FLORIDA SOLAR ENERGY CENTER®

Final Report

Further Investigation of Energy and Performance Impacts of Whole-House Dehumidifier Duct Configurations

DBPR Project #B3EB44
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Submitted to:

Mo Madani
Florida Department of Business and Professional Regulation
2601 Blair Stone Road
Tallahassee, Florida 32399

Submitted by:
Charles R. Withers, Jr.

1679 Clearlake Road, Cocoa, FL 32922-5703 • Phone: 321-638-1000 • Fax: 321-638-1010
www.fsec.ucf.edu

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Executive Summary

Dehumidifiers (DHU) are the most commonly relied upon appliance, supplemental to air conditioning, used to help control indoor relative humidity (RH) in Florida homes. They can offer the lowest first-cost, are well-established in the market, and are often easier to install than other alternatives; however they have the potential to use a lot of energy and first-cost and life-cycle cost can vary widely. Dehumidifiers may be designed to be ducted or un-ducted. DHU with ducts are sometimes referred to as whole-house or ducted dehumidifiers. Dehumidifiers that are not designed to be ducted may be known as room or space dehumidifiers and sometimes as stand-alone dehumidifiers. This project evaluated potential energy impacts of ducted DHU duct configurations.

There are several different options from ducted DHU manufacturers on how to duct these systems without any guidance on performance impacts. Also sorely missing are expanded DHU performance metrics to help professionals and consumers determine the appropriate DHU capacity and predicted real application efficiency.

The Florida Solar Energy Center Building Science Lab was used to conduct performance testing. The central cooling was controlled by a thermostat set at 76°F and a rated 70 pint per day DHU was controlled by a wall-mounted dehumidistat set at 50% RH. This report covered lab-based experimental study conducted in two phases to evaluate AC and DHU performance based upon how a DHU was integrated with central AC system ducts and compared this to a DHU run without ducting to central system ducts. A total of five different DHU lab test configurations were completed in two phases. The first three tests occurred during the first phase with initial results reported on June 2018 (Withers et al. 2018) and the last two tests were completed late May 2019. All ducted DHU test configurations were as follows:

1. Test 1 DHU air ducted from/to the central main body of building.
2. Test 2 DHU air ducted from/to return side of central air conditioner (AC).
3. Test 3 DHU air ducted from/to supply side of AC.
4. Test 4 DHU air ducted from central main body and ducted to supply side of AC.
5. Test 5 DHU air ducted from outdoors + central main body and ducted to supply side of AC.

Tests 1-4 were the most directly comparable since they had tighter control over the latent load of the test building. Test 5 ran mechanical ventilation through the DHU since this is used by some homebuilders as a means to ventilate and control indoor RH using one appliance.

Although some DHU manufacturers show it as an option, Test 2 is not recommended. This project found two highly significant detrimental impacts of Test 2 upon dehumidification performance. First, DHU supply air into the central return duct upstream decreased central AC latent performance by 28% when the DHU and AC ran simultaneously. Second, the DHU evaporated moisture off of the central cooling coil at a rate of 2-3 pounds of water per hour when the AC was cycled off. This was about the average rate of the measured DHU moisture removal under actual test conditions. Under these specific circumstances, the Test 2 DHU configuration performed about as well as electric strip heat with the inefficient leverage of increased temperature that lowers RH and increases cooling load.

In addition to the poorer dehumidification performance, the annual predicted space conditioning (DHU + AC) energy use was about 308 kWh/year (4%) more than Test 1. This would cost a resident about $37 per year more. The DHU used in this project was at least twice the rated efficiency for a 70 pint per day capacity DHU compared to the lowest available 70 pint per day DHU. A Less efficient DHU, would result in substantially more DHU and AC energy use and the energy impact from Test 2 could double to 8% and about $80 per year increased energy cost if the lowest efficiency DHU were used.

There was very little difference in annual predicted energy use of Tests 1, 3, 4. Test 5 indicated about 8% more energy use than Test 1 after adjusting for the fan energy use associated with mechanical ventilation that occurred in Test 5 that did not occur in the other tests. Some of the weather that occurred in Test 5 caused a latent load about 9% higher during warmer weather periods than occurred during the controlled latent loads of Tests 1-4. This would account at least for most of the 8% difference from Test 1.

The steady-state and longer-term test findings show that DHUs should not be ducted from/to a central cooling system return upstream of the central cooling evaporator coil (Test 2). The DHU from/to central supply ducts (Test 3) steady-state testing showed a decrease in DHU power consumption, and improved latent performance, however this improved DHU performance only occurred when both AC and DHU operated simultaneously for at least 15-20 minutes. Longer-term testing, that included cycling-related performance, showed an insignificant difference between the annual energy use of Test 3 and Test 1.

The lab test results reported here are from specific environmental conditions and at specific control setpoints. The severity of impact in homes will depend upon particular cooling temperature setpoints and DHU humidistat setpoints. Higher temperature and humidity setpoints will result in less AC and DHU operations and generally lower impact. This study did not evaluate impacts at different setpoints. It should also be considered that the ducted DHU used in this research was about the smallest and most efficient DHU ducted unit on the market. A larger rated pint capacity DHU or a lower efficiency 70 pint DHU ducted from/to the same tested central return system would have had worse results upon the DHU from/to central return compared to the DHU used in the tests.

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Introduction

As home energy efficiency increases, cooling loads decrease and the total hours of air conditioning also decrease. This raises the potential for elevated indoor RH during low cooling load periods without some form of supplemental dehumidification. Supplemental dehumidification can require a significant amount of energy use. In some low load homes with very efficient air conditioning (AC), supplemental dehumidification may use as nearly as much annual energy as central cooling (Withers 2018). Another concern is that there is no standard for addressing the energy use of supplemental dehumidification leaving consumers no way to expect the impact of different options. As stated by Vieira and Beal 2017, "*…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*."

There has been very little study of energy impacts of integrating dehumidifiers with central cooling. A review of a few ducted dehumidifier unit (DHU) manufacturer installation manuals found several different recommended ways to duct DHU. Installation manuals and a lack of third-party published research do not address any potential DHU duct configuration impacts upon central cooling or DHU energy performance.

Some fundamental qualities of DHU and AC are offered here that may help some readers better understand why this research project was conducted.

- DHU consume electric power. Smaller ducted units consume about 500-600 Watts.
- DHU is controlled by a dehumidistat that senses relative humidity (RH).
- DHU removes moisture from the air.
- More sensible heat leaves the DHU than cooling (warm/hot air is delivered into home).
- The combination of removing some moisture and adding heat helps lower indoor RH.
- The DHU performance is affected by the moisture and temperature conditions that enter the DHU. There is very little published DHU performance data at different entering conditions.
    - Based on one manufacturer's published data, DHU moisture (latent) removal performance decreases about 20% from entering air at 80°F/60%RH to 70°F/60%RH.
- Central air conditioning performance is also affected by moisture and temperature conditions entering the evaporator coil.
    - A review of several AC manufacturer expanded performance data tables show that latent performance decreases with less moisture (lower wet bulb temperature) in air entering evaporator coil than with higher entering moisture.

In the fall of 2017, the Florida Building Commission approved an initial phase of research for FSEC to evaluate the impacts of three ducted DHU configurations. Further investigation was approved for FSEC to evaluate two additional ducted DHU configurations one year later. The work was sponsored by Florida Department of Business and Professional Regulation. This report focusses on the two additional ducted DHU performance tests, provides updated analysis from the previous 3 tests (Withers et al. 2018), summarizes results from all five tests, and provides recommended building code modification.

The primary research purpose of this project was to determine which whole-house ducted DHU configurations provide the best performance and lowest energy use. The electric energy use of central air space conditioning and DHU as well as latent dehumidification performance was evaluated and compared to last year's research results (Withers et al. 2018).

Contracted Scope of Work

The Florida Department of Business and Professional Regulation (DBPR) established a contract with the Florida Solar Energy Center (FSEC) to perform a comparative study of attic performance between a lab home with a vented attic and same home with sealed attic vents.

The contracted scope of work covered in this final report is summarized below in the following items:

1. FSEC shall alternate the method of DHU duct configurations identified in Scope of Work item (2 a.) and (2 b.). Testing will be completed using the Building Science Lab building on the FSEC campus. A wall-hung dehumidistat control will be used to control the DHU at the indoor RH setpoint.
2. In a laboratory, alternate method of dehumidifier air distribution for specified test cases. DHU ducts will be configured in the following ways:
    a) DHU entering air (DHU return) from conditioned space and DHU supply air ducted into central heat/cool supply duct (Referred to as Test 4 in this report Figure 4)
    b) DHU entering air with about 70 cfm airflow rate from outdoors mixed with 90 cfm from indoors. The DHU supply air will be ducted into central heat/cool supply duct (Test 5, Figure 5).
3. Write a final report (to be delivered by June 1) with results, combine with previous Withers et al. 2018 study results, and offer additional recommendations if warranted.
    a) Results will include a predicted annual DHU and space cooling energy use based on energy monitoring that can be compared to previous Withers et al. 2018 results.
    b) Results will include an evaluation of duct configuration upon DHU performance.
    c) Update Withers et al. 2018 report summary with new findings in final report.

This document is the final report deliverable that includes measurement, analysis and reporting on recent Tests 4 and 5 as well updated analysis for Tests 1-3 conducted last year.

Background

This background was modified from the report for Tests 1-3 (Withers et al. 2018) and is relevant to all five DHU duct configuration tests. It provides added context to better understand the ducted DHU test experimental set-up and results and why it should be considered as a building code matter.

Central cooling systems that are designed and installed well, work generally well at cooling and dehumidifying air as long as there is adequate sensible load to cause the system to run long enough to remove moisture close to the rate of generation. The need for supplemental humidity control arises as the sensible cooling load (dry bulb temperature) decreases relative to the latent load (water vapor). Sensible cooling loads are lowest during overnight periods as well as during spring and fall seasonal conditions. Latent loads are influenced by internal and external moisture sources.

Sensible and latent loads can be independent from each other and are affected by several factors. Such factors include variability in outdoor dry bulb temperature and moisture levels, natural and mechanically induced air infiltration/ventilation rates, internal generation rates of sensible and latent loads, sensible and latent capacitance of materials, and the cooling performance of HVAC used within a home. The first three items address the load rate. The next item, materials capacitance, addresses how well interior materials acquire and release load to the indoor air, and the last item, HVAC performance, addresses how well equipment removes sensible and latent heat.

Internal latent comes from activities such as cooking, bathing, as well as from respiration and perspiration. External latent is primarily transported through natural infiltration, and induced from mechanical equipment. Mechanically-induced latent may be transported into home through the use of exhaust fans, mechanical ventilation, and even by air distribution duct leakage.

Consider an example of a home mechanically ventilated overnight during warm moist weather. About 85% of the cooling load associated with air entering from outdoors is latent heat and only about 15% is sensible load. The cooling loads are lowest overnight resulting in less cooling and less moisture removal at a time when moisture loads may be increasing. Infiltration or intended mechanical ventilation steadily increases the moisture load overnight and the indoor RH increases as a result. As cooling load increases, additional cooling begins removing more moisture and indoor RH drops.

Even internal moisture sources from cooking, bathing, and dishwashing as well as from occupant perspiration and respiration may be large enough to be significant sources of moisture that must be removed from the home. There can be enough moisture from these sources to result in elevated indoor RH during mild swing seasons when cooling loads are low (Hendron and Engebrecht 2010).

As homes are built to reduce external sensible cooling loads, and become adequately ventilated, the probability increases for more annual hours with elevated indoor relative humidity (RH) at or above 60% RH (Martin et al. 2018), (Withers 2016), (Henderson and Rudd 2014), (Rudd et al. 2005). This increases the likelihood that supplemental dehumidification will be needed to maintain indoor RH below 60%. The fact that many homes in hot humid climates do not use supplemental DHUs does not mean occupants are always satisfied with their indoor humidity. Indoor RH is low enough most of the time likely due that homes have less ventilation than recommended by ASHRAE 62.2-2013 standards. Controlled lab study found that indoor RH increased significantly during some overnight periods and required a DHU to maintain indoor RH below 60% when the FSEC Manufactured House Lab was mechanically ventilated according to ASHRAE 62.2-2013 (Withers 2016).

DHUs are the most commonly relied-upon device used to help control RH in homes. This is because they offer the lowest first-cost, are well-established in the market, and may be easier to install than other alternatives (Withers and Sonne 2014), (Rudd et al. 2002). Whole-house DHUs, while effective at controlling RH, can result in a significant amount of energy use that is not typically considered (Withers 2018), (Mattison and Korn 2012). Current Florida energy code does not consider whole-house DHU energy use in space conditioning compliance. This may be worth considering in the future. One past study showed significant difference in HVAC energy depending upon how and where a DHU was installed (Rudd et al. 2005), however this study occurred in Texas and did not focus specifically on the research focus of this project.

Experimental Method

Lab and Equipment Description

This section discusses details about the test building, equipment details and data collection procedures. All test configurations were conducted within the Building Science Lab building located on the Florida Solar Energy Center campus in Cocoa, Florida.

Lab Test Method

Previous Tested DHU Duct Configurations

Three DHU configuration experiments were completed last year to evaluate the energy performance of each test configuration. Lab test configurations evaluated were:

1) Test 1 DHU air ducted from/to the central main body of building.
2) Test 2 DHU air ducted from/to return side of central cooling (AC).
3) Test 3 DHU air ducted from/to supply side of AC.

Conceptual illustrations of Test 1-3 are shown in Figures 1-3 respectively. Greater detail on Tests 1-3 can be found in the final report (Withers et al. 2018).

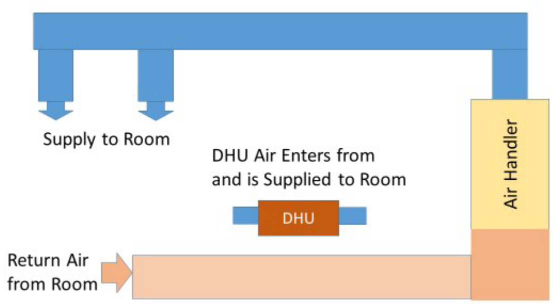

Figure 1. DHU air directly from and back into conditioned space. (Test 1)

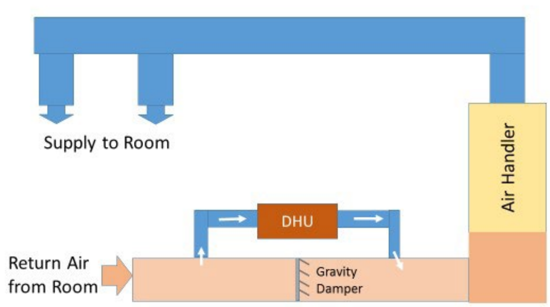

Figure 2. DHU ducted to the main central return duct. Gravity damper only opens when central system on. When closed, it blocks short-circuiting of DHU supply air straight back into DHU return. (Test 2)

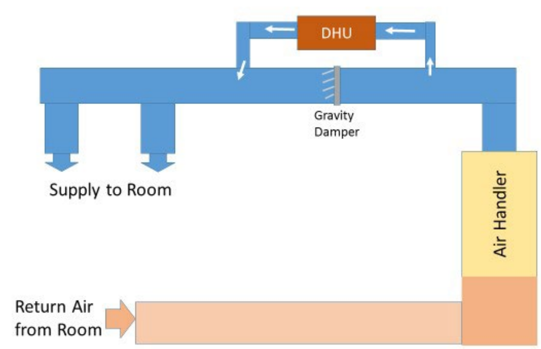

Figure 3. DHU ducted to the main central supply duct. Gravity damper only opens when central system is on. When closed, it blocks short-circuiting of DHU supply into DHU return. (Test 3)

New DHU Test Configuration Tests

The two recent DHU test configurations tested are referred to Test 4 and Test 5.

