Electrifying Large Commercial Central Plants: Demonstration of TIER and Program Delivery Solutions

Final Report

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Prepared by:

Hwakong Cheng, Curtis Fong, Titus Ellison, Brandon Gill
Taylor Engineers

Bryan Boyce, Rezvan Mohammadiziazi, Sara Sultan
Energy Solutions

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Executive Summary

Commercial HVAC systems have long relied on gas-fired heating paired with separate chilled-water, but the industry has yet to establish a clear, effective pathway for fully electrifying large central plants. Engineers face steep learning curves with new equipment, and the complexity of integrating all-electric components has slowed progress at a time when rapid building decarbonization is urgently needed.

Early all-electric designs have typically relied on air-to-water heat pumps (AWHPs), but these systems often struggle to meet performance and cost objectives. AWHPs have limited cold-weather efficiency, narrow operating ranges, and large space requirements that make them difficult to retrofit into existing buildings. Heat recovery chillers (HRCs) can achieve higher efficiencies—but only when simultaneous heating and cooling loads are available—leading to common design oversights and misapplications on first-generation projects.

Time Independent Energy Recovery (TIER) is an innovative approach that integrates heat-recovery chillers, thermal energy storage (TES), and AWHPs into a coordinated, all-electric plant. Unlike traditional TES applications, which focus on shifting peak cooling loads, TIER uses TES to store and deploy heat recovery, enabling a cascading system that maximizes overall efficiency. TIER is estimated to reduce energy use by up to 40 percent compared with standard AWHP plants, while also reducing space requirements, first costs, and operating costs and supporting grid-interactive building strategies.

This project evaluated the first operational TIER installation, which serves a 1.2 million square foot office building in Sunnyvale, California (Climate Zone 4). The evaluation confirmed that the system delivers higher performance than both conventional AWHP plants and its own modeled predictions. The installed TIER system achieved an average combined heating and cooling COP of 5.5, compared to 4.0 for a comparable baseline AWHP plant. These results reinforce TIER's advantages in energy efficiency, compactness, cost-effectiveness, and grid support.

Because current building-simulation platforms cannot model complex central plants like TIER, the project team developed simplified spreadsheet tools to analyze TIER configurations using either hot-water or condenser-water storage. Modeling across three California climates and two building load profiles showed that TIER consistently reduces energy costs relative to chiller/boiler baselines, overcoming typical all-electric "spark gap" cost penalties. One of the project goals was to provide generalized guidance on TIER plant design for different applications. However, a life-cycle cost analysis of different TES media and TES sizing did not indicate any strongly preferred approaches for the different building loads or climate zones evaluated; optimal solutions depend on site-specific factors such as load profiles, part-load equipment performance, utility rates, and opportunities to repurpose existing firewater tanks for storage.

While TIER has clear performance advantages, barriers to widespread adoption include system complexity, modeling software limitations, lack of standard design approaches, and lack of understanding and awareness. To address this, the project produced a design guide, consisting of example schematics, sequences of operation, and modeling tools to support engineers considering

TIER systems. Lessons learned from early installations and from the analytical work included in this report can further improve implementation success.

Finally, the report outlines a coordinated market transformation strategy to accelerate adoption. Interventions include workforce training, improved modeling tools, manufacturer engagement, design guidance, incentives, and supportive policy pathways. These actions focus on reducing perceived risk, simplifying design, and increasing industry familiarity—creating a roadmap for evolving TIER from an emerging solution to a mainstream, standardized approach for large all-electric HVAC plants.

Abbreviations and Acronyms

Acronym	Meaning
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
AWHP	Air to water heat pump
BAS	Building automation system
BEM	Building energy models
BPS	Building performance standard
Cal TF	California Technical Forum
CBECC	California Building Energy Code Compliance
CDD	Cooling degree days
CEC	California Energy Commission
CEDA	California Energy Design Assistance Program
CHW	Chilled water
COP	Coefficient of performance
COPc	Coefficient of cooling performance
COPh	Coefficient of heating performance
CPUC	California Public Utilities Commission
CW	Condenser water
EE	Energy efficiency
ET	Emerging technology
HDD	Heating degree days

Acronym	Meaning
HRC	Heat recovery chiller
HW	Hot water
HVAC	Heating, ventilation, and air conditioning
IOU	Investor-owned utility
IPLV	Integrated part-load value
kBtu/h	Thousand British thermal units per hour
kW	Kilowatt
LCC	Life cycle cost
NMEC	Normalized Metered Energy Consumption
PA	Program Administrator
PG&E	Pacific Gas & Electric
PLR	Part load ratio
QPL	Qualified product list
RA	Resource acquisition
SCE	Southern California Edison
SGIP	Self-Generation Incentive Program
TES	Thermal energy storage
TIER	Time independent energy recovery
TSB	Total system benefit
TOU	Time of use
3P	Third-party

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Introduction

As interest and demand grows for electrifying and decarbonizing buildings, the heating, ventilating, and air-conditioning (HVAC) industry faces new challenges and barriers in achieving these goals for large commercial buildings. Typical all-electric plant solutions are expensive, require a large footprint, and are not particularly efficient. Many early all-electric installations have also faced challenges from design and installation issues due to equipment design failures, misapplication of rapidly evolving equipment, unfamiliar design considerations, and lack of experience with new equipment constraints. This project aims to advance a novel all-electric central heating and cooling plant design approach that promises to reduce first cost, footprint, and improve energy efficiency compared to alternative approaches. Performance and operational data from the first installation of the time independent energy recovery (TIER) plant are evaluated to confirm the design concept and illustrate advantages over alternative all-electric strategies. A parametric life cycle cost (LCC) analysis will help illustrate design conditions that favor certain system approaches. This project will also develop a design guide, simplified simulation tools, and market transformation recommendations to support the HVAC industry in effectively developing all-electric central plant solutions.

This report presents the project background and objectives, a description of the project methodology and approach, and a summary of findings and recommendations for future steps.

Background

Standard HVAC designs for large commercial buildings rely on separate gas-fired hydronic heating and chilled water (CHW) loops. This technology has slowly been refined with incremental energy efficiency (EE) improvements over several decades and there are existing resources for design and control best practices (Taylor, 2017; ASHRAE, 2024; Cheng, Wendler, & Raftery, 2024). Nevertheless, for hot water (HW) plants, recent studies have uncovered significantly lower operating efficiencies than commonly understood, and some common pitfalls in the design and operation of boiler plants (Raftery, Geronazzo, Cheng, & Paliaga, 2018; Raftery, Singla, Cheng, & Paliaga, 2024; Cheng, Wendler, & Raftery, 2024). For all-electric plants, equipment offerings and design considerations are new, and the HVAC industry has not yet established how to serve heating loads effectively and energy efficiently for large commercial buildings (e.g., generally 150,000 square foot and larger). Many engineers lack the expertise and time to keep up with new and rapidly evolving equipment needed to overcome the novel design challenges to combine the plant equipment into functional systems. To meet the urgent need to rapidly decarbonize buildings, the HVAC industry needs targeted support to overcome these design challenges and barriers around system complexity and integration.

The first generation of all-electric large buildings has relied upon air-to-water heat pumps (AWHPs), an approach that has had difficulty meeting design objectives and is cost prohibitive for most building owners.

- AWHPs are inherently inefficient during cold weather. While a design coefficient of heating performance (COPh) above two is possible in mild West Coast climates, efficiency and capacity both drop rapidly as ambient temperature falls due to higher compressor lift requirements and the need to run the heat pump cycle in reverse to defrost the evaporator coil.

- AWHPs generally have limited operating ranges, with most units only able to produce 130–140 °Fahrenheit (F) HW supply temperatures, which is often incompatible with existing HW distribution and equipment in buildings. Furthermore, COPh decreases as AWHPs generate higher HW temperatures.
- AWHPs are expensive per unit of capacity (roughly $150 to $200 per thousand British thermal units per hour [kBtu/h] of capacity, compared to $15 to $30 per kBtu/h for condensing gas boilers), and have a large footprint, which makes that approach infeasible for existing buildings with limited space.

Heat recovery chillers can operate at higher efficiencies to generate both hot and CHW, but only when simultaneous heating and cooling loads are available.

Many early projects have struggled with the misapplication of new all-electric equipment and critical design oversights. For example, the design of a plant serving an academic laboratory building with two two-pipe AWHPs sized sufficiently for peak heating and cooling conditions did not properly consider that a single AWHP would need to meet cooling loads under most conditions because the other would be operating in heating mode. Another all-electric plant was designed with an AWHP and electric boiler backup. The electric boiler was needed more often than expected, but the return water temperatures from the boiler were too high for the AWHP to operate, so the equipment could only operate in non-integrated mode, rather than in series, where the electric boiler was not sized for the peak heating demand. Control-related issues are also a common challenge. A modular AWHP with primary-only flow distribution had pumps controlled to maintain a pressure setpoint. Though the pressure setpoint was maintained and the supply temperature was maintained at its minimum, the coils were starved despite additional modules being available for staging. A recent study has also documented lower than expected operating efficiency and implementation and control issues with early all-electric heat pump plants (Weitze, 2024).

Time Independent Energy Recovery

TIER is an innovative all-electric central heating and cooling plant concept (Figure 1) that integrates heat recovery chillers (HRCs), thermal energy storage (TES), and AWHPs to overcome the shortcomings of alternative all-electric plant configurations (Gill, 2021) (Stein & Gill, 2024). Traditional TES is used to shift or reduce peak cooling loads, whereas TIER leverages TES for heat recovery. The result is a cascading all-electric system that maximizes heat recovery and effectively deploys plant equipment to maintain the highest system efficiencies. TIER is estimated to provide energy savings of 40 percent compared to the current state-of-the-art all-electric central plants. The first implementations of the TIER plant concept use condenser water (CW) as the TES medium but alternative configurations may use HW storage instead (Figure 2).

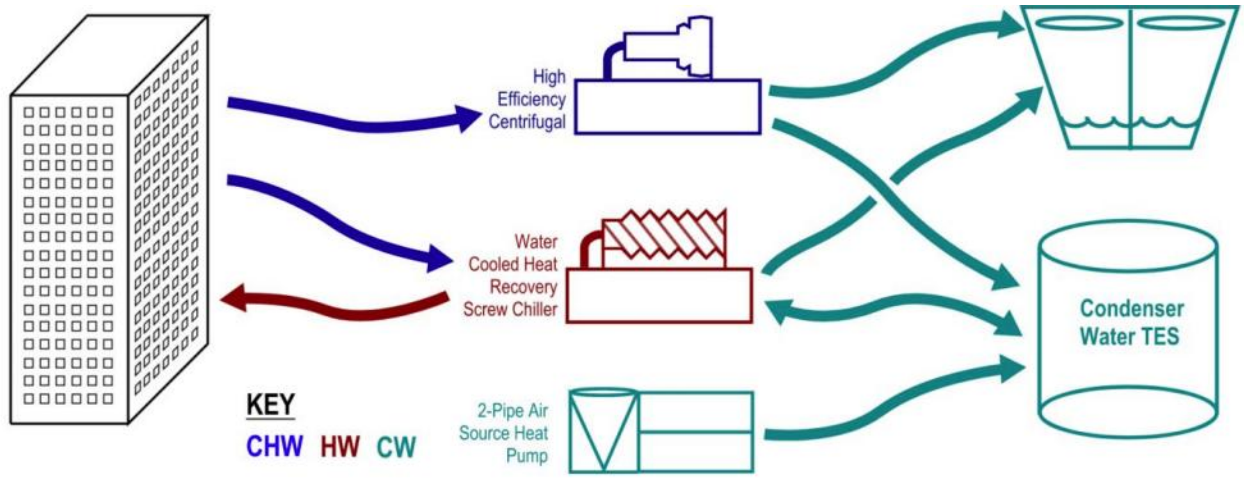

Figure 1: Condenser water TIER system diagram.

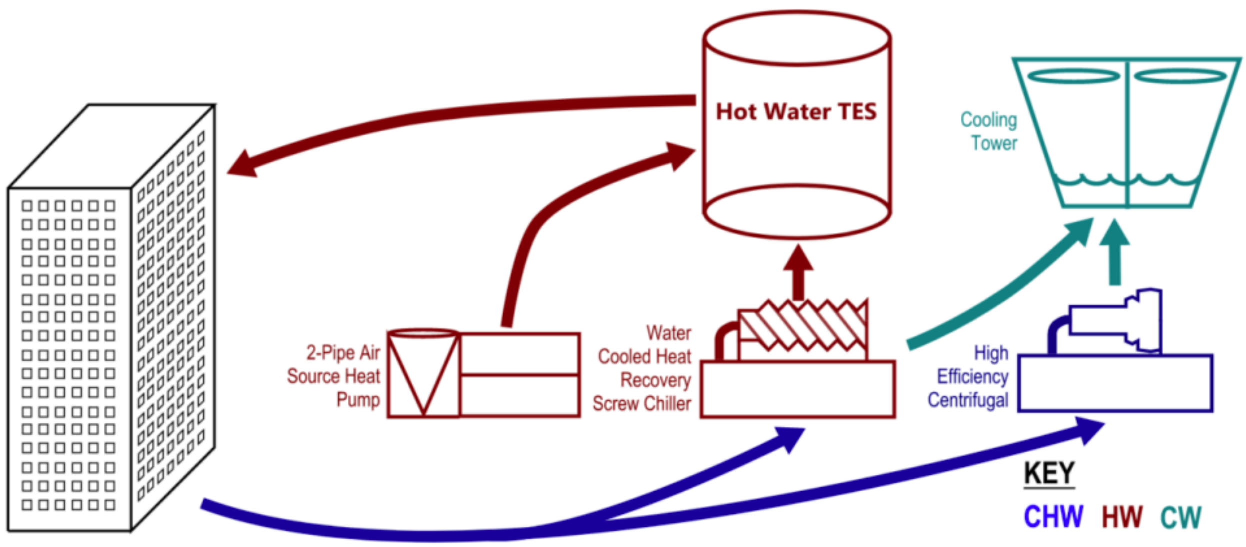

Figure 2: Hot water TIER system diagram.

Though TES can have large space requirements, when sized appropriately in a TIER plant, the heat recovered in storage allows for dramatic reductions in required AWHP capacity, significantly reducing plant footprint and first cost (Gill, 2021). In California's mild climates, the energy recovered from cooling loads alone can satisfy heating loads for most of the year. The AWHPs only operate when needed and up to an intermediate temperature (for the CW approach), which allows for higher efficiencies. Overall, the TIER design offers the opportunity to save space, improve EE, support grid-interactive efficient building initiatives, and reduce first costs and operating costs compared to alternative all-electric plant approaches.

Each of the equipment types in a TIER plant are commercially available, however, the proper design and control of the built-up central plants is complex and challenging to implement. Validation of the novel design concept in real-world applications is needed to confirm its theoretical advantages and support the successful deployment in future plant designs.

Objectives

The primary goal of this project is to provide direct industry support for a rapid and successful shift to decarbonizing large commercial buildings. Specifically, this project leverages an in-depth performance review of the first-of-its-kind TIER plant for a new large commercial building to confirm its real-world performance and to evaluate the application of the novel design concept. Operating data show significant energy efficiency gains over a comparable baseline all-electric central plant. Limitations with existing building energy simulation software are a significant barrier to evaluating TIER plant opportunities. This project aims to help overcome those barriers by developing simplified modeling tool for designers to evaluate alternative plant configurations and energy performance, and by using those tools to conduct parametric LCC analyses. Findings from the LCC analyses, recent research, and from project experiences are incorporated into a design guide that will support the HVAC industry with all-electric plant applications. Lastly, market transformation research has identified barriers to adoption, interventions that will help overcome those barriers, and recommendations for utility program opportunities that will further encourage the application of TIER designs and help with scaling.

Methodology and Approach

The project was comprised of three main tasks:

Task 1: In-Depth Performance Review of TIER Plant

The Project Team reviewed performance data from a TIER plant in operation to help confirm the overall design concept and validate the complex control sequences under a wide range of operational and seasonal conditions. Traditional commissioning efforts generally only verify performance during the seasonal conditions immediately following installation. This in-depth review provided an opportunity to evaluate the TIER plant performance through other seasonal conditions to confirm that the plant control is effective under different modes of operation. Issues observed during the review led to adjustments to the control sequences and tuning parameters. The review of the plant performance was led by a third party to avoid any potential conflict of interest.

One metric used to evaluate the performance of the TIER plant is the coefficient of performance (COP), which is the unitless ratio of a system's output to energy input. In this project, the heating COP and the cooling COP of the TIER plant and the baseline plant were used to evaluate the performance of the TIER plant as function of various factors and conditions. The performance of the TIER plant was compared against multiple reference AWHP plants as well as a spreadsheet model (described in Task 2). of the expected performance

As the system energy input and energy output are often asynchronous because of the heat recovery TES, COP was evaluated on a daily basis. The heating COP was calculated by dividing the average daily heating output required to meet building heating demand by the average electrical input to all heating-related system components. These components include AWHPs, HRCs when operating in heat recovery mode, chillers operating for heat recovery (if applicable), CW pumps, and HW distribution pumps. This approach captures plant-level heating efficiency by accounting for both primary equipment and associated auxiliary energy consumption. Similarly, the daily cooling COP was determined by dividing the average daily cooling output by the average electrical input to all cooling-related system components. The total cooling system power includes chillers operating for cooling, HRCs operating in cooling mode, cooling towers, CHW pumps, and cooling tower pumps. All required input parameters, including temperatures, flow rates, loads, and equipment power were obtained from the TIER plant and baseline plant building automation system (BAS).

To evaluate performance under varying weather conditions, outdoor air temperature was obtained from the building automation system (BAS) and used to calculate daily heating degree days (HDD, base 65°F), cooling degree days (CDD, base 65°F), and average daily outdoor air temperature. These metrics allowed for assessment of COP trends as a function of outdoor temperature and weather severity.

In addition, normalized peak load ratios were developed by dividing daily total heating and cooling loads by their respective maximum observed values during the study period. This facilitated load-binned analyses to evaluate part-load and peak-load performance behavior.

Task 2: Development of a Design Guide

The Project Team developed a design guide for large all-electric heating and cooling plants. In general, all-electric heating and cooling plants involve design considerations that are relatively new and unfamiliar to the HVAC industry. Many early plant designs have failed or struggled due to unanticipated conditions or misunderstandings with how to apply new HVAC products, including misapplication of heat recovery equipment. Many other new plants have been built at high installation costs and with relatively low EE. To support the rapid industry pivot to all-electric designs, the Project Team developed a design guide that addresses key design issues and describes shortcomings and lessons learned from early design failures.

The design guide addresses lessons learned and design considerations for all-electric plants in general based on interviews with designers, commissioning providers, and building owners. Unlike other mature HVAC equipment markets where equipment capabilities have converged across different manufacturers and are well characterized and understood, the AWHP market is still immature and rapidly developing. Product lines from different manufacturers have widely varying capabilities and limitations, and effective control of the varying offerings is not yet well established. Although demand for all-electric HVAC equipment is rapidly growing, manufacturers recognize these challenges of effectively conveying equipment capabilities and limitations to designers and ensuring that new equipment is being deployed in appropriate applications.

The guide provides direction on potential applications for the TIER plant and generalized guidance on key electrification design concepts such as equipment and thermal energy storage sizing based on a LCC study of different plant configurations for a range of building types and climates. The guide includes a case study of the TIER field demonstration from Task 1, including reporting of performance metrics and an illustration of its operation under varying conditions.

Conventional building energy simulation tools like EnergyPlus are not yet capable of modeling complex all-electric central plants like the TIER plant. This presents a significant barrier for applying TIER on real projects. To help overcome this challenge, simplified spreadsheet modeling tools were developed as a companion to the design guide for CW and HW versions of TIER. These tools were leveraged to support LCC analyses and develop recommendations for a range of plant configurations and applications. The spreadsheet models use annual CHW and HW load profile data that may either come from measured data from real buildings or from separate energy simulations. The tools incorporate key plant design factors including detailed equipment sizing and staging data. Equipment performance of water-cooled chillers, heat recovery chillers, cooling towers, AWHPs, and pumps is modeled with detailed performance curves or look up tables. Simplified equipment staging logic are used to deploy the various components at each hour of the year.

The Project Team used the spreadsheet tools to conduct a parametric LCC analysis of different TIER plant configurations for a range of California climates and building types with varying load profiles. The following factors were varied:

- California climate zones: 3 (Oakland), 9 (Los Angeles), and 12 (Sacramento)
- Building types: office, office + 50-ton IT baseload
- TES media: CW, HW
- TES capacity (tradeoff with AWHP capacity): large, medium, small

For each plant configuration, the Project Team developed a concept plant layout that was used for developing construction cost estimates and engaged experienced contractors to support the costing efforts. Energy performance of each plant configuration was modeled to estimate annual energy use and costs. Results of the TIER plant LCC analyses were used to provide generalized guidance on TIER plant application in the design guide.

Task 3: Market Development

The market development task will primarily consist of two main efforts: manufacturer outreach to encourage scaling and a detailed list of recommendations for utility program design.

Task 3.1: Manufacturer Outreach

The design guide will be a valuable resource on all-electric plants for a wide range of stakeholders, but it is only practical to develop custom built-up TIER plants for larger commercial applications (around 200,000 ft^2 and larger) which have sophisticated engineering teams and dedicated facilities operators.

To expand applicability to medium-sized commercial buildings, a packaged equipment solution is ultimately needed to reduce the dependence on sophisticated project engineering teams and building operators. These would be modular equipment solutions that are pre-assembled and integrated, complete with controls from the factory. The project team led outreach to HVAC manufacturers to support and encourage the development of packaged equipment solutions for all-electric central plants to simplify future integration in medium-sized buildings.

Task 3.2: Develop Utility Program Recommendations for Electrification of Large Commercial Buildings

The project team engaged key stakeholders from California investor-owned utilities (IOU) programs, the design community, and manufacturers to identify industry-barriers related to deployment of TIER. Key stakeholders under California IOU programs targeted for this effort included the electric IOUs themselves, where their custom and market support programs were discussed, including the California Energy Design Assistance Program (California Energy Design Assistance Program [CEDA]/Willdan), and the California Technical Forum (Cal TF). In addition, the Project Team discussed Normalized Metered Energy Consumption (NMEC) to understand the implications for potential offerings for the existing building market.

Together, this broad industry engagement will guide detailed recommendations on programmatic implementation barriers for each stakeholder group and include potential mitigation strategies for California IOU programs to consider in their current and future program designs.

Additionally, as many California IOUs are interested in large-building electrification, the Project Team will engage the broader community for additional technology transfer to ensure findings are transferred downstream to building code and equipment standards activities.

Findings

Overview

The Findings section presents a summary of the results and analyses of the main project tasks. The Performance Review of TIER Plant subsection describes the process and findings of the detailed review of the first TIER plant installation, including summaries of plant energy performance, comparisons of TIER to other all-electric plant performance, and lessons learned from these early installations. The Performance Review of TIER Plant subsection also compares the measured field performance between the TIER plant and other all-AWHP plants, and includes a comparison between measured and modeled TIER performance. The Simplified Tool subsection describes the development of simplified spreadsheet tools for modeling the energy performance of two alternative TIER plant configurations, one with CW TES and one with HW TES, and the Energy Analysis subsection describes the results of a parametric energy analysis. In the energy analyses, the performance of the TIER plant is evaluated against a conventional chiller and gas-fired boiler plant. The Life Cycle Cost Study subsection after that presents the LCC study of TIER, evaluating the performance of different TES media and equipment sizing. In the LCC analysis, the various TIER configurations are compared against the TIER approach with the lowest LCC. Information gathered from this study, from other recent studies, and industry experience are compiled in a new design guide for all-electric central plants, described in the Design Guide subsection. Finally, the Market Transformation subsection identifies barriers to adoption and potential remedies.

Performance Review of TIER Plant

Two new office buildings in the California Bay Area (California Climate Zone 4 and ASHRAE Climate Zone 3C) completed construction in 2025 with CW TIER plants (Table 1). One building completed construction with an operational TIER plant for most of 2025 and was the focus of this analysis. The second building completed construction but both HRCs were damaged during startup by frozen evaporator coils. The subsequent mitigation efforts to repair and safely restore the HRCs prevented the central plant from operating in the TIER capacity until late 2025. In the interim, the central plant was reconfigured so that the AWHPs could directly serve the building heating loads, temporarily isolating the TES and HRCs. Though the two buildings and plants differ significantly in scale, they were constructed around the same time, serve similar functions and have similar equipment and equipment configurations. Because of these similarities, the second building was treated as a "baseline" central plant representing conventional practice with an AWHP plant serving the building heating loads and an independent water-cooled chiller plant serving the building cooling loads. Both buildings and central plants were already heavily instrumented for monitoring and control through their BAS. Comparisons between the plants are done using metrics like COP which are not impacted by the size differences.

Table 1: Study site details.

	TIER Plant	Baseline
Location	CA climate zone 4 (Sunnyvale)	CA climate zone 4 (Santa Clara)
Building Occupancy Type	Office	Office
Building Size	1,200,000 ft^2	300,000 ft^2
Cooling-Only Chillers	Two 595-ton variable speed centrifugal chillers	One 400-ton variable speed centrifugal chiller
Heat Recovery Chillers	One 600-ton variable speed centrifugal chiller Two 385-ton screw chillers	Two 190-ton screw chillers (not operational during study period)
Air-to-Water Heat Pumps	Two 2,400-kBtu/h AWHPs	Two 800-kBtu/h AWHPs
Cooling Towers	Three 1,010-ton Cooling Towers	Two 460-ton Cooling Towers
Thermal Energy Storage Tank	147,000 gallons	42,000 gallons (not operational during study period)

TIER Plant Study Site

The cooling-only chillers serve as the primary source of CHW for space cooling, meeting the building's cooling demand under normal operating conditions. Depending on system demand, one or more HRCs may operate to supplement cooling capacity during peak load conditions.

The HRCs are designed to provide all the HW needed for space heating by sourcing heat from either the CHW loop directly during high cooling load operating conditions or from the CW loop. Heat within the condenser loop is sourced from the cooling-only chillers, HRCs operating in cooling-only mode, and AWHPs.

The TES tank plays a critical role in temporally decoupling heating production and heating demand, thereby maximizing daily heat recovery potential and enabling time-independent energy management. Thermal energy added to the condenser loop is stored in a thermally stratified TES tank instead of being rejected immediately to cooling towers like in a conventional water-cooled chiller plant. The tank stores 80°F CW at the top of the tank and 60°F at the bottom of the tank. HRCs operating in heating-only mode source CW at 80°F from the top of the tank as a heat source for their evaporators to satisfy building heating demand and return CW at 60°F to the bottom of the tank. Chillers operating in cooling-only mode source 60°F from the bottom of the tank as a heat sink for their condensers to satisfy building cooling demand and return CW at 80°F to the top of the tank. When there are simultaneous building cooling and heating loads, heat rejected from chillers in cooling-only mode is directly used as a heat source by HRCs operating in heating-only mode. Any heat surplus is used to charge the tank and any heat deficit is made up by discharging the tank. If all 80°F water is drained from the top of the tank and replaced with 60°F water, the tank and chillers all shift to use 60°F at the top of the tank and 42°F at the bottom of the tank instead.

The AWHPs provide supplemental heating capacity to enhance system resiliency and support heating operation. Their primary function is to charge the TES tank during periods when recovered heat from building is not sufficient to keep up with tank discharge rates or charge the TES tank to full capacity by the start of the next day.

Finally, the cooling towers are responsible for rejecting excess heat from the chillers in cooling-only mode when the TES tank is fully charged with 80°F water. Once building's heating demands exceed the building's cooling demands, the cooling towers are disabled and the TES tank starts discharging.

A more thorough description of the CW TIER plant operating modes and system schematic is provided in Appendix A.

Baseline Plant Study Site

In the baseline plant, the heating and cooling production are independent processes as shown in Figure 3, and opportunities for simultaneous heat recovery and load shifting are non-existent.

The chiller serves as the primary source of CHW for space cooling. It operates to meet the building's cooling demand by removing heat from the CHW loop and transferring it to the CW loop. The rejected heat is then discharged to the atmosphere via the cooling towers. Since the system does not include heat recovery capability, condenser heat is not reclaimed for useful heating and is instead fully rejected to ambient air.

The cooling towers provide heat rejection for the chiller's condenser loop. They operate as required to maintain appropriate CW temperatures and ensure stable chiller performance.

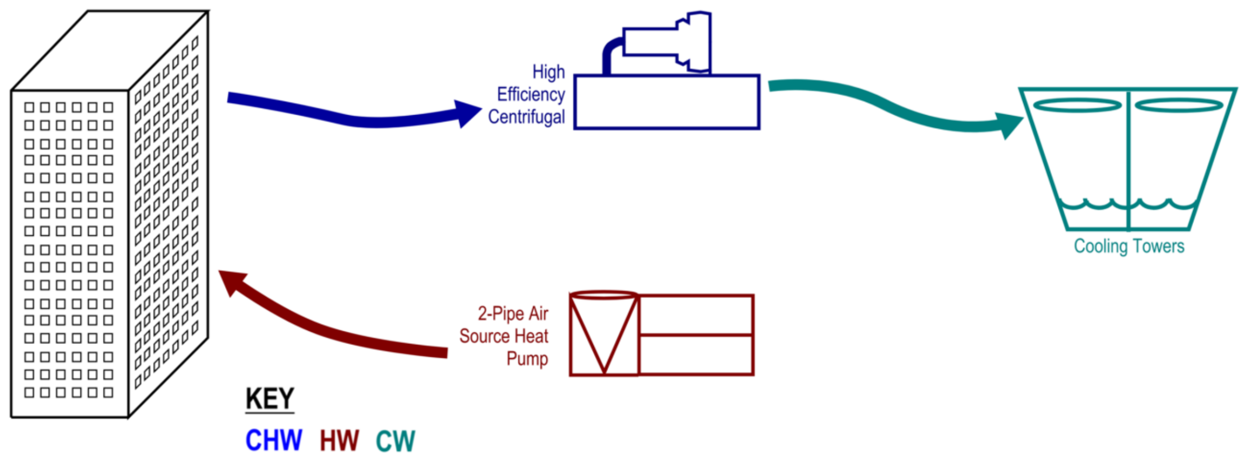

Figure 3: Baseline plant diagram.

Heating in the baseline plant is provided by the AWHPs alone. The AWHPs generate HW at elevated temperatures compared to the TIER plant to meet space heating demands independently of the cooling system. Because there is no thermal storage, all of the heating production must synchronously track real-time heating demand. Because there is no integrated heat recovery, the system cannot internally recover and reuse waste heat during periods of simultaneous heating and cooling loads, which results in higher overall energy consumption compared to the TIER configuration.

Overall, the baseline plant represents a conventional HVAC central plant design in which cooling and heating systems operate independently, without integrated heat recovery or energy storage capabilities. The configuration provides a clear benchmark for evaluating the operational and energy performance benefits of the TIER plant.

Data Collection

Data used to evaluate the performance of the TIER plant were obtained from the BAS of the two study sites. One building is equipped with the TIER plant, while the second building operates with a conventional AWHP system. The latter serves as the baseline system for comparative performance analysis.

Two categories of data were extracted from the BAS:

1. Measured Data: These data were recorded directly by field-installed instrumentation, including temperature sensors, flow meters, and power meters located at various points throughout the HVAC plants. These measurements provide the primary basis for calculating thermal energy transfer, system efficiencies, and operational performance
2. Calculated/Estimated Data: Certain parameters were not directly metered but were calculated or estimated within the BAS. Examples include chiller power consumption, AWHP power consumption, and other derived performance metrics. These values are typically generated using predefined algorithms within the control system, based on measured variables and equipment specifications.

Data obtained from the BAS were recorded on a change of value basis or at intervals ranging from one to ten minutes. All data were aggregated into five-minute intervals before the analysis was undertaken to facilitate the calculation of performance metrics while maintaining sufficient temporal resolution for system performance assessment.

Comparison: TIER Plant and Baseline Plant

To illustrate the TIER plant operation during the heating season, a representative day in February was examined to demonstrate the interaction between TES tank state of charge, equipment power (i.e., HRCs, cooling chillers, AWHPs, and cooling towers), and overall system sequencing at an hourly resolution. The profile shown in Figure 4 demonstrates how the tank is discharged throughout the day by the HRCs, while simultaneously recharged by the cooling chillers, and topped off by the AWHPs to 100 percent charge (TES tank full of 80°F water) a few hours before the next day's morning warm-up period when the source of heat recovery from the CHW loop runs out. The profiles demonstrate how each piece of mechanical equipment interacts with the storage system asynchronously throughout the day to maximize system efficiency.

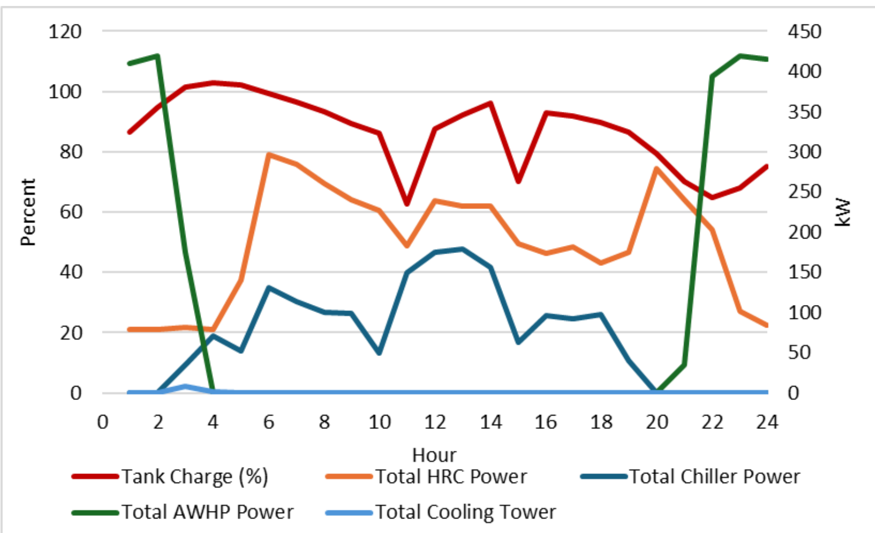

Figure 4: Hourly profile of the TIER plant for one representative day in February.

X-axis represents hour of day, Y-axis represents tank state of charge (%), AWHPs, HRCs, chillers, and cooling towers power (kW).

Figure 5 displays an analogous hourly profile for one day in August, as a representative day in the cooling season. Total chiller power increased substantially from late morning through the afternoon, reaching its highest levels during peak cooling hours (3:00 p.m. through 5:00 p.m.), while cooling tower power followed a similar pattern in response to increased condenser heat rejection. In contrast, AWHP power remained at zero throughout the day, indicating that heating demand was minimal and met entirely through heat recovered from the CHW loop. The system's behavior is consistent with expected summer sequencing, where chillers serve as the primary equipment for thermal energy production.

The tank state of charge remained relatively stable, fluctuating within a narrow band around the mid- to upper 70 percent range (the TES tank charge was manually limited to 80 percent instead of 100 percent on this day). This suggests limited charging and discharging activity compared to heating-dominated days. HRC power shows moderate activity in the morning and again briefly in the late evening which correlates with space heating demands.

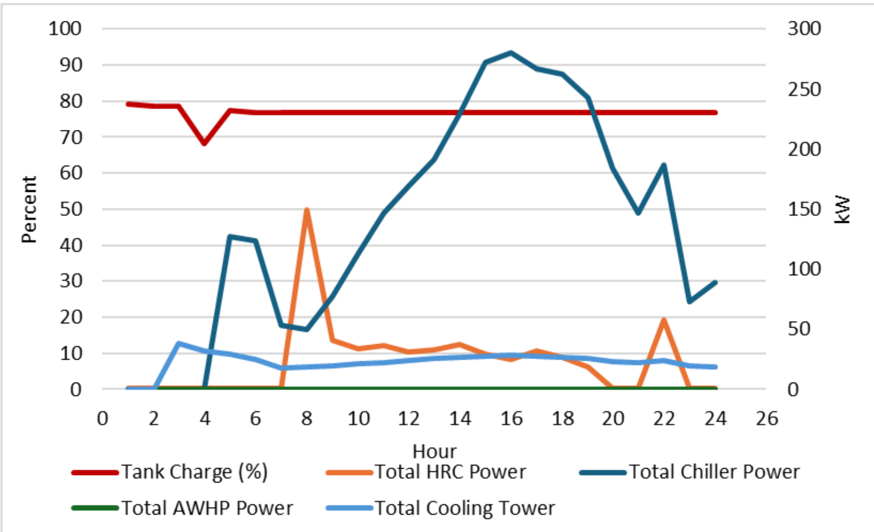

Figure 5: Hourly profile of the TIER plant for one representative day in August.

X-axis represents hour of day, Y-axis represents tank state of charge (%), AWHPs, HRCs, chillers, and cooling towers power (kW).

As discussed earlier in this report, a key component of the TIER plant performance evaluation involved benchmarking its operation against a baseline plant served by a conventional AWHP system. This comparative analysis provides a consistent basis for assessing relative system efficiency under similar operating conditions. It should be noted that the data used for this comparative analysis spans from April to August 2025 for both plants.

As summarized in Table 2, the TIER plant demonstrates superior energy performance over the analysis period, achieving higher overall average COPs in both heating and cooling modes as well as overall combined COP. During the April through August period when both plants were operating, the average heating COP for the TIER plant was 4.2, compared to 3.2 for the baseline plant, representing approximately a 31 percent improvement. In winter conditions, the difference in COPh between the two plant types may be expected to increase due to decreasing COPs for AWHPs in colder weather, but data were not available during that period because the Baseline Plant was restored to TIER operation in the late fall. Similarly, the TIER plant achieved an average cooling COP of 6.0, compared to 4.4 for the baseline system (which was effectively a water-cooled chiller plant), which is an improvement of roughly 36 percent. In addition, the combined COP, which is estimated as total heating and cooling loads divided by the total power used by all plant equipment, was higher in the TIER plant compared to the baseline plant. Average COPs are also provided for different seasonal periods in Table 2. Evaluating COPs for heat recovery equipment is not straightforward because the work performed can simultaneously provide heating and cooling. For the comparisons throughout this project, the convention used is to assign chiller power to the heating COP calculation whenever it is rejecting heat to the TES and to the cooling COP calculation whenever rejecting heat to the cooling towers. When comparing to reported COPs for other systems, the TIER heating COPs will be relatively depressed and the TIER cooling COPs will be relatively elevated based on these conventions.

Note that both buildings include small process cooling loads that continuously require CHW. These processes were each added late in the design stages and the central plants are not well equipped to handle such low loads overnight when there are no other CHW loads. As the process loads are well below the minimum turndown capability of the lead chiller in each plant, the resulting cycling of the lead chillers in each plant reduces the overall COPs. This impacts even the heating COP based on the convention of applying chiller power to the heating COP calculation when heat is rejected to the TES, as discussed below. The addition of pony chillers is being considered for the actual buildings to address this issue.

Table 2: Average COPs for the TIER and baseline plants.

	Heating COP		Cooling COP		Combined COP	
Averaging Period	**Baseline Plant**	**TIER Plant**	**Baseline Plant**	**TIER Plant**	**Baseline Plant**	**TIER Plant**
Winter	–	2.5	–	4.5	–	2.3
Spring	3.0	4.0	3.8	6.0	3.2	5.2
Summer	3.4	4.9*	5.2	5.9*	4.8	5.7
Fall	3.2*	3.5	4.3*	7.4	3.6*	5.0
Average (Spring through Summer)	3.2	4.2	4.4	6.0	4.0	5.5
Annual Average	–	3.6	–	6.0	–	4.4

Notes:
Winter = January through March, Spring = April through June, Summer = July through September, Fall = October through December
Baseline plant did not operate in AWHP configuration in December through March months.
Convention for COP calculation for TIER plant assigns chiller power to heating COP calculation whenever rejecting heat to the TES and to the cooling COP calculation whenever rejecting heat to the cooling towers.
* Reported averages exclude 09/01 to 10/07 data for TIER plant and December data for baseline plant

Figure 6 and Figure 7 illustrate the relationship between total daily heating energy and HDD (base 65°F), and total daily cooling energy and CDD (base 65°F), respectively, for the TIER building. Figure 8 and Figure 9 present the corresponding relationships for the baseline building.

A comparison of Figure 6 and Figure 8 indicates that daily heating energy in the baseline building exhibits a more clearly defined linear relationship with HDD than in the TIER building. The less linear relationship between total daily heating energy and HDD in the TIER building suggests that additional operational factors, such as occupancy, building characteristics, etc., may influence heating energy use beyond simple weather dependence.

For cooling, both buildings demonstrate a relatively strong positive linear relationship between total daily cooling energy and CDD (Figure 7 and Figure 9). As cooling degree days increase, daily cooling energy increases in a generally proportional manner for both systems.

Both buildings exhibit non-zero total daily heating energy use even on warm days when outdoor air temperatures never drop below 65°F. The same is true for the TIER building on the cooling side where at least some daily cooling energy is expended on cold days when outdoor air temperatures never rise above 65°F. These base loads are largely attributed to user behaviors such as 24/7 occupancy and 24/7 plug loads.

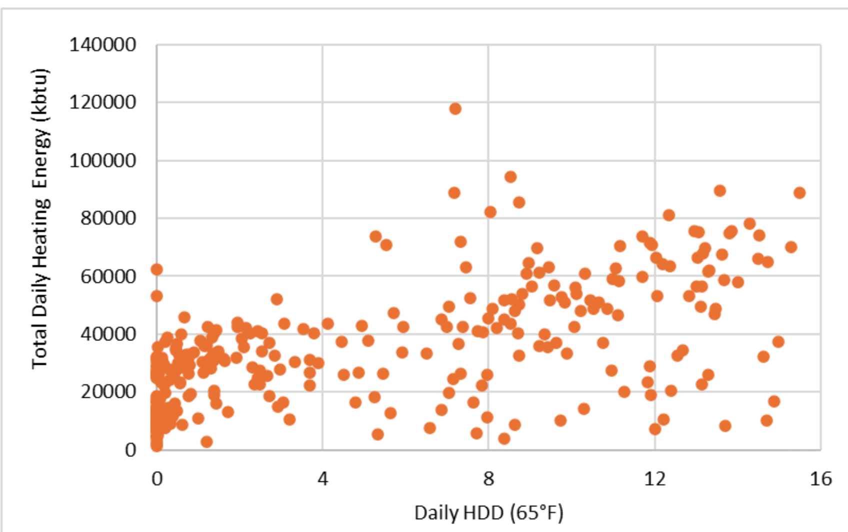

Figure 6: Correlation between total daily heating energy and HDD in the TIER building.

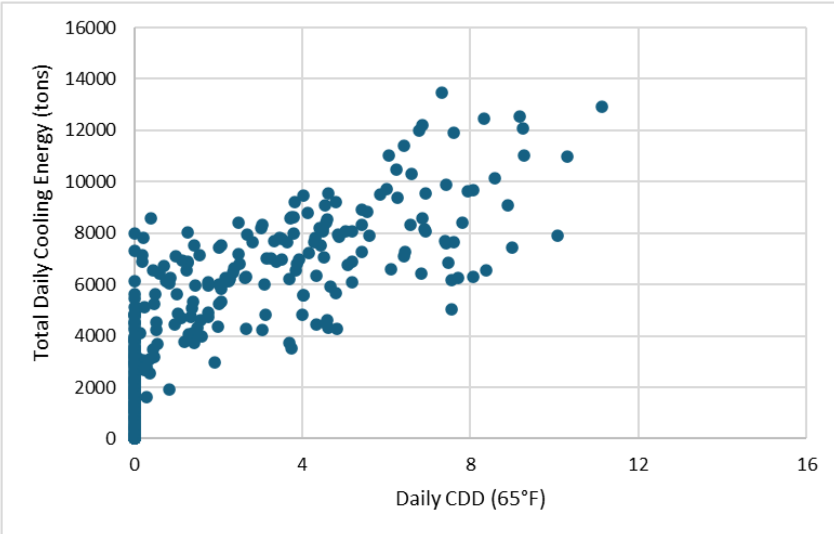

Figure 7: Correlation between total daily cooling energy and CDD in the TIER building.

