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Systems Evaluation at the Cool Energy House

J. Williamson and S. Puttagunta
Consortium for Advanced Residential Buildings

September 2013

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Prepared for:

The National Renewable Energy Laboratory

On behalf of the U.S. Department of Energy's Building America Program

Office of Energy Efficiency and Renewable Energy

15013 Denver West Parkway

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NREL Contract No. DE-AC36-08GO28308

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Prepared under Subcontract No. KNDJ-0-40342-03

September 2013

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Definitions

A/C	Air conditioner
AHU	Air handler unit
BEopt™	Building Energy Optimization software
Btu	British thermal unit
CARB	Consortium for Advanced Residential Buildings
CEH	Cool Energy House
CFM	Cubic feet per minute
COP	Coefficient of performance
DHW	Domestic hot water
EF	Energy factor
ERWH	Electric resistance water heater
HERS	Home Energy Rating System
HPWH	Heat pump water heater
HVAC	Heating, ventilation, and air conditioning
kWh	Kilowatt hour
NREL	National Renewable Energy Laboratory
RH	Relative humidity
S/T	Sensible heat to total heat ratio
SWA	Steven Winter Associates, Inc.
WHD	Whole-house dehumidifier

Executive Summary

Steven Winter Associates, Inc. monitored several advanced mechanical systems within a 2012 deep energy retrofitted home in the small Orlando suburb of Windermere, Florida. This report provides performance results of one of the home's heat pump water heaters (HPWHs) and the whole-house dehumidifier (WHD) over a six-month period. In addition to assessing the energy performance of these systems, this study sought to quantify potential comfort improvements over traditional systems. This information is applicable to researchers, designers, plumbers, and heating, ventilation, and air conditioning contractors. Though builders and homeowners can find useful information within this report, the corresponding case studies are likely better references for these audiences.

Heat Pump Water Heaters

Over recent years, HPWHs have grown in both market interest and product capability. These units harness the heat transfer benefits of the vapor compression heat pump cycle to extract heat from surrounding air space and supply it to water for domestic hot water (DHW) needs. In doing so, an HPWH can provide DHW more efficiently than a standard electric resistance tank water heater. Furthermore, the unit's operational byproduct of cooling and dehumidification can be beneficial in hot-humid climates.

Energy simulation analysis was performed with Building Optimization Energy Plus (BEoptE+) v1.3 software. Energy modeling predicts that an HPWH can annually save the Cool Energy House test home approximately 64% (or $113/yr) on DHW utility costs over a standard electric resistance tank water heater (at the local electricity rate). There is an interesting inverse effect with HPWHs in the hot-humid climate zone. While the cooling/dehumidification benefits and higher coefficients of performance of an HPWH would be more advantageous in the hot-humid climate, mains water temperature tends to be higher. The mains water temperature ranged from 75°–85°F over the June through October monitoring period. This means less water heating is required (less than a cold-climate location), resulting in the overall water heating cost being minimized. Therefore, the cost benefit of the HPWH was diminished.

The long-term monitored data show that water draw profile (both volume and frequency) has a strong influence on the unit's operating efficiency. If too great a volume is drawn in a short period of time, the system will revert to electric resistance backup elements to provide supplemental heat. Table 1 displays a summary of HPWH operating and performance conditions that existed during the monitoring period.

Table 1. HPWH Operating Condition and Performance Study (June 23, 2012 to October 30, 2012)

Hot Water Set Point	120°F
Average Water Inlet Temperature[1]	82°F
Average Water Outlet Temperature[1]	117°F
Total COP	2.2
% Electric Resistance [2]	23%
Average Hot Water Use[3]	48.8 gal/day

[1] Average estimated with 15-min periods containing near-continuous flow
[2] % electric resistance = % of total kWh consumed by resistance
[3] Average of daily averages

Whole-House Dehumidifier

In hot-humid climates, it becomes a challenge to simultaneously control moisture levels (latent heat) and temperatures (sensible heat) within a living space. As a result, it is often beneficial to utilize a WHD to specifically address the building's latent load while controlling the sensible load with the home's central air conditioning (A/C) unit. Once the control of each of these units is independently decoupled, the homeowner can achieve improved comfort and cost savings.

BEoptE+ energy modeling predicts that a combined A/C-WHD system, set at 78°F and 55% relative humidity (RH), will be able to provide similar indoor humidity ratio levels to an A/C only system at 75°F. However, the combined system will accomplish this while maintaining an increased moisture-comfort level and energy savings. Modeling shows that the combined system has the potential to support RH levels below the defined 60% comfort limit for 15.8% more of the year while saving 8.2% annually (or $53/yr) over the 75°F A/C only base case.

Some previous research on dehumidifiers (primarily stand-alone dehumidifiers) has been done, but it has been mostly focused on lab testing or analysis of the annual dehumidifier use based on estimates from initial measurements and manufacturer's performance data. This study sought to provide long-term field data to validate this energy modeling.

Long-term testing shows that the combined system, at the homeowner's preference of 76°–78°F and 65% RH, was able to maintain acceptable indoor summer conditions for a majority of the monitoring period. Indoor temperature was maintained between 74°F and 80°F for 91.5% of the time while RH did not exceed the 60% limit for 99.9% of the period. The dehumidifier's average performance was 3.56 pints/kWh over the monitoring period. Short-term scenario testing that compared three set point configurations (78°F and 60%, 78°F and 50%, and 75°F with no dehumidifier) suggest that a paired A/C-WHD system at 78°F and 60% can maintain indoor comfort in the most cost-effective manner.

These results indicate that utilizing a WHD to specifically control a building's latent load while supporting the sensible load with a central A/C unit can provide multiple benefits. Most importantly, the homeowner can see improved comfort that comes with heightened moisture control. Additionally, the system will maintain this comfort while saving energy and utility costs.

1 Introduction

In 2012, the Consortium for Advanced Residential Buildings (CARB) provided the technical engineering and building science support for a highly visible demonstration home in connection with the National Association of Home Builder's International Builders Show. This project, which was unveiled at the 2012 International Builders Show in Orlando on February 9, is known as the deep energy retrofit Cool Energy House (CEH). The CEH began as a mid-1990s two-story traditional spec house of about 4,000 ft^2 in the upscale Orlando, Florida suburb of Windermere. The homeowners, who were planning some interior renovations and a small addition, were recruited for this research project through their contractor, Southern Traditions.

The initial objectives of the CEH project included reducing simulated annual source energy consumption by 50% compared to the existing pre-retrofit home. The project team realized from the start that reaching this performance goal in the context of a relatively light (not gut-rehab) renovation was a significant challenge, and a challenge that might not be fully accomplished. However, by aggressively pushing the performance level, this project was able to achieve an estimated 49% reduction in source energy use compared to the existing home.

Nearly all the interior wall surfaces and all the exterior brick veneer remained in place. The two-story insulated frame walls were insulated further by blowing fiberglass from the interior through 6-in. horizontal slots in the gypsum board. The partially occupied attic space was sealed, finished, and encapsulated with R-30 closed-cell spray polyurethane foam at the roof deck. Downsized air-source heat pumps replaced the two older space conditioning systems, while most of the existing ducts were sealed with mastic and reused (Zoeller et al. 2013). Figure 1 shows the front façade of the post-retrofit home.

Figure 1. Front elevation (west) of the CEH post-retrofit

The 49% simulated reduction in energy consumption over the pre-existing conditions translates to approximately 12,687 kWh or $1,500 in anticipated annual electricity bill savings (see Appendix C for more information). The retrofit also provided an improvement to indoor air quality by supplying whole-house and local ventilation, as well as dehumidification, improvements to thermal comfort by air sealing, duct sealing, and reductions in solar heat gain through windows.

The home also features advanced mechanical systems that are the focus of this research report. CARB monitored a heat pump water heater (HPWH) and a whole-house dehumidifier (WHD) over a six-month period to assess their performance in an occupied setting. These two technologies address two common gaps in this region—high efficiency electric water heating and effective humidity control in a hot-humid climate zone.

Domestic hot water (DHW) is a major source of energy consumption in almost every residential building. In most cases, DHW is supplied through fossil-fuel combustion units (e.g. gas, oil, and propane) or electric resistance water heaters (ERWHs) when fossil fuels are not available or desired. In recent years, HPWH technology has grown in both product capability and market demand. The high efficiency potential of this technology makes it a desirable alternative compared to other DHW technologies—specifically ERWHs.

A heating, ventilation, and air conditioning (HVAC) system's ability to maintain indoor environmental quality levels within human comfort preferences is one of the most important features of a building. In hot-humid climates, such as Orlando, this is not always a trivial task. Year-round high outdoor relative humidity (RH) levels create a unique situation for comfort control. Since moisture removal (latent heat) and temperature maintenance (sensible heat) are not always needed simultaneously, comfort conditions can be truly optimized only when each is addressed by separate systems. By utilizing a WHD to address the latent load and a separate central air conditioner (A/C) to address the sensible load, indoor comfort should be improved.

1.1 Heat Pump Water Heater

Water heating is the third largest contributor to residential energy consumption in the United States, after consumption attributed to space conditioning and lighting, appliances, and miscellaneous electric loads. In the United States, residential water heating consumes 2.11 quadrillion Btu of site energy per year, which is 20% of total residential site energy (EIA 2005). The vast majority of water heaters are powered by natural gas (58.4 million households) and electricity (46.7 million households), but fuel oil (3.6 million households) and propane (4.2 million households) also have sizable shares of the water heating market (EIA 2009). Fortunately, more efficient water heaters for both major water heating fuels are becoming more readily adopted in the marketplace. In terms of electric heating options, HPWHs are a promising technology that has the potential to reduce water heater energy consumption by around 50% compared to traditional ERWHs.

HPWH technology promises to significantly reduce energy consumption for DHW over traditional ERWHs. While ERWHs perform with energy factors (EFs) around 0.9, new HPWHs boast EFs upward of 2.0. High EFs are achieved by combining a vapor compression system to extract heat from the surrounding air at high efficiencies; electric resistance element(s) are still present to meet large demands. Looking at Table 2, swapping ERWHs to HPWHs could result in roughly 50% reduction in water heating energy consumption for 41.1% of all households. This impact is even greater in hot-humid climates in the South, where ERWHs make up more than half of all installed systems.

Table 2. 2009 Residential Energy Consumption Survey Data Sample of Households With ERWHs

Census Region	Fraction of Households With ERWHs by Region
Northeast	24.5%
Midwest	29.3%
South	64.6%
West	27.8%
National	41.1%

A key feature of an HPWH unit is that it is a hybrid system. When conditions are favorable, the unit will operate in heat pump mode (using a vapor compression system, which extracts heat from the surrounding air) to efficiently provide DHW. Yet it does not require homeowners to adjust their behavior to conform to the heat pump's capabilities. If a heat pump is unable to meet a higher water draw demand, the heater will seamlessly switch to electric resistance to provide a higher heating rate. Thus, hybrid mode provides the energy savings of heat pump mode (when possible) while being able to perform as an ERWH during periods of high DHW demand (Shapiro et al. 2013).

CARB monitored the in-situ performance of 14 HPWHs from 2010–2012 in homes in the heating-dominated climates of Massachusetts and Rhode Island. Data collected during this study have been used to address issues facing researchers of residential water heating systems. Usage patterns and environmental characteristics can vary dramatically between households, and researchers must be able to determine the correlation between various household characteristics and HPWH performance. The results provide considerable insight into the sensitivity of HPWHs to environmental variables, such as mains water temperature, set point temperature, ambient temperature, ambient humidity, and water draw profiles. This monitoring was done on HPWHs located primarily in unconditioned basements. The results of the evaluation were encouraging (Table 3). Most importantly, all customers were satisfied with the supply of hot water and noticed a reduction in their utility bills (Shapiro et al. 2013).

Table 3. Performance Summary of Monitored HPWHs by Model

Model	Capacity (gal)	Rated EF	First Hour Rating (gal/h)	Measured Average COP*	COP Range
General Electric	50	2.35	63.0	1.82	1.5–2.1
A.O. Smith	60/80	2.33	68.0/84.0	2.12	2.1
Stiebel Eltron	80	2.51	78.6	2.32	2.0–2.6

* Coefficient of performance, the ratio of heat provided per unit of electrical energy used.

A tradeoff that is often overlooked is that if the HPWH is located within the home (rather than an attached garage), the heat pump is not collecting “free” energy to heat water, as is often touted for this technology. It is removing heat from the home, which may need to be replenished by the space conditioning system. During the cooling season, this serves as an added benefit, but during the heating season, this is a concern. Although many desire to integrate this cooling and dehumidification byproduct into the space conditioning system, it is not a simple system to integrate. Since a HPWH’s space conditioning impact is sporadic, due to its being a function of hot water demand and not room air conditions, the quantity of heating/dehumidification that can occur on any day is inherently variant.

This report analyzes the system performance of an HPWH installed within the thermal boundary of a hot-humid climate test home. The intent is to examine the extent to which geographical install location influences DHW heating performance. In addition, the study analyzes the effects that this system has on space conditioning.

1.2 Whole-House Dehumidifier

One of the most common complaints that homeowners have in a hot-humid climate is that they are too cold when indoors. Often the temperature set point is set lower than the desirable cooling level to try to increase moisture removal (latent cooling), so that the interior air is not humid or “muggy.” Even when a thermostat with humidity control installed, the humidity control logic is still based on dropping the dry-bulb temperature (sensible cooling) set point by a couple of degrees to have the system run more frequently. However, this method is not always effective in maintaining indoor RH or human comfort. By separating the systems that control the latent and sensible cooling, this specific complaint can be eliminated.

Figure 2 provides an illustration that gauges human comfort based on the RH of the ambient air. As seen from the range of perceived temperatures at a summer set point of 75°F, the perceived temperature can range from a maximum of 80°F at 100% RH to a minimum of 69°F at 0% RH. This justifies the importance of humidity control to maintain comfort levels and provides an insight into how comfort can be maintained by removing latent energy with a dehumidifier rather than a more energy-intensive central A/C unit.

WHAT THE AIR FEELS LIKE

HOW HOT THE HEAT-HUMIDITY COMBINATION MAKES IT FEEL.
EXAMPLE: AIR AT 90ºF WITH 50% RH FEELS LIKE 96ºF TO THE HUMAN BODY!

RELATIVE HUMIDITY (PERCENTAGE)

AIR TEMPERATURE (DEGREES FAHRENHEIT)	0	5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100
100	91	93	95	97	99	101	104	107	110	115	120	126	132	138	144						
95	87	88	90	91	93	94	96	98	101	104	107	110	114	118	124	130	136				
90	83	84	85	86	87	88	90	91	93	95	96	98	100	102	106	109	113	117	122		
85	78	79	80	81	82	83	84	85	86	87	88	89	90	91	93	95	97	99	102	105	108
80	73	74	75	76	77	77	78	79	79	80	81	81	82	83	85	86	86	87	88	89	91
75	69	69	70	71	72	72	73	73	74	74	75	75	76	76	77	77	78	78	79	79	80
70	64	64	65	65	66	66	67	67	68	68	69	69	70	70	70	71	71	71	71	71	72

EXTREME DANGER
DANGER
EXTREME CAUTION
CAUTION

SOURCE: TEMPERATURE - HUMIDIY INDEX WAS DERIVED BY R.G. STEADMAN, JOURNAL OF APPLIED METEOROLOGY, JULY 1979. M27328

Figure 2. Perceived air temperatures based on actual temperature and RH

(courtesy of Honeywell; Honeywell 2008a)

Indoor comfort, as defined by ASHRAE, is achieved when indoor humidity ratio is controlled below 0.012 lb_m water/lb_m dry air (ASHRAE 55-2010). For common indoor summer set points of 75°–80°F, these conditions are achieved while RH is below the range of 55%–65%. In addition to improved comfort, maintaining indoor RH at < 60% also can reduce: (1) insects like earwigs and millipedes; (2) occupant health issues with allergies, asthma, and diseases from fungal toxins; and (3) damage to building contents from corrosion or warping (Aprilaire 2009). For this research, an upper limit of 60% was used as the acceptable RH threshold within a conditioned living space. This upper RH limit was chosen to support human comfort at common set point temperatures, create a health-promoting environment, and maintain interior building conditions that support the durability of construction components.

