FLORIDA SOLAR ENERGY CENTER®

Final Report

Investigation of Energy Impacts of Ducted Dehumidifier Duct Configurations and Location

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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 offer the lowest first-cost, are well-established in the market, and often easier to install than other alternatives; however they have the potential to use a lot of energy. Dehumidifiers may be designed to be ducted or unducted. 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 location and duct configuration.

There are several different suggestions from whole-house DHU manufacturers on how to duct these systems. This project was conducted to look for answers to two primary research questions:

1. Are there measureable space cooling and DHU energy performance impacts depending upon how or if a DHU is ducted to a central system?
2. What are potential heat gain/loss impacts related to DHU and duct location?

This project sought answers to the first question through an evaluation of three common DHU configurations that were tested in a controlled lab building. Answers to the second question were evaluated through simulation work.

The three DHU lab test configurations were:

1. DHU ducted directly from/to room
2. DHU ducted from/to the central air return duct
3. DHU from/to the central air supply duct

The three different duct location heat gains/losses evaluated were ducts in:

1. Conditioned space
2. Attic
3. Garage

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 70 pint/day DHU was controlled by an off-board wall-mounted dehumidistat set at 50% RH. Monitoring occurred from mid-December through May 2018. Following are answers in response to the first research question.

- DHU ducted from/to central return had the highest daily energy use and resulted in two primary causes of latent performance degradation.
    - DHU air degraded central latent cooling performance during simultaneous operations of both AC and DHU appliances. Temporary 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 DH was operating at the same time.
    - DHU air re-evaporated water off of warm central coil when AC was cycled off. Temporary steady-state testing just after the central system cycled off, with the DHU operating 28 continuous minutes after, measured a total 1.5 lbs of water re-evaporated off of the central cooling coil (rate of 3.2 lb/h back into condition space).
    - During one 15 minute period observation of uninterrupted monitoring, the moisture pulled out of the room air by the DHU was at about the same rate that was being re-evaporated off of the central cooling coil while the AC was cycled off. In this instance

the DHU coil rate of latent removal was -1.8 lb/h and the latent heat due to evaporation from the central cooling coil was +1.9 lb/h into the space while the DH was operating steady and the central cooling system had remained naturally cycled off 1.25 hours prior during very low cooling load period in the early morning.

- In regards to the DHU performance, under short-term steady-state testing the best latent performance and lowest measured electric power occurred when the DH was ducted to/from the central supply.
  - Compared to DH from/to conditioned space, DH from central supply had a 5% improvement in latent (Btu/h) performance, decrease of 15% electric power, and latent efficiency (pints/kWH) improved 22%.

Least-squares regression analysis of daily total space conditioning energy (central cooling + DHU) versus daily average temperature difference between outdoors and indoors was performed. Regression models were used with TMY3 data for the three Florida cities of Miami, Orlando, and Jacksonville to develop annual energy estimates for three tested DHU configurations. The annual energy are shown in Table ES-1 along with energy differences of each test configuration compared to DHU ducted from and to the conditioned space (room). Negative values indicate decreased energy and positive values indicate an increase to the basis of comparison.

Table ES-1. Predicted Annual Central Cooling and DHU Energy for Three Tested DHU Configurations at Three Florida Cities

	Miami			Orlando			Jacksonville		
	DHU room	DHU supply	DHU return	**DHU room**	DHU supply	DHU return	**DHU room**	DHU supply	DHU return
Annual kWh	8569	8464	9615	6774	6669	7576	5661	5566	6322
Delta kWH from DHU room	0	-105	1046	0	-105	802	0	-95	661
Delta % from DHU room	0	-1.2%	12.2%	0	-1.6%	11.8%	0	-1.7%	11.7%

The steady-state and longer term test findings show that DHUs should not be ducted from/to central cooling system returns upstream of the cooling evaporator coil. Regarding DHU from/to central supply ducts, steady-state testing shows some decrease in power consumption, and improved latent performance of DHU ducted from/to central supply, however these only occur when both AC and DH operate simultaneously for about 20 minutes. Longer-term testing, that includes cycling behavior, shows there is very little predicted annual difference between DHU ducted from/to room and central supply.

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 operations and generally lower impact. This study did not evaluate impacts at different setpoints. It should also be considered that the ducted DHU used was about the smallest (appropriate for the testing) ducted unit on the market. A larger DHU ducted from/to the same tested central return system would have had even worse results upon the DHU from/to central return.

The second component of the research was to develop a model that would examine a different location scenario: If a dehumidifier has an independent duct system, what is the potential impact of the location of those ducts and the location of the dehumidifier? A model was developed that proved good accuracy when compared against the measured data at steady state conditions. The model is capable of providing the effect of a dehumidifier with insulated sealed ducts independent of the space conditioning system. The ducts and DHU can be located in a space at any temperature. Six scenarios are presented: All ductwork and DHU in attic, all in conditioned space and all in garage, each for a probable summer and winter condition. The amount of heat gain or loss prior to the dehumidifier effects the performance of the dehumidifier as the capacity and efficiency are temperature dependent. By having the unit and ductwork in the attic for the characteristics and assumed conditions presented, there is a likely decrease of dehumidification capacity by about 304 Btu/h and an increased total heat gain of 219 Btu/h. A garage location would have a smaller impact than the attic.

Based on these results, the energy code should allow a ducted dehumidifier in any space in the house. Ducts should be insulated to at least R-6 and the dehumidifier box should be insulated to at least R-2 if located outside of conditioned space. It is recommended that the reference house system be specified so as for simulations to capture the difference, albeit small, between methods. The simplest and likely most common installation will be a stand-alone unducted system (or DHU located and ducted within conditioned space) and that would be the most logical to specify for the reference home as well.

Work by Vieira and Beal 2017 previously made recommendations for changes to the Energy Conservation Code. These covered mechanical ventilation and dehumidification systems and have been included in Appendix A of this final report for reference. The work completed for this project has the following recommendations to be made in addition to those of Vieira and Beal 2017:

- The code should not permit DHUs to be ducted from/to central air returns upstream of the cooling coil due to increased energy use and latent performance degradation of the central AC.
- The performance path should have an unducted stand-alone dehumidifier as its base case. The proposed home should be as installed, including any effect of ductwork in unconditioned space.

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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. 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*." This report touches on a portion of the questions posed by Vieira and Beal 2017. Portions of recommendations and questions from that report have been included in Appendix A of this report for reference.

Air conditioning (AC) alone cannot guarantee indoor RH below 60% all hours of the year. Desire for tighter controlled indoor relative humidity (RH) increases the need for supplemental dehumidification most often provided by a dehumidifier (DHU). A review of a few ducted DHU manufacturer installation manuals found several different recommended ways suggested to duct DHU. Noise level, need for a condensate drain, and preference for how air is distributed, are considerations that dictate how a whole-house ducted DHU is installed. Installation manuals and a lack of third-party published research don't address if DHU duct configuration impacts central cooling or DHU energy performance.

Before going any further, 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 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 addition to equipment performance related to how DHU is integrated with central AC, there may also be sensible load gains/losses from DHU distribution ducts that depend upon location. The most likely locations for ducted DHU in newer homes are indoors in a conditioned utility room or closet, in unconditioned attic, or unconditioned garage.

There were several possible duct installation scenarios that could be considered in this project, but only three DHU configurations and three duct locations could be evaluated in the allotted project timeline.

The configurations evaluated were those considered common with reasonable potential to have space conditioning energy impacts that could be conducted within the project timeline. The Florida Department of Business and Professional Regulation (DBPR) has contracted with the Florida Solar Energy Center (FSEC) with the primary objective to identify if specific DHU duct configurations and its locations can result in energy impacts in Florida homes; and further more determine if such impacts warrant recommendation of any energy code changes.

The contracted scope of work is summarized below in the following items:

a) Alternate the method of DHU air distribution for the three cases identified in Scope of Work item (b). Testing to be completed in FSEC Building Science Lab.
b) DHU distribution shall be configured to do the following 3 tests:
    1) DHU air from/to return side of central cooling (AC) system with gravity damper to avoid short-circuiting of DHU air. Concept illustrated in Figure 1.
    2) DHU air from/to supply side of AC system with gravity damper to avoid short-circuiting of DHU air. Concept illustrated in Figure 2.
    3) DHU air from/to the central main body of building. Concept illustrated in Figure 3.
c) For each configuration tested, the following measurements shall be made:
    1) Temperature and humidity of the entering and leaving air of the DHU and AC system.
    2) Energy use of the DHU and AC system.
    3) Condensation removal of DHU and AC system.
    4) Outdoor air temperature and humidity
d) FSEC shall simulate energy use based upon the physical location of the DHU located in a garage, conditioned indoor space, and attic location accounting for any duct gain/loss effects. FSEC will modify EnergyGauge to use it for simulations.

All contracted work according to the scope of work has been completed, however a mathematical model was implemented using Excel instead of EnergyGauge simulations for looking at steady state ducted dehumidifier performance. This final report discusses the materials, methods, analysis and results of this research project and is the final deliverable of the contract.

Background

Central cooling systems 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 (drybulb 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 drybulb 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 to the fact that some occupants may feel satisfied with homes with 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 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 questions of this project.

The remainder of this report will detail two different primary tasks. The first task was to manage controlled lab experiments to evaluate energy performance impacts from three different DHU duct configurations with respect to the central conditioning system examining the combined impact. The second primary task was to complete simulation efforts to evaluate energy impacts on independently ducted dehumidifier systems located in attics, garages or conditioned space.

Task 1: Lab Experimental Work to Determine AC and DHU Performance Impacts Based Upon DHU Duct Configuration to and Independent from Central Ducts

Experimental Materials and Equipment

This section discusses details about the test building, equipment details and data collection procedures. All DHU configuration experiments were conducted within the Building Science Lab building located on the Florida Solar Energy Center campus. This lab has a conditioned floor area of 2000 ft^2 with concrete masonry block walls having R-5 unfaced foam board insulation located on the interior side of the wall. Windows are 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 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 and return ducts resulted in measured delivered cooling at about 2.3 tons. The heat pump system was controlled by a thermostat located on an interior wall in the large open central room.

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.

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.

Interior latent loads were delivered at three different target rates. Target rates of 15, 30 and 60 pounds of water each day were evaporated into the building and distributed within the central area of building by a small circulation fan.

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 December 16-February 10. 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.

The delivered latent load was monitored throughout the project by means of tipping bucket or flow meter that provided a pulse output proportional to the volume of water passing through the evaporation assembly.

Latent heat removed as condensate drained from evaporator coils was measured using tipping buckets 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 where the number of tips were measured for a given measured mass of water. 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.

All data from sensors were collected using a Campbell Scientific, Inc. CR10 datalogger, where 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 datalogger 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. No missing scans occurred during the data used in final analysis.

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 DHU and AC system was measured by calibrated tipping buckets at each appliance. 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 Vaisala HMP60 sensors.
- Airflow stations measured central AC system airflow and DHU airflow. These were measured using digital manometers with stated pressure accuracy of +/- 1%. DHU airflow calibration was performed using a TSI Model 8390 Bench Top WindTunnel accuracy of +/- 2%.

Lab Test Method

Three primary DHU configuration experiments were conducted to evaluate the energy performance of each test configuration. Conceptual illustrations of these three lab DHU test configurations are shown in Figures 1-3. Lab test configurations evaluated were:

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

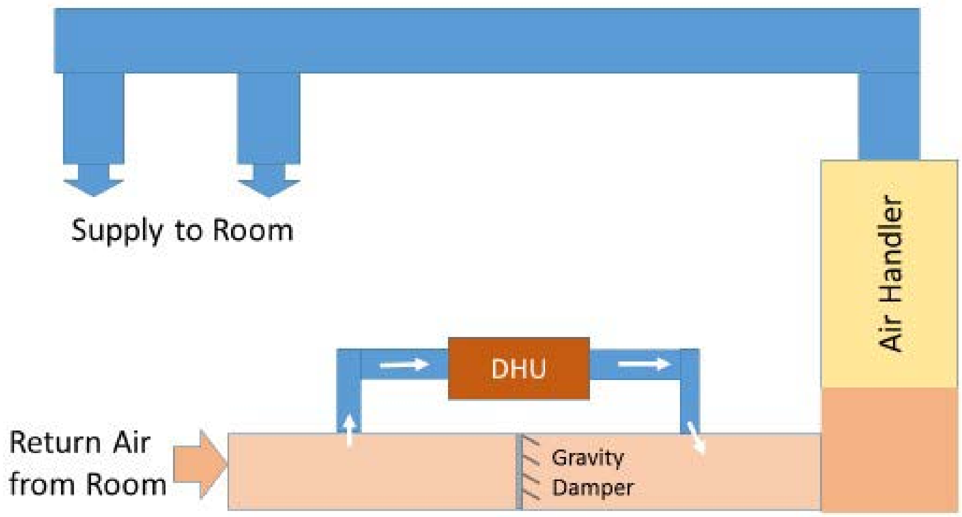

Figure 1. Conceptual illustration of DHU ducted to the main central return duct. Gravity damper only opens when central system on. When closed, it blocks short-circuiting of DHU supply air straight back into DHU return.

