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Measured Space-Conditioning Energy and Humidity in a Mechanically-Ventilated House Lab with Fixed and Variable-Capacity Cooling Systems Located in a Hot and Humid Climate

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ABSTRACT

Residential whole-house mechanical ventilation has become more important as air tightness requirements have increased under certain building programs and codes. Generally, homes mechanically ventilated during warm and humid weather will have elevated indoor relative humidity (RH) during low cooling load periods. Supplemental dehumidification has been relied upon to control RH. Herein lies a challenge to balance acceptable RH with minimal energy use.

Three specific types of space cooling equipment configurations were tested in a controlled research lab home. The central Florida 3 bedroom, 2-bathroom lab home was furnished, had automated internal sensible and latent loads, and was ventilated in accordance with ASHRAE 62.2-2013. The focus of the testing was to evaluate space cooling and dehumidifier energy use as well as the resulting indoor RH throughout the home. The three primary test configurations covered in this paper involved: 1) a central ducted fixed-capacity SEER 13 rated system 2) a central ducted variable-capacity SEER 22 rated system, and 3) a SEER 21.5 ductless variable-capacity minisplit. The minisplit was operated as the primary cooling system with central system used for cooling backup during near peak cooling load periods.

The project found that the SEER 22 central system configuration used 20% less energy than the SEER 13 central system, and the minisplit configuration used 25% less energy than the SEER 13 system under typical seasonal conditions. Limited supplemental dehumidification was needed to maintain indoor RH below 60% during some low cooling load periods. When needed, dehumidifier use was typically only 1-3 cycles per day. This paper will share greater details on the variability of indoor RH among the test configurations, factors that resulted in the very limited need for supplemental dehumidification, and recommendations to improve latent performance of variable capacity cooling systems.

INTRODUCTION

This paper focuses on investigating potential energy-efficient methods of cooling and dehumidifying a research lab home mechanically ventilated in accordance with ASHRAE 62.2-2013 (ASHRAE 2013a). Three primary cooling systems were used. The first was a minimum efficiency fixed capacity central ducted system, the second was a very high efficiency variable capacity central ducted system, and the third was a single ductless minisplit system. Maintaining good indoor relative humidity (RH) and simultaneously providing adequate mechanical ventilation can be challenging during warm and humid weather, particularly during low cooling load periods. During warm and humid

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weather, mechanical ventilation introduces moisture into a home that must be removed, otherwise the indoor RH may increase beyond acceptable levels during certain hours of the year. The fundamental problem with relying solely on central cooling systems to manage moisture during low sensible load periods is they are oversized for cooler periods of the year despite being "properly sized" for a hot design cooling day. Operation of air conditioning relies on set points that are lower than the room temperature. Lowering the cooling set point during cooler weather increases runtime, but during very low cooling load periods, the space can become overcooled and runtime is not adequate to remove much moisture from the air. This can result in cool, humid (cave-like) uncomfortable conditions. Some studies determined that supplemental dehumidification is required to meet RH control targets when ventilation is supplied at ASHRAE 62.2 levels (Rudd et al. 2013a; Rudd and Henderson 2007; Walker and Sherman 2007).

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

TEST DESCRIPTION

Testing was conducted in a 1620 ft² (150.5 m²) manufactured house lab in Cocoa, Florida with a measured house tightness of 5 air changes per hour at 0.200 in wc (50 Pa), also stated as 5 ACH50. The house was mechanically ventilated with a constant monitored air supply of 56 cubic feet per minute (26.4 L/s) delivered into the living room of the house using a dedicated fan. The tests presented here were conducted with the goal of answering two questions: 1) What are potential cooling energy savings and resulting indoor RH of operating variable capacity versus fixed (lower efficiency) systems?, and 2) Can a ductless minisplit adequately manage moisture in a mechanically ventilated home without the use of a dehumidifier? Three test configurations covered in this paper are summarized as shown in Table1. Capacity and efficiency shown in Table 1 are based on manufacturer rated values.