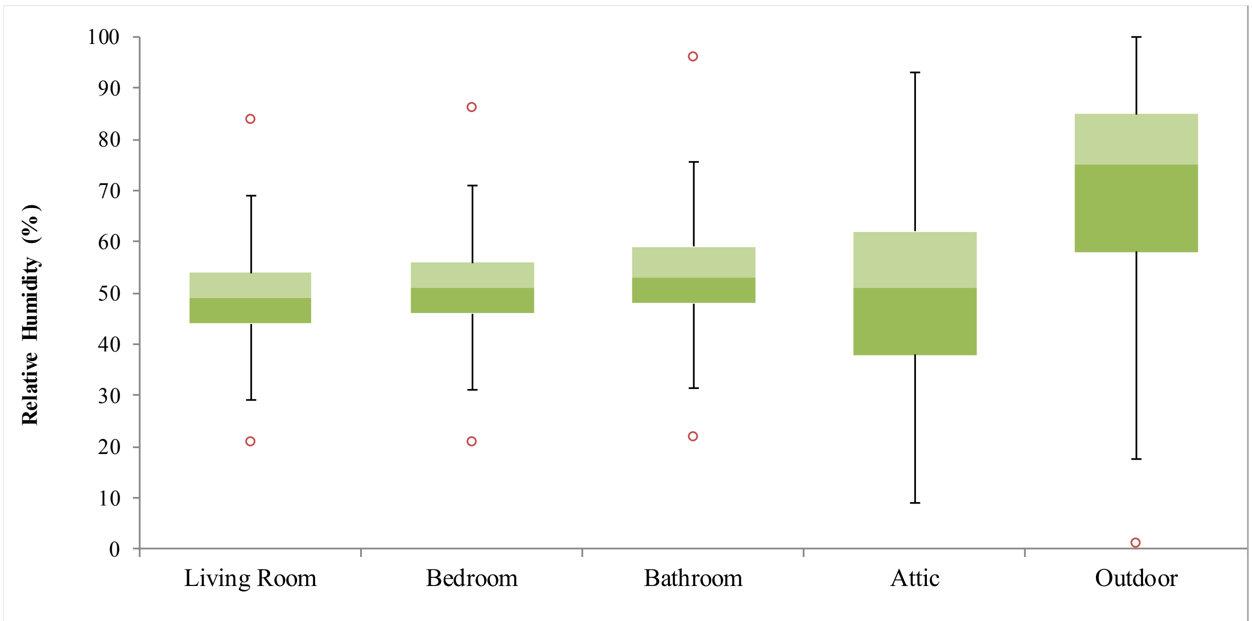
A WHD was installed in the CEH with its own dedicated compact distribution system to better address latent cooling. This allows the A/C system to have a smaller capacity such that it will be sized to specifically address the sensible cooling needs during the summer. One added benefit of this configuration is that the dehumidifier also has a lower power draw than the air conditioner—strictly considering energy used for moisture removal. Additionally, with the combined capability of maintaining better indoor RH, the thermostat cooling set point can often be set higher (~78°F), resulting in additional energy savings while maintaining (or improving) occupant comfort.

The climate pattern in a hot-humid climate, such as Orlando, Florida, typically shows conditions in which the morning outdoor RH will often sustain moisture levels upward of 80% throughout the entire year. However, the outdoor dry-bulb temperature exceeds comfortable levels only during summer months. Even though latent energy removal will be needed year-round, sensible cooling is needed only from roughly May through October. This brings forth the need for a cooling system in which moisture removal can be accomplished without decreasing indoor air temperature. Hence, an independent WHD in conjunction with a central A/C system can be more effective in maintaining indoor comfort than an A/C alone. Even though the central A/C unit may have the mechanical capability of controlling latent load, it is unlikely that it will be able to do so without inducing uncomfortably low temperatures in the space.

The latent (moisture) load in homes is traditionally addressed by the central A/C unit, but how well these systems will actually control humidity levels varies drastically. One major issue with traditional A/C systems is the common industry practice of oversizing system capacity. With an A/C unit, the greatest moisture removal will occur once the unit has reached steady-state operating conditions. Steady-state performance generally begins 5–15 min after the initial compressor startup (Shirey et al. 2006). Hence, if the unit is oversized and can drop the indoor temperature below the set point very quickly (without spending much time in steady state), the latent cooling effects will be minimal since the evaporator coil has not achieved the coil temperatures where dehumidification performance is optimized.

Additionally, the initial period of a system's startup provides a condition where moisture can re-enter the airstream as warm air is blown across the not fully chilled evaporator coil (Katipamula and O'Neal 1991). Likewise, this re-evaporation effect can also occur with homeowners who chose to alternate between cooling and continuous fan-only mode. As a result, HVAC practices of oversizing system capacities and homeowner preferences of A/C settings can strongly influence the effectiveness of the A/C dehumidification process.

Unfortunately, there is an insufficient amount of measured data available on actual indoor humidity levels in U.S. households. Steven Winter Associates, Inc. (SWA) collected one full year of indoor temperature and humidity data for a sample of 60 homes across three different climate regions—the hot and humid Southeast, the cold Northeast, and the marine Northwest (Arena et al. 2010). Figure 3 shows RH box plots in various interior locations throughout a home for the hot-humid climate study homes. All whiskers represent 1.5 times the inter quartile range, whereas, inter quartile range is the 75th percentile minus the 25th percentile.



	Living Room	Bedroom	Bathroom	Attic	Outdoor
Min	21.0	21.0	22.0	9.0	1.0
Max	84.0	86.0	96.0	93.0	100.0
Mean	49.0	51.2	53.2	49.9	70.8
Median	49.0	51.0	53.0	51.0	75.0
% Outliers	0.51%	2.64%	1.43%	0.00%	0.09%

Figure 3. RH box plots for zone 2, hot-humid climate for each sensor location

(Arena et al. 2010)

Though the mean RH was within acceptable indoor levels, there were numerous cases of RH levels exceeding 60% for significant portions of the time. Additionally, RH levels of up to 96% were measured inside of the home's conditioned space (bathroom 96%, bedroom 86%, and living room 84%). This suggests that a central A/C unit alone does not always maintain adequate RH levels within the living space.

In 2005, a report was published that describes research done by Building Science Corporation on residential dehumidification in hot-humid climates (Rudd et al. 2012).This study monitored the indoor temperature and RH, as well as energy use of various dehumidifiers and ventilation systems. However, the report does not directly address the impact that the dehumidifier and ventilation units have in reducing cooling requirements of the building's central A/C system.

As follow-up research, Building Science Corporation performed a study in 2012 that evaluated the construction of new homes in the hot-humid climate of New Orleans (Osser and Kerrigan 2012). This study provides some evaluation of the supplemental use of a dehumidifier in a cooling system in hot-humid climates. However, since supplemental dehumidification could not be modeled in BEopt or Energy Gauge USA software at the time of the research, it is not included in the report. Rather than long-term monitoring with various cooling scenario set points, this study analyzes the annual dehumidifier use with estimates from initial measurements and manufacturer's performance data.

Further supporting the need for supplemental dehumidification, the National Renewable Energy Laboratory (NREL) performed energy modeling to analyze indoor RH in various home types. This study concluded that supplemental dehumidification is needed to maintain interior RH levels < 60% in a hot-humid climate home. (Fang et al. 2011) Again, this 60% RH threshold is based on providing suitable comfort and health to occupants as well preserving the home and the contents within it. Additionally, in 2011, NREL performed laboratory research on the performance of six dehumidifiers (Winkler et al. 2011). This study analyzed the lab-controlled operational performance of units at an array of conditions to determine performance curves. As this study does provide extensive results on the in-laboratory performance, it does not directly evaluate the performance of the units in an occupied setting.

Aprilaire, a manufacturer of dehumidifiers, performed Transient System Simulation Tool modeling on a 2,000-ft^2 home in Miami, Florida, to demonstrate the benefit of an A/C and dehumidification system (Aprilaire 2009). When comparing a conventional A/C set at 75°F versus an A/C set at 78°F and dehumidification system set at 59% RH, the latter system maintained indoor RH below 60% for the entire year and resulted in 18% annual energy savings. The conventional A/C system resulted in 1,641 hours of RH >60% or 18.7% of the time.

In an alternative case, the conventional A/C was set to a 72°F set point as homeowners may lower the thermostat setting further in an attempt to reduce the moisture levels. In this scenario, the dual-system approach was even more beneficial. Though the occurrence of RH >60% was reduced to 280 hours for the conventional A/C or 3.2% of the year, the energy consumption was greatly increased. The dual-system approach resulted in a 44% annual energy savings. This modeling suggests significant cost savings that homeowners could achieve while improving the overall comfort of their homes.

Overall, this report addresses each of the previously mentioned topics while detailing the benefits of decoupling the humidity and temperature control such that a more comfortable living environment may be provided while minimizing electric consumption. It examines the system performance of the combined A/C and WHD to analyze energy and costs savings, as well as improved comfort conditions.

2 Methodology

2.1 Research Focus

The primary questions addressed by this research were:

- What is the expected efficiency of an HPWH located within the conditioned volume of hot-humid climate homes?
- What types of space conditioning implications are associated with utilizing a HPWH in the conditioned space of the home?
- Can a more energy-efficient home be created by separating the mechanical systems that address sensible and latent cooling?
- Can indoor comfort levels be improved by primarily addressing sensible load with a central A/C and latent load with a WHD?

2.2 Equipment Monitored

There were two main equipment sets that were remotely monitored in order to provide the data for this study: (1) an HPWH; and (2) a WHD. In addition, the air handler units' (AHU) runtime and indoor/outdoor conditions (temperature and RH) were monitored with Onset's HOBO data loggers.

2.2.1 Heat Pump Water Heater

Figure 4 shows the HPWH that was monitored in this study. This unit is a General Electric GeoSpring 50-gal hybrid water heater (first generation) and is located in the unvented attic of the CEH. The hot water that is provided by this unit is used in the upstairs bathrooms of the home. There is an additional HPWH that services the lower level water fixtures that was not monitored due to logistical reasons.

Figure 4. Cover removed from top of the monitored HPWH

2.2.2 Whole-House Dehumidifier

A dehumidifier utilizes refrigeration (vapor-compression cycle) to cool the inlet air below its dew point so that condensate forms on the evaporator coil and drains to the condensate pan. The air first passes by the evaporator coil to remove moisture and cool the inlet air, then flows over the condenser coil which reheats the air. The net result is warmer, drier outlet air to be supplied back to the home. Figure 5 provides terminology for the various components of the dehumidifier.

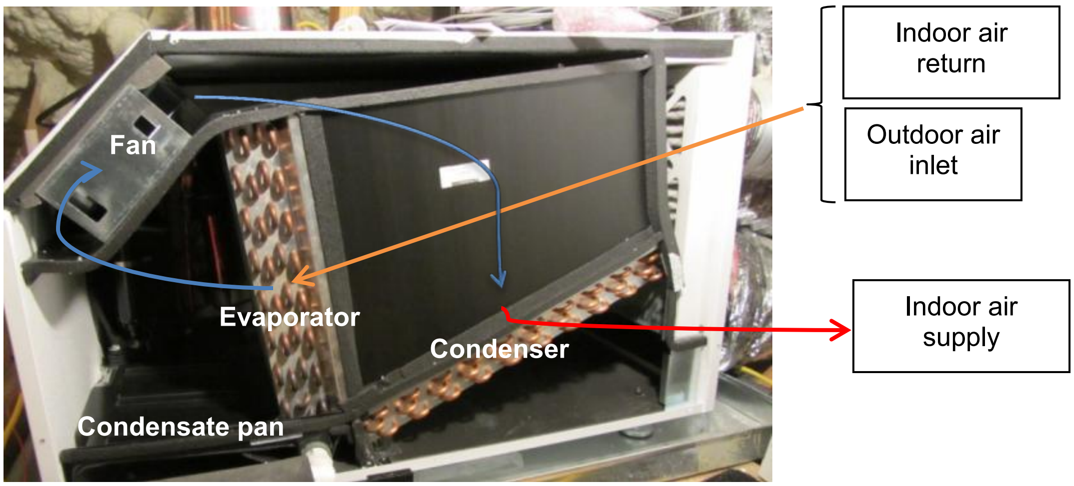


Figure 5. How the monitored dehumidifier works

The WHD that was monitored in this study was a Honeywell TrueDRY DH150 dehumidifier (with a daily condensate removal capacity of 150 pints/day @ 80°F/60% RH inlet air and an energy factor of 3.56 L/kWh). The sizing of this unit followed Honeywell's recommendation based on the square footage of the home.

This unit is located in the unvented attic of the CEH. This dehumidifier also provides whole-house supply-only ventilation. A 6-in. duct brings in outdoor air (~69 CFM), mixes with air from a 10-in. return duct (~330 CFM) from the stairwell of the home, is dehumidified as needed, and is supplied back to the home via a 10-in. supply duct (~399 CFM). For the CEH duct layout, the 10-in. supply duct feeds three branches going to the master bedroom, play room, and family room. Figure 6 shows a simplified schematic of the dehumidifier port configuration.

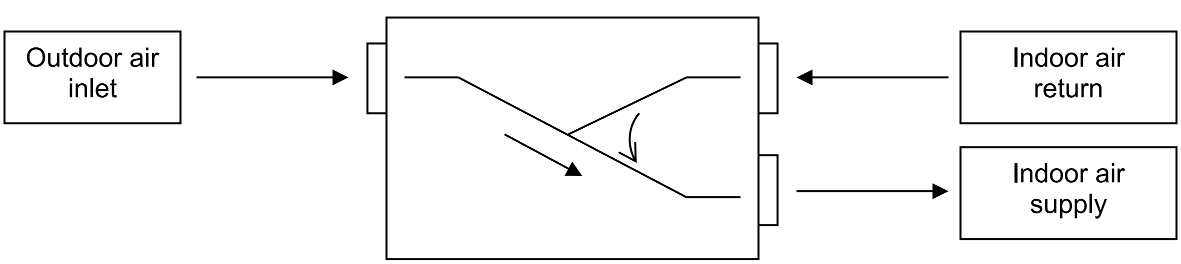


Figure 6. Schematic of dehumidifier ports

Control of this unit was specified to be Honeywell's TrueIAQ digital control, which can display indoor and outdoor temperatures and RH, the desired humidity setting, maintenance/service reminders, and can control whole-house ventilation to meet ASHRAE 62.2-2010 requirements. Unfortunately, the system was installed with a basic dial humidistat, so whole-house ventilation is provided only when RH levels in the home exceed the desired humidistat set point.

2.2.3 Central Air-Conditioning System

Ideally, CARB would have fully monitored both A/C units to be able to have a direct comparison between the dehumidifier and the A/C system, but the equipment layout and timing of this project did not allow for these A/Cs to be fully monitored. Only the runtimes of the two AHUs were able to be monitored. The entire central A/C system consists of two units that provide a combined cooling capacity of 4 tons (2 tons per unit). Figure 7 shows the home's outdoor condenser units (left image) and one of the indoor AHUs (right image).

Figure 7. Outdoor condensers and the AHUs for the second floor

2.3 Measurements

The monitoring of the mechanical equipment (HPWH, WHD, and central A/C) was carried out over a two-stage process. This process included gathering initial one-time readings at the start of data monitoring and continuously collecting various measurements in increments of 10 s—while outputting sums and averages in 15-min intervals—throughout the entire testing period. Although house temperature and RH were measured within the return of the WHD, Onset HOBO remote temperature/RH sensors were installed to quantify these conditions in the central stairwell. An additional temperature/RH sensor was located outdoors. See Appendix A of this report for the detailed list of measurements taken and calculations that were performed for this field monitoring.

3 Energy Modeling

Hourly energy simulation was performed with NREL's BEopt EnergyPlus v.1.3 software (BEoptE+) to estimate annual energy usage, utility bill costs, indoor conditions, and general system performance for various control configurations of the mechanical systems. As individual mechanical systems are being evaluated, it was assumed that the full cost of each system would be paid in cash rather than financed. Installation costs were selected from the National Residential Efficiency Measures Database (NREL 2012). The economic values used in the energy simulations are summarized in Table 4.

Table 4. Inputs of Economic Analysis

Economic Variables	Modeling Inputs
Electricity Rate*	$0.11/kWh + $8.00 monthly charge
Natural Gas Rate*	$1.32/therm + $8.00 monthly charge

* State average for Florida

3.1 Predicted Performance of the Heat Pump Water Heater

The costs of various DHW options were evaluated over their respective projected life expectancies. Table 5 displays a breakdown of lifetime costs that the energy simulation software uses to determine the finances of each system. It is important to note that the HPWH is modeled with a shorter lifetime and with a significantly higher initial cost than other tank units. The shorter estimated lifetime cost of the HPWH is due to uncertainties that develop because the technology is relatively new on the market and the unit's slightly more complicated system components than common ERWHs—with the main component difference being its compressor. The higher initial cost is because these HPWHs use a newer technology and HPWHs may require two technicians to install due to the added weight of the heat pump components and the additional routing of a condensate drain line. Note: Installed cost includes purchase price and installation cost.

Table 5. Lifetime Costs of Various DHW Technologies

Unit	Installed Cost	Annual Utility Cost	Lifetime (years)	Lifetime Cost	Cost per Year (Lifetime Cost/Lifetime)
Electric Tank (0.92 EF)	$708	$177	13	$3,015	$232
HPWH (2.35 EF)	$2,100	$64	10	$2,737	$274
Gas Tank (0.59 EF)	$797	$138	13	$2,589	$199
Gas Tankless (0.82 EF[1])	$1,160	$81	20	$2,774	$139

[1] A one-time energy factor derate of 8% is included due to cycling inefficiencies (Davis Energy Group 2006).

It is likely that an HPWH will also be able to reliably supply hot water to a home for 13 years, like other tank water heaters. If the HPWH does last this long, the technology becomes more financially competitive with traditional tank water heaters. A 13-year HPWH life expectancy would give the unit a lifetime cost/year metric of $225/yr. At this rate, the HPWH can be a cost-effective alternative to traditional ERWHs. However, to be the least-cost option based on this cost analysis, the first cost for the HPWH would have to be $753 or less, which is not likely even once the technology is market mature.

Annual lifetime costs show that a gas tankless water heater offers the greatest financial benefit for the homeowner. However, these financial benefits come with a few caveats in performance. With a higher EF rating and no standby losses, it would seem that there is no reason not to install a tankless gas water heater (assuming natural gas is available to the home). Even though it is true that tankless water heaters can be beneficial, there are some potential concerns:

- Tankless water heaters eliminate the standby loss of tank water heaters, but they do not provide hot water any quicker to the faucet. In fact, they may take 5–15 s longer as the unit senses the call for hot water and turns on the burner.
- Tankless water heaters typically require a minimum flow rate of 0.5 gpm to fire, based on their ability to modulate burner capacity. This limitation is designed to prevent continual operation in the case of a leak or during the event of a faucet not being fully shut off. Additionally, a minimum allowable flow rate helps to control the risks of the heat exchanger overheating. With low-flow fixtures/aerators being more standard water-saving features, this flow rate may not always be met.
- If the water source has a high mineral content, scaling of the low-mass heat exchanger can occur. The heating coils need to be flushed with a descaling solution per manufacturer's specifications, but most homeowners are not aware of this requirement.
- Tankless water heaters require venting for combustion air and to get rid of hot combustion exhausts. This venting introduces an additional cost over electric water heaters. Additional care also must be taken to ensure that carbon monoxide from the exhaust does not leak into the living space
- The gas line usually needs to be upsized from conventional ½-in. diameter piping to ¾-in. diameter piping because of the increased capacity required to heat the water more quickly.