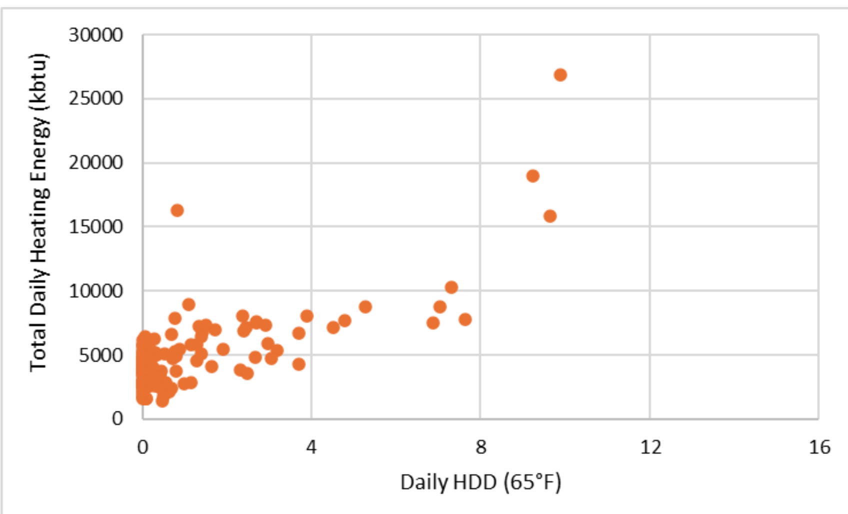

Figure 8: Correlation between total daily heating energy and HDD in the baseline building.

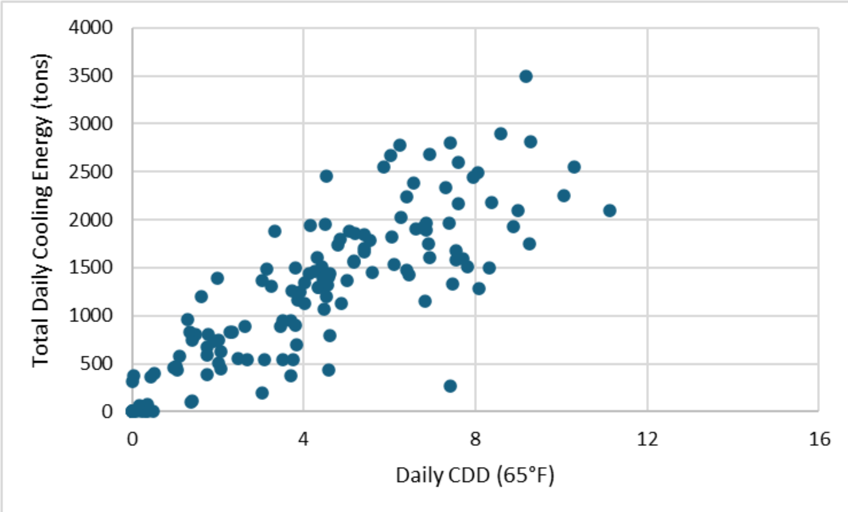

Figure 9: Correlation between total daily cooling energy and CDD in the baseline building.

Figure 10 presents the daily average heating COP for the TIER plant and the baseline plant from April through August. Over the analysis period, the TIER plant operated at a slightly higher heating COP than the baseline system. The TIER plant generally maintains heating COP in the range of approximately 3.5 to 5.5, with a few days approaching 6.0. In contrast, the baseline plant more frequently operates between approximately 2.5 and 4.0 heating COP, with periodic drops below 2.0. While the TIER plant demonstrates a higher overall efficiency trend, there are isolated days where the baseline plant achieves equal or slightly higher COP values. The study period comparing the TIER and baseline plants does not include many cold weather days as shown in Figure 7 where nominal AWHP efficiencies are expected to drop to 2.0 and below. Both systems exhibit day-to-day variability, likely reflecting changes in load conditions, outdoor air temperature, and equipment cycling.

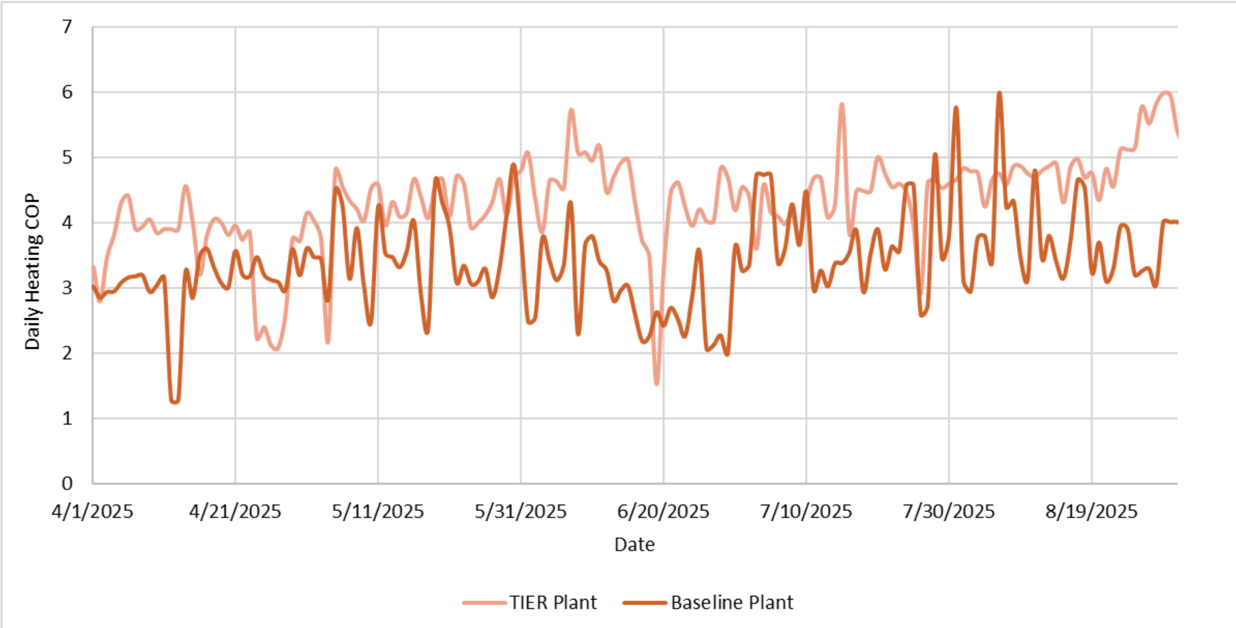

Figure 10: Daily average heating COP for the TIER plant and the baseline plant from April through August.

Figure 11 presents the daily average cooling COP for the TIER plant and the baseline plant from April through August. Over this timeframe, the TIER plant consistently demonstrates higher cooling efficiency compared to the baseline system except for a period in April. The TIER plant generally operates within a COP range of approximately 5.0 to 7.5. In contrast, the baseline plant more commonly operates between approximately 3.5 and 6.0, with multiple short duration drops below 3.0 observed from May to July. In addition, between May and July, the conventional AWHP system (baseline plant) shows noticeably lower daily cooling COP compared to TIER plant.

In both plants, the lead chiller is relatively oversized compared to the base cooling load of the respective buildings. The lead chiller must cycle at cooling loads below their minimum unloading threshold. However, both buildings have process load requirements that demand minimal interruptions in CHW service. To the extent possible, the rate of chiller cycling was intentionally increased to meet this demand, despite the negative impact on efficiency. The lead chillers in both plants have centrifugal compressors that are not only inefficient at their minimum unloading threshold but also incur a substantial power penalty during each startup sequence. Because the TIER plant counts chiller energy as heating power in the heating COP calculations whenever the rejected heat is used to charge the TES tank instead of sent to the cooling towers, the cooling COP of the TIER plant is not penalized for equipment cycling to the same extent as cooling COP is penalized in the baseline plant. The heating COP of the TIER plant is penalized as shown by the lower COP for TIER in the coldest bin in Figure 9.

Another compounding reason for the lower daily cooling COP in the baseline plant is the relative size of the lead cooling chiller in the baseline plant compared to the TIER plant. The 400-ton lead chiller in the baseline plant serving a 300,000 ft² building is 2.7 times larger on a Btu/hr·ft² design capacity basis relative to the 600-ton lead chiller in the TIER plant designed to a 1,200,000 ft² building. The oversized baseline chiller is even more prone to short cycling, leading to higher transient losses and lower COP. In contrast, the TIER plant benefits from two smaller chillers and more granular staging. The baseline plant could have been designed with two cooling chillers like the TIER plant, but the additional chiller was determined to not be cost-justified. However, in real operation the minimum unloading threshold was much higher than the advertised turndown and the building's base cooling load was much lower than anticipated.

As outdoor air temperatures increase in late July through August and the overall cooling loads become higher and more sustained, the frequency of equipment cycling decreases and the baseline plant COP improves. This improvement is consistent with reduced cycling.

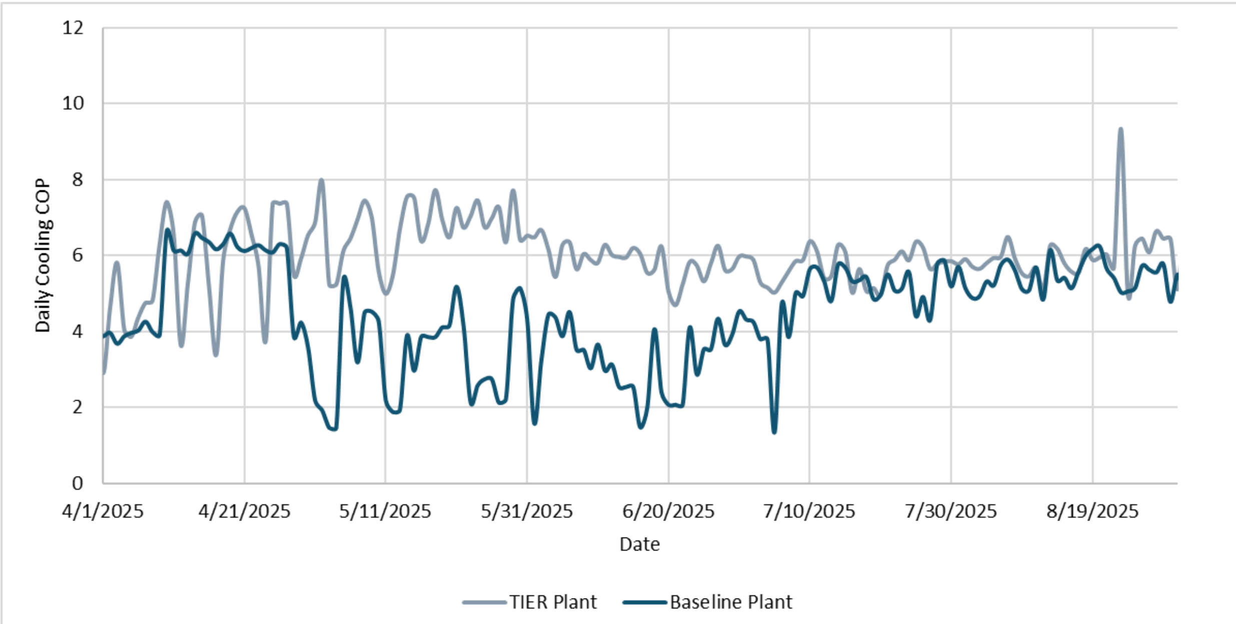

Figure 11: Daily average cooling COP for the TIER plant and the baseline plant from April through August.

In addition to benchmarking and comparing the TIER plant against the baseline plant, it is important to evaluate how each plant's heating efficiency varies with outdoor air temperature. Figure 12 presents the average daily heating COP of both plants for different average daily outdoor air temperature bins. As expected, the heating COP of both systems generally improves as average daily outdoor temperature increases, reflecting reduced compressor cycling and more favorable heat pump operating conditions.

The baseline plant is less sensitive to outdoor air temperature, with its average daily heating COP maintaining a comparatively narrow range across the evaluated temperature bins. In contrast, the TIER plant shows a more considerable temperature dependence, with heating COP dropping by a more pronounced margin as average daily outdoor air temperature falls. This behavior is consistent with the characteristics of the TIER system and COP calculation conventions used here. Again, the TIER plant attributes chiller compressor cycling penalties to the plant's heating COP, so lower heating COP is expected at lower outdoor temperatures when cooling demand is reduced. Conversely at higher outdoor temperatures, higher heat recovery availability and reduced cooling chiller cycling allow the TIER plant to produce HW more efficiently, leading to an upward trend in COP relative to the baseline system.

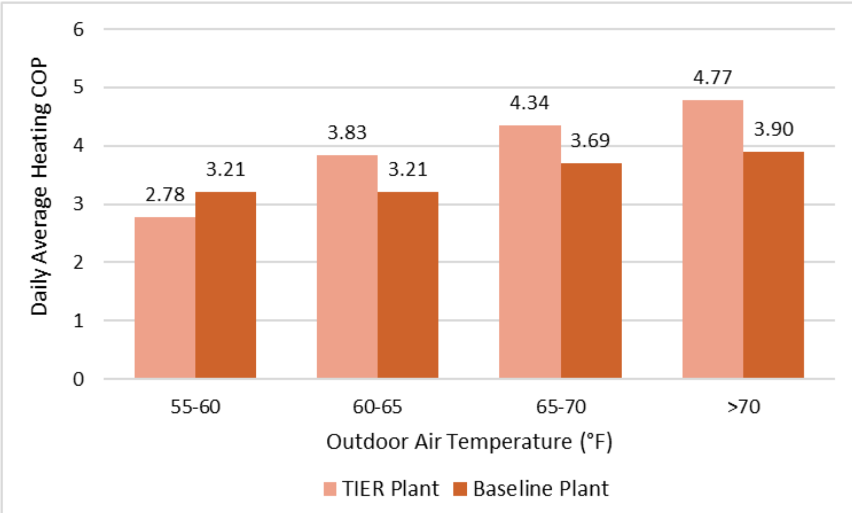

Figure 12: Daily average heating COP of the TIER plant and the baseline plant by average daily outdoor air temperature bins.

The Project Team also examined the cooling performance of both plants at different outdoor temperature bins. Figure 13 shows that across all average daily outdoor air temperature bins, the TIER plant consistently achieves a higher daily average cooling COP than the baseline plant. The differences are due to findings shown in Figure 9 documenting how the chiller cycling penalty is absorbed into the heating COP for the TIER plant and how the oversized chiller in the baseline plant exacerbates the penalty.

The increase in average daily cooling COP with rising average daily outdoor air temperature is observed in both plants (see Figure 13). As outdoor air temperature increases, cooling demands in both plants become higher and more sustained, allowing equipment to operate more consistently above its 15 to 20 percent minimum unloading threshold. At lower outdoor air temperatures, cooling demand is lighter and more intermittent, leading to increased cycling losses. Consequently, both the TIER and baseline plants show higher average daily cooling COP values at elevated average daily outdoor air temperatures. However, centrifugal chillers are less efficient at full load compared to part load which is expressed by the daily average cooling COP differences between the 65–70°F bin and the >70°F bin. The proportionally oversized chiller in the baseline plant is operating more at part load compared to the lead chiller in the TIER plant hence the modest COP increase in the >70°F bin as opposed to the modest decrease in the baseline plant.

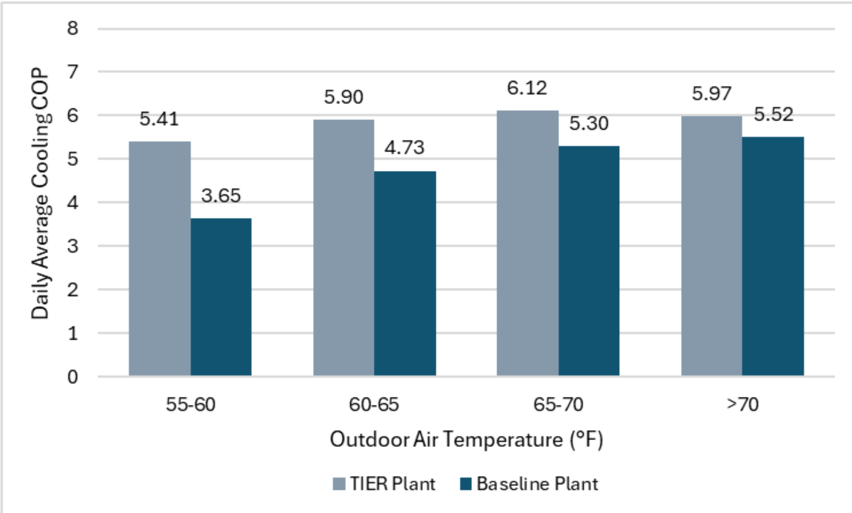

Figure 13: Daily average cooling COP of the TIER plant and the baseline plant by average daily outdoor air temperature bins.

One limitation of the temperature-binned analysis shown in Figure 12 and Figure 13 is the unequal number of days represented in each temperature bin during the April through August analysis period. The counts of daily observations in the 55–60°F, 60–65°F, 65–70°F, and >70°F bins are 8, 28, 65, and 52 days, respectively. The relatively small observation count in the 55–60°F bin makes the average COP in that range more sensitive to operational variability, anomalous days, and transient control behavior, potentially resulting in greater noise compared to bins with larger observation counts. In contrast, the 65–70°F and >70°F bins contain substantially more data points, providing more statistically stable and representative averages. Therefore, while the overall trends across temperature bins remain informative, the results in the lowest temperature bin should be interpreted with caution due to the limited number of observations.

In addition, the analysis results comparing the two plants in this section are constrained by the absence of winter-month data, as the period for this comparison spans only April through August. Consequently, lower outdoor temperature conditions typical of peak heating season are not represented.

Performance Assessment of TIER Plant

While comparing the TIER plant with the baseline system is beneficial for understanding the efficiency of TIER design against a conventional AWHP system, further insight can be gained by examining the TIER plant's performance independently. This section provides a more detailed evaluation of TIER-specific operational trends. This deeper assessment helps identify the drivers of efficiency and operational dynamics. All data, metrics, and discussions presented in this section are based on one year of plant operation from January 1, 2025, through January 27, 2026, excluding the period from September 1, 2025, to October 7, 2025. Data from September 1, 2025, to October 7, 2025, was excluded from analysis in this section of the report because the plant was undergoing active testing during this time. Unless otherwise noted, reported performance indicators and metrics reflect normal operational conditions during the evaluation period (January 1, 2025, to January 27, 2026).

[Figure 14](#) illustrates the daily average heating COP of the TIER plant over a period slightly exceeding one year (January 2025 through January 2026), providing insight into seasonal performance dynamics. The trend shows a clear seasonal pattern, with lower daily average heating COP values during the colder winter months (January, February, and December) and comparatively higher COP values during the shoulder and cooling seasons. This behavior aligns with the design and operational characteristics of the TIER system. During colder months, the plant relies more on the AWHPs to meet heating demand, which inherently lowers system efficiency. Additionally, chiller energy that is expended while rejecting heat to the TES tank is accounted as heating energy and increased compressor cycling during colder months results in depressed daily average heating COP values. Data from September 1 through October 7, 2025, were excluded from the analysis due to active functional performance testing activities conducted at the TIER plant during this period. The testing introduced atypical operating conditions and control overrides that were not representative of normal day-to-day plant operation, resulting in COP behavior that does not reflect day-to-day performance. This period appears as a gap in Figure 14. To ensure the calculated performance metrics reflect typical operating conditions, this time span was removed from the analytical dataset. However, it is intentionally retained in the figure to clearly show the excluded interval.

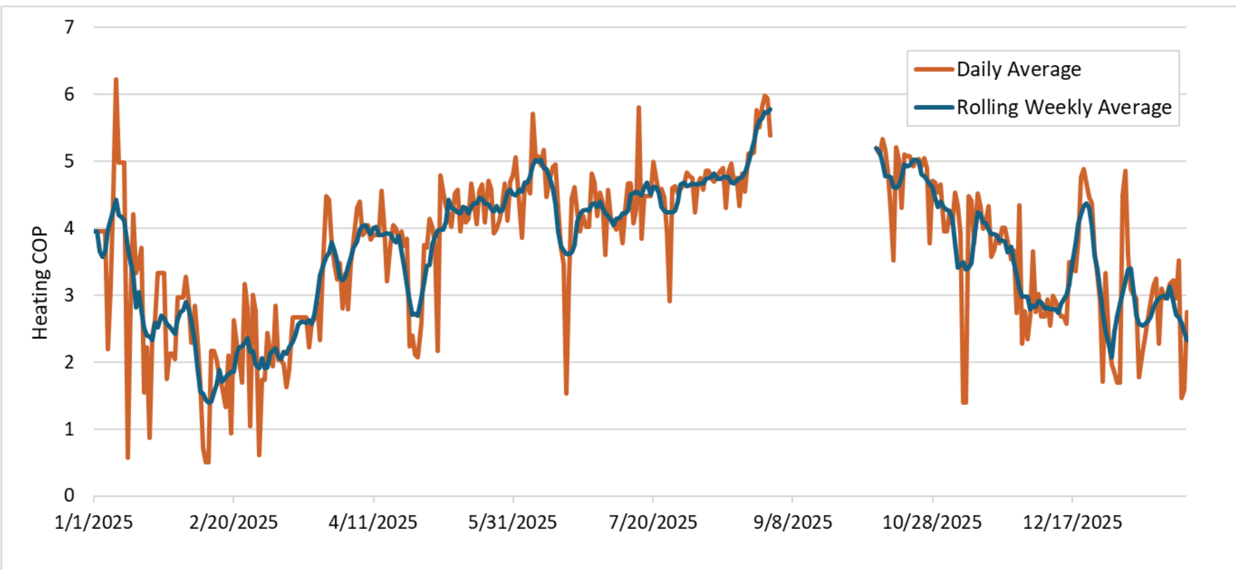

Figure 14: Daily average heating COP of the TIER plant from January 2025 to January 2026.

To better understand the operational role of the AWHPs within the TIER plant, the project team quantified the portion of the total daily heating load (kBtu/h) supplied by the AWHPs. The thermal output of the AWHPs was calculated using measured entering and leaving water temperatures across the units in conjunction with water flow rates obtained from installed flow meters. Using these measurements, the AWHP heating energy was estimated and aggregated on a daily basis. This value was then divided by the total daily building heating energy to estimate the fraction of heating demand met by the AWHPs. Figure 15 presents this fraction as a function of average daily outdoor air temperature. As shown, the AWHP contribution is negligible on most days, with the fraction of heating load met by the AWHPs remaining near zero under moderate and warmer conditions. However, the AWHP contribution increases at lower outdoor temperatures. Figure 16 shows the average fraction of heat load met by AWHP by OA temperature bins. In lower OA temperature bins, the average fraction is higher, while, in higher OA temperature bins the fraction is close to zero or zero. This behavior is consistent with the intended operation of the TIER plant. During colder weather, cooling demand decreases, limiting the amount of recoverable heat available from the CHW loop to recharge the TES tank. At the same time, building heating demand increases. Under these conditions, the system relies more on the AWHPs to supplement heating production, resulting in a higher fraction of heating load met by these units.

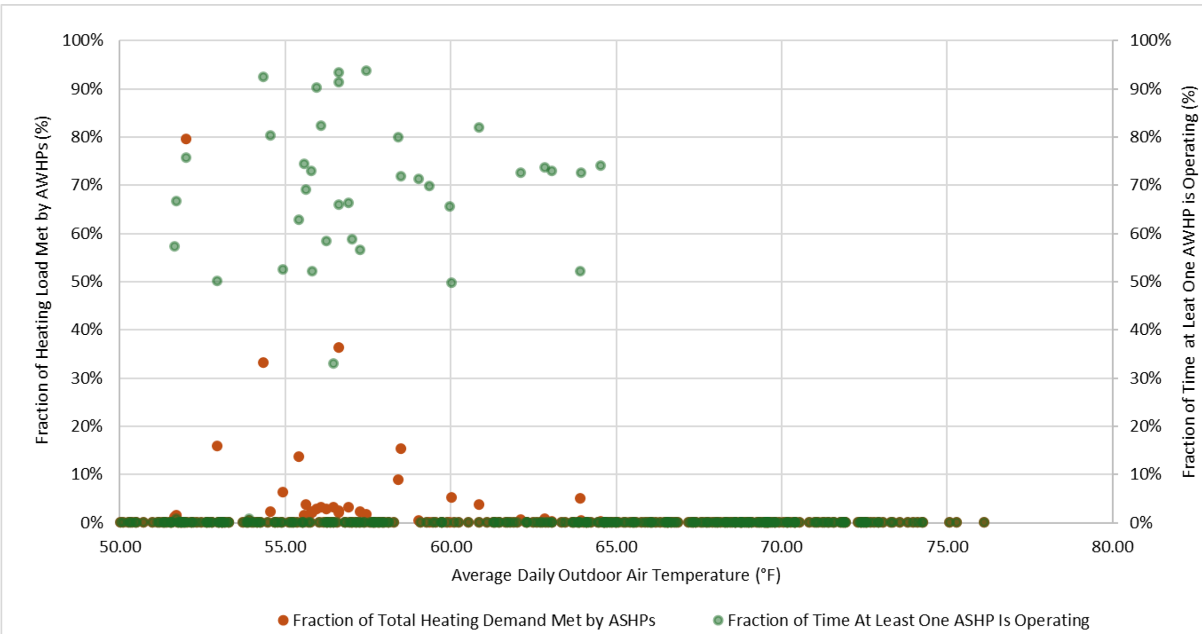

Figure 15: Fraction of daily heating load met by AWHPs and daily fraction of time that at least one AWHP was operating in TIER plant as a function of average daily outdoor air temperature.

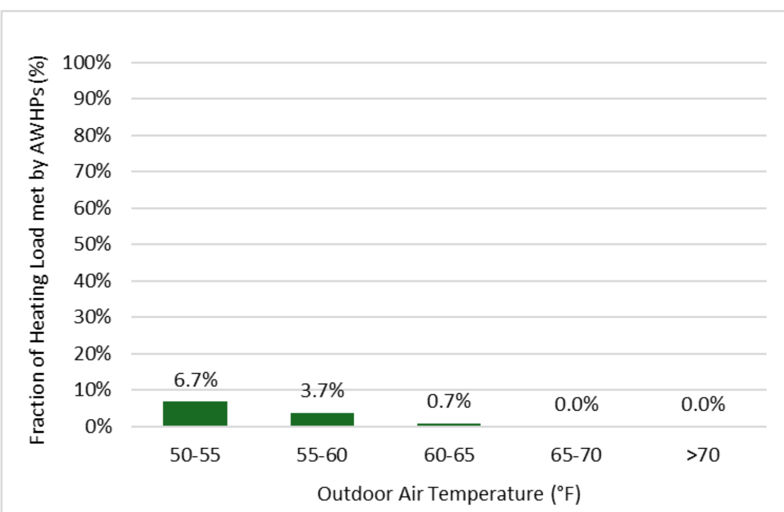

Figure 16: Daily average fraction of heating load met by AWHPs by average outdoor air temperature bins.

Figure 17 and Figure 18 present the variation of the TIER plant's daily average heating COP and cooling COP as a function of daily HDD (base 65°F) and CDD (base 65°F), respectively. As shown in Figure 17, the daily average heating COP decreases with increasing HDD, with the highest heating COP at the lowest HDD bin. As HDD increases, cooling demand falls causing the chillers, which are the primary equipment used to generate heat for the HRCs, to cycle more frequently. The increased cycling at low cooling loads reduces efficiency, and this reduces the heating COP because of the convention of applying chiller power to the heating COP calculation when rejecting heat to the TES tank. As HDD increases, heating demand also rises causing the system to rely more on running AWHPs less efficiently at lower ambient conditions, though this is likely only a minor impact since the AWHPs seldom run. This is partially offset by the HRCs operating more efficiently at higher load fractions as screw chillers. Collectively, these factors result in lower heating COP at higher HDD bins.

In contrast, Figure 18 indicates that the daily average cooling COP remains relatively stable across different CDD bins, generally ranging between approximately 5.9 and 6.5. The daily average cooling COP only accounts for chiller, pump, and cooling tower energy when the TES tank is fully charged, which predominantly occurs after cooling demands have increased beyond chiller minimum unloading thresholds that cause compressor cycling. Equipment efficiencies of the lead centrifugal compressors are highest between 30 and 50 percent part load and decrease at low loads and high loads. This theoretical performance is reflected in Figure 16 below as there is a slight downward trend in daily average cooling COP at higher daily CDD days.

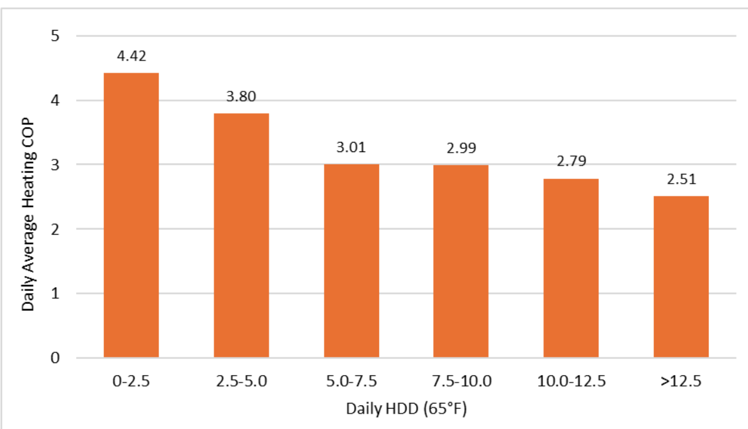

Figure 17: Daily average heating COP of the TIER plant by HDD.

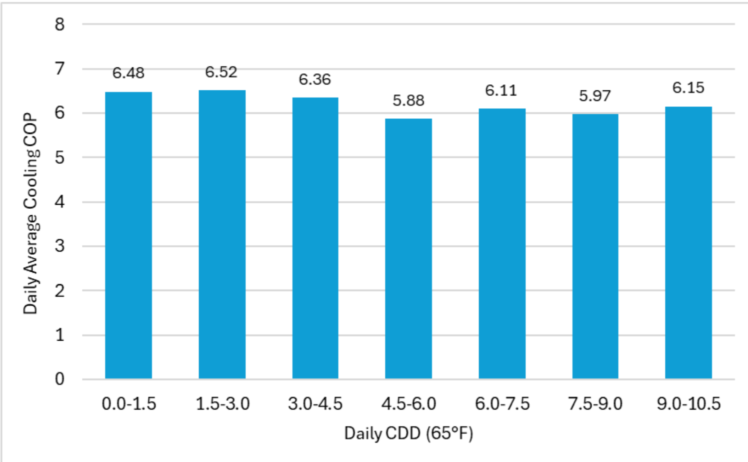

Figure 18: Daily average cooling COP of the TIER plant by CDD.

The efficiency and performance of the TIER plant is a function of not only weather conditions but also the heating and cooling loads. Thus, the Project Team also investigated how the heating and cooling COPs of the TIER plant correlated with heating and cooling load. To evaluate the relationship between heating COP and heating load (kBtu), a normalized peak load metric was developed (see equation 1). For each day within the analysis period (January 2025 through January 2026), the total daily heating load was divided by the maximum observed total daily heating load during the study period. This ratio represents the fraction of peak daily heating load and was used to categorize days into defined peak daily heating load bins. The daily average heating COP was then calculated for each load bin to assess load-dependent performance.

$$Peak\ Daily\ Heat\ Load\ Fraction_d = \frac{\sum_{t=1}^{Nd} Q_{d,t}}{max(\sum_{t=1}^{Nd} Q_{d,t})} \qquad \text{(equation 1)}$$

$Q_{d,t}: daily\ heat\ load\ at\ time\ interval\ t\ on\ day\ d$

$N_d: number\ of\ time\ intervals\ in\ day\ d$

As shown in Figure 19, heating efficiency varies with the peak daily load fraction. Again, the same couple of dominant factors come through. There is less chiller compressor cycling in the lower daily heating load bins, so the average daily heating COP values are higher. The HRC screw compressors are more efficient at full load, so the average daily heating COP values generally improve at higher load fractions (60 to 100 percent).

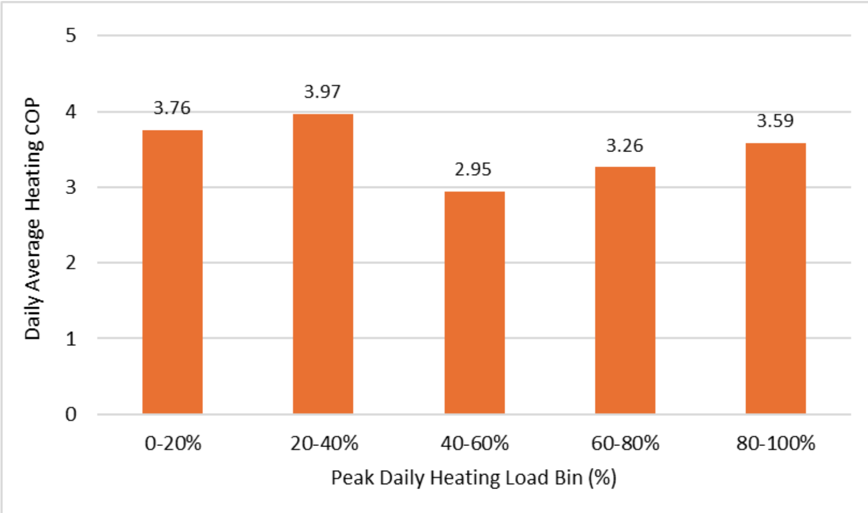

Figure 19: Average daily heating COP of the TIER plant by peak heating load range.

A similar methodology was applied to evaluate the relationship between average daily cooling COP and peak daily cooling load. The total daily cooling load was normalized by the maximum observed daily cooling load during the analysis period to define peak daily cooling load bins. The daily average cooling COP was then calculated for each bin to assess load-dependent cooling performance. Figure 20 presents the resulting relationship between peak cooling load fraction and average daily cooling COP.

$$Peak\ Daily\ Cool\ Load\ Fraction_d = \frac{\sum_{t=1}^{Nd} Q_{d,t}}{max(\sum_{t=1}^{Nd} Q_{d,t})} \quad \text{(equation 2)}$$

$Q_{d,t}: daily\ cool\ load\ at\ time\ interval\ t\ on\ day\ d$

$N_d: number\ of\ time\ intervals\ in\ day\ d$

The highest cooling COP occurs within the 20 to 40 percent peak load bin, indicating that the TIER plant achieves optimal cooling efficiency under moderate loading conditions. At very low load fractions (0 to 20 percent), the cooling COP decreases to the lowest value. This reduction is likely attributable to the increased frequency of chiller short cycling even during instances when the TES tank is fully charged. At higher load fractions (40 to 100 percent), the cooling COP stabilizes within a relatively narrow range, remaining slightly below the peak value observed at moderate load. This is consistent with the part load performance of the lead chiller’s centrifugal compressor where efficiencies are highest near 30 percent to 50 percent of the chiller’s maximum capacity. Similarly cooling tower fans are much more efficient at part load since they are variable speed and power decreases exponentially with load.

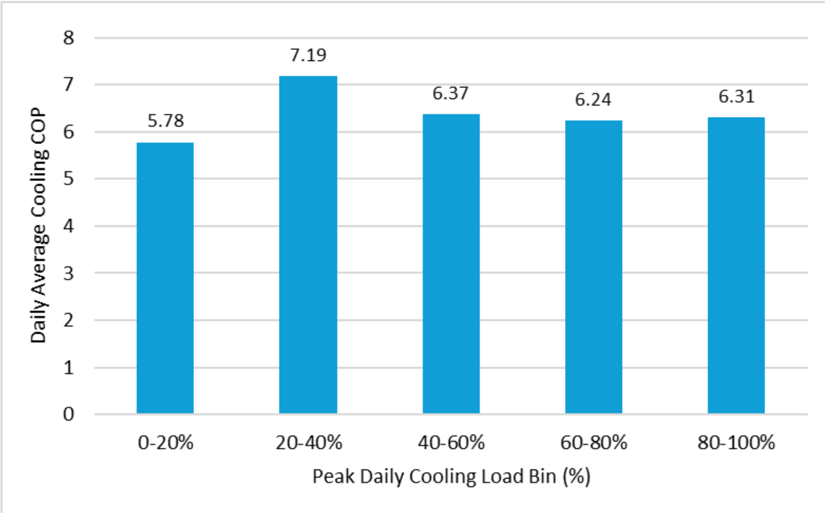

Figure 20: Average daily cooling COP of the TIER plant by peak cooling load range.

In addition to the peak load-performance analysis presented above, it is also beneficial to examine TIER system performance over representative summer and winter months. Figure 21 and Figure 22 present daily cooling and heating COPs, respectively, alongside their corresponding loads (kBtu/h) for July, providing a clearer view of how plant efficiency responds to day-to-day variations in demand under different operating conditions.

As expected for a summer month, cooling loads are consistently high throughout July, generally ranging between approximately 180 and 420 tons (see Figure 21). In contrast, heating loads are relatively low and intermittent, with most days below roughly 400 and 500 kBtu/h, aside from one noticeable spike during the middle of July (see Figure 22).

The daily average cooling COP remains relatively stable throughout July, generally between 5.0 and 6.0. Even during periods of higher cooling load, the daily average cooling COP does not drastically degrade. This suggests that the TIER plant maintains efficient cooling performance under higher summer loads. Although daily average cooling load fluctuates over the month, the daily cooling COP does not show a strong shift with load. In fact, on some higher-load days, COP remains comparable to or slightly higher than moderate-load days. This indicates that the plant does not experience significant efficiency penalties at higher cooling load levels, and part-load inefficiencies are likely minimized during this cooling-dominated season. This is consistent with findings from investigating the relationship between the daily average cooling COP and the peak cooling load fraction presented in Figure 20.

According to Figure 22, the daily average heating COP shows greater variability compared to daily average cooling COP. Since July's daily average heating loads are low and intermittent, the daily average heating COP fluctuates more noticeably, likely reflecting short-duration operation of heating components, low-load cycling, and varying levels of heat recovery potential. The sudden increase in daily heating load mid-July is probably caused by testing at the TIER plant, which temporarily changed normal operating condition and resulted in very high daily heating load.

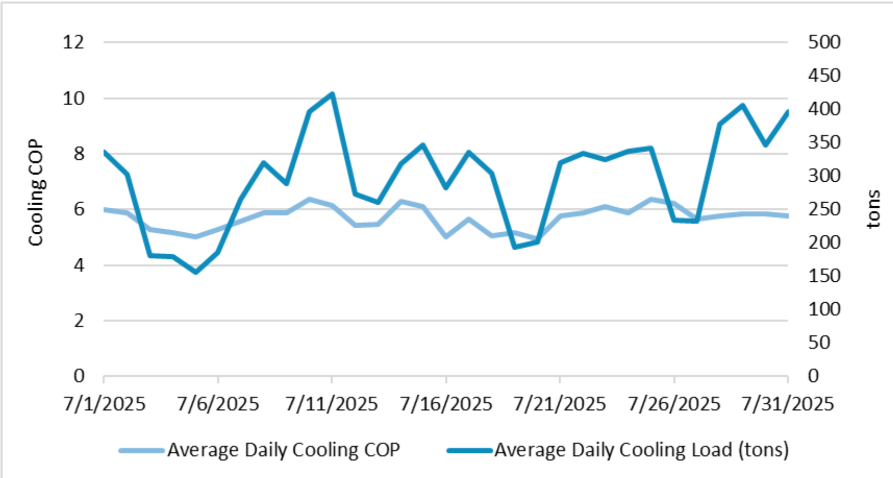

Figure 21: Trend of daily average cooling COP and daily average cooling load in July for the TIER plant.

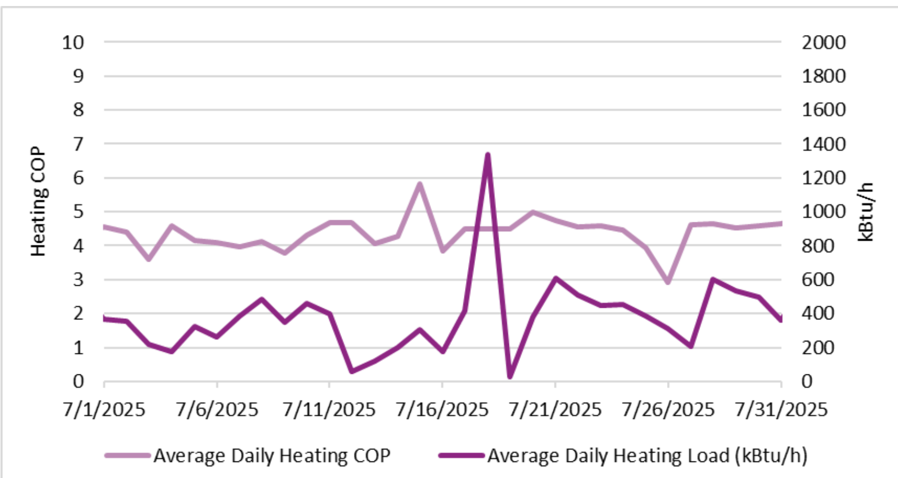

Figure 22: Trend of daily average heating COP and daily average heating load in July for the TIER plant.

Similarly, Figure 23 and Figure 24 present the relationship between daily average heating and cooling COPs and their corresponding daily loads for the month of December, reflecting winter operating conditions. As shown in Figure 23, the daily average heating COP exhibits a slight improvement as daily heating load decreases. This trend may be attributed to reduced demand for HRCs operating for heat recovery at lower heating demand levels.

Figure 24 indicates that daily average cooling COP in December fluctuates considerably, which is expected given the relatively low and intermittent cooling demand during winter months. At lower daily cooling loads, the cooling COP decreases, likely due to short cycling of chillers. This behavior is consistent with the earlier analysis of peak cooling load and daily cooling COP.

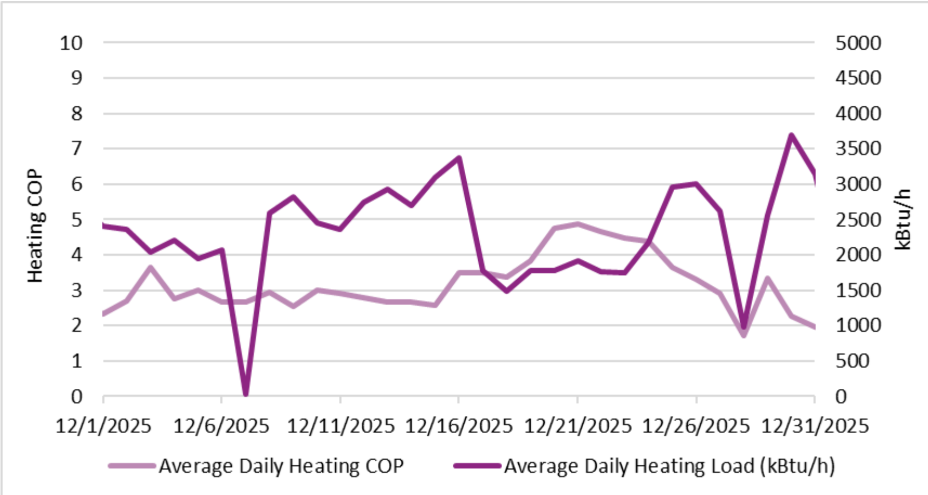

Figure 23: Trend of daily average heating COP and daily average heating load in December for the TIER plant.

