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Residential Performance Code Methodology for Crediting Dehumidification and Smart Vent Applications Final Report

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Executive Summary

As energy efficient building enclosures, lights, and appliances reduces the hours when air conditioning is called for, and as codes require outdoor air to be brought into homes, humidity levels in homes may rise to the point where supplemental dehumidification is required to control comfort. However, there are currently no standards in Florida's Residential Energy Conservation Code for dehumidification. This report explores rules that could be proposed for dehumidification systems in a future code should builders choose to install them.

Another strategy to reduce space heating, cooling, and dehumidification loads is to allow flexible hours of mechanical ventilation. Research shows potential to save some energy by controlling when ventilation occurs, or "smart ventilation." However, the Energy Conservation Code will need to have a more complete ventilation reference. Furthermore the current performance code is based on a set amount of energy regardless of the type of ventilation system. This report explores an option whereby the proposed home is compared against a standard reference energy use based on type of ventilation system.

A number of references on dehumidification and smart ventilation were examined prior to making recommendations. The references 1)describe a number of ways of removing humidity; 2)describe the effect of humidity removal with respect to runtime of the air conditioner; 3)and many references share results of using building simulations to show the number of hours where various levels of relative humidity are exceeded based on home location and efficiency. The literature also describes some of the issues with whole house mechanical ventilation and some recent research regarding the potential benefits of smart ventilation systems.

Based on the literature, draft recommendations were first made in the February 2017 interim report. After testing those recommendations the authors altered the recommendations. The authors recommend the changes to Energy Conservation Code Table R405.5.2(1) shown in report Table Ex-1 in order to accommodate dehumidification and ventilation.

In addition to Florida Code Table R405.5.2(1) proposed changes, a change to the Energy Conservation Code should be made to indicate the minimum requirements of any dehumidifier installed:

R403.# Dehumidifiers (Mandatory): If installed a dehumidifier:

1. Shall be sized in accordance with ACCA Manual S.
2. Shall have a minimum rated efficiency greater than 1.7 Liters/ kWh if the total dehumidifier capacity for the house is less than 75 pints/day and greater than 2.38 Liters/kWh if the total dehumidifier capacity for the house is greater than or equal to 75 pints/day.
3. Shall operate without requiring operation of the cooling system air handler fan.
4. If connected into the return side of the cooling system, shall include a backdraft damper installed in the return air duct between the inlet and outlet of the dehumidifier.
5. Shall be controlled by a dehumidistat that is installed in a location where it is exposed to mixed house air and does not receive undue direct influence from mechanical ventilation air or supply air from the home's cooling or heating system(s).

Table Ex-1. Recommended changes to pertinent sections of Table R405.5.2(1)

Building Component	**Standard Reference Design**	**Proposed Design**
Mechanical ventilation	None, except where mechanical ventilation is specified by the proposed design, in which case: Type of system modeled: Balanced if balanced or ERV in proposed home, exhaust if exhaust in proposed home, supply if supply system in proposed home Annual vent fan energy use: Where proposed home has a supply or exhaust only system: kWh/yr=0.35*fanCFM*8.76 kWh/y Where proposed home has a balanced system: kWh/yr=0.70 *fanCFM*8.76 kWh/y Where proposed home has predominantly a balanced system with energy recovery: kWh/yr=01.0 *fanCFM*8.76 kWh/y ~~kWh/yr =0.03942xCFA+29.565x(Nbr+1)~~ ~~where:~~ ~~CFA = conditioned floor area~~ ~~Nbr = number of bedrooms~~ Airflow Schedule: Same as proposed average airflow rate but not to exceed requirement of ASHRAE 62.2-2016. Airflow Frequency: Continuous	As proposed As proposed As proposed As proposed As proposed
Dehumidification Systems	None, except where dehumidification equipment is specified by the proposed design Fuel Type: Electric Capacity: Sufficient to maintain humidity at setpoint all hours Efficiency: 1.7 Liters/ kWh if proposed total capacity is less than 75 pints/day. 2.38 Liters/kWh if proposed house total capacity is greater than or equal to 75 pints per day. Location: In conditioned space	As proposed As proposed Sufficient to maintain humidity at setpoint all hours As proposed As proposed
Dehumidistat	None, except where dehumidification equipment is specified by the proposed design Setpoint turn on = 60% relative humidity Setpoint turn off= 55% relative humidity	Same as standard reference

It is also recommended that the Florida Mechanical and/or Residential codes include a section as follows to avoid water damage from the dehumidifier:

[Section #] Dehumidifier drainage. Dehumidifiers shall automatically drain condensate to the outdoors and have a flow switch that shuts off operation when the retaining capacity of the dehumidifier is reached in the event of a clogged drain.

The Florida Building Commission could alternatively reference all or part of the Florida Residential Code Section M1411.3.1, which for convenience is included in the Appendix of this report.

[Section #] Dehumidifier drainage: Drain system shall be in accordance with the applicable provisions of Florida Residential Code Section M1411.3.1.

If these recommendations are incorporated, the code should include all necessary definitions. The ENERGY STAR® specification for dehumidifiers contains definitions for dehumidifier, capacity and energy factor. A definition for residential ventilation balanced airflow will also be required. Suggested language:

[Residential Definition] Balanced mechanical ventilation: A mechanical ventilation system in which the difference between supply and exhaust fan air flows is less than 20% of exhaust air flow.

The recommendations were tested in a version of EnergyGauge® USA to determine the energy impact. A typical home built at minimum code level with tight construction, ACH50 of 3, and ASHRAE 62.2-2013 compliant mechanical ventilation rate had small dehumidification load as the air conditioner ran often enough to maintain moisture level. The reference home dehumidifier increased overall heating, cooling, ventilation and dehumidification (HVCD) use by about 1% (Miami) to 6% (Jacksonville) compared with the same home simulated without dehumidification. However, a low load home (much more efficient than minimum code level) showed a different story with the dehumidifier representing over 25% of the total HVCD energy use.

Current code simulation guidelines provides hours of the year when window operation for natural ventilation and large air exchange is required. In order to model homes with dehumidifiers enabled year round, window operation should be disabled in the simulation program. Otherwise, the dehumidification load becomes excessive and unrealistic.

This study also examined the proposed changes for whole-house mechanical ventilation. As shown in Table Ex-1 the energy specification for the reference home is currently dependent on whether the proposed home has a supply, exhaust, balanced or energy recovery ventilator system. There are also requirements for continuous flow to be modeled in the reference home. One of the objectives of this study was to show the potential energy impact of controlling the mechanical ventilation system in such a manner as to reduce the heating and cooling energy use. A “smart ventilation” strategy based on outdoor temperature was examined to see the impact of such systems (see Figure Ex-1).

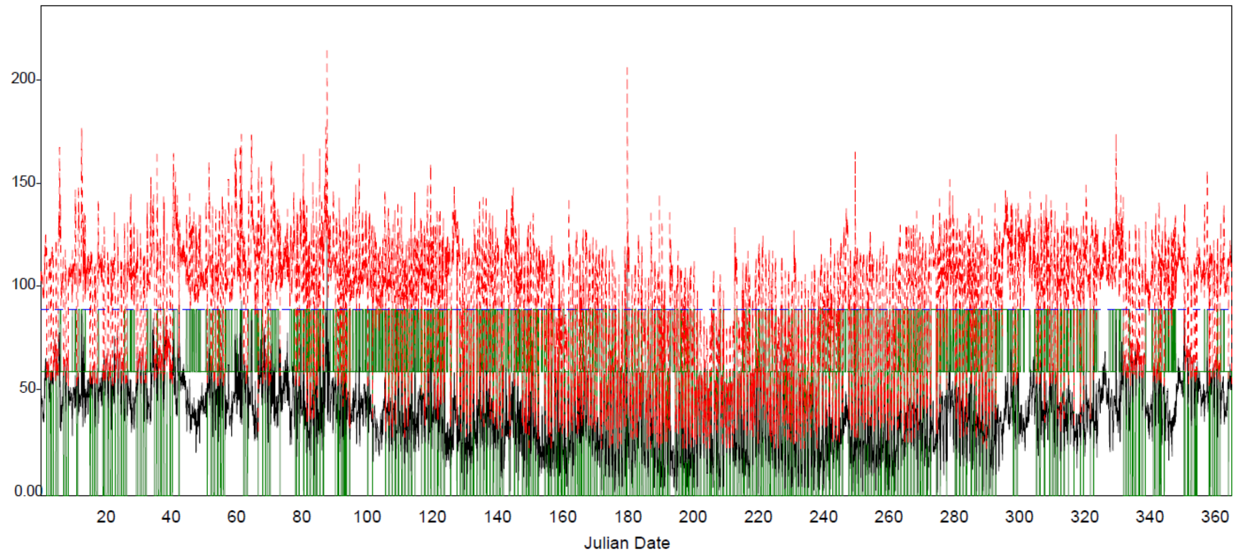

Figure Ex-1. Tallahassee balanced system with outdoor temperature-based smart ventilation showing infiltration airflow in black and the total airflow in red including the varying mechanical ventialtion in green. Fan ventilates at 0, 60 or 90 cfm based on outdoor temperature. Goal is to achieve an annual average of 90 cfm per ASHRAE 62.2-2013 total outside air requirements (blue dashed line).

An outdoor temperature strategy tends to save the most in the heating season and when there is a fair amount of temperature swing through day and night. Simulations were run for Miami, Orlando and Tallahassee. Total heating, cooling, ventilation and dehumidification savings ranged from about 2% to 8.6%. Figure Ex-1 demonstrates how the simulated smart ventilation strategy works throughout the year by ventilating above the ASHRAE 62.2-2013 fan requirement rate of 60 cfm during favorable times and shutting off the fan during unfavorable times and venting at the required rate during moderate times. The report includes the parameters used and also shows how the smart vent simulation compares to the continuous vent method using the method of "equivalency" for intermittent fan use described in ASHRAE 62.2-2016 Appendix C. Exhaust system average annual airflows for the smart ventilation system were equal or greater than those of the non-smart vent systems.

Savings from a hybrid vent system were also estimated. The system was assumed to bring in four times the required ventilation air amount when the system was heating or cooling with a cut off of 25% runtime. If the heating/cooling systems ran less time, then a backup exhaust fan ran. Such a system saves fan energy compared to central runtime systems that would force the system to run 25% of the time each hour. For code purposes such systems would be compared against a continuous exhaust system and were shown to save 1.7% to 2% of HVCD energy use.

Although this report addresses the impact of proposed rules on energy use, questions remain on best methods of implementing dehumidification and whole-house ventilation strategies in Florida homes. Ventilation and dehumidification installations practices, fan flow measurements, and possible discrepancies with ventilation standards remain issues to explore. A discussion section in the report provides a number of issues to consider.

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Residential Performance Code Methodology for Crediting Dehumidification and Smart Vent Applications Final Report

Rationale

As energy efficiency reduces the hours where air conditioning is called for, and as codes require outdoor air to be brought into homes, humidity levels in homes may rise to the point where dehumidification is required. However, there are currently no standards in Florida's Energy Conservation Code for dehumidification. Thus, a home that invests in a heat pipe or low volume technology in order to dehumidify and save energy receives little benefit relative to another home that installs an inefficient dehumidifier. A reference home dehumidification strategy needs to be established. Another strategy to reduce interior moisture loads is to allow flexible hours of mechanical ventilation. Research being conducted by LBNL, FSEC and others are showing potential to save some energy by controlling when ventilation occurs, or "smart ventilation." However, the energy conservation code will need to have a strategy for providing an appropriate baseline for a code reference home.

Overview

The performance method (R405) is the most popular compliance method in Florida. The method requires a software vendor to virtually create a baseline reference home the same size as the home to be permitted and insulate and equip the baseline to a set of parameters spelled out in Table R405.5.2.1. This table includes the temperature that both the to-be-permitted home and the baseline must be maintained to simulate heating and cooling. It also has rules on energy use of the ventilation system for the baseline home. What needs to be added are the following parameters:

1. The interior humidity set point required to be maintained, and whether this applies all year or only at certain times of year. Also, is this set point constant or does it start dehumidifying at one set point and shut off at another like many portable dehumidifiers?
2. The energy use of the dehumidifier in the baseline home. Is using a constant Liter of moisture removed per kWh a sufficient methodology and what should the baseline value be?
3. For simulations that allow smart ventilation, what level of ventilation must be maintained, and if that smart ventilation reduces ventilation during peak times, does the baseline stay constant in its ventilation rate?

Work Performed

The first task was to conduct literature review. Second was to develop draft recommendations. Third is to determine the impact of the draft code recommendations.

Task 1: Literature review of dehumidification strategies, devices and controls.

The literature review task was required at a minimum to include searching databases of NREL, LBNL, ASHRAE, DOE Building America and general search with key words of home or residential dehumidification.

A number of references have been reviewed and the relevant ones are included here as an annotated bibliography. Italics are used to indicate direct quotes from the referenced publication.

This report begins with earlier literature study work by FSEC, specifically:

Charles R. Withers, Jr., Jeff Sonne, "Assessment of Energy Efficient Methods of Indoor Humidity Control for Florida Building Commission Research," June, 2014 http://www.floridabuilding.org/fbc/commission/FBC_0614B/Energy/Energy_Efficient_RH_control_Draft_Final_06_15_14.pdf

This report conducted for the Florida Building Commission, had two parts: a literature review to determine the energy efficiency and cost-effectiveness of various residential latent load approaches; and an experiment measuring the humidity and energy performance of four latent load management approaches at various levels of mechanical ventilation. Key parts of that literature review are copied here so as to avoid repetitive work.

Approximately 30 articles, research reports, presentations and code documents were reviewed by Withers and Sonne. Information sources included the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), Building Science Corporation (BSC), CDH Energy Corp., Florida Solar Energy Center (FSEC), International Code Council (ICC) and Oak Ridge National Laboratory (ORNL).

Indoor Humidity Level Limits

An important first step in determining appropriate latent control approaches is determining what constitutes appropriate indoor humidity levels. In a 2002 publication, Joe Lstiburek (Lstiburek 2002) notes the variety of factors that go into determining proper RH levels:

> *…determining the correct range depends on where the home is located (climate), how the home is constructed (the thermal resistance of surfaces determines surface temperatures), the time of year (the month or season determines surface temperatures), and the sensitivity of the occupants.*

A recent Building America Expert Meeting report (Rudd 2013a) that included input from BSC, CDH Energy Corp., FSEC and IBACOS summarized several publications.

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 its Answers to Research Questions section, the same publication further addresses this topic.

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. …

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.

The EPA Indoor airPLUS program is designed for *improved indoor air quality compared to homes built to minimum code*. This program specifies using equipment that will keep the indoor RH <60% (EPA 2013). The authors consider 60% RH as a reasonable recommended indoor control point for supplemental dehumidification in Florida homes. It is low enough to protect building degradation and a fair balance between energy conservation and comfort. Furthermore, it is an easy setting to find on controllers lacking set point markings on the control knob. While we recognize 60% as reasonable, individual comfort should be allowed to be accommodated. What constitutes comfort varies by individual and even varies in specific individuals over time. Occupants with health issues may have more specific requirements that must be considered.

Rising Indoor Humidity Levels

While, there are some factors that tend to increase indoor RH in new construction and other factors that tend to decrease RH, a 2014 ASHRAE publication (Henderson and Rudd 2014) indicates that overall RH levels are increasing.

Conventional air conditioners have traditionally been deemed adequate for controlling space humidity levels in residential applications. However, as homes in humid climates have become more energy efficient, there is evidence that relative humidity levels in homes have been increasing (Rudd and Henderson 2007). This implies that sensible heat gains to the building have been reduced more than moisture loads, leaving a mix of latent and sensible loads that is poorly matched to the sensible heat ratio of conventional air-conditioning systems.

The 2013 Building America Expert Meeting report noted above (Rudd 2013a) lists the influences modeling has shown to most effect indoor RH in high performance, warm-humid climate homes:

- Internal moisture generation
- Internal sensible heat generation
- Heating setpoint temperature
- Air distribution system duct location.

Regarding air distribution system duct location, the 2014 ASHRAE publication (Henderson and Rudd 2014) explains that moving ducts from the attic to the conditioned space reduces sensible heat gains more than it reduces latent loads, resulting in higher relative humidity levels.

Mechanical ventilation also has a significant impact on indoor RH. A recent monitored FSEC study (Parker et. al. 2014) found mechanical ventilation added to a tight (ACH50 2.2) central Florida lab home to raise summertime moisture levels by 2% - 5%.

Modeling results summarized in the 2013 Building America Expert Meeting report (Rudd 2013a) show this need.

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 paper discussed methods of dehumidifying in some detail. Table 1 from that work is included here:

The experimental work consisted of using a mini-split to bring in outside air with a high efficiency central cooling system and comparing it to just bringing in the outside air to the return area of the central system. Each configuration included a dehumidifier set to 60%RH. Outside air was introduced at 60 cfm, and later repeated at 130 cfm. The 60 cfm was what the IMC2012 would require for a three bedroom home. The 130 cfm represents what ASHRAE 62.2-2013 (or ASHRAE 62.2 2016) would require for an extremely tight home of 0.5 ACH50 with 3025 ft^2 and 5 bedrooms. The mini-split configurations were set to use the mini-split to cool to 74°F and only when it could not meet demand did the central unit kick on at 77°F. This strategy has since been shown effective in most existing homes to save energy, however our lab had a SEER 21 central system and the mini-splits used more energy as they cooled and dehumidified more than just the central system due to the lower set point. All four configurations maintained the relative humidity below 60% during our tests so the dehumidifier did not turn on.

Table 1. Supplemental Dehumidification Options (cost sources: Rudd 2013b and FSEC research). [from Charles R. Withers, Jr., Jeff Sonne," Assessment of Energy Efficient Methods of Indoor Humidity Control for Florida Building Commission Research," June, 2014]

Supplemental Dehumidification System	First-Cost Estimate Including Labor	Pros	Cons
Overcooling	**$0**	Low first cost. User control.	Results in cold clammy comfort. No help in swing season. Energy inefficient
Lowering fan speed	$0-$75	Improved dehumidification. Owner may be able to do this.	Some loss in cooling efficiency. No help in swing season.
Heat pipes	$3000	Long life, low maintenance	May not have room to install. No help in swing season.
Enthalpy recovery ventilation	$700-$1400	Can reduce load from ventilation. Balanced house pressure possible.	Extra energy to run the two fans needed. No help in swing season.
Dual capacity air conditioner	$1800*	Low speed can result in lower energy use while saving energy	Higher first cost. Better than single cap., but still some hour's swing season it will not operate.
Variable capacity air conditioner ventilation	$3700*	Excellent efficiency. Longer run times. Good RH control. Good ventilation mixing.	High first cost. New on residential market, so more to learn.
Dedicated outdoor air system	$7000	Good RH control. Excellent ventilation effectiveness potential.	High first cost.
Mini-split Dedicated outdoor air system	$3200	Good RH control. High-efficiency.	Hard to size solely for low flows. Some localized overcooling may occur at times. Good mixing depends upon central fan cycling.
Stand-alone Dehumidifier with Remote Dehumidistat	$500-$2000**	Works with or without AC. Good RH control.	Energy -inefficient. Adds heat, some RH dead bands can be excessive. Noise may be issue.
Integrated Ducted Dehumidifier	$1,000-2000**	Works with or without AC. Good RH control. Air is distributed better than stand-alone. Noise issue less likely than stand-alone	Energy inefficient. Adds heat, some RH dead bands have been found excessive
Sub-cooling Reheat	$1,600	Good RH Control. More efficient than dehumidifiers.	Overcools and then heats, using energy for both. High first cost.
Full-condensing Reheat	$1,750	Good RH Control. More efficient than dehumidifiers.	Overcools and then heats, using energy for both. High first cost.
Desiccant Dehumidifier	$2,000	Good RH control. Has potential to be recharged by solar or gas	Higher first cost,

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ASHRAE References

Charles R. Withers, Jr., "Measured Space-Conditioning Energy and Humidity in a Mechanically-Ventilated House Lab with Fixed and Variable-Capacity Cooling Systems Located in a Hot and Humid Climate," ASHRAE IAQ Conference, 2016.

This paper presents results of lab research on three methods of cooling and dehumidifying a home mechanically ventilated in accordance with ASHRAE 62.2-2013 (ASHRAE 2013a). The first method *was a minimum efficiency fixed capacity central ducted system, the second was a very high efficiency variable capacity central ducted system, and the third was a single ductless minisplit system.*

The author describes some of the challenges of controlling humidity. Maintaining good indoor relative humidity (RH) and simultaneously providing adequate mechanical ventilation can be challenging during warm and humid weather, particularly during low cooling load periods. During warm and humid weather, mechanical ventilation introduces moisture into a home that must be removed; otherwise the indoor RH may increase beyond acceptable levels during certain hours of the year. The fundamental problem with relying solely on central cooling systems to manage moisture during low sensible load periods is they are oversized for cooler periods of the year despite being "properly sized" for a hot design cooling day. Operation of air conditioning relies on set points that are lower than the room temperature. Lowering the cooling set point during cooler weather increases runtime, but during very low cooling load periods, the space can become overcooled and runtime is not adequate to remove much moisture from the air. This can result in cool, humid (cave-like) uncomfortable conditions.