Energy simulation was used to determine potential space conditioning benefits that can be achieved by providing DHW heating through an HPWH. When operating in heat pump mode, air that passes through the HPWH will be both cooled and dehumidified. Table 6 shows the estimated influence that an HPWH has on various segments of space conditioning. Since Orlando is a cooling-dominated climate, the greatest space conditioning benefits are realized through cooling (both sensible and latent). Most directly, it will provide a "free" added benefit in cooling months by both cooling and dehumidifying. Still the overall impact on space conditioning based on current energy models is minimal.

Table 6. Space Conditioning Utility Costs of DHW Heating Methods

DHW Heating Method	Cooling ($)	Heating ($)	AHU Fan ($)	Total ($)
Electric Tank	570	115	53	738
HPWH	562	118	52	732
Gas Tank	575	111	54	740
Gas Tankless	568	116	53	737

Finally, the utility costs of each DHW heating method were analyzed as the combined cost that resulted from water heating and space conditioning utility costs (Table 7). This table indicates that the HPWH is lower than each of the other DHW options. As a result, the HPWH provides the lowest total operating cost alternative for supplying hot water to a home under these assumptions.

Table 7. Combined Water Heating and Space Conditioning Utility Costs for Various DHW Methods

DHW Heating Method	Cost ($)
Electric Tank	915
HPWH	795
Gas Tank	878
Gas Tankless	818

Table 8 once again looks at the cost of various DHW options evaluated over their respective projected life expectancy, but in this case, the space conditioning impact was included. The space conditioning with the electric tank cost was considered the base case for assessing the space conditioning impact of the alternative heating methods. It is unlikely that an HPWH will ever compete with a gas tankless water heater, but if comparing to traditional ERWHs, this technology has promise once the HPWH market matures.

Table 8. Lifetime Costs of Various DHW Technologies Including Space Conditioning Impact

Unit	Installed Cost ($)	Annual DHW Cost ($)	Annual Space Conditioning Impact ($)	Lifetime (years)	Lifetime Cost ($)	Lifetime Cost/Year ($)
Electric Tank (0.92 EF)	708	177.45	–	13	3,015	$232
HPWH (2.35 EF)	2,100	63.68	–6.05	10/13	2,676/ 2,849	268/ 219
Gas Tank (0.59 EF)	797	137.86	2.20	13	2,618	201
Gas Tankless (0.82 EF*)	1,160	80.69	–0.59	20	2,762	138

* A one-time EF derate of 8% is included due to cycling inefficiencies (Davis Energy Group 2006).

Currently many utilities are providing incentives of $400–$1,000 to homeowners who install HPWHs. Additionally, a federal tax credit of $300 is available. With a ~$1,000 incentive/credit included in the costs analysis, the lifetime cost per year reduces to $168 or $142 for a 10- or 13-yr life expectancy, respectively. Again, this analysis does not include fuel price escalation, so the future cost of fuels will have a major impact on whether this technology is an ideal solution.

There is an interesting inverse effect with HPWHs in the hot-humid climate zone. While the cooling/dehumidification benefits and higher COPs of an HPWH would be more advantageous in the hot-humid climate, mains water temperature tends to be higher. The mains water temperature ranged from 75°–85°F over the monitoring period. This means less water heating is required than in a cold climate, resulting in the overall water heating cost being minimized. Therefore, the cost benefit of the HPWH is slightly diminished.

3.2 Predicted Performance of the Whole-House Dehumidifier

Table 9 presents the total percentage of time that various cooling configurations are predicted to allow indoor conditions to exceed ASHRAE-recommended humidity ratio levels. This table shows the simulated percent frequency that this humidity ratio threshold is exceeded throughout both the entire year and just the summer months of June through August. It is useful to analyze each configuration as both annual and summer only cases, as the dehumidifier will be likely to run year round while the A/C will typically run only in sensible-cooling demand months. The hourly results from the energy simulation suggest that a WHD and A/C control configuration of 55% RH and 78°F, respectively, can adequately manage indoor humidity ratio levels to acceptable conditions for the entire year. Alternatively, to provide comparable (1.44% worse) conditions with an A/C only configuration, the unit's set point would need to be maintained below at least 75°F.

Table 9. Percent Frequency of Conditions > 0.012 lb_m Water/lb_m Dry Air

Control Scenario		> 0.012 lb_m Water/lb_m Dry Air		> 60% RH	
A/C (°F)	Dehumidifier (%)	Entire Year (%)	6/1–9/1 (%)	Entire Year (%)	6/1–9/1 (%)
78	65	17.03	19.59	14.98	17.21
78	60	13.85	15.42	5.48	3.58
78	55	0.00	0.00	0.09	0.00
78	50	0.00	0.00	0.03	0.00
78	off	17.85	19.59	15.98	17.66
75	off	1.44	0.00	15.87	18.25
72	off	0.56	0.00	19.95	31.79
70	off	0.56	0.00	23.40	44.79

Even with indoor air humidity ratios being maintained below the recommended threshold, it is also essential that RH is controlled below ~60%. Table 9 shows energy simulation results of RH maintenance for each cooling configuration during both the entire year and the summer months of June through August. An A/C and WHD at 78°F and 55% RH has the capability of providing sub-60% RH conditions for > 99.91% of the entire year. In contrast, even though an exclusive A/C configuration can control the humidity ratios to acceptable ASHRAE levels, it is not a viable option for satisfactory 60% RH control. A primary reason for this occurrence is that the RH of air will naturally increase as the air temperature decreases—colder air can hold less moisture resulting in higher RH (even with the same humidity ratio).

Figure 8 shows the annual energy costs of each space conditioning configuration. For this analysis, space conditioning is categorized between cooling, heating, and HVAC fan/pump use. In general, each of the acceptable A/C only configurations (that are able to maintained ASHRAE acceptable humidity ratio levels) will generate moderately higher utility costs than their equivalently comfortable combined system counterparts. Table 10 shows the percentage and total cost savings that can be achieved by providing cooling with the combined A/C and WHD system set to 78°F/55% RH system over just the A/C system set to 75°F. More impressively, each of these monetary savings will come as an added benefit to the improved comfort and heightened living conditions that the dehumidifier provides. Note: Columns with titles containing temperature and RH represent combined A/C-WHD configurations and columns titled with temperature only represent A/C only configurations.

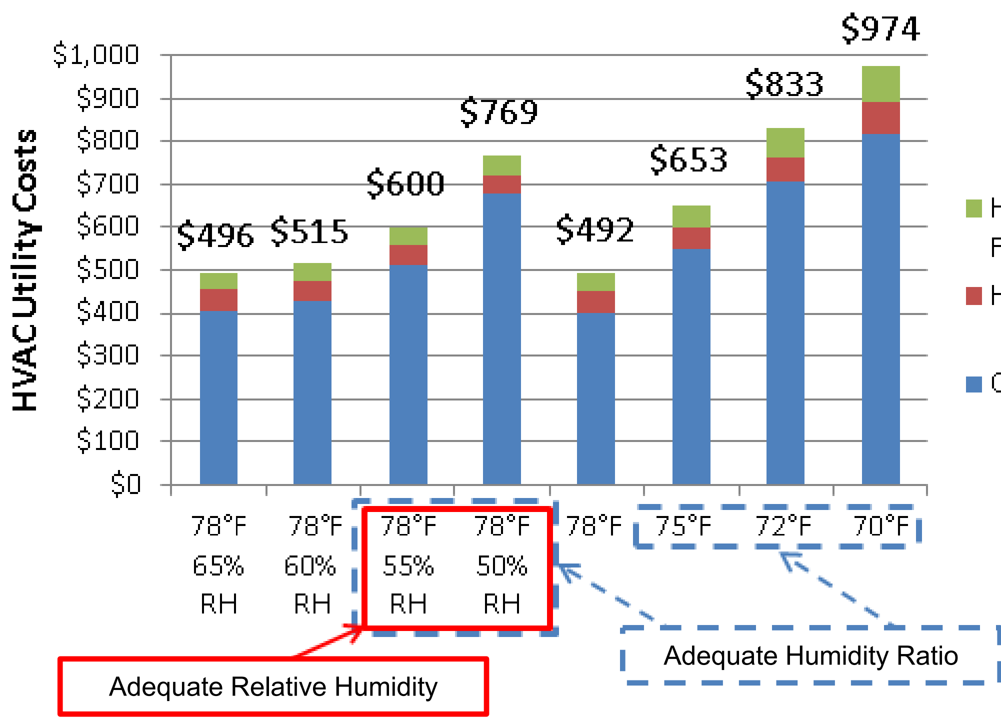


Figure 8. Annualized space conditioning costs of building with various cooling configurations

Table 10. Space Conditioning Savings of 78°F/55% RH Case Over 75°F Case

Space Conditioning	Savings (%)	Savings ($)
Cooling/Dehumidification	6.7	37
Heating	10.2	5
HVAC Fan	21.4	11
Total	8.2	53

- Cooling utility bill savings are a result of the A/C unit not operating as often due to the WHD handling the majority of the latent load.

- Heating utility bill savings result from the dehumidification sensible heat byproduct offsetting some of the heating load requirements.
- HVAC fan utility bill savings are a result of the minimized operation of the AHU related to the reduced cooling and heating requirements.

It is useful to consider these utility costs in a metric that accounts for the moisture-related comfort that they provide. The 78°F and 55% RH arrangement will support acceptable RH levels for 8,752 h/yr while the A/C only case at 75°F will only support acceptable RH conditions for 7,360 h/yr. With this information, a metric that describes the annual building energy cost per number of comfortable hours per year can be derived for each. The combined A/C system with a WHD is predicted to provide comfortable moisture conditions at the rate of $0.21/h, while the A/C system is predicted to carry a rate of only $0.23/h. This suggests that the combined A/C and WHD setup can provide far more moisture-adequate hours per year at a lower annual cost.

As seen in Table 11, the A/C system likely cannot be downsized significantly enough to offset the added cost of the WHD. By optimizing the A/Cs for sensible cooling, it would be possible to reduce each 2-ton heat pump to a 1.5-ton heat pump. The initial cost difference was estimated to be $1,652 ($2,600 for the dehumidifier and –$948 for downsizing the two heat pumps). Based on an 11-yr life expectancy of the dehumidifier, the lifetime cost is 60% higher for the combined A/C-WHD configuration. This cost analysis doesn't account for the added comfort associated with complete (temperature and RH) space conditioning control provided be the combined A/C-WHD configuration.

Table 11. Combined A/C and WHD Cost Analysis

Cooling Configuration	Initial Cost ($)	Lifetime (yr)	Additional A/C ($/yr)	Lifetime Cost ($)
A/C only (75°F)	–	11	54	589
A/C-WHD (78°F/55%)	1,652	11	–	1,652

4 Results From Field Monitoring

4.1 Heat Pump Water Heater

Long-term monitoring was performed to assess the on-site performance of the HPWH unit in an occupied setting. Table 12 shows a data summary of the performance results that the unit achieved between June 22, 2012 and October 31, 2012. It is important to consider that this data summary provides results for only one of the two HPWHs that supplied DHW needs to the house (though it was the primary water heater for the bathrooms). As seen from the data summary, the unit effectively provides DHW via the heat pump for the majority of the needs of the house while only deferring 23% of energy use (not thermal fraction) toward the electric resistance heating elements.

Table 12. HPWH Monitored Performance Summary (June 22, 2012 to October 31, 2012)

Hot Water Set Point	120°F
Average Water Inlet Temperature[1]	82°F
Average Water Outlet Temperature[1]	117°F
Total COP	2.2
% Electric Resistance[2]	23%
Average Hot Water Use	48.8 gal/day
Average Inlet Air Temperature	74°F
Average Inlet Relative Humidity	55%

[1] Average estimated with 15 min. periods containing near-continuous flow.
[2] % electric resistance = % of total kWh consumed by resistance.

Figure 9 shows the daily COP of the HPWH plotted against the measured water draw for the day's period. As seen from the plot, there are a number of days where a large electric resistance load was present and as a consequence, the unit operated with a lower COP value. The system switched over to electric resistance heating due to large volumes of hot water being drawn over a short period of time. Still the efficiency for days when the electric resistance percentage exceeded 4% was higher than a standard ERWH. Additionally, standby losses from the tank and piping also have an effect on the HPWH unit's performance. This impact is especially pronounced at low daily hot water consumption (< 20 gal/day).

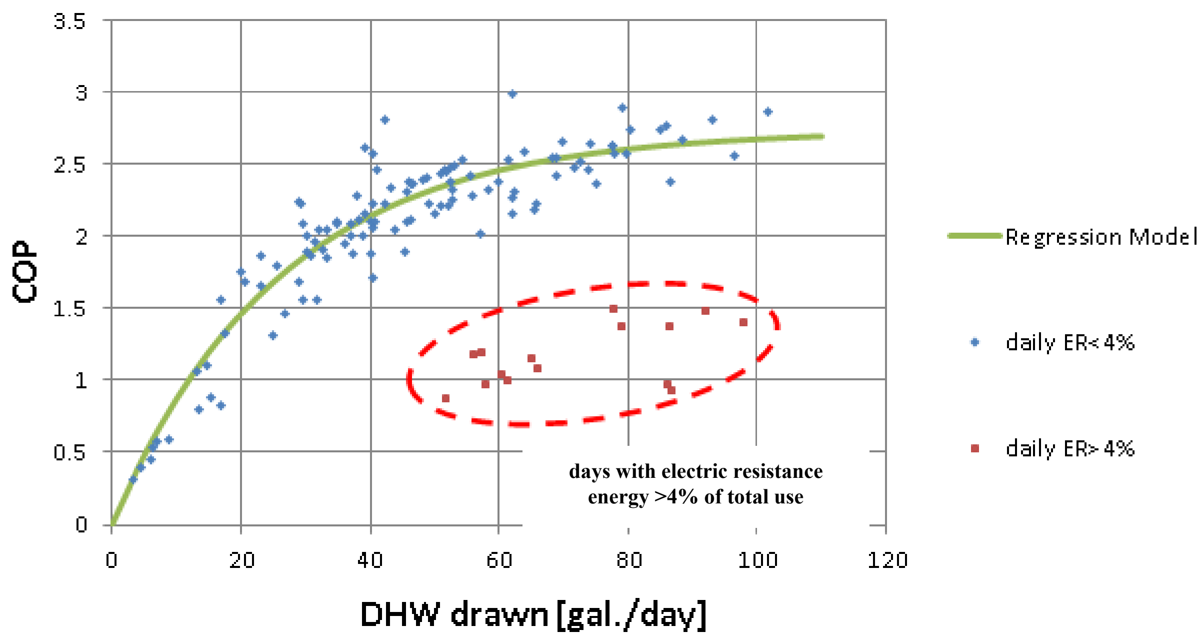


Figure 9. HPWH COP versus water draw

The unit operated with an average COP of 2.2, while achieving a maximum COP of 3.0 over the course of the monitoring period. In comparison, the manufacturer-rated EF for this HPWH is 2.35. This difference is not surprising because the EF is calculated under lab conditions with an unrealistic draw profile, while our monitored data are specific to the conditions and loads observed. However, on average, this HPWH operated at a 144% higher efficiency than the rate efficiency of a comparable ERWH (with a COP of 0.9) despite the fact that the hot water loads are smaller than the EF test loads.

To quantify the performance of the heat pump based on hot water demand, the daily COP of the unit was plotted against the total water drawn for that day. To develop a performance model for instances where the unit was running in heat pump mode, days with > 4% electric resistance total energy consumption were excluded from the regression. A performance curve for the HPWH was fitted using the exponential form of Equation 1.

$$COP = a\left(1 - e^{bV}\right) \tag{1}$$

The fitting coefficient *a* represents the COP of the tank without thermal losses ($COP_{adiabatic}$). The fitting coefficient *b* represents the losses of the tank as a function of water draw. The variable *V* corresponds to the total volume of water drawn for one day. The fitting coefficients that described the performance of this General Electric HPWH unit are displayed in Table 13.

Table 13. COP Regression Fitting Coefficients

Fitting Coefficients	
a	2.73
b	–0.038

The hot water draw profile is a primary factor on the unit's performance. The average draw profile for this unit over the monitoring period is provided in Figure 10. The draw profile is compiled as the average water drawn during each 15-min time period starting at 12:00 a.m. each day. Since there is a often a significant water draw at midnight, daily COP values were calculated starting at 3:00 a.m. and going to 3:00 a.m. the next day to avoid major recovery periods extending into the next day averages.