Table 1. Test Configurations Shown with Primary Space Conditioning Information

Test	Description	Nominal Cooling Capacity	Capacity Range	Efficiency EER (COP)	Supplemental Dehumidification
1	Central ducted, fixed capacity SEER13	23.9 kBtu/h (7.0 kW)	Single capacity	11.6 (3.4)	Dehumidifier
2	Central ducted, variable capacity SEER 22	23 kBtu/h (6.7 kW)	11.3-26.9 kBtu/h (3.3-7.9 kW)	14 (4.1)	Dehumidifier
3	Ductless MSHP, variable capacity with central ducted SEER 13	14.5 kBtu/h (4.2 kW)	3.1-18.4 kBtu/h (0.9-5.4 kW)	12 (3.5)	None

EER is the Energy Efficiency Ratio = Btu cooling output / watt-hour electrical power input determined at specific test conditions.
COP is the coefficient of performance = heat energy output / heat energy input

The first two tests are common arrangements, however the third test configuration used a ductless minisplit heat pump (MSHP). The MSHP was undersized to meet design load as primary cooling system and used the existing central ducted system as secondary backup cooling on the hottest days as well as for improved thermal and ventilation air distribution. The reason the third test was set up this way was because MSHP were known to typically be installed to supplement, not replace ducted systems in existing homes. Being ductless they are challenged to distribute conditioned air around the home. An already existing ducted system can be used to circulate air around the home on a schedule. Since the variable capacity MSHP has long run-times, supply outside ventilation air delivered near the unit return enabled it to condition air and mix it with indoor air during much of the cooling season. This can work well when the ductless unit is in a large central open plan area with a ducted central return located in the same large area. A previous lab study (Withers and Sonne 2014) demonstrated good potential for energy savings and RH control in a lab building with a high mechanical ventilation rate. Another recent study of this measure in six occupied Florida homes, without mechanical ventilation, measured cooling savings of 37% and heating savings of 59% (Sutherland et al. 2016). The MSHP chosen for the controlled lab study had a specified range in cooling capacity from 3,100–18,400 Btu/h (0.9-5.4 kW). This seemed well-suited at low load and for cooling loads most of the year. However, it was observed that the lower delivered capacity sustained over 15 minute periods was about 7-8kBtu (2.0-2.3 kW). This is about 2.25 times greater than the stated low capacity. The lowest capacity may have been delivered during transitional periods shorter than our 15 minute data intervals.

The calculated design day ACCA Manual J cooling load (Rutkowski 2006) was 21.0 kBtu/h (6.1 kW) of cooling capacity. The first two test configurations had a dehumidifier enabled to operate if the living room RH exceeded 60%. The dehumidistat was located in the living room next to the thermostats about 12 feet (3.7 m) away from the mechanical ventilation supply air discharge. Mechanical ventilation air was delivered into the living room space near the dehumidifier. The dehumidifier was an EnergyStar unit rated to remove 70 pints (33 L)/day.

The third test configuration was conducted to see if a variable capacity MSHP having a very low-end rated capacity (ideal during low cooling load) could control indoor RH without the use of a dehumidifier. Since the upper range capacity was too low to meet the design load, the MSHP was operated as the primary cooling unit and a central ducted system was enabled to cool if the MSHP could not maintain an interior condition at 76°F (24.4°C). The central system was set two degrees above the MSHP set-point of 74°F (23.3°C). Since the minisplit indoor unit was located in a main central open area of the living room, the central ducted system was also used to circulate house air on a fan recirculation cycle of 20 minutes on and 20 minutes off to improve thermal distribution. Humidity control was improved by waiting 20 minutes after a central system cooling cycle had ended before enabling a fan recirculation cycle. This minimized evaporation of condensed water off of the evaporator coil.

This project was conducted during a period covering late spring through early winter. In central Florida this period tends to have weather conditions that vary between warm dry, warm moist, hot moist, cool moist, and cool/cold dry (short heating periods). Days with low cooling load and high outdoor moisture content (dewpoint or W) present the greatest moisture control challenge. Such conditions occurred sporadically throughout the test period.

Occupancy was simulated with added internal sensible and latent heat that was controlled by automation schedules. The internal sensible heat load daily average was 3,398 Btu/h (994 W) and latent heat average daily rate was 446 Btu/h (131 W). More than 100 channels of data were used to characterize indoor and outdoor temperatures, RH, solar radiation, system airflows, condensate, and energy consumption for about seven months to obtain a variety of seasonal data. Data was sampled every ten seconds and then stored at fifteen minute intervals by dataloggers. A central computer system periodically transferred data from dataloggers several times per day, screened data for errors or missing values, flagged such data, then processed and stored it in a secure data account.