Withers points out the importance of dehumidistat location: Dehumidifiers can effectively control indoor RH but at lower efficiency than air conditioners. Dehumidifiers that short-cycle or operate with fan run-on at the end of cycles operate very inefficiently (Winkler et al. 2014). Furthermore, dehumidifier operation may occur more than is necessary if the dehumidistat is located in a confined space where mechanical ventilation air is delivered, such as a closet. A dehumidifier with dehumidistat control contained within an isolated mechanical ventilation closet or other location where untreated outdoor air comes in direct contact with dehumidistat control could use 10 times more energy than necessary to maintain acceptable indoor RH (Withers 2015). This stems from the fact that outside air in places like Florida (climate zones 1a and 2a) have RH greater than 60% RH for about 80%-85% of the hours in a year based on TMY3 data. Allowing mechanical ventilation air to mix with dry indoor air before it comes in contact with dehumidistats will decrease the RH and help optimize good RH control and energy conservation. Therefore locating dehumidistat controls and mechanical ventilation delivery should be carefully considered.

The experimental configuration compared a SEER 13 central ducted single speed unit with a backup dehumidifier, a SEER 22 variable capacity central ducted unit with a dehumidifier, and a ductless mini-split heat pump with a SEER 13 central ducted single speed unit as backup. During summer the mini-split and SEER 22 units averaged 52% relative humidity while the base SEER 13 averaged 50% RH. The dehumidifier did not need to run for the SEER 13 unit and only ran 2% of the time for the SEER 22 test. The mini-split ran 95% of the time only requiring the central unit to run 9% of the time. The RH went slightly above 60% RH in this configuration some of the time, between 3am and 8am when sensible loads were low.

During low cooling load periods (Some fall and winter days) tests were limited to one system. The mini-split system maintained 58% to 64% RH in normal mode. Using the manufacturer's dry mode improved performance slightly.

Energy savings for the high efficiency central unit were evaluated at 23.5% and the mini-split at 27%. Each system handled summertime conditions with mechanical ventilation without a great need for additional dehumidification.

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Don B. Shirey III, Hugh I Henderson Jr., "Dehumidification at Part Load," ASHRAE Journal, April 2004.

The paper quantifies the latent removal degradation of vapor compression air conditioning systems under part load. Vapor compression air conditioning systems will re-evaporate moisture on the coil once the system is off as shown in their Figure 2 reproduced here.

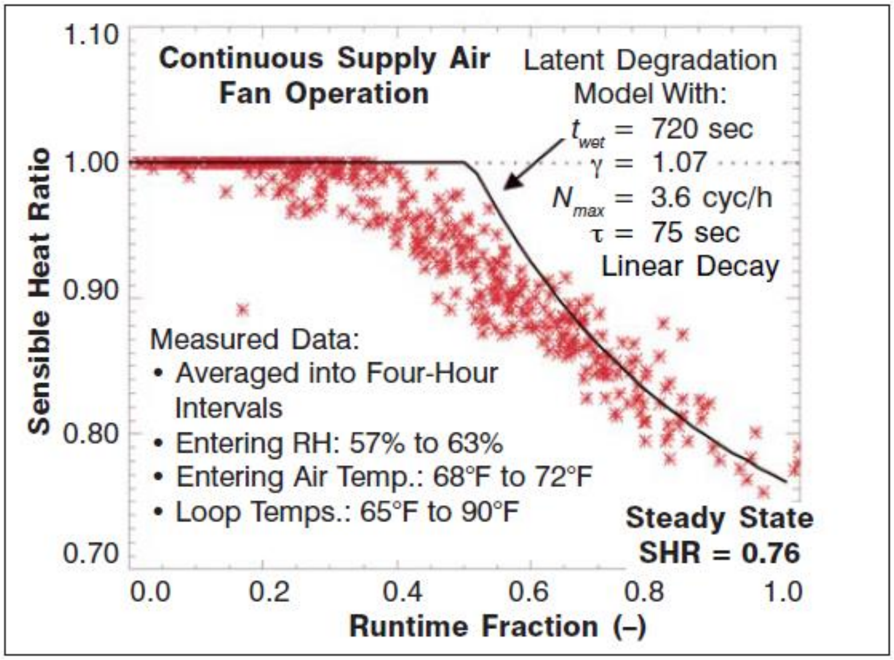

Figure 1. From ASHRAE Journal, 2004, Shirey and Henderson's Figure 2.

Thus the moisture removal capacity is related to the run cycle. Part load latent performance is severely degrade for continuous running fans and still present in auto fan mode. Performance will be closer to steady state if multistage systems are used so at the smaller size the system will have longer runtime fractions.

The amount of time the fan runs after the coil cooling has stopped will only assure that more of the water on the coil will evaporate.

Tested four different coils one of them at two airspeeds determining that the time for condensate to first fall from the coil varied from 12 minutes to 33 minutes for the lab test coils at nominal conditions. The authors provide an equation for.

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Lewis G. Harriman III, Dean Plager, and Douglas Kosar, "Dehumidification and Cooling Loads from Ventilation Air," ASHRAE Journal, November, 1997

The authors introduce a method of characterizing latent and sensible loads from 1 cfm of ventilation air. The proposed "ventilation load index" (VLI) is the total load generated by one cubic foot per minute of fresh air brought from the weather to space-neutral conditions over the course of one year. It consists of two numbers, separating the load into its dehumidification and cooling components: latent ton-hours per cfm per year and sensible ton-hours per cfm per year. For example, a ventilation air load index of 6.7 + 1.1 means that the total annual latent load is 6.7 ton-hours per cfm, and the annual sensible load is 1.1 ton-hours per cfm.

They avoid counting hours where the humidity or sensible loads would be beneficial. They use 75°F and 50% relative humidity for their indoor conditions at which to base the VLI. As can be seen below, Miami has the higher annual loads from 1 cfm of ventilation than those from other states that they analyzed using TMY2 weather data.

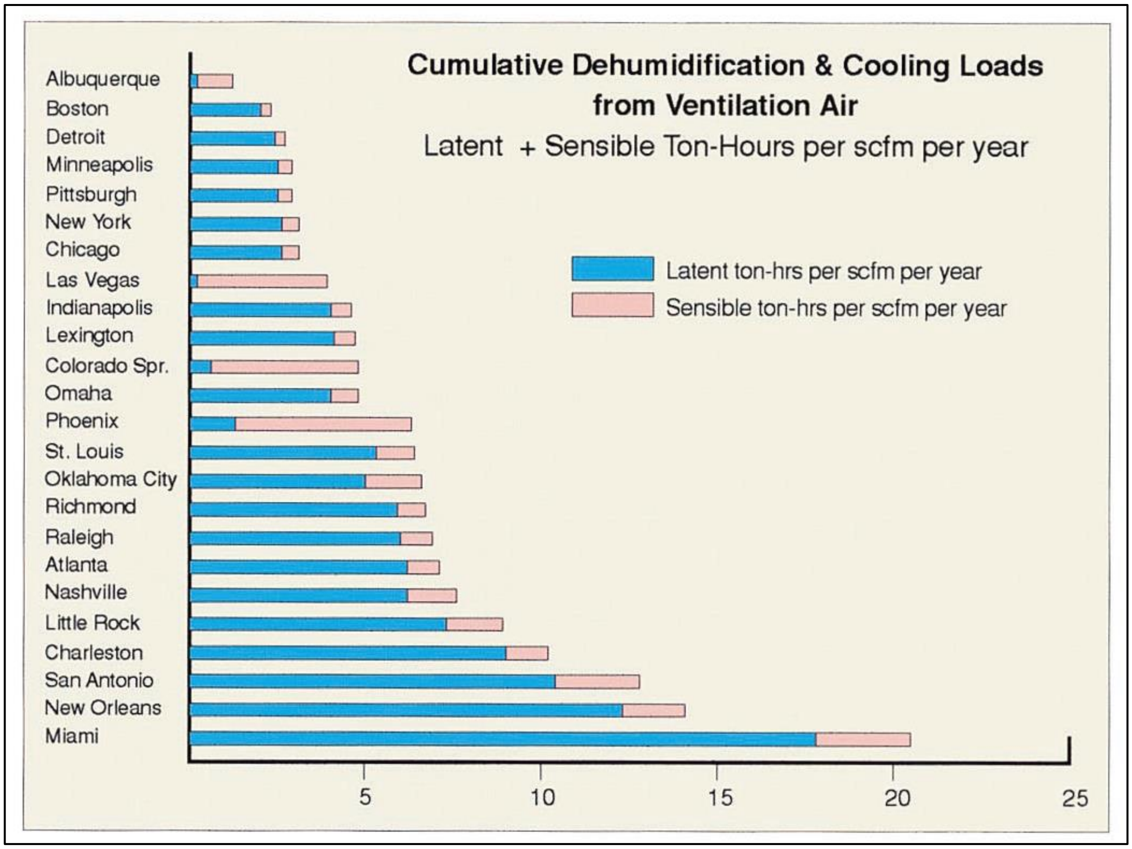

Figure 2. From ASHRAE Journal, November 1997, Harriman, et. al. Figure 2.

Other Florida cities (Excepted from their Table 2 showing latent ton-hrs per scfm and sensible ton-hrs per scfm, respectively)

- Daytona Beach 12.3 1.7
- Jacksonville 12.2 1.8
- Key West 21.6 3.5
- Miami 17.8 2.7
- Tallahassee 11.6 1.7

- Tampa 14.2 2.3
- West Palm Beach 17.0 2.3

Their analysis is helpful in viewing the amount of annual latent load due to each cfm of ventilation.

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Environmental Health Committee (EHC) Emerging Issue Report: Note: Emerging Issue Reports are developed and approved by the ASHRAE Environmental Health Committee (EHC). The Energy Efficient Humidity Control in Hot-Humid Climates Emerging Issue Report was approved by EHC in June 2007.

Energy Efficient Humidity Control in Hot-Humid Climates
This committee provides a summary of issues, largely addressing commercial buildings in humid climates, but focusing on research that is needed on the topic of how to keep buildings dry without overcooling them.

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NREL/DOE/BUILDING AMERICA References

Arlin Burdick, IBACOS, "Strategy Guideline: Accurate Heating and Cooling Load Calculations," prepared for DOE Building America, 2011. http://www.nrel.gov/docs/fy11osti/51603.pdf

The authors indicate the intent of the guide:

This guide presents the key criteria required to create accurate heating and cooling load calculations and offers examples of the implications when inaccurate adjustments are applied to the HVAC design process.

The guide addresses safety factors that are often applied to sizing residential HVAC equipment. By applying safety factors to a house in Orlando they were able to show an almost 3-ton increase in design load.

Combining several adjustments only compounds the inaccuracy of the calculation results. The results of the combined manipulations to outdoor/indoor design conditions, building components, ductwork conditions, and ventilation/infiltration conditions produce significantly oversized calculated loads. The Orlando House example showed a 33,300 Btu/h (161%) increase in the calculated total cooling load, which may increase the system size by 3 tons (from 2 tons to 5 tons) when the ACCA Manual S procedures are applied. Not only does this oversizing impact the heating and cooling equipment costs, but duct sizes and numbers of runs must also be increased to account for the significantly increased system airflow.

The authors summarize the moisture issues associated with oversizing.

In the cooling season in humid climates, cold clammy conditions can occur due to reduced dehumidification caused by the short cycling of the equipment. The cooling system removes moisture from the air by passing the air across a condensing coil. The system must run long enough for the coil to reach a temperature where condensation will occur and an oversized system that short cycles may not run long enough to sufficiently condense moisture from the air. Excess humidity in the conditioned air delivered to a space may lead to mold growth within the house.

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Jon Winkler and Chuck Booten, "Procedures for Calculating Residential Dehumidification Loads," National Renewable Energy Laboratory (NREL), NREL/TP-5500-66515, June 2016 http://www.nrel.gov/docs/fy16osti/66515.pdf

The authors modeled dehumidification requirements for code level, ENERGY STAR LEVEL and what they indicated as BUILDING AMERICA level characteristics. The level of efficiency for envelope and tightness increased for each of these goals –the researchers selected 7, 4 and 1 ACH50 for climate zones 1 and 2 air infiltration. Each home was modeled with continuous ventilation of 50 cfm (authors do not indicate the method of ventilation) and internal moisture gain of about 11 lbs/day. Although the authors do not indicate the meth of ventilation, based on the following it appears it would be exhaust or supply only.

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 (ELA4) 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 ASHRAE 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.

The authors concentrated on how best to size the air conditioning systems and the dehumidifiers. They used two different sizing calculations, similar to ACCA Manual J but not exactly. Their 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*. Their second method used a delta T for summer cooling load through opaque surfaces. They also differed in the treatment of adjoining spaces with a summer type (Solar loaded) procedure for method 1 and a non-solar loaded procedure for method 2. These differences led to larger cooling systems for method 1 than method 2.

The unmet latent load was determined from using steady state performance of the cooling system such that the unmet load was the total latent load minus the product of the cooling system run time fraction and the system latent capacity. Next the unmet moisture load was used to estimate the capacity of a whole-house dehumidifier necessary to meet the load. The dehumidification requirements were modeled three different ways for each of the three homes and two cooling system measures.

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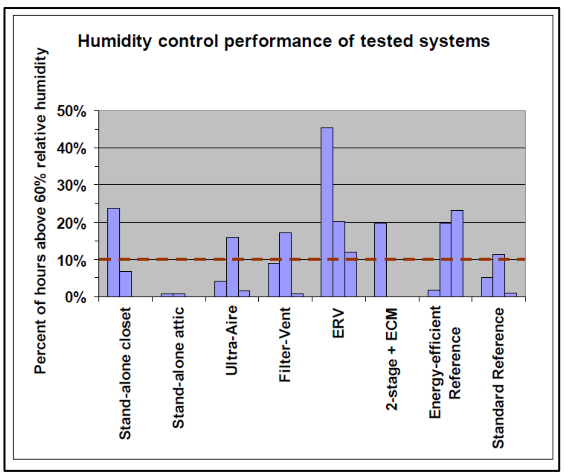
Armin Rudd, Joseph Lstiburek, Kohta Ueno, "Residential Dehumidification Systems Research for Hot-Humid Climates," NREL/SR-550-36643, February 2005 http://www.nrel.gov/docs/fy05osti/36643.pdf also same title at https://buildingscience.com/documents/bareports/ba-0219-residential-dehumidifications-systems-research-hot-humid-climates/view as BA-0219

The authors present results of a Houston, Texas monitored study of twenty homes. Three code level homes had neither ventilation nor dehumidification. Three other homes were built to high efficiency level with controlled mechanical ventilation, but no dehumidification separate from cooling. The other fourteen homes were built to the high efficiency level and had both mechanical ventilation and dehumidification. Two houses had standard dehumidifiers placed in a hall closet, two other placed in the attic, three houses had an ultra-air system, three others an ERV, another three had premixing of outside air with inside air along with a dehumidifier, and one house had a two-stage cooling system with variable fan motor and a “Thermidistat control was both a temperature and humidity controller.” The authors indicate the fan cycling control was set to 33% duty cycle (on for 10 min if it had not been on for 20 min) to intermittently average air conditions throughout the house and distribute ventilation air. Running the fan may have helped the uniformity of air, but it also may have evaporated any moisture remaining on the coil.

The authors present analysis of runtime, energy use and relative humidity. Although the stand alone dehumidifier in a hall closet was not the least energy consuming (The two speed compressor with ECM motor and control was), the authors concluded it may be the best value.



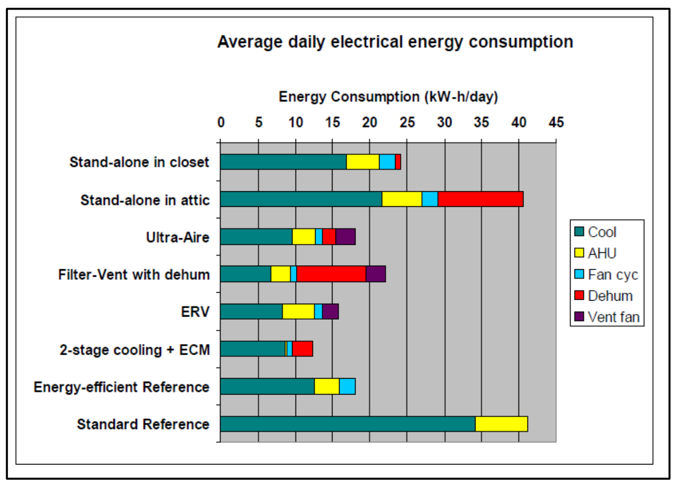


Figure 3. Humidity frequencies and electrical use in homes with six different dehumidification strategies (from Armin Rudd, Joseph Lstiburek, Kohta Ueno, “Residential Dehumidification Systems Research for Hot-Humid Climates,” Figures 10 & 14).

The system providing the best overall value, including humidity control, first cost, and operating cost, involved a standard dehumidifier located in a hall closet with a louvered door and central-fan integrated supply ventilation with fan cycling.

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Dave Korn, John Walczyk, Cadmus “Exactly What Is a Full Load Cooling Hour and Does Size Really Matter?,*” ACEEE Summer Study, 2016*

The authors showed different sizing factors but the most relevant part of their research of metered homes are repeated here from their paper.

To show the impact that system sizing has on humidity, we analyzed meter data of 60 air conditioners operating for an entire cooling season. This controlled sample includes only central air conditioners with single-speed compressors operating in the Midwest—a region with high temperatures and oftentimes high relative humidity.… Conventional wisdom suggests that oversized air conditioners lead to indoor humidity problems. Using a population of 60 directly metered air conditioners, we compared indoor humidity to the operating coincidence factors, directly testing if we could see a difference in humidity in oversized units that ran at low frequencies (short cycle times) at high temperatures. The authors did not see any clear trend in increasing humidity with decreasing run frequency. The description of the Midwest homes used in the study is sparse and they do not mention whether the homes were mechanically ventilated.

Task 2: Literature review of dehumidification set point recommendations and studies of energy use associated with various set points.

Jeff Ihnen, "Keys to Efficient Dehumidification," Engineered Systems Magazine, May, 2009, http://www.esmagazine.com/articles/93776-keys-to-efficient-dehumidification?v=preview

This is a nicely organized guide to dehumidification, explaining some of the key terms and then listing strategies and systems for controlling moisture. Although it appears the article is geared more at commercial buildings, most of the suggestions apply to residential and commercial buildings even if all the systems don't. The four strategies listed are to *only cool to the desired dew point when necessary, control cooling using variable volume as much as possible* [this refers to using low volume flow to remove more moisture], *keep the building positively pressured, shut down outside air when the building is unoccupied.*

The systems suggested by the author are:

- Dedicated outdoor air system (DOAS)
- Precool and reheat outdoor ventilation air with heat recovery.

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Joe Lstiburek, "What relative humidity should I have in my house?" RR-R203, Building Science Corporation, April, 2002

After an introduction to ASHRAE recommendations the author concludes:

Keeping relative humidity in the 25 percent to 60 percent range tends to minimize most health issues – although opinions vary greatly…The range of 40 percent to 60 percent relative humidity is commonly incorrectly recommended for health and comfort reasons. As we will see, there is a big difference between 25 percent as a lower limit rather than 40 percent – particularly in very cold and cold climates.

The author discusses heating climates and then following about mold growth in cooling climates.

In cooling climates, interior mold growth also occurs because interior surfaces are typically cold and then exposed to moisture levels that are too high. The cold surfaces in cooling climates arise from the air conditioning of enclosures. When exterior hot air is cooled, its relative humidity *increases. If the exterior hot air is also humid, cooling this air will typically raise its relative humidity above the point at which mold growth can occur (70 percent).*

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Philip Fairey, Danny Parker, Robin Vieira and Eric Martin, "Vent Right and Then? Mechanical Ventilation, Dehumidification and Energy Use in Humid Climates," FSEC-PF-460-14, August, 2014 *http://www.fsec.ucf.edu/en/publications/pdf/FSEC-RR-505-14.pdf*

This paper points out other Building America simulation work that used 60% relative humidity as a dehumidification set point. It also indicates that home ventilation in the southeast makes humidity control a challenge that otherwise would be minimized: ASHRAE 55-2013 on Thermal Environmental Conditions for Human Occupancy intends that indoor dew point temperatures be maintained below 62°F. The operating characteristic of typical air conditioning equipment is such that indoor dew point temperatures are normally near 55°F during the summer air conditioning season. In hot, humid southeastern and gulf coast climates where summertime outdoor average dew point temperatures reach 75°F, ventilation can introduce significant quantities of excess moisture into homes, presenting indoor comfort and moisture control issues that do not exist in other climates.

The paper reports on side-by-side identical unoccupied labs that have internal heat and moisture generation. One has had an air leakage of 8 ACH50 and the other 2 ACH50. At one point a supply ventilation system was installed in the tight home delivering 63 CFM of air consistent with ASHRAE 62.2-2013 for a 3-bedroom home while the leakier home remained unventilated. The tight home ran in two week cycles of ventilation system on and off in order to examine indoor conditions and energy use under two different circumstances. The research included injecting CO2 into the unoccupied homes at a constant interval to measure the infiltration using the CO2 as a tracer gas.