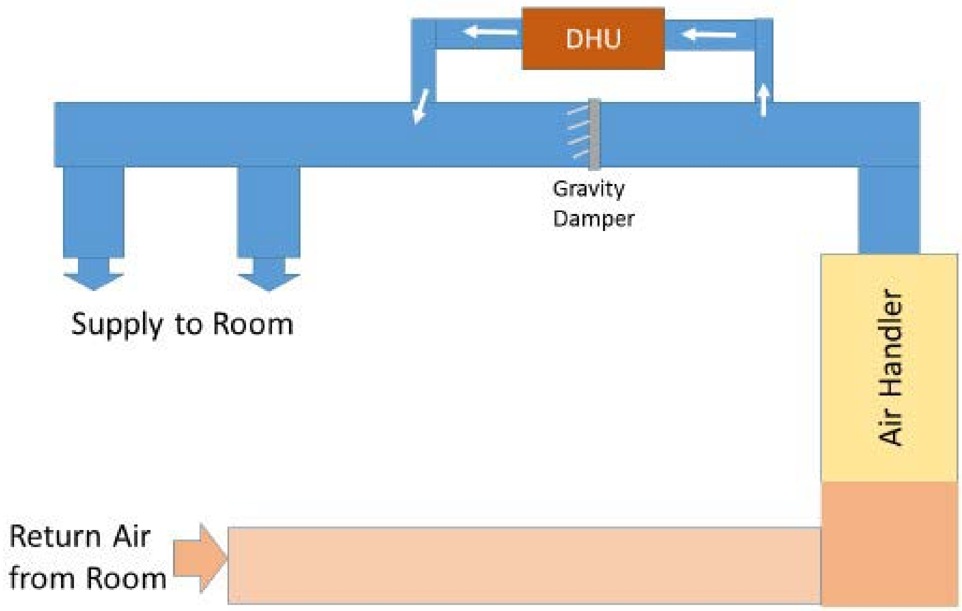

Figure 2. Conceptual illustration of DHU ducted to the main central supply duct. Gravity damper only opens when central system is on. When closed, it blocks short-circuiting of DHU supply air straight back into DHU return.

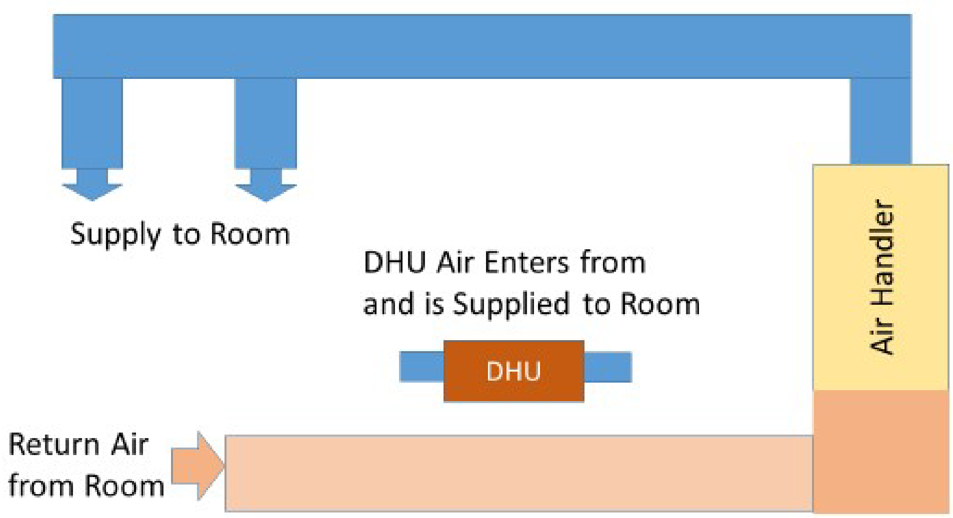

Figure 3. Conceptual illustration with DHU not ducted to central cooling ducts. DHU air directly from and back into conditioned space.

A primary reason for ducting both sides of the DHU to the same duct section is to minimize static pressure impacts across the DHU distribution fan. One of a manufacturers installation guides (not the one we tested) showed an installation option where the DHU return duct pulls against the negative static central return duct and supplies DHU air into a positive static pressure supply duct. This would not generally be a good idea unless operational static pressure is known and the manufacturer can verify this option is acceptable for their product. The Ultra-Aire 70H unit tested in this project came with installation instructions not to install to anything greater or equal to +0.5 in wg pressure. The installations tested were well below this. This project did not have the scope to evaluate potentially extreme static impacts upon DHU flowrates.

The project test method used a long straight section of central duct board return duct with the DHU supply introduced about 8 ft. upstream of cooling coil. The gravity damper was about equal distance from DHU return duct and DHU supply duct. There was about 10 ft. of distance between each DHU duct connection. Test configurations with DHU ducted from/to central ducts had gravity dampers installed within the central supply and central return to avoid short-circuiting of DHU air when the central system was not operating.

A similar DHU duct installation method to test 1 was applied to the second test where the DHU was ducted from / to the central supply duct. In this case the DHU return duct was installed about 12 ft. downstream from the central coil and the DHU supply duct was connected another 12 ft. downstream from the DHU return connection. A gravity damper was located between each DHU duct connection within the central supply.

Figure 4 shows a photo of the DHU with short straight metal duct sections connected to the unit where entering and leaving air conditions were measured. The round collar on the return side is an iris flow damper used to measure entering airflow to the DHU. Ducts with R-6 insulation were used during testing. A portion of the central return duct can be seen in Figure 4 on the floor behind the DHU. Flex ducts at each end of the DHU go to the central supply duct in the attic space and were connected to the DHU during DHU ducted from/to central supply testing.

Figure 4. Dehumidifier shown with return air flow station and condensate drain to tipping bucket on a stand.

The central system fan operation was not integrated to coincide with DHU operation. Forcing central air circulation to coincide with DHU operation is sometimes done to improve circulation of DHU air around the home, however the authors do not recommend this, since air through a wet warm central cooling coil will evaporate back into the home increasing latent load (Henderson 1990; Shirey et al. 2006) and the central fan energy use significantly increases energy use.

Lab Test Results

Evaluations were performed through limited short-term steady-state testing as well as longer-term naturally occurring testing. Short-term testing was performed to evaluate if there were any significant potential central cooling and supplemental DHU performance impacts.

Longer-term testing was designed to account for realistic operational conditions that occur over a range of outdoor weather conditions that include naturally occurring central cooling and DHU cycling impacts.

Short-Term Test Results

Short-term testing involved running central air conditioning and the DHU for prolonged periods of 30 minutes or more in order to make airside entering and leaving evaporator coil performance measurements. These tests were intended for the purpose of making comparisons and do not represent controlled rated conditions.   Tests were done to see if DHU from/to central return impacted central AC performance and to see if DHU from/to central supply impacted DHU performance.

Central Cooling System Performance

Table 1 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 86F during this testing.

Table 1. Central AC Steady-State Evaporator Coil Performance Comparisons

Test 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
% diff from AC only to AC&DHU from/to return						1.7%	14.3%	-27.9%	12.4%

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. 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 significant finding was that moisture left on the cooling coil was re-evaporated when the DHU was ducted from/to the central return and the cooling system was cycled off. During one 15 minute period of uninterrupted monitoring, the DHU coil rate of latent removal was -1.8 lb/hr and the latent heat of evaporation from the central cooling coil was +1.9 lb/hr while the DHU was operating steady and the central cooling system had remained cycled off 1.25 hours prior and during this 15 minute period. A controlled test conducted with a fully wet central cooling coil just after cycling off and DHU run for 28 minutes after measured 1.5 lbs of moisture evaporated from the central cooling coil and delivered down the central supply eventually back into the conditioned space. This 28 minute test can be seen in Figure 5. Negative values indicate that sensible or latent heat was removed from the airstream by the central AC coil. Positive values mean that heat was added. 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.

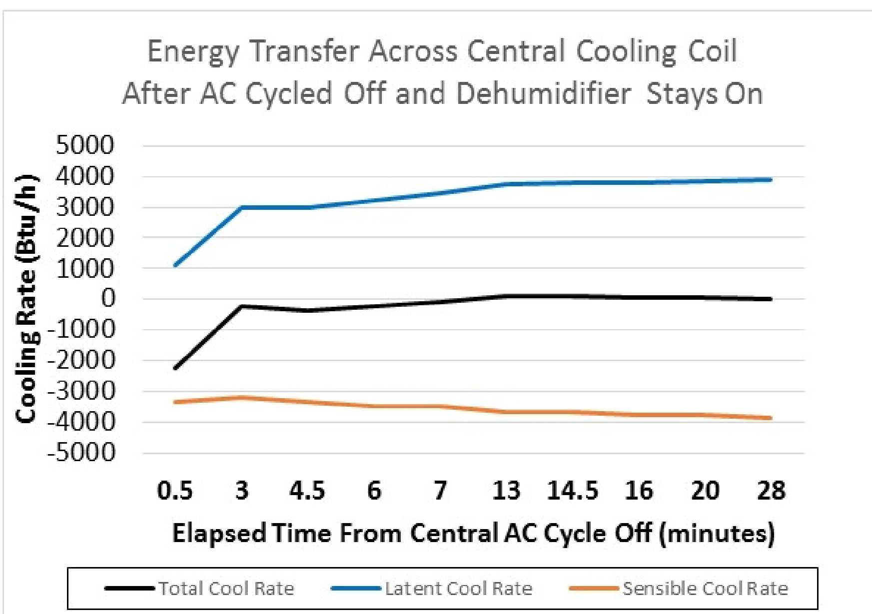

Figure 5. Moisture evaporated off a warming wet AC coil as DHU stayed on and blew through central AC coil.

These findings show that DHUs ducted from/to central cooling system returns upstream of the coiling coil can have significant performance degradation impacts upon the central cooling system.

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 2 shows manufacturer data shipped with the DHU. The data shows a 33% drop in latent capacity from rated conditions (entering air 80F/60%RH/ 69.6Fwb) to when the entering air is cooler and drier (70F/60%/58.4Fwb). The latent efficiency (pints/kWh) drops by 20%.

Table 2. Ultra-Aire 70H Manufacturer DHU Performance Data

Test Condition	Enter T (°F)	Enter RH (%)	Capacity Pints/day	Pints/kWh
Warm/moist	80	60	70	5
cool/dry	70	60	47	4
% diff from rated cond.			33%	20%

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

The DHU rated conditions (80F, 60% RH) are at higher temperature and RH than typically maintained in occupied Florida homes. Indoor average temperature and RH was 76.2F 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 3 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 3 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 80F 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 3. 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 3.

Table 3. Dehumidifier Steady-State Test Performance Comparisons

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
% diff from a) **(enter DHU warm/moist) to** b) **(enter DHU cool/dry)**						-13.3%	-27.4%	-33.8%	-11.2%	-25.5%
% diff from a) **(enter DHU warm/moist) to** c) **(enter DHU cold/dry)**						-4.5%	-22.2%	-30.3%	-24.6%	-9.1%
% diff from b) **(enter DHU cool/dry) to** c) **(enter DHU cold/dry)**						10.2%	7.2%	5.3%	-15.1%	22.0%

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 3 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 3 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.

Long-Term Test Results

The long-term testing was performed to determine if there were measureable energy differences between the three test configurations. Each test configuration was established for several days in a row before reconfiguring to the next test. This method of testing rotation continued throughout the monitoring period from December 15, 2017 – May 29, 2018. The purpose of this was to acquire as much variability in cooling and dehumidification data as possible in an effort to be able to model an annual estimate of each configuration. No significant heating data is available.

The central cooling system was controlled by a thermostat set at 76°F. The ducted DHU was run independently from a dehumidistat set at 50% RH. Both control devices were located on the same internal wall in a large central open room. Each system was left to cycle naturally.

Predicted Annual Energy

The three test configurations 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. Figure 6 shows a basic plot of daily total space conditioning (DHU + cooling) energy versus the daily average temperature difference (outdoor temperature minus indoor temperature). This shows the general trend of higher energy use at higher dT as well as higher energy at greater latent load. The trend becomes more obvious when showing one latent load rate at a time with the three test configurations

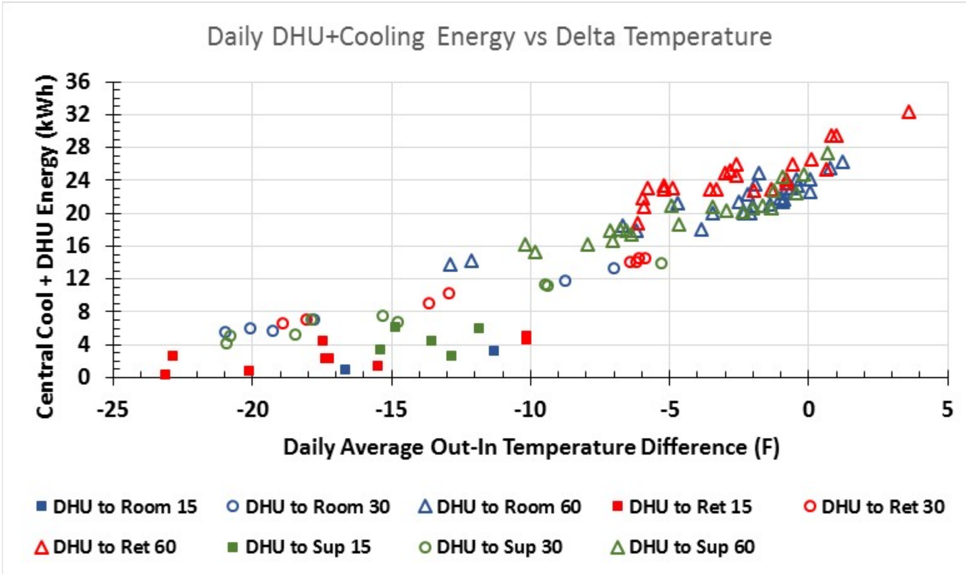

Figure 6. All three DHU configurations shown together and identified by the latent load rated during test.

