

Evaluation of Condensation Risk Mitigation Methods for Residential Radiant Ceiling Cooling Systems

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ABSTRACT

Air-to-water heat pumps (AWHPs) using hydronic radiant ceiling panels were tested from 2015 through 2019 at one of the Central Valley Research Houses (CVRH) in Stockton, CA. These unoccupied laboratory houses are used to evaluate emerging technologies and potential code compliance options. CVRH has been sponsored by utilities, primarily PG&E and SCE, and the California Energy Commission for almost a decade.

A 2018 ACEEE Summer Study paper based on CVRH work compared radiant ceiling systems with air-to-air ducted systems. This paper evaluates three methods of avoiding condensation on radiant ceiling panel surfaces and compares energy use and comfort.

1. Using a 55°F panel water supply temperature without supplemental dehumidification.
2. Using a 48°F panel water supply temperature with a standalone dehumidifier controlled to maintain a dewpoint below 48°F.
3. Using a 52°F panel water supply temperature with a ductless hydronic fan coil controlled to maintain a dewpoint below 56°F (lowest recorded panel surface temperature in previous tests). Fan coil water supply temperature was 45°F.

Electric heaters and humidifiers were used to mimic internal heat and moisture gains representative of an occupied home, according to prescribed schedules.

Using Method 1, condensation risk was minimal, but comfort suffered. Using Method 2, comfort improved and condensation risk was nonexistent, but the dehumidifier used almost as much energy as the AWHP. Method 3 performed the best for comfort and energy use, while providing adequate dehumidification. However, during dehumidification operation the fan coil contributed sensible cooling equal to the radiant panels, and the system regularly overcooled the space.

Introduction

Radiant systems offer an alternative solution to reduce distribution losses while providing equal or superior comfort. These systems use large heated or cooled surfaces, typically panels containing tubes to transport heated and chilled water, to directly serve thermal loads through radiative heat transfer. Traditional forced air systems instead use heated and cooled air distributed by ducts to indirectly serve thermal loads through convective heat transfer.

Figure 1 provides a representation of heat transfer effects in a room with a radiant ceiling during cooling operation. Objects inside the room radiate heat to the ceiling panel surface, which is cooled by chilled water circulated through channels in the panel. Some heat is also transferred to the panel surface via natural convection where rising warm air meets the panel surface.

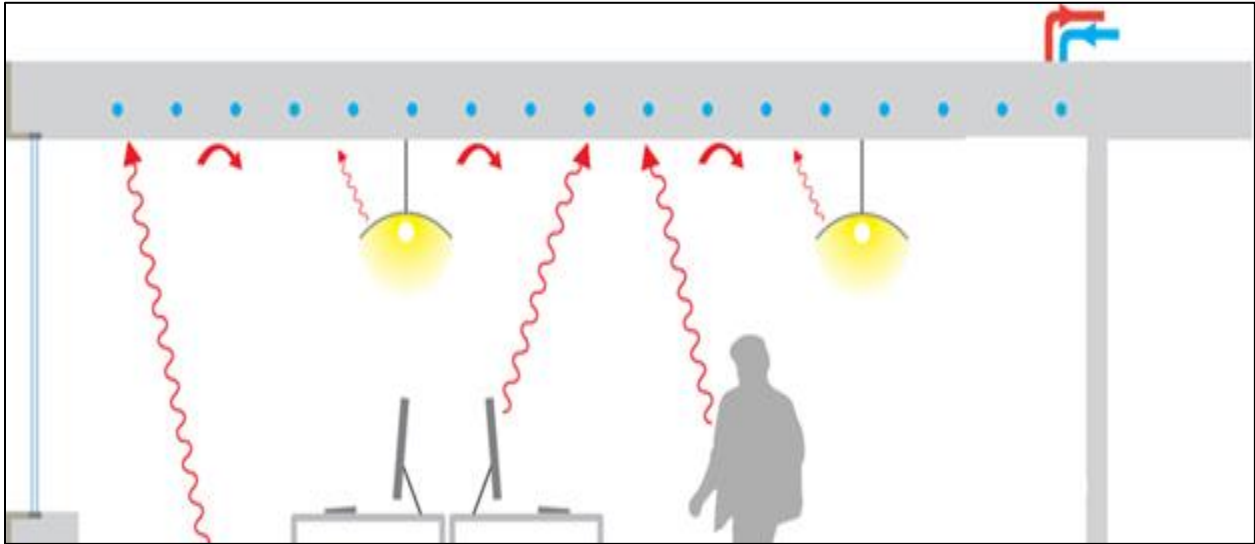


Figure 1. Representation of heat transfer effects in an office with a radiant ceiling in cooling.
Image credit: Caroline Karmann, Center for the Built Environment at UC Berkeley.

Radiant systems impact the mean radiant temperature (MRT), thus minimizing the effects of drafts and non-uniform distribution within rooms. Because radiative heat transfer is more direct than convection, comfort can be delivered at more moderate air temperatures, which can translate to improved equipment efficiency.