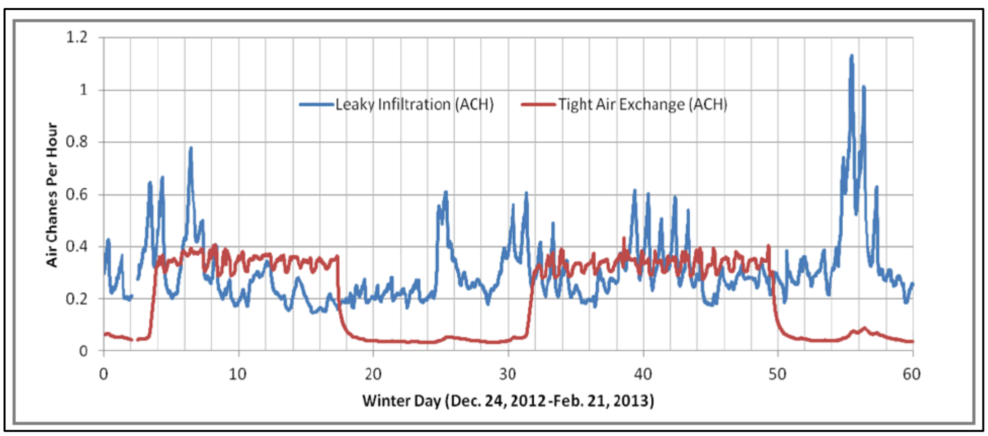

Figure 4. Winter measured air exchange between leaky 8ACH50 home (Blue line) and tight 2ACH50 home (Red line). Tight home ran for two weeks with 63 cfm of supply ventilation air and then was 0 ventilation for two weeks.

In summer there was little to no difference in energy use between the tight home and leakier home during periods where the tight home was not ventilated. However, the authors found that mechanical ventilation of the tight home increased *cooling energy use by 20-38% or about 4 kWh per day compared with the leaky home. Mechanical ventilation of the tight home increased indoor RH modestly by 2-5%. However, mechanical ventilation increased the comparative* *quantity of air conditioner moisture removal significantly by 27%.* This last result indicates that the air conditioning system was able to remove the majority of the ventilated moisture.

The authors also report on a total of 864 simulations were run for new home configurations using two building archetypes (1 story and 2 story), two building leakage rates (1.5 and 3 ACH50), two building orientations, three ventilation system types, three ventilation rates, and 12 climates. Results of the number of hours above threshold levels are shown below. At 60% relative humidity there are over 1500 hours for an exhaust or fan integrated ventilation system; most of these hours occur during milder weather where neither the heat or cooling systems are working and the number of hours that exceed 65% relative humidity fall to about 500 while only about 100 hours are greater than70% relative humidity.

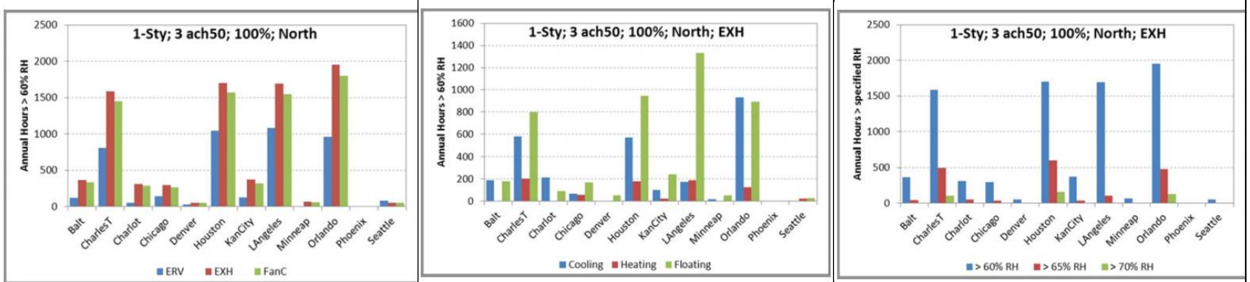

Figure 5. [Left]Simulation results of number of hours above 60% relative humidity in different climates for three types of ventilation systems, energy recovery (ERV), exhaust only (EXH) and fan integrated (FanC).[Center] Those hours occur primarily during floating periods where the sensible load is insufficient for air conditioner to run. [Right] Simulation resultsof hours above 60%, 65% and 70% relative humidity for exhaust only ventilation.

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Mattison, L. and D. Korn (2012). "Dehumidifiers: A Major Consumer of Residential Electricity." ACEEE Summer Study on Energy Efficiency in Buildings, 2012. The Cadmus Group, Inc. http://www.aceee.org/files/proceedings/2012/data/papers/0193-000291.pdf.

The Cadmus Group metered 21 dehumidifiers operating in 19 homes in Massachusetts, New York, Maryland and Virginia. Metering of each unit began between mid September and early October 2011 and continued for one to 12 weeks. \

The authors found that the dehumidifiers used a considerable amount of energy, did not perform as efficiently as their rating under real time conditions and had difficulty with the accuracy of the humidity control. Here is a list of their conclusions:

- The average metered active power was 459 Watts.
- The average metered runtime was 8.9 hours/day. At 8 months/year, the average unit would operate 2,160 hours annually.
- Eleven of the units drew standby power between 0.4 and 1.9 Watts.

- The average metered electricity consumption was 4.2 kWh/day, or 1,000 kWh/year based on 8 months/year of operation. This is equal to 9% of the electricity consumption in an average home.
- For the 15 manually emptied units, the average water removal was 4.9 pints/day and the average EF was 0.8 L/kWh.
- The humidity controls on some units did not function properly, as some units did not operate when a separate meter showed ambient RH exceeding the setpoint.
- The measured EF was lower than the rated EF for all but two units. This lower operating efficiency is believed to be in part because most units in this study were operating in spaces with lower temperature and RH than the standard test conditions.
- User operation is a key factor in effectiveness and energy consumption of dehumidifiers, including frequency of emptying tubs for units that don't drain directly.

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Lawrence Berkeley National Laboratory (LBNL)

Henry Willem, Camilla Dunham Whitehead, Chun Chun Ni, Venessa Tavares, Thomas Alan Burke, Moya Melody, and Sarah Price, 2013. Field-Monitoring of Whole-Home Dehumidifiers: Initial Results of a Pilot Study; November 2013

LBNL Monitored three Wisconsin homes with whole house dehumidifiers located in the basement. One system dehumidified the basement, another house and the third basement and house. The units used 8 to 9kWh/day on average with set points ranging from 40% RH to 50% RH. RH varied some during standby mode with each system. Two of the systems took air from the basement causing negative pressure which might mean more air was infiltrating to the basement from either the main house or the outside. *A decrease of RH in the range of 18-34% (mean, daily) was recorded among the study sites. However, the effect was associated with elevated air temperature in the range of 11°F to18°F (mean, daily).*

Danny S. Parker, FSEC for LBNL, "Determining Appropriate Heating and Cooling Thermostat Set Points for Building Energy Simulations for Residential Buildings in North America," May, 2013, http://fsec.ucf.edu/en/publications/pdf/fsec-cr-2010-13.pdf

This document relates as it has a good literature review of studies in which residential temperatures were measured. Higher temperatures in houses can handle more absolute moisture before the relative humidity exceeds a certain level. The report showed that in heating climates the measured temperatures during heating periods were often 68°F or less; and for Florida the measured temperature during cooling was 78°F with a nighttime set lower at 77°F.

ENERGY STAR Specification for Dehumidifiers Version 4

The specification document is included in the appendix to this report. It includes definitions of dehumidifiers, capacity, and energy factor. It also contains energy efficiency and test requirements. The efficiency requirements are for units with capacity < 75 pints per day, the energy factor must be > 2.00 liters/kWh. For units with capacity 75 pints per day to 185 pints per day the ENERGY STAR requirement is 2.80 liters/Kwh. There is no ENERGY STAR labeling for units larger than 185 pints per day. The standard went into effect October 2016.

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Task 3: Literature review of smart ventilation strategies and recent developments at ASHRAE and LBNL regarding allowances.

This includes reviewing papers from the most recent ASHRAE conferences and searches for resources with keywords of smart ventilation, temperature controlled ventilation, and humidity controlled ventilation.

ASHRAE's annual 2016 Conference was held in St. Louis MO from June 25 to June 29. Review of the conference guide revealed no pertinent papers. Searches of LBNL and NREL/FSEC websites found thirteen applicable papers written from 2014 to 2016 summarized in the following annotated bibliography. No references were found that dealt directly with code modifications or allowances.

The following conclusions can be generalized from these papers

- *Smart ventilation controls were effective at reducing indoor humidity levels, and they maintained air quality equivalent to or better than a continuous fan sized to 62.2-2013.*
- The majority of information regarding the energy and moisture impacts of mechanical ventilation is based on simulations using one of two software packages, LBNL's REGCAP or FSEC's EnergyGauge USA.
- Low-load efficient houses in Florida will have significant periods of interior humidity above 60%RH regardless of ventilation systems due to interior generated moisture load at times of minimal or no cooling system operation.
- Health impacts of ventilation are not studied in any significant detail.
- Mechanical ventilation in Florida will increase interior humidity and require more HVAC energy.
- Natural infiltration in a Florida home built to 8 ACH50 will not provide the necessary ventilation rate to comply with 62.2-2013 due to Florida's mild climate and the resulting reduced infiltration drivers.
- *High indoor humidity generally does not occur during cooling system operation and most problems occur during winter and shoulder season transitions or during late evening and early morning hours.*
- *Sensible cooling load drives cooling system moisture removal.*
- *Ventilation has non-negligible but secondary impacts on indoor humidity levels.*
- Very tight construction risks excessive and potentially damaging indoor moisture levels.
- FSEC's simulation work indicates that application of an Enthalpy Recovery Ventilator in lieu of the exhaust ventilation will significantly reduce indoor humidity.
- Simulation results in all California climates using LBNL's RIVEC controller show that smart ventilation control systems can reduce the energy penalty from ventilation by more than 40% without compromising long-term and short-term exposure to indoor pollutants, however this includes the impact of California's time-of-use electrical charges.
- Several studies, including a recent FSEC study, show significant failures of ventilation systems in the field, ranging from dirty, clogged filters to fan failure.

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ASHRAE Standing Standard Project Committee 62.2, ASHRAE Standard 62.2-Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings. American Society of Heating Refrigeration and Air-Conditioning Engineers.

62.2-2010-Among much else this standard defines minimum cubic feet per minute (CFM) requirements for ventilation systems. CFM requirements are based on the number of bedrooms and the area (NOT volume) of the conditioned space. Continuous ventilation rate (CFM) = (conditioned floor area (CFA)*0.01)+(7.5*number of bedrooms (Nbr)+1) A default natural infiltration level of 2 CFM per 100 ft2 of floor space is used. The only implied requirement for Smart Ventilation Control is an effective ventilation rate of intermittent systems that provides a flow equal to the HOURLY requirement of a continuously operating fan

62.2-2013-This standard updates Standard 62.2-2010. A major change is a replacement of the default natural infiltration credit with the actual, measured annual average infiltration rate. The calculation is now Required CFM=(CFA*0.03)+(7.5*Nbr+1), required mechanical ventilation is equal to the Required CFM minus the calculated infiltration CFM. This results in a significant increase in the mechanical ventilation rate for more air tight buildings. The updated standard further defines intermittent mechanical ventilation systems, requiring ventilation operation at a minimum of every 3-hours, or a daily equivalent flow. A further enhancement to the standard is a definition of equivalent ventilation, allowing a smart ventilation controller to provide an ANNUAL exposure rate (level or amount of indoor pollutants) less than or equal to that provided by continuously operated ventilation systems.

62.2-2016-This Standard updates 62.2-2013. This standard makes major changes and clarifies the intent of 62.2-2013's intermittent ventilation requirements. Short-term average ventilation is defined to be a 3-hour based equivalent ventilation rate. The Standard further defines scheduled ventilation systems based on annual relative exposure to indoor pollutants. A new definition, "Real Time Control" calls for active ventilation control that provides equivalent exposure based on a minimum of daily to a maximum of yearly equivalent exposure rates.

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Brennan Less and Iain Walker, Nov. 2016, Smart Ventilation Control of Indoor Humidity in High Perfromance Homes in Humid U.S. Climates, Ernest Orlando Lawrence Berkeley National Laboratory. LBNL-1006980, https://eta.lbl.gov/sites/all/files/publications/1006980.pdf

This paper summarizes recent simulation efforts and field studies and presents the results of simulation work that looks at 13 different smart ventilation control strategies. The objective of the simulations was to reduce the number of hours of high indoor humidity (greater than 60% RH).The simulation evaluated high performance, single-family homes that meet the U.S. DOE Zero Net-Energy Ready home requirements, using three house sizes: 100 m3, 200m3, and 300m3, three internal moisture gains: 3, 6.5 and 11.8 kg/day, and six hot-humid climates-two in Florida, Miami and Orlando.

Key findings of past work summarized in the paper are: *High indoor humidity generally does not occur during cooling system operation and most problems occur during winter and* *shoulder season transitions or during late evening and early morning hours; Internal moisture generation has a strong impact on indoor humidity; Sensible cooling load drives cooling system moisture removal, in particular duct location (house vs. attic) and thermostat setting; Mechanical ventilation has non-negligible but secondary impacts on indoor humidity levels; Supplemental dehumidification is required in high performance homes in humid climates, irrespective of mechanical ventilation rates; Homes using supplemental dehumidification strategies are able to reduce, but not eliminate hours of indoor relative humidity above60% (on average from around 30% of annual hours to 15% of hours >60%; dehumidifier capacity and set points interact such that all high humidity hours are not eliminated). Supplemental humidity control strategies have mixed effectiveness and first costs from $150 to $2,000 Research estimated that supplemental dehumidification in high performance homes requires approximately 170 kWh per year with a 60% RH set point and estimated that dehumidifiers operate 10% of the year in high performance homes with annual energy use of 976 kWh/year. Field research in conventional homes suggests that dehumidifiers use between 300 and 2,000 kWh annually, averaging 1,000 to 1,200 kWh per year.*

The simulation compares the results from the smart control algorithms to baseline simulations using a constant fan to provide 62.2-2013 ventilation rates. Control algorithms were of four generic types: scheduled, sensor-based, relative dose target, and cooling system tie-ins, and hybrids of these. Simulations are performed by REGCAP - *LBNL's in-house residential building energy and ventilation simulation tool with mass, heat, and moisture transport models. (extracted from LBNL-5969E, Commissioning Residential Ventilation Systems July 2012, Walker et.al.)*

The paper concluded:

- *High indoor humidity was not an issue in many combinations of location, house size and moisture gains. The most problematic cases were small homes with high moisture gains, where between 5 and 40% of annual hours were >60% RH.*
- *Smart ventilation controls were effective at reducing indoor humidity levels, and they maintained air quality equivalent to or better than a continuous fan sized to 62.2-2013. The best performing strategy used both indoor and outdoor sensors and a cooling system tie-in. It was able to* ***reduce 16% of annual hours <60% RH in a small Miami home using under 300 kWh****.*
- *Estimated energy use for smart controls was in the same range as that used by mechanical supplemental dehumidification strategies.*
- *In the most challenging cases, indoor humidity remained >60% for 20 to 25% of annual hours despite use of smart controls, and use of supplemental dehumidification in humid climates may be necessary to achieve acceptable levels in these high performance homes. Our next steps are to evaluate how smart ventilation controls interact with and compare to a supplemental mechanical dehumidification strategy.*

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W. Turner and I. Walker, Dec. 2012, Advanced Controls and Sustainable Systems for Residential Ventilation. Ernest Orlando Lawrence Berkeley National Laboratory. LBNL-5968E. https://buildings.lbl.gov/sites/all/files/lbnl-5968e.pdf

This paper looks at ventilation energy use in all 16 California climates. It uses a baseline house with no ventilation and compares, using simulations, standard ASHRAE 62.2-ventilation and a controlled ventilation system, "Residential Integrated Ventilation Controller" (RIVEC). RIVEC monitors all of the house's ventilation devices, bath and kitchen fan, dryers, etc. and occupancy. One of the main objectives of RIVEC is to eliminate vent fan operation during peak demand periods. RIVEC control shifts the ventilation load away from peak demand periods.

Simulations used REGCAP - *LBNL's in-house residential building energy and ventilation simulation tool with mass, heat, and moisture transport models. (extracted from LBNL-5969E, Commissioning Residential Ventilation Systems July 2012, Walker et.al.)* Baseline for simulation comparisons is no whole house ventilation. Simulations looked at all 16 California climates. Three different house sizes and constructions were evaluated. Each house was simulated using three different infiltration levels. House shells met CEC Title 24 Package D. Ventilation equipment simulated was taken from the Home Ventilation Institute's 2011 Directory.

The results show that RIVEC systems can reduce the energy penalty from ventilation by more than 40% without compromising long-term and short-term exposure.

- Strategy 1-Whole-house fan. RIVEC control reduced annual vent energy from 38% to 52%, mean of 46% or 592kWh.
- Strategy 2-Heat Recovery Ventilator. RIVEC control savings range from 25% to 38% with means of 31% or 876 kWh (note that HRV operation includes running air handler fan at the same time for distribution of vent air).
- Strategy 3-Central Fan Integrated Supply and whole house exhaust fan. RIVEC control resulted in 34% to 52% savings with a mean of 43% or 573kWh.
- Predictions of the impact ventilation would have on California housing range from 5% to 32% of total building load. REVIC is assumed to reduce this by at least 25%. This exercise is continued to its end; predicting State-wide saving if implementing REVIC of 3010 GWh.

Conclusions regarding use of the RIVEC controller are:

- Reduce whole-house ventilation energy by at least 40% while in compliance with 62.2-2010.
- No acute exposures
- Energy reductions are robust across climate, house size and leakage rates.
- Predicted household savings of 500 to 7500 kwh/year based on climate.
- Reduce peak power by up to 2kW for a typical house.

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P. Fairey et.al., August 2014, Vent Right and Then? Mechanical Ventilation, Dehumidification and Energy Use in Humid Climates. Florida Solar Energy Center, FSEC-PF-460-14. http://fsec.ucf.edu/en/publications/pdf/FSEC-PF-460-14.pdf.

This paper covers the impacts of mechanical ventilation and includes simulation results and preliminary results from two monitored full-scale lab homes with simulated occupancy, designed to be typical existing Florida homes. The monitored homes were configured to compare tight and leaky envelopes with and without mechanical ventilation. The simulations were conducted using

EnergyGauge USA, a residential building energy analysis and rating program developed by the Florida Solar Energy Center. The simulations were of new, high-performance homes with mechanical ventilation in 12 American cities, including Orlando FL, as well as older homes in Orlando FL with and without air tightening and mechanical ventilation.

Conclusions reached from the monitored lab homes include:

- Very tight construction (2 ACH50) risk excessive and potentially damaging indoor moisture levels without ventilation
- Summertime indoor humidity levels for a tight home (2 ACH50) employing 62.2-2013 exhaust ventilation will be greater than found in a loose (8 ACH50) unvented home. The simulation work indicates that application of an ERV in lieu of the exhaust ventilation will significantly reduce indoor humidity.
- The loose (8 ACH50) home may not achieve 62.2-2013 ventilation levels due to small infiltration driving forces.
- Standard air conditioning summertime usage removes significant moisture from the house, reducing summer interior RH concerns

Conclusions from the simulation results are:

- When air-tightening an unvented existing home from 11 ACH50 to 5 ACH50 with dehumidification and 62.2-2013 ventilation the energy use for the ventilation and dehumidification may be larger than the potential heating and cooling energy saved.
- In humid climates tight, high-performance homes with ventilation experience significant periods of interior humidity above 60%. The majority of the high humidity situations occur during floating hours with no space conditioning requirements. High interior humidity is worst when an ERV is employed.
- If the desired maximum interior humidity level is raised from 60% to 65% a large fraction of the hours of concern are eliminated.
- Modeling of both new and existing homes show operating costs are not significantly impacted by choice of ventilation system

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Brennan Less, Walker, I., and Tang, Y. 2014. Development of an Outdoor Temperature-Based Control Algorithm for Residential Mechanical Ventilation Control. Ernest Orlando Lawrence Berkeley National Laboratory.https://publications.lbl.gov/islandora/object/ir%3A1005599

This is a summary of methodology and simulations used to develop a simple, outdoor temperature based control strategy to reduce the energy impact of ASHRAE 62.2-2013 ventilation. One and two story, 2100 ft2 buildings were simulated in fifteen different U.S. DOE climate zones, including Miami and Houston (1A and 2A). The building's insulation varied by climate zone. Six different infiltration levels, from 0.6 ACH50 to 10 ACH50 were modeled. Four different temperature-based ventilation control strategies were modeled, a fixed temperature of 5 C, a fixed percentile or two methods based infiltration using the enhanced ventilation model from ASHRAE Fundamentals. Simulations used REGCAP - *LBNL's in-house residential building energy and ventilation simulation tool with mass, heat, and moisture transport models.*

(extracted from LBNL-5969E, Commissioning Residential Ventilation Systems July 2012, Walker et.al.)

