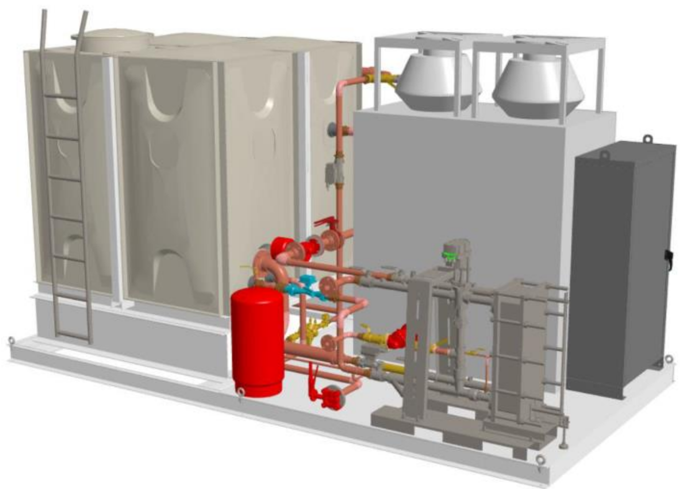
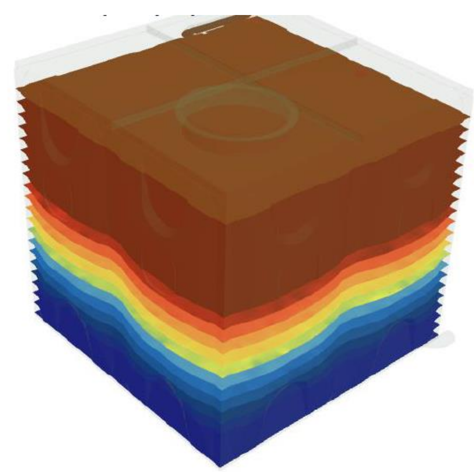
Central Heat Pump Water Heater Unpressurized Storage Design Optimization

Final Report

ET23SWE0059





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Acknowledgements

Thank you to the manufacturer for sharing the ins and outs of their novel water heating approach and for informing the modeling effort. Thank you to the expert modelers for all their heat transfer advice and diligent efforts to calibrate an accurate model, all while working around an indeterminate scenario with more independent variables than one could expect. Thank you to the various engaged stakeholders for their expert input, feedback, and interest in our work.

Executive Summary

Central heat pump water heaters for multifamily and non-residential buildings have become a high-priority technology for energy-efficiency and decarbonization programs in recent years. There are many completed and in-progress projects aimed at understanding, development, adoption, and program opportunities for this important measure. Manufacturers, researchers, end users, and program administrators are all eager to find paths towards increased cost-effectiveness and adoption of these systems. One such opportunity is to reduce the first cost of the products through simplification of the thermal energy storage volume of these systems. To that end, unpressurized storage has been proposed as a way to reduce first costs and expand market options.

Unpressurized storage volumes for central heat pump water heater systems are thermal energy storage tanks at atmospheric pressure for direct or indirect heating of the domestic hot water loop. The first generation of this design uses water as the storage medium in a modular, insulated, rectangular tank in both open- and closed-loop versions. The closed-loop option uses indirect heating of the domestic hot water distribution loop through a plate-and-frame heat exchanger, while the open-loop has direct fluid exchange with the distribution loop. Furthermore, the design uses a return-to-primary system configuration which may have energy, demand, and cost benefits over the more common swing tank configuration. However, this novel design is unproven and the dynamics of atmospheric thermal storage are not well understood. This project takes a computational fluid dynamics modeling approach to assess the effective storage capacity, sizing, and ability of the product design to meet typical building load profiles.

The team conducted computational fluid dynamics modeling of the new technology in one- and three-dimensions. Modeling confirmed the ability of the thermal storage to stratify into an effective, usable hot water volume despite concerns about the rectangular form factor and flow disruptions at the inlets and outlets. Modeling across a test matrix of varying loads, locations, and operating conditions over 24-hour average and peak design day profiles shed light on the maximum building size and loads the system can satisfy. With some load profile assumptions based on empirical data from multifamily buildings, modeling results suggest that the product with 2,000 gallons of storage volume can satisfy a building of up to 300 to 350 occupants in 167 to 195 apartments on the peak design day in Los Angeles wintertime. The system could easily supply a larger building on the average day load profile. However, the open-loop design likely will require a small electric resistance heat source to maintain the hot water supply temperature during certain low load and high demand moments. The system can meet the demand of the modeled buildings with roughly the same storage volume and heating capacity of an equivalent swing tank configuration while providing benefits unique to the unpressurized design.

The modeling results suggest that there is potential for the product to fill in domestic hot water decarbonization market gaps, reduce equipment costs, and help realize the annual 1.7 million tons of avoidable greenhouse gas emissions in the California multifamily sector. The project team is optimistic about the technology’s potential and has some recommendations for product line development and inclusion into existing program pathways. The most urgent next steps include lab and field demonstrations and development of a user-friendly, return-to-primary sizing and modeling tool that includes an unpressurized storage approach.

Abbreviations and Acronyms

Acronym	Meaning
1D	One-dimensional
3D	Three-dimensional
Btu	British thermal units
COP	Coefficient of performance
DAC	Disadvantaged community
DHW	Domestic hot water
GHG	Greenhouse gas
GPM	Gallon per minute
GWP	Global warming potential
HTR	Hard-to-reach
HX	Heat exchanger
kWh	Kilowatt-hour
PCM	Phase change material
PG&E	Pacific Gas & Electric
PSIG	Pounds per square inch – gauge
SCE	Southern California Edison
SOC	State of charge
SDG&E	San Diego Gas & Electric
sysCOP	System COP
TES	Thermal energy storage

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Introduction

Central heat pump water heaters for recirculating and distributed domestic hot water (DHW) have become a high priority in recent years for those seeking hot water energy efficiency and decarbonization solutions. Central DHW systems, found broadly in the market from multifamily buildings to offices and education, often have energy-intensive load profiles, high losses, and low-efficiency gas-fired heat sources. High-efficiency, electrified central heat pump water heater replacement systems come in a variety of flavors, each of which has its own benefits, applications, and design considerations. For instance, a central heat pump water heater could have an electric swing tank or not, use a low-global warming potential (GWP) natural refrigerant or legacy refrigerants, be return-to-primary or not, single-pass or multi-pass, custom engineered and site-built or skid-delivered in a factory-built package. Any of these choices can be successful and reliable in both retrofit and new construction situations for reducing emissions and improving efficiency of central DHW loads.

These central heat pump water heater systems can provide electrified, efficient hot water for both new and existing buildings while minimizing grid impact and energy costs with load shifting capabilities. However, additional production options and design solutions are necessary to overcome remaining barriers to widespread market adoption. Hot water industry stakeholders, decarbonization programs, and researchers are looking for ways to optimize central heat pump water heater offerings and encourage product development towards a range of efficient, cost-effective options for any application. To that end, this study was initiated to explore the abilities of a novel storage tank design that could have benefits over incumbent products and address some adoption barriers. This product line and production is in development by a California manufacturer and is moving toward applications testing and field demonstration in the next year.

Background

Thermal energy storage (TES) volume is historically one of the most expensive components in any type of central heat pump water heater system design. Current central heat pump water heater products store the hot, potable water directly in pressurized storage tanks rated to American Society of Mechanical Engineers (ASME) standards. Despite being the standard approach, this has several inherent issues: They are expensive, use building space poorly due to their cylindrical shape and necessary plumbing, are less than ideal for maintaining thermal stratification, and are size limited by access corridors in existing buildings. Additionally, the common central heat pump water heater products with pressurized storage cannot accommodate freeze protection chemicals and can pose challenges in high-rise buildings with high water pressure when they are configured to hold potable water that directly passes through the heat pump water heaters.

Pressurized TES tanks in central heat pump water heater systems need to be relatively large to hold adequate hot water to supply peak demand events and even larger if the system is designed to provide grid services (i.e. load shifting). The TES is often designed to be stratified with a thermocline to maximize efficiency and load shift potential with single-pass designs. In some configurations, temperature maintenance recirculation systems and design of the piping into and out of the storage tanks can degrade thermal stratification and inefficiently use the entire volume, thus reducing effective storage volume and requiring a larger number of pressurized tanks. The temperature sensor penetrations necessary for monitoring and control are also expensive in a pressure-rated cylindrical tank. These tanks are heavy, round, and come in limited aspect ratios so space utilization is typically poor. Furthermore, in retrofit applications the tanks must be sized to be able to get into the building and through any mechanical room doors, hallways, elevators, or stairwells.

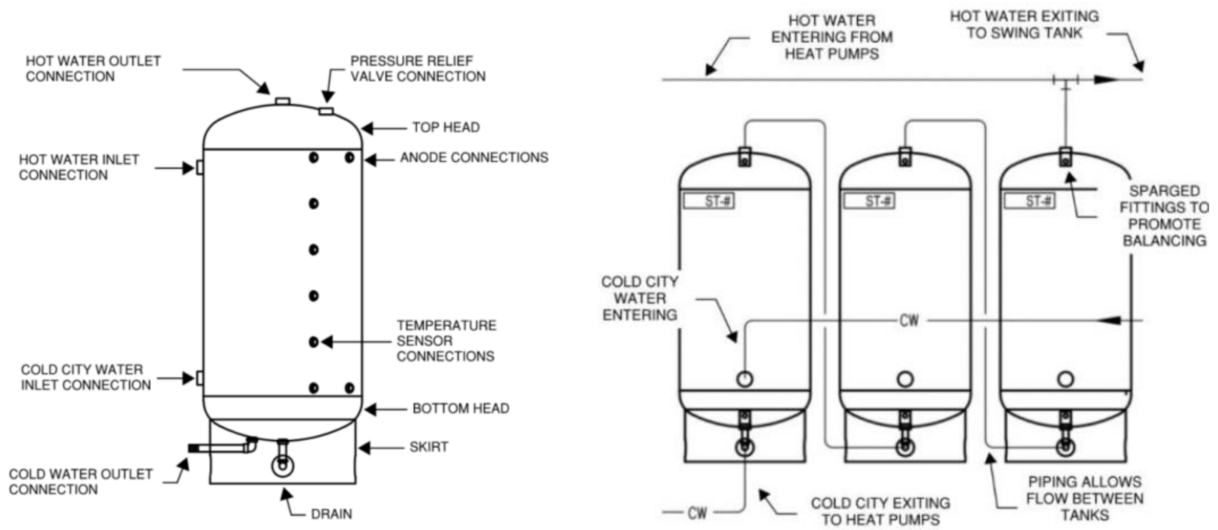

Figure 1: Typical pressurized tank and example series arrangement (Spielman, Banks and Kintner, Thermal Storage Performance in Heat Pump Water Heating Systems 2022).

Central heat pump water heater systems with unpressurized (i.e. "atmospheric") storage could mitigate these issues. An unpressurized tank design would have one or more stratified atmospheric tanks that have a heat source loop with one or more heat pump water heaters and a load side loop for the distribution system. These loops could have heat exchangers (HX) between the tank and the source or load or not. Figure 2 shows a theoretical version of this tank with venting to the atmosphere and heat exchangers between the tank and the heat pump water heater loop and between the tank and the DHW load side (Spielman, Banks and Kintner 2022). Figure 3 shows an alternate design with helical coil heat exchangers within the tank itself instead of externally. While the differences between various heat exchange options were not studied for this effort, future laboratory studies could test different heat exchange options to assess their impacts on thermal performance, controllability, stratification, effective storage capacity, and capacity to meet load profiles with high variability. For instance, an external heat exchanger may be more difficult to control and have an operating range that can meet an unpredictable load profile with low and high instantaneous loads. On the other hand, internal helical coils could affect the stratification of the storage volume by providing a conductive heat transfer path vertically through the tank.

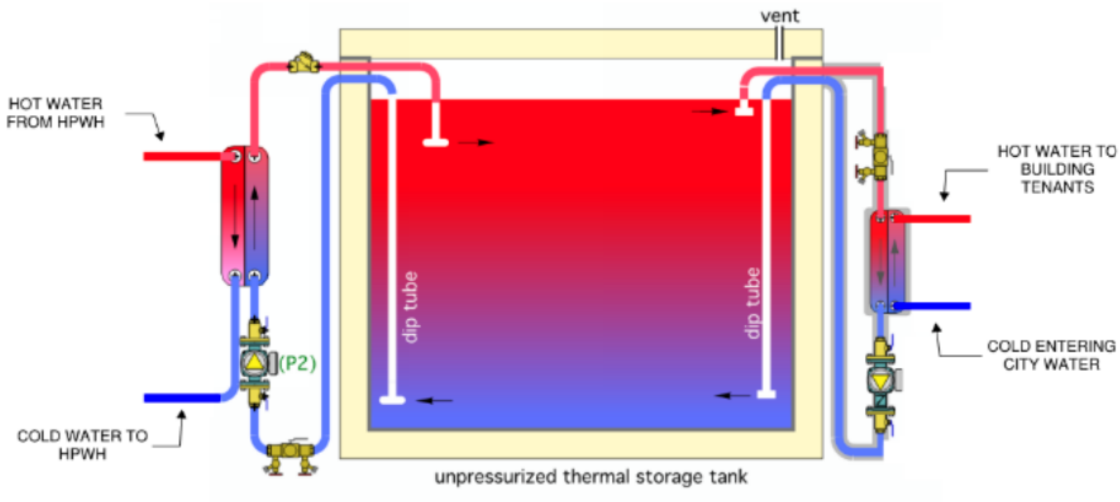

Figure 2: Example unpressurized storage tank with heat exchangers on both heat pump water heater and load side — not representative of modeled system (Spielman, Banks and Kintner, Thermal Storage Performance in Heat Pump Water Heating Systems 2022).

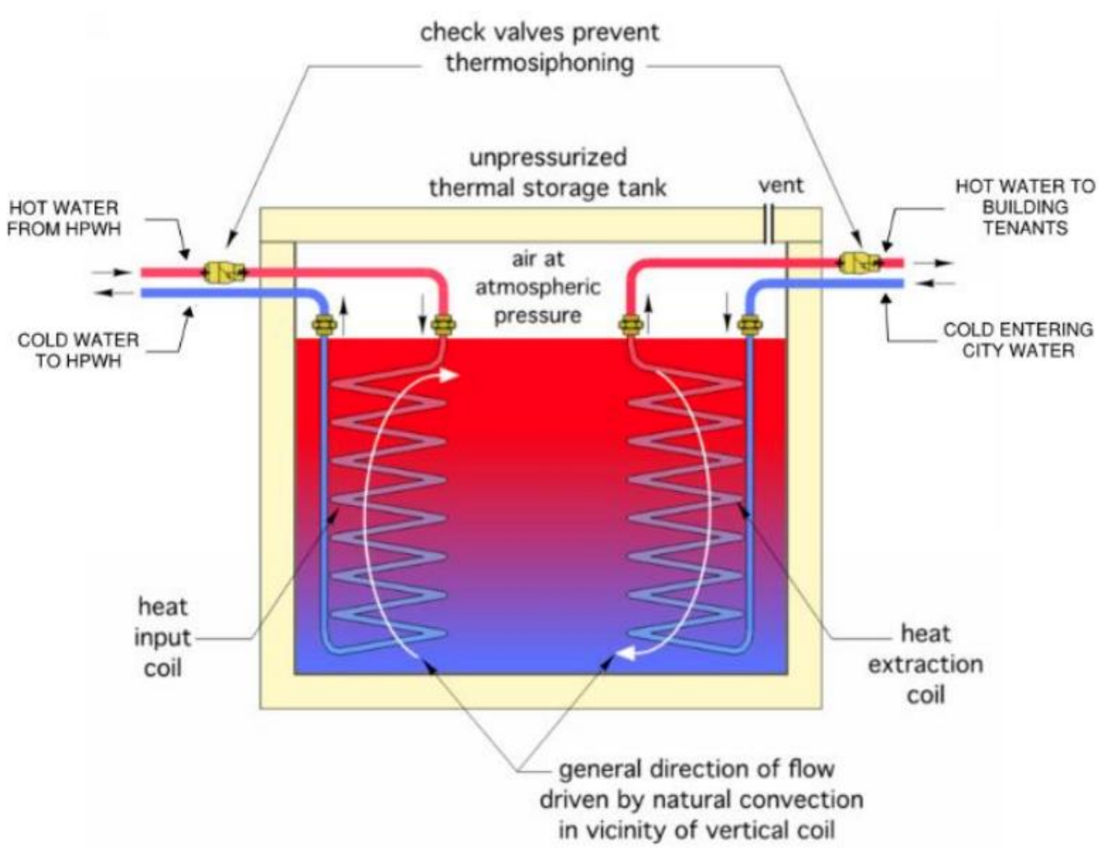

Figure 3: Example unpressurized storage tank with helical coil heat exchangers within the tank — not representative of modeled system (Spielman, Banks and Kintner, Thermal Storage Performance in Heat Pump Water Heating Systems 2022).

