Technical Evaluation of Air-to-Water Heat Pumps with Thermal Storage

Final Report

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Disclaimer

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Abbreviations and Acronyms

Acronym	Meaning
AAHP	air-to-air heat pump
AC	air conditioning
AHRI	The Air Conditioning, Heating, and Refrigeration Institute
AHU	air handling unit
ASHP	air-source heat pump
AWHP	air-to-water heat pump
BAAQMD	Bay Area Air Quality Management District
BTU	British thermal unit
CARB	California Air Resources Board
COP	coefficient of performance
CPUC	California Public Utilities Commission
DHW	domestic hot water
EER	energy efficiency ratio
GHG	greenhouse gas
HPWH	heat pump water heater
HVAC	heating, ventilation, and air conditioning
kWh	kilowatt-hour
M&V	measurement and verification
NYSERDA	New York State Energy Research and Development Authority
NOx	nitrogen oxides
PG&E	Pacific Gas and Electric

Acronym	Meaning
RMI	Rocky Mountain Institute
SCAQMD	South Coast Air Quality Management District
TES	thermal energy storage

Executive Summary

This report provides results from a market assessment and pilot study of efficient residential air-to-water heat pump technologies with load-shift capabilities through coupled thermal energy storage and supplemental air-source heat pump cooling. Results quantify the energy and cost savings, cost effectiveness, and emission reduction potential of shifting space heating loads to off-peak times, an essential strategy to alleviate grid constraints as more homes in California electrify their end uses. The market assessment is informed by a literature review of the available technologies and components, as well as publicly available results from studies conducted on packaging and controls technologies, system performance models and field testing of storage systems integrated with heating, cooling and domestic hot water heating. Development of the pilot study stemmed from the market assessment results and data gaps identified in the market. The team recruited four single family households in California to participate in a field testing of the technology during typical household operation. A commercially available heating and hot water system consisting of an air-to-water heat pump with integrated thermal energy storage, supplemented with an air-source heat pump for cooling and supplemental heating was installed in each home. Data were collected while the system operated in both a baseline mode and load-shift mode to understand differences in energy use and to compare costs based on utility time-of-use rates for peak, partial peak and off-peak times.

The market assessment identified numerous configurations, but each contain four primary components:

- <u>Heat Pumps</u> – the primary heat transfer component that allows energy from ambient conditions outside to be transferred through a medium to the interior.
- <u>Thermal Storage</u> – water or phase change materials with higher specific heat can be pre-heated during off-peak times and stored on-site in a buffer tank. The energy can then be extracted when heating demand occurs during peak times, thus reducing demand on the grid during peaks.
- <u>Controls</u> – used to enable optimization and maximization of potential for load-shifting by allowing real-time monitoring and control of charging/discharging of the thermal sink material.
- <u>Distribution</u> – this component is needed to distribute the thermal energy where it is needed within the household, either for space heating/cooling or domestic hot water needs.

Prior field studies have shown that system performance is often lower than the product ratings. Additionally, in retrofit applications, heat pumps systems appear to be more cost effective than installing new air conditioning units or expanding natural gas service for home heating. When sized appropriately, they can be used to reduce peak demand by shifting energy use to off-peak times and can potentially reduce homeowner costs, depending on the rate structure.

Key metrics evaluated from the metered data in the pilot study include system coefficients of performance, domestic hot water as percentage load, percentage peak shifted and average utility savings. The key metrics are shown in Table 1 below.

Table 1: Key Metrics

Key Metrics - Pilot Study Results	Site 1	Site 2	Site 3	Site 4
COP – Average baseline	4.26	4.51	3.08	4.24
COP – Average control	3.61	3.84	2.52	3.35
DHW as % of total load during summer	61%	14%	24%	57%
DHW as % load during winter	10%	11%	19%	26%
% Peak load shift during summer	44%	58%	66%	28%
% Peak load shift during winter	60%	17%	60%	82%
% Peak load shift – annual	56%	36%	63%	66%
Average utility cost savings*	($40.43)	($115.84)	$216.27	$322.56

*compared to measured baseline. Values in parentheses indicate cost increase.

Key Findings and Outcomes

- **Peak shift savings** are achievable for residential, water-based, thermal storage systems for combined space heating and domestic hot water loads relative to systems without thermal storage. The shift depends on the overall system load, which can vary based on building performance and occupant behavior/usage.
- **Energy savings** observed are likely attributed to domestic hot water load profile changes between baseline and control mode periods.
- A 15 percent to 21 percent decrease in coefficient of performance is the **efficiency penalty** of operating a hot water-based thermal storage system for peak-load shifting.
- **Utility cost savings and payback** can vary depending on load profile, utility rate and installation costs.

Introduction

Contemporary air-to-water heat pump technology can efficiently serve home space heating, space cooling, and domestic water heating loads. Air-to-water heat pump systems serving hydronic space heating and domestic hot water heating can shift load from peak to off-peak demand periods with thermal energy storage.

This study evaluated residential air-to-water heat pump systems serving space heating loads and domestic hot water loads with thermal energy storage and supplemental heating and cooling from an air-to-air heat pump in single family homes in California. The system that was selected to pilot is comprised of a commercially available CO_2-refrigerant air-to-water heat pump, a domestic hot water storage tank, an air handler (i.e., fan coil unit) containing both a hydronic coil and a refrigerant coil, a supplemental air-to-air heat pump for cooling and heating, and a proprietary control system.

As part of the field study, the project team performed measurement and verification to evaluate the efficiency, load shift potential, and cost-effectiveness of a water-based thermal energy storage system for space heating and domestic water heating in single family, residential buildings in California. The evaluation quantified the potential whole-building energy efficiency, peak demand reduction, and owner economics of the integrated air-to-air and air-to-water system.

The study assessed the effectiveness of the system for existing and new California single family homes in terms of the cost to owners, the cost to utilities, and public benefits. The results of the study will be used to inform utility-sponsored programs for energy efficiency and load shift or load reduction.

Background

Developing readily available and commercially attractive solutions to electrify and decarbonize existing single family homes is critical for California to meet its energy goals. The majority of California's residential space heating and domestic hot water supply is fueled by natural gas. When cooling is included, these three uses make up 56 percent of energy used in the typical California home. Simple electrification with heat pump technology is gaining adoption, however, and current solutions require two separate systems: 1) a heating, ventilation, and air conditioning (HVAC) heat pump and 2) a heat pump water heater, causing duplicative sales and installation cycles. California electrical infrastructure and carbon goals require innovative solutions to address peak demand, the impacts of high penetration solar (e.g., the duck curve), and refrigerant emissions. New combined systems that streamline system design and field installations can overcome these barriers and integrate thermal storage for load shift to periods of lower utility electric energy cost and lower greenhouse gas emission intensity of the utility grid.

The incumbent technologies combine three parallel systems to handle the hot water, space heating, and cooling. The most common technology for heating is gas furnaces. The most common technology for cooling is non-reversible heat pumps (aka air conditioners) which operate coincidentally with the summer afternoon peak electric demand on the grid. The most common technology for domestic hot water is natural gas-fired tank heaters.

This project produced a California-specific cost and benefit analysis and technology evaluation, which is a critical first step to inform utility-sponsored programs for energy efficiency, electrification, and load shift or load reduction. It also supported the market commercialization of air-to-water heat pumps with thermal energy storage through field demonstrations.

The findings from this project have the potential to accelerate the adoption of combined domestic hot water and space conditioning with integrated water-based thermal energy storage, especially for single family homes where time-of-use energy rates and demand response programs are available.

Objectives

The objectives of this study included the following:

1. Conducted a market scan to better understand the performance and market opportunity for integrated thermal energy storage and heat pump technologies with load-shifting control capabilities.
2. Assessed through pilot study the operating performance of a residential combination space heating and domestic hot water heating air-to-water heat pump integrated with thermal energy storage and a supplemental air-to-air heat pump for heating and cooling.
3. Produced actionable recommendations for California utility programs (energy efficiency, electrification, and workforce education and training) and codes and standards for:
    a. Energy savings, cost impacts and load shifting and demand reduction
    b. Program design
    c. Measure development

Methodology and Approach

Market Scan – Literature Review of Existing Technology and Prior Studies

The focus of this literature review was to understand the current knowledge of air-to-water heat pump technologies with load shifting capabilities using thermal energy storage for use in peak demand avoidance. The literature review identified a variety of single family residential air-to-water heat pump system configurations used for space conditioning and domestic hot water. Only air-to-water heat pumps with a nameplate capacity of 60,000 British thermal units (BTU) per hour (five tons) or less were assumed for single family installation. The literature review does not include multifamily or commercial air-to-water heat pump applications. A summary of relevant field installations, pilot and demonstration projects, and case studies of air-to-water heat pump systems provided insight into the real performance of different systems with thermal energy storage capabilities to help identify applications where load-shifting could occur. Modeled data is also mentioned but the focus of this review was on installed systems.

The research team identified studies utilizing air-to-water heat pumps for heating and heating and cooling, as well as studies focused on heating, cooling and domestic hot water. Those without the capability for thermal storage are mentioned but excluded from further investigation for this study. A unique demonstration of hot water storage involving solar panels, a biomass boiler system, and a large underground buffer tank to heat the water during the summer for use in the winter is also mentioned, but the thermal energy storage applications in this study focused on daily peak reductions rather than seasonal peak reductions. Opportunities and challenges within the design process, installation and operation of the various projects are outlined in the overview of findings.

Pilot Study

Recruitment of Field Test Homes

The recruitment of field test homes was based on the project team's knowledge of the compatibility of the installed system with existing home HVAC systems, general customer interest in thermal storage, available incentives for homeowners, and the project team's criteria for this pilot study. The criteria for pilot study were single family homes with a design heating load of 25 to 30 kBtu/h with indoor space to accommodate an air handler and hot water storage, and outdoor space for outdoor heat pump units. Depending on the thermal performance of the building envelope, homes sizes were in the range of 1,000 to 2,500 square feet. Manual J load calculations, the national standard for producing HVAC equipment sizing, were performed to ensure heating loads are appropriate. Household occupants were not away from the home for more than five days during the baseline monitoring and verification periods to ensure adequate system loads. The team utilized social media outreach and direct marketing to drive participant enrollment. Participants have allowed installation of monitoring and verification devices, have allowed household data network connection for monitoring and verification data upload, and have allowed access to the smart thermostat setpoint and sensor data of space temperature and relative humidity.

Recruitment and Training of Installation Contractors

The selection of field test homes included consideration of local contractors capable of effectively installing the control system and retrofitting existing HVAC and domestic hot water systems in a single family home. Installing contractors were hired directly by the homeowners and included in their scope of work the installation of both the control system and the monitoring and verification metering equipment. Monitoring and verification metering equipment included both hardware onboard the control system as well as equipment furnished by the project team. Training of installation contractors was conducted by the project team and included live webinar training for estimating the costs of installation and for executing the installation.

Stakeholder Engagement

The project team conducted outreach to experts in thermal energy storage markets, technology deployment, and utility programs to gain insight on current opportunities and barriers. Lawrence Berkeley National Laboratory (LBNL) was contacted to understand parallel research efforts with integrated thermal energy storage systems in California and beyond. The project team continued to engage with LBNL throughout the course of the project to both collect and provide research findings.

Additionally, the team engaged with Peninsula Clean Energy (PCE), a Californica-based clean choice aggregator (CCA) implementing a host of incentive programs for customers, who shared insights regarding a similar field study they conducted, including obstacles their team had encountered. The project team later contacted PCE to discuss opportunities that could stem from a potential measure package.

Technical Evaluation and Reporting

The technical evaluation and reporting followed a monitoring and verification plan, consistent with the requirements of the International Performance Measurement and Verification Protocol Option B: Retrofit Isolation, All Parameter Measurement (Efficiency Valuation Organization 2022). The evaluation used a Normalized Energy Savings approach to account for weather effects on the load and energy savings during the evaluation period. The project team deployed metering devices for measurement of loads and energy consumption, downloaded energy data remotely from metering devices, and performed calculations of the energy efficiency and load shift of the thermal storage system. The project team provided recommendations to efficiency program administrators for quantifying the performance of residential scale thermal storage systems and provided recommendations for maximizing the benefits of thermal storage systems at scale in the California marketplace. This included recommendations on incentives and technical support for such systems.

M&V OBJECTIVES

The objectives of the measurement and verification of the pilot study installations were:

1. Quantify the loads for domestic hot water heating, space heating, and space cooling.
2. Verify heating and cooling set points are satisfied by the system.
3. Quantify the electric energy consumption of the system.
4. Quantify the energy savings and efficiency of the thermal energy storage system.
5. Quantify the energy cost savings of shifting load from the utility peak demand period.
6. Evaluate the efficiency gain of supplemental heating and cooling via air-source heat pumps.

DATA COLLECTION

Data was collected as described in the monitoring and verification plan during a period spanning the heating (winter), cooling (summer), and shoulder (spring) seasons. During the data collection period, the system operation was switched back-and-forth between baseline and control modes. The length of time spent in each mode varied throughout the data collection period and between sites, as seen in Table 2, Table 3, Table 4, and Table 5. Approximately five months of data (late December or January through early May) were collected in heating mode at each site except for Site 1, which utilizes four months of heating data (January through April). Approximately three months of data were collected in cooling mode at each site (late May through late August) except for Site 1, which utilized data points from four months (May through August).

Prior to analysis, data collected from the various data streams were merged, cleaned, and formatted. Data collected during reported unoccupied time periods were removed from the dataset. Additionally, the team removed data exceeding established expected voltage tolerances and thermostat setpoint thresholds. The expected voltage was 120V with a +/- 10V tolerance and the minimum expected heating setpoint was 60°F. Other anomalies in the data were found due to system outages and these data were also removed. Time periods of data removed spanned from one minute to multiple days. In one case, Site 1 was unoccupied for nearly 64 continuous days during the data collection period leading to substantial datapoint omissions from the final dataset. Table 2,

Table 3, Table 4, and Table 5 show the mode switching schedule for each site. The number of days of data used for modeling annual energy use from both baseline and control modes are presented in Table 6.

Table 2: Summary of Mode Switching at Site 1

Start Date	End Date	Mode	Total Days	Days Omitted	Days Captured in Analysis
12/14/2023	12/18/2023	Control	4	4	0
12/19/2023	1/2/2024	Baseline	14	14	0
1/3/2024	1/16/2024	Control	13	12	1
1/17/2024	2/1/2024	Baseline	15	0	15
2/2/2024	2/15/2024	Control	13	3	10
2/16/2024	2/29/2024	Baseline	13	4	9
3/1/2024	3/18/2024	Control	17	0	17
3/19/2024	4/18/2024	Baseline	30	6	24
4/19/2024	5/17/2024	Control	28	4	24
5/18/2024	6/24/2024	Baseline	37	15	22
6/25/2024	7/16/2024	Control	21	21	0
7/17/2024	8/15/2024	Baseline	29	27	2

Table 3: Summary of Mode Switching at Site 2

Start Date	End Date	Mode	Total Days	Days Omitted	Days Captured in Analysis
1/19/2024	1/31/2024	Control	12	0	12
2/1/2024	2/15/2024	Control	14	0	14
2/16/2024	3/1/2024	Baseline	14	1	13

Start Date	End Date	Mode	Total Days	Days Omitted	Days Captured in Analysis
3/2/2024	3/18/2024	Baseline	16	0	16
3/19/2024	4/18/2024	Control	30	0	30
4/19/2024	5/17/2024	Baseline	28	14	14
5/18/2024	6/24/2024	Control	37	2	35
6/25/2024	7/16/2024	Baseline	21	6	15
7/17/2024	8/15/2024	Control	29	0	29

Table 4: Summary of Mode Switching at Site 3

Start Date	End Date	Mode	Total Days	Days Omitted	Days Captured in Analysis
1/10/2024	1/24/2024	Control	14	0	14
1/25/2024	2/7/2024	Control	13	0	13
2/8/2024	2/21/2024	Baseline	13	0	13
2/22/2024	3/6/2024	Baseline	13	0	13
3/7/2024	3/18/2024	Control	11	0	11
3/19/2024	4/18/2024	Baseline	30	0	30
4/19/2024	5/17/2024	Control	28	0	28
5/18/2024	6/24/2024	Baseline	37	9	28
6/25/2024	7/16/2024	Control	21	0	21
7/17/2024	8/15/2024	Baseline	29	0	29

Table 5: Summary of Mode Switching at Site 4

Start Date	End Date	Mode	Total Days	Days Omitted	Days Captured in Analysis
1/20/2024	2/21/2024	Control	32	2	30
2/22/2024	3/6/2024	Control	13	0	13
3/7/2024	3/18/2024	Baseline	11	0	11
3/19/2024	4/18/2024	Baseline	30	0	30
4/19/2024	5/17/2024	Control	28	0	28
5/18/2024	6/24/2024	Baseline	37	1	36
6/25/2024	7/16/2024	Control	21	0	21
7/17/2024	8/15/2024	Baseline	29	0	29

The number of days of data utilized in each operation mode to develop the annual energy use models for each of the sites is shown in Table 6. The number of days representing control mode and cooling for Site 1 were significantly reduced by the extended period of unoccupancy. Although this negatively impacts the uncertainty of this mode for this site, cooling operation is less significant to the thermal storage performance than is space heating.

Table 6: Number of Days of Data Utilized in Each Operational Mode to Develop Annual Energy Use Models

	Site 1	Site 2	Site 3	Site 4
Baseline heating (days used)	51	41	63	60
Control heating (days used)	36	94	61	29
Baseline cooling (days used)	13	14	20	26
Control cooling (days used)	3	24	23	24

Hourly weather data for the sites corresponding with the metering periods was downloaded from OpenWeatherMap. The typical meteorological year (TMY) hourly weather data used for annualizing the performance estimates was sourced from the California Measurement Advisory Council

(CALMAC). The linear regressions were applied to the CZ2022 weather files nearest to each of the four sites (California Weather Files 2024).

ELECTRIC ENERGY CONSUMPTION

For each pilot installation, the electric energy consumption of the system during the reporting period and baseline periods was the sum of the component metered electric energy, as shown below in Equation 1. The total energy consumption was annualized, and weather normalized by developing multi-variable linear regression models with independent variables for outdoor air temperature.

Equation 1

$$E_T = E_{AWHP} + E_{AAHP} + E_{fan}$$

where:

$E_T = Total\ electric\ energy\ [kWh]$

$E_{AWHP} = Air\ to\ water\ heat\ pump\ electric\ energy\ [kWh]$

$E_{AAHP} = Air\ to\ air\ heat\ pump\ electric\ energy\ [kWh]$

$E_{fan} = Fan\ electric\ energy\ [kWh]$

The metered fan electric energy also included the energy for low voltage controls of the air handler.

A multi-variable linear regression model was developed using independent measured variables of daily outdoor air temperature and daily consumption of domestic hot water, as shown in Equation 2.

Equation 2

$$E_{daily-pred} = b_0 + b_1(T_h - T_{daily})^+ + b_2(T_{daily} - T_c)^+ + b_3(gal_{dhw})$$

where:

$E_{daily-pred} = Daily\ predicted\ total\ electric\ energy\ [kWh/day]$

$T_{daily} = Daily\ average\ outdoor\ air\ dry - bulb\ temperature\ [°F]$

$T_h = Space\ heating\ changepoint\ temperature\ [°F]$

$T_c = Space\ cooling\ changepoint\ temperature\ [°F]$

$gal_{dhw} = DHW\ daily\ volume\ at\ fixed\ temperature\ set\ point\ [gallons]$

$b_0 = Linear\ regression\ coefficient, constant\ [kWh/day]$

$b_1, b_2 = Linear\ regression\ coefficients\ [kWh/°F - day]$

$b_3 = Linear\ regression\ coefficient\ [kWh/gal - day]$

The linear regression model was developed for the baseline periods with thermal storage and discharge deactivated, and for reporting periods with the thermal storage active. The predicted consumption without thermal storage was compared with the predicted consumption with thermal storage to calculate the energy savings or losses.

THERMAL LOADS

The thermal loads of the system include domestic hot water heating, space heating, and space cooling. Load values are used for the periods when the set points are satisfied for space temperature and domestic hot water temperature.

Domestic Hot Water Heating Load

The domestic hot water heating load is the heating for hot water delivered to the plumbing fixtures (see Equation 3 and Equation 4).

Equation 3

$$\dot{Q}_{DHW} = 500 * (T_{DHW} - T_{DW}) * \dot{V}_{DHW}$$

where: $\dot{Q}_{DHW} = Domestic\ hot\ water\ heating\ load, instantaneous\ [BTU/h]$

$500 = water\ density\ and\ unit\ conversion\ factor\ [\frac{BTU-minute}{deg.\ F-gallon-hour}]$

$T_{DHW} = Domestic\ hot\ water\ temperature\ [°F]$

$T_{DW} = Domestic\ water\ supply\ temperature\ [°F]$

$\dot{V}_{DHW} = Domestic\ hot\ water\ flowrate\ [gallons\ per\ minute]$

Equation 4

$$Q_{DHW} = \sum_{t=0}^{T} [\dot{Q}_{DHW_t} * t]$$

where: $Q_{DHW} = Domestic\ hot\ water\ heating\ load\ [BTUs]$

$\dot{Q}_{DHW_t} = Domestic\ hot\ water\ heating\ load, instantaneous\ at\ time\ t\ [BTU/h]$

$\mathrm{t} = Time\ step\ of\ instantaneous\ load\ [hours]$

$T = Duration\ of\ load\ [hours]$

Space Heating Load

The space heating load is the sum of the space heating provided by the air-to-water heat pump and air-to-air heat pump via the air handler (see Equation 5 and Equation 7).

Equation 5

$$\dot{Q}_{heat} = 1.08 * (T_{SA} - T_{RA}) * \dot{V}_{AHU}$$

where: $\dot{Q}_{heat} = Space\ air\ heating\ load, instantaneous\ [BTU/h]$

$1.08 = air\ density\ and\ unit\ conversion\ factor\ [\frac{BTU-minute}{deg.\ F-ft^3-hour}]$

$T_{SA} = Supply\ air\ temperature\ [°F]$

$$T_{RA} = Return\ air\ temperature\ [°F]$$

$$\dot{V}_{AHU} = Air\ handler\ unit\ air\ flowrate\ [cubic\ feet\ per\ minute]$$

The air temperatures were measured in the supply air duct and return air duct. The air handler unit air flowrate was estimated from the metered fan motor CT amps, the metered external static pressure of the fan, and the manufacturer's fan curve data for external static pressure vs. airflow. The measured static air pressure coincident with the maximum measured motor amps was plotted on the fan curve for the maximum airflow at full fan speed. The airflow at speeds less than full speed was interpolated along an approximate system curve line defined by the fan affinity laws (see Equation 6).

Equation 6

$$\dot{V}_{AHU} = \dot{V}_{max} * \sqrt{\frac{ESP}{ESP_{max}}}$$

where: $\dot{V}_{max} = Air\ handler\ unit\ maximum\ air\ flowrate\ [cubic\ feet\ per\ minute]$

$$ESP = External\ static\ pressure\ [in.wg]$$

$$ESP_{max} = External\ static\ maximum\ pressure\ [in.wg]$$

Equation 7

$$Q_{heat} = \sum_{t=0}^{T} [\dot{Q}_{heat_t} * t]$$

where: $Q_{heat} = Space\ air\ heating\ load\ [BTUs]$

$$\dot{Q}_{heat_t} = Space\ air\ heating\ load, instantaneous\ at\ time\ t\ [BTU/h]$$

Space Cooling Load

The space cooling load is the cooling provided by the air-to-air heat pump via the air handler (see Equation 8 and Equation 9).

Equation 8

$$\dot{Q}_{cool} = 4.5 * (h_{RA} - h_{SA}) * \dot{V}_{AHU}$$

where: $\dot{Q}_{cool} = Space\ air\ cooling\ load, instantaneous\ [BTU/h]$

$$4.5 = air\ density\ and\ unit\ conversion\ factor\ [\frac{lb-minute}{ft^3-hour}]$$

$$h_{SA} = Supply\ air\ enthalpy\ [BTU/lb]$$

$$h_{RA} = Return\ air\ enthalpy\ [BTU/lb]$$

The supply air enthalpy and return air enthalpy were estimated from the measured supply air and return air temperatures and relative humidities based on the psychrometric properties of air.

Equation 9

$$Q_{cool} = \sum_{t=0}^{T} [\dot{Q}_{cool_t} * t]$$

where: $Q_{cool} = Space\ cooling\ load\ [BTUs]$

$\dot{Q}_{cool_t} = Space\ cooling\ load, instantaneous\ at\ time\ t\ [BTU/h]$

ENERGY SAVINGS AND EFFICIENCY

Heating

The heating efficiency of the system is defined by the operating coefficient of performance (see Equation 10).

Equation 10

$$COP_H = \frac{Q_{heat} + Q_{DHW}}{\left(E_{AWHP} + E_{AAHP_H} + E_{fan_H}\right) * (3412\ BTUs/kWh)}$$

where: $COP_H = Coefficient\ of\ performanc, heating\ [unitless]$

$E_{AWHP} = Air\ to\ water\ heat\ pump\ electric\ energy\ [kWh]$

$E_{AAHP_H} = Air\ to\ air\ heat\ pump\ electric\ energy\ for\ heating\ [kWh]$

$E_{fan_H} = Fan\ electric\ energy\ for\ heating\ [kWh]$

The predicted heating electric energy consumption for a baseline, non-thermal energy storage operation will be estimated from a linear regression model developed from baseline period data and applied to TMY weather data. Similarly, the predicted heating electric energy consumption for a thermal energy storage operation will be estimated from a linear regression model developed from reporting period data and applied to TMY weather data (see Equation 11).

Equation 11

$$E_{daily-pred_H} = b_0 + b_1(T_h - T_{daily})^+ + b_2(gal_{dhw})$$

where: $E_{daily-pred_H} = Daily\ predicted\ heating\ electric\ energy\ [kWh/day]$

$T_{daily} = Daily\ average\ outdoor\ air\ dry-bulb\ temperature\ [°F]$

$T_h = Space\ heating\ changepoint\ temperature\ [°F]$

$gal_{dhw} = DHW\ daily\ volume\ at\ fixed\ temperature\ set\ point\ [gallons]$

$b_0 = Linear\ regression\ coefficient, constant\ [kWh/day]$

$$b_1 = Linear\ regression\ coefficient\ [kWh/°F - day]$$

$$b_2 = Linear\ regression\ coefficient\ [kWh/gal - day]$$

The energy savings or losses are the difference between the predicted energy consumption for thermal storage operation and the predicted energy consumption of the system without thermal energy storage (see Equation 12).

