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# Modeling Assessment of Residential Air-to-Water Heat Pumps Coupled with Cooling Thermal Storage

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Building America

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# Modeling Assessment of Residential Air-to-Water Heat Pumps Coupled with Cooling Thermal Storage

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# List of Acronyms

| | |
|---|---|
| ASHP | air-source heat pump |
| AWHP | air-to-water heat pump |
| BEopt | Building Energy Optimization tool |
| COP | coefficient of performance |
| CVRH | Central Valley Research House |
| HVAC | heating, ventilating, and air conditioning |
| IECC | International Energy Conservation Code |
| RH | relative humidity |
| TES | thermal energy storage |
| TOU | time of use |
| ZERH | Zero Energy Ready Home |

# Executive Summary

This study explored the performance and operating cost viability of air-to-water heat pumps (AWHPs) coupled with thermal energy storage (TES) in efficient new residential construction. AWHPs are an emerging technology in the United States, but offer promise in terms of high efficiency, fully contained and factory-charged outdoor refrigeration systems, and hydronic delivery capabilities, which facilitate zoning, ducts in conditioned space, and TES integration for summer load shifting. Although this AWHP+TES strategy is not yet mainstream, the authors feel that in 10 years as decarbonization efforts proceed and time-of-use (TOU) rates become more common, strategies such as this will be more accessible.

The Frontier Energy team developed EnergyPlus® simulation models based on detailed monitoring data collected over several years at Pacific Gas and Electric's Central Valley Research House (CVRH) laboratory test homes located in Stockton, California. One of the CVRH test homes (1,962 ft$^2$; two-story) had been used for testing various AWHP systems and configurations over the past six years.[1] The validated model was then updated with high-efficiency International Energy Conservation Code (IECC) Zero Energy Ready Home (ZERH) envelope and component requirements for climate zones 1–5, including ducts in conditioned space thermal distribution. The Frontier team completed simulations for the 1,962 ft$^2$ home in each climate zone for a minimum efficiency air-source heat pump (ASHP), an AWHP coupled with a fan coil, and an AWHP coupled with TES sized to eliminate summer on-peak compressor operation. To maintain consistency in reporting energy use estimates, all cases were run with a similar indoor thermostat control strategy to pre-cool the house below the nominal 76°F set point prior to the on-peak period and float slightly above the set point during the peak period.

The AWHP+TES configuration was controlled to alternately condition the indoor space or to charge the TES tanks prior to the beginning of the on-peak period. Three composite TOU rates were developed based on existing TOU rates across the United States to provide differing economic scenarios to evaluate customer bill impacts throughout the summer. Two of the TOU rates had short 3-hour peak periods, while the third rate had a longer 7-hour duration peak period.

AWHP modeling projections were based on the observed field performance of the Chiltrix CX34 variable-speed unit. Other products on the market or entering the market in the near term would likely perform differently.

Key conclusions from the EnergyPlus simulation effort include:

- In comparing ASHP to AWHP summer cooling energy usage without TES, the AWHP usage was projected to be 11%–15% higher in the humid climates, 6% higher in Phoenix, but 6% lower in the dry Denver climate.

[1] Most of the CVRH testing had been completed with the Chiltrix CX34 AWHP, which was then used as the validation reference for this study.

- The EnergyPlus AWHP+TES control strategy involved alternate hours of charging the TES tank (beginning in the early a.m.) and conditioning the indoor zone. (This alternating hour control strategy was necessary due to Energy Plus modeling limitations and is not reflective of real-world control strategies.) In addition, several hours before the peak period began, the indoor cooling set point was reduced from 76°F to 72°F to provide additional stored energy in the house. TES tank size was iteratively evaluated to allow for virtually all the on-peak cooling to be provided by the TES tank without compressor operation. For all but the Phoenix 7-hour duration peak case, 210 gallons of storage was adequate to meet peak loads. For the Phoenix case, 525 gallons of storage, a 15% larger capacity compressor, and a modified tank charging schedule was needed.
- The AWHP+TES load-shifting strategy was highly effective at moving cooling energy use to non-peak hours. For the shorter 3-hour on-peak rates, 55% of non-storage AWHP cooling energy usage was shifted from the on-peak period; for the longer 7-hour peak, 66% was shifted.
- In comparing the AWHP+TES summer utility bills for the 3-hour duration TOU rates to non-storage AWHP bills with similar cooling thermostat set points, homeowner cost savings were essentially zero. This can be largely attributed to the effectiveness of the advanced ZERH building shell assumptions coupled with pre-cooling. However, for the 7-hour peak period (Rate 2), summer cooling utility bill savings averaging 11% were realized. This suggests that when coupled with advanced construction methods, a short duration peak period may not be the best application for this technology. TOU rate structures and other forms of utility incentives for load-shifting may result in different conclusions.
- This initial scoping effort shows favorable results for this technology option pending the structure and configuration of TOU rates. Utilities will need to focus increasing attention on TOU rate development in the future as summer temperatures continue to climb, air conditioning saturations increase, and peak loads continue to rise.
- As the AWHP technology matures in this country, a more detailed application assessment of preferred designs and system costing should be completed.

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# 1 Introduction

The predominant residential heating, ventilating, and air conditioning (HVAC) equipment installed in most new U.S. single-family homes over the last several decades has been ducted forced air heat pumps or gas furnaces coupled with an air conditioner. Alterative systems such as minisplit heat pumps (either ducted or non-ducted) have made a significant jump in sales in recent years as the systems are easily installed and offer increased installation flexibility. With an increasing national focus on decarbonizing the building sector, ground source heat pumps and air-to-water heat pumps (AWHPs) have also been gaining traction. AWHPs are an emerging technology that offer an alternative to mainstream HVAC equipment with potential to provide improved energy efficiency and reduced summer peak demand. Although prominent in other parts of the world, the AWHP market is currently small in the United States. Like conventional equipment, AWHPs consist of an outdoor unit that contains a compressor, a finned refrigerant coil, and a fan. What is different with AWHPs is that they have a fully contained and factory-charged refrigerant system in the outdoor unit and use water (or water glycol mix in cold climates) to convey heating or cooling to the indoor distribution system. The indoor hydronic heat exchange facilitates compact zoned thermal delivery and alternative distribution opportunities (e.g., fan coils, radiant floors, radiant ceiling panels) and also simplifies the integration of thermal storage tanks for load-shifting capability. Figure 1 details the basic components of the AWHP outdoor unit with hydronic supply and return piping shown penetrating an exterior wall for indoor delivery; this configuration shows the system operating in cooling mode.

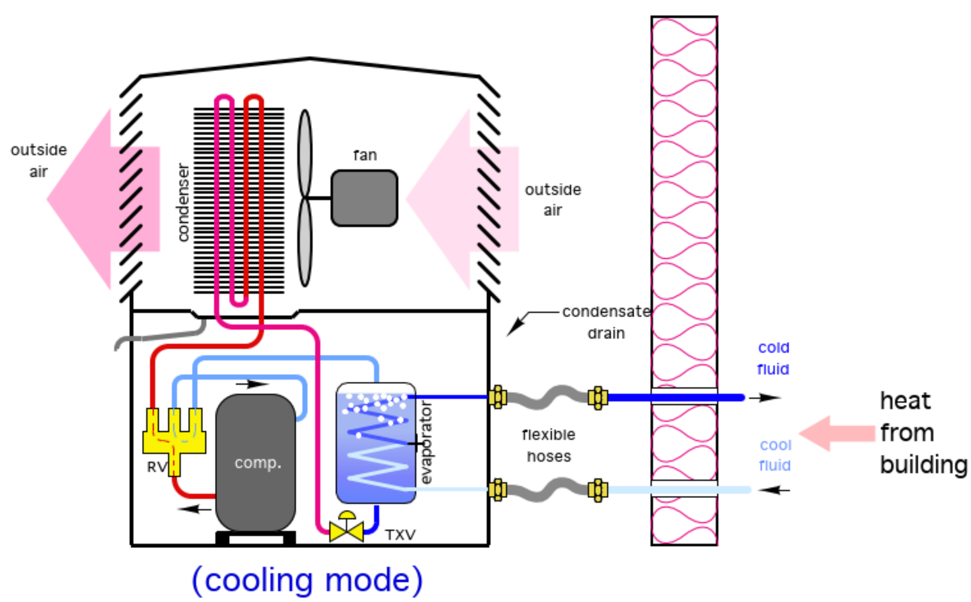


**Figure 1. AWHP outdoor unit schematic (cooling mode)**

Image credit: John Siegenthaler

Hydronic delivery from the outdoor unit facilitates locating piping within conditioned space to the indoor fan coils and also simplifies zoned delivery and facilitates short ducting solutions. In addition to providing year-round space conditioning, several AWHPs currently on the market allow for dedicated domestic water heating. This "three function" operation is attractive because it not only allows for the unit's high-capacity compressor to be used to satisfy water heating loads,[2] but it also simplifies electrical service requirements given that an AWHP would occupy a single breaker on an electrical panel rather than two breakers for a conventional air-source heat pump and a unitary heat pump water heater. Although the focus of this study is on the new construction market, the single electrical breaker feature is a significant advantage for the existing home replacement market where electrical panel constraints often present an installation barrier. In the retrofit case, the existing indoor air handler (or furnace) would be replaced with a hydronic fan coil unit.

As the United States moves toward electrification of the building sector to help combat climate change, it is critically important that added electrical loads associated with heat pump technologies operate synergistically with renewable generation resources that are becoming increasingly common on the electric grid. The term "duck curve" has gained prominence in recent years with the growth of renewable electric generation in much of the country. This term refers to the impact of renewable generation on the grid's hourly demand profile. As renewables increase, the shape of the curve follows that of a duck with the midday low being the lower portion of duck's back (due to maximum solar generation) and the high late afternoon peak (when the reduction in solar output late in the day coincides with the afternoon increase in electrical demand) being the duck's head. Increasingly extreme grid demand fluctuations occur as PV ramps up and down during the course of a day. Though long considered a "California problem" due to the significant share of renewable generation in the state,[3] the duck curve effect has started to appear in other parts of the country.

As AWHPs and other heat pump technologies gain market share, optimizing how these systems integrate with the grid becomes increasingly important to ensure that added electrical demand occurs during the most favorable times of day. Hydronic systems such as AWHPs are particularly amenable to load-shifting because they can be easily coupled to thermal energy storage (TES) tanks. TES systems can satisfy on-peak cooling load using off-peak compressor operation to chill the tanks. Indoor air pre-cooling prior to the peak period using conventional air-to-air systems is prone to comfort issues due to the need for large indoor temperature swings during the day (German & Hoeschele, 2014). Storing thermal energy in insulated storage tanks allows for the potential of storing more energy for discharge during the peak period. This study completed detailed EnergyPlus simulation modeling to evaluate the potential of AWHPs coupled

[2] Common unitary heat pump water heaters currently available on the market have a compressor capacity < 0.5 tons, while most AWHPs have a 2-ton (or larger) compressor.
[3] https://www.energy.ca.gov/data-reports/energy-almanac/california-electricity-data/2020-total-system-electric-generation

with TES to provide summer load-shifting benefits both to homeowners (in the form of reduced utility costs) and to utilities.

Prior research supporting the development and validation of the EnergyPlus models used in this study is based on ongoing activities at the Central Valley Research House (CVRH) project located in Stockton, California. The four CVRH laboratory test houses have been in use for energy efficiency experiments for over a decade. The houses serve as test beds for evaluating various technologies and to support enhancements to California's Title 24 residential energy building standards. Three of the houses have been used most recently to evaluate minisplit heat pump performance, and the fourth house has been used for evaluating AWHPs in heating and cooling operation using either radiant ceiling panels or fan coils for delivery. The AWHP lab house is a 1,962 $ft^2$ two-story, slab-on-grade home built in 1996 but extensively upgraded about ten years ago as part of an aggressive energy efficiency retrofit demonstration project. In addition to the AWHP, the house (as-built building parameters highlighted in Table 1) included two high-efficiency forced-air systems, a heat pump and an air conditioner, both with ducts in conditioned space. These conventional systems have each been used as reference systems for comparison with the AWHP system.[4]

**Table 1. Modeled As-Built House Characteristics**

| House Characteristic | Modeled Value |
|---|---|
| Wall Construction | R-13, 2x4, 16-inch on center |
| Slab Insulation | Uninsulated slab |
| Ceiling Insulation | R-49 at ceiling plane in vented attic |
| Envelope Leakage | 5.2 air changes/hour at 50 Pascals |
| Duct Location | Conditioned space |
| Window Solar Heat Gain Coefficient | 0.25 |
| Window U-Factor | 0.30 Btu/hr-$ft^2$-°F |

[4] These systems are fully redundant installations. One will remain with the house once the CVRH research project concludes. The other is a second reference system with air handler and all ducts located within the living space.

# 2 Technical Approach

## 2.1 Overview

This modeling study was developed to provide a detailed assessment of the energy and operating cost performance of AWHPs coupled with TES under different utility rate structures and different climates. The work builds from ongoing Pacific Gas and Electric supported research activities at one of the four CVRH laboratory houses located in Stockton, California. An initial assessment of summer TES performance was completed in 2021 (Haile, Springer, & Hoeschele, 2021), with a follow-on study underway at the time of this report. The AWHP lab test house was one of four CVRH homes extensively retrofitted from 2013–2014 (Proctor, Wilcox, & Chitwood, 2016) with significant attention to air sealing, integration of ducts in conditioned space, and upgrading of accessible envelope components and HVAC equipment. Thorough high-resolution monitoring has been ongoing since the initial project as different AWHP experiments have been completed (Haile, Springer, & Hoeschele, 2016).

As both AWHPs and residential TES are emerging technologies, the authors decided to frame this modeling effort from the perspective of a high-performance home that might realistically be built by the end of this decade. The home specifications were derived from Zero Energy Ready Home (ZERH) requirements.[5] A key ZERH feature is the requirement for ducts within conditioned space. This feature is an impactful measure for efficient new homes and for the purposes of this study provides the added benefit of completely isolating equipment performance differences from any thermal distribution delivery differences.