This totally isolated version would essentially act as a stratified liquid heat exchanger between the heat pump water heater and DHW loops. This isolation through heat exchangers could be used if a glycol-water solution is necessary in the heat pump water heater loop or if there are *Legionella* concerns at the bottom of the tank. However, neither heat exchanger is strictly necessary and are merely design options; future unpressurized storage products are likely to come in different variations with or without heat exchangers for the paired primary (heat pump water heater) and secondary (DHW) water loops.

Unpressurized tanks are not new technology and have been used for water and thermal storage in a wide range of systems for many years, but very rarely in DHW applications (e.g., solar thermal hot water). The manufacturer of the product under study has a long history of delivering unpressurized tanks with various penetrations and options that seamlessly integrate with the heat exchangers necessary for packaging with a central heat pump water heater. Some technical questions that need to be answered include: Determining the best methods for moving the heat into and out of the thermal storage tanks, sizing, assigning best use cases, and evaluating performance to optimize product offerings. The forthcoming products are anticipated to come in 1,000- or 2,000-gallon skid-mounted options paired with one or two heat pump water heaters. Two alternate product designs are closed- or open-loop approaches with single-pass, return-to-primary heat pump water heater configurations.

Using a package of readily available components, the unpressurized system must balance the following goals:

- Optimal efficiency of the central heat pump water heater system overall (including temperature maintenance).
- Simplest overall system possible with minimal components, maintenance, and control points.
- Adequate heat transfer to the potable water to serve the high instantaneous hot water demand seen in commercial and multifamily buildings.
- Effective stratification and thermocline maintenance.
- Integrated controls for heat pump, mixing valve, temperature maintenance, and load shifting.

The project team estimates that unpressurized storage volume is 50 to 75 percent less expensive than pressurized storage volume per gallon (Spielman, Banks and Kintner, Thermal Storage Performance in Heat Pump Water Heating Systems 2022). Unpressurized storage can be assembled onsite or in packaged skids with maximum utilization of available space and can easily isolate the heat pumps and DHW loops from each other. This isolation can allow for the use of freeze protection chemicals that prevent catastrophic heat pump failure and avoid challenges from the high water pressures often seen in mid- and high-rise buildings. These benefits will allow for wider adoption of central heat pump water heater systems at a lower cost, higher reliability, and reduced plumbing complexity.

The technology has many potential benefits (Spielman 2022), including:

- Less expensive, including with lower-cost temperature probe taps.
- No ASME pressure certification required.
- Can be assembled on-site making access into mechanical rooms easy.
- Can be built into the corner of a room or on a packaged skid requiring much less space than round tanks.
- Can be built with higher levels of insulation relatively inexpensively.
- Can include freeze protection chemicals.
- No expansion tank required.
- Protects the heat pumps from high pressures present in high-rise buildings.

Unpressurized storage central heat pump water heater systems would expand the available market by enabling installs at otherwise inaccessible buildings and could reduce costs of implementation. From this, utility programs involved with central heat pump water heaters would see larger impact and savings. A higher volume of installations could be possible with the same overall amount of funding and the potential market size would increase. Additionally, the unpressurized storage approach may allow for simplified, more effective load shifting with larger available water storage volumes. This innovation would be directly applicable to expanding PG&E's WatterSaver program to the multifamily and commercial building sector, for instance.

Unpressurized storage can enable central heat pump water heater installations at disadvantaged community (DAC) multifamily housing by reducing installation costs for building owners who are already cash limited. The installation of central heat pump water heaters in multifamily housing would bring energy efficiency and decarbonization to hard-to-reach (HTR) renters. Furthermore, the enhanced load shifting capabilities of the innovative storage design could help ensure that DAC and HTR ratepayers are not negatively impacted by the switch from natural gas to electricity for water heating, as encouraged by ongoing electrification efforts.

Eventually, unpressurized storage may be appropriate for including in Title 24 sections for central heat pump water heater systems and central heat pump water heater workpapers. Modeling, testing, and demonstrations are the necessary first steps towards that eventual goal.

Objectives

The main objective of this study is an assessment of the proposed unpressurized storage system designs through the modeling of heat transfer, controls, and fluid dynamics:

- Characterize the performance of the most likely first-generation design of unpressurized storage central heat pump water heater systems.

- Model the unpressurized storage system in computational fluid dynamics software to assess its ability to sufficiently stratify and serve a representative building load profile.
- Confirm the effectiveness of an unpressurized storage, return-to-primary configuration to meet building loads.
- Predict the size of multifamily building that the proposed product can serve based on peak design load profiles.
- Develop recommendations for manufacturer product development and program pathways.
- Disseminate of findings to the public through a final report and possible conference or event presentations.

The original project scope also included the stretch goal of studying phase change material (PCM) additions to the tank as a means of increasing heat capacity, heat transfer, and enforce stratification. However, the complexity of the primary modeling of the technology without PCMs was complicated and time-consuming enough that the team could not meet this stretch goal. Further, the manufacturer is pursuing a separate PCM product design that does not include water as a thermal storage medium — a divergence from the original PCM scope targeted by the project team.

Methodology and Approach

To achieve the objectives, the project team has progressed through several successive tasks:

1. Defined an unpressurized storage central heat pump water heater system design for study.
2. Defined boundary conditions for storage control volume.
3. Conducted validated three-dimensional (3D) modeling of the storage control volume.
4. Used 3D modeling to calibrate a one-dimensional (1D) model for further analysis.
5. Developed 24-hour load profiles and operating conditions for 1D modeling tests.
6. Ran 1D models over 24-hour load profiles to assess system effectiveness and performance.
7. Estimated costs of equivalent unpressurized and pressurized storage central heat pump water heater systems.
8. Recommended actions for future product development, programs, and research.

Unpressurized Storage Central Heat Pump Water Heater System Design

The team defined the geometries and system specifications for model development using the first-generation design of a California manufacturer of unpressurized storage central heat pump water heater systems. The product is a factory-built packaged unit delivered on a skid. It comprises a low-GWP R513A heat pump water heater, a rectangular unpressurized hot water storage tank fiberglass reinforced plastic walls, plumbing connections to the storage tank, primary loop pumping, control valves, and onboard control hardware. A closed-loop option includes a plate-and-frame heat exchanger between the primary and secondary DHW loops, while an open-loop option directly draws from and returns to the tank on the load side. The product comes in two sizes: a 1,000-gallon tank paired with a single heat pump water heater and a 2,000-gallon tank paired with two heat pump water heaters. Figure 4 shows a computer aided design rendering of the packaged system.

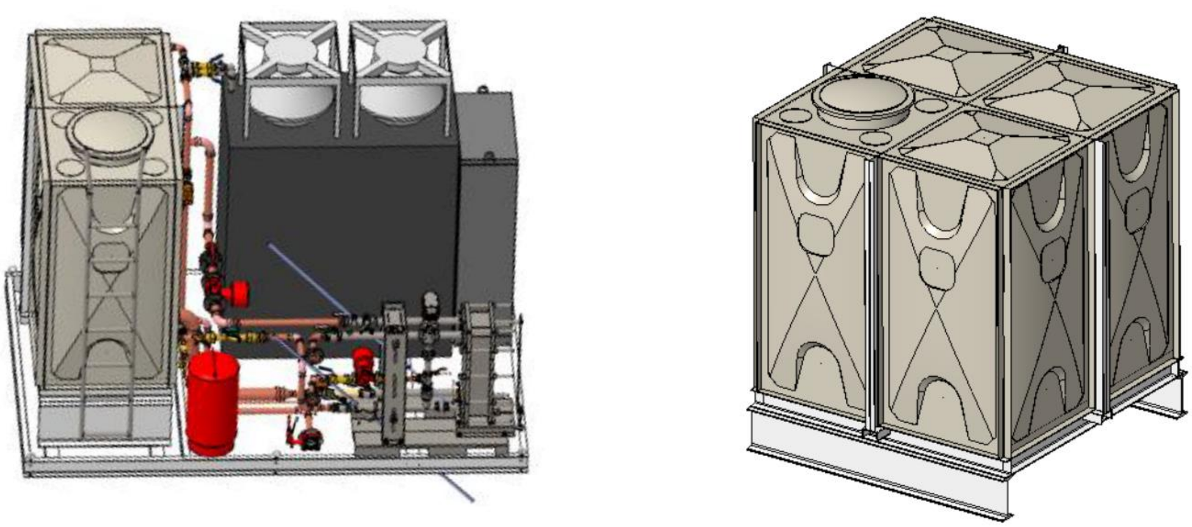

Figure 4: Packaged central heat pump water heater with 1,000 gallons of unpressurized storage and a larger, 2,000-gallon tank.

In the closed-loop option there are three plumbing loops, as shown in Figure 5:

- The heat pump water heater loop, which draws water from the bottom of the tank, heats it in a single pass and returns it to the top of the tank.
- The primary loop, which circulates hot water from the top of the storage tank to the heat exchanger and returns it to the bottom of the tank.
- The secondary DHW loop, which pumps mixed recirculation and cold make-up water through the heat exchanger, mixes this heated water with return recirculating water, and pumps the water at the distribution setpoint throughout the building to satisfy the various end-use loads.

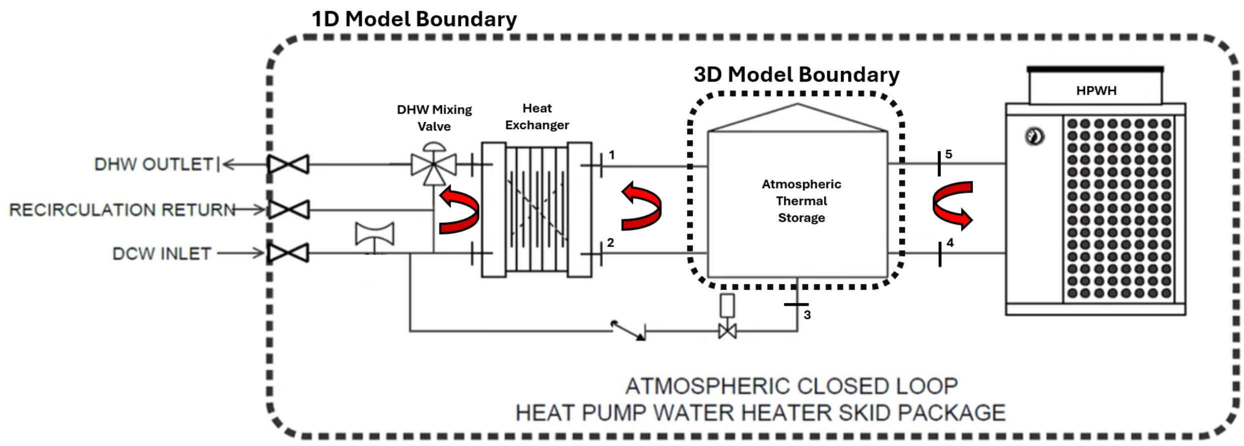

Figure 5: Plumbing diagram of closed-loop heat pump water heater package.

The plate-and-frame heat exchanger has a design point as shown in Table 1 with listed gallons per minute (GPM) and operating pressure per square inch gauge (PSIG).

Table 1: Heat Exchanger Design Point Specifications

	DHW Side	Heat Pump Water Heater Side
Fluid medium	Domestic water	Storage hot water
Flow rate (GPM)	60	56
Inlet temp (°F)	50	140
Outlet temp (°F)	120	65
Operating pressure (PSIG)	150 (Max)	150 (Max)
Pressure drop (PSIG)	9	9
Design temp (°F)	210	220
Effectiveness	98%	98%

In the open-loop design there are two plumbing loops with no heat exchanger as seen in Figure 6:

- The heat pump water heater loop that draws water from the bottom of the tank, heats it in a single pass, and returns it to the top of the tank.
- The primary loop circulates hot water from the top of the storage tank to a mixing valve (cold water make-up, recirculation water, and hot water) and then pumps it to the building's distribution system to satisfy the various end-use loads.
- A booster pump is required to overcome the added pressure drop of the storage volume.

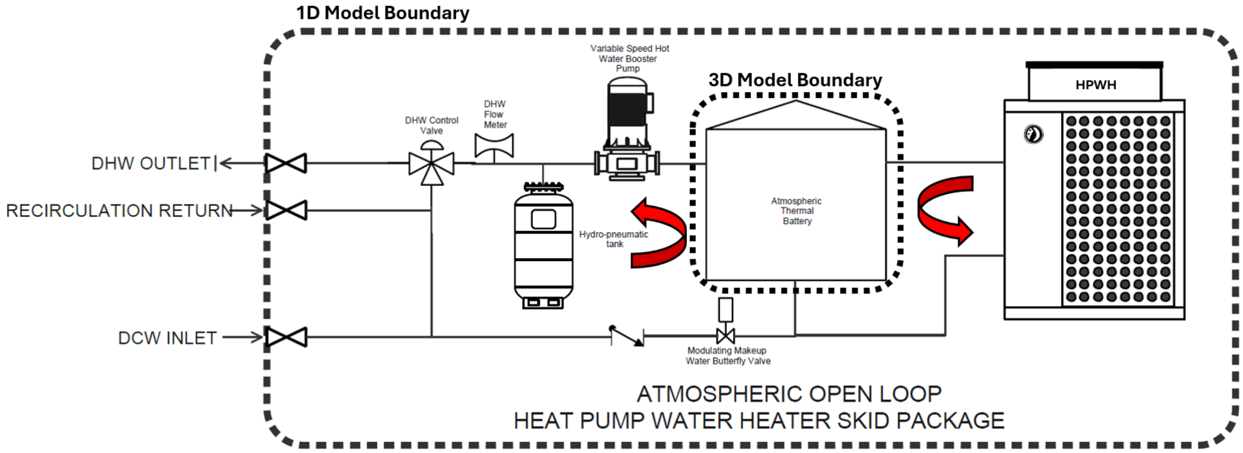

Figure 6: Plumbing diagram of open-loop heat pump water heater package.

Certain boundary conditions, system operating conditions, and limits had to be defined for the modeling phase. These conditions were established based on engineering decisions, product operating principles, equipment specifications, and assumed building and ambient conditions. The tank walls are fiberglass reinforced plastic modeled with R-6 insulation, thermal conductivity of 0.04167 W/m-°K, density of 2,000 kg/m^3, and a specific heat of 1,000 J/kg-°K. In general, 3D modeling only required boundary conditions for the storage tank control volume while 1D modeling required those plus operating conditions at various points in the system (heat exchanger flows and load profiles, for example). Table 2 lists these boundary conditions.