The work only looks at controlling ventilation based on a minimum temperature threshold. The recommendations for Miami, in Climate zone 1A, were to do nothing, and the results for Houston showed very small savings (best case 250 kWh to 480 kWh). The paper concluded that in approximately 35% of the test cases they would recommend no temperature-based control. Controlling for maximum temperatures, humidity differences or other strategies appropriate for hot, humid climates was not investigated in this paper.

In 3 to 10 ACH50 test homes, substantial energy savings have been shown to result from the smart control of ventilation systems based on outdoor temperature, while maintaining equivalence with ASHRAE 62.2-2013 through fan oversizing. Limited savings were realized in milder climates for tighter homes. Energy reductions generally increased with climate severity, and in nearly all cases, they were greatest in airtightnesses 3 and 5 ACH50. Simulations demonstrated annual HVAC energy savings ranging from approximately 100 kWh to 4,000 kWh. Using a sequential optimization tool, fans were oversized by an average of 34% (ranging from approximately 5% to 150%), and equivalence with 62.2-2013 was maintained in all of these cases. Temperature controlled ventilation is not recommended in climate zone 1 or in most of the very airtight cases (i.e., 1.5 and 0.6 ACH50).

As a general guiding principle, energy savings increased with reductions in mechanical fan runtime, resulting from higher cut-off temperatures. These reductions in runtime required larger fan sizes in order to maintain equivalence with 62.2. This dynamic was not consistent in more airtight homes, where higher cut-off temperatures often necessitated substantially larger fans to maintain equivalence, which led to increased HVAC energy use. The simplest strategy (a 5°C cut-off) was in fact the most effective across a variety of climate zones, though it was not effective in all cases where savings were identified.

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Eric Martin et.al, August 2014. Measured Cooling Season Results Relating the Impact of Mechanical Ventilation on Energy, Comfort, and Indoor Air Quality in Humid Climates, Florida Solar Energy Center. FSEC-PF-461-14 http://fsec.ucf.edu/en/publications/pdf/FSEC-PF-461-14.pdf

Ten homes in Gainesville FL were studied to evaluate the impact of ASHRAE 62.2-2010 ventilation. The homes were U.S. DOE Builders Challenge complaint (HERS <65). Homes were three to four years old. All homes had an existing central fan integrated supply ventilation system (CFIS) providing approximately 20% of the ASHRAE 62.2-requirements. Larger bath exhaust fans, capable of meeting 62.2-vent rates, were installed. Six houses flip-flopped, or ran alternating ventilation systems for two week periods all summer- two weeks with CFIS, two weeks with continuous exhaust ventilation (CEV) from June 28 till October 15, 2013. As controls two houses each were run continuously with either CFIS or CEV.

The report concluded:

- *The continuous exhaust ventilation systems result in approximately 9% more space conditioning energy use on average to maintain the desired temperature set points in the homes*
- *Resulting RH and dew point are higher in the homes while under continuous exhaust.*
- *Preliminary analyses of the data indicate that concentrations of acetaldehyde and nitrogen dioxide… exhibiting decreased concentrations with increased ventilation rate.*
- *In some cases, concentrations of VOCs and formaldehyde increased significantly from the runtime ventilation condition to the continuous exhaust condition in the flip-flop homes.*
    - *It is hypothesized that this may be a result of the exhaust-only ventilation method pulling make-up air through the building envelope and increasing emission rates of any solvents or other volatile chemicals contained in the materials used to construct the envelop…,further data collection and analysis are necessary to…confirm this hypothesis.*

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Lubliner, Michael, Paul Francisco, Eric Martin, Iain Walker, Brennan Less , Robin Vieira, Rick Kunkle, Zachary Merrin, <u>Practical Applications and Case Study of Temperature Smart Ventilation Controls – DRAFT3</u>, ASHRAE Transactions *draft*, May 2016 [likely to be published in Jan. 2017]

Paper presents both simulation and whole-house monitored results studying smart ventilation control. The monitored houses are in cold and marine climates. The marine climate home was a renovated 1640 ft2 two-story building with a 5 ACH50. The cold climate house was a 900 ft2 single-story house on an unfinished basement, 9 ACH50. Simulations were carried out with LBNL's REGCAP and a beta version FSEC's EnergyGauge USA.

Both homes had whole-house exhaust fans installed in the bathrooms. Both homes' ventilation operation alternated weekly between continuous ventilation fan operation and an outdoor temperature controlled smart ventilation controller. The larger, tighter marine climate house's fan operated at 40 CFM when running continuously and when in "smart" operating mode provided 90 CFM when the outdoor temperature was above 57 F. The cold climate house's fan provided 30 CFM when running continuously or 80 CFM above 55 F when using "smart" controls.

The paper estimates the cost of simple outdoor temperature-based ventilation controller to be $80 installed. EnergyGauge simulations project savings of $7 to $23 per year in the monitored houses. The impact of the smart control system on the homes' CO2 level and interior humidity was not as significant as other factors beyond control.

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William J.N. Turner, Jennifer M. Logue, Craig P. Wray, July 2012 <u>Commissioning Residential Ventilation Systems: A Combined Assessment of Energy and Air Quality Potential Values</u> Ernest Orlando Lawrence Berkeley National Laboratory. https://buildings.lbl.gov/sites/all/files/lbnl-5969e.pdf

The paper presents an effort to quantify and monetize the health impacts of poor indoor air quality. The report examines the costs of addressing poor IAQ through insuring that ventilation systems are commissioned to insure they are operating as desired. Costs include Time Dependent Valuation (TDV) of energy costs and Disability Adjusted Life Year (DALY) for health costs.

The paper concludes: *Our results show that health benefits dominate over energy benefits when converted to US dollars using DALY and TDV approaches. This was independent of house size and climate. The potential health impacts were large when ventilation rates were insufficient to dilute the emitted indoor contaminants. Providing minimum airflow rates to comply with ASHRAE Standard 62.2-alone is not a sufficient metric for commissioning whole-house ventilation systems and ideally, decisions about tuning should be made with knowledge on indoor pollutant emission rates, ventilation airflow rates, and outdoor air quality. The metric should be NPV of the combined energy and IAQ benefits to the consumer and commissioning cost decisions should be made relative to that value even if that means ventilating to exceed the ASHRAE 62.2-minimum. Identifying that diagnostics are needed to quantify emission rates will hopefully spur industry to develop an appropriate tool for the commissioning community. Identification of low emission products contained within the home via labeling schemes could be part of the commissioning process. As a consequence of combining energy costs with monetized IAQ costs we now have the beginnings of an approach to optimize the ventilation rates of homes*.

The paper's applicability to Florida's climate is found in the conclusion that there are substantial health benefits from ventilating a house at a minimum of ASHRAE 62.2-2010 levels. What is not examined is the potential for health impacts from injecting hot, humid air into the house, potentially raising the humidity indoors to a level conducive to fungal, mold and bacteria growth. The authors' method of monetizing life and health are debatable, but the conclusions seem clear, ventilation can produce a healthier indoor environment.

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Martin, Eric. January 2014. Impact of Residential Mechanical Ventilation on Energy Cost and Humidity Control. NREL-60675. Golden, CO: National Renewable Energy Laboratory. http://www.fsec.ucf.edu/en/publications/pdf/NREL-60675.pdf.

Paper is twofold; a multi-point review of codes, modeling work, and Building America Teams' experience combined with new simulation work relating to energy and humidity impacts from ASHRAE 62.2-2013 ventilation.

The code review outlines ventilation requirements in International, state-wide, and Canadian jurisdictions. The BA Team review discusses previous BA Team's ventilation system recommendations, which were 62.2-2010 compliant but, were operated to provide approximately one third to one half 62.2. Discussed modeling results compare a home meeting the required ventilation through infiltration to standard and high-performance houses in multiple climates and ventilation systems. Further work looked at the impact of duct system location, and thermostat set-points.

Simulation work used 12 cities in 5 U.S.DOE climate zones including Houston and Orlando. Two building types were modeled to determine the energy and humidity impacts of the ventilation. They feature U.S. DOE Zero Energy Ready Home program compliant construction, two leakage rates, two orientations, three ventilation systems (exhaust fan (EX), Energy

Recovery Ventilator (ERV), and Central Fan Integrated (CFI-uses Air handler fan)), three ventilation rates (100%,75%, and 50% 62.2-2013), and with and without dehumidification to RH <60% . Simulations used EnergyPlus V7.1 (E+) and EnergyGauge USA V3.0.01P (EG)

Total annual operating costs are for the buildings are reported. Results show that the impact of different vent systems are fairly irrelevant in Orlando and Houston, and that ALL simulated results were within $90/ year in operating costs. Supplement dehumidification for the hot, humid climates was projected to be $10-$58/year with a dehumidifier EF of 1.47L/kWh (probably unrealistically low).

RH above 60% was reduced by ERVs by one third to one half compared to CFI and EX (EGUSA). Hours above 60% in Orlando occurred mainly during "floating" (no space conditioning) hours. Mechanical ventilation in a tight house is projected to raise the RH by almost 10% compared to a leakier unventilated house in Orlando. Supplemental dehumidification is also needed in the unventilated house to maintain RH below 60% in Orlando.

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Iain Walker, Max Sherman and Brennan Less, May 2014 Houses are Dumb without Smart Ventilation Lawrence Berkeley National Laboratory. LBNL-6747E https://publications.lbl.gov/islandora/object/ir%3A1005394

Paper is based on California Title 24 concerns, and thus looks at California homes and California time-of-use power rates. The paper discusses smart ventilation controller requirements, practicalities of different control strategies, and show examples of actual controllers. They conclude the technology is absent from the residential market place due to: mechanical ventilation being a fairly new idea, controls add first cost, homeowner unwilling, and existing equipment is not always appropriate. They posit the impact of 62.2-2010 ventilation adds around 10% to HVAC energy use versus a similar house with no ventilation. Paper proposes smart ventilation control based on one or several of: outdoor air quality, outdoor thermal conditions, utility rates, occupancy, exogenous (other) ventilation fans, key contaminates, and infiltration. Sensors available could measure: occupancy, humidity, temperature, or third party signals. Currently indoor pollutant sensors are not appropriate for control of smart ventilation due to high cost in confusion as to the best pollutant to sample.

Paper concludes smart ventilation control can reduce the ventilation related energy use by 40% while maintaining or improving indoor air quality, however this includes time-of-use factors. The existing systems to control ventilation are rudimentary or overly complex, and not really applicable to residential ventilation control. Viability of the technology would be advanced by adjustments to codes and standards crediting smart controllers, better/cheaper sensors, and more software and communications hardware to improve cloud and network communications of smart controllers.

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J. Sonne, Withers, C., Vieira, R. June 2015. Investigation of the Effectiveness and Failure Rates of Whole-House Mechanical Ventilation Systems in Florida. Florida Solar Energy Center FSEC-CR-2002-15 http://www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-2002-15.pdf

Paper reports on a 21 home field study investigating the failure rate and effectiveness of whole-house ventilation systems. Study encompassed a home-owner survey and testing of the ventilation system.

Key testing results:

- Of the 21 houses tested only three were found to have ventilation systems with performance approaching the designed amount of airflow. Of these three two were turned off, meaning only one of the houses of the 21 was delivering the required ventilation rate.
- Of the 21 houses nine had inoperable ventilation systems
- Of the 12 operable systems five were deemed to have significant performance issues.
- *Performance issues were identified including failed controllers and dampers, partially disconnected or crushed ducts, dirty filters, and outdoor air intakes installed directly over the air conditioning condenser unit hot air discharge.*

Key survey results: *When asked if they are satisfied with the overall performance of the ventilation system, 10 of the 21 homeowners answered "yes," two answered "I guess," eight answered "I don't know" or similar and one answered "no".*

Specific code-related recommendations include:

- General labeling of components
- Written summary documents for homeowner
- Some kind of failure alarm
- No filter access that requires ladders to access.
- Reduce code specified house tightness to 7 ACH50 for all of Florida.
- Builder test report for ventilation system.

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Danny Parker et al., September 2016. Flexible Residential Test Facility: Impact of Infiltration and Ventilation on Measured Cooling Season Energy and Moisture Levels FSEC-CR-2038-16 http://www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-2038-16.pdf

The report summarizes the summer of 2012's experimental work in two identical, side-by-side residential buildings on FSEC's Cocoa FL facility. One building was configured as a "leaky" building, with an ACH50 of 8, the other building was "tight" with an ACH50 of 2.2. The tight building had mechanical ventilation which was switched on and off for approximately 2-week periods.

When not ventilating the tight building there was virtually no difference in A/C energy use, and minimal differences in interior RH compared to the leaky building. When the tight building was mechanically ventilated at ASHRAE 62.2-2013 rates there was a significant increase in cooling energy (20-38%) combined with modest increases in interior RH (2%-5%) and dewpoint.

We found that building tightness, mechanical ventilation, and infiltration all operate in concert with the outdoor conditions and indoor moisture generation rates to produce indoor moisture conditions. Sometimes low infiltration lowered indoor moisture levels (during moist/rainy periods) and sometimes high infiltration, whether from a leaky envelope or mechanical venting, was beneficial (such as during periods with "free" cooling or dehumidification due to diurnal *weather patterns). The issue then becomes, on balance, which conditions predominate in a given climate and during which seasons. Also critical is how this interacts with AC operation, which can counteract most moisture variation, even doubling indoor moisture generation rates.*

We saw that mechanical venting operates similarly to natural venting, in that under moist outdoor conditions it leads to higher indoor humidity, but this same effect in Florida's winter would operate in reverse with drier outdoor air. We also saw indication that mechanical venting seems to have a slightly different effect than natural ventilation to a similar rate, although such a hypothesis would need more rigorous experimentation.

Consistent with past findings and simulation estimates, the introduction of mechanical ventilation will generally increase the energy usage of an airtight home and may affect indoor humidity, but this is necessary because of the highly variable and often insufficient air ventilation rate provided by even a fairly leaky home in a hot-humid climate, which can have limited natural driving forces during the cooling season.

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Sonne, J.; Vieira, R. June 2014. A Review of Home Airtightness and Ventilation Approaches for Florida Building Commission Research. FSEC-CR-1977-14. http://www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-1977-14.pdf

This report, written for the Florida Building Commission, *presents a* Florida specific *literature review, examination of experimental data, and calculations of energy impacts of using or not using various types of ventilation systems and presents alternative approaches to achieving acceptable levels of ventilation while avoiding the risks associated with tight home enclosures and potential mechanical system failures.*

The first task comprised a literature review consisting of over 40 sources. The review reports on:

- Measured airtightness data
- Airtightness and whole house ventilation requirement trends
    - Energy use considerations.
    - Moisture considerations
- Ventilation options
- Industry ventilation recommendations
- Ventilation system failure concerns
- Health-based ventilation considerations

The second task presented alternative approaches to providing acceptable levels of ventilation. Specific conclusions are:

- No code requirements for further tightening of buildings beyond the 2012 IECC mandate of 5 ACH50 in Florida.
- There is limited information regarding the health impacts of whole house ventilation.
- System design for Florida should include:
    - Flexible flow rate
    - Efficient fans
    - Positive pressure
    - Air intake properly located

  - Provide dehumidifaction.
- Promote balanced systems.
- Limited field studies have shown significant failure rates of installed ventilation systems.

FSEC unpublished work

FSEC is investigating smart ventilation algorithms designed for humid climates that rely on combination of temperature and dew point. Testing to date has yielded 2 – 12 % savings of cooling energy use by altering the time of venting during the day and seasonally. Unlike some climates with more diurnal swings in temperature or humidity, it is difficult to save a good deal in coastal Florida while trying to meet a daily goal, therefore annual average compliance is targeted.

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Task 4: Based on the literature search, develop draft rules.

Draft rules will be of a form that can fit into the code document. The rules will describe how to treat the proposed home as well as establishing parameters for the standard reference home.

Dehumidifier Draft Rules

The most important criteria for modeling the performance of humidity control devices is the dehumidification setpoint. Based on the research, it appears that a majority of studies focus on maintaining indoor relative humidity to 60% or less. For energy-efficient homes this level will tend to be exceeded during mild weather if there is no dehumidification, and will be exacerbated during many time of the year by mechanical ventilation. The literature provides some insight into the working of common dehumidifiers even though that tended not to be the purpose of the studies. The control and sensors used for many low cost dehumidifiers did not function accurately. This work focuses on rules for modeling and not accuracy of the current devices on the market.

Based on the results of this study it is the recommendation that the proposed design and the standard reference design have the dehumidification setpoint turn on at 60% relative humidity and turn off at 55% relative humidity. This differs from the interim report where the recommendation was 60% relative humidity only and we have highlighted this new criteria in red in the recommended change in Table 2. The dehumidifier would only be modeled in the reference home if a dehumidifier is installed in the proposed home. An alternative is to recommend 65% relative humidity which is still below the threshold of 70% where most materials may start to from mold. Studies show far fewer hours requiring humidity control at 65%. Similar to residential thermostats, the code will have no way of mandating the actual dehumidistat set point used by occupants.

The second key criteria is what to use for the reference house dehumidifier efficiency. The Cadmus group showed that often rated efficiencies are not achieved. Furthermore, standby power of stand-alone dehumidifiers is of concern, with 11 of 21 units they measured drawing standby power of 0.4W to 1.9W. Energy factors are determined under test conditions. ENERGY STAR[1] has a required level of efficiency of >2 L/kWh for units less than 75 pints/day and > 2.8 L/kWh for units with capacity 75 to 185 pints/day. ENERGY STAR.gov indicates that labeled dehumidifiers “that have earned the ENERGY STAR are 15% more efficient than non-certified

[1] https://www.ENERGY STAR.gov/products/appliances/dehumidifiers/key_efficiency_criteria

models."[2]At this time there are 165 products available on the ENERGY STAR list, with 150 of the products falling under 75 pints/day. ENERGY STAR indicates even very damp conditions can be met with dehumidifiers of under 50 pints/day for homes up to 2500 square feet.[3] However, some very large homes are built in Florida so both capacity levels will need to be accounted for. For homes requiring total capacity larger than 75 pints/day, the standard reference design level should be equal to 2.38 L/kWh, which would be the base for claiming the ENERGY STAR level is 15% more efficient. Similarly for equipment less than 75 pints/day, the standard reference should be modeled with 1.7 L/kWh. These values changed slightly from the interim report as the 15% improvement was incorrectly applied. Recommended language also changed slightly from the interim report based on house total capacity so as to reducing 'gaming" the system by installing multiple low capacity systems for the purpose of energy code compliance.

The third criteria are how the heat from the dehumidifier should be handled in the performance simulation program. For the standard reference design where a portable dehumidifier efficiency is being used, the recommendation is that the heat from running the dehumidifier be released into the conditioned space. For the proposed design the heat should be modeled released to the space where it is located.

The fourth criteria are when that equipment runs and the capacity of the equipment. It is recommended that the criteria simply state the capacity shall meet the load during all hours. However, this has some implications. Historically the Florida code and the software used has offered credit for natural ventilation (opening windows) and for whole house fans designed for cooling. When conditions are good for natural ventilation the air change increases in the house. If the house has achieved the code criteria for cross ventilation credit, then the air exchange is even greater. See Appendix A that contains language excerpted from the 2014 FBC Technical Assistance Manual (TAM) for Software Developers. Applying this criteria while dehumidifying greatly increases the dehumidification load. Either new natural ventilation criteria needs to be developed for modeling, or the model for the proposed and reference homes should assume no windows open mode all year if dehumidification is implemented. This criteria should be included in the TAM if the dehumidification recommendations become part of the code.

Some HVAC equipment is designed to reduce humidity load when the system runs through modification of the fan speed or other mechanism. However, often the system is not running during hours of high humidity. Those systems will need to be supplemented by a device that runs on a humidistat in order to invoke the standard reference design to employ a dehumidifier.[4] The point is that to obtain credit for a system that may perform better for humidity control, that humidity control needs to be guaranteed through a device designed to maintain the control at all times throughout the year: during floating hours where there is no heating or cooling and during hours of heating or cooling.

[2] https://www.ENERGY STAR.gov/products/appliances/dehumidifiers
[3] https://www.ENERGY STAR.gov/products/appliances/dehumidifiers/dehumidifier_basics
[4] This is not to imply that all cooling systems should have a dehumidistat. Installing control systems that turn on the sensible cooling system whenever an RH setpoint is exceeded can induce problems during times of very low sensible loads. Take a case where a house relative humidity exceeds the setpoint in a mild time of year where perhaps the outside weather is rainy and 65ºF. The system may cool continuously. Even though it will remove moisture, it will quickly bring the temperature lower and the relative humidity will remain high.

It is recommended that these criteria be included in Table R405.5.2 (see below with additions underlined) that performance code compliance software vendors will then have to implement for homes that have a dehumidifying device. The excepted partial Table R405.5.2 with changes is shown as Table 2.