Test 4 is similar to Test 3 from last year with the DHU supply air ducted into the central supply, except the return air to the DHU comes directly from the indoor central room. An illustration of this test concept is shown in Figure 4. This test configuration has been reported as a preferred option used by some Florida contractors.

Test 5 in this project is different from all other tests since part of the DHU entering air comes from outdoors and mixes with indoor air before entering the DHU. The DHU supply air was ducted to the central air duct in the same manner as Test4. An illustration of this test concept is shown in Figure 5. This test mode was chosen since some builders in Florida are using this method as a means to provide adequate whole-house mechanical ventilation in tight homes and maintain acceptable indoor humidity using one appliance.

Test 5 representing the ventilating dehumidifier concept was operated in a way that would comply with ASHRAE 62.2-2016. ASHRAE62.2-2016 accounts for the number of occupants a home is designed for and allows the estimated natural infiltration (Qinf) to be applied towards the total required ventilation. The balance of the requirement must then be provided mechanically (Qfan). Test 5 was run assuming a design occupancy of 4 persons and the laboratory home had a measured airtightness of 2.4 ACH50. Based upon this, Qinf was calculated as 20 cfm and the required continuous fan flowrate for ventilation (Qfan) was 70 cubic feet per minute (cfm). The average dehumidifier total airflow rate when ducted was about 160 cfm. The remaining 90 cfm of dehumidifier air came directly from the central indoor room.

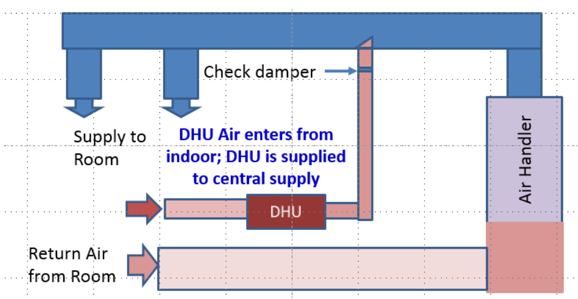

Figure 4. Illustration of DHU supply ducted to the main central supply duct. Gravity damper only opens when DHU is on. DHU air enters directly from indoor central room. (Test 4)

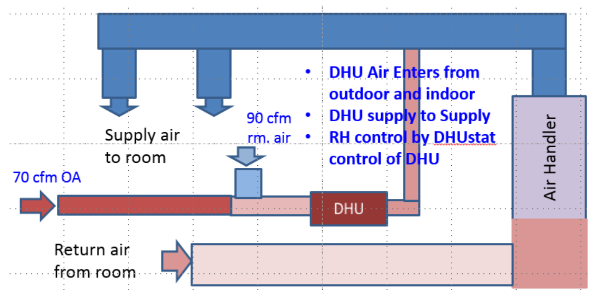

Figure 5. Illustration of a ventilating DHU design. DHU supply ducted to the main central supply duct. A portion of DHU air enters from indoor central room and the rest from outdoors. (Test 5)

Duct modifications suitable for both Test 4 and Test 5 began in September 2018 and then provisional testing of Test 5 (ventilating DHU) was begun. Some modifications were made during the provisional testing period to assure the correct airflows from outdoors and indoors were acquired. Each test configuration was conducted over time with rotations between both tests in an effort to acquire as much variation in weather conditions for each test as possible. Rotation through Test 4 and Test 5 continued well into May 2019 to enable as much weather variation as possible for each test.

Lab Building

This lab has a conditioned floor area of 2000 ft^2 with concrete masonry block walls having R-5 un-faced foam board insulation located on the interior side of the wall. Windows were single pane clear glass set in metal frame. Ceiling insulation was R-19 batt. Building airtightness was tested using a blower door and measured a normalized air leakage rate of 2.4 ACH50. There was no measurable duct leakage to outdoors (CFM25out=0). A manual J8 load calculation on the tested building calculated a summer 99% design total cooling load of 2.3 tons. The floor plan is shown in Figure 6.

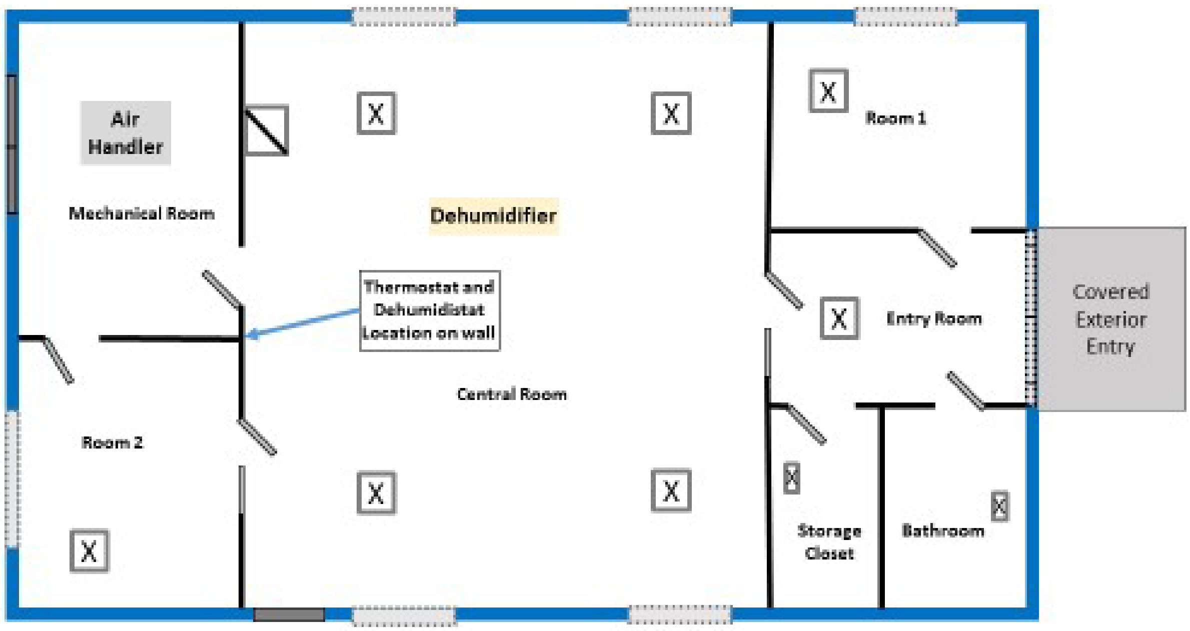

Figure 6. Building Science Lab floorplan.

Interior Loads

Internal loads were established using some 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 about 4,200 Btu/h. Based upon a Manual J8 sizing calculation, this is an amount appropriate for the installed central air conditioner during the testing configurations on a design day.

Latent loads were generated at three different target rates using a humidifier or hot plate. Fifteen lb/day interior generated latent was used to represent the base daily interior generated latent associated with occupancy. This amount is within the range of what is currently considered for simulation and lab research (Fan et al. 2011), (Hendron and Engebrecht 2010). Target rates of 15, 30, and 60 pounds of water each day were evaporated into the building and mixed within the central area of building by a small circulation fan. Latent delivery was monitored using calibrated water meters or tipping buckets on water supply lines. The rationale of when the three different rates of latent load generation were used is discussed later in the section, Space Conditioning Energy.

Tests 1-4 were intentionally conducted with no mechanical ventilation particularly since outdoor latent load varies significantly from fall through spring when the tests had to be completed. A longer test period of about a full year would be needed to better assure each test had similar latent load representation in the testing with mechanical ventilation. The controlled interior latent generation was a means to provide varying levels of latent load across a wide range of sensible cooling load. Since Test 5 intentionally ventilated and was conducted during through a range of outdoor latent load, the interior latent generation was held constant at 15 lb/day.

Based upon the measured building tightness for a 3 bedroom 2,000 ft^2 home, ASHRAE 62.2-2013 would call for a total ventilation rate of 90 cfm, of which 70 cfm would come from mechanical ventilation and 20 cfm from infiltration. Due to the highly variable moisture content in outdoor air in east central Florida during winter and spring, mechanical ventilation was not utilized. Instead, moisture was generated internally at a rate of 60 pounds per day. This rate was delivered as long as outdoor temperatures averaged around 68°F or greater. This moisture rate represented 48 pounds per day that would have come in from mechanical ventilation (at 70°F dp) and another 12 pounds per day internally generated by occupant activities.

Because 60 pounds of latent is abnormally high during cool weather, internal latent generation was reduced during cooler weather periods. Internal moisture was generated at a rate of 15 pounds per day generally when daily average outdoor temperatures were about 65°F or colder. Internal moisture was generated at a target of 30 pounds per day when daily average outdoor temperatures were between about 60°F-72°F.

Latent heat removed from the building as condensate drained from evaporator coils was measured using tipping buckets calibrated at the expected rates of flow for each application. The basis of determining tipping bucket calibration was by supplying a drip rate of water to each bucket where the number of tips were measured for a given measured mass of water (pounds). Latent coil performance was also evaluated using Vaisala HMP 60 temperature and relative humidity sensors before and after coils (aka entering and leaving conditions) along with the measured flow rate. Indoor and outdoor conditions were measured using Type T thermocouples and Vaisala HMP 60 temperature and relative humidity sensors. Temperature and RH were compared to a handheld Vaisala HM34 temperature and humidity sensor with NIST traceable calibration to verify that sensors were operating within manufacturer specifications.

Lab Space Conditioning Equipment

The central heat pump and ducted dehumidifier used in this project are the same units used in the first 3 tests from last year's project (Withers et al. 2018). The central ducted system was a SEER13 heat pump with a nominal rated cooling output of 2.7 tons, however fan operation at low flow setting and addition of gravity dampers within supply ducts resulted in measured delivered cooling at about 2.3 tons. The capacity was appropriate for the design building load.   The heat pump system was controlled by a thermostat located on an interior wall in the large open central room. The thermostat was set to maintain an indoor average of 76°F.

The whole-house ducted DHU used was an Ultra-Aire 70H model with rated efficiency of 2.4 liters/kWh and rated moisture removal of 70 pints per day at 80°F and 60% RH. This was 20% more efficient than the minimum qualifications of an ENERGY STAR® dehumidifier at 2.0 L/kWh. It was also about twice the efficiency of 70 pint/day DHU at the lower end of efficiency. This means that the energy use in the lab tests can be expected to be lower than homes that use lower efficiency DHU. Supplemental dehumidification was controlled by a dehumidistat located on a central interior wall near the central thermostat. The RH setpoint was at 50% RH. This activated the DHU at 50% RH until RH reached about 45% RH at which point DHU cycled off.

Average daily indoor as well as outdoor conditions from September 24, 2018 through May 19, 2019 are shown in Figure 7. It can be seen that the central heat pump system was able to maintaining setpoint throughout testing. The supplemental dehumidifier was also able to maintain indoor RH at or below 50%. There is one two-day period where indoor RH exceeded 50% about 65 days into testing. This particular period had occurred due to isolated condensate line leaks that wetted the floor around the leak area for two days. Extra air circulation and some local heat was used to dry the area. The latent load humidifier was turned off for about one day. Days impacted from this event were eliminated from any analysis.

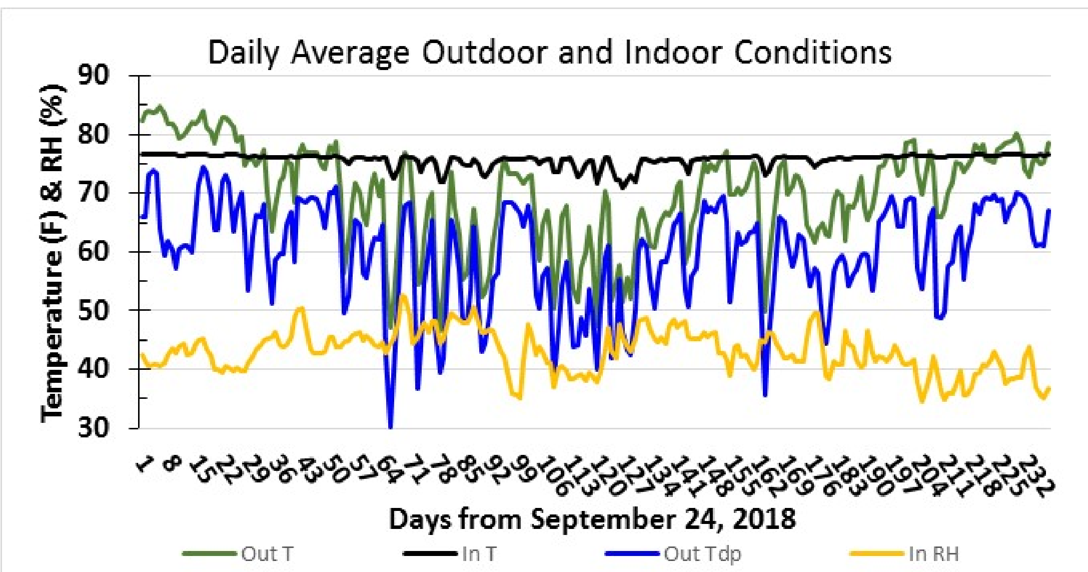

Figure 7. Daily average temperatures and RH September 24, 2018 through May 19, 2019.

Lab Sensors

All sensors for this project were installed and verified to be in good working order. Temperature and relative humidity (RH) sensors measured indoor, attic, and outdoor conditions. Power meters measured internal loads and space conditioning energy.

A summary of manufacturer stated accuracy of meters and sensors are below:

- Vaisala Temperature and relative humidity HMP60 sensors were installed. These sensors have a manufacturer stated accuracy of +/- 3% RH of RH reading and +/- 0.9 °F for temperature. Type T Thermocouples were also used to measure temperatures. These have accuracy of +/- 0.2°F.
- Continental Control Systems Wattnode power meters have a manufacturer stated accuracy of +/- 1% were installed to measure DHU energy, central AC system, and internal generated sensible loads.
- Condensate removal of AC system was measured by calibrated tipping bucket. Tipping buckets were calibrated by mass of water measurement collected along with the pulse output signal. Stated accuracy was 3% or better.
- Outdoor air temperature and humidity were measured by thermocouple and Vaisala sensors.

In all, 37 channels of data from sensors were collected using a Campbell Scientific, Inc. CR10 data logger. Data was gathered several times each day from FSEC's 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 data logger was screened for out of bound errors and then processed for terminal collection in the main project database account. Errors or missing scans were marked and noted within the main database. This is used to help avoid using any unsuitable data in analysis. Days during transition between test configurations or other interruptions to testing were also screened out from analysis.

Lab Test Results

Space Conditioning Energy

The total space conditioning energy use impacts of the two different test configurations was evaluated. The daily total energy use of the DHU and the central cooling system was combined to represent total daily space conditioning energy. This was plotted against the daily average temperature difference between outdoors and indoors. The temperature difference is sometimes noted as delta temperature (dT). Use of dT enables one to later predict energy use at specific indoor and outdoor temperatures. Data collection for Tests 4 and 5 began September 2018 and continued through May 19, 2019.

Figure 8 shows data from Test 4 having 100% of the DHU return air come from indoor central room. The DHU supply air was directed into the central supply main duct before any supply branches. The data were separated into three groups having different interior latent loads. This is consistent with the testing method during Tests 1-3 last year. The first group is with indoor latent load of 15 lb/day that represented the base interior latent associated with occupancy. The 15 lb/day latent load was reserved for weather that varied from cold up to mild cooling load. The second group was with a latent load of 30 lb/day during weather cool mild up to warm conditions. The last group was with indoor latent of 60 lb/day. The 60 lb/day latent delivery was reserved for days with warm to hot weather. Interior latent load rates were permitted to occur during overlapping weather periods to improve trend analysis.