Simulation models were developed for reference forced-air air-source heat pumps (ASHPs), AWHPs without TES, and AWHPs coupled with TES. The AWHP performance was based on one of the AWHPs that had been installed and monitored at the lab house (the Chiltrix CX34).

Simulations were completed in International Energy Conservation Code (IECC) climate zones 1 through 5 using the representative cities of Miami, Phoenix, Dallas, Washington, D.C., and Denver, respectively. Because TES operation involves shifting compressor operation from on-peak to off-peak periods, time-of-use (TOU) utility rates were applied to the simulations to project operating cost impacts.

## 2.2 Develop Time-of-Use Utility Rates

A performance evaluation of an advanced load-shifting strategy such as AWHPs coupled with TES involves applying utility rates to define load-shifting potential from both an energy use perspective, as well as a homeowner cost viewpoint. TOU rates are becoming increasingly common in many parts of the country as electric utilities are trying to modify homeowner behaviors through economic signals to add electrical load during favorable grid times and shed loads during peak periods when the grid is challenged to meet load without overloading

[5] https://www.energy.gov/sites/default/files/2019/04/f62/DOE%20ZERH%20Specs%20Rev07.pdf

distribution systems. In California, electric utility rate reform enacted in 2013 started the process of moving residential customers to mandatory TOU rates by 2019.[6] In other states, utilities have limited enrollment pilot TOU rates to better assess the impacts of future roll-out of TOU rates.

For this study, the authors reviewed 22 TOU residential rates from utilities across the country. Thirteen of the rates were not used to inform this study due to either rate complexity, the rate being a short-term pilot, or for overly long on-peak period duration (12 hours or more). The remaining rates highlighted in Table 2 present a sampling of current TOU rate offerings.

In reviewing all the TOU rates, it was decided to develop three prototype TOU rates that provide a range of peak period lengths, a range in utility costs consistent with the actual utility TOU rates, and the presence of a mid-peak period in one of the three rates. In addition, the TOU tariffs were configured to represent a range of electric rates from high rates common to much of California and the mid-Atlantic and lower rates common to the Midwest and Southeast. The rate differentials between TOU periods were influenced by the rate differentials observed in the actual rates. Each of the three composite rates shown in Table 3 were applied to each of the simulation runs. Figure 2 graphically represents the rate transitions by time of day.

**Table 2. Existing Sample Utility TOU Rates**

| Utility/Location | Rate | On-Peak Period |
|---|---|---|
| Southern California Edison (CA) | TOU-D | 4–9 or 5–8 p.m. |
| Dominion Energy (VA) | Off-peak Plan | 3–6 p.m. |
| Georgia Power (GA) | TOU-RD-5 | 2–7 p.m. |
| Xcel (CO) | RE-TOU | 3–7 p.m. |
| Arizona Public Service (AZ) | TOU-E | 3–8 p.m. |
| Ameren (MO) | R-TOU | 2–7 p.m. |
| Sacramento Municipal Utility District (CA) | Standard rate | 5–8 p.m. |
| Salt River Project (AZ) | EZ-3 rate | 3–6 or 4–7 p.m. |
| PEPCO (Washington, DC) | R-TOU | 2–6 p.m. |

[6] https://www.cpuc.ca.gov/industries-and-topics/electrical-energy/electric-rates/residential-rate-reform-r-12-06-013

**Table 3. Summer TOU Rate Assumptions for Parametric Analysis**

| Rate | On-Peak | Off-Peak | Mid-Peak |
|---|---|---|---|
| Rate 1: Short Peak Period, High Rates (5–8 p.m. peak period) | $0.376/kWh | $0.140/kWh | n/a |
| Rate 2: Broad Peak Period, Lower Rates (1–8 p.m. peak period) | $0.220/kWh | $0.045/kWh | n/a |
| Rate 3: Short Peak + Mid-Peak (5–8 p.m. peak, 1–5 p.m. & 8 p.m.–12a.m. mid-peak) | $0.260/kWh | $0.090/kWh | $0.151/kWh |

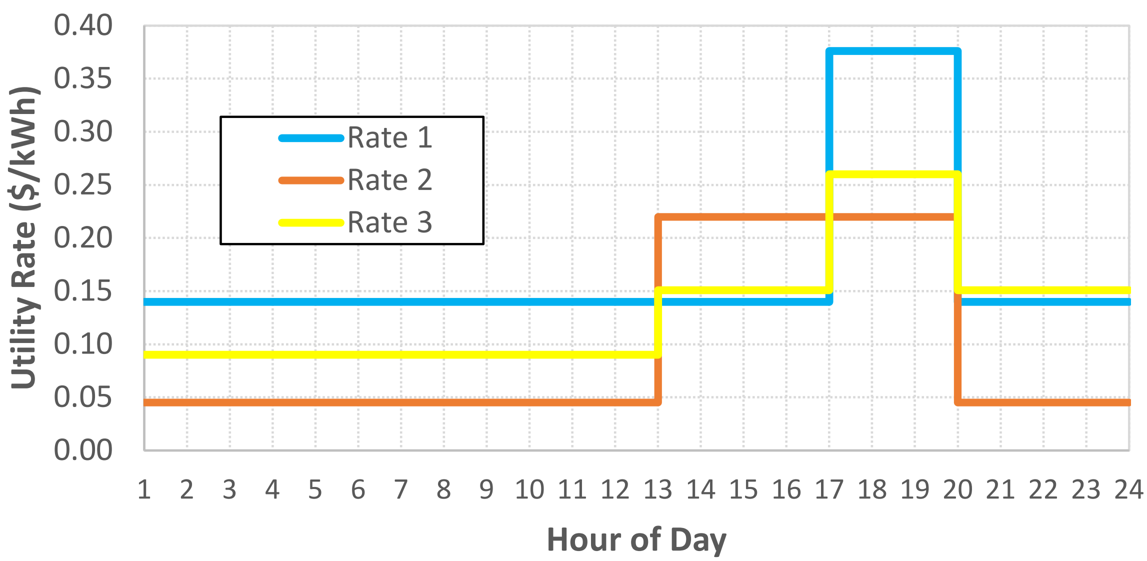


**Figure 2. TOU rate representation**

## 2.3 House Loads Model Validation

The CVRH laboratory test house utilizes controlled electric resistance heaters in each room to simulate occupants and other internal gains, as well as humidifiers to simulate latent load sources. In early phases of the CVRH project, these electric heaters were also used to maintain winter heating set points as a simple way to capture heating load without bringing equipment efficiencies into play. The electric resistance heaters represent an ideal model validation case, because electric resistance heating efficiency is known and stable (100% efficient) and temperature control in each of the rooms is highly consistent. Actual meteorological year weather data from Stockton, California, during the corresponding monitoring period was used to drive the "as-built" Building Energy Optimization Tool (BEopt™) house model, assuming electric resistance heating and no duct losses. Resistance heater monitored and modeled output data was compared for the period from December 15 to December 25, 2015, with several days removed when the resistance heaters were not being used. For three 48-hour resistance heating periods in this time frame, the final validated model was projected to consume 208.5 kWh, while monitored energy totaled 240.8 kWh during the same period (13.4% deviation).

## 2.4 HVAC Equipment Model Validation

With the basic house model reasonably validated with the electric resistance heating data, the validation process advanced to making sure that the HVAC equipment modeling aligned with the monitoring data. This involved comparison for both the forced ASHP and the AWHP installed at the CVRH test house.

During the winter of 2016 and the summer of 2017, the CVRH test house utilized a two-stage ASHP to condition the home. This reference heat pump system (Amana/Goodman ZSZC160241AE outdoor unit and AVPTC313714AA air handler) was modeled in BEopt in place of the resistance heaters to validate the model's cooling loads and provide a point of comparison for the AWHP. (This step was taken prior to modeling the AWHP system because the BEopt ASHP model is well established, while the AWHP must be modeled in EnergyPlus™ outside of BEopt.) The specifications shown in Table 4 were used in the BEopt model.

**Table 4. ASHP Modeled Characteristics**

| Specification | Modeled Value |
|---|---|
| Number of Speeds | 2 |
| Rated HSPF | 9.5 |
| Rated SEER | 16.0 |
| Rated EER | 12.5 |
| COP at 47°F Outdoor Air Dry Bulb | 3.8 |
| Nominal Capacity | 24,000 Btu/hour |
| Indoor Fan Power Efficacy | 0.15 Watts/cfm |

The reference ASHP monitoring and modeled data was compared for a period from January 19 to February 5, 2016, and a period from July 1 to August 26, 2017. Days during this period when the house was not conditioned by the reference heat pump were removed from the comparison. In the winter of 2016, the resulting periods of comparison are a 12-day period in January and a 4-day period in February. In the summer of 2017, the resulting periods of comparison are for two 24-hour periods and five 55-hour periods in July and August. The monitoring data used for comparison was sampled at a frequency of 10 minutes. During these periods, the model and existing building were controlled by the reference heat pump (heating set point of 68°F and a cooling set point of 76°F).

For the heat pump heating comparison, the final validated model used 107.3 kWh during the comparison period, while monitored energy totaled 92.6 kWh during the same period (15.9% deviation). For the heat pump cooling comparison, the final validated model used 215.3 kWh during the comparison period, while monitored energy totaled 201.0 kWh during the same period (7.2% deviation). Because this study was focused on cooling performance, the authors concluded that the 7.2% deviation was acceptable.

Once the reference heat pump model validation was completed, the model was fully transitioned to EnergyPlus version 9.6 so that the ASHP model objects could be replaced with the required EnergyPlus objects that characterize an AWHP system including modeled representations of the AWHP, hydronic thermal distribution loops, water pump, and fan coil. This step was necessary because BEopt does not have the ability to model AWHP systems. The AWHP system was modeled using the AWHP objects HeatPump:PlantLoop:EIR:Cooling and HeatPump:PlantLoop:EIR:Heating. The characteristics of the AWHP shown in Table 5 were defined using manufacturer data tables. The Chiltrix CX34 unit is an inverter driven, variable capacity unit with nominal capacities of 2.0 tons of cooling and 2.75 tons heating. Internal controls vary the speed of the compressor, pump, and fan based on a desired entering water temperature and water temperature difference across the heat exchanger. Coefficient of performance (COP) values reported in Table 5 include the AWHP outdoor unit electrical demand but not the pump energy for supplying the indoor fan coil.

**Table 5. AWHP Modeled Characteristics**

| Specification | Modeled Value |
|---|---|
| Condenser Type | Air Source |
| Outdoor Reference Temperature Cooling [°F] | 95 |
| Leaving Water Temperature Cooling [°F] | 44.6 |
| Reference Capacity Cooling [Btu/hr] | 26,100 |
| Reference COP Cooling at 95°F | 3.23 |
| Outdoor Reference Temperature Heating [°F] | 47 |
| Leaving Water Temperature Heating [°F] | 95 |
| Reference Capacity Heating [Btu/hr] | 33,800 |
| Reference COP Heating at 47°F | 3.92 |
| Load Side Reference Flow Rate (gal/min) | 5.85 |
| Source Side Reference Flow Rate (gal/min) | 5.85 |

Manufacturer data tables specifying the performance and part load curves were also defined in the AWHP heating and cooling objects. The AWHP objects are each connected to two separate hydronic loops, one for heating and one for cooling. These hydronic loops contain a constant speed pump upstream of the AWHP object on the supply side, and a hydronic fan coil on the demand side that delivers thermal energy to conditioned space.

The AWHP monitored and modeled data were compared for a period from July 17 to October 10, 2020 (total of 78 days). During this period, the model and existing building were maintained by the reference heat pump to a cooling set point of 75°F. The modeling presented here assumes a deadband of 2.5°F below the set point for triggering both heating and cooling, consistent with how the AWHP was controlled in the CVRH field research project.

Review of the initial results suggested that the model was achieving much higher efficiencies relative to the monitored data. As a result, changes to the modeled AWHP part load curve were made to better match the efficiencies seen in the monitored data. The part load curve used by EnergyPlus was adjusted until monitored and modeled daily energy use at different daily average temperatures aligned. The part load modeling is a critical piece in trying to accurately represent the variable-speed performance of the AWHP system. This remains the biggest challenge working within the EnergyPlus framework for this system type. In addition to the part load curve changes, the modeled fan coil fan efficacy (W/cfm) and pumping efficiency were changed to match monitored data. For the cooling comparison, the final validated model projected 1,224.6 kWh usage during the comparison period, while monitored energy totaled 1,278.4 kWh (4.2% deviation). It is important to note that although the AWHP modeled energy use was very close to actual, the EnergyPlus model is not able to fully mimic the variable-speed operating characteristics of the Chiltrix unit.[7]

After the validation of the AWHP model was complete, a 500-gallon TES tank was coupled with the EnergyPlus AWHP model to mimic the 2019 CVRH lab house installation configuration. The AWHP model was configured so that the unit could either charge the TES tank or condition the home independently, and also so that the TES tank could discharge independently of AWHP operation. In order to mimic actual system operation in EnergyPlus, the TES was connected in parallel with the AWHP on the supply side of the hydronic loop. This hydronic loop was connected to the use side of the TES tank. The source side inlet and outlet of the TES tank were connected to a second hydronic loop that contained the TES tank on the demand side of the loop and a copy of the AWHP object on the supply side of the hydronic loop. The resulting configuration is detailed in Figure 3.

This configuration allowed for the TES element to exist in parallel with the AWHP to either condition the zone or charge the TES tank. This parallel conditioning approach (AWHP 1 to condition zone, and AWHP 2 to charge storage) was necessary because EnergyPlus does not allow for TES to be modeled both in parallel and in series with the AWHP within the same hourly time step. The lower limit supply water temperature for TES tank charging was set at 46°F, consistent with CVRH findings.

Although field performance of the Chiltrix CX34 was relied on to characterize performance of the AWHP for this modeling study, there are other products that are (or might) be available in the market in the coming years. For example, Chiltrix is coming out with a new larger capacity

[7] Operating cycles begin with high-speed operation for several minutes, at which time the unit adjusts compressor speed to an appropriate level. The model assumes averaged electrical demand during a cycle.

AWHP in the fall of 2022,[8] Enertech has an available three function unit,[9] Mitsubishi has a three function R32 refrigerant AWHP[10] available in the United Kingdom (but not yet in the United States), and PHNIX has a three function R290 (propane) refrigerant unit[11] that is also not yet approved in the United States.

