# Energy and Latent Performance Impacts from Four Different Common Ducted Dehumidifier Configurations

**Charles R. Withers, Jr.**


## ABSTRACT

*Dehumidifiers (DHUs) are the second most-selected equipment, after air conditioning (AC), used to manage indoor relative humidity (RH) in homes. They can offer the lowest first-cost, are well-established in the market, and are often easier to install than other supplemental dehumidification alternatives. However, DHUs have the potential to use significant amounts energy and may impact the performance of the central ducted cooling system under certain conditions. Dehumidifiers may be designed to be ducted or un-ducted. Dehumidifiers with ducts are sometimes referred to as whole-house or ducted dehumidifiers.*

*Manufacturer manuals offer several different options for ducting DHUs, but they do not provide adequate information about potential performance impacts. Also sorely missing are expanded DHU performance metrics to help professionals and consumers determine the appropriate DHU capacity and to help predict operational efficiency for specific realistic applications. The Florida Building Commission initiated a research project to determine if some common ducted DHU configurations had significant energy and moisture impacts and whether any configurations should be not allowed in Florida Building Code. The research evaluated measured energy use and latent heat removal rates of AC and DHU for four different common ducted DHU duct configurations. Testing occurred during variable weather and interior latent loads common in a warm moist climate. A highly-instrumented building lab was used to evaluate AC and DHU performance based upon how a DHU was ducted to a central AC ductwork and compared these to a DHU ducted to and from an open central room.*

*There was less than 1% difference in annual space conditioning energy (DHU + AC) among two different methods of DHU ducted to the central supply duct and the DHU ducted directly to indoors. However, steady-state and longer-term test findings showed that a DHU ducted to and from a central cooling system return upstream of the central cooling evaporator coil resulted in the annual predicted space conditioning energy use of 308 kWh/year (4%) more than the DHU ducted directly to indoors. It also resulted in decreased central AC latent performance by 28% when the DHU and AC ran simultaneously. This DHU duct configuration further degraded dehumidification performance by causing moisture to be evaporated off of the central cooling coil at a rate as high as 2-3 pounds of water per hour when the AC was cycled off and DHU operated. This paper discusses the experimental method, results and recommended practice of ducting DHU for optimum performance.*


## INTRODUCTION

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

Charles Withers, Jr. is a Sr. Research Analyst at the Florida Solar Energy Center, a research institute of the University of Central Florida, Cocoa, Florida.

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. Supplemental dehumidification can require a significant amount of energy use, so design should be carefully considered. In some low load homes with very efficient air conditioning (AC), dehumidifier units (DHU) may use nearly as much annual energy as central cooling (Withers 2018), (Mattison and Korn 2012). Lab controlled studies of DHU performance of several different DHU found that DHU power and efficiency varied significantly as a function of varying entering air conditions Winkler et al. (2011), DOE (2014).

Ducted DHU are becoming more popular over ductless stand-alone DHU as they enable better distribution of dehumidified air. They can also be installed in unconditioned space which does not take up conditioned space and reduces operational noise in occupied space. A review of different ducted DHU manufacturer installation manuals showed several different options how to connect the DHU ducts to/from central ducted AC and to/from indoor spaces. There was no guidance in manuals on how different ducting may impact AC or DHU performance other than some indicating DHU operational static pressure limits. Until 2019, DHU appliances were only required to rate capacity and efficiency at a single entering air condition of 26.7°C (80°F) and 60% relative humidity (RH). The DOE has established an updated rule that now requires ducted DHU to be tested at 18.3°C (65°F) and 60% RH DOE (2014). While this new test condition is closer to real applications than the previous one, it is still inadequate to help contractors or consumers to choose the suitable capacity and expected performance given the potential variety of entering air conditions. DHU capacity guidance is typically based upon the area of the space to be dehumidified.