Table 2: Boundary and System Operating Conditions

Boundary Point	Limits	Definition	Notes
1,2 – HX primary side flow (GPM)	Max of 90 GPM	Determined by modeling	Model includes control scheme that sufficiently supplies heat to load side at minimum flow on primary side
1 – HX primary supply temp (°F)	Max limited by heat pump water heater operating limits (up to 150)	Determined from modeling, drawn from top of tank	Should typically be equal to heat pump water heater outlet setpoint but could decrease as storage depletes
HX primary return temp (°F)	n/a	Determined by modeling	N/A
City make-up water temp (°F)	[50, 70]	Defined by project team based on engineering judgement towards conservative cases. Selected 55 and 60°F for test model runs	Constant
Heat pump water heater flow (GPM)	n/a	For 140°F and 150°F setpoints, respectively: $\#HPWHs * (-2.36198 + 0.0658 * HeatPumpReturnTemp + 0.076678 * AmbientTemp)$ $0.966 * \#HPWHs * (-2.36198 + 0.0658 * HeatPumpReturnTemp + 0.076678 * AmbientTemp)$	Regression based on heat pump water heater specs. One heat pump water heater for the 1,000-gallon model, two heat pump water heaters for the 2,000-gallon model
Heat pump water heater supply temp (°F)	150	Heat pump water heater specs, temperature of water coming from heat pump water heater to top of tank	Heat pump water heater setpoints were tested at 140 and 150°F
Heat pump water heater return temp (°F)	Max of 115	As model predicts, drawn from bottom of tank, with upper bound of 115°F	To maintain necessary HP efficiency and lift, if heat pump water heater return temp location reaches 115°F, HP flow should stop
Ambient temp (°F)	[30, 100]	Ambient temps over a 24-hour profile were derived for a summer and winter day for Los Angeles and Sacramento as test cases, shown in Figure 33	Based on CZ2022 standard weather data (CALMAC 2022)

Table 3 lists additional system points that the project team defined for the modeling runs.

Table 3: Additional System Point Conditions

Boundary Point	Limits	Definition	Notes
HX load side return temp (°F)	[mixed temp,115]	DHW load side inlet to HX. Dependent on amount of city water draw to make-up for end-use loads and load profile. Maximum of 115°F representing no load, recirc only. Otherwise defined by mixed temperature after city water makeup	n/a
HX load-side supply temp (°F)	125	Design point delivery temperature to DHW load side circulation	Necessary load design point (DHW setpoint)
Recirculation flow (GPM)	n/a	$(3.41 * Recirc\ losses \frac{watt}{apt} * \#apts)/500/RecircDeltaT$	Defined for assumed recirc losses per apartment (up to 150 watt/apt)
Recirc return temp (°F)	115	System controlled to 115°F recirculation return temp	Assumed design point
Heat pump water heater efficiency (COP)	n/a	$-0.000071 * AmbientTemp^2 + 0.02907 * AmbientTemp + 1.7255$	Regression to ambient temp from heat pump water heater specs

3D Modeling of the Storage Tank Design

These system and boundary conditions were used to develop a 3D model in computational fluid dynamics software. The 3D rendering of the closed-loop plumbing, heat pump water heaters, 2,000-gallon storage volume, and heat exchangers is shown in Figure 7.

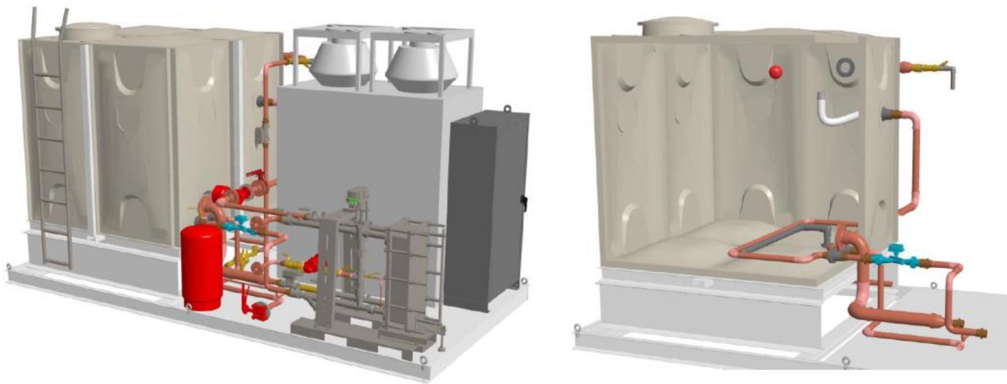

Figure 7: 3D model mock-up of closed-loop system.

The team conducted 3D modeling of the storage tank under a simple constant supply and return flow through the heat pump water heater. This 3D modeling was initially validated by comparing it to an accepted 1D formula describing thermocline development and stratification (the Boussinesq approximation). The 3D modeling was run with a simple heat pump water heater flow rate of 10 and 20 GPM at the maximum heat pump water heater outlet temperature to confirm that stratification could indeed develop in the design geometries and specifications. As shown in Figure 8, these thermoclines develop in the absence of load conditions. This confirmation was necessary to validate the ability of the unique geometries and plumbing to stratify and satisfy DHW loads through the various plumbing placements. Note that in the presence of DHW loads where the return water from the HX reinforces the stratification by injecting cold water into the bottom of the tank, the thermocline becomes more pronounced and ideal. The modeling shown in Figure 8 was done purely for model validation purposes and for calibration of the 1D model.

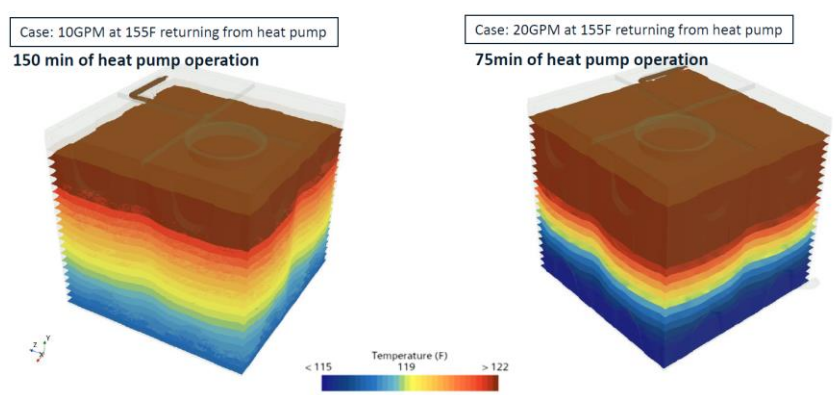

Figure 8: Thermocline development under 10 GPM and 20 GPM heat pump water heater flow rates.

Calibrated 1D Models

The 3D model takes a very long time to run and is not economical to use it for 24-hour model runs or to assess various sets of design and operating conditions. Rather, the 3D model was developed primarily to 1) confirm that a thermocline will indeed develop with and without flow through the load side heat exchangers and 2) to produce results for an unpressurized storage tank that can be used to then calibrate a more computationally efficient 1D model in GT-Suite. The validated, calibrated 1D model was then be used for various model runs in shorter amounts of time. Furthermore, the 1D model can incorporate other components of the system beyond the tank (such as mixing valves and heat exchangers) for more comprehensive system results. Figure 9 and Figure 10 show the computational elements in the 1D models, each of which has adjustable parameters that could be used for calibration to the validated 3D model. Table 7 in the Appendix shows the final calibration results with less than five percent error between the 1D and 3D models across the thermocline for both 10 and 20 GPM flow rates through the heat pump water heater loop.

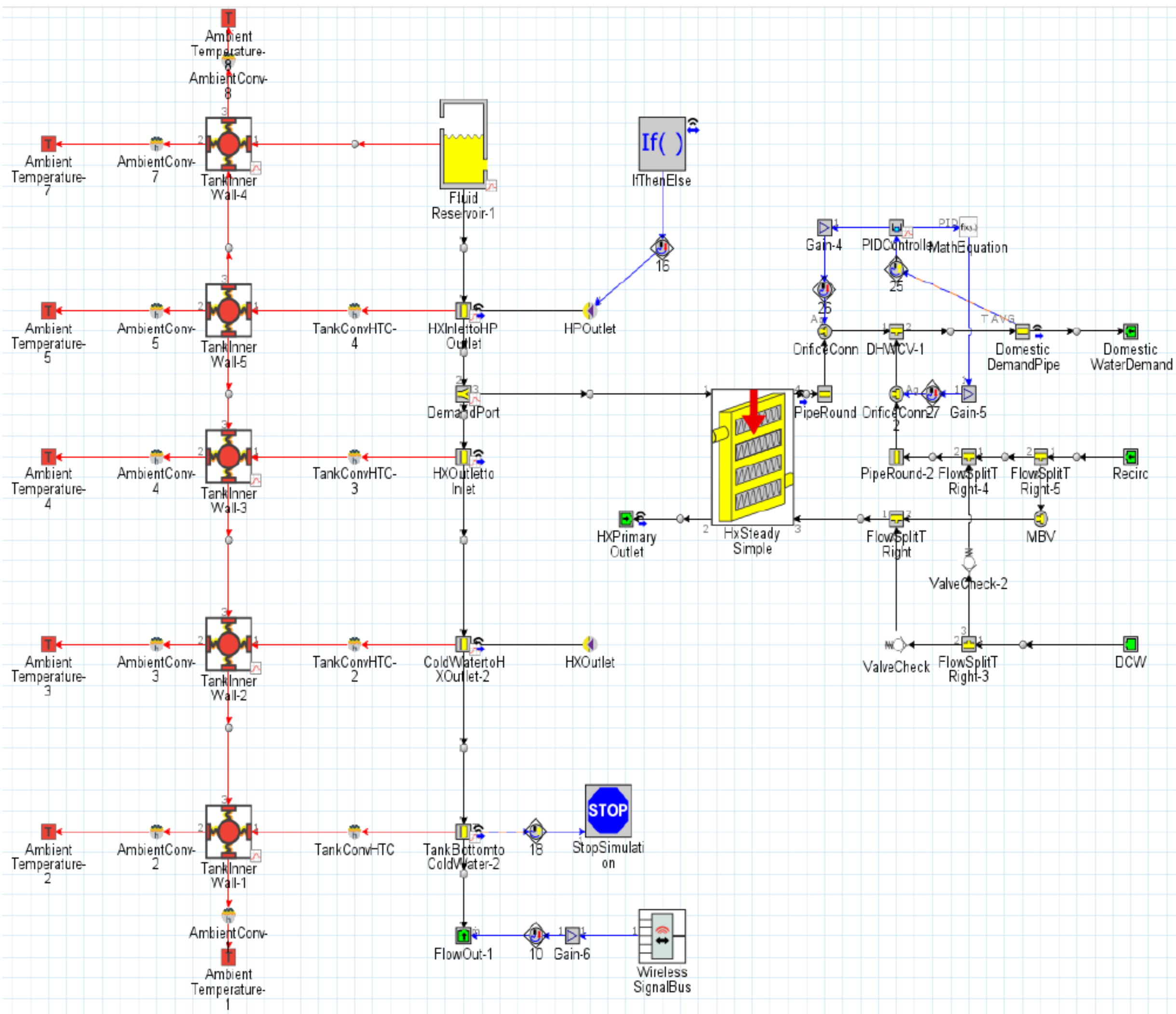

Figure 9: Closed-loop 1D model element schematic.

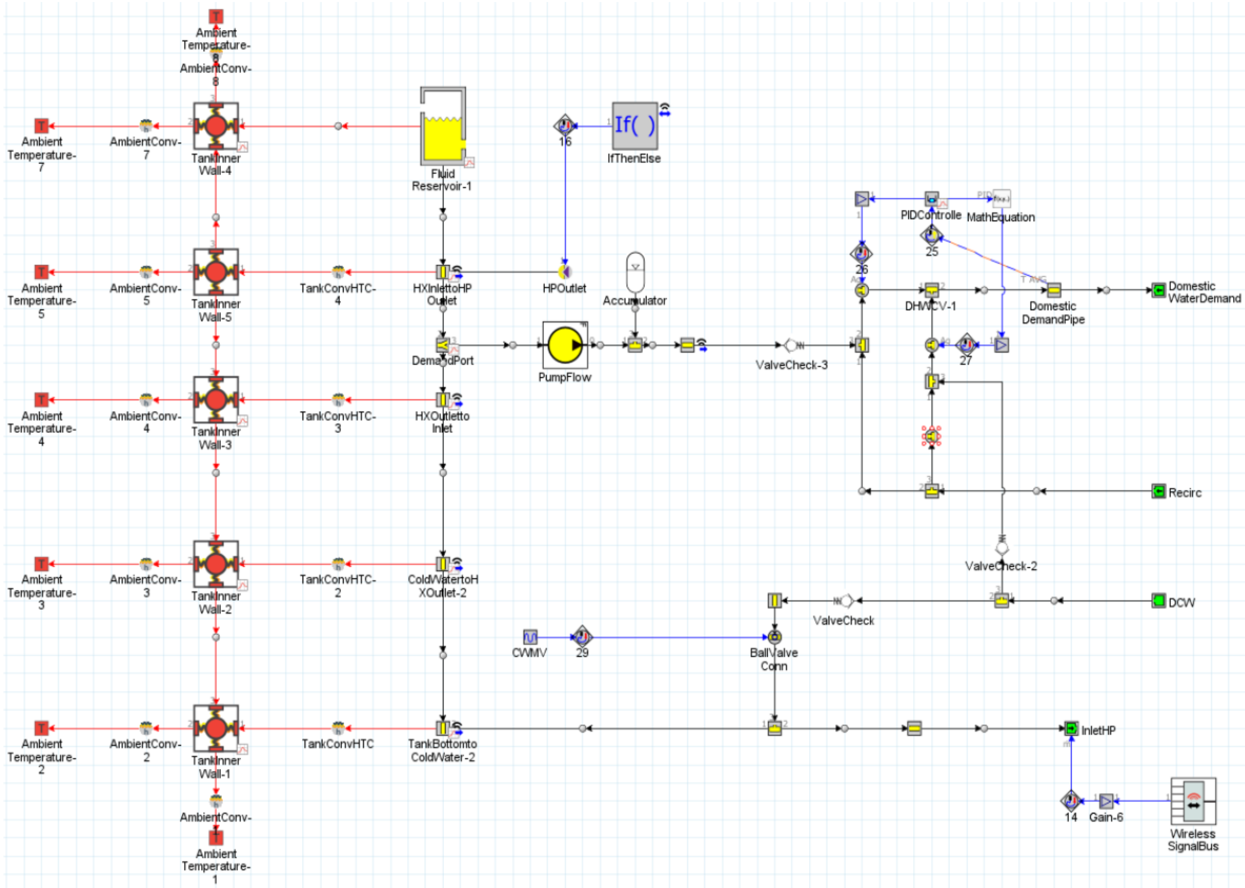

Figure 10: Open-loop 1D model element schematic.

Twenty-four Hour Load Profiles

The team developed 24-hour load profiles for use in the 1D modeling of an entire day's operation. These load profiles were based on empirical data collected from recirculating central heat pump water heater systems in California and Washington apartment buildings in other central heat pump water heater field studies. These empirical datapoints were selected because they provide the minute-by-minute data that is particularly useful for the CFD modeling approach, rather than hourly load profiles such as those found in most standard DHW references. Measured hot water draw data from these buildings was simplified to a per-occupant basis so that it could be scaled to a range of building sizes for both the average day and a peak design day. Peak load design day was determined based on the methodology in the Ecosizer central heat pump water heater sizing tool which selects representative design days from empirical data based on the 98th percentile daily load and "peakiness" of the load shape (Price, et al. 2024). Figure 11 shows these per-person DHW load profiles.