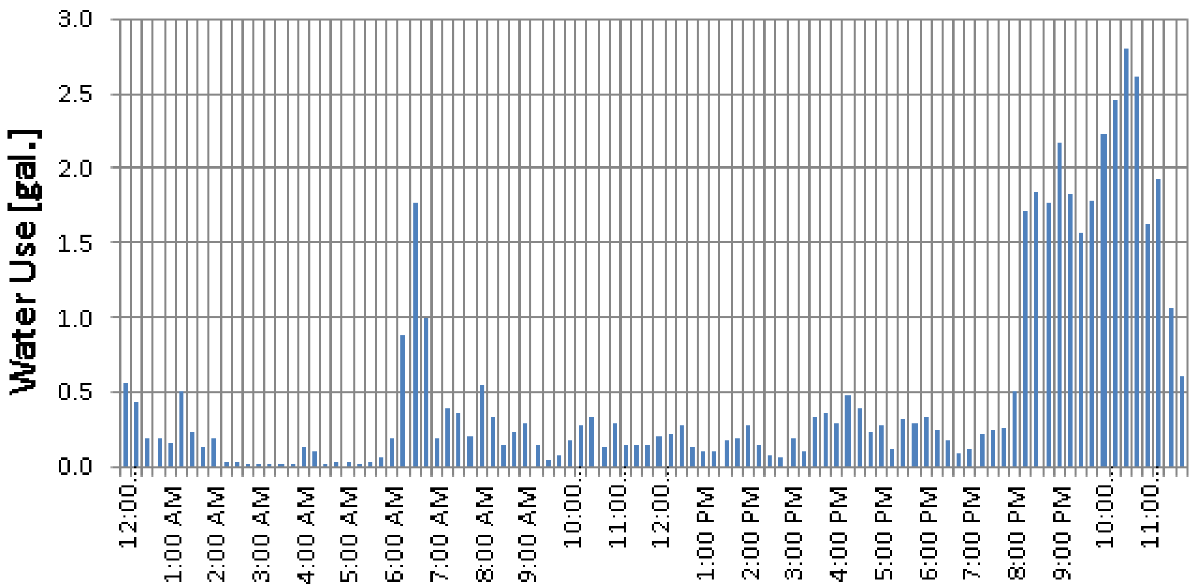


Figure 10. HPWH average water draw

The water draw profile of this occupied home is shifted more to the evening hours. Figure 11 provides a comparison of the draw profile of this occupied home, the Building America normalized draw profile (for a water heater servicing four bedrooms and three bathrooms with weather data from the Orlando International Airport location), and the draw profile used for the EF rating test. Variations in draw profiles are the hardest attribute of HPWHs to model as they are highly dependent on individual homeowner's behavior. Even though these hybrid water heaters provide hot water regardless of homeowners' usage patterns, the efficiency of these units is greatly impacted based on the total quantity and frequency of hot water draws.

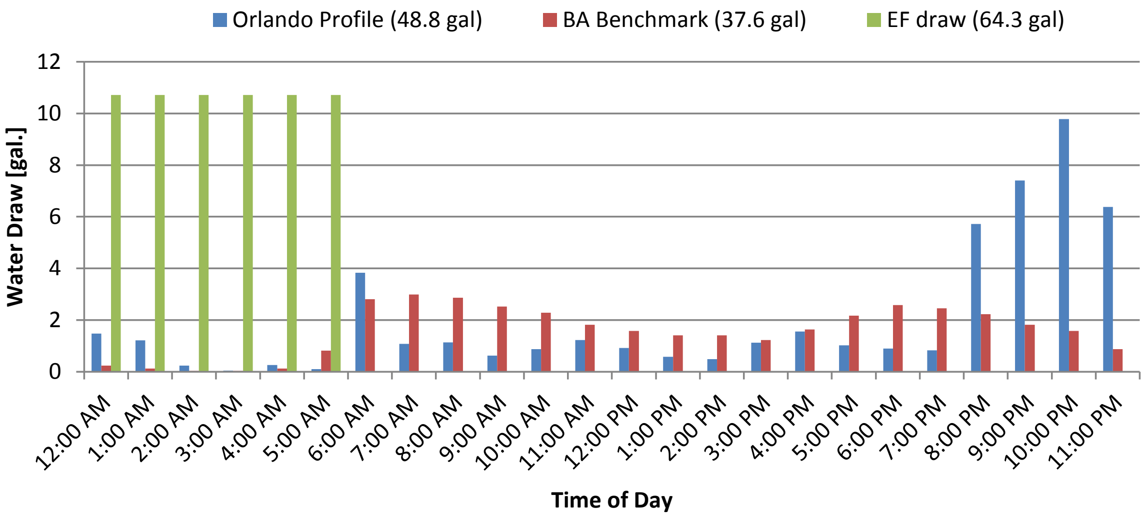


Figure 11. Comparison of various DHW draw profiles

The HPWH unit will call upon electric resistance heat when the heat pump mode cannot supply heat to the water at a fast enough rate to replenish the hot water being drawn from the tank. As a result, when large quantities of water are drawn in short period of time, the heat pump cannot transfer enough heat to the tank to support the high demand. Figure 12 shows the water draw profile in which 97.7 gal were drawn over the course of the day and the unit performed with a COP of only 1.4. This low COP was largely influenced by the high electric resistance energy use between the hours of 9:30 p.m. and 10:30 p.m. The HPWH required the use of the electric resistance elements in order to recover from the series of hot water draws that occurred shortly before (8:45 p.m. to 10:15 p.m.). Note: Hot water draws are shown as gal/15-min periods.

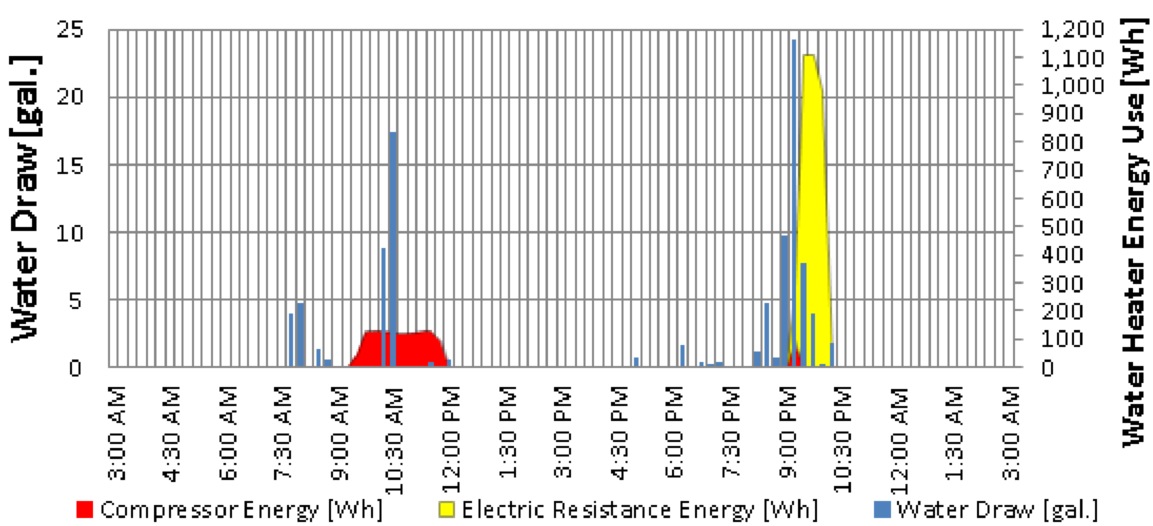


Figure 12. Water draw profile resulting in low daily COP (COP = 1.4)

If the draw profile follows a more distributed pattern of use throughout the day, rather than high concentrations of use in short periods of time, it is likely that the HPWH will be more able to support the entire DHW load with heat pump operation. As a result of avoiding the use of electric resistance elements, a higher COP can be achieved. The unit will not need to revert to electric resistance as a supplementary heat source; it will be able to comfortably replenish heat to the tank with the vapor compression cycle alone. Figure 13 displays the water draw profile and energy consumption for an alternative day in which the total water consumption was only 1.3 gal less than the previous "low COP draw profile case." However, due to the spaced out water draw pattern, the HPWH was able to operate with a daily COP of 2.6—which is more than 85% more efficient than the previous case's daily COP of 1.4. Even though roughly the same daily amount of hot water was used, the heat pump was able to replenish the tank's hot water without utilizing the electric resistance elements. As a result, all of the heat transfer to the tank was accomplished through the heat pump and more efficient performance operation was achieved. This comparison speaks to the significant impact of load patterns as well as the sensitivity of the HPWH's control logic.

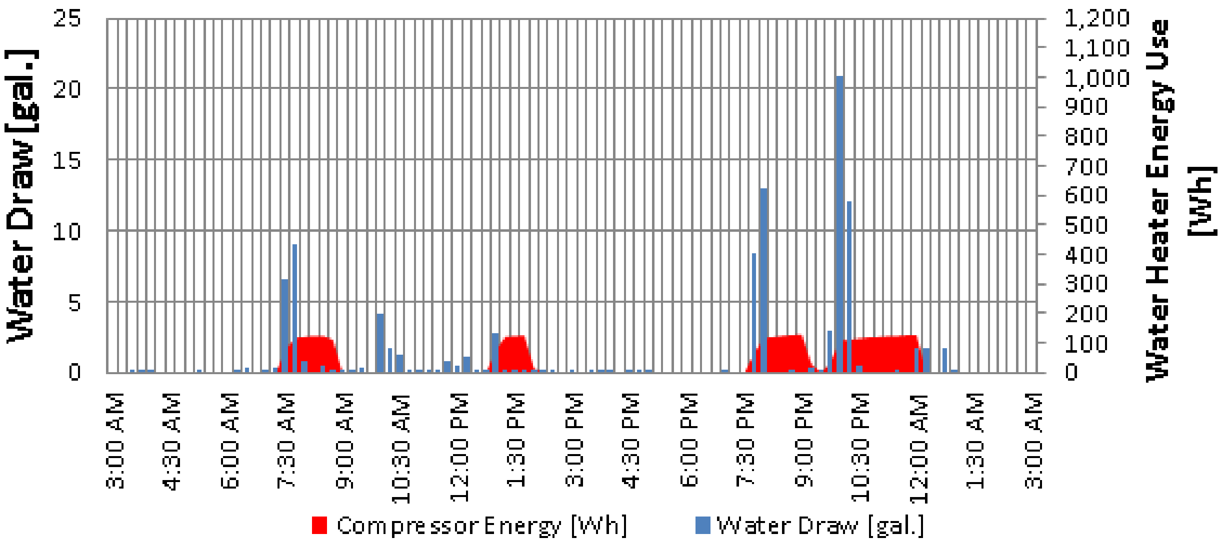


Figure 13. Water draw profile resulting in high daily COP (COP = 2.6)

Figure 14 displays the water draw profile and energy consumption from a low total water usage day. On this day, the homeowner used only 3.3 gal of hot water over the course of the entire day. The day contained a series of very small draws, with none of them exceeding more than 1 gal over the course of a 1-h period. As a result, the unit was able to operate with a daily COP of only 0.3. As seen from Figure 14, the heat pump cycle runs for a period of time even though there is essentially insignificant amount of hot water drawn over the course of the day. This pattern indicates that the heat pump is running to replenish heat that is lost into the surrounding space through the tank walls, referred to as *standby losses*.

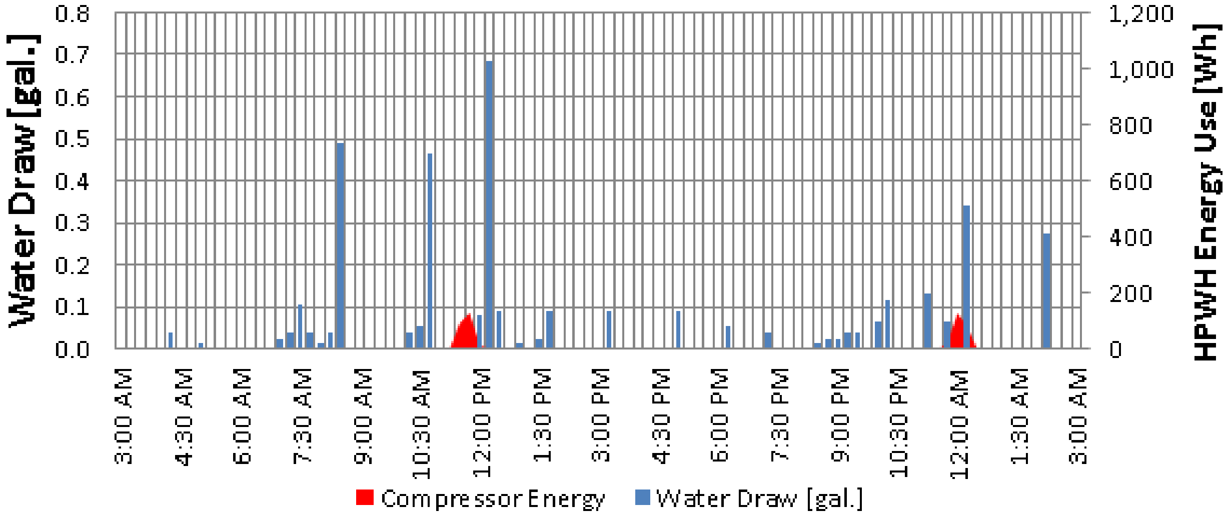


Figure 14. HPWH performance with minimal water draw (COP = 0.3)

An HPWH also affects the temperature and humidity of the ambient air surrounding the unit. As the ambient airstream passes through the HPWH at a rate of roughly 175–200 CFM, the air temperature is decreased and moisture is extracted through condensation. Figure 15 displays the change in temperature that occurred during typical HPWH operation. On average, the airsteam temperature was decreased from ~75°F to ~60°F. In a hot-humid climate, such as Orlando, this will serve as an added benefit for a majority of the year.

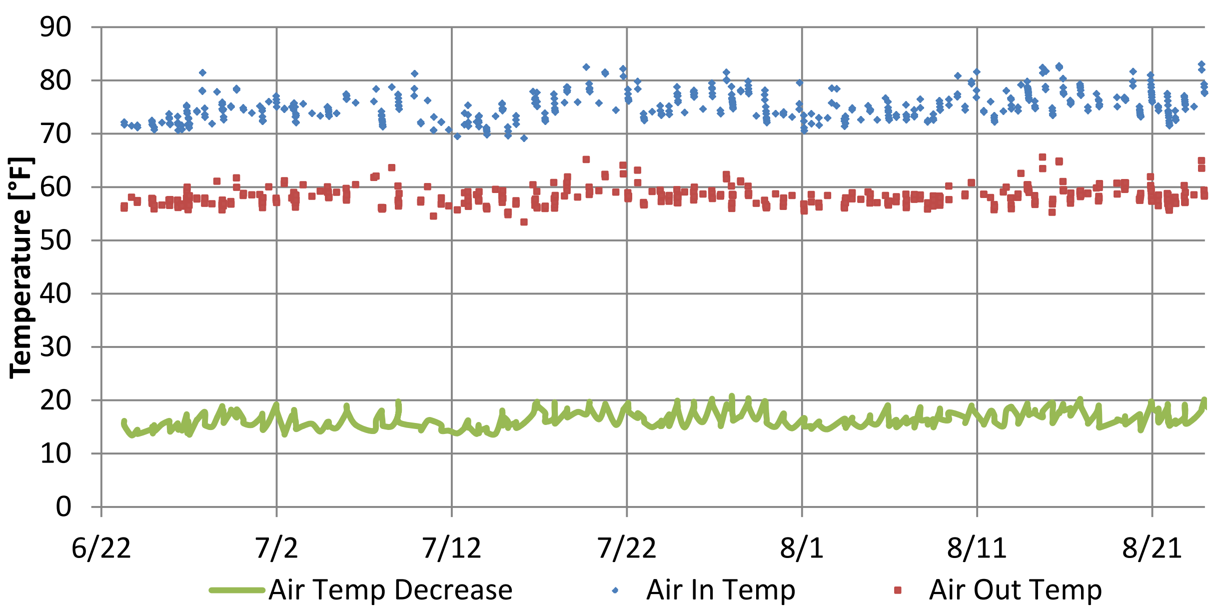


Figure 15. Air temperature change through HPWH

In addition to temperature change, humidity properties of the airstream are also affected. Figure 16 shows the change in humidity ratio that occurred as air passed through the heat pump. As moisture is extracted in the form of condensate on the HPWH evaporator coil, the humidity ratio of the airstream decreases from ~0.011 to ~0.009 lb_m water/lb_m dry air. Even with moisture being removed, the RH of the airstream increases due to the decrease in air temperature (from ~60% to ~90%). As air temperature decreases, it possesses less of an ability to hold moisture. Although the outlet airstream has a higher RH, once this air mixes with the surrounding air, the lower moisture content of this air will result in dehumidification of the surrounding space.

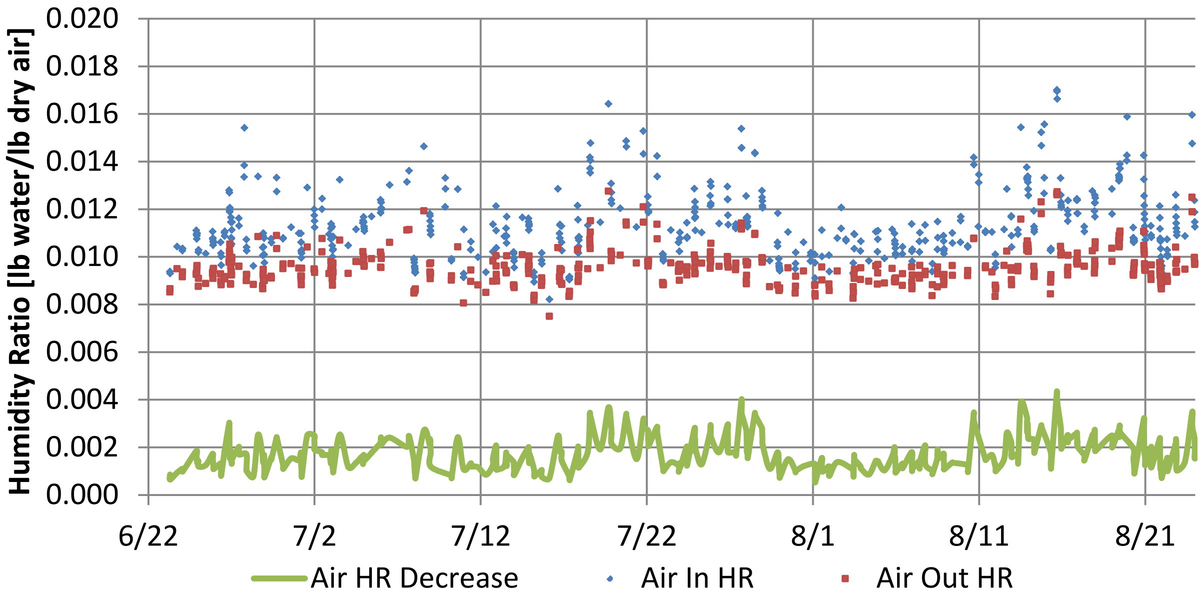


Figure 16. Humidity ratio change through the HPWH

The impacts of the HPWH on the space conditioning systems depend heavily on climate, home configuration, HPWH location, and the space conditioning systems used. Analysis of the air enthalpy change through the HPWH showed that cooling capacity averaged 5,540 Btu/h (with a maximum of 7,696 Btu/h) when the unit was operating continuously over a 15-min logging period. This cooling capacity is sufficient to impact the room in which the HPWH is located. Many manufacturers are offering ducting kits (typically less than 10 ft combined between return and supply ductwork) to allow designers to better integrate HPWHs into the control of the space conditioning load of the home.

