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Air-to-Water Heat Pumps With Radiant Delivery in Low-Load Homes

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Alliance for Residential Building Innovation

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Air-to-Water Heat Pumps With Radiant Delivery in Low-Load Homes

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Unless otherwise noted, all figures were created by the ARBI team.

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Definitions

ACH	Air changes per hour
ACM	Alternative Calculation Methodology
AH	Air handler
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
AMY	Actual meteorological year
ARBI	Alliance for Residential Building Innovation
AWHP	Air-to-water heat pump
BA	Building America
BEopt	Building Energy Optimization
Btu	British thermal unit
C&C	Cool and coast
CEC	California Energy Commission
CI	Confidence interval
COP	Coefficient of performance
CZ	Climate zone
DEG	Davis Energy Group
DH	Dehumidifier
DHW	Domestic hot water
DOE	U.S. Department of Energy
EAT	Entering air temperature
EER	Energy efficiency ratio
EIA	U.S. Energy Information Administration
EPS	expanded polystyrene
ERV	Energy recovery ventilator
EWBT	Entering wet-bulb temperature
EWT	Entering water temperature
HERS	Home Energy Rating System
HP	Heat pump
HSPF	Heating seasonal performance factor
IECC	International Energy Conservation Code
kWh	Kilowatt-hour

LWT	Leaving water temperature
MM	Mixed-mode
OAT	Outdoor air temperature
PV	Photovoltaic
PWR	Power
RCC	Reverse cycle chiller
RH	Relative humidity
S.E.E.D.	Super Energy Efficient Design
SEER	Seasonal energy efficiency ratio

Executive Summary

Vapor compression heating and cooling system performance varies substantially with changing operating conditions. Performance can be improved by reducing the "thermal lift" (the difference between condenser and evaporator temperatures) of the system, reducing compressor cycling, and improving distribution efficiency. Air-to-water heat pumps (AWHPs) that substitute a refrigerant-to-water heat exchanger for the customary refrigerant-to-air indoor air coil and that use hydronic distribution can be used to facilitate these potential efficiency improvements.

Space conditioning represents nearly 50% of average residential household energy consumption (U.S. Energy Information Administration 2009), highlighting the need to identify alternative cost-effective, energy efficient cooling and heating strategies. As homes are built better, there is an increasing need for strategies that are particularly well suited for high performance, low-load homes. Due to their efficiency advantages, AWHP systems are particularly well suited for low-load homes.

This research evaluates air-source AWHPs applied to radiant floor and mixed-mode (MM) distribution systems. The MM strategy consists of hydronic distribution using a small fan coil connected in series with and upstream of a radiant floor system. Two monitoring projects in hot-dry climates were initiated in 2010 to test this strategy. One of the projects, the Cana house, is a three-bedroom, 3,270-ft^2 straw-bale house located in Chico, California, which uses a three-function Altherma heat pump. The second project is the Super Energy Efficient Design (S.E.E.D.) house, which is a 1,935-ft^2, single-story spec home in Tucson, Arizona. The heat pump at the S.E.E.D. house is a built-up system with an Aqua Products refrigerant-to-water heat exchanger. The systems in each test house were fully instrumented and monitored over 1 year to capture complete performance data over the cooling and heating seasons. Results are used to quantify energy savings, cost effectiveness, and system performance using different operating modes and strategies. A calibrated TRNSYS model was developed and used to evaluate performance in various climate regions.

Monitoring results demonstrated seasonal space heating coefficients of performance over the full monitoring period of 3.26 and 4.18 at the S.E.E.D. house and Cana house, respectively. Measured heating performance of the Altherma heat pump was very comparable to manufacturer specifications; however, water heating performance was much lower than expected as a result of poor heat transfer between the heat pump supply loop and the storage tank and regular operation of the electric resistance backup heater. Seasonal energy efficiency ratios over the monitoring period in space cooling were 11.2 and 10.8 at the S.E.E.D. house and Cana house, respectively. This is a substantial improvement over measured performance in the field of residential air conditioners with ducted air delivery of 5.5 to 8.5 energy efficiency ratios (Proctor et al. 2011). Performance was most dependent on outdoor air conditions with less than expected sensitivity between efficiency and entering water temperature on the load side. Data were not able to confirm expected performance improvements of the hydronic system due to reduced thermal lift from high supply temperatures in cooling and lower supply temperatures in heating.

Measured distribution efficiencies of the radiant floor distribution averaged 96%. This is approximately equivalent to a ducted distribution system with ductwork located inside

conditioned space (94%), but is much higher than the 76% estimated for typical attic-located tight ducts (≤ 6% air leakage).

TRNSYS modeling estimates up to 31% annual HVAC energy savings compared to an air-to-air heat pump with tight ducts located in the attic and up to 28% compared to the same base case with ducts located within conditioned space. Percent savings are higher for cold climates or hot climates with high heating loads as modeling results indicate higher radiant distribution effectiveness for heating than for cooling. Some form of dehumidification is required in all but the driest climates. Monitoring and modeling results from the dry Central Valley of California indicate that neither floor condensation nor interior comfort is a concern with radiant-only cooling distribution. In other dry climates, such as Tucson, Arizona, some dehumidification is required during the humid monsoon season, which can be accomplished with the MM distribution strategy. A control strategy to optimize performance could incorporate a humidistat control on the fan coil to switch from floor cooling to MM cooling only during periods of rising indoor moisture conditions. Due to high latent loads, the model found that radiant cooling is not appropriate in humid climates.

Substantial cooling energy savings can be achieved from a precooling operating strategy that shifts air-conditioner daytime operation to cooler nighttime hours and utilizes the house thermal mass to ride out most peak afternoon cooling events. Monitoring results from the S.E.E.D. house show 27% savings from precooling operation at a daily maximum outdoor temperature of 90°F and up to 40% savings at a maximum temperature greater than 100°F. TRNSYS modeling estimates 17%–43% seasonal cooling energy savings from precooling in hot-dry climates. Seasonal percent savings can be lower than daily savings on hot days due to overcooling on milder days. An optimized solution could minimize this by employing a “smart” precool strategy that monitors weather conditions and changes the precooling set point accordingly.

AWHPs with radiant or MM delivery are an effective and efficient means of providing space heating and cooling in residential buildings in certain climates. This strategy presents a viable alternative to locating ductwork in conditioned space, which may not be feasible in all homes due to architectural challenges, while providing the comfort, thermal storage, improved distribution, and reduced noise benefits of radiant slab delivery. Based on a cost benefit analysis over a 30-year mortgage using mature market costs, the strategy was found to be cost effective only in cold climates and not in hot-dry climates. However, there are various benefits provided by the AWHP radiant system that are not factored into a cost analysis. These include increased occupant comfort, improved distribution, noise reductions, and peak load reduction. In addition, as electricity prices increase this may move the technology into cost effectiveness for additional climates.

Current system costs are high; however, there is justification to anticipate lower incremental costs as this strategy gains wider market acceptance. Cost reductions can be expected with increased contractor familiarity and reductions in manufactured equipment costs from volume production. Further research focused on development of packaged AWHPs as well as packaged controls for zoned systems is necessary. This will be a driver for cost reductions and simplified installation procedures, as well as for ensuring consistent levels of quality and gaining market acceptance from contractors and installers.

Acknowledgments

Davis Energy Group would like to acknowledge the U.S. Department of Energy Building America program for its funding and support of development of this technical report as well as the research that informed it. In addition, we would like to acknowledge builders Michael Ginsburg of La Mirada Homes and Robin Trenda of Chico Green Builders, Mark Sadler at Daikin AC, and Michael Erickson at Hydronic Heat Pumps for their cooperation throughout the design, construction, and monitoring stages of this project.

1 Introduction

1.1 Background and Motivation

Space conditioning represents nearly 50% of average residential household energy consumption according to the *2009 Residential Energy Consumption Survey* (U.S. Energy Information Administration [EIA] 2009). Identifying cooling and heating strategies that address the need for more cost-effective, energy efficient residential systems will lead to reductions in overall residential energy consumption and help further progress toward Building America (BA) goals.

Vapor compression heating and cooling system performance varies significantly with changing evaporator and condenser temperatures, evaporator airflow, equipment cycling, and other factors such as system charge and airflow. Installed heat pump (HP) system performance can be improved by reducing "thermal lift" (the difference between condenser and evaporator temperatures) of the system, reducing compressor cycling, and improving distribution efficiency. Air-to-water heat pumps (AWHPs) that substitute a refrigerant-to-water heat exchanger for the customary refrigerant-to-air indoor air coil and that use hydronic distribution can be used to facilitate these potential efficiency improvements.

As homes are built better, there is an increasing need for heating and cooling strategies suited for high performance, low-load homes. Forced-air furnace systems are typically oversized for these applications, resulting in reduced operating efficiencies and delivery effectiveness. Ductwork located in attics results in large distribution losses due to conduction and air leakage.

AWHP systems are an energy efficient space conditioning solution that, through additional field research and coordination with manufacturers, has the potential to lead to a market-ready product that cost-effectively provides comfort in homes with efficient, safe, and durable operation. Radiant delivery increases distribution efficiencies and hydronic systems with higher cooling supply temperatures and lower heating supply temperatures improve system efficiency and are well designed for low-load homes. Radiant cooling distribution is most appropriate to hot-dry climates; however, when combined with a dehumidification strategy it may be applicable in humid climates. The technology can be implemented in both single and multifamily residences. While it may be appropriate for some deep-retrofit projects, the focus of this research is new construction.

Critical path milestones related to high performance HVAC and delivery systems identified by the BA Space Conditioning Standing Technical Committee members include the following:

- Identify low-cost space conditioning distribution strategies with negligible conductive, radiant, and leakage losses
- Demonstrate market-ready, high-efficiency, small-capacity heating and cooling equipment for low-load situations.

Through detailed monitoring of two test houses and TRNSYS modeling, this research project evaluates air-source AWHPs applied to mixed-mode (MM) distribution systems that are capable of moderating condensing/evaporating temperatures and reducing thermal lift. The MM distribution strategy consists of hydronic distribution using a small fan coil connected in series with and upstream of a radiant floor system (see Figure 1). Chilled water is piped first to the small fan coil, which provides latent and sensible cooling and significantly reduces the size of ducting needed. The water is then delivered to the radiant floor tubing, which will provide the bulk of the sensible cooling. Piping chilled water to the fan coil first warms the water entering the slab and removes moisture in the supply airstream, reducing the risk of condensation on the floor surfaces.

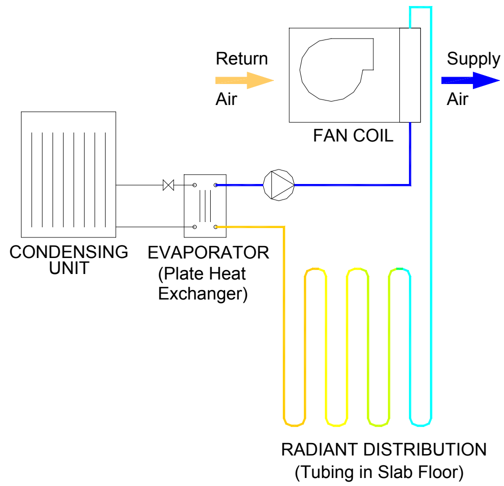

Figure 1. Schematic of air-to-water cooling system with MM distribution

The radiant system provides primary distribution, thermal storage, and zoned comfort. Due to the thermal mass requirements, houses must be built on slab foundations with exposed or tile floors (minimal carpeting) or incorporate other strategies of adding sufficient mass to the house.

The effectiveness of a MM forced-air/radiant cooling system was evaluated in a previous BA study in Borrego Springs, California, with results showing significant improvements in cooling efficiencies (Springer et al. 2008). Two nearly identical homes were equipped with the same model 13 seasonal energy efficiency ratio (SEER) condensing unit, one connected to a conventional direct expansion evaporator coil and ducted distribution system, and the other to a refrigerant-to-water heat exchanger with MM distribution. Over the test period from July through September 2007, energy efficiency ratios (EERs) of 5.1 and 10.3 were measured for the standard system and the chilled water system, respectively. It was theorized that the reduced thermal lift resulting from the relatively high evaporator temperature of the chilled water system was responsible for the substantial reduction in compressor power. Because the slab underside was uninsulated, little benefit was seen in the way of seasonal energy savings, resulting in distribution inefficiencies due to the significant downward energy loss.

1.2 Objectives and Research Questions

The primary objectives of this study are to evaluate air-source AWHPs applied to MM systems that use radiant cooling and heating as the primary means of distribution, along with an upstream forced-air cooling coil for humidity control (latent cooling), and determine how well this strategy performs in various climates.

Efforts are made to provide conclusions to the following research questions in this report:

1. What are the average effective heating coefficients of performance (COPs)? What is the efficiency of the integrated water heating/space heating system in heating mode (Altherma) and how does it compare to manufacturer's specifications?
2. What are the average effective cooling EERs, and can the dramatic improvement in performance relative to typical forced-air-only systems seen in previous testing be replicated?
3. How effective is nighttime precooling in improving HP efficiencies and reducing cooling energy use?
4. How does the distribution efficiency of the MM system compare to that of a typical forced-air delivery system with ducts in unconditioned space?
5. Is the fan coil and the latent cooling it provides necessary for dehumidification and to prevent floor condensation in a hot-dry climate, or, can the forced-air delivery be eliminated completely?
6. Can TRNSYS reliably predict performance of this HVAC strategy?
7. In what climate zones is this strategy applicable?

2 Technology and Project Description

2.1 Air-to-Water Heat Pump Products

AWHPs operate on the same mechanical principles as air-to-air HPs, but instead of connecting outdoor units to an indoor refrigerant-to-air heat exchanger coil as split-system air-to-air HPs do, AWHPs employ a refrigerant-to-water heat exchanger and generate hot or chilled water. To distribute heating and cooling they circulate the water through fan coils or radiators.

Numerous manufacturers offer AWHPs. These may be offered as "packaged" products that incorporate both the refrigerant-to-water heat exchanger and the outdoor unit, such as the Daikin Altherma,[1] the Multiaqua MACH,[2] the Unico UniChiller,[3] and the SpacePak Chiller.[4] However, the Multiaqua models provide cooling only, not heating. Other companies, such as Aqua Products,[5] manufacture individual refrigerant-to-water heat exchangers that can be coupled to any commercially available outdoor unit. The Daikin Altherma product can be configured as a split system in which the indoor unit includes the refrigerant-to-water heat exchanger, pump, controls, and other hydronic components, or as a monobloc-package unit with all of the components located with the outdoor unit and supply and return piping run to the building. The monobloc-package units are factory charged and only require water and control connections. The Daikin units have inverter-driven compressors and variable-speed outdoor fans. These systems can produce water that is typically in the range between 40°F and 130°F.

Of the systems discussed above, all are currently available in the United States and several more options are sold in Asian and European markets. Other manufacturers include LG and Fujitsu. The two systems evaluated under this research are the Aqua Products reverse cycle chiller (RCC) installed in the S.E.E.D. house and the Daikin Altherma Monobloc installed in the Cana house.

Advantages of these systems over air-to-air HPs include the following:

- With the use of inverter-driven compressors (Daikin) or buffer tanks they can operate at a very wide range of capacities and accommodate low-load buildings.
- They can be operated down to very low outdoor temperatures and when used with buffer tanks or radiant floor slab distribution do not require resistance heat during defrost cycles.
- They can utilize building mass, particularly radiant floor slabs, to shift load and improve performance by operating when the outdoor temperature is lower in summer and higher in winter.
- Systems are easily zoned without the penalties experienced with air-based systems.

Daikin and others offer a domestic water heating option that allows the full capacity of the HP to be applied to water heating. A three-way valve switches between space heating and cooling, and water heating.

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[1] See www.daikinac.com/residential/altherma-system-configuration.asp?sec=productsandpage=53.
[2] See www.multiaqua.com/index.htm.
[3] See www.unicosystem.com/Home/Products/UniChillerRC/tabid/80/Default.aspx.
[4] See http://spacepak.com/air-conditioning-products.asp#chb.
[5] See www.aquaproducts.us/products/reverse-cycle-chiller.html

2.2 Hydronic Distribution Options

AWHPs open up several options for distribution. As with air-to-air split system HPs, fan coils and ducting can be used for distribution. They can also deliver heating and cooling to a variety of hydronic distribution systems including radiant floor, ceiling, and wall panels; ductless fan convectors; traditional radiators; and baseboard convectors. To capitalize on the opportunity to improve performance through reduced thermal lift and load shifting, their best application is with radiant floor systems. The Alliance for Residential Building Innovation (ARBI) team researched various hydronic distribution systems to determine their market potential in a feasibility study report (Springer et al. 2012).

Because of the potential for moisture damage from condensation, radiant floor cooling must only be used with exposed concrete slabs or slabs with ceramic tile or stone coverings. Carpeting and wood floors increase the risk of floor condensation and provide a better medium for mold growth, and vinyl flooring acts as a vapor barrier to trap the condensed moisture. The recommended minimum floor surface temperature is about 65°F. The risk of condensation is reduced when used in well-insulated, tightly constructed houses in dry climates. Since radiant cooling is strictly “sensible,” moisture removal can be accomplished by introducing a small fan coil in series and upstream of the radiant panels to provide latent cooling and moisture removal. In all but the driest climates some method for dehumidification is highly advised because indoor moisture sources can elevate dew point temperatures even in well-constructed and properly ventilated houses. Delivering chilled water to the fan coil first facilitates moisture removal, and increases the temperature of the water entering the radiant panel, making it less likely to approach the dew point temperature. The relative percentage of cooling provided by the coil and floor can be varied through adjustment of the air handler fan speed. This strategy was shown to provide a dramatic improvement in condensing unit EER (~200% increase in EER compared to a 13 SEER unit with conventional ducted distribution) in prior BA research conducted at Borrego Springs (Springer et al. 2008).

2.3 Test House Measure Details

Two projects were initiated in 2010 to test MM AWHP systems. One of the projects, the Cana house, is a three-bedroom, 3,270-ft^2 straw-bale house located in the hot-dry Northern California climate of rural Chico (see Figure 2). The owners wanted radiant floor heating and an integrated system to effectively heat and cool their home. The home is located in a rural area with no access to natural gas on site. The Cana house uses the Daikin Altherma inverter-driven three-function AWHP for space heating and cooling as well as domestic water heating. By substituting the Altherma for a conventional furnace, air conditioner, and water heater, the need for propane gas was eliminated, and electric heating allows more of the building energy use to be met by on-site photovoltaic (PV) generation.

Figure 2. Completed Cana test house

The second test house is the Super Energy Efficient Design (S.E.E.D.),[6] which is a 1,935-ft^2, single-story spec home located in the hot-dry climate of Tucson (German et al. 2012; see Figure 3). The builder, Michael Ginsburg of La Mirada Homes, developed this prototype design with the goal of providing exceptionally efficient yet affordable homes. The S.E.E.D. test house uses an Aqua Products RCC for space heating and cooling only. The RCC packages a conventional Ruud 13 SEER HP with a refrigerant-to-water heat exchanger.

Figure 3. Completed S.E.E.D. test house

[6] More information on the S.E.E.D. house can be found at the builder's website: http://lamiradahomes.net/lamirada_homes_seed.htm

Both test homes were completed in 2011 and are currently occupied. Both systems have been fully instrumented and are monitored over 1 year to capture complete performance data over the cooling and heating seasons. The two test sites also provide information on costs, installation, equipment and system operation and durability, and contractor training needs. Results are used to quantify energy savings, cost effectiveness, and system performance using different operating modes and strategies. Table 1 lists the basic home characteristics and the specifications for the AWHP systems installed in the two test houses. For additional details on the S.E.E.D. house see ARBI's "Super Energy Efficient Design (S.E.E.D.) Home Evaluation" (German et al, 2012).

Table 1. HVAC Measure Specifications

Measure	Cana Test House	S.E.E.D. Test House
Basic Building Characteristics		
Building Type/Stories	Single family, one story	Single family, one story
Conditioned Floor Area, Ft^2	3,270	1,935
Number of Bedrooms	3	4
Heating/Air-Conditioning Type and Efficiency	Altherma Monobloc HP: COP 4.37 and EER 9.4	Ruud HP w Aqua Products RCC: heating seasonal performance factor (HSPF) 8.4 and EER 11 (Ruud rating)
Heating Distribution	MM radiant floor and fan coil	Radiant floor
Cooling Distribution	MM radiant floor and fan coil	MM radiant floor and fan coil
Duct Location and Insulation	Attic R-6, < 6% total leakage	Conditioned space, R-6

At both test houses space heating and cooling are provided by the AWHP, with primarily delivery through a radiant floor and additional delivery via a fan coil (for cooling only at S.E.E.D.; see Figure 4). The fan coil is sized to provide about half the required cooling capacity. During the winter, in the Cana house the fan coil is operated at low speed with approximately 75% of heating being delivered through the radiant floor. At the S.E.E.D. house, a bypass valve at the fan coil sends the hot supply water directly to the floor and the fan is only used if requested for air movement purposes. A dedicated mechanical ventilation system is still required in both scenarios. In both houses the fan coil is located in the mechanical room adjacent to the outdoor condensing unit, so pipe run lengths are short and pipe losses are not significant.

The Daikin Altherma has not been tested according to the U.S. Department of Energy (DOE) testing procedures for central air conditioners and HPs (Title 10 of the *Code of Federal Regulations* part 430, subpart B, appendix M) because the test procedure does not account for operational characteristics of air-to-water pumps or the integrated domestic hot water (DHW) component. In March 2011, DOE approved an alternative testing method based on the European testing methods and standards (European Standards EN 14511 and EN 15316) for rating the Altherma equipment based on EER and COP. Daikin has received approval from the California Energy Commission (CEC) for a compliance option that allows for modeling within the Title 24 Building Energy Code (CEC 2012). Cooling efficiencies were tested at outdoor conditions of

95°F and a supply water temperature of 64°F while heating effiencies were tested at outdoor conditions of 45°F and a supply temperature of 95°F. Under these conditions operational efficiencies are 9.42 EER for cooling and 4.37 COP for heating.