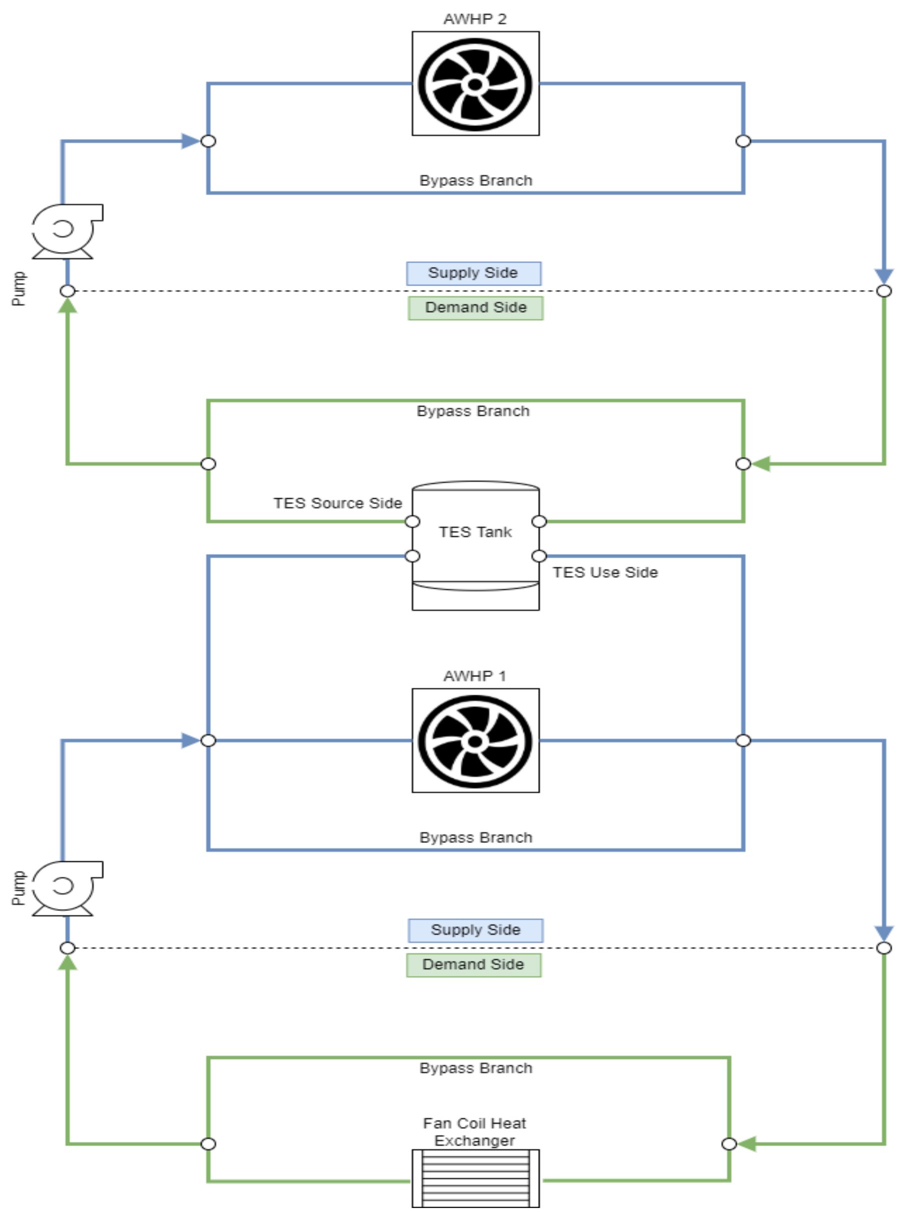


**Figure 3. AWHP + TES schematic**

[8] https://www.chiltrix.com/CX50-air-to-water-heat-pump/

[9] https://enertechusa.com/products/air-source

[10] https://les.mitsubishielectric.co.uk/products/residential-heating/outdoor/ecodan-r32-ultra-quiet-puz-monobloc-air-source-heat-pump

[11] https://www.phnix-e.com/r290-greentherm-heat-pump.html

A wide range of potential tank options was considered, including large-volume American Society of Mechanical Engineers-rated tanks, atmospheric site-built membrane lined tanks, and other options, but for simplicity modular 105-gallon insulated polybutene tanks were selected.[12] These tanks are (effectively) lifetime non-corroding tanks that, coupled with their familiarity for plumbers, are an attractive feature for builders, while also providing storage size customization capability by ganging multiple tanks together. TES tank sizing for the different climates and utility peak periods were completed in each case. Sizing runs completed in this study found 210 gallons of storage appropriate for all but the Phoenix 7-hour on-peak rate case, in which case 525 gallons of storage and a compressor size 15% larger than the Chiltrix compressor were determined to be appropriate.[13]

## 2.5 Development of Simulation Models for HVAC System Evaluations

The 1,962 ft$^2$ CVRH lab house configuration was the basis for all simulation evaluations. The house envelope characteristics were adjusted by climate zone to meet the current ZERH requirements (version 1, revision 7). Table 6 summarizes the key modeled envelope and mechanical specifications by IECC climate zone. Full year performance for base case ASHP, AWHP, and AWHP +TES was projected for each climate zone. ASHP and AWHP cases without storage were initially completed with a fixed indoor cooling set point of 76°F.

**Table 6. Modeled Home Characteristics by IECC Zone**

| House Characteristic | Zones 1 and 2 | Zones 3 and 4 | Zone 5 |
|---|---|---|---|
| Wall Construction | R-20, 2x6, 16-inch o.c | R-20, 2x6, 16-inch o.c | R-20, 2x6, 16-inch o.c |
| Slab Insulation | Uninsulated slab | 3- Uninsulated slab<br>4- R-10 to 2 ft depth | R-10 to 4 ft depth |
| Ceiling Insulation | 1- R-30<br>2- R-38 | 3- R-38<br>4- R-49 | R-49 |
| Envelope Leakage | 3.0 ACH50 | 2.5 ACH50 | 2.0 ACH50 |
| Window Solar Heat Gain Coefficient | 0.25 | 0.25 | Any |
| Window U-Factor | 0.40 Btu/hr-ft$^2$-°F | 0.30 Btu/hr-ft$^2$-°F | 0.27 Btu/hr-ft$^2$-°F |
| Mechanical Ventilation | As per 2019 ASHRAE 62.2 requirement | | |
| Duct Location | Conditioned Space | Conditioned Space | Conditioned Space |

[12] https://cdn.globalimageserver.com/FetchDocument.aspx?ID=AE473595-F539-4FA2-B755-8CE908501260
[13] It should be noted that storage sizing for on-peak compressor avoidance is dependent on climate, house size and peak cooling load, compressor capacity, and storage volume. In addition, operating a variable-speed AWHP at a fixed low demand level all through the peak period is another control option, which unfortunately was not within EnergyPlus modeling capabilities.

For the AWHP+TES case, utility rates 1 and 3 feature a 3-hour duration peak period (5–8 p.m.) while rate 2 is a 7-hour peak (1–8 p.m.). Given the modeling constraints whereby the AWHP must either be directed to charge storage or pre-cool the indoor space, operational configurations were developed for describing set points and conditions for the two on-peak rate durations. Table 7 describes the indoor cooling set points and time periods when the AWHP can be used to either condition the zone or charge the TES tank, and discharge time period when the TES tank is available to condition the zone for the 3-hour on-peak rate cases, while Table 8 provides the same information for the 7-hour peak period. For utility rate 2, the peak duration is 7 hours (1–8 p.m.), which is a more challenging load-shifting scenario, especially for high cooling load climates such as Phoenix. A series of runs were completed for the Phoenix rate 2 case to arrive at a system sizing of 525 gallons of thermal storage and a compressor sized at 8,792 W (30,000 Btu/hr), 15% larger than the nominal Chiltrix compressor size.

The thermostat schedules show a lowering of the thermostat set point from 76°F to 72°F for the 2 hours preceding the peak period in an effort to increase consumption of lower cost electricity. During the peak, the cooling set point is relaxed 2°F above the normal 76°F set point, again to improve customer utility costs. As the AWHP+TES runs were being completed for this project, the authors decided to apply the modified cooling set point schedule to all the simulation cases to maintain a direct comparison between results. This is representative of homeowners without TES being responsive to their TOU rate. Additionally, the morning TES charging was timed to begin roughly around sunrise for the 3-hour on-peak cases to better align with the beginning of photovoltaic contributions to the grid.

**Table 7. Cooling Set Points and Operating Capabilities for 5–8 p.m. Peak Period**

| Hour of Day | Cooling Set Point | Available AWHP Operating Mode | TES Tank Mode |
|---|---|---|---|
| Mid–8 a.m. | 76°F | Condition Zone (AWHP 1) | Idle |
| 8–9 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 9–10 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 10–11 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 11–12 Noon | “” | Condition Zone (AWHP 1) | Idle |
| 12–1 p.m. | “” | Charge TES (AWHP 2) | Charging |
| 1–2 p.m. | “” | Condition Zone (AWHP 1) | Idle |
| 2–3 p.m. | “” | Charge TES (AWHP 2) | Charging |
| 3–4 p.m. | 72°F | Condition Zone (AWHP 1) | Idle |
| 4–5 p.m. | 72°F | Condition Zone (AWHP 1) | Idle |
| 5–8 p.m. | 78°F | No compressor operation, unless Tin>81°F | Discharge |
| 8 p.m.–Mid | 76°F | Condition Zone (AWHP 1) | Idle |

**Table 8. Cooling Set Points and Operating Capabilities for 1–8 p.m. Peak Period**

| Hour of Day | Cooling Set Point | Available AWHP Operating Mode | TES Tank Mode |
|---|---|---|---|
| Mid–6 a.m. | 76°F | Condition Zone (AWHP 1) | Idle |
| 6–7 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 7–8 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 8–9 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 9–10 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 10–11 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 11–12 Noon | 72°F | Condition Zone (AWHP 1) | Idle |
| 12–1 p.m. | 72°F | Condition Zone (AWHP 1) | Idle |
| 1–8 p.m. | 78°F | No compressor operation, unless Tin>81°F | Discharge |
| 8 p.m.–Mid | 76°F | Condition Zone (AWHP 1) | Idle |

In the Phoenix 525 gallon 7-hour on-peak scenario, the charging schedule shown in Table 9 was used to better accommodate the larger tank size. This schedule begins tank charging several hours earlier in the morning, adds an extra hour of tank charging, and provides a longer uninterrupted period to condition the zone before pre-cooling.

**Table 9. Cooling Set Points and Operating Capabilities for Phoenix 525-Gallon TES Case With 1–8 p.m. Peak Period**

| Hour of Day | Cooling Set Point | Available AWHP Operating Mode | TES Tank Mode |
|---|---|---|---|
| Mid –2 a.m. | 76°F | Condition Zone (AWHP 1) | Idle |
| 2–3 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 3–4 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 4–5 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 5–6 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 6–7 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 7–8 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 8–9 a.m. | “” | Charge TES (AWHP 2) | Charging |
| 9–10 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 10–11 a.m. | “” | Condition Zone (AWHP 1) | Idle |
| 11–12 Noon | 72°F | Condition Zone (AWHP 1) | Idle |
| 12–1 p.m. | 72°F | Condition Zone (AWHP 1) | Idle |
| 1–8 p.m. | 78°F | No compressor operation, unless Tin>81°F | Discharge |
| 8 p.m.– Mid | 76°F | Condition Zone (AWHP 1) | Idle |

# 3 Simulation Results

The study results focus on energy use comparisons of the alternative HVAC systems, HVAC energy use breakdown by TOU period, and operating costs under the three prototype TOU utility rates. It is important to highlight that the AWHP findings are based on the observed performance of the Chiltrix CX34 AWHP.[14] As previously noted, new AWHP product offerings entering the market in the coming years will feature different performance characteristics, and therefore the study's findings should be viewed as a current snapshot of technology potential.

The findings focus on the summer utility period (June–September) when load shifting is most beneficial for near-term utility grid needs in most of the United States. In the future, as building decarbonization efforts progress, winter load-shifting performance will become equally important to ensure that added heat pump electrical demand does not exacerbate winter peak grid demands.

Table 10 provides a comparison of ASHP and AWHP energy usage during the June to September period in each climate zone. Energy usage is broken down into the projected outdoor unit use, indoor fan, and pumping energy use (the latter, only for the AWHP case). For the simulated climates, summer total ASHP cooling energy usage ranges by a factor of more than four between Denver and Phoenix. AWHP projected summer HVAC usage is 11%–15% higher than ASHP for the Dallas, Washington, D.C., and Miami climates, 6% higher in Phoenix, but 6% lower in Denver. It is not clear if the project performance variability between climates is dependent to some degree on modeling algorithm differences where AWHP performance is based on "validated" field performance, while the ASHP relies on default (non-field validated) assumptions.

**Table 10. Comparison of Summer Cooling Energy Usage (kWh)**

| End Use | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| **ASHP** | | | | | |
| Outdoor Unit | 1,974 | 2,928 | 1,931 | 977 | 747 |
| Indoor Fan | 597 | 943 | 647 | 333 | 173 |
| Pumping | 0 | 0 | 0 | 0 | 0 |
| *Total kWh* | *2,571* | *3,871* | *2,578* | *1,310* | *920* |
| | | | | | |
| **AWHP (no storage)** | | | | | |
| Outdoor Unit | 2,396 | 3,181 | 2,298 | 1,160 | 611 |
| Indoor Fan | 454 | 741 | 459 | 249 | 203 |
| Pumping | 113 | 173 | 113 | 63 | 50 |
| *Total kWh* | *2,963* | *4,095* | *2,870* | *1,471* | *864* |

[14] The Chiltrix CX34 cooling capacity and application to the 1,932 ft$^2$ home imply a sizing relationship.