However, a notable concern for using radiant systems for cooling is humidity build-up and condensation on the panel surfaces. Unlike traditional residential air conditioning systems, which provide sensible and latent cooling, a radiant cooling system should only provide sensible cooling. Latent cooling, or dehumidification, requires condensation of water vapor in the air, which occurs when a surface temperature is less than or equal to the dewpoint temperature. Condensation on a radiant ceiling system during cooling operation will eventually cause mildew and mold growth, which leads to indoor air quality issues and structural damage, as well as water damage to objects within the conditioned space (Watson and Chapman 2002). Therefore, developing and evaluating methods for preventing condensation and reducing condensation risk are essential for radiant ceiling systems.

A project using a test house in Stockton, CA with a radiant ceiling cooling system evaluated three methods of avoiding condensation:

1. Using a 55°F panel water supply temperature without supplemental dehumidification.
2. Using a 48°F panel water supply temperature with a standalone dehumidifier controlled to maintain a dewpoint below 48°F.

3. Using a 52°F panel water supply temperature with a ductless hydronic fan coil controlled to maintain a dewpoint below 56°F (lowest recorded panel surface temperature in previous tests). Fan coil water supply temperature was 45°F.

Evaluations included in this paper compare comfort performance, energy use, and level of condensation risk for each of the three dehumidification methods. Reports are available on the Emerging Technology Coordinating Council (ETCC) website that document other evaluations of the radiant ceiling system in detail (Haile, Springer and Hoeschele 2016).

Project Overview

From summer 2015 and through summer 2019, air-to-water heat pumps (AWHPs) with radiant ceiling panels were tested at one of the four Central Valley Research Houses (CVRH) in Stockton, CA (Wilcox 2014). The CVRH project operates four laboratory houses of different vintages and envelope configurations in Stockton, California (Climate Zone 12). Three have been used to evaluate the field performance of mini-split heat pumps (Wilcox 2016). The houses serve as a useful test bed for evaluating various building technologies and systems. The fourth house, pictured in Figure 2, was used for the radiant system evaluations. Built in 1996, the house is a 1,690 ft², two-story home with slab on grade construction and a mid-level entry.



Figure 2. CVRH Radiant House.

The house was equipped with resistance heaters and fans in each room and a humidifier in the kitchen to mimic internal heat and moisture gains due to occupants and appliances, according to schedules used by California energy compliance models (CEC 2012).

Radiant System

An AWHP delivered conditioned water to the radiant ceiling panels to heat or cool the house. During cooling operation, radiant ceiling panels are provided with cooled water to chill the panel surface for heat transfer to the space below.

The radiant system consisted of two types of panels: site-built heat transfer plates and prefabricated modular panels. The prefabricated panels were installed only in the Master Bedroom, as they were added late in the initial installation phase of the project, while the site-built panels were installed in all other rooms.

The site-built panels consisted of a 26-gauge steel lath, 16" wide with grooves for ½" PEX tubing spaced 8 in. The lath was screwed to the ceiling and lath and tubes were covered with a layer of standard ½" drywall. The specified capacities were 15.0 Btu/ft²-hr at 55 °F mean water temperature (MWT) (cooling) and 21.9 Btu/ft²-hr at 95 °F MWT (heating) (Newberry 2012).¹

The prefabricated panels consisted of 8 mm PEX tubing bonded to an 18-gauge aluminum sheet, backed by 1½" thick EPS foam insulation, and covered by specially formulated drywall glued to the aluminum sheet. The specified capacities were 23.8 Btu/ft²-hr at 55 °F MWT (cooling) and 22.3 Btu/ft²-hr at 95 °F MWT (heating) (Messana 2017).²

Panels were selected based on availability and ease of installation in existing homes. Table 1 lists basic properties of each room and panel array, including average flow rates with all zones calling, installed panel type and area, and the building insulation R-value above the panels. Total house sensible load per square foot of panel is 20.5 Btu/h·ft² in cooling and 25.4 Btu/h·ft² in heating. Per square foot of floor area, the total house sensible load is 6.9 Btu/h·ft² in cooling and 8.6 Btu/h·ft² in heating.

Table 1. Room and panel properties.

Name	Installed Panel Type	Floor Area (ft ²)	Panel Area (ft ²)	Average Flow (gpm)	ACCA Manual J Load (Btu/h)		Insulation R-value Above Panel
					Cooling	Heating	
Bonus	Site-Built	222	120	1.00	2,008	2,742	4.5
Bedroom 1	Site-Built	119	88	1.29	1,236	2,270	4.5
Bedroom 2	Site-Built	119	72	1.16	1,214	1,370	4.5
Bedroom 3	Site-Built	104	72	1.00	1,411	2,358	38
Bath	Site-Built	38	27	1.82	530	949	38
Living/Dining	Site-Built	312	187	1.77	3,237	2,717	49
Kitchen	Site-Built	131	108	0.79	2,307	1,424	49
Master Suite	Prefab.	267	125	1.37	2,307	2,345	49

One aspect of delivering cooling via the panels is the potential for condensation on the panel surface, depending upon the panel surface temperature and the dewpoint temperature of the house air.

Three methods of mitigating condensation risk were evaluated with the radiant ceiling panels:

1. Using a 55°F panel water supply temperature without supplemental dehumidification.

¹ At 76 °F (cooling) and 70 °F (heating) indoor temp., film coefficients are 0.71 and 0.88 Btu/hr-ft²-°F, respectively.