Figure 17 displays a bar graph of the volume of the condensate removed for daily periods. During the case of maximum moisture removal, the HPWH is responsible for extracting more than 4.5 pints/day (or ~4.9 kBtu of latent heat removal). However, this number is somewhat deceptive toward relaying the HPWHs dehumidification capabilities since the unit is operating in a home that is already being dehumidified by a WHD.

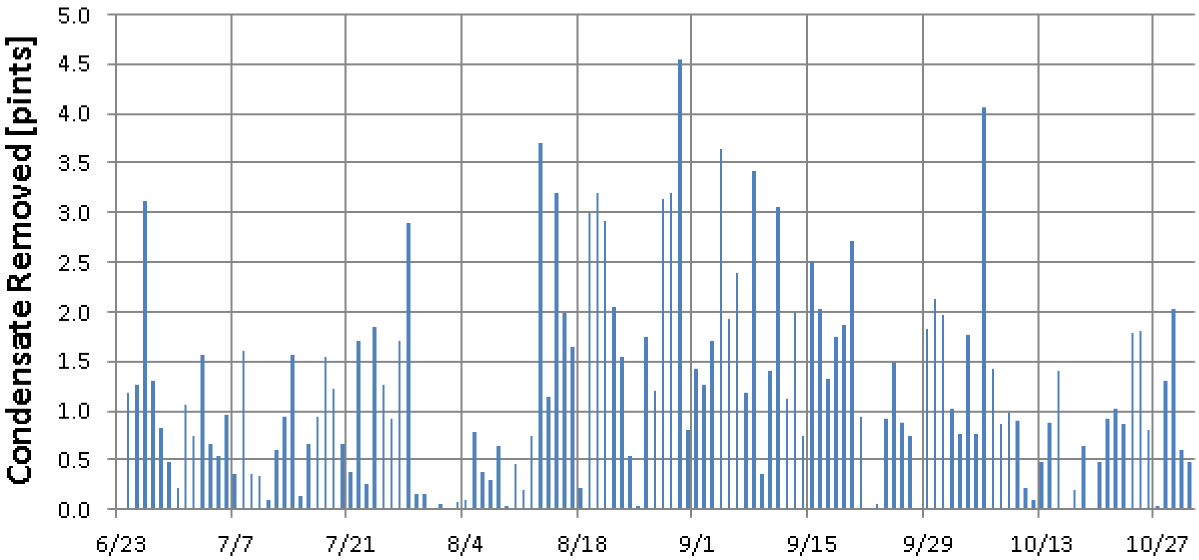


Figure 17. HPWH condensate removal

Since the HPWH is operating only while there is a demand to replenish hot water, a "pints/day" metric provides only small insight into the latent performance of the unit. Figure 18 shows a scatterplot that describes the "pints/kWh" removal rate of the unit on a daily basis. The best fit line resulting from a linear regression fit (though the coefficient of determination is not strong) reveals that this unit was operating at an average removal rate of 1.17 pints/kWh.

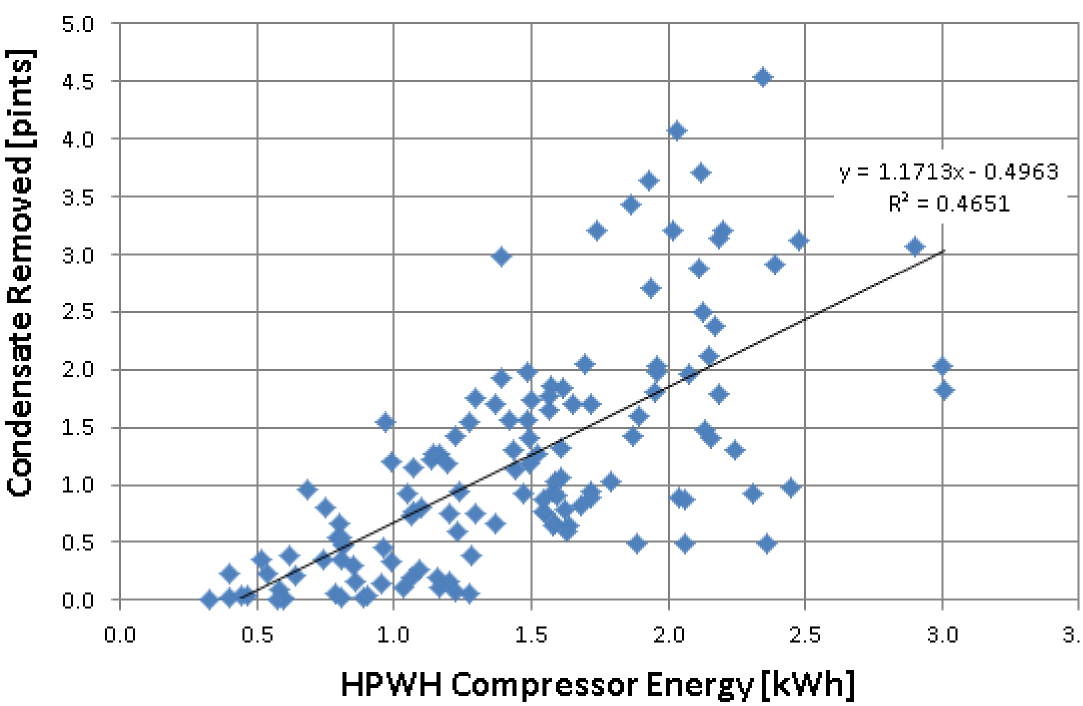


Figure 18. HPWH condensate removal performance

4.2 Whole-House Dehumidifier

Initial power measurements for each of the mechanical systems, which are responsible for controlling the sensible and latent cooling load in the house, are shown in Table 14. In this test home, the WHD draws approximately 830 W during normal operation, while the two central A/C systems draw a combined load of 3,330 W. As seen from the power consumption values, the home's A/C system uses more than four times as much power as the WHD, so minimizing the A/C system operation as much as possible could be advantageous in terms of energy savings.

Table 14. Power Draw of HVAC System Components

Mechanical System	Power (W)
WHD	830
Condenser Unit 1	1,460
Condenser Unit 2	1,460
AHU 1	173
AHU 2	237

As seen from Table 14, there is a difference in power consumption between the two AHUs. The higher power draw in AHU 2 can be attributed to greater static pressures that exist in this system's duct configuration. As air resistance in the ducts increases, the AHU's electronically commutated motor compensates by increasing power consumption to maintain the desired airflow rate.

In order to quantify the latent heat removal efficiency of each of these units, it is useful to look at their performance when running at maximum capacity. The energy performance quantifies the volume of condensate that is removed from the airstream per kWh of dehumidifier power consumed by the fan and compressor. The product capacity denotes the maximum amount of condensate that the unit can remove (pints/day) while running continuously for 24 h. According to the manufacturer's product specification sheet, the Honeywell DH150 WHD has a capacity of 150 pints/day and an energy factor of 3.56 L/kWh (or 7.51 pints/kWh). These metrics are determined under lab conditions with a constant 80°F, 60% RH supply airstream.

Even though standard A/C units are often used for moisture removal purposes, manufacturers do not regularly publish their rated latent energy performance (pints/kWh) and moisture removal capacity (pints/day). Rather, the manufacturer will publish a table of sensible-to-total capacity (S/T) ratios that describe A/C performance under various conditions. The expanded performance table for the Lennox AHU and condensing unit that were used in this study were used for this analysis.

In order to compare A/C latent energy removal rates with published dehumidifier latent rates, the potential A/C moisture removal performance was calculated at similar conditions to the WHD's performance data. The equivalent capacity metrics for the two 2-ton A/C systems combined are displayed in Table 15. For these calculations, the S/T ratio was selected under conditions of outdoor dry bulb of 85°F, AHU flow of ~720 CFM, entering dry bulb of 80°F, and entering wet bulb of 67°F.

The energy performance (pints/kWh) was determined by calculating total latent capacity (at an S/T of 0.74) and assuming the average heat of vaporization (at ~75°F) to be 1,050 Btu/lb_m. Power consumption values from initial HVAC measurements were used to determine kWh usage of the equipment over a continuous run scenario of a 1-h period. The condensate capacity is calculated as the volume of condensate that could potentially be removed if the system were running at maximum capacity for 24 h.

Table 15. Equivalent A/C Capacity Metrics

Equivalent A/C Capacity Metrics	
Total Capacity	48,000 Btu/h
S/T ratio	0.74
Condensate Capacity	274 pints/day
Energy Performance	3.42 pints/kWh

As seen from calculated values, the A/C system's condensate capacity of 274 pints/day is almost double the dehumidifier's capacity of 150 pints/day. Even though the A/C may theoretically be able to extract a greater volume of condensate, it will be removed at a less efficient rate. The unit's calculated EF of 3.42 pints/kWh is significantly lower than the WHD's published value of 7.51 pints/kWh. As a result, the A/C carries a condensate removal energy performance that is 54% lower than the WHD's when operating at capacity. It is important to note that these energy metrics will be much smaller for daily use as typical outdoor conditions will not generally require the system to run at maximum capacity for entire-day spans. Overall, these results suggest that specifically controlling the latent cooling load with a WHD and addressing the sensible cooling load with a central A/C unit can maximize cooling system efficiency.

Long-term performance monitoring of these systems was also carried out in order to examine their operation in an installed environment. For a monitoring period between mid-June and mid-November, the home's indoor set points were maintained with the A/C thermostat at 76°–78°F and the dehumidifier set to 65% RH. Indoor temperature and RH levels were monitored in an upstairs living room for the entirety of this period. Figure 19 displays the frequency distribution of RH occurrences. The RH within the home was maintained at levels < 60% for 99.86% of the monitored period.

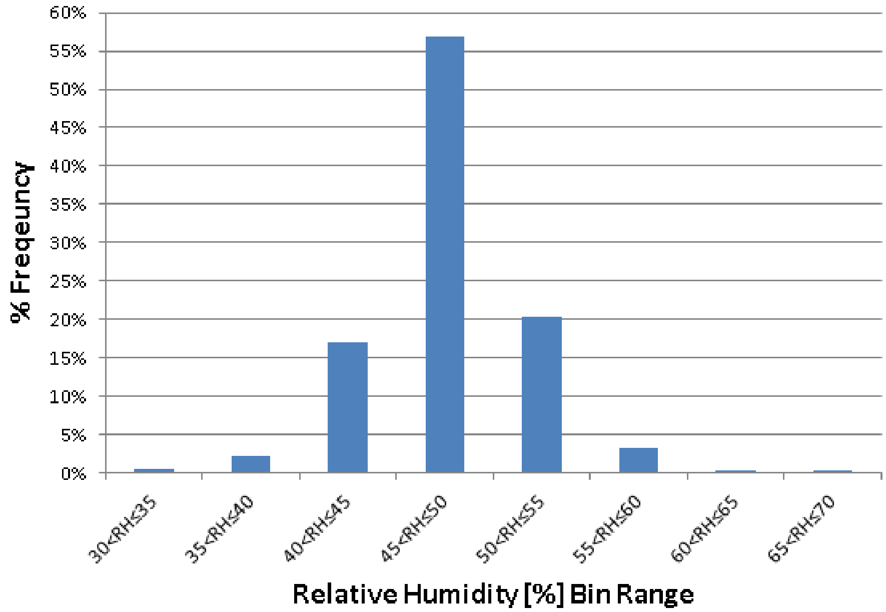


Figure 19. RH frequency distribution chart

Maintaining adequate temperature levels within the building environment is equally important for achieving human comfort. Figure 20 shows the frequency distribution of temperature levels maintained within the second-floor living room of the house. The ASHRAE summer comfort zones suggest human comfort at summer conditions can be achieved at higher temperatures as long as RH is maintained at a lower level. While keeping adequate indoor RH levels (< 60%), the home's indoor temperature was kept within the comfort zone (~74°–80°F) for 91.5% of the data monitoring period.

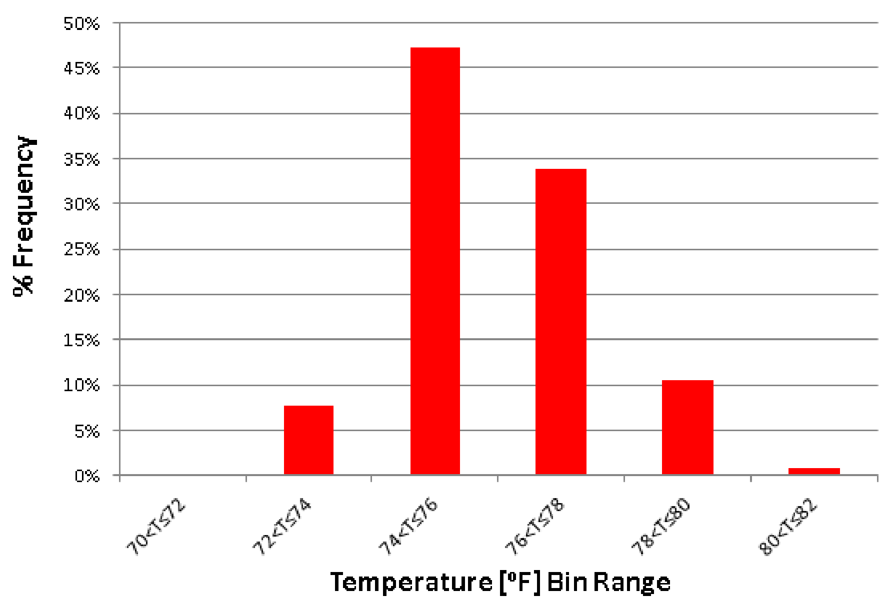


Figure 20. Temperature frequency distribution chart

It is important to note that a majority of the occurrences where indoor temperature was beyond the comfort design range was during times when the outdoor temperature was colder than A/C set point requirements. During the monitoring period, outdoor temperature dipped to as cold as 45°F on some nights. Therefore, since the A/C would not be called on at this temperature, the sub-74°F indoor conditions should not be directly attributed to the A/C running more frequently than needed in order to maintain RH. Therefore, the home was kept within the comfort zone temperature range for 99.3% of the cooling portion of the data monitoring period. Overall, these temperature data support the idea that the presence of a dehumidifier allows the A/C to be fixed at a higher set point. As a result, the A/C is no longer required to run more often (in order to address latent loads) than required to achieve the desired indoor temperature.

The total volume of condensate removed was monitored to assess the moisture-removal effectiveness of the dehumidifier. The maximum condensate volume removed over the course of the monitoring period was 96 pints in one day. This high condensate removal occurred during a period of atypical conditions with long periods of heavy rainfall. If an A/C had been solely responsible for removing this amount of moisture, it would have needed to run for roughly 35% of the day to pull the same amount of moisture out of the air. Unless sensible cooling was also required, this could have resulted in the space overcooling.

On November 12, 2012, a water alarm sensor that SWA installed as a precautionary measure was tripped, suggesting that water was leaking onto the wood platform that the dehumidifier and drain pan sat on. The condensate pipe connection to the dehumidifier was leaking and condensate was dripping into the drain pan. This was immediately corrected by the contractor. During the data analysis, it was estimated that the leak may have started as early as August 24, 2012 and continued to worsen over time until our water alarm was triggered. Unfortunately, we were measuring the condensate only through the condensate pipe and did not plumb the drain pan to our condensate measurement device.

Figure 21 shows three distinct time periods of our monitoring. The red data points show that even though the dehumidifier is running, we were not capturing the condensate that was being produced. This was confirmed by evaluating the enthalpy differential during operation based on airside measurements. The enthalpy differential remained fairly consistent over these three periods, so the condensate data from August 24, 2012 through November 5, 2012 were ignored.

The manufacturer rated this dehumidifier with a moisture removal rate of 7.51 pints/kWh (at 80°F and 60% RH inlet air). Figure 21 shows actual moisture removal performance metrics achieved throughout the monitoring period. A linear regression fit of the pre-August 24 places the average performance slope at 4.95 pints/kWh. This is lower than the manufacturer's rated performance, but as the rated performance is at a single condition, it is not surprising.

