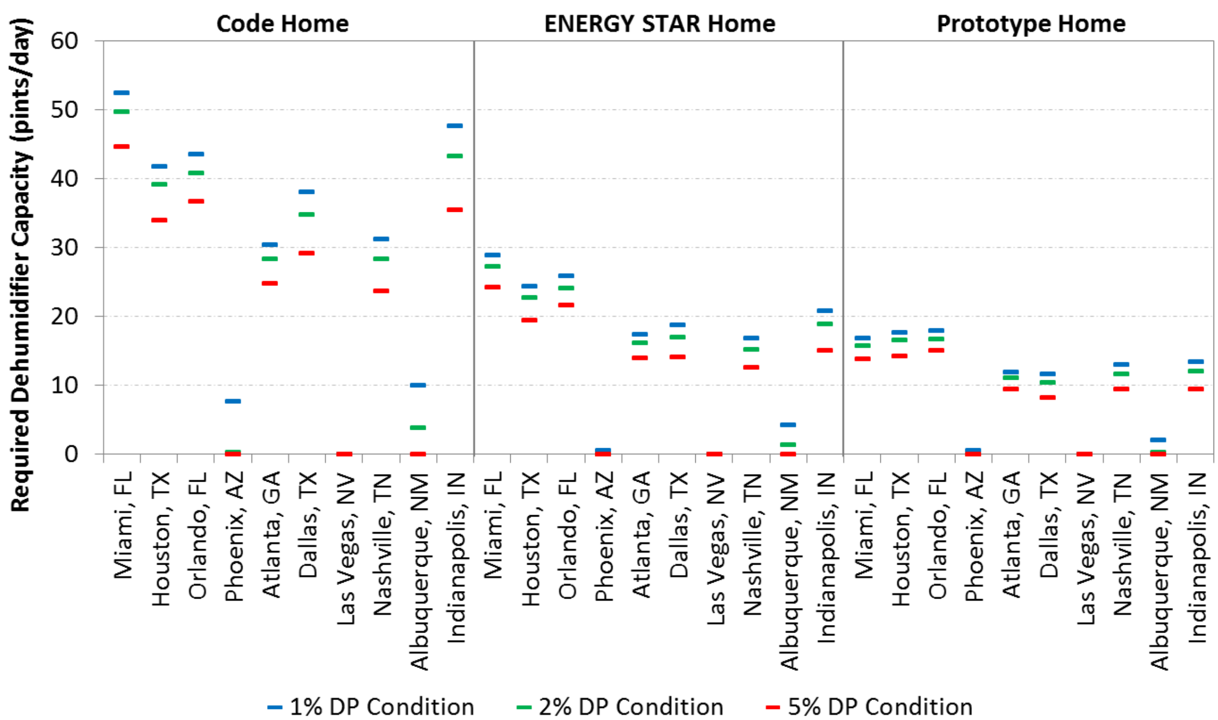

Figure ES-1. Required whole-house supplemental dehumidification capacity based on different DP design conditions for three new construction house efficiency levels

As part of this project, we developed a methodology to size residential supplemental dehumidification equipment which relies on ACCA Manual J cooling load calculation procedures. The key steps in sizing supplemental dehumidification include:

1. Calculation of peak cooling sensible and latent loads and sizing of primary cooling equipment (standard ACCA Manual J and S procedures using Building America House Simulation Protocols sensible and latent internal gains)
2. Calculation of part-load sensible and latent loads using the ASHRAE 2% DP design condition with a slightly modified Manual J load calculation procedure (see Section 2.2.1)
3. Prediction of primary cooling equipment moisture removal at the DP design condition
4. Calculation of unmet moisture load at part-load cooling conditions and sizing of required supplemental dehumidification equipment (see Section 2.2.2).

There are two key advantages of this approach that will ease adoption and improve the likelihood of implementation:

1. The supplemental dehumidification equipment sizing procedure closely aligns with current Manual J calculations
2. ASHRAE 2% DP design conditions are readily available for all the cities listed in Manual J.

A summary of the procedure for sizing supplemental dehumidification equipment is shown in Figure ES-2.

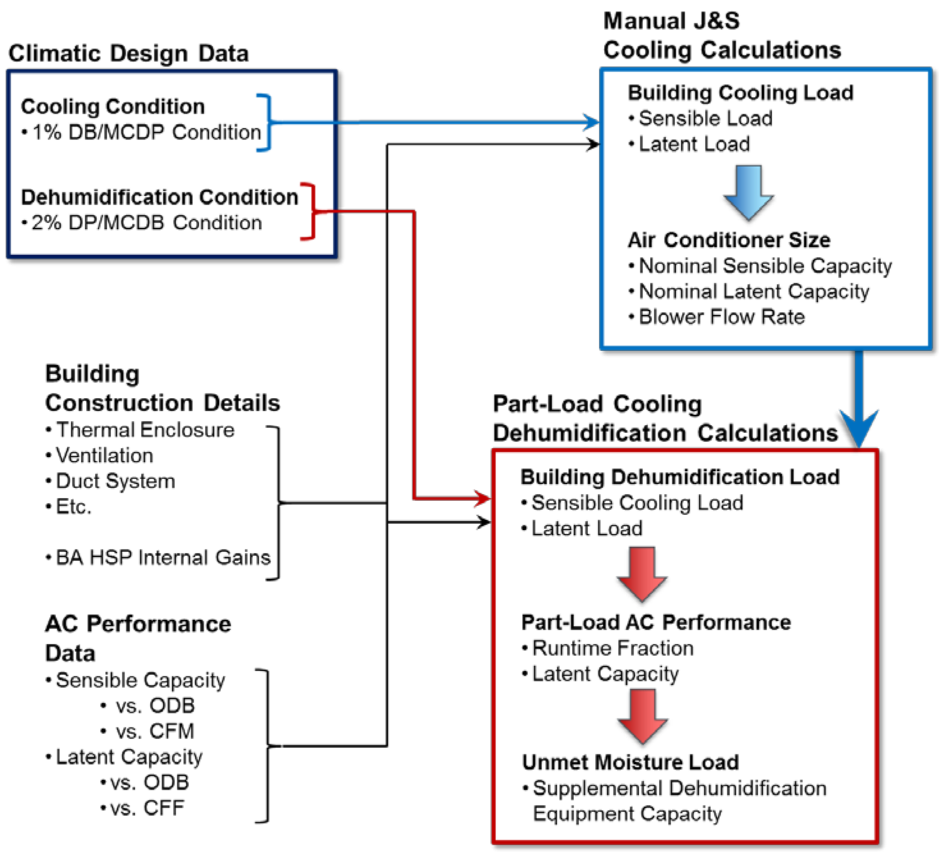

Figure ES-2. Overview of supplemental dehumidification equipment sizing procedure

Definitions

ACCA	Air-Conditioning Contractors of America
ACH_{50}	air changes per hour at 50 Pascals
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BEopt	Building Energy Optimization
CDF	cumulative distribution function
CLTD	cooling load temperature difference
DB	dry-bulb
DP	dew-point
DSE	distribution system effectiveness
EMPD	effective moisture penetration depth
EPT	expanded performance table
EPA	Environmental Protection Agency
EWB	entering wet-bulb temperature
BA HSP	Building America House Simulation Protocols
IECC	International Energy Conservation Code
ICC	International Code Council
MCDB	mean coincident dry-bulb
MCDP	mean coincident dew-point
MCWB	mean coincident wet-bulb
PLF	part-load factor
PLR	part-load ratio
RH	relative humidity
RTF	runtime fraction
SEER	seasonal energy efficiency ratio
SHGC	solar heating gain coefficient
SHR	sensible heat ratio
TMY	typical meteorological year
WB	wet-bulb
WMO	World Meteorological Organization
ZERH	Zero Energy Ready Home

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1 Overview

Residential building codes and voluntary labeling programs are continually increasing the energy-efficiency requirements of residential buildings. Improving a building's thermal enclosure and installing energy-efficient appliances and lighting can result in significant reductions in sensible cooling loads, thus leading to smaller air conditioners and shorter peak cooling seasons. However, due to fresh air ventilation requirements and internal gains, latent cooling loads are not reduced by the same proportion. Thus, it's becoming more challenging for typical air conditioning equipment to control indoor humidity at part-load cooling conditions, which poses the potential risk of high indoor humidity in moist climates.

Conventional air-conditioning equipment is typically controlled based on space sensible cooling loads. As a result, the amount of moisture removed by the air conditioner depends on the space sensible load and latent loads are often not explicitly controlled. Thus, indoor humidity can reach high levels during periods with small sensible cooling loads that are common in today's newly constructed homes. However, cooling equipment is selected based on meeting peak sensible and latent cooling loads and part-load humidity control is not considered during the design and equipment selection process. As we progress toward more energy-efficient construction, whole-house humidity control during periods with part-load sensible cooling loads will need to be considered when designing a home's space conditioning system.

1.1 Background

1.1.1 Design Conditions

Best-practice residential building heating and cooling load calculations follow procedures listed in Air-Conditioning Contractors of America (ACCA) Manual J (Rutkowski 2006) which states peak summer cooling loads should be based on 1% dry-bulb (DB) and mean coincident wet-bulb (MCWB) temperatures for a given location. Design conditions are typically attained from American Society of Heating, Ventilating, and Air-Conditioning Engineers (ASHRAE) data sources. The 1% cooling DB temperature condition corresponds to the 1% annual cumulative frequency of occurrence from historical hourly weather data typically spanning 25 years (ASHRAE 2013a). Thus, the 1% cooling DB temperature is exceeded by 88 hours per year on average and outdoor temperatures are typically cooler than the 1% DB temperature for 99% of the year. The MCWB temperature is the mean WB temperature coinciding with the design DB temperature and is determined by double-binning hourly historical weather data (ASHRAE 2013a).

Despite summer cooling design conditions typically being specified as DB and MCWB temperatures, latent loads are calculated using the difference between indoor and ambient humidity ratios. Thus, it can be convenient to express summer cooling design conditions using two independent properties of moist air such as DB and mean coincident humidity ratio or mean coincident dew-point (MCDP) temperature.

Design conditions based on DB temperature represent peak sensible cooling loads and are suitable for sizing cooling equipment. Similarly, design conditions based on DP temperature represent periods of peak latent loads due to weather and are useful when selecting humidity control equipment. According to ASHRAE (2013a), design conditions based on DP temperature are useful when analyzing cooling equipment performance at part-load conditions, which is important for a majority of residential cooling equipment because dehumidification is not the equipment's primary function. Periods of high outdoor DP temperature can occur at moderate DB temperatures, resulting in little air conditioner runtime and high indoor humidity.

