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# Assessment of Energy Efficient Methods of Indoor Humidity Control for Florida Building Commission Research

Florida Solar Energy Center

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# Final Report

## Assessment of Energy Efficient Methods of Indoor Humidity Control for Florida Building Commission Research

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# Executive Summary

The Florida 5$^{th}$ Edition (2014) Code will require that houses be tested for envelope air leakage and will not permit leakage in excess of 5 ACH50. At the same time, the new International Mechanical Code requires that mechanical ventilation, also known as outdoor air (OA), be provided for any house that has less than 5 ACH50. Given that natural forces that drive natural infiltration in Florida buildings are relatively small, there is a reasonable expectation that new mechanically vented Florida homes will experience greater air change rates and increased latent loads. Of further consideration is the trend for homes to be built with more energy efficiency and thus lower sensible loads resulting in less runtime of central cooling systems, particularly during swing seasons. With increased latent load from required OA, there is the concern of adequate indoor relative humidity (RH) control for at least periods of time. Supplemental dehumidification may therefore be necessary in some homes where RH is desired to be controlled.

The Florida Department of Business and Professional Regulation (DBPR) has contracted with the Florida Solar Energy Center (FSEC) with the primary objective to identify approaches and technologies which can achieve energy-efficient latent (indoor RH) control in light of requirements that may increase overall ventilation rates in Florida homes. The work has been proposed to be completed in two phases. This report covers the work proposed in the first phase to be completed June 15, 2014.
The primary scope of work of phase 1 consists of two tasks:

1. Assess energy efficiency and cost-effectiveness of various approaches to manage the latent load in homes from a literature review and existing experimental data.
2. Complete a minimum of four different experiments in a lab building to assess resulting indoor RH and energy consumption. The experiments are to use various latent load management approaches (including a dehumidifier) at various levels of mechanical ventilation.

Review of the most recent existing literature on indoor RH control in homes in hot and humid climates has identified several supplemental dehumidification options, but there are several factors that have to be considered in selecting one, therefore no single measure can simply standout. Factors to be considered are:

- Health needs of occupants
- House design
    - Variability of solar exposure throughout the day and year
    - Availability of physical space to locate supplemental dehumidification equipment
    - Type of construction and interior finishes- Homes with minimal internal moisture capacitance will have faster rates of change in interior RH (finished concrete or tile floors and minimal soft cloth-type furniture and furnishings).
- Occupant driven operations
    - Indoor temperature set point preferences -Low cooling set point can result in high wall surface humidity, under certain conditions, even if occupant is comfortable.
    - Excessive runtime of bath or large kitchen exhaust system
    - Large internal latent load generation
- Personal comfort / ascetic preferences
    - No supplemental dehumidification may be desired or needed if house interior is kept relatively warm and there is a preference for elevated humidity.
    - Preference for equipment to be unseen and unheard may make some options less desirable.
- First cost and operational cost

Cost is a significant factor and varies widely. The lowest first cost options, overcooling and reduced fan speed will not adequately control RH for all hours of the year. Dehumidifiers may have the lowest first cost of options that can control RH year round, but are least energy efficient of all other mechanical measures. Full-condensing and sub-cooling reheat integrated with the central cooling system will operate more efficiently than dehumidifiers at an estimated cost just under $2000. Another potentially effective technology near the $2000 range is the regenerated desiccant dehumidifier, which can use waste heat from any source such as the DX cooling system, or even renewable energy resources. What is lacking is available monitored performance data from the full-condensing, sub-cooling and desiccant dehumidifier systems in hot humid climates.

Given the costs, one must seriously consider how important it is to maintain humidity within a specific range. An ideal specific level of RH control is debatable and specific needs of occupants must be considered given variability in personal comfort preferences and health considerations. Generally, RH maintained at or below 60%RH is considered reasonable for Florida. Consideration of specific supplemental dehumidification equipment options shown in this report should also consider the differences in climate within Florida. For instance, Miami has much more annual cooling driven hours and shorter swing season than Tallahassee, therefore sensible driven supplemental dehumidification options will offer better RH control than in northern Florida.

Experimental work completed, fulfilling task 2 of this project, evaluated the resulting indoor RH and energy consumption of four different test configurations. The configurations focused on two potentially energy-efficient configurations at two different ventilation rates with a dehumidifier controlled by a remote dehumidistat set at 60%RH. The four different test configurations were:

- Test 1: involved 130 cfm OA ducted to the return of a SEER 19 variable capacity minisplit (MS) with central space cooling provided by SEER 21 variable capacity central ducted heat pump.
- Test 2: same as Test 1 except with OA at 60 cfm.
- Test 3: Turn off the MS and duct 60 cfm OA to the return of a high efficiency (SEER 21) variable capacity system.
- Test 4: same as the Test 3 except with OA at 130 cfm.

All four test methods were able to maintain indoor RH in the range from 47%-52%RH with outdoor dewpoints near 70°F, therefore the dehumidifier never operated. This experimental research has found promising results for using very high efficiency (SEER 21) variable capacity central ducted heat pumps with mechanical ventilation ducted to the return intake. The SEER21 variable capacity central system offers an energy-efficient method of combined space cooling and good RH control in mechanically ventilated buildings. At a high OA flow rate of 130 cfm (about twice the rate for typical homes) captured at the return of the central unit, the highest hourly indoor RH reading on a warm moist day was 54.0%. This is not to say that there would never be any hours with indoor RH>60% throughout the year, but they are expected to be few given that these systems can operate as low as 40% of nominal capacity. This method used about 14% less seasonal daily energy than the method where OA is captured at the MS return and the SEER21 central system provides cooling as needed.

Work in this first phase had only enough time to evaluate two different methods at two different ventilation rates. The weather during the experiments offered very limited days with elevated dewpoints to work with. A second phase already proposed is very important to evaluate more latent load approaches and collect more data for experiments during more variable weather. Four additional test configurations are discussed in more detail in the Conclusion section of this report.

# Introduction

The new Florida 5th Edition (2014) Code will require that houses be tested for envelope air leakage and will not permit leakage in excess of 5 ACH50. At the same time, the new International Mechanical Code requires that mechanical ventilation be provided for any house that has less than 5 ACH50. This practically means that new Florida houses will be required to have mechanical ventilation. Because infiltration forces such as wind and temperature are substantially lower in Florida than in other parts of the country, natural infiltration often does not provide the required air change rate. Mechanical ventilation reintroduces a considerable portion of the outdoors-to-indoors air exchange that was eliminated by house tightening. Florida code ventilation requirements follow the 2012 International Residential Code (IRC) or 2012 International Mechanical Code (IMC). The differences between these two are discussed further in the Background section of this report.

Given that Florida home natural ventilation rates are much lower than required mechanical ventilation, there is an expected overall increase in latent load from ventilation. With an increase in latent load and the variability in sensible loads throughout the cooling season that primarily drive cooling and dehumidification, it is of concern that some air conditioning systems may not adequately control indoor relative humidity (RH) with continuous mechanical ventilation.

The primary objective of this work is to identify approaches and technologies which can achieve energy-efficient latent control in light of requirements that may increase overall ventilation rates in Florida homes. The work has been proposed to be completed in two phases. This report covers the work proposed in the first phase to be completed by June 15, 2014.

The primary scope of work of phase 1 is summarized below in the following two tasks:

1) Assess energy efficiency and cost-effectiveness of various approaches to managing the latent load in homes from a literature review and existing experimental data.
2) Complete a minimum of four different experiments in a lab building to assess resulting indoor RH and energy consumption. The experiments are to use various latent load management approaches (including a dehumidifier) at various levels of mechanical ventilation.

The second phase is proposed to continue the experimental work of task 2 above evaluating more latent load approaches and collecting more data for experiments during higher outdoor dewpoint periods. Good RH control becomes more challenging during the cooling season on days when the outdoor dewpoint is greater than 65° F. Typical summer outdoor dewpoints run between 72°F-76° F, but outdoor latent load is just beginning to increase May through early June with lower dewpoints in the range 60°F-70°F. It is imperative that the second phase be completed to provide more expansive and meaningful results through November. This would provide much needed testing under the challenging outdoor conditions of lower outdoor sensible, but high latent loads such as a cooler day with outdoor dewpoint of 72°F.

# Background

The thermal efficiency of homes has improved in recent decades. Most of those improvements have reduced the amount of sensible heat load on the house (sensible heat is associated with temperature rise, while latent heat is associated with energy embodied in water vapor in the air). Wall and attic insulation, improved duct insulation, and improved windows have reduced sensible loads on homes, which have brought about smaller air conditioning systems and caused these systems to operate less. Reduced operation can lead to a reduction in water vapor removal, and the potential for an increase in indoor humidity. While external sensible loads have been reduced, there has been a growing trend of increased internal sensible loads over time. More electronic devices are operating in homes producing additional internal sensible heat gains.

Another trend is that natural air infiltration has declined in recent decades. House air tightness is measured as an air flow rate at 50 pascals of pressure. The result is normalized by dividing by the house volume and is represented as air changes per hour at 50 pascals or ACH50. The work of (McIlvaine et al. 2013) show a trend of increased tightening in each of the decades from the 1960s to 2000s. House air tightness has declined from an average 18.2 ACH50 (n=12) in the 1960s to an average 6.3 ACH50 (n=7) in homes built in the first decade of 2000 (McIlvaine et al. 2013). A study of house air tightness of homes built under the Florida 2009 code found homes about 11% tighter than the 2000 decade homes of the McIlvaine study with an average of 5.6 ACH50 (n=31) (Withers et al. 2012).

Since about 85% of the cooling load associated with air entering from outdoors during hot and humid weather is latent heat (water vapor) and only about 15% is sensible heat, the tightening of homes has greatly reduced the amount of water vapor that must be removed by the air conditioning system. Tightening of duct systems has also substantially reduced the amount of water vapor entering the house.

However, codes that require very tight homes need to address IAQ. Thus the need arose to provide treated OA to homes by means of mechanical ventilation. Nationally ASHRAE Standard 62.2 is considered by some as an indoor air quality standard for residential ventilation, however Florida code (2012 International Energy Conservation Code) specifically references that ventilation meet the requirements of the International Residential Code (IRC) or International Mechanical Code (IMC). The IMC 2012 simply applies 15cfm to each bedroom plus another 15 cfm. The IRC 2012 requirements are indicated in Table M1507.3.3(1) and have a prescribed ventilation rate based upon different ranges of house area and number of bedrooms. They both have the exact same 60 cfm requirement for 3 bedroom homes between 1501ft$^2$-3000 ft$^2$ (which covers most homes), however there is variance in requirements between the two depending upon house size and number of bedrooms. Take for example a 1510 ft$^2$ retirement home with 2 bedrooms. IMC 2012 would require 45cfm OA whereas IRC 2012 requires 60cfm. Calculating OA using ASHRAE 62.2 is more involved requiring inputs such as airtightness test data, floor area, number of bedrooms, and height of ceiling. A 2000 ft$^2$ home with 3 bedrooms and airtightness of about 5 ACH50 would require 60 cfm. However, the mechanical ventilation requirement increases as the house tightness increases assuming less natural ventilation occurs.

Of particular concern is how to maintain acceptable RH when latent load from OA becomes a greater fraction of the overall load at times such as overnight or long periods of low heat gain such as swing seasons.  Even without OA, there are several hours of the year when a “properly sized” cooling system will be oversized and have limited runtime. The most common way consumers have traditionally tried to address high indoor humidity is to either lower the cooling set-point of the central air or operate a dehumidifier as needed, neither of which is energy efficient and depending upon the equipment still may not result in satisfactory comfort.

The internal latent gains from occupancy driven activity is another important load on the cooling system that must be addressed effectively to maintain acceptable indoor RH.  Key internal moisture sources come from cooking, bathing, and dishwashing as well as from occupant perspiration and respiration. Work has been done to characterize these types of loads in homes (Hendron 2010). There can be enough moisture from these sources to result in elevated indoor RH during mild swing seasons where cooling loads may be low.