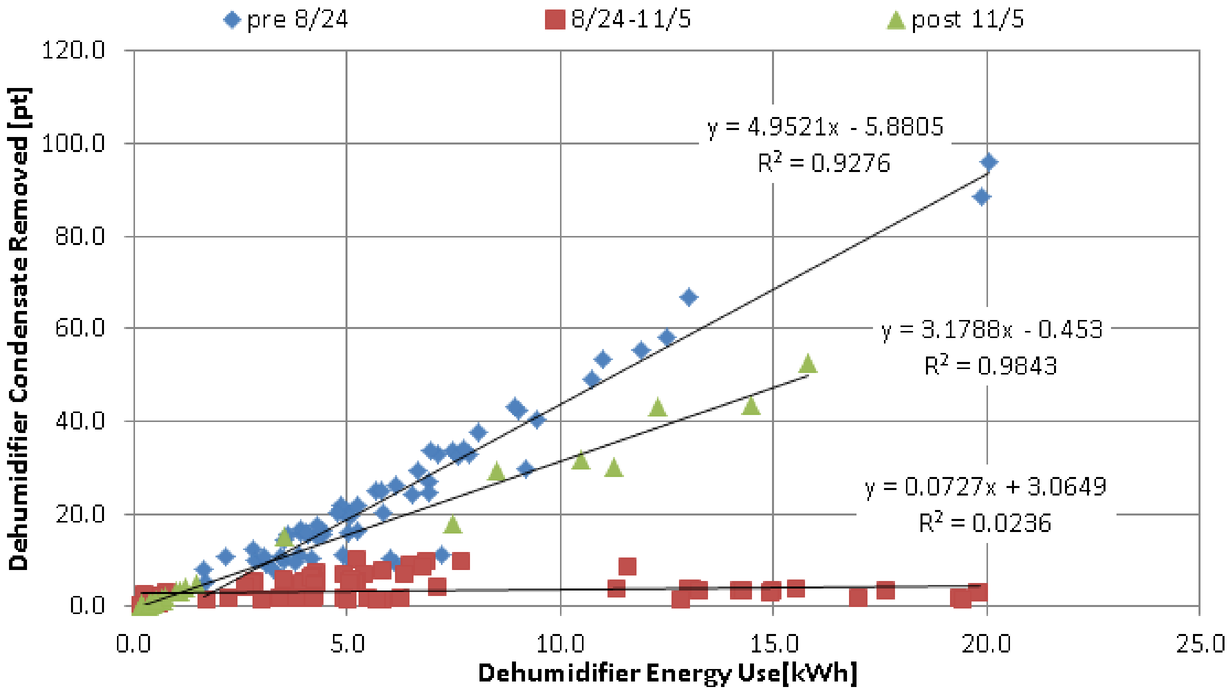


Figure 21. Dehumidifier condensate removal versus energy performance

The immediate goal of the dehumidifier is to extract moisture from the air and remove it from the confines of the building envelope. The RH of indoor air can be an effective indicator of moisture-related comfort conditions. Figure 22 shows that the dehumidifier reduces humidity ratio as the mixture of fresh outdoor air and building return air passes across its evaporator coil. As condensate is removed from the airstream, the humidity ratio of the airstream decreases from ~0.011 to ~0.008 lb_m water/lb_m dry air. The RH of the airstream is reduced from ~55% to ~20%. However, this RH decrease is attributed to the temperature increase that occurs as heat is transferred from the condenser coil. Essentially, the dehumidifier is converting latent heat energy contained in the moist inlet stream to condensate and sensible heat in the warmer-dryer outlet airstream. Figure 23 displays the effect that the dehumidification process has on the temperature of the airstream. On average, the mixed inlet air, with a temperature of ~75°F, exits the unit at an increased temperature of ~95°F.

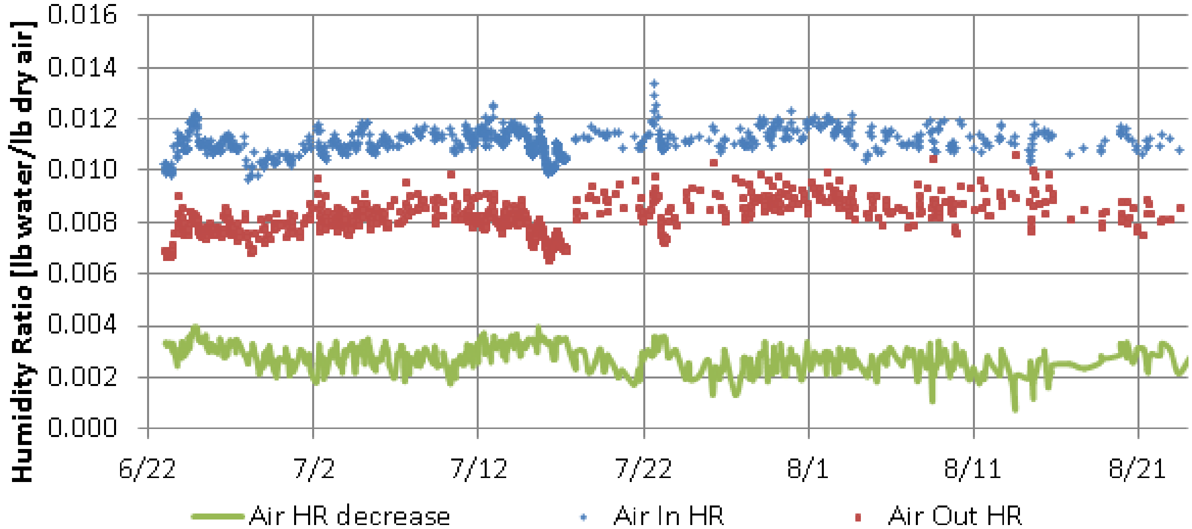


Figure 22. Humidity ratio change of airstream through dehumidifier

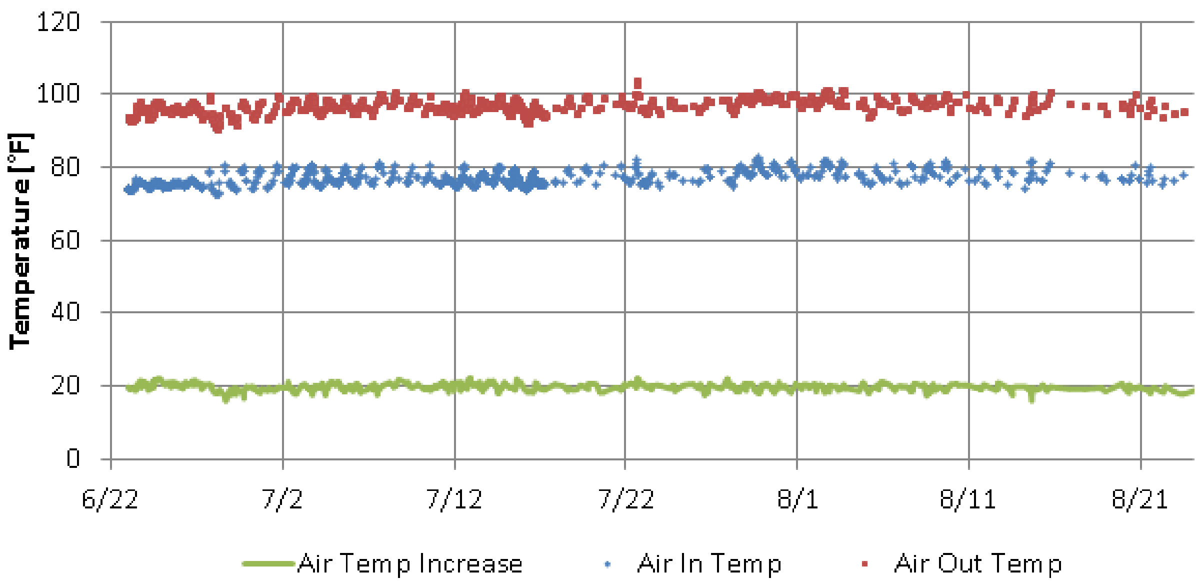


Figure 23. Temperature change of airstream through dehumidifier

From November through April, half of the year, this temperature increase can serve as a small added benefit to the moisture removal process. Heat that may have been otherwise required to be supplied by an additional source (furnace, heat pump, etc.) is acquired as a "free" byproduct of the WHD's operation. However, during summer months, the central A/C system will need to address this sensible heat gain.

Sensible, latent, and total heat changes that occurred as air passed through the dehumidifier were calculated for the entire monitoring period. Heat removal values were calculated using the differential enthalpy changes that resulted as a function of temperature increase and RH decrease. Figure 24 shows the daily totalized heat transfer into and out of the airstream over the monitoring period through August 24. It is important to note that these calculations serve only as estimates of the true heat transfer effects that occur in the dehumidification process. The sum of sensible heat gain over the entire period equates to approximately 25% of the total sensible cooling load predicted by energy simulations. The estimates seem fairly reasonable with the heat of vaporization accounting for roughly 50% of the sensible heat gain.

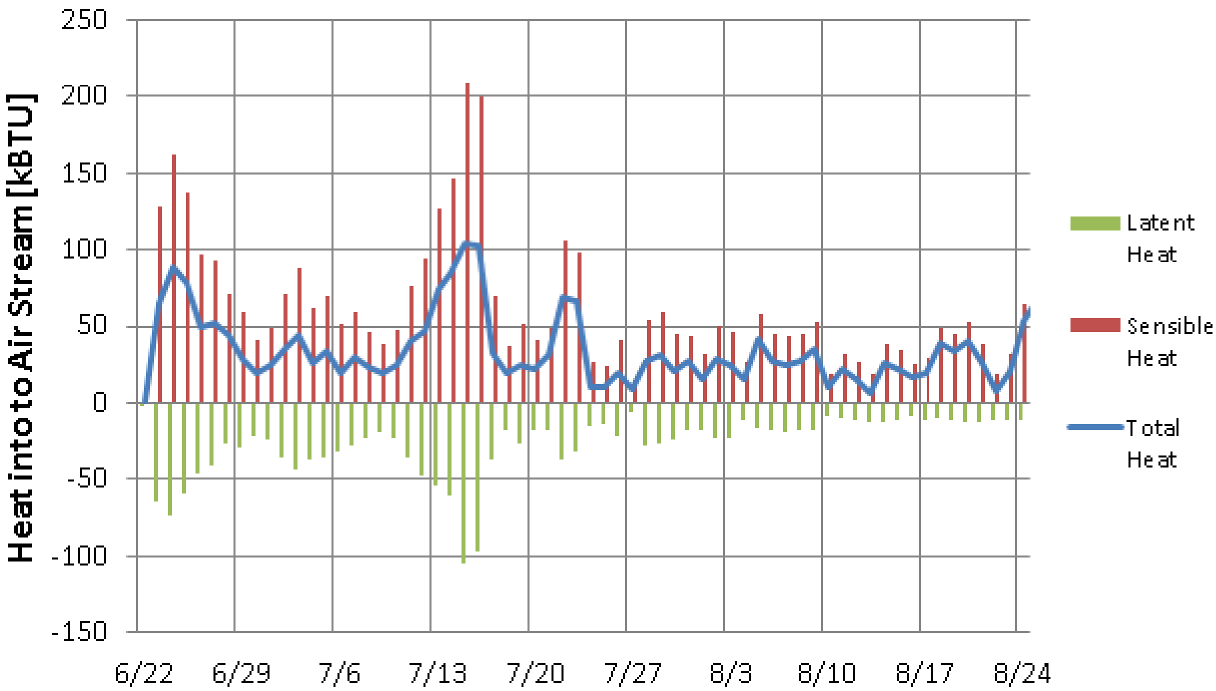


Figure 24. Daily heat into dehumidifier airstream

Additionally, short-term scenario testing was performed during November and December 2012 to evaluate how well various set point configurations of the combined A/C and WHD system fared. Three short-term testing cases, each 4–10 days in total length, were run and the system's performance and home's comfort conditions were continuously monitored. Table 16 shows various control configurations of A/C and WHD that could be used to provide comfortable conditions within a home. This table shows the total monitoring time of each scenario (days), total energy used by the A/C system and WHD (kWh), average outdoor conditions during the period (°F and % RH), and the daily energy to provide cooling and dehumidification to the building (kWh/day). It is important to note that the cooling energy rate is not normalized per outdoor temperature conditions and serves only as an estimate of cooling energy use per day. In general, WHD and AHU runtime is often highly dependent on factors other than outdoor temperature and humidity. Indoor temperature and RH levels are influenced strongly by solar heat gains and internal heat gains (occupants, cooking, lights, appliances, etc.). Therefore, weather normalization for a cooling-dominated climate does not always provide an apples-to-apples comparison between varying outdoor conditions. As a result, it was not performed for this section of the analysis.

Table 16. Scenario Test Summary

Test Scenario	78°F, 50%	75°F, Dehumidifier Off	78°F 60%
Total Time (days)	8.0	4.0	10.3
Dehumidifier Energy (kWh)	72.80	0.00	10.59
AHU #1 Energy (kWh)	1.85	1.24	4.06
AHU #2 Energy (kWh)	4.88	3.87	1.79
Condenser #1 Energy (kWh)[1]	15.57	10.49	34.29
Condenser #2 Energy (kWh)[1]	30.05	23.85	11.05
Average Outdoor Air Temperature (°F)	60.88	63.02	67.93
Average RH (%)	73.31	77.45	81.75
Dehumidifier Energy Rate (kWh/day)	9.10	0.00	1.02
A/C System Energy Rate (kWh/day)[2]	6.54	9.76	4.95
Cooling Energy Rate (kWh/day)[3]	15.60	9.76	5.97

[1] Assumed condenser power draw of 1.46 kW while AHU was operating
[2] Combined AHU energy use and condenser energy use
[3] Combined A/C system energy use and dehumidifier energy use

Cooling energy rates for each of the scenarios support the same configuration ranking that was generated in the BEoptE+ hourly simulation models. Both scenario testing results and energy modeling simulations rank the scenarios in the following order (from most energy consumptive to least energy consumptive): (1) 78°F and 50% RH; (2) 75°F and dehumidifier off; and (3) 78°F and 60% RH. Even though the model and scenario tests are in agreement, it is important to recognize that the energy modeling is simulating usage over the course of an entire year, while the scenario test only accounts for a short time period during November/December 2012. Thus, it is important to mention that A/C energy use would be greater during summer months when a higher sensible cooling demand is needed. However, these scenario test results effectively demonstrate that a WHD can be effective and efficient in controlling latent loads during swing seasons in hot-humid climates. Appendix B of this report contains detailed plots of the indoor and outdoor data conditions during each of the scenario tests.

Occupant evaluations are generally the best indicators of true comfort, since comfort is truly a subjective metric. Following several months of living with the combined A/C and WHD system, the homeowner provided an unsolicited evaluation of the comfort improvement that took place in the home. The homeowner stated, “We are happy with the systems and surprised that we can set the thermostat so much higher than before and still be very comfortable.” This statement supports the notion that a home that addresses sensible cooling loads with an A/C and primary latent cooling loads with a WHD can maintain or improve indoor comfort levels while saving energy.

5 Conclusion

The results of this energy modeling and field monitoring study show support for the use of HPWHs as well as combined A/C and WHD cooling systems in hot-humid climates. The HPWH was able to achieve high COPs while providing some space conditioning benefits to the home. Additionally, by adding a WHD to the home's cooling system, the home was able to achieve improved indoor comfort while saving energy.

5.1 Heat Pump Water Heater

What is the expected efficiency of an HPWH located within the conditioned volume of hot-humid climate homes?

The HPWH operated at an average COP of 2.2 over the monitoring period. Performance trends indicated that the efficiency of the unit was primarily a function of the volume of DHW drawn and the distribution pattern of each draw. The optimal draw pattern (for the highest COP) includes draws that are spaced just within the heating rate limit of the heat pump, such that the heat pump can replenish all hot water to the tank before the next significant water draw. These performance conclusions will vary with differences in climate (cold water inlet temperature effects and evaporator inlet air), unit location, and hot water loads.

There is an interesting inverse effect with HPWHs in the hot-humid climate zone. While the cooling/dehumidification benefits and higher COPs of an HPWH would be more advantageous in the hot-humid climate, mains water temperature tends to be higher. The mains water temperature ranged from 75°–85°F over the monitoring period. This means less water heating is required, resulting in the overall water heating cost being minimized. Therefore, the cost benefit of the HPWH is diminished.

What types of space conditioning implications are associated with utilizing an HPWH in the conditioned space of the home?

The impact of a HPWH on the space conditioning systems rely heavily on climate, home configuration, HPWH location, and the space conditioning systems used. In hot-humid climates, the HPWH's cooling and dehumidification byproduct is desired for a large portion of the year. During operation, the HPWH affected the airstream in the following ways: (1) decreases temperature from 75°F to 60°F; (2) increases RH from 60% to 90%; and (3) decreases humidity ratio from 0.011 to 0.009 lb_m vapor /lb_m dry air. Analysis of the air enthalpy change through the HPWH showed that cooling capacity averaged 5,540 Btu/h (with a maximum of 7,696 Btu/h) when the unit was operating continuously over a 15-min logging period. For this test home, the unit operated with an average heat removal rate of 16,266 Btu/day. Based on power consumption of this home's A/C system running at capacity (4 ton) and local utility rates, this heat removal could save the homeowner ~$45/yr. This cooling capacity is sufficient to impact the room in which the HPWH is located. In this case, there wasn't a significant impact on comfort as the HPWH was located in the unvented attic.

5.2 Whole-House Dehumidifier

Can indoor comfort levels be improved by primarily addressing sensible load with a central A/C and latent load with a WHD?

Modeling results from the energy simulation (Table 17) show that a combined A/C paired with a WHD cooling system at 78°F/55% RH can maintain more comfortable humidity levels within the building than a standard A/C only at 75°F. These results were also confirmed with limited data from short-term scenario testing.