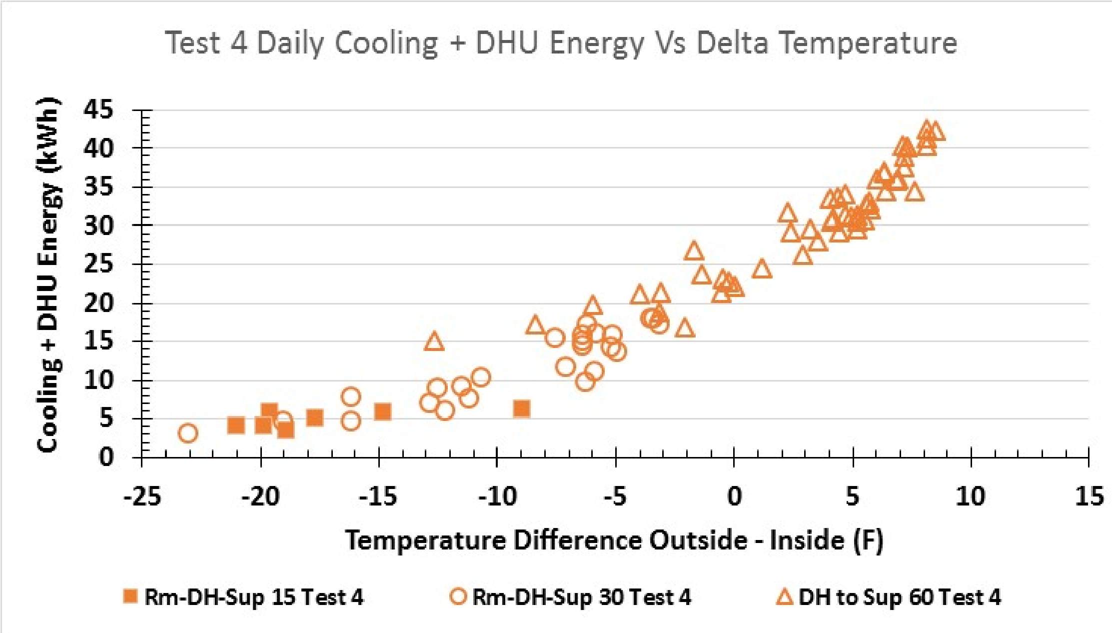

Figure 8. Test 4 data with DHU air entering from indoor central room and supplied into the central cooling system main supply. Data is differentiated at three different indoor latent loads.

Each test configuration had a least-squares best-fit regression analysis completed using daily total space conditioning energy versus the daily average dT. Figure 9 shows data from Test 5 with a second-order polynomial best-fit line.

Test 5 had 70 cfm air from outdoors mixed with 90 cfm of indoor room before entering the DHU. The DHU supply air was directed into the central supply main duct. Indoor latent was maintained at a constant 15 lb/day for this test configuration. The data show a representation from very cool to warm days.

Test 5 was similar to Test 4 except it was mechanically ventilated to ASHRAE 62.2 minimum ventilation requirement. This means Test 5 included 1.8 kWh/day in mechanical fan energy that Tests 1-4 did not. While Test 5 did maintain a minimum 15 lb/day interior latent generation, the actual outdoor latent load could not be controlled. Review of the data indicated that the outdoor portion of latent exceeded the controlled latent of Tests 1-4 by about 9% during warmer weather. So Test 5 results shown in Figure 9 include 1.8 kWh/day and at least 9% greater latent load during warmer weather with dT of 1 or greater. Therefore the regression equations of Test 5 should not be directly compared to those of Tests 1-4 without some adjustment. The impact of additional cooling load imposed by Test 5 was accounted for in the final annual energy analysis to enable a more equitable comparison.

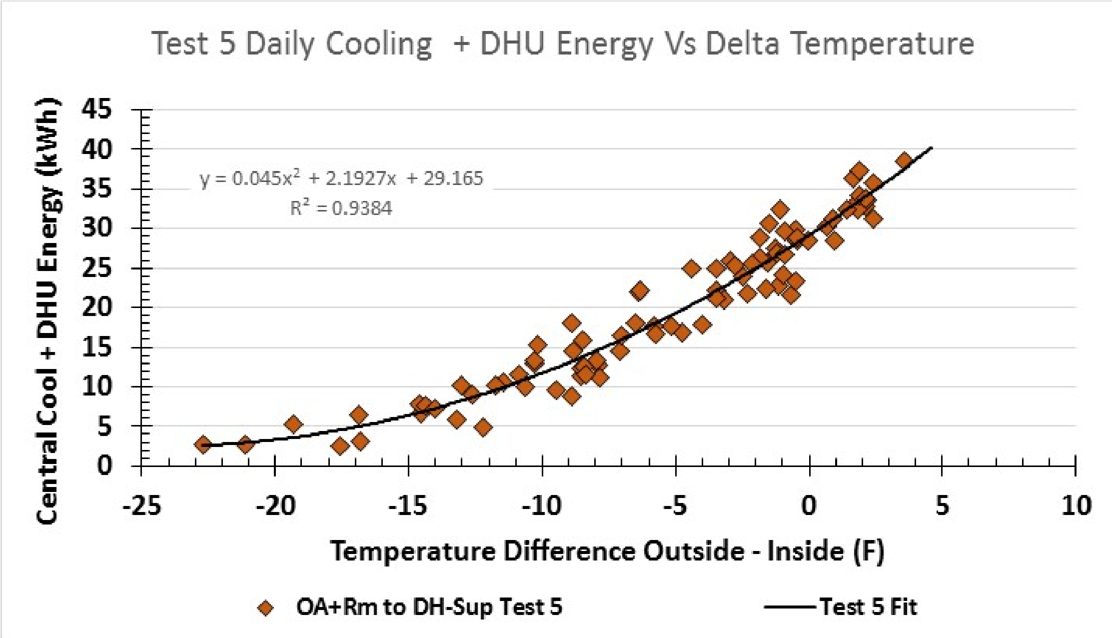

Figure 9. Test 5 data with DHU air entering from outdoors and mixing with indoor air. The DHU supply air is delivered to the central cooling main supply. Indoor latent was maintained at a constant 15 lb /day.

In these lab tests with a SEER13 rated cooling system, cooling energy represented the greatest fraction of the energy use over the year. Figure 10 shows all Test 1-4 configurations as one group to compare how the daily DHU energy as a % of the total (dotted black line) drops as the outdoor temperature increases. The total energy includes standby energy so at very cool temperatures, the cooling did not operate but still had about 0.75 kWh/day so DHU is not indicated as 100% in these cases.

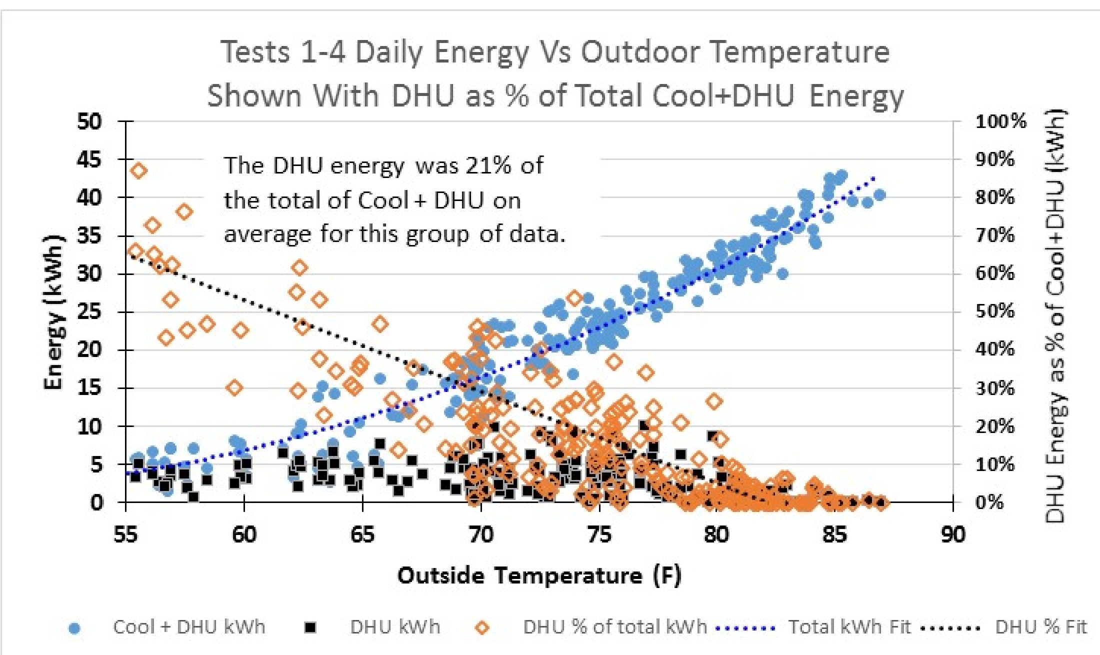

Figure 10. Tests 1-4 as one group distinguishing daily Cool + DHU energy use from the DHU energy.

The data in Figure 10 is more representative of a minimum code home. The DHU % of total would be higher for high performance homes with very high SEER cooling that operates at a high sensible heat ratio and had mediocre dehumidification. Under this scenario, annual DHU may represent as much as the space conditioning energy.

Predicted Annual Energy

Least-squares best-fit regression analysis was performed to characterize the cooling and DHU energy consumption (kWh/day) versus delta-T (outdoor temperature minus indoor temperature) of Tests 1-5. Greater detail on the regression analysis and method to develop the representative energy equations are provided in Appendix B. The resulting regression best-fit line and equations for Tests 1-5 are shown in Figure 11. Test 5 result in Figure 11 reflects adjustments made to account for extra cooling load imposed at dT>= 1 that did not occur during Tests 1-4.

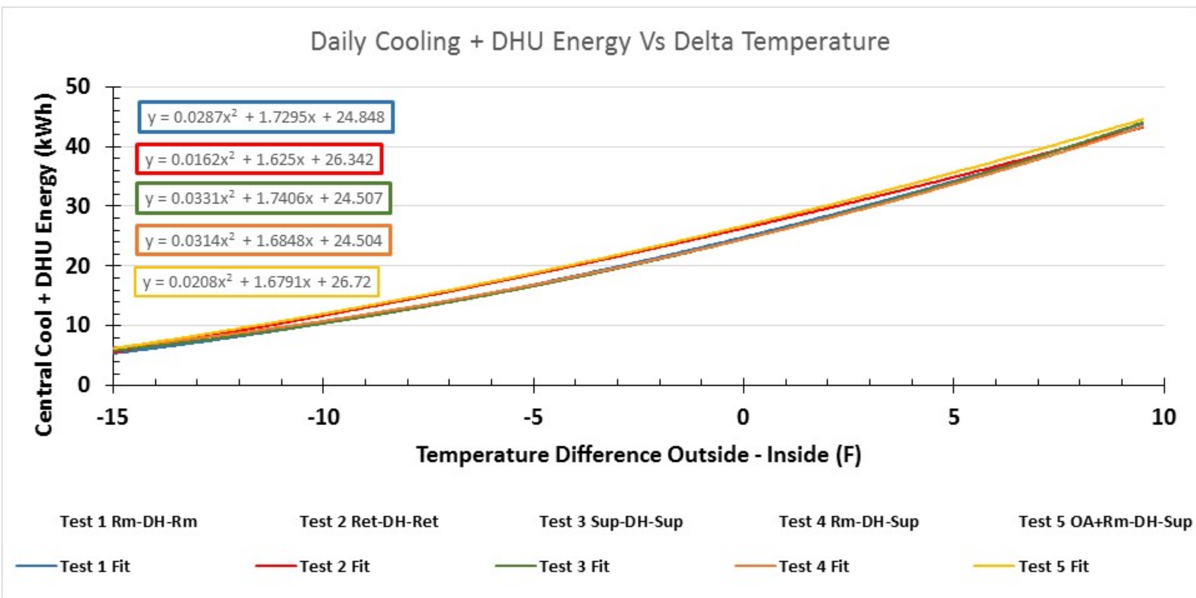

Figure 11. Resulting equations used to predict annual energy used with TMY3 data and indoor T=75°F.

There is little distinction between Tests 1, 3, and 4. Tests 2 and 5 have similar profiles indicating higher energy use than the other tests. Test 2 showed greater energy use within the middle range of dT between -8°F to 1°F (outdoor daily average temperature range between 68°F to 77°F). The reason for this will be fully explained in the next section Dehumidification Performance.

The final regression equations from all tests were used with Typical Meteorological Year 3 (TMY3) outdoor temperature to predict an annual cooling and DHU energy use. Annual predictions were made for Florida cities of Miami, Orlando and Jacksonville. An indoor cooling setpoint of 75°F was used for indoor temperature. The indoor temperature of 75°F was subtracted from the TMY3 outdoor temperature to calculate a daily temperature difference for each day of the year. Only days with temperature differences equal to or greater than -15°F were used. The results for Tests 1-4 are shown in Table 1. The results show that the energy impact of Test 2, DHU ducted to the central return, increased for cities in central or northeast Florida compared to Miami. The average of all three cities for Tests 1-4 are shown in Table 2. On average, Test 2 used about 4% more energy than Test 1 (DHU pulls air directly from and supplies DHU directly to a central indoor room). The extra energy used in Test 2 configuration represents approximately $40 per year in increased energy cost. Tests 3 and 4 showed similar results of using approximately 1% less energy than Test 1.

Table 1. Predicted Annual Central Cooling and DHU Energy for Tests 1-4 at Three Florida Cities With Comparisons to Test 1

	Miami			
	Test 1	Test 2	Test 3	Test 4
	Rm-DH-Rm	Ret-DH-Ret	Sup-DH-Sup	Rm-DH-Sup
Annual kWh	10245	10534	10192	10133
Delta kWH	0	290	-53	-112
Delta % from DHU return	0	2.8%	-0.5%	-1.1%
	Orlando			
	Test 1	Test 2	Test 3	Test 4
	Rm-DH-Rm	Ret-DH-Ret	Sup-DH-Sup	Rm-DH-Sup
Annual kWh	7661	8007	7610	7609
Delta kWH	0	346	-51	-52
Delta % from DHU return	0	4.5%	-0.7%	-0.7%
	Jacksonville			
	Test 1	Test 2	Test 3	Test 4
	Rm-DH-Rm	Ret-DH-Ret	Sup-DH-Sup	Rm-DH-Sup
Annual kWh	6339	6626	6303	6308
Delta kWH	0	287	-36	-30
Delta % from DHU return	0	4.5%	-0.6%	-0.5%

Table 2. Average of Three Florida Cities Predicted Annual Central Cooling and DHU Energy for Tests 1-4 With Comparisons to Test 1

	Average 3 Florida Cities			
	Test 1	Test 2	Test 3	Test 4
	Rm-DH-Rm	Ret-DH-Ret	Sup-DH-Sup	Rm-DH-Sup
Annual kWh	8082	8389	8035	8017
Delta kWH	0	308	-46	-65
Delta % from DHU return	0	3.8%	-0.6%	-0.8%

The results of Test 5 after adjustments were made to subtract for extra cooling load brought in by mechanical ventilation that did not occur during Tests 1-4 are shown in Table 3 along with Average results of Tests 1-4. The results show Test 5 with 70 cfm of continuous mechanical ventilation had about 6% more predicted annual energy use. The energy use of mechanical ventilation fan accounts for the difference from Test 1. Therefore the annual predicted energy of Test 5 is expected to be very similar to Tests 1, 3, and 4 if the mechanical fan energy is accounted for.

Table 3. Average of Predicted Annual Central Cooling and DHU Energy With Adjustments to Test 5 Made to Account for Mechanical Ventilation Energy Difference Between Tests 1-4 and Test 5 With Comparison to Test 1.