## TEST DESCRIPTION

### Test Rationale

The primary research question for this project was to determine the AC and DHU energy and moisture removal performance of some common DHU duct configurations. There are at least seven potential duct configurations, of which research funding could accommodate four tests. The four DHU duct configurations tested were: Test 1 DHU air ducted from/to the central main body of building; Test 2 DHU air ducted from/to main return duct of central cooling (AC); Test 3 DHU air ducted from/to main supply duct of AC. Test 4 is similar to Test 3 with the DHU supply air ducted into the central supply duct, except the return air to the DHU comes directly from the indoor central room. Figure 1 shows an illustration of the four DHU tests. Test 1 was assumed likely to have minimal detrimental performance impacts and was used as a baseline of comparison. External static pressure (ESP) limits of DHU must be considered when choosing duct configuration. DHU should not be ducted to central supply and/or return ducts if the (ESP) acting on DHU will exceed DHU manufacturer limits. Tests 2 and 3 provide a method for DHU air to be distributed through central system without significant DHU ESP. A gravity damper should also be used as shown in Tests 2 and 3 in Figure 1 to avoid DHU supply air from being short-circuited backwards through the DHU return whenever the DHU is on and central heat and cooling system is off. It is also important that the central system duct design can accommodate an appropriately sized gravity damper to avoid detrimental static or air flow issues. Test 4 was requested by the Florida Energy Technical Advisory Committee to be evaluated as a common method used in south Florida.

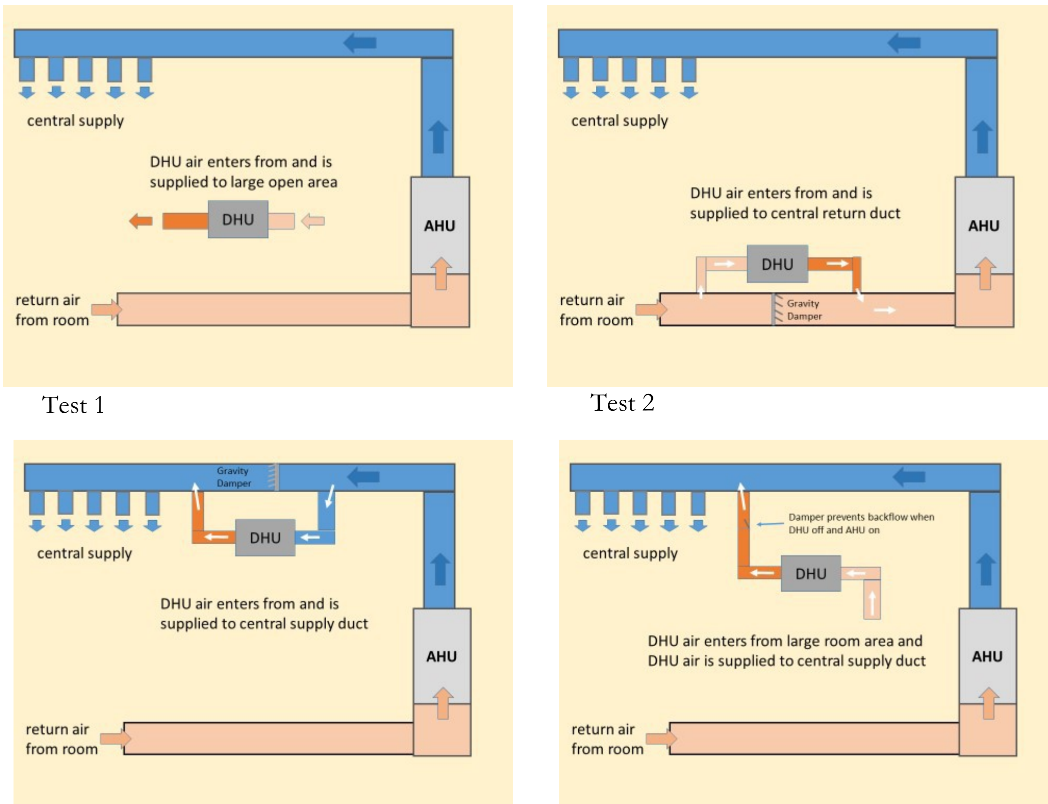
### Test Building

Testing was conducted using an unoccupied single-story 2,000 $ft^2$ research lab building with one large central room, four rooms, storage closet and bathroom. Testing occurred in east central Florida. The exterior walls were concrete masonry block 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).