² At 76 °F (cooling) and 70 °F (heating) indoor temp., film coefficients are 1.08 and 0.88 Btu/ft²-hr-°F, respectively.

2. Using a 48°F panel water supply temperature with a standalone, ENERGYSTAR-rated, dehumidifier controlled to maintain a dewpoint at or below 46°F.
3. Using a 52°F panel water supply temperature with a ductless hydronic fan coil controlled to maintain a dewpoint below 56°F (lowest recorded panel surface temperature in previous tests). Fan coil water supply temperature was 45°F.

Over the years the hydronic test system at the CVRH Radiant Test House has evolved to allow testing new configurations and equipment. Method 1 and 2 were both evaluated during summer 2015 from July through September using the single zone for the radiant panels and a single speed AWHP. Method 3 was evaluated in summer 2019 from July through September using the same radiant panels, but zoned upstairs and downstairs, and using a variable capacity AWHP.

Figure 3 shows a simplified schematic for the 2015 configuration presenting piping, pumps, valves, and temperature (T) and flow (F) sensing locations, used by Methods 1 and 2. The dehumidifier used in Method 2 is also shown, but was not used during testing of Method 1. Both configurations were operated as a single zone.

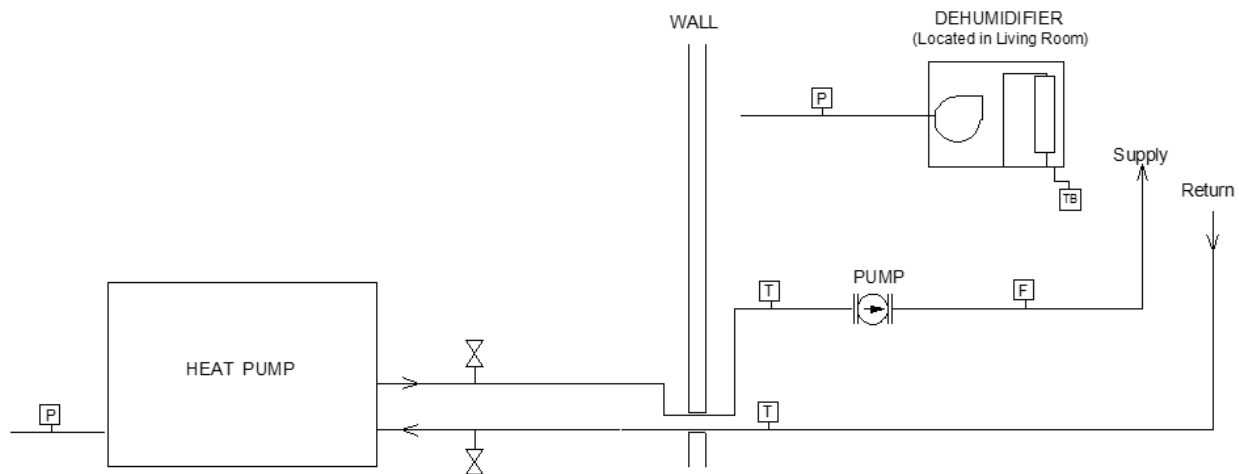


Figure 3. System schematic for summer 2015 (dehumidification provided by a free-standing dehumidifier, summer 2015).

Figure 4 shows a simplified schematic for the radiant system with Method 3. The radiant ceiling and fan coil were operated using 52°F and 45°F supply water temperatures, respectively. The lower supply temperature for the fan coil was needed to improve the latent cooling capacity of the fan coil. The fan coil was controlled using a 56°F dewpoint setpoint with a 5°F deadband and minimum 10-minute run time, based on the upstairs dewpoint. This means the fan coil would begin dehumidification if dewpoint went over 56°F and stop when dewpoint reached 51°F. If dewpoint went over 58°F, flow to the radiant panels would be shut off.

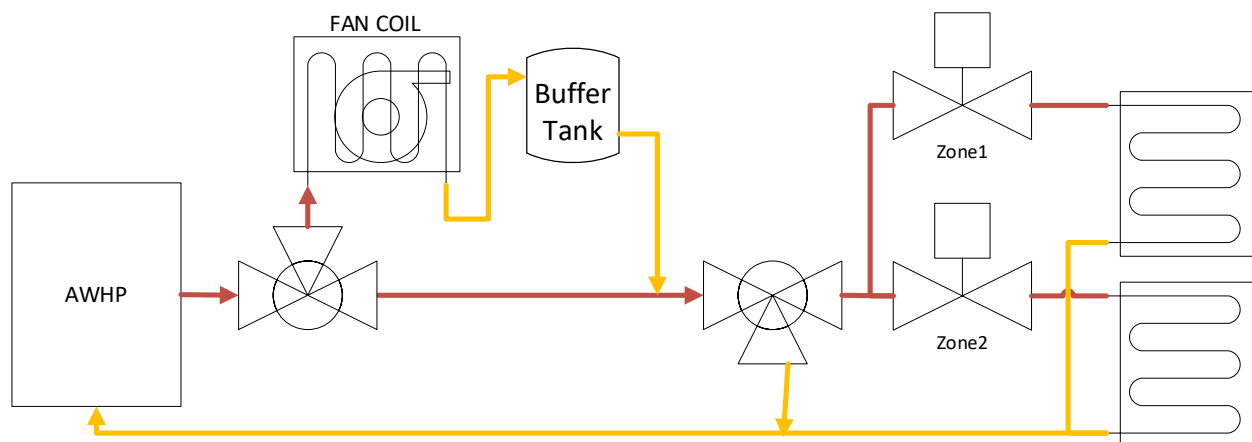


Figure 4. System schematic for Method 3 (dehumidification provided by ductless hydronic fan coil, summer 2019).