	Average 3 Florida Cities				
	Test 1	Test 2	Test 3	Test 4	Test 5 Mech.Vent.
	Rm-DH-Rm	Ret-DH-Ret	Sup-DH-Sup	Rm-DH-Sup	OA&Rm-DH-Sup
Annual kWh	8082	8389	8035	8017	8569
Delta kWH	0	308	-46	-65	487
Delta % from DHU return	0	3.8%	-0.6%	-0.8%	6.0%

Mechanical Ventilation Fan Power

Test 5 shows a significant efficiency advantage of DHU with an efficient supply air fan compared to a central system runtime ventilation design. A runtime ventilation system is one where the outdoor air is ducted into the central system return and air is pulled in using the central system supply air fan. Mechanical ventilation provided through the DHU only required about 76 watts of power compared to 300 watts up to about 600 watts or more if a central air handler fan was used instead. Figure 12 shows the low end power used when only the DHU fan was operating without dehumidification. The red box highlights the power when only the DHU fan was operating at about 76 watts. The power increased to about 530 watts on average when dehumidification occurred.

There are less efficient DHU where the DHU fan power could be greater than indicated here. The DHU tested here had a fan efficiency of about 0.46 W/cfm (2.2 cfm/W) under real testing conditions with static pressures below manufacturer stated limit of 0.5 in WC.

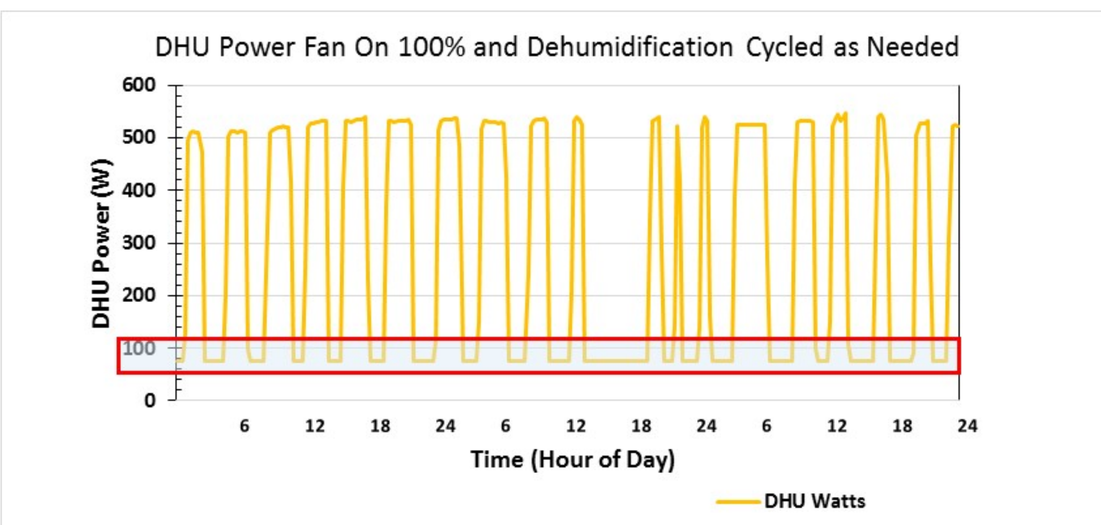

Figure 12. DHU power during three days of Test 5.

Dehumidification Performance

Dehumidification performance was evaluated in short steady-state tests as well as over longer periods of time. As this report is providing new tests and updated results from the previous Tests 1-3, significant findings from Test 1-3 (Withers et al. 2018) are shared here as well as the most recent findings. This section on dehumidification performance focused primarily on impacts upon space cooling and upon supplemental dehumidification appliances. A broader evaluation of comparing the daily moisture (condensate) removal performance of each appliance is covered in Appendix C.

Central Cooling Dehumidification Performance

Test 2 was the only test configuration that had a significantly detrimental impact upon the central cooling latent performance. This is because it is the only test configuration where the hot dry air leaving the DHU entered directly upstream of the central cooling coil. Table 4 shows a summary of central cooling system evaporator coil entering and leaving conditions along with the measured energy transfer characteristics across the coil. Outdoor conditions at the condensing unit averaged 86°F during this testing.

Table 4. Central AC Steady-State Evaporator Coil Performance Comparisons During Test 2

Test 2 Condition	Entering T (°F)	Entering RH (%)	Leaving T (°F)	Leaving RH (%)	Airflow cfm	Total Btu/h	Sensible Btu/h	Latent Btu/h	SHR
AC On; DHU Off	75.8	51.7	56.9	81.7	947	-27983	-19616	-8366	0.701
AC On; DHU On; DHU ducted to AC return duct	79.3	42.1	57.7	75.9	947	-28446	-22415	-6031	0.788
% difference from when only AC on compared to when both AC&DHU on						1.7%	14.3%	-27.9%	12.4%

Source: Table 1. (Withers et al. 2018)

Temporary controlled steady-state testing with both the central cooling system and DHU operating at the same time found that the central cooling latent performance was decreased by 28% compared to when no DHU was operating at the same time during Test 2 Sensible cooling increased by 14%. The increase in SHR (decreased latent ratio) is the opposite performance characteristic desired during when trying to remove indoor moisture and control indoor RH.

Another detrimental dehumidification performance impact from Test 2 configuration occurred when moisture left on the cooling coil was re-evaporated from the DHU blowing hot dry air through the warm wet central cooling coil. For example, during one 15 minute period of uninterrupted monitoring, the measured DHU coil rate of latent removal was -1.8 lb/h and the measured latent heat of evaporation from the central cooling coil was +1.9 lb/h. This occurred while the DHU had operated the full period and the central cooling system had remained cycled off 1.25 hours prior to and during this 15 minute period.

The observed events of central coil moisture evaporated back into the space from a dehumidifier supply into the central return was repeatable. A controlled test conducted with a wet central cooling coil just after cycling off and DHU operated for 28 minutes after the central cooling had cycled off. A total 1.5 pounds of moisture evaporated from the central cooling coil was measured. This moisture then gets pushed slowly down the central supply eventually making its way back into the conditioned space. This 28 minute test can be seen in Figure 13. Negative values indicate that sensible or latent heat was removed from the airstream at the central AC coil. Positive values mean that heat was added to the air after the AC coil. The total cooling appears small because the latent heat was positive. This means that moisture was coming off from the AC coil into the central supply air duct. The evaporation of water provided a small amount of sensible coiling.

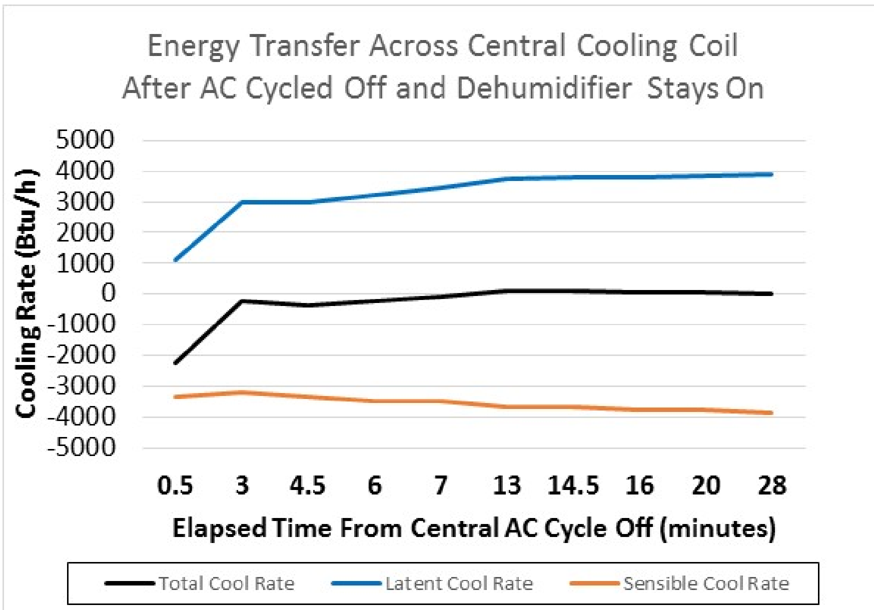

Figure 13. Test 3 moisture evaporated off wet AC coil from DHU air supplied into central return.

These results show that ducting DHU into the central cooling return will have periods when the moisture evaporated off the central warm wet coil will be nearly at the same rate of DHU moisture removal. In this case the only benefit to lowering RH is the waste heat of DHU that helps lower RH and increase the cooling load. This particular condition is similar to an inefficient strip heater.

Higher Energy Impacts of Test 3 Explained

These findings show that DHUs ducted from/to central cooling system returns upstream of the coiling coil (Test 2) can have significant performance degradation impacts upon the central cooling system. They also explain why Test 2 energy use was greater than Tests 1, 3, and 4, particularly around the middle range of delta temperature (refer back to Figure 10 and 11). The energy impact is greatest during a period of cycling by both the AC and DHU. At higher dT, (daily outdoor average of 77°F or greater), cooling load in the lab tests was high enough to remove enough moisture requiring little to no DHU operation and there would be no negligible impact. At lower dT, (daily outdoor average around 68°F or less) the cooling load was very low and there was very little cooling cycles. This means the coil would not be as wet for evaporation to occur and fewer cycles for DHU to run coincidently, thereby limiting the potential frequency of degraded latent performance.

Ducted Dehumidifier Performance

Manufacturer data from the Ultra-Aire 70H DHU tested shows that a change in DHU performance can be expected based upon entering air conditions. Table 5 shows manufacturer data shipped with the DHU. The data shows a 33% drop in latent capacity from rated conditions (entering air 80°F/60%RH/ 69.6°F wb) to when the entering air is cooler and drier (70°F/60%/58.4°Fwb). The latent efficiency (pints/kWh) drops by 20% according to manufacturer data.

Table 5. Ultra-Aire 70H Manufacturer DHU Performance Data

Test Condition	Entering T (°F)	Entering RH (%)	Capacity Pints/day	Efficiency Pints/kWh
Rated warm/moist	80	60	70	5
cool/dry	70	60	47	4
cool/dry % difference from rated condition			-33%	-20%

Source: Therma-Stor LLC , Ultra-Aire Installation Instruction Manual 4/27/16

The DHU rated conditions (80°F, 60% RH) are at higher temperature and RH than typically maintained in occupied Florida homes. Indoor average temperature and RH was 76.2°F and 53.4 % RH based on hourly measurements taken over several months to a year within 81 central Florida homes (Withers et al. 2012). Since DHU performance can be expected to be impacted by different entering conditions, short-term testing was conducted to evaluate three possible scenarios. Table 6 shows three different sets of entering conditions for the DHU and the calculated total, sensible and latent heat as well as measured electric power and a latent efficiency metric of pints/kWh. Positive heat values within Table 6 indicate more heat leaving the DHU unit than entered, whereas negative heat values indicate less heat leaving than entered. For example, positive total heat Btu/h means the net energy leaving DHU is greater than DHU entering conditions. Negative latent Btu/h means that less latent exited the unit than entered. This occurs from a cold evaporator coil that collects moisture from indoor air and drains the condensate out of the unit.

- The first condition a) is with warm-moist air entering the DHU. This condition is the closest one to a rated condition of 80°F and 60% RH and could represent conditions that might occur during extended periods of home vacancy when a DHU is used to help control humidity. This could also be a common entering condition in some homes if the DHU was ducted from/to the central return with the central system off. Some periods of entering air elevated above room temperature were observed during normal testing and were believed to be due to a small amount of damper leakage and some possible radiative heating of the metal damper into the upstream side of central return when the central AC was cycled off.
- The second set of entering conditions b) are more representative of typical room air conditions that would enter a DHU if ducted directly to the room or from the return when the central system is on.
- The last set of conditions c) were measured with the DHU on and the central AC on when the DHU was ducted from/to the central supply. This offers the coldest and driest set of entering conditions into the DHU.

A summary of relative DHU impacts are shown as % differences at the bottom of Table 6. Negative % values indicate a decrease, positive % values indicate an increase. All tests indicate that more heat energy leaves the DHU than entered, this is expected. Another observation of electric power shows that power drops as the entering air temperature drops. This is not a surprise since the hardest working component, the compressor is directly impacted by the entering air temperature. A review of manufacturer split dx AC performance data will also show higher energy use with higher outdoor temperatures, where the condensing unit and compressor are located. Additional results are summarized immediately following Table 6.

Table 6. DHU Steady-State Test Performance Comparisons at Three Different Entering Conditions

DHU Test Condition	Enter T (°F)	Enter RH (%)	Leave T (°F)	Leave RH (%)	Airflow cfm	Total Btu/h	Sensible Btu/h	Latent Btu/h	DHU Elect. Watts	Pints/kWh
a) **Air enter DHU warm/moist**	82.8	57.3	110.6	17.0	165	1583	5041	-3458	581	5.5
b) **Air enter DHU cool/dry (typical room condition)**	75.7	49.1	96.0	18.0	165	1372	3661	-2289	516	4.1
c) **Air enter DHU cold/dry (enter from AC supply; AC On)**	54.8	71.9	75.9	19.4	171	1512	3923	-2411	438	5.0
% difference from a) **(enter DHU warm/moist) to** b) **(enter DHU cool/dry)**						-13.3%	-27.4%	-33.8%	-11.2%	-25.5%
% difference from a) **(enter DHU warm/moist) to** c) **(enter DHU cold/dry)**						-4.5%	-22.2%	-30.3%	-24.6%	-9.1%
% difference from b) **(enter DHU cool/dry) to** c) **(enter DHU cold/dry)**						10.2%	7.2%	5.3%	-15.1%	22.0%

Source: Table 3. (Withers et al. 2018)

Since DHU efficiency is rated by latent removed per electric energy input, this should also be considered. The results here are more mixed. While the electric energy clearly decreased with decreasing entering conditions, the latent removal at the evaporator coil decreased with lower entering dewpoint temperature from test a) to b), and test a) to c) but not from test b) to c).

Table 6 can be summarized as follows:

From **a) (enter DHU warm/moist) to b) (enter DHU cool/dry)**

- This shows an expected trend of decreasing latent as the manufacturer data indicated.
- The electric power consumption dropped 11%.
- The latent energy removed decreased by 34%
- The latent efficiency (pints/kWh) dropped 26%.
- Total energy (net sensible) leaving the DHU decreased by 13%.

From **a) (enter DHU warm/moist) to c) (enter DHU cold/dry)**

- The electric power consumption dropped 25%.
- The latent energy removed decreased by 30%.
- The latent efficiency (pints/kWh) dropped 9%.
- Total energy (net sensible) leaving the DHU decreased by 5%.

From **b) (enter DHU cool/dry) to c) (enter DHU cold/dry)**

- The electric power consumption dropped 15%.

- The latent energy removed increased 5%.
- The latent efficiency (pints/kWh) increased 22%.
- Total energy (net sensible) leaving the DHU increased 10%.

In practical application, the most appropriate DHU performance comparison would be between tests b) (room air) and c) (supply air). Solely based upon DHU electric power and latent performance, test c) (cold dry air from AC system) uses the least DHU energy and offers reasonable latent efficiency compared to pulling air from inside room or return duct. The benefits shown in Table 6 only apply to steady-state conditions shown and do not include impacts of normal operated conditions where the DHU and AC would cycle on and off independent from each other. The maximum benefit of ducting a DHU from/to the central AC supply would only occur when both systems have operated simultaneously for about 20 minutes.