Equation 12

$$E_{savings_H} = E_{daily-pred_H-baseline} - E_{daily-pred_H-reporting}$$

where: $E_{savings_H} = Heating\ energy\ savings\ [kWh/day]$

$$E_{daily-pred_H-baseline} = Baseline\ daily\ predicted\ heating\ electric\ energy\ [kWh/day]$$

$$E_{daily-pred_H-reporting} = Reporting\ daily\ predicted\ heating\ electric\ energy\ [kWh/day]$$

Energy savings or losses are attributable to the operation of thermal storage and supplemental heating.

Cooling

The cooling efficiency of the system is defined by the seasonal energy efficiency ratio (see Equation 13).

Equation 13

$$SEER = \frac{Q_{cool}}{\left(E_{AAHP_C} + E_{fan_C}\right) * \left(\frac{1000\ w}{kW}\right)}$$

where: $SEER = Seasonal\ energy\ efficiency\ ratio\ [BTUs/watt - hours]$

$$E_{AAHP_C} = Air\ to\ air\ heat\ pump\ electric\ energy\ for\ cooling\ [kWh]$$

$$E_{fan_C} = Fan\ electric\ energy\ for\ cooling\ [kWh]$$

Systems installed with economizer cooling had savings attributable to staging off the air-to-air heat pump compressors (see Equation 14 and Equation 15). The linear regressions are applied to TMY weather data.

Equation 14

$$E_{daily-pred} = b_0 + b_1(T_{daily} - T_c)^+$$

where: $E_{daily-pred} = Daily\ predicted\ total\ electric\ energy\ [kWh/day]$

$$T_{daily} = Daily\ average\ outdoor\ air\ dry-bulb\ temperature\ [°F]$$

$$T_c = Space\ cooling\ changepoint\ temperature\ [°F]$$

$$b_0 = \text{Linear regression coefficient, constant } [kWh/day]$$

$$b_1 = \text{Linear regression coefficient } [kWh/°F - day]$$

Equation 15

$$E_{savings_C} = E_{daily-pred_C-baseline} - E_{daily-pred_C-reporting}$$

where: $E_{savings_C} = \text{Cooling energy savings } [kWh/day]$

$E_{daily-pred_C-baseline} = \text{Baseline daily predicted cooling mode electric energy } [kWh/day]$

$E_{daily-pred_C-reporting} = \text{Reporting daily predicted cooling mode electric energy } [kWh/day]$

ENERGY COST SAVINGS

Energy cost savings were estimated based on the time-of-use prices from the residential utility rate structures of the pilot study participants. The energy cost savings were the difference in the predicted energy cost of the system operating without energy storage and the energy cost with thermal storage in operation.

LOAD SHIFT

The load shift of the systems was the quantity of energy saved during the peak period. The load shift was calculated as the difference between the predicted peak-period energy consumption of the system operating without controlled load shifting or energy storage used for heating and the peak-period energy consumption with thermal storage in operation.

GREENHOUSE GAS EMISSIONS SAVINGS

The greenhouse gas (GHG) emissions savings was estimated based on the time of use emission factors per unit of kilowatt-hour (kWh) for the utility electric energy supply. The GHG emissions savings was the difference between the predicted GHG emissions of the system operating without energy storage and the GHG emissions with thermal storage in operation.

Findings

Existing Systems and Technology

This section outlines available air-to-water heat pump technologies with thermal energy storage for peak demand avoidance for both heating and cooling applications. The potential technical advantages of this type of system included consolidation of heat pump equipment, reduction in the heat pump capacity required to meet loads, reduction in circuit ampacity (compared to other all-electric pathways), shifting the time of day for electric demand, and the improved coefficient of performance by shifting heat pump operation to periods of favorable ambient temperatures (RMI 2018).

Overview of System Configuration

Air-to-water heat pump systems were configured in numerous ways to meet a variety of heating and cooling loads, but each system included four main components: 1) a heat pump unit, 2) thermal storage, 3) a controller, and 4) a distribution system.

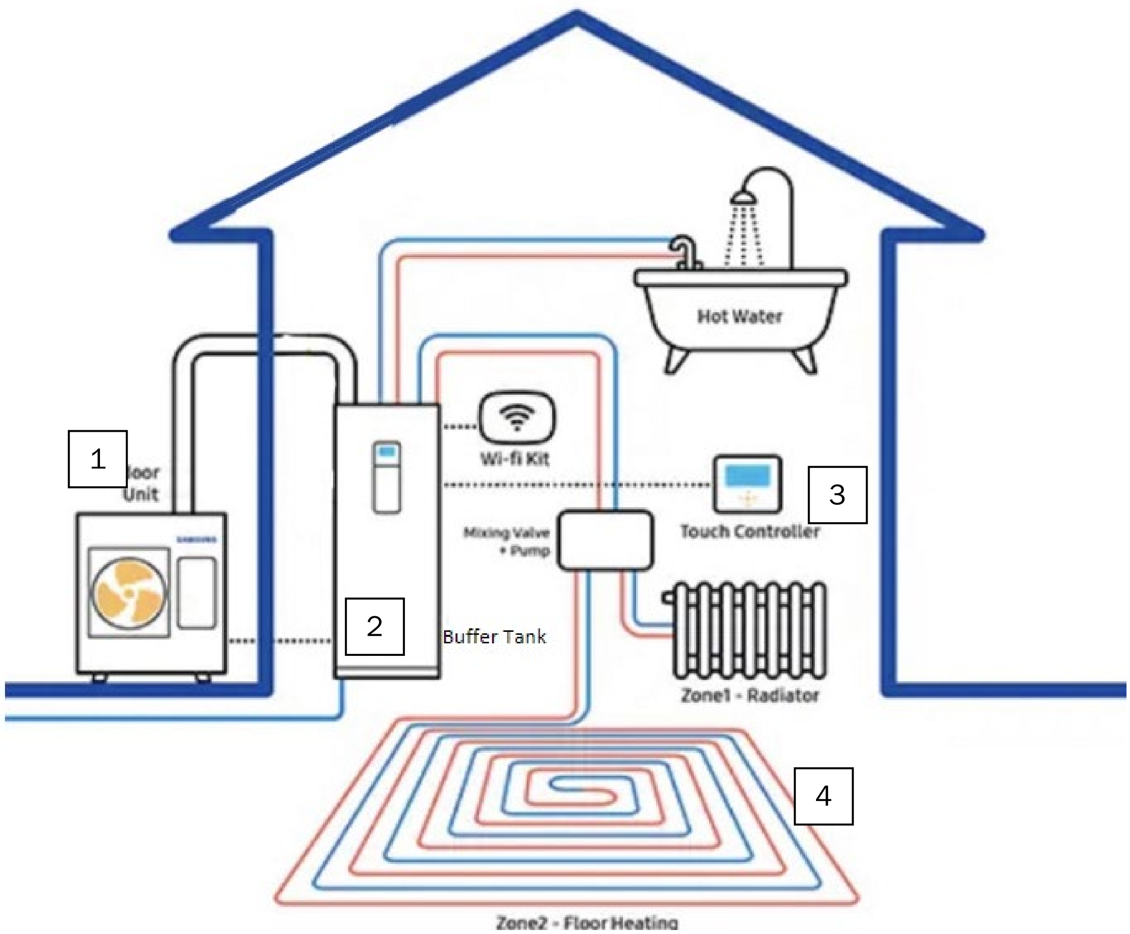


Figure 1: An example schematic of air-to-water heat pump technology modified for use in this report.

Source: (Samsung 2023)

Each system component has different options to choose from to meet the home residents' needs, the system components and their variations, as described in the sections below.

HEAT PUMPS

The heat pump component of the system transfers heat either from outside to inside (heating) or from inside to outside (cooling). The heat pump contains a compressor, finned coil, and fan for air heat exchange to the refrigerant line. The two configurations of heat pumps are monobloc and split systems. In a monobloc heat pump, the outdoor unit is a single-package system with the air and hydronic heat exchangers bundled in the housing. A hydronic line feeds to an indoor thermal storage tank and distribution system. These systems are factory charged so there is no risk of refrigerant leakage from field connections or accidental discharge (Frontier Energy 2021). The monobloc systems do not require refrigerant certified contractors for installation and are typically used in areas where freezing outdoor air temperatures are not expected, or where additional fail safes are employed, such as heat trace and drain valves, to prevent the water that circulates outside from freezing. In a split system, the outdoor unit is comprised of the air heat exchanger and the compressor, with a refrigerant line running to a hydronic heat exchanger located within the building envelope, which is further piped to the thermal storage tank and distribution system. The split system heat pump's hydronic lines would remain in the building envelope to avoid freezing temperatures. A split system requires a refrigerant certified contractor for installation.

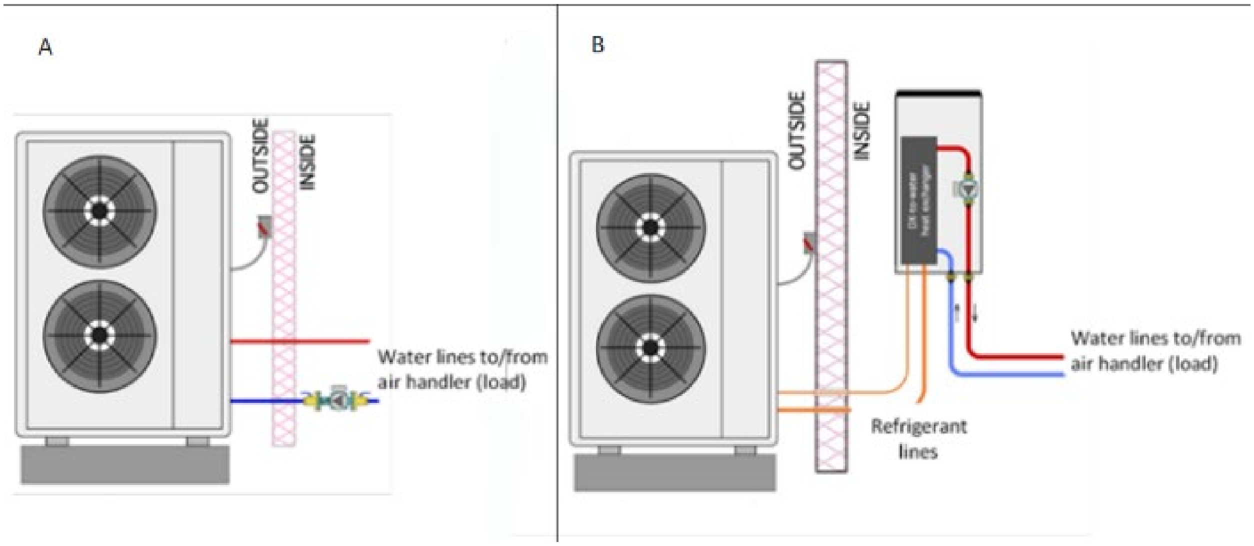


Figure 2: Monobloc (A) and split (B) air-to-water heat pump system schematics.

Source: Frontier Energy 2021.

THERMAL STORAGE

Water and phase-change materials are most commonly used for thermal energy storage in HVAC applications. Water is superior to air for energy storage and energy transfer because water has a higher specific heat capacity, greater density, and higher heat transfer coefficient than air. The thermodynamic characteristics of water make hydronic heating systems favorable for peak load shifting without impacting individual comfort. This is especially true when water can be preheated to high temperatures (e.g., 150–160°F) and stored to provide hot water to the user for many hours later, without the need to use additional energy from the grid during a peak demand period (RMI 2018). The maximum supply temperature in US residential air-to-water heat pump systems ranges from 100–150°F; European models are available with supply temperatures as high as 167°F using natural refrigerants, but as of 2023 they had not reached the US market.

The thermal storage or "buffer" tanks are used in series with the distribution system components, whereby they are directly connected at the supply or return of the distribution loop (see Figure 3). They can also be connected to both the supply and the return between the air-to-water heat pump and distribution system components to create a primary and secondary loop, decoupling the indoor and outdoor units. The latter allows for complex zoning and simultaneous space conditioning and domestic hot water production (Frontier Energy 2021).

For single family residential use, water is typically stored in interior buffer tanks ranging in size from 13 to 120 gallons, and the energy storage capacity can be increased in increments of additional water storage tanks. Additional storage is crucial to increased electrification as demand increases, and variable renewable energy production continues to grow to meet demand. Electricity markets will be more likely to experience large price differentials across seasons and times of day, including more periods of near-zero or negative wholesale pricing, and storage with load shift can help improve the economics of demand (RMI 2018). The additional storage offers the potential to reduce costs by decreasing peak demand. Because energy sources with the greatest carbon emissions are used in the highest quantities during peak demand, decreasing energy use during peaks using thermal storage allows customers to use less carbon intensive energy and to even increase the use of non-dispatchable renewable energy sources like wind and solar (RMI 2018).

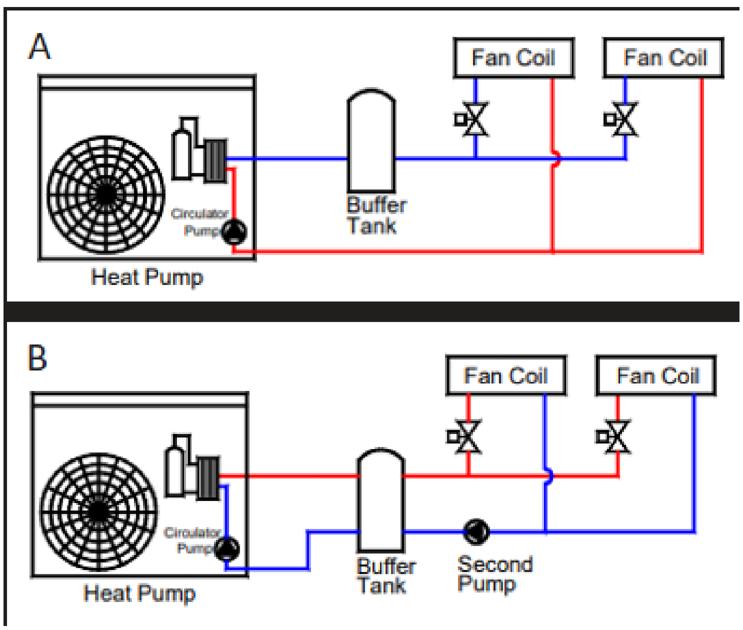


Figure 3: Buffer tank configuration: A) in-line and B) connected to both supply and return (primary and secondary loops).

Source Frontier Energy 2021.

CONTROLS

Controls help to optimize and maximize the potential of load shifting. Both passive and active thermal storage can achieve load shifting. An example of passive thermal storage involves heating the ambient indoor air above the target temperature setpoint during a non-peak time and allowing the temperature to drift during peak events or utilizing building materials as thermal heat sinks to allow for radiative heat transfer. The disadvantage is that ambient temperatures in the building are likely to vary greatly and the heat cannot be discharged in a controlled manner.

Active thermal storage is a more controlled use of the energy by storing energy as thermal heat in an insulated tank, and using controls, releasing that heat to achieve desired set points. A control system in active thermal storage allows real-time monitoring and control of charging and discharging of the thermal sink. The primary benefit of active storage is the ability to optimize the shift in peak electricity loads in a more precise way and discharge energy with greater control to reduce temperature fluctuations and maintain end user comfort (LBNL 2022).

HEATING AND COOLING DISTRIBUTION SYSTEMS

Space heating in an air-to-water heat pump system can be achieved through a hydronic radiant floor (or ceiling), low-temperature wall radiators, or in duct coils. To achieve the greatest efficiency, the heat pump is often sized to the space heating load, which does not account for domestic hot water. Therefore, some systems utilize a decoupled system to serve both heating and domestic hot water loads, in which the domestic hot water will contain back-up electric resistance heating to accommodate for times of simultaneous space heating and domestic hot water needs (Davis Energy Group 2013). Hydronic distribution to radiant flooring is best achieved with slab-on-grade construction, and thus is difficult to achieve as a retrofit, but, it is possible to retrofit a home with radiant flooring without a slab.

Most existing hydronic baseboard heating systems served by a conventional fossil fuel boiler are designed to operate with a supply water temperature above 170°F, which is higher than the output temperature of the air-to-water heat pump system. Therefore, retrofits require additional system upgrades to the emitters prior to installation. Cooling is typically achieved using ducted coils to avoid condensation build up in humid climates. In air-to-water heat pump systems coupled with domestic hot water, the domestic hot water is typically stored in buffer tanks that can be utilized for load shifting and on-demand hot water but may also be stored in an additional domestic hot water tank.

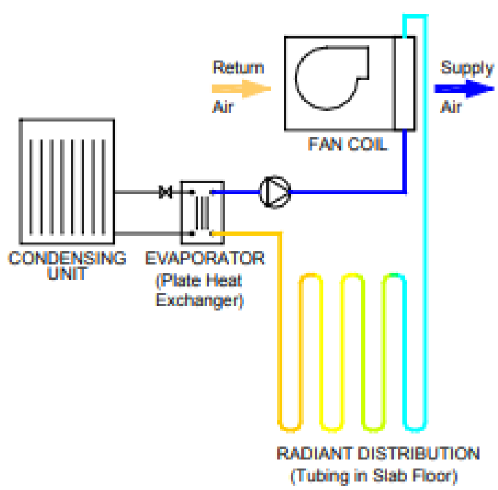


Figure 4: Schematic of air-to-water heating and cooling system with radiant floor and fan coil distribution.

Source: Davis Energy Group 2013.

Economics & Energy Modeling Outcomes

This section includes details of prior pilot projects and case studies to illustrate different configurations of air-to-water heat pump systems with thermal storage and their performance outcomes in residential applications. Both modeled and in-use systems are included in this overview.

According to a review conducted by researchers at University College Dublin, heat pump field studies conducted in the European Union have demonstrated that air-to-water heat pumps have an average seasonal performance 40 percent lower than their product ratings (O'Hegarty R. 2022). However, this review found the variation between installed performance and the manufacturers specifications was much closer. In this study, only air-to-water heat pumps with a nameplate capacity up to 60,000 Btu/h are considered for single family residential applications. As such, the air-to-water heat pump systems have a nominal coefficient of performance of as high as 4.6 at 47°F, and an energy efficiency ratio as high as 10.3 at 95°F. The coefficient of performance and energy efficiency ratio vary with ambient temperature and load. Installed performance will be discussed in the sections below.

Economics of Load Shifting in Single Family Homes

The Rocky Mountain Institute investigated the economic impacts of heating electrification in single family homes in four climate regions. The results indicated that the use of standard or flexible heat pumps for space conditioning and water heating in a retrofit application was more cost-effective than installing a new air conditioning unit and expanding natural gas service to a home for heating. However, it was still less expensive to expand natural gas for heating alongside an existing air conditioning unit. The flexible heat pump has the capability for load shifting by preheating or precooling space and water, and when load shifting is used, the energy costs are less than that of natural gas. The load shifting should be used to reduce peak demand electricity costs by utilizing high solar energy generation midday for space and water heating (RMI 2018). One strategy employed shifting most of the energy use for water heating to the nighttime, which reduced homeowner energy costs by more than two thirds due to a specific time-of-use rate structure. This study confirmed that energy costs can be optimized using air-to-water heat pumps especially with load shifting controls, and that the air-to-water heat pump systems can be cost competitive against the first cost of equipment (labor and materials) and operational costs of new natural gas installations; flexible load management of these systems should be encouraged.

Load-Shift Modeling Outcomes

The California Public Utilities Commission led a Load Shift Working Group to determine viable product designs to help shift the load away from peak periods to reach a carbon free grid. Their report illustrates interest in shifting load to maximize renewable energy use by avoiding renewable power over-generation and curtailment. It is mentioned that by 2025 in California as much as two to five percent of the daily load could be shifted, with nearly half of that attributed to shift in HVAC usage (CPUC Working Group 2019).

A 2022 study from Lawrence Berkley National Laboratory simulated the performance of an air-to-water heat pump system using phase change materials as thermal energy storage, designed to provide all heating, cooling, and domestic hot water with a single heat pump and no auxiliary resistance heating. Compared with the baseline system, researchers found that the proposed system could comfortably be downsized to 60 percent capacity, while meeting space conditioning setpoints within a max deviation of 2°C (LBNL 2022). Despite an overall increase of around 30 percent in the total annual energy (kWh) used with this system, the energy use calculated during load-shedding periods ranged from four to12 hours and was reduced by nearly 50 percent. The annual energy use is greater in this system because a higher setpoint is needed, which lowers the coefficient of performance. The decrease in energy use during load-shedding periods is advantageous because it reduces usage during high use rates and reduces load on the grid when the cost of electricity is the greatest.

A team at Frontier Energy simulated energy use in detailed energy models for residential air-to-water heat pumps coupled with cooling thermal storage using detailed data collected over several years at Pacific Gas & Electric's (PG&E) Central Valley Research Homes laboratory in, Stockton, California. The study simulated energy use across various climate zones for a minimum-efficiency air-source heat pump, an air-to-water heat pump couple with a fan coil, and an air-to-water heat pump coupled with thermal energy storage size to eliminate summer on-peak compressor operation. The air-to-water heat pump plus thermal energy storage was controlled to alternately condition indoor space or to charge the thermal energy storage tank before peak period. The air-to-water heat pump plus thermal energy storage load-shifting strategy was highly effective at moving cooling energy use to non-peak hours. For the shorter three-hour on-peak rates, 55 percent of non-storage air-to-water heat pump-cooling energy usage was shifted from the on-peak period; for the longer seven-hour peak, 66 percent was shifted (Frontier Energy 2023).

Air-To-Water Heat Pump Pilots, Case Studies, and Performance Outcomes

This section provides a thorough review of six pilots and case studies of air-to-water heat pump systems found during the literature review; all were installed in single family residential households in a variety of climate zones. Table 7 provides a summary of those projects along with the system types and important takeaways. Greater detail of the system configuration, setpoints and performance are provided in the subsequent sections. The level of detail provided by the pilots or case studies varied greatly, and for awareness, the last project description is for a future project in the planning process in Wisconsin. The studies did not provide information about payback time. This may be attributed to the varying utility and project costs.

Table 7: Summary of Pilot Projects and Key Takeaways

Entity/Project	System Type	Important Notes/Takeaways
Stash Study (Stash Energy 2023)	Heat Pump, thermal storage heat pump, and thermal storage heat pump with solar photovoltaic	The incorporation of thermal storage can provide energy and cost savings through the ability to load shift away from higher electricity rates. This benefits the customer and the grid, by reducing electricity costs and reducing GHG emissions associated peak electricity use.
Cold Climate Housing Research Center: Thermal Storage Demonstration (CCHRC 2013)	Solar thermal panels with a biomass boiler system and buffer tank	The potential for thermal storage in cold climates is unknown to homeowners and was successful in Arkansas. Thermal storage systems can be designed to hold heat for hours to a season and are paired well with renewables.
Massachusetts Clean Energy Center: Whole Home Installation Pilot (Mass CEC 2019)	Air-to-water heat pump (AWHP) with hydronic radiant floor heating and low temperature heating and cooling fan coils	It is important to have enough thermal storage for the system to cut down system cycling. Retrofit opportunities will likely require modifications to meet space requirements.
Center for Energy and the Environment (CEE) Minnesota: Cold Climate Case Study (MN CEE 2023)	Split system	The greatest heat loss experienced by the system was small compared to the total heating load.
	Monobloc AWHP with a ducted coil	There were limitations of existing ductwork that resulted in changing the cooling coil temperature from 50°F to 42°F. There was a low daily coefficient of performance (COP) because more than half of the run cycles were short cycles.
	Monobloc AWHP with a ducted coil and DHW	A significant decrease in power consumption of the domestic hot water (DHW) system occurred after the addition of the AWHP DHW system. Short cycling was very frequent.

Entity/Project	System Type	Important Notes/Takeaways
New York State Energy Research and Development Authority : Combined Heating/Cooling Demonstration (NYSERDA 2019)	AWHP for space heating and DHW with a buffer tank	Noted there were very frequent defrosts of the system. Combination system heat pump had a COP of 1.5–1.9 in the winter and 3+ for water heating
UC Davis: AWHP with space heating, cooling and DHW & AWHPs with Radiant Deliver in Low-Load Homes (Davis Energy Group 2013)	AWHP for space heating and cooling with DHW and radiant flooring	Heating performance aligned with the manufacturer, cooling performance was close to manufacturer (small sample), and the heat pump was unable to meet the storage tank setpoint of 130°F for DHW. Additional research on these systems includes modeling performance in different climate zones.

Stash Study

Stash Energy studied the energy use in four nearly identical homes using four different heating system configurations. These configurations included a control system with original electric baseboard heaters, an air-to-air heat pump, a Stash Energy thermal storage heat pump, and a Stash Energy thermal storage heat pump and solar photovoltaic. The Stash Energy thermal storage heat pumps are designed to operate when electricity prices are low to store heat for utilization during time-of-day peak periods when prices are higher. The system included a two-ton outdoor unit that is connected to an indoor thermal storage radiator housing a phase change material, heat exchanger, and fan to provide heating and cooling options. The study, performed in collaboration with various entities in Nova Scotia, reinforced the knowledge that heat pumps reduce energy use to provide cost saving to the customer compared to resistive heating, but also demonstrated that the incorporation of thermal storage can provide enhanced energy and cost savings by shifting time of use in areas where time-of-use and peak rates apply (Groszko 2021). While quantified energy savings from the study were not published, the system specifications listed a nominal load shifting capability of two kW, with a discharge time of four hours at 100 percent output (Stash Energy 2023).

Massachusetts Clean Energy Center: Whole Home Installation Pilot

Massachusetts Clean Energy Center has introduced a pilot program for whole-home air-source heat pump projects. There were 17 existing building retrofits. In 2019, they received the first application for air-to-water heat pumps connected to hydronic radiant floor heating and installation of low-temp heating and cooling fan coils (Mass CEC 2019). Neither the specifications of the air-to-water heat pump system nor the quantitative performance metrics were included as part of this pilot.

Cold Climate Housing Research Center: Thermal Storage Demonstration

The Cold Climate Housing Research Center demonstrated the use of seasonal thermal storage with a solar hybrid system that combined solar thermal panels with a biomass boiler system and a buffer tank. In 2013, 16 solar thermal collectors were installed along with a 25,000-gallon in-ground water tank. During the summer, the panels store heat in the tank until winter. Overall, the system was able to produce as much as 50 million Btu annually during summer for winter use, taking advantage of renewable energy during a peak production period (CCHRC 2013).

CEE Minnesota: Cold Climate Case Study

ASHRAE performed a case study in Minnesota of cold climate air-to-water heat pumps retrofitted to boiler heating systems (MN CEE 2023). Three sites with three different air-to-water heat pump systems were analyzed: a split system, a monobloc system with ducted coil and radiant in floor heat, and a monobloc system with ducted coil and domestic hot water. They all had concrete slabs and varying compressor speed capability.