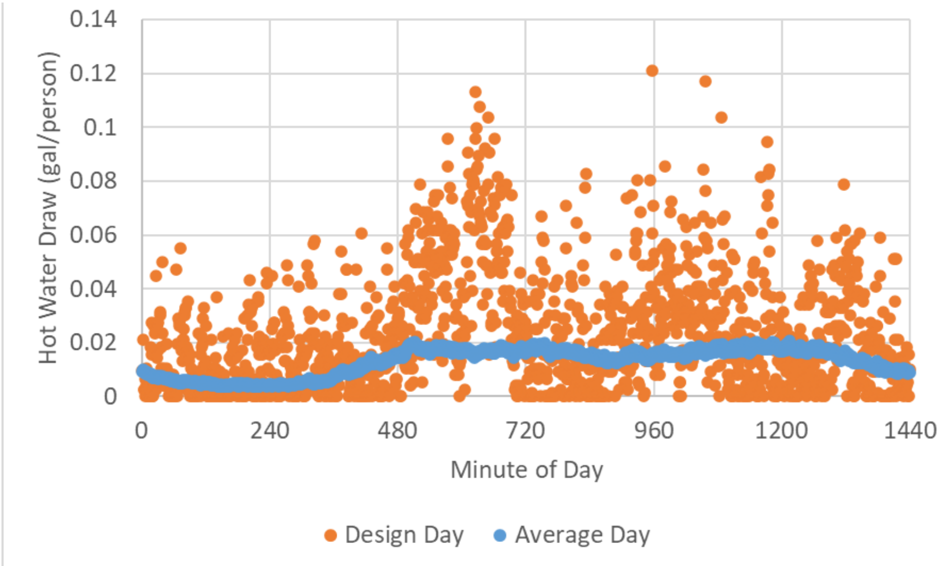

Figure 11: Average and peak design day hot water draw profiles on a per-person basis.

Recirculation flow must also be added to the load profile based on the assumed building size and distribution losses. Recirculation flow rate was based on assumed distribution system loss rates of 60 and 100 watts/apartment for new construction and existing multifamily buildings, respectively. These assumed values are recommended rules of thumb for sizing central DHW systems. From this assumption, the recirculation flow rate required to overcome distribution losses can be defined by:

$$Recirc\ GPM = \frac{\#Apartments * 3.41\frac{Btu/h}{watt} * watts/Apt}{8.34\ \frac{lbs}{gal} * 60\frac{min}{hour} * (T_{HWS} - T_{HWR})}$$

where T_{HWS} is the hot water supply temperature (125°F), and T_{HWR} is the hot water recirculation return temperature (115°F).

Assuming an occupancy of 1.8 people per apartment, the recirculation and hot water draw profiles can be combined for any building size. For example, a building with 300 occupants (~167 apartments) and 60 watts/apartment distribution losses will have a recirculation flow rate of 6.82 GPM. This recirculation load combined with the profile in Figure 11 scaled to 300 occupants will yield the total building load profile. Figure 12 and Figure 13 show the combined load profiles for this example.

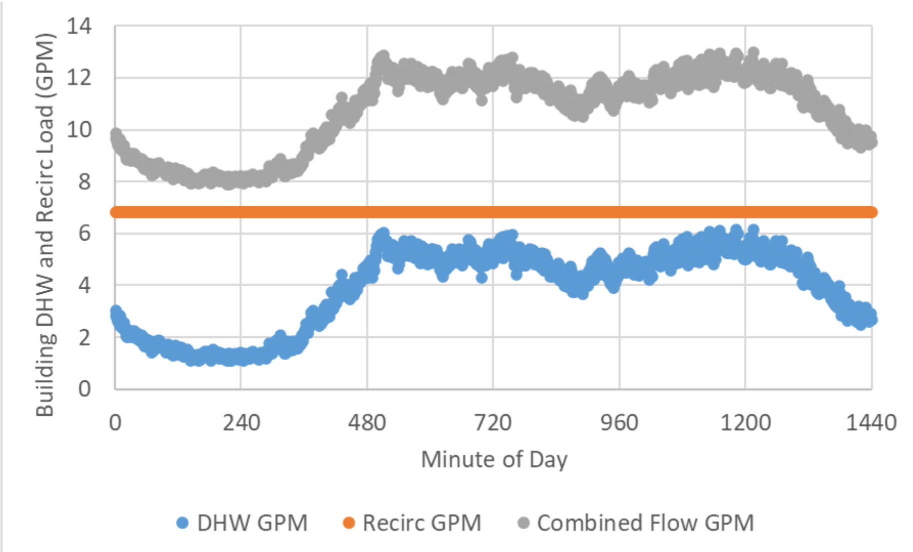

Figure 12: Example average day profiles for a 300-person, 167-unit multifamily building.

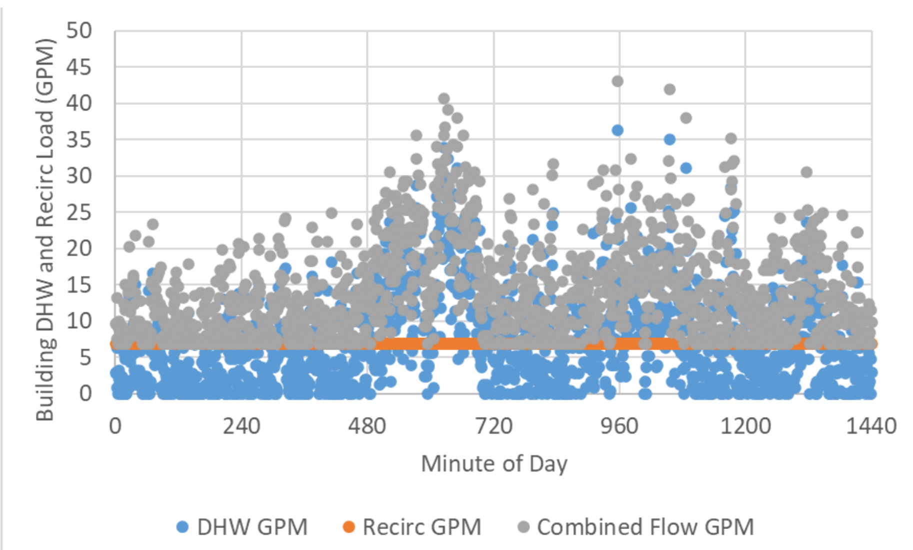

Figure 13: Example peak design day profiles for a 300-person, 167-unit multifamily building.

To evaluate the central heat pump water heater system across a range of applications, a test matrix was developed that covers variations in location, ambient temperature, load profile, city make-up water temperature, people/apartment, and building size. This test matrix was designed to assess the performance of the system and match the system capabilities to building size for future sizing decisions. For instance, modeling the system across a range of 100 to 500 occupants allows the team to determine the largest building that the system will be able to reliably serve under the various assumed conditions, component specifications, and load profiles. This test matrix of 36 cases is listed in Appendix A under the Modeling Test Matrix section. The first seven test cases are designed to determine the maximum building size that the system can reliably serve; the remainder of the test cases are based on this maximum size with variations on location, time of year, outside air temperature, load profile (e.g., design versus average day), building occupancy, city water temperature, and recirculation losses.

Findings

Sizing

One of the primary goals of the study was to understand what size building and load magnitude the new technology can satisfy. This sizing match was determined by two criteria in the modeled results for the defined peak design day loads: heat pump run time and hot water supply temperatures to the building. The heat pump manufacturer recommendation is to keep the daily run times under 16 hours. If the loads on the heat pump become great enough that they need to run more than 16 hours a day, the modeled runs imply that the system is undersized for the application. In parallel with that sizing metric, the ability of the system to meet peak loads while delivering hot water to the building at the setpoint can indicate proper sizing. If the building size and loads are so great that the system cannot provide water at the 125°F supply setpoint, the system may again be undersized.

The sizing model runs suggested that the packaged system with two heat pumps with 2,000 gallons of atmospheric storage can likely supply a building under the assumed load conditions with about 300 to 350 occupants in 167 to 195 apartments. This should be taken only as an example since the appropriate sizing of any actual situation will depend heavily on the hot water draw profiles and recirculation loads at a specific building. The new product may very well be able to serve buildings larger than the assumed 300- to 350-occupant building described above, especially if peak load day profiles differ. However, for these purposes, the team assumes that the modeled building is representative of typical conditions in an existing multifamily building.

Table 4 shows the sizing criteria for the modeled 300- and 350-occupant buildings. With the 300-occupant building the heat pumps operated below the total recommended daily run time and did not have any DHW supply temperature deficiency. In the 350-occupant building, however, the model run suggested that the heat pumps would have to operate 17.2 hours in the day (above the 16 hour recommended limit) and had a 1.7 hour time period during which the DWH supply temperature was below the 125°F setpoint, indicating the system could not keep up with the load.

Table 4: Sizing Criteria for 300- and 350-Occupant Building (Peak Design Day Load Profile)

Building Size	Daily Runtime (hr)	Time with Supply Temp Below Setpoint (hr)	Avg Supply Temp During Cold Water Event (°F)
300 Occupants	14.8	0	n/a
350 Occupants	17.2	1.7	116

Figure 14 shows these results in more detail. The peak design day load profile, cumulative run times, and DHW supply temperatures are shown.

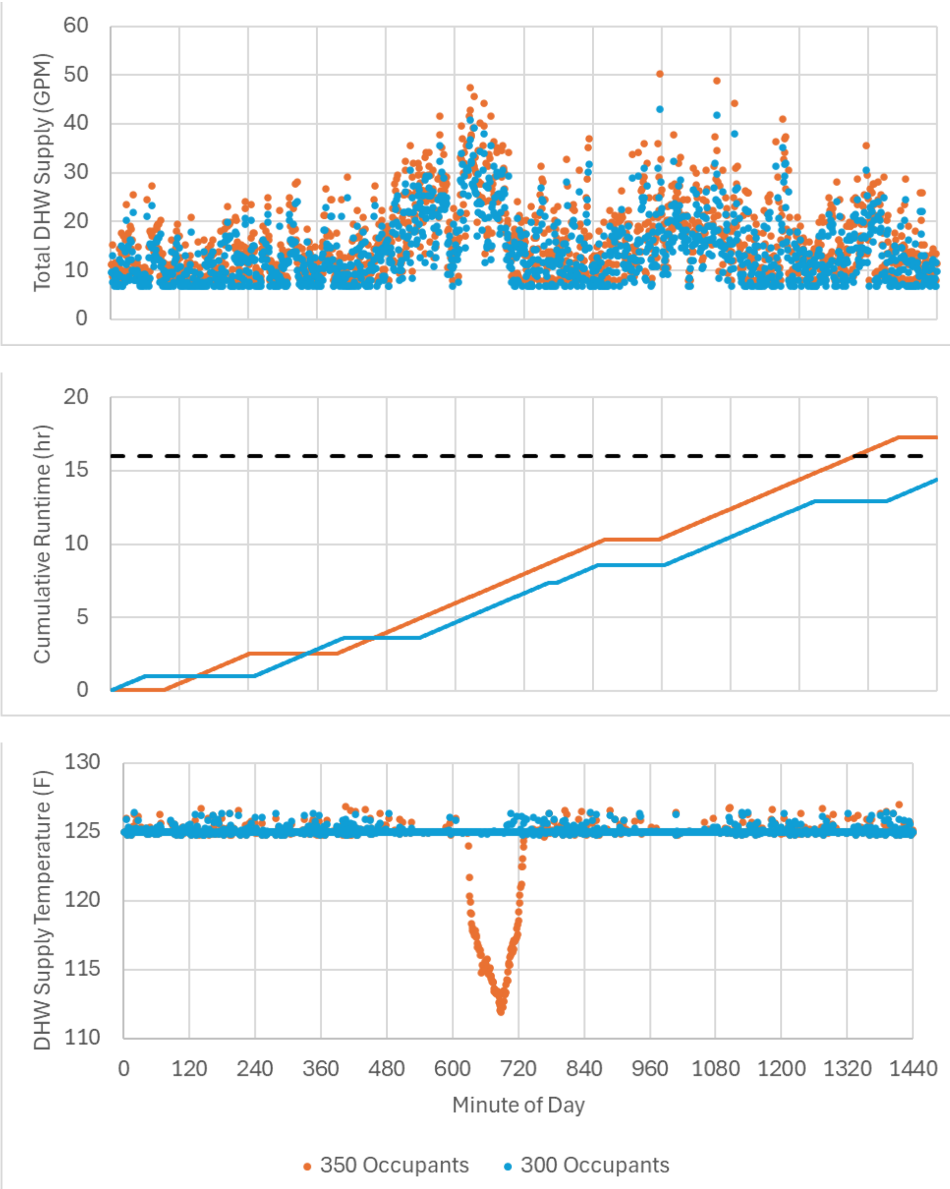

Figure 14: Loads and sizing criteria for model runs of buildings with 300 and 350 occupants.

These results provide sizing perspective but are not fully definitive regarding sizing limits. It may very well be that the brief hot water supply temperature dip would not cause unacceptable discomfort or issue for the building operators or occupants. The heat pumps may be able to withstand run times above the manufacturer recommendation on rare occasions. The supply temperature only dips to 112°F and has an average of 116°F during the observed hot water supply reduction time. This supply temperature dip would only occur during the roughly two percent of peak days in a calendar year. So, it is reasonable to conclude that the modeled system could be well-suited to a building of 300 to 350 occupants in 167 to 195 residences under assumed conditions.

It is useful to compare the emerging technology to the current, most-typical central heat pump water heater alternative: a single-pass swing tank configuration. The Ecosizer modeling and sizing tool allows for convenient sizing of a swing tank configuration for an equivalent 300- to 350-occupant building. As shown in Figure 16 and Figure 15, this tool suggests that a single-pass, swing tank configuration would need 1,766 to 2,056 gallons of storage volume with 424 to 495 kBtu/hr of heat pump capacity on the design day and an additional 120-gallon swing tank with 17.5 to 20.5 kW of electric resistance heating. This sizing very closely matches the 2,000-gallon emerging product with about 410 to 528 kBtu/hr of heat pump capacity at the design day conditions.

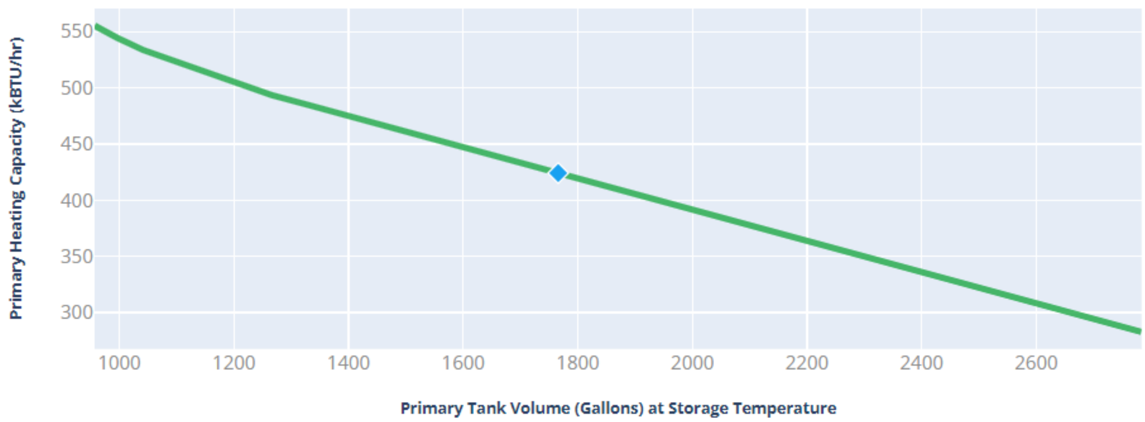

Figure 15: Ecosizer sizing for a 300-occupant building with additional 120-gallon, 17.5 kW swing tank

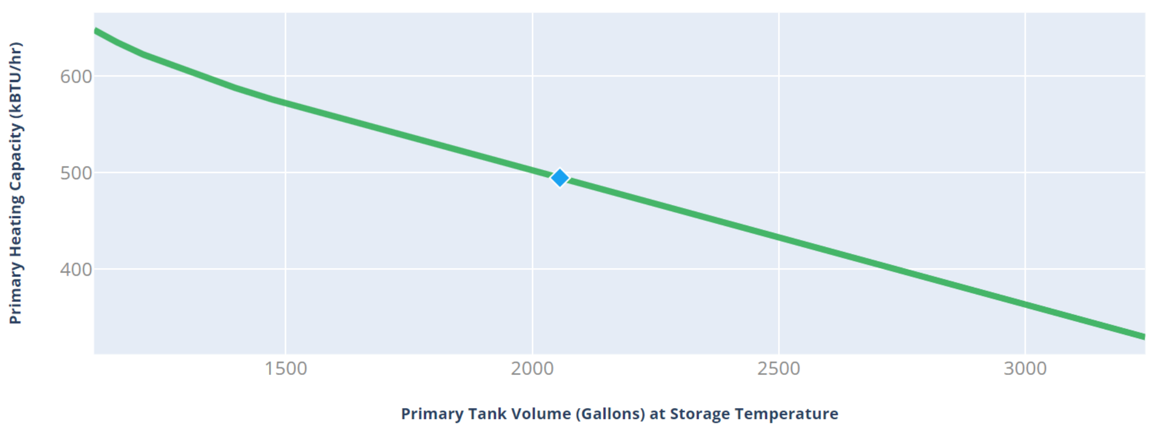

Figure 16: Ecosizer sizing for a 350-occupant building with additional 120-gallon, 20.5 kW swing tank

This suggests that the emerging technology can supply DHW to a building with roughly the same primary storage volume and heat pump capacity as a single-pass, swing tank configuration. It can do this with a smaller physical footprint and without the additional swing tank necessary for temperature maintenance.