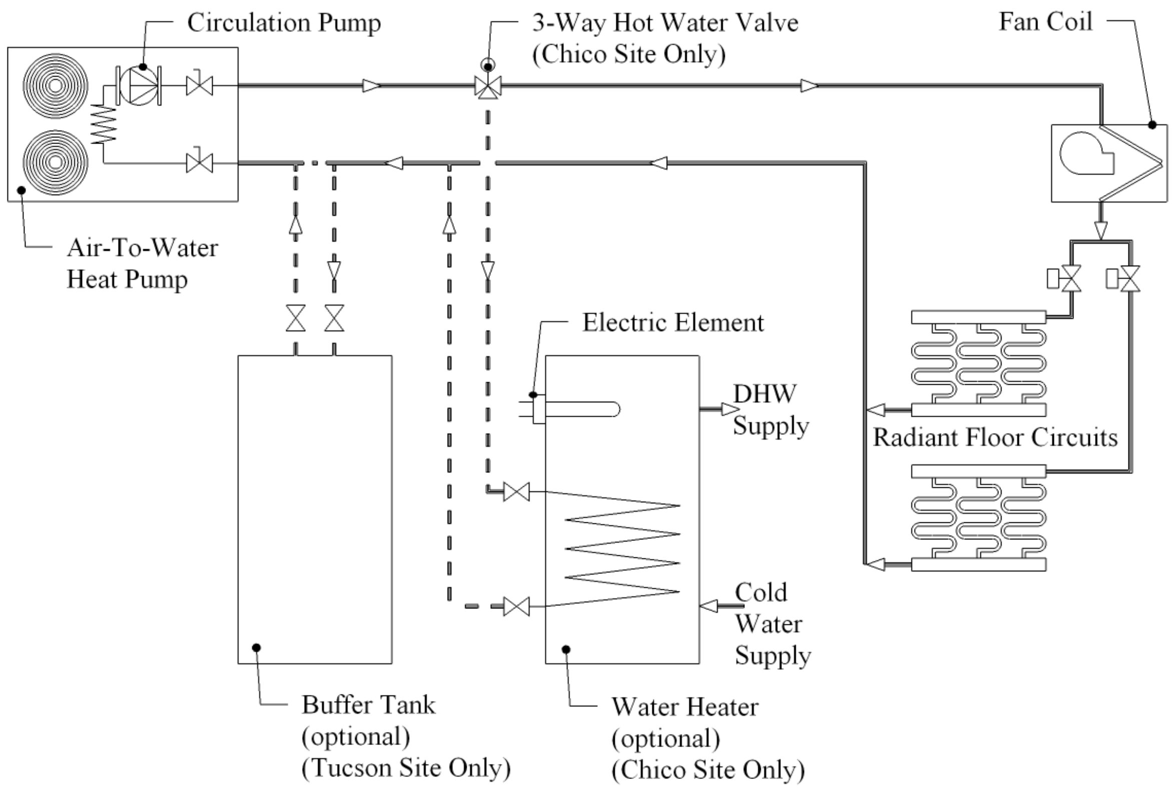

Figure 4. Schematic of AWHP with MM distribution

Figure 5 shows the attached DHW tank and three-way valve for switching between space conditioning and DHW mode for the system in Chico. No buffer tank was installed because the thermal mass of the slab and the variable capacity capabilities of the Altherma prevent the HP from short cycling. Figure 6 shows a picture of the installed unit at the Cana house and in Figure 7 the hydronic distribution system can be seen.

Figure 5. The Monobloc Altherma installed at the Cana house

Figure 6. The hydronic installation at the Cana house. The mechanical room on the left shows the storage tank for DHW and piping to both the radiant floor and the fan coil. On the right is a manifold for the radiant floor.

In the S.E.E.D. test house the AWHP, manufactured by Aqua Products, consists of a standard efficiency 13 SEER Ruud HP perched on a module that contains the evaporator coil and temperature controls. As is the case with the Altherma, because the Ruud is installed with nonmatched heat exchanger coils it is not rated by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI). Average rated efficiencies for this unit with a matched standard indoor evaporator coil are 11 EER and 8.4 HSPF. The Aqua Products HP does not have the variable capacity capabilities of the Altherma. A 30-gal buffer tank was installed in the hydronic loop to minimize equipment short cycling. Figure 7 shows a picture of the installed unit. The fan coil is located in an insulated closet and all ductwork is in conditioned space. Figure 8 shows the installed hydronic equipment and piping. The small tank on the far right is the buffer tank for space conditioning. The large storage tank in the middle with the drainback tank above it is for the solar DHW. The hydronic fan coil can be seen on the far left in the closet.

Figure 7. West elevation of S.E.E.D. house showing sun screen over window and AWHP

Figure 8. S.E.E.D. house hydronic equipment and piping installed in the garage

2.4 System Costs

At the initiation of the research study, Daikin was one of the only manufacturers selling packaged AWHPs in the United States market, the first of which was installed in 2008. While other companies have since entered the market, no major U.S. manufacturer has yet begun to make AWHPs. Currently, these systems are priced significantly higher than standard HPs of similar efficiencies. For systems like the Altherma that also provide DHW, part of the incremental cost can be attributed to this other end use.

Current high costs of packaged equipment are largely the result of an emerging market and limited consumer demand for this technology. Additionally, the controls and capabilities of the Altherma are quite sophisticated, adding costs that may or may not be warranted for residential applications. For single- or dual-zone, low-load homes with high building thermal mass, expensive inverter-driven compressor technology with modulating capacity does not provide the magnitude of savings that might be expected in other applications. The same is true of outdoor air reset in milder climates. Costs are expected to come down as these systems gain market acceptance through increased contractor familiarity and reductions in manufactured equipment costs due to volume production. Control simplifications should result in a much more cost-effective product without significantly compromising efficiencies. Costs can also be reduced through the strategy used in the S.E.E.D. home in which a standard HP is used in conjunction with a refrigerant-to-water heat exchanger. The incremental cost in this case is attributed only to the heat exchanger, the radiant floor system, and any accessories. However, the limitation of this strategy is that there are no AHRI-certified matched combinations of commercially available HPs with refrigerant-to-water heat exchangers, limiting credit under building energy codes for high efficiency equipment. However, if these systems gain market acceptance, manufacturers may move to develop an AHRI testing method or identify other similar acceptance for codes, such as the Daikin CEC compliance option for the Altherma unit (CEC 2012). Previously, the federal minimum efficiencies for space conditioning HPs and electric resistance storage water heaters had to be used as the efficiency descriptors.

There are also a number of other nonenergy benefits of radiant floor delivery that are difficult to incorporate into the cost-benefit analysis, including occupant comfort, noise reductions, and peak load reduction. Radiant distribution provides additional comfort over air distribution system since it better regulates the mean radiant temperature within the space. Often with forced-air systems the air may be within a reasonable temperature range but the surfaces within the space may not be, causing occupants to feel too hot in the summer and too cold in the winter. In poorly insulated homes this can result in the occupant turning the thermostat down in the summer and up in the winter to compensate.

Complete as-built cost information for the MM delivery AWHP system was provided by the builder for both the S.E.E.D. home and the Cana home and are presented in Table 2 and Table 3, respectively. The base case is assumed to be an air-to-air HP of standard efficiency (federal minimums) with ducted forced-air delivery (tight ducts, R-6). Base case system costs are estimated from a combination of Davis Energy Group's and Building Energy Optimization's (BEopt's) cost databases. For the S.E.E.D. house, total as-built HVAC equipment costs were higher than expected primarily due to the high costs from Aqua Products for the packaged air-to-water condensing unit. Costs could be significantly lowered by purchasing only the refrigerant-to-water heat exchanger from Aqua Products, sourcing the HP through the contractor's regular supplier, and assembling the unit on site.

Table 2. S.E.E.D. House System Total and Incremental Costs

Building Component	Base Case Specifications	As-Built Specifications	Base Case Cost	As-Built Costs	Incremental Costs
Slab	4-in. slab mono pour	4-in. slab footing & stem + R-10 underslab & edge insulation	$15,000	$19,020	$4,020
Radiant Floor, Manifolds, Zone Controls, and Valves	None	Per plan	–	$9,844	$9,844
HVAC Equipment	7.7 HSPF/13 SEER, 4-ton, R-6 ducts in attic	Aqua Products RCC, 2-ton	$7,624	$15,182	$7,558
Total Costs			$22,624	$44,047	$21,423

Table 3. Cana House System Incremental Costs

Building Component	Base Case Specifications	As-Built Specifications	Base Case Cost	As-Built Costs	Incremental Costs
Slab	–	2-in. expanded polystyrene (EPS) edge + 1¼-in. EPS underslab	–	$4,510	$4,510
Radiant Floor, Manifolds, Zone Controls, and Valves	none	Per plan	–	$9,083	$9,083
HVAC Equipment	7.7 HSPF/13 SEER, 5-ton + storage water heater	Altherma Monobloc HP, 4-ton	$11,810	$30,818	$19,008
Total Costs			$11,810	$44,411	$32,601

Slab insulation and hydronic system costs were quite comparable between the two test houses. The HVAC equipment costs at the Cana house are significantly more due to the high cost of the Altherma system. As-built costs include the DHW portion, and therefore a storage water heater is added to the base case costs. More HP downsizing potential was possible but at the time of installation, a 4-ton Altherma was the smallest unit available.

In dry climates where dehumidification may not be necessary, the incremental cost could be reduced through elimination of the fan coil and all ductwork. Coupling this with the potential mature market cost savings including increased contractor familiarity and reductions in manufactured equipment costs due to volume production, Table 4 presents estimated mature market incremental costs for this system installed in a 2,400-ft^2 BA Benchmark[7] house. Following are justifications for the proposed reductions.

- **Slab:** No cost savings are expected for this component (International Energy Conservation Code [IECC] Climate Zones 1–3). IECC climate zones 4–8 prescriptively require slab edge insulation, which would reduce incremental costs.
- **Radiant Floor:** Based on 2011 RSMeans (RSMeans, 2010) pricing, $6,400 was estimated. This is about 65% of the cost incurred at test houses and is justified primarily by reduced labor costs in a production environment. This cost includes the incremental costs for manifolds, pumping, and controls.
- **HVAC Equipment:** The primary cost savings here are attributed to the elimination of the ducted system. The cost savings of $4,100 assume $1,200 for the refrigerant-to-water heat exchanger, $860 for the high efficiency HP over standard efficiency less $6,100 for elimination of ductwork and the air handler (based on BEopt and local contractor pricing) and $100 for HP downsizing by ½ ton due to elimination of load associated with ducts in an attic.

Table 4. Mature Market Incremental Cost Estimates (Dry Climate[a])

Building Component	Proposed Specification	Incremental Cost
Slab	R-10 underslab and edge insulation	$4,000
Radiant Floor, Manifolds, Zone Controls, and Valves	Radiant floor system with 30-gal buffer tank	$6,400
HVAC Equipment	High efficiency HP (12.7 EER, 8.8 HSPF), ductless	($4,100)
Total Incremental Cost		$6,300

[a] Assumes elimination of air handler and ducted distribution.

[7] Benchmark as defined by the *Building America House Simulation Protocols* (Hendron and Engebrecht 2010).

3 Methodology

3.1 General Technical Approach

The general approach of this research plan is to employ system commissioning, short-term tests, long-term monitoring, and detailed analysis of results including model calibration and simulations to identify the performance attributes and cost effectiveness of AWHPs with radiant or MM distribution. Long-term monitoring continued for a minimum of 1 full year. Monitoring commenced in the second quarter of 2011 for the S.E.E.D. house and the third quarter of 2011 for the Cana house and concluded for both projects in the fourth quarter of 2012.

The specific approach for evaluating the efficiency of the two AWHP systems is to measure heating and cooling energy delivered by the HPs, electrical energy consumed, and the seasonal operating conditions under which they are functioning, including outdoor air temperature, indoor air temperature, and HP entering and leaving water temperatures. These data allow for development of equipment performance maps that can be used in modeling, and for comparing performance to conventional systems. Monitoring data are used to calibrate TRNSYS models, which have been used to develop seasonal estimates of energy savings in various climates and under various conditions. Of key interest is whether the fan coils are needed to prevent floor condensation, and whether the mass of the floor slabs can be used to improve performance by shifting times of operation. Indoor temperature and relative humidity (RH) measurements are used to determine the dew point temperature, and the surface temperature of the slab is monitored to determine whether condensation may occur. Supply and return air enthalpies are calculated at both sites to identify latent cooling.

Control settings for the heating, cooling, and ventilation systems were verified, and the operations of the HP, controls, zone valves, fan, and other components were checked. Long-term monitoring was also used to provide “continuous commissioning” and to identify failure of any components.

The specific approach to evaluate the effect of using the floor slab mass to shift cooling operation to night and early morning to improve system performance was tested in the S.E.E.D. home during the 2011 and 2012 cooling seasons. This “cool and coast” (C&C) strategy takes advantage of the exposed floor mass and cooler outdoor nighttime temperatures to operate the HP at more favorable outdoor conditions. Table 5 describes the operating strategies tested and identifies thermostat set points.

Table 5. Evaluated Cooling Strategies

Cooling Strategy	Thermostat Set Point
C&C, Precooling	78°F with 73°F setback from 12:00 a.m. to 6:00 a.m.
Constant Set Point	Fixed 76°F

3.2 Measurements

The two sites are equipped with data loggers and modems for continuously collecting, storing, and transferring data via telephone lines or cellular communications. Sensors are scanned every 15 s, and data are summed or averaged (as appropriate) and stored in data logger memory every 15 min. Automated scripts are used for dialup, data retrieval, range checking, and cleaning. A minimum of 1 year of 15-min interval data was collected for each test site.

3.2.1 Monitoring Data Points

Table 6 and Table 7 list all data points for the S.E.E.D. test house and the Cana test house, respectively. Key HP water side monitoring data points are shown in the piping diagram in Figure 9, and are nearly identical for the two test sites. As indicated in Figure 9, there are some differences in the design of the systems, most notably that the Cana HP produces DHW in addition to hot and chilled water for space conditioning, whereas the Tucson system has stand-alone DHW using a solar water heater with electric resistance backup for domestic water heating. Additionally, the Tucson system has a buffer tank installed in the loop to reduce compressor short cycling during low-load conditions, whereas the Altherma system at the Chico site has an inverter-driven compressor that modulates HP capacity to match the load, eliminating the need for a buffer tank.

Flow and temperature sensors in the piping allow for separately calculating heating and cooling delivery via the radiant floor and the fan coil. In addition to the sensors shown in Figure 9, temperature and RH sensors are included in the supply and return air plenums for calculation of sensible cooling, total cooling from measured enthalpies, and by subtraction, latent cooling supplied by the air system. Condensed water retained on the coil and reevaporated during fan operation makes it difficult to determine whether any latent cooling is actually occurring. A tipping bucket rain gauge installed at the Tucson site measures the volume of any water that leaves the condensate drain.

Indoor temperature and RH sensors are located in the individual zones (two for Cana and three for S.E.E.D.). Each of the houses is also equipped with sensors near the surface of the slab, at the bottom of the slab, and below the underslab insulation. By comparing the floor temperature with the calculated dew point temperature it can be determined whether there is any potential for condensation to occur on the floor and when the floor temperature will be at its lowest. These sensors are also used to estimate the rate of heat transfer from the slab to the ground below.

Table 6. S.E.E.D. House Monitoring Points List

Abbreviation	Description	Location	Sensor Type	Sensor Manufacturer/Model
OAT **RHO**	Temperature, air, outdoor RH, air, outdoor	Northwest side of covered rear patio, in shade, on underside of patio roof	Resistance temperature detector (RTD), 4–20 mA RH, 4–20 mA	R.M. Young 41372LF
TAI1 **RHI1**	Temperature, air, indoor, east RH, air, indoor, east	West wing, next to T1, outside bath 2, mount approx. 4 ft, 6 in. high	RTD, 4–20 mA RH, 4–20 mA	Vaisala HMW60
TAI2 **RHI2**	Temperature, air, indoor, living RH, air, indoor, living	Great room, next to T2, on west wall of dining area, mount approx. 4 ft, 6 in. high	RTD, 4–20 mA RH, 4–20 mA	Vaisala HMW60
TAI3 **RHI3**	Temperature, air, indoor, master bedroom RH, air, indoor, master bedroom	Master bedroom, next to T3, on south wall, mount approx. 4 ft, 6 in. high	RTD, 4–20 mA RH, 4–20 mA	Vaisala HMW60
TAS **RHS**	Temperature, air, air handler (AH) supply RH air, AH supply	Supply plenum, mechanical room	RTD, 4–20 mA RH, 4–20 mA	Vaisala HMD60
TAR **RHR**	Temperature, air, AH return RH air, AH return	Return plenum, mechanical room	RTD, 4–20 mA RH, 4–20 mA	Vaisala HMD60
TWHL	Temperature, water, HP leaving	AH, mechanical room	Immersion TT	Thermex
TWFS	Temperature, water, floor supply	AH, mechanical room	Immersion TT	Thermex
TWHE	Temperature, water, HP return	Mechanical room	Immersion TT	Thermex
TSF1	Floor surface temperature—zone 2	Living, floor surf near t-stat	Contact TT	Omega
TSF2	Slab bottom temperature—zone 2	Above insulation near t-stat	Contact TT	Omega
TSF3	Below-slab insulation temperature—zone 2	Below insulation near t-stat	Contact TT	Omega
EHP	Energy, HP	At outdoor unit	Power meter	Wattnode/ WNB-3D-240-P

Abbreviation	Description	Location	Sensor Type	Sensor Manufacturer/Model
EHSE	Energy, total house	Main service panel	Power meter	Wattnode/ WNA-1P-240P-PV
EFAN	Energy, AH fan	AH, laundry	Power meter	Wattnode/ WNA-1-P-240P
EPV	Energy, PV system	Main service panel	Power meter	Wattnode/ WNA-1P-240P-PV
FWS	Flow, HP system	Mechanical room	Flow meter	Onicon F-1300
EGEN	Energy, house to grid	Main service panel	Power meter	Wattnode/ WNA-1P-240P-PV
FWC	Condensate flow	Mechanical room	RainGauge	
SZ1	Zone 1 status	Mechanical room	Current status meter	Hawkeye
SZ2	Zone 2 status	Mechanical room	Current status meter	Hawkeye
SZ3	Zone 3 status	Mechanical room	Current status meter	Hawkeye

Table 7. Cana House Monitoring Points List

Abbreviation	Description	Location	Sensor Type	Sensor Manufacturer/Model
OAT RHO	Temperature, air, outdoor RH, air, outdoor	Mount on north side of building, mount in shade	RTD, 4–20 mA RH, 4–20 mA	R.M. Young 41372VF
TAI1 RHI1	Temperature, air, indoor, zone 1 RH, air, indoor, zone 1	Near Z1 t-stat	RTD, 4–20 mA RH, 4–20 ma	Vaisala HMW60Y
TAI2 RHI2	Temperature, air, indoor, zone 2 RH, air, indoor, zone 2	Near Z2 t-stat	RTD, 4–20 mA RH, 4–20 mA	Vaisala HMW60Y
TAS RHS	Temperature, air, AH supply RH, air, AH supply	Supply plenum-mechanical room	RTD, 4–20 mA RH, 4–20 mA	GenEastern MRHT3-2-1
TAR RHR	Temperature, air, AH return RH, air, AH return	Return plenum-mechanical room	RTD, 4–20 ma RH, 4–20 mA	GenEastern MRHT3-2-1
TWHL	Temperature, water, HP leaving	Mechanical room	Immersion TT	Thermex
TWFS	Temperature, water, floor supply	Mechanical room	Immersion TT	Thermex
TWHE	Temperature, water, HP returning	Mechanical room	Immersion TT	Thermex
TWCS	Temperature, water, cold water supply	Mechanical room	Immersion TT	Thermex
TWHO	Temperature, water, DHW supply	Mechanical room	Immersion TT	Thermex
TWMS	Temperature, water, master shower hot water supply	Master bath shower hot water supply	Surface TT	Thermex
TSF1	Floor surface temperature—zone 2	Gallery - On floor surface near t-stat	Contact TT	Omega
TSF2	Slab bottom temperature—zone 2	Gallery - Above insulation near t-stat	Contact TT	Omega
TSF3	Below slab insulation temperature—zone 2	Gallery - below insulation near t-stat	Contact TT	Omega
TSF4	Floor surface temperature—zone 1	MBed - on floor surface near t-stat	Contact TT	Omega
TSF5	Slab bottom temperature—zone 1	MBed - above insulation near t-stat	Contact TT	Omega
TSF6	Below slab insulation	MBed - below insulation	Contact TT	Omega

Abbreviation	Description	Location	Sensor Type	Sensor Manufacturer/Model
	temperature—zone 1	near t-stat		
PAS	Pressure, air supply plenum	Supply plenum-mechanical room	Ptd, 4–20 mA	
SDMP	Status, Nightbreeze damper	NB control panel-mechanical room	24 VAC relay	Omron
SZD1	Status, zone 1 damper	NB control panel-mechanical room	24-VAC relay	Omron
SZD2	Status, zone 2 damper	NB control panel-mechanical room	24-VAC relay	Omron
SVDW	Status, valve, DHW three-way	Mechanical room	24-VAC relay	Omron
EFAN	Energy, AH ran	Mechanical room-NB	Power meter	Wattnode/WNA-1P-240P
EHP	Energy, HP	At outdoor unit	Power meter	Wattnode/WNA-1P-240-P
FWS	Flow, HP system	Mechanical room	Flow meter	Onicon F-1300
FWD	Flow, DHW	Mechanical room	Flow meter	Dwyer
SPC	Status, HW recirculation pump	Mechanical room	Current status meter	Hawkeye
EWH	Energy, water heater electric element	Mechanical room	Power meter	Wattnode/WNA-1P-240P
EHSE **EPV** **EGEN**	Energy, total house Energy, PV Energy, generated to grid	Main service panel Main service panel Main service panel	Power meter Power meter Power meter	Wattnode/ WNB-3D-240P(PV)