The combination of Florida homes becoming more energy-efficient and airtight together with the advent of mandatory mechanical ventilation will create new moisture related challenges for the state.  A 2007 Building Science Corporation (BSC) study involving 43 warm-humid and mixed-humid climate US homes (Rudd and Henderson 2007) summarizes these challenges.  The study found that adding continuous mechanical ventilation to standard builder practice homes did not consistently raise indoor humidity but did in high-performance (low sensible gain) Building America program homes.  The BSC report concludes:

*The combination of high-performance, low sensible heat gain buildings and continuous mechanical ventilation has significantly increased the number of hours in the year that require dehumidification without sensible cooling. Humidity loads in these high-performance homes cannot always be met by conventional or enhanced cooling systems, but instead require separate dehumidification. These load conditions— which have not typically been observed in standard homes— do not occur during peak summer cooling conditions but mostly occur in the swing seasons and may occur during summer nights.*

## Task 1: Literature Review, Examination of Data and Cost-Effectiveness

Task 1 of this report involves a literature review, examination of existing experimental data and assessment of the energy efficiency and cost-effectiveness of various approaches to managing the latent load in homes.  The primary goal is to identify approaches and technologies which can achieve energy-efficient latent cooling in light of requirements that increased ventilation rates be implemented in Florida homes.

Approximately 30 articles, research reports, presentations and code documents were reviewed for Task 1.  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%.

Another recent monitored FSEC study involving six Gainesville Florida homes (Martin et. al. 2014) also noted in FSEC's concurrent 2014 DBPR ventilation report (Sonne, Vieira 2014) found continuous exhaust ventilation (CEV) at approximately ASHRAE 62.2-2010 rates to raise summertime indoor RH by 5% compared with runtime only, central fan integrated supply (CFIS) ventilation that provided approximately 20% of ASHRAE 62.2-2010 requirements, which is substantially lower than the CEV rates. So, one cannot simply conclude from this that CEV raised RH 5% compared to CFIS. Figure 1 from the same study shows significantly more hours of indoor RH > 60% for the homes during continuous exhaust ventilation (right bars for sites 1, 2, 4, 6, 8 and 9) versus during runtime ventilation periods. The data shown was collected from about June 28 through October 15, 2013.

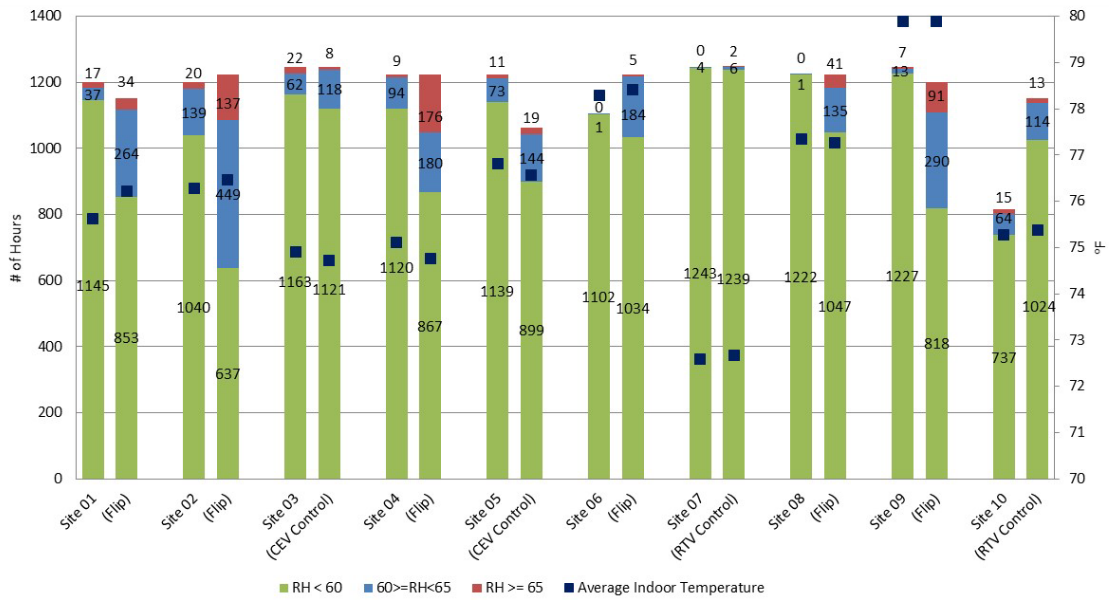


*Figure 1: Distribution of hours at various % RH ranges, separated into runtime vent (left bar) and continuous exhaust vent (right bar) periods, each corresponding to the left axis (# of hours). Numeric labels correspond to hours, black squares correspond to the right axis (average indoor temperature). Sites 3 and 5 were always operated with continuous exhaust ventilation, and sites 7 and 10 were always operated with runtime ventilation. (figure and caption from Martin et. al. 2014)*

The 2014 Building America Expert Meeting report noted above also indicates mechanical ventilation caused increases in indoor RH and observes the reasons for the higher level:

> *…mechanical ventilation, operated at the ASHRAE 62.2-2010 addendum r rate, in a 3 ach50 house, raises the annual median indoor RH by almost 10% RH compared to a 7 ach50 house without mechanical ventilation in Orlando. That is because infiltration drivers are generally weak in that climate during floating hours (when it is still humid outside and the cooling system is not removing moisture), but mechanical ventilation forces a minimum air exchange.*

An earlier monitored study (Rudd and Henderson 2007) similarly found continuous whole-house ventilation combined with infrequent cooling demand to cause high humidity levels. Conversely, the study report noted that due to low driving forces during mild conditions, a naturally ventilated home would not see much ventilation and might as a result have consistently lower humidity.

As residential mechanical ventilation use has increased over the past several decades, means of addressing increasing RH were developed. A 2014 Building America report (Martin 2014) includes a summary of historically successful methods, by several measures of providing mechanical ventilation to homes while still maintaining comfort and minimizing moisture issues:

> *BA-PIRC (formerly BAIHP) worked with site and factory builders constructing custom, production, affordable, and multifamily homes to implement supply-based mechanical ventilation through the introduction of outdoor air into the return side of centrally ducted, forced-air, space conditioning systems. This approach, combined with rightsized heating/cooling systems and properly operating bathroom and kitchen exhaust fans*

*(ducted to the outdoors) has been implemented in thousands of homes, primarily in the southeastern United States, since 1997 and has effectively controlled odors, maintained comfort, and proven effective at minimizing wintertime moisture buildup (Chandra et al. 2008). Similar to BSC's approach, these systems draw outdoor air from a known fresh air location, filter the air, temper the air by mixing it with central system return air, and fully distribute the air. Systems have been commissioned to deliver approximately 30%–70% of ASHRAE Standard 62.2-2010 rates, enough to create a slight positive pressure in the home with respect to outdoors; however, only while the central HVAC system is running to satisfy a heating or cooling requirement. Therefore, operation of the ventilation system is intermittent, especially during periods of limited to no HVAC runtime. In the Southeast, these periods typically coincide with increased natural ventilation through more frequent window operation, and the system has gained the acceptance of homeowners and builders alike in terms of comfort, durability, energy consumption, and perceived odor and moisture control. However, most of these systems do not meet the whole-house mechanical ventilation requirements of ASHRAE 62.2-2010*.

While these results are very informative, as homes continue to become more energy efficient, likely with more stringent airtightness and higher ventilation requirements, additional latent control strategies will be required. 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.*

## Ventilation Strategies

Florida Code permits mechanical ventilation to be provided in a variety of ways. It may be provided by exhaust only, supply only or combinations of both. It is important to realize that some ventilation methods may result in un-intended consequences under certain conditions. While it may have worked in many homes without any observed consequences, an exhaust only method of ventilation in Florida is not recommended here. An exhaust dominated building will operate under negative pressure. Air will come into the home wherever leaks are located. Ventilation air in an exhaust only home will consist of a blend of outdoor air through window and doors, air from an attached garage, and attic air. The fact is there is no way to know where the air is coming in from or the quality of it. Depressurization in a tight home can enhance the transport of soil gases such as radon into the home or pollutants from attached garages into the home. In Florida's warm-humid climate, exhaust ventilation is also not recommended as it creates negative pressure in the home with respect to outdoors which in turn can pull water vapor into building materials. If the home is kept cold enough or a vapor barrier such as vinyl wall paper is on a wall exposed to high moisture content, then high wall surface humidity may occur creating mold and possibly building damage. Therefore an exhaust only method of ventilation is not recommended in this report and has the potential to be more detrimental than an under-ventilated home. The topic of airtightness and potential consequences is covered in more detail in a parallel home airtightness and ventilation DBPR research report (Sonne and Vieira 2014).

The best method of ventilation is through either a balanced or positive pressure approach. The balanced approach may be used by simply supplying the same amount of OA as is used in an exhaust method. The supply air should at least be filtered and ideally conditioned before being dumped into the home. Energy Recovery Ventilators (ERV) exchange some of the cooling energy content of conditioned exhaust air to the OA supply and can provide a balanced house air pressure. It is important that ERV be designed and installed well as well as maintained to ensure that the air flow balance occurs over time. Another method simply providing supply air can also be used. Best practice in using this method is to filter it and drop it at the return of a central cooling or other space conditioning system. Dropping the supply OA into the conditioned space away from space conditioning return will result in elevated indoor RH (during cooling season) in at least the vicinity of the OA termination. This area may be warm and humid depending upon conditions of the OA and the size of the space it terminates into. Terminating close to a thermostat may result in overcooling areas of the home.

## Latent Control Approaches and Technologies

While the goal of this research is to identify energy efficient and cost-effective approaches to managing latent loads, it should be noted that even highly energy-efficient solutions will add some energy use as removing latent energy requires energy.

The following section provides a listing of dehumidification technology options identified with an overview and performance and cost effectiveness discussion for each.  Note that several of the options require a sensible (heat) cooling load so are not seen as “stand alone” supplemental dehumidification as much as air conditioning latent enhancement.

### Overcooling

**Overview:** This strategy involves reducing the thermostat set point temperature below the desired temperature by two to three degrees while at the same time lowering the airflow rate to increase air conditioner run times and in turn remove additional moisture.

**Performance and Cost Effectiveness:** Simply overcooling a space will increase the runtime and remove moisture, but may not reduce the RH much resulting in a cold and clammy feeling. Modeling work reported in a 2013 Building America Expert Meeting publication (Rudd 2013a) showed this strategy (reducing the set point by 3°F and at the same time lowering the airflow rate from 375 cfm/ton to 210 cfm/ton) to reduce hours above 60% RH by 95% in Miami and 50% in Orlando.  The key here is that the airflow rate was reduced to make the coil colder which lowers sensible heat ratio (SHR). Drawbacks include increased energy use, the possibility of comfort issues from overcooling and inadequate moisture removal during swing seasons and other mild temperature periods when air conditioning isn’t called for.  In his 2013 supplemental dehumidification publication, Rudd (Rudd 2013b) estimates the cost of lowering the setpoint of a HERS Index 50 house by 2°F together with lowering the airflow rate at less than $10/yr. While the cost appears low, it should be kept in mind that this is not completely effective at keeping all hours of the year under 60% RH.

### Fan Airflow Adjustments

**Overview:** Air handler fan adjustments that can potentially reduce moisture levels in homes include 1) reducing airflow and 2) controlling fan end times. Reducing the airflow of variable fan speed air conditioners reduces the cooling coil temperature which in turn increases the latent capacity and decreases the sensible capacity of these systems, increasing run times and reducing moisture levels in homes. Controlling air hander fan end times to minimize evaporation off of the coil (e.g. not running the fan for a period of time after the compressor shuts off) also reduces indoor moisture levels.