This fair sizing comparison also allows for estimation of cost impacts. Based on past project experience, a swing tank central heat pump water heater system costs about $1,500 per apartment with a 1.6 multiplier for labor and installation costs. So, for a 300- to 350-occupant building, an installed swing tank configured system can be expected to cost around $720,000 to 840,000. The project team and manufacturer expect a materials cost reduction of about $60,000 by using unpressurized storage for the 2,000-gallon central heat pump water heater. This amounts to a very attractive seven to eight percent reduction in first cost. Additional cost savings may be realized from the reduction of labor and electrical infrastructure necessary for the swing tank heaters. The dollar value that could be assigned to reduced physical footprint and freed-up building square footage is also not included here. Note that there are still very few central heat pump water heater installations around the country and typical measure costs still have a high degree of uncertainty and can vary significantly. These costs and savings estimates should be taken as an example rather than as sufficiently definitive for budgeting or program measure package development purposes.

Representative Model Results

One of the primary outcomes of this project is a dataset of modeled performance across a range of conditions. This dataset is available for future work such as for the development of public-facing sizing or modeling tools. These modeling results could be used to validate user-friendly sizing and modeling tools such as a new return-to-primary, atmospheric storage configuration option in Ecosizer. Results from several of the modeled runs are presented in detail here.

Closed-Loop Design

The closed-loop design was modeled for a 250-occupant building across the test matrix shown in Table 8. The energy usage and storage tank temperatures for the models in Los Angeles for 1) the peak design day in February, 2) an average load day in February, and 3) an average load day in August are shown below.

Figure 17 plots the central heat pump water heater energy consumption, equivalent gas central water heater energy consumption, thermal storage volume state of charge (SOC), and system efficiency (sysCOP) for the peak design day. Gas energy consumption was calculated for a central, recirculating gas-fired water heater with a Title 24 code-minimum thermal efficiency of 86 percent. SOC is defined as the total useful energy in the thermal storage volume. Any water below 130°F is not useful due to the limits of the heat exchanger and mixing valve. In other words, an SOC of 0 means that all water in the storage volume is at or below 130°F and an SOC of 1 would indicate that the entire volume is at 150°F (the heat pump water heater outlet temperature). Note that the maximum modeled SOC is 0.7, indicating that the effective useful volume is about 70 percent of the total storage tank with the remainder dedicated to the thermocline.

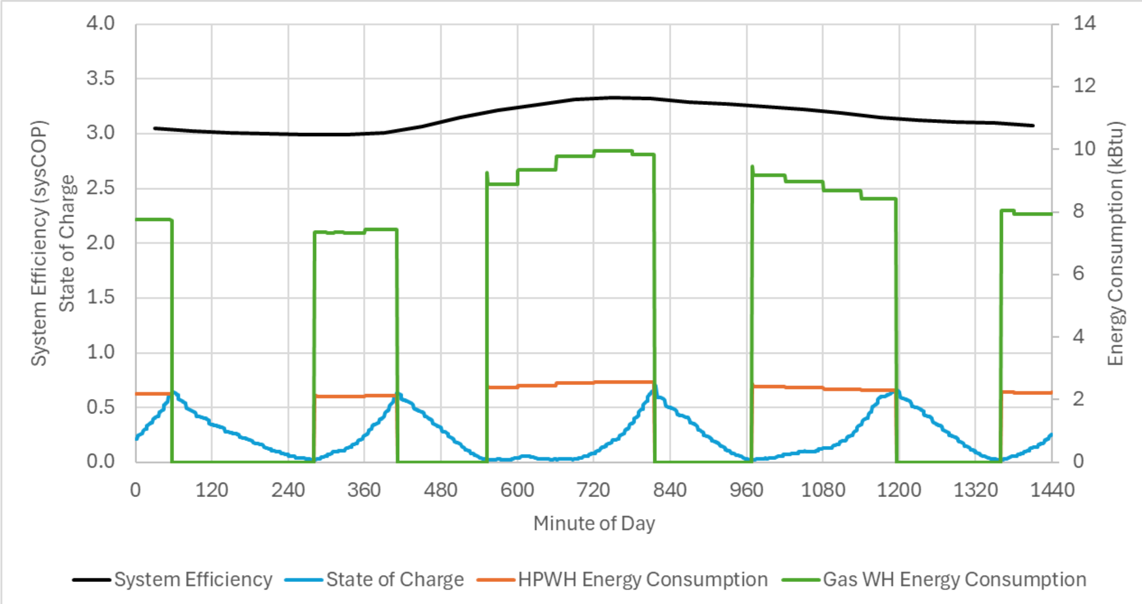

Figure 17: Energy usage, SOC, and sysCOP (Los Angeles, closed-loop, 250-occupant building, peak day).

Figure 18 shows the hot water draw profile for the building on the peak design day as well as the temperatures across the height of storage volume. The tank temperatures swing up and down as the volume is depleted and recharged. A tighter vertical spread indicates the smaller thermocline (typically seen closer to maximum SOC) while a wider spread indicates a large thermocline and less available useful hot water (typically seen at minimum SOC). Peak transient loads are especially disruptive to the thermocline stability due to the higher velocity tank outflow and inflow.

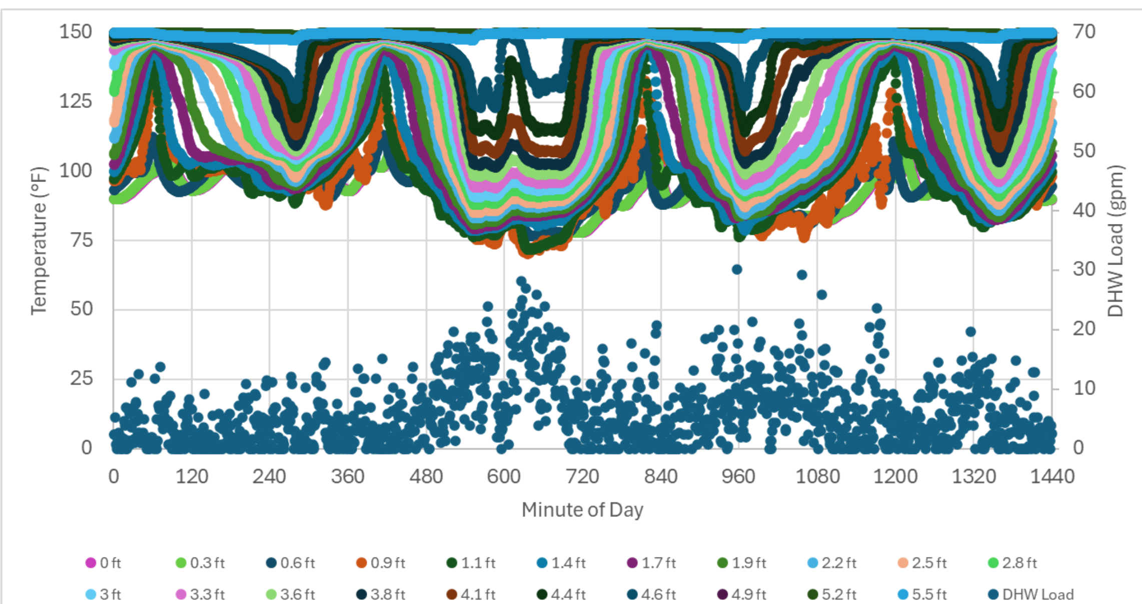

Figure 18: Thermal stratification and DHW load (Los Angeles, closed-loop, 250-occupant building, peak day).

Figure 19 and Figure 20 show the same results for the average load day in February. Figure 21 and Figure 22 show the same for August. The smoother curves are primarily due to the use of an average, representative load profile — any actual single day is likely to have more peaks and variation from minute to minute.

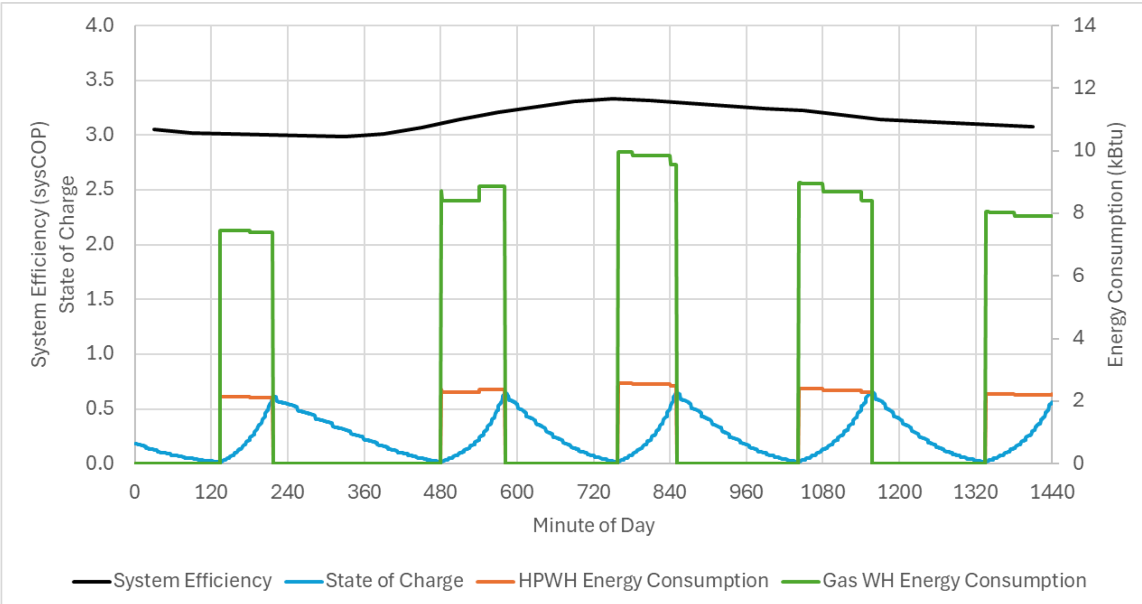

Figure 19: Energy usage, SOC, and sysCOP (Los Angeles, closed-loop, 250-occupant building, avg Feb day).

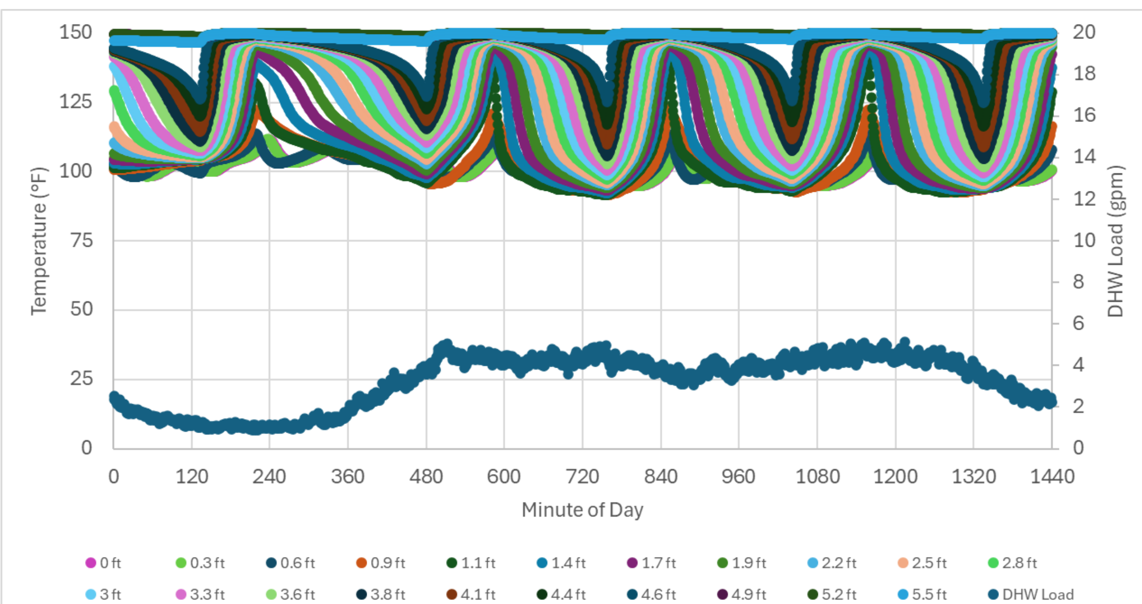

Figure 20: Thermal stratification and DHW load (Los Angeles, closed-loop, 250-occupant building, avg Feb day).

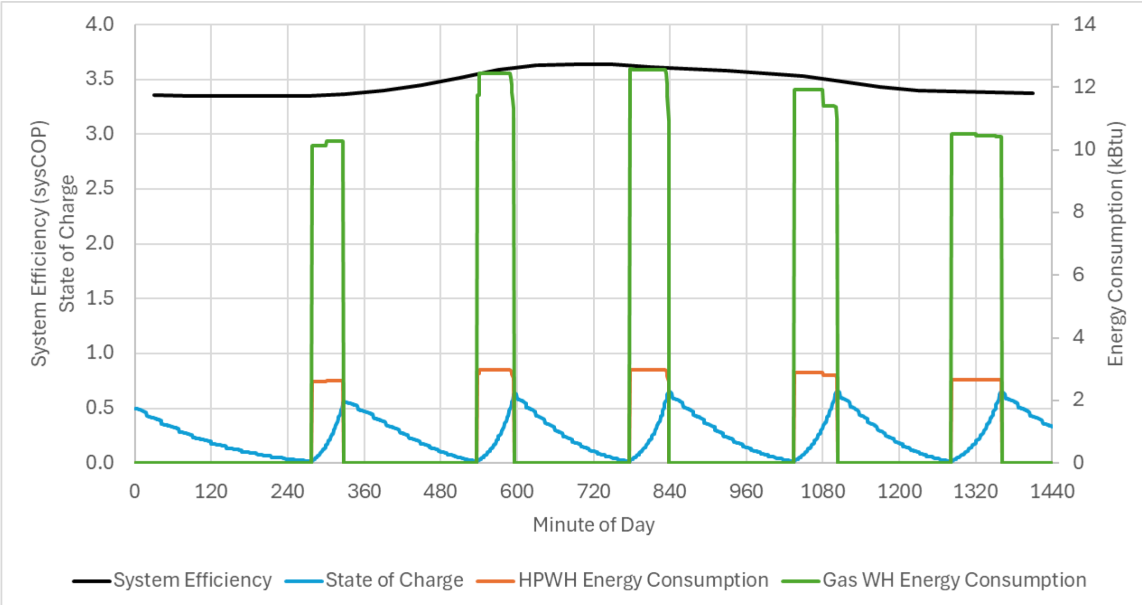

Figure 21: Energy usage, SOC, and sysCOP (Los Angeles, closed-loop, 250-occupant building, avg Aug day).