The daily data shown in Figure 6 was broken down into three different groups of latent generation. Least-squares regression analysis was performed to characterize the cooling and DHU energy consumption (kWh/day) versus delta-T (outdoor temperature minus indoor temperature) of the three different test configurations at three different latent loads. Figure 7 shows the plot for the 15 lb/day latent load, Figure 8 shows results for the 30 lb/day latent rate and Figure 9 shows the results for 60 lb/day latent load. The best-fit line equations and coefficient of determination (r^2) are shown in the colored text boxes.

Figure 7 shows energy use at the lowest latent generation of 15 lb/day and during the coolest weather periods with at least some cooling and DHU energy use. DHU to room indicates lower energy use in this plot, however only two days were available in this group. Uncertainty is highest within this group, but it represents days with very low energy use.

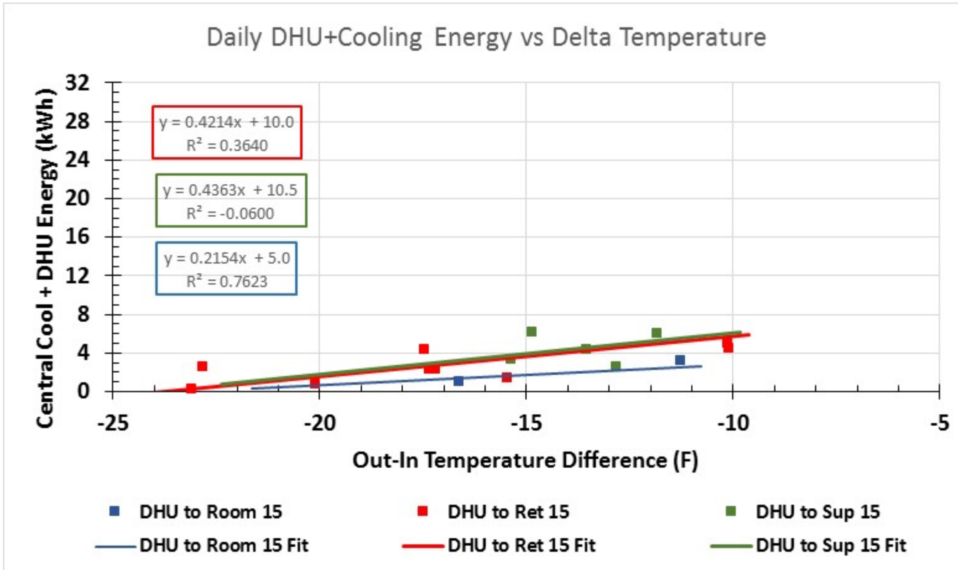

Figure 7. Daily total space conditioning energy versus dT at 15 lb/day latent load.

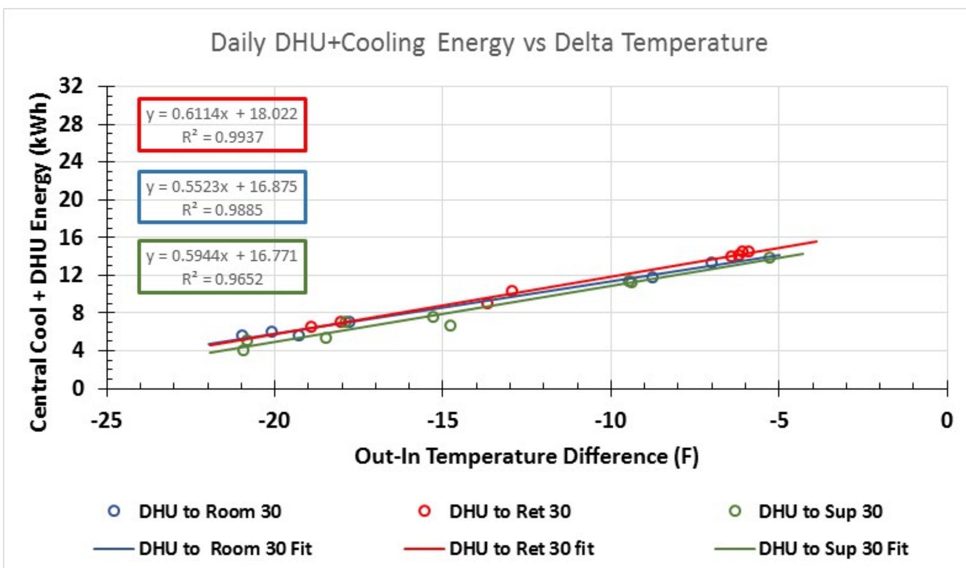

Figure 8. Daily total space conditioning energy versus dT at 30 lb/day latent load.

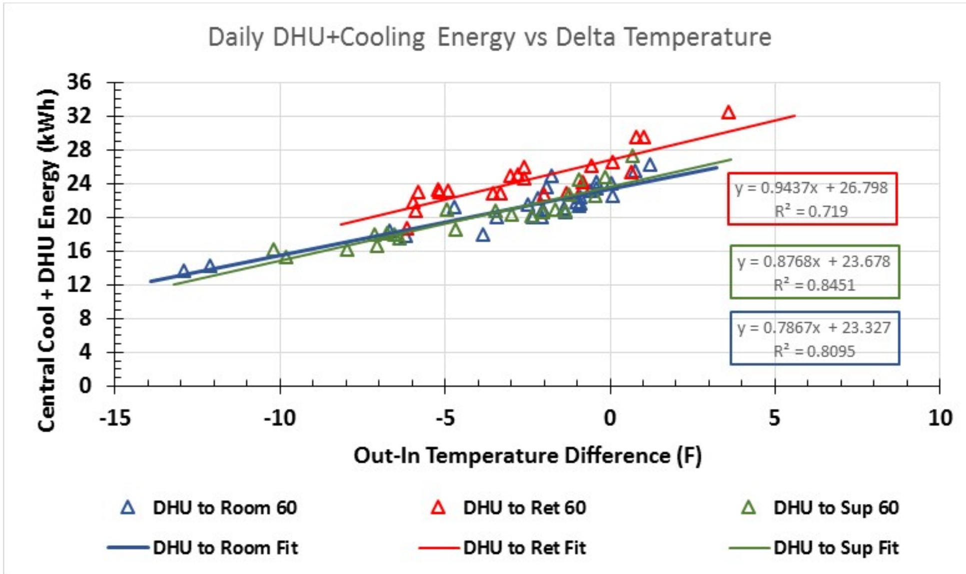

Figure 9. Daily total space conditioning energy versus dT at 60 lb/day latent load.

The impact of DHU from/to central return become most pronounced at the highest latent rate occurring during warmer weather. This makes sense as the central cooling makes up the largest fraction of space conditioning energy and DHU impact upon central cooling becomes compounded with more runtime and latent load. Recall from the steady-state testing mentioned earlier that DHU from/to central cooling return decreased central latent performance and also re-evaporates water from central coil when AC cycles off and DHU is operating.

A least squares best-fit regression model was then developed for each of the three DHU duct test configurations based on the different groups of latent loads across the full range of dT occurring during testing. The resulting linear best-fit for each test is shown in Figure 10.

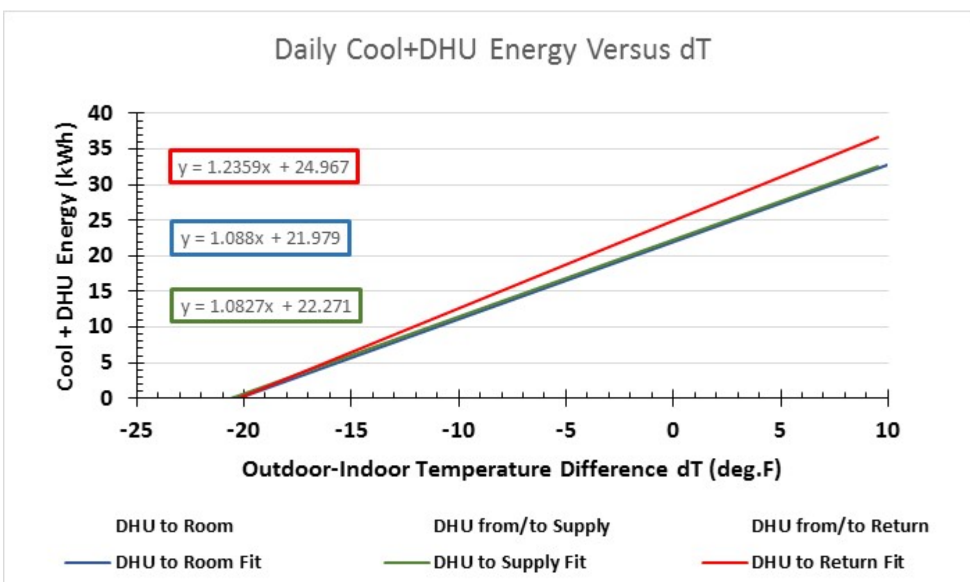

Figure 10. Least-squares best-fit results for the three DHU configurations created to predicted annual energy.

The test data analysis has shown the combination of DHU and central cooling together since there is better correlation looking at the sum. As the outdoor temperature becomes warmer (increased dT), sensible cooling load increases and the central cooling system removes more latent moisture. This results in less DHU runtime. Figure 11 shows the trend of decreasing DHU energy as a % of the total space conditioning energy (DHU+ Cooling). Additional data and analysis showing the relationship between cooling runtime and outdoor temperature as well as relationship between DHU and central cooling runtime can be found in Appendix B.

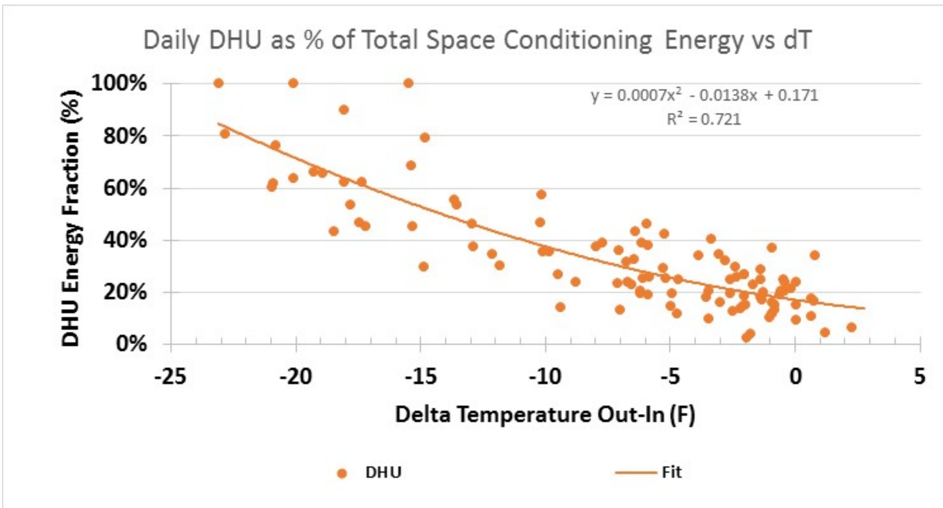

Figure 11. DHU daily energy as % of total space conditioning versus dT.

Annual space conditioning energy use was predicted using the regression fit equations shown in Figure 8 for three Florida cities of Miami, Orlando, and Jacksonville. The temperature difference was calculated based upon TMY3 outdoor temperature minus an indoor temperature of 75°F. Results for the three cities are shown in Table 4. An average of the three cities is summarized in Table 5. Negative values indicate decreased energy and positive values indicate an increase to the basis of comparison.

Table 4. Predicted Annual Central Cooling and DHU Energy for Three Tested DHU Configurations at Three Florida Cities

	Miami			Orlando			Jacksonville		
	DHU room	DHU supply	DHU return	**DHU room**	DHU supply	DHU return	**DHU room**	DHU supply	DHU return
Annual kWh	8569	8464	9615	6774	6669	7576	5661	5566	6322
Delta kWH from DHU room	0	-105	1046	0	-105	802	0	-95	661
Delta % from DHU room	0	-1.2%	12.2%	0	-1.6%	11.8%	0	-1.7%	11.7%

Table 5. Average of Three Florida Cities Predicted Annual Central Cooling and DHU Energy for Three Tested DHU Configurations

	DHU room	DHU supply	DHU return
Annual kWh	7001	6900	7838
Delta kWH from DHU room	0	-101	836
Delta % from DHU room	0	-1.4%	12.0%

The predicted annual estimated central cooling and DHU energy is lowest for the DHU ducted from/to central supply, but only by about 1% less on average than the DHU ducted to room. The DHU ducted from/to central return did show measureable significant increase of 12% more energy than compared to DHU to room.

The results here are from specific environmental conditions and at specific control setpoints. The severity of impact in individual homes will depend upon particular cooling temperature setpoints and DHU dehumidistat setpoints. Higher temperature and humidity setpoints will result in less operations and generally lower impact. This study did not evaluate impacts at different setpoints.