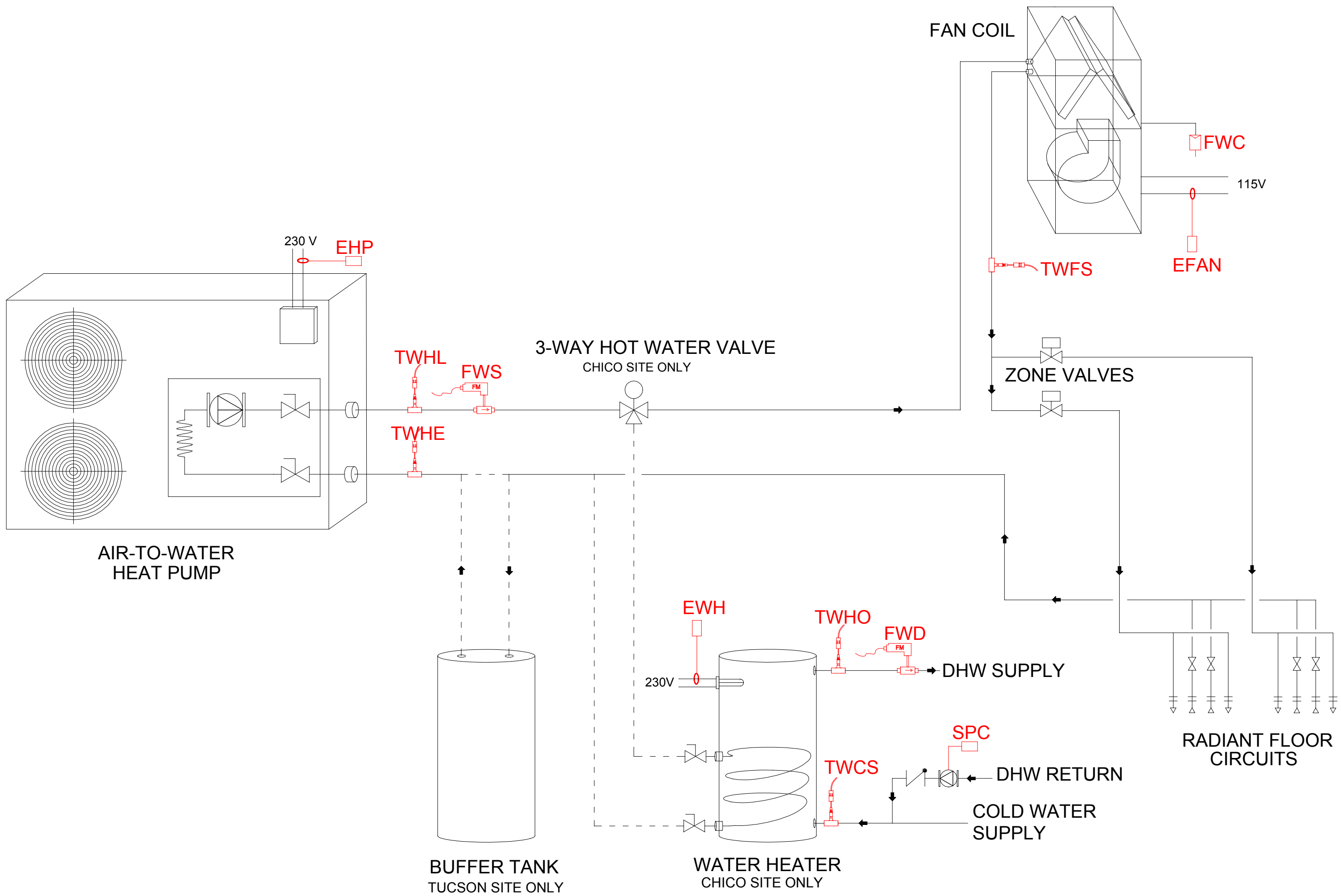

Figure 9. Sensor locations for measuring HP system performance (water side measurements only)

3.2.2 Short-Term Tests

Davis Energy Group and the project Home Energy Rating System (HERS) raters completed short-term tests. These tests are listed below:

- A duct pressurization test measures duct leakage at 25 Pa using an Energy Conservatory duct blaster.
- Air handler air flow is measured using an Energy Conservatory fan flow meter. Data from this test are used to establish a relationship between fan power consumption rates and airflow delivered.
- A water flow test measures flows with different zone valves operating.

A single airflow measurement was made at the Tucson site, which uses a single-speed permanent split capacitor blower motor. Multiple measurements were made at the Chico site to correlate airflow rate with both supply plenum pressure and fan watt draw, the latter of which was monitored continuously. This site uses a variable-speed electrically commutated motor blower, which changes speed depending on cooling stage and modulates torque to maintain a constant airflow.

3.3 Equipment

3.3.1 Data Logger and Sensor Types and Specifications

Data Electronics data loggers are used to collect and store monitoring data. A Model DT-800 is used for both sites. Standard specifications for the sensor types used are listed in Table 8. Sensor selection was based on functionality, accuracy, cost, reliability, and durability. Signal ranges for temperature sensors correspond to listed spans.

Table 8. Sensor Specifications

Type	Application	Mfg/Model	Signal	Span	Accuracy
RTD	Outdoor temperature and RH	R.M. Young 41372LF	4–20 mA	14°–140°F 0%–100%	± 0.5°F + 2% RH
RTD	Indoor/duct temperature/RH	Vaisala HM*60	4–20 mA	23°–131°F 0%–100%	± 0.36°F + 2% RH
Type T Thermocouple	Immersion water temperatures Surface/air temperatures	Gordon Watlow Type T special limits Omega	~11 mV @ 500°F	Range = –328° to 662°F –99° to 500°F	0.4% Special limits of error
24 VAC Relay	Fresh air damper status, zone damper status	Hawkeye	Dry contact	n/a	n/a
Small Power Monitor	Fan and condenser power	WattNode WNA-1-P-240-P	Pulse	Current Transformer Amps (CTA)/40	± 0.5%
Large Power Monitor	Total house power, PV production	Watt Node WNB-3D-240-P	Pulse	CTA/60 CTA/120	± 0.5%
Flow Meter	Water flow	Onicon F-1300	Pulse	Varies by meter	± 0.5%
Pyranometer	Insolation	LiCor	Analog	Varies by sensor	± 5%

3.4 Computation of Monitoring Variables

3.4.1 Heat Pump Performance

Heating and cooling energy delivery are measured using a water flow meter and supply and return temperature sensors. HP delivery efficiencies and seasonal performance for both heating and cooling are then calculated. A performance map of HP sensible and total capacity and power relative to outdoor temperature and supply water temperature is to be developed and compared to manufacturer data.

Total HP heating and cooling delivered is computed by the data logger program on 15-s intervals, using Equation 1. Heating and cooling delivered to the fan coil and the radiant floor system is calculated using water side measurements according to Equation 2 and Equation 3. For the Altherma system the status of the DHW three-way valve, SVDW, is monitored to identify whether the unit is in water heating or space conditioning mode. When in water heating mode all energy is directed to that load and the heating energy delivered to the storage tank is calculated with Equation 1. Values are positive for heat addition (heating) and negative for heat extraction (cooling).

Equation 1: $Q_{HP\_total} = FWS * |TWHL - TWHE| * 8.33\ (Btu)$

Equation 2: $Q_{HP\_fan\ coil} = FWS * |TWHL - TWFS| * 8.33\ (Btu)$

Equation 3: $Q_{HP\_radiant\ floor} = FWS * |TWFS - TWHE| * 8.33\ (Btu)$

where

FWS	= HP system flow (gallons / monitored time period)
TWHL	= HP supply temperature (°F)
TWHE	= HP return temperature (°F)
TWFS	= radiant floor supply temperature (°F).

The value of 8.33 in Equation 1 represents the product of the specific heat of water, 1.0 Btu/°F-lb, and the density of water, 8.33 lb/gal, at a representative water temperature. Over the range of expected temperatures the <0.5% variation is considered to be within acceptable measurement error.

Energy delivered by the fan coil is also calculated using air side measurements. Temperature and RH sensors in supply and return plenums are used to measure sensible, latent, and total cooling delivery by the forced-air components. The density of air is calculated using the supply air temperature in Equation 4. Equation 5 calculates sensible cooling as well as total heating capacity of the fan coil.

Equation 4: $D_{air} = (518.67 / (459.67 + TAS)) * 0.075028$

Equation 5: $\dot{Q}_{air\_sensible} = cfm * |TAS - TAR| * D_{air} * 0.24\ Btu/°F\text{-}lb * 60\ min/h\ (Btu/h)$

where

D_{air}	= density of air at 500 ft altitude (ft^3/lbm)
cfm	= calibrated airflow rate (cubic feet per minute)
TAS	= supply air temperature (°F)
TAR	= return air temperature (°F).

Total fan coil cooling (sensible plus latent) is calculated based on calculated enthalpies of the supply and return air streams. Equation 6 through Equation 10 represent a noniterative approximation of enthalpy based on supply/return air temperature and RH.

Equation 6: $X = (18.678 - T_c / 234.5) * T_c / (T_c + 257.14)$

Equation 7: $Y = 1 + X + 0.5*X^2 + 0.16393*X^3 + 0.041667*X^4 + 0.0123457*X^5$

Equation 8: $P_w = RH * 6.112 * Y / 100$

Equation 9: $W = 0.6219 * P_w / (1013.26 - P_w)$

Equation 10: $h = 0.24 * T_f + W * (1060.9 + 0.443 * T_f)$

where

T_c = supply or return air temperature in °C
T_f = supply or return air temperature in °F
RH = percent relative humidity of the supply or return air
P_w = water vapor partial pressure (hPa)
W = humidity ratio
h = enthalpy of supply or return air (Btu/lbm).

Total cooling load is calculated according to Equation 11.

Equation 11: $\dot{Q}_{total} = CFM * (h_{supply} - h_{return}) \times D_{air}$ *(Btu/h)*

Due to moisture reevaporated from the indoor coil during off cycles, total air-side cooling calculations may not provide an accurate representation of latent cooling. In the Tucson house, to determine whether the fan coil is removing any moisture from indoor air, flow from the condensate drain, FWC, is measured using a tipping bucket rain gauge. Net latent cooling is calculated using Equation 12.

Equation 12: $Q_{lat} = 1080.84 * m_{H2O}$ *(Btu)*

where

m_{H2O} = mass of condensate (lb)
1080.84 = latent heat of condensation at 45°F (Btu/lb).

Total power input to the HP system is the sum of the outdoor compressor unit, the circulation pump, and the indoor fan as shown in Equation 13. In the Tucson house, system pump energy is calculated from the one-time measurement of power draw and filtering power on HP system flow, FWS. The Altherma system uses an internal pump, the energy use of which is captured in total unit energy, EHP (i.e., EPUMP = 0). Equation 14 and Equation 15 are then used to calculate EER and COP, respectively.

Equation 13: E_{hp} = EHP + EFAN + EPUMP (*kWh*)

Equation 14: **Cooling Operation:** $EER_{cooling} = (Q_{HP\_total})_{cool} / E_{hp}$

Equation 15: **Heating Operation:** $COP_{heating} = (Q_{HP\_total})_{heat} / (E_{hp} * 3{,}412)$

where

EHP	= HP energy (kWh)
EFAN	= air handler fan energy (kWh)
EPUMP	= HP circulation pump energy (kWh).

3.4.2 Heat Pump Water Heating Efficiency

For the Chico house, heat delivered to the DHW storage tank by the HP and heat energy supplied from the storage tank are measured. Losses from the hot water recirculation system are treated as part of the DHW load. Electrical energy use from both the HP and the backup electric element is monitored and used to calculate seasonal water heating efficiency. A performance map of capacity and COP relative to delivered water and outside air temperatures is to be developed.

For DHW delivered to the house, the following equation will be used.

Equation 16: $Q_{delivered}$ = FWD * (TWHO – TWCS) * 8.33 *(Btu)*

where

FWD	= DHW flow (gallons/monitored time period)
TWHO	= DHW supply temperature (°F)
TWCS	= cold water supply temperature (°F.)

The status of the DHW three-way valve, SVDW, is monitored to determine when the unit is in water heating mode. Energy consumption of the Altherma and backup element is converted to Btu using Equation 17. Total system efficiency, Equation 18, is calculated using total energy delivered to the house and all the energy inputs into the system. Recirculation pump energy is calculated from the current status meter, SPC, and one-time measurement of pump power draw. Heating delivered from the Altherma to the storage tank can also be quantified with Equation 1.

Equation 17: E_{dhw} = (EHP + EWH + ERPUMP) * 3,412 *(Btu)*

where

EHP	= HP energy (kWh)
EWH	= electric use of water heater (kWh)
EPUMP	= recirculation pump energy (kWh).

Equation 18: Total System Efficiency = $Q_{delivered} / E_{dhw}$

3.4.3 Indoor Comfort and Dew Point

Indoor air temperature and RH are continuously monitored. Temperature sensors are placed below the slab insulation, above the slab insulation, and at the surface of the floor to estimate heat losses and gains across the slab and identify the dew point temperature of the slab.

The dew point of the air in each zone is calculated using interior temperature and RH measurements and Equation 19 and Equation 20. The dew point temperature is then compared to the temperature of the zone floor surface to determine if and how often condensation is occurring at the floor during cooling.

Equation 19: $\alpha = \ln (P_w * 0.01450377)$

Equation 20: $T_{dp} = 100.45 + 33.193 * \alpha + 2.319 * \alpha^2 + 0.17074 * \alpha^3 + 1.2063 * (P_w * 0.01450377)$ ^(0.1984)

Heat loss or gain through the bottom of the slab and the slab perimeter is estimated according to Equation 21 and Equation 22, respectively. The distribution efficiency of the radiant slab is calculated in Equation 23. In cooling mode $Q_{HP\_radiant\ floor}$ will be negative and it is expected that Q_{Cond} will be positive.

Equation 21: Q_{Cond_under} = (A_slab * |TSF3 – TSF2| / R) * *t (Btu)*

Equation 22: Q_{Cond_perim} = (P_slab * |OAT – TSF2| / R) * *t (Btu)*

where

R	= thermal resistance of insulation (h*ft^2*°F/Btu)
A_slab	= total slab area (ft^2)
P_slab	= exposed (insulated) area of slab perimeter (ft^2)
TSF3	= temperature at underside of underslab insulation (°F)
TSF2	= temperature of concrete immediately above underslab insulation (°F)
OAT	= outdoor air temperature (°F)
t	= time (hour).

Equation 23: $\eta_{Dist} = (Q_{HP\_radiant\ floor} + (Q_{Cond\_gain\_under} + Q_{Cond\_gain\_perim}))/ Q_{HP\_radiant\ floor}$

3.4.4 Additional Data

Outdoor temperature and RH are monitored on site. Local weather data including insolation and wind speed are obtained through online sources (e.g., National Weather Service) to qualify the results.

3.5 Modeling Methodology

An AWHP with MM delivery was modeled in TRNSYS v.17. Monitoring data from the S.E.E.D. house were used to calibrate the AWHP model in order to evaluate the feasibility and applicability of such a system in various climates compared to a ducted forced-air system. The calibration process was conducted as described below. Of primary importance was calibration of the mechanical system operation. While calibration of the S.E.E.D. house was only a secondary objective, it was important to calibrate certain components to ensure that the delivery system, primarily the interaction between the slab and the house, was correct. The calibrated model was then utilized to evaluate heating and cooling energy use in four climates via both MM and radiant-only delivery. The energy impact of using a precooling control strategy compared to a standard thermostat control was also evaluated. The base case developed in TRNSYS assumes a minimum efficiency air-to-air HP with ducted distribution located within the attic. Because this AWHP strategy is designed for high performance, low-load homes, all scenarios were evaluated using a high performance, low-load home with the floor plan of the S.E.E.D. house and similar insulation levels.

3.5.1 Calibration Process

3.5.1.1 Heat Pump Calibration

The TRNSYS AWHP model Type 941 was utilized to calibrate performance against monitored data. Type 941 takes a user-specified input file that maps HP power and capacity based on entering air and water temperatures. This performance map was generated using a linear regression tool that looked at full-load monitoring data for the S.E.E.D. house over the 2011 summer. Data from the Cana house were not used because the variable capacity function of the compressor made it difficult to normalize data and develop accurate performance maps, and also because the Type 941 model is only for a single-speed HP. See Appendix A for a description of the linear regression method. The circulation pump was defined as follows according to actual operating specifications (Table 9).

Table 9. HP Flow Parameters

Parameter	Value
Flow Rate	5.6-gpm, one-speed , 150-W pump

The HP model was validated by controlling entering water temperature, outdoor air temperature, and the water flow rate by using monitoring data as inputs to the model, while observing the leaving water temperature and HP power. The results of the analysis showed that over the monitoring period (April 2011–August 2012) the temperature differential across the heat exchanger was within 1.6% of the observed data during heating and 9.7% during cooling. The modeled power was on average 8% different in heating operation and 1% different in cooling operation. Samples of heating and cooling events are graphed in Figure 10 and show the leaving water temperature (LWT) and power (PWR) for given outdoor air temperature (OAT) and entering water temperature (EWT).

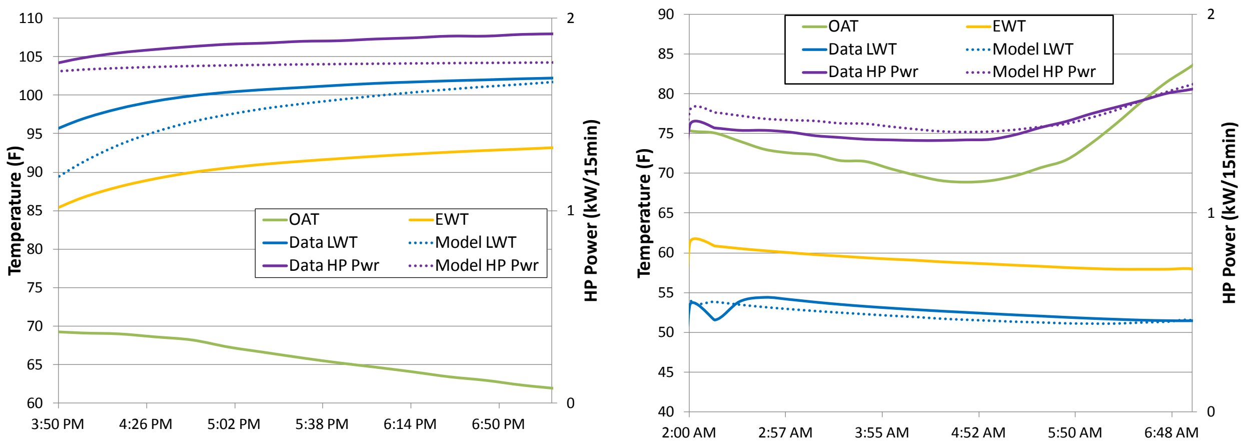

Figure 10. HP calibration step showing alignment between model and monitoring data for a heating event (left) and cooling event (right)

The temperature variation between modeled and observed at the beginning of the heating event is partially due to the data being truncated to only include full-load events over a 15-min period. In reality the HP did have some part-load run time that "warmed up" the LWT. This cannot be captured in the model on a 15-min simulation time step. Smaller time steps were used in later analysis to better evaluate the impacts of part-load performance.

3.5.1.2 Fan Coil Calibration

The coil is located within conditioned space and is set to operate whenever there is a call for cooling. The fan coil bypass fraction was modified in order to achieve similar water and air temperature splits as was observed during the monitored operation. Table 10 lists the parameters that define the validated model.

Table 10. Fan Coil Parameters

Parameter	Value
Fan Flow Rate	831 cfm
Fan Power	0.25 kW

3.5.1.3 House and Slab Calibration

A single-zone Type-56 model house in TRNSYS was constructed to approximate the construction characteristics of the S.E.E.D. house. Operating characteristics such as occupancy and internal loads from the monitored period were used in the initial calibration. The house model was calibrated by modifying radiant floor characteristics, internal gains (latent and sensible), and house capacitance. Ultimately these variables were chosen to minimize the deviations between modeled and observed interior and slab exiting loop temperatures. The monitored flow rate and inlet temperature were used as inputs to drive heating and cooling events for the radiant floor model, and actual meteorological year (AMY) data, from August 2011 through July 2012, were used in the model.

While details of the radiant floor design are known, pipe spacing and depth of pipes in the slab were varied to produce slab temperature responses similar to those seen in monitored data. Table 11 lists the initial and post calibration specifications.

Table 11. Slab and House Characteristics

Parameter	Initial Specification	Postcalibration Specification
Slab Depth	8 in.	8 in.
Radiant Floor Spacing	12 in. on-center, tubes 4 in. below slab surface	15 in. on-center, tubes 5 in. below slab surface
House Capacity	5,986 kJ/°K	8,986 kJ/°K

In Figure 11 the slab exiting water temperature during a series of heating and cooling events is shown. During this calibration step EWT, OAT, and water flow rate remain controlled by inputs from monitoring data. This comparison was used to calibrate modeled energy delivered to the slab with monitored data. Throughout the monitoring period, occupancy, thermostat set points, window operation, energy recovery ventilator (ERV) operation, and cooling mode (radiant floor versus fan coil delivery) varied. The slab calibration focused on specific monitoring periods that were identified to have only radiant floor delivery with documented interior set points.