The central ducted system was a SEER13 fixed-capacity split-DX 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, which exactly matched the Manual J8 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 had a rated efficiency of 2.4 liters/kWh and rated moisture removal of 70 pints per day at 26.7°C (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 about twice the efficiency of 70 pint/day DHU at the lower end of efficiency. This means that the DHU energy use in the lab building 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 set point was at 50% RH. This activated the DHU at 50% RH until RH reached about 45% RH, at which point DHU cycled off. The DHU and AC systems were not interlocked and cycled independent from each other. DHU ESP was well below manufacturer stated limit of 125 Pa (0.500 in wc) with DHU airflow within 4% of each other during all test configurations.




Test 3 Test 4

Figure 1. Illustrations of four different DHU duct test configurations.

## Space Conditioning Loads

Interior-generated sensible and latent loads were automated on a daily schedule. Interior sensible loads were at an average rate of about 1,230 W (4,200 Btu/h). Internal moisture was generated at three different rates of 6.8 kg, 13.6 kg, and 27.2 kg (15, 30, and 60 pounds) of water per day using a commercial-grade humidifier distributed throughout different periods of each day. Different rates were used to approximate variable latent loads associated with warmer and cooler weather periods. Tests were intentionally conducted with no mechanical ventilation particularly since outdoor latent load varies significantly from fall through late spring when the tests had to be completed. The controlled interior latent generation was a means to provide varying levels of latent load across a wide range of sensible cooling load similarly for all tests. Based upon the measured building tightness for a 3 bedroom 186 $m^2$ (2,000 $ft^2$) home, ASHRAE 62.2-2013 would call for a total ventilation rate of 42.5 L/s (90 cfm), of which 33.0 L/s (70 cfm) would come from mechanical ventilation and 9.4 L/s (20 cfm) from infiltration. Moisture was generated internally at 27.2 kg (60 lb) per day as long as outdoor temperatures averaged around 20°C (68°F) or greater. This moisture rate represented 21.8 kg (48 lb) per day that would have come in from mechanical ventilation at 21.1°C (70°F) dp and another 5.4 kg (12 lb) per day internally generated by occupant activities.

Because 27.2 kg (60 lb) of latent is abnormally high during cooler weather, internal latent generation was reduced during cooler weather periods. Internal moisture was generated at 13.6 kg (30 lb) per day when daily average outdoor temperatures were between about 15.6°C- 22.2°C (60°F-72°F). Internal moisture was generated at a rate of 6.8 kg (15 lb) per day generally when daily average outdoor temperatures were about 18.3°C (65°F) or colder.

### Monitoring Sensors

All sensors for this project were verified to be functioning within manufacturer stated accuracy. Temperature and relative humidity (RH) sensors measured indoor, attic, and outdoor conditions as well as entering and leaving air conditions. Power meters measured internal loads and space conditioning energy. Air flow stations measured AHU and DHU air flow rates throughout all testing. Condensate was measured using calibrated tipping buckets.

## RESULTS

Performance was evaluated based on short steady-state tests as well as long term testing that allowed equipment to cycle on and off as interior set point and conditions dictated. Long term testing rotated through each test approximately 2-3 weeks over several months. This permitted all tests to be evaluated over variable weather conditions.

### Steady-State Test Results

Test 2 was the only configuration that showed detrimental impact upon the central cooling latent performance when the DHU and AC operated simultaneously. This is attributed to hot dry air leaving the DHU that entered directly upstream of the central cooling coil. 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 when DHU and AC operated simultaneously. Outdoor conditions at the condensing unit averaged 30°C (86°F) during this testing.