Table 2 [partial] Table R405.5.2(1)
Specifications for the Standard Reference and Proposed Designs including Dehumidification

Building Component	**Standard Reference Design**	**Proposed Design**
Heating systems	Efficiency: in accordance with prevailing Federal minimum standards. Capacity: sized in accordance with Section R403.6. Fuel type: same as proposed.	As proposed As proposed As proposed
Cooling systems	Fuel type: Electric Capacity: sized in accordance with Section R403.6. Efficiency: in accordance with prevailing Federal minimum standards.	As proposed As Proposed As Proposed
Dehumidification Systems	None, except where dehumidification equipment is specified by the proposed design Fuel Type: Electric Capacity: Sufficient to maintain humidity at setpoint all hours Efficiency: 1.7 Liters/ kWh if proposed total capacity is less than 75 pints/day. 2.38 Liters/kWh if proposed house total capacity is greater than or equal to 75 pints per day. Location: In conditioned space[5][6]	As proposed As proposed Sufficient to maintain humidity at setpoint all hours As proposed As proposed
Service water Heating	Fuel type: Use: same as proposed design. Efficiency: in accordance with prevailing Federal minimum standards.	As proposed gal/day = 30 + (10 x N_{br}) As proposed
Thermal distribution systems	Distribution System Efficiency: 0.88 Duct location: entirely within the building thermal envelope. Air Handler location: entirely within the building thermal envelope. Duct insulation: R6	Thermal distribution system efficiency shall be as tested in accordance with Section 803 of RESNET Standards or as specified in Table R405.5.2(2) if not tested. As proposed As proposed As proposed
Thermostat	Type: Manual, cooling temperature setpoint = 75°F; Heating temperature setpoint = 72°F	Same as standard reference
Dehumidistat	None, except where dehumidification equipment is specified by the proposed design Setpoint turn on = 60% relative humidity Setpoint turn off= 55% relative humidity	Same as standard reference

In addition to Table R405.5.2, a change to the Energy Conservation code should be made to indicate the minimum rated energy requirement level of any dehumidifier installed regardless of

[5] The performance modeling software should apply heat gain from the dehumidifier to the space specified.
[6] The performance modeling software should apply heat gain from the dehumidifier to the space specified.

method of compliance. It is also recommended that the dehumidifier be required to drain to the outside.

New section in energy conservation code:

Dehumidifier (mandatory): If installed a dehumidifier:

R403.# Dehumidifiers (Mandatory): If installed a dehumidifier:

1. Shall be sized in accordance with ACCA Manual S.
2. Shall have a minimum rated efficiency greater than 1.7 Liters/ kWh if the total dehumidifier capacity for the house is less than 75 pints/day and greater than 2.38 Liters/kWh if the total dehumidifier capacity for the house is greater than or equal to 75 pints/day.
3. Shall operate without requiring operation of the cooling system air handler fan.
4. If connected into the return side of the cooling system, shall include a backdraft damper installed in the return air duct between the inlet and outlet of the dehumidifier.
5. Shall be controlled by a dehumidistat that is installed in a location where it is exposed to mixed house air and does not receive undue direct influence from mechanical ventilation air or supply air from the home's cooling or heating system(s).

It is also recommended that the mechanical or residential codes include a section as follows to avoid water damage from the dehumidifier:

[Section #] Dehumidifier drainage. Dehumidifiers shall automatically drain condensate to the outdoors and have a flow switch that shuts off operation when the retaining capacity of the dehumidifier is reached in the event of a clogged drain.

The Florida Building Commission could alternatively reference all or part of the Florida Residential Code Section M1411.3.1, which for convenience is included in the Appendix of this report.

[Section #] Dehumidifier drainage: Drain system shall be in accordance with the applicable provisions of Florida Residential Code Section M1411.3.1

Ventilation Control Rules

The other component of the report was to recommend mechanical ventilation changes. Manufacturers and researchers are exploring methods for controlling ventilation to minimize loads.

The interim report recommended the following criteria for the reference home. Adoption would allow the proposed home to alter controls to save energy and would not penalize balanced or ERV systems. The goal of any change is to create appropriate credit for measures that will reduce moisture issues and/or save energy. The current code provides a standard reference design energy use of the fan but does not actually require outside air to be modeled. In the event a builder installs a whole house mechanical ventilation system there should be outside air brought into both the proposed and standard reference design home to avoid the proposed home being penalized for something done for the mechanical code or for perceived health benefits. This is also important for when dehumidifiers are used in conjunction with mechanical ventilation systems so that the humidity removal from ventilation is accounted for in the reference design. It is recommended that the quantity of air brought into the standard referenced design be the same as the average amount brought into the proposed home. This “average” allows the proposed home to provide smart ventilation control.

There are two more criteria required and determining a rule set for these is not straight forward. When modeling a ventilation system the type of system matters. If an exhaust fan is providing the ventilation then the heat from the fan is exhausted and does not heat up the conditioned space. A supply fan system will heat up the space slightly, increasing the cooling sensible load and slightly reducing the relative humidity. A balanced system uses two fans which can double fan energy use and one of the fans provides internal gains to the conditioned space. There are ERV systems that recover the heat and moisture at some rated effectiveness level, but due to increased pressure drop use more energy than the balanced system. Further complicating the matter is the tendency recently to have hybrid systems that use the mechanical system when the unit calls for heating or cooling but augments with an exhaust or supply fan at other times. Placing the same type of system in the standard reference home as the proposed home might negate the extra effort for air quality a builder puts into the proposed home. The interim report indicated airflow would be modeled as balanced in the reference home. Based on results, that is being changed to the same flow-type of system as the proposed. Results showed savings from balanced to an exhaust or supply only system is enormous because of forced and natural airflow interactions with unbalanced flow.

The only document that tries addressing fan power relative to the system used in the proposed home is the ANSI/RESNET ICC 301-2014 standard. Their method of addressing energy use in the standard home is recommended so as to not penalize a house with an ERV. Proposed changes are given in Table 3 for changes to Table R405.5.2(1) with new language underlined and removed language crossed out.

Table 3. Mechanical Ventilation portion of Table R405.5.2(1)
Specifications for the Standard Reference and Proposed Designs including Ventilation Changes

Building Component	Standard Reference Design	Proposed Design
Mechanical ventilation	None, except where mechanical ventilation is specified by the proposed design, in which case: Type of system modeled: Balanced if balanced or ERV in proposed home, exhaust if exhaust in proposed home, supply if supply system in proposed home Annual vent fan energy use: Where proposed home has a supply or exhaust only system: kWh/yr=0.35*fanCFM*8.76 kWh/y Where proposed home has a balanced system: kWh/yr=0.70 *fanCFM*8.76 kWh/y Where proposed home has predominantly a balanced system with energy recovery: kWh/yr=01.0 *fanCFM*8.76 kWh/y ~~kWh/yr =0.03942xCFA+29.565x(Nbr+1)~~ ~~where:~~ ~~CFA = conditioned floor area~~ ~~Nbr = number of bedrooms~~ Airflow Schedule: Same as proposed average airflow rate but not to exceed requirement of ASHRAE 62.2-2016. Airflow Frequency: Continuous	As proposed As proposed As proposed As proposed As proposed

If these recommendations are incorporated, the code should include all necessary definitions. A definition for residential ventilation balanced airflow will also be required. Suggested language:

[Residential Definition] Balanced mechanical ventilation: A mechanical ventilation system in which the difference between supply and exhaust fan air flows is less than 20% of exhaust air flow.

Another consideration is if a minimum level of ventilation should be required. The Florida Home Builders Association, Florida legislature and Florida Building Commission have weighed in on this question and current code only mandates mechanical ventilation if the home ACH50 is less than 3. The energy code change as written does not address a minimum flow for the proposed home, as the standard home is to be modeled with the *same ventilation rate as the proposed but does not exceed ASHRAE 62.2-2016*. ASHRAE 62.2-2016 formulas for fan ventilation are the same as ASHRAE 62.2-2013. However running a fan intermittingly does not necessarily deliver the same air quality as continuous operation. ASHRAE 62.2-2016 also has Normative Appendix C –Relative Exposure. Relevant exposure equivalency is designed as a numeric metric for air quality based on air flow and is used to determine equivalency for intermittent running ventilation systems. By referencing ASHRAE 62.2-2016 directly in future codes instead of simply the fan formula or tables based on formulas, the Normative Appendix C can be included allowing more flexibility in controlling ventilation. Specific language for such a Florida Mechanical or Residential Code awaits the code modification cycle as more research may lead to refinement in what is best for Florida regarding ventilation rates.

Task 5: Test draft rules in a simulation program.

This task consisted of two simulation efforts. One was to test the recommended dehumidifier rules to determine the impact. The other task was to test the mechanical ventilation strategy.

Dehumidification Results

As indicated in the literature search task and Table 1, there are numerous techniques for controlling humidity. Mathematically evaluating each was beyond the scope of this project. For our analysis we ran four building simulations for Florida cities Miami, Orlando and Jacksonville:

1. Code level house with mechanical ventilation and no dehumidifier
2. Same but with a standard reference design dehumidifier (EF=1.74 liter/kWh)[7]
3. Same but with an Energy Star qualified dehumidifier (EF=2 liter/kWh)
4. Same but with the "Best" available dehumidifier (EF=2.47 liter/kWh)

EnergyGauge USA was used for this task as it already has the ability to add smart ventilation and dehumidification. The homes modeled were 1 story, 40'X50', 2000 ft2, 3-bed, slab-on-grade frame houses with attached garages. The homes featured a R-38 attic, R-13 frame wall, R-0 slab insulation package and 300 ft2 of Low-E windows (U=0.4, SHGC=0.25). HVAC consisted of 2.5 ton SEER 14 HSPF 8.2 HVAC with 100% supply ventilation complying with ASHRAE 62.2-2013. The infiltration was set to 3 ACH50 and duct leakage was fixed at a Qn(out) of 0.04. To eliminate the effects of natural ventilation windows were modeled as closed at all times. Homes had the heating and cooling sized based on Manual J 8th edition. For the initial modeling

[7] This value is slightly higher than the proposed minimum efficiency of 1.70 due to a formula error. This change is insignificant for the purpose of this study.

the dehumidifiers' on and off set-points was 60%RH. In operation dehumidifiers have a deadband around their set-point. To represent this deadband a second simulation was performed using an on set-point of 60% RH and an off set-point of 55% RH.

Table 4 compares the results of the 60% on/off set-point analysis. Table 5 presents the results of the deadband analysis, with a 60% RH on set-point and a 55% RH off set-point.

Table 4. Dehumidification results of code level home with supply ventilation at 62.2-2013 level with dehumidification set point of 60% RH on and 60% RH off.

City	Dehumidifier	TotalCooling	TotalHeating	TotalMV	Total DH	Total HVCD	Total Savings from no DH	Total Savings% from no DH	Dehumidifier Savings from Default	Dehumidifier Savings from Default
		kWh	kWh	kWh	kWh	kWh	kWh	%	kWh	%
Jacksonville										
	None	2925	1788	210	0	4923				
	Default	2968	1747	210	159	5084	-161	-3.27%		
	EnergyStar	2965	1748	210	142	5065	-142	-2.88%	19	0.37%
	Best	2962	1750	210	118	5040	-117	-2.38%	44	0.87%
Orlando										
	None	3742	593	211	0	4546				
	Default	3788	570	210	136	4704	-158	-3.48%		
	EnergyStar	3786	571	210	121	4688	-142	-3.12%	16	0.340%
	Best	3782	572	210	101	4665	-119	-2.62%	39	0.829%
Miami										
	None	5884	94	210	0	6188				
	Default	5897	93	211	28	6229	-41	-0.66%		
	EnergyStar	5896	93	211	25	6225	-37	-0.60%	4	0.064%
	Best	5896	93	211	21	6221	-33	-0.53%	8	0.128%

Table 5. Dehumidification results of code level home with supply ventilation at 62.2-2013 level with dehumidification set point of 60% RH on and 55% RH off.

City	Dehumidifier	TotalCooling	TotalHeating	TotalMV	Total DH	Total HVCD	Total Savings from no DH	Total Savings% from no DH	Dehumidifier Savings from Default	Dehumidifier Savings from Default
		kWh	kWh	kWh	kWh	kWh	kWh	%	kWh	%
Jacksonville										
	None	2925	1788	210	0	4923				
	Default	3016	1747	210	251	5224	-301	-6.11%		
	EnergyStar	3011	1745	210	228	5194	-271	-5.50%	30	0.57%
	Best	3005	1745	210	193	5153	-230	-4.67%	71	1.36%
Orlando										
	None	3742	593	211	0	4546				
	Default	3833	572	210	224	4839	-293	-6.45%		
	EnergyStar	3828	572	210	198	4808	-262	-5.76%	31	0.641%
	Best	3822	575	210	163	4770	-224	-4.93%	69	1.426%
Miami										
	None	5884	94	210	0	6188				
	Default	5909	94	210	50	6263	-75	-1.21%		
	EnergyStar	5909	94	210	46	6259	-71	-1.15%	4	0.064%
	Best	5896	93	211	21	6221	-33	-0.53%	42	0.671%

At a steady 60% RH the default dehumidifier uses 161, 158, 41 kWh in Jacksonville, Orlando, and Miami, respectively. At 60% RH on and 55% RH off setting, energy use increases to 301, 293, 75 kWh in Jacksonville, Orlando, and Miami, respectively. These values represent increased dehumidification of over 80% reflecting the sensitivity to the setpoint.

The default dehumidifier at 60%on/55% off adds 75kWh in Miami to 301 kWh in Jacksonville. In Jacksonville that represents 6% of the no dehumidifier simulated house total HVAC energy use. The dehumidifier energy use in Jacksonville ranges from a high of 251 kWh for the default dehumidifier to a low of 193 for the best available unit. Although that represents a 23% reduction in dehumidification energy use, it represents slightly less than 1.36% of total HVCD energy use. The effect on the code EPI once hot water loads are added would be about 1% for "best" cases run.

Studies referenced in the literature search concluded dehumidification is not needed when the air conditioning system (A/C) is running as the A/C system tends to keep the humidity level below 60%. The Florida code also requires simulating with an air conditioner set point at 75 F which increases the hours the air conditioner is running compared to simulations using a set-point of 78 F (RESNET standard). This 3 F difference reduces excess moisture significantly by increasing A/C runtime and associated dehumidification.

The simulations show that little dehumidification is required in a Florida *code level* home to maintain 60% relative humidity. Dehumidifier operation in Jacksonville accounts for from 4.7% to 6.1% of total heating, cooling, ventilation and dehumidification (HVACD) energy. As the simulation shows climates with higher cooling loads require less dehumidification, a home in Miami adding only 1.2% energy use to their HVACD energy use.

Figures 6, 7 and 8 show the houses' interior relative humidity when no dehumidification is present. A line is shown at 60% relative humidity on each graph. Relative humidity is represented on the Y-axis as a number between 0 and 1. The plots represent an entire year of operation, with the days on the X-axis shown as Julian days (1-365).

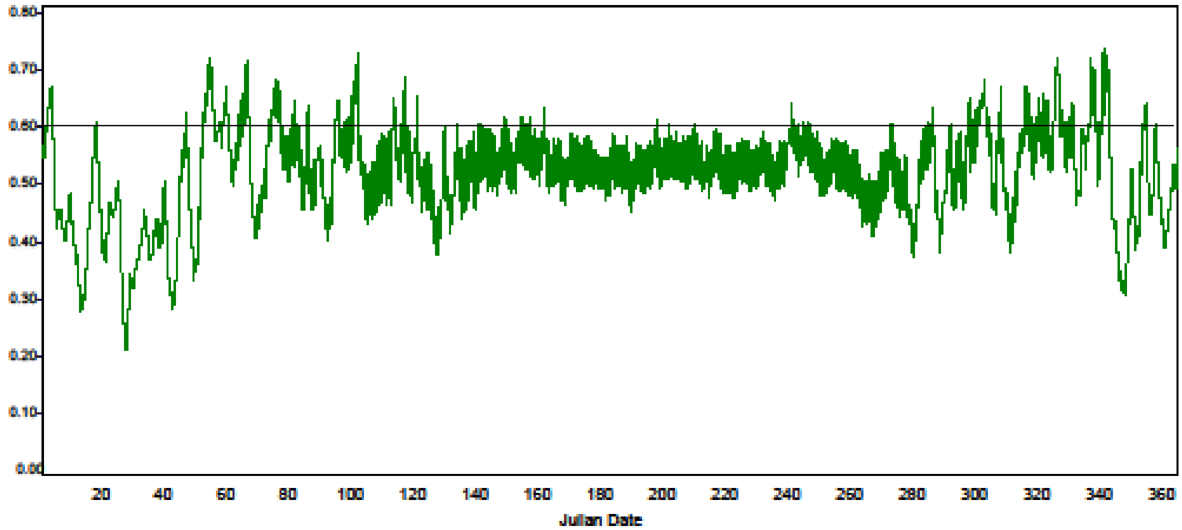

Figure 6. Relative humidity in simulated Jacksonville home with no dehumidifier.

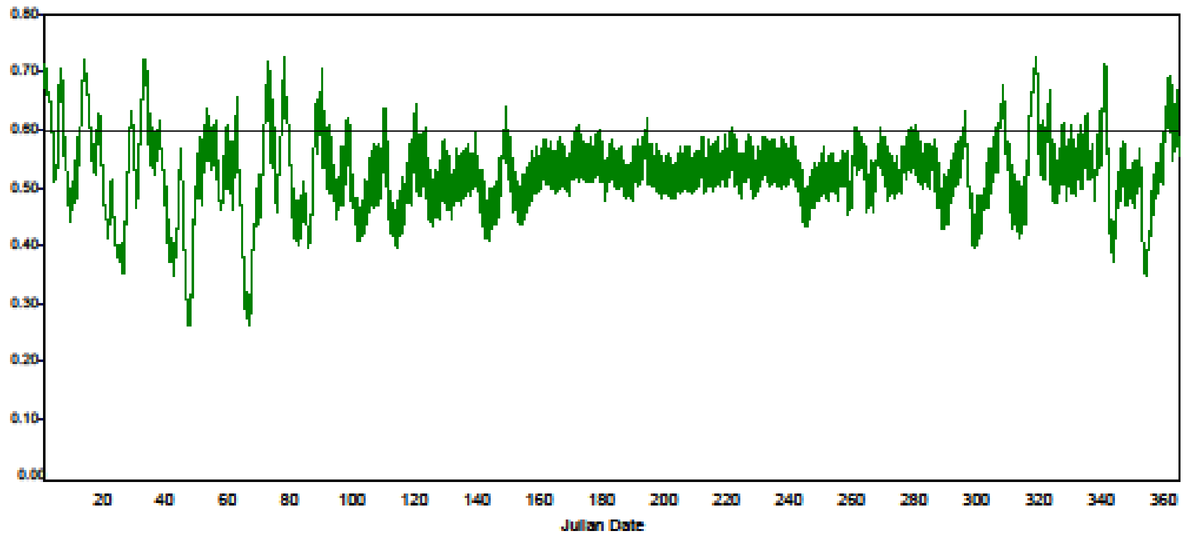

Figure 7. Relative humidity in simulated Orlando home with no dehumidifier.

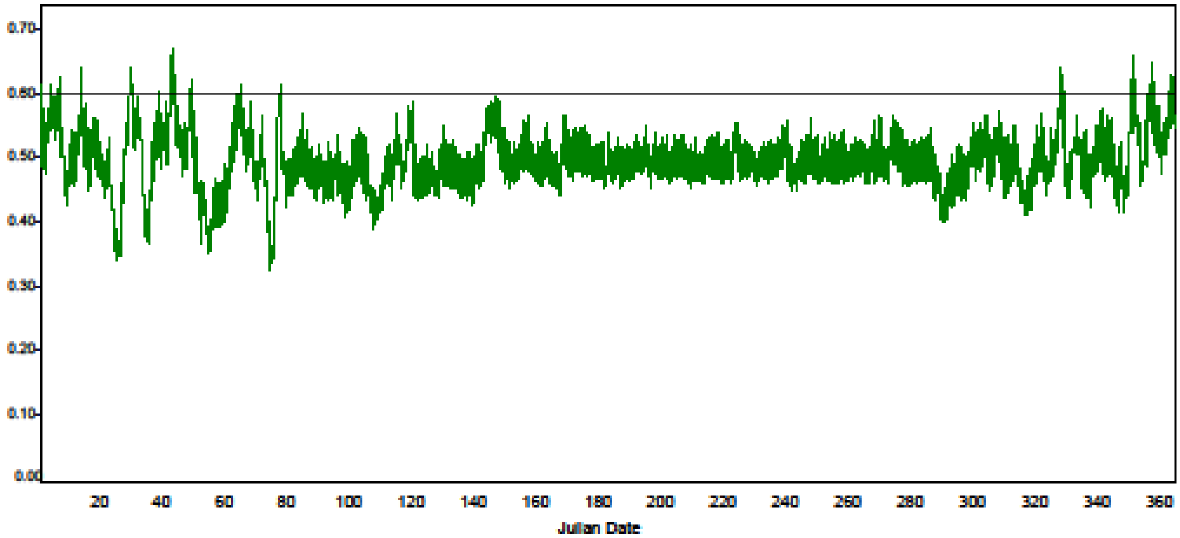

Figure 8. Relative humidity in simulated Miami home with no dehumidifier.