The split system functioned at a maximum coefficient of performance of 2.6 at 48°F, and at temperatures below 10°F the coefficient of performance approached one. Notably, the lowest coefficient of performance still equates to greater efficiency than the electric resistance boiler-only system, and the results can be seen in Figure 5 below. The greatest heat loss was 900 btu/h, and this was a small fraction, compared with the heating load. In this case, because the system design relied on a backup boiler system to run concurrently with the air-to-water heat pump at low outdoor air temperatures when needed, the air-to-water heat pump was not sized to fully support the heating load at low outdoor air temperatures and therefore short cycled (that is, turned on and off too often) in order to meet the load if the boiler did not kick on.

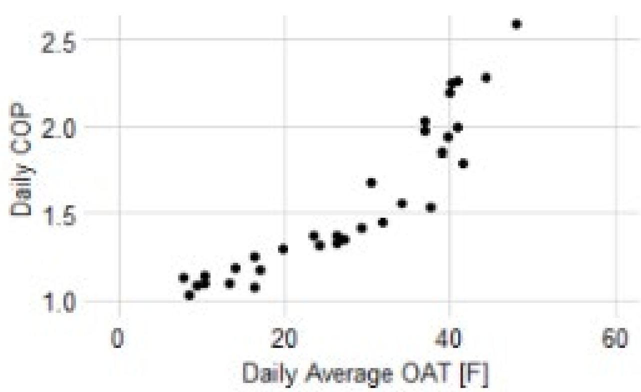


Figure 5: Daily coefficient of performance for heating over a range of outdoor air temperatures.

Source: MN CEE 2023.

The monobloc system serving a ducted hydronic coil and radiant in floor heat had a target coil temperature of 42°F. However, the existing ductwork at the site limited the target coil temperature. The average daily cooling coefficient of performance for this system was 2.5 when the outdoor air temperature was above 70°F. The values are displayed as negative to illustrate cooling. The unit's ability to cool was not strongly influenced by the outdoor air temperature and can be seen in Figure 6 below.

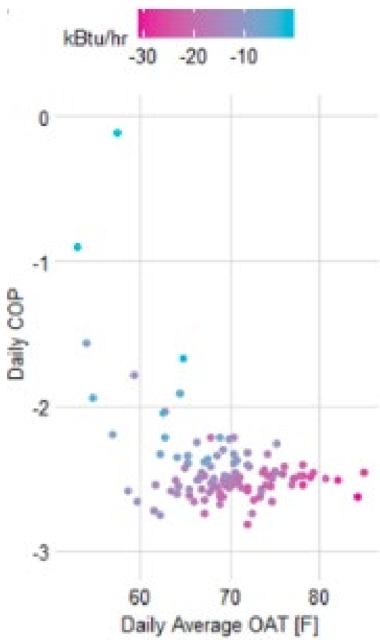


Figure 6: Daily coefficient of performance of cooling over a range of outdoor air temperatures.

Source: MN CEE 2023.

The monobloc serving a ducted coil and domestic hot water system was the most similar to the project system planned for use in this project. The system had daily heating coefficient of performances ranging from 1.5 to 3.4, and the cooling coefficients of performance were not strongly influenced by the outdoor air temperature. The domestic hot water component of the system measured coefficient of performances approaching five. The performance of the system can be seen below in Figure 7.

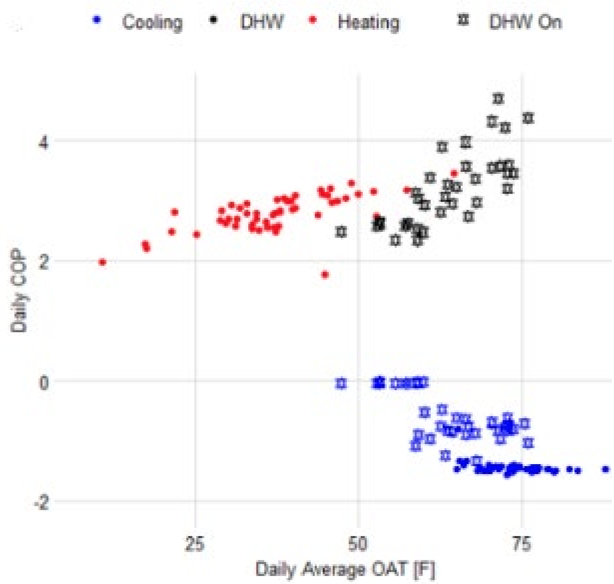


Figure 7: Daily coefficient of performance of cooling over a range of outdoor air temperatures for heating cooling and domestic hot water.

All systems experienced short cycling, which the authors suggest could be reduced with integrated sensors to prevent the compressors from ramping up and down. Overall, all test sites demonstrated an improvement of efficiency, compared with an electric resistance boiler system, but the measured coefficient of performance varied drastically between test sites and outdoor air temperatures. The cooling coefficient of performance also varied significantly between the two sites outfitted with hydronic A-coils and did not reach the level of efficiency comparable with a 13 SEER air conditioning unit.

NYSERDA: Combined Heating/Cooling Demonstration

NYSERDA examined 15 different air-to-water heat pump units and installed two types coupled with a buffer tank for testing across five sites. One air-to-water heat pump unit chosen was designed for both space heating and domestic hot water, and the product required that heating be combined with domestic hot water, but that space heating could not occur at outdoor air temperatures below 27°F. Additionally, when installed in this pilot each system included the use of controls. The systems available in this study did not operate at return water temperatures above 130°F.

It was found that the seasonal coefficient of performance for the units was on average lower than the available published data from the manufacturer. The space heating coefficient of performance ranged from 1.5 to 1.9 in the winter, while the specifications listed a coefficient of performance of 1.5 to three from temperatures ranging from 10 to 40°F for heating. See Figure 8 for the full range of coefficient of performances from the manufacturer versus the measured values.

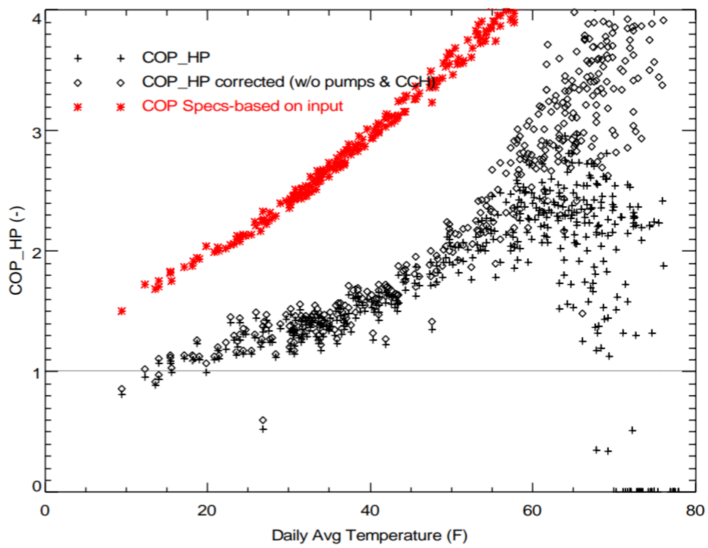


Figure 8: Rated coefficient of performance from specs vs. actual measured coefficient of performance at various daily average termpatures.

The actual coefficient of performance for the water heating was greater than three, but still less than the specification coefficient of performance of 4.5 (Taitem Engineering, Frontier Energy 2019). Overall, the highest efficiencies are achieved with lower return water temperatures, constant loads, and limited cycling, which explains the need both for a buffer tank and for the use of controls to trigger the backup heating source.

UC Davis Study: Air-To-Water Heat Pump with Space Heating, Space Cooling and Domestic Hot Water

A 2013 study by the Alliance for Residential Building Innovation evaluated air-to-water heat pump performance of two different mixed-mode distribution systems within two single family, one-story homes over a one-year period. The systems were modeled to evaluate how the systems would function in different climate zones. Both systems utilized the air-to-water heat pump for space heating and cooling and radiant floor for additional heating. One of the systems also utilized a fan coil for heating.

The “Tucson house” utilized a SEER 13 air-to-water heat pump heat pump for space heating and cooling only with a refrigerant-to-water heat exchanger and a 30-gallon buffer tank to prevent short cycling when load was low. The “Chico house” utilized an inverter-driven three-function air-to-water heat pump for space heating and cooling and domestic hot water. The system does not have a desuperheater. The inverter-driven compressor helps to reduce the cycling during times of variable load and the lack of a desuperheater restricts the system to heating only, cooling only or domestic hot water only at a given time.

The measured heating and cooling performance aligned closely with the rated specifications of the units. In the Tucson house, large variations were noted in the observed data for air-to-water heat pump performance, likely due to a variation in water temperature for the supply loop, caused by zone dynamics across the three zones, and by other factors causing load fluctuations. In the Chico house, the heat pump could not meet water storage-temperature setpoints and this is believed to be due to close supply and storage temperature setpoints. The heating, cooling and domestic hot water performance of the Chico house can be seen in Figure 9 below.

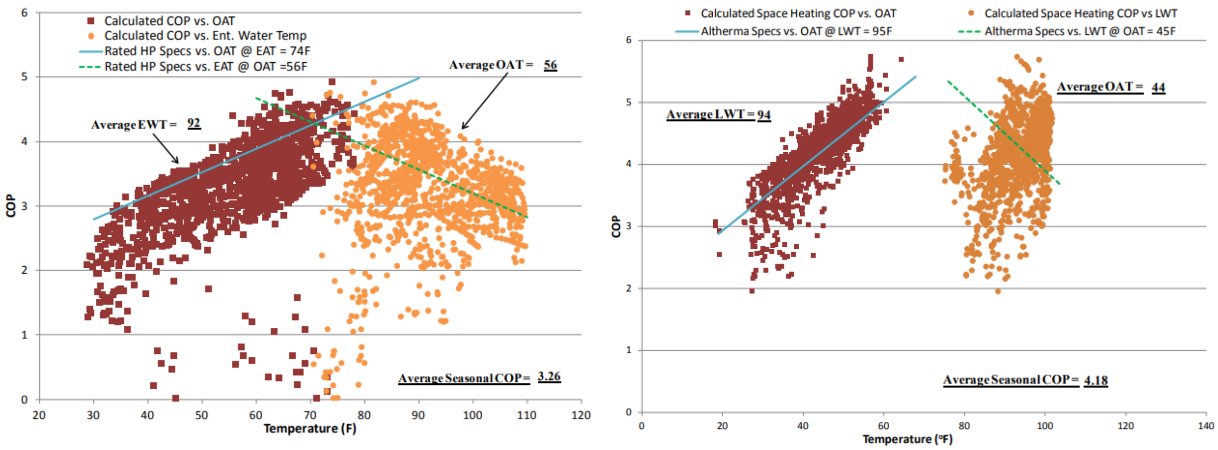


Figure 9: Calculated average full-load coefficient of performance vs OAT and LWT compared to manufacturer specs for the Tucson house (left) and the Chico house (right).

Source: Davis Energy Group 2013.

The seasonal heating coefficients of performance measured were 3.26 and 4.18, which are comparable to manufacturer specifications. The seasonal energy efficiency ratios for space cooling were 11.2 and 10.8, and the domestic hot water performance was much lower than expected with a COP of 1.63, attributed to poor heat transfer between the heat pump supply loop and the storage tank (Davis Energy Group 2013).

Wisconsin Focus on Energy: Air-To-Water Heat Pump Field Study

Wisconsin's energy efficiency program, Focus on Energy, is scoping a field study to assess air-to-water heat pump retrofits in existing homes and multifamily properties and in residential new construction. The pilot will install air-to-water heat pumps in three single family buildings and one multifamily building and seek to identify the installation barriers for air-to-water heat pumps in Wisconsin, the comfort impacts of air-to-water heat pumps, the fuel consumption and coefficient of performance curves of air-to-water heat pumps, the load shifting potential of smart air-to-water heat pump systems with storage tanks, and whether they provide sufficient hot water and space heating. The project does not currently have a contract signed but intends to launch in 2023 and complete in 2025.

Market Opportunities, Benefits and Barriers to Implementation

Market Opportunities & Benefits

Air-to-water heat pump sales are seeing growth globally, particularly in Europe where they grew by 49 percent in 2022 (Monschauer 2023). This growth rate more than doubled that of air-to-air heat pumps, with sales that grew by 19 percent in 2022 (CPUC 2022). Air-to-water heat pumps are a very small but emerging portion of California's residential HVAC marketplace. While respondents of the

CPUC 2022 Heat Pump market study acknowledged that heat pumps would simplify meeting both heating and cooling needs, “lack of awareness of heat pumps is a concern because” mainstream residential HVAC contractors install primarily gas furnaces with a split system air conditioning unit. The reason is that cooling is still considered by many to be a luxury benefit rather than a necessity (CPUC 2022). Hydronic systems featuring electrically driven HVAC equipment are currently even less prevalent, but the increased focus on all-electric strategies will be furthering this effort to some degree. Another significant driver of electrification in the coming years will be California municipalities that have adopted all-electric reach codes (Frontier Energy 2022).

With electrification at the forefront of the energy transition and the grid becoming cleaner, heat pump technologies are increasingly expected to meet residential heating, cooling and domestic hot water loads reducing fossil fuel use. According to a California Public Utilities Commission (CPUC) study, air-to-air or air-source heat pumps were just four percent of the 12.2 million space heating systems in California in 2020, but that number is expected to grow (CPUC 2022). Data from AHRI showed that furnace sales have decreased while air-source heat pump sales have shown annual growth, but air-source heat pumps are still only three percent of residential shipments of heating and cooling equipment in California (AHRI 2022). Growth in electric heating is increasing with electric heating reaching 20 percent of the stock in 2020 versus five percent in 2009. While the pace of adoption of air-source heat pumps is lagging that of conventional electric heating systems in single family residential homes, the total stock of air-source heat pumps has nearly doubled over the 10-year period (CPUC 2022).

Air-to-water heat pumps for space conditioning are much less common and no official data on the number of units installed annually in California was available. However, they have greater potential for load shifting, due to the higher thermal storage capacity of water relative to air. A case study in a Canadian residential home demonstrated that shifting electricity load to off-peak energy use periods and utilizing storage with air-to-water heat pumps could decrease peak energy use by up to 25 percent for heating and up to 45 percent for cooling (Erdemir 2022).

According to the EIA, California households use 53.7 MMBtu annually, with 27 percent attributed annually to space heating, 10.4 percent attributed to space cooling, and 30 percent attributed to water heating (Residential Energy Consumption Survey (RECS) 2020). As households continue to electrify, if 1.3 percent of residential single family homes were to install a water-based system with thermal storage and controls, a two- to five-percent load shift of HVAC loads could be realized per household (CPUC Working Group 2019). This would amount to nearly 200,000 households in California and a total of 41 to 103 GWh energy shifted annual from peak or partial peak periods to off-peak periods.

The average GHG emissions during peak and partial peak periods is 0.44 $MTCO_2e/MWh$, and during off-peak periods the average GHG emissions is 0.35 $MTCO_2e/MWh$ (CPUC 2022). The emissions savings between off-peak and peak times is 0.09 $MTCO_2e/MWh$, which could result in emissions savings of between 3,720 and 9,300 $MTCO_2e$ annually from thermal storage shifting electric demand from the grid to off-peak periods.

NEW AND UPCOMING REGULATIONS IN CALIFORNIA

The Bay Area Air Quality Management District (BAAQMD) imposed new regulations on furnaces, boilers and water heaters that emit nitrogen oxides (NOx). Only zero NOx water heaters can be sold or installed in the Bay Area beginning 2027. Zero NOx furnaces can be sold or installed beginning in 2029, and zero NOx large commercial water heaters in 2031 (BAAQMD 2023). Additionally, the California Air Resources Board (CARB) and the South Coast Air Quality Management District (SCAQMD) are also currently developing regulations to regulate NOx emissions in space and water heating. Along with these regulations proposed in its 2022 Air Quality Management Plan, SCAQMD is also developing incentives programs to help encourage the adoption of zero emission appliances (SCAQMD 2023). CARB committed to exploring and developing zero-emission GHG standards for new space and water heaters sold in California in its 2022 State Implementation Plan (CARB 2022). These new and upcoming regulations indicate the trend towards zero-emission water heating in California and indicate a large market opportunity for all types of heat pumps to fill the gap left by gas water heaters.

REDUCTION IN ENERGY USE OVER FOSSIL EQUIVALENT TECHNOLOGIES

Air-to-water heat pumps offer an opportunity for building retrofits seeking to replace or supplement existing gas, oil, or steam domestic hot water systems (Building Energy Exchange 2021). They are compatible with radiators, floor radiant, and air handlers with hydronic coils, which increases the opportunity for adoption. However, due to lower supply water temperatures, they are not typically considered a drop-in replacement for systems with conventional emitters designed for 180°F hot water. A research study conducted through Efficiency Maine installed and tested air-to-water heat pump systems plus thermal energy storage in houses in Northern Maine and demonstrated that the field-verified required source water temperature on the coldest day was under 140°F when the team had anticipated a source water temperature closer to 170°F (Gridworks Consulting 2024). The optimal system design for a house with a maximum source water temperature of 140°F results in significantly lower installation and operation costs, and saves on energy use over design temperatures of 170°F. Furthermore, when installed, air-to-water heat pumps can significantly reduce energy use, particularly for multifamily buildings, hotels, and dorms with centralized domestic hot water systems (Building Energy Exchange 2021). In the single family market, the ability to shift load can help avoid energy consumption when the grid is at peak and shift that consumption to times when over-generation of renewable energy may occur (CPUC 2022).

The Harvest Thermal pilot study conducted by Peninsula Clean Energy evaluated the energy savings of the thermal storage system relative to a baseline of gas-fired hot water heating and gas-furnace space heating (Peninsula Clean Energy 2021) (TRC 2022). The results of the study show annual cost savings of $100 to $350 in four residential installations (Peninsula Clean Energy 2024). The savings are attributable to the efficiency of the heat pumps relative to the existing gas systems and load shift from the peak-period utility rates. The savings attributable specifically to the thermal storage and the efficiency impact of load shift are not included in the analysis.

CONSOLIDATION OF APPLIANCES, DEMAND FLEXIBILITY AND PEAK REDUCTION

Air-to-water heat pump systems are compact and could consolidate air and water heating, reduce refrigerant emissions, reduce electric circuit ampacity, and increase electric demand shifting. The technology helps avoid electric distribution infrastructure upgrades which, along with its compact size, can help streamline multifamily retrofits and installations. The Berkeley National Laboratory tested the impact of heat pumps with phase-change material thermal energy storage in a multifamily residential apartment building (LBNL 2022). The study found that the technology offered a promising pathway to reducing first costs and operational costs and improving demand flexibility in cold climates.

LIFECYCLE COST REDUCTIONS

For most existing natural gas customers, air-source heat pump space heating is more expensive than gas heating systems, due to the favorable cost of natural gas and the existing infrastructure. However, the Rocky Mountain Institute (RMI) found that there are scenarios in which home heating electrification can compete with fossil systems. Those scenarios are: a) when replacing heating oil or propane systems; b) when installing heat pumps to replace natural gas-, propane- or oil-heating system replacements and air conditioning or domestic hot water replacements; c) in new construction, where additional gas service lines would increase overall infrastructure costs (RMI 2018). They also found that when operating with a time-of-use cost differential of 3:1, load shifting can produce meaningful cost savings. Coupling electrified heating systems with utility demand response programs can RMI's study highlights that lifecycle cost savings of air-to-water heat pump plus thermal energy storage can be realized when strategic electrification is employed.

Barriers to Implementation

Potential barriers to implementation included equipment sizing requirements, lack of contactor experience, high installation costs, and noise associated with heat pump technologies. In the following paragraphs summarize the details of these barriers.

UNDERSTANDING OPTIMIZATION OF HEAT PUMP SIZING

To maximize the efficiency of the system with multiple functions (heating, cooling and domestic hot water), a contractor must properly size the system to the expected or known loads. Each function has a different load requirement and can add additional nuance to sizing in these types of systems. A 2019 report by the Bonneville Power Administration on residential heat pump commissioning found that there was a 70 percent probability that a heat pump contractor would not set the auxiliary heat lockout properly, resulting in a coefficient of variation for sizing of 0.5 (SBW Consulting 2019). Auxiliary heat controls ensure that users are getting the greatest energy efficiency from a unit. If the unit is switching to backup heating, the maximum heat pump potential efficiency is not being realized. Proper sizing ensures the greatest heating efficiency, while an oversized unit results in reduced seasonal efficiencies. While this study focuses on heat pump use in air-conditioning applications, a combined heating, cooling and domestic hot water system may have additional barriers to proper sizing due to greater variations in load. An air-to-water heat pump system coupled with domestic hot water often cannot meet maximum combined load unless the air-to-water heat pump is oversized for the heating load. This results in a decreased efficiency overall, as mentioned above.

LACK OF CONTRACTOR EXPERIENCE WITH AIR-TO-WATER HEAT PUMPS

An existing residential HVAC system will often need extensive retrofit work because the systems are not currently compatible with high-temperature hydronic system replacement. Contractors may not be familiar with connecting to a buffer tank or properly sizing the distribution systems based on a low-temperature output requirement. This can result in difficulty finding contractors for regular system maintenance as well. Customer pressure on contractors for services on these systems will drive increased service in this area (Taitem Engineering, Frontier Energy 2019).

INCREASED FIRST COSTS FOR IMPLEMENTATION

Because these systems are not direct replacement solutions, the existing infrastructure design and sizing of the heating system may not be sufficient for the new system. Additional design and sizing measurements may disrupt successful implementation, due to the increased costs to right-size a new system based on the technical specs of the air-to-water heat pump and accompanying components (MN CEE 2023). If the design is not conducive to utilizing existing duct work, then the cost could increase substantially. Additionally, if there is not space to fit the addition of an air handling unit or new required ductwork, retrofitting with this technology may not be possible without additional infrastructure changes (Frontier Energy 2021).

POTENTIAL INCREASE IN NOISE ASSOCIATED WITH INSTALLATION LOCATION

Placement of the unit at the home is important to consider. If the installation location available is in proximity to bedrooms and if the house is not well insulated from noise, noise from compressor cycling could adversely affect the homeowner and they may be less interested in replacing their existing system (Davis Energy Group 2013).

Pilot Study System Description

Equipment and System Arrangement

The installed systems consisted of five primary devices which work together to deliver hot water: a high-efficiency air-to-water heat pump, a water tank for thermal storage, a control device, an air handler, and an air conditioning condenser.

The control device used integrates the entire system and facilitates the storage and delivery of hot water. It optimizes system efficiency by operating heat pump during the cheapest off-peak times, while ensuring that users' heat and hot water needs are always met. It is either wall mounted, flat above the tank, or on a shelf. It is connected to the heat pump's signal box via a cat5 cable and to the heat pump's power box via a current transducer, which also connects to the cat5.

The heat pump installed allows the system to function by heating water to 150°F.

The air handler installed can be located horizontally in a basement, crawl space or attic, or vertically in a garage or closet with appropriate mounting and bracing. When hot water is used in the home, the control device directs water through the air handler for air heating. Heat is transferred to air through the air handler's heat exchanging coil.

Water tanks can be located indoors in closets, garages, basements, or in an outdoor closet. The tank stores water at 150°F to allow the system to deliver higher heat than the heat pump's instantaneous capacity.

The air conditioning condenser is an outdoor fully modulating reversible air conditioning heat pump unit. With the outdoor air conditioning unit, an evaporator coil module is added to the air handler.

A/C Add On

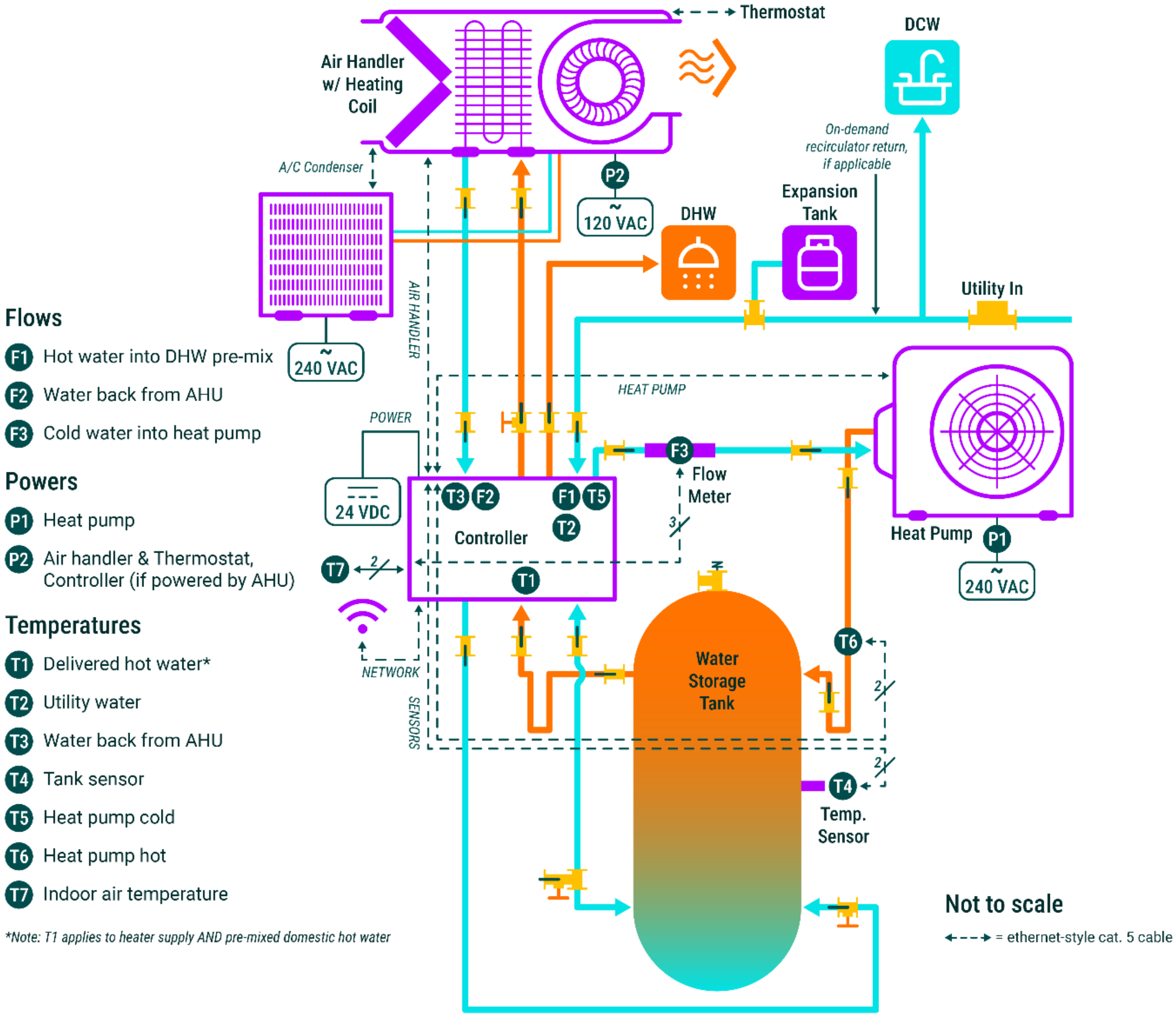


Figure 10: Diagram of installed system with supplemental heating and cooling air-to-air heat pump.