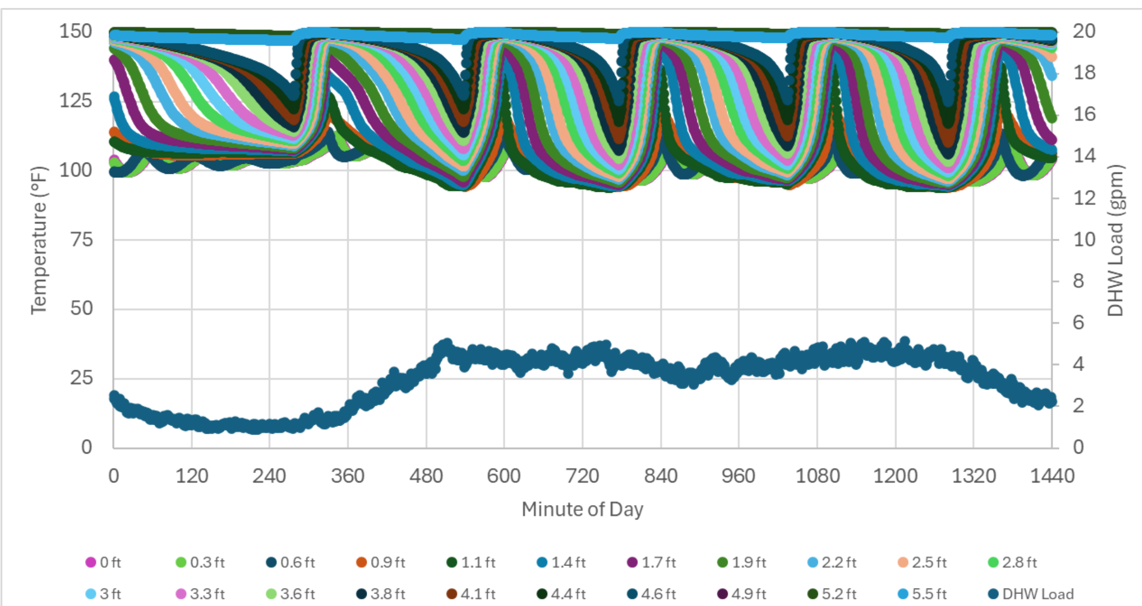

Figure 22: Thermal stratification and DHW load (Los Angeles, closed-loop, 250-occupant building, avg Aug day).

Open-Loop Design

The open-loop design was also modeled for a 250-occupant building across the test matrix shown in Table 8. The energy usage and storage tank temperatures of the models in Los Angeles for 1) the peak design day in February, 2) an average load day in February, and 3) an average load day in August are shown in the plots below.

In the open-loop design, the team observed that there were periods of time during which the heat pumps and thermal storage could not meet the load. This appeared to occur mainly at low load times when the make-up and return water volumes back to the storage volume were low (also often when the thermal storage was recovering). Under these low load conditions, insufficient hot water is drawn from the top of the storage tank to raise the outlet temp of the mixing valve to the supply temperature, even if the top of the tank is at its maximum temperature (150°F). One potential solution is to this is to add an inline electric resistance heater at the recirculation pipe to raise the temperature sent back to the mixing valve. This approach would be similar to a swing tank, but with much lower power requirements.

Figure 23 shows the supply temperature of the open-loop system on the design day without any supplemental electric resistance heat. Where it dips below the 125°F supply setpoint, the system will require some supplemental heat. The green line inverse to the supply temperature shows the amount of supplemental heat that would be needed.

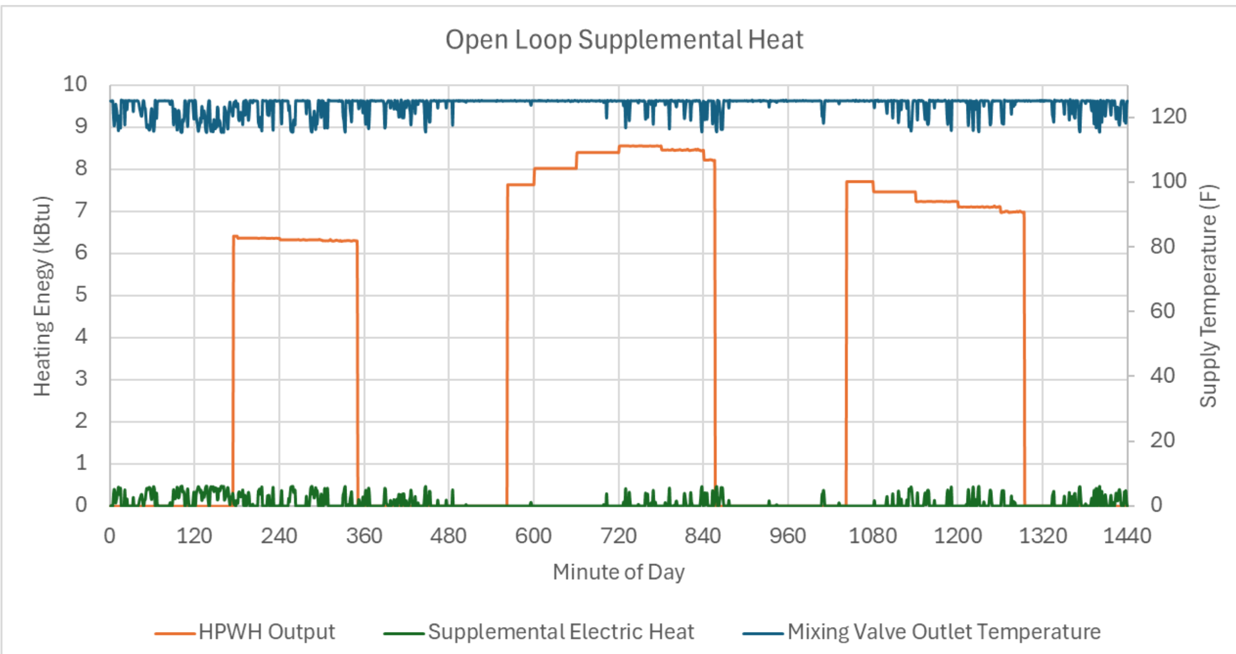

Figure 23: Supply temperature without supplemental electric resistance heat and the necessary additional heat input (Los Angeles, open-loop, 250-occupant building, peak day).

Figure 24 plots the central heat pump water heater system energy consumption, SOC, and sysCOP for the peak design day as well as gas consumption for an equivalent central gas water heater. For the open-loop design, the total energy consumption and sysCOP must include the additional booster pump and inline electric resistance heater that is not present in the closed-loop design. The resistance heater accounts for the small spikes and the low periods of sysCOP around 1 during which the resistance heater is the only active heat source (even though hot water is being drawn from the tank that was heated by the heat pump previously).

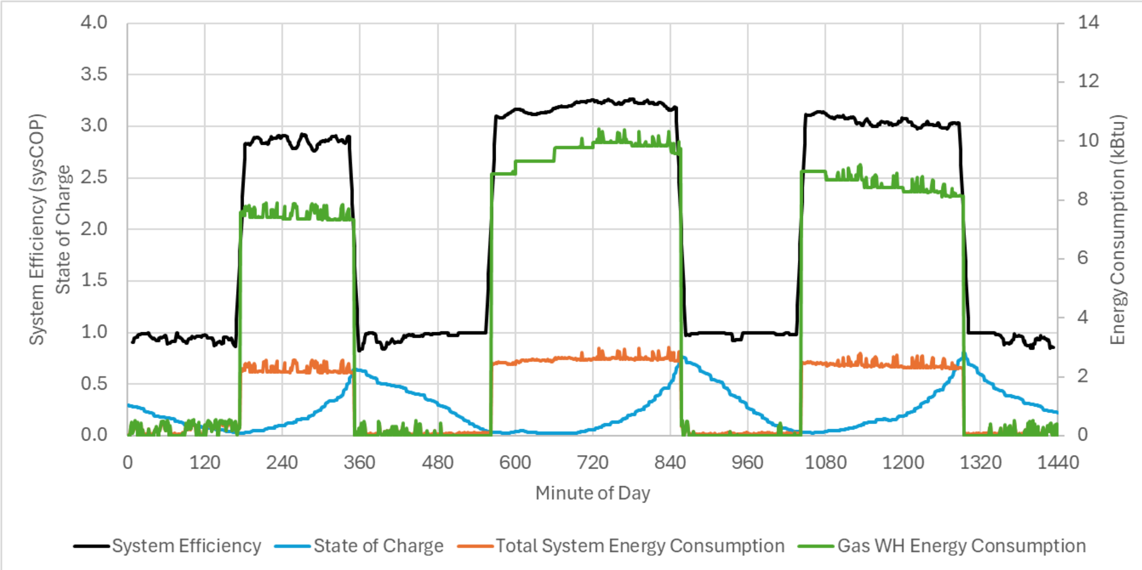

Figure 24: Energy usage, SOC, and sysCOP (Los Angeles, open-loop, 250-occupant building, peak day).

Figure 25 again shows the DHW draw profile and the temperature stratification across the storage volume. The open-loop design stratification differs from the closed-loop design, especially by the range of temperatures in the tank. Since the open-loop design has cold city make-up water entering the bottom of the tank, the lower height temperatures can be as low as the incoming water temperature (55 °F in this case). However, this lower cold temperature in comparison to the closed-loop design does not appear to be particularly detrimental to the maximum SOC that can be achieved.

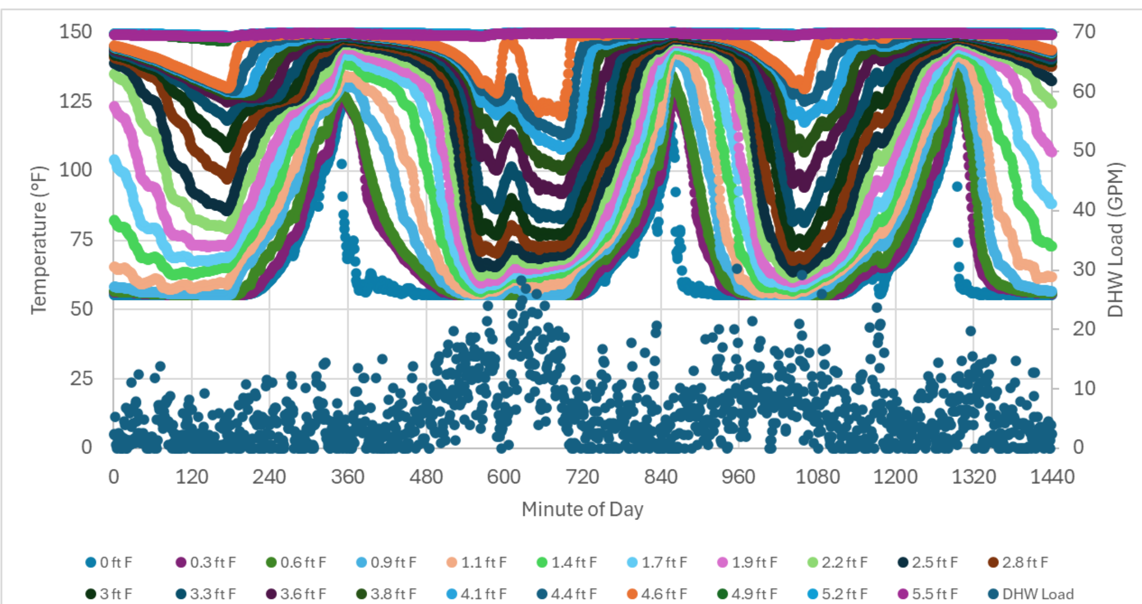

Figure 25: Thermal stratification and DHW load (Los Angeles, open-loop, 250-occupant building, peak day).

Figure 26 and Figure 27 show the same plots for the average load day in February; Figure 28 and Figure 29 shows the same for August. Again, curves are smoothed due to the use of an average, representative load profile.

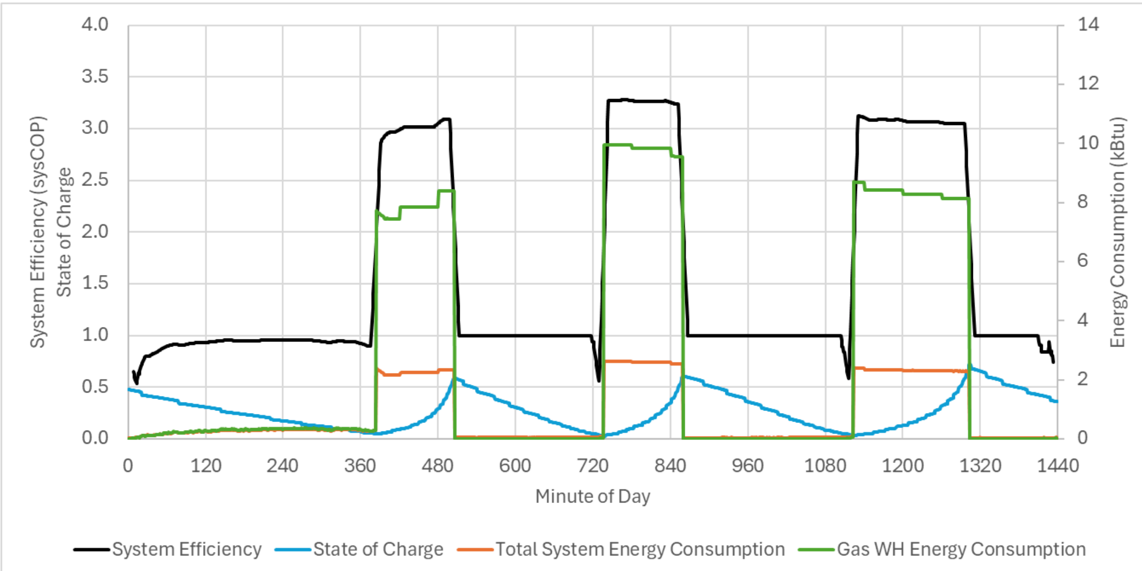

Figure 26: Energy usage, SOC, and sysCOP (Los Angeles, open-loop, 250-occupant building, avg Feb day).

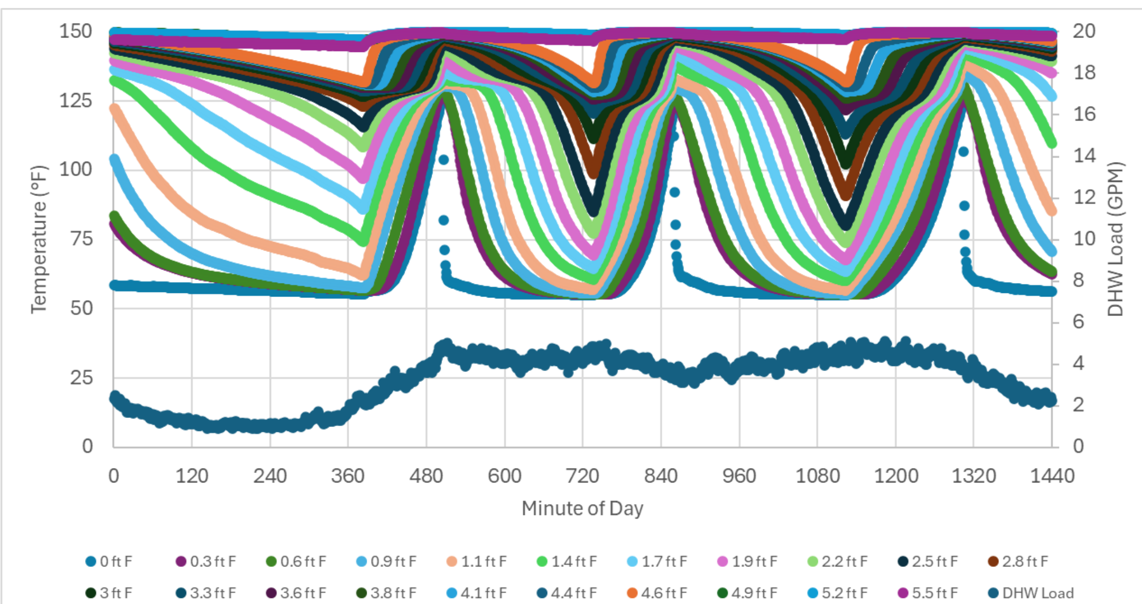

Figure 27: Thermal stratification and DHW load (Los Angeles, open-loop, 250-occupant building, avg Feb day).