Task 2: Simulation to Determine Energy Impact of DHU and DHU Duct Location

Independently Ducted Dehumidification-System Model

This model represents a ducted dehumidification system that is independent of the homes central air conditioning system. This model is meant to address possible other concerns of addressing dehumidifiers in an energy or mechanical code. The dehumidification system has supply and return side ducts connected to a dehumidification unit (DHU). This model is developed such that the DHU can be installed in a conditioned, attic or garage space. The model allows supply and return ducts and the DHU cabinet to exchange heat with the environment air where they are located. The dehumidification system model assumes the following: (1) constant, uniform R-value of the ducts (2) constant, uniform R-value of the DHU cabinet, (3) constant environmental air temperature for each time step, (4) no air leakage into or from the dehumidification system, (5) dehumidistat is always located in well-mixed conditioned space regardless of location of dehumidifier unit, and (6) constant air thermo-physical properties. Figure 12 shows schematics of a ducted dehumidification system installed in attic space. The dehumidification principle is that the DHU cools entering system air below its dewpoint such that part of the moisture in the air condenses out, and rejects the net heat back to the cooled-dehumidified system air. The model calculates net heat rejected from the latent heat removed and electrical energy consumed by the DHU. A single fan serves as the supply and condenser fan.

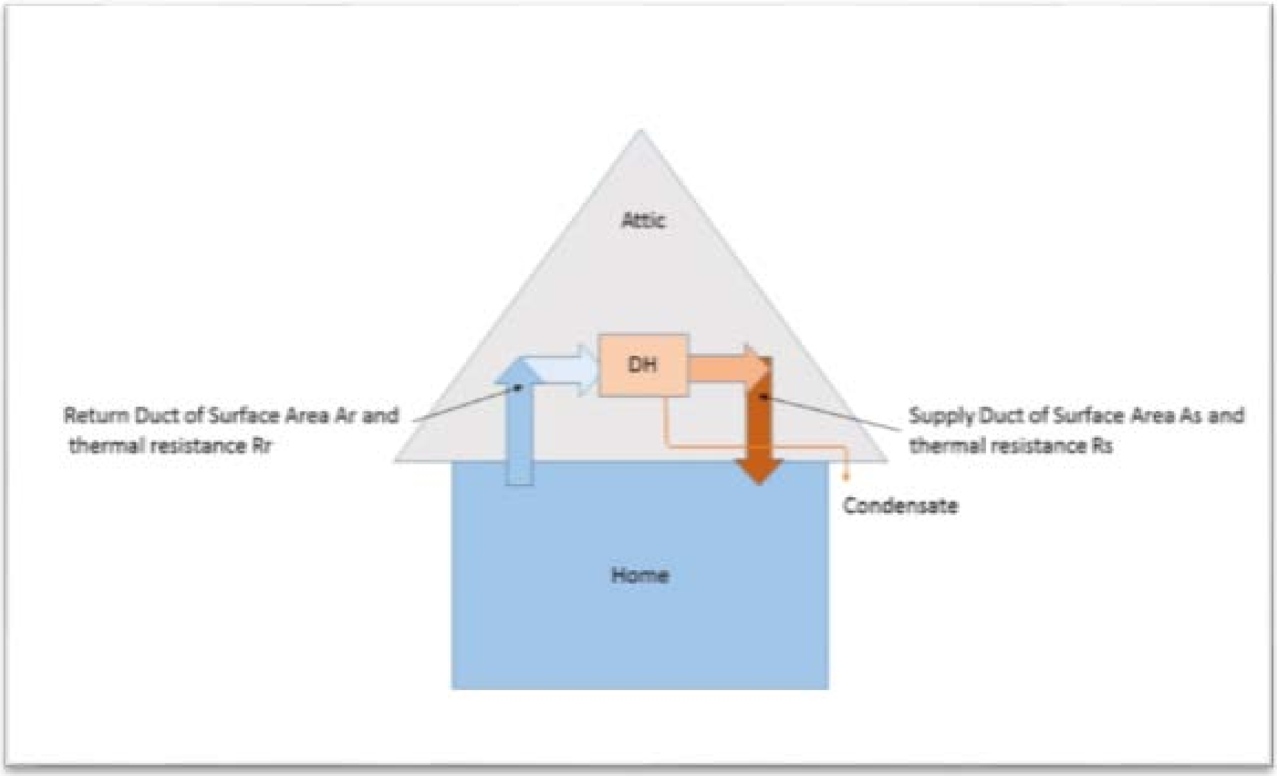

Figure 12. Schematic of independently ducted DHU located in attic space.

The model implementation procedure is presented in seven sections: model initialization, return-side heat transfer, DHU model, supply-side heat transfer, net sensible heat transfer, DHU performance curve data, and sample simulation inputs assumption and results.

Model Initialization

First the model initializes the return and supply duct surface area, R-value, DHU cabinet surface area and R-value, and thermodynamic properties of air based on known return air condition, which is the previous time step conditioned space air condition. The model calculates the supply and return ducts UA-value, and the DHU cabinet surface UA-value. The DHU cabinet surface is split into pre-coil and post-coil fractions for proper representation of the DHU operation. The current time dehumidification load is calculated based on the conditioned space relative humidity, relative humidity setpoint, and dehumidification system full supply air flow rate.

Return Side Heat Transfer

The condition of air entering the DHU depends on the location of the return ducts and where the DHU is installed. If the return ducts and DHU is installed in a conditioned space, then the condition of air entering the DHU is that of the conditioned space air; however, if the return ducts or DHU is installed in the attic or garage space, then the return air exchanges heat with the air in attic or garage space. Therefore, the return air temperature entering the DHU can be different from the conditioned space air temperature due to sensible heat gain or loss depending upon the air temperature of the attic or garage space. The model accounts for sensible heat exchange with the environment air where the DHU is installed and assumes no air leakage. The return air temperature change due to heat exchange with attic or garage space air is estimated by:

$$T_{RDLE} = T_{RDEN} + (T_{AG} - T_{RDEN}) \cdot \left(1 - e^{\left(-\frac{UA_{RD}}{\dot{m}_{SA} Cp_{SA}}\right)}\right) \qquad 1$$

Where,

$\dot{m}_{SA}$	=	system supply dry air mass flow rate, (kg/s)
Cp_{SA}	=	specific heat of air, (kJ/kg·°C)
T_{RDEN}	=	return-duct entering air dry-bulb temperature, (°C)
T_{RDLE}	=	return-duct leaving air dry-bulb temperature, (°C)
T_{AG}	=	attic or garage space air temperature, (°C)
UA_{RD}	=	return duct UA-value, (kW/m2·°C)

The return duct and the DHU cabinet pre-coil leaving air temperatures are calculated sequentially. The return duct entering air temperature is the conditioned air dry-bulb temperature, and the return duct leaving air dry-bulb temperature will be the DHU cabinet pre-coil section entering air temperature. The DHU cabinet pre-coil section temperature change due to heat exchange with attic or garage space is estimated by:

$$T_{PRCLE} = T_{RDLE} + (T_{AG} - T_{RDLE}) \cdot \left(1 - e^{\left(-\frac{UA_{PRC}}{\dot{m}_{SA} Cp_{SA}}\right)}\right) \qquad 2$$

Where,

T_{PRCLE}	=	DHU cabinet pre-coil leaving air dry-bulb temperature, (°C)
UA_{PRC}	=	DHU cabinet pre-coil UA-value, (kW/m2·°C)

Dehumidification Unit Model

This section is the core of the DHU model. The model at each time step (hourly) calculates the DHU available capacity, water removal load, the part load ratio (*PLR*), actual moisture removed, DHU runtime fraction, electric energy consumed, the latent heat removed, DHU air outlet temperature and humidity ratio, and aggregates the electric energy consumed and latent heat removed, which results in the net heat rejected by the DHU to the dehumidified supply air. Water removal capacity of the DHU at current entering air condition is given by:

$$DHCAP = DHCAP_{Rated} \cdot DHWR_{Mod} \qquad 3$$

Where,

$DHCAP_{Rated}$ = DHU rated capacity, (kg/s)

$DHWR_{Mod}$ = DHU rated capacity modifier normalized curve value at current DHU entering air dry-bulb temperature and relative humidity, (-)

The water removal capacity modifier normalized curve is calculated from a bi-quadratic curve function of the DHU entering air dry-bulb temperature (°C) and relative humidity (%) and is given by:

$$DHWR_{Mod} = a + b \cdot T_{PRCLE} + c \cdot T_{PRCLE}^2 + d \cdot RH_{PRCLE} + e \cdot RH_{PRCLE}^2 + f \cdot T_{PRCLE} \cdot RH_{PRCLE} \tag{4}$$

Where the bi-quadratic curve coefficients *a*, *b*, *c*, *d*, *e* and *f* are usually generated from manufacturer's performance data using equation fit but the coefficients used in this analysis and provided in Table came from EnergyPlus (US Department of Energy, 2018). T_{PRCLE} and RH_{PRCLE} are the dry-bulb temperature and relative humidity of the return air entering the DHU, respectively, are the independent variables. The water removal capacity modifier normalized curve adjusts the rated capacity based upon the rated DHU entering air condition. The DHU rated condition is 80°F (26.67°C) and 60% relative humidity.

Dehumidification Load: dehumidification load is calculated from the DHU full (dry air) mass flow rate and the current conditioned space humidity ratio and humidity ratio setpoint values as follows:

$$\dot{m}_{WaterREM} = \dot{m}_{SA} \cdot (\omega_Z - \omega_{ZSP}) \tag{5}$$

Where,

$\dot{m}_{WaterREM}$ = water removal load, (kgWater/s)

ω_Z = conditioned space humidity ratio, (kgWater/kgDryAir)

ω_{ZSP} = conditioned space humidity ratio set point, (kgWater/kgDryAir)

DHU Part Load Ratio: part load ratio (*PLR*) is defined as follows:

$$PLR = \frac{\dot{m}_{WaterREM}}{DHCAP} \tag{6}$$

$$PLR = MAX(MIN(PLR, 1.0), 0) \tag{7}$$

Actual moisture removed by DHU is the product of *PLR* and the DHU current full capacity; it is determined by:

$$\dot{m}_{WaterREM} = PLR \cdot DHCAP \tag{8}$$

DHU Latent Heat Removed: latent heat removed by the DHU is determined by:

$$\dot{Q}_{LAT} = \frac{\dot{m}_{WaterREM} \cdot h_{fg}}{1000} \quad 9$$

Where,

$\dot{Q}_{LAT}$	=	latent heat removed by DHU, (kW)
h_{fg}	=	latent heat of vaporization of water, (J/kgWater)

Electric Power Input: electrical energy consumption rate is calculated as follows:

$$\dot{Q}_{ELEC} = 3600 \cdot \frac{DHCAP}{DHEF} \cdot DHRTF \quad 10$$

$$DHEF = DHEF_{Rated} \cdot DHEF_{Mod} \quad 11$$

Where,

$\dot{Q}_{ELEC}$	=	total electric power input to DHU, (kW)
$DHEF$	=	DHU energy factor at current entering air condition, (kgWater/kWh)
$DHEF_{Rated}$	=	DHU energy factor at rated condition, (kgWater/kWh)
$DHEF_{Mod}$	=	DHU rated energy factor modifier normalized curve value at current DHU entering air dry-bulb temperature and relative humidity, (-)

The energy factor modifier normalized curve is calculated from a bi-quadratic curve function of the DHU entering air dry-bulb temperature (°C) and relative humidity (%) and is given by:

$$DHEF_{Mod} = a + b \cdot T_{PRCLE} + c \cdot T_{PRCLE}^2 + d \cdot RH_{PRCLE} + e \cdot RH_{PRCLE}^2 + f \cdot T_{PRCLE} \cdot RH_{PRCLE} \quad 12$$

The energy factor modifier normalized curve adjusts the rated energy factor based upon the rated DHU entering air condition. Where the bi-quadratic curve coefficients *a*, *b*, *c*, *d*, *e* and *f* are generated from manufacturer's performance data using equation fit but the coefficients used in this analysis and provided in Table came from EnergyPlus (US Department of Energy, 2018). T_{PRCLE} and RH_{PRCLE} are the dry-bulb temperature and relative humidity of the return air entering the DHU, respectively. The DHU rated condition is 80°F (26.67°C) and 60% relative humidity. The DHU run time fraction (RTF) is calculated from current *PLR* and part load fraction (*PLF*):

$$DHRTF = \frac{PLR}{DHPLF} \quad 13$$

$$DHRTF = MAX(MIN(DHRTF, 1.0), 0) \quad 14$$

The DHU *PLF* is calculated from normalized linear curve as a function of *PLR* and is given by:

$$DHPLF = a + b * PLR \quad 15$$

The sensible heat added to the dehumidified supply air by the DHU ($\dot{Q}_{DHSENS}$) is the sum of the total electrical power input rate and the latent heat removed by the DHU and is given by:

$$\dot{Q}_{DHSENS} = \dot{Q}_{LAT} + \dot{Q}_{ELEC} \quad 16$$

The sensible heat added by the DHU to the dehumidified air results in an increase in DHU leaving air dry-bulb temperature and is calculated as follows:

$$T_{DHLE} = T_{PRCLE} + \left(\frac{\dot{Q}_{DHSENS}}{\dot{m}_{SA}Cp_{SA}}\right) \quad 17$$

Humidity ratio of the dehumidified air leaving the DHU is determined by:

$$\omega_{DHLE} = \omega_{DHEN} - \left(\frac{\dot{m}_{WaterREM}}{\dot{m}_{SA}}\right) \quad 18$$

Where,

ω_{DHLE}	=	DHU leaving air humidity ratio, (kgWater/kgDryAir)
ω_{DHEN}	=	DHU entering air humidity ratio, (kgWater/kgDryAir)
T_{DHLE}	=	DHU leaving air dry-bulb temperature, (°C)