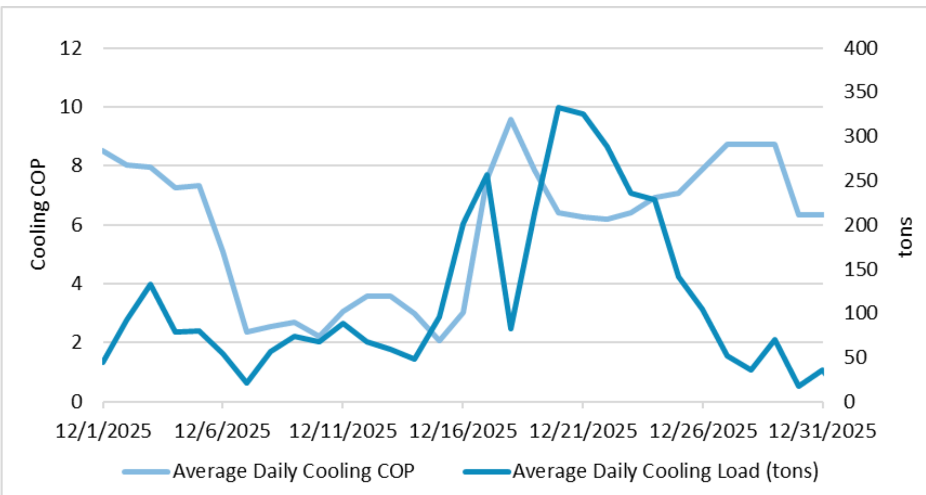

Figure 24: Trend of daily average cooling COP and daily average cooling load in December for the TIER plant.

Comparison: TIER Plant and Model

This section compares the performance of the real-world TIER plant to the as-designed performance from a simplified spreadsheet model. As discussed earlier, the TIER plant refers to an office building located in CA that has an operational TIER system (see TIER Plant Study Site section for more details). The spreadsheet model reflects the as-designed TIER configuration and was developed in design to estimate the annual heating and cooling performance based on hourly analyses. The spreadsheet model uses the measured building heating and cooling loads (HW and CHW loads), and represents plant components, control logic, and equipment efficiency curves. A full description of the spreadsheet model structure, inputs, and assumptions is provided in the Simplified Tool section of this report.

Though the measured building loads are used in the spreadsheet model, the intent of this effort was to compare the actual and as-designed performance, rather than to calibrate the model to the performance of the actual TIER plant installation. The model uses generalized performance assumptions consistent with design stage analysis and simplified staging and control logic. The equipment performance curves used in modeling are for similar equipment types in the EnergyPlus library, but do not necessarily reflect the performance of the installed equipment. Therefore, differences between modeled and estimated performance reflect both real-world operational effects such as TES utilization, part- and full-load behavior, and control strategies implemented in the site and the simplified nature of the modeling framework.

The comparison between the estimated (based on data obtained from BAS) and modeled daily average heating COP, presented in Figure 25, indicates that the TIER plant performs slightly better than the model. While both trends follow a similar pattern, showing gradual improvement from spring to summer, the actual plant achieves higher efficiency levels and exhibits slightly greater variability. This suggests that the model conservatively estimated heating performance, possibly due to different assumptions such as thermal energy storage utilization, or part-load control refinements implemented in the field. One contributing factor is the known difference between the modeled and actual part load performance of the lead chiller. The spreadsheet model uses performance curves for a variable speed centrifugal chiller of similar size from EnergyPlus library (Figure 39). For the modeled temperature conditions, the part load curve shows severe degradation in efficiency at low loads, up to 0.8 kW/ton at 20 percent part load. In contrast, the submittal for the installed equipment reports 0.37 kW/ton at 20 percent part load and slightly lower kW/ton values throughout the range of part load conditions. Given the frequency of operation of the first stage chiller at low part loads, this factor is likely a significant contributor to the differences between the actual and modeled performance. Based on the COP conventions used, the differences between modeled and actual part load performance affect both heating and cooling.

A similar pattern is observed in Figure 26 on the cooling side. The TIER plant's estimated daily average cooling COP generally exceeds the modeled daily average cooling COP throughout the study period. Although both curves show similar seasonal trends, the TIER plant's cooling COP shows higher fluctuations, which may be attributable to variability in load or chiller cycling. On the other hand, the modeled daily cooling COP trend is smoother with less fluctuation.

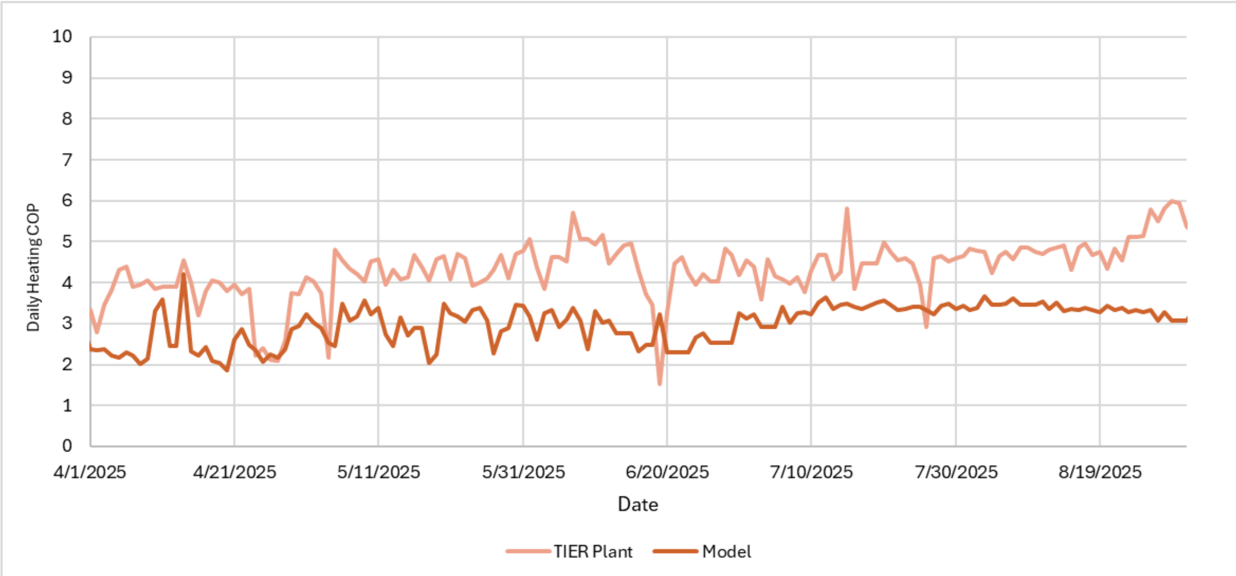

Figure 25: Daily average heating COP for the TIER plant and model from April through August.

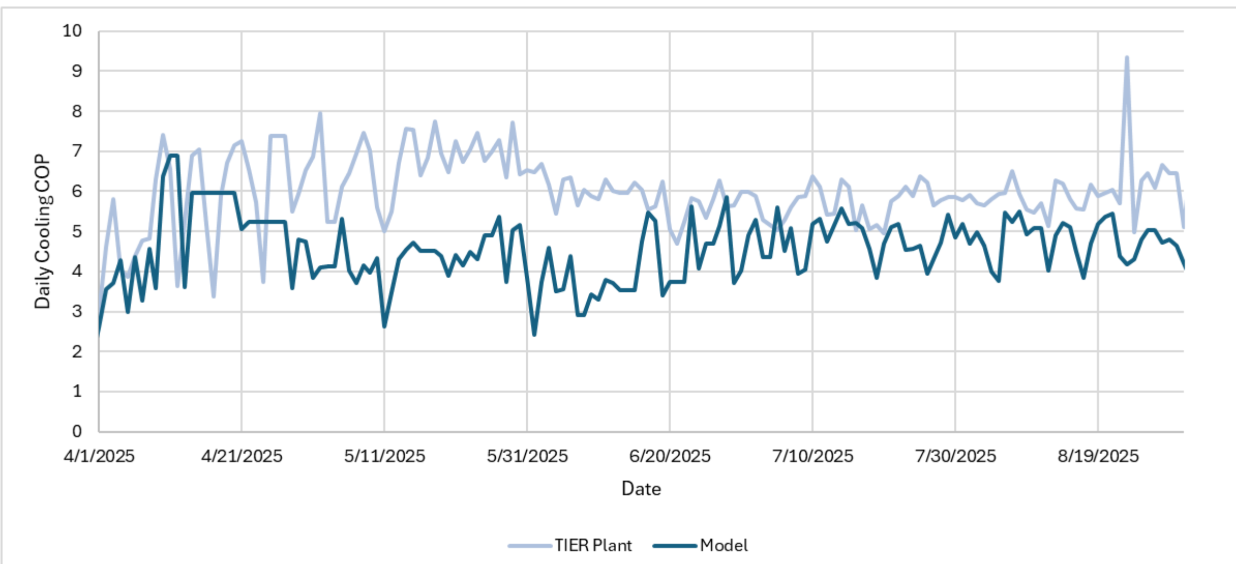

Figure 26: Daily average cooling COP for the TIER plant and model from April through August.

Figure 27 shows that combined COP decreases as HDD and heating demand increase for both the TIER plant and the model, indicating less favorable operating conditions during colder periods. However, the TIER plant consistently achieves higher combined COP across all HDD bins.

As shown in Figure 28 combined COP increases with higher CDD, which is consistent with improved efficiency at higher and more stable cooling loads. The TIER plant maintains a higher combined COP across all CDD bins, especially in low to moderate weather conditions. At higher CDD levels, the gap between measured combined COP and modeled combined COP narrows, which implies that the model better captures near-full-load cooling operation. This is consistent with known differences between the modeled and actual part load performance of the centrifugal chiller.

Overall, the degree day analysis confirms that the TIER plant outperforms the model, which is consistent with the time series view of heating and cooling COPs. The performance gap is most pronounced under moderate weather conditions, where operational strategies such as reduced cycling and effective use of thermal energy storage likely play a larger role than is reflected in the model assumptions.

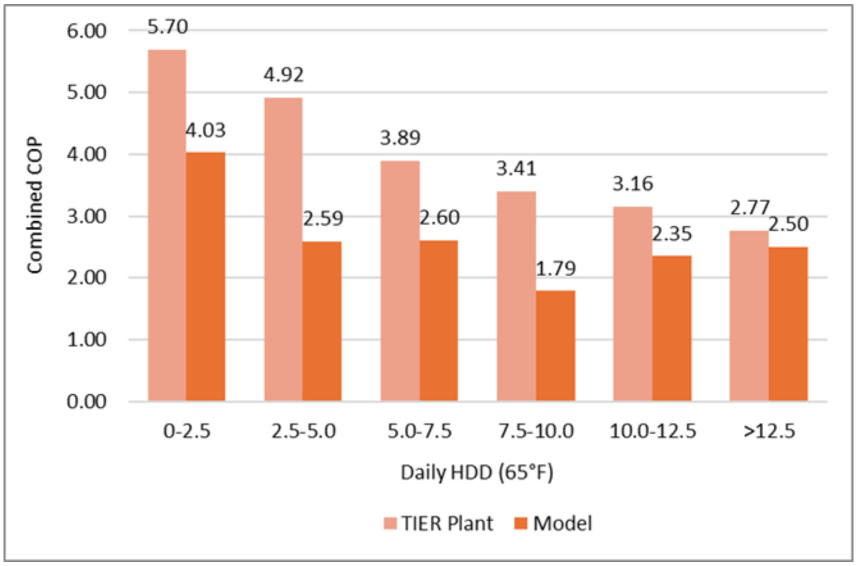

Figure 27: Combined COP of the TIER plant and the model by HDD.

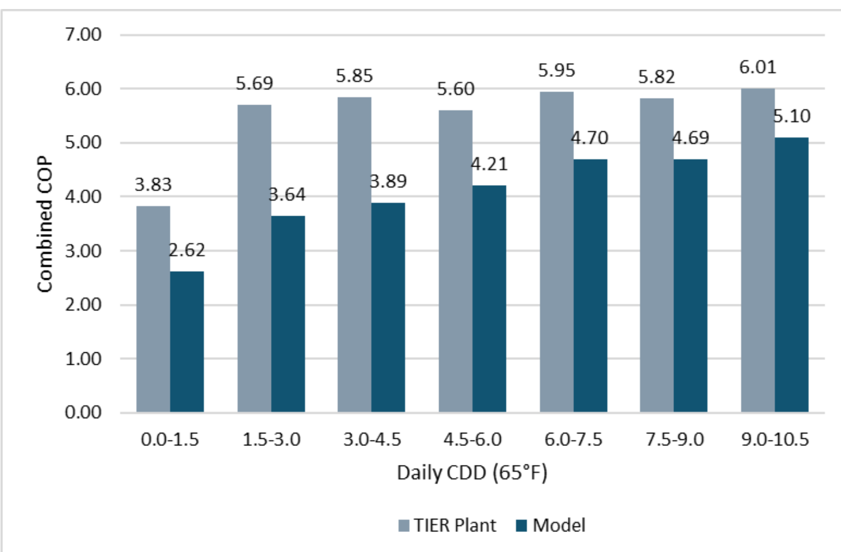

Figure 28: Combined COP of the TIER plant and the model by CDD.

The heating load–binned results show that the TIER plant achieves a higher combined COP than the model across all peak daily heating load bins (see Figure 29). The TIER plant exhibits its highest efficiency at moderate heating loads (20 to 40 percent). The combined COP for the TIER plant decreases at higher load bins as heating demand increases. The model follows a similar pattern but remains generally lower and flatter across bins, which indicates limited sensitivity of the model to changes in heating load (see Figure 29). This suggests that the model may underrepresent efficiency gains at moderate loads and may not fully capture part-load advantages or operational strategies that improve performance under typical heating conditions.

A clearer and more monotonic trend is observed in the cooling load–binned results presented in Figure 30. Combined COP increases steadily with peak daily cooling load for both the TIER plant and the model, which shows improved efficiency at higher and more stable cooling loads. The TIER plant outperforms the model across all cooling load bins, and the largest relative differences are observed at low to moderate cooling loads. This pattern indicates that the model more closely represents near-to-full-load cooling operation but may underestimate efficiency during part-load conditions.

In summary, the peak load analysis shows that the TIER plant's performance advantage is most pronounced under moderate and part-load operating conditions, while the model better approximates behavior at higher loads. These results suggest that incorporating more detailed representations of part-load control, cycling effects, and operational flexibility could improve the model's ability to replicate observed plant performance across the full range of operating conditions.

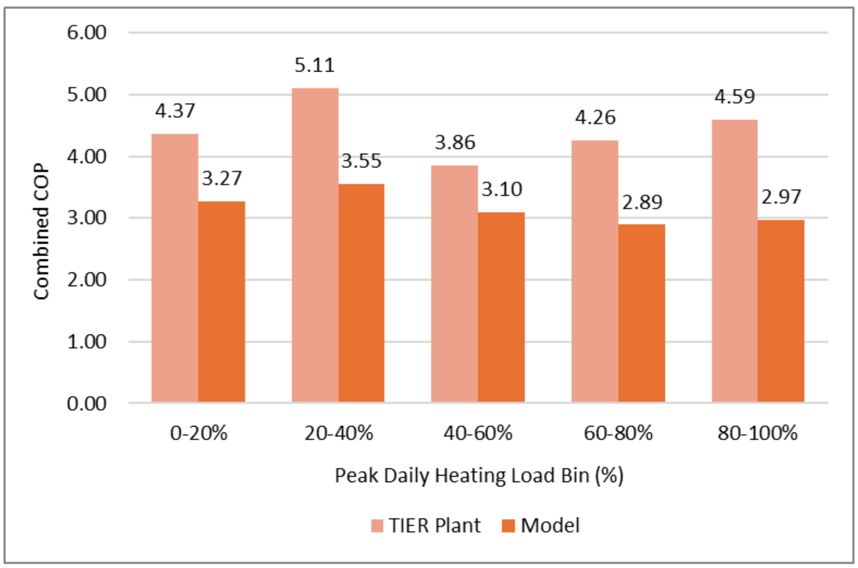

Figure 29: Combined COP of the TIER plant and the model by peak heating load range.

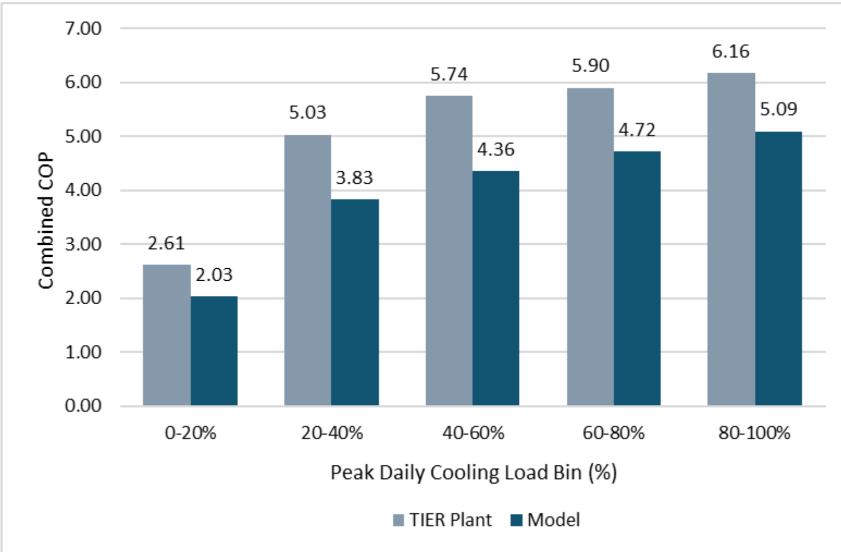

Figure 30: Combined COP of the TIER plant and the model by peak cooling load range.

Performance Comparison with Other All-Electric Plants

There are few published studies reporting on the as-installed performance of AWHPs. The project team reviewed published data and compiled additional data to provide a point of reference for comparison with the TIER plants. Though building types and load profiles will vary, the intent is to gather sufficient data to better understand typical operating efficiency ranges and to allow for comparison of plant efficiencies under different operating conditions and seasons. Individual equipment have reported efficiencies at rated conditions, but most HVAC equipment seldom operate at steady-state, rated conditions. Rather, HVAC systems typically spend most of the time operating at part-load conditions, where efficiencies can be widely variable for different system types and configurations. Many all-electric plants also employ heat recovery capabilities, though the heat recovery efficiencies are generally based on rated conditions where heating and cooling loads are perfectly matched. In reality, loads are highly variable and often imbalanced, which can limit both heat recovery capacity and efficiency. Large all-electric HVAC plants also comprise many different components that are incorporated into custom built-up systems that must be controlled together as a system. How well each of the components are controlled and staged can have a significant impact on energy performance.

Table 3 presents a summary of the compiled all-electric central plants and the properties of the associated buildings and HVAC systems served. All the plants included in Table 3 consist of fixed speed AWHPs, though some units have two or four compressor stages. One system is four-pipe with heat recovery, whereas the others are two-pipe systems. Information and findings from systems #1 through #3 are from a Pacific Gas & Electric Company (PG&E) study (Weitze, 2024).

Table 3: Description of reference all-electric central plant systems.

No.	Building Type and Year	Climate Zone	Building Size (ft^2)	All-Electric Plant	HVAC System Served
1	Library (2018)	ASHRAE 3C CA CZ 3	22,000	One 2-pipe AWHP, 2 fixed speed scroll compressors. Heating-only. 139 kBtu/h.	Dedicated outdoor air system + radiant
2	Office (2018)	ASHRAE 3B CA CZ 12	68,200	One 4-pipe AWHP with heat recovery, 4 fixed speed scroll compressors. 79 tons, 576 kBtu/h.	Radiant floor + fan coils
3	Office (2018)	ASHRAE 3B CA CZ 12	68,200	One 2-pipe AWHP, 4 fixed speed scroll compressors. 66 tons, 513 kBtu/h.	Dedicated outdoor air system
4	Academic Science (2019)	ASHRAE 3C CA CZ 3	38,300	Two 2-pipe AWHPs, each with 4 fixed speed scroll compressors. Each: 85 tons, 736 kBtu/h.	VAV air handling units with 100% outside air, 24/7 operation, 4-pipe VAV terminals

Note: Source for Systems 1 through 3: (Weitze, 2024)

A comparison between the design and measured coefficient of cooling performance (COPc) and COPh values from each study site is shown in Figure 31. Across the different plants, measured COPs are sometimes significantly higher or lower than the design values. Design COPs are generally evaluated at full load and at the extreme ambient conditions, whereas equipment spend the majority of operation time at part loads and off-design ambient conditions.

The heating-only, two-pipe AWHP in System #1 had a design COPh of 2.7 at 37°F outdoor air temperature, but the load-weighted average COPh was measured at 1.85. Weitze reported decreasing COPh at lower part-load factors and showed that the system spent more than half of its operating time below 25 percent load factor and more than 86 percent of the time below 50 percent (Weitze, 2024). This system only had two compressor stages, so excessive cycling at low loads is apparently a primary factor for the low measured COPh. However, even at higher part-load factors and off-peak weather conditions, measured COPh values were considerably lower than the design; for example, in one instance, the measured 2.1 COPh at 40°F outdoor air temperature while operating between 75 percent and 99 percent load factors.

The four-pipe AWHP in System #2 had measured COPc and COPh values (5.9 and 4.6, respectively) that were considerably higher than design (2.9 and 2.6, respectively), despite reporting that the unit typically operated below 50 percent load factor in cooling mode and about half of the time below 20 percent load factor in heating mode. The 2-pipe AWHP in System #3 also had measured cooling COPs well above design (4.7 vs. 2.8), despite typically operating below the 25 percent load factor. Though System #2 spent a considerable amount of time heating at low loads and System #3 cooling at low loads, there appears to be minimal negative impact on COP, particularly in comparison to System #1. It may be that a more granular review of how much time and how far below the minimum turndown is needed to understand this impact better. The primary hypothesis for the higher than design COPs for System #2 and #3 is that the units primarily operate under ambient conditions that are much milder compared to the design conditions for California CZ 12. In heat recovery mode, (Weitze, 2024) reported that System #2 operated at a combined COP of 4.2, well below the design COP of 9.74, with the reason given that the design efficiency assumes that the unit is recovering heat at the full design cooling and heating capacities, which never occurs. In practice, there were relatively few hours with simultaneous heating and cooling and heat recovery loads were generally less than 10 percent of the design capacity.

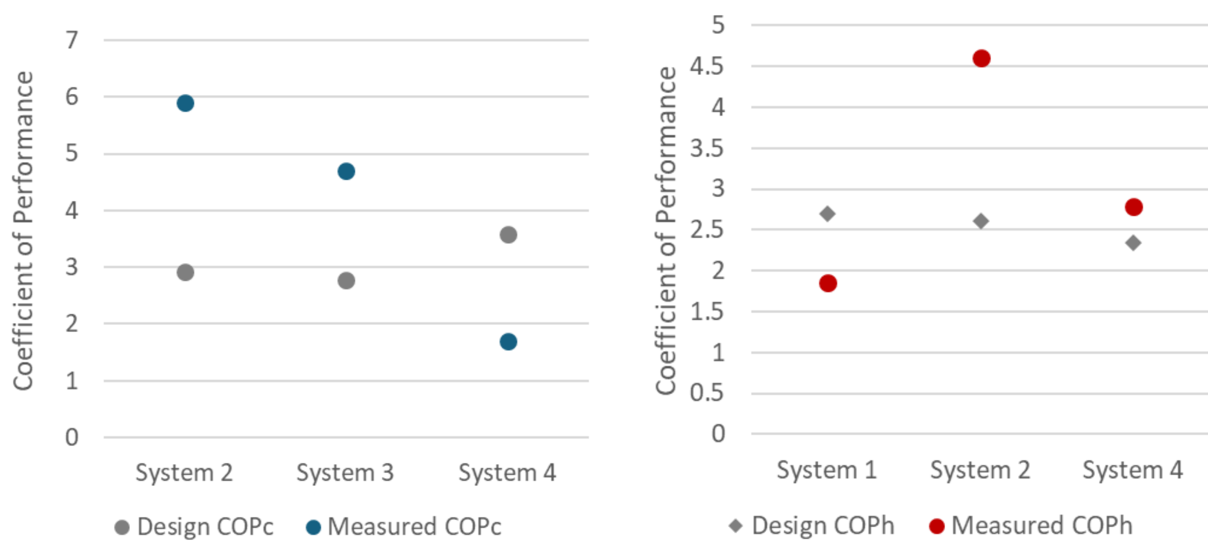

Figure 31: Comparison of design and measured cooling and heating COPs for reference all-electric central plants.

The two-pipe AWHPs in System #4 operated fairly close to the design COPh (2.78 vs 2.35), where the part load distribution suggests that the units were able to operate in heating above the minimum turndown limit for the majority of the time. However, the measured COPc of 1.7 was less than half of the design COPc of 3.6. The cooling load distribution was heavily skewed toward very low part loads, with more than 50 percent of the time below 5 percent part load and more than 75 percent below 10 percent part load, where full load is considered to be the nominal capacity of one unit. The mismatch in peak heating versus cooling loads suggests that the two-pipe changeover system may not have been the right fit for this application but the extremely low cooling part loads were exacerbated by an issue with the staging control logic. The control logic would stage on the lag heat pump if there was demand for more cooling from the zones, regardless of the actual load and whether an additional unit was needed. In practice, there were many hours where both units were running in cooling mode at very low part loads because some zones demanded more cooling, even though the CHW temperature and pumping were at their design conditions.

Overall, the TIER plant's measured COPc of 5.96 and COPh of 4.2 significantly outperform most of the other AWHP plants shown in Table 3 and Figure 31 as expected. The 4-pipe AWHP in System #2 achieved heating-only and cooling-only COPs that closely matched that of the TIER plant. Note, however, that the reported TIER plant COPs are inclusive of the entire plant, including pumping energy, whereas the performance for Systems #1 through 4 represent heat pumps alone. The energy from the constant pumping in System #2 was separately reported to nearly match the heat pump energy use, which would essentially halve its COPs if included (Weitze, 2024). Further, for a direct comparison against the TIER plant metrics, the System #2 performance in heat recovery mode should be integrated in its COPs, which would also decrease those values slightly.

All of the reference AWHP plants evaluated used fixed speed scroll compressors. As newer modular, variable speed AWHPs are installed and operated, operational COPs are expected to be higher than reported at some of the sites here, because of vastly improved part load performance.

Detailed Findings and Discussion

The buildings evaluated in this study represent the first two fully constructed TIER central plants, providing the earliest real-world data for a system that had previously only been examined in concept and through simulation. Across both installations, the measured performance showed that TIER consistently outperformed the AWHP baseline, validating the core premise that coordinated heat recovery and TES can deliver higher annual efficiencies than conventional all-electric strategies.

A notable trend across both the TIER sites and the AWHP reference buildings was the large number of hours with very low heating and cooling loads. At both study sites, specialty programs require 24/7 low load chilled water service for dehumidification and temperature control. When cooling-chillers cycled off, humidity and temperature levels quickly exceeded strict limits, so the cooling-chillers were tasked to reduce downtime to the maximum extent possible. Typical building occupancies have more relaxed humidity and temperature requirements, allowing chillers to remain off for longer periods to accumulate sufficient load for sustained operation, which minimizes adverse impacts on equipment efficiency and longevity. The unique specialty program environmental quality requirements were introduced when the TIER plants were finishing installation, so pony chillers or dedicated cooling systems could no longer be easily added to the projects.

Operation below equipment turndown limits led to frequent cycling of chillers and HRCs, reducing overall COPs and underscoring the importance of designing for realistic load profiles and adequate turndown capability. Because of these load characteristics, caution is needed when comparing COPs to values reported from other plants or studies; differences in calculation conventions can lead to misleading comparisons, and total COP is often the most appropriate metric for integrated systems like TIER.

The actual TIER plant also performed better than the modeled results. This is likely due to differences between the part-load performance curves used in the simulation and the actual behavior of the installed equipment. Given how often most buildings can operate at low loads, small deviations in part-load assumptions can significantly affect annual energy predictions.

Finally, the implementation of the first operational TIER plant produced numerous practical lessons related to system integration, controls coordination, commissioning complexity, and operator training. These experiences highlight that while TIER is capable of high performance, successful deployment requires careful engineering and attention to operational realities observed during startup and early operation.

Detailed review of the system installation and early performance offered the opportunity to learn from challenges faced and improve on future plant designs. Some of the detailed findings include:

- Chillers may not have the internal safeties in their factory software necessary to safeguard the machines from operating at sub-freezing refrigerant saturated suction temperatures—a condition that risks sub-freezing water temperatures within the evaporator heat exchanger and subsequent tube ruptures from ice formation in systems without glycol. Water-to-water heat recovery chillers are particularly susceptible to running out of a source of heat especially since a source of heat is typically not plentiful during times when HW must be produced. The TIER plant leverages a TES tank to ensure that HRCs have a reliable heat source, but AWHP failures and insufficient heat recovery can cause consistent low day to day TES tank charges. In the event that the TES tank charge is depleted of 80°F and 60°F water, the BAS must include supplemental safeties for refrigerant low saturated suction temperatures since the factory chiller software may not have these safeties at all. BAS control instabilities and transient conditions can also cause low source water temperatures to the HRC evaporators even with a charged TES tank so safeguards beyond TES tank charge alarms that directly look at refrigerant low saturated suction temperatures are necessary. We have also observed saturated suction temperatures drop for no external reason at all when steady warm source water was being provided to the HRC. These BAS safeties are addressed in the updated CW TIER sequences of operation included with the Design Guide (provided as a supplemental document).
- CW TIER plants can experience rapid temperature fluctuations in the CW loop. The condenser barrels of cooling chillers are interconnected with the evaporator barrels of heat recovery chillers so changes in leaving water temperatures into the shared CW loop from either set of barrels can directly impact the entering water temperatures of the other chillers and cause equipment trips and failures. On top of chiller capacity control instabilities during operation that can cause temperature fluctuations in the CW loop, chillers can also entirely bypass their CW during startup while they prove flow before turning on and during shutdown when they require that flow be maintained without any active heat exchange. The CW loop is connected to the AWHPs, TES tank, and cooling towers, each of which are engaged during different modes of operation. Switching between modes of operation causes temperature and flow fluctuations in the CW loop that can prove challenging for maintaining stable chiller operation. Carefully re-tuning loops and imposing ramp rates on valve modulation, temperature setpoint changes, and flow setpoint changes without impeding the ability to provide cold or hot CW when needed by the equipment is critical. Other chillers may also be less sensitive than the chillers installed at the study sites.
- The CW TES tank may instigate more biological growth throughout the plant's hydraulically interconnected CW, building CHW, and building HW loops. Contractors must be tasked with carefully performing flush-out and passivation activities prior to startup. Also, depending on the

heat load during warmer months, a large volume of water in the TES tank and even heat recovery chillers may remain stagnant and held at 80°F which are ideal conditions for biological growth. A rigorous chemical treatment plan must be developed for a CW TES application including exercise modes.

- Cooling chillers must be able to operate with low CW flows without issue or else a recirculating bypass is needed to achieve target temperature differentials in the CW loop. Temperature differential control is critical since the cooling chillers are directly connected to the heat recovery chillers. Without the ability to throttle CW flow through cooling chillers at minimum load, the cooling chillers cannot generate enough heat to match TES tank temperatures and feed the heat recovery chillers. CW minimum flow requirements for the cooling chillers can also exceed heat recovery chiller design evaporator flows even when the building's heating load greatly exceeds the building's cooling load. Whenever flow through the cooling chiller condenser barrels exceed the flow through the heat recovery chiller evaporator barrels, the net effect is that flow from the top of the TES tank reverses. The heat recovery chillers source potentially cold water entirely from the cooling chillers if target temperature differentials are not maintained. Cold water also feeds into the top of the tank which can destabilize the tank's thermal stratification. Manufacturers must be pressed during chiller selections to note the true minimum CW flow rate requirement of their equipment and given any realistic limitations, designers must be prepared to design a recirculating loop as needed to achieve proper temperature differential control.
- Staging chillers on and off in the highly interconnected CW TIER plant can prove abnormally complicated when trying to sequence chillers, pumps, and valves. When a heat recovery chiller switches which loop its evaporator or condenser barrel belongs to, it may be preferable to simply shut down all chillers before reconfiguring the plant and restarting the chillers to avoid trips.
- Deciding when and how to accelerate TES tank charging is complex. There are two methods: (1) manually disable airside economizers to amplify the heat source for cooling chillers to reject heat to the TES tank (more energy efficient than running AWHPs), and (2) enable AWHPs to supplement the recovered heat from the cooling chillers. Allowing the tank charge to drop too far before disabling airside economizers results in a lower plant heating COP since the missing heat needs to be produced using the AWHPs which are comparatively less efficient than a centrifugal chiller. Conversely, not allowing the tank charge to drop enough before triggering either of these control sequences can result in additional heat that must get rejected after the TES tank is already fully charged. Cooling towers must then turn on only to reject the heat that the plant either artificially created through false loading in the first place or inefficiently created with AWHPs. Disabling airside economizers is only efficient when the extra heat can be used for heating duty. Tuning the parameters associated with false loading the plant and enabling AWHPs is critical. Self-tuning algorithms can be used, but the initial value on each self-tuned parameter should be initialized at values that appropriately reflect the active seasonal load requirements at the time of startup.
- There can be a discrepancy between the AWHPs internal temperature sensor and the corresponding BAS temperature sensor. As the less efficient secondary tank recharging device, AWHPs are typically controlled to operate at maximum capacity and as close in time as possible to the building's morning warm-up period to allow the more efficient heat recovery system as much time as it can have. The AWHPs can be forced to run at maximum capacity by

recirculating just enough cold inlet water to prevent the AWHPs from reaching their target leaving water temperature setpoint. However, if the internal temperature sensor reads high, the AWHPs will back off and recharge the tank at a slower rate which will result in low TES tank charges to meet the next day's heating demands.

Simplified Tool

Conventional building energy simulation software like EnergyPlus are not capable of modeling complex and novel HVAC solutions like the TIER plant. Though efforts are underway to expand the capabilities of EnergyPlus in this area, there is significant complexity with the staging and control of equipment in a TIER plant that would be difficult to simulate in EnergyPlus. Recent research developments in Modelica and the control description language have included implementations of the TIER plant and control logic but these simulation environments are currently too complex to be used other than for large research efforts (Lawrence Berkeley National Laboratory, n.d.).

The Project Team developed simplified spreadsheet models of two TIER plant configurations to provide a more practical way for designers to explore and analyze the energy performance of TIER plants. Figure 61 in Appendix A illustrates the system diagram for a TIER plant that uses CW as the storage medium with cascading refrigeration cycles. Figure 62 in Appendix A illustrates an alternative TIER plant approach using HW as the storage medium. The spreadsheet models are hourly simulations of the TIER plant. Model inputs are clearly shaded in yellow and include:

- Annual building HW and CHW load profiles (from separate building simulations or measured data from real buildings)
- Equipment performance data, such as part load curves (Figure 32) or lookup tables (Figure 33)
- HVAC equipment selection data
- HVAC equipment staging details (Figure 34)
- Utility rates
- Carbon emissions factors

Separate spreadsheet models were developed for a CW TIER plant and HW TIER plant. Detailed descriptions of the plant models are provided in Appendix B. The simulations allow for detailed modeling of equipment staging and charging/discharging of TES tanks, while accounting for system dynamics, part load performance, and equipment control. The model outputs include annual HVAC energy, energy cost, and emissions (Figure 35).

For ease of use, each simulation model is pre-populated with input data, including equipment details for real-world selections. The models also include a limited number of user-selectable options for exploring different building climates, load profiles, utility rates, and marginal emissions factors, or users may also have the option of entering their own custom data (Table 4).

The CW TIER plant spreadsheet model is available online here. The HW TIER plant spreadsheet model is available online here.

Each model compares the proposed case TIER plant with a baseline plant under the same load profile and ambient conditions. The baseline plants were modeled entirely in EnergyPlus, and consist of water-cooled chillers and natural gas boilers. Baseline energy use was determined for each CZ and cooling baseload configuration for a total of six load profile and energy use pairings (Table 4).

Energy use outputs for these baselines are delineated by use type, allowing the team to individually account for space cooling, heating, pump energy, fan energy, lighting, and miscellaneous equipment usage over the course of the year.

The baseline load profiles and energy use outputs are referenced in the spreadsheet models such that the proposed TIER plant serves the exact same load profile as the baseline. Determination of final energy use in the proposed TIER plant relies on both the internal spreadsheet calculations and the baseline energy use data. Central plant energy, including cooling, heating, and pump energy, is calculated from scratch in the spreadsheet model since these are heavily dependent on plant configuration, then combined with the air handler, lighting, and miscellaneous equipment energy from the baseline to compile the full proposed case energy use. Lighting, air handler, and miscellaneous energy use are equal between the baseline and the proposed cases for a given scenario but were still included in the analysis so that demand charges could be assessed on a whole-building basis.

Both TIER plant concepts include a control strategy which throttles the airside economizer to artificially increase the mechanical cooling load. This seems non-intuitive because it increases the total plant cooling load and sacrifices “free” cooling energy, but this is not the case if we consider the implications for heating. When the airside economizer is throttled to provide minimum outside air and the additional mechanical cooling load is served by a heat recovery chiller, this cooling can still be considered “free” because the plant is recovering the heat from the additional cooling load. Whether the additional cooling load is handled by the airside economizer or a heat recovery chiller, there is no explicit energy use associated with this load since the energy consumption of the heat recovery chiller is considered heating energy. In fact, throttling the economizer and recovering heat from the mixed air is more energy efficient than the alternative of relying on the AWHP to source heat instead from colder ambient air.

Conventional efficiency metrics such as COP are difficult to quantify in heat recovery plants, since a single machine can simultaneously heat and cool. Rather than reporting a combined COP which can potentially be misleading, the project team decided to standardize the plant level COP reporting based on the “free cooling” phenomenon described above. Any time a chiller creates CHW and the cooling tower loops are off, this energy is accounted for in the plant level heating COPh since this load will either charge the TES tank or serve the building heating loop directly. When the tower water loop is active, chillers with their evaporators indexed to the CHW loop are accounted for in the plant level cooling COPc. The only exception to these rules is that the HRC in the HW model is always considered heating energy since its condenser is always indexed to the HW loop and tank.

CH-1 Curves

	Capf(CHWST,CWRT)	EIRf(CHWST,CWRT)	EIRf(CWRT,PLR)
c	7.77E-01	1.13E+00	4.30E-02
x	-2.15E-02	-4.20E-02	1.74E-02
x2	-1.08E-03	7.57E-04	-2.73E-05
y	2.75E-02	-2.04E-02	-5.45E-01
y2	-7.94E-04	7.68E-04	2.16E+00
xy	1.67E-03	-2.75E-04	-1.60E-02
x3			0.00E+00
y3			-6.72E-01
x2y			0.00E+00
xy2			0.00E+00
minx	4.4	4.4	17.6
maxx	8.9	8.9	35.3
miny	21.3	21.3	0.09
maxy	41.1	41.1	1.01
Value	0.96	1.06	1.00

NomCap	**395.6 tons**
NomEIR	**0.1357**

AHRI Condition Check

CHWST	6.7°C
CWRT	35.0°C
Capf(CHWST,CWRT)	0.97
Cap	**382.1 tons**
PLR	99.4%
EIRf(CHWST, CWRT)	1.05
EIRf(CWRT,PLR)	0.99
EIR	**0.1408**

54.44126074

Figure 32: Example chiller part load performance curves from simplified tool.

ASHP-1,-2 Performance Table

OAT	HWST	Heating Capacity	Power	COPh
23.0°F	77.0°F	1,189,009 Btu/hr	118.6 kW	2.9
32.0°F	77.0°F	1,273,015 Btu/hr	119.2 kW	3.1
41.0°F	77.0°F	1,554,022 Btu/hr	122.1 kW	3.7
44.6°F	77.0°F	1,726,868 Btu/hr	124.0 kW	4.1
50.0°F	77.0°F	1,905,627 Btu/hr	125.7 kW	4.4
59.0°F	77.0°F	2,062,428 Btu/hr	127.1 kW	4.8
68.0°F	77.0°F	2,172,616 Btu/hr	128.3 kW	5.0
77.0°F	77.0°F	2,247,356 Btu/hr	129.2 kW	5.1
86.0°F	77.0°F	2,297,808 Btu/hr	130.0 kW	5.2
100.0°F	77.0°F	2,297,808 Btu/hr	130.0 kW	5.2
23.0°F	84.0°F	1,174,294 Btu/hr	126.1 kW	2.7
32.0°F	84.0°F	1,253,589 Btu/hr	126.4 kW	2.9
41.0°F	84.0°F	1,528,953 Btu/hr	129.1 kW	3.5
44.6°F	84.0°F	1,699,282 Btu/hr	130.9 kW	3.8
50.0°F	84.0°F	1,879,891 Btu/hr	133.0 kW	4.1
59.0°F	84.0°F	2,040,223 Btu/hr	134.6 kW	4.4
68.0°F	84.0°F	2,153,885 Btu/hr	136.1 kW	4.6
77.0°F	84.0°F	2,232,042 Btu/hr	137.3 kW	4.8
86.0°F	84.0°F	2,285,855 Btu/hr	138.4 kW	4.8
100.0°F	84.0°F	2,285,855 Btu/hr	138.4 kW	4.8
23.0°F	95.0°F	1,151,171 Btu/hr	137.8 kW	2.4
32.0°F	95.0°F	1,223,062 Btu/hr	137.8 kW	2.6
41.0°F	95.0°F	1,489,560 Btu/hr	140.1 kW	3.1
44.6°F	95.0°F	1,655,933 Btu/hr	141.7 kW	3.4
50.0°F	95.0°F	1,839,448 Btu/hr	144.4 kW	3.7
59.0°F	95.0°F	2,005,329 Btu/hr	146.5 kW	4.0
68.0°F	95.0°F	2,124,451 Btu/hr	148.4 kW	4.2
77.0°F	95.0°F	2,207,976 Btu/hr	150.1 kW	4.3
86.0°F	95.0°F	2,267,071 Btu/hr	151.6 kW	4.4

Figure 33: Example AWHP performance data from simplified tool.

Chiller Staging Tables

CH-1

Chiller Status Table (0 = OFF, 1 = ON)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	0	1	1	1	
	2,273 MBH	0	1	1	1	
	8,000 MBH	0	1	1	1	
	7,500 MBH					
	12,000 MBH					

HRC-1

Chiller Status Table (0 = OFF, 1 = ON)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	0	0	1	1	
	2,273 MBH	1	1	1	1	
	8,000 MBH	1	1	1	1	
	7,500 MBH					
	12,000 MBH					

Evaporator Index Table (0 = CHW, 1 = CW)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	1	1	0	0	
	2,273 MBH	1	1	1	0	
	8,000 MBH	1	1	1	0	
	7,500 MBH					
	12,000 MBH					

Condenser Index Table (0 = CW, 1 = HW)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	1	1	0	0	
	2,273 MBH	1	1	1	1	
	8,000 MBH	1	1	1	1	
	7,500 MBH					
	12,000 MBH					

HRC-2

Chiller Status Table (0 = OFF, 1 = ON)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	0	0	0	1	
	2,273 MBH	0	0	1	1	
	8,000 MBH	1	1	1	1	
	7,500 MBH					
	12,000 MBH					

Evaporator Index Table (0 = CHW, 1 = CW)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	1	1	0	0	
	2,273 MBH	1	1	0	0	
	8,000 MBH	1	1	0	0	
	7,500 MBH					
	12,000 MBH					

Condenser Index Table (0 = CW, 1 = HW)		CHW Load				
		0 tons	342 tons	463 tons	999 tons	1,200 tons
HW Load	0 MBH	1	1	0	0	
	2,273 MBH	1	1	0	0	
	8,000 MBH	1	1	1	1	
	7,500 MBH					
	12,000 MBH					

Figure 34: Example equipment staging table from simplified tool.