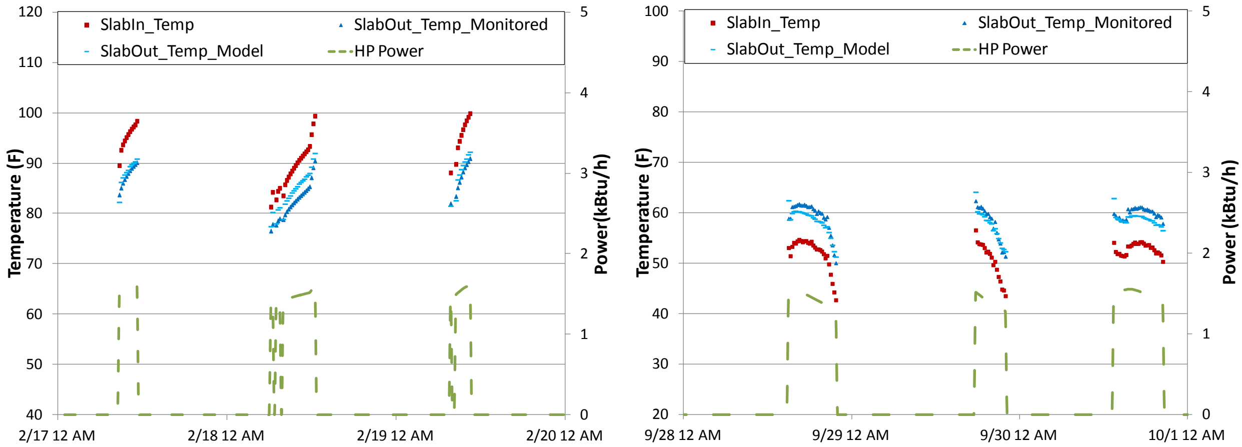

Figure 11. Heating event (left) and cooling (right) temperature comparison during flow events

The house capacitance was adjusted to minimize the deviations between interior air temperatures and slab loop exiting water temperatures both when the house was "floating" and during radiant space conditioning events. Floating implies that the heating and cooling systems are inactive and the house simply responds to exterior conditions and internal loads. Initially, the house in TRNSYS responded quicker to changes in environmental conditions than reflected in monitoring data. An increase in capacitance of the house airnode (to 8,986 kJ/K) dampened and delayed the house's response, resulting in a model that produced similar daily temperature change and coincident peak temperature times as those monitored. Table 11 lists slab and house characteristics before and after this calibration step.

Figure 12 shows monitored and modeled house interior temperature over 15-min simulation time steps. The green line indicates when cooling events occur and it can be seen that the modeled interior temperature of the house responds similarly to the monitored data during these events.

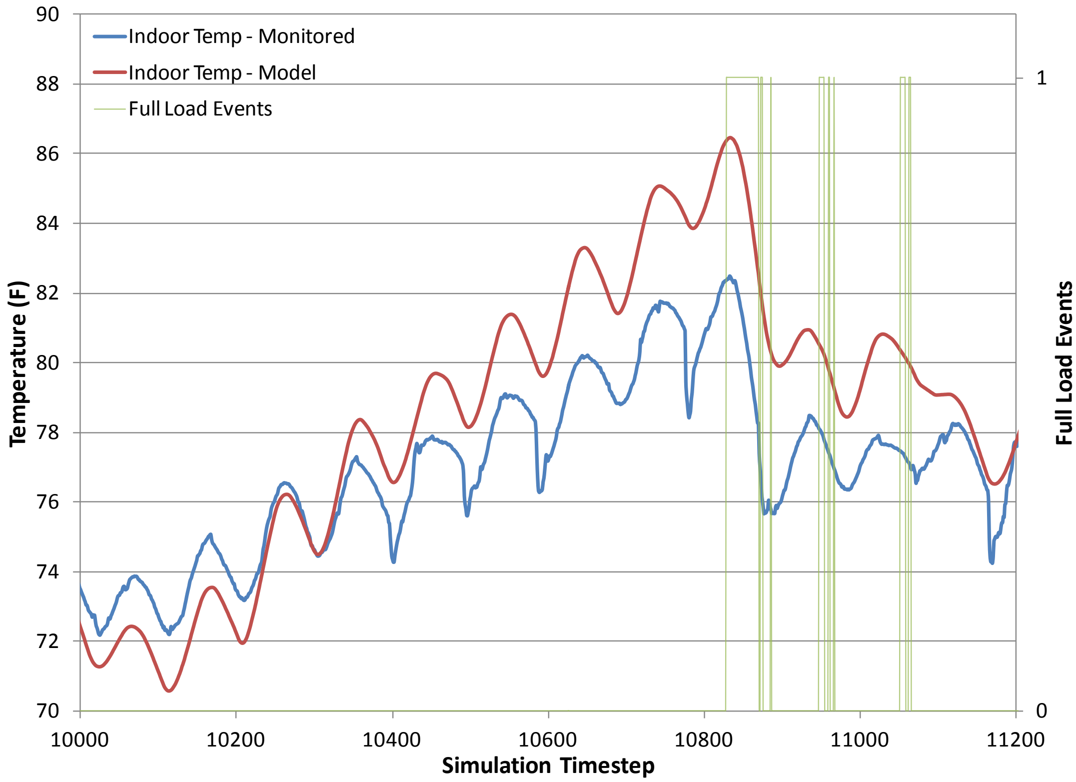

Figure 12. Interior temperature comparison

At this point in the calibration process the controlled variables (EWT, OAT, and flow rate) are no longer driven by monitoring data and the model is allowed to operate based on the imposed loads and the thermostat set point. Heating and cooling performance were compared on a seasonal basis, using AMY data in the TRNSYS model. Heating and cooling events were defined as steady-state operation of the HP over a minimum time period of ~45 min. The response of the slab over the course of these events was evaluated and compared for the model and monitoring data. Figure 13 demonstrates similar changes in slab temperature over an event for a given amount of delivered energy. The modeled data exhibit a greater number of events delivering higher cooling energy than the monitoring data, largely a result of HP cycling in the monitored data. If the HP ran for 2 h, then stopped for 15 min and ran again for 2 h more, this registered as two small events with correspondingly smaller cooling energy delivered and slab temperature change. Long run times were observed in both the model and the monitored data during radiant-only delivery on very hot days due to the slab's slow response. However, HP run times in the TRNSYS model were longer than in the monitoring data on these hot days. This may suggest that the house internal loads assumed in the model are greater than those in actuality. Detailed information on occupant loads and load scheduling was very difficult to ascertain. As part of the calibration process, the internal loads in the model were adjusted to provide the best match to the monitored slab response.

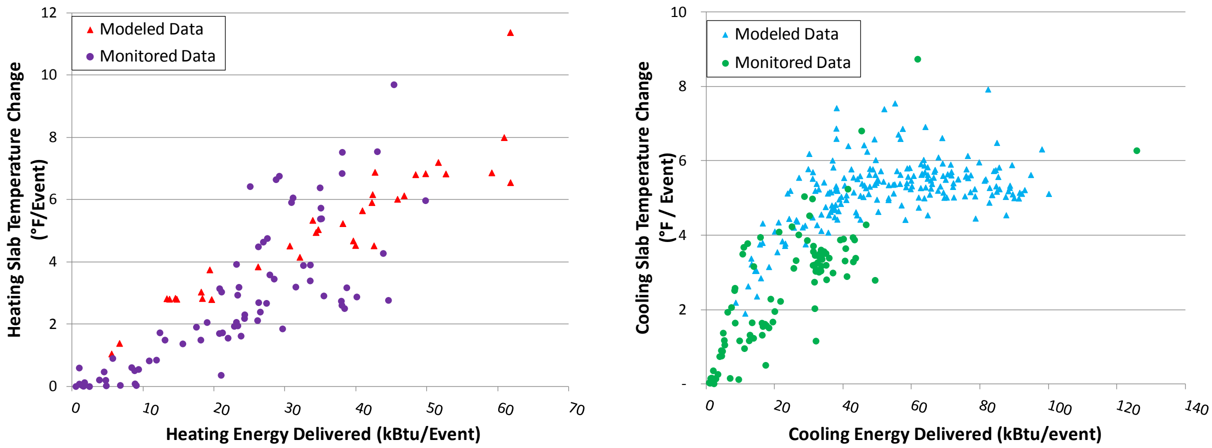

Figure 13. Energy input required to change slab temperature for heating (left) and cooling (right)

3.5.2 Climate Zone Evaluation

Using the calibrated model, the four cities shown in Table 12 were selected as representative of the major climate zones where the AWHP MM strategy may be appropriate.

A high performance, low-load home was utilized for this analysis with the properties presented in Table 13. This house is based on the S.E.E.D. house built for TRNSYS. HP equipment sizing was adjusted by climate based on Air Conditioning Contractors of America Manual J load calculations for each climate zone (CZ) (ACCA 2006). Design heating and cooling loads are shown in Table 14.

Table 12. Evaluated Cities and Climate Zones

IECC and BA CZ	Representative City
CZ 2 Hot-Dry	Tucson
CZ 3 Hot-Dry	Sacramento
CZ 5 Cold	Denver
CZ 2 Hot-Humid	Houston

Table 13. House Characteristics as Modeled in TRNSYS

	Specification
Conditioned Floor Area	1,935 ft2
Walls	R-23 equivalent (U = 0.25 W/m2-K)
Roof	R-49 (U = 0.114 W/m2-K) White (0.2 absorptivity)
Infiltration	5.2 ACH50 (0.0003 specific leakage area@ 50 Pa)
Ventilation	50 cfm, per ASHRAE 62.2
Internal Gains	Occupants: 2 @ 13,008 Btu/day (57% sensible) Lights: 100% compact fluorescent: 7,572 Btu/day (100% sensible) Miscellaneous electric load /appliance load: 28,475 Btu/day (84% latent) Total: 49,055 Btu/daya

[a] Total daily internal gain was calculated as 20,000 + 15*CFA (Btu/day) from the CEC Title-24 2008 Alternative Calculation Methodology (ACM) (CEC 2010). Breakdown was taken from a combination of the ACM and the Building America House Simulation Protocols (Hendron and Engebrecht 2010).

Table 14. Design Heating and Cooling Loads (Btu/h)

	Tucson	Denver	Houston	Sacramento
Heating Design Load	13,449	25,340	14,890	13,993
Cooling Design Load	16,471	13,777	14,735	14,913

The base case system for this analysis was an air-to-air HP with forced-air delivery. The duct system was assumed to be located in a vented unconditioned attic. The duct leakage was assumed to be 6% of total air flow. HP cooling and heating efficiencies are 11 EER and 2.8 COP, respectively, and are based on the BA benchmark HP. Two proposed alternative HVAC systems were evaluated using an AWHP and a circulation pump. The first alternative uses radiant floor distribution only. The second uses the MM distribution design and therefore includes the hydronic fan coil downstream of the HP and before the radiant floor. Both alternative system designs were run with and without external dehumidification. Table 15 lists the equipment specifications used in this analysis.

Table 15. Equipment Assumptions Used in the TRNSYS Model

Equipment	Specification	Value	Unit
Hydronic Fan Coil	Fan efficacy	0.33	W/cfm
Circulation Pump	Pump flow Pump power	5.6 150	Gpm W
AWHP[a]	Rated cooling EER Rated heating COP	10.4 3.1	Btu/Wh W/W
Dehumidifier	Efficiency rating	1.2	L/kWh

[a] Rated conditions were selected to match AHRI conditions for air-to-air HP as best as possible.
Cooling: 95°F outdoor air dry bulb and 75°F entering evaporator water temperature.
Heating: 45°F outdoor air dry bulb and 85°F entering condenser water temperature.

Two control strategies were investigated in this study: 1) constant heating/cooling set point control and 2) C&C. In the first case, fixed thermostat set points of 71°F for heating and 76°F for cooling were assumed. The C&C strategy utilized the same heating set point (71°F) but during the cooling season, the thermostat was set to precool the house overnight. The thermostat was set back to 73°F at night (12 a.m. to 6 a.m.) and up to 78°F during the day (6 a.m. to 12 a.m.). While both heating and cooling were available anytime throughout the year, C&C operation was restricted to May 1 through October 31.

4 Results

This section presents monitoring results from the two test houses, as well as TRNSYS modeling results. Results from the commissioning process are presented in Appendix B.

4.1 Monitoring Results and Discussion

This section reports on monitored performance of the AWHP systems at both test houses. All presented results are based on full-load data. Full load for all analyses in this report is defined as whenever the HP was running for a minimum of 13 min in steady-state operation. Unless otherwise noted, calculated efficiencies include HP compressor and pumping energy. Where appropriate, fan energy is also included for MM cooling operation.

4.1.1 Heating Performance

Figure 14 shows calculated full load heating COP of the HP (outdoor unit + circulation pump) at the S.E.E.D. house compared to both OAT and EWT. Average seasonal COP over the monitoring period of the 2011–2012 heating season was 3.26.

For reference, a comparison is made to the published engineering data for the standard (air-to-air) Ruud HP (data points are referenced to entering air temperature [EAT] instead of EWT). In the curve versus OAT an average entering air temperature of 74°F was selected to represent typical air-to-air HP operation. The manufacturer data shown represent a best-case scenario for supply air systems based on research that has shown field performance of HPs to be much lower than expected from unit ratings (Proctor et al. 2011). There is substantial spread in the monitoring data, which may be explained by the variation in water temperatures in the supply loop due to load variations and zoning (there are three zones). The data presented are relative to each dependent variable; for example, the data relative to OAT are not normalized to a constant EAT.

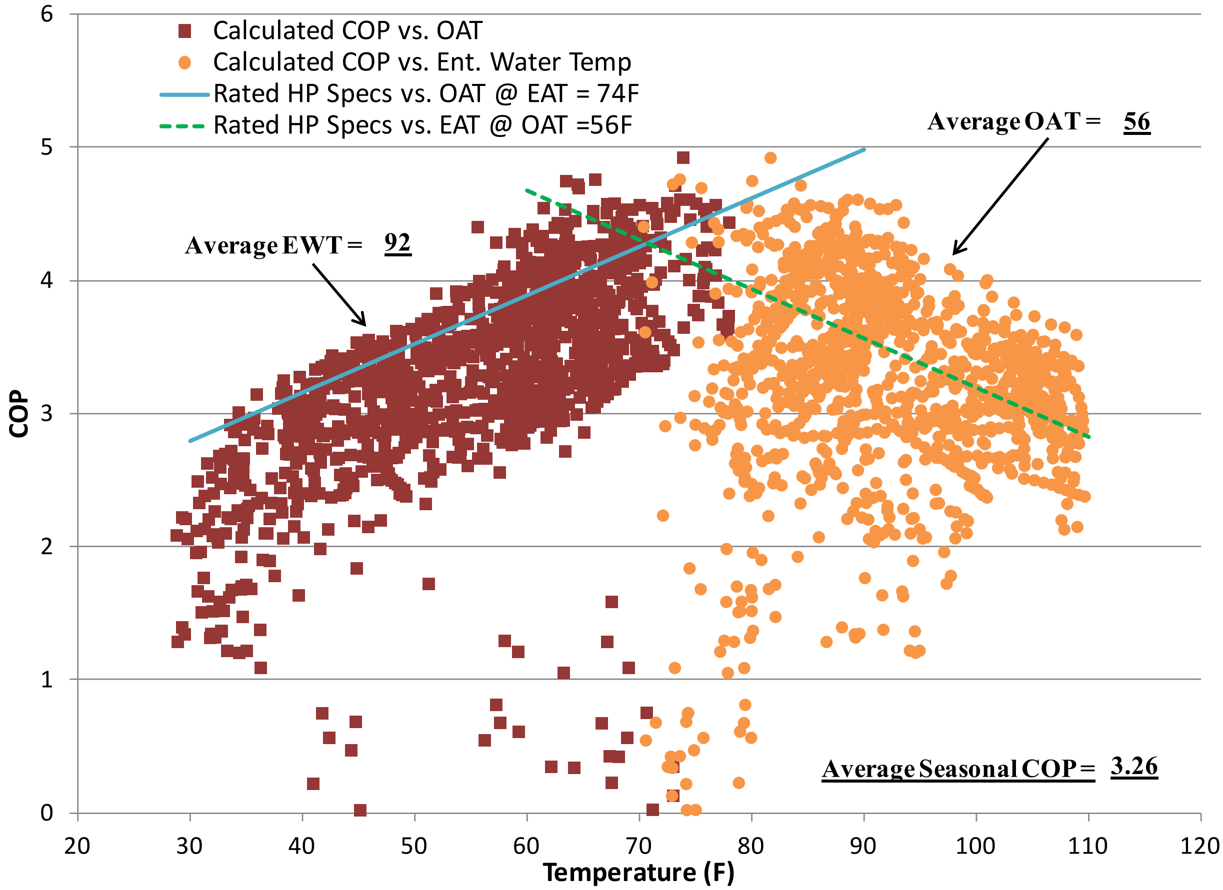

Figure 14. Calculated full load COP of the S.E.E.D. house HP in space heating versus outdoor dry-bulb temperature and EWT and compared to manufacturer-rated specifications (n = 1,241)

Figure 15 shows calculated full load heating COP at the Cana house compared to both outdoor air and LWT.[8] The chart also includes curves representing the Altherma engineering specifications at various operating conditions. Field performance of the Altherma tracks laboratory performance quite well with respect to OATs. Monitoring data show a strong correlation between efficiency and outdoor air and little correlation between efficiency and load conditions (LWT). This is partially a factor of the narrower band of temperature conditions on the load side as well as the variable capacity of the inverter-driven compressor. The Altherma is an inverter-driven compressor that adjusts compressor frequency and ultimately output capacity based on load. If the water temperature is overshooting the set point the compressor will throttle down, and likewise if it is falling short it will ramp up. This allows the system to better supply the building load while improving occupant comfort and maintaining efficiencies by reducing the temperature lift and reducing cycling. While the Altherma has the capability for outdoor air reset in heating mode, this functionality was not enabled for this data monitoring period. The Altherma specifications are generated for high-speed operation only, and do not reflect

[8] Note that Altherma engineering tables are related to LWT when performance is ultimately dependent on EWT.

performance at various compressor frequencies. If compressor frequency could have been captured within the monitoring data, the information may have led to a more defined trend with respect to LWT and improved correlation with the Altherma specifications.

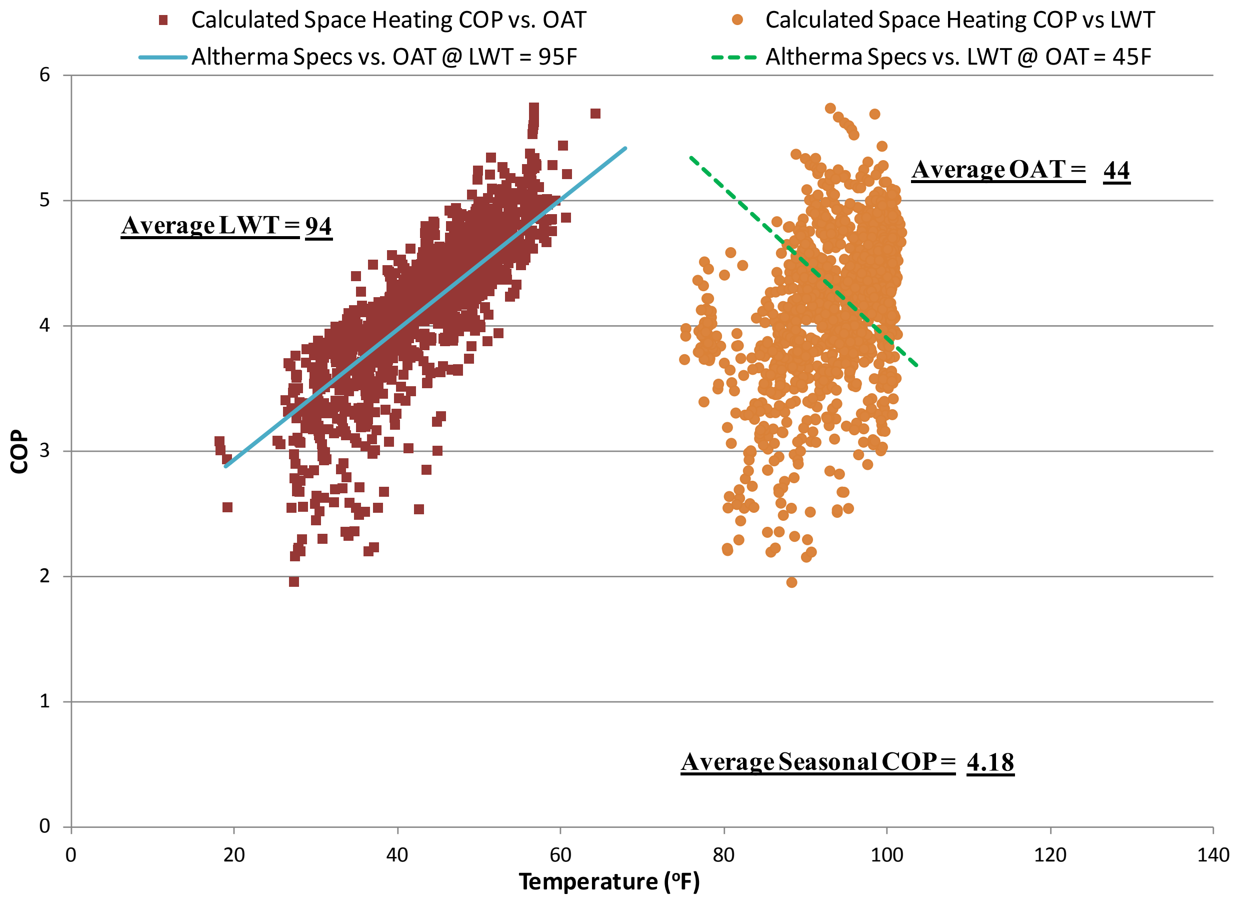

Figure 15. Calculated full load COP of the Cana house HP in heating mode versus outdoor dry-bulb temperature and LWT and compared to manufacturer-rated specifications (n = 1383)

4.1.2 Domestic Hot Water Performance

Figure 16 shows heat pump COPs at the Cana house for DHW operation that range from approximately 1 to 3. These COPs are considerably below manufacturer-rated data with significant variation in the data set (see Altherma specs in figure). Poor heat transfer from the HP to the storage tank was observed and is thought to be the cause of lower performance. Temperature differentials of only 2°F were seen on average across the DHW heat exchanger coil, resulting in the low heating capacities. As a result, the HP was not able to meet the storage tank set point of 130°F, and the auxiliary electric resistance heat was activated to satisfy water heating demand. ARBI worked with the Altherma representative and the installer to identify the cause of the low performance, and several program (operational) adjustments have been made to the Altherma unit based on initial monitoring results (including lowering the tank set point). The 4-ton Altherma unit is rated at a water flow rate of 10.6 gpm. Monitoring data showed that the average flow rate through the storage tank was 9.8 gpm. With just 15.6 ft^2 of surface area,[9] the DHW tank heat exchanger is undersized compared to other indirect water heaters. It is hypothesized that this combination of high flow rate and small effective area is the primary cause of reduced efficiencies. Because of the low delta-T through the heat exchanger, the inverter-driven compressor ramps down its capacity. Over the 13- month monitoring period, data showed that the HP supplied 76% of the energy to the DHW tank, just slightly lower than the 78% estimate from the Altherma engineering manual for a tank set point temperature of 120°F. Seasonal total system COP[10] over 1 year (November 2011 – October 2012) was 1.05. This low value was a result of electric resistance heat operation, poor HP efficiencies, and storage losses.