Comfort-related metrics reported in this paper are limited to temperature and RH. These two measurements are most commonly measured when addressing indoor comfort and moisture concerns given the low cost and simplicity of measurement compared to other comfort-related metrics established in ASHRAE 55-2013 (ASHRAE 55 2013b). While there is no current single upper RH standard limit, some high-performance home programs have established an upper level of indoor RH at 60% (EPA 2013) (Rudd 2013b). RH levels below this have low potential for indoor moisture-related issues. Outdoor moisture content will be discussed in terms of dewpoint temperature (dpt).

RESULTS

Cooling Energy

The combined energy of space cooling and dehumidifier was plotted against the daily average difference in temperature (Δt) between outdoors and indoors. The indoor temperature is represented by the average of five locations around the home: three bedrooms, a central hallway, and at the thermostat location in the living room. A least-squares best-fit analysis was used to develop equations of space-conditioning energy versus Δt using measured data. The coefficient of determination, R^2, for daily space cooling plus daily dehumidifier energy versus Δt ranged between 0.92 up to 0.99, which indicated very strong correlation. The daily energy was almost entirely cooling energy as there were most days that the dehumidifier did not run, and when it did, the energy use represented very little of the combination. The central ducted cooling energy and fan recirculation energy are included in test case three where the minisplit was operated as the primary air conditioner. The best-fit equations were used in conjunction with typical meteorological year (TMY3) data for Daytona, Florida to predict annual cooling energy consumption for each test configuration. The results are representative for a climate region with significant hot and humid weather. The annual energy and predicted savings results are shown in Table 2. Peak power is predicted for a Δt of 15.0°F (8°C) which represents an outdoor condition of 91.0°F (32.8°C) and indoor temperature of 76.0°F (24.4°C). The peak power results are based on linear regression least-squares analysis of measured hourly cooling power versus hourly average Δt. Test Case 1 is used as the baseline for comparison for annual savings and peak reduction.

Table 2. Predicted Annual Cooling Energy, Peak Cooling Power, Use and Savings

Test Case	Annual kWh (MBtu)	Savings kWh/yr (MBtu), %	Peak kW (kBtu/h)	Peak Reduction kW (kBtu/h), %
1 Central ducted SEER 13	4820 (16.45)	---	2.04 (6.97)	---
2 Central ducted SEER 22	3743 (12.77)	1078 (3.68) 22.4%	1.56 (5.33)	0.48 (1.64) 23.5%
3 Ductless MSHP with central ducted SEER13	3224 (1.10)	1596 (5.45) 33.1%	1.49 (5.09)	0.55 (1.88) 27.0%

The ductless variable capacity MSHP test showed predicted annual savings of about 1596 kWh (5.45 MBtu) which was about 33% lower than the standard base efficiency central fixed capacity system. The MSHP also indicated about 0.55 kW (1.9 kBtu/h) peak cooling reduction which should be attractive to summer peak dominated electric utilities. The variable capacity system experienced a greater percent savings since it has about twice the runtime and therefore greater conductive gains. The winter heating peak reduction could be much greater where electric strip heat is used in central heating systems.

Indoor Comfort

Individual comfort varies widely based upon several factors as is accounted for by ASHRAE Standard 55 (ASHRAE 2013b). There is no single temperature or moisture limit at which everyone will be satisfied. Daily averages are useful for summarizing conditions of a period of time and may be useful as a simplified general representation that could be used to consider potential consequences upon building materials. Individual responses to comfort do not respond on a daily basis, but on some smaller interval. Indoor and outdoor conditions are shown here as daily and hourly representations.

Seasonal summer days

Seasonal summer days represented here use days with some variable outdoor dry bulb temperature where the outside dewpoint temperature was about 70°F (21°C) or greater. Indoor daily temperature and humidity were controlled well during hot humid weather conditions for Test 1 and 2, and fairly well for Test 3 using the MSHP. Table 3 shows a summary of outdoor and indoor daily conditions as well as average daily runtime of the different space conditioning systems. The reader is reminded that the dehumidifier was only used in Tests 1 and 2, and that Test 3 utilized the MSHP as the primary cooling system and had the dehumidifier disabled. The last row shows data during a short test to observe the "dry" cooling mode of the MSHP. The data from this row does not represent a seasonal summer day, but is shown here to demonstrate that variable capacity systems can still have long runtime and help improve comfort during low cooling load periods. This test was run similar to Test 3 during very low cooling load period of a few weeks. There were only two days available that had low cooling load and elevated outdoor moisture content, which present a challenge for typical cooling systems to adequately dehumidify.