Figure 1a plots the outdoor DB temperature cumulative distribution function (CDF) and the MCDP for Houston, Texas, taken from ASHRAE Weather Data Viewer v5.0 (ASHRAE 2013b). As indicated in the figure, the 1% DB temperature is 95°F and the MCDP (plotted on the right y-axis) is 69.3°F. The load sensible heat ratio (SHR) (red line) for a new construction home based on the 2009 International Energy Conservation Code (IECC) was calculated using procedures described in Manual J. Based on DB and MCDP conditions, the minimum load SHR of approximately 0.7 occurs at a DB temperature of 77.5°F, which is nearly 15% lower than the load SHR calculated at the 1% DB design condition.

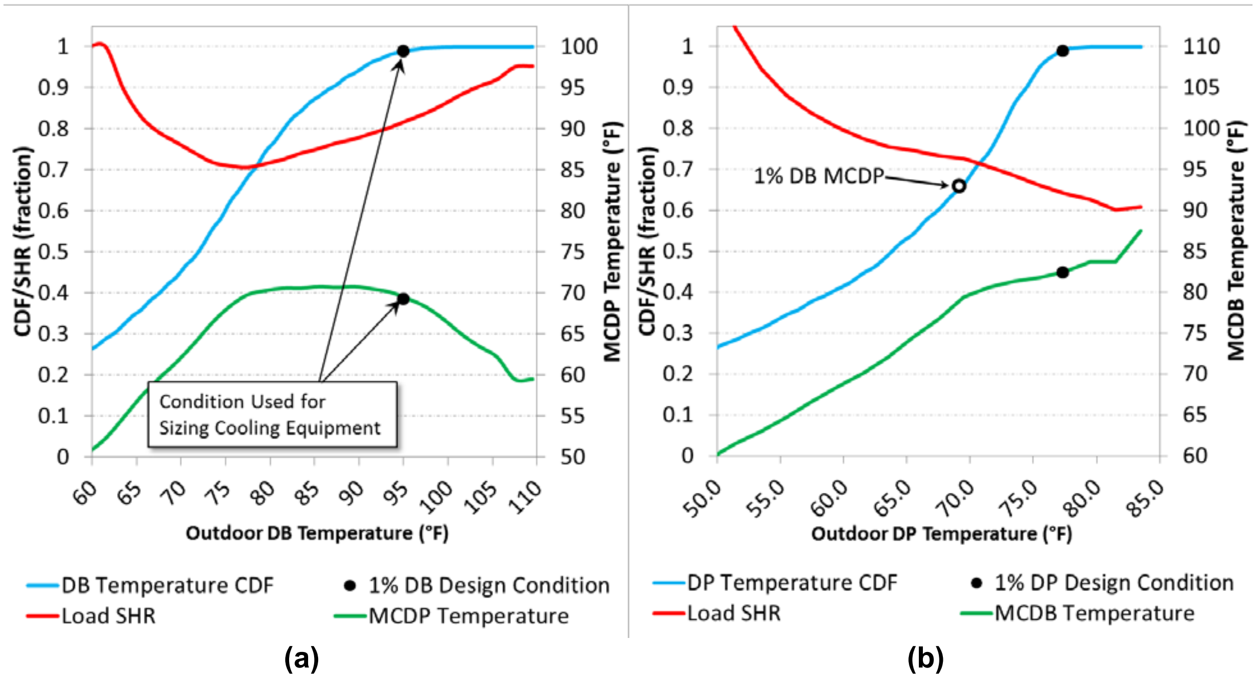

Figure 1. (a) Outdoor DB temperature CDF and MCDP and (b) outdoor DP temperature CDF and MCDB for Houston, Texas

Figure 1b plots the outdoor DP temperature CDF and the MCDP also for Houston, Texas. The 1% DP temperature is 77.3°F, which is significantly higher than the 1% DB MCDP temperature of 69.3°F. Additionally, the MCDB temperature corresponding to the 1% DP temperature is 82.4°F, which is significantly lower than the 1% DB temperature of 95°F. The load SHR at the 1% DP and MCDB temperatures is nearly 22% lower than the SHR at the 1% DB and MCDP temperatures. Since cooling equipment is selected based on the 1% DB temperature, cooling equipment will not run much at the 1% DP MCDB temperature, resulting in little moisture removal.

Figure 1b also shows that though cooling equipment is sized to satisfy loads at the 1% DB condition, latent cooling loads will be higher for approximately 30% of the year.

1.1.2 Typical Building Load Profile

Building sensible cooling loads are primarily driven by outdoor DB temperature, solar load, and internal sensible gains. Figure 2a plots the sensible and latent loads as a function of outdoor DB temperature for a 2,000-sq.-ft., 2009 IECC construction home in Houston, Texas. (Additional details on assumptions used to develop Figure 2 can be found in Section 2.) Assuming constant internal gains, the sensible load can vary at a given outdoor temperature based on the level of sky cover. Notice the latent load does not vary much compared to the sensible load (due to internal gains being a significant percentage of the total latent load and the MCDP remaining relatively constant over a range of outdoor DB temperatures) (shown in Figure 1a).

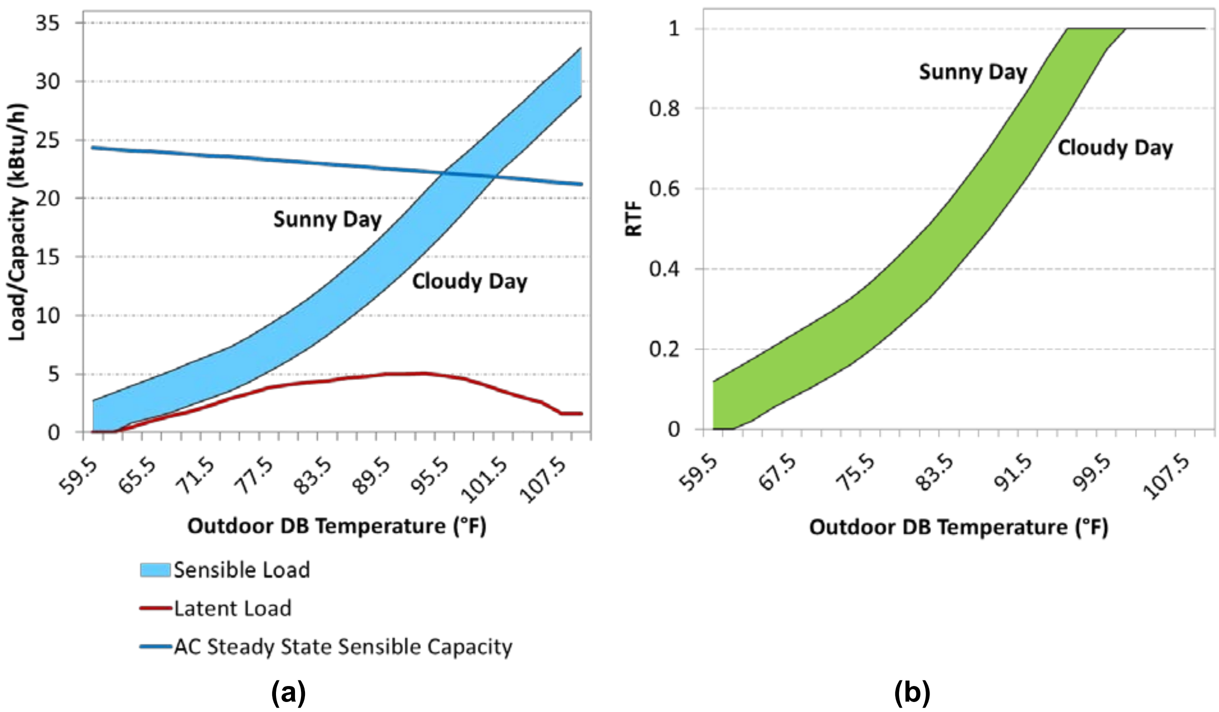

Figure 2. (a) A typical building load profile based on outdoor DB temperature and (b) estimated air conditioner RTF for a typical new construction home in Houston, Texas

Figure 2a also plots the air conditioner's steady-state sensible capacity as function of outdoor DB temperature. (Notice the steady-state sensible capacity intersects the sunny day sensible load line at 95°F, which is the 1% DB temperature for Houston, Texas.) The steady-state sensible capacity increases at cooler outdoor DB temperatures and at temperatures below 95°F the air conditioner must cycle on and off to prevent over-cooling the space.

Figure 2b plots the air conditioner runtime fraction (RTF) as a function of outdoor DB temperature for sunny and cloudy days. As expected on a sunny, 95°F day, the air conditioner must run 100% of the time to meet the sensible cooling load. Note that at the 1% DP MCDB temperature of 82.4°F (shown in Figure 1a) and under a cloudy sky, the air conditioner only needs to run approximately 35% of the time to meet the sensible cooling load, potentially leading to high indoor humidity.

1.2 Objective

The objective of this project was to investigate the impact the chosen design condition has on the calculated part-load cooling moisture load. Procedures for sizing whole-house supplemental dehumidification equipment have yet to be formalized; however, simple modifications to current Manual J load calculation procedures are appropriate for calculating residential part-load cooling moisture loads. Though ASHRAE 1% DP design conditions are commonly used to determine the dehumidification requirements for commercial buildings, an appropriate DP design condition for residential buildings has not been investigated.

Section 2 details the simulation-based approach used to meet this objective, which consists of implementing Manual J load calculation procedures and conducting a parametric study to assess the appropriateness of the dehumidification equipment sizing procedure. The simulation results are presented and discussed in Section 3.

2 Approach

This study investigated the impact DP design conditions have on the calculated part-load cooling moisture load in different residential buildings and how calculated moisture loads compare to whole-building simulation results. Section 2.1 discusses a parametric study that was conducted on different house efficiency levels and different climates. Two similar approaches adapted from ACCA Manual J to calculate residential building part-load cooling dehumidification needs are presented in Section 2.2. Both approaches are tested with different DP design conditions with three house types located in ten cities.

2.1 House Designs and Assumptions

Three thermal enclosures were analyzed:

1. 2009 IECC Code Home – A home using specifications in the Building America House Simulation Protocols (BA HSP) (Wilson et al. 2014), which is consistent with the 2009 IECC (ICC 2009)
2. ENERGY STAR® Home – A home using ENERGY STAR v3.1 prescriptive path requirements (EPA 2015)
3. Building America Prototype Home – A high-efficiency home optimized using the Building Energy Optimization (BEopt)™ software (NREL 2014).

Details on the three homes are presented in Section 2.1.2 and varied based on climate. The three house types were selected to span energy savings levels in new construction homes. Using the DOE Zero Energy Ready Home (ZERH) specification was also considered, but the thermal enclosure specifications did not significantly vary from the ENERGY STAR home. In terms of energy savings, the ENERGY STAR home was a closer midpoint between the code home and Building America prototype home.