Longer-Term Dehumidifier Performance

Dehumidifier performance was evaluated by energy use, capacity (pints or pounds of water removed), and efficiency as L / kWh. The dehumidifier performance can be impacted by entering air conditions and other factors such as static pressure which can impact airflow rate. Monitoring has found that the DHU has been operating within manufacturer specifications under all test configurations. The DHU has a manufacturer stated efficiency of 2.4 L/kWh under rated test conditions of entering air at 80°F and 60% RH using 580 watts of power. The actual entering air test conditions can be quite different from rated conditions, and the reader is reminded that the 15 minute interval data shown here are often not at steady state conditions. Figure 14 shows the DHU power consumption during 15 minute intervals only for periods where the unit operated the full 15 minutes. Power was plotted against dry bulb and dewpoint temperature. The result shows DHU power increases with increases in Tdb and Tdp as expected. It also shows that entering dewpoint temperature is a better single variable predictor of power use. Figure 14 shows that DHU power can increase by about 27% depending upon entering dewpoint temperatures.

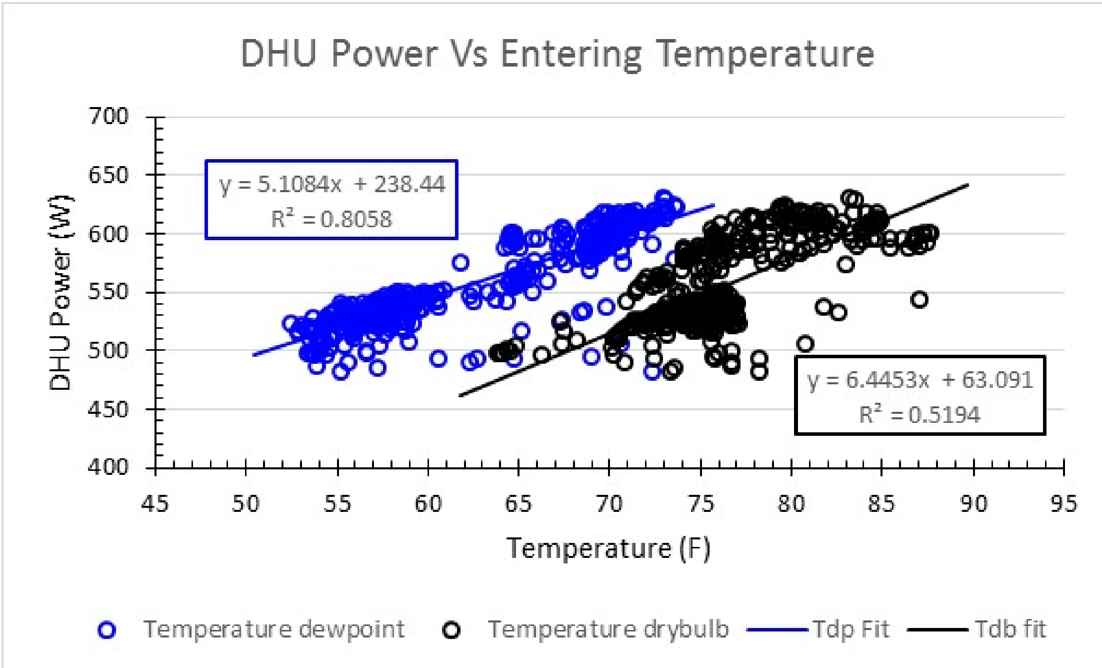

Figure 14. DHU Power versus entering temperature.

Testing has found the efficiency during full 15 minute periods of dehumidification operation to be near proximity of the rated efficiency (2.4 L/kWh). Since the test conditions are cooler and drier than rated conditions (80°F, 60% RH) it is reasonable to expect the efficiency to be lower than rated conditions. Figure 15 provides an example of the efficiency during three consecutive days of Test 4 (entering air from indoors). The data shown are a continuous stream of 15 minute interval data, but periods with DHU runtime with less than 100% are shown as a non-value. The entering conditions as well as the outdoor temperature are also shown in Figure 15. This is because during partial runtime periods the energy use is much lower and resulted in abnormally high efficiency values.

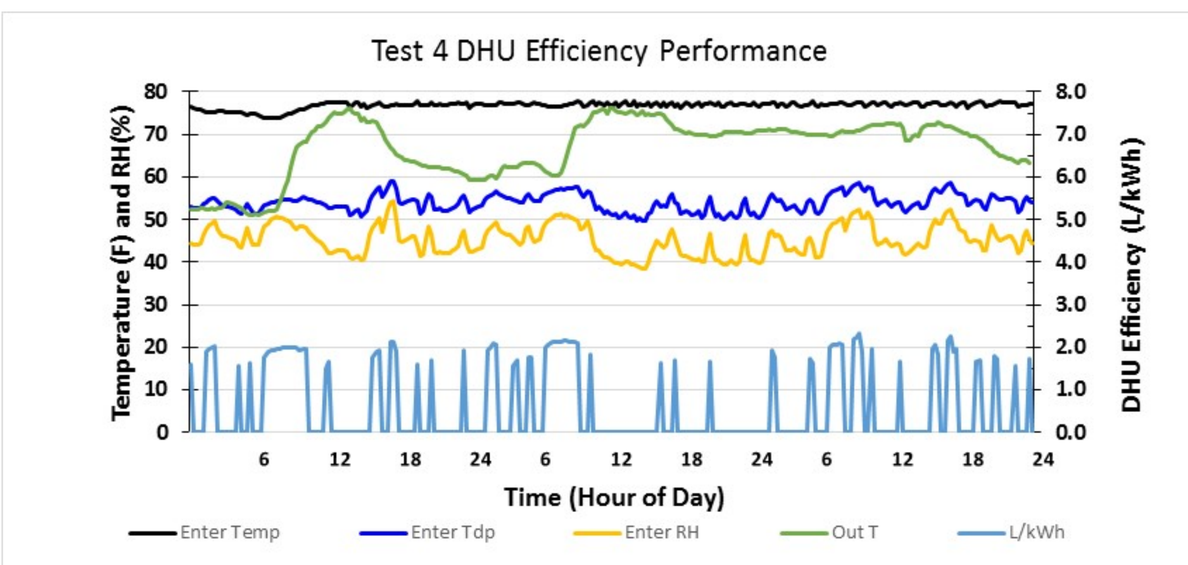

Figure 15. DHU efficiency as L/kWh during December 7-9, 2019 during 100% DHU runtime.

Figure 16 shows the DHU capacity performance during the same three-day period as Figure 15.

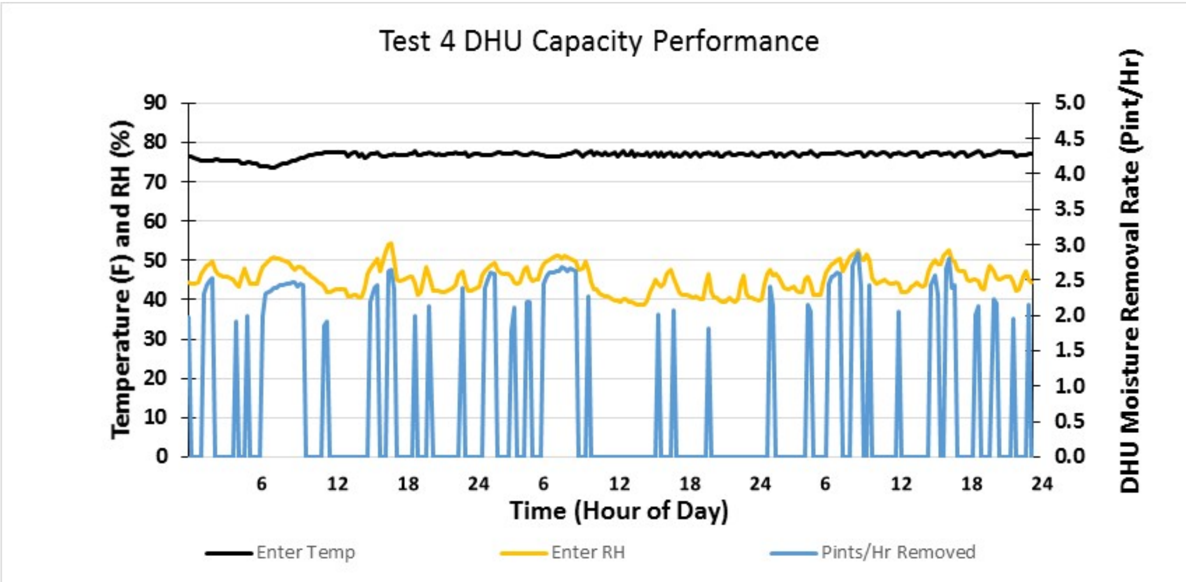

Figure 16. DHU moisture capacity performance during December 7-9, 2019 during 100% DHU runtime.

Since Test 5 brought some outdoor air mixed with indoor air into the DHU, there was some opportunity to evaluate DHU performance with a little wider variation in entering conditions particularly with more moisture some of the time. The outdoor air represented about 43% of the total entering air. DHU efficiency was higher on average compared to Test 4 due to higher entering moisture levels. DHU efficiency is shown in Figure 17 and DHU capacity is shown in Figure 18.

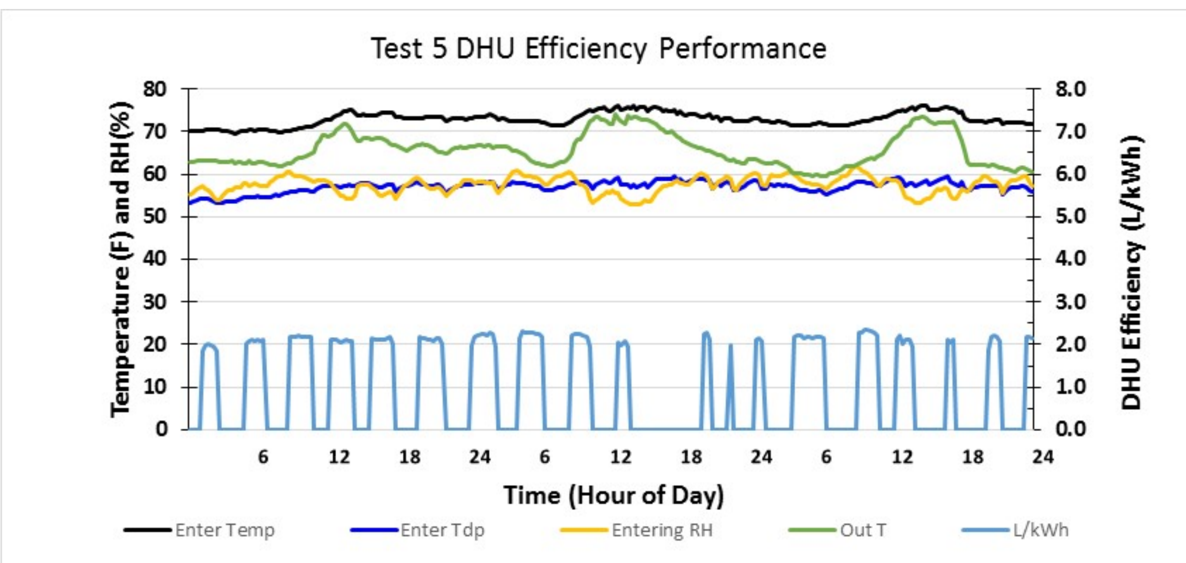

Figure 17. DHU efficiency during February 1-3, 2019 during 100% DHU runtime.

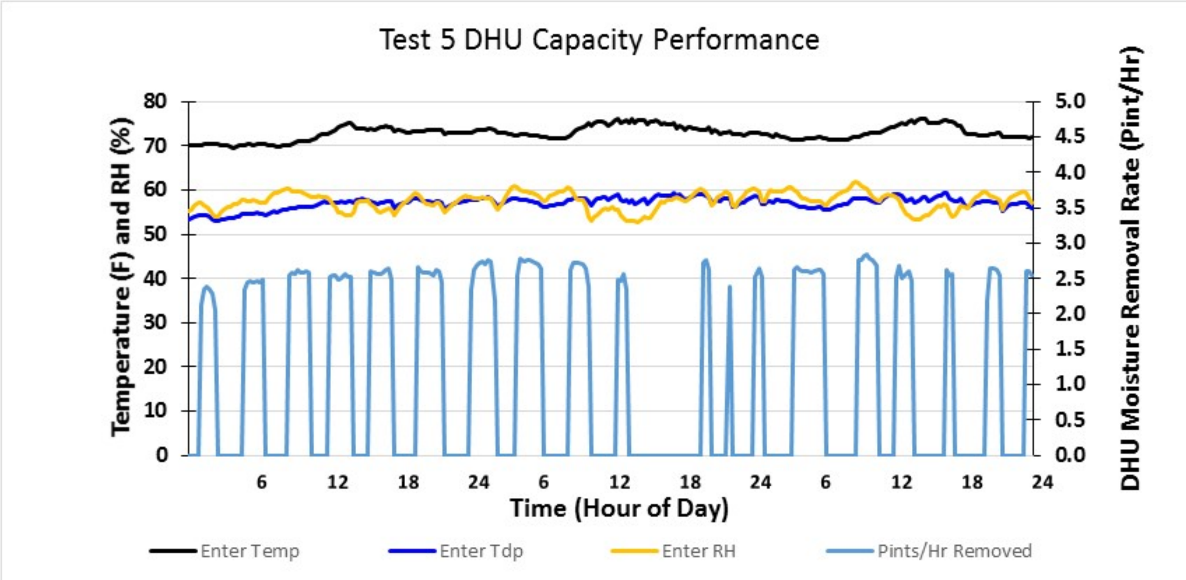

Figure 18. DHU moisture capacity performance during February 1-3, 2019 during 100% DHU runtime.

Short Test of Wider Variability of Entering DHU Air Conditions

A wider variety of entering DHU air was sought after to be able to observe a wider range in DHU performance. This also represents another ventilating DHU configuration that some contractors have used in larger homes when higher ventilation rates are needed. The five days of this testing were not represented in any Test 4 or Test 5 analysis, but are offered here as an opportunity observe the potential for wider variation in performance metrics. DHU efficiency is shown in Figure 19 and DHU capacity is shown in Figure 20.

The DHU was ducted so that 100% of all entering air came directly from outdoors to provide a wider range in entering conditions. Testing was conducted to see if DHU performance varied significantly from Tests 4 and Test 5. As expected, a wider range of efficiency due to greater variability in entering conditions was observed and is shown in Figure 19. It can be seen that generally DHU efficiency improved with more moist (higher dewpoint temperature) entering air and decreased as entering temperature increased during similar dewpoint temperature. The range of observed DHU efficiency was from 1.8 L/kWh up to 3.7 L/kWh. While the weather conditions in Figure 19 do not represent design cooling conditions, there are periods that represent weather that occurs much of the year in Florida.