Table 3. Daily Seasonal Summer Average Indoor Temperatures and RH

Test Case	Outdoor dpt °F (°C)	Out Temp. °F (°C)	Daily Solar kBtu/ft² (kWh/m²)	Living Rm. °F (°C)	3 Bedrooms °F (°C)	Living Rm. RH %	House RH %	Ducted Unit Runtime % of day	MSHP or Dehumidifier Runtime % of day
1 Central SEER 13	71.7 (22.1)	82.3 (27.9)	231 (6.3)	77.3 (25.2)	77.0 (25.0)	49.8	50.3	64.8	0.0
2 Central SEER 22	72.6 (22.6)	84.0 (28.9)	231 (6.3)	77.2 (25.1)	76.9 (24.9)	51.3	52.2	87.3	2.0
3 Ductless MSHP with central SEER13	71.1 (21.7)	80.0 (26.7)	184 (5.0)	75.4 (24.1)	78.2 (25.7)	53.3	52.3	9.2	95.0
* MSHP "dry" mode with central SEER13	67.5 (19.7)	74.5 (23.6)	125 (3.4)	75.0 (23.9)	76.5 (24.7)	55.9	56.1	2.8	97.7

*Temporary testing during very low cooling load period; not part of primary testing.

Indoor temperature is shown for the living room and the average of the three bedrooms. There is very little difference between these two with the central ducted systems used in Tests 1 and 2. As expected there is more difference between the living room and bedrooms with Test 3 since most of the cooling is first delivered into the open plan living room area, then distributed by central fan cycling and some limited central cooling cycles. This average daily difference was 2.8°F (1.5°C). The hourly peak difference between the living room and each bedroom did not exceed this. The living room RH is shown since this is where the mechanical ventilation air was delivered. The daily average shows that it is similar to the overall house average for all tests. While these are similar within given tests, the variable capacity systems had greater indoor RH than the fixed capacity system, and the MSHP tests had the highest RH. Reasons for this will be discussed later. The fixed capacity system ran about 65% on average for hot humid days while the central variable system cooled for about 87% of the day. Test 3 shows the MSHP cooling operated for about 95% of the day and the central ducted system cooled about 9% of the time. The dry mode of the MSHP during very low load shows it still operated 98% of the time with only 3% runtime of the central system. By design, the daily average indoor RH could not exceed 60% for Tests 1 and 2 since the dehumidifier would operate as needed. Test 3 was run without the dehumidifier to see if the RH would exceed 60% and if so by how much.

Based on hourly averages, Tests 1 and 2 did not have indoor RH exceed 60% during seasonal summer weather. Test 3 had rare occurrences where the hourly average RH was between 60%-62% RH in the living room, and sometimes in the master bedroom. Elevated RH, when it did occur, was during early morning hours between 3am-8am during. This is during the lowest cooling loads and two hot showers automated in the master bedroom bathroom between 7 am- 8:15am. Under moderate summer conditions the living room and master bedroom RH exceeded 60% RH about 4% of 216 hours evaluated. Figure 1 demonstrates how indoor RH varied depending upon MSHP cooling load during a five-day period in September. This figure shows hourly interval data with the lowest living room RH occurring during greater cooling loads and colder supply air temperature (SAT). The elevated RH occurred with lower cooling load and warmer SAT. The indoor RH varied in direct proportion to the MSHP SAT. A least-squares linear regression analysis found a coefficient of determination R^2=0.77. These brief periods of elevated RH may be acceptable to some where no modification, such as enabling a dehumidifier or activating dry or humidity cooling control modes, may be necessary. The ducted systems also faced the same challenge of adequately removing moisture during very low cooling load periods. The indoor RH had higher peaks just over 60% during overnight lower cooling load periods seen in the first three days of Figure 1. The RH was lower on relatively higher cooling load periods as shown in the last two days of Figure 1.