**Performance and Cost Effectiveness:** While the increased latent capacity increases overall AC energy use (Parker et. al. 1997) for both fan adjustment strategies, there is no extra up front equipment cost. There would be a contractor cost of about 1 hour to implement these changes. Controlling fan run on time may not be an option with many systems. These strategies cannot be used during mild weather when no air conditioning is called for (or would require overcooling the space under these conditions) they are not adequate latent control approaches by themselves.

### Energy Recovery Ventilation

**Overview:** Energy recovery ventilators (ERVs) use air-to-air heat exchangers to recover sensible and latent energy from building exhaust air and supply pre-conditioned outdoor air to the conditioned space. They could be considered to be a supplementary dehumidification device in that they are able to reduce a portion of the latent load in ventilation, but not all of it at all times. This measure will only work when the indoor air is maintained at dry conditions (dewpoints around 55°F or so) by a central cooling system.

**Performance and Cost Effectiveness:** A 2007 monitored study (Rudd and Henderson 2007) showed indoor humidity to be controlled "reasonably well" by an ERV during summer cooling periods, but also that during mild conditions the ERV provided practically no dehumidification. Similarly, modeling work reported in the 2014 Building America mechanical ventilation report (Martin 2014) showed ERVs to nearly eliminate indoor hours above 60% RH during cooling periods for a DOE Challenge Home level efficiency home but were not found to be effective in limiting indoor RH during floating periods with small differences between indoor and outdoor dew point temperatures. Comparing balanced ERV ventilation to exhaust ventilation as a reference, the modeling work showed ERV use in a DOE Challenge Home level home to, on average, add $20/yr in energy cost at 100% of the ASHRAE 62,2-2010 ventilation rate in an ACH50 3 home, but to *save* $20/yr at higher ventilation rates required for an ACH50 1.5 home. Costs for ERV can range from about $700-$1400.

### Heat Pipes

**Overview:** Heat pipes, also called run-around coils, circulate heat transfer fluid through heat exchangers on either side of the cooling coil to precool air entering the coil and reheat air leaving the coil. This allows the coil to stay colder and in turn remove more moisture from the air while overcooling the conditioned space is also reduced. This is uncommon in residential applications, but is used effectively in commercial buildings.

**Performance and Cost Effectiveness:** Modeling work reported in the same 2013 Building America Expert meeting publication reference above (Rudd 2013a) showed heat pipes were less effective in reducing indoor humidity than the overcooling strategy with low airflow but still able to reduced hours above 60% RH significantly. One model showed RH above 60% to be reduced by about 90% in Miami and 30% in Orlando while another model showed Orlando hours above 60% RH to be reduced by almost 50%. It was estimated that increased static pressure and fan energy use for the heat pipe system resulted in a 20% to 25% increase in space conditioning cost. Since, like the fan flow adjustment and overcooling options discussed above, heat pipes only operate while the air conditioner is running, they also are not adequate latent cooling approaches by themselves since they can't be used during mild weather when no air conditioning is called for. Costs for residential applications may be somewhere in the range from $2000-$3000.

### Dehumidifiers

**Overview:** The most common type of dehumidifier uses the refrigeration cycle to remove moisture from the air. Since the condenser coil is packaged with the unit and air is not exhausted to outdoors, the space is also heated as part of the process. Dehumidifier types include stand-alone, ducted and gas-fired desiccant varieties.

**Performance and Cost Effectiveness:** A 2002 monitored study that studied a range of dehumidification strategies including several dehumidifier configurations, ERVs, and a two-stage compressor (Rudd et. al. 2002) concluded that a hall closet located standard dehumidifier provided the best overall value (considering humidity control, first cost and operating cost). Modeling work reported in the 2013 Building America Expert Meeting publication (Rudd 2013a) showed all three types of dehumidifiers identified above to eliminate hours over 60% RH in Orlando and Miami. The gas-fired desiccant option had the lowest operating cost but also has the highest installed cost, estimated by Rudd (Rudd 2013b) at $2,000. The ducted dehumidifier had the next lowest operating cost. The estimated installed cost of a ducted dehumidifier can vary widely from about $1,000 - $2,000 depending upon capacity, space requirements and if a condensate pump is needed. The stand-alone unit had the highest operating cost of the three, but is the least expensive. The estimated installed cost can be as low as $400, but can quickly climb to as much as $2000 for quiet high capacity systems with a remote dehumidistat.

The 2014 Building America report (Martin 2014) indicates a $10-$30 annual cost for supplemental dehumidification to control indoor RH to 60% in the warm-humid climates of Charleston, Houston, Orlando and the marine climate of Los Angeles. The report also provides some additional dehumidifier performance considerations however that suggest actual operating costs will be higher:

*However, a caveat was provided that indicates that this value is predicated on an operating dehumidifier energy factor of 1.47 L/kWh, and recent field data indicate that conventional dehumidifiers operate closer to 0.8 L/kWh (Mattison and Korn 2012), which would  tend to double this cost. Additionally, dehumidifiers tend to operate on a large humidity dead band, which means that maintaining humidity below 60% would likely require humidity set points near 55%, which could dramatically increase dehumidification costs.*

### Dual Capacity Air Conditioners

**Overview:** Dual capacity air conditioners are similar to traditional single-speed air conditioners except they provide cooling at two specific capacities, which allows them to run at a lower capacity most of the time increasing runtime, then go to a higher capacity during peak cooling conditions. Some manufacturers set the first stage at 50% of total capacity and others around 75% of total capacity.

**Performance and Cost Effectiveness:** Based on modeling results, the 2014 ASHRAE humidity control options report (Henderson and Rudd 2014) indicated no appreciable reductions in hours above 60% RH for two-speed air conditioning unless coupled with reduced airflow and/or overcooling. Contrary to the(Henderson and Rudd 2014) modeling results, the author has observed improved indoor RH control using dual capacity systems compared to a single capacity system without overcooling, but it will not eliminate all hours of RH>60%. How well the first stage capacity meets swing season loads is an important factor in how much it will operate an limit hours of RH>60%.

### Variable Speed Air Conditioners

**Overview:** Variable speed air conditioners and heat pumps are very high efficiency (SEER 21 to 24+) space conditioning systems which achieve much of their efficiency operating below their nominal rated capacity and limited cycling on and off. They run about twice as long as fixed capacity systems (Cummings and Withers 2011) and can modulating capacity from approximately 40% to over 100% of nominal.

**Performance and Cost Effectiveness:** As reported above for two-speed air conditioners, based on modeling results, the 2014 ASHRAE humidity control options report (Henderson and Rudd 2014) reported no appreciable reductions in hours above 60% RH for variable speed air conditioning unless coupled with reduced airflow and/or overcooling.  These modelled results are surprising given substantial testing results in an FSEC lab. Extensive monitoring of a three-ton variable capacity system maintained daily average indoor RH typically between 53%-55% over a variety of weather conditions (Cummings and Withers 2011). The three-ton variable capacity system was about 90% oversized when used with an indoor duct system. There was no mechanical ventilation in the lab home at the time of the study, but the home was fairly leaky with an ACH50 of about 10. Additional experimental results using a variable capacity system in a mechanically ventilated building will be discussed in further detail in the Task 2 section of this report. Variable capacity systems are well-suited for swing seasons since they can provide cooling down to 40% of the nominal rated capacity. These systems also come with an RH control feature which will allow operation at lower cfm/ton for periods and also permit overcooling past the set point of 2 degrees. The cost of a variable capacity heat pump is approximately $3700 more than the cost of a SEER 13 single capacity system.

## Subcooling Reheat

**Overview:** Subcooling reheat is a cooling system enhancement technology that combines overcooling and reduced airflow with a heating coil downstream of the evaporator coil to reheat the air being delivered to conditioned space with heat recovered from the refrigerant line.

**Performance and Cost Effectiveness:** Modeling results reported in the 2013 Building America Expert Meeting publication (Rudd 2013a) showed subcooling reheat technology to eliminate all hours over 60% RH in Miami and nearly eliminate hours above 60% RH in Orlando. However, Rudd's 2013 Supplemental Dehumidification report (Rudd 2013b) notes that since this technology incorporates cooling air below the desired set-point, it "is not a full supplemental dehumidification option because it cannot continue to operate indefinitely to control indoor humidity without unacceptably overcooling the conditioned space." A cost estimate provided by (Ruud 2013b) is $1,600.

## Full Condensing and Subcooling Reheat

**Overview:** This technology is very similar to subcooling reheat, but as described in Rudd 2013b, by using modulating hot gas reheat to bring cooled and dehumidified air back to room-neutral temperature, can operate with greater efficiency and also run indefinitely without overcooling.

**Performance and Cost Effectiveness**: Rudd's 2013 Supplemental Dehumidification publication (Rudd 2013b) states that this strategy "is the most energy efficient option for effectively controlling indoor relative humidity near 50% RH" and that while it doesn't perform quite at the level of the stand-alone or ducted dehumidifiers, only a "relatively small number of hours … remain slightly above the desired RH set point…." The cost estimate for this technology is about $1,750 (Ruud 2013b).

## Dedicated Outdoor Air System

**Overview:** Dedicated outdoor air system (DOAS) technology preconditions ventilation air before it is introduced into a building, providing filtration, cooling and moisture removal. This is an effective practice in commercial buildings. It is a more difficult application for residential OA which will typically only be about 60 cfm. There are some systems currently available through Unico for residential applications that are well-suited to handle 100% OA in hot humid climates. The system includes a variable speed motor and intelligent controller programmed to handle cooling humidity control and ventilation needs. This system essentially uses a deep coil designed for low flow rates to maintain a colder coil and lower SHR when needed during higher indoor RH periods. Higher duct velocity design has also been shown to increase ventilation mixing effectiveness throughout the home. Cost for the entire installed ducted system is about $7000.

Currently FSEC has been experimenting with using a variable capacity minisplit as part of an OA system. A residential mini-split dedicated outdoor air (MSDOA) system can be used to pre-condition OA before being distributed in a home. For best RH control the OA should be ducted to the return intake at low velocity to encourage effective capture into the return. Cost of an installed MSDOAS is estimated to be about $3200.

**Performance and Cost Effectiveness:** Currently variable capacity mini-splits with SEER ratings around 19-20 are available. There is no known published data on using this method, but it is currently under examination and results will be discussed in further detail in Task 2. The cost of an installed MSDOA is estimated to be about $3200. In cases where a SEER 19 mini-split is used in a home with a SEER14 central system, the mini-split will operate more efficiently than central cooling system so there is an efficiency advantage to trying to use it as a primary stage of cooling. The MSDOA thermostat can be set one to two degrees below the set-point of the central cooling system. If the MSDOA can't keep up with the load, the central system can turn on and run as needed.

## Discussion

A two-part latent load management strategy is appropriate. While one specific air conditioning latent enhancement recommendation cannot be made for all Florida homes and homeowners, using some levels of overcooling and fan airflow adjustment together with dual or variable capacity systems (if so equipped) will typically be helpful in the absence of having any other additional measure of RH control.

A 2007 BSC research report (Rudd and Henderson 2007) summarized equipment control strategies

> *Several equipment manufacturers offer equipment with enhanced moisture removal at part load. These systems typically vary blower speed, operate at lower compressor stages, or lower cooling setpoints to provide more dehumidification. The increasing use of variable-speed fan motors, lower-cost humidity sensors, and embedded electronic controls in higher-end cooling systems have made these control approaches practical. Typical control algorithms lower airflow at low-load conditions to reduce coil temperatures and provide more moisture removal (Krakow et al. 1995; Andrade and Bullard 2002).*

However, as noted above in the discussion of several of the latent control options, since enhanced control strategies cannot be used during mild weather when no air conditioning is called for (or would require overcooling the space under these conditions) they are not adequate latent cooling approaches by themselves for high-performance homes and a second, separate dehumidification strategy will typically be needed.