Source: Project Team.

INSTALLATION

Air-to-water heat pump systems were installed in four households. The site characteristics are shown below. Each installation site is in the San Francisco Bay Area which is California Climate Zone 3. The clustering of the sites in this area is a result of the recruitment process and the higher availability of contractors experienced with installing this thermal energy storage product, relative to other regions in the state.

Table 8: Installation Site Characteristics

	Site 1	Site 2	Site 3	Site 4
Locality	East Palo Alto	San Carlos	Millbrae	Piedmont
Floor area (square feet)	2,190	1,930	1,670	1,418
Home type	4 bd, 3 ba	3 bd, 2 ba	3 bd, 2 ba	2 bd, 2 ba
Average number of occupants In next 12 months	4 occupants: 2 adults, 2 children	3 occupants: 2 adults + 1 child	3 occupants: 3 adults	3 occupants: 2 adults + 1 child
Year built or last whole house renovation	2000	1965	1959	1960
Design heating load (kBtu/h)	29	29	30	23
AHU location	Attic	Garage / utility room	Crawlspace	Crawlspace
AHU orientation	Horizontal	Vertical	Horizontal	Horizontal
Economizer installed	yes	no	yes	yes
Number of HVAC zones	2	1	1	1
Average/max/min heating setpoint (°F)	68 / 69.5 / 66.2	68 / 72 / 62.1	67 / 74.5 / 64	67 / 72.5 / 62
Average/max/min cooling setpoint (°F)	76 / 76 / 76	72 / 80 / 68	73 / 82 / 68	75 / 78 / 65

The project team opted to install the system over five days to minimize hot water and heating downtime for occupied homes. Local building permits for water heater installation or replacement and for mechanical system installation or replacement were required for the mechanical contractor's HVAC and water heater retrofits.

The heat pump was installed on the first day, outdoors on either a pad or hung on an exterior wall with heavy duty L brackets. When hung on a wall, rubber risers were used to avoid sound transmission through the wall. When located on a pad, the heat pump was set on pressure-treated lumber, prefabricated pump or heat pump risers on top of the pad. There were at least three inches underneath for the drain plug. The heat pump was located as close as possible to the tank but must be within 66 feet of pipe length and 23 feet of vertical separation. The heat pump was installed on the east, southeast, or south side of the house, if possible, for best coefficient of performance. Pipes under the heat pump cover were insulated with a half-inch foam sleeve and foam tape. The pipes were clamped on the outside of a minimum of one inch of insulation. A current transducer was installed in the heat pump's power box around a live wire and fished to the signal box. The heat pump had a clearance of six inches behind, two feet in front, one foot to the right (waterside) and six inches to the left.

The control device was installed on the first day with the heat pump. The control device was hung near the hot water tank's location, ideally between the tank and the house domestic hot water pipe. It was also flat-mounted and secured on top of the tank, however, this required the tank to be installed first, and the installer must ensure there is two feet of service access above the control device. If the control device is hung above the tank, the minimum clearance above the control device is five inches, while the bottom of the control device can be level with the tank's insulation, so long as there is enough space for pipes to drop next to the tank.

On day two, the existing storage tank was removed, and the new tank was installed. The new tank was placed on a high-density foam or cork insulation pad that is a minimum of one-inch thick. The tank was wrapped in R-8+ insulation and any exposed pipes, valves, and couplings coming from the top of the tank were insulated, including the temperature and pressure valve. There was a two-to-three-inch gap around the entire tank for insulation. The tank had a temperature sensor which is connected to the control device via a two-wire cable.

The existing furnace and the installation of air handler and thermostat took place on the third day. The air handler came pre-assembled with modules connected with latches. The modules contained a minimum of a hydronic coil, a fan, and a supply and return plenum. There were options to add on a V-bank filter, and an air conditioning coil. It can be suspended from the floor using a hanging frame in a crawl space, rested on a platform in an attic, or placed vertically in a garage or closet. If placed in a garage or closet, it should allow space to open filter box door and be in a position for outside air intake. The reversible air conditioning unit was installed during this period as well. The air conditioning unit was located outdoors on a pad that is one to two inches larger than unit on all sides. The unit also required a minimum of five feet of discharge area above the unit, one foot clearance on one side and two feet on the side adjacent to the access panel. The unit was installed near bedrooms and was located to avoid precipitation falling on unit. If necessary, because of snow accumulation, the unit was elevated three to 12 inches above the pad and a snow drift barrier was installed.

Once everything was in place, the installer set up the control device and commissioned the system. System controllers were connected to the home's Wi-Fi network or an Ethernet cable for remote monitoring and control.

PRINCIPLE OF OPERATION

The control device monitors how much hot water in the storage tank and activates the heat pump when the most renewable energy is available and energy costs are lowest, while also ensuring hot water is available. The tank stores water in a stratified manner with the hottest water at the top of the tank. The control device activates the heat pump to draw the cold water from the bottom of the tank and reheats it to 150°F before sending it back to the top of the tank. The heat pump continues heating until the control device projects that there is enough hot water for the projected heating and hot water needs of the 12 to 24 hours. On the first day of every month, the heat pump heats the entire tank to 150°F to sanitize the system.

If the control device is powered off due to a circuit breaker trip or other reason, the system reverts to standard heat pump operations. When the tank sensor senses the water dropping below 113°F the heat pump activates until the return water from the bottom of the tank reaches 126°F.

When the thermostat calls for heat, the circulator in the control device turns on and then the air handler turns on after allowing the heating coil to warm up. Once the thermostat setpoint has been reached, the circulator in the control device turns off, the water in the heating loop cools down and the control device shuts off the air handler. During the non-heating season, the control device activates the circulator once per day to flush the water in the heating loop to avoid stagnating water.

Economizers were installed with the air handlers at three of the four sites (see Table 8). The economizer only operates when the thermostat calls for cooling and the outdoor air temperature is lower than the indoor air temperature by 2°F.

Pilot Systems Evaluation

Loads

In the winter and shoulder seasons the system utilizes the hot water tank as much as possible in control mode to satisfy the space heating load and the domestic hot water load. In the summer season, the system utilizes the hot water tank solely for domestic hot water loads, with cooling loads satisfied by the auxiliary air source heat pump.

Figure 11 shows the daily average load for summer and winter by end use. These values are the metered average load served on days that call for space heating or space cooling at each site. The loads include metered data for both the baseline mode periods and the control mode periods. The figures show significantly higher space heating loads and cooling loads for Site 2, which also has the highest domestic hot water load. The domestic hot water load is the same in both seasons, given that the domestic hot water load profile was averaged across all days. Site 1 had the second-highest space heating load and only the third-highest cooling load relative to Site 2. This is due in part to the fact that both Sites 1 and 2 had the same average space-heating set point, and Site 1 had an average space cooling set point 4 degrees warmer than Site 2 (see Table 8). Site 3 had a higher cooling load than Site 4, along with a lower cooling set point than Site 4. Site 3 had the smallest domestic hot water load and space heating loads, while Site 4 had the smallest cooling load.

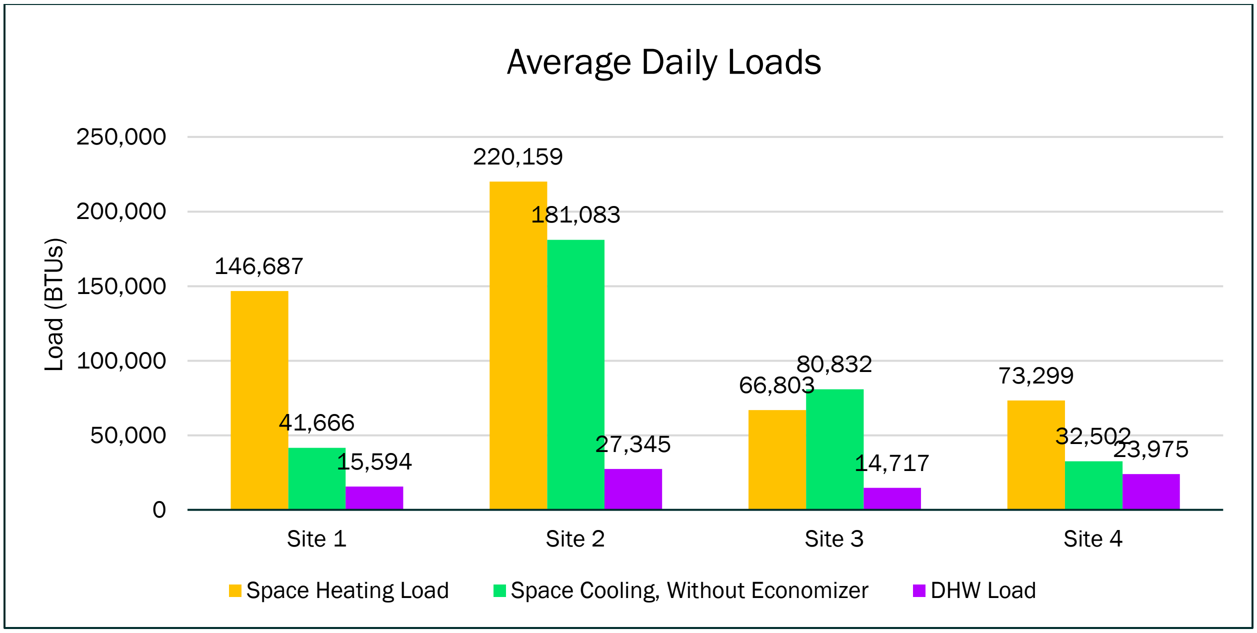


Figure 11: Average daily heating, cooling, and domestic hot water loads.

Figure 12 presents the additional cooling provided by economizer operation for those sites equipped as such. Site 3 had some unmet load in economizer mode, with an average space temperature approximately 1.5 degrees above cooling set point vs. approximately 0.5 degrees for other sites. Investigation found that the smart thermostat at Site 3 underestimated the actual outside air temperature at times and would call for economizer mode, while the outside temperature was greater than the inside temperature.

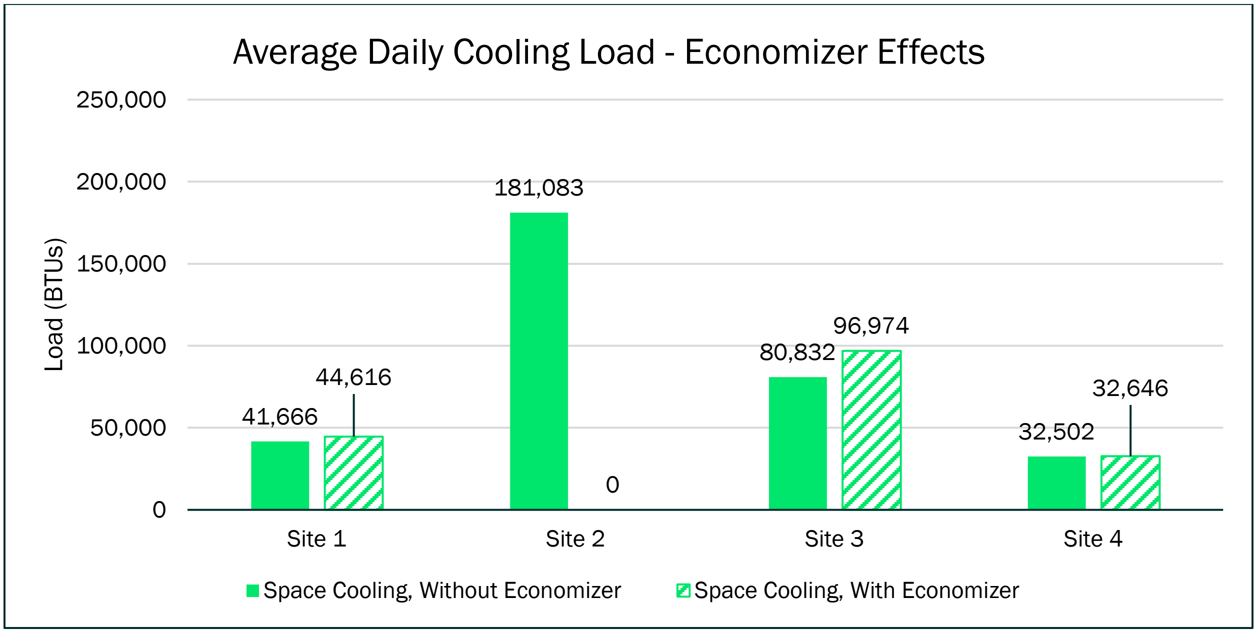


Figure 12: Average daily cooling load with and without an economizer.

At the four sites studied, the estimated annual domestic hot water load ranged from 5.3 to 9.9 MMBtu/yr, which is significantly lower than the residential average domestic hot water load in California of 16.3 MMBtu/yr (Residential Energy Consumption Survey (RECS) 2020). It is unknown how much and in what manner the hot water fixtures and occupancy of the sites differ from average households represented in the Residential Energy Consumption Survey. However, it is worth noting that the performance of the systems may differ with higher domestic hot water load. Table 9 shows the domestic hot water as a percent of the seasonal system load.

Table 9: Domestic Hot Water Load As a Percentage of Seasonal Load and Annual Domestic Hot Water Energy Consumption

	Site 1	Site 2	Site 3	Site 4
DHW as % load during summer	61%	14%	24%	57%
DHW as % load during winter	10%	11%	19%	26%
Annual energy consumption From DHW [MMBtu/yr]	5.69	9.98	5.37	8.75

The winter and summer load profiles for all sites are presented in Figure 13 through Figure 20. The load profiles compare the baseline and control-mode electric energy consumption against the backdrop of domestic hot water load, space heating or cooling load, and the utility rate peak and partial peak periods. The summer loads show a noticeable shift in electric energy from the peak periods. In the winter, much of the load is space heating which occurs in the off-peak period. Significantly more load would need to be shifted if the utility winter peak included a period in the morning. Sites 1 and 3 show a small load during winter peaks while Site 4 shows a more significant load shift.

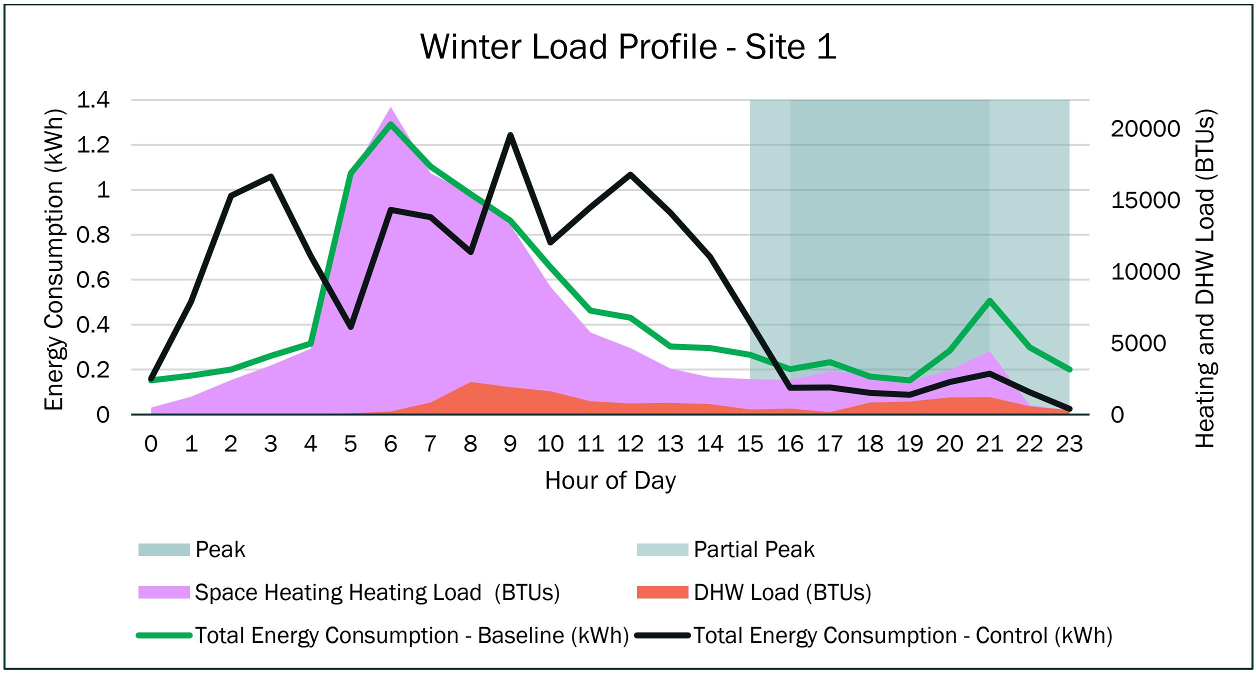


Figure 13: Site 1 average daily winter (months 1–4) load profile.

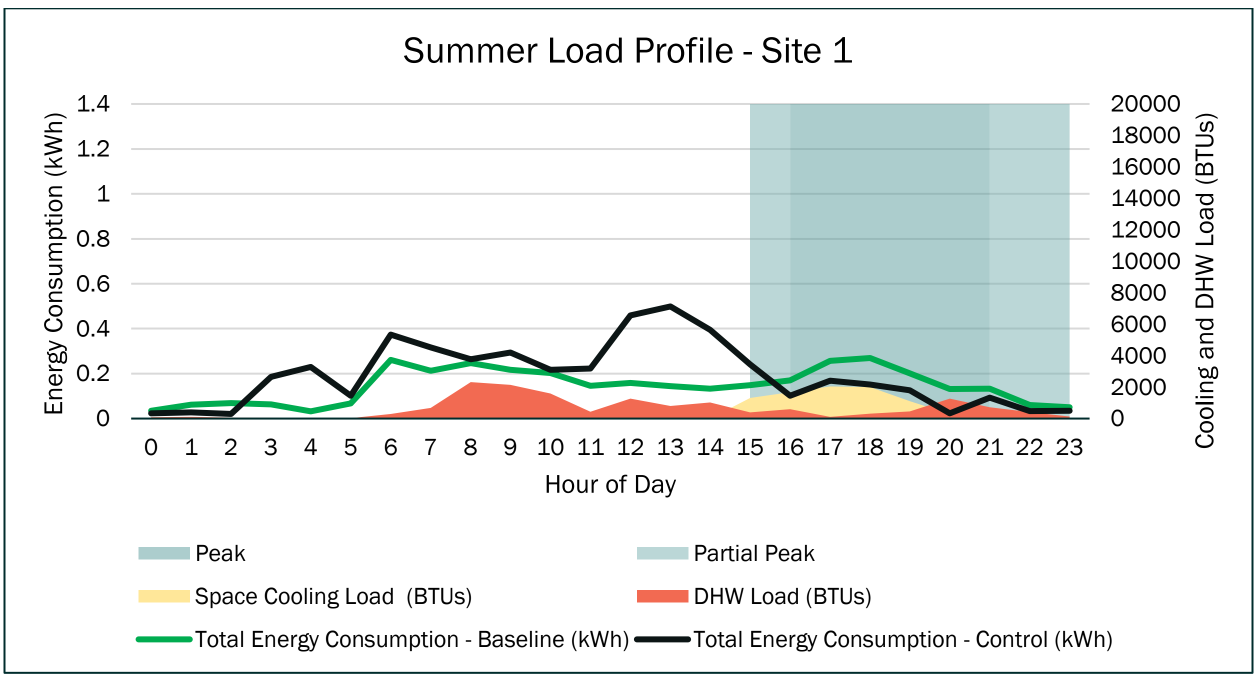


Figure 14: Site 1 average daily summer (months 5–6) load profile.

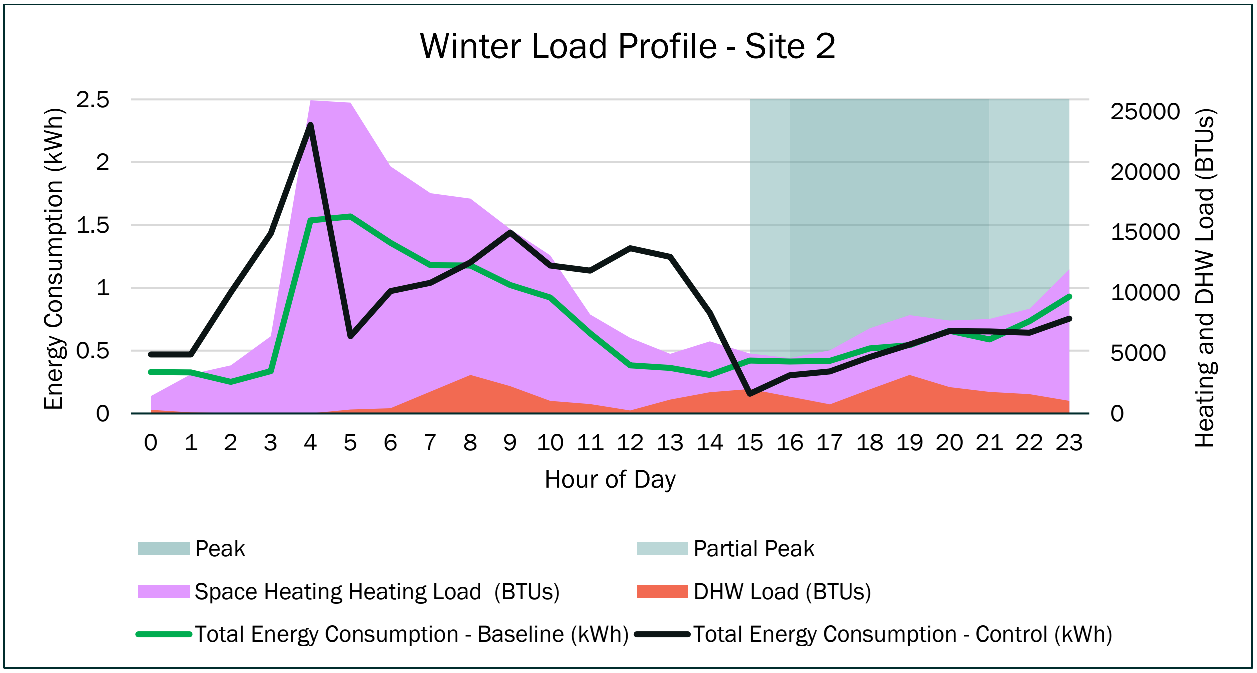


Figure 15: Site 2 average daily winter (months 1–5) load profile.

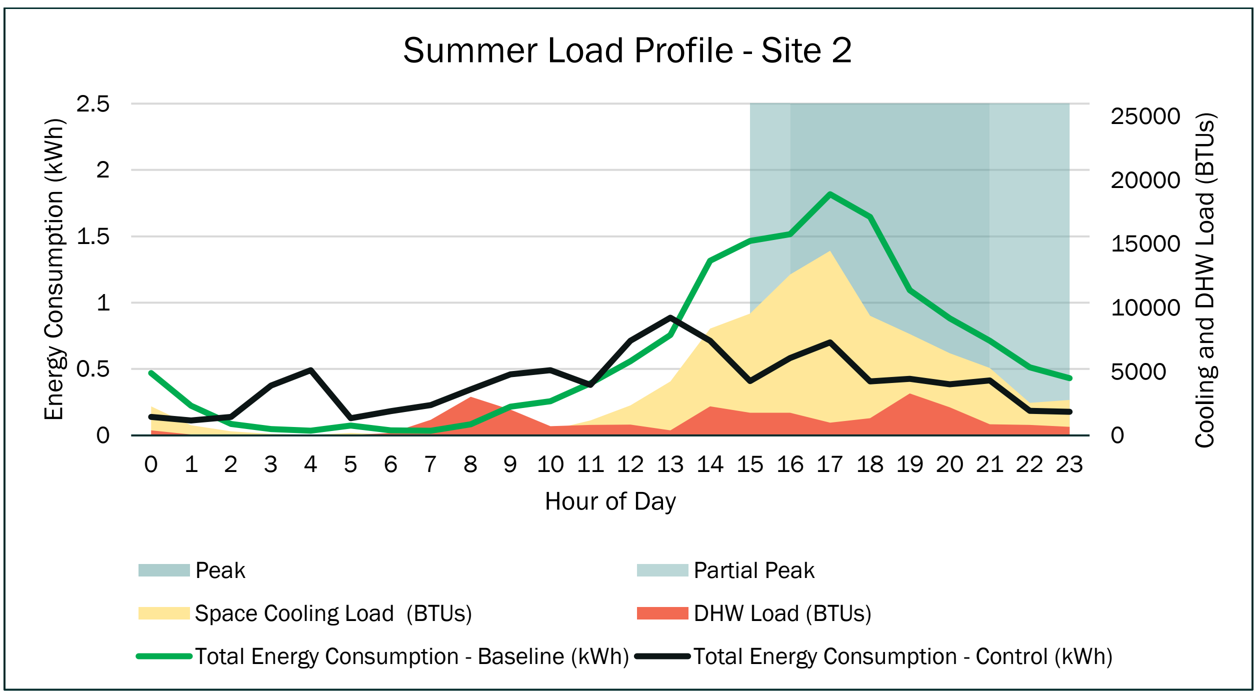


Figure 16: Site 2 average daily summer (months 6–8) load profile.