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

| Test 2 Condition | Entering T (°C) (°F) | Entering RH (%) | Leaving T (°C) (°F) | Leaving RH (%) | Airflow (L/s) (cfm) | Total (W) (Btu/h) | Sensible (W) (Btu/h) | Latent (W) (Btu/h) | SHR |
|---|---|---|---|---|---|---|---|---|---|
| **AC On and DHU Off** | 24.3<br>75.8 | 51.7 | 13.8<br>56.9 | 81.7 | 447<br>947 | -8,194<br>-27,983 | -5,744<br>-19,616 | -2,450<br>-8,366 | 0.701 |
| **AC On, DHU On; DHU ducted to AC return duct** | 26.3<br>79.3 | 42.1 | 14.3<br>57.7 | 75.9 | 447<br>947 | -8,329<br>-28,446 | -6,563<br>-22,415 | -1,766<br>-6,031 | 0.788 |
| **% difference between when AC & DHU on same time and when only AC on** | | | | | | 1.7% | 14.3% | -27.9% | 12.4% |

This shows that although the sensible cooling capacity increased 14%, the latent capacity decreased 28%. The sensible heat ratio (SHR) increased 12%. When space dehumidification is needed, this is the opposite impact that should occur.

Test 2 exhibited one more negative impact upon space dehumidification. This occurred when moisture subsequently remaining on the cooling coil was re-evaporated by 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 -0.82 kg/h (-1.8 lb/h) and the measured latent heat of evaporation from the central cooling coil was 0.86 kg/h (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 0.68 kg (1.5 lb) of moisture evaporated from the central cooling coil was measured. This moisture then moves down the central supply, eventually making its way back into the conditioned space. This 28 minute test can be seen in Figure 2. Negative values indicate that sensible or latent heat was removed from the airstream across the central AC coil. Positive values indicate that latent heat vapor was added to the air after the AC coil. The positive latent rate (Figure 2 blue line) indicates moisture evaporated from the AC coil and went into the central supply air duct. The evaporation of water from the AC coil provided a small amount of sensible coiling (orange line).

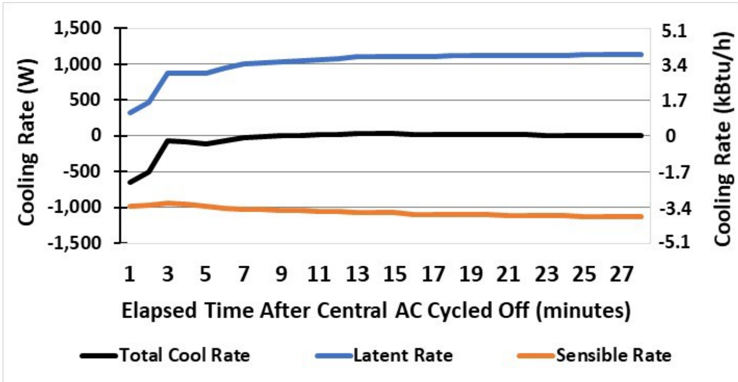



Figure 2. Test 2 cooling rates across the central AC coil just after the AC cycled off and DHU remained on.

## DHU Performance

Dehumidifier energy and moisture performance demonstrated significant variations depending upon entering air conditions. DHU manufacturer data showed that a 33% drop in latent capacity could be expected from changing entering air from rated conditions at 26.7°C (80°F) and 60% RH to cooler and drier air at 21.1°C (70°F) and 60% RH). The stated latent efficiency dropped by 20% from 2.4 L/kWh to 1.9 L/kWh. Most manufacturers only provide performance data at the single rated condition.

As with most real applications, DHU entering air test conditions were different than rated conditions. Short tests were run to evaluate the DHU performance at three very different sets of entering air conditions: Near-rated condition (warm and moist), air conditioned room (cool and dry), and air from central cold air supply (cold and dry). The test results are summarized in Table 2 and show performance trends supported by the manufacturer data.