The conclusion section of the 2002 monitored BSC dehumidification study (Rudd et. al. 2002) summarizes this well:

> *All of the systems with dehumidification of recirculated air, separate from the cooling system, exhibited much better humidity control than those with dehumidification of ventilation air only (ERV system) and those with dehumidification only as part of the cooling system. Therefore, the problem of high humidity does not lie with mechanical ventilation, and the solution does not lie with the cooling system. The problem of elevated humidity in energy-efficient homes in hot-humid climates is a result of interior moisture generation and lowered sensible heat gain. High-performance windows and insulation, and*

*locating air distribution ducts inside conditioned space reduces sensible heat gain to the extent that the fraction of latent cooling load to total load is often outside the capacity range of even the best currently available mass-market cooling equipment. The solution, for now, is to employ dehumidification separate from cooling in hot-humid locations.*

Rudd concludes the abstract of the article by stating:

*Some options are less expensive but may not control indoor humidity as well as more expensive and comprehensive options. The best performing option is one that avoids overcooling (cooling below the requested set point) and avoids adding unnecessary heat to the space by using waste heat from the cooling system to reheat the cooled and dehumidified air to room-neutral temperature.*

The 2013 Supplemental Dehumidification publication (Rudd 2013b) provides a listing of the most effective options that have “relatively low operating cost and essentially eliminating indoor humidity above 60% RH:”

- Full condensing and subcooling reheat integrated with the central cooling system
- Ducted dehumidifier
- Stand-alone dehumidifier with central system mixing
- DX condenser-regenerated desiccant dehumidifier.

While he doesn’t break out operating energy use by technology in this summary, Rudd also estimates that supplemental dehumidification will use about 170 kWh/yr for a HERS Index 50 house with ducts in conditioned space and a 60% RH set point.

The 2013 Building America Expert Meeting report (Rudd 2013a) also provides a general cost for supplemental dehumidification for high performance homes:

*The supplemental dehumidification energy consumed to keep indoor RH below 60%, taken as the difference in total HVAC cost for the same building with and without supplemental dehumidification, is relatively small. It is in the range of 250 kWh/yr or less ($30/yr or less), but necessary to enable deep cuts in sensible heat gain without incurring long periods of elevated indoor RH.*

Several supplemental dehumidification options have been discussed thus far with much of the most recent work cited by Rudd2013a and Rudd2013b. Following here is Table 1 which summarizes these options with estimates on first cost along with some positive and negative attributes of each. These are presented here as a general summary. Costs shown can vary widely, but an attempt is made to provide some idea on general expectations.

*Table 1. Supplemental Dehumidification Options (cost sources: Rudd 2013b and FSEC research).*

| 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** | $3,000 | Long life, low maintenance | May not have room to install. No help in swing season. |
| **Enthalpy recovery ventilation** | $700-$1,400 | 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** | $1,800* | Low speed can result in lower energy use while saving energy | Higher first cost. Better than single cap., but still some hours swing season it will not operate. |
| **Variable capacity air conditioner ventilation** | $3,700* | 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** | $7,000 | Good RH control. Excellent ventilation effectiveness potential. | High first cost. |
| **Mini-split Dedicated outdoor air system** | $3,200 | 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-$2,000** | 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** | $1,000-2,000** | 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, |

* cost increase compared to single capacity S13 system.

** Wide variability in cost depending upon capacity, availability of space, and if condensate pump is needed.

## Task 2: Experimental Work

Task 2 of this project is to complete a minimum of four different experiments in a lab building to assess resulting indoor RH and energy consumption. The experiments are to use various latent load management approaches (including a dehumidifier) at various levels of mechanical ventilation. It should be noted that the original proposed scope of this work is larger than evaluating four different experiments, however there was only time to evaluate four within the timeframe of the current work contract. A second phase of this work is proposed to continue evaluation of other experiments.

### Experimental Method

Four experiments were conducted within the Building Science Lab building located on the Florida Solar Energy Center campus. It has a conditioned floor area of 2000 $ft^2$ with concrete masonry block walls having R5 unfaced foam board insulation located on the interior side of the wall. Windows are single pane clear glass set in metal frame. Ceiling insulation is R19 batt. Building airtightness was tested using a blower door and measured a normalized air leakage rate of 4.5 ACH50. This tightness would meet the 2012 IECC requirement of 5 ACH50 or tighter and would therefore require mechanical ventilation. Duct airtightness testing measured 53 CFM25 total, but since the ducts are located within the primary air barrier of the building, the measured CFM25out=0. A manual J8 load calculation on the building calculated a summer 99% design total cooling load of 35,780 btuh based on 130 cfm of mechanical outdoor ventilation air (our maximum ventilation rate test) and the internal sensible loads

The building is currently configured with a central ducted variable capacity SEER21 heat pump (S21) with a nominal rated 3 ton cooling capacity. This system is able to vary cooling from 40% up to 118% of its nominal capacity. It does so by varying refrigerant flow as well as varying the system airflow. This means that the system is able to deliver cooling from about 1.2 tons up to about 3.5 tons. The S21 system is controlled by a thermostat that monitors subtle changes between the room temperature and set point, then determines the stage of cooling (or heating) to operate at. The system is capable of running at eleven different stages of cooling. Its nominal capacity operation is at about the eighth stage and stages 10 and 11 only occur if the room temperature has exceeded the set point by two degrees for several minutes. The thermostat also offers a humidity control option and fan cycle option which are well-suited for improving humidity control and mixing of mechanical OA.

Internal loads were established using guidance from a Building America report on internal residential loads (Hendron and Engebrecht 2010). Internal cooling loads were maintained consistently throughout all experiments by keeping the building unoccupied and providing internal sensible and latent heat through controlled measures. Sensible heat was added primarily through interior lighting, space heater and mechanical fans. The interior sensible loads were monitored using power meters during the entire project to ensure consistency was maintained for each experiment. The average interior sensible load delivered per day was at a rate of 3851btu/h. An interior latent load of about 9.8 pounds of water each day was evaporated into the building (an average latent heat rate of about 397btu/h) and distributed within the building by a floor circulation fan. The latent load was also monitored throughout the project by means of a tipping bucket that provides a pulse output proportional to the volume of water passing through the evaporation assembly.

Electric energy was measured using Continental Controls Wattnode power meters and were calibrated using comparison to a manufacturer calibrated Dranetz Power Visa power meter analyzer, using model TR-2510B CT's. Measurements were within 2% agreement of each other. Latent heat added and removed (as liquid phase water evaporated or water drained from cooling coils) from the building was measured using tipping buckets carefully calibrated at the anticipated rates of flow for each application. The basis of determining tipping bucket calibration was by supplying a drip rate of water to each bucket used where the number of tips were measured for a given measured mass of water. Indoor and outdoor conditions were measured using Type T thermocouples and Vaisala temperature and relative humidity sensors. These were calibrated against a manufacture calibrated Vaisala HM34 temperature and humidity sensor.

All data from sensors were collected using a Campbell Scientific, Inc. CR10 datalogger where data was gathered several times each day from a FSECs central computer terminal. Data from sensors were sampled at 10 second intervals, then processed and stored at 15 minute intervals. Upon collection by the central computing terminal, the raw data from the datalogger was screened for out of bound errors and then processed for terminal collection in the main project database account. Errors or missing scans are marked and noted within the main database. No missing scans occurred during the data used in analysis.

**Mechanical Ventilation**

As covered in the fourth paragraph of the Background section of this report, Florida code references both the IMC 2012 and IRC 2012 ventilation requirements. They have the same 60 cfm requirement for 3 bedroom homes between 1,501ft$^2$-3000 ft$^2$, however they may differ depending upon the house area and number of bedrooms. While Florida code does not specifically reference ASHRAE Standard 62.2, a very tight home could potentially have twice the mechanical ventilation rate as Florida code requirements.

Mechanical ventilation was provided through an in-line fan in a supply terminal application where air is pulled from outdoors through a south side wall access port and ducted into the building to a desired terminal location. To ensure consistent air delivery occurred, a Continental Fan iris damper with differential pressure measurement across the iris orifice was used to measure the actual delivered air flow rate into the building throughout the entire project. The iris damper was calibrated using a TSI model 8390 Windtunnel.

Two ventilation rates were chosen to be applied to two different RH control strategies. The first OA rate chosen was 60 cfm. This rate is what the IMC 2012requires for a three bedroom home (15cfm/bedroom plus 15 cfm) and is also what IRC 2012 requires for 3 bedroom homes between 1,501ft$^2$-3000 ft$^2$.

The second rate chosen was 130 cfm. This rate is on the extreme range of what would be expected in residential construction, but is plausible. This is much higher than would occur in any typical home based strictly upon IMC 2012. Based upon IRC 2012, 130cfm would be required in a home >7,500 ft$^2$ with 4-5 bedrooms. This rate would also be required in ASHRAE 62.2 for an extremely tight home of 0.5 ACH50 with 3025 ft$^2$ and 5 bedrooms. Testing at 130 cfm also allowed the dehumidification strategies to be challenged more.

**Test Configuration Descriptions**

Test configuration 1: 130 cfm OA to MS; Central SEER21; Dehumidifier backup at 60%RH

Test 1 consisted of running a variable capacity Fujitsu minisplit heat pump at a 74°F cooling setpoint with 130 cfm OA ducted to a constructed return air plenum with an air flow station inside. The Fujitsu 18RLXFW mini-split heat pump has a nominal capacity of 1.5-tons but maximum capacity of 1.92 tons (23,000 Btu/h). Figure 2 shows the minisplit system near the top right of figure and a view of the metal ducted ventilation system. The primary purpose of the ducted return plenum was to enable an accurate air flow measurement through a calibrated flow station inside. The plenum also effectively captures all the OA when the MS is on. The other sections of duct leaving the minisplit can be used to distribute OA to other locations as needed. This MS only allows setpoint in increments of 2°F. It was found that the 74°F setpoint enabled adequate handling of OA without overcooling the space during the conditions the experimental period.

The SEER21 central system was set at 77°F setpoint so it was able to turn on to maintain comfortable conditions if the MS was not providing enough cooling. Interestingly the MS unit temperature control tolerates a long slow rise in interior temperatures gradually increasing capacity over time without jumping up into a high cooling capacity. The typical pattern as shown in figures later is for the MS to handle most of the cooling load, then the central system activates cooling for a while during the middle part of the day. But the MS is not operating at its highest capacity at the time the central system comes on. This appears to be a normal operation of this type of variable capacity MS as observed in a previous proprietary project.

*Figure 2. Minisplit with OA duct termination at return intake plenum.*

A dehumidifier rated to remove 70 pints/day was placed within the central room with a remote dehumidification control set on the wall near the central system thermostat shown in Figure3. The dehumidistat control was set to 60% RH. As is common with many dehumidistat controls, the markings on the controllers can be off by 5%RH. A calibrated Vaisala HM34 was used to confirm activation at the

60%RH level. The dehumidifier could then cycle on if any of the dehumidification control measures could not maintain 60% RH indoors. Energy and condensate of the dehumidifier was monitored, although the measures used adequately controlled RH and therefore it never operated.

*Figure 3. View of thermostat and remote dehumidistat on wall at left and view of OA duct termination just above the central intake return. Dehumidifier is on a stand on the right side.*

Test configuration 2: 60 cfm OA to MS; Central SEER21; Dehumidifier backup at 60%

- Test configuration 2 was essentially the same as described in the previous Test 1 description except at a lower ventilation rate of 60 cfm instead of 130 cfm was delivered to the MS.

Test Configuration 3: 60 cfm to SEER21 central system; MS off; DH at 60%

This test operated with only the SEER21 central ducted cooling system set at 77°F setpoint and the dehumidifier activated at the 60% control point. The OA rate of 60 cfm was ducted to the central return. The OA termination can be seen just above the central return intake in Figure 3. The box assembly in front of the central return was used to verify a visualization of the effectiveness of OA capture into the return. Figure 4 shows visualization of the OA capture using fog injected into the OA intake from outdoors. This image was taken while the central system was running at its lowest stage of cooling and low flow rate which makes capture more difficult. While the image shows some concentration slightly to the right of the box, it was quickly captured into the return at the edge. Tissue paper placed at the front edge of the box also shows the airflow direction from the room into the central return. Use of tracer gas in the building also indicated complete capture into the return with the central system in the lowest stage of cooling. The capture potential is greater as the central system airflow increases.