Supply-Side Heat Transfer

The supply air leaving the DHU also exchanges heat with space air in which it is located so-long as there exists temperature difference across the DHU cabinet and the supply duct. Just like the return-side heat transfer calculation, the supply-side DHU cabinet post coil and the supply duct section conduction heat gain or loss calculations are performed sequentially. The DHU leaving air dry-bulb temperature will be supply-side DHU cabinet post coil entering air dry-bulb temperature. The DHU cabinet post-coil section leaving air dry-bulb temperature will be entering air dry-bulb temperature for the supply duct heat transfer calculation. The supply air dry-bulb temperature leaving the DHU cabinet post-coil section is estimated by:

$$T_{POCLE} = T_{DHLE} + (T_{AG} - T_{DHLE})\left(1 - e^{\left(-\frac{UA_{DHPOC}}{\dot{m}_{SA}Cp_{SA}}\right)}\right) \quad 19$$

Where,

T_{DHLE}	=	DHU leaving air dry-bulb temperature, (°C)
T_{POCLE}	=	DHU cabinet-post coil section leaving air dry-bulb temperature, (°C)
UA_{DHPOC}	=	DHU cabinet-post coil UA-value, (kW/m2·°C)

The supply-duct leaving air dry-bulb temperature is estimated by:

$$T_{SDLE} = T_{POCLE} + (T_{AG} - T_{POCLE})\left(1 - e^{\left(-\frac{UA_{SD}}{\dot{m}_{SA}Cp_{SA}}\right)}\right) \quad 20$$

Where,

T_{SDLE} = supply-side leaving air dry-bulb temperature, (°C)
UA_{SD} = supply duct UA-value, (kW/m2·°C)

Net Sensible Heat Transfer

Net sensible heat gain or loss of the system ($\dot{Q}_{SENS}$) is the sum of the heat transfer to the system air across the return-side, the DHU, and the supply-side of the ducted dehumidification system and is determined by:

$$\dot{Q}_{SENS} = \dot{Q}_{RSSENS} + \dot{Q}_{DHSENS} + \dot{Q}_{SSSENS} \quad 21$$

Sensible heat gain or loss of system at the return-side of the DHU ($\dot{Q}_{RSSENS}$) is determined by:

$$\dot{Q}_{RSSENS} = \dot{m}_{SA} \cdot Cp_{SA} \cdot (T_{RDEN} - T_{PRCLE}) \cdot PLR \quad 22$$

Sensible heat gain or loss of system at the supply-side of the DHU ($\dot{Q}_{SSSENS}$) is determined by:

$$\dot{Q}_{SSSENS} = \dot{m}_{SA} \cdot Cp_{SA} \cdot (T_{SDLE} - T_{DHLE}) \cdot PLR \quad 23$$

Net sensible heat removed or added to the attic or garage space ($\dot{Q}_{AG}$) is determined by:

$$\dot{Q}_{AG} = -\dot{Q}_{SSSENS} - \dot{Q}_{RSSENS} \quad 24$$

DHU Performance Data

The capacity modifier normalized curve coefficients, the Energy Factor modifier normalized curve coefficients, and part load fraction curve coefficients used in the model are provided in Table 6. These factors came from the EnergyPlus model (US Department of Energy, 2018).

Table 6. Performance Curves of the DHU Model

Capacity(DHWR) and Energy Factor(DHEF) Modifier Normalized Bi-quadratic Curves		
$Curve_{Mod} = a + b \cdot X_1 + c \cdot X_1^2 + d \cdot X_2 + e \cdot X_2^2 + f \cdot X_1 \cdot X_2$		
Coefficients / Variables	*DHWR*	*DHEF*
a	-2.724880	-2.388320
b	0.100712	0.093048
c	-0.000990	-0.001370
d	0.050053	0.066534
e	-0.000200	-0.000340
f	-0.0003400	-0.000560
X_1	T_{DHEN}	T_{DHEN}
X_2	RH_{DHEN}	RH_{DHEN}
Part Load Fraction Normalized Linear Curve		
$PLF = a + b \cdot X$		
Coefficients / Variables	*Part Load Fraction (PLF)*	
a	*0.95*	
b	0.05	
X	*PLR*	

Experimental and Simulation Results Comparison

To validate the model the steady-state performance of the DHU model is compared with the measured data. In the experimental model validation set-up the DHU and the return and supply ducts were entirely in the conditioned space. The return and supply ducts were 6.0 ft and 7.0 ft long, respectively. The other model input assumptions are summarized in Table 10. Otherwise, simulation model input assumptions and experimental set-up for these three test cases can be assumed to have reasonably “identical” operating condition.

Good steady-state test periods were infrequent. Three data points were selected. Return duct entering air condition and supply air flow rates for the three test cases are summarized in Table 7.

Table 7. Measured Return Duct Entering Air Condition of Ducted DHU

Test No.	Return Duct Entering Temperature, °F	Return Duct Entering Relative Humidity, %	Supply Air Flow Rate, cfm
1	82.7	57.6	165.6
2	82.8	57.3	164.6
3	82.8	57.3	164.5

Measurements of the performance variables were recorded after the DHU reached steady state operating condition. Measured and predicted supply air dry-bulb temperature and relative humidity leaving the DHU for the three test cases are summarized in Table 8. The model predicted DHU leaving air temperature and relative humidity are within 1.8% and 6.1% of the measured values, respectively.

Table 8. Measured and Predicted DHU Leaving Air Condition Comparison

Test No.	DHU Leaving Air Temperature, °F			DHU Leaving Air Relative Humidity, %		
	Measured	Predicted	Difference, %	Measured	Predicted	Difference, %
1	110.1	112.1	1.8	17.4	16.3	-6.1%
2	110.6	112.3	1.5	17.1	16.2	-5.4%
3	110.6	112.3	1.5	17.0	16.2	-4.9%

Table 9 summarizes the DHU measured and predicted performance for the three test cases under steady state condition. The predicted sensible heat addition rate (Btu/h) to the conditioned space by the DHU is within 2.5% of the measured value, and the predicted moisture removal rate (lb/h) is within 5.1% of the measured value.

Table 9. Measured and Predicted DHU Performance Comparison

Test No.	Sensible Heat added by DHU, Btu/h			Moisture Removed Rate, lb/h		
	Measured	Predicted	Difference, %	Measured	Predicted	Difference, %
1	4999	5122.2	2.5%	-3.20	-3.047	-4.8%
2	5041	5110.7	1.4%	-3.20	-3.036	-5.1%
3	5041	5110.6	1.4%	-3.20	-3.036	-5.1%

It can be inferred that under a steady state operating condition the model predicts the measured performance variables within a reasonable accuracy. The small difference observed in the predicted and measured variables can be attributed to: (1) constant, uniform R-values of DHU cabinet and ducts, (2) simplifying assumptions introduced in the algorithm such as zero air leakage and well-mixed space air model, (3) constant thermo-physical properties, (4) constant space air condition at each time step (hourly), (5) instrumentation measurement accuracy, and (6) using generic performance curve from EnergyPlus instead of specific for the unit tested.

Simulation Inputs Assumptions and Results

Attic, Conditioned Space and Garage Located DHU Simulation Results

We have run the model for six sets of test conditions for a ducted DHU, with return and supply ducts located in attic, conditioned, and garage spaces; one with summer operating condition, and another with winter operation conditions for each of the DHU location. Table 10 summarizes model input assumptions that are common for all the six test cases. These model input parameters include: DHU design flow rate, DHU rated capacity and energy factor, DHU cabinet surface area and R-values, duct geometry and duct R-values, Conditioned Space Volume, etc.

Table 10. Inputs Assumptions for Model Testing

Input Parameters	Units	Value
DH Flow Rate	cfm	150
DHU Capacity	pints per day	70
DHU Energy Factor Rated	liters/kWh	2.4
DHU Energy Factor Rated	pints/kWh	5.07
DHU Cabinet Surface Area	ft^2	8.64
DHU Box R-Value Insulated	$ft^2 \cdot °F \cdot h/Btu$	2.00
Insulated DHU Box Portion	-	0.999
DHU Box R-Value Uninsulated	$ft^2 \cdot °F \cdot h/Btu$	0.01
Uninsulated DHU Box Portion	-	0.001
DHU Cabinet Air-To-Air UA-Value	$Btu/h \cdot °F$	2.73
Pre-Coil Fraction of DHU Cabinet Surface Area	-	0.100
Post-Coil Fraction of DHU Cabinet Surface Area	-	0.900
Return and Supply Duct Circumference	ft	2.8798
Return Duct Length	ft	30
Return Duct Area	ft^2	86.39
Return Duct R-Value	$ft^2 \cdot °F \cdot h/Btu$	6.0
Return Duct Air-To-Air R-Value	$ft^2 \cdot °F \cdot h/Btu$	7.1
Supply Duct Length	ft	30.0
Supply Duct Area	ft^2	86.39
Supply R-Value	$ft^2 \cdot °F \cdot h/Btu$	6.0
Supply Duct Air-To-Air R-Value	$ft^2 \cdot °F \cdot h/Btu$	7.1
Duct Inside Convection Heat Transfer Coefficient	$Btu/ft^2 \cdot °F \cdot h$	4.40
Duct Outside Convection Heat Transfer Coefficient	$Btu/ft^2 \cdot °F \cdot h$	1.06
Conditioned Space Air Volume	ft^3	20000
Barometric Pressure	psia	14.69
Specific Heat of Air	$Btu/lb \cdot °F$	0.241
Density of Air	$lbDryAir/ft^3$	0.0765

Table 11 shows the conditioned, attic and garage space conditions and set-points for summer and winter test operating conditions.

Table 11. Conditioned Space and Attic Space Conditions for Model

Parameters	Units	Summer	Winter
Conditioned Space Air Relative Humidity	%	65.0	65.0
Conditioned Space Air Temperature	°F	75.0	70.0
Attic Space Air Temperature	°F	125	50.0
Garage Space Air Temperature	°F	85	60.0
DH Relative Humidity On Set Point	%	65.0	60.0
DH Relative Humidity Off Set Point	%	50.0	50.0

Figure 13 and Figure 14, respectively, show summer and winter operating conditions performance results for DHU located in an attic space. Figure 15 and Figure 16, respectively, show summer and winter operating conditions performance results for DHU located in a conditioned space. Figure 17 and Figure 18, respectively, show summer and winter operating conditions performance results for DHU located in a garage space.

Comparing performance of the DHU located in attic space with that of conditioned space, the DHU in attic has reduced moisture removal rate and increased electric energy use due to higher DHU entering air temperature. The return-duct heat gain, which results in 3.6 °F DHU entering air temperature increase for the characteristics and assumed summer test conditions presented, reduced performance of the DHU located in the attic. This results in about 304 Btu/h dehumidification capacity decrease and a total (sensible heat gain from the operation minus latent removed) increased heat gain of 219 Btu/h. For similar characteristics and assumed summer test conditions of dehumidifier located in garage space, the DHU entering air temperature rise is 0.7°F only. Hence, DHU located in garage space and conditioned space for the modeled conditions show marginal performance difference.

The part load ratio (*PLR*) calculated and reported for the six sample test cases should be interpreted as fraction of the full capacity moisture removed in order to keep the conditioned space at 50% relative humidity setpoint. Thus, the PLR is an indicator of the DHU run-time-fraction required for the DHU to maintain the conditioned space relative humidity set point and can be slightly different for each of the test cases depending on DHU location and environment air temperature. The total sensible heat added to the conditioned space is the net cooling load in summer that the air conditioning system expected to remove had the AC system run in tandem with ducted DHU. The total sensible heat added is also different for each test case depending on the DHU location and test season.

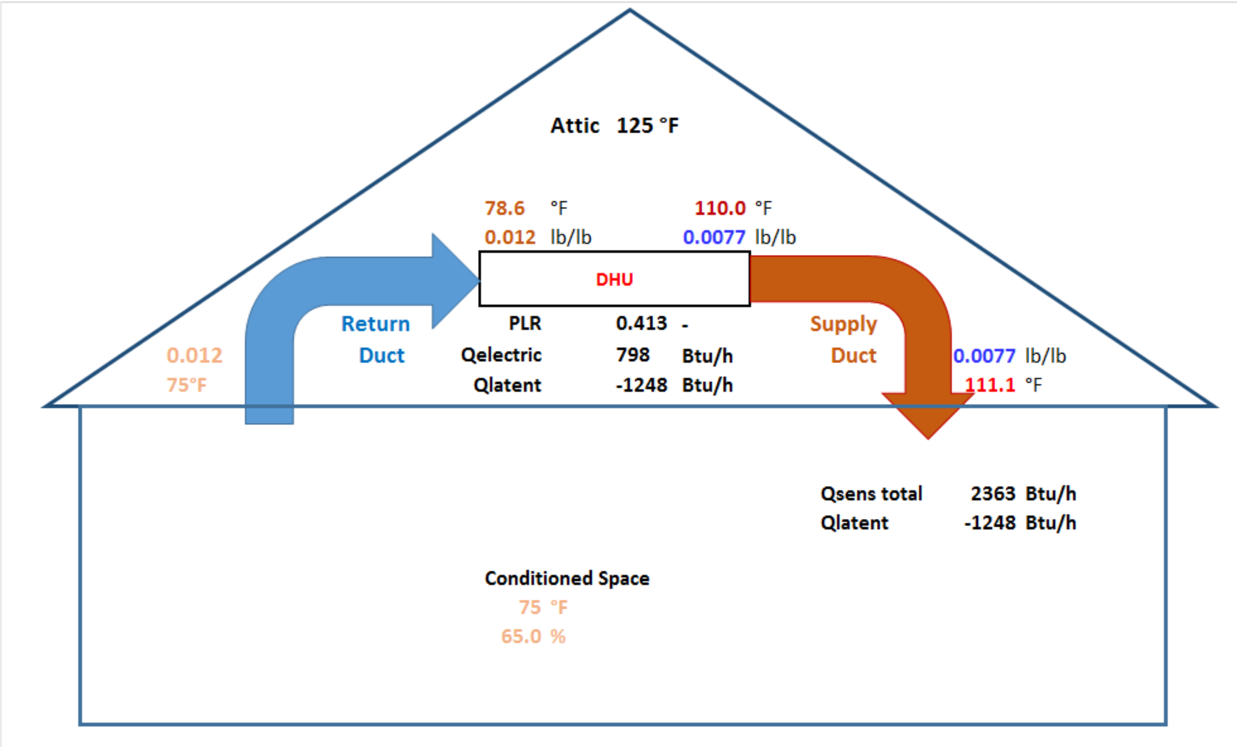

Figure 13. Schematic of DHU attic space in summer operating condition.