	Energy Cost ($)	Marginal Emission	Gas Use (therms)
Baseline	$169,480	698,330	38,801
Proposed	$172,246	217,152	-
Savings	-$2,766	481,178	

Annual Heating Loads	
AWHP Heat (kBTU)	206,315
Recovered Heat (kBTU)	3,630,517
Total Heating Loop Load (kBTU)	3,836,832
Heat Recovery %	95%

Annualized Energy Use (kWh)	
Total plant energy	1,005,185
Cooling Equipment energy	660,781
Heating Equipment energy	344,404
Pump energy	78,070
Cooling COPc	4.52
Heating COPh	3.27

Annual Cooling Loads	
Cooling Only Chiller Load (ton-hr	696,751
HRC cooling load (ton-hrs)	153,149
Total Cooling Loop Load	849,900
Economizer Throttling %	1%

Figure 35: Example output from simplified tool.

Table 4: User inputs for simplified tools.

Input	Tool Options
Annual Heating/Cooling Load Profile	• MOB - CA CZ 3 • MOB - CA CZ 9 • MOB - CA CZ 12 • MOB + 50-ton load - CA CZ 3 • MOB + 50-ton load - CA CZ 9 • MOB + 50-ton load - CA CZ 12 • Custom hourly load profile
Utility Rate	• PG&E – Secondary Voltage (B-20) • Peninsula Clean Energy – Industrial Secondary Voltage (B-20-S) • Silicon Valley Power – Non-Time of Use (CB-1) • Silicon Valley Power – Time of Use (CB-1) • Sacramento Municipal Utility District (SMUD) – Commercial & Industrial Time-of-Day (CI-TOD4) • Southern California Edison (SCE) – General Service – Large, Time-of-Use (TOU-8) • Custom utility rate profile
Marginal Emissions Factors	• CAISO 2030 Midcase • Custom marginal emissions factors

Energy Analysis

Summary of Parametric Analysis

Spreadsheet modeling tools were used to conduct a parametric analysis assessing TIER plant performance in a variety of climates and utility rates structures across California. The following parameters account for a total of (36) unique modeling outputs:

- Storage Media (2):
  - [CW] Condenser Water
  - [HW] Hot Water
- Regions: Climate zones and associated utility (3):
  - [CZ03] Oakland – PG&E
  - [CZ09] Los Angeles – SCE
  - [CZ12] Sacramento – SMUD
- IT Baseload Conditions (2):
  - [] 0-ton baseload
  - [50] 50-ton baseload
- TES Tank Sizes (3): TES tank volume and AWHP plant size
  - [1] Intermediate tank size, AWHPs sized to fully recharge the tank throughout the year
  - [2] Larger tank size, AWHPs sized at half of the intermediate tank size
  - [3] Smaller tank size, AWHPs sized at double the intermediate tank size

Each run receives a unique index based on the combination of the bracketed abbreviations defined in the above criteria which identifies the CZ, IT baseload condition, and TES tank sizing. For example, CZ09 with 0-ton baseload and half module configuration is noted as “CZ09-2,” while the same run with a 50-ton baseload is named “CZ09-502.”

In this energy analysis section, performance of the TIER plant is evaluated relative to the performance of a baseline representing a conventional central plant with water-cooled chillers and condensing boilers.

Both HW and CW storage TIER plant configurations performed similarly on an energy cost savings basis, typically yielding 10–30 percent operational energy cost savings compared to the water-cooled chiller and boiler baseline (Figure 36). The HW storage plant yielded slightly better savings than the CW storage plant in most scenarios due to a combination of lowered energy use and lowered energy demand charges.

The delta in demand charges between the models can be traced back to differences in AWHP load. The CW model runs the heat pumps in CZ 03 slightly more often than the HW model to keep the tank charged because the HRCs in the CW storage plant require a sufficient source of heat to operate safely. When the tank charge is depleted and the AWHPs cannot generate a sufficient heat source for the HRCs, the HRCs must cycle off and the building will lose zero heat can be provided to the building when the tank charge is depleted. It is a high priority for CW storage plants to maintain sufficient tank charge. Because it is impossible to accurately predict how much heat recovery will be available from cooling loads before the next morning warmup period, the control strategy for supplemental tank charging will use air-source heat more often to keep the tank charged. Conversely, the HW storage plant can use AWHPs to still meet some of the building heating demands even when the TES tank charge is depleted. Because of the lower risks to equipment safety and losing plant heating capacity outright, HW storage plants can afford to allow tank charge to be depleted more which decreases heat pump usage and increases how much heat recovery can be utilized.

Thermal Energy Storage Media

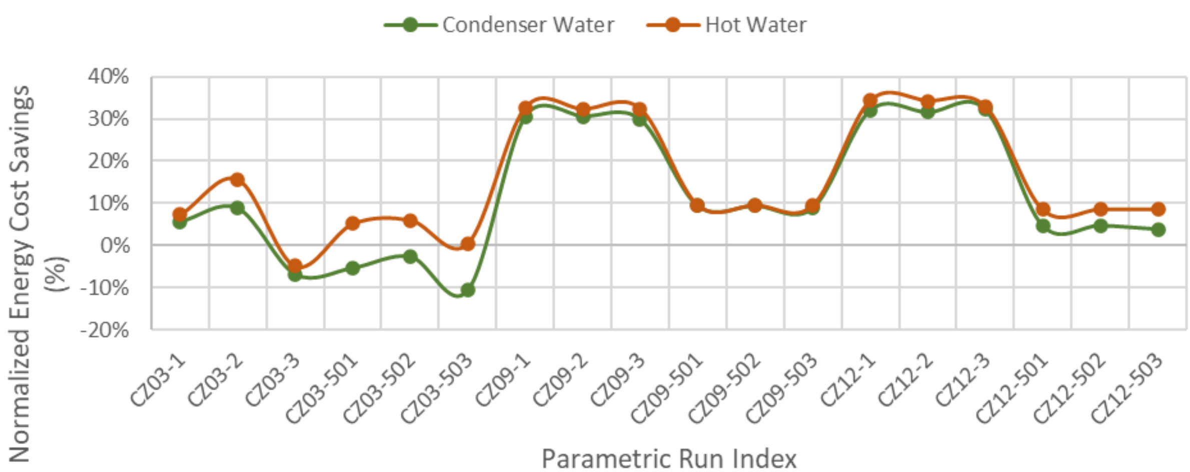

Figure 36: Parametric energy cost savings vs. baseline plant.

Additional supplemental charging in the CW model, as well as differences in efficiency between the individual heat pump performance curves lead to increased electricity use, and thus demand charges. The effect on energy costs is less pronounced for all-electric plants in other CZs because the demand charge penalties for the other utilities are much more relaxed than for PG&E (Figure 45).

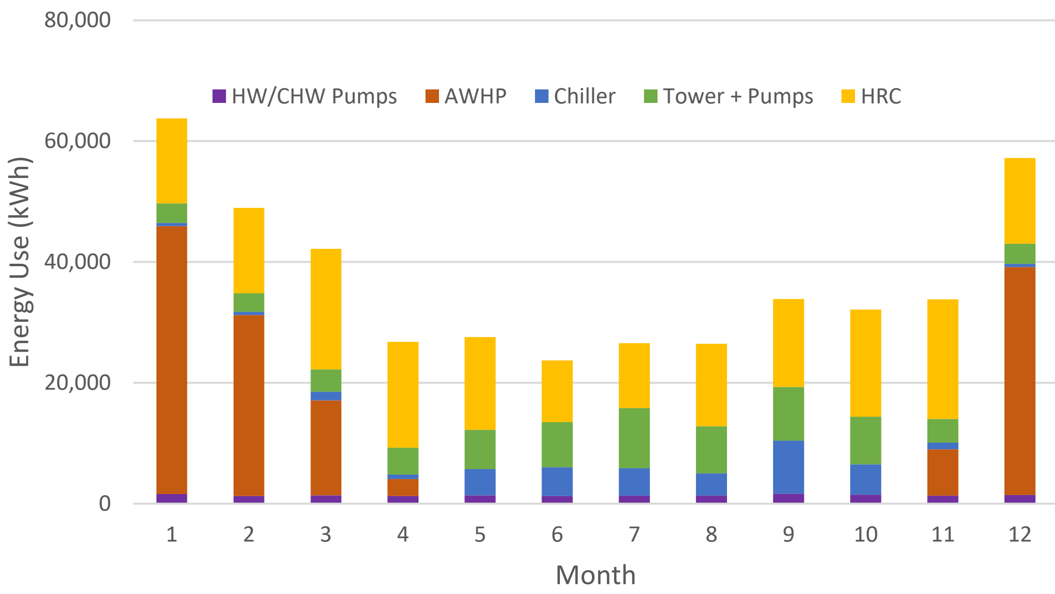

Figure 37: Energy use by month: hot water storage CZ03-01

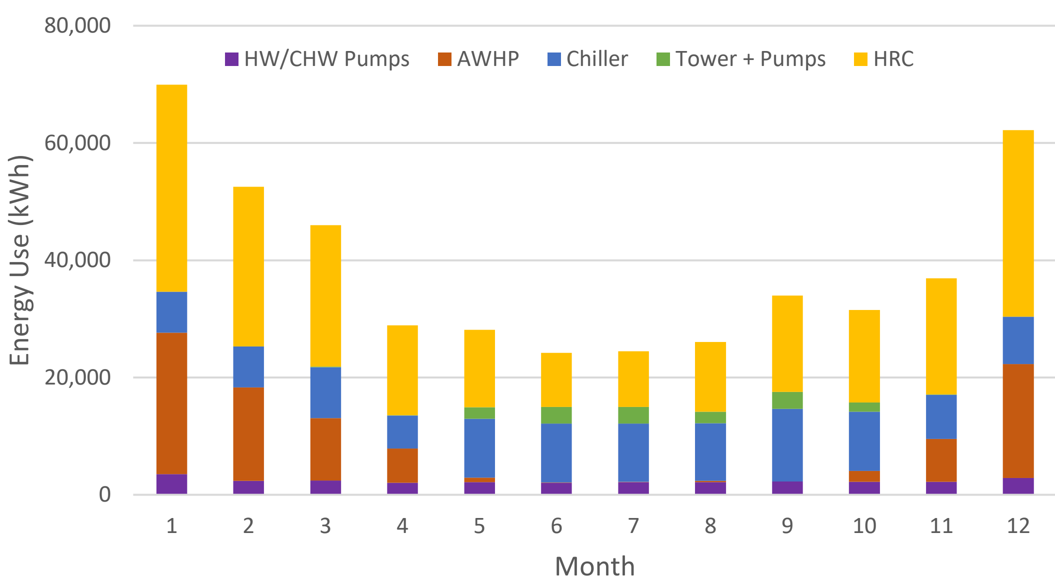

Figure 38: Energy use by month: condenser water storage CZ03-01

The differences in energy efficiency (and thus energy charges) between the condenser water and hot water plant designs are primarily driven by variations in chiller part load efficiencies and staging orders between the models. The modeled condenser water plant uses a large centrifugal chiller as the first stage cooling machine because heat can be rejected to the TES tank at 80°F whereas the modeled HW plant uses a screw HRC as the first stage cooling machine. Because centrifugal chillers are much more efficient than positive displacement screw chillers at most part loads, energy usage during higher cooling load months of the year is lower for CW storage plants as shown in Figure 37and Figure 38. Figure 39 presents the part load performance curves for both types of equipment at the standard lifts used in the models, with the centrifugal chiller curve evaluated at 42°F CHW and 80°F CW and the HRC curve evaluated at 42°F CW and 130°F CW. Chiller curves were selected from the EnergyPlus library based on the closest match available. Minimum part load ratios of 20 percent were applied to all parametric model runs such that the energy use scales linearly with load below 20 percent as if the units are cycling on and off. A linear relationship between energy and load represents constant efficiency below 20 percent part load, as shown in Figure 39.

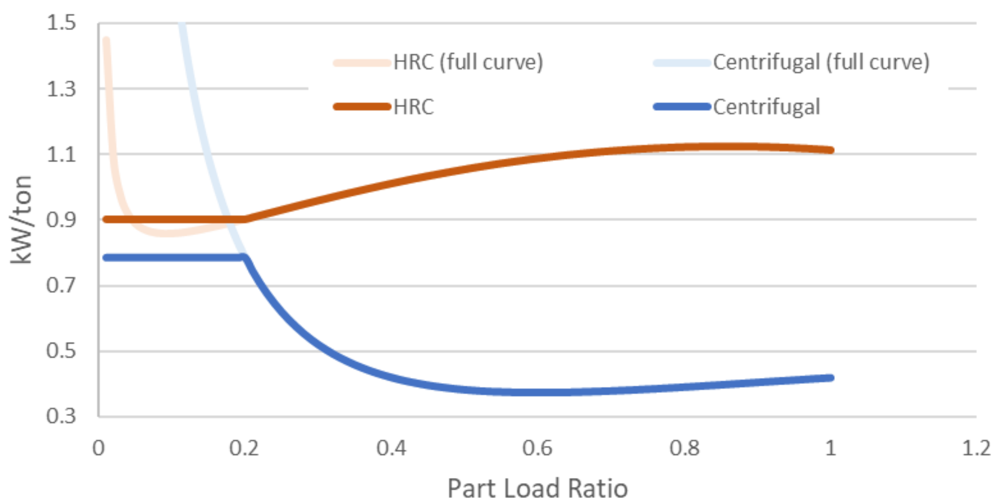

Figure 39: Part load efficiency curves for centrifugal and heat recovery chillers

Chiller performance data are evaluated at 42°F CHW and 80°F CW for the centrifugal chiller and at 42°F CW and 130°F CW for the HRC.

The chillers in both models spend the majority of the year operating below 20 percent part load at the ~0.9 kW/ton efficiency point for screw chillers and ~0.8 kW/ton for centrifugal chillers. While the CW storage plant is able to leverage a centrifugal chiller with slightly better efficiency for heating operation, the CW plant efficiency as a whole is lower than the HW plant. The HRC in the HW model produces HW from a single machine operating with a lift envelope of 42°F to 130°F, while the CW model requires two machines. The centrifugal machine operates with a lift envelope of 42°F to 80°F and the HRC operates with a lift envelope of 60°F to 130°F. While both the centrifugal chiller and the HRC at these conditions are individually more efficient than a single HRC handling the full lift, the 20°F of lift overlap in the CW model means that a single less-efficient machine is more efficient overall for heating operation. As shown in Figure 37 and Figure 38, energy usage during higher heating load months of the year is higher for CW storage plants. Note that the energy use breakdown differences by equipment type is not easily compared since the AWHPs in the CW model must rely on HRC energy use to boost the heat they produce. Running multiple machines simultaneously also sets the demand peaks higher in the condenser water models, especially in regions with utilities like PG&E with high demand charges (Figure 45).

To further understand how the differences in these chiller curves manifest as energy cost savings between condenser water and hot water models, refer to the charts in Figure 40. Equipment efficiency curves and part load histograms are presented using data from the CZ 09 intermediate tank size runs which support the above conclusions. In the CW TIER plant, the centrifugal chiller is the lead chiller and operates at its highest efficiencies above 0.3 part load ratio. Nearly all of the centrifugal chiller's operation, however, is below 0.1 part load ratio and at high lift conditions where the kW/ton is more than twice as high at 0.8 kW/ton. For the HW TIER plant, the HRC is the lead chiller where its high efficiencies are at low part loads. The majority of the HRC operation is also below 0.1 part load ratio at 0.9 kW/ton. Though the centrifugal chiller's kW/ton at these low load conditions is slightly lower than the HRC's, the centrifugal chiller only handles part of the lift for generating HW in the CW TIER plant; the additional HRC power causes the CW TIER plant to consume more power than the HW TIER plant under low load conditions. Note, however, that if the lead chillers operated more frequently at higher part loads, the performance curves would strongly favor the CW TIER plant, as the centrifugal chiller efficiency would significantly increase and the HRC efficiency in the HW TIER plant would decrease.

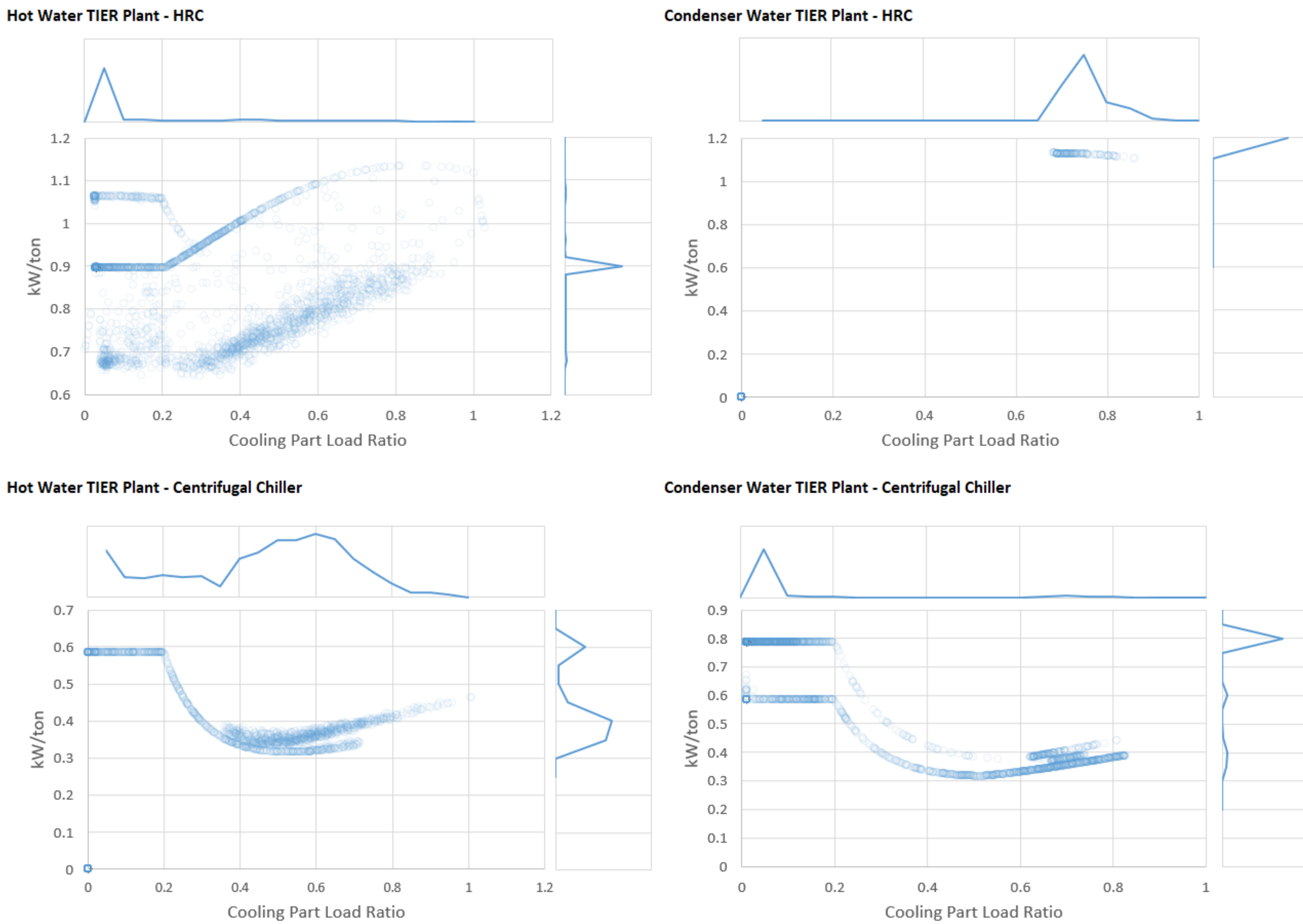

Figure 40: Chiller efficiency as a function of part load ratio for centrifugal and heat recovery chillers

Line charts above and to the right of each plot represent frequency distribution for modeled part load ratio and efficiency (kW/ton)

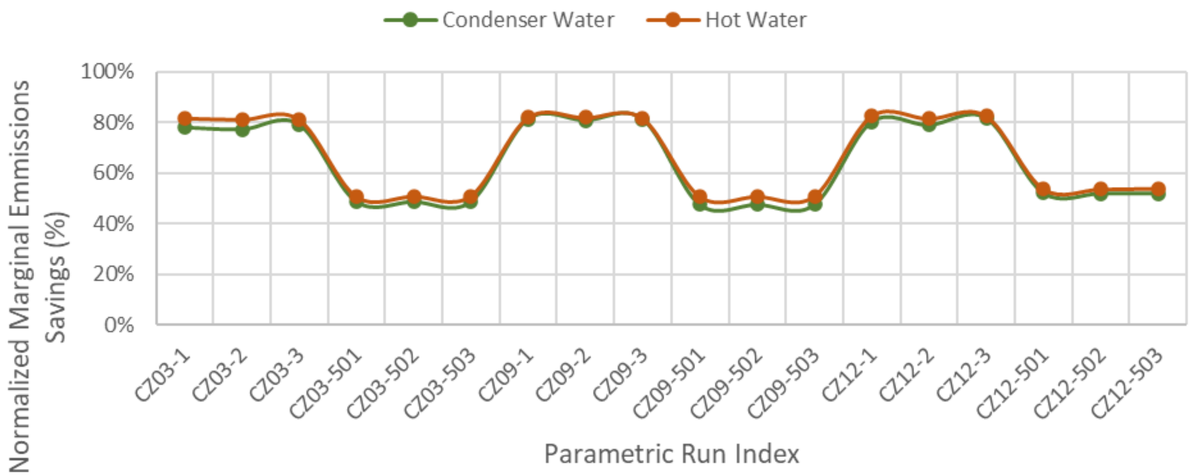

Figure 41: Parametric marginal emissions savings vs. baseline.

Both TIER plant configurations were effective at reducing total energy consumption and therefore marginal emissions (Figure 41). Long range marginal emissions factors were generated using Cambium data sets through the National Laboratory of the Rockies (NLR) Scenario Viewer (National Laboratory of the Rockies, n.d.). Since the marginal emissions factors are representative of the entire California ISO grid, there are no obvious regional differences between parametric runs. Minor changes can be seen as the tank size and the AWHP sizing ratios are varied, but the marginal emissions savings are clearly most sensitive to the IT baseload.

In runs with a 50-ton baseload, the marginal emissions savings are roughly 30 percent less than the equivalent runs without a baseload. Long range marginal emissions factors depend on many factors outside the scope of this analysis, but it stands to reason that the relative savings are lower at high baseloads; high cooling baseload scenarios which achieve near 100 percent heat recovery see much more heat rejection through the towers, bringing the overall plant efficiency closer to that of the water-cooled chiller baseline.

Therefore, it stands to reason that the emissions savings relative to the baseline plant are lower in these runs when compared to scenarios with comparable heating and cooling loads that see a larger efficiency gain from heat recovery.

While both TIER plant configurations are more efficient than the baseline system, the COPs for heating and cooling differ substantially between the plant types (Figure 42 and Figure 43).

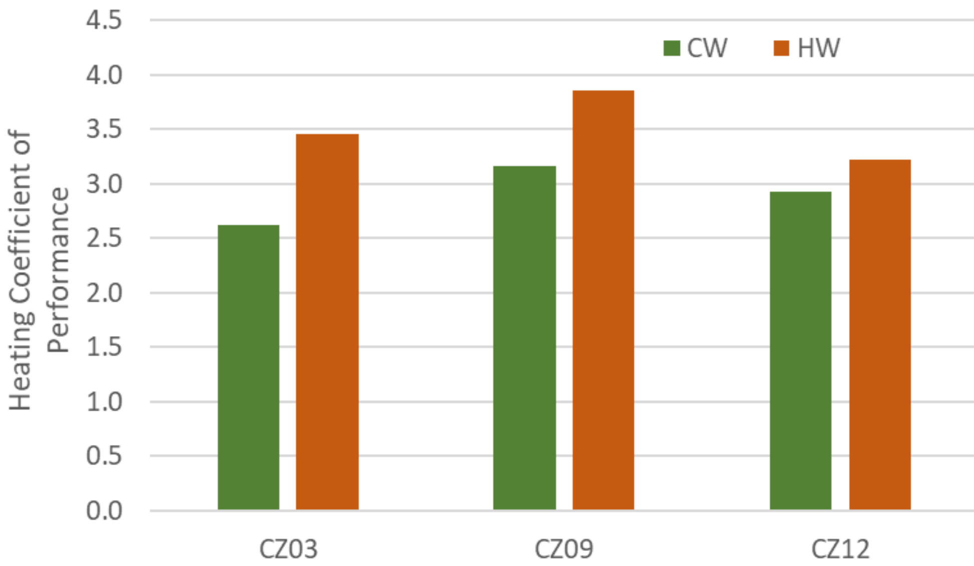

Figure 42: Heating COPh condenser water vs. hot water – intermediate tank with 0-ton baseload.

Heating COPh is considerably higher for HW storage scenarios when compared to the CW heating COPh. Theoretically, the COPh should always be higher for a HW storage plant since the HRCs in this configuration are indexed to the CHW and HW loops except when heating is not needed, allowing heat recovery from a single machine with a single lift envelope (e.g. 42°F evaporator leaving temperature to 130°F condenser leaving temperature). In the CW storage plant, HRCs typically source heat from the CW loop and the TES tank except when cooling demands are high.

While the HRCs operate with a lift envelope of 60°F to 130°F, cooling chillers and AWHPs also have to be engaged to charge the tank. Cooling chillers operate with a lift envelope of 42°F to 80°F, meaning that the heating system in the CW storage plant must do 20°F more lift, or work, because of the overlapping lift envelopes to move the same amount of heat from the CHW loop to the HW loop. The nominal efficiency differences are small between a screw chiller handling the full lift to generate HW in the HW storage plant and a screw chiller handling less lift but in an overlapping lift configuration with a more efficient centrifugal chiller in the CW storage plant. However, over the course of the year the single lift heat recovery screw approach proves to be more efficient due to variations of part load behavior for the different compressor types (Figure 39 and Figure 40). Note that these COP differences are also influenced by the conventions chosen by the project team: all chiller energy including energy used by cooling chillers is considered heating energy when heat from the chiller condensers is not rejected through the towers.

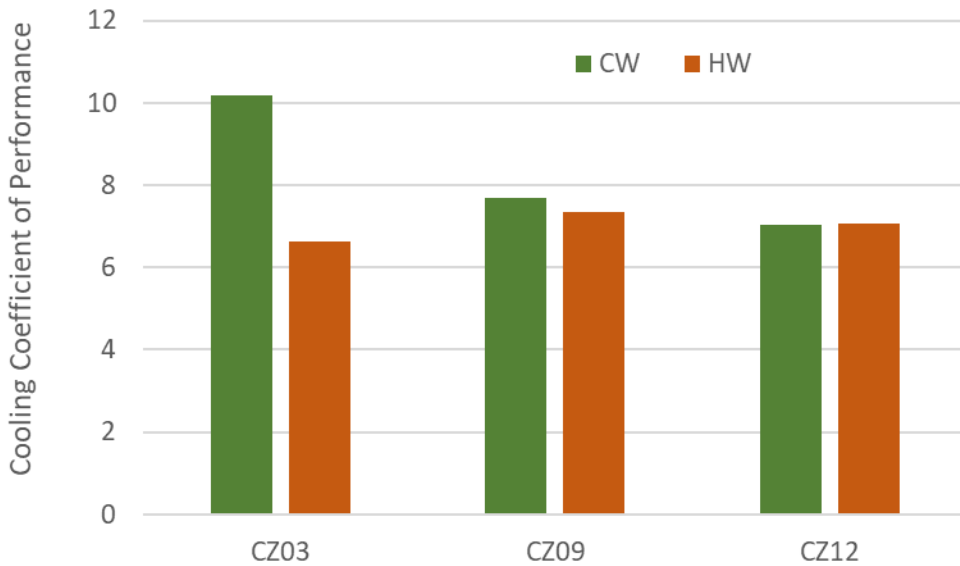

Figure 43: Cooling COPc condenser water vs. hot water – intermediate tank with 0-ton baseload.

Relative cooling performance between the HW and CW plant configurations is highly climate dependent (Figure 43). The CW storage COPc is substantially higher in CZ 03 than in CZ 09. This mild coastal climate with low cooling loads relative to the heating loads keeps the storage tank in a constant state of flux, where the tank is discharged quickly each morning to meet heating loads, then slowly recharged throughout the day with the limited heat recovery from the cooling loads. Warmer climates with higher cooling loads quickly maximize heat recovery potential, and the tank remains fully charged during more hours of the year. The intermediate CW storage tank in CZ 03 spends 9 percent of the year at full charge, while the same scenario in CZ 09 spends 20 percent of the year at full charge. The COPc versus COPh conventions chosen for this analysis provide a larger cooling efficiency credit to CW storage plants in CZ 03 at a system level, since compressor energy is only categorized as cooling energy when heat is rejected through the towers. Chiller energy is considered heating energy otherwise and included in the heating COPh calculation, allowing differences in annual tower loop run hours to inflate the COPc regardless of part load efficiency differences between CW models in different climates.

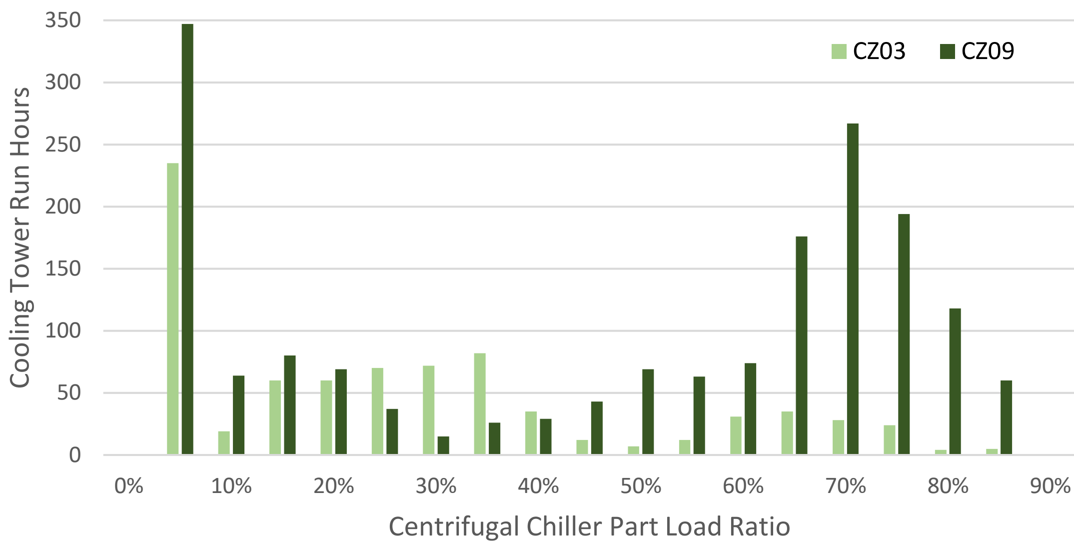

Figure 44 Hours per year with TES tank fully charged - condenser water model

Figure 44 presents the total tower loop run hours for CZ 03 and CZ 09 binned by chiller part load ratio. Even though centrifugal chillers are more efficient at higher part loads (Figure 39), there are more cooling tower run hours in CZ 09 at almost every PLR bin when compared to CZ 03. So, while the increased hours at high part loads may improve instantaneous cooling efficiencies in CZ 09, it is still more efficient at a system level to run the towers as little as possible based on the COPc conventions.

In CZs 09 and 12, the cooling loads are much higher than those of climate zone 03. This leads to a higher heat recovery fraction because these zones often meet the heating loads with entirely recovered heat and avoid running the AWHPs altogether. High heat recovery percentages negatively impact COPc by these conventions because there are more hours where heat must be rejected through the cooling tower loop.

Contrary to the CW model where cooling towers only run at full tank charge conditions, the COPc convention has less of an impact on the HW model which runs the cooling towers whenever the centrifugal chillers are running, up to 95 percent of the annual operating hours. For the HW model, this is the case across each region, which is why there is less delta between the COPc of each CZ for the HW model. Both CW and HW plant types leverage centrifugal chillers when the tank is full and heat is rejected through the towers, leading to similar cooling performance between the plants in climates with high heat recovery fractions. Therefore, in climate zones with higher cooling loads relative to heating loads, the COPc advantages of the CW storage plant are less pronounced than they are in mild climates.

Cooling Baseloads

Heat recovery equipment has generally been considered most desirable for buildings with high coincident heating and cooling loads. TIER plants allow non-coincident heat recovery through the storage tank, allowing a building to reap the benefits of heat recovery even when coincident loads are small or non-existent. Even so, base cooling or heating loads which do not vary much throughout the year provide a consistent source and sink for heat recovery, where cooling baseloads may take the form of IDF/MDF rooms or kitchen condensing units and heating baseloads may take the form of domestic HW preheat. For the modeled commercial building application, there is negligible domestic HW load, so plant performance was assessed as a function of cooling baseload to represent buildings with a steady source of heat recovery.

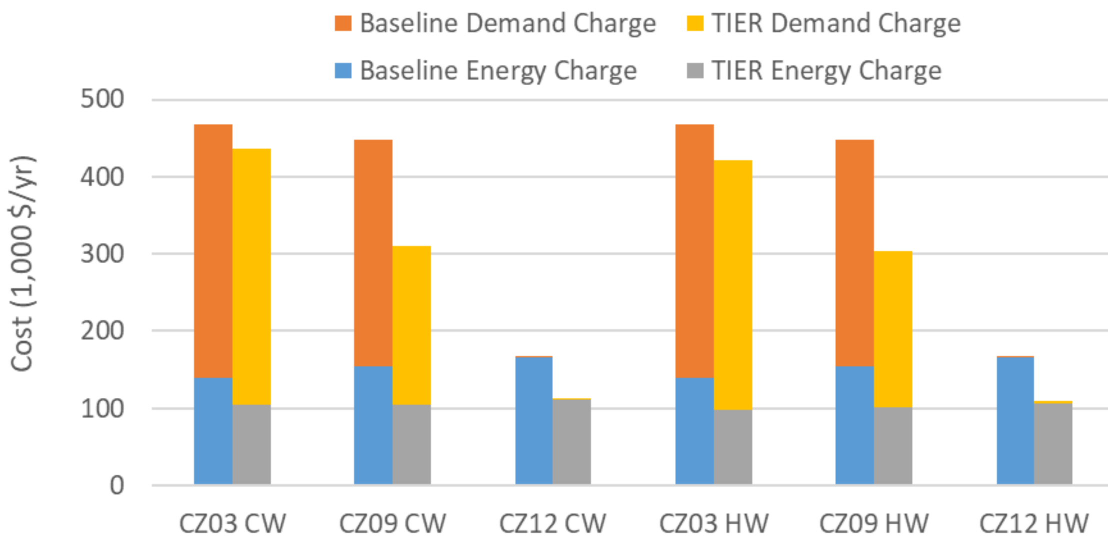

Figure 45: Energy cost breakdown by region and storage media 0-ton baseload.

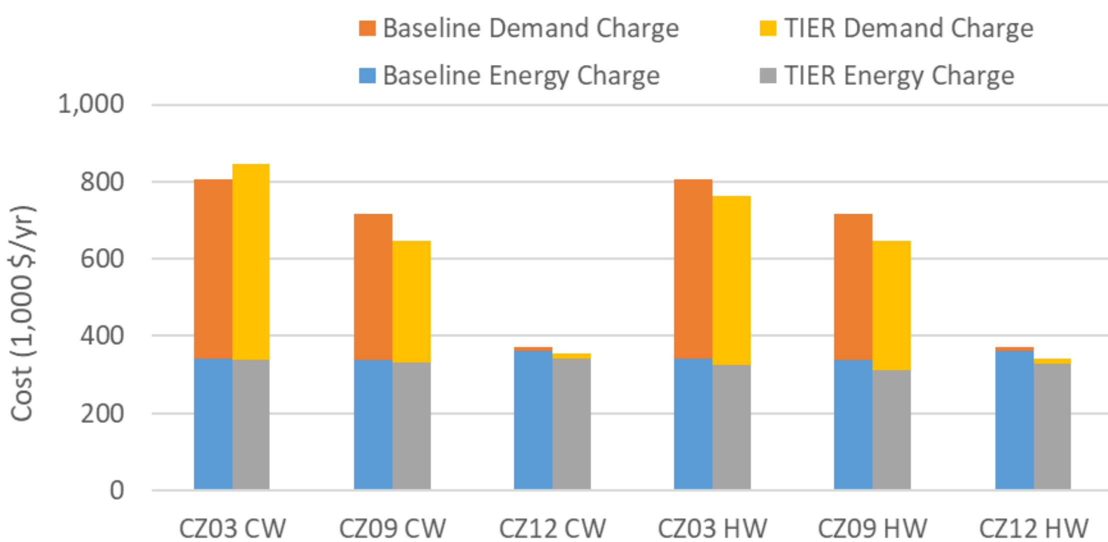

Figure 46: Energy cost breakdown by region and storage media 50-ton baseload.

With all other variables held constant, Figure 45 and Figure 46 show the variations in baseline energy costs and TIER plant energy costs as a function of CZ and storage media. Note that all data presented in these figures are for intermediately sized storage tanks. Energy cost savings over the baseline are less pronounced with additional baseloads, with the 50-ton runs returning only 10 percent energy cost savings on average versus 30 percent in the runs with no baseload. In the case of the CW plant in CZ 03, the TIER plant performs worse than the baseline at high cooling baseloads. While the 0-ton runs show significant energy charge savings between the baseline and TIER, the 50-ton runs show very little energy charge reduction by comparison. This means that the TIER plant configurations are only slightly more energy efficient than their baselines when a 50-ton baseload is applied since overall cooling loads begin to eclipse the heating loads.

As previously discussed in the Thermal Energy Storage Media section, TIER plants incur cooling efficiency penalties at higher cooling loads, since there is only so much heat that can be recovered until it must be rejected to the cooling towers. CZs 09 and 12 already have high cooling loads relative to CZ 03 and must rely on cooling towers during much more of the year, leading these two climates to have lower cooling COPs by comparison. Cooling efficiency is further reduced when additional baseloads are applied because the model building becomes cooling load dominated across all three climates, causing the cooling towers to run during more hours of the year. As cooling loads rise and heating loads stay constant, the TIER plant begins behaving much more like the conventional water-cooled CHW plant baseline. There is still the benefit of near 100 percent heat recovery, leading to substantial emissions savings for all proposed cases compared to the baseline design where boilers are required for heating, but the energy cost savings are more modest due to the low relative cost of natural gas in the baseline.

In general, the hot water storage plant tends to perform better than the condenser water storage plant at higher cooling baseloads due to relative sizing of the dedicated cooling chillers and differences in chiller lift. Where the CW plant uses a single large centrifugal chiller at high lift (80°F condenser water), the HW plant uses 2 smaller centrifugal chillers at lower lift (60°F condenser water). At the 50-ton baseload, smaller chillers have more favorable efficiencies since they are closer to their best efficiency point, while the large centrifugal chiller in the condenser water plant will operate much closer to its cycling point at 20 percent part load. When combined with the fact that the condenser water model runs at higher lifts for most of the year to charge the tank, the plant level cooling efficiency across all climate zones becomes better in a hot water storage plant configuration (Figure 48).

Note that the condenser water model in CZ 03 incurs more demand charges than the baseline, and thus a higher total operating cost (Figure 46). The efficiency penalties of running chillers at high lift and low part loads are magnified in regions with strict time of use charges since they can influence the demand peaks. The baseline water cooled chiller plant does not see as many peaks, and thus lower demand charges, because the centrifugal chillers are running at a lower lift and more optimal part load ratios (Figure 39 and Figure 40).

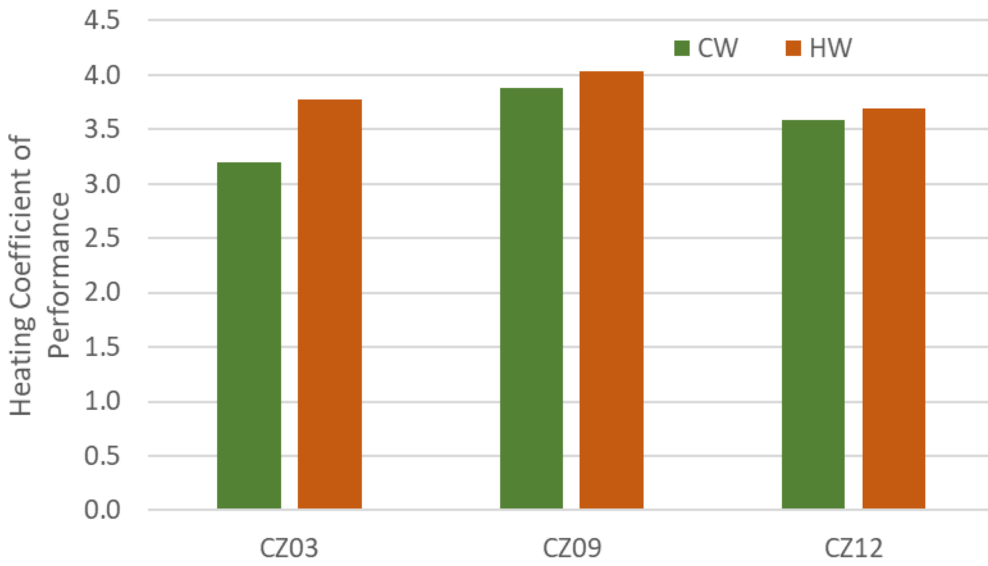

Figure 47: Heating COPh condenser water vs. hot water – intermediate tank with 50-ton baseload.

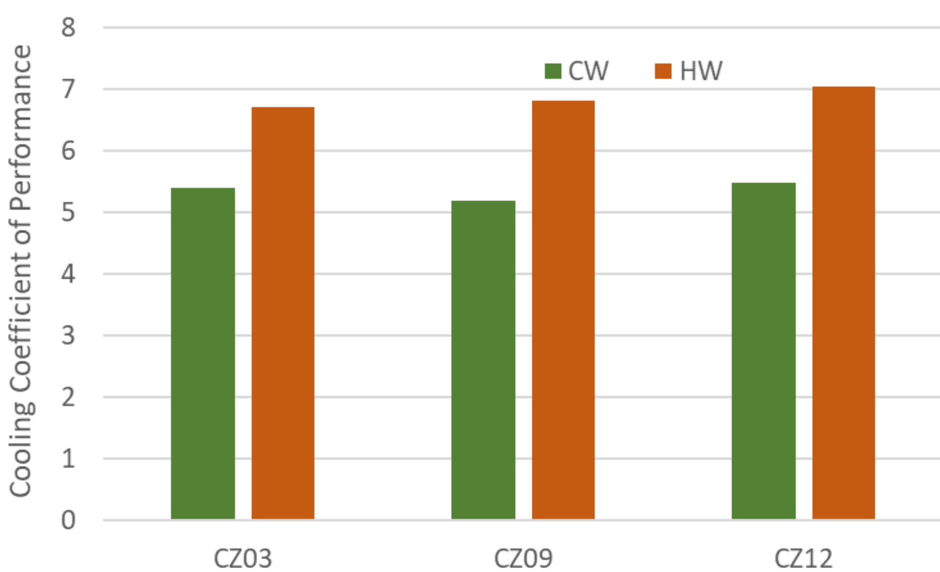

Figure 48: Cooling COPc condenser water vs. hot water – intermediate tank with 50-ton baseload.