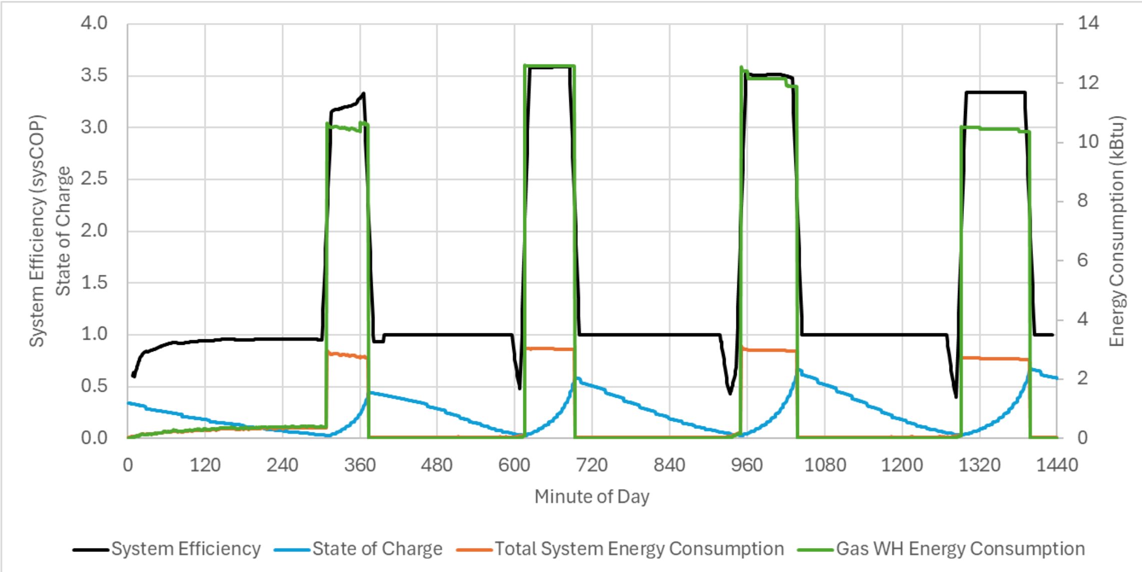

Figure 28: Energy usage, SOC, and sysCOP (Los Angeles, closed-loop, 250-occupant building, avg Aug day).

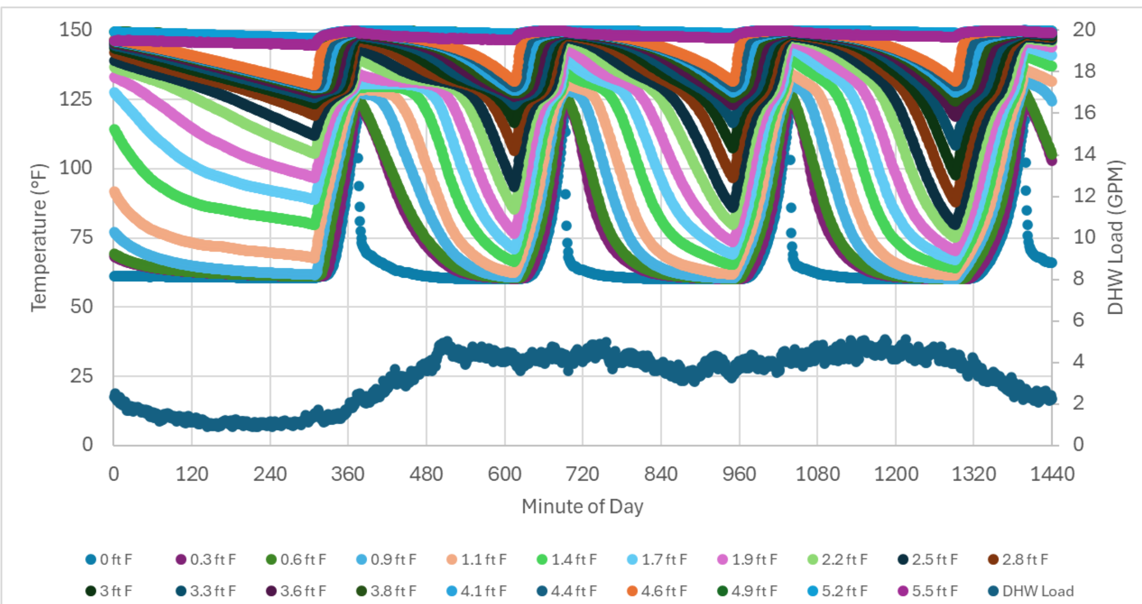

Figure 29: Thermal stratification and DHW load (Los Angeles, open-loop, 250-occupant building, avg Aug day).

Topline energy efficiency, impacts, and greenhouse gas (GHG) emissions savings over a natural gas baseline are shown in Table 5. GHG savings of 83-85% percent are calculated using hourly long-run marginal emissions factors for CAISO grid electricity (NREL 2024) and piped natural gas from the California Air Resources Board (California Air Resources Board n.d.). These GHG reductions are similar to what a swing tank central heat pump water heater would achieve, as well (Valmiki, et al. 2023). However, the return-to-primary atmospheric storage central heat pump water heater design reduces peak electrical demand and infrastructure needs with the elimination of the swing tank. The closed-loop design completely eliminates the swing tank (17.5 to 20.5 kW reduction) while the open-loop system would likely require supplemental electric resistance heat which results in less peak demand reduction (9.5 to 12.5 kW). The booster pump and electric resistance heater account for four to five percent and five to nine percent of total energy consumption for the open-loop central heat pump water heater, thereby reducing the overall sysCOP and GHG savings in comparison to the closed-loop design.

Table 5: Efficiency, Energy Usage, and Savings

	Closed Loop	Open Loop	Notes
Average daily sysCOP	3.2–3.5	2.9–3.1	Daily sysCOP captures full, representative draw and recovery cycles and includes ancillary energy from pumping and electric resistance for the open-loop design
Added pumping energy (%)	n/a	4–5%	For open-loop design booster pump
Booster electric resistance heater size (kW)	n/a	8	For open-loop design supplemental heater
Booster electric resistance heater energy (%)	n/a	5–9%	For open-loop design supplemental heater
Peak demand reduction compared to swing tank configuration (kW)	17.5–20.5	9.5–12.5	Peak demand and electrical infrastructure reductions compared to equivalent swing tank configuration central heat pump water heater
Average central heat pump water heater energy consumption per Person (kWh/day-person)	1.0–1.4	1.3–1.4	Total system energy consumption for average load days
Equivalent energy consumption for code-efficient gas water heater (therm/day-person)	0.15–0.17	0.15–0.16	Equivalent natural gas energy consumption for a code-efficiency central DHW system
GHG savings over gas baseline (CO_2e ton/day-person)	85%	83%	GHG emissions reduction over an equivalent natural gas code-efficient central DHW system.

Stakeholder Feedback

This report was circulated to central heat pump water heater subject matter experts and stakeholders to gather feedback on findings. Some recommendations and changes were made to the report based on their feedback (e.g., editorial changes and clarification). Some general impressions included:

- "It's interesting to see manufacturers bringing low-pressure central heat pump water heater systems to the market and to see the potential cost reductions as well as avoidance of swing tanks."
- "In general, I think this new technology is promising. Innovations such as atmospheric storage that reduce first costs of equipment and reduce barriers to retrofit installations will be beneficial."
- "For the issue of stratification dynamics with multiple incoming flows, we are aware of another project that created a sort of 'diffuser' to minimize disturbances that would cause destratification."

Additional stakeholder outreach after publication will include wider report dissemination and possible presentation in a conference or webinar such as through the Advanced Water Heating Initiative.

Discussion and Recommendations

The modeling showed that a return-to-primary configuration central heat pump water heater system with an unpressurized thermal storage tank may be an effective solution and warrants further research, including lab testing and actual installation. The project team can confidently recommend use of the product and specification from the design engineers based on these findings. Under a very specific set of assumed operating conditions and load profiles, the 2,000-gallon system was shown to be effective for a building with up to 300 or 350 occupants in 167 to 195 apartments.

The project team recommends that the product be demonstrated in a building, ideally in a scenario with peak loads somewhat under the expected maximum sizing to mitigate risk without compromising research potential. If possible, measurement plans for any future lab or field testing should be designed to show stratification profiles alongside the typical central heat pump water heater datapoints (e.g., power and boundary conditions of flow and temperatures. Lab and field testing should be done for both the 2,000-gallon option as well as the smaller 1,000-gallon form factor; the smaller form factor may show improved thermocline dynamics and stratification – although that is not currently seen as a barrier to use of the 2,000-gallon option. The team also recommends that sysCOP be quantified and compared between the closed- and open-loop versions. While the elimination of the secondary HX in the open-loop design is a cost and efficiency benefit, it comes at the sake of additional pumping energy and potential increased use of an electric resistance heat source.

Future lab and field demonstrations should also include assessment of load shift capabilities and performance. The load shifting performance of the emerging technology is particularly uncertain due to relatively poor understanding of the stratification dynamics of a rectangular storage volume with flow disturbances from both load-side and primary-side penetrations. Empirical data from load shift testing of this new design is necessary. Future central heat pump water heater best practices, guidelines, and standards will most certainly include load shift controls and research should anticipate that need.

The team also recommends additional research and development of sizing and modeling tools that include open- and closed-loop atmospheric storage in a return-to-primary configuration. For instance, the Ecosizer tool (and underlying model that is used elsewhere, such as for California energy code compliance software) could be updated to include this novel configuration. In general, there is an urgent need for updated modeling and sizing tools that include return-to-primary systems. As part of the inevitable updates to these sizing and compliance models towards return-to-primary inclusion, atmospheric storage should be added.

In general, the modeling efforts and results did not suggest that any of the emerging technology benefits listed in the Background section are at risk. The technology can reduce first costs, physical footprint, electrical capacity and peak demand, and barriers to central heat pump water heater adoption in some buildings. It can also provide advantages such as convenient separation of the heat pump water heater and DHW loops which reduces complications regarding freeze protection and high building pressure. The emerging technology can help unlock the total market potential for decarbonization of DHW in multifamily and commercial buildings through expansion of available central heat pump water heater options. However, the team has some recommendations for manufacturers to consider in product development:

- Total storage volume flexibility is important for design engineers so they can specify systems that match building loads. Product lines with modular volumes or options to mix-and-match volume and heat pump water heater capacity will allow for selection of a system that is more effective and efficient for a given building, load shifting needs, and load profile.
- Similarly, having a range of HX sizing options for the closed-loop design will allow for the specifying engineer to match system capacity and available heat rate to varying building needs, especially peak instantaneous loads. Options for HX selection that can accommodate higher flow rates may expand product applications and possible use cases.
- Manufacturers should include at least three temperature measurements across the tank height and corresponding load shift control sequences as a standard feature. In versions that include multiple heat pumps or electric resistance heaters for supplemental heat, lockouts and staging should also be a standard control capability.
- Design features that enhance stratification of the storage volume may be valuable. Consider a moving baffle, inlet/outlet flow disturbance mitigation, and reduction of vertical heat transfer through the tank walls.
- Optimize the return pipe location to the storage volume from the DHW side (HX loop or direct recirculation return). Further lab testing or modeling may be needed for this goal.
- Include measurement and monitoring capabilities in future installations for research, development, and maintenance purposes. Data from future applications will provide the

opportunity for further product development, case study publication, and research opportunities.

Finally, the project team recommends that energy program designers and administrators take steps to include return-to-primary atmospheric storage central heat pump water heaters in existing and future programs. Programs should explicitly identify this new type of central heat pump water heater to encourage adoption and market expansion since the energy and GHG benefits are apparent and similar to other central heat pump water heater options with potential for lower measure costs and other benefits. To that end, programs can provide support through:

- Additional funding for laboratory testing (including testing load side heat exchangers, internal helical coil heat exchangers, and closed-loop designs) and field demonstration. Both laboratory and field demonstration should also include load shifting tests to empirically assess load shift performance and controls.
- Funding for expansion of modeling and sizing tools (including compliance software).
- Using sizing and modeling tools to establish measure impacts for return-to-primary and atmospheric storage central heat pump water heater systems that may have lower cost and larger kW impacts over swing tank configurations.
- Addition of return-to-primary and atmospheric storage central heat pump water heater systems as an explicit measure in existing workpapers and program definitions.

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Appendix A: Supplemental Information

Heat Pump Performance Map and Regressions

Each R513A heat pump water heater on the packaged product skid has capacity, efficiency, and flow rates according to the following empirical manufacturer data:

Table 6: Heat Pump Water Heater Performance Map

140° F Leaving Water Temperature (LWT)										
H2O inlet temperature 40°F										
Ambient DB °F	**10**	**24**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**
Heating capacity (kbtu/h)	98	126	141	171	205	241	287	333	351	351
Cooling capacity (kbtu/h)	60	76	88	112	140	171	211	252	269	269
Unit heating COP	2.4	2.4	2.5	2.8	3.0	3.2	3.4	3.6	3.8	3.9
H2O flow rate (gpm)	2.0	2.5	2.8	3.4	4.1	4.8	5.7	6.6	7.0	7.0
H2O inlet temperature 50°F										
Ambient DB °F	**10**	**24**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**
Heating capacity (kbtu/h)	98	126	141	171	205	241	287	333	351	351
Cooling capacity (kbtu/h)	60	76	88	112	140	171	211	252	269	269
Unit heating COP	2.4	2.4	2.5	2.8	3.0	3.2	3.4	3.6	3.8	3.9
H2O flow rate (gpm)	2.2	2.8	3.1	3.8	4.5	5.3	6.4	7.4	7.8	7.8
H2O inlet temperature 60°F										
Ambient DB °F	**10**	**24**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**
Heating capacity (kbtu/h)	98	126	141	171	205	241	287	333	351	351
Cooling capacity (kbtu/h)	60	76	88	112	140	171	211	252	269	269
Unit heating COP	2.4	2.4	2.5	2.8	3.0	3.2	3.4	3.6	3.8	3.9
H2O flow rate (gpm)	2.4	3.1	3.5	4.3	5.1	6.0	7.2	8.3	8.8	8.8
H2O inlet temperature 70°F										
Ambient DB °F	**10**	**24**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**
Heating capacity (kbtu/h)	98	126	141	171	205	241	287	333	351	351
Cooling capacity (kbtu/h)	60	76	88	112	140	171	211	252	269	269
Unit heating COP	2.4	2.4	2.5	2.8	3.0	3.2	3.4	3.6	3.8	3.9
H2O flow rate (gpm)	2.8	3.6	4.0	4.9	5.8	6.9	8.2	9.5	10.0	10.0

Note: All conditions are at 50% relative humidity.

From this data regressions for capacity, efficiency, and flow rate were used to help define the boundary conditions of control volume for 1D and 3D modeling. The regression in Figure 30 establishes the relationship between ambient temperature and the heating capacity of each heat pump.

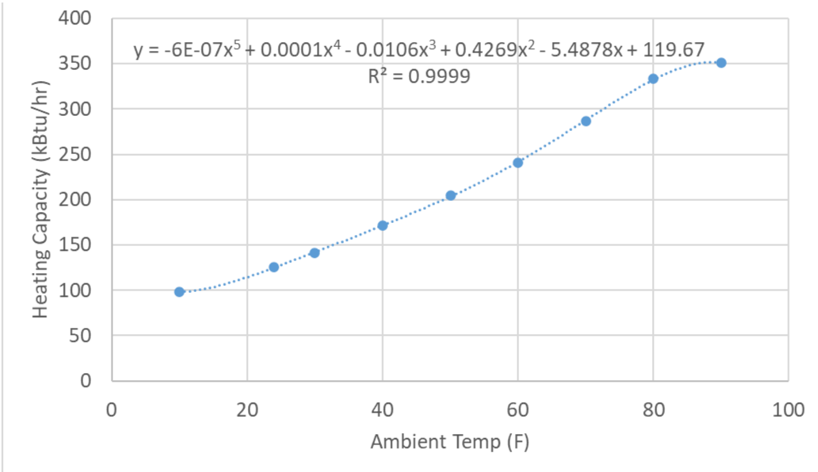

Figure 30: Heat pump water heater capacity regression to ambient temperature.