*Figure 4. Fog injected into the OA intake shows Capture of OA at 60cfm into central return during the central system's lowest stage of cooling and low system flow rate. Tissue paper indicates air flow direction from room into return.*

Test Configuration 4: 130 cfm to S21 central system; MS off; DH at 60%

- Test configuration 4 was the same as described in the previous Test 3 description except at a higher ventilation rate of 130 cfm instead of 60 cfm was delivered to the central system return.

## Results

The experiments were conducted May 2-June 12, 2014. The outdoor and indoor conditions are summarized in Table 2 for each of the four test configurations. The overall average of every day is shown along with the range from lowest daily average to the highest daily average in parentheses. The indoor temperatures shown in this report are an average of five different locations distributed around the building.

*Table 2. Daily average conditions during each test configuration with the total number of days for each set.*

| Test Configuration | Out temp. (°F) (range) | Out dewpoint (°F) (range) | Indoor temp. (°F) (range) | Indoor RH (%) (range) | # days |
|---|---|---|---|---|---|
| 1 MS, OA 130cfm +S21 | 75.3 (70.0-79.3) | 63.4 (53.7-68.2) | 76.5 (75.3-77.1) | 47.3% (44.4%-51.9%) | 10 |
| | | | | | |
| 2 MS, OA 60cfm +S21 | 75.2 (70.0-79.6) | 62.5 (52.8-71.0) | 76.6 (75.8-77.2) | 47.3% (42.9%-48.7%) | 10 |
| | | | | | |
| 3 S21, OA 60cfm | 79.9 (77.1-82.5) | 67.8 (62.2-70.5) | 76.3 (76.1-76.4) | 49.2% (46.2%-51.6%) | 10 |
| | | | | | |
| 4 S21, OA 130cfm | 78.8 (75.4-81.7) | 66.0 (61.4-69.3) | 76.3 (76.1-76.4) | 50.9% (49.7%-52.5%) | 9 |

While variability in outdoor conditions can be seen for all test configurations in Table 2, it is perhaps more evident in Figure 5. Figure 5 shows the hourly average outdoor temperature and dewpoint temperature throughout the testing period. The X-axis is continuous time with the test configuration and beginning date of monitoring. T1-M2 on the axis indicates when Test configuration 1 began on May 2 (T2-M12 indicates when Test 2 began on May12 and so on). The most notable observations are that the first two test configurations occurred with cold fronts passing through. This is not unusual variability for this time of year. The temperatures in mid- June through mid- September typically have much less variability. It is the weather that occurs in August through November that would be most useful to evaluate indoor RH control effectiveness. This can involve days with high outdoor dewpoints, but mild sensible loads when indoor RH will rise due to less air conditioner run-time. Given that Test 1 configuration had the least amount of dewpoints near 70°F, an attempt was made to collect more data beginning June 11 (T1-J11 on Figure5), however the weather did not favor our effort as there was another dive in outdoor dewpoint during this period.

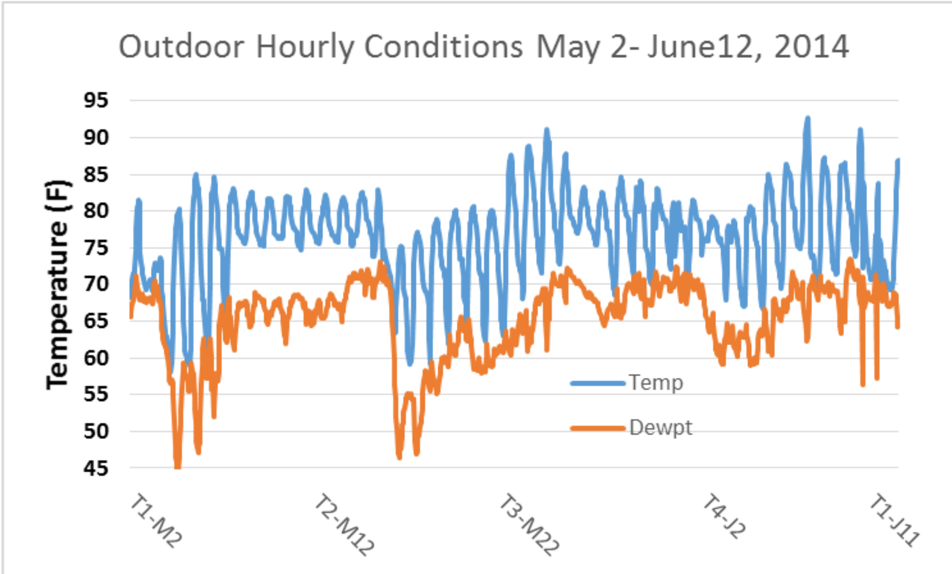


*Figure 5. Outdoor Temperature and Dew Point (Hourly Averages)*

## Cooling Energy and Latent Removal

Although there were many relatively dry days, each test had at least one or two days when the average outdoor dewpoints were 68°F-71°F. This section highlights the daily pattern of energy use and resulting indoor conditions using days with the most available similarity in warm moist outdoor conditions. Table 3 shows a summary the outdoor drybulb and dewpoint temperatures along with the indoor drybulb temperature, RH, and latent (condensate) removal from the building. The average for the day is shown along with range from lowest to highest hourly average in parentheses. The resulting indoor temperature and RH are also shown for the same day. The average daily indoor temperatures are very close to each other, but test configurations 1 and 2 using the MS show a wider band from the lowest to the highest temperature. While overcooling did not occur in this testing, there is potential to overcool a space where the MS is located depending upon the size of the room and set-points of cooling equipment.

There were no days in which the indoor RH was equal to or exceeded 60% and therefore there was no dehumidifier operation. The latent removal shown in Table 3 represents the total from all space conditioning equipment. Using the MS unit clearly removes much more moisture as seen in Table 3. The average daily removal using the MS vs S21 alone is 23.9 pints (88.6 pints-64.7 pints= 23.9 pints) (36.9%) more. Using the MS results in an average difference in indoor RH of 4.3%RH (51.4%RH-47.1%RH= 4.3%RH) which is 8.4% lower.

*Table 3. Daily average conditions during each test configuration during similar outdoor conditions.*

| Test Configuration | Out temp. (°F) (range) | Out dewpoint (°F) (range) | Indoor temp. (°F) (range) | Indoor RH (%) (range) | Latent heat removed (Pints/day) |
|---|---|---|---|---|---|
| 1 MS, OA 130cfm +S21 | 79.3 (76.2-83.4) | 66.5 (62.0-68.7) | 76.5 (75.3-77.1) | 47.2% (44.7%-50.2%) | 96.8 |
| | | | | | |
| 2 MS, OA 60cfm +S21 | 79.4 (75.6-84.7) | 68.1 (67.1-71.1) | 76.6 (75.8-77.2) | 47.0% (44.7%-51.1%) | 80.3 |
| | | | | | |
| 3 S21, OA 60cfm | 78.7 (70.8-85.9) | 69.3 (65.6-71.6) | 76.3 (76.1-76.4) | 51.1% (48.5%-52.5%) | 52.7 |
| | | | | | |
| 4 S21, OA 130cfm | 79.6 (69.8-90.6) | 69.3 (68.0-71.2) | 76.3 (76.1-76.4) | 51.7% (49.7%-54.0%) | 76.6 |

Following are several daily plots (see Figures 3-12) that represent the same days shown in Table 3. Some Figures show the cooling energy used and latent heat (condensate) removed during each hour of the day. The energy is in kWh on an hourly basis and could be also be considered the average kW for each hour. There are also Figures that show the daily profile of indoor and outdoor conditions for each configuration.

Test Configuration 1 with 130 cfm OA ducted to the MS return and using SEER 21 Central System has the energy use and latent heat removed for each system shown in Figure 6. Data shown occurred on May 10, 2014. This configuration used a total of 21.6 kWh of cooling energy (MS+S21) for the day and removed a total of 96.8 pints (12.1 gallons) of latent load. It is also of interest to see the combined impact by adding the energy and condensate of both systems as shown in Figure 7.

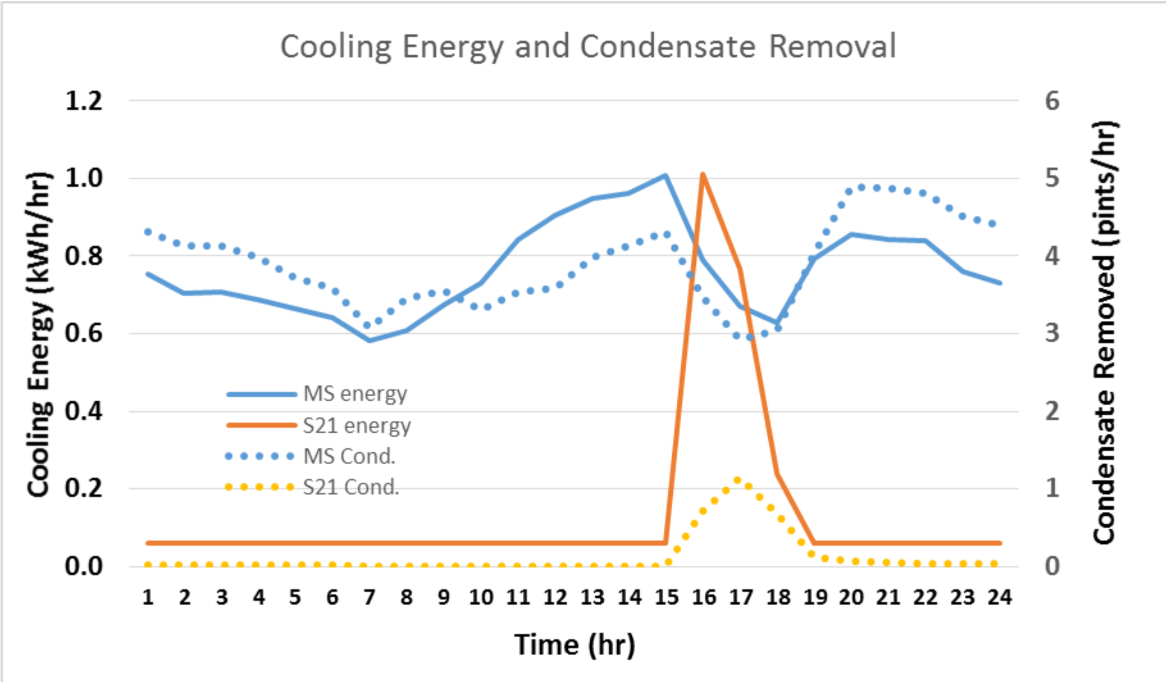


*Figure 6. Test 1 cooling energy consumption and condensate removal per hour for each system shown for May 10; 130 cfm OA to MS and S21 central system active*