In contrast to the results for a 0-ton baseload where the two plant configurations offer a tradeoff between cooling and heating COP, the HW storage plant is clearly more efficient for buildings with higher cooling baseloads (Figure 47 and Figure 48). These efficiency improvements are manifested as operating cost savings in CZs 03 and 12 but not in CZ 09. The high baseload HW storage plant in CZ 09 was more efficient than the CW storage plant under the same conditions, but it incurred higher demand charges in the 50-ton baseload runs. Coincidentally, the increase in demand charges was roughly enough to offset the increased efficiency, leading both plants to perform similarly on an energy cost basis in these scenarios.

Thermal Energy Storage Tank Sizing

The last significant variable in this parametric analysis is the ratio of thermal energy storage size to AWHP capacity. Three different sizing configurations were tested for each CZ, baseload, and storage media type. The intermediate tank size and associated AWHP plant size heat pump sizing were determined by first calculating the amount of heat that would be necessary on the highest heating load day of the year observed in the energy model to supplement the available heat recovery from the CHW loop and still replenish the tank to full charge prior to the start of the next day's morning warm-up period. The AWHP plant was sized to equal the total supplemental heat required on the design day divided by 24 hours.

The tank was then sized to exceed the highest cumulative net heat deficit observed during the highest heating load day. The largest tank size and smaller associated AWHP plant size (1/2x AWHP) were determined by halving the AWHP capacity of the intermediate tank size design and upsizing the tank as needed to never fully drain given the smaller supplemental heat capacity and slower tank recharge rate. In the 1/2x AWHP model, the tank may not fully recharge prior to the start of each day's morning warm-up period but would never fully discharge across the highest heating load multi-day periods during the year. The smallest tank size and larger associated AWHP plant size (2x AWHP) were determined by doubling the AWHP capacity of the intermediate tank size design and reducing the tank size to the extent possible while still exceeding the highest cumulative net heat deficit observed during the highest heating load day.

Variations in storage tank volume and air-to-water heat pump size had little to no effect on energy cost savings in CZs 09 and 12 (Figure 49 and Figure 50). These figures are normalized to the plant energy usage at the intermediate tank size. Plant level energy use, demand charges, and AWHP loads remained relatively consistent at different sizes, as well as heating and cooling COPs. This makes sense because those climate zones are cooling dominated, meaning that they achieve near 100 percent annualized heat recovery and spend many hours during the year with the tank fully charged and the cooling towers running. The size of the tank and AWHP plant becomes largely inconsequential when the majority of the heating load is served with recovered heat from the chillers, as long as there is enough storage capacity to meet peak heating loads when discharging.

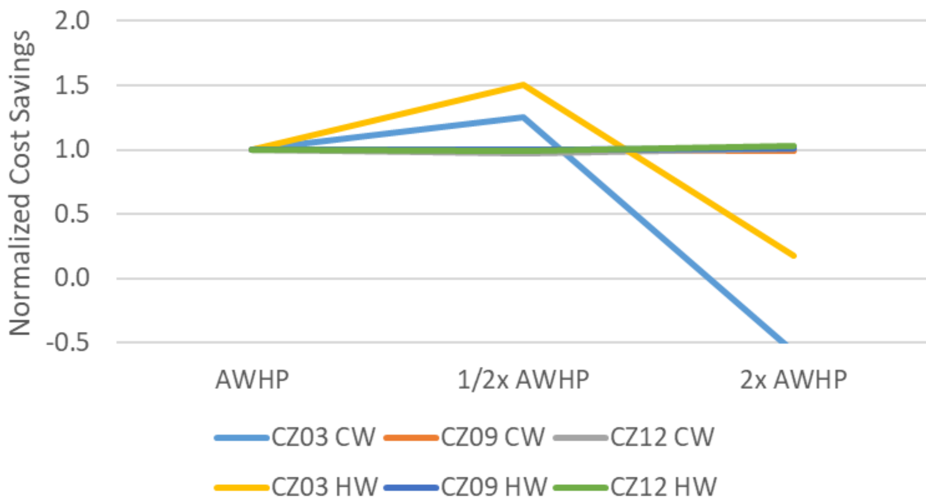

Figure 49: Normalized cost savings at varying TES volumes and AWHP capacities 0-ton baseload.

Once again, CZ 03 was the exception showing marked improvement with a larger tank. Energy use and plant efficiencies in this region remain relatively unchanged between tank sizes, meaning that lower demand charges are driving the cost savings. The heat recovery fraction is much lower at around 66 percent because this climate is not cooling dominated, meaning that the remainder of the heating load must come from the AWHPs. A larger tank is advantageous in this scenario since it gives more runway for cooling chillers to recharge the tank later in the day before requiring supplemental charging. Not only do the AWHPs run less often, but they are also smaller and incur a lower demand charge penalty than designs with larger less efficient AWHP capacities.

Compared to other climates, the demand charges in CZ 03 are highly sensitive to the AWHP size, the frequency at which they run, and the times of day at which they run. These compounding factors are responsible for lowering the demand charges in both HW storage and CW storage configurations when AWHP capacity is decreased. Conversely, demand charges are increased significantly when the AWHP capacity is doubled and the tank is smaller, because the heat pumps need to run more often earlier in the day while the heat recovery chillers are running to prevent a large charge deficit. Large AWHPs running simultaneously with heat recovery chillers set the demand charge for the winter months.

The effects of varying tank and AWHP size are similar when cooling baseloads are applied. Higher cooling baseloads generally reduce the AWHP loads when compared to an equivalent 0-ton baseload scenario, but there are still significant differences in the AWHP loads of each tank size with a baseload. A larger tank results in around 40 percent less AWHP load compared to the intermediate tank, while the smaller tank size increases the AWHP load by about 150 percent. Decreasing the load on the AWHPs is desirable for avoiding time of use charges, leading to lower energy costs overall.

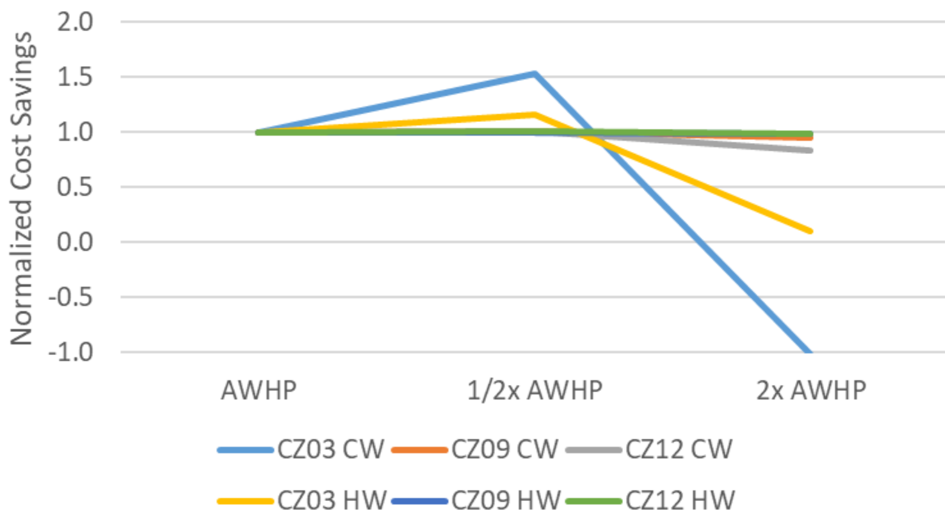

Figure 50: Normalized cost savings at varying TES volumes and AWHP capacities 50-ton baseload.

Life Cycle Cost Study

The Project Team conducted a LCC analysis across the 36 parametric runs defined in the Summary of Parametric Analysis section. The physical layout from the real study site was used as the basis for the LCC analysis, with equipment resized for different plant approaches and applications. Mechanical and electrical contractors familiar with the costs from the real building provided construction cost estimating support. In this LCC analysis, performance of each TIER plant configuration is only evaluated against the performance of other TIER plants. The expectation is that a conventional chiller/boiler plant would have far lower LCC compared to a TIER plant because of the significant first cost premium of a TIER plant, and that TIER would have lower LCC compared to an AWHP plant because of the lower first and operating costs.

Summary of TIER Plant Equipment Capacities

Each parametric run's TIER plant design and mechanical equipment capacities are summarized in Table 5 and Table 6. Instead of keeping the building design constant and resizing the plant entirely for each climate zone, the total design cooling capacity of the plant (including HRCs operating in cooling mode) was held constant by scaling down the building and loads as needed while the plant's design heating capacity was adjusted for each climate zone. Normalizing all parametric plant designs for design cooling capacity results in equal total chiller tonnage in every run and facilitates better comparisons across climate zones since all evaluated climate zones are cooling dominated for equipment sizing.

Contractors helped to gather equipment and labor costs for all equipment. Labor costs include mechanical and electrical labor rate multipliers for the representative cities in each climate zone. Where equipment was not priced due to the quantity of unique sizes needed, cost regression curves were developed by equipment type to interpolate where needed.

Table 5: Condenser water storage plant mechanical equipment.

Run ID	Centrifugal Cooling Chiller Size [tons]	Screw HRC Size [tons]	AWHP Size [tons]	TES Tank Size [feet diameter | gallons]	CHW Loop Pump Size 85' [gpm]	HW Loop Pump Size 140' [gpm]	CW Loop Pump Size (Chiller Evaporator Barrels) 55' [gpm]	CW Loop Pump Size (Chiller Condenser Barrels)] 90' [gpm]	Tower Water Heat Exchanger Size [gpm]	Cross-Flow CT & Tower Water Pump Size 55' [gpm]
03-CW1	1 @ 380	2 @ 199	2 @ 133	16.75 | 96,600	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
03-CW2	1 @ 380	2 @ 199	2 @ 75	21.75 | 162,900	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
03-CW3	1 @ 380	2 @ 199	2 @ 200	12.75 | 56,000	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
03-CW501	1 @ 380	2 @ 199	2 @ 133	14.25 | 69,900	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
03-CW502	1 @ 380	2 @ 199	2 @ 67	18.25 | 114,700	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
03-CW503	1 @ 380	2 @ 199	2 @ 183	11.50 | 45,500	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
09-CW1	1 @ 525	2 @ 127	2 @ 75	12.25 | 51,700	2 @ 380	2 @ 135	2 @ 150	2 @ 720	2 @ 720	2 @ 720
09-CW2	1 @ 525	2 @ 127	1 @ 75	18.25 | 114,700	2 @ 380	2 @ 135	2 @ 150	2 @ 720	2 @ 720	2 @ 720
09-CW3	1 @ 525	2 @ 127	2 @ 100	9.25 | 29,500	2 @ 380	2 @ 135	2 @ 150	2 @ 720	2 @ 720	2 @ 720
09-CW501	1 @ 525	2 @ 127	1 @ 117	10.50 | 38,000	2 @ 380	2 @ 135	2 @ 150	2 @ 720	2 @ 720	2 @ 720
09-CW502	1 @ 525	2 @ 127	1 @ 67	13.00 | 58,200	2 @ 380	2 @ 135	2 @ 150	2 @ 720	2 @ 720	2 @ 720
09-CW503	1 @ 525	2 @ 127	2 @ 92	7.00 | 16,900	2 @ 380	2 @ 135	2 @ 150	2 @ 720	2 @ 720	2 @ 720
12-CW1	1 @ 380	2 @ 199	2 @ 150	16.75 | 96,600	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
12-CW2	1 @ 380	2 @ 199	2 @ 83	24.25 | 202,500	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
12-CW3	1 @ 380	2 @ 199	2 @ 150	12.50 | 53,800	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720

Run ID	Centri-fugal Cooling Chiller Size [tons]	Screw HRC Size [tons]	AWHP Size [tons]	TES Tank Size [feet diameter | gallons]	CHW Loop Pump Size 85' [gpm]	HW Loop Pump Size 140' [gpm]	CW Loop Pump Size (Chiller Evap-orator Barrels) 55' [gpm]	CW Loop Pump Size (Chiller Condenser Barrels)] 90' [gpm]	Tower Water Heat Exch-anger Size [gpm]	Cross-Flow CT & Tower Water Pump Size 55' [gpm]
12-CW501	1 @ 380	2 @ 199	2 @ 150	14.50 | 72,400	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
12-CW502	1 @ 380	2 @ 199	2 @ 75	18.50 | 117,800	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720
12-CW503	1 @ 380	2 @ 199	2 @ 150	10.25 | 36,200	2 @ 380	2 @ 210	2 @ 230	2 @ 720	2 @ 720	2 @ 720

Table 6: Hot water storage plant configurations mechanical equipment.

Run ID	Centri-fugal Cool-ing Chil-ler Size [tons]	Scr-ew HRC Size [tons]	AWHP Size [tons]	TES Tank Size [feet diameter | gallons]	CHW Loop Pump Size 85' [gpm]	HW Loop Pump Size 140' [gpm]	HRC Evap-orator Pump Size 20' [gpm]	HRC Cond-end-ser Pump Size 20' [gpm]	Heat Pump Water Loop Pump Size 10' [gpm]	TES Water Loop Pump Size 35' [gpm]	Tower Water Heat Exch-anger Size [gpm]	Cross-Flow CT & Tower Water Pump Size 55' [gpm]
03-HW1	2 @ 270	1 @ 240	2 @ 140	16.25 | 90,900	2 @ 380	2 @ 175	1 @ 575	1 @ 635	2 @ 150	2 @ 120	1 @ 450	2 @ 720
03-HW2	2 @ 270	1 @ 240	1 @ 140	21.75 | 162,900	2 @ 380	2 @ 175	1 @ 575	1 @ 635	2 @ 75	2 @ 120	1 @ 450	2 @ 720
03-HW3	2 @ 270	1 @ 240	3 @ 140	11.25 | 43,600	2 @ 380	2 @ 175	1 @ 575	1 @ 635	2 @ 220	2 @ 150	1 @ 450	2 @ 720
03-HW501	2 @ 270	1 @ 240	2 @ 105	18.50 | 117,800	2 @ 380	2 @ 175	1 @ 575	1 @ 635	2 @ 110	2 @ 120	1 @ 450	2 @ 720
03-HW502	2 @ 270	1 @ 240	1 @ 105	22.75 | 178,200	2 @ 380	2 @ 175	1 @ 575	1 @ 635	2 @ 55	2 @ 120	1 @ 450	2 @ 720
03-HW503	2 @ 270	1 @ 240	3 @ 105	14.00 | 67,500	2 @ 380	2 @ 175	1 @ 575	1 @ 635	2 @ 165	2 @ 120	1 @ 450	2 @ 720
09-HW1	2 @ 280	1 @ 225	1 @ 140	14.25 | 69,900	2 @ 380	2 @ 110	1 @ 540	1 @ 595	2 @ 75	2 @ 110	1 @ 420	2 @ 720
09-HW2	2 @ 280	1 @ 225	1 @ 70	21.00 | 151,800	2 @ 380	2 @ 110	1 @ 540	1 @ 595	2 @ 35	2 @ 110	1 @ 420	2 @ 720

Run ID	Centri-fugal Cool-ing Chil-ler Size [tons]	Scr-ew HRC Size [tons]	AWHP Size [tons]	TES Tank Size [feet diameter | gallons]	CHW Loop Pump Size 85' [gpm]	HW Loop Pump Size 140' [gpm]	HRC Evap-orator Pump Size 20' [gpm]	HRC Cond-end-ser Pump Size 20' [gpm]	Heat Pump Water Loop Pump Size 10' [gpm]	TES Water Loop Pump Size 35' [gpm]	Tower Water Heat Exch-anger Size [gpm]	Cross-Flow CT & Tower Water Pump Size 55' [gpm]
09-HW3	2 @ 280	1 @ 225	2 @ 70	10.00 | 34,400	2 @ 380	2 @ 110	1 @ 540	1 @ 595	2 @ 110	2 @ 110	1 @ 420	2 @ 720
09-HW501	2 @ 280	1 @ 225	1 @ 140	13.25 | 60,400	2 @ 380	2 @ 110	1 @ 540	1 @ 595	2 @ 75	2 @ 110	1 @ 420	2 @ 720
09-HW502	2 @ 280	1 @ 225	1 @ 70	18.50 | 117,800	2 @ 380	2 @ 110	1 @ 540	1 @ 595	2 @ 35	2 @ 110	1 @ 420	2 @ 720
09-HW503	2 @ 280	1 @ 225	2 @ 70	9.50 | 31,100	2 @ 380	2 @ 110	1 @ 540	1 @ 595	2 @ 110	2 @ 110	1 @ 420	2 @ 720
12-HW1	2 @ 290	1 @ 200	2 @ 140	17.75 | 108,500	2 @ 380	2 @ 175	1 @ 480	1 @ 530	2 @ 150	2 @ 100	1 @ 375	2 @ 720
12-HW2	2 @ 290	1 @ 200	1 @ 140	24.75 | 210,900	2 @ 380	2 @ 175	1 @ 480	1 @ 530	2 @ 75	2 @ 100	1 @ 375	2 @ 720
12-HW3	2 @ 290	1 @ 200	3 @ 140	12.50 | 53,800	2 @ 380	2 @ 175	1 @ 480	1 @ 530	2 @ 220	2 @ 140	1 @ 375	2 @ 720
12-HW501	2 @ 290	1 @ 200	2 @ 140	17.25 | 102,400	2 @ 380	2 @ 175	1 @ 480	1 @ 530	2 @ 150	2 @ 100	1 @ 375	2 @ 720
12-HW502	2 @ 290	1 @ 200	1 @ 140	22.00 | 166,600	2 @ 380	2 @ 175	1 @ 480	1 @ 530	2 @ 75	2 @ 100	1 @ 375	2 @ 720
12-HW503	2 @ 290	1 @ 200	3 @ 140	12.00 | 49,600	2 @ 380	2 @ 175	1 @ 480	1 @ 530	2 @ 220	2 @ 140	1 @ 375	2 @ 720

Electrical infrastructure matched the electrical system design of the baseline plant study site when the building was operating as a TIER plant. Only the capacity of electrical equipment impacted by differences in mechanical equipment connections across the 36 parametric runs are noted in Table 7 and Table 8.

Table 7: Condenser water storage plant impacted electrical equipment.

Run ID	20A Connect-ion Count	30A Connect-ion Count	40A Connect-ion Count	200A Connect-ion Count	400A Connect-ion Count	600A Connect-ion Count	Distribution Panel-board Size [A]	Main Switch-board Size [A]	Main Switch-board Trans-former Size [kVA]
03-CW1	2	4	2	0	5	0	2,000	4,000	3,000
03-CW2	2	4	2	2	3	0	2,000	4,000	3,000
03-CW3	2	4	2	0	5	0	2,500	5,000	4,000
03-CW501	2	4	2	0	5	0	2,000	4,000	3,000
03-CW502	2	4	2	2	3	0	2,000	4,000	3,000
03-CW503	2	4	2	0	5	0	2,500	5,000	4,000
09-CW1	4	2	2	2	2	1	2,000	4,000	3,000
09-CW2	4	2	2	1	2	1	2,000	4,000	3,000
09-CW3	4	2	2	2	2	1	2,000	4,000	3,000
09-CW501	4	2	2	0	3	1	2,000	4,000	3,000
09-CW502	4	2	2	1	2	1	1,600	4,000	3,000
09-CW503	4	2	2	2	2	1	2,000	4,000	3,000
12-CW1	2	4	2	0	5	0	2,000	4,000	3,000
12-CW2	2	4	2	2	3	0	2,000	4,000	3,000
12-CW3	2	4	2	0	6	0	2,500	5,000	4,000
12-CW501	2	4	2	0	5	0	2,000	4,000	3,000
12-CW502	2	4	2	2	3	0	2,000	4,000	3,000
12-CW503	2	4	2	0	6	0	2,500	5,000	4,000

Table 8: Hot water storage plant impacted electrical equipment.

Run ID	20A Connect-ion Count	30A Connect-ion Count	40A Connect-ion Count	200A Connect-ion Count	400A Connect-ion Count	600A Connect-ion Count	Distribution Panel-board Size [A]	Main Switch-board Size [A]	Main Switch-board Trans-former Size [kVA]
03-HW1	6	4	0	0	2	3	2,500	5,000	4,000
03-HW2	6	4	0	0	2	2	2,000	4,000	3,000
03-HW3	6	4	0	0	2	4	2,500	5,000	4,000
03-HW501	6	4	0	0	4	1	2,000	4,000	3,000
03-HW502	6	4	0	0	3	1	2,000	4,000	3,000
03-HW503	6	4	0	0	5	1	2,500	5,000	4,000
09-HW1	8	2	0	0	3	1	2,000	4,000	3,000
09-HW2	8	2	0	0	4	0	1,600	4,000	3,000
09-HW3	8	2	0	0	5	0	2,000	4,000	3,000
09-HW501	6	4	0	0	3	1	2,000	4,000	3,000
09-HW502	6	4	0	0	4	0	1,600	4,000	3,000
09-HW503	6	4	0	0	5	0	2,000	4,000	3,000
12-HW1	6	4	0	0	3	2	2,500	5,000	4,000
12-HW2	6	4	0	0	3	1	2,000	4,000	3,000
12-HW3	6	4	0	0	3	3	2,500	5,000	4,000
12-HW501	6	4	0	0	3	2	2,500	5,000	4,000
12-HW502	6	4	0	0	3	1	2,000	4,000	3,000
12-HW503	6	4	0	0	3	3	2,500	5,000	4,000

First Cost Comparison: Thermal Energy Storage Tank Sizing

Larger AWHP plants require more electrical infrastructure, but TES tanks can be smaller. The first cost tradeoffs are highlighted in Figure 51. In CZs 3 and 12, where higher heating loads require larger AWHP plant capacities, the mechanical and electrical first cost savings of cutting AWHP plant capacity in half (1/2x AWHP) more than offsets the incremental first costs of the larger TES tank that is approximately double in volume. Where heating loads are lower overall, like in CZ 9, the first cost savings of a smaller AWHP plant are less impactful and therefore outweighed by the higher tank costs.

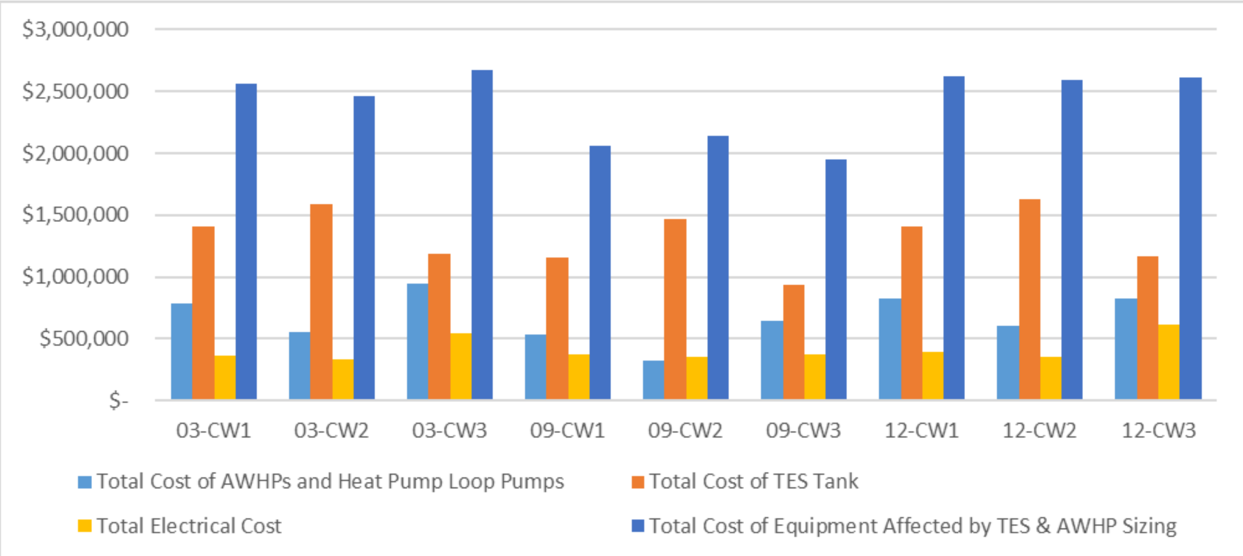

Figure 51: First cost comparison of equipment affected by TES tank and AWHP sizing by climate zone.

First Cost Comparison: Thermal Energy Storage Media

Thermal energy storage media has relatively minimal impact on the TES versus AWHP first cost tradeoffs across climate zones and cooling baseload quantity. As shown in Figure 52, each equipment cost category has similar magnitude and behavior between the CW and HW storage plants.

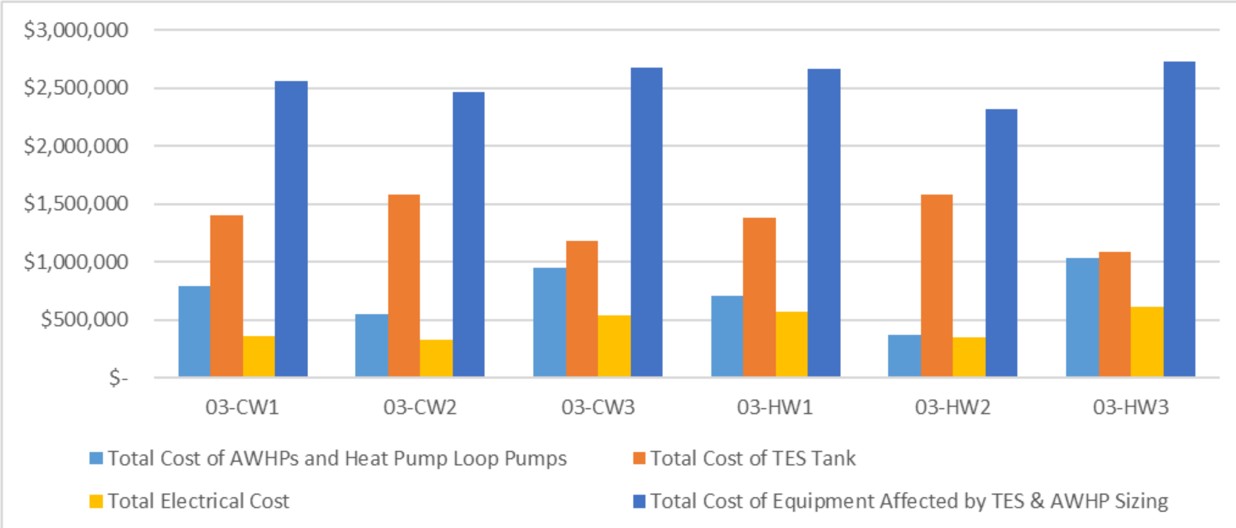

Figure 52: First cost comparison of equipment affected by TES tank and AWHP sizing by storage media.

While the total cost of equipment affected by TES tank and AWHP sizing including electrical infrastructure is marginally higher in HW storage plants compared to CW storage plants, the HW storage plant offers marginal mechanical first cost savings for equipment not affected by TES tank and AWHP sizing. The net impact is general first cost parity between TIER plants with HW vs. CW storage media.

The centrifugal chillers priced as part of this study are generally 25 percent more cost effective per ton compared to screw HRCs. The heat recovery screw chiller in the HW storage plant is sized for only the coincident simultaneous heating and cooling load in the building with the remaining heating capacity met by the AWHPs. On the other hand, the screw HRCs in the CW storage plant are sized to meet the entirety of the building's heating load since the AWHPs are only configured to charge the TES tank and not serve the building HW loop directly. The HW storage plant therefore has a relatively larger centrifugal compressor fraction of total chiller capacity which is more cost effective and results in ~$0.30 per square foot savings.

Another notable first cost difference between the two plant designs is the tower loop heat exchangers. Because the cooling chillers in the CW storage plant are configured to reject heat to both the TES tank and the cooling towers, they must be hydraulically isolated from the cooling tower loop. The cooling chillers in the HW storage plant only reject heat to the cooling towers so the tower loop heat exchanger only needs to be sized for the HRC, which is generally less than a third of the total chiller heat rejection capacity. The heat exchanger savings result in another ~$0.30 per square foot first-cost reduction.

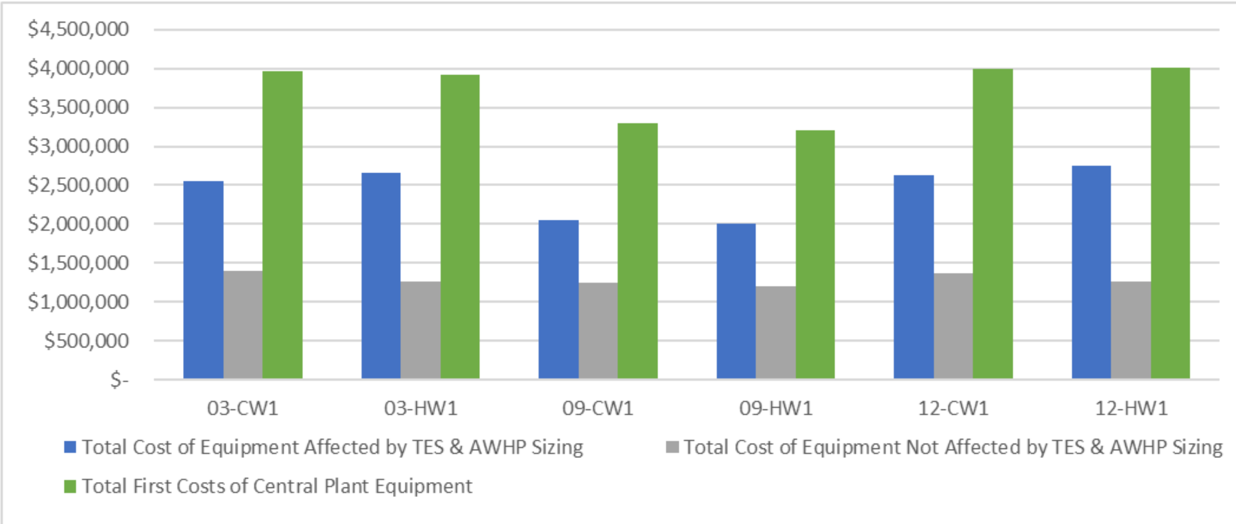

Figure 53: First cost comparison by storage media and climate zone.

Net Present Energy Cost

The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 90.1 scalar ratio method was developed to provide a simplified, objective tool for evaluating the economic viability of energy efficiency improvements relative to a baseline building. The method collapses a full LCC analysis into a single threshold value. It incorporates parameters such as useful life (30 years), variable fuel escalation rates (from National Institute of Standards and Technology/U.S. Department of Energy), federal and state tax rates, inflation (2.38 percent), discount rates (9.34 percent), and loan interest rates (7 percent). The scalar ratio itself is calculated as the incremental first cost of a measure divided by its annual energy cost savings; if the resulting ratio is lower than the scalar ratio, the measure is deemed cost-effective. By extension, the scalar ratio can be applied as a multiplier to incremental annual energy costs and then added to incremental first costs to calculate the incremental net present cost or savings of an alternative design. Assuming 30-year equipment life, a scalar ratio of 16.2 was applied.

Energy cost differences across the 36 parametric analyses were discussed in the Energy Analysis section and the scalar ratio does not change the relative energy cost performance between the different plant design options.

Life Cycle Cost Comparison

Incremental LCCs were calculated by adding incremental first costs to incremental net present energy costs. Figure 54 through Figure 59 show the incremental first and energy cost breakdowns in each climate zone and baseload variation evaluated in the parametric analysis.

In CZ 3, both first costs and net present energy costs align and favor smaller AWHP plants (1/2x AWHP), larger TES tanks, and HW storage over CW storage primarily due to the high demand costs associated with higher AWHP usage given the utility rate structure assumed in CZ 3. HW storage runs with smaller AWHPs show 7 percent to 19 percent savings relative to most of the other plant configurations evaluated.

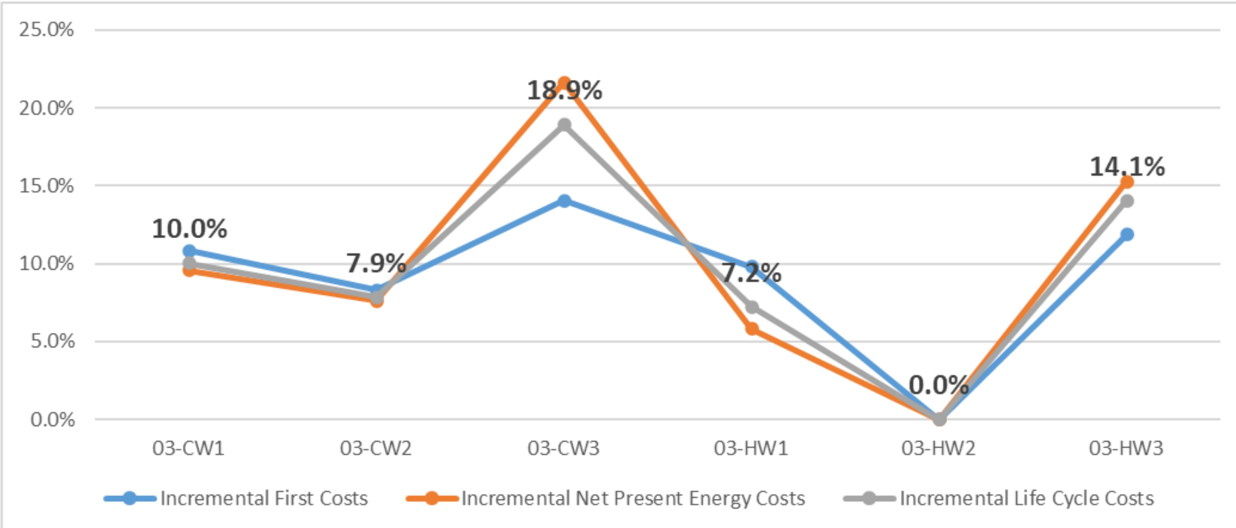

Figure 54: Incremental life cycle cost comparison—Climate Zone 3 with 0-ton baseload.

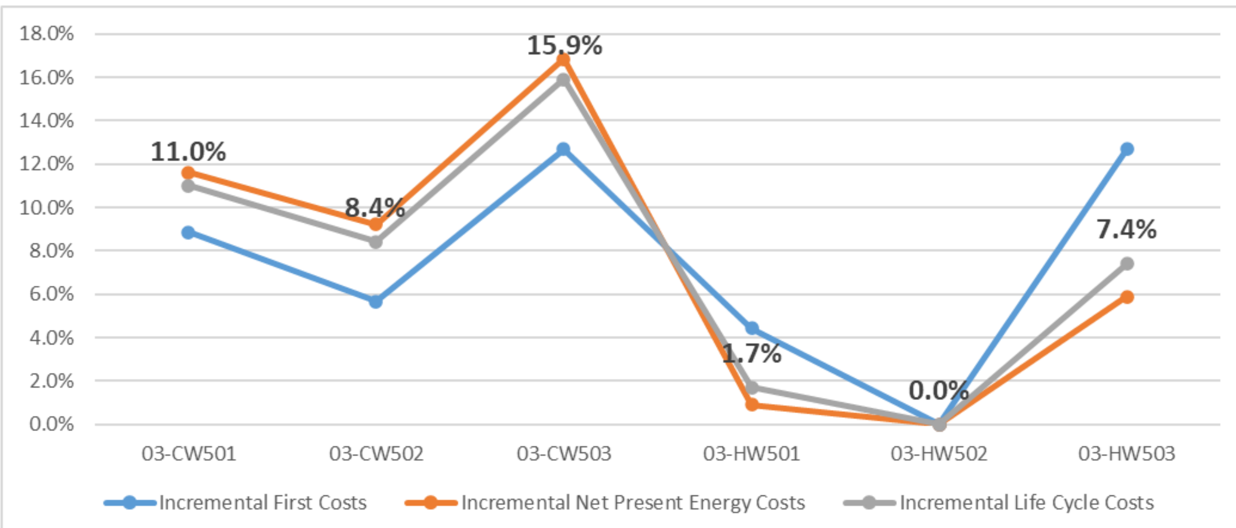

Figure 55: Incremental life cycle cost comparison—Climate Zone 3 with 50-ton baseload.

Energy cost differentials between the different plant configurations are small in CZ 9 given the small heating load, so LCCs are highly first cost driven. Since the AWHP plant is relatively small as well, LCCs are particularly driven by tank costs and therefore favors the 2xAWHP plant design option with small TES tank in a HW storage TIER plant.

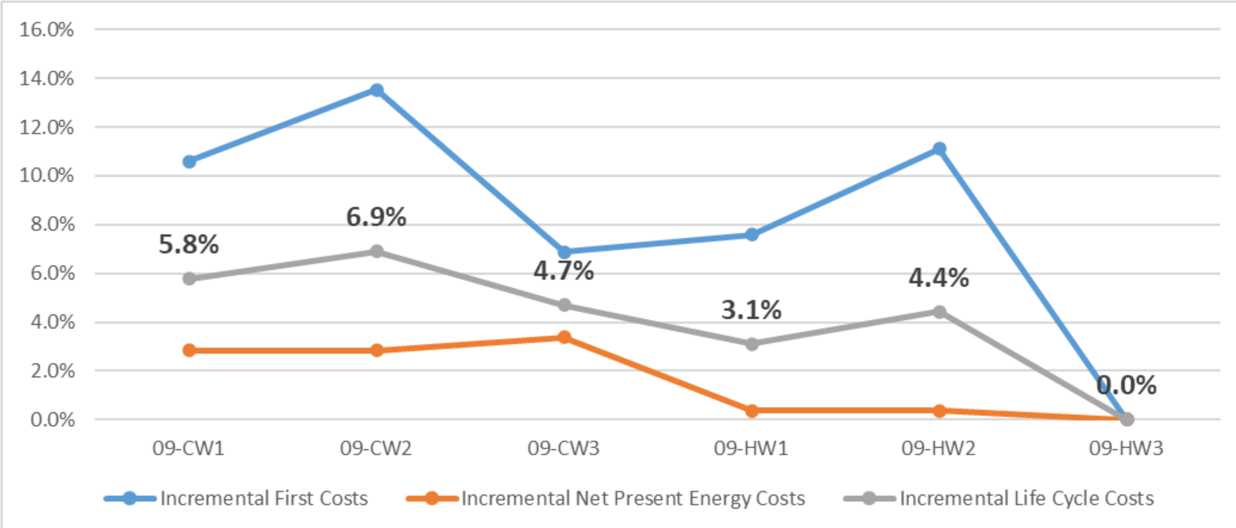

Figure 56: Incremental life cycle cost comparison—Climate Zone 9 with 0-ton baseload.

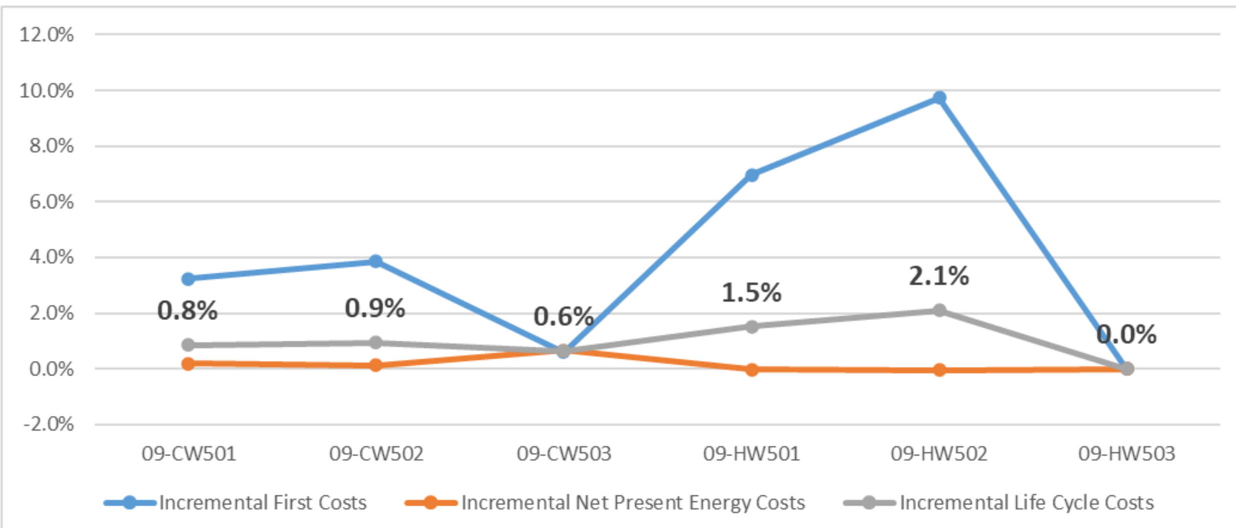

Figure 57: Incremental life cycle cost comparison—Climate Zone 9 with 50-ton baseload.

CZ 12 favors smaller AWHP plants and HW storage for lowest LCCs by a consistent 4 to 8 percent margin compared to all the other plant configurations evaluated. Although lower demand charges muddy the energy cost differences between the different AWHP plant size options, first costs are the main driver of LCCs in CZ 12.

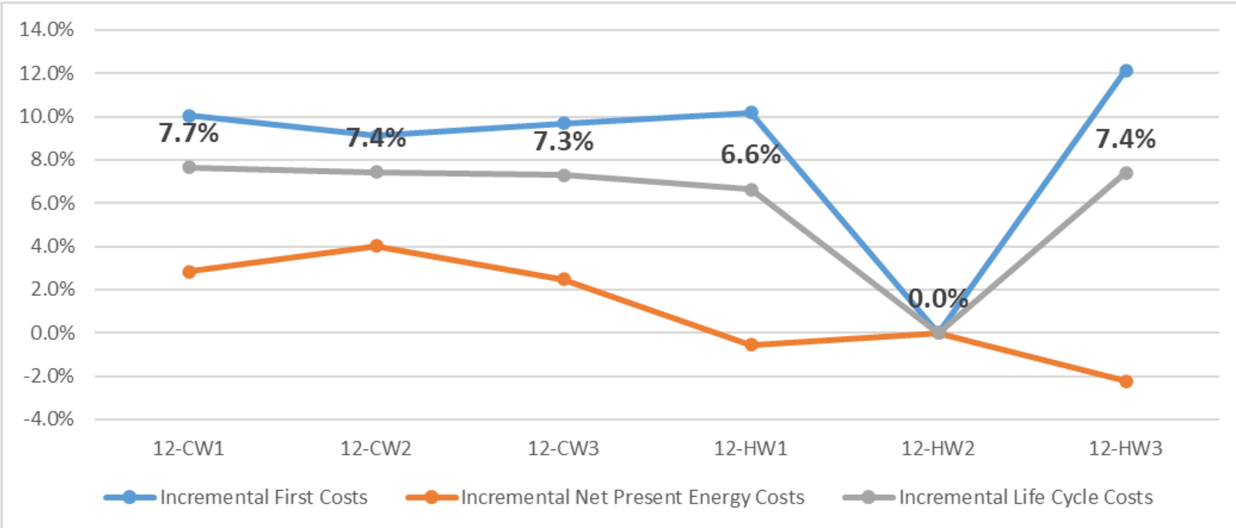

Figure 58: Incremental life cycle cost comparison—Climate Zone 12 with 0-ton baseload.