The regression in Figure 31 establishes the relationship between heat pump water heater efficiency and ambient temperature.

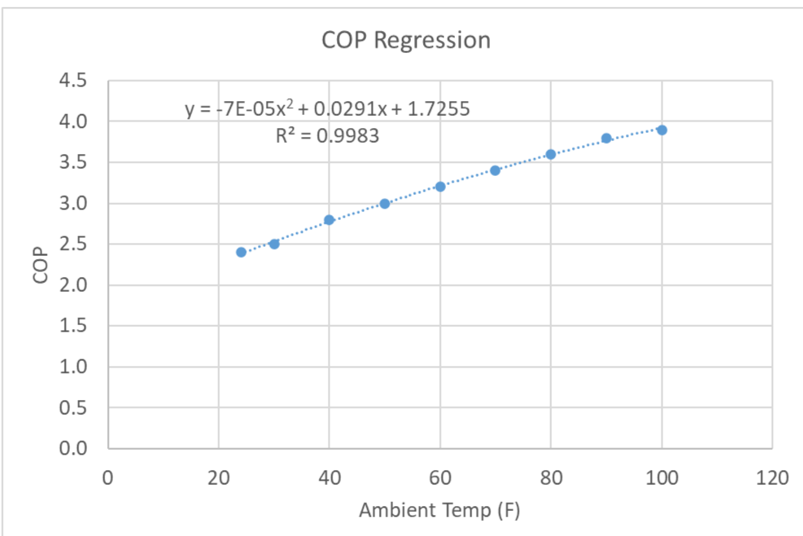

Figure 31: Heat pump water heater efficiency regression to ambient temperature.

Finally, a multivariable regression with an R^2 of 0.96 establishes the relationship between heat pump water heater flow rate and the ambient temperature and the inlet temperature in the following form:

$$HPWH\ Flowrate\ (gpm) = -2.361976304 + 0.0658 * Inlet\ Temp\ (F) + 0.076678273 * Ambient\ Temp\ (F)$$

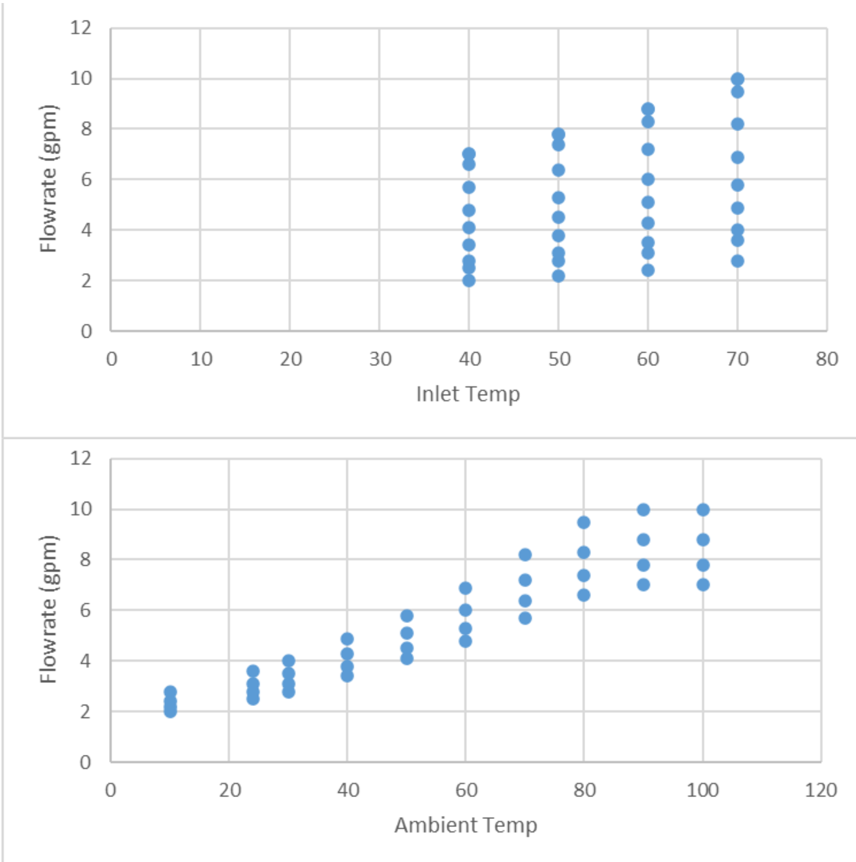

Figure 32: Heat pump water heater flow rate dependence on inlet temperature (°F) and ambient temperature (°F).

1D Model Calibration

Table 7 shows the comparison between the 3D and 1D models, demonstrating the validation of the 1D model. Each point in the vertical axis of the tank varied less than five percent on average over the validation test runs.

Table 7: Comparison Between the 3D and 1D Models Under Identical Operating Conditions

% Error in 10GPM Tank Charging																					
Section height (in m)/Physical time (s)	0.00	0.11	0.19	0.28	0.36	0.45	0.54	0.62	0.71	0.79	0.88	0.96	1.05	1.14	1.22	1.31	1.39	1.48	1.57	1.65	1.74
0	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
1800	-4%	-5%	-5%	-4%	-5%	-6%	-7%	-7%	-8%	-9%	-10%	-9%	-4%	4%	10%	2%	-5%	-1%	5%	2%	-5%
3600	-13%	-13%	-12%	-11%	-11%	-12%	-12%	-13%	-13%	-13%	-11%	-9%	-5%	4%	13%	11%	8%	7%	8%	6%	0%
5400	-15%	-15%	-13%	-12%	-12%	-12%	-12%	-13%	-12%	-12%	-10%	-7%	-2%	5%	13%	11%	10%	9%	9%	7%	1%
7200	-13%	-13%	-12%	-10%	-10%	-10%	-10%	-10%	-10%	-9%	-8%	-6%	-1%	5%	11%	10%	9%	8%	8%	6%	1%
9000	-11%	-11%	-10%	-8%	-8%	-9%	-9%	-8%	-8%	-8%	-6%	-5%	-1%	3%	8%	7%	6%	6%	5%	4%	0%
9600	-11%	-10%	-9%	-8%	-8%	-8%	-8%	-8%	-8%	-7%	-6%	-5%	-2%	3%	7%	6%	6%	5%	4%	3%	-1%

% Error in 20GPM Tank Charging																					
Section height (in m)/Physical time (s)	0.00	0.11	0.19	0.28	0.36	0.45	0.54	0.62	0.71	0.79	0.88	0.96	1.05	1.14	1.22	1.31	1.39	1.48	1.57	1.65	1.74
0	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
600	0%	0%	0%	0%	0%	0%	-1%	-1%	-2%	-3%	-4%	-5%	-1%	8%	12%	2%	-8%	-16%	-7%	-6%	-11%
1200	-2%	-3%	-4%	-3%	-4%	-6%	-7%	-8%	-10%	-10%	-8%	-4%	0%	5%	7%	-1%	-2%	-1%	-3%	-7%	-10%
1800	-8%	-9%	-9%	-8%	-10%	-11%	-12%	-12%	-11%	-10%	-7%	-5%	-1%	5%	10%	6%	2%	0%	-2%	-4%	-8%
2400	-12%	-13%	-13%	-11%	-12%	-13%	-12%	-12%	-10%	-9%	-7%	-4%	1%	7%	10%	6%	3%	2%	0%	-3%	-6%
3000	-14%	-14%	-13%	-12%	-12%	-12%	-11%	-11%	-9%	-8%	-5%	-2%	2%	6%	9%	6%	4%	2%	1%	-2%	-5%
3600	-14%	-14%	-12%	-11%	-11%	-10%	-10%	-9%	-7%	-6%	-4%	-2%	2%	5%	7%	5%	4%	2%	1%	-2%	-5%
4200	-12%	-12%	-11%	-9%	-9%	-9%	-8%	-7%	-6%	-5%	-3%	-1%	1%	4%	6%	4%	3%	1%	0%	-2%	-5%
4800	-10%	-10%	-9%	-8%	-8%	-7%	-7%	-6%	-5%	-4%	-3%	-1%	1%	3%	4%	3%	2%	0%	-1%	-3%	-5%
5400	-9%	-9%	-8%	-7%	-7%	-7%	-6%	-6%	-5%	-4%	-3%	-2%	0%	1%	2%	1%	0%	-1%	-2%	-4%	-5%

Ambient Temperature Profiles for Selected Model Locations and Months

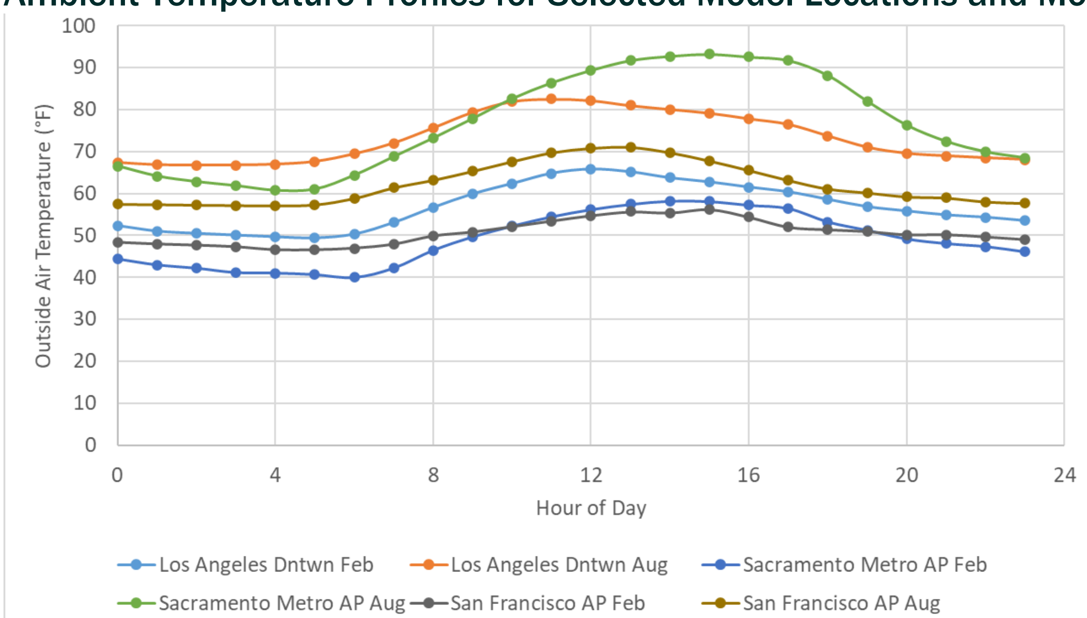

Figure 33: Outside air temperature profiles for test locations (CALMAC 2022).

Modeling Test Matrixes

Table 8: Test Matrix for 24-hour Modeling Across Varying Conditions

2,000 Gal HPWH Test Matrix									
Test Run #	Building Size/# of Occupants	City Water Temp (°F)	Recirc Losses per/apt	Draw Profile (Peak Design or Average Day)	Temperature Day	People/apt	Recirc Temp (°F)	Supply Temp (°F)	Notes
1	200	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
2	250	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
3	300	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
4	350	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
5	400 (stop if can't meet demand)	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
6	450 (stop if can't meet demand)	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
7	500 (stop if can't meet demand)	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Establish Peak Design Capacity at worst conditions
8	Max Capacity based on above results	50	60 Watts	Peak Design	Sacramento, Feb	1.8	115	125	Test Peak Conditions at Different Temperature Day
9	Max Capacity based on above results	50	60 Watts	Peak Design	San Francisco, Feb	1.8	115	125	Test Peak Conditions at Different Temperature Day
10	60% Capacity based on above results	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Test Peak Conditions at 60% Max Building Size
11	60% Capacity based on above results	50	60 Watts	Peak Design	Sacramento, Feb	1.8	115	125	Test Peak Conditions at 60% Max Building Size
12	80% Capacity based on above results	55	60 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Test Peak Conditions at 80% Max Building Size
13	80% Capacity based on above results	50	60 Watts	Peak Design	Sacramento, Feb	1.8	115	125	Test Peak Conditions at 80% Max Building Size
14	Max Capacity based on above results	55	60 Watts	Peak Design	Los Angeles, Feb	2.6	115	125	Test Peak Conditions while increasing People/apt
15	Max Capacity based on above results	50	60 Watts	Peak Design	Sacramento, Feb	2.6	115	125	Test Peak Conditions while increasing People/apt
16	Max Capacity based on above results	55	60 Watts	Avg Day	Los Angeles, Feb	1.8	115	125	Avg Day Demand Performance
17	Max Capacity based on above results	50	60 Watts	Avg Day	Sacramento, Feb	1.8	115	125	Avg Day Demand Performance
18	Max Capacity based on above results	55	60 Watts	Avg Day	Los Angeles, Feb	2.6	115	125	Test Peak Conditions while increasing People/apt
19	Max Capacity based on above results	50	60 Watts	Avg Day	Sacramento, Feb	2.6	115	125	Test Peak Conditions while increasing People/apt
20	Max Capacity based on above results	55	100 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Test at 100 Watts/Apt Recirc Losses
21	Max Capacity based on above results	50	100 Watts	Peak Design	Sacramento, Feb	1.8	115	125	Test at 100 Watts/Apt Recirc Losses
22	60% Capacity based on above results	55	100 Watts	Peak Design	Los Angeles, Feb	1.8	115	125	Test at 100 Watts/Apt Recirc Losses
23	60% Capacity based on above results	50	100 Watts	Peak Design	Sacramento, Feb	1.8	115	125	Test at 100 Watts/Apt Recirc Losses
24	Max Capacity based on above results	60	60 Watts	Peak Design	Los Angeles, Aug	1.8	115	125	August Temp Day Performance
25	Max Capacity based on above results	60	60 Watts	Peak Design	Sacramento, Aug	1.8	115	125	August Temp Day Performance
26	60% Capacity based on above results	60	60 Watts	Peak Design	Los Angeles, Aug	1.8	115	125	August Temp Day Performance
27	60% Capacity based on above results	60	60 Watts	Peak Design	Sacramento, Aug	1.8	115	125	August Temp Day Performance
28	Max Capacity based on above results	60	60 Watts	Avg Day	Los Angeles, Aug	1.8	115	125	Avg day runs in August for load shifting insight
29	Max Capacity based on above results	60	60 Watts	Avg Day	Sacramento, Aug	1.8	115	125	Avg day runs in August for load shifting insight
30	Max Capacity based on above results	55	60 Watts	Avg Day	San Francisco, Aug	1.8	115	125	Avg day runs in August for load shifting insight
31	60% Capacity based on above results	60	60 Watts	Avg Day	Los Angeles, Aug	1.8	115	125	Avg day runs in August for load shifting insight
32	60% Capacity based on above results	60	60 Watts	Avg Day	Sacramento, Aug	1.8	115	125	Avg day runs in August for load shifting insight
33	60% Capacity based on above results	55	60 Watts	Avg Day	San Francisco, Aug	1.8	115	125	Avg day runs in August for load shifting insight
34	80% Capacity based on above results	60	60 Watts	Avg Day	Los Angeles, Aug	1.8	115	125	Avg day runs in August for load shifting insight
35	80% Capacity based on above results	60	60 Watts	Avg Day	Sacramento, Aug	1.8	115	125	Avg day runs in August for load shifting insight
36	80% Capacity based on above results	55	60 Watts	Avg Day	San Francisco, Aug	1.8	115	125	Avg day runs in August for load shifting insight