The following tables compare the AWHP+TES configuration to the non-storage AWHP runs to quantify the impacts of integrating TES with an AWHP. Table 11 and Table 12 provide the comparison of the non-storage AWHP case with the AWHP+TES configuration. Two sets of AWHP+TES results are shown, one for the 3-hour on-peak period (utility Rates 1 and 3) and one for the 7-hour peak (Rate 2). Energy use comparisons between the Rate 1/3 simulations and Rate 2 simulations indicate lower usage for the longer peak period cases, presumably due to more favorable outdoor ambient temperature (lower condensing temperatures) during pre-cooling operation. Table 12 provides percentage differences for the AWHP+TES cases relative to the non-storage AWHP configuration. The addition of TES results in a 4%–19% increase in summer HVAC energy use for rates 1/3, with slightly improved relative performance for Rate 2 (3% savings, up to 17% increase in kWh). Interestingly, the generally dry Phoenix and Denver climates show the greatest increase in HVAC use, rather than the more humid Dallas, Miami, and Washington, D.C. climates.

**Table 11. Comparison of Summer Cooling kWh Usage (AWHP and AHWP+TES)**

| Case | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| **AWHP (no TES)** | | | | | |
| Rate 1/3 kWh | 2,963 | 4,095 | 2,870 | 1,471 | 864 |
| Rate 2 kWh | 2,950 | 3,990 | 2,820 | 1,451 | 837 |
| | | | | | |
| **AWHP (with TES)** | | | | | |
| Rate 1/3 kWh | 3,080 | 4,858 | 2,988 | 1,576 | 975 |
| Rate 2 kWh | 2,865 | 4,673 | 2,804 | 1,485 | 908 |

**Table 12. Percent Difference in Summer Cooling kWh (AWHP+TES vs. AHWP)**

| Case | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| **AWHP (with TES, utility rates 1 & 3)** | | | | | |
| Total kWh | 4% | 19% | 4% | 7% | 13% |
| | | | | | |
| **AWHP (with TES, utility rate 2)** | | | | | |
| Total kWh | -3% | 17% | -1% | 2% | 8% |

Table 13 disaggregates the summer usage into on-peak and off-peak consumption, and in the case of Rate 3 includes mid-peak period consumption. Table 14 shows the percentage change in usage for the on-peak and off-peak periods (and mid-peak for Rate 3) relative to the non-storage AWHP case. The benefit of moving compressor operation off-peak is clear, as summer HVAC on-peak usage is projected to be reduced by 38%–65% for the shorter 3-hour peak period and 54%–71% for the longer 7-hour peak.

**Table 13. Summer Cooling kWh by Utility Rate Period (AWHP vs. AHWP+TES)**

| TOU kWh | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| **AWHP (no storage)** | | | | | |
| Rate 1: On-peak | 93 | 557 | 166 | 10 | 15 |
| Off-peak | 2,869 | 3,538 | 2,704 | 1,462 | 849 |
| Rate 2: On-peak | 577 | 1,339 | 670 | 218 | 143 |
| Off-peak | 2,372 | 2,651 | 2,150 | 1,233 | 694 |
| Rate 3: On-peak | 93 | 557 | 166 | 10 | 15 |
| Mid-peak | 1,587 | 2,034 | 1,639 | 1,013 | 665 |
| Off-peak | 1,282 | 1,504 | 1,066 | 448 | 184 |
| | | | | | |
| **AWHP (with TES)** | | | | | |
| Rate 1: On-peak | 39 | 195 | 62 | 6 | 7 |
| Off-peak | 3,041 | 4,663 | 2,926 | 1,570 | 968 |
| Rate 2: On-peak | 173 | 392 | 190 | 75 | 67 |
| Off-peak | 2,693 | 4,270 | 2,614 | 1,410 | 841 |
| Rate 3: On-peak | 39 | 195 | 62 | 6 | 7 |
| Mid-peak | 1,575 | 2,473 | 1,626 | 1,036 | 706 |
| Off-peak | 1,465 | 2,190 | 1,300 | 534 | 262 |

**Table 14. Percent Reduction in Summer Cooling TOU On-Peak Usage (AWHP+TES vs. AHWP)**

| TOU kWh | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| On-peak | | | | | |
| Rate 1/3 | 58% | 65% | 62% | 38% | 53% |
| Rate 2 | 70% | 71% | 72% | 66% | 54% |
| Off-peak | | | | | |
| Rate 1 | -6% | -32% | -8% | -7% | -14% |
| Rate 2 | -14% | -62% | -22% | -14% | -21% |
| Rate 3 | -14% | -46% | -22% | -19% | -42% |
| Mid-peak | | | | | |
| Rate 3 | 1% | -22% | 1% | -2% | -6% |

Load-shifting operation is characterized by pre-peak charging of the TES tank coupled with pre-cooling of the indoor space to provide a small cushion for the AWHP's on-peak compressor avoidance strategy. With a short 3-hour peak period, the combination of pre-cooling the indoor space by several degrees and also charging the storage tank down to a 46°F lower limit temperature proved effective in successfully meeting most on-peak loads throughout the summer. The longer 7-hour peak associated with Rate 2 is clearly more challenging, especially in the extremely hot Phoenix summer climate. For Phoenix simulation cases, a 525-gallon TES tank with a 15% larger compressor size was modeled for the 7-hour peak. Figure 4 plots the daily indoor and outdoor temperature profile for the 111°F Phoenix summer peak day to compare the performance of the AWHP+TES operation relative to the non-storage AWHP case. During the post-midnight hours, both systems are maintaining indoor conditions slightly below the 76°F set point due to thermostat hysteresis effects. Beginning at 2 a.m., the AWHP+TES case is applying alternating hours of operation to charge the TES tank to a 46°F lower limit temperature, leading to slightly warmer indoor temperatures than the AWHP case. By around 8 a.m. the tank temperature is near its lower limit. During the early hours of the peak period, indoor temperatures for both cases float upward until the higher on-peak thermostat set point temperature is reached around 2 p.m. When the peak period begins, the TES tank also begins discharging, causing the tank temperature to steadily increase and the fan coil therefore becomes less effective at cooling the zone due to diminishing cooling capacity. In the later part of the peak period, the TES system starts to lose ground as the tank temperature is warming to a high of 67°F. At the conclusion of the peak, both the AWHP and the AWHP+TES systems recover back to the nominal cooling set point.

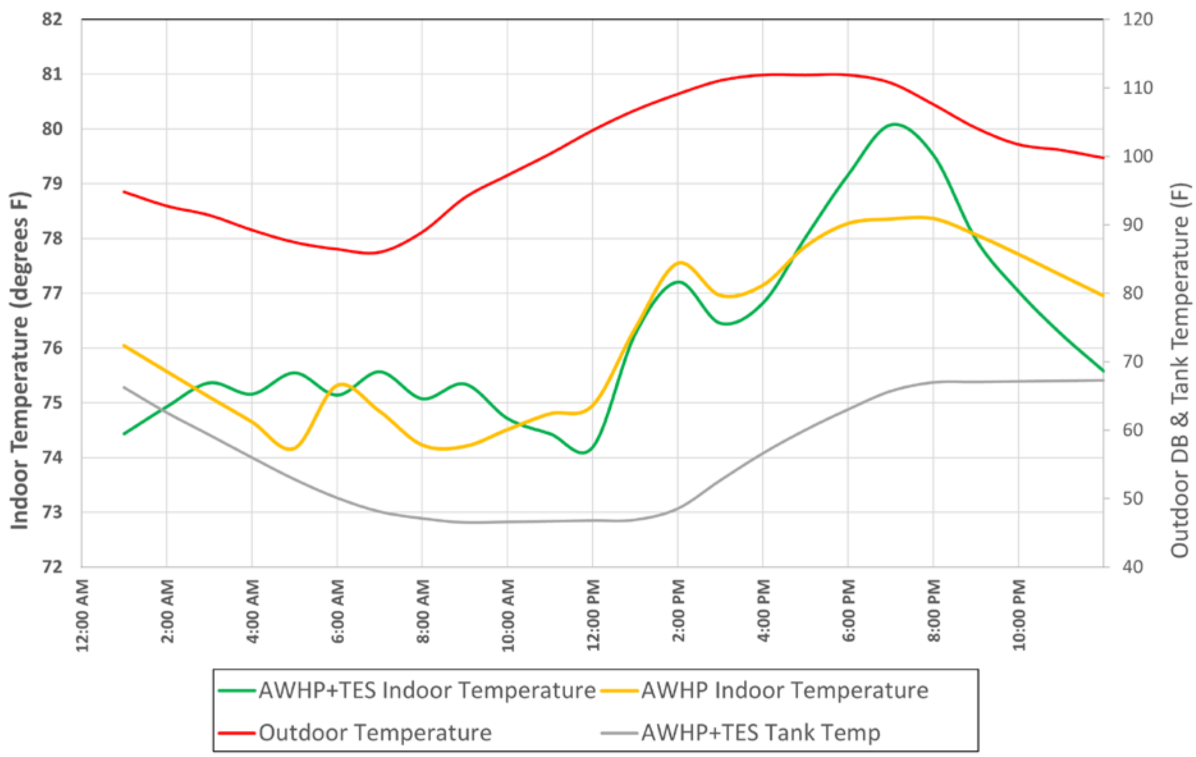


**Figure 4. Peak day AWHP and AWHP+TES indoor temperature profile (Phoenix 7-hour peak)**

Figure 5 plots similar hourly data showing the hourly cooling demand profile of the two system types. The ability of the AWHP+TES to shift cooling demand more effectively is evident.

Although the AWHP+TES consumes more energy over the course of the day (47.9 versus 40.4 kWh), the on-peak usage is reduced by 71% relative to the AWHP non-storage case (3.8 versus 13.1 kWh).

As observed in Figure 4, the AWHP+TES control strategy does involve some occurrences above a normal comfort condition. On the peak day, indoor temperature was projected to reach 80.5°F. For the full summer period, 43 hours were projected to exceed 79°F for this 7-hour peak period in the Phoenix climate. A control strategy allowing limited low-speed compressor operation to mitigate these higher indoor temperatures would be implemented at the expense of some on-peak usage, but the EnergyPlus model was unable to support that simulation case.

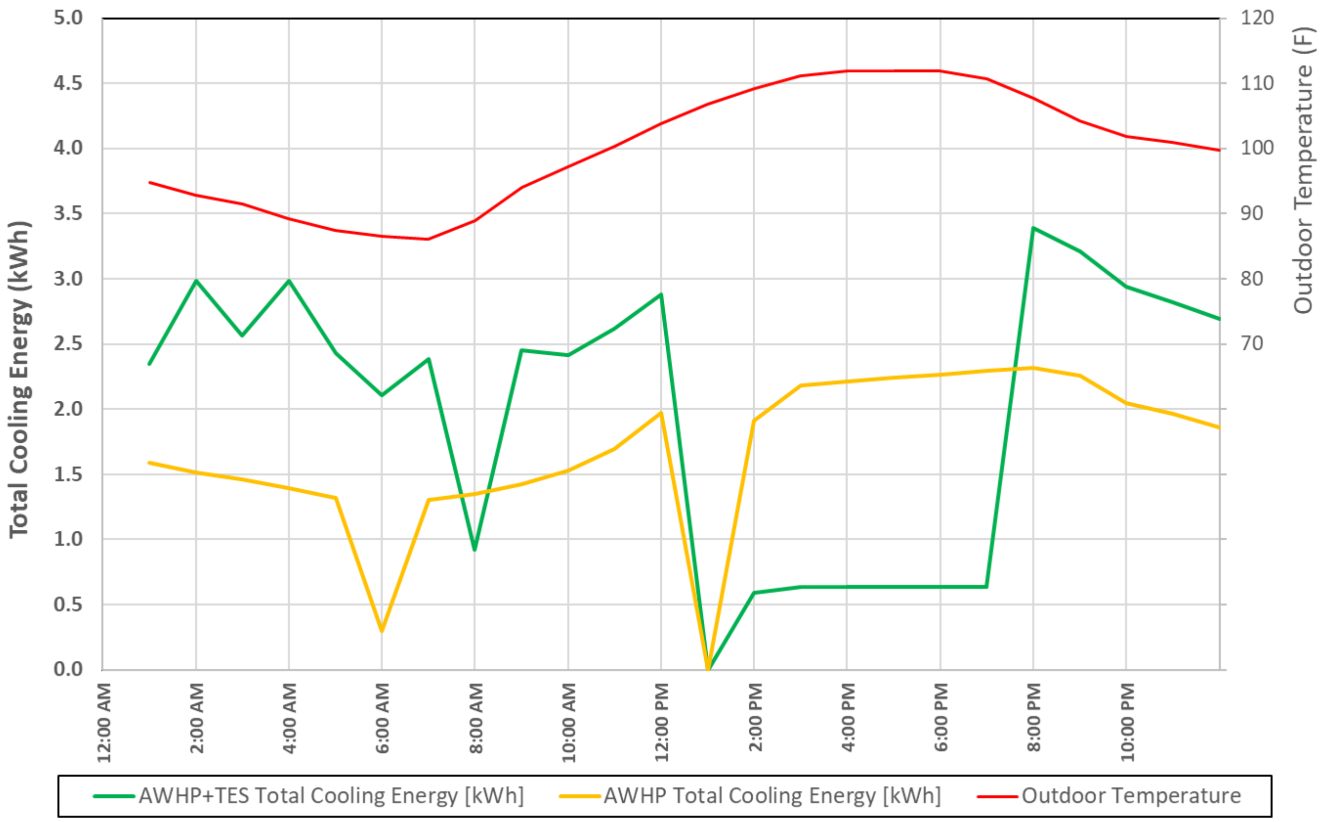


**Figure 5. Peak day AWHP and AWHP+TES hourly demand profile (Phoenix 7-hour peak)**

The impressive load-shifting impacts of the AWHP+TES cases provide benefits to electric utilities by sharply reducing on-peak usage and adding load during off-peak hours. Increasingly, electric utilities are providing TOU electric rates to incentivize customers to support the utility's load management goals. Some TOU rates are more beneficial to the customer than others as utilities are balancing fixed and marginal costs and assessing the impacts of different rate scenarios. This study developed three TOU rates to be applied to the hourly EnergyPlus kWh output data. Table 15 summarizes the summer utility cost projections for the three TOU rates for the ASHP case, the AWHP case without storage, and the case with TES. Rates 1 and 3 are the 3-

hour on-peak rates, with Rate 3 having a mid-peak shoulder period on each side of the peak. Rate 2 is the longer 7-hour peak. Although summer utility cost projections are included for the ASHP case in Table 15, ASHP utility cost projections will not be compared because the focus of this study is to compare the AWHP case without storage to the case with TES.