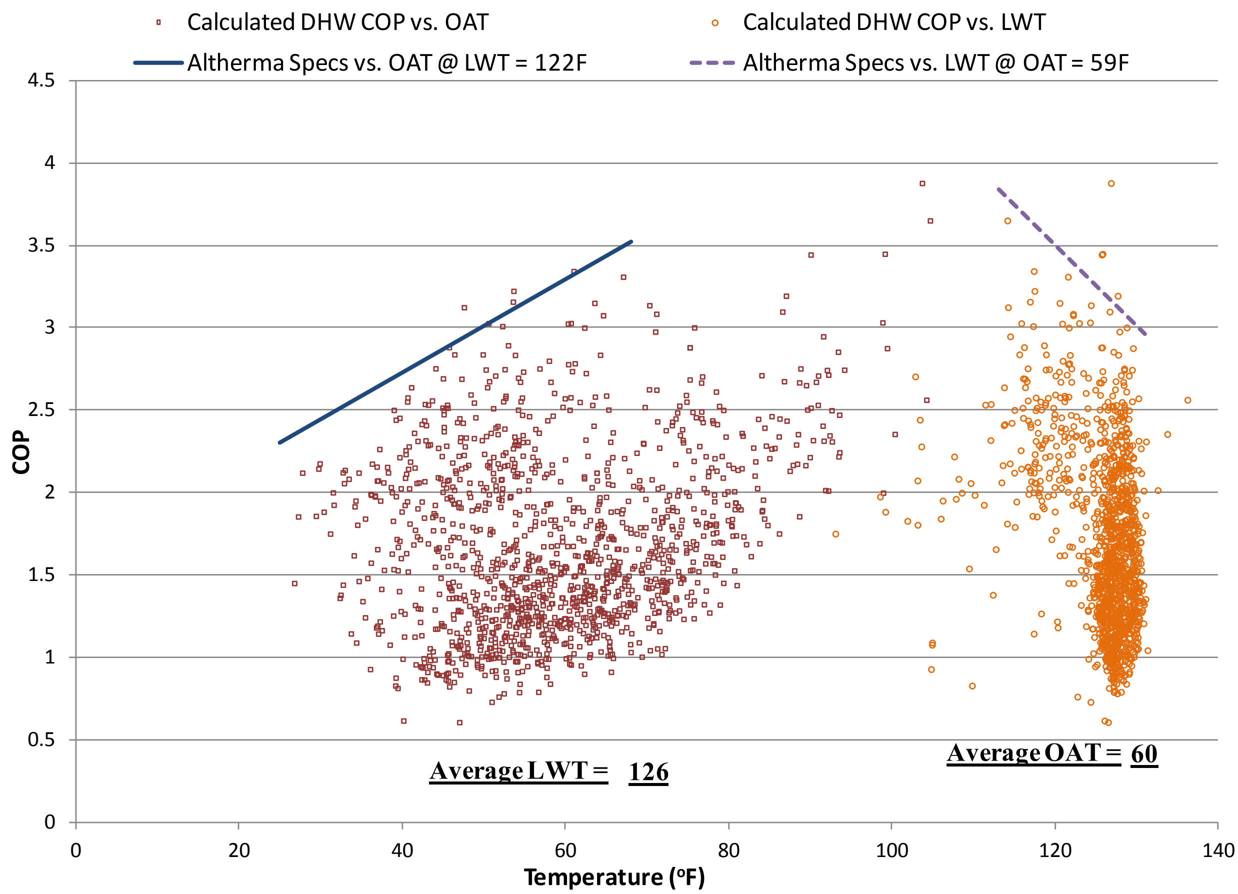

Figure 16. Calculated full load COP of the Cana house HP in DHW mode versus outdoor dry-bulb temperature and LWT and compared to manufacturer-rated specifications (n = 1607)

4.1.3 Cooling Performance

Heat pump EER using water side energy delivery and total electricity input including condenser and pump energy use, relative to operating conditions during full load operation, were reviewed to identify performance and operating efficiency. Figure 17 and Figure 18 show calculated full load EER at the S.E.E.D. house and Cana house, respectively, compared to both OAT and EWT

[9] See http://www.daikinac.com/content/residential/whole-house/daikin-altherma/faq-2/

[10] Recovery load divided by electrical input to both the HP and the backup electric resistance heater

for the S.E.E.D. system and LWT for the Cana system.[11] The chart also includes curves representing the manufacturer's engineering specifications at various operating conditions.

The S.E.E.D. house data illustrate a strong correlation between efficiency and OAT. However, as was observed in the space heating data, performance is not as dependent on load-side conditions (EWT). Supply water temperatures within the hydronic loop continuously decreased throughout cooling events. This resulted in reduced efficiencies over periods of prolonged cooling operation due to lower return water temperatures to the HP. In addition, latent cooling was delayed until the loop reached the indoor air dew point temperature. This partially explains the spread in the data for a given OAT.

A comparison to published engineering data for the Ruud HP is also shown in Figure 17. Manufacturer's data assume that the HP is air-to-air and paired with a standard air handler and all data points are referenced to entering air wet-bulb temperature (EWBT) instead of EWT. In the curve of rated performance versus OAT an average entering air wet-bulb temperature of 60°F and dry-bulb temperature of 78°F were selected to represent typical air-to-air HP operation in a dry climate.

Because of cooler evenings and the effect of nighttime ventilation cooling at the Cana house, the Cana cooling hours of operation were significantly reduced as compared to the S.E.E.D. house. Perhaps due to the small sample size it is difficult to make a comparison to the Altherma engineering specifications. However, on average efficiencies are close or higher than rated values.

Results from a 2009 California study show measured equipment efficiencies well below manufacturer-rated efficiencies (Proctor et al. 2011). The study evaluated 80 residential single-family and multifamily units built and occupied in 2007. Efficiency and capacity measurements were taken on ten of these systems before and after refrigerant and airflow repairs were made as appropriate. Sensible EERs (corrected to standard AHRI conditioned) ranged from 4 to 7 prior to repairs. After repairs, efficiency increase was on average 24% with normalized sensible EERs ranging from 5.5 to 8.5.

[11] The Altherma engineering tables present LWT when performance is ultimately dependent on EWT.

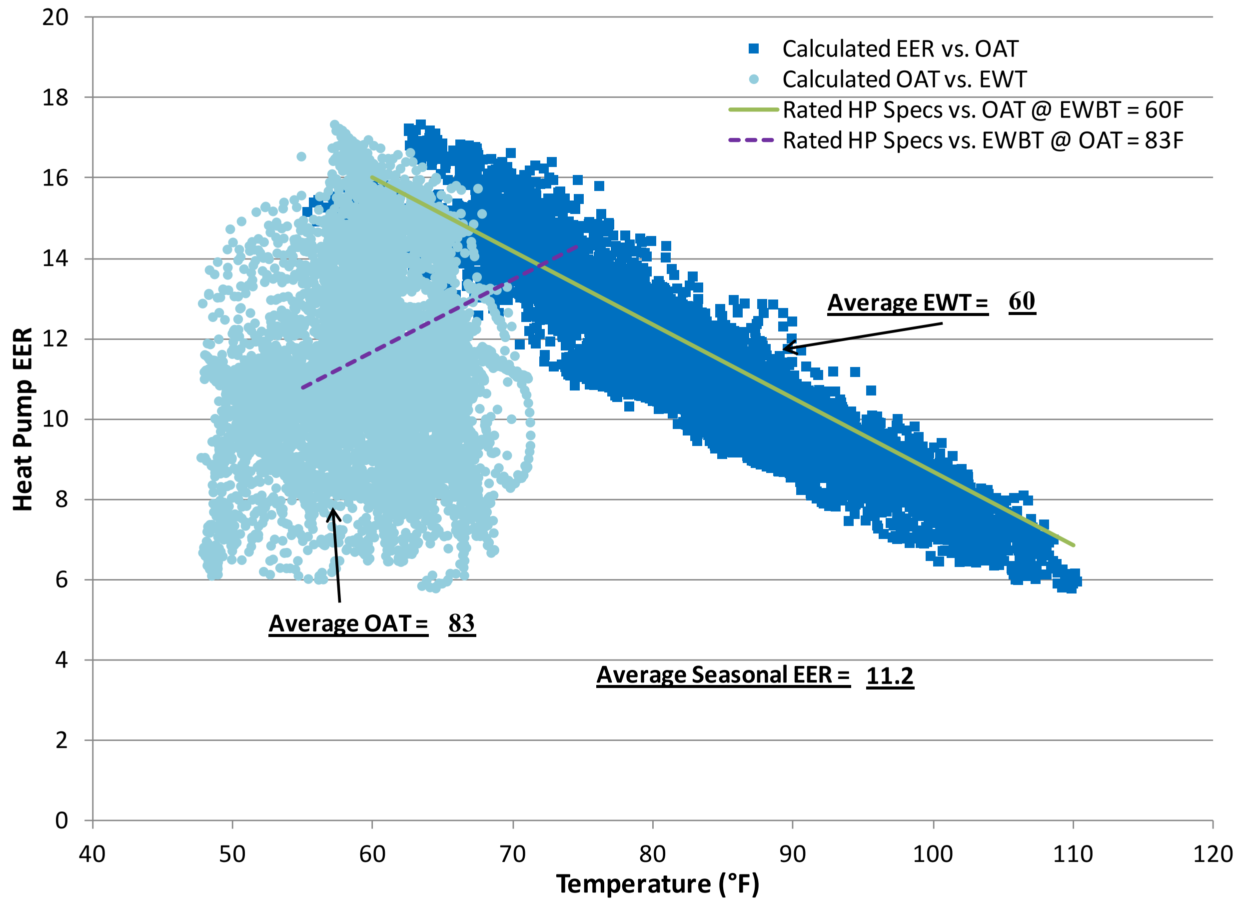

Figure 17. Calculated full load EER of the S.E.E.D. house HP (condenser + pump) in space cooling versus outdoor dry-bulb temperature and LWT and compared to manufacturer-rated specifications (n = 8,208)

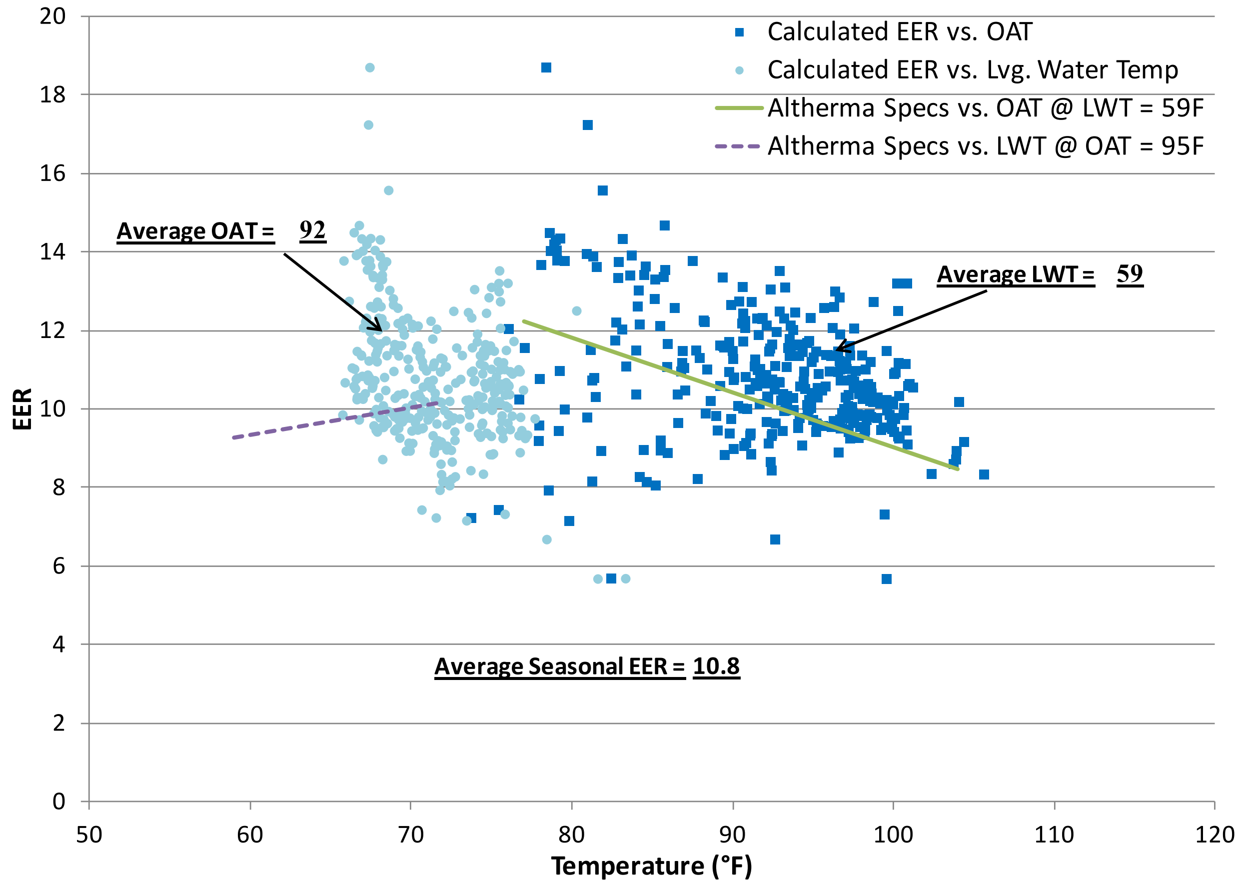

Figure 18. Calculated full-load EER of the Cana house HP (condenser + pump) in space cooling versus outdoor dry-bulb temperature and LWT and compared to manufacturer-rated specifications (n = 337)

4.1.4 Nighttime Precooling Strategy

The C&C precooling strategy was tested at the S.E.E.D. house to evaluate its effectiveness in reducing cooling energy use by improving HP efficiencies and shifting load from on to off peak. Figure 19 shows a comparison of daily average EER[12] and daily HVAC energy use as a function of maximum daily OAT for the two control strategies operating in radiant floor delivery mode (no fan coil).

Shifting C&C air-conditioner operation to cool nighttime periods results in higher average efficiencies than with constant set point operation and a much less performance degradation impact with outdoor temperature. Daily C&C energy use is clearly lower than constant set point operation, with an average 27% savings at a daily maximum outdoor temperature of 90°F and up to 40% savings at maximum temperatures greater than 100°F. However, below a certain OAT, C&C operation can result in HVAC energy use that may have been avoided by conventional

[12] Calculated as the average of all full-load operating points during a 24-h daytime period. Average temperatures within the living space under all scenarios remain relatively constant at close to 76°F. However, results are normalized to a daily internal average temperature of 76°F.

thermostat operation. Overcooling during milder conditions could be avoided by including a control strategy that monitors weather conditions and alters the nighttime setback based on the anticipated cooling load for the next day.

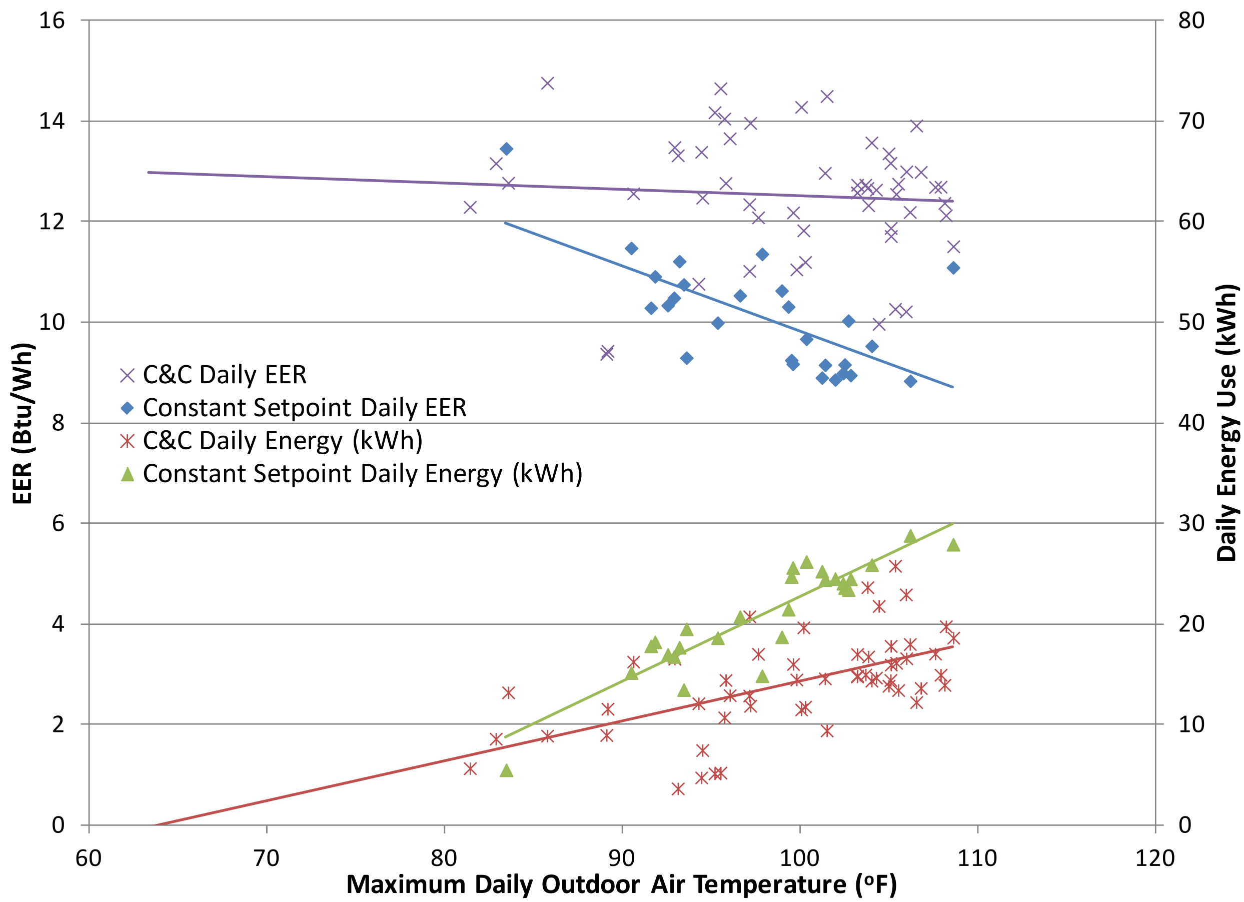

Figure 19. Daily average EER and energy use comparison for C&C (n = 82) versus constant set point (n = 30) operating strategies in radiant floor delivery mode

Figure 20 compares indoor and outdoor temperature and HP power for the constant set point mode of operation and the C&C mode. The shape of the HP electrical demand profile during the course of the day shows the performance sensitivity to temperature with a roughly 30% increase in cooling demand from the nighttime minimum to the daytime maximum. Also, running the HP for extended periods of time to satisfy the reduced temperature setback reduces cycling, thus increasing steady-state operation and ultimately system efficiency. As a result of the long evening run time and the associated low nighttime cooling loads, the HP EWTs are often lower than during peak daytime operation. EWTs gradually decline over the run period, increasing the thermal lift on the HP and lowering efficiency slightly. However, any reduction in efficiency due to this effect is overshadowed by the more favorable OATs and reduced cycling. This result corresponds well with the efficiency curves presented in Figure 17 and Figure 18, which demonstrate that performance is most influenced by source conditions.

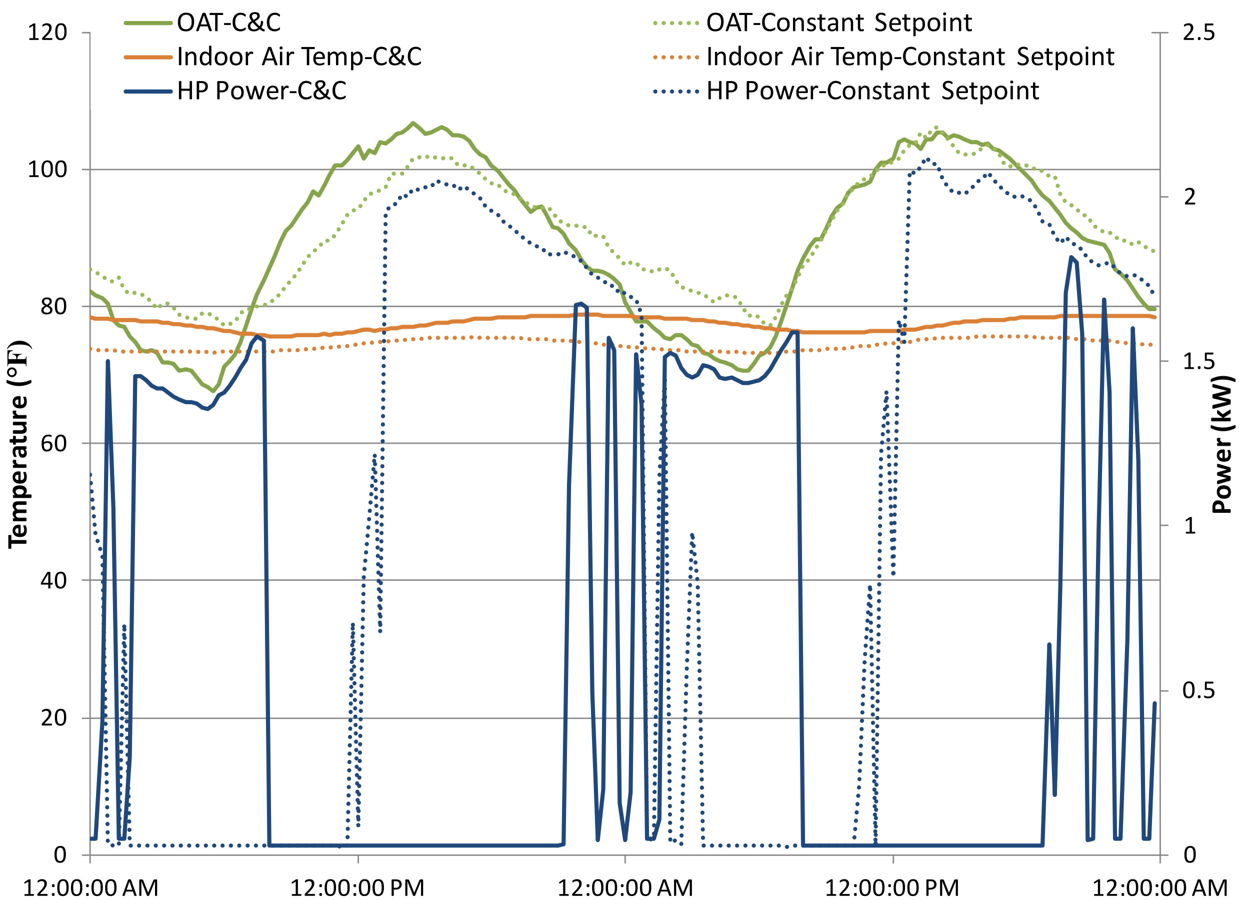

Figure 20. Interior temperature and HP operation for two hot days comparing C&C versus constant set point operating strategies in radiant floor delivery mode

4.1.5 Distribution System Performance

The total distribution efficiency of radiant floor delivery was calculated based on estimated heat losses and gains between the slab and ground and the exposed slab perimeter and ambient air (see Equation 21 and Equation 23). Pipe losses from pipes outside of the slab and conditioned space were not considered in this evaluation. Slab temperature was also assumed to be uniform throughout the slab. Figure 21 shows the radiant slab distribution efficiency in both heating and cooling for the two test houses over the monitoring periods. Data points represent calculated efficiencies based on 15-min data intervals during full-load operation.