Though the fan coil is plumbed in series with the panels, a motorized three-way bypass valve was used to allow operating both the fan coil and panels, panels only, or fan coil only, depending on the combination of cooling and dehumidification calls. The fan speed was varied depending on the need for increased dehumidification. Though the sensible heat ratio was higher at higher fan speeds, the total latent removal was also higher.

During commissioning, many problems were encountered with this configuration. The fan coil, which was produced by the same manufacturer as the AHP, was too restrictive for the AHP's internal pump, even when operating in fan coil only mode. This was resolved by adding a second variable speed pump to boost flow rates.

However, despite the increased flow rates, running in fan coil only mode resulted in excessive short cycling and very limited dehumidification. The AHP would only run for a few minutes, as it could not modulate enough to prevent the supply temperature from dropping quickly and hitting the freeze protection limit. After consulting with the manufacturer, it was found that the total volume of the fan coil only loop was too low for the AHP to operate efficiently. The installed AHP requires a minimum system volume in all zonal combinations of at least 15 gallons. Therefore, a buffer tank was plumbed in-line with the fan coil return, not as a traditional buffer tank, but to simply increase the volume of the system when only the fan coil zone was open. This paper only considers data from after the buffer tank and additional pump were installed when evaluating Method 3.

Test Conditions

Internal heat gains from occupants and appliances were simulated using electric heaters and humidifiers. The heaters were cycled every five minutes for the time necessary to produce the desired average internal heat gain rate for the hour. Similarly, the humidifier was turned on every 15 minutes for the time necessary to produce the desired average latent heat rate for the hour.

The sensible and latent profiles are shown in Figure 5 and were varied each month using the multipliers in Table 2. The algorithms used to develop the profiles come from the 2013 Title 24 Alternative Calculation Method (ACM) Manual.

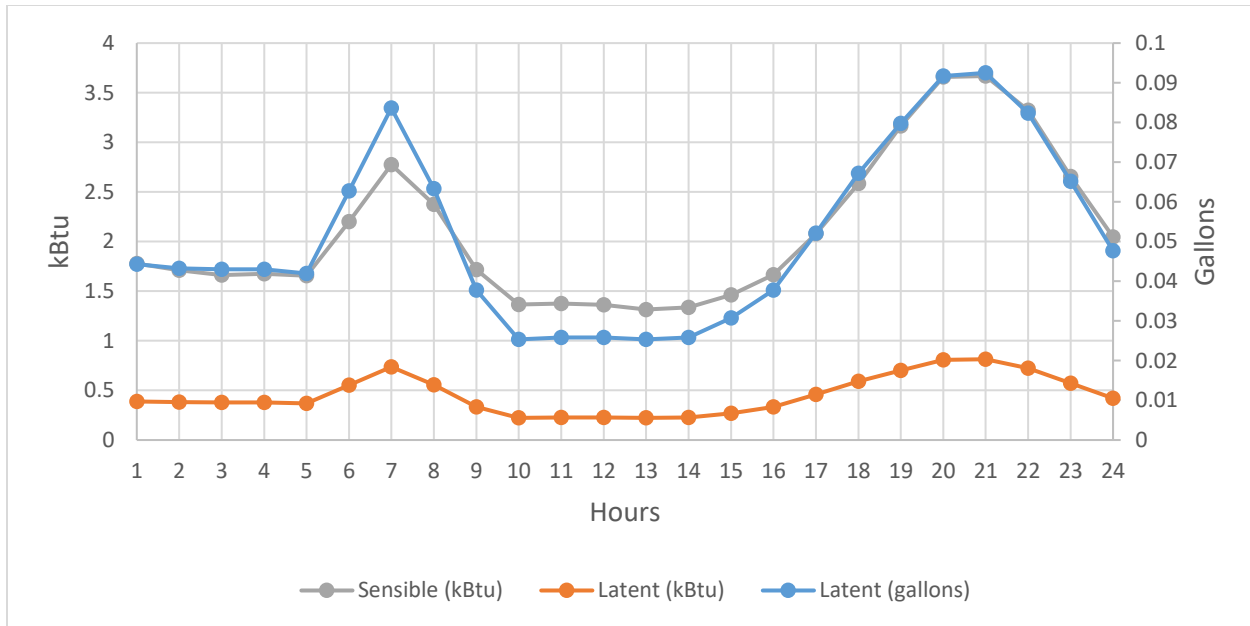


Figure 5. Internal heat gain profiles.