**Table 15. Projected Summer Utility Costs (AWHP vs. AHWP+TES)**

| TOU kWh | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| **ASHP** | | | | | |
| Rate 1 | $714 | $950 | $683 | $517 | $463 |
| Rate 2 | $406 | $546 | $417 | $293 | $267 |
| Rate 3 | $594 | $795 | $610 | $440 | $393 |
| | | | | | |
| **AWHP (no storage)** | | | | | |
| Rate 1 | $767 | $1,035 | $771 | $538 | $455 |
| Rate 2 | $434 | $615 | $445 | $304 | $263 |
| Rate 3 | $642 | $850 | $650 | $459 | $384 |
| | | | | | |
| **AWHP (with TES)** | | | | | |
| Rate 1 | $771 | $1,056 | $763 | $552 | $468 |
| Rate 2 | $327 | $478 | $360 | $280 | $253 |
| Rate 3 | $643 | $884 | $642 | $469 | $396 |

Table 16 calculates the summer utility bill savings percentage for the TES case relative to the non-storage AWHP case. The 3-hour peak period TOU rates (Rates 1 and 3) result in negligible savings for the AWHP+TES relative to the AWHP non-storage reference case. This is partly attributable to the increased energy usage of the TES load-shifting operation (Table 12) combined with the efficient building shell to support pre-cooling for this short duration peak. The longer peak duration Rate 2 shows more significant summer bill reductions, ranging from around 20% for Miami, Phoenix, and Dallas to 4%–8% for Denver and Washington, D.C. Clearly the design of the TOU rate and the duration of the peak period are two factors that impact customer economics. Utility rate design is a complicated endeavor, and the simplistic approach presented here may not be reflective of future expected TOU rates.

**Table 16. Projected Summer Utility Cost Savings (AWHP+TES vs. AHWP)**

| TOU kWh | Miami | Phoenix | Dallas | Wash D.C. | Denver |
|---|---|---|---|---|---|
| Rate 1 | 0% | -2% | 1% | -3% | -3% |
| Rate 2 | 25% | 22% | 19% | 8% | 4% |
| Rate 3 | 0% | -4% | 1% | -2% | -3% |

One aspect of the EnergyPlus modeling that warrants additional attention relates to the maintenance of indoor comfort conditions between the direct expansion vapor compression ASHP and the hydronic AWHP. Under most operating conditions, the ASHP will maintain a colder evaporator coil temperature than the hydronic fan coil, given that achieving AWHP fan coil supply water temperatures below 45°F with water as the heat transfer fluid (in an AWHP system) limits the supply water temperature to avoid freezing of the evaporator coil. Appendix A provides some details on the EnergyPlus modeling of the two system types in the five climates.

# 4 Discussion

The AWHP+TES strategy is projected to be highly effective at reducing the on-peak energy usage in all the climates evaluated in comparison to an AWHP operated under a similar pre-cooling thermostat control strategy. Combining indoor air pre-cooling prior to the peak with limited on-peak "floating" above set point can be accomplished in most climates with roughly 210 gallons of storage. Ideally the storage tank(s) could be installed within conditioned space (e.g., built into a mechanical closet between garage and indoors) to gain the thermal benefit of any tank storage heat transfer.

One aspect of the TES strategy that would need to be addressed is indoor humidity control and the need for indoor air movement, especially during the longer 7-hour peak period. Air stagnation can be addressed by ceiling fan operation. Humidity control in more humid climates could either be accomplished by supplemental dehumidification or by limited on-peak AWHP operation with lower supply water temperatures and low fan coil airflow levels.

Ideally, some of the 7-hour peak duration cases would have benefited from being simulated with limited AWHP operation to help mitigate the indoor temperature deviations above set point, as shown in Figure 4. Unfortunately, EnergyPlus is unable to model this mode of operation.

These issues would tend to slightly increase the AWHP+TES on-peak energy usage projections in some of the modeled cases. The impacts are expected to be small and should not detract from the favorable findings.

# 5 Conclusions and Recommendations

This study explored the performance and operating cost viability of AWHPs coupled with TES in efficient new residential construction. AWHPs are an emerging technology in the United States but offer promise in terms of high efficiency and a simplified installation (fully contained and factory-charged refrigeration system) and hydronic delivery capabilities, which facilitate zoning, ducts in conditioned space, and TES integration for summer load-shifting. Although the AWHP+TES strategy is not yet a mainstream HVAC strategy, the authors feel that in ten years as decarbonization efforts proceed and TOU rates become more common, strategies such as this will be more accessible.

How TOU rates evolve in the years ahead will have significant influence on the future customer benefits of this AWHP approach. As the modeling demonstrates, short 3-hour peak periods are fairly well served by controlled pre-cooling as efficient building shells can better coast through the on-peak period. Increasing summer temperatures, the greater saturation of air conditioning, and the level of renewable generation in the local grid will all play a role in influencing how future TOU rates are developed for residential customers. Incentives outside of the utility rate structure may be one approach to provide additional cost savings to advanced load-shifting technologies.

In terms of the work completed in this study, validated EnergyPlus simulation models were developed based on detailed field monitoring data. The models were then updated with IECC ZERH envelope and component requirements for climate zones 1–5 (Miami, Phoenix, Dallas, Washington, D.C., and Denver), and simulations were completed for the 1,962 $ft^2$ home in each climate zone for a minimum efficiency ASHP, an AWHP coupled with a fan coil, and an AWHP coupled with TES sized to eliminate on-peak compressor operation. Hourly simulation energy use outputs were then applied to three sample TOU rates to estimate customer utility cost impacts.

AWHP modeling projections were based on the observed field performance of the Chiltrix CX34 variable-speed unit. Other products on the market or entering the market in the near term would likely perform differently.

Key conclusions from the EnergyPlus simulation effort include:

- In comparing ASHP to AWHP summer cooling energy usage without TES, the AWHP usage was projected to be 11%–15% higher in the humid climates, 6% higher in Phoenix, but 6% lower in the dry Denver climate.
- The EnergyPlus AWHP+TES control strategy involved alternate hours of charging the TES tank (beginning in the early a.m.) and conditioning the indoor zone. In addition, several hours before the peak period began, the indoor cooling set point was reduced from 76°F to 72°F to provide additional stored energy in the house. TES tank size was iteratively evaluated to allow for virtually all the on-peak cooling to be provided by the

TES tank without compressor operation. For all but the Phoenix 7-hour duration peak case, 210 gallons of storage was adequate to meet peak loads. For the Phoenix case, 525 gallons of storage, a 15% larger capacity compressor, and a modified tank charging schedule.

- The AWHP+TES load-shifting strategy was highly effective at moving cooling energy use to non-peak hours. For the 3-hour duration peak rates studied, summer on-peak cooling energy use was reduced by 55% relative to a non-storage AWHP (with similar pre-cooling and peak set points) across all climates. For the 7-hour peak duration TOU rate, the AWHP+TES on-peak reduction averaged 66%.
- In comparing the AWHP+TES summer utility bills for Rates 1 and 3 to non-storage AWHP bills with similar cooling thermostat set points, homeowner cost savings were essentially zero. This can be largely attributed to the effectiveness of the advanced building shell assumptions coupled with pre-cooling. However, for the 7-hour peak period (Rate 2), cooling savings averaging 11% were realized. This suggests that when coupled with advanced construction methods, a short duration peak period may not be the best application for this technology. TOU rate structures and other forms of utility incentives for load-shifting may result in different conclusions.

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# Appendix A: EnergyPlus Output Summary

The combination of EnergyPlus objects used to model the AWHP and ASHP differ particularly when it comes to the indoor cooling coil objects. In BEopt, the cooling coil is packaged as part of the larger object that simulates an ASHP (Coil:Cooling:DX:SingleSpeed). However in the AWHP model, the cooling coil object (Coil:Cooling:Water) is separate from the AWHP object (HeatPump:PlantLoop:EIR:Cooling). The two objects that model the cooling coils contain different algorithms and inputs that determine how each object models latent loads. The Coil:Cooling:DX:SingleSpeed object has inputs for "Ratio of Initial Moisture Evaporation Rate and Steady State Latent Capacity" and a "Latent Capacity Time Constant," while the AWHP's Coil:Cooling:Water object has inputs for "Design Inlet Air Humidity Ratio" and "Design Outlet Air Humidity Ratio." In addition, the previously mentioned AWHP inputs were auto sized in EnergyPlus, while the inputs for the ASHP were auto sized using BEopt. The differences between the auto sizing feature and the cooling coil objects themselves led to differences in how each model responded to latent loads. The effects of the ASHP and AWHP models' distinctive latent load responses are explored in this appendix.

Table A-1 presents projected June through September EnergyPlus latent and total cooling loads for the different climates and equipment cases (ASHP and AWHP without storage). A resulting sensible heat ratio (sensible cooling load as a fraction of total cooling load) is also reported. Latent loads are much higher in the more humid climates, while the dry Phoenix and Denver climates have considerably lower seasonal latent loads. In all cases, the AWHP performs less latent cooling, ranging from 4%–73% less, depending upon climate. A reduction in latent cooling with the AWHP is expected as a warmer coil temperature with the AWHP hydronic fan coil reduces the potential for condensing moisture from the return air passing over the coil. Sensible heat ratios in the more humid climates are in the 0.70–0.80 range, while the drier climates are over 0.90. The one case that stands out is the Denver ASHP case, which is projected to have a 0.82 sensible heat ratio. The authors have reviewed the input file in detail but were unable to identify why this case would align more closely with the projected humid climate performance rather than the dry climate cases.

A final comparison was made by looking at the total June–September sensible cooling delivered. In four of the five climates, the AWHP had a slightly lower total sensible cooling load, ranging from 0.8% to 1.9% lower. In the fifth case (Denver), the AWHP sensible load was slightly higher. Although the differences are relatively small, the expectation is that the difference in sensible loads should be closer to zero. Fan efficacy for the two system types will be slightly different and will contribute to small differences in terms of the fan motor heat transferred to indoor air. The authors will communicate with the EnergyPlus help desk staff at the U.S. Department of Energy to relay these findings.

**Table A-1. Summer Cooling Load Summary**

| Case | Summer Cooling Latent MBTU | Summer Cooling Total MBTU | Calculated Sensible Heat Ratio | % Difference in Summer Cooling Sensible MBTU |
|---|---|---|---|---|
| Phoenix ASHP | 2,448 | 41,644 | 0.94 | |
| Phoenix AWHP | 1,819 | 40,288 | 0.95 | |
| % difference | 26% | 3% | | -1.9% |
| | | | | |
| Dallas ASHP | 7,155 | 32,644 | 0.78 | |
| Dallas AWHP | 6,738 | 31,681 | 0.79 | |
| % difference | 6% | 3% | | -2.1% |
| | | | | |
| Wash DC ASHP | 4,517 | 17,916 | 0.75 | |
| Wash DC AWHP | 4,167 | 17,461 | 0.76 | |
| % difference | 8% | 3% | | -0.8% |
| | | | | |
| Denver ASHP | 2,196 | 12,477 | 0.82 | |
| Denver AWHP | 597 | 10,969 | 0.95 | |
| % difference | 73% | 12% | | +0.9% |
| | | | | |
| Miami ASHP | 9,280 | 34,287 | 0.73 | |
| Miami AWHP | 8,876 | 33,666 | 0.74 | |
| % difference | 4% | 2% | | -0.9% |

Table A-2 summarizes summer indoor comfort conditions for the different cases. The average indoor temperature maintained by the ASHP and AWHP systems during the June–September summer period were within 0.2°F of each other, with all but the Phoenix case maintaining identical average indoor temperatures. Similarly, peak indoor temperatures were nearly identical, with only the Phoenix case showing a 0.5°F increase over the ASHP case. Indoor relative humidity is impacted by the latent removal characteristics of the two system types. As shown in Table A-1, the AWHP is projected to provide less moisture removal, which will result in higher indoor relative humidity (RH) levels. For the dry Phoenix and Denver climates, average indoor RH is projected to be 1.8% to 3.2% higher than the ASHP. Given the low indoor humidities in those climates, the comfort impact is likely beneficial. In more humid climates, the AWHP indoor RH is projected to be 1.7% to 1.9% higher. In the extreme Miami climate, even a small change in indoor RH may contribute to a feedback response from the occupant, such as reducing indoor cooling set point. The significance of this impact in terms of occupant response is difficult to quantify and would be best evaluated with actual field comparative studies.

**Table A-2. Summer Projected Indoor Comfort Conditions**

| Case | Summer Maximum Indoor Temp (°F) | Summer Average Indoor Temp (°F) | Summer Maximum Indoor RH (%) | Summer Average Indoor RH (%) |
|---|---|---|---|---|
| Phoenix ASHP | 77.5 | 75.1 | 51.2 | 40.5 |
| Phoenix AWHP | 78.0 | 75.3 | 54.2 | 43.7 |
| Difference | +0.5°F | +0.2°F | +3.0% | +3.2% |
| Dallas ASHP | 77.7 | 75.1 | 60.5 | 51.4 |
| Dallas AWHP | 77.7 | 75.1 | 62.3 | 53.1 |
| Difference | +0.0°F | +0.0°F | +1.8% | +1.7% |
| Wash DC ASHP | 77.7 | 74.8 | 67.6 | 53.0 |
| Wash DC AWHP | 77.7 | 74.8 | 68.1 | 54.8 |
| Difference | +0.0°F | +0.0°F | +1.5% | +1.8% |
| Denver ASHP | 77.7 | 74.6 | 51.9 | 33.4 |
| Denver AWHP | 77.7 | 74.6 | 58.0 | 41.5 |
| Difference | +0.0°F | +0.0°F | +1.5% | +1.8% |
| Miami ASHP | 77.7 | 75.1 | 61.9 | 54.7 |
| Miami AWHP | 77.7 | 75.1 | 64.2 | 56.6 |
| Difference | +0.0°F | +0.0°F | +2.3% | +1.9% |

Figures A-1 through A-15 provide three daily graphical comparisons between the ASHP and AWHP cases for each climate zone. The first graph compares each model's hourly average latent cooling load and total cooling load for the month of July. The second and third graph compare the comfort conditions for each system. The second graph shows the average hourly indoor temperature and relative humidity for the month of July, while the third graph shows these values for the hottest day of the year. The graphs comparing each system's hourly cooling loads show that both the total and latent cooling loads for the ASHP and AWHP remain very close throughout the day in humid climates such as Miami and Dallas. However, in a dry climate like Denver, the latent and total cooling loads of the ASHP are consistently higher than the AWHP's respective cooling loads. During the pre-cooling portion of the day prior to the peak period, both the latent and total cooling loads are around 3,000 BTUs lower for the AWHP. The second and third graphs for the Denver climate zone show that the result of the AWHP's lower latent load is an indoor relative humidity that is around 10%–12% higher. In the more humid climate zones, the difference between each systems relative humidity ends up being a much smaller 1%–2%.