Heat losses and gains through the slab at the Cana house are driven by the ground temperature given the larger area for heat exchange and reduced underslab insulation (R-5) as compared to slab edge conditions (R-8). During the summertime there were many instances when ground temperatures were lower than that of the slab, which actually contributed to slab cooling, albeit negligibly since temperature differentials across the underslab insulation were minimal. Because of cool ground temperatures and minimal compressor cooling operation, average seasonal cooling distribution efficiency at the Cana house was greater than 100%

Average seasonal distribution efficiencies for both the Cana and S.E.E.D. test houses are presented in Table 16 and compared to ducted air supply systems. With the exception of the Cana house during cooling operation, the measured distribution efficiencies are roughly equivalent to a ducted distribution system with ductwork located inside conditioned space. Distribution efficiencies are improved by almost 50% from the benchmark case with ductwork in the attic at a 15% leakage rate.

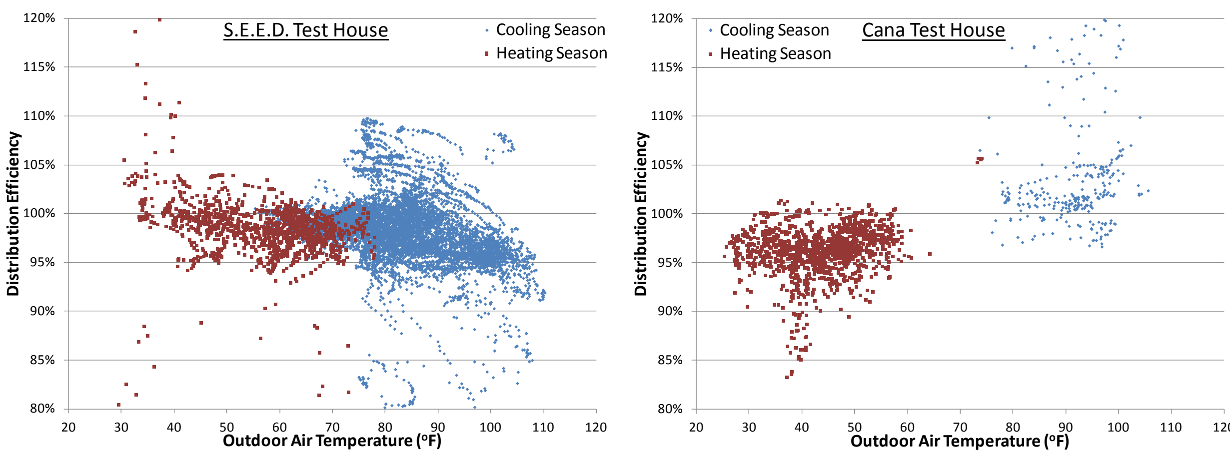

Figure 21. Radiant floor distribution efficiency versus outdoor dry-bulb temperature for S.E.E.D house (left) and Cana house (right)

Table 16. Distribution Efficiency Comparison

Case		Distribution Efficiency			
Monitoring Data		**Heating**		**Cooling**	
		S.E.E.D.	**Cana**	**S.E.E.D.**	**Cana**
		96%	**96%**	**97%**	**108%**
Model Results[a]	Benchmark forced-air unit (15% leakage)	68%		61%	
	Benchmark w/tight ducts (6% leakage)	78%		74%	
	Benchmark w/ducts in conditioned space	94%		94%	

[a] Seasonal distribution efficiencies are calculated according to ASHRAE Standard 152 (ASHRAE, 2004) assuming a 2,400-ft^2 house in Sacramento.

Performance of the AWHP was also compared using radiant-floor-only distribution and MM distribution. This comparison is presented in Figure 22, which demonstrates clear energy savings from removal of the fan coil. MM operation provides latent cooling ability and the potential for improved compressor efficiency by raising entering HP water temperatures and reducing thermal lift, but parasitic energy use from the fan in MM resulted in higher daily energy use for cooling relative to floor cooling delivery alone.

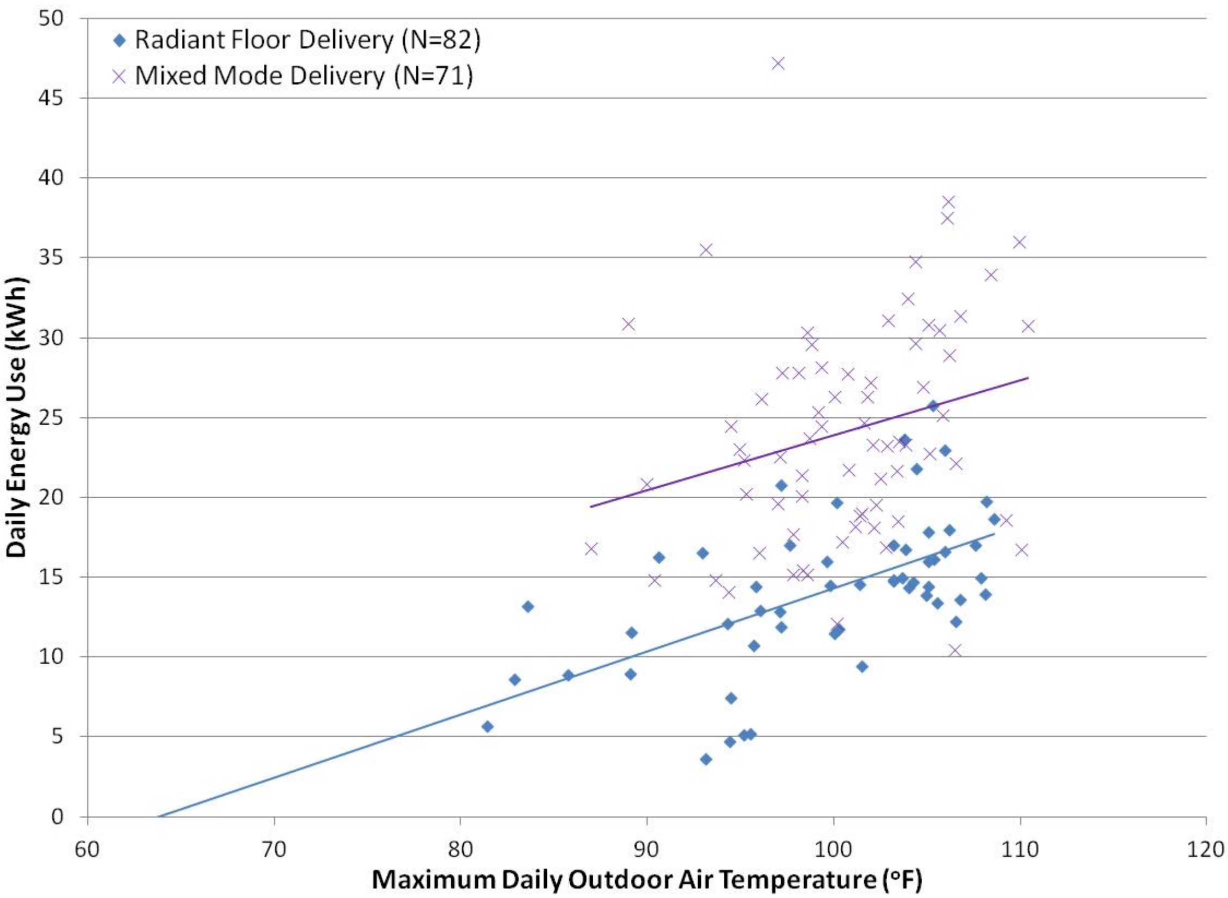

Figure 22. Daily average HP (condenser + pump + fan) energy use comparison for radiant floor delivery (n = 82) versus MM delivery (n = 71) modes in C&C operation

4.1.6 Latent Cooling, Dehumidification, and Condensation

To evaluate the potential and risk for condensation on the floor, the temperature of the slab surface was monitored inside the living room. The dew point temperature of the air inside the zone is calculated and compared to the slab surface temperature (see Equation 19 and Equation 20). All cooling operation at the Cana house was with the fan coil active; therefore, ARBI was unable to test conditions with radiant delivery only. However, during cooling operation the indoor air dew point remained on average 17°F above the floor temperature with a minimum difference of 8°F, and interior RH remained within the ASHRAE comfort range.[13] These data suggest that in a very dry climate such as Chico floor condensation in a radiant floor cooling application should not be a concern as long as internal latent loads from cooking and showers are adequately removed with intermittent ventilation. High latent loads could potentially present a concern if not properly addressed. While not presented in this report, radiant floor cooling has been successfully demonstrated in other California research projects within hot-dry climates (CEC 2003).

[13] ASHRAE *Standard 55-2010: Thermal Environmental Conditions for Human Occupancy* allows a wider range from about 25%– to 80% (ASHRAE 2010).

In climates with increased levels of humidity during the summertime, such as in Tucson, which experiences a summer monsoon season, some level of dehumidification, primarily for occupant comfort, is required, even if for only a short period of the cooling season. The occupant of the S.E.E.D. house commented that comfort was compromised during heavy rains due to the sensation of high interior RH. Figure 23 compares indoor and outdoor conditions (temperature and RH), dew point temperature and floor surface temperature during a week in the monsoon season with outdoor RH exceeding 80%. During this period the fan coil was operational and the floor surface temperature approached to within 5°F above the dew point temperature of the indoor air. Indoor RH exceeded 60% on one day. The condensate measurement indicated negligible amounts of condensate off the evaporator coil except for during the monsoon season. The correlation between dehumidification and outdoor moisture is shown in Figure 24. During the nonrainy season the system operated in radiant-floor-only mode without any comfort or condensation concerns.

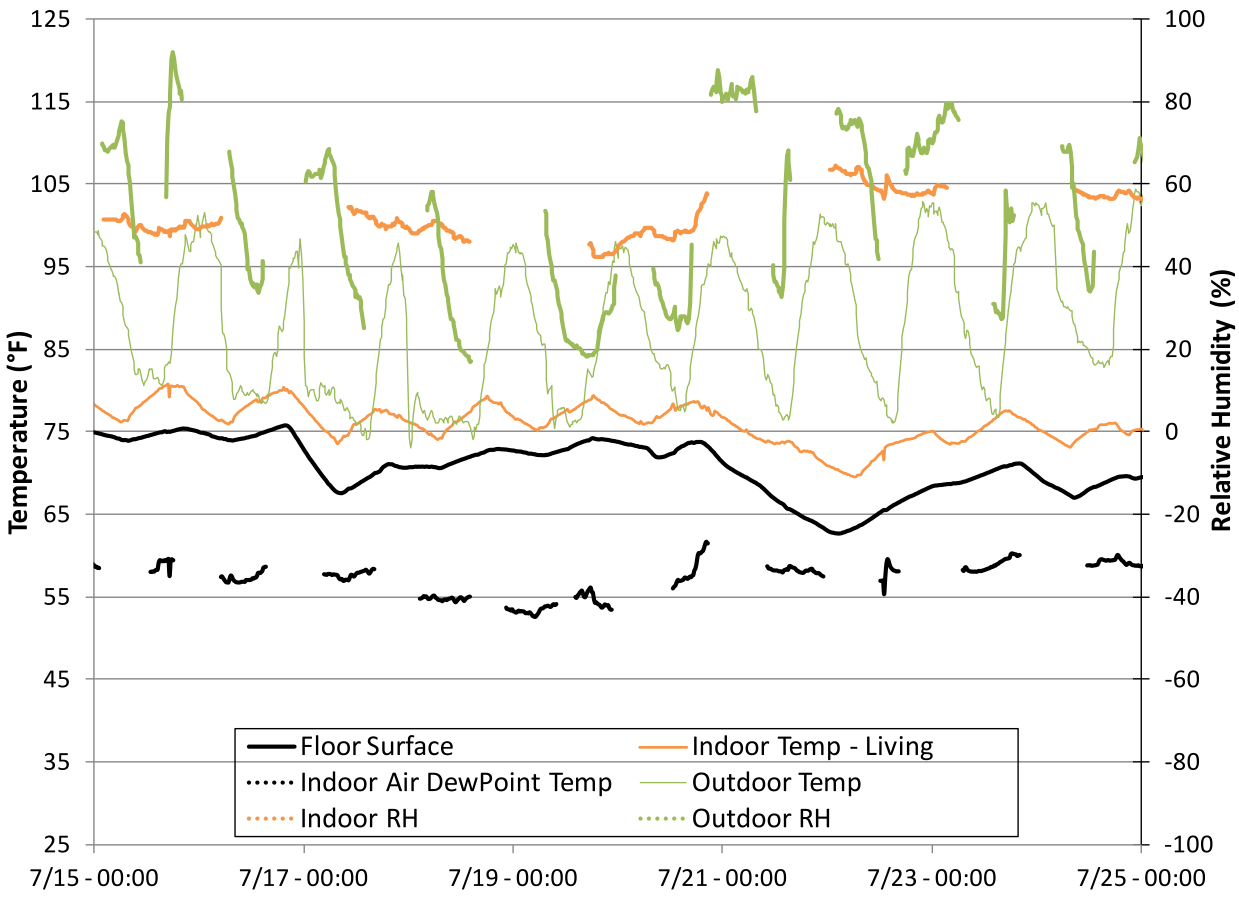

Figure 23. Evaluation of S.E.E.D. house radiant slab condensation potential during 2012 monsoon season (MM operation)

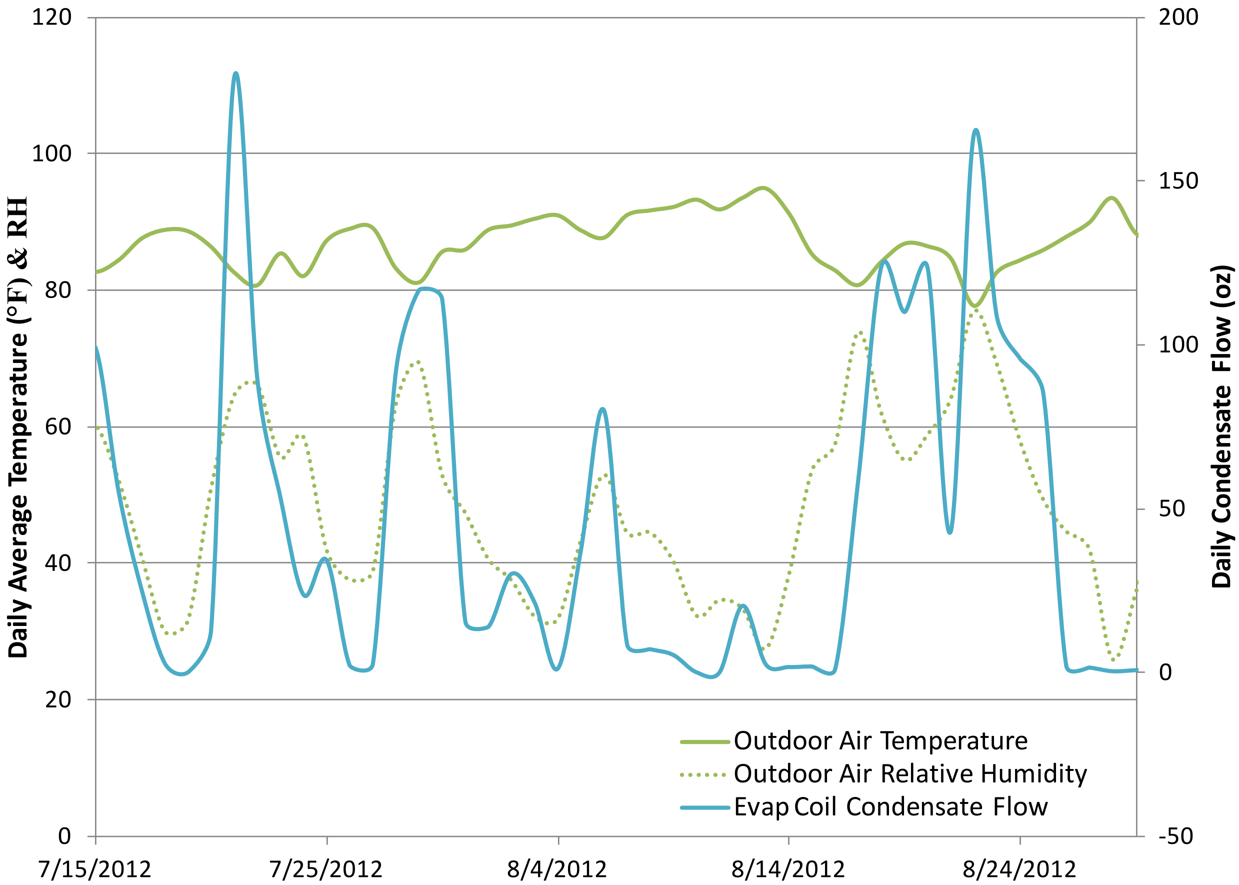

Figure 24. Monitored S.E.E.D. house evaporator coil condensate flow (MM operation)

4.2 TRNSYS Modeling Results

The validated AWHP TRNSYS model was used to project annual energy use for the two operating modes: radiant-only and MM, and the two control strategies: constant set point and C&C. The strategies were applied to the validated S.E.E.D. house model while modifying the internal gains and ventilation to reflect BA benchmark assumptions. Comparisons have been made to a base case with an air-to-air HP and forced-air delivery with ductwork located in the attic (6% leakage). An alternative case is also evaluated, which is identical to the base case except duct losses are eliminated by moving all ductwork inside the conditioned envelope. Four climates were evaluated as described in Table 12 using Typical Meteorological Year 3 weather files.

The Tucson climate was evaluated to assess the effect of a stand-alone dehumidifier (DH) on energy use and interior comfort (Table 17 and Figure 25). The stand-alone dehumidifier was controlled such that interior RH never exceeded 60%. As was discussed previously in the monitoring data evaluation, there were comfort concerns during the summer with radiant-only operation. The model validates this with floor condensation occurring when the radiant-only mode was applied without dehumidification. While the MM strategy did not have floor condensation errors, the internal RH did exceed 60% and fell outside the ASHRAE comfort regime (see Figure 25) according to ASHRAE Standard 55-2010 (ASHRAE 2010). This comfort regime is defined by a range of temperature and humidity at which 80% of sedentary or slightly active people would find the environment comfortable. The ranges are defined by clothing level, where CLO 1 is equivalent to a winter business suit and is commonly used to represent winter comfort ranges, and CLO 0.5 is equivalent to short sleeves and trousers, representing summer comfort. Data in Figure 25 reference ASHRAE comfort range CLO 0.5.

Table 17. TRNSYS Model Predictions for Tucson, AZ.

Mode	Control	Supp. DH	Condensation Errors	HP Heating (kWh)	HP Cooling (kWh)	Pump (kWh)	Fan (kWh)	DH (kWh)	Total HVAC (kWh)
Base Case: Ducts in Attic	Set point	Yes	No	205	3,155	0	895	69	4,324
BC + Ducts in Cond. Space	Set point	Yes	No	198	3,094	0	881	28	4,201
Radiant-Only	Set point	Yes	No	277	3,312	334	0	287	4,210
Radiant-Only	**Set point**	**No**	**Yes[a]**	277	3,126	316	0	0	3,719
Radiant-Only	C&C	Yes	No	288	2,812	302	0	209	3,611
Radiant-Only	**C&C**	**No**	**Yes[b]**	288	2,693	291	0	0	3,272
Mixed Mode	Set point	Yes	No	246	3,074	281	434	159	4,194
Mixed Mode	Set point	No	No	246	3,009	273	422	0	3,950
Mixed Mode	C&C	Yes	No	253	2,581	270	415	108	3,626
Mixed Mode	C&C	No	No	253	2,531	266	407	0	3,457

[a] The TRNSYS model predicted 28 h in which condensation formed on the slab surface.
[b] The TRNSYS model predicted 3 h in which condensation formed on the slab surface.