DHU capacity is stated as pints liquid moisture removed per day at rated conditions. The DHU tested was rated at 33.1 L/day (70 pt/day). This rated latent capacity can also be stated as 897 W (3,063 Btu/h). The test results showed a 21%-25% reduction in DHU latent removal for configurations 1-4 when compared to rated conditions. The latent reduction to Tests 1-4 indicated even greater diminished latent capacity of 30%-34% when compared to actual measured warm/moist condition (b near-rated). This shows that the diminished latent capacity under the most likely real operating conditions should be taken into account when selecting DHU capacity.

**Table 2. DHU Steady-State Performance Comparisons for Three Entering Conditions**

| DHU Entering Air Test Condition | Entering T (°C) (°F) | Entering RH (%) | Leaving T (°C) (°F) | Leaving RH (%) | Airflow (L/s) (cfm) | Total Capacity (W) (Btu/h) | Sensible Capacity (W) (Btu/h) | Latent Capacity (W) (Btu/h) | DHU Power (W) (Btu/h) | Latent Efficiency L/kWh |
|---|---|---|---|---|---|---|---|---|---|---|
| a) Manuf. Rated (warm/moist) | 26.7<br>80.0 | 60 | NA | NA | 170 | NA | NA | -897<br>-3,063 | 580<br>1,981 | 5.1 |
| b) Near-rated condition (warm/moist) | 28.2<br>82.8 | 57.3 | 43.7<br>110.6 | 17.0 | 165 | 464<br>1,583 | 1,476<br>5,041 | -1,012<br>-3,458 | 581<br>1,984 | 5.5 |
| c) Tests 1,2,4 (cool/dry) | 24.3<br>75.7 | 49.1 | 35.5<br>96.0 | 18.0 | 165 | 402<br>1,372 | 1,072<br>3,661 | -670<br>-2,289 | 516<br>1,762 | 4.1 |
| d) Test 3 (cold/dry) | 12.7<br>54.8 | 71.9 | 24.4<br>75.9 | 19.4 | 171 | 443<br>1,512 | 1,149<br>3,923 | -706<br>-2,411 | 438<br>1,496 | 5.0 |
| % difference from b) (near-rated condition) to a) (manufacturer rating) | | | | | | | | 12.9% | 0.2% | 7.8% |
| % difference from c) (cold/dry) to a) (manufacturer rating) | | | | | | | | -25.3% | -11.0% | -19.6% |
| % difference from d) (cold/dry) to a) (manufacturer rating) | | | | | | | | -21.3% | -24.5% | -2.0% |

Rated conditions would rarely occur in most occupied homes, except when the DHU is used as a ventilating DHU. Rated conditions had the greatest power consumption, but best overall performance as indicated by latent capacity and efficiency. The near-rated conditions were slightly warmer and drier than actual rated conditions, and had measured latent efficiency within 8% of the rated efficiency. Cold dry entering air from central supply had the second-best performance and lowest DHU power use. The real-time data observed during several months of testing showed variation in DHU power that followed the same trend in Table 2, where DHU power increases in direct proportion with an increase in entering dewpoint temperature. DHU measured electric power varied from down around 440 W with cold dry air up to 630 W with very warm moist air during normal monitoring including hundreds of cycles not part of short-term steady-state testing.

One other DHU performance study found similar results as this study. In an independent lab test of six different dehumidifiers, Winkler et al. (2011) found that all DHU met or exceeded manufacturer performance at rated conditions, however DHU performance varied significantly when tested with entering air at different conditions other than rated conditions. DHU latent performance and power use also dropped as the entering air dewpoint temperature dropped as found by this research.

## Space Conditioning Energy

The total space conditioning energy use impacts of the four different test configurations was evaluated under normal operation that included equipment cycling. 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 also known as delta temperature (dT).