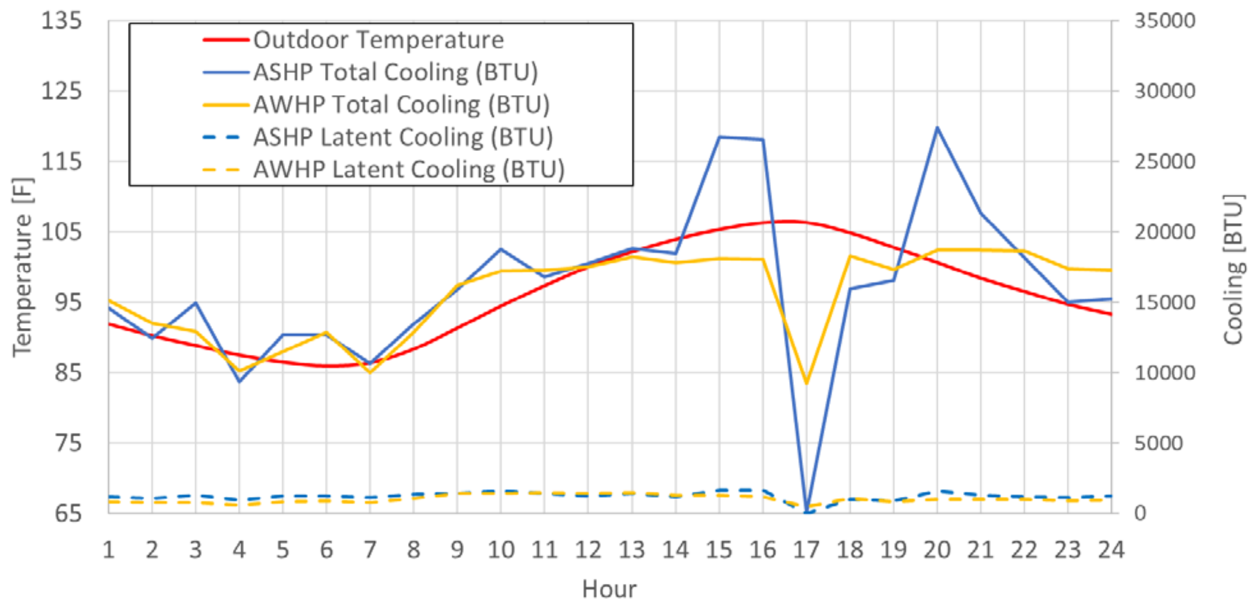


**Figure A-1. Average Phoenix load profile for July**

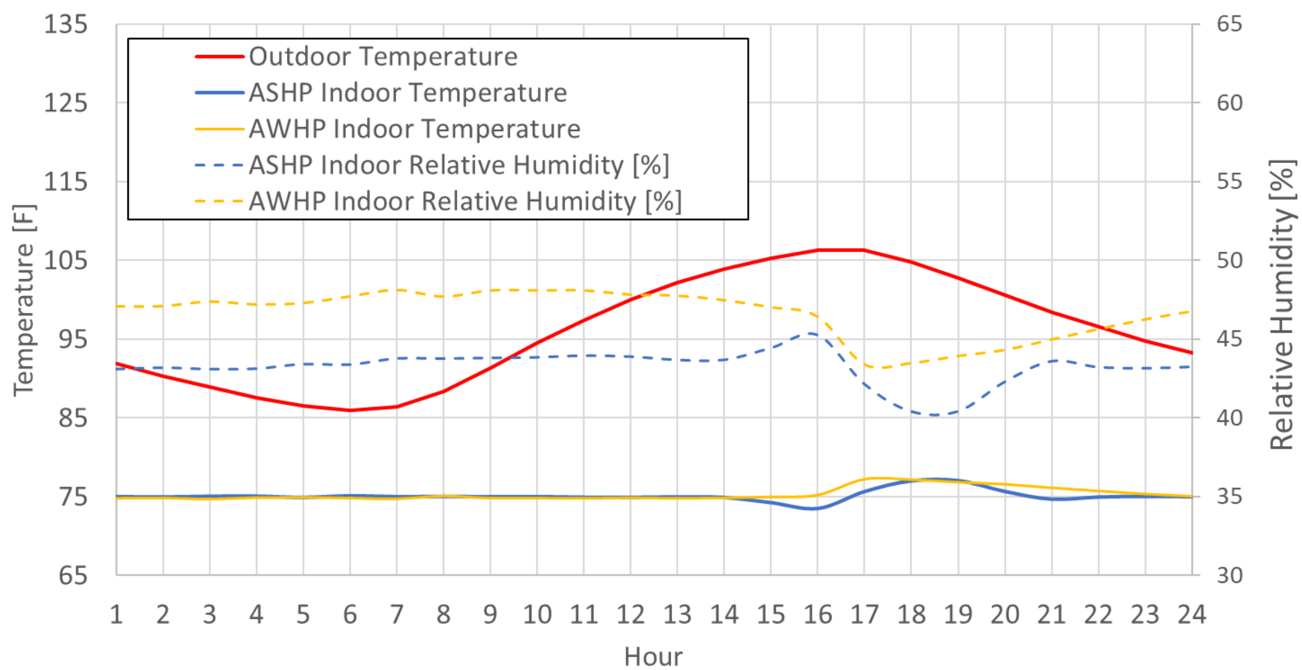


**Figure A-2. Average Phoenix temperature and RH conditions for July**

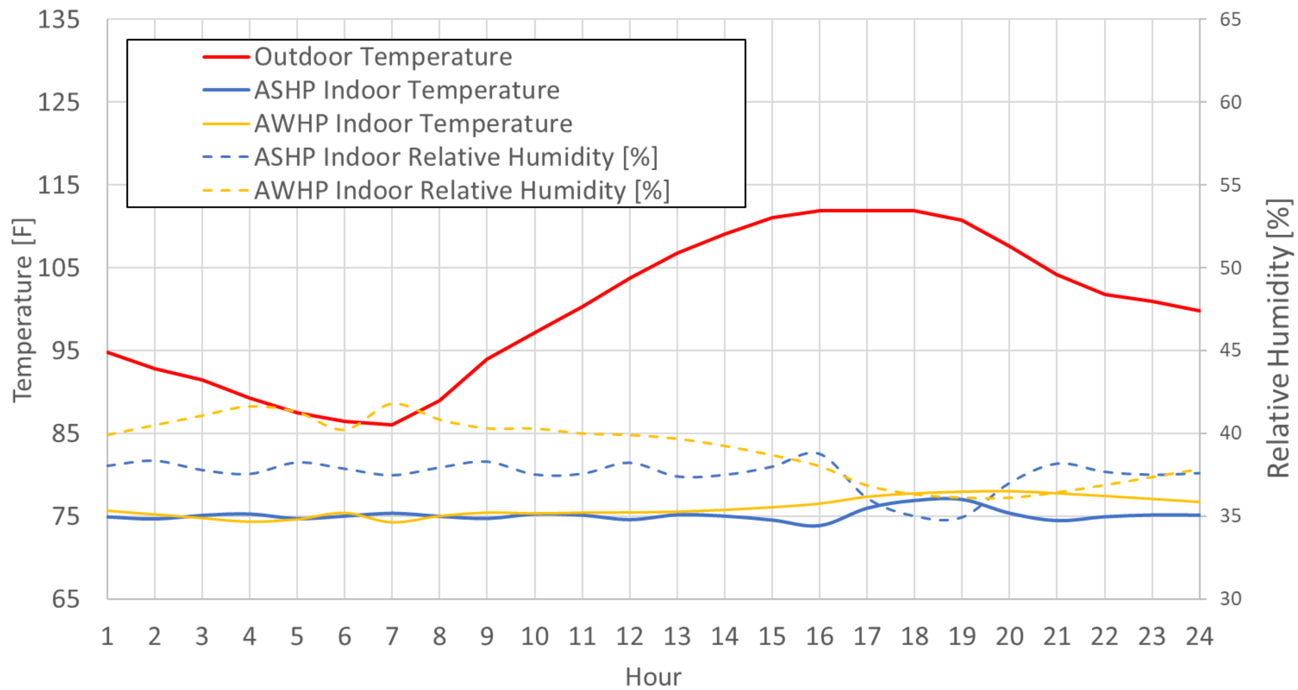


**Figure A-3. Phoenix peak day temperature and RH conditions**

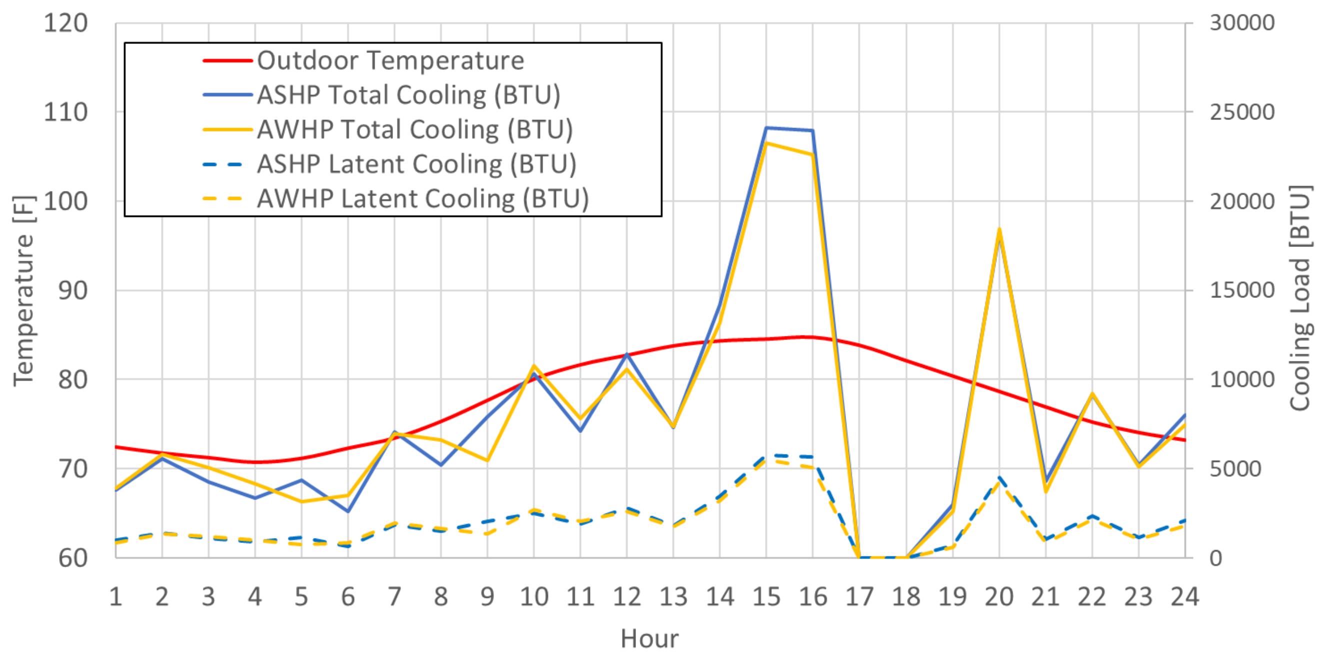


**Figure A-4. Average Washington, D.C. load profile for July**

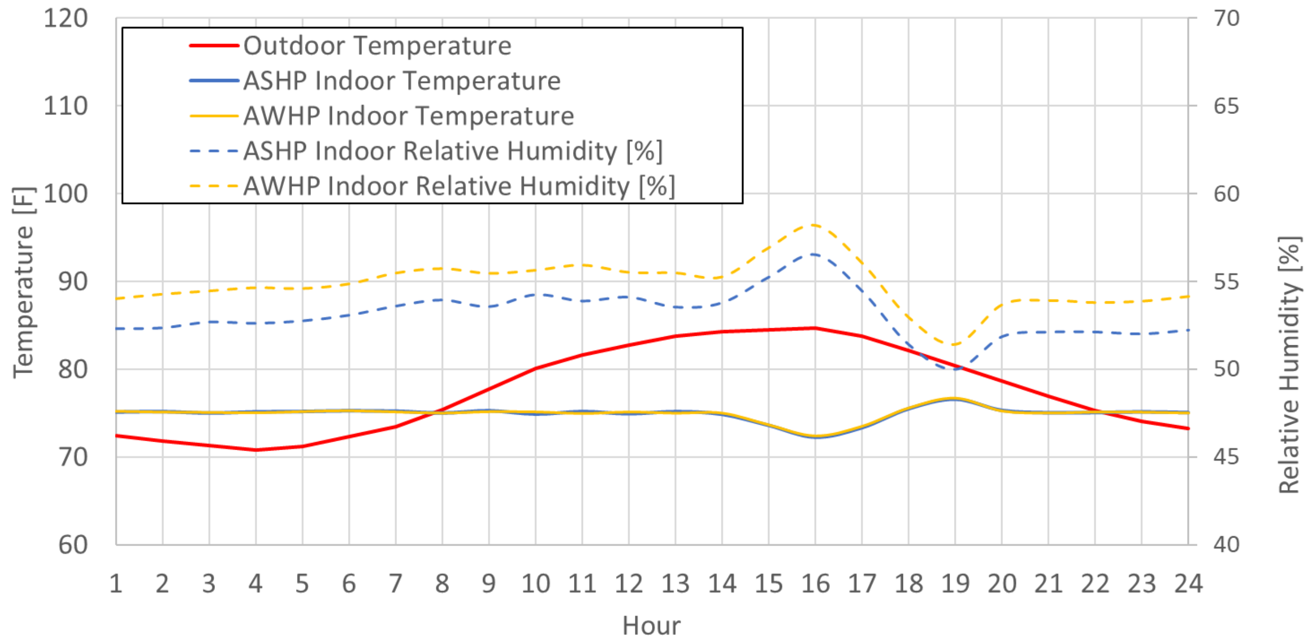


**Figure A-5. Average Washington, D.C. temperature and RH conditions for July**

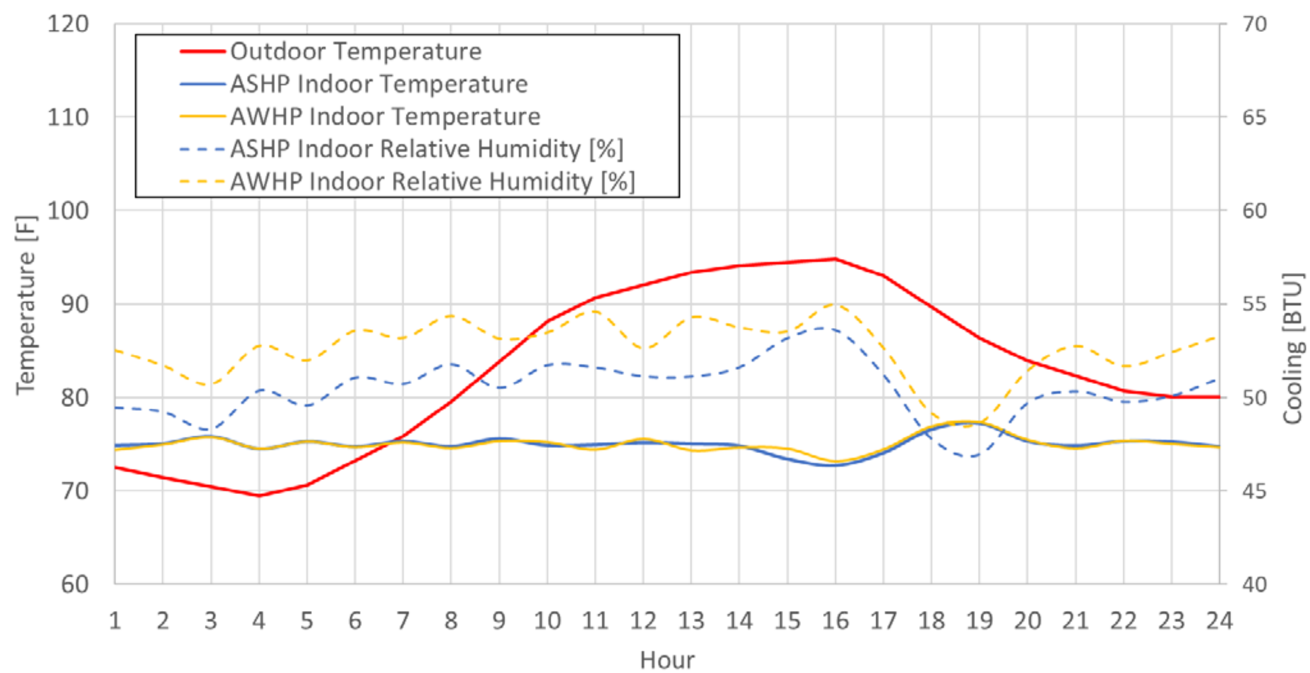