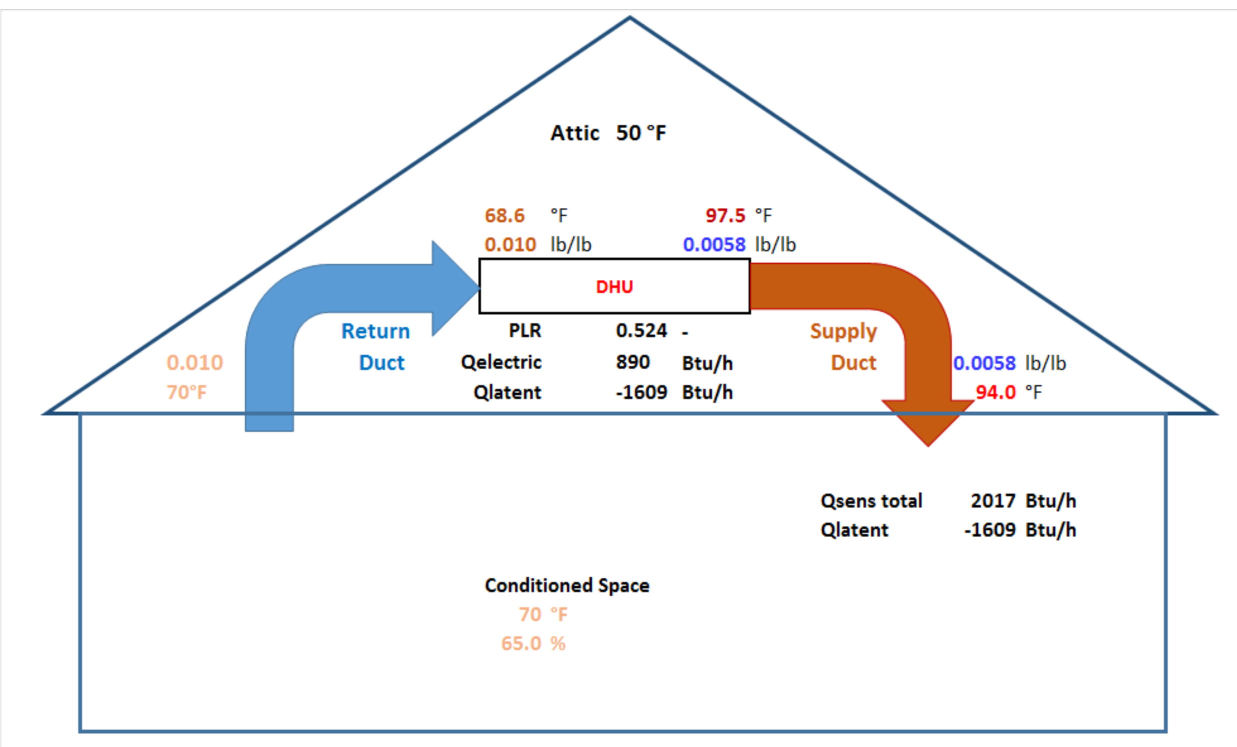

Figure 14. Schematic of DHU in attic space in winter operating condition.

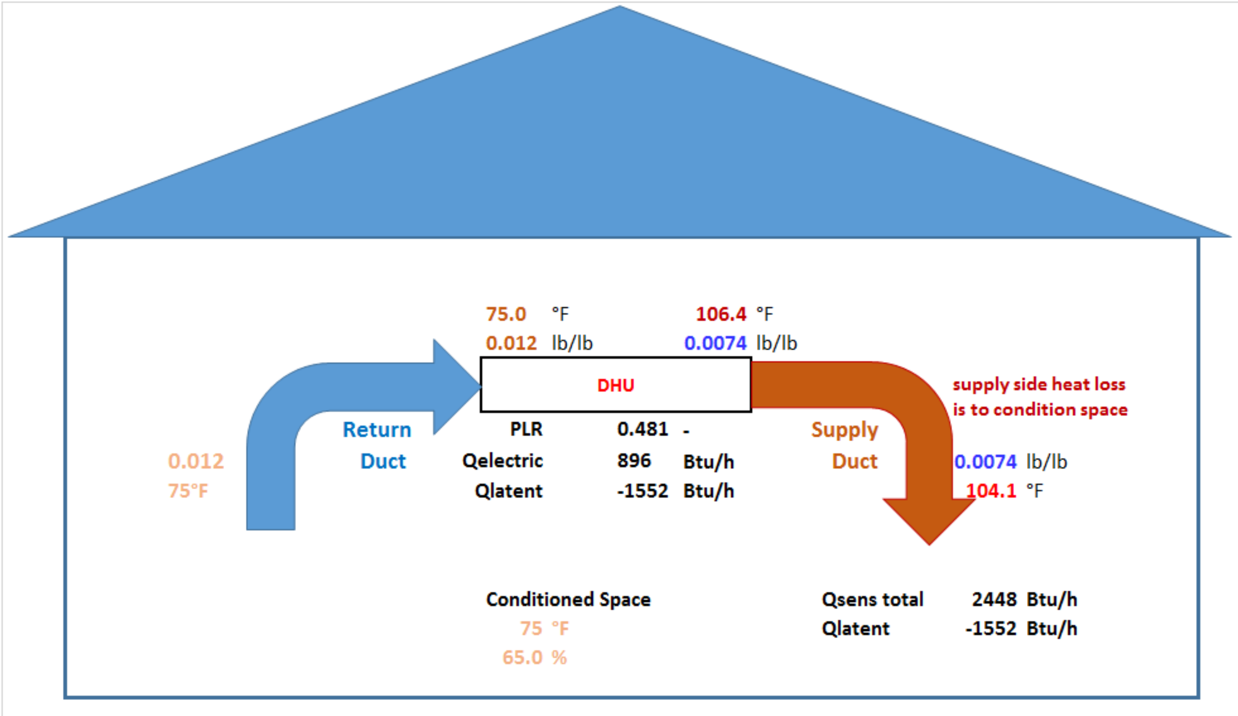

Figure 15. Schematic of DHU in conditioned space in summer operating condition.

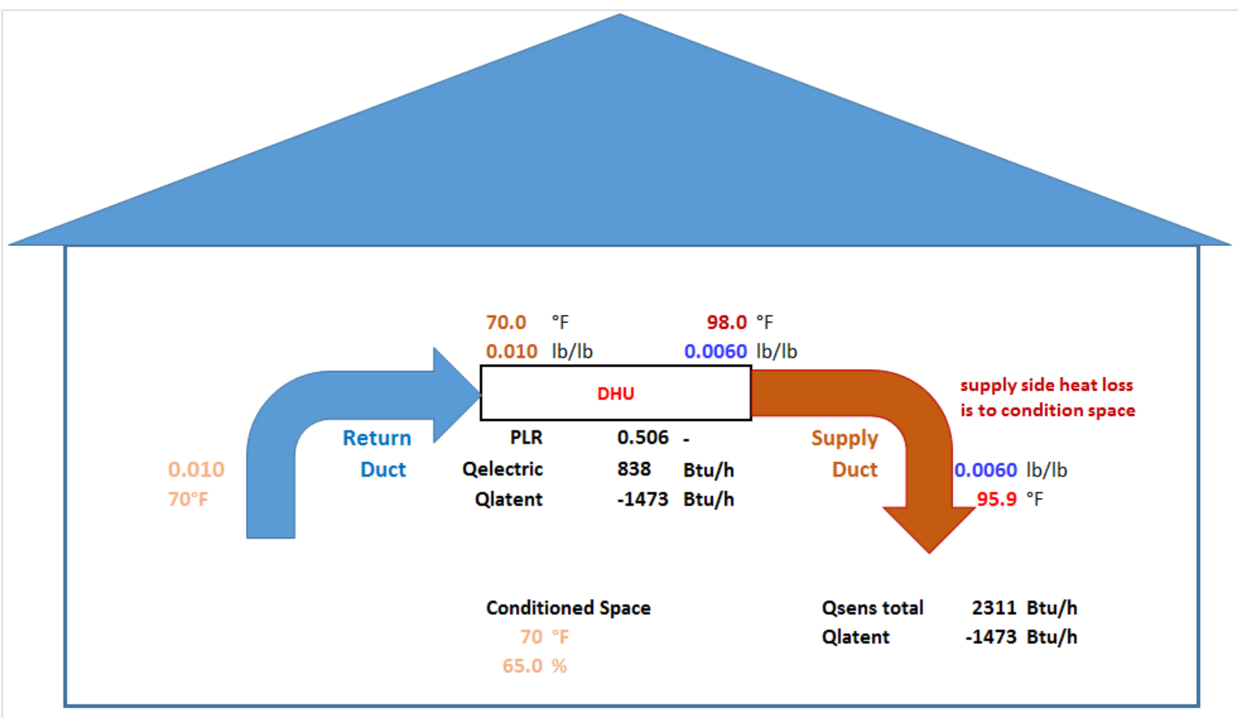

Figure 16. Schematic of DHU in conditioned space in winter operating condition.

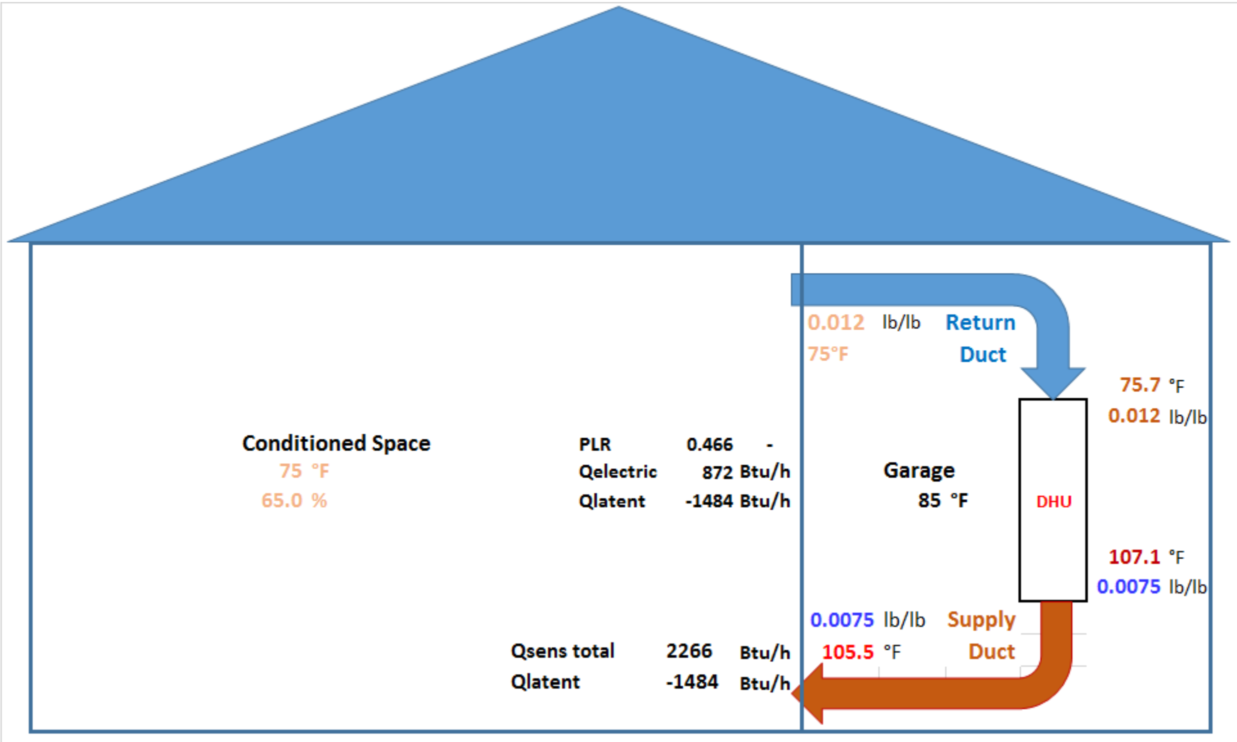

Figure 17. Schematic of DHU in garage space in summer operating condition.