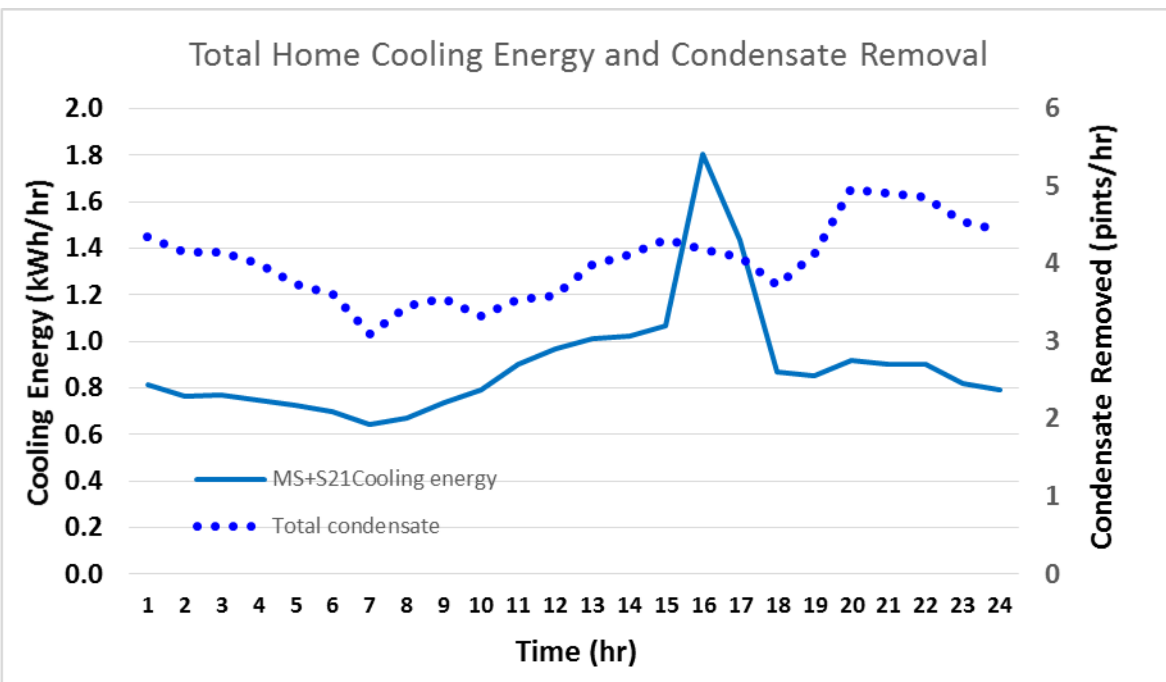


*Figure 7. Test 1 total combined cooling energy and condensate removal per hour shown for May 10; 130 cfm OA to MS and S21 central system active*

The latent removal is primarily from the mini-split. The variability in the latent removal throughout the day is due to the variable capacity delivered and variability in the SHR. The increase in latent removal starting at about 8pm is a result of the minisplit SHR dropping to about 0.65 compared to 0.76 at 6:45 am when the latent removal was at its lowest. The daily average SHR on this day was 0.71. The daily average indoor and outdoor conditions are shown in Figure 8.

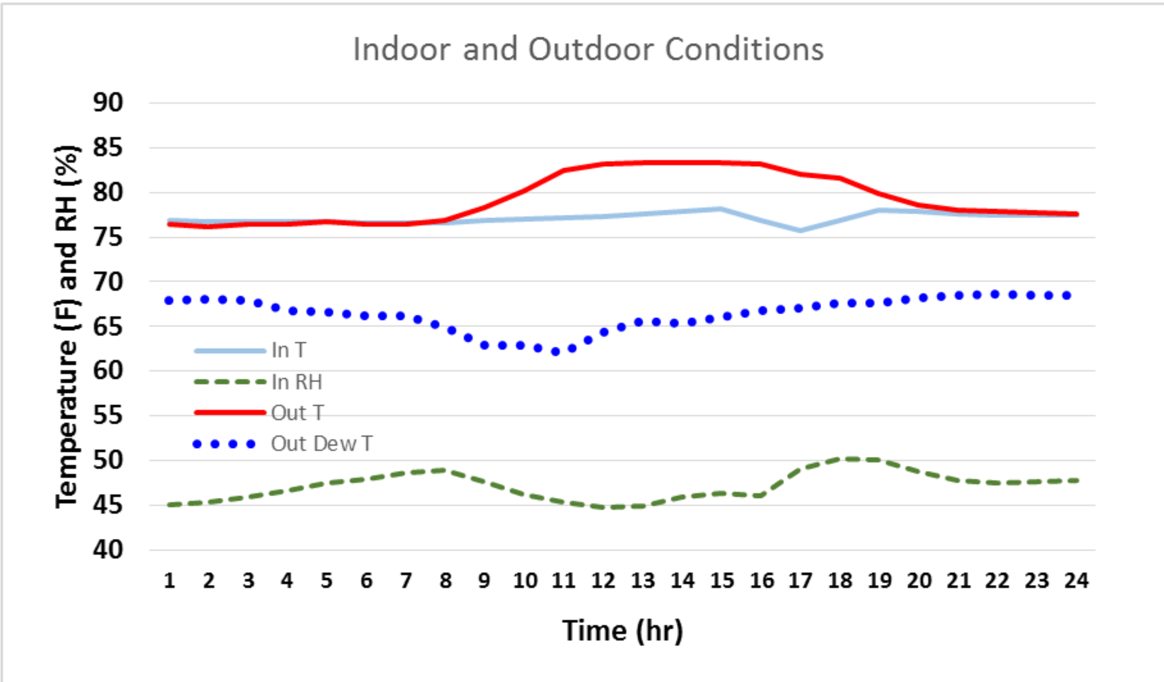


*Figure 8. Test 1 hourly average indoor and outdoor temperature, indoor relative humidity and outdoor dewpoint shown for May 10; 130 cfm OA to MS and S21 central system active*

Daily energy and latent removal results for test Configuration 2 with 60 cfm OA to the MS return and SEER 21 central system has the energy use and latent heat removed for each system shown in in Figure 9. The combined impact of both systems is shown in Figure 10. Data shown occurred on May 13, 2014. This configuration used a total of 20.8 kWh of cooling energy (MS+S21) for the day and removed a total of 80.3 pints (10.0 gallons) of latent load. The daily average indoor and outdoor conditions are shown in Figure 11.

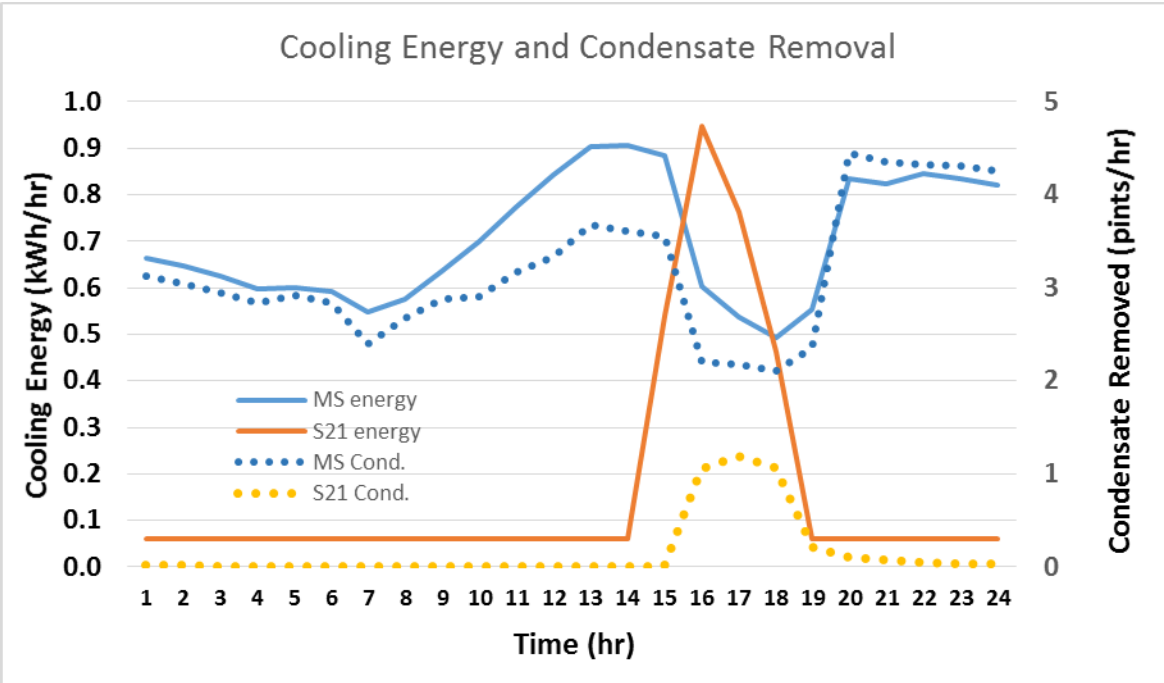


*Figure 9. Test 2 cooling energy and condensate removal per hour for each system shown for May 13; 60 cfm OA to MS and S21 central system active*

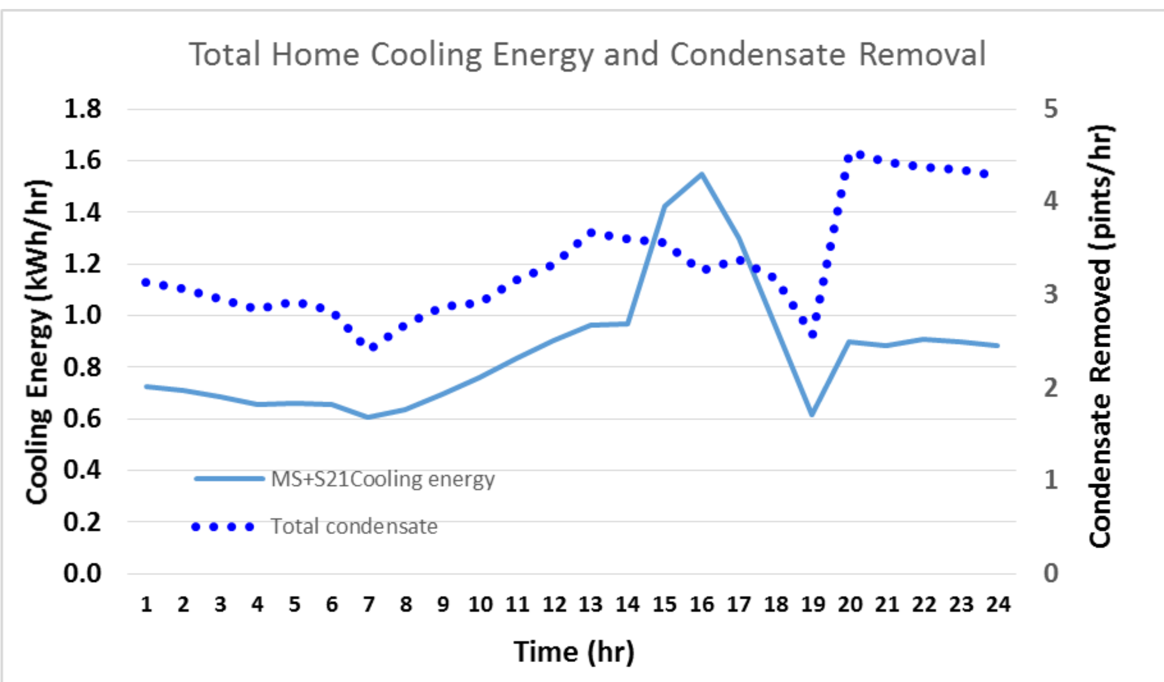


*Figure 10. Test 2 total cooling energy and condensate removal per hour shown for May 13; 60 cfm OA to MS and S21 central system active*