Table 2. Internal heat gain monthly multipliers

Month	Multiplier
1	1.19
2	1.11
3	1.02
4	0.93
5	0.84
6	0.80
7	0.82
8	0.88
9	0.98
10	1.07
11	1.16
12	1.21

Test Methods

Relative humidity, air temperature, and panel surface temperature measurements were recorded at 1-minute intervals. Instrumentation by room is shown in Table 3 below.

Table 3. Summary of instrumentation by room.

Room	Instrumentation
Bonus	• Relative Humidity (Vaisala HMP60) • IR Panel Surface Temp. (Omega OS211 LT) • Air Temperature (Aspirated Type T Thermocouple)
Bedroom 1	• IR Panel Surface Temp. (Omega OS211 LT) • Air Temperature (Aspirated Type T Thermocouple)
Bedroom 2	• IR Panel Surface Temp. (Omega OS211 LT) • Air Temperature (Aspirated Type T Thermocouple)
Bedroom 3	• Air Temperature (Aspirated Type T Thermocouple)
Bath	• Relative Humidity (Vaisala HMP60) (Method 1 and 2 only) • Air Temperature (Aspirated Type T Thermocouple)
Living/Dining	• Relative Humidity (Vaisala HMP60) • IR Panel Surface Temp. (Omega OS211 LT) • Air Temperature (Aspirated Type T Thermocouple)
Kitchen	• IR Panel Surface Temp. (Omega OS211 LT) • Air Temperature (Aspirated Type T Thermocouple)
Master Suite	• Relative Humidity (Vaisala HMP60) • IR Panel Surface Temp. (Omega OS211 LT) • Air Temperature (Aspirated Type T Thermocouple)

Due to the open floor plan of the house, a relative humidity (RH) sensor was not necessary for every room, only upstairs and downstairs. This was verified in 2018 by comparing monitored values for RH between two upstairs rooms and two downstairs rooms. In 2015, RH was measured in the Bonus Room (downstairs) and the Living Room (upstairs). In 2019, RH was measured in the Bonus Room, Downstairs Bathroom, Living Room, and Master Bedroom (upstairs).

Infrared averaging temperature sensors were used to measure the panel surface temperatures, but not all panels could be monitored. Due to the similarity of the bedroom panels, as well as observed mean water temperatures and air temperatures, the panel surface temperature in Bedroom 3 was not instrumented. The panel in the downstairs Bath also did not receive an IR sensor. As the house is unoccupied, there were no latent loads coming from the Bath and it was the smallest panel surface area. The Bath was therefore considered a low priority for measuring panel surface temperature.

Other instrumentation in the house and on the hydronic system included flow meters, supply temperature, and return temperature for panels in each room, as well as the hydronic fan coil. In 2015, pulse flow meters and pulse electricity meters were used to calculate thermal energy flows and electrical energy use by the AWHP and dehumidifier. In 2019, a Modbus data logger was added to collect data directly from the AWHP, as well as from revenue-grade BTU meters and electricity meters on each component of the system.

RH and air temperature were used to calculate the dewpoint for each room other than Bedroom 3 and the Bath. The air temperatures in each room were paired with the RH measurement closest to them, or an average of measurements, and the RH and dry bulb air

temperatures were used with the methods and psychrometric equations provided in ASHRAE Handbook Fundamentals. (ASHRAE 1985)

The differences between the panel surface temperatures and room dewpoints were calculated to evaluate the risk of condensation occurring on the panel surface. These differences are referred to in this paper as “surface and dewpoint deltas” (SADD). Condensation on the panel surface would be expected if the panel surface temperature were less than or equal to dewpoint ($SADD \leq 0$).

The comfort performance was compared in aggregate for entire seasons, using the criteria provided in ACCA Manual RS for air temperature deviation from setpoint (ACCA 1997). ACCA Manual RS also provides an upper and lower bound for RH (25 to 55 %RH) for evaluation at the Manual J design condition. RH that is too low causes dry skin, increases the risk of respiratory illness, makes allergies more acute, and other concerns. RH that is too high can promote bacteria and mold growth, and cause condensation which can damage building components and contents. In the 2020 ASHRAE Handbook HVAC Systems and Equipment, ASHRAE cites several studies on these issues and notes the optimum indoor RH range is likely between 30 and 60% (ASHRAE 2020). This RH range is used in this paper to evaluate comfort performance in addition to the temperature deviation from setpoint from ACCA Manual RS.

These metrics are provided in Table 4.

Table 4. Comfort criteria.

	Room-to-Setpoint Temperature Difference (°F)	Allowable RH Range (%RH)
Multi-Zone (Method 3)	2	30 to 60
Single Zone (Method 1 and 2)	3	

Additional details on the instrumentation and detailed test procedures for work at the radiant test house are provided in the Phase 1³ and Phase 2⁴ Emerging Technologies reports published by PG&E.

Monitoring Results

Condensation Risk

Table 5 shows the smallest SADD for each dehumidification method and supply water temperature. The smallest SADD was always with the prefabricated panels, which tended to operate at much lower surface temperatures than the site-built panels.