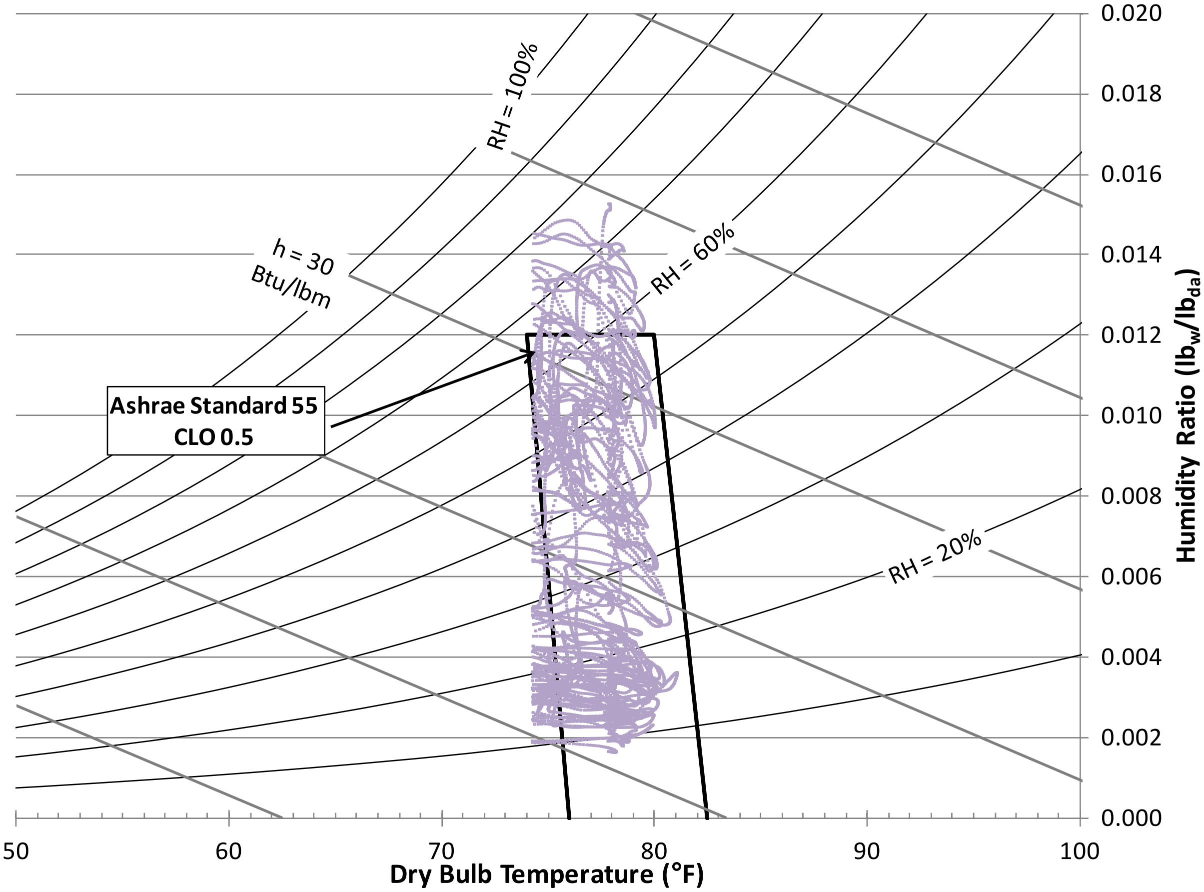

Figure 25. Tucson modeled indoor conditions during cooling season

Because dehumidification was found to be necessary in Tucson under certain operational strategies, a stand-alone DH was included in all subsequent simulations, except for climates where it was deemed unnecessary. A DH was applied to the model in both Tucson and Houston. Decoupling the two functions of sensible and latent cooling avoids overcooling in humid climates and can expand the feasibility of radiant cooling to a wider range of climates, although it does add first cost. Due to minimal latent loads in Sacramento and Denver and provided that both climates do not commonly employ dedicated humidification systems, a DH was not included in the simulations for these climates.

Results for all evaluated cases in each climate zone are presented in Table 18 through Table 23.

Table 18 presents energy consumption and savings model results for Tucson. The results show AWHP savings of 3% of annual HVAC energy compared to the base case air-to-air HP with ducts in the attic for the set point control strategy and similar energy consumption compared to the base case with ducts in conditioned space. However, 16% annual savings over the base case are achieved by combining the AWHP and the C&C control strategy. C&C seasonal cooling savings (May–October) are 17% for the MM case and 18% for the radiant-only case.

There are some noted differences in these results compared to the monitoring of the S.E.E.D. house. TRNSYS results do not demonstrate any energy advantage with the set point strategy from removal of the fan coil and operating in radiant-only delivery mode, while the monitoring data show clear savings. It is expected this principally has to do with longer run times on very hot days in the TRNSYS model with radiant-only delivery, as was discussed in the calibration results presented above. On very hot days the HP in TRNSYS is projected to operate the entire day. While long run times were also observed in the monitoring data on hot days, they were not seen to the extent that they were in the TRNSYS model. Additionally, daily energy savings for C&C compared to set point strategy on hot days were not as high as observed in monitoring data[14] and energy use did not trend with daily maximum temperature as well as in the monitoring data (Figure 19). This was more apparent with radiant-only delivery and may also indicate reduced effectiveness of the slab delivery model.

Table 18. Tucson TRNSYS Results Comparison

Mode	Control Strategy	Energy Use (kWh)					% Savings Versus Base Case	C&C Savings Versus Set Point
		HP	Pump	Fan	DH	Total		
Base Case: Ducts in Attic	Set point	3,360	0	895	69	4,324	–	–
BC + Ducts in Cond. Space	Set point	3,292	0	881	28	4,201	3%	–
Radiant-Only	Set point	3,589	334	0	287	4,210	3%	–
Radiant-Only	C&C	3,100	302	0	209	3,611	16%	14%
MM	Set point	3,320	281	434	159	4,194	3%	–
MM	C&C	2,833	270	415	108	3,626	16%	14%

Table 19 presents results for Sacramento. Similar to Tucson, Sacramento is also in the hot-dry climate zone but has fewer cooling degree days, drier summers, and greater summer diurnal temperature swings. Because humidity is not a significant issue in Sacramento, results are presented for the case without a DH.[15] Percent savings in this climate are much higher than in Tucson with AWHP savings of 21%–23% over the base case (set point strategy). This increase can be partially attributed to the elimination of the DH and associated energy consumption. It is expected that the higher heating loads are also a factor with the results indicating that the radiant distribution effectiveness is higher for heating than for cooling.

Since the Sacramento cooling season is shorter than in Tucson and the nights are much cooler, there were no observed savings for the C&C strategy as previously defined in this report. An alternative precooling strategy was evaluated that moved the setback period to 4 a.m.–6 a.m. to take advantage of nighttime building losses due to cool temperatures and shortened the season to June through September. In the model minimal cooling is observed in May and October. With this strategy, C&C annual savings over the base case are 2%–4%. Seasonal C&C cooling savings

[14] As reported earlier, monitoring at the S.E.E.D. house demonstrated 27% savings from precooling operation at a daily maximum outdoor temperature of 90°F and up to 40% savings at maximum temperature greater than 100°F

[15] A comfort evaluation was conducted and the simulation run with a stand-alone DH and results are presented in Appendix D.

(June–September) are even higher than in Tucson at 21% for the MM case and 43% for the radiant-only case.

Table 19. Sacramento TRNSYS Results Comparison

Mode	Control Strategy	Energy Use (kWh)				% Savings Versus Base Case	C&C Savings Versus Set Point
		HP	Pump	Fan	Total		
Base Case: Ducts in Attic	Set point	2,946	0	627	3,573	–	–
BC + Ducts in Cond. Space	Set point	2,799	0	596	3,395	5%	–
Radiant-Only	Set point	2,606	214	0	2,820	21%	–
Radiant-Only	C&C	3,213	277	0	3,489	2%	–24%
MM	Set point	2,507	203	23	2,733	23%	–
MM	C&C	2,674	224	49	2,946	18%	–8%

Table 20. Sacramento TRNSYS Results Comparison With C&C Operation June–September From 4:00 a.m. to 6:00 a.m.

Mode	Control Strategy	Energy Use (kWh)				% Savings Versus Base Case	C&C Savings Versus Set Point
		HP	Pump	Fan	Total		
Radiant-Only	C&C	2,493	205	0	2,698	24%	4%
MM	C&C	2,457	202	22	2,682	25%	2%

Table 21 shows results for the heating-dominated climate of Denver. The low cooling load requires almost no cooling energy use and therefore this study did not analyze the C&C control strategy. Similar to Sacramento, because of negligible latent cooling loads, a DH was not included in this simulation (see Appendix D for results with a stand-alone DH). In Denver and other cold climates, radiant delivery with an AWHP is an effective strategy that saves 31% of annual HVAC energy. The higher distribution effectiveness in heating is more evident in the cold climate of Denver where percent savings are almost 50% higher than in Sacramento.

Table 21. Denver TRNSYS Results Comparison

Mode	Control Strategy	Energy Use (kWh)					% Savings Versus Base Case
		HP	Pump	Fan		Total	
Base Case: Ducts in Attic	Set point	7,439	0	1,502		8,941	–
BC + Ducts in Cond. Space	Set point	7,136	0	1,441		8,577	4%
Radiant-Only	Set point	5,787	398	0		6,184	31%
MM	Set point	5,759	395	10		6,164	31%

This study also evaluated the feasibility of this strategy for hot-humid climates. Table 22 presents simulation results for Houston with a stand-alone DH similar to that used in the Tucson model. Due to high latent cooling loads in Houston, DH energy use is almost 50% of total

HVAC energy use for the AWHP cases. TRNYSYS modeling indicates that there is no savings potential for AWHPs with radiant or MM delivery in hot-humid climates. C&C operation is not evaluated for Houston since no savings were observed for the AWHP.

It was also of interest to evaluate the case in which dehumidification is accomplished in such a way that the system heat is not rejected back into the house. Results are presented in Table 23.[16] In this case the DH system also contributes substantially to sensible cooling load and represents the majority of total HVAC energy use for all cases. While total energy use for the AWHP strategies is reduced for this scenario compared to the previous scenarios, positive savings over the base case are still not achieved.

Table 22. Houston TRNSYS Results Comparison (Stand-Alone DH, DH Waste Heat to Conditioned Space)

Mode	Control Strategy	Energy Use (kWh)					% Savings Versus Base Case
		HP	Pump	Fan	DH	Total	
Base Case: Ducts in Attic	Set point	3,182	0	820	1,669	5,671	–
BC + Ducts in Cond. Space	Set point	3,049	0	787	1,711	5,547	2%
Radiant-Only	Set point	4,118	378	0	3,394	7,890	–39%
MM	Set point	3,370	298	359	2,608	6,635	–17%

Table 23. Houston TRNSYS Results Comparison (DH Waste Heat Rejected to the Outside)

Mode	Control Strategy	Energy Use (kWh)					% Savings Versus Base Case
		HP	Pump	Fan	DH	Total	
Radiant-Only	Set point	2,035	165	0	4,153	6,354	–12%
MM	Set point	1,659	132	15	4,209	6,015	–6%

Results for the four climate zones are summarized in Table 24 including total HVAC energy savings and utility cost savings. All results are presented for the radiant-only delivery mode.

The greatest annual savings are seen in Denver, a cold climate, with an estimated $310 per year utility bill savings over the base case and $269 compared to ducts within conditioned space. Savings for C&C operation are greatest for Sacramento,[17] with 24% HVAC energy savings and $99 utility bill savings. Note that time-of-use rates may significantly increase utility bill savings for the C&C strategy.

[16] This could be accomplished by using a chilled water coil similar to that in the MM scenario, which is decoupled from the primary cooling delivery system.
[17] C&C results for Sacramento are presented for the alternative precooling strategy with a setback from 4 a.m.–6 a.m. June through September.

Table 24. Climate Zone Annual HVAC Energy and Utility Cost Savings[a]

Climate Zone	AWHP vs. Base Case		AWHP Versus BC + Ducts in Cond. Space		AWHP + C&C Versus Base Case	
	kWh Savings	Utility Savings	kWh Savings	Utility Savings	kWh Savings	Utility Savings
Tucson (Hot-Dry)	3%	$13	0%	$0	16%	$80
Sacramento (Hot-Dry)	21%	$85	17%	$65	24%	$99
Denver (Cold)	31%	$310	28%	$269	n/a	n/a
Houston (Hot-Humid)	0%	$0	0%	$0	0%	$0

[a] Utility costs are based on a national average cost of $0.1126/kWh.

4.3 Cost Effectiveness

Table 25 shows results of a cost-effectiveness analysis for this strategy in the three climates that demonstrated energy savings. This analysis assumes that the incremental cost of the energy efficiency measures will be financed at an interest rate of 4.5% and a loan term of 30 years. The mature market system incremental cost of $6,400 as presented in Table 4 is used. Annualized utility savings are estimated using an escalation rate for electricity of 4% and a real discount rate of 3%.

Denver is the only climate with a positive annual cash flow at $108 annually. The break-even point, which results in a neutral cash flow, is achieved with first-year utility savings of $217, or about 1,925 kWh of annual electricity savings. For the AWHP strategy to be cost effective in Sacramento, provided $85 annual savings, the total incremental cost must come down to about $2,500. This may be achievable if the costs of the radiant floor are removed from the mature market incremental cost estimate, assuming that radiant distribution is desirable from a comfort perspective.

Table 25. Cost-Effectiveness Evaluation

Climate Zone	AWHP Versus Base Case		
	First-Year Utility Savings	Annualized Utility Savings	Average Annual Cash Flow
Tucson (Hot-Dry)	$13	$15	($235)
Sacramento (Hot-Dry)	$85	$98	($152)
Denver (Cold)	$310	$358	$108

5 Conclusions and Recommendations

AWHPs with radiant or MM delivery are an effective and efficient means of providing space heating and cooling in residential buildings in certain climates. This strategy presents a viable alternative to locating ductwork in conditioned space, which may not be feasible in all homes due to architectural challenges, while providing the comfort, thermal storage, improved distribution, and reduced noise benefits of radiant slab delivery. TRNSYS modeling estimates up to 31% HVAC energy savings compared to a standard HP with tight ducts located in the attic and up to 28% compared to the same base case with ducts located within conditioned space. Current system costs are high; however, there is justification to anticipate lower incremental costs as this strategy gains wide market acceptance. Cost reductions can be expected with increased contractor familiarity and reductions in manufactured equipment costs from volume production. Further research focused on development of packaged AWHPs as well as packaged controls for zoned systems is necessary. This will drive cost reductions and simplified installation procedures, in addition to ensuring consistent levels of quality and gaining market acceptance from contractors and installers.

The following are conclusions to the research questions posed in this study.

1. What are the average effective heating COPs? What is the efficiency of the integrated water heating/space heating system in heating mode (Altherma) and how does it compare to manufacturer specifications?

Seasonal space heating COPs over the full monitoring period of 3.26 and 4.18 were observed at the S.E.E.D. house and Cana house, respectively. Measured heating performance of the Altherma HP was very comparable to the manufacturer's specifications. However, water heating COPs with the Altherma were much lower than expected (COP = 1.63). It is anticipated that this is a result of poor heat transfer between the HP supply loop and the storage tank. The seasonal water heating system COP of 1.05 was also lower than expected due to regular operation of the electric resistance back-up heater to satisfy water heating demand.

2. What are the average effective cooling EERs, and can the dramatic improvement in performance relative to typical forced-air-only systems seen in previous testing be replicated?

Seasonal EER over the monitoring period in space cooling was 11.2 and 10.8 at the S.E.E.D. house and Cana house, respectively. This is a substantial improvement over measured performance in the field of residential air conditioners with ducted air delivery of 5.5 to 8.5 EER (Proctor et al. 2011). Performance was most dependent on outdoor air conditions with less than expected sensitivity between efficiency and EWT on the load side. Data were not able to confirm expected performance improvements of the hydronic system due to reduced thermal lift from high supply temperatures in cooling and lower supply temperatures in heating.

3. How effective is nighttime precooling in improving HP efficiencies and reducing cooling energy use?

Substantial cooling energy savings can be achieved from a precooling operating strategy that shifts air conditioner daytime operation to cooler nighttime hours and utilizes the house thermal mass to ride out most peak afternoon cooling events. Monitoring results from the S.E.E.D. house show 27% savings from precooling operation at a daily maximum outdoor temperature of 90°F and up to 40% savings at maximum temperature greater than 100°F. TRNSYS modeling estimates up to 18% seasonal cooling energy savings from precooling in the Tucson climate and up to 43% cooling savings in Sacramento with an alternative precooling strategy more appropriate to that climate. Seasonal percent savings can be lower than daily savings on hot days due to overcooling on milder days. An optimized solution could minimize this by employing a "smart" precool strategy that monitors weather conditions and changes the precooling set point accordingly. A precooling strategy is recommended in homes with available thermal mass (i.e,. exposed slab) for energy storage. Primary advantages result from more efficient operation during cool nighttime temperatures and increased efficiency due to reduced cycling. Daytime air-conditioner operation is delayed or eliminated, reducing peak load demand and utility bills for those customers on time-of-use electricity rates. The success of this strategy relies on occupants being comfortable with cooler interior temperatures during the night and early mornings, and the compressor being located such that nighttime operation does not affect occupants sleeping. To be effective, a minimum nighttime setback of 5°F is recommended.

4. How does the distribution efficiency of the MM system compare to that of a typical forced-air delivery system with ducts in unconditioned space?

Measured distribution efficiencies of the radiant floor distribution averaged 96%. This is approximately equivalent to a ducted distribution system with ductwork located inside conditioned space (94%) but is much higher than the 76% estimated for typical attic-located tight ducts (<=6% air leakage). Ductwork distribution efficiencies can commonly be much lower still, with 61% estimated for attic ducts with 15% leakage. In certain climates, underfloor slab losses may be eliminated in the cooling season due to cool ground temperatures. In cooling-dominated climates with temperate ground temperatures it may be possible to eliminate slab insulation without significantly affecting heating season performance.

5. Is the fan coil and latent cooling it provides necessary for dehumidification and to prevent floor condensation in a hot-dry climate, or can the forced-air delivery be eliminated completely?

Some form of dehumidification is required in all but the driest climates. Monitoring and modeling results from the dry Central Valley of California indicate that neither floor condensation nor interior comfort is a concern with radiant only cooling distribution. In other dry climates, such as Tucson, some dehumidification is required during the monsoon season, which can be accomplished with the MM distribution strategy. A control strategy to optimize performance could incorporate a humidistat control on the fan coil to switch from floor cooling to MM cooling only during periods of rising indoor moisture conditions.

6. Can TRNSYS reliably predict performance of this HVAC strategy?

Calibration of the AWHP was able to achieve results very close to that from monitored data. The calibrated model was able to match monitoring data over the period of April 2011–August 2012, with temperature differentials across the heat exchange within 1.6% of the observed data during heating season and within 9.7% during the cooling season. HP power was on average 8% different in heating operation and 1% different in cooling operation. The TRNSYS AWHP module can be used with confidence to evaluate additional climates and applications.

There is less confidence in the radiant slab delivery component of the model and future work could contribute to improved calibration. Detailed information on occupant loads and load scheduling was very difficult to ascertain. As part of the calibration process, the internal loads in the model were adjusted to provide the best match to the monitored slab response. Long run times were observed in both the model and the monitored data during radiant-only delivery on very hot days due to the slab's slow response. However, HP run times in the TRNSYS model were longer than in the monitoring data on these hot days. The model delivery system and interaction between the slab and the house may have reduced accuracy or these results may suggest that the house internal loads assumed in the model are greater than those in actuality.

7. In what climate zones is this strategy applicable?

The calibrated TRNSYS model was used to predict energy savings for air-to-water systems in four locations for radiant-only and MM operating modes and using both fixed set point and C&C night setback control strategies. Long-term monitoring confirms TRNSYS modeling results that AWHPs with radiant delivery are an effective, energy saving strategy for hot-dry climates with negligible latent cooling loads. Modeling indicates that when whole-house dehumidification is properly addressed, such as via a stand-alone DH, radiant cooling is a viable strategy in dry climates with seasonal humidity loads. TRNSYS modeling estimates 3%–21% annual HVAC energy savings in a hot-dry climate compared to a standard HP with tight ducts located in the attic, with the higher savings in milder hot-dry climates. Savings may be as high as 24% with a precooling strategy. In cold climates, estimated annual HVAC savings are even higher at 31% due to high distribution effectiveness during radiant heating mode.

Due to high latent loads, the TRNSYS model found that radiant cooling is not appropriate in humid climates. A stand-alone DH was necessary to avoid significant operation outside the ASHRAE Standard 55-2010 comfort zone (ASHRAE 2010) and to prevent floor condensation in the Houston climate. It is possible that a strategy that combines radiant cooling and a decoupled dehumidification strategy could be applicable in mixed-humid climates. One possible solution would be to operate the MM scenario in such a way that chilled water could be delivered just to the fan coil and bypass the radiant floor when latent loads are high. This single system could serve both the sensible and latent cooling loads and may be more viable. However, this would only be appropriate for climates that do not have dehumidification needs during the heating season. Field testing is necessary to test future research ideas in hot-humid climates.

Based on a cost benefit analysis over a 30-year mortgage using current projections for mature market costs, the strategy was found to be cost effective only in cold climates and not in hot-dry climates. System incremental costs would have to come down by a factor of three to make the strategy cost effective in a hot-dry climate, or utility rates would have to triple. Combining the AWHP with a precooling strategy can greatly increase cost effectiveness, especially if time-of-use utility rates are in use. There are various benefits provided by the AWHP radiant system that are not factored into a cost analysis. These include increased occupant comfort, improved thermal distribution, noise reductions, and peak load reduction. In addition, as electricity prices increase, this may move the technology to be cost effective for additional climates.

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Appendix A: Analysis Details

A multiple linear regression analysis was conducted to develop two predicted relationships between HP performance and outdoor dry-bulb temperature and EWT. The two performance parameters that were used were condenser power and capacity. The DOE2 simulation approach, which assumes a bi-quadratic relationship between the variables of interest, was used. The bi-quadratic equation is of the following form:

$$c_1+c_2*OAT+c_3*OAT^2+c_4*TWHE+c_5*TWHE^2+c_6*OAT*TWHE$$

c_n are constants that are determined through the regression, OAT is outdoor air temperature, and TWHE is HP EWT.

The function *linest* in Excel was used to estimate the coefficients of the curve based on full-load monitoring data. Full-load data for this analysis is quantified as any 15-min period during which the HP is operating greater than 92% of the time and had been operating greater than 92% of the time during the previous 15-min period

Statistical outputs from this analysis are presented in Table 26 and Table 27. The F probability distribution was evaluated to test the hypothesis that the observed relationship between the dependent and independent variables occurred by chance and resulted in a 0% chance in all cases.