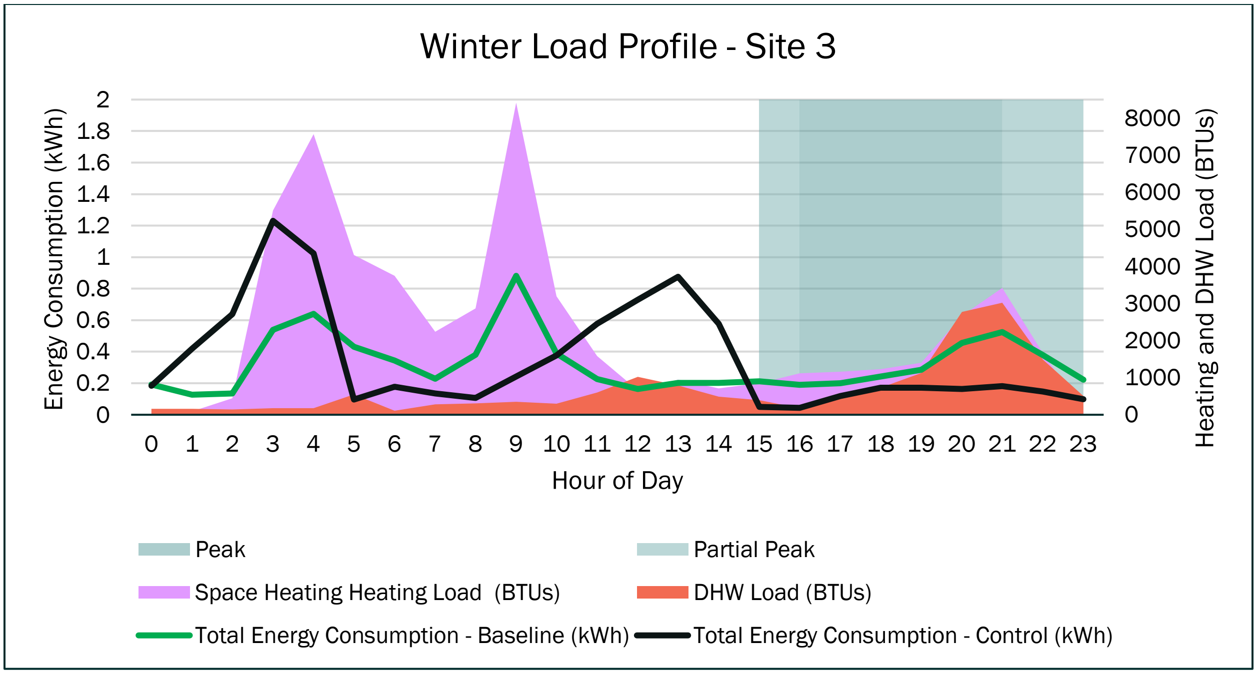


Figure 17: Site 3 average daily winter (months 1–5) load profile.

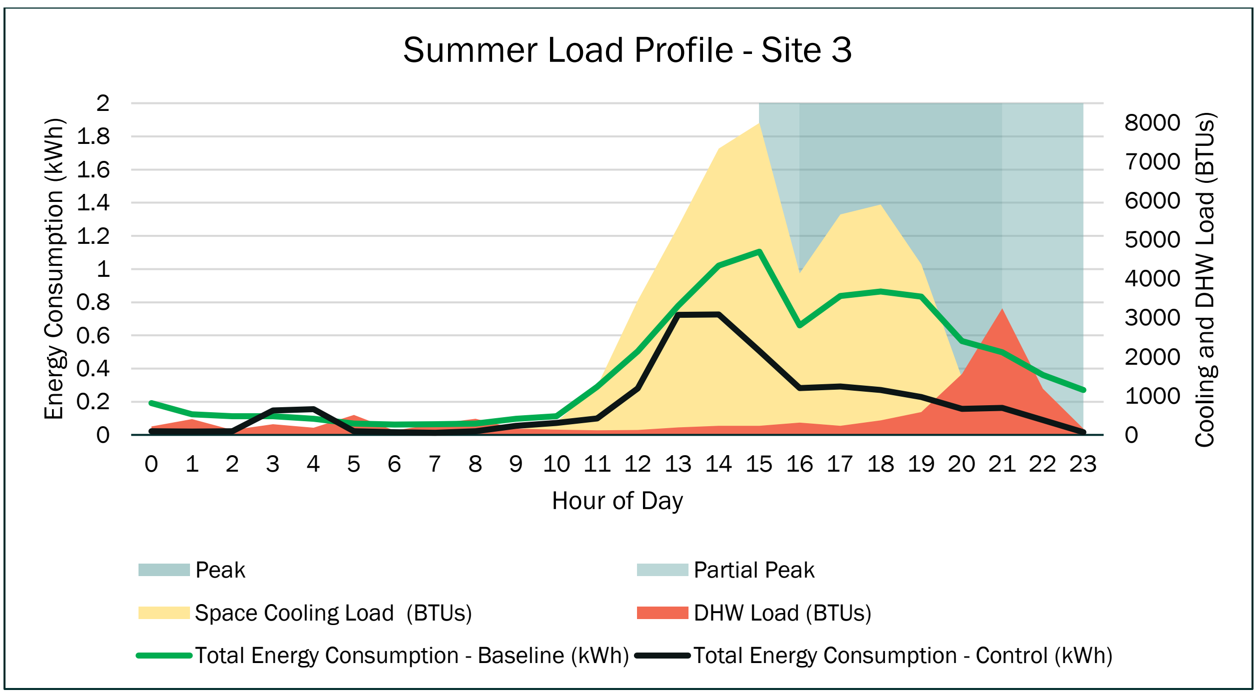


Figure 18: Site 3 average daily summer (months 6–8) load profile.

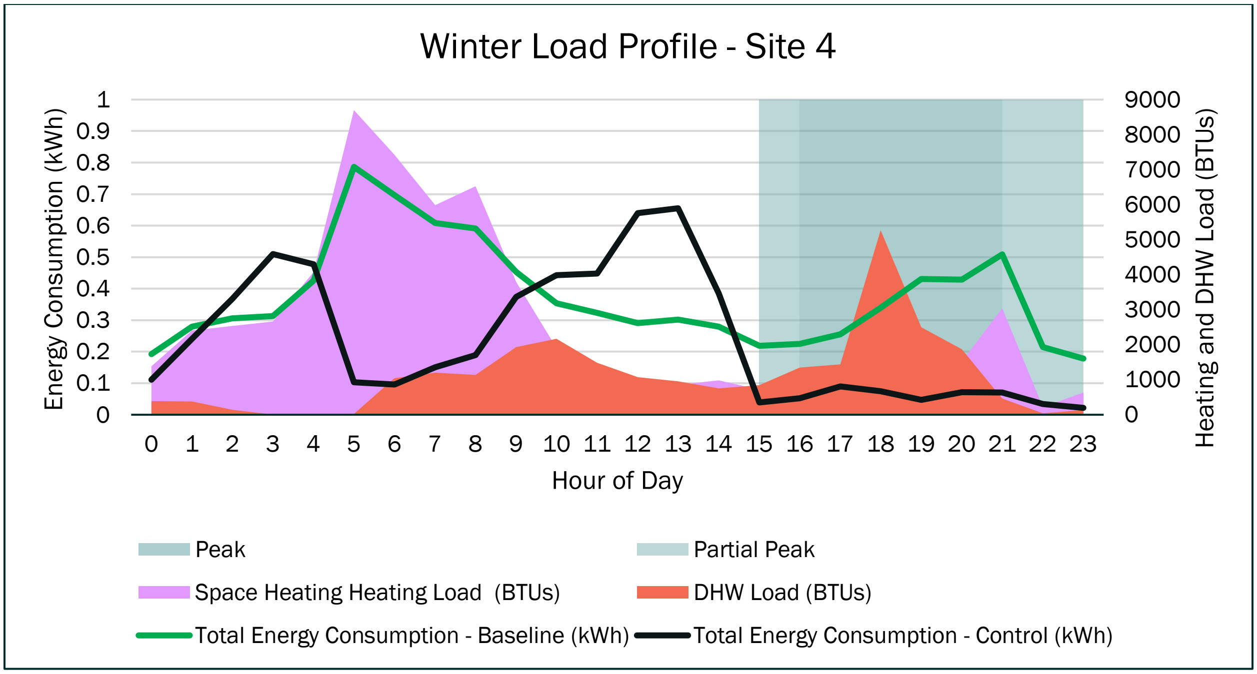


Figure 19: Site 4 average daily winter (months 1–5) load profile.

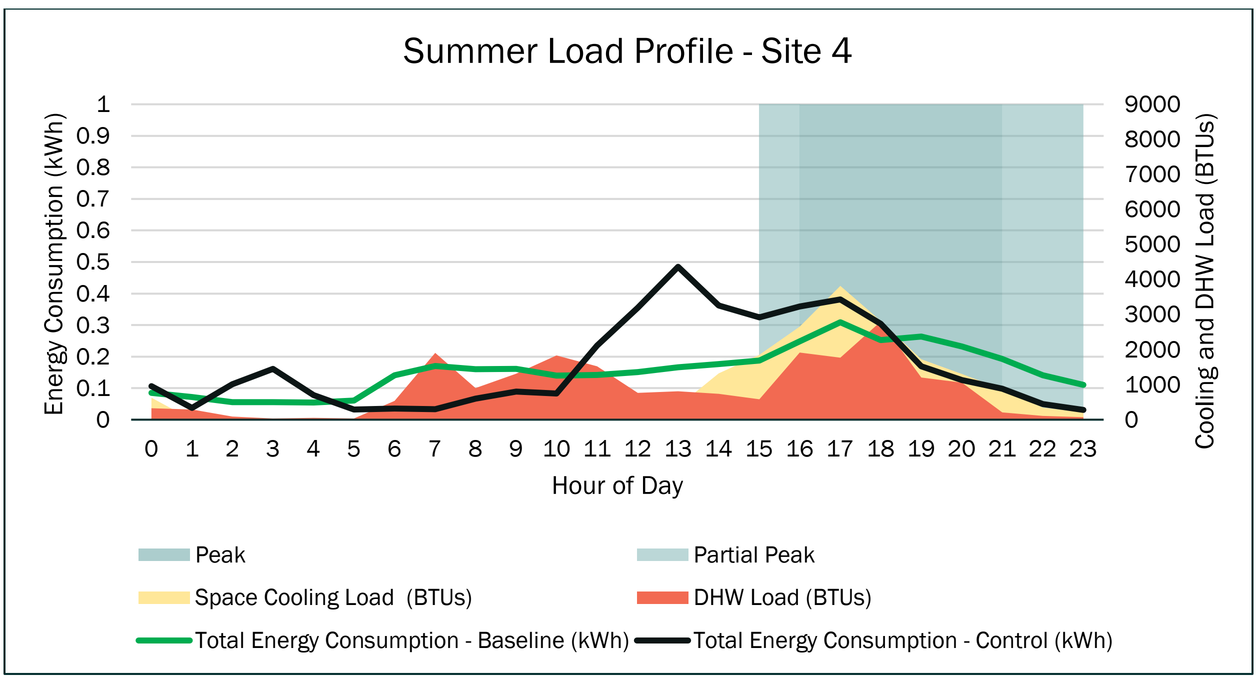


Figure 20: Site 4 average daily summer (months 6–8) load profile.

Energy Efficiency

The energy efficiency of the systems is represented by the coefficient of performance in the heating mode and the energy efficiency ratio in the cooling mode. Coefficient of performance and energy efficiency ratio values for each site or mode are calculated, based on the data collected during the

Table 10 and cooling season efficiencies, with and without the economizer, are presented in Table 11. The coefficient of performance and energy efficiency ratio values shown represent an average over the monitoring period and the daily maximum and minimums. The coefficient of performance values capture the total power input to the air-to-water heat pump, air-to-air heat pump, and the air handling unit and the total heating output for space heating and domestic water heating. Therefore, the coefficients of performance represent the efficiency of the system as a whole, inclusive of any operating cycles for refrigerant temperature management. For reference, the manufacturer listed coefficients of performance for the air-to-water heat pump are 2.8 @ 17°F OAT (outdoor air temperature), 4.2 @ 43°F OAT, and 5.5 @ 80°F OAT. The manufacturer listed coefficients of performance for the air-to-air heat pump are 2.16 @ 17°F OAT and 3.28 @ 47°F OAT. The listed cooling efficiency of the air-to-air heat pump is a SEER of 18 and advertised as “up to 20 SEER.”

Table 10: Heating Season Efficiencies by Control Mode

Mode	Efficiency Metric	Site 1	Site 2	Site 3	Site 4
Baseline	COP – Average	4.26	4.51	3.08	4.24
	COP – Max	5.53	8.99	8.38	12.77
	COP – Min	1.98	2.25	1.05	1.14
Control	COP - Average	3.61	3.84	2.52	3.35
	COP – Max	4.12	6.17	8.21	9.43
	COP – Min	2.96	2.14	1.16	1.99

For all sites, the coefficient of performance was less while operating in control mode, indicating decreased heating efficiency. The percent decrease in average coefficient of performance for thermal storage operation was between 15 percent and 21 percent. The lower efficiency is attributable to the effect of higher return water temperatures from the energy storage on the efficiency of the heat pump, as well as the thermal losses of energy storage. The results provide insight on the degree of this combined effect. Site 2 has the highest coefficient of performance of all sites in both baseline and control modes, while Site 3 has the lowest coefficient of performance for both baseline and control modes.

Table 11: Cooling Season Efficiencies By Mode With And Without Economizer

Mode	Efficiency Metric	Site 1		Site 2	Site 3		Site 4	
		W/ Econ	No Econ	No Econ	W/ Econ	No Econ	W/ Econ	No Econ
Baseline	EER – Average	61.4	20.4	19.0	19.0	15.3	22.3	26.8
	EER – Max	114.3	24.0	22.7	41.9	25.0	33.6	33.6
	EER – Min	19.5	18.2	15.9	4.1	11.7	2.0	21.7
Control	EER – Average	71.2	12.6	20.6	22.0	15.6	17.2	22.2
	EER – Max	97.0	17.6	25.5	53.7	21.3	27.8	28.0
	EER – Min	19.4	3.1	16.5	5.1	4.3	2.8	19.3

The most notable variations in cooling efficiency are between the economizer operation versus the non-economizer operation. Sites and modes with the lowest daily minimum energy efficiency ratio, (non-economizer control mode for Site 1, economizer control mode and baseline mode for Site 3, and economizer control mode and baseline mode for Site 4), had relatively lower average energy efficiency ratios. The project team has not found a cause or correlation for these lower instances of cooling efficiency, which warrants further investigation. In commercial building HVAC systems, where economizers are more commonly applied, an economizer operation is known to be prone to failure due to sensor error and damper actuator slippage. An economizer was not part of the Site 2 system.

For sites 1 and 3, the energy efficiency ratio was greater with economizer operation in control mode than in baseline mode. The energy efficiency ratio for Site 1 and Site 4 decreased between baseline and control mode when not accounting for the economizer. The energy efficiency ratio for Site 3 remained nearly constant with a slight increase between baseline and control mode. None of the variations in cooling efficiency between modes are attributable to the thermal storage system.

Sites 1 and 4 have fewer data points in cooling mode, either with or without economizer. Site 4 also had an issue with the installation location of the return air enthalpy sensor, which measures entering outdoor air temperature when an economizer is active, thereby underestimating the cooling delivered in the economizer mode.

Model Metrics

The project team explored several techniques to model and predict the annual energy consumption of the thermal storage system in baseline and control modes at each site. The team calculated standard statistical metrics to evaluate the fit of each model. These metrics include R-squared, coefficient of variation root mean squared error (CV-RMSE), fractional savings uncertainty (FSU), an

F-test for model P-value, and net determination bias error (NBE). The team referenced ASHRAE Guideline 14 to evaluate whether model metrics fell within the acceptable thresholds. Statistical metrics for each model are summarized in Table 12 below.

Table 12: Model Metric Thresholds

Metric	Threshold
CV-RMSE	Less than 25% for daily data
R-squared	Greater than 50%
FSU	Less than 50% for a 90% confidence interval
F-test for model P-value	Less than 10%
Net determination bias error	Less than 0.005%

Two of the modeling techniques the team explored included generating linear piecewise regression models and using a gradient boosting machine learning method to produce optimized hourly models. Both techniques used hourly outdoor dry-bulb temperature and domestic hot water load to predict hourly energy consumption. And the linear piecewise regression models also included time of the day and hour of the week as dependent variables. The model metrics for the hourly models produced with both techniques were very poor, far beyond acceptable thresholds. Model fit for residential HVAC and domestic hot water heating applications can be difficult, due to multiple confounding factors, such as internal heat gains from occupants, solar heat gains, and wind-driven infiltration, which are not readily available independent variables for individual sites.

In addition to these two techniques, the team generated multi-variable linear regression models for each site and control mode. Using daily average outdoor dry-bulb temperature and domestic hot water load as dependent variables, the team predicted hourly energy consumption with hourly TMY weather data. The models were manually tuned to optimized R-squared and CV-RMSE values while not overfitting the model to the data. The statistical metrics for each model are presented in Table 13 below.

Table 13: Statistical Metrics of Model Performance for Each Site for Baseline and Control Modes

Metric	Site 1		Site 2		Site 3		Site 4	
	Baseline	Control	Baseline	Control	Baseline	Control	Baseline	Control
CV-RMSE	22%	17%	15%	18%	26%	33%	35%	27%
R-squared	84%	94%	87%	92%	46%	81%	70%	53%

Metric	Site 1		Site 2		Site 3		Site 4	
	Baseline	Control	Baseline	Control	Baseline	Control	Baseline	Control
FSU	43%	34%	40%	25%	72%	45%	73%	72%
F-test for model P-value	0%	0%	0%	0%	0%	0%	0%	0%
NBE	0%	0%	0%	0%	0%	0%	0%	0%
Pearson's Autocorrelation	0.12	0.09	−0.30	−0.04	0.13	0.04	−0.38	0.31
Number of data points	55	49	43	88	85	92	85	43

The CV-RMSE is lowest for both control and baseline models of Sites 1 and 2, indicating strong model fits. The R-squared values for these four models are also well above the acceptable threshold. However, Site 1 models predicted periods of negative energy consumption over the course of the year.

The CV-RMSE for Site 3 baseline and control models and Site 4 control model is just beyond the acceptable threshold, indicating the model fit is not as strong. While the R-squared is above the acceptable threshold for Site 4 baseline and control models, the FSU for Site 4 is well above the acceptable limits, indicating large uncertainty in model predictions.

Given the poor Site 4 model fit and the periods of negative energy consumption predicted by the Site 1 model, results of predicted energy and cost savings, cost effectiveness, and GHG emission reductions are presented for Sites 2 and 3 only.

In addition to linear regression modeling, the team calculated average hourly energy consumption and savings from the metered data and extrapolated to annual estimates. Results from the metered average analysis are presented along with the results from the linear regression modeling.

Energy Consumption and Savings

Predicted annual energy consumption and savings potential are estimated with the linear regression models and a metered average analysis for each site and mode. Figure 21 presents the predicted annual energy consumption for each site.

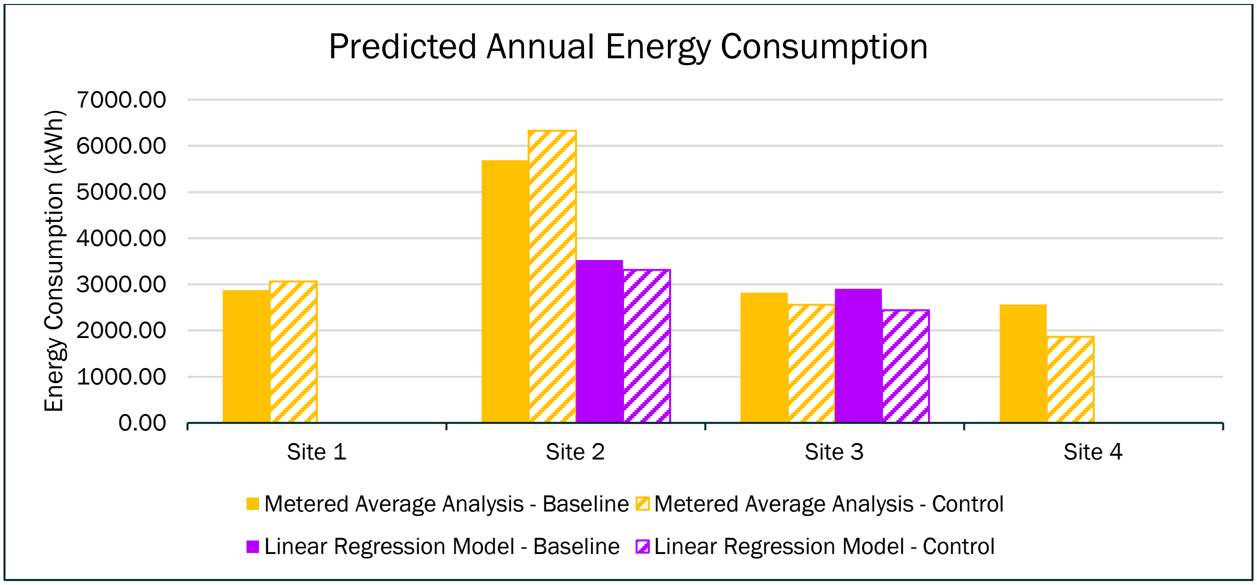


Figure 21: Predicted annual energy consumption in each mode at each site.

Site 2 has the highest energy consumption in baseline and control model for both analysis methods, which aligns with the daily average load calculations (see Figure 15). The large discrepancy in magnitude between the metered average analysis and linear regression model for Site 2 is not explainable from the metered data or model metrics. Possible causes of this discrepancy include the effect of microclimate (site weather conditions varying significantly from the TMY weather dataset), and the effect of using a daily average model to predict hourly data. The metered analysis method shows energy consumption in control mode increases at Sites 1 and 2 while it decreases at Sites 3 and 4 compared to baseline mode. However, the linear regression modeling method shows energy consumption decreases at Site 2 as well. From these estimations of energy consumption in each mode, the team estimated annual energy savings for these sites, presented in Figure 22 below. Site 4 shows the greatest energy savings potential followed by Site 3. Site 2 shows positive savings with the linear regression model but negative savings with the metered average analysis.

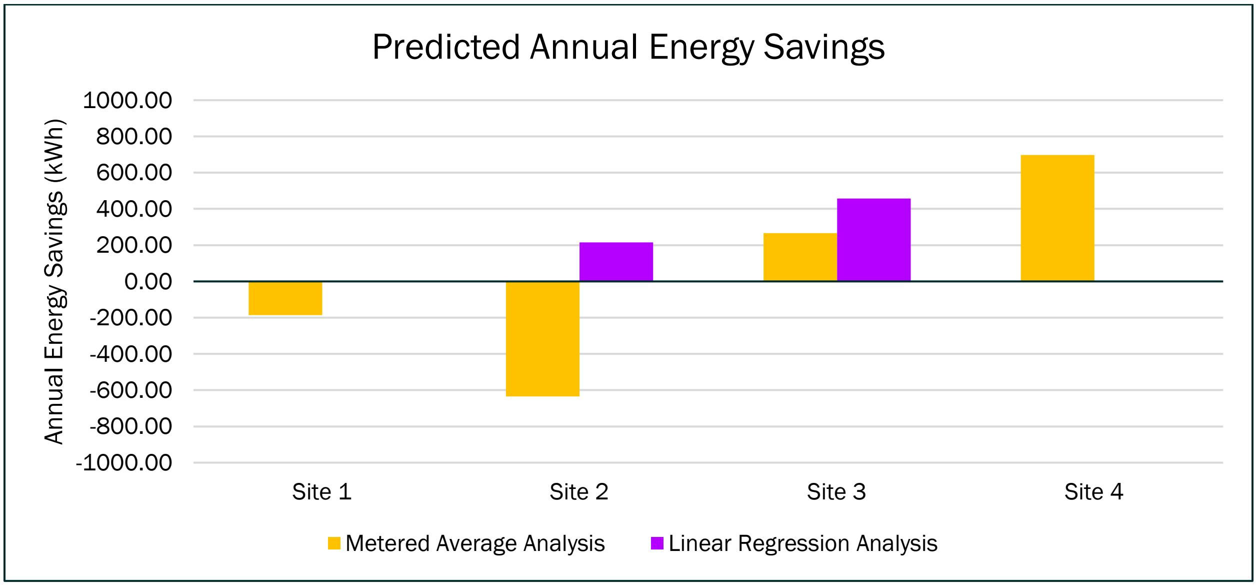


Figure 22: Predicted annual energy savings at each site.

Heat maps of the average hourly predicted energy savings by month of the year are shown in Figure 23 through Figure 26 for each site. Heatmaps were generated, using results from the metered average analysis. Sites 2 and 3 heat maps show savings in the mid-day summer hours when the systems are providing load shift for domestic hot water only. Site 1 shows very small savings during mid-day summer hours while Site 4 shows slightly negative savings through 6pm. All sites show negative savings (increase in energy use) in the winter months with the greatest increase in energy use in the winter overnight hours and mid-morning to mid-afternoon. This can be attributed to the decrease in efficiency and thus coefficient of performance of the combined heat pump and thermal energy storage system (see Table 10).

Site 1	Month of the Year											
	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec
Hour of Day 0	-0.018	-0.018	-0.018	-0.018	-0.018	0.013	0.013	0.013	-0.018	-0.018	-0.018	-0.018
1	-0.198	-0.198	-0.198	-0.198	-0.198	0.040	0.040	0.040	-0.198	-0.198	-0.198	-0.198
2	-0.518	-0.518	-0.518	-0.518	-0.518	0.054	0.054	0.054	-0.518	-0.518	-0.518	-0.518
3	-0.482	-0.482	-0.482	-0.482	-0.482	-0.141	-0.141	-0.141	-0.482	-0.482	-0.482	-0.482
4	-0.160	-0.160	-0.160	-0.160	-0.160	-0.191	-0.191	-0.191	-0.160	-0.160	-0.160	-0.160
5	0.616	0.616	0.616	0.616	0.616	-0.031	-0.031	-0.031	0.616	0.616	0.616	0.616
6	0.385	0.385	0.385	0.385	0.385	0.071	0.071	0.071	0.385	0.385	0.385	0.385
7	0.437	0.437	0.437	0.437	0.437	0.042	0.042	0.042	0.437	0.437	0.437	0.437
8	0.162	0.162	0.162	0.162	0.162	-0.044	-0.044	-0.044	0.162	0.162	0.162	0.162
9	-0.256	-0.256	-0.256	-0.256	-0.256	-0.097	-0.097	-0.097	-0.256	-0.256	-0.256	-0.256
10	-0.050	-0.050	-0.050	-0.050	-0.050	-0.030	-0.030	-0.030	-0.050	-0.050	-0.050	-0.050
11	-0.382	-0.382	-0.382	-0.382	-0.382	-0.061	-0.061	-0.061	-0.382	-0.382	-0.382	-0.382
12	-0.541	-0.541	-0.541	-0.541	-0.541	-0.323	-0.323	-0.323	-0.541	-0.541	-0.541	-0.541
13	-0.490	-0.490	-0.490	-0.490	-0.490	-0.374	-0.374	-0.374	-0.490	-0.490	-0.490	-0.490
14	-0.260	-0.260	-0.260	-0.260	-0.260	-0.317	-0.317	-0.317	-0.260	-0.260	-0.260	-0.260
15	-0.018	-0.018	-0.018	-0.018	-0.018	-0.120	-0.120	-0.120	-0.018	-0.018	-0.018	-0.018
16	0.154	0.154	0.154	0.154	0.154	0.079	0.079	0.079	0.154	0.154	0.154	0.154
17	0.155	0.155	0.155	0.155	0.155	0.112	0.112	0.112	0.155	0.155	0.155	0.155
18	0.087	0.087	0.087	0.087	0.087	0.152	0.152	0.152	0.087	0.087	0.087	0.087
19	0.066	0.066	0.066	0.066	0.066	0.101	0.101	0.101	0.066	0.066	0.066	0.066
20	0.124	0.124	0.124	0.124	0.124	0.119	0.119	0.119	0.124	0.124	0.124	0.124
21	0.377	0.377	0.377	0.377	0.377	0.047	0.047	0.047	0.377	0.377	0.377	0.377
22	0.248	0.248	0.248	0.248	0.248	0.028	0.028	0.028	0.248	0.248	0.248	0.248
23	0.167	0.167	0.167	0.167	0.167	0.022	0.022	0.022	0.167	0.167	0.167	0.167
Monthly Sum	-12.2	-11.1	-12.2	-11.8	-12.2	-25.5	-26.3	-26.3	-11.8	-12.2	-11.8	-12.2

Figure 23: Predicted energy savings at Site 1 by hour of day and month of year.