³ [Field Assessment of Residential Radiant Ceiling Panel Space Conditioning Systems | ETCC \(etcc-ca.com\)](https://www.etcc-ca.com/Field-Assessment-of-Residential-Radiant-Ceiling-Panel-Space-Conditioning-Systems)

⁴ [Central Valley Research Homes: Phase 2 Assessment of Residential Radiant Ceiling Panel Space Conditioning Systems | ETCC \(etcc-ca.com\)](https://www.etcc-ca.com/Central-Valley-Research-Homes-Phase-2-Assessment-of-Residential-Radiant-Ceiling-Panel-Space-Conditioning-Systems)

Table 5. Summary of condensation risk results.

Dehumidification Method	Radiant Supply Water Temp. (°F)	Smallest SADD ($\Delta^{\circ}\text{F}$)	Corresponding Surface Temp. (°F)	Corresponding Dewpoint Temp. (°F)	Dewpoint Setpoint (°F)
None (2015)	55	4.6	68.2	63.7	N/A
Dehumidifier (2015)	48	12.7	62.4	49.7	46
Fan Coil (2019)	52	6.6	61.2	54.6	56

The SADD seen without dehumidification was small, but not critical. However, if the supply water temperature had been several degrees lower (such as 48°F), assuming the same dewpoints, the panel surface temperature could have gone below the dewpoint.⁵ The closeness of the panel surface temperature approach to dewpoint using Method 1 and Method 2 can be visualized by plotting the 1-minute interval panel surface temperatures by dewpoint temperatures. This is shown in Figure 6, in which the straight diagonal line is where panel surface temperature would equal dewpoint. Any data points above this line would result in condensation on the panel surface.

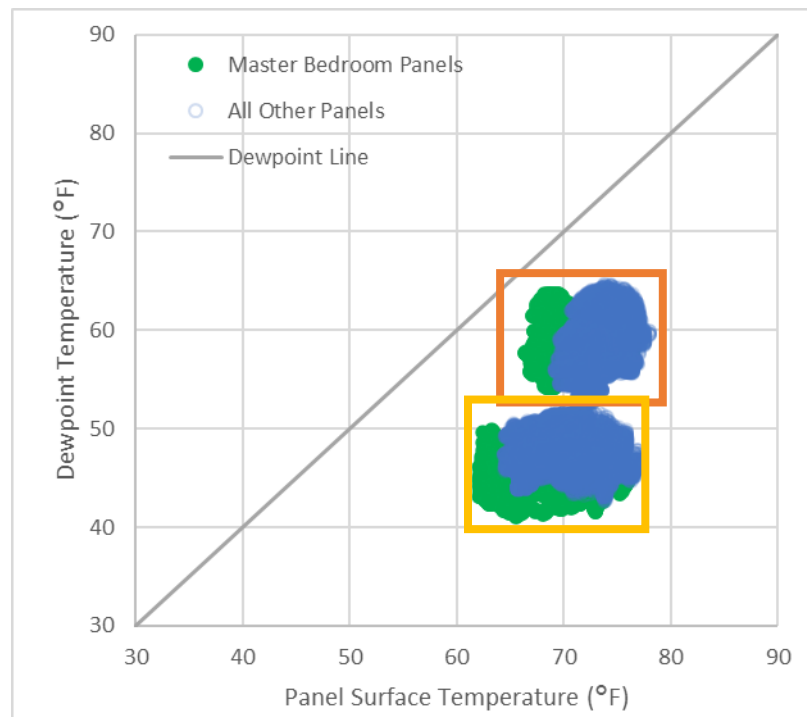


Figure 6. Panel surface temperature by dewpoint for Methods 1 (orange box) and 2 (yellow box).

⁵ As seen in Table 5, lowering the supply water temperature by 7 °F resulted in a 6 °F lower surface temperature. This is greater than the 4.5 $\Delta^{\circ}\text{F}$ SADD at the higher water supply temperature. Because the same latent gains schedule was used, it is reasonable to assume that the dewpoint without dehumidification would have been similar.

The same visualization can be seen for Method 3 in Figure 7 below.

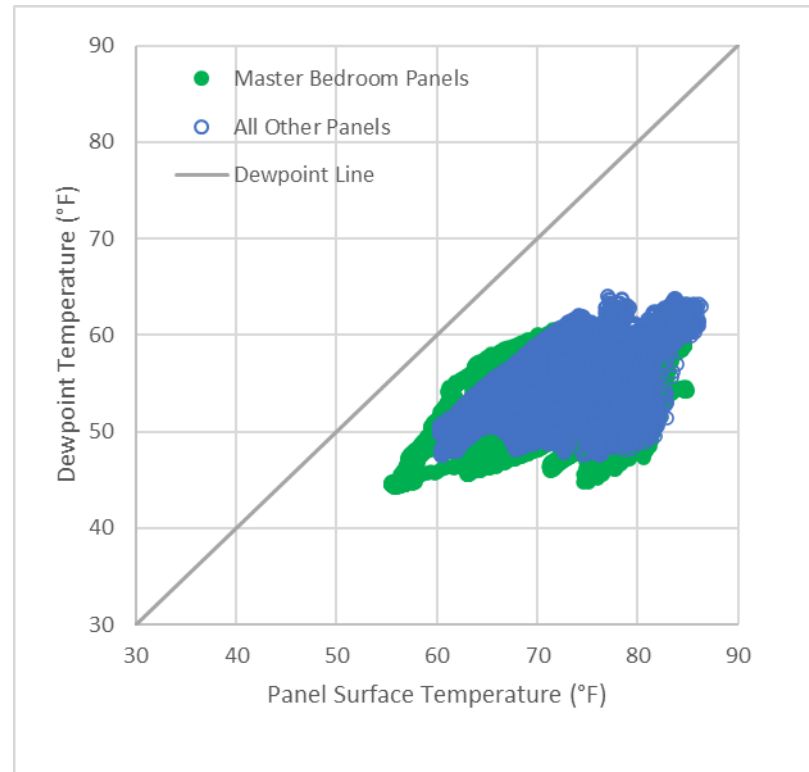


Figure 7. Panel surface temperature by dewpoint for Method 3.