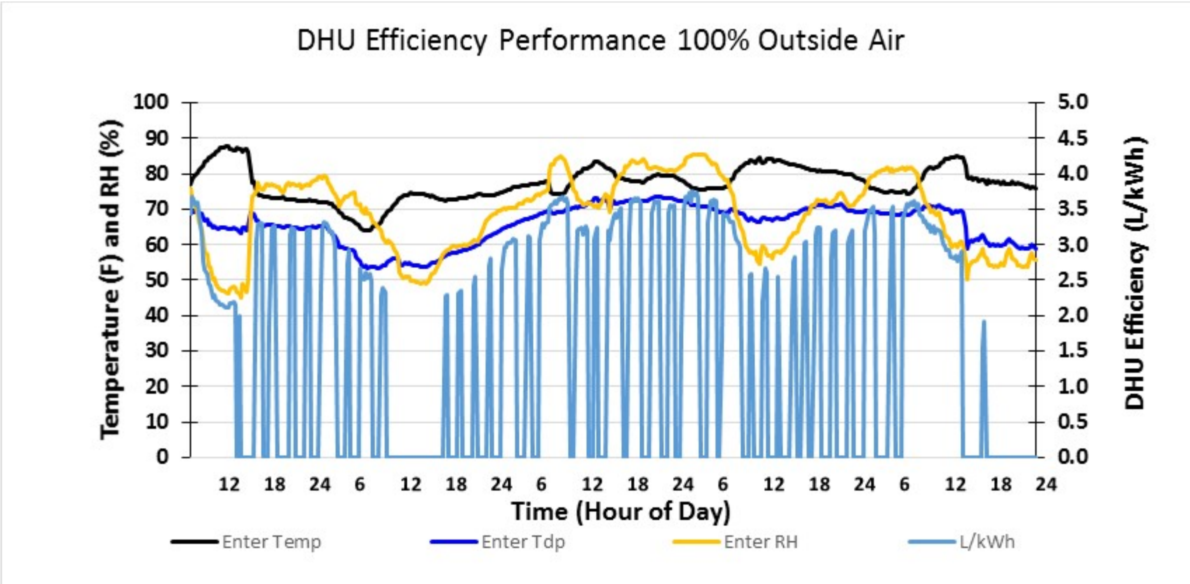

Figure 19. Period during 100% outdoor air entering DHU.

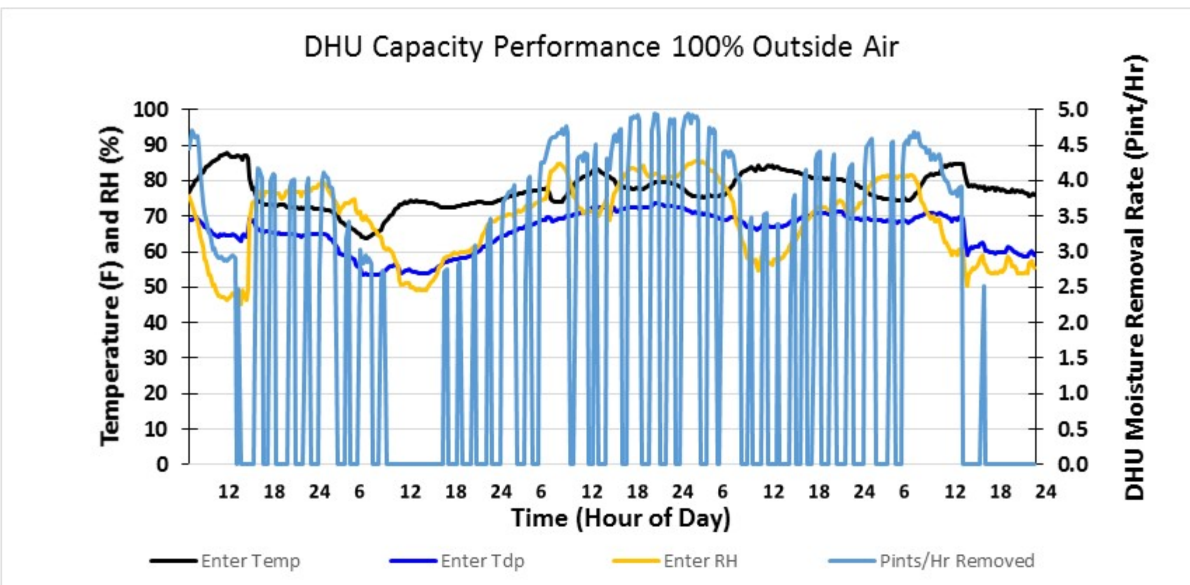

Figure 20. Period during 100% outdoor air entering DHU.

Longer term evaluation of DHU dehumidification performance was summarized for groups of days having both cooling and DHU operations previously shown in Figures 15-20. This is a closer representation of actual performance over time. Only data with DHU operation during 100% of each 15 minute period was used in the summary to eliminate unreasonable efficiency values. The entering conditions and resulting DHU capacity (Pints/h) and efficiency (L/kWh) were averaged over several days of four different types of entering conditions. These are the four most-likely types of entering conditions for a ducted DHU. The results are shown in Table 7. The first row of entering conditions represents very cool and dry conditions as tested in Test 3. The longer-term DHU performance is lower than the steady-state test shown in Table 6 row c as expected. While cold supply air entering the DHU was found to significantly improve DHU efficiency shown in Table 6, this only occurred when the AC and DHU happen to both be cycled on at the same time. Often the DHU cycles on without the AC running so the entering during Test 3 is frequently similar to indoor room conditions. The occasion of both operating at the same time is reflected at Test 3 in Table 7 as it does have a higher efficiency and capacity rate than Test 4 (entering air directly from indoor central room).

Higher entering moisture, as relatively indicated by dewpoint temperature, shows that DHU efficiency and capacity rate increase. This was well demonstrated in the test with 100% outside air entering the DHU.

Table 7. Long-Term Average DHU Efficiency and Capacity Shown With Average Entering Conditions.

DHU Entering Air From	Temperature (°F)	RH (%)	Temperature dp (°F)	RH (%)	Efficiency (L/kWh)	Capacity (pints/h)
central AC supply (Test 3)	70.7	56.1	54.3	56.1	2.3	2.7
Indoor central rm. (Test 4)	75.8	51.8	56.8	51.8	1.9	2.4
Mix 43% OA + 57% indoor (Test 5)	72.9	57.5	57.0	57.5	2.1	2.6
100% outside air	77.2	68.7	66.1	68.7	3.1	4.0

Summary

This report covered lab-based experimental study conducted in two phases to evaluate AC and DHU performance based upon how a DHU was integrated with central AC system ducts and compared this to DHU run without ducting to central system ducts. A total of five different DHU lab test configurations were completed in two phases. The first three tests occurred during the first phase with results reported on June 2018 (Withers et al. 2018) and the last two tests were completed late May 2019. All ducted DHU test configurations were as follows:

6. Test 1 DHU air ducted from/to the central main body of building.
7. Test 2 DHU air ducted from/to return side of central cooling (AC).
8. Test 3 DHU air ducted from/to supply side of AC.
9. Test 4 DHU air ducted from central main body and ducted to supply side of AC.
10. Test 5 DHU air ducted from outdoors + central main body and ducted to supply side of AC.

Tests 1-4 were the most directly comparable since they had tight control over the latent load of building. Test 5 ran mechanical ventilation through the DHU since this is used by some homebuilders as a means to ventilate and control indoor RH.

Although some DHU manufactures show it as an option, Test 2 is not recommended. This project found two highly significant detrimental impacts of Test 2 upon building dehumidification performance. First, DHU supply air into the central return duct upstream decreased central AC latent performance by 28% when the DHU and AC ran simultaneously. Second, the DHU evaporated moisture off the central cooling coil at a rate of 2-3 lb/h when the AC was cycled off. This was about the rate of measured DHU moisture removal under actual test conditions. Under these specific circumstances, the Test 2 DHU configuration performed about as well as electric strip heat with the inefficient leverage of increased temperature lowering RH and increasing cooling load. In spite of the poorer dehumidification performance, the annual predicted space conditioning (DHU + AC) energy use was about 308 kWh/year (4%) more than Test 1. This would cost a resident about $40 /year more. The DHU used in this project was at least twice the rated efficiency for a 70 pint/day capacity compared to the lowest available 70 pint/day DHU. Less efficient DHU, would result in substantially more DHU and AC energy use and the energy impact from Test 2 could double to 8% and about $80/year increased energy cost if the lowest efficiency DHU were used.

There was very little difference in annual predicted energy use of Tests 1, 3, 4. Test 5 indicated about 6% more energy use than Test 1. This difference is attributed to the mechanical ventilation fan energy that was did not occur during Tests 1-4.

Currently manufacturers only indicate rated DHU capacity (pints) and efficiency (L/kWh) at a single entering air condition of 80°F and 60%. This is sorely lacking to help engineers or consumers to choose the suitable capacity and expected performance. At best, the only publicly available DHU capacity guidance is based upon the area of the space to be dehumidified. The dehumidification performance of the DHU was evaluated for Tests 1-5 and also a short period with 100% outside air entering the DHU to better understand how much entering conditions impact the rated metrics of DHU capacity and DHU efficiency. The testing was limited to one DHU and specific test periods, but it was able to demonstrate the potential for a wider variety of performance to be considered. Tables 6 and 7 of this report show that the tested DHU performance operated within expectations for entering air similar to conditioned indoor air. The DHU performance improved with lower dry bulb temperature or with higher moisture content indicated in this report as dewpoint temperature.

In tight homes where whole-house mechanical ventilation is required, mixing the incoming outside air with indoor air before entering the DHU (Test4) shows a promising alternative to running runtime ventilation using a central ducted system fan operated to meet ASHRAE 62.2 standards. This is because the fan of the DHU uses much less power if used to bring in outside air compared to a central ducted system fan used to do so in a runtime ventilation strategy. The DHU fan power was 75% less than the central fan used in this testing. The DHU fan power could be as much as 90% less or more if central systems larger than the nominal 3 ton system were used in this project. This is based on a straight-up comparison of the total power to operate each fan. A useful way to look at this is in terms of fan efficiency measured as watts of power used per cfm air delivered (W/cfm).

While the case for using this DHU fan for mechanical ventilation looks good compared to a runtime ventilation method, other fans exist that may have similar efficiency when evaluated on a W/cfm of ventilation airflow delivered. A runtime vent fan requiring the use of the central ducted system may be between 4-8 W/cfm ventilation delivered, which is very high. The DHU tested in this project was only 0.46 W/cfm ventilation air delivered (76 W/165 cfm). Some energy recovery ventilation (ERV) units can also ventilate with fans using less than 1 W/cfm.

Selecting the lowest operating cost ventilating system will come down to more than just the vent fan efficiency. An important thing to remember about using an ERV in Florida is that it will not be able to maintain indoor RH below 60% RH all hours of the year without supplemental dehumidification or expensive cooling equipment with special dehumidification modes that can operate at very low sensible heat ratios and at low cooling capacity. Some may not mind indoor humidity around 60%-65% and would not require supplemental dehumidification. In this specific case, an ERV alone may be a suitable mechanical ventilation method. For those requiring tighter control over indoor RH, and diminished moisture-related issues, total space conditioning and ventilation equipment first cost and operational costs must also be considered.

Conclusion

The primary focus of this project was to test different ducted DHU configurations, evaluate the performance of each and make recommendations for Florida Building Code modifications if warranted.

As a result of this research of ducted DHU configurations, Florida Building Code Modifications were submitted for consideration. The submitted code modification language can be found in Appendix A.

It is recommended that Test 2 with DHU ducted from and to a central ducted return be not permitted. While the predicted annual energy increase of Test 2 was modest, the energy impact with a lower efficiency DHU would be greater than 4% compared to a DHU ducted to and from the conditioned space (Test 1). The primary merit in eliminating this option is due to the significant potential for poor space dehumidification performance that could be considered similar to simply adding electric strip heat during specific periods.

The gravity or check dampers used for Tests 3 and 4 during the research testing were a critical part of each duct configuration. These allow airflow to move as needed during operations and stop airflow from moving during undesirable conditions. The damper of Test 3 stops short circuiting of DHU air directly from the DHU supply back into the DHU return particularly when the central system is cycled off. The damper of Test 4 stops central cool air from moving backwards through the DHU when the DHU is off.

Another important matter to consider when connecting any ducts to DHU is that the static pressure acting upon the DHU should not exceed manufacturer recommendations. This may result in some ducted DHU configurations that should not be used based upon manufacturer guidance. Excessive static pressure may adversely impact the DHU performance due to limited DHU airflow. This project did not evaluate impact of static and airflow rate upon the DHU. Tested DHU duct designs were in compliance with manufacturer recommendations.

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Appendix A

Recommended Code Modification

Based upon research test results Florida Building Code modifications were submitted. As of June 1, 2019, the recommended code modification to Chapter 4 Residential Energy Efficiency was as follows:

R403.13 Dehumidifiers (Mandatory).

If installed, a dehumidifier shall conform to the following requirements:

1. The minimum rated efficiency of the dehumidifier shall be 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.

2. The dehumidifier shall be controlled by a sensor that is installed in a location where it is exposed to mixed house air.

3. Any dehumidifier unit located in unconditioned space that treats air from conditioned space shall be insulated to a minimum of R-2.

4. Condensate disposal shall be in accordance with Section M1411.3.1 of the *Florida Building Code, Residential*.

R403.13.1 Ducted Dehumidifiers.

Ducted dehumidifiers shall, in addition to conforming to the requirements of Section R403.13, conform to the following requirements:

1. If a ducted dehumidifier is configured with return and supply ducts both connected into the supply side of the cooling system, a backdraft damper shall be installed in the supply air duct between the dehumidifier inlet and outlet duct.

2. If a ducted dehumidifier is configured with only its supply duct connected into the supply side of the central heating and cooling system, a backdraft damper shall be installed in the dehumidifier supply duct between the dehumidifier and central supply duct.

3. A ducted dehumidifier shall not be ducted to or from a central ducted cooling system on the return duct side upstream from the central cooling evaporator coil.

4. Ductwork associated with a dehumidifier located in unconditioned space shall be insulated to a minimum of R-6.

{*Recommended code modification to* Table R405.5.2(1) *continued on next page.*}

[Add following two sections to Table R405.5.2(1); no other changes to table.]

Building Component	Standard Reference Design	Proposed Design
Dehumidification Systems	None, except where dehumidification equipment is specified by the proposed design, in which case: 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 Dehumidifier Ducts: None Dehumidifier Duct Location: N/A Dehumidifier Duct R Value: N/A Dehumidifier Duct Surface Area: N/A	As proposed As proposed Sufficient to maintain humidity at setpoint all hours As proposed As proposed As proposed As proposed As proposed As proposed
Dehumidistat	None, except where dehumidification equipment is specified by the proposed design, in which case: Setpoint turn on = 60% relative humidity Setpoint turn off= 55% relative humidity	Same as standard reference design

Appendix B

Space Conditioning Energy Analysis Method

Test configurations 1-4 were conducted at three different rates of interior latent generation. The purpose and method of using three different amounts of interior latent generation was to approximate the variability in latent load from outdoors and was previously described in more detail in the main body of this report. There was inadequate heating weather to evaluate heating. Each test configuration plotted daily total space conditioning (DHU + cooling) energy versus the daily average temperature difference (outdoor temperature minus indoor temperature) for each interior latent generation rate at 15, 30, and 60 lb/day. Latent load generation rates were conducted over a wide range of dT allowing some overlapping in ranges to improve trend analysis. Data at cooler temperatures was much more limited due to weather patterns. Figure B-1 shows all available data from Tests 1-4 together to offer a single observation of the range and variability of these tests.

It became clearer after isolating east test by latent load generation groups that data are not distributed evenly across dT for the same for each test. Scheduling tests to target particular weather conditions relied on weather forecast which posed some difficulty particularly during the cooler weather that experienced some wide variability day to day at times.