Site 2	Month of the Year												Daily
Hour of Day	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec	Average
0	-0.160	-0.160	-0.160	-0.160	-0.160	0.360	0.360	0.360	-0.160	-0.160	-0.160	-0.160	-0.017
1	-0.179	-0.179	-0.179	-0.179	-0.179	0.139	0.139	0.139	-0.179	-0.179	-0.179	-0.179	-0.027
2	-0.819	-0.819	-0.819	-0.819	-0.819	-0.039	-0.039	-0.039	-0.819	-0.819	-0.819	-0.819	-0.032
3	-1.092	-1.092	-1.092	-1.092	-1.092	-0.324	-0.324	-0.324	-1.092	-1.092	-1.092	-1.092	-0.035
4	-0.699	-0.699	-0.699	-0.699	-0.699	-0.457	-0.457	-0.457	-0.699	-0.699	-0.699	-0.699	-0.038
5	0.536	0.536	0.536	0.536	0.536	-0.044	-0.044	-0.044	0.536	0.536	0.536	0.536	-0.038
6	0.235	0.235	0.235	0.235	0.235	-0.139	-0.139	-0.139	0.235	0.235	0.235	0.235	-0.031
7	0.203	0.203	0.203	0.203	0.203	-0.187	-0.187	-0.187	0.203	0.203	0.203	0.203	-0.025
8	0.145	0.145	0.145	0.145	0.145	-0.258	-0.258	-0.258	0.145	0.145	0.145	0.145	-0.013
9	-0.348	-0.348	-0.348	-0.348	-0.348	-0.237	-0.237	-0.237	-0.348	-0.348	-0.348	-0.348	0.009
10	-0.266	-0.266	-0.266	-0.266	-0.266	-0.210	-0.210	-0.210	-0.266	-0.266	-0.266	-0.266	0.042
11	-0.500	-0.500	-0.500	-0.500	-0.500	0.029	0.029	0.029	-0.500	-0.500	-0.500	-0.500	0.078
12	-0.920	-0.920	-0.920	-0.920	-0.920	-0.105	-0.105	-0.105	-0.920	-0.920	-0.920	-0.920	0.115
13	-0.869	-0.869	-0.869	-0.869	-0.869	-0.048	-0.048	-0.048	-0.869	-0.869	-0.869	-0.869	0.135
14	-0.482	-0.482	-0.482	-0.482	-0.482	0.706	0.706	0.706	-0.482	-0.482	-0.482	-0.482	0.143
15	0.284	0.284	0.284	0.284	0.284	1.126	1.126	1.126	0.284	0.284	0.284	0.284	0.139
16	0.144	0.144	0.144	0.144	0.144	0.851	0.851	0.851	0.144	0.144	0.144	0.144	0.110
17	0.100	0.100	0.100	0.100	0.100	1.189	1.189	1.189	0.100	0.100	0.100	0.100	0.083
18	0.092	0.092	0.092	0.092	0.092	1.307	1.307	1.307	0.092	0.092	0.092	0.092	0.038
19	0.030	0.030	0.030	0.030	0.030	0.717	0.717	0.717	0.030	0.030	0.030	0.030	-0.002
20	0.034	0.034	0.034	0.034	0.034	0.531	0.531	0.531	0.034	0.034	0.034	0.034	-0.009
21	-0.070	-0.070	-0.070	-0.070	-0.070	0.341	0.341	0.341	-0.070	-0.070	-0.070	-0.070	-0.013
22	0.147	0.147	0.147	0.147	0.147	0.363	0.363	0.363	0.147	0.147	0.147	0.147	-0.014
23	0.143	0.143	0.143	0.143	0.143	0.270	0.270	0.270	0.143	0.143	0.143	0.143	-0.017
Monthly Sum	-51.1	-37.3	-4.8	-10.7	55.5	61.1	45.8	95.9	67.8	48.8	-10.1	-45.1	215.8

Figure 24: Predicted energy savings at Site 2 by hour of day and month of year.

Site 3		Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec	Daily Average
		Month of the Year												
Hour of Day	0	-0.001	-0.001	-0.001	-0.001	-0.001	0.172	0.172	0.172	-0.001	-0.001	-0.001	-0.001	0.193
	1	-0.206	-0.206	-0.206	-0.206	-0.206	0.109	0.109	0.109	-0.206	-0.206	-0.206	-0.206	0.183
	2	-0.458	-0.458	-0.458	-0.458	-0.458	0.094	0.094	0.094	-0.458	-0.458	-0.458	-0.458	0.179
	3	-0.654	-0.654	-0.654	-0.654	-0.654	-0.026	-0.026	-0.026	-0.654	-0.654	-0.654	-0.654	0.175
	4	-0.383	-0.383	-0.383	-0.383	-0.383	-0.050	-0.050	-0.050	-0.383	-0.383	-0.383	-0.383	0.172
	5	0.258	0.258	0.258	0.258	0.258	0.048	0.048	0.048	0.258	0.258	0.258	0.258	0.172
	6	0.082	0.082	0.082	0.082	0.082	0.048	0.048	0.048	0.082	0.082	0.082	0.082	0.186
	7	0.039	0.039	0.039	0.039	0.039	0.053	0.053	0.053	0.039	0.039	0.039	0.039	0.192
	8	0.172	0.172	0.172	0.172	0.172	0.050	0.050	0.050	0.172	0.172	0.172	0.172	0.192
	9	0.577	0.577	0.577	0.577	0.577	0.044	0.044	0.044	0.577	0.577	0.577	0.577	0.187
	10	0.066	0.066	0.066	0.066	0.066	0.045	0.045	0.045	0.066	0.066	0.066	0.066	0.359
	11	-0.353	-0.353	-0.353	-0.353	-0.353	0.190	0.190	0.190	-0.353	-0.353	-0.353	-0.353	0.553
	12	-0.565	-0.565	-0.565	-0.565	-0.565	0.235	0.235	0.235	-0.565	-0.565	-0.565	-0.565	0.729
	13	-0.675	-0.675	-0.675	-0.675	-0.675	0.057	0.057	0.057	-0.675	-0.675	-0.675	-0.675	0.866
	14	-0.377	-0.377	-0.377	-0.377	-0.377	0.302	0.302	0.302	-0.377	-0.377	-0.377	-0.377	0.712
	15	0.163	0.163	0.163	0.163	0.163	0.624	0.624	0.624	0.163	0.163	0.163	0.163	0.530
	16	0.147	0.147	0.147	0.147	0.147	0.380	0.380	0.380	0.147	0.147	0.147	0.147	0.411
	17	0.082	0.082	0.082	0.082	0.082	0.560	0.560	0.560	0.082	0.082	0.082	0.082	0.339
	18	0.076	0.076	0.076	0.076	0.076	0.616	0.616	0.616	0.076	0.076	0.076	0.076	0.269
	19	0.126	0.126	0.126	0.126	0.126	0.623	0.623	0.623	0.126	0.126	0.126	0.126	0.227
	20	0.303	0.303	0.303	0.303	0.303	0.410	0.410	0.410	0.303	0.303	0.303	0.303	0.201
	21	0.365	0.365	0.365	0.365	0.365	0.332	0.332	0.332	0.365	0.365	0.365	0.365	0.182
	22	0.221	0.221	0.221	0.221	0.221	0.272	0.272	0.272	0.221	0.221	0.221	0.221	0.190
	23	0.140	0.140	0.140	0.140	0.140	0.255	0.255	0.255	0.140	0.140	0.140	0.140	0.195
Monthly Sum		98.6	88.7	160.1	143.6	328.3	246.9	230.1	462.4	269.6	499.9	137.5	123.0	2788.7

Figure 25: Predicted energy savings at Site 3 by hour of day and month of year.

Site 4	Month of the Year											
	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec
Hour of Day 0	0.040	0.040	0.040	0.040	0.040	-0.029	-0.029	-0.029	0.040	0.040	0.040	0.040
1	0.004	0.004	0.004	0.004	0.004	0.030	0.030	0.030	0.004	0.004	0.004	0.004
2	-0.087	-0.087	-0.087	-0.087	-0.087	-0.063	-0.063	-0.063	-0.087	-0.087	-0.087	-0.087
3	-0.204	-0.204	-0.204	-0.204	-0.204	-0.113	-0.113	-0.113	-0.204	-0.204	-0.204	-0.204
4	-0.053	-0.053	-0.053	-0.053	-0.053	-0.028	-0.028	-0.028	-0.053	-0.053	-0.053	-0.053
5	0.652	0.652	0.652	0.652	0.652	0.026	0.026	0.026	0.652	0.652	0.652	0.652
6	0.518	0.518	0.518	0.518	0.518	0.101	0.101	0.101	0.518	0.518	0.518	0.518
7	0.364	0.364	0.364	0.364	0.364	0.132	0.132	0.132	0.364	0.364	0.364	0.364
8	0.337	0.337	0.337	0.337	0.337	0.093	0.093	0.093	0.337	0.337	0.337	0.337
9	0.064	0.064	0.064	0.064	0.064	0.070	0.070	0.070	0.064	0.064	0.064	0.064
10	-0.110	-0.110	-0.110	-0.110	-0.110	0.054	0.054	0.054	-0.110	-0.110	-0.110	-0.110
11	-0.136	-0.136	-0.136	-0.136	-0.136	-0.100	-0.100	-0.100	-0.136	-0.136	-0.136	-0.136
12	-0.351	-0.351	-0.351	-0.351	-0.351	-0.203	-0.203	-0.203	-0.351	-0.351	-0.351	-0.351
13	-0.381	-0.381	-0.381	-0.381	-0.381	-0.327	-0.327	-0.327	-0.381	-0.381	-0.381	-0.381
14	-0.112	-0.112	-0.112	-0.112	-0.112	-0.198	-0.198	-0.198	-0.112	-0.112	-0.112	-0.112
15	0.179	0.179	0.179	0.179	0.179	-0.123	-0.123	-0.123	0.179	0.179	0.179	0.179
16	0.175	0.175	0.175	0.175	0.175	-0.117	-0.117	-0.117	0.175	0.175	0.175	0.175
17	0.169	0.169	0.169	0.169	0.169	-0.072	-0.072	-0.072	0.169	0.169	0.169	0.169
18	0.269	0.269	0.269	0.269	0.269	-0.051	-0.051	-0.051	0.269	0.269	0.269	0.269
19	0.364	0.364	0.364	0.364	0.364	0.094	0.094	0.094	0.364	0.364	0.364	0.364
20	0.336	0.336	0.336	0.336	0.336	0.103	0.103	0.103	0.336	0.336	0.336	0.336
21	0.394	0.394	0.394	0.394	0.394	0.094	0.094	0.094	0.394	0.394	0.394	0.394
22	0.164	0.164	0.164	0.164	0.164	0.088	0.088	0.088	0.164	0.164	0.164	0.164
23	0.114	0.114	0.114	0.114	0.114	0.076	0.076	0.076	0.114	0.114	0.114	0.114
Monthly Sum	84.1	76.0	84.1	81.4	84.1	-13.9	-14.3	-14.3	81.4	84.1	81.4	84.1

Figure 26: Predicted energy savings at Site 4 by hour of day and month of year.

Load Shift

The system shifts domestic hot water and heating load in the winter and domestic hot water load in the summer by pre-charging the thermal energy storage tank during off-peak times. The load shift of the systems is defined as the reduction in predicted energy use during the peak and partial peak utility rate periods for control mode, versus the baseline mode. The predicted results for summer, winter, and annually are shown in Table 14, Table 15, and Table 16, respectively. The tables summarize the load shift presented in the preceding heat maps of predicted energy savings. The highest magnitude of total peak load shift occurs at Site 4 in the winter, when there is a combination of space heating and domestic hot water heating. The lowest load shift in the winter occurs at Site 2 and is due to a high heating load which could only be shifted partially with 119-gallon storage. The system vendor noted that they improved their algorithm following this pilot.

Table 14: Summer Peak, Partial Peak, and Total Peak Load Shift

Summer Load Shift	Peak Load Shift (kWh/yr)	Partial Peak Load Shift (kWh/yr)	Total Peak Load Shift (kWh/yr)	Percent Total Peak Shift
Site 1 – metered average analysis	69.5	21.1	90.5	44%

Summer Load Shift	Peak Load Shift (kWh/yr)	Partial Peak Load Shift (kWh/yr)	Total Peak Load Shift (kWh/yr)	Percent Total Peak Shift
Site 2 – metered average analysis	434.8	208.3	643.0	58%
Site 3 – metered average analysis	260.1	163.1	423.3	66%
Site 4 – metered average analysis	35.6	37.9	73.5	28%
Site 2 – linear regression analysis	63.1	44.3	107.3	67%
Site 3 - linear Regression Analysis	67.5	61.0	336.5	76%

Table 15: Winter Peak, Partial Peak, and Total Peak Load Shift

Winter Load Shift	Peak Load Shift (kWh/yr)	Partial Peak Load Shift (kWh/yr)	Total Peak Load Shift (kWh/yr)	Percent Total Peak Shift
Site 1 – metered average analysis	142.3	188.2	330.5	60%
Site 2 – metered average analysis	97.2	122.3	219.5	17%
Site 3 – metered average analysis	178.6	216.0	394.6	60%
Site 4 – metered average analysis	319.2	207.0	526.2	82%
Site 2 – linear regression analysis	18.0	−9.0	9.0	1%
Site 3 – linear regression analysis	138.50	99.1	597.8	48%

Table 16: Annual Peak, Partial Peak, and Total Peak Load Shift

Annual Load Shift	Peak Load Shift (kWh/yr)	Partial Peak Load Shift (kWh/yr)	Total Peak Load Shift (kWh/yr)	Percent Total Peak Shift
Site 1 – metered average analysis	211.76	209.25	421.02	56%
Site 2 – metered average analysis	532.0	330.6	862.6	36%
Site 3 – metered average analysis	438.7	379.1	817.9	63%
Site 4 – metered average analysis	354.8	244.9	599.7	66%
Site 2 – linear regression analysis	81.0	35.3	116.4	10%
Site 3 – linear regression analysis	531.4	402.9	934.3	55%

Cost savings

Cost savings of the load shift from peak to non-peak periods were examined by the project team, using three different rates available to PG&E (Pacific Gas and Electric) residential customers, as listed in Table 17 below. The E-TOU-C is a common, default rate for residential customers and one with relatively less differential between peak and non-peak rates. The rate shown in the table is PG&E's Total Usage Energy Charge, which does not include PG&E's baseline credit, because the baseline credit is typically exhausted by non-HVAC and hot water energy uses in the home. The Total Usage rate is appropriate for comparing the marginal effects of more or less energy use by the thermal storage systems. The E-Elec utility rate was designed for residential customers operating heat pumps and not specifically with thermal storage. The rate has a reduced winter peak rate, to reduce the burden of operating heat pumps in heating mode. The reduced peak rate can reduce the cost savings of thermal storage that shifts heating load from the winter peak period. The E-Elec rate also has higher off-peak prices than the EV2 rate. The EV2 utility rate was originally designed for EV charging but has been opened to heat pump customers as well. The advantage of this rate is it has a high cost differential between the peak and off-peak rates, resulting in advantages to customers with storage systems that can shift load to off-peak hours. California customers have the ability to pick the best rate for their consumption pattern. PG&E has a rate comparison tool that can account for a customer's usage (PG&E 2024).

Table 17: Summary of PG&E Utility Rate Structures Used for Analysis

Utility Rate	E-TOU-C 1/1/2024	E-Elec 1/1/2024	EV2 1/1/2024
Summer period	June – September	June – September	June – September
Winter period	October – May	October – May	October – May
Peak period	4 pm – 9 pm	4 pm – 9 pm	4 pm – 9 pm
Partial peak period	n/a	3 pm – 4 pm; 9 pm – 12 am	3 pm – 4 pm; 9 pm – 12 am
Total rate summer peak [$ per kWh]	0.61806	0.6358	0.65713
Total rate summer partial peak [$ per kWh]	n/a	0.47392	0.54664
Total rate summer off peak [$ per kWh]	0.53462	0.41724	0.34462
Total rate winter peak [$ per kWh]	0.51536	0.40429	0.53002
Total rate winter partial peak [$ per kWh]	n/a	0.3822	0.51332
Total rate winter off peak [$ per kWh]	0.48701	0.36834	0.34462

The results of the annual utility cost savings using each of the utility rates are shown in Table 18. The relative favorability of the E-Elec rate versus the EV2 rate is variable and due in large part to the differences in the partial peak and winter peak rates and how much load is saved within those periods. Overall, the annual utility cost savings can vary significantly by rate and no single rate is superior to the other for all load profiles. It should be noted that PG&E rate schedules change multiple times per year, which would further change the relative economics of the system operation.

Table 18: Annual Utility Cost Savings Modeled by Utility Rate

Utility Rate	E-TOU-C 1/1/2024	E-Elec 1/1/2024	EV2 1/1/2024	Average
Site 1 – metered average analysis	−$85.37	−$52.38	$16.47	−$40.43
Site 2 – metered average analysis	−$245.32	−$97.14	−$5.07	−$115.84
Site 3 – metered average analysis	$179.43	$194.08	$275.30	$216.27
Site 4 – metered average analysis	$347.76	$271.01	$348.90	$322.56
Site 2 – linear regression analysis	$119.60	$102.73	$101.42	$107.92
Site 3 – linear regression analysis	$473.19	$376.48	$400.45	$416.71

Cost-effectiveness

The project team evaluated the cost-effectiveness of the systems with respect to owner's utility cost savings, program incentives, incremental cost to owners, and simple payback for owners. A summary of cost-effectiveness is presented in Table 19 below. Baseline equipment costs were queried from TECH Clean California data for residential installations of split unitary systems (i.e. traditional air-to-air heat pumps) and heat pump water heaters (TECH Clean California Heat Pump Data 2024). The project team selected a heat pump water heater storage volume of 83 gallons to capture the cost corresponding with the installed air-to-water heat pumps for domestic hot water heating only. The average costs in the San Francisco Bay Area (counties of Alameda, Contra Costa, Marin, San Francisco, Santa Clara, Solano, and Sonoma) for these two system types, $24,284 and $12,633 respectively, are added together to represent the type of HVAC and domestic hot water system that would normally be installed, absent a water-based thermal storage system. The efficient cost of the equipment and installation was based on the actual invoiced costs to the owners. The costs of the equipment and installation varied as a function of different installing contractors, variations in system arrangement, supplemental scope, and warranty. Site 1 included costs for two HVAC zones and a 10-year labor warranty in the contract. Site 4 included $5,700 for an outdoor tank enclosure and upgrades to existing ductwork. All installation contracts included labor for installing the metering hardware described in the M&V plan. One of the contracts itemized this cost with an amount of

$800. The incremental cost shown in Table 19 is the difference between the costs without incentives and the baseline system costs. The utility cost savings are based on the E-Elec utility rate.

Table 19: Summary of Cost-Effectiveness

Metric	Site 1	Site 2	Site 3	Site 4
Baseline cost of equipment/installation	$36,917	$36,917	$36,917	$36,917
Efficient cost of equipment/installation	$58,000	$39,000	$49,800	$51,500
Incremental capital cost	$21,083	$2,083	$12,883	$14,583
Utility cost savings ($/yr) – metered average analysis	−$52.40	-$97.10	$194.10	$271.00
Utility cost savings ($/yr) – linear regression analysis	--	$102.70	$376.50	--
Payback (years) – metered average analysis	none	none	66.4	53.8
Payback (years) – linear regression analysis	--	20.3	27.2	--
Incentives	$11,500	$5,300	$5,300	$5,300
Payback w/incentives (years) – metered average analysis	none	none	39.1	34.3
Payback w/incentives (years) – linear regression analysis	--	immediate	16.0	--

All sites received a $5,300 heat pump water heater incentive from TECH Clean California, which includes a $3,100 base incentive plus $700 for a large tank and $1,500 for low GWP refrigerant. Site 1 also received a $4,500 Peninsula Clean Energy incentive, a $900 Golden State incentive, and a $800 BayREN incentive. In addition to the established program incentives, the pilot participants each had the direct cost of their equipment reduced by a Southern California Edison or CalNEXT participation incentive of $28,768. The cost reduction of the incentives was regarded by the project team as essential for securing participation in the pilot study. The incentives may have had a customary effect of higher contractor mark-up in installation costs; however, there is no data to confirm this is the case. In any case, the cost of installation with or without incentives may reduce over time as more contractors include air-to-water heat pump systems in their business model, resulting in economies of scale and increased competition. More than 20 contractors in California have installed more than 140 systems of this type.

The installation process could be streamlined by reducing field labor with factory integration and fabrication. One way to achieve this is to physically integrate the control module with the storage tank, which is expected to be available for systems installed next year. A second opportunity to streamline the installation is to utilize a single reversible air-to-water heat pump in lieu of an air-to-water heat pump for heating only, with an air-to-air heat pump for supplemental heating and for cooling. Further streamlining and field labor reductions could be achieved through factory plumbing and wiring extending from the components. Costs may be lower for new construction installations as compared to the retrofit installations of this study. New construction installations can avoid the cost and complexity of removing and retrofitting existing mechanical systems and upgrading thermal shell performance to achieve an acceptable design load.

The most significant barrier to adoption is the higher differential cost, compared with air-to-air heat pump systems. This is partially addressed by the incentives currently available for the air-to-water heat pump systems in this study and could be addressed further by incentives that value load shift, low GWP refrigerants, and GHG savings. Another barrier to adoption is the high space heating loads. Heating loads exceeding what can be stored and shifted from the peak period can lead to higher consumption in the peak period and lower or negative savings. This can be addressed by thermal shell performance upgrades, higher amounts of storage capacity, and combining with space temperature set-backs for heating during peak periods.

Total System Benefit and Avoided GHG Emissions

The project team estimated the GHG savings of the thermal storage systems by applying the annualized kWh loads to hourly emissions factors. The project team obtained an extract of hourly emissions factors of metric tons of CO_2e/kWh for the year 2024 from the CPUC Avoided Cost Calculator (DER Cost-Effectiveness n.d.). The GHG emissions factors were then applied to the hourly MWh savings from the hourly analyses to obtain an annual estimated GHG savings (see Table 20). The project team also obtained an extract of hourly avoided cost factors in units of $/MWh from the CPUC Avoided Cost Calculator for the year 2030. This year represents the average lifetime cost factors for an approximate expected life of 10 to 15 years. The CPUC-adopted service life of heat pumps and heat pump hot water heaters is 10 years, to be increased to 15 years. The cost factors were then applied to the hourly MWh savings from the linear regressions to obtain an annual and lifetime estimated total system benefit (TSB), as recognized by the CPUC. The lifetime avoided costs represent a public benefit of the thermal storage for consideration in incentive programs.

Table 20: Annual GHG Savings, TSB, and TDV by Site

	Site 1	Site 2	Site 3	Site 4	Average
Annual GHG savings (metric tons CO_2e/yr) – metered average analysis	–0.003	–0.119	0.189	0.324	0.10
Annual GHG savings (metric tons CO_2e/yr) – linear regression analysis	–0.333	0.075	0.360	0.622	0.18

	Site 1	Site 2	Site 3	Site 4	Average
Annual TSB ($/yr) – metered average analysis	$20.42	$164.31	$129.61	$53.07	$91.86
Annual TSB ($/yr) – linear regression analysis	–$51.87	$38.19	$83.85	$110.07	$45.06
Lifetime TSB – metered average analysis	$204.25	$1,643.09	$1,296.15	$530.72	$918.55
Lifetime TSB – linear regression analysis	–$518.70	$381.91	$838.51	$1,100.74	$450.61
Annual TDV (kBtu/yr) – metered average analysis	3713	–2974	15505	25912	10539
Annual TDV (kBtu/yr) – linear regression analysis	–23181	4611	25678	43256.01	12590.94

Table 20 also presents the annual time dependent valuation (TDV) of savings for the sites. The project team obtained an extract of hourly data of kBtu/kWh (UC Davis 2024). The hourly factors (30 year electric residential, climate zone 3) were applied to the hourly kWh savings to obtain annual estimated time dependent valuation. The regressions used CALMAC CZ2022 TMY weather data (California Weather Files 2024), consistent with the application of time dependent valuation to Title 24 compliance calculations in new construction. For Site 1, the total system benefit and the time dependent valuation are opposite in sign to the GHG savings for the metered average analysis. This is due to the relative differences in magnitude of the hourly factors and the load profile for Site 1. A similar result exists for Site 2 GHG savings and total system benefit.

Conclusions

According to prior research studies of residential retrofit applications, standard or flexible heat pumps systems appear to be more cost effective than installing new air conditioning units or expanding natural gas service for home heating. When sized appropriately, they can be used to reduce peak demand by shifting two to five percent of the total energy use to off-peak times and can potentially reduce homeowner costs, but savings depending heavily on the time-of-use rate structure.

The pilot study research findings show that residential, water-based, thermal storage systems can achieve load shift for combination space heating and domestic hot water heating systems, relative to systems without thermal storage. The pilot systems evaluated in this study show a range of annual average peak load shift of 36 percent to 66 percent.

Over the next few years, adoption of electric HVAC and domestic hot water systems is expected to increase and as many as 200,000 homes or 1.3 percent of households in California could adopt water-based heat pump technologies with thermal storage and controls. Thermal storage systems could result in a shift of 41 to 103 GWh of energy annual from peak to off-peak periods, which would result in annual emissions savings between 3,720 and 9,300 MTCO2e.