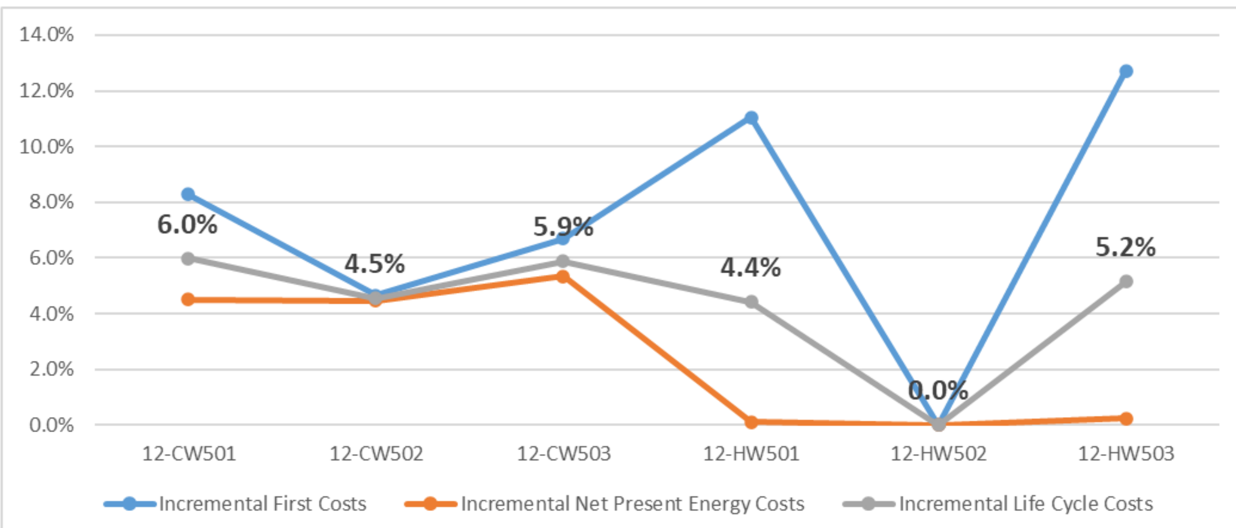

Figure 59: Incremental life cycle cost comparison—Climate Zone 12 with 50-ton baseload.

Study Limitations

Below is a summary of some of the limitations with this study:

- TES tank insulation and losses. The construction cost estimates for both the CW and HW tanks assume 2-inches of insulation but thicker insulation might be warranted for the HW tank given the larger temperature differential to ambient. However, the 2-inch assumption matches the code-minimum insulation thickness for piping based on the applicable fluid temperatures. More study is needed to evaluate appropriate tank insulation for the HW application.
- TES tank envelope losses. Envelope losses are ignored in the energy analysis for simplicity. The CW tank is insulated and stores energy at relatively neutral temperatures so losses may be relatively minor. However, there is a larger temperature differential to ambient for the HW tank

and the envelope losses may have a larger performance impact that is not accounted for in this analysis.

- Utility rate structures. Utility demand charges account for a major portion of the energy costs under some rate tariffs in geographic areas where some nearby locations may instead be served by municipal utilities or community choice aggregators may be available. This analysis co-varied the California climate zone along with an associated utility rate structure for that geographic location. However, given that rate structures may differ significantly within a narrow geographic region, more study is needed evaluate the impact of rate structures in isolation to properly evaluate cost effectiveness.
- Equipment part load performance. The TIER modeling used equipment part load performance curves from the EnergyPlus library that were selected to match the target equipment type and approximate size. However, detailed review of the centrifugal chiller curves found significant part load performance differences, more than a factor of two, between the modeled curves and the submittal data from the real TIER plant, even though the curves were normalized to the actual full load efficiencies. Given that the centrifugal chiller in the CW TIER plant model operated at very low load conditions for most of the year, the curve selection may have a large impact on the overall findings. More study of equipment part load performance curves is needed to understand this potential impact.

Overall Findings and Discussion

The TIER central plant concept provides meaningful first-cost and energy efficiency advantages for large commercial facilities when compared to conventional AWHP-based all-electric plants. However, TIER systems are inherently complex, and current building energy modeling platforms cannot accurately simulate their operation. To address this gap, this study developed two simplified spreadsheet-based simulation tools capable of annual performance modeling for both CW and HW TIER configurations. These tools were used to evaluate energy use and LCCs across three California climates, two building load profiles, and a range of TES media and TES sizing options.

Across most modeled configurations, TIER reduced annual energy costs relative to a chiller/boiler baseline. This is a notable result given the common expectation that all-electric systems carry higher operating costs. However, TIER plants still have significantly higher first costs, and under current economic conditions they are unlikely to achieve LCC parity with chiller/boiler plants solely on financial metrics.

The LCC analysis did not identify a single preferred TES medium or storage size. The HW TIER configuration generally matched or slightly outperformed the CW TIER configuration, but these outcomes appear to be driven by the specific load profiles, climates, and equipment assumptions in the modeled cases. As such, they should not be generalized. Several application-specific and climate-dependent patterns were also observed and can be explained through differences in load profiles, equipment cycling, and TES utilization.

Other factors may exert even stronger influence on the optimal TIER configuration. For example, access to an existing firewater storage tank suitable for CW storage can dramatically improve the economics of the CW TIER approach–an advantage that HW storage plants cannot access and would outweighing the other differences evaluated here. The anticipated reduction in controls system complexity to operate the HW storage plant compared to the CW storage plant lowers construction and commissioning costs and time and expands the applicability of HW storage plants to smaller buildings with fewer dedicated facility operations staff. Building heating and cooling load profiles, equipment sizing and part load efficiency characteristics, and local utility rate structures can all substantially impact energy performance and shift design recommendations. Because plant behavior is highly dynamic, generalized guidance is difficult to establish; however, the simplified modeling tools developed through this project now enable detailed, project-specific evaluation.

COP metrics for systems with significant heat recovery must also be interpreted carefully, as different conventions can lead to misleading comparisons without a consistent basis.

Finally, the study underscores the need for improved characterization of equipment part load performance. Many of the available performance curves—particularly for high-lift HRCs and AWHPs—do not adequately represent real-world behavior, contributing to discrepancies between modeled and measured plant performance. Expanded performance curve libraries for emerging equipment types will be critical for improving future TIER analyses.

Design Guide

The project team developed a design guide that addresses key design issues and concepts unique to all-electric central plants. The guide builds upon research and findings from this project on TIER plant applications, as well as lessons learned and design considerations for all-electric plants in general based on interviews with designers, commissioning providers, and building owners. Major sections in the design guide include:

- All-electric central plant equipment, including electric boilers, air-to-water heat pumps and heat recovery chillers. This section provides overviews of and discussions of challenges for each type of equipment.
- New design considerations. This section of the design guide discusses paradigm shifts – design considerations that are new with all-electric central plants. New challenges include codes and regulations, the latest restrictions on refrigerants, evolving equipment product lines, and heat recovery.
- All-electric plant approaches. Different plant approaches are described in this section, including a basic AWHP plant; plants with heat recovery, storage, and trim heat sources; and hybrid/partial electrification solutions. Potential applications for the TIER plant and generalized guidance are presented based on results from parametric energy analyses and LCC studies. The design guide includes links to simplified spreadsheet modeling tools for two TIER plant configurations to support the exploration and analysis of future TIER applications.
- Case studies, including both a conventional AWHP plant and a TIER plant.

The design guide is primarily targeted toward engineers and designers but will be valuable to a broad audience that also includes contractors, building owners, policy makers, and equipment manufacturers. The All-Electric Central Plant Design Guide is included as a supplemental document to this report.

Market Transformation

It is well known that the state of California is seeking to rapidly decarbonize its built environment. For many years, HVAC in the medium-to-large commercial building segment has provided a unique challenge because (apart from carbon emissions and pollution) the most popular incumbent technology, the natural gas boiler, is an excellent solution. Natural gas boilers are space efficient, energy dense, and relatively affordable for building owners. Boilers have been in place for many decades and there is a high degree of workforce knowledge around how to install and operate the equipment. As noted elsewhere, many all-electric replacement options are very expensive, take up a lot of space, and are challenging to operate effectively by the current workforce. While there remain many barriers that need to be dealt with before they can be widely deployed in the market, TIER systems address some of the challenges outlined.

TIER presents several significant benefits over other all-electric options. TES reduces the need for AWHP equipment, which saves space and lowers upfront costs. Low lift chillers provide energy efficiency. Waste heat storage instead of rejection saves water. These advantages over other all-electric options have been documented in the Performance Review of TIER Plant section of this report. The purpose of this discussion is to highlight the barriers to adoption and strategies that would be effective in dealing with them.

Market transformation involves the deliberate, long-term strategy of intervening in a market to force a superior disruptive technology to replace an inferior incumbent technology. It is not a foregone conclusion that a superior technology will automatically dominate the market if there are broader market factors delaying or preventing its adoption. There are numerous factors that can accelerate or hold back a given technology from being adopted, many of which have nothing to do with technological prowess.

Market transformation begins with an assessment of the market and technology landscape and identification of all barriers to adoption of the measure. Once barriers are understood, intervention strategies can be crafted and executed. Interventions can come from a variety of market actors, including both private and public sectors. For example, an intervention could look like an incentive program, federal tax credit, or a manufacturer's business plan for making their product more appealing to the market.

Market transformation involves a continuous re-assessment of the technology's current status in the market, since the economy is highly dynamic. As early barriers are addressed, new problems will arise, requiring incremental intervention strategies. Hopefully, as time goes on, the market adoption rate steadily increases, and the number and magnitude of the barriers structurally declines. Eventually, once sufficient awareness and adoption have been achieved, the technology may become a candidate for a prescriptive code requirement or other regulatory lever that could lock in its status in the market as the standard of care. A premature code measure could backfire because it would eliminate the possibility of other forms of market development (e.g., incentive programs) while still likely suffering from poor compliance rates due to insufficient workforce education.

At this stage, TIER is a compelling but still relatively unfamiliar option to the HVAC industry. Awareness and appreciation for the benefits of TES is catching on gradually, but a lot of work is still needed. The HVAC industry is conservative, and market actors tend to be very cautious when trying new measures. This stems from the fact that HVAC is a comfort industry first and foremost. Occupant comfort is the top priority of designers and installers, with energy efficiency and carbon emissions considerations lower down on the list of priorities. If any novelty or complexity could potentially interfere with comfort, then it will likely be ignored by much of the market. Only when a compelling value proposition can be presented to the customer, with minimal risk to comfort, will a new measure be adopted on a widespread basis.

A significant benefit for TIER is the fact that its subcomponents are not new, it's simply the combination and overall architecture of the system that is different. AWHPs, HRCs, cooling towers, and sensible TES tanks are all existing pieces of equipment, but the combination is new. The primary challenge with TIER is the complexity of controlling the system and novelty with the way that the components are applied.

Barriers to Adoption

Table 9 and Table 10 illustrate our understanding of barriers to more widespread adoption in two lists: one focused on technology barriers and the other focused on market barriers. The technology barriers more directly relate to limitations or progress needed on the hardware, software, controls, modeling, and design aspects of TIER. Market barriers relate more to the supply chain, customer, and economic aspects. There is overlap between the two categories but we feel that it is helpful to separately consider them since the interventions will also correspondingly tend to distinctly address each bucket of barriers. Note that numbering does not reset across the "technology" and "market" categories to create a single count.

Table 9: Technology barriers to TIER adoption.

No.	Technology Barrier	Description
1	Energy modeling software limitations	Energy modeling limitations inhibit the ability to quantify TIER benefits and achieve code compliance or incentives. The Project Team is aware that EnergyPlus is currently being developed to be able to model TIER. Modelica is further along but is a more niche option.
2	Code compliance limitations	Initial sites installing TIER require an "exceptional method" to achieve Title 24 Part 6 code compliance. Code compliance needs to be straightforward for TIER to become more popular. Furthermore, we want the EE-related advantages of TIER to be able to show up in a performance model to make the technology more attractive to designers/owners.
3	Lack of a standard design approach	By now, several publicly available pieces of content (e.g., an ASHRAE Journal article, an ACEEE Summer Study paper) have conceptually described the system. More concrete information is going to be required to expect additional TIER projects.
4	HVAC design software limitations	In contrast to energy modeling for code compliance & other 8760 analyses, typical load calculation software only partially supports design considerations for TIER projects. Some nascent tools exist but the broader community may be unaware of how to quantify TIER systems for their designs.

No.	Technology Barrier	Description
5	Novelty with how typical system components are applied	This relates to specifying water-cooled chillers (WCC) & AWHPs in a non-standard manner for the TIER system (e.g., the HRC operating envelope being CW-HW, how the AWHP delivers tepid CW, the WCC having CW pass through its evaporator). The technical capabilities exist but it is rarely done so manufacturers and distributors may not be able to assist or work with these jobs as easily as they would on a standard WCC or AWHP project.
6	Standard understanding around TIER performance parameters	TIER and components currently lack a standard terminology and framework for specifying, designing, and applying TES equipment. For TES, items to standardize include capacity, discharge rate, round trip efficiency, stratification, sizing, etc. For AWHP/HRCs, part load performance is currently a black box since there is no heating integrated part-load value (IPLV) being published.
7	Limited manufacturer-generated TIER materials	Not all manufacturers are currently describing or promoting their equipment in ways that would align with TIER designs. To be clear, some are, but it is not universal yet. There are not many examples of robust application guides or manufacturer support resources (such as application engineering support) to help troubleshoot issues with the equipment for sites that may be interested in pursuing TIER.
8	Facility engineer ability to operate	This barrier is related to the shortcoming around the facility engineer workforce's level of familiarity & ability to maintain TIER plants. It is one thing for technology to be effectively designed and installed, but if the workforce is incapable of maintaining it effectively then it will not succeed.
9	Controls complexity	The controls sequence for operating this system is novel and has been time consuming for even experts to debug and troubleshoot issues that arise. This is a key barrier that relates to other technical & market barriers.
10	Electric panel impacts	For large non-residential buildings, it may be possible to avoid panel upgrades, but if this cannot be avoided then this could be a very time consuming (from the perspective of utility agreements/electric equipment retrofit) and costly barrier to adopting all-electric systems. The TIER approach of leveraging "low lift" chillers operating between CHW and CW or CW and HW envelopes could lead to different conclusions around whether the electrical engineer is operating off a worst-case scenario or taking load diversity/operating schedules into account. Different types of TES could exacerbate or minimize this issue (e.g., HW TES could be less impactful).

No.	Technology Barrier	Description
11	Existing HVAC system limitations	HW supply temperature, state of controls, hydronic piping layout, and other legacy HVAC system design features may need to be changed or updated prior to making a TIER plant a viable option for existing buildings.
12	Lack of AWHP and HRC performance data for modeling software	AWHP and HRC chiller equipment requires representative performance data to create accurate energy models in software such as EnergyPlus. This information can be generated using manufacturer-developed selection software at a wide range of inlet/outlet water temperatures and load levels, but it is time consuming and complex to generate and sometimes market actors are reluctant to provide it.

Table 10: Market barriers to TIER adoption.

No.	Market Barrier	Description
13	Building owner awareness	Our assumption is that it is extremely unlikely that the average building owner is aware of the opportunity to apply TIER-style designs to their planned projects or retrofits of existing buildings.
14	Building owner willingness to adopt	Even if a building owner knows about the benefits of TIER systems, there are concerns that building owners may be unwilling to take a risk on a newer system design option at this stage of TIER's maturity in the market.
15	Lack of market penetration	This barrier relates to the straightforward fact that a new technology is going to be hard to sell when there are so few successful examples of it functioning in the real world. It is reasonable that building owners are cautious and hesitant to embrace a brand new technology without a lot of other examples to point to.
16	Lack of appreciation for TIER benefits at architect, designer, and distributor level	If architects, distributors, contractors and designers are not excited about TIER, then it's unlikely that they would be able to articulate and sell the technology to customers or be a resource for problems that arise for sites that do make the decision to pursue TIER.
17	Lack of a supply chain around TES equipment	This barrier relates specifically to the TES component of the heat recovery/TIER system plant. Ice TES equipment has a supply chain, but it's primarily focused on cooling-related use cases today. Custom fabricated sensible water TES tanks are not a difficult piece of equipment to design, but the waste heat recovery use case is not common today.

No.	Market Barrier	Description
18	Perception of a complicated process	This barrier is related to a few others on both the market & technical sides. The "perception" is grounded in reality. There is currently significant complexity and uncertainty with regard to TIER design and installation. And the effect of this barrier is that it will limit customer participation until it can be overcome.
19	Space for TES in an existing building	TIER plants are more "space effective" overall relative to a full AWHP plant. But that doesn't negate the fact that when compared to an existing building with a boiler, it's possible that a TIER plant retrofit would take up more physical space in the building. For new construction - when comparing two all-electric systems - TES is clearly more space efficient, but it's a different scenario in retrofits.
20	Misguided perceptions around NR HVAC TES	This barrier is related to the historical impressions regarding ice TES systems in HVAC designs. The industry seems to have a mixed view on the equipment based on anecdotal conversations. It is surmised that this historical barrier may map onto the TIER / waste heat TES potential market as well.
21	Upfront cost of TIER relative to boiler retrofit	Although there is a negative material incremental measure cost for TIER relative to an all AWHP baseline, when compared to a gas boiler, a TIER plant is more expensive. Furthermore, the "soft costs" to deal with new technology are typically very conservative. Contractors and designers will likely charge more for a design with a less proven track record due to the possibility that extended installation and commissioning is needed.
22	Potential for on-bill impacts to the customer from electrification	This barrier refers to the "spark gap" between gas and electricity prices for many customers. There are going to be a lot of site-specific nuances and considerations but broadly it's true that electricity is not cheap enough to guarantee on-bill savings for customers seeking to electrify their HVAC system.

The tables of barriers are presented to attempt to itemize the full suite of challenges facing TIER market adoption to clarify and strategize opportunities to overcome the barriers. What is missing in the list is a notion of causality or sequencing of the barriers. Certain barriers are more important to address earlier in the market transformation process, while others may only be important to actively address later. It is quite possible that as certain barriers are addressed, other unforeseen barriers will arise and need to be dealt with. This is why it is important to regularly reassess the state of the market and update barriers as more knowledge and experience is gained.

Interventions to Address Barriers

After identifying and internalizing the full array of technology and market barriers, the project team developed a set of intervention proposals. These potential interventions are grouped by categories and shown in Table 11 through Table 16. Interventions could be led by IOU programs, state or local governments, or the private sector. The suggestions in Table 11 could be tasks within a single study or broken out into separate studies.

Table 11: Further ET research interventions.

No.	Title	Description
1	Study on additional TIER plants	Additional studies would be valuable if they intentionally target different building types, TIER configurations, and climate zones beyond the focus of this project.
2	Study for TIER and electric panel upgrades/demand response	A study that specifically looks at the electric panel upgrade/service impacts and how TES can mitigate or avoid this need. A study could also factor in a demand response/load flexibility component to research those potential benefits.
3	Study on TIER and space constraints	A study that focuses on the space constraint of TIER in existing buildings, potentially including a real world retrofit, tracking costs, and how space can be made.
4	Study on TIER and project costs	A study of TIER project costs to seek to understand up-front and ongoing bill impacts for various offerings of TIER retrofits.
5	Create a database of TIER projects	The purpose of this intervention would be to establish and maintain a public-facing database of TIER projects. This would address market barriers related to stakeholder unwillingness to pursue TIER if they have the impression that it is a rare and bespoke system concept. This could also provide a mechanism for stakeholders to get in touch with sites already operating TIER plants to help spread knowledge and lessons learned.

Table 12: Upstream supply chain interventions.

No.	Title	Description
6	Government-sponsored TIER "challenge"	The Project Team envisions a potential formal government-sponsored "challenge" that could be issued to the manufacturers to create products and materials that align with TIER. A product specification and other directions that could be broadcast out for them to all adhere to. The specification could ask for certain set of information or data in their product documentation. For example, require that the chiller operating envelopes need to include settings that are uncommon in standard hydronic plants but needed to properly size a TIER plant.

No.	Title	Description
7	Encourage equipment vendor competition	This intervention would involve encouraging competition in the marketplace among manufacturers. This would be contingent on accelerating market demand, but actively relaying market progress and momentum, if it picks up, to encourage additional manufacturers to promote their products in this way.
8	TIER "Challenge lite"	This would be a "lighter" version of a challenge, something more informal, such as conveying the opportunity to manufacturers and encouraging them to develop their own materials around TIER. That could be accomplished with targeted outreach or by collaborating in other venues, such as Air-Conditioning, Heating, and Refrigeration Institute (AHRI) committees or ASHRAE working groups.
9	Encourage controls vendor to program TIER sequences	This intervention would involve collaboration specifically with controls vendors and manufacturers to encourage them to create guides and pre-programmed sequences in their equipment that can cover sequences required for TIER systems.

Table 13: Standardization/modeling interventions.

No.	Title	Description
10	TIER in physics BEM software	This intervention would add TIER modeling capabilities to physics building energy models (BEM) software, particularly EnergyPlus, to enable modeling opportunities for designers and energy modelers. EnergyPlus is the basis for the California Public Utilities Commission (CPUC) and California Energy Commission (CEC) incentive programs and code compliance software, respectively, and therefore an EnergyPlus object is crucial for market transformation progress in California.
11	TIER in Title 24 performance compliance software	In addition to adding TIER capabilities to physics BEM software, TIER should be added to the Alternative Calculation Method Reference Manual and code compliance software (e.g., the California Building Energy Code Compliance [CBECC] software, which is based on EnergyPlus).
12	TIER ASHRAE content	Work with ASHRAE to develop content such as a guideline, method of test, design guide, etc. Potentially initiate this action with Technical Committee (TC) 6.9, “Thermal Energy Storage” or TC 6.8, “Geothermal Heat Pump and Energy Recovery.” Consider a new section in a relevant chapter of a Handbook textbook (perhaps Chapter 50 “Thermal Storage” HVAC Systems and Equipment) that lays out system design parameters, sizing best practices, controls sequences.

No.	Title	Description
13	AHRI standards enhancements	Work with the AHRI hydronics STC regarding test procedure and ratings enhancements for equipment that contributes to a TIER system, particularly standards 550/590, “Performance Rating of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle” and 900, “Performance Rating of Thermal Storage Equipment Used for Cooling.” Promote new ratings or adjustments to the existing ratings to better describe AWHP and HRC equipment being used in TIER configurations. Work to create a heating IPLV for AWHP and HRC equipment. The AHRI 900 TES test procedure should be broadened to include heating TES in addition to its current focus on cooling TES.
14	TIER early design assistant	This intervention would create a streamlined early design tool to give building owners a rough idea around whether their site is a good candidate for TIER. This tool would not be intended to provide a full design but rather serve as a screening and early-stage design assistant.
15	TIER in HVAC design software	This intervention would be focused on creating rigorous, in-depth HVAC design software that can be used to fully lay out TIER plants. Existing major products such as Trane TRACE and Carrier HAP could be engaged for expansion to include TIER. Additionally, a project to release a standalone TIER design tool could be undertaken. This product could focus on equipment sizing and layouts, as well as controls design.
16	TIER qualified product list (QPL)	The purpose of this intervention would be to develop and maintain a QPL or equipment list of all system components that can be applied to TIER designs. This would assist designers and other stakeholders with understanding which pieces of equipment can be leveraged for TIER systems. It would also assist with incentive program development.
17	Collect AWHP and HRC data for BEM	Work with manufacturers and/or distributors to collect expanded performance maps across a range of inlet/outlet temperatures and load levels that would yield more accurate energy modeling runs for TIER plants in BEM software such as EnergyPlus.

Table 14: Workforce education interventions.

No.	Title	Description
18	TIER case studies	Create simple, visually compelling literature that would highlight TIER benefits in real buildings. The documents would be intended for a wide audience, including building owners, designers, architects, distributors, and program administrators. The intent is to provide third party validation of TIER plant performance.
19	TIER Design Guide	In contrast to a case study, this literature would be significantly more in-depth and technical and focused more exclusively on the designer, distributor, and contractor workforce. This work product would be intended to assist designers and installers with understanding the granular details of TIER designs.
20	Designer training	The purpose of this intervention would be to train designers in a formal classroom setting. These could be sponsored by CA IOUs, manufacturers, or national organizations such as ASHRAE.
22	Contractor/facility engineer training	This intervention would focus on a formal training for the contractor, installer, and equipment operator/facilities engineer audience.
23	Conference/trade group awareness	This intervention would be focused on broadly raising awareness of TIER in public settings such as conferences, trade shows, building owners' trade groups, and other opportunities.

Table 15: Financial incentive interventions.

No.	Title	Description
24	Electrification readiness incentives	This intervention would target “readiness” for TIER by incentivizing all of the modifications that are expected for many existing buildings as precursors to the actual installation of a TIER plant. Measures such as HVAC controls modernization, lowering the HW supply temperature, building envelope improvements, and potentially electric panel upgrades that would enable a future TIER project.

No.	Title	Description
25	TES incentives	This intervention would be focused on TES incentives in particular. There is currently the IRS Tax Credit 48E[1] available for commercial thermal energy storage. The California self generation incentive program (SGIP) is potentially available if the 5 kg/kWh threshold of carbon reduction can be achieved. Other CPUC-based incentive opportunities are less obvious due to the rules around EE impact requirements and the perception that TES is a load flexibility technology.
26	Market Support Incentives	Market support programs are more flexible than resource acquisition (RA) programs which may make them a compelling candidate for TIER incentives in the near term. They are generally undertaken for purposes other than total system benefit (TSB) impact accomplishments. Market support programs could be ideal for TIER if the objective is to help induce early adopters to select this system, improve familiarity throughout the workforce, etc. The Project Team's understanding is that the CEDA program is a new construction market support program and could work to promote TIER systems. New construction could be a more straightforward near-term TIER opportunity due to the "blank slate" aspect of these buildings as compared to existing buildings with legacy systems.
27	RA Incentives (future)	At some point, if TIER systems become well understood and impacts can be readily quantified, RA programs could become a possibility. The custom delivery type is almost certainly the only viable choice for the foreseeable future, since deemed measures must be highly repeatable and occur in large volumes to make sense. But at some point, it may be possible to create deemed offerings for components of a TIER system and then assemble a custom or hybrid measure out of a few deemed components.
28	Spark gap incentives	This intervention is included to acknowledge the potential for the ongoing differential between gas and electricity utility rates. The Project Team does not know of any existing incentives that deal with this, and this is likely a large lift to get through regulatory approval prior to roll-out. There may be a special rate tariff for all-electric sites, which has been accomplished outside of California.
29	Flexible load incentives	This intervention would focus on incentivizing the demand response and/or load flexibility potential of TIER systems in particular. There are existing demand response programs and unique tariff structures offered by the IOUs that would reward load flexibility.

[1] https://www.irs.gov/credits-deductions/clean-electricity-investment-credit.

Table 16: Policy interventions.

No.	Title	Description
30	Voluntary/reach (and eventually prescriptive) building codes for TIER	Further out in the future building codes and TIER should be considered. This could initially start with voluntary or reach code measures that could be adopted by a smaller number of local jurisdictions, and then further out, a prescriptive statewide base code measure in Title 24 Part 6 and ASHRAE Standard 90.1. Measures targeting very large buildings could be introduced sooner and then broadened to smaller buildings in the future. The Project Team is aware that national model codes are introducing TES-oriented “credits” to encourage TIER-like systems.
31	Building performance standards (BPS) and TIER	This intervention would be more indirect in the sense that future BPS requirements could encourage TIER based on its efficiency benefits that would help achieve the given BPS targets. The policy would not directly require TIER, but presumably if aggressive efficiency targets are set, then efficient systems such as TIER would be favored
32	Improve AWHP and HRC representation in ASHRAE Standard 90.1	This intervention would be focused on enhancing the equipment rating requirements in ASHRAE Standard 90.1 Table 6.8.1 for AWHPs and HRC to include more representative heating, heat recovery, and part load performance ratings, such as heating IPLV. It may also be possible to include TES performance requirements as well.

Just as with barriers, there is a sequencing aspect to the interventions that is not fully reflected in these tables. Certain interventions are more important to focus on earlier in the market transformation process, while some are inappropriate (e.g., deemed measures) before more information has been collected and early-stage barriers have been overcome.

Market transformation theory involves segmenting the market into categories such as “early adopters,” “early majority,” “late majority,” “laggards,” etc. For an innovative technology like TIER, it is important to focus early-stage interventions on the early adopters who are willing to take more of a risk for a technology with a less proven track record. These are building owners who may be motivated to electrify space heating for mission or public-facing reasons, or buildings with some type of corporate or government mandate to electrify. In addition, buildings with severe space constraints preventing an all-AWHP plant or another economic reason to be driven toward TIER should be a priority for early program efforts.

It is worth noting that this study has attempted to advance several of the interventions noted above. Specifically, the case study, simplified tool, design guide, and initial manufacturer engagement components of this study are all interventions intended to advance the state of knowledge and ease of application of TIER in the market.

Matrix of Barriers & Interventions

After envisioning a list of intervention opportunities based on the identified barriers to adoption, the project team then cross referenced the two lists to understand which interventions could potentially be more impactful or important relative to others. This matrix of barriers and interventions to TIER adoption is shown in Figure 60. This is an imperfect approach because of the notion that the magnitude and sequencing of the barriers is not displayed, but this exercise is meant to directionally show which interventions are more likely to provide a meaningful return on investment. In a perfect world, all interventions will be undertaken, but since funds are scarce, prioritization is important. Key intervention opportunities are described in some more detail later in this section.

		Barrier Type	Technical												Market										
		Barrier No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	
Intervention Type	Intervention No.	Intervention\Barrier	Energy Modeling software limitations	Code compliance limitations	Lack of a standard design approach	HVAC Design software limitations	Novelty with how typical system components are applied	Standard understanding/language around TIER concepts	Limited manufacturer-generated materials	Facility engineer ability to operate	Controls complexity	Electric panel impacts	Existing HVAC system limitations	Lack of AWHP and HR chiller performance data for BEM	Building owner awareness	Building owner willingness to adopt	Lack of market penetration	Lack of at architect, designer, distributor appreciation	Lack of a supply chain around TES equipment	Perception of a complicated process	Space for TES in an existing building	Misguided perceptions around NR HVAC TES	Upfront cost of TES project	Potential for on-bill impacts to the customer	Count of barriers addressed
Further Research and Studies	1	ET study on additional TIER plants											✓		✓	✓		✓		✓					5
	2	ET study for TIER and electric panel upgrades/DR										✓	✓			✓		✓		✓					5
	3	ET study on TIER and space constraints											✓			✓					✓				3
	4	ET study on TIER and project costs														✓							✓	✓	3
	5	Create database of TIER projects													✓	✓		✓				✓			4
Upstream supply chain	6	Government-sponsored TIER "challenge"					✓		✓					✓					✓	✓		✓			6
	7	Encourage equipment vendor competition							✓						✓	✓	✓		✓				✓		6
	8	TIER "challenge lite"					✓		✓										✓			✓			4
	9	Work with controls vendor to program TIER sequences								✓	✓														2
Standardization/ modeling	10	Add TIER to EnergyPlus	✓					✓																	2
	11	Add TIER to Title 24 Performance Compliance Software		✓																					1
	12	Develop TIER ASHRAE content			✓			✓												✓	✓				4
	13	AHRI standards enhancements						✓																	1
	14	TIER early design assistant	✓			✓	✓											✓		✓					5
	15	TIER in design software				✓	✓		✓																3
	16	TIER Qualified Product List							✓																1
	17	Collect AWHP and HR chiller performance data for BEM	✓	✓		✓		✓						✓											5
Workforce Education	18	TIER Case study													✓			✓		✓		✓			4
	19	TIER Design guide			✓	✓	✓			✓	✓							✓		✓	✓	✓			9
	20	Designer training					✓			✓	✓					✓		✓							5
	21	Contractor & facility engineer training					✓			✓	✓		✓		✓	✓									6
	22	Conference/trade group awareness					✓			✓	✓				✓	✓		✓		✓		✓			8
Incentive-related	23	Electrification readiness incentives											✓		✓	✓	✓			✓			✓		6
	24	TES Incentives													✓	✓	✓				✓		✓		5
	25	Market support incentives			✓		✓						✓		✓	✓	✓			✓	✓		✓		9
	26	Resource acquisition incentives (custom or deemed)											✓		✓	✓	✓			✓	✓		✓		7
	27	Spark gap incentives														✓							✓	✓	3
	28	Flexible load incentives														✓							✓	✓	3
Policy	29	Voluntary (and eventual prescriptive) codes for TIER															✓								1
	30	BPS and TIER															✓								1
	31	Improve HR chiller 90.1 values					✓		✓								✓								3
		Count of intervention overlap	3	2	3	4	10	4	6	5	5	1	7	2	10	15	8	8	3	11	6	6	8	3	

Figure 60: Matrix of barriers and interventions for TIER.

It is hoped that this analysis can form the basis of future research and programmatic efforts. Periodically, this exercise can be repeated to attain an up-to-date understanding of the technology's position in the market and revisit the most appropriate interventions for TIER at that time.

Market Engagement

As part of this project, a variety of market engagement efforts were undertaken. These efforts are summarized in the following sections.

MANUFACTURER OUTREACH

The Project Team has communicated with HVAC equipment manufacturers on the challenges with the rapid industry shift to all-electric central plants. In addition to the rapid growth in demand for all-electric central plant equipment, manufacturers have also recently faced severe supply chain disruptions and new restrictions on refrigerants that have required development of new product lines. Anticipation that the global warming potential of refrigerants will be further restricted in the near future has also led major manufacturers to pursue different approaches for product development. One end result of these dynamics is that product lines for all-electric central plant equipment are generally new, rapidly evolving, and fairly different in capability from one manufacturer to another. Coupled with relatively unfamiliar design considerations for designers, these rapid changes have presented challenges for manufacturers to educate designers on new equipment offerings, their limitations, and appropriate applications. In many cases, manufacturers are also learning lessons and evolving their recommendations as they gain experience with new equipment installations.

Some HVAC equipment manufacturers have expressed significant interest in the concept of turnkey packaged solutions for all-electric central plants to overcome common market barriers with system design and operational complexity. Packaged equipment solutions would include modular equipment options with integrated system controls for staging and deploying equipment and TES through the range of heating, cooling, and heat recovery conditions. Whereas built-up solutions require in-depth expertise by designers, installers, and operators and are practically limited to larger buildings and owners with more resources, packaged solutions offer the potential of reducing the need for this expertise, reducing the risk of improper equipment applications, and overall costs.

Packaged solutions potentially expand the applicable market to a wider range of building sizes, which would increase equipment sales for the manufacturers. Current practice often limits the purview of manufacturer support to their narrow equipment selections, which presents a risk of improper applications, particularly with new and rapidly evolving all-electric equipment offerings, and varying capabilities and approaches across different equipment product lines. Offering packaged solutions would expand manufacturers' purview to the full central plant, which may help ensure that equipment is applied appropriately. Centralizing the engineering knowledge of the full plant concept at the manufacturer level also streamlines the design processes.

At least one manufacturer to date is exploring the development of a packaged equipment offering based on the Project Team's outreach.

CALIFORNIA PROGRAM ADMINISTRATOR ENGAGEMENT

In early 2026, the project team met with several key influential entities within the California program landscape, including separate meetings with: Cal TF, San Diego Gas & Electric, SCE, and PG&E (collectively referred to as program administrators or PAs throughout this sub-section). The objectives of the interviews were twofold: 1) educate the PAs on the TIER technology and program opportunities, and 2) educate the project team on the barriers and opportunities for the various program options. The interviews were clarifying and educational for both parties. The meetings underscored the challenges with implementing custom measures in today's environment and outlined key opportunities, as described below.

Each meeting followed a consistent agenda: the project team educated the PA team on the TIER system and its benefits, then ran through the list of barriers and intervention opportunities described earlier in this report, followed by a group discussion on how the PA sees the TIER technology fitting into existing utility program options. The key outcomes of these discussions have been captured in the paragraphs that follow.

The consistent message across all meetings was that the traditional RA-based custom program pathway is seen as time consuming, heavily regulated by CPUC, and consequently, not commonly leveraged by third-party (3P) programs anymore. Additionally, in each meeting deemed measure packages were noted in passing but all parties agreed that this is certainly not a near-term opportunity, and due to the site-specific nuances of TIER, is unlikely to ever be an appropriate option.[2]

The top two near-term program opportunities that were uncovered in the interviews include 1) market support programs and 2) site-level normalized meter energy consumption (NMEC), which is a subcategory of custom but was described as possibly more viable than traditionally-modeled custom. The CEDA new construction program, overseen by PG&E, is classified as "market support", meaning that it is not regulated in the same way as a traditional RA program subject to the heavy CPUC oversight framework (Pacific Gas and Electric Company, 2026). Market support measures are more flexible than RA measures.

Market support programs are generally seeking some ancillary objective such as equity or market transformation, which is in line with how TIER may need to be promoted in the near term. As shown in the list of market barriers in Table 10, customer awareness, a stronger supply chain, and more willingness to adopt are needed for TIER to gain traction. Due to the novelty and complexity, it is not expected that TIER will be sited in a large quantity of buildings in the near future, and therefore RA impacts are likely to be limited for several years to come. Market support, however, can focus on objectives aside from TSB impacts. Rather than TSB, a market support program that promotes TIER could instead focus on overcoming market barriers identified above. This strikes the project team as the most compelling near-term opportunity.

CEDA is already providing support for technologies which contribute to a TIER plant on their "High Performance Measures" section of the program website. CEDA highlights two technologies that contribute to a TIER system: Space Heating Hydronic Heat Pump and Heat Recovery Chiller. These measure summaries highlight details about the technologies, how the measures align with program goals, how customers can leverage the technology to receive inducements, eligibility information, and other technical details such as pertinent codes and standards information. There are opportunities to add TIER to this list of high-performance measures.

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[2] A nuance regarding deemed is that it may be possible to "deemify" various sub-components and pieces of equipment that comprise the TIER plant, e.g., a deemed offering for the AWHP, another offering for the HR chiller, etc., and then allow sites to build up an overall incentive with multiple deemed "building blocks." This could result in a streamlined custom or hybrid incentive framework.

In March 2026, Project Team conducted interviews and knowledge sharing sessions with the two key market support programs in the utility portfolio: CEDA for nonresidential and high-rise multifamily buildings four stories or greater, and the California Energy Smart Homes program for residential and low-rise multifamily buildings less than four stories. In both sessions, program managers expressed enthusiasm about how their programs could become instruments to further popularize and promote TIER plants. In particular, CEDA noted the applicability of TIER to the larger building segment. California Energy Smart Homes could be a more appropriate program for designs that leverage heat recovery or thermal storage on a smaller scale, but those buildings would not be an ideal candidate for the TIER plant that was the focus of this study. CEDA noted that their program, as a market support program, could be a good mechanism to promote TIER plants due to the flexibility of their incentive opportunities and connection to the PG&E code readiness pipeline. CEDA staff echoed the notion that TIER is a novel system and that future efforts to make its complexity more understandable and actionable by the program and building owners would be valuable. CEDA staff agreed that a future TIER "high performance measure" guide would be a valuable next step.

Though custom RA programs are challenging, the PA interviews did note that there could be some uptake in the measures if site-level NMEC is pursued. Site-level NMEC would only be applicable to existing buildings and would rely on changes to the meter data pre- and post-installation to confirm the efficacy of the measure. A PA engineer noted that submetering plant equipment would enable stronger measurement and verification outcomes.

The distinction between existing buildings and new construction came up on multiple calls. All PAs agreed that new construction seems more appealing due to the complexity of retrofitting an existing system over to TIER. However, it is also the case that the impact potential for existing buildings is far more significant than new construction, of course. As illustrated above, there are numerous barriers to adoption for TIER. Impacts are unlikely to be an immediate focus near term, instead, raising awareness, highlighting early adopter buildings, streamlining the complexity of installation (including code compliance and modeling), and educating the workforce are among the near-term priorities. Since impacts are going to be limited regardless, RA programs are simply too burdensome to engage with relative to the market support option.

As a new construction market support program, CEDA should be actively engaged in the future as a mechanism to promote TIER. This is a significant finding of this engagement exercise.

Other aspects of these calls are worth briefly calling out:

- Efforts to streamline: Cal TF has an ongoing effort to better document and streamline the custom program process.[3] The Project Team is of the view that while TIER could make sense for custom, it would be significantly more effective to work within a streamlined framework as opposed to the current process. It may be beneficial to specifically leverage TIER technology as a pilot measure for ongoing Cal TF custom enhancement work.
- SGIP: The CPUC-led could potentially incentivize thermal energy storage if it can demonstrate a minimum carbon reduction rate of 5 kg/kWh, per CPUC decision D. 19-08-001.[4] At this stage,

[3] https://www.caltf.org/tools-custom.

[4] CPUC Decision 19-08-001: https://docs.cpuc.ca.gov/PublishedDocs/Published/G000/M310/K260/310260347.PDF.

the Project Team has not deeply investigated whether TIER plants can meet this threshold, and this is an item for future research.

- Custom load shapes: This parameter was highlighted by a PA as an important enabling piece of the puzzle prior to custom programs becoming a viable option for TIER. According to PAs, IOU staff are needed to manually enter the data into CEDARS. Custom load shapes are considered important by the PAs for TES and load shifting measures, which are key components of TIER.
- Code compliance & SB1414: PA staff noted that code compliance will need to be demonstrated. TIER currently has some additional barriers to code compliance due to its absence from performance compliance software now.
- TIER as a platform or framework: TIER as a platform or framework whose elements can be modified based on climate and building conditions rather than a rigid system layout. With one PA, it was discussed that in southern California, heating loads are so minimal that TIER may be achievable without the need for a trim heating source (e.g., the AWHP component). Note that this would still treat TIER as a HP-based system due to the continued presence of a HRC in the layout, which is effectively a water-to-water heat pump.
- 3P custom programs: a PA described 3P custom programs as being less focused on ambitious, complex, time consuming efforts such as TIER retrofits and are more focused on more straightforward measures that can be accomplished more quickly and do not require significant “back and forth” with CPUC. This indicates that additional barriers noted above would likely need to be addressed prior to TIER becoming a compelling candidate for custom programs.

Regardless of whether RA or market support program framework is pursued, all PAs agreed that additional support and tools for the 3P programs are crucial. This includes content that can educate the 3P programs about TIER and its applicability, simplified modeling and screening tools, literature that can be shared with potential building owners, and improved education and incentives surrounding the technology. This “toolkit” of materials is a compelling next step for the technology transfer process from the Project Team’s perspective, to advance TIER beyond emerging technology (ET) and into the incentive program space.

Key Near Term Intervention Opportunities

As noted above, since resources are scarce, prioritization is critical. Though introduced in the tables above, some more consideration is provided regarding the reasons why these interventions were targeted as the most important near-term intervention opportunities. Of course, highlighting these opportunities does not minimize the importance of other intervention concepts.

ADDITIONAL TIER ET STUDIES

Though the Project Team views this research and study as a meaningful step forward, there is more emerging technology work to be done. As noted, the HVAC industry is cautious and conservative, and overwhelming evidence of a new technology must be demonstrated and proven before a significant degree of uptake can be expected. In addition to the open questions identified earlier in the report (e.g., demand management/load flexibility potential, more data or confidence about TIER project costs, a better understanding of the relationship between TIER retrofits and electric panel impacts), there is value in simply seeing more studies and proof points that demonstrate the benefits of TIER described in ET research products. For example: TIER performance in various climate zones, building types, or other nuances. TIER with an electric resistance boiler, TIER with a large TES tank, TIER serving a campus, TIER with no AWHP, or TIER paired with a ground loop.

TIER DESIGNER SUPPORT

A critical need for TIER and market transformation is a highly educated workforce, which starts with the design community. The design guide advanced in this project is an important first step, but the Project Team's view is that more work can be advanced in this area. As noted, ASHRAE may be an appropriate vehicle to advance TIER-related content. In addition to a design guide, designer education efforts at ASHRAE conferences and other technical events would be helpful.