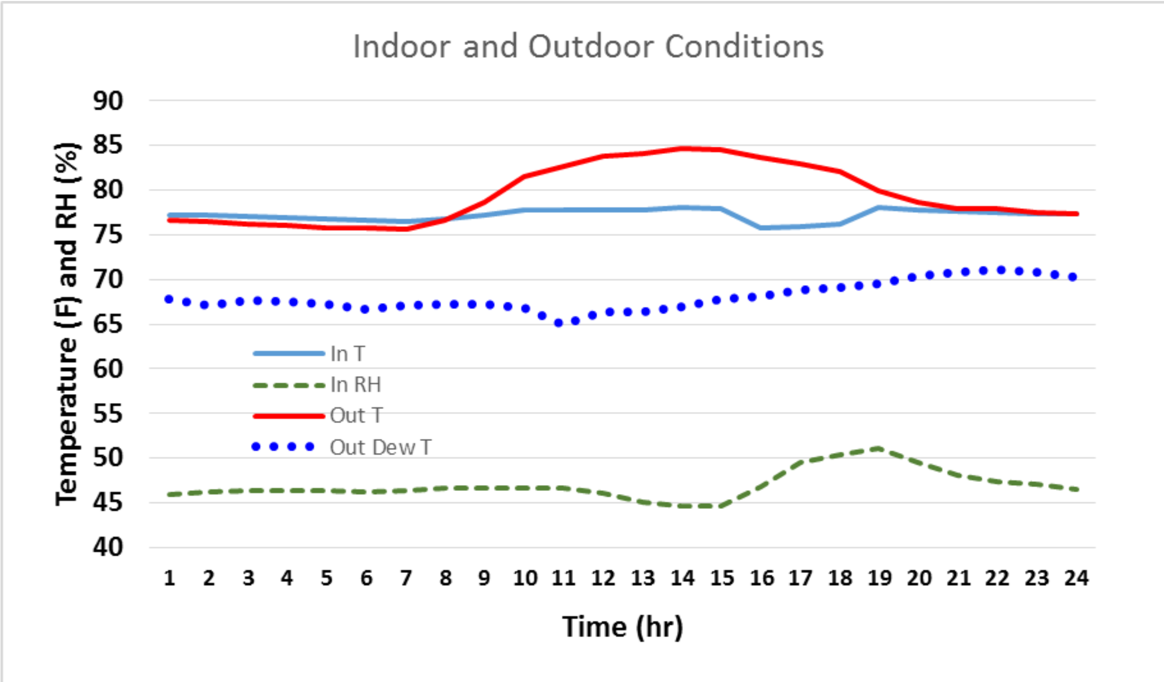


*Figure 11. Test 2 hourly average indoor and outdoor temperature, indoor RH, and outdoor dewpoint shown for May 13; 60 cfm OA to MS and S21 central system active*

Daily energy and latent removal results for test Configuration 3 with MS off (not permitted to turn on) and 60 cfm OA captured at the SEER 21 central system return has the energy use and latent heat removed the central system shown in in Figure 12. Data shown occurred on May 29, 2014. This configuration used a total of 18.5 kWh of cooling energy for the day and removed a total of 52.7 pints (6.6 gallons) of latent load. The daily average indoor and outdoor conditions are shown in Figure 13.

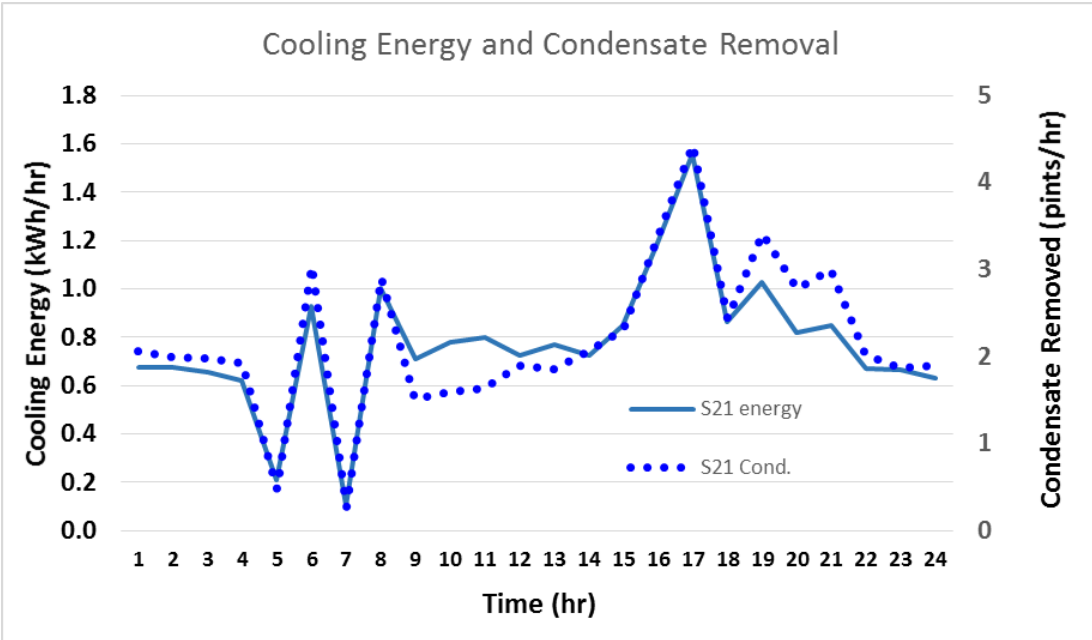


*Figure 12. Test 3 cooling energy and condensate removal per hour shown for May 29; 60 cfm OA to S21 central system*

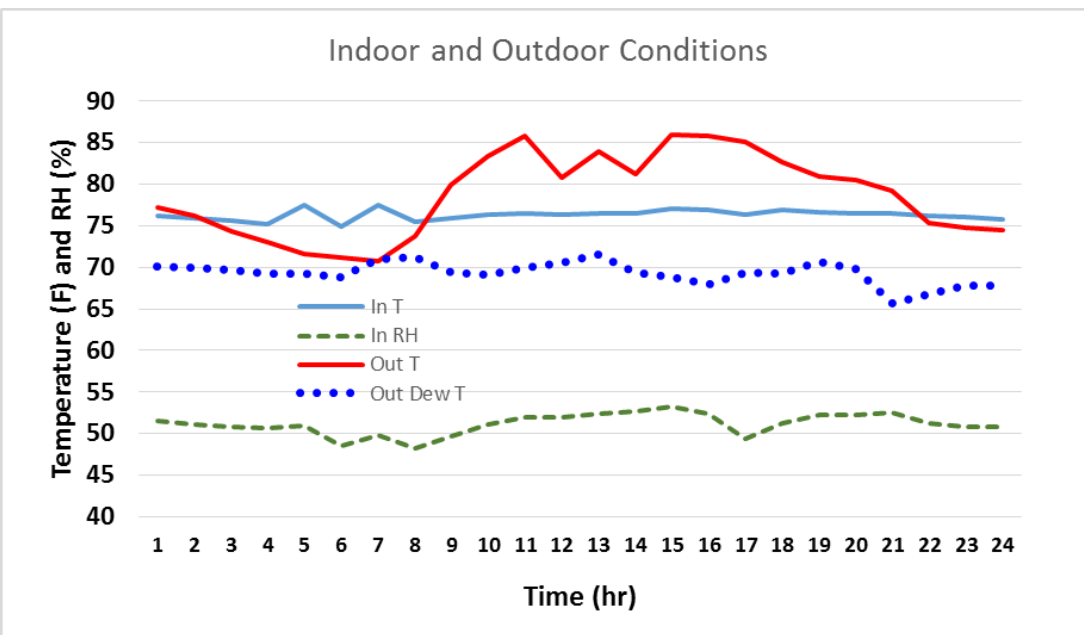


*Figure 13. Test 3 hourly average indoor and outdoor temperature, indoor RH, and outdoor dewpoint for May 29; 60 cfm OA to S21 central system*

Daily energy and latent removal results for test Configuration 4 with MS off and 130 cfm OA captured at the SEER 21 central system return has the energy use and latent heat removed the central system shown in in Figure 14. Data shown occurred on June 8, 2014. This configuration used a total of 20.1 kWh of cooling energy for the day and removed a total of 76.6 pints (9.6 gallons) of latent load. The daily average indoor and outdoor conditions are shown in Figure 15.

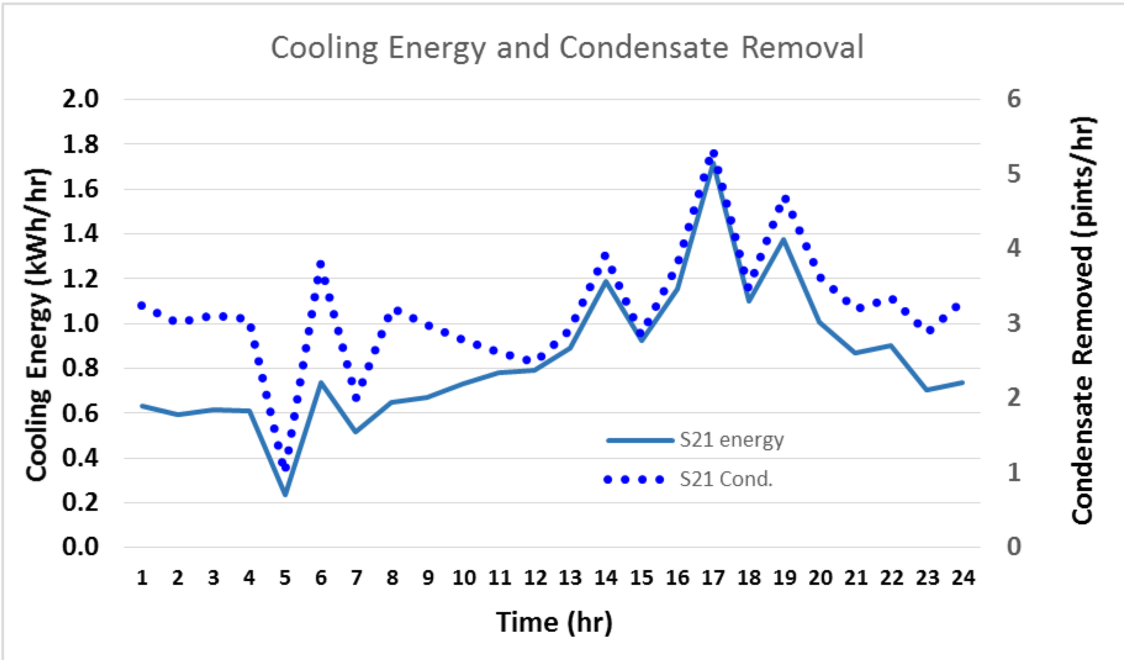


*Figure 14. Test 4 cooling energy and condensate removal per hour shown for June 8; 130 cfm OA to S21 central system*

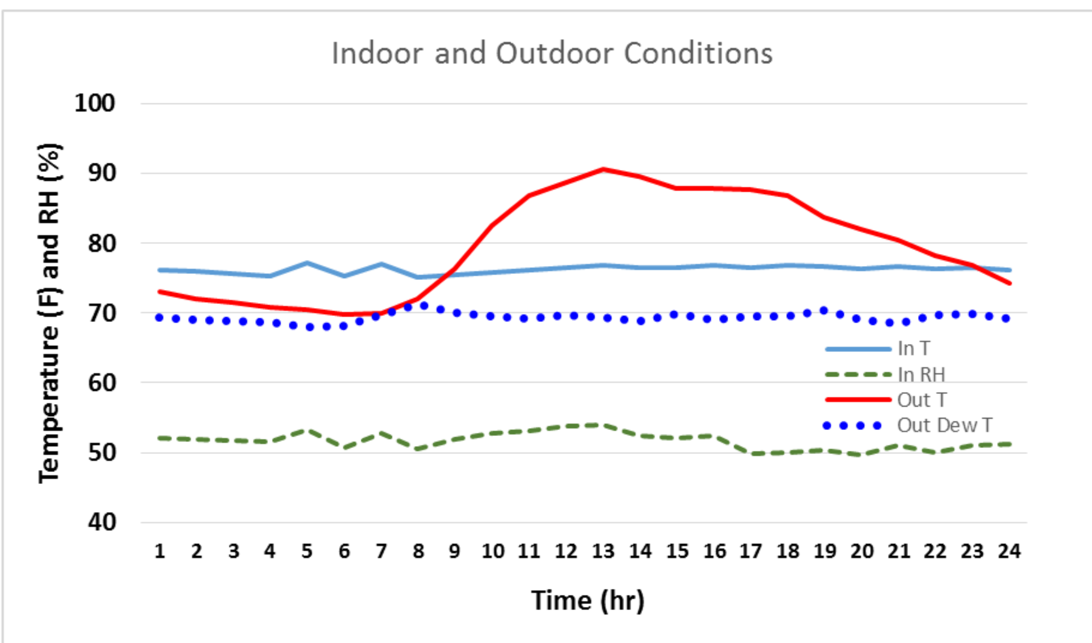


*Figure 15.Test4 hourly indoor and outdoor temperature, indoor RH, and outdoor dewpoint shown for June 8; 130 cfm OA to S21 central system*

Predicted Seasonal Energy

Linear regression analysis has been performed to characterize the relative cooling energy consumption (kWh/day) versus delta-T (outdoor temperature minus indoor temperature) of the four different test configurations.