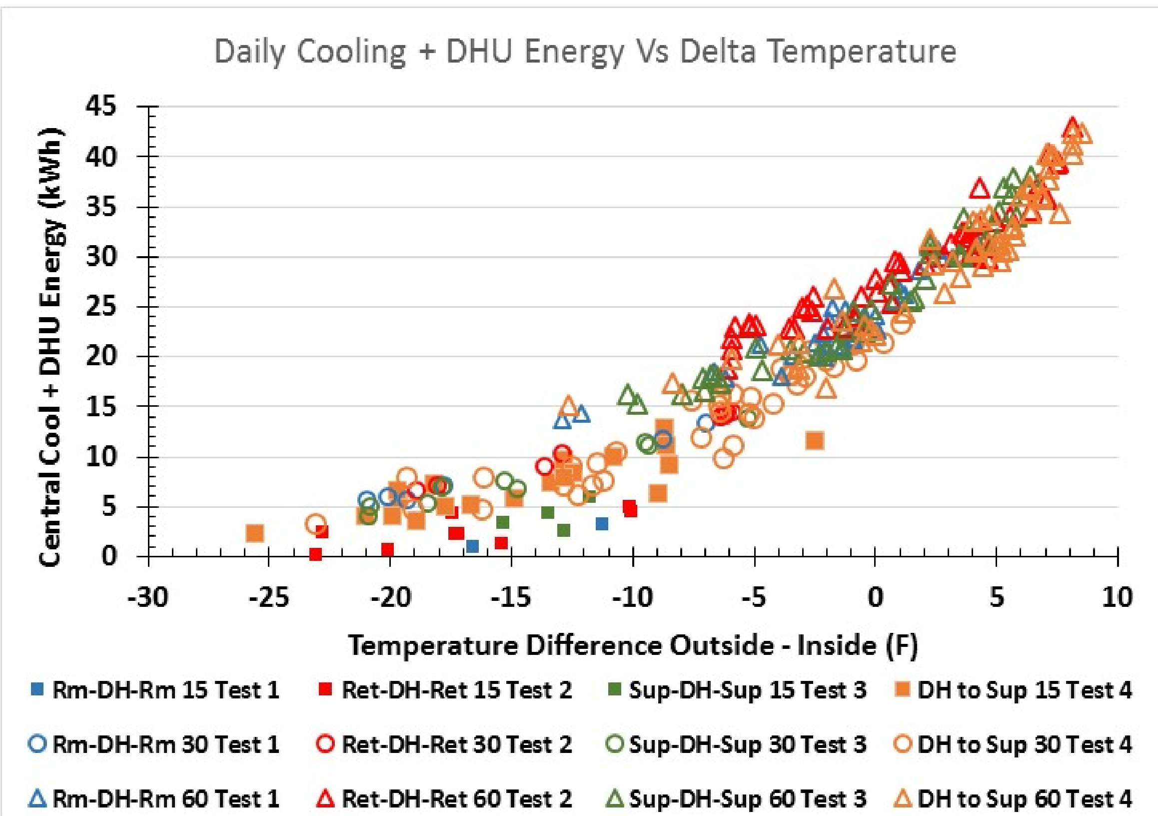

Figure B-1. Data for Tests 1-4 each at three different interior latent moisture load rates.

Least-squares best-fit regression analysis was performed to characterize the cooling and DHU energy consumption (kWh/day) versus delta-T (outdoor temperature minus indoor temperature) of all test configurations. This analysis was performed with each test isolated by the latent generation rate. The best-fit line equations and coefficient of determination (r^2) are shown in the colored text boxes of each figure.

Figure B-2 shows data for Tests 1-4 at 60 lb/day latent generation. Tests 1-3 were updated with more data from the previous year. Figure B-2 shows the general trend of higher energy use at higher dT as well as higher energy at greater latent load. Figure B-2 shows that Test 2 starts a trend of greater energy use around dT=2 that increases as dT decreases. This is explained by increased frequency of when both the DHU and central cooling operate more often thereby requiring DHU and AC to operate longer due to diminished latent cooling performance explained in the main body of this report. At the highest dT cooling loads are high enough to remove enough moisture to maintain indoor RH below 50%. Therefore, only air conditioning is running on these hotter days and all test configurations can be seen pulling together to about the same energy use.

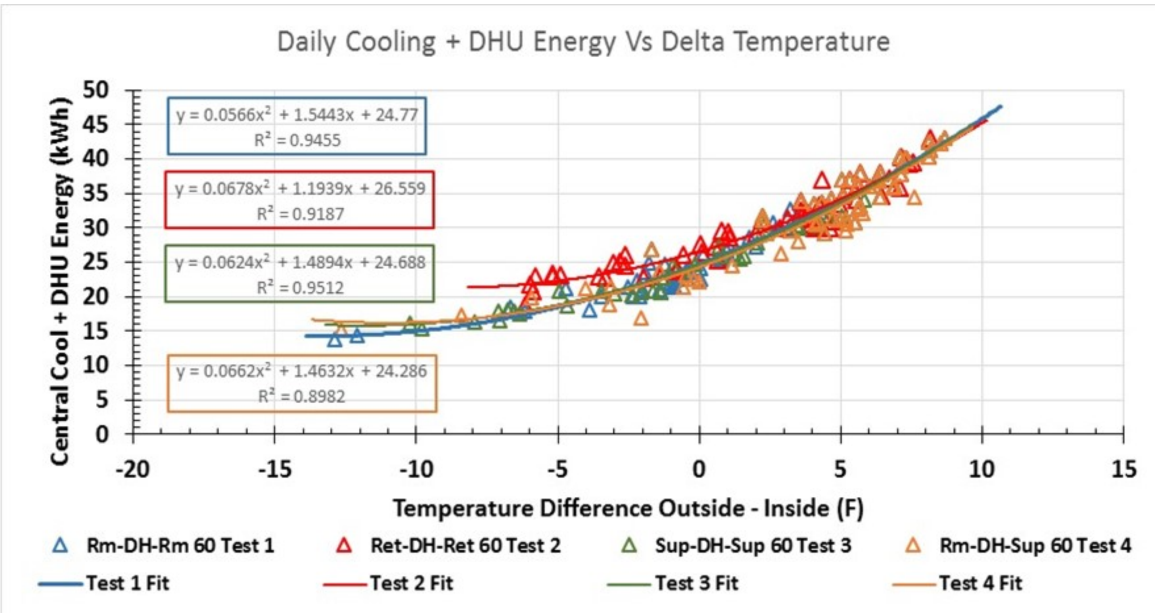

Figure B-2. Tests 1-4 at the 60 lb/day interior latent load rate.

Figure B-3 shows the results for Tests 1-4 at 30 lb/day.

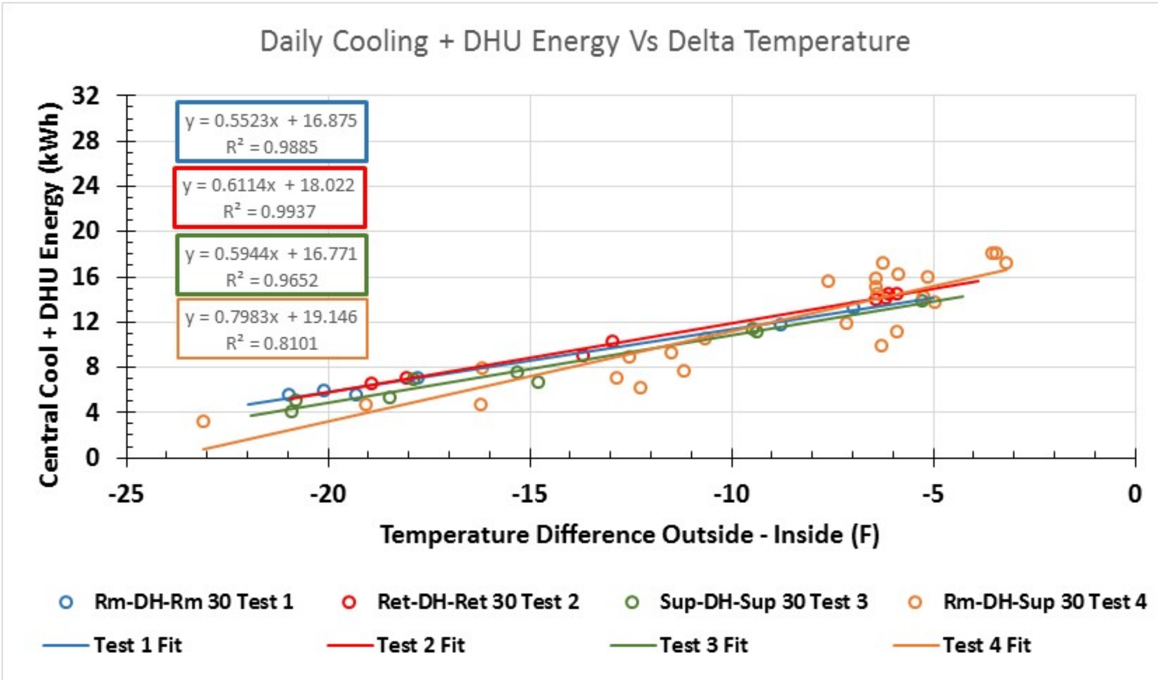

Figure B-3. Tests 1-4 at the 30 lb/day interior latent load rate.

Figure B-4 shows the results for Tests 1-4 at 15 lb/day. It would have been expected that Test 4 would have less energy at the lowest dT. The data were carefully reviewed and no clear cause was identified other than due to the very little data available at 15 lb/day, there is higher uncertainty of a relatively low energy use. It is noted that annual cooling energy predictions did not occur for days with dT less than -15 °F so the impact on the total annual energy prediction over the course of a year is very low.

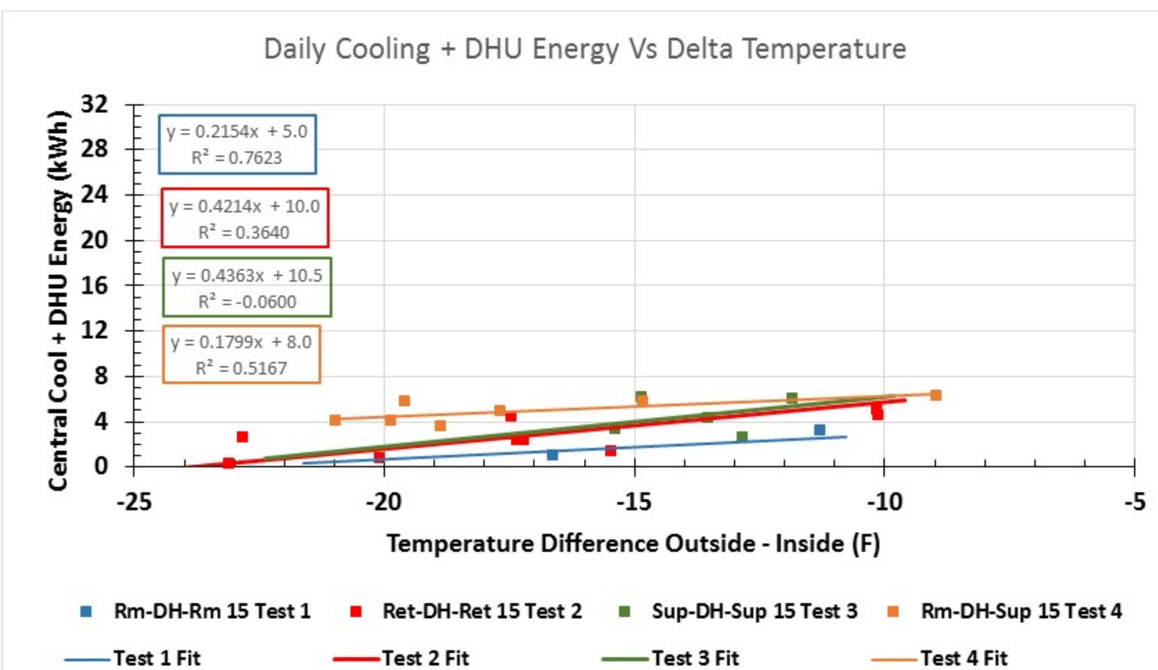

Figure B-4. Tests 1-4 at the 15 lb/day interior latent load rate.

The regression equations developed in Figures B-2 through B-4 were used to create a single best-fit equation for each test for a cooling season that was limited to a low dT of -15°F. The result is shown in Figure B-5. The 60 lb/day equations were used for dT from -5 to 10°F, 30 lb/day equations for dT between -14 to -5°F, and 15 lb/day for dT between -15 to -13°F. The 60 lb/day represented a 15 lb/day interior latent with about 45 lb/day due to ventilation related latent load that is common in Florida over the range in dT chosen. The 30 lb/day rate represented 15 lb/day interior latent and 15 lb/day ventilation latent load that generally decreases with lower outdoor temperatures.

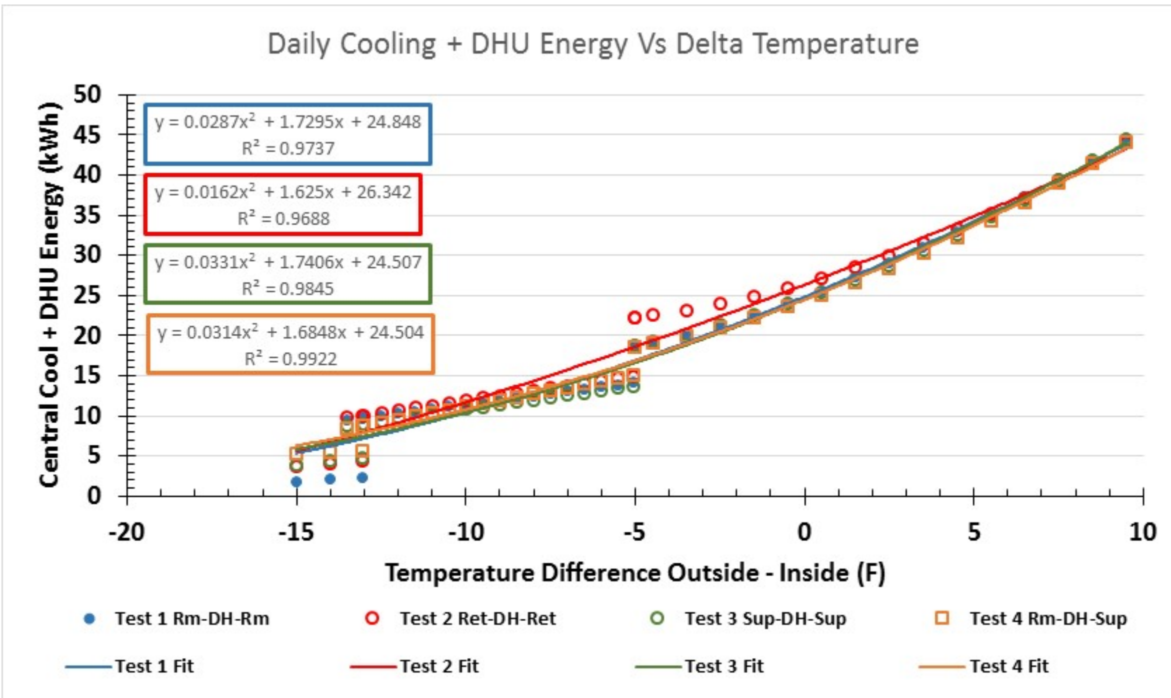

Figure B-5. Tests 1-4 showing a spliced-fit across the three different latent load rates.

Test 5 always had a 15 lb/day internal latent load and was conducted with mechanical ventilation so the actual ventilation latent load varied according to the weather conditions. The least-squares best-fit regression analysis results for Test 5 are shown in Figure B-6. Test 5 differs from Test 1-4 due to the mechanical ventilation fan energy and some higher cooling loads on some days associated with bringing in outdoor air at dT greater than or equal to 1. The final analysis accounted for this.