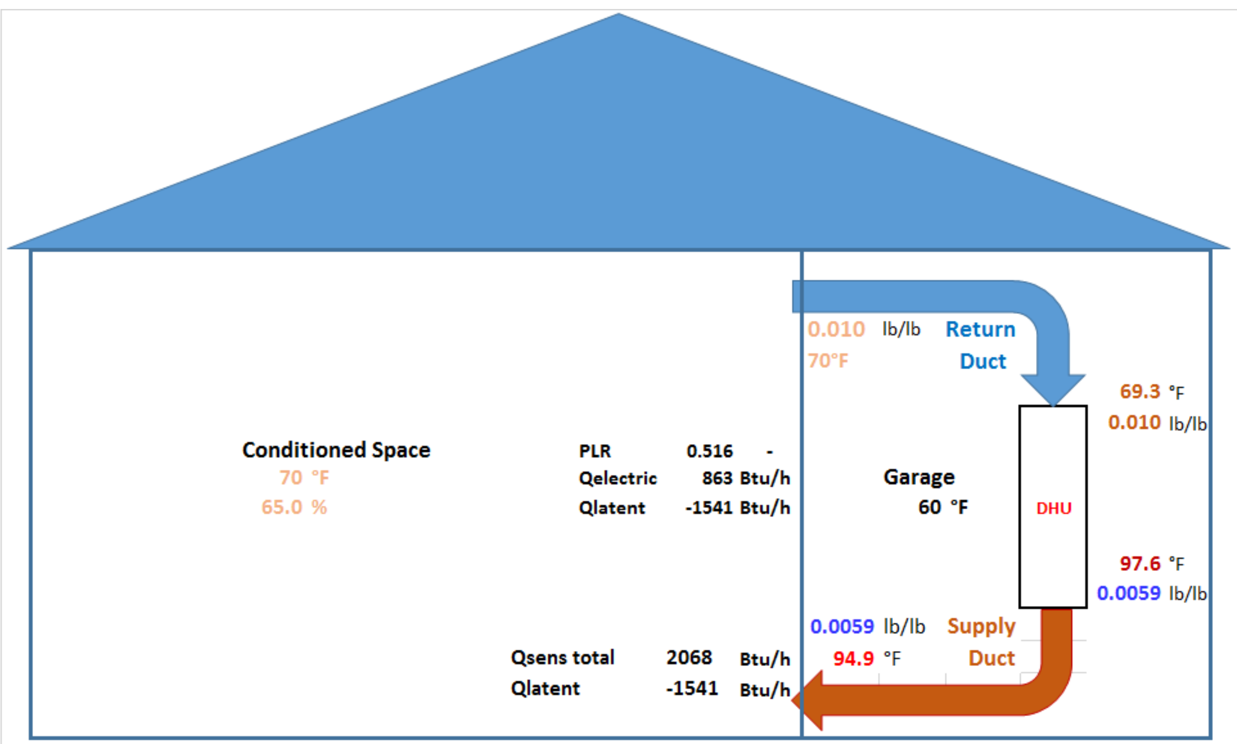

Figure 18. Schematic of DHU in garage space in winter operating condition.

Task 2 Simulation Conclusion

A model was developed that proved good accuracy when compared against measured data at steady state conditions. The model is capable of providing the effect of insulated sealed ducts in a space at any temperature. Six scenarios are presented: All ductwork and DH unit in attic, all in conditioned space and all in garage, each for a probable summer and winter condition. The amount of heat gain or loss prior to the dehumidifier effects the performance of the dehumidifier as the capacity and efficiency are temperature dependent. By having the unit and ductwork in the attic for the characteristics and assumed conditions presented, there is a likely decrease of dehumidification capacity by about 304 Btu/h and an increased total heat gain of 219 Btu/h. A garage location would have a smaller impact than the attic.

Based on these results, the energy code should allow a ducted dehumidifier in any space in the house. Ducts should be insulated to at least R-6 and the dehumidifier box should be insulated to at least R-2 if located outside of conditioned space. It is recommended that the reference house system be specified so as for simulations to capture the difference, albeit small, between methods. The simplest and likely most common installation will be a stand-alone unducted system and that would be the most logical to specify for the reference home as well.

Conclusion

This project conducted two primary tasks set out to determine answers to some basic questions regarding ducted dehumidifiers. Questions such as:

- Is there cooling or dehumidifier performance degradation based upon how a dehumidifier is ducted to central cooling systems?
- Is it better to have DHU air enter from room and discharge directly back into room?
- Are there significant heat gains/losses associated with DHU and DHU ducts in attic, garage and within conditioned space locations?
- Should the code mandate rules or allow penalties/credits based on location in performance code while limiting options in prescriptive?

A lab-based experiment study was used 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. The three DHU lab test configurations were:

1. DHU ducted directly from/to room
2. DHU ducted from/to the central air return duct
3. DHU from/to the central air supply duct

DHU ducted from/to central return had the highest daily energy use and resulted in two primary causes of latent performance degradation.

- DHU air degraded central latent cooling performance during simultaneous operations of both AC and DHU appliances. Temporary 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 DH was operating at the same time.
- DHU air re-evaporated water off of warm central coil when AC was cycled off. Temporary steady-state testing just after the central system cycled off, with the DHU operating 28 continuous minutes after, measured a total 1.5 lbs of water re-evaporated off of the central cooling coil (rate of 3.2 lb/h back into condition space).
- During one 15 minute period observation of uninterrupted monitoring, the moisture pulled out of the room air by the DHU was at about the same rate that was being re-evaporated off of the central cooling coil while the AC was cycled off. In this instance the DHU coil rate of latent removal was -1.8 lb/h and the latent heat due to evaporation from the central cooling coil was +1.9 lb/h into the space while the DH was operating steady and the central cooling system had remained naturally cycled off 1.25 hours prior during very low cooling load period in the early morning.

In regards to the DHU performance, under short-term steady-state testing the best latent performance and lowest measured electric power occurred when the DH was ducted to/from the central supply.

- Compared to DH from/to conditioned space, DH from central supply had a 5% improvement in latent (Btu/h) performance, decrease of 15% electric power, and latent efficiency (pints/kWH) improved 22%.

The predicted annual estimated central cooling and DHU energy is lowest for the DHU ducted from/to central supply, but only by about 1% less on average than the DHU ducted to room. The DHU ducted from/to central return did show measureable significant increase of 12% more energy than compared to DHU to room.

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. Excessive static pressure may adversely impact the DHU performance. This project did not evaluate impact of static and airflow rate upon the DHU. Tested DHU duct designs were in compliance with manufacturer recommendations.

A simulation-based effort was performed to determine potential heat impacts associated with DHU and DHU locations. The three different duct location heat gains/losses evaluated were ducts in:

1. Conditioned space
2. Attic
3. Garage

Based upon the lab experiments and simulation work, the following recommendations are made:

- The code should not permit DHUs to be ducted from/to central air returns upstream of the cooling coil due to increased energy use and latent performance degradation of the central AC.
- The performance path should have an unducted stand-alone dehumidifier as its base case. The proposed home should be as installed, including any effect of ductwork in unconditioned space.
- The energy code should allow a ducted dehumidifier in any space in the house (note that condensate drainage requirements may be a challenge in some locations).
- DHU ducts should be insulated to at least R-6 and the dehumidifier cabinet should be insulated to at least R-2 if located outside of conditioned space.

In addition to these changes, we still recommend changes made previously that can be found in Appendix A which includes a summary of the previous and new recommendations from this report.

The state budget period required the project to end before we have peak summer weather. FSEC will continue to collect some data during summer at its own expense. FSEC will also take the steady-state model and input it in EnergyGauge and run annual simulations. These efforts will be completed in time to submit any new developments and recommendations in time for consideration before the next code cycle.

Acknowledgments

The authors would like to thank Mr. Andrew Ask, P.E., for his assistance in developing the scope of work and for helping to secure the donation of the Therma-Stor Ultra-Aire dehumidifier. We thank Therma-Stor for donating the new DHU that was used in testing. Thanks also to the Florida Building Commission and Mo Madani of the Florida Department of Business and Professional Regulation Office of Codes and Standards for supporting this work.

References

ANSI/ASHRAE Standard 62.2-2013 (2013).*Thermal Environmental Conditions for Human Occupancy*. Accessed May 2018:

ANSI/ASHRAE Standard 55-2013 (2013).*Thermal Environmental Conditions for Human Occupancy*. Accessed May 2018: https://www.ashrae.org/resources--publications/bookstore/standard-55

"EnergyGauge USA: Code Compliance and Home Energy Rating Software." (1996-2018). Cocoa, FL: Florida Solar Energy Center. http://www.energygauge.com/usares/.

Henderson H.; Rudd, A. (2014). "Energy Efficiency and Cost Assessment of Humidity Control Options for Residential Buildings." *ASHRAE Transactions*, (120) Part 1. NY-14-013 (RP-1449).

Hendron, R.; Engebrecht, C. (2010). *Building America House Simulation Protocols*. NREL/TP-550-49426. Golden, CO: National Renewable Energy Laboratory. Accessed June 2014: http://www.nrel.gov/docs/fy11osti/49246.pdf

Henderson, H.I. 1990. "An Experimental Investigation of the Effects of Wet and Dry Coil Conditions on Cyclic Performance in the SEER Procedure." Proceedings of USNC/IIR Refrigeration Conference at Purdue University, July 1990, West Lafayette, IN.

Martin, E., C. Withers, J. McIlvaine, D. Chasar, and D. Beal. (2018). "Evaluating Moisture Control of Variable-Capacity Heat Pumps in Mechanically Ventilated, Low-Load Homes in Climate Zone 2A". Cocoa, FL; Building America Partnership for Improved Residential Construction (BA-PIRC). DOE/EE-1702. Accessed May 2018: https://doi.org/10.2172/1421385

Mattison, L.; Korn, D. (2012). "Dehumidifiers: A Major Consumer of Residential Electricity.*"Proceedings of the 2012 ACEEE Summer Study on Energy Efficiency in Buildings.* The Cadmus Group, Inc. Accessed May 2018: www.aceee.org/files/proceedings/2012/data/papers/0193-000291.pdf.

Rudd, A., J. Lstiburek, and K. Ueno. (2005). "Residential Dehumidification Systems Research for Hot-Humid Climates." Golden, CO. National Renewable Energy Laboratory. NREL/SR-550-36643.

Rudd, A.; Lstiburek, J.; Eng, P.; Ueuno, K. (2002). *Residential Dehumidification Systems Research for Hot-Humid Climates*. Building Science Corporation: Westford, MA. Accessed June 2014: http://www.buildingscience.com/documents/bareports/ba-0219-residential-dehumidifications-systems-research-hot-humid-climates

Shirey, D., Henderson, H., and Raustad, R. (2006). Understanding the Dehumidification Performance of Air-Conditioning Equipment at Part-Load Conditions. Cocoa Lake, FL: Florida Solar Energy Center, FSEC-CR-1537-05: www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-1537-05.pdf.

US Department of Energy. 2018. Input Output Reference Documentation – EnergyPlus, Version 8.9. Washington, DC: U. S. Department of Energy.

Vieira, R. and D. Beal. (2017). "Residential Performance Code Methodology for Crediting Dehumidification and Smart Vent Applications Final Report" FSEC-CR-2067-17 Cocoa, FL: Florida Solar Energy Center.

Withers, C. (2018). "Optimizing Energy Efficiency and Improved Dehumidification Performance of Variable Capacity Air Conditioning Systems." *Proceedings of the 2018 ACEEE Summer Study on Energy Efficiency in Buildings.* Available after August 18, 2018, currently undergoing peer review.

Withers, C. (2016). "Energy-Efficient Management of Mechanical Ventilation and Relative Humidity in Hot-Humid Climates" National Renewable Energy Laboratory. DOE/GO-102016-4766. Accessed May 2018: http://fsec.ucf.edu/en/publications/pdf/fsec-cr-2049-16.pdf

Withers, C. and J. Sonne. (2014). "Assessment of Energy Efficient Methods of Indoor Humidity Control for Florida Building Commission Research" FSEC-CR-1976-14 Cocoa, FL: Florida Solar Energy Center.

Withers, C.; Cummings, J.; Nelson, J.; Vieira, R. (2012). *A Comparison of Homes Built to the 2009 and 1984 Florida Energy Codes*. FSEC-CR-1934-12. Cocoa, FL: Florida Solar Energy Center. Accessed May 2018: http://www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-1934-12.pdf

Appendix A

Previous Code-Related Questions and Recommendations Regarding Supplemental Dehumidification **quoted below from Vieira and Beal 2017**

Discussion

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

- *What kind of energy savings or penalty occurs from advanced central cooling systems that are designed to dehumidify more effectively?*
- *Should dehumidifier efficiency be evaluated at conditions other than 80F, 60%, and how should simulation models handle standby energy use of dehumidifiers? What are realistic static pressures of whole house dehumidifiers that are installed in duct systems? Should the dehumidifier appliance rating change to accommodate?*
- *Where is it best to bring in a supply outside air intake if the cooling system is off?? What if the home has a whole-house dehumidifier?*
- *Where should a whole house dehumidifier be located –stand alone, on return side of central system, on supply side of central system? If in cooling mode it would make more sense on the supply side but will that hurt the life of the unit? Should the code mandate rules or allow penalties/credits based on location in performance code while limiting options in prescriptive?*
- *Where should a dehumidistat be located?*
- *What type of savings will other smart vent options produce? Ones based on dew point as well as dry bulb temperature?*
- *What are interior latent generation rates and how can we improve modeling to reflect real world?*
- *What kind of accuracy increase would be achieved by requiring HVAC system mapping (performance at many test points) as inputs to code performance software tools?*
- *How does moisture and smart ventilation impact results for very low load homes that have larger latent/total cooling ratios?*
- *What rules should be applied to smart ventilation controls? Does the relative exposure have to be 1.0 or less as recommended in ASHRAE 62.2-2016? Should there be a requirement for minimum airflow per day or per week? Should average annual airflow or relative exposure have to be the same as the baseline? Should fan ventilation rates be the same for unbalanced and balanced flow rates as currently in ASHRAE 62.2-2016 even though balanced flow is estimated to exchange about 30% more total air?*

Dehumidifier Recommendations from Vieira and Beal 2017 with New Recommendations

This section shows previous dehumidifier-related recommendations in black underlined text and updates based upon this final report in red underlined text.