TIER INCENTIVES

Incentives are an important mechanism to induce reluctant customers and owners to take a risk with a newer system. The incentive landscape was described in-depth and based on the research conducted under the auspices of this project, our tentative conclusion is that market support and electrification readiness incentives are the most important near-term opportunities to pursue. The key market support program identified in our research is CEDA. The concept of advancing a "toolkit" of materials that can assist 3P programs with their understanding and ability to promote TIER is a compelling next step in the Project Team's view. As noted above, CEDA is already providing support for hydronic heat pumps and heat recovery chillers as standalone pieces of equipment. The Project Team recommends expanding this support to also cover TIER, which is a combination of these constituent technologies.

An important opportunity to highlight is the ongoing IRS TES tax credit (48E). PAs should be aware that this tax credit can enable the building owner to reduce the project cost by 40 percent for thermal storage equipment. This tax credit was introduced with the Inflation Reduction Act of 2022 and remains active as of early 2026.

UPSTREAM (MANUFACTURER) COORDINATION

As highlighted above, manufacturers can play a very influential role in positioning their equipment to be more easily leveraged in TIER systems. Manufacturers could work to produce more explicit performance data for equipment being installed in CW-level TES TIER layouts. They could produce their own application guides. For the medium-sized building segment, as noted above, manufacturers could work toward a packaged solution that makes TES-based HVAC systems more of an "off the shelf" solution, comparable to how VRF systems are currently promoted. For controls vendors, programming TIER sequences into their libraries is a key intervention opportunity to make additional installations more straightforward in the future.

TIER IN PHYSICS-BASED MODELING SOFTWARE

Adding TIER into physics-based BEM software such as EnergyPlus will enable more widespread modeling opportunities for this system. CPUC and CEC impacts are generally derived using EnergyPlus software now and are expected to continue to be for the foreseeable future. TIER in EnergyPlus, supported by accurate AWHP and HRC performance data, will enable more widespread incentive and code compliance opportunities for TIER and should be a high priority.

PROMOTE TIER AT CONFERENCES

Awareness of TIER and its benefits needs to increase for it to be more regularly considered for building designs. Promoting TIER, including the outcomes of this research, should be a priority for future conferences, trade shows, ASHRAE dinners, building owner consortia, etc.

Recommendations

The project team recommends the following actions to advance the development and deployment of energy-efficient, all-electric central plants for large commercial buildings:

1. Prioritize TIER Strategies for Complex Central Plants: TIER configurations should be strongly considered for large central plants operated by teams with the sophistication and capacity to manage advanced, integrated systems. Early field results and modeled analyses demonstrate that TIER can achieve significantly higher efficiencies than conventional all-electric alternatives.
2. Develop Simplified and Packaged TIER Approaches: Current TIER implementations can be complex. There is a clear need for simplified TIER variants—either packaged, vendor-supported solutions or built-up configurations using fewer components and more straightforward control strategies. Actual building load profiles may limit the realization of theoretical efficiency gains, making simpler, more robust designs valuable for wider market adoption.
3. Design for Realistic Load Profiles and Turndown Requirements: Both measured and simulated TIER plants showed a high number of operating hours with extremely low heating and cooling loads, resulting in frequent cycling of chillers and heat recovery chillers (HRCs). This cycling significantly reduced annualized efficiency and mirrors findings from other AWHP studies. Large plants with expected low load periods should evaluate options such as pony chillers, modular chillers, or other low capacity equipment to maintain stable operation and avoid excessive cycling.
4. Expand TIER Demonstrations and Performance Evaluations: More field installations and comprehensive demonstrations are needed to deepen understanding of TIER performance, refine design guidelines, and validate modeling assumptions. Newly developed spreadsheet-based modeling tools can support detailed evaluations of TIER performance where conventional building simulation platforms cannot currently model these systems.
5. Select Equipment Based on Application-Specific Part Load Behavior: Part load performance must be scrutinized during design and equipment selection, as it has a dominant influence on annual plant performance.

    - Centrifugal chiller performance variability: In the CW TIER model, the lead centrifugal chiller operated below 20 percent part load ratio for most of the year—below its turndown threshold—and exhibited roughly double the kW/ton compared to operation above 40 percent part load ratio. Different performance curves for the same equipment type can produce dramatically different outcomes. Under different load distributions where the chiller operates at higher PLRs, a CW TIER configuration may outperform a HW TIER system.
    - Performance curve selection limitations: The variable speed centrifugal chiller curve from the EnergyPlus library was selected to match capacity and normalized to the site chiller's nominal efficiency. However, the library curve's kW/ton at 20 percent part load ratio was nearly twice that of the actual chiller. This discrepancy significantly influenced modeled annual energy use, demonstrating that selecting curves solely by equipment class is insufficient.
    - Need for improved performance curve libraries: More research and curated data libraries are needed to characterize part load performance, particularly for emerging equipment types. Current EnergyPlus libraries lack representative curves for HRCs at higher lift

conditions. Developing custom project-specific curves is resource-intensive and rarely performed, yet this study shows that curve selection greatly affects predicted performance.

6. Strengthen Technical Transfer and Market Support. To enable broader adoption of TIER and other advanced all-electric plant designs, the industry will require:
    - Improved and more flexible modeling tools capable of simulating complex central plants
    - Utility incentive programs aligned with heat recovery and high-efficiency electrification strategies
    - Manufacturer engagement to support packaged or semi-packaged TIER solutions
    - Workforce training and design guidance materials for engineers and operators

    These interventions will reduce risk, improve design quality, and help accelerate market adoption of high-efficiency, all-electric plant configurations.

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Appendix A

System Schematics for Condenser Water and Hot Water TIER Plants

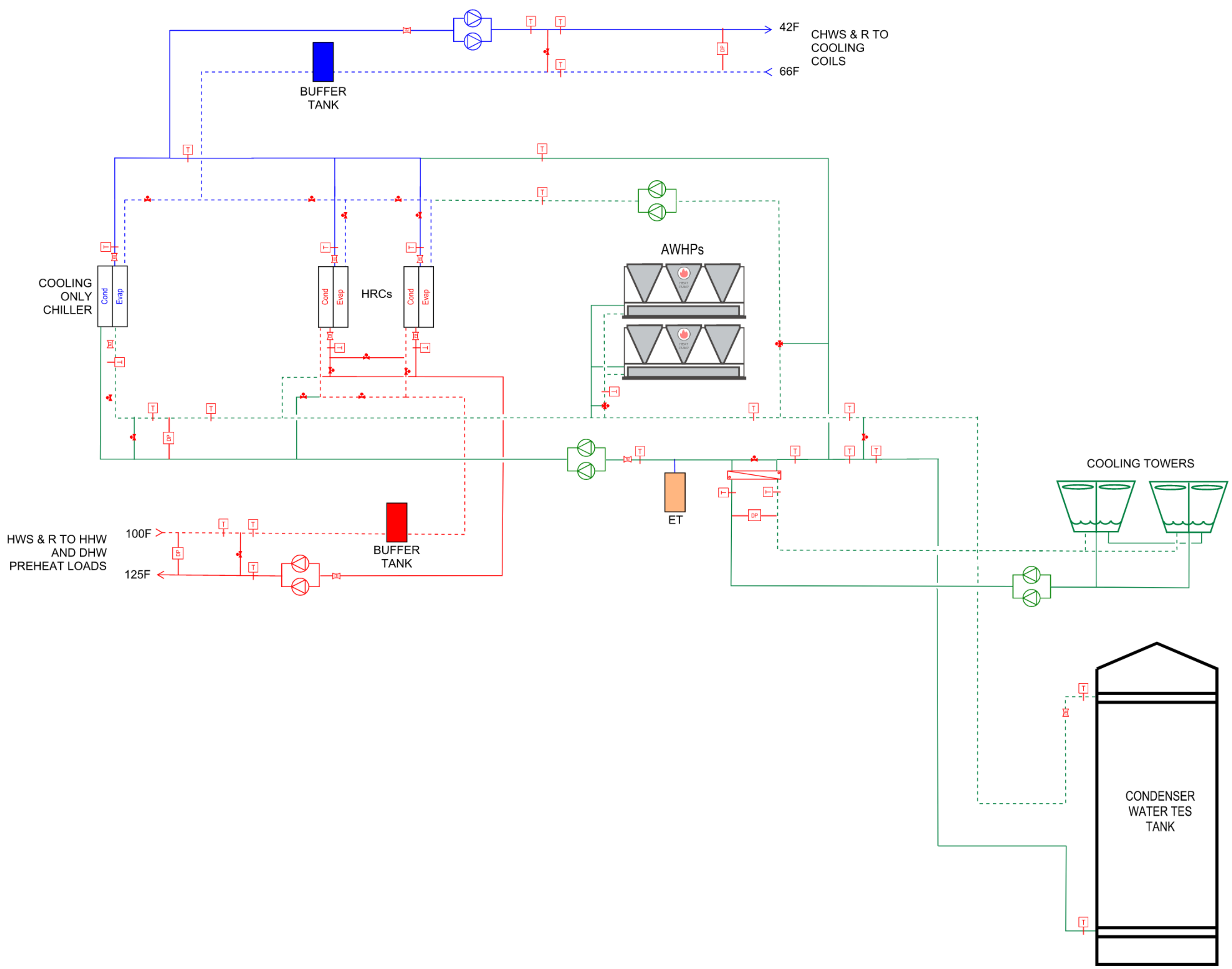

Figure 61: Condenser water TIER system schematic.

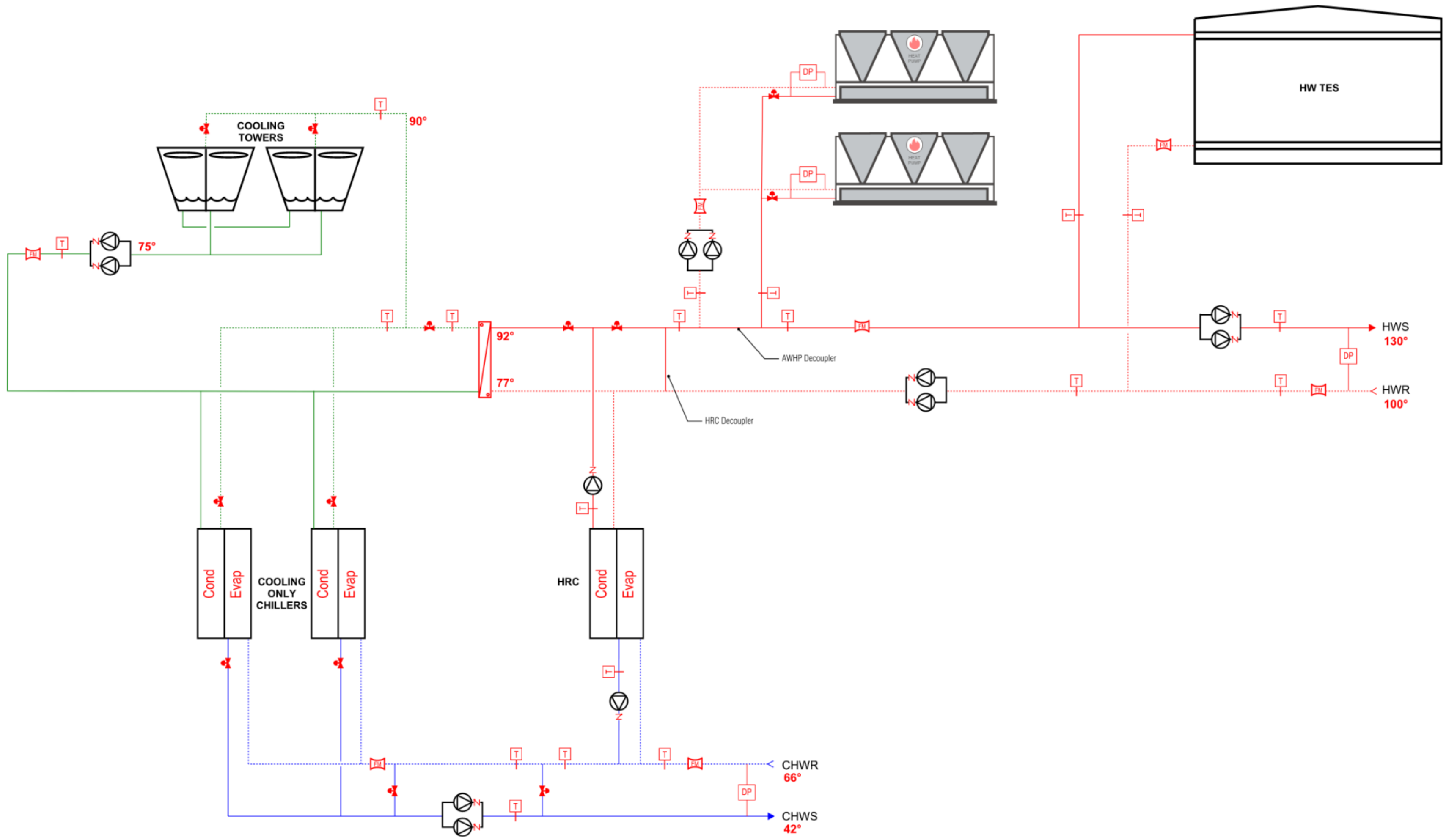

Figure 62: Hot water TIER system schematic.

Condenser Water TIER Plant: Description of System Operating Modes

The CW TIER system consists of multiple components designed to meet both cooling and heating demands, including chiller/s, HRC/s, TES, heat exchanger/s, cooling tower/s, and AWHP/s. Chillers provide CHW for space cooling, serving as the central source to meet the cooling demand. The purpose of the HRCs is to provide HW by reclaiming heat from the condenser side, enhancing overall system efficiency, and reducing the need for separate heating. TES tank stores excess heat rejected from the chiller or HRC. The heat exchanger facilitates energy transfer between hydronic loops—specifically the condensing water loop and cooling tower loop without direct mixing of fluids, ensuring operational flexibility and system protection. The cooling tower rejects excess heat from chillers and on some occasions HRCs to the ambient air. Finally, the AWHPs provides supplemental heating to charge the TES tank when recovered heat is not sufficient. Below is a summary of the operational modes and details from the TIER plant.

1. Heating only, TES tank is fully charged

In this operating mode, the system functions in heating-only mode with the TES tank fully charged. The chillers remain off because the CHW load is zero, and the cooling tower is also off since no heat rejection from CW loop to the ambient air is required. Similarly, the AWHP is off. The TES tank operates in a discharging mode, supplying stored CW for the HRCs to meet the building's heating needs. The HRCs' operation is dependent on the amount of HW load. In this operation mode, one HRC is on when there is a non-zero HW load, while the other two HRCs remain off unless the heating load is large enough to trigger their operation.

It is worthwhile mentioning that HRCs are not used for cooling in this operation mode. The AWHP is off and the net heat from TES tank is negative, which means that the tank is discharging.

2. Heating only, TES tank is not fully charged

In this operation mode, the system is in heating-only mode, with the TES tank not fully charged. The chillers are off since the CHW load is zero, and the cooling towers are also off because there is no requirement for heat rejection to the ambient air. The AWHP may turn on and operate to charge the TES tank, while the HRCs could be on or off, depending on the CHW load anticipated later in the day and the depletion rate of the TES tank charge. The TES tank simultaneously discharges for the HRCs to meet the HW load, and heat from the AWHP contributes to recharging the tank.

Overall, this mode represents a heating-dominant condition in which the system operates without cooling demand. The AWHP provides supplemental heat to recharge the TES tank if needed while the HRCs deliver heating to the building. The cooling tower and chiller remain off.

3. Heating larger than cooling (heating > cooling)

In this operation mode, the system operates under heating-dominant conditions with some cooling demand. One or both chillers are active to meet the CHW load, while the HRCs operate to provide heating to the building. The TES tank may either charge or discharge, depending on the balance between heating supply (rejected heat from chillers to CW plus heat provided by AWHP) and demand. The cooling towers remain off since all rejected heat from the cooling process can be utilized within the system for heating, eliminating the need for external heat rejection.

4. Cooling larger than heating (cooling > heating), TES tank is not fully charged

In this operation mode, the system operates under cooling-dominant conditions where cooling demand is greater than heating demand, and the TES tank is not fully charged. The chillers (at least one) are on to meet the CHW load. The HRCs could be in cooling or heat recovery modes depending on CHW and HW loads. The AWHPs may also operate intermittently, depending on system conditions. The cooling towers may be on or off depending on the level of excess heat rejection required. The TES tank is in charging mode, storing thermal energy from excess cooling capacity.

Overall, this mode represents a mixed but cooling-dominant operating condition, where the chiller provides the primary cooling, and part of the recovered energy is stored in the TES tank. The system efficiently balances cooling production, heat rejection, and limited heat recovery while maintaining optimal energy use through selective operation of the AWHPs and cooling towers

5. Cooling only, TES tank is not fully charged

In this operation mode, the system operates under cooling-only conditions with the TES tank not fully charged. One or two chillers are on and supply CHW to meet the building cooling load. In some hours, one or more HRCs operate in cooling mode to assist chillers in meeting the cooling load. The cooling tower may be on or off depending on the CW loop heat rejection needs, and the AWHP could either be off or operate intermittently based on prior hours' heat removal from the CW loop. The TES tank is in charging mode.

Overall, this mode represents a partial charging condition during a cooling-dominant period. The chiller operates as the primary cooling source, with the TES tank storing surplus cooling energy. The cooling tower supports heat rejection as needed, while the AWHPs remain largely inactive to optimize efficiency and prioritize chiller-driven cooling and TES charging.

6. Cooling only, TES tank is fully charged

In this operation mode, the system functions under cooling-only conditions, with the TES tank fully charged. The chiller/s operates to meet the CHW load. In addition to chillers, in some hours, one or more HRC operates in cooling mode to assist chillers in meeting the cooling load. The cooling towers are on to reject heat from the CW loop to the ambient. The AWHPs are off, and the TES tank remains idle, not charging or discharging, since the system's thermal storage capacity is already at maximum.

As mentioned above, the HRCs can be either on or off, depending on the cooling load. When on, the HRCs operate as chillers, using their evaporators to extract heat from CHW loop and using their condensers to reject heat to the CW loop. The CW circulates between the chiller and the heat exchanger that connects to the cooling tower, with heat rejected to the ambient air.

7. Cooling larger than heating (cooling > heating), TES tank is fully charged

In this operation mode, the system operates under conditions where cooling demand exceeds heating demand, and the TES tank is fully charged. The chillers are active to meet the CHW load. HRCs could be in cooling or heat recovery modes depending on CHW and HW loads. One or more cooling tower operates in all hours to reject excess heat from the CW loop to the ambient air. The AWHP remains off, and the TES tank is neither charging nor discharging, as it is already at full capacity.

Overall, this mode represents a cooling-dominant mixed condition, where the chiller and cooling tower manage the primary cooling load, and limited heating is supplied by HRC. The TES tank remains idle due to its fully charged state, ensuring operational stability and allowing the system to maintain comfort conditions efficiently without engaging additional heating or storage components.

Hot Water TIER Plant: Description of System Operating Modes

The HW TIER system has many of the same components as its CW equivalent, but they are arranged differently to allow heat recovery directly from the chilled water loop to the hot water loop. Chillers are piped in series, with the HRC in a side stream configuration as the lead cooling machine, and cooling-only centrifugal chillers downstream of the HRC. The HRC has its own dedicated pumps on the CHW and HW side stream loops, to maintain flow as required by the part load. The HW TES tank is piped directly into the HW loop, and the charging / discharging state is controlled by the relative speeds of the two sets of HW pumps. The AWHPs are piped directly to the HW loop in series and downstream of the HRC, as opposed to the CW model where the heat pumps are piped into the CW loop. Cooling towers serve the centrifugal chillers in parallel, as well as an additional heat exchanger with the HW loop. This heat exchanger allows the HRC to function as a cooling only chiller and reject heat from the HW loop to the cooling tower loop if cooling loads demand a third chiller stage.

1. Heating only, TES tank is fully charged

In this mode, the heating load is served by the TES tank and the AWHPs, since there is no possibility of heat recovery from the CHW loop. The HRC remains off, as do the centrifugal chillers since there are no cooling loads. The HW pumps with their suction side closest to the top of the TES tank speed up faster than the other set of pumps, allowing 130°F HW to run from the top of the tank out to the building. If the tank charge falls below a predetermined threshold, the AWHPs are enabled to supplement the heating load and recharge the tank. The quantity of heat pumps enabled and part loads at which they run are determined by the BAS based on how fast the tank is discharging, and how soon it needs to be charged for the next heating peak. During a design peak heating scenario, all heat pumps may be enabled at full load to supplement the heating from the tank discharge.

2. Heating only, TES tank is not fully charged

When the tank is partially discharged, it continues discharging as heating loads require until the supplemental charging control logic is triggered. Supplemental tank charging can be triggered by a number of factors, such as rate of tank-charge change, proximity to a predicted heating peak, or falling below a fixed tank charge percentage. When the plant is in a supplemental charge state, heat pumps are enabled to refill the tank while continuing to meet the building heating loads.

3. Heating larger than cooling (heating > cooling)

When there are some cooling loads present but heating loads dominate, the HRC is enabled as the lead cooling machine and constrained by the cooling load. This means that the HRC controls to the CHW supply temperature setpoint while rejecting heat to the HW loop, but additional heating is still required. The remainder of the heating load not met with recovered heat is served by the tank, or the heat pumps if the plant is in a supplemental charge state. If the cooling loads exceed the capacity of the HRC but the heating loads are still larger, the centrifugal chillers and cooling towers run to handle the remainder of the cooling load, while the HRC continues running at full load to recover as much heat as possible.

4. Cooling only, TES tank is fully charged

When there are no building heating loads and the tank is charged, the HRC remains disabled since there are no opportunities for heat recovery. The cooling loads are served by the more efficient centrifugal chillers, with all heat rejected to the towers in the exact same way as a typical water-cooled chilled water plant.

5. Cooling only, TES tank is not fully charged

If there are no building heating loads but there are still opportunities to recover heat by charging the tank, the HRC will run as the lead cooling equipment. The HRC will be heating load constrained such that it controls to a 130°F HW supply temperature setpoint suitable for charging the tank, while cooling as much as possible on the CHW side. The HRC runs if there is excess capacity in the tank, while the cooling centrifugal chillers and cooling towers meet the remainder of the CHW load. Once the tank is full, the HRC is disabled and the centrifugal chillers meet the entirety of the cooling load.

6. Cooling larger than heating (cooling>heating), TES tank is fully charged

When cooling loads are high and coincident with heating loads, the HRC remains the first stage heating and cooling machine. If the tank is fully charged, then the HRC part load is constrained by the building heating load. The HRC controls to the HW supply temperature setpoint while recovering as much heat as possible from the CHW loop. The remaining cooling loads are served by the centrifugal chillers and cooling towers. If the heating loads exceeds the heating capacity of the HRC but are still less than the cooling loads, the balance of the heating load will be discharged from the tank.

When cooling loads are at their peak and centrifugal chillers are pushed close to design capacity, the HRC runs in cooling only mode and rejects heat through the CW/HW heat exchanger. Cooling only mode in this case means that the HRC runs at a standard lift for cooling, with leaving condenser water temperatures of ~90°F which transfer heat from the HW loop to the tower water loop. If the HRC is staged on as the third cooling chiller, the balance of building heating loads are met by the tank and the AWHPs.

7. Cooling larger than heating (cooling>heating), TES tank is not fully charged

The plant operates similarly in this mode as it did in the previous. If the tank is partially charged, it will charge and discharge based on the net balance of the recovered heat and the building heating loads. Since the HRC is running as the lead cooling machine, it will recover as much heat as possible and is only constrained by the sum of the heating load required to charge the tank and the building heating loop load. If cooling loads are high and present a source for heat recovery which exceeds the building heating load, the excess heat is used to charge the tank. If the heating loads exceed the HRC's capacity for heat recovery, then the tank is discharged to meet the remaining load in addition to the HRC's contribution.

The HRC can also enter a cooling only mode if cooling loads are high enough, and which point the entirety of the heating loads will be served by the tank, and eventually by the heat pumps when the tank runs out and supplemental charging logic is triggered within the BAS.

Appendix B

Condenser Water TIER Plant: Simplified Spreadsheet Modeling Tool Description

The spreadsheet energy analysis of the TIER plant is conducted using the following procedure:

1. Develop and run an annual performance simulation of an EnergyPlus model representing the building that the TIER plant is sized to serve with the airside economizer disabled.
2. Structure the spreadsheet as an 8,760 model and not a bin analysis.
3. Export the following parameters from the EnergyPlus model using an hourly timestep.
    a. Ambient dry bulb temperature
    b. Ambient wet bulb temperature
    c. Air handler return air dry bulb temperature
    d. Air handler mixed air dry bulb temperature
    e. Air handler mixed air wet bulb temperature
    f. Air handler supply air dry bulb temperature
    g. Air handler supply air wet bulb
    h. HW loop load
    i. HW supply temperature
    j. HW return temperature
    k. HW flow rate
    l. CHW loop load
    m. CHW loop supply temperature
    n. CHW return temperature
    o. CHW loop flow rate
4. Determine the portion of the CHW loop load that the TIER plant serves. The remaining CHW loop load is served by the airside economizers.
    a. If the TES tank has less than 95 percent of its maximum heating capacity (where 0 percent is a tank full of 42°F water and 100 percent is a tank full of 80°F water), the TIER plant will serve 100 percent of the CHW loop load, keep the airside economizers locked out, and maximize heat recovery.
    b. If the ambient dry bulb temperature exceeds 75°F or the ambient dry bulb temperature exceeds the air handler return air dry bulb temperature, the TIER plant will serve 100 percent of the CHW loop load.
    c. If the TES tank has more than 95 percent of its maximum heating capacity and the ambient dry bulb temperature is less than both 75°F and the air handler return air dry bulb temperature, the TIER plant will serve the greatest of the following calculations which subtract the CHW loop load by the airside economizer energy potential:
        i. CHW loop load × (Ambient air dry bulb temperature – Air handler supply air dry bulb temperature) ÷ (Air handler mixed air dry bulb temperature – Air handler supply air dry bulb temperature)
        ii. CHW loop load × (Ambient air enthalpy – Air handler supply air enthalpy) ÷ (Air handler mixed air enthalpy – Air handler supply air enthalpy)

5. Identify operating chillers and whether the evaporator and condenser barrels are indexed to the CHW loop, the CW loop, or the HW loop per Table 17.

Table 17: Equipment staging table.

Hot Water Load (kBtu/h)	Chilled Water Load (tons) 0	0 – 360	360 – 470	470+
0	—	CH-1 CHW-CW	CH-1 CHW-CW HRC-1 CHW-CW	CH-1 CHW-CW HRC-1 CHW-CW HRC-2 CHW-CW
0 – 2300	HRC-1 CW-HW	CH-1 CHW-CW HRC-1 CW-HW	CH-1 CHW-CW HRC-1 CW-HW HRC-2 CHW-CW	CH-1 CHW-CW HRC-1 CHW-HW HRC-2 CHW-CW
2300+	HRC-1 CW-HW HRC-2 CW-HW	CH-1 CHW-CW HRC-1 CW-HW HRC-2 CW-HW	CH-1 CHW-CW HRC-1 CW-HW HRC-2 CHW-HW	CH-1 CHW-CW HRC-1 CHW-HW HRC-2 CHW-HW

The left subscript denotes the index of the evaporator barrel and the right subscript denotes the index of the condenser barrel.

6. Distribute the building HW loop load and the plant CHW loop among operating devices per the following rules:
    a. For all chillers with their evaporators indexed to the CHW loop, distribute the plant CHW loop load proportionally based on chiller nominal capacity.
    b. For any chillers with their evaporators indexed to the CHW loop and their condensers indexed to the HW loop, add the chiller's chilled loop water load and its compressor heat output (chiller input energy) to determine the chiller's HW loop load.
    c. For any chillers with their evaporators indexed to the CW loop and their condensers indexed to the HW loop, subtract the HW loop load covered by the chillers in the preceding clause from the building HW loop load to determine the remaining HW loop load. Where there are multiple chillers operating in this mode, distribute the remaining HW loop load proportionally based on nominal chiller heating capacity.
7. Estimate the leaving CW temperature setpoint of chillers rejecting heat to the CW loop as follows:
    a. If the TES tank is currently cycled through to 60°F at the top of the tank and 42°F at the bottom, assume chillers have a leaving CW temperature setpoint of 60°F.
    b. If the TES tank is currently cycled through to 80°F at the top of the tank and 60°F at the bottom and not fully charged (less than 100 percent of its maximum heating capacity), assume chillers have a leaving CW temperature setpoint of 80°F.
    c. If the TES tank is fully charged (100 percent of its maximum heating capacity), assume chillers have a leaving CW temperature setpoint that resets from a minimum of CHW loop

supply temperature plus 18°F (minimum chiller lift) at 78 tons of plant CHW load to a maximum of the CHW loop supply temperature plus 51°F (design chiller lift) at 780 tons of plant CHW load.

8. Equate the leaving evaporator water temperature setpoint of chillers with evaporators indexed to the CHW loop to the CHW loop supply temperature.
9. Determine the leaving HW temperature setpoint of chillers with condensers indexed to the HW loop as follows:
    a. If only HRC-1 is operating with a condenser indexed to the HW loop, the leaving HW temperature setpoint is the HW loop supply temperature.
    b. If both HRC-1 and HRC-2 are operating with condensers indexed to the HW loop (HRC-1 and HRC-2 condensers are in series), the upstream condenser's (HRC-2) leaving HW temperature setpoint equals (HW loop supply temperature plus HW loop return temperature) ÷ 2 and the downstream chiller's (HRC-1) leaving HW temperature setpoint equals the HW loop supply temperature.
10. Determine the leaving evaporator water temperature setpoint of chillers with evaporators indexed to the CW loop as follows:
    a. If the TES tank is currently cycled through to 60°F at the top of the tank and 42°F at the bottom, all chiller leaving evaporator water temperature setpoint equals al 42°F.
    b. If the TES tank is currently cycled through to 80°F at the top of the tank and 60°F at the bottom, all chiller leaving evaporator water temperature setpoint equals 60°F.
11. Calculate power draw of chillers with condensers indexed to the CW loop given: current CHW load per chiller (see 6.a), leaving CW temperature and leaving evaporator water temperature (see 7 and 8), full load chiller efficiency, the EnergyPlus Chiller:Electric:ReformulatedEIR model, and the following chiller performance curves normalized to the full load performance of each chiller:
    a. ChillerCapFTemp
       = a
       + b × (Leaving evaporator water temperature)
       + c × (Leaving evaporator water temperature)^2
       + d × (Leaving CW temperature)
       + e × (Leaving CW temperature)^2
       + f × (Leaving evaporator water temperature) × (Leaving CW temperature)
        i. a = 1.033403
        ii. b = 0.03515774
        iii. c = 0.0004288264
        iv. d = -0.005737406
        v. e = -0.00002647149
        vi. f = -0.0002222927
    b. ChillerEIRFTemp
       = a
       + b × (Leaving evaporator water temperature)
       + c × (Leaving evaporator water temperature)^2
       + d × (Leaving CW temperature)
       + e × (Leaving CW temperature)^2
       + f × (Leaving evaporator water temperature) × (Leaving CW temperature)
        i. a = 0.4008316

ii. b = -0.006141937
iii. c = 0.00081138
iv. d = 0.005982224
v. e = 0.0005345503
vi. f = -0.001142919

c. ChillerEIRFPLR
= a
+ b × (Leaving CW temperature)
+ c × (Leaving CW temperature)^2
+ d × (Part load ratio)
+ e × (Part load ratio)^2
+ f × (Leaving CW temperature) × (Part load ratio)
+ g × (Part load ratio)^3
    i. a = 0.551946
    ii. b = -0.01406889
    iii. c = 0.000003843199
    iv. d = 0.07950267
    v. e = 0.8798848
    vi. f = 0.0137948
    vii. g = -0.5070156

d. Chiller power draw = (Chiller nominal capacity) × (ChillerCapFTemp at current conditions) ÷ (ChillerCapFTemp at design conditions) × (Chiller nominal electric input ratio) × (ChillerEIRFTemp at current conditions) × (ChillerEIRFPLR at current conditions) ÷ (ChillerEIRFTemp at design conditions) ÷ (ChillerEIRFPLR at design conditions)

12. Calculate power draw of chillers with condensers indexed to the HW loop draw as follows:
    a. For chillers with evaporators indexed to the CHW loop and condensers indexed to the HW loop, follow the same procedure as in 11, albeit use leaving HW temperature (see 9) as the leaving CW temperature in the chiller performance curves.
    b. For chillers with evaporators indexed to the CW loop, calculations are complicated by the fact that chiller models take evaporator load as an input to calculate chiller power. In this case, we instead know evaporator barrel load per the procedure in 6.c, so the process is iterative as follows:
        i. Guess that the initial chiller heating COPh equals 4.0.
        ii. Using the HW loop load served by the chiller (see 6.c) and initial COPh, estimate evaporator load as (Condenser load) – (Condenser load ÷ Initial COPh).
        iii. Use the chiller model from 11 to estimate chiller power draw based on CHWST, HWST, full load chiller efficiency, evaporator part load ratio, and the Energy Plus chiller curves.
        iv. Calculate iterated COPh as (Evaporator load + Chiller power draw) ÷ Chiller power draw.
        v. Repeat 12.b.ii and 12.b.iii using iterated COPh as the new initial COPh until chiller power draw converges to within a 0.1 percent.

13. Calculate excess HW loop load that needs to be transferred to the CW loop as the condenser load from chillers with evaporators indexed to the CW loop minus building HW loop load. If this value is greater than zero, this heat is transferred to the condenser loop.

a. In rare occasions when all chillers have evaporators indexed to the CHW loop (near cooling design condition) but there is still a small amount of heating load for some chillers with condensers to be indexed to the HW loop, the amount of heat rejected to the HW loop may exceed the HW loop load. In these cases, that heat gets transferred to the CW loop by bleeding CW into the HW loop.

14. Calculate the net heat added to or removed from the CW loop without supplemental heat as follows:
    a. Heat gain from chillers with condensers indexed to the CW loop.
    b. Plus excess heat from the HW loop (see 13.a).
    c. Minus heat extracted by chillers with evaporators indexed to the CW loop.
    d. Plus net gain or removal from any other building loads connected to the CW loop.
15. If heat is added to the CW loop, determine whether that heat should be added to the TES tank or rejected via cooling towers draw as follows:
    a. If the TES tank is not fully charged (less than 100 percent of its maximum heating capacity), and the hourly net heat added to or removed from the CW loop without supplemental heat is less than the remaining available storage capacity in the tank, then the hourly net heat added to or removed from the CW loop without supplemental heat is added to or removed from the TES tank.
    b. If the TES tank is not fully charged (less than 100 percent of its maximum heating capacity), but the hourly net heat added to the CW loop without supplemental heat exceeds the remaining available storage capacity in the tank, then the TES tank is fully charged and the remaining hourly net heat added to or removed from the CW loop without supplemental heat is rejected through the cooling towers.
    c. If the TES tank is fully charged (100 percent of its maximum heating capacity), then all net heat added to or removed from the CW loop without supplemental heat is rejected through the cooling towers.
16. Determine whether the AWHPs operate during a given hour draw as follows:
    a. Both AWHPs operate at full load if any of the following conditions is true:
    b. Net heat removal from the CW loop without supplemental heat exceeds:
        i. 20 percent of TES tank maximum heating capacity during the previous hour,
        ii. 15 percent of TES tank maximum heating capacity in each of the previous 2 consecutive hours, or
        iii. 10 percent of TES tank maximum heating capacity in each of the previous 3 consecutive hours.
    c. Or the current TES tank heating capacity plus 6 percent for each hour between the current hour and 6AM is less than the TES tank maximum heating capacity and the current TES tank heating capacity is less than 97 percent of TES tank maximum capacity.
    d. Otherwise, the AWHPs do not operate.
17. Determine the AWHP supply temperature setpoint draw as follows:
    a. If the TES tank is currently cycled through to 60°F at the top of the tank and 42°F at the bottom, AWHP supply temperature setpoint equals 77°F (the minimum leaving water temperature allowed by the AWHP manufacturer).
    b. If the TES tank is currently cycled through to 80°F at the top of the tank and 60°F at the bottom, AWHP supply temperature setpoint equals 84°F.

18. Calculate AWHP capacity output and power draw as follows:
    a. AWHP capacity is primarily a function of ambient dry bulb temperature and supply water temperature. Power draw is a function of the same variables and load. Since the AWHPs only operate at full load when enabled, capacity and power draw calculations can ignore part load ratio and just reference and ambient dry bulb temperature supply water temperature as inputs.
    b. Use a lookup table from the manufacturer with ambient dry bulb temperature and supply water temperature as inputs and capacity and power draw as outputs, to perform linear interpolations for each hour when the AWHPs are operating.
19. Calculate the net heat added to or removed from the TES tank using the AWHPs and adjust AWHP power draw as follows:
    a. Net heat added to or removed from the TES tank equals the net heat gain to the CW loop without any supplemental heat (per 14) less heat rejected through the cooling towers (per 15) plus heat added by the AWHPs (per 18).
    b. If both AWHPs do not need to run the full hour to finish charging the tank, multiply the AWHP capacity output and power draw for that hour by the fraction of the hour that the AWHPs need to run to finish charging the tank.
20. Calculate the flow rate and power draw of the CW pumps serving the chiller condenser barrel loop as follows:
    a. If the TES tank is currently cycled through to 60°F at the top of the tank and 42°F at the bottom:
        i. Apply an 18°F delta-T to the total condenser load from the chillers with condensers indexed to the CW loop to determine flow rate. However, if the AWHPs are enabled, apply an 18°F delta-T to the maximum possible AWHP heating capacity according to the lookup table developed in step 18.b to determine a minimum flow rate limit.
    b. If the TES tank is currently cycled through to 80°F at the top of the tank and 60°F at the bottom and cooling towers are not operating during the hour:
        i. Apply a 20°F delta-T to the total condenser load from chillers with condensers indexed to the CW loop to determine flow rate. However, if the AWHPs are enabled, apply a 20°F delta-T to the maximum possible AWHP heating capacity according to the lookup table developed in step 18.b to determine a minimum flow rate limit.
    c. If the TES tank is currently cycled through to 80°F at the top of the tank and 60°F at the bottom and cooling towers are operating during the hour:
        i. Each enabled chiller with its condenser indexed to the condenser loop shall operate at design CW flow.
        ii. Additionally, add flow for any excess heat rejected from the HW loop to the CW loop (see 13.a). Assume this heat is rejected with a 48°F delta-T for the purposes of calculating flow rate. This is inherently conservative since delta-T will be even higher than HW supply temperature less design entering CW temperature during most hours.
    d. Calculate power draw assuming pump head varies as $(\text{flow rate})^{1.8}$, pump efficiency, NEMA premium motor efficiency, and 98% VFD efficiency.

21. Calculate the flow rate and power draw of the CW pumps serving the chiller evaporator barrel loop as follows:
    a. If the TES tank is currently cycled through to 60°F at the top of the tank and 42°F at the bottom:
        i. Apply an 18°F delta-T to total evaporator load from the chillers with evaporators indexed to the CW loop to determine flow rate.
    b. If the TES tank is currently cycled through to 80°F at the top of the tank and 60°F at the bottom and cooling towers are not operating during the hour:
        i. Apply a 20°F delta-T to current evaporator load from the chillers with evaporators indexed to the CW loop to determine flow rate.
    c. In neither scenario can flow rate be less than 50% of the lowest heat recovery chiller's design evaporator flow rate.
    d. Calculate power draw assuming head varies as $(\text{flow rate})^{1.4}$, pump efficiency, NEMA premium motor efficiency, and 98% VFD efficiency.
22. Calculate the flow rate and power draw of the tower water pumps as follows:
    a. Tower water pump flow rate equals the flow rate of the CW pumps serving the chiller condenser barrel loop (see 20) when the CW pumps serving the chiller condenser barrel loop are enabled but shall be no less than 30% of design cooling tower flow.
    b. Calculate power draw using cooling tower static head and assuming with the remainder of design head varies as $(\text{flow rate})^{1.4}$, pump efficiency, NEMA premium motor efficiency, and 98% VFD efficiency.
23. Calculate cooling tower temperatures and power draw as follows:
    a. When the CW pumps serving the chiller condenser barrel loop are on, cooling tower leaving temperature equals the leaving CW temperature setpoint of chillers rejecting heat to the CW loop (see 7.b), minus the delta-T resulting from the tower heat rejection load applied to the current flow rate of the CW pumps serving the chiller condenser barrel loop, minus 2°F.
        i. This assumes a fixed heat exchanger approach of 2°F from the open tower water loop to the closed CW loop, which is conservative. In practice, approach decreases as flow rate decreases, but those dynamics are not worth modeling.
    b. Cooling tower entering temperature is calculated as cooling tower leaving temperature plus the delta-T resulting from the tower heat rejection load applied to the current tower water pumps flow rate.
    c. Using tower water pump flow rate from 22.a, cooling tower entering and leaving water temperatures, and ambient wet bulb temperature, calculate cooling tower fan power draw using the following CoolTools empirical model assuming both tower cells always run:
        i. Calculate cooling tower approach temperature at 10% air flow rate ratio, 50% air flow rate ratio, and 100% air flow rate ratio assuming that air flow rate ratio equals cooling tower fan speed:

$$
\begin{aligned}
&= \text{Coeff}(1) \\
&+ \text{Coeff}(2) \times (\text{Air flow rate ratio}) \\
&+ \text{Coeff}(3) \times (\text{Air flow rate ratio})^2 \\
&+ \text{Coeff}(4) \times (\text{Air flow rate ratio})^3 \\
&+ \text{Coeff}(5) \times (\text{Water flow rate ratio})
\end{aligned}
$$

+ Coeff(6) × (Air flow rate ratio) × (Water flow rate ratio)
+ Coeff(7) × (Air flow rate ratio)^2 × (Water flow rate ratio)
+ Coeff(8) × (Water flow rate ratio)^2
+ Coeff(9) × (Air flow rate ratio) × (Water flow rate ratio)^2
+ Coeff(10) × (Water flow rate ratio)^3
+ Coeff(11) × (Ambient wet bulb temperature)
+ Coeff(12) × (Air flow rate ratio) × (Ambient wet bulb temperature)
+ Coeff(13) × (Air flow rate ratio)^2 × (Ambient wet bulb temperature)
+ Coeff(14) × (Water flow rate ratio) × (Ambient wet bulb temperature)
+ Coeff(15) × (Air flow rate ratio) × (Water flow rate ratio) × (Ambient wet bulb temperature)
+ Coeff(16) × (Water flow rate ratio)^2 × (Ambient wet bulb temperature)
+ Coeff(17) × (Ambient wet bulb temperature)^2
+ Coeff(18) × (Air flow rate ratio) × (Ambient wet bulb temperature)^2
+ Coeff(19) × (Water flow rate ratio) × (Ambient wet bulb temperature)^2
+ Coeff(20) × (Ambient wet bulb temperature)^3
+ Coeff(21) × (Tower range temperature)
+ Coeff(22) × (Air flow rate ratio) × (Tower range temperature)
+ Coeff(23) × (Air flow rate ratio)^2 × (Tower range temperature)
+ Coeff(24) × (Water flow rate ratio) × (Tower range temperature)
+ Coeff(25) × (Air flow rate ratio) × (Water flow rate ratio) × (Tower range temperature)
+ Coeff(26) × (Water flow rate ratio)^2 × (Tower range temperature)
+ Coeff(27) × (Ambient wet bulb temperature) × (Tower range temperature)
+ Coeff(28) × (Air flow rate ratio) × (Ambient wet bulb temperature) × (Tower range temperature)
+ Coeff(29) × (Water flow rate ratio) × (Ambient wet bulb temperature) × (Tower range temperature)
+ Coeff(30) × (Ambient wet bulb temperature)^2 × (Tower range temperature)
+ Coeff(31) × (Tower range temperature)^2
+ Coeff(32) × (Air flow rate ratio) × (Tower range temperature)^2
+ Coeff(33) × (Water flow rate ratio) × (Tower range temperature)^2
+ Coeff(34) × (Ambient wet bulb temperature) × (Tower range temperature)^2
+ Coeff(35) × (Tower range temperature)^3

1. Coeff(1) = 0.52049709836241
2. Coeff(2) = -10.6170464
3. Coeff(3) = 10.72929747
4. Coeff(4) = -2.749883772
5. Coeff(5) = 4.736299439
6. Coeff(6) = -8.257597009
7. Coeff(7) = 1.576409381
8. Coeff(8) = 6.511196438
9. Coeff(9) = 1.504335252
10. Coeff(10) = -3.288852929
11. Coeff(11) = 0.025778615

12. Coeff(12) = 0.182464289
13. Coeff(13) = -0.081894729
14. Coeff(14) = -0.215010004
15. Coeff(15) = 0.018674131
16. Coeff(16) = 0.053682418
17. Coeff(17) = -0.00270969
18. Coeff(18) = 0.001122775
19. Coeff(19) = -0.001277585
20. Coeff(20) = 0.0000760421
21. Coeff(21) = 1.436000883
22. Coeff(22) = -0.519869591
23. Coeff(23) = 0.117339577
24. Coeff(24) = 1.504928108
25. Coeff(25) = -0.135898906
26. Coeff(26) = -0.152577582
27. Coeff(27) = -0.053384383
28. Coeff(28) = 0.004932949
29. Coeff(29) = -0.007962604
30. Coeff(30) = 0.00022262
31. Coeff(31) = -0.0543952
32. Coeff(32) = 0.004742669
33. Coeff(33) = -0.018585467
34. Coeff(34) = 0.001156677
35. Coeff(35) = 0.000807371

ii. Determine cooling tower fan speed by interpolating between cooling tower approach temperatures at different air flow rate ratios using required cooling tower approach temperature calculated as the cooling tower leaving water temperature minus the ambient wet bulb temperature.

iii. Calculate power draw assuming design brake horsepower varies as (fan speed)$^{2.9}$, NEMA premium motor efficiency, and 98% VFD efficiency.