Figure 3 shows an example of daily total AC cooling + DHU energy versus dT for all tests at the three different latent loads. The DHU represents the majority of energy shown on Figure 3 for dT < -5.6°C (-10°F). Least-squares regression analysis of space conditioning energy versus dT was used to develop an equation for each test configuration. The sets of equations were used to create a single best-fit equation representing all test configurations over the range of interior latent loads. The results are applicable during cool to hot weather. The final best-fit results for four tests are shown in Figure 4. The outdoor air temperature of data shown in Figures 3-4 can be estimated by adding the x-axis dT°C value to 24.4°C indoor temperature (dT°F value + 76°F).

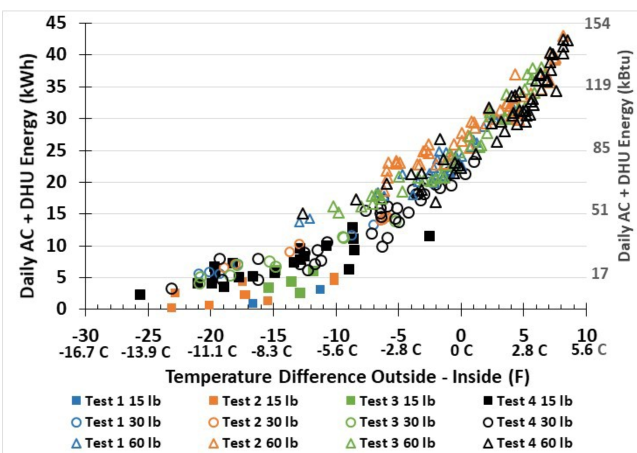



Figure 3. Daily conditioning electric energy use plotted against dT for all tests at three different latent loads.

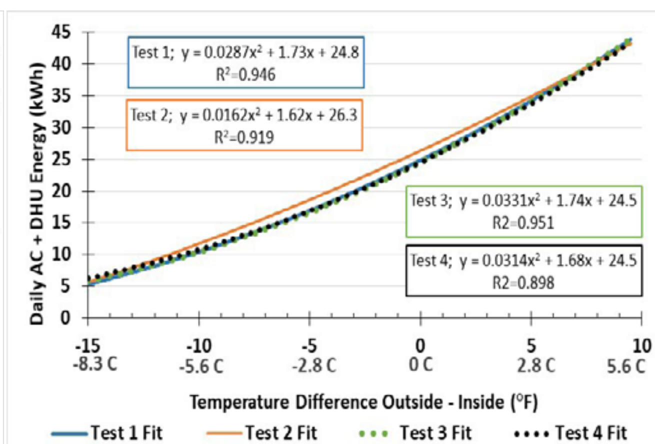



Figure 4. Final best-fit models of daily conditioning electric energy use versus dT used to predict annual energy.

The long term energy testing results showed no significant energy difference between tests at the lower and upper ranges of dT. There was only AC operation at upper dT, and mostly DHU and very little AC operation at low dT. Only Test 2 showed higher energy use for dT from about -5.6°C to 2.8°C (-10°F to 5°F). This is the range in which the detrimental dehumidification performance shown earlier was more likely to occur.

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 as well as Houston, Texas. The predictive equations are most-applicable for locations similar to IECC hot-humid climate zones of 1A and 2A. An indoor cooling set point of 23.9°C (75°F) was subtracted from the TMY3 average daily outdoor temperature to calculate a daily temperature difference for each day of the year. Due to inadequate space heating weather, heating season energy was not evaluated. While this does not represent all conditioning energy for a full year, it still enabled a relative comparison between different tests and represents a majority of annual space conditioning used in hot humid climates.

Using the equations shown in Figure 4 and calculated dT, space conditioning energy was calculated for each day of the year for dT≥ -8.3°C (-15°F). This limitation eliminated energy predictions during days when heating would be likely.