**Figure A-6. Peak day Washington, D.C. temperature and RH conditions**

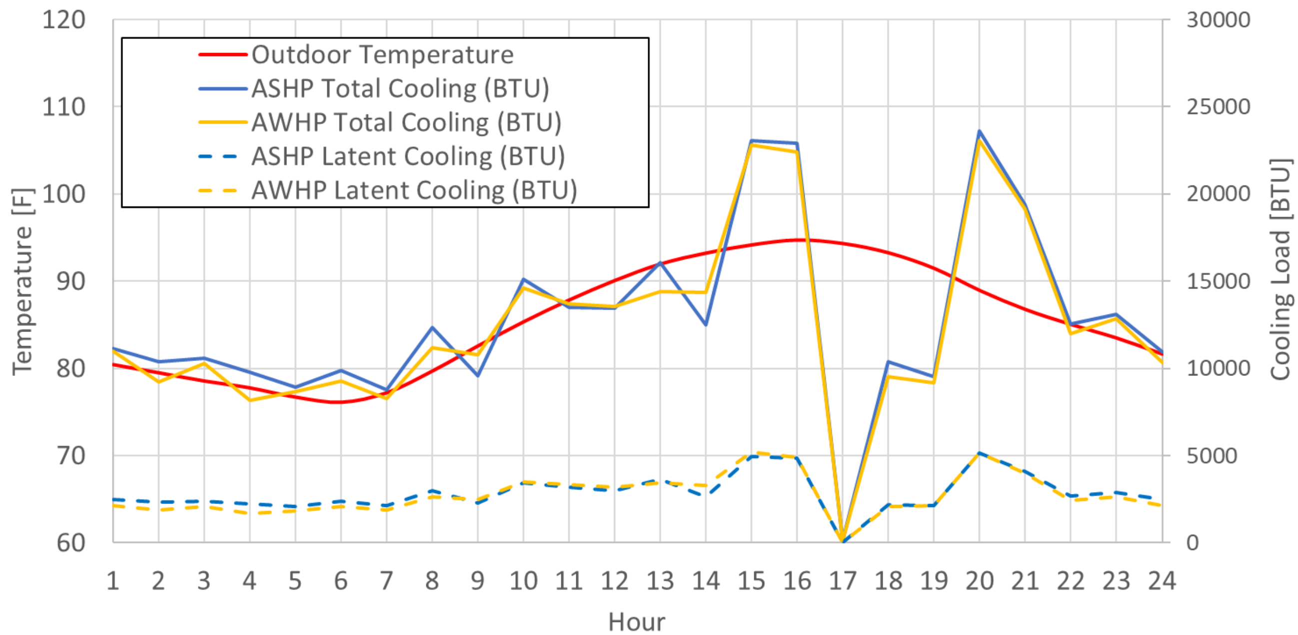


**Figure A-7. Average Dallas load profile for July**

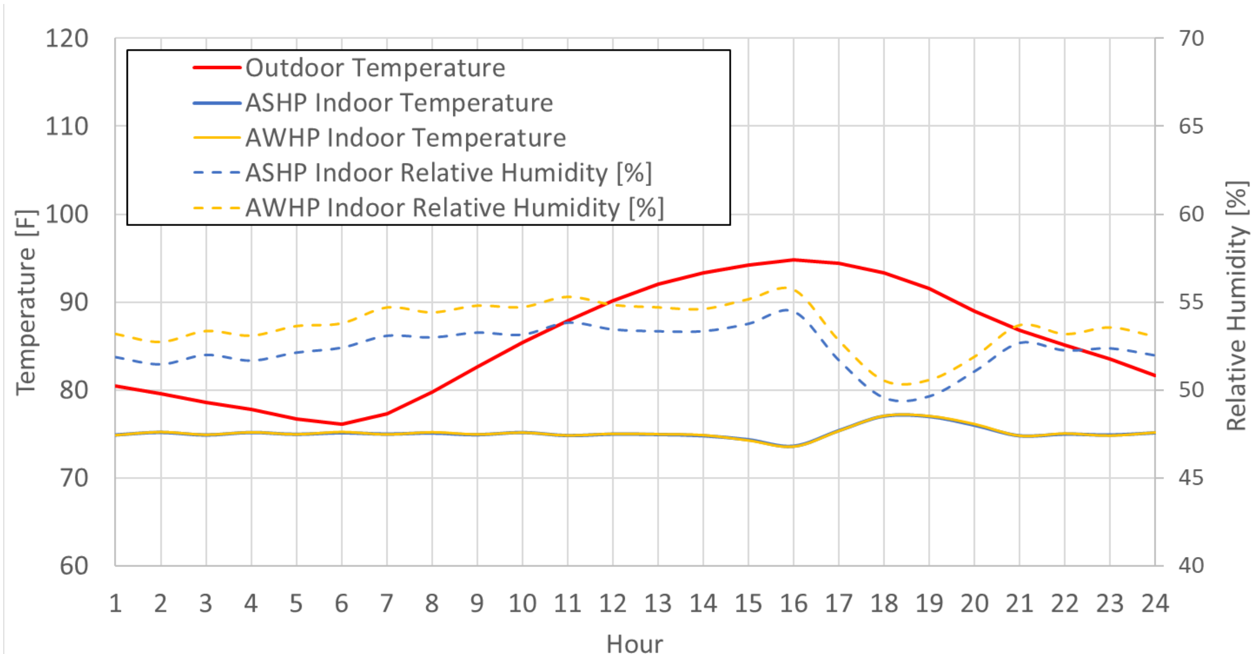


**Figure A-8. Average Dallas temperature and RH conditions for July**

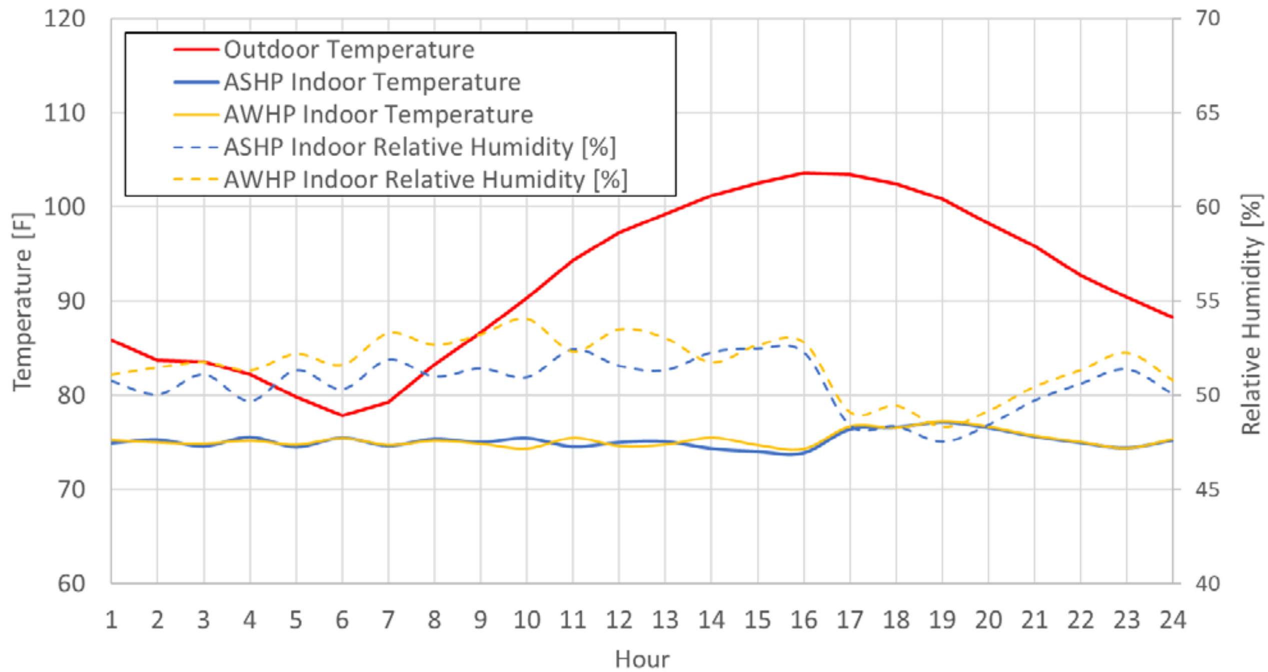


**Figure A-9. Peak day Dallas temperature and RH conditions**

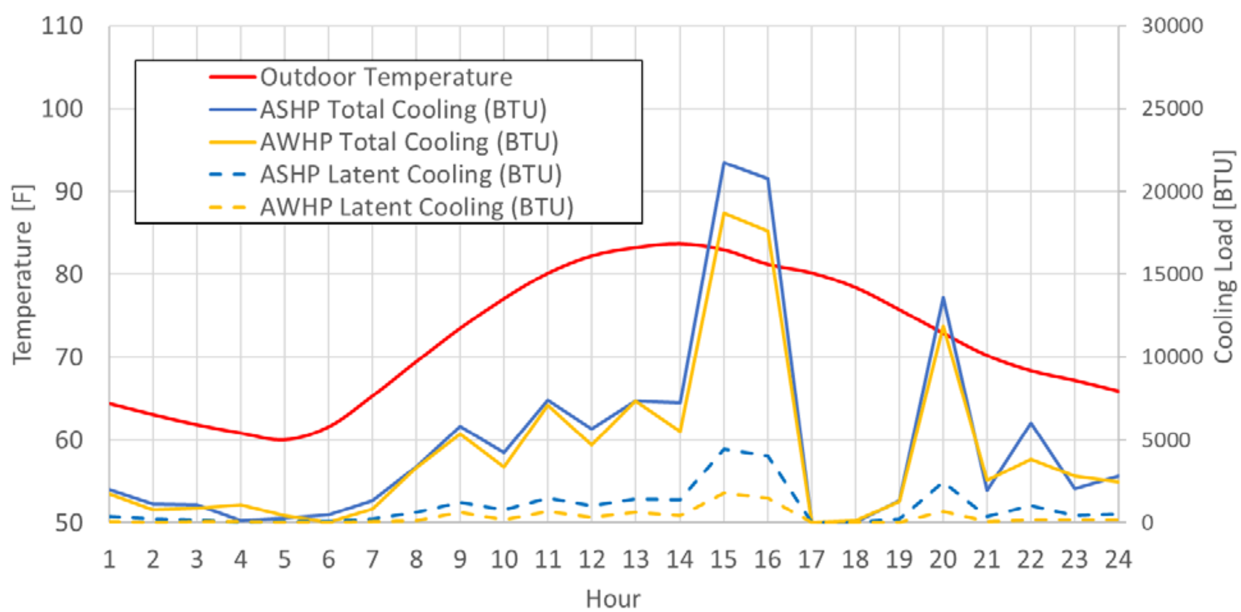


**Figure A-10. Average Denver load profile for July**

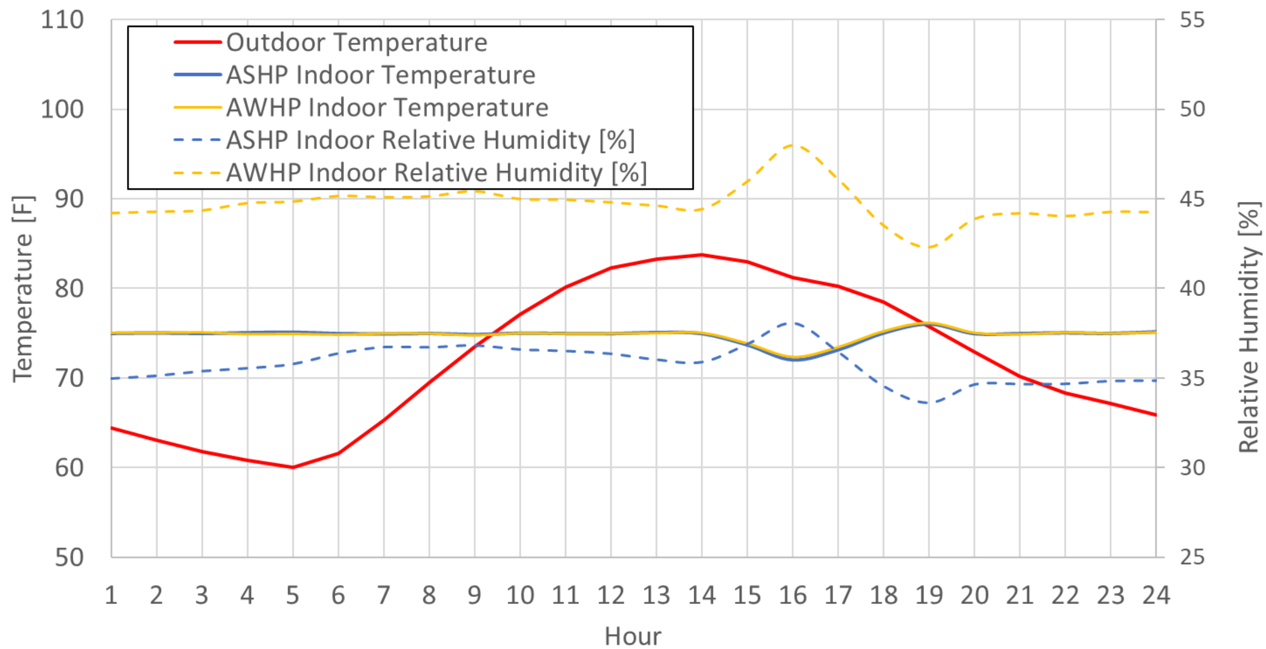