The SADD seen with the free-standing dehumidifier (Table 5) is substantially larger than necessary, and this is the smallest observed value. Upstairs RH with the free-standing dehumidifier ranged between 32% and 39%, which is risking being too dry. This was due to the aggressive dewpoint target used with the dehumidifier, which was chosen to keep the dewpoint below the panel supply water temperature, which is standard practice (Watson and Chapman 2002).

However, observed panel surface temperatures were always considerably higher than the water supply temperature. The SADD seen with the fan coil is much smaller than that of the dehumidifier, using a less aggressive dewpoint target, based on years of observed differences between the supply water temperature and panel surface temperatures. Using this less aggressive target still provided a reasonable buffer between the panel surface temperature and dewpoint, while also not excessively drying the air. Upstairs RH with the fan coil was between 45% and 55%, a much more reasonable level.

Energy Consumption

Table 6 shows performance characteristics for the evaluated dehumidification methods.

Table 6. Performance characteristics of evaluated dehumidification methods.

Dehumidification Method	Avg. Power (W)	Avg. Daily Energy Use (kWh)	Average Condensate Removal (gal/day)	Cooling Provided by Method (kBtu/day)	Radiant Cooling Provided (kBtu/day)	Avg. Indoor Temp. (°F)
Dehumidifier (2015)	470	11.1	2.0	-38	212	73
Fan Coil (2019)	29	0.6	1.1	77	115	74.8

The fan coil used significantly less energy than the dehumidifier. This was due to a higher dewpoint target, avoiding excessive dehumidification, and because the fan coil's only electrical components consisted of basic controls and a small fan motor.⁶ Because the fan coil was allowed to operate at the same time as the radiant panels, it is impossible to disaggregate the AWHP energy use for dehumidification from the AWHP energy use for radiant cooling.

However, the same AWHP that was used in the 2019 evaluation of Method 3 was also used for other work with the radiant system in 2018. By looking at similar days with (2019) and without (2018) fan coil dehumidification for the same AWHP, we can get a sense of this disaggregation. For two hot summer days with similar outdoor conditions the AWHP consumed 12.2 kWh without using the fan coil for dehumidification, while it consumed 15.8 kWh with using the fan coil for dehumidification. That's an increase of 3.6 kWh, or about 29.5% over AWHP energy use for just the radiant panels.

The average daily energy use of the AWHP in 2019 was 15.1 kWh and the average daily energy use of the fan coil was 0.6 kWh. Based on this, the addition of a fan coil for dehumidification costs approximately 3.4 kWh in AWHP operation on average. Adding the average energy use of the fan coil itself (0.6 kWh/day), the total energy cost of using a fan coil for dehumidification with a radiant system is approximately 4 kWh. This is still much lower than the energy use of the standalone dehumidifier (11 kWh), indicating that a fan coil is the best method of providing dehumidification for a radiant panel system.

Even though the dehumidifier removed a greater condensate volume than the fan coil, this excess removal was unnecessary, and contributed significant thermal load to the space, requiring greater output from the AWHP. In contrast, the fan coil helped cool the indoor space.

Simulated latent gains were approximately 1.6 gallons/day. This is lower than the average daily condensate volume removed by the dehumidifier and higher than the volume removed by the fan coil. The lack of realistic internal mass (furnishings, books, etc.) which would normally absorb and release moisture and the use mechanical ventilation (ASHRAE 62.2-compliant IAQ in both 2015 and 2019, and a whole house fan operated in the mornings in 2015) may have affected the level of moisture in the indoor air during these evaluations.

⁶ The installed free-standing dehumidifier included compressor energy, which is attributed to the AWHP in the fan coil configuration.

Comfort Performance

Figure 8 shows the seasonal fraction of operating time that temperatures were within, above (undercooling), or below (overcooling) ACCA room-to-setpoint criteria by room. As seen from the ACCA RS results, the zoned radiant system (Method 3) provided better comfort than the single zoned radiant configuration (Methods 1 and 2). This is apparent in Method 2 where the upstairs rooms remained within the ACCA RS temperature ranges, but the downstairs rooms were overcooled. This was due to the dehumidifier's exhaust air warming the upstairs living room where the thermostat was located. This caused the single zone panels to overcool the downstairs while keeping the upstairs rooms at setpoint.