Table 26. Statistical Results from Multiple Linear Regression of Condenser Power and Capacity in Space Cooling in Relation to OAT and LWT

Statistic	Result—Power	Result—Capacity
n	3,607	3,607
r^2	98.1%	87.4%
Standard Error	0.0298	785
95% Confidence Interval (CI)	(–0.058, 0.058)	(–1,539, 1,539)
99% CI	(–0.084, 0.084)	(–2,204, 2,204)

Table 27. Statistical Results from Multiple Linear Regression of Condenser Power and Capacity in Space Heating in Relation to OAT and LWT

Statistic	Result—Power	Result—Capacity
n	1,105	1,105
r^2	98.8%	95.8%
Standard Error	0.0201	793
95% CI	(–0.039, 0.039)	(–1,553, 1,553)
99% CI	(–0.057, 0.057)	(–2,225, 2,225)

Appendix B: Short-Term Testing and Commissioning Results

S.E.E.D.

Major commissioning tasks were conducted over 2 days in April 2011. The focus of the effort was to verify correct operation of the mechanical systems (specifically the HP), verify correct operation of the monitoring equipment including sensors and communications, and take one-time measurements of pertinent data points.

Integrating the zoning, the hydronic and air systems, and the HP in both heating and cooling required some creative on-site revisions to the original design as well as troubleshooting. Although efforts were made to use off-the-shelf components, this was not possible in all instances. A custom control box was constructed to communicate between the zone thermostats, the zone valve controller, the HP, and the fan coil. The Taco brand zone controller is designed for heating-only systems and is not capable of controlling both heating and cooling. A residential zone control may have worked, but the defrost control may not have been compatible with the AWHP equipment. A relay was installed to open all floor zone valves when a defrost signal is received from the HP, allowing the HP to absorb heat from the entire slab instead of from a single zone.

The Aqua Products controls require that a heating call be received from a single zone to activate the reversing valve in heating. The living room zone was selected as the master zone and has to call for heating for either of the other two zones to receive heating. For cooling, the controls were able to be set up such that a call for cooling from any zone would initiate HP operation.

There was some difficulty wiring the pump into the Aqua Product controls. It was expected that if the pump relay output was wired directly to the unit it would operate whenever there was a call for heating or cooling; however, at startup it was found that the pump was not operating at all. A second relay was installed between the zone controller and the HP. A control schematic can be found in Appendix C.

A zone bypass was installed on the radiant floor system to maintain a minimum flow rate through the HP when only one zone is calling. The monitoring equipment was used to verify both pump flow and power under different operating modes. Both flow and power remain relatively constant regardless of the number of zones calling.

One-time HERS tests, including duct and building envelope leakage testing were completed by HERS raters contracted by the local utility (Tucson Electric Power). ARBI also conducted testing to verify fan coil airflow, fan coil power, and pump power.

Cooling operation began with using only the floor for delivery (air handler turned off). When the air handler was later enabled to test MM delivery performance, the builder discovered that the condensate pan was slanted, so that the condensate often pooled in the pan. This water then reevaporated into the supply airstream resulting in the reintroduction of humidity to the space. The builder fixed the problem by leveling the pan, which facilitated proper drainage, and the monitored indoor RH decreased. Indoor RH never exceeded 60% during this period, and the occupants did not express any discomfort.

The following is the verification checklist used during commissioning with select results from the process.

System/Sensor Commissioning

Preparation and Base Load Measurement

- ☒ Shut off breakers for water heater element, refrigerator, microwave, range, washing machine, and PV system.
- ☐ Run hot water tap (bathtub) to deplete solar storage (did not do).
- ☒ Shut off the ERV.
- ☒ Unplug solar pump.
- ☒ Unplug recirc pump.
- ☒ Wait 5 min while base load is being measured.

Verify Hydronic System Operation

- ☒ Install flow meters and recharge system.
- ☒ Review control wiring and verify transformer in zone control is disconnected.
- ☒ Disconnect HP compressor from contactor.
- ☒ Unplug HP from switched outlet and plug into live outlet.
- ☒ Set all thermostats to cool and power up HP.
- ☒ Verify pump operation.
- ☒ Set up logger to read flow and set logger spans correctly.
- ☒ Manually open Zone 1 (guest bedrooms) and wait 5 min while pump power and flow are recorded.
- ☒ Manually open Zone 2 (living) and wait 5 min.
- ☒ Manually open Zone 3 (master bedroom) and wait 5 min.
- ☒ Adjust bypass valve to ensure a minimum flow of 5 gpm in all zone-calling scenarios.

Table 28 documents the measured flow rates during the radiant system for various zone calling scenarious.

Table 28. Hydronic System Flow Rate With Various Zones Calling

Zones Calling	Flow Rate (gpm)
All Open	6.21
Zone 1 Only	5.72
Zone 2 Only	5.69
Zone 3 Only	5.69
Zone 1 and 2 Only	5.74
All Closed	5.65

DHW Pump Power Tests

Solar Pump (could not test – leak in solar collector)

- ☐ Plug in the solar pump.
- ☐ Wait 5 min (after pump starts running).
- ☐ Unplug and turn off water.

Recirc Pump

- ☒ Plug in recirc pump and set to "manual on."
- ☒ Wait 5 min while pump power is recorded.

Air Handler Airflow Tests

To check supply/return temperature calibration:

- ☒ Turn on switch at fan coil.
- ☒ Unplug the pump from live outlet.
- ☒ Close coil bypass valve (noting original position).
- ☒ Activate living room thermostat in cooling mode (verify fan operation).
- ☒ Wait 5 min while supply/return sensors are checked.

To measure airflow:

- ☒ Plug the pump into the switched outlet.
- ☒ Set living room thermostat to heat and raise temperature until it turns on the HP.
- ☒ Manually close the zone valve for the living room.
- ☒ Wait 15 min or until supply/return temperature difference stabilizes.

Restore System

- ☒ Set living room thermostat to 78°F (cooling mode).
- ☒ Reset the position of the coil bypass valve.
- ☒ Turn on breakers that were turned off in Step 1, Preparation.
- ☒ Plug in the solar pump.
- ☒ Turn on the ERV.
- ☒ Set recirc pump to "timer."

HERS Tests

Duct Tightness Test

- ☒ Test total duct leakage: **24 supply/22 return: 46 cfm total leakage**

Blower Door

- ☒ Test building leakage with blower door (CFM_{50} < 1935): **750 CFM_{50}**

One-Time Measurements

- ☒ Verify fan coil airflow: **831 cfm**
- ☒ Measure solar pump power: **137 W**
- ☒ Measure HP circulation pump power: **150 W**
- ☒ Measure DHW recirculation pump power: **38 W**

Cana

Commissioning of the Cana house was conducted over several visits to ensure that all issues were resolved and the Altherma system was functioning properly. An initial visit was scheduled with the Daikin Altherma representative, who verified operation and programmed the HP based on the design criteria. During the first visit, several basic installation mistakes were identified and brought to the attention of the builder and HVAC contractor. They included the following:

- Incorrect zone valve models were installed. Normally open valves were specified, but normally closed valves were installed.
- Zone dampers were wired incorrectly, resulting in incorrect operation.
- HP went off on a “7H” error, indicating insufficient water flow. The sound of air in the lines could be heard.
- Duct leakage was just below 6% of design airflow.

The contractor returned and corrected the above items, including purging of the lines to remove air. ARBI returned to the site to complete the commissioning process. The commissioning process included the following procedures:

1. Test duct leakage.
2. Verify water heating mode and operation.
3. Verify systems in heating and cooling modes.
4. Verify adequate pump flow in each mode of operation.
5. Verify adequate flow through each floor circuit.
6. Calibrate airflow measurements relative to fan power and supply plenum pressure.

The data logging equipment was used to assist in commissioning and readings from handheld equipment were taken to calibrate and verify correct operation of all monitoring sensors. A blower door test was also performed to measure building envelope leakage at 50 Pa.

At the time of commissioning, the HP was operating correctly in all modes. Once data collection began, it was noticed that the HP was not operating during calls for cooling or water heating. The HVAC contractor returned several times to reset the HP and flush the loop, but the heat pump continued to turn off after short periods of operation. On September 21, 2011, when ARBI was on site for a walk-through with the homeowner and the Altherma representative, the HP was again disabled due to a flow error. The filter screen on the Altherma unit was found to be almost completely plugged with plumber’s dope (see Figure B-1). The HVAC contractor cleaned out the filter screen, reflushed each circuit individually, cleaned the screen one more time, and then bled air out of the loop. The HP was then restarted and it has been operating reliably since.

Figure 26. Plugged Altherma filter screen

HERS Tests

Duct Tightness Test

- ☒ Test with outside air and relief louvers in normally closed position to evaluate leakage from dampers.
- ☒ Retest with outside and relief louvers sealed: **92 cfm**

Blower Door

- ☒ Test now and evaluate need for test after ceiling insulation is installed (SLA < 3.5, CFM_{50} < 2995): **2637 CFM_{50}**

System/Sensor Commissioning

- ☒ Verify NightBreeze control settings.

Verify Water Heating Mode and Sensors

- ☒ Run down water temperature (as needed) to initiate a water heating call (heat pump and resistance) and leave tap open, start HP.
- ☒ Verify change in three-way valve position.
- ☒ Measure/verify: FWD: **4.2** gpm FWS **11** gpm EWH **4.8** kW
- ☒ Time to fill cup: **10.5** s for **3.5** cups

Verify Systems in DHW Mode

- ☒ Verify HP operation.
- ☒ Verify mode status:

  TWHL **132.3** TWHO **100** FWS______

Verify Systems in Cooling Mode

- ☒ Set zones 1 and 2 to cooling.
- ☒ Wait 15 min and measure temperatures and power:

  TWHS **66.3** TWHL **52.7** TWFS **56.7** TWCS **75.8**

EFAN __42____ EHP__________

☒ Test individual zones:

Zone 1 calling: FWS__6.0_____ gpm (> 4.5) Verify SZD2 = 1
EFAN__56____

Zone 2 calling: FWS___5.6____ gpm (>4.5) Verify SZD1 = 1
EFAN__56____

Verify Vent Cooling Mode and Status Sensor

☒ Set manual fan to “outside air.”

☒ Verify airflow at relief (verified damper operation).

☒ Verify SDMP status and signal: **0** = Recirc **1** = OA

Flow Balance Check

☒ While system is operating in heating or cooling mode, check flow at each circuit. Adjust and balance if necessary and double check system flow (FWD).

Table 29 presents results from the flow balance check process.

Table 29. Hydronic System Flow Balance Test Results

	Design Heating (gpm)	Design Cooling (gpm)	Measured (gpm)
MANIFOLD 1 – MASTER BEDROOM CLOSET			
Loop 1: M Bath / Closet	0.28	0.34	0.75
Loop 2: M Bedroom	0.33	0.40	0.75
MANIFOLD 2 – CLOSET AT END OF HALL			
Loop 3: Hallway	0.32	0.39	0.5
Loop 4: Bed 2	0.21	0.26	0.5
Loop 5: Office	0.17	0.21	0.75
Loop 6: Office	0.16	0.20	0.75
Loop 7: Gallery	0.4	0.47	0.5
Loop 8: Gallery	0.4	0.47	0.5
Loop 9: Gallery	0.4	0.47	0.5
MANIFOLD 3 – CABINET OFF KITCHEN NEAR PANTRY			
Loop 10: Kitchen / Laundry	0.47	0.58	0.75
Loop 11: Kitchen / Pantry	0.53	0.65	0.9

Airflow Calibration and Balance

☒ Install True-Flow grid.

☒ Install pitot tube in plenum.

☒ Set manual fan to “on” and verify pressure sensor operation.

☒ Measure system airflow at various settings to correlate with supply plenum pressure (PAS) and fan power (EFAN). (See Table 30)

Table 30. NightBreeze Airflow Test Results

NightBreeze Airflow Setting (cfm)	Measured Airflow Reading (cfm)	PAS (Pa)	EFAN (W)
200	N/A	2.7	23
500	540	5.7	45
1,000	860	9.3	113
1,500	1,350	18.0	360
1,800	1,610	32.0	608

Plenum Pressure Normal (Pan) = 9.3 Pa w/ TrueFlow (Pa_{TF}) = 8.9 Pa

☒ Measure room-by-room airflows in cooling mode. (SeeTable 31)

Table 31. Room-by-Room Airflow Test Results

Room	Measured Airflow Reading (cfm)
Master Bed	130
Master Bath	95
Bath 2	23
Bed 2	132
Office	143
Gallery-1 (NW)	169
Gallery-2 (SW)	146
Gallery-3 (NE)	97
Gallery-4 (SE)	97
Kitchen-1 (N)	211
Kitchen-2 (S)	121
Laundry	71
Pantry	65

ZV1 closed = living zone only

ZV2 closed = sleeping zone only

Appendix C: S.E.E.D. House Mechanical Systems Control

The control diagram and associated description for the HP and zone control are shown in Figure 27.

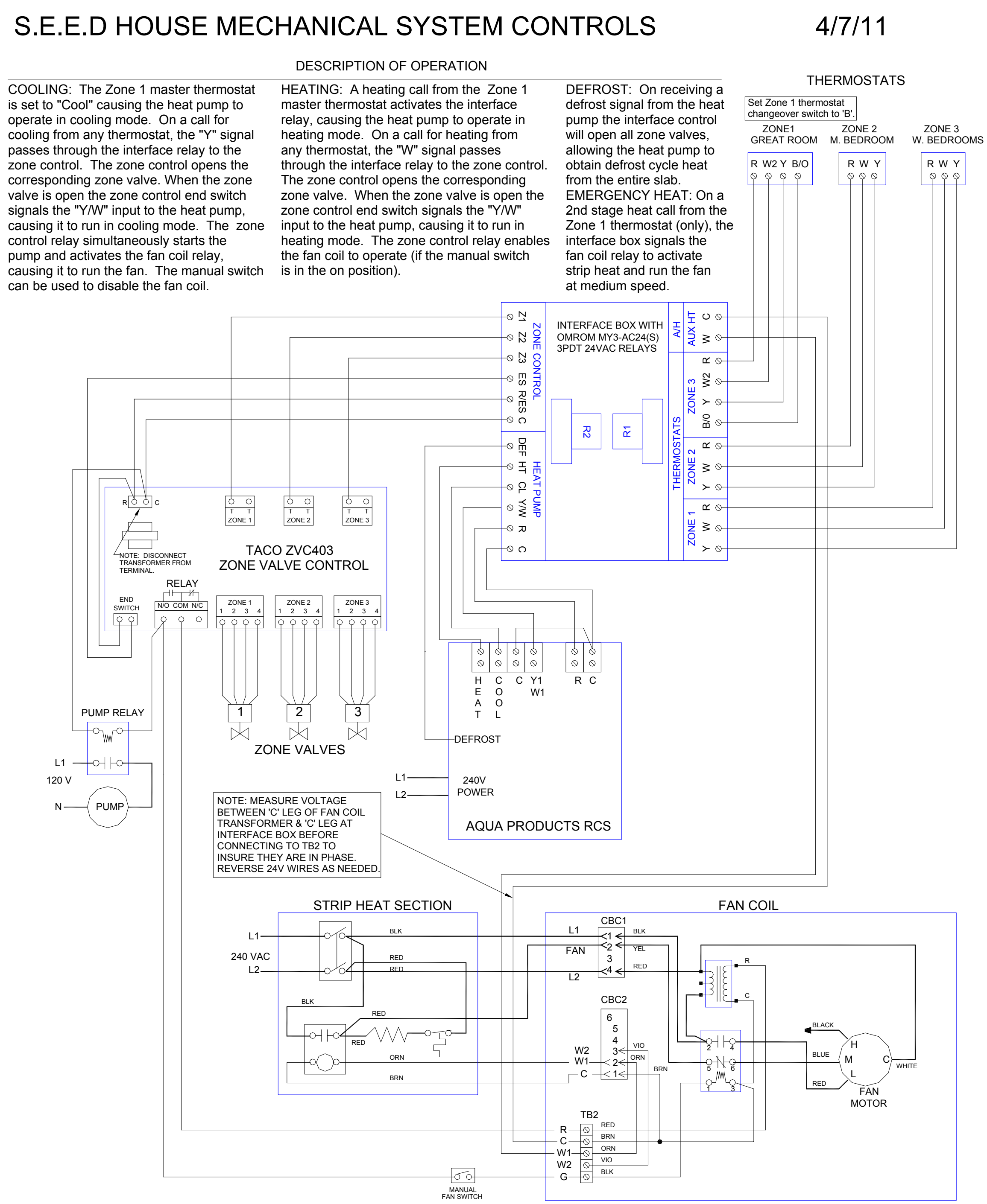

Figure 27. Mechanical system controls schematic

Appendix D: Additional TRNSYS Model Results

Following are additional TRNSYS results not reported in the main body of the report.

Table 29 presents TRNSYS results for Sacramento with a stand-alone DH. None of these strategies produced condensation warnings during floor cooling. The Sacramento radiant-only indoor conditions are graphed in Figure 28 for each time step of the case without external dehumidification. There was some operation above the ASHRAE 55-2010 comfort standard of 0.012 humidity ratio (ASHRAE 2010) but given ARBI's experience in this region, DHs would not be needed with a well-designed radiant cooling system.

Table 32. Sacramento TRNSYS Results Comparison with Dehumidification Control

Mode	Control Strategy	Energy Use (kWh)					% Savings Versus Base Case
		HP	Pump	Fan	DH	Total	
Base Case: Ducts in Attic	Set point	2,927	0	623	79	3,630	
BC + Ducts in Cond. Space	Set point	2,774	0	591	79	3,443	5%
Radiant-Only	Set point	2,587	212	0	117	2,916	20%
MM	Set point	2,521	204	26	71	2,823	22%

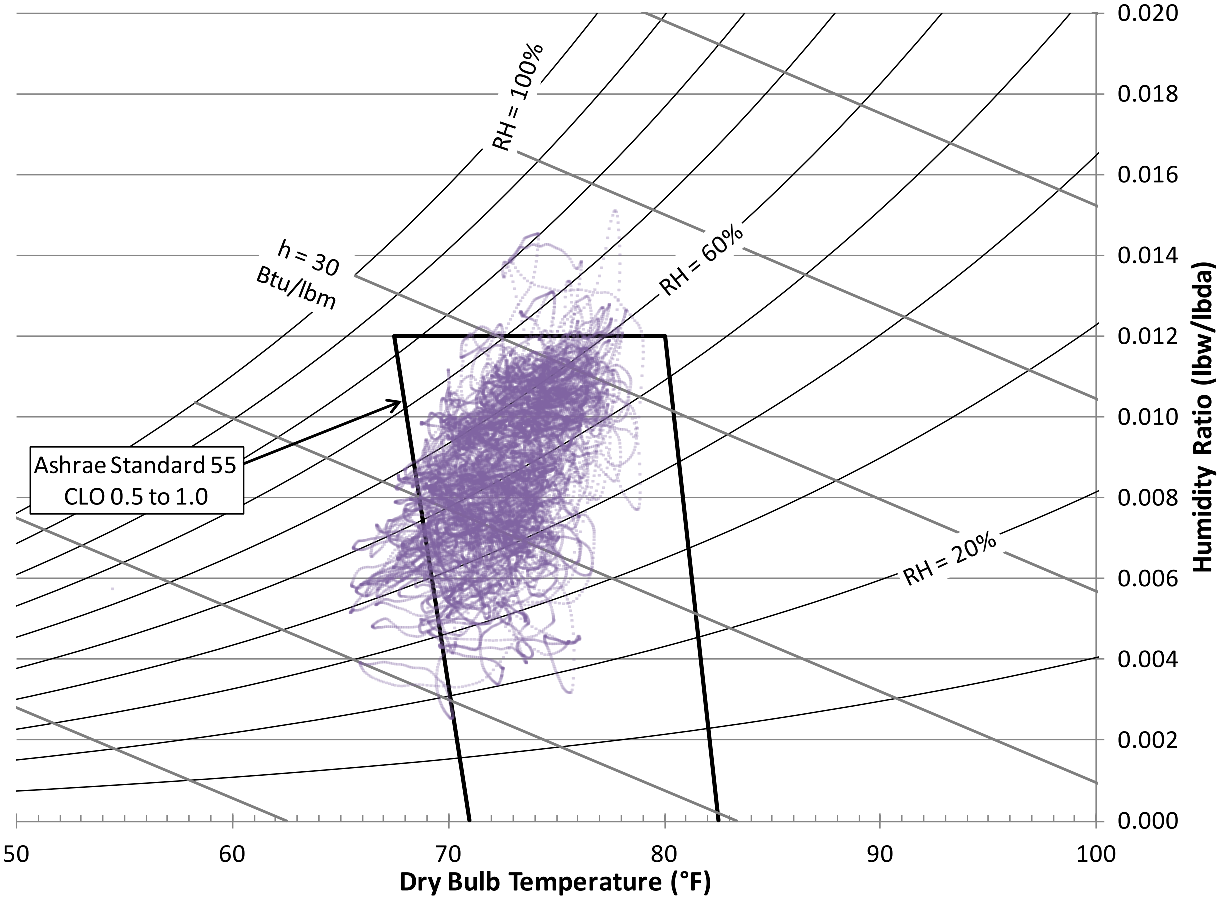

Figure 28. Sacramento indoor conditions over the course of the year as compared to ASHRAE 55-2010 comfort

Table 30 presents TRNSYS results for Denver with a stand-alone DH. DH operation is very minimal.

Table 33. Denver TRNSYS Results Comparison With Dehumidification Control

Mode	Control Strategy	Energy Use (kWh)					% Savings Versus Base Case
		HP	Pump	Fan	DH	Total	
Base Case: Ducts in Attic	Set point	7,426		1,500	41	8,967	–
BC + Ducts in Cond. Space	Set point	7,116		1,438	41	8,595	4%
Radiant Only	Set point	5,796	399		35	6,230	31%
MM	Set point	5,737	393	9	36	6,175	31%

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