**Table 3. Predicted Annual Central Cooling and DHU Energy Use for Tests 1-4 with Comparisons to Test 1 for Four Cities in Hot Humid Climate Zone**

| | Test 1 | Test 2 | Test 3 | Test 4 |
|---|---|---|---|---|
| | | **Miami, FL** | | |
| Annual kWh | 10,245 | 10,534 | 10,192 | 10,133 |
| Annual MBtu | 34.96 | 35.94 | 34.56 | 34.58 |
| Delta kWh from Test 1 | 0 | 290 | -53 | -112 |
| Delta % from Test 1 | 0 | 2.8% | -0.5% | -1.1% |
| | | **Orlando, FL** | | |
| Annual kWh | 7,661 | 8,007 | 7,610 | 7,609 |
| Annual MBtu | 26.14 | 27.32 | 25.97 | 25.96 |
| Delta kWh from Test 1 | 0 | 346 | -51 | -52 |
| Delta % from Test 1 | 0 | 4.5% | -0.7% | -0.7% |
| | | *Jacksonville, FL* | | |
| Annual kWh | 6,339 | 6,626 | 6,303 | 6,308 |
| Annual MBtu | 21.63 | 22.61 | 21.51 | 21.52 |
| Delta kWh from Test 1 | 0 | 287 | -36 | -30 |
| Delta % from Test 1 | 0 | 4.5% | -0.6% | -0.5% |
| | | *Houston, TX* | | |
| Annual kWh | 7145 | 7357 | 7120 | 7093 |
| Annual MBtu | 24.38 | 25.10 | 24.29 | 24.20 |
| Delta kWh from Test 1 | 0 | 211 | -25 | -52 |
| Delta % from Test 1 | 0 | 3.0% | -0.4% | -0.7% |

## CONCLUSION

Experiments were conducted to determine the AC and DHU energy and moisture removal performance of four common DHU duct configurations. Ducting DHU supply air into a central return upstream of the evaporator coil (Test 2) had the poorest performance. This was the only test configuration that showed significant reduction in dehumidification due to detrimental impacts upon the central AC performance. It was also the only configuration that used more space conditioning energy compared to three other configurations. When the DHU and AC ran simultaneously, the AC latent capacity decreased 28% due to warm drier air entering the evaporator coil. The AC sensible heat ratio (SHR) increased 12%. When space dehumidification is needed, this is the opposite impact that should occur. Furthermore, the DHU evaporated residual moisture from the AC coil into the central duct system, then into the conditioned space whenever the DHU operated following a cooling cycle. One monitoring period measured a total accumulative mass of 0.68 kg (1.5 lb) of moisture evaporated from the central cooling coil while the DHU operated for 28 minutes after the central cooling had cycled off. It was also observed that during some periods, the rate of evaporation (humidification) occurring off the AC coil was nearly the same as the DHU latent removal rate of moisture from the indoor air at 0.82 kg/h (1.8 lb/h) thereby negating dehumidification by the DHU. While the results here are based solely upon one specific AC and DHU, one could expect poor performance to apply using other AC and DHU manufacturer models configured as Test 2. Based upon results and basic performance principles of refrigerant-based systems, Test 2 is not a recommended practice. One added caution is offered here regarding ducting DHU entering air from central return and supplying

DHU leaving air into central AC supply. Although this specific test was not evaluated, this DHU duct configuration may decrease DHU airflow below acceptable limits due to high ESP acting upon the DHU.

Latent removal capacity of DHU decreased as conditions became cooler and drier than rated conditions. The test results showed a 21%-25% reduction in DHU latent removal for test configurations 1-4 when compared to rated conditions. The high potential of diminished latent capacity under the most likely real operating conditions should be taken into account when selecting DHU capacity. The latest DOE Whole-house DHU test standard should be expanded to be able to determine DHU latent performance at two or more realistic entering air conditions instead of just one. This would enable more informed decisions on determining DHU capacity for anticipated specific DHU entering air conditions.

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