A plot of the data and regression results are shown in Figure 16. The best-fit line equations and coefficient of determination ($r^2$) are shown in the colored text boxes and are in the order from test 1 through test 4 from highest to lowest respectively. Table 4 presents the predicted energy for a seasonal summer day having a dT of 5°F using the best-fit line equations. A dT of 5°F is used based upon an average 77°F indoor condition and daily summer outdoor average temperature of 82°F (82-77=5). Also shown in Table 4 are the standard error and $r^2$.

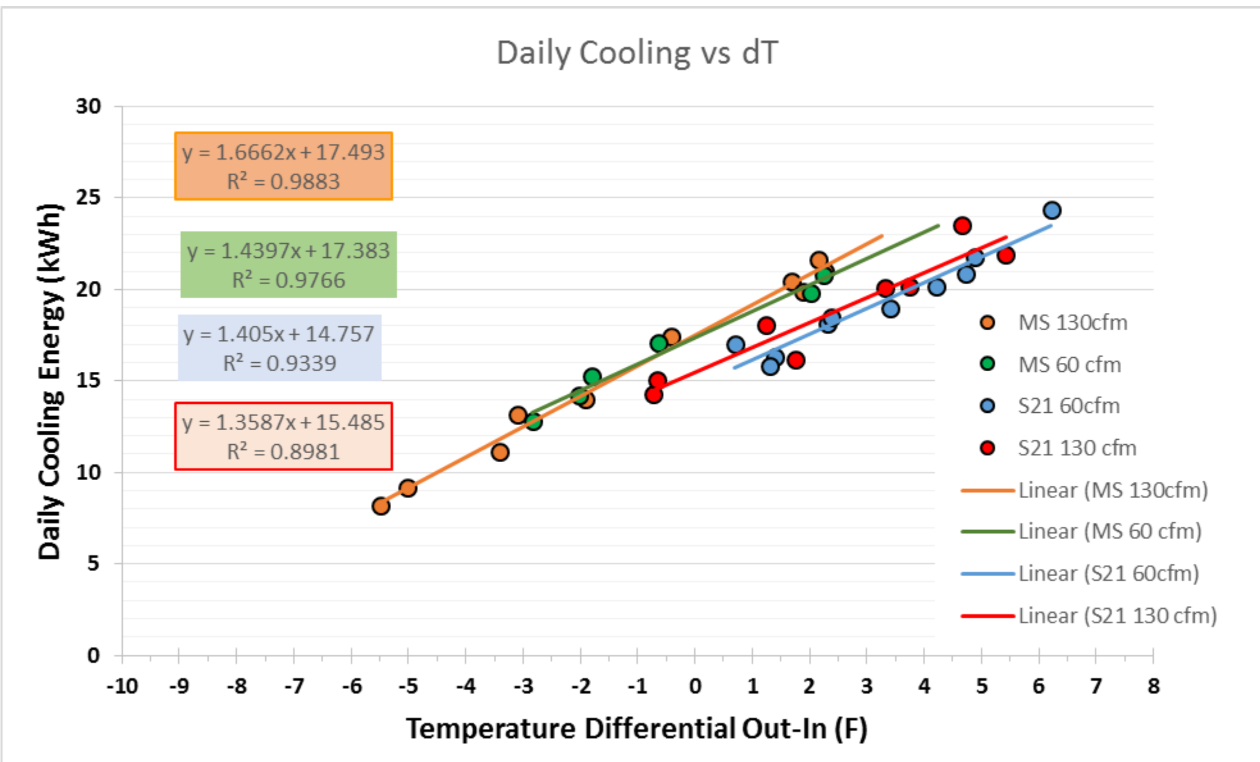


*Figure 16. Daily cooling energy versus the daily average temperature difference between outdoors and indoors.*

*Table 4. Predicted seasonal energy for a typical summer day at dT=5 shown with standard error and $r^2$*

| Test configuration | Predicted Daily Cooling Energy (kWh) | Standard Error | Coefficient of determination ($r^2$) |
|---|---|---|---|
| 1 MS 130 cfm + S21 | 25.82 | 0.59 | 0.988 |
| 2 MS 60 cfm + S21 | 24.58 | 0.54 | 0.977 |
| 3 S21 60 cfm | 21.78 | 0.72 | 0.934 |
| 4 S21 130 cfm | 22.28 | 1.15 | 0.898 |

Based upon the four test experiments the following summary can be made:
Impact of changing ventilation rates

- For the test configurations 1 and 2 which had ducted OA to the mini-split and used the central SEER21 cooling system, decreasing ventilation rate from 130 cfm to 60 cfm resulted in 1.24 kWh/day (4.8%) lower space conditioning energy use. Conversely it could also be stated that increasing the OA rate from 60 cfm to 130 cfm increased energy by 5.1%
- For the test configurations 3 and 4 which had ducted OA to the return of the central SEER21 cooling system, decreasing ventilation rate from 130 cfm to 60 cfm resulted in 0.50 kWh/day (2.2%) lower space conditioning energy use. Conversely it could also be stated that increasing the rate from 60 cfm to 130 cfm increased energy by 2.3%.
- There is an indicated difference in impact of ventilation rates on the MS+S21 test about 2.5 times that of just using the S21 central system (1.24 kWh/day / 0.5 kWh/day=2.5), however given the small percent differences, average standard error of 0.75, and the lack of test 1 and 2 data at dT=5, the difference may not be significant.

Difference between latent control methods

- Both methods worked well in controlling indoor RH below 60% RH.
- Using the minisplit with the SEER 21 central ducted system provided lower indoor RH, but used more energy.
- At the 130 cfm level of OA, test 1 (minisplit with the SEER 21 central system) used 3.55 kWh/day (15.9%) more energy than test 4 where the OA was captured at the return of the central unit and MS was off.
- At the 60 cfm level of OA, test 2 (MS + S21) used 2.80 kWh/day (12.8%) more energy than just test 3 (S21 and MS off).
- It is believed there is more energy in using the MS primarily due to shift of most of building cooling load to MS operating at a lower avg. efficiency, but at the benefit of better latent removal. From Table 3 it was shown that using the MS had an average daily removal of 23.9 pints (36.9%) more than using OA ducted to the SEER 21 unit. Using the MS results in an average difference in indoor RH 4.3%RH (51.4%RH-47.1%RH= 4.3%RH) which is 8.4% lower.

## Conclusion

A literature review of indoor RH control methods in homes was completed. Resources most relevant to addressing Florida's climate were of primary focus. From this review several dehumidification options have been identified, but there are several factors that have to be considered in selecting one, therefore no single measure can simply standout. Factors to be considered are:

- Health needs of occupants
- House design
  - Variability of solar exposure throughout the day and year
  - Availability of physical space to locate supplemental dehumidification equipment
  - Type of construction and interior finishes- Homes with minimal internal moisture capacitance will have faster rates of change in interior RH (finished concrete or tile floors and minimal soft cloth-type furniture and furnishings).
- Occupant driven operations
  - Indoor temperature set point preferences (low cooling temperature can result in high wall surface humidity under certain conditions even if occupant is comfortable)
  - Excessive runtime of bath or large kitchen exhaust
  - Large internal latent load generation
- Personal comfort / ascetic preferences
  - No supplemental dehumidification may be desired if house interior is kept relatively warm and there is a preference for elevated humidity.
  - Preference for equipment to be unseen and unheard may make some options less desirable.
- First cost and operational cost

Overcooling and reducing fan speed are the simplest and lowest first cost RH control strategies, but are not adequate to maintain indoor RH below 60% all the time. This may not be an issue for some occupants or buildings materials more tolerant of elevated indoor RH part time. Other technologies such as sub-cool reheat, and full-condensing reheat, have been proven to work in commercial applications, but only work as long as the central cooling system operates and therefore some hours >60%RH can be expected. DX condenser-regenerated desiccant dehumidifier may also work very well at minimizing, if not eliminating, all hours of RH>60%. While the last three technologies mentioned are expected to control RH reasonably well, there was inadequate published data found on measured operational performance in residential applications in hot humid climates.

These technologies should be tested and evaluated to collect data in residential applications occurring through Florida's cooling seasonal conditions to obtain more confidence in the operational costs.

It appears that there is much promise for SEER21 variable capacity central systems to offer an energy-efficient method of combined space cooling and supplemental dehumidification in mechanically ventilated buildings. At a high OA flow rate of 130 cfm captured at the return of the central unit, the highest indoor RH reading on a warm moist day was 54.0%. This is not to say that there would never be any hours with indoor RH>60% throughout the year, but they are expected to be few given that these systems can operate as low as 40% of nominal capacity. This method used about 14% less seasonal daily energy than the tested method where OA is captured at the minisplit return and the SEER21 central system provides cooling as needed.

Work in this first phase had only enough time to evaluate two different methods at two different ventilation rates. The weather during the experiments offered very limited days with elevated dewpoints to work with. A second phase already proposed is important to evaluate more latent load approaches and collect more data for experiments during more variable weather. As discussed earlier, good RH control becomes more challenging during the cooling season on days when the outdoor dewpoint is greater than 65° F. Typical summer outdoor dewpoints run between 72°F-76°F, but outdoor latent load is just beginning to increase May through early June with lower dewpoints in the range 60°F-70°F. It is imperative that the second phase be completed to provide more expansive and meaningful results through November. This would provide much needed testing under the challenging outdoor conditions of lower outdoor sensible, but high latent loads such as a cooler day with outdoor dewpoint of 72°F.

Based upon phase 1 results and input from the Florida Energy Technical Advisory Committee meeting held on June26, the next recommended test configurations for Phase 2 would be to evaluate the effectiveness of:

1. 60 cfm OA ducted to MS with a low efficiency SEER 13 central ducted fixed capacity system. Phase 1 experiments found operating the MS with the SEER21 central system used more energy but, given the much lower efficiency of the SEER13, this proposed Phase 2 Test 1 may result in lower operating cost than running the S13 unit alone. Furthermore, based on results so far, the MS operation will result in lower indoor RH which could eliminate the need for an inefficient dehumidifier.
2. MS off and 60 cfm OA to return of SEER 13 central ducted fixed capacity system
3. SEER 13 central ducted fixed capacity system with 60 cfm OA ducted to central zone (not captured by return). This will be an important base efficiency test to compare other results.
4. Collect swing season data for at least Phase 1 Test 3 at the 60 cfm OA rate. This test uses a vent rate close to average size homes and was the most efficient of Phase 1 testing.
5. SEER 13 central ducted fixed capacity with continuous exhaust ventilation. The exhaust ventilation rate would be at a rate greater than 60 cfm designed to evaluate the combined impact of mechanical OA ventilation along with operation of other exhaust systems such as clothes dryers and high flowrate kitchen exhaust systems.

*The actual number of test configurations that could be effectively evaluated in Phase 2 depend upon being able to begin testing in July 2014 and run through November 2014. Effective evaluations are also dependent upon adequate swing-season weather periods for each test configuration. Should limitations of time or available weather exist, then we recommend that priority be given to three tests. The three tests to be given priority are proposed above as Phase 2 Test 1, Test 2, and Test 5*.

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