Recommended changes to the Energy Conservation Code should be made to indicate the minimum requirements of any dehumidifier installed:

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

1. *Shall be sized in accordance with ACCA Manual S.*
2. *Shall have a minimum rated efficiency greater than 1.7 Liters/ kWh if the total dehumidifier capacity for the house is less than 75 pints/day and greater than 2.38 Liters/kWh if the total dehumidifier capacity for the house is greater than or equal to 75 pints/day.*
3. *Shall operate without requiring operation of the cooling system air handler fan.*
4. *If connected into the return side of the cooling system, shall include a backdraft damper installed in the return air duct between the inlet and outlet of the dehumidifier.*
5. *Shall be controlled by a dehumidistat that is installed in a location where it is exposed to mixed house air and does not receive undue direct influence from mechanical ventilation air or supply air from the home's cooling or heating system(s).*
6. Shall not be ducted to or from a central ducted cooling system on the return duct side upstream from the central cooling evaporator coil.
7. Ductwork associated with the dehumidifier located in non-conditioned space shall be insulated to a minimum of R-6.
8. Any dehumidifier unit located in non-conditioned space shall be insulated to R2.

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

Building Component	Standard Reference Design	Proposed Design
Dehumidification Systems	None, except where dehumidification equipment is specified by the proposed design	As proposed
	Fuel Type: Electric	As proposed
	Capacity: Sufficient to maintain humidity at setpoint all hours	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.	As proposed
	Location: In conditioned space	As proposed
	Dehumidifier Ducts: None	As proposed

Dehumidistat	None, except where dehumidification equipment is specified by the proposed design Setpoint turn on = 60% relative humidity Setpoint turn off= 55% relative humidity	Same as standard reference

Appendix B

Supplemental Data and Analysis from Lab Testing

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

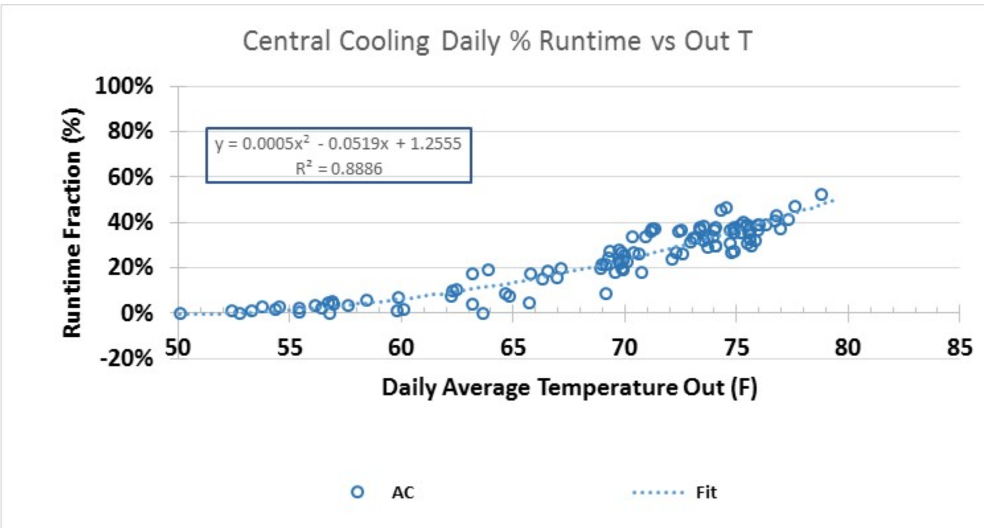

Figure B-1. Test lab central cooling daily runtime % versus daily average outdoor Temperature.

DHU runtime is impacted by both AC runtime and latent load. Figure B-2 demonstrates the measured decrease in DHU runtime as central cooling increased during lab testing with 60lb/day latent load. The 60lb/day latent load was shown here since it was the largest tested latent rate with the greatest tested influence upon DHU operation.

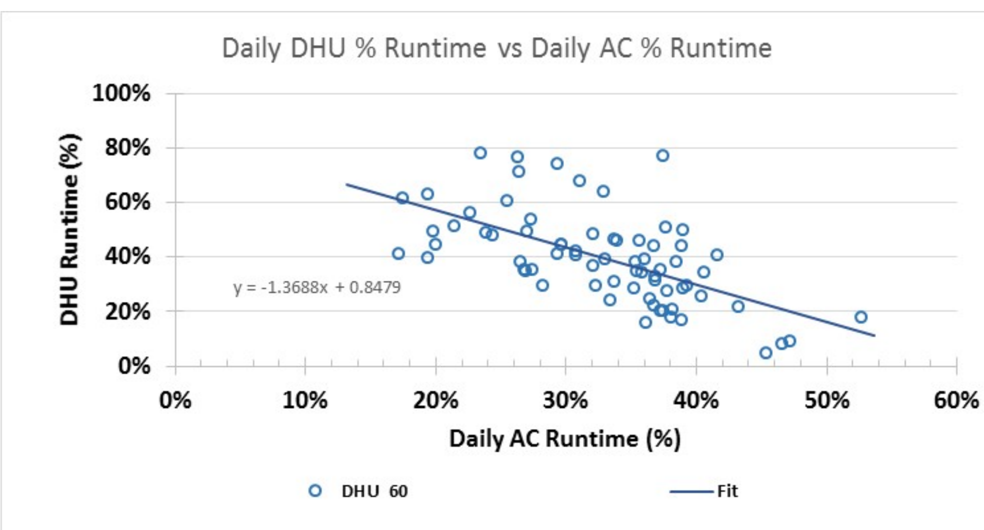

Figure B-2. Test lab daily DHU runtime % versus the daily central cooling runtime % with 60 lb/day latent load.

Figure B-3 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 all DHU configurations and the three different latent load rates of 15 lb/day, 30 lb/day, and 60lb/day. It is expected that the central AC condensate would 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.

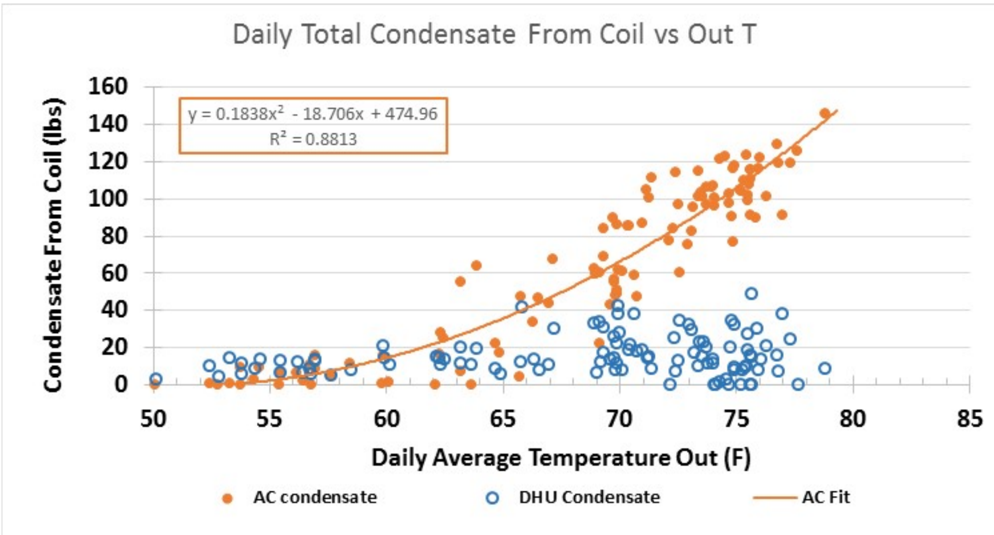

Figure B-3. Daily total AC condensate and DHU condensate versus outdoor temperature for all three latent loads and all DHU configurations.

Figure B-4 is uses the same data as in Figure B-3 except the latent load rate is identified. No overly 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 70F 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 63F-79F.

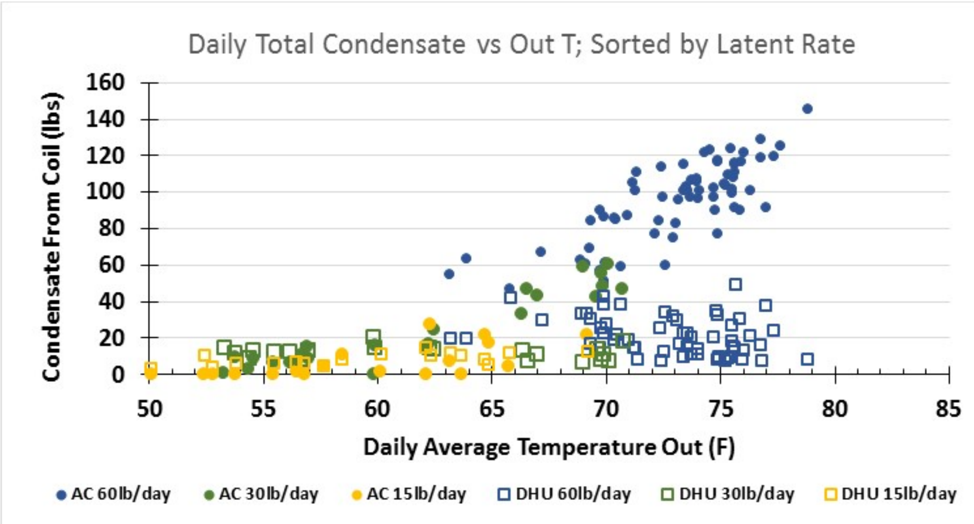

Figure B-4. Daily total AC condensate and DHU condensate versus outdoor temperature identified by each latent load rate.

Somewhat similar to Figure B-2, Figure B-5 shows the DHU condensate versus central cooling runtime for only the 60 lb/hr latent load rate. This shows a moderate correlation.

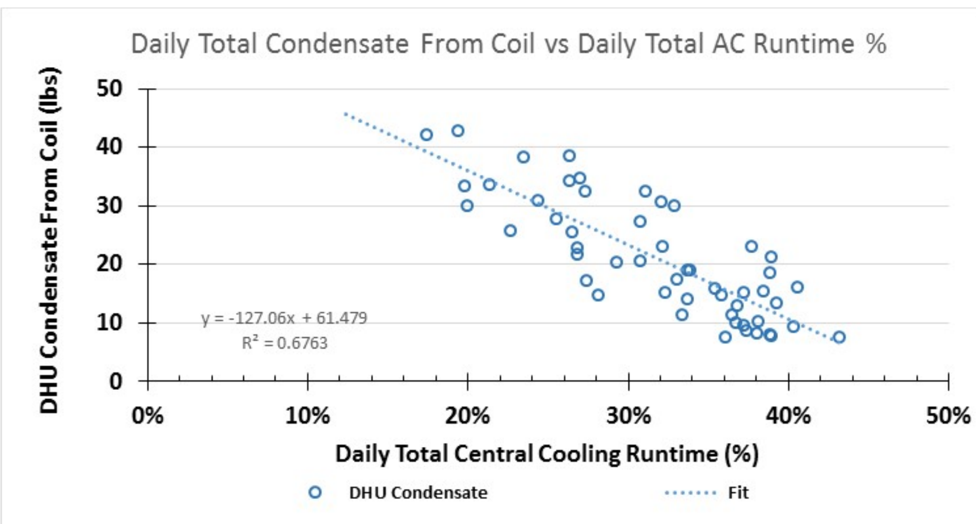

Figure B-5. Daily total DHU condensate versus daily total cooling runtime at the 60 lb/day latent load.

Figure B-6 shows the daily total DHU condensate versus daily total DHU runtime at the 60 lb/day latent load. It is expected to see much better correlation here since the longer the DHU runs, the more condensate that will be removed. Some of the variance can be due to the different DHU duct configurations.

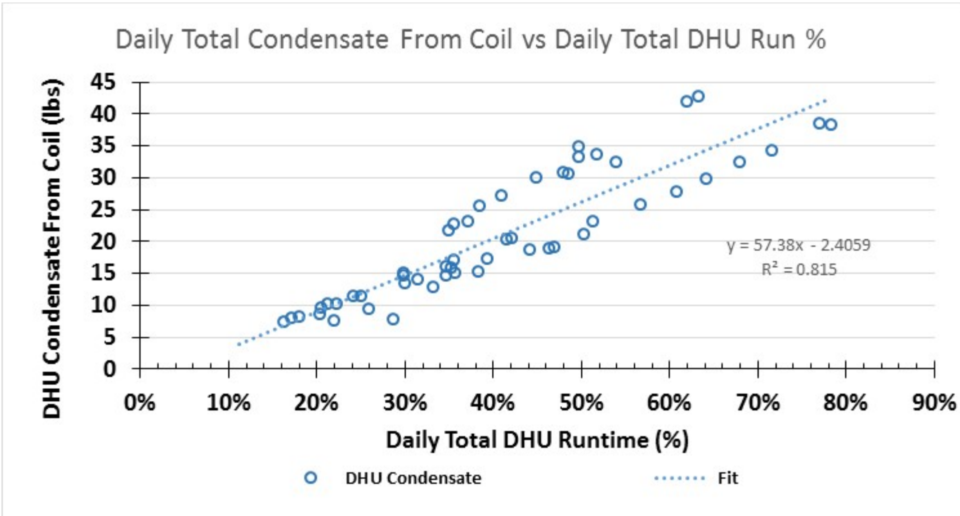

Figure B-6. Daily total DHU condensate versus daily total DHU runtime at the 60 lb/day latent load.