2.1.1 Climate Zones

The three homes were modeled in the cities listed in Table 1 with the IECC climate zones shown in Figure 3. Load calculations were conducted using data from the ASHRAE Weather Data Viewer v5.0 (ASHRAE 2013b) for the corresponding World Meteorological Organization (WMO) station number. Simulations were performed using the corresponding Typical Meteorological Year 3 (TMY3) weather data. Part-load cooling moisture loads were the focus of the study. However, dry climates were included to investigate the latent load calculations in areas where total latent loads are known to be small or non-existent. Multiple cities within climate zones with high latent loads were included for comprehensiveness.

Table 1. Cities Included in the Analysis

City	IECC Climate Zone	WMO Station ID	1% DB Temperature (°F)	1% DB MCDP Temperature (°F)
Miami, FL	1	722020	90.7	72.6
Houston, TX	2A	722430	95.0	69.3
Orlando, FL	2A	722050	92.5	69.8
Phoenix, AZ	2B	722780	108.1	47.1
Atlanta, GA	3A	722190	91.4	66.6
Dallas, TX	3A	722590	98.4	64.4
Las Vegas, NV	3B	723860	106.2	43.4
Nashville, TN	4A	723650	92.1	67.8
Albuquerque, NM	4B	723270	92.8	37.4
Indianapolis, IN	5A	724380	88.7	68.3

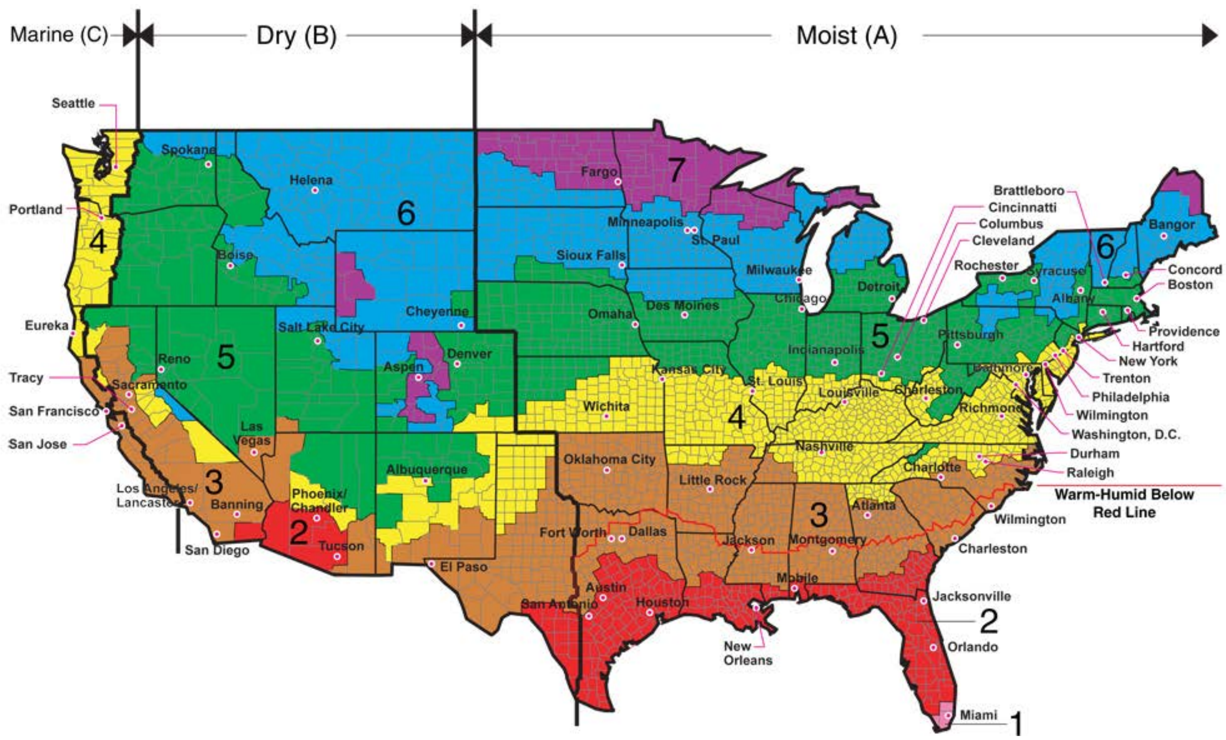

Figure 3.IECC climate zones

2.1.2 Building Construction Details

A three-bedroom, two-bathroom, 2,000-sq.-ft., slab-on-grade house was used for the analysis. Slab-on-grade construction was selected since it is the most common type of foundation in warm climates and was used in all climate zones for consistency. The house was assumed to have an unfinished attic with insulation on the attic floor, a north orientation, and a window-to-wall area ratio of 15% with the windows being evenly distributed based on the corresponding wall exterior surface area. Ducts were assumed to be located in the attic in the code home but in conditioned space for the ENERGY STAR and Building America prototype homes. All three homes were assumed to have ASHRAE 62.2 2010 continuous whole-house ventilation rate of 50 cfm (ASHRAE 2010).

Construction details for the three homes are listed in Table 2 based on IECC climate zone. The Building America prototype home was determined by running a BEopt optimization for each IECC climate zone. Though the construction of this home may be beyond current practice, it was intended to exemplify residential building energy efficiency and serve as a practical upper bound on energy efficiency. Duct specifications for the code home are based on the BA HSP (Wilson et al. 2014).

Table 2. House Construction Details

House Type	IECC Climate Zone	Ceiling R-Value	Frame Wall R-Value	Slab R-Value, Depth	Air Leakage (ACH_{50})	Window U-Value, SHGC	Duct R-Value, % Leakage
Code	1-3	30	13	0	7	0.37, 0.30	8, 15%
	4	38	13	10, 2 ft	7	0.35, 0.44	8, 15%
	5	38	13+5[a]	10, 2 ft	7	0.35, 0.44	8, 15%
ENERGY STAR	1	30	13	0	4	0.40, 0.25	N/A[b]
	2	38	13	0	4	0.40, 0.25	N/A
	3	38	13+5	0	3	0.30, 0.25	N/A
	4	49	13+5	10, 2 ft	3	0.30, 0.40	N/A
	5	49	13+5	10, 2 ft	3	0.27, 0.40	N/A
BA Prototype	1-2	49	19+10	10, 2 ft	1	0.29, 0.26	N/A
	3	49	19+10	10, 2 ft	1	0.29, 0.31	N/A
	4-5	60	19+10	15, 4 ft	1	0.21, 0.40	N/A

[a] "13+5" means R-13 cavity insulation combined with R-5 continuous insulating sheathing on the exterior of the wall.

[b] Duct R-value and percent leakage are not applicable since ducts were located in conditioned space.

2.1.3 Internal Gains

Hourly sensible and latent internal gains were modeled based on the BA HSP. The average daily internal moisture generation rate was 11.2 lbm/day for the code house and 11.0 lbm/day for both the ENERGY STAR and Building America Prototype homes. Figure 4 plots the hourly internal gain profiles for the code home. The three blue lines (plotted on the left y-axis) represent the hourly total, sensible, and latent internal gains, respectively. The red line (plotted on the right y-axis) is the moisture generation rate corresponding to the internal latent gain.

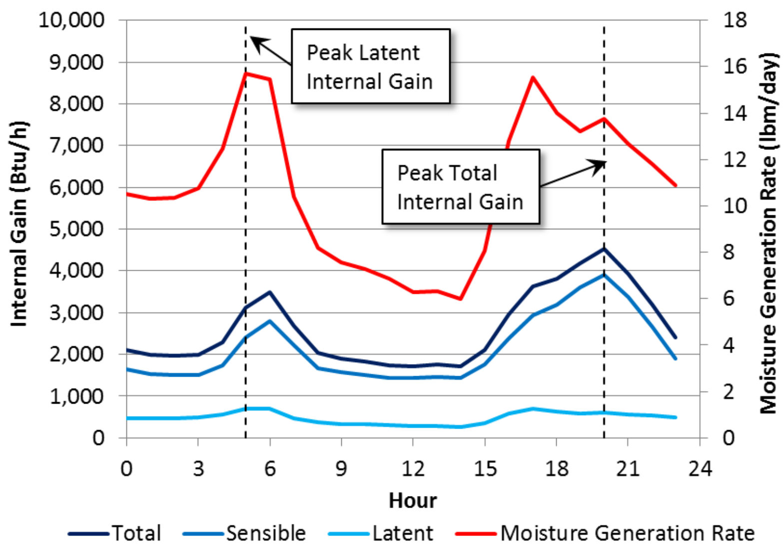

Figure 4. Hourly internal gain profiles for the code home

The average daily sensible internal gain was 23 kWh/day for the code home and 21.2 kWh for both the ENERGY STAR and Building America Prototype homes. Both the ENERGY STAR and Building America prototype homes included energy-efficient appliances, resulting in slightly less internal gains.

Identical BA HSP internal gain profiles were used in the building load calculations and the whole-building simulation. Internal gains impact both the cooling and supplemental dehumidification equipment capacity. However, investigating the impact internal gains have on the necessary equipment capacity was beyond the scope of this study. Since the internal gains used in the load calculations matched the internal gains in the building simulation, results from the two should be comparable.

2.1.4 Equipment

Each house was simulated with a single-stage air conditioner with a seasonal energy efficiency ratio (SEER) of 13. Though high-efficiency homes in warm climates will have higher-efficiency equipment, the focus of the study was not on cooling energy or air conditioner latent performance. The purpose of the study was to compare calculated moisture loads and the required dehumidification capacity using modified Manual J procedures to whole-building simulations independent of the selected equipment. Thus, a single-stage air conditioner and whole-house dehumidifier were selected as the space-conditioning equipment for simplicity. Results are unlikely to change significantly with a two-stage or variable-speed air conditioner since that equipment has not been shown to significantly impact indoor humidity (Rudd et al. 2013). The air conditioner was simulated using inputs presented by Cutler et al. (2013) and the whole-house dehumidifier was simulated using performance curves developed by Winkler et al. (2012).

The air conditioner in each home was sized using ACCA Manual J and S procedures with the BA HSP internal gains described in Section 2.1.3; it was assumed that the properly sized air conditioner would be available on the market (Rutkowski 2006; Rutkowski 1995). Table 3 lists the rated total capacity for all homes studied. The rated SHR of the air conditioner was assumed to be 0.74.