The pilot system evaluation measured the coefficient of performance efficiency penalty of operating a hot water-based thermal storage system for shifting heating load. The efficiency penalty to the coefficient of performance for thermal storage operation was 15 percent to 21 percent for the evaluated sites. The measured operating coefficient of performances of 2.52 to 3.84 with thermal storage are overall more efficient that a minimally efficient 7.5 HSPF2 heat pump (2.19 average coefficient of performance) combined with a minimally efficient 3.0 UEF heat pump hot water heater. Energy savings were measured for some of the sites, which was most likely attributable to a change in the domestic hot water and cooling load profiles between the baseline and control mode periods.

Customer economics of utility cost savings and payback can vary considerably depending on utility rate, installation cost, and load profile. Providing feedback to system users on the operating cost performance of the load shift could help users identify needed interventions in managing load and selection of the most favorable utility rate. The degree of load shift and cost savings depends on the load of the system which can vary depending on the thermal performance of the building and occupant behavior or usage.

System performance is constrained by the ability to meet heating loads, of which space heating is the largest load. These loads would foreseeably be greater in colder climate zones, depending on thermal envelope performance. Additional research is needed in colder climate zones for evaluating broader application of the system type in California.

Recommendations

The following are recommendations for maximizing the energy efficiency, maximizing the load shift, minimizing the financial costs with respect to system installation and operation, and maximizing the public benefit of the thermal storage systems.

Recommendations for System Vendors

1. Reduce field labor cost and total system cost by integrating control hardware with tank hardware in the factory and by increasing factory packaging of piping and wiring.
2. Reduce complexity, equipment cost, and field labor by integrating heating and cooling in a single air-to-water heat pump.
3. Offer a modular system of storage expansion to enable the installation of additional storage to systems installed with inadequate capacity for load shift, or for which additional load shift duration is desired.
4. Provide a means of performance feedback to users on the operating cost savings, GHG savings, and load shift. A visual display of this data would help users identify any adjustments needed in their load profile to achieve desired savings.

Recommendations for Codes and Standards

1. Develop a performance rating standard for capacity, load shift, and efficiency of air-to-water-based thermal storage systems. Such a standard would enable a competitive market of heat pump thermal storage products and would ensure system selections are sized appropriately.
2. Consistent with the development of a performance rating standard, establish a thermal storage efficiency requirement in the CA Title 24 Building Energy Efficiency Standards for air-to-water-based thermal storage systems. Thermal storage can reduce heating efficiency, and therefore a minimum level of performance should be established. Such a requirement would also support adoption of the technology in new construction projects.

Recommendations for Programs

Utility programs supporting residential thermal storage systems should include incentives of weatherization programs to ensure space heating loads do not exceed the storage capacity for load shift from peak periods. Weatherization program incentives can increase the market of suitable homes for thermal storage systems.

1. Support research to assess the economics of implementing air-to-water heat pump thermal storage systems in colder climate zones where thermal shell performance would foreseeably need to be higher. The systems evaluated in this research were in a relatively mild climate zone for California and with homes able to meet the design heating load limit. Homes requiring thermal shell upgrades would incur higher costs that may warrant additional support from utility programs.

2. Begin the process of sponsorship for measure package development, which would include the discussions and decisions on modelling needed to characterize system performance. Definition of representative load profiles accounting for building types and occupancy will likely be a key challenge, considering the variability of load profiles measured in this study.
3. Link prescriptive incentives to the public benefit of load shift and GHG savings if not already accounted for in existing incentives. Incentive levels could be tied to the total system benefit of the energy shift of the thermal storage operation.
4. Consider incentive support for low GWP refrigerant residential heating and cooling systems, which can be extended to air-to-water heat pump thermal storage systems. Air-to-water heat pumps offer natural refrigerants such as CO_2 in lieu of synthetic refrigerants with much higher GWPs. The GHG savings of leaked refrigerants are not explicitly accounted for in programs and could be in air-to-water heat pump thermal storage programs.
5. Integrate a utility demand response signal for thermostat set point setbacks to supplement load shift capacity and a reduction of load during peak periods. A demand response signal would foreseeably occur in only a fraction of the peak periods each year yet could improve reduction of load during peak periods when most needed by the utility. This can be implemented with a third-party thermostat capable of responding to a demand response signal and through enrollment in the existing demand response program.

Appendix A: M&V Plan Supplemental Information

Introduction

Project Overview

This measurement and verification (M&V) plan is designed to evaluate the efficiency, load shift potential, and cost-effectiveness of water-based thermal energy storage systems for integrated space heating and domestic hot water heating in single family, residential buildings in California. The systems comprise a commercially available air-to-water CO2-refrigerant heat pump, domestic hot water storage tank, air handler (i.e., fan coil unit) containing both a hydronic coil and a refrigerant coil, a supplemental air-to-air heat pump for cooling and heating, and a control system. The evaluation will quantify the potential whole-building energy efficiency, peak demand reduction, and owner economics of the integrated systems.

Intent of Energy Efficiency Measure

Home space heating, space cooling, and domestic hot water heating loads can be served efficiently by contemporary air to water heat pump technology. Higher operating efficiencies and a shift of loads from peak to off-peak demand periods can be achieved through hydronic piping coupled to thermal energy storage.

The control system consists of five primary devices which work together to deliver hot water; a high efficiency air-to-water heat pump, a water tank for thermal storage, a control device, an air handler, and an air conditioning condenser.

1. The control device integrates the entire system and facilitates the storage and delivery of hot water. It optimizes system efficiency by operating the heat pump during the cheapest off-peak times while ensuring that users' heat and hot water needs are always met.
2. The heat pump installed allows the system to function by heating water to 150°F.
3. The air handler installed can be located horizontally in a basement, crawl space or attic, or vertically in a garage or closet with appropriate mounting and bracing. The air handler contains a hydronic coil that is served by the control device for heating, as well as a refrigerant coil that is served by the air conditioning condenser for supplemental heating and cooling.
4. The water tanks installed can be located indoors in closets, garages, basements, or in an outdoor closet. The tank stores water at 150°F to allow the system to deliver more heat than the heat pump's instantaneous capacity.
5. The air conditioning condenser installed is an outdoor fully modulating reversible air conditioning heat pump unit. With the outdoor air conditioning unit, an evaporator coil module is added to the air handler. The air-to-air heat pump will operate to serve peak loads above 24 kBtu/h and to serve space heating loads while the air-to-water heat pump system is charging the thermal storage before the morning and evening peaks. The air-to-air heat

pump will also operate as an air conditioner to serve space cooling loads which are not served by the heating-only air-to-water heat pump system.

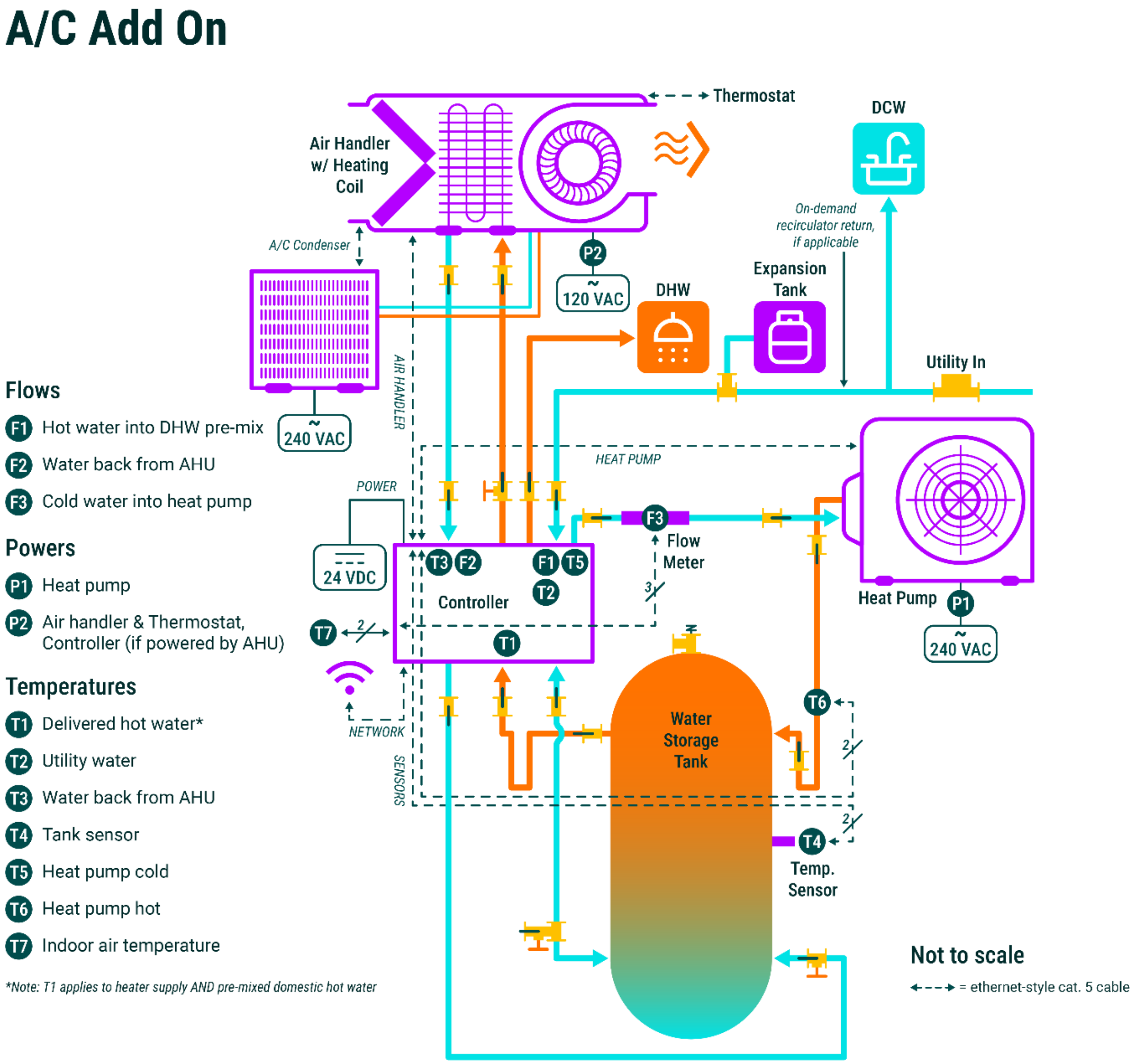


Figure 27: Diagram of control system with supplemental heating and cooling air-to-air heat pump.

Source: Project Team.

We will evaluate the energy efficiency and load shift of thermal storage for combined space heating and domestic hot water heating loads. The air-to-water heat pump and thermal storage will be installed and operated as described in the previous sections.

The load shift of the heat energy storage and discharge will be measured directly by the metering devices. The energy efficiency will be calculated from the measurement of delivered thermal energy and measured input electric energy. The energy savings of the systems will be estimated in comparison to a baseline without thermal storage (i.e. native controls for air-to-water heat pumps without operation of thermal battery).

Some of the installed pilot systems may include an economizer option, which provides cooling via outdoor air ventilation when indoor air temperature is above set point and when the outdoor air temperature is less than indoor air temperature. The energy savings of economizer cooling would be estimated in comparison to a baseline without economizer cooling (i.e. air-to-air heat pump serving cooling loads).

Objectives

The objectives of the measurement & verification of the pilot study installations are:

a. Quantify the loads for domestic hot water heating, space heating, and space cooling.
b. Verify heating and cooling set points are satisfied by the system.
c. Quantify the electric energy consumption of the system.
d. Quantify the energy savings and efficiency of the thermal energy storage system.
e. Quantify the energy cost savings of shifting load from the utility peak demand period.
f. Evaluate the efficiency gain of supplemental heating and cooling via air-source heat pumps.

Methodology & Approach

Our measurement and verification are based on the International Performance Measurement and Verification Protocol (IPMVP) Option B: Retrofit Isolation, All Parameter Measurement (Efficiency Valuation Organization 2022). Our evaluation will use a Normalized Energy Savings approach to account for weather effects on the load and energy savings during the evaluation period. Savings will be estimated based on the development of linear regression models of baseline (non-thermal storage) operation and applied to reporting period independent variables of outdoor air temperature and domestic hot water demand, to calculate the difference in predicted energy consumption between a non-thermal storage system and a thermal storage system.

Data Collection

This section presents the data collection of the evaluation, including the metering points, installation of meters, operational verification of metering, the reporting period, and the baseline period.

Metering Points

Table 21: Metering Points

Parameter	Location	Make and Model	Units	Sampling Interval
AWHP electric power	@ electrical panel tie-in or disconnect	EGauge EG4015	kW, PF	1-minute
AHU electric power	In-line at AHU 120v AC receptacle	EGauge EG4015	kW, PF	1-minute
AAHP electric power	@ electrical panel tie-in or disconnect	EGauge EG4015	kW, PF	1-minute
Static pressure differential	In ducting @ supply and return of AHU	Ashcroft CXLdp	PSI	1-minute
AHU supply temp/rh	In ducting @ supply of AHU	E+E EE160	°F, %RH	1-minute
AHU return temp/rh	In ducting @ return of AHU	E+E EE160	°F, %RH	1-minute
Water flow rate @F1	Hot water into DHW pre-mix	Kamstrup FlowIQ 2100	GPM	64 sec
Water flow rate @F2	Water return from AHU	Kamstrup FlowIQ 2100	GPM	64 sec
Water flow rate @ F3	Cold water into heat pump	Kamstrup FlowIQ 2100	GPM	64 sec
Water temp @T1	Delivered Hot Water	Littelfuse Thermistor	°F	64 sec
Water temp @T2	Utility water	Littelfuse Thermistor	°F	64 sec

Parameter	Location	Make and Model	Units	Sampling Interval
Water temp @T3	Water back from AHU	Littelfuse Thermistor	°F	64 sec
Water temp @T4	Tank sensor	Littelfuse Thermistor	°F	64 sec
Water temp @T5	Heat pump cold	Littelfuse Thermistor	°F	64 sec
Water temp @T6	Heat pump hot	Littelfuse Thermistor	°F	64 sec
Space temp set point	Main Home Thermostat	Ecobee Smart Thermostat	°F	5 minutes
Space temp measured	Main Home Thermostat	Ecobee Smart Thermostat	°F	5 minutes
Space rh measured	Main Home Thermostat	Ecobee Smart Thermostat	RH	5 minutes

Diagram and Description of Metering Points

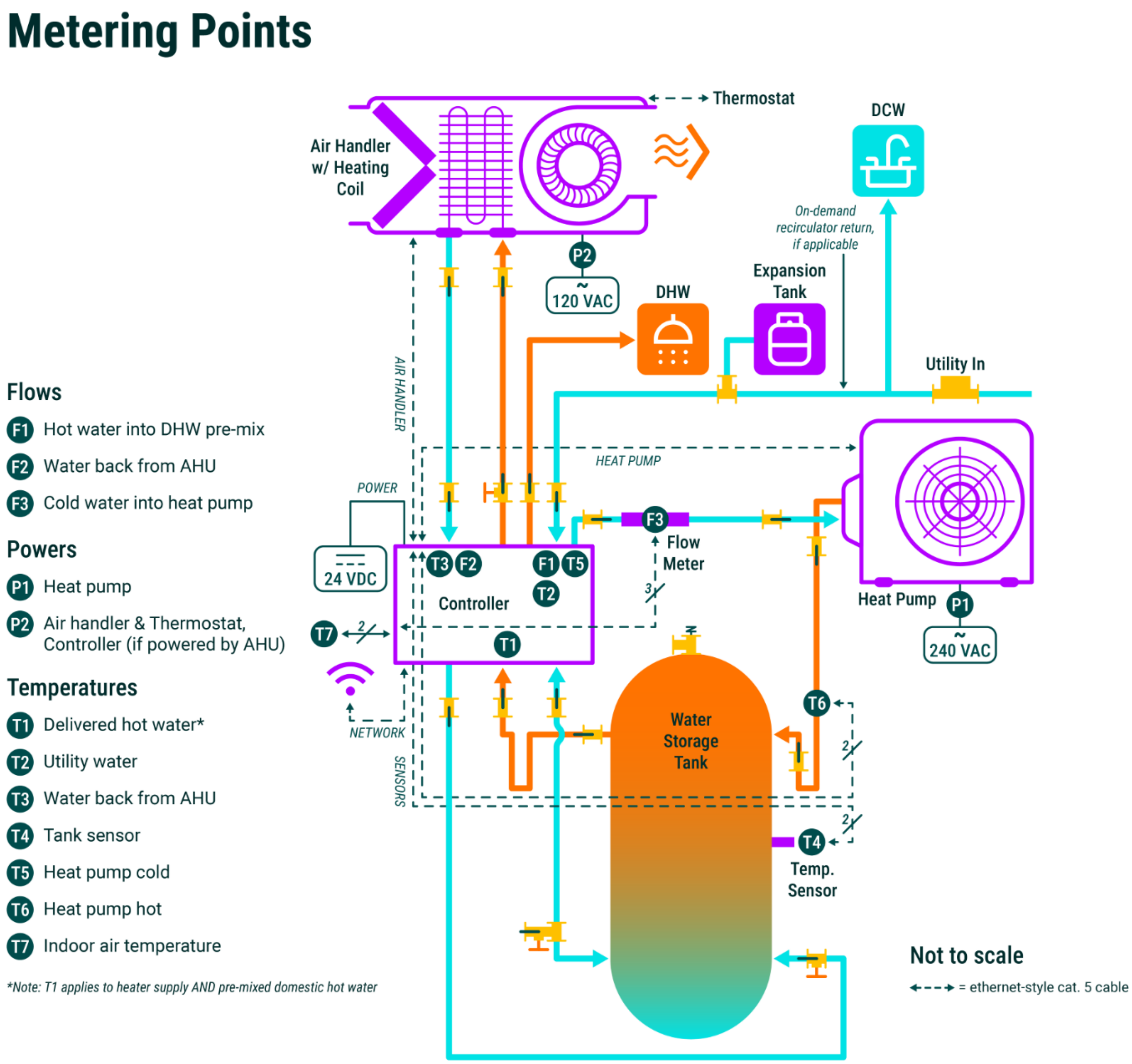


Figure 28: Diagram and Description of Onboard Metering Points.

Source: Project Team.

Description of Meters

- The eGauge Core EG4015 which will be used for power metering, is FCC and UL-61010 compliant, as well as lead-free and RoHS-compliant, with 0.5 percent revenue grade accuracy compliance. The EG4015 has 15 channel ports, and can log V, A, W, Wh, Hz, VA, VAr, THD, and deg.
- The E + E EE160 will be used to capture temperature and relative humidity in the supply and return plenum of the HVAC system. At 23°C, accuracy is ±2.5 percent RH, and ± 0.3 ΔT.
- The Ashcroft CXLdp Differential Pressure Transmitter will be used to capture the pressure differential between the supply and return of the air handling unit. At 21°C it has an accuracy class of ±0.25 percent, ±0.4 percent, ±0.8 percent of span.

Installation of Meters

EGauge 4015

The eGauge 4015 must be installed by a licensed electrician according to all applicable local, national, and international codes. This meter will be installed in a NEMA UL listed enclosure which provides a screw-on and locking cover to prevent accidental contact with eGauge power connections. This meter will be capturing and uploading all data collected.

The eGauge will be powered by a 15A 2-pole breaker in a standard split-phase residential electrical panel.

50A CTs each will be used to meter the hot conductors feeding:

a) the air-to-water heat pump,

b) the air-to-air heat pump, and

c) the air handler.

This metering will be done at the electrical panel where dedicated circuits are tied in for a) and b), and at the receptacle where the air handler is plugged in.

Per the NEC, all CT-leads should be run in conduit between the eGauge enclosure and the electrical panel, or mechanical equipment junction box.

The eGauge will be connected to the internet via a wired LAN connection.

The eGauge will also connect to sensors that are outside of the room containing the eGauge enclosure via shielded Category-5 cable (for example, the CT connected to the air handling unit power supply).

E + E EE160

The E+E EE160 temperature and relative humidity probes will be installed in both the supply and return plenum of the air handling unit. A ½" pilot hole will be drilled into the ducting and a gasketed mounting flange will attach the meter to the ducting.

The 4-20mA data from these probes will be routed to the eGauge via an eGauge EC420 current sensor, and an eGauge Sensor Hub component.

Ashcroft CXLdp

The Ashcroft CXLdp will be installed in both the supply and return plenum of the air handling unit. A ¼" barbed hose fitting will be attached to both plenums. ¼" ID hoses will be run from both plenums to the Pressure Transmitter.

The 4-20mA data will be routed to the eGauge via an eGauge EC420 current sensor, and an eGauge Sensor Hub component.

Diagram and Installation Instructions for Metering Points

There will be two main installation locations for the meters. One will be located in proximity to the Air Handler Unit, and one will be located in proximity to the Electrical Panel where the air-to-water heat pump and air-to-air heat pump are connected to the electrical system.

Detailed installation instructions are located in **Appendix A.**

Below are the installation one-lines for the two metering locations:

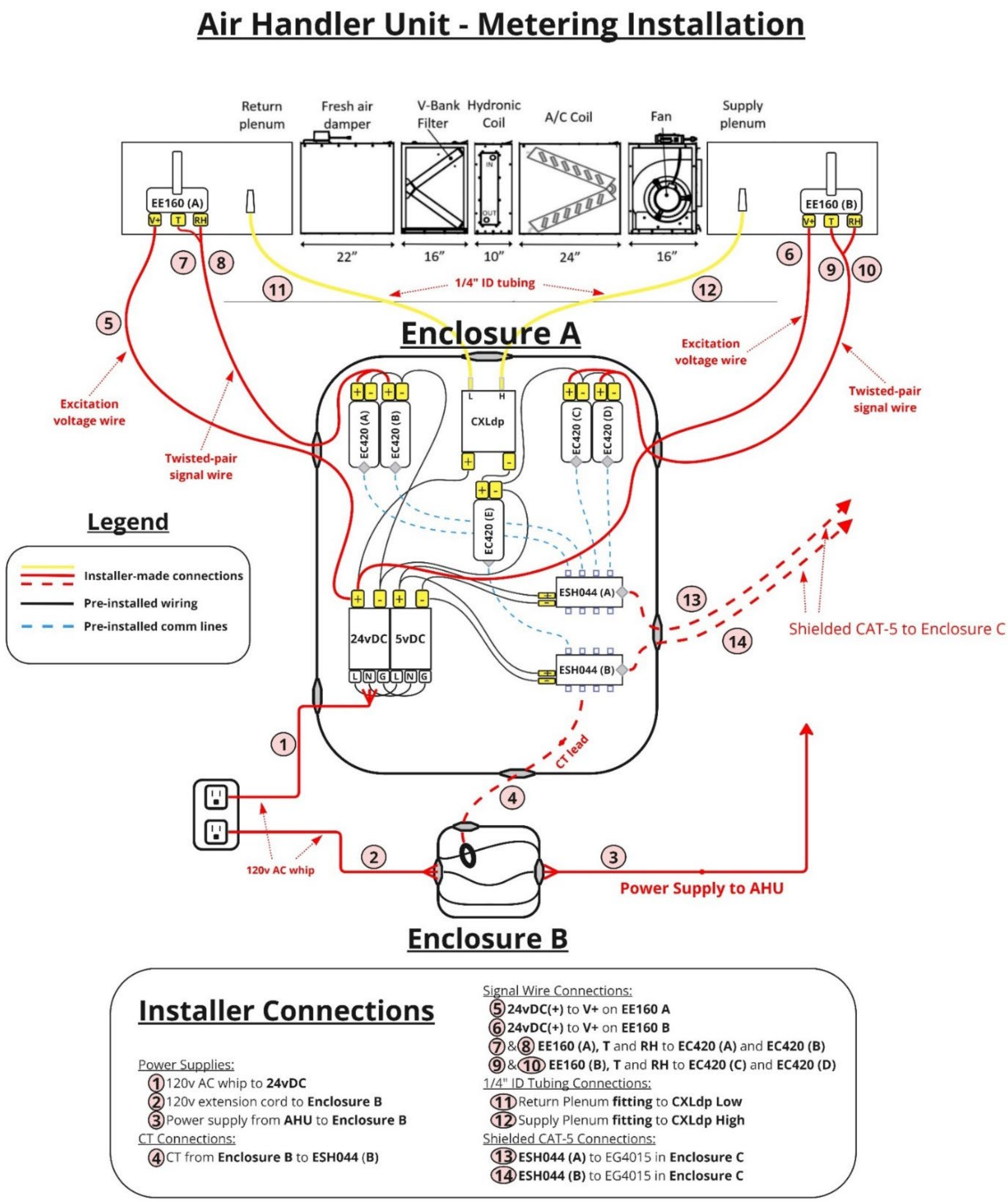
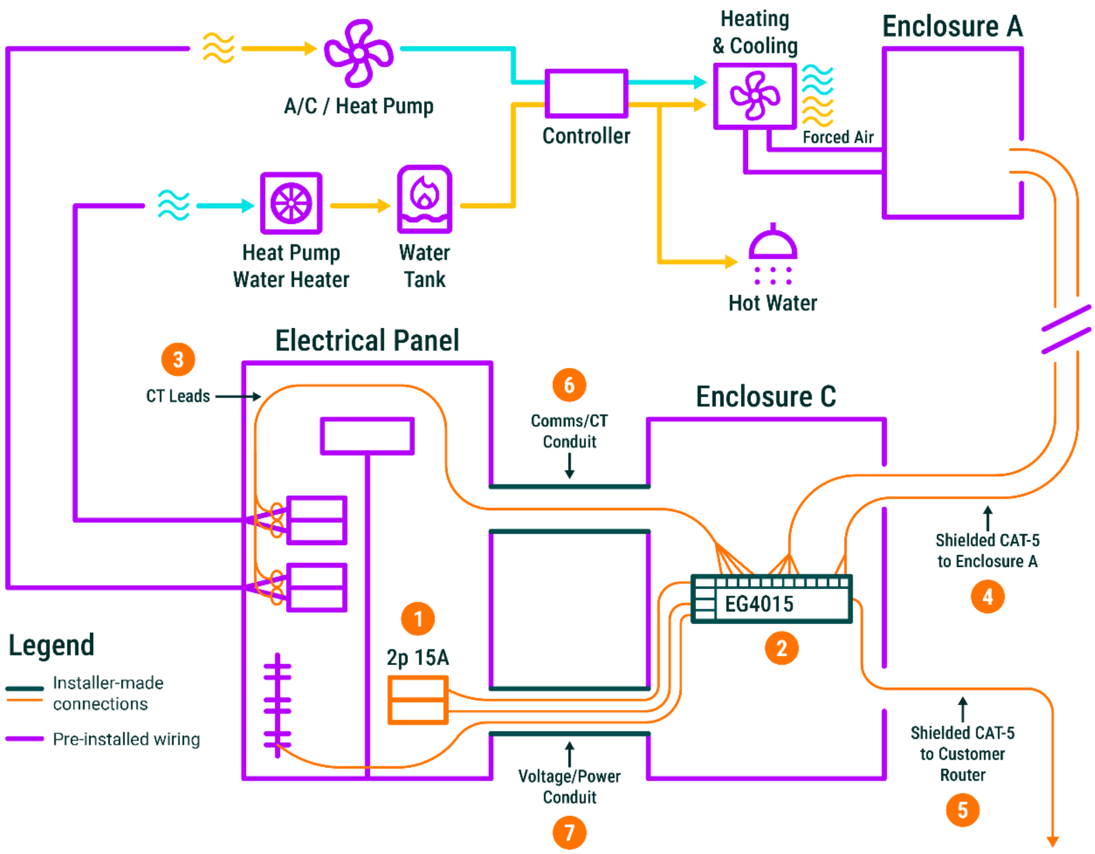


Figure 29: Air handler unit – metering installation.