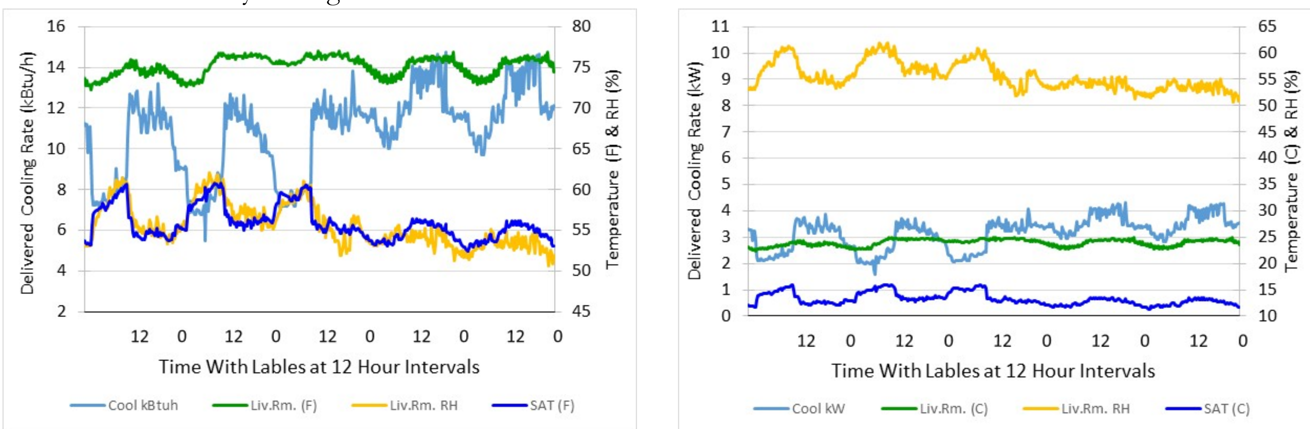

Figure 1. Test 3 data illustrates impact of MSHP cooling load and supply air temperature upon indoor RH

Low load cooling period

There was only a short period of a few weeks available to evaluate indoor RH control under very low cooling loads. Since very little is published on this for MSHP systems, this system was chosen to be evaluated during this time. Operation of MSHP in the standard cooling mode during very low load periods resulted in living room RH between 58%-64%. This system also has a "dry" control mode designed for lower cooling load periods. The dry mode operated within a smaller range of capacity with a colder coil on average, however it appears that there is room to improve the moisture removal in this mode. Figure 2 shows a composite day made up of hourly intervals from a nine-day period that began December 27, 2014. This period had average outdoor temperature of 70.3°F (21.3°C), and average outdoor dew point of 65°F (18.3°C). While Figure 2 composite shows that humidity was not out of control, it was borderline elevated during late mornings. Two days of this period, January 1-2, had very low cooling loads. This period offered an opportunity to see that even though the dry mode improved humidity control, more improvement could be made to the RH control mode performance.

The MSHP was monitored to be able to measure delivered cooling. Figure 3 shows the measured indoor RH and the sensible heat ratio (SHR) at 15 minute intervals. The indoor RH was highest when supply air temperature was warmer, as seen earlier in Figure 1. It was also much higher when the SHR was greater than 0.8. Typical SHR under rated conditions is usually around 0.75. The SHR values greater than 1 indicate that moisture was being added to the supply air. This happened when the compressor turned off, the coil warmed up, and air blew through the indoor evaporator coil, evaporating water off the coil and delivering it back into the space. The fan was set to auto, so this was not due to a continuous fan setting. Such episodes shown in Figure 3 occurred due to frequent indoor unit fan circulation periods between cooling cycles where the fan cycles on for about 20 seconds every 2.5 minutes. During one three-hour period when the MSHP stayed off, about 0.26 pints (0.123L) of moisture were re-evaporated back into the air. While this is not a lot of moisture, it is detrimental to good humidity control and should not occur during humidity control modes.