Table 3. Simulated Air Conditioner Total Rated Capacity (tons)

City	House Type		
	Code	ENERGY STAR	BA Prototype
Miami, FL	2.6	1.6	1.2
Houston, TX	2.7	1.7	1.3
Orlando, FL	2.5	1.6	1.2
Phoenix, AZ	3.4	2.2	1.6
Atlanta, GA	2.3	1.3	1.2
Dallas, TX	2.9	1.5	1.4
Las Vegas, NV	3.2	1.7	1.6
Nashville, TN	2.5	1.4	1.3
Albuquerque, NM	2.1	1.3	1.2
Indianapolis, IN	2.2	1.3	1.2

2.2 Dehumidification Load Calculation Procedure

Current residential load calculation and equipment selection procedures can be easily modified to account for part-load cooling moisture loads. Since Manual J does not explicitly prescribe a part-load cooling load calculation procedure, assumptions were made to determine the primary cooling equipment moisture removal capacity, which can have a significant impact on the calculations. It is important that the air-conditioner moisture removal at part-load conditions be considered to prevent oversizing the supplemental dehumidification equipment.

Two similar sets of assumptions are described below to estimate dehumidification loads and the supplemental dehumidification equipment capacity; one closely follows the Manual J cooling load procedure and the other makes more conservative assumptions to calculate the sensible load more closely aligning with the Manual J heating load procedure. The procedure starts with selecting the primary cooling equipment based on the 1% DB condition as prescribed by Manuals J and S. Only after the cooling equipment has been sized can the part-load cooling dehumidification requirements be determined. Part-load cooling dehumidification requirements are dependent on the selected part-load design condition. Sensitivity of the chosen DP design condition on the required moisture removal capacity will be presented in Section 3.

Determining the part-load cooling moisture removal requirements involves three main steps:

1. Calculate the sensible and latent cooling loads at the part-load design condition
2. Estimate the latent capacity and total moisture removal of the primary cooling equipment at the part-load condition
3. Determine the unmet moisture load and required supplemental dehumidification equipment capacity.

2.2.1 Part-Load Moisture Load Calculation

Two methods largely based on ACCA Manual J cooling load calculation procedures were used to estimate the part-load cooling moisture load. Table 4 lists the key differences between the two methods for portions of the load calculation procedure that differ from the standard Manual J peak cooling load calculations. Both methods assumed cloudy conditions when calculating window solar loads and used the ASHRAE distribution system effectiveness (DSE) formulas from ASHRAE Standard 152 to estimate the duct losses for the code home (ASHRAE 2004). (Both other homes had ducts located in conditioned space and the duct losses were neglected.)

Table 4. Part-Load Moisture Load Calculation Method Assumptions

	Method 1	Method 2
Wall Conduction	UA·CLTD method based on DP MCDB temperature[a]	UA·ΔT method using DP MCDB temperature[b]
Buffer Space Temperatures	Summer calculation based on DP MCDB temperature	Winter calculation based on DP MCDB temperature
Window Solar Loads	Cloudy Conditions	Cloudy Conditions
Internal Gains[c]	Peak Hourly Total (see Figure 4)	Peak Hourly Latent (see Figure 4)
Duct Losses	RTF adjusted ASHRAE DSE calculation	RTF adjusted ASHRAE DSE calculation
Indoor Design Conditions (DB; RH)	75°F; 55%	75°F; 55%

[a] CLTD method based on Manual J procedure.
[b] Calculated using wall UA and difference between the outdoor and indoor DB temperatures.
[c] Internal gains were based on the BA HSP procedures (Wilson et al. 2014) and are described in Section 2.1.3.

The key difference between the two methods is the approach used to calculate sensible conduction loads. Method 1 uses the cooling load temperature difference (CLTD) calculation method to calculate the opaque panel cooling load which accounts for the panel solar load and thermal mass. (The CLTD calculation method is deemed as a reasonable compromise between accuracy and simplicity for the purpose of sizing cooling equipment.) Manual J prescribes a procedure to adjust the CLTD based on the outdoor DB temperature for non-summer conditions. Method 2 uses a UA·ΔT approach, similar to heating load conduction calculations, where the ΔT is the temperature difference across the opaque panel. The Method 1 approach will result in higher opaque panel sensible loads and coincidently, more air conditioner runtime/moisture removal and smaller supplemental dehumidification capacity. Additionally, the Method 1 approach to estimate the buffer space (attic, garage, etc.) temperatures will result in larger temperature differences than Method 2, also resulting in higher sensible loads. Method 1 more closely aligns with the Manual J summer cooling load calculation.

Manual J allows for an indoor relative humidity (RH) of 55% for calculating latent loads in humid climates. Since the focus is on calculating part-load cooling moisture loads, which is a more important design step in humid climates, 55% was used.

Infiltration air flow rates were calculated using the Component Leakage Area Method included in Manual J where the assumed ACH50 value was converted into an aggregate 4-Pascal leakage area (ELA_4) value using equations in Chapter 16 of ASHRAE 2013a. Stack and wind coefficients were selected from Table 5D of Manual J for a 2-story building and a shielding class of 4 for a typical suburban location. Mechanical ventilation rates, calculated based on ASHREA 62.2 (ASHRAE 2010), were added in quadrature to the calculated infiltration rate to determine the total ventilation rate (ASHRAE 2013a), which was used to calculate the sensible and latent ventilation loads at the given design condition.

2.2.2 Estimating Cooling Coil Part-Load Moisture Removal

After calculating sensible and latent loads ($\dot{Q}_{load,sens}$, $\dot{Q}_{load,lat}$) at the part-load design condition, the cooling coil RTF should be calculated to estimate the moisture removal. The cooling coil steady-state sensible and latent capacity at the part-load condition can be determined using manufacturer expanded performance table (EPT) data. For this study, the steady-state sensible and latent capacity were determined using the total cooling capacity performance curve provided by Cutler et al. (2013) and the coil bypass factor/apparatus DP method for calculating the coil SHR (DOE 2013).

The coil RTF can be estimated using the steady-state sensible capacity ($\dot{Q}_{cap,sens}$) at the part-load condition. The part-load ratio (PLR) is defined as

$$PLR = \frac{\dot{Q}_{load,sens}}{\dot{Q}_{cap,sens}} \tag{1}$$

The coil RTF can be estimated using the PLR and part-load fraction (PLF)

$$RTF = \frac{PLR}{PLF} \tag{2}$$

where

$$PLF = 1 - C_D(1 - PLR) \tag{3}$$

where C_D is the part-load degradation coefficient. Equation 3 is often used when estimating part-load efficiency of air conditioners and heat pumps in building simulation tools (Henderson et al. 1999; Cutler et al. 2013) and has compared well to theoretical models (Henderson et al. 1999). For the SEER 13 equipment modeled in this study, a C_D of 0.07 was assumed.

Latent degradation due to air-conditioner cycling was not considered and is known to be less significant when the blower cycles with the compressor. Thus, the steady-state latent capacity ($\dot{Q}_{cap,lat}$) was used with the coil RTF fraction to determine the unmet moisture load ($\dot{Q}_{moist,unmet}$) at the part-load design condition.

$$\dot{Q}_{moist,unmet} = \dot{Q}_{laod,lat} - RTF \cdot \dot{Q}_{cap,lat} \tag{4}$$

The calculated unmet moisture load was then used to determine the required capacity of the supplemental dehumidification equipment.

2.2.3 Procedure Overview

Figure 5 provides an overview of the procedure used to calculate part-load cooling moisture loads and how the supplemental dehumidification equipment selection procedure was tested using a whole-building simulation tool. Climatic design data, along with building and air conditioner performance data, are first used to size the primary cooling equipment based on Manual J and S procedures (blue box). Next, various DP design conditions were applied to a modified Manual J procedure, and after estimating the air conditioner RTF and moisture removal, the supplemental dehumidification equipment capacity was calculated (red box).

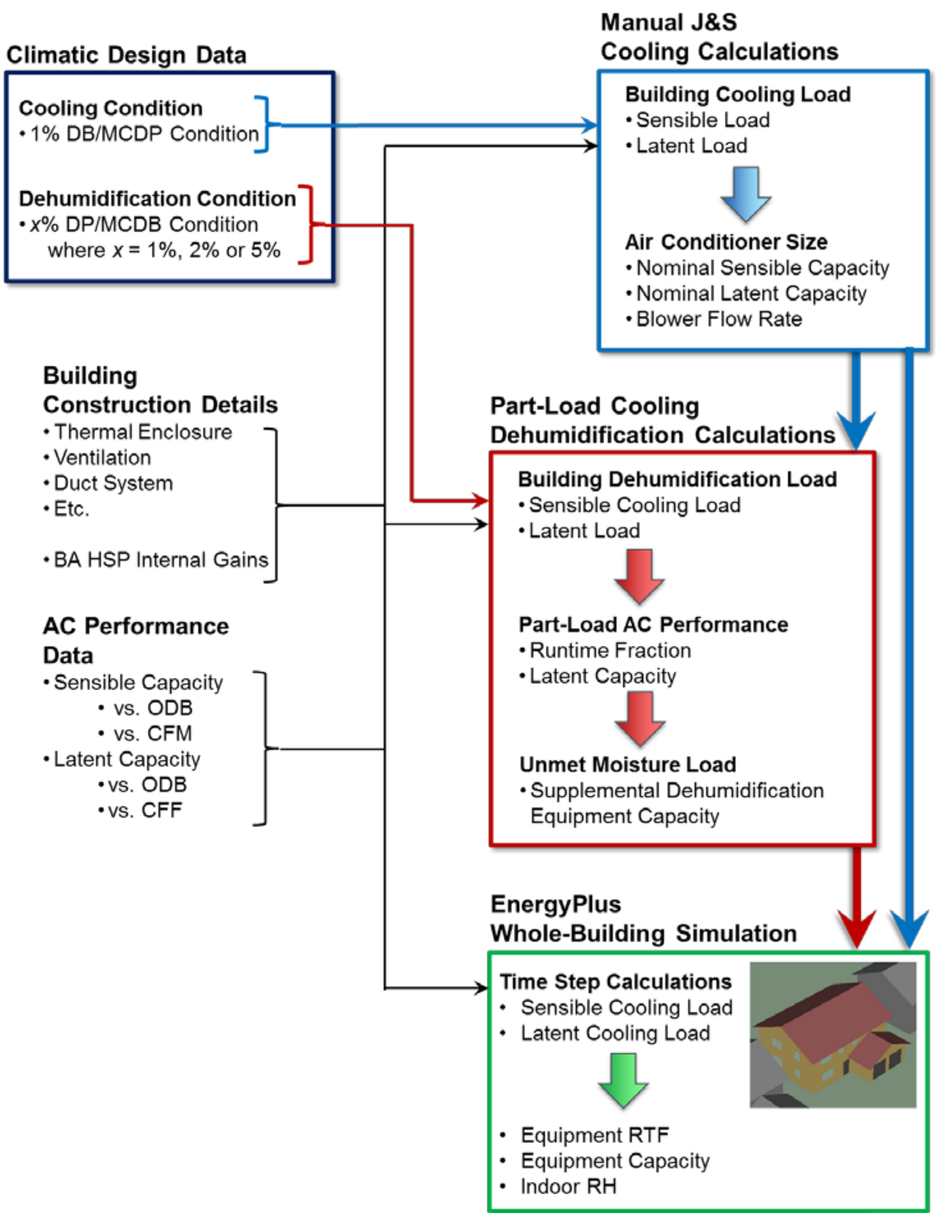

Figure 5. Overview of the part-load cooling moisture load calculation and testing procedure

The accuracy of the selected supplemental dehumidification equipment capacity was tested using a whole-house dehumidifier in the EnergyPlus whole-building simulation tool (green box). The

EnergyPlus building model was simulated using BEopt and BA HSP procedures based on the inputs listed in Table 2. The air conditioner was simulated using inputs presented by Cutler et al. (2013) and the whole-house dehumidifier was simulated using performance curves developed by Winkler et al. (2012). An effective capacitance multiplier of 15 was used to account for the moisture buffering of building materials (Woods et al. 2013). Woods et al. (2013) concluded that the Effective Capacitance model has several shortcomings compared to the Effective Moisture Penetration Depth (EMPD) model. However, a generalized set of EMPD model inputs have yet to be developed for residential buildings, limiting the model's usefulness. Given the current analysis is investigating the dehumidifier's ability to meet the building moisture load, the effective capacitance model was deemed sufficient for the current study.