Table 17. Percent Frequency of Conditions Over 60% RH

A/C (°F)	Dehumidifier (%)	Entire Year (%)	June–September (%)
78	**55**	0.09	0.00
75	**off**	15.87	18.25

Additionally, long-term monitoring of the coupled system showed that RH within the home was maintained at levels < 60 % RH for 99.86% of the monitoring period. As human comfort is truly a subjective measure, it is important to note that the homeowner was satisfied with the comfort levels in the home and took notice of the fact that comfort could be achieved at higher A/C set point temperatures when the dehumidifier was running.

Can a more efficient home be created by separating the mechanical systems that address sensible and latent cooling?

Modeling predicts that a home with separated mechanical systems for sensible and latent cooling can see an 8.2% annual decrease in space conditioning (heating, cooling, and dehumidification) electrical energy costs. Additionally, manufacturer rating specifications of the dehumidifier and A/C unit reveal that WHD can remove moisture from the air at a 54% more efficient rate when running at capacity under similar conditions (7.51 vs 3.42 pints/kWh). Actual monitored data resulted in an average performance at 4.95 pints/kWh, which is lower than the manufacturer's rated performance for the dehumidifier, but is expected due to the varying operating conditions over the monitoring period (versus the single point conditions of the rated performance). Short-term testing results demonstrated that the combined A/C and WHD configuration at 78°F/60% RH is able to use less energy per day than a standard A/C setup at 75°F.

5.3 Next Step

This study indicates favorable results for both of these mechanical systems; however, further in-depth research is needed to develop a more robust understanding of their operational performance. More research on the space conditioning effects of HPWHs is being pursued by SWA to access how these units can affect indoor conditions in various climate regions and space configurations. Even though the space conditioning impact of the HPWH in the CEH appears to be relatively minimal, homeowners may realize a more pronounced effect in tighter homes or in buildings that are located in heating-dominate climates.

Additional experimental studies are also needed on WHD-A/C arrangements to further validate the effectiveness of this type of combined cooling system and to access other set point configurations that can yield acceptable indoor environmental conditions. In addition, two manufacturers have recently released whole-house dehumidifiers that warrant investigation, as they eliminate the sensible heat gain issue typical of dehumidifiers and in the case of one of the units, actually provides some sensible cooling.

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Appendix A. Measurements and Calculations

A1. Heat Pump Water Heater

The following one-time measurements were made for the HPWH:

- Air Inlet (CFM), this was done by using a duct blaster as a powered capture hood.
- Power Draw (W)
- Current (A)
- Voltage (Vac)
- Power Factor.

The following HPWH parameters were measured every 10 seconds:

- Inlet Water Temperature (°F)
- Outlet Water Temperature (°F)
- Inlet Air Temperature (°F)
- Inlet Air Relative Humidity (%)
- Outlet Air Temperature (°F)
- Outlet Air Relative Humidity (%)
- Hot Water Flow (gals)
- Condensate (pints)
- Compressor Energy Consumption (Wh)
- Heating Element Energy Consumption (Wh)
- Entire System Energy Consumption (Wh).

An additional ambient air temp/RH sensor was located at 5 ft downstream of the HPWH air outlet. Based upon the measured parameters, the following data were calculated for each 15-minute logging period:

- Average Water Inlet Temperature (°F)
- Average Water Outlet Temperature (°F)
- Minimum Water Inlet Temperature (°F)
- Maximum Water Outlet Temperature (°F)
- Average Inlet Air Temperature (°F)

- Average Inlet Air Relative Humidity (%)
- Average Outlet Air Temperature (°F)
- Average Outlet Air Relative Humidity (%)
- Total Domestic Hot Water Usage (gal)
- Total Latent Moisture Removal (pints)
- Domestic Hot Water Energy (Btu)
- Total Compressor Only Energy Consumption (Wh)
- Total Upper Heating Element Energy Consumption (Wh)
- Total Lower Heating Element Energy Consumption (Wh)
- Total System Energy Consumption (Wh), not including condensate pump
- Total Heat Pump Energy (Wh)
- Total Standby Energy Consumption (Wh)
- Sensible Heat Added (Btu)
- Latent Heat Removed (Btu).

Note that the inlet and outlet temperature and RH measurements were analyzed only when the unit is operating; so these are conditional temperature and RH values, that is, conditional upon the unit operating. Conditional averages were also calculated for the inlet and outlet water temperatures. In addition to daily sums, averages, minima, and/or maxima of the 15-min data, the following data were also calculated on a daily basis:

- Coefficient of Performance
- Pints Removed
- Latent Cooling (Btu/h)

The COP has been defined as the net heat delivered to the hot water system divided by the total electrical energy consumed over a period of time:

$$COP = \frac{useful\ heating\ energy}{net\ energy\ input} = \frac{Q_{DHW}}{W_{DHW} \times 3.413\ \ Btu/Wh}$$

where:

COP = coefficient of performance (dimensionless)

Q_{DHW} = useful heat energy (Btu)

W_{DHW} = energy consumed by the HPWH (Wh)

The water heating energy (Q_{DHW}) was calculated in the datalogger every ten seconds using measured data. These energy values will be summed and recorded at 15 minute intervals.

$$Q_{DHW} = (\Delta T_{T_{Out} - T_{In}} \times \dot{V} \times C_p \times \rho)$$

where:

$\Delta T_{T_{Out} - T_{In}}$ = T_{Out} minus T_{In} (°F)

$\dot{V}$ = hot water volumetric flow rate (ft^3/h)

C_p = specific heat of water ($Btu/lb_m \cdot °F$)

ρ = density of water (lb_m/ft^3)

A2. Whole-House Dehumidifier

During the time of initial component inspection the following one-time measurements were made for the whole-house dehumidifier:

- Power Draw (W)
- Current (A)
- Voltage (Vac)
- Power Factor
- Outdoor Air Inlet (CFM), measured with a hot-wire anemometer
- Indoor Air Return (CFM), measured with a hot-wire anemometer
- Indoor Air Supply (CFM), measured with a hot-wire anemometer.

The following dehumidifier parameters will be measured every 10 s and saved as average or summed values at 10-min time steps.

- Outdoor Air Inlet Temperature (°F)
- Outdoor Air Inlet Relative Humidity (%)
- Indoor Air Supply Temperature (°F)

- Indoor Air Supply Relative Humidity (%)
- Indoor Air Return Temperature (°F)
- Indoor Air Relative Humidity (%)
- Energy Consumption of Unit (Wh)
- Condensate (pints)

Based upon the measured parameters, the following data was calculated for each 10-15 minute logging period:

- Average/Min/Max Air Temperatures (°F)
- Average/Min/Max Air Relative Humidities (%)
- Total Latent Moisture Removal (pints)
- Total System Energy Consumption (Wh)
- Total Standby Energy Consumption (Wh)

In addition to daily sums, averages, minima, and/or maxima of the data, the following data were also calculated on a daily basis:

- Pints Removed
- Latent Cooling (Btu/hr)

Two methods were used to measure the dehumidification performance of the dehumidification system: (1) measuring condensed water exiting the dehumidifier; and (2) calculating the system latent capacity using the psychrometric properties of air entering/leaving the unit and the measured air flow rate.

The first method uses the measured condensate flow multiplied by the latent heat of condensation for water (1,065 Btu/lb) to calculate the latent cooling rate.

$$Q_{cond} = Cond \times 1{,}065 \ Btu/lb \times 60 \ \min/hr$$

where:

$Q_{standbyloss}$	= Latent cooling capacity based upon condensate
Cond	= Measured condensate (lb/min)

The psychrometric method uses the measured humidity of air entering and leaving the coil along with the measured air flow rate across the coil to calculate the dehumidification (latent cooling) rate. It was calculated as follows:

The saturation pressure over liquid water p_{ws} (psia) was found as a function of dry bulb temperature T (°R).

$$\ln p_{ws} = C_8 / T + C_9 + C_{10} T + C_{11} T^2 + C_{12} T^3 + C_{13} \ln T \,,$$

where

C_8 = -1.0440397×10^4,

C_9 = -1.1294650×10^1,

C_{10} = $-2.7022355 \times 10^{-2}$,

C_{11} = 1.2890360×10^{-5},

C_{12} = $-2.4780681 \times 10^{-9}$, and

C_{13} = 6.5459673×10^0.

The partial pressure of water vapor p_w (psia) was found as a function of the saturation pressure over liquid water and relative humidity ϕ (fraction).

$$p_w = p_{ws} \phi \,.$$

The humidity ratio W was calculated as a function of the partial pressure of water vapor and atmospheric pressure p (14.696 psia).

$$W = 0.621945 \frac{p_w}{p - p_w} \,.$$

The specific enthalpy of dry air h_{da} (Btu/lb$_{da}$) was approximated as a function of dry bulb temperature t (°F).

$$h_{da} = 0.240 t \,.$$

The specific enthalpy of saturated water vapor h_g (Btu/lb$_w$) was similarly approximated as a function of dry bulb temperature.

$$h_g = 1061 + 0.44 t \,.$$

The specific enthalpy of moist air h (Btu/lb$_{da}$) was calculated as a function of the specific enthalpy of dry air, humidity ratio, and the specific enthalpy of water vapor.

$$h = h_{da} + Wh_g \,.$$

The total cooling capacity $\dot{Q}_{total}$ (Btu/h) was calculated as a function of flow rate $\dot{V}$ (CFM), supply enthalpy $h_{\sup ply}$ (Btu/lb$_{da}$), return enthalpy h_{return} (Btu/lb$_{da}$), and the density of dry air ρ_{da} (0.076474 lb$_{da}$/ft^3).

$$\dot{Q}_{total} = \dot{V}\rho_{da}\left(h_{\sup ply} - h_{return}\right) \times 60 \frac{\text{minutes}}{\text{hour}} .$$

The latent cooling capacity $\dot{Q}_{latent}$ (Btu/h) was calculated similarly to the total cooling capacity, but just uses the supply and return enthalpies of saturated water vapor at the measured dry bulb temperature.

$$\dot{Q}_{latent} = \dot{V}\rho_{da}\left(h_{g,\sup ply} - h_{g,return}\right) \times 60 \frac{\text{minutes}}{\text{hour}} .$$

Finally the moisture removal efficiency of the dehumidifier was calculated. This dehumidification EF is the daily amount of condensate removed (commonly reported in liters) divided by the daily energy consumption (kWh).

$$EF = \dot{V}_{cond} / P_{DH}$$

where:

$\dot{V}$ = Volumetric flow rate (L/day)

P_{DH} = Energy consumption (kWh/day)