**Figure A-11. Average Denver temperature and RH conditions for July**

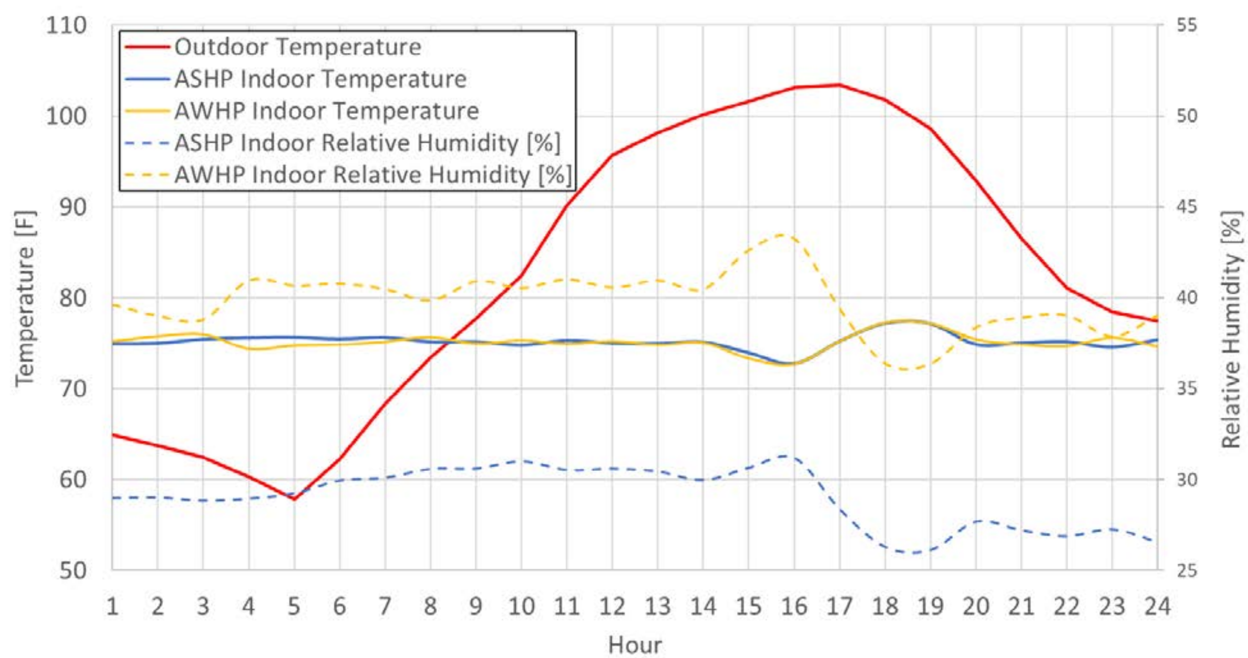


**Figure A-12. Peak day Denver temperature and RH conditions**

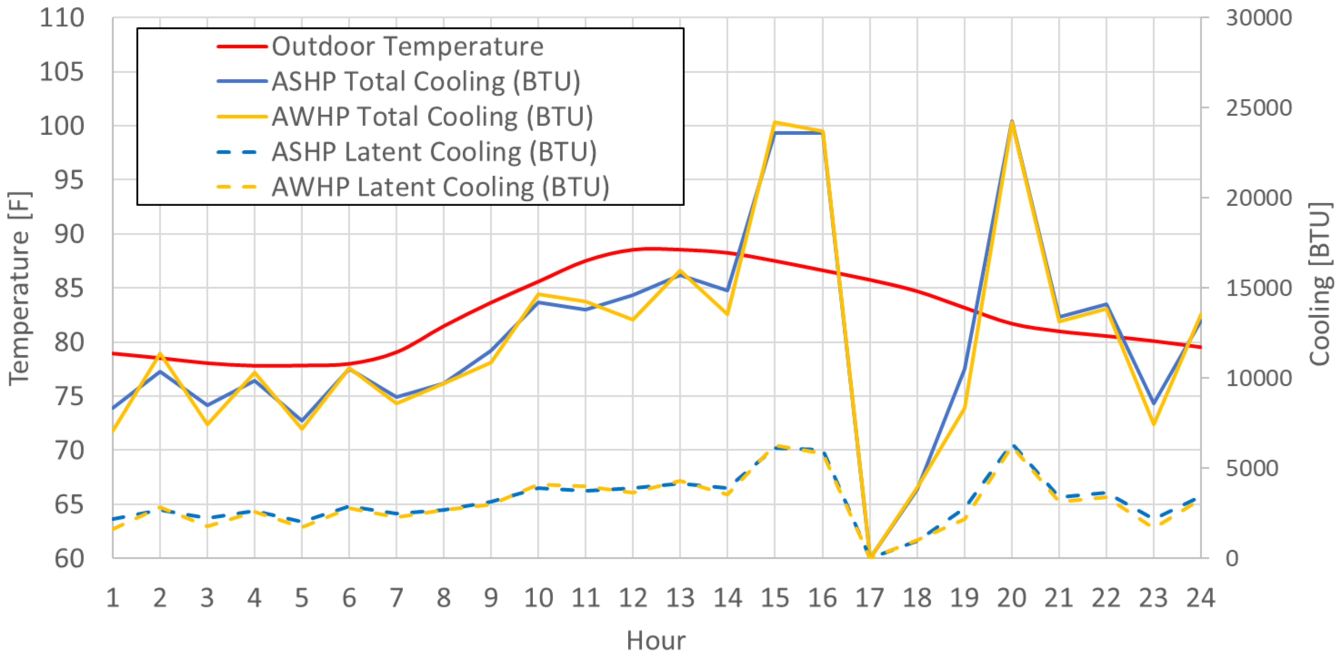


**Figure A-13. Average Miami load profile for July**

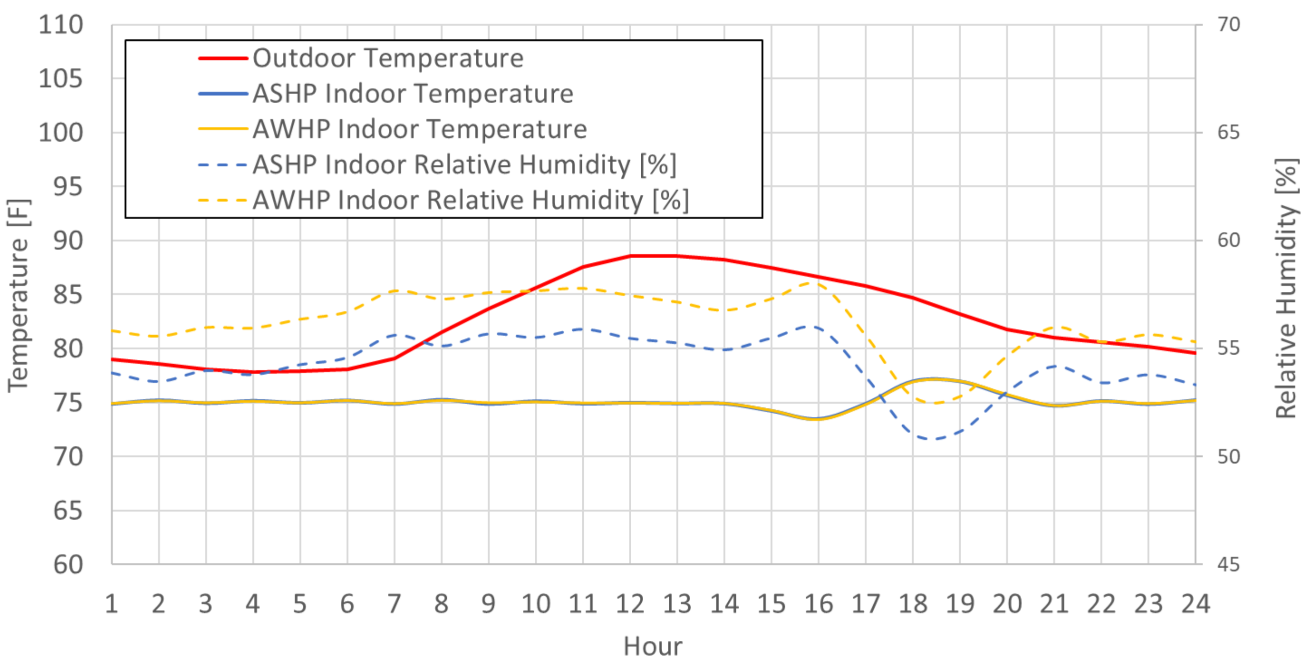


**Figure A-14. Average Miami temperature and RH conditions for July**

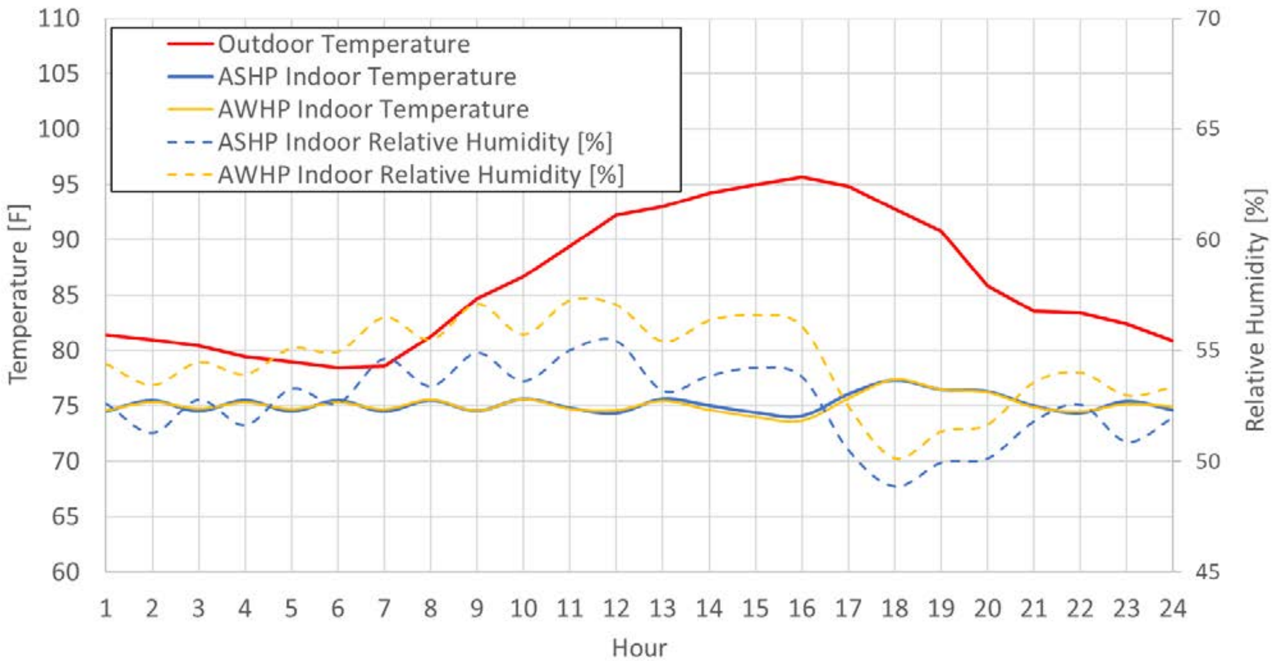


Figure A-15. Peak day Miami temperature and RH conditions

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