3 Results

The dehumidification load procedure presented in Section 2.2 was applied using 1%, 2%, and 5% DP design conditions. For each house type, located in all ten cities, the unmet moisture load was calculated and used to estimate the capacity of a whole-house dehumidifier necessary to meet the load.

3.1 Dew-Point Design Conditions for Moisture Load Calculations

Table 5 lists the 1%, 2%, and 5% DP design conditions for each city included in the analysis. As expected, the DP temperatures are significantly higher than the 1% DB condition MCDP temperatures listed in Table 1. In general, there is not a significant difference between the 1% and 2% DP temperatures in the humid cities. However, in the dry cities, the difference between the 1% and 2% DP temperatures can be several degrees.

Table 5. 1%, 2%, and 5% DP Design Conditions

City	1% DP Condition		2% DP Condition		5% DP Condition	
	DP (°F)	MCDB (°F)	DP (°F)	MCDB (°F)	DP (°F)	MCDB (°F)
Miami, FL	77.6	83.3	77.1	83.2	76.1	82.8
Houston, TX	77.3	82.4	76.7	82.2	75.5	81.8
Orlando, FL	76.9	81.4	76.2	80.9	75.2	80.3
Phoenix, AZ	69.4	84.5	67.6	86.6	64.1	90.0
Atlanta, GA	73.3	80.2	72.6	79.6	71.3	78.6
Dallas, TX	74.4	82.8	73.5	82.1	72.3	81.9
Las Vegas, NV	63.2	84.6	60.4	87.5	54.6	88.0
Nashville, TN	74.0	81.8	73.1	81.0	71.6	79.9
Albuquerque, NM	60.4	68.6	58.8	69.5	55.8	71.5
Indianapolis, IN	74.0	82.0	72.7	80.4	70.3	77.6

As mentioned in Section 2.2.1, the indoor design conditions for load calculations were assumed to be 75°F DB and 55% RH, which is equivalent to a DP temperature of 57.8°F (at sea level). Despite being considered dry climates, Phoenix, Las Vegas, and Albuquerque all have positive weather-related latent loads at both the 1% and 2% DP condition. However, the latent loads in the drier cities tended to be small enough to be met by the air conditioner, even at part-load operation. Thus, as the results will show, these cities tend not to need supplemental dehumidification equipment to control moisture loads.

3.2 Latent Loads

Figure 6 plots the latent load breakdown for the three homes by depicting the latent load associated with internal gains, return duct leakage, and ventilation. Mechanical ventilation and infiltration latent loads are challenging to represent in a bar plot given mechanical ventilation and infiltration air flow rates are summed in quadrature, as explained in Section 2.2.1. Thus, the total ventilation load due mechanical ventilation and infiltration is represented by the solid and hashed blue region (Figure 6) with the mechanical ventilation portion calculated using the ASHRAE 62.2 ventilation rate and the infiltration load being the remainder. Given all three homes were mechanically ventilated based on ASHREA 62.2 rates, all three homes in a given city had equal mechanical ventilation latent loads. Ducts were assumed to be in conditioned space for the ENERGY STAR and Building America prototype homes and therefore, these homes did not have a latent load due to return duct leakage.

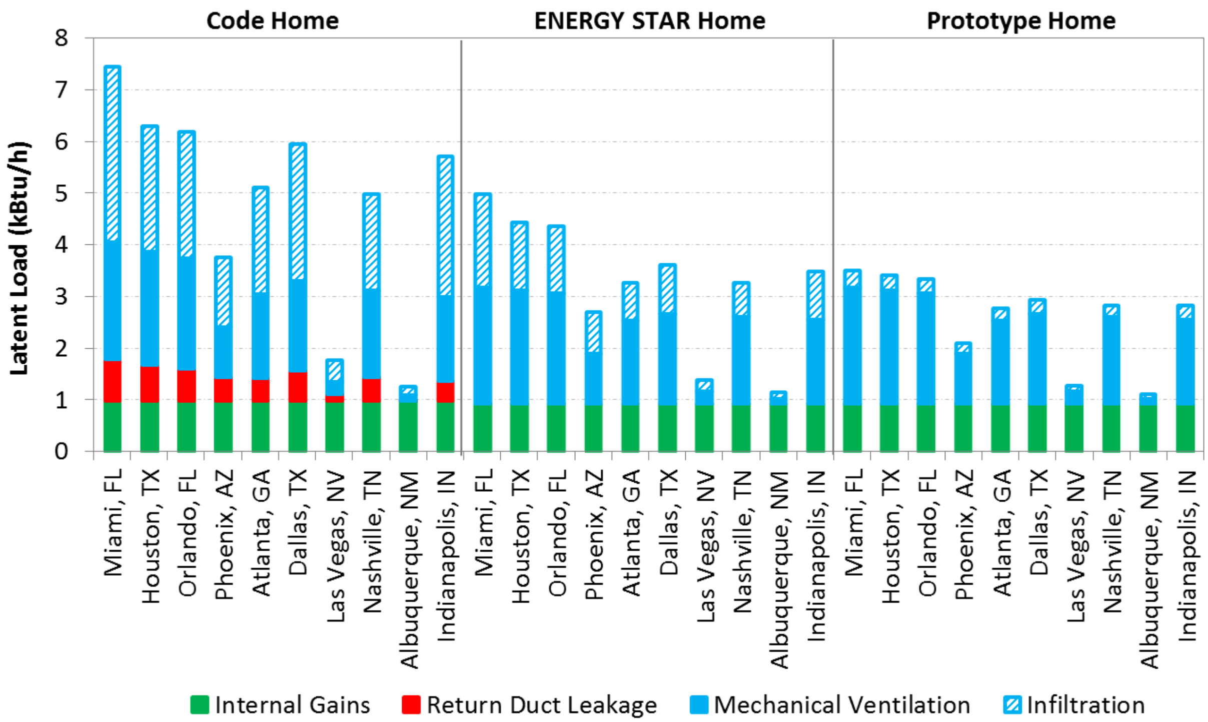

Figure 6. Total latent load breakdown at the 2% DP design condition

3.3 Unmet Moisture Loads – Method 1

Figure 7 plots the unmet moisture load at 1%, 2%, and 5% DP conditions calculated using Method 1 described in Section 2.2.1. As expected, the unmet moisture load is always highest at the 1% DP condition. Although Phoenix and Albuquerque are in dry climates, the air conditioner cannot meet the moisture load at the 1% DP condition; however, the unmet moisture loads are small enough to not be a concern. As the house envelope improves, the unmet moisture load is less dependent on the ambient condition and more dependent on the internal gains.

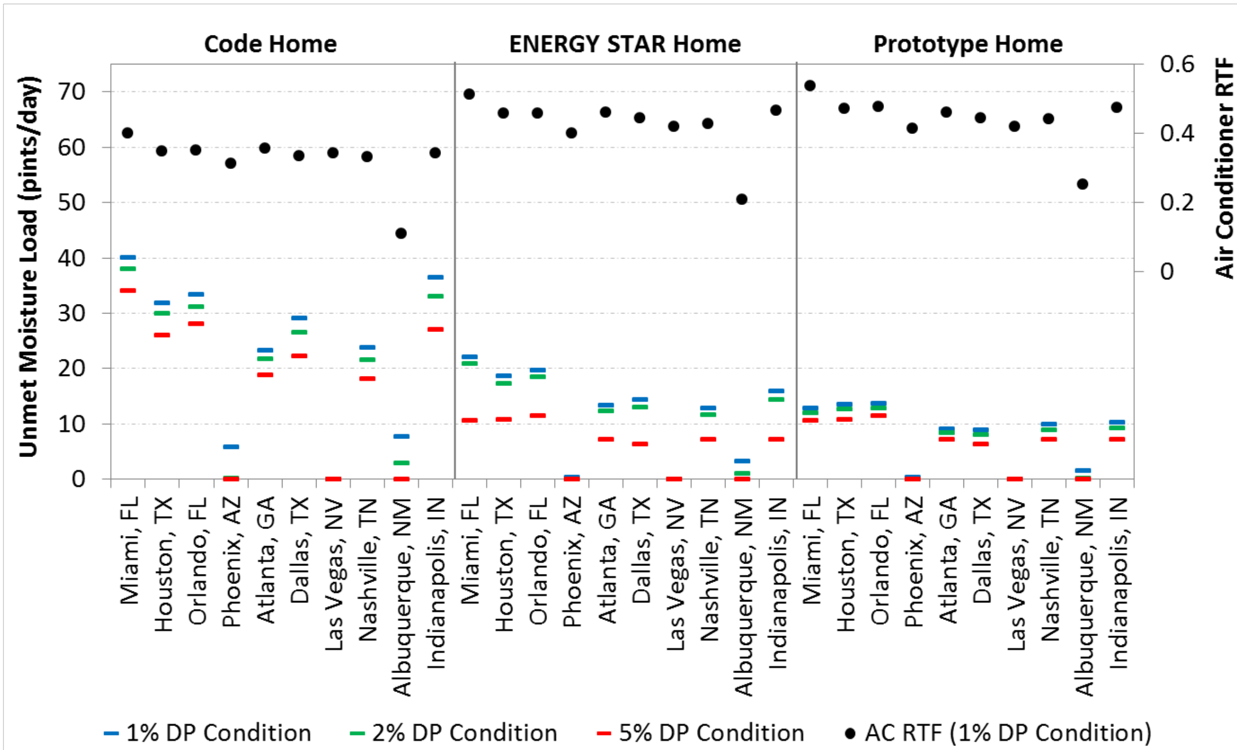

Figure 7. Unmet moisture loads at 1%, 2%, and 5% DP conditions and air conditioner RTF (using Method 1)

In Figure 7, the air conditioner RTF at the 1% DP condition is plotted on the right y-axis, which was calculated using Equations 1 through 3. The RTF is fairly consistent across a majority of climates but the RTF increases with house efficiency level. Air conditioner runtime is thought to decrease in energy-efficient houses; however, the results in Figure 7 indicate higher air conditioner runtimes in the more efficient homes. This trend is explained in the next section.