Method 3 shows the most desirable results, with lower undercooling compared to Method 1. Method 3 shows more overcooling compared to Method 1 due to the fan coil providing additional cooling over that provided by the radiant panels, including cooling when there were dehumidification calls outside normal cooling calls. These dehumidification calls were necessary, as radiant ceilings continue to be at a lower surface temperature after a cooling call, continuing to present a condensation risk even outside of active cooling.

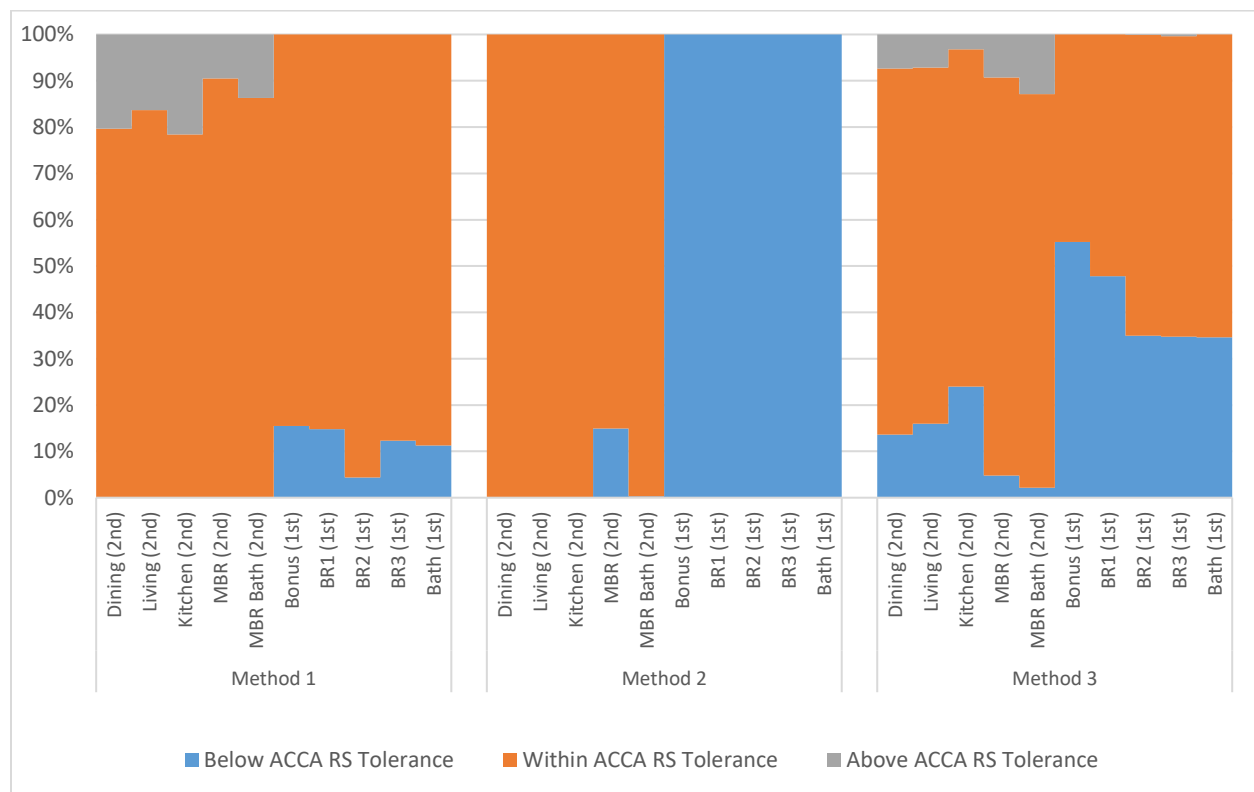


Figure 8. ACCA RS room-to-setpoint noncompliance rates by room.

Without the dehumidifier, RH was often close to 60%. However, as previously noted, with the dehumidifier, upstairs RH ranged between 32% and 39%, which is risking being too dry. Upstairs RH with the fan coil was between 45% and 55%, a much more reasonable and comfortable level.

Conclusions and Next Steps

Some dehumidification is prudent with radiant systems and dehumidification needs must be assessed as part of the radiant design process, especially if operating at low water temperatures to accommodate higher outdoor design temperatures. Given the closeness of the panel surface temperatures approach to dewpoint without active dehumidification, while not providing a means of active dehumidification is feasible, not providing a means of emergency dehumidification may not be wise. However, the 2019 evaluation of the integrated hydronic fan coil confirms that options exist for dehumidification with radiant systems that are more efficient and effective than standalone dehumidifiers.

This evaluation also demonstrated that the industry accepted rule of thumb to set a dehumidification target to maintain dewpoints less than panel supply water temperature needs to be re-evaluated. Using this method resulted in unhealthily dry air, and excessive energy use. Ideally, radiant system controls would directly monitor panel surface temperature and dewpoint, then adjust supply water temperature and integrated dehumidification methods accordingly. More research needs to be done to develop a new rule of thumb for dehumidification targets where such controls are not available, and standards for such controls should be developed.

Additionally, while a fan coil can provide adequate dehumidification, it also provides significant sensible cooling in the process, resulting in overcooling and duplication of function with the radiant panels. Using a ducted fan coil as the primary cooling method would be simpler, less expensive, would face fewer market barriers than radiant ceilings, and could serve both latent and sensible loads without the risk of condensation on interior surfaces

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