As can be seen from Figure 8 Miami homes with A/C set-points of 75°F rarely exceed 60% RH, and never exceed 70% RH. A/C operation alone is able to meet the dehumidification needs of the house for a majority of the time. Figure 9 contrasts the HVAC operation with the RH in the Miami house. The blue line is the RH (same as Figure 8) and the red line indicates HVAC operation, 0 is off, 1 is cooling and 2 is heating. Note continuous cooling from day 121-305. Although the air conditioner may be running ten or fifteen minutes during an hour, it is still running each hour. This is for a simulation with the setpoint at 75°F and the internal and ventilation loads imposed in the simulation.

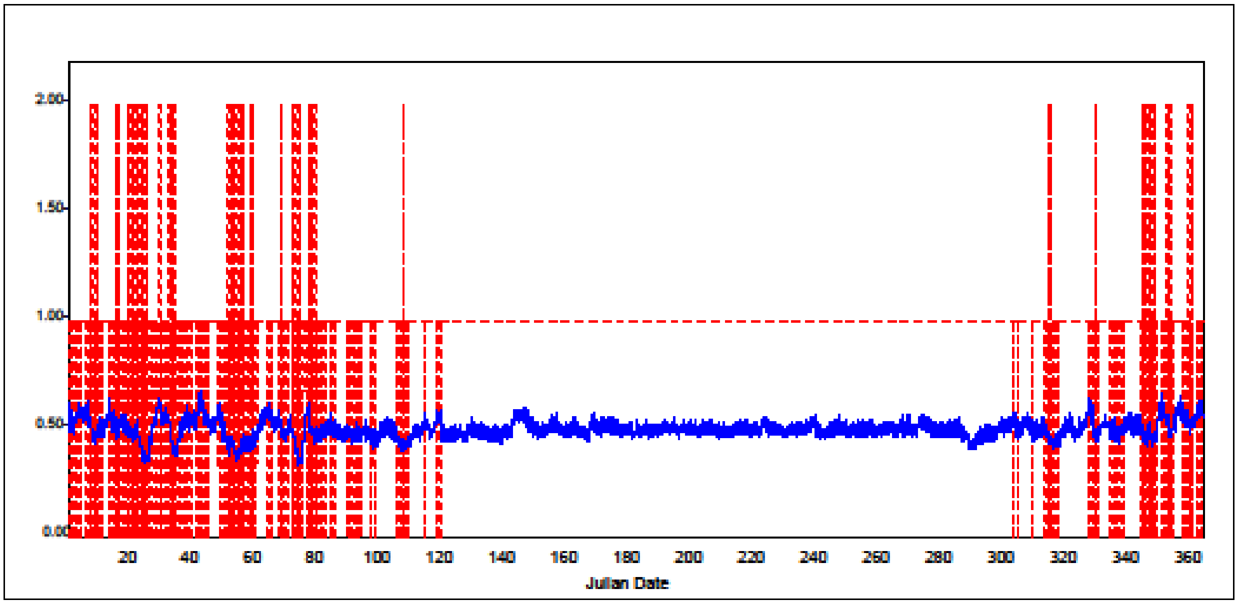

Figure 9. Simulated Miami house with no dehumidifier-RH (Blue) and HVAC operation (0=off, 1=A/C, 2=Heat).

The likely times that dehumidification would be needed are swing season nights when A/C operation is minimal. This conclusion is borne out by the reviewed literature in Task1 regarding hot-humid climates. Figure 10 illustrate the dehumidifier operation in the Miami house on an annual timeline. Supplemental dehumidification (indicated by a value of 1 for the green line) is only needed in late fall and winter, a swing season in Miami, and very seldom then.

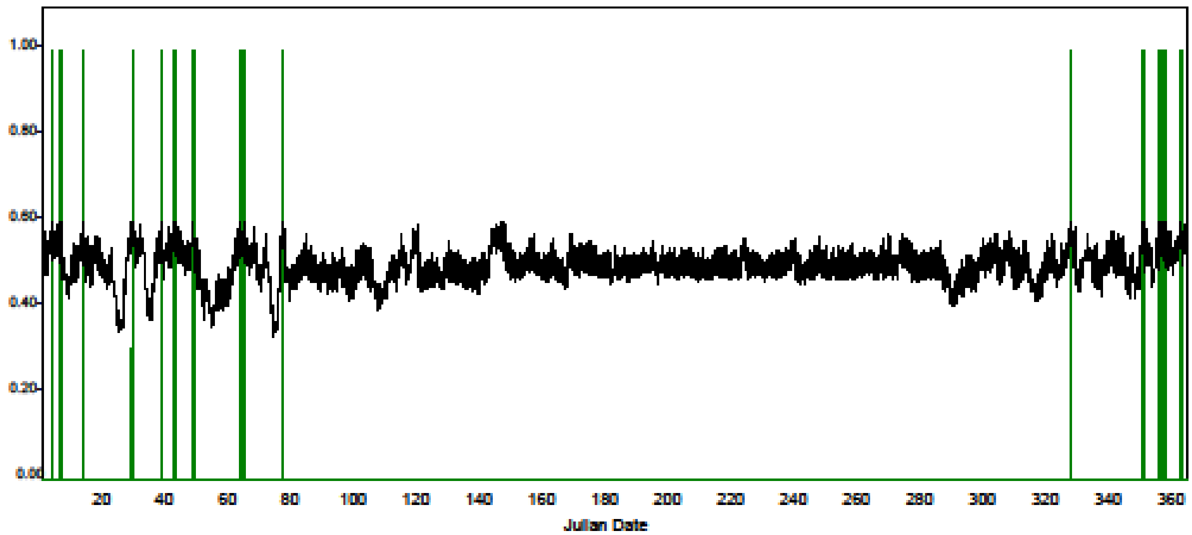

Figure 10. Simulated Miami house with dehumidification. Black=RH, Green indicates dehumidifier operation.

Of concern is the perceived internal heat load resulting from dehumidifier operation. Dehumidifiers generate surplus heat which is dumped into the house when in operation. This heat could add to the heat load the A/C has to remove. Conversely, it can reduce the amount of heat the heating system has to generate.

Figure 11 show the daily sensible and latent loads imposed by the dehumidifier as well as the supply-only mechanical ventilation. The bars on the charts show the sum of the entire year's hourly data represented in Btu/year. As such, at times the ventilation load may be cooling the house during times of non-heating or cooling. So the annual graph does not represent loads actually removed by the air conditioner. The latent and sensible loads from the mechanical ventilation are shown as MVlat (black) and MVsen (green). The surplus heat generated by the dehumidifier is shown as DHsen (blue), and the latent load REMOVED by the dehumidifier is shown as DHlat (red). In all cases the latent load from the mechanical ventilation dwarfs the impact of the ventilation sensible load and the dehumidifier's loads. In these models the humidity is controlled to 60% by a dehumidifier. The majority of the mechanical ventilation latent loads are dealt with by the A/C or room conditions, not the dehumidifier. As can be seen the warmer, wetter climate of Miami imposes a significantly higher mechanical ventilation load then more northern cities.

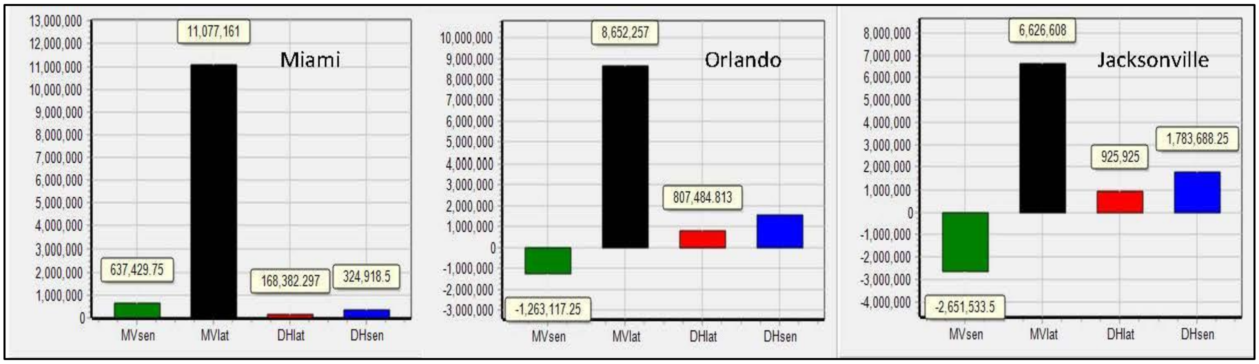

Figure 11 Annual (Btu/year) mechanical venting added latent and sensible loads and dehumidifier removed latent load and added sensible load.

Figure 12 breaks out the data for the month of March and again for the month of July for the simulation which uses typical meteorological year (TMY3) data. In Jacksonville in March the mechanical ventilation cools the home and on average is neutral for latent loads. In Orlando and Miami the mechanical ventilation latent loads are more than 5000 Btu/day average in March while the sensible load on average provides cooling.

Figure 13 shows data points for when the dehumidifier ran during those months. It was off all hours in July in Miami and Orlando while sparsely running in Jacksonville. The dehumidifier tends to run, if at all, in AM hours –from midnight to noon in March in each city and also after 8pm in Orlando and Jacksonville.

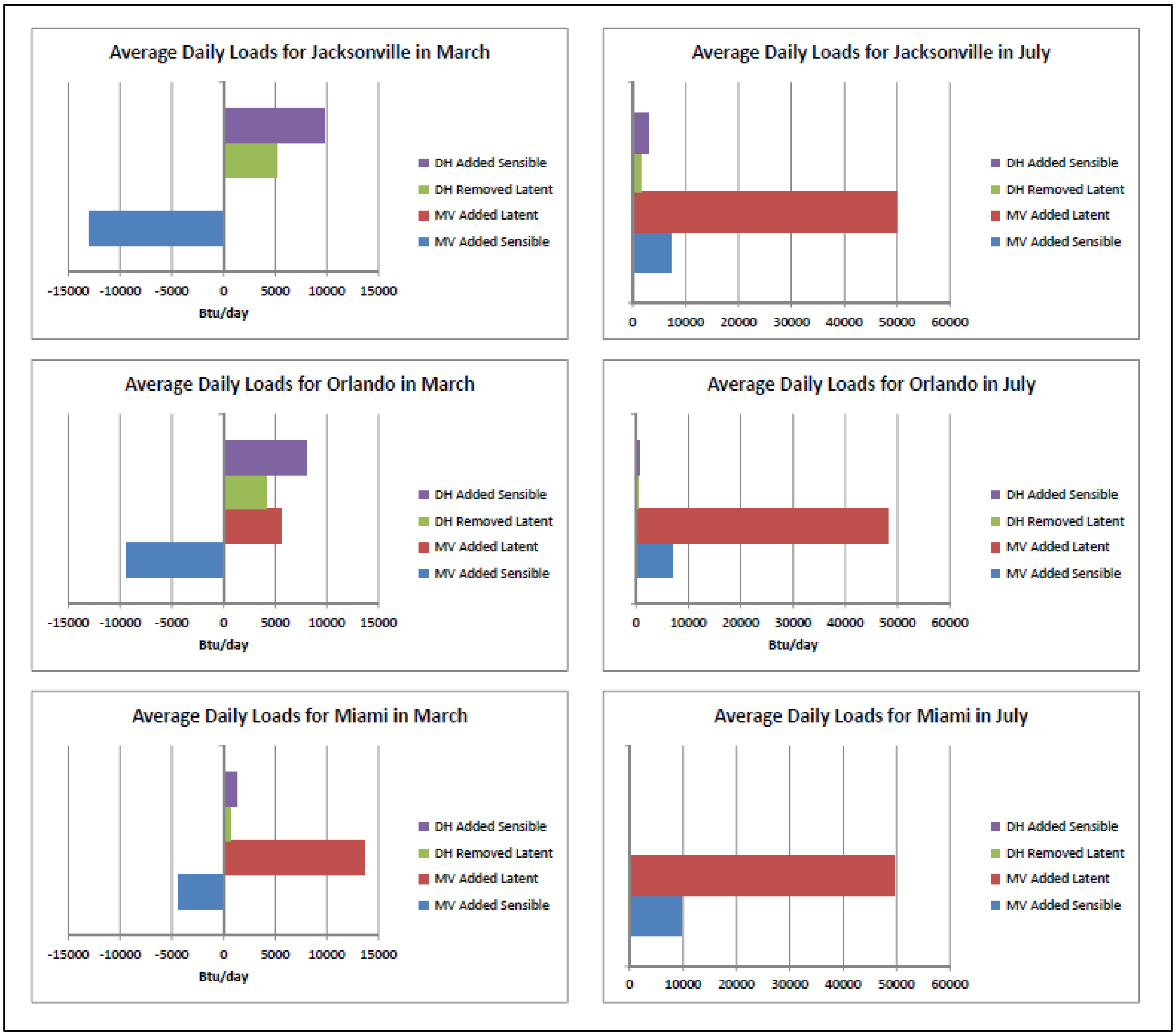

Figure 12. Average daily gains/losses to the house for March and July in each of the three cities for the default dehumidifier and the supply mechanical ventilation system.

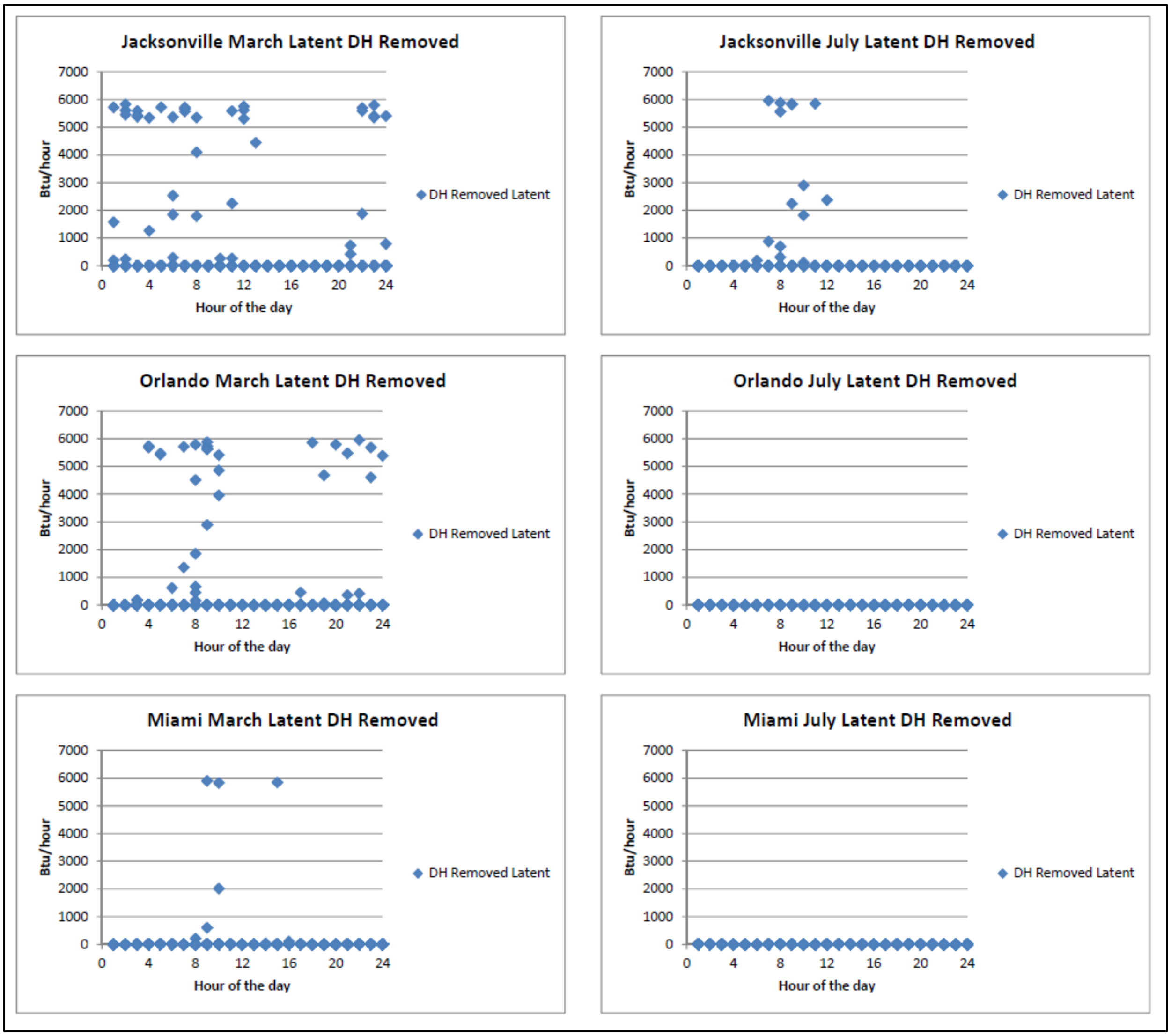

Figure 13. The March and July data points of latent heat removed by the dehumidifier in each of three cities.

Although the dehumidification energy use is small in the simulation it is still important to include a baseline in the code. The cases run do not represent all homes or dehumidification strategies. One of the driving features of this modeling is the Florida code requirement to use a cooling set-point of 75°F. Most of the documents reviewed use a cooling set-point of 78°F. Since the majority of the modeled dehumidification occurs due to A/C operation reducing the cooling set-point increases the runtime of the A/C, resulting in more dehumidification from A/C operation. The ventilation system was modeled with supply only fan ventilation. The small heating load added by the supply ventilation fan motor reduces the relative humidity. Although this is a small effect it is one of many that may alter the dehumidification load.

As homes continue to reduce sensible loads through better insulation and better or fewer windows there will not be similar reductions in latent loads due to internal generation or increased ventilation. As sensible loads are reduced A/C operation will be reduced and dehumidification requirements will increase as the ratio between sensible and latent load increases.

As an example of a home with reduced sensible load, the geometry of an energy-efficient home that was built in Florida was modeled in Orlando. The home is 1290 square feet of conditioned floor area with just 113.7 square feet of 0.22 SHGC window area and three bedrooms. So the internal load to envelope load is much higher than with the code base case. The home has R15 CBS walls and an R21 sealed attic with ducts in the sealed attic space. The house was modeled with 3 ACH50 and ASHRAE 62.2-2013 level of ventilation (55.5 cfm) to be consistent with the previous analysis. The equipment installed in the house was modeled, a 1.5 ton SEER 15.5, HSPF 10 heat pump. Appliances were standard with 75% high efficacy lighting. The modeled house has a Florida e-Ratio of 0.59. Results are shown in Table 6.

Table 6. Dehumidification comparison for the simulated code level home and the low load home in Orlando modeled at 60% on/55% off relative humidity.

Orlando Efficeincy Level	Dehumidifier	TotalCooling	TotalHeating	TotalMV	Total DH	Total HVCD	Total Savings from no DH	Total Savings% from no DH	Dehumidifier Savings from Default	Dehumidifier Savings from Default
		kWh	kWh	kWh	kWh	kWh	kWh	%	kWh	%
Code Leve	None	3742	593	211	0	4546				
Home	Default	3833	572	210	224	4839	-293	-6.45%		
	EnergyStar	3828	572	210	198	4808	-262	-5.76%	31	0.64%
	Best	3822	575	210	163	4770	-224	-4.93%	69	1.43%
Low Load	None	1888	301	211	0	2400				
Home	Default	2084	279	210	515	3088	-688	-28.67%		
	EnergyStar	2072	278	211	458	3019	-619	-25.79%	69	2.23%
	Best	2057	277	210	380	2924	-524	-21.83%	164	5.31%

With this low load home, the default dehumidifier increases total HVCD energy use by over 28%, as it consumes 515 kWh and the heat it generates increases the cooling by 196 kWh while reducing heating by 22 kWh for a total increase of 688 kWh. The "best" dehumidifier reduces these effects slightly and would represent a savings in total HVCD by over 5% relative to the default system.

Integration of concurrent detailed systems operation into existing modeling programs would allow differentiation between systems that might have the same performance at standard reference conditions but considerably different moisture removal capabilities at other operating conditions. The dehumidifier and air conditioner performance need to be modeled as a system. Such modeling will require more inputs on performance of mechanical systems on the part of those completing code forms. The recommended inclusion of dehumidifier base parameters allows for more options for code models to become more sophisticated.

Based on the results the energy and mechanical Technical Advisory Committees might recommend that the dehumidistat set point in simulations be modeled with 60% RH on and 55% RH off when a dehumidification system is included in the proposed home. The other recommendations under the draft rules should be considered as well.