Based on the DP design conditions listed in Table 5, the unmet moisture in the Indianapolis code home (Figure 7) was higher than expected. The unmet moisture load aligns closer with expectations for the other two house types. Other than in Miami, the Indianapolis code home had the least amount of excess air conditioner latent cooling capacity, increasing the unmet moisture load. Indianapolis also had the second-lowest load SHR at the 1% DB design condition.

3.3.1 House Efficiency Impact on Air Conditioner Runtime Fraction

Figure 8 compares the building load profile as a function of outdoor DB temperature for the code house and ENERGY STAR house in Houston, Texas. The slope of the sensible load region explains higher air conditioner RTFs in the more efficient homes. In both houses, the air conditioner is sized to meet the sensible cooling load at the 1% DB condition of 95°F. Due to higher insulation levels and tighter construction, the ENERGY STAR house is less sensitive to outdoor DB temperature. Since the air conditioner RTF is approximately equal to the ratio of sensible load to air conditioner sensible capacity, the RTF fraction in the ENERGY STAR house will be higher at cooler temperatures. The difference in sensible load due to solar loads is smaller in the ENERGY STAR home, which is attributed to lower solar heat gain coefficient (SHGC) windows. Additionally, the latent load is only slightly smaller in the ENERGY STAR house despite the tighter construction since a majority of the latent load is due to internal gains and mechanical ventilation.

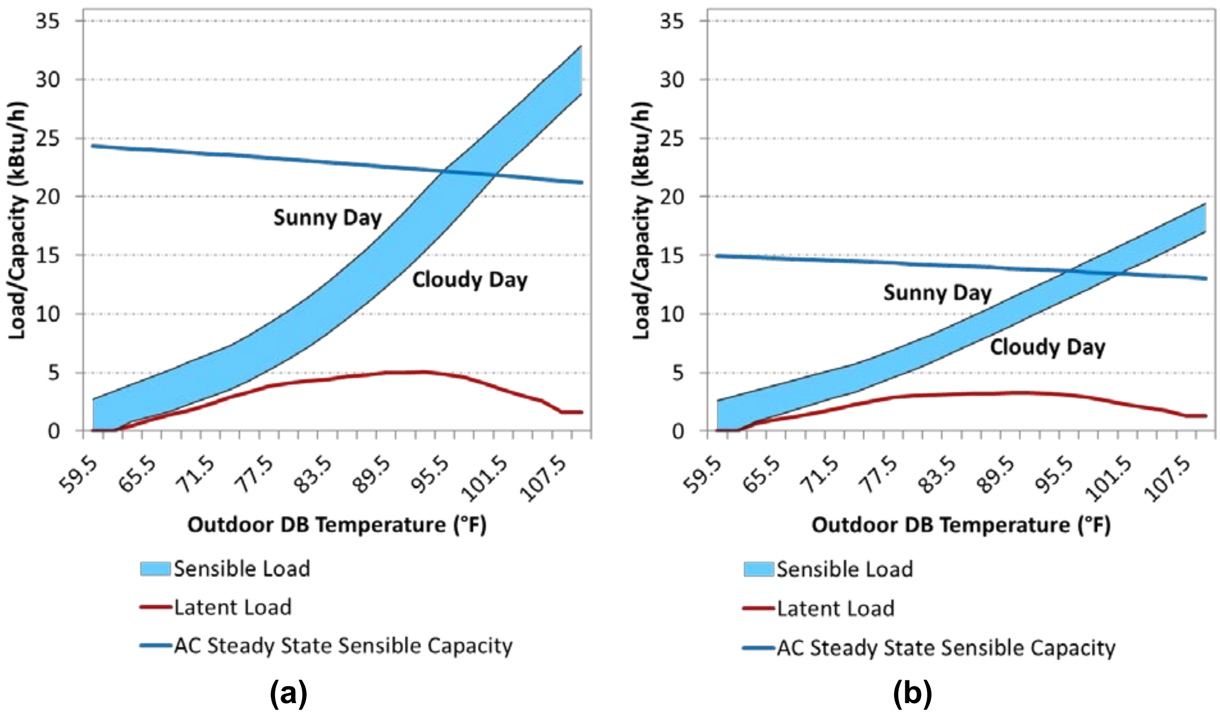

Figure 8. Building load profile based on outdoor DB temperature for the (a) code home and (b) ENERGY STAR home in Houston, Texas

As shown, less sensitivity in the ENERGY STAR home to outdoor temperature explains high air conditioner RTFs.

Thus in general, if an air conditioner is appropriately sized based on the design cooling load, it will run for more hours throughout the year in an energy-efficient home compared to a code home.

3.3.2 Air Conditioner Moisture Removal

Figure 9 plots the percentage of the Method 1 moisture load at 1%, 2%, and 5% DP conditions that is removed by the air conditioner. As expected, the air conditioner handles a larger percentage of the moisture load in the more efficient homes since the air conditioner RTF is higher and the loads are less dependent on the ambient conditions. Figure 9 highlights the importance of accounting for the air conditioner moisture removal when sizing supplemental dehumidification equipment. Despite operating at part-load at the DP design conditions (see RTFs plotted in Figure 7), the air conditioner handles approximately 50-65% of the moisture load in humid climates. If unaccounted for when selecting a dehumidifier, the dehumidifier could be oversized by 50-100%.

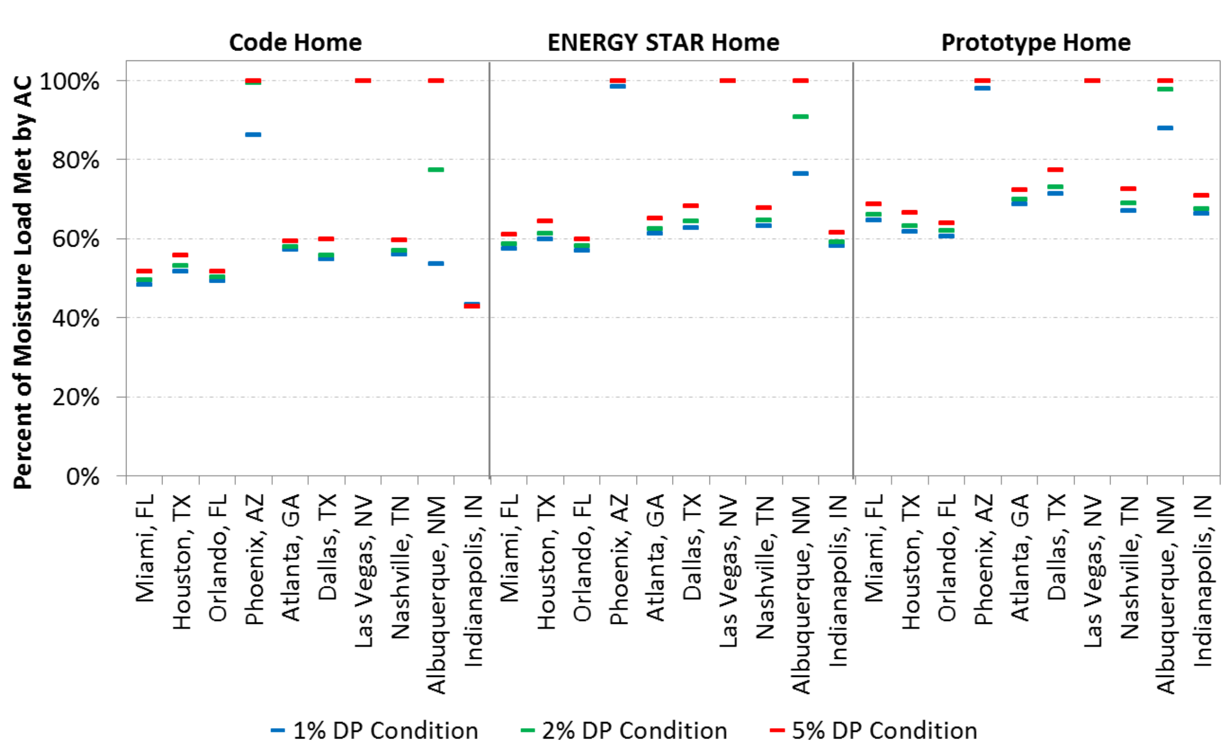

Figure 9. Percentage of the Method 1 moisture load removed by the air conditioner at 1%, 2%, and 5% DP conditions

3.4 Unmet Moisture Loads – Method 2

Figure 10 plots the unmet moisture load at 1%, 2%, and 5% DP conditions calculated using Method 2 described in Section 2.2.1. As expected, the unmet moisture load is higher than Method 1 values (previously shown in Figure 7) for all cases, which is due to lower air conditioner RTFs and higher assumed internal latent gains. The high unmet moisture loads in the dry climate cities indicates Method 2 is likely too conservative to calculate part-load cooling moisture loads. Additionally, given Method 1 closely aligns with the current Manual J cooling load calculation procedure, Method 1 is the preferred approach to calculating part-load cooling moisture loads.