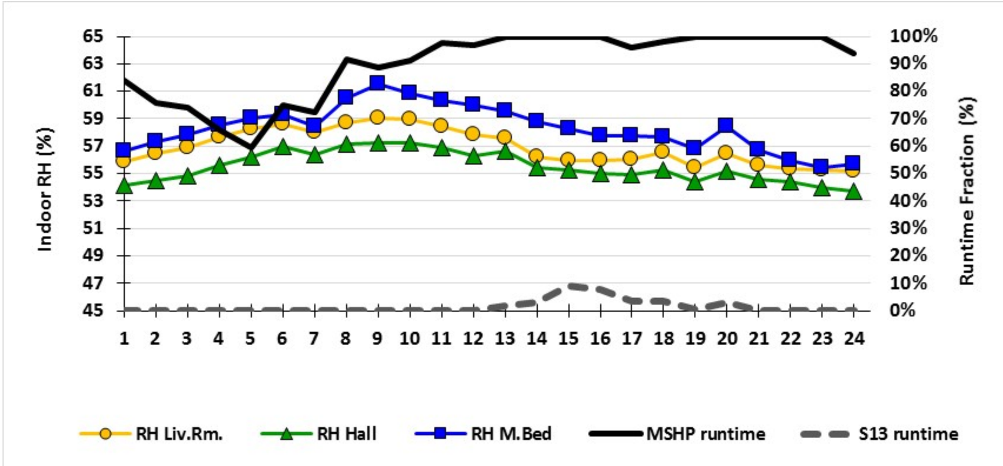

Figure 2. Indoor RH and cooling runtimes with MSHP operated in dry mode and secondary SEER 13 central cooling

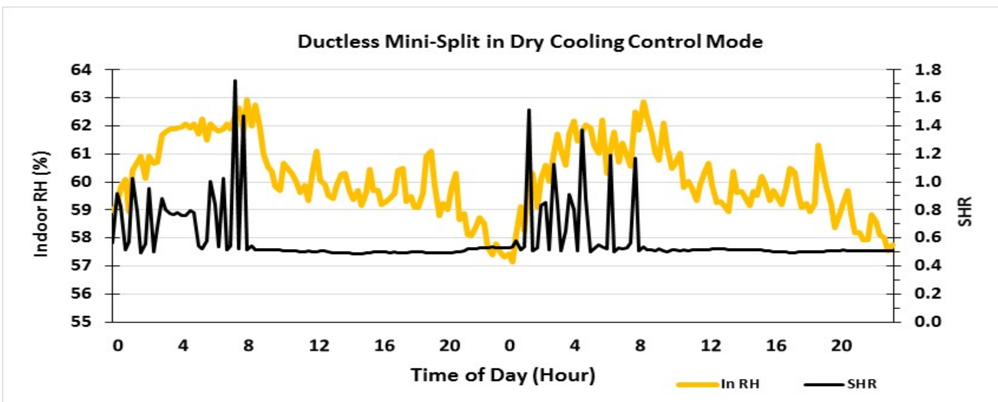

Figure 3. Fan recirculation cycles resulted in periods of high SHR contributing to high indoor living room RH

CONCLUSION

Variable capacity systems demonstrated significant energy savings of 22%-33% compared to a central ducted fixed capacity system. The ductless MSHP demonstrated the best potential annual energy savings due to very high mechanical efficiency and improved distribution efficiency due to significantly decreased conductive heat gain associated with ducts in attic space. More research is needed to determine how well indoor comfort and energy use is controlled through the use of multiple ductless MSHP systems or a ductless multi-split system in place of central ducted systems in mechanically ventilated homes during hot and humid weather conditions.

All three test configurations demonstrated reasonably acceptable thermal distribution for a small single-story home during low to high cooling load periods. Homes with more than one central heating and cooling system would likely have a greater challenge maintaining uniform thermal conditions with a single MSHP unit. All configurations could maintain an average daily RH throughout the home below 60%, but hourly intervals indicate some periods when RH exceeded 60% in at least some parts of the home during lower cooling load periods. Indoor hourly average

RH did not exceed 65% during the testing periods. In tests with the dehumidifier enabled, the average daily dehumidifier runtime was shorter than 0.1% on average summer days for the fixed capacity configuration, and only about 2% dehumidifier runtime for central ducted variable capacity configuration. On days with dehumidifier operation, the energy use was limited to 0.5 kWh -0.9 kWh (1.7 kBtu – 3.1 kBtu) per day. The low dehumidifier use is, in part, due to good dehumidistat placement in main body living room away from direct impact from mechanical ventilation air.

It appears that some variable capacity air conditioners with humidity control modes are close to being able to cool and adequately dehumidify newer mechanically ventilated homes at least under specific circumstances. The variable capacity systems tested could likely further improve dehumidification using the existing equipment if humidity control mode algorithms were improved to sustain colder coils at lower cooling capacities and allowed longer operation at the lowest capacity stages instead of operating the lowest capacities during very short durations and cycling off. It is not suggested to alter standard or economy cooling control modes. This would maintain the highest cooling efficiency during standard or economy modes, and allow occupants to enable an improved dehumidification cooling mode as needed. There is also a need for better operational manuals that provide information to users to better understand what to expect from dehumidification modes, and when and how they should be utilized.

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