Smart Ventilation Results

For this report eight simulations were run in each of three Florida cities for a 2000 square foot 3 bedroom one story home with 5 ACH50 envelope leakage rate (this is higher infiltration than would be required for mandatory ventilation but many builders use ventilation for other reasons). Tallahassee was substituted for Jacksonville in this analysis as climates with more diurnal temperature swings may prove more advantageous to some strategies. The strategies and (abbreviations) were:

1. Exhaust only system with flow input at ASHRAE 62.2-2013 level and power at 0.35W/cfm (Exhaust)
2. Same as #1 but with a temperature controlled smart ventilation system (Exhaust SVT)
3. A balanced system with supply and exhaust flow each set to ASHRAE 62.2-2013 level and power at 0.70 W/cfm (BAL).
4. Same as #3 but with a temperature controlled smart ventilation system (BAL SVT)
5. An ERV balanced system with supply and exhaust flow each input at ASHRAE 62.2-2016 level and power at 1 W/cfm and ERV effectiveness of 60% (ERV).
6. Same as #5 but with a temperature controlled smart ventilation system (ERV SMT)
7. A runtime ventilation system that uses the central heating/cooling fan with flow set at 4 times the ASHRAE 62.2-2013 level and controlled to provide supply ventilation for 25% of each hour. Extra power for ventilation for this case is the power used each hour between the fraction of the hour the system ran for cooling or heating and the 25% ran for ventilation. Thus power consumption was none when cooing or heating was required 25% or more of the hour, while power consumption was large for times when no heating or cooling was needed (RTV25).
8. Similar to number 7 with runtime flow set to 4 times ASHRAE 62.2-2013 and ventilation time limited to 25% but the runtime system only runs when system calls for heating or cooling. A **backup** exhaust ventilation system meeting ASHRAE 62.2-2013 runs to make up any air needed hour (RTV BU). The purpose of this system is to compare it to system number 7 as well as system number 1.

Although it may be difficult to currently find the smart vent controllers for the SVT systems, they are likely to be available soon. The modeling of the SVT was as follows for Tallahassee and Orlando:

<u>For non-cooling hours,</u> the ventilation system would run at 1.5 times the entered ventilation rate unless the outside temperature dropped to below 65°F at which time it would run at the 62.2-level unless the outside temperature dropped below 45°F at which time it shuts off.

<u>For cooling hours</u> the ventilation system will run at 1.5 times the entered ventilation rate if the temperature is less than 75°F. It will run at the entered ventilation rate between 75 and 80. It will shut off at 80°F.

The control strategy led to similar average annual total flow as the comparative cases without smart ventilation, and did so without extended periods of no ventilation. Specifically the average annual ventilation flow in each of the three cities was slightly greater for the smart ventilation exhaust cases, such that it was slightly conservative in estimated savings. The balanced case smart ventilation ended up with a lower average annual ventilation rate than the comparison balanced average annual flow rate but still was slightly above the required average total air flow of ASHRAE 62.2-2013.

In Miami, to obtain those similar average annual flows as the non-smart ventilation case, the cooling hours control strategy was altered slightly. The ventilation system will run at 1.5 times the entered ventilation rate if the temperature is less than 78°F. It will run at the entered ventilation rate between 78 and 80. It will shut off at 80°F.

The power for the 1.5 times flow used a consistent power/cfm as was entered for the system. The development version of EnergyGauge USA software used for this study has other power options but this one was selected. Flows and power for the simulation runs for the smart ventilation cases are shown in Table 7 and Table 8.

Table 7. Non-Cooling Hour Mechanical Ventilation Fan Flow for Smart Vent Simulations

City	Outdoor Temperature	Fan Flow (cfm)	Power (Watts)
Tallahassee	>65 F	90	31.5/63/90
	>45 - 65 F	60	21/42/60
	<= 45 F	0	0
Orlando	>65 F	87.6	30.7/61.3/87.6
	45 - 65 F	58.4	20.4/40.9/58.4
	<= 45 F	0	0
Miami	>65 F	85	29.8/59.5/85
	>45 - 65 F	56.7	19.85/39.7/56.7
	<= 45 F	0	0

Table 8. Cooling Hours Mechanical Ventilation Fan Flow for Smart Vent Simulations

City	Outdoor Temperature	Fan Flow (cfm)	Exhaust/Balanced/ERV (Watts)
Tallahassee	< 75	90	31.5/63/90
	=>75 but <80 F	60	21/42/60
	=>80	0	0
Orlando	< 75	87.6	30.7/61.3/87.6
	=>75 but <80 F	58.4	20.4/40.9/58.4
	=>80	0	0
Miami	<78	85	29.8/59.5/85
	=>78 but < 80	56.7	19.85/39.7/56.7
	=> 80	0	0

An example of how smart ventilation works is illustrated in Figures 14 and 15. Figure 14 show the baseline balanced airflow case in Tallahassee and Figure 15 illustrates the temperature-based smart control ventilation balanced airflow case. The green line in Figure 15 fluctuates between the three possible fan rates (0, 60, 90) whereas it is a constant 60 cfm in Figure 14. By careful examination of Figure 15 it seems the fan almost never went to high speed in the peak summer months but still ventilated some most days.

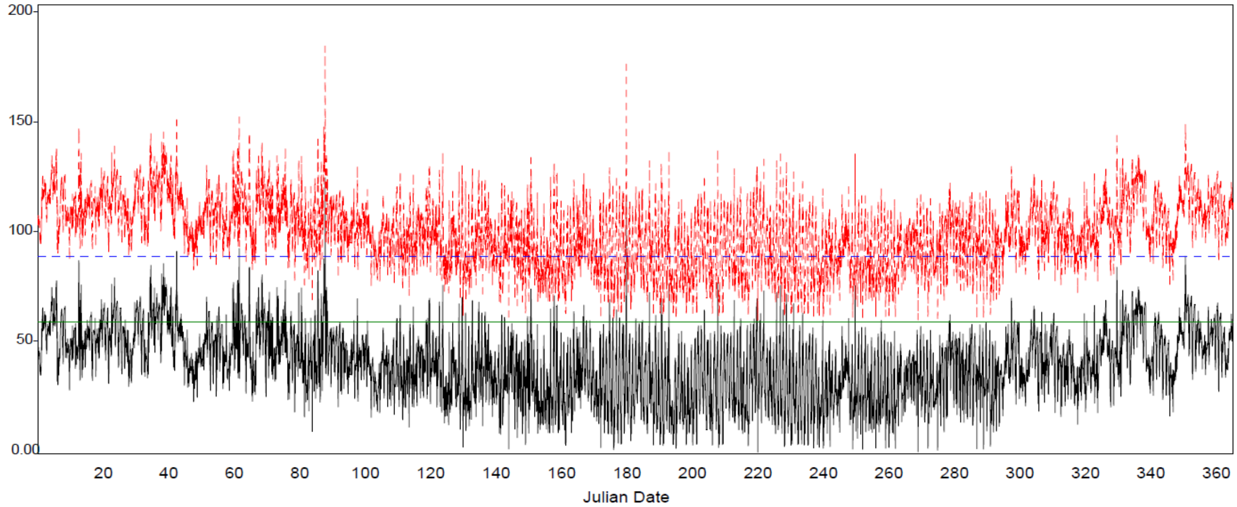

Figure 14. Tallahassee balanced system airflow showing the infiltration in black and the total airflow in red including the constant 60 cfm of mechanical ventilation (green line) designed to achieve the 90 cfm ASHRAE 62.2-2013 goal (blue dashed line).

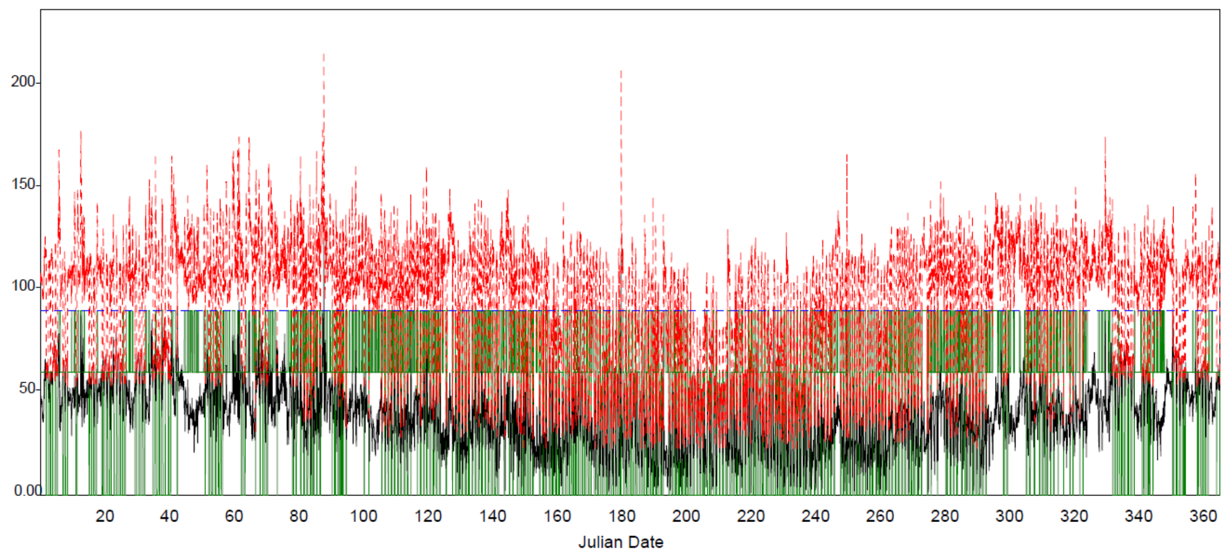

Figure 15. Tallahassee balanced system with outdoor temperature-based smart ventilation showing infiltration airflow in black and the total airflow in red including the varying mechanical ventilation in green. Fan ventilates at 0, 60 or 90 cfm based on outdoor temperature. Goal is to achieve an annual average of 90 cfm per ASHRAE 62.2-2013 total outside air requirements (blue dashed line).

Another measure of ventilation equivalency is given in ASHRAE 62.2-2016 Normative Appendix C –Relative Exposure. Relevant exposure equivalency is designed as a numeric metric for air quality based on air flow and is used to determine equivalency for intermittent running systems. Using the hourly total flow and a time series calculation provided in ASHRAE 62.2-2016, relevant equivalency was determined for each of the smart vent simulations. Figure 16 is an example of how the smart ventilation deployed increases relative exposure during peak summer and winter periods with minor reductions during some of the milder months.

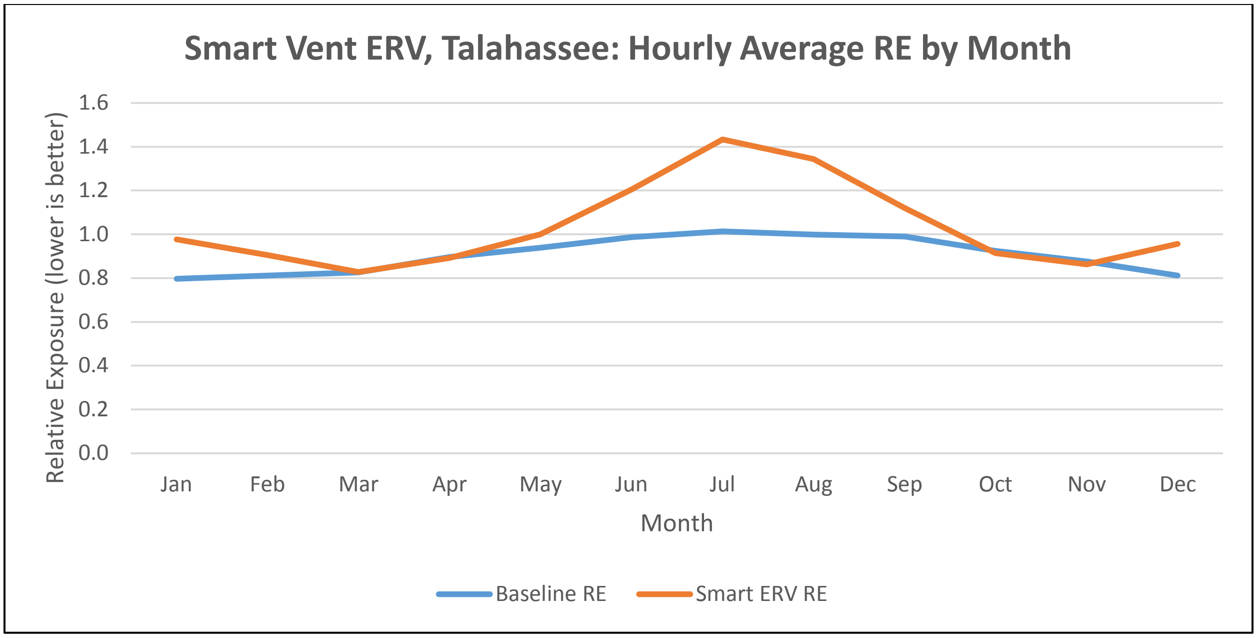

Figure 16. The average annual RE is ideally 1 or less. The smart ventilation strategy employed averaged RE of 1.04 but never went greater than 2.5 for the balanced and ERV ventilation cases (it is recommended all strategies keep the RE below 5 all hours). The highest monthly average from hourly simulation data was 1.43 in July when total airflow averaged 67 cfm. The continuous vented reference case had an average annual RE of 0.91 and a July average of 1.01 when total airflow averaged 90 cfm.

Tables 9 - 11 shows average annual values, including maximums and minimums for a given hour. Overall these control strategies would be deemed acceptable or close to acceptable. The acceptability of any mechanical ventilation system that shuts off is partially dependent on the infiltration level. These houses were run with 5 ACH50 air leakiness level. A lower air leakiness level would necessitate more ventilation hours.

Table 9. Ventilation Flows and Relative Exposure from Tallahassee Hourly Simulations

Tallahassee		Baseline				Smart Vent			
		Natural	Fan	Total CFM	Ri/RE	Natural	Fan	Total CFM	Ri/RE
ERV	min	0.83	60	60.83	0.57	1.33	0.00	22.91	0.50
	max	125.76	60	185.76	1.30	125.73	90.00	215.73	2.50
	average	40.95	60	100.95	0.91	40.98	52.48	93.68	1.04
BAL	min	0.83	60	60.83	0.57	1.33	0.00	22.91	0.50
	max	125.76	60	185.76	1.30	125.73	90.00	215.73	2.50
	average	40.95	60	100.95	0.91	40.95	52.48	93.43	1.04
EXH	min	1.03	60	60.01	0.81	1.33	0.00	22.91	0.72
	max	125.74	60	139.32	1.48	125.73	90.00	154.62	2.52
	average	40.98	60	73.75	1.23	40.98	52.48	74.19	1.26

Table 10. Ventilation Flows and Relative Exposure from Orlando Hourly Simulations

Orlando		Baseline				Smart Vent			
		Natural	Fan	Total CFM	Ri/RE	Natural	Fan	Total CFM	Ri/RE
ERV	min	1.65	58.40	60.05	0.48	1.66	0.00	22.89	0.45
	max	141.53	58.40	199.93	1.31	141.51	87.60	221.48	2.54
	average	41.99	58.40	100.39	0.91	41.99	54.17	96.16	1.02
BAL	min	1.65	58.40	60.05	0.48	1.67	0.00	22.89	0.42
	max	141.47	58.40	199.87	1.31	141.44	87.60	225.90	2.54
	average	41.95	58.40	100.35	0.91	41.95	54.03	95.98	1.02
EXH	min	1.65	58.40	58.42	0.66	1.66	0.00	22.89	0.62
	max	141.53	58.40	153.11	1.50	141.52	87.60	163.72	2.57
	average	42.02	58.40	73.09	1.24	41.98	54.15	74.82	1.26

Table 11. Ventilation Flows and Relative Exposure from Miami Hourly Simulations

Miami		Baseline				Smart Vent			
		Natural	Fan	Total CFM	Ri/RE	Natural	Fan	Total CFM	Ri/RE
ERV	min	0.99	56.70	57.69	0.62	0.95	0.00	22.80	0.53
	max	114.25	56.70	170.95	1.33	114.26	85.05	178.24	2.86
	average	42.96	56.70	99.66	0.92	42.96	47.95	90.91	1.15
BAL	min	0.93	56.70	57.63	0.62	0.84	0.00	22.80	0.53
	max	114.25	56.70	170.95	1.33	114.26	85.05	178.24	2.86
	average	42.94	56.70	99.64	0.92	42.94	47.86	90.79	1.15
EXH	min	0.96	56.70	56.71	0.87	0.89	0.00	22.80	0.75
	max	114.25	56.70	127.55	1.54	114.26	85.05	126.17	2.88
	average	42.96	56.70	72.18	1.26	42.96	47.92	72.71	1.34

Results of the 24 simulations are summarized in Table 12.

Table 12. Ventilation energy use and savings relative to similar system without smart control

City	Vent System	Total Cooling	Total Heating	Total MV	Total DH	Total HVCD	Total Savings	Total Savings
		kWh	kWh	kWh	kWh	kWh	kWh	%
Tallahassee								
	Exhaust	3125	2046	184	151	5506		
	Exhaust_SVT	3032	1911	161	210	5314	192	3.49%
	Balanced	3360	2365	368	238	6331		
	Balanced_SVT	3139	2025	322	301	5787	544	8.59%
	ERV60	3074	2004	526	111	5715		
	ERV60_SVT	2962	1880	462	130	5434	281	4.92%
	RTw25Min25Max	3176	2023	342	141	5682		
	RTVwBU	3125	2046	91	151	5413	269	4.73%
Orlando								
	Exhaust	4145	476	179	100	4900		
	Exhaust_SVT	4036	460	166	145	4807	93	1.90%
	Balanced	4431	576	358	186	5551		
	Balanced_SVT	4190	532	331	245	5298	253	4.56%
	ERV60	4117	461	512	83	5173		
	ERV60_SVT	3989	446	475	95	5005	168	3.25%
	RTw25Min25Max	4191	465	260	94	5010		
	RTVwBU	4145	476	79	100	4800	210	4.19%
Miami								
	Exhaust	6125	73	174	18	6390		
	Exhaust_SVT	5975	73	147	43	6238	152	2.38%
	Balanced	6470	97	348	52	6967		
	Balanced_SVT	6135	96	294	98	6623	344	4.94%
	ERV60	6103	71	496	0	6670		
	ERV60_SVT	5938	70	420	20	6448	222	3.33%
	RTw25Min25Max	6161	70	134	16	6381		
	RTVwBU	6125	73	45	18	6261	120	1.88%

Some observations from the results:

- Savings are largest in Tallahassee and smallest in Miami for smart ventilation as it has a larger effect in winter than summer.
- As this smartvent strategy was outside temperature based, the hours of extra ventilation tend to increase dehumidification.
- Balanced systems use more energy for cooling, heating and ventilation than exhaust only systems. Unbalanced mechanical ventilation is not simply additive with infiltration. Thus, the total air exchange rate for unbalanced ventilation is less than the simple sum of the two separately considered air flows. Exhaust only runs had an average total air flow of 72

- 75 cfm whereas the balanced and ERV runs averaged 90+ cfm even though the fan flow entered for each is set to meet ASHRAE 62.2-2013 fan flow requirement. This is a potential problem in codes and standards that simply have an equation for fan rate without relevance to whether the flow is balanced or unbalanced. Equation C-7 in ASHRAE 62.2-2016 could be used as an alternative.

- Within the eight simulations run in each city, the ERV with smart vent temperature control was always the one using the lowest cooling energy.
- The runtime vent with backup system for code purposes would save 1.7% - 2% compared against the first row in Table 12. The Table shows savings versus the current typical method of runtime vent using the central fan to make up air (RTw25Min25Max). Savings from Runtime Vent with Backup occurs from using less energy for ventilation. That system uses the least energy for ventilation. As the climate switches further South, less energy is used for the extra ventilation as the runtime of the air conditioner increases. In Miami, the model indicated an average runtime for cooling of 34% for the RTw25Min25Max simulation (see Figure 17). The heating and cooling loads are identical to the exhaust only case as the ventilation rate remains the same. Our example used a very high rate of runtime ventilation that may be unrealistic to achieve (e.g., 240 cfm in Tallahassee versus a required continuous rate of 60 cfm) for when the system is on. If for example the runtime vent rate was 60 cfm, the exhaust backup system would have to make up the difference and the savings would be about 25% the value shown.
- Runtime vent with backup uses somewhat more energy for heating and somewhat less energy for cooling than the fixed runtime vent case as the central fan runs less reducing heat load from the fan motor.
- The simple outdoor temperature strategy simulated saves from 1.9% to 8.6% of the total HVCD energy use depending on the city and the ventilation system.
- All savings shown are for systems running at the same efficiency (CFM/Watt) as the reference case so as to show the benefits of the timing of the smart vent strategy without confounding variables. . Greater savings are achievable by using more efficient fans.

The effect on the EPI would be somewhat smaller than the savings shown in Table 12 as hot water is not included in this analysis. However, there are other effects than smart controller that will be impacted by the proposed change. ERVs for instance, would now be compared against an ERV level of mechanical ventilation fan energy (1 W per CFM for the standard reference design) allowing energy-efficient ERV systems to outperform the standard reference house.

This scope of work was not addressing the level of mechanical ventilation required in the proposed home as that falls under the residential and mechanical scope of the codes. The energy criteria were written to handle whatever flows would be decided there by addressing the reference home flow as: *Airflow rate: Same as proposed average airflow rate but not to exceed requirement of ASHRAE 62.2-2016.* This clause could be modified if needed.