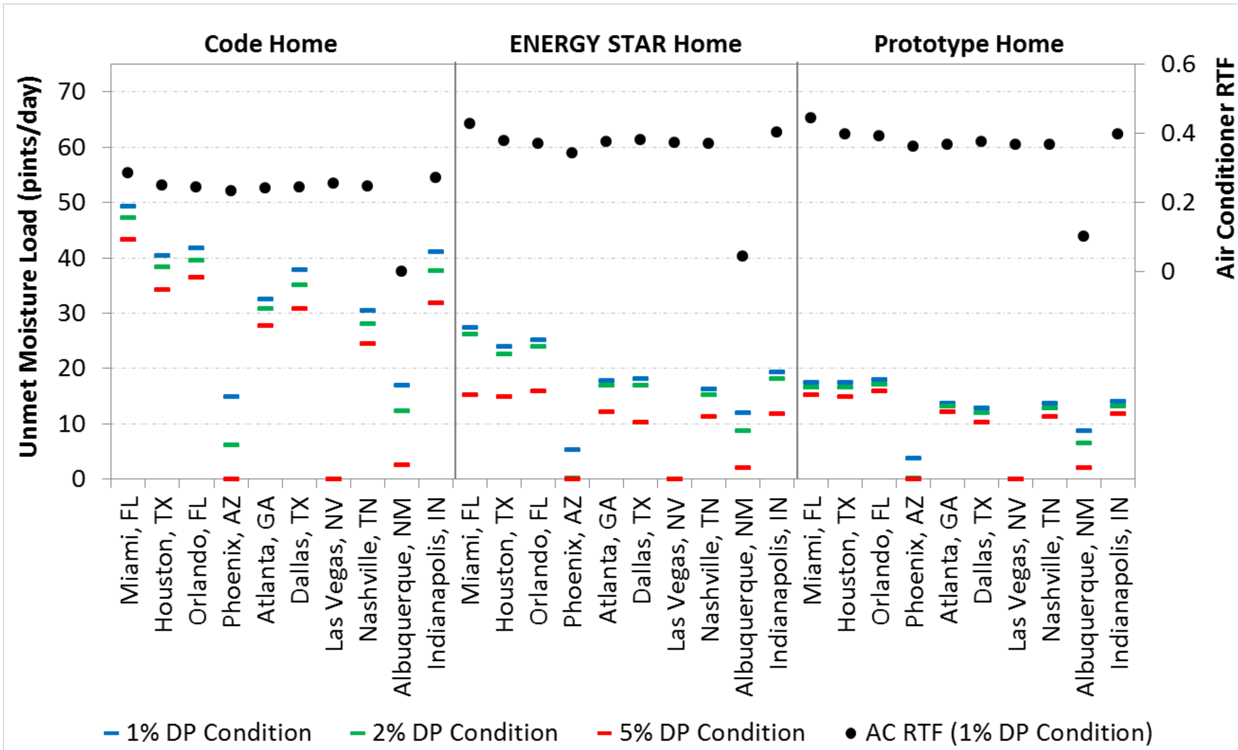

Figure 10. Unmet moisture loads at 1%, 2%, and 5% DP conditions and air conditioner RTF (using Method 2); unmet moisture loads are higher than Method 1 (Figure 7)

3.5 Calculated Dehumidifier Sizes

Figure 11 displays the estimated rated whole-house dehumidifier capacity required to control the unmet moisture load calculated using Method 1, which was plotted in Figure 7. The required dehumidifier capacity was calculated using the unmet moisture load with the dehumidifier performance curves provided by Winkler et al. (2012) to account for the indoor design conditions used for load calculations. Other equipment types exist that can control the moisture load at part-load cooling conditions but the capacity of a standalone dehumidifier is easy to conceptualize and the technology is easily simulated in EnergyPlus for comparison. The calculated dehumidifier sizes are significantly smaller than those used in similar simulation and field-based studies (Rudd et al. 2013; Fang et al. 2011; Kerrigan and Norton 2014). However, as the following section will show, the calculated dehumidifier capacity sufficiently handled the moisture load in the annual simulation, highlighting the need for a more rigorous approach to calculating supplemental dehumidification loads.

The 1% DP design condition leads to a nonzero dehumidifier in the Phoenix and Albuquerque code homes, which indicates the 1% DP design condition may be too conservative in sizing supplemental dehumidification equipment. However, there is not a sizeable difference in the required dehumidifier capacity between the 1% and 2% design conditions.

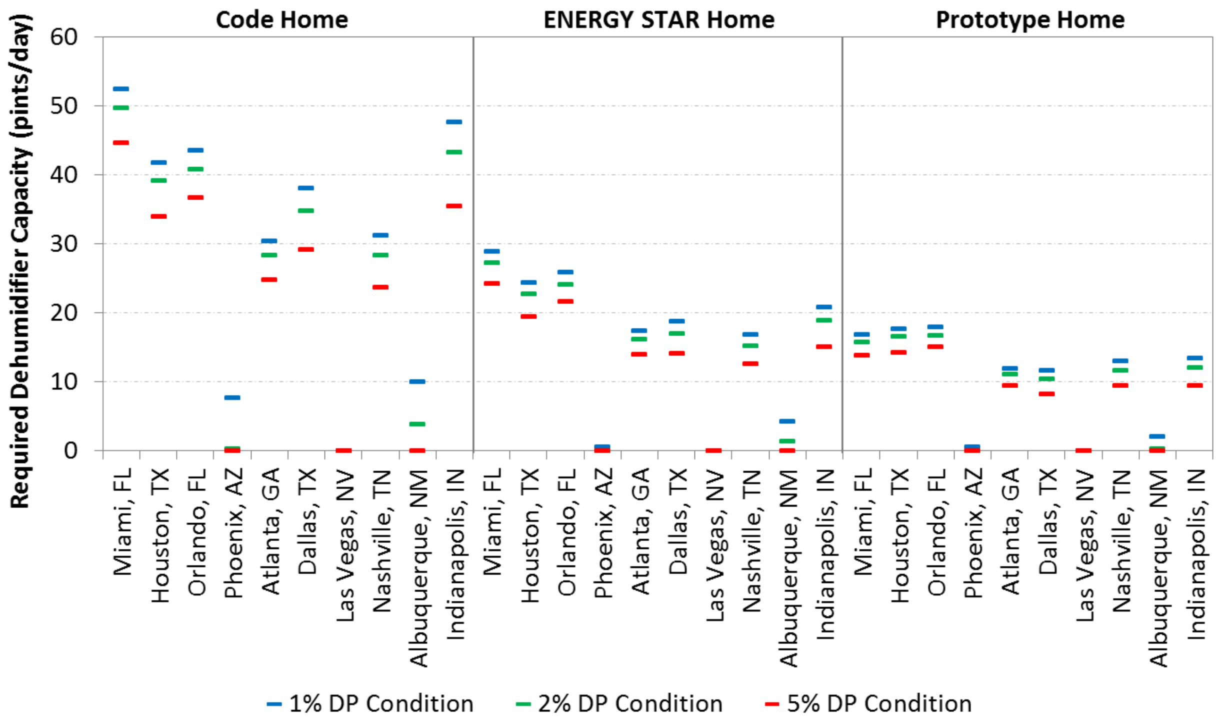

Figure 11. Estimated rated dehumidifier capacity required to meet the unmet moisture load from Figure 7 (Method 1) using 1%, 2%, and 5% DP conditions

Figure 12 displays the calculated dehumidifier capacities when using Method 2. As expected, Method 2 results in larger dehumidifiers but the dehumidifier sizes for the humid cities are still reasonable and don't vary much compared to Method 1. Method 2 predicts up to a 20-pint/day dehumidifier for the code home in Phoenix and Albuquerque, which is another indication that Method 2 is overly conservative in its assumptions.

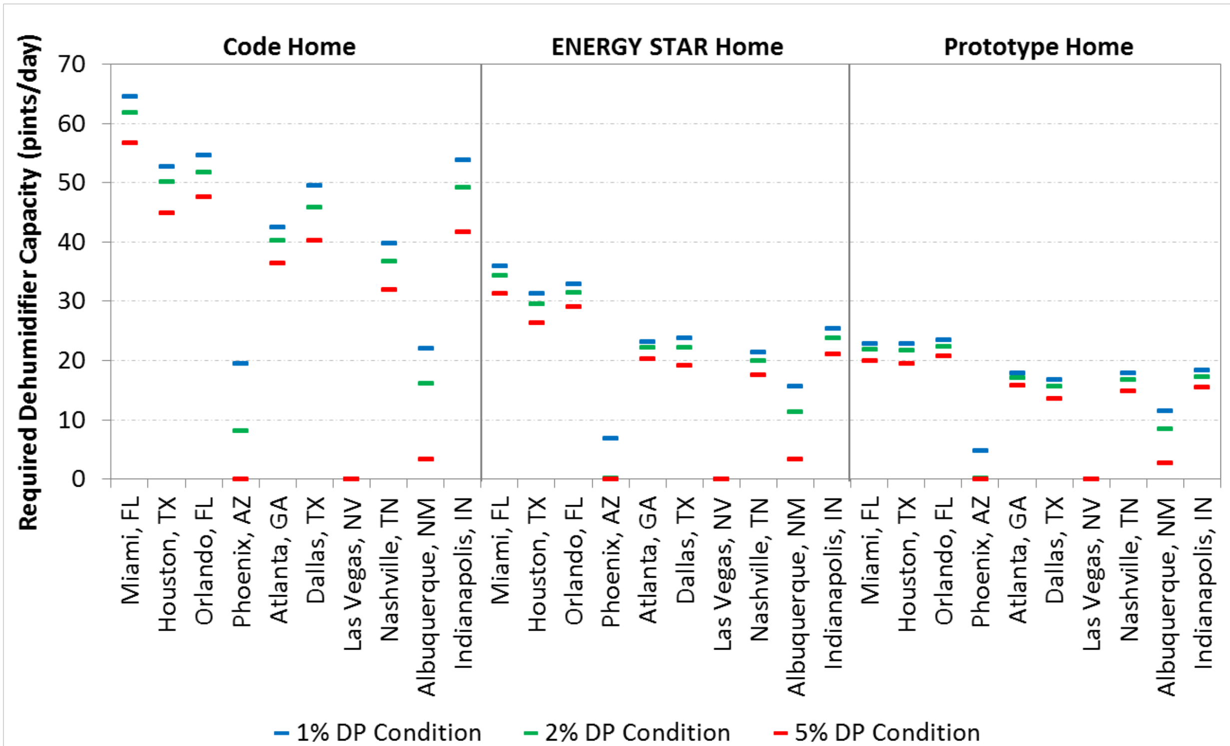

Figure 12. Estimated rated dehumidifier capacity required to meet the unmet moisture load from Figure 10 (Method 2) using 1%, 2%, and 5% DP conditions

3.6 Whole-Building Simulation Results

Houses were simulated using the EnergyPlus whole-building simulation tool with TMY3 data for the corresponding city. Whole-house dehumidifier capacities were based on the load-calculated capacities shown in Figure 11 and Figure 12 for both dehumidifier sizing Methods 1 and 2, respectively. The EnergyPlus simulations gave cooling priority to the air conditioner, thus the supplemental dehumidifier was only run to control the indoor humidity after accounting for the air conditioner latent removal.

Figure 13 plots the percent of time the dehumidifier sized using Method 1 (displayed in Figure 11) was able to maintain the RH set point. Since the dehumidifier was sized based on an indoor RH of 55%, an RH set point of 55% was used in the simulation. Ideally, a dehumidifier sized using the 1% DP condition should meet the moisture load for 99% of an average year. Results for several cases show the RH was controlled for 100% of the time, indicating the dehumidifier in that case may have been oversized or the ASHRAE DP design conditions were more humid than the data included in the TMY3 data. However, when using the 2% design condition to size the dehumidifier, the indoor RH never exceeded 60% RH when using a set point of 55% RH. Thus, using Method 1 with the 2% DP design condition is a suitable approach to calculating the part-load cooling supplemental dehumidification load.