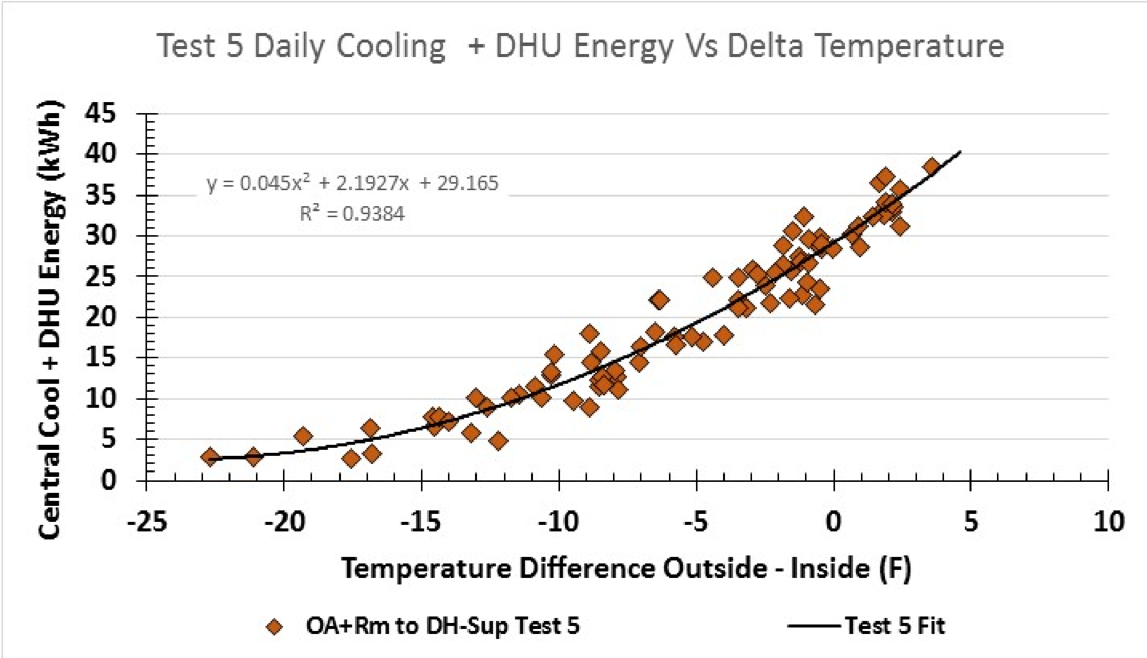

Figure B-6. Test 5 energy versus delta T with the internal latent load kept at 15 lb/day.

Annual cooling and supplemental dehumidification energy (combined) was predicted using the equations from Figures B-5 and B-6 along with Typical Meteorological Year 3 (TMY3) weather data for Three Florida cities of Miami, Orlando, and Jacksonville. The dT was calculated using in indoor temperature maintained at 75°F subtracted from the daily average TMY3 outdoor Temperature for each day of the year. Since inadequate heating weather occurred, no heating testing could be conducted. The cooling energy and associated DHU energy were calculated for TMY3 days having dT = -15°F or greater.

Final results were shown in the main body of this report.

Appendix C

Space Dehumidification Performance

The energy impact at lower cooling sensible and latent loads can be explained by the fact that most of the impact on performance occurred during simultaneous operation of both the DHU and central cooling system. Figure C-1 shows the trend for the central cooling system runtime to increase as the outdoor air increased and latent load increased. This includes Tests 1-4 at all interior latent generation rates. As the sensible load increased and cooling increased, more latent was removed by the central system and less supplemental DHU operation was needed.

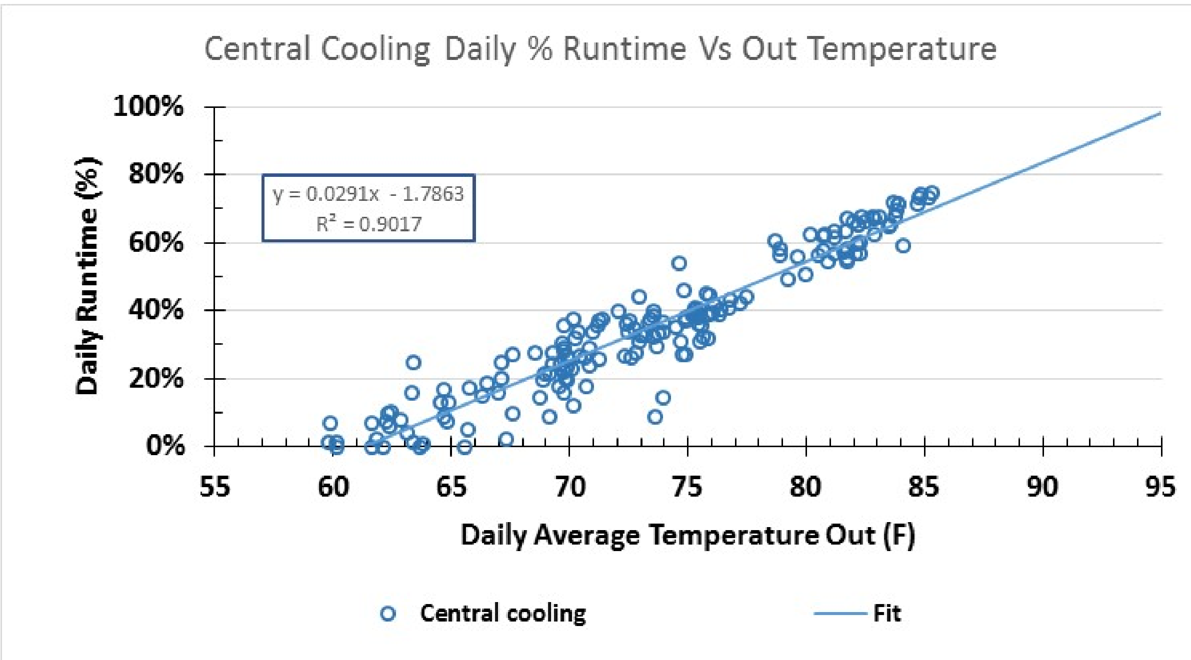

Figure C-1. Central cooling runtime based upon outdoor temperature.

Figure C-2 shows the daily total condensate removed from the evaporator coil versus the daily average outdoor temperature for the DHU and central AC system. This data includes DHU Test configurations 1-4 at three different latent load rates of 15 lb/day, 30 lb/day, and 60 lb/day. It is expected that the central AC condensate would have a good correlation to outdoor temperature since it is controlled by a thermostat that only senses sensible load. The DHU operates off of a dehumidistat controlled by relative humidity. RH setpoint was 50% in this testing and interior temperature setpoint at 76 °F.

On average central cooling removed 2.7 times more moisture than a DHU for daily outdoor average temperature of 67°F and 13.9 times more than DHU for outdoor average 77°F.

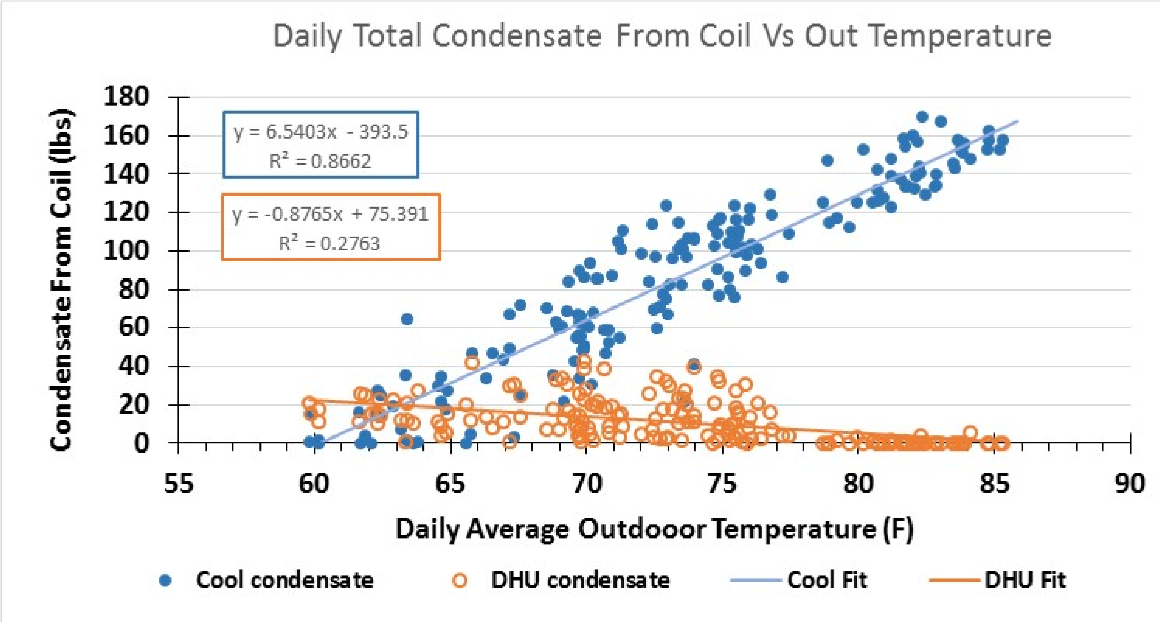

Figure C-2. Daily total condensate removed by indicated appliance based upon outdoor temperature.

Figure C-3 is used the same data as in Figure C-2 except the latent load rate is identified. No surprising characteristics can be seen. Generally, the higher latent occurred during the higher outdoor temperatures. A relatively flat range in DHU condensate is seen between about 1 lb/day up to 19 lb/day during the coolest weather up until daily outdoor temperature reached 70°F for latent rates of 15 lb/day and 30 lb/day. There was significantly more variability of DHU condensate at the 60 lb/day latent load rate that varied from between 7 lb/day up to 49 lb/day within the daily average outdoor temperature range between 63°F-79°F.

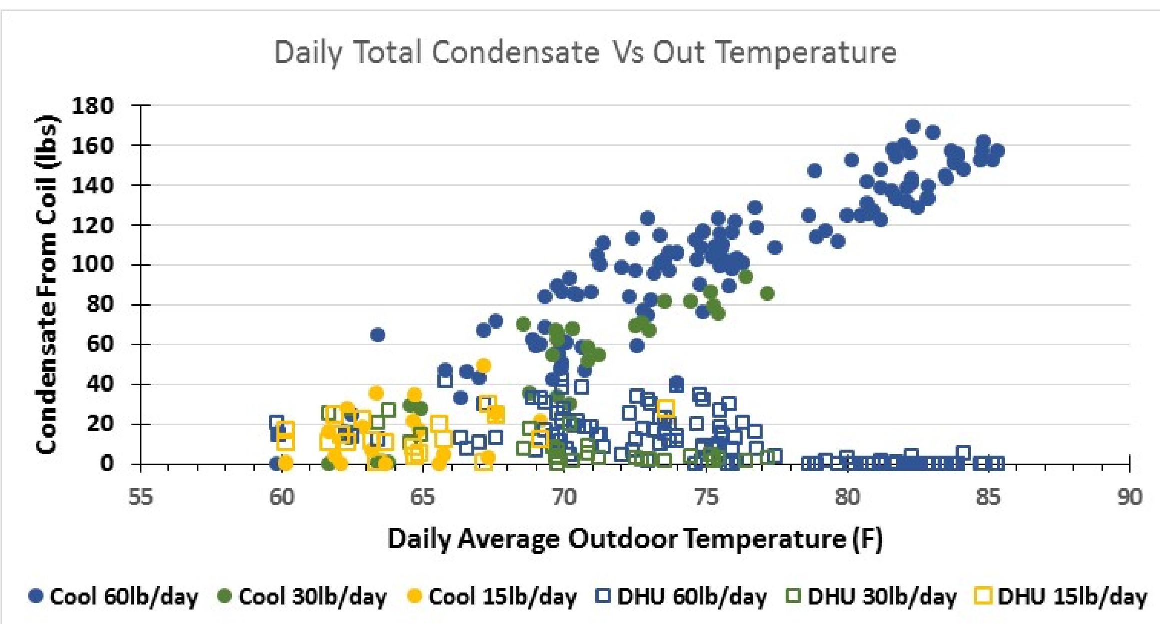

Figure C-3. Daily condensate based upon outdoor temperature with specific latent load rates indicated.

Figure C-4 shows the daily total DHU condensate versus daily total DHU runtime for Tests 1-4 at the 60 lb/day latent load. A least-squares linear regression line was plotted for the 60 lb/day data set since it had the widest range in daily DHU runtime. It showed an excellent coefficient of determination as expected. This data showed that there was no significant difference in the daily DHU indoor moisture removal rate based upon the internal latent load. The average indoor conditions of all days represented in this data set had a temperature of 76.9F and 45% RH.

Figure C-4 offers an opportunity to compare the current standard performance metric of *pints of liquid water removed / day rated at 80°F and 60 % RH* to actual measured moisture removal performance at more realistic entering conditions for Florida or other hot humid climate regions. The linear regression line indicates an extrapolated DHU of 53 lbs/day (51 pints/day) of moisture removed if the DHU ran 100% for the day with entering conditions of about 76.9°F, 45% RH, 62.6F wb, 54.0°F dp. This result is reasonable given that the as-tested entering conditions were between the manufacturer stated capacity range between 70 to 47 pints per day depending upon entering air conditions. The manufacturer stated capacity of 70 pints per day at 80°F, 60% RH, 69.6 wb, 64.9F dp, and only 47 pints per day at 70°F, 60% RH, 61.0F wb, 55.5°F dp.

The lack of more detailed published DHU performance and reality that actual entering conditions are likely going to be much different than the current standard rated test at 80°F, 60% RH makes it difficult to know how to correctly size a DHU for specific applications. DHU need to be required to be evaluated at more realistic entering air conditions to enable more consistent and accurate DHU sizing methodology. Manufacturer guidance typically uses the floor area of a space to be dehumidified to select the capacity of DHU. This clearly does not account for any load considerations.

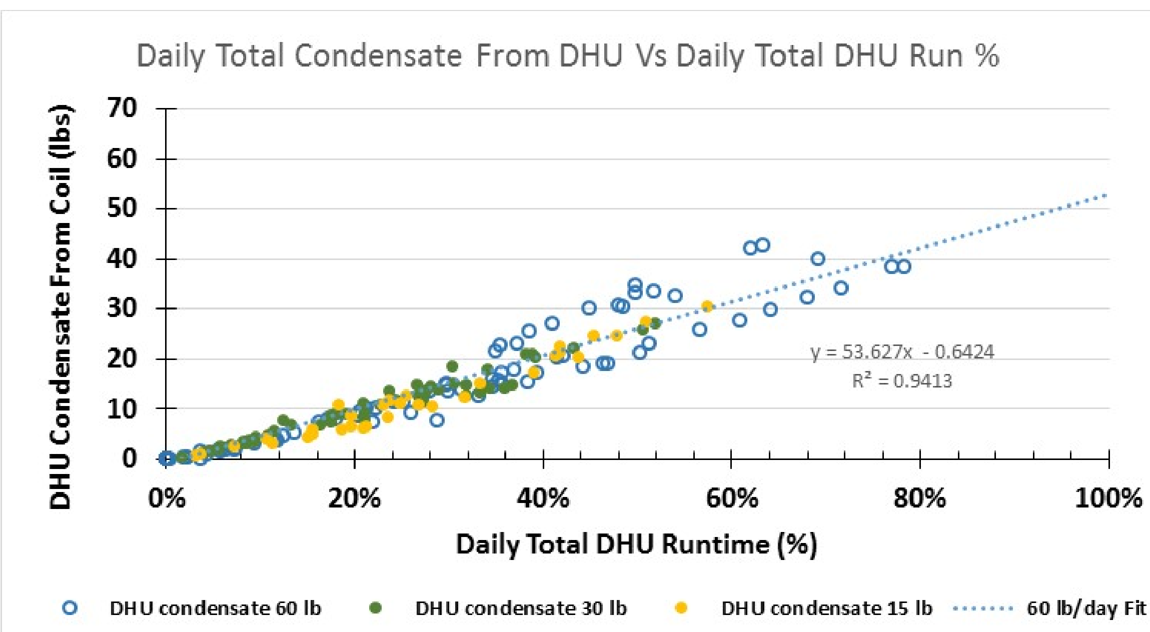

Figure C-4. Daily DHU condensate removed based upon DHU runtime at different latent loads.