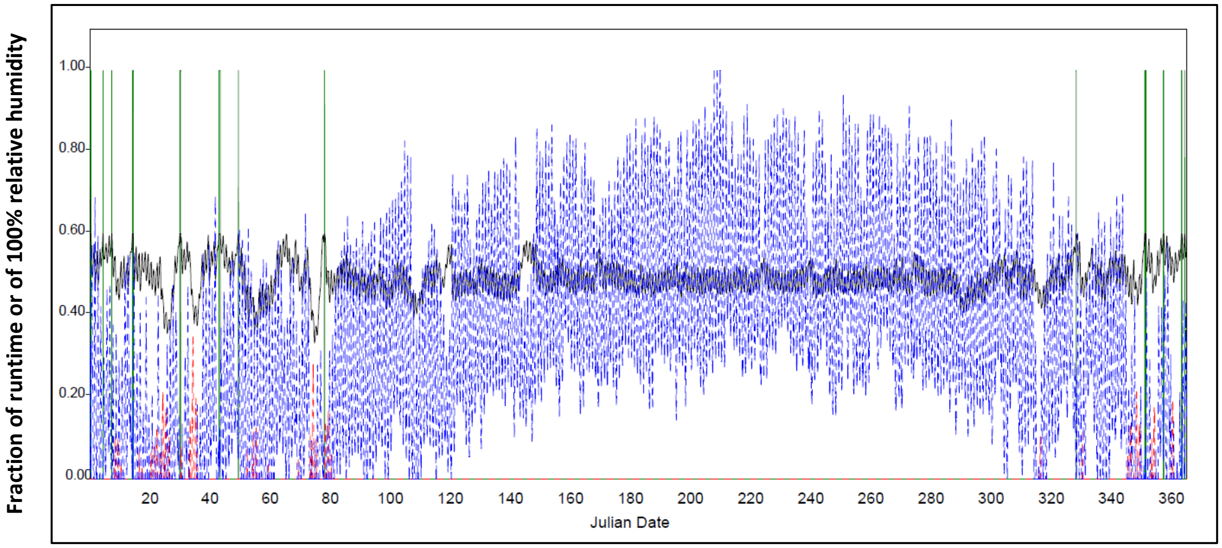

Figure 17. Relative humidity (expressed as a fraction in Black) and part load ratios for heating in Red and cooling in Blue. As the humidity spikes during mild weather the dehumidifier (Green line) comes on –shown as a 1 for any hour when it is on). This graph is for the RTw25Min25Max case in Miami. Average annual value for relative humidity was 49% and the dehumidifier only ran during 51 of the 8760 hours. The annual cooling part load ratio was 34% calculated for all hours including hours when it did not run.

Discussion

This document provides recommended changes to Florida's energy code along with some related changes to other parts of the Building Code for the purpose of creating fair options for builders choosing to install dehumidification or mechanical ventilation equipment. This project presented a handful of results of implementing those options. There is much more information that could be explored for the purpose of determining the impact of such changes as well as obtaining a better understanding of these topics and their interactions for potential code changes:

- What kind of energy savings or penalty occurs from advanced central cooling systems that are designed to dehumidify more effectively?
- Should dehumidifier efficiency be evaluated at conditions other than 80F, 60%, and how should simulation models handle standby energy use of dehumidifiers? What are realistic static pressures of whole house dehumidifiers that are installed in duct systems? Should the dehumidifier appliance rating change to accommodate?
- Where is it best to bring in a supply outside air intake if the cooling system is off?? What if the home has a whole-house dehumidifier?
- Where should a whole house dehumidifier be located –stand alone, on return side of central system, on supply side of central system? If in cooling mode it would make more sense on the supply side but will that hurt the life of the unit? Should the code mandate rules or allow penalties/credits based on location in performance code while limiting options in prescriptive?
- Where should a dehumidistat be located?
- What type of savings will other smart vent options produce? Ones based on dew point as well as dry bulb temperature?
- What are interior latent generation rates and how can we improve modeling to reflect real world?

- What kind of accuracy increase would be achieved by requiring HVAC system mapping (performance at many test points) as inputs to code performance software tools?
- How does moisture and smart ventilation impact results for very low load homes that have larger latent/total cooling ratios?
- What rules should be applied to smart ventilation controls? Does the relative exposure have to be 1.0 or less as recommended in ASHRAE 62.2-2016? Should there be a requirement for minimum airflow per day or per week? Should average annual airflow or relative exposure have to be the same as the baseline? Should fan ventilation rates be the same for unbalanced and balanced flow rates as currently in ASHRAE 62.2-2016 even though balanced flow is estimated to exchange about 30% more total air?

Deliverables Update

Deliverable #1 Interim Report

Completed with submission of February 15, 2017 interim report.

Deliverable #2 Draft of Calculation Procedures (Task 4)

Completed with submission of February 15, 2017 interim report.

Deliverable #3 Draft rules test results

Submitted May 15, 2017 and incorporated and expanded upon in this report

Deliverable #4 Report providing for summary of the literature review, technical information on the problem background, results, final recommendations for code changes, and expected impact for example homes.

Completed with submission of this June 1, 2017 final report.

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Appendix A: Excerpts from 2014 Software Technical Assistance Manual regarding Residential Natural Ventilation

6.5.5.4 Cross Ventilation Option	Normal open window ventilation shall be modeled at 5 air changes per hour, or adjusted based on open area (see Equation 6-1), whenever the following conditions are met: • Outdoor temperature is between 71°F and 75°F • Indoor temperature remains below 75°F Use an algorithm that only allows ventilation to begin after some time period (for example, three hours) after heating or cooling has been called or until the outdoor temperature is reasonably below the cooling set point. If modeling is done in a simple fashion for projects achieving the criteria in the Florida Energy Code, Section R405.7.4, increase the window ventilation from 5 air changes per hour to 7 air changes per hour. The ventilation condition (windows open or closed) shall be set to not change between midnight and 6 a.m. to reflect most typical operating conditions.
6.5.4.4.1 Programs Using DOE2 to ModelCross Ventilation	In DOE2-based software, apply the undocumented method of adding a -4 to the end of the schedule to allow DOE2 to determine typical conditions prior to opening windows: VENTING = SCHEDULE THRU DEC 31 (ALL) (1,24) (-4). **6-1** $FVA = \left(0.25 \times \frac{A_W}{A_{cfa}}\right) \cdot (0.85 \times Discoef)$ **Where:** FVA = the fraction of ventilation area A_w = the sum of all the window areas in the conditioned part of the home A_{cfa} = the sum of all the conditioned areas in the home Discoef = the coefficient of the discharge rate of air, set to 0.60 for standard ventilation, 0.25 and 0.85 are factors for window area open and screens In DOE2 programs, the vent method should be set to use the Sherman and Grimsrud method: (VENT-METHOD= S-G) and the max vent rate should be set to 20 (MAX-VENT-RATE= 20). If other hourly modeling engines are used, they should use the model closet to the DOE2 method described here. When the specified code criteria for cross ventilation credit is met, the software should increase the window ventilation discharge coefficient from 0.6 to 0.75 compared to standard window ventilation.
6.5.4.5 Whole House Fan Option	When the specified code criteria in R405.7.5 for whole house fan is met, either a default of 300W per hour, or a user specified and reported energy use value from the installed whole house fan unit, shall be included in the cooling energy performance when the unit runs. The software shall check to make sure the entered power use and cfm are within the range of current fans available. An air change rate of 20 air changes per hour shall be modeled during times when the whole house fan is operated or a larger value is entered by the user. The operation (on or off) of the unit shall not change from midnight to 6 a.m.

Appendix B. 6th Edition Florida Residential Code Section M1411.3.1

M1411.3 Condensate disposal.
Condensate from all cooling coils or evaporators shall be conveyed from the drain pan outlet to an *approved* place of disposal. Such piping shall maintain a minimum horizontal slope in the direction of discharge of not less than $^1/_8$ unit vertical in 12 units horizontal (1-percent slope). Condensate shall not discharge into a street, alley or other areas where it would cause a nuisance.

M1411.3.1 Auxiliary and secondary drain systems.
In addition to the requirements of Section M1411.3, a secondary drain or auxiliary drain pan shall be required for each cooling or evaporator coil where damage to any building components will occur as a result of overflow from the *equipment* drain pan or stoppage in the condensate drain piping. Such piping shall maintain a minimum horizontal slope in the direction of discharge of not less than $^1/_8$ unit vertical in 12 units horizontal (1-percent slope). Drain piping shall be a minimum of $^3/_4$-inch (19 mm) nominal pipe size. One of the following methods shall be used:

1. An auxiliary drain pan with a separate drain shall be installed under the coils on which condensation will occur. The auxiliary pan drain shall discharge to a conspicuous point of disposal to alert occupants in the event of a stoppage of the primary drain. The pan shall have a minimum depth of 1.5 inches (38 mm), shall not be less than 3 inches (76 mm) larger than the unit or the coil dimensions in width and length and shall be constructed of corrosion-resistant material. Galvanized sheet steel pans shall have a minimum thickness of not less than 0.0236-inch (0.6010 mm) (No. 24 Gage). Nonmetallic pans shall have a minimum thickness of not less than 0.0625 inch (1.6 mm).

2. A separate overflow drain line shall be connected to the drain pan installed with the *equipment*. This overflow drain shall discharge to a conspicuous point of disposal to alert occupants in the event of a stoppage of the primary drain. The overflow drain line shall connect to the drain pan at a higher level than the primary drain connection.

3. An auxiliary drain pan without a separate drain line shall be installed under the coils on which condensation will occur. This pan shall be equipped with a water level detection device conforming to UL 508 that will shut off the *equipment* served prior to overflow of the pan. The pan shall be equipped with a fitting to allow for drainage. The auxiliary drain pan shall be constructed in accordance with Item 1 of this section.

4. A water level detection device conforming to UL 508 shall be installed that will shut off the *equipment* served in the event that the primary drain is blocked. The device shall be installed in the primary drain line, the overflow drain line or the *equipment* supplied drain pan, located at a point higher than the primary drain line connection and below the overflow rim of such pan.

M1411.3.1.1 Water-level monitoring devices.
On down-flow units and all other coils that have no secondary drain or provisions to install a secondary or auxiliary drain pan, a water-level monitoring device shall be installed inside the primary drain pan. This device shall shut off the equipment served in the event that the primary drain becomes restricted. Devices shall not be installed in the drain line.

M1411.3.2 Drain pipe materials and sizes.

Components of the condensate disposal system shall be cast iron, galvanized steel, copper, polybutylene, polyethylene, ABS, CPVC or PVC pipe or tubing. All components shall be selected for the pressure and temperature rating of the installation. Joints and connections shall be made in accordance with the materials specified in Chapter 30. Condensate waste and drain line size shall be not less than $^{3}/_{4}$-inch (19 mm) internal diameter and shall not decrease in size from the drain pan connection to the place of condensate disposal. Where the drain pipes from more than one unit are manifolded together for condensate drainage, the pipe or tubing shall be sized in accordance with an *approved* method.

Appendix C. ENERGY STAR Dehumidifier Specifications Version 4

ENERGY STAR® Program Requirements for Dehumidifiers

Partner Commitments

Following are the terms of the ENERGY STAR Partnership Agreement as it pertains to the manufacture and labeling of ENERGY STAR qualified products. The ENERGY STAR Partner must adhere to the following partner commitments:

Qualifying Products

1. Comply with current ENERGY STAR Eligibility Criteria, which define performance requirements and test procedures for dehumidifiers. A list of eligible products and their corresponding Eligibility Criteria can be found at www.energystar.gov/specifications.
2. **Prior to associating the ENERGY STAR name or mark with any product**, obtain written certification of ENERGY STAR qualification from a Certification Body recognized by EPA for dehumidifiers. As part of this certification process, products must be tested in a laboratory recognized by EPA to perform dehumidifier testing. A list of EPA-recognized laboratories and Certification Bodies can be found at www.energystar.gov/testingandverification.

Using the ENERGY STAR Name and Marks

3. Comply with current ENERGY STAR Identity Guidelines, which define how the ENERGY STAR name and marks may be used. Partner is responsible for adhering to these guidelines and ensuring that its authorized representatives, such as advertising agencies, dealers, and distributors, are also in compliance. The ENERGY STAR Identity Guidelines are available at www.energystar.gov/logouse.
4. Use the ENERGY STAR name and marks only in association with qualified products. Partner may not refer to itself as an ENERGY STAR Partner unless at least one product is qualified and offered for sale in the U.S. and/or ENERGY STAR partner countries.
5. Provide clear and consistent labeling of ENERGY STAR qualified dehumidifiers.

    5.1. The ENERGY STAR mark must be clearly displayed on the top/front of the product, in product literature (i.e., user manuals, spec sheets, etc.), on the product packaging, and on the manufacturer's Internet site where information about ENERGY STAR qualified models is displayed.

Verifying Ongoing Product Qualification

6. Participate in third-party verification testing through a Certification Body recognized by EPA for dehumidifiers, providing full cooperation and timely responses. EPA/DOE may also, at its discretion, conduct tests on products that are referred to as ENERGY STAR qualified. These products may be obtained on the open market, or voluntarily supplied by Partner at the government's request.

Providing Information to EPA

7. Provide unit shipment data or other market indicators to EPA annually to assist with creation of ENERGY STAR market penetration estimates, as follows:

    7.1. Partner must submit the total number of ENERGY STAR qualified dehumidifiers shipped in the calendar year or an equivalent measurement as agreed to in advance by EPA and Partner.

Partner shall exclude shipments to organizations that rebrand and resell the shipments (unaffiliated private labelers).

7.2. Partner must provide unit shipment data segmented by meaningful product characteristics (e.g., type, capacity, presence of additional functions) as prescribed by EPA.

7.3. Partner must submit unit shipment data for each calendar year to EPA or an EPA-authorized third party, preferably in electronic format, no later than March 1 of the following year.

Submitted unit shipment data will be used by EPA only for program evaluation purposes and will be closely controlled. If requested under the Freedom of Information Act (FOIA), EPA will argue that the data is exempt. Any information used will be masked by EPA so as to protect the confidentiality of the Partner.

8. Report to EPA any attempts by recognized laboratories or Certification Bodies (CBs) to influence testing or certification results or to engage in discriminatory practices.
9. Notify EPA of a change in the designated responsible party or contacts within 30 days using the My ENERGY STAR Account tool (MESA) available at www.energystar.gov/mesa.

Performance for Special Distinction

In order to receive additional recognition and/or support from EPA for its efforts within the Partnership, the ENERGY STAR Partner may consider the following voluntary measures, and should keep EPA informed on the progress of these efforts:

- Provide quarterly, written updates to EPA as to the efforts undertaken by Partner to increase availability of ENERGY STAR qualified products, and to promote awareness of ENERGY STAR and its message.
- Consider energy efficiency improvements in company facilities and pursue benchmarking buildings through the ENERGY STAR Buildings program.
- Purchase ENERGY STAR qualified products. Revise the company purchasing or procurement specifications to include ENERGY STAR. Provide procurement officials' contact information to EPA for periodic updates and coordination. Circulate general ENERGY STAR qualified product information to employees for use when purchasing products for their homes.
- Feature the ENERGY STAR mark(s) on Partner website and other promotional materials. If information concerning ENERGY STAR is provided on the Partner website as specified by the ENERGY STAR Web Linking Policy (available in the Partner Resources section of the ENERGY STAR website), EPA may provide links where appropriate to the Partner website.
- Ensure the power management feature is enabled on all ENERGY STAR qualified displays and computers in use in company facilities, particularly upon installation and after service is performed.
- Provide general information about the ENERGY STAR program to employees whose jobs are relevant to the development, marketing, sales, and service of current ENERGY STAR qualified products.
- Provide a simple plan to EPA outlining specific measures Partner plans to undertake beyond the program requirements listed above. By doing so, EPA may be able to coordinate, and communicate Partner's activities, provide an EPA representative, or include news about the event in the ENERGY STAR newsletter, on the ENERGY STAR website, etc. The plan may be as simple as providing a list of planned activities or milestones of which Partner would like EPA to be aware. For example, activities may include: (1) increasing the availability of ENERGY STAR qualified products by converting the entire product line within two years to meet ENERGY STAR guidelines; (2) demonstrating the economic and environmental benefits of energy efficiency through special in-store displays twice a year; (3) providing information to users (via the website and user's manual) about energy-saving features and operating characteristics of ENERGY STAR qualified products; and (4) building awareness of the ENERGY STAR Partnership and brand identity by collaborating with EPA on one print advertorial and one live press event.
- Join EPA's SmartWay Transport Partnership to improve the environmental performance of the company's shipping operations. The SmartWay Transport Partnership works with freight carriers,

ENERGY STAR® Program Requirements
Product Specification for Dehumidifiers

Eligibility Criteria
Version 4.0

Following is the Version 4.0 product specification for ENERGY STAR certified dehumidifiers. A product shall meet all of the identified criteria if it is to earn the ENERGY STAR.

1) **Definitions**: Below are the definitions of the relevant terms in this document.

   A. Dehumidifier: A product, other than a portable air conditioner, room air conditioner, or packaged terminal air conditioner, that is a self-contained, electrically operated, and mechanically encased assembly consisting of: (a) a refrigerated surface (evaporator) that condenses moisture from the atmosphere; (b) a refrigerating system, including an electric motor; (c) an air-circulating fan; and (d) means for collecting or disposing of the condensate[1].

      a. Stand Alone: Portable unit designed to provide dehumidification within the confined living space where it is placed and plugged into an electrical outlet.

      b. Whole House: Unit designed to be incorporated into the home's HVAC system, or installed with its own duct system, providing dehumidification for all conditioned spaces within the building enclosure.

   B. Capacity[2]: A measure of the ability of a dehumidifier to remove moisture from its surrounding atmosphere, measured in pints collected per 24 hours of continuous operation. Capacity shall be measured according to the test standard referenced in Section 4, below.

   C. Energy Factor (EF)[2]: A measure of energy efficiency of a dehumidifier calculated by dividing the water removed from the air by the energy consumed, measured in liters per kilowatt hour (L/kWh).EF shall be calculated according to the test standard referenced in Section 4, below.

   D. Basic Model Group[1]: All units of a given type of product (or class thereof) manufactured by one manufacturer, having the same primary energy source, and which have essentially identical electrical, physical, and functional (or hydraulic) characteristics that affect energy consumption, energy efficiency, water consumption, or water efficiency.

2) **Scope**:

   A. Included Products: Products that meet the definition of a dehumidifier as specified herein are eligible for ENERGY STAR qualification, with the exception of products listed in Section 2.B. Stand alone and whole house units with capacities measuring less than or equal to 185 U.S. pints (87.5 liters) are eligible for ENERGY STAR.

   B. Excluded Products: Dehumidifiers with daily water-removal capacities greater than 185 U.S. pints (87.5 liters) are not eligible for ENERGY STAR.

3) **Qualification Criteria**:

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[1] 10 CFR Part 430, Subpart A
[2] 10 CFR Part 430, Subpart B, Appendix X

A. Energy Efficiency Requirements: To qualify for ENERGY STAR, dehumidifiers shall meet the EF requirements provided in Table 1, below.

Table 1: Performance Criteria for ENERGY STAR Certified Dehumidifiers

Product Capacity (Pints/Day)	**Energy Factor Under Test Conditions (L/kWh)**
< 75	≥ 2.00
75 ≤ 185	≥ 2.80

B. Other Requirements:

Qualifying units shall be equipped with an adjustable humidistat control or shall require a remote humidistat control to operate.

C. Significant Digits and Rounding:

a. All calculations shall be carried out with directly measured (unrounded) values.

b. Unless otherwise specified in this specification, compliance with specification limits shall be evaluated using directly measured or calculated values without any benefit from rounding.

c. Directly measured or calculated values that are submitted for reporting on the ENERGY STAR website shall be rounded to the nearest significant digit as expressed in the corresponding specification limit.

4) Test Requirements:

A. One of the following sampling plans shall be used for purposes of testing for ENERGY STAR certification:

1) A single unit is selected, obtained, and tested. The measured performance of this unit and of each subsequent unit manufactured must be equal to or better than the ENERGY STAR specification requirements. Results of the tested unit may be used to certify additional individual model variations within a basic model group as long as the definition for basic model group provided in Section 1, above, is met; or

2) Units are selected for testing and results calculated according to the sampling requirements defined in 10 CFR Part 429, Subpart B § 429.36. The certified rating must be equal to or better than the ENERGY STAR specification requirements. Results of the tested unit may be used to certify additional model variations within a basic model group as long as the definition for basic model group provided in Section 1, above, is met. Further, all individual models within a basic model group must have the same certified rating based on the applicable sampling criteria this rating must be used for all manufacturer literature, the qualified product list, and certification of compliance to DOE standards.

B. When testing dehumidifiers, the following test methods shall be used to determine ENERGY STAR qualification:

Table 2: Test Methods for ENERGY STAR Qualification

ENERGY STAR Requirement	**Test Method Reference**
Capacity and Energy Factor	10 CFR Appendix X to Subpart B of Part 430

5) **Effective Date**: This ENERGY STAR Dehumidifier Specification shall take effect on **October 25, 2016**. To qualify for ENERGY STAR, a product model shall meet the ENERGY STAR specification in effect on the date of manufacture. The date of manufacture is specific to each unit and is the date on which a unit is considered to be completely assembled.

6) **Future Specification Revisions**: EPA reserves the right to change the specification should technological and/or market changes affect its usefulness to consumers, industry, or the environment. In keeping with current policy, revisions to the specification are arrived at through industry discussions. In the event of a specification revision, please note that the ENERGY STAR qualification is not automatically granted for the life of a product model.