24. Calculate HWP power draw as follows:
    a. Calculate power draw using HW flow rate from EnergyPlus assuming head varies as (flow rate)$^{1.4}$, pump efficiency, NEMA premium motor efficiency, and 98% VFD efficiency.
25. Calculate CHWP power draw as follows:
    a. Calculate power draw using CHW flow from EnergyPlus, scaled linearly per the adjusted CHW loop load from 4 assuming head varies as (flow rate)$^{1.4}$, pump efficiency, NEMA premium motor efficiency, and 98% VFD efficiency.
26. Sum power draw from all end uses for the hour.

Hot Water TIER Plant: Simplified Spreadsheet Modeling Tool Description

This section divides the methodology of the spreadsheet model into several functional groups. In developing the model, it may be helpful to group calculations and functions in the same way.

Energy Plus Data

Structure the spreadsheet as an 8,760 model, not a bin analysis, and export the following parameters from EnergyPlus/CBECC-Com on an 8,760 basis:

1. Ambient Dry Bulb
2. Ambient Wet Bulb
3. CHW load
4. CHW load with the airside economizer disabled
5. CHW supply temperature
6. CHW return temperature
7. CHW flow
8. HW load
9. HW supply temperature
10. HW return temperature
11. HW flow

System Inputs (Sheet 1)

The spreadsheet model shall include all of the following inputs on a single System Inputs sheet:

1. HW System Variables:
    a. Design building loop HWST Setpoint (°F)
    b. Design Primary Loop Flow (GPM)
    c. Design Primary Loop Differential Pressure Drop (ft)
    d. Design Building Loop Flow (GPM)
    e. Design Building Loop Differential Pressure Drop (ft)
    f. Design Building loop DP Setpoint (psi)
2. CHW System Variables:
    a. Design CHWST Setpoint (°F)
    b. Design CHW Loop Flow (GPM)
    c. Design CHW Loop Differential Pressure Drop (ft)
    d. Design CHW Loop DP Setpoint (psi)
3. Thermal Energy Storage (TES) Tank Variables:
    a. TES Tank Height (ft)
    b. TES Tank Diameter (ft)
    c. TES Tank estimated thermocline height (ft)
    d. TES Tank estimated top and bottom heel heights (ft)
4. Air to Water Heat Pump (AWHP) Inputs:
    a. AWHP capacity at worst case conditions (kBTU/hr)
    b. AWHP staging capacity
    c. AWHP minimum turndown ratio (%)
    d. AWHP module count (#)
    e. AWHP module minimum flow (GPM)

f. AWHP module maximum flow (GPM)
g. Design AWHP water pressure drop and a corresponding flow point (ft)
h. EnergyPlus regression model curves (x2):
    i. Module capacity as a function of HWST and ambient dry bulb
    ii. Electrical demand as a function of HWST, ambient dry bulb, and part load ratio

5. Heat Recovery Chiller (HRC) Inputs:
    a. HRC minimum turndown ratio (%)
    b. HRC module minimum HW flow (GPM)
    c. Side stream HW pump design flow (GPM)
    d. HRC module minimum CHW flow (GPM)
    e. Side stream CHW pump design flow (GPM)
    f. Design CHW barrel pressure drop and a corresponding flow point (ft)
    g. Design HW barrel pressure drop and a corresponding flow point (ft)
    h. EnergyPlus regression model curves (x4):
        i. Module heating capacity as a function of HWST and CHWST
        ii. Electrical demand as a function of HWST, CHWST, and the heating part load ratio
        iii. Module cooling capacity as a function of CHWST and HWST
        iv. Electrical demand as a function of CHWST, HWST, and cooling part load ratio
6. Cooling Only Chiller Inputs:
    a. Chiller capacity at worst case conditions (tons)
    b. Chiller staging capacity percentage (%)
    c. Chiller minimum turndown ratio (%)
    d. Design chiller module CW flow (GPM)
    e. Condenser barrel pressure drop at design CW flow (ft)
    f. Chiller module minimum CHW flow (GPM)
    g. Design CHW barrel pressure drop and a corresponding flow point (ft)
    h. EnergyPlus regression model curves (x2):
        i. Chiller cooling capacity as a function of CHWST and CWRT
        ii. Electrical demand as a function of CHWST, CWRT, and the cooling part load ratio
7. Cooling Tower / CW Loop Inputs:
    a. HW / CW heat exchanger approach (°F)
    b. Tower minimum flow (for both cells running at once) (GPM)
    c. Cooling Tower Static Lift (ft)
    d. Design CW Loop Differential Pressure Drop (ft)
    e. EnergyPlus regression model for cooling tower fan power using the CoolTools empirical model covered in the EnergyPlus Engineering Reference

Plant Staging (Sheet 1 + Sheet 2)

1. Hourly TES tank charge tracking:
    a. Based on the usable volume of the tank, heel height, and thermocline height, an hourly tally of volume above and below the thermocline shall be tracked. The variable TankCharge shall be set equal to usable volume above the thermocline divided by total usable volume.
        i. Tank charge shall also be expressed as a separate variable in energy units, kBTU

ii. Energy required to charge the tank shall be expressed as a separate variable in units if kBTU, based on the total tank volume and current tank charge.

b. The temperature above the thermocline, TankTempAboveTC, shall be assumed equal to the Design HWST Setpoint.

c. The temperature below the thermocline, TankTempBelowTC, shall be a volume weighted average of the existing tank temperature below the thermocline and the temperature of flow entering the bottom of the tank during the hour. E.g., if the volume below the thermocline at the beginning of hour 5 is 4000 gallons at a temperature of 100°F, and 1000 gallons enters the bottom of the tank during hour 5 at 105°F, the temperature below the thermocline at the end of hour 5 shall equal 101°F.

d. Initialize the model, i.e., set conditions prior to the first hour, assuming HW storage is fully charged.

2. Determine HRC cooling output based on cooling load:
   a. Current hour HRStagingCapacity shall be calculated with the following inputs to the regression:
      i. Use the current hour instantaneous CHW supply temperature from the E+ outputs for the evaporator leaving temperature.
      ii. Use the design HW supply temperature setpoint for the condenser leaving temperature.
   b. Determine whether the air side economizer shall be throttled this hour to maximize heat recovery:
      i. If the tank is less than 80% (user adjustable) charged, set HRCoolingOutput1 equal to the lesser of instantaneous cooling load with economizer <u>disabled</u> and HRStagingCapacity. In this condition the economizer is being throttled.
      ii. Otherwise, set HRCoolingOutput1 equal to the lesser of instantaneous cooling load with economizer <u>enabled</u> and HRStagingCapacity. In this condition the economizer is <u>not</u> being throttled.
      iii. Indicate economizer throttling status in a separate TRUE of FALSE column. Wherever cooling load is referenced later in the model, ensure that the correct cooling load is referenced based on the throttling condition.
3. Determine HRC heating output:
   a. Use the values of HRCoolingOutput1 and HRStagingCapacity to determine CoolingPLR, the cooling part load ratio.
   b. Use the CoolingPLR for the current hour to determine the HRCdemand1 using the electrical demand regression for the heat recovery chiller along with the following inputs:
      i. Use the current hour instantaneous CHW supply temperature from the E+ outputs for the evaporator leaving temperature.
      ii. Use the design HW supply temperature setpoint for the condenser leaving temperature.
      iii. If the part load ratio (PLR) is less than the minimum turndown ratio for this equipment, calculate the demand at the minimum turndown ratio, then multiply the output by the actual PLR divided by the minimum turndown. Power * (CoolingPLR/min turndown). This effectively models the equipment cycling throughout the hour if the required PLR is unrealistically low. This same logic

applies to all calculations where PLR is used in a regression curve to calculate power, for the HRC, AWHPs, and chillers.

   c. Determine HRHeatingOutput1 as the sum of HRCoolingOuput1 and the HRCdemand1 for the current hour.

4. Determine whether the HRC needs to run in cooling only mode.
   a. Determine the maximum potential heating load achievable through heat recovery for the current hour as the lesser of:
      i. (instantaneous heating load + energy required to charge the tank)
      ii. Nominal HRC heating capacity for the current hour
   b. Convert the heating load from the previous step to cooling load by multiplying the value (COPh-1)/COPh, where COPh is a nominal heating COP.
   c. Subtract the calculated cooling load from the preceding step from the hourly building cooling load. If the resulting value is >90% of the nominal capacity of both cooling only chillers, then the HRC should be enabled in cooling only mode. Otherwise, the HRC shall run in heat recovery mode.
5. Determine the Plant Heating Loads and Tank Charge Rate:
   a. The heating plant load does not equal the instantaneous heating load because the behavior of TES tank must be considered.
   b. Based on a set of user definable rules, determine whether the HW storage requires charging with supplemental—i.e., non-recovered—heat during the hour. If yes, the logic variable TankChargeReq will be set to true and false otherwise. Default triggers will include the following; others may be added as modeling progresses:
      i. TankCharge < 30% and rate of charge change of TankCharge during previous hour was < -10%.
      ii. TankCharge < 50% and rate of charge change of TankCharge during previous hour was < -20%.
      iii. TankCharge < 90% and the ambient dry bulb of the current hour is higher than the average temperature over the previous 24 hours, indicating that conditions are more favorable for AWHP heating
      iv. TankCharge < 50% and the occupied start time for the building is less than X (default 3) hours from the current hour.
   c. If TankChargeReq is True:
      i. Plant HWST setpoint is equal to design HWST
      ii. Set the variable HWPlantCapacity equal to HRHeatingOutput1 + AWHP-1 Capacity + AWHP-2 Capacity
         1. AWHP capacities for this step shall be determined based on the plant HWST setpoint and the current hour ambient dry bulb.
      iii. Compare HWPlantCapacity with the energy required to fully charge the tank in one hour.
         1. If HWPlantCapacity > energy required to fully charge the tank and serve the building load, set HWPlantLoad equal to (energy required to fully charge the tank) + instantaneous building heating load. TankChargeRate is set equal to the energy required to fully charge the tank within the hour.

        2. If HWPlantCapacity < energy required to fully charge the tank and serve the building load, set HWPlantLoad equal to HWPlantCapacity. TankChargeRate becomes HWPlantCapacity – instantaneous building heating load
  d. If TankChargeReq is False:
    i. Set TankChargeRate equal to HRHeatingOutput1 – instantaneous building heating load subject to the following constraints:
        1. If TankChargeRate is positive, plant HWST shall be set to design HWST
        2. If TankChargeRate is negative, plant HWST is equal to loop HWST from E+ outputs
        3. TankChargeRate may not exceed the energy required to charge the tank within the hour. If the above calculation exceeds this upper bound:
            a. Set TankChargeRate equal to the energy required to fully charge the tank within the hour.
            b. Re-evaluate the HRC heating output such that HRHeatingOutput is equal to (energy required to fully charge tank) + (instantaneous building heating load). This heating output variable should be stored in a separate column.
            c. Calculate HeatingStagingCapacity using the heating capacity regression with current hour CHWST and plant HWST as determined in the previous steps.
            d. Calculate HeatingPLR
            e. Using the regression, calculate HRCdemand as a function of HeatingPLR, CHWST and plant HWST
            f. Calculate HRCoolingOutput as a HRHeatingOutput – HRCdemand.
        4. Otherwise, if TankChargeRate does not exceed the upper bound described in step 5.d.i.3 above, set HRCoolingOutput = HRCoolingOutput1, and HRHeatingOutput = HRHeatingOutput1.
        5. The absolute value of TankChargeRate when discharging must be < the amount of energy stored in the tank
    ii. If TankChargeRate is negative, set the TankChargeStatus indicator to "Discharging." If it is positive, set TankChargeStatus to "Charging."
    iii. Set HWPlantLoad equal to instantaneous building heating load + TankChargeRate.

6. Calculate TES tank flow:
  a. If the tank is charging:
    i. Use TankChargeRate from the previous step, TankTempAboveTC, and TankTempBelowTC to calculate TESFlow, the tank charge / discharge flow. The flow should be positive when charging.
  b. If the tank is discharging:
    i. Use TankChargeRate from the previous step, TankTempAboveTC, and building HW return temperature to calculate TESFlow. The flow should be negative when discharging.

7. Equipment Staging Table:

a. It may be helpful to group this staging table with the system inputs if it comfortably fits on a single sheet. This table identifies which equipment will be enabled based on the HW and CHW loads of each hour. The heat recovery chiller can run in heat recovery mode or cooling only mode, denoted by the subscripts HR and CO respectively. Some areas of the staging table are less likely to occur in practice since they would require large simultaneous loads, but may occur in the energy modeling process:

Table 18: Equipment staging table.

	CStage /Cooling load not met by HRC (tons)				
		1 / 0	2 / 0-250	3 / 250-575	4 / 575 +
Heating Stage/Heating Load not met by HRC (kBtu/hr)	1 / 0	HRC_{HR}	HRC_{HR} CH-1	HRC_{HR} CH-1 CH-2	HRC_{CO} CH-1 CH-2 AWHP-1
	2 / 0-2500	HRC_{HR} AWHP-1	HRC_{HR} CH-1 AWHP-1	HRC_{HR} CH-1 CH-2 AWHP-1 *edge case	—
	3 / 2500 +	HRC_{HR} AWHP-1 AWHP-2	HRC_{HR} CH-1 AWHP-1 AWHP-2 *edge case	—	—
	4	—	—	—	—

8. Determine Cooling Plant Load:
    a. The cooling plant load is straightforward for this model, since the instantaneous cooling load is equal to CHWPlantLoad.
9. Determine Primary Loop Plant Flows:
    a. The primary CHW loop flow is equal to the instantaneous CHW flow from the E+ outputs subject to the following constraints:
        i. PrimaryCHWFlow >= sum of minimum flows for # of enabled cooling only chillers
    b. The primary HW loop flow, PrimaryHWFlow, is equal to instantaneous HW flow from the E+ outputs plus TESFlow.
10. Determine Heat Recovery Chiller Flows:
    a. The CHW flow through the HRC evaporator, FlowHRCc is equal to the primary CHW loop flow subject to 2 constraints:

        i. FlowHRCc >= minimum HRC CHW flow
        ii. FlowHRCc <= design CHW side stream pump flow
    b. The HW flow through the HRC condenser FlowHRCh is equal to the primary HW loop flow subject to 2 constraints:
        i. FlowHRCh >= minimum HRC HW flow
        ii. FlowHRCh <= design HW side stream pump flow

11. Determine HRC constraints (HRC Constraints only apply in heat recovery mode):
    a. If HRHeatingOutput >= HWPlantLoad, then the HRC is HeatingConstrained in heat recovery mode.
    b. If HRHeatingOutput < HWPlantLoad, and HRStagingCapacity > CHWPlantLoad, then the HRC is CoolingConstrained in heat recovery mode.
    c. If HRHeatingOutput < HWPlantLoad, and HRStagingCapacity <= CHWPlantLoad, then the HRC is QuantityConstrained. This scenario means that we are running the HRC at design cooling capacity and getting as much recovered heat as possible while still operating in heat recovery mode.
        i. Indicate the type of constraint in a separate variable column.
12. Assign Load to the Heat Recovery Unit depending on the constraint defined in the previous step. This section only applies when the unit is in heat recovery mode.
    a. When HeatingConstrained, set the HW load equal to the HWPlantLoad. The following steps outline an iterative calculation method.
        i. Begin the iteration with the assumption that the chilled and HW loads on the HRC are equal to HRCoolingOutput and HRHeatingOutput.
        ii. Calculate CHWST1 using CHW load, and the adjust flow and return temperatures used in the calculation based on the following:
            1. If FlowHRCc > PrimaryCHWFlow, use the primary CHW flow, and primary CHW return temperature in the calculation.
            2. If FlowHRCc <= PrimaryCHWFlow, use FlowHRCc and primary CHW return temperature in the calculation.
        iii. Calculate heating capacity using the heating capacity regression curve based on CHWST1 and HWST setpoint.
        iv. Calculate heating PLR based on HW load and heating capacity from the previous regression.
        v. Use heating PLR, HWST setpoint, and CHWST1 to calculate demand.
        vi. Subtract demand from HW load to get CHW load.
        vii. Calculate CHWST2 using CHW load, and the adjust flow and return temperatures used in the calculation based on the following:
            1. If FlowHRCc > PrimaryCHWFlow, use the primary CHW flow, and primary CHW return temperature in the calculation.
            2. If FlowHRCc <= PrimaryCHWFlow, use FlowHRCc and primary CHW return temperature in the calculation.
        viii. Calculate heating capacity using the heating capacity regression curve, HWST setpoint, and CHWST2
        ix. Calculate heating PLR based on HW load and heating capacity from the previous regression.

x. Use heating PLR, HWST setpoint, and the assumed CHWST2 to calculate demand.
xi. Subtract demand from HW load to get final CHW load.
xii. This final CHW load may be applied to the HRC in the model, or this iteration may be repeated as many times as needed by calculating CHWST3, CHWST4, etc. if the agreement between CHW load results of each iteration is not yet sufficient.

b. When CoolingConstrained, set the CHW load equal to HRCoolingOutput, then step through the following iterative calculation.
    i. Begin the iteration with the assumption that the chilled and HW loads on the HRC are equal to HRCoolingOutput and HRHeatingOutput.
    ii. Calculate HWST1 using the HW load, and adjust the flow and return temperature values used in the calculation based on the following:
        1. Return Temperature (After Blending with the TES Tank):
            a. When the TES tank is charging, the HW return temperature is determined by an energy balance using the building HWRT, building flow, TESFlow, and TankTempBelowTC.
            b. When the TES tank is discharging or neutral, the HW return temperature in this calculation is equal to the building HWRT.
        2. Flow:
            a. If FlowHRCh > PrimaryHWFlow, use the primary HW flow in the calculation.
            b. If FlowHRCh <= PrimaryHWFlow, use FlowHRCh in the calculation.
    iii. Calculate cooling capacity using the regression curve as a function of the active CHWST setpoint and HWST1.
    iv. Calculate the cooling PLR as HRCoolingOutput divided by the capacity from the previous regression.
    v. Use cooling PLR, CHWST setpoint, and HWST1 to calculate demand.
    vi. Add the calculated demand to the CHW load to get the HW load.
    vii. Calculate HWST2 using the HW load, and adjust the flow and return temperature values used in the calculation based on the following:
        1. Return Temperature (After Blending with the TES Tank):
            a. When the TES tank is charging, the HW return temperature is determined by an energy balance using the building HWRT, building flow, TESFlow, and TankTempBelowTC.
            b. When the TES tank is discharging or neutral, the HW return temperature in this calculation is equal to the building HWRT.
        2. Flow:
            a. If FlowHRCh > PrimaryHWFlow, use the primary HW flow in the calculation.
            b. If FlowHRCh <= PrimaryHWFlow, use FlowHRCh in the calculation.
    viii. Calculate cooling capacity using the regression curve as a function of the active CHWST setpoint and the calculated value, HWST2.
    ix. Calculate the cooling PLR as HRCoolingOutput divided by the capacity from the previous regression using HWST2.
    x. Use cooling PLR, CHWST setpoint, and HWST2 to calculate demand.

xi. Add the calculated demand to the CHW load to get the final HW load.
xii. This final HW load may be used, or this iteration may be repeated as many times as needed by calculating HWST3, HWST4, etc. if the agreement between HW load results of each iteration is not yet sufficient.

c. When QuantityConstrained, use the following steps:
  i. Begin the iteration with the assumption that the chilled and HW loads on the HRC are equal to HRCoolingOutput and HRHeatingOutput.
  ii. Calculate HWST1 using the HW load, and adjust the flow and return temperature values used in the calculation based on the following:
    1. Return Temperature (After Blending with the TES Tank):
      a. When the TES tank is charging, the HW return temperature is determined by an energy balance using the building HWRT, building flow, TESFlow, and TankTempBelowTC.
      b. When the TES tank is discharging or neutral, the HW return temperature in this calculation is equal to the building HWRT.
    2. Flow:
      a. If FlowHRCh > PrimaryHWFlow, use the primary HW flow in the calculation.
      b. If FlowHRCh <= PrimaryHWFlow, use FlowHRCh in the calculation.
  iii. Calculate CHWST1 using CHW load, and the adjust flow and return temperatures used in the calculation based on the following:
    1. If FlowHRCc > PrimaryCHWFlow, use the primary CHW flow, and primary CHW return temperature in the calculation.
    2. If FlowHRCc <= PrimaryCHWFlow, use FlowHRCc and primary CHW return temperature in the calculation.
  iv. Calculate cooling capacity using the regression curve as a function of the previously calculated CHWST1 and HWST1. This capacity becomes the new CHW load.
  v. Use cooling PLR=1, CHWST1, and HWST1 to calculate demand.
  vi. Add the calculated demand to the new CHW load to get the HW load.
  vii. Calculate HWST2 using the HW load, and the same flow / return temperature considerations outlined previously for HWST1 calculations.
  viii. Calculate CHWST2 as a function of the most recently calculated module CHW load, and the same flow considerations outlined previously for CHWST1 calculations.
  ix. Calculate cooling capacity using the regression curve as a function of CHWST2 and HWST2.
    1. Set the CHW load equal to this capacity if this is the final time cooling capacity will be calculated in these iterative steps.
  x. Use cooling PLR=1, CHWST2 and HWST2 to calculate demand.
  xi. Add the calculated demand to the CHW load to get the final HW load.
  xii. This final HW load may be used, or this iteration may be repeated as many times as needed by calculating CHWST3 and HWST3, etc. if the agreement between HW load results of each iteration is not yet sufficient.

13. Once the HW and CHW loads on the HRC have been determined, assign the remainder of the CHWPlantLoad by splitting it evenly among the appropriate equipment in stage. The following exceptions apply:
    a. Chiller staging capacity shall be determined as a function of the worst case capacity multiplied by the user adjustable staging percentage (initially 80%) to determine staging capacity. Divide the remaining CHW load by the staging capacity and round up to determine the number of operating chillers.
    b. If CHWPlantLoad is large enough to operate in stage 4 cooling based on the staging table, the HRC shall run in cooling only mode, and the HW and CHW loads for heat recovery established in the previous section shall be overridden.
        i. CH-1 and CH-2 loads shall be set to their current hour capacities multiplied by the staging capacity %, where current hour capacity is based on the regression output using CWRT and CHWST set point.
        ii. The following steps should be taken to find the new HRC loads when in cooling only mode:
            1. Since the loads on CH-1 and CH-2 are now established, use the CHWST setpoint, chiller load, and primary loop flow to calculate the water temperature leaving the HRC and entering the cooling only chillers.
            2. If CHWPrimaryFlow > FlowHRCc, use the CHWRT from E+ outputs, temperature output from the preceding step, CHWPrimaryFlow, and FlowHRCc in an energy balance which calculates the actual heat recovery chiller leaving water temperature, HRCLWT.
                a. Otherwise if CHWPrimaryFlow <= FlowHRCc, then HRCLWT is equal to the output of step 1 in this process.
            3. Calculate HRC cooling capacity as a function of HRCLWT for the evaporator leaving temp, and (CWRT+HX approach) for the condenser leaving temp.
            4. After CH-1 and CH-2 are fully loaded, assign the remaining cooling load to the HRC and calculate the cooling PLR using the capacity established in the preceding step.
            5. Calculate demand as a function of cooling PLR, HRCLWT, and (CWRT + HX approach)
            6. Calculate HRC heating output as HRC CHW load + demand
        iii. Document any remaining cooling load which is not met by the chillers or HRC in a column for unmet cooling load.
14. AWHP module staging capacity shall be determined as a function of the worst case capacity multiplied by the user adjustable staging percentage (initially 80%) to determine staging capacity. Divide the remaining HW load by the staging capacity and round up to determine the number of operating heat pump modules. Then, use the following steps to determine AWHP capacity (which will be used later for power calculations).
    a. AWHP loop flow is equal to PrimaryHWFlow subject to the following constraints:
        i. AWHPFlow >= sum of minimum flows per module for all modules in stage
        ii. AWHPFlow <= sum of maximum flows per module for all modules in stage
    b. Determine the HWST intermediate, which is located in the supply line between the AWHP and TES tank connections.

i. When the tank is charging, HWSTinter is equal to the design HWST

ii. When the tank is full or discharging, use the following energy balance to determine HWSTinter:

$$HWST_{inter} = \frac{HWST_{sp} * (Building\ HW\ FLow) - TESFlow * (TankTempAboveTC)}{PrimaryHWFlow}$$

c. If AWHPFlow is >= to HWPrimaryFlow:

i. The HWST set point for the AWHPs, HWSTawhp, is equal to the HWSTinter

d. If AWHPFlow is < HWPrimaryFlow the calculation is more complicated, and involves several energy balances to determine the temperature in the AWHP decoupler, which is required to determine the HWSTawhp:

i. First calculate the return temperature at the inlet of the primary loop pumps after the connection with the TES tank:

1. When the tank is discharging or full, the HWRTprimary is equal to the building HWRT.
2. When the tank is charging, the temperature is determined by the following energy balance.

$$HWRT_{primary} = \frac{HWRT_{B} * (Building\ HW\ FLow) + TESFlow * (TankTempBelowTC)}{PrimaryHWFlow}$$

ii. If the HRC is in cooling only mode:

1. The AWHP decoupler temperature is equal to the building HW return temperature after blending with the TES tank, HWRTprimary, as described in the preceding steps.

iii. If the HRC is in heat recovery mode:

1. The AWHP decoupler temperature is equal to HWRTprimary + HRC HW load/(490*PrimaryHWFlow). Note that the HRC HW load used here is dependent on the HRC constraint type.

iv. Once the temperature in the AWHP decoupler has been determined, use another energy balance to determine the required HWSTawhp.

$$HWST_{AWHP} = \frac{HWST_{inter} * (PrimaryHWFlow) - DecoupTemp_{AWHP} * (PrimaryHWFlow - AWHPFlow)}{AWHPFlow}$$

e. Once the HWST of the AWHPs is determined, use the capacity regression, HWSTawhp, and the outside air temperature to determine the capacity of the modules in stage.

15. Divide the remaining HWPlantLoad which is not served by the HRC evenly between the AWHPs in stage, provided the assigned load does not exceed the capacity of any module.
    a. Document any remaining heating load which is not met by the AWHPs or the HRC in heat recovery mode in a column for unmet heating load.
16. CW to HW Heat Exchanger:
    a. CW return temperature (CWRT) is reset based on CHWPlantLoad, from CHWST + 20°F when CHWPlantLoad = 120 tons or below, to CHWST + 50°F when CHWPlantLoad = 700 tons or above.
    b. Calculate the CWST based on the current hour chiller CWRT and the chiller CW design flow.
    c. Calculate the CW flow through the HX based on the heat exchanger load, calculated CWST, and CWRT.

    i. If the HRC is in cooling only mode, the heat exchanger load, HXLoad, is equal to the heating load on the HRC.
    ii. If the HRC is in heat recovery mode, the heat exchanger load is equal to 0 and the CW return temp leaving the heat exchanger.

17. Cooling Towers
    a. Calculate total cooling tower flow as the sum of CW flow through the HX and the CW flow of any operating chillers.
    b. Using cooling tower flow, CWRT, CWST, and ambient wet bulb, calculate cooling tower power demand.
18. Calculate Chiller Power
    a. Calculate current hour chiller capacity using the building CHWST and CWRT.
    b. Calculate cooling PLR based on the load assigned to the cooling only chillers and the capacity calculated in the preceding step.
    c. Using CHWST set point, CWRT, and cooling PLR, calculate cooling only chiller power demand for each enabled chiller
19. Calculate AWHP power
    a. Calculate heating PLR based on the load assigned to each AWHP and the capacity established in step 14
    b. Using OAT, HWSTawhp, and heating PLR, calculate the heat pump power demand for each enabled heat pump.
20. HRC Power:
    a. HRC power demand for all constraint scenarios and cooling only mode have been previously calculated throughout steps 12 and 13. Reference the appropriate calculation depending on plant mode and HRC constraint.
21. Calculate Power for Remaining Pumps
    a. CHW Pumps (PrimaryCHWFlow)
        i. Head loss through chiller evaporator shall be calculated as a function of chiller module flow assuming pressure drop varies as $Flow^{1.8}$.
        ii. Head loss through the piping circuit upstream of the DP sensor shall be calculated as a function of loop flow assuming pressure drop varies as $Flow^{1.8}$. Design pressure drop upstream of the DP sensor is the difference between design loop DP and design DP sensor setpoint.
        iii. Head loss downstream of the sensor shall be assumed equal to the DP setpoint.
        iv. Assume DP setpoint resets downward as a function of loop load flow along a user definable linear reset using inputs of design DP setpoint, minimum DP setpoint, max reset flowrate, and min reset flowrate.
        v. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.
    b. Primary HW Pumps (PrimaryHWFlow)
        i. Head loss through the primary loop shall be calculated as a function of loop flow assuming pressure drop varies as $Flow^{1.8}$.
        ii. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.
    c. Building Loop HW Pumps (Instantaneous HW flow from E+)

i. Head loss through the secondary piping circuit upstream of the DP sensor shall be calculated as a function of loop flow assuming pressure drop varies as $Flow^{1.8}$. Design pressure drop upstream of the DP sensor is the difference between design loop DP and design DP sensor setpoint.
ii. Head loss downstream of the sensor shall be assumed equal to the DP setpoint.
iii. Assume DP setpoint resets downward as a function of loop load flow along a user definable linear reset using inputs of design DP setpoint, minimum DP setpoint, max reset flowrate, and min reset flowrate.
iv. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.

d. AWHP Side Stream Pumps
   i. Only perform this calculation if one or more AWHPs are enabled.
   ii. Flow is equal to PrimaryHWFlow subject to the following constraints:
      1. AWHP loop flow >= sum of minimum flows per module for all modules in stage
      2. AWHP loop flow <= sum of maximum flows per module for all modules in stage
   iii. Head loss through AWHP condensers shall be calculated as a function of condenser flow relative to the design flow of each heat pump, assuming pressure drop varies as $Flow^{1.8}$.
   iv. Head loss through the side stream loop shall be calculated as a function of AWHP loop flow assuming pressure drop varies as $Flow^{1.8}$.
   v. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.
e. HRC HW Side Stream Pump
   i. HW and CHW flows for these pumps are calculated in step 10.
   ii. Head loss through HRC condenser shall be calculated as a function of condenser flow assuming pressure drop varies as $Flow^{1.8}$.
   iii. Head loss through the HW side stream loop shall be calculated as a function of loop flow assuming pressure drop varies as $Flow^{1.8}$.
   iv. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.
f. HRC CHW Side Stream Pump
   i. HW and CHW flows for these pumps are calculated in step 10.
   ii. Head loss through HRC evaporator shall be calculated as a function of evaporator flow assuming pressure drop varies as $Flow^{1.8}$.
   iii. Head loss through the CHW side stream loop shall be calculated as a function of loop flow assuming pressure drop varies as $Flow^{1.8}$.
   iv. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.
g. CW Pumps (use cooling tower flow)
   i. Use the larger of the following for equipment based pressure drop in the CW loop.
      1. Head loss through chiller condenser shall be calculated as a function of CW flow assuming pressure drop varies as $Flow^{1.8}$. CW flow per cooling only chiller is constant at design CW flow.

2. Head loss through the heat exchanger assuming pressure drop varies as $Flow^{1.8}$. Use CW heat exchanger flow.

ii. Head loss through the CW loop shall be calculated as a function of loop flow assuming pressure drop varies as $Flow^{1.8}$.

iii. Tower static head must be included in the total pressure drop calculations.

iv. Calculate pumping power using head, flow, pump efficiency, motor efficiency, and VFD efficiency.

Equipment Performance Curves

Establishing Heating Regression curves using Only the Cooling regression curve, no pre-existing data:

1. Use cooling capacity and power curves to populate a large data set at a variety of ELT, CLT, and cooling PLRs.
2. For each unique combination of CLT, ELT, and PLR, find heating capacity = cooling capacity + power
3. At each point in the data set where cooling PLR is less than 1, calculate heating PLR as the heating capacity at that cooling PLR / heating capacity at the same set of inputs when PLR is 1. This establishes heating PLR for the full data set.
4. Use CLT, ELT, and heating capacities to develop heating regression curve.
5. Use CLT, ELT, and heating PLR to develop power regression curve. Power outputs are the same for both heating and cooling based power curves.

Appendix C

Additional Analysis to Assess the Performance of the TIER Plant

Figure 63 and Figure 64 show the combined COP by peak daily heating and cooling loads. Combined COP was estimated as the sum of total daily heating and cooling loads divided by the sum of total daily equipment power (i.e., heating equipment power and cooling equipment power). For both plants, the combined COP decreases at higher heating load bins and lower cooling load bins (see Figure 63 and Figure 64). The combined COP is generally higher for the TIER plant compared to the baseline plant. The gap between the TIER and baseline plants is especially larger at high heating load, which confirms the better performance of the TIER plant on heating mode.

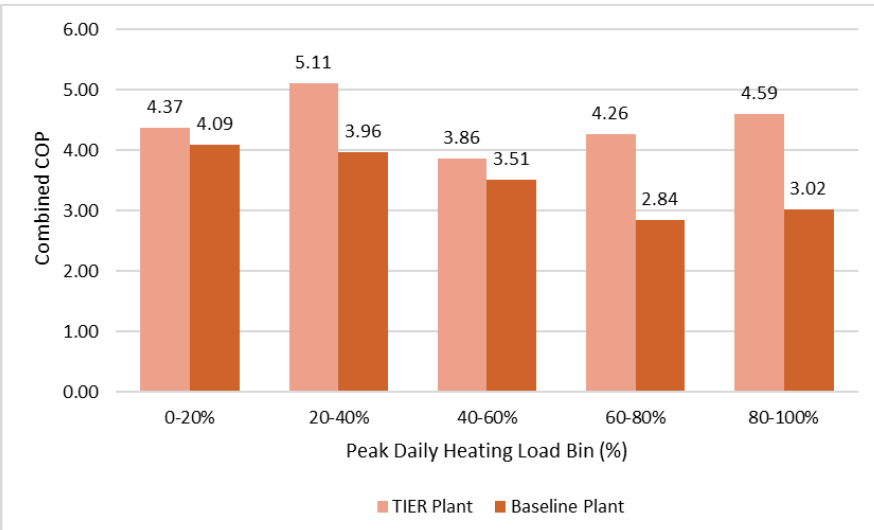

Figure 63: Combined COP by peak daily heating load range for the TIER plant and baseline plant.

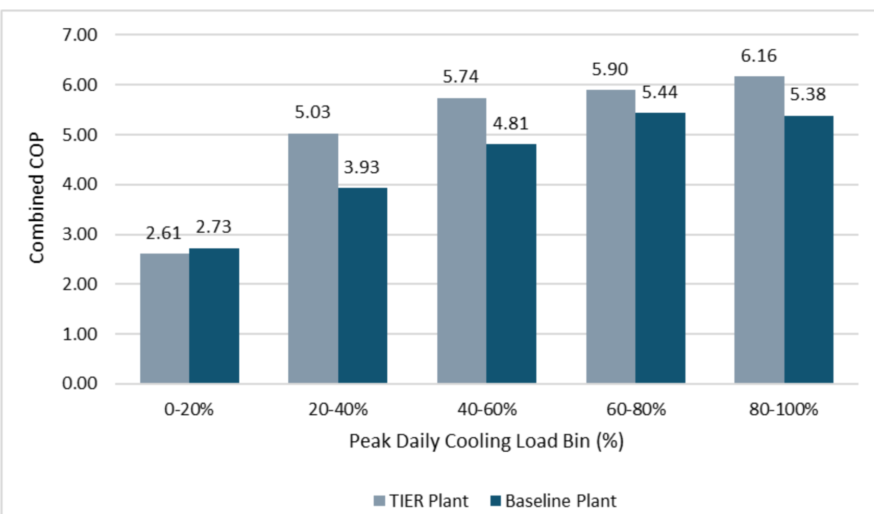

Figure 64: Combined COP by peak daily cooling load bin for the TIER plant and baseline plant.

Figure 65 and Figure 66 present the daily average number of cooling chiller starts by HDD and CDD for both plants from April to August. The average daily start of the cooling chiller for the baseline plant is higher than the TIER plant mainly because the chiller is oversized in the baseline plant. Thus, the chiller cycles more frequently.

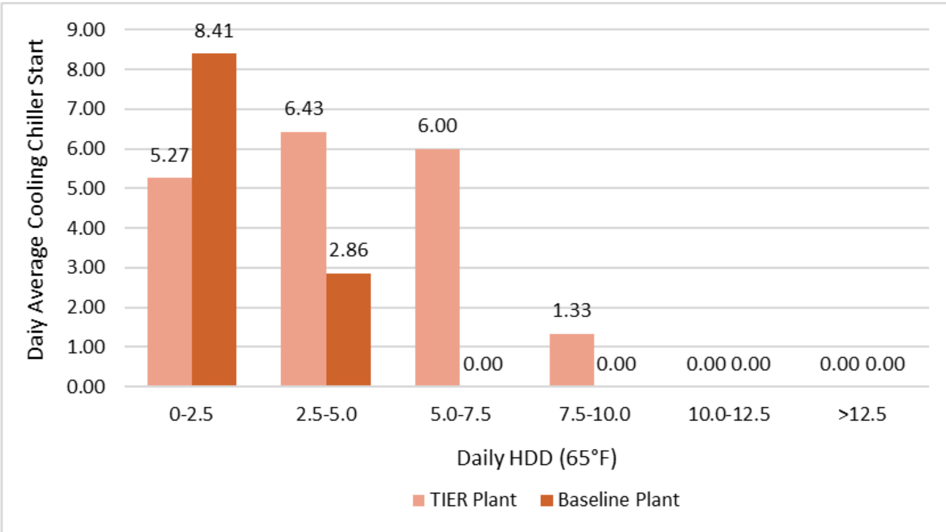

Figure 65: Daily average cooling chiller starts by daily HDD for the TIER plant and the baseline plant from April to August.

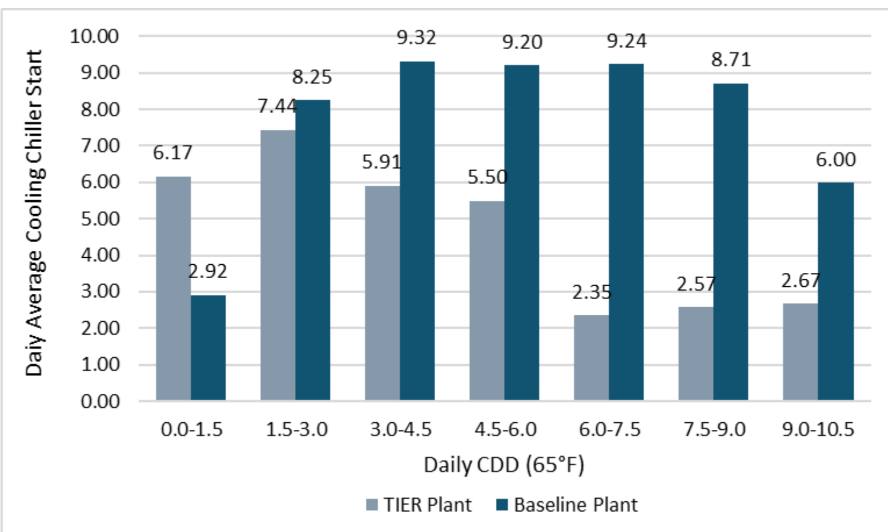

Figure 66: Daily average cooling chiller starts by daily CDD for the TIER plant and the baseline plant from April to August.

According to Figure 67, the daily average cooling chiller start reduces as the cooling load increases (60 to 100 percent peak cooling load bins). This shows that while cooling chillers run more at higher loads, there is less chiller cycling.

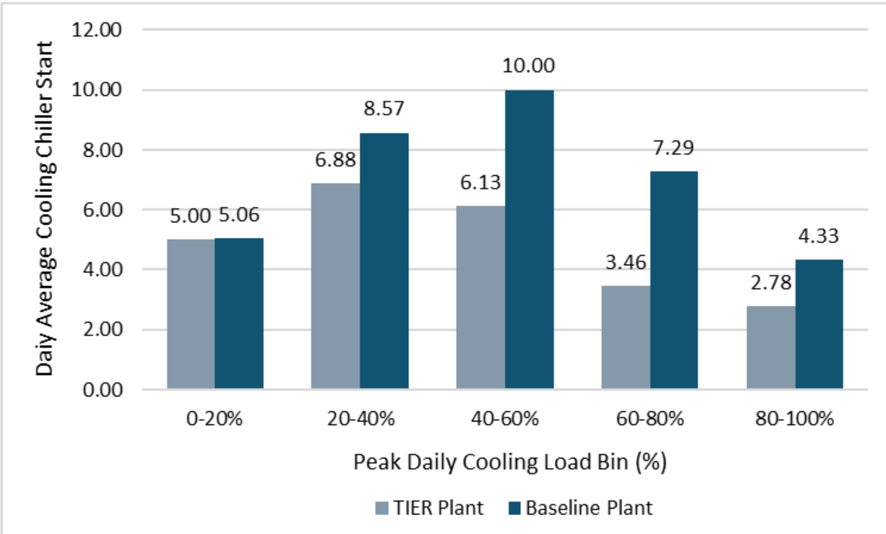

Figure 67: Daily average cooling chiller starts by peak daily cooling load range for the TIER plant and the baseline plant from April to August.