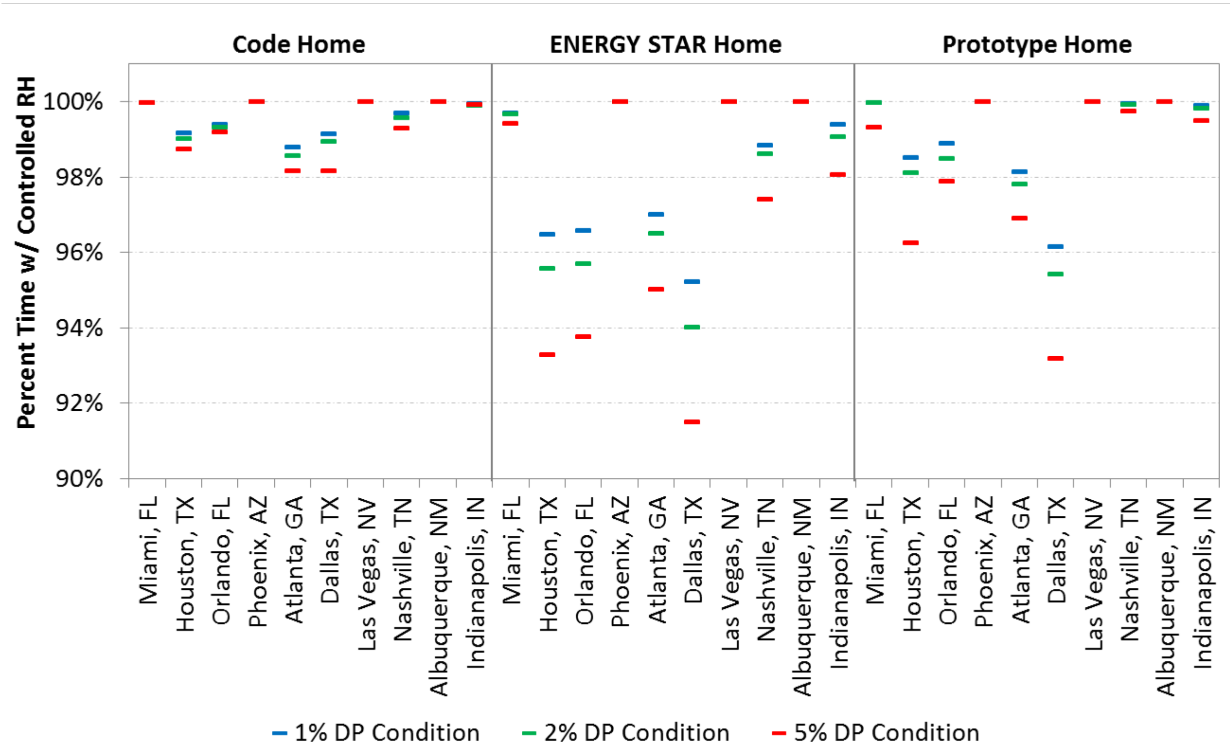

Figure 13. Dehumidifier simulation results when sized based on 1%, 2%, and 5% DP conditions using the Method 1 sizing procedure (Note y-axis minimum value of 90%)

Based on Figure 13, it appears the dehumidifier was under-sized for several cases. For example, a 19-pint/day dehumidifier in the ENERGY STAR home in Dallas, which was sized based on the 1% DP condition, only controlled the RH for a little over 95% of the year. This is partially due to discrepancies between the ASHRAE design data and the hourly data included in the TMY3 weather data. Table 6 compares the ASHRAE DP design temperatures to corresponding values calculated using hourly TMY3 data.

Table 6. Comparison of 1% and 2% DP Design Temperatures to TMY3 Hourly Data

City	1% DP Condition		2% DP Condition	
	ASHRAE Design Data	Hourly TMY3	ASHRAE Design Data	Hourly TMY3
Miami, FL	77.6	75.9	77.1	75.9
Houston, TX	77.3	77.0	76.7	75.9
Orlando, FL	76.9	77.0	76.2	75.9
Phoenix, AZ	69.4	66.9	67.6	64.9
Atlanta, GA	73.3	73.0	72.6	72.0
Dallas, TX	74.4	75.0	73.5	73.9
Las Vegas, NV	63.2	60.1	60.4	57.9
Nashville, TN	74.0	73.9	73.1	73.0
Albuquerque, NM	60.4	60.1	58.8	57.9
Indianapolis, IN	74.0	73.9	72.7	72.0

In Dallas, the hourly DP temperature in the TMY3 data exceeds the ASHRAE 1% DP temperature 1.5% of the time, indicating the weather conditions in the TMY3 data are more humid than the weather data used to determine the design conditions. In Miami the opposite is true; the TMY3 conditions are less humid than the weather data used to determine the design conditions, which partially explains why the selected dehumidifiers in the Miami homes appear to be oversized when compared to EnergyPlus simulation results using TMY3 data.

Despite the dehumidifier appearing slightly undersized for several cases, it is worthwhile to note that the indoor RH was maintained below 60% for all hours for all cases, which is a common metric to quantify a system's ability at controlling indoor humidity (Rudd et al. 2013; Kerrigan and Norton 2014). This is partly due to using a 55% RH set point to control the dehumidifier and the moisture buffering of the building.

Figure 14 plots the percent of time the dehumidifiers sized using Method 2 (displayed in Figure 12) were able to maintain the RH set point. Since the dehumidifiers sized using Method 2 were always larger than the dehumidifiers sized using Method 1, the percent of time with controlled indoor RH increased.

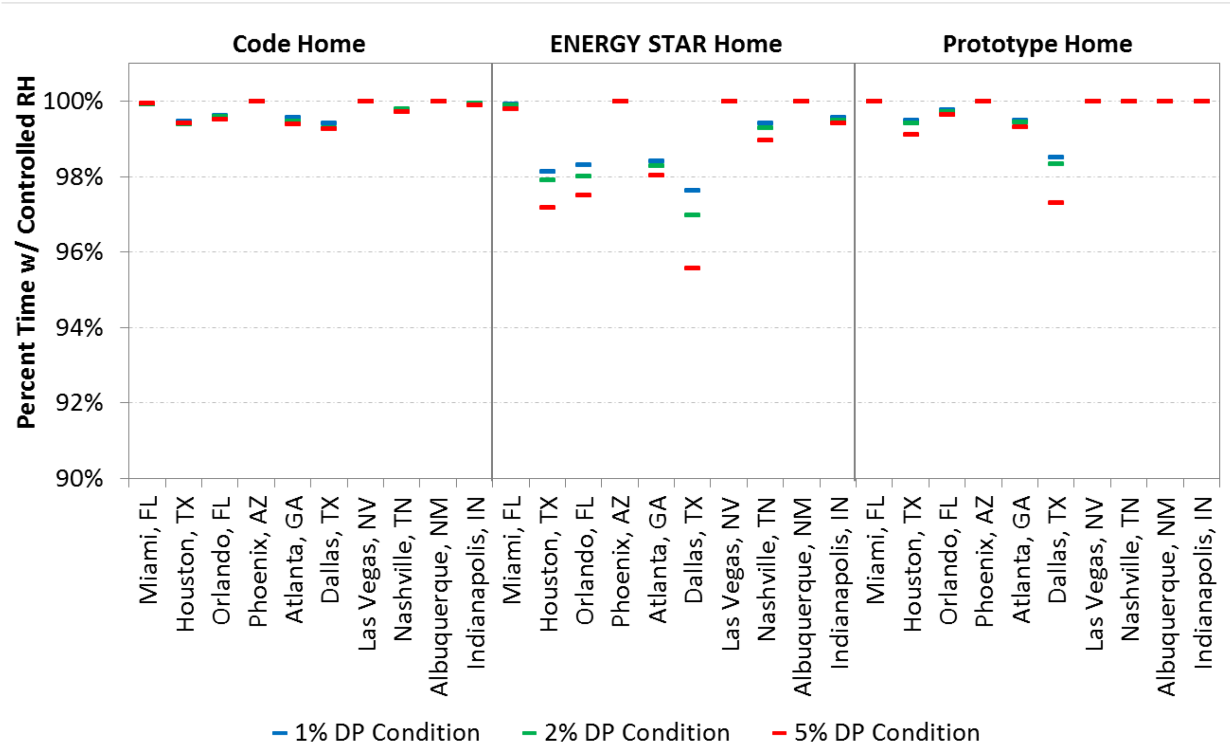

Figure 14. Dehumidifier simulation results when sized based on 1%, 2%, and 5% DP conditions using the Method 2 sizing procedure (Note y-axis minimum value of 90%)

4 Conclusions

Parametric study results using EnergyPlus for three house efficiency levels in ten U.S. cities indicated residential dehumidification equipment can be appropriately sized by slightly modifying ACCA Manual J cooling load procedures. Two methods of sizing residential dehumidification equipment were developed and tested against annual whole-building simulations. Method 1 closely followed the ACCA Manual J cooling load calculation procedure whereas Method 2 included more conservative assumptions related to sensible and latent loads. Humid and dry climates were included in the analysis to test the methods' ability to predict when supplemental dehumidification equipment would not be required.

The results indicate using the 2% DP design condition with Method 1 to size supplemental dehumidification equipment in residential buildings is appropriate. Using the 1% DP design condition resulted in oversized dehumidifiers in many of the cases and using the 2% DP design condition to calculate the part-load cooling moisture load correctly identified which locations would not require supplemental dehumidification. Method 1 more closely followed Manual J cooling load calculation procedures easing adoption and improving the likelihood of implementation.

When using the 2% DP design condition to size supplemental dehumidification equipment, the result indicated:

- Indoor humidity was controlled to 55% RH for more than 94% of the year for all cases without being oversized
- Part-load moisture loads in dry climates were small enough to be handled by the primary cooling equipment
- When using an RH set point of 55%, the indoor RH never exceeded the 60% level.

As part of this project, a methodology was developed to size residential supplemental dehumidification equipment; it relies on ACCA Manual J cooling load calculation procedures. The key steps in sizing supplemental dehumidification include:

1. Calculation of peak cooling sensible and latent loads and sizing of primary cooling equipment (standard ACCA Manual J and S procedures)
2. Calculation of part-load sensible and latent loads using ASHRAE 2% DP design condition with modified Manual J load calculation procedure
3. Prediction of primary cooling equipment moisture removal at part-load conditions
4. Calculation of unmet moisture load at part-load cooling conditions and sizing of required supplemental dehumidification equipment.

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