Source: Project Team



Installer Connections

Power Supplies

1. Install **2-pole 15A breaker**
2. Connect **EG4015** to **2p 15A breaker** and **ground/neutral bar**

CT Connections

3. **4x CTs** from **Heat Pump** and **A/C** conductors to **EG4015**

Sheilded CAT-5 Connections

4. **2x CAT-5** runs from **Enclosure A** to **EG4015**, using provided **Rj-45 to 2-pin adapter**
5. **CAT-5** from **EG4015** to **Customer Router**

Conduit Installations

6. Install **dedicated Comms/CT Conduit** from Electrical Panel to Enclosure C
7. Install **dedicated Voltage/Power Conduit** from Electrical Panel to Enclosure C

Figure 30: Electrical panel – metering installation.

Source: Project Team.

Operational Verification

Upon installation of the Vmeters, commissioning and live verification of all meters will occur. Live data readouts will be verified against secondary meters where possible.

Electrical consumption will be verified against a calibrated ammeter for accuracy, while temperature and humidity sensors will come pre-calibrated from the factory. Differential pressure will be confirmed by comparing the signal output when the air handling unit is turned off, demonstrating no differential pressure, with the signal output when the air handling unit is turned on, indicating a differential pressure.

The eGauge will be connected to an online eGauge dashboard where power readings, Air handling unit temp/RH and differential pressure data will be stored at one-minute frequency. This data will be accessible remotely and can be downloaded at any time the eGauge is connected to the internet.

Reporting Period

The reporting period will include the heating season (winter), the cooling season (summer) and the shoulder season (spring). Reporting will be continuous upon installation of the system and will capture at least two months of data in each season for a total reporting period length of eight months. Installation of the systems is planned for December 2023.

Baseline Period

The baseline periods will occur within the reporting period. The baseline system is one without thermal energy storage. In each of the seasons of the reporting period, the thermal energy storage system will be bypassed for a period of one month.

Analysis

For a description of the analysis procedures and calculations for electric energy consumption, thermal loads, energy savings and efficiency, greenhouse gas (GHG) emissions savings, and energy cost savings, see the study report section Technical Evaluation and Reporting.

Appendix B: Metering Installation Procedure

This is a summary of the installation procedure for the metering being done by the project team.

There are two separate metering installations needed, one being for HVAC/mechanical, and one for electrical. However, all metered device signals will be routed through the eGauge Core 4015 meter and data logger.

HVAC/Mechanical Metering

The HVAC system will need additional metering done by a qualified HVAC/Mechanical contractor.

These are the measures required:

- 1x **Temperature & Relative Humidity** probe installed in the AHU Supply plenum.
- 1x **Temperature & Relative Humidity** probe installed in the AHU Return plenum.
- 1x **¼" male barbed hose fitting** installed in the supply plenum.
- 1x **¼" male barbed hose fitting** installed in the return plenum.

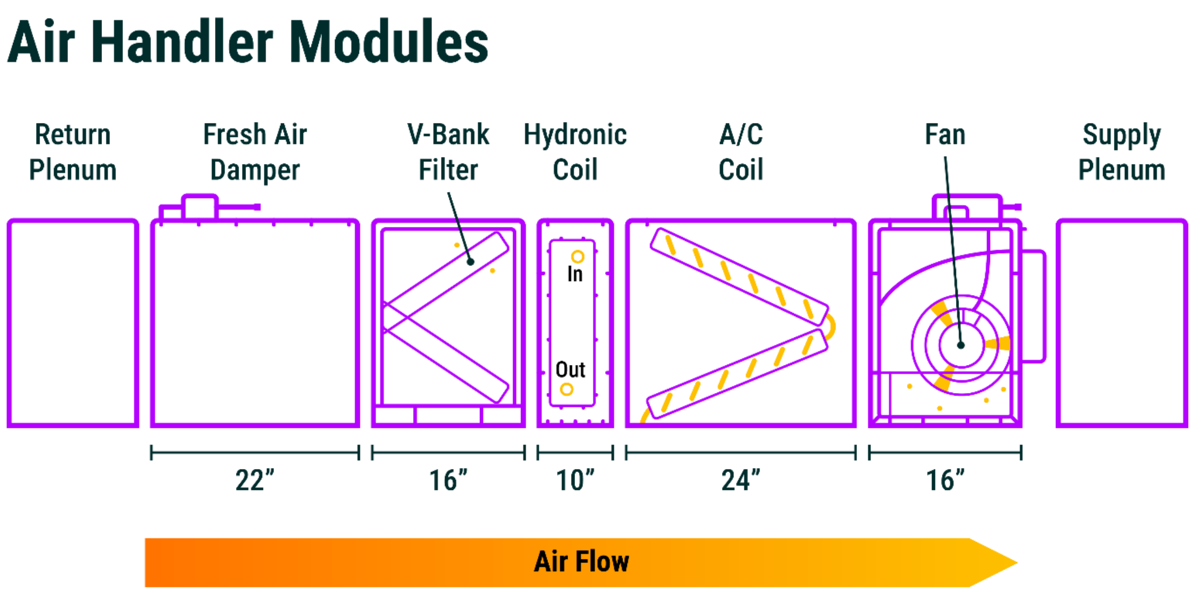


Figure 31: Air handler modules.

Temp & Relative Humidity probe – EE160

This dual function probe installation can be replicated on both the supply and return sides of the AHU.

The EE160 duct-mounted probe should be installed near the center of the ducting, downstream of the fan (on supply side), before any T’s or branches. The probe requires a 16mm (5/8”) pre-drilled hole, and 2x 6mm sheet metal screws. ***Wiring not required.*** Details below (mm/inches):

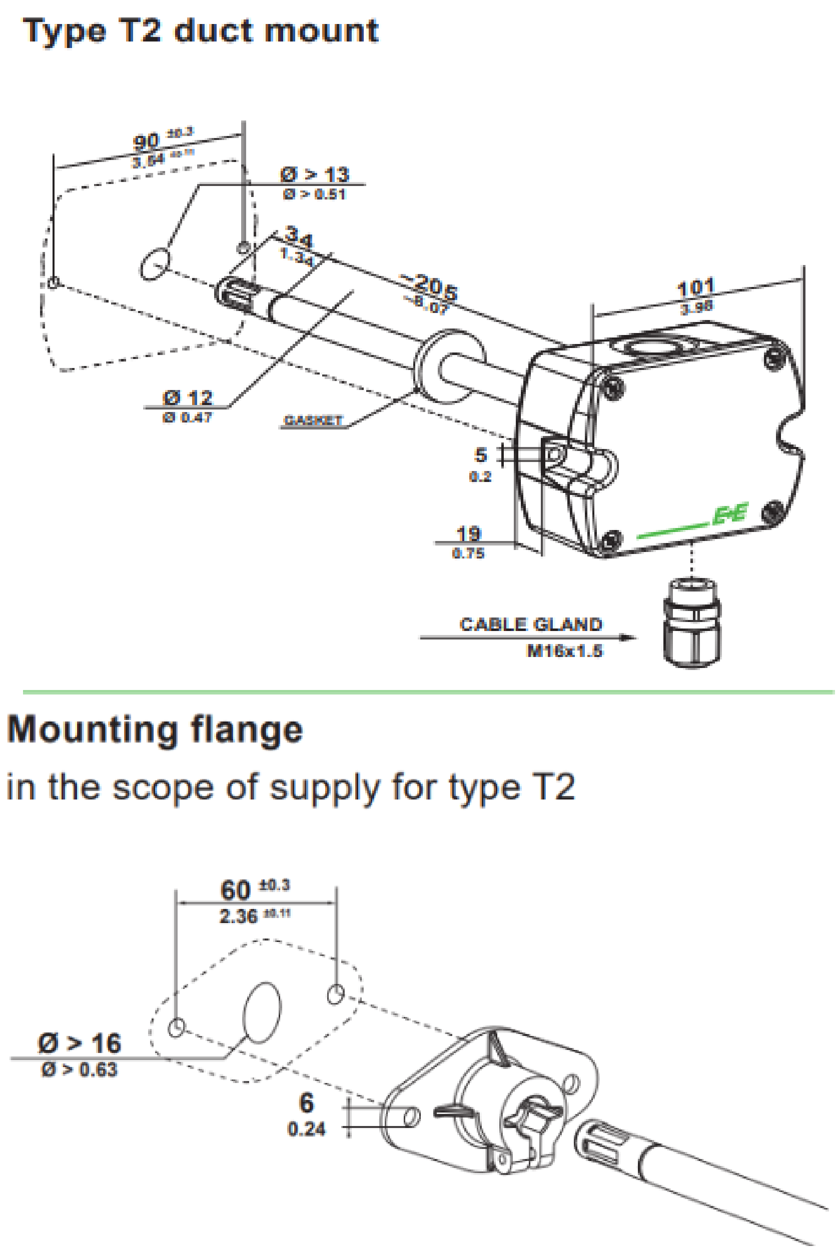


Figure 32: Type T2 duct mount and mounting flange.

¼" ID Male Hose Barb Fitting Installation

For the barbed male hose fitting, install one (1) in both the supply and return plenums (*2 total*) of the AHU with the hose barb facing out. *No tubing will be required.*

Install this within 1' of the Temp/RH probe.

Materials list:

Provided:

- 2 (two) x EE160 Temperature and Relative Humidity duct-mount probes.

- 2 (two) x ¼" ID Male Hose Barb Fittings as described above.

Installing Contractor Provided:

- Time and Materials.

Electrical Metering

Overview

The electrical contractor will be provided with three (3) enclosures.

- Two enclosures (Enclosure A and Enclosure B) will be installed at the Air Handler Unit.
- The third enclosure (Enclosure C) will be installed next to the electrical panel where the heat pump and air conditioning unit are tied in.

***NOTE: If the air-to-water heat pump and air-to-air heat pump AC unit are connected electrically at different locations, a shielded CAT-5 run will be needed to extend the 2x CT leads from the remote tie-in to eGauge Enclosure C. ***

Metering Points

Six) measures will be made at the AHU (Enclosures A and B), and this data will be sent to the eGauge power meter and data logger located in Enclosure C via parallel shielded CAT-5 runs.

Four (4) measures will be made at the electrical panel next to Enclosure C.

Table 22: Metering Points

Parameter	Metered By	Location	Make and Model	Measure	Enclosure
AWHP electric power	Project Team	@ electrical panel tie-in	EGauge EG4015	2x 50A CTs	C
AAHP electric power	Project Team	@ electrical panel tie-in	EGauge EG4015	2x 50A CTs	C
AHU electric power	Project Team	In-line at AHU 120v AC receptacle	EGauge EG4015	1x 50A CT	B
Static pressure differential	Project Team	In ducting @ supply and return of AHU	Ashcroft CXLdp	1x 4-20mA signal	A
AHU supply temp/rh	Project Team	In ducting @ supply of AHU	E+E EE160	2x 4-20mA signals	A
AHU return temp/rh	Project Team	In ducting @ return of AHU	E+E EE160	2x 4-20mA signals	A

The data from these ten (10) measures will be transferred to the internet via a shielded CAT-5 run from the eGauge to the router of the customer.

AHU Enclosures A and B Installation:

Install Enclosures A and B in an accessible space near the Air Handling Unit according to all NEC and AHJ codes.

Enclosure A is 16”x14”x7” and houses:

- An Ashcroft CXLdp differential pressure transmitter
- A 24vDC Autonics power supply
- A 5vDC Autonics power supply
- 5x eGauge EC420 4-20mA signal sensors
- 2x eGauge ESH044 Sensor Hubs

Enclosure B is a 4”x4”x4” box to intercept the power supply for the AHU, to enable metering the power usage of the AHU via a CT within this enclosure.

See installation diagram below.

Connections List

Power supplies:

- **(1)** Connect a **120v AC whip** from an available receptacle to the **24vDC power supply.**
- **(2)** Connect a **120v AC extension cord** from an available receptacle to the **“Line”** side of Enclosure B (this will be a male receptacle).
- **(3)** Connect **existing power supply from** AHU (that was plugged into a receptacle) into the **“Load”** side (female receptacle) on Enclosure B.

CT Connections:

- **(4)** Connect the **CT** that comes in Enclosure B to **Sensor Hub B** located in Enclosure A.

  *******Note – CT leads are 7’ long, so plan enclosure spacing accordingly.***

Signal Wire Connections:

Six (6) total connections are needed. Please use twisted pair for the Temperature and RH signal wire connections.

- **(5) + (6)** Connect a voltage conductor from the **(+) terminal block on the 24vDC** to the **V+** terminal block on the **EE160** unit that is installed on the **Return** Plenum of the AHU. Duplicate this connection for the **EE160** unit that is installed on the **Supply** Plenum of the AHU.
- **(7) + (8)** Connect a twisted-pair signal wire from the **T** and **RH** terminal blocks on the EE160 at the **Return** Plenum to the (+) terminal blocks on **EC420 A** and **EC420 B** in Enclosure A.

- **(9) + (10)** Connect a twisted-pair signal wire from the **T** and **RH** terminal blocks on the EE160 at the **Supply** Plenum to the (+) terminal blocks on **EC420 C** and **EC420 D** in Enclosure A.

¼" ID tubing Connections:

- **(11)** Connect **¼" ID tubing** from the ¼" barbed male fitting on the **Return** Plenum to the Ashcroft CXLdp **Low** fitting in Enclosure A.

- **(12)** Connect **¼" ID tubing** from the ¼" barbed male fitting on the **Supply** Plenum to the Ashcroft CXLdp **High** fitting in Enclosure A.

Shielded CAT-5 Connections:

- **(13) +(14)** A parallel run of **2x shielded CAT-5** (RJ-45) connections needs to be made between the Sensor Hubs A + B in Enclosure A, and the eGauge 4015, located in **Enclosure C**.

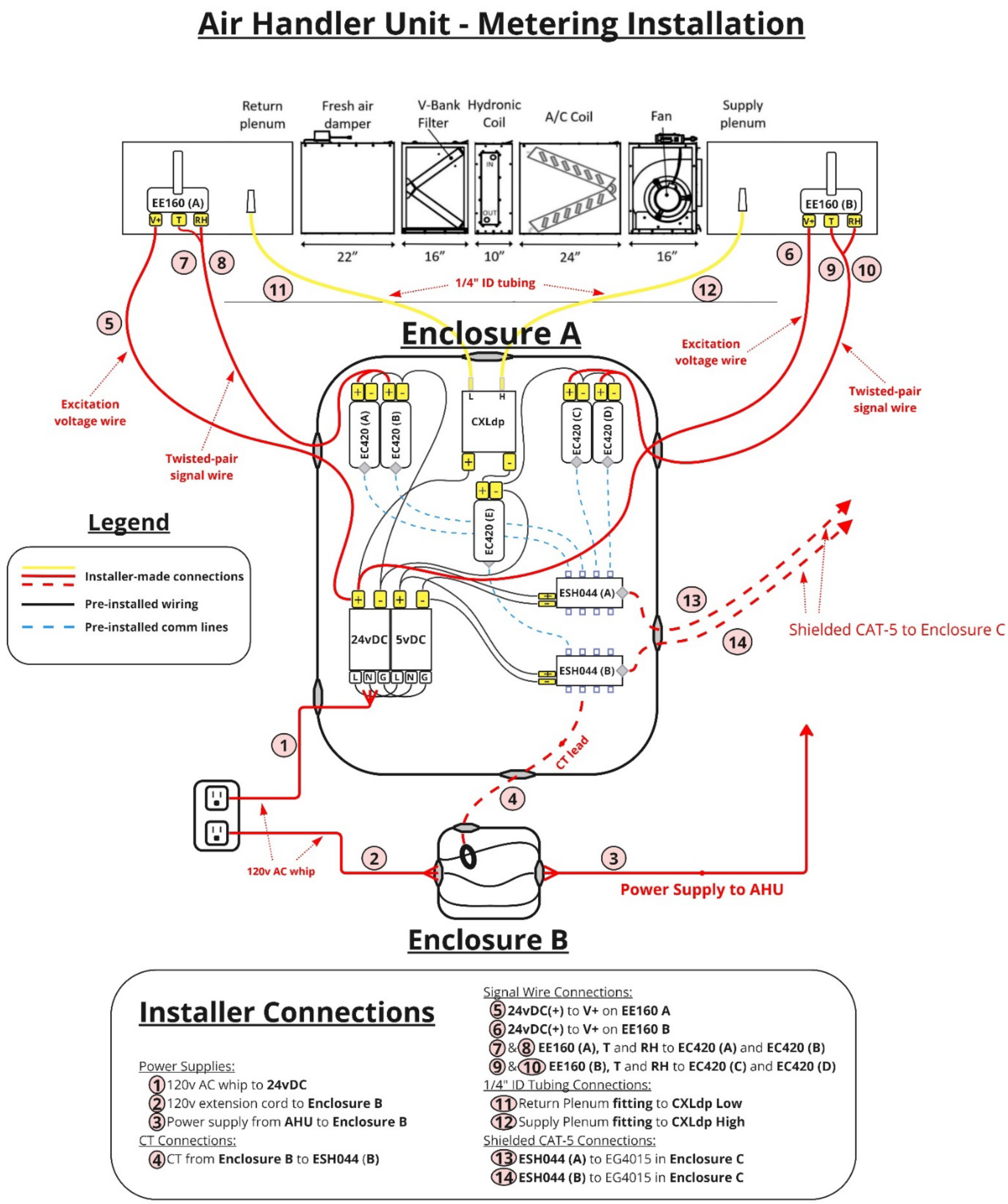


Figure 33: Air handler unit – metering installation.

Power Metering Enclosure C Installation:

Enclosure C should be installed at the MSP or Subpanel where either or both of the air-to-water heat pump or air-to-air heat pump AC unit dedicated circuits are connected to the electrical system.

Install Enclosure C at an accessible space near the electrical panel according to all NEC and AHJ codes.

Enclosure C is 10”x8”x4” and houses:

- The eGauge 4015 Power Meter and Data Logger
- Whip connections that convert 2x CAT-5 runs from Enclosure A to 2-pin terminal block connectors.

Connections list:

Power Supply:

- Install a dedicated 2-pole 15A breaker in the electrical panel being tied-in to. Run appropriate sized conductors from this breaker, and from the ground OR neutral bar to Enclosure C via a **dedicated conduit.**
- Connect the Phase A conductor to the L1 terminal on the eGauge, Phase B conductor to L2, and the neutral or ground conductor to N. Leave the L3 terminal on the eGauge unused.

CT Connections:

- Install 2x 50A CTs in the conductors feeding the 2-pole 15A breaker of the air-to-water heat pump and the on conductors feeding the 2-pole 40A breaker of the AC unit.
- Install a separate dedicated conduit for the four (4) CT leads from the electrical panel to Enclosure C.

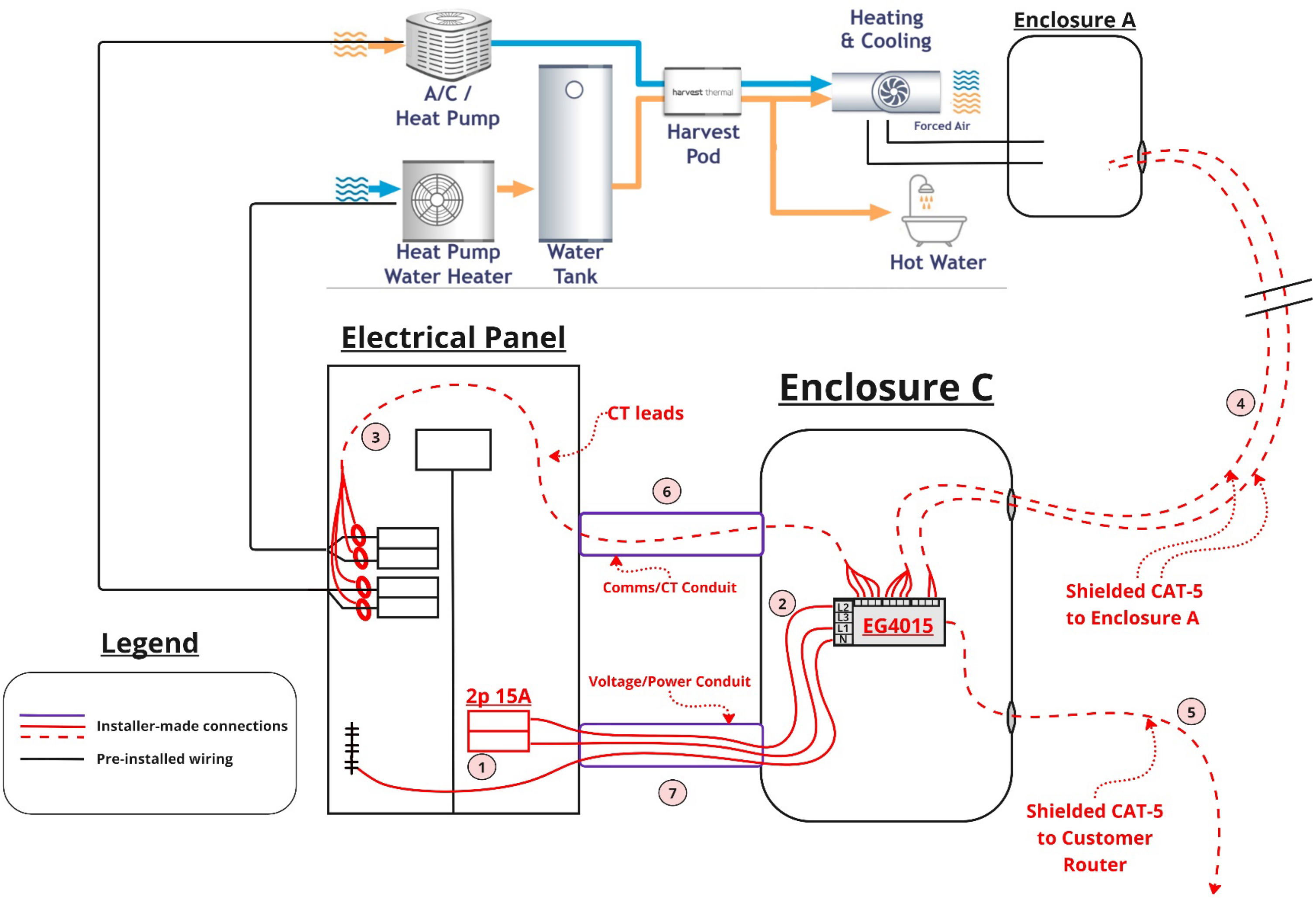
***NOTE - If the air-to-water heat pump and air-to-air heat pump AC unit are connected electrically at different locations, a shielded CAT-5 run will be needed to extend the CT leads from the remote tie-in to eGauge Enclosure C. ***

Shielded CAT-5 Connections:

- A parallel run of **2x shielded CAT-5** connections will be needed between the AHU **Enclosure A** and the eGuage **Enclosure C.**
- **Converters from CAT-5 to 2-pin** terminal blocks will be provided to enable landing the signals from the Sensor Hubs in Enclosure A to be attached to the sensor ports of the eGauge in Enclosure C.
- **One (1) shielded CAT-5** run will need to be made from the eGauge 4015 Ethernet-port to an available LAN port on the router of the customer.

- As noted above: If the air-to-water heat pump and air-to-air heat pump AC unit are connected electrically at different locations, a shielded CAT-5 run will be needed to extend the CT leads from the remote tie-in to eGauge Enclosure C.

Electrical Panel - Metering Installation



Installer Connections

Power Supplies:

1. Install **2-pole 15A breaker**
2. Connect **EG4015** to **2p15A breaker** and **ground/neutral bar**

CT Connections:

3. **4x CTs** from **Heat Pump** and **A/C** conductors to **EG4015**

Shielded CAT-5 Connections:

4. **2x CAT-5** runs from **Enclosure A** to **EG4015**, using provided **RJ-45 to 2-pin adapter**
5. **CAT-5** from **EG4015** to **Customer Router**

Conduit Installations:

6. Install **dedicated Comms/CT Conduit** from Electrical Panel to Enclosure C
7. Install **dedicated Voltage/Power Conduit** from Electrical Panel to Enclosure C

Figure 34: Electrical panel – metering installation.

Materials list:

Provided:

- Enclosures A, B and C, with meters and wires installed and labeled (as per diagrams).
- All meters and CTs.
- EE160 Temp/RH probes will be installed in AHU supply and return by HVAC contractor, but not wired.
- ¼" Barbed Male fittings for pressure differential will be installed in AHU supply and return by HVAC contractor, but not connected with tubing.
- Tubing for electrical contractor to install after Enclosure A is installed.
- AHU will have an existing power cable to be plugged in to Enclosure B.
- Two (2), 19" RJ-45 to 2-pin breakout cables for connecting CAT-5 runs from Enclosure A to 2-pin sensor inputs on the eGauge 4015.

Installing Contractor Provided:

- **Shielded CAT-5** for three (3) runs, potentially four (4) if Heat Pump and AC Unit are tied-in at separate panels.
- **Conduit** for two (2) separate runs from electrical panel to eGauge (one for power conductors, one for CT twisted-pair wires).
- **2-pole 15A breaker** for dedicated eGauge power.
- **Conductors** from dedicated 2-pole 15A eGauge breaker and neutral/ground to the eGauge line-in ports.
- **120vAC power whip** from receptacle to 24vDC power supply in **Enclosure A**.
- **120VAC extension cord** from receptacle to Enclosure B.
- Fasteners, tools.

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