Appendix B. Cooling Scenario Data Detailed Plots

B1. Cooling Scenario Outdoor Condition Comparison

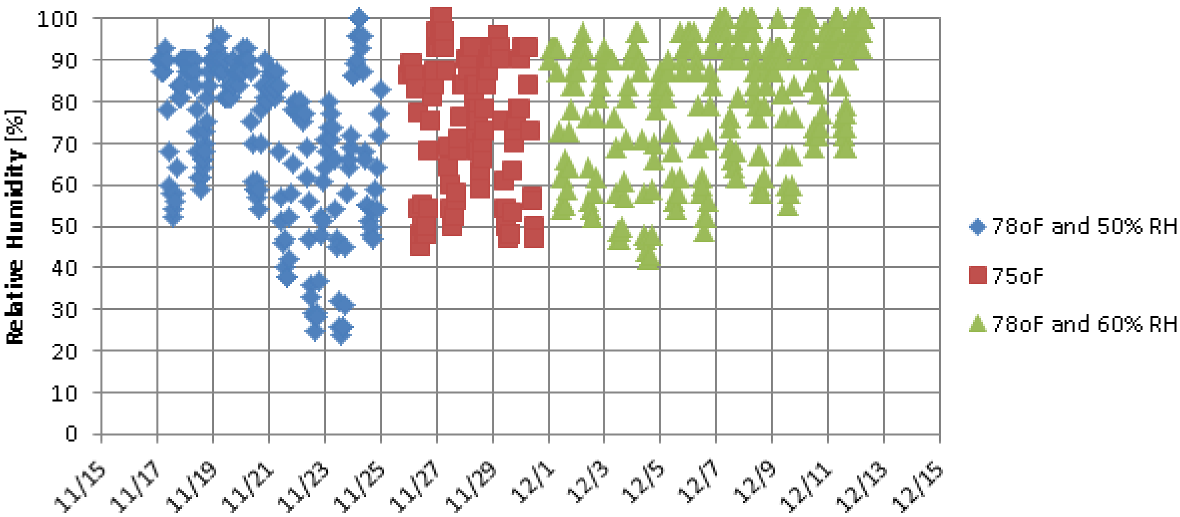

Figure 25. Scenario comparison of outdoor RH

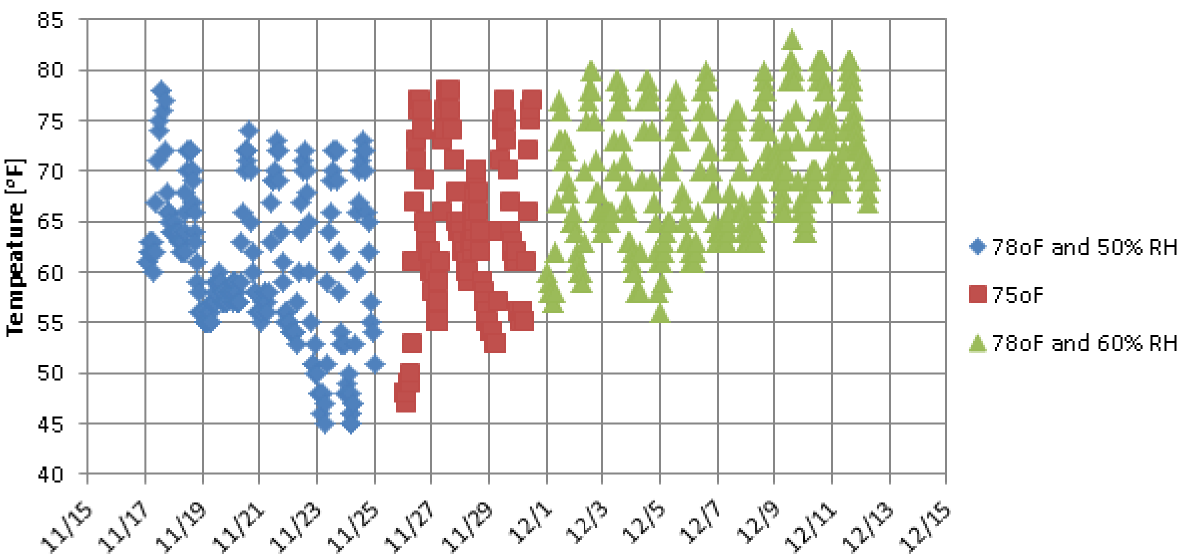

Figure 26. Scenario comparison of outdoor temperatures

B2. Cooling Scenario Indoor Air Conditions Maintenance

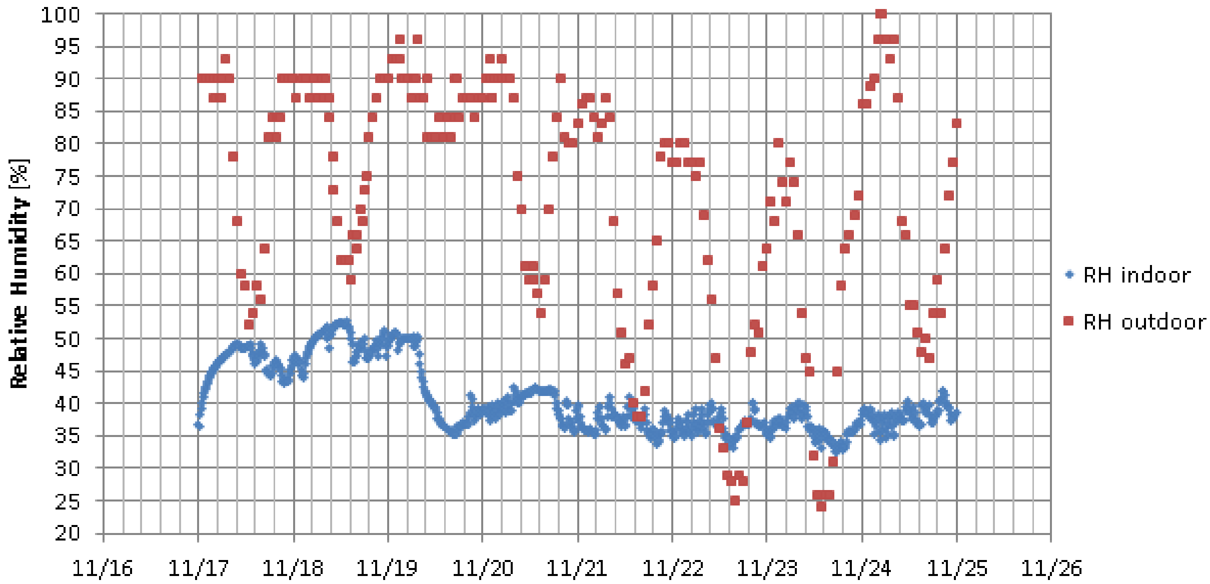

Figure 27. Scenario 1 (78°F, 50% RH) RH maintenance

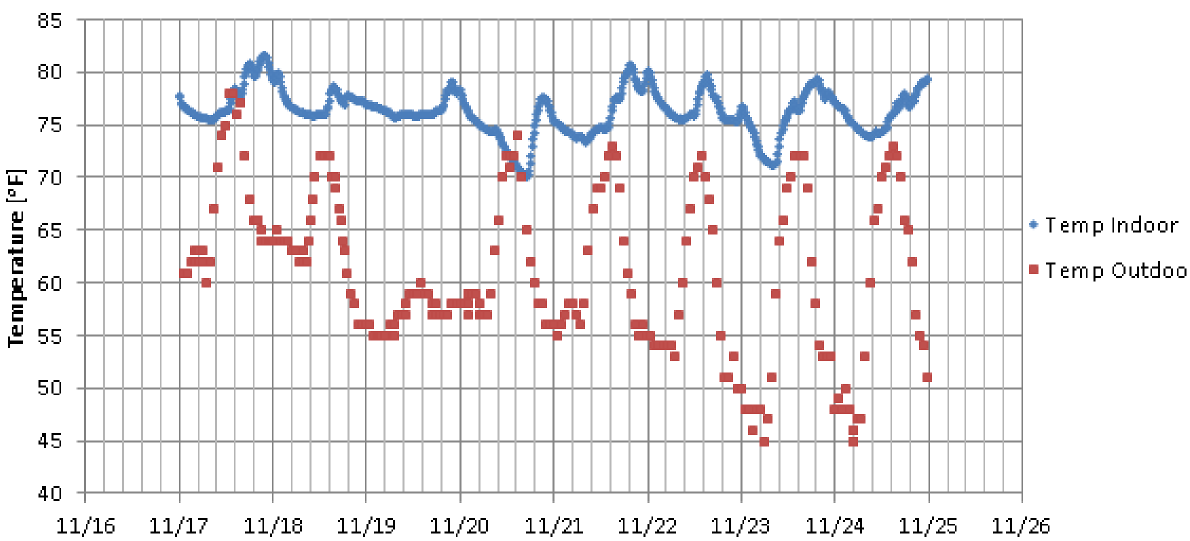

Figure 28. Scenario 1 (78°F, 50% RH) temperature maintenance

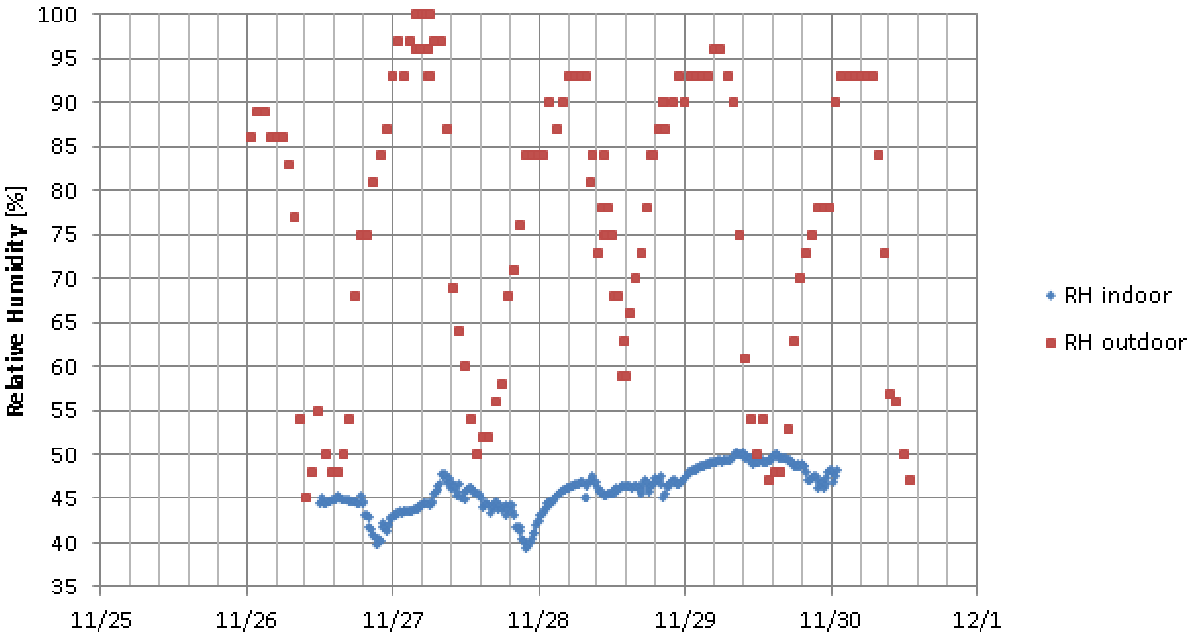

Figure 29. Scenario 2 (75°F, dehumidifier off) RH maintenance

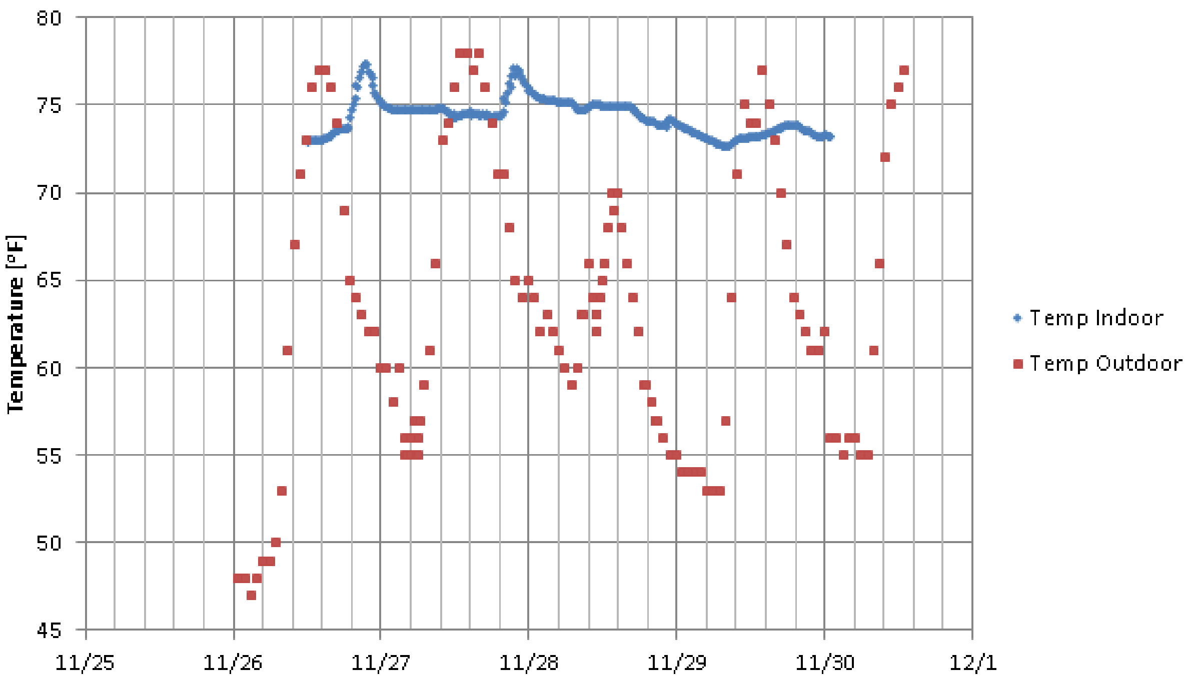

Figure 30. Scenario 2 (75°F, dehumidifier off) temperature maintenance

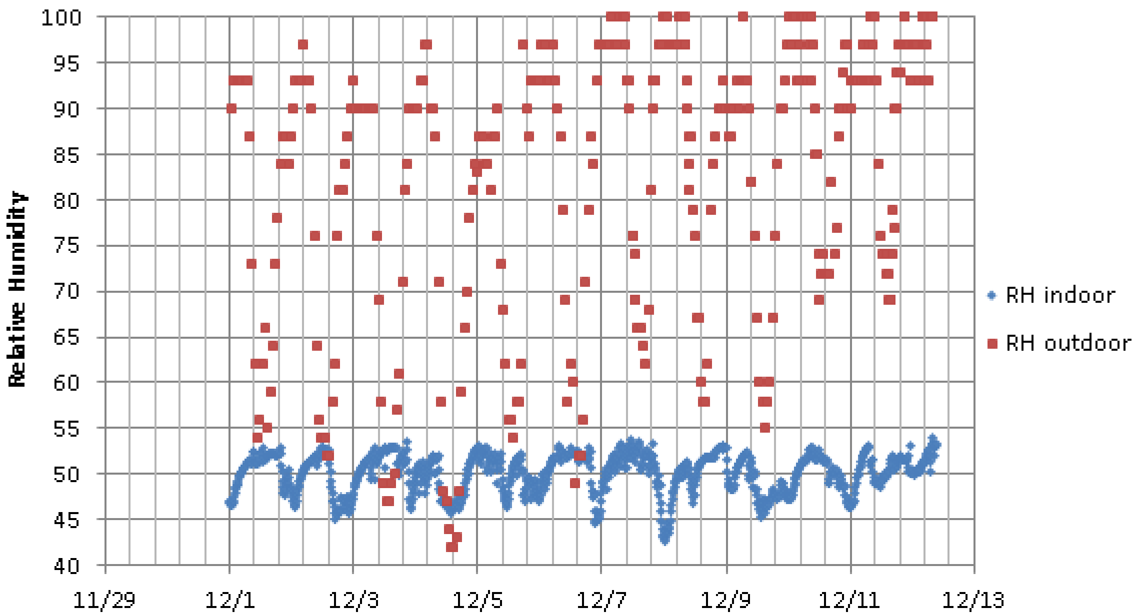

Figure 31. Scenario 3 (78°F, 60% RH) RH maintenance

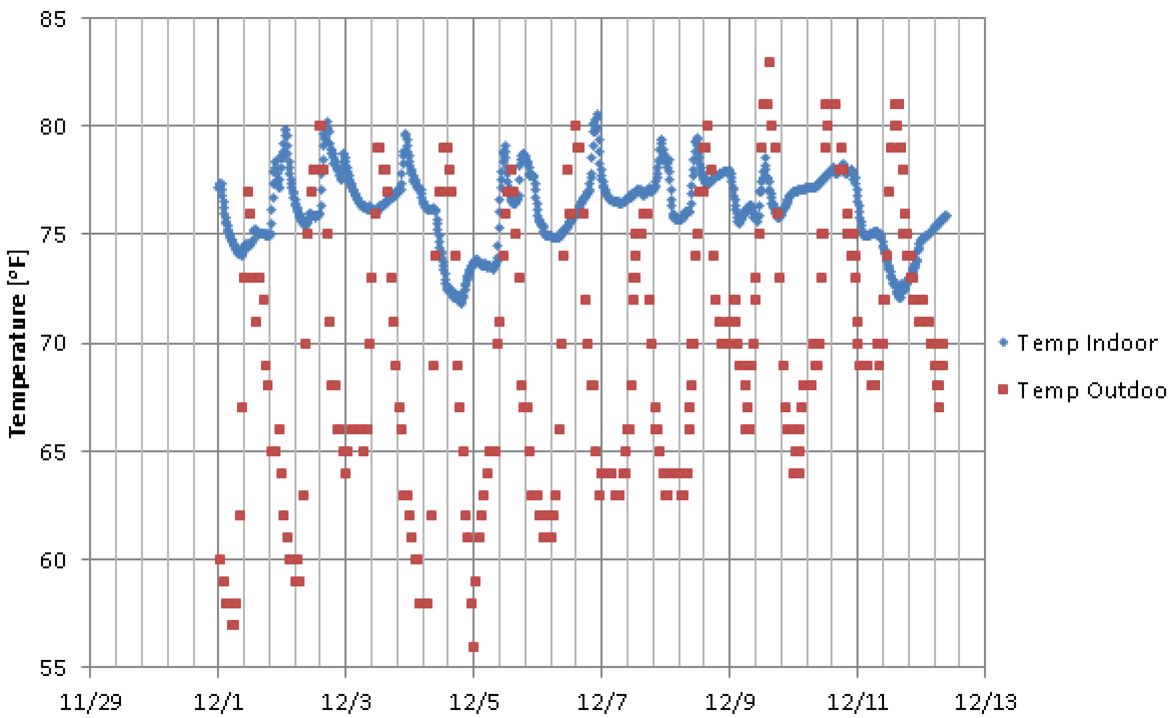

Figure 32. Scenario 3 (78°F, 60% RH) temperature maintenance

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Appendix C. Energy Performance of the Cool Energy House

Energy efficiency measures implemented in the CEH are provided in Table 18. The desire to preserve existing brick façade and minimize impact to interior finishes figured prominently in CARB's recommendations. The first column represents building components in the existing home based on findings from the energy audit.

Table 18. Existing, Proposed, and Post-Retrofit Specifications

Component	Existing	Post-Retrofit
Foundation Assembly	Uninsulated	Same as existing
Above Grade Wall Assembly	1st floor: R-19, Grade III, 2 × 6 16-in. o.c. 2nd floor: R-13, Grade III, 2 × 4 16-in. o.c.	1st floor: R-21, Grade I, 2 × 6 16-in. o.c. 2nd floor: R-15, Grade I, 2 × 4 16-in. o.c.
Ceiling/Attic Assembly	R-19 blown-in fiberglass, 2 × 8 at ceiling, vented attic	R-30 at roof, 5-in. closed-cell spray foam, unvented attic
Window Glazing	Rear (east): single pane, U = 0.869, SHGC = 0.619 Elsewhere: double pane, U = 0.447, SHGC = 0.547 wood, aluminum	All: low-e, double pane, U = 0.28, SHGC = 0.21 vinyl
Building Infiltration	Measured 6 ACH50	Measured 4.8 ACH50[1]
Space Conditioning System	Two 2.5 ton ~SEER 10, HSPF* 6.2 heat pumps	Two 2 ton SEER 17.5, HSPF 9.5 heat pumps, whole-house dehumidifier (150 pints/day; 2.02 L/kWh)
Ductwork	R-4 insulation, measured 7% leakage to outside	Ducts in conditioned space, sealed with mastic
Whole House Ventilation	None	Whole-house supply-only ventilation controlled by humidistat (part of WHD)
Local Ventilation	Spot vent only	Spot vent with delay off timers
Water Heating	Two ERWHs, 50-gal, EF = 0.91	Two HPWHs, 50 gal, EF = 2.35
Lighting	14% fluorescent lighting	75% high efficacy lighting
Roofing Material	dark asphalt shingles, abs = 0.92, e = 0.91	Same as existing
Appliances	ENERGY STAR refrigerator, standard dishwasher	New ENERGY STAR refrigerator and dishwasher, induction stove

* HSPF = heating season performance factor
[1] Proposed 4.0 ACH 50
[2] Proposed ASHRAE 62.2 ventilation rates

Table 19 shows the BEoptE+ v1.2 simulated energy consumption by end use for the existing and post-retrofit Cool Energy House.

Table 19. Simulated Energy Use Distribution

	Existing		Post-Retrofit		Savings
Loads	Site Energy (kWh)	Source Energy (MMBtu)	Site Energy (kWh)	Source Energy (MMBtu)	Percent (%)
Cooling	10,295	118	3,337	38	68
Heating	1,148	13	292	3	75
Hot Water	2,116	24	699	8	67
HVAC Fan	2,524	29	340	4	87
Lights	3,813	44	2,512	29	34
Appliances	1,257	14	1,220	14	3%
Ventilation	33	0	98	1	–198
MELs	4,622	53	4,622	53	0
Total	25,807	296	13,120	151	49

Figure 33 shows the cumulative energy savings by measure and their individual impacts on the whole-house energy savings. The new air source heat pumps contributed the most; the unvented "conditioned" attic and HPWHs came in second and third, respectively. Improvements to lighting and windows contributed a little more than 5% each to the total 49% savings.

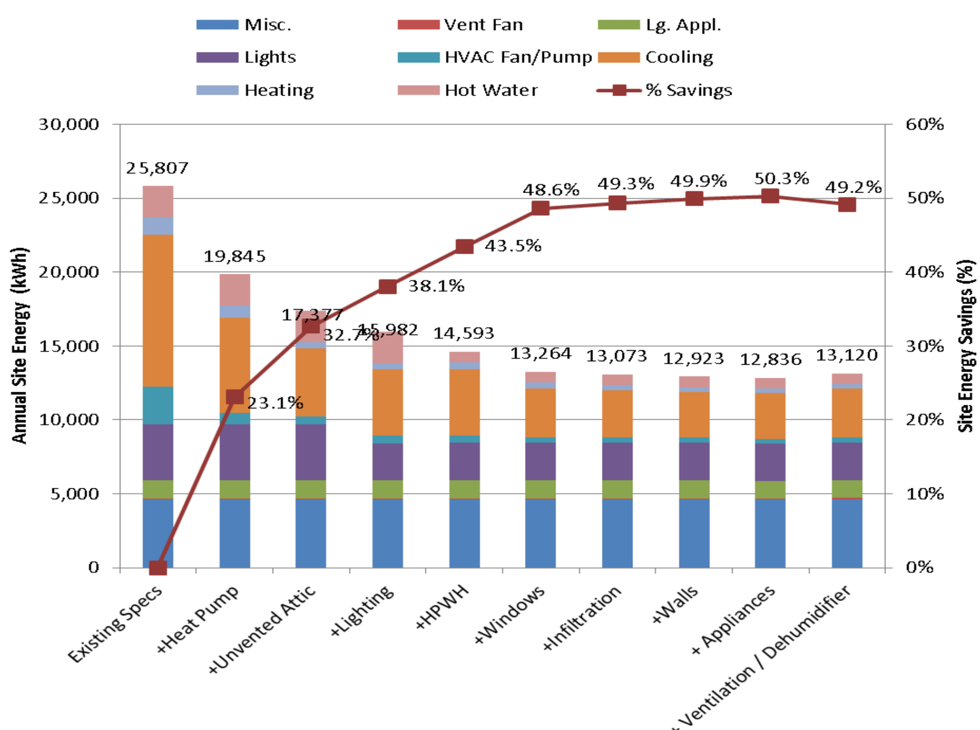

Figure 33. Cumulative contribution to total simulated